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Wet/Dry Cooling for Cycling Steam-Electric Plants

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Prepared by
Dynatech R/D Company
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ELECTRIC POWER RESEARCH INSTITUTE

Wet/Dry Cooling for Cycling Steam-Electric Plants

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Research Project 1182-1

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ABSTRACT

The economics of water-conserving wet/dry cooling systems for steam-electric plants used for cycling or intermediate-load duty have been evaluated using a system simulation and design optimization computer model. Conclusions are based on the case-study evaluation of a separate wet and dry tower system which provides cooling for a nominal 500 MWe coal-fired plant for two site locations - Boston, Massachusetts, and Phoenix, Arizona. Historical utility load and meterological data which are coincidental have been used along with representative utility system generation models to describe the economic and physical environment for the operation of the wet/dry systems.

Optimum cooling system costs and designs assuming economically-dispatched cycling operation schedules are presented and compared to the costs and designs determined for base-load operation. All-wet and all-dry cooling systems have been evaluated in addition to wet/dry systems of different total annual water consumption. Consideration has been given to the impact of the design plant capacity factor on the incremental power production cost due to heat rejection, the selection of the steam turbine, the determined value of makeup water, and the economic consequences of operation at an off-design capacity factor.

EPRI PERSPECTIVE

PROJECT DESCRIPTION

Dynatech R/D Company has developed and demonstrated a method to include electrical supply and production economics in both design procedures and operating strategies for wet/dry cooling systems. A computer program that incorporates the essential features of this methodology for design and operation simulation has been formulated for applications to a wet/dry system. In an earlier effort (EPRI FP-1096), the computer program was applied to a wet/dry system in the case study evaluation of the design and operation for a typical large fossil-fired baseload plant. In the effort described in this report, the economics of wet/dry cooling were evaluated for the types of cycling steam-electric plants projected for use in the near future.

The computer code provides a methodology for determining in a quantitative manner the optimum economic trade-off between loss of performance and the sizing of the wet/dry cooling capacity. This methodology is necessary because the use of wet/dry cooling with a steam-electric plant can result in a significant reduction of electrical capacity and electrical energy production to levels below those obtained with the use of conventional cooling systems.

PROJECT OBJECTIVES

The objective of this project has been the development and demonstration of a computerized method to incorporate electrical capacity supply and electrical energy production economics of a utility system into both the design procedures and operational control strategies for wet/dry cooling systems.

The specific objective of this effort has been the determination of the impact of the power plant capacity factor on the design and economics of wet/dry cooling systems.

PROJECT RESULTS

This study provides several significant insights on design of wet/dry cooling systems for baseload and cycling plants.

An important conclusion is that the optimum design for all-wet and all-dry systems is essentially independent of the assumed plant capacity factor. Because of this, the incremental cost of power production due to cooling costs for the cycling plants is significantly greater, typically 60-70%.

It was also found that for small amounts of water consumption for cooling (250 acre-feet/year or less), there is essentially no difference in the optimum cooling system design for cycling and baseload operating schedules. This is because there is not sufficient water to impact the loss-of-energy production penalty, which strongly influences the size of the cooling system. Alternatively, when water is readily available, optimum cooling system design for cycling plants requires significantly less dry-cooling capacity than baseload plants. Nevertheless, the cost of wet/dry cooling per unit power production is at least 50% greater for cycling plants. Consequently, the value of cooling water is more than 50% greater for cycling plants.

Consideration should be given to designing the cooling system for a baseload capacity factor, regardless of the anticipated capacity factor. This approach may be justified in some cases because the additional total cost could be significantly less than the cost penalty associated with the underdesign of the cooling system.

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NOMENCLATURE

afcr	= Annual fixed charge rate.
B(n)	= Economic benefit of operating the wet/dry cooled plant during time increment n.
$B(n)_{ref}$	= Economic benefit of operating reference plant during time increment n.
B_T	= Total annual economic benefit of operating the wet/dry plant.
$B_{T_{ref}}$	= Total annual economic benefit of operating the reference plant.
C	= Present worth cost of heat rejection system over the plant lifetime.
C_c	= Present worth of revenue requirement for investment and associated costs.
C_I	= Capital cost of cooling system.
C_o	= Present-worth cost of operating heat rejection system over lifetime of plant.
$C_{o,m}$	= Annual operating and maintenance cost.
Cp_k	= Cost of power production at plant k.
Cp_o	= Total cost of power production at the wet/dry plant.
Cp_T	= Total cost of power production in utility system.
e_i	= Incremental power production cost resulting from the cost of heat rejection (mills/Kwhr).
F	= Availability factor.
G	= Total annual generation from reference ideally cooled plant.
I	= Operation state of wet cooling (0, 1, 2, ...S).
K_b	= Incremental cost of base-load type of capacity ($/kw).
k=1,2....K	= Generating unit in utility system other than wet/dry unit.

$O_k(t)$ = Outage process for plant k.

$O_o(t)$ = Outage process affecting wet/dry plant other than meterology and water-use scheduling.

$P_{g,o}$ = Base or reference power generation or capability for conventionally cooled power plant.

$P_g(t)$ = Actual power generation or capability for conventionally cooled plant.

$P_{max,k}$ = Rated output of plant k in utility system.

$P_L[t, m(t)]$ = Utility system electrical load demand at time of meteorological condition m(t).

$P_o(n)$ = Dispatched power level of the reference plant.

P_{min} = Capacity of plant at maximum ambient temperature condition with maximum wet cooling operation.

$P_{o,d}(n)$ = Actual generation of wet/dry plant during time period n, includes effects of metereology, water-use scheduling, and economic dispatch of plant.

P_o = Base or reference capacity of wet/dry cooled plant.

$P_{o,d}(t)$ = Actual generation capability of wet/dry plant at time t (includes effect of meteorology and water-use scheduling).

$P_o(n,I)$ = Plant power output during time period n at wet cooling state I.

P_s = Power requirement for operating wet/dry heat rejection system.

Pwf_i^m = The present-worth factor for a uniform series of annual payments at interest rate i for m years.

$r(n,I)$ = Water consumption during time period n at wet cooling state I.

$S_{fuel}(n)$ = Fuel saving due to operations at reduced thermal power during time period n.

$T_{db}(n)$ = Ambient dry bulb temperature during time period n.

$T_{wb}(n)$ = Ambient wet bulb temperature during time period n.

$u_k(t)$ = Dispatch parameter for plant k.

W_{max} = Maximum annual water consumption at wet/dry plant.

W_T = Maximum annual water consumption.

W_{ij} = Allowable water use for the period beginning at time i and ending at time j.

$\gamma(n,I)$ = Incremental water-use benefit function for time period n and wet cooling state I.

Λ_o = Present-worth of loss of energy production penalty.

Λ_c = Present-worth of loss of generating capacity penalty.

$\Lambda_{c,s}$ = Capacity penalty for power requirements for operating wet/dry cooling facility.

δ_m = Equivalent supplemental capacity required to offset reliability impact of meterologically-induced power losses.

δ_s = Power requirement to operate heat rejection system.

λ_p = Average power cost (fuel and operating only)for wet/dry cooled plant.

$\lambda(n)$ = Utility system incremental power production cost during time period n.

SUMMARY AND CONCLUSIONS

Combined wet and dry cooling for heat rejection from steam-electric plants is currently being examined by the electric utility industry as an alternative to conventional cooling systems. The use of wet/dry cooling can result in a significant reduction in water requirements for condenser cooling. In water short regions, reduction of the water requirement for electric power production can have important economic, environmental, and political consequences. However, the capital costs of wet/dry cooling systems are significantly in excess of conventional alternatives. Also, the use of wet/dry cooling can result in reduced electrical generation due to increased condensing temperatures and power requirements. Because of this increased capital cost and potential impact on plant generating performance, there is a strong incentive to optimize the design and operation of a wet/dry system for each specific application.

This report presents the results of a detailed evaluation of the economics of wet/dry cooling with large steam-electric plants designed and implemented for cycling or intermediate-load duty. The objective of this work is the determination of the impact of the power plant capacity factor on the design and economics of wet/dry cooling systems. To date, studies of wet/dry cooling have focused on the economics of such systems used with base-load steam-electric plants.

Current generation expansion plans for many utilities designate large coal-fired steam-electric plants as the most economical means for power generation in the intermediate load duty range (about 2000 to 5000 hours of plant operation annually). The optimum design of wet/dry cooling systems is an economic design problem involving both capital and operating costs. Thus, the projected plant load duty or capacty factor can potentially have an important influence on the total cost and optimum configuration of a wet/dry cooling system in a specific application.

The approach taken to achieve the project objectives has been the formulation of a wet/dry cooling system economic evaluation and design optimization model and the subsequent application of the model to two case-study siting situations. The computer model WDCSIM II, which has been formulated for this analysis of wet/dry cooling, is a modification of the earlier WDCSIM computer program (Ref. 1). The case study situations examined are for sites near Phoenix, Arizona and Boston, Massachusetts. At each site all-wet systems, all-dry systems and wet/dry systems with different water requirements have been evaluated.

The WDCSIM II wet/dry cooling system economic model is based on the simulation of the economic operation of a wet/dry-cooled plant as a component of an integrated electric utility system. Input to the model includes power production cost data formulated from the local utility system load and generation characteristics, meteorological data, and power plant and cooling systems thermal performance and cost data. The plant operation simulation routine in the WDCSIM II program accounts for the economic dispatch of the wet/dry plant and optimizes the operating schedule for the wet/dry cooling system in order to maximize the benefit of the available water supply. The optimum design methodology incorporated in the WDCSIM II model utilizes the criterion of minimizing the total cost of heat rejection to define the optimum system design. The total cost of heat rejection is defined as the present-worth sum of capital investment costs, equipment operating costs, loss of electrical capacity penalty costs, and loss of electrical energy production costs.

The wet/dry cooled plant examined in the case studies is a nominal 500 MWe coal-fired unit cooled using separate banks of dry and wet cooling modules arranged in a series flow arrangement. Optimum cooling system designs have been defined for this plant for a baseload operating schedule (capacity factor=0.75) and an economically-dispatched cycling operating schedule (capacity factor=0.40).

Initial design studies in this effort focused on the economics on all-wet and all-dry cooling systems. The important conclusion from these studies is that the optimum design of all-wet and all-dry systems is essentially independent of the assumed plant capacity factor. For the all-wet cooling systems, this result can be attributed to the fact that the cost of wet cooling capacity is relatively low. Cycling operating schedules, although leading to relatively low ratio of operating cost to capital cost in comparison to base-load schedules, do not warrant a reduced cooling capacity. Specifying a cooling capacity which is less than the optimum design for base-load operation leads to capital cost savings that are less than the additional energy replacement and capacity supply penalty costs. For the all-dry systems, the optimum cooling system capacity for a base-load plant is that which results in the maximum allowable condensing pressure at full thermal power at the peak ambient temperature. Again, cycling operating schedules, although leading to relatively low operating to capital cost ratio in comparison to base-load schedules, do not justify reduced cooling capacities. The capacity penalty which would be experienced with a reduced-size design is found to be greater than the capital cost savings. Since the optimum all-dry and all-wet systems designs (hence capital costs) are essentially identical for base-load and cycling operating schedules, the incremental cost of power production due to cooling costs (mills/Kwhr) for the cycling plants is significantly greater - up to 60 to 70 percent depending on the actual capacity factor.

For the wet/dry cooling system evaluations, optimum system designs have been determined for the cases of annual water availability of 250 and 1000 arce-ft. These amounts of water consumption correspond to approximately 10 and 40 percent of the annual consumption for all-wet cooling of the cycling plant. An important conclusion is that for small amounts of water consumption (about 250 a.f./year or less) there is essentially no difference in the optimum design for cycling and base-load operating schedules. The basic reason for this conclusion is that, with this level of available water, there is not sufficient water to significantly impact on the loss of energy production penalty - the major determinant for the sizing of the dry cooling section of the system. Alternatively, for the case of about 40 percent wet cooling it has been found that the assumption of cycling operation as opposed to base-load operation can lead to optimum wet/dry designs which call for significant reductions in the required dry cooling capacity. Nevertheless, for all cases examined, the optimized cost of wet/dry cooling per unit power production (mills/Kwhr) for cycling plants is at least 50 percent greater than the cost of wet/dry cooling with base-loaded plants. This conclusion remains valid even when alternative turbine exhaust end designs are considered for use with cycling plants. This greater cost of wet/dry cooling for cycling plants leads to a greater value, or breakeven cost, of water for use as makeup supply to a conventional all-wet system. Depending on the actual capacity factor of a cycling plant, the value of water for makeup supply may be more than 50 percent greater for a cycling plant in comparison to similar base-loaded plant.

For the cases where there is a significant difference in the optimum design for cycling and base-load operating schedules, the economic impact of operation at off-design capacity factor has been examined. This is of interest since there may be significant uncertainty in the anticipated plant capacity factor. In terms of providing adequate cooling capacity under all conditions, designing for a base-load capacity factor, irrespective of the anticipated capacity factor, would be the conservative approach. The analyses performed indicate such a design approach may be justified in some cases since the additional total cost may be significantly less than the cost penalty which would be encountered if the actual capacity factor were significantly in excess of the design capacity factor.

Other considerations relevant to the design and economics of wet/dry systems for cycling plants which have been addressed include the requirements for makeup water storage and wet cooling control capability. The wet cooling control requirements for optimum water utilization with wet/dry systems operating with cycling plants have been found to be essentially identical to that required for wet/dry systems operated with base-load plants. It has also been determined that for a specific water availability situation, the makeup storage requirement of a plant operating in a cycling mode would in many cases be similar to the storage requirements of the identical plant operated in a base-load mode.

Section 1
INTRODUCTION

WET/DRY COOLING WITH CYCLING STEAM PLANTS

The significant use of wet/dry cooling of steam-electric plants for water conservation has been projected for the United States. The basic motivation for the use of wet/dry cooling with steam-electric power plants is the lack of an available water supply which can be economically consumed in the operation of an entirely evaporative cooling system in an environmentally acceptable manner. Important related considerations involve the site selection process. Decreased dependence on water for heat rejection will, in many cases, allow a greater flexibility in plant siting.

Because of the large capital costs and operational penalties associated with wet/dry cooling, the optimum design and operation of this type of cooling system is a matter of significant economic consequence. Optimum design and operation of wet/dry systems can be achieved only through the correct accounting of both the heat transfer processes occurring in the wet/dry heat rejection system and the economic environment in which the wet/dry cooled plant will operate. The general problem of wet/dry cooling system economics has been reviewed in Reference 1..

The objective of the work reported herein is the evaluation of the economics of wet/dry cooling with large steam-electric plants designed and operated for intermediate load duty operation. To date, analyses of the economics of wet/dry cooling have focused on its application to base-load steam-electric plants. Currently, a significant number of large coal-fired steam plants are being designed for operation with a relatively low capacity factor - so-called "cycling" operation. A recent survey of 60 new plants ordered by utilities found that 22 of these plants are designed for cycling load duty (Ref. 8).

Application of wet/dry cooling to this type of plant can represent a significantly different design situation than that encountered when base-load duty is assumed. As a general rule, designing power systems with a relatively low utilization leads to economic optimum designs which involve minimum capital investment while compromising efficiency. In this report a detailed approach to the evaluation of the economics of wet/dry cooling with cycling plants is presented. Also presented are the results of several case study analyses of the potential impact of low capacity factor on the design and economy of wet/dry cooling.

In the case studies discussed in Section 4, two wet/dry cooled plant siting situations are examined. The specific sites are Phoenix, Arizona, and Boston, Massachusetts, and both involve the siting of a nominal 500 MWe pulverized-coal steam plant. A general discussion of the utilization and design of steam-electric plants for intermediate load duty is presented in Appendix B.

Section 2

GENERAL ECONOMIC MODEL OF WET/DRY COOLING SYSTEMS FOR CYCLING STEAM PLANTS

A comprehensive methodology for evaluating the economics of wet/dry cooling of base-loaded steam-electric plants has previously been developed (Ref. 1). The evaluation methodology is based on the modeling of operation of the wet/dry cooled plant as a component of an integrated electric utility system.

This existing economic evaluation and design methodology can be applied to an economic evaluation of cycling steam plants using wet/dry cooling. Some modifications, however, are required. In this section an economic evaluation methodology for evaluating the costs of wet/dry cooling for cycling steam plants is presented. This economic evaluation is based largely on that presented in Reference 1.

2.1 The General Heat Rejection System Cost Equation

The wet/dry heat rejection system economic design process involves a consideration of the economic tradeoffs between the cost of increased cooling system size and the costs of penalties resulting from inadequate cooling. The optimum design in each case is the cooling system design which results in the least total cost: a sum of the actual capital and operating costs of the cooling facility plus the energy loss and capacity loss penalties resulting from it use. In comparing the costs of alternative system designs the total cost of each design is best expressed as a total present worth of revenue requirements (Ref. 5). The goal of economic design is thus the minimization of:

$$C = C_c + C_o + \Lambda_c + \Lambda_o \qquad (2.1)$$

where C = present worth of revenue requirements for heat rejection over the plant lifetime

C_c = present worth of investment associated costs

C_o = present worth cost of operation of wet/dry system

Λ_c = present worth of loss of generating capacity penalty

Λ_o = present worth of loss of energy production penalty

The above costs are most easily developed in reference to a zero cost steam condensing system which supplies sufficient cooling to maintain the plant at its maximum electrical output at all times.

2.2 C_c, Investment Associated Costs

The heat rejection system capital costs include all costs associated with the acquisition and construction of all plant equipment "downstream" of the turbine exhaust flange. For an annual fixed charge rate, afcr, and a discount rate, i, the present worth of the revenue requirements for the investment associated costs is

$$C_c = Pwf_i{}^m \text{ (afcr) } C_I \qquad (2.2)$$

where C_I = investment cost. The annual fixed charge rate includes the minimum acceptable return on the invested capital, depreciation, income tax, other taxes, insurance expenses and other general and administrative expenses.

2.3 C_o, Operation Cost

The term "C_o" in Eq. 2.1 is the present-worth of all costs incurred in the operation of the wet/dry heat rejection facility. This includes both the cost of electrical power to operate the system pumps and fans and normal operation and maintenance expenses. The applicable equation is

$$C_o = Pwf_i^m \left\{ F \cdot \int_{1\ \text{year}} P_s(t) \lambda_p \, dt + C_{o,m} \right\} \qquad (2.3)$$

where Pwf_i^m = present-worth factor for annual series of payments for m years with a discount rate of i

F = availability factor for plant

$P_s(t)$ = power requirement of wet/dry system (taking into consideration economic shutdowns and maintenance scheduling)

λ_p = average power production cost for wet/dry plant

$C_{o,m}$ = annual operating and maintenance cost

Implicit in the above equation is the assumption that the annual cost of operation of the plant is constant over the lifetime of the plant.

2.4 Λ_o, The Loss of Energy Production Penalty

In Reference 1, a method for evaluating the loss of energy production penalty is developed which is based on the modeling of the optimum operation of the wet/dry cooling system. Incoporated into this evaluation methodology is the assumption that the plant is base-loaded and is off-line only for forced outage and maintenance. To apply the methodology to cycling wet/dry plants it is necessary to additionally account for the economic dispatch of the plant.

The approach taken herein for the evaluation of the loss of energy penalty for cycling wet/dry plants is based on the assumption that the problem of the economic dispatch of the plant by the utility is separable from the problem of the economic allocation of the available cooling water. Simply stated, the approach will be 1) to define the operating schedule of the plant independent of the operation of the wet/dry cooling system, and then 2) to determine the optimum wet/dry system operating schedule for this defined plant operating schedule. This approach is based on the assumption that, on a routine operational basis, water-use considerations do not sufficiently impact the cost of power production of the wet/dry plant to affect its dispatch by the utility.

A general statement of the problem of loss of energy production penalty for cycling steam plants can be developed by first stating the economic objective for the operation of the utility system. The objective is the selection of $u_k(t)$, the dispatch parameter, to minimize the total cost of power production in the utility system:

$$C_{PT} = \sum_{k=1}^{K} C_{Pk}(t) + C_{Po}(t), \qquad (2.4)$$

subject to the constraints that the electrical load demand is satisfied at all times,

$$P_L[t,m(t)] - \sum_{k=1}^{K} u_k(t) O_k(t) P_{max,k} - \left(u_o(t) O_o(t) P_{o,d}(t)\right) = 0 \qquad (2.5)$$

and the water use limitation is not exceeded:

$$\sum_{n=1}^{N} r(n) \leq W_{max} \quad \text{or} \quad \sum_{n=i}^{n=j} r(n) \leq W_{ij} \qquad (2.6)$$

where C_{PT} is the total cost of meeting the load, $P_L[t,m(t)]$, with plants k=1....K and the wet/dry cooled plant. The variables in the above equation are defined in the nomenclature.

The first step in determining the loss of energy production penalty is the modeling of the optimum operation of the planned electric generating system assuming no meteorological impacts on the operation of the wet/dry plant. Quantitatively this involves the optimal selection of $u_k(t)$, k=0....K. From this result the time variation in the utility system incremental power production costs, $\lambda[t,\underline{m}(t)]$, can be calculated. Importantly, this also defines the dispatch schedule for the wet/dry cooled plant.

The second step in determining the loss of energy production penalty is thus the optimization of the operation of the wet/dry cooling facility to meet the condensing load specified by the plant dispatch schedule.

The operation optimization task involves the scheduling of the operation of the wet/dry cooling system such that the economic benefit to the operation of the power plant is maximum. Quantitatively, the cooling system operation optimization problem for a plant of average power production cost, λ_p is:

$$B_T = \sum_{n=1}^{N} B(n) = \sum_{n=1}^{N} [\lambda(n)]P_{o,d}(n) = \text{maximum} \qquad (2.7)$$

subject to the constraint

$$\sum_{n=1}^{N} r(n) \leq W_{max} \quad \text{or} \quad \sum_{n=i}^{n=j} r(n) \leq W_{ij} \tag{2.8}$$

The term $[\lambda(n)]P_{o,d}(n)$ in Equation 2.7 is the value of power $P_{o,d}(n)$ supplied to the utility grid during time period n when the incremental fuel cost in the utility system is $\lambda(n)$. Hence it represents the value of operating the wet/dry cooling system to achieve the power level $P_{o,d}(n)$. For some time periods, the optimum operation of the wet/dry cooling system may require reduction in the thermal power (i.e., fuel burning rate) of the plant. In such cases, credit must be given to the fuel savings as well as the value of the actual power produced. Thus Eq. 2.7 written in complete form is:

$$B_T = \sum_{n=1}^{N} B(n) = \sum_{n=1}^{N} [\lambda\ (n)]\ P_{o,d}(n) + S_{fuel}(n) = \text{maximum}$$

where $S_{fuel}(n)$ is the fuel savings resulting from operation at power $P_{o,d}(n)$. The term $S_{fuel}(n)$ accounts for the fact that, if the plant thermal power is reduced to minimize water consumption, the actual penalty for the lost electrical power resulting from the thermal power reduction is the difference between the fuel cost of the wet/dry plant and the utility system incremental fuel cost.

The third step in determining the loss of energy production penalty involves the actual calculation of the loss of energy production penalty using the equation:

$$\Lambda_o = Pwf_i^{\,m} \cdot F \cdot \left[B_{Tref} - B\right]_{min} \tag{2.9}$$

where

$$B_{T_{ref}} = \sum_{n=1}^{N} B_{ref}(n) = \sum_{n=1}^{N} \lambda(n) P_o(n) \tag{2.10}$$

The quantity $P_o(n)$ is the dispatched power level of the reference plant and B_{Tref} is the total annual economic benefit of the dispatched operation of the reference power plant of capacity P_o.

Thus, this final step involves finding, for the same dispatch schedule, the difference between the energy production benefit of the wet/dry cooled plant and the energy production benefit of the reference plant.

2.5 Λ_c, Loss of Capacity Penalty

The evaluation of the loss of capacity penalty for wet/dry cooled cycling steam plants can be undertaken in a manner identical to that outlined in Reference 1.

The utility system generation reliability will, in many cases, be adversely affected by the meterologically-induced capacity loss from a wet/dry cooled plant. The loss-of-capacity penalty as developed in the following paragraphs represents the investment costs associated with sustaining the utility system power generation reliability at the required level. Also, wet/dry cooling systems will require a considerable amount of power to operate the many fans and pumps and thus these generating capacity costs must also be considered.

The capacity penalty is thus divided between two parts:

$$\Lambda_c = \Lambda_{c,m} + \Lambda_{c,s} \tag{2.11}$$

where $\Lambda_{c,m}$ = capacity penalty due to meterologically-induced power losses, and

$\Lambda_{c,s}$ = capacity penalty for power requirments for operating the wet/dry cooling facility

A method for assessing the generation reliability impact resulting from meterological effects on the performance of wet/dry heat rejection systems is presented in Reference 1. The goal of this reliability assessment is the determination of the "equivalent supplemental" generating capacity additions which are required to maintain the required level of power generation reliability. Once this "equivalent supplemental capacity", δ_m, for offsetting the wet/dry plant losses due to meterological fluctuations has been determined, the associated capacity penalty would be found from:

$$\Lambda_{c,m} = Pwf_i^m \text{ (afcr) } K_p \delta_m \tag{2.12}$$

where K_p is the incremental capital cost of peaking type of generating units (i.e., \$/kw). A cost representative of peaking type units has been chosen since the frequency and magnitude of the meteorology-induced capacity losses is most similar to the electrical demand normally assigned to peaking types of generating units.

The capacity penalty associated with the power requirement of the heat rejection system itself would be determined from:

$$\Lambda_{c,s} = Pwf_i^m \text{ (afcr) } K_b \delta_s \tag{2.13}$$

where δ_s is the power requirement of the cooling system and K_b is the cost of base-load or intermediate-load duty generating plants. (Note that the cost of power [mills/Kwhr] for the auxillaries is correspondingly taken as λ_p and not $\lambda(n)$.)

In some cases a wet/dry plant may utilize a so-called "high-back pressure" turbine which has a reduced base efficiency (as shown in Figure 4.1). In this case, the difference in the maximum output of "high-back pressure" turbine and a conventional turbine would be included in the quantity δ_m.

2.6 Incremental Cost of Power Production

In comparing the costs of alternative systems it is of interest to express the typically large values calculated for C in more readily understandable terms. This can be achieved by use of the equation:

$$e_i = \frac{\left(\begin{array}{c}\text{Equivalent Uniform}\\ \text{Annual Cost}\end{array}\right) \times 1000}{G} = \frac{C \cdot 1000}{(Pwf_i^m)G} \tag{2.14}$$

where e_i = incremental power production cost resulting from the cost of heat rejection (mills/Kwhr)

Pwf_i^m = the present worth factor for a uniform series of annual payments at interest rate i for m years

G = total annual generation from ideally-cooled reference plant (mwhrs)

For a given plant both Pwf_i^m and G would be constant for all heat rejection system designs.

Section 3

WDCSIM II - ECONOMIC EVALUATION MODEL FOR WET/DRY COOLING WITH CYCLING STEAM PLANTS

The general economic model presented in Section 2 provides the basis for the economic evaluation of wet/dry cooling with cycling steam plants. This section presents the specific approaches and assumptions which are used in implementing the cost model in the WDCSIM II computer program for a specific set of case-study situations. The computer program WDCSIM II is a modified and expanded version of the WDCSIM program presented in Reference 1.

In this section the specific approaches which are incorporated into WDCSIM II to evaluate the loss of energy production penalty, the loss of capacity penlty, and the wet/dry system capital and operating costs are presented.

3.1 Computation of Loss of Energy Penalty in WDCSIM II

The procedure in Reference 1 for evaluating the loss of energy penalty is based on simulation of the wet/dry cooled plant as an operating component of an integrated utility system. An operation simulation is used to determine the wet/dry cooling system operating schedule which will result in the maximizing of the economic benefit of the available water. The basis for the optimum operation simulation is the computation of a so-called "incremental water-use benefit" for each feasible operating condition of the wet/dry system for a representative year of plant operation. The incremental water-use benefit is defined as

$$\gamma\,(n,I) = \frac{B(n,I) - B(n,I-1)}{r(n,I) - r(n,I-1)} = \frac{\$}{\text{gallons of water consumed}} \qquad (3.1)$$

where $B(n,I) = [\lambda(n)]\ P_o(n,I)$

I = operating state of the wet/dry cooling system during time period n

$P_o(n,I)$ = power output of wet/dry plant at time period n of dry bulb temperature $T_{db}(n)$ and of wet bulb temperature $T_{wb}(n)$ and at wet/dry cooling system operating state I,

$r(n,I)$ = rate of water consumption of wet/dry plant at operating state I during time period n

Having determined the $\gamma(n,I)$ functions for each hour of the year it is then possible to determine the optimum state of operation for each hour of the year. The operation optimization procedure involves the repetitive searching of the $\gamma(n,I)$ function for the various possible states of operation, I, of the wet cooling system for all hours of the year. The intent of the searching is to identify the hours of the year, and the specific operational states at those hours, which have the highest incremental water-use benefit. Starting with the hour and the operational state which has the greatest value of $\gamma(n,I)$, the available water is allocated for consumption at the appropriate rate $r(n,I)$. The water allocation proceeds by continually allocating a portion of the remaining available water to the hours n and states I with successively lower values of γ. The procedure continues until the available water is completely allocated. The operation state, I, defined for each hour, n, by this procedure is the optimum wet/dry operational scheme.

This optimization procedure has been incorporated in the WDCSIM computer program presented in Reference 1. This procedure, as described in Reference 1, assumes base-load plant operation with scheduled maintenance outage and random forced outage. To apply this optimum allocation technique to a cycling wet/dry plant it is necessary, in addition, to account for the scheduled or dispatched outage of the power plant.

The WDCSIM program has been modified by the addition of a routine which can function to economically dispatch the wet/dry plant. This dispatch is done according to the current value of $\lambda(n)$. If $\lambda_p > \lambda(n)$ the plant is shut down and if $\lambda_p < \lambda(n)$ the plant operates at full thermal power. This routine can also function to dispatch the wet/dry plant on a fixed time sequence.

The simulation procedure subroutine SIMULE in the WDCSIM II program computes the annual loss of energy production penalty. The SIMULE subroutine determines the optimum operating state for each of the 2920 three-hour time periods occuring during the simulated year of operation. These optimum operating states define the loss of energy production at each time period. The loss of energy production penalty for each time period for which the plant is scheduled for operation is computed by multiplying this loss of energy by the concurrent utility system incremental power production cost (i.e., system "lambda").

3.2 Computation of Loss of Capability Penalty in WDCSIM II

Similar to the original WDCSIM program, the WDCSIM II program does not have the capability to evaluate the impact of meterologically-induced capacity losses on utility system capacity supply requirements. Rather, it is necessary to make some assumptions as to the relationship between capacity loss characteristics of the plant and penalty for lost capacity.

The assumptions incorporated in the WDCSIM II program are essentially those employed in the WDCSIM program. They are that the cost penalty for meterologically-induced capacity losses are directly proportional to the maximum capability reduction of the plant at extreme ambient conditions and that the cost penalty for pump and fan capacity is proportional to the maximum auxiliary power requirement. Thus, with reference to Eq. 2.12 and 2.13,

$$\delta_m = P_o - P_{min} \tag{3.3}$$

and

$$\delta_s = P_{aux} \tag{3.4}$$

where P_o = reference capacity of plant

P_{min} = capacity of plant at maximum ambient temperature condition with maximum wet cooling operation

P_{aux} = maximum auxiliary power requirement of wet/dry cooling system

3.3 Wet/Dry Cooling System Performance and Capital Cost Models in WDCSIM II

3.3.1 Performance Models for Wet/Dry Heat Rejection System

3.3.1.1 General.

The heat rejection system model incorporated in the WDCSIM II program is based on a separate wet and dry tower system in a series flow arrangement. Thermal performance models employed for this system are consistent with those employed in WDCSIM with the exception of the condenser calculations which have been upgraded. In this system, the cooling water is pumped through the condenser, through a connecting pipe system, and is equally distributed among the mechanical draft dry tower modules. After being partially cooled by the dry towers a fraction of the flow is diverted to the booster pumps where it is pumped through the wet tower modules. The mixed outlet from the cooling tower is then fed back to the circulating water pumps. This system is illustrated in Figure 3.1.

The system design variables for this wet/dry cooling system are:

(1) the number of dry cooling modules,
(2) the number of wet cooling modules,
(3) the condenser flow rate, and
(4) the water loading of the wet tower modules.

The following paragraphs are descriptions of the models for the individual components of the wet/dry heat rejection system.

3.3.1.2. Condenser.

The condenser is constructed with a stainless steel two pass waterbox and 1 in O.D. 20 BWG 304 stainless steel welded tubing.

A heat balance is performed on the condenser for the conditions of maximum steam flow with a turbine exhaust pressure of 2.5 in Hg. From this balance the heat exchanger tube length and the number of tubes are sized to achieve a 5°F terminal temperature difference (TTD).

3.3.1.3 Dry Tower Modules.

The dry towers are mechanical draft dry towers of the cross-flow type. The heat exchanger banks are 61 feet long and have one inch O.D. tubes arranged in a two-pass flow configuration.

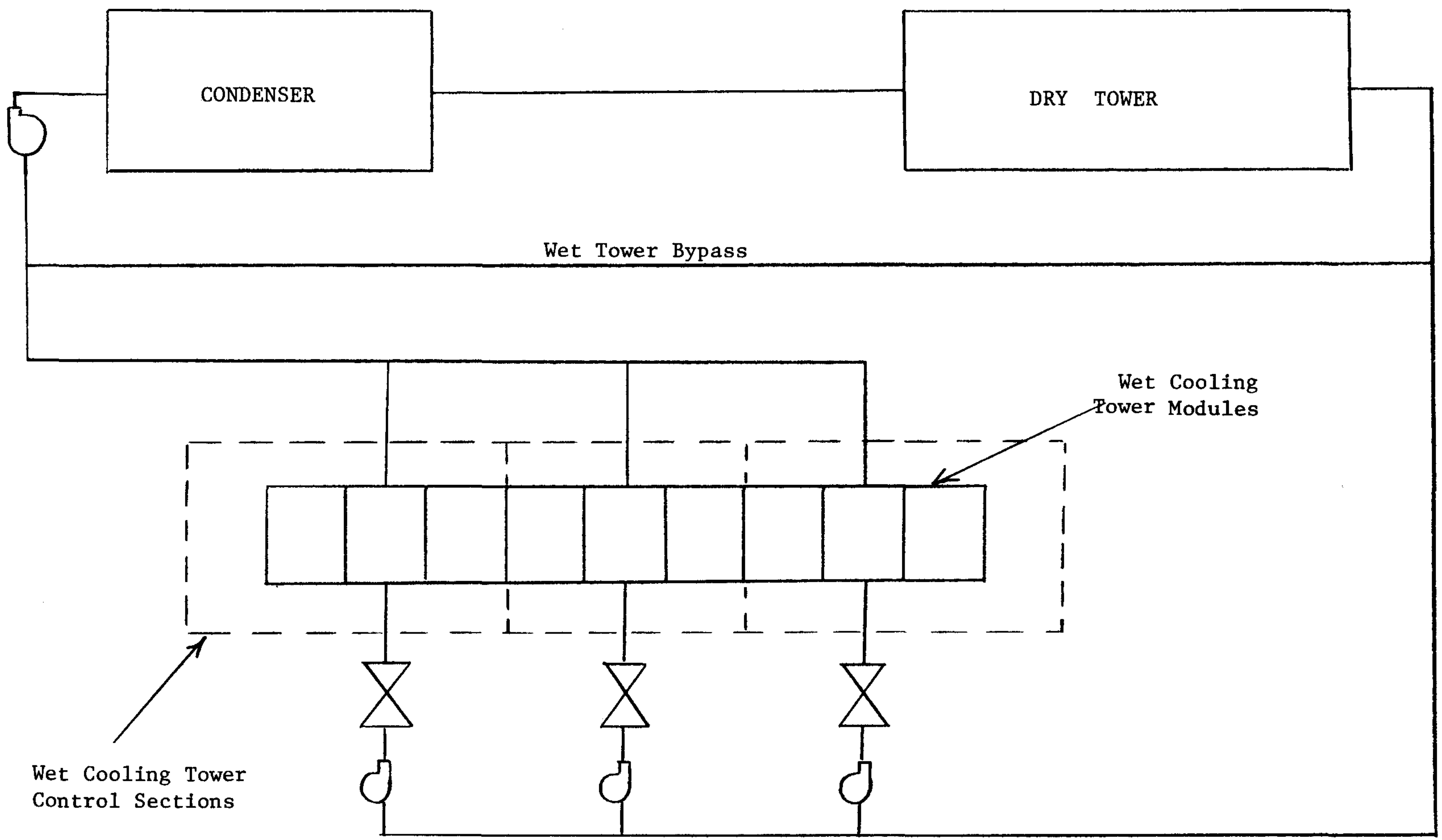

Figure 3.1

Schematic of Series-Flow Separate
Wet and Dry Tower Heat Rejection System

Each dry tower module has an overall heat transfer coefficient of 1,150,000 BTU/hr - °F and an air flow rate of 8,050,000 lbs/hr at 60°F. The heat transfer effectiveness of the dry tower is corrected for variations in air flow rate due to changes in the ambient temperature.

3.3.1.4 Wet Tower Modules.

The wet tower module is a cross-flow type and contains two fill sections each 21 feet wide, 41 feet long, and 41 feet high. The total air flow rate through the tower fill is assumed to be 3,444,000 lbs/hr.

The temperature of the water and the specific humidity of the air leaving the wet tower is assumed to be a function only of the inlet water temperature, the ambient air wet bulb temperature, and the water flow rate. The heat and mass transfer model used to compute the module cooling performance and evaporation rate is that given by Croley, et al. (Ref. 2). A tower fill transfer coefficient of 0.034 is assumed.

3.3.1.5 Circulating Water Pipelines.

Water is distributed to the cooling towers via a network of steel circulating water pipelines. Both the wet and dry tower module header lines feed off the main circulating water pipeline. Each pipe is sized to yield a 12 ft/sec water velocity.

The dry towers are arranged in arrays with two linear banks of modules each. This grouping allows both banks to share the same pipelines, thus permitting the use of more cost effective larger diameter piping (Ref. 3).

For simplicity the wet tower modules are assumed to be arranged in a single linear bank of modules.

Figure 3.2 illustrates the assumed piping and cooling tower layout.

3.3.1.6 Circulating Water Pumps.

Both the circulating water pumps and the wet tower booster pumps are capable of delivering 3,000 brake-horsepower. The main circulating water pumping power requirements are established from the overall head loss through the system and flow rate, assuming a pump efficiency of 88 percent and a pump utilization factor of 0.75. The wet tower booster pumps are sized for a pumping head of 100 feet.

3.3.2 Capital Cost Assumptions for the Wet/Dry Cooling System

The cost algorithms used in this study are similar to those used in the WDCSIM computer model. However, several aspects of the cost model have been upgraded to better represent the effect of the system design variables on the equipment capital costs.

Costs are computed in 1985 dollars and are based on the number of dry modules (ND), number of wet modules (NW) and the circulating water flow rate (WF). The condenser cost is a function of the area (AREA), length (LTUBE), and number (NTUBE) of condenser tubes which were computed based on WF in subroutine COND.

Each cost equation has been compared with information from References 3 and 4 and other sources to arrive at representative relationships.

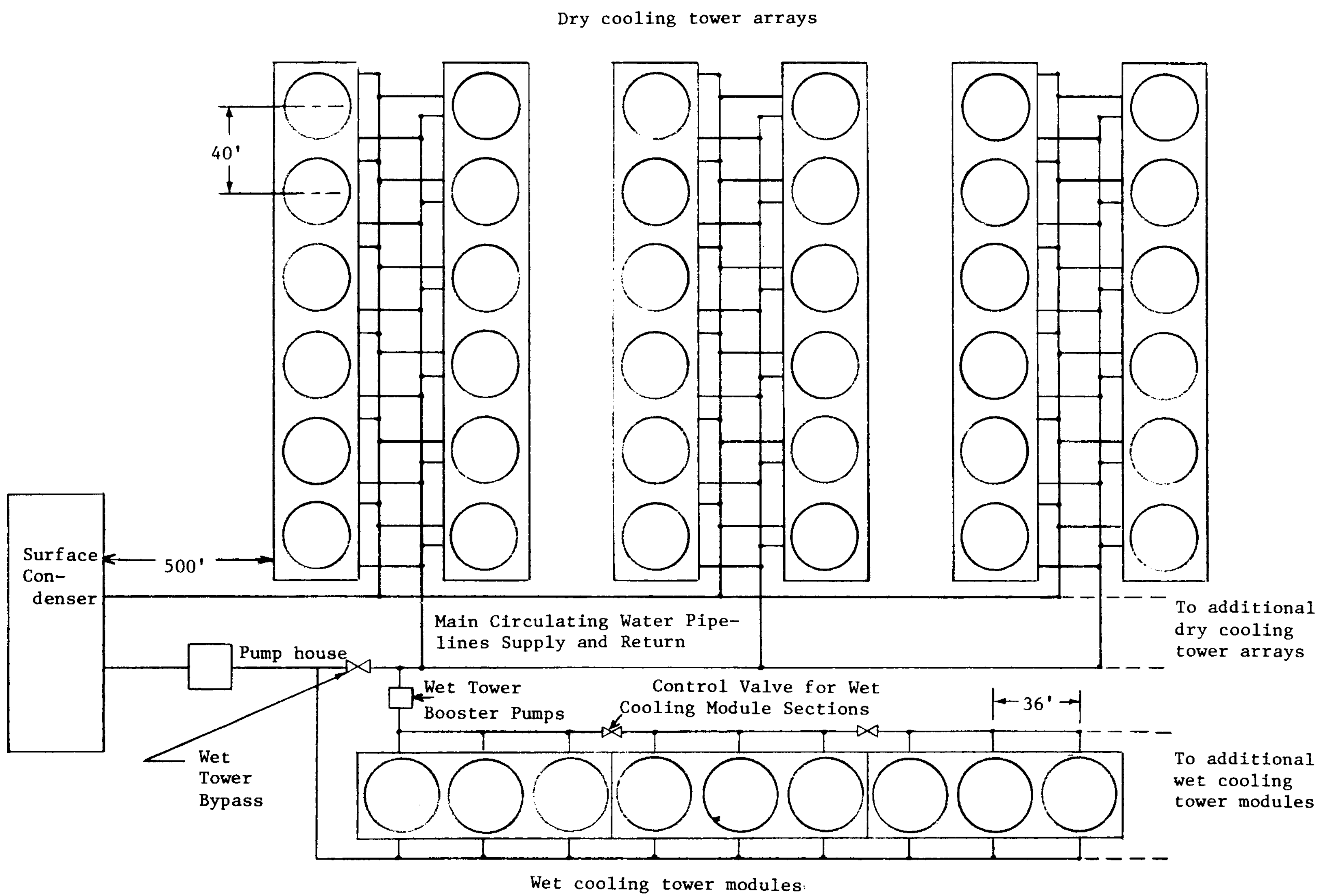

Figure 3.2 Series Flow Separate Wet and Dry Tower Heat Rejection System and Piping Layout

The specific capital cost relationships are as follows:

1) Dry tower modules and foundations:

$$C_1 = 349{,}000 \times ND \qquad (3.5)$$

2) Wet tower modules and foundations:

$$C_2 = 590{,}000 \times NW \qquad (3.6)$$

3) Surface condenser (condensed from Reference 3):

$$C_3 = CCS + CCT + CER \qquad (3.7)$$

$$CCS = NCS\ (CBS\ [CFL + CFCS] + CAUX)$$

where NCS = No. of shells = 2

CBS = Basic shell cost

$= 338{,}605 + 10.07 \times NTUBE$ (3.9)

CFL = Length correction factor

$= 0.1572 \times LTUBE + 0.58$

CFCS = Factor account for condenser design and materials

$= 0.00997 \ln \frac{NTUBE}{2} - 0.417$ (3.10)

CAUX = Auxiliary equipment cost per shell

$= 0.645 \times AREA$ (3.11)

CCT = Cost of tubes = $6.88 \times AREA$ (3.12)

CER = Erection costs = $4.66 \times AREA$ (3.13)

4) Electrical equipment

$$C_4 = 57{,}000 \times ND + 87{,}700 \times NW \qquad (3.14)$$

5) Steel pipelines

Each component of the piping system described in Section 3.3.1.5 is costed using cost algorithms which approximate the cost data given in Reference 1.

For the main circulating water pipeline and the wet module piping, each pipe, valve, tee, and reducer is sized, costed, and summed to give the total cost.

For the dry tower module piping, a single tower array is costed out in a similar fashion. The cost per module is assumed to be constant for all dry tower arrays and equal to the cost per array divided by the number of modules in the array. This cost times the total number of dry tower modules is the dry tower modules piping cost.

The total piping cost is:

$$C_5 = CC + CD + CW \tag{3.15}$$

where CC = main circulating pipeline cost

CD = dry module piping cost

CW = wet module piping cost

The cost of each component in the piping system is evaluated as a function of the pipe diameter (D). The actual cost algorithms are as follows:

Main Circulating Water Piping System (for $D \gtrsim 75$ inches)

Pipe costs = CCP = $23.5 \times D - 612$	\$/ft	(3.16)
Tee Costs = CCT = $2142 \times D - 105910$	\$	(3.17)
Reducer Costs = CCR = $14.3 \times D^2 - 78125$	\$	(3.18)

Module Piping System

Pipe Costs = CMP = $7.05 \times D + 42$	\$/ft	(3.19)
Tee Costs = CMT = $169 \times D - 388$	\$	(3.20)
Reducer Costs = CMR = $55.7 \times D + 348$	\$	(3.21)
Valve Costs = CMV = $929 \times D - 8954$	\$	(3.22)

6) Circulating water pump structures

$$C_6 = 0.362 \times C_7 \tag{3.23}$$

7) Circulating water pumps and motors

$$C_7 = 500 \times P_{main} + 600 \times P_{booster} \tag{3.24}$$

where 500 = \$/Hp for main circ. pump

P_{main} = required pumping power capability (Hp)

= CUNIT x CFS x PL/(0.75 x 0.88) (3.25)

CUNIT = unit conversion factor = 0.2618

CFS = circ. water flow rate (ft^3/sec)

PL = pressure drop through piping system (lb/in^2)

0.75 = pump utilization factor

0.88 = pump efficiency

600 = \$/Hp for wet booster pumps

$P_{booster}$ = desired pumping power capability (Hp)

= CUNIT x WL x NW x P/(0.75 x 0.88) (3.26)

where WL = water flow to each wet module

P = 43.35 (psia)

8) Indirect capital costs (3.27)

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

9) Total capital costs are thus, (3.28)

$$C = \sum_{i=1}^{8} C_i$$

3.3.3 Operating Cost for Fans and Pumps

The annual cost of operating the pumps and fans is computed as:

$$C_{p+f} = \lambda_b \cdot E_t \tag{3.29}$$

where E_t = total annual energy required to operate the fans and pumps.

The power consumed in operating the heat rejection system is assumed to be as follows:

1) Fan power for dry tower = 0.112 MWe x ND (3.30)
2) Fan power for wet tower = 0.149 MWe x NW (3.31)
3) Booster pumping power = 0.150 MWe x NW (3.32)
4) Circulating water pumping power = $\dfrac{4.35 \times 10^5 \cdot Q_t \cdot \Delta P}{\eta}$ (3.33)

whete Q_t = circulating water flow rate (Gpm)

η = pump efficiency = 0.88

ΔP = overall pressure loss through circulating water system

$$= 21.9 + 3.087 \times 10^{-7} \left(\frac{Q_t}{ND}\right)^2 + 1.482 \times 10^{-2} \left(15.34 + 1.141 \times 10^{-3} \left(\frac{Q_t}{ND}\right)^2\right) \frac{lbf}{in^2}$$

and ND = number of dry modules

NW = number of wet modules

3.4 Use of WDCSIM II Program for Design Optimization

The design variables considered in optimizing wet/dry system designs for various case study situations are those listed in Section 3.3.1.1. The condenser size in all cases is fixed by assuming a 5°F terminal temperature difference at a condenser pressure of 2.5 in H_g.

The procedure used to determine the optimum combination of design variables in each case is a simple success-guided grid search of feasible combinations. This procedure is seen to be adequate for present purposes and quickly leads to optimized costs within about 2 percent of the true minimum. Greater accuracy can be obtained with additional effort.

Section 4

CASE STUDIES OF WET/DRY COOLING WITH CYCLING STEAM-ELECTRIC PLANTS

In this section, the results of the case-study evaluation of wet/dry cooling for cycling steam-electric plants are presented. Included in this section is the description of the specific site, power plant, and utility system characteristics input to the WDCSIM II computer program to perform the case studies.

4.1 Power Plant Characterization

The power plant modeled in the case studies is a 500 MWe coal-fired plant designed for cycling service. The fossil-steam turbine operating characteristics are based on data from Reference 7 for a 500 MWe tandem-compound 3600 RPM, 2400 psig, 1000°F/1000°F turbine with a 5 in. Hg back pressure limit. The heat rate curve for this turbine has been extrapolated to 8 in. Hg. Also, a heat rate curve for a hypothetical high back pressure turbine is used to investigate the benefits of applying this design to wet/dry cooling. The heat rate curves for these various turbine designs are given in Figure 4.1.

The maximum thermal power level of the plant is 1130 MWt. This results in the reference power output of 500 MWe at a condensing pressure of 2.5 in Hg. The maximum output of this plant using the high back pressure turbine is 465 MWe.

In the case studies, the plant is modelled for both economically-dispatched cycling operating schedules and base-load operation schedules. Forced outage rates of 18 percent and 25 percent and maintenance periods of two weeks and four weeks are assumed for the cycling and base-load plants, respectively.

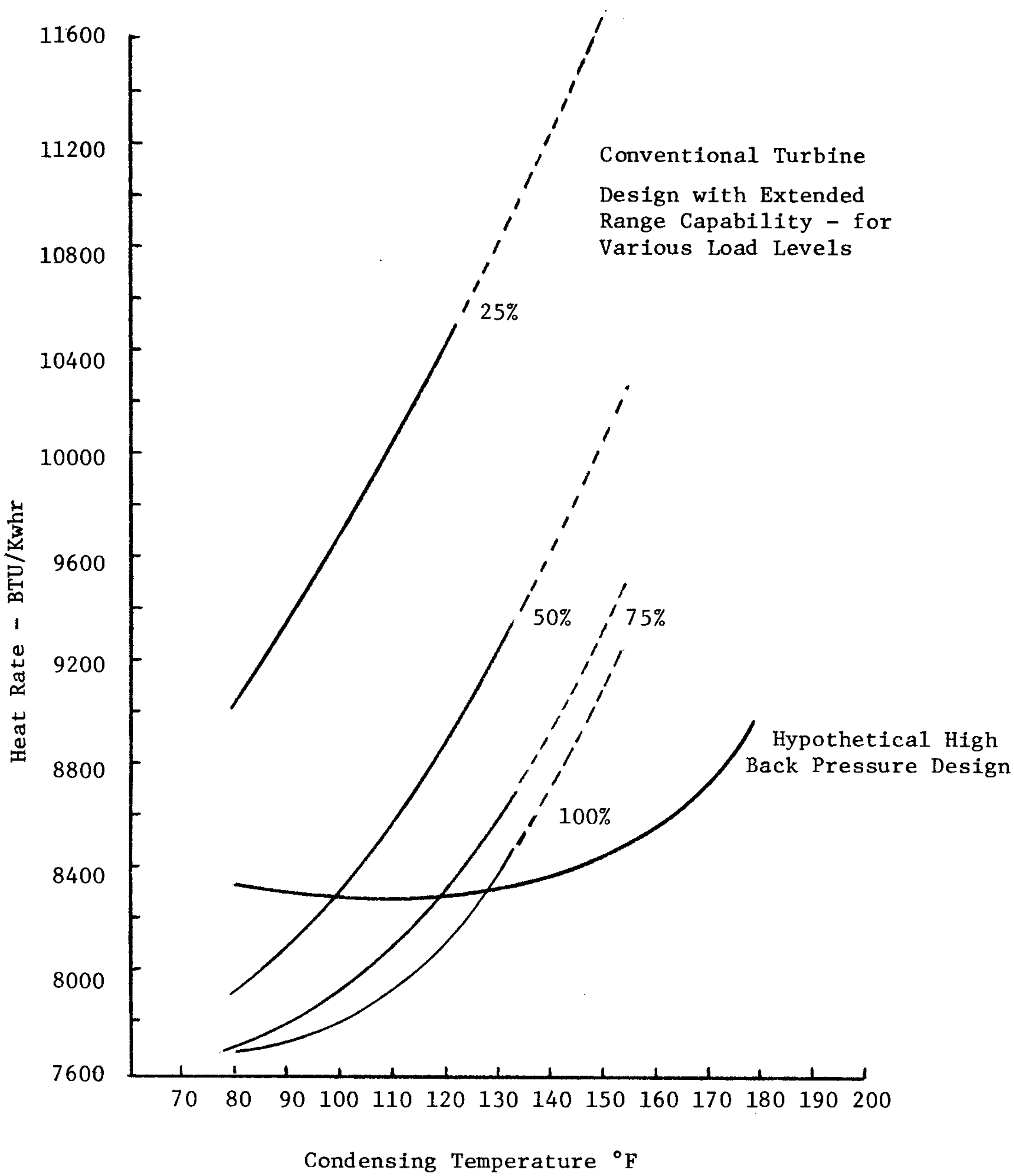

Figure 4.1 Turbine-Generator Heat Rate BTU/Kwhr

All-wet, all-dry, and wet/dry cooling system designs are evaluated for the 1129 MWt plant for two siting situations - Boston, Massachusetts, and Phoenix, Arizona. The assumed fuel cost for the Boston and Phoenix plants is 21.9 and 10.9 mills/kwhr, respectively. When considering the high back pressure turbine design cases, these values are corrected by the ratio of the base heat rate of the conventional and high back pressure turbines.

4.2 Utility System and Economic Models

Use of the WDCSIM II program requires the input of a chronological time series of utility system incremental power production cost data and site meterological data. To model the time-variation in the incremental power production cost, electrical load data for 1975 have been obtained for the Boston and Phoenix region utility systems. These load data are consistent with the meteorological data which are also for 1975. From these load data, data sets for the time-varying incremental power production cost were formulated based on the assumption of a representative mix of generating plants in each utility system.

The specific procedure employed to develop the incremental power production cost data in each case is as follows:

(1) assume mix of generating plants of sufficient capacity to meet the load (Figures 4.6 and 4.7)

(2) use the capacity of plants along with their associated heat rates and fuel costs to formulate a relationship between the incremental power production cost and the electrical load, and

(3) use this relationship to assign to each load data point an appropriate incremental fuel cost.

The cumulative load-duration curves for the assumed Arizona and Massachusetts region electric utility systems are shown in Figure 4.2 and 4.3, respectively. The cumulative temperature-time duration curves for these same two localities for the same year (1975) are presented in Figure 4.4 and 4.5. The Massachusetts case represents a utility system with a summer peak load but with high winter peaks and a relatively high average incremental fuel cost. The Arizona case is a summer peaking situation in which there is a strong correlation between ambient temperature and utility system electrical load and in which the average incremental fuel cost is relatively low.

The representative mix of generating plants presented in Reference 1 for the Arizona and Massachusetts locations has been modified to reflect the expansion of these systems with a coal-fired steam-electric unit designed for cycling load duty. The mix of generating plants assumed for each of these utility systems is presented in Table 4.1 and 4.2, respectively. The fuel costs presented in these tables are in 1979 dollars and are escalated at 8 percent annually to yield 1985 costs. The "system lambda" versus system electrical load graphs presented in Figures 4.6 and 4.7 for the two utility systems have been developed assuming a very simple logic for dispatching the available plants. Using this assumed "system lambda" versus load function and the historical load data, a chonological time series of "system lambda" data have been generated for use in the wet/dry plant operations modeling. For the simulated year of operation, the two designated coal-fired plants in Tables 4.1 and 4.2 have an average capacity factor of about 40 percent.*

For the purpose of evaluating loss of capacity penalties, it is assumed that all meterologically-induced capacity losses are charged at a rate of $208/Kw to reflect the cost of peaking-type of capacity and that all auxiliary power requirements are charged at a rate of $1,190/Kw

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* In the case studies, the plant is dispatched to operate at zero or the rated thermal power only. The capacity factor is defined here as the percentage of time at full thermal power.

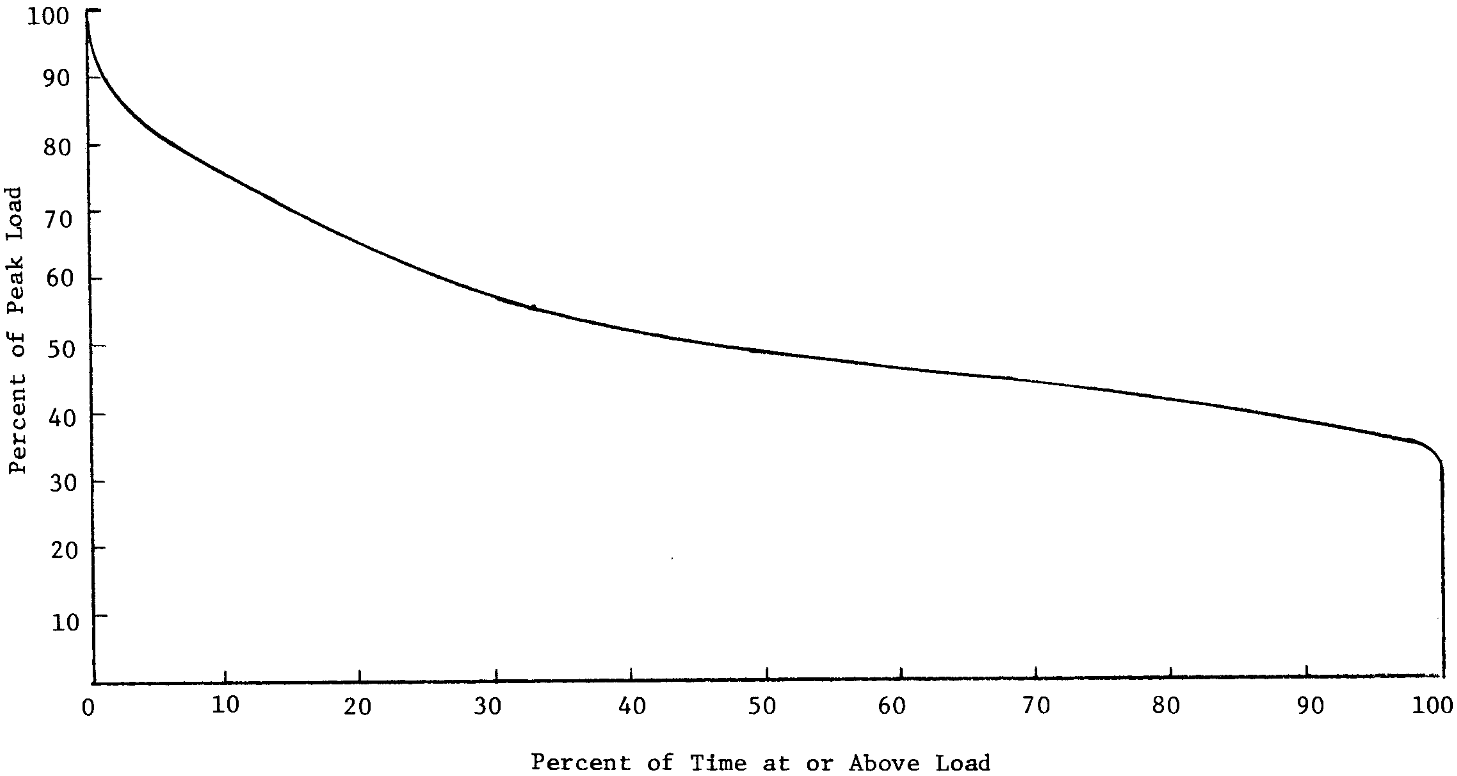


Figure 4.2 Cumulative Load Duration Curve- Arizona Region

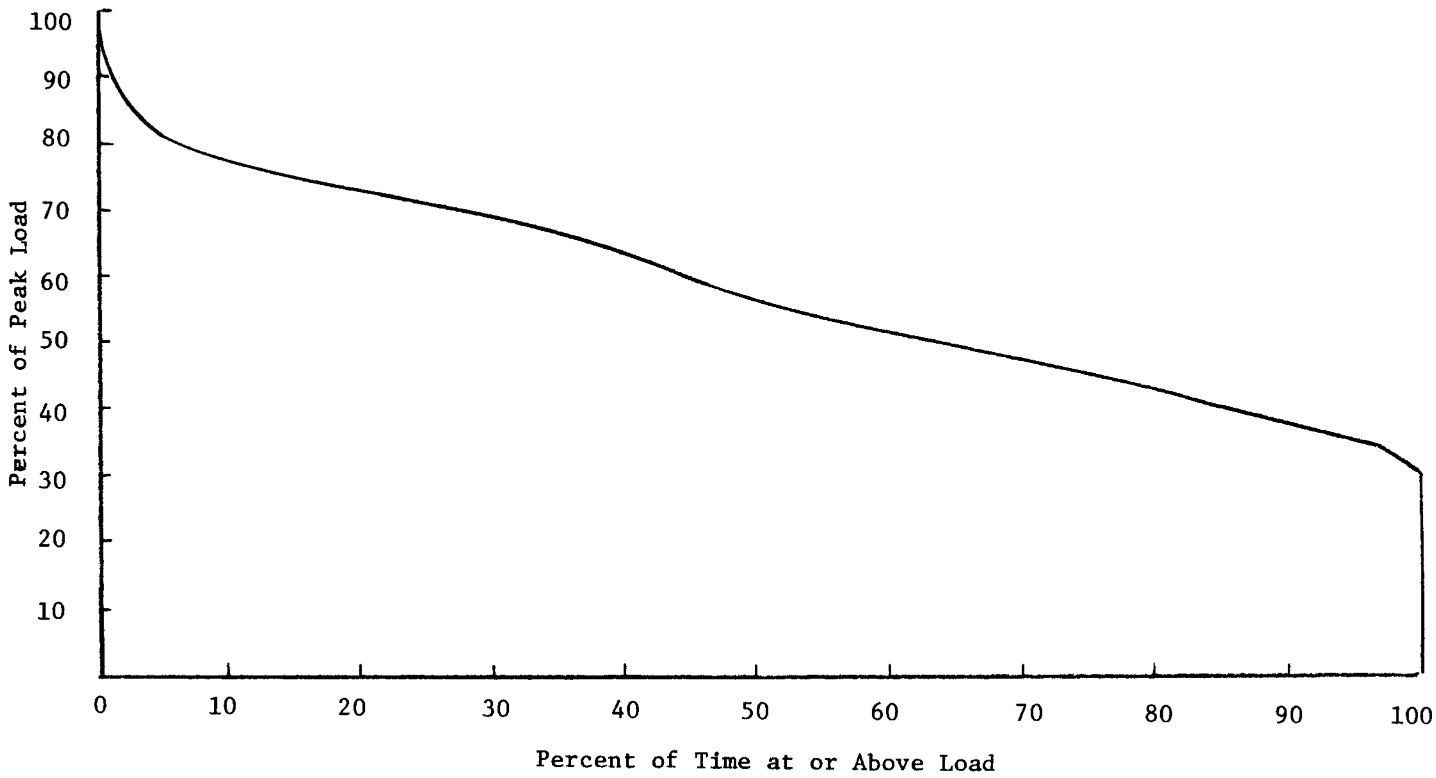


Figure 4.3 Cumulative Load Duration Curve - Massachusetts Region

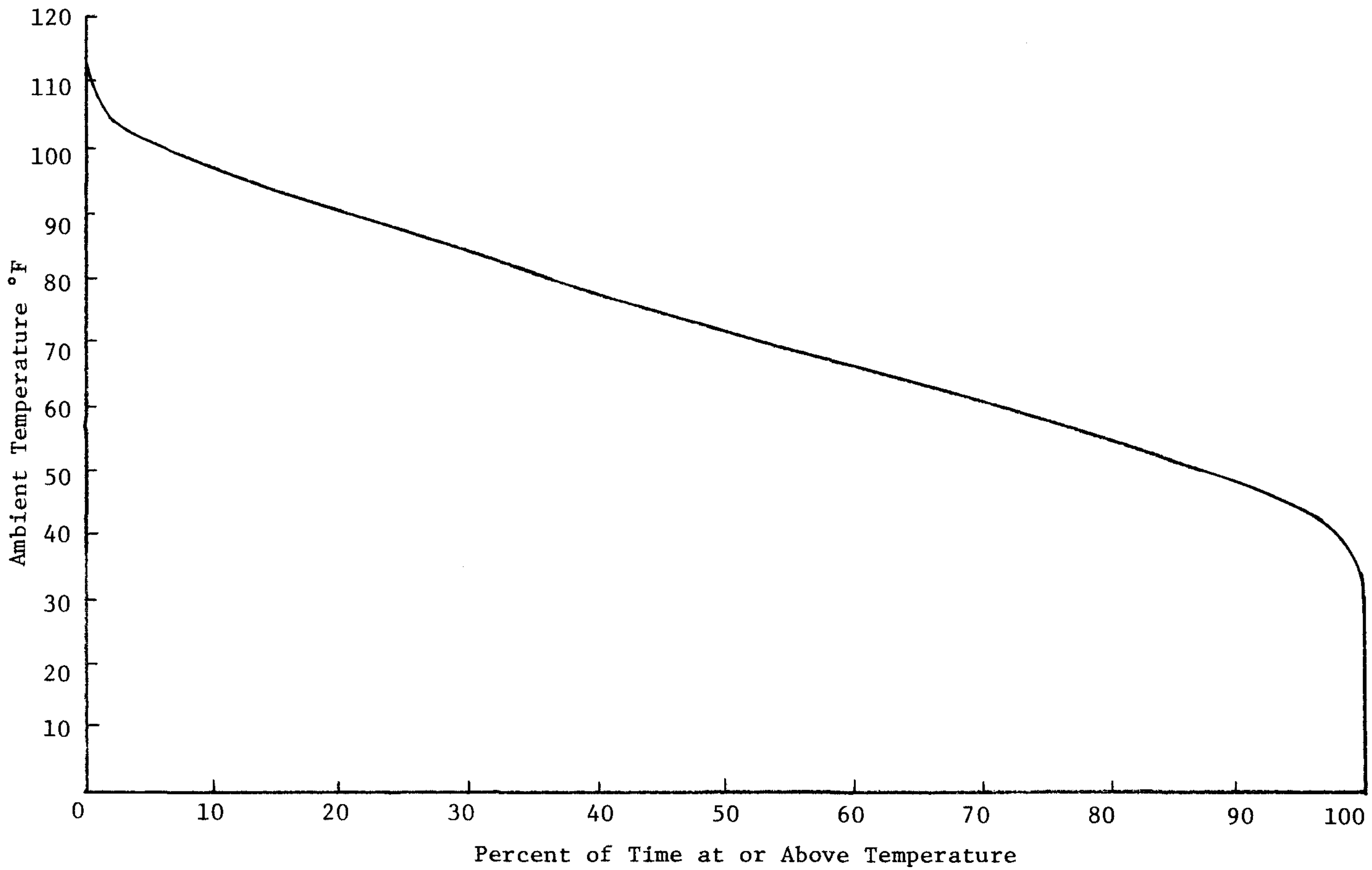


Figure 4.4 Ambient Temperature Duration Curve - Phoenix

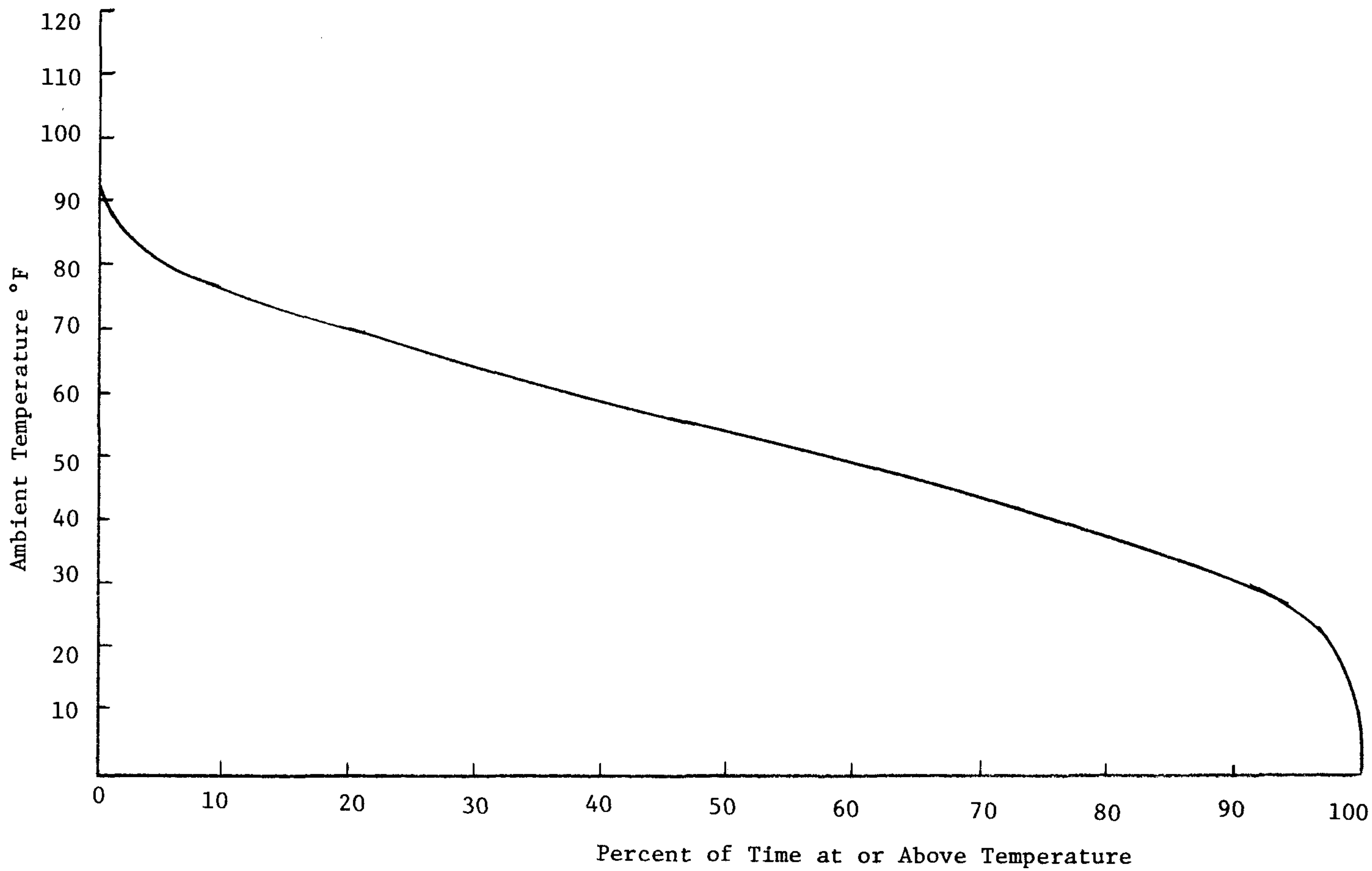
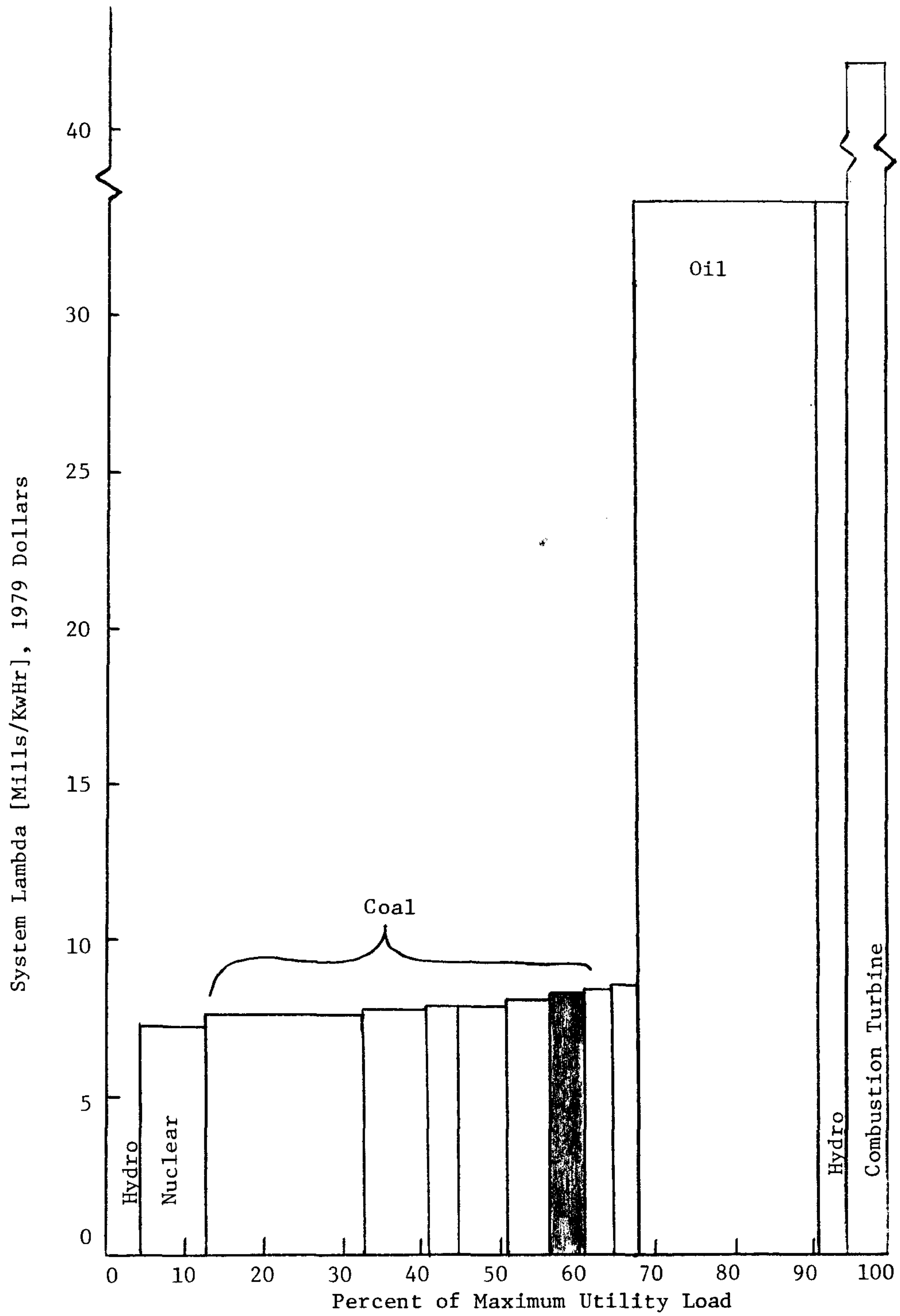


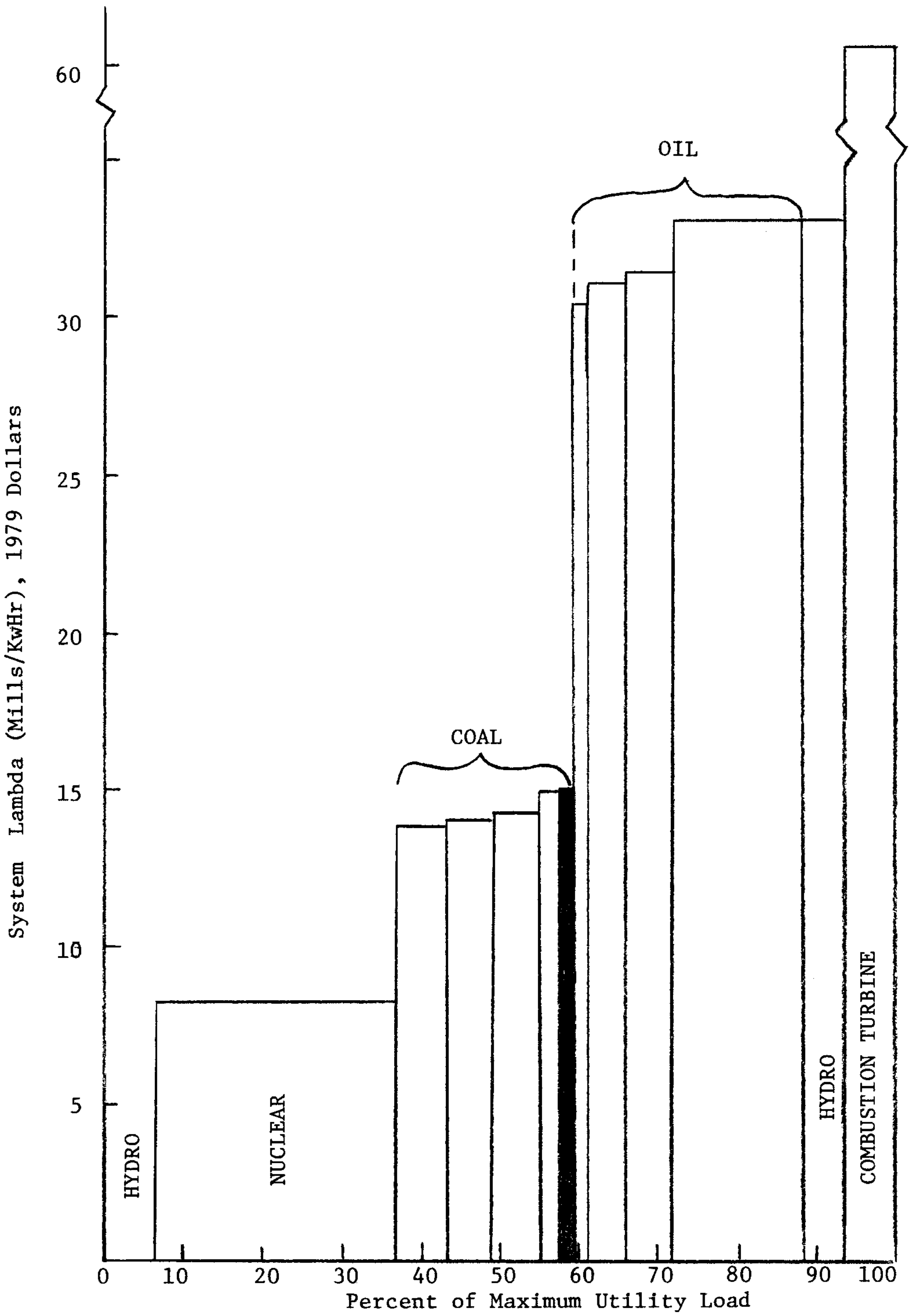
Figure 4.5 Ambient Temperature Duration Curve - Boston *

* National Weather Service data for Logan International Airport



Shaded area denotes cycling plant studied.

Figure 4.6 Load-lambda Curve for Arizona Utility System



Shaded area denotes cycling plant studied.

Figure 4.7 Load-lambda Curve for Eastern Massachusetts Utility System

Table 4.1

Massachusetts Utility System

Unit Type	Number of Units	Unit Size	Incremental Fuel (1) Costs (Mills/KwHr)	
Hydro	32	50	0.	(2)
Nuclear	3	1200	8.3	
Nuclear	3	1000	8.3	
Nuclear	5	600	8.3	
Coal	2	800	13.9	
Coal	5	600	14.1	
Coal	5	400	14.2	
Coal	4	200	15.0	
Coal (3)	1	500	15.0	
Oil	1	800	30.5	
Oil	3	600	31.1	
Oil	5	400	31.4	
Oil	24	200	33.1	
Pumped-storage Hydro	8	200	33.1	(2)
Comb. Turbine	32	50	60.7	

(1) 1979 Dollars derived from Reference 6.

(2) Equivalent incremental fuel cost based on energy production limitations.

(3) Cycling plant to be modeled.

to reflect the cost of high efficiency units with higher capacity factors. The capacity penalty for the higher base heat rate of the high back pressure turbine systems is also charged at a rate of $208/Kw.

The values of the general economic parameters assumed for the case studies are as follows:

1. Annual fixed charge rate = 18 percent
2. Interest rate = 9 percent
3. Real fuel escalation rate = 4 percent
4. Plant lifetime = 40 years

Table 4.2

Arizona Utility System

Unit Type	Number of Units	Unit Size	Incremental Fuel Cost (Mills/KwHr (1)
Hydro	1	350	0.0
Pumped-storage hydro	1	20	0.0
Nuclear	1	1000	7.0
Coal	2	1000	7.3
Coal	1	800	7.3
Coal	1	600	7.4
Coal	1	500	7.45
Coal (3)	1	500	7.5
Coal	1	400	7.8
Coal	1	200	7.8
Hydro	1	500	7.8
Oil	2	400	33.1
Oil	7	200	33.1
Hydro	1	500	33.1
Pumped-storage hydro	1	50	33.1
Comb. Turb.	12	50	46.1

(1) 1979 Dollars derived from Reference 6.

4.3 Case Study Results

4.3.1 General

A total of 24 design optimization case studies of wet, dry, and wet/dry cooling have been completed. The following sections present results of these design optimizations along with the results of a number of supporting analyses. The detailed results obtained for each of the 24 design optimizations using the WDCSIM II Program are presented in Appendix A. In the following sections, individual case study results are referred to by the "System" numbers of Appendix A as listed in Table 4.3

Generally, the reported optimum designs have been determined within the following limits;

1. Number of dry cooling modules $\pm$ 5 percent.
2. Number of wet cooling modules $\pm$ 5 percent.
3. Circulating water flow rate $\pm$ 10 percent.

Actual optimum total costs have been determined to within one percent of the true optimum as required. The approximate $\pm$ 10 percent convergence on the optimum circulating water flow rate has been found to be adequate because of the relative insensitivity of the total cost to this parameter.

4.3.2 Baseline System

A number of design optimizations have been performed for all-wet and all-dry cooling systems in order to establish a reference point for results of the wet/dry analyses. A summary of these results for the Boston and Phoenix sites are presented in Tables 4.4 and 4.5, respectively.

Table 4.3

Summary of Design Optimization

System No.	Sites	Water Availability (acre-feet)	Operating Mode	Turbine Type	Cost of Heat Rejection (mills/KwHr)
1	Boston	unlimited	Base-load	Conv.	3.04
2	Boston	unlimited	Cycling	Conv.	4.97
3	Boston	0	Base-load	E.R.Conv.	7.34
4	Boston	0	Cycling	E.R.Conv.	12.40
5	Boston	0	Base-load	HBP	8.26
6	Boston	0	Cycling	HBP	13.10
7	Phoenix	unlimited	Base-load	Conv.	2.76
8	Phoenix	unlimited	Cycling	Conv.	4.70
9	Phoenix	0	Base-load	E.R.Conv.	9.53
10	Phoenix	0	Cycling	E.R.Conv.	16.80
11	Phoenix	0	Base-load	HBP	7.25
12	Phoenix	0	Cycling	HBP	11.90
13	Phoenix	250	Base-load	E.R.Conv.	7.98
14	Phoenix	250	Cycling	E.R.Conv.	13.80
15	Phoenix	1000	Base-load	E.R.Conv.	6.41
16	Phoenix	1000	Cycling	E.R.Conv.	10.20
17	Boston	250	Base-load	E.R.Conv.	6.57
18	Boston	250	Cycling	E.R.Conv.	10.70
19	Boston	1000	Base-load	E.R.Conv.	5.61
20	Boston	1000	Cycling	E.R.Conv.	8.34
21	Phoenix	250	Base-load	HBP	6.61
22	Phoenix	250	Cycling	HBP	10.60
23	Phoenix	1000	Base-load	HBP	5.83
24	Phoenix	1000	Cycling	HBP	8.88

Turbine Type

Conv. = Conventional with 5" Hg. Back Pressure Limit.
E.R. Conv. = Extended range conventional with 8" Hg limit.
HBP = High back pressure design.

Table 4.4

Reference Optimum All Wet and Dry Systems
for Boston

	System Number 1	2	3	4	5	6
Cooling System Type	Wet	Wet	Dry	Dry	Dry	Dry
Plant Operating Mode	Base-load	Cycling	Base-load	Cycling	Base-load	Cycling
Turbine Type	Conv.	Conv.	E.R.Conv.	E.R.Conv.	HBP	HBP
Water Consumption (acre-feet)	4230	2250	0	0	0	0
Optimum Design						
#Dry Modules	0	0	80	80	50	50
#Wet Modules	11	11	0	0	0	0
Condenser Flow (cfs)	550	550	600	600	400	400
Present Worth Total Cost ($X 10^6)	108.0	94.9	260.3	236.6	293.9	250.0
Incremental Power Production Cost (mills/KwHr)	3.04	4.97	7.34	12.4	8.26	13.1

Table 4.5

Reference Optimum All Wet and All Dry System
for Phoenix

	System Number 7	8	9	10	11	11
Cooling System Type	Wet	Wet	Dry	Dry	Dry	Dry
Plant Operating Mode	Base-load	Cycling	Base-load	Cycling	Base-load	Cycling
Turbine Type	Conv.	Conv.	E.R.Conv.	E.R.Conv.	HBP	HBP
Water Consumption (arce-feet)	5040	2895	0	0	0	0
Optimum Design						
#Dry Modules	0	0	120	120	60	60
#Wet Modules	10	10	0	0	0	0
Condenser Flow	550	550	700	700	500	500
Present Worth Total Cost (X10^6$)	98.1	91.8	338.2	326.4	257.3	233.2
Incremental Power Production Cost (mills/KwHr)	2.76	4.70	9.53	16.8	7.25	11.9

The important conclusion suggested by the results in Tables 4.4 and 4.5 is that the optimum design of all-wet and all-dry cooling systems is independent of the assumed mode of plant operation. Assumption of either base-load operation or economically dispatched cycling operation for otherwise identical design situations has been found to lead to the specification of the same cooling system design (within the limits of the design optimization convergence).

For the all-wet cooling systems, this result can be attributed to the fact that the cost of wet cooling capacity is relatively low. The lower operating and energy penalties for the cycling mode of operation suggest the possibility of a justified reduction in capital costs. However, the computed results show that, for such a reduction, the relatively low cost of wet cooling capacity does not result in capital cost savings which are greater than the additional penalty costs.

For the case of the all-dry systems, the situation with respect to the impact of the assumed operating mode is similar. For each of the cases of dry cooling, the optimum design for base-load mode of operation results in the maximum condensing pressure (at full thermal power) at the maximum ambient. Reduction in the size of the dry cooling capacity, as might be suggested by the lower operating penalties of cycling operation, would therefore lead to necessary reductions in thermal power at the maximum ambient condition. The capacity penalty experienced with a reduced size design is clearly larger than the capital cost savings and thus a reduction in size with cycling operation is not justified.

Also note that, with all-dry cooling, the optimum turbine design for Boston is different from that which is optimum for Phoenix. For Boston, lowest costs are obtained with the extended-range conventional turbine for both cycling and base-load assumed modes of operation. For Phoenix, lowest costs are obtained with the high-back-pressure turbine for both the cycling and base-load modes of operation. This result can be attributed to the much higher fuel costs and lower annual average and lower maximum ambient temperature in the Boston region. The annual average incremental fuel cost for the Boston region utility is 32.4 mills/KwHr while the average for Phoenix is 17.6 mills/KwHr. The maximum ambient dry bulb temperature for Phoenix is 13°F higher and the annual average ambient temperature is 17°F higher. For the Boston site, the higher fuel cost causes the high back pressure design, with its lower base efficiency, to lead to very high energy replacement costs. The comparatively lower ambient temperatures of Boston reduces the dry cooling capacity needed to maintain the 8 in. Hg (153°F) condenser pressure limit. In Figure 4.8 is shown the duration of operation of the extended range conventional turbines above the conventional back pressure limit of 5 in. Hg for the dry-cooled Boston plant.

The last two lines of Tables 4.4 and 4.5 present the total present worth costs of owning and operating the heat rejection system over its lifetime of 40 years and the lifetime average incremental cost of power production due to heat rejection as defined in Equations 2.1 and 2.14, respectively. Generally, the present worth cost of heat rejection is about 10 to 15 percent less for cycling operation of the plant. However, since the energy production of the cycling plants (40 percent capacity factor) is nearly 50 percent less than the comparable base-load plants (75 percent capacity factor) the incremental cost of power production due to heat rejection for the cycling plants is large. Generally, the incremental cost (mills/KwHr) is 60 to 70 percent higher for the cycling plants. The increase in cost has important implications with respect to the value or breakever cost of consumable water as discussed in the next section.

4.3.3 Wet/Dry Cooling with Cycling Plants

Optimum wet/dry cooling systems have been determined for different amounts of available water. Optimum systems for use with both the extended-range conventional turbine and the high back pressure turbine have been determined for the Phoenix sited plant. Presented in Table 4.6 are the results for the Boston site and the results for Phoenix site are

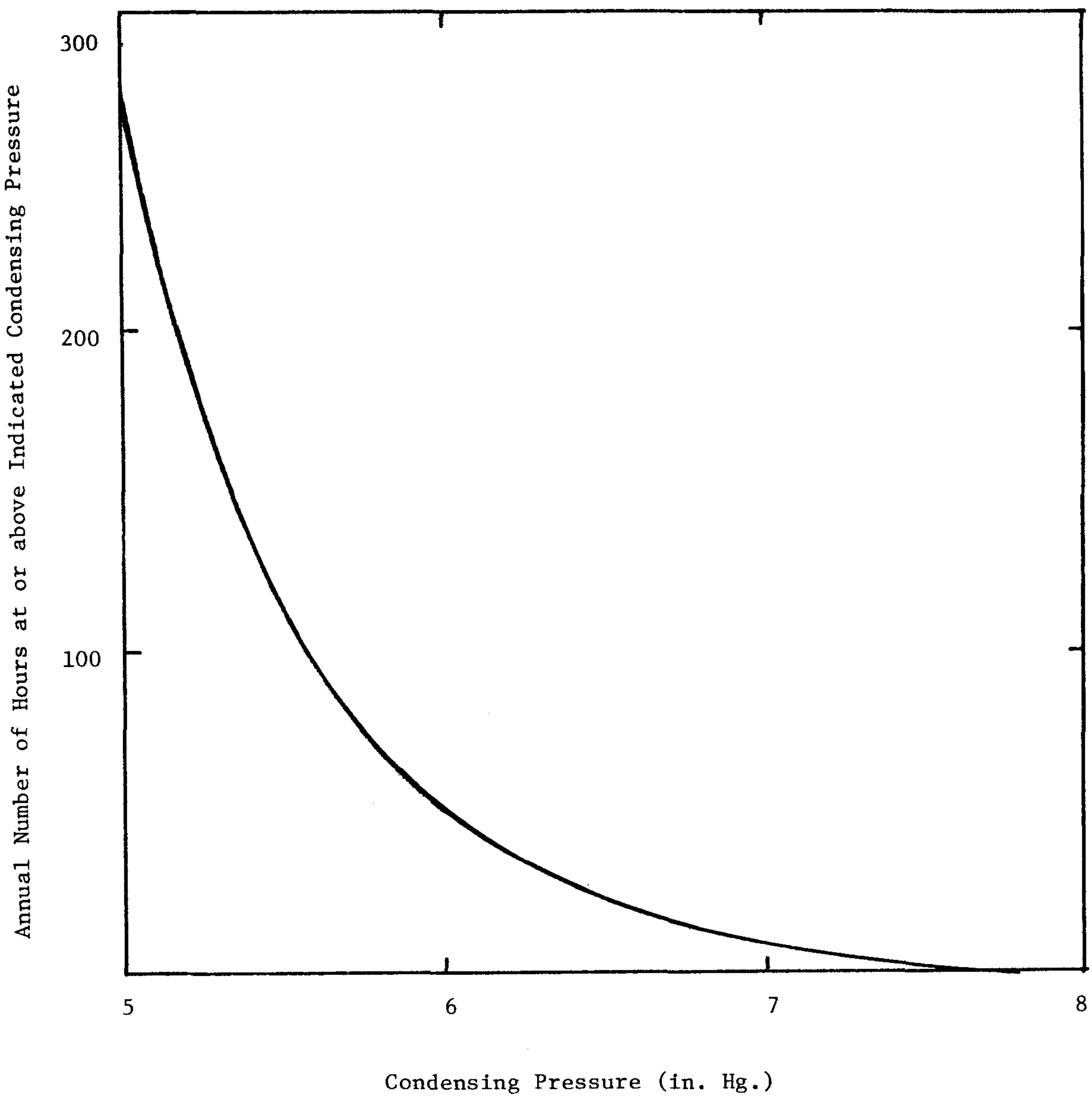


Figure 4.8 Duration of Condenser Operation Above 5 in. Hg. for Boston Dry-cooled Plant

presented in Tables 4.7 and 4.8. In each of these tables, optimum designs for annual water availabilities of 250 and 1000 acre-feet are presented. These amounts of available water correspond to approximately 5 percent and 20 percent annual average wet cooling for a base-load plant or 10 percent and 40 percent of the annual makeup requirements of the case-study cycling plant.

Important observations which can be made with respect to results presented in Tables 4.6 through 4.8 are:

1. For small amounts of water consumption (about 250 a.f./year or less) the assumption of base-load or cycling operation leads to similar wet/dry system designs.

2. Optimum designs for cycling operation at moderate levels of water consumption (about 1000 a.f.) specify significantly less dry cooling capacity than would be specified with assumption of base-load operation.

3. For the Phoenix plant, the high back pressure turbine option remains the economic choice even for case of 1000 a.f./year available water.

4. For all the cases examined, the cost of wet/dry cooling per unit power production with cycling operation is 50 to 70 percent higher than for base-load operation.

The first result listed above can be attributed to the fact that, with small amounts of water consumption, the effect of the water use is mainly to minimize the amount of dry cooling capacity needed to maintain the condenser pressure limit. There is not sufficient water to significantly impact on the energy penalty - the major determinant for the size of the dry cooling capacity in wet/dry systems. With moderate amounts of water consumption (about 35 percent of that needed for all wet cycling plant), the use of evaporative cooling becomes important with respect to the magnitude of the

Table 4.6

Optimum Wet/Dry Cooling Systems for Boston Plant

(All use extended-range Conventional Turbine)	System Number 17	18	19	20
Plant Operating Mode	Base-load	Cycling	Base-load	Cycling
Water Consumption (acre-feet)	250	250	1000	1000
Optimum Design				
#Dry Modules	65	60	50	37
#Wet Modules	5	5	8	9
Circulating Water Flow (CFS)	450	450	400	400
Fraction to Wet Towers	.47	.47	.85	.95
Present Worth Cost ($\$X10^6$)				
Capital	110.7	105.1	97.2	85.4
Operating	32.5	19.0	29.3	17.0
Loss of Energy Production	44.7	32.6	34.7	21.6
Loss of Capacity	45.1	44.6	37.9	34.9
Total	233.1	204.4	199.2	158.9
Incremental Power Production	6.57	10.7	5.61	8.34

Table 4.7

Optimum Wet/Dry Cooling Systems for Phoenix Plant

(all use extended-range conventional turbine)	System Number 13	14	15	16
Plant Operating Mode	Base-load	Cycling	Base-load	Cycling
Water Consumption (acre-feet)	250	250	1000	1000
Optimum Design				
#Dry Modules	75	70	55	35
#Wet Modules	6	6	9	10
Circulating Water Flow (CFS)	500	500	425	425
Fraction to Wet Towers	.51	.51	.90	1.00
Present Worth Cost ($\$X10^6$)				
Capital	125.3	119.6	106.1	86.6
Operating	21.2	13.7	17.6	10.2
Loss of Energy Production	87.8	86.4	64.4	65.9
Loss of Capacity	48.8	47.8	39.1	34.6
Total	283.2	267.7	227.2	197.4
Incremental Power Production Cost (mills/KwHr)	7.98	13.8	6.41	10.2

Table 4.8

Optimum Wet/Dry Cooling Systems for Phoenix Plant
Using High Back Pressure Turbine

	System Number 21	22	23	24
Plant Operating Mode	Base-load	Cycling	Base-load	Cycling
Water Consumption (acre-feet)	250	250	1000	1000
Optimum Design				
#Dry Modules	45	45	35	27
#Wet Modules	2	2	4	4
Circulating Water Flow (CFS)	400	400	300	300
Fraction to Wet Towers	.21	.21	.56	.56
Present Worth Cost ($\$X10^6$)				
Capital	79.2	79.2	67.7	58.8
Operating	14.9	9.9	17.2	7.39
Loss of Energy Production	98.5	76.0	92.0	72.8
Loss of Capacity	41.9	41.9	34.6	33.6
Total	234.5	207.1	206.5	172.6
Incremental Power Production Cost (mills/KwHr)	6.61	10.6	5.83	8.88

loss of energy production penalty. The available 1000 a.f./year of water contributes to a significantly greater fraction of the total cooling requirements of the cycling plant in comparison to a base-load plant, thereby reducing the required dry cooling capacity.

The economic advantage of the high back pressure turbine option for the Phoenix site for the situations of 250 and 1000 a.f./year water availability is again attributable to the relatively low cost of power for that utility system.

Importantly, the cost of wet/dry cooling with a a cycling plant in terms of power production cost is shown to be significantly higher than for a base-load plant. This directly indicates that the "break-even" cost for water for economically justifying the use of wet/dry cooling is much greater for cycling plant compared to a similar base-load plant. Figure 4.9 graphically presents the economics of water conservation for the Boston siting situation. The incremental cost for water conservation for different amounts of water conservation is presented along with the break-even cost of water for all dry cooling. The costs given in Figure 4.9 represent the equivalent uniform annual cost of water for the 40 year life of the plant. Each of these costs can be intrepreted as the value of water to the utility system. With the value of water to the cycling plant being 50 percent or more higher than it is to a similar base-load plant, there is a proportionately greater incentive for the utility to secure the needed water supply for the all-wet cooling of plants intended for cycling operation.

4.3.4 Impact of Off-Design Capacity Factor

Discussion with various utilities currently installing steam-electric plants with cycling capability has revealed that, in many cases, there is significant uncertainty with regard to expected load duty of a specific plant. This raises the question of the economic and water consumption impact of having to operate a wet/dry plant at a capacity factor which is significantly different from that used to optimize the wet/dry cooling system.

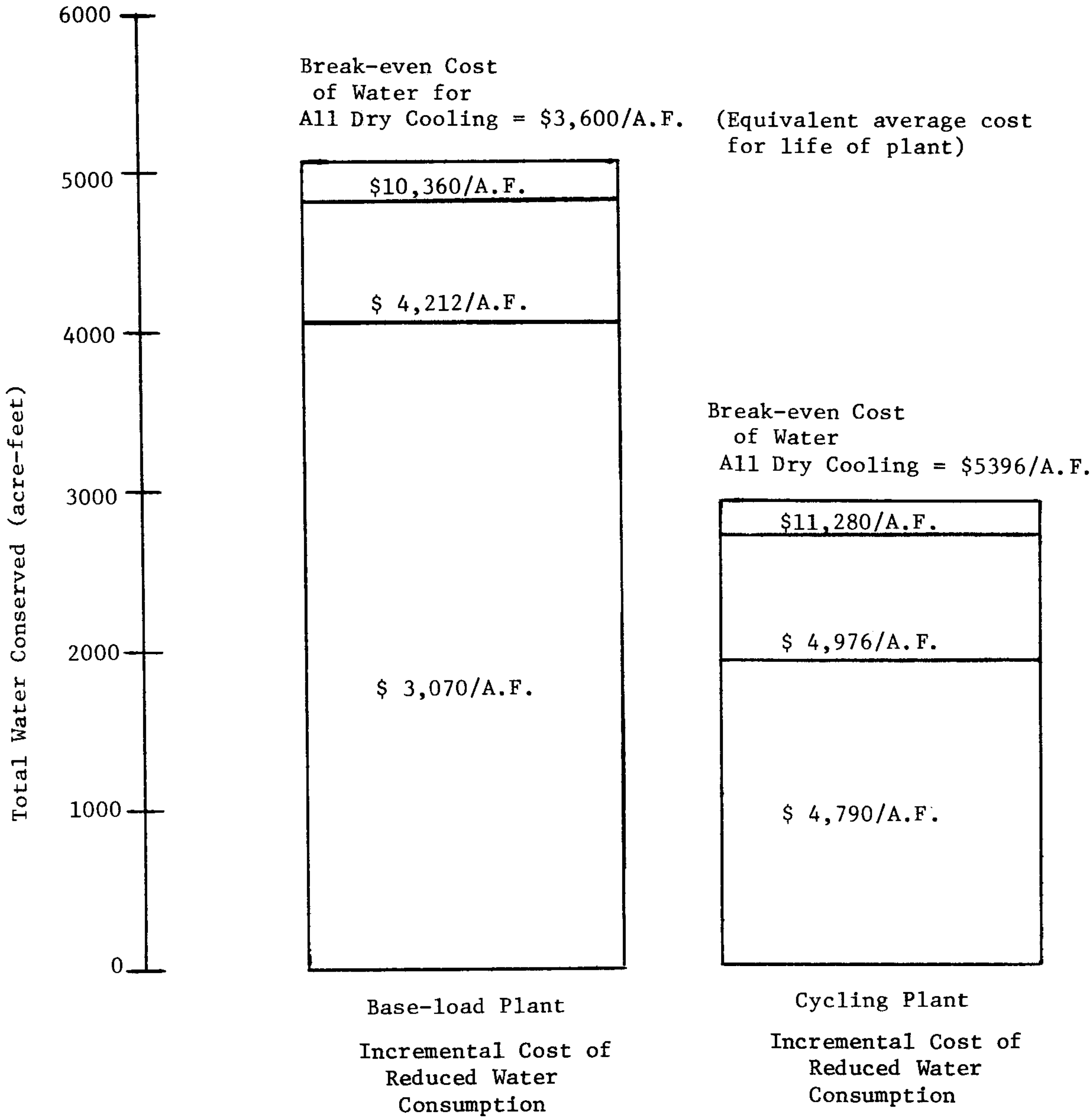


Figure 4.9 Cost of Water Conservation for Base-load and Cycling Plants

Two situations can be envisioned. One is that the wet/dry system is designed assuming base-load operation and later operated with a low capacity factor. The other situation is that the wet/dry system is designed assuming a low capacity factor, but once on-line the plant must operate with a base-load capacity factor. In the first situation, the economic penalty (if any) would be in capital cost of the oversized cooling system. In the second case, the penalty (if any) would be in increased operational penalties.

Clearly for the cases of all-wet, all-dry and wet/dry systems with small amounts of water consumption the economic impact of off-design operation is zero or minimal - the optimum designs for cycling and base-load are, or are nearly, identical. The largest impact is for the cases where there is a significant difference between the design which is optimum for a low capacity and the design which is optimum for a high capacity factor - such as the designs for the case of 1000 a.f./year of available water.

Table 4.9 summarizes the results of the evaluation of total cost penalty for Systems # 19, 20, 23, and 24 for operation at an off-design capacity factor. The penalties are larger for the Boston plant because of sensitivity of the performance of extended-range conventional turbine to an undersized cooling system and the high cost of replacement power. For the conservative design assumption of base-load operation for the Boston plant the penalty is \$14.9 million if the actual capacity factor is 40 percent. This is about 10 percent of the total system cost. For the Phoenix site, making the conservative assumption of base-load operation leads to a maximum penalty for off-design operation of \$1.5 million or about 1 percent of the cooling system cost.

Table 4.9

Economic Penalties for Operation at Off-Design Capacity Factor

	System Number	Design Capacity Factor	Actual Capacity Factor	Penalty for Incorrect Design Assumption	
				mills/KwHr	Present Worth ($\$ \times 10^6$)
Boston Plant	19	75% (base-load)	40% (cycling)	0.77	14.9
	20	40%	75%	0.87	30.8
Phoenix Plant	23	75%	40%	0.07	1.5
	24	40%	75%	0.18	6.5

4.3.5 Impact of Load-weather-dispatch Correlation on Design Optimization

The WDSCIM II Program, in simulating the operation of a wet/dry system for a one year period, directly accounts for the time-correlation of the utility system electrical demand, the utility system incremental power production cost, the economic dispatch of the wet/dry plant, and the site meterology. The simulation technique accounts for the time-correlation of the operating parameters in an attempt to make a realistic estimate of the loss of energy production penalty. A simpler approach, based on earlier approaches to the economic evaluation of wet/dry cooling systems, would be: 1) to assume a constant incremental power production cost, 2) to assume the plant operates 8760 hours per year and, 3) to then compute the energy penalty by multiplying the specified capacity factor times the energy penalty computed for 8760 hours of operation.

For the two case study situations, this simpler approach leads to loss of energy production penalties which are significantly less than those computed using the simulation approach. For the optimum design system for the cycling Phoenix plant for 1000 acre-feet (System #24) the difference in the computed present worth energy penalty is \$25.7 million (\$72.8 million - \$47.1 million). Similar results have also been obtained for other Phoenix design situations. The difference for the Boston situations is somewhat less. This can be attributed to the fact that correlation between ambient temperature and the electrical load is not as strong in the Boston region as it is in the Phoenix region.

With respect to the specification of an optimum design, the impact of this difference in the evaluated loss of energy production penalty is on the sizing of the dry cooling section. Accounting for the correlation of load-weather-dispatch results in optimum dry cooling sections which are about 10 to 15 percent larger than those specified as optimum with the simpler approach to computing the energy penalty.

4.3.6 Water Storage Requirements for Wet/Dry Cooled Cycling Plants

A wet/dry cooling system designed and operated with a cycling plant will result in different water consumption schedules than experienced with a base-load plant even though the total annual consumption is identical. Figure 4.10 depicts the water consumption schedules for the month of July for wet/dry systems Number 19 and Number 20 as described in Table 4.6.

Generally, for the same total water consumption, a cycling plant would consume makeup water over a somewhat longer period during the warmer seasons than a comparable base-load plant. Thus depending on whether the water availability is highly seasonal (spring runoff) or continuous at a low rate of supply (wells) a cycling plant with wet/dry cooling may require more or less on-site water storage capacity. General quantitative conclusions are difficult. However, based on the results for both the Boston and Phoenix

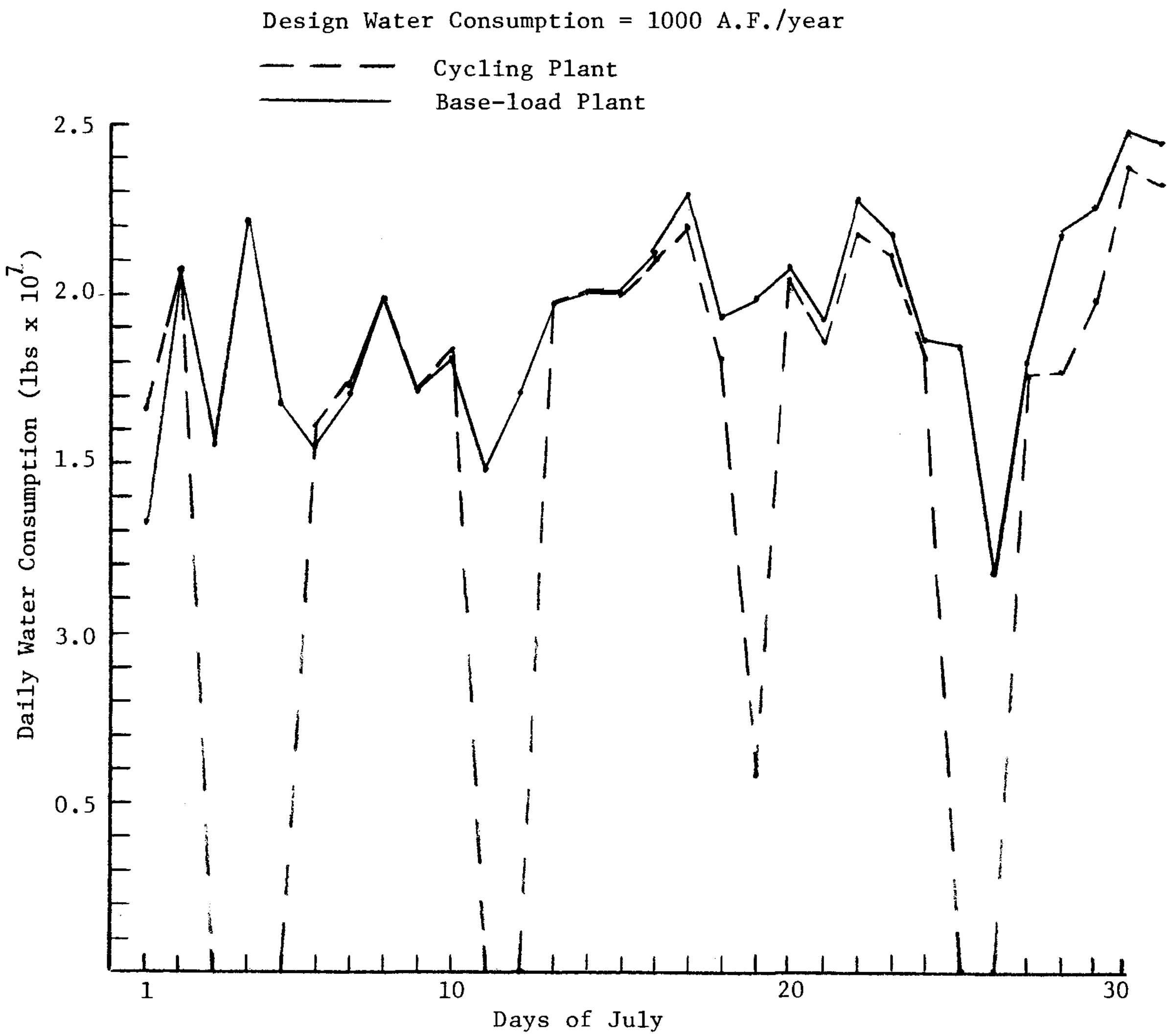


Figure 4.10 July Water Consumption Patterns of Wet/Dry Cycling and Base-load Plants Sited in Boston

sites, it appears that the difference between the optimum water storage capacity of a cycling plant and a base-load plant (of the same total annual consumption) would be only about 10 to 20 percent. Thus, the actual capacity factor of the plant would not appear to be critical to the sizing of a water impoundment for cooling tower makeup supply.

4.3.6 Wet/Dry System Control Capability Requirements

Earlier analysis of wet/dry cooling systems has shown that the control capability requirements and control strategies necessary for optimum water use are practical (Reference 1). This earlier work demonstrated that, for base-load plants, the wet cooling capacity should have the capability to be controlled (on-off) in increments no greater than one-third of the maximum capacity. Evaluation of the control capability requirements for wet/dry cooled planted operating in cycling duty indicates that a similar amount of control capability is desirable.

Section 5

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APPENDIX A

SYSTEM # 1

SITE: BOSTON
PLANT: 1129 MWT FOSSIL
OPERATING MODE: BASE-LOAD
TURBINE: CONVENTIONAL
AVAILABLE WATER: (UNLIMITED) ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	0	
2. NUMBER OF WET TOWER MODULES	11	
3. CIRCULATING WATER FLOW RATE	500	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	100%	
5. NUMBER OF WET TOWER CONTROL UNITS	1	
6. CONDENSER HEAT EXCHANGE AREA	286,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	3.00	IN. HG
8. MINIMUM PLANT CAPACITY	481.7	MW
9. PUMP AND FAN POWER	6.25	MW
10. ANNUAL ENERGY PRODUCTION LOSS	43,300	MWHRS
11. ANNUAL FAN AND PUMP POWER	50,600	MWHRS
12. CAPACITY FACTOR	75	%

CAPITAL AND OPERATING COST($X10^6$)		
1. DRY TOWERS AND FOUNDATION	$ 0.00	
2. WET TOWERS AND FOUNDATION	$ 6.49	
3. CONDENSER	$ 5.95	
4. ELECTRICAL EQUIPMENT	$ 0.96	
5. STEEL PIPELINES	$ 1.84	
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.36	
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.95	
8. INDIRECT CAPITAL	$ 6.58	
9. TOTAL CAPITAL	$24.15	
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 1.44	
11. ANNUAL COST OF PUMP AND FAN POWER	$ 1.11	
12. PLANT FUEL COST	21.9	MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 46.7
OPERATING	$ 18.2
LOSS OF ENERGY PRODUCTION	$ 20.7
LOSS OF CAPACITY	$ 22.3
TOTAL	$ 108.0

COST OF HEAT REJECTION = 3.04 MILLS/KWHR

SYSTEM # 4

SITE: BOSTON
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: CYCLING
TURBINE: CONVENTIONAL
AVAILABLE WATER: (NONE) ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	80	
2. NUMBER OF WET TOWER MODULES	0	
3. CIRCULATING WATER FLOW RATE	600	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	N/A	
5. NUMBER OF WET TOWER CONTROL UNITS	N/A	
6. CONDENSER HEAT EXCHANGE AREA	298,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	7.23	IN. HG
8. MINIMUM PLANT CAPACITY	427.7	MW
9. PUMP AND FAN POWER	13.0	MW
10. ANNUAL ENERGY PRODUCTION LOSS	46,700	MWHRS
11. ANNUAL FAN AND PUMP POWER	53,400	MWHRS
12. CAPACITY FACTOR	40	%

CAPITAL AND OPERATING COST($X10^6$)	
1. DRY TOWERS AND FOUNDATION	$ 27.90
2. WET TOWERS AND FOUNDATION	$ 0.00
3. CONDENSER	$ 6.18
4. ELECTRICAL EQUIPMENT	$ 4.61
5. STEEL PIPELINES	$ 4.77
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.62
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.71
8. INDIRECT CAPITAL	$ 17.19
9. TOTAL CAPITAL	$ 62.99
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 2.22
11. ANNUAL COST OF PUMP AND FAN POWER	$ 1.17
12. PLANT FUEL COST	21.9 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 121.9
OPERATING	$ 22.8
LOSS OF ENERGY PRODUCTION	$ 31.8
LOSS OF CAPACITY	$ 59.9
TOTAL	$ 236.6

COST OF HEAT REJECTION = 12.4 MILLS/KWHR

SYSTEM # 5
SITE: BOSTON
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: BASE-LOAD
TURBINE: HIGH BACK PRESSURE
AVAILABLE WATER: (NONE) ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	50	
2. NUMBER OF WET TOWER MODULES	0	
3. CIRCULATING WATER FLOW RATE	400	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	N/A	
5. NUMBER OF WET TOWER CONTROL UNITS	N/A	
6. CONDENSER HEAT EXCHANGE AREA	248,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	13.6	IN. HG
8. MINIMUM PLANT CAPACITY	437.6	MW
9. PUMP AND FAN POWER	8.39	MW
10. ANNUAL ENERGY PRODUCTION LOSS	312,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	55,122	MWHRS
12. CAPACITY FACTOR	75	%

CAPITAL AND OPERATING COST ($X10^6$)	
1. DRY TOWERS AND FOUNDATION	$17.43
2. WET TOWERS AND FOUNDATION	$ 0.00
3. CONDENSER	$ 5.23
4. ELECTRICAL EQUIPMENT	$ 2.88
5. STEEL PIPELINES	$ 2.82
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.42
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.16
8. INDIRECT CAPITAL	$11.24
9. TOTAL CAPITAL	$41.21
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$10.27
11. ANNUAL COST OF PUMP AND FAN POWER	$ 2.13
12. PLANT FUEL COST	23.6 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 79.8
OPERATING	$ 32.2
LOSS OF ENERGY PRODUCTION	$135.8
LOSS OF CAPACITY	$ 45.1
TOTAL	$292.9

COST OF HEAT REJECTION = 8.26 MILLS/KWHR

SYSTEM # 6

SITE: BOSTON
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: CYCLING
TURBINE: HIGH BACK PRESSURE
AVAILABLE WATER: (NONE) ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	50	
2. NUMBER OF WET TOWER MODULES	0	
3. CIRCULATING WATER FLOW RATE	400	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	N/A	
5. NUMBER OF WET TOWER CONTROL UNITS	N/A	
6. CONDENSER HEAT EXCHANGE AREA	248,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	13.6	IN. HG
8. MINIMUM PLANT CAPACITY	437.6	MW
9. PUMP AND FAN POWER	8.39	MW
10. ANNUAL ENERGY PRODUCTION LOSS	166,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	29,400	MWHRS
12. CAPACITY FACTOR	40	%

CAPITAL AND OPERATING COST($X10^6$)	
1. DRY TOWERS AND FOUNDATION	$ 17.43
2. WET TOWERS AND FOUNDATION	$ 0.00
3. CONDENSER	$ 5.23
4. ELECTRICAL EQUIPMENT	$ 2.88
5. STEEL PIPELINES	$ 2.82
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.42
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.16
8. INDIRECT CAPITAL	$ 11.24
9. TOTAL CAPITAL	$ 41.21
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 7.57
11. ANNUAL COST OF PUMP AND FAN POWER	$ 1.06
12. PLANT FUEL COST	23.6 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 79.8
OPERATING	$ 18.8
LOSS OF ENERGY PRODUCTION	$106.1
LOSS OF CAPACITY	$ 45.1
TOTAL	$250.0

COST OF HEAT REJECTION = 13.1 MILLS/KWHR

SYSTEM # 7

SITE: PHOENIX
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: BASE-LOAD
TURBINE: CONVENTIONAL
AVAILABLE WATER: (UNLIMITED) ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	0	
2. NUMBER OF WET TOWER MODULES	10	
3. CIRCULATING WATER FLOW RATE	550	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	100%	
5. NUMBER OF WET TOWER CONTROL UNITS	1	
6. CONDENSER HEAT EXCHANGE AREA	286,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	2.8	IN. HG
8. MINIMUM PLANT CAPACITY	483	MW
9. PUMP AND FAN POWER	6.10	MW
10. ANNUAL ENERGY PRODUCTION LOSS	65,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	40,000	MWHRS
12. CAPACITY FACTOR	75	%

CAPITAL AND OPERATING COST($X10^6$)	
1. DRY TOWERS AND FOUNDATION	$ 0.00
2. WET TOWERS AND FOUNDATION	$ 5.90
3. CONDENSER	$ 5.95
4. ELECTRICAL EQUIPMENT	$ 0.87
5. STEEL PIPELINES	$ 1.78
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.36
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.95
8. INDIRECT CAPITAL	$ 6.31
9. TOTAL CAPITAL	$ 23.1
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 1.52
11. ANNUAL COST OF PUMP AND FAN POWER	$ 0.72
12. PLANT FUEL COST	10.9 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 44.8
OPERATING	$ 11.7
LOSS OF ENERGY PRODUCTION	$ 20.1
LOSS OF CAPACITY	$ 21.3
TOTAL	$ 98.1

COST OF HEAT REJECTION = 2.76 MILLS/KWHR

SYSTEM # 8
SITE: PHOENIX
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: CYCLING
TURBINE: CONVENTIONAL
AVAILABLE WATER: (UNLIMITED) ACRE-FEET

DESIGN AND OPERATING PARAMETERS			CAPITAL AND OPERATING COST($X10^6$)	
1. NUMBER OF DRY TOWER MODULES	0		1. DRY TOWERS AND FOUNDATION	$ 0.00
2. NUMBER OF WET TOWER MODULES	10		2. WET TOWERS AND FOUNDATION	$ 5.90
3. CIRCULATING WATER FLOW RATE	550	FT^3/SEC	3. CONDENSER	$ 5.95
4. FRACTION OF FLOW TO WET TOWERS	100%		4. ELECTRICAL EQUIPMENT	$ 0.87
5. NUMBER OF WET TOWER CONTROL UNITS	1		5. STEEL PIPELINES	$ 1.78
6. CONDENSER HEAT EXCHANGE AREA	286,000	FT^2	6. CIRCULATING WATER PUMP STRUCTURE	$ 0.36
7. MAXIMUM CONDENSING PRESSURE	2.8	IN. HG	7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.95
8. MINIMUM PLANT CAPACITY	483	MW	8. INDIRECT CAPITAL	$ 6.31
9. PUMP AND FAN POWER	6.10	MW	9. TOTAL CAPITAL	$ 23.14
10. ANNUAL ENERGY PRODUCTION LOSS	44,400	MWHRS	10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 1.30
11. ANNUAL FAN AND PUMP POWER	21,900	MWHRS	11. ANNUAL COST OF PUMP AND FAN POWER	$ 0.36
12. CAPACITY FACTOR	41	%	12. PLANT FUEL COST	10.9 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 44.8
OPERATING	$ 7.3
LOSS OF ENERGY PRODUCTION	$ 18.2
LOSS OF CAPACITY	$ 21.4
TOTAL	$ 91.8

COST OF HEAT REJECTION = 4.70 MILLS/KWHR

SYSTEM # 9

SITE: PHOENIX
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: BASE-LOAD
TURBINE: CONVENTIONAL
AVAILABLE WATER: (NONE) ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	120	
2. NUMBER OF WET TOWER MODULES	0	
3. CIRCULATING WATER FLOW RATE	700	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	N/A	
5. NUMBER OF WET TOWER CONTROL UNITS	N/A	
6. CONDENSER HEAT EXCHANGE AREA	320,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	7.8	IN. HG
8. MINIMUM PLANT CAPACITY	420	MW
9. PUMP AND FAN POWER	17.9	MW
10. ANNUAL ENERGY PRODUCTION LOSS	135,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	117,000	MWHRS
12. CAPACITY FACTOR	75	%

CAPITAL AND OPERATING COST($X10^6$)	
1. DRY TOWERS AND FOUNDATION	$ 41.85
2. WET TOWERS AND FOUNDATION	$ 0.00
3. CONDENSER	$ 6.63
4. ELECTRICAL EQUIPMENT	$ 6.92
5. STEEL PIPELINES	$ 6.35
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.68
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.88
8. INDIRECT CAPITAL	$ 24.12
9. TOTAL CAPITAL	$ 88.44
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 4.28
11. ANNUAL COST OF PUMP AND FAN POWER	$ 2.07
12. PLANT FUEL COST	10.9 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 171.2
OPERATING	$ 36.0
LOSS OF ENERGY PRODUCTION	$ 56.7
LOSS OF CAPACITY	$ 74.3
TOTAL	$ 338.2

COST OF HEAT REJECTION = 9.53 MILLS/KWHR

SYSTEM # 10

SITE: PHOENIX
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: CYCLING
TURBINE: CONVENTIONAL
AVAILABLE WATER: (NONE) ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	120	
2. NUMBER OF WET TOWER MODULES	0	
3. CIRCULATING WATER FLOW RATE	700	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	N/A	
5. NUMBER OF WET TOWER CONTROL UNITS	N/A	
6. CONDENSER HEAT EXCHANGE AREA	320,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	7.8	IN. HG
8. MINIMUM PLANT CAPACITY	420	MW
9. PUMP AND FAN POWER	17.9	MW
10. ANNUAL ENERGY PRODUCTION LOSS	118,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	64,000	MWHRS
12. CAPACITY FACTOR	41	%

CAPITAL AND OPERATING COST($X10^6$)	
1. DRY TOWERS AND FOUNDATION	$ 41.85
2. WET TOWERS AND FOUNDATION	$ 0.00
3. CONDENSER	$ 6.63
4. ELECTRICAL EQUIPMENT	$ 6.92
5. STEEL PIPELINES	$ 6.35
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.68
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.88
8. INDIRECT CAPITAL	$ 24.12
9. TOTAL CAPITAL	$ 88.44
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 4.09
11. ANNUAL COST OF PUMP AND FAN POWER	$ 1.06
12. PLANT FUEL COST	10.9 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 171.2
OPERATING	$ 23.4
LOSS OF ENERGY PRODUCTION	$ 57.3
LOSS OF CAPACITY	$ 74.2
TOTAL	$ 326.4

COST OF HEAT REJECTION = 16.8 MILLS/KWHR

SYSTEM # 11

SITE: PHOENIX
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: BASE-LOAD
TURBINE: HIGH BACK PRESSURE
AVAILABLE WATER: (NONE) ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	60	
2. NUMBER OF WET TOWER MODULES	0	
3. CIRCULATING WATER FLOW RATE	500	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	N/A	
5. NUMBER OF WET TOWER CONTROL UNITS	N/A	
6. CONDENSER HEAT EXCHANGE AREA	278,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	14.0	IN. HG
8. MINIMUM PLANT CAPACITY	430	MW
9. PUMP AND FAN POWER	10.2	MW
10. ANNUAL ENERGY PRODUCTION LOSS	330,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	67,000	MWHRS
12. CAPACITY FACTOR	75	%

CAPITAL AND OPERATING COST($X10^6$)	
1. DRY TOWERS AND FOUNDATION	$ 20.92
2. WET TOWERS AND FOUNDATION	$ 0.00
3. CONDENSER	$ 5.78
4. ELECTRICAL EQUIPMENT	$ 3.46
5. STEEL PIPELINES	$ 3.64
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.53
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.47
8. INDIRECT CAPITAL	$ 13.43
9. TOTAL CAPITAL	$ 49.26
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 6.60
11. ANNUAL COST OF PUMP AND FAN POWER	$ 1.30
12. PLANT FUEL COST	11.8 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 95.4
OPERATING	$ 22.0
LOSS OF ENERGY PRODUCTION	$ 87.4
LOSS OF CAPACITY	$ 52.5
TOTAL	$257.3

COST OF HEAT REJECTION = 7.25 MILLS/KWHR

SYSTEM # 12

SITE: PHOENIX
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: CYCLING
TURBINE: HIGH BACK PRESSURE
AVAILABLE WATER: (NONE) ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	60	
2. NUMBER OF WET TOWER MODULES	0	
3. CIRCULATING WATER FLOW RATE	500	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	N/A	
5. NUMBER OF WET TOWER CONTROL UNITS	N/A	
6. CONDENSER HEAT EXCHANGE AREA	278,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	14.0	IN. HG
8. MINIMUM PLANT CAPACITY	430	MW
9. PUMP AND FAN POWER	10.2	MW
10. ANNUAL ENERGY PRODUCTION LOSS	189,900	MWHRS
11. ANNUAL FAN AND PUMP POWER	36,600	MWHRS
12. CAPACITY FACTOR	41	%

CAPITAL AND OPERATING COST($X10^6$)	
1. DRY TOWERS AND FOUNDATION	$20.92
2. WET TOWERS AND FOUNDATION	$ 0.00
3. CONDENSER	$ 5.78
4. ELECTRICAL EQUIPMENT	$ 3.46
5. STEEL PIPELINES	$ 3.64
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.53
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.47
8. INDIRECT CAPITAL	$13.43
9. TOTAL CAPITAL	$49.26
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 5.08
11. ANNUAL COST OF PUMP AND FAN POWER	$ 0.66
12. PLANT FUEL COST	11.8 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 95.4
OPERATING	$ 14.0
LOSS OF ENERGY PRODUCTION	$ 71.3
LOSS OF CAPACITY	$ 52.4
TOTAL	$233.2

COST OF HEAT REJECTION = 11.9 MILLS/KWHR

SYSTEM # 13

SITE: PHOENIX
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: BASE-LOAD
TURBINE: CONVENTIONAL
AVAILABLE WATER: 250 ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	75	
2. NUMBER OF WET TOWER MODULES	6	
3. CIRCULATING WATER FLOW RATE	500	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	51%	
5. NUMBER OF WET TOWER CONTROL UNITS	3	
6. CONDENSER HEAT EXCHANGE AREA	273,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	4.8	IN. HG
8. MINIMUM PLANT CAPACITY	457.7	MW
9. PUMP AND FAN POWER	13.6	MW
10. ANNUAL ENERGY PRODUCTION LOSS	251,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	71,200	MWHRS
12. CAPACITY FACTOR	75	%

CAPITAL AND OPERATING COST($X10^6$)		
1. DRY TOWERS AND FOUNDATION	$26.15	
2. WET TOWERS AND FOUNDATION	$ 3.54	
3. CONDENSER	$ 5.70	
4. ELECTRICAL EQUIPMENT	$ 4.80	
5. STEEL PIPELINES	$ 4.50	
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.50	
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.82	
8. INDIRECT CAPITAL	$17.65	
9. TOTAL CAPITAL	$64.74	
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 6.12	
11. ANNUAL COST OF PUMP AND FAN POWER	$ 1.04	
12. PLANT FUEL COST	10.9	MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 125.3
OPERATING	$ 21.2
LOSS OF ENERGY PRODUCTION	$ 87.8
LOSS OF CAPACITY	$ 48.8
TOTAL	$ 283.2

COST OF HEAT REJECTION = 7.98 MILLS/KWHR

SYSTEM # 14

SITE: PHOENIX
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: CYCLING
TURBINE: CONVENTIONAL
AVAILABLE WATER: 250 ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	70	
2. NUMBER OF WET TOWER MODULES	6	
3. CIRCULATING WATER FLOW RATE	500	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	51%	
5. NUMBER OF WET TOWER CONTROL UNITS	3	
6. CONDENSER HEAT EXCHANGE AREA	272,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	4.9	IN. HG
8. MINIMUM PLANT CAPACITY	457.3	MW
9. PUMP AND FAN POWER	13.0	MW
10. ANNUAL ENERGY PRODUCTION LOSS	217,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	39,600	MWHRS
12. CAPACITY FACTOR	41	%

CAPITAL AND OPERATING COST($X10^6$)		
1. DRY TOWERS AND FOUNDATION	$ 24.41	
2. WET TOWERS AND FOUNDATION	$ 3.54	
3. CONDENSER	$ 5.70	
4. ELECTRICAL EQUIPMENT	$ 4.56	
5. STEEL PIPELINES	$ 4.36	
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.51	
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.84	
8. INDIRECT CAPITAL	$ 16.85	
9. TOTAL CAPITAL	$ 61.80	
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 6.03	
11. ANNUAL COST OF PUMP AND FAN POWER	$ 0.54	
12. PLANT FUEL COST	10.9	MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 119.6
OPERATING	$ 13.7
LOSS OF ENERGY PRODUCTION	$ 86.4
LOSS OF CAPACITY	$ 47.8
TOTAL	$ 267.7

COST OF HEAT REJECTION = 13.8 MILLS/KWHR

SYSTEM # 15

SITE: PHOENIX
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: BASE-LOAD
TURBINE: CONVENTIONAL
AVAILABLE WATER: 1000 ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	55	
2. NUMBER OF WET TOWER MODULES	9	
3. CIRCULATING WATER FLOW RATE	425	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	90%	
5. NUMBER OF WET TOWER CONTROL UNITS	3	
6. CONDENSER HEAT EXCHANGE AREA	251,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	3.6	IN. HG
8. MINIMUM PLANT CAPACITY	472	MW
9. PUMP AND FAN POWER	11.9	MW
10. ANNUAL ENERGY PRODUCTION LOSS	347,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	59,000	MWHRS
12. CAPACITY FACTOR	75	%

CAPITAL AND OPERATING COST($X10^6$)		
1. DRY TOWERS AND FOUNDATION	$19.18	
2. WET TOWERS AND FOUNDATION	$ 5.31	
3. CONDENSER	$ 5.30	
4. ELECTRICAL EQUIPMENT	$ 3.96	
5. STEEL PIPELINES	$ 3.77	
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.44	
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.88	
8. INDIRECT CAPITAL	$14.9	
9. TOTAL CAPITAL	$54.8	
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 4.49	
11. ANNUAL COST OF PUMP AND FAN POWER	$ 0.86	
12. PLANT FUEL COST	10.9	MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 106.1
OPERATING	$ 17.6
LOSS OF ENERGY PRODUCTION	$ 64.4
LOSS OF CAPACITY	$ 39.1
TOTAL	$ 227.2

COST OF HEAT REJECTION = 6.41 MILLS/KWHR

SYSTEM # 16

SITE: PHOENIX
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: CYCLING
TURBINE: CONVENTIONAL
AVAILABLE WATER: 1000 ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	35	
2. NUMBER OF WET TOWER MODULES	10	
3. CIRCULATING WATER FLOW RATE	425	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	100%	
5. NUMBER OF WET TOWER CONTROL UNITS	3	
6. CONDENSER HEAT EXCHANGE AREA	251,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	3.3	IN. HG
8. MINIMUM PLANT CAPACITY	476.4	MW
9. PUMP AND FAN POWER	10.6	MW
10. ANNUAL ENERGY PRODUCTION LOSS	641,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	37,400	MWHRS
12. CAPACITY FACTOR	41	%

CAPITAL AND OPERATING COST($X10^6$)	
1. DRY TOWERS AND FOUNDATION	$ 12.20
2. WET TOWERS AND FOUNDATION	$ 5.90
3. CONDENSER	$ 5.30
4. ELECTRICAL EQUIPMENT	$ 2.89
5. STEEL PIPELINES	$ 3.46
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.54
7. CIRCULATING WATER PUMPS AND MOTORS	$ 2.22
8. INDIRECT CAPITAL	$ 12.20
9. TOTAL CAPITAL	$ 44.73
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 4.60
11. ANNUAL COST OF PUMP AND FAN POWER	$ 0.41
12. PLANT FUEL COST	10.9 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 86.6
OPERATING	$ 10.2
LOSS OF ENERGY PRODUCTION	$ 65.9
LOSS OF CAPACITY	$ 34.6
TOTAL	$ 197.4

COST OF HEAT REJECTION = 10.2 MILLS/KWHR

SYSTEM # 17

SITE: BOSTON
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: BASE-LOAD
TURBINE: CONVENTIONAL
AVAILABLE WATER: 250 ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	65	
2. NUMBER OF WET TOWER MODULES	5	
3. CIRCULATING WATER FLOW RATE	450	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	47%	
5. NUMBER OF WET TOWER CONTROL UNITS	3	
6. CONDENSER HEAT EXCHANGE AREA	266,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	4.8	IN. HG
8. MINIMUM PLANT CAPACITY	458.6	MW
9. PUMP AND FAN POWER	12.0	MW
10. ANNUAL ENERGY PRODUCTION LOSS	102,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	85,900	MWHRS
12. CAPACITY FACTOR	75	%

CAPITAL AND OPERATING COST($X10^6$)	
1. DRY TOWERS AND FOUNDATION	$ 22.6
2. WET TOWERS AND FOUNDATION	$ 2.95
3. CONDENSER	$ 5.57
4. ELECTRICAL EQUIPMENT	$ 4.18
5. STEEL PIPELINES	$ 4.02
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.48
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.71
8. INDIRECT CAPITAL	$ 15.6
9. TOTAL CAPITAL	$ 57.2
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 3.12
11. ANNUAL COST OF PUMP AND FAN POWER	$ 1.88
12. PLANT FUEL COST	21.9 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 110.7
OPERATING	$ 32.5
LOSS OF ENERGY PRODUCTION	$ 44.7
LOSS OF CAPACITY	$ 45.1
TOTAL	$ 233.2

COST OF HEAT REJECTION = 6.57 MILLS/KWHR

SYSTEM # 18
SITE: BOSTON
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: CYCLING
TURBINE: CONVENTIONAL
AVAILABLE WATER: 250 ACRE-FEET

DESIGN AND OPERATING PARAMETERS

1. NUMBER OF DRY TOWER MODULES	60	
2. NUMBER OF WET TOWER MODULES	5	
3. CIRCULATING WATER FLOW RATE	450	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	47%	
5. NUMBER OF WET TOWER CONTROL UNITS	3	
6. CONDENSER HEAT EXCHANGE AREA	266,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	4.9	IN. HG
8. MINIMUM PLANT CAPACITY	457.1	MW
9. PUMP AND FAN POWER	11.6	MW
10. ANNUAL ENERGY PRODUCTION LOSS	54,700	MWHRS
11. ANNUAL FAN AND PUMP POWER	43,900	MWHRS
12. CAPACITY FACTOR	40	%

CAPITAL AND OPERATING COST($X10^6$)

1. DRY TOWERS AND FOUNDATION	$ 20.02
2. WET TOWERS AND FOUNDATION	$ 2.95
3. CONDENSER	$ 5.57
4. ELECTRICAL EQUIPMENT	$ 3.90
5. STEEL PIPELINES	$ 3.88
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.49
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.74
8. INDIRECT CAPITAL	$ 14.8
9. TOTAL CAPITAL	$ 54.2
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 2.48
11. ANNUAL COST OF PUMP AND FAN POWER	$ 0.96
12. PLANT FUEL COST	21.9 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)

CAPITAL	$ 105.1
OPERATING	$ 19.0
LOSS OF ENERGY PRODUCTION	$ 35.6
LOSS OF CAPACITY	$ 44.6
TOTAL	$ 204.4

COST OF HEAT REJECTION = 10.7 MILLS/KWHR

SYSTEM # 19

SITE: BOSTON
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: BASE-LOAD
TURBINE: CONVENTIONAL
AVAILABLE WATER: 1000 ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	50	
2. NUMBER OF WET TOWER MODULES	8	
3. CIRCULATING WATER FLOW RATE	400	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	85%	
5. NUMBER OF WET TOWER CONTROL UNITS	3	
6. CONDENSER HEAT EXCHANGE AREA	243,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	4.0	IN. HG
8. MINIMUM PLANT CAPACITY	469.9	MW
9. PUMP AND FAN POWER	10.9	MW
10. ANNUAL ENERGY PRODUCTION LOSS	85,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	78,000	MWHRS
12. CAPACITY FACTOR	75	%

CAPITAL AND OPERATING COST($X10^6$)	
1. DRY TOWERS AND FOUNDATION	$ 17.43
2. WET TOWERS AND FOUNDATION	$ 4.72
3. CONDENSER	$ 5.16
4. ELECTRICAL EQUIPMENT	$ 3.58
5. STEEL PIPELINES	$ 3.43
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.42
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.74
8. INDIRECT CAPITAL	$ 13.6
9. TOTAL CAPITAL	$ 50.2
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 2.42
11. ANNUAL COST OF PUMP AND FAN POWER	$ 1.71
12. PLANT FUEL COST	21.9 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 97.2
OPERATING	$ 29.3
LOSS OF ENERGY PRODUCTION	$ 34.7
LOSS OF CAPACITY	$ 37.9
TOTAL	$ 199.2

COST OF HEAT REJECTION = 5.61 MILLS/KWHR

SYSTEM # 20

SITE:	BOSTON
PLANT:	1129 MWT-FOSSIL
OPERATING MODE:	CYCLING
TURBINE:	CONVENTIONAL
AVAILABLE WATER:	1000 ACRE-FEET

DESIGN AND OPERATING PARAMETERS

1.	NUMBER OF DRY TOWER MODULES	37	
2.	NUMBER OF WET TOWER MODULES	9	
3.	CIRCULATING WATER FLOW RATE	400	FT^3/SEC
4.	FRACTION OF FLOW TO WET TOWERS	95%	
5.	NUMBER OF WET TOWER CONTROL UNITS	3	
6.	CONDENSER HEAT EXCHANGE AREA	243,000	FT^2
7.	MAXIMUM CONDENSING PRESSURE	3.6	IN. HG
8.	MINIMUM PLANT CAPACITY	472.8	MW
9.	PUMP AND FAN POWER	10.1	MW
10.	ANNUAL ENERGY PRODUCTION LOSS	33,700	MWHRS
11.	ANNUAL FAN AND PUMP POWER	40,700	MWHRS
12.	CAPACITY FACTOR	40	%

CAPITAL AND OPERATING COST($X10^6$)

1.	DRY TOWERS AND FOUNDATION	$ 12.90
2.	WET TOWERS AND FOUNDATION	$ 5.31
3.	CONDENSER	$ 5.16
4.	ELECTRICAL EQUIPMENT	$ 2.92
5.	STEEL PIPELINES	$ 3.33
6.	CIRCULATING WATER PUMP STRUCTURE	$ 0.47
7.	CIRCULATING WATER PUMPS AND MOTORS	$ 1.97
8.	INDIRECT CAPITAL	$ 12.03
9.	TOTAL CAPITAL	$ 44.1
10.	ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 1.51
11.	ANNUAL COST OF PUMP AND FAN POWER	$ 0.89
12.	PLANT FUEL COST	21.9 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)

CAPITAL	$ 85.4
OPERATING	$ 17.0
LOSS OF ENERGY PRODUCTION	$ 21.6
LOSS OF CAPACITY	$ 34.9
TOTAL	$ 158.9

COST OF HEAT REJECTION = 8.34 MILLS/KWHR

SYSTEM # 21

Site: Phoenix
Plant: 1129 MWt-Fossil
Operating Mode: Base-load
Turbine: High Back Pressure
Available Water: 250 acre-feet

DESIGN AND OPERATING PARAMETERS		
1. Number of Dry Tower Modules	45	
2. Number of Wet Tower Modules	2	
3. Circulating Water Flow Rate	400	FT^3/SEC
4. Fraction of Flow to Wet Towers	21%	
5. Number of Wet Tower Control Units	3	
6. Condenser Heat Exchange Area	248,000	FT^2
7. Maximum Condensing Pressure	11.1	IN. Hg
8. Minimum Plant Capacity	446.4	MW
9. Pump and Fan Power	8.56	MW
10. Annual Energy Production Loss	363,000	MWHrs
11. Annual Fan and Pump Power	64,800	MWHrs
12. Capacity Factor	75	%

CAPITAL AND OPERATING COST($X10^6$)	
1. Dry Towers and Foundation	$ 15.69
2. Wet Towers and Foundation	$ 1.18
3. Condenser	$ 5.23
4. Electrical Equipment	$ 2.77
5. Steel Pipelines	$ 3.09
6. Circulating Water Pump Structure	$ 0.44
7. Circulating Water Pumps and Motors	$ 1.35
8. Indirect Capital	$ 11.16
9. Total Capital	$ 40.93
10. Annual Loss of Energy Production Penalty	$ 6.87
11. Annual Cost of Pump and Fan Power	$ 0.76
12. Plant Fuel Cost	11.8 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
Capital	$ 79.2
Operating	$ 14.9
Loss of Energy Production	$ 98.5
Loss of Capacity	$ 41.9
TOTAL	$ 234.5

Cost of Heat Rejection = 6.61 MILLS/KWHR

SYSTEM # 22

SITE: PHOENIX
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: CYCLING
TURBINE: HIGH BACK PRESSURE
AVAILABLE WATER: 250 ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	45	
2. NUMBER OF WET TOWER MODULES	2	
3. CIRCULATING WATER FLOW RATE	400	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	21%	
5. NUMBER OF WET TOWER CONTROL UNITS	3	
6. CONDENSER HEAT EXCHANGE AREA	248,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	11.1	IN. HG
8. MINIMUM PLANT CAPACITY	446.4	MW
9. PUMP AND FAN POWER	8.56	MW
10. ANNUAL ENERGY PRODUCTION LOSS	213,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	34,900	MWHRS
12. CAPACITY FACTOR	41	%

CAPITAL AND OPERATING COST($X10^6$)	
1. DRY TOWERS AND FOUNDATION	$ 15.69
2. WET TOWERS AND FOUNDATION	$ 1.18
3. CONDENSER	$ 5.23
4. ELECTRICAL EQUIPMENT	$ 2.77
5. STEEL PIPELINES	$ 3.09
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.44
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.35
8. INDIRECT CAPITAL	$ 11.66
9. TOTAL CAPITAL	$ 40.93
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 5.31
11. ANNUAL COST OF PUMP AND FAN POWER	$ 0.43
12. PLANT FUEL COST	11.8 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 79.2
OPERATING	$ 9.9
LOSS OF ENERGY PRODUCTION	$ 76.0
LOSS OF CAPACITY	$ 41.9
TOTAL	$207.1

COST OF HEAT REJECTION = 10.6 MILLS/KWHR

SYSTEM # 23

SITE: PHOENIX
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: BASE-LOAD
TURBINE: HIGH BACK PRESSURE
AVAILABLE WATER: 1000 ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	35	
2. NUMBER OF WET TOWER MODULES	4	
3. CIRCULATING WATER FLOW RATE	300	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	56%	
5. NUMBER OF WET TOWER CONTROL UNITS	3	
6. CONDENSER HEAT EXCHANGE AREA	211,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	8.3	IN. HG
8. MINIMUM PLANT CAPACITY	457.0	MW
9. PUMP AND FAN POWER	7.3	MW
10. ANNUAL ENERGY PRODUCTION LOSS	375,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	52,100	MWHRS
12. CAPACITY FACTOR	75	%

CAPITAL AND OPERATING COST($X10^6$)	
1. DRY TOWERS AND FOUNDATION	$ 12.20
2. WET TOWERS AND FOUNDATION	$ 2.36
3. CONDENSER	$ 4.61
4. ELECTRICAL EQUIPMENT	$ 2.37
5. STEEL PIPELINES	$ 2.40
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.32
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.18
8. INDIRECT CAPITAL	$ 9.54
9. TOTAL CAPITAL	$ 35.01
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 6.42
11. ANNUAL COST OF PUMP AND FAN POWER	$ 0.61
12. PLANT FUEL COST	11.8 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 67.7
OPERATING	$ 12.2
LOSS OF ENERGY PRODUCTION	$ 92.0
LOSS OF CAPACITY	$ 34.6
TOTAL	$ 206.5

COST OF HEAT REJECTION = 5.83 MILLS/KWHR

SYSTEM #24

SITE: PHOENIX
PLANT: 1129 MWT-FOSSIL
OPERATING MODE: CYCLING
TURBINE: HIGH BACK PRESSURE
AVAILABLE WATER: 1000 ACRE-FEET

DESIGN AND OPERATING PARAMETERS		
1. NUMBER OF DRY TOWER MODULES	27	
2. NUMBER OF WET TOWER MODULES	4	
3. CIRCULATING WATER FLOW RATE	300	FT^3/SEC
4. FRACTION OF FLOW TO WET TOWERS	56%	
5. NUMBER OF WET TOWER CONTROL UNITS	3	
6. CONDENSER HEAT EXCHANGE AREA	211,000	FT^2
7. MAXIMUM CONDENSING PRESSURE	8.0	IN. HG
8. MINIMUM PLANT CAPACITY	456	MW
9. PUMP AND FAN POWER	6.7	MW
10. ANNUAL ENERGY PRODUCTION LOSS	329,000	MWHRS
11. ANNUAL FAN AND PUMP POWER	26,200	MWHRS
12. CAPACITY FACTOR	41	%

CAPITAL AND OPERATING COST($X10^6$)	
1. DRY TOWERS AND FOUNDATION	$ 9.41
2. WET TOWERS AND FOUNDATION	$ 2.36
3. CONDENSER	$ 4.61
4. ELECTRICAL EQUIPMENT	$ 1.90
5. STEEL PIPELINES	$ 2.15
6. CIRCULATING WATER PUMP STRUCTURE	$ 0.36
7. CIRCULATING WATER PUMPS AND MOTORS	$ 1.29
8. INDIRECT CAPITAL	$ 8.29
9. TOTAL CAPITAL	$ 30.40
10. ANNUAL LOSS OF ENERGY PRODUCTION PENALTY	$ 5.08
11. ANNUAL COST OF PUMP AND FAN POWER	$ 0.31
12. PLANT FUEL COST	11.8 MILLS/KWHR

PRESENT WORTH TOTAL COSTS ($X10^6$)	
CAPITAL	$ 58.8
OPERATING	$ 7.39
LOSS OF ENERGY PRODUCTION	$ 72.8
LOSS OF CAPACITY	$ 33.6
TOTAL	$172.59

COST OF HEAT REJECTION = 8.88 MILLS/KWHR

APPENDIX B

CYCLING STEAM-ELECTRIC PLANTS

B.1 Background

Traditionally, utilities have purchased new additions of the largest and most efficient steam cycle available. This relegated the system's previously base-loaded plants to cyclic or lower load factor operation. In comparison to a base-load plant which operates essentially continuously (except for maintenance and forced outage), a cycling plant operates between about 2,000 and 5,000 hours per year and is dispatched as the utility system electrical load varies. Historically, the system's largest unit would be expected to operate at a capacity factor equal to its availability for the first several years of commercial operation, while the remaining years would find the unit capacity factor continually decreasing until the unit was no longer required except in emergencies (Reference B1).

During the 1960's the electrical utility industry entered an era of diminishing improvements in heat rates. New generating units designed specifically for base-load operation began forcing older, but only slightly less efficient, units into cyclic duty.

This trend, along with other factors such as the introduction of significant amounts of nuclear power, resulted in the implementation of new methods of system expansion. These new methods of expansion provide for adding capacity to the generating capability in three general time-based segments. These have been designated as:

(1) high-efficiency base-load plants,
(2) medium load factor or cycling plants, and
(3) minimum load factor or reserve plants

In each case, the plant would be optimized for minimum lifetime power production cost for the designated load factor.

The types of thermal power plants which have been considered for cycling operation are steam plants and combined gas-turbine and steam plants. Before the dramatic rise in fuel prices in 1974, the specific types of plants favored for cycling service were simple steam cycles fired by oil or gas with turbine inlet conditions of about 1850 psig, 950°F with reheat to 950°F. Because of the large capital costs of fuel handling, coal-firing of these plants was avoided (Reference B2). Also, combined cycle gas-turbine and steam-turbine plants were seen to have economic advantages during this time period for cycling duty. One major architect-engineer offered a predesigned 200 Mwe combined cycle package which utilized dry cooling (Reference B3).

However, since the advent of high energy prices, the 1850 psig/950°F/950°F plant has nearly become extinct and has been superseded by more efficient 2400 and 2600 psig/1000°F/1000°F designs for cycling plants. Electrical World's 15th Steam Station Design Survey (1978) found that, of 60 new plants ordered by utilities, 22 were designed for cycling service (Reference B4). Of these, only three were not designed for coal-firing. No combined cycle plants are reported in the recent survey, but there remains an active interest in technical developments of combined cycle plants which could utilize coal derived fuels (Reference B5). Becaue of low fuel costs, high capital costs, and fuel integrity considerations, nuclear power plants are not used for cycling services.

Cycling service may be defined as the use of a steam generator for repetitive periodic operation to supplement base power production of other units in the electric utility system. Different modes of cyclic operation have been defined. Among these are (Reference B6):

(a) Week mode--the unit will be base-loaded during weekdays and removed from service each weekend.

(b) Peak mode or two-shifting--the boiler plant will operate weekedays at full load and be off-line every night for 8 to 10 hours. In addition, the unit will be removed from service each weekend.

(c) Coal cycling mode--the boiler plant will operate weekdays at full load during the day and at minimum load at night. The unit will be removed from service each weekend.

For the 30-year life of a steam generator, cycling service, if continuous for this full life period, could result in the following type of operation (Reference 7):

a) 6000 daily starts after 10 to 16 hours overnight shutdowns;
b) 1500 weekly starts after 50 to 60 hour weekend shutdowns;
c) 45,000 cumulative load changes based on 1500 per year of 50 percent magnitude or greater.

B.2 Conventional Pulverized Coal Steam Plants for Cycling Duty

The design of coal-fired cycling steam plants is generally similar to that of a base-load unit. There are, nevertheless, some special features which are not normally found in base-load units. These include (Reference B8, B14, B15, B16):

(1) A drainable superheater to prevent cold water blockage during startup

(2) Upgraded superheater metals

(3) Steam temperature control systems to maintain boiler temperature balance

(4) A steam bypass before the primary superheater directly to the condenser to facilitate steam turbine temperature matching.

Also, cycling units are usually designed for variable pressure operation to gain a better heat rate at low load operation and allow the turbine to stay hotter during shutdown so that more rapid startups can be accomplished.

These additional features have led to total plant costs which are estimated to be about 5 percent greater than the cost of a comparable unit designed for base-load operation.

With a drainable superheater and a steam bypass, startup times of approximately one hour (ten hours or overnight shutdown) and two hours (55 hours or weekend shutdown) have been reported for a 600 MWe unit (Reference B8).

Steam turbine generators have been developed and marketed for cycling service. Special design features have been incorporated to allow for rapid starting and loading. For large units, designs for cycling service result in about a 0.75 percent increase in the turbine heat rate in comparison to a standard design (Reference B9).

Currently, cycling coal-fueled steam-electric units as large as 800 MWe are being purchased by electric utilities. In most cases, the design basis for the plant would be an operating schedule similar to that listed above. However, during the actual operation of the plant, the load duty may be considerably different. Depending upon the utility's ability to complete its generation expansion program and its ability to predict plant outages and load demands, a unit designed for cycling service may be operated as essentially a base-loaded plant.

B.3 Combined Cycle Gas Turbine and Steam Turbine Plants for Cycling Duty

In the early 1970's several different design concepts for combined cycle gas turbine and steam turbine plants were advanced as viable options for intermediate load duty (Reference B120, B11). These concepts were based on existing gas turbine technology and would be fired using premium fuels. However, as pointed out earlier, the rapid rise in the price of premium fuels during the 1970's has thwarted the implementation of combined cycle units which were designed using conventional technology.

There are, nevertheless, development efforts currently underway which should eventually lead to commercially viable combined cycle units using coal as the primary fuel. Design concepts under consideration are (Reference B12, B13):

(1) integrated coal gasifier and combined-cycle plants
(2) pressurized fluidized-bed combined cycle units
(3) combined cycle units using coal-derived liquid fuels.

Advanced combined-cycle concepts using coal or coal-derived fuels call for gas turbines with inlet temperatures as high as 3000°F and for steam bottoming cycles similar to current 2400 psig/1000°F/1000°F designs.

Any of these systems could be implemented for cyclic load duty. Some concepts would have basic economic advantages over other for this mode of operation.

Because of the development status of coal-fueled combined cycles, the impact of wet/dry cooling on combined cycle plants were not investigated in detail in this project. However, several points are worth mentioning. First, and most obvious, is that overall high system efficiency and resultant smaller exhaust of waste heat in the turbine exhaust stream leads to significantly reduced condenser cooling requirements per Kw of electrical output. Thus the higher cost implications of wet/dry cooling become proprotionately less significant in terms of cost of cooling per unit power operation. Second, the impact of ambient air temperature will generally have a greater effect on the power output of the gas turbine than on the output of the steam bottoming cycle. As shown in Figure B.1, the impact of the variation in the condenser pressure (over the range of interest) on the net output of the plant is considerably smaller than the power variation due to the impact of varying ambient temperatures on the gas turbine output.

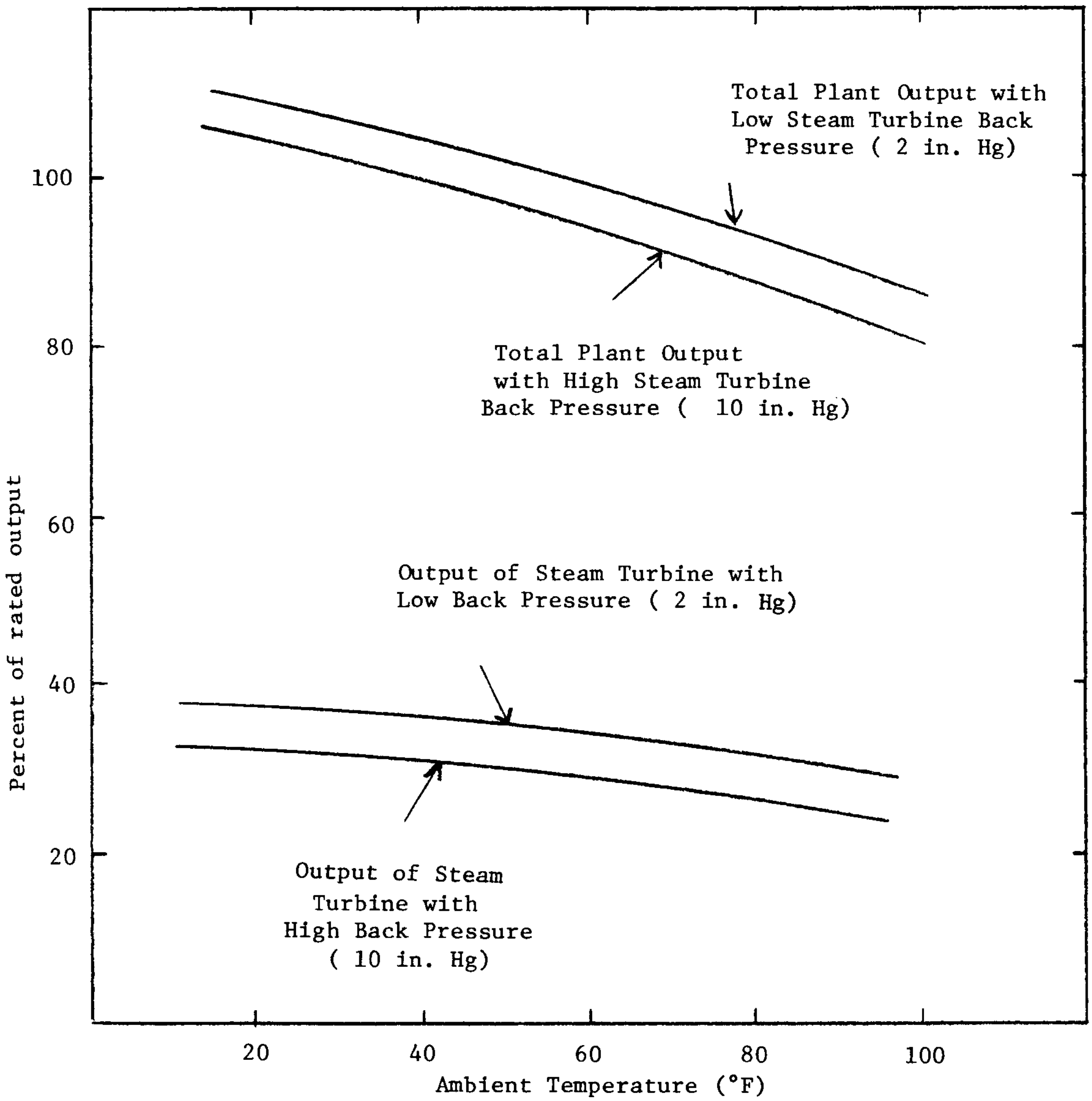

Figure B.1 Effect of Ambient Temperature on Combined-Cycle Plant

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