MITNE-191

AN ENGINEERING AND ECONOMIC EVALUATION OF SOME MIXED-MODE WASTE HEAT REJECTION SYSTEMS

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October, 1976

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Submitted to the Department of Nuclear Engineering on September 22, 1976, in partial fulfillment of the requirements for the degree of Doctor of Science.

ABSTRACT

A survey of potential mixed-mode waste heat rejection systems for large central power stations has been undertaken in an effort to develop new waste heat rejection system design options. A mixed-mode waste heat rejection system is defined as a waste heat rejection system in which more than one type of heat rejection device or more than one method of system operation is utilized. All currently available waste heat technology has been reviewed for its applicability to the mixed-mode concept with the exception of "once-through" cooling systems. The literature concerning the mathematical modeling of the thermal performance of waste heat dissipation systems is reviewed and recommendations are made for modeling the thermal performance of natural and mechanical draft evaporative cooling towers, spray canals. cooling ponds, and natural and mechanical draft dry cooling towers.

An initial survey of some mixed-mode options indicates that waste heat system utilization considerations would, at most sites, not provide sufficient economic justification for the design of systems composed of devices with different ratios of operational to capital cost. Also, a study of combined evaporative cooling tower-cooling pond systems yields some simple design recommendations. Investigations of a variety of cyclically-operated storage pond-cooling tower systems reveals that this concept is an attractive solution to the problem of the coincidental occurrence of the maximum daily power loss due to high ambient temperatures and maximum daily utility-system electrical demand. Several applications of this concept with regard to evaporative cooling towers are examined and are shown to be worthy of consideration as design options in more detailed analyses of specific plant sites. The most important result of this study of cyclically-operated waste heat rejection systems has been the identification of significant economic benefits of the combined thermal storage pond and dry cooling tower system.

The engineering feasibility of constructing and operating a thermal storage pond which would exhibit the type of thermal-hydraulic behavior necessary for the efficient operation of the combined thermal storage pond and dry cooling tower system has been established through the use of an experimental physical model of the proposed water storage pond. Based on the results of the modeling studies, density-induced flows which tend to short-circuit the pond are seen to be the major design constraint, and recommendations are made for the design of an efficient and economical storage pond.

The economics of the thermal storage pond-dry cooling tower system have been evaluated through the use of a design-optimization, system-simulation model. Various plant sites, steam turbine types, and system operational schemes are considered. A potential 15 to 20% savings in the cost of dry cooling is seen for nuclear plants in the western United States using the proposed high-exhaust-pressure, modified-conventional steam turbine. Also, the thermal storage pond-dry cooling tower system utilized with the conventional steam turbine is seen to be economically superior at many sites to simple dry cooling utilized with any of the proposed advanced high-exhaust-pressure turbines.

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ACKNOWLEDGMENTS

The author has enjoyed the help, encouragement, and direct assistance of many people during the course of this work.

The author appreciates having had the opportunity to work under his thesis supervisor, Michael Golay. His efforts to obtain financial support for this work were crucial to its timely completion.

The technical assistance of the staff of the Ralph M. Parsons Laboratory for Water Resources and Hydrodynamics in connection with the hydraulic modeling reported herein is also greatly valued.

Special thanks goes to Drs. Leon Glicksman and Gerhard Jirka for their comments and encouragement.

The Electric Power Research Institute and Battelle Northwest Laboratory are acknowledged for their interest in and support of this study. Also I am fully appreciative of the personal financial support provided by the United States Atomic Energy Commission and the National Science Foundation through their engineering traineeship programs.

The author's wife, Deborah, not only contributed invaluable and cheerful moral support, but was active in helping perform some of the more arduous tasks of data collection and presentation. The author is also indebted to his entire family for their wholehearted support and encouragement.

TABLE OF CONTENTS

List of Figures

List of Tables

CHAPTER 1

MIXED-MODE WASTE HEAT DISSIPATION AT CENTRAL POWER STATIONS

1.1 The Problem

Within the scope of present technology there are a number of waste heat dissipation methods available to the designer of a central power station. In all but a few instances, it has been engineering practice to employ only a single method at any particular station. Also, to date little consideration has been given to dynamically operated systems which attempt to maximize operational economies in the presence of changing meterology and electrical generation requirements. This practice of using single-mode waste heat dissipation systems is the result of current design procedures and not the direct result of a comprehensive engineering and economic evaluation of the entire range of options which can be fabricated given the current waste heat technology.

The currently available methods for the dissipation of waste heat include once-through cooling, cooling ponds, spray canals, mechanical and natural draft evaporative cooling towers and - to a limited extent - mechanical and natural draft dry cooling towers. The primary goal of this research has been the identification and subsequent engineering and economic evaluation of mixed-mode waste heat rejection systems which can be constructed using the above devices as the basic system components. The general term "mixed-mode" waste heat rejection system has, in this work, as its meaning any waste heat rejection system which achieves the desired cooling through the use of more than one device and/or method of operation. A single-mode system is thus any one of the above listed devices operated in a continuous and non-varying manner. It should be clear that the intention of this work is not the specification of new heat disposal equipment, but rather the intention is to reveal how the presently available heat rejection technology can be utilized to construct mixed-mode systems which are economically superior to single-mode systems.

The waste heat system evaluation problem being addressed in this work is different from that which confronts the power plant design engineer. Typically, the plant engineer has a fairly well defined waste heat rejection problem in that the site economics and physical characteristics are fixed. He need only identify the system which is the most economical with respect to the governing environmental constraints given the site characteristics.

In the present task, it is of interest not to be constrained to a specific set of site characteristics since the goal of this work is to determine what benefits might be gained by employing mixed-mode waste heat dissipation systems <u>and</u> under what circumstances these benefits are likely to be obtained. Thus, rather than being constrained to a specific set of site characteristics the present work is constrained only to the range of site characteristics which the United States electric utility industry is expected to encounter in the future.

1.2 Limitations on the Investigation

In order for this investigation to be compatible with the available financial and human resources two important limitations were placed on the scope of the proposed work. The first limitation is that no consideration is given to mixed-mode systems involving the use of once-through cooling. Second, no effort has been made to explore the area of combined evaporative/non-evaporative (i.e. wet/dry) systems.

Once-through systems have not been considered due to the extremely site-specific constraints on the use of this type of cooling. For economic reasons, the plant designer would almost always attempt to maximize the use of once-through cooling. More often than not, the limits on the use of once-through cooling are environmentally and not economically or physically based and any attempt to improve combined systems involving a once-through component would necessitate direct consideration of environmental matters [F3]. Reconciliation of environmental constraints is difficult or impossible within the context of quantitative waste heat system design and performance analysis.

For similar reasons the use of wet/dry system is not considered. Wet/dry systems would only be employed in those instances when total wet cooling is not acceptable due to water consumption or fogging.[L1] Water consumption and fogging are constraints which are environmental in nature and are highly site dependent. Even a generalization of the economic-environmental relationships between system performance and these constraints is at best tenuous. In any event, other investigators have given wet/dry systems considerable attention.[D4][H6][L2]

1.3 Approach to the Problem

Although minimum cost is the fundamental criterion by which all waste heat rejection systems are judged, this criterion in itself does not provide an adequate guide for the conceptualization of mixed-mode systems. What is needed is a set of avenues of approach to the conceiving of mixed-mode waste heat systems. The required avenues of approach can be formulated by considering the means by which mixed-mode systems could result in economic improvements. Then one may consider how the available waste heat technology might be adapted and utilized to realize the potential benefits.

Surveying the field of waste heat dissipation, mixed-mode systems appear to offer potential with regard to

1) making greater advantage of the time varying meteorological and cost characteristics of the site,

2) utilizing to the maximum extent cooling resources available on site, and

3) utilization of coupled systems such that the total performance of the combined system is superior to that of the individual components.

Certainly these three routes to the conceptualization of mixed-mode waste heat systems are not entirely independent of each other or completely comprehensive. It is not reasonable to assume that all the potentially beneficial mixed-mode systems will be recognized by this type of problem approach. Indeed, this type of approach has as its basis an inventor-type of recognition of problem solution. Consideration of more structured approaches which would attempt to develop a more analytical framework for mixed-mode system recognition has, nevertheless, lead to an understanding of their decided impracticality.

1.4 Outline of Presentation

The basic mathematical models of the thermal performance of the various component waste heat rejection systems are discussed in Chapter 2. Then using the approach mentioned in the previous section a survey of several possible mixed-mode options is presented in Chapter 3. The results of this survey reveal the large benefits which may be obtained through the use of an adaptation of the conventional cooling pond - the thermal storage pond - in conjunction with dry cooling towers. In Chapter 4 is discussed the experimental design study of the proposed thermal storage pond and the following chapter includes a detail discussion of thermal storage pond/dry cooling tower system economics and engineering.

CHAPTER 2

MATHEMATICAL MODELS FOR PREDICTING THE THERMAL PERFORMANCE OF CLOSED-CYCLE WASTE HEAT DISSIPATION SYSTEMS

2.1 Introduction

The literature concerning the dissipation of waste heat from central power stations has grown rapidly in the last decade. All areas within the general category - from biological effects to heat transfer developments - have been the subject of an increasing number of technical reports, journal articles, and trade magazine articles.

The two fundamental reasons for the rapid growth of this literature are the imposition of environmentally-motivated governmental regulations on the traditional "once-through" cooling system and the increasing unavailability of adequate sources of "once-through" cooling water at otherwise attractive central power station sites.

However, there is as yet no definitive source of information from which one can independently construct reliable thermal behavior and economic models of waste heat dissipation systems. The few studies which have addressed the general problem of developing the independent capability of evaluating the thermal performance of alternative waste heat dissipation systems are either out-of-date [D5] or lacking in the details [H4][S1] and thus can not be directly applied to the present task. Thus, considerable effort was required to review the available information and compile it into a useful tool for evaluating the costs/benefits of various alternative waste heat dissipation schemes.

The available literature concerning the mathematical modeling of the economics and thermal behavior of waste heat systems has been authored primarily by 1) the vendors of waste heat dissipation equipment, 2) the electric utility industry, and 3) various research institutes and universities. In view of the present task of developing accurate mathematical models of conventional waste heat rejection devices some general comments can be made about the literature with regard to its authorship.

Although there has been a tremendous increase in the waste heat dissipation equipment vendor sector in both size and diversity, the publications of these vendors are generally qualitative in nature. With a few notable exceptions, the literature published does not deal quantitatively with thermal behavior analysis, but, rather, describes qualitatively the particular vendors present capabilities and highlights the economic advantages of the particular vendors devices. Little of this information is of value to those interested in developing an independent analysis capability.

The dearth of substantial information published by equipment vendors is, of course, understandable since their proprietary interests are not well served by the free-flow of their costly research and development results.

The literature on this topic authored by the electric utility industry has come from the electric utilities themselves as well as their consultants - mainly the large architectural engineering firms. As is the case above, little substantive information has been published with regard to the mathematical modeling of the thermal behavior of various heat rejection systems by this sector. However, valuable government-sponsored information has been reported by architectural engineering forms. Many trade journal articles which review the waste heat dissipation solutions applied to specific sites have been authored by utility system engineers, but these findings are usually of little value to the present task.

Much useful information concerning the mathematical modeling of the thermal performance of heat rejection systems has been authored by various research institutes and universities under the sponsorship of federal and state agencies and electric utilities. In applying some of this information, however, difficulty is encountered in attempting to relate the published results to the actual thermal performance of modern, well-designed waste heat dissipation systems.

2.2 Mechanical Draft Evaporative Cooling Towers

2.2.1 Literature Review

Croley et al. [C8] have recently addressed the problem of developing an accurate thermal and economic model of conventional cross-flow mechanical draft evaporative cooling towers. Their review of the literature led then to the use of a thermal analysis model based on a simple straightforward finite-difference solution of the well-known Merkel [M3] evaporative heat transfer differential equation.

The Merkel formulation of evaporative heat transfer combines the mass transfer (evaporation) and the sensible heat transfer coefficient into a single coefficient. The approximate net energy transfer is then a product of the coefficient and the enthalpy potential difference between the water and the air streams. The standard "Merkel" equation is as follows:

$$\frac{KaV}{L} = \int_{T_2}^{T_1} \frac{dT}{(h''-h)} \qquad (2.1)$$

where K = overall transfer coefficient, lb/(hr)(ft^2 of interface)(lb of water/lb of dry air)

a = interfacial contact area (ft^2/ft^3 of tower fill)

V = planar volume (ft^3/ft^2 of plan area,)

L = water flow rate ($lb/hr\text{-}ft^2$ of plan area),

T_1 = inlet water temperature,

T_2 = exit water temperature,

dt = water temperature differential,

h'' = enthalpy of saturated air at the water temperature, and

h = enthalpy of the main air stream (BTU/lb of dry air) .

Derivation of this relationship may be found in several references [K2][M5]. Physically the quantity KaV/L in the above equation represents an effective heat transfer ability or "number of transfer units" for a particular cooling tower. This coefficient is dependent on the relative amounts of water and air flow in the tower and must be determined experimentally.

Croley et al. [C8] have applied this differential equation in finite-difference form to solve the two-dimensional heat exchange problem of the widely-utilized induced draft crossflow evaporative cooling tower for known inlet air and water boundary conditions. The finite-difference approximation to the Merkel equation consists basically of the division of the energy transfer volume into a number of equal sized blocks over which the energy transfer potential (enthalpy) is averaged.

The conclusions of Croley et al. concerning the utility of the basic Merkel formulation for the predicting of the energy transfer in a cooling tower has since been substantiated by the recommendation of Hallet [H1]. Hallet, representing a leading cooling tower vendor, has suggested that the best approach (for a non-vendor) to the problem of evaluating the thermal performance of wet cross-flow towers is a finite-difference solution of the basic Merkel equation. This author also points out that, although many improvements in the theory of simultaneous heat and mass transfer at water/air interfaces have been suggested, the basic Merkel formulation is the only widely accepted and proven theory.

The analysis technique suggested by Hallet is essentially identical to that of Croley et al. except that Hallet recommends the inclusion of a temperature dependence in the expression for the tower fill energy transfer coefficient:

$$Ka = f(T_1) \tag{2.2}$$

where T_1 is the tower inlet water temperature. It is interesting to note that no physical justification is given by Hallet for this "temperature effect". Consideration of recent works which address the errors inherent to the Merkel equation suggest that this "temperature effect" fixup is necessary because of errors in the Merkel approximate formulation for evaporative heat transfer.

The investigations of Nahavandi [N1] and Yadigaroglu [Y1] have been concerned with an evaluation of the errors inherent to Merkel equation. The results of Yadigaroglu are based on a comparison of the predictions of the Merkel theory and a more exact and complete theory which treats the mass and sensible heat processes separately. This investigator found that the effect of the various approximations of the Merkel theory tends to be small since the different approximations of the Merkel theory result in partially cancelling positive and negative errors. The conclusion is that, given the other errors associated with cooling tower performance predictions (uniform air and water flow rates, for example) and performance verifications (experimental uncertainties), the added complexity of performing the more exact energy transfer analysis is not justified. Nevertheless, it is of interest to note that Yadigaroglu found that the net positive error in predicting the cooling range increased with increasing air inlet temperature and humidity. This error could be corrected by arbitrarily decreasing the value of KaV/L by the appropriate amount as the water inlet temperature increased. Indeed, this is the same, but unjustified, approach recommended by Hallet. Examining the magnitude of the over-prediction resulting from the use of Merkel theory (on the order of 5%), it is found that the Merkel theory error is consistent with the suggested "temperature effect" correction of KaV/L (about 5% per 10 °F rise in inlet water temperature for inlet water temperatures in excess of 90 °F).

2.2.2 Selection of a Model

The mathematical model to be used in the prediction of the thermal performance of mechanical draft evaporative cooling towers is the finite-difference approximation of the Merkel equation. The finite-difference approximation to the Merkel equation can be stated as [C8]

$$G(h_o-h_i) = \frac{KaV}{N}\left[\frac{h_i'-h_i + h_o'-h_o}{2}\right] \tag{2.3}$$

where h_i' and h_o' = saturated air enthalpies at the inlet and outlet of an incremental element,

h_i and h_o = saturated air enthalpies at the temperature of the water entering and leaving the incremental element,

G = air flow rate per incremental element,

N = square root of the number of incremental elements, and

KaV = transfer coefficient.

The application of this equation to the cross-flow problem of a conventional induced draft cross-flow cooling tower is shown in Fig. 2.1.

In addition to the above equation, the energy balance equation

$$G(h_o - h_i) = c_p L(t_i - t_o) = L(h_i' - h_o') \qquad (2.4)$$

is needed to completely describe the temperature history of the air and water as it passes through the tower fill. In the above equation:

L = water loading per incremental element,

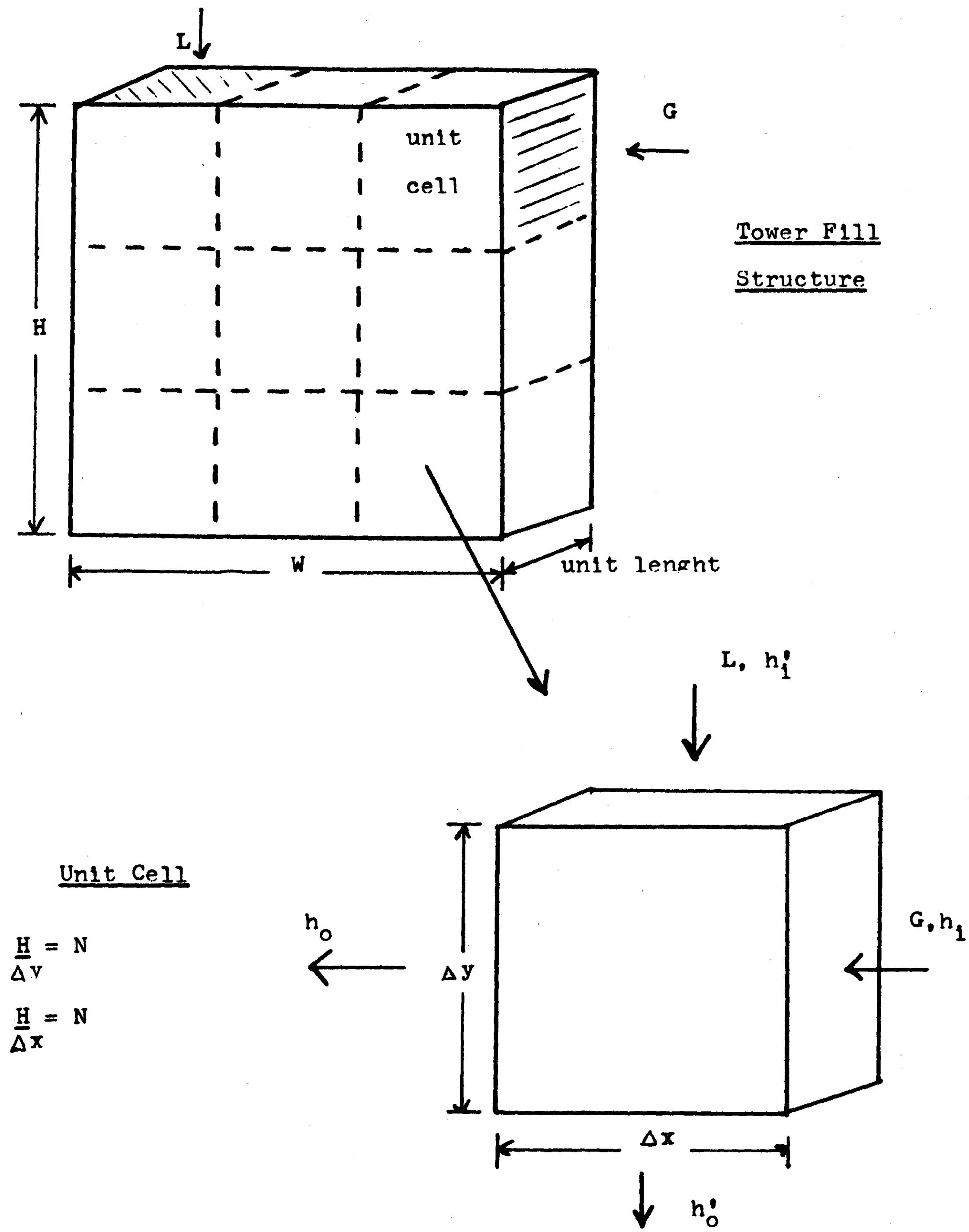
t_i and t_o = inlet and outlet water temperatures for an incremental element, and

c_p = specific heat capacity of water.

Equations 2.3 and 2.4 form a set of coupled equations with unknown variables h_o and h_o' which must be solved for iteratively. The algorithm for calculating the average outlet water temperature and average outlet air temperature is given in Fig. 2.2. Note that, for practical purposes, the water and air flow rates are fixed by the tower design and to a good approximation can be assumed to be uniform and constant throughout the tower. Note, also, that the algorithm is for calculating the performance of a given tower design. If we wish to find the size of the tower needed to meet a

Fig. 2.1

Illustration of Tower Fill Finite-Difference Calculation



specific cooling requirement, a trial and error calculation may be performed.

The saturated air enthalpy used as the driving potential in the Merkel equation depends on both the dry bulb temperature and the humidity of the air. However, a good approximation to the enthalpy which depends solely on the thermodynamic wet bulb temperature may be derived. From Marks [M2] we have the relationship,

$$E = 0.24Td + W(1062.0 + 0.44Td) \tag{2.5}$$

and

$$W = \frac{W^* - (0.24 + 0.44W^*)(Td - Twb)}{(1094 + 0.44Td - Twb)} \tag{2.6}$$

where E = enthalpy of moist air,

Td = dry bulb temperature,

W = specific humidity,

Twb = wet bulb temperature, and

W* = specific humidity for saturation at Twb.

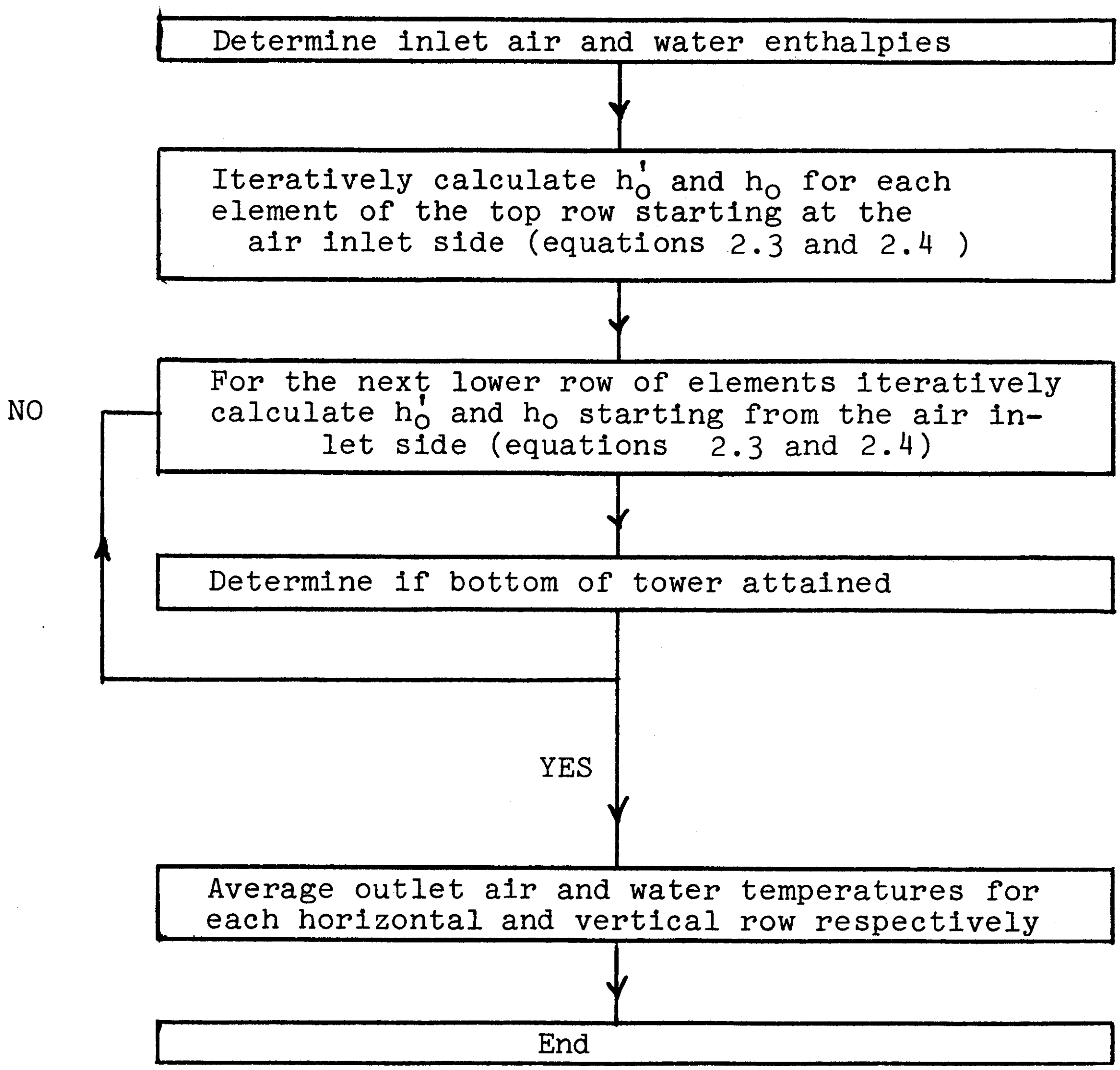
Substituting the latter into the former we have

$$E = 0.24Td + W^*(1062 + .44Td) -$$

$$- \frac{(0.24 + 0.44W^*)(Td-Twb)(1062 + 0.44Td)}{(1094 + 0.44Td-Twb)} \tag{2.7}$$

Fig. 2.2

Calculational Algorithm for Predicting the Performance of Mechanical Draft Cross-flow Evaporative Cooling Tower (MECDRAFT Program)



Now assuming that in the deminator we can make the approximation

$$32 - Twb \approx 0 \qquad (2.8)$$

and expressing the saturation himidity in terms of saturation pressure we have

$$E \approx 0.24Twb + \frac{0.622P_{sa}}{P_{atm}-P_{sa}}(1062.0 + 0.44Twb) \qquad (2.9)$$

where P_{atm} = total atmospheric pressure, and

P_{sa} = saturation pressure of water vapor at Twb.

The above assumption is a good one in this particular circumstance since the error affects the ratio of large numbers. An error of 50 °F in magnitude in the denominator would be typical with the total resultant error being about 5%. However, in all applications of the approximate enthalpy equation the equation is ultimately used to find the difference of two enthalpies and thus the resultant error in the difference is minimal.

2.2.3 Application of Model

To achieve the goal of obtaining an accurate thermal performance model of a conventional cross-flow induced draft evaporative cooling tower module the physical dimensions and empirical heat transfer and air friction data for a typical module must be acquired. Croley et al. [C8] have modeled the thermal behavior of such modules and reported the results. From the published information the physical dimensions of the tower fill are readily obtainable. They are

height = 60 feet

width = 36 feet, and

length = 32.

However, the air friction factors for this fill is not directly obtainable from the published results. Nevertheless, an energy balance on the modeled tower based on the published information indicates an average air flow rate of 2.4×10^3 lb_m/ hr-ft^2. It will suffice for the purposes of this study to assume the air flow is constant and equal to this value. Croley et al. do not report the values of the energy transfer coefficient used in their study since empirical proprietary information was used in evaluating the energy transfer coefficient. However, sufficient calculational results using this proprietary information are reported to allow a regression of the required information.

The Cooling Tower Institute [C6] states that the dependency of the energy transfer coefficient Ka on the air and water flow rates in a tower can be well expressed by a relationship of the form

$$Ka = \alpha G^{\beta} L^{1-\beta} \tag{2.10}$$

where α depends on the fill configuration and β is, to a good approximation, equal to 0.6. Using the following expression

$$\alpha = 0.065 - (T_1 - 110.0)*(0.000335) \quad T_1 > 90°F$$

and (2.11)

$$\alpha = 0.0715 \quad T_1 \leq 90°F$$

where T_1 is the inlet water temperature, the performance predictions of Croley et al. based on proprietary data can be closely matched as shown in Fig. 2.3. This value of α is consistent with the type of fill used in modern towers and the values of α experimentally determined by Lowe and Christie [L4].

2.3 Spray Systems

2.3.1 Literature Review

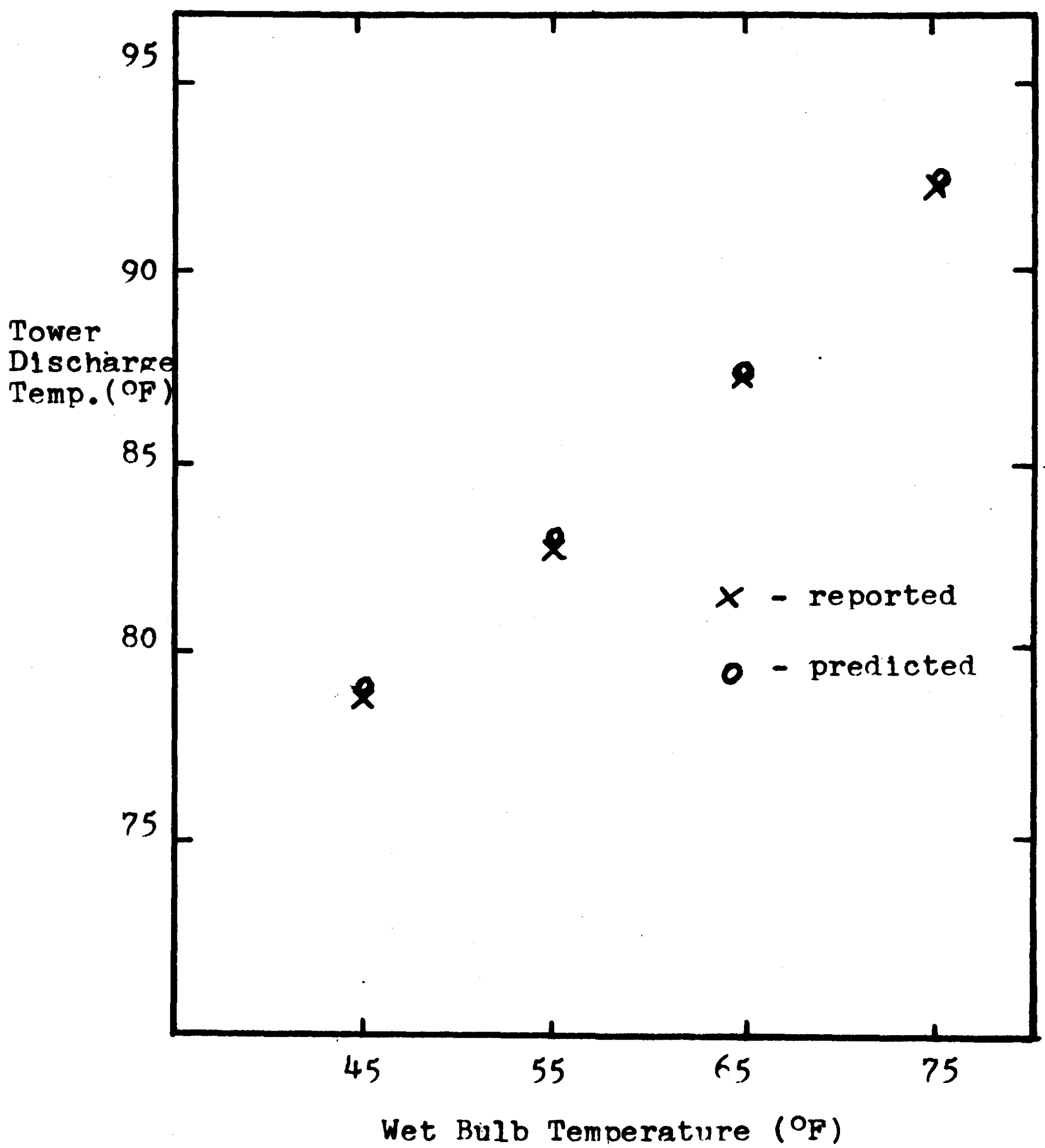
Spray cooling systems for the dissipation of waste heat at large central power stations are a relatively new concept [H9]. As a consequence, the development of thermal analysis techniques for these systems is presently incomplete. The development of reliable mathematical prediction models has not been achieved and has been hindered by the complexity of the problem.

Fig. 2.3

Comparison of Reported and Predicted Mechanical Draft

Cooling Tower Performance

Water Loading = 6200 lbm/hr - ft^2
Air Loading = 1692 lbm/hr - ft^2



As opposed to cooling towers, the water-air interfacial area and relative air to water flow rates are not well defined for spray systems. Open to the atmosphere, variations in the ambient wind result in different spray patterns, different air flows through the sprays both in magnitude and direction, and different interference effects between the individual sprays. The spray canal system also presents a channel hydraulics problem in that the behavior of the water in the canal must be understood to insure optimum spray system performance.

Porter et al. [P2][P3] have authored the only two presently available detailed works on the thermal performance of spray canals. The two papers represent two different approaches to the problem, one analytical and one numerical. Both models, however, are based on the same limited data which according to the authors result in optimistic predictions [P1].

Richards of Rockford [K8] have published some limited information concerning the application of their spray modules. They indicate that an empirical "NTU" approach is used in the basic heat transfer calculation. Most interesting, however, is their description of the flow requirements of the channel in which the spray modules are utilized since this description indicates their recognition of the importance of the channel thermal-hydraulics in the overall performance of the system.

2.3.2 Selection of Model

For the purposes of survey-type analyses, the numerical prediction of the thermal performance of spray canals as suggested by Porter et al. [P2] is most advantageous. In this model the heat transfer ability of each spray module is defined by an empirical "NTU" or number of transfer units which is dependent on the ambient wind speed. The effects of air interference between individual sprays is considered through the use of an empirical air humidification coefficient. Given the ambient meterological conditions and inlet water temperature and flow rate, the calculational procedure is to march down the canal taking into account the cooling effect of each spray module as it is encountered. The basic calculational algorithm is given in Fig. 2.4.

The heat transfer equation used in the model is

$$NTU = \frac{C_p (T_n - T_s)}{\frac{(h(T_s) + h(T_n))}{2} - h(Twb)} \qquad (2.12)$$

where C_p = specific heat capacity of liquid water,

T_n = temperature of water exiting spray nozzle,

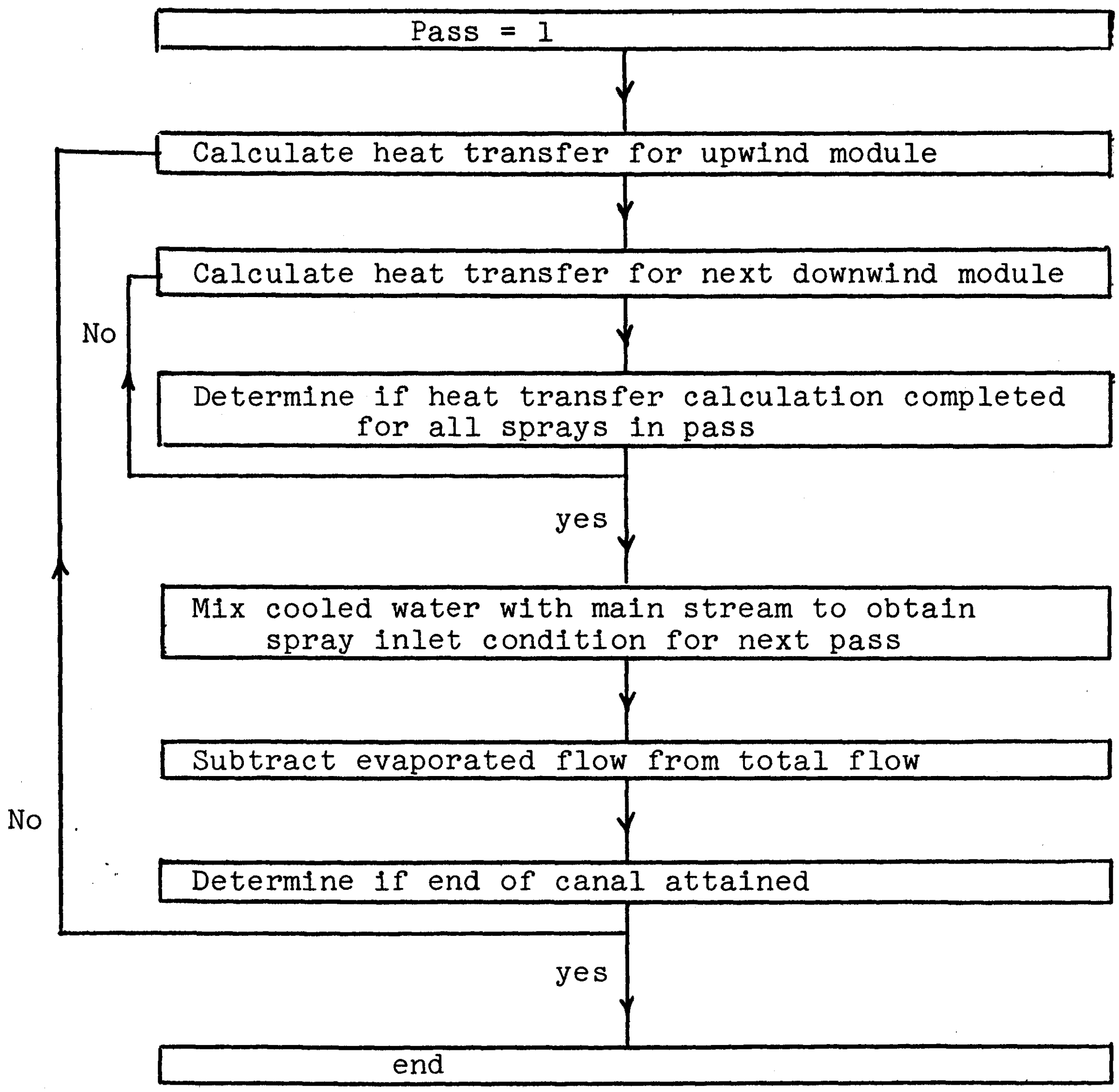
T_s = final spray temperature,

$h(T)$ = total heat or sigma function as defined by Marks [M2],

Twb = local wet bulb temperature, and

Fig. 2.4

Computational Algorithm for Spray Canal Thermal Performance Model (SPRANAL Program)



NTU = number of transfer units of an individual module.

The total heat or sigma function used as the driving potential for the energy transfer is defined by Marks [M2] as

$$\Sigma = h_m^* - W^* h_f^* \tag{2.13}$$

where h_m^* = enthalpy of moist air at the wet bulb temperature,

W^* = specific humidity for saturation at the wet bulb temperature, and

h_f^* = enthalpy of liquid water at the wet bulb temperature.

However, comparison of the sigma function and the enthalpy indicates that, for the temperature range and temperature differences of interest the following is a good approximation;

$$\Delta\Sigma(twb) \approx \Delta h(Twb) \tag{2.14}$$

where h is the enthalpy of saturated air at temperature Twb.

Since we are attempting to determine T_s by using Eq.(2.12) and T_s is a term in the same equation an iterative solution is necessary. The evaporated water loss is calculated using the expression of Porter [P2]. It is

$$\alpha = C_p(T_n - T_s)/i_{fg}(1 + B) \qquad (2.15)$$

where α = fraction of water evaporated in each spray,

i_{fg} = specific heat of vaporization of water, and

B = so-called Bowen ratio of sensible to evaporative heat transfer.

In the application of the above equations, the Bowen ratio can be conservatively set equal to zero, since, in any case, the effect of water evaporation on the spray canal thermal performance is minimal.

From the data given by Porter the relationship between the NTU and windspeed has been deduced to be approximated by

$$NTU = 0.16 + 0.053*V \qquad (2.16)$$

where V is the windspeed in miles per hour.

In this model no direct account is made of the thermal-hydraulic behavior of the water in the channel. However, Porter has made some simple arguments in favor of assuming that the channel is vertically fully-mixed between successive passes of sprays.

2.4 Natural Draft Evaporative Cooling Towers

2.4.1 Literature Review

Conceptually, the thermal analysis of natural draft evaporative cooling towers is a straightforward extension of the mechanical draft cooling tower analysis developed in this chapter. However, from a practical standpoint the problem is considerably more complex since the heat transfer characteristics and the air flow in the tower are dynamically coupled. Also, in addition to needing to know the empirical heat transfer coefficient of the fill, one also needs to know the empirical air friction factors for the tower structures and the fill. Further, a more exact determination of the psychrometric condition of the air exiting the fill is desirable since this condition ultimately determines the overall performance of the tower.

In the past, attempts have been made, notably by Chilton[C1] to simplify the performance prediction for natural draft evaporative cooling towers by applying an empirical relationship for the overall thermal behavior. These efforts, however, were not well received and presently the suggested approach to the thermal analysis problem is based on a detailed evaluation of the important physical phenomena.

Keyes [K3] has outlined the necessary steps for the construction of a thermal behavior model of natural draft cooling towers. Essentially, the mathematical modeling of a natural draft tower requires the solution of three coupled equations. The equations are 1) an energy balance between the air and water streams, 2) an energy transfer equation for the combined evaporative and sensible heat transfer, and 3) an energy equation for the density induced air flow through the tower. Keyes only reviews the general problem and discusses the empirical information which is available for accomplishing the modeling task.

Winiarski et al. [W4] have developed a computer model of the thermal behavior of a natural draft cooling tower based on the three equations mentioned above. The author notes, however, that the model presented awaits final verification based on reliable test data from actual towers.

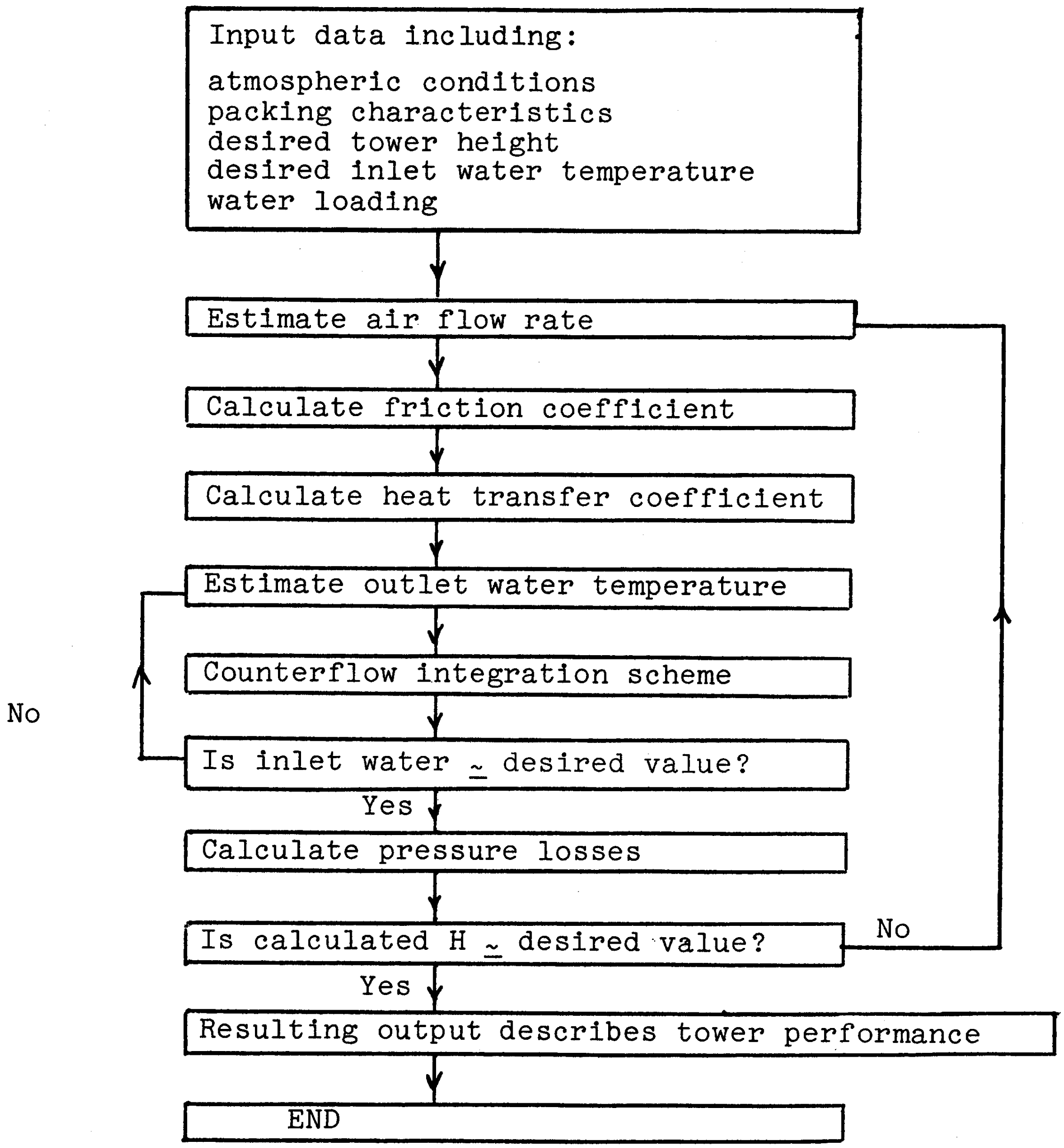
2.4.2 Selection of Model

The model of Winiarski et al. [W4] has been chosen as the basis for the development of a thermal behavior model of natural draft evaporative cooling towers. The thermal analysis calculational procedure is reported in the form of a computer program. The basic computational algorithm is given in Fig. 2.5. The major remaining task in the model development was, thus, the acquisition of the necessary empirical information which would enable the computer program application. In this regard all domestic vendors of natural draft evaporative cooling towers were contacted and sufficient information was obtained.

The data obtained was not typical heat transfer coefficients and air flow friction factors for a modern natural draft tower but instead consisted of a set of typical perfor-

Fig. 2.5

Calculational Algorithm for Natural Draft Evaporative Cooling Tower Performance Model (NATDRAFT Program)



mance curves and tower and fill structural dimensions. Thus, is was required to fit the computer model to the performance curves by a trial and error selection of appropriate heat transfer coefficients and friction factors. The performance data are known to be based on roughened-surface parallel-plate-type tower fill with counter air/water flow. Rish [R1] has reported an empirical relationship for the heat transfer coefficient and friction factors for smooth parallel plate packing. They are;

$$C_f = 0.0192(L/G)^{0.5} \quad , \tag{2.17}$$

and

$$h = \frac{C_p C_f G}{2 + 71.6 C_f (\frac{L}{G})^{-0.25}} \tag{2.18}$$

where C_f = friction factor,

C_p = specific heat capacity of liquid water,

G = air flow rate lbm/ft^2-hr,

L = water flow rate lbm/ft^2-hr, and

h = heat transfer coefficient for evaporative and sensible heat transfer based on enthalpy difference potential.

It was assumed that the effect of the roughened surface of the parallel plates could be simply accounted for by a friction factor multiplier F_m. That is;

$$C_{fa} = F_m * C_f \qquad (2.19)$$

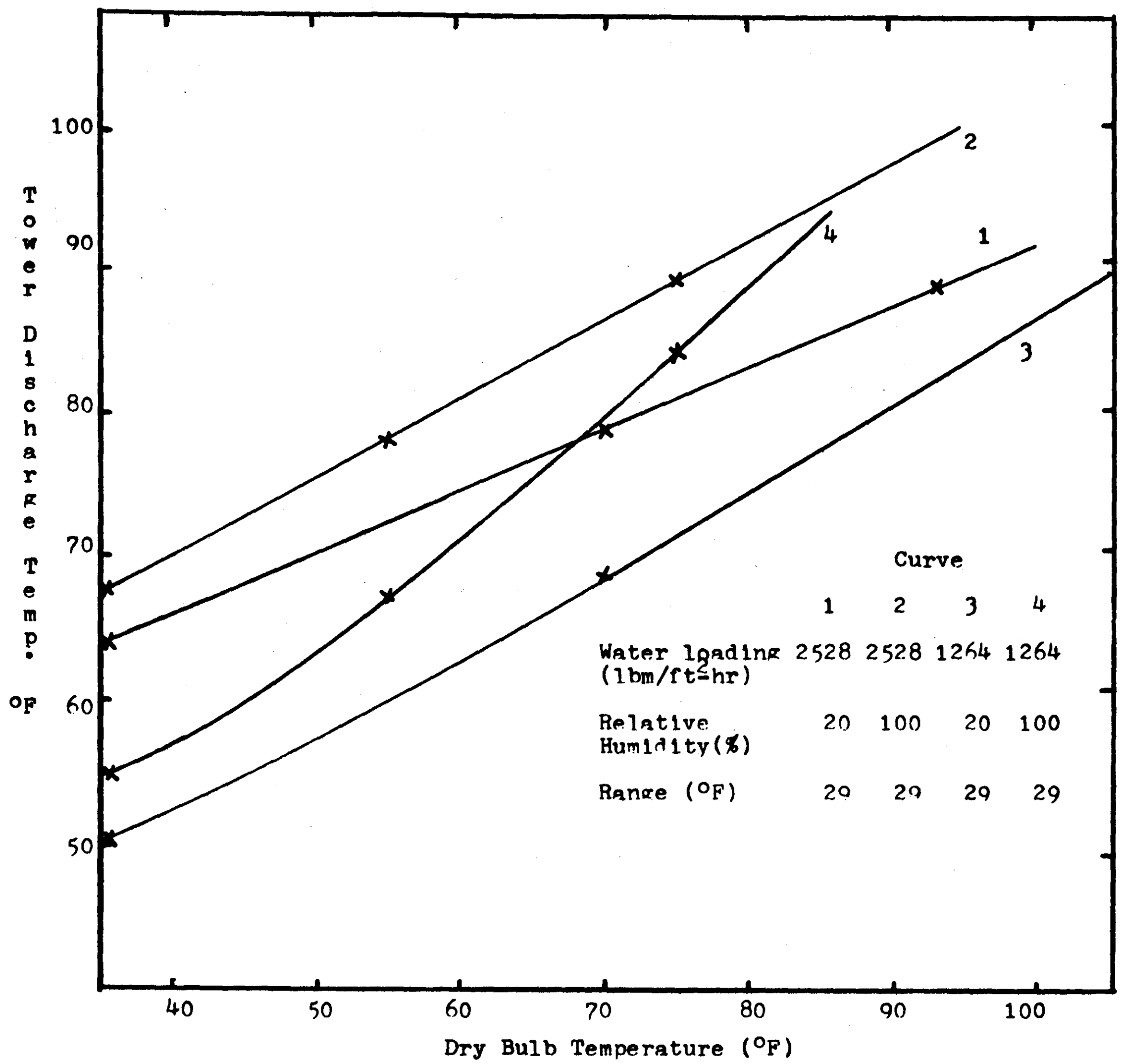
where C_{fa} is the actual friction factor. The relationship between the heat transfer coefficient and the friction factor was assumed to remain the same.

A trial and error approach to determining F_m was used and, as Fig. 2.6 indicates, a value of F_m of 3.2 gives excellent results over a representative range of operating temperatures and flow rates. In the determination of F_m all other air friction effects other than that of the fill were neglected.

All the details of the computer model will not be discussed here, but may be found in the original report. Nevertheless, some important points are worth mentioning. In this model, water vapor saturation of the air stream is not a basic assumption as was the case for the heat transfer model developed for the mechanical draft tower. Instead, the sensible heat transfer is calculated in addition to the total heat transfer due to both evaporation and sensible heat transfer. As in the mechanical draft tower model the transfer calculation is based on a finite-difference approximation to the Merkel Equation, but in this case the counter-flow of the air and water streams necessitates only a one-dimensional calculation.

Fig. 2.6

Comparison of Reported and Predicted Natural Draft Cooling Tower Performance



The calculation of both the total energy transfer and the sensible heat transfer allows the determination of the exact psychrometric condition (both dry bulb and humidity) of the air stream leaving each "cell" of the finite difference integration. The assumption of water vapor saturation, if it in fact did not exist, would result in an underestimate of the fill air exhaust dry bulb temperature and hence an underestimate of the induced draft.

To complete the thermal model of a natural draft tower a relationship between the tower height and the tower base diameter needed to be established for different sized towers. This was necessary because while a mechanical draft tower may be sized to a particular cooling duty by varying the number of tower modules, a natural draft tower is sized by varying the tower size. Flangan [F2] has published data concerning the ratio of height to diameter for 16 large natural draft towers which indicates an average ratio of 1.248.

2.5 Cooling Ponds

2.5.1 Literature Review

The task of mathematically modeling the thermal-hydraulic behavior of a cooling pond is a problem which is substantially different from the problem of modeling cooling towers. This is because actual cooling ponds are not physically well-defined in the sense that the important parameters which determine their thermal behavior can not be assigned values which are representative of all, or even most. cooling ponds. In fact different cooling ponds may exhibit completely different types of thermal-hydraulic behavior each of which require different analysis approaches and techniques.

There are two idealized cases of pond thermal-hydraulic behavior which yield themselves to very simple analytical treatment [L3]. These are termed the plug-flow and fully-mixed models. In plug flow there is no mixing between the discharge into the pond and the receiving water and the surface temperature, for steady-state conditions, decreases exponentially from the pond inlet to the pond outlet. The fully-mixed pond represents an extremely high degree of mixing of the discharge and the receiving water. Thus a uniform temperature over the entire pond results. In reality, the behavior of most ponds would fall between these two extreme cases. The plug flow pond represents the best possible heat dissipation situation since the temperature of the discharge is kept as high as possible. Conversely, the fully-mixed pond represents a lower bound on the heat transfer performance of the pond. The "worst case" performance, however, is a short-circuited pond. For either the plug-flow or fully-mixed model both steady-state and transient behavior can be readily calculated.

Ryan [R10] reported the development of a transient cooling pond thermal-hydraulic model which was the first attempt to realistically mathematically model the actual physical process occurring in a cooling pond. Watanabe [W1] extended the model and reported criteria for its applicability. This model is recommended for use as a design tool or means of evaluating the performance of cooling ponds relative to alternative waste heat disposal systems. However, since the model is not fully developed into a documented computer program its application appears difficult. Also, for the purposes of most surveys the computational time is excessive.

2.5.2 Selection of a Model

The task of formulating a representative thermal-hydraulic model of a cooling pond can be considered to be different from the task of formulating a model of a cooling pond which is to be used for design purposes. The present interest is in mathematically representing the approximate thermal-hydraulic behavior of a representative cooling pond. It is perceived that this limited goal can be accomplished through the use of a plug-flow, vertically-mixed pond model capable of accounting for variable meterological conditions, variable inlet temperatures, and variable flow rates. For a given cooling requirement such a model would tend to predict pond sizes which are smaller than would be normally required. Thus, if the model were to be used in a detailed economic comparison of alternative waste heat disposal systems the pond economics would be unduly favored.

The vertically-mixed, plug-flow model predicts the transient pond behavior by following a slug of water of uniform temperature through the pond and calculating the average heat loss for each successive day of residence in the pond. The heat transfer correlations used in this model are those recommended by Ryan [R10]. The basic equation of the net energy flux from a water surface exposed to the environment is

$$\phi_n = \phi_r - \left[4.0\text{x}10^{-8}(T_s+460) + FW[(es-ea) + 0.25(T_s-T_a)]\right] \tag{2.20}$$

where $FW = 17*W$ for an unheated water surface,

$FW = 22.4(\Delta\theta)^{1/3} + 14*W$,

$\Delta\theta = T_{sv} - T_{av}$ (°F),

W = wind speed at 2 meters (MPH),

T_{sv} = virtual temperature of a thin vapor layer in contact with the water surface,

$= (T_s + 460)/(1 - .378\ es/P)$,

T_{av} = virtual air temperature,

$= (T_a + 460)/(1 - .378\ ea/P)$,

es = saturated vapor pressure at T_s (mm Hg),

ea = saturated vapor pressure at T_a (mm Hg),

P = atmospheric pressure (mm Hg),

T_s = bulk water surface temperature (°F),

T_a = air dry bulb temperature (°F),

ϕ_n = net heat from pond surface (BTU/day-ft^2)

$\phi_r = \phi_{sn} + \phi_{an}$ = net absorbed radiative energy,

ϕ_{sn} = net absorbed solar radiation,

$= .94(\phi_{sC})(1 - 0.64C^2)$

ϕ_{sC} = incident solar radiation,

C = fraction of sky covered by clouds,

ϕ_{an} = net absorbed longwave radiation, and

$= 1.16\text{x}10^{-13}(460 + T_a)^6(1 + 0.17C^2)$.

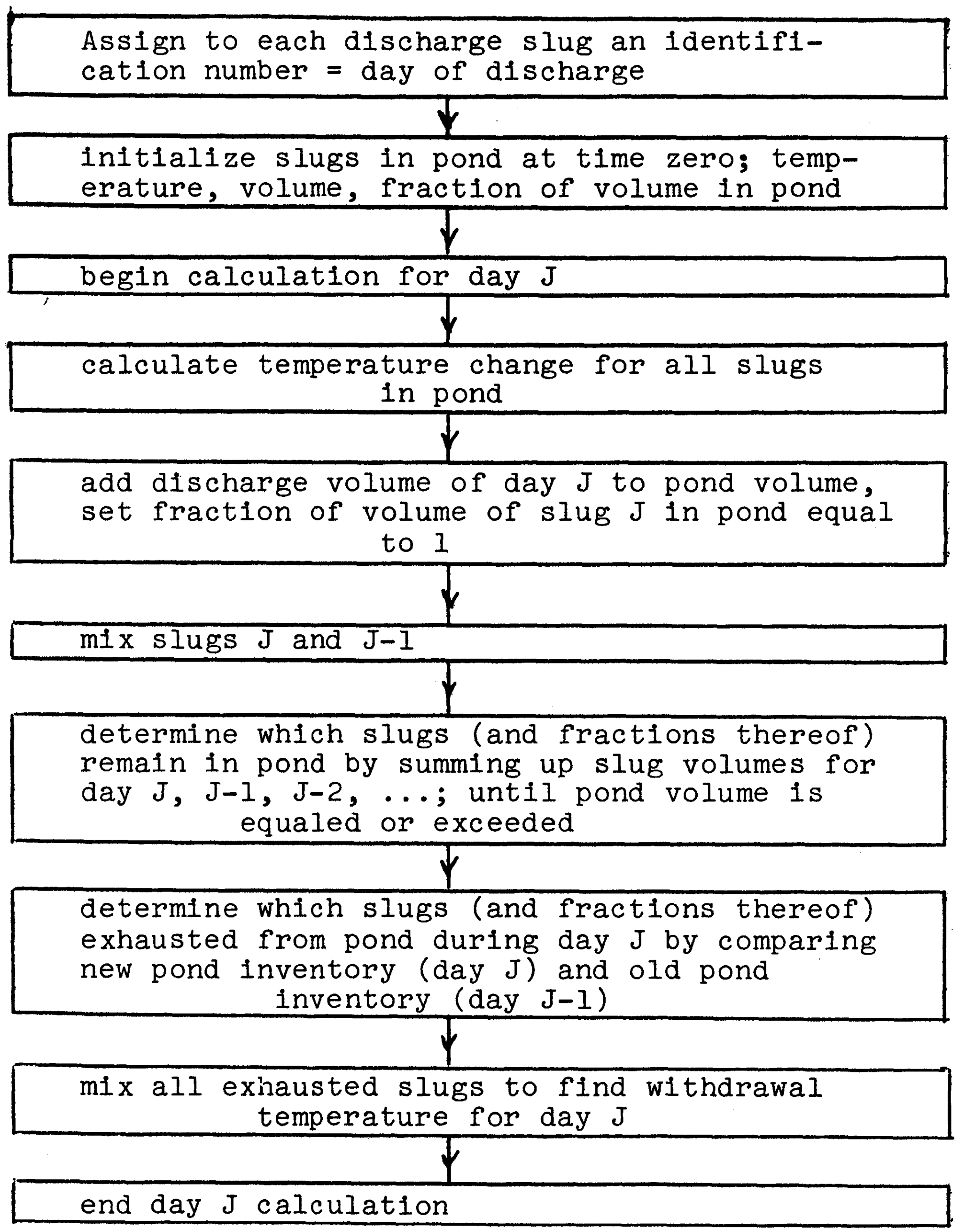
The computational algorithm for the plug-flow model is given in Fig. 2.7. Note that the model is not a perfect plug-flow model in that each plug of water entering the pond is assumed to be mixed with the slug immediately preceeding it. This mixing qualitatively accounts for the effect of entrance mixing.

2.6 Dry Cooling Towers

In relation to the other waste heat dissipation systems, the development of a reliable performance model of dry cooling towers is simple. The amount of heat rejected by a mechanical draft dry tower can be shown to be directly proportional to

Fig. 2.7

Cooling Pond Model Computational Algorithm



the difference between the inlet water temperature and inlet air temperature for a fixed dry tower design. With reference to Fig. 2.8

$$Q = UA\Delta T_{lm}F_g \tag{2.21}$$

where Q = heat rejection rate,

A = heat transfer surface area,

U = effective heat transfer coefficient,

F_g = cross-flow correction factor, and

ΔT_{lm} = log mean temperature difference.

$$\Delta T_{lm} = \frac{(T_o - T_i') - (T_i - T_o')}{\ln \frac{(T_o - T_i')}{(T_i - T_o')}} \tag{2.22}$$

where $(T_o - T_i') > (T_i - T_o')$

T_i = water inlet temperature,

T_o = water outlet temperature,

T_i' = air inlet temperature, and

T_o' = air outlet temperature.

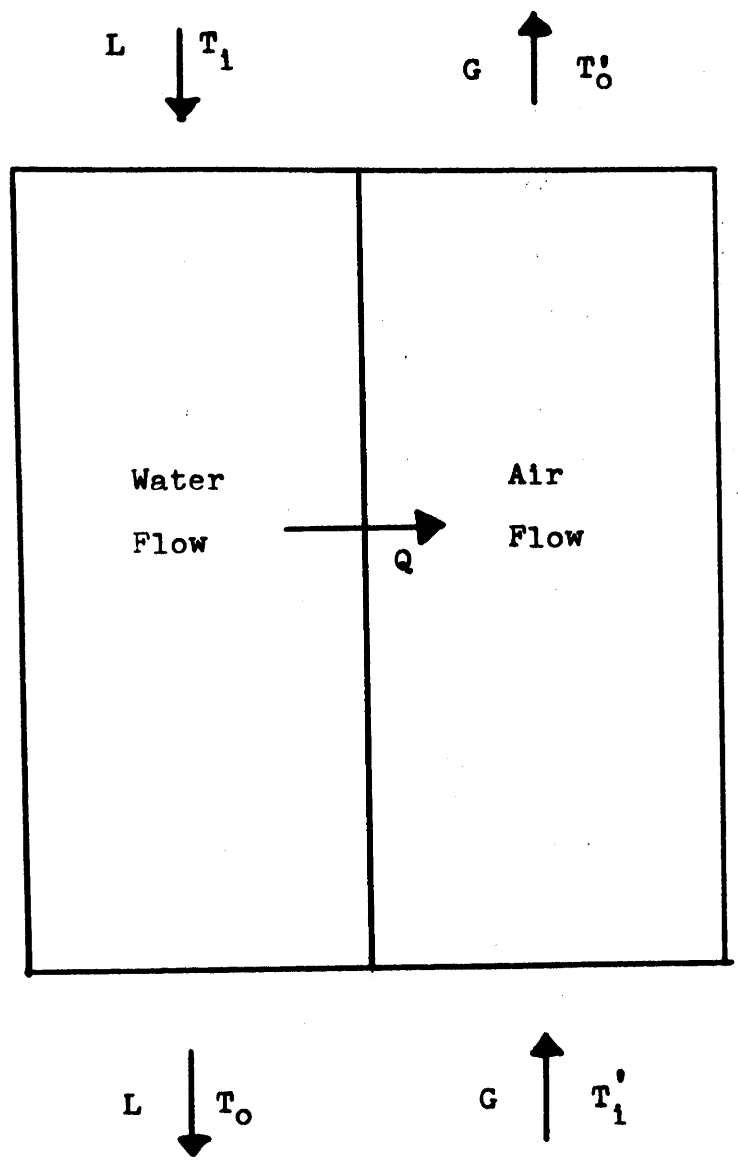
A heat balance on the tower gives

$$LC_w(T_i - T_o) = GC_a(T_o' - T_i') \tag{2.23}$$

Equations (2.21), (2.22) and (2.23) may be combined to yield

Fig. 2.8

Dry Tower Schematic Drawing



$$Q = \frac{ITD(e^x - 1)}{\frac{e^x}{GC_a} - \frac{1}{LC_w}} \tag{2.24}$$

where $\quad ITD = T_i - T_i'$,

and

$$x = F_g UA \left[\frac{1}{GC_a} - \frac{1}{LC_w} \right] .$$

Now note that, for fixed values of the parameters U, A, F_g, G and L,

$$Q \; \alpha \; ITD \tag{2.26}$$

This result has been found by Rossie [R6] to be experimentally verified. Further, Rossie has found that the thermal performance of natural draft dry cooling towers may be reasonably expressed by a relationship of the form

$$Q \; \alpha \; ITD^b \tag{2.27}$$

where b is a constant for a given tower. A typical value of b is 1.33.

CHAPTER 3

SURVEY OF SOME MIXED-MODE WASTE HEAT REJECTION SYSTEM OPTIONS

3.1 INTRODUCTION

The task of evaluating possible mixed-mode waste heat dissipation options has been approached as discussed in Chapter 1. The results of this survey are included in three sections of this chapter. Each section discusses a particular application of the mixed-mode concept. One particular system - the thermal storage pond and dry cooling tower system -- is shown to offer a substantial economic benefit.

3.2 IMPROVEMENTS IN WASTE HEAT SYSTEM UTILIZATION ECONOMICS THROUGH THE USE OF MIXED-MODE SYSTEMS

Because some waste heat rejection systems have higher ratios of operational to capital cost and because the conditions which determine the required amount of heat rejection capability are variable, it is of interest to examine the potential benefit of constructing waste heat dissipation systems composed of two component systems. One component system would have a lower ratio of operating cost to capital cost and would be used continuously. The other component system would have a substantially higher ratio of system operating to capital costs and would be used only as conditions required. Two such systems would be a natural draft evaporative cooling tower and a mechanical draft evaporative cooling tower. Based on the results of the economic studies presented in WASH 1360 for typical optimal tower systems for a 1000MWe nuclear power station the cost ratio for these two types of heat rejection systems for a 40-year-lifetime plant are;

Natural draft tower

$$R_n = \frac{\text{Capitalized Annual Operating Cost}}{\text{Capital Cost}} = 0.78$$

Mechanical Draft Tower

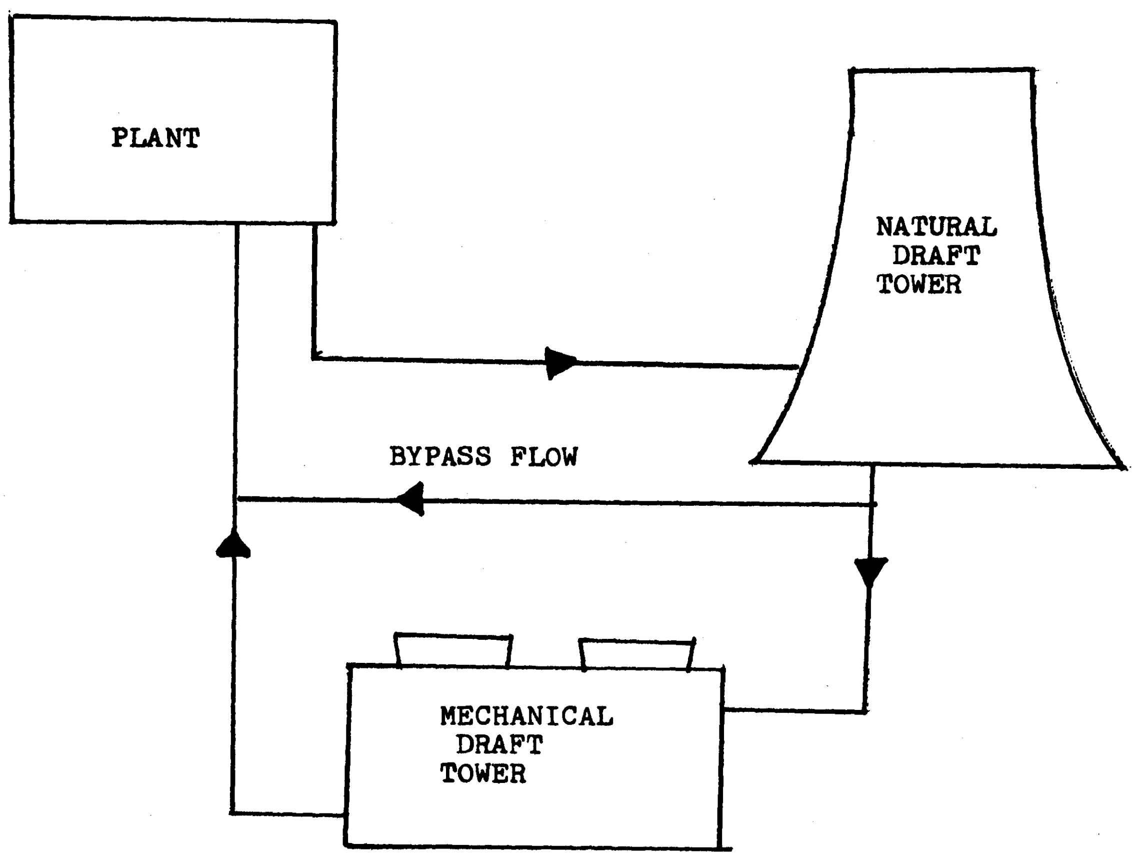
$$R_m = 1.98$$

To evaluate the potential benefit of combining these two cooling tower types a hypothetical plant/combined cooling tower system has been designed. The system is composed of a dual unit station (1000MWe each) cooled by a series flow combination of a 514 ft. high natural draft evaporative cooling tower and a 44-cell mechanical draft evaporative cooling tower as shown in Fig. 3.1.

For this system the most desirable condenser inlet temperature is 70°F. Higher condenser inlet temperatures result when ambient temperatures are high. When the temperature is low, however, the condenser inlet temperature

FIG. 3.1

Combined Mechanical Draft Tower - Natural Draft Tower System



is not allowed to fall below this value by shutting off some of the mechanical draft tower cells and bypassing the flow directly to the condenser intake. For the hypothetical plant-towers system the $70^{o}F$ condenser intake temperature occurrs when the ambient dry bulb temperature equals $50^{o}F$ and the ambient wet bulb temperature equals $47^{o}F$. This relationship of the ambient temperature and optimal condenser intake temperature can be considered to representative of closed-cycle evaporative cooling systems.

To determine the approximate utilization factor of the mechanical draft towers, defined by

$$f = \frac{\text{Yearly average \# of tower cells operating}}{\text{Total \# of cells}} ,$$

the number of cells, N_i, required at each $10^{o}F$ increment of the dry bulb temperature from 10 to $90^{o}F$ has been determined. Using the annual temperature duration curves for ERDA's Middletown site as shown in Fig. 3.2, f is determined by

$$f = \frac{\sum_{i=10}^{i=90} D_i \cdot N_i}{N_t \sum_{i=10}^{i=90} D_i} \tag{3.1}$$

where i = dry bulb temperature, increment of $10^{o}F$,

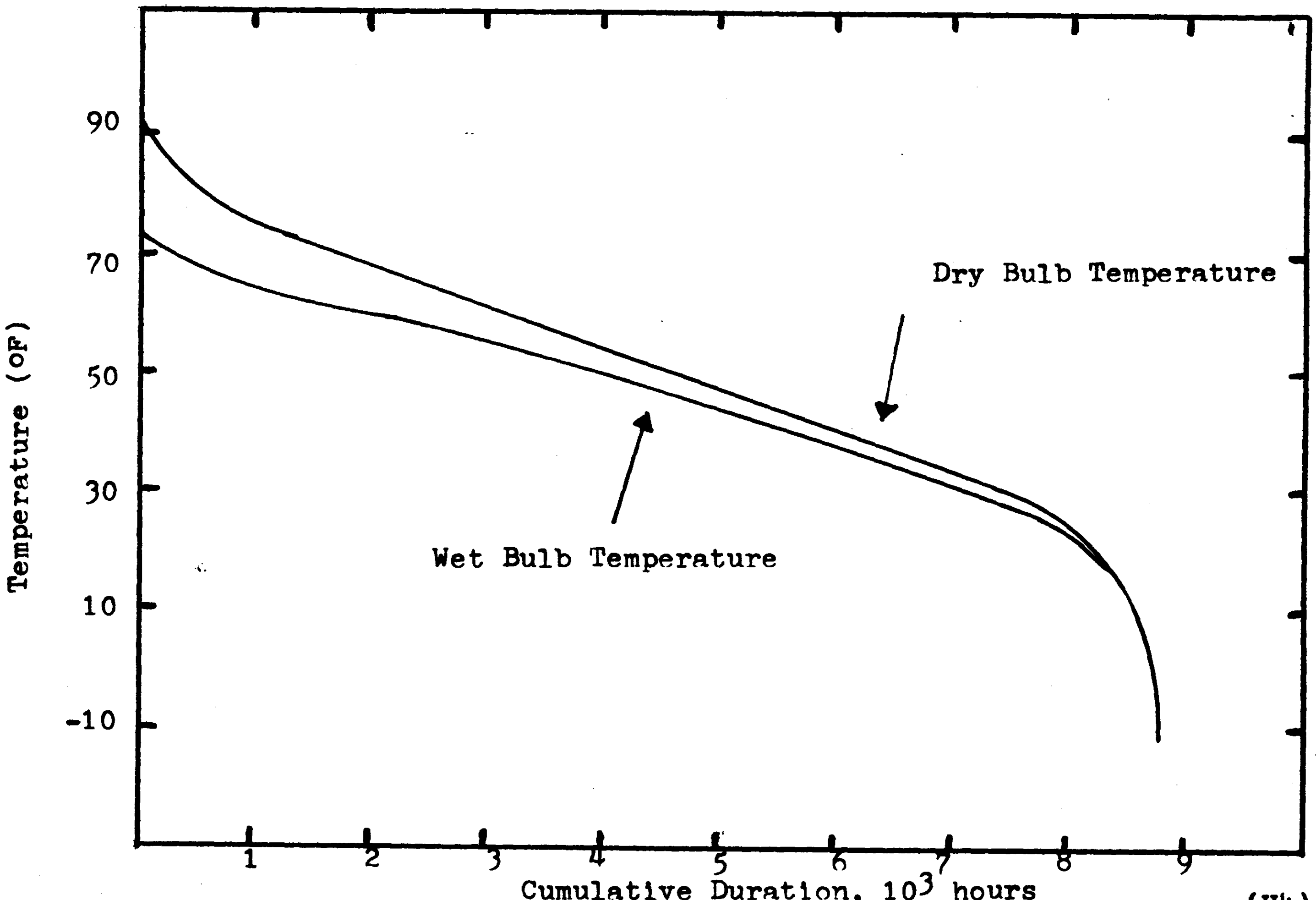


Fig. 3.2 Annual Temperature Duration Curves, Middletown, USA (H4)

N_i = number of cells required for plant operation at temperature i,

D_i = the annual duration of temperatures above i-5 and below i+5, and

N_t = total number of mechanical draft tower cells.

Actually, the performance of the waste heat rejection system depends on both the dry bulb temperature and the wet bulb temperature both of which are given in Fig. 3.2 and both of which are essentially independent variables. Nevertheless, for the purposes of this preliminary evaluation it is sufficient to assume a one to one correspondance of the wet bulb temperature to the dry bulb temperature based on equal cumulative duration. This assumption can be construed as the assignment of an average wet bulb temperature to each value of the dry bulb temperature.

Completion of the coupled plant-waste heat system performance calculation for the hypothetical station yields the results presented in Fig. 3.3. This figure shows the number of mechanical draft tower cells which are needed to maximize the plant power output as a function of the ambient dry bulb temperature. Note that at all ambient temperatures greater than 55^{o}F all the mechanical draft tower cells are required and some loss of plant generating capability occurrs. The shifing of the main heat

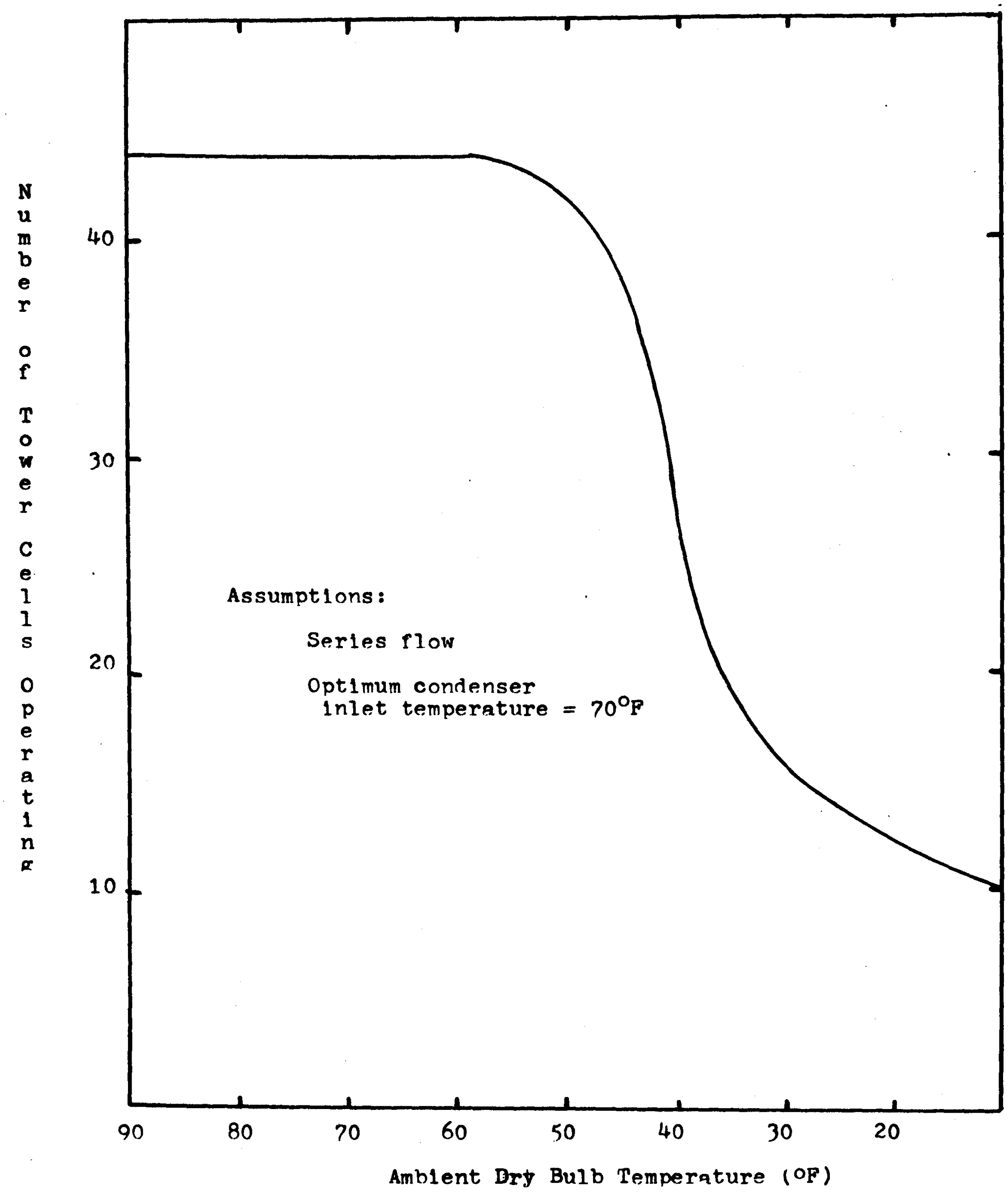


Fig. 3.3 Number of Mechanical Draft Cells Required as a Function of Ambient Dry bulb Temperature

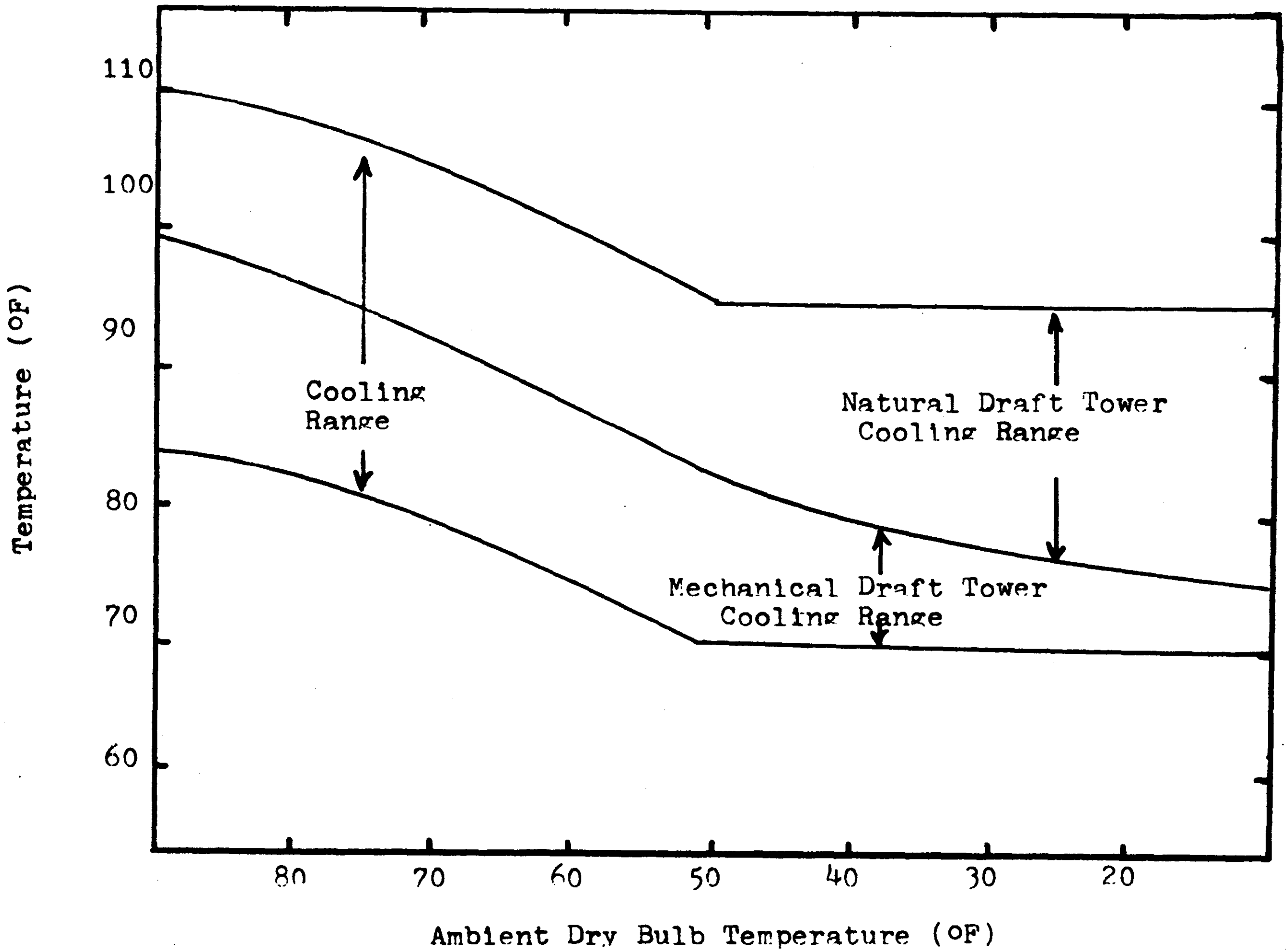


Fig. 3.4 Illustration of Increased Natural Draft Tower Loading at Low Ambient Temperatures

rejection load to the natural draft tower at ambient temperatures below 50^{o}F is shown in Fig. 3.4.

Now using the information in Figs. 3.2 and 3.3 to determine the mechanical draft tower utilization factor defined earlier we find that

$$f = 0.92$$

Thus, the annual average number of tower cells in operation is about 40 out of 44 and thus only about 10% of the maximum mechanical draft tower opeation cost could be saved. The total cost of the combined waste heat rejection system can be adequately expressed as

$$C_t = C_{cn} + C_{cm} + C_{on} + C_{om}$$

where C_{cn} , C_{cm} = capital cost of natural draft and mechanical draft towers, respectively, and

C_{on} , C_{om} = capitalized annual operating cost of the natural and mechanical draft towers, respectively.

Now inputing values of these costs (based on WASH 1360 (H4)) to determine the total cost of the combined waste heat system and comparing this cost to the calculated mechanical draft tower operation savings it is found that the saving resulting from the utilization of the combined waste heat system is only about 3% of the total waste heat system cost.

This cost savings is not significant and would be easily outweighed by the economics of scale inherent to a single-mode waste heat system. Thus, it is possible to conclude that waste heat system utilization considerations should not provide, in most circumstances, a basis for the design of mixed-mode waste heat rejection systems. Although this conclusion is based on the analysis of the particular case of combined mechanical and natural draft evaporative cooling towers, it is perceived that this conclusion would be valid for combined systems composed in whole or in part of cooling ponds (low cost ratio) or spray systems (high cost ratio). Additionally, waste heat system utilization considerations do not appear to offer significant savings for the case of combined natural and mechanical draft dry tower systems as indicated by Fig. 3.5. Figure 3.5 illustrates the extent to which the mechanical draft tower component system for a combined natural draft dry tower and mechanical draft dry tower system must be utilized to maximize the plant power generation. The hypothetical dry cooling system is composed of one natural draft dry cooling tower capable of rejecting one-half of the total waste heat load with an initial temperature difference (ITD) of $60^{o}F$

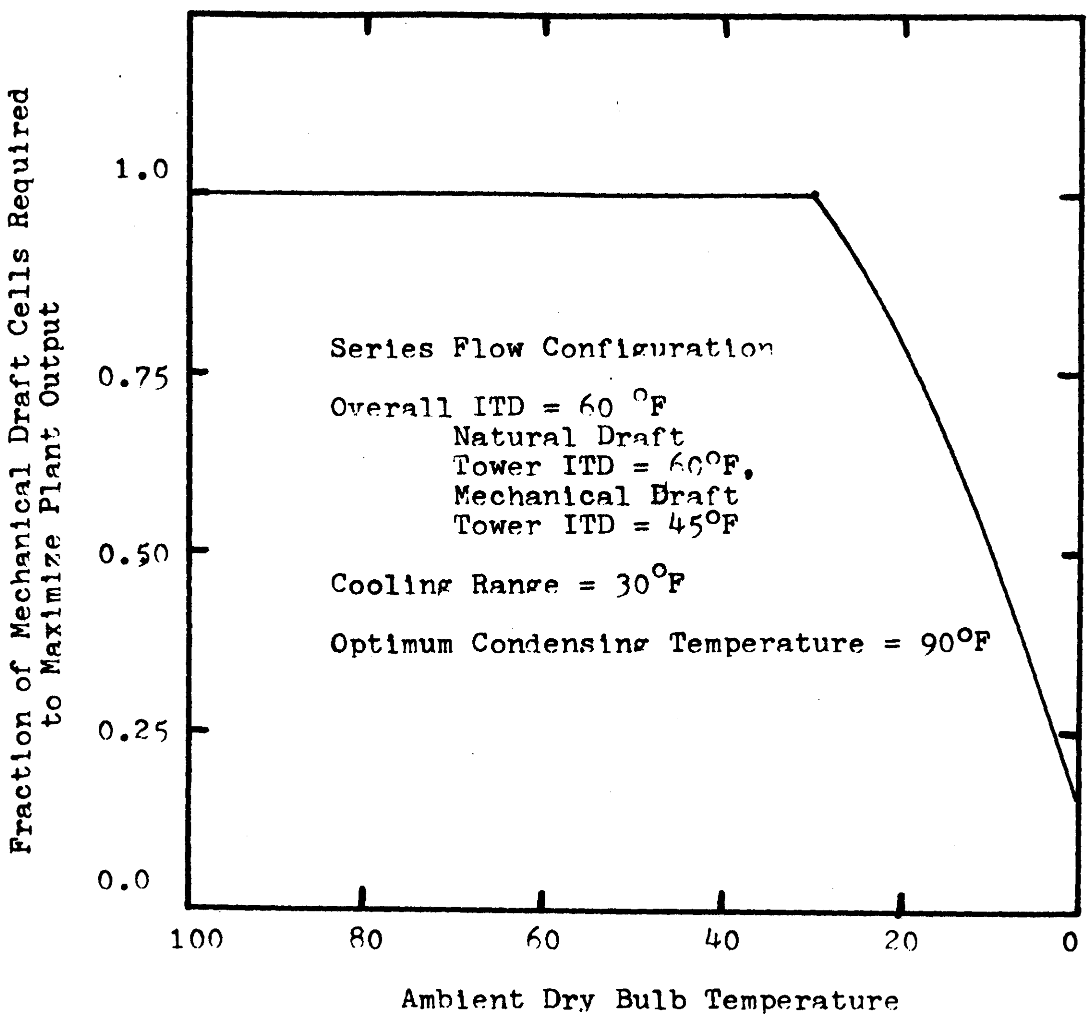


Fig. 3.5 Combined Natural Draft and Mechanical Draft Dry Cooling Tower Operation

and a multi-cell mechanical draft dry cooling tower capable of rejecting one half of the waste heat load with an initial temperature difference of 60°F.

In considering the above conclusions, it should be understood that the general concept of utilizing combined waste heat rejections systems is not ruled out. We have only illustrated that the potential savings resulting from the application of combined systems to achieve waste heat rejection "load-following" are minimal. Also, nothing in this work indicates that mechanical draft assisted natural draft cooling towers are an unattractive concept. However, the results of this work do indicate that the savings obtained through the shutting-off of the assisting fans in this type of tower during favorable heat rejection conditions would in most cases be small in relation to the total costs of the waste heat system. The summary capital and operation economics for fan assisted natural draft towers reported in WASH 1360 substantiate this conclusion.

3.3 The Design of Mixed-Mode Systems for Sites With a Limited Cooling Pond Resource

3.3.1 Introduction

Because of their large thermal inertia and large land requirements, cooling ponds may represent an attractive but limited heat rejection option at many central power station sites. Utilization of the avilable cooling pond resource would therefor necessitate the combination of the cooling pond with an alternative heat rejection device in order to meet the total heat rejection requirements of the station. Fan (F1) has examined such combinations but the findings of this study are inconclusive.

In designing such a combined system there are several system configurations options which the waste heat rejection system designer might consider. These design options would include:

1) a series combination with the pond receiving the condenser discharge and feeding and alternative system,
2) a series combination in which the alternative system discharges into the pond and the pond feeds directly to the condenser,
3) a parallel combination in which the discharge from the condenser is split between the pond and the alternative system

The objective of this work is the evaluation of the comparative economics of these system configuration options.

In attempting to perform this evaluation it readily becomes apparent that there is considerable difficulty in formulating both the thermal performance and economic models of cooling ponds. Simply stated, this difficulty arises from the strong dependency of the pond thermal-hydraulic behavior and cost on the physical and economic characteristics of the site. This difficulty may be overcome, however, by structuring the evaluation such that the economics of the pond itself are divorced from the resultant conclusions and by modeling the thermal-hydraulic behavior of the pond with a representative model.

The pond economics are separated from the evaluation by stating the problem in the following manner: given a pond of X acres at no cost, which alone will not provide sufficient yearround cooling for a hypothetical power station, what is the best method of incorporating the pond in a combined waste heat system such that the total cost of the waste heat system is minimal. The cost of the system is the sum of the capital cost of the alternative system and the operational cost (including loss of capability penalties) of the entire waste heat rejection system. For the present evaluation the alternative system is taken to be a mutiple-cell, induced-mechanical-draft, cross-flow, evaporative cooling tower. The operational costs of the combined waste heat systems are based on the computer simulation of the plant-tower-pond system performance for one year periods.

3.3.2 The Model

3.3.2.1 Plant

Since the interaction of the waste heat system with the power plant is important in determining the total economics of the waste heat rejection system, a representative plant model is incorporated into the simulation model. The plant modeled is a dual unit nominal 2000 MWe nuclear power station with a conventional nuclear steam turbine and surface condenser. The two units are assumed to operate at a combined full thermal power of 6000Mwt continuously.

3.3.2.2 Cooling Pond

The cooling pond thermal-hydraulic model employed in the evaluation is the simple vertically-fully-mixed, plug-flow model. Although this model is not accurate in representing the behavior of actual cooling ponds, this model should be adequate since our basic interest in this work is not the sizing of a cooling pond. Rather, our interest is in representing the special heat transfer and thermal capacitive characteristics of cooling ponds such that a survey of the economic implications of the transient behavior of various plant-cooling pond-cooling tower systems is possible.

The details of the cooling pond model are found in Sec. 2.5.

3.3.2.3 Cooling Tower

The cooling tower performance model used in the evaluation is identical to that discussed in Sec. 2.2. The total tower is composed of multiple cells of fixed dimensions. The number of cells comprising the total tower is varied according to the cooling demand of the system.

3.3.2.4 The Site

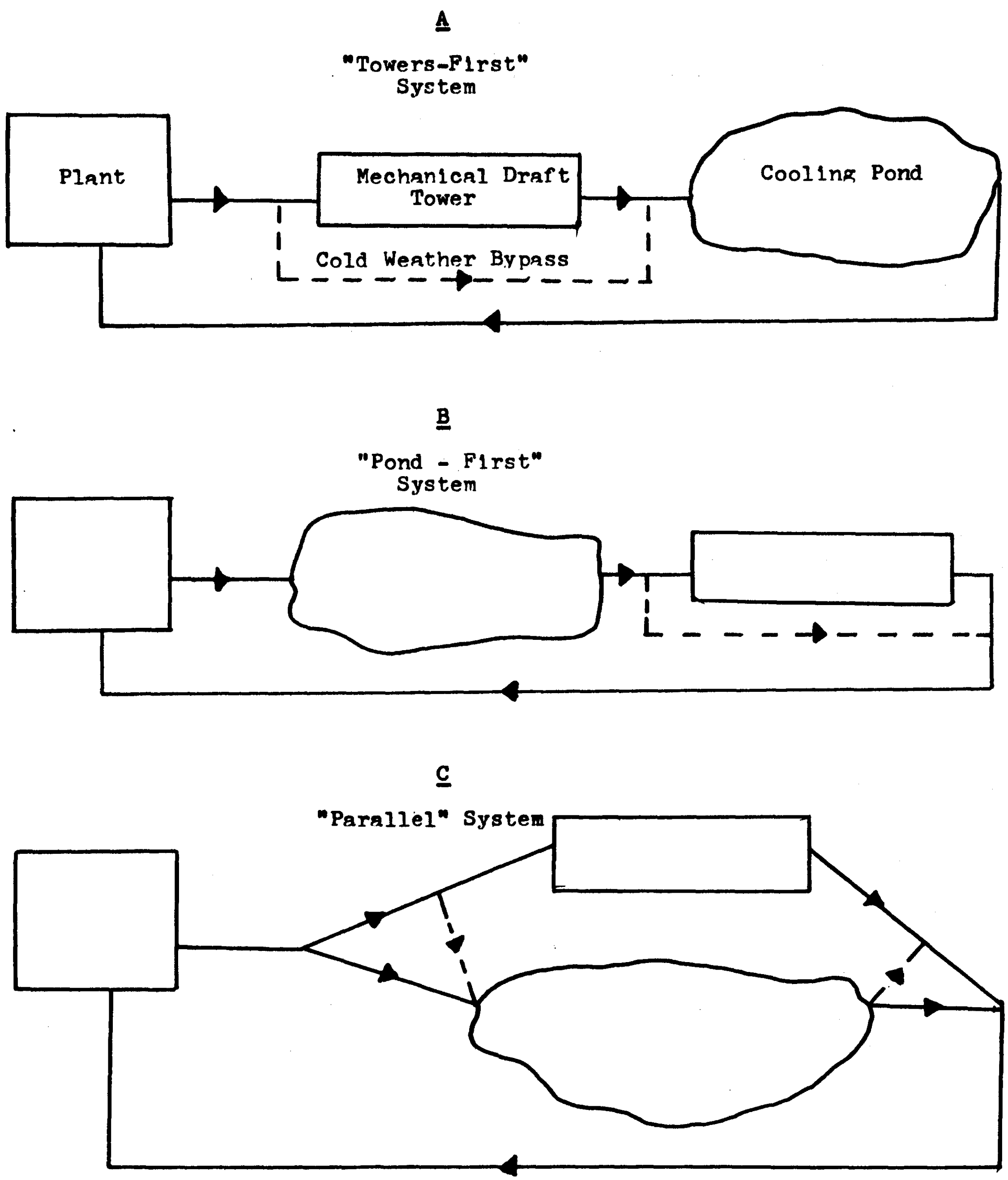
The steady-state heat transfer performance of combined systems is independent of the site. However, in order to determine optimal system configurations based on economic evaluations it is necessary to perform year-long simulations of the combined system behavior. The daily calculations of the plant response to variable meteorology are necessary to evaluate operational costs and penalties for the different combined system configurations. The meteorological data input included daily average dry bulb temperature and relative humidity and monthly average windspeed, cloud cover, and insolation. As previously stated, no cooling pond capital costs are introduced into the system optimization and, thus, no information concerning the physical characteristics of the site as they relate to the capital cost of the pond need be specified. The meteorology input to the model is that for Boston, Mass., 1974.

3.3.2.5 Combined System Options

The three basic combined system configuration options are shown in Fig. 3.6. Fig. 3.6a shows a "towers-first" series flow system. In this system the full circulating waster flow is routed first through the tower and then through the pond. During periods of favorable heat rejection conditions part of the condenser discharge can be directly routed to the pond and some of the towers removed from service. Fig. 3.6b, the "pond-first" system, is the reverse of the above. Again during periods of favorable heat rejection conditions, the tower can be partially bypassed with the pond discharging a fraction of the flow directly to the condenser. Fig. 3.6c is a simple "parallel" flow configuration. In this system cold weather tower operation expenses can be saved by increasing the flow to the pond and removing from service the appropriate number of cooling tower cells.

For all the above configurations the water loading of the tower cells (lb_m / ft^2-hr) is constant. Reduction of the total flow to the entire tower is achieved by reducing the

Fig. 3.6 Combined Cooling Pond - Tower System Configurations



number of towers in operation.

Since the optimal design evaluation performed in this work is based on a given pond size and a given tower cell design, the design variables for the series flow combinations are simply the circulating water flow rate and the number of tower cells. For the parallel case an additional design variable must be considered which determines the flow split between the pond and the tower. Including the tower cell water loading along with the number of tower cells and the circulating water flow rate as design variables satisfies this requirement.

3.3.2.6 Combined System Economics

The optimal combined waste heat system is found by comparing the costs of the least-cost "towers-first" system, the least-cost "pond-first" system , and the least-cost "parallel" system. The least-cost for each of the different configurations is found by varying the design variables over the range of feasible values and calculating the total cost of the waste heat rejection system for each set of design variables. The cost of waste heat system for each set of values of the design variables is based on a year-long operation simulation (one day time increments) and is given by the following equation;

$$C_t = C_c + C_a \;/\; afcr$$

where

C_t = Total evaluated cost,

C_c = capital cost,

C_a = annual operating cost and penalties, and

afcr = annual fixed charge rate.

The term C_a / afcr is an effective capitalization of the annual costs. The annual fixed charge rate is defined as the percentage of a capital investment which must be paid out each year for interest on borrowed capital, retirement of capital, taxes, etc. The total evaluated cost C_t is an approximate measure of the lifetime cost of the system and is useful for system comparisons. A more detailed present worth analysis would be of little additional benefit although it would be more precisely accurate.

The capital cost of the waste heat system is composed of the cost of the tower cells, the cost of the circulating water pumps, the tower booster pumps, and the cost of replacement generating capacity. The capital cost of the condenser, the circulating water piping, and all other waste heat system structures is assumed to be constant for all system designs and thus is not directly considered in the comparative analysis.

The loss of capability capital cost is determined by the annual minimum average power output for a one day period.

The annual operating costs include costs for operating the main circulating water pumps, the tower booster pumps, and the tower fans in addition to maintenance costs and energy replacement costs due to loss of generating capability. The main circulating water pumping cost and the booster pumping cost are evaluated at 30 and 50 feet of head, respectively. In all cases, fan and booster pumping power credits are taken for the decreased utilization of the tower during cold weather. The maintenance cost of the system is taken to be directly proportional to the number of tower cells. The energy replacement cost is evaluated by determining the difference between the theoretical maximum annual station electricity generation and the actual annual generation of electricity. The difference is multiplied by a fixed replacement generation cost per kilowatt-hour to yield the annual cost.

The unit costs used to calculate each of the above costs are mainly based on the results published in WASH 1360 (H4) and are given in Table 3.1.

3.3.2.7 Optimization Technique

Since the series combination optimization involved only two design variables the complexities associated with the use of a guided search of the feasible design space to find the global minimum of a nonlinear,numerically evaluated objective function were avoided in favor of simply constructing a two-dimensional matrix of discrete values of the design variables (circulating water flow rate and the number of tower cells)

TABLE 3.1

UNIT COST SUMMARY
FOR COOLING POND-TOWER SYSTEM

1.0 Capital Costs	
a) Tower cell	= \$162,000
b) Pumps	= \$125/hp
c) Peaking Capacity	= \$200/Kw
2.0 Operating Costs	
a) Energy replacement	= 8.5 mills/Kwh
b) Maintenance	= \$39,000/cell-yr
Annual Fixed Charge rate	= 15%

and calculating the previously defined total cost for each set of values. The discrete values of the variables are taken to be regular increments over a limited range. Past experience with the design of waste heat rejection systems allows the design variables to be restricted to a limited range.

The three variable problem of the "parallel" combination is solved by simply performing the 2-dimensional calculation as indicated above and then determing the effect of the selection of different values of the tower water loading.

3.3.3 Results of Combined System Evaluation

The principal results of the optimization study are shown in Fig. 3.7. The meaning of the labels "tower-first", "pond-first", and "parallel" have been previously defined. As indicated, the results are based on the use of a 450 acre pond. However, this number is of no real physical significance since an idealized pond thermal-hydraulic behavior (plug-flow) has been assumed. Of importance, nevertheless, is the fact that the pond is representative of a cooling pond which is capable of meeting about 1/2 of the total heat rejection requirements of the station. For clarity, Fig. 3.7 shows only the total cost as a function of the number of tower cells. The circulating water flow rates indicated are those which result in the least-cost for each of the particular configuration options. The values of the combined system design variables of ;

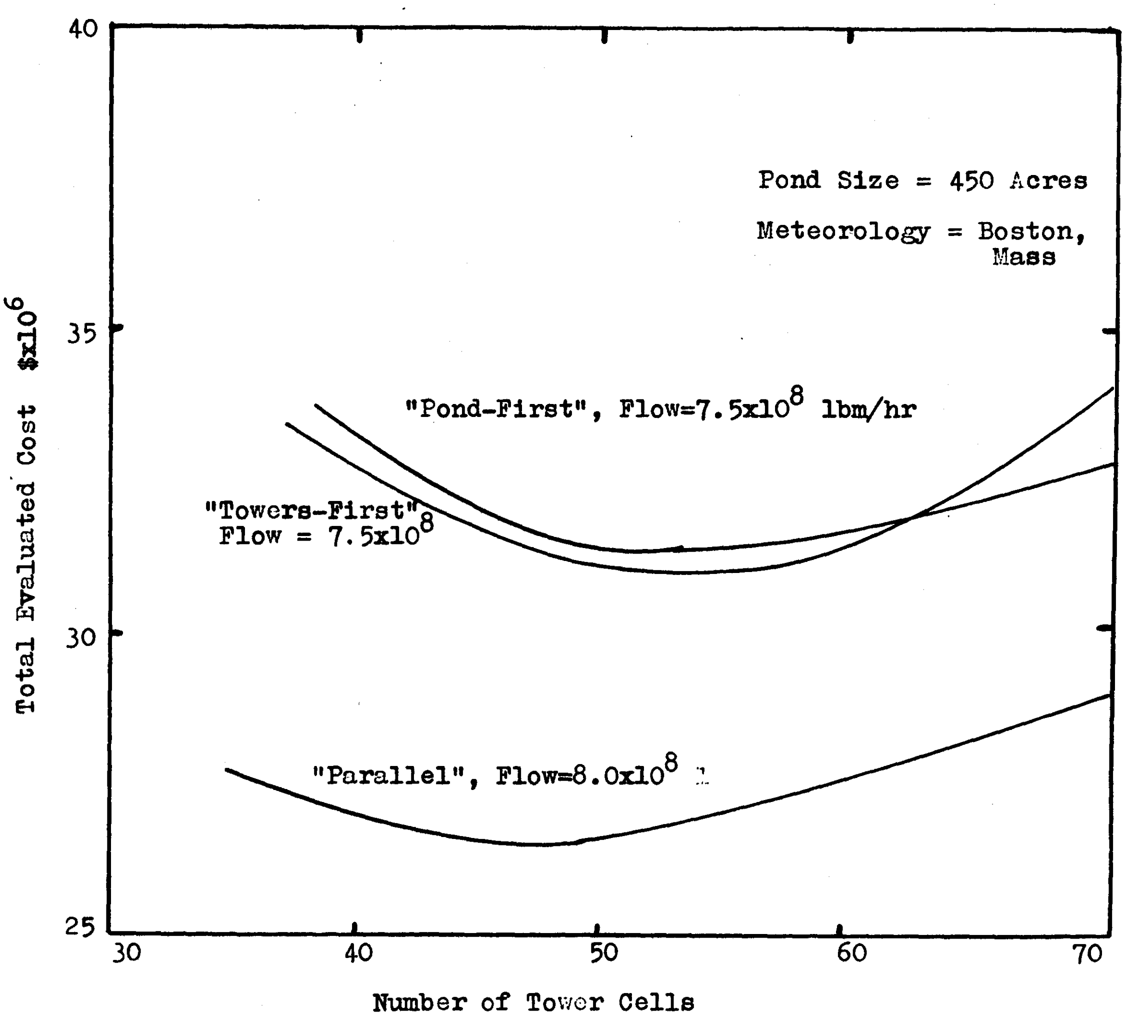


Fig. 3.7 Combined Cooling Pond - Tower System Optimization Results

Configuration = "parallel",

No. of Tower cells = 50,

Circulating water flow rate = 800,000,000 lb_m/hr,

and water loading = 5200 lb_m /ft^2-hr

result, to a good approximation, in the optimal combined waste heat system. The total evaluated cost for the system is $26,500,000.

The "parallel" system configuration is by far the best design choice. The difference between the least-cost "pond-first" system and the "parallel" system is about $4,500,000 or about 17% of the total system cost. The greater cost of both the series flow options is most readily attributed to the increased cost of operating the tower booster pumps and the somewhat reduced heat transfer performance of the combined system as shown in Fig. 3.8.

Since the above conclusion concerning the economic superiority of the "parallel" system configuration has resulted from an examination of the specific case of a pond-tower combination in which each of the devices rejects approximately 1/2 of the total heat load it is of interest to investigate the sensitivity of this conclusion to this assumption. Figure 3.9 summarizes the results of this investigation. It is clear that for combined systems in which the pond is the predominate

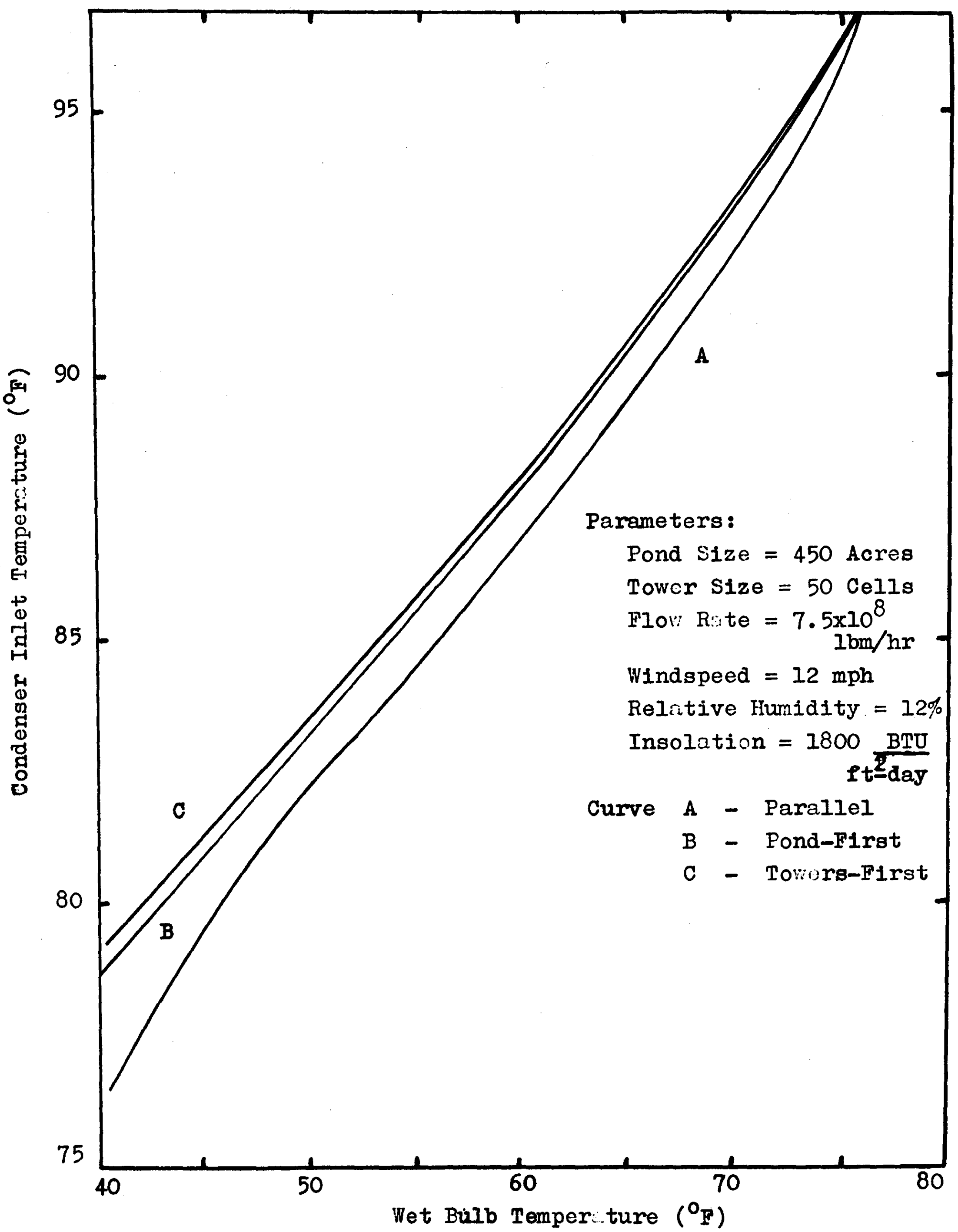


Fig. 3.8 Comparative Steady-State Heat Transfer Performance of Combined Cooling Pond-Tower System

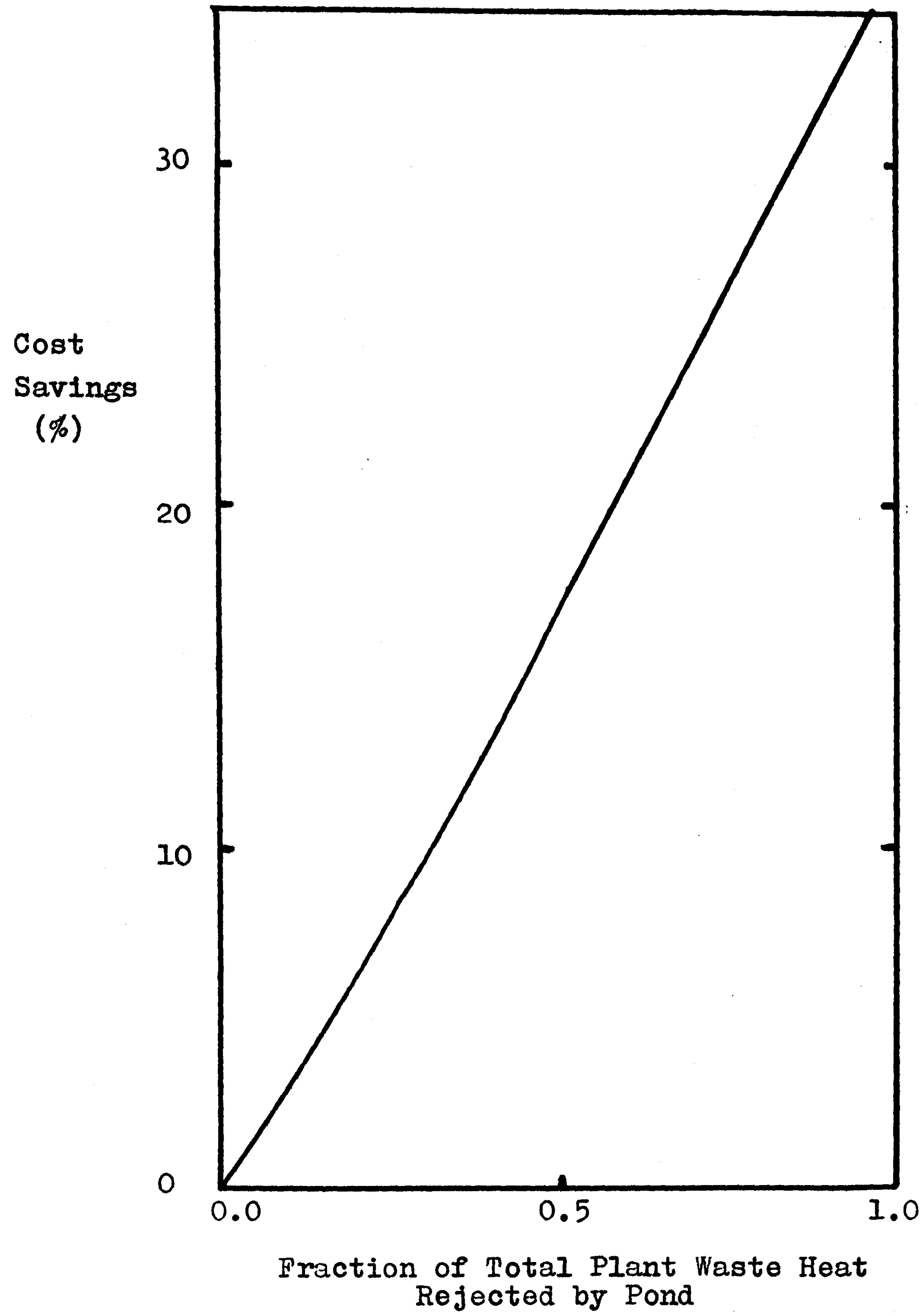


Fig. 3.9 Percentage Cost Savings of Parallel System Over Series System

heat rejection device the use of a series configuration is highly undesirable since extremely high tower water loadings result and hence high booster pumping power per unit heat rejected in the tower. For combinations in which the tower is the predominate heat rejection device the cost difference between parallel and series combinations diminishes since in either case the loading of the tower is approximately the same.

3.3.4 Conclusions

Although the economic evaluation of this study is based on the meteorology for a specific site, the result that the "parallel" combined system is the optimal configuration can be taken as a general conclusion. The generalization is possible since the steady-state heat transfer of series flow systems has been found typically to be slightly inferior to that of parallel flow systems. Certainly, for a series system to be economically advantageous the overall heat transfer performance of the series combinations would have to be substantially superior to that of the parallel system in order to justify the considerable expense of pumping the full circulating water flow through the cooling tower.

3.4 Cooling Tower-Pond Systems with Variable Operational States

3.4.1 Introduction

The loss of capability of tower cooled central power stations during high ambient temperatures is a significant problem since, in many cases, the periods of peak utility system electrical demand and high ambient temperatures are coincidental (H10)(T1) . This loss of capability results from the inability of the towers to recool sufficiently the condenser discharge water.

In order to improve this situation, it is proposed that a small cooling pond be used as a capacitive heat sink with the towers during the peak electrical demand periods of the day (G3). The specific type and function of the pond would depend on its particular application. Four different types of tower-capacitive pond systems have been evaluated. They are:

1) The evaporative cooling tower and supplemental cooling pond system,

2) The salt-water evaporative cooling tower and supplemental cooling and makeup storage pond system,

3) The evaporative cooling tower and thermal storage pond system, and

4) The dry cooling tower and thermal storage pond system.

3.4.2 The Evaporative Cooling Tower/Supplemental Cooling Pond System

3.4.2.1 The System

The operation of the evaporative cooling tower/supplemental cooling pond system is best explained by examining Fig.3.10. During the greater part of the day when the utility system demand is significantly below the daily peak electrical demand the cooling tower would provide the sole source of condenser cooling water (valves A open B and C closed). However, when the system electrical demand becomes high the tower would discharge to the pond and the condenser cooling water would be withdrawn from the pond (valves A closed, B and C open).

Since the pond would be utilized for only a relatively short period of the day (less than 12 hours) and since the great portion of the heat rejection would still take place through the towers, the pond should provide a relatively constant heat sink temperature with respect to variations in the ambient wet bulb temperature. Although the heat discharge to the pond is intermittent, the heat transfer from the pond would be continuous.

A variation of the above system would consist of having the condenser discharge directly to the pond during periods of high ambient temperature. The condenser cooling water during the coincidental periods of high ambient temperature and

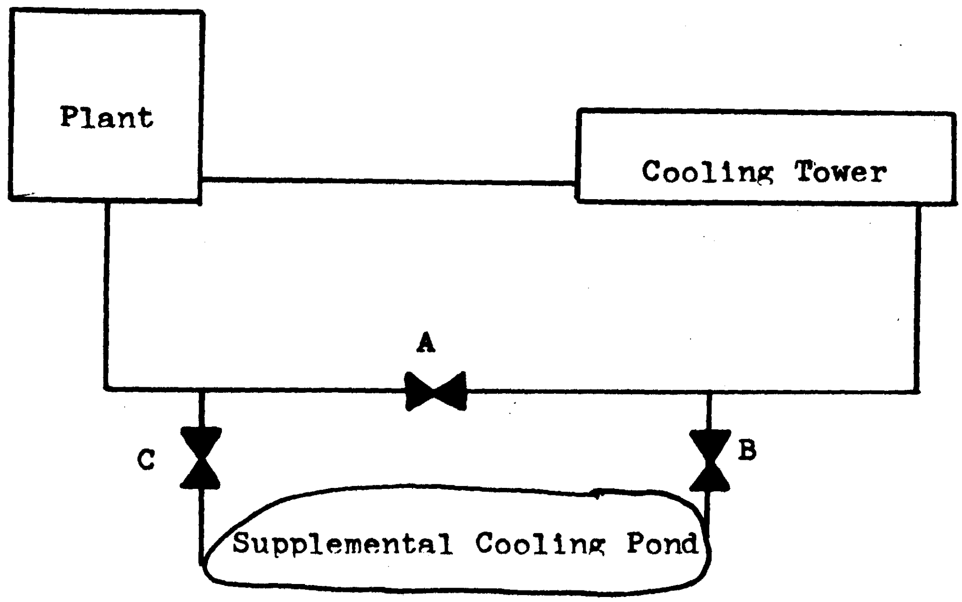


Fig. 3.10 Evaporative Cooling Tower/Supplemental Cooling Pond System

high system electrical demand would be gotten by discharging from the pond, via the cooling tower, to the condenser intake. The advantage of this system would be that, since the pond is typically hotter than in the previously discussed case, heat transfer from the pond would be greater. However, the heat rejection from the towers would be decreased during the combined system operation since the inlet temperature to the towers would be low. Also, there is the disadvantage of having the condenser temperature directly coupled to the ambient wet bulb temperature.

3.4.2.2 Component System Models

In order to make a preliminary determination of the potential benefits of the proposed system a computer model has been constructed which simulates the day to day operation of the proposed system. The system modeled is composed of a 3000 MWe nuclear power station with a conventional nuclear turbine, a cross-flow induced-mechanical-draft evaporative cooling tower, and a fully-mixed supplemental cooling pond. The fully-mixed pond model is chosen as the pond thermal-hydraulic model since it represents a lower bound on the performance of a well-designed cooling pond.

In each of the evaluations of the concept for the different geographical locations, the cooling tower model is representative of a conventionally optimally-sized cooling tower.

3.4.2.3 Description of Case Studies

The benefits of the proposed supplemental cooling pond have been evaluated for four different geographical locations-- Boston, Mass; Winslow, Arizona; Atlanta, Georgia; Minneapolis, Minn. The meteorolgy for each of these sites is for the year 1974 and was obtained from the National Climatic Records Center. A description of the cooling tower systems for each of these sites is given in Table 3.2.

The use of both a "hot" and "cold" pond was evaluated. A "hot" pond system is one in which during the period of pond operation the condenser discharges directly to the pond and the pond discharges through the towers to the condenser inlet. The "cold" pond system is one in which during the period of pond operation the tower discharges to the pond and the pond discharges directly to the condenser inlet. The benefits of the supplemental cooling system were evaluated in terms of average daily and summer capability savings for both the "hot" and "cold" systems, for all four sites, and for supplemental pond sizes of 20,50, and 80 acres (all 20 feet deep). The periods of operation of the supplemtntal pond were taken to be either 3 hours (3 PM to 6 PM) or 6 hours (noon to 6 PM).

The heat transfer performance of the towers was evaluated at the average wet bulb temperature for the supplemental

TABLE 3.2

Description of Cooling Tower Systems

Location	"Design" Wet Bulb	Number of Tower Cells	Circulating Water Flow Rate
Boston, Mass.	73^0F	44	1157 ft^3/sec
Winslow, Ariz.	63	39	1023
Atlanta, Geo.	75	46	1206
Minneapolis, Minn.	74	45	1179

1 Cell = Fill volume of 32' by 36' by 60',Station Power at "Design wet bulb = 1027 MWe, Maximum Station Power = 1052 MWe

3.4.2.4 Presentation and Discussion of Results

The summer average power savings during the periods of operation of the combined pond-tower system as a function of the supplemental pond size is shown in Figure 3.11 for the four different sites. In examining the figure, it is interesting to note that a pond size as small as 20 acres results in average savings of almost 1% of the total plant capability. Also, note that the effect of the different meteorologies at the different sites on the resultant pond benefit is not large.

Because of the summer peaking problem experienced by many utilities the information in Figure 3.12 may be a more significant indicator of the supplemental pond benefit. With ponds which are small in comparison to typical 1000 acre nuclear plant sites savings of nearly 2% of the plant power output on the hottest summer days are indicated for some sites.

Tables 3.3 and 3.4 present the results of the investigation of the comparative benefits for different supplemental pond utilization schemes. Recalling the definition of the "hot" and "cold" supplemental ponds presented earlier, these results show that , in general, some additional summer average power savings may be gained by utilizing a "hot" pond instead of a "cold" pond. With one exception, the

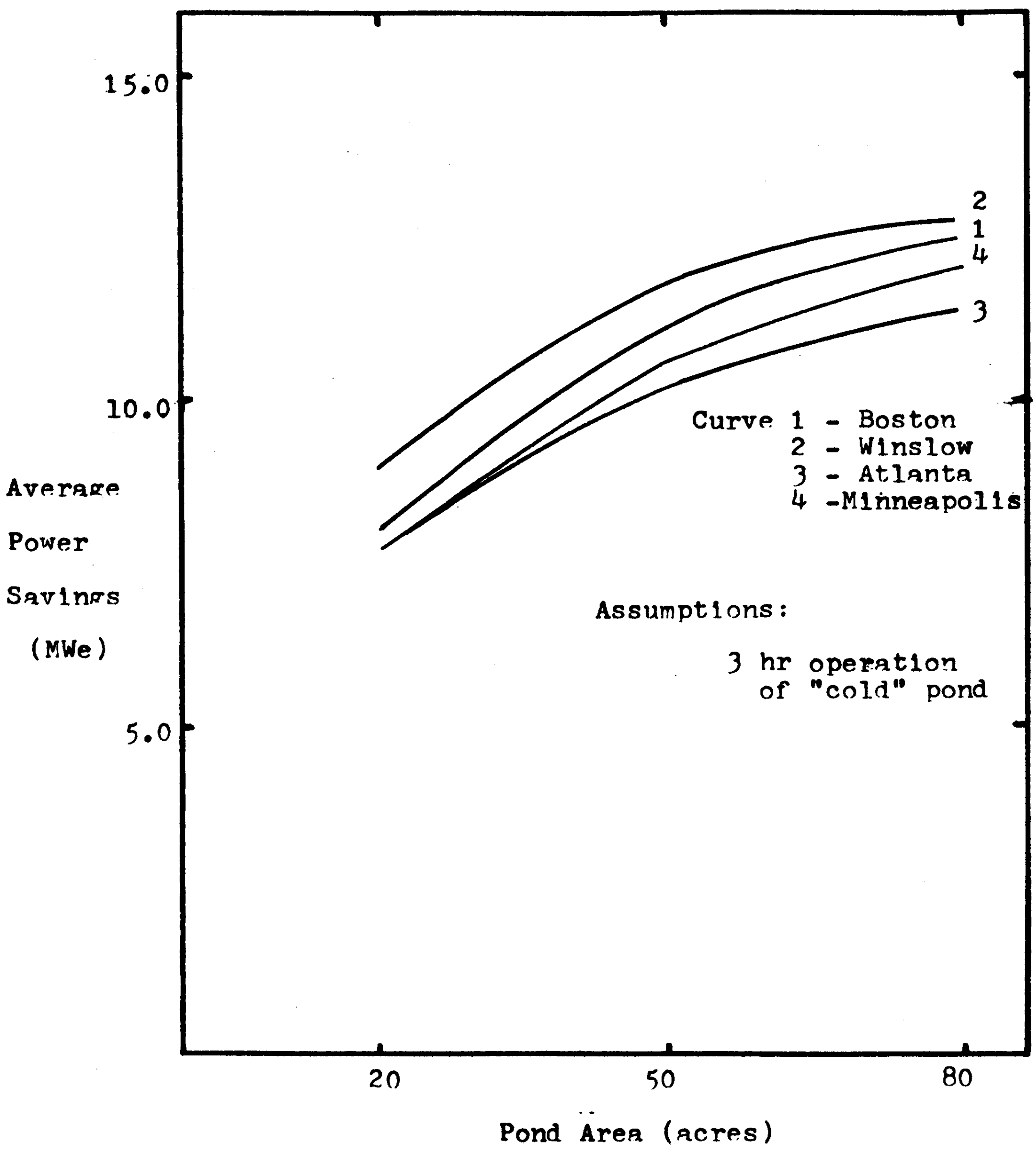


Fig. 3.11 Effect of Supplemental Cooling Pond Size On Summer-Average Power Savings for Different Sites

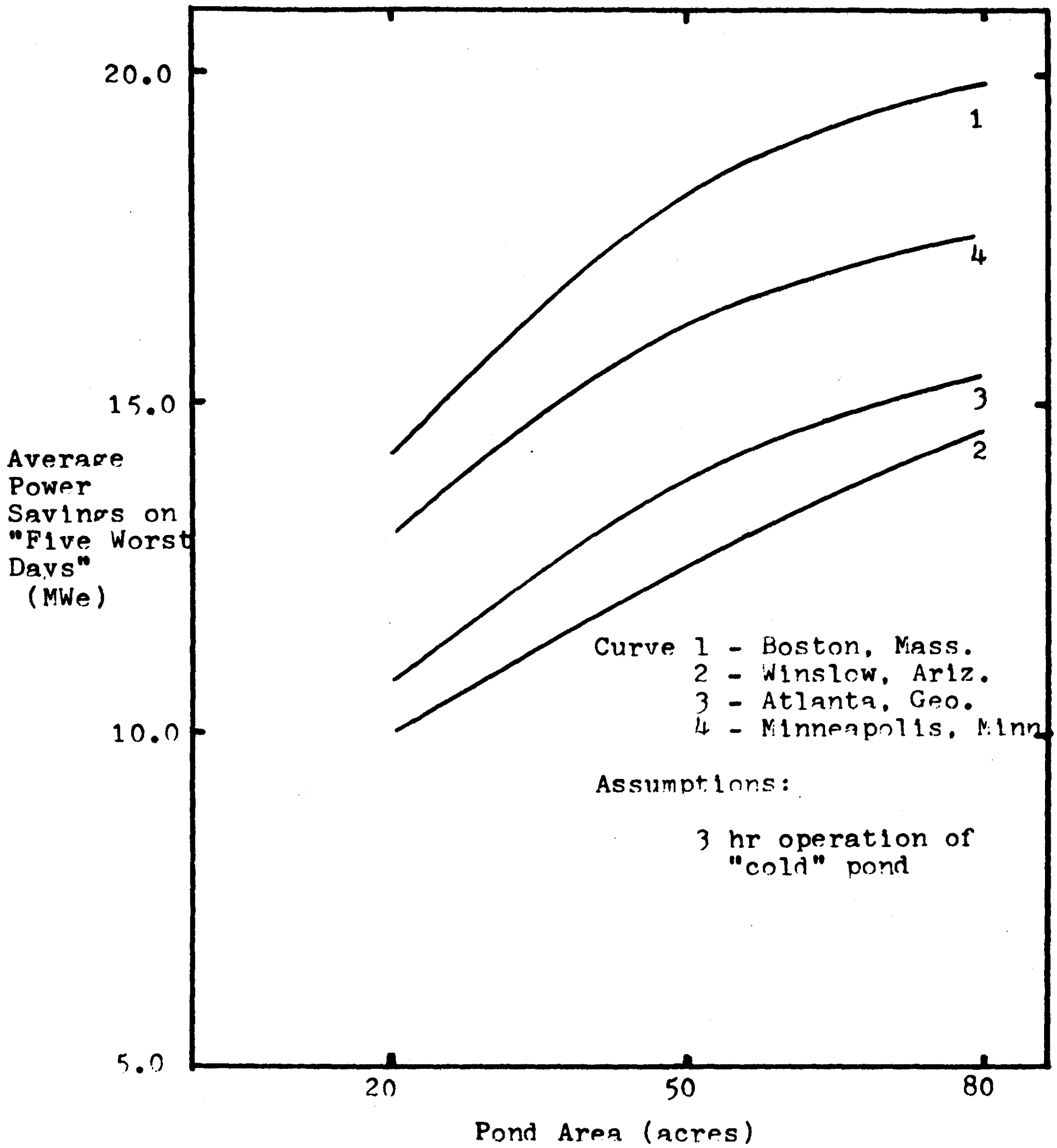


Fig. 3.12 Effect of Supplemental Cooling Pond Size on "Five Worst Days" Average Power Savings for Different Sites

TABLE 3.3

Effect of Alternative Supplemental Cooling Pond Utilization Schemes on Average Summer Power Savings

	20 Acres		80 Acres	
	Cold	Hot	Cold	Hot
Boston				
3 hour	8.0 MWe	8.6	12.5	13.0
6 hour	5.6	5.9	10.3	10.9
Winslow				
3 hour	9.0	11.0	12.8	15.6
6 hour	6.1	7.9	11.1	13.0
Atlanta				
3 hour	7.7	8.8	11.4	12.5
6 hour	5.0	6.0	9.6	10.6
Minneapolis				
3 hour	7.7	8.3	12.0	12.4
6 hour	5.3	5.7	9.9	10.5

TABLE 3.4

Effect of Alternative Supplemental Cooling Pond on "Worst-5-Day" Average Power Savings

	20 Acres		80 Acres	
	Cold	Hot	Cold	Hot
Boston				
3 hour	14.2 MWe	14.0	19.8	18.4
6 hour	11.0	9.8	17.3	16.2
Winslow				
3 hour	10.0	12.4	14.6	16.9
6 hour	6.1	8.0	12.9	14.6
Atlanta				
3 hour	10.8	10.7	15.4	14.9
6 hour	6.8	6.8	12.4	12.8
Minneapolis				
3 hour	13.0	12.5	17.5	16.9
6 hour	8.9	8.7	16.0	14.8

"worst-5-days" capability savings appears to be comparable for the two pond utilization schemes.

Noting the comparative benefits of the 3 hour pond versus the 6 hour pond it is generally indicated that the longer utilization period results in a lower capability savings (MWe), but a larger total electrical energy production savings (MWhrs).

It is difficult at this point to determine a general cost-benefit of implementing this supplemental cooling system. Indeed, an accurate cost-benefit analysis can only be achieved through a total cooling system design optimization. The cost of constructing the supplemental pond is highly site-dependent and thus any determination of the cost/benefit ratio would be site-dependent. Nonetheless, some comments concerning the costs of a supplemental cooling pond are worthwile.

First, since the size of the proposed pond is small in comparison to typical nuclear power station sites the cost of the land for the pond may not be a relevant consideration. Second, no extra pumps or pumping power would be required if the pond is at the elevation of the tower discharge. Third, credit may be taken for symbiotic use of the supplemental cooling pond with regard to blowdown control, emergency cooling, and makeup storage.

Although a complete cost-benefit analysis is not being attempted at this point, it is of interest to examine the cost of the obvious alternative to the supplemental cooling pond. Essentially, we are interested in determining the capital cost of the additional mechanical draft tower cells which would be needed to obtain the same performance achieved through the use of a supplemental pond. This can be simply accomplished by determining the number of extra cells needed for a towers-only cooled plant to have an annual minimum power equal to that of a typical combined cooling system cooled plant. For the 50 acre-3 hour supplemental pond in Boston, Mass. the minimim plant capacity is 1042 MWe. For the towers-only cooling system consisting of 44 mechanical draft tower cells the minimum power is 1025 MWe. To raise this minimal power level to 1042 MWe the addition of approximately 13 tower cells would be required. Thus, the increase in the capital cost of the towers would be approximately 30%.

3.4.3 The Salt-Water Evaporative Cooling Tower/ Supplemental Cooling and Makeup Storage Pond System

3.4.3.1 Introduction

The use of cooling towers has been considered for several coastal-sited central power stations of the 1000 Mwe class.

The application of conventionally optimal evaporative cooling towers at such sites would result in a loss of generating capability of about 30MWe during periods of high ambient temperature. During such periods the tower discharge water (condenser intake) would typically be about $90^{0}F$. This is far in excess of the local sea water temperature which for northern coastal sites may average about $60^{0}F$ during the summer months. Thus, it would be highly desirable to make use of this cooling potential during the peak electrical demand periods. Obviously, one cannot simply advocate a conventional "once-through" type of supplemental cooling since the required use of the towers is predicated on the environmental undesirability of "once-through" cooling. However, if one considers the amount of makeup water which will be required to operate the saltwater towers (R3) and the amount of sea water needed to dilute the tower blowdown it is realized that significant cooling could be achieved through the proper application of this resource.

Essentially, the goal of this system is to utilize the water resource required for makeup and blowdown dilution in such a fashion that during peak electrical demand periods plant performance typical of that of "once-through" cooling could be attained while, on the other hand, the hydrothermal impact be on the order of that associated with cooling towers.

3.4.3.2 Description of Proposed System

To achieve the desired supplemental cooling it is proposed that the makeup water and blowdown dilution water be withdrawn steadily from the ocean and stored in a small pond. The stored water would then be used for condenser cooling water during peak electrical demand periods. The details of this concept are shown in Figure 3.13 and can be best explained by illustrating its application to a typical ocean sited 1000 MWe station.

During periods of non-peak electrical demand the cooling towers would operate in the conventional manner with valves B and C closed. However, makeup water M_m would come from the pond as would the dilution flow M_d for the blowdown flow M_b. Note that for a typical 1000 MWe station saltwater cooling tower operating at two cycles of concentration (about 70,000 ppm)

$$M_e = \text{evaporation rate} \approx 20 \text{ ft}^3/\text{sec},$$

$$M_b = 20 \text{ ft}^3/\text{sec}, \text{ and}$$

$$M_m = 40 \text{ ft}^3/\text{sec}.$$

Now if we assume a blowdown dilution flow of 2 to 1 then

$$M_d = 40 \text{ ft}^3/\text{sec}, \text{ and}$$

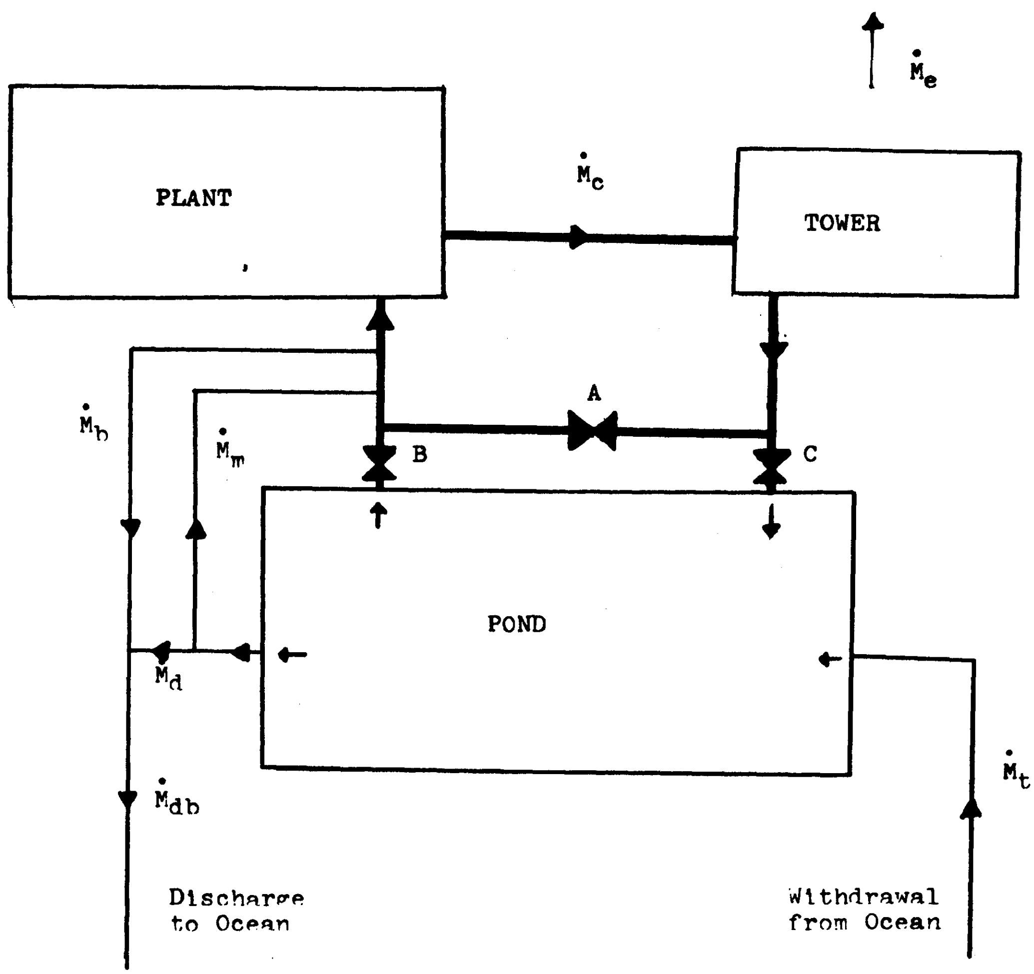


Fig. 3.13 The Saltwater Evaporative Cooling Tower/Supplemental Cooling and Makeup Storage Pond System

$$M_t = 80 \text{ ft}^3/\text{sec} .$$

At this point it is important to note that M_t the total rate at which water is withdrawn from the ocean, is the same as if the dilution flow was obtained directly from the ocean.

Now assume for the moment that the temperature of the pond is higher than the ambient ocean temperature as a result of having received the tower discharge during some previous time period. Since the temperature of M_t is less than the pond temperature the pond will be gradually cooling towards the ocean temperature. During this operation stage and all others the pond volume is constant since

$$M_t = M_m + M_d .$$

The temperature of M_m the makeup water flow, will be slightly higher than if it were obtained directly from the ocean. However, the significance of this temperature increase is small since the makeup flow is only about 4% of the total circulating water flow rate. The net effect would be less than a one degree F change in the condensing temperature.

The above mode of operation of the combined plant-tower supplemental pond will be termed the "cooldown" mode of operation and would continuous except for periods of peak electrical demand. If the pond were required to contain 3 hours worth of condenser cooling water the necessary pond volume would be

$$V_p = \dot{M}_c \cdot 3hr$$

$$V_p = (1100\ ft^3/sec)\ (3hr)\ (3600)$$

$$= 272\ \text{acre-ft}$$

Thus for $\dot{M}_t = 80\ ft^3/sec$ the ratio for the daily flushing volume to the pond volume about 0.5. For a smaller pond which would supply only ½ the condenser cooling water (the other ½ coming directly from the towers) the same ratio would be approximately unity.

The second mode of operation will be termed the "heatup" mode. This "heatup" mode would continue for the period of utility system peak electrical demand, say from 3 to 6PM. The operational scheme would consist of closing valve A and opening valves B and C. The tower discharge would be directly routed to the pond and the condenser cooling water would be supplied by the pond. Variations of this operation scheme would include the partial opening and closing of the valves such that the condenser inlet water was supplied in part by the pond and in part by the tower discharge.

During the "heatup" mode of operation the temperature of the water entering the condenser would be considerably lower than if the tower discharged directly to the condenser. Once the supplemental cooling capacity of the pond had been exausted or the peak electrical demand period had passed the system would then revert to the "cooldown" mode of operation.

3.4.3.3 Description of Evaluation Model

A preliminary quantitative evaluation of the combined system concept has been performed in order to reveal the potential benefits of implementing such a system. The basis of the evaluation is the simulation of the operation of a representative system for the months of June, July and August. The representative system is a nominal 1000MWe nuclear power station cooled with a conventionally optimal 44 cell mechanical draft tower. The characteristics of the site (meteorology and ocean temperature) are typical of those of coastal Massachusetts.

The mathematical models of the plant and the wet tower are identical to those described in Sec. 3.3.2. However, substantial difficulty is encountered when attempting to formulate a mathematical model of a storage pond. Nevertheless, the following assumption allows for an adequate appraisal of the concept:

> complete mixing of $\dot{M}_t$ at all times and plug flow of the tower discharge into the pond during the "heatup" mode of operation.

3.4.3.4 Results of Evaluation

The benefits of several different makeup storage schemes have been calculated. The system design variables which which essentially determine the effect of the pond are the pond size, the dilution flow rate $\dot{M}_d$, and the pond utilization rate. The pond utilization rate is defined here as the fraction of the condenser cooling water supplied by the pond during the "heatup" mode of operation.

A complete description of the system modeled is given in Table 3.5 and the results of the simulation studies for three different cases are shown in Table 3.6. An explanation of the various column headings in Table 3.6 follows;

<u>pond size</u> - the size of the pond in acres; all ponds are twenty feet deep,

total withdrawal flow- the rate at which water is withdrawn from the ocean and mixed with the pond inventory,

dilution flow- rate at which water is withdrawn from the pond to dilute the tower blowdown,

average power difference- the average of the difference between the plant power output of a towers only system and the combined system during the "heatup" mode of operation for the entire summer,

temperature excess

with pond - average difference between the diluted blowdown temperature and sea temperature for the entire summer- dilution flow obtained from pond,

without pond- average difference between the diluted blowdown temperature and the sea temperature for the entire summer-dilution flow obtained directly from ocean,

power difference- ten worst days- same as average power difference except evaluated for 10 hottest (wet bulb) days of the summer only,

pond utilization scheme - "1/2 flow" indicates that the pond supplies only 1/2 of the condenser cooling water during the heatup mode, the rest comes from the tower: "fullflow" indicates that the pond supplies the total condenser cooling water requirements during the "heatup" mode,

minimum power

with pond - minimum power of plant during the peak demand period with combined cooling system for the entire summer (peak demand period = "heatup" period and,

without pond - minimum power of plant during the peak demand period for towers-only cooling system.

In addition to the thermal performance calculations

TABLE 3.5

Description of Plant-Cooling Tower-Site

for Saltwater Evaporative Cooling Tower/Supplemental

Cooling and Makeup Storage Pond System Evaluation

1.0 Plant

-3000 MWt
-conventional nuclear turbine
-5°F terminal temperature difference in condenser
-circulating water flow rate = 1200 ft^3/sec
-temperature rise in condenser = 25° F
-plant power at design wet bulb of 73° F = 1027 MWe
-condenser inlet temperature at design condition = 91°F

2.0 Mechanical Draft Cooling Tower

-type-crossflow, induced draft
-number of cells = 44
-fill dimensions = 32 by 36 by 60
-water loading = 5130 lbm/hr-ft^2
-air loading = 1692 lbm/hr-ft^2
-cycles of concentration = 2

3.0 Meteorology

-Boston, Massachusetts-1974

4.0 Ocean Temperature; salinity = 30,000 ppm

-June = 55°F
-July = 60
-August = 65

5.0 Operation Schedule

-all combined systems operate in "heatup" mode from 3 to 6 PM

TABLE 3.6

PERFORMANCE OF THE SALTWATER EVAPORATIVE TOWER/SUPPLEMENTAL COOLING AND MAKEUP STORAGE POND SYSTEM

		Case #1	Case #2	Case #3
1.0 Pond Size	Acres	8	8	16
2.0 Total withdrawal Flow	Ft^3/sec	80	120	160
3.0 Dilution Flow-Ft^3/sec		40	80	120
4.0 Average Power Difference	MWe	12.6	15.0	17.4
5.0 Temperature Excess				
With Pond	^oF	18.2	14.0	13.9
W/O Pond	^oF	8.8	5.3	3.7
6.0 Power Difference Ten Worst Days	MWe	15.6	17.9	22.6
7.0 Pond Utilization Scheme		1/2 flow	Full Flow	Full Flow
8.0 Minimum Power-MWe				
With Pond		1043	1045	1051
W/O Pond		1027	1027	1027

For all the above, raw blowdown = 20 ft^3/sec
Diluted blowdown flow = 20ft3/sec = Dilution Flow

summarized in Table 3.6, calculations have been performed with regard to the effect of the dual mode of tower operation on the buildup of dissolved salts in the system. These calculations indicate that, although the tower would be purged of its high salt concentration during the "heatup" mode of operation, blowdown of the tower would have to be initiated within a few hours after the end of the "heatup" mode in order that the maximum salt concentration not be exceeded. The net salt blowdown from the system is identical for both the tower-only systems and the combined systems since the net salt blowdown depends only on the rate of evaporation which is essentially constant for all the systems considered.

3.4.3.5 Conclusions

As in the case of the evaporative tower-supplemental cooling pond system the determination of the benefit-cost of the system is difficult due to the site-specificity of the pond construction cost. However, in terms of conventional replacement capability pricing at $150.00/Kw the capibility replacement savings would be on the order of several million dollars using the results of Table 3.6. The costs of the pond construction (based on general excavation and piping costs) would be lower at many sites.

3.4.4 Cooling Tower/Thermal Storage Pond System

3.4.4.1 Introduction of the Concept

A fundamental disadvantage of wet and dry cooling tower systems is the small thermal inertia of these systems and the sensitivity of their performance to changes in the ambient meteorology. As a means of avoiding this loss of generating capability during peak electrical demand periods it is proposed that wet tower or dry tower systems be combined with a small thermal storage pond. In this combined system the thermal storage pond would function as a heat capacitive component to take advantage of the diurnal variation of the ambient meteorology. For many inland regions of the United States the daily range of the dry bulb temperature regularly exceeds 30°F, and a 15° F average variation in the wet bulb temperature is common.

To evaluate the benefits of this combined system concept a computer model which simulates the operation of a nominal 1000 MWe nuclear power station has been constructed. The utilization of an idealized plug-flow thermal storage pond capable of holding 3 hours of condenser cooling water (approximately 300 acre-feet) with both wet and dry cooling towers has been investigated.

Typical results of these calculations are shown in Fig. 3.14. The cases shown are based on daily variations of the wet bulb temperature (wet tower) and dry bulb temperature (dry tower) of 65 to 80°F and 70 to 100°F, respectively.

In the proposed system the thermal storage pond would hold the relatively warm tower discharge and would supply cool water to the condenser intake during periods of coincidental peak electrical demand and high ambient temperature. During the night when the electrical demand is low and the ambient temperature has declined the relatively warm water in the pond would be used for condenser intake water and the now relatively cool tower discharge water would be used to "recharge" the pond. Between periods of pond utilization the cooling tower would discharge directly to the condenser intake.

As Fig. 3.14 indicates, the most likely candidate for the addition of a thermal storage pond is the plant cooled with dry cooling towers. The typical benefits which are gained with the application of a thermal storage pond in conjunction with a wet cooling tower are only minimal. The greater impact of the thermal storage pond on the performance of the dry tower cooled plant is due to 1) the significantly greater typical daily range of the

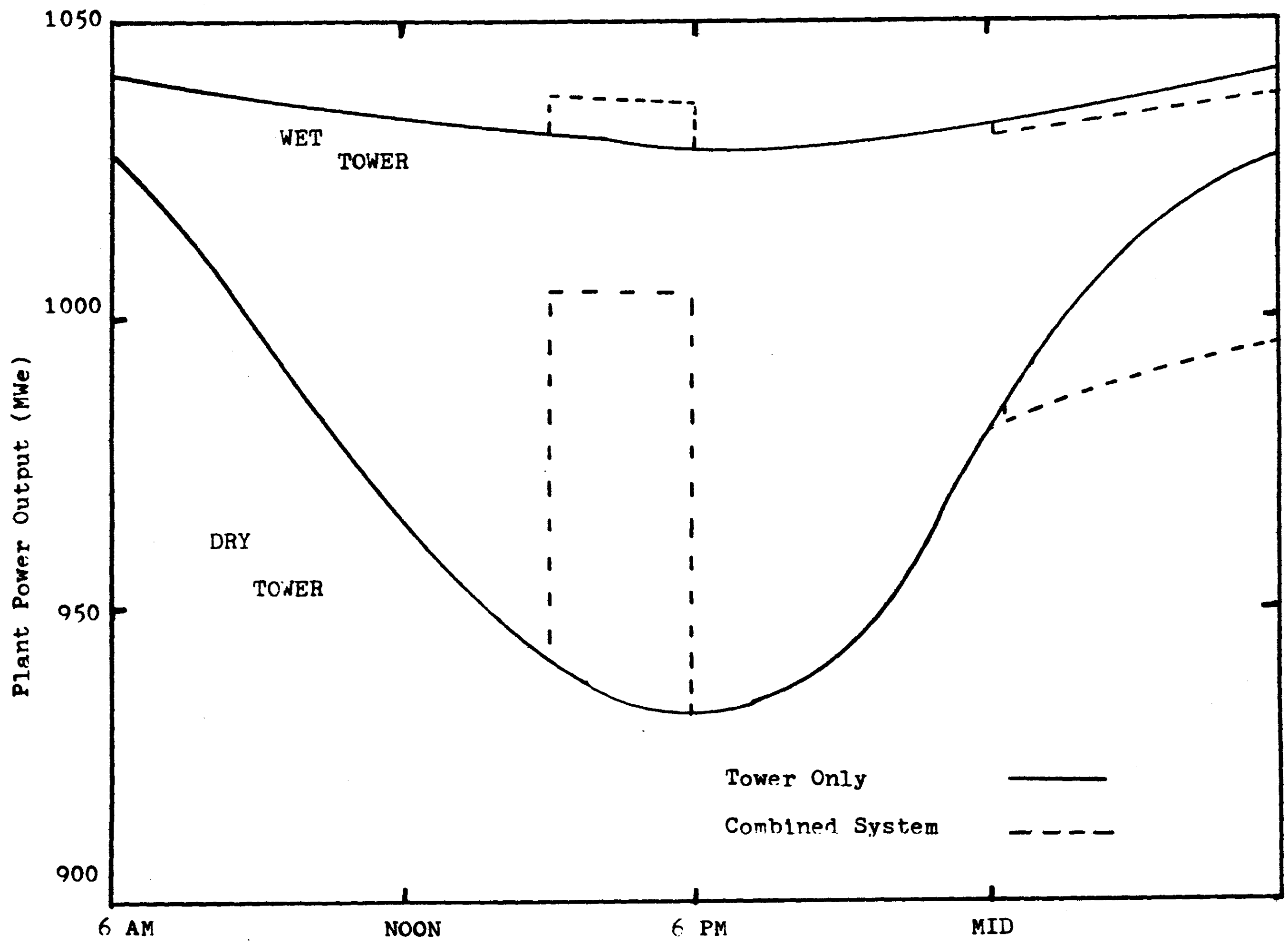


Fig. 3.14 Plant Performance for Combined Thermal Storage Pond/ Cooling Tower Systems

dry bulb temperature and 2) the greater rate of change of the plant heat rate per degree change in the condenser inlet temperature for the condenser inlet temperature ranges typical of dry tower cooled power plants.

Thus, in view of the greater potential benefit of the thermal storage pond when combined with a dry tower system, it was decided that a detailed examination of this particular combination should constitute the major portion of this cooling tower-thermal storage pond system evaluation.

3.4.4.2 Significance of High Ambient Temperatures On the Performance of a Dry-Tower-Cooled Nuclear Power Station

The relative heat rates as a function of condensing temperature of several turbine designs are shown in Fig.3.15. Several important observations concerning these curves are:

1) the modified basic nuclear turbine heat rate curve is nonlinear and varies substantially in magnitude over the range of condensing temperatures of 90 to 160 ^{o}F. (corresponds approximately to an ambient temperature range of 30 to 100^{o}F)

2) the above situation is somewhat improved with the intermediate annulus and high back pressure turbine designs in that a

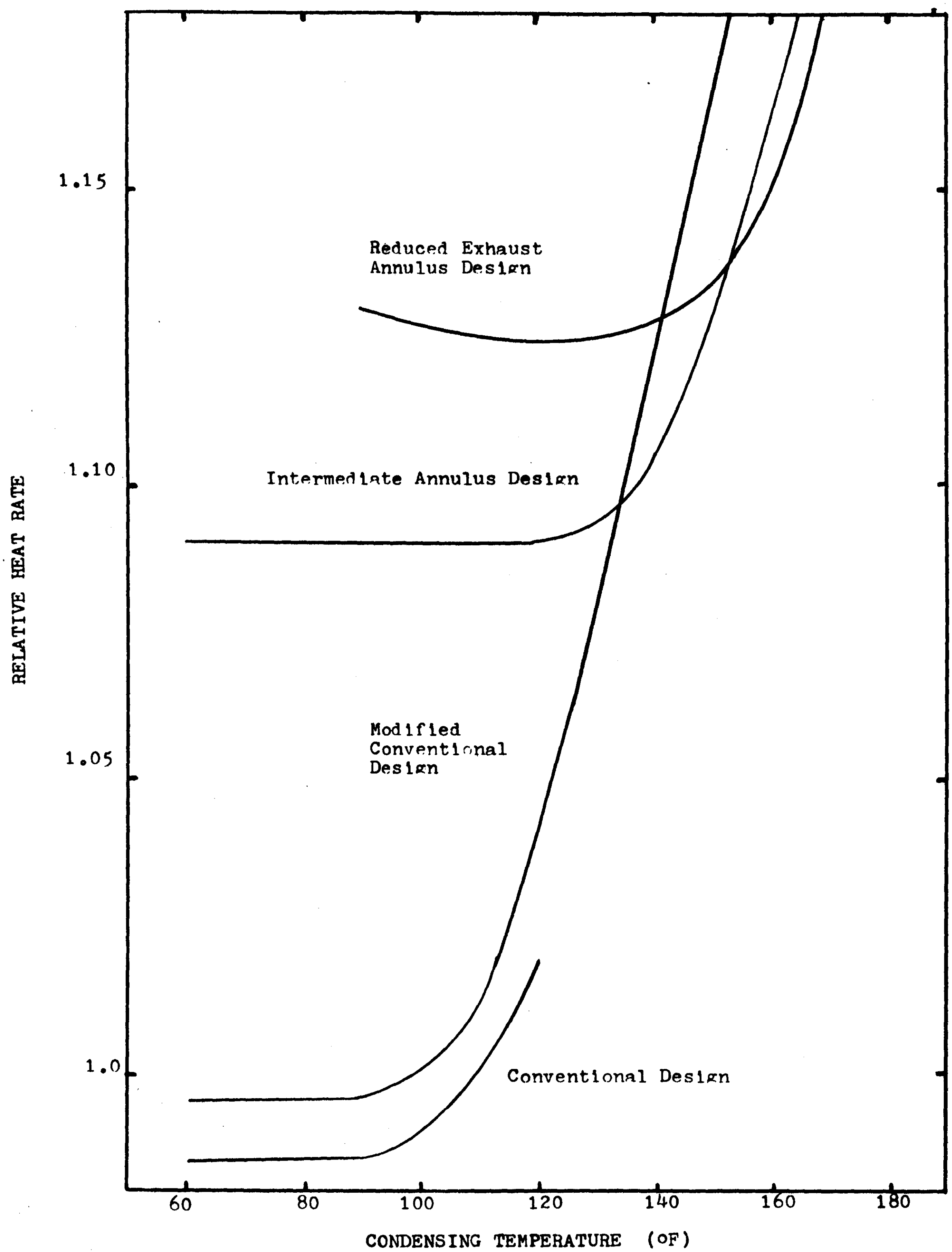


Fig. 3.15 Relative Heat Rate Curves for Nuclear Turbines (R7)

less severe loss of performance at high condensing temperatures, but loss of performance in comparison to the modified nuclear turbine design occurs at all condensing temperatures less than $144^{o}F$ and $152^{o}F$ respectively.

3) the performance of the modified basic nuclear turbine is near that of the conventional basic nuclear turbine at low condensing temperatures.

Although substantial loss of generating capability would be encountered during the great portion of the year for many areas of the United States if an intermediate annulus nuclear turbine or a high back pressure nuclear turbine were to be employed at a dry cooled power station, the incentive for considering such application is strong. The impetus is derived from the fact that if a modified nuclear turbine in conjunction with a dry cooling tower system were required to generate full load at temperatures exceeding approximately $90^{o}F$ the dry tower would be unduly larger and more expensive (or the loss of capability capital cost penalty would be considerably greater) than that associated with either of the other two turbine designs.

However, it is important to note that for both the intermediate annulus design and the high back pressure design the <u>slope</u> of the heat rate curve is nearly constant at high condensing temperatures and is approximately equal to the slope of the heat rate curve for the modified conventional turbine at condensing temperatures 10 to 20^{o}F less. Thus if it were possible to reduce the design condition (high ambient temperature) condenser inlet temperature by a similar amount, the design condition performance of the modified conventional turbine would be approximately equal to the design condition performance of the other two applicable turbines. The incentive for affecting such a reduction of the condenser inlet temperature during the design condition is the substantial performance margin of the modified conventional turbine over the other two at low and average condensing temperatures.

The above argument for reducing the peak load time condenser inlet temperature can be extended to the case where the turbine design has been established. Now, the incentive for reducing the peak time condenser inlet temperature is simply the reduction in the loss of capability penalty. With the use of a thermal storage pond it may be possible to shift the maximum expected loss of capability to the late evening hours. Certainly, it seems reasonable that the same cost penalty used to penalize the loss of production at peak demand times should not be used to penalize the loss during off-peak night-time hours. (a more detailed discussion of the use of dry cooling with nuclear steam-electric plants is presented in Sec. 5.1.4).

3.4.4.3 Details of Operational Cycle

There are a variety of possible ways in which a small pond might be used in conjunction with a dry cooling tower to produce a matching of peak load and peak cooling ability, but some simple arguments lead us to consider first the system mentioned previously. A more detailed discussion of the system operation is now given.

In the late afternoon when electrical demand is highest and the ability to reject heat at desirable temperatures is lowest because of the high ambient temperature, cooling water for the condenser would be drawn from the pond. After passing through the condenser the water would be passed through the towers and returned to the opposite side of the pond. This operation would continue until all the relatively cool water had been withdrawn from the pond. At the end of this period the pond would contain water which is hotter than when the pond cycling sequence began. Note, however, that a considerable amount of waste heat would still be rejected by the towers. In fact, the pond temperature may still be considerably lower than the temperature of the tower discharge for steady-state operation at these high ambient temperatures without a thermal storage pond. Nevertheless, the tower would not be operating very efficiently since the ITD (tower inlet temperature-ambient air temperature) would be low in comparison to the required average ITD.

When the peak load period has passed the discharge from the tower would be routed directly to the condenser bypassing the pond. This operation would continue until the ambient temperature had declined sufficiently. Some cooling of the pond may take place during this time period.

Now once the ambient temperature has declined to near its minimal value and the electrical demand was low (say 12 midnight to 6 AM) the pond would be recooled in a manner analogous to the previously discussed heatup operation. With the plant operating at full thermal capacity, the now relatively warm water in the pond would be circulated through the condenser, through the tower and then back to the pond. During this operational mode the towers would essentially be rejecting the presently required waste heat energy in addition to that energy which the dry tower was not able to reject earlier during the heatup mode of operation. This large amount of heat rejection is possible since during the cooldown mode of operation the tower ITD would be increased over its normal value.

Once the entire pond inventory had been recooled the pond again would be bypassed until its now relatively cool water was needed during the peak electrical demand time later that day.

In describing the above opeational sequence of the thermal storage pond-dry tower system it was assumed that the discharge into the pond was not mixed with water already in the pond. This is the so-called "plug-flow" assumption. From the stand-point of the performance of the combined system this is a favorable, but realistic assumption. (See Chapter 4)

3.4.4.4 Preliminary Quantitative Evaluation of the Thermal Storage Pond Concept

In order to evaluate the feasibility of the combined thermal storage pond-dry cooling tower concept a computer model has been constructed which simulates the operation of a 3000 MWt nuclear power station coupled with a dry tower-thermal storage pond heat rejection system. The dry cooling tower is sized to produce a net electrical output of 928 MWe at a 93°F ambient air temperature with a modified conventional nuclear turbine. The thermal storage pond has a volume of 305 acre-feet with a surface area of 14.3 acres and is capable of holding 3 hours worth of condenser cooling water supply. This system will be refered to as the standard system. Important assumptions incorporated into this model are the following:

1) Condenser type---surface;

   Condenser terminal temperature difference---5°F;
2) Circulating water flow rate----277,499,900 lbs/hour;
3) Dry tower thermal performance is described by the equation

$$Q = C * ITD$$

where Q = heat rejected,

ITD = temperature difference between tower inlet water and ambient air, and

C = 137445000.0 BTU/hr-F;

4) Plug flow during heatup and cooldown of pond;
5) Pond becomes completely mixed (i.e. homogeneous in temperature) during period of pond bypass;

6) The pond is open to the atmosphere and subject to heat transfer to the environment as described by Ryan;(R10)

7) Water loss from the pond occurrs only as a result of evaporation;

8) The water inventory of the dry tower is small in comparison to the pond inventory (based on information presented by Rossie).(R7)

For the purpose of evaluating the thermal storage pond concept a standard set of meteorological and operational conditions have been assumed for a basis of comparison. The standard conditions (as they will be called) are 1) a 3 hour pond heatup time from 3PM to 6PM, 2) a 6 hour cooldown period from 12 midnight to 6 AM, and 3) a daily variation in the dry bulb temperature of 70°F to 100°F with a constant relative humidity of 30%. For the standard conditions listed above and for all other conditions examined in this section the "steady-state" behavior of the plant-thermal storage pond-dry tower system is presented. The term "steady-state" in this case meaning that the pond temperature assumed to exist at the beginning of the day is identical to that calculated for the end of the cycling sequence (i.e. beginning of the next day). It should be emphasized that the pond cycling sequence mentioned above is only a qualitative attempt to define an optimal application of the TSP concept. Consideration of alternative operational sequences and TSP utilization schemes will be mandatory in a more detailed and complete evaluation of the TSP concept.

3.4.4.4a Significance of Pond Size

The computer model has first used to evaluate the benefit derived from ponds of various sizes. The meteorology and dry tower are the standard given earlier. The results are summarized in Table 3.7 and by Fig. 3.16 which shows the plant output as a function of the time of day. The curves in Fig. 3.16 represent the following situations;

Curve A--no thermal storage pond,

Curve B--3 hour pond,

Curve C--6 hour pond, and

Curve D--9 hour pond,

In each case the cooldown period is equal to the heatup period in length. Table 3.7 indicates, as one would expect that the benefit of an incremental increase in the pond size diminishes as the pond becomes larger. Nevertheless, the magnitude of the benefit per incremental increase in pond size is large over the range examined.

TABLE 3.7

Comparison of the Performance of Different Size

Thermal Storage Ponds

Pond Size- Hours of Capacity	Daily Total Extra MWhrs	Peak Load Time Extra MWhrs	Average Peak Extra Capacity MWe
3	153	280	93.3
6	308	513	85.5
9	506	656	72.8

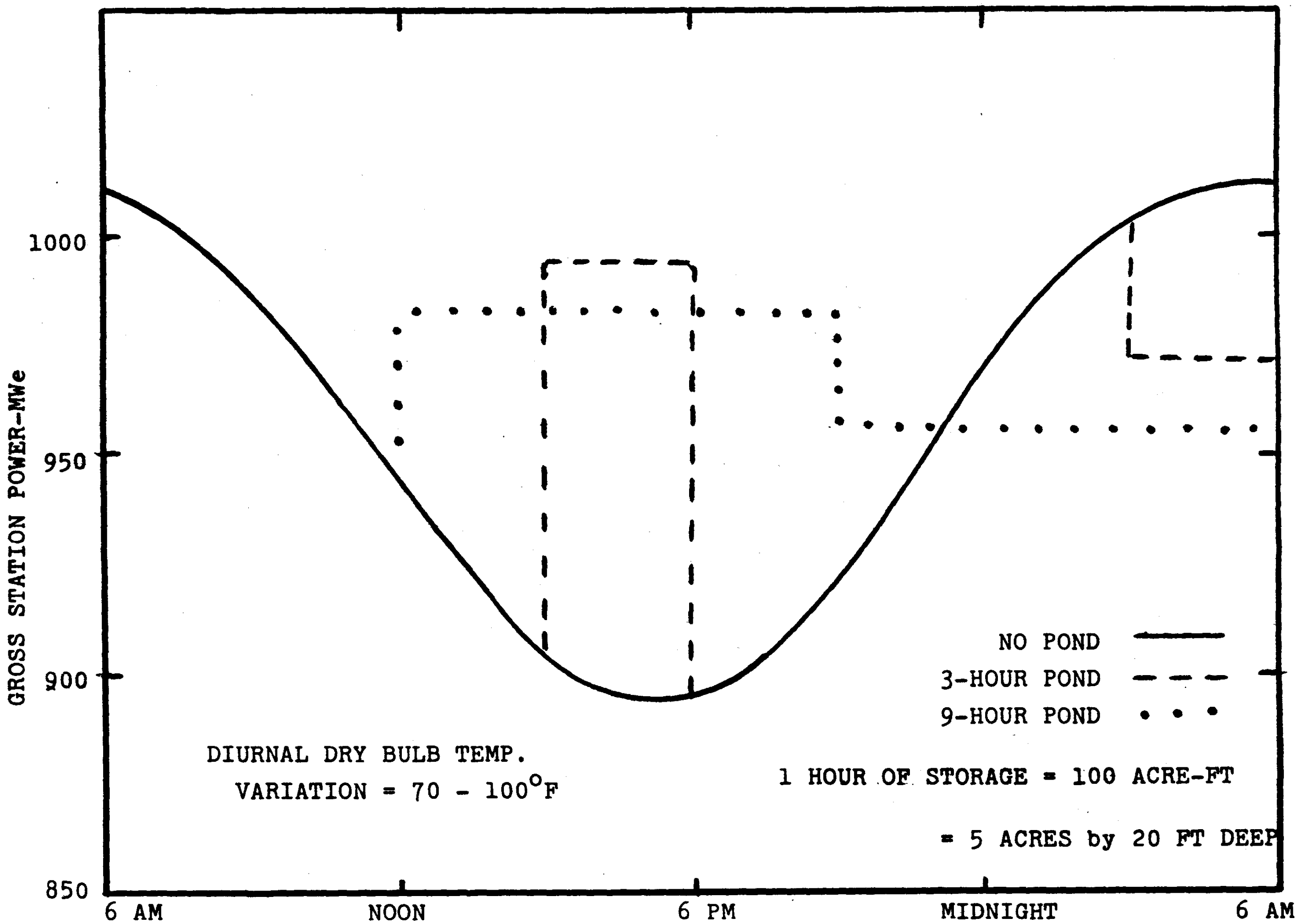


Fig. 3.16 Effect of Pond Size on the TSP/Dry Tower System Performance

3.4.4.4b Effect of Variations in the Ambient Daily Range

The effect of variations of the ambient temperature on the combined system performance at different absolute temperatures is shown in Fig. 3.17. A general conclusion which can be derived from this figure is that the excess capacity due to the use of a thermal storage pond increases with increases in both the magnitude of the peak daily temperature and the range of the daily temperature. Fig. 3.18 shows the total extra daily electrical energy generated at different temperature ranges at different peak daily temerpatures. The increase of the total daily plant output is due to 1) cooling of the water stored in the pond 2) the nonlinearity of the turbine heat rate vs. condensing temperature curve. The nonlinearity effect seems to be predominant. The slope of the heat rate versus condensing temperature curve is greater at high condensing temperatures and thus a unit change in the condensing temperature at high condensing temperatures results in a greater change in the plant power output. Note, however, that deviation from this behavior seems to occur at hign peak ambient temperature and low daily range. This is a result of cooling of the pond water since for such conditions the pond temperature

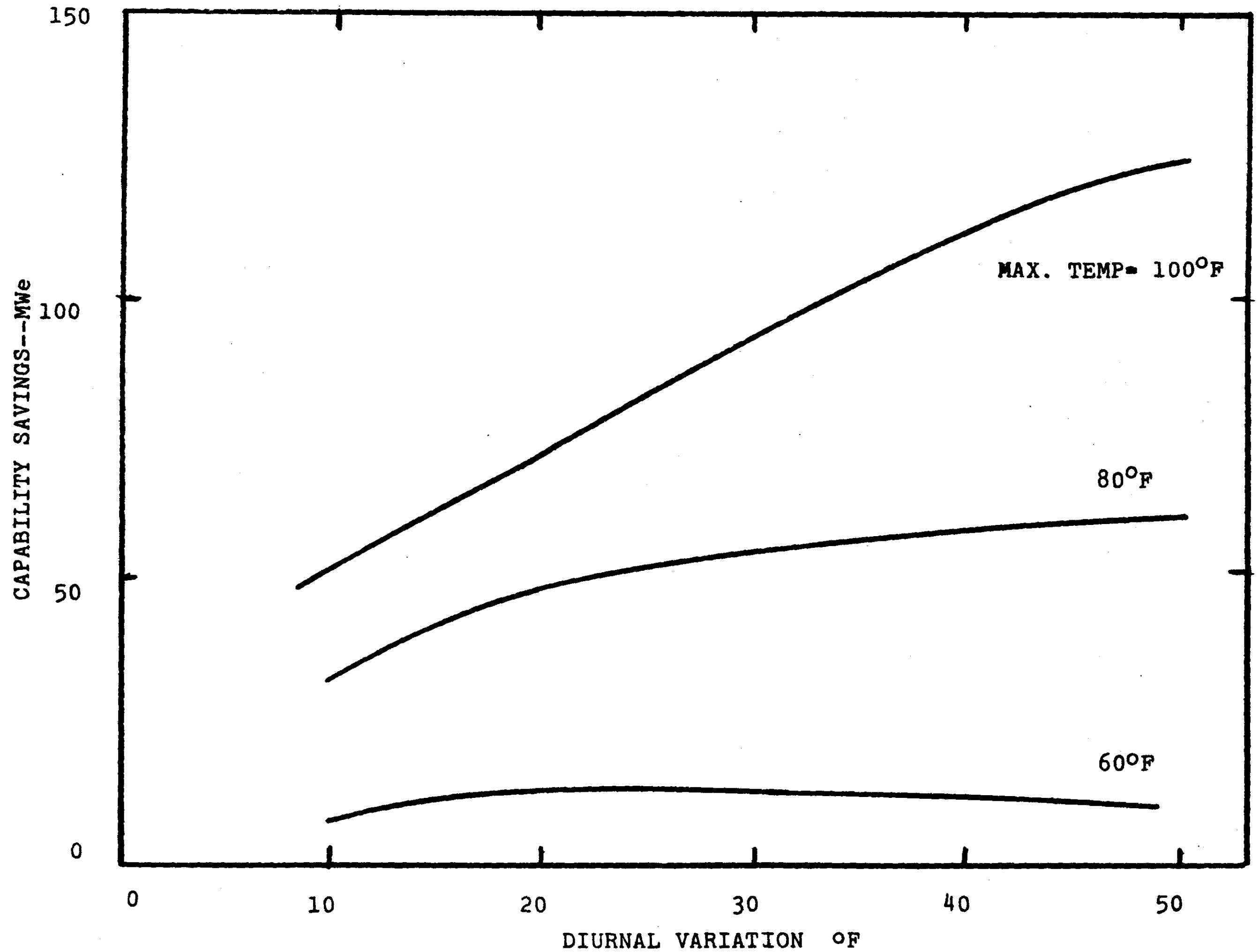


Fig. 3.17 Effect of Daily Variation of Ambient Temperature on the Average Extra Peak Capacity

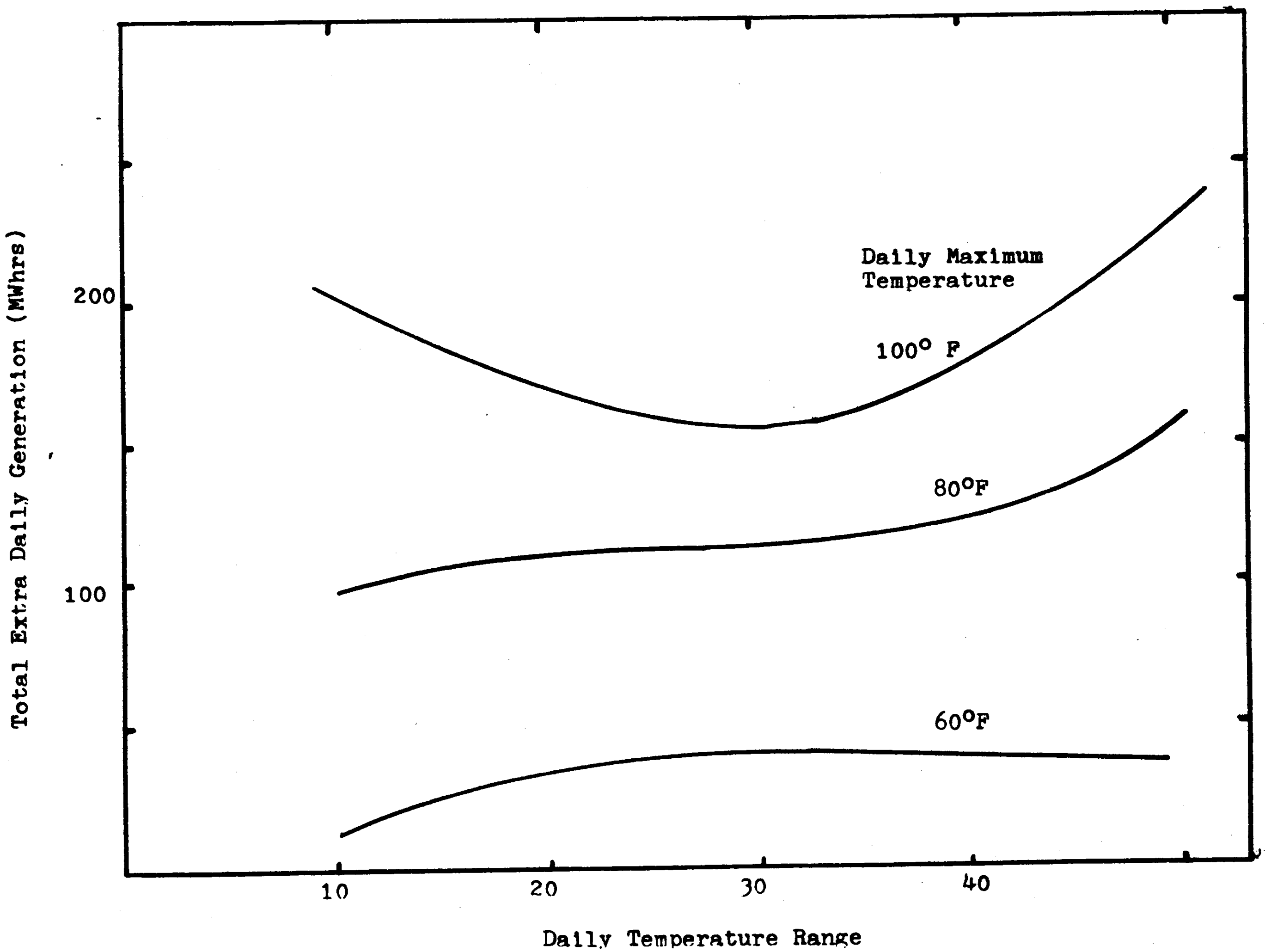


Fig. 3.18 Effect of Daily Temperature Range on Total Extra Power Generation

becomes quite high (greater than about 120^{o}F). The effect of pond cooling on the performance of the combined system will be examined in a later section.

3.4.4.4c <u>Significance of Alternative Pond Utilization Schemes</u>

The standard combined system as described earlier calls for the utilization of the pond solely for condenser inlet water supply for a time period which will result in all the pond inventory passing once through the condenser. For such an operational scheme the temperature of the pond water at the end of the heatup period may still be substantially lower than that which would result from the use of the towers only. So there exists the option of cycling water in the pond through the condenser and towers again to achieve additional generating capacity gains while the system electrical demand is still high. Another alternative to the standard utilization scheme would be the partial utilization of the pond over an extended period of time. In this operational mode water from the pond would be mixed with the tower discharge to supply cooling water for the condenser. For the standard 3 hour capacity pond mixed in a 1 to 1 ratio with the tower discharge the benefits of the pond could be extended to six hours.

There are, however, penalties associated with the increased utilization of the pond. The need to reject additional quantities of heat during the pond cooldown period would result in an increased pond temperature at the end of the cooldown period and hence a loss of benefit during the next peak load period. Also such a system utilization would result in a greater sensitivity to daily variations in the ambient temperature range. Figure 3.19 shows the power histories of following:

Curve A-- the standard operational procedure

Curve B-- a double cycling of the pond

Curve C-- a partial utilization of the pond of the standard pond over a six hour period

3.4.4.4d <u>Evaporation from the Thermal Storage Pond</u>

Significant amounts of water loss from the thermal storage pond by evaporation would occur at high ambient temperatures. Covering of the pond would prevent this loss. Whether or not such a cover would be economical would depend on the economic and hydrological characteristics of the site.

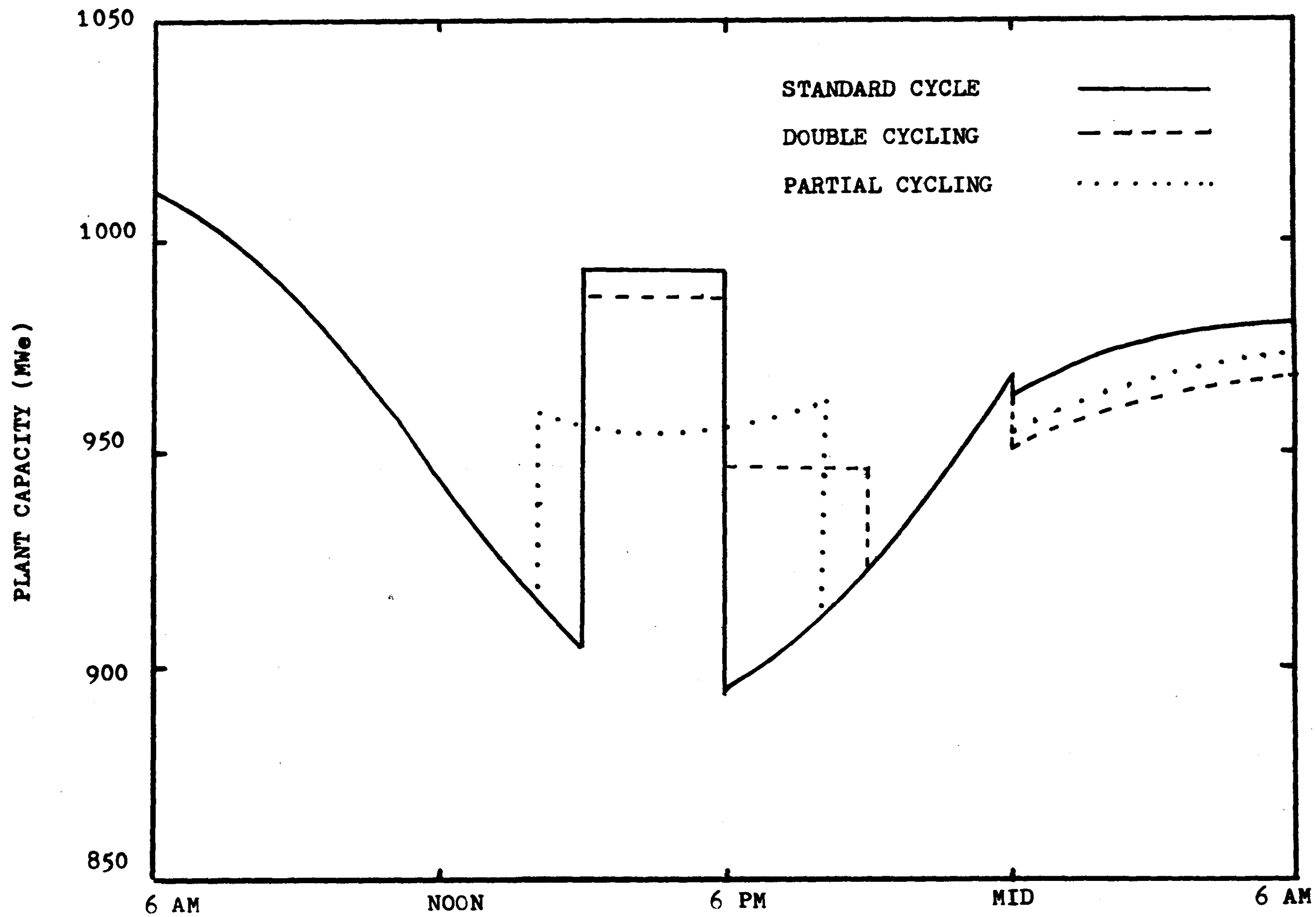


Fig. 3.19 Effect of Alternative Cycling Sequences on TSP/Dry Tower System Performance

The rate of evaporation from the pond for different daily temperature ranges at different peak daily temperature is shown in Fig. 3.20. This may be compared to an equivalent once-thru cooling utilization of about 1100 ft^3/sec and a cooling tower evaporative consumption of about 20 ft^3/sec.

3.4.4.4e Significance of Heat Transfer From the Pond

The significance of heat transfer from the pond to the environment can be determined by comparing the system performance for the cases of heat transfer and no heat transfer from the pond. The calculated performance for each of these cases for the standard conditions is summarized in Table 3.8. General conclusions derivedfrom these results are:

1) the effect of heat transfer from the pond increases with increasing daily average ambient temperature,
2) the performance of the thermal storage pond-dry tower system is not strongly dependent on the amount of heat transfer from the pond-thus, covered ponds appear to be feasible.

3.4.4.5 Simulation of Combined System Performance for an Entire Year

TABLE 3.8

Significance of Heat Transfer from the Thermal Storage Pond

Case 1-- Heat Transfer from Pond Surface

Case 2-- No Heat Transfer from Pond Surface

Daily Temperature Range	Case #	Daily Total Extra MWhrs	Peak Load Time Extra MWhrs	Average Extra Peak Capacity -MWe
70-100°F	1	155	283	94.3
	2	10	232	77.3
50-100°F	1	233	373	124.3
	2	155	358	119.3
50-80°F	1	112	160	53.3
	2	90	154	51.3
30-80°F	1	154	187	62.3
	2	142	187	62.3

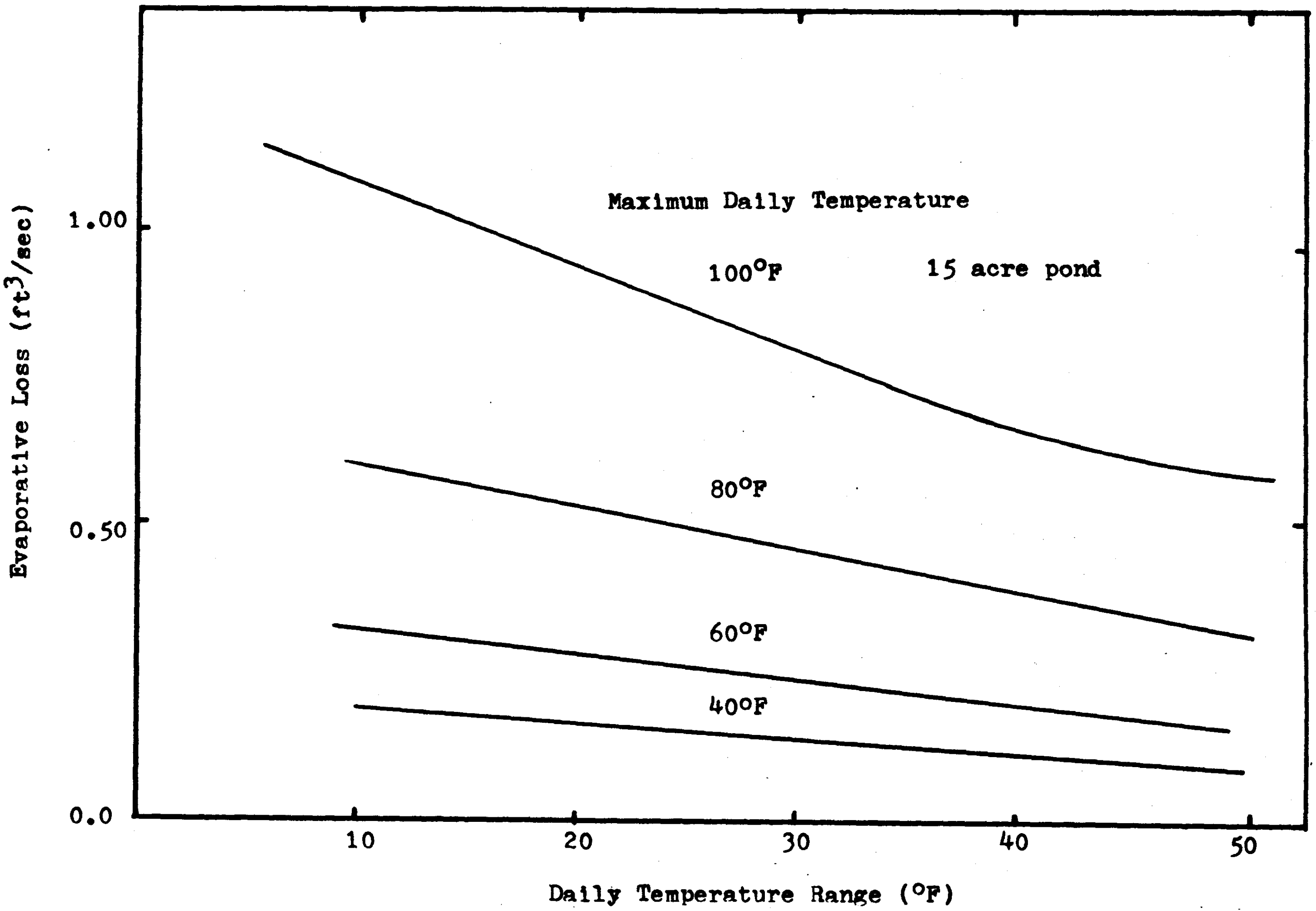


Fig. 3.20 Evaporation from a Thermal Storage Pond

Although the previous section gives much insight into the performance characteristics of thermal storage ponds, the crucial test of such a system is the evaluation of the performance for an entire year. Thus, in this section the results of the computer simulation of the annual performance of the earlier described standard thermal storage pond-dry tower system is reported.

The standard thermal storage pond-dry tower system design used in this study is arbitrarily defined and hence does not result in optimal performance. The dry tower component of this combined system, however, matches those found to be optimal for the western regions of the United States. The meteorological variations used are those for the town of Winslow, Arizona for the year 1974 and should be representative of arid, moderate altitude, western regions.

The daily maximum and minimum temperatures for the entire year were input into the simulation model. It is assumed that the maximum temperature occurs at 5 PM and the minimum temperature occurs at 5 AM. The temperature variation for the periods between the maximum and minimum is assumed to follow a sine curve.

Year-long simulations were performed for the following three cases:

Case 1) Dry Tower only ("simple" dry cooling),

Case 2) Combined thermal storage pond-dry tower system with an open pond (i.e. heat transfer from the pone), and

Case 3) Combined thermal storage pond-dry tower system with covered pond (heat transfer from the pond assumed to be negligible),

Tables 3.9,3.10,and 3.11 summarize the important results of these simulation studies.

The results show that there is a substantial incentive for considering the utilization of a thermal storage pond in conjunction with dry cooling towers. Both the covered and open ponds result in great savings in the summer time loss of peak time generating capability. The difference between the required peaking facility capability for Case 1 (no pond) and Case 2 (open pond) is 97 megawatts.

In summary, the thermal storage pond concept appears to offer a very attractive solution to the summer peaking problem of dry cooling towers. The system performance is, however, strongly dependent on the local meteorology.

TABLE 3.9

Comparative Peak Time Performance For Alternative Dry Cooling Systems

Peak demand time = 3 to 6 PM

	Yearly Minimum Peak Time Capacity-MWe	Annual Average Peak Time Capacity -MWe	Summer Average Peak Time Capacity -MWe
Case 1 Tower only	889	991	926
Case 2 Combined System-Open Pond	986	1037	1019
Case 3 Combined System Closed Pond	967	1032	1010

TABLE 3.10

Comparative Gross Power Generation

For Alternative Dry Cooling Systems

	Annual Energy Production-MWhrs	Summer Energy Production-MWhrs
Case 1 Tower Only	8,930,920	2,195,050
Case 2 Combined System-Open Pond	8,963,447	2,210,663
Case 3 Combined System-Closed Pond	8,951,354	2,202,186

TABLE 3.11

Comparative Average Monthly Peak Time Capacity

For Alternative Dry Cooling Systems

	Jan.	Feb.	Mar.	Apr.	May	Jun.
Case 1 Tower only	1042 MWe	1037	1020	1009	964	917
Case 2 Combined System Open Pond	1044	1044	1044	1044	1035	1018
Case 3 Combined System Closed Pond	1044	1044	1044	1043	1033	1010

	Jul.	Aug.	Sep.	Oct.	Nov.	Dec.
Case 1	930	933	965	1007	1031	1042
Case 2	1016	1027	1029	1039	1044	1044
Case 3	1006	1014	1023	1028	1044	1044

3.4.4.6 Applicability of Thermal Storage Pond Concept to Alternative Sites

Using averaged historical meterological data, the applicability of the thermal storage pond concept to sites other than the Winslow,Arizona site has been examined as a part of the preliminary evaluation of the concept. Table 3.12 illustrates the theoretical average summer capability savings during the peak demand period for a 1000 MWe plant with a modified conventional steam turbine obtained by multiplying the summer average daily range of the ambient dry bulb temperature by the slope of the heat rate versus temperature for the modified conventional turbine (Fig. 3.15) at high condensing temperatures - 0.4% per $^{\circ}$F. As evidenced by the results of the simulation calculation for the Winslow, Arizona site reported in the previous section it is expected that the actual savings would be about 75% of the theoretical values.

In general, the areas offering the greatest potential for the utilization of a thermal storage pond are the interior western regions. Russell (R9) remarked that in the United States, east of the Mississippi, the daily range of temperature varies from 12 to 20°F, with a 15°F average while west of the Mississippi to the Rocky Mountains it varies from 20° to 35°F. In a classical treatise on American weather Greely (G1) states,

TABLE 3.12

THEORETICAL AVERAGE SUMMER CAPABILITY SAVINGS DURING PEAK ELECTRICAL DEMAND PERIODS FOR A 1000 MWe NUCLEAR POWER STATION*

Reno, Nevada	123 MWe
Winslow, Arizona	115
Albuquerque, New Mexico	99
Boise, Idaho	97
Lander, Wyoming	93
Billings, Montana	92
Denver, Colorado	91
Salt Lake City, Utah	85
Walla Walla, Washington	85
Atlanta, Georgia	68
Boston, Massachusetts	61
Los Angeles, California	67
San Francisco, California	30

* Modified-Conventional Turbine

Meteorological Data taken from Reference (B1)

"The daily ranges over the Rocky Mountain and plateau regions are extraordinary. At Fat Apache, Arizona (elevation, 5050 feet), the mean daily range is no less than 42.6^{o}F Even as remarkable as are these ranges they are exceeded at Campo, California (elevation, 2710 feet), where the mean range for September is 45.4^{o}F, and from June to October inclusive averages 44.8^{o}F"

It is also observed that the days of maximum yearly temperature are usually coincidental with the occurance of greater than average daily ranges. This can be attributed to the fact that the clear sky conditions which lead to the maximum temperature normally persist until the evening thus creating favorable conditions for radiative cooling.

CHAPTER 4

DESIGN OF A THERMAL STORAGE POND

4.1 Introduction

4.1.1 Statement of Basic Problem

The basic function of a thermal storage pond is to provide simultaneously a relatively low temperature condenser cooling water source and a holding facility for the same water after it has passed through the condenser and has been recooled partially by the dry cooling tower. Thus, the fundamental fluid dynamics problem in designing the pond becomes apparent. It can be briefly stated as follows: design a container for the storage of a large volume of water such that the simultaneous withdrawal of the total initial inventory at rate Q and the introduction of water at an equal rate and different temperature is possible, with minimal mixing of the incoming water with that initially in place.

The type of thermal-hydraulic behavior required in the pond is exactly that of the well-known plug-flow. However, exact plug-flow in the pond may not be required since the dry cooling tower/TSP system may still perform effectively - in an economic sense - in spite of some undesirable mixing in the pond. The extreme case of mixing of the initial pond inventory with incoming flow is the case of fully-mixed behavior.

A further limit beyond that of the idealized fully-mixed case - in terms of overall system performance degradation - is that of the short-circuited pond. This situation is highly undesirable since any fraction of the pond which is isolated from the flow circuit becomes lost as a medium for the storage of waste heat with the net result that the effective storage volume of the pond is reduced accordingly.

Before considering the constraints in designing a TSP a note regarding the attitude taken in this investigation is useful. One can imagine that the TSP design problem can be solved by using some sort of mechanical separation of the initial pond inventory from the inlet flow into the pond. Mechanical separation could be achieved by utilizing two ponds, or by use of some type of movable mechanical barrier in a single pond. The former solution is unattractive economically, and the latter solution is seen to be unattractive if one considers the required size of the proposed pond, the duration of pond operation, and the range of available pond lining and roofing materials. Thus, the goal of this investigation has been to design a thermal storage device without mechanical segregation such that a significant fraction of the possible benefits of the thermal storage concept can be obtained at reasonably low costs.

It should also be pointed out that the TSP design effort is directed towards achieving an efficient and economical design for a particular application of the TSP concept. The particular application of interest is the use of the TSP to achieve a "matchup" of the daily periods of peak ability to reject heat with the daily period of peak utility system electrical demand. An alternative application of the TSP concept might be the use of a pond on a continual basis (no operation mode switching) to damp out daily ambient temperature variations. The design of a TSP which will allow the efficient "matchup" operation nevertheless has been selected as the design problem of most interest because of the following factors:

1) Most electric utilities experience difficulty in meeting the summer daily peak electrical demand. The utility systems incremental cost of power production (mills/kwhr) is highest during the peak demand period due to the required operation of peaking units with high heat rates. The "matchup" TSP will result in maximum plant output during this period thus keeping peaking unit operation to a minimum.

2) The "matchup" system could operate effectively with as little as 3 hours of storage. A continuously utilized pond in the "averaging" system would require at least about 12 hours of water storage since the magnitude of the characteristic response time of the pond would

necessarily be on the order of that of the period of the ambient temperature oscillation (24 hours).

Finally it is important to emphasize that although a preliminary evaluation has shown that the loss of TSP benefit is not large for the cases of a full-mixed pond as opposed to a plug-flow pond the correct design of the pond is extremely important. This is because the idealized fully-mixed behavior is not the "worst-case" which can be realized for the performance of a TSP. The "worst-case" is that of a short-circuited pond. As is discussed in Section 4.2.1, short-circuiting is likely to occur in a poorly designed TSP due to density-induced flow.

4.1.2 Design Constraints and Requirements

In attempting to formulate some preliminary design options for the plug-flow TSP the designer must consider the engineering and economic constraints or requirements which will be imposed on any design concept. A list of some of the desirable features of a well-designed pond is the following:

1) low pond head loss during operation,
2) low pond flow velocity,
3) minimal bottom and surface area,
4) not excessively deep,
5) smooth response of the TSP outlet temperature to changes in the inlet temperature,

6) effective operation during both "heatup" mode and "cooldown" mode of operation, and

7) modest inlet flow velocity.

The requirement of low pond head loss during operation stems simply from the fact that during the standby stage of the TSP operation the surface elevation of the pond will seek its natural level. If the pond structure must also contain the extra water elevation due to a large head loss through the flow circuit of the pond then storage capacity will be wasted. A low pond flow velocity is also desirable since the pond lining and covering will likely be fabricated from flexible plastic or rubber membranes.

The requirements of minimal bottom and surface areas and reasonable depth are contradictory when attempting to design a TSP for a fixed storage volume. Thus, some compromise is required. Minimal surface area is desirable for either the open or covered pond. In the case of the open pond it is of interest to minimize evaporation, and in the case of the covered pond it is of interest to minimize the cost of covering and lining the structure. The need to maintain a reasonable depth is connected with the difficulty of constructing high dikes for an above-ground level pond, and of excavating to large depths for a below ground level pond.

Additionally, it is a basic design constraint that the

TSP outlet temperature respond smoothly to the inlet temperature. Large and high frequency variations in the TSP outlet temperature (condenser temperature) as the storage capacity of the pond nears exhaustion could result in unacceptable transients in the turbine-generator behavior.

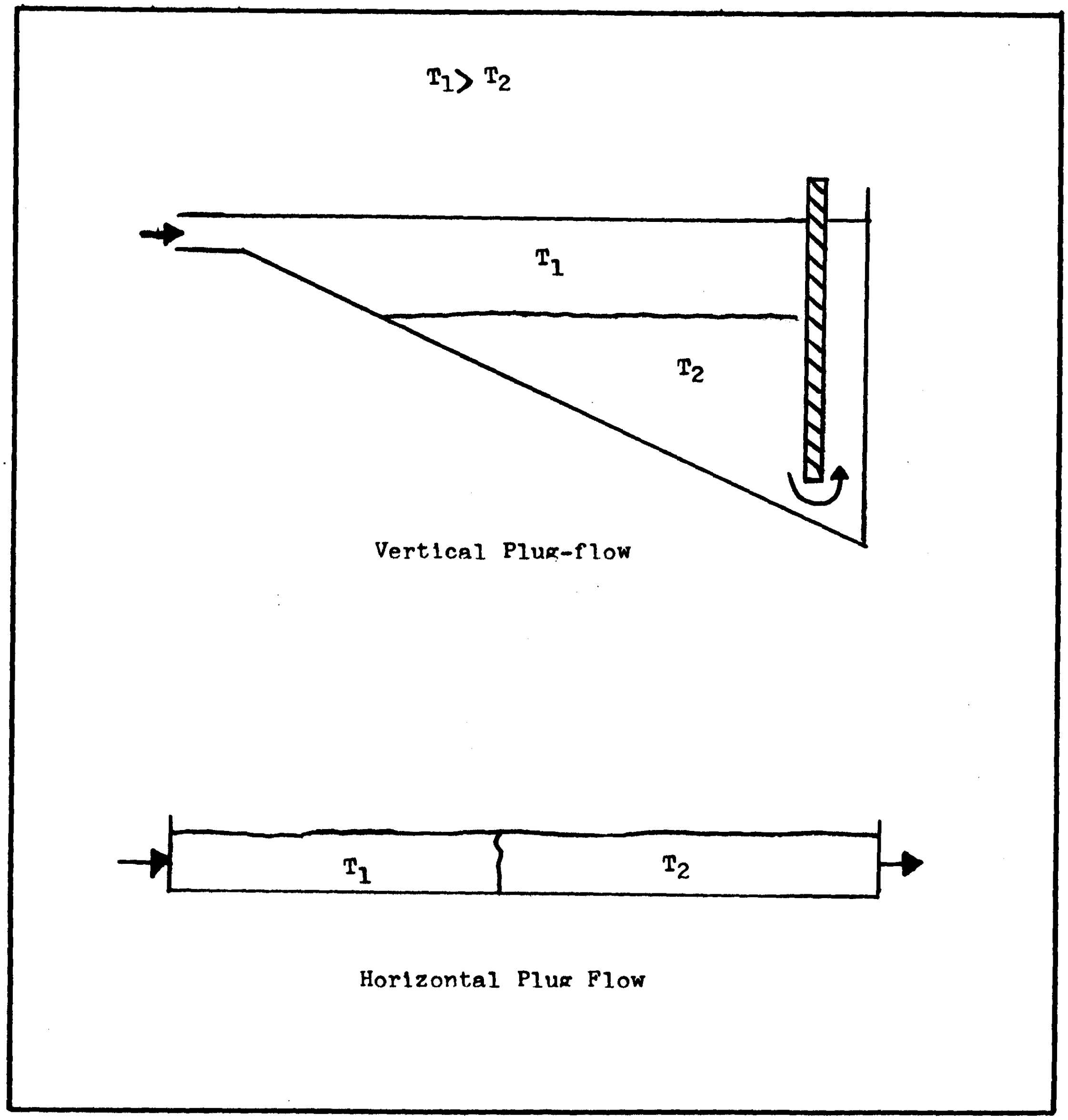
A most fundamental requirement is that the TSP functions effectively during both the "heatup" and "cooldown" modes of operation. Designing for one mode of operation will not automatically result in a correct design for the other due to the potential for the occurrence of density-induced flows in the TSP.

Finally, it is desirable to minimize the inlet velocity into the pond since the velocity head of the inlet flow is nonrecoverable and requires an additional amount of circulating water pumping power above that required for the operation of the tower only.

4.1.3 Feasible Solutions to the TSP Design Problem

There are two general methods by which the desired plug-flow behavior may be achieved. They are by means of horizontal plug-flow and vertical plug-flow. Figure 4.1 schematically demonstrates how the two types of behavior might be obtained in situations in which the reservoir is being filled with warm water. In the vertical plug-flow case one would attempt to take advantage of the tendency for stable vertical

Fig. 4.1
Horizontal and Vertical Plug-flow Concepts



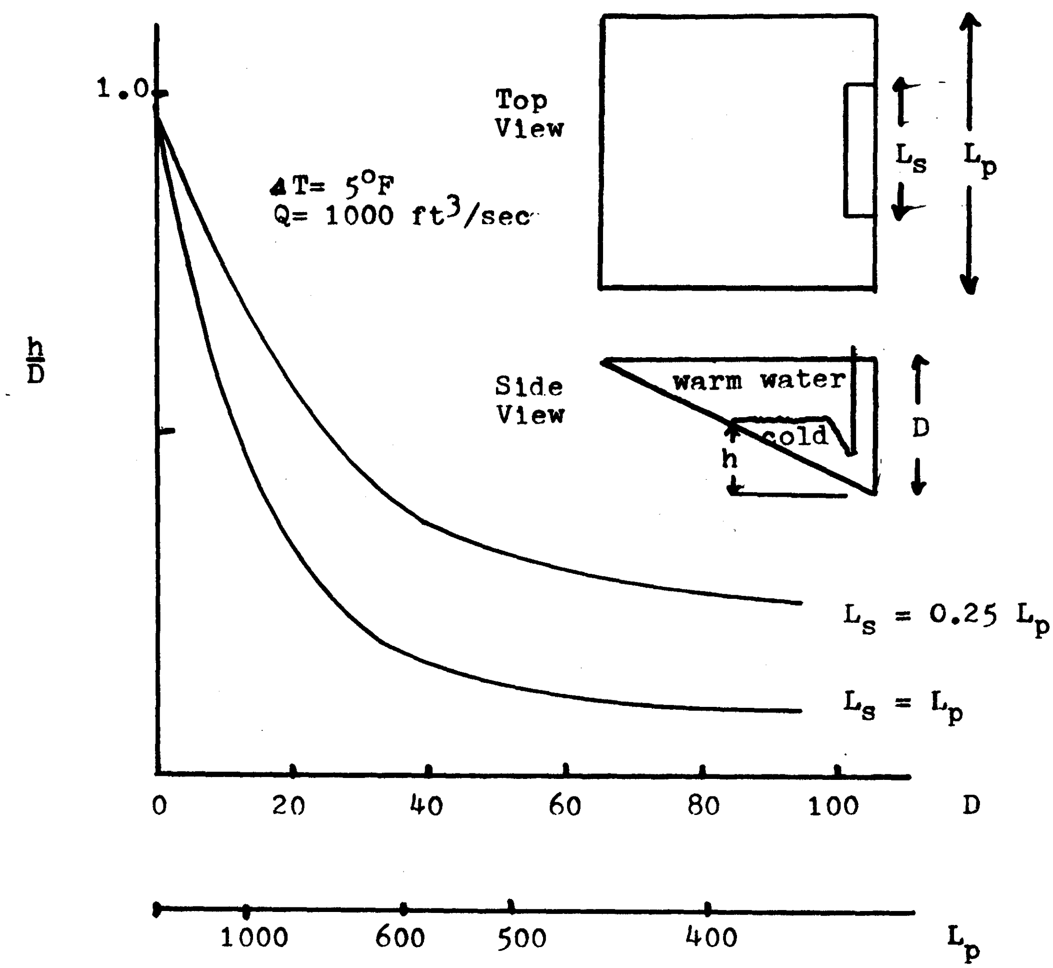
stratification, and in the case of the horizontal plug-flow one would have to guard against the tendency for vertical stratification (with the warm water floating above the colder water). As an alternative, one could consider the possibility of designing a fully-mixed pond, but this does not appear to be a practical alternative. To achieve uniform mixing the inlet flow discharge would have to be injected with high mixing (either jet induced or mechanical) at various points throughout the entire pond volume and the mixed condenser cooling water would have to be selectively withdrawn from the pond in a manner such that only the mixed pond water, and not the direct inlet flow, is withdrawn. Such a discharge and withdrawal system would be expensive and it is difficult to envision how one could guard against inadvertant short-circuiting.

Both the vertical and horizontal plug-flow design options have particular advantages and disadvantages with regard to the previously discussed economic and engineering constraints and performance requirements. Based on a somewhat qualitative comparison of the advantages and disadvantages of the two options it has been decided that an attempt would be made to design a horizontal plug-flow pond. This decision is justified by examining the applicability of both the vertical plug-flow and horizontal plug-flow pond concepts to the problem at hand. It is important to note that precise quantitative justification for the selection of either options based on their comparative economics is not possible, given the current state of available information. The economic benefits of the pond will largely depend on its thermal-hydraulic behavior, and in the cases of both the horizontal and vertical plug-flow ponds one if faced with the need for resolution of a previously unresolved and complex problem in transient non-homogeneous fluid flow. Additionally, the capital cost of the pond structure will depend strongly upon the particular geologic and economic characteristics of the site. Thus the preliminary option selection is made under some uncertainty.

For the vertical plug-flow design to be feasible it would be necessary to introduce the discharge into the pond in a manner such that little mixing with the initial inventory can occur. Later, as the water discharged into the pond begins to comprise a significant fraction of the pond volume selective withdrawal would be required. The problems which might be encountered in achieving a weakly-mixed inlet flow situation and selective outlet flow withdrawal can be appreciated by examining a hypothetical 300 acre-ft (about 3 hours of storage capacity) TSP. The pond is assumed to be square in shape, to utilize a skimmer wall for selective withdrawal, and to have a 5 °F temperature difference between the hot and cold water layers. The minimum depth of the cold lower layer before entrainment (drawdown) of the hot upper layer commences for such a pond has been calculated as a function of the pond geometry based on the work of Harleman [H3]. These results are shown in Fig. 4.2. Examining these data one notes that for practical man-made pond depths of 20 to 40 feet an extensive skimmer wall structure would be required to secure a satisfactory amount of selective withdrawal before drawdown (say greater than 80% of the initial cold water inventory).

An evaluation of the practicality of floating the discharge into the pond and over the initial cold water pond inventory is difficult to achieve. An analytical model for predicting the transient spreading of a confined buoyant surface discharge does not appear to be available currently. However, since it is of interest to minimize the entrance mixing so that the discharge would not interact directly with the selective withdrawal, a shallow, low-velocity inlet geometry is indicated. These combined requirements of low velocity and shallowness in turn indicate that a wide and shallow discharge structure is required in order to obtain the desired 1000 ft^3/sec discharge flow.

Other apparent problems with a vertical plug-flow pond include 1) operation during the "cooldown" mode, and 2) off-design condition performance. Since in the "cooldown" operational mode the discharge into the pond is denser than the initial pond inventory it would be necessary to reverse the



D = Pond Depth

h = Height of interface at drawdown for optimized skimmer wall withdrwal

L_p= Horizontal dimension of square pond

L_s= Lenght of Skimmer wall

Fig. 4.2 Interface Drawdown Height for Vertical Plug-flow TSP pond in order to obtain a stable density front. The off-design condition performance of a vertical plug-flow pond may also be a problem in that with a decreasing daily range of the ambient air temperature the thermal-hydraulic efficiency of the pond would decrease. The efficiency would decrease since a smaller daily ambient temperature range would result in a decreasing density difference between the "hot" and "cold" pond fluids, and thus decreased pond stratification stability with an attendant decrease in the pond's selective withdrawal performance.

In view of the difficulties encountered with the vertical plug-flow concept as outlined above, the horizontal plug-flow concept has been selected for detailed evaluation. This design appears to be superior in that no elaborate discharge and withdrawal structures are required and operation during both the "heatup" and "cooldown" modes would be identical. Additionally, the depths for typical man-made water storage facilities (about 20-40 feet) favor the horizontal plug-flow design. A detailed discussion of this concept follows.

4.1.4 Initial Design Concept

4.1.4.1 Concept and Design Goal

In the situation where the density of the inlet flow into a rectangular pond is equal to the ambient density, and where the inlet flow in uniformly introduced along one end and withdrawn uniformly at the opposite end, the inlet flow would form a nearby vertical interface with the original pond fluid and this interface would pass through the pond at the plug-flow velocity

$$V_p = Q/A , \tag{4.1}$$

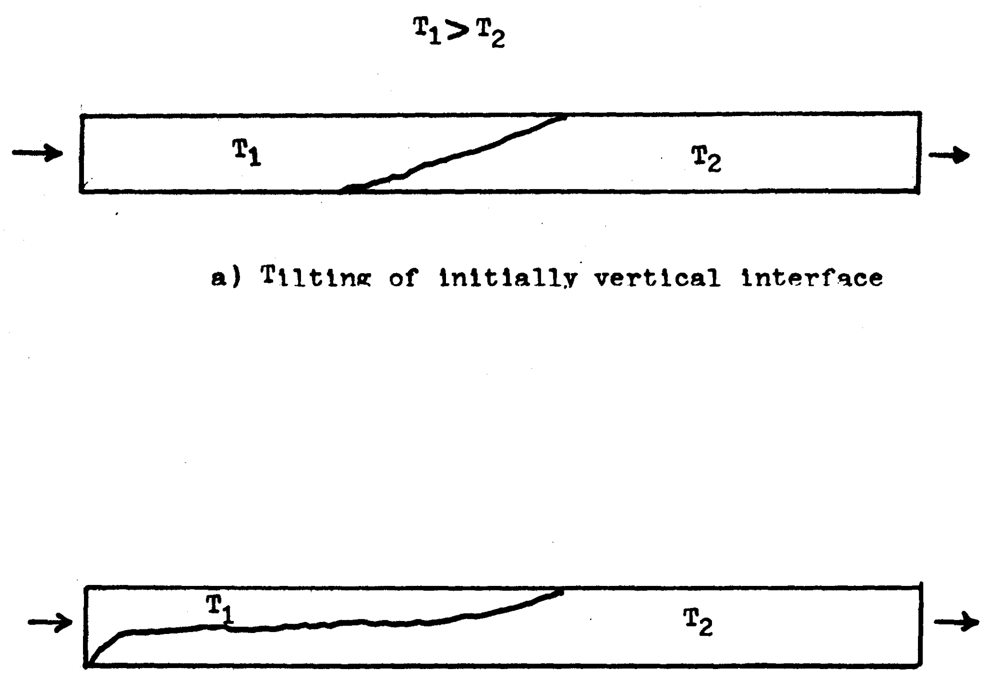
where Q = discharge flow rate ft^3/sec and,
A = pond cross-sectional area.

This behavior is shown in Figure 4.1. This is the ideal TSP thermal-hydraulic behavior. In the case of the actual TSP where the density of the inlet flow is different from that of the initial pond inventory the propagation of the front would be perturbed by the density-induced flow instability at the interface. Two general cases of possible behavior due to density effects are shown in Fig. 4.3. The first, shown in Fig. 4.3a, would arise when there is only a weak tendency for stable stratification and the second, shown in Fig. 4.3b would result when there is a strong tendency for stable stratification. The second case is highly undesirable in terms of the desired TSP performance, while the acceptability of the first case would depend on the rate of propagation of the incoming fluid to the pond withdrawal structure. The initial design concept utilizing the horizontal plug-flow behavior is based on an attempt to design a TSP such that the inclination of the density front is minimized.

Fig. 4.3

Effects of Density-induced Flows in

Horizontal Plug-flow TSP



a) Tilting of initially vertical interface

b) Developement of two-layered structure

4.1.4.2 Assessment of Performance - Basic Design Tradeoff

A review of the pertinent literature has revealed no previous work which directly addresses the problem of achieving transient horizontal plug-flow in the presence of density differences. However, some quantitative arguments based on available information can be employed to characterize the general type of pond required to achieve the desired performance.

A determination of the magnitude of the density-induced velocity at a density interface for an initially static, vertical density interface is a classical problem in stratified flow. This problem is termed the lock-exchange problem due to its similarity to the transient flow which arise from the opening of a channel lock interfacing between initially static fresh and salt water. Simple conservation of the available potential energy resulting from the density difference across the interface is shown in Fig. 4.4 and results in the following prediction for the initial density front velocity:

$$V_F = \frac{1}{2}\left(g \frac{\Delta\rho}{\rho} H\right)^{1/2} , \qquad (4.2)$$

where $\Delta\rho$ = the density difference

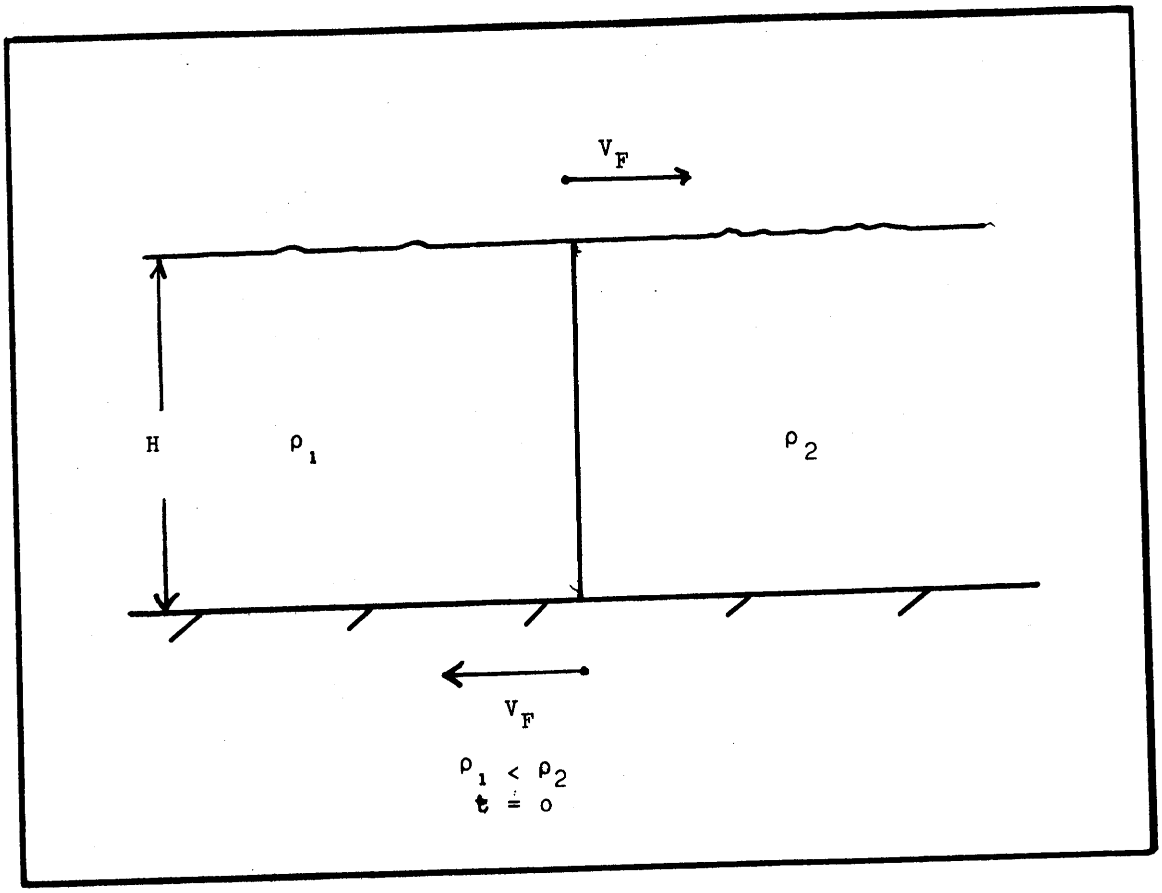
ρ = average density,

H = depth, and

g = gravitational acceleration.

Fig. 4.4

Unstable Vertical Density Interface



Assuming that this relationship should also provide an approximate estimate of the additional density-induced velocity of an interface, which was initially vertical, and moving at a velocity V_p, the total velocity of the leading edge of the density front would be

$$V_F = V_p + \frac{1}{2}\left(g \frac{\Delta\rho}{\rho} H\right)^{1/2} \quad . \tag{4.3}$$

Thus, in order that the pond not be badly short-circuited by the density-induced flow it would be required that

$$\frac{V_p}{\frac{1}{2}\left(g \frac{\Delta\rho}{\rho} H\right)^{1/2}} \gtrsim 5 \text{ to } 10 \quad . \tag{4.4}$$

This is equivalent to requiring that

$$F_D = \frac{V_p}{\left(g \frac{\Delta\rho}{\rho} \frac{H}{2}\right)^{1/2}} \gtrsim 3.5 \text{ to } 7.0 \quad , \tag{4.5}$$

where F_D is the pond densimetric Froude number. For a typical pond flow of 1000 ft^3/sec and ΔT = 10 °F a narrow and shallow TSP is indicated to be desirable. Using F_D = 2.5 as the design Froude number, the required pond dimensions would be:

depth = 10 feet,

width = 60 feet, and

length = 21,000 feet for a 3 hour pond.

Such a design would not be acceptable due to the large ratio of surface area to stored volume, the large flow velocity, and the extremely long length of the channel.

However, it may be possible to reduce the design Froude number substantially since some effects which may retard the motion of the density front have not been taken into consideration. Nevertheless, the fundamental design tradeoff has been established in this example -- increasing the pond (channel) design densimetric Foude number would lead to increased pond construction costs. The design task is thus the specification of the minimal design Froude number such that an accept able level of performance is achieved.

4.1.4.3 Information Requirements for Accurate TSP Performance Assessment

The correct design of the horizontal plug-flow TSP will require 1) the design of a pond discharge structure such that the initial condition of a near-vertical density front is achieved, and 2) the prediction of the behavior of the density front as it propagates downstream.

4.1.4.3a Entrance Region

The design of the pond inlet flow structure can be based on the work of Jirka [J1] et al. This author has presented criteria for the stability of submerged multiport discharges which can be used to design an inlet structure for a highly unstable submerged multiport jet discharge into the TSP. Unstable submerged jets are those which do not result in a stable stratified, two-layered flow in the mixing region (near-field) of the jet, but rather which produce vertical mixing in the vicinity of the jet. In the actual TSP it will not be possible to establish initially a distinct interface between the hot (or cold) inlet flow discharge and the ambient pond water as a result of jet-induced mixing. However, the mixing region should be flushed rapidly by the inlet flow discharge at a velocity equal to the plug-flow velocity. Therefore a zone transition from ρ_1 to ρ_2 moving at V_p wil be established initially which will be short with respect to the entire length of the pond, and which will display a minimal amount of vertical stratification. This postulated behavior has been observed in the experiments discussed later in this chapter.

The criterion of Jirka is given in Fig. 4.5. Definition of the parameters constituting the criterion are as follows:

$$F_S = \left(\frac{4\ell}{D\pi} \right)^{1/2} \frac{V_J}{\left(g \frac{\Delta\rho}{\rho} D\right)^{1/2}} \tag{4.6}$$

= equivalent slot jet densimetric Froude number,

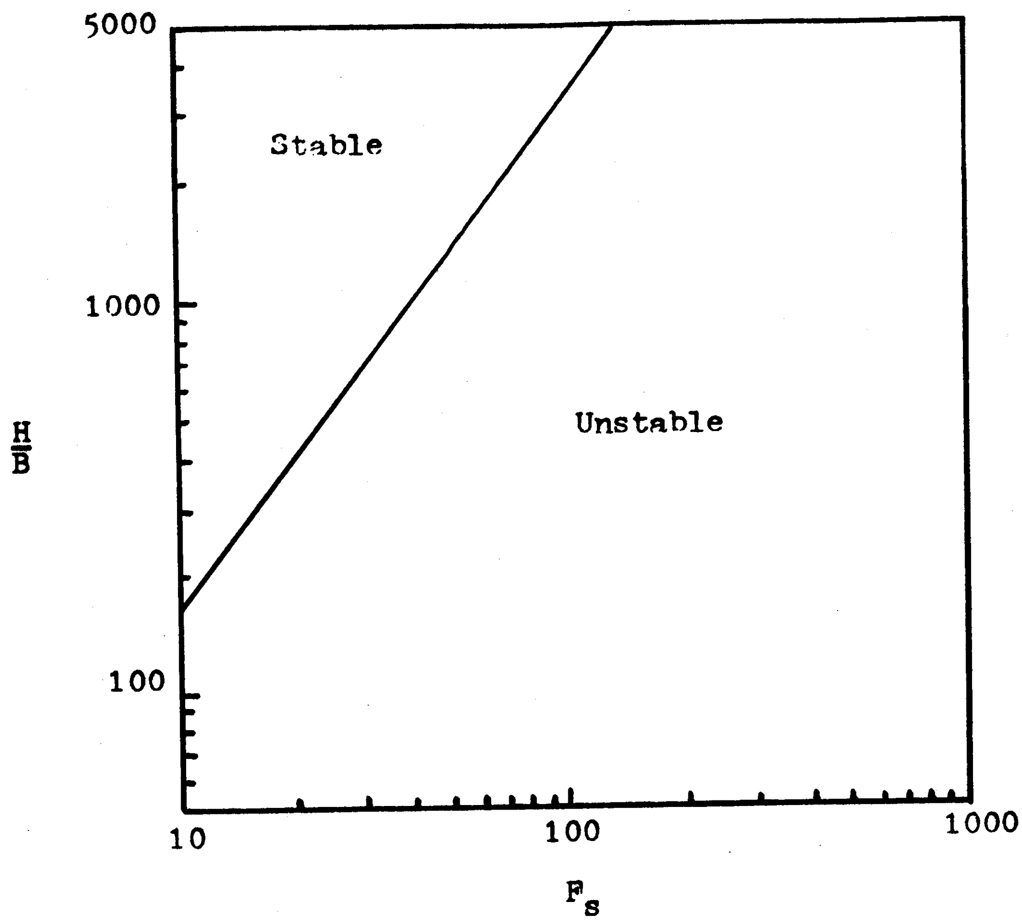


Fig. 4.5 Stability of TSP Inlet Region (J1)

where V_J = jet velocity,

D = diameter of multiport jets,

ℓ = centerline spacing between jets,

H = depth of pond,

$\Delta\rho$ = initial density difference, and

$B = \frac{D^2\pi}{4\ell}$.

The design of the highly mixed entrance region is aided by the fact that a horizontal plug-flow pond will in general be narrow and shallow with a small initial temperature difference. For the range of TSP depths and widths of practical interest calculations indicate that reasonable discharge designs would fall well below the line of transition from stable to unstable jets shown in Fig.4.5

4.1.4.3.b Density Front Propagation

A literature survey was unable to discover any published theoretical or experimental information which would enable the confident prediction of the propagation behavior of the density front as it passes through TSP. As indicated in Section 4.1.4.2 some simple arguments can be made regarding the general magnitude of the density-front velocity, but these are at best speculative. A more accurate model is required along with experimental verification.

A steady-state criterion has suggested the establishment of fully-vertical mixed flow in ponds [J1]. This steady-state criterion predicts that the expulsion of any underlying cold wedge will occur in cases for which $F_D \geq 1.0$. However, this criterion, which is based on a solution of a limiting case of the two layer shallow water equations, is not applicable in any respect to the thermal-hydraulic behavior of a TSP since it does not account for the withdrawal flow behavior exhibited at the outlet of the TSP. In fact, if the withdrawal from the pond is taken over the full depth of the pond, one would expect that at long time (i.e. steady-state) no cold underlying wedge would exist at any densimetric Froude number due to the interfacial shear flow acting on the cold wedge. The cold bottom layer would gradually diminish in thickness since there is no mechanism for its renewal. But, as a practical matter, the ultimate condition of vertically homogeneous flow may not be important since such cold wedge explosion would probably occur only long after the plug-flow flushing time, $V/Q = T_{flush}$ (V - pond volume, Q = flow rate).

Koh and Fan [K7] have addressed the problem of transient stratified flow for the case of a surface discharge layer spreading over an infinite quiesent body of colder water, but likewise the results are not applicable to the case of interest in this work.

To provide the required ability to describe the behavior of a propagating density front in a TSP a control-volume type fluid dynamic model has been derived, and its predictions have been matched to experimental results. The experiment has been designed such that it serves the dual purposes of providing the data necessary to construct the semi-empirical mathematical model of transient stratified flow in shallow channels for a limited range of densimetric Froude numbers, and of providing a physical model of a prototype thermal storage pond.

4.2 Modeling of the Initial Design Concept

The modeling of the thermal-hydraulic behavior of the initial TSP design concept has been based on the experimental examination of the behavior of a TSP model. Subsequently, a simple control-volume analytical model has been fitted to the experimental data by means of appropriate adjustment of one free parameter. Assessment of the thermal-hydraulic behavior of the initial design concept in relation to the practical implementation of the dry cooling tower-TSP system indicates the need for modification of the initial concept.

4.2.1 Physical Model

4.2.1.1 Similarity Requirements

Total physical similarity between model and prototype TSP's requires similarity of flow in three regions. These are the entrance mixing region, the main storage volume, and the withdrawal region. The intent in physically modeling a horizontal plug-flow TSP is the replication of the behavior of the entrance mixing and main storage regions. Modeling of the withdrawal region is not crucial in this work, since if the horizontal plug-flow concept is to be viable, vertical stratification in the withdrawal region will necessarily be minimal (or at least this would be true for the greater part of the TSP operational time).

It is important to note that no attempt has been made to delineate the positions of the boundaries between the different density regions. As has been indicated previously, a horizontal plug-flow TSP will likely have a channel-type geometry with a large length/width ratio. In any situation, the boundaries of the entrance and withdrawal regions will not likely exceed a length of more than several pond widths, and thus will comprise only a small fraction of the total pond volume. Therefore, accurately specifying these boundaries does not appear to be essential.

Similarity to the entrance region of a prototype TSP is obtained in the model in an integral sense by assuming that any submerged jet discharge with a highly unstable near-field region will produce essentially the same vertical temperature mixing effect in a geometrically similar region, without regard for the geometrical details of the discharge structures. Since the volume of the entrance region is small in relation to the total storage volume this assumption should be adequate. In any event, the near-field mixing region would be flushed rapidly by the inlet flow, and would quickly approach the locally steady-state condition that the inlet discharge temperature would be equal to the ambient water temperatures (i.e. no back flow from the storage region to the entrance mixing region would occur). In both the model and the prototype an unstable near-field flow can be obtained by using a horizontal multiport submerged jet discharge with the jets uniformly spread over the width of the pond (see Fig. 4.6).

To insure similarity between the main storage volume flows of the model and the prototype, the following requirements are met:

1) geometric similarity,
2) Froude Law similarity,
3) densimetric Froude Law similarity,
4) satisfaction of Reynolds criterion, and
5) similarity of the ratio of inertial to friction forces.

The requirement of geometric similarity is fundamental to all hydraulic scale-models. The remaining similarity requirements can be derived from the differential equation of motion for channel flow with buoyant acceleration. The momentum equation is

$$\frac{\partial u}{\partial t} + \frac{\partial u^2}{\partial x} + \frac{\partial uv}{\partial z} = - \frac{1}{\rho} \frac{\partial p}{\partial x} + \nu\left(\frac{\partial^2 u}{\partial z^2}\right) \tag{4.7}$$

where u = component of velocity in x direction,

v = component of velocity in z direction,

ρ = density,

ν = kinetic viscosity, and

p = hydrostatic and dynamic pressure = $\rho g x + P_D$.

Note also that

$$\frac{\partial p}{\partial x} = g\rho + gx\,\frac{\partial \rho}{\partial x} + \frac{\partial P_D}{\partial x} \quad . \tag{4.8}$$

Strictly speaking the force of gravity would not act in the direction x, but is included in Eq. (4.7) to show the derivation of the basic dimensionless groups. For geometric similarity,

$$K_l = x/x_1 \quad , \tag{4.9}$$

where the subscripted variable refers to the model. For dynamic similarity [R4],

$$K_\rho = \rho/\rho_1$$

$$K_\nu = \nu/\nu_1$$

$$K_t = t/t_1$$

$$K\frac{dP_D}{dx} = \left(\frac{dP_D}{dx}\right)/\left(\frac{dP_D}{dx}\right)_1$$

$$K_u = u/u_1,\ v/v_1$$

$$K_{\Delta\rho} = \Delta\rho/\Delta\rho_1$$

$$K_g = g/g_1$$

where $\Delta\rho$ equals the characteristic density difference. Now rewriting Eq. (4.7) in terms of the subscripted quantities and the scaling ratios:

$$\frac{K_u}{K_t}\left(\frac{\partial u_1}{\partial t_1}\right) + \frac{K_u^2}{K_L}\left(\frac{\partial u_1^2}{\partial x_1}\right) + \frac{K_u^2}{K_L}\left(\frac{\partial u_1 v_1}{\partial z_1}\right) =$$

$$\left(\frac{K_g K_L K_{\Delta\rho}}{K_\rho K_L}\right)\frac{g_1 x_1 \partial\rho_1}{\rho_1 \partial x_1} - K_g g_1 - \left(\frac{K_{\frac{dP_D}{dx}}}{K_\rho}\right)\left(\frac{\partial P_D}{\partial x}\cdot\right)_1 \frac{1}{\rho_1} +$$

$$+ \left(\frac{K_\nu K_u}{K_L^2}\right)\nu_1\left(\frac{\partial^2 u_1}{\partial z_1^2}\right) \qquad (4.10)$$

From the above we find the following must be true

$$\frac{K_L}{K_t K_u} = 1 \qquad (4.11)$$

$$\frac{K_g K_L}{K_u^2} = 1 \qquad (4.12)$$

$$\frac{K_L K_g K_{\Delta\rho}}{K_\rho K_u^2} = 1 \qquad (4.13)$$

$$\frac{K_L K_{\frac{dP_D}{dx}}}{K_\rho K_u^2} = 1 \qquad (4.14)$$

$$\frac{K_\nu K_u/K_L^2}{K_u/K_L} = 1 \tag{4.15}$$

Equation (4.11) simply indicates the ratio of characteristic times for the model and the prototype, i.e.,

$$\frac{t}{t_1} = \frac{x/u}{x_1/u_1} \quad . \tag{4.16}$$

Equation (4.12) yields the Froude law similarity requirement

$$F = \frac{u}{\sqrt{gL}} \tag{4.17}$$

and Equation (4.13) gives the densimetric Froude number

$$F_d = \frac{u}{\sqrt{g \ \Delta\rho/\rho \ L}} \tag{4.18}$$

for the characteristic flow depth L. Equation (4.14) gives the dimensionless quantity defined as the friction factor

$$f = \frac{(\frac{dP_D}{dx})L}{\rho u^2} \tag{4.19}$$

Lastly, Eq. (4.15) gives the well-known Reynolds number:

$$Re = \frac{uL}{\nu} \tag{4.20}$$

In Froude models it is impossible to meet both Froude similarity and Reynolds similarity exactly. However, exact similarity of Reynolds number for turbulent flows is not required. It is generally satisfactory to meet the criterion

$$Re_{model} > Re_c \tag{4.21}$$

where Re_c is the critical Reynolds number for fully developed turbulent flow.

Although it is not essential to meet Reynolds number similarity, it is important, as noted by Jirka [J1] that the ratio of inertial forces to bottom friction forces be equal for the model and the prototype. Thus in the physical modeling of the prototype TSP it is desirable to distort the vertical dimension of the model such that the ratio of friction forces to inertial forces in the prototype is approximately replicated in the model. This ratio can be expressed by

$$RFI = \frac{f_o \rho \nu^2 (L/D_H)}{\rho V^2} \tag{4.22}$$

where f_o = friction factor, f(Re),

V = velocity,

L = characteristic horizontal dimension, and

D_H = hydraulic radius.

Cancelling terms one obtains the result

$$\mathrm{RFI} = f_o \frac{L}{D_H} \quad . \tag{4.23}$$

Examining Fig. 4.6 we see that over a limited range of aspect ratios, the hydraulic diameter of a rectangular channel is approximately directly proportional to the aspect ratio.

Since D_H, the hydraulic diameter, is approximately equal to $\alpha D/W$ over a limited range we have for a constant w, f_o, and L the relationship

$$\frac{\mathrm{RFI}_2}{\mathrm{RFI}_1} \cong \frac{f_o \; L/D_{H_2}}{f_o \; L/D_{H_1}} \cong \frac{f_o L/\alpha(D_2/w)}{f_o L/\alpha(D_1/w)} \; , \text{ or} \tag{4.24}$$

$$\frac{\mathrm{RFI}_2}{\mathrm{RFI}_1} \cong \frac{D_1}{D_2} \quad . \tag{4.25}$$

Thus, as long as the aspect ratio of the model remains within the limited range the ratio of frictional to inertial forces may be taken into account simply through its inverse dependence upon depth. The required distortion is given as

$$\mathrm{RFI}_{\text{prototype}} = \mathrm{RFI}_{\substack{\text{distorted}\\ \text{model}}} = \frac{D_{\text{undistorted}}}{D_{\text{distorted}}} \mathrm{RFI}_{\text{undistorted}}$$

In general RFI in the geometrically undistorted model is greater than that of the prototype. Thus, an increase in the model depth is required.

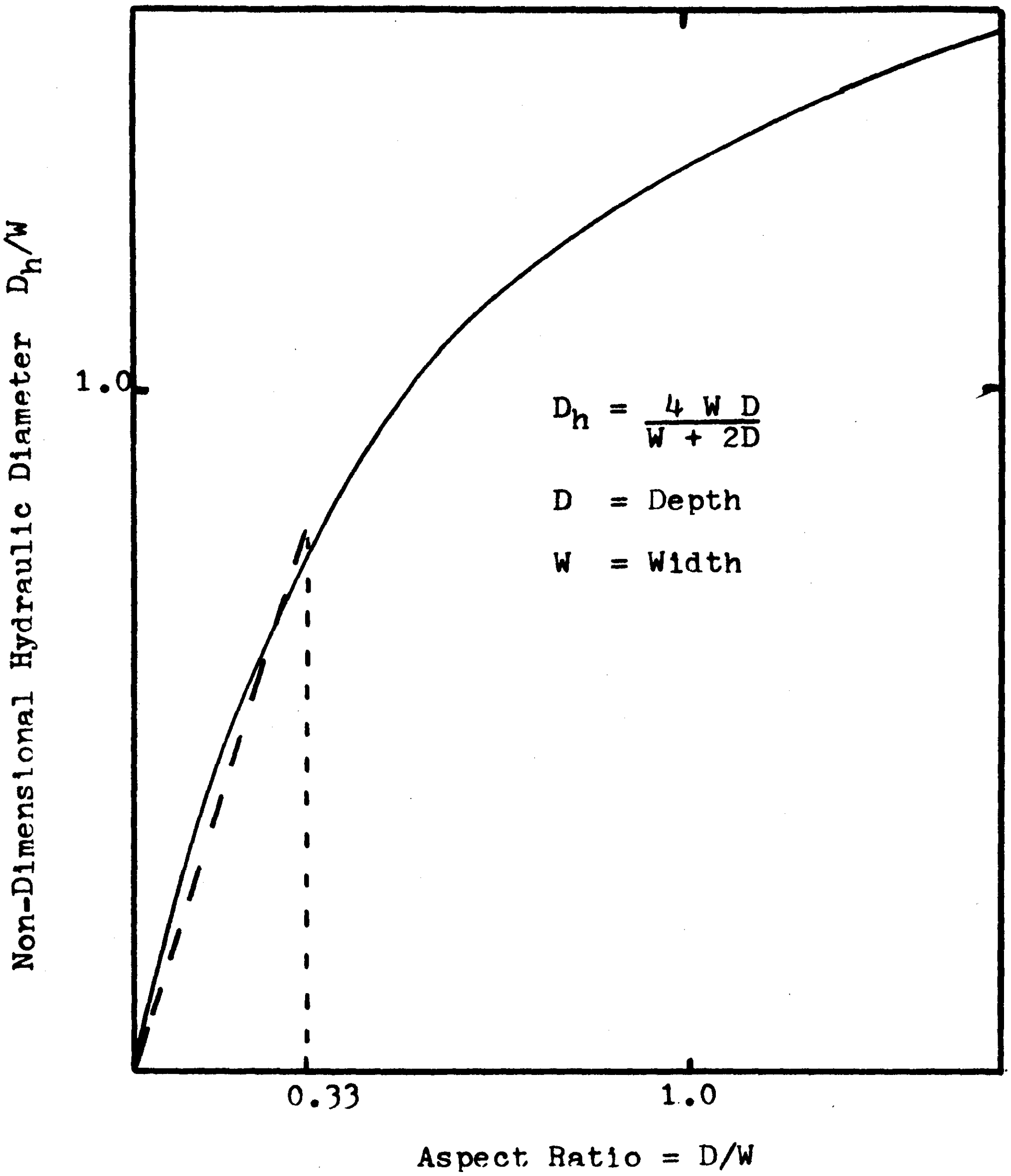


Fig. 4.6 Hydraulic Diameter as a Function of Aspect Ratio

Intuitively, this ability to distort the geometry of the model to obtain RΦI similarity is justifiable. For small values of the aspect ratio it seems reasonable that the total effect of friction is largely independent of the channel width. The role of geometric similarity in modeling the TSP prototype is that of simulating the boundary effects (friction effects). The above discussion simply indicates that these boundary effects are, over a limited range, dependent only on depth. As a practical matter, the limiting value of the aspect ratio will be taken to be 0.25.

Finally, it should be noted that similarity of surface heat loss is not important in TSP modeling because of the short operation period of the prototype in relation to the time required to produce significant temperature changes via heat transfer to the environment.

In meeting all the above similarity requirements it is often desirable to use a distorted temperature difference. An increase in the temperature difference requires an offsetting increase in the mean plug-flow velocity to sustain densimetric Froude law similarity. The increase in velocity is advantageous in maintaining a high (turbulent) Reynolds number. However, the distortion of the temperature difference results in an inability to meet exactly normal Froude law similarity. A temperature distortion of four leads to a Froude number change by a factor of two. Nonetheless this alteration of the Froude number value does not appear to be significant in the present case since the magnitude of the Froude number is small (approximately 10^{-2}). A small Froude number is characteristic of weak or tranquil flow and density-induced flows are most strongly dependent in their character on the densimetric Froude number. Thus, it can be reasoned that for variations of the Froude number of the order of magnitude stated above the resultant effect on the overall behavior of the density induced flow is small. This deduction has been verified by observations of the density induced flow behavior at two different Froude numbers.

4.2.1.2. Experimental Apparatus and Observational Techniques

The experimental model of the prototype TSP has been constructed at the Ralph M. Parsons Laboratory for Water Resources and Hydrodynamics at M.I.T. The model fabrication consisted mainly of the modification of an existing flume. The flume, shown in Figs. 4.13 and 4.14, is constructed of 1/2" glass mounted on an aluminum frame. The nominal flume dimensions are 8" deep, 18" wide and 64 feet long. The piping and storage tanks which were added to the facility to enable the desired inlet discharge and withdrawal flowrates to be maintained in the model are shown schematically in Fig. 4.7.

The inlet flow holding tank and the flume are filled with water at the desired temperatures prior to the operation of the experiment by mixing streams of hot and cold

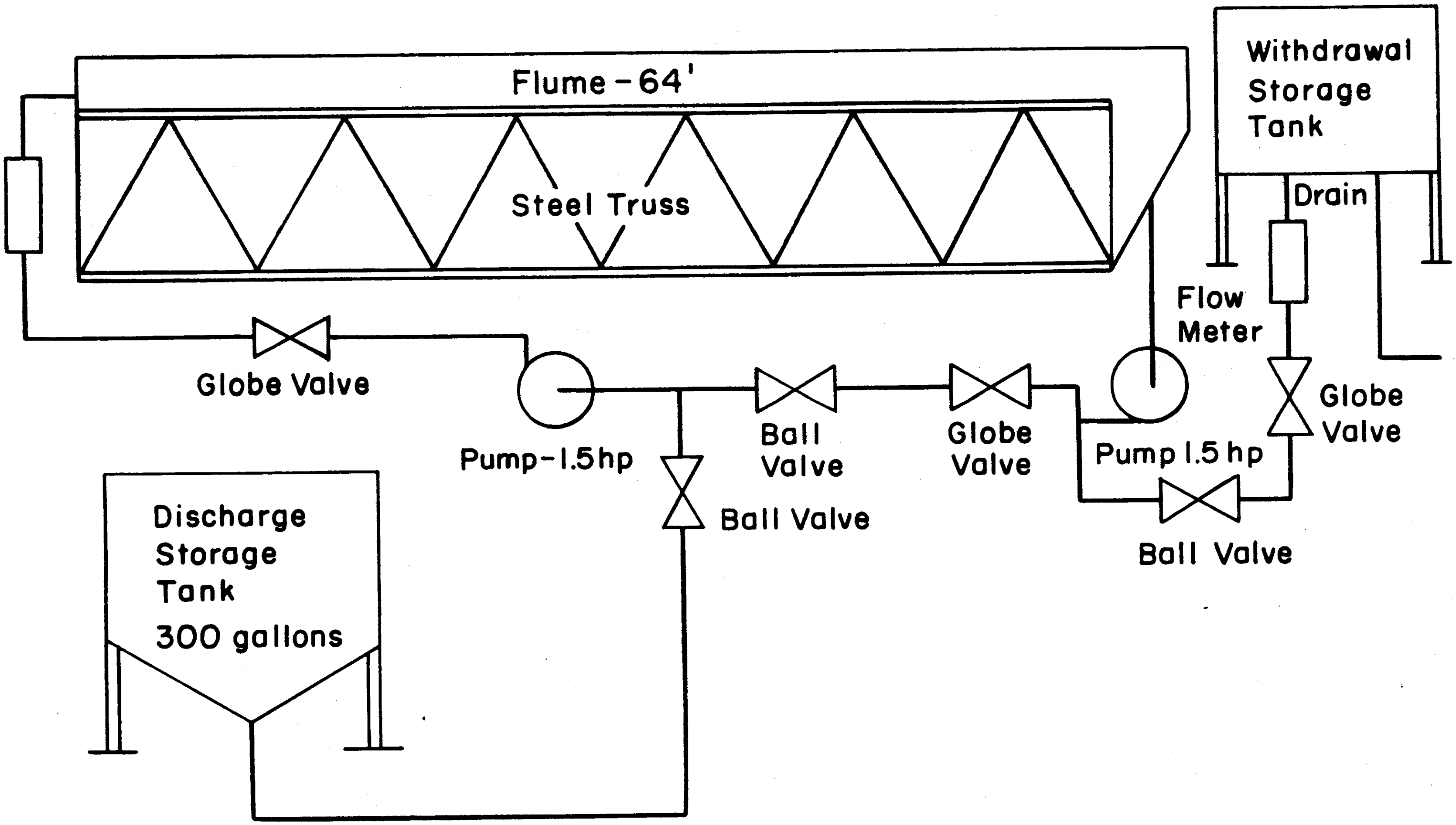


Fig. 4.7 Schematic of Thermal Storage Pond Model Experiment

tap water (controlled by globe valves), and by monitoring the mixed stream temperature. Careful monitoring of the filling stream temperature allows the establishment of initial discharge tank and flume temperatures within 1 ^{o}F of the desired values.

As indicated in Fig. 4.7, a flume recirculation line was installed to allow the initial pond inventory to be continuously cycled through the flume in order to diminish any temperature differences which may develop just prior to the actual experiment. Also the recirculation flow allows the plug flow velocity field to be established in the flume prior to the introduction of hot inlet water. The inlet and withdrawal flow rates are measured by rotor-meters with maximum capacities of 67 gpm. The discharge structure consists of four 2 in. threaded nozzles which could be reduced in size by application of the appropriate reducing fittings. Various withdrawal schemes were utilized during different experimental runs depending on the current modeling requirements.

The following observational techniques were tested for recording the thermal-hydraulic behavior of the model:

1) verbal recording of a description of the progress of the dyed density-front as a function of time using a cassette recorder,
2) photographic recording of the dyed density-front configuration at known instants,

3) videotape recording of the dyed density-front displacement, and
4) measurement of vertical temperature profiles with thermistor probes at fixed stations.

In general, the photographic and video recording methods have not been found to be superior to the simple verbal recording of the motion since the characteristic velocities are small (less than 0.20 ft/sec). Temperature measurements were found to be necessary in determining the effects of mixing in the pond upon the outlet flow temperature. The temperature dependent resistance of the fast response (70 millisec) thermistor probes was measured using a compensating Wheatstone bridge circuit coupled to a Sanborn amplifier and strip chart recorder.

The flow visualization dye, FD+C Blue Food Color #1, was used to color the entire contents of the inlet flow holding tank just prior to the operation of the experiment. The addition of the dye has an inconsequential effect on the density difference between the inlet flow and the initial flume inventory. The required dye concentration is 0.5 grams/gallon.

4.2.2 Analytical Model

4.2.2.1 Approach to the Problem

An approximate analytical model of the behavior of a density front in a horizontal plug-flow pond has been developed based on a control-volume momentum conservation equation. The resultant equation contains one empirical constant, the value of which is determined by experiment. Essentially the model utilizes the superposition of the density-induced "lock-exchange" flow field upon the unperturbed homogenous fluid velocity profile in the pond. It also includes interfacial and bottom friction effects. The formulation of the control-volume equation for the case of interest is similar to the treatment by Abraham and Vreugdenhil [A2] of the exchange flow problem.

4.2.2.2 Derivation of Density-Front Propagation Equation

The equation of motion of the leading and trailing edges of the density front in a horizontal plug-flow pond is obtained from a force balance on the double-lumped moving control volume shown in Fig. 4.8. The forces affecting the motion of the leading and trailing edges of the density front with respect to the mean plug-flow velocity V_p are:

1) F_p = pressure force due to $\rho_2 > \rho_1$,
2) F_i = interfacial friction force,
3) F_b = bottom friction force, and
4) F_m = inertial force, where

F_i, F_b, and F_m all act to resist the pressure force F_p. Thus, one obtains the result

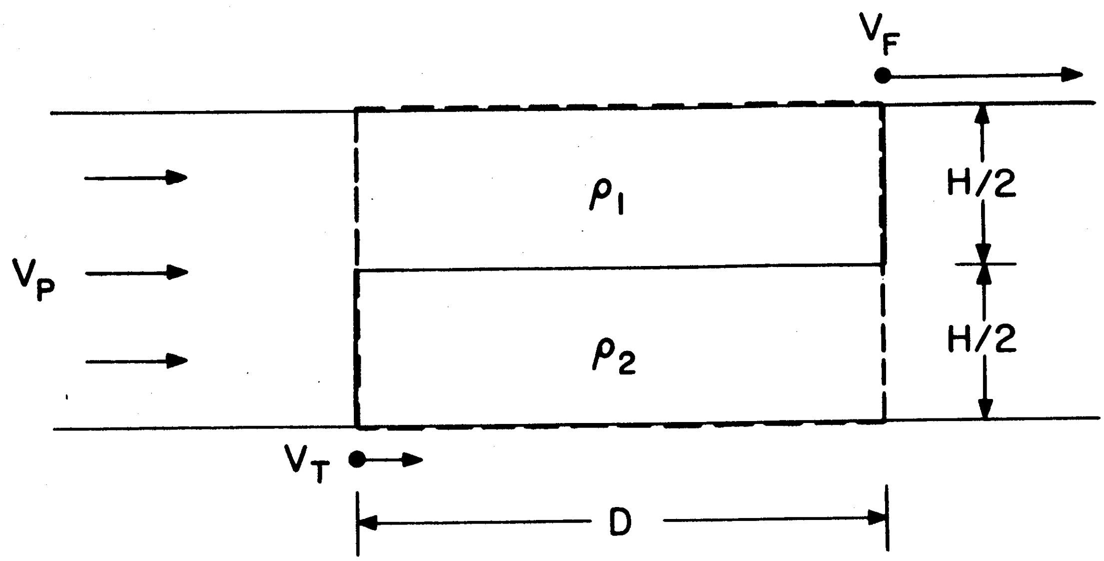


Fig. 4.8 Density-Induced Flow Control Volume

$$F_p = F_i + F_b + F_m \quad . \tag{4.26}$$

The approximate analytic expression for V_f, the leading edge velocity, and V_T, the trailing edge, is obtained by expressing Eq. (4.26) in terms of $\frac{dD}{dT}$, the rate of growth of the horizontal interfacial length. Thus, under the assumption of equal depths of the two overlying layers, and from the requirement of continuity one obtains the result

$$V_f - V_p = V_p - V_T = \frac{dD}{dt}/2 \quad . \tag{4.27}$$

The total static pressure due to density difference acting horizontally across any point on the interface is

$$(P_2 - P_1)_h = h(\rho_1 - \rho_2)\,\frac{g}{g_c} \tag{4.28}$$

where h is the depth into the pond.

Since this pressure difference is a linear function of depth the total pressure acting across the interface is given as

$$F_p = \frac{(P_2 - P_1)_H - 0.0}{2}\;A \quad , \tag{4.29}$$

where $(P_2 - P_1)$ is the pressure difference at the bottom of the pond, and A is the cross-sectional area of the pond, ($A = L\cdot H$, where L = width). The average pressure can be envisioned to act (see Fig. 4.8) to the right on the top fluid segment in the control volume and to the left on the bottom fluid segment. Substitution of Eq. (4.28) into Eq. (4.29) gives the result

$$F_p = \frac{\beta}{2} LH^2 \Delta\rho \frac{g}{g_c} \quad , \tag{4.30}$$

where the parameter β has been introduced to account for the fact that the actual pressure difference may be different from that deduced from the simple model.

The friction force terms in Eq. (4.26) can be expressed as

$$\begin{aligned} F_i &= \tau_i A_i, \text{ and} \\ F_b &= \tau_b A_b \quad , \end{aligned} \tag{4.31}$$

where $A_i \simeq A_b = D \cdot L$, and τ_i and τ_b are the interfacial and bottom shear forces, respectively. The interfacial shear is given by

$$\tau_i = \frac{f_i}{8} \rho V_R^2 = \frac{f_i}{8} \rho \left(\frac{dD}{dt} \right)^2 \quad , \tag{4.32}$$

where f_i is the interfacial friction factor. The variation of f_i as deduced by Abraham and Eysink [A1] is shown in Fig. 4.9 as a function of the Reynolds number.

The bottom shear force will be effected by the magnitude of the plug-flow velocity. The resultant bottom shear force is

$$\tau_b = \frac{f_b}{8} \rho \left(\left(\frac{dD}{dt}\right) \frac{V_p}{2} - \frac{1}{4}\left(\frac{dD}{dt}\right)^2 \right) \quad . \tag{4.33}$$

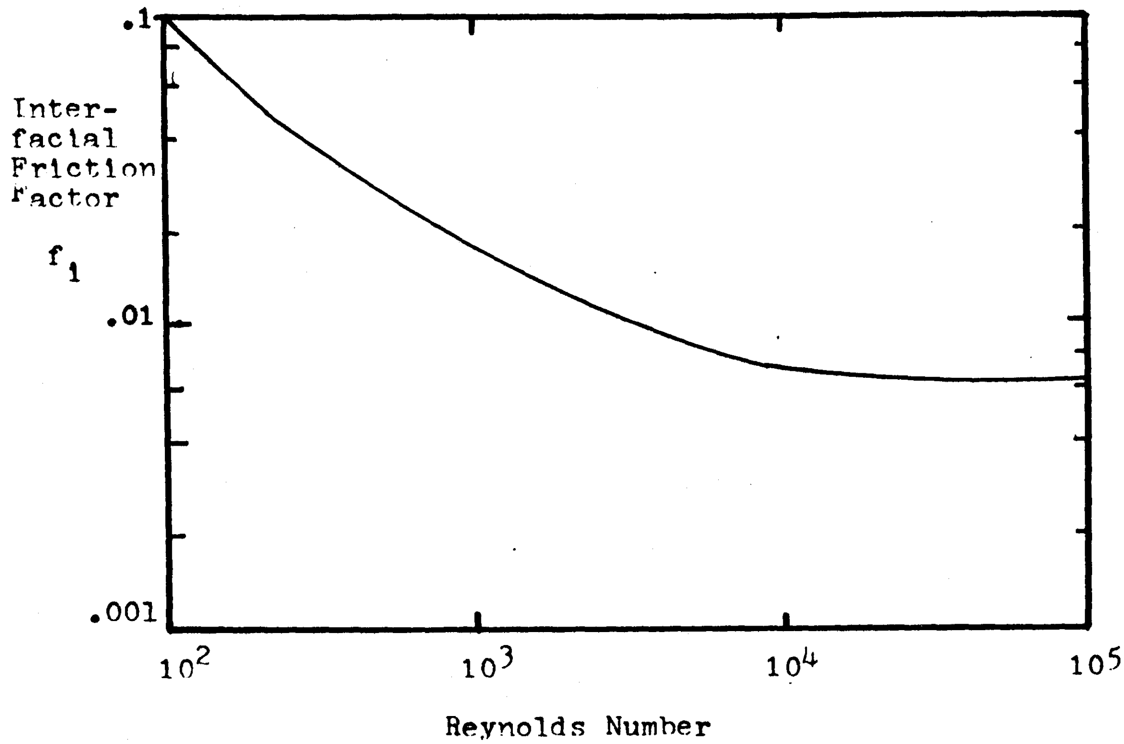


Fig. 4.9 Variation of Interfacial Friction Factor with Reynolds Number (A1)

The inertial force acting on the control volume is

$$F_m = \frac{d(M\overline{V})}{g_c dt} \quad , \tag{4.34}$$

or

$$F_m = \frac{M}{g_c} \left(\frac{d\overline{V}}{dt}\right) + \frac{\overline{V}}{g_c} \frac{dM}{dt} \ , \tag{4.35}$$

when $M = DLH\rho$, and $\overline{V}$ is the average relative velocity of the control volume segment with respect to V_p. Since the acceleration of the control volume is expected to be small the term $M \cdot \left(\frac{d\overline{V}}{dt}\right) / g_c$ may be neglected. Noting that

$$\frac{dM}{dt} = \frac{dD}{dt} \ LH\rho \quad , \tag{4.36}$$

and

$$\overline{V} = \frac{1}{2} \ \left(\frac{dD}{dt}\right) \quad , \tag{4.37}$$

we obtain the result

$$F_m = \frac{1}{2g_c} \left(\frac{dD}{dt}\right)^2 \ LH\rho \tag{4.38}$$

In deriving the terms F_i, F_b and F_m it has been assumed that the density front velocity similarity profile is that shown in Fig. 4.8. Thus, to account for the fact that the actual velocity profile may be different from that assumed the parameter γ is introduced as follows:

$$\left(\frac{dD}{dt}\right)_{Actual} = \gamma \left(\frac{dD}{dt}\right) \qquad (4.39)$$

where $\left(\frac{dD}{dt}\right)_{Actual}$ is the actual rate of growth of the distance D defined in Fig. 4.8. The entire equation of motion may now be written as

$$\frac{\beta}{2} LH^2 \Delta\rho \frac{g}{g_c} = \frac{f_i}{g_c 8} \rho\gamma^2 \left(\frac{dD}{dt}\right)^2 D \cdot L + \frac{f_b}{g_c 8} \rho\left(\frac{\gamma}{2}\left(\frac{dD}{dt}\right)V_p - \frac{\gamma^2}{4}\left(\frac{dD}{dt}\right)^2\right) D \cdot L + \frac{\gamma^2}{2g_c}\left(\frac{dD}{dt}\right)^2 L H \rho \quad . \qquad (4.40)$$

The above equation is quadratic in $\frac{dD}{dt}$, with the solution being

$$\frac{dD}{dt} = \frac{\frac{-\gamma^2{}_b \rho V_p D}{8} \pm \sqrt{(\gamma^2{}_b \rho V_p D)^2 + 4\beta H^2 \Delta\rho g\left(\frac{\gamma^2 f_i}{4}\rho D - \frac{\gamma^2 f_b D}{16} + \gamma^2 H\right.}}{2\left(\gamma^2 \frac{f_i}{4}\rho D - \frac{\gamma^2 f_b \rho D}{16} + \gamma^2 H\rho\right)} \qquad (4.41)$$

Now if typical values of the various parameters are used to evaluate Eq. (4.41) ($\gamma = \beta = 1$) it becomes apparent that the effect of V_p, for the range of Froude numbers of interest, is small. Thus, the terms containing V_p can be neglected in Eq. (4.41). Equation (4.41) is then simplified to the form

$$\frac{dD}{dt} = \left(\frac{\beta H^2 \Delta\rho g}{\gamma^2 \left(H\rho + \left(\frac{f_i + f_b/4}{4}\right) \rho D \right)} \right)^{1/2} , \qquad (4.42)$$

or

$$\frac{dD}{dt} = \alpha \left(\frac{H^2 \Delta\rho g}{H\rho + \left(\frac{f_i + f_b/4}{4}\right) \rho D} \right)^{1/2} , \qquad (4.43)$$

where $\alpha = \beta/\gamma^2$.

Thus, the expression for the rate of growth of the horizontal projection of the interface is given by an equation with a single free parameter to be determined by experiment. The total velocity of the leading edge of the density front is now expressed as

$$V_F = C_o V_p + \frac{\alpha}{2} \left(\frac{H^2 \Delta\rho g}{H\rho + \left(\frac{f_i + f_b/4}{4}\right) \rho D} \right)^{1/2} \qquad (4.44)$$

The displacement of the leading edge at time t from D = 0 at t = 0 is

$$X_F = C_o V_p t + \frac{1}{2} D(t), \qquad (4.45)$$

where D(t) is obtained by integrating $\frac{dD}{dt}$ (Eq. 4.43) from time-zero to t. The resulting expression for D(t) is

$$D(t) = \frac{4H^2\Delta\rho g}{\rho(f_i+f_b/4)}\left(\left(\frac{\rho}{H\Delta\rho g}\right)^{3/2} + \left(\frac{3\rho(f_i+f_b/4)}{8H^2\Delta\rho g}\right)(\alpha T)\right)^{2/3}$$

$$- \frac{4H}{(f_i+f_b/4)} \quad . \qquad (4.46)$$

The position of the trailing edge at time t would be

$$X_T = C_oV_pt - \frac{1}{2}D(t). \qquad (4.47)$$

It should be realized that since no specific assumption has been made regarding the exact density-induced velocity distribution the above equation only predicts the position of the leading edge or trailing edge of the density front. The parameter C_o corrects for the fact that the channel velocity at the height of the leading edge (or trailing edge) is, in general, different from the mean plug-flow velocity. The value of C_o can be determined, by assuming that the unperturbed velocity profile in the pond is given by the relationship [R4]

$$V(y) = V_s\left(\frac{y}{y_o}\right)^{1/6} , \qquad (4.49)$$

where V_s is the surface velocity. The value of C_o at any particular height y in the pond is seen to be

$$C_o(y) = \frac{V_s\left(\frac{y}{y_o}\right)^{1/6}}{V_p} \tag{4.50}$$

At $y = 0$, $C_o = 0$ and at $y = y_o$, $C_o = 1.15$. The fact that at $y = 0$, $C_o = 0$ poses somewhat of a dilemma since if some simple density induced flow similarity profile is assumed (linear for example) Eq. (4.48) predicts a negative trailing edge velocity for the case of hot water discharged into an initially cold pond. However, keeping in mind that the present interest is primarily in predicting the position of the leading edge, the position of the trailing edge is (for the case of the hot discharge into the cold pond) assumed to be given by

$$V_T = (1.0 - (C_o(y_o) - 1.0))\cdot V_p + \frac{D(t)}{2} \tag{4.51}$$

This relationship satisfies continuity (i.e. Eq. 4.45)) if a linear density front profile is assumed. As is discussed in the next section this assumption appears to be adequate for describing the bulk density flow behavior outside the boundary layer at the bottom of the pond.

The case of the advancing cold front presents a different problem. In this case the maximum density-induced forward velocity should occur at or near the bottom. Certainly it would not occur at the bottom wall since the wall no-slip condition always holds.

In any event, the maximum advancing cold front velocity would be less than the maximum advancing hot-front velocity since in the latter case the plug-flow velocity correction factor C_o is greater than 1 while in the former case it would be less than 1. The advancing cold front case thus represents a more desirable situation than with the advancing hot front in terms of achieving a near vertical density front velocity profile. In view of this fact no effort has been made to quantify the behavior of the advancing cold front other than to say that for TSP design purposes the advancing hot front is the more restrictive design condition. This conclusion has been verified by experiment.

4.2.2.3 Summary

An expression for prediction of the position of the leading edge of an advancing two dimensional "hot" density front in shallow water channels has been derived and is repeated here:

$$V_F = C_o V_p + \frac{\alpha}{2}\left(\frac{H^2 \Delta\rho g}{H\rho + \left(\frac{f_i + f_b/4}{4}\right)\rho D}\right)^{1/2} \tag{4.52}$$

where $C_o \simeq 1.15$. For small values of D, Eq. (4.52) reduces to

$$V_F = V_P \left(C_o + \frac{\alpha\sqrt{2}}{2F_D}\right) \quad , \tag{4.53}$$

where F_D is the pond densimetric Froude number

$$F_D = \frac{V_p}{(g\frac{\Delta\rho}{\rho}\frac{H}{2})^{1/2}} \tag{4.54}$$

The velocity of the trailing edge is approximated by the equation:

$$V_T = (2.0-C_o)V_p - \frac{\alpha}{2}\left(\frac{H^2\Delta\rho g}{H\rho + \frac{f_i+f_b/4}{4}\rho D}\right)^{1/2}, \tag{4.55}$$

where again $C_o \simeq 1.15$. Equations (4.52) and (4.55) are valid only for flows $F_D \gtrsim 1.0$ since at lower F_D values Eq. (4.55) will predict a negative trailing edge velocity. No quantitative analysis of the advancing cold front has been found to be necessary since the advancing hot front represents the more conservative TSP design situation.

4.2.3. Results of Initial Design Concept Evaluation

Three prorotype TSPs have been designed to span the range of design densimetric Froude numbers from 0.5 to 1.5, as is shown in Table 4.1. The densimetric Froude number value of 1.5 can be considered to be an upper limit for practical designs. Three model designs which simulate these prototype designs, and which can be realized in the available flume are summarized in Table 4.2. All three designs result in unsatisfactory thermal hydraulic performance of the pond.

TABLE 4.1

Thermal Storage Pond Prototype Designs

	#1	#2	#3
1. Densimetric Froude Number	1.5	1.0	.50
2. Depth (feet)	15	15	15
3. Width (feet)	72	109	218
4. Aspect ratio (depth/width)	0.20	0.14	0.07
5. Mean velocity (ft/sec)	0.93	0.61	0.30
6. f_o (L/D)	0.048	0.072	0.144
7. Reynolds Number	4.3×10^6	2.8×10^6	1.5×10^6
8. Discharge	Highly unstable submerged multiport type		
9. Temperature Difference (°F)	10	10	10

TABLE 4.2

Thermal Storage Pond Model Designs

	#1	#2	#3
1. Densimetric Froude Number	1.5	1.0	.50
2. Depth (inches)	4.5	4.5	4.0
3. Width (inches)	18	18	18
4. Aspect ratio	0.25	0.25	0.22
5. Mean Velocity (ft/sec)	0.28	0.19	0.088
6. Total flow (GPM)	70.4	47.2	19.5
7. Reynolds Number	30,000	20,100	8,830
8. Scale factor	48.4	72.7	147.0
9. Temperature difference (°F)	40	40	40
10. Vertical Distortion	1.21	1.80	3.33
11. $\frac{f_o(l/D)\text{model}}{f_o(l/D)\text{prototype}}$	1.83	1.38	0.94
12. Discharge	highly unstable submerged multiport		

4.2.3.1 Qualitative Discussion of Thermal-Hydraulic Behavior

All three of the TSP designs summarized in Table 4.2 have significant density-induced flows which partially reduce the pond thermal capacitance. As is shown in Fig. 4.15 the highly unstable inlet flow jets were successful in establishing initially a nearly vertical density front. However, as the density front propagates a short distance downstream from the inlet mixing region as shown in Fig. 4.16 significant vertical stratification begins to develop. In fact, except for the case of a densimetric Froude number of 1.5, the trailing edge of the density front was not observed to travel a significant distance beyond the inlet mixing region.

Figure 4.16 also indicates that a short time after the initiation of the inlet flow, the overlying hot wedge is somewhat diluted as a result of inlet flow mixing. This small dilution nevertheless is rapidly overcome by the continuing intrusion of the unmixed inlet flow, and a distinct density interface develops rapidly.

Accurate visual determination of the position of the trailing edge of the upper fluid layer in all cases was difficult since in none of the experiments was the inlet flow able to displace rapidly or to mix with the initial cold water in the boundary layer at the bottom of the channel. Figure 4.10a shows qualitatively the profile of the density front for F_D greater than about 1.00. The interface is

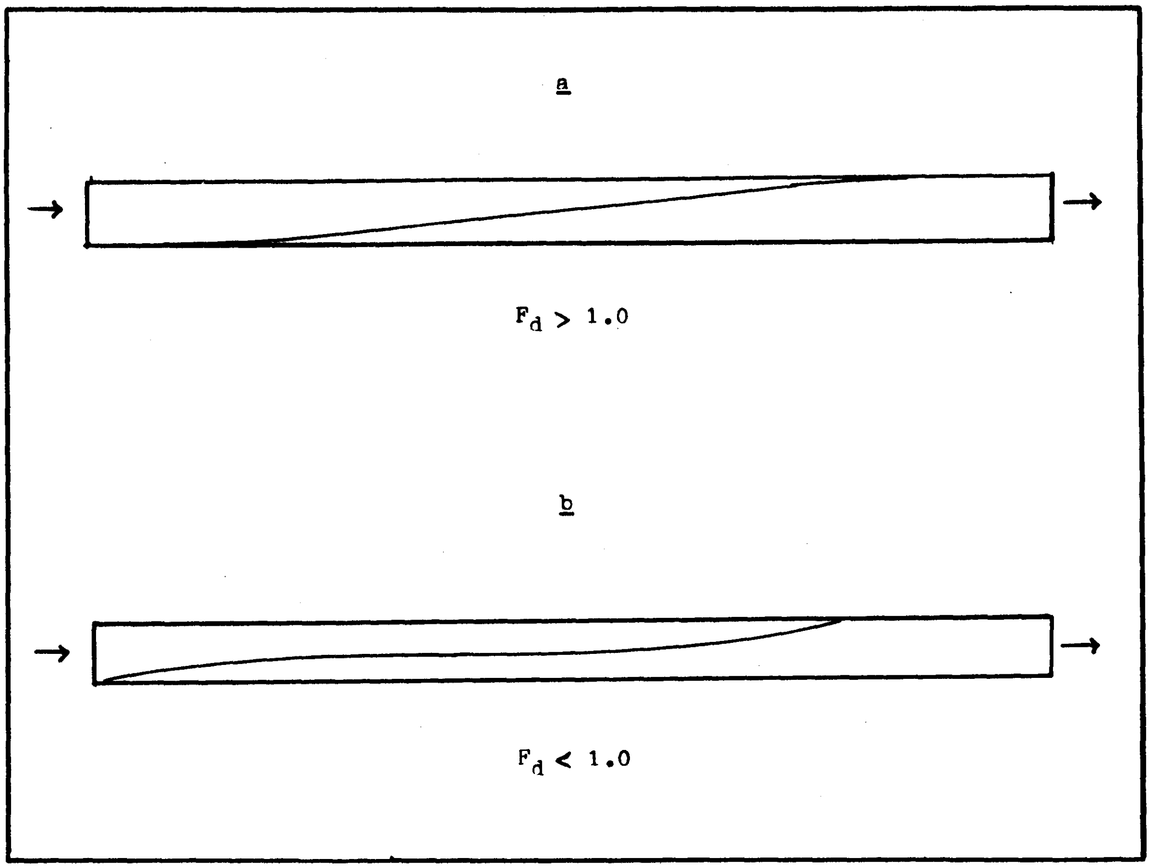


Fig. 4.10

Illustration of Density Front Profile -

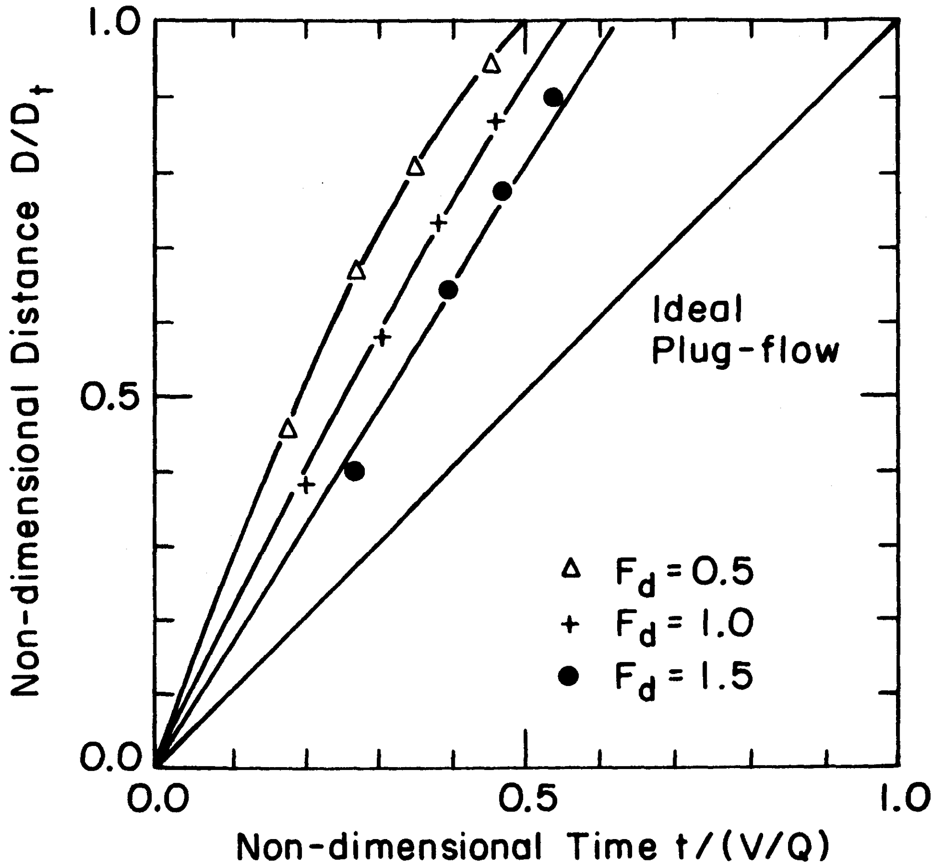
Advancing Hot Front

nearly linear except at the leading and trailing edges. At lower densimetric Froude numbers density front profiles developed as shown approximately in Fig. 4.10b.

In addition to the three experimental runs performed at densimetric Froude numbers of 0.5, 1.0, and 1.5 several runs were made at higher densimetric Froude number values to see if any gross changes in the flow behavior would occur. In all cases the flow behavior was observed to remain qualitatively unchanged.

4.2.3.2 Quantitative Evaluation of Thermal-Hydraulic Behavior

The positions of the leading edge of the advancing density front as a function of time for each of the TSP designs of Table 4.2 are shown in Fig. 4.11. In each case the position is plotted in a non-dimensional manner as a function of the fraction of the plug-flow residence time V/Q, which had elapsed. The design parameters are equal for all of the ponds except for the densimetric Froude number values. As Fig. 4.11 indicates, even a TSP with a Froude number of 1.5 (which corresponds to a channel only 72 feet wide and 15 feet deep) is observed to short-circuit long before the plug-flow residence time has elapsed. Clearly, this problem becomes worse at even lower design densimetric Froude number values. Once the density front has reached the withdrawal region, the pond would be effectively short-circuited, since as is discussed in section 4.1.3, selective withdrawal of the remaining cold layer is not possible.



D = Actual Position of Leading Edge of Density Front

D_t = Total Lenght of Pond

Fig. 4.11 Propogation of Density Front as a Function of TSP Design Densimetric Froude Number

A total of 8 experimental runs were made to determine the value of the empirical constant α of Eq. (4.52). In each experimental run the position of the advancing hot front was observed visually and recorded as a function of time. Some of these runs reflected variations of the design parameter values of the three initial model designs summarized in Table 4.2. Table 4.3 summarizes the values of the relevant parameters for each run, and all of the resulting data points are plotted in Fig. 4.12. The coordinate axes in Fig. 4.12 are chosen such that the slope of the straight line drawn through the data points is equal to the empirical coefficient α. The abscissa variable is time, while the ordinate variable is the result of integration of Eq. (4.52), followed by solution for αT. The ordinate variable is

$$y^* = \frac{1}{C} \left(\frac{D+E}{A}\right)^{3/2} - \frac{B}{C} \quad , \qquad (4.56)$$

where D = observed position of the leading edge of the advancing hot front at time t,

$$A = \frac{4H^2 \Delta\rho g}{\rho(f_i + f_b/4)} \quad ,$$

$$B = \left(\frac{\rho}{H \Delta\rho g}\right)^{3/2} \quad ,$$

$$C = \frac{3}{8} \; \frac{\rho}{\Delta\rho} \; \left(\frac{f_i + f_b/4}{H^2 g}\right) \quad , \text{ and}$$

$$E = \frac{4H}{f_i + f_b/4} \quad .$$

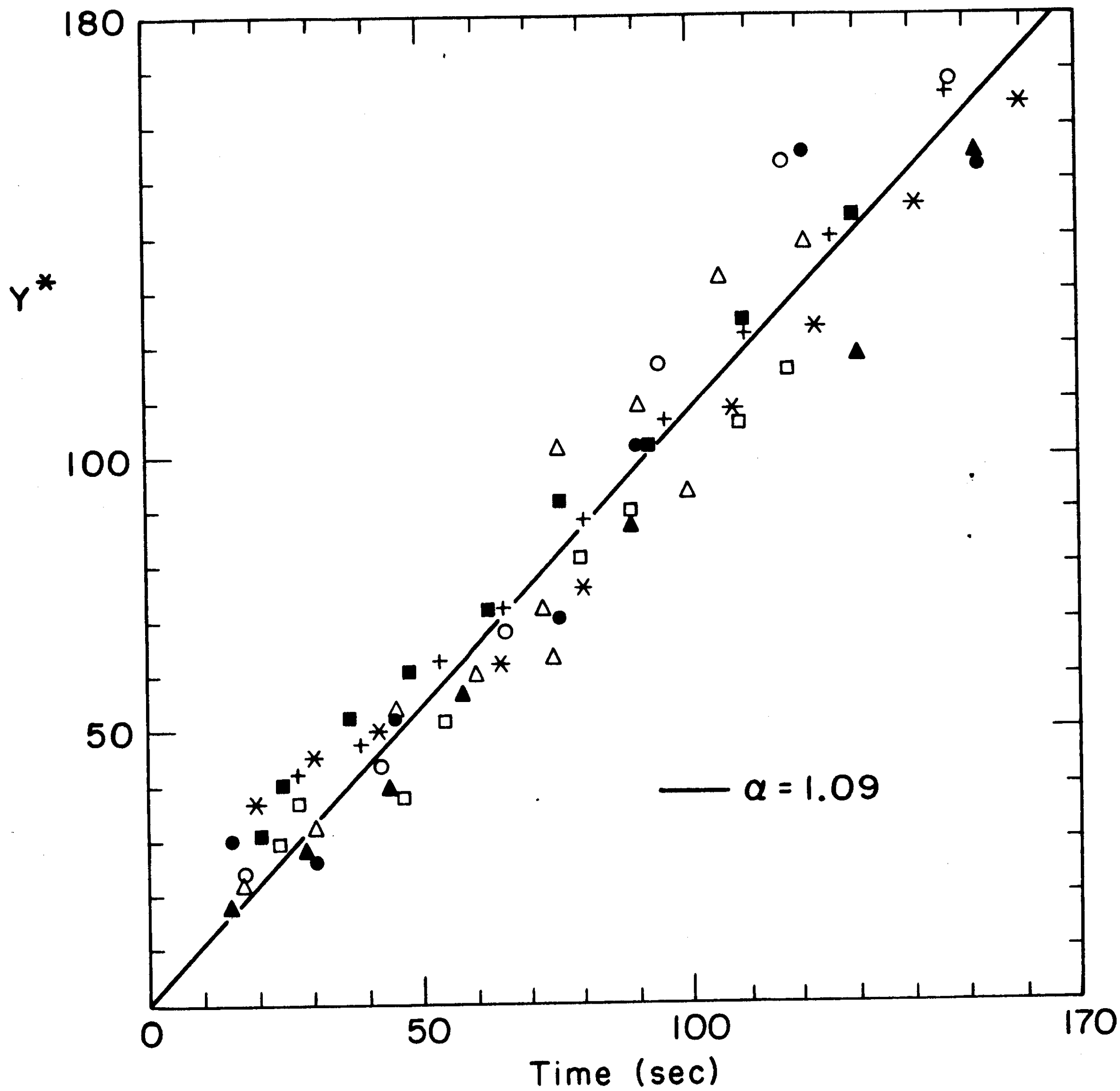


Fig. 4.12 Data for α Determination

TABLE 4.3

Summary of Experiments for α Determination

No.	F_D	ΔT (°F)	V_p (ft/sec)	Depth (Inches)	α
1	1.46	37	0.26	4.5	1.16
2	2.99	9	0.26	4.5	1.20
3	1.30	32	0.25	4.75	0.93
4	1.48	36	0.26	4.5	1.00
5	2.58	10	0.25	4.75	1.11
6	1.05	37	.19	4.5	1.20
7	1.28	40	.16	2.0	1.11
8	.85	40	.113	2.2	1.03

Fig. 4.13 TSP Experimental Model - Side View

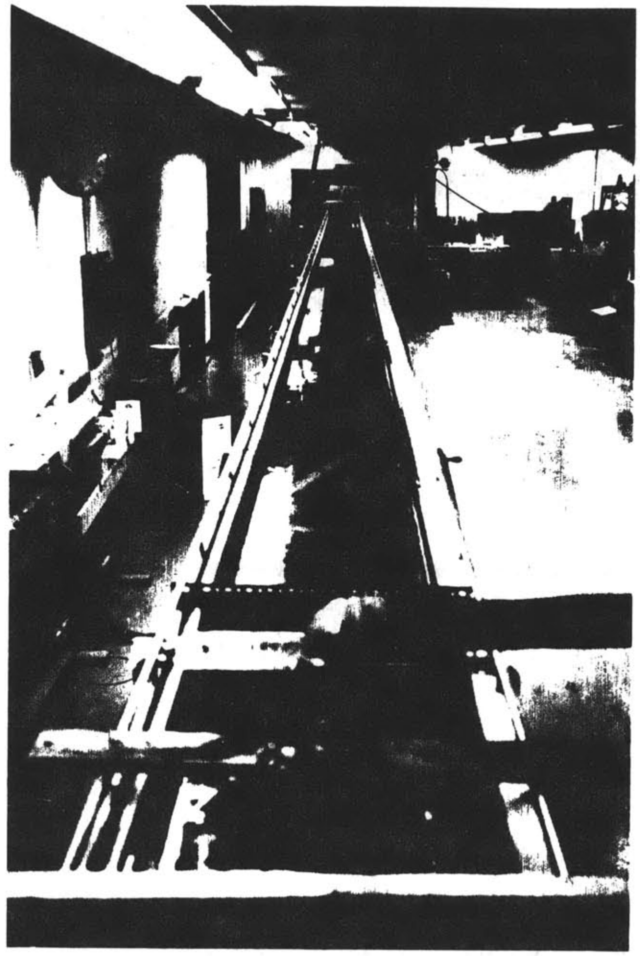


Fig. 4.14 TSP Experimental Model - Top View

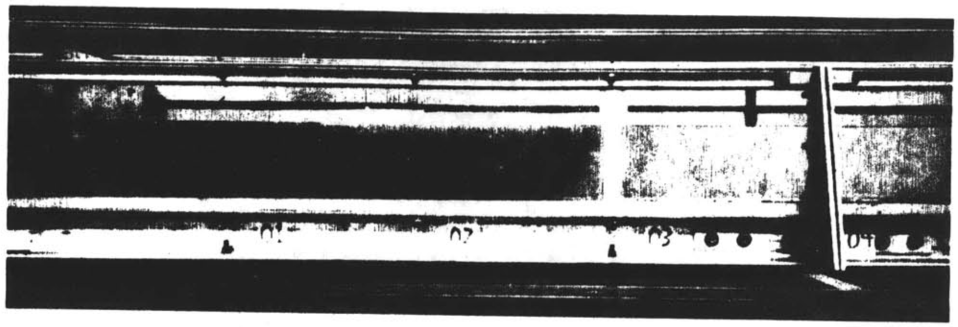


Fig. 4.15 Initial Density Interface

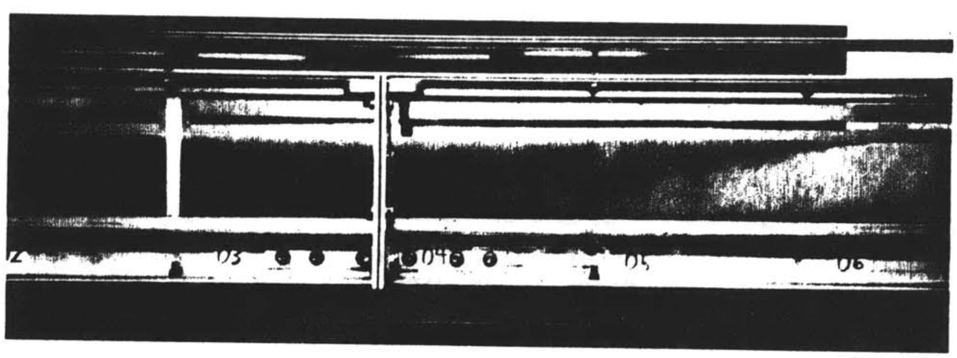


Fig. 4.16 Advancing Density Interface

An approximate best-fit (eyeball) straight line was drawn through each set of data points, and a value of α determined. These values are summarized in Table 4.3 along with the average value of 1.09 for all runs. Using the value of α the data points for all 8 experimental runs can be predicted with an average error of 4.0%. The average error for the run with the poorest agreement is 7.2%. Note that most runs show substantially greater downstream propagation of the density front at small times than is predicted by Eq. (4.52). This deviation is expected since the effective volume of the injected hot water (and hence the apparent downstream propagation of the front) is increased due to the inlet flow mixing with the cold pond inventory.

4.3 EVALUATION OF DESIGN MODIFICATIONS

4.3.1 Survey of Design Modification Options

As Section 4.2.3.2 has indicated, an efficient and economical horizontal plug-flow TSP cannot be designed as envisioned in the initial design concept. If the development of the TSP design on the basis of a horizontal plug-flow pond is to be pursued then some measures must be taken to retard the advancing density front, and the resultant short circuiting. Thus, some simple modifications of the initial design concept have been investigated by testing the effects of design changes in the experimental model.

In evaluating different possible designs all the previously discussed constraints must be considered. No exhaustive examination of all possible design modifications has been attempted. The development effort has been directed mainly towards achieving experimentally a workable design with a simple and hopefully economical modification.

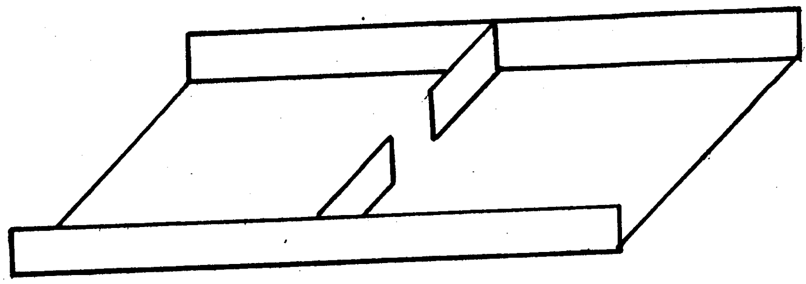
There appear to be two different means of retarding the density-induced flow in a TSP. The first is simply the correct placement of barriers in the pond such that the advance of the leading edge of the front is retarded. The second is the placement of barriers in the pond such that internal mixing jets are created which have the effect of reducing the local density differences, and consequently of retarding any density-induced flow. Actually, any barrier or construction placed in the pond will have, to varying degrees, both of the above effects. Any constriction produces a jet as a result of the locally increased flow velocity, and a barrier will retard the leading edge of either a hot or cold advancing front. Thus, the design task has been to select empirically an appropriate pond constriction design and to determine the required number of such constrictions in order to yield the desired result of adequately diminishing the density-induced flow short-circuiting.

4.3.2 Additional Modeling Considerations

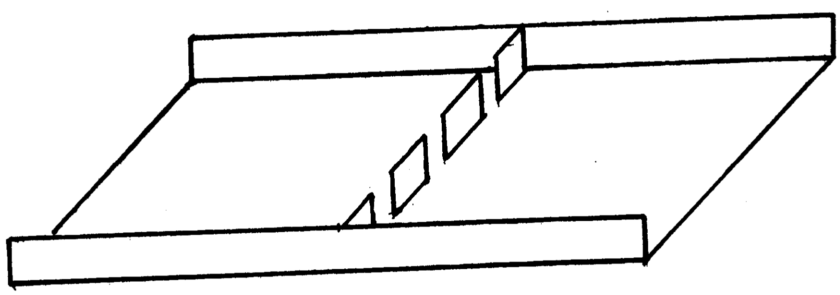
In modifying the basic flow processes occuring in the model TSP due to changes in the model geometry one needs to insure that the identical geometry changes in the prototype will yield the same effect on the prototype flow behavior. Of particular importance are the effects of jets created by constrictions in the flow channel. As noted by Jirka [J1], modeling of flows with jets requires undistorted geometry. This is because the width of a turbulent jet flows as a linear function of the downstream distance. The rate of growth of a jet is independent of the Reynolds number [W3]. All experiments reported herein have a distorted vertical dimension to correct for the friction factor dependence on Reynolds number (the distortion also helps in maintaining high Reynolds number values).

Consequently, any jet produced in the model which grows vertically will not be modeled correctly. However, the practical consequence of this modelling flaw does not appear to be important. This is because the type of constriction which appears to offer the best combination of pond performance enhancement, and ease of construction, is a symmetrical full depth barrier as shown in Fig. 4.17a and 4.17b. Such constrictions produce laterally two dimensional jets which are largely unaffected by the vertical distortion. Of course the bottom does interact with such jets as a result of the requirement of a zero bottom wall velocity (i.e. friction) and thus they are not entirely independent of the vertical dimension.

Fig. 4.17 Horizontal Barriers



(a)



(b)

4.3.3 Comparative Performance of Design Modifications

4.3.3.1 Performance of Barriered Ponds

4.3.3.1a Horizontal Versus Vertical Barriers

Initially a determination of the relative merits of horizontal versus vertical barriers was made. A horizontal barrier is of the type shown in Fig. 4.17a and 4.17b while a vertical barrier would have the general characteristic of blocking the entire width of the pond over a part of the water depth as shown in Fig. 4.18.

As is shown in Fig. 4.19, the horizontal barrier concept was found to be superior. Figure 4.19 compares the thermal capacitance performance of identical ponds (except for barrier geometries) on the basis of percentage of the theoretical cooling potential recovered from the pond as a function of time. The time variable is plotted in a non-dimensional manner by dividing by the plug-vlow residence time V/Q where V is the pond volume and Q equals the flow rate. Mathematically, the percentage of the theoretical cooling potential recovered, R(t). is defined as follows:

$$R(t) = 100\% \int_0^t \frac{T_i - T_{oa}}{T_i - T_{ot}} \cdot \frac{d\tau}{(V/Q)} \qquad (4.57)$$

where T_i = pond inlet temperature,

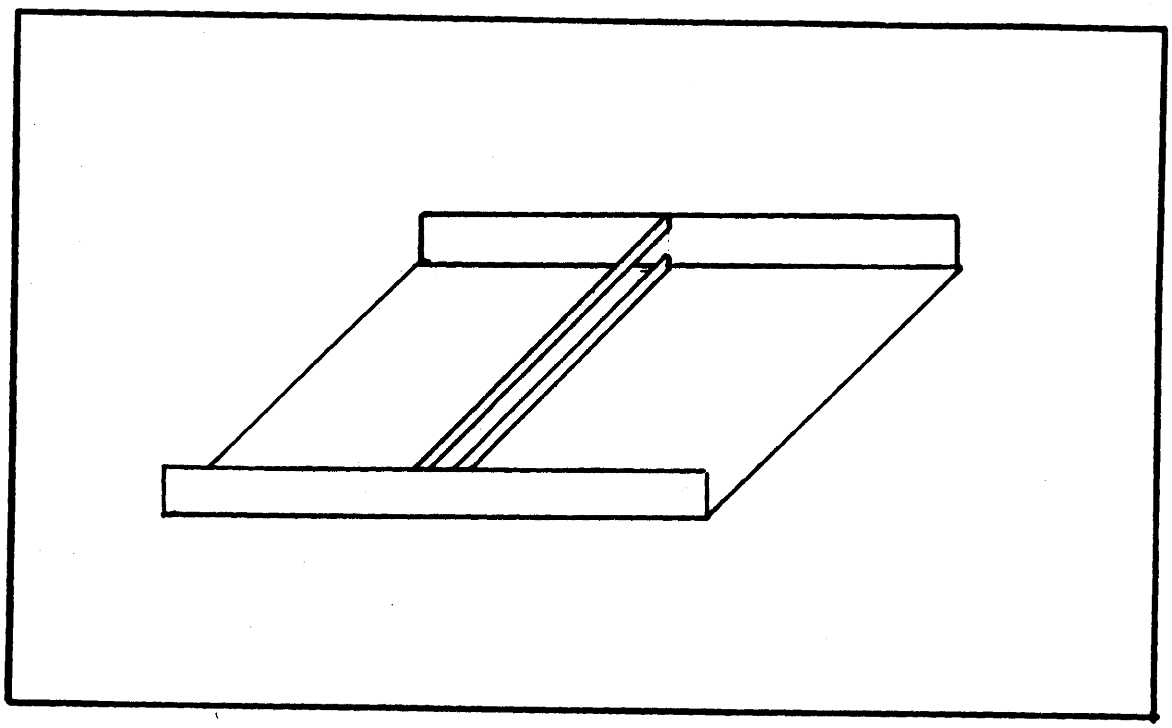
T_{oa} = actual outlet temperature,

T_{ot} = outlet temperature for theoretical plug-flow, and

t = elapsed time of operation.

Fig. 4.18

Vertical Barrier



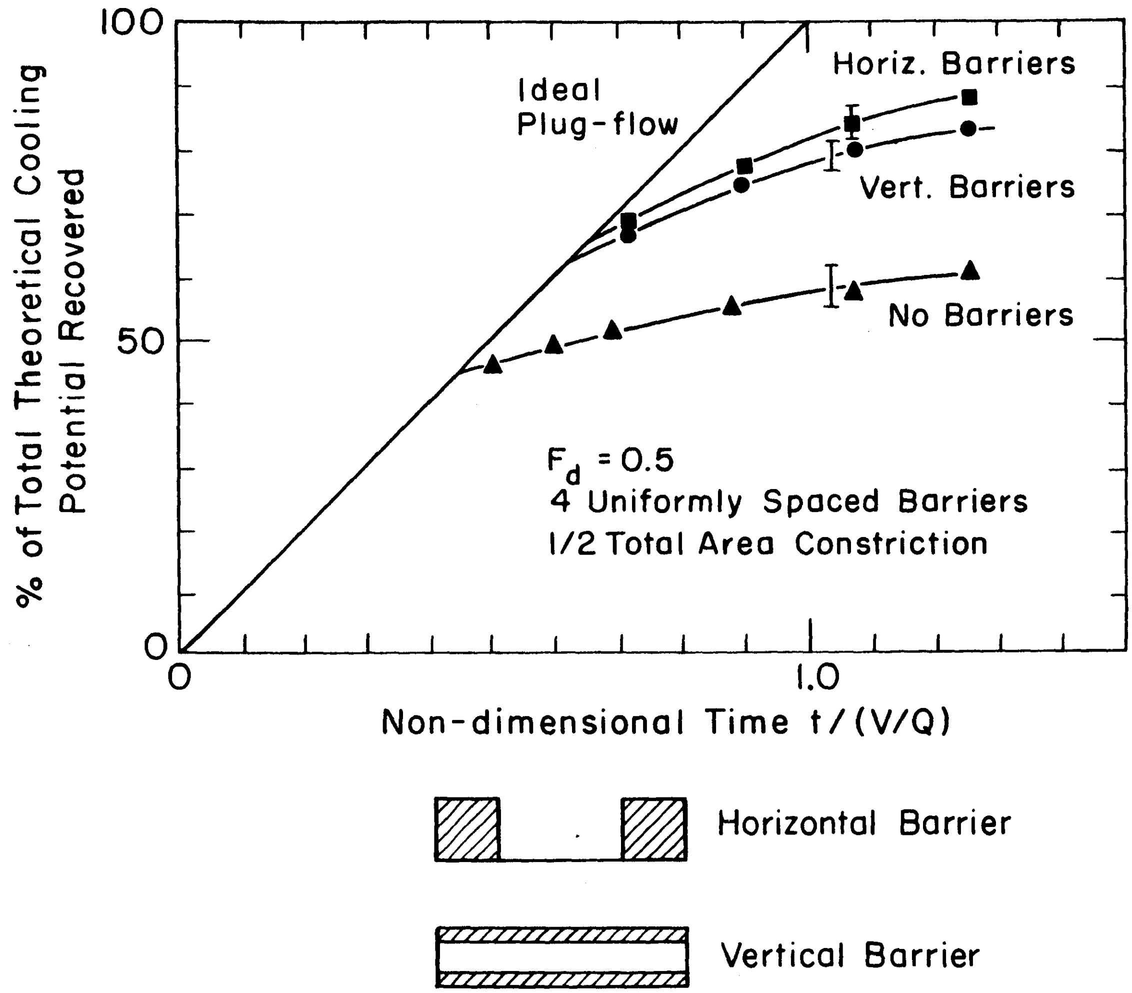


Fig. 4.19 Performance of Barriered TSP

Also shown in Fig. 4.19 is the idealized plug-flow performance curve and the performance curve for a TSP with no flow constrictuions.

All the curves in Fig. 4.19, except that for the idealized plug-flow case, are deduced from actual temperature measurements in the model at the point of flow withdrawal. In all cases the withdrawal structure consisted of a horizontal slot 1" high centered at mid-depth of the TSP model. The experimental error in the curves in Fig. 4.19 is estimated to increase from 0.0 (at $t/(\frac{V}{Q}) = 0$) to about $\pm 3\%$ (at $t/(\frac{V}{Q}) = 1.0$). Correction was made for water surface heat loss from the flume based on an experimental determination of the water to air heat transfer rate.

The performance curves in Fig. 4.19 indicate that in maintaining the total amount of flow constriction constant at 1/2 the total cross-sectional area (i.e. constant barrier pressure drop) the horizontal barriers appear to be slightly more efficient than the vertical barrier. If one considers the comparative difficulty in constructing the vertical barrier as opposed to that involved with the horizontal barrier, the horizontal is also seen to be the more desirable option.

It is important to note that all the curves in Fig. 4.19 are for a thermal storage pond with a design densimetric Froude number of 0.5. From Tables 4.1 and 4.2 it is seen that this corresponds to a prototype TSP design 15 ft. deep and 218 ft. wide. This geometry is practical for the storage of large volumes of water. A three-hour capacity TSP of these dimensions would require a channel about 3300 ft. long, and would have a total surface area of about 16 acres. Additionally, the flow head loss through the barriers is small. The 1/2-total area construction results in an additional head loss of approximately 0.05 inches per constriction in the prototype. This loss is negligible even for multiple barriers.

4.3.3.1b Observations of Thermal-Hydraulic Behavior of Horizontal-Barriered TSP

In performing the TSP model experiments the injected discharge into the pond was dyed in order that visual observations could be made of the flow processes. For the case of the barriered ponds this visualization of the density front behavior yielded substantial insight into the effects of the different types of barriers in retarding the density-induced motion. A series of photographs which characterize the behavior of an advancing hot front are shown in Figs. 4.26 through 4.29. The pertinent parameters describing the particular experiment are:

$F_D = 0.5$,	type of barrier = 1/2 total area horizontal type
Number of barriers = 8,	modeled storage capacity = 6 hrs.

Figure 4.26 shows the density front as it approaches the first barrier. Due to the thinness of the hot stratum it is not affected significantly by the barrier. Recall

that for the case of $F_D = 0.5$ the trailing edge (not visible in the photograph) never advances far beyond the inlet mixing zone. As the thickness of the density front passing through the barrier increases, however, the influence of the barrier becomes very strong as is shown in Fig. 4.27.

At lease two processes appear to be contributing to the vertical mixing of the hot and cold fluids. The first process in the downward movement of the hot upper layer along the upstream surface of the barrier, followed by streaming around the barrier and through the constriction. Passing through the constriction is an apparently nearly homogeneous flow which in the second process mixes with the ambient water in the downstream pond segment. The downward movement of the upstream hot layer behind the barrier can be observed in Fig. 4.27. However, the mean temperature of the flow through the constriction is necessarily less than that of the upstream hot layer since, as can be seen in Figs 4.27 and 4.28, the underlying cold layer is gradually drawn through the constriction.

The same processes occur at succeeding barriers downstream. In fact, succeeding barriers are more efficient in mixing the stratified layers since the density front at each succeeding barrier is progressively less stable than previously. In the latter portions of the pond the dye tracer indicates that little or no vertical stratification exists, as can be seen in Fig. 4.29 (the dark area to the extreme right is a shadow).

4.3.3.2 Refinement of Horizontal Barrier Concept

In attempting experimentally to optimize the design of a barriered TSP two general problems need investigation. The first is the design of the barrier itself, and the second is the specification of the number and spacing of barriers required to obtain good pond performance. Beyond these two problems it would be desirable to investigate the tradeoffs between decreasing the pond design densimetric Froude number value, and increasing the number of barriers. Additionally, the effect of a floating roof needs to be evaluated.

All these problems have been given some consideration by making appropriate modifications to the TSP experimental model. The goal of this series of experiments has been to converge to a spectrum of workable designs and within that spectrum to define a good design.

4.3.3.2a Barrier Geometry

Horizontal barrier design variations from that shown in Fig. 4.19 would include different total area constrictions, and a distribution of the constriction over the width of the channel such that multiple smaller internal jets would be created at each barrier. Increasing the total constriction (i.e. more blockage of flow) has the advantage of creating more vigorous jet mixing. However, the experiments have shown decreased jet size may lead to undesirable preferential attachment of the jet to the pond wall resulting in partial short-circuiting. This phenomenon, termed the Coanda effect [K6], is similar to that occurring in bistable fluidic switching devices. In the TSP model this effect was observed at a design $F_D = 0.5$, and a barrier geometry at 5/6 total constriction with a 1/6-width slot at the center of the pond. In this case the jet issuing from the first barrier became attached to the pond wall, bypassing a large part of the pond volume between the barriers. A large recirculating eddy was formed in each segment as is shown in Fig. 4.20. The result of this behavior is that no increase in the pond performance was noted beyond that obtained in the case of a 1/2 total area constriction. This attachment effect appears to be adequately counteracted by using smaller multiple jets to obtain the same total flow constriction.

Since this attachment phenomenon was evident in some experiments and not in others some critical condition for this attachment to occur is suggested. Review of the work by Krischner and Katz [K6] however, indicates that prediction of the critical condition in the particular situation of interest is an unsolved problem. Only some simple cases of isolated jet attachment have been investigated, with the results not being applicable to the present case.

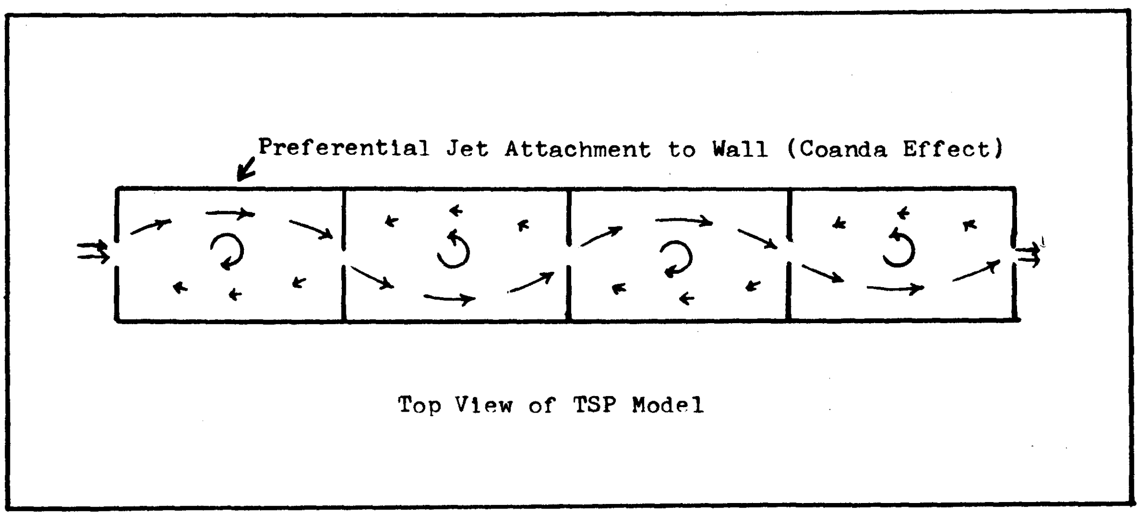


Fig. 4.20 Flow Field Resulting From Attached Jet

4.3.3.2b Number and Spacing of Barriers

Several experiments were performed to examine the effect of varying pond length on the number of barriers required to achieve a certain level of performance. The results are summarized in Table 4.4. Essentially, these results indicate that the number of barriers required to achieve a certain level of performance (percent of the theoretical cooling potential recouped at the plug-flow residence time V/Q may be largely independent of the length (storage volume) of the pond. To see this, compare the results of case 1 to those of cases 2 and 3 in Table 4.4. Intuitively the conclusion can be related to the need in the smaller (shorter) ponds to diminish the density induced flow more quickly than in the larger (longer) ponds. One would also expect that there would be a diminishing rate of return for the addition of barriers to the pond. Such behavior has been observed, as is shown in Fig. 4.21.

Throughout the experiments uniform spacing of the barriers was employed. Ideally, since the strength of the density-induced flow decreases as the mixed front passes through the pond it may be advantageous to have the barriers more closely spaced near the TSP inlet.

4.3.3.2c Design Densimetric Froude Number

As is discussed in Section 4.2.3.1 with regard to the initial design concept (unbarriered pond), the tendency

TABLE 4.4

Experiments to Examine Effect of Number of Barriers on Pond Performance

Case	F_D	Number of Barriers	Prototype Storage Volume	% of Total Theoretical Cooling Potential Recovered at t/(V/Q)=1.0
1	0.5	8	6	87%
2	0.5	4	3	81%
3	0.5	8	3	89%

All barriers 1/2 Total Area Constriction, 3 slot horizontal type

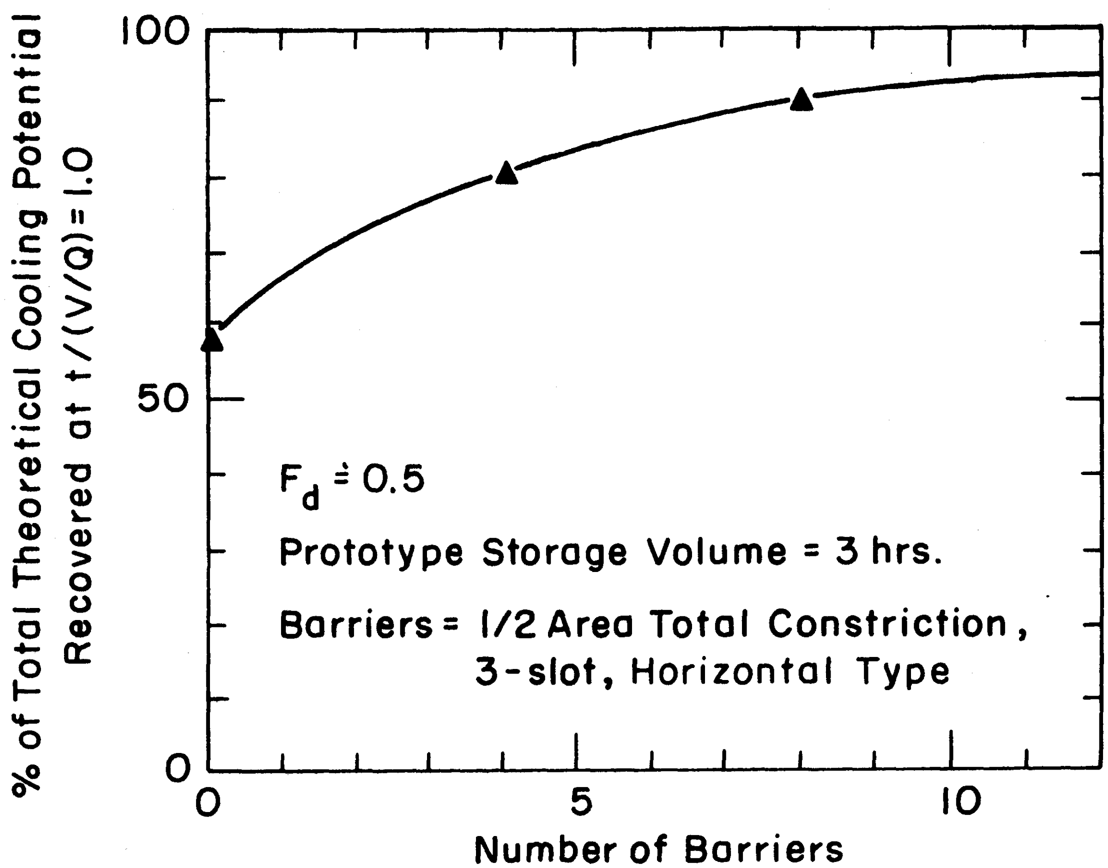


Fig. 4.21 TSP Performance as a Function of the Number of Barriers

for short-circuiting increased with decreasing design densimetric Froude number value. Qualitatively this same result holds in the case of the barriered pond. Decreasing the design densimetric Froude number value allows for a deeper and wider pond, but to maintain high performance additional (or more effective) barriers need to be added. Table 4.5 gives the performance loss for a decrease in design densimetric Froude number from 0.5 to 0.25 (cases 1 and 2). However, in comparing cases 1 and 3 in Table 4.5 it is apparent that a highly efficient pond can still be realized with the same number of barriers by increasing the effectiveness of the barriers. The barrier effectiveness is increased by increasing the total constriction, which in turn increases the jet mixing effect.

The experiments were limited to a minimum densimetric Froude number of 0.25 due to the necessity of maintaining a high (turbulent) Reynolds number. However, decreasing the design Froude number much below this value would probably not result in substantial savings since the number and width of the required barriers would become excessive.

In addition to the modeling of the behavior of the advancing hot front at the design condition temperature difference of 10 ^{o}F, experimental runs were made to examine the off-design behavior. One run was made which is identical to case 1 in Table 4.4 except instead of a hot discharge into a cold pond, cold water was discharged into an initially

TABLE 4.5

Experiments to Determine Effect of Design Densimetric Froude Number on TSP Performance

Case	F_D	Number of Barriers	Fraction of Cross-Sectional Area blocked by Barrier	% of Total Theoretical Cooling Potential Recovered at t/(V/Q)=1.0
1	0.5	8	1/2	87%
2	0.25	8	1/2	75%
3	0.25	8	5/6	86%

All barrier horizontal type, 3 slot, Prototype Storage Volume = 6 hrs.

hot pond. This simulates the "cooldown" mode of operation. As anticipated the performance increased from 87% to 92% (percentage of theoretical cooling potential recouped at $t=V/Q$). Also as anticipated an evaluation of the off-design performance ($\Delta T=5\ ^{o}F$) for a TSP of the same design revealed an increase in the performance from 87% to 90%.

4.3.4 Numerical Prediction Model for TSP Thermal Behavior

4.3.4.1 Analytical Modeling Difficulties

It has been demonstrated that the thermal performance of the thermal storage pond need not be known exactly for adequate evaluation of the plant lifetime economics. It is only required that the actual performance fall between the bounds of plug and fully-mixed flow. Nevertheless, if the TSP concept is to find application there will be a strong desire for a means of predicting the thermal behavior for all conceivable operating conditions.

In attempting to develop a predictive model of a barriered TSP one is immediately confronted with the problem of mathematically modeling the complex flow behavior of a density front as it intercepts and passes through a barrier. The fluid dynamic mechanisms which result in the observed behavior are not obvious. In fact, as is discussed in Section 4.3.3.1b, there may be several phenomena which contribute to the overall effect of the barrier. Additionally, there is the problem of predicting the motion of the density front between the barriers.

4.3.4.2 Approximate TSP Behavior Model

In spite of these difficulties some effort has been directed towards developing a model of TSP thermal behavior based on simple approximations to the density induced fluid dynamic behavior. It was believed that such a model might result in some improvement over the oversimplified plug-flow and fully-mixed models. The results obtained from the model are not very accurate.

The approximate numerical model used in attempting to predict the performance of a barriered TSP is based on a time-marching calculation of the position of the density front in each segment of the pond. The assumptions made are the following:

1) full vertical mixing occurs in all flows passing through a barrier,
2) a distinct linear interface develops in each TSP segment sequentially and its motion is given by Eqs. (4.52) and (4.55). (If the trailing edge velocity $V_T < 0.0$, then set $V_T = 0.0$)
3) full-mixing occurs at each time step of the water on the "hot" side of the linear density front,
4) if the leading edge of the density front intercepts the next barrier, then the amount of the

remaining cold wedge in that segment expelled through the barrier during that time step is equal to the volume decrease resulting from the imaginary projection of the leading edge of the density front past the barrier. (See Fig. 4.22)

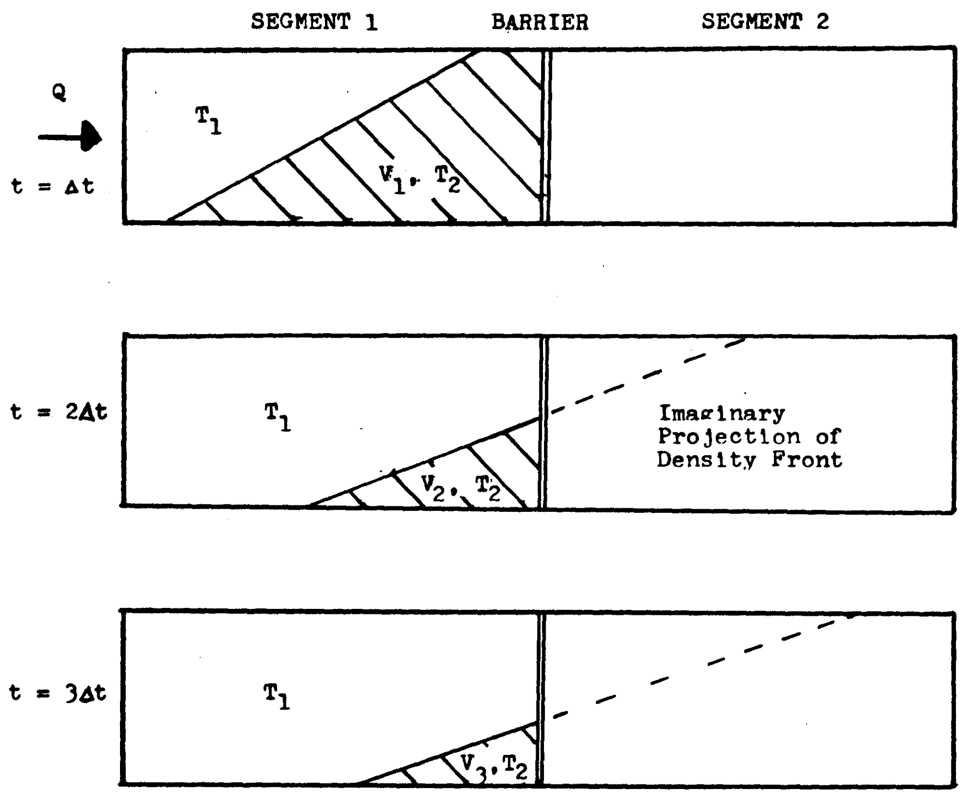
An illustration of the calculational procedure is given in Fig. 4.23.

A first test of the model is that it be able to predict the temperature of the discharge from an unbarriered pond. The agreement between the numerical model and the experimental result is better than that of the oversimplified models of plug-flow and full-mixing. A typical case is shown in Fig. 4.24. In applying the numerical model to a barriered pond some improvement over the idealized cases of flow behavior in behavior prediction was again noted as is shown in Fig. 4.25. Nevertheless, the errors are still large and it is apparent that a more realistic model of the fluid dynamics is required for confident predictions.

4.4 SUMMARY AND CONCLUSIONS

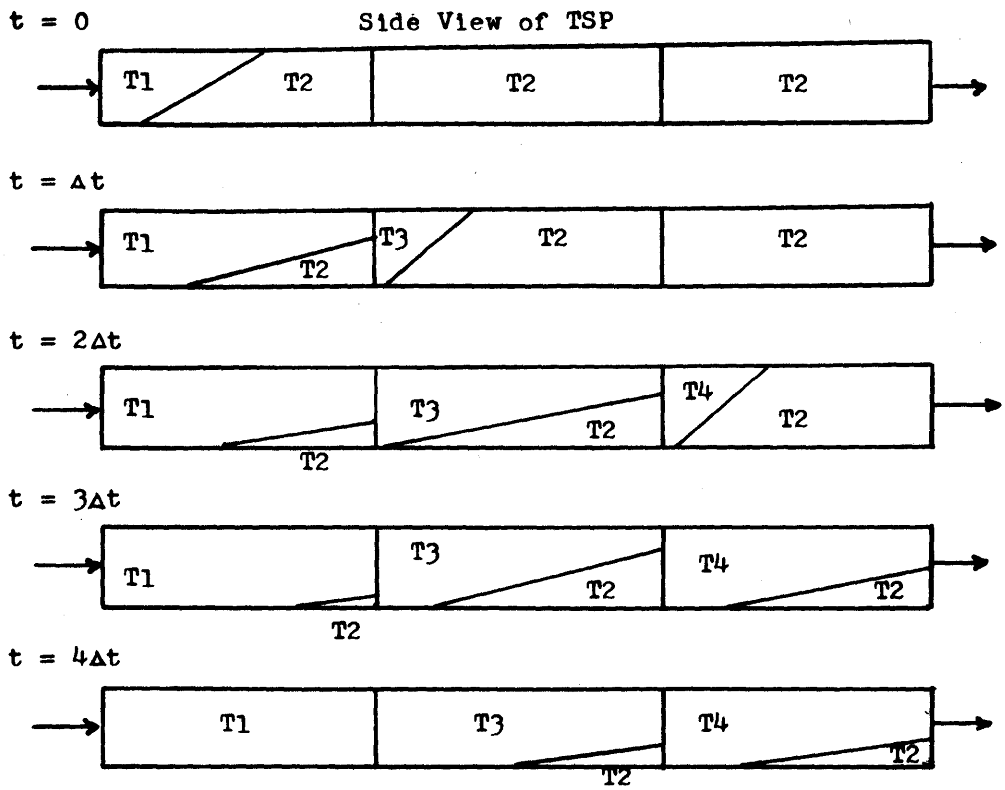
The two basic options for the thermal storage pond design - horizontal and vertical plug-flow - have been evaluated semi-quantitatively with regard to their

Fig. 4.22 Illustration of Barrier Flow Calculation



Temperature of discharge from segment 1 to segment 2 for $t=\Delta t$ to $t=2\Delta t$ $= \dfrac{(V_1-V_2)T_2 + (Q-(V_1-V_2))T_1}{Q}$

Temperature of discharge from segment 1 to segment 2 for $t=2\Delta t$ to $t=3\Delta t$ $= \dfrac{(V_2-V_3)T_2 + (Q-(V_2-V_3))T_1}{Q}$



T1 and T2 are constant

T3 and T4 increase with time depending on the temperature of the discharge from the upstream segment

Fig. 4.23 Illustration of Density Front Calculation

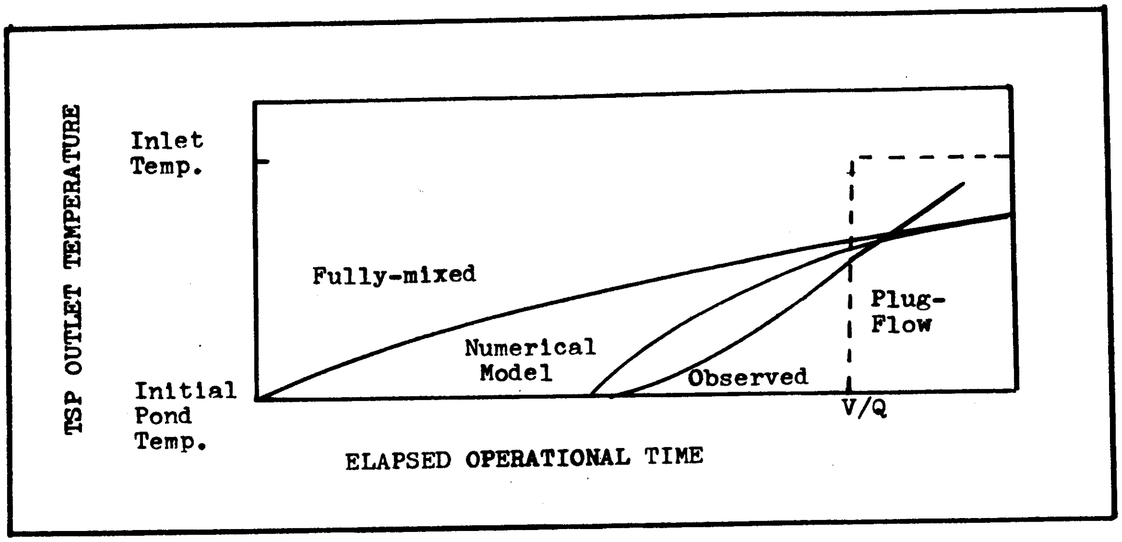


Fig. 4.25 Comparison of Observed and Predicted TSP Outlet Temperature -- 4 barriers, F_d=1.0

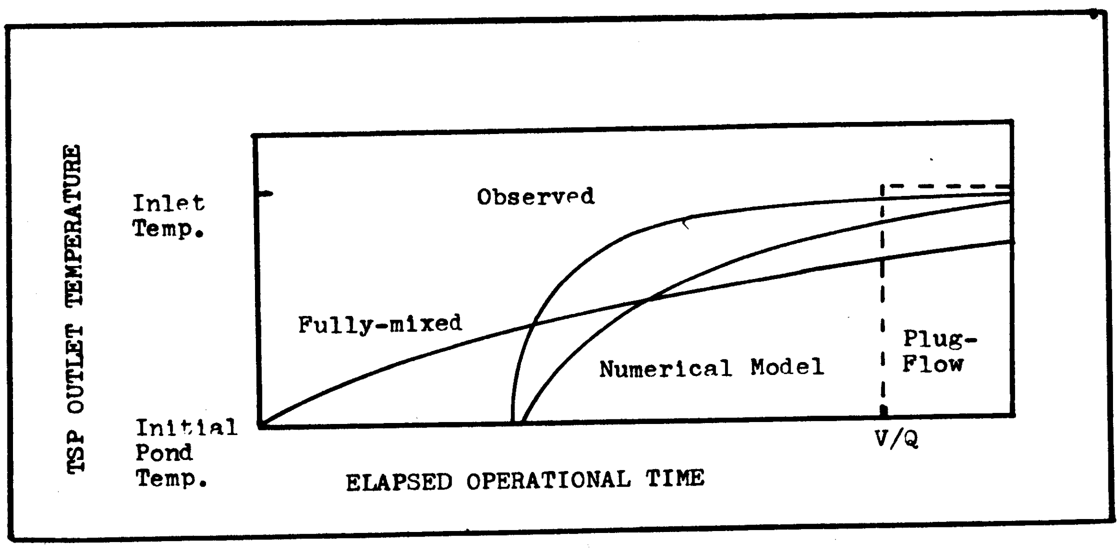


Fig. 4.24 Comparison of Observed and Predicted TSP Outlet Temperature -- No barriers, F_d=0.5

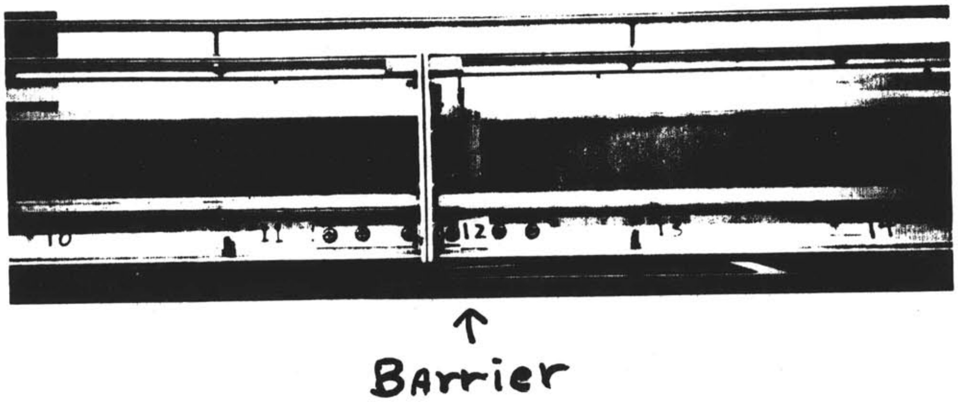


Fig. 4.26 Density Front
Intercepting Barrier

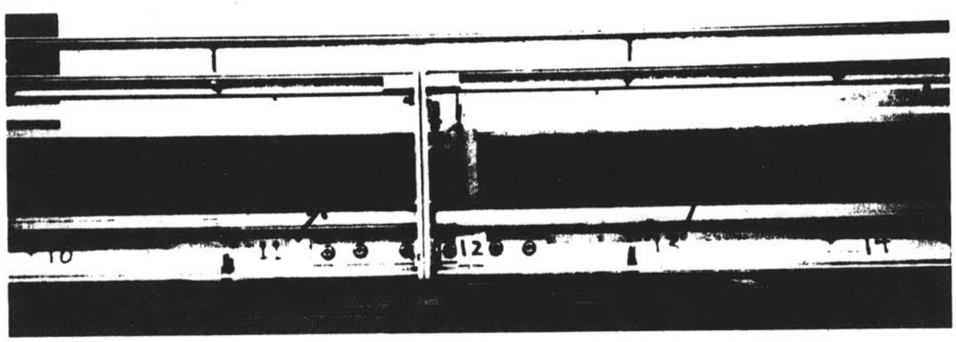


Fig. 4.27 Mixing of Density
Front at Barrier

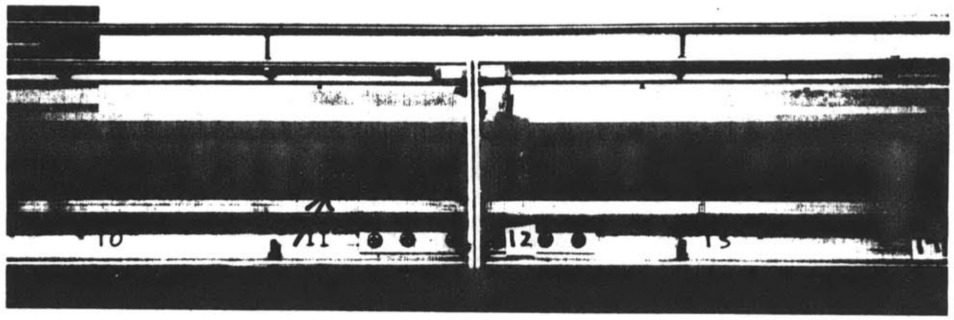


Fig. 4.28 Expulsion of Cold Wedge thru Barrier

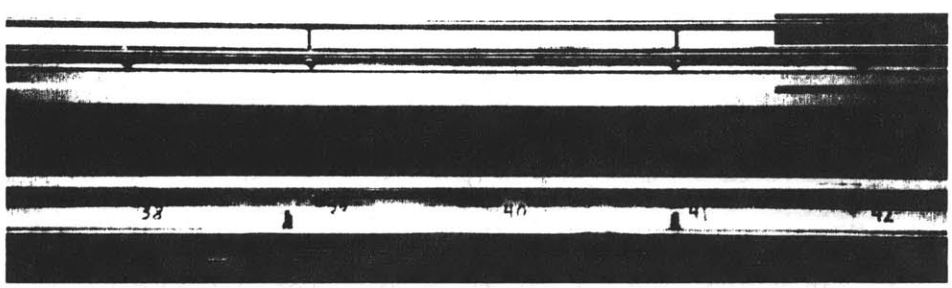


Fig. 4.29 Vertically Homogeneous Flow near Point of Withdrawal

value in TSP applications. The horizontal plug-flow pond was selected for detailed evaluation based on its relative merits. Subsequently, an initial design concept was formulated for the horizontal plug-flow pond and its performance evaluated by the use of an experimental model. The result was found to be unsatisfactory due to the magnitude of the density-induced flows. Thus, modification of the initial design concept was required. Various flow constrictions which induced flow mixing were evaluated in model studies in order to determine their relative merits. A simple full-depth barrier with vertical slot jets proved to be a very successful in achieving control of the density-induced currents.

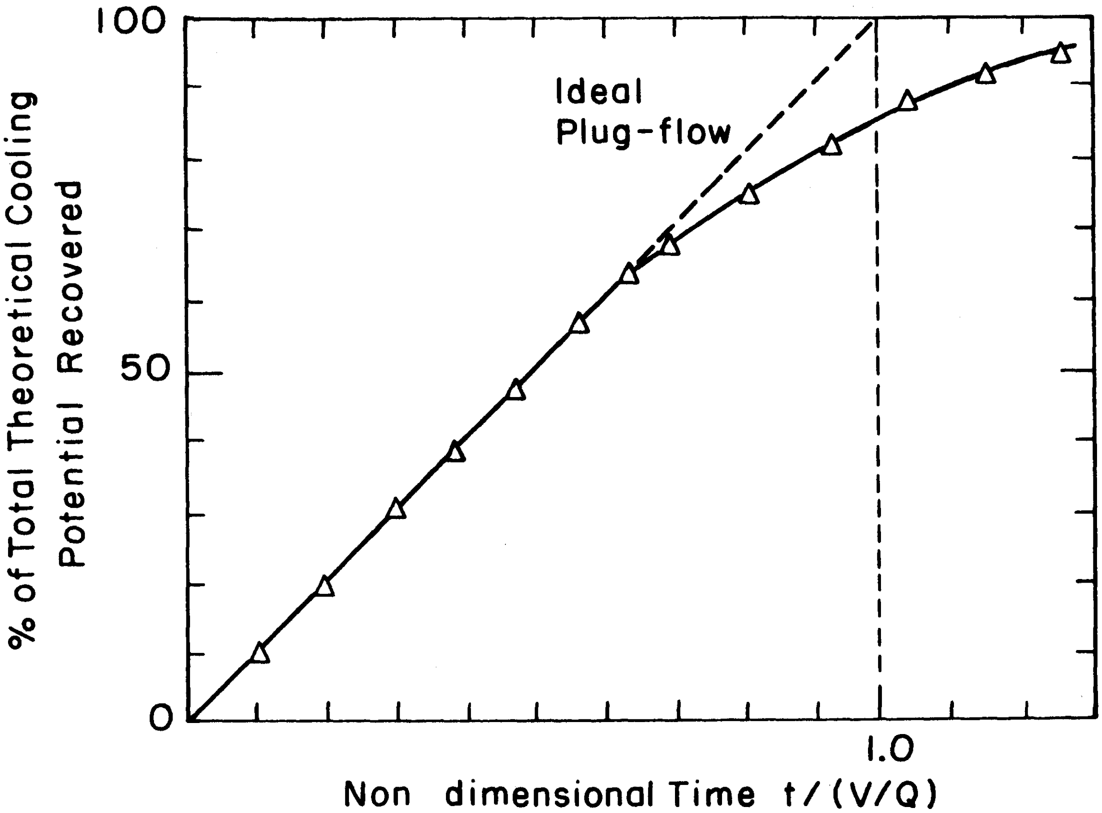
Experiments were performed to examine the sensitivity of the pond performance to variations in the design densimetric Froude number, and to the number and geometry of the flow constrictions. In addition the off-design performance was evaluated. An attempt to develop a simple mathematical model to predict the behavior of a barriered TSP proved to be inconclusive. Accurate and predictive mathematical modeling of the TSP thermal-hydraulic behavior will require development of more precise models of the transient non-homogeneous flows occurring in the TSP.

The main conclusion of this design investigation is that a horizontal plug-flow thermal storage pond of apparently reasonable geometry and cost can be designed such that the ideal plug-flow behavior is approximated well. The following TSP specification results in the excellent design-condition performance shown in Fig. 4.30. This case may be used to evaluate TSP construction costs:*

Design F_D = 0.25,
Depth = 20 ft.,
Width = 240 ft.,
# of barriers = 8
Length = (dependent on desired storage volume),
Barrier geometry = 5/6 total constriction, horizontal type, 3 slot
Discharge structure = 10 uniformly spaced 4' dia. submerged nozzles,
Withdrawal structure = no specific structure required, and
Cover = (floating cover will enhance performance).

It should be emphasized that many other designs may also prove to be efficient thermally, and may ultimately be more economical. Qualitatively, tradeoffs between the above parameters may be made on the following basis:

*For design flow = 1000 cfs, design ΔT = 10 ^{o}F



$F_d = 0.25$

Prototype Storage Volume = 6 hrs.

Barriers = 8, 5/6 Total Area Constriction
3-slot, Horizontal Type

Fig. 4.30 Performance of Recommended Design

Increasing design F_D → increasing performance

Increasing depth → decreases performance

Increasing width → decreases performance

Increasing number of barriers → increases performance

Increasing total constriction → increases performance

CHAPTER 5

ENGINEERING DESIGN AND ECONOMIC EVALUATION OF TSP/DRY COOLING TOWER SYSTEMS

5.1 Engineering Design Considerations

5.1.1 System Configuration

In general, there are two locations in the circulating water flow circuit of a dry dooling tower system at which one might consider incorporating a thermal storage pond. These locations are the dry tower outlet and the condenser outlet. Some simple analysis indicates that the tower outlet is the preferable location. The relative merits of the two configurations shown in Fig. 5.1 are best shown by a simple numerical example.

The performance of the idealized TSP/dry tower system shown in Fig. 5.1 resulting rom a 30 ^{o}F daily range in the ambient dry bulb temperature (70 ^{o}F to 100 ^{o}F) is shown in Table 5.1. Note that system A has been termed the "cold" pond system and system B the "hot" pond system. In both systems A and B in Fig. 5.1 the pond does produce the desired capacitive effect and a greater rate of heat rejection occurs during the "cooldown" mode of operation. As a consequence, the condenser outlet temperature (approximately equal to the condensing temperature) is less in

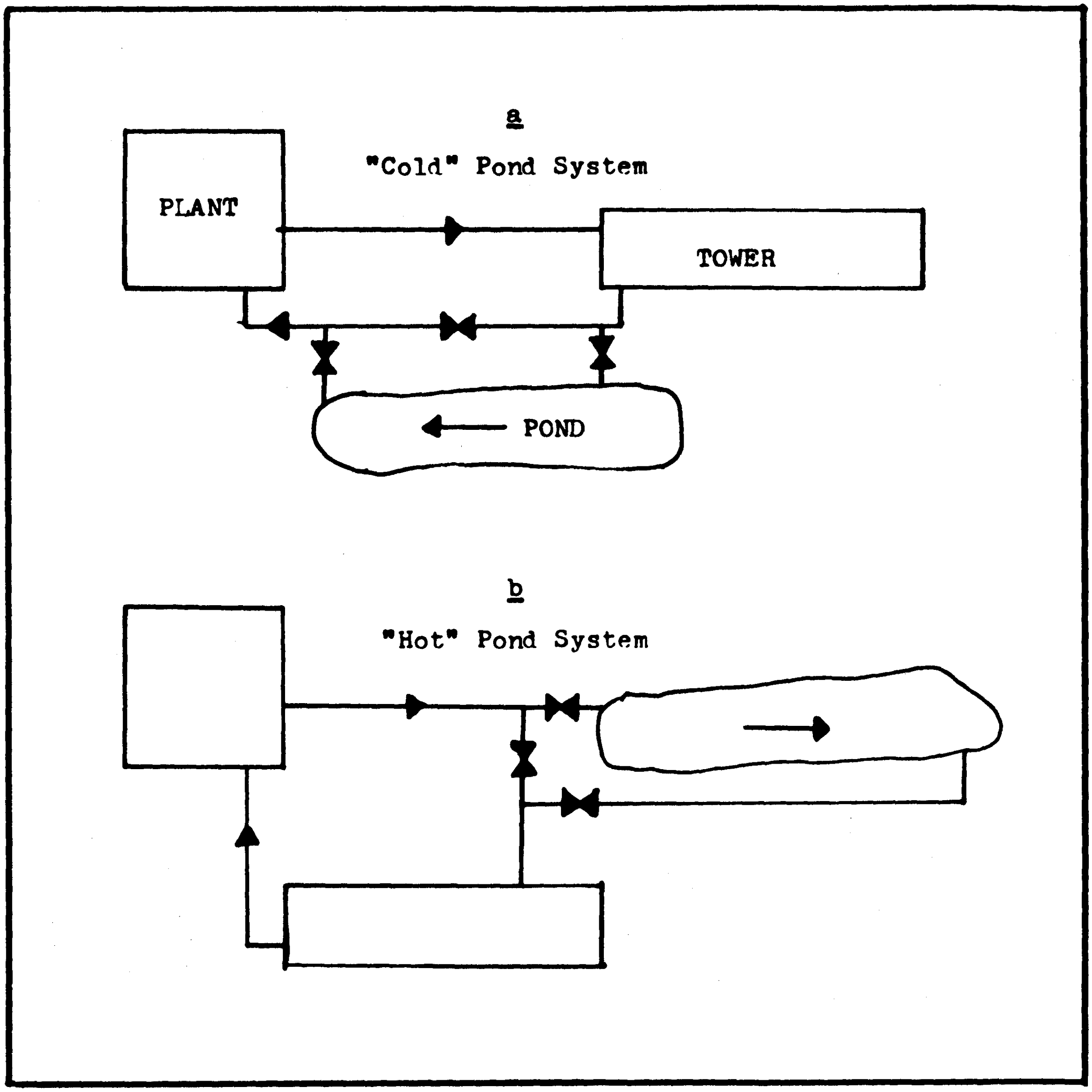


Fig. 5.1 Alternative TSP/Dry Tower System Component Arrangements

both systems during the "heatup" period in comparison to that occurring in a system with no capacitive pond. However, the condenser outlet temperature for system A during the heatup mode is substantially less than that for system B. The superiority of system A, the "cold" pond system, can be attributed to the fact that the condensing temperature in system A during the "heatup" mode is not directly coupled to the maximum ambient temperature as it is in system B.

The component configuration of system A also has other advantages. System A would allow the circulating water pumps to be located in their normal position just ahead of the condenser. System B would require that a set of pumps be located downstream of the pond in order to overcome the pressure drop occurring in the tower. Additionally, the "cold" pond concept is attractive since the temperature of the pond is lower than that of system B. High pond temperatures would not be advantageous with regard to the selection of a TSP lining and roofing material.

5.1.2 Mode Switching Transients

Knowledge of the thermal transients occurring in the TSP-dry tower-plant system as a result of operational mode switching is necessary for the prediction of the power generation transients. Also one needs to be assured that the turbine-condenser is not subject to excessive thermal-

Table 5.1

Comparative Performance of Alternative TSP / Dry Tower Configurations

	Condenser Inlet Temperature	Condenser Outlet Temperature	Tower Inlet Temperature	Tower Outlet Temperature
System A (TSP located at tower outlet)				
Heatup Mode	110°F	140	140	120
Cooldown Mode	120	150	150	110
System B (TSP located condenser outlet)				
Heatup Mode	120	150	140	120
Cooldown Mode	110	140	150	110
System C (Simple Dry Tower System)				
Afternoon*	130	160	160	130
Early Morning*	100	130	130	100

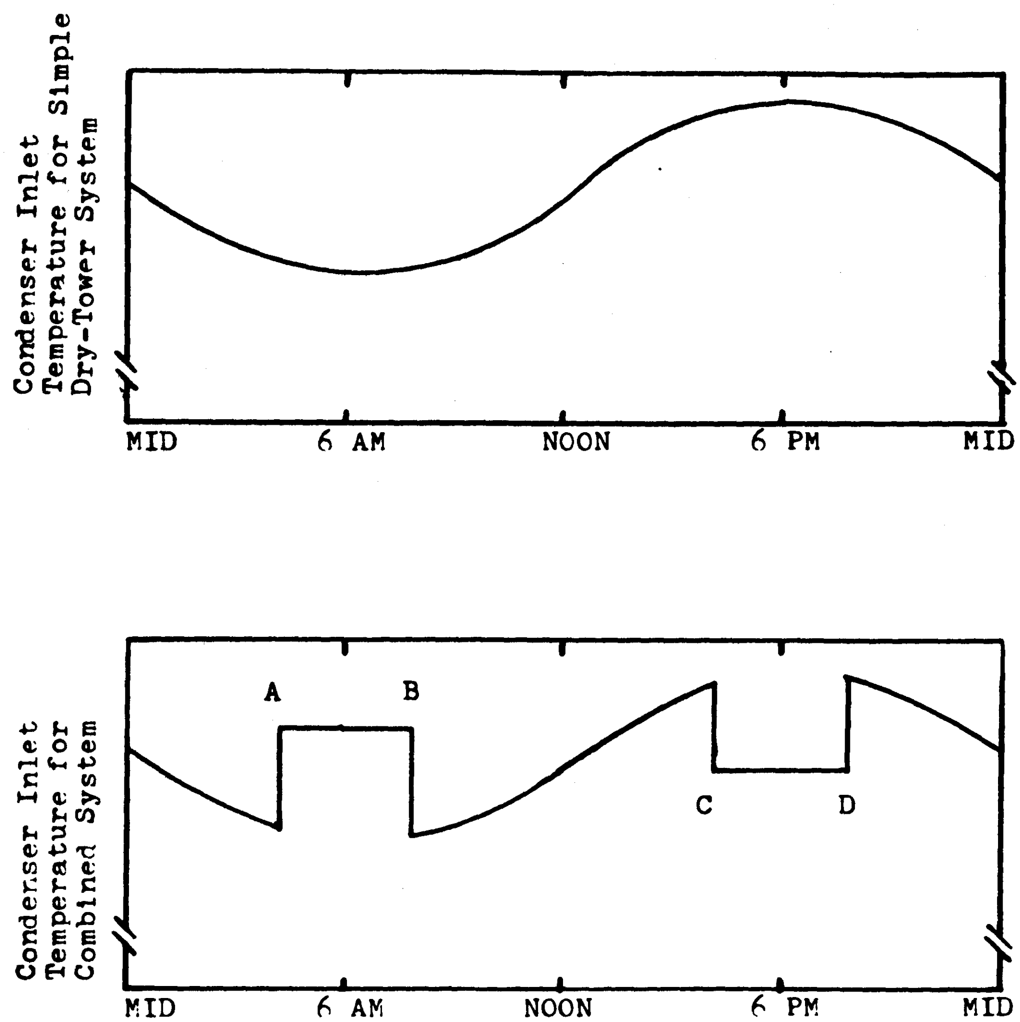
*Afternoon and Early Morning periods identical to Heatup and Cooldown Modes Based on Heatup Mode period ambient temperature of 100°F and a Cooldown Mode period temperature of 70°F mechanical stresses. The character of the thermal transients in the system will be determined by 1) the circulating water valve opening and closing time, 2) the water inventory of the dry tower-condenser system, and 3) the thermal inertia of the dry tower system.

5.1.2.1 <u>Condenser Inlet Temperature Transient</u>

The condenser inlet temperature as a function of time for an idealized TSP-dry tower-plant system in which the valving times, tower water inventory, and tower thermal inertia are equal to zero is shown in Fig. 5.2. Figure 5.3 is representative of the actual condenser inlet temperature transient which should occur at the "heat-up" mode startup time. The time interval τ_0 indicated in Fig. 5.3 is simply the time required for the simultaneous opening of the pond isolation valves and the closing of the valve on the pipe leading directly from the tower outlet to the condenser inlet. The condenser temperature would closely follow the condenser inlet temperature and thus for a 5-minute valving time the turbine-generator would experience a power transient (about 5 to 10%) with a period of about 5 minutes.

A power transient of this magnitude appears to be acceptable with regard to the allowable thermal transients in the steam turbine and with regard to the stability of the electric utility system. Discussion with a turbine

Fig. 5.2 Idealized TSP/Dry Tower System Performance



A - "Cool-down" Mode Startup Time
B - "Cool-down" Mode Shutdown Time
C - "Heatup" Mode Startup Time
D - "Heatup" Mode Shutdown Time

manufacturer (S3) revealed that the design thermal transients include those resulting from major load changes and load rejection. Transients resulting from TSP/dry tower mode switches are clearly much less severe.

The anticipated electrical power production transient also appears to be acceptable. The rate of change of load in a large utility system is about 1 to 2% per minute. Further a system must be able to withstand the instantaneous loss of the largest plant and remain operable. A 5 minute 50 MWe TSP induced transient (5% of 1000 MWe) in a 10,000 MWe system (i.e. 0.1% per minute) would not exceed these limits. Of course, the transients resulting from the TSP operation would be predictable by simple monitoring of the pond temperature and controllable by regulating the valving action.

5.1.2.2 Pond Inlet Temperature Transients

A condenser-piping-tower system has a finite water inventory and resistance to temperature change. Changes in the inlet condenser temperature will not be reflected in the tower discharge temperature (pond inlet temperature) until a time

$$\tau_R = \frac{V_{Tower}}{Q}$$

has elapsed where V_{Tower} equals the volume of the tower water inventory. Thus, as shown in Fig. 5.4, the pond inlet temperature (temperature of the water discharged

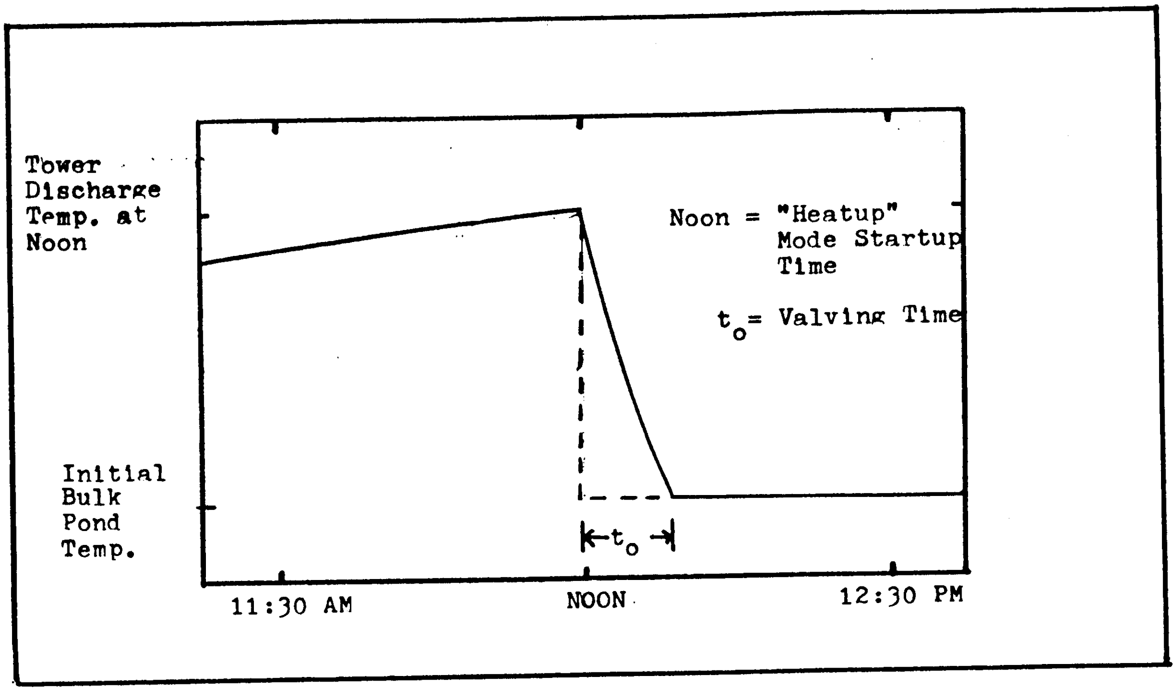


Fig. 5.3 Condenser Inlet Temperature Transient

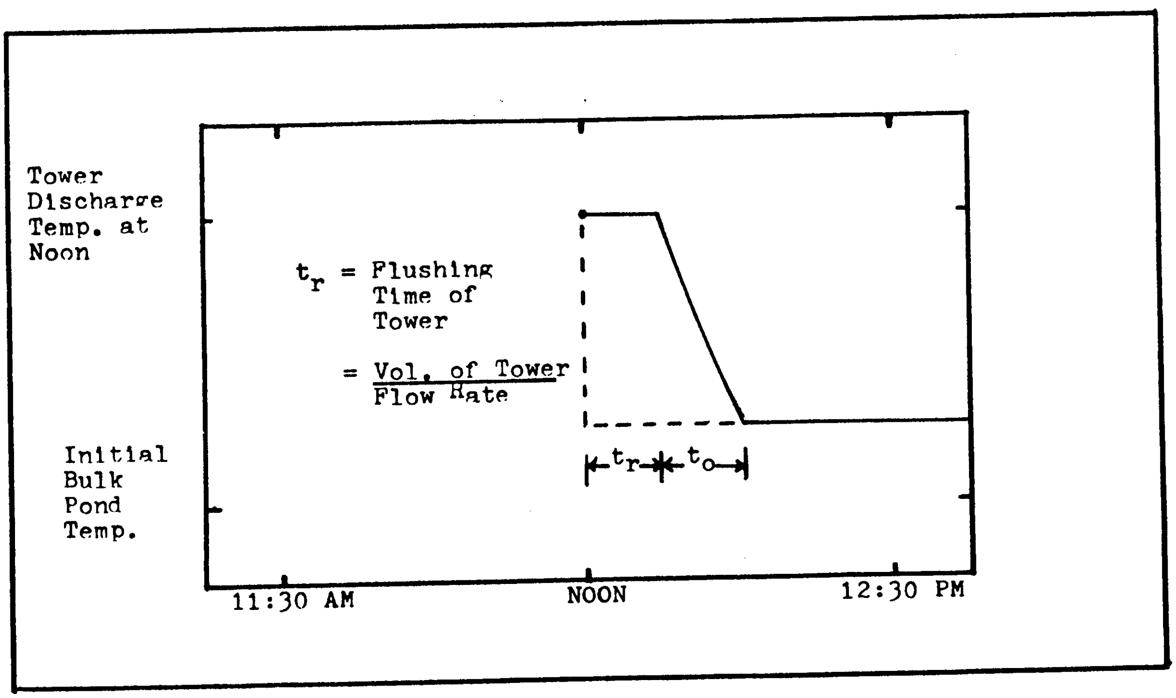


Fig. 5.4 Pond Inlet Temperature Transient

into the pond) will initially be equal to the steady-state tower outlet temperature existing just prior to the mode change. This pond inlet temperature will persist until the entire pipe-dry tower water inventory has been flushed. The time period for the subsequent decrease would be approximately τ_0, the previously mentioned valving time. Actually, the period of decrease would differ somewhat with the actual valving time period due to mixing in the condenser and the thermal inertia of the tower. However, both of these effects should be small. Condenser volumes are typically equivalent to only about 2 minutes of flow [D3]. A lumped parameter analysis has shown that the characteristic thermal response time of a typical dry-tower finned-tube bank is about 0.5 minutes.

5.1.3 Thermal Storage Pond Construction

The use of a thermal storage pond in conjunction with a dry cooling tower will be practical only if the water loss from the pond due to seepage and evaporation can be controlled. If the condenser cooling system is to be totally "dry", it will be necessary to line the pond bottom to stop seepage and cover the pond to inhibit evaporation. In some cases it may be desirable to leave the pond uncovered and thus receive the added benefit of evaporative cooling at the expense of a significant amount of water consumption. In almost all cases, control of seepage would be required since it results in no cooling effect. However, the covered pond system is qualitatively more attractive in that it provides the only totally "dry" solution to the performance problems of conventional dry towers. Thus, the available techniques and materials for lining as well as covering storage reservoirs will be discussed. The purpose of this discussion is to review the available literature on the subject and justify the conclusion that the construction of a TSP would not involve the development of a substantially new water impoundment technology.

5.1.3.1 Linings for Thermal Storage Ponds

A variety of different materials have been examined with regard to the lining of large irrigation canals and "finished" water storage reservoirs. The Bureau of Reclamation of the Department of the Interior has been particularly active in examining canal lining techniques many of which would be applicable to the construction of a TSP [B2] [H7] [H8]. A summary of the types of canal linings placed on Bureau operated projects is given in Table 5.2. Since it would be desirable to maintain high water quality, use of either buried membrane linings or earth linings would probably be undesirable. Also seepage rates through simple earth linings would be intolerable. In fact, as is discussed in Reference [B2], seepage can be significant in many of the asphaltic and cement linings due to post-

TABLE 5.2

TSP Lining Options

I Exposed Linings

a) Asphaltic Concrete

b) Asphaltic macadams

c) Asphaltic surface membranes

d) Reinforced Portland cement concrete

e) Unreinforced Portland cement concrete

f) Pneumatically applied Portland cement mortar

g) Soil cement

h) Plastic and rubber membranes

II Buried Membrane Lining

a) Asphalt

b) Bentonite

c) Plastic

III Earth linings

a) Thick compacted earth

b) Thin compacted earth

c) Loosely-placed earth blankets

d) Soil sealants

construction cracking. Only plastic and rubber membrane linings appear to offer the potential for zero seepage.

In addition to providing for near-zero seepage membrane linings are seen to be lowest in cost. Recent applications of calendered polyvinylchloride sheet have included a 215 million gallon, 40 acre sewage lagoon and a 447 acre solar evaporation pond for the recovery of potash from brine [K1]. Day [D2] has presented a manual for the details of brine disposal ponds which in many respects are similar to the proposed thermal storage pond. Recently a DuPont Co. synthetic butyl-rubber called Hypalon has seen numerous reservoir lining applications [P4]. A 1975 cost of lining a 14 acre reservoir with 45 mil. Hypalon was $\$0.69/ft^2$ with a projected lifetime of 40 years [R5]. Discussion with a vendor of this product indicates that the highest temperature which might occur in a TSP (about 115 ^oF) and the flow velocity (less than 1 ft/sec) should not degrade this type of lining [K2].

5.1.3.2 Covers for Thermal Storage Ponds

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 [C9] and floating rafts [T2] appear less attractive.

Chin [C2] 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. Projected costs for steel and pre-cast concrete coverings supported on columns ranged from 2 to 3 times those of flexible floating coverings. 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 [P4] [R2]. Currently vendors of the patented "Roofloat" system [K2] are anticipating 40 year lifetimes for conventional storage reservoir applications. This cover, which is made from nylon reinforced Hypalon synthetic rubber appears to be satisfactory for use in covering a TSP and currently (1976) has a cost of between \$1.50 and \$2.50/Ft^2. Economies of scale and the simultaneous installation of a similar lining should favor a minimal unit cost of covering. The drag on a floating cover which would result during the operation of a TSP has been evaluated and found to be insignificant. The placement of the suggested barriers in the TSP should provide convenient anchor points for a floating cover. Consideration of the effect of pond freezing on the cover, as well as lining and barriers, will be essential. The rubber membrane lining and covers, however, have demonstrated resistance to freezing of the impounded water.

5.1.4 Steam Turbines for Use With Dry Cooling Systems

The maximum allowable condenser pressure of conventional steam turbine systems is about 5" HgA. With dry cooling systems condensing pressures far in excess of this value would be routinely encountered. Although dependent on the size of the dry tower system, a condenser pressure range of about 3 to 15 HgA is not unlikely.

The main problem in designing a steam turbine to operate at high back pressure is the design of the blading of the last few stages of the low pressure turbine. High back pressure results in a substantially decreased steam specific volume and thus a substantially decreased axial steam velocity. Normally, high axial steam velocities are required to assure that inlet steam velocities relative to each blade row are subsonic, that negative reaction static pressure rise across the rotating blades will not occur, and that adequate stage efficiency is obtained (S2).

At present, no nuclear-steam turbine which can be operated at elevated back pressure is marketed domestically. However, one manufacturer has developed a special low pressure stage design suitable for operation up to 15" HgA for use with high-quality fossil-steam [M6]. The poorer nuclear-steam quality and the resulting generally larger physical size of the nuclear turbines necessitate considerable additional development work. Currently, it appears that the domestic turbine vendors are not pressing forward with this development effort (S3).

The hesitancy of vendors to invest in the high-back-pressure nuclear turbine appears to stem from the following two factors. First, the electrical generating industry simply has not demonstrated a near-term need for such devices. Second, the actual design of the turbine will be influenced by the yet-to-be-resolved economics of dry-tower condenser cooling. The industrial need has not occurred since to date the availability of makeup water for evaporative cooling systems has not yet severely constrained generating capacity expansion.

The actual turbine design will be influenced by the economics of dry cooling because different types of high back pressure designs will result in different performance curves as shown in Fig. 5.5. In general, two types of high back pressure turbine designs appear to be feasible each of which has its particular performance characteristics. Curve A is representative of a design which has a small low-pressure stage exhaust annular area. Reducing this area (in comparison to the conventional design-curve C) allows high axial velocities to be maintained even at high condensing pressures. However, at low condensing pressure

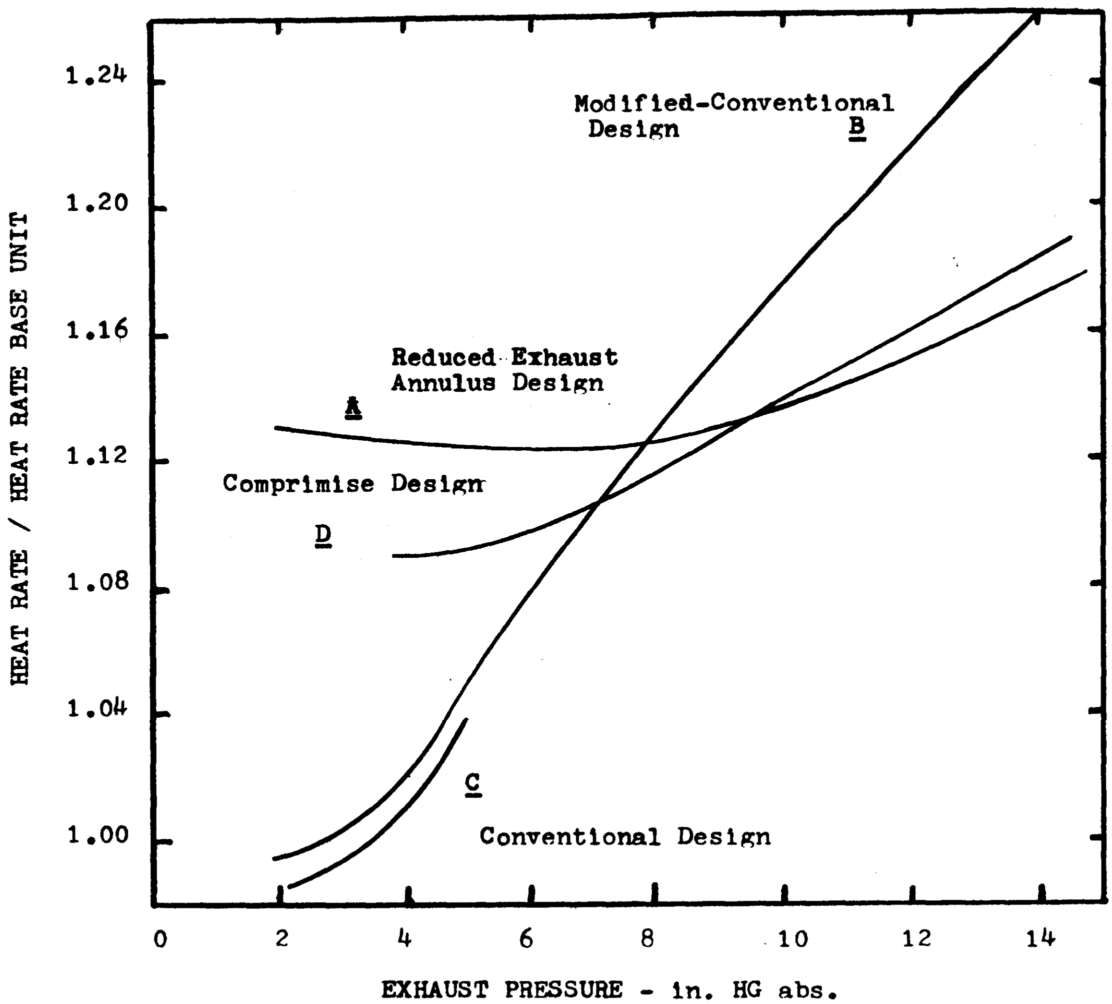


Fig. 5.5 Exhaust Pressure Corrections for Nuclear Steam Turbine Heat Rates (R7)

the flow through the last stage of the turbine becomes choked and further decreasing the condenser pressure does not result in increasing performance. Thus, although the performance is relatively good at very high pressures the performance is relatively poor at the more frequently occurring low pressures. Curve B shows the performance of a design which is similar to the conventional design (approximately same exhaust annulus area) but which is suitable for operation at the higher pressures. In this case the low pressure operation is good while at high pressures the decreasing specific volume (axial velocity) of the steam results in inferior performance.

To date no decision has been made by the industry as to which type of design would yield the most favorable overall system economics. Studies by a turbine manufacturer have suggested that a compromise design, the performance of which is shown by curve D in Figure 5.5, is most desirable [M7].

Since the performance characteristics of high back pressure steam turbines are not as yet resolved this study of the combined dry-tower/TSP system will individually consider the application of the TSP system to plants with the following turbine types:

1) Conventional with 5.0" HgA limit (Curve C).
2) Modified Conventional type (Curve B)
3) Reduced exhaust annulus type (Curve A)

Since the TSP concept works better when the plant performance is more sensitive to variations in the condensing temperature it is apparent that a plant with a modified conventional design turbine (Curve B) would be more greatly benefited by the incorporation of a TSP into the dry cooling system. The conventional turbine design will also be considered since this represents a proven and readily available technology and is projected to be less' expensive. Estimated costs for a high-back-pressure design turbine are about 15% higher than for conventional designs (C3). This difference is substantial considering the cost of a 1000 MWe steam turbine -- about $50 million.

5.1.5 Secondary Uses of a Thermal Storage Pond

Although the primary use of a TSP at a central power station would be the maintenance of favorable temperatures in the main steam condenser, the TSP may also play an important role in meeting other heat rejection needs of the dry-tower cooled plant. For both fossil and nuclear plants the TSP may aid in the meeting of the service cooling water requirement. For the nuclear plant, the TSP could provide a reliable backup system for emergency cooling of the reactor. In this section the possible use of a TSP in these applications is examined.

5.1.5.1 Utilization of a TSP for Emergency Cooling

All nuclear power plants require a reliable backup cooling system for the dissipation of reactor core decay heat for an extended period of time. At several sites using evaporative cooling towers the emergency cooling system requirement has been met by constructing a small pond [H5]. A thermal storage pond could be utilized similarly at a station cooled by dry cooling towers.

The waste heat rejection requirements of a nuclear plant due to decay heat is well described by [E1].

$$q = 0.095\ P_o t^{-0.26} \tag{5.1}$$

when

P_o = Power before shutdown

t = time in seconds.

This equation applies only to t>200 sec and to infinite (usually a good approximation) operation with uranium fuels. Assuming a fully-thermally-mixed TSP an energy balance for a TSP during the dissipation of decay heat would be

$$V \rho c_p \frac{dTp}{dt} = P_o (o.095) t^{-0.26} - KA\ (T_p T_E) \tag{5.2}$$

where

T_p = pond temperature,

V = pond volume,

A = pond surface area,

C_p = specific heat of water,

ρ = density of water,

K = pond surface heat transfer coefficient,

T_E = pond equilibrium temperature, and

t = time.

This equation contains the additional simplification that K, the heat transfer coefficient, is independent of the pond temperature. The above equation can be simply solved for the limiting case of

$$\frac{dT_p}{dt} = 0$$

and the results of such a solution are plotted in Fig. 5.6.

The limiting temperature of the pond as a function of the pond size is given in Fig. 5.6. The limitimg temperature is defined as being that temperature of the pond above which there will not be a rise of the pond temperature due to the introduction of decay heat. For all initial pond temperatures less than this value the pond temperature will at first rise and then later decline, but the maximum attained will be less than the limiting temperature. A TSP would likely not be any less than 10 acres in size. The approximate 105^{o}F limiting tempera-

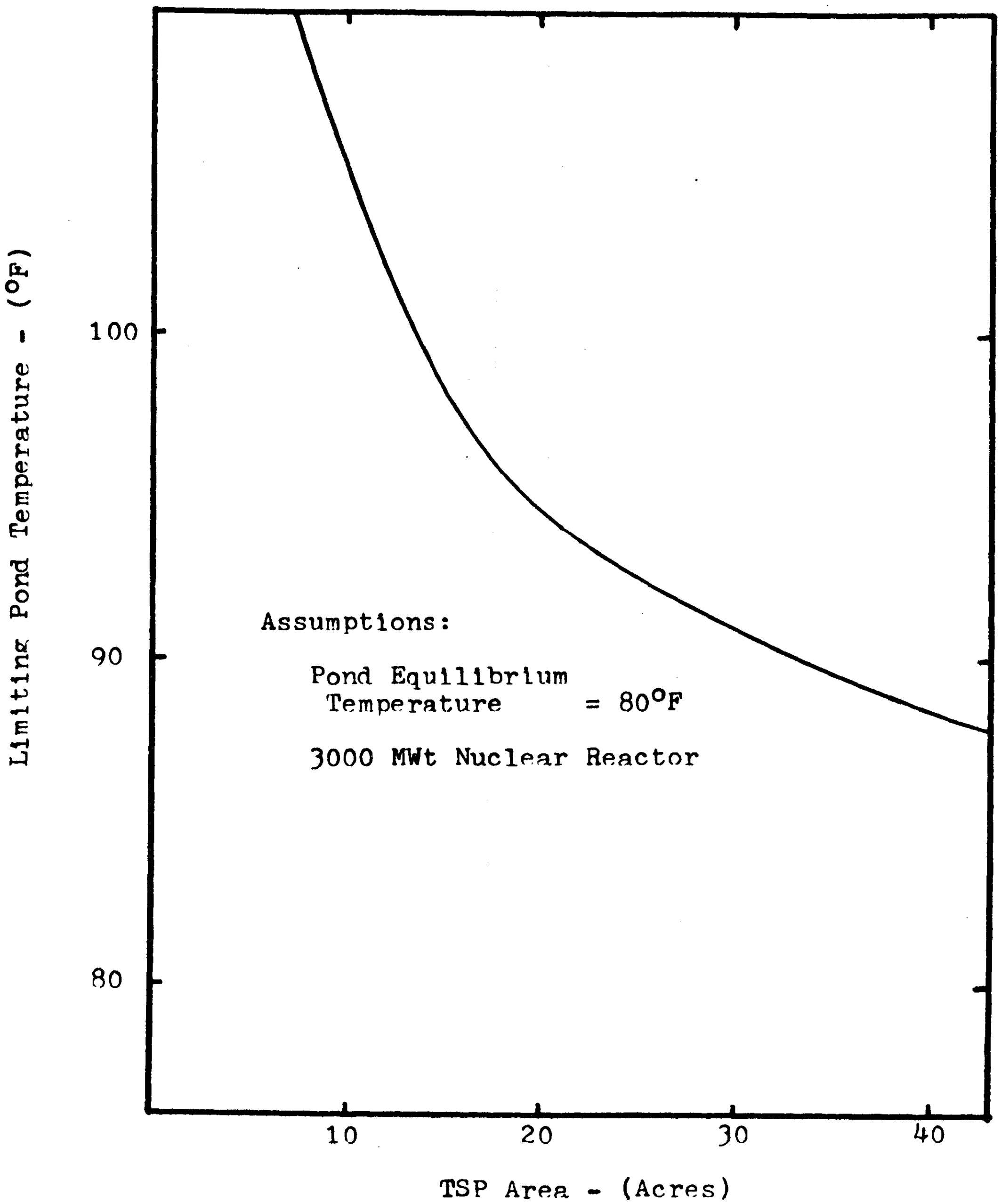


Fig. 5.6 Limiting Temperature for Decay Heat Rejection

ture for a pond of this size is well within the range of temperature for cooling water supply. This analysis is based on an uncovered pond with evaporative heat transfer taken into account. In the case of a covered pond, ample time would exist for the removal of the pond cover, if necessary, since the adiabatic heatup rate of the pond due to decay heat accumulation would be only about 1 ^{o}F per hour. The total amount of water evaporated from the pond over a period of several months of decay heat rejection would be small in comparison to the TSP volume.

In conclusion, a thermal storage pond should provide an excellent source of cooling water for long-term core decay-heat removal. This conclusion is significant since the use of a TSP as an emergency cooling system may help justify the required capital expenditure for the construction of the pond. Other studies of dry-cooling at nuclear power stations have, to date, not taken into consideration emergency cooling requirements [R7].

5.1.5.2 Use of a TSP for Service Water Cooling

The service water cooling requirement for a nominal 1000 MWe nuclear power station is about 150 x 10^6 BTU/hr with a maximum inlet temperature of 95 ^{o}F. The 95 ^{o}F limit arises from temperature limits in the hydrogen cooling of the generator and the cooling of the turbine bearing oil. The amount of cooling required is about 2% of that of the total plant heat rejection and consideration has been given to use of a covered TSP as a means of meeting this requirement. Application of an uncovered TSP would indicate that small amounts of evaporative water consumption are tolerable and in such cases a small, mechanical-draft, evaporative cooling tower would be the logical means of satisfying the service cooling water requirements.

For the TSP/dry tower system to be the sole source of service water cooling, water of a temperature less than 95 °F must be available from the tower discharge or from the water stored in the pond. Based on the computer simulation of optimally-designed TSP/dry tower systems for two different sites this was not found to be the case. For both sites, Winston, Arizona and Billings, Montana, during a significant length of time in the summer the temperature of both the tower discharge and the pond exceeded 95 °F. The quantitative results of the simulation calculation are given in Table 5.3. At both sites the TSP can extend the use of the main cooling system as a supply of service water only slightly and does not negate the need for some intermittent evaporative cooling.

5.2 Economics of TSP/Dry Cooling Tower Systems

Having established the magnitude of the potential benefit and engineering practicality of the thermal storage pond concept it is necessary at this point to consider the comparative economics of the TSP/dry tower system with respect to simple dry-tower cooling systems.

Table 5.3

Service Cooling Water Availability in a TSP/Dry Tower Cooling System

	Site	
	Winslow, Arizona	Billings, Montana
Number of Hours* the Temperature of the Tower Discharge is Less Than 95°F	6585	7590
Number of Hours the Temperature of the Tower Discharge is Greater Than 95°F and the TSP Temperature is Less Than 95°F	838	572
Number of Hours the Temperature of the Tower Discharge is Greater Than 95°F and the TSP Temperature is Greater Than 95°F	1337	598

* Hours per Year

5.2.1 Basis for Economic Comparison

Previous studies concerned with the economics of dry cooling have presented final cost determination in terms of the incremental cost of utilizing dry cooling per unit of electrical energy production. The generating plant is assumed to have a fixed capital and operational cost independent of the specific cooling system design. Annualized capital and operating expenses for cooling systems are divided by the annual electrical power production to yield the incremental cost in mills/KWHR.

An expression for the incremental cost of dry cooling can be derived from the simplified electrical energy cost equation. The cost equation is

$$e = \frac{(\phi I+f+O)\ 1000}{(K*8760)\ L\cdot} \tag{5.3}$$

where

e = cost of electricity (mills/KWHR),

I = total capital cost of station ($),

ϕ = annual fixed charge rate,

f = annual fuel cost ($),

O = annual operating cost ($),

L = plant capacity factor, and

K = net plant capacity (KWe)

In the above equation the quantities I,f, and o can be expressed as a sum of two parts -- one associated with the costs of the basic plant and site and one associated with the use of dry cooling,

Thus,

$$e = \left[\frac{(I_p+I_d)\,\phi+f_p+f_d + O_p+O_d}{(K \cdot 8460)\, L} \right] 1000 \qquad (5.4)$$

where the subscripts p and d refer to the basic plant-site and the dry cooling system respectively. The basic plant-site cost component of the total cost can be expressed as

$$e_p = \left[\frac{I_p\ \phi + f_p + O_p}{(K \cdot 8460)\ L} \right] 1000 \qquad (5.5)$$

and the cost of the dry cooling system (i.e. incremental cost of dry cooling) as

$$e_d = \left[\frac{I_d\ \phi+f_d+O_d}{(K \cdot 8460)\ L} \right] 1000 \qquad (5.6)$$

where

$$e = e_d + e_p . \qquad (5.7)$$

The determination of the quantities I_d, f_d, and O_d necessarily include a consideration of the capital and operation expense of the dry tower itself as well as a consideration of the electrical generation losses imposed on the plant.

The dry tower capital cost includes the direct cost of the dry tower and associated pumps and piping in addition to the cost of replacement electrical generation capability. A fuel cost f_d associated with the dry cooling system arises from the power requirements of the dry tower itself and the need to replace the plant power losses resulting from elevated condensing temperatures. The operating cost of the dry tower would be equal to the operating cost of producing the energy replacement in addition to the cost of maintaining the dry tower system.

The economics of central power station cooling systems has been based on the assumption that any loss of capacility below that of the maximum station rating due to cooling system inadequacies must be replaced by supplemental electrical generation. This assumption is founded on the fact that in many utility systems the maximum electrical demand is coincidental with the maximum power generation losses due to cooling system inadequacies (i.e. highest ambient temperatures). During such periods there would not be routinely large amounts of idle capacity in excess of the required reserve capacity and therefore supplemental generation in an amount equal to the maximum loss must be included in the total cost of dry cooling.

The TSP/dry tower system represents an effort to correct this unfavorable coincidental occurrence of the peak condensing temperature and the peak electrical demand

and thus the simple assumption concerning the necessity of penalizing all lost capability and energy production irrespective of its time of occurrence is no longer adequate or rational. The specific assumption made with regard to loss of capability penalties which should allow for the realistic accounting of the TSP benefit and the justification for these assumptions are found in the next section.

5.2.2 Evaluation of Condenser-Cooling-System Induced Power Generation Losses

5.2.2.1 Approach to Cost Evaluation and Justification

In previous economic analyses of condenser cooling systems no accounting has been made for the routine variation with time of the worth of electrical generation. Typically, in a large utility during the early morning hours there is a large excess of generating capacity and the incremental cost of obtaining the next kilowatt-hour of electrical energy is small. During the afternoon, especially during summer peak demand periods, there would typically be a dearth of excess capacity with an incremental kilowatt-hour of electrical energy obtained at considerable expense (i.e. from lightly-loaded units with high heat rates). Previous studies have assumed that all the replacement energy can be evaluated at a constant cost [H4] [R7]. Further, as implied in the previous section, the capital cost of the replacement capability in these studies is evaluated by a simple calculation of the maximum power loss occurring during the year.

Clearly these assumptions are adequate when the peak load is coincidental with the maximum loss of capability. A more realistic approach to cost evaluation which can be applied to plants in which the maximum loss of capability is not coincidental with the peak electrical system load will be based on the following assumptions:

1) the capability replacement cost is evaluated by a determination of the annual maximum power loss occurring during the daily period of peak electrical demand and,

2) all replacement energy costs are to be evaluated at the incremental power production cost in the utility system at the time of the loss.

The first assumption can be justified by examining the load curve of a typical large utility system for the summer day on which the yearly peak demand occurs as shown in Fig. 5.7. Ideally, the utility's generation capability (minus the required reserve capacity) would just equal the peak load indicated in the figure. Any capability loss due to dry cooling occurring at the peak demand time must be included as an economic penalty since supplemental generation would in fact be needed to meet the peak demand. For the simple dry cooling system the loss of capability penalty must be assessed as the absolute maximum loss since the maximum loss and peak load are coincidental. However, for a plant with a well-designed TSP/dry tower system the maximum loss should occur at a time other than the peak

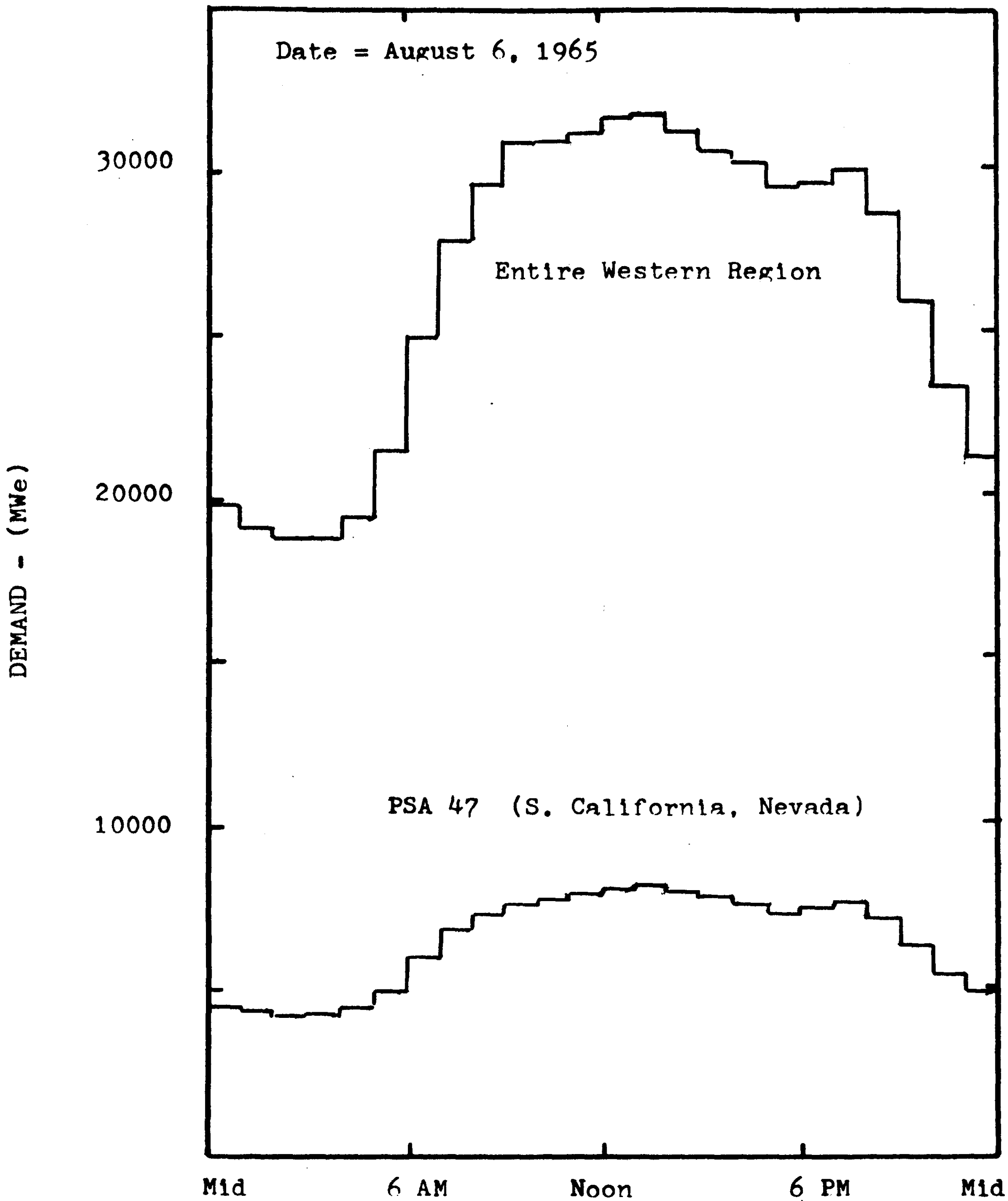


Fig. 5.7 West Region Summer Daily Load Curve (F5)

load time as shown in Fig. 5.8. Figure 5.8 shows the excess system generating capability as a function of time based on the maximum required capability shown in Fig. 5.7. Also shown in Fig. 5.8 is the loss of capability as a function of time for a hypothetical 1000 MWe plant cooled by a TSP/dry tower system. For a "well-designed" TSP/dry tower cooled plant the maximum loss of capability should occur when there is ample excess system capacity to accommodate the loss. The capability replacement penalty would be assessed only as the actual capability loss occurring at the peak load time. Since this capability savings at the peak load time is the main purpose of implementing a TSP the criterion of "well-designed" can be construed to mean that the length of the operation of the TSP is sufficiently long such that the peak demand time will always fall within the period of the TSP operation.

The second part of the operational penalty is the cost of replacement energy. The incremental cost of electricity production is defined as the cost of obtaining the next kilowatt-hour of electricity given the present status of the utility system and forms the basis for assessing the penalty. This incremental power production cost, termed λ, is important with regard to utility generating scheduling. At any given time, the generating units called-up or shut-down are those whose value of λ is most nearly that of the

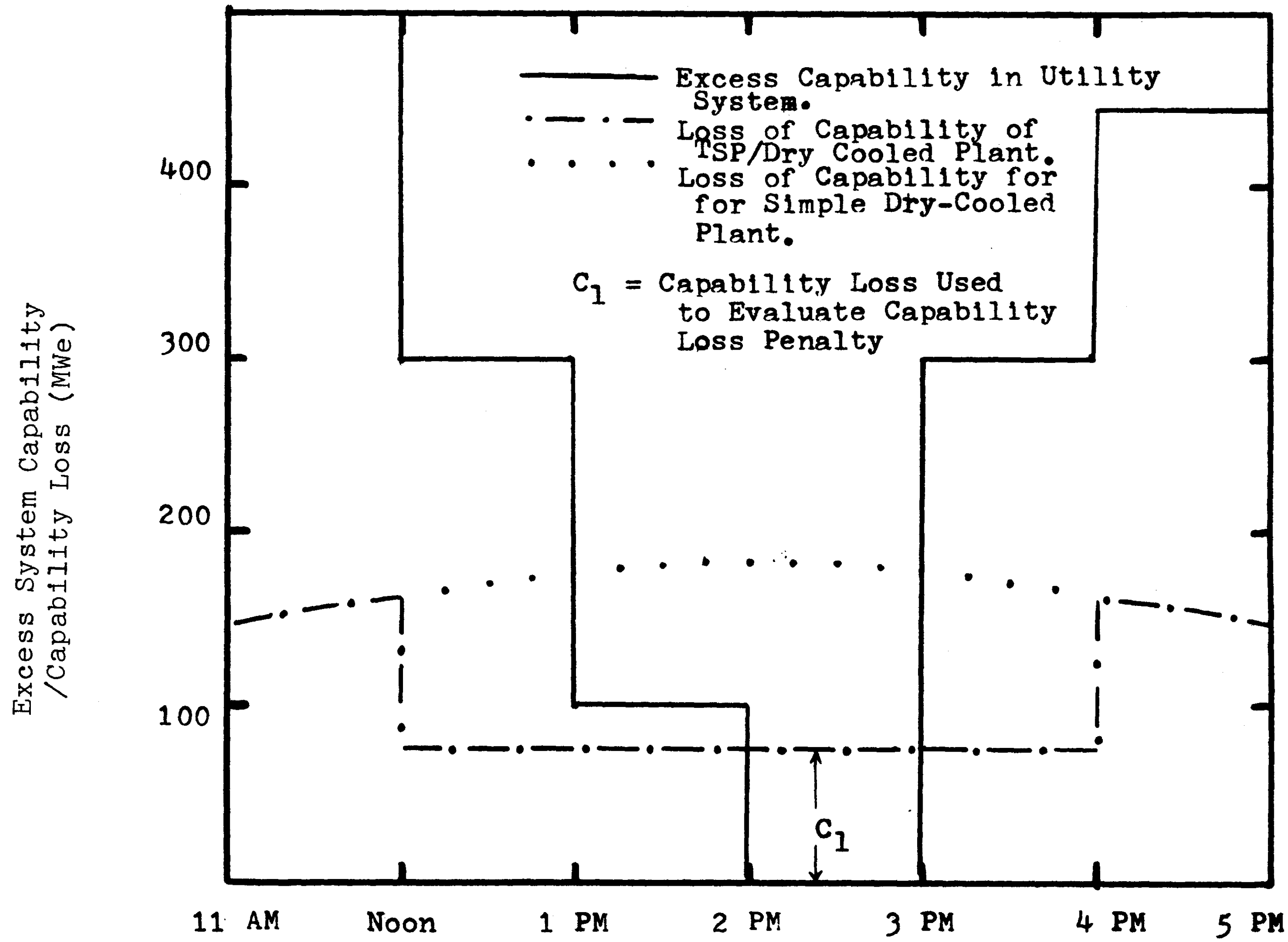


Fig. 5.8 Evaluation of Loss of Capability Penalty

system λ. The system λ equals the value of λ for the plant currently operating with the largest value of λ.

To a good approximation, the penalty for not generating power as a result of cooling system inadequacies can be based on the system λ and thus will vary with the time of day. Weekly and seasonal variations in the system λ could be important. In general, however, the system λ would be minimal during the early morning hours reflecting the fuel and operating costs of large base loaded units and be at a maximum during the afternoon reflecting the operation of peaking units with high heat rates.

Utilizing the system λ as the economic basis, the evaluation of the energy replacement penalty is straightforward. The annual energy replacement cost would simply be

$$ERC = \int_{1\ \text{year}} \lambda(t)\ C_L(t)\ dt \qquad (5.8)$$

where

$\lambda(t)$ = the incremental power production cost of the utility system at time t, and

$C_L(t)$ = the capability loss experienced by the plant at time t.

In terms of the expression for the incremental cost of dry cooling (Eq. 5.6)

$$ERC = f_d + O_d \qquad (5.9)$$

since the value of λ incorporates both the fuel and operation cost of producing an incremental unit amount of electricity.

5.2.2.2 Practical Aspects of Determining Capacity Replacement and Energy Replacement Costs

As Eg. 5.8 suggests, the correct determination of the energy replacement cost requires the simulation of the plant-TSP-dry tower system performance for at least one year with a maximum time step of one hour. Such a calculation is the basis for the economic case studies of TSP/dry tower cooling reported in Chapter 6. Larger time steps would not allow for an accurate determination of the economic benefit of the TSP while smaller time steps would be computationally prohibitive.

Since in the case studies a single year of historical meteorology was used to evaluate the system performance some question arose as to what temperature (or what day) should be used to evaluate the maximum loss of capability. The year of historical meteorology for a given site was chosen at random. Evaluation of the replacement capability penalty solely on the basis of the hotest day for the particular year chosen is not an adequate approach since 1) the yearly single highest temperature occurring at a particular site is subject to substantial variation from year to year and 2) the economics of the TSP system are strongly dependent on the temperature used to evaluate the capability replacement penalty. Therefore the approach used was to determine the average maximum temperature of the ten hotest days for the chosen meteorological year. This average maximum temperature is used to directly evaluate the maximum loss of capability for the simple dry tower systems considered in the case studies. The determination of the peak demand time loss of capability for the TSP/dry tower cooled plant involved additionally the determination of the average minimum temperature on the ten hotest days and the averages of the maximum and minimum temperatures of the ten days preceding the ten hotest . The preceding day's average maximum and minimum temperatures were used to calculate iteratively an initial TSP temperature. This TSP temperature was subsequently used to determine the loss of capability of the plant at the peak load time subject to the average temperature extremes of the ten hotest days.

The accurate evaluation of the energy replacement cost at a given site is more difficult since the value of λ as function of time is highly dependent on the utility system generating-unit mix and load curve. Efforts were made to secure directly from electrical utilities historical information which would detail the value of the system λ for a specific area at one-hour increments for a one year period. This effort was unsuccessful. Apparently, in practice, the system λ is not routinely recorded by most utilities in scheduling the production of electricity from various units in the system.

Nevertheless, after some discussion with utility operation personnel it was concluded that the best approach to the problem of specifying a system λ function for use in a survey of TSP/dry tower economics would be to assume that the value of λ varies sinusoidally from a minimum at the time of minimum utility system load to a maximum at the time of maximum system demand. The maximum and minimum values are chosen to represent the fuel and operating cost of based loaded steam-electric plants and peaking combustion turbines or deisels respectively. The specific form of the assumption is shown in Fig. 5.9.

Considering all aspects, this approach may be preferable to use of site-specific historical data since 1) rapidly changing fuel costs would negate much of the benefit of using historical costs and 2) using an identical economic model at the different sites of the case-studies allows a clearer resolution of the significance of the major performance-determining site variable -- the site meteorology. In any event, the case studies include an evaluation of the sensitivity of the results to this and other economic assumptions.

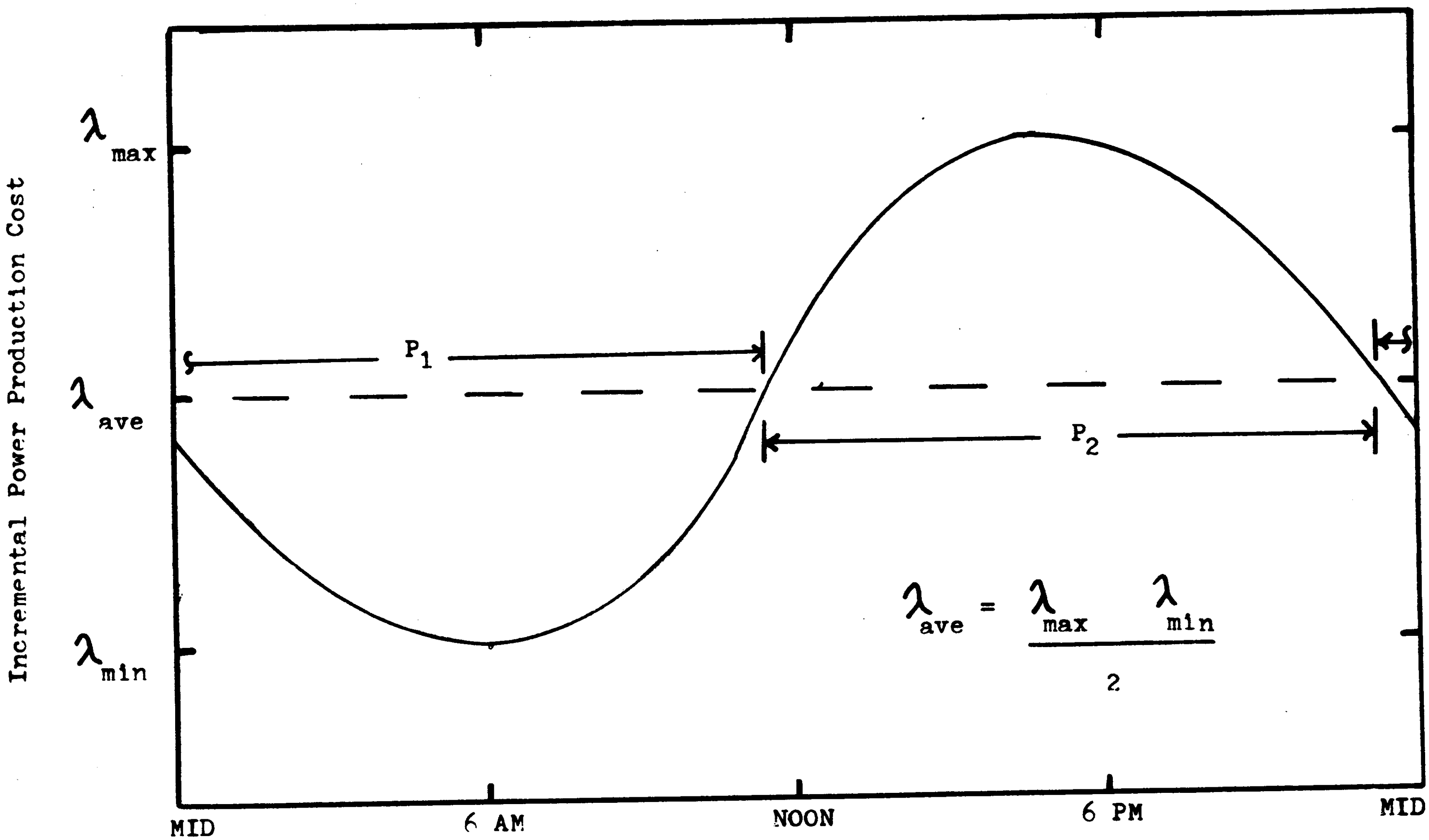


Fig. 5.9 Assumed Form of Energy Replacement Cost Function

5.3 Methods for Determining Optimal Thermal Storage Pond Dry Cooling Tower - Plant System Designs

Any decisions concerning the economic viability of the combined thermal storage pond - dry cooling tower system must be based on a comparison of the economics of this system with the various design alternatives. To insure a meaningful comparison, the TSP-dry tower and the alternative systems chosen for this comparison should in each case represent optimal system designs. As in all economic studies, optimum is defined as "least-cost."

The main emphasis of this work is a comparison of the economics of combined TSP-dry tower waste heat rejection systems and conventional dry tower heat rejection systems for large steam-electric plants.

5.3.1 Design of Optimum Simple Dry Tower - Plant Systems

For the purposes of this study it is assumed that the dry-tower unit cell design is fixed and the present task is simply a specification of the system design. Thus, the design of optimal simple dry-tower cooled stations requires only a determination of the particular circulating water flow rate and the number of dry tower unit cells which will yield the smallest incremental cost of dry-cooling. The numerical search procedure termed the Rosenbrock is utilized to determine optimal designs. The details of this method are presented in section 5.3.5.

Similar to the combined TSP/dry tower system, the determination of the cost of dry cooling for a specific design is based on a one year plant performance simulation with one hour time steps.

5.3.2 Design of Optimum TSP-Dry Tower-Plant Systems

Determination of optimum TSP-dry tower-plant designs is a more formidable problem. This is because in addition to a determination of the actual design of the system (i.e. number of tower cells, flow rates) it is necessary to determine the optimal method of operating the system. The full set of design and operational variables which needs to be considered in arriving at the least-cost TSP/dry tower cooling system design is given in Table 5.4.

The decision variables included in Table 5.4 are separated into two groups -- discrete value integer operational variables and continuous design variables. The need to constrain the operational variables to integer values stems from the computational requirement of simulating the plant performance.

To solve the optimization problem indicated in Table 5.4 a two stage optimization method has been utilized. The integer variables are treated by means of a simple grid search. Then for each set of integer operational variables determined by the grid search the optimal set of continuous design variables is determined by the Rosenbrock method. Thus, a sub-optimal determination of

TABLE 5.4

TSP/DRY TOWER SYSTEM OPTIMIZATION DECISION VARIABLES

I) Discrete Value Integer Operational Variables

- A) heatup mode startup time
- B) heatup mode shutdown time
- C) cooldown mode startup time
- D) cooldown mode shutdown time

II) Continuous Design Variables

- A) dry tower size
- B) circulating water flow rate
    - 1) coupled modes
    - 2) uncoupled modes
- C) fraction of flow bypassing pond
- D) size of pond (volume)

the continuous design variables is nested in the grid search of the integer operational variables.

The cost of performing the nested optimization has been greatly reduced by making some simple but adequate assumptions. First, it is assumed that the length of the "heatup" mode of operation is equal to that of the "cooldown" mode. Second it is assumed that the "heatup" mode operation period and the "cooldown" mode operation period are centered on the times of the daily maximum and minimum ambient temperatures respectively. This simply has the effect of maximizing the temperature difference between the "hot" and "cold" states of the pond. Since the times of the maximum and minimum system electrical demand are typically nearly coincidental with the times of the maximum and minimum ambient temperatures. This assumption also has the effect of maximizing the benefits of the TSP with regard to energy replacement cost savings. These assumptions reduce the set of operational variables to a single variable — the length of pond operation.

The number of continuous design variables also can be reduced by assuming that the plug-flow flushing time of the pond ($V/_Q$) is equal to the period of pond operation. Ponds with a storage volume greater than that required to contain the flow during the heatup or cooldown mode would be wasteful. Operation of the pond for periods longer than the plug flow flushing time may have some additional benefit. However, since the empirical thermal performance model derived from the experiments reported in Chapter 4 is only valued up to the plug-flow flushing time the above assumption is retained. The probable additional benefit of extended operation is nevertheless examined in Chapter 6 based on a grossly simplified pond thermal-hydraulic model. Additionally, the bypass flow fraction has not been included as a decision variable in the case studies in order to minimize the computational effort. The bypass flow is set to zero for all the design studies reported in Chapter 6 except that addressing the economic significance of non-zero bypass flow. The bypass flow fraction is defined as that fraction of the tower discharge flow routed directly from the tower discharge to the condenser inlet during the combined system operational modes.

5.3.3 Optimization Methods for TSP-Dry Tower-Plant Design

As has been noted in the previous section a nested optimization procedure is utilized in determining the least-cost TSP/dry tower system. A simple grid search is utilized for the integer variable part of the problem, but the grid method is not attractive for the continuous design variable part due to the large number of function evaluations which would be required.

The more efficient direct-search methods for handling multivariable, constrained, nonlinear problems which have found application to engineering problems are based on multivariable unconstrained methods combined with techniques for handling the constraints. The basic exploration technique in many of these unconstrained methods involves sequencial single direction searches guided by successes as they are achieved [F4]. Some methods (which are not properly classified as direct search methods) are based on finite-difference evaluation of the gradient of the non-analytic objective function. However, caution must be exercised in using these methods since truncation and/or concellation errors may lead the search astray so that it converges very slowly or not at all [S5]. Additionally, the present case may be particularly unsuited to such schemes since the evaluation of the objective function will in part be based on the use of tabluar data to approximate continuous functions.

The techniques for handling the constraints for the direct search methods are grouped in two categories -- feasibility check and modified objective function. The constrained decision variable problems are computed in a manner identical to the unconstrained problem for the feasibility check methods except that, if the search leads to non-feasible space, the search is redirected to the feasible region in a prescribed manner. The modified objective function techniques incorporate the constraints into the objective function thus producing an unconstrained problem. With regard to these methods, preference has been indicated for interior penalty or barrier functions in which the objective function is modified within the feasible region and all non-feasible points are rejected as search failures [S5].

5.3.4 Selection of Optimization Method

The direct search method of Rosenbrock has proved to be particularly successful in solving constrained optimization problems and has been selected for use in determining optimal TSP-Dry tower-plant system designs. This method which handles the constraints via objective function modification and which does not require derivative evaluations has been noted to be fairly slow with respect to more advance techniques but has proven to be very reliable and has seen widespread use in practical engineering applications [D1]. Since it is expected that the present problem will be characterized by broad optima and since close convergence on the true optimum will not be required, the disadvantages resulting from this relative slowness should be more than offset by the method's robustness and reliability.

The Rosenbrock method consists of sequencial explorations along a set of orthogonal directions which are rotated periodically according to the results of the explorations. Each rotation is prescribed such that the successive sets of directions generated converge on the principal axes of the quadratic function which approximates the objective function in the neighborhood of the optimum. The constraint handling strategy is based on the use of barrier functions within narrow boundaries of the constraints to modify the objective function. The function is altered in such a way that the modified function has a minimum in the boundary region. The method requires the selection of a starting point which is in the feasible region but not within one of the boundary zones.

Kuester and Mize [K9] have detailed the Rosenbrock method algorithm and presented a FORTRAN computer program for its convient application. The algorithm is presented in the next section.

5.3.5 Constrained Rosenbrock Method Algorithm

The basic unconstrained search method as outlined by Kuester [K9] is given first and then the necessary modifications for constrained problems given second. A logic diagram for the method is given in Fig. 5.10 and a two-dimensional illustration of the search procedure is discussed in Fig. 5.11.

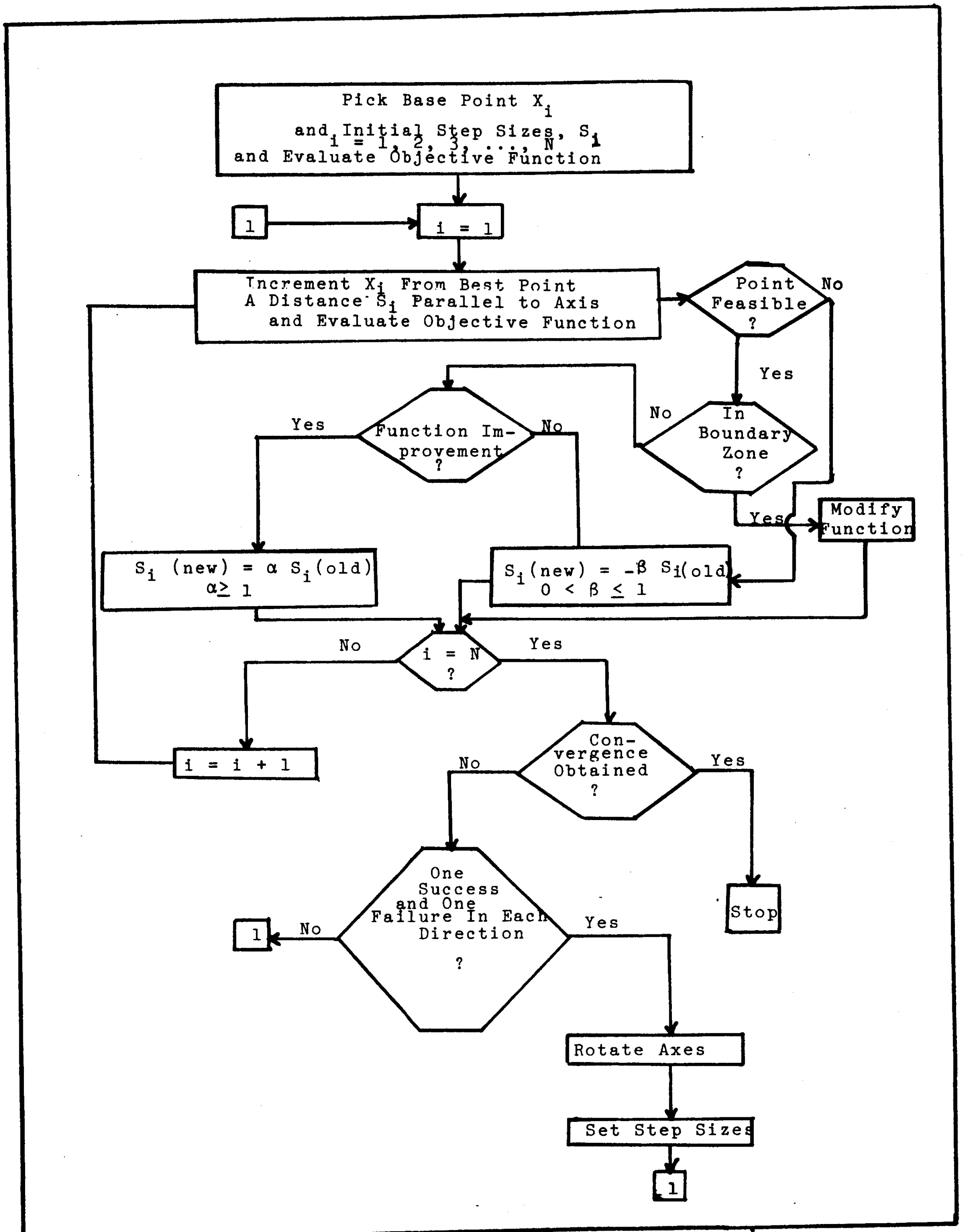


Figure 5.10. Constrained Rosenbrock Logic Diagram [K9]

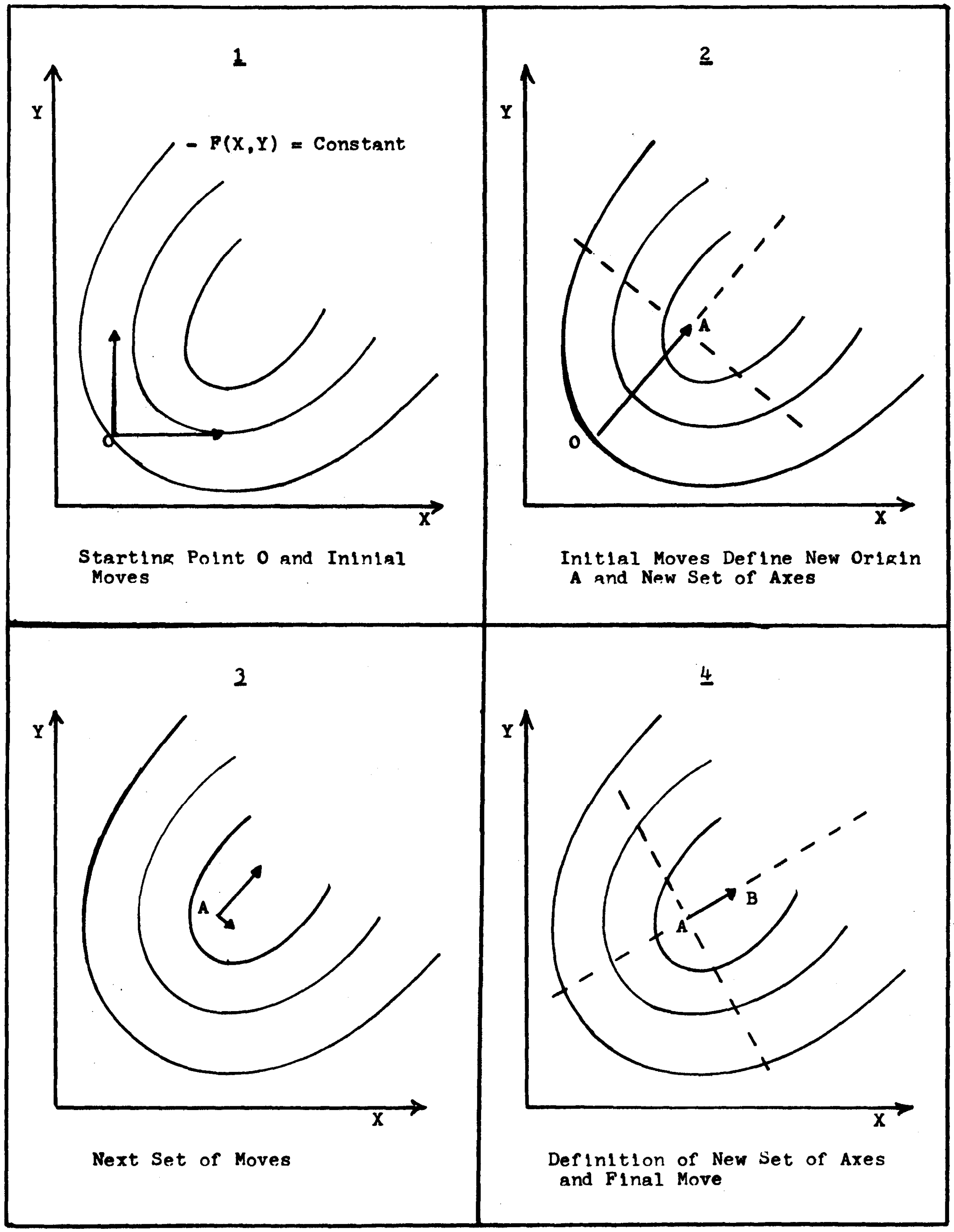


Fig. 5.11 2-Dimensional Illustration of Rosenbrock Method

It is worthwhile to point out that, in practice, the constraints on the design variables do not actively limit the search of the feasible design space. In this study, the flow rates and number of tower cells are constrained only to be non-negative. Consideration of design constraints may be important in later studies and thus the constraint handling procedure is included.

5.3.5.1 <u>The Unconstrained Search Routine</u>

1) A starting point and initial step sizes, S_i, i = 1, 2, ..., N, are picked and the objective function evaluated.
2) The first variable X_1 is stepped a distance S_1 parallel to the axis, and the function evaluated. If the value of F decreased, the move is termed a success and S_1 increased by a factor α, $\alpha \geq 1.0$. If the value of F increased the move is termed a failure and S_1 decreased by a factor β, $0 < \beta \leq 1.0$, and the direction of movement reversed.
3) The next variable, X_i, is in turn stepped a distance S_i parallel to the axis. The same acceleration or deceleration and reversal procedure is followed for all variables in consecutive repetitive sequences until a success (decrease in F) and failure (increase

in F) have been encountered in all N directions.

4) The axes are then rotated by the following equations. Each rotation of the axes is termed a <u>stage</u>.

$$M_{i,j}^{(k+1)} = \frac{D_{i,j}^{(k)}}{\left[\sum_{\ell=1}^{N} (D_{\ell,j}^{(k)})^2\right]^{1/2}} \qquad (5.10)$$

where

$$D_{i,1}^{(k)} = A_{i,1}^{(k)} \qquad (5.11)$$

$$D_{i,j}^{(k)} = A_{i,j}^{(k)} - \sum_{\ell=1}^{j-1} \left[\left(\sum_{n=1}^{j} M_{n,\ell}^{(k+1)} \cdot A_{n,j}^{(k)}\right) \cdot M_{i,\ell}^{(k+1)}\right], \quad j=2,3....N \qquad (5.12)$$

$$A_{i,j}^{(k)} = \sum_{\ell=j}^{N} d_\ell^{(k)} \cdot M_{i,\ell}^{(k)} \qquad (5.13)$$

where

i = variable index = 1, 2, 3, ..., N

j = direction index = 1, 2, 3, ..., N

k = stage index

d_i = sum of distances moved in the i direction since last rotation of axes

$M_{i,j}$ = direction vector component (normalized).

5) Search is made in each of the X directions using the new coordinate axes:

$$\text{new } X_i^{(k)} = \text{old } X_i^{(k)} + S_j^{(k)} \; M_{i,j}^{(k)} \qquad (5.14)$$

6) The procedure terminates when the convergence criterion is satisfied.

5.3.5.2 Constrained Variable Modifications

The search computations are the same for the constrained method escept that after each function evaluation, the following steps are required:

1) Define by F^o the current best objective function value for a point where the constraints are satisfied, and F^* the current best objective function value for a point where the constraints are satisfied and in addition the boundary zones are not violated. F^o and F^* are initially set equal to the objective function value at the starting point.

2) If the current point objective function evaluation, F, is worse than F^o or if the constraints are violated, the trial is a failure and the unconstrained procedure is continued.

3) If the current point lies within a boundary zone, the objective function is modified as follows:

$$F(\text{new}) = F(\text{old}) - (F(\text{old}) - F^*)\;(3\lambda - 4\lambda^2 + 2\lambda^3)$$

where

$$\lambda = \frac{\text{distance into boundary zone}}{\text{width of boundary zone}}$$

$$= \frac{G_k + (H_k - G_k)\cdot 10^{-4} - X_k}{} \quad \text{(lower zone)}$$

$$= \frac{X_k - (H_k - (H_k - G_k)\cdot 10^{-4})}{(H_k - G_k)\cdot 10^{-4}} \quad \text{(upper zone)}$$

At the inner edge of the boundary zone, $\lambda = o$, i.e., the function is unaltered (F(new) = F(old). At the constraint, $\lambda = 1$, and thus F(new) = F*. Thus the function value is replaced by the best current function value in the feasible region and not in a boundary zone. For a function which improves as the constraint is approached, the modified function has an optimum in the boundary zone.

4) If an improvement in the objective function has been obtained without violating the boundary zones or constraints, F* is set equal to F^o and the procedure continued.

5) The search procedure is terminated when the convergence criteria is satisfied.

5.4 The MITDAS Code

A FORTRAN computer program has been written and utilized to perform the case studies reported in Chapter 6. The MITDAS code (Model for the Investigation of the Thermal Storage Pond/Dry Cooling Tower Advanced Heat Rejection System) is listed with sample output data in Appendix B. All important assumptions incorporated in the code are discussed in this chapter. Details of the program's use are found in Appendix B.

CHAPTER 6

ECONOMICS OF TSP/DRY COOLING TOWER WASTE HEAT REJECTION SYSTEMS

6.1 Approach to the Problem and Assumption

A survey of the economics of combined thermal storage pond/dry cooling tower waste heat rejection systems has been completed using the TSP/dry tower-plant simulation design-optimization model MITDAS described in Appendix B. This effort has been directed mainly towards a comparison of the economics of the combined TSP/dry cooling tower system with the economics of the simple dry cooling tower system

Throughout these studies, a consistent set of economic parameters which should reasonably reflect present day costs have been employed. Nevertheless, the uncertainties in performing economic studies on undeveloped and unconventional technology are fully recognized and an approximate examination of the sensitivity of the results to the basic economic assumptions has been attempted. The details of the basic economic assumptions utilized in the studies are given in Table 6.1a. The dry cooling tower, pumps, and piping costs are based on costs reported in WASH 1360 [H4]. The costs of the thermal storage pond are estimated from references [C7], [R7], and [K2]. The cost breakdown of the TSP is given in Table 6.1b.

Table 6.1a

Cost Assumptions Used in Economic Studies

I. Dry Cooling Tower *	
1) Unit Tower Cell Cost	\$165,000/cell
2) Pump and Piping Cost	\$37,500 * R_1^2 * R_2
II. Thermal Storage Pond	
1) Covered Pond	\$515,000 + \$0.175 * VOLUME(ft^3)
2) Open Pond	\$515,000 + \$0.100 * VOLUME(ft^3)
III. Capability and Energy Replacement	
1) Electrical Generation Capability Replacement	\$150/KWe
2) Energy Replacement	
a) Daily Minimum	4.0 mills/KWHr
b) Daily Maximum	20.0 mills/KWHr
3) Utility System Electrical Load	
a) Daily Minimum	3 AM
b) Daily Maximum	2 PM
IV. Annual Fixed Charge Rate	15%
V. Plant Capacity Factor	0.75

R_1 = Ratio of flow rate (ft^3/sec) in system to that in reference system (Q/1154)
R_2 = Ratio of number of Tower cells to number of Tower cells in reference system (#/141)
VOLUME = volume of TSP based on design recommended in Chapter 4
All costs in current dollars (1976)
* Physical parameters of Tower cell described in Appendix B

Table 6.1b

Thermal Storage Pond Capital Costs

I.	Fixed Costs	
	a) Valves	\$25,000
	b) Piping	\$100,000
	c) Discharge Structure	\$100,000
	d) Withdrawal Structure	\$100,000
	e) 8 Barriers	\$140,000
II.	Stored-Volume Dependent Costs	
	a) Excavation	$\$0.05/ft^3$
	b) Lining	$\$0.05/ft^3$
	c) Cover	$\$0.075/ft^3$

6.2 Base Case Study

6.2.1 Base Comparison

The plant-site for the base comparison between the use of "combined" TSP/dry tower cooling and "simple" dry tower cooling is as follows:

1) Meteorology -- Winslow, Arizona, 1974
   Weather Service Observatory

2) Plant -- 3000 Mwt Boiling Water Reactor with a conventional nuclear steam turbine. (Curve C, Fig. 5.5)

Optimum systems have been designed for both the "combined" and "simple" heat rejection systems. The TSP thermal-hydraulic model is based on Fig. 4.30 and is as follows:

If $t < 0.63(\frac{V}{Q})$,

$$T_{po} = T_{pi} \quad \text{or,} \tag{6.1}$$

If $t > 0.63(\frac{V}{Q})$,

$$T_{po} = T_{pi} + 0.33\,(T_o - T_{pi}) \tag{6.2}$$

where

V = pond volume (ft^3),

Q = flow rate (ft^3/sec),

t = time since initiation of pond operation,

T_{po} = TSP outlet temperature,

T_{pi} = initial temperature of TSP, and

T_o = pond inlet temperature at $t = 0$.

In the case of the combined TSP/dry tower system the cost of the pond covering has been included but no credit was taken for heat transfer from the covered surface. A description of the optimized designs for the base plant-site is given in Table 6.2.

In comparing the "combined" and "simple' dry cooling systems some important observations and deductions are the following:

1) utilization of a TSP results in a net dry cooling savings of about 15%,
2) the greatest cost saving obtained through the use of a TSP is the saving in the capability replacement cost,
3) the tower size and circulating water flow are very similar for both systems,
4) the ratio of the economic benefit of the pond to the cost of the pond is approximately 4 to 1.

The "worst-ten-day" maximum loss of capability (defined in Sec. 5.2.2.2) for the "simple" system is 149.2 MWe while for the combined system the maximum loss is 70.2 MWe. The average capability loss during the heatup mode is somewhat less (about 65 MWe(with the maximum loss occurring near the end of the heatup mode due to partial short-circuiting of the pond. Figure 6.1 illustrates the behavior of the sub-optimal cost of the TSP/dry tower system as a function of the length of pond

Table 6.2

Comparison of Optimized Designs for "Combined" and "Simple" Systems for the Basic Plant/Site

	"Combined" System	"Simple" System
I. System Design Parameters		
a) Number of Tower cells	165	176
b) Flow rate during uncoupled modes(ft^3/sec)	1225	1237
c) Flow rate during coupled modes(ft^3/sec)	1233	-
d) Pond Area(acres)	30.6	-
e) Heatup Mode Startup Time	Noon	-
f) Heatup Mode Shutdown Time	6 PM	-
g) Cooldown Mode Startup Time	1 AM	-
h) Cooldown Mode Shutdown Time	7 am	-
II. System Costs		
a) Initial Costs		
1) Dry Tower System (tower cells, pipes pumps)	\$35,400,000	\$38,400,000
2) Thermal Storage Pond	\$5,170,000	-
3) Replacement Capability	\$10,500,000	\$22,400,000
b) Annual Costs		
1) Fan and Pump Operation	\$2,310,000	\$2,420,000
2) Replacement Energy	\$1,820,959	\$2,430,000
c) Incremental Cost of Dry Cooling (mills/kwhr)	1.71	2.04
d) Incremental Annual Cost of Dry Cooling	\$11,800,000	\$14,100,000

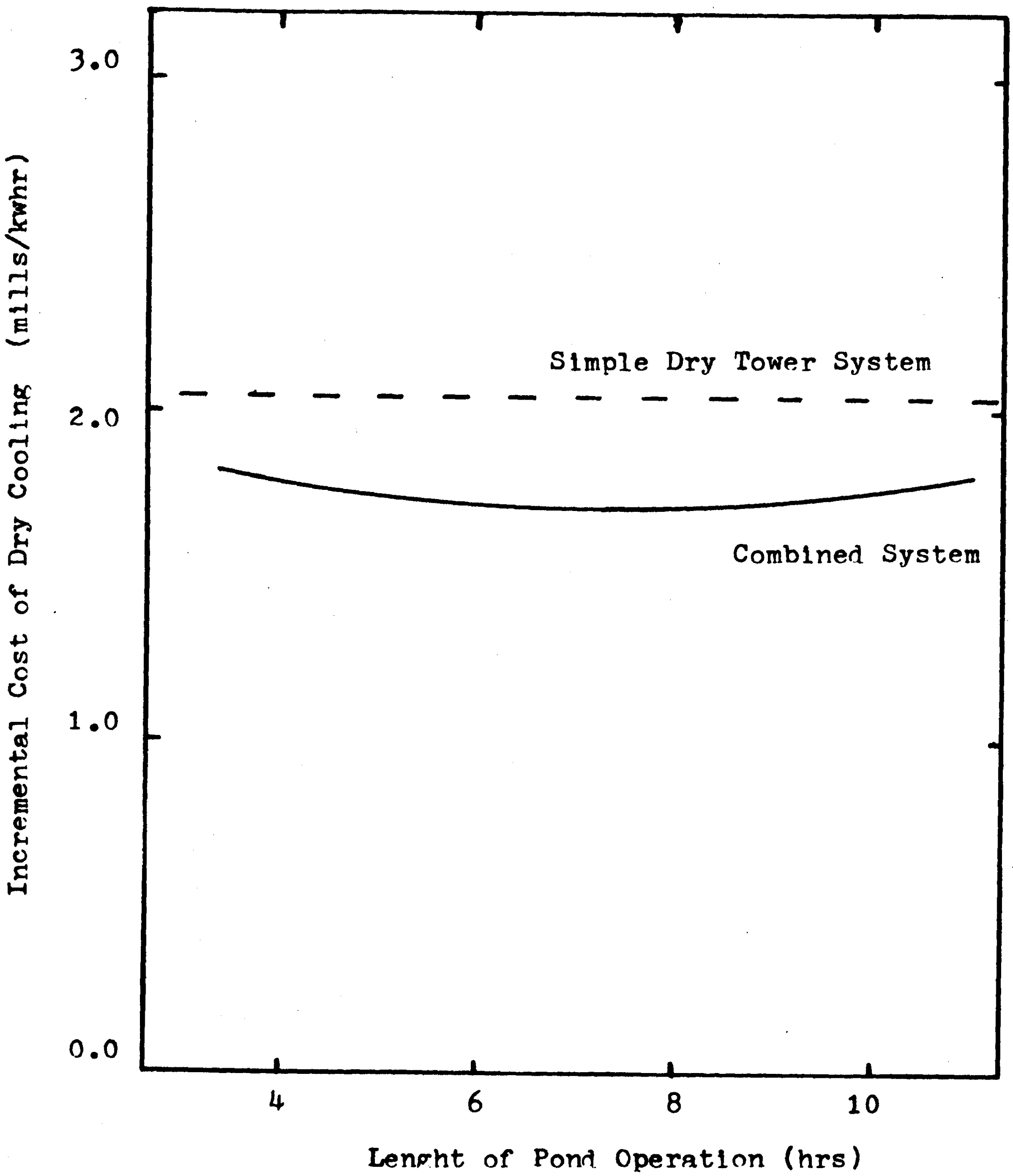


Fig. 6.1 Cost of Combined System as a Function of Pond Operation Period Lenght

operation (i.e. pond size). Note the relative insensitivity of the incremental cost of dry cooling to the size of the TSP. This broad minimum is to a degree a reflection of some of the simplifying assumptions build into the simulation model. Nevertheless the basic tradeoffs leading to a minimum can be generally characterized as follows:

increasing pond size $\rightarrow$ decreases capability replacement savings

increasing pond size $\rightarrow$ increases energy savings

increasing pond size $\rightarrow$ increases cost of pond

One assumption which tends to result in unrealistically low costs for long pond operation periods (say greater than 8 hours) is that the pond becomes fully-mixed during the standby modes of operation. For standby mode periods much less than 3 hours this assumption is not realistic. Another assumption which tends to result in unrealistically low costs at short operational periods is the assumption that the length of pond operation adequately covers the period of peak utility-system electrical demand. For pond operation times less than 2 hours this assumption seems unjustifiable.

6.2.2. Effects of Heat Transfer from the TSP

Heat transfer from the surface of the TSP aids to a limited extent in the rejection of waste heat. Radiative and conductive heat transfer will occur at the surface of a pond with a floating membrane cover. Additionally, evaporative heat transfer will occur if the pond surface is uncovered. Table 6.3 compares the total cooling costs for the two types of ponds with the base case. The savings obtained by using the uncovered pond amount to 20% of the total cost of "simple' dry cooling. However, taking into account the fact that a more expensive corrosion resistant tube material would be needed in the dry tower would, in part, nullify this apparent increase in savings. Also, no penalty has been assessed for the cost of water treatment and makeup water supply. The results for the reflective pond (all evaporation suppressed and all solar radiation reflected) indicates that the heat transfer from a covered pond will have a small effect on the total system economics.

6.2.3. Effect of Pond Bypass Flow

The potential benefit of bypassing a fraction of the tower discharge directly to the condenser inlet during both the "heatup" and "cooldown" modes of operation has been examined and the results are shown in Table 6.4 in comparison to the base case (bypass flow fraction = 0.0). Included in Table 6.4 is the benefit/cost ratio of the TSP for the different systems. Note that the relatively small capital expenditures for the 15.4 acre pond results in a substantial reduction in the incremental cost

Table 6.3

Significance of Heat Transfer from the Pond Surface

	Base Case (Adiabatic Pond)	Reflective Pond (Cover)	Exposed Pond (No Cover)
I. Incremental Cost of Dry Cooling $(\frac{mills}{kwhr})$	1.71	1.70	1.61
II. Heat Rejection from Pond Surface (% of Total)	0.0	1.8	3.3
III. Annual Water Consumption (Acre-feet)	0.0	0.0	673

Table 6.4

Effect of TSP Flow Bypass

		Bypass Flow Fraction*	
	0.0	0.25	0.50
1) Incremental Cost of Dry Cooling (mills/kwhr)	1.71	1.72	1.74
2) Pond Area (Acres)	30.6	23.0	15.4
3) Pond Cost	$5,170,000	$4,020,000	$2,870,000
4) Pond Benefit/Cost Ratio	4.0	4.6	5.8
5) Savings over "Simple" Dry Cooling	15.0%	14.4%	13.1%

All the above systems operate in the "Heatup" mode from Noon to 6PM and in the "Cooldown" mode from 1 AM to 7AM

* Fraction of Total circulating water flow routed directly from the tower to the condenser inlet during the coupled modes of operation

of dry cooling.

This result is important in that it indicates that a detailed design optimization of a TSP/dry tower system should include the bypass flow fraction as design variable. Qualitatively, the increase in the pond benefit/cost ratio with increasing pond bypass flow fraction can be attributed to more efficient utilization of the pond (i.e. greater difference between the temperatures of the "hot" and "cold" pond states).

6.2.4 Alternative Pond Utilization Schemes

The experimental model studies of the TSP thermal-hydraulic behavior have been limited to the extent that, once the pond became partially short-circuited, it was not possible to maintain the correct temperature of the discharge into the model TSP. Consequently, the empirical thermal-hydraulic model obtained from the experimental results is only valid over the pond operational time $\tau=0$ to $\tau = (\frac{V}{Q})$ or until the plug-flow flushing time. The base case has been formulated with this restriction in mind and thus does not consider the additional benefits which may be gained by operating the pond for a time longer than the plug-flow flushing time in either the heatup or cooldown modes.

The most obvious benefit of extended operation of the pond would be in decreasing the initial temperature of the pond at the beginning of the heatup mode of operation. This could be done by beginning the cooldown of the pond prematurely in the evening and thus allowing the bulk pond temperature to more closely approach the minimum ambient temperature occurring the next morning.

As a first approximation to the determination of the added benefit of an extended cooldown operation a simulation calculation was performed based on a fully-mixed TSP thermal-hydraulic model. The fully-mixed model was found to result in total dry-cooling cost predictions very near those resulting from use of the empirical pond model. Therefore, the extension of the fully-mixed model to pond operation times in excess of the plug-flow flushing time appears reasonable. Table 6.5 summarizes the results of the simulation calculation for an extended cooldown period based on the fully-mixed model. As indicated in Table 6.5 the added benefits are substantial and strongly suggest that careful consideration be given in the design of a prototype TSP to more complex operational schemes than that employed in the base case.

Table 6.5

Performance and Cost of TSP / Dry Tower System with Extended "Cooldown" Period

1)	Heatup Mode Startup Time	1 PM
2)	Heatup Mode Shutdown Time	5 PM
3)	Cooldown Mode Startup Time	10 PM
4)	Cooldown Mode Shutdown Time	7 AM
5)	Maximum Capability Loss	45.8 MWe
6)	Incremental Cost of Dry Cooling	1.62 mills/kwhr
7)	TSP Benefit/cost Ratio	6.7
8)	Savings over "Simple" Dry Cooling System	21%

6.2.5 Sensitivity of the Results to the Economic Assumptions

The quantitative economic results presented in the preceding section of this chapter are only as valid as the economic assumptions upon which they are based. Uncertainties in the economics used to evaluate the incremental cost of dry cooling systems is found in four areas. They are 1) the capital cost of the dry tower system, 2) the energy replacement cost, 3) the capablity replacement penalty, and 4) the capital cost of the thermal storage pond.

Examination of the sensitivity of the predicted benefit of the thermal storage pond (mills/kwhr) to changes in the cost of the dry cooling towers indicates that an increasing capital cost of the tower system (i.e. cost per unit heat rejection capability) favors an increasing TSP benefit. However, a 50% increase in the cost of the towers results in only an 8% increase in the pond benefit over that of the base case.

One of the largest uncertainties in the base economic model is associated with the energy replacement cost. The savings resulting from the utilization of a TSP, however, appears to be largely independent of the energy replacement cost as shown in Table 6.6. This can be attributed to the fact that, at low energy replacement costs, the capability replacement savings dominates the total savings resulting from the use of a TSP. Increasing the energy replacement

Table 6.6

Sensitivity of TSP Benefit to Energy Replacement Cost Variations

	Daily Range of Utility System Incremental Power Production Cost (λ) (mills/kwhr)		
	4 to 10	4 to 20	4 to 40
1) Daily Average λ	7	12	22
2) "Combined System Incremental Cost (mills/kwhr)	1.43	1.71	2.21
3) "Simple" System Incremental Cost (mills/kwhr)	1.70	2.04	2.67
4) Savings (mills/kwhr)	0.27	0.33	0.46
5) Percentage Savings	19%	16%	17%

cost increases both the cost of "simple" dry cooling and the "combined" system dry cooling with ratio of the two costs remaining about the same.

The uncertainity with regard to the evaluation of the capability replacement penalty lies not with the unit cost of replacement -- this is fairly well established at about $150/KW -- but rather with how the maximum loss of capability should be determined. The method for assessing the maximum loss for both the "simple" and "combined" system is discussed in detail in Sec. 5.2.2. A different approach which has been used in other studies of simple dry cooling is to assess the maximum loss at some design temperature condition. The design temperature is usually defined as some temperature which is exceeded (historically), on the average, a certain percentage of the time during the year [C5]. For the Winslow, Arizona site, 91 $^{\circ}$F is the dry-bulb temperature exceeded 5% of the time during the summer months. Using this temperature to evaluate the maximum loss capability for a conventional dry tower system results in a total cost of simply dry-cooling of 1.89 mills/KWHR for the base case. Using the same design temperature and an appropriate average minimum daily temperature to evaluate the maximum loss of capability for the "combined" system results in a total cost of dry cooling of 1.59 mills/KWHR. The total benefit of the TSP of 0.30 mills/KWHR under the above assumption is only slightly less than the 0.33 mills/KWHR benefit determined earlier for the base case.

Finally, it is clear that even if the cost of the TSP itself is substantially underestimated in this study the qualitative conclusion regarding its benefit is still valid. The calculated benefit/cost ratio (as large as 6.7) demonstrates that a considerably more expensive pond (say 100% greater) would still be economically attractive.

In summary, the qualitative and general conclusion that the combined TSP/Dry tower system is economically superior to simple dry tower systems is judged to be justified for the plant-site examined. For the base case of the Winslow, Arizona site meteorology and the modified conventional turbine plant the anticipated savings resulting from the use of a TSP falls in the range of 15 to 20% of the total cost of dry cooling.

6.3 Use of Conventional Steam Turbines With TSP/Dry Cooling Tower Systems

Conventional nuclear or fossil fired steam electric plants are not adaptable to dry cooling systems since in many areas the yearly maximum ambient dry bulb temperature would exceed the maximum allowable condenser inlet temperature. Thus before dry-cooling, as it is presently perceived, can succeed a new turbine design must be evolved as discussed in Sec. 5.1.4.

However, consideration of the use of a conventional steam turbine plant with a combined TSP/dry cooling tower system has revealed that at many sites the conventional turbine is economically preferable to any of the new proposed designs.

Table 6.7 summarizes the results of the design-optimization simulation calculation for the three types of nuclear steam turbines (discussed in Sec. 5.1.4 (for the base site.) The most important result contained in Table 6.7 is the near equality of the cost of TSP/dry tower cooling for both the conventional turbine and modified-conventional turbine. The total cost of an optimally-designed TSP/dry tower system is less than one-half of that of simple dry cooling for a conventional turbine plant but is also less than that of a modified conventional turbine using simple dry cooling. The benefit/cost ratio for the utilization of a TSP with a conventional turbine cooled by dry cooling towers is a dramatic 17.5. As expected, the thermal storage pond is of little benefit to the dry-cooled plant with a reduced-exhaust-annulus turbine since the heat rate of this proposed turbine design is relatively independent of the condensing temperature.

The conventional turbine simulation model includes a plant thermal-power derating option for use when the

Table 6.7

Performance and Cost of Alterntive Turbines with TSP/Dry Tower System

	Turbine Type(Nuclear)		
	Conventional *	Modified Conventional	Reduced Exhaust Annulus
1) Number of Tower Cells	173.	165	108
2) Pond Size(acres)	30.3	30.6	28.3
3) Maximum loss of Capability(MWe)	58.4	70.2	133.5
4) Capability Replacement Capital Cost	$8,700,000	$10,500,000	$20,090,000
5) Energy Replacement Cost(Annual)	$2,905,000	$1,820,000	$11,100,000
6) TSP Benefit/Cost Ratio	17.5	4.0	1.6
7) Incremental Cost of Simple Dry Cooling(mills/kwhr)	3.53	2.04	2.87
8) Incremental cost of TSP/Dry Tower Cooling(mills/kwhr)	1.72	1.71	2.80

* Credit Taken for Less Expensive Coventional Turbine = 0.13 mills/kwhr

condenser temperature exceeded the maximum temperature limit. For the Winslow, Arizona site derating would, at times, be necessary during the cooldown period in the early morning and during the pond-standby modes of operation. As evidenced by the maximum loss of capability for the conventional turbine plant given in Table 6.7, however, even on the hotest days of the year full thermal power could be achieved during the peak utility electrical load period.

The effect of variations of the base economic parameters (as is discussed in Sec. 6.2.5 with regard to the modified-conventional turbine system analysis) has been considered in connection with the results for the conventional turrine. Significant perturbations have been made in each of the basic unit costs as is detailed in Table 6.8 and the fundamental conclusion is unchanged.

6.4 Significance of Site Meteorology in Determining TSP Economics

6.4.1 Modified Conventional Steam Turbine Plants

In addition to the Winslow, Arizona site, four additional sites were examined with regard to the applicability of the TSP concept. The four sites are as follows:

Needles, California;
Billings, Montana;
Atlanta, Georgia; and
Boston, Massachusetts.

The Winslow, Arizona site was chosen since it repre-

Table 6.8

Economic Parameter Variation Sensitivity Study
for
Conventional Nuclear Turbine Plant

	Combined Cooling with Conventional Turbine	Simple Dry Cooling with Conventional Turbine	Simple Dry Cooling with Modified Conventional Turbine
1) 50% Increase in Cost of Dry Tower	2.03 mills/kwhr	4.66	2.41
2) 100% Increase in Cost of Pond	1.82	3.53	2.04
3) 50% Increase in Replacement Capability Cost	1.79	4.57	2.22
4) 50% Increase in Average Cost of Replacement Energy	1.99	3.85	2.17
5) "Base" Case	1.72	3.53	2.04

sents an arid region of moderately high summer temperatures (about 103 ^{o}F) with a substantial daily range. Needles, California is similar but the summer maximums are considerably higher (about 120 ^{o}F). Billings, Montana represents a cool climate with considerable variation of the ambient temperature during the summer. Atlanta, Georgia and Boston, Massachusetts typify coastal-northeastern and southeast meteorologies both of which are characterized by relatively small daily temperature variations. Table 6.9 gives the relative frequencies of occurrence of daily average temperature groups and corresponding daily ranges for the five sites for the year 1974.

For each of the sites the design-optimization calculation for both the "simple" and "combined" dry cooling systems have been performed. The results are shown in Table 6.10. The economic assumptions are identical for all these calculations and equal to the base case values. Figure 6.2 demonstrates the correlation between the economic savings resulting from the use of a TSP and the average daily temperature range for the ten hottest days of the year at each site. The economic savings do not correlate well with the yearly average tempeature range since, as in the case of Billings, Montana, the winter daily range (which does not greatly affect the economics) may be substantially less than the summer daily range.

Table 6.9

Temperature Frequency Distribution for Various Sites - 1974 Meteorology

Daily Average	Daily Range *	Winslow, Arizona	Needles, California	Billings, Montana	Atlanta, Georgia	Boston, Massachusetts	
0	15	22	1	104	12	90	Days/Year
to	30	57	3	17	6	6	
40	45	13	0	1	0	0	
40	15	21	75	81	61	127	
to	30	51	54	64	67	19	
60	45	48	0	4	10	0	
60	15	8	25	7	148	100	
to	30	87	59	76	65	18	
80	45	42	6	8	10	0	
80	15	0	22	1	4	4	
to	30	13	116	2	0	1	
100	45	3	4	0	0	0	

* 15 indicates range falls between 0 and 22.5^{o}F, 30 indicates between 22.5 and 37.5^{o}F, and 45 indicates 37.5^{o}F and above.

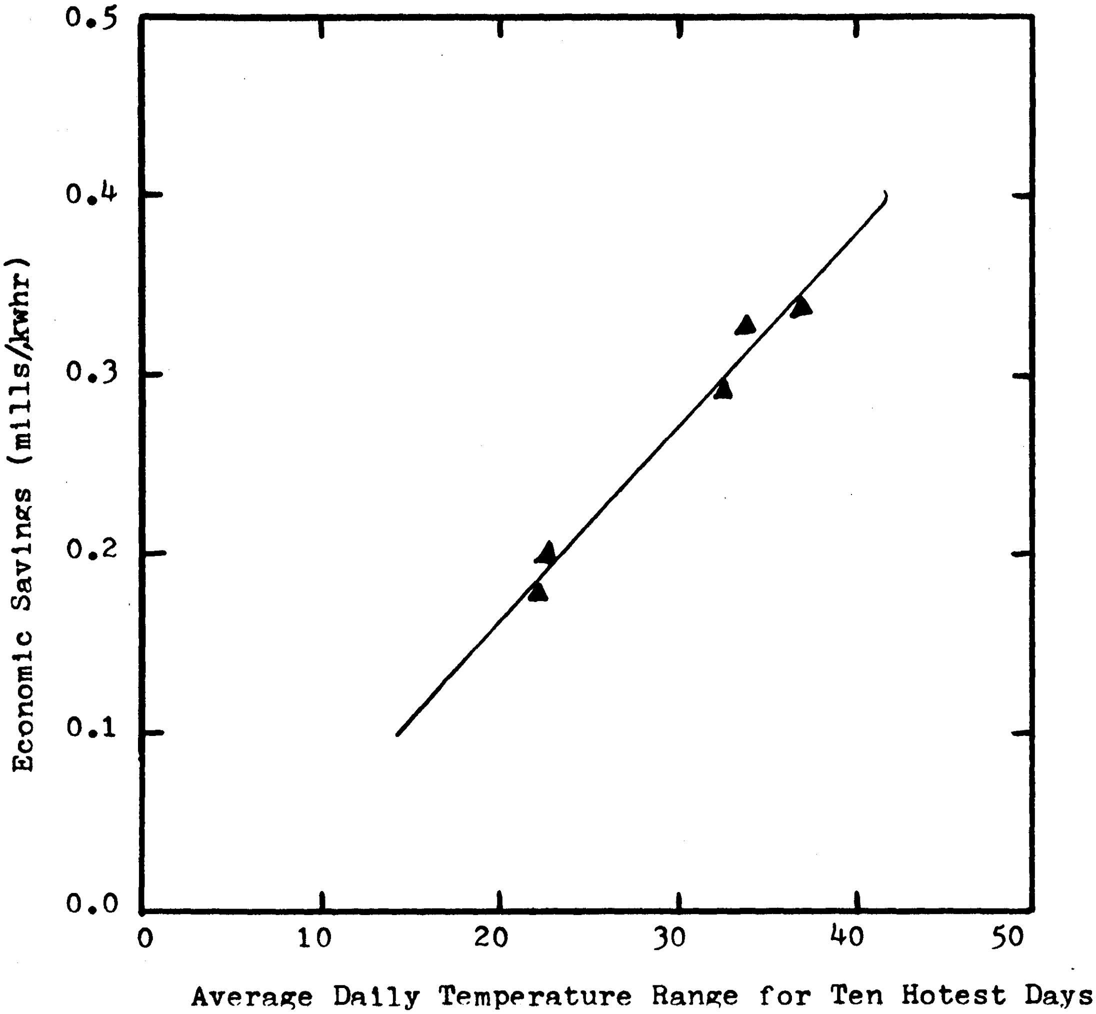


Fig. 6.2 Correlation of TSP/Dry Tower Savings with Average Ambient Temperature Range

Table 6.10

Cost Comparison of TSP/Dry Tower System and Simple Dry Tower System for Modified Conventional Nuclear Steam Turbine

	Combined TSP/ Dry Tower System	Simple Dry Tower System
1) Winslow, Arizona	1.71 mills/kwhr	2.04
2) Needles, California	2.33	2.66
3) Billings, Montana	1.50	1.79
4) Atlanta, Georgia	1.74	1.92
5) Boston, Massachusetts	1.53	1.73

The optimized designs for the "combined" systems for the different sites are similar to that determined for the Winslow, Arizona site in that there is very little difference in the number of tower cells and flow rates between the "combined" and "simple" dry tower systems.

6.4.2 Conventional Steam Turbine Plants

To investigate further the possible use of the TSP/dry tower system with a conventional nuclear steam turbine the MITDAS code has been used to determine the economics of TSP/dry tower cooling with a conventional nuclear turbine plant for the four sites listed in the previous section. The results are shown in Table 6.11.

In all cases, the use of a TSP results in a substantial savings. More importantly, however, the cost of TSP/dry tower cooling with the conventional turbine is nearly identical to that of TSP/dry tower cooling with the advanced turbine design and is less than that of simple dry cooling with the advance turbine design for all but one site. The exception, the Needles, California site, arose from the fact that, although the TSP/dry tower system could sustain full thermal power during the peak demand period, considerable plant thermal derating(up to 50%) is required at other times leading to excessive energy replacement

Table 6.11

Cost Comparison of TSP/Dry Tower System and Simple Dry Tower System for Conventional Nuclear Steam Turbine

	Combined TSP/ Dry Tower System	Simple Dry Tower System
1) Winslow, Arizona	1.72 mills/kwhr	3.53
2) Needles, California	2.82	6.86 #
3) Billings, Montana	1.51	2.72
4) Atlanta, Georgia	1.77	2.38
5) Boston, Massachusetts	1.55	2.40

Credit Taken for Less Expensive Conventional Turbine = $0.13 \frac{\text{mills}}{\text{kwhr}}$

Total Plant Shutdown During Hottest Summer Days Unaviodable Due to Extreme Ambient Temperatures (Approximately 120°F)

costs.

6.5. Use of a TSP With Alternative Plants and/or Dry Cooling Systems

6.5.1 Use of a Natural-Draft Dry Cooling Tower

A power station utilizing a natural - draft cooling tower will be benefited more by the use of a TSP than the mechanical-draft tower cooled plant. This fact is readily established by comparing the basic performance relations for the two-types of towers. For a fixed-design mechanical draft tower the rate of heat rejection is given by

$$q = c\ (ITD)^{1.0}$$

while for the natural draft tower of fixed design

$$q = c_1\ (ITD)^{1.33}$$

as previously discussed in Chapter 2. Now consider two dry tower facilities each of which rejects the same amount of heat at a 60 ^{o}F ITD. If the two towers are operated equal lengths of time a 40 ^{o}F and then 80 ^{o}F ITD (representing the "heatup" and "cooldown" modes) the natural-draft tower will reject a greater amount of heat to the atmosphere than the mechanical-draft tower.

This is simply due to the non-linear nature of the natural-draft tower performance equation which results in the average heat rejection over such a cycle being greater than that which would be calculated using the average ITD. Table 6.12 illustrates the comparative thermal performance

Table 6.12

Comparative Performance of Natural Draft Dry Tower/TSP System and Mechanical Draft Dry Tower/ TSP System

	Natural Draft Tower/TSP	Mechanical Draft Tower/ TSP
1) Heatup Mode Operation (Ambient Temperature=100°F)		
a) Condenser Inlet	106.5°F	110
b) Condenser Outlet	136.5	140
c) ITD	36.5	40
d) Tower Outlet	121.0	120
e) Tower Cooling Range	15.4	20
2) Cooldown Mode Operation (Ambient Temperature=70°F)		
a) Condenser Inlet	121.0°F	120
b) Condenser Outlet	151.0	150
c) ITD	81.0	80
d) Tower Outlet	106.5	110
e) Tower Cooling Range	44.6	40

ITD for steady-state heat rejection = 60°F , and the cooling range at 60°F ITD is 30°F for both systems of the two types of TSP/tower systems each of which is sized to yield a 30 ^{O}F water cooling range for a 60^{O} F ITD.

6.5.2 Use Of a TSP With Dry-Cooled Fossil-Fired Stations

One vendor of large steam turbines has marketed a turbine for dry-cooled fossil-fired plant applications [M7]. The heat rate curve for this turbine and that for the conventional fossil-fired plant turbine are shown in Fig. 6.3. Using the base economic and meteorological model an evaluation of the economics of utilizing TSP/dry cooling and simple dry cooling has been performed for both of these types of turbines. The results are shown in Table 6.13. The TSP is of little benefit to the plant with the high back pressure turbine design. However, there is a large incentive for using a TSP with the dry-cooled conventional fossil turbine as there is in the case of the conventional nuclear turbine. The relatively high cost of dry-cooling with the advanced turbine is due to its much higher heat rate even at low condensing temperatures.

TABLE 6.13

Use of TSP/Dry Tower System With Fossil Fueled Plants

	TSP/Dry Tower System	Dry Tower System
Conventional Turbine	0.976 mills/KWHR	1.74
High Back Pressure Design Turbine	2.50	2.65

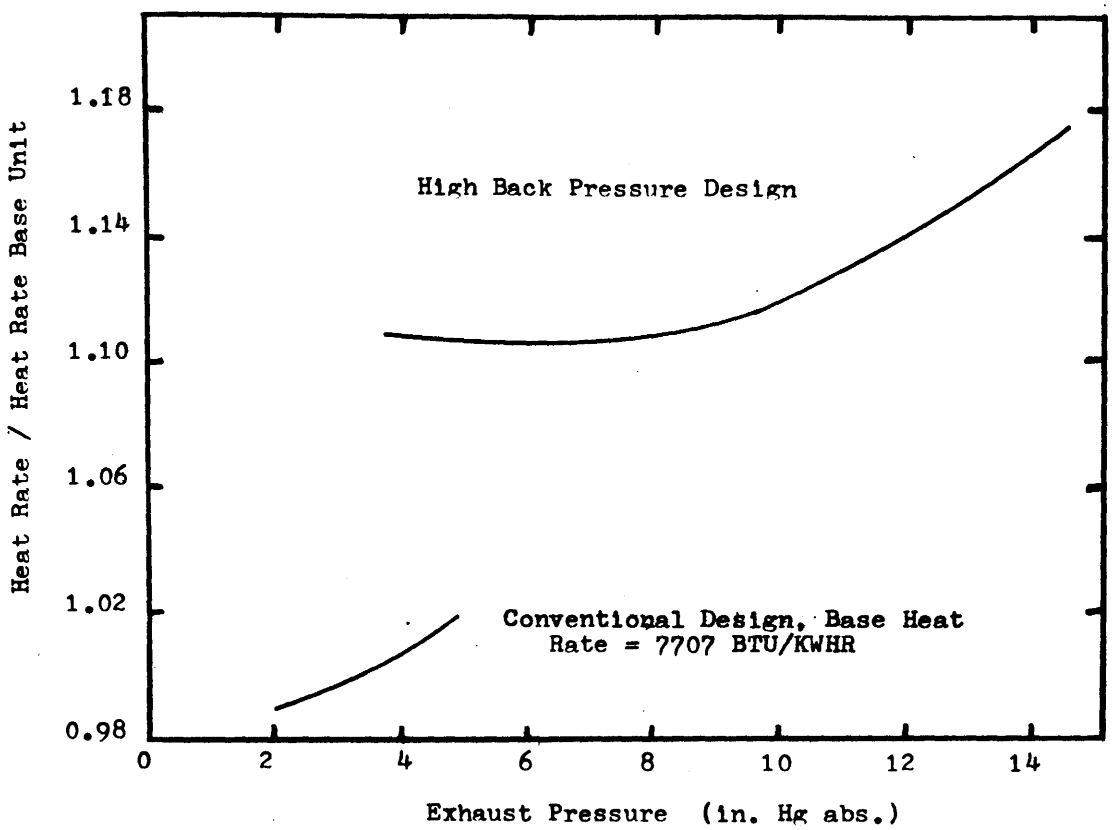


Fig. 6.3 Heat Rates for Fossil-Steam Turbines

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

7.1 Applicability of the Mixed-Mode Concept

A survey of mixed-mode waste heat dissipation systems aimed at improving waste heat system utilization economics indicates that no substantial benefit may be gained through the use of a combined system composed of component systems with different ratios of capital to operating cost. Thus, waste heat system utilization considerations should not be a basis for mixed-mode waste heat system design. Qualitatively this result can be attributed to the fact that optimized waste heat system designs typically call for "undersized" systems which result in partial loss of plant generation capability at high ambient temperatures. Excess heat rejection capability exists only at relatively low ambient temperatures and even at the low ambient temperatures the amount of excess cooling capacity is not large for evaporative systems since, simply stated, the performance of water-air evaporative heat exchangers becomes limited by the low water-vapor saturation specific humidity occurring at low temperatures. This conclusion that system utilization considerations should not provide a basis for system design also appears to be applicable to dry cooling tower systems.

One goal of this investigation of mixed-mode systems is to survey the use of mixed-mode systems which would extent limited cooling resources avialable on the plant site. Since this investigation of mixed-mode systems does not include a consideration of "once-through" cooling, the combined evaporative cooling tower / cooling pond system was selected as being representative of this application of the mixed-mode system concept. The conclusion derived from the examination of this system is simply that combined cooling pond / evaporative cooling tower waste heat rejection systems should be designed with a parallel water flow circuit with the ability to route a larger fraction of the flow to the pond during periods of cold weather. Any alternative component arrangement has not been seen to provide superior thermal performance. Thus, the parallel flow system results in the least total system cost since it minimizes the required pumping power for the water flow through the tower.

The major portion of this investigation of mixed-mode systems has involved an examination of cooling towers combined with several types of small, intermittently-used cooling and/or storage ponds. All these systems are proposed as solutions to the problem of coincidental occurrance of the maximum loss plant generation capability due to high ambient temperatures and the maximum utility system electrical demand. In all the proposed systems the pond serves as a source of condenser cooling water supply for the daily period of peak utility system electrical demand. The major difference among these systems is the method of cooling the pond. Three possible alternatives for evaporative cooling tower / intermittently-used pond systems which have been examined are the following:

1) the evaporative cooling tower / supplemental cooling pond system,

2) the evaporative saltwater cooling tower / supplemental cooling and makeup storage pond system, and

3) the evaporative cooling tower / thermal storage pond system.

All three types of ponds provide simultaneously a source of condenser cooling water and a storage reservoir for the tower outflow during the daily period of peak utility system electrical demand. In the first system the pond temperature is maintained at a desirable level simply as a result of heat transfer from the pond surface. The second system, the saltwater evaporative cooling tower / supplemental cooling and makeup storage pond system, is applicable only to coastal-sited plants and in this system the pond temperature is maintained at a low value by flushing the pond with tower makeup water and blowdown dilution flow obtained directly from the ocean. The potential benefit of both these systems has been demonstrated and indicates that these mixed-mode alternatives should be given consideration when contemplating the use of evaporative cooling systems. A detailed benefit-cost determination is not possible for these two systems due to the strong site dependency of the cost of constructing the required pond. Nevertheless, the 10 to 20 acre size of these ponds is noteworthy since it is small in comparison to the typical 1000 acre nuclear power plant site.

The third evaporative system combination, the evaporative cooling tower and thermal storage pond system, is considered in an attempt to take advantage of the daily variation in the ambient wet bulb temperature. In this system the pond is recooled each day by cycling the entire volume of the pond back through the condenser and cooling tower during the early morning hours of minimum daily ambient wet bulb temperature and minimum daily utility system electrical demand. A survey examination of the performance of this combined system indicates that the potential benefit of the system is small due to the small and highly variable daily range of the wet bulb temperature occurring at most locations in the United States. However, recognition of the large and consistent variation of the ambient dry bulb temperature in many of the arid regions of the country has led to a consideration of using a thermal storage pond in an exactly analagous manner with a dry cooling tower. A simple analysis of the combined thermal storage pond / dry cooling tower system and the plant power generation performance resulting from the cyclic operation of this waste heat rejection system reveals that this system offers great potentail for the solution of the problem of coincidental occurrance of maximum loss of plant generating capability and the maximum utility system electrical demand.

For this resason and the fact that this system is currently the only proposed solution to the performance problems of dry-cooled plants which does not require supplemental evaporative cooling it was concluded that this concept was worthy of a detailed analysis and developement effort. Indeed such and effort has been directed towards achieving an accurate determination of the system economics and a resolution of the engineering design problems which would be encountered in the practical implementation of the system.

7.2 Economics of Thermal Storage Pond / Dry Cooling Tower Systems

The cost-benefit of the TSP/dry cooling tower system for various plant types and site meteorologies has been ascertained by constructing a computer model of the basic TSP - dry tower - plant system and utilizing the model to simulate the dynamic thermal and power generation behavior of the system. An accurate description of the thermal-hydraulic behavior of the TSP is crucial to the prediction of the overall system performance and has been obtained from a physical model of the proposed TSP.

The initial design concept of the TSP, simply a long and narrow channel, proved inadequate due to strong density-induced flows which tended to short-circuit the pond. The flow stratification problem was resolved by installing a series of flow constricting barriers in the pond experimental model in order to induce vertical mixing. The recommended design results in a good approximation to the idealized case of plug-flow behavior. The structure, geometry, and operation of the recommended design appear to be compatable with available low-cost water impoundment technology.

Using the empirical thermal-hydraulic model of the pond which was deduced from the experiments, the economics of the TSP/dry tower system were evaluated on the basis of an incremental cost of dry cooling(mills/kwhr). The benefit of the TSP has been determined by the differnce between the total cost of optimized TSP/dry tower and optimized simple dry cooling. As expected, the benefit of utilizing a TSP depends greatly on the type of generating plant and the site meteorology. In general, a 15 to 20% savings in the cost of dry cooling with a modified-conventional nuclear steam turbine appears to be possible at sites with a large daily range of the ambient dry bulb temperature. This result is to a large degree dependent on the assumptions incorporated in the economic model with regard to loss of generation capability and loss of power production pricing. These assumptions, nevertheless, are judged to be reasonable and appropriate extensions of current waste heat system design practices. Sensitivity studies based on variations of the several economic parameters used in the model add confidence to at least the qualitative nature of these conclusions concerning the magnitude of the pontential benefit.

The most important conclusion of the economic studies is that TSP/dry tower cooling with a conventional nuclear steam turbine plant is, in many geographical areas, less costly than simple dry cooling with any of the proposed advanced turbine designs. The advance turbine designs have been proposed to allow for plant operation at the high condensing pressures which would be commonly experienced with simple dry cooling. Use of a TSP has shown that a condensing pressure less than the conventional steam turbine limit of 5 in.Hg can be maintained during the afternnon period of peak utility-system electrical demand even during the summer months. It is imperative to emphasize that this conclusion is strongly dependent on the site meteorology. However, the locations at which this conclusion is most appropriate are those regions where dry cooling is most attractice - the arid western regions of the United States. This conclusion concerning the economic practicality of dry cooling with a conventional nuclear steam turbine is important since the application of dry cooling to nuclear plants has been hindered by the commercial unavailability of a suitable turbine.

Two TSP construction options are possible - the covered and the uncovered pond. The covered pond would be decidedly more expensive since the pond cover would likely constitute a major part of the pond construction cost. An uncovered pond would enhance heat transfer from the pond surface but would cause degradation of the quality of the water inventory. In either case, the total heat transfer from the pond surface would be small in relation to the total heat rejected by the station. Review of the water impoundment technology suggests that synthetic rubber or polyvinylchloride membrane reservior lining and covering materials are adaptable to the proposed TSP.

Investigation of the possible secondary uses of the TSP in addition to its main function as an aid in condenser cooling indicates that a TSP will not provide a solution to the service water cooling problem of dry-cooled plants. Generally, the temperature of the pond would exceed the service water temperature criterion frequently during the summer. Evaluation of the TSP as a means of rejecting reactor core decay heat under reactor accident conditions does indicate that the TSP could be a reliable and effective means of meeting this cooling requirement. To date, the problem of long-term emergency cooling at dry-cooled nuclear power stations has not been addressed. A TSP can provide a low-temperature heat sink for a period of several months if necessary. Application of the TSP in this regard would help justify the required capital expenditure for the pond construction.

In the economic studies, a cost and thermal performance model of a mechanical-draft, multi-cell, dry cooling tower was used to model the tower system. Comparing the simple thermal performance models of mechanical and natural draft dry cooling towers indicates that use of a TSP in conjunction with a natural draft dry tower would yield somewhat larger economic benefits. Also, analysis indicates, as in the case of the nuclear steam plant, that the use of a conventional fossil-steam turbine with the TSP/dry tower system may be preferable to simple dry cooling with advance high-back-prssure fossil-steam turbines.

In designing the TSP/dry tower system the most important consideration is the correct sizing of the pond. The cost of the pond is approximately directly proportional to the stored volume of water. This is because the pond depth is fixed by thermal-performance requirements and the major cost of constructing the pond would be the cost of excavating, lining, and covering the pond. No additional pumps or pumping power would be required above that required for simple dry cooling systems and the cost of the additional piping, the pond discharge and withdrawal structures, and the pond mixing barriers are judged to be secondary to the cost of excavating, lining, and covering the pond. There is no requirement that the pond be constructed without bends in the flow circuit and thus a labyrinth type of pond appears attractive as it would minimize land usage and piping costs.

If a covered TSP is utilized, the dry cooling tower design need not be different from that of a conventional dry cooling system. However, if the TSP is open to the air and subject to evaporative losses use of corrosion resistant tube material in the dry tower may be required and some method of water treatment would be mandatory. The TSP/dry tower concept is predicated on the use of a conventional surface condenser. Application of the TSP concept to the proposed "Heller" dry cooling system which utilizes a jet condenser does not seem practical.

Careful consideration of the various feasible methods of operating the TSP/dry tower system is important in arriving at the most economic design. Specification of the optimal TSP/dry tower system necessitates the inclusion of operational variables in the design procedure in addition to the system design varialbles,such as component sizes and water flow rates, which are common to traditional waste heat system design procedures. The operational variables to be considered are the daily startup and shutdown times of the various modes of operation. Also, it has been recognized that diversion of the total flow from the tower to the pond during the coupled modes of operation may be economically unjustified. Thus, the fraction of the flow bypassing the pond and routed directly to the condenser during the coupled modes of operation should be treated as a basic system design varialbe.

As expected, the site-specific factor which is most significant in determining the economic attractiveness of the TSP concept is the daily range of the ambient dry bulb temperature. All other factors being equal, the economic benefit of the TSP increases with increasing average daily range of the ambient dry bulb temperature. The quantitative nature of the relationship between the ambient temperature range and the economic benefit is dependent on the type of steam turbine plant under consideration. Another site-specific factor which is important is the daily range of the incremental cost of electrical power production in the local utiltiy system.

7.3 The TSP / Dry Tower System Compared to Alternative Enhanced Performance Dry Tower Systems

To date, the most widely discussed method of enhancing the heat rejection performance of dry cooling towers is the addition of some evaporative cooling. Tower cells composed of both wet and dry parts with series and/or parallel air and water paths have been proposed as well as the utilization of separate dry and wet tower structures and the deluge cooling of the dry tower extended surface. If minimal water consumption is the primary motive for the use of a dry tower such systems are an unattractive compromise.

The TSP / dry tower system has been shown to be effective in maintaining desirable condensing temperatures without any evaporative cooling. A TSP with a liner and cover would theoretically result in no water loss. Filling of the pond could be accomplished over an extended period of time during the plant construction. For the case of the uncovered pond, evaporation from the pond would be at a near constant rate. Typically, the rate of evaporation for the Winslow, Arizona site examined in this study would be about 0.3% of the pond volume per day or about 220 GPM for a 15 acre pond.

The makeup flow of a supplemental evaporative cooling tower which would result in the same thermal performance improvement obtained with the TSP for the Winslow site would be approximately 5500 GPM. Admittedly, this is the maximum water flow demand. The total annual consumption would be about equal to that of the uncoverd TSP. Greater performance improvement than that obtained from the TSP would be possible through the use of supplemental evaporative cooling but the water consumption would increase markedly.

In short, it appears that the relative attractiveness of the TSP system as opposed to supplemental evaporative cooling will be dependent on the availability of makeup water for supplemental evaporative cooling or on the necessity of maintaining high water quality in the cooling system. A general translation of these constraints into economic terms is difficult. For sites at which no water consumption is allowable the covered thermal storage pond is the only option for improving the plant performance (other than increasing the size of the dry tower). For sites at which the consumptive use of water is limited to a small but constant flow rate (i.e. instantaneous flow demand and not total consumption is limiting) the uncovered TSP would find appropriate application. For sites with greater water availability the most economical system can only be determined by a detailed design and operation optimization taking into account the cost of water. A such sites the TSP system, the supplemental evaporative system, or some combination of the two systems would be feasible.

The combination of the TSP and supplemental evaporative cooling presents an interesting compromise. In this system the large instantaneous water flow demand for evaporative cooling could be met by the TSP which would also function to reduce the required amount of supplemental evaporative cooling.

7.4 Recommendations for Future Investigations of the TSP/ Dry Tower System Concept

Future developement of the TSP / dry tower concept should encompass the following tasks.

1) System Economics

Economic studies should be performed for proposed plant sites using loss of capability and energy replacement pricing which reflects actual anticipated costs of the local utility. Further, effort should be directed towards a detailed design of the thermal storge pond for these sites in order that a more exact correlation between the cost of the TSP and the stored water volume be established. The cost of water should be introduced into the economic model if uncovered ponds are considered.

2) Performance Modeling

The range of applicability of the empirical thermal performance relationship derived from the TSP experimental model should be extended to pond operational periods greater than the plug flow residence time of the pond. The assumptions of complete mixing during the standby modes of pond operation and the significance of startup thermal transients in the pond should be quantified. Correlation of the pond thermal performance with the pond geometry and the number and type of barriers would aid in the design of optimal TSP / dry tower systems.

3) Combined TSP - Supplemental Evaporative - Dry Tower Cooling Systems

The economics and design of combined thermal storage pond - supplemental evaporative cooling - dry cooling tower systems should be examined. In such systems the pond would serve as a storage device for both makeup water for the supplemental evaporative cooling and waste heat for the capacitive cooling system.

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Appendix A

SUMMARY OF EXPERIMENTAL RESULTS

Figures A.1 through A.8 in this Appendix summarize the results of the experiments for the determination of the empirical coefficient α defined in Sec. 4.2.2.2. Figures A.9 through A.21 summarize the results of the experiments concerning the effect of flow-constricting barriers on the thermal performance of the thermal storage pond. The parameters describing each experiment are defined below:

ΔT = Temperature difference between the discharge into the model and the initial temperature of the model (^{o}F)

Q = Flow rate in model (GPM)

D = Depth of water in model (inches)

L = Length of model (feet)

W = Width of model = constant = 1.5 ft.

T_r = Plug-flow flushing time of model = V/Q (min.)

V = Volume of water (gallons)

A description of the "horizontal" and "vertical" barriers mentioned in connection with Figures A.8 through A.21 is found in Sec. 4.3.

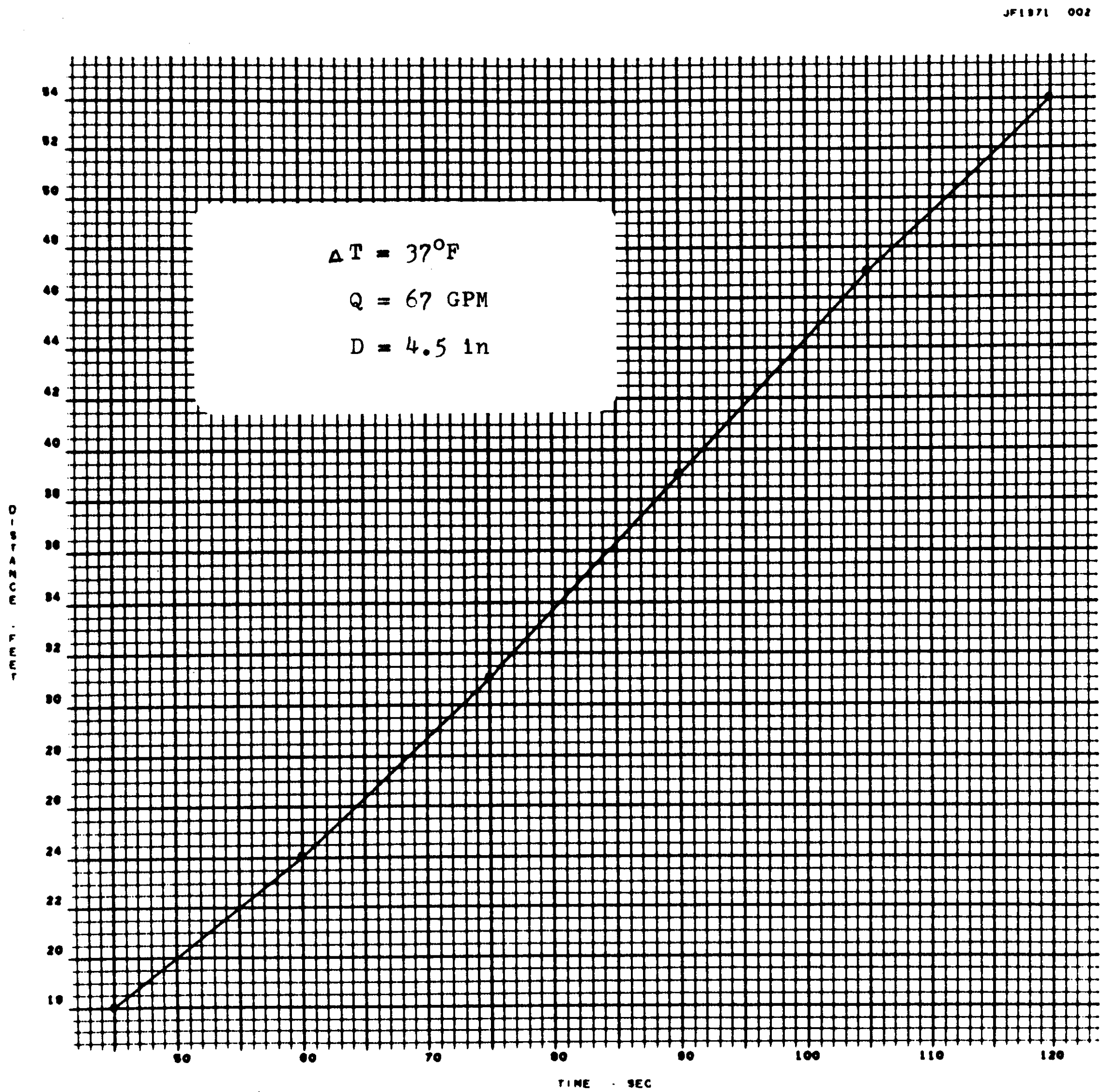

Fig. A.1

Position of Density Front
as a Function of Elapsed
Operational Time

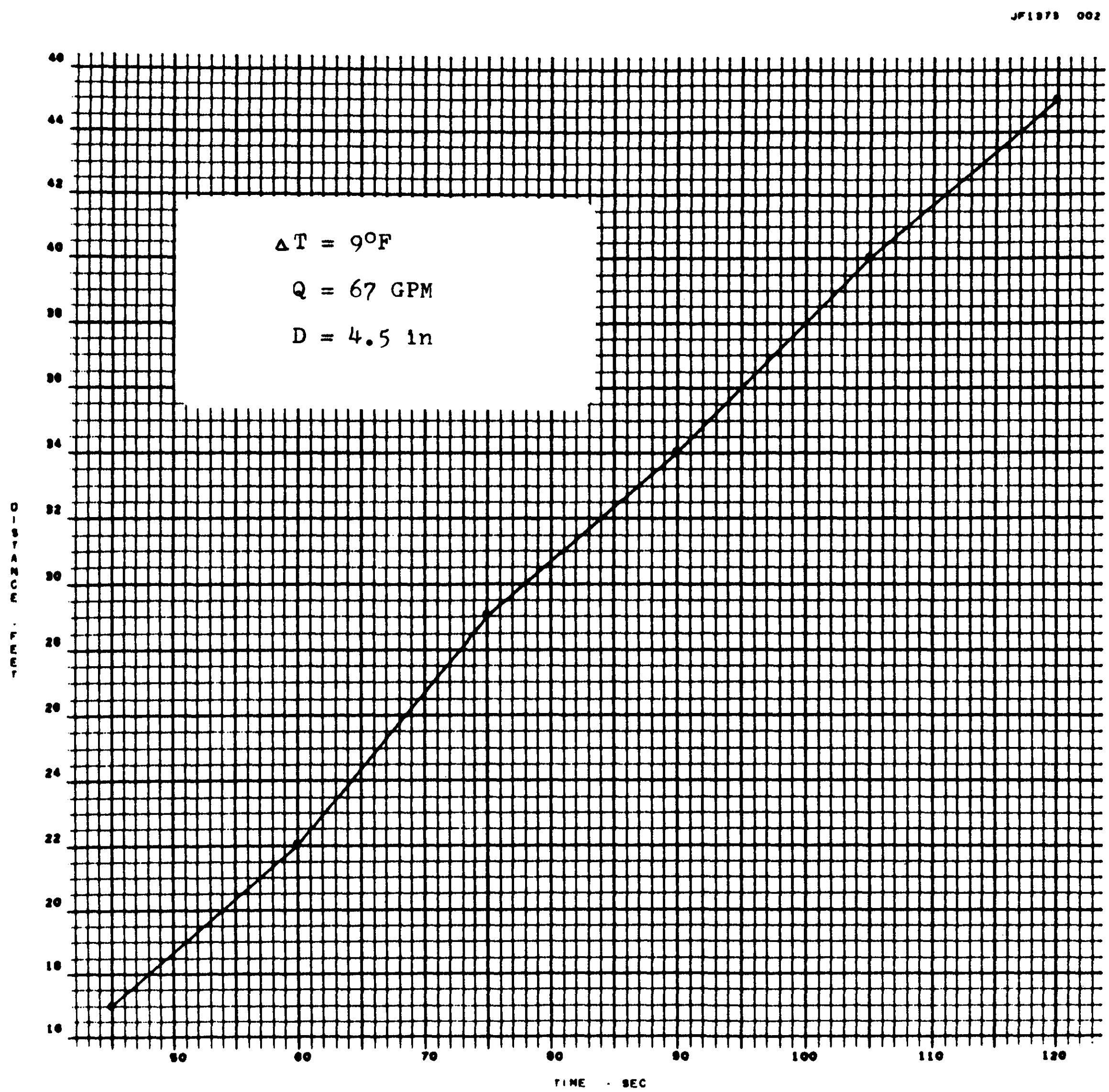

Fig. A.2

Position of Density Front as a Function of Elapsed Operational Time

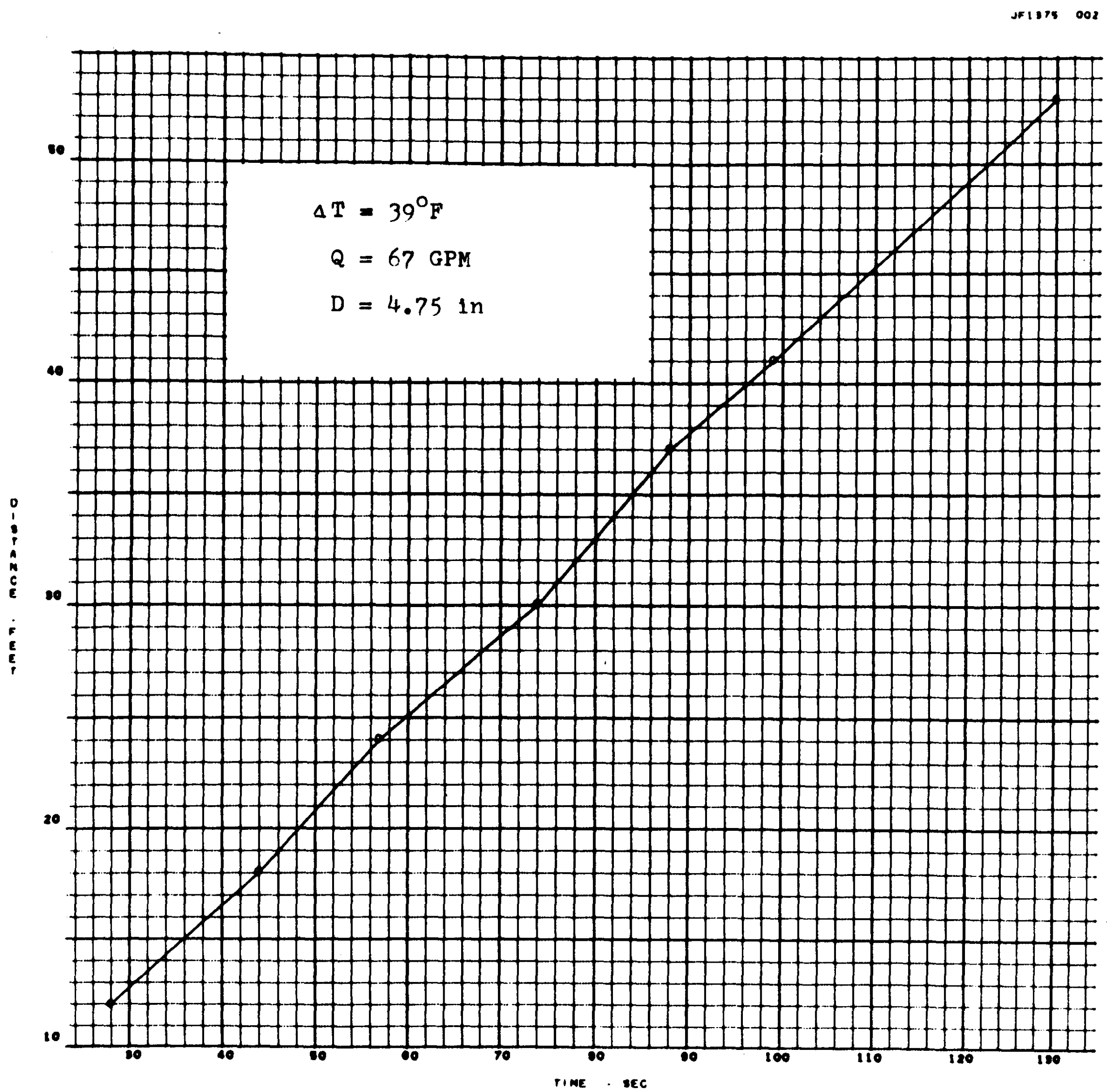

Fig. A.3

Position of Density Front
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Operational Time

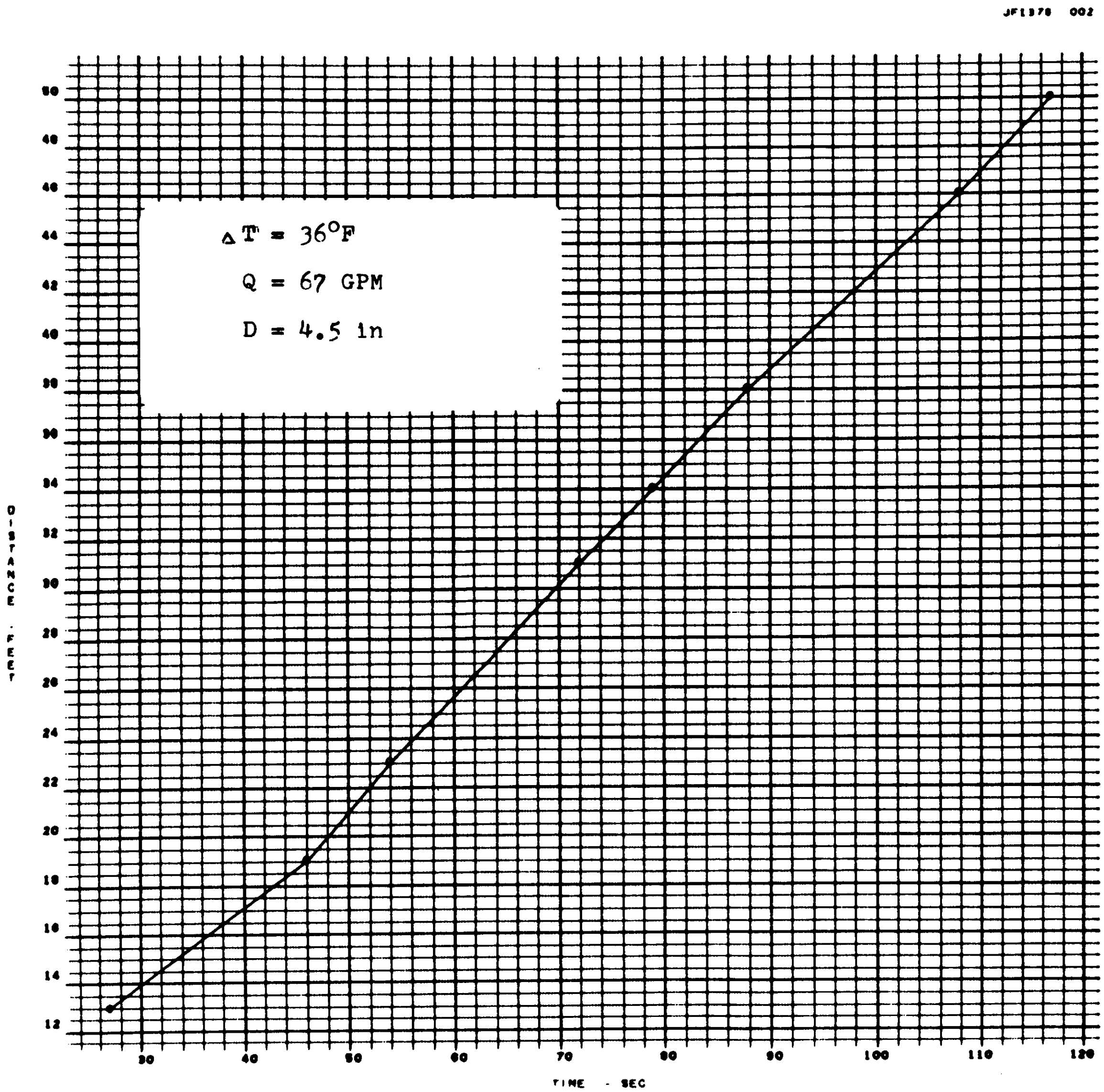

Fig. A.4

Position of Density Front as a Function of Elapsed Operational Time

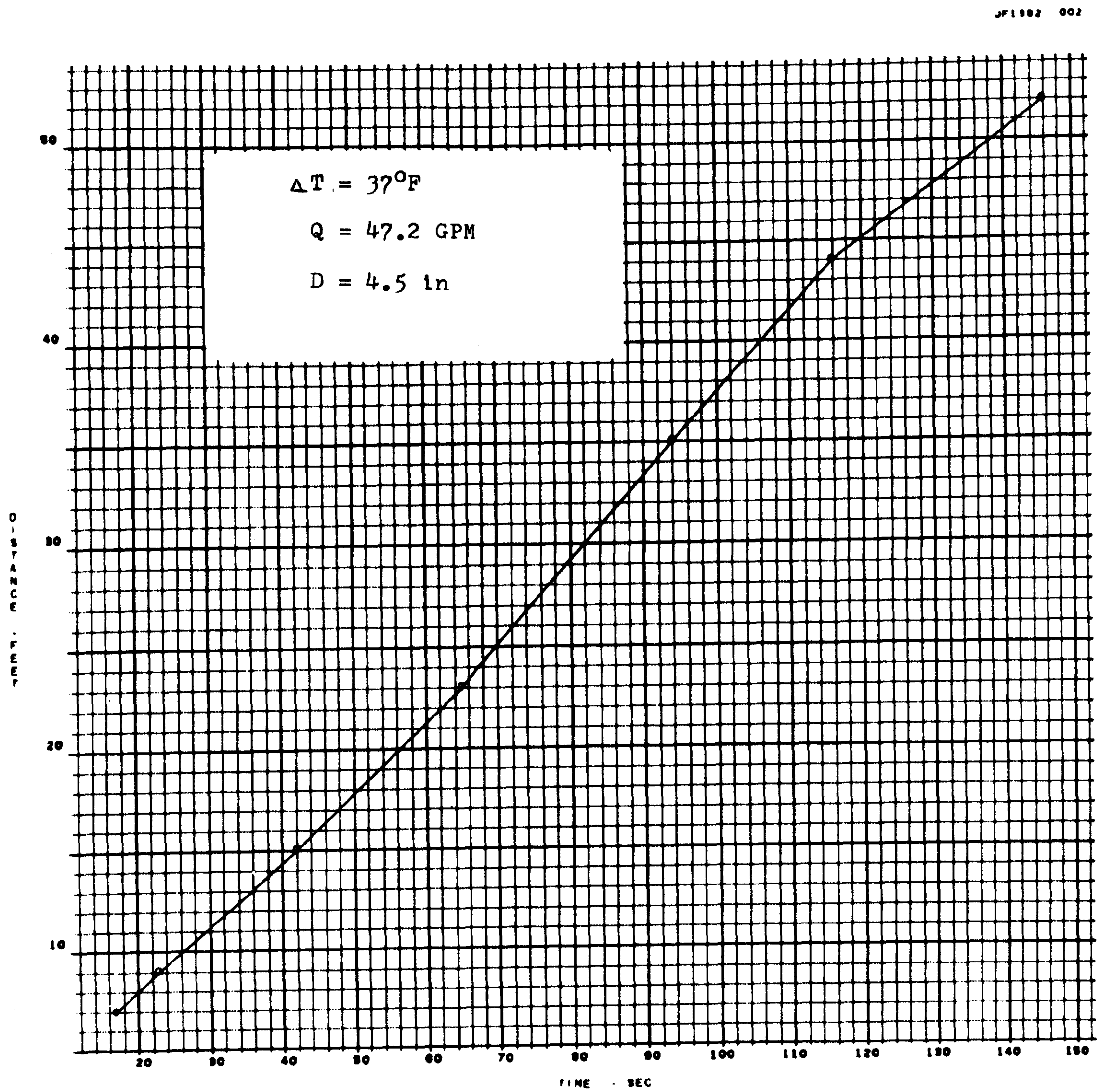

Fig. A.5

Position of Density Front
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Operational Time

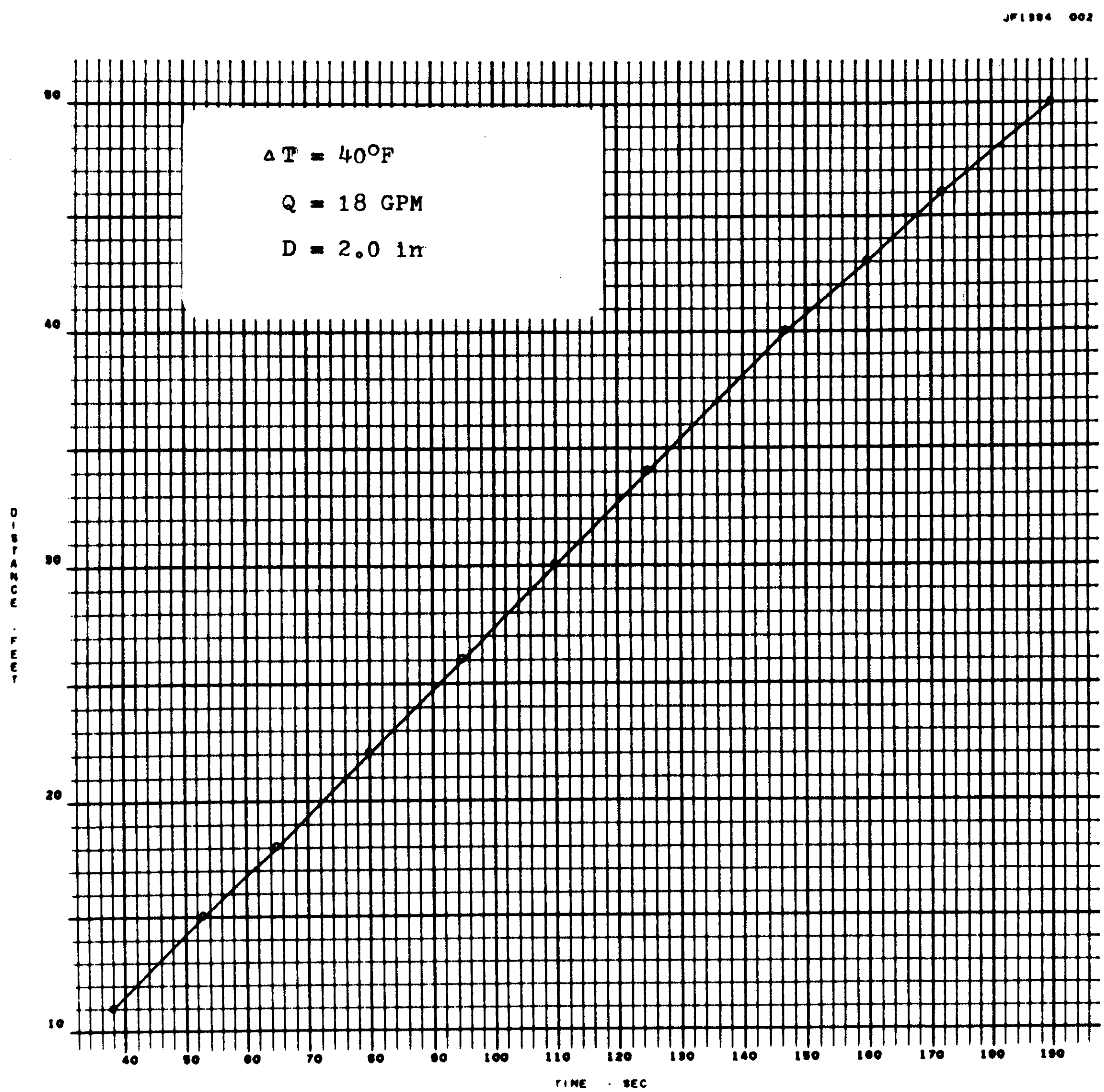

Fig. A.6

Position of Density Front as a Function of Elapsed Operational Time

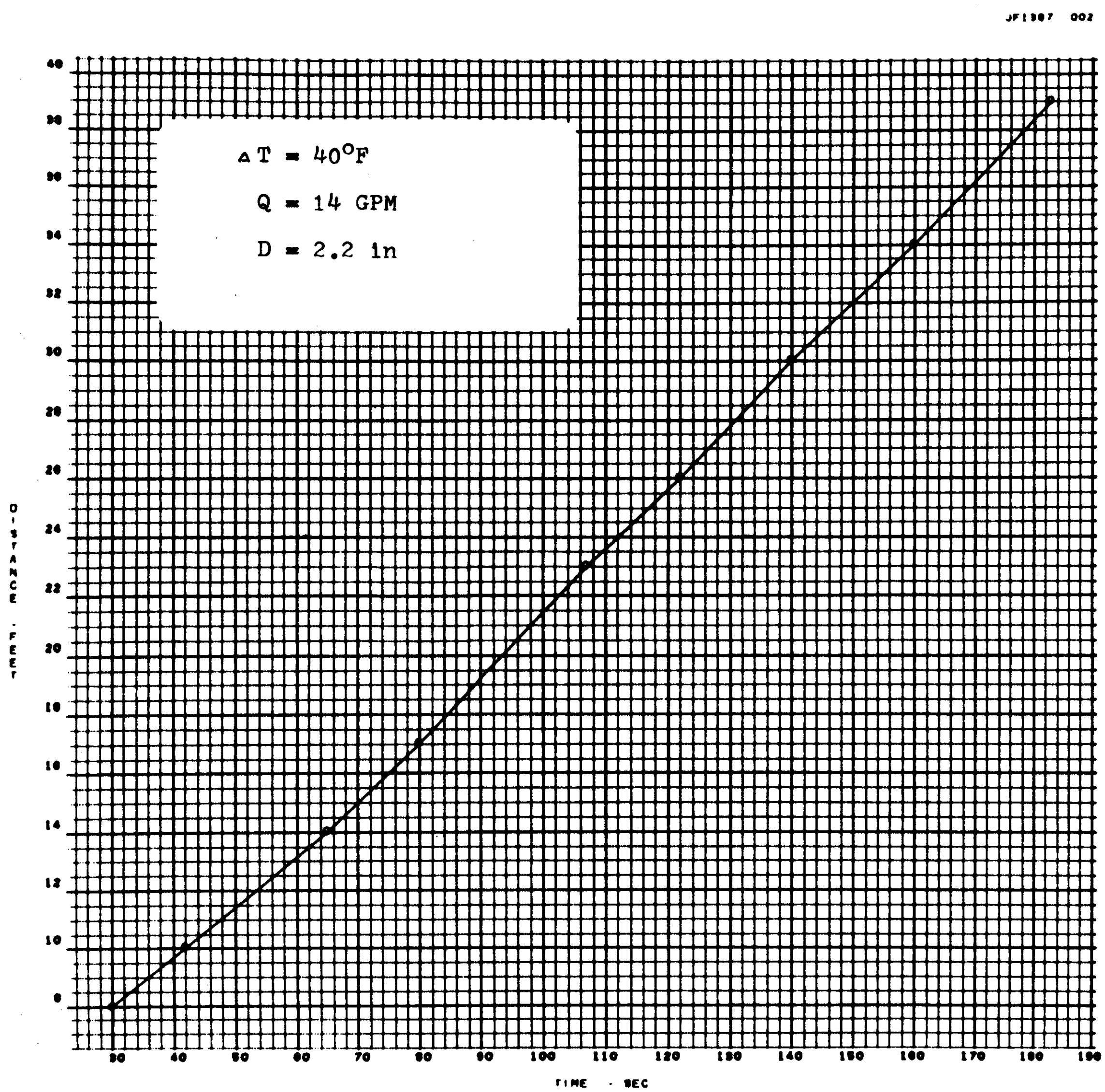

Fig. A.7

Position of Density Front as a Function of Elapsed Operational Time

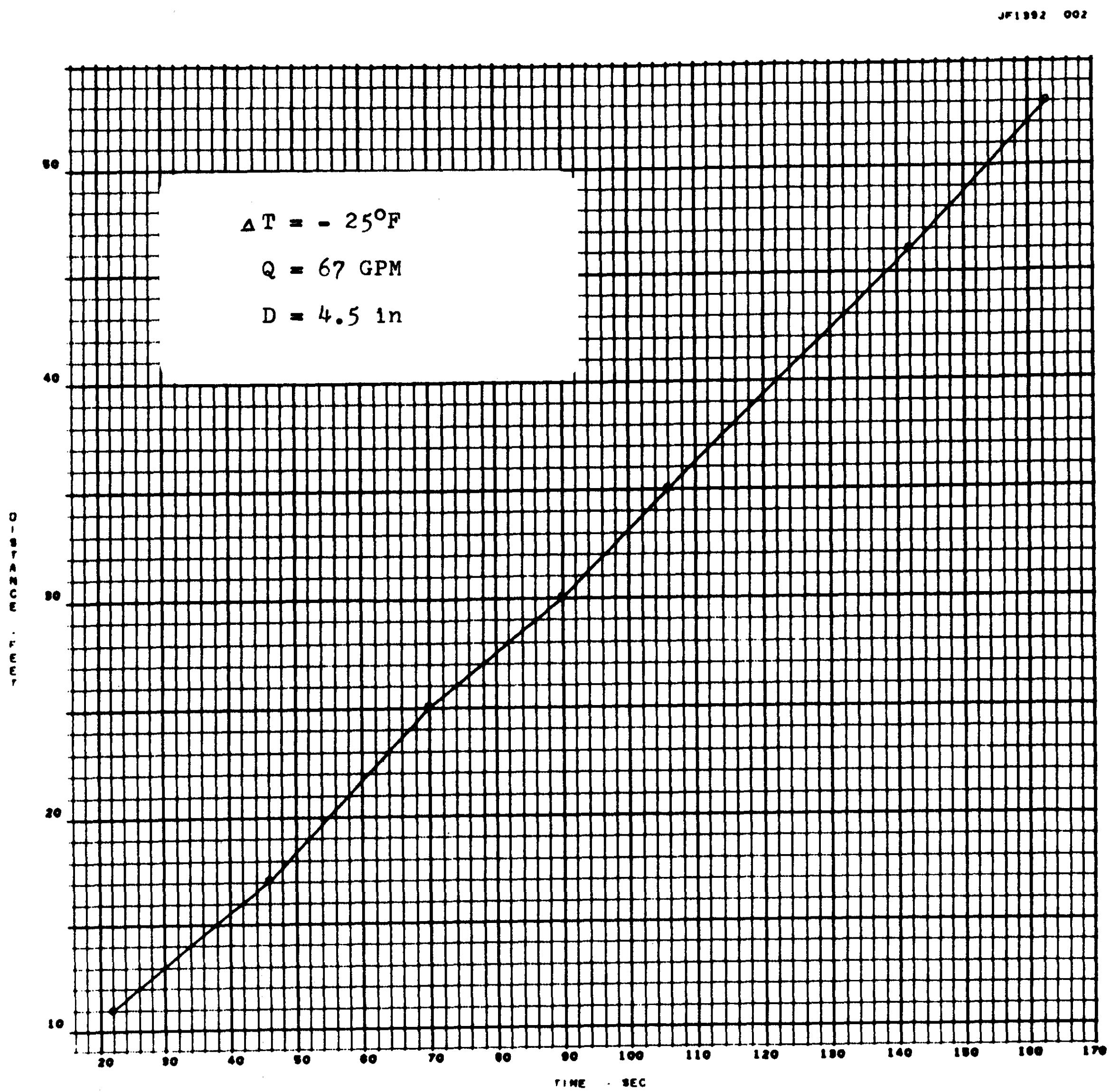

Fig. A.8

Positon of Density Front as a Function of Elapsed Operational Time

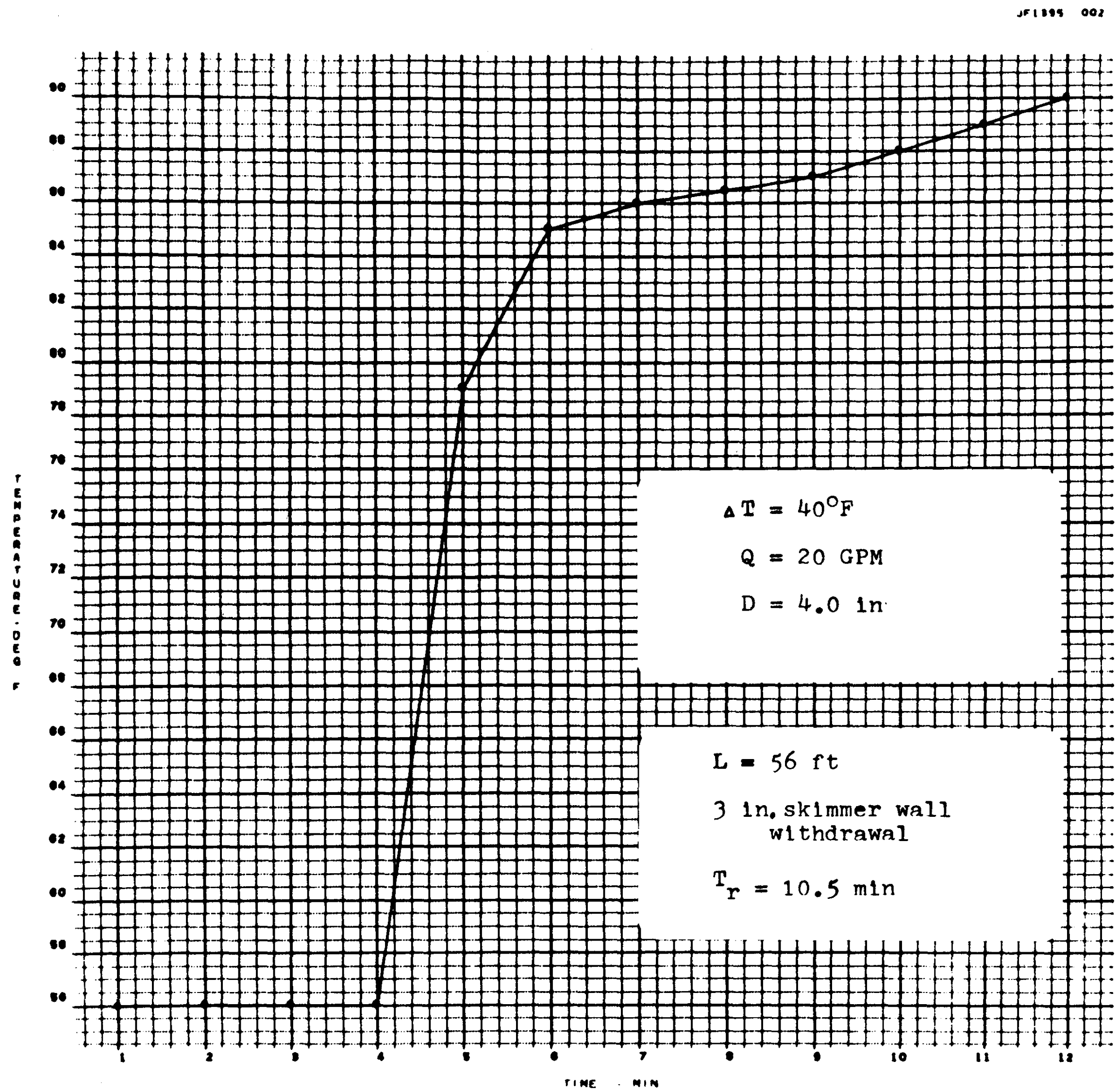

Fig. A.9

Temperature of Withdrawal

Flow as a Function of

Elapsed Operational Time

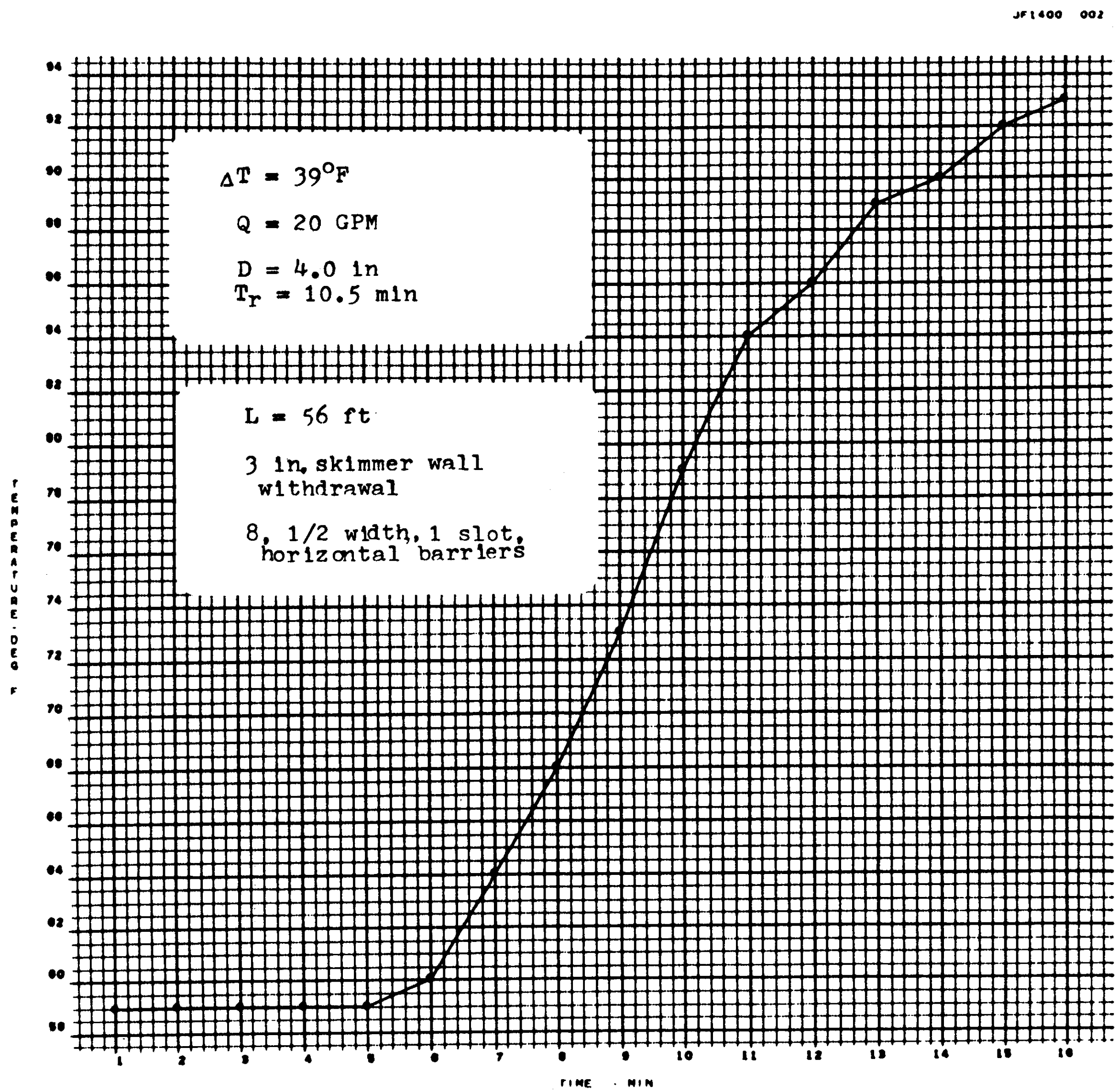

Fig. A.10

Temperature of Withdrawal

Flow as a Function of

Elapsed Operational Time

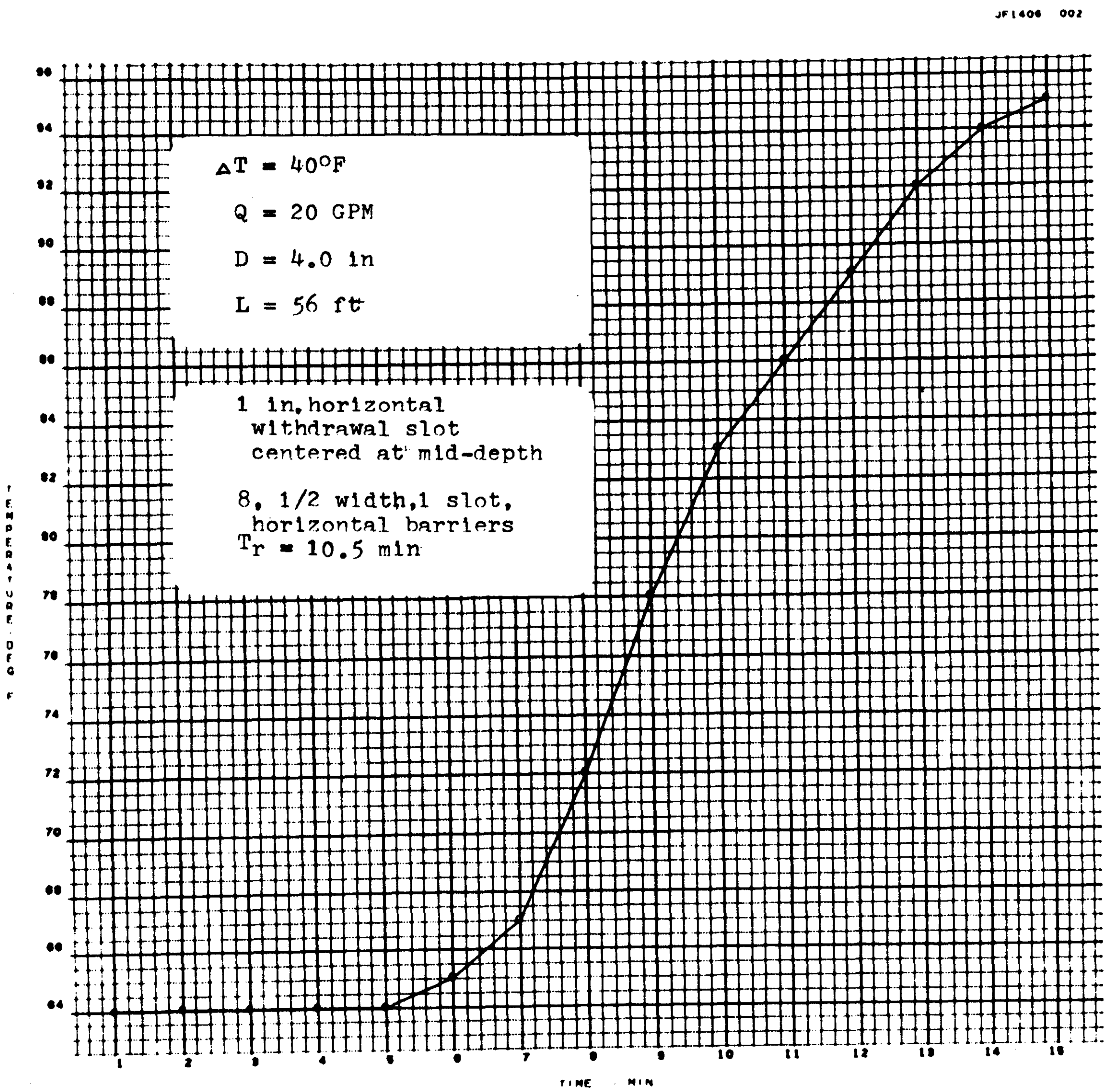

Fig. A.11

Temperature of Withdrawal Flow as a Function of Elapsed Operational Time

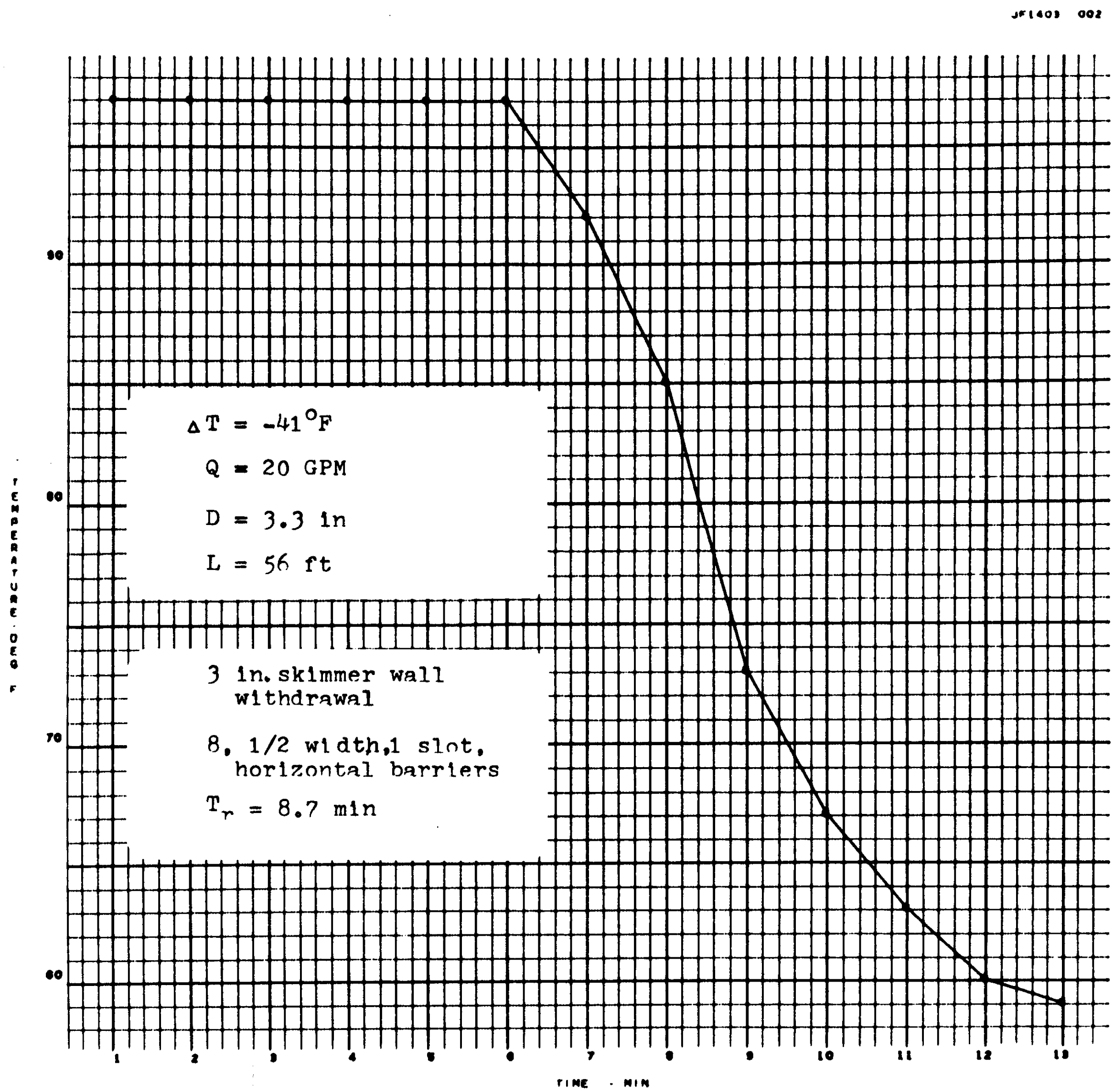

Fig. A.12

Temperature of Withdrawal Flow as a Function of Elapsed Operational Time

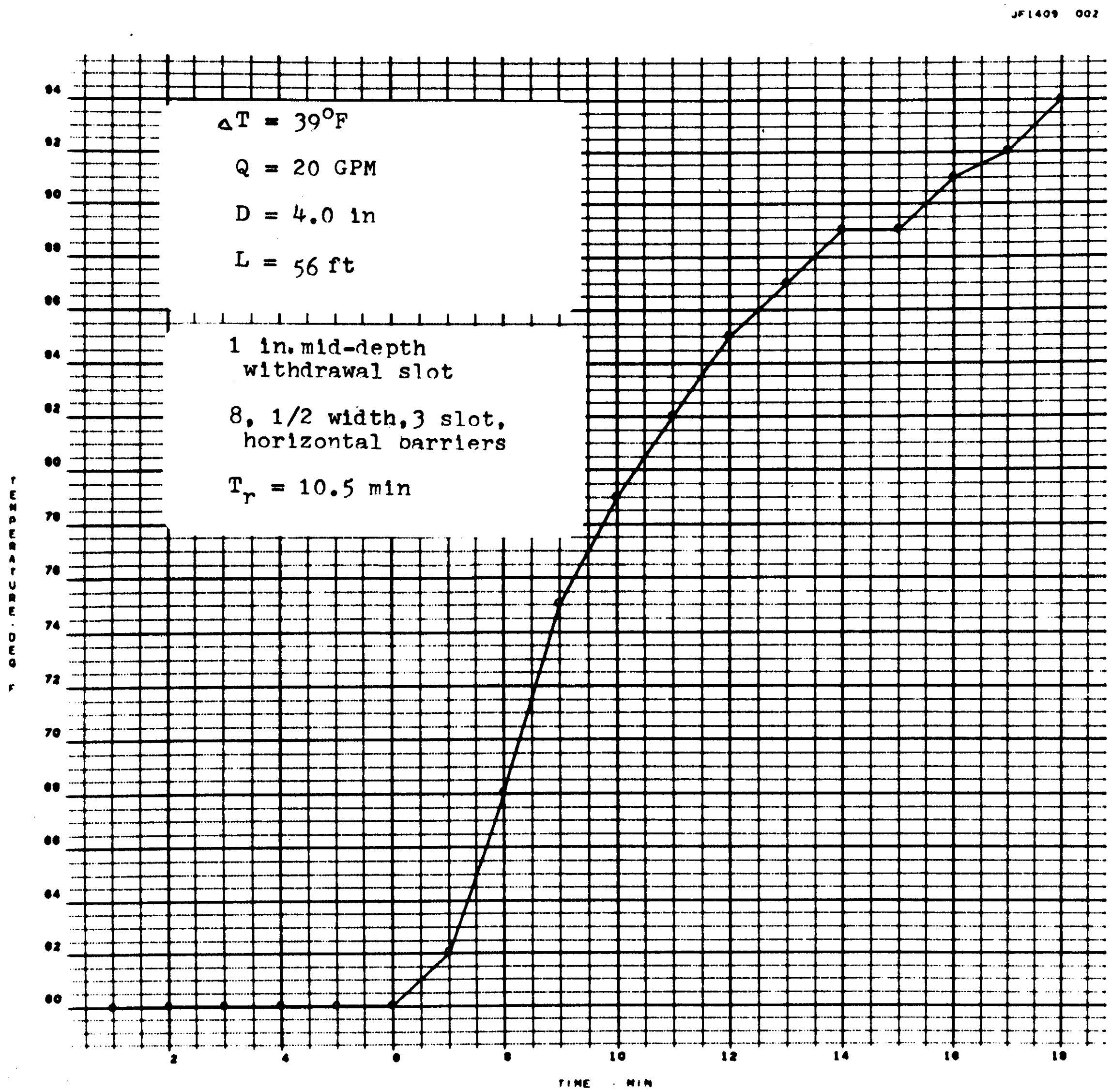

Fig. A.13

Temperature of Withdrawal

Flow as a Function of

Elapsed Operational Time

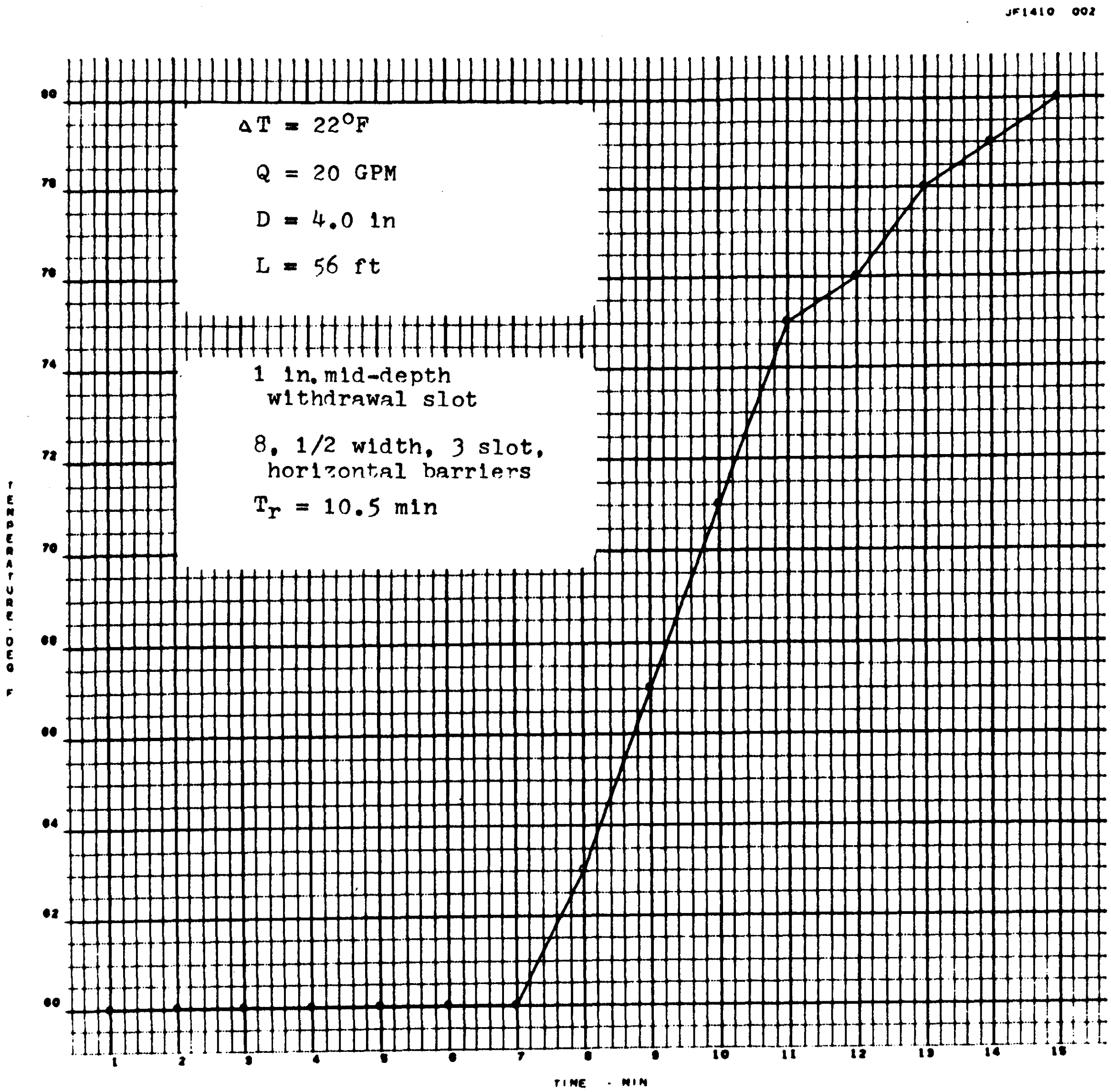

Fig. A.14

Temperature of Withdrawal

Flow as a Function of

Elapsed Operational Time

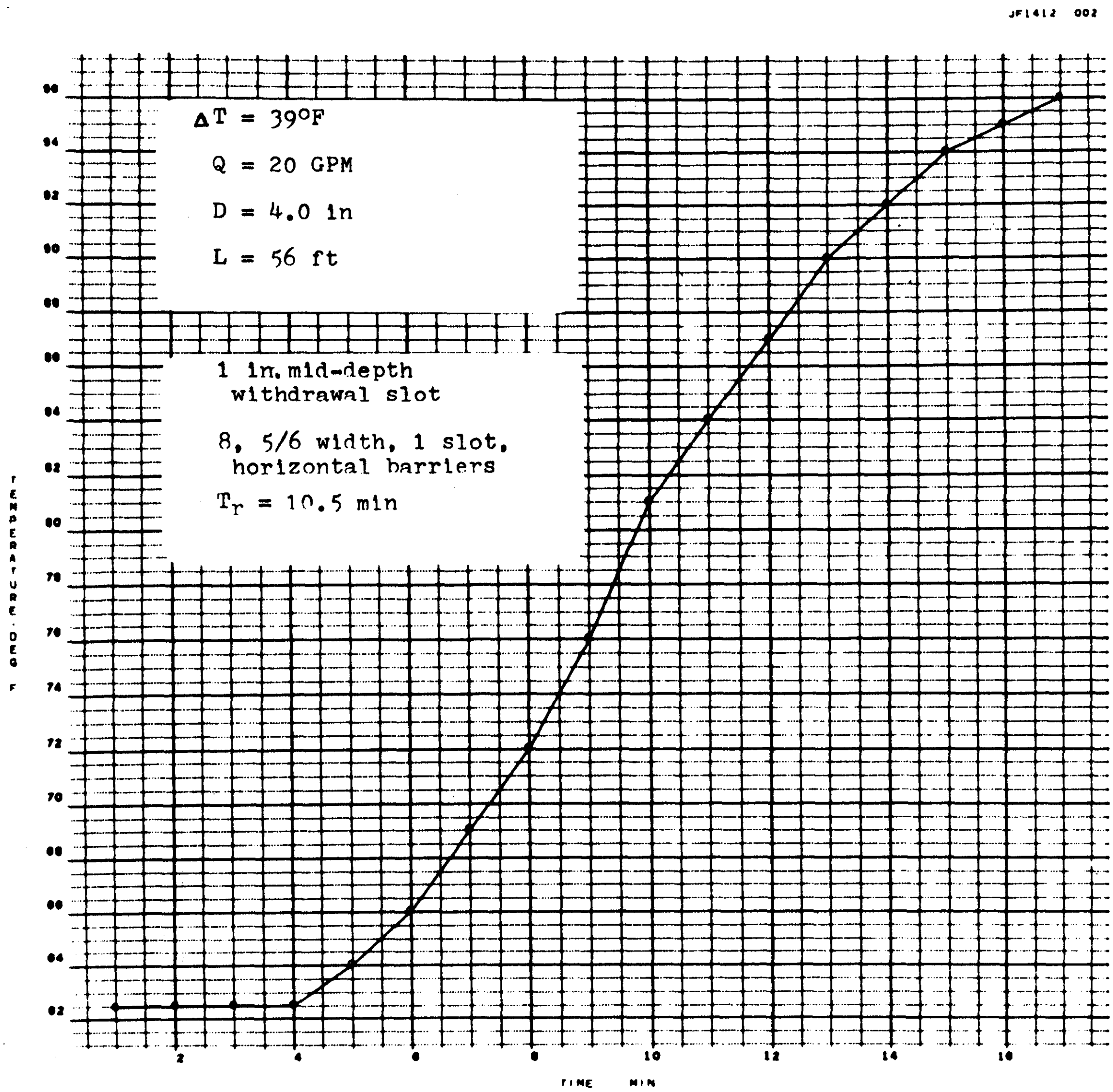

Fig. A.15

Temperature of Withdrawal

Flow as a Function of

Elapsed Operational Time

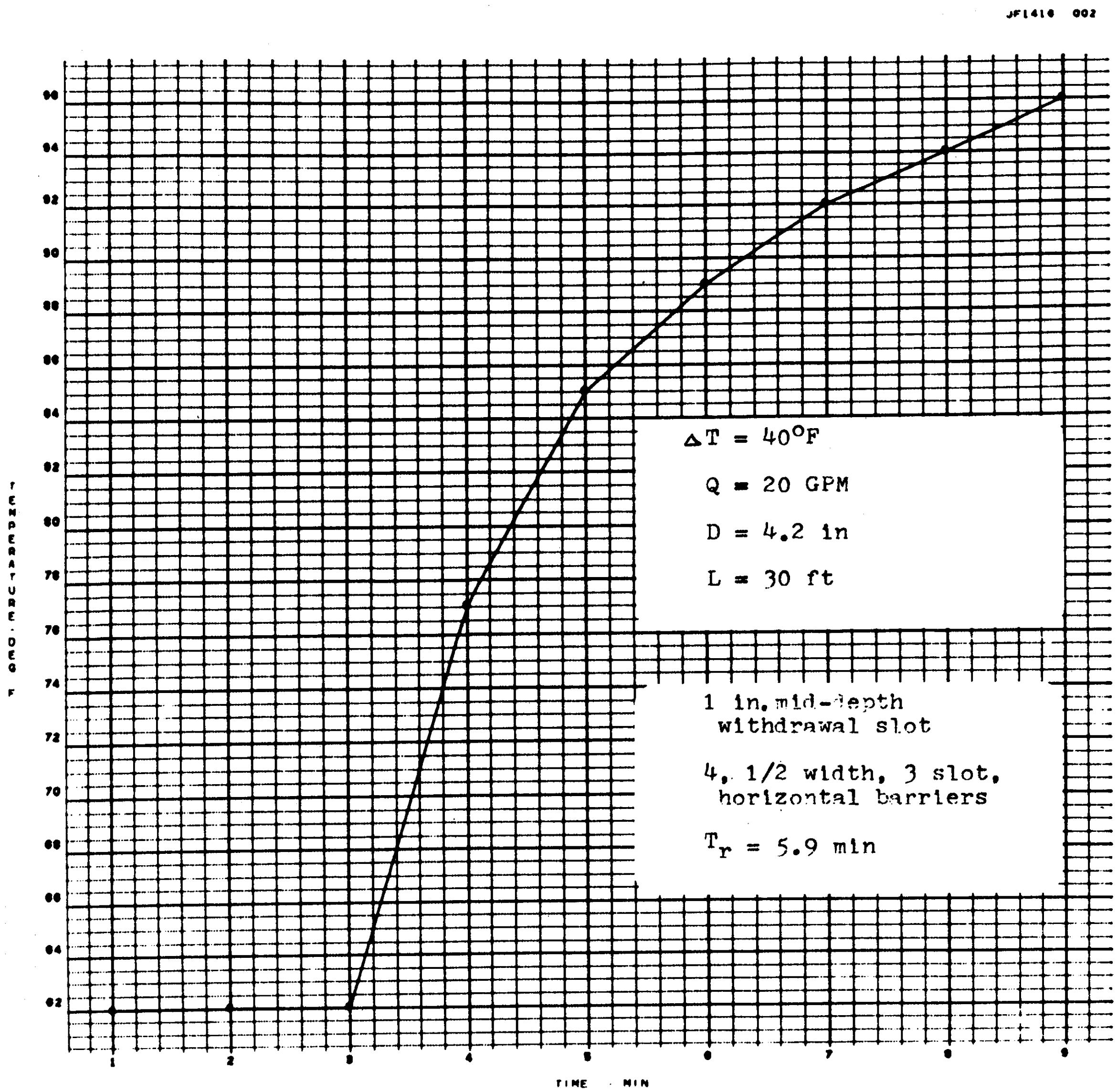

Fig. A.16

Temperature of Withdrawal

Flow as a Function of

Elapsed Operational Time

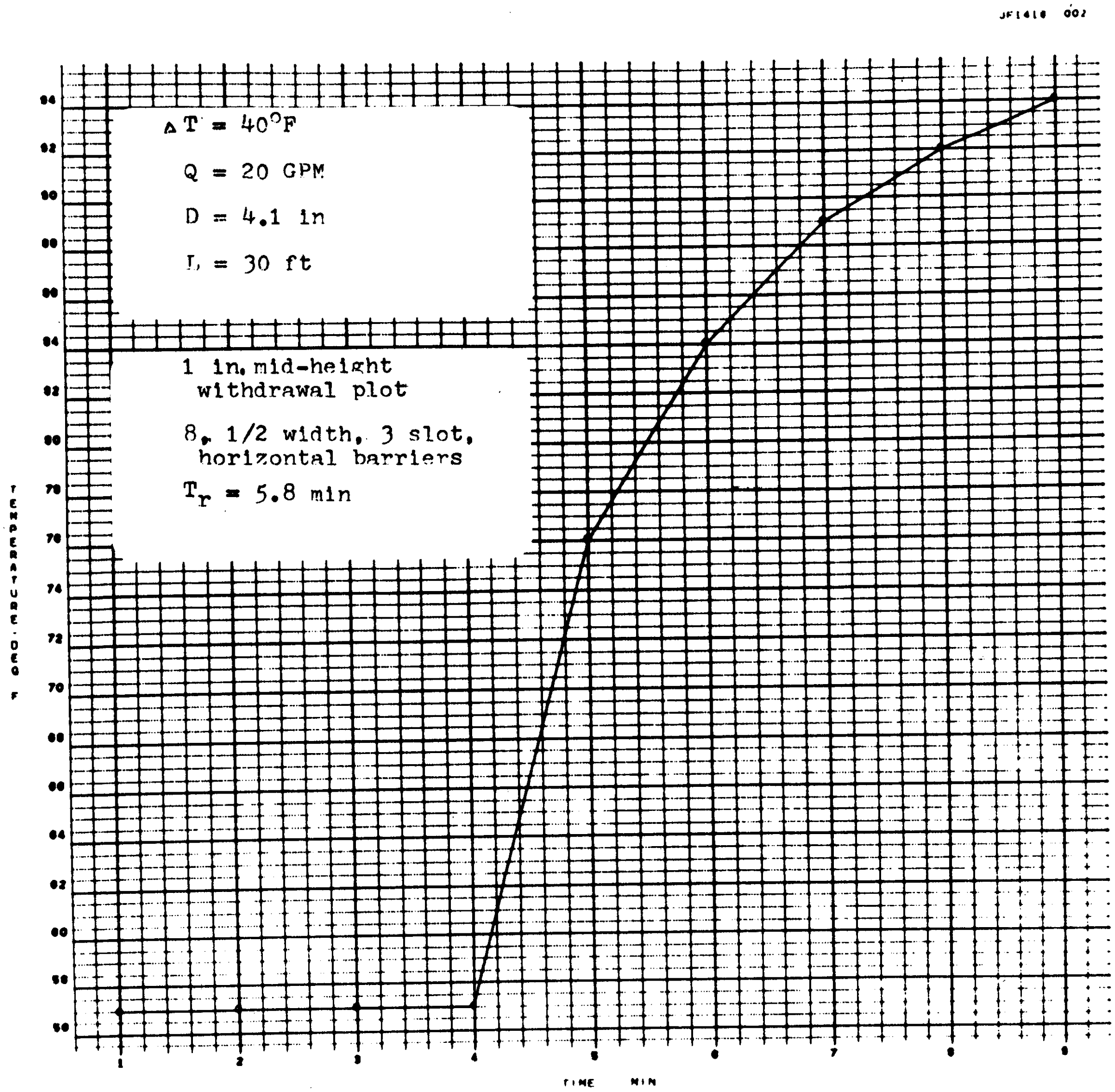

Fig. A.17

Temperature of Withdrawal Flow as a Function of Elapsed Operational Time

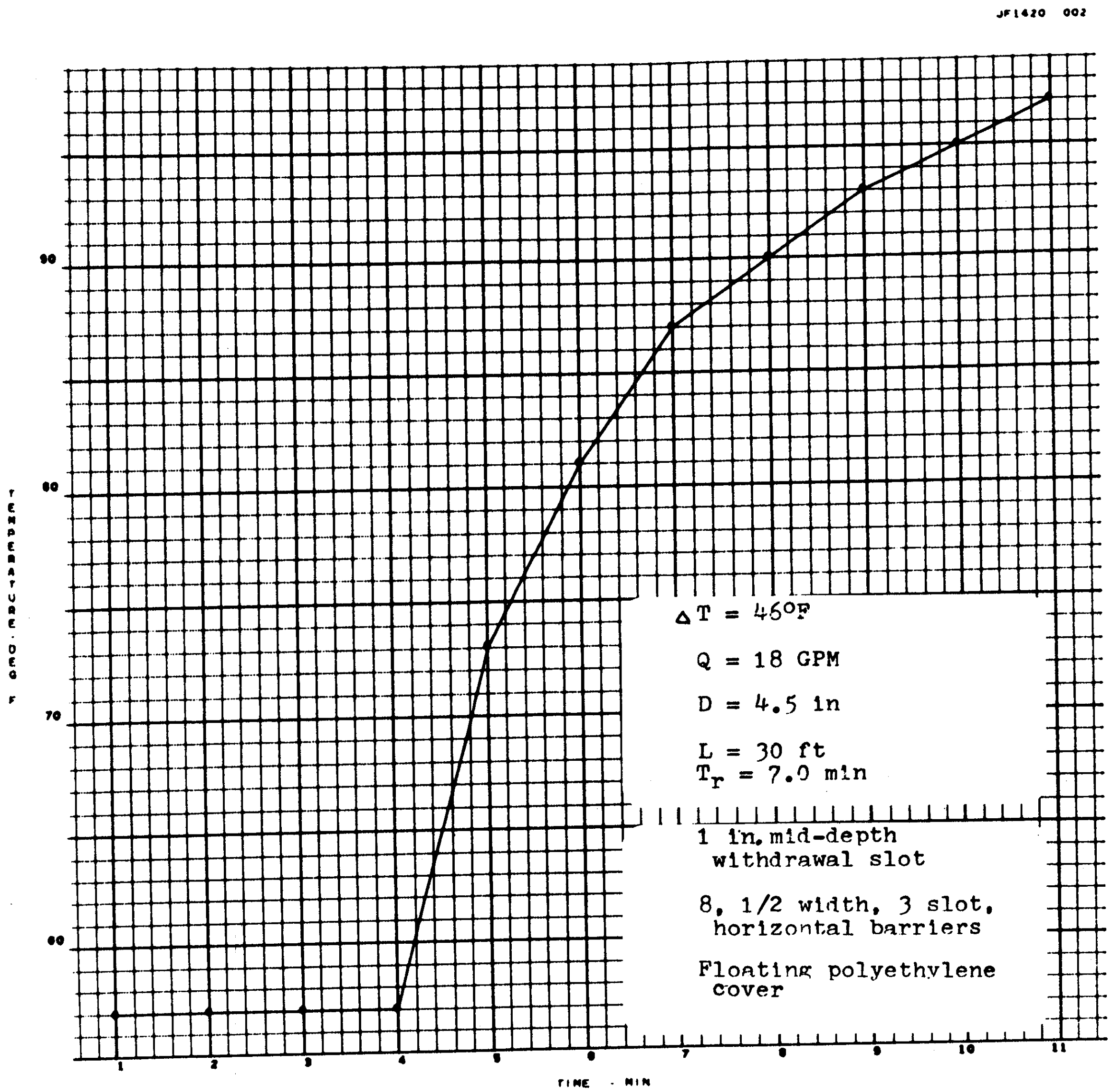

Fig. A.18

Temperature of Withdrawal Flow as a Function of Elapsed Operational Time

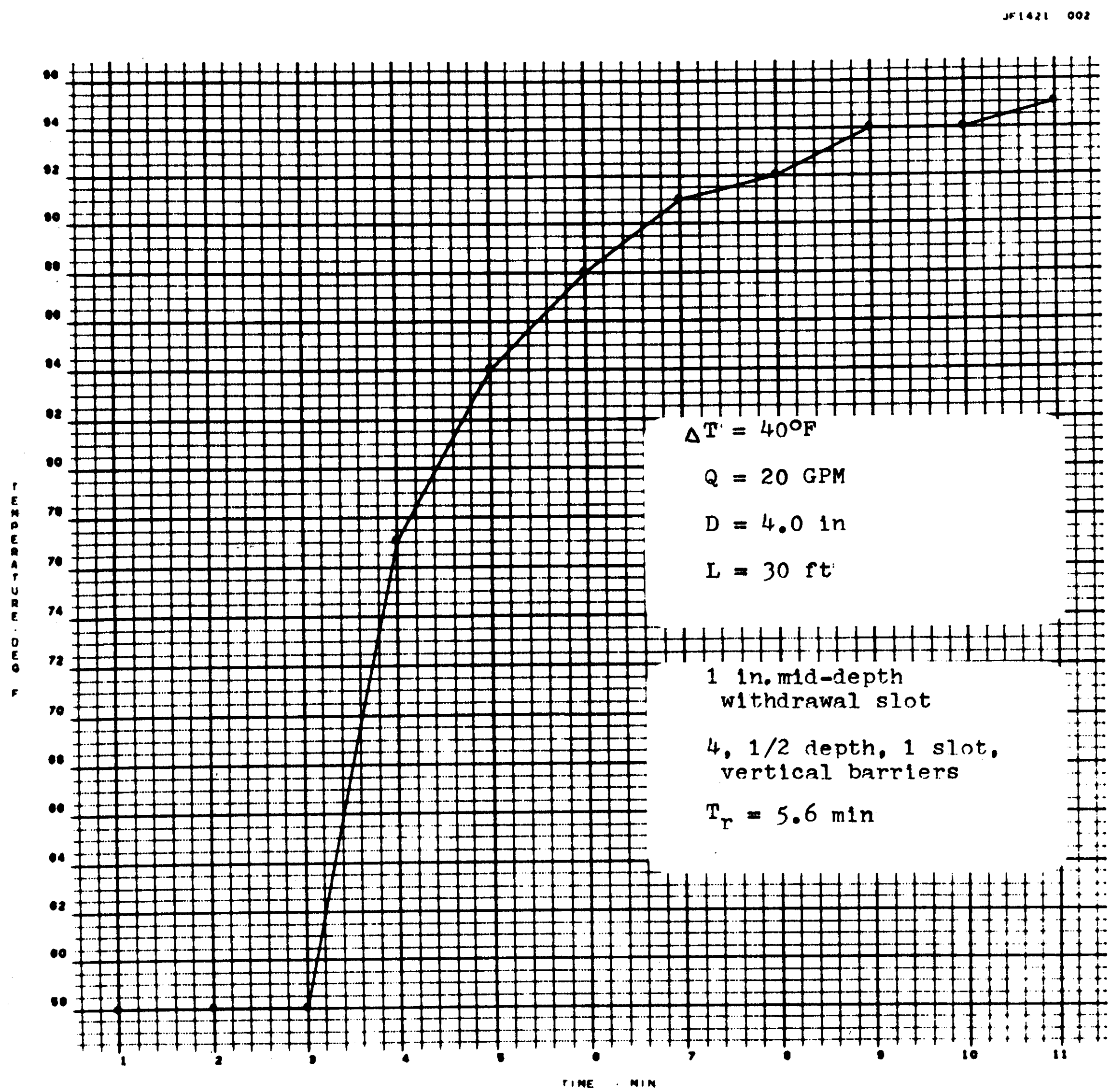

Fig. A.19

Temperature of Withdrawal

Flow as a Function of

Elapsed Operational Time

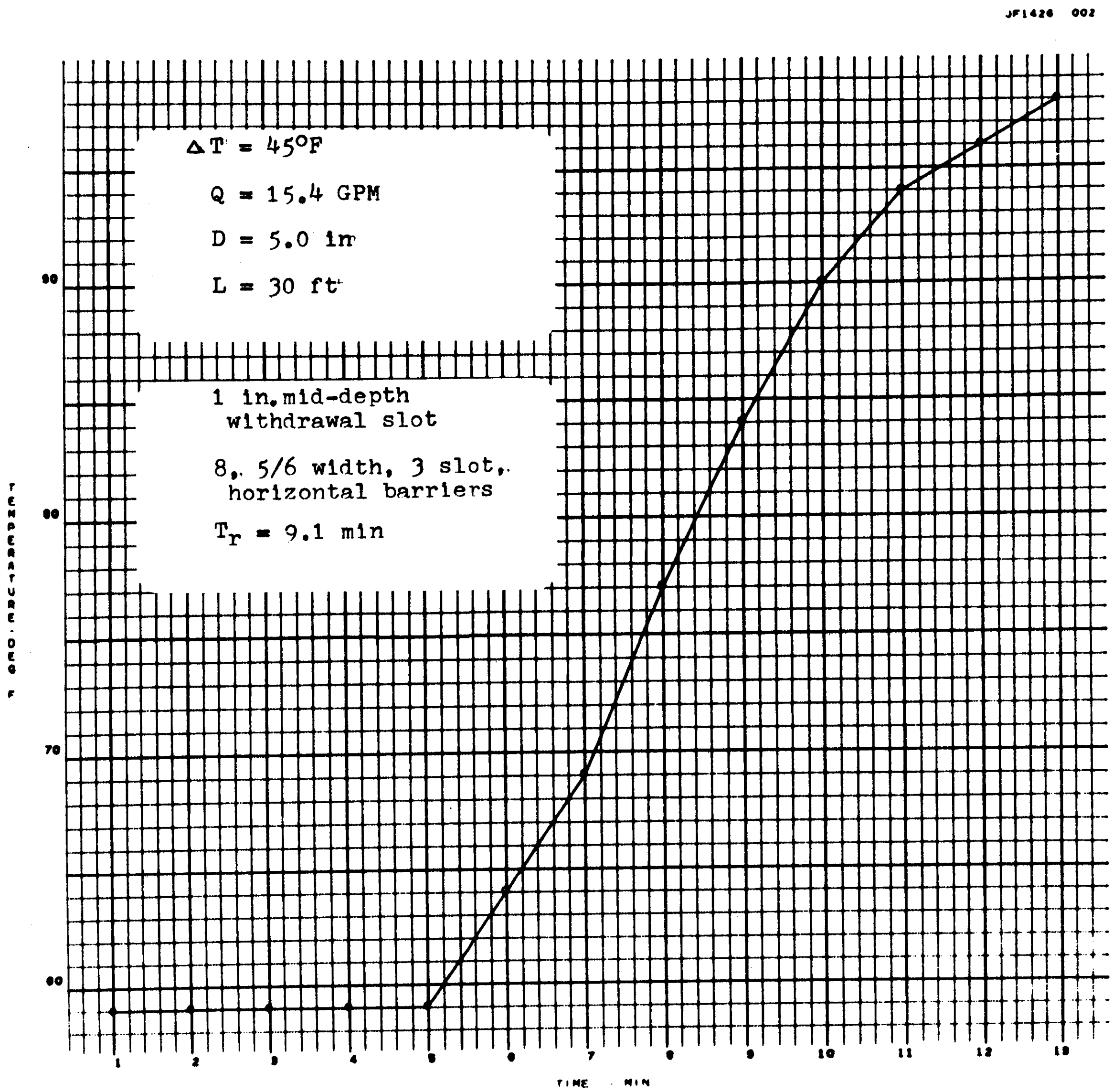

Fig. A.20

Temperature of Withdrawal

Flow as a Function of

Elapsed Operational Time

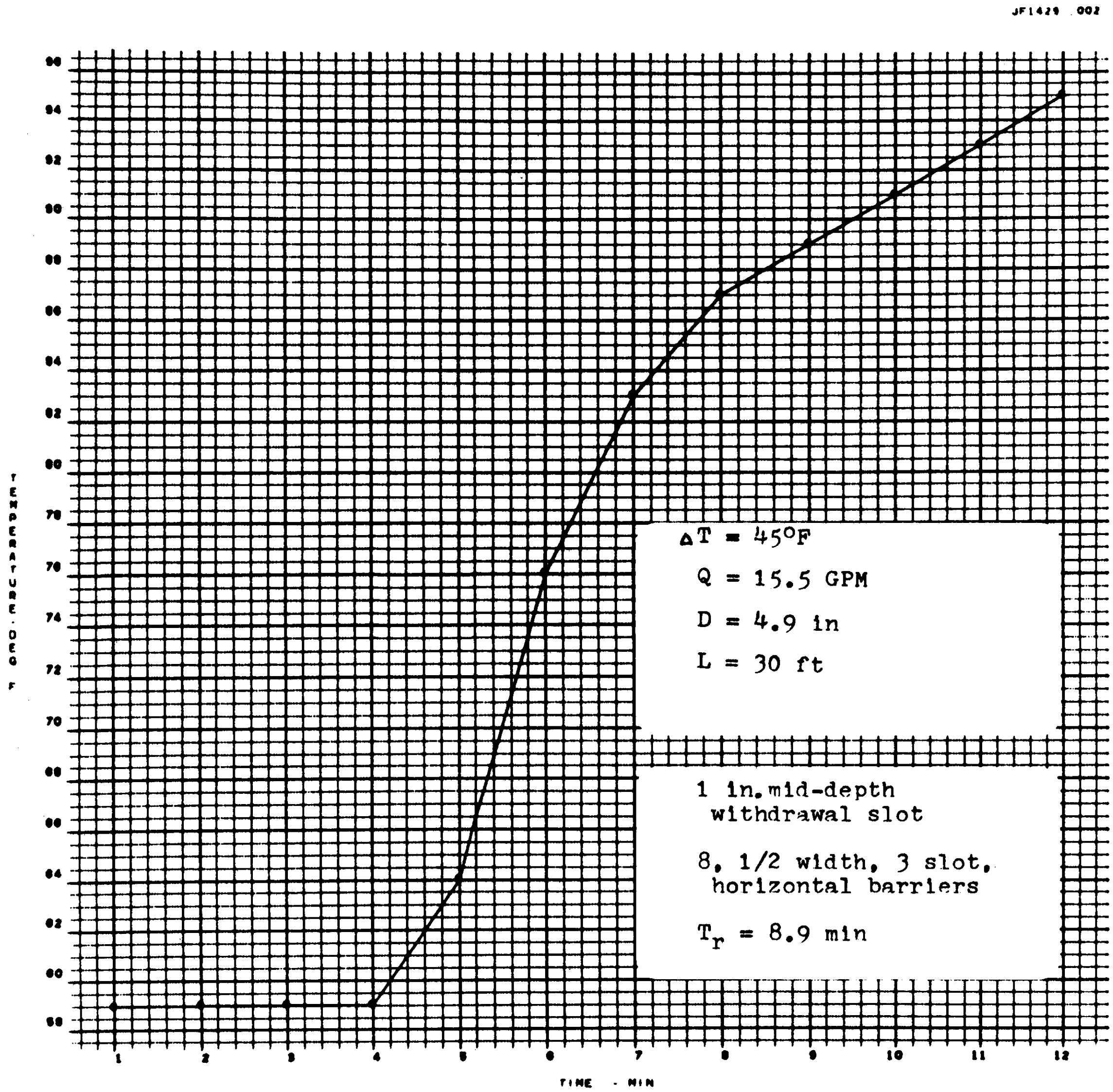

Fig. A.21

Temperature of Withdrawal

Flow as a Function of

Elapsed Operational Time

APPENDIX B

MODEL FOR THE INVESTIGATION OF THE THERMAL STORAGE POND/DRY COOLING TOWER ADVANCED WASTE HEAT REJECTION SYSTEM

The MITDAS program calculates the incremental cost of dry cooling for steam-electric plants for both simple dry tower systems and combined TSP/dry tower systems. Included in the program is a routine for the determination of optimal system design and operation.

The dry cooling system operational costs are determined for the optimum system design by performing a plant-TSP-dry tower system thermal performance simulation calculation for a one year period with a one year period with a one hour time step. A complete simulation calculation is not performed for each design specified by the optimization routine as it searches for the optimal system design. To do so would be prohibitively expensive. Instead, operational costs are determined by a pseudo-simulation calculation which is based on the grouping of days (from the chosen meteorological year) of similar ambient temperature range. The "steady-state" performance of the system and the associated daily operational costs are determined for each of the averaged daily temperatures and ranges listed in Table B.1. The daily cost for each average temperature and range is then multiplied by its frequency of occurrence (days/year) and then the sum of weighted costs is found to yield the annual cost.

The basic logic diagram for the program is given in Fig. B.1 and the required input data is described in Table B.2 and Fig. B.2. Included in this Appendix is a listing of the MITDAS program and a sample output. The sample run is for an adiabatic TSP-dry cooling tower-conventional nuclear steam turbine plant sited at Winslow, Arizona.

Table B.1

Average Daily Ambient Temperature and Temperature Range Groups

30,15	50,15	70,15	90,15
30,30	50,30	70,30	90,30
30,45	50,45	70,45	90,45

x,y ≡ Representation of average daily temperature (°F) and daily temperature range (°F) assigned to any day with an average temperature less than x+10 and greater than x-10, and with a range less than y+7.5 and greater than y-7.5.

Table B.2

MITDAS Input Variable Definitions

(Variables listed in input sequence)

ITHRMB = TSP thermal-hydraulic model

PONTYP = Type of thermal storage pond cover

BYPASS = fraction of circulating water flow bypassing TSP

ITHTMN = Minimum temperature of heat rate data input

ITHTMX = Maximum temperature of heat rate data input

ITHTIN = Temperature increment of heat rate data

IPOHMN = Minimum power of heat rate data input (tenths)

IPOHMX = Maximum power of heat rate data input (normally = 10)

IPOHIN = Power increment of heat rate data input (tenths)

HTRATE(ITHTR,IPOHTR) = net heat rate of turbine (BTU/KwHr) at condensing temperature ITHTR and thermal power IPOHTR

TDBMAX(J) = maximum day bulb temperature on day J (°F)

TDBMIN(J) = minimum dry bulb temperature on day J (°F)

NOPSET = number of operational variables sets to be read in

TURMAX = maximum allowable condensing temperature (°F)

MPRATE = minimum plant thermal power (tenths)

MWTHRM = thermal power of plant (MW)

TIMMAX = daily time of peak ambient temperature (integer hour)

Table B.2(continued)

TIMMIN	=	daily time of minimum ambient temperature (integer hour)
DEMDPT	=	time of day, following time of maximum system lambda, at which the average system lambda occurs (integer hour)
DEMDMT	=	time of day, following time of minimum system lambda, at which the average system lambda occurs (interger hour)
LAMMAX	=	daily maximum system lambda ($)
LAMMIN	=	daily minimum system lambda ($)
PRINT1	=	printout option (T or F)
PRINT2	=	printout option (T or F)
TOWONL	=	Simple dry tower system simulation option (T or F)
NOPTIM	=	optimization routine bypass option (T or F)
WLG	=	initial geuss at circulation water flow during uncoupled modes (ft^3/sec)
WLCG	=	initial geuss at circulating water flow during coupled modes (ft^3/sec)
TOWSZG	=	initial geuss at number of tower cells
TBHUOP	=	time begun heatup operation (integer hour)
TEHUOP	=	time end heatup operation (integer hour)
TBCDOP	=	time begin cooldown operation (integer hour)
TECDOP	=	time end cooldown operation (integer hour)

Fig. B.1 Simplified Flowchart for MITDAS Program

MAIN PROGRAM

Input data

Interpolate heat rate data to $1^{o}F$ increments

↓

Determine hourly values of lambda

↓

Determine temperature frequency distribution

↓

Find ten hotest day's maximum and minimum temperatures

↓

Find maximum and minimum temperatures of ten days preceding the tem hotest

↓

Printout input data

↓

Make initial geuss of optimal design

↓

Determine cost of specified design by calling FUNCTION F(X) (using pseudo-simulation)

↓

Select improved design by Rosenbrock method

↓

Determine change in system cost from improved design and previous design by calling FUNCTION F(X)

↓

Convergence criterion satisfied

NO

YES

↓

Determine exact cost of improved system design by calling FUNCTION(X) (using exact simulation)

↓

Printout system design summary and costs

↓

End program

Fig. B.1 (cont.)

FUNCTION F(X)

Determine condenser outlet temperature as a function of inlet temperature for 1^{o}F temperature increments for coupled operational mode

Determine tower performance coefficients for both coupled and uncoupled modes

Determine condenser outlet temperature as a function of inlet temperature for 1^{o}F temperature increments for uncoupled operational modes

Determine plant power output as a function of ambient dry bulb temperature for uncoupled modes

Is this cost determination to be based on pseudo-simulation

YES

Determine "steady-state" behavior of system for each of the representative ambient temperature averages and daily ranges---determine daily operational cost

NO

Determine loss of capability penalty based on ten hotest day's average maximum and minimum ambient temperatures

Weigh each daily cost by its frequency of occurrance and sum over all temperature average and range groups to determine annual operational cost

Calculate capital cost based on system design parameters and add to operational cost to find total cost

Return to MAIN PROGRAM

Determine annual operational costs by calculating the system performance for a one year period with one hour time steps

Calculate capital cost based on system design parameters and add to operational cost

Return to MAIN PROGRAM

last record ********* TBHUOP TEHUOP TBCDOP TECDOP : FORMAT(4I10)

WLG WLCG TOWSZG : FORMAT(3F10.3)

TOWONL NOPTIM : FORMAT(2L5)

PRINT1 PRINT2 : FORMAT(2L5)

DEMDPT DEMDMT LAMMAX LAMMIN : FORMAT(2I10,2F10.7)

TIMMAX TIMMIN : FORMAT(2I4)

TURMAX MPRATE MWTHRM : FORMAT(F10.2,I10,F10.2)

NOPSET : FORMAT(I4)

(TDBMAX(J),TDBMIN(J),J=1,365) : FORMAT(20F4.0)

((HTRATE(ITHTR,IPOHTR),ITHTR=ITHTMN,ITHMXITHTN),IPOHTR=IPOHMN,IPOHMX,IPOHIN):FORMAT(8F10.1)

ITHTMN ITHTMX ITHTIN IPOHMN IPOHMX IPOHIN : FORMAT(6I10)

ITHRMB PONTYP BYPASS : FORMAT(2I10,F10.5) **************first record

Fig. B.2 MITDAS Input FORMAT

```
C
C                        THE 'MITDAS' PROGRAM
C
C    MODEL FOR THE INVESTIGATION OF THE THERMAL STORAGE POND/DRY COOLING
C       TOWER ADVANCED WASTE HEAT REJECTION SYSTEM
C
C
C
C   AVBCKX = AVERAGE MAXIMUM TEMPERATURE ON DAYS PRECEDING THE TEN HOTEST
C   AVBCKN = AVERAGE MINIMUM TEMPERATURE ON DAYS PRECEDING THE TEN HOTEST
C   AVTENX = AVERAGE MAXIMUM TEMPERATURE ON TEN HOTEST DAYS
C   AVTENN = AVERAGE MINIMUM TEMPERATURE ON TEN HOTEST DAYS
C   AFCR = ANNUAL FIXED CHARGE RATE
C   AVPCLD = AVERAGE POWER DURING COOLDOWN MODE
C   AVPHTU = AVERAGE POWER DURING HEATUP MODE
C   AREAP = AREA OF POND
C   AUF = HEAT TRANSFER AREA * HEAT TRANSFER COEFFICIENT * CROSS FLOW
C          FACTOR FOR REPRESENTATIVE DRY TOWER CELL
C   BYPASS = FRACTION OF CIRCULATING WATER FLOW BYPASSING THE POND
C   CAPRC = CAPITAL COST OF REPLACEMENT CAPABILITY
C   CAPTSP = CAPITAL COST OF THERMAL STORAGE POND
C   CAPTOW = CAPITAL COST OF TOWER
C   CAPLMX = MAXIMUM CAPABILITY LOSS
C   CONOT = CONDENSER OUTLET TEMPERATURE
C   CITCM = MAXIMUM INLET CONDENSER TEMPERATURE FOR COUPLED MODES
C   CUNIT = TOWER PERFORMANCE COEFFICIENT FOR UNCOUPLED MODE
C   CUNITC = TOWER PERFORMANCE COEFFFICIENT FOR COUPLED MODE
C   CONDEN(XCONIN) = CONDENSER OUTLET TEMPERATURE FOR CONDENSER INLET
C          TEMPERATURE XCONIN FOR COUPLED MODE
C   CAPLUM (XCONIN) = LOSS OF GENERATING CAPABILITY FOR UNCOUPLED MODE OF OPERAT
C         ION WITH CONDENSER INLET TEMPERATURE XCONIN
C   CAPLCM (XCONIN) = LOSS OF GENERATING CAPABILITY FOR  COUPLED MODE OPERATION
C         WITH CONDENSER INLET TEMPERATURE XCONIN
C   CONDN1(XCONIN)= CONDENSER OUTLET TEMPERATURE FOR CONDENSER INLET TEMPERATURE
C         XCONIN FOR UNCOUPLED MODE
C   DELY = CONVERGENCE CRITERION
C   DELPR2 = PRESSURE DROP IN TOWER, COUPLED MODE
C   DELPR1 = PRESSURE DROP IN TOWER, UNCOUPLED MODE
C   DEMDPT = TIME OF DAY, FOLLOWING TIME OF MXIMUM SYSTEM LAMBDA, AT WHICH THE
C          AVERAGE SYSTEM LAMBDA OCCURS
C   DEMDMT = TIME OF DAY, FOLLOWING TIME OF MINIMUM SYSTEM LAMBDA, AT WHICH THE
C          AVERAGE SYSTEM LAMBDA OCCURS
C   E( ) = DIRECT SEARCH STEP SIZES
C   ETA = PLANT EFFICIENCY
C   EVAPD = DAILY EVAPORATIVE LOSS FORM POND
C   EVAPT = TOTAL EVAPORATION FORM POND IN ONE YEAR
C   FREQ(X,Y) = FREQUENCY OF DAYS PER YEAR WITHIN AVERAGE TEMPERATURE GROUP X
C        AND AVERAGE RANGE GROUP Y
C   FLTOWC = FLOW THRU TOWER CELL, COUPLED MODE
C   FLTOWU = FLOW THRU TOWER CELL, UNCOUPLED MODE
C   FANPOW = FAN POWER PER UNIT DRY TOWER CELL
C   HTRATE (A,B) = HEAT  RATE AT CONDENSER TEMPERATURE A AND POWER LEVEL B = POW
C          ER/MAXIMUM POWER
C   IPOW = PLANT THERMAL POWER RATING (TENS OF PERCENT)
```

```
C     IPOW = POWER LEVEL = (MWT) = PER CENT
C     ITHTR = CONDENSING TEMPERATURE AT WHICH HEAT RATE IS EVALUATED
C     IPOHTR = THERMAL POWER AT WHICH HEAT RATE IS EVALUATED
C     ITHTMN = MINIMUM CONDENSING TEMPERATURE INPUT OF HTRATE
C     ITHTMX = MAXIMUM CONDENSING TEMPERATURE INPUT OF HTRATE
C     IPOHMN = MINIMUM POWER LEVEL OF HTRATE INPUT
C     IPOHMX = MAXIMUM POWER LEVEL OF HTRATE INPUT
C     IPOHIN = POWER INTERVAL FOR IPOHTR INPUT OF HTRATE
C     ITHRMB = THERMAL-HYDRAULIC BEHAVIOR OF POND,1=EMPIRICAL MODEL,2=FULLY-MIXED
C     IOUROP = HOUR OF OPERATIONAL DAY
C     JDAY = DAY OF YEAR
C     LOSCPU (ITTDB) = LOSS OF PLANT GENERATION CAPABILITY FOR UNCOUPLED OPERATION
C            AT AMBIENT TEMPERATURE IITDB
C     LAMBDA(I) = SYSTEM INCREMENTAL GENERATION COST FOR HOUR I
C     TDBMAX (S) = MAXIMUM DRY BULB TEMPERATURE ON DAY J
C     MODSYS = SYSTEM OPERATIONAL  ODE, 1=COLD STANDBY, 2=HEATUP, 3=HOT
C            STANDBY, 4=COOLDOWN
C     MASSP = MASS OF WATER IN POND
C     MLSPKW = INCREMENTAL COST OF DRY COOLING
C     NOVSRI = NUMBER OF OPERATIONAL VARIABLE SETS READ IN
C     NOPSET = NUMBER OF SETS OF OPERATIONAL VARIABLES
C     OPUMP = PUMP OPERATIONAL COST
C     OPFAN = FAN OPERATIONAL COST
C     OPPEN (I,J) = LOSS OF CAPABILITY PENALTY FOR DAY J, HOUR I
C     PKTEM(X,Y) = MAXIMUM TEMPERATURE FOR AVERAGE TEMPERATURE GROUP X AND AVERAGE
C     PONTYP = (INTERGER) IF 1- OPEN POND, IF 2- REFLECTIVE POND, IF 3- ADIBATIC P
C     POWFRC =  FRACTIONAL THERMAL POWER
C     PHISCC = DAILY INSOLATION
C     PRINT2 = PRINTING OPTION
C     PRINT1 = PRINTING OPTION
C     PCF = PLANT CAPACITY FACTOR
C     POWFRC = FRACTION OF MAXIMUM THERMAL RATING
C     POWMAX = MAXIMUM POWER RATING OF PLANT
C     PHOTSB = LENGHT OF HOT STANDY OPERATION
C     PCOOLD = LENGHT OF COOLDOWN OPERATION
C     PHEATU = LENGHT OF HEATUP OPERATION
C     PCOLSB = LENGHT OF COLD STANDBY OPERATION
C     QJECT = HEAT TRANSFER RATE FORM TOWER
C     RELHUM = RELATIVE HUMIDITY
C     SMTEM(X,Y) = MINIMUM TEMPERATURE FOR AVERAGE TEMPERATURE GROUP X AND AVERAGE
C            AVERAGE RANGE GROUP Y
C     SUNSSS = FRACTION OF CLOUD COVER
C     TOTANC = TOTAL ANNUAL COST OF COOLING SYSTEM
C     TOPC = TOTAL YEARLY OPERATIONAL COST
C     TPOND = POND TEMPERATURE
C     TBDA = AVERAGE AMBIENT TEMPERATURE DURING PERIOD OF HEAT TRANSFER FROM POND
C     TIMEX = PERIOD OF HEAT TRANSFER FORM POND
C     TURMAX = MAXIMUM TURBINE EXHAUST TEMPERATURE, I.E. CONDENSING TEMPERATURE
C     TIMMIN = HOUR OF DAY WHEN DAILEY MINIMUM AMBIENT TEMPERATURE OCCURRS
C     TECDOP = (INTERGER) = TIME COOLDOWN OPERATION ENDS
C     TBCDOP = (INTERGER) = TIME COOLDOWN OPERATION BEGINS
C     TIMMIN = TIME OF DAY WHEN MINIMUM DAILY TEMPERATURE OCCURRS
C     TIMMAX = TIME OF DAY WHEN MAXIMUM DAILY TEMPERATURE OCCURRS
C     TIMMAX = HOUR OF DAY WHEN DAILY MAXIMUM AMBIENT TEMPERATURE OCCURS
C     TPONDH = HOT POND TEMPERATURE
```

```
C     TPONDC = COLD POND TEMPERATURE
C     TITD = TOWER INITIAL TEMPERATURE DIFFERENCE
C     TC = CONDENSING TEMPERATURE
C     THTOUT = HEAT TRANSFER FROM POND DURING ONE DAY
C     TLSOGN = YEARLY GENERATION LOSS
C     TOWSZ = NUMBER OF TOWER CELLS
C     TOWSZG = INITIAL GEUSS AT NUMBER OF TOWER CELLS
C     TWASTH = TOTALL WASTE HEAT REJECTION AT POWMAX
C     TAUF = TOTAL AUF FOR A GIVEN TOWER SIZE
C     TPONDI = INITIAL CONDITION OF POND
C     TEHUOP = TIME HEATUP OPERATION ENDS
C     TBHUOP = TIME HEATUP OPERATION BEGINS
C     TDBMIN (J) = MINIMUM DRY BULB TEMPERATURE ON DAY J
C     VOLUM = POND VOLUME
C     W2 = WIND SPEED
C     WLC = CIRCULATING WATER FLOW FOR COUPLED MODES
C     WL = CIRCULATING WATER FLOW FOR UNCOUPLED MODES
C     WLCG = INITIAL GEUSS AT CIRCULATING WATER FLOW FOR COUPLED MODES
C     WLG = INITIAL GEUSS AT CIRCULATING WATER FLOW FOR UNCOUPLED MODES
C     X( ) = OPTIMIZATION DECISION VARIABLE
C     XLOSS = LOSS OF PLANT GENERATING CAPABILITY
C     YMELGN = YEARLY TOTAL POWER GENERATION
C     YTOWOP = YEARLY TOWER OPERATIONAL COST
C     YCAPEN = ANNUAL LOSS OF GENERATION PENALTY
C
C
C
          DIMENSION X(3),E(3),V(3,3),SA(3),D(3),G(3),H(3),AL(3),
     1    PH(3),A(3),B(3,3),BX(3),DA(1),VV(3,3),EINT(3),VM(3)
          DIMENSION HTRATE(300,10),LAMBDA(25),TDBMAX(370),TDBMIN(370)
          DIMENSION WINTIL(400),IWTEN(10)
          DIMENSION FREQ(6,6),PKTEM(4,4),SMTEM(4,4)
          DIMENSION XSAVE(3)
          DATA ((PKTEM(IIAV,IIRAN),IIRAN=1,3),IIAV=1,4)/37.5,45.0,52.5,57.
     15,65.0,72.5,77.5,85.0,92.5,97.5,105.0,112.5/


          DATA ((SMTEM(JJAV,JJRAN),JJRAN=1,3),JJAV=1,4)/22.5,15.0,7.5,42.5
     1,35.0,27.5,62.5,55.0,47.5,82.5,75.0,67.5/


          REAL LAMMAX,LAMMIN
          REAL LAMBDA
          REAL LOSCPU,MLSPKW,MWTHRM
          INTEGER PONTYP,TIMMIN,TIMMAX
          INTEGER DEMOPT,DEMOMT
          INTEGER TBHUOP,TEHUOP,TBCDOP,TECDOP
          COMMON/HOTDAY/WINTIL
          LOGICAL TOWONL,NOPTIM
          LOGICAL PRINT1,PRINT2
          COMMON KOUNT
```

```
      INTEGER P,PR,R,C
      REAL LC
      COMMON/OMEGA/MWTHRM
      COMMON/TOPANT/TIMMIN,TIMMAX
      COMMON/CAPCIT/CAPLMX,CAPSAV
      COMMON/PHTRMO/PONTYP,ITHRMB
      COMMON/CAPLDT/AVBCKX,AVBCKN,AVTENX,AVTENN
      COMMON/SEQE/NOPTIM
      COMMON/PMAX/POWMAX
      COMMON/PRINTO/PRINT1,PRINT2
      COMMON/WAS/TWASTH
      COMMON/TURBIN/HTRATE,TURMAX
      COMMON/OPMODE/TOWONL
      COMMON/MP/MPRATE
      COMMON/FJLCOS/LAMBDA
      COMMON/METEOR/TDBMAX,TDBMIN
      COMMON/PARTFL/BYPASS
      COMMON/TIMES/TBHUOP,TEHUOP,TBCDOP,TECDOP
      COMMON/COST/MLSPKW
      COMMON/SWITCH/IFINAL
      COMMON/TEMFRQ/FREQ
C   DATA INPUT
C***************************************************************************
C***************************************************************************
 30   CONTINUE
      READ(5,99) ITHRMB,PONTYP,BYPASS
 99   FORMAT(2I10,F10.5)
      READ(5,101)ITHTMN,ITHTMX,ITHTIN,IPOHMN,IPOHMX,IPOHIN
101   FORMAT(6I10)
      READ(5,102)((HTRATE(ITHTR,IPOHTR),ITHTR=ITHTMN,ITHTMX,ITHTIN),
     1  IPOHTR=IPOHMN,IPOHMX,IPOHIN)
102   FORMAT(8F10.1)
      READ(5,104)(TDBMAX(JJJJ),TDBMIN(JJJJ),JJJJ=1,365)
104   FORMAT(20F4.0)
      READ(5,111) NOPSET
111   FORMAT(I4)
      READ(5,112) TURMAX,MPRATE,MWTHRM
112   FORMAT(F10.2,I10,F10.2)
      READ(5,113) TIMMAX,TIMMIN
113   FORMAT(2I4)
      READ(5,117) DEMDPT,DEMDMT,LAMMAX,LAMMIN
117   FORMAT(2I10,2F10.7)
C***************************************************************************
C      PRINT1=TRUE   DESIGN VARIABLES SUMMARY ONLY
C      PRINT2=TRUE   DAILY RESULTS FOR FINAL DESIGN
C***************************************************************************
      READ(5,131) PRINT1,PRINT2
131   FORMAT(2L5)
      NOVSRI=0
      PIE=3.141
C   IF TOWONL = TRUE -- OPTIMIZED DRY TOWER SYSTEM ONLY
C   IF NOPTIM = TRUE,  NO OTIMIZATION OF SYSTEM DESIGN
      READ(5,177) TOWONL,NOPTIM
177   FORMAT(2L5)
      AMPLAM=(LAMMAX-LAMMIN)/2.0
```

```
            PER1=DEMDPT-DEMDMT
            PER2=24.0-PER1
            RDMDPT=DEMDPT
            RDMDMT=DEMDMT
C     CALCULATE HOURLY VARIATION IN THE INCREMENTAL POWER PRODUCTION COST
            DO 499   KKK=1,24
            RKKK=KKK
            IF(KKK.LE.DEMDMT)      LAMBDA(KKK)=-AMPLAM*SIN((PIE/(PER2))*
     1      (RKKK+(PER2-RDMDMT)))+LAMMIN+AMPLAM
            IF(KKK.GT.DEMDMT.AND.KKK.LT.DEMDPT)       LAMBDA(KKK)=AMPLAM*
     1       SIN((PIE/(PER1))*(RKKK-RDMDMT))+LAMMIN+AMPLAM
            IF(KKK.GE.DEMDPT)      LAMBDA(KKK)=-AMPLAM*SIN((PIE/(PER2))
     1      *(RKKK-RDMDPT))+LAMMIN+AMPLAM
  499       CONTINUE
            RTHTIN=ITHTIN
            LESS1=ITHTMX-ITHTIN
C     INTERPOLATE HEAT RATE DATA INPUT TO 1 DEGREE F INCREMENTS FOR ALL POWER
C       LEVELS
            DO 601   IXPOW=IPOHMN,IPOHMX,IPOHIN
            DO 599   IXI=ITHTMN,LESS1,ITHTIN
            DELHTR=HTRATE(IXI+ITHTIN,IXPOW)-HTRATE(IXI,IXPOW)
            INCM1=ITHTIN-1
            IF(INCM1.EQ.0) GO TO 599
            DO 598    JXJ=1,INCM1
            RJXJ=JXJ
            KXK=IXI+JXJ
            HTRATE(KXK,IXPOW)=HTRATE(IXI,IXPOW)+(RJXJ/RTHTIN)*DELHTR
  598       CONTINUE
  599       CONTINUE
            DO 602   ITM3=ITHTMX,250
            HTRATE(ITM3,IXPOW)=HTRATE(ITHTMX,IXPOW)
  602       CONTINUE
  601       CONTINUE
            POWMAX=(3412.0/HTRATE(ITHTMN,10))*MWTHRM
            TWASTH=(MWTHRM-POWMAX)*81888000.0/100.0
C     DETERMINE TEMPERATURE AVERAGE-RANGE FREQUENCY
            DO 2602   IJKA=1,4
            DO 2601 KJIR=1,3
            FREQ(IJKA,KJIR)=0.0
 2601       CONTINUE
 2602       CONTINUE
            DO 2620   ITEMI=1,365
            TAVEGE=(TDBMAX(ITEMI)+TDBMIN(ITEMI))/2.0
            TAVFIX=TAVEGE-40.0
            IF(TAVFIX.LE.0.0) IAVET=1
            IF(TAVFIX.LE.0.0) GO TO 2618
            TAVFIX=TAVFIX-20.0
            IF(TAVFIX.LE.0.0) IAVET=2
            IF(TAVFIX.LE.0.0) GO TO 2618
            TAVFIX=TAVFIX-20.0
            IF(TAVFIX.LE.0.0) IAVET=3
            IF(TAVFIX.LE.0.0) GO TO 2618
            IAVET=4
 2618       CONTINUE
            TAVEGE=(TDBMAX(ITEMI)+TDBMIN(ITEMI))/2.0
```

```
      RANG=TDBMAX(ITEMI)-TDBMIN(ITEMI)
      RANG=RANG-22.5
      IF(RANG.LE.0.0) IRANG=1
      IF(RANG.LE.0.0) GO TO 2619
      RANG=RANG-15.0
      IF(RANG.LE.0.0) IRANG=2
      IF(RANG.LE.0.0) GO TO 2619
      IRANG=3
 2619 CONTINUE
      FREQ(IAVET,IRANG)=FREQ(IAVET,IRANG)+1.0
 2620 CONTINUE
      DO 2833 IXYZI=1,365
      WINTIL(IXYZI)=0.0
C   DETERMINE THE TEN HOTEST DAYS OF THE YEAR
 2833 CONTINUE
      DO 2844 IFINDT=1,10
      TAXFIN=0.0
      DO 2846 IDAFIN=1,365
      IF(WINTIL(IDAFIN).EQ.1.0) GO TO 2846
      IF(TDBMAX(IDAFIN).GE.TAXFIN) TAXFIN=TDBMAX(IDAFIN)
      IF(TDBMAX(IDAFIN).EQ.TAXFIN)IWTEN(IFINDT)=IDAFIN
 2846 CONTINUE
      WINTIL(IWTEN(IFINDT))=1.0
 2844 CONTINUE
      AVTENN=0.0
      AVBCKN=0.0
      AVTENX=0.0
      AVBCKX=0.0
      DO 2863  ITZ=1,10
      AVBCKN=AVBCKN+TDBMIN(IWTEN(ITZ)-1)/10.0
      AVBCKX=AVBCKX+TDBMAX(IWTEN(ITZ)-1)/10.0
      AVTENN=AVTENN+TDBMIN(IWTEN(ITZ))/10.0
      AVTENX=AVTENX+TDBMAX(IWTEN(ITZ))/10.0
 2863 CONTINUE
      WRITE(6,1315)
 1315 FORMAT(50X,19H INPUT DATA SUMMARY)
      WRITE(6,2631)
 2631 FORMAT(35H TEMPERATURE FREQUENCY DISTRIBUTION)
      WRITE(6,2633)((PKTEM(JZA,JZR),SMTEM(JZA,JZR),FREQ(JZA,JZR),
     1 JZR=1,3),JZA=1,4)
 2633 FORMAT(11H PEAK TEMP=,F5.1,15H  MINIMUM TEMP=,F5.1,12H  FREQUENC
     1Y=,F8.1)
      WRITE(6,1318)
 1318 FORMAT(1H0)
      DO 1317   KUZK=1,24
      XAMFIX=LAMBDA(KUZK)*1000.0
      WRITE(6,1319) KUZK,XAMFIX
 1319 FORMAT(25X,4H AT ,I3,36H HOURS THE INCREMENTAL FUEL COST IS ,
     1 F8.4,12H MILLS/KWHR )
 1317 CONTINUE
      IF(ITHRMB.EQ.1) WRITE(6,1321)
 1321 FORMAT(38H0THIS RUN IS FOR A PLUG FLOW TYPE POND)
      IF(ITHRMB.EQ.2) WRITE(6,1322)
 1322 FORMAT(35H0THIS RUN IS FOR A FULLY-MIXED POND)
      WRITE(6,1344) BYPASS
```

```
1344     FORMAT(48HOTHE FRACTION OF THE FLOW BYPASSING THE POND IS ,F8.4)
         IF(PONTYP.EQ.3) WRITE(6,1323)
1323     FORMAT(34HOTHIS RUN IS FOR AN ADIABATIC POND)
         IF(PONTYP.EQ.2) WRITE(6,1324)
1324     FORMAT(34HOTHIS RUN IS FOR A REFLECTIVE POND)
         IF(PONTYP.EQ.1) WRITE(6,1325)
1325     FORMAT(34HOTHIS RUN IS FOR AN UNCOVERED POND)
         XPRATE=MPRATE*10
         WRITE(6,1326) TURMAX,XPRATE
1326     FORMAT(31HOTHE MAXIMUM CONDENSING TEMP IS,F8.1,33H AND THE MINIM
      1UM THERMAL POWER IS,F5.1,9H PER CENT)
         WRITE(6,1327) MWTHRM,POWMAX
1327     FORMAT(22H THE THERMAL POWER IS ,F10.2,41H MEGAWATTS AND THE MAX
      1IMJM GENERATION IS ,F10.2,10H MEGAWATTS)
         WRITE(6,1328)
1328     FORMAT(64HOTHE MAXIMUM TEMPERATURES OF THE TEN HOTEST  DAYS ARE
      1AS FOLLOWS)
         WRITE(6,1329)(TDBMAX(IWTEN(ITZZZ)),ITZZZ=1,10)
1329     FORMAT(10F10.1)
         WRITE(6,1339)
1339     FORMAT(78HOTHE MAXIMUM TEMPERATURES OF THE DAYS PRECEDING THE TE
      1N HOTEST  ARE AS FOLLOWS)
         WRITE(6,1340)(TDBMAX(IWTEN(ITXZ)-1),ITXZ=1,10)
1340     FORMAT(10F10.1)
         WRITE(6,1341)AVTENN,AVTENX
1341     FORMAT(46HOAVERAGE TEMPERATURES OF TEN HOTEST --MINIMUM=,F5.1,
      1  10H  MAXIMUM=,F5.1)
         WRITE(6,1342) AVBCKN,AVBCKX
1342     FORMAT(53H AVERAGE TEMPERATURES OF PRECEDING TEN DAYS--MINIMUM=,
      1  F5.1,9H MAXIMUM=,F5.1)
         IF(TOWONL) WRITE(6,1330)
1330     FORMAT(38HOTHIS RUN IS FOR TOWERS ONLY OPERATION)
C   STATE INITIAL GEUSS FOR DESIGN VARIABLES
         READ(5,133)WLG,WLCG,TOWSZG
133      FORMAT(3F10.2)
         WLG=WLG*224640.0
         WLCG=WLCG*224640.0
         WL=WLG
         WLC=WLCG
         TOWSZ=TOWSZG
2000     CONTINUE
         M=-1
         P=3
         L=3
         PR=1
         NSTEP=1
         LOOPY=50
C   THE SUBSCRIPTED VARIABLES E ARE OPTIMIZATION ROUTINE STEPPING SIZES,
C     ADJUST IF REQUIRED
         E(1)=15000000.0
         E(2)=15000000.0
         E(3)=7.0
         IF(TOWONL) E(2)=7.0
         IF(TOWONL) P=2
         IF(TOWONL) L=2
```

```
C   FOR EACH SET OF OPVAR INITIAL GEUSS IS RESULT FROM LAST SET
          IF(NOVSRI.EQ.NOPSET) GO TO 9999
          READ(5,121) TBHUOP,TEHUOP,TBCDOP,TECDOP
  121     FORMAT(4I10)
          NOVSRI=NOVSRI+1
C**************************************************************************
C   IF FINAL =0 FINAL RUN BEING MADE
C**************************************************************************
          IFINAL=1
C**************************************************************************
          IF(.NOT.TOWONL) GO TO 178
          X(1)=WLG
          X(2)=TOWSZG
          GO TO 179
  178     CONTINUE
          X(1)=WLG
          X(2)=WLCG
          X(3)=TOWSZG
  179     CONTINUE
C
C      MAIN LINE PROGRAM FOR ROSENBROCK HILLCLIMB
C
          LAP=PR-1
          LOOP=0
          ISW=0
          INIT=0
          KOUNT=0
          TERM=0.0
          DELY=0.005
          F1=0.0
          NPAR=1
          N=L
          DO 40 K=1,L
   40     AL(K)=(CH(X,N,K)-CG(X,N,K))*0.0001
          DO 60 I=1,P
          DO 60 J=1,P
          V(I,J)=0.0
          IF(I-J) 60,61,60
   61     V(I,J)=1.0
   60     CONTINUE
          DO 65 KK=1,P
          EINT(KK)=E(KK)
   65     CONTINUE
C
C
 1000     DO 70  J=1,P
          IF(NSTEP.EQ.0) E(J)=EINT(J)
          SA(J)=2.0
   70     D(J)=0.0
          FBEST=F1
   80     I=1
          IF(INIT.EQ.0) GO TO 120
   90     DO 110 K=1,P
  110     X(K)=X(K)+E(I)*V(I,K)
          DO 50  K=1,L
```

```
 50     H(K)=F0
C
C
 120    F1=F(X)
        IF(NOPTIM) GO TO 485
        F1=M*F1
        IF(ISW.EQ.0) F0=F1
        ISW=1
        IF(ABS(FREST-F1)-DELY) 122,122,125
 122    TERM=1.0
        GO TO 450
 125    CONTINUE
C
C
        J=1
C
 130    XC=CX(X,N,J)
        LC=CG(X,N,J)
        UC=CH(X,N,J)
        IF(XC.LE.LC) GO TO 420
        IF(XC.GE.UC) GO TO 420
        IF(F1.LT.F0)  GO TO 420
        IF(XC.LT.LC+AL(J)) GO TO 140
        IF(XC.GT.UC-AL(J)) GO TO 140
        H(J)=F0
        GO TO 210
C
C
 140    CONTINUE
C
        BW=AL(J)
C
        IF(XC.LE.LC.OR.UC.LE.XC) GO TO 150
        IF(LC.LT.XC.AND.XC.LT.LC+BW) GO TO 160
        IF(UC-BW.LT.XC.AND.XC.LT.UC) GO TO 170
        PH(J)=1.0
        GO TO 210
C
C
 150    PH(J)=0.0
        GO TO 190
 160    PW=(LC+BW-XC)/BW
        GO TO 180
 170    PW=(XC-UC+BW)/BW
 180    PH(J)=1.0-3.0*PW+4.0*PW*PW-2.0+PW*PW*PW
C
 190    F1=H(J)+(F1-H(J))*PH(J)
C
 210    CONTINUE
        IF(J.EQ.L) GO TO 220
        J=J+1
        GO TO 130
C
 220    INIT=1
        IF(F1.LT.F0) GO TO 420
```

```
      D(I)=D(I)+E(I)
      E(I)=3.0*E(I)
      F0=F1
      IF(.NOT.TOWONL) GO TO 227
      XSAVE(1)=X(1)
      XSAVE(2)=X(2)
      GO TO 228
  227 CONTINUE
      XSAVE(1)=X(1)
      XSAVE(2)=X(2)
      XSAVE(3)=X(3)
  228 CONTINUE
      CAPSAV=CAPLMX
      IF(SA(I).GE.1.5) SA(I)=1.0
C
  230 DO 240  JJ=1,P
      IF(SA(JJ).GE.0.5) GO TO 440
  240 CONTINUE
C
C     AXES ROTATION
C
      DO 250 R=1,P
      DO 250 C=1,P
  250 VV(C,R)=0.0
      DO 260 R=1,P
      KR=R
      DO 260 C=1,P
      DO 265 K=KR,P
  265 VV(R,C)=D(K)*V(K,C)+VV(R,C)
  260 B(R,C)=VV(R,C)
      BMAG=0.0
      DO 280 C=1,P
      BMAG=BMAG+B(1,C)*B(1,C)
  280 CONTINUE
      BMAG=SQRT(BMAG)
      BX(1)=BMAG
      DO 310 C=1,P
  310 V(1,C)=B(1,C)/BMAG
C
      DO 390 R=2,P
C
      IR=R-1
      DO 390 C=1,P
      SUMVM=0.0
      DO 320 KK=1,IR
      SUMAV=0.0
      DO 330 KJ=1,P
  330 SUMAV=SUMAV+VV(R,KJ)*V(KK,KJ)
  320 SUMVM=SUMAV*V(KK,C) + SUMVM
  390 B(R,C)=VV(R,C)-SUMVM
      DO 340 R=2,P
      BBMAG=0.0
      DO 350 K=1,P
  350 BBMAG=BBMAG+B(R,K)*B(R,K)
      BBMAG=SQRT(BBMAG)
```

```
            DO 340 C=1,P
      340   V(R,C)=B(R,C)/BBMAG
            LOOP=LOOP+1
            LAP=LAP+1
            IF(LAP.EQ.PR) GO TO 450
            GO TO 1000
C
      420   IF(INIT.EQ.0) GO TO 450
            DO 430 IX=1,P
      430   X(IX)=X(IX)-E(I)*V(I,IX)
            E(I)=-0.5*E(I)
            IF(SA(I).LT.1.5) SA(I)=0.0
            GO TO 230
C
      440   CONTINUE
            IF(I.EQ.P) GO TO 80
            I=I+1
            GO TO 90
C
      450   WRITE(6,003)
      003   FORMAT(//,2X,5HSTAGE,8X,8HFUNCTION,12X,8HPROGRESS,9X,
         1  16HLATERAL PROGRESS )
            WRITE(6,004) LOOP,FO,BMAG,BBMAG
      004   FORMAT(1H ,I5,3E20.8)
            WRITE(6,014) KOUNT
      014   FORMAT(/,2X,33HNUMBER OF FUNCTION EVALUATIONS = ,I8)
            WRITE(6,005)
      005   FORMAT(/,2X,25HVALUES OF X AT THIS STAGE )
C         PRINT CURRENT VALUES OF X
            WRITE(6,006)(JM,XSAVE(JM),JM=1,P)
      006   FORMAT(/,2X,3(2HX(,I2,4H) = ,1PE14.6,4X))
C
            LAP=0
            IF(INIT.EQ.0) GO TO 470
            IF(TERM.EQ.1.0) GO TO 480
            IF(LOOP.GE.LOOPY) GO TO 480
            GO TO 1000
C
      470   WRITE(6,007)
      007   FORMAT(///,2X,81HTHE STARTING POINT MUST NOT VIOLATE THE CONSTRA
         1INTS.  IT APPEARS TO HAVE DONE SO.)
      480   CONTINUE
C*****************************************************************************
            IFINAL=0
            EXACT=F(XSAVE)
            WRITE(6,9998) EXACT
      9998  FORMAT(43H THE EXACT COST OF THE OPTIMIZED DESIGN IS ,F7.3,
         1 15H MILLS PER KWHR)
            WLG=XSAVE(1)
            IF(TOWONL) TOWSZG=XSAVE(2)
            IF(.NOT.TOWONL) WLCG=XSAVE(2)
            IF(.NOT.TOWONL) TOWSZG=XSAVE(3)
            GO TO 2000
      485   CONTINUE
            XSAVE(1)=X(1)
```

```
      XSAVE(2)=X(2)
      IF(.NOT.TOWONL) XSAVE(3)=X(3)
      IFINAL=0
      EXACT=F(XSAVE)
      WRITE(6,9997) EXACT
 9997 FORMAT(34H THE NON-OPTIMIZED DESIGN COST IS ,F7.3)
      GO TO 2000
 9999 CONTINUE
      STOP
      END
```

```
      FUNCTION F(X)
      DIMENSION X(3)
      DIMENSION HTRATE(300,10),LAMBDA(25),TDBMAX(370),TDBMIN(370)
      DIMENSION CAPLCM(300),CAPLUM(300),TDB(30)
      DIMENSION FREQ(6,6),PKTEM(4,4),SMTEM(4,4)
      DIMENSION PLNTPF(300),LOSCAP(300),CONDEN(300),CONDN1(300)
      DIMENSION LOSCPU(300)
      DIMENSION WINTIL(400)
      DATA ((PKTEM(IIAV,IIRAN),IIRAN=1,3),IIAV=1,4)/37.5,45.0,52.5,57.
     15,65.0,72.5,77.5,85.0,92.5,97.5,105.0,112.5/

      DATA ((SMTEM(JJAV,JJRAN),JJRAN=1,3),JJAV=1,4)/22.5,15.0,7.5,42.5
     1,35.0,27.5,62.5,55.0,47.5,82.5,75.0,67.5/

      REAL LOSCPU,MLSPKW,MWTHRM
      LOGICAL PRINT1, PRINT2
      REAL MASSP
      REAL LAMBDA
      INTEGER TBHUOP,TEHUOP,TBCDOP,TECDOP
      LOGICAL TOWONL
      LOGICAL NOPTIM
      INTEGER XCONIN,XXIN,TIMMIN,TIMMAX,PONTYP
      COMMON KCOUNT
      COMMON/TURBIN/HTRATE,TURMAX
      COMMON/CAPLDT/AVBCKX,AVBCKN,AVTENX,AVTENN
      COMMON/CAPCIT/CAPLMX,CAPSAV
      COMMON/PHTRMO/PONTYP,ITHRMB
      COMMON/PRINTO/PRINT1,PRINT2
      COMMON/TOPANT/TIMMIN,TIMMAX
      COMMON/SEQE/NOPTIM
      COMMON/PMAX/POWMAX
      COMMON/HTPOND/HTOUT
      COMMON/METEOR/TDBMAX,TDBMIN
      COMMON/FULCOS/LAMBDA
      COMMON/HOTDAY/WINTIL
      COMMON/OPMODE/TOWONL
      COMMON/OMEGA/MWTHRM
      COMMON/SWITCH/IFINAL
      COMMON/TIMES/TBHUOP,TEHUOP,TBCDOP,TECDOP
      COMMON/WAS/TWASTH
      COMMON/MP/MPRATE
      COMMON/PARTFL/BYPASS
      COMMON/COST/MLSPKW
      COMMON/POND/MASSP,AREAP
      COMMON/TEMFRQ/FREQ
      WL=X(1)
      WLC=X(2)
      IF(TOWONL) TOWSZ=X(2)
      IF(.NOT.TOWONL) TOWSZ=X(3)
```

```
      SPCHET=1.0
      TPONDI=50.0
      YCAPEN=0.0
      EVAPT=0.0
      TLOSGN=0.0
      EVAPD=0.0
      HTOUT=0.0
      THTOUT=0.0
      IF(TOWONL) GO TO 2026
      PCOLSB=TBHUOP-TECDOP
      PHEATU=TEHUOP-TBHUOP
      PCOOLD=PHEATU
      PHOTSB=24.0-PCOLSB-PHEATU-PCOOLD
      VOLUM=WLC*(1.0-BYPASS)*PHEATU/62.4
      MASSP=VOLUM*62.4
      AREAP=VOLUM/20.0
      ITCHAN=0.63*PHEATU
      ITIMHF=TBHUOP+ITCHAN
      ITIMCF=TBCDOP+ITCHAN
C***************************************************************************
C   DERTERMINE CONDENSER HT MATRIX FOR COUPLED MODE
C***************************************************************************
      CITCM=250.0
      DO 2020  XCONIN=1,250
      IPOW=10
      POWFRC=1.0
 2009 ETA=0.33
 2010 ETA1=ETA
      CONIN=XCONIN
      CONOT=(POWFRC*MWTHRM*1000.0*3412.0*(1.0-ETA)+WLC*SPCHET*
     1 CONIN)/(WLC*SPCHET)
      TC=CONOT+5.0
      IF(TC.LE.85.0) TC=85.0
      IF(TC.GE.TURMAX.AND.IPOW.EQ.MPRATE)CITCM=XCONIN
      IF(TC.GE.TURMAX.AND.IPOW.EQ.MPRATE)GO TO 2021
      ITC=TC
      HEATR=HTRATE(ITC,IPOW)
      ETA=3412.0/HEATR
      DIFETA=ABS(ETA1-ETA)
      IF(DIFETA.LE.0.0001) GO TO 2015
      ETA=(ETA+ETA1)/2.0
      GO TO 2010
C   PLANT DERATES IN 10, INCREMENTS AT EXCESSIVE CONDENSING TEMPERATURES
 2015 CONTINUE
      IF(TC.LE.TURMAX) GO TO 2017
      POWFRC=POWFRC-0.1
      IPOW=IPOW-1
      GO TO 2009
 2017 CONDEN(XCONIN)=CONOT
      POWRR=POWFRC*MWTHRM
      CAPLCM(XCONIN)=POWMAX-ETA*POWRR
      IF(IFINAL.EQ.1) GO TO 2020
 2020 CONTINUE
 2021 CONTINUE
      ICITCM=CITCM
```

```
      DO 2025  IITEM=ICITCM,250
      CAPLCM(IITEM)=POWMAX
      CONDEN(IITEM)=IITEM
 2025 CONTINUE
 2026 CONTINUE
      NIDWC=2
      IF(TOWONL) NIDWC=1
      DO 2044  IDWC=1,NIDWC
      IF(IDWC.EQ.1) XWL=WL
      IF(IDWC.EQ.2) XWL=WLC
C*******************************************************************************
C    DETERMINE TOWER PERFORMANCE COEFFICIENT
C*******************************************************************************
C    ALTERNATIVE TOWER DESIGNS MAY BE CONSIDERED BY CHANGING THE NEXT TWO CARDS
      AUF=1120000.0
      AIRFLO=11466000.0
      ARTFLO=AIRFLO*TOWSZ
      TAUF=TOWSZ*AUF
C    PICK ARBITARY ITD, TAIR
      ARITD=60.0
      ARTAIR=95.0
C    GEUSS AT TOWER OUTLET TEMPERATURE
      ARTOWO=ARTAIR+30.0
      ADD=1.0
 2030 CONTINUE
      ARQJET=XWL*(ARTAIR+ARITD-ARTOWO)
      ARTARO=ARQJET/(ARTFLO*0.24*TOWSZ)+ARTAIR
      DELTAA=ARTAIR+ARITD-ARTARO
      DELTAB=ARTOWO-ARTAIR
      IF(DELTAA.LE.0.0.OR.DELTAB.LE.0.0) ARTOWO=ARTAIR+ADD
      IF(DELTAA.LE.0.0.OR.DELTAB.LE.0.0) ADD=ADD+1.0
      IF(DELTAA.LE.0.0.OR.DELTAB.LE.0.0) GO TO 2030
      ARQJT1=TAUF*(DELTAA-DELTAB)/ALOG(DELTAA/DELTAB)
      ARTWO1=(ARTAIR+ARITD)-ARQJT1/XWL
      ARDIF=ABS(ARTOWO-ARTWO1)
      IF(ARDIF.LE.0.1) GO TO 2040
      ARTOWO=(ARTOWO+ARTWO1)/2.0
      GO TO 2030
 2040 CONTINUE
      IF(IDWC.EQ.1)CUNIT=ARQJT1/ARITD
      IF(IDWC.EQ.2) CUNITC=ARQJT1/ARITD
      IF(IFINAL.EQ.1) GO TO 2045
      IF(.NOT.PRINT2) GO TO 2045
      IF(IDWC.EQ.1) WRITE(6,2041) CUNIT
 2041 FORMAT(53H TOWER PERFORMANCE COEFFICIENT FOR UNCOUPLED MODES = ,
     1 F20.4)
      IF(IDWC.EQ.2) WRITE(6,2042) CUNITC
 2042 FORMAT(51H TOWER PERFORMANCE COEFFICIENT FOR COUPLED MODES = ,
     1 F20.4)
 2045 CONTINUE
 2044 CONTINUE
C*******************************************************************************
C    DETERMINE CONDENSER HT MATRIX FOR UNCOUPLED MODE
C*******************************************************************************
      DO 2049 XCONIN=32,200
```

```
            IPOW=10
            POWFRC=1.0
      2051  ETA=0.33
      2052  ETA1=ETA
            CONIN=XCONIN
            XCONOT=(POWFRC*MWTHRM*1000.0*3412.0*(1.0-ETA)+WL*SPCHET*CONIN)/
     1        (WL*SPCHET)
            TC=XCONOT+5.0
            IF(TC.LE.85.0) TC=85.0
            IF(TC.GE.TURMAX.AND.IPOW.EQ.MPRATE)CITUCM=XCONIN
            IF(TC.GE.TURMAX.AND.IPOW.EQ.MPRATE)GO TO 2069
            ITC=TC
            HEATR=HTRATE(ITC,IPOW)
            ETA=3412.0/HEATR
            DIFETA=ABS(ETA1-ETA)
            IF(DIFETA.LE.0.0001) GO TO 2054
            ETA=(ETA+ETA1)/2.0
            GO TO 2052
      2054  CONTINUE
            IF(TC.LE.TURMAX) GO TO 2053
            POWFRC=POWFRC-0.1
            IPOW=IPOW-1
            GO TO 2051
      2053  CONDN1(XCONIN)=XCONOT
            POWRR=POWFRC*MWTHRM
            CAPLUM(XCONIN)=POWMAX-ETA*POWRR
      2049  CONTINUE
      2069  CONTINUE
            ICTUCM=CITUCM
            FAKDIF=CONDN1(ICTUCM-1)-CITUCM-1.0
            DO 2070  IITEM=ICTUCM,250
            RIITEM=IITEM
            CONDN1(IITEM)=RIITEM+FAKDIF
            CAPLUM(IITEM)=POWMAX
      2070  CONTINUE
C*****************************************************************************
C    DETERMINE PERFORMANCE MATRIX FOR UNCOUPLED MODE
C*****************************************************************************
            DO 2060   IITDB=1,120
            RITDB=IITDB
            XXIN=IITDB+30
            IF(IITDB.LE.40) RITDB=40.0
            IF(IITDB.LE.40) XXIN=70
            RXXIN= XXIN
      2055  CONTINUE
            IRXXIN=RXXIN
            IRXXN1=IRXXIN+1
            RIRXXN=IRXXIN
            CONOT=CONDN1(IRXXIN)+(RXXIN-RIRXXN)*(CONDN1(IRXXN1)-CONDN1(IRXXI
     1N))
            TITD=CONOT-RITDB
            QJECT=CUNIT*TITD
            TOWOT=CONOT-QJECT/WL
            RXXIN1=TOWOT
            XXDIF=ABS(RXXIN1-RXXIN)
```

```
          IF(XXDIF.LE.0.1) GO TO 2057
          RXXIN=(RXXIN+RXXIN1)/2.0
          GO TO 2055
 2057     CONTINUE
          LOSCPU(IITDB)=CAPLUM(IRXXIN)+(RXXIN-RIRXXN)*(CAPLUM(IRXXN1)-
     1CAPLUM(IRXXIN))
 2060     CONTINUE
C    INITIALIZE VARIABLES FOR SIMULATION
          MODSYS=1
          TPOND=TPONDI
          TPONDH=TPONDI
          TPONDC=TPONDI
          JDAY=0
          PHASE1=TIMMIN
          PHASE2=TIMMAX
          CYC1=PHASE1+24.0-PHASE2
          CYC2=24.0-CYC1
          PIE=3.141
          EVAP=0.0
          FANPOW=139.6*TOWSZ
          FLTOWU=WL/TOWSZ
          DELPR1=42.0*(FLTOWU/1.835E6)**2
          PPUMPU=WL*DELPR1/2.67E6
          YTOWOP=0.0
          IF(TOWONL) CAPLMX=0.0
          IF(TOWONL) GO TO 10000
          FLTOWC=WLC/TOWSZ
          DELPR2=42.0*(FLTOWC/1.835E6)**2
          PPUMPC=WLC*DELPR2/2.67E6
          MIN111=TEHUOP-1
          MIN222=TECDOP-1
          IF(IFINAL.EQ.1) GO TO 9000
          IF(.NOT.NOPTIM) CAPLMX=CAPSAV
          AVPCLD=0.0
C***********************************************************************
C    IF NOT FINAL COST DETERMINATION - GO TO PSEUDO SIMULATION
C***********************************************************************
C    EXACT SIMULATION - STATEMENTS 3000 TO 4000
C***********************************************************************
C    ENTER DAILY LOOP
C***********************************************************************
 3000     CONTINUE
          JDAY=JDAY+1
          JBEFOR=JDAY-1
          JNEXT=JDAY+1
          IF(JBEFOR.EQ.0) JBEFOR=1
          IF(JNEXT.EQ.366) JNEXT=365
          AMP1=-TDBMIN(JDAY)+TDBMAX(JBEFOR)
          AMP2=TDBMAX(JDAY)-TDBMIN(JDAY)
          AMP3=-TDBMIN(JNEXT)+TDBMAX(JDAY)
          IOUROP=0
 3050     CONTINUE
C***********************************************************************
C    ENTER HOURLY LOOP
C***********************************************************************
```

```
      IOUROP=IOUROP+1
      ROUROP=IOUROP
      IF(IOUROP.LE.TIMMIN) TDB(IOUROP)=-AMP1*SIN((PIE/(2.0*CYC1))*
     1 (ROUROP+(CYC1-PHASE1)))+AMP1+TDBMIN(JDAY)
      IF(IOUROP.GT.TIMMIN.AND.IOUROP.LE.TIMMAX) TDB(IOUROP)=AMP2*
     1 SIN((PIE/(2.0*CYC2))*(ROUROP-PHASE1))+TDBMIN(JDAY)
      IF(IOUROP.GE.TIMMAX) TDB(IOUROP)=-AMP3*SIN((PIE/(2.0*CYC1))*
     1 (ROUROP-PHASE2))+TDBMAX(JDAY)
      IF(TDB(IOUROP).LE.1.0) TDB(IOUROP)=1.0
      IPTDB=TDB(IOUROP)
      RIPTDB=IPTDB
      FPTDB=TDB(IOUROP)-RIPTDB
      IPTDB1=IPTDB+1
      IF(TOWONL) GO TO 10100
      IF(MODSYS.EQ.1) GO TO 3100
      IF(MODSYS.EQ.2) GO TO 3200
      IF(MODSYS.EQ.3) GO TO 3300
      IF(MODSYS.EQ.4) GO TO 3400
C************************************************************************
C   CALCULATIONS FOR COLD STANDBY MODE
C************************************************************************
 3100 CONTINUE
      IF(IOUROP.EQ.TBHUOP) MODSYS=2
      IF(IOUROP.EQ.TBHUOP) GO TO 3200
      XLOSS=LOSCPU(IPTDB)+FPTDB*(LOSCPU(IPTDB1)-LOSCPU(IPTDB))
      TLOSGN=TLOSGN+XLOSS
      OPPEN=LAMBDA(IOUROP)*XLOSS*1000.0
      YCAPEN=YCAPEN+OPPEN
      GO TO 5000
C************************************************************************
C   CALCULATIONS FOR HEATUP MODE
C************************************************************************
 3200 CONTINUE
      IF(IOUROP.EQ.TEHUOP) MODSYS=3
      IF(MODSYS.EQ.3) GO TO 3300
      IF(IOUROP.NE.TBHUOP) GO TO 3225
      TPONDC=TPOND
      IF(PONTYP.EQ.3) GO TO 3220
      IF(JDAY.NE.1) GO TO 3216
      HAFTIM=0.5*PCOLSB+0.25*(PCOOLD+PHEATU)
      BCDOP=TBCDOP
      TFCEV=BCDOP+0.5*PCOOLD+HAFTIM
      ITFCEV=TFCEV
      TIMEX=2.0*HAFTIM
 3216 CONTINUE
      TDBA=TDB(ITFCEV)
      CALL PONTEM(PONTYP,TPOND,EVAP,TDBA,TIMEX)
      EVAPT=EVAPT+EVAP
 3220 CONTINUE
 3225 CONTINUE
      JCTC=0
      TGEUSS=TPOND
 3241 IGEUSS=TGEUSS
      RIGEUS=IGEUSS
      IGEUS1=IGEUSS+1
```

```
          COLT=CONDEN(IGEUSS)+(TGEUSS-RIGEUS)*(CONDEN(IGEUS1)-CONDEN(IGEUS
     1S))
          TLET=COLT-(CUNITC*(COLT-TDB(IOUROP))/WLC)
          IF(IOUROP.EQ.TBHUOP) TLET1=TLET
          IF(IOUROP.LE.ITIMHF.AND.ITHRMB.EQ.1) TGEUSN=TPOND*(1.0-BYPASS)+
     1    TLET*BYPASS
          IF(IOUROP.GT.ITIMHF.AND.ITHRMB.EQ.1) TGEUSN=(TPOND+0.33*
     1    (TLET1-TPOND))*(1.0-BYPASS)+TLET*BYPASS
          IF(ITHRMB.EQ.2) TGEUSN=TPOND*(1.0-BYPASS)+TLET*BYPASS
          DFXX=ABS(TGEUSN-TGEUSS)
          IF(DFXX.LE.0.3) GO TO 3229
          JCTC=JCTC+1
          IF(JCTC.GE.20) WRITE(6,9242) TGEUSS
          IF(JCTC.GE.25) GO TO 3229
          TGEUSS=(TGEUSS+TGEUSN)/2.0
          GO TO 3241
 3229     XLOSS=CAPLCM(IGEUSS)+(TGEUSS-RIGEUS)*(CAPLCM(IGEUS1)-CAPLCM(IGEU
     1SS))
          IF(IOUROP.EQ.TBHUOP) TPONDS=0.0
          IF(ITHRMB.EQ.1) TPONDS=TPONDS+TLET*((WLC*(1.0-BYPASS))/MASSP)
          IF(ITHRMB.EQ.1.AND.IOUROP.EQ.MIN111) TPOND=TPONDS
          IF(ITHRMB.EQ.2) TPOND=(TPOND*(MASSP-WLC*(1.0-BYPASS))+TLET*
     1    (WLC*(1.0-BYPASS)))/MASSP
          TLOSGN=TLOSGN+XLOSS
          OPPEN=LAMBDA(IOUROP)*XLOSS*1000.0
          YCAPEN=YCAPEN+OPPEN
          IF(IOUROP.EQ.TBHUOP) THTOUT=THTOUT+HTOUT/TWASTH
          IF(.NOT.PRINT2) GO TO 5000
          IF(IOUROP.EQ.TBHUOP) AVPHTU=0.0
          AVPHTU=((POWMAX-XLOSS)/PHEATU)+AVPHTU
          IF(IOUROP.EQ.TBHUOP) EVAPD=EVAP+EVAPD
          GO TO 5000
C*****************************************************************************
C    CALCULATIONS FOR HOT STANDBY MODE
C*****************************************************************************
 3300     CONTINUE
          IF(IOUROP.EQ.TBCDOP) MODSYS=4
          IF(MODSYS.EQ.4) GO TO 3400
          XLOSS=LOSCPU(IPTDB)+FPTDB*(LOSCPU(IPTDB1)-LOSCPU(IPTDB))
          TLOSGN=TLOSGN+XLOSS
          OPPEN=LAMBDA(IOUROP)*XLOSS*1000.0
          YCAPEN=YCAPEN+OPPEN
          GO TO 5000
C*****************************************************************************
C    CALCULATIONS FOR COOLDOWN MODE
C*****************************************************************************
 3400     CONTINUE
          IF(IOUROP.EQ.TECDOP) MODSYS=1
          IF(IOUROP.EQ.TECDOP) MODSYS=1
          IF (MODSYS.EQ.1) GO TO 3100
          IF(IOUROP.NE.TBCDOP) GO TO 3425
          TPONDH=TPOND
          IF(PONTYP.EQ.3) GO TO 3420
          IF(JDAY.NE.1) GO TO 3416
          HAFTIM=0.5*PHOTSB+0.24*(PCOOLD+PHEATU)
```

```
      BHUOP=TBHUOP
      TFCEV1=BHUOP+0.5*PHEATU+HAFTIM
      ITFCV1=TFCEV1
      TIMEX2=2.0*HAFTIM
      IF(ITFCV1.GT.24) ITFCV1=ITFCV1-24
 3416 CONTINUE
      TDBA=TDB(ITFCV1)
      CALL PONTEM(PONTYP,TPOND,EVAP,TDBA,TIMEX2)
      EVAPT=EVAPT+EVAP
 3420 CONTINUE
 3425 CONTINUE
      JCTC=0
      TGEUSS=TPOND
 3441 IGEUSS=TGEUSS
      RIGEUS=IGEUSS
      IGEUS1=IGEUSS+1
      COLT=CONDEN(IGEUSS)+(TGEUSS-RIGEUS)*(CONDEN(IGEUS1)-CONDEN(IGEUS
     1S))
      TLET=COLT-(CUNITC*(COLT-TDB(IOUROP))/WLC)
      IF(IOUROP.EQ.TBCDOP) TLET1=TLET
      IF(IOUROP.LE.ITIMCF.AND.ITHRMB.EQ.1) TGEUSN=TPOND*(1.0-BYPASS)+
     1 TLET*BYPASS
      IF(IOUROP.GT.ITIMCF.AND.ITHRMB.EQ.1) TGEUSN=(TPOND+0.33*
     1 (TLET1-TPOND))*(1.0-BYPASS)+TLET*BYPASS
      IF(ITHRMB.EQ.2) TGEUSN=TPOND*(1.0-BYPASS)+TLET*BYPASS
      DFXX=ABS(TGEUSN-TGEUSS)
      IF(DFXX.LE.0.3) GO TO 3429
      JCTC=JCTC+1
      IF(JCTC.GE.20) WRITE(6,9242) TGEUSS
      IF(JCTC.GE.25) GO TO 3429
      TGEUSS=(TGEUSS+TGEUSN)/2.0
      GO TO 3441
 3429 XLOSS=CAPLCM(IGEUSS)+(TGEUSS-RIGEUS)*(CAPLCM(IGEUS1)-CAPLCM(IGEU
     1SS))
      IF(IOUROP.EQ.TBCDOP) TPONDS=0.0
      IF(ITHRMB.EQ.1) TPONDS=TPONDS+TLET*((WLC*(1.0-BYPASS))/MASSP)
      IF(ITHRMB.EQ.1.AND.IOUROP.EQ.MIN222)TPOND=TPONDS
      IF(ITHRMB.EQ.2) TPOND=(TPOND*(MASSP-WLC*(1.0-BYPASS))+TLET*
     1 (WLC*(1.0-BYPASS)))/MASSP
      TLOSGN=TLOSGN+XLOSS
      OPPEN=LAMBDA(IOUROP)*XLOSS*1000.0
      YCAPEN=YCAPEN+OPPEN
      IF(IOUROP.EQ.TBCDOP) THTOUT=THTOUT+HTOUT/TWASTH
      IF(.NOT.PRINT2) GO TO 5000
      AVPCLD=AVPCLD+((POWMAX-XLOSS)/PCOOLD)
      IF(IOUROP.EQ.TBCDOP) EVAPD=EVAP+EVAPD
      GO TO 5000
 5000 CONTINUE
C***********************************************************************
C    SUM DAILY PARAMETERS - PRINT OUT DAILY PERFORMANCE
C***********************************************************************
      OPFAN=FANPOW*LAMBDA(IOUROP)
      IF(MODSYS.EQ.1.OR.MODSYS.EQ.3) OPUMP=PPUMPU*LAMBDA(IOUROP)
      IF(MODSYS.EQ.2.OR.MODSYS.EQ.4) OPUMP=PPUMPC*LAMBDA(IOUROP)
      YTOWOP=YTOWOP+OPUMP+OPFAN
```

```
            IF(IOUROP.LT.24) GO TO 3050
            IF(.NOT.PRINT2) GO TO 5500
            IF(JDAY.EQ.1) LCOUNT=0
            IF(LCOUNT.EQ.45) LCOUNT=0
            IF(LCOUNT.NE.0) GO TO 5150
            LCOUNT=LCOUNT+1
            WRITE(6,5099)
 5099       FORMAT(1H1)
            WRITE(6,5101)
 5101       FORMAT(120H    DAY       MAXIMUM   MINIMUM    AVERAGE    AVERAGE
       1COLD       HOT         EVAP        HEAT TR
       2 )
            WRITE(6,5102)
 5102       FORMAT(120H                 AMBIENT   AMBIENT   POWER -    POWER -
       1POND       POND        LOSS        FROM
       2 )
            WRITE(6,5103)
 5103       FORMAT(120H                   TEMP       TEMP      HEATUP     COOLDOWN
       1TEMP       TEMP     (LBS/DAY)    POND(%)
       2 )
 5150       CONTINUE
            WRITE(6,5155) JDAY,TDBMAX(JDAY),TDBMIN(JDAY),AVPHTU,AVPCLD,
       1    TPONDC,TPONDH,EVAPD,THTOUT
 5155       FORMAT(I5,2X,6F10.1,F13.1,F10.1)
            EVAPD=0.0
 5500       CONTINUE
            IF(JDAY.EQ.1) TOTHET=0.0
            TOTHET=TOTHET+THTOUT/365.0
            THTOUT=0.0
            AVPCLD=0.0
            IF(JDAY.EQ.365) GO TO 7000
            GO TO 3000
 7000       CONTINUE
C***************************************************************************
C    COST DETERMINATIONS
C***************************************************************************
            IF(TOWONL) VOLUM=0.0
            XWLX=WL
            IF(WL.LT.WLC) XWLX=WLC
            XWLX=XWLX/224640.0
C    CHANGES IN BASIC ECONOMIC PARAMETERS MAY BE ACCOMPLISHED BY APPROPRIATLY
C      ADJUSTING THE CONSTANTS IN THE NEXT TEN CARDS
            PCF=0.75
            AFCR=0.15
            CAPTOO=132000.0*TOWSZ
            CAPPUP=30000.0*TOWSZ*((XWLX/1154.0)**2.0)*(TOWSZ/141.0)
            CAPTOW=CAPTOO+CAPPUP
            CAPTOW=1.25*CAPTOW
            CAPTSP=515000.0  +  0.175*VOLUM
            IF(PONTYP.EQ.1) CAPTSP=515000.0+0.10*VOLUM
            IF(TOWONL) CAPTSP=0.0
            CAPRC=CAPLMX*150000.0
            TCAPC=CAPTOW+CAPTSP+CAPRC
            TOPC=(YTOWOP+YCAPEN)*PCF
            TOTANC=(TCAPC*AFCR)  +  TOPC
```

```
              YMELGN=POWMAX*1000.0*8760.0*PCF
              IF(TOWONL) GO TO 8500
              IF(.NOT.PRINT1) GO TO 8000
              WRITE(6,7100)
       7100   FORMAT(1H0,20X,30HOPERATION AND DESIGN VARIABLES)
C*******************************************************************************
C     P?INTOUT OF SIMULATION RESULTS
C*******************************************************************************
              WRITE(6,7101)
       7101   FORMAT(80H   TIME       TIME       TIME       TIME    # TOWER     F
           1LOW      FLOW      POND    )
              WRITE(6,7102)
       7102   FORMAT(80H HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COU
           1P M3DE UNCO MODE   AREA    )
              WLCCUF=WLC/224640.0
              WLCUF=WL/224640.0
              PONARA=A?EAP/43560.0
              WRITE(6,7103) TBHUOP,TEHUOP,TBCDOP,TECDOP,TOWSZ,WLCCUF,WLCUF,PON
           1ARA
       7103   FORMAT(I5,3I10,4F10.1)
              GO TO 8700
       8500   CONTINUE
              WLCUF=WL/224640.0
              WRITE(6,8501) TOWSZ,WLCUF
       8501   FORMAT(20H NUMBER OF TOWERS = ,F10.3,14H  FLOW RATE = ,F20.4)
       8700   CONTINUE
              MLSPKW=(TOTANC/YMELGN)*1000.0
              WRITE(6,7700) MLSPKW
              IF(IFINAL.EQ.1) GO TO 8900
              WRITE(6,8801)
       8801   FORMAT(1H1,30X,21H FINAL DESIGN SUMMARY)
              IF(.NOT.TOWONL) GO TO 8319
              TBHU3P=0
              TEHUOP=0
              TBCDOP=0
              TECDOP=0
              WLCCUF=0.0
              PONARA=0.0
              TOTHET=0.0
       8319   CONTINUE
              WRITE(6,8802)
       8802   FORMAT(1H0)
              WRITE(6,7100)
              WRITE(6,7101)
              WRITE(6,7102)
              WRITE(6,7103) TBHUOP,TEHUOP,TBCDOP,TECDOP,TOWSZ,WLCCUF,WLCUF,
           1  PONARA
              WRITE(6,8803)
       8803   FORMAT(1H0,45X,13H SYSTEM COSTS)
              WRITE(6,8813)
       8813   FORMAT(28X,13HCAPITAL COSTS,35X,24HOPERATING COSTS (ANNUAL))
              WRITE(6,8804)
       8804   FORMAT(5X,13H TOWER CELLS ,5X,13H STORAGE POND,5X,27HREPLACEMENT
           1 CAPABILITY     ,5X,14HFANS AND PUMPS,5X,11HENERGY REPL)
              YTOWOP=PCF*YTOWOP
```

```
               YCAPEN=PCF*YCAPEN
               WRITE(6,8805) CAPTOW,CAPTSP,CAPRC,YTOWOP,YCAPEN
      8805     FORMAT(8X,F10.1,8X,F10.1,13X,F10.1,17X,F10.1,9X,F10.1)
               WRITE(6,8806) CAPLMX
      8806     FORMAT(1H0,28H MAXIMUM CAPABILITY LOSS IS ,F10.2,7H MWS(E))
               WRITE(6,8807) TLOSGN
      8807     FORMAT(38H  TOTAL LOST ELECTRICAL GENERATION IS ,F10.1,6H MWHRS)
               WRITE(6,8810) EVAPT,TOTHET
      8810     FORMAT(32H  THE EVAPORATIVE WATER LOSS IS ,F18.2,60H LBS   AND T
           1HE TOTAL HEAT TRANSFER FROM THE POND SURFACE IS ,F5.1,3H % )
      8900     CONTINUE
      7700     FORMAT(20H INCREMENTAL COST = ,F8.3,11H MILLS/KWHR)
      8000     CONTINUE
               F=(TOTANC/YMELGN)*1000.0
               KCOUNT=KCOUNT+1
               RETURN
      C*****************************************************************************
      C  BEGIN AVERAGED-METEOROLOGICAL-CONDITION PSEUDO SIMULATION
      C*****************************************************************************
      9000     CONTINUE
               TPOND=TPONDI
               HAFTIM=0.5*PHOTSB+0.24*(PCOOLD+PHEATU)
               BHUOP=TBHUOP
               TFCEV1=BHUOP+0.5*PHEATU+HAFTIM
               ITFCV1=TFCEV1
               TIMEX2=2.0*HAFTIM
               IF(ITFCV1.GT.24) ITFCV1=ITFCV1-24
               HAFTIM=0.5*PCOLSB+0.25*(PCOOLD+PHEATU)
               BCDOP=TBCDOP
               TFCEV=BCDOP+0.5*PCOOLD+HAFTIM
               ITFCEV=TFCEV
               TIMEX=2.0*HAFTIM
               SMTEM(4,4)=AVBCKN
               PKTEM(4,4)=AVBCKX
               DO 9500 IAVE=1,4
               NORNGI=3
               IF(IAVE.EQ.4) NORNGI=4
               DO 9499  IRANG=1,NORNGI
               KKCOUT=0
      9025     CONTINUE
               YCAPET=0.0
               YTOWOT=0.0
               TPCHEK=TPOND
               AMP=PKTEM(IAVE,IRANG)-SMTEM(IAVE,IRANG)
      9026     CONTINUE
               IOUROP=0
      9099     CONTINUE
      C   ENTER HOURLY LOOP
               IOUROP=IOUROP+1
               ROUROP=IOUROP
               IF(IOUROP.LE.TIMMIN) TDB(IOUROP)=-AMP *SIN((PIE/(2.0*CYC1))*
           1    (ROUROP+(CYC1-PHASE1)))+AMP  + SMTEM(IAVE,IRANG)
               IF(IOUROP.GT.TIMMIN.AND.IOUROP.LE.TIMMAX) TDB(IOUROP)=AMP *
           1    SIN((PIE/(2.0*CYC2))*(ROUROP-PHASE1))+SMTEM(IAVE,IRANG)
               IF(IOUROP.GE.TIMMAX) TDB(IOUROP)=-AMP *SIN((PIE/(2.0*CYC1))*
```

```
     1    (ROUROP-PHASE2))+PKTEM(IAVE,IRANG)
          IF(TDB(IOUROP).LE.1.0) TDB(IOUROP)=1.0
          IPTDB=TDB(IOUROP)
          RIPTDB=IPTDB
          FPTDB=TDB(IOUROP)-RIPTDB
          IPTDB1=IPTDB+1
          IF(MODSYS.EQ.1) GO TO 9100
          IF(MODSYS.EQ.2) GO TO 9200
          IF(MODSYS.EQ.3) GO TO 9300
          IF(MODSYS.EQ.4) GO TO 9400
C***************************************************************************
C    CALCULATIONS FOR COLD STANDBY MODE
 9100     CONTINUE
          IF(IOUROP.EQ.TBHUOP) MODSYS=2
          IF(IOUROP.EQ.TBHUOP) GO TO 9200
          XLOSS=LOSCPU(IPTDB)+FPTDB*(LOSCPU(IPTDB1)-LOSCPU(IPTDB))
          GO TO 9489
C***************************************************************************
C    CALCULATIONS FOR HEATUP MODE
C***************************************************************************
 9200     CONTINUE
          IF(IOUROP.EQ.TEHUOP) MODSYS=3
          IF(MODSYS.EQ.3) GO TO 9300
          IF(IOUROP.NE.TBHUOP) GO TO 9225
          TPONDC=TPOND
          IF(PONTYP.EQ.3) GO TO 9220
          TDBA=TDB(ITFCEV)
          IF(IAVE.EQ.4.AND.IRANG.EQ.4) TDBAS=TDBA
          CALL PONTEM(PONTYP,TPOND,EVAP,TDBA,TIMEX)
 9220     CONTINUE
 9225     CONTINUE
 9242     FORMAT(8H TGEUSS=,F10.1)
          JCTC=0
          TGEUSS=TPOND
 9241     IGEUSS=TGEUSS
          RIGEUS=IGEUSS
          IGEUS1=IGEUSS+1
          COLT=CONDEN(IGEUSS)+(TGEUSS-RIGEUS)*(CONDEN(IGEUS1)-CONDEN(IGEUS
     1S))
          TLET=COLT-(CUNITC*(COLT-TDB(IOUROP))/WLC)
          IF(IOUROP.EQ.TBHUOP) TLET1=TLET
          IF(IOUROP.LE.ITIMHF.AND.ITHRMB.EQ.1) TGEUSN=TPOND*(1.0-BYPASS)+
     1    TLET*BYPASS
          IF(IOUROP.GT.ITIMHF.AND.ITHRMB.EQ.1) TGEUSN=(TPOND+0.33*
     1    (TLET1-TPOND))*(1.0-BYPASS)+TLET*BYPASS
          IF(ITHRMB.EQ.2) TGEUSN=TPOND*(1.0-BYPASS)+TLET*BYPASS
          DFXX=ABS(TGEUSN-TGEUSS)
          IF(DFXX.LE.0.3) GO TO 9229
          JCTC=JCTC+1
          IF(JCTC.GT.40) GO TO 9229
          TGEUSS=(TGEUSS+TGEUSN)/2.0
          GO TO 9241
 9229     XLOSS=CAPLCM(IGEUSS)+(TGEUSS-RIGEUS)*(CAPLCM(IGEUS1)-CAPLCM(IGEU
     1SS))
          IF(IOUROP.EQ.TBHUOP) TPONDS=0.0
```

```
      IF(ITHRMB.EQ.1) TPONDS=TPONDS+TLET*((WLC*(1.0-BYPASS))/MASSP)
      IF(ITHRMB.EQ.1.AND.IOUROP.EQ.MIN111) TPOND=TPONDS
      IF(ITHRMB.EQ.2) TPOND=(TPOND*(MASSP-WLC*(1.0-BYPASS))+TLET*
     1 (WLC*(1.0-BYPASS)))/MASSP
      XLOSAV=XLOSS
      GO TO 9489
C***********************************************************************
C    CALCULATIONS FOR HOT STANDBY MODE
C***********************************************************************
 9300 CONTINUE
      IF(IOUROP.EQ.TBCDOP) MODSYS=4
      IF(MODSYS.EQ.4) GO TO 9400
      XLOSS=LOSCPU(IPTDB)+FPTDB*(LOSCPU(IPTDB1)-LOSCPU(IPTDB))
      GO TO 9489
C***********************************************************************
C    CALCULATIONS FOR COOLDOWN MODE
C***********************************************************************
 9400 CONTINUE
      IF(IOUROP.EQ.TECDOP) MODSYS=1
      IF(IOUROP.EQ.TECDOP) MODSYS=1
      IF (MODSYS.EQ.1) GO TO 9100
      IF(IOUROP.NE.TBCDOP) GO TO 9425
      TPONDH=TPOND
      IF(PONTYP.EQ.3) GO TO 9420
      TDBA=TDB(ITFCV1)
      CALL PONTEM(PONTYP,TPOND,EVAP,TDBA,TIMEX2)
 9420 CONTINUE
 9425 CONTINUE
      JCTC=0
      TGEUSS=TPOND
 9441 IGEUSS=TGEUSS
      RIGEUS=IGEUSS
      IGEUS1=IGEUSS+1
      COLT=CONDEN(IGEUSS)+(TGEUSS-RIGEUS)*(CONDEN(IGEUS1)-CONDEN(IGEUS
     1S))
      TLET=COLT-(CUNITC*(COLT-TDB(IOUROP))/WLC)
      IF(IOUROP.EQ.TBCDOP) TLET1=TLET
      IF(IOUROP.LE.ITIMCF.AND.ITHRMB.EQ.1) TGEUSN=TPOND*(1.0-BYPASS)+
     1 TLET*BYPASS
      IF(IOUROP.GT.ITIMCF.AND.ITHRMB.EQ.1) TGEUSN=(TPOND+0.33*
     1 (TLET1-TPOND))*(1.0-BYPASS)+TLET*BYPASS
      IF(ITHRMB.EQ.2) TGEUSN=TPOND*(1.0-BYPASS)+TLET*BYPASS
      DFXX=ABS(TGEUSN-TGEUSS)
      IF(DFXX.LE.0.3) GO TO 9429
      JCTC=JCTC+1
      IF(JCTC.GT.40) GO TO 9429
      TGEUSS=(TGEUSS+TGEUSN)/2.0
      GO TO 9441
 9429 XLOSS=CAPLCM(IGEUSS)+(TGEUSS-RIGEUS)*(CAPLCM(IGEUS1)-CAPLCM(IGEU
     1SS))
      IF(IOUROP.EQ.TBCDOP) TPONDS=0.0
      IF(ITHRMB.EQ.1) TPONDS=TPONDS+TLET*((WLC*(1.0-BYPASS))/MASSP)
      IF(ITHRMB.EQ.1.AND.IOUROP.EQ.MIN222)TPOND=TPONDS
      IF(ITHRMB.EQ.2) TPOND=(TPOND*(MASSP-WLC*(1.0-BYPASS))+TLET*
     1 (WLC*(1.0-BYPASS)))/MASSP
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      GO TO 9489
 9489 CONTINUE
      OPPENT=LAMBDA(IOUROP)*XLOSS*1000.0
      OPFAN=FANPOW*LAMBDA(IOUROP)
      IF(MODSYS.EQ.1.OR.MODSYS.EQ.3) OPUMP=PPUMPU*LAMBDA(IOUROP)
      IF(MODSYS.EQ.2.OR.MODSYS.EQ.4) OPUMP=PPUMPC*LAMBDA(IOUROP)
      YCAPET=YCAPET+OPPENT
      YTOWOT=YTOWOT+OPUMP+OPFAN
      IF(IOUROP.LT.24) GO TO 9099
      DIFPON=ABS(TPOND-TPCHEK)
      IF(DIFPON.LE.0.3) GO TO 9490
      TPOND=(TPOND+TPCHEK)/2.0
      KKCOUT=KKCOUT+1
      IF(KKCOUT.GT.30) GO TO 9490
      GO TO 9025
 9490 CONTINUE
      IF(IAVE.EQ.4.AND.IRANG.EQ.4) TPCPLO=TPONDC
      IF(IAVE.EQ.4.AND.IRANG.EQ.4) GO TO 9499
      YCAPEN=YCAPEN+FREQ(IAVE,IRANG)*YCAPET
      YTOWOP=YTOWOP+FREQ(IAVE,IRANG)*YTOWOT
 9499 CONTINUE
 9500 CONTINUE
C*****************************************************************************
C   DETERMINE MAXIMUM CAPABILITY LOSS
C*****************************************************************************
      IOUROP=TBHUOP
      TPOND=TPCPLO
      IF(PONTYP.EQ.3) GO TO 9725
      TDBA=TDBAS
      CALL PONTEM(PONTYP,TPOND,EVAP,TDBA,TIMEX)
 9725 CONTINUE
      CAPLMX=0.0
      AMP=AVTENX-AVTENN
      DO 9811 IOUROP=TBHUOP,MIN111
      ROUROP=IOUROP
      IF(IOUROP.LE.TIMMIN) TDB(IOUROP)=-AMP *SIN((PIE/(2.0*CYC1))*
     1 (ROUROP+(CYC1-PHASE1)))+AMP   + AVTENN
      IF(IOUROP.GT.TIMMIN.AND.IOUROP.LE.TIMMAX) TDB(IOUROP)=AMP *
     1 SIN((PIE/(2.0*CYC2))*(ROUROP-PHASE1))+AVTENN
      IF(IOUROP.GE.TIMMAX) TDB(IOUROP)=-AMP *SIN((PIE/(2.0*CYC1))*
     1 (ROUROP-PHASE2))+AVTENX
      IF(TDB(IOUROP).LE.1.0) TDB(IOUROP)=1.0
      JCTC=0
      TGEUSS=TPOND
 9741 IGEUSS=TGEUSS
      RIGEUS=IGEUSS
      IGEUS1=IGEUSS+1
      COLT=CONDEN(IGEUSS)+(TGEUSS-RIGEUS)*(CONDEN(IGEUS1)-CONDEN(IGEUS
     1S))
      TLET=COLT-(CUNITC*(COLT-TDB(IOUROP))/WLC)
      IF(IOUROP.EQ.TBHUOP) TLET1=TLET
      IF(IOUROP.LE.ITIMHF.AND.ITHRMB.EQ.1) TGEUSN=TPOND*(1.0-BYPASS)+
     1 TLET*BYPASS
      IF(IOUROP.GT.ITIMHF.AND.ITHRMB.EQ.1) TGEUSN=(TPOND+0.33*
     1 (TLET1-TPOND))*(1.0-BYPASS)+TLET*BYPASS
```

```
      IF(ITHRMB.EQ.2) TGEUSN=TPOND*(1.0-BYPASS)+TLET*BYPASS
      DFXX=ABS(TGEUSN-TGEUSS)
      IF(DFXX.LE.0.3) GO TO 9729
      JCTC=JCTC+1
      IF(JCTC.GT.40) GO TO 9729
      TGEUSS=(TGEUSS+TGEUSN)/2.0
      GO TO 9741
 9729 XLOSS=CAPLCM(IGEUSS)+(TGEUSS-RIGEUS)*(CAPLCM(IGEUS1)-CAPLCM(IGEU
     1SS))
      IF(XLOSS.GE.CAPLMX) CAPLMX=XLOSS
      IF(ITHRMB.EQ.2) TPOND=(TPOND*(MASSP-WLC*(1.0-BYPASS))+TLET*
     1 (WLC*(1.0-BYPASS)))/MASSP
 9811 CONTINUE
      GO TO 7000
10000 CONTINUE
C***************************************************************************
C   SIMULATION  ROUTINE FOR TOWERS-ONLY SYSTEM
C***************************************************************************
      GO TO 3000
10100 CONTINUE
      XLOSS=LOSCPU(IPTDB)+FPTDB*(LOSCPU(IPTDB1)-LOSCPU(IPTDB))
      IF(WINTIL(JDAY).EQ.1.0.AND.IOUROP.EQ.TIMMAX) CAPLMX=CAPLMX+
     1 XLOSS/10.0
      OPPEN=XLOSS*LAMBDA(IOUROP)*1000.0
      YCAPEN=YCAPEN+OPPEN
      YTOWOP=YTOWOP+(FANPOW+PPUMPU)*LAMBDA(IOUROP)
      TLOSGN=TLOSGN+XLOSS
      IF(IOUROP.LT.24) GO TO 3050
      IF(JDAY.EQ.365) GO TO 7000
      GO TO 3000
      END
```

```
      SUBROUTINE PONTEM(PONTYP,TPOND,EVAP,TDBA,TIME)
C   THIS SUBROUTINE DETERMINES HEAT TRANSFER FROM THE POND SURAFACE
C   ADJUST METEOROLOGICAL PARAMETERS IF REQUIRED
      COMMON/HTPOND/HTOUT
      COMMON/POND/MASSP,AREAP
      INTEGER PONTYP
      REAL MASSP
      PTEMP=TPOND
      AREA=AREAP
      RELHUM=0.40
      AIRTMP=TDBA
      SUNSSS=0.0
      PHISCC=3000.0
      W2=6.0
      SPCHET=1.0
      PTSAVE=PTEMP
      NCHOCK=0
      ICOUNT=0
15    CONTINUE
      NCHOCK=NCHOCK+1
      IF(NCHOCK.GE.20) GO TO 50
      ES=PSA(PTEMP)
      EA=PSA(AIRTMP)*RELHUM
      TSV=(PTEMP+460.0)/(1.0-(0.378*ES/14.7))
      TAV=(AIRTMP+460.0)/(1.0-(0.378*EA/14.7))
      DELTV=TSV-TAV
      IF(DELTV.LE.0.0) DELTV=0.0001
      FW2=22.4*(DELTV**.33)+14.0*W2
      ES=ES*51.8
      EA=EA*51.8
      IF(PONTYP.EQ.2) ES=0.0
      IF(PONTYP.EQ.2) EA=0.0
      CE=SUNSSS/100.0
      PHISN=0.94*PHISCC*(1.0-0.65*(CE**2.0))
      IF(PONTYP.EQ.2) PHISN=0.0
      PHIAN=(1.16E-13)*((460.0+AIRTMP)**6)*(1.0+0.17*(CE**2))
      PHIR=PHISN+PHIAN
      PHIN=PHIR-((4.0E-8)*((PTEMP+460.0)**4)+FW2*((ES-EA)+0.25*(PTEMP-
     1 AIRTMP)))
      HTOUT=PHIN*AREA*TIME/24.0
      DELTP=HTOUT/(MASSP*SPCHET)
      PTEMP1=PTSAVE+DELTP
      ICOUNT=ICOUNT+1
      IF(ICOUNT.GT.1) GO TO  30
      PTMPAV=(PTEMP1+PTEMP)/2.0
      PAVES=PTMPAV
      IF(ICOUNT.EQ.1) PTEMP=PTMPAV
      IF(ICOUNT.EQ.1) GO TO 15
30    CONTINUE
      PTMPAV=(PTEMP1+PTSAVE)/2.0
      DIF=ABS(PTMPAV-PAVES)
      IF(DIF.LE.0.2) GO TO 50
      PTEMP=(PTMPAV+PAVES)/2.0
      PAVES=PTEMP
      GO TO 15
```

```
 50     CONTINUE
        PTEMP=PTSAVE
        TPOND=PTEMP1
        EVAP=HTOUT/1060.0
        IF(PONTYP.EQ.2) EVAP=0.0
        RETURN
        END
```

```
        FUNCTION PSA(T)
C   THIS FUNCTION SUPPLIES VAPOR PRESSURE OF WATER
        DIMENSION V(181)
        DATA M/0/
       DATAV/.08854,.09223,.09603,.09995,.10401,.10821,.11256,.11705,.121
      170,.12652,.13150,.13665,.14199,.14752,.15323,.15914,.16525,.17157,
      2.17811,.18486,.19182,.19900,.20642,.2141,.2220,.2302,.2386,.2473,.
      32563,.2655,.2751,.2850,.2951,.3056,.3164,.3276,.3390,.3509,.3631,.
      43756,.3886,.4019,.4156,.4298,.4443,.4593,.4747,.4906,.5069,.5237,.
      55410,.5588,.5771,.5959,.6152,.6351,.6556,.6766,.6982,.7204,.7432,.
      67666,.7906,.8153,.8407,.8668,.8935,.9210,.9492,.9781,1.0078,1.0382
      7,1.0695,1.1016,1.1345,1.1683,1.2029,1.2384,1.2748,1.3121,1.3504,1.
      83896,1.4298,1.4709,1.5130,1.5563,1.6006,1.6459,1.6924,1.7400,1.788
      98,1.8387,1.8897,1.9420,1.9955,2.0503,2.1064,2.1638,2.2225,2.2826,2
      *.3440,2.4069,2.4712,2.5370,2.6042,2.6729,2.7432,2.8151,2.8886,2.96
      *37,3.0404,3.1188,3.1990,3.281,3.365,3.450,3.537,3.627,3.718,3.811,
      *3.906,4.003,4.102,4.203,4.306,4.411,4.519,4.629,4.741,4.855,4.971,
      *5.090,5.212,5.335,5.461,5.590,5.721,5.855,5.992,6.131,6.273,6.471,
      *6.565,6.715,6.868,7.024,7.183,7.345,7.510,7.678,7.850,8.024,8.202,
      *8.383,8.567,8.755,8.946,9.141,9.339,9.541,9.746,9.955,10.169,10.38
      *5,10.605,10.830,11.058,11.290,11.526,11.769,12.011,12.262,12.512,1
      *.771,13.031,13.300,13.568,13.845,14.123,14.410,14.696/
        NT=T
        PSA=0.0
        IF(NT.GT.31) GO TO 5
        PSA=V(1)
        RETURN
 5      CONTINUE
        IF(NT.GE.212) PSA=V(212)
        IF(NT.GE.212) RETURN
        PSA=V(NT-31)+(V(NT-30)-V(NT-31))*(T-NT)
        RETURN
        END
```

```
      FUNCTION CX(X,N,K)
C   IMPLICIT CONSTAINTS ON VARIABLES (USUALLY NONE)
      DIMENSION X(N)
C
      CX=X(K)
C
      RETURN
      END
C


      FUNCTION CG(X,N,K)
C   LOWER LIMIT CONSTRAINTS
      DIMENSION X(N)
C
      CG=0.0
C
      RETURN
      END


      FUNCTION CH(X,N,K)
C   UPPER LIMIT CONSTRAINTS
      DIMENSION X(N)
      LOGICAL TOWONL
      COMMON/OPMODE/TOWONL
C
      IF(.NOT.TOWONL) GO TO 10
      GO TO (1,2),K
10    CONTINUE
      GO TO (1,1,2),K
C
1     CH=1000000000.0
      GO TO 3
2     CH=3000.0
C
3     RETURN
      END

$ENTRY
```

INPUT DATA SUMMARY

TEMPERATURE FREQUENCY DISTRIBUTION

PEAK TEMP= 37.5	MINIMUM TEMP= 22.5	FREQUENCY= 22.0
PEAK TEMP= 45.0	MINIMUM TEMP= 15.0	FREQUENCY= 57.0
PEAK TEMP= 52.5	MINIMUM TEMP= 7.5	FREQUENCY= 13.0
PEAK TEMP= 57.5	MINIMUM TEMP= 42.5	FREQUENCY= 21.0
PEAK TEMP= 65.0	MINIMUM TEMP= 35.0	FREQUENCY= 51.0
PEAK TEMP= 72.5	MINIMUM TEMP= 27.5	FREQUENCY= 48.0
PEAK TEMP= 77.5	MINIMUM TEMP= 62.5	FREQUENCY= 8.0
PEAK TEMP= 85.0	MINIMUM TEMP= 55.0	FREQUENCY= 87.0
PEAK TEMP= 92.5	MINIMUM TEMP= 47.5	FREQUENCY= 42.0
PEAK TEMP= 97.5	MINIMUM TEMP= 82.5	FREQUENCY= 0.0
PEAK TEMP=105.0	MINIMUM TEMP= 75.0	FREQUENCY= 13.0
PEAK TEMP=112.5	MINIMUM TEMP= 67.5	FREQUENCY= 3.0

AT	1	HOURS THE INCREMENTAL FUEL COST IS	5.0725 MILLS/KWHR
AT	2	HOURS THE INCREMENTAL FUEL COST IS	4.1219 MILLS/KWHR
AT	3	HOURS THE INCREMENTAL FUEL COST IS	4.1211 MILLS/KWHR
AT	4	HOURS THE INCREMENTAL FUEL COST IS	5.0702 MILLS/KWHR
AT	5	HOURS THE INCREMENTAL FUEL COST IS	6.8549 MILLS/KWHR
AT	6	HOURS THE INCREMENTAL FUEL COST IS	9.2599 MILLS/KWHR
AT	7	HOURS THE INCREMENTAL FUEL COST IS	11.9953 MILLS/KWHR
AT	8	HOURS THE INCREMENTAL FUEL COST IS	13.6630 MILLS/KWHR
AT	9	HOURS THE INCREMENTAL FUEL COST IS	15.2533 MILLS/KWHR
AT	10	HOURS THE INCREMENTAL FUEL COST IS	16.7015 MILLS/KWHR
AT	11	HOURS THE INCREMENTAL FUEL COST IS	17.9443 MILLS/KWHR
AT	12	HOURS THE INCREMENTAL FUEL COST IS	18.9274 MILLS/KWHR
AT	13	HOURS THE INCREMENTAL FUEL COST IS	19.6078 MILLS/KWHR
AT	14	HOURS THE INCREMENTAL FUEL COST IS	19.9559 MILLS/KWHR
AT	15	HOURS THE INCREMENTAL FUEL COST IS	19.9564 MILLS/KWHR
AT	16	HOURS THE INCREMENTAL FUEL COST IS	19.6093 MILLS/KWHR
AT	17	HOURS THE INCREMENTAL FUEL COST IS	18.9298 MILLS/KWHR
AT	18	HOURS THE INCREMENTAL FUEL COST IS	17.9475 MILLS/KWHR
AT	19	HOURS THE INCREMENTAL FUEL COST IS	16.7053 MILLS/KWHR
AT	20	HOURS THE INCREMENTAL FUEL COST IS	15.2577 MILLS/KWHR
AT	21	HOURS THE INCREMENTAL FUEL COST IS	13.6676 MILLS/KWHR
AT	22	HOURS THE INCREMENTAL FUEL COST IS	12.0000 MILLS/KWHR
AT	23	HOURS THE INCREMENTAL FUEL COST IS	9.2643 MILLS/KWHR
AT	24	HOURS THE INCREMENTAL FUEL COST IS	6.8585 MILLS/KWHR

THIS RUN IS FOR A PLUG FLOW TYPE POND

THE FRACTION OF THE FLOW BYPASSING THE POND IS 0.0000

THIS RUN IS FOR AN ADIABATIC POND

THE MAXIMUM CONDENSING TEMP IS 133.0 AND THE MINIMUM THERMAL POWER IS 50.0 PER CENT
THE THERMAL POWER IS 3000.00 MEGAWATTS AND THE MAXIMUM GENERATION IS 1050.17 MEGAWATTS

THE MAXIMUM TEMPERATURES OF THE TEN HOTEST DAYS ARE AS FOLLOWS

103.0	102.0	102.0	102.0	102.0	102.0	101.0	101.0	101.0	101.0

THE MAXIMUM TEMPERATURES OF THE DAYS PRECEDING THE TEN HOTEST ARE AS FOLLOWS

```
   100.0      102.0      101.0      100.0      100.0      103.0      101.0      101.0      101.0      102.0

AVERAGE TEMPERATURES OF TEN HOTEST --MINIMUM= 64.7  MAXIMUM=101.7
AVERAGE TEMPERATURES OF PRECEDING TEN DAYS--MINIMUM= 63.5 MAXIMUM=101.1

                     OPERATION AND DESIGN VARIABLES
  TIME      TIME       TIME       TIME     # TOWER      FLOW       FLOW       POND
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12        18          1          7     160.0     1100.0     1100.0      27.3
INCREMENTAL COST =     1.921 MILLS/KWHR

                     OPERATION AND DESIGN VARIABLES
  TIME      TIME       TIME       TIME     # TOWER      FLOW       FLOW       POND
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12        18          1          7     160.0     1166.8     1100.0      28.9
INCREMENTAL COST =     1.936 MILLS/KWHR

                     OPERATION AND DESIGN VARIABLES
  TIME      TIME       TIME       TIME     # TOWER      FLOW       FLOW       POND
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12        18          1          7     167.0     1100.0     1100.0      27.3
INCREMENTAL COST =     1.900 MILLS/KWHR

                     OPERATION AND DESIGN VARIABLES
  TIME      TIME       TIME       TIME     # TOWER      FLOW       FLOW       POND
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12        18          1          7     167.0     1100.0     1300.3      27.3
INCREMENTAL COST =     1.904 MILLS/KWHR

                     OPERATION AND DESIGN VARIABLES
  TIME      TIME       TIME       TIME     # TOWER      FLOW       FLOW       POND
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12        18          1          7     167.0     1066.6     1100.0      26.4
INCREMENTAL COST =     1.909 MILLS/KWHR

                     OPERATION AND DESIGN VARIABLES
  TIME      TIME       TIME       TIME     # TOWER      FLOW       FLOW       POND
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12        18          1          7     188.0     1100.0     1100.0      27.3
INCREMENTAL COST =     1.877 MILLS/KWHR

                     OPERATION AND DESIGN VARIABLES
  TIME      TIME       TIME       TIME     # TOWER      FLOW       FLOW       POND
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12        18          1          7     188.0     1100.0      999.8      27.3
INCREMENTAL COST =     1.920 MILLS/KWHR

                     OPERATION AND DESIGN VARIABLES
  TIME      TIME       TIME       TIME     # TOWER      FLOW       FLOW       POND
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12        18          1          7     188.0     1116.7     1100.0      27.7
INCREMENTAL COST =     1.886 MILLS/KWHR

                     OPERATION AND DESIGN VARIABLES
  TIME      TIME       TIME       TIME     # TOWER      FLOW       FLOW       POND
```

```
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12         18          1          7   251.0    1100.0    1100.0      27.3
INCREMENTAL COST =     2.127 MILLS/KWHR

                          OPERATION AND DESIGN VARIABLES
  TIME       TIME       TIME       TIME    # TOWER      FLOW       FLOW      POND
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12         18          1          7   188.0    1100.0    1150.1      27.3
INCREMENTAL COST =     1.883 MILLS/KWHR

                          OPERATION AND DESIGN VARIABLES
  TIME       TIME       TIME       TIME    # TOWER      FLOW       FLOW      POND
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12         18          1          7   188.0    1091.7    1100.0      27.1
INCREMENTAL COST =     1.875 MILLS/KWHR

                          OPERATION AND DESIGN VARIABLES
  TIME       TIME       TIME       TIME    # TOWER      FLOW       FLOW      POND
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12         18          1          7   156.5    1091.7    1100.0      27.1
INCREMENTAL COST =     2.295 MILLS/KWHR

                          OPERATION AND DESIGN VARIABLES
  TIME       TIME       TIME       TIME    # TOWER      FLOW       FLOW      POND
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12         18          1          7   188.0    1091.7    1075.0      27.1
INCREMENTAL COST =     1.882 MILLS/KWHR

                          OPERATION AND DESIGN VARIABLES
  TIME       TIME       TIME       TIME    # TOWER      FLOW       FLOW      POND
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12         18          1          7   188.0    1066.6    1100.0      26.4
INCREMENTAL COST =     1.886 MILLS/KWHR


 STAGE        FUNCTION            PROGRESS          LATERAL PROGRESS
   1     -0.18752450E 01     0.15116730E 08     0.28000000E 02

 NUMBER OF FUNCTION EVALUATIONS =       14

 VALUES OF X AT THIS STAGE

 X( 1) =    2.471039E 08    X( 2) =    2.452290E 08    X( 3) =    1.880000E 02

                          OPERATION AND DESIGN VARIABLES
  TIME       TIME       TIME       TIME    # TOWER      FLOW       FLOW      POND
HU BEGINS  HU ENDS CLD BEGINS CLD ENDS    CELLS    COUP MODE UNCO MODE   AREA
  12         18          1          7   193.2    1090.1    1112.4      27.0
INCREMENTAL COST =     1.887 MILLS/KWHR

 STAGE        FUNCTION            PROGRESS          LATERAL PROGRESS
   1     -0.18752450E 01     0.15116730E 08     0.28000000E 02

 NUMBER OF FUNCTION EVALUATIONS =       15
```

DAY	MAXIMUM AMBIENT TEMP	MINIMUM AMBIENT TEMP	AVERAGE POWER - HEATUP	AVERAGE POWER - COOLDOWN	COLD POND TEMP	HOT POND TEMP	EVAP LOSS (LBS/DAY)	HEAT TR FROM POND(%)
1	45.0	23.0	1050.2	0.0	50.0	50.0	0.0	0.0
2	24.0	9.0	1050.2	1050.2	41.2	57.4	0.0	0.0
3	17.0	11.0	1050.2	1050.2	35.8	42.1	0.0	0.0
4	29.0	13.0	1050.2	1050.2	34.6	36.2	0.0	0.0
5	37.0	26.0	1050.2	1050.2	44.9	42.8	0.0	0.0
6	39.0	30.0	1050.2	1050.2	50.9	51.7	0.0	0.0
7	42.0	32.0	1050.2	1050.2	53.6	55.3	0.0	0.0
8	50.0	31.0	1050.2	1050.2	54.3	58.1	0.0	0.0
9	43.0	31.0	1050.2	1048.8	56.2	62.7	0.0	0.0
10	44.0	25.0	1050.2	1049.9	51.4	59.4	0.0	0.0
11	44.0	20.0	1050.2	1050.2	47.9	57.8	0.0	0.0
12	44.0	20.0	1050.2	1050.2	47.3	56.3	0.0	0.0
13	52.0	28.0	1050.2	1050.2	52.0	56.3	0.0	0.0
14	51.0	21.0	1050.2	1049.1	50.6	62.5	0.0	0.0
15	33.0	22.0	1050.2	1049.4	50.4	61.3	0.0	0.0
16	31.0	27.0	1050.2	1050.2	49.0	51.5	0.0	0.0
17	36.0	29.0	1050.2	1050.2	49.5	50.0	0.0	0.0
18	44.0	28.0	1050.2	1050.2	50.4	53.0	0.0	0.0
19	52.0	23.0	1050.2	1050.2	49.6	57.5	0.0	0.0
20	58.0	25.0	1050.2	1049.3	52.6	61.6	0.0	0.0
21	54.0	31.0	1049.9	1047.7	58.0	66.4	0.0	0.0
22	40.0	18.0	1050.2	1048.1	50.2	66.0	0.0	0.0
23	39.0	17.0	1050.2	1050.2	44.9	54.9	0.0	0.0
24	42.0	14.0	1050.2	1050.2	42.1	52.1	0.0	0.0
25	53.0	11.0	1050.2	1050.2	40.8	52.5	0.0	0.0
26	51.0	18.0	1050.2	1050.1	47.2	58.2	0.0	0.0
27	37.0	18.0	1050.2	1049.8	47.6	59.8	0.0	0.0
28	42.0	16.0	1050.2	1050.2	43.2	52.2	0.0	0.0
29	47.0	13.0	1050.2	1050.2	42.1	53.1	0.0	0.0
30	55.0	20.0	1050.2	1050.2	47.2	55.5	0.0	0.0
31	58.0	16.0	1050.2	1049.3	47.7	62.0	0.0	0.0
32	61.0	25.0	1050.2	1048.5	53.8	64.1	0.0	0.0
33	47.0	20.0	1050.2	1047.3	52.4	68.1	0.0	0.0
34	49.0	13.0	1050.2	1049.8	44.8	59.5	0.0	0.0
35	54.0	12.0	1050.2	1050.2	43.5	57.4	0.0	0.0
36	45.0	15.0	1050.2	1049.8	46.1	59.8	0.0	0.0
37	39.0	11.0	1050.2	1050.2	41.9	55.8	0.0	0.0
38	40.0	9.0	1050.2	1050.2	38.7	50.7	0.0	0.0
39	38.0	8.0	1050.2	1050.2	37.8	49.8	0.0	0.0
40	45.0	7.0	1050.2	1050.2	36.7	48.3	0.0	0.0
41	51.0	10.0	1050.2	1050.2	40.0	51.8	0.0	0.0
42	59.0	12.0	1050.2	1050.2	43.3	56.6	0.0	0.0
43	62.0	31.0	1049.9	1048.7	56.8	63.0	0.0	0.0
44	53.0	27.0	1050.0	1045.7	57.3	70.2	0.0	0.0
45	51.0	20.0	1050.2	1048.3	51.1	65.2	0.0	0.0
46	55.0	16.0	1050.2	1049.5	47.4	61.3	0.0	0.0
47	62.0	13.0	1050.2	1049.3	46.1	61.9	0.0	0.0
48	56.0	25.0	1050.2	1048.1	54.4	65.5	0.0	0.0
49	48.0	19.0	1050.2	1048.2	50.8	65.6	0.0	0.0
50	63.0	24.0	1050.2	1049.8	51.3	59.8	0.0	0.0
51	42.0	24.0	1050.2	1047.1	54.8	68.4	0.0	0.0

52	45.0	17.0	1050.2	1050.2	46.2	57.9	0.0	0.0
53	62.0	13.0	1050.2	1050.2	43.5	55.9	0.0	0.0
54	44.0	15.0	1050.2	1048.6	48.0	64.2	0.0	0.0
55	45.0	5.0	1050.2	1050.2	38.6	55.8	0.0	0.0
56	51.0	7.0	1050.2	1050.2	38.5	52.5	0.0	0.0
57	64.0	9.0	1050.2	1050.2	41.4	55.9	0.0	0.0
58	68.0	25.0	1050.0	1048.3	54.3	64.7	0.0	0.0
59	72.0	23.0	1049.7	1044.3	56.3	72.3	0.0	0.0
60	70.0	35.0	1047.6	1040.8	64.5	75.7	0.0	0.0
61	64.0	40.0	1046.0	1038.2	68.3	78.2	0.0	0.0
62	60.0	29.0	1048.9	1040.7	61.0	76.1	0.0	0.0
63	50.0	19.0	1050.2	1045.7	52.8	70.5	0.0	0.0
64	60.0	14.0	1050.2	1049.4	46.3	61.4	0.0	0.0
65	69.0	23.0	1050.1	1048.4	53.0	64.5	0.0	0.0
66	59.0	29.0	1049.3	1043.9	59.6	72.5	0.0	0.0
67	54.0	29.0	1049.8	1046.0	58.2	69.7	0.0	0.0
68	47.0	30.0	1050.1	1047.7	57.3	56.4	0.0	0.0
69	51.0	22.0	1050.2	1049.2	50.9	61.9	0.0	0.0
70	57.0	22.0	1050.2	1049.4	50.8	61.5	0.0	0.0
71	66.0	26.0	1050.0	1048.3	54.7	64.9	0.0	0.0
72	68.0	24.0	1049.8	1044.9	56.4	71.4	0.0	0.0
73	72.0	24.0	1049.7	1043.5	57.2	73.2	0.0	0.0
74	73.0	33.0	1047.9	1040.6	63.6	76.0	0.0	0.0
75	81.0	37.0	1045.6	1036.5	67.3	79.3	0.0	0.0
76	79.0	34.0	1045.3	1026.2	68.2	85.3	0.0	0.0
77	73.0	39.0	1043.5	1026.9	70.6	84.6	0.0	0.0
78	63.0	46.0	1041.5	1030.0	73.5	82.6	0.0	0.0
79	64.0	40.0	1046.1	1038.3	68.1	78.2	0.0	0.0
80	64.0	30.0	1048.6	1040.7	61.6	76.1	0.0	0.0
81	65.0	37.0	1047.7	1042.6	64.6	73.5	0.0	0.0
82	65.0	40.0	1046.6	1040.7	67.1	75.5	0.0	0.0
83	67.0	37.0	1047.1	1040.0	65.8	76.5	0.0	0.0
84	72.0	29.0	1048.5	1040.2	61.5	76.7	0.0	0.0
85	68.0	32.0	1048.0	1039.1	63.6	77.7	0.0	0.0
86	69.0	31.0	1048.3	1040.4	62.5	76.4	0.0	0.0
87	69.0	29.0	1048.6	1040.5	61.4	76.4	0.0	0.0
88	78.0	34.0	1047.3	1040.5	64.2	76.1	0.0	0.0
89	77.0	40.0	1043.5	1030.8	70.4	82.5	0.0	0.0
90	66.0	29.0	1047.7	1029.0	64.6	84.1	0.0	0.0
91	65.0	28.0	1049.0	1041.3	60.3	75.5	0.0	0.0
92	59.0	34.0	1048.4	1042.8	62.8	73.4	0.0	0.0
93	55.0	23.0	1050.1	1045.3	55.3	70.9	0.0	0.0
94	56.0	20.0	1050.2	1048.2	51.3	65.4	0.0	0.0
95	71.0	17.0	1050.2	1048.6	49.4	64.2	0.0	0.0
96	77.0	33.0	1048.3	1043.9	62.1	72.2	0.0	0.0
97	68.0	31.0	1047.7	1034.4	64.4	80.8	0.0	0.0
98	78.0	25.0	1049.1	1040.6	59.2	76.5	0.0	0.0
99	73.0	43.0	1043.0	1034.4	71.2	80.4	0.0	0.0
100	53.0	33.0	1047.5	1031.7	65.9	82.5	0.0	0.0
101	63.0	33.0	1049.1	1046.4	60.3	69.1	0.0	0.0
102	71.0	35.0	1048.1	1043.6	63.1	72.4	0.0	0.0
103	55.0	22.0	1049.9	1039.8	57.7	77.6	0.0	0.0
104	61.0	23.0	1050.2	1047.8	53.5	66.4	0.0	0.0
105	69.0	21.0	1050.1	1047.4	53.3	68.0	0.0	0.0
106	81.0	21.0	1049.7	1044.3	55.3	72.3	0.0	0.0

107	77.0	34.0	1046.5	1035.3	66.1	80.1	0.0	0.0
108	77.0	34.0	1046.0	1031.5	67.0	82.5	0.0	0.0
109	64.0	38.0	1045.2	1030.1	69.2	83.0	0.0	0.0
110	67.0	25.0	1049.4	1040.8	58.9	76.3	0.0	0.0
111	72.0	32.0	1048.4	1042.5	62.0	73.9	0.0	0.0
112	79.0	33.0	1047.1	1038.8	64.5	78.0	0.0	0.0
113	79.0	44.0	1041.1	1028.9	73.0	83.3	0.0	0.0
114	77.0	42.0	1040.9	1022.4	73.3	86.8	0.0	0.0
115	75.0	47.0	1038.7	1023.5	75.7	85.9	0.0	0.0
116	77.0	36.0	1044.5	1025.5	69.3	85.6	0.0	0.0
117	78.0	28.0	1047.4	1030.1	64.0	83.6	0.0	0.0
118	75.0	36.0	1045.7	1031.9	67.9	82.2	0.0	0.0
119	73.0	34.0	1046.4	1032.1	66.7	82.1	0.0	0.0
120	75.0	36.0	1046.1	1034.5	67.2	80.6	0.0	0.0
121	82.0	34.0	1045.9	1032.6	66.7	81.8	0.0	0.0
122	82.0	42.0	1041.0	1024.3	73.0	85.8	0.0	0.0
123	81.0	40.0	1041.1	1019.4	72.9	88.4	0.0	0.0
124	80.0	42.0	1040.4	1020.5	73.8	87.8	0.0	0.0
125	83.0	36.0	1043.3	1021.9	70.3	87.4	0.0	0.0
126	80.0	40.0	1041.4	1020.8	72.7	87.7	0.0	0.0
127	81.0	36.0	1043.6	1022.8	70.0	86.9	0.0	0.0
128	87.0	44.0	1038.7	1022.4	74.6	86.6	0.0	0.0
129	89.0	56.0	1023.2	1007.4	83.9	92.3	0.0	0.0
130	86.0	49.0	1027.3	995.6	81.9	97.2	0.0	0.0
131	89.0	45.0	1033.5	1003.4	78.5	94.5	0.0	0.0
132	87.0	50.0	1027.9	1001.6	81.5	94.8	0.0	0.0
133	83.0	55.0	1022.8	999.9	84.4	95.2	0.0	0.0
134	85.0	53.0	1025.6	1002.9	82.8	94.2	0.0	0.0
135	85.0	51.0	1027.3	1002.3	81.9	94.6	0.0	0.0
136	84.0	53.0	1025.2	1001.4	83.1	94.8	0.0	0.0
137	79.0	47.0	1033.2	1004.4	79.2	94.0	0.0	0.0
138	83.0	41.0	1040.1	1016.9	73.9	89.4	0.0	0.0
139	69.0	37.0	1043.0	1017.9	71.4	89.1	0.0	0.0
140	68.0	31.0	1047.9	1035.5	64.0	80.2	0.0	0.0
141	76.0	33.0	1047.6	1040.1	63.8	76.6	0.0	0.0
142	85.0	44.0	1041.4	1032.6	72.2	81.3	0.0	0.0
143	87.0	55.0	1026.2	1013.0	82.4	90.2	0.0	0.0
144	84.0	44.0	1034.7	1001.5	78.2	95.3	0.0	0.0
145	89.0	44.0	1035.9	1010.7	76.8	91.7	0.0	0.0
146	93.0	45.0	1033.2	1004.7	78.4	94.0	0.0	0.0
147	98.0	50.0	1024.4	995.9	82.7	97.0	0.0	0.0
148	92.0	51.0	1021.9	912.5	84.3	101.6	0.0	0.0
149	88.0	47.0	1027.8	988.7	81.6	98.9	0.0	0.0
150	86.0	49.0	1028.6	1000.0	81.2	95.5	0.0	0.0
151	85.0	49.0	1029.7	1003.2	80.7	94.3	0.0	0.0
152	87.0	49.0	1029.7	1003.8	80.6	94.1	0.0	0.0
153	91.0	47.0	1030.9	1002.9	79.7	94.5	0.0	0.0
154	89.0	46.0	1030.8	998.4	79.9	96.4	0.0	0.0
155	91.0	47.0	1030.2	1000.8	80.1	95.3	0.0	0.0
156	90.0	57.0	1017.7	995.3	86.4	96.9	0.0	0.0
157	91.0	64.0	1005.6	964.7	91.2	99.3	0.0	0.0
158	87.0	64.0	1005.0	907.9	91.6	101.9	0.0	0.0
159	76.0	47.0	1029.9	960.4	81.2	99.4	0.0	0.0
160	82.0	36.0	1043.1	1020.1	70.5	88.3	0.0	0.0
161	93.0	42.0	1038.9	1021.4	73.8	87.3	0.0	0.0

162	95.0	50.0	1026.4	1000.9	81.8	95.1	0.0	0.0
163	98.0	50.0	1023.1	945.1	83.3	99.6	0.0	0.0
164	100.0	54.0	1016.3	910.8	86.3	102.0	0.0	0.0
165	103.0	56.0	1013.2	822.0	87.5	104.5	0.0	0.0
166	102.0	60.0	1007.5	724.3	89.8	106.8	0.0	0.0
167	100.0	60.0	1008.1	704.4	89.7	107.3	0.0	0.0
168	100.0	55.0	1015.8	730.0	86.5	106.0	0.0	0.0
169	98.0	55.0	1015.2	825.3	87.0	104.6	0.0	0.0
170	100.0	58.0	1011.7	829.5	88.4	103.7	0.0	0.0
171	102.0	59.0	1009.6	763.8	89.0	105.5	0.0	0.0
172	98.0	57.0	1012.9	724.8	88.0	106.8	0.0	0.0
173	100.0	65.0	1001.2	801.3	92.3	104.4	0.0	0.0
174	100.0	60.0	1008.2	702.7	89.7	107.3	0.0	0.0
175	102.0	65.0	1000.9	719.3	92.3	106.3	0.0	0.0
176	101.0	73.0	954.0	605.1	96.7	108.8	0.0	0.0
177	101.0	73.0	953.7	575.4	96.8	109.8	0.0	0.0
178	101.0	66.0	999.0	612.9	93.1	109.7	0.0	0.0
179	101.0	62.0	1006.2	623.6	90.3	108.2	0.0	0.0
180	102.0	68.0	996.2	672.6	93.9	107.2	0.0	0.0
181	102.0	65.0	1000.6	614.1	92.4	109.3	0.0	0.0
182	98.0	63.0	1005.2	616.3	90.9	108.5	0.0	0.0
183	94.0	58.0	1012.9	764.3	88.3	105.5	0.0	0.0
184	93.0	56.0	1015.5	907.6	87.2	102.0	0.0	0.0
185	97.0	56.0	1015.5	913.6	86.9	101.0	0.0	0.0
186	99.0	56.0	1014.9	841.9	87.0	103.1	0.0	0.0
187	92.0	67.0	999.4	800.4	93.5	104.6	0.0	0.0
188	83.0	62.0	1010.0	819.8	90.0	103.4	0.0	0.0
189	91.0	61.0	1012.6	994.7	88.5	96.9	0.0	0.0
190	93.0	59.0	1012.9	914.4	88.3	100.6	0.0	0.0
191	92.0	53.0	1019.9	912.9	85.3	101.4	0.0	0.0
192	95.0	55.0	1017.8	947.2	86.0	99.6	0.0	0.0
193	93.0	57.0	1014.5	911.5	87.7	101.6	0.0	0.0
194	96.0	69.0	996.7	883.7	94.3	101.6	0.0	0.0
195	88.0	61.0	1010.1	727.3	89.7	106.0	0.0	0.0
196	85.0	63.0	1008.5	944.4	90.3	99.6	0.0	0.0
197	84.0	62.0	1010.3	987.1	89.9	98.8	0.0	0.0
198	87.0	65.0	1006.5	992.4	91.0	97.5	0.0	0.0
199	92.0	64.0	1006.1	935.5	90.9	99.7	0.0	0.0
200	83.0	61.0	1010.6	887.7	89.8	102.3	0.0	0.0
201	89.0	60.0	1014.7	995.2	87.9	96.8	0.0	0.0
202	85.0	61.0	1011.4	963.7	89.3	99.3	0.0	0.0
203	92.0	61.0	1011.4	992.5	88.9	97.7	0.0	0.0
204	91.0	63.0	1006.2	910.4	91.0	101.5	0.0	0.0
205	97.0	60.0	1009.4	910.6	89.4	101.7	0.0	0.0
206	97.0	63.0	1004.8	801.7	91.1	104.4	0.0	0.0
207	97.0	60.0	1008.9	793.1	89.6	105.1	0.0	0.0
208	96.0	63.0	1004.9	801.5	91.2	104.5	0.0	0.0
209	98.0	63.0	1004.4	801.0	91.2	104.7	0.0	0.0
210	99.0	63.0	1004.6	741.8	91.1	105.8	0.0	0.0
211	97.0	67.0	999.2	704.5	93.2	106.4	0.0	0.0
212	95.0	67.0	999.7	710.1	93.2	106.3	0.0	0.0
213	94.0	62.0	1006.4	800.3	90.7	105.0	0.0	0.0
214	88.0	66.0	1003.2	804.3	92.2	103.4	0.0	0.0
215	86.0	64.0	1006.6	913.3	91.1	100.8	0.0	0.0
216	86.0	64.0	1006.6	974.7	91.1	99.2	0.0	0.0

217	89.0	63.0	1007.6	977.4	90.5	99.1	0.0	0.0
218	91.0	62.0	1009.0	914.4	89.9	100.5	0.0	0.0
219	88.0	58.0	1014.8	912.8	87.9	101.2	0.0	0.0
220	89.0	60.0	1012.2	987.7	88.8	98.7	0.0	0.0
221	89.0	58.0	1015.4	944.2	87.6	99.6	0.0	0.0
222	91.0	56.0	1016.9	986.6	86.7	99.0	0.0	0.0
223	92.0	56.0	1017.2	937.8	86.5	99.7	0.0	0.0
224	91.0	56.0	1017.5	916.3	86.4	100.2	0.0	0.0
225	91.0	57.0	1016.1	943.7	87.1	99.6	0.0	0.0
226	91.0	60.0	1012.5	916.7	88.6	100.0	0.0	0.0
227	95.0	59.0	1012.5	914.3	88.4	100.7	0.0	0.0
228	94.0	58.0	1013.4	861.2	88.1	102.7	0.0	0.0
229	96.0	61.0	1007.8	899.4	90.1	102.1	0.0	0.0
230	98.0	58.0	1012.0	827.4	88.4	104.0	0.0	0.0
231	91.0	59.0	1012.0	803.4	88.8	104.4	0.0	0.0
232	91.0	58.0	1014.7	914.4	87.8	100.7	0.0	0.0
233	91.0	52.0	1022.6	917.4	84.1	100.0	0.0	0.0
234	91.0	54.0	1019.7	990.0	85.4	98.5	0.0	0.0
235	91.0	54.0	1019.3	983.1	85.6	99.1	0.0	0.0
236	91.0	57.0	1015.8	969.9	87.2	99.2	0.0	0.0
237	93.0	54.0	1019.6	926.5	85.3	99.9	0.0	0.0
238	96.0	51.0	1022.7	917.6	83.6	100.0	0.0	0.0
239	95.0	48.0	1025.5	915.4	82.3	100.8	0.0	0.0
240	95.0	52.0	1021.4	929.3	84.3	99.8	0.0	0.0
241	94.0	50.0	1023.6	915.7	83.3	100.7	0.0	0.0
242	95.0	54.0	1018.9	931.2	85.4	99.8	0.0	0.0
243	96.0	59.0	1011.3	911.8	88.8	101.4	0.0	0.0
244	93.0	56.0	1016.0	840.8	87.0	103.4	0.0	0.0
245	92.0	60.0	1011.0	912.9	89.1	101.1	0.0	0.0
246	90.0	60.0	1011.0	911.7	89.2	101.4	0.0	0.0
247	92.0	59.0	1013.5	915.3	88.2	100.4	0.0	0.0
248	90.0	56.0	1017.0	914.0	86.7	100.9	0.0	0.0
249	88.0	55.0	1018.7	978.3	86.1	99.1	0.0	0.0
250	91.0	56.0	1018.1	993.1	86.2	97.8	0.0	0.0
251	94.0	59.0	1013.0	946.4	88.3	99.6	0.0	0.0
252	96.0	61.0	1007.9	894.3	90.1	102.2	0.0	0.0
253	96.0	60.0	1009.8	809.4	89.3	104.1	0.0	0.0
254	91.0	63.0	1006.6	803.6	90.8	103.8	0.0	0.0
255	88.0	55.0	1018.1	912.3	86.3	101.5	0.0	0.0
256	87.0	54.0	1021.2	993.3	85.0	97.8	0.0	0.0
257	78.0	53.0	1024.6	996.5	83.8	96.7	0.0	0.0
258	75.0	49.0	1034.1	1012.0	79.0	91.0	0.0	0.0
259	74.0	47.0	1038.2	1021.0	76.2	87.3	0.0	0.0
260	75.0	52.0	1035.1	1023.3	78.4	85.7	0.0	0.0
261	78.0	49.0	1036.4	1021.0	77.4	87.2	0.0	0.0
262	67.0	55.0	1030.7	1017.3	81.2	88.5	0.0	0.0
263	72.0	49.0	1038.9	1027.1	75.8	84.0	0.0	0.0
264	75.0	50.0	1037.5	1026.0	76.7	84.4	0.0	0.0
265	79.0	49.0	1036.6	1022.3	77.1	86.4	0.0	0.0
266	77.0	49.0	1035.6	1017.5	78.1	88.8	0.0	0.0
267	81.0	50.0	1034.6	1019.0	78.4	88.1	0.0	0.0
268	82.0	49.0	1033.5	1013.4	78.9	90.4	0.0	0.0
269	81.0	51.0	1030.8	1011.1	80.3	91.2	0.0	0.0
270	82.0	51.0	1030.6	1010.9	80.4	91.3	0.0	0.0
271	74.0	38.0	1041.4	1012.1	72.9	91.5	0.0	0.0

272	81.0	34.0	1045.6	1029.4	67.4	83.7	0.0	0.0
273	81.0	36.0	1044.1	1025.9	69.4	85.3	0.0	0.0
274	82.0	42.0	1040.8	1023.3	73.2	86.3	0.0	0.0
275	85.0	47.0	1036.0	1017.9	77.1	88.7	0.0	0.0
276	81.0	57.0	1023.2	1007.2	84.3	92.3	0.0	0.0
277	81.0	47.0	1033.9	1007.3	78.8	92.9	0.0	0.0
278	81.0	46.0	1036.3	1013.7	77.2	90.4	0.0	0.0
279	65.0	51.0	1034.1	1014.4	79.5	89.9	0.0	0.0
280	68.0	50.0	1039.5	1030.5	75.5	82.1	0.0	0.0
281	74.0	46.0	1041.0	1031.2	73.3	82.0	0.0	0.0
282	75.0	44.0	1041.0	1027.0	73.3	84.3	0.0	0.0
283	81.0	48.0	1037.5	1024.3	76.1	85.5	0.0	0.0
284	78.0	46.0	1037.2	1016.3	76.7	89.4	0.0	0.0
285	61.0	48.0	1038.6	1019.5	76.9	88.0	0.0	0.0
286	62.0	45.0	1043.6	1037.3	71.1	78.6	0.0	0.0
287	72.0	38.0	1046.6	1039.9	66.4	76.5	0.0	0.0
288	71.0	36.0	1046.4	1035.5	66.9	80.0	0.0	0.0
289	74.0	34.0	1046.9	1036.4	65.6	79.5	0.0	0.0
290	75.0	36.0	1046.0	1034.4	67.3	80.7	0.0	0.0
291	78.0	40.0	1043.6	1031.6	70.2	82.1	0.0	0.0
292	77.0	44.0	1040.6	1025.4	73.7	85.1	0.0	0.0
293	77.0	45.0	1039.7	1023.4	74.7	86.1	0.0	0.0
294	70.0	44.0	1040.5	1022.8	74.1	86.5	0.0	0.0
295	60.0	46.0	1041.8	1030.3	73.3	82.4	0.0	0.0
296	63.0	44.0	1044.8	1039.6	69.7	76.6	0.0	0.0
297	65.0	37.0	1047.2	1040.0	65.8	76.5	0.0	0.0
298	70.0	40.0	1046.1	1039.8	67.6	76.5	0.0	0.0
299	70.0	47.0	1041.5	1035.3	73.0	79.7	0.0	0.0
300	60.0	45.0	1042.4	1031.3	72.6	82.0	0.0	0.0
301	58.0	43.0	1045.7	1040.0	68.9	76.2	0.0	0.0
302	50.0	37.0	1048.1	1042.7	64.2	73.4	0.0	0.0
303	48.0	37.0	1049.0	1047.4	61.4	67.0	0.0	0.0
304	53.0	33.0	1049.9	1048.2	58.2	64.6	0.0	0.0
305	51.0	26.0	1050.2	1048.0	54.8	65.8	0.0	0.0
306	58.0	24.0	1050.2	1048.8	52.7	63.2	0.0	0.0
307	59.0	26.0	1050.1	1047.9	55.2	66.2	0.0	0.0
308	65.0	24.0	1050.0	1047.4	54.8	67.7	0.0	0.0
309	74.0	36.0	1047.8	1044.1	63.5	71.8	0.0	0.0
310	62.0	34.0	1047.5	1035.8	65.6	79.8	0.0	0.0
311	62.0	31.0	1048.8	1042.6	61.2	73.9	0.0	0.0
312	65.0	37.0	1047.9	1043.7	64.0	72.3	0.0	0.0
313	68.0	35.0	1047.7	1040.8	64.4	75.7	0.0	0.0
314	67.0	28.0	1048.8	1040.3	60.8	76.7	0.0	0.0
315	73.0	30.0	1048.6	1042.0	61.2	74.6	0.0	0.0
316	74.0	40.0	1045.2	1037.8	68.6	78.5	0.0	0.0
317	62.0	35.0	1046.8	1032.3	67.0	82.0	0.0	0.0
318	66.0	30.0	1048.8	1042.1	61.0	74.5	0.0	0.0
319	54.0	35.0	1048.2	1042.0	63.6	74.3	0.0	0.0
320	50.0	33.0	1049.4	1046.6	60.0	68.8	0.0	0.0
321	48.0	33.0	1049.9	1048.1	58.4	65.1	0.0	0.0
322	45.0	21.0	1050.2	1048.9	50.6	63.0	0.0	0.0
323	47.0	20.0	1050.2	1050.2	48.0	58.0	0.0	0.0
324	49.0	18.0	1050.2	1050.2	47.0	58.0	0.0	0.0
325	52.0	32.0	1050.2	1049.9	55.5	59.1	0.0	0.0
326	51.0	28.0	1050.2	1048.4	55.3	64.2	0.0	0.0

327	53.0	22.0	1050.2	1048.8	51.5	63.3	0.0	0.0
328	60.0	21.0	1050.2	1049.0	50.9	62.8	0.0	0.0
329	64.0	24.0	1050.1	1047.8	54.3	66.5	0.0	0.0
330	61.0	25.0	1050.0	1045.8	56.3	70.1	0.0	0.0
331	62.0	32.0	1049.2	1046.1	60.0	69.5	0.0	0.0
332	54.0	25.0	1050.0	1044.9	56.7	71.4	0.0	0.0
333	52.0	39.0	1048.6	1047.7	62.3	66.0	0.0	0.0
334	49.0	33.0	1049.6	1047.4	59.3	67.2	0.0	0.0
335	46.0	21.0	1050.2	1048.7	51.0	63.9	0.0	0.0
336	51.0	26.0	1050.2	1050.0	51.9	58.9	0.0	0.0
337	54.0	20.0	1050.2	1049.3	49.8	61.9	0.0	0.0
338	55.0	30.0	1050.1	1048.8	56.0	62.9	0.0	0.0
339	52.0	30.0	1050.0	1047.8	57.2	66.1	0.0	0.0
340	48.0	26.0	1050.2	1048.3	54.3	64.8	0.0	0.0
341	49.0	21.0	1050.2	1049.5	50.0	61.2	0.0	0.0
342	45.0	22.0	1050.2	1049.7	50.0	60.0	0.0	0.0
343	41.0	14.0	1050.2	1050.2	44.4	57.5	0.0	0.0
344	44.0	10.0	1050.2	1050.2	40.2	52.8	0.0	0.0
345	47.0	14.0	1050.2	1050.2	42.6	52.9	0.0	0.0
346	48.0	19.0	1050.2	1050.2	46.6	55.7	0.0	0.0
347	54.0	21.0	1050.2	1050.2	48.8	58.0	0.0	0.0
348	38.0	14.0	1050.2	1049.3	46.3	62.0	0.0	0.0
349	48.0	16.0	1050.2	1050.2	43.3	52.1	0.0	0.0
350	51.0	18.0	1050.2	1050.2	46.5	56.6	0.0	0.0
351	61.0	21.0	1050.2	1049.8	49.6	59.6	0.0	0.0
352	44.0	15.0	1050.2	1048.0	48.8	66.2	0.0	0.0
353	49.0	23.0	1050.2	1050.2	49.2	56.6	0.0	0.0
354	51.0	19.0	1050.2	1049.8	48.3	59.6	0.0	0.0
355	52.0	19.0	1050.2	1049.7	48.6	60.3	0.0	0.0
356	56.0	27.0	1050.2	1049.3	53.6	61.2	0.0	0.0
357	27.0	11.0	1050.2	1048.5	45.7	65.0	0.0	0.0
358	18.0	-4.0	1050.2	1050.2	30.6	45.1	0.0	0.0
359	21.0	-7.0	1050.2	1050.2	26.0	33.8	0.0	0.0
360	27.0	9.0	1050.2	1050.2	31.5	34.1	0.0	0.0
361	27.0	3.0	1050.2	1050.2	30.5	39.7	0.0	0.0
362	31.0	-3.0	1050.2	1050.2	28.2	39.0	0.0	0.0
363	35.0	10.0	1050.2	1050.2	35.0	40.6	0.0	0.0
364	34.0	5.0	1050.2	1050.2	34.4	46.1	0.0	0.0
365	32.0	9.0	1050.2	1050.2	36.1	44.8	0.0	0.0

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                              FINAL DESIGN SUMMARY


                      OPERATION AND DESIGN VARIABLES
  TIME       TIME       TIME       TIME     # TOWER      FLOW       FLOW       POND
HJ BEGINS  HU ENDS  CLD BEGINS CLD ENDS     CELLS   COUP MODE  UNCO MODE    AREA
  12         18          1          7      188.0    1091.7     1100.0      27.1

                                         SYSTEM COSTS
                          CAPITAL COSTS                                        OPERATING COSTS (ANNUAL)
     TOWER CELLS        STORAGE POND     REPLACEMENT CAPABILITY             FANS AND PUMPS     ENERGY REPL
       39560830.0         4641447.0              8346130.0                    2457541.0          2750809.0

 MAXIMUM CAPABILITY LOSS IS       55.64 MWS(E)
 TOTAL LOST ELECTRICAL GENERATION IS    310553.1 MWHRS
 THE EVAPORATIVE WATER LOSS IS               0.00 LBS    AND THE TOTAL HEAT TRANSFER FROM THE POND SURFACE IS    0.0 %
THE EXACT COST OF THE OPTIMIZED DESIGN IS    1.897 MILLS PER KWHR

MEMORY USAGE       OBJECT CODE=   48272 BYTES,ARRAY AREA=   26820 BYTES


COMPILE TIME=     1.85 SEC,EXECUTION TIME=    28.12 SEC,
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