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ABSTRACT

The selection of waste heat rejection systems for steam-electric power plants involves a trade-off among environmental, energy and water conservation, and economic factors. This study compares four general types of cooling systems on the basis of these factors. The cooling systems chosen for study are: once-through systems including surface canals and submerged multiport diffusers; shallow closed cycle cooling ponds; mechanical and natural draft evaporative cooling towers; and mechanical draft dry towers.

The cooling system comparison involves, first, an optimization of each cooling system and then a comparison among optimal systems. Comparison is made for an 800 MWe fossil unit and a 1200 MWe nuclear unit located at a hypothetical midwestern river site. A set of models has been developed to optimize the components of each cooling system based on the local meteorological and hydrological conditions at the site in accordance with a fixed demand, scalable plant concept. This concept allows one to compare the costs of producing the same net power from each plant/cooling system. Base case economic parameters were used to evaluate the optimum system for each of the four general cooling systems followed by a sensitivity study for each parameter. Comparison of energy and water consumption follows from the results of the performance model, while comparison of environmental impacts is mostly qualitative. Some quantitative modelling was performed for the environmental effects of thermal discharges from once-through systems, fogging from wet cooling towers and water consumption from the ponds, wet towers and once-through.

The results of the optimization models of each of the systems are compared on the basis of: performance - discrete distributions of environmental conditions and transient simulation; economics - using base case scenarios and sensitivity values to arrive at costs expressed in terms of production costs, annualized costs and present value costs; energy and water consumption; and environmental effects. The once-through systems were found to be the least expensive of the four systems, the most energy efficient, but potentially the most environmentally damaging. On the other extreme, dry cooling towers are the most environmentally sound while being the most expensive and least energy efficient. Finally, the results of the economic optimization are compared with results from previous comparative studies.

ACKNOWLEDGMENTS

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"Computer Optimization of Dry and Wet/Dry Cooling Tower Systems for Large Fossil and Nuclear Plants," by Choi, M., and Glicksman, L.R., MIT Energy Laboratory Report No. MIT-EL 79-034, February 1979.

"Computer Optimization of the MIT Advanced Wet/Dry Cooling Tower Concept for Power Plants," by Choi, M., and Glicksman, L.R., MIT Energy Laboratory Report No. MIT-EL 79-035, September 1979.

"Operational Issues Involving Use of Supplementary Cooling Towers to Meet Stream Temperature Standards with Application to the Browns Ferry Nuclear Plant," by Stolzenbach, K.D., Freudberg, S.A., Ostrowski, P., and Rhodes, J.A., MIT Energy Laboratory Report No. MIT-EL 79-036, January 1979.

"An Environmental and Economic Comparison of Cooling System Designs for Steam-Electric Power Plants," by Najjar, K.F., Shaw, J.J., Adams, E.E., Jirka, G.H., and Harleman, D.R.F., MIT Energy Laboratory Report No. MIT-EL 79-037, January 1979.

"Economic Implications of Open versus Closed Cycle Cooling for New Steam-Electric Power Plants: A National and Regional Survey," by Shaw, J.J., Adams, E.E., Barbera, R.J., Arntzen, B.C., and Harleman, D.R.F., MIT Energy Laboratory Report No. MIT-EL 79-038, September 1979.

"Mathematical Predictive Models for Cooling Ponds and Lakes," Part B: User's Manual and Applications of MITEMP by Octavio, K.H., Watanabe, M., Adams, E.E., Jirka, G.H., Helfrich, K.R., and Harleman, D.R.F.; and Part C: A Transient Analytical Model for Shallow Cooling Ponds, by Adams, E.E., and Koussis, A., MIT Energy Laboratory Report No. MIT-EL 79-039, December 1979.

"Summary Report of Waste Heat Management in the Electric Power Industry: Issues of Energy Conservation and Station Operation under Environmental Constraints," by Adams, E.E., and Harleman, D.R.F., MIT Energy Laboratory Report No. MIT-EL 79-040, December 1979.

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Chapter I

INTRODUCTION

1.1 Background

The continuously increasing demand for electric power in the United States, both in absolute terms and as a fraction of the total energy consumption, documents the attractiveness of this energy form for domestic, commercial and industrial consumers (Crow, 1977). Presently, the generation of electric energy requires about 29% of the nation's overall energy usage. According to the National Electric Reliability Council (1977), this is expected to approach 40% by the year 1980 and 50% by the year 2000.

Table 1.1 shows the nature of this increasing energy demand for both total energy consumption and electricity consumption in the U.S. The large variance in the projected demand among different groups reflects the difficulty in predicting the nation's energy needs. It is clear nonetheless that many more power facilities will be required to meet the growing demand for electricity.

The two principal sources of electric energy are (1) by the conversion of heat in central steam-electric generating stations (presently about 84% of the total national generation) and, (2) by kinetic energy conversion of falling water in hydroelectric power stations (about 13% presently). This study is concerned with steam-electric power generation where the increase in the number of power plants inherently means large costs (both capital and operating), increased fuel and water consumption, and more environmental impacts.

Table 1.1

ENERGY FORECASTS

SOURCE		TOTAL ENERGY (10^{15} BTU)		ELECTRICITY (10^{12} KWH)		ELECTRICITY* Share (%)	
ACTUAL - 1975		79.7		1.90		24.4	
		1985	2000	1985	2000	1985	2000
Chapman, et al. (1972)	High				9.890		
	Med.				3.450		
	Low				2.010		
Dupree-West (1972)		116.6	191.9	4.140	9.010	36.4	48.1
Bureau of Mines (1973)**				4.378	10.432		
Hudson-Jorgenson (1974)		108.2	164.5	3.363	6.981	31.8	43.4
ERDA-48 (1975)	Scenario: 0	107.3	165.5	3.455	6.903	33.0	42.7
	I	96.9	122.5	3.199	4.152	33.8	34.7
	II	107.3	165.4	3.455	6.792	33.0	42.0
	III	106.7	161.2	3.747	8.236	36.0	52.3
	IV	107.0	158.0	3.334	4.694	31.7	30.4
	V	98.1	137.0	3.217	4.335	33.6	32.4
ERDA (1976)	Import Dependence	100.0	156.2	3.321	5.860	34.0	38.4
	Domestic Development	96.7	135.9	3.321	6.349	35.2	47.8
FERC (1977)**		103.7	163.4	4.070	9.332	40.3	58.5
EPRI (1977)		100.9	142.4	2.880	5.030	29.2	36.2
EPRI (1978)	High	104.8	196.0	3.889	9.200	38.0	48.1
	Base	97.6	159.0	3.655	7.400	38.3	47.7
	Low	94.4	146.0	3.544	6.600	38.4	46.3

* Assuming heat rate = 10,238 BTU/KWH

** As reported by U.S. Water Resource Council, 1977

In steam-electric power plants the chemical energy of the prime mover, either fossil or nuclear fuels, is ultimately converted into electric energy. The overall conversion efficiency of these stations, however, is low; on the order of 33% to 40% for modern facilities. This means that about two thirds of the energy of the prime mover is lost in the form of "waste heat" discharged into rivers, lakes and the atmosphere. In view of the national goal of conservation of energy resources, this appears to be a highly wasteful process and suggests that any effort to improve this efficiency should be pursued. Also, since all large steam-electric power plants use water for steam condensation, there are environmental impacts as well as large water requirements associated with the cooling process. The management of waste heat from steam-electric power plants is thus significant with regard to environmental impacts and the potential for energy and water conservation. This study deals with one area in which all these factors come together -- namely the selection of the waste heat rejection system.

The primary goals of this thesis are:

(1) to identify and compare costs (capital, operating and penalty) associated with the use of various cooling systems for new base-loaded steam-electric power plants. In this way the true differences in costs for various cooling system alternatives can be ascertained;

(2) to examine the fuel and water conservation issues associated with cooling system selection, and

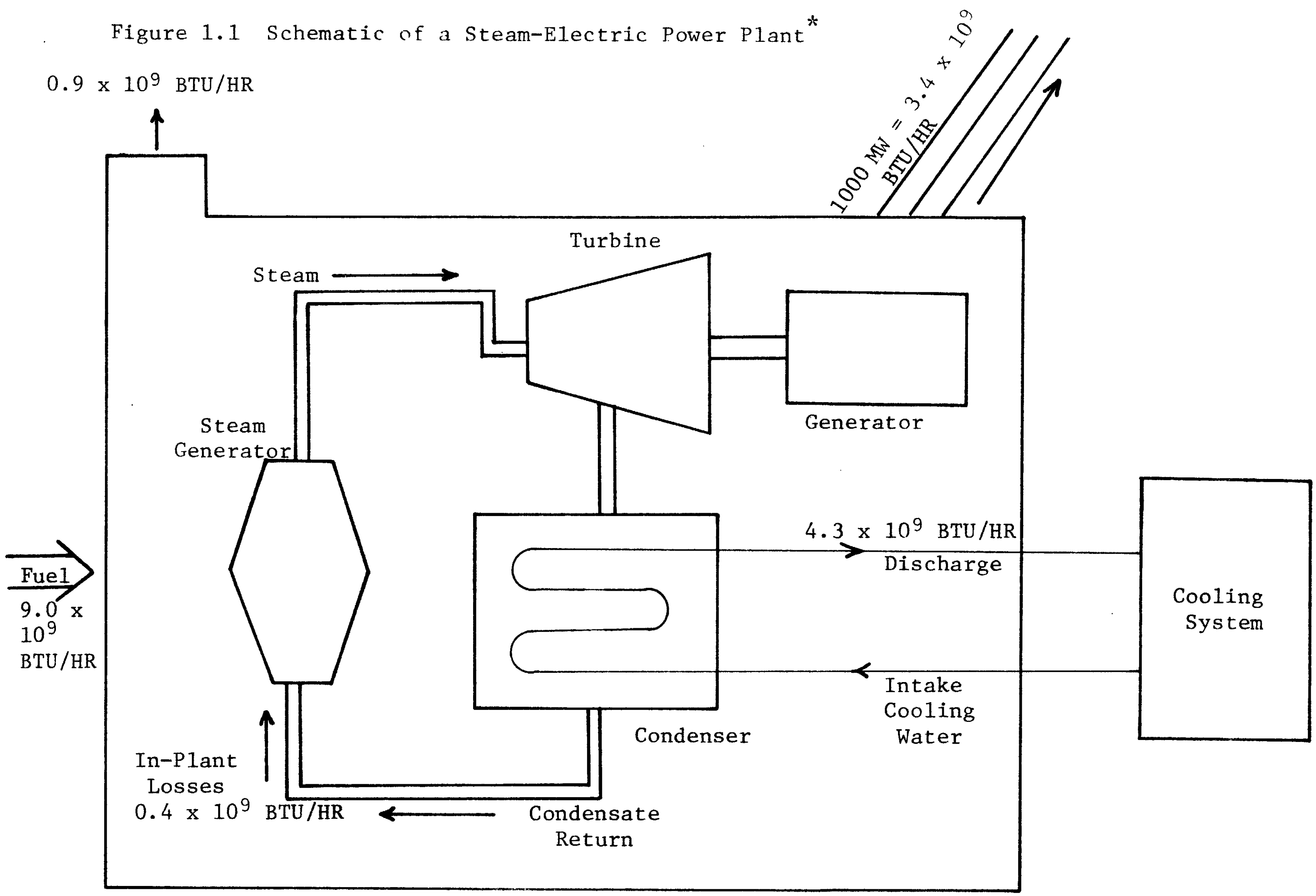
(3) to analyze various environmental factors associated with the different methods and policies of waste heat rejection.

Four general types of cooling systems are chosen for study: once-through systems including surface canals and submerged multi-port diffusers; shallow closed cycle cooling ponds; mechanical and natural draft evaporative cooling towers; and mechanical draft dry towers. These systems were chosen to provide a representative range of alternatives for comparison and do not include all possible cooling systems. Several mixed-mode cooling systems which are either in use or are being considered for use in large power plants will be discussed briefly.

1.2 Power Plant Cooling System

Steam-electric power plants operate on the basis of a thermodynamic cycle which converts heat into work. The major conversion steps in the steam-electric process are: chemical energy of the fuel → heat → mechanical energy → electrical energy. Heat is produced by combustion of coal, oil or gas for fossil-fuel plants and from controlled atomic fission of nuclear fuel in a reactor for nuclear plants. This heat is then used to turn boiler water into steam, which is harnessed at high temperature and pressure to move a turbine. The turbine turns a generator, thereby converting mechanical energy into electricity which is then transmitted from the plant to the eventual user. The steam meanwhile, leaves the turbine and enters a condenser, where it gives off its remaining heat to continuously circulating cooling water. Once the steam has condensed back to water, it is returned to the boiler to begin the next cycle. Figure 1.1 shows a general schematic of a steam-electric power plant where either a fossil or nuclear fuel source can supply the energy to the steam generator.

Figure 1.1 Schematic of a Steam-Electric Power Plant*



* Heat fluxes represent a 1000 MWe fossil plant

The critical phase in the process is the conversion from heat into mechanical energy by means of the heat engine. A fundamental efficiency expression for this conversion is the Carnot (ideal) efficiency

$$\eta = 1 - \frac{T_2}{T_1} \tag{1.1}$$

where T_1 is the temperature of the heat source and T_2 is the temperature of the heat sink, both measured on an absolute scale. In the more practical Rankine Cycle engines, the efficiencies are lower than given by Equation (1.1), although it still holds qualitatively. In the steam engine the heat source can be represented by the steam temperature in the boiler and the heat sink by the water temperature in the condenser.

The heat source temperature T_1 is governed by the choice of the prime mover and technological constraints (materials etc.) on the combustion or reactor processes. Typical values are 1000^oF (550^oC) for fossil-fueled plants and 600^oF (320^oC) for nuclear plants.

The focus of this study deals with the heat sink or steam condensing temperature T_2 which has an equally important effect on conversion efficiency as seen from Equation (1.1). The steam condensing temperature (SCT) can be written as the sum of the environmental background temperature T_{ENV} and the temperature differential ΔT_{CS} of the cooling system (including condenser) which is used to reject the waste heat

$$SCT = T_2 = T_{ENV} + \Delta T_{CS} \tag{1.2}$$

T_{ENV} is governed by the choice of the cooling medium (the atmosphere or water body) and by its characteristic variability due to seasonal or weather effects. The value of ΔT_{CS} is dependent on the choice, size and design of the cooling system.

The efficiency is viewed more clearly by defining the overall heat balance within the power plant. The heat balance example in Figure 1.1 for a 1000 MWe fossil plant shows that a heat rate of 3.4×10^9 BTU/hr of 9.0×10^9 BTU/hr supplied is used for the generation of electric power (η = 37.8%). Note that for this fossil plant waste heat rejected in the cooling system accounts for about 48% of the input energy, in-plant losses account for about 4%, 10% of the energy is lost in the stacks, leaving about 38% for generation. A nuclear plant, however, converts only about 32% of its energy while about 5% is lost within the plant. About 63% is then lost in waste heat. Thus, nuclear plants require considerably more cooling water, as well as more fuel, when compared with fossil plants.

Open cycle (or once-through) systems usually have the lowest T_2 and therefore the highest efficiency. Thus, they have traditionally been the choice of power plant designers. However, once-through systems have large water withdrawal requirements (between 500 and 2000 cubic feet per second for a 1000 MWe plant) and may possess significant environmental impacts (e.g., thermal pollution and intake entrainment). In view of increasingly stringent environmental standards and the decrease in cooling water supply, closed-cycle systems are becoming more popular.

Closed cycle systems (ponds, wet towers, dry towers) recycle the cooling water, thereby minimizing the water withdrawal (20 to 50 cfs for a 1000 MWe plant using wet cooling and negligible amounts for one using dry cooling), while substantially reducing the thermal and intake burden on the aquatic ecology. Another advantage is increased siting flexibility; the freedom to locate a power plant on smaller water bodies which may be closer to the fuel source or electrical load center contributes to reduced generating costs. The disadvantages of closed cycle cooling lie mostly in the higher capital and operating costs. Lower thermal efficiency due to warmer intake temperatures (higher T_2 caused by recycling the cooling water) leads to greater fuel use and more waste heat produced per KWH of power generated. In addition, other environmental impacts including fogging, noise, land use, drift, chemical blowdown, aesthetics, etc., may be encountered. While water withdrawal has decreased substantially, there is increased water consumption (by evaporation, drift, etc.) from the use of some closed cycle systems.

Clearly, the availability of water is one of the most important issues in the selection of a cooling system. The total national water use (withdrawal and consumption) by steam-electric power plants in the U.S. is shown in Table 1.2 for 1975 and for projections to the years 1985 and 2000.

These forecasts are derived from WRC's (1977) capacity and generation estimates for steam-electric plants according to mode of cooling. These modes reflect utility projections for power demand, water availability for each of 21 regions and anticipated constraints on thermal discharges from once-through cooling. The totals for these WRC regions is given in Table 1.3. In examining the table it is worth noting the following:

(1) Steam-electric power plants are projected to produce 94% of the total national generation by the year 2000 while hydroelectric

Table 1.2

WATER USE BY STEAM-ELECTRIC POWER PLANTS

	WITHDRAWAL (MGD)			CONSUMPTION (MGD)		
	Fresh	Saline	Ground	Fresh	Saline	Ground
1975	92,342	46,683	259	1,208	326	811
1985	86,547	78,653	357	3,491	785	157
2000	69,912	93,815	151	9,061	2,320	87

Table 1.3 ESTIMATED STEAM-ELECTRIC AND HYDROELECTRIC GENERATION AND USE OF COOLING METHODS BY STEAM-ELECTRIC PLANTS IN THE UNITED STATES (From U.S. Water Resource Council, 1977)

STEAM ELECTRIC	1975		1985		2000	
Cooling Method	Generation GWH	% Total Steam Generation	Generation GWH	% Total Steam Generation	Generation GWH	% Total Steam Generation
Once-Through						
Fossil	1,025,996	59	983,675	28	688,855	8
Nuclear	105,219	6	323,185	9	590,933	7
Total	1,131,215	65	1,306,860	37	1,279,788	15
Cooling Ponds						
Fossil	190,787	11	362,862	10	459,679	5
Nuclear	31,236	2	154,338	4	477,062	6
Total	222,023	13	517,200	14	936,741	11
Wet Towers						
Fossil	292,984	17	761,000	21	1,705,660	19
Nuclear	40,846	2	917,064	26	4,276,799	49
Total	333,830	19	1,679,064	47	5,982,459	68
Dry Towers						
Fossil	170	~	3,465	~	30,351	~
Nuclear	0	0	0	0	343,726	4
Total	170	~	3,465	~	374,077	4
Combination						
Fossil	29,025	2	48,874	$1\frac{1}{2}$	36,653	~
Nuclear	16,200	1	17,247	1/2	172,810	2
Total	45,225	3	66,121	2	209,463	2
TOTAL STEAM-ELECTRIC	1,732,463	100	3,572,710	100	8,782,528	100
TOTAL HYDRO	248,235	-	268,235	-	305,437	-
TOTAL STEAM & HYDRO	1,980,698	-	3,840,945	-	9,087,965	-

plants will produce 3%.

(2) Steam-electric energy generation is projected to increase five-fold from 1975 to 2000.

(3) Nuclear plants, which use more water, are projected to produce 67% of total steam-generation in 2000 compared with 11% in 1975.

According to the table, the fraction of generation using once-through cooling will rapidly decline over time while that fraction using wet towers will increase just as significantly. The generation using cooling ponds will increase over time but the percent of the distribution will remain about the same. The usage of dry towers and combined cooling systems will continue to be small to the year 2000.

1.3 Objectives

This thesis is part four of a five-part study which examines the trade-offs among cost, environmental impact and conservation issues associated with cooling system choices for new base-load power plants. The first three parts deal with various cooling modes including (1) the optimization of dry and wet/dry towers for closed cycle cooling, (2) the optimization of artificial cooling ponds for closed cycle cooling and (3) the intermittent use of evaporative cooling towers to supplement once-through cooling for purposes of meeting environmental constraints. Note that the three cooling systems chosen for study are each alternatives to the more conventional wet tower.

This thesis (part 4 of the study) attempts to integrate the results of the first three parts by providing a unified comparison of cooling system performance. The study considers a single hypothetical site and uses detailed performance models to design four major cooling modes: once-through, cooling ponds, wet tower and dry tower. The systems are then compared with respect to issues of performance and cost (under both transient and long term conditions), environmental impacts, and fuel and water comsumption. Throughout the study an effort is made to relate the procedures and results to those found in previous comparative studies including United Engineers and Constructors (1974), Croley (1975), Technekron (1976), Sebald (1976), Fryer (1976), and Rossie, *et al*. (1972).

In part five of the study, the results of this comparison are used along with various scenarios of energy demand, to address several national issues associated with cooling system selection. In particular, for areas in which once-through cooling is possible, the national costs of future thermal discharge controls are estimated, while for areas with less water, an estimate is made of the relative contributions which can be made by cooling ponds and wet cooling towers.

1.4 Outline of Presentation

The purpose of this thesis is to compare costs, environmental impacts and the energy and water consumption associated with the choice of cooling system for a large base-loaded power plant using a hypothetical site as a case study. The present chapter has discussed the overall picture of waste heat management as it pertains to these trade-offs. Chapter II describes the specific procedures used for the study including the general design procedure, assumptions regarding costs, lost capacity, etc., and a description of the study site. Chapters III through VI present detailed descriptions of each cooling system as well as design details and results which pertain to the individual cooling system. The final chapter compares the systems with respect to cumulative operating performance (i.e., integrated over the year) and transient performance. Also included are comparisons of the systems' environmental effects, and water and energy consumption. Finally, these results are compared with previous findings.

Chapter II

APPROACH

2.1 Introduction

The cooling system comparison involves, first, an optimization of each cooling system and then a comparison among optimal systems. The cooling system is assumed to include the ultimate heat exchanger (tower, pond, outfall, etc.), the condenser, and connecting pumps and pipes, for a fixed power plant design (boiler, turbine-generator, etc). Comparison is made for an 800 MWe fossil unit and a 1200 MWe nuclear unit located at a hypothetical midwestern river site.

For each cooling system type, an optimal configuration is determined by varying the design (size) of one of more system components and searching for that configuration with the lowest combination of capital, operating and penalty costs. These may be expressed as annual costs ($/year), production costs (mills/KWH), present-valued cost ($), etc. In general, larger systems have higher capital cost but are more efficient and therefore may have lower operating and penalty costs. An optimum can usually be found at some intermediate size as suggested in Figure 2.1.

The number of design variables which are considered depends on the cooling system. In particular dry towers are by far the most expensive and are, in practice, the least utilized of the cooling systems which are considered. Because operating experience has not been sufficient to allow cost-effective sub-optimization or modularization, a considerable amount of effort has been expended in dry tower optimization. The optimization for dry towers follows Andeen et al. (1973) and Choi et al. (1978), and involves

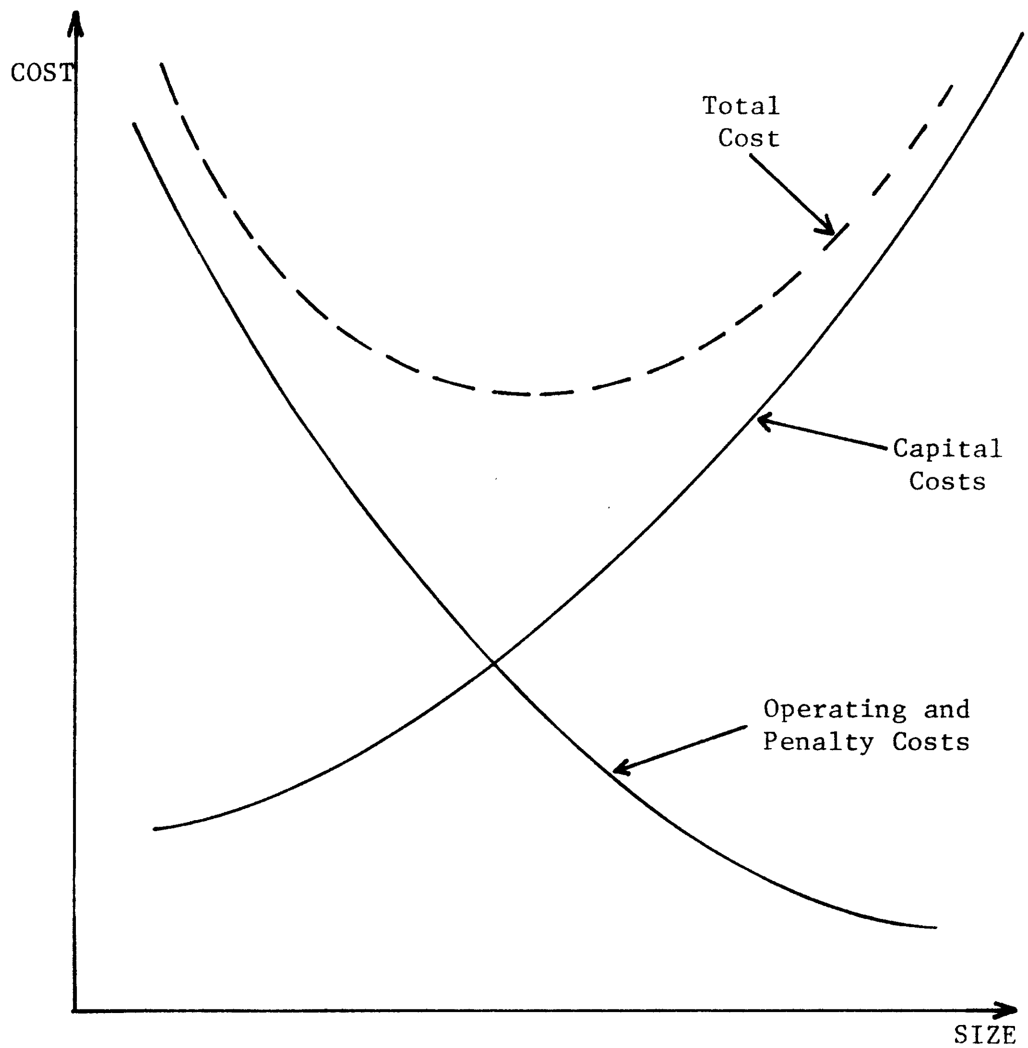


Figure 2.1 Concept of Optimization

the simultaneous design of six components; thus the axis labeled system size in Figure 2.1 really consists of six dimensions for the dry tower design. Additional details concerning the optimization routine are found in Chapter VI. The design codes for the remaining systems are based on a modification of Croley _et al._(1975). In each case at most two components are varied with remaining optimization performed externally.

2.2 System Components

This section discusses, briefly, the assumptions which have been made regarding the turbine-generator, condenser, pump and piping systems as they apply to all cooling systems.

2.2.1 Turbine

Two types of GE turbine-generators were considered for both fossil (General Electric, 1974) and nuclear (General Electric, 1973) units:

(1) A conventional steam turbine with a maximum allowable back pressure of 5 inches HgA (all cooling systems). Nuclear turbine used is model #TC 6F-38 and the fossil turbine is model # CC 6.

(2) A "high back pressure" design which has short last-stage buckets and is capable of operating up to a back pressure of 15 inches HgA (dry system only).

The heat rate ratio versus back pressure curves for these four turbines are shown in Figures 2.2 and 2.3 for the fossil and nuclear plants, respectively. The conventional unit has a rating pressure of 3.5 inches HgA. The heat rates at this pressure are a) 6.35447×10^9 BTU/hr for the fossil turbine, and b) 12.210376×10^9 BTU/hr for the nuclear turbine. The rating

Figure 2.2 Fossil Turbine Characteristics Curves

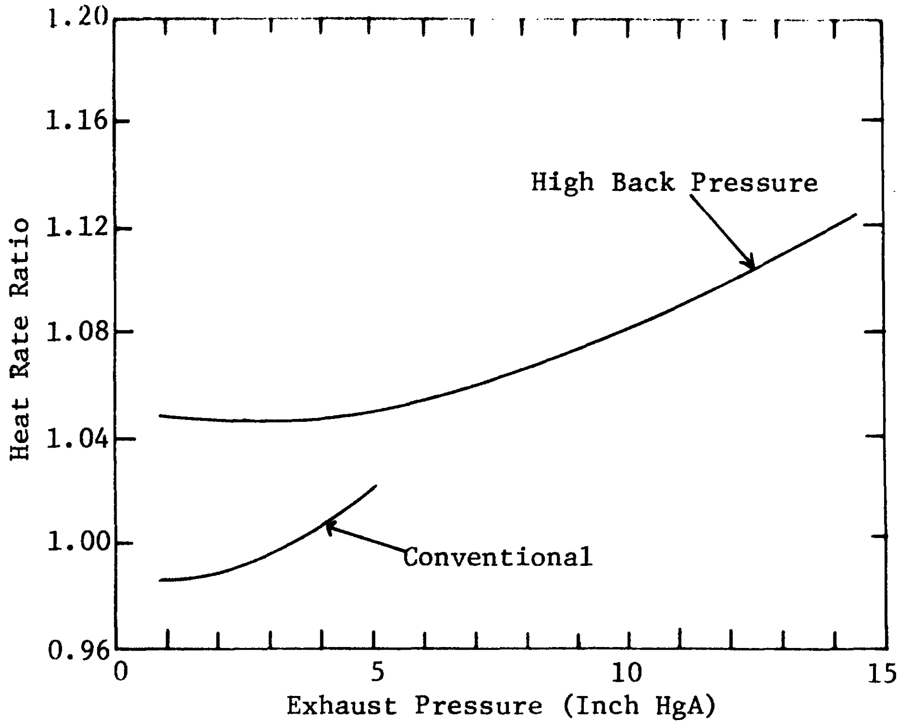
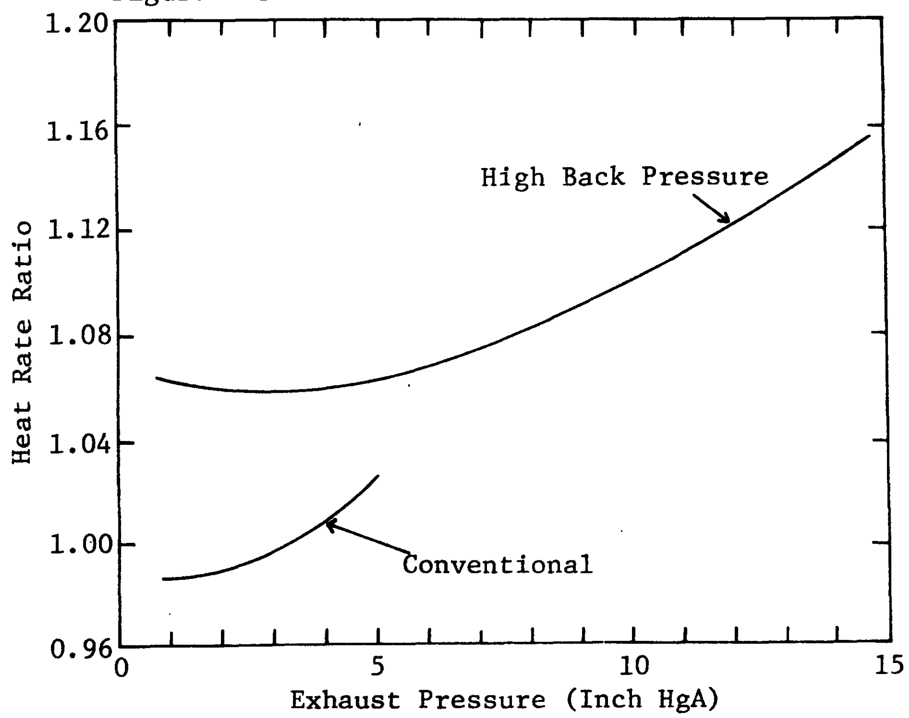


Figure 2.3 Nuclear Turbine Characteristics Curves



pressure for the high back pressure turbine is 8 inches HgA. The associated heat rates are: a) 6.79928×10^9 BTU/hr for the fossil and, b) 13.18721×10^9 BTU/hr for the nuclear. As indicated, the high back pressure unit would require larger steam supply systems than the conventional turbine to produce the rated output. Using these curves the heat rate at any pressure other than the rating exhaust back pressure is obtained by multiplying the rating back pressure by the corresponding heat rate ratio for that specific pressure.

It was reported by Rossie et al. (1973) that the cost of fossil-fueled high back pressure turbines would be the same as conventional turbines while nuclear-fueled high back pressure units would cost 15% more than conventional units. Because the turbine is not considered as part of the cooling system, only the extra 15% for high back pressure nuclear turbines has been attributed to the cooling system capital cost; the remaining cost is considered part of the total plant cost (see Section 2.3).

2.2.2 Condensers

The quantity of condenser surface is the most significant factor in connection with the initial cost of the condenser and is dependent upon the quantity of water, number of passes, and tube material and gauge. The condensers considered in this study are single pressure surface type. Tube material is #18 gauge admiralty with 1 inch outer diameter costing $\$8/ft^2$. A water velocity through the tubes of 7 ft/sec has been selected based on head loss and heat transfer characteristics. The number of tubes is determined from the condenser flow rate, Q_o, velocity and inside pipe diameter (.902 in. for #18 admiralty).

For any Q_o, the required condenser surface area is determined based on heat transfer coefficients recommended by the Heat Exchange Institute (1970) and a terminal temperature difference (TTD) of 5^oF. The length of the tubes is determined from the required area and the outside pipe perimeter. If the calculated tube length exceeds 50 ft, two passes are required. Condenser head losses are calculated based on a friction factor of 0.0125 and include minor losses due to additional passes.

2.2.3 Pump and Pipe Design

Pumping power is computed based on condenser flow rate, head loss as incurred in the condenser, cooling system and connecting pipes, and pump efficiency. A pumping efficiency of 82.2% was selected based on data compiled by Sebald et al. (1976). A linear relationship between capital cost and pumping power, cost (\$) = 1476 + 315 x power (MW), was determined from the same reference.

With the exception of dry towers, a separation of 1000 ft is assumed between condenser and cooling system. A distance of 500 ft is assumed between condenser and the dry tower. The connecting pipes are sized according to the condenser flow rate and an externally optimized pipe velocity of 9 fps. Only diameters between 6' and 20' are considered; for large flow rates requiring larger cross-sectional areas, two equal-sized smaller pipes are assumed. A linear relationship between pipe diameter and capital and installation, cost ($\$/ft^2$) = 16.6 x diam (inches) - 567, was developed from data of Vitro Engineering as reported by Ard et al. (1976).

2.3 Comparison of System Performance

The systems are compared by evaluating the net generating costs, including capital costs, operating costs and penalty costs (for replacement energy and capacity), but excluding transmission costs, in accordance with a fixed demand, scalable source concept. This concept was discussed by Fryer (1976) and is illustrated in Figure 2.4. Transmission costs as they relate to siting decisions will be discussed in Chapter VII.

A fixed demand is assumed throughout the year. For nuclear plants this is 1200 MWe and for fossil plants this is 800 MWe. Due to variations in the environmental parameters which govern plant/cooling system performance (e.g., water temperature for once-through, dry bulb temperature for dry towers, etc.), the net generating capability of any system (gross power minus all auxiliary power requirements) will vary throughout the year. Consider a system which is capable of supplying the net output given by the solid line in the Figure and designated $P_o(t)$. Associated with this system is a capital cost and an operating cost. Because the output from this system will not, in general, equal the target output, it is necessary to account for those periods of the year for which generation is below target and for those periods of the year (if any) for which generation exceeds demand. Some of the deviation between target demand and generation capability may be reduced by scaling up or down the size of the plant and the associated generation (dashed line in Figure 2.4); thus in effect, adding or subtracting base-load power. The remaining energy deficit (small triangular-shaped region in the upper left hand corner of Figure) is accounted for by charging a capability penalty ($/MW) for lost capacity, ΔP_r, and an energy charge ($/MWH) for lost energy. While this deficit can be made up in a number of ways (e.g., additional baseloaded power, peaking power, purchased power, etc.), in this study the costs assigned to these

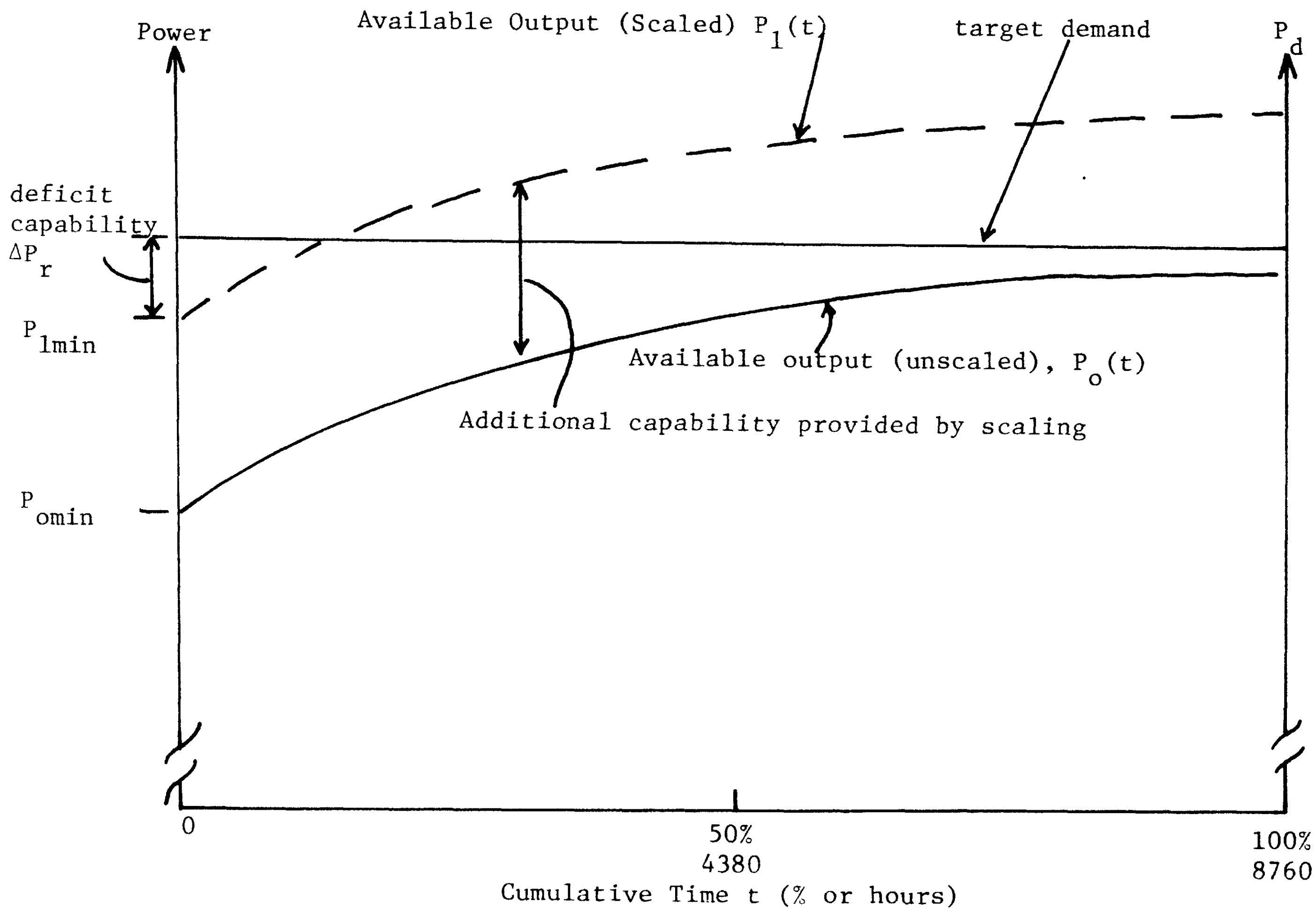


Figure 2.4 Definition Sketch Showing Scaling Concept Used in Study

losses correspond to those for gas turbines. Other types of make-up power are considered in more detail for the case of a dry tower in Choi et al. (1978). During periods when the scaled output exceeds the larger demand, a fuel cost credit is allowed. For a given plant/cooling system, the degree of scaling is chosen to minimize the annual generating costs.

The annualized costs can now be written as

$$AC = (CCP + CCS)\cdot f\cdot AFCR + \int_0^{8760} OPC\cdot \min(P_d, P_1(t))\cdot CF\cdot f\cdot dt$$

$$+ CCR\,\Delta P_r\cdot AFCR + \int_0^{T'} (P_d - P_1(t))\cdot REC\cdot CF\cdot dt \qquad (2.1)$$

where

AC = annual cost ($)

CCP = capital cost of plant (exclusive of condenser and cooling system) ($/MW)

CCS = capital cost of unscaled cooling system ($/MW)

f = scaling factor (of order 1) = P_1/P_o

$AFCR$ = annual fixed charge rate

OPC = operating costs (fuel, water, maintenance, etc.) ($/MWH)

P_d = target demand (MW)

$P_o(t)$ = potential (unscaled) net generation (MW)

$P_1(t)$ = potential (scaled) net generation (MW)

CF = capacity factor

t = time (hrs)

CCR = replacement capacity cost ($/MW)

ΔP_r = $P_d - P_{1\,min}$ (MW)

T' = number of hours per year in which scaled generation is less than target demand

REC = replacement energy cost (\$/MWH)

It can be seen from this formulation that cooling system costs can differ, in general, due to differences in capital costs, differences in operating costs (e.g., auxiliary power requirements) and differences in unscaled net generation. The last differences are reflected in differential capability losses (including scaled capital costs of base-load power and replacement capacity costs) and energy costs (including scaled energy costs and replacement energy costs).

This annualized cost is one of several ways to represent electric generating cost. Annual cost (\$/yr) as expressed in Equation (2.1) allows comparison of plants operating with alternative cooling systems while presupposing knowledge of amount of electric energy produced per year. Net production cost per unit of electricity (e.g. mills/KWH) is another way of representing cost that facilitates comparison of plant/cooling system combination while normalizing electric energy production. The production cost (mills/KWH or \$/MWH) can be arrived at from annual cost by

$$PC = \frac{AC}{8760 \cdot CF \cdot P_d} \tag{2.2}$$

A third way, total present value costs, can be used to measure total cost for any cooling system used mostly for the purpose of a utility's evaluation of alternative investments. This is arrived at by adding capital costs to the sum of all operating and penalty costs discounted over the plant's lifetime.

It should be mentioned, finally, that the scaling concept and the consideration of an optimal combination of base-load and peaking power, are employed so that various systems can be compared relative to a fixed demand. It does not mean that a utility would necessarily build a base-load plant which would not meet its expected demand. This concept is most relevant to dry and wet/dry towers where annual performance is most variable.

2.4 Evaluation of Net Power

Net power for the unscaled systems, $P_o(t)$, is determined by evaluating the cooling system for a discrete set of environmental conditions which are expected to occur at the site (e.g., river temperatures for once-through systems, combinations of wet and dry bulb temperature for natural draft evaporative towers, etc.). The annual performance is then computed by weighting the discrete performances in accordance with the frequency with which each combination of environmental condition occur.

The performance of a cooling system design under any environmental condition is evaluated in conjunction with the concurrent performance of the power plant turbine. For every turbine back pressure and associated heat rate (Figures 2.2 and 2.3) there is a corresponding saturated steam temperature. At a given turbine throttle opening, there is a one-to-one relationship between this turbine steam condensing temperature (SCT) and the turbine heat rejection (BTU/hr). For each environmental condition, an equilibrium is assumed to exist between the rate of heat rejection at the turbine and in the cooling system. The cooling system performance then is measured in terms of the turbine performance at that SCT at which the turbine heat rejection equals the cooling system heat rejection. Determination of this equilibrium involves an iteration as suggested in Figure 2.5. At this point the steam condensing temperature (SCT) can be computed as

$$SCT = T_{ENV} + \Delta T_{CS}$$

$$\Delta T_{CS} = \Delta T_{APP} + \Delta T_o + TTD \tag{2.3}$$

In this equation, the cooling system approach temperature (ΔT_{APP}) represents the difference between the cold water temperature entering the condenser and the environmental temperature (T_{ENV}). The range (ΔT_o) represents the increase in temperature across the condenser and is related to the condenser flow rate (Q_o). The terminal temperature difference (TTD) is the difference between the steam condensing temperature and the hot water temperature leaving the condenser, and is a direct measure of the heat transfer characteristics of the condenser. In general, an increase in the SCT, by increases of any of the above temperature components, increases the turbine back pressure which decreases the work produced in the last stages of the turbine, and thus lowers the efficiency of electricity production ("efficiency derating"). The turbine back pressure, P, is estimated using standard steam tables to be an exponential function of the SCT (in absolute units) according to the following relation:

$$P = \exp(17.168 - 9240/SCT) \tag{2.4}$$

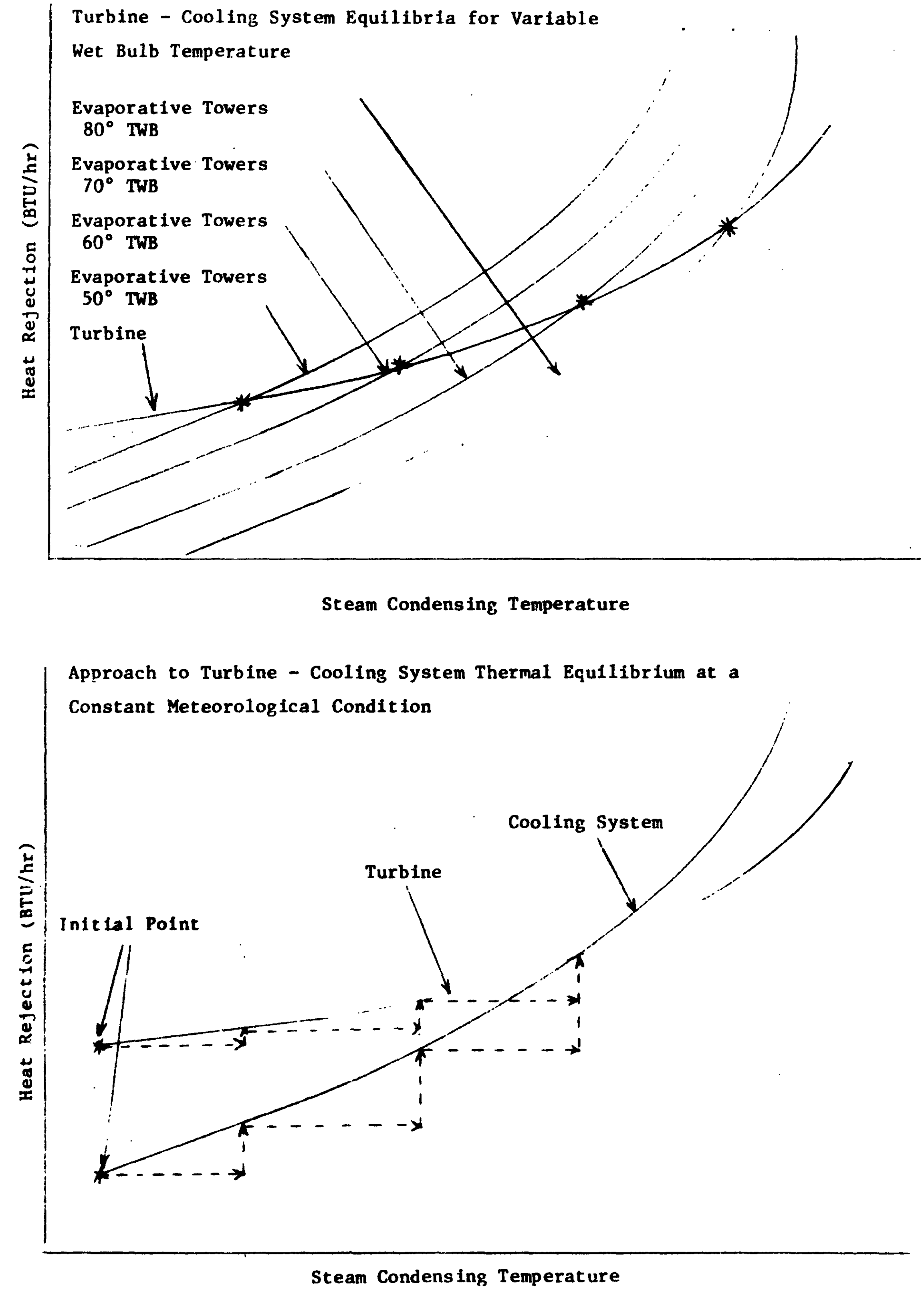


Fig. 2.5 Matching of Turbine and Cooling System Heat Rejections

Figure 2.6 shows the efficiency derating for the 1200 MWe nuclear plant using the conventional turbine. The true net power for this unscaled system is obtained by subtracting from the gross power the auxiliary power requirements (e.g., fans, pumping, etc.) of the cooling system.

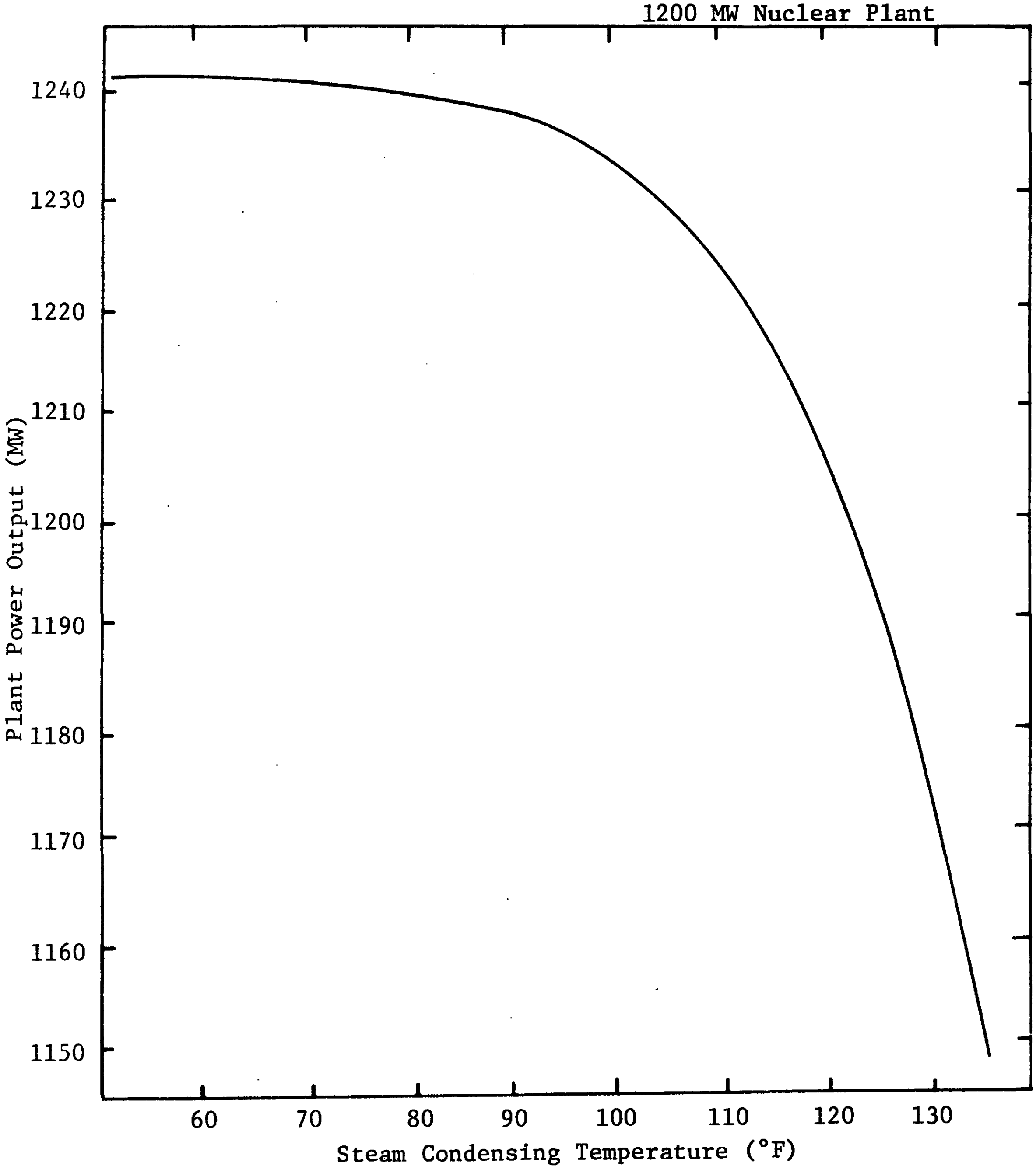
Because most cooling systems being considered (all but ponds) respond rapidly to changes in environmental conditions, this quasi-steady approach provides an acceptable evaluation of cooling system performance and the accuracy can be improved by increasing the resolution of the distribution of environmental temperatures. Special consideration has been given to cooling ponds, in this respect, to account for their large thermal inertia. (See Chapter IV.) In order to compare performance based on discrete distribution with acutal transient calculations, the distribution of environmental conditions were compiled from time series data (see Section 2.5).

2.5 Site Selection

The site chosen for study corresponds hydrologically and meteorologically to that of the Quad Cities Nuclear Power Plant on the Mississippi River (on the border of Iowa and Illinois). This study, however, does not relate to the actual plant at that site. This site was chosen because it was a typical site at which any of the possible cooling systems could be built. The generic nature of this study extends itself to possible application at other representative sites such as large lakes or coastal sites.

The distributions of environmental temperature used in the modeling were compiled from data for stations near Quad Cities. The meteorological data used in this study was obtained from the National Climatological Center for the station at Moline, Illinois ($90^{\circ}31'$ W Longitude, $41^{\circ}27'$

Figure 2.6 Plant Output vs Turbine Exhaust Temperature
1200 MW Nuclear Plant



N Latitude, elevation of 582 feet). The data includes dry bulb temperature, wet bulb temperature, wind speed and cloud cover recorded every three hours for ten years (1961-1970). Figure 2.7 shows a cumulative distribution compiled from this data for the wet bulb temperature and the dry bulb temperature used as input to the wet and dry cooling tower models, respectively. Chapter IV describes how the remaining data is processed for use in the cooling pond optimization model. The distribution of equilibrium temperature (T_E) used in the cooling pond analysis is also shown in Figure 2.7.

The hydrologic data was obtained from the United States Geological Survey for the station at Fulton, Illinois, which is located 25 miles upstream from Quad Cities on the Mississippi River. The data includes daily river temperatures and daily river flow rates for five years (1970-1974) and was used to evaluate the performance of the once-through cooling systems. The cumulative distribution of river temperature is also shown in Figure 2.7.

2.6 Costs Used in the Study

Design comparisons were made for a set of base case economic factors and a number of sensitivity runs were also made. A list of the major factors used in this study is included in Table 2.1; economic details pertaining to individual cooling systems are included in Chapters III-VI. Note that coal was chosen for use as the prime mover in the fossil fuel plant and thus those fuel costs correspond to coal. Also, the power from the nuclear plant was assumed to be generated with a boiling water reactor (BWR).

Figure 2.7 Cumulative Distributions of Environmental Temperatures (for Moline, Illinois site)

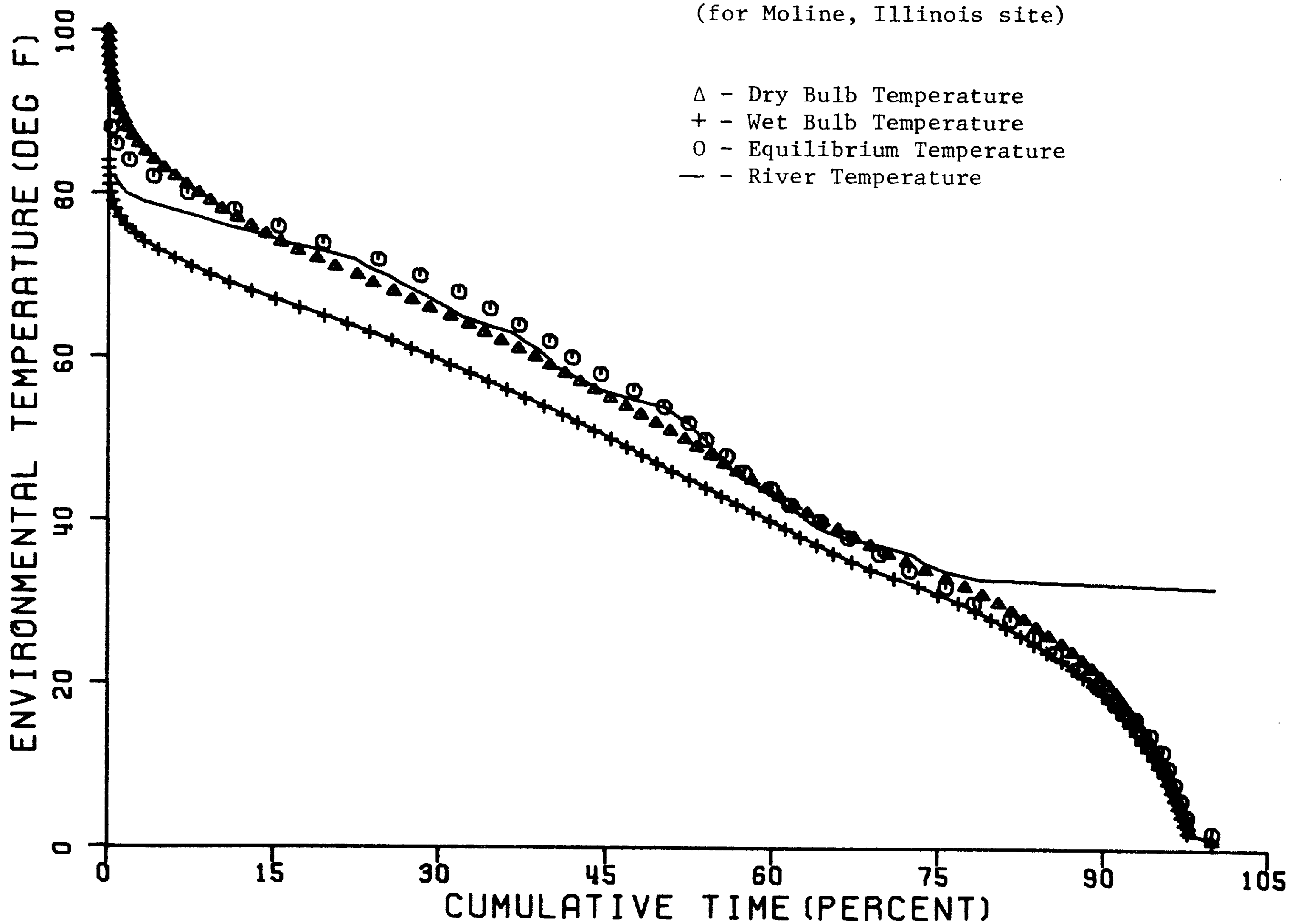


Table 2.1

Basic Economic Factors

Item	Base Case Value	additional values used in Sensitivity Study
Year of Pricing	1977	
Capacity Factor	75%	50
Fixed Charge Rate	17%	15, 20
Annual Operation and Maintenance Cost	1% of all capital costs	
Indirect Costs	25% of all capital costs	
Fuel Cost - Fossil	$0.0031/KWH	$0.0023, $0.0046, $0.0061
- Nuclear	$0.0016/KWH	$0.0012, $0.0024, $0.0032
Plant Construction Costs		
- Fossil	$500/KW	$375, $750
- Nuclear	$600/KW	$450, $900
Replacement Capacity Cost (Gas Turbines)	$160/KW	$120, $240
Replacement Energy Cost (Gas Turbines)	$.03/KWH	$.0225, $.0375 $.0450, $.0600
Operation Horizon (↔ AFCR)	35 years	
Water Cost	$0/1000 gal	$.10, $.50, $1.00
Waste Water Treatment Cost*	$.10/1000 gal	$.05, $.25, $.50
Cooling System Cost Multiplier	1.00	0.75, 1.50

*Once-through system uses $0.0/1000 gal as base case.

Chapter III

ONCE-THROUGH SYSTEMS

3.1 Introduction

In open cycle, or once-through, cooling systems, water is removed from its source, is pumped through the condenser in one or more passes to receive rejected heat, and is then returned to the water source. The waste heat is ultimately transferred to the atmosphere from the water body by a combination of radiation, conduction and evaporation. In a well-designed system none of the warm water which is discharged to the receiving water recirculates to the intake, eliminating the approach temperature (ΔT_{APP}) inherent in closed cycle cooling systems and thus leading to greater operating efficiencies. Because their efficiency is generally higher, and their operating and capital costs are generally lower than the equivalent closed cycle system, once-through systems are economically preferable for sites where sufficient water is available. Thus, in general, other systems have been used only when sufficient water for once-through cooling is not available or environmental considerations have prevented once-through cooling.

Flows for a single base-load unit range from 200,000 to 1,000,000 gpm. Pumps required to circulate this water through the condenser are normally located near the intake structure. Usually there are several pumps for each unit due to the large flows and the requirement of providing a high degree of flexibility and safety in the plant operation. The discharge from the condenser can be returned to the source via a canal or a pipe depending on the location and/or the degree of mixing which is desired. The circulation of condenser cooling water, and the resulting heated discharge can impact the aquatic environment in a number of ways as discussed in the following section.

3.2 Environmental Factors

3.2.1 Ecological Effects

The impact of a once-through cooling system on the aquatic environment can be broken into two distinct categories: (1) effects due to organism impingement and entrainment at the plant intake and (2) biological, physical and chemical effects which result from elevated temperatures within the plume. Figure 3.1 shows the potential physical locations for biological damage from a once-through system.

On the intake side, entrainment is the passage of relatively small organisms (e.g. eggs and larvae) through the condenser cooling system. Entrainment mortality is not caused by the intake structure but rather thermal, physical and chemical effects within the cooling system. The number of organsims that are entrained is a function of intake design and location as well as condenser flow rate. An important reference on the effects of organism entrainment is a recent book edited by Schubel and Marcy (1978). Impingement, on the other hand, is the forcing of nektonic species and, in some cases, benthic shellfish such as clams and shrimp, against a screen mesh by velocity forces produced by the water flowing through the screen. For the intake as a whole, impact is a function of a number of variables including the age and species distribution of

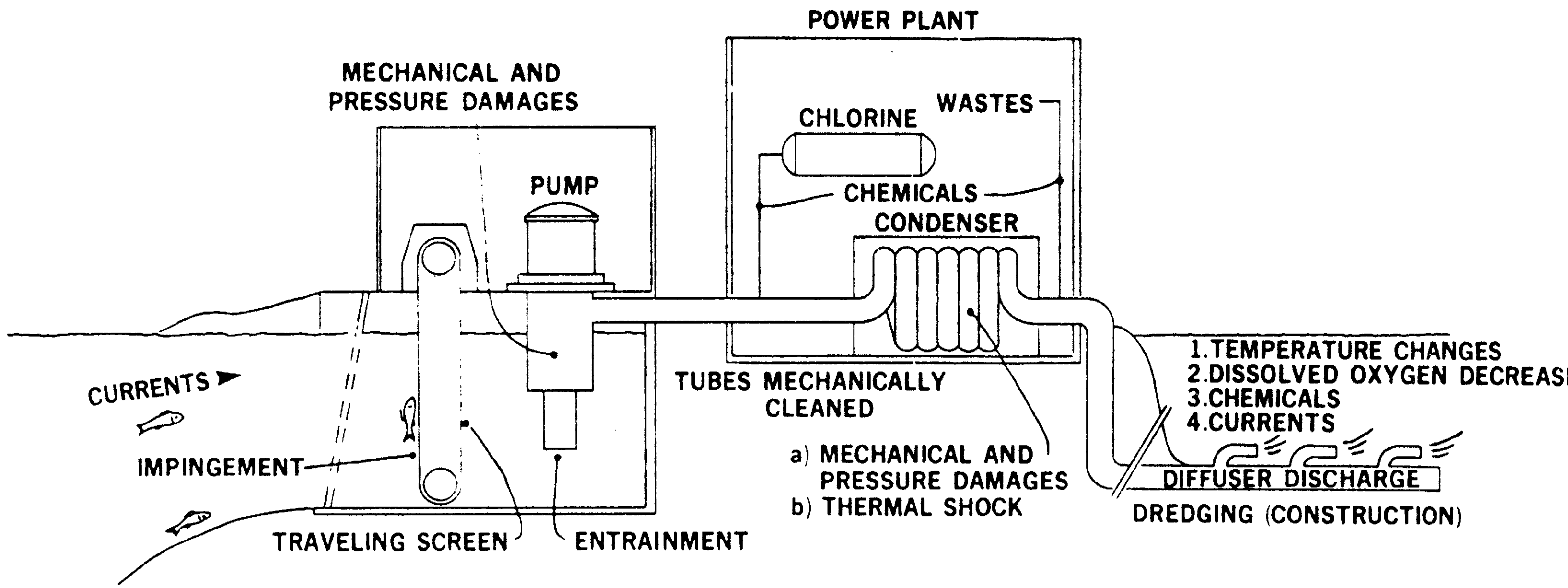


Figure 3.1 Sources of Potential Biological Damages from Once-Through Cooling (from MacLaren, 1975)

of organisms in the receiving water, the volume flow rate and intake approach velocity of the circulating water, and the internal design of, and in particular the time of travel, associated with the cooling system. From the standpoint of intake considerations alone, the preferable system would involve a low flow rate (high ΔT_o), a low intake velocity (relatively large intake structure) and a short outfall pipe or channel to minimize the time of travel.

On the discharge side, impact is associated primarily with organism entrainment into the discharge plume which, in turn, is controlled largely by the design of the outfall structure. Organisms respond to a temperature rise with increased metabolism, lowered resistance to toxic substances and greater need for oxygen. For long enough exposure at high temperatures, mortality occurs. While thermal stresses have received the most attention, physical and chemical stress may be present as well and the types of biological assays used for intake entrainment analysis would also be appropriate for plume entrainment.

It is clear that an understanding of the effects at both the intake and the discharge depends on the specific ecological environment near the plant and that therefore, the most desirable design may vary from site to site. However, as a generalization, low flow rates are preferable from the standpoint of intake impact, while a high flow rate results in a lower ΔT_o and thus lower plume temperatures. Thus in selecting a condenser flow rate, a trade-off exists between intake and outfall considerations as suggested in Figure 3.2a.

Plume temperatures are also very much affected by outfall design. The simplest design, consisting of a low velocity surface discharge, would not

Figure 3.2 a Qualitative Variation of Once-Through Impacts Versus Condenser Flow Rate

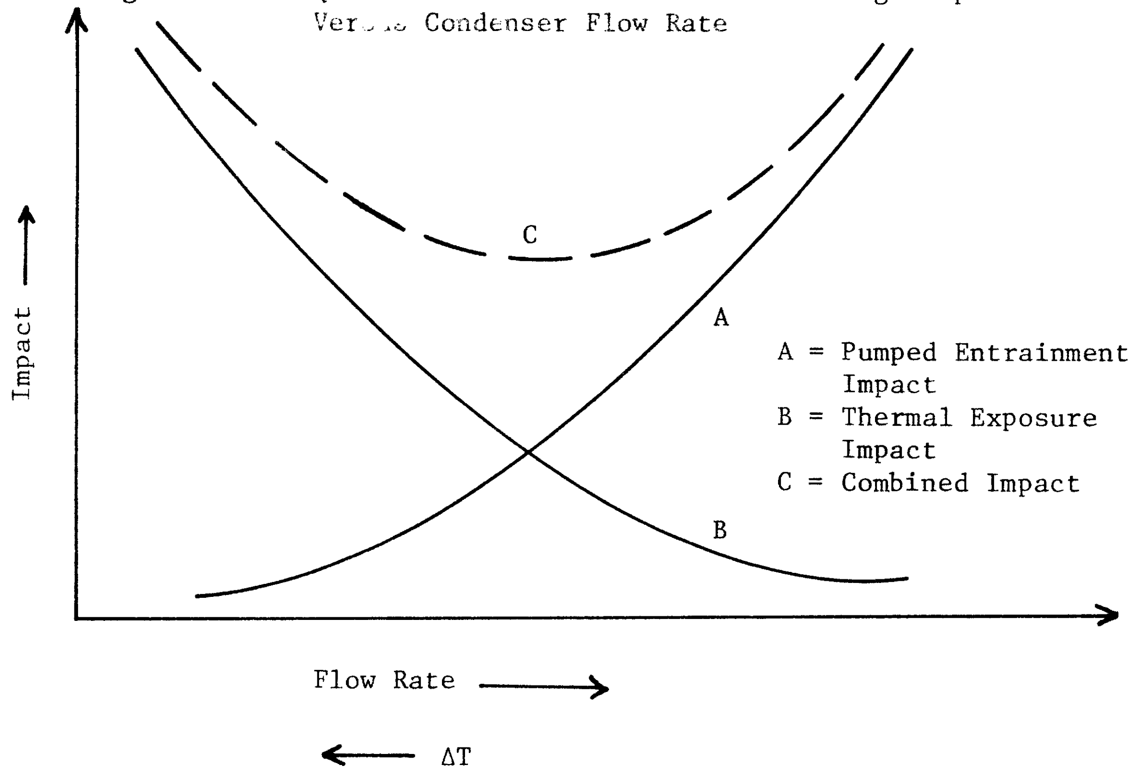
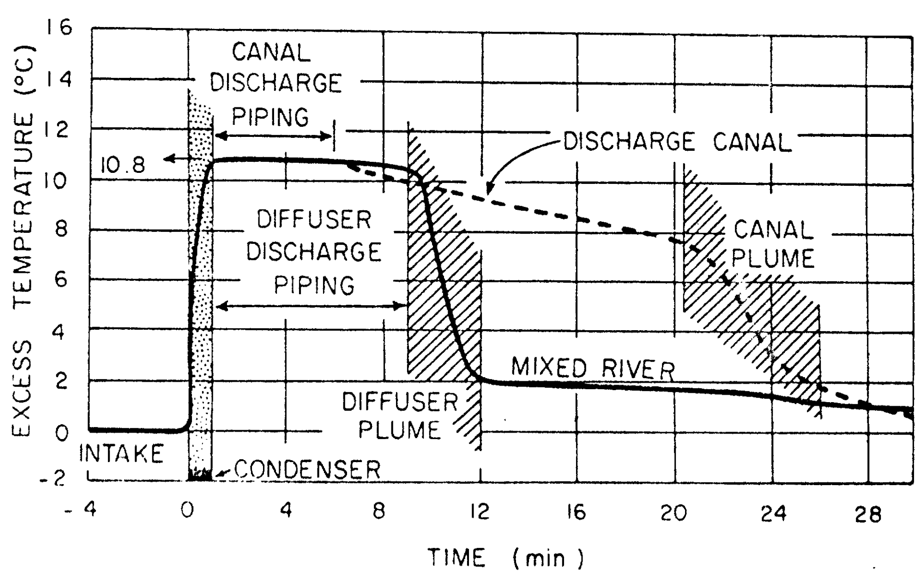


Figure 3.2 b Example Relationship Between Excess Temperature and Exposure Time as a Function of Discharge Mixing (from Coutant, 1970 as reported in Schubel, 1978)



induce much mixing and thus would result in relatively high surface temperatures near the outfall. At the other extreme, discharge through a submerged multi-port diffuser may result in appreciable mixing yielding lower induced temperatures, but involving much greater volumes of water. Alternatively, if one were to compute temperature versus time of exposure for an organism entrained in the plume, the relationship may look like Figure 3.2b. The choice between relatively small volumes of water (or short exposure time) at relatively high temperature, versus larger volumes (and long exposure time) at lower temperatures involves a number of site specific factors.

3.2.2 Legal Aspects

The National Environmental Policy Act of 1969 (NEPA) sets the stage for national consideration of thermal as well as other effluents. In what may be termed an environmental Bill of Rights, NEPA sets forth a broad national policy to "encourage harmony between man and his environment" (PL91-190). NEPA takes the major step of requiring all Federal agencies to consider values of environmental preservation in their spheres of activity.

The Federal Water Pollution Control Act Ammendments of 1972 (FWPCAA), Public Law 92-500, which followed closely behind NEPA, has as its objectives "to restore and maintain the chemical, physical and biological integrity of the Nation's waters." The Act provides in Section 301(a), under Subchapter III - Standards and Enforcement, "that the discharge of any pollutant is unlawful unless it is in compliance with conditions or effluent limitations contained in a permit issued under Section 402.

The effluent guidelines and standards under Subchapter III of the Act have been designed for issuance on three separate levels: (1) The Best Practical Control Technology Currently Available (BPCTCA or BPT), which existing plants should have met by 1977, (2) The Best Available Demonstrated Control Technology (BADCT), which new plants must meet upon startup; and (3) The Best Available Control Technology Economically Achievable (BACTEA), which all plants must meet by July 1, 1983.

The intent of FWPCAA in setting up these increasingly stringent restrictions on discharge of contaminants is to attain the Act's goal of zero-pollutant discharge, for at least some source types, by 1985. It remains to be seen whether thermal discharges are one of the source types capable of achieving zero discharge by 1985.

With specific regard to thermal discharges, Section 316(a) allows a particular power plant, on an ad hoc basis, exemption from thermal control requirements. The 316 exemptions are permitted by the Environmental Protection Agency (EPA) for those plants whose owners can demonstrate that any effluent limitations proposed for the control of the thermal component will require effluent limitations "more stringent than necessary to assure the protection and propagation of a balanced, indigenous population of shellfish, fish and wildlife in and on the body of water." In that case, the Administrator will impose a specific effluent limitation on thermal discharges for that particular plant.

The Act also states in Section 316(b): Any standard applicable to a point source shall require that the location, design, construction and capacity of cooling water intake structures reflect the best technology available for minimizing adverse environmental impact. Thus, it is clear that Section 316 of the Act is of particular importance for the continuation and future design of uses of once-through cooling systems.

In addition, the EPA in 1974 enacted Effluent Guidelines and Standards for Steam Electric Power Generating (40 CFR 423) and Regulations on Thermal Discharges (40 CFR 122) as supplements to the FWPCAA to better control the specific discharges (heat, blowdown, etc.) from steam electric power plants.

These federal effluent limitation regulations are determined by the EPA and are applied alongside state "standard-setting" regulations (previously established) where the water quality criteria in any state will be stricter of the two. In general, the established temperature standards set by state regulatory agencies all permit a "reasonable" but undefined area for mixing beyond the point of discharge to be exempted from the established standards.

As a case study of temperature standards, our site on the Mississippi River at Quad Cities bordering both Iowa and Illinois will be investigated. Since all waste heat discharged from a power plant at this site must comply with the thermal criteria from both these states, combining their regulations yields the following (Parr, 1976):

Definition: The mixing zone is the area of diffusion of an effluent in the receiving water and Water Quality Standards shall be applied beyond the mixing zone.

Regulations: The mixing zone may not contain more than 25 percent of the cross-sectional area or volume of flow at any cross-section, and temperature increases outside the mixing zone may not exceed $5^{\circ}F$.

The Illinois regulations specifically state, in addition, that no mixing zone shall exceed the area of a circle with a 600 ft radius (approximately 26 acres).

Other Specifications: The rate of temperature change shall not exceed $2^{o}F$ per hour.

Maximum River Temperature: Water temperature shall not exceed the maximum monthly limits shown in the table below during 1% of the hours in the 12-month period ending with any month. Moreover, at no time shall the water temperature at such locations exceed the maximum limits in the table by more than $3^{o}F$.

Table 3.1 Temperature Standards at Study Site

Month	Temperature (^{o}F)
January	45
February	45
March	57
April	68
May	78
June	85
July	86
August	86
September	85
October	75
November	65
December	52

3.2.3 Control of Environmental Impacts

From the previous discussion it is clear that the optimization of a cooling system under environmental constraints requires site specific biological information and extensive analytical tools. While this type of optimization was not done in this study, intake and outfall controls were considered as follows.

The intake structure and canal are designed to minimize organism impingement by keeping the approach velocity low. This is generally accepted as 1 fps up to the intake screens and 0.5 fps up to the canal in the fish escape passages (MacLaren, 1975), although intake velocity should more specifically be based on values determined for the "important" species at each site. As flow rates increase, the size of the intake structure and canal increase to keep the velocities low at the expense of higher capital costs. Since the environmental impact at the outfall is a function of the condenser flow rate and the outfall design, several different flow rates and outfall designs were considered. Figure 3.3 shows the relationship of ΔT_o vs. $\rho c Q_o$ for single fossil and nuclear units as calculated by the model.

The type of outfall should be selected as a function of the desired temperature distribution. A surface discharge canal, as shown in Figure 3.4 provides the most economical means of discharge. The induced temperature rise due to discharge through a surface canal was calculated using the three-dimensional heated surfaced discharge model developed by Stolzenbach, et al. (1972). For the river site considered in this study, the discharge structure was assumed to be a rectangular open channel oriented 90 degrees to the river flow as shown in Figure 3.4. The model treats the discharge as a buoyant surface jet characterized by a reduction in vertical entrainment and an increase in lateral gravitational spreading. This results in a velocity and temperature distribution which is much wider than deep with increased surface area which may lead to significant surface heat loss.

For a given condenser flow rate, Q_o, river velocity, V, temperature rise across the condenser, ΔT_o, and discharge canal velocity, u_o, the model can

Figure 3.3 ΔT_0 vs Q_0 for Once-Through Systems

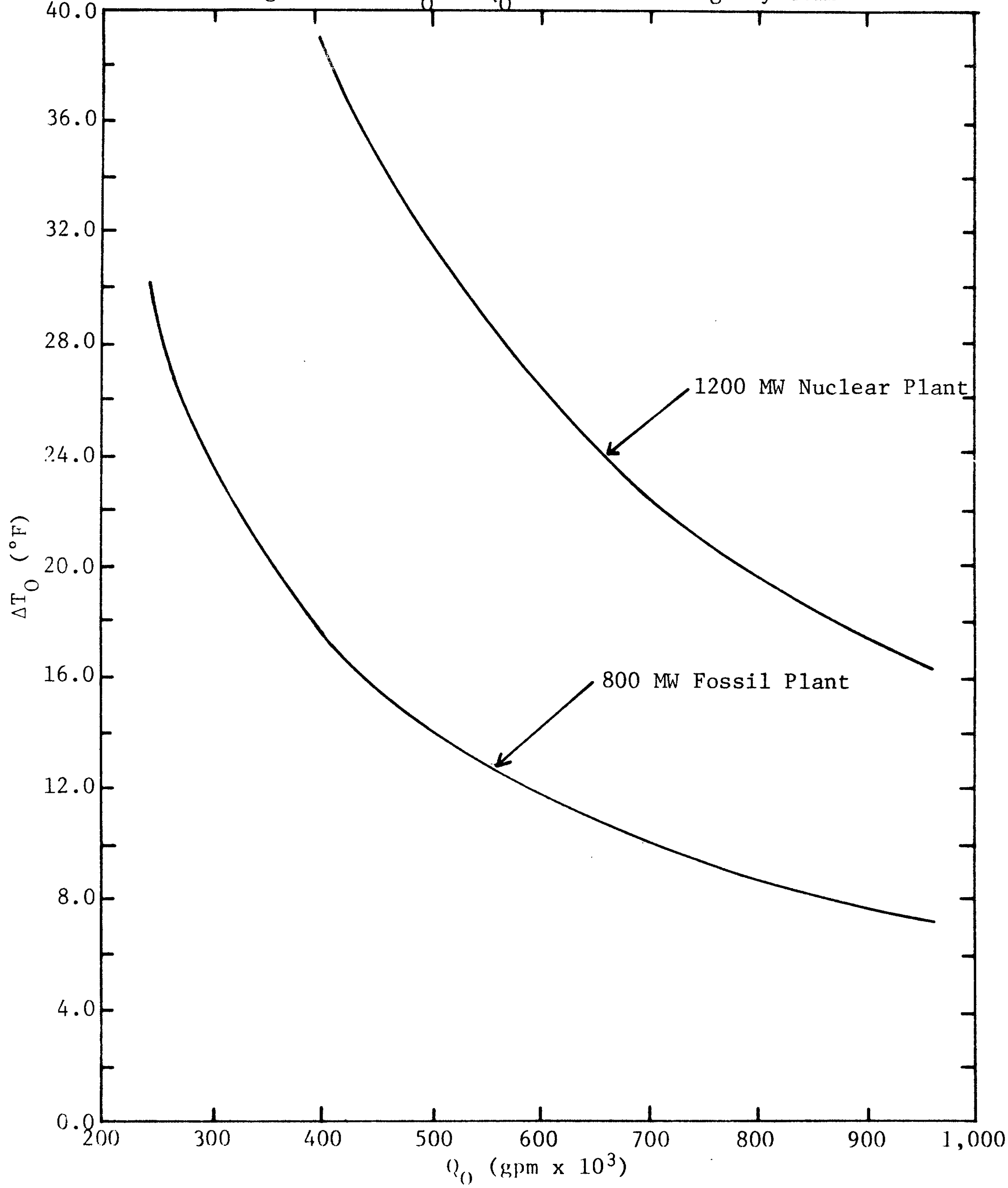
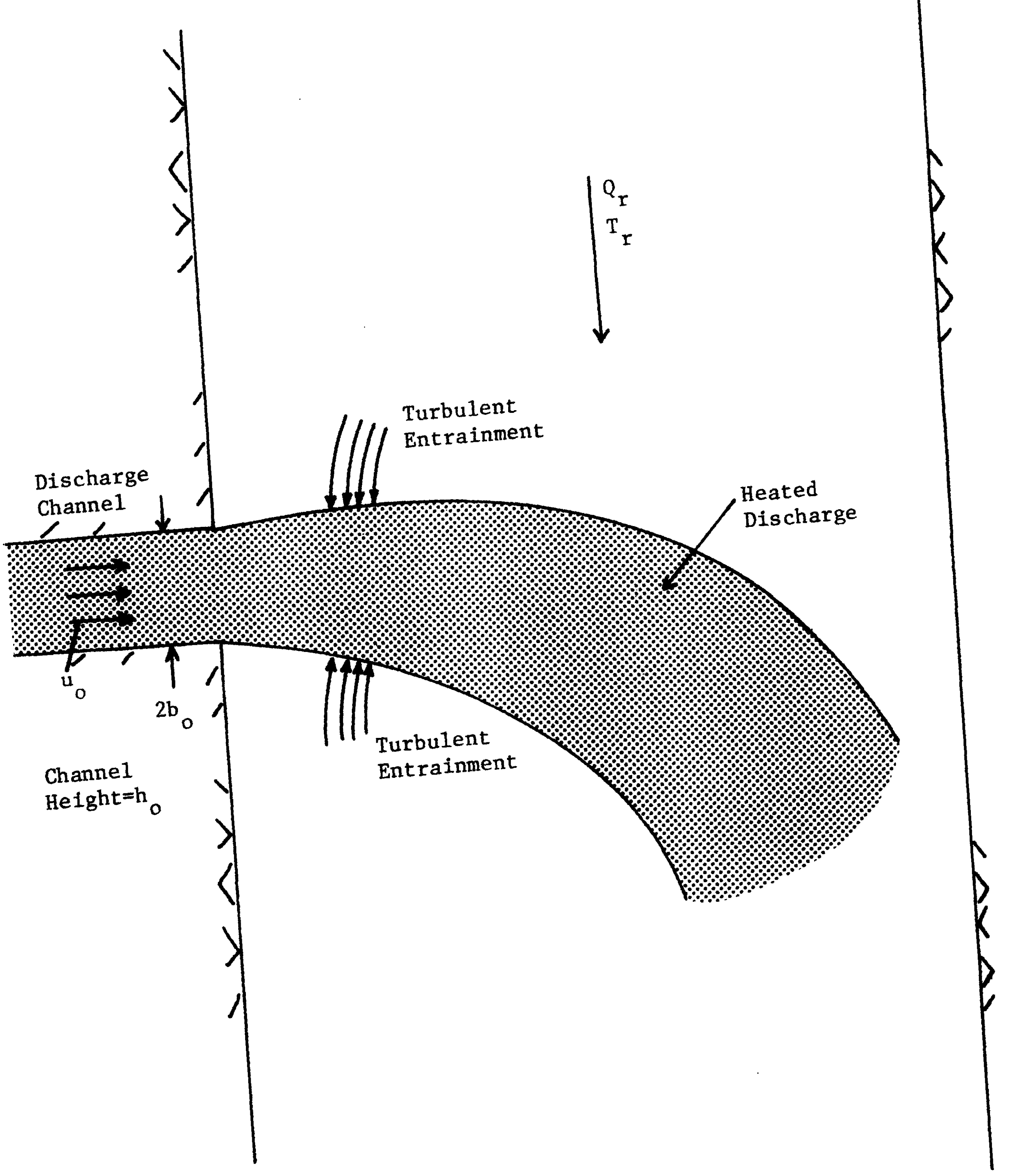


Figure 3.4 Schematic of Heated Surface Discharge



be used to compute the mean field temperature distribution. However, for purposes of comparison with the submerged diffuser predictions, only the temperature at the edge of the mixing zone was computed. This temperature is designated T_m and is computed as

$$T_m = T_r + \Delta T_m \tag{3.1}$$

with

$$S_m = \frac{\Delta T_o}{\Delta T_m} = \begin{cases} 1.4\sqrt{\mathbb{F}_o'^2 + 1} & \text{for } S_m Q_o < Q_r \\ \dfrac{Q_r}{Q_o} & \text{for } S_m Q_o > Q_r \end{cases} \tag{3.1a}$$

where

ΔT_m = averaged temperature rise after mixing (°F)

T_r = ambient (river) temperature (^oF)

$$\mathbb{F}_o' = \text{a densimetric "Froude number"} = \frac{u_o}{\sqrt{g\ \beta\ \Delta T_o (h_o b_c)^{1/2}}} \tag{3.1b}$$

g = gravitational acceleration (ft/sec^2)

h_o = canal depth (ft)

b_o = canal half-width (ft)

β = coefficient of thermal expansion ($^oF^{-1}$) = a function of T_r and ΔT_o.

The predicted mixed river temperatures are evaluated for a canal with exit velocity, u_o=2.5 fps. For their calculations, F_o' is based on an average river temperature, T_r = 57 °F.

Submerged multi-port diffusers provide a more efficient means of diluting the heated discharge and thus minimizing the size and temperature of the mixing zone. The actual size and temperature of the diluted plume depends on the hydrological characteristics of the water body and the diffuser design. For river sites, the preferable design is a co-flowing diffuser in which the diffuser pipe extends across a portion of the bottom of the river and the many discharge nozzles point downstream. Figure 3.5 shows a sketch of this type of diffuser along with the type of induced temperature patterns.

The effectiveness of co-flowing diffuser designs depends on the ΔT_o leaving the condenser, the river flow rate Q_r, condenser flow rate Q_o, diffuser length L, diffuser exit velocity u_o, and the river cross-sectional characteristics. These relate to the mixed temperature rise, ΔT_m according to a formula given by Adams (1972):

$$S_m = \frac{\Delta T_o}{\Delta T_m} = \begin{cases} \frac{1}{2}\left\{\frac{u_r HL}{Q_o} + \sqrt{\left(\frac{u_r HL}{Q_o}\right)^2 + \frac{2HLu_o}{Q_o}}\right\} & \text{for } S_m Q_o < Q_r \\ \frac{Q_r}{Q_o} \quad \text{for } S_m Q_o > Q_r \end{cases} \qquad (3.2)$$

where

H is the average river height

u_r is the average river velocity (related to Q_r and the river cross-section)

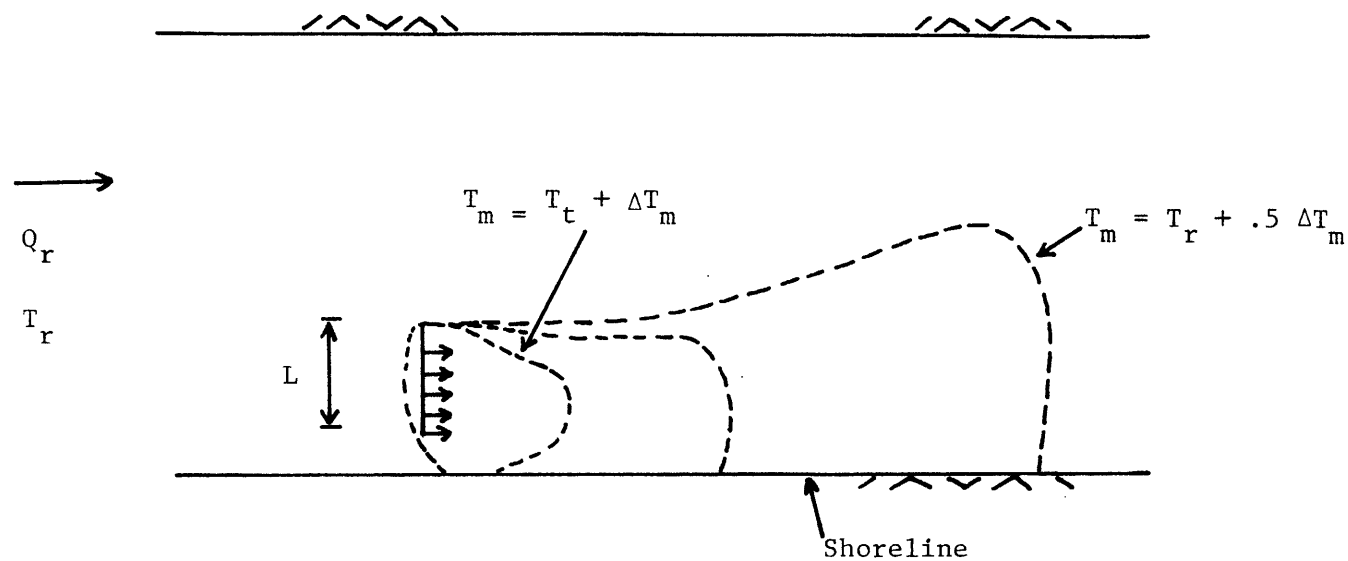


Figure 3.5 Unidirection Diffuser in a River

S_m is the dilution defined as the ratio of condenser rise to induced temperature rise at the end of the mixing zone

By developing a bi-variate distribution of ambient river temperature and flow rate using historical hydrological data, one can evaluate the effect on the expected distribution of ΔT_m for various diffuser and condenser designs (different u_o, L, Q_o and ΔT_o).

This has been done for four different diffusers using two different flow rates. The lengths and velocities of the four diffusers were:

$L = 500$ ft, $u_o = 10$ fps

$L = 500$ ft, $u_o = 20$ fps

$L = 1{,}500$ ft, $u_o = 10$ fps

$L = 1{,}500$ ft, $u_o = 20$ fps

The flow rates were

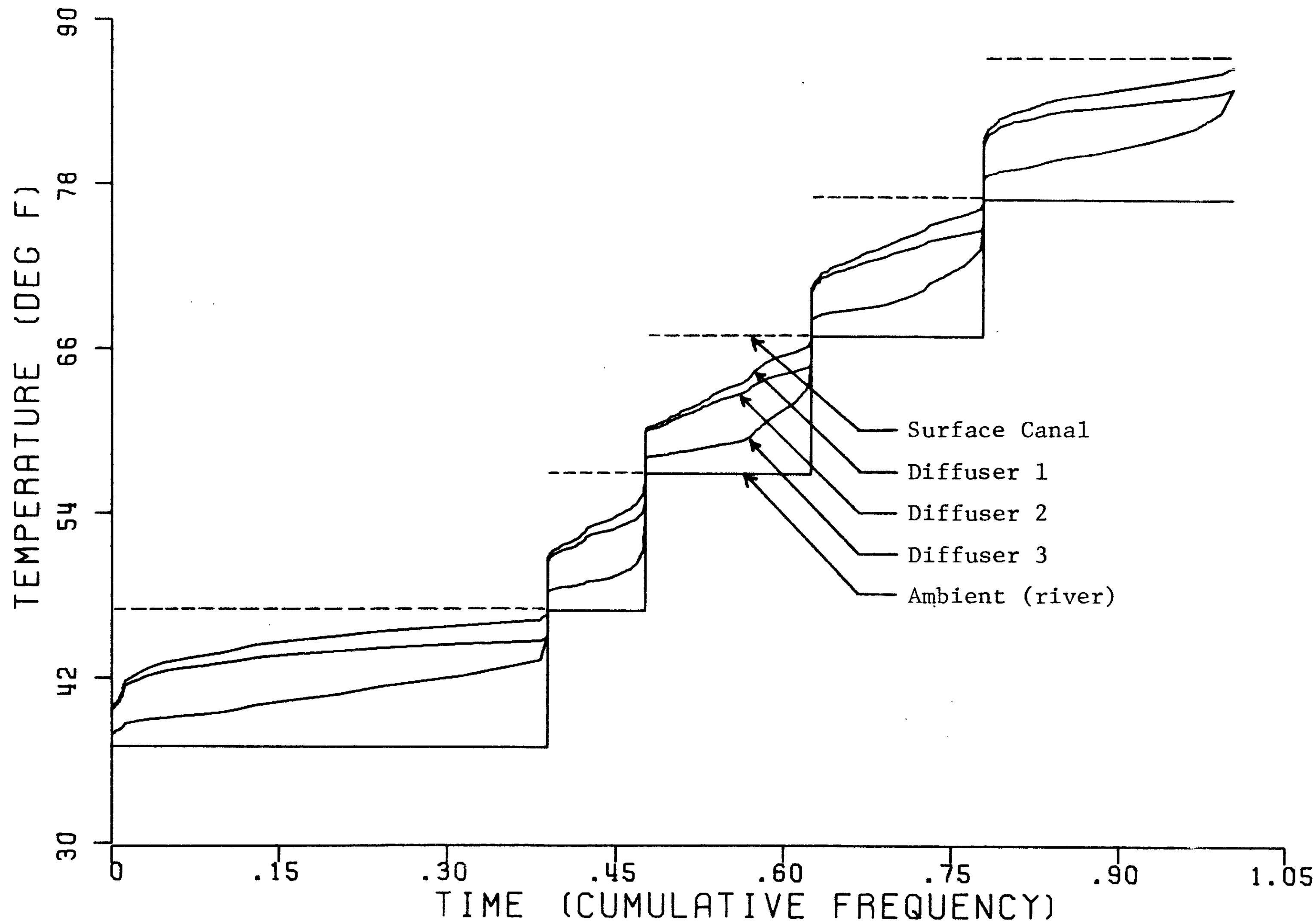
$Q_o = 1{,}760{,}000$, $\Delta T_o = 35^oF$

$Q_o = 2{,}800{,}000$, $\Delta T_o = 22^oF$

These values correspond approximately to the heat rejection from four 1200 MW unclear units as considered in this study. To compile the distribution of river flow rate and ambient temperature, increments of 5000 cfs and 10^oF were used.

The resulting cumulative distributions of ambient and induced river temperatures for several of the design combinations as well as for the surface discharge canal are shown in Figure 3.6 and the information is

Figure 3.6 Open Cycle Induced Temperatures



summarized in Table 3.2. This information allows one to tell if ΔT_m ever exceeds an induced temperature standard and for how long it is exceeded. It can also determine if an upper limit on temperature in the river is ever exceeded and for what duration. It is clear that the diffusers, in general, provide better dilution than the surface discharge and that for the diffusers, increasing Q_o, L and u_o all serve to lower the induced temperatures. It will also be clear in Section 3.4 that the cost of the diffusers is greater than the surface canal and that the cost increases monotonically with increasing u_o and L while there is an economically desirable intermediate value of Q_o. Of these three variables the diffuser length L has the greatest effect on lowering ΔT_m because a longer diffuser is able to intercept more river flow as well as to induce more momentum.

3.3 Thermodynamic Performance Model

Thermodynamically the once-through system is the most efficient, the simplest and the most predictable in response to environmental input. For this site the river temperature is input directly as the environmental temperature (T_{ENV}) feeding the condenser and represents the lowest temperature in the Rankine cycle. The T_{ENV} and the turbine steam condensing temperature (SCT) are related on a one-to-one basis:

$$SCT = T_{ENV} + RANGE + TTD \tag{3.3}$$

Table 3.2 Open Cycle Outfall Designs

	Ambient (river)	Surface Discharge	Diffuser 1	Diffuser 2	Diffuser 3
Length, L (ft)*	-	1,000	500	500	1,500
Exit Velocity, u_o(fps)**	-	2.5	10	10	10
Condenser Flow Rate, Q_o (gpm)	-	2,800,000	1,760,000	2,800,000	2,800,000
Condenser Temperature Rise ΔT_o (°F)	-	22.5	22.5	22.5	35
Lowest Mixed Temperature (°F)	32	38.8	34.7	34.7	32.9
Maximum Mixed Temperature (°F)	82	92.6	91.6	90.1	90.0
Maximum Temperature Rise (°F)	0	10.6	9.6	8.1	8.0
Amount of time exceeding 5°F temperature rise	-	100%	86%	82%	14%

* channel length for surface discharge; diffuser length for diffusers

** canal velocity for surface discharge

The terminal temperature difference (TTD) is a property of the condenser design and is considered a constant, 5^oF. The range (ΔT_o) for a specific power plant is related simply by

$$\Delta T_o = \frac{J}{\rho C_p Q_o} \tag{3.4}$$

where

J = waste heat rejection rate

Q_o = condenser flow rate

ρC_p = heat capacity of water

ΔT_o = range (^oF)

Figure 3.3 shows the relationship between ΔT_o and Q_o for the nuclear and fossil units. Note that because plant efficiency varies slightly with SCT, J is not constant.

3.4 Optimization Model

The cooling system costs for the once-through system include intake structure, intake canal, condenser pumps, and either dishcarge canal and structure or discharge pipe and diffuser. The scaled optimization procedure has been described in Chapter II. The chief design parameter is the condenser flow rate, Q_o, which is optimized economically on the basis of trade-offs between capital and operating costs, reflecting thermodynamic efficiencies and power requirements.

The intake structure and canal were sized at each flow rate for the design velocity of 1 fps and 0.5 fps, respectively. Hence, the intake capital costs as well as the pumping costs increase with flow rate. The length of intake canal as well as outfall line was assumed to be 1,000 ft. each way to be consistent with the other models. The cost of constructing an intake structure and canal was on the order of 4 million dollars for the nuclear plant and 3 million dollars for the fossil plant comprising about 20% of total cooling system cost.

The condenser size as described in Chapter II increases with increasing flow rate across it. It is priced by surface area ($8/ft) and for the optimal once-through system is $8.5 million for nuclear and $4.5 million for fossil. This constitutes about 45% of the total once-through cooling system costs.

The outfall canal and structure has less complexities in construction (no screens, etc.) and are smaller than the intake canal. These are therefore less expensive and cost about $1.5 million for the 1,000 ft canal length for both fossil and nuclear plants (Shiers, 1973; EPA, 1976).

The pump capital costs for the once-through system are about $3.6 million for nuclear and $2.8 million for fossil plants at the optimum flow rates without diffusers. This is about 20% of the total cooling system capital costs; replacing the discharge canal with a 1,000 ft pipe and various lengths of diffuser makes the capital costs of pumps 2% to 5% higher. Capital and operating costs of pumps increase in the presence of diffusers due to larger frictional and exit losses.

The diffuser pipe was to be made of corrugated steel and to be semi-buried along the river bottom. Installed costs, based on Acres

American (1976), were \$3,000 to \$5,000/ft for pipe diameters ranging from 6 to 20 feet. The diffusers were sized according to the condenser flow rate keeping velocity through the diffuser constant at 10 fps (optimized externally). The costs were arrived at through the summation of material and construction costs for all the diffuser parts.

To accurately cost the once-through system with diffusers, it is necessary to evaluate the energy losses through the diffuser. These include pipe friction and exit losses and were computed according to French (1972), considering a constant diameter pipe and assuming an infinite number of ports. The total head at the entrance to the diffuser can be expressed as:

$$E(0) = E(L) + \frac{1}{3} \frac{V_o^2}{2g} f \frac{L}{D} \tag{3.5}$$

where

$$E(L) = \frac{u_o^2}{C_D 2g} \tag{3.6}$$

and
E(0) = total head through the diffuser measured at x = 0 (ft)
E(L) = head due to losses at the exit, x = L (ft)
u_o = diffuser exit velocity at the port (ft/sec)
V_o = initial velocity through diffuser (ft/sec)
f = Darcy pipe friction factor, assumed constant
L = length of diffuser (ft)
D = diameter of diffuser (ft)
g = gravitational acceleration (ft/sec^2)
C_D = discharge coefficient, assumed constant

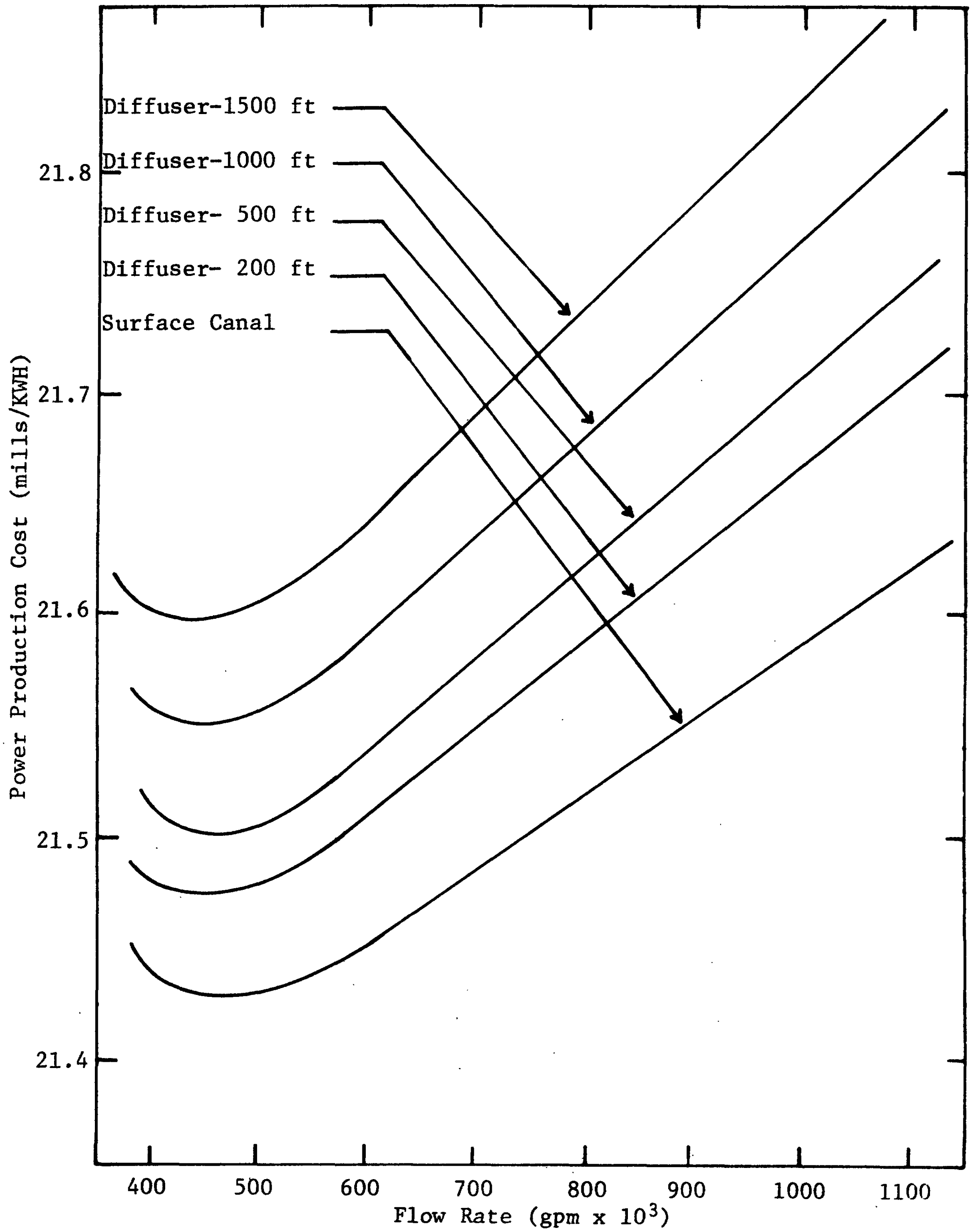
Equation 3.5 is arrived at through our analysis which shows that despite a constant diameter diffuser, there is not much difference in flow between the first and last nozzle (x = 0 and x = L, respectively). Discharge coefficient, C_D, is evaluated as .63 for a sharp-edged port chosen for design in this study (Pearce, 1968). The friction factor, f, is evaluated from Daily and Harleman (1970) as:

$$1/f \quad = \quad -2*\mathrm{Log}(\frac{0.00015}{D}) \quad + 1.14 \tag{3.7}$$

The capital and operating costs involved with overcoming this head is illustrated in Figure 3.7 for the unscaled nuclear plant with once-through cooling modified with diffusers ranging in length from 200 ft to 1,5000 ft. An exit velocity of 10 fps is assumed in each case. This is consistent with the diffuser model results shown in Section 3.2.3. Since each diffuser design induces a different mixed river temperature, the choice of which diffuser is used ultimately depends on the environmental constraints applied at the site.

While it does not play a part in the optimization, water loss due to forced evaporation for a once-through system was calculated for comparison with the other cooling systems. In principle this water loss should be computed by first computing the total (forced plus natural) evaporation and then subtracting the computed natural evaporation. The first calculation, in turn, requires that the induced temperature field resulting from the condenser water discharge be computed. However, because the ability to use once-through cooling presupposes a large water supply, water consumption is not as critical as for closed cycle

Figure 3.7 Once-Through System with Various Outfall Designs
Unscaled Nuclear Plant



systems and therefore a simpler approach is taken.

The approach follows Stolzenbach (1971) and computes the evaporative flux Q based on the concept of the surface heat exchange coefficient K (discussed further in Section 4.2). The results are expressed in Figure 3.8, which give Q_E in terms of the discharge variables Q_o and T_o and the meterological variables of wind speed and water surface temperature. The monthly average evaporation rates for the 1200 MWe nuclear plant was computed based on the monthly average water surface temperatures and wind speeds for 1970 and are presented in Chapter VII for comparison with the other cooling systems. The computed annual average forced evaporation was 16 cfs. This is about 50% of the total of approximately 33 cfs which would have been necessary had the total heat load been dissipated by evaporation. The latter figure was evaluated by dividing the heat load per pound of water, $CQ_o\Delta T_o$, by the latent heat of vaporization given by

$$L = 1087 - .54T_S \qquad \text{(BTU/lb, } T_S \text{ in } {}^{o}\text{F)} \qquad (3.9)$$

Although heat loss through evaporation from once-through cooling is slightly less when using submerged diffusers than when using surface canals, it is assumed here that the forced evaporation is the same.

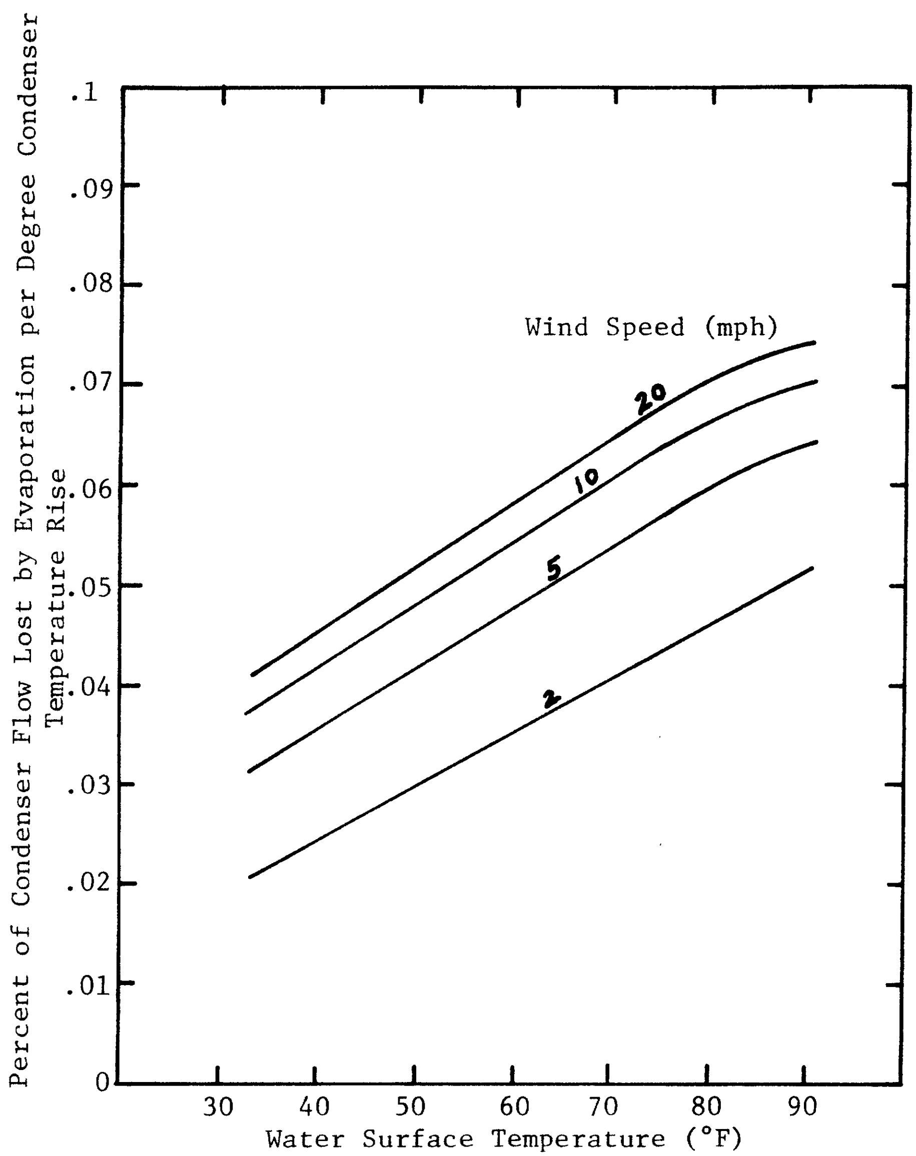


Figure 3.8 Evaporative Losses Induced by Heated Discharges (from Stolzenbach, 1971)

3.5 Results

The primary design variables which we have considered in the evaluation of the optimum once-through system are condenser flow rate Q_o, and outfall type. A surface discharge canal and several multi-port diffuser designs were analyzed for both environmental and economic preference. Figure 3.7 has shown that on the basis of economics alone, the surface discharge canal is the most desirable outfall type over the range of flow rates. Using minimization of cost as our focus, the remaining results in this section will consider a constant outfall type (surface canal) and will vary only the flow rate.

Tables 3.3 and 3.4 show the total power production cost through the summation of various cost components for the once-through system for the nuclear and fossil plants, respectively. Minimum cost is reached by changing the primary variable Q_o, and summing the cost components. The optimum flow rates were found to be: nuclear plant Q_o = 675,000 gpm; fossil plant, Q_o = 375,000 gpm.

As Q_o is increased in the tables, the capital cost of the once-through cooling system clearly increases while the replacement capacity and the operating costs (base-load fuel and replacement energy) are seen to decrease. At the higher flow rates the power plant is thermodynamically more efficient but more energy is needed for auxiliary (e.g. pumping) power. Thus, as Q_o increases, the base-load fuel cost may increase or decrease depending on which factor is more significant; however, the replacement energy and capacity costs will decrease due to the greater efficiencies of the base load plant at extreme hydrologic conditions.

Table 3.3 Power Production Cost Versus Flow Rate-1200 MW Nuclear Plant

Cost Components (mills/KWH)	Flow Rate, Q_o (gpm x10^3)				
	400	500	600	700	800
Plant Construction	14.929	14.934	14.934	14.944	14.949
Cooling System	0.306	0.342	0.375	0.407	0.438
Replacement Capacity	0.184	0.101	0.061	0.041	0.024
Fuel	4.828	4.830	4.832	4.833	4.835
Replacement Energy	0.255	0.112	0.057	0.032	0.021
Maintenance	0.680	0.678	0.678	0.679	0.680
Total Power Production Cost	21.182	20.997	20.940	20.936	20.947

Table 3.4 Power Production Cost Versus Flow Rate-800 MW Fossil Plant

Cost Components (mills/KWH)	Flow Rate, Q_o (gpm x10^3)				
	200	300	400	500	600
Plant Construction	12.753	12.722	12.696	12.670	12.676
Cooling System	0.308	0.354	0.397	0.434	0.470
Replacement Capacity	0.075	0.047	0.042	0.045	0.039
Fuel	7.876	7.857	7.849	7.843	7.846
Replacement Energy	0.118	0.062	0.059	0.072	0.060
Maintenance	0.579	0.579	0.579	0.580	0.582
Total Power Production Cost	21.709	21.623	21.622	21.644	21.673

The plant construction costs, which represent approximately 71% of the cost of the nuclear plant and approximately 60% of the cost of the fossil plant, are found to vary slightly with the change inflow rate.

The sensitivity of power production cost to condenser flow rate under variation of all the economic parameters discussed in Chapter II was performed for all cooling systems for both the nuclear and the fossil plants. However, only the more interesting sensitivity studies are shown graphically. For the once-through system using a surface canal outfall Figures 3.9 through 3.12 show sensitivity to replacement energy cost, replacement capacity cost, capacity factor and cooling system multiplier, all for the nuclear plant. The sensitivities for the fossil plant are qualitatively similar and are not shown.

Aside from the obvious increase or decrease in the power production cost which occurs when each of the factors is changed it is worthwhile observing how the optimal flow rates vary as well. Increasing the replacement energy costs or the cost of replacement capacity penalizes poor efficiency and thus causes cooling systems to optimize at higher flow rates as shown in Figures 3.9 and 3.10. But since replacement capacity and replacement energy are only a small part of the total production cost, large increases in these parameters result in only relatively small increases in generating cost. Figures 3.11 shows that increasing the plant capacity factor also manadates more efficient operation, and thus a large flow rate, while Figure 3.12 shows that an increase in the cooling system cost, which is dependent on flow rate, suggests a lower optimal flow rate. (Note similarly from Figure 3.7

Figure 3.9 Sensitivity Study for Replacement Energy Costs
Once-Through System 1200 MW Nuclear Plant

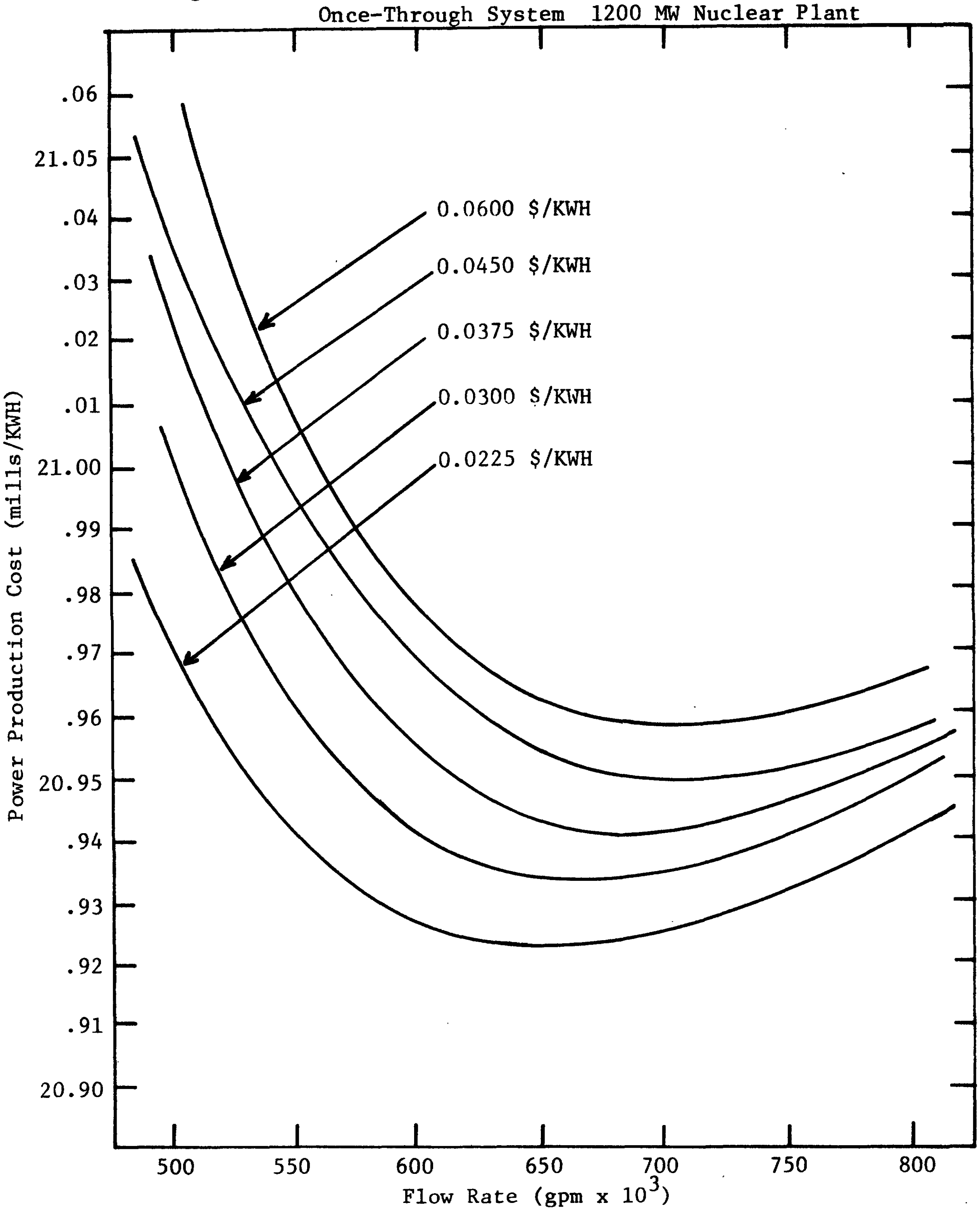


Figure 3.10 Sensitivity Study for Replacement Capacity
Once-Through System
1200 MW Nuclear Plant

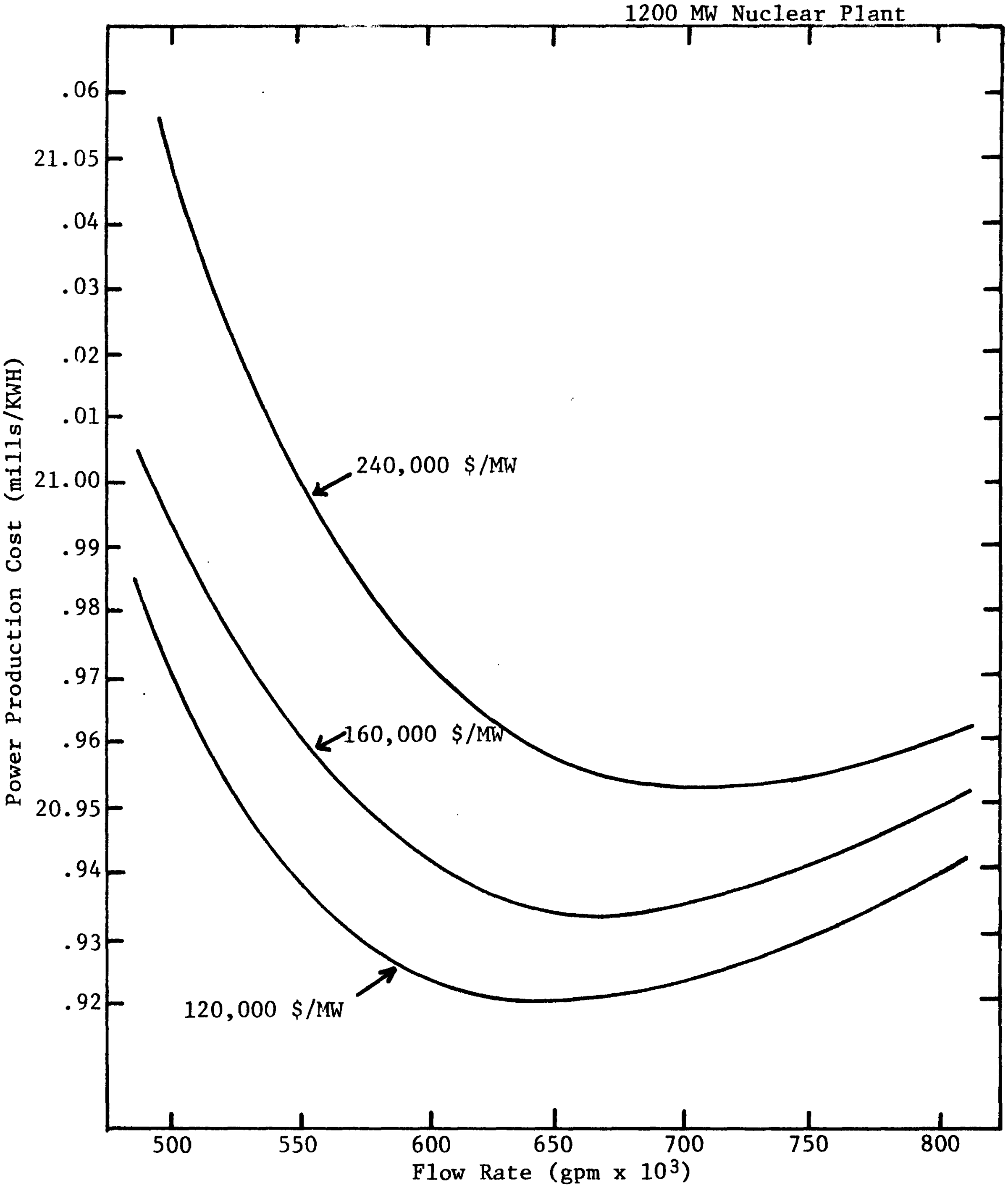


Figure 3.11 Sensitivity Study for Capacity Factor
Once-Through System
Nuclear Plant

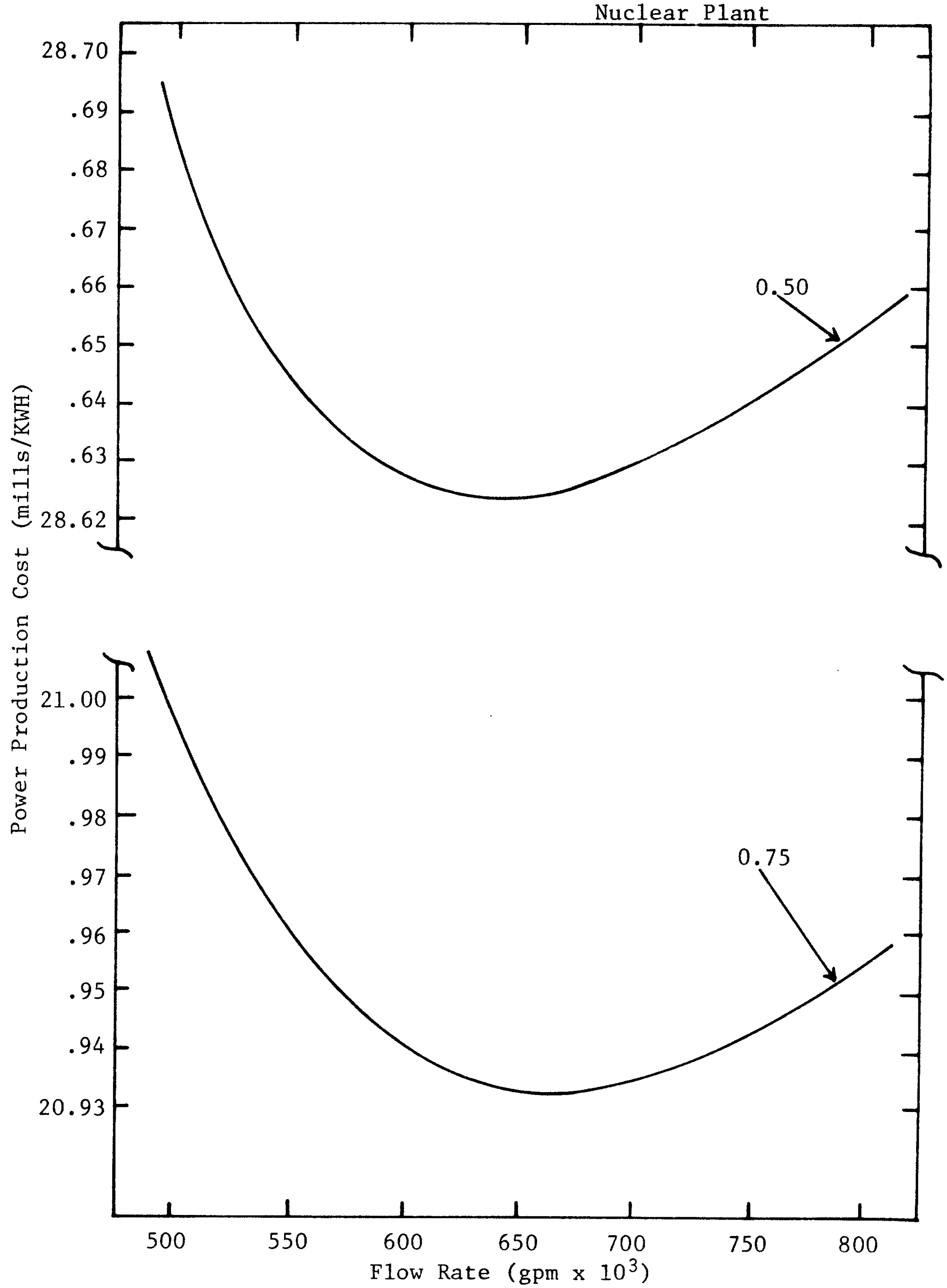
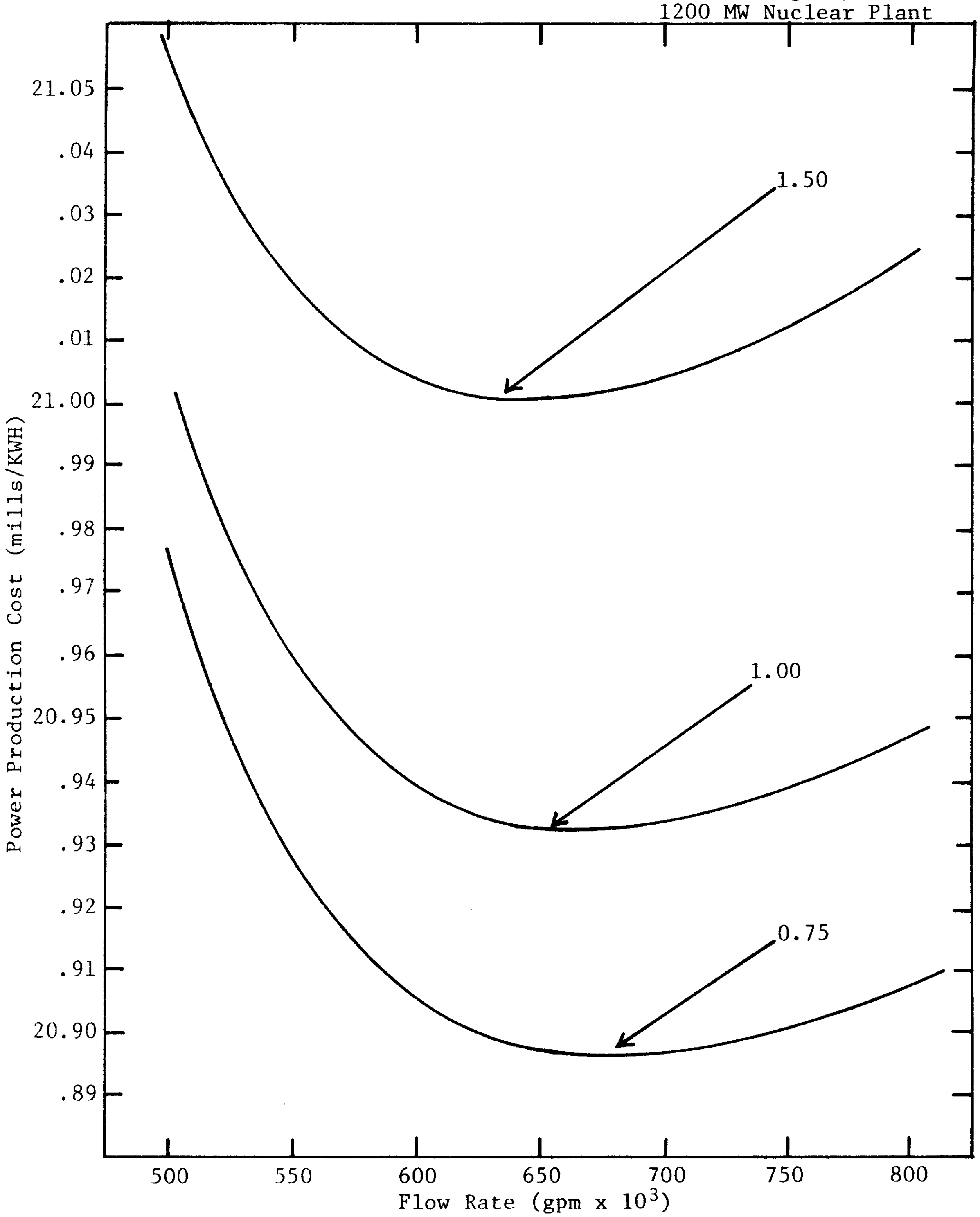


Figure 3.12 Sensitivity Study for Cooling System Multiplier
Once-Through System
1200 MW Nuclear Plant



that an increase in the length or discharge velocity of a multi-port diffuser also suggests a decreasing optimal flow rate.) The cost of fuel, the fixed charge rate and the plant construction cost each have a relatively minor influence on the optimal flow rate. This is because these dominant factors have a greater effect on the total production cost and thus micrify any changes in cooling system design. Finally, for the once-through system the cost of water and water treatment, two variables mentioned in Chapter II, were assumed to be zero for the base case calculations and no sensitivity runs were made. This is because the ability to use once-through cooling pre-supposes a fairly large source of cheap water. Water Treatment costs are negligible compared to the treatment of chemical blowdown which might be required for wet cooling systems and thus no sensitivity was considered.

Tables 3.5 and 3.6 summarize the results for the once-through simulation by breaking down total operating costs into capital cost in 1977 dollars for plant construction, cooling system and replacement capacity , and operating cost in mills/KWH for fuel, maintenance, and replacement energy. Also summarized in the table is the sensitivity to variation in plant cost, fuel cost, fixed charge rate, capacity factor, cooling system multiplier, replacement capacity and replacement energy.

In summarizing the sensitivity study, for this table, the cooling system size (flow rate) was maintained at a constant value equal to the optimal value obtained using base case economic parameters. This is referred to as a transferred system, and a quantitative comparison of this system, and one in which the cooling system is optimized for each

Table 3.5 Cost Sensitivity Study for Once-Through Systems - 1200 MW Nuclear Plant

	Plant Construction ($\$10^6$)	Cooling System ($\$10^6$)	Replacement Capacity ($\$10^6$)	Replacement Energy ($\frac{Mills}{KWH}$)	Fuel Cost ($\frac{Mills}{KWH}$)	Maintenance ($\frac{Mills}{KWH}$)	Water Treatment ($\frac{Mills}{KWH}$)	Water Cost ($\frac{Mills}{KWH}$)	Total Production Cost ($\frac{Mills}{KWH}$)
* Base Case:	693.004	18.484	2.043	0.035	4.833	0.679			20.933
	14.943 $\frac{Mills}{KWH}$	0.399 $\frac{Mills}{KWH}$	0.044 $\frac{Mills}{KWH}$						
Sensitivity:									
Plant Cost									
$450/KW	519.753	18.484	2.043	0.035	4.833	0.514			17.032
* $600/KW	693.004	18.484	2.043	0.035	4.833	0.679			20.933
$900/KW	1039.507	18.484	2.043	0.035	4.833	1.008			28.734
Fuel Cost									
$0.0012/KWH	693.004	18.484	2.043	0.035	3.625	0.679			19.724
* $0.0016/KWH	693.004	18.484	2.043	0.035	4.833	0.679			20.933
$0.0024/KWH	693.004	18.484	2.043	0.035	7.249	0.679			23.349
$0.0032/KWH	693.004	18.484	2.043	0.035	9.666	0.679			25.765
Fixed Charge Rate									
15%	693.004	18.484	2.043	0.035	4.833	0.679			19.123
* 17%	693.004	18.484	2.043	0.035	4.833	0.679			20.933
20%	693.004	18.484	2.043	0.035	4.833	0.679			23.648
Capacity Factor									
0.50	693.004	18.484	2.043	0.035	4.833	0.679			28.625
* 0.75	693.004	18.484	2.043	0.035	4.833	0.679			20.933
Cooling System Multiplier									
0.75	693.004	16.932	2.043	0.035	4.833	0.678			20.899
* 1.00	693.004	18.484	2.043	0.035	4.833	0.679			20.933
1.50	693.004	21.580	2.043	0.035	4.833	0.680			21.004
Replac. Capac.									
$120/KW	693.004	18.484	1.532	0.035	4.833	0.678			20.921
* $160/KW	693.004	18.484	2.043	0.035	4.833	0.679			20.933
$240/KW	693.004	18.099	3.478	0.041	4.833	0.680			20.958
Replac. Energy									
$0.0225/KWH	693.004	18.484	2.043	0.026	4.833	0.679			20.924
* $0.0300/KWH	693.004	18.484	2.043	0.035	4.833	0.679			20.933
$0.0375/KWH	693.004	18.484	2.043	0.044	4.833	0.679			20.941
$0.0450/KWH	693.004	18.484	2.043	0.053	4.833	0.679			20.950
$0.0600/KWH	695.704	18.556	1.303	0.012	4.833	0.681			20.960

Table 3.6 Cost Sensitivity Study for Once-Through Systems - 800 MW Fossil Plant

	Plant Construction ($$10^6$)	Cooling System ($$10^6$)	Replacement Capacity ($$10^6$)	Replacement Energy (Mills/KWH)	Fuel Cost (Mills/KWH)	Maintenance (Mills/KWH)	Water Treatment (Mills/KWH)	Water Cost (Mills/KWH)	Total Production Cost (Mills/KWH)
* Base Case:	392.501	11.891	1.375	0.066	7.849	0.579			21.618
	12.695 Mills/KWH	0.385 Mills/KWH	0.044 Mills/KWH						
Sensitivity:									
Plant Cost									
$375/KWH	295.782	11.948	0.771	0.018	7.861	0.440			18.298
* $500/KWH	392.501	11.891	1.375	0.066	7.849	0.579			21.618
$750/KWH	585.939	11.834	1.980	0.143	7.828	0.856			28.226
Fuel Cost									
$0.0023/KWH	392.501	11.891	1.375	0.066	5.823	0.579			19.592
* $0.0031/KWH	392.501	11.891	1.375	0.066	7.849	0.579			21.618
$0.0046/KWH	391.563	11.862	1.678	0.101	11.633	0.578			25.414
$0.0061/KWH	390.626	11.834	1.980	0.144	15.404	0.577			29.206
Fixed Charge Rate									
15%	393.438	11.919	1.073	0.038	7.856	0.580			20.073
* 17%	392.501	11.891	1.375	0.066	7.849	0.579			21.618
20%	391.563	11.862	1.678	0.101	7.840	0.578			23.933
Capacity Factor									
0.50	390.626	11.834	1.980	0.144	7.828	0.577			28.171
* 0.75	392.501	11.891	1.375	0.066	7.849	0.579			21.618
Cooling System Multiplier									
0.75	392.501	10.738	1.375	0.066	7.849	0.578			21.580
* 1.00	392.501	11.891	1.375	0.066	7.849	0.579			21.618
1.50	392.501	14.295	1.375	0.066	7.849	0.580			21.701
Replac. Capac.									
$120/KW	392.501	11.891	1.032	0.065	7.849	0.578			21.606
* $160/KW	392.501	11.891	1.375	0.066	7.849	0.579			21.618
$240/KW	392.501	11.891	1.661	0.066	7.849	0.580			21.641
Replac. Energy									
$0.0225/KWH	390.626	11.834	1.980	0.108	7.828	0.577			21.594
* $0.0300/KWH	392.501	11.891	1.375	0.066	7.849	0.579			21.618
$0.0375/KWH	393.438	11.919	1.073	0.048	7.856	0.580			21.630
$0.0450/KWH	394.375	11.948	0.770	0.027	7.861	0.581			21.637
$0.0600/KWH	395.313	11.976	0.468	0.010	7.865	0.582			21.645

comparison was made for the dry cooling towers (Choi and Glicksman, 1978) This type of "optimized system" could be made from the sensitivity studies by interpolating between flow rates for the optimum and then costing that system. But since it was found in the sensitivity study for dry towers that the difference in generating cost between the "optimized" and "transferred" systems was small, the transferred system is tabulated for all the cooling systems.

It is apparent from these tables that changes in the economic parameters have a much greater effect on the overall production cost than do changes in the system design (varying Q_o). This is especially true for the major economic components (plant cost, fuel cost, fixed charge rate, capacity factor) where varying these leads to increases (or decreases) of 5 to 40%. Changes in the system design generally amount to changes much less than 5% of the total production cost.

Chapter IV

COOLING PONDS

4.1 Introduction

Cooling ponds are large waterbodies, typically several hundred to several thousand acres in surface area, used for closed cycle cooling. Heated water from the plant condensers flows through the pond, loses heat to the atmosphere through evaporation, radiation and conduction, and is recirculated through the plant intake. Because of the finite surface area, the condenser intake temperature is generally higher, and thus the thermodynamic efficiency lower, than for an equivalent once-through system. However, these differences can be reduced if the pond size is increased. As opposed to cooling lakes which are created by the damming of a stream, cooling ponds are artificially constructed, usually by the erection of earth dikes, and recieve their make-up water from nearby surface or subsurface supplies. Internal dikes as shown in Figure 4.1 are often added to direct the flow and prevent recirculation.

The objective in designing cooling ponds is to maximize surface heat transfer while minimizing construction and operational costs. A systematic evaluation of pond depth, areal geometry, internal baffling, condenser flow rate, etc. is being performed in conjunction with the present study (Adams *et al*. 1978). The results suggest that in many cases a cost-effective pond is shallow and involves a sufficient number of baffles to create an essentially one-dimensional flow. (Again, see Figure 4.1).

One of the characteristics which distinguishes cooling ponds from other closed cycle cooling systems is their "thermal-inertia." Thermal

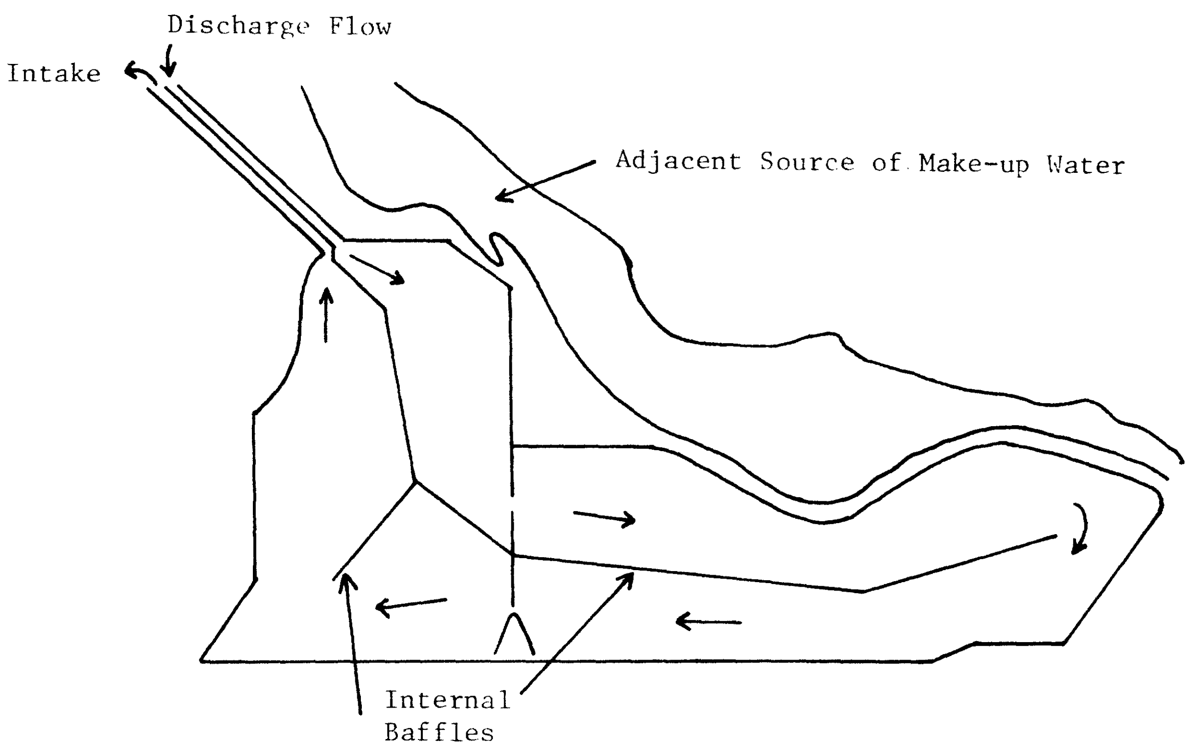


Figure 4.1 Plan View of a Typical Cooling Pond (Dresden, Illinois, Cooling Pond)

inertia arises from the large mass of water involved and allows the pond to damp out high temperature peaks, induced by variations in plant loading or meteorological conditions, which would adversely affect the performance of cooling towers. It also dictates that true pond performance be simulated with a transient model using a time series of meteorological data. However, in the design state, where many alternatives must be evaluated, a transient simulation is rather costly and so a simpler, yet physically meaningful, analytical design model has been developed for this thesis.

4.2 Environmental Factors

Cooling ponds are closed cycle cooling systems that qualify as a technological control of thermal dishcarges. The EPA, in its Effluent Guidelines and Standards for Steam Electric Power Generating (40 CFR 423, 1974), makes the distinction between cooling ponds and cooling lakes. A cooling pond is an "off-stream" water impoundment which does not impede the flow of a navigable stream and is used to remove waste heat from power plants. Cooling lakes are "on-stream" water impoundments created by damming a small stream, hence impeding its flow. While thermal legislation can be applied to cooling lakes, it only applies to the heat from the blowdown of a cooling pond and thus the waste heat load is often far greater for ponds than for lakes.

The EPA defines "blowdown" (40 CFR 423, 1974) as "the minimum discharge of recirculating water for the purpose of discharging materials contained in the process, the further buildup of which would cause concentrations exceeding limits established by best engineering practice." In addition to treatment requirements for these high pollutant concentrations, the EPA designates that "heat may be discharged in blowdown from cooling ponds provided the temperature at which the blowdown is discharged does not exceed at any time the lowest temperature of recirculated cooling water prior to the addition of make-up water." Although blowdown contains only 1 to 2% of the condenser water flow, the regulations are necessary since the discharge heat from blowdown can have relatively large effects in small streams.

Because the contribution of forced evaporation to total surface heat transfer increases with increasing surface temperature above equilibrium temperature, cooling ponds consume more water, and have more fogging potential, than once-through systems. They may also consume somewhat more water than other closed cycle cooling system (e.g., towers), due to natural evaporation (that which would occur in the absence of artificial heating) and seepage, although this is not always the case. The rate of water loss will be calculated later in this Chapter and compared with the other systems in Chapter VII.

Make-up water is needed to replace that lost by evaporation, blowdown and seepage. Since the quantity of water requried is low (3 to 5% of condenser flow) these intakes are much smaller than those for once-through systems and therefore the effects of intake impingement and entrainment are smaller. And because of their storage, cooling ponds hold an advantage over other forms of evaporative cooling (e.g., towers) in that they need not withdraw their make-up water continuously. Thus in regions of hydrologic variability, the utility can store water during low flow and replace it during periods of high flow. In (predominantely western) states where water usage is governed by appropriation rights, this allows a utility to purchase fewer, or more "junior", water rights. A related advantage is that the level of blowdown treatment, which usually must be geared to conditions of low flow, can be reduced by discharging primarily during periods of higher river flow.

Other environmental impacts may be associated with seepage losses from cooling ponds. Infiltration may affect the surrounding groundwater reservoir by (1) contaminating the groundwater supply or (2) recharging the local groundwater structure, thus raising the water table elevation and affecting surrounding land use.

While water consumption, blowdown, fogging and infiltration may be important environmental problems, the major environmental consideration in the selection of cooling ponds is their large land requirements (typically 3/4 to 2 acres/MWe) making them unattractive where either land (or land development) costs are high or local land use policies restrict their siting.

It should be mentioned that spray modules can be added to a cooling pond to form a type of mixed-mode cooling system which can significantly reduce land requirements. Spray devices aerate the water by shooting it over the surface of the pond thus increasing the effective surface area of water exposed to the air and the relative velocity between the water droplets and the air, thereby accelerating cooling. Spray ponds allow a reduction in surface area by as much as a factor of twenty (20), (HEDL, 1972), at the expense of additional capital and operating expenses for the spray units. Due largely to the inability to accurately predict their hydrothermal performance, spray units have not been widely used and they will not be considered explicitly in this study.

4.3 Performance Model

This section presents the methodology by which the transient characteristics of a shallow cooling pond can be treated in a quasi-steady hydrothermal model. In this way different cooling pond designs can be compared through simulation with a cumulative distribution of meteorological data in much the same manner as other cooling systems (e.g., towers). The material in this section has been derived largely from recent cooling pond studies at MIT (e.g., Jirka *et al.*, 1978) and a cooling pond optimization study (Adams, *et al.*, 1978) which is being conducted in parallel with the present effort.

4.3.1 A Transient Simulation Model

A transient, mathematical model for shallow, one-dimensional type cooling ponds, has been developed by Watanabe and Jirka (1977); the essential features are indicated in Figure 4.2. The pond is schematized by its length, L, its average surface width, W, its average depth, H, and the circulating water flow, Q_o. The jet entrance mixing region is a small fraction of the total pond area; the major throughflow portion of the pond is characterized by a longitudinal dispersion process.

Following Taylor (1954), the longitudinal dispersion of heat is written as a one-dimensional bulk diffusion equation with cross-sectionally averaged variables

$$\frac{\partial T}{\partial t} + U\frac{\partial T}{\partial x} = E_L\frac{\partial^2 T}{\partial x^2} - \frac{\phi_n}{\rho c H} \tag{4.1}$$

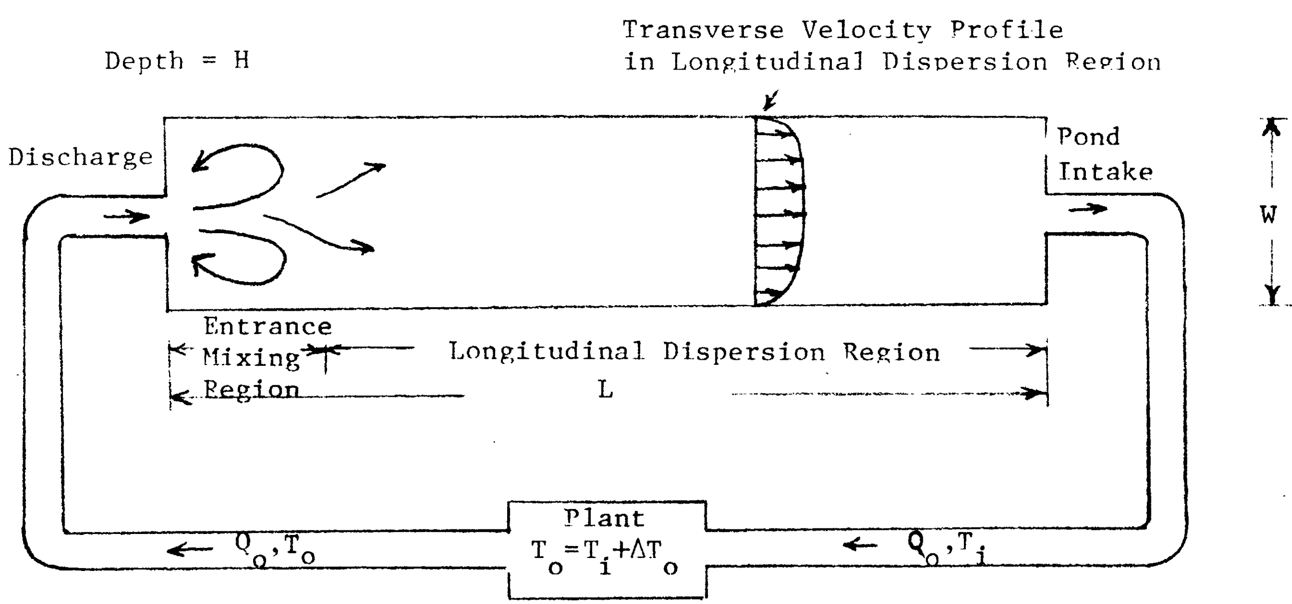


Figure 4.2 Schematic View of One-Dimensional Type Cooling Pond

where T = cross-sectional mean temperature, U = cross-sectional mean velocity Q/WH, x = longitudinal distance, t = time, E_L = longitudinal dispersion coefficient, ϕ_n = net heat flux across the surface, and ρc = heat capacity of water per unit volume. Equation (4.1) assumes a channel of constant W, H and E_L. The extension to variable values of W, H and E_L is readily made, but is not needed for this generic design study. Equations of the above type have been frequently used to model the dispersion of tracers and pollutants in natural streams and rivers (e.g., Fischer, 1967, McQuivey and Keefer, 1976).

Equation (4.1) is valid within the pond length L, that is, in the domain of $0 < x < 1$. The boundary conditions which apply in dispersive fluid systems of finite length are given as follows:

At the inflow:

$$T\Big|_{x=0} - \frac{E_L}{UL} \frac{\partial T}{\partial x}\Big|_{x=0} = T_o \tag{4.2}$$

where T_o = inflow temperature

At the outflow:

$$\frac{E_L}{UL} \frac{\partial T}{\partial x}\Big|_{x=0} = 0 \tag{4.3}$$

The inflow boundary condition expresses continuity between the purely convective transport in the inflow channel and the sum of convective and dispersive transport just within the pond. The outflow boundary condition eliminates any dispersive flux into the outflow channel.

A numerical solution of Equation (4.1) has been developed using a finite difference scheme with the Crank-Nicholson method. The two parameters which have to be defined in order to predict the behavior of the cooling pond are the dispersion coefficient, E_L, and the total heat flux through the water surface, ϕ_n. The dispersion coefficient follows Fischer (1967) and is given by

$$E_L = \frac{0.3 \sqrt{\frac{f}{8}}\, U \left(\frac{W}{2}\right)^2}{\kappa^2 H} \tag{4.4}$$

where κ = von Karman's constant (0.4) and f = friction coefficient.

The net surface heat flux can be broken into the following components:

$$\phi_n = \underbrace{\phi_{sn} + \phi_{an}}_{\phi_r} - \phi_L \tag{4.5}$$

where ϕ_{sn} = net incident solar radiation (incident minus reflected)

ϕ_{an} = net incident atmospheric radiation (incident minus reflected)

ϕ_r = net radiation term

ϕ_L = net heat loss term

The net heat loss, ϕ_L in turn can be written as

$$\phi_L = \phi_{br} + \phi_e + \phi_c \tag{4.6}$$

where ϕ_{br} = long wave (back) radiation from the water surface

ϕ_e = evaporative heat flux

ϕ_c = conductive heat flux

The complete non-linear expression for surface heat transfer is given by Ryan and Harleman (1973) as

$$\phi_n = \phi_r - \{\underbrace{4x10^{-8}\ (T+460)}_{\phi_{br}}{}^4 + f(W)[\underbrace{(e_s - e_a)}_{\phi_e} + \underbrace{0.255(T-T_a)}_{\phi_c}]\} \tag{4.7}$$

where $\phi_r = \phi_{sc}(1-.65C^2) + 1.16x10^{-13}(460+T_a)^6(1+0.17C^2)$

ϕ_{sc} = clear sky radiation

$f(W) = 22.4(\Delta\theta_v)^{1/3} + 14W_2$

e_s = saturated vapor pressure (mmHg) of air at the average water surface temperature T_s

e_a = actual vapor pressure (mmHg) of the ambient air at air temperature

T_a = air temperature (^oF)

C = cloud cover (0 to 1)

W_2 = wind velocity (mph) measured two meters above the water surface

$\Delta\theta_v$ = virtual temperature difference = $T_{s_v} - T_{a_v}$

$$T_{s_v} = \overline{T}_s/(1-0.378e_s/p)$$

$$T_{a_v} = T_a/(1-0.378e_a/p)$$

p = atmospheric pressure (mmHg)

The rate of water loss by evaporation is computed by dividing the evaporative heat loss term, $f(W)(e_s-e_a)$, by the latent heat of vaporization.

A comparison of model prediction with field data from the Dresden, Illinois cooling pond shown in Figure 4.1 is reported by Watanabe and Jirka (1977). Agreement was generally within $1^{\circ}F$ and the transient nature of the temperature fluctuations from the plant, as well as the long term weather and some diurnal changes, were exhibited by the model when run with a time step of three hours. The model also compared favorably with two simpler models which are commonly used in cooling pond design: a plug flow model in which E_L is effecitvely zero and a fully-mixed model in which E_L is effectively infinite.

4.3.2 The Need for Long Term Simulation

In order to evaluate the performance of a particular cooling pond, it is necessary to cover a wide range in meteorological conditions which might occur during the pond's life time. A brute force way to do this involves running a transient numerical model with time-varying meteorological conditions for a number of years. From these simulation results, the frequency distribution of the plant intake temperature can be obtained and the effect of the plant performance can be evaluated.

A disadvantage of a long term transient simulation, however, is the considerable computation time and effort which is involved; at the design stage, where a number of alternative designs must be evaluated, such a simulation is impractical. Therefore, it is necessary to develop a simpler, approximate, model to be used for the purpose of initial pond design. In particular it would be desirable to use a steady state model so that, as with the design of cooling towers and once-through systems, a frequency distribution of meteorological data, rather than a long time series, can be used. The more accurate transient simulation model can then be used to evaluate the chosen design.

4.3.3 Development of a Quasi-Steady State Model

The quasi-steady model uses the following differential equation

$$U \frac{\partial T}{\partial x} = E_L \frac{\partial^2 T}{\partial x^2} - \frac{K}{\rho c H} (T - T_E) \tag{4.8}$$

along with boundary conditions given by Equations (4.2) and (4.3). Equation (4.8) differs from Equation (4.1) only in the use of a linearized excess temperature representation for surface heat transfer (see below) and the fact that the time dependent term is missing. The model is quasi-steady in the sense that the input parameters governing the pond performance (plant operating conditions and meteorology) are assumed to be constant over a period of time and the pond temperature is assumed to be in instantaneous equilibrium with these parameters. The constant input parameters are derived by averaging the real parameters over the time interval. Clearly this procedure is an approximation of true pond behavior. By averaging the input data one is filtering high frequency fluctuations and by assuming "instant response" one is ignoring the "thermal inertia" known to characterize ponds. The intent is to adjust the averaging interval such that the effects cancel as much as possible in their influence on the cumulative distribution of intake temperatures.

The solution to Equations (4.8), (4.2) and (4.3) was first given by Wehner and Wilhelm (1956), and the outflow temperature T_i at $x = L$ can be written as:

$$\frac{T_i - T_E}{T_o - T_E} = \frac{4a\exp\{1/2E_L^*\}}{(1+a^2)\exp\{a/2E_L^*\} - (1-a^2)\exp\{-a/2E_L^*\}} \tag{4.9}$$

where $\quad a = \sqrt{1+4rE_L^*}$

$$r = \frac{KA}{\rho cQ}$$

and $\quad E_L^* = \dfrac{E_L}{UL}$

Since the cooling pond is a closed system, dishcarge temperature T_o can be written as $T_o = T_i + \Delta T_o$. Substituting T_o into Equation (4.9) gives

$$\frac{T_i - T_E}{\Delta T_o} = \frac{4a\exp\{1/2E_L^*\}}{(1+a)^2\exp\{a/2E_L^*\} - (1-a)^2\exp\{-a/2E_L^*\} - 4a\exp\{1/2E_L^*\}} \tag{4.10}$$

In order to predict the intake temperature T_i, the equilibrium temperature, T_E, and the heat exchange coefficient, K, have to be defined.

The former is defined as the water temperature at which the net heat flux $\phi_n = 0$. Therefore T_E can be obtained iteratively by solving the following equation,

$$\phi_n = \phi_r - \{4\text{x}10^{-8}(T_E+460)^4 + f(W)[(e_s - e_a) + 0.255(T_E - T_a)]\} = 0 \tag{4.11}$$

The linearized surface heat exchange coefficient K is defined as follows:

$$K = -\frac{\partial\phi_n}{\partial T_s} = 23.0 + [14W_2 + 22.4(\Delta\theta_v)^{1/3}](\beta_s + 0.255)$$
$$+ 7.5(\Delta\theta_v)^{-\frac{2}{3}}[e_s - e_a + 0.255(T_s^* - T_a)] \tag{4.12}$$

where $\beta_s = 0.255 - 0.0085T_s + 0.000204T_s^2 (\text{mmHg}/^oF)$.

In order to give the correct value of the total heat transfer ϕ_n through the relationship $\phi_n = -K(T_s - T_E)$, K is evaluated at T_s^* which lies between the average water surface temperature, T_s, and the equilibrium temperature, T_E (see Figure 4.3). For this analysis it is assumed that

$$T_s^* = \frac{1}{2}(T_s + T_E) \tag{4.13}$$

The average water surface temperature is given by

$$T_s = T_E + \frac{\Delta T_o}{r} \tag{4.14}$$

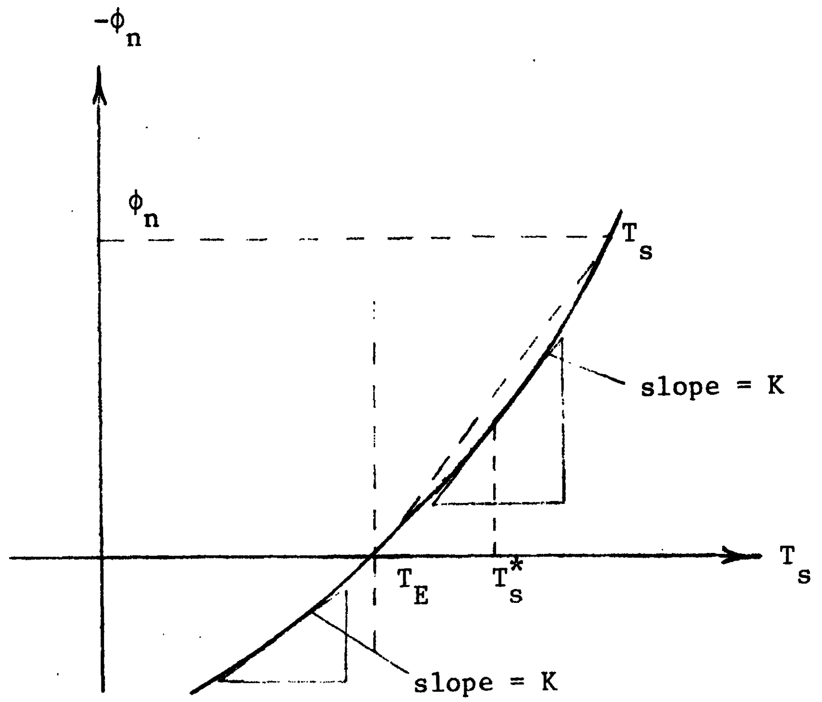


Figure 4.3 Variation of Heat Transfer Coefficient K with Water Surface Temperature T_s

so that

$$T_s^* = T_E + \frac{\Delta T_o}{2r} \tag{4.15}$$

Since the value of r is not known a priori, it must be determined by iteration.

4.3.4 Comparison of Quasi-Steady State Model and Transient Model

Cumulative distributions of predicted intake temperatures using both the quasi-steady and the transient models were compared using a schematic pond (similar to the Dresden pond, but scaled to a size appropriate for 1200 MW) with the following characteristics: L = 21750', W = 1500', H =10', Q_o = 1260 cfs and T_o = 23^{o}F (constant heat rate assumed). Steady plant operation was assumed so that the discharge temperature was obtainable from the intake temperature at the previous time step. The transient model was run for two summers (May-September, 1966 and 1967) using three hour meteorological data from Argonne National Laboratory; the cumulative distribution of intake temperatures predicted with this model are shown in Figure 4.4 as a solid line. Quasi-steady calculations were also made for the same pond and time period by averaging the meteorological data over different averaging intervals, computing values of K and T_E for each time interval, and then using Equation (4.10) to compute intake temperature. Distributions of intake temperatures are plotted in Figure 4.4 for averaging intervals of 1, 3, and 5 days.

Comparison of the various graphs indicates that reasonably good

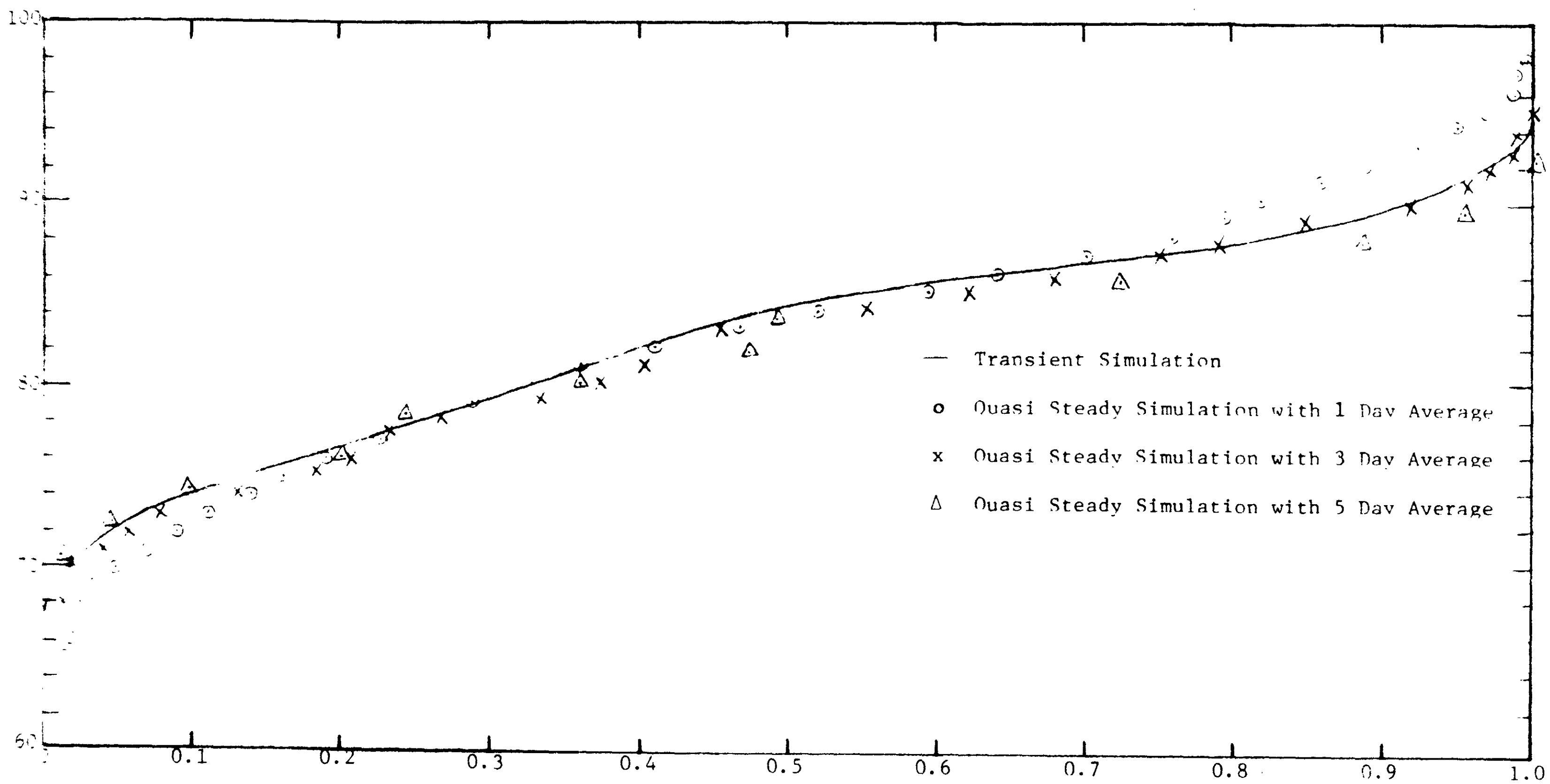


Figure 4.4 Cumulative Distribution of Intake Temperatures Using Transient Model and Quasi Steady Models with Various Averaging Times

agreement is obtained between the transient model and the 3-day averaged model. By contrast, results for 1-day averaging show greater extremes in temperatures suggesting that the averaging has not adequately filtered the high frequency fluctuations, while the distributions resulting from 5-day averaging is the flattest, suggesting that the averaging of input data provides more filtering than the transient model. These results indicate that, for this site and pond, an averaging of 3 days seems appropriate. This figure seems reasonable as it corresponds roughly to the time constant, $\frac{\rho c H}{K}$, which governs the response of a shallow water body to a step change in T_E. Because all of our pond designs will be based on H = 10', and will involve a similar climate to that used in the example, an averaging time of 3 days will be used throughout.

4.4 Optimization Model

4.4.1 Design Variables

The optimal pond is found by finding that combination of design variables which minimizes the total cooling system cost (operation and construction) in accordance with the procedures discussed in Chapter II.

The pond is assumed to be constructed of dikes and baffles to provide a one-dimensional vertically well-mixed flow. The solution for intake temperature, Equation (4.11), and the supporting discussion in the previous section, suggests that pond performance may depend on condenser flow rate Q_o (or temperature rise ΔT_o), and pond dimensions L, W and H. Noting from Equation (4.8) that intake temperature depends primarily on ΔT_o and surface area (A = L*W), and only weakly on L, H and W independently (e.g. through their influence on E_L^*), it is concluded that A and ΔT_o are the primary design variables which should be studied. Therefore pond depth and width were set to constant values of 10' and 2000', respectively.

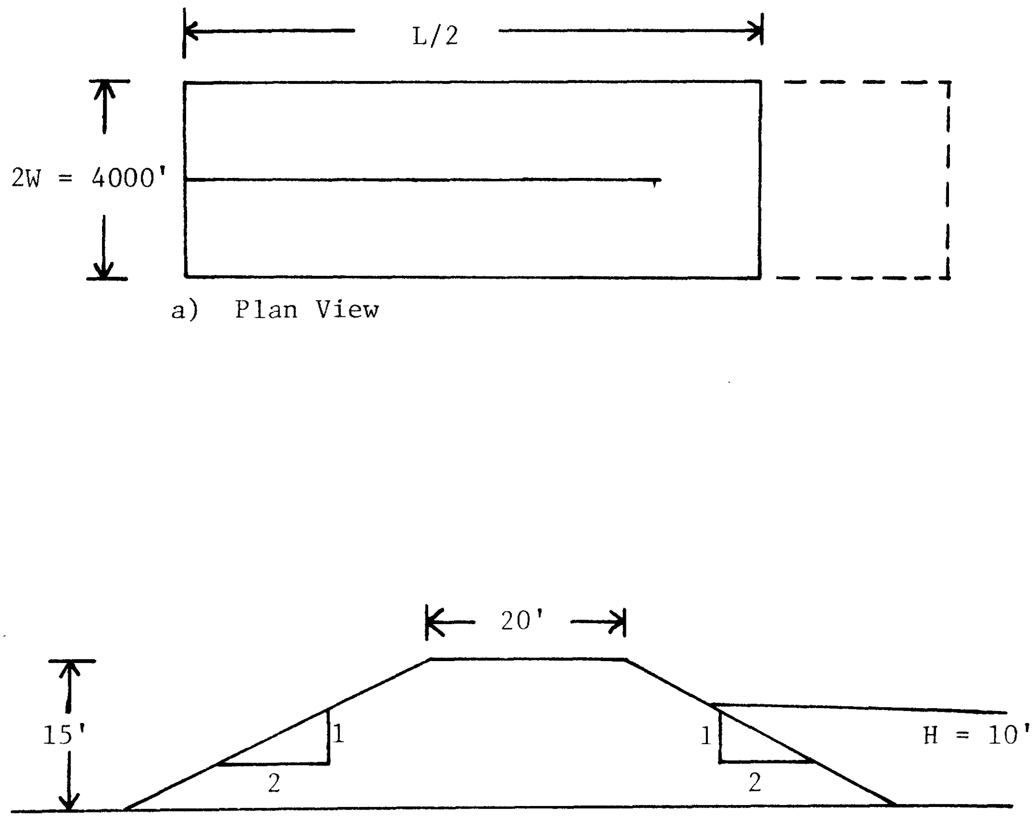
It should be noted that, although temperature rise and pond area are primary design variables, program calculations were made using flow rate and residence time $\frac{V}{Q_o}$ as independent variables. Flow rates for both nuclear and fossil-fueled plants, were selected such that ΔT_o ranged between nominal values of 10^oF and 40^oF while residence times of between 1 and 8 days were used.

4.2.2 Data Aggregation

The ten years of meteorological data (1961-1970) discussed in Chapter II were used for the design evaluation. As a first step each of the relevant variables was averaged over 3-day intervals and values of K and T_E were computed for each interval. These variables were then stored in a bi-variate distribution of K and T_E using intervals of 30 BTU/ft^2-^{o}F-day for K and 10^oF for $T_E \leq 70^oF$ and 2^oF for $T_E > 70^oF$.

4.4.3 Cooling Pond Costing

In determining pond costs, it was assumed that the pond would be U-shaped (similar to the Dresden, Illinois pond) with a constant width as shown in Figure 4.5a. The central baffle and the perimeter dikes would be constructed with earth fill obtained from local excavation. Cross-sectional dimensions are indicated in Figure 4.5b. Costs for excavation, fill, core and erosion protection were assumed to be \$3.5/yd^3 or \$97/linear foot. Since the pond width is constant, total baffle/dike cost is \$780,000 + \$6160/acre. Cost of land purchase and preparation was assumed



a) Plan View

b) Cross Section of Dikes and Baffles

Figure 4.5 Cooling Pond Dimensions

to be \$3000/acre so that total pond construction costs is \$780,000 + \$6160/acre. To study sensitivity, total costs of 1.5 and .75 times this value were also considered.

4.4.4 Water Balances

A water balance must be written to compute the make-up water requirements and water consumption for a cooling pond. The former can be computed by treating the cooling pond as a control volume and accounting for all inflows and outflows to this volume. Over a sufficient period of time the inflows and outflows must be equal, so referring to Figure 4.6,

$$Q_M + Q_P = Q_E + Q_B + Q_S \tag{4.16}$$

where

Q_M = make-up water flow (cfs)

Q_P = precipitation applied directly to pond surface (cfs)

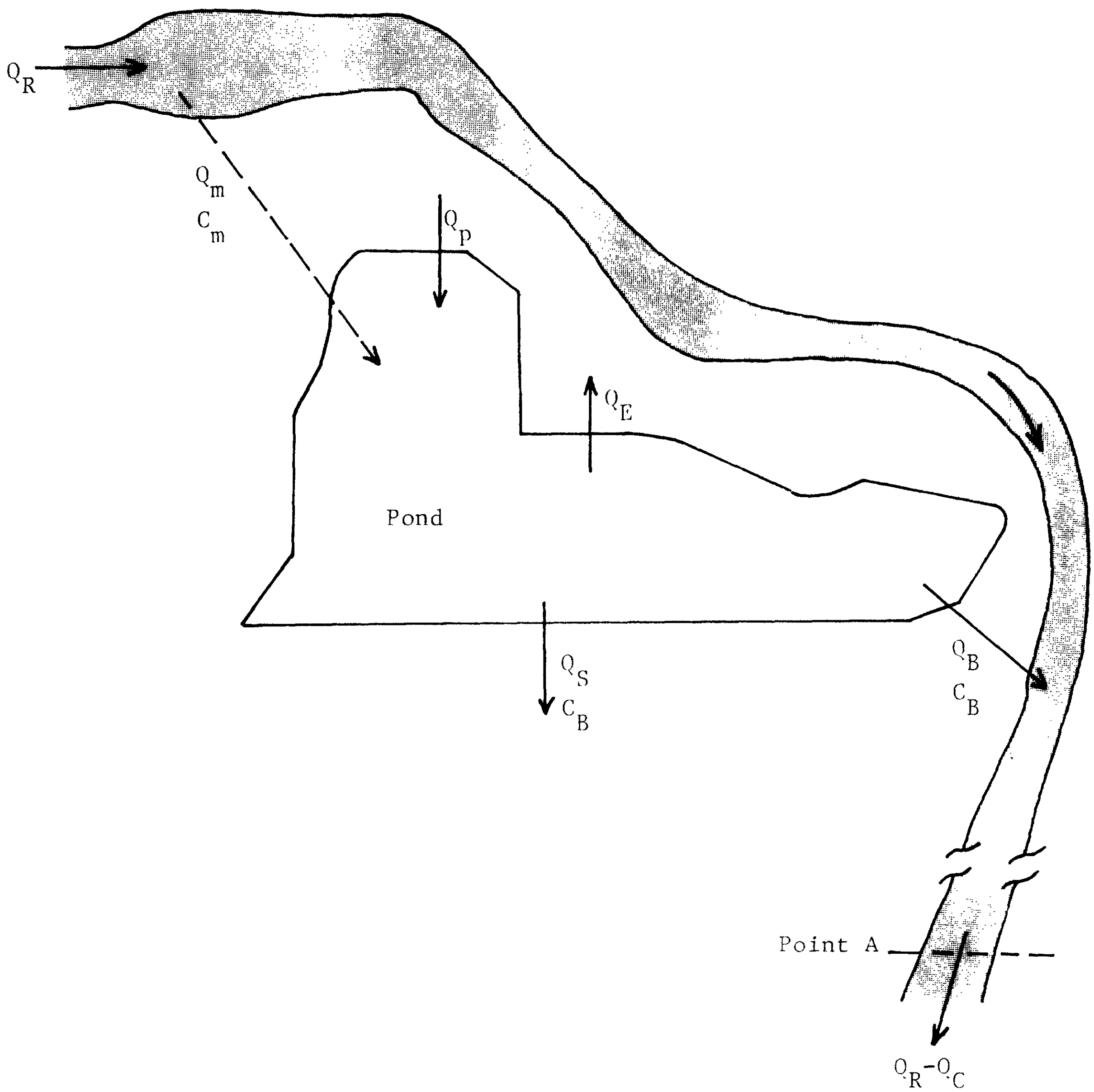
Q_E = total water evaporation losses (natural plus forced)

Q_B = water blowdown (cfs)

Q_S = water seepage (cfs)

The evaporation rate, Q_E, consists of both natural and forced evaporation and can be computed from the evaporative heat flux given in Equation 4.7. For the 1200 MWe nuclear plant with the optimized pond (area = 1770 acres, ΔT_o = 20^{o}F) the average evaporation for the year 1970,

Figure 4.6 Water Balance for Cooling Ponds



in which a transient simulation was performed, was 27.7 cfs. While both natural and forced evaporation are included in this calculation, the natural evaporation can be estimated separately by running the model without heat loading. The average natural evaporation for the 1770 acre pond in 1970 was 8.2 cfs. This can be compared with an annual average natural evaporation of 6.5 cfs computed from the U.S. Climatic Atlas (U.S. Department of Commerce, 1968) based on an evaporation rate of 32 inches/yr. Both the natural and forced components of evaporation are plotted as a function of time in Chapter VII. The annual average precipitation rate, also obtained from the Climatic Atlas was 35 inches/year. The seepage rate is expected to vary considerably from site to site and no specific values were computed for our site. Instead, a value of 5 cfs/1000 acre was selected as representative of values compiled from HEDL (1972).

The remaining two terms, Q_M and Q_B must be determined based on maximum concentration allowances for dissolved solids. Denoting C_M as the concentration of the make-up water and C_B as the concentration of the blowdown (and also any seepage) then,

$$Q_M C_M = (Q_B + Q_S) C_B \tag{4.17}$$

Combining with Equation 4.16 the blowdown flow can be given as

$$Q_B = \frac{(Q_E + Q_S - Q_P) C_M - Q_S C_B}{C_B - C_M} \tag{4.18}$$

The value of C_M is site specific while C_B is dependent on either environmental constraints, where no treatment is applied, or constraints on permissible concentrations of various dissolved solids (Ca, Mg, Si, etc.) designed to prevent scaling, fouling, etc., within the condenser. Since costs for water and treatment are zero for the study site, Q_B and Q_M are evaluated only for the sensitivity study. Values of C_B and C_M were selected as: C_M = 100 ppm, C_B = 200 ppm which corresponds to a cycle of concentration (C_B/C_M) of 2. The resulting flow rates given by Equations 4.18 and 4.17 are: Q_B = 11.7 cfs, Q_M = 41.2 cfs.

The computation of water consumption is not as straightforward as the computation of water requirements. From a chemical standpoint, water is not consumed by the cooling process; instead it merely changes state and may be transferred from a surface to a groundwater supply or vice versa. Following the guidelines of Espey and Huston (1977) the rate of "water consumption" may be viewed as the flow of water denied to a particular water resource (e.g., the make-up source), at a point downstream from the hydrologic influence of the pond, as a result of the existence of the pond. Because the construction of a cooling pond involves hydrological changes in the areas surrounding the pond as well as in the pond itself, consumption should be evaluated by computing the flow of the make-up water source without the pond and subtracting from this the computed flow with the pond. Assuming that the make-up source is a stream, one might evaluate the consumption at a point A in Figure 4.6 as

$$Q_C = Q_M - Q_B - Q_S R_S + Q_P R_P \tag{4.19}$$

where

Q_C = water consumption rate (cfs)

R_S = fraction of cooling pond seepage flow which enters the river and

R_P = fraction of precipitation which would have entered the stream via groundwater or surface runoff had the pond not been built.

Substituting for Q_M-Q_B from Equation 4.16 yields

$$Q_C = Q_E - Q_P(1-R_P) + Q_S(1-R_S) \tag{4.20}$$

Evaluation of R_P and R_S depends on the hydrologic characteristics of the area. For cooling ponds located near the banks of a river, it is reasonable to assume that both may be nearly one. At any rate, Q_E is the largest term and the only one which is evaluated for the present purposes.

The make-up water costs and blowdown treatment costs for the values of Q_M and Q_B respectively are evaluated at the rate given in Section 2.6 where the base case was assumed \$0.00/1000 gal. and \$0.10/1000 gal respectively for the plant site on the Mississippi River. However, the sensitivity to water and waste water treatment costs was evaluated from \$0.05/1000 gal to \$1.00/1000 gal and are shown in the next section.

4.5 Results

To evaluate the optimum pond, power production cost is plotted against the two design variables, ΔT_o and Area (A). The examples for the nuclear unit is shown in Figure 4.7 which indicates that the minimum cost is associated with a ΔT_o of 20°F (Q_o = 800,000 gpm) and an area of 77 $x10^6$ ft^2 (1770 acres, 0.68 MWe per acre, residence time of 5 days). The ΔT_o of 20°F and residence time of 5 days also correspond to the optimal cooling pond for the 800 MWe fossil plant (not shown) where Q_o = 360,000 gpm and A = 33 million ft^2. Because generating costs appear more sensitive to area than temperature rise, the remaining results in this section will consider a constant temperature rise of 20°F and will vary only the pond area.

Tables 4.1 and 4.2 show the total power production cost for varying cooling pond designs for the nuclear and fossil plants, respectively. As with the once-through, the cost of generation is broken down into various components of capital, operating and penalty costs.

The sensitivity of power production cost to cooling system size (pond area, for a constant temperature rise of 20°F) is shown for variation of the cooling system multiplier in Figure 4.8. Sensitivity to the other economic parameters is not shown graphically since their behavior is similar to that discussed with the once-through system; however the sensitivity is included in Tables 4.3 and 4.4. As with the once-through system, increasing the pond multiplier results in a shift towards a smaller cooling system (pond area) at the expense of lower operating efficiency. However, this shift is more prominent with the

Figure 4.7 Power Production Cost vs. Pond Area
Cooling Pond
1200 MW Nuclear Plant

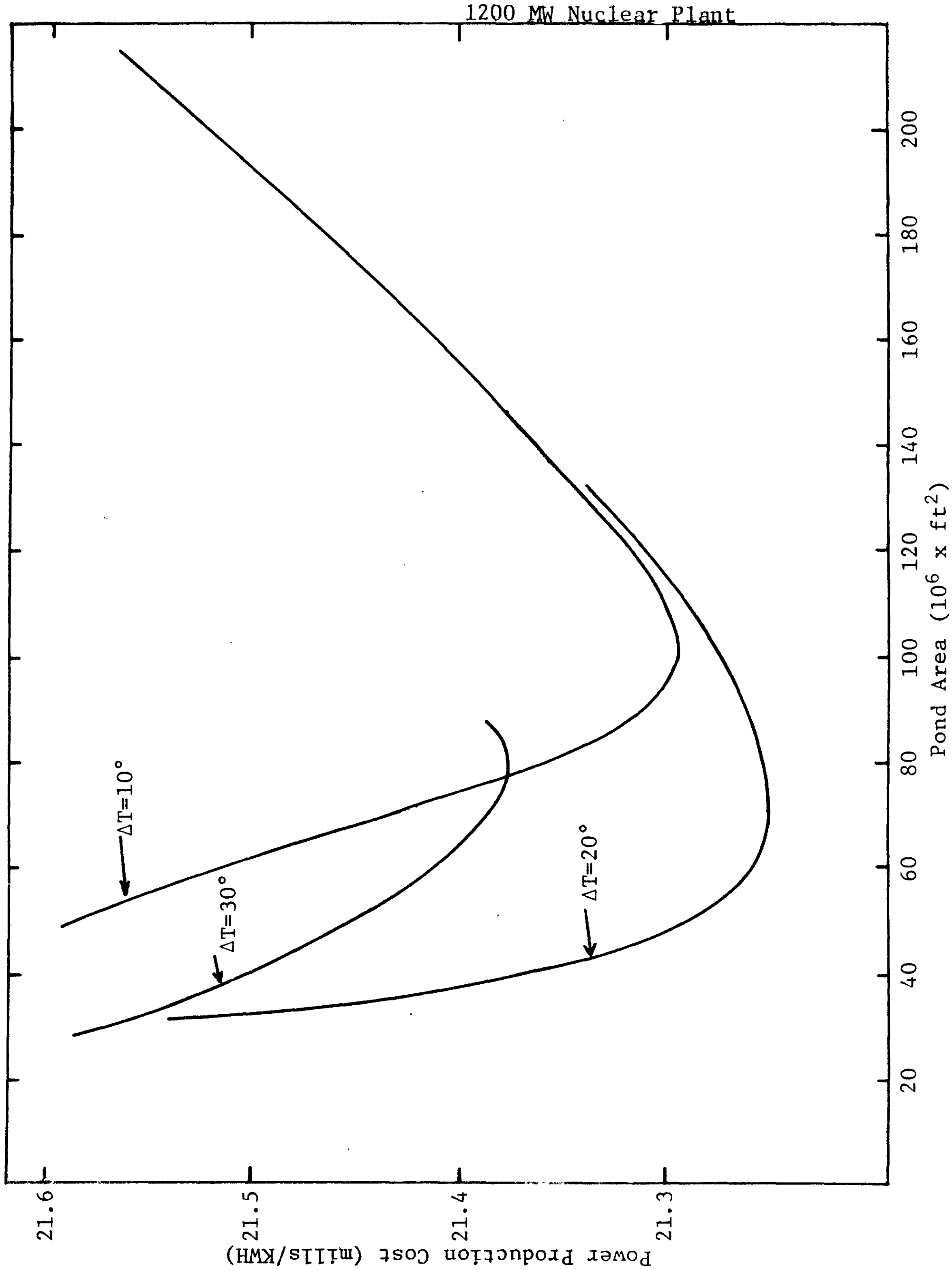


Figure 4.8 Sensitivity Study for Pond Multiplier
Cooling Pond
1200 MW Nuclear Plant

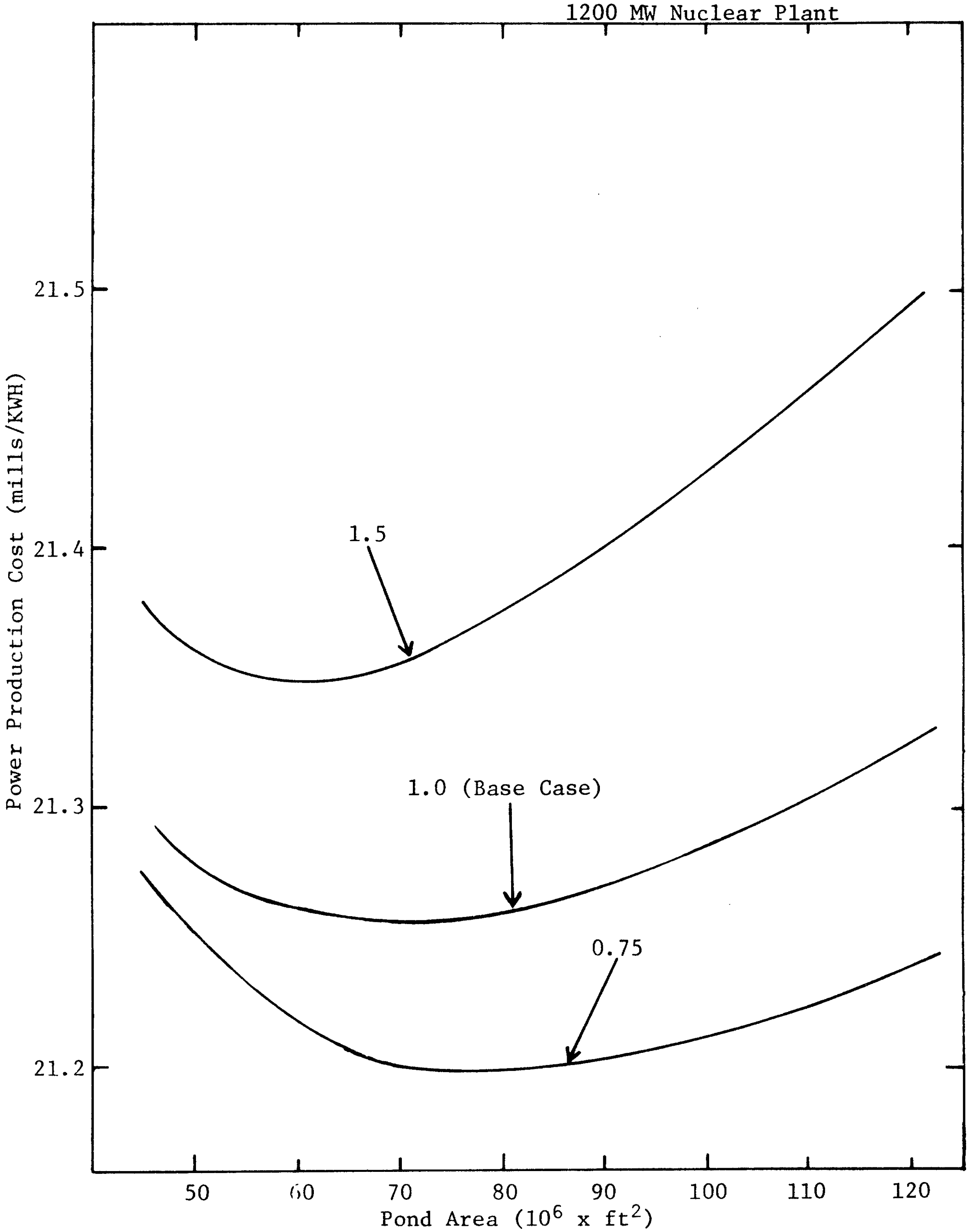


Table 4.1 Power Production Cost Versus Pond Area-1200 MW Nuclear Plant

Cost Component (mills/KWH)	Pond Area (10^6 x ft^2)				
	46.5	62.0	77.5	93.0	108.4
Plant Construction	15.003	14.945	14.945	14.945	14.945
Cooling System	0.519	0.555	0.594	0.632	0.670
Replacement Capacity	0.130	0.116	0.099	0.087	0.079
Fuel	4.843	4.834	4.834	4.834	4.834
Replacement Energy	0.103	0.100	0.075	0.062	0.053
Maintenance	0.691	0.689	0.690	0.691	0.692
Water Treatment	0.026	0.026	0.026	0.026	0.026
Total Power Production Cost	21.315	21.265	21.262	21.277	21.299

Table 4.2 Power Production Cost Versus Pond Area-800 MW Fossil Plant

Cost Component (mills/KWH)	Pond Area (10^6 x ft^2)				
	20.7	27.6	34.5	41.4	48.3
Plant Construction	12.788	12.768	12.760	12.755	12.724
Cooling System	0.420	0.445	0.471	0.497	0.522
Replacement Capacity	0.070	0.065	0.061	0.059	0.066
Fuel	7.897	7.883	7.877	7.873	7.864
Replacement Energy	0.090	0.075	0.065	0.059	0.082
Maintenance	0.586	0.586	0.586	0.587	0.586
Water Treatment	0.019	0.019	0.019	0.019	0.019
Total Power Production Cost	21.870	21.841	21.839	21.849	21.863

cooling pond, since the cooling system cost (land and construction) are greater than for the once-through system. Sensitivity to water and water treatment costs, which were performed for the cooling pond, showed little influence on the optimal pond area.

Tables 4.3 and 4.4 summarize the cooling pond design and sensitivity analysis to variation of economic parameters for the nuclear and fossil plants, respectively, using the same format as for the once-through system. As with the other cooling systems, these tables summarize the costs for only the optimum cooling pond size (flow rate and pond area) obtained using base case economic parameters.

Table 4.3 Cost Sensitivity Study for Cooling Ponds
- 1200 MW Nuclear Plant

	Plant Construction ($\$10^6$)	Cooling System ($\$10^6$)	Replacement Capacity ($\$10^6$)	Replacement Energy ($\frac{Mills}{KWH}$)	Fuel Cost ($\frac{Mills}{KWH}$)	Maintenance ($\frac{Mills}{KWH}$)	Water Treatment ($\frac{Mills}{KWH}$)	Water Cost ($\frac{Mills}{KWH}$)	Total Production Cost ($\frac{Mills}{KWH}$)
* **Base Case:**	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.000	21.262
	14.945 $\frac{Mills}{KWH}$	0.594 $\frac{Mills}{KWH}$	0.099 $\frac{Mills}{KWH}$						
Sensitivity:									
Plant Cost									
$450/KW	521.849	27.636	3.849	0.034	4.840	0.527	0.026	0.000	17.358
* $600/KW	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.000	21.262
$900/KW	1039.663	27.530	4.576	0.075	4.834	1.019	0.026	0.000	29.065
Fuel Cost									
$0.0012/KWH	693.109	27.530	4.576	0.075	3.630	0.690	0.026	0.000	20.054
* $0.0016/KWH	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.000	21.262
$0.0024/KWH	693.109	27.530	4.576	0.075	7.281	0.690	0.026	0.000	23.709
$0.0032/KWH	693.109	27.530	4.576	0.075	9.698	0.690	0.026	0.000	26.126
Fixed Charge Rate									
15%	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.000	19.423
* 17%	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.000	21.262
20%	693.109	27.530	4.576	0.075	4.834	0.090	0.026	0.000	24.022
Capacity Factor									
0.50	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.000	29.081
* 0.75	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.000	21.262
Cooling System Multiplier									
0.75	693.109	25.126	4.576	0.075	4.834	0.687	0.026	0.000	21.208
* 1.00	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.000	21.262
1.50	693.109	32.337	4.576	0.075	4.834	0.694	0.026	0.000	21.371
Replac. Capac.									
$120/KW	693.109	27.530	3.432	0.075	4.834	0.688	0.026	0.000	21.237
* $160/KW	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.000	21.262
$240/KW	693.109	27.530	6.864	0.075	4.834	0.692	0.026	0.000	21.314
Replac. Energy									
$0.0225/KWH	693.109	27.530	4.576	0.056	4.834	0.690	0.026	0.000	21.243
* $0.0300/KWH	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.000	21.262
$0.035/KWH	693.109	27.530	4.576	0.094	4.834	0.690	0.026	0.000	21.281
$0.0450/KWH	695.798	27.636	3.849	0.051	4.840	0.692	0.026	0.000	21.291
$0.0600/KWH	695.798	27.636	3.849	0.068	4.840	0.692	0.026	0.000	21.308
Water Cost									
* $0.00/1000 gal.	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.000	21.262
$0.10/1000 gal.	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.092	21.353
$0.50/1000 gal.	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.462	21.724
$1.00/1000 gal.	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.925	22.187
Water Treatment									
$0.05/1000 gal.	693.109	27.530	4.576	0.075	4.834	0.690	0.013	0.000	21.249
* $0.10/1000 gal.	693.109	27.530	4.576	0.075	4.834	0.690	0.026	0.000	21.262
$0.25/1000 gal.	693.109	27.530	4.576	0.075	4.834	0.690	0.066	0.000	21.302
$0.50/1000 gal.	693.109	27.530	4.576	0.075	4.834	0.690	0.133	0.000	21.366

Table 4.4 Cost Sensitivity Study for Cooling Ponds - 800 MW Fossil Plant

	Plant Construction ($\$10^6$)	Cooling System ($\$10^6$)	Replacement Capacity ($\$10^6$)	Replacement Energy ($\frac{Mills}{KWH}$)	Fuel Cost ($\frac{Mills}{KWH}$)	Maintenance ($\frac{Mills}{KWH}$)	Water Treatment ($\frac{Mills}{KWH}$)	Water Cost ($\frac{Mills}{KWH}$)	Total Production Cost ($\frac{Mills}{KWH}$)
* Base Case:	394.503	14.567	1.892	0.065	7.877	0.586	0.019	0.000	21.839
	12.760 $\frac{Mills}{KWH}$	0.471 $\frac{Mills}{KWH}$	0.061 $\frac{Mills}{KWH}$						
Sensitivity:									
Plant Cost									
$375/KW	296.617	14.604	1.577	0.038	7.885	0.446	0.019	0.000	18.508
* $500/KW	394.503	14.567	1.892	0.065	7.877	0.586	0.019	0.000	21.839
$750/KW	587.315	14.458	2.838	0.188	7.848	0.863	0.019	0.000	28.473
Fuel Cost									
$0.0023/KWH	394.503	14.567	1.892	0.065	5.849	0.586	0.019	0.000	19.811
* $0.0031/KWH	394.503	14.567	1.892	0.065	7.877	0.586	0.019	0.000	21.839
$0.0046/KWH	393.516	14.309	2.208	0.100	11.686	0.585	0.019	0.000	25.658
$0.0061/KWH	392.530	14.494	2.523	0.139	14.926	0.584	0.019	0.000	29.469
Fixed Charge Rate									
15%	394.503	14.567	1.892	0.065	7.879	0.586	0.019	0.000	20.277
* 17%	394.503	14.567	1.892	0.065	7.877	0.586	0.019	0.000	21.839
20%	393.516	14.531	2.208	0.099	7.871	0.585	0.019	0.000	24.185
Capacity Factor									
0.50	392.530	14.494	2.523	0 139	7.861	0.584	0.019	0.000	28.472
* 0.75	394.503	14.567	1.892	0.065	7.877	0.586	0.019	0.000	21.839
Cooling System Multiplier									
0.75	394.503	13.363	1.892	0.065	7.879	0.585	0.019	0.000	21.801
* 1.00	394.503	14.567	1.892	0.065	7.877	0.586	0.019	0.000	21.839
1.50	394.503	16.976	1.892	0.065	7.874	0.590	0.019	0.000	21.923
Replac. Capac.									
$120/KW	393.516	14.531	1.656	0.099	7.871	0.584	0.019	0.000	21.824
* $160/KW	394.503	14.567	1.892	0.065	7.877	0.586	0.019	0.000	21.839
$240/KW	395.490	14.604	2.365	0.039	7.880	0.589	0.019	0.000	21.874
Replac. Energy									
$0.0225/KWH	392.530	14.494	2.523	0.104	7.860	0.584	0.019	0.000	21.814
* $0.0300/KWH	394.503	14.567	1.892	0.065	7.877	0.586	0.019	0.000	21.839
$0.0375/KWH	395.490	14.604	1.577	0.048	7.886	0.587	0.019	0.000	21.855
$0.0450/KWH	396.476	14.640	1.261	0.028	7.891	0.588	0.019	0.000	21.864
$0.0600/KWH	396.476	14.640	1.261	0.037	7.891	0.588	0.019	0.000	21.873
Water Cost									
* $0.00/1000 gal.	394.503	14.567	1.892	0.065	7.877	0.586	0.019	0.000	21.839
$0.10/1000 gal.	394.503	14.567	1.892	0.065	7.877	0.586	0.019	0.064	21.903
$0.50/1000 gal.	394.503	14.567	1.892	0.065	7.877	0.586	0.019	0.318	22.157
$1.00/1000 gal.	394.503	14.567	1.892	0.065	7.877	0.586	0.019	0.636	22.475
Water Treatment									
$0.05/1000 gal.	394.503	14.567	1.892	0.065	7.877	0.586	0.010	0.000	21.830
* $0.10/1000 gal.	394.503	14.567	1.892	0.065	7.877	0.586	0.019	0.000	21.839
$0.25/1000 gal.	394.503	14.567	1.892	0.065	7.877	0.586	0.048	0.000	21.868
$0.50/1000 gal.	394.503	14.567	1.892	0.065	7.877	0.586	0.095	0.000	21.915

Chapter V

WET TOWERS

5.1 Introduction

The wet (or evaporative) cooling towers commonly used by the power industry include both natural draft and mechanical draft types. In both, water from the condenser is distributed in fine droplets over an internal fill. Circulating air is brought into direct contact with the water to promote heat transfer, primarily by evaporation. The cooled water is then returned to the condenser. Because they do not have the large land requirements of cooling ponds or the considerably lower thermodynamic efficiency of dry towers, evaporative towers are perhaps the simplest cooling alternative in situations where water supply or thermal standards prevent once-through cooling.

It should be pointed out that while we are considering wet towers only for use as a closed cycle cooling system, they may also be used to supplement once-through cooling, i.e., the condenser cooling water can be circulated through a wet tower, before being discharged to the receiving water body, in order to reduce the thermal impact. In a parallel effort, included as part of the present project, a case study of TVA's Browns Ferry Nuclear Power Plant cooling system has been undertaken. The cooling system consists of a submerged diffuser and banks of mechanical draft wet towers. The objective of the research has been to determine the optimal use of open cycle, closed cycle or helper cycle modes in order to meet prescribed constraints on induced temperature rise.

Like ponds, wet towers consume large amounts of water due primarily to evaporation and are thermodynamically less efficient than once-through systems. Furthermore, evaporative towers respond more quickly to changes in the ambient meteorology than do ponds or once-through and thus may exhibit poorer performance during extreme environmental conditions. Finally, the auxiliary power required to pump the circulating water to the top of the fill imposes an additional operating cost.

The two types of wet towers studied here are physically very different, yet use the same heat transfer processes. The natural draft cooling tower is a tall hyperbolic chimney with a height up to about 500 feet and a base diameter up to about 450 feet. The mechanical draft tower is less than 100 feet high and uses fans to circulate air through the tower. Mechanical draft towers can either be forced draft (fan located at the bottom of the tower section) of induced draft (fan located at the top of the tower).

Heat transfer in the wet tower occurs when the free energy content available for exchange (or enthalpy) of the water is greater than that of the air. It is this differential in enthalpies which determines the tower's capacity to remove waste heat. The enthalpy of the cooling water is a function of its temperature. Because the dominant component in evaporative tower heat transfer is evaporation, the "effective" enthalpy of the air is determined primarily by its wet-bulb temperature. Therefore, the rate of heat transfer is governed by the difference between the temperature of the hot water passing through the tower and the wet-bulb temperature of the air entering the tower.

Three additional factors influence the cooling performance of a tower. The first is the total area of the air-water interface available for heat transfer. The cooling water is passed down through an internal fill of baffles or plates, whose purpose is to break the flow into droplets (baffles) or to spread it into thin sheets (plates).

The second factor affecting cooling is the ratio of air flow to water flow within the tower. A high air to water ratio serves two purposes: first, it creates a larger effective sink into which waste heat may be transferred; second, it insures unheated ambient air is quickly replacing heated air, thus maintaining the large enthalpy differential necessary for cooling. Natural draft towers take advantage of the fact that heated, vapor-laden air leaving the fill has a lower density than the surrounding ambient air, establishing a buoyancy force. The product of this buoyant force and shell height represents the energy available for circulating air through the tower. For a given shell height and buoyant force the maximum air flow is a function of the head loss of the air mass passing through the tower.

The third factor to affect the performance of a tower is the direction of the air flow ("counterflow" vs. "cross-flow") relative to the direction of the water flow. In counterflow towers, the two flows pass in opposite directions with the water passing down through the fill and the air passing up through the fill. The advantage of this arrangement is that while the circulating air is becoming warmer as it approaches the top, it is coming into contact with progressively warmer water. This flow configuration serves to maintain a fairly constant enthalpy differential along the length of the two flows. This configuration has the disadvantage of larger head loss through the fill, thereby diminishing the air flow through the tower. In "cross-flow" towers, the air stream passes at right angles to the downward path of the water. While this arrangement does not offer the thermodynamic advantage of maintaining a constant enthalpy differential along the air flow path, it avoids some of the head loss effects present in the counterflow.

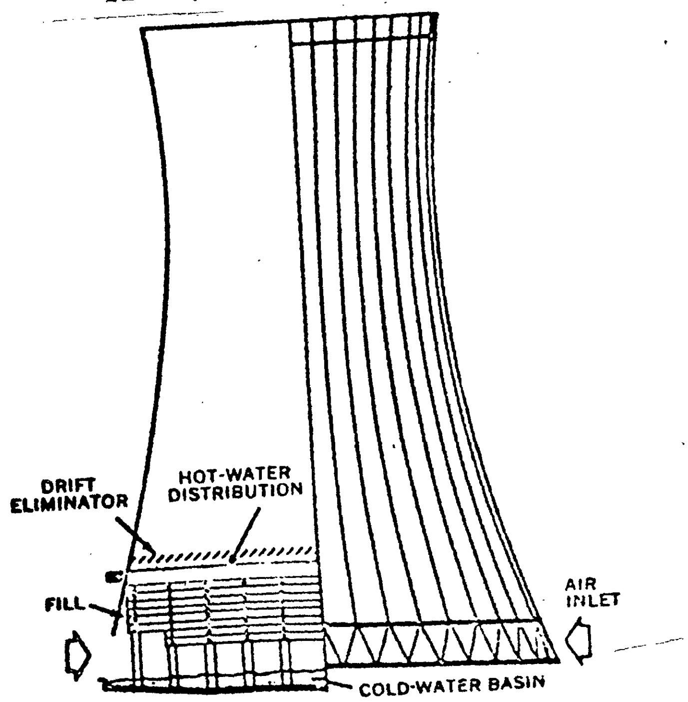
To illustrate the types of wet tower configurations available, Figure 5.1 shows a counterflow natural draft tower and a cross flow mechanical draft tower.

5.2 Environmental Factors

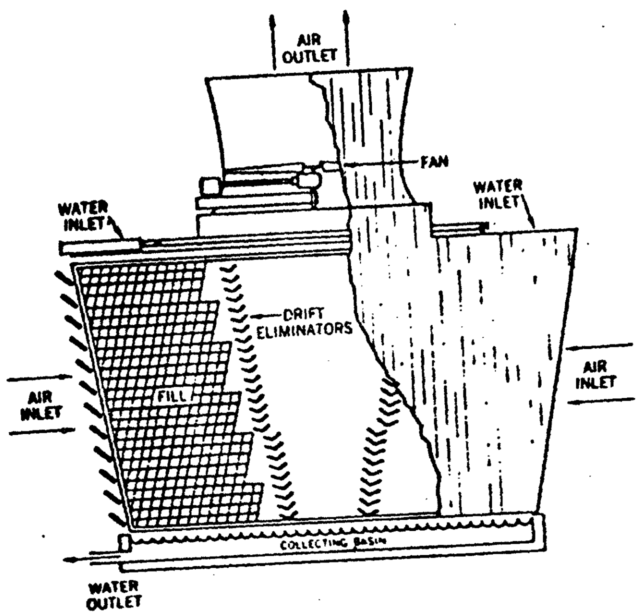
Like cooling ponds, wet towers eliminate much of the adverse impact caused by the heated discharge and the intake of once-through systems. The restrictions on heat discharged by blowdown as well as the treatment of blowdown are given by the Effluent Guidelines (40 CFR 423, 1974) and are similar to the guidelines given for ponds.

Evaporative cooling towers were considered by the EPA to be the best available technology for abating the thermal impact from steam-electric power plants. However, while wet towers may eliminate some of the adverse effects of once-through cooling, they are not without their own impacts. In addition to higher costs, wet towers consume considerably more water than once-through systems. This is due mostly to evaporation which constitutes about 75% of the heat transfer from the wet tower. Make-up water is also required due to water loss from drift (about 0.2% of condenser flow) and blowdown (about 1 to 2%).

Figure 5.1 Wet Cooling Towers (a) Natural Draft Counterflow (b) Mechanical Draft Crossflow (from Burns and Roe, 1973)



(a)



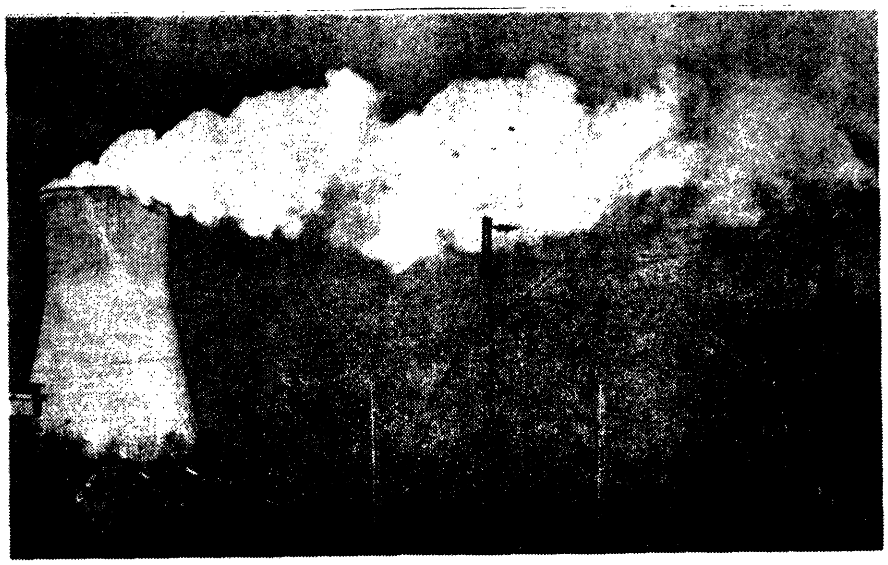
(b)

The remaining environmental impacts associated with wet towers are specific to either the mechanical draft or the natural draft tower. Fogging and icing caused by plume condensation, and particle drift are more of a problem for the mechanical draft tower since it releases its plume less than 100 feet above ground level. In addition, noise impacts are most significant in the mechanical draft tower due to fan operation. While natural draft operation requires no fans and can reduce ground level fogging by releasing its plume at a much greater height, it has greater aesthetic impacts. Both types of towers can have land use impacts but these are much less than for the cooling pond, with natural draft towers, generally having smaller land requirements. Finally, mechanical draft towers must be carefully designed against hot air recirculation and air flow interference while the natural draft needs to avoid aviation problems and hurricane threat.

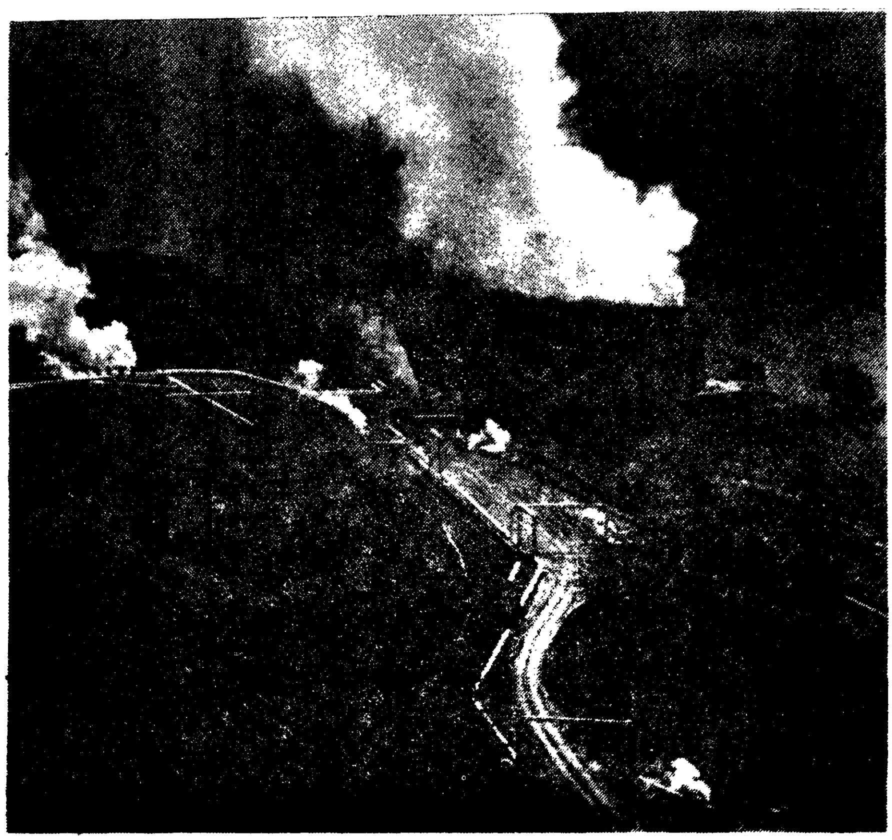
Evaluation of the environmental effects from wet towers is generally difficult due to the site specific nature of the impacts. However, some quantification of the fogging impact will be made here and evaluation of the water consumption is made in Section 5.4 since these are the more significant effects associated with evaporative cooling.

Figure 5.2 illustrates fogging from both a natural and a mechanical draft wet tower. While the aesthetic impact of the natural draft tower is obvious from these photographs, the fogging impact is more serious with the mechanical draft tower where the vapor plume is more likely to diffuse to the ground level.

Figure 5.2 Fogging from Wet Cooling Towers



(from Bogh, 1974)



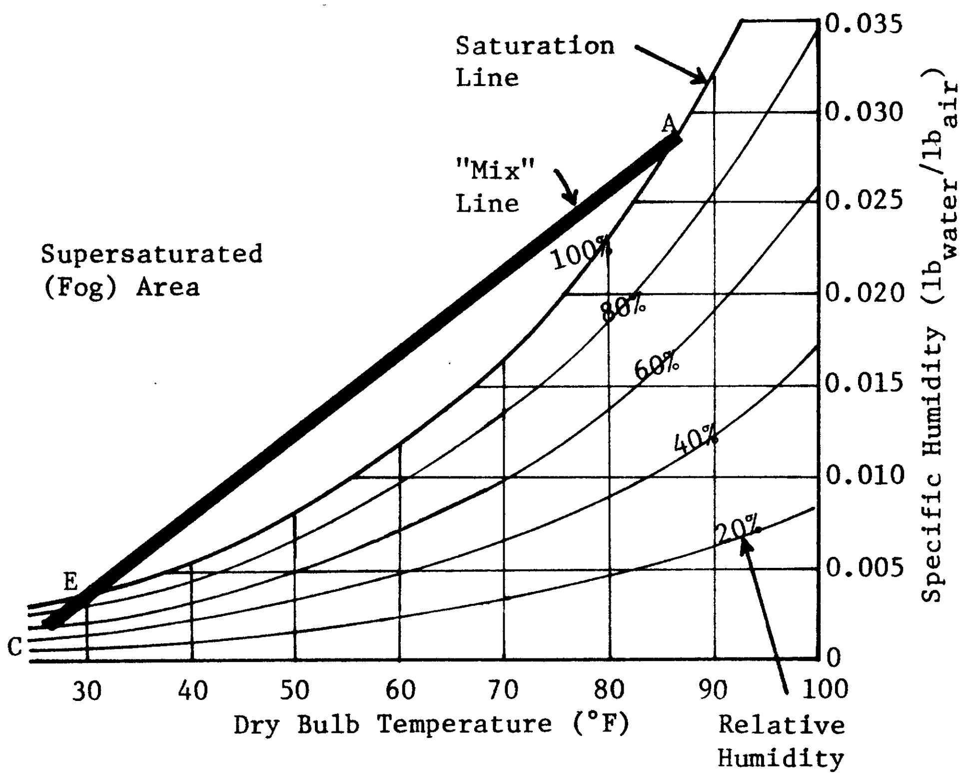
(from Pacific Gas and Electric Co. as reported in Wilson and Jones, 1974)

Fog is produced when the warm, almost saturated air from the tower mixes with the cooler ambient air. As the air becomes cooler, saturation and supersaturation with respect to water vapor content occurs resulting in vapor condensation into droplets of fog.

The modeling of fogging plumes used in this study follows Croley et al. (1975) for the mechanical draft evaporative tower, where fogging considerations are limited to (1) occurrence of visible plume and (2) the severity of the plume. The path of the plume which determines whether ground fogging will occur is dependent upon wind direction and velocity and is not considered here. But it is recognized that in the absence of wind, the buoyant force which causes the plume to rise is the major force acting on the plume.

Fogging is measured by indicators based on the saturation curve. A linear "mix" line on the psychrometric chart shown in Figure 5.3 was assumed to apply for the plume temperature and humidity as the tower exhaust returns to ambient conditions. The saturation line describes the locus of points where air is just saturated with water vapor. At points above this line, the air is supersaturated producing a visible fog condition. Thus, whenever the mix line crosses the saturation line a visible plume occurs (e.g. from A to E). Point A in the figure is the saturated state that is assumed to represent the cooling tower exhaust while point C corresponds to a possible ambient atmospheric condition. The assumed thorough mixing of cooling tower effluent and atmospheric air presumably follows the straight line from A to C. The fogging severity is defined as the area between the saturation curve and the mix line that

Figure 5.3 Plume/Atmosphere Interaction (From Reisman, 1973)



lies in the supersaturated portion of the psychrometric chart.

The results of the plume model are shown in Table 5.1. Using the discrete bivariate distribution of wet bulb and dry bulb temperatures that characterize the study site , four different tower designs (represented by tower lengths) applied with the 1200 MW nuclear plant are evaluated and compared for (1) fogging severity (lb water $\cdot$ ^{o}F/lb air) at a select meteorological condition (30^{o} wet bulb, 30^{o} dry bulb, probability of occurrence = 12%) and (2) frequency at which the fogging severity exceeds a selected value (.10 lb water $\cdot$ ^{o}F/lb air).

5.3 Performance Model

The model examines splash fill, cross-flow configurations for both the mechanical draft and natural draft towers. The decision to examine the cross-flow configuration rather than a counterflow arrangement was due to the availability of a flexible thermodynamic program for the former and not due to any a priori assumption that the cross-flow was economically preferable to the counterflow.

Merkel developed the governing equations for heat transfer between the cooling water and the circulating air in 1925. It is the solution of these equations, with a computer algorithm, which essentially determines the model used to predict a cooling tower's thermodynamic performance.

Examining an elementary volume of the fill (Figure 5.4) we observe that water enters with a temperature "t_i" and leaves at temperature "t_o". The rate of heat loss by the water is equal to:

$$\frac{L\ \Delta x \Delta y}{x\ z}\ (dt)\ k \tag{5.1}$$

Table 5.1 Results of Fogging Model

Tower Length (ft)	Fogging Severity at 30° DBT, 30° WBT (lb water·°F/lb air)	Frequency at which Fogging Severity > 0.10 %	days/year
800	0.257	53	193
1000	0.160	42	153
1200	0.108	25	91
1400	0.074	0	0

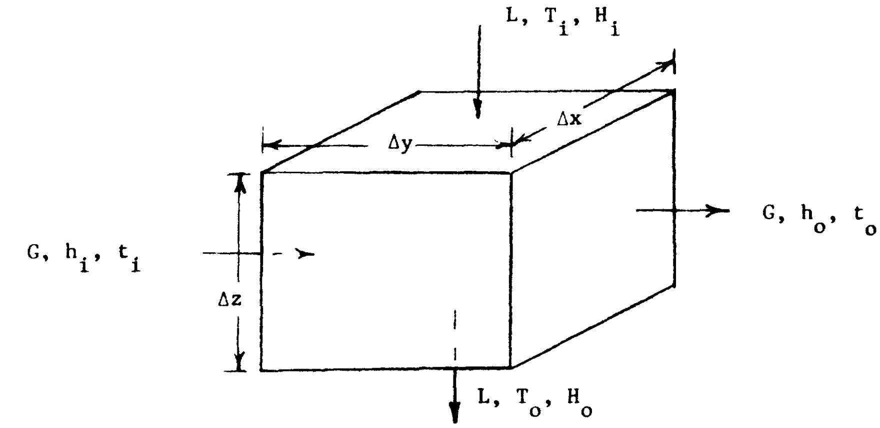


Fig. 5.4 Differential Control Volume Used to Describe Heat Transfer in an Evaporative Tower

Nomenclature

L: Tower circulating water flow (lb/hr)

G: Tower circulating air flow (lb/hr)

y: Width of tower fill (feet)

x: Length of tower fill (feet)

z: Depth of tower fill (feet)

Δy: Incremental width of fill (feet)

Δx: Incremental length of fill (feet)

Δz: Incremental depth of fill (feet)

k: Specific heat of water (BTU/lb/^{o}F)

t: Circulating water temperature within incremental fill volume, $T_i \leq t \leq T_o$ (^{o}F)

h_i: Enthalpy of air entering incremental fill volume (BTU/lb)

h_o: Enthalpy of air leaving incremental fill volume (BTU/lb)

K: Effective mass transfer between water and air within tower (lb/hr/ft^2)

a: Effective area available for heat transfer per unit volume of fill (ft^2/ft^3)

h: Enthalpy of air within incremental fill volume, $h_i \leq h \leq h_o$ (BTU/lb)

H_i: Enthalpy of water entering incremental fill volume (BTU/lb)

H_o: Enthalpy of water leaving incremental fill volume (BTU/lb)

h': Enthalpy of water within incremental fill volume $H_o \leq h' \leq H_i$ (BTU/lb)

T_i: Temperature of water entering incremental fill volume, $\tau_2 \leq T_i \leq \tau_1$ (^{o}F)

V: Volume of tower fill ($V = x \cdot y \cdot z$) (ft^3)

t_i: Wet-bulb temperature of air entering incremental fill volume, $T_2 \leq t_i \leq T_1$ (^{o}F)

t_o: Wet-bulb temperature of air leaving incremental fill volume, $T_2 \leq t_o \leq T_1$ (^{o}F)

τ_1: Temperature of Circulating water entering tower (^{o}F)

τ_2: Temperature of Circulating water leaving tower (^{o}F)

T_1: Wet-bulb temperature of air entering tower (^{o}F)

T_2: Wet-bulb temperature of air leaving tower (^{o}F)

At the same time, air enters this volume with an enthalpy "h_i" and leaves with an enthalpy "h_o". The rate of heat gain by the air is equal to:

$$\frac{G}{x\ z} \Delta z\ \Delta x\ (dh) \tag{5.2}$$

For a condition of thermodynamic equilibrium the following equality must hold:

$$\frac{L}{x\ y} \Delta x\ \Delta y\ (dt) = \frac{G}{x\ z} \Delta x\ \Delta z\ (dh) \tag{5.3}$$

This equality alone is not sufficient for a prediction of a particular tower configuration's cooling performance. We recall that the rate of heat transfer is proportional to the area of the air-water interface available for heat transfer. This area is a function of the efficiency of the fill in breaking up the water stream. It follows, then, that we require an additional term, in the equality above, which represents the area available for heat transfer. Merkel showed that within a reasonable approximation, the driving force for heat transfer across the interface is proportional to the difference between the average enthalpy of saturated air at the bulk water temperature (T_i) and the average enthalpy of saturated air at the air wet-bulb temperature (t_i). The constant of proportionality, K, has the units (lb/hr/ft^2). If we define a variable "a" which defines the area available for heat transfer per unit volume of fill (ft^2/ft^3), we can add an additional term to the equality in Equation (5.3) above:

$$\frac{L}{x\ y} \Delta x \Delta y dt = \frac{G}{x\ z} \Delta x\ \Delta z (dh) = Ka \Delta x \Delta y \Delta z (h'-h) \tag{5.4}$$

$$\frac{L}{x}\frac{\Delta x\ \Delta y\ dT}{y(h'-h)} = Ka\Delta x\Delta y\Delta z \qquad (5.4a)$$

$$\frac{G}{x}\frac{\Delta x\ \Delta z}{z}\ \frac{dh}{h'-h} = Ka\Delta x\Delta y\Delta z \qquad (5.4b)$$

The temperature of the water leaving the fill, τ_2, is found by integration of Equation (5.4) or

$$\frac{L}{x\ y} \int_o^x \Delta x \int_o^y \Delta y \int_{\tau_1}^{\tau_2} \int \frac{dT}{h'-h} = Ka \int_o^x \Delta x \int_o^y \Delta y \int_o^z \Delta z \qquad (5.5a)$$

$$\frac{G}{x\ z} \int_o^x \Delta x \int_o^z \Delta z \int_{h_1}^{h_z} \frac{dh}{h'-h} = Ka \int_o^x \Delta z \int_o^y \Delta y \int_o^z \Delta z \qquad (5.5b)$$

$$\int_{\tau_1}^{\tau_2} \frac{dt}{(h'-h)} = \frac{KaV}{L} \qquad (5.6a)$$

$$\int_{h_1}^{h_2} \frac{dt}{h'-h} = \frac{KaV}{G} \qquad (5.6b)$$

Equations (5.6a-b) are the integral forms of the Merkel equations. The computer algorithm developed by the Iowa Institute of Hydraulic Research (Croley et al., 1975/ hereafter referred to as the Iowa Model) approximates these integrals by solving Equation (5.4) for a number of elementary volumes of size Δx, Δy, and Δz within the fill.

For each volume

$$h'-h \approx 1/2(H_i + H_o) - 1/2(h_i + h_o) \tag{5.7}$$

$$dT = (T_i - T_o) \tag{5.8}$$

$$dh = (h_o - h_i) \tag{5.9}$$

$$\Delta z/\Delta y - z/y \text{ or } z/\Delta z - y/\Delta y - N' \tag{5.10}$$

Equation (5.10) says we assume, for computational purposes, that there are an equal number of elementary volumes sequenced horizontally as there are sequenced vertically. Substituting these approximations into Equation (5.4), we find:

$$h_o - h_i = \frac{KaV}{GN'} \frac{(H_i + H_o - h_i - h_o)}{2} \tag{5.11}$$

$$G(h_o - h_i) = L(T_1 - T_o) \tag{5.12}$$

If we think of the water as traveling down through the pile shown in Figure 5.5 and air as traveling from left to right across the pile (cross-flow configuration) then we see that:

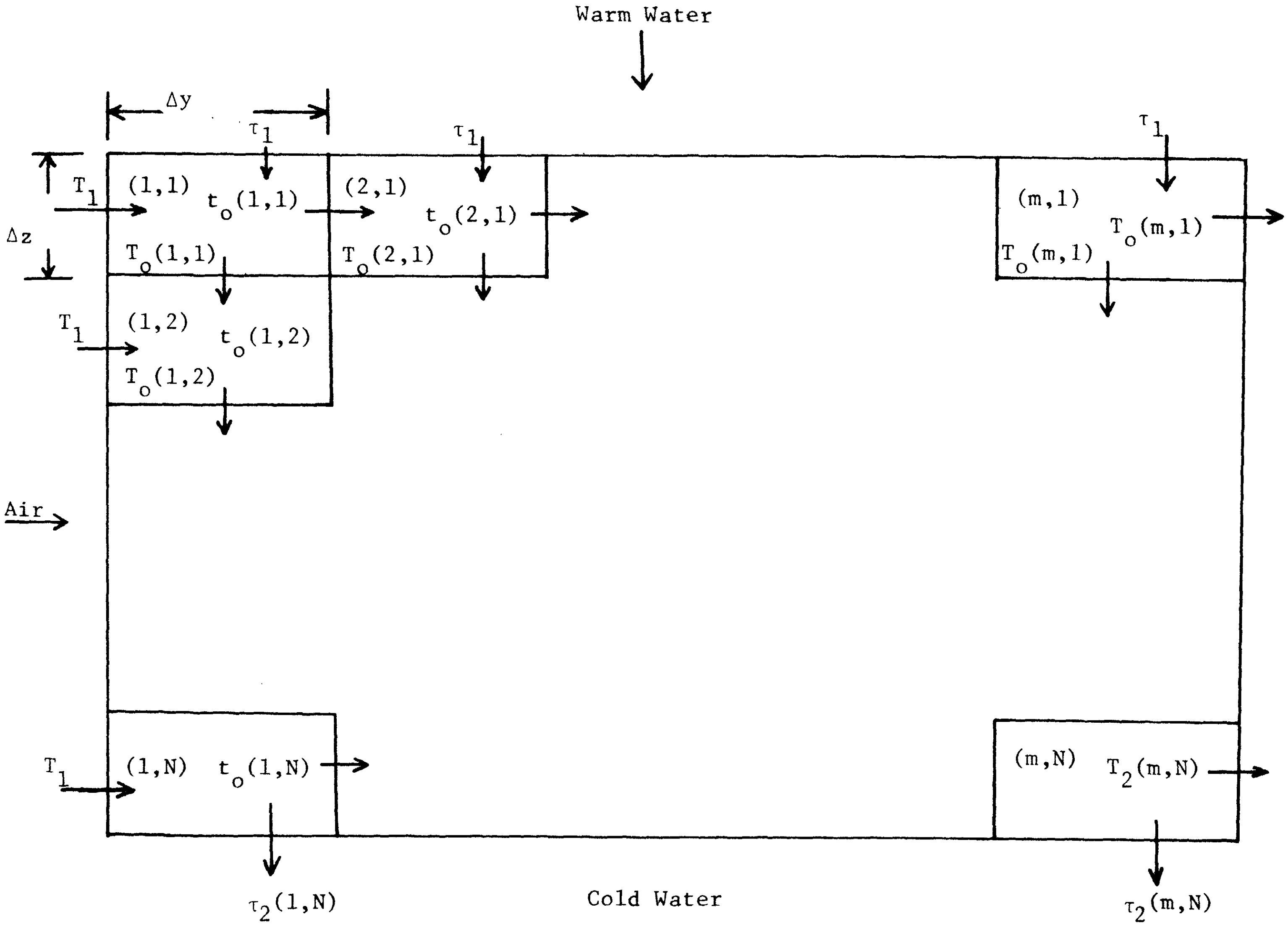


Figure 5.5 Schematization of Incremental Volumes within Fill

$$t_i(j,k) = t_o(j-1,k) \qquad 1 \leq j \leq m \quad ; \quad m = N \tag{5.13}$$

$$T_i(j,k) = T_o(j,\ k-1) \qquad 1 \leq k \leq N \tag{5.14}$$

where T_1 is the wet bulb temperature of the air entering the tower fill and τ_1 is the temperature of the hot water entering the tower fill.

Given T_1 and τ_1, the entrance conditions for the element (1,1), we can use Equations (5.11) and (5.12) to solve for the exit conditions from this element. The air exit condition from element (1,1) is the air entrance condition for the adjacent element (2,1). The water exit condition from element (1,1) is the water entrance condition for the adjacent element (1,2). Given the water entrance conditions for the first row (τ_1) and given the air entrance conditions for the first column (T_1) the computer algorithm uses this recursive procedure to solve the exit condition from every element.

The temperature of the "cold" water leaving the tower fill is:

$$\tau_2 = 1/m \sum_{j=1}^{m} \tau_2(j,N) \tag{5.15}$$

The temperature of the "hot" air leaving the tower fill is:

$$T_2 = 1/N \sum_{k=1}^{N} T_2(m,k) \tag{5.16}$$

As we have seen, the heat rejection capability of a tower can be written:

$$HR = f(\tau_1, T_1, G, L, Ka) \tag{5.17}$$

T_1, the current ambient wet bulb temperature, is known at any time. Recalling from Chapter II that there is a one to one relation between the temperature of the hot water leaving the condenser, τ_1, and the turbine heat rejection, the computer iterates on τ_1 until the turbine heat rejection is equal to the tower heat rejection. Thus, τ_1 is also a "given" value at any time.

The water flow through the tower, L, is determined by the plan water loading (gpm/ft^2/min) and the total plan area of the fill (the x-y plane). We use a constant water loading for all towers of 13 gal/min/ft^2. Therefore, for a given tower configuration L is a constant.

The air flow through the tower is a constant value for mechanical draft towers and is the product of a constant tower air loading (lb/ft^2/hr) in the x-z plane and the inlet area in this plane. Therefore the air flow is considered constant in the mechanical draft towers.

In natural draft towers the air loading is determined by the density difference between the warm, moist air leaving the fill and the cooler dry ambient air, the tower height, and the head losses within the tower. For a given shell height, under given meteorological conditions (ambient air density is known), the density difference is established by iteration. An initial density of the exhaust air is estimated, giving a first approximation for the air loading. The tower cooling and a new exhaust air density are then computed. This procedure is repeated with the new air density and these iterations are continued until the model converges on an air flow rate.

Head loss due to passage through the fill and due to entrance and exit losses restricts the flow of air passing through a tower. In mechanical draft towers fans are used to overcome this resistance. In natural draft towers the potential for flow due to the chimney effect balances this resistance. The models use values for pressure drop vs. inlet velocity which are appropriate for cross-flow evaporative towers (Croley, 1975). The final performance variable, Ka, was determined by experimental values offered by Lowe and Christie (1962). Guyer and Golay (1976) found that performance predicted by the Iowa Model using proprietary head loss data was consistent with the results of the Lowe and Christie experiments.

5.4 Optimization Details

The optimal cooling tower — whether natural or mechanical draft — is found by determining that system which minimizes the total cooling system cost (operation and construction) in accordance with the procedures discussed in Chapter II. For mechanical draft towers the primary design variable is tower length, while for natural draft towers the primary design variable is tower height.

The procedure for determining the capital cost of a wet mechanical draft tower employs the concept of a "tower unit." The tower unit is an index which represents the tower's cooling efficiency of a known "design" wet bulb temperature. The larger the number of tower units, the more efficient the tower. However, since the tower efficiency is a function of the complexity of the fill, amount of fill and air delivery capacity of the fan, the greater the number of tower units, the higher the capital cost of the tower. Costs per tower unit were estimated at \$10.00 based on Dickey and Cates (1973). Our sensitivity studies examined tower unit costs ranging from \$7.50 - \$15.00.

Natural draft towers are constructed at the site of use, unlike mechanical draft towers which come to the site largely pre-fabricated. Consequently, natural draft tower capital costs are more variable, depending on local labor and materials costs. Nevertheless, good correlation between the tower shell height and the cost per tower was found based on the data presented by Sebald (1976). Assuming a ratio of shell height to base diameter of 1:1 this relationship is \$/tower = \$38.630 x shell height (ft) - \$5,005,000.

Evaluation of operating costs for wet towers is similar to that used for once-through systems and cooling ponds. Tower performance is evaluated for each combination of dry bulb and wet bulb temperature found in a bi-variate distribution compiled from the site meteorological data. It should be noted that while the thermodynamic performance of the mechanical draft tower is dependent only on the wet bulb temperature, dry bulb temperatures are used to compute water loss through evaporation as a part of the total water balance of the system.

The water balance for wet cooling towers is computed in a manner similar to that done for cooling ponds in Equation 4.16. Thus the make-up flow is:

$$Q_M = Q_E + Q_D + Q_B \tag{5.18}$$

where Q_D = water drift losses. As with the cooling ponds the blowdown quantity is assumed to return to the make-up source after any treatment and thus is not included as water consumption. Also drift losses are small (about .2% of condenser flow) for towers with modern drift eliminators and thus will be neglected. Water consumption for wet towers, then, is due mainly to forced evaporation. This is computed in the process of evaluating the thermodynamic performance of the towers by assuming that the vapor which leaves the towers is fully saturated. Monthly evaporation rates for the 1200 MWe nuclear plant with mechanical draft towers during 1970 are presented in Chapter VII for comparison with the other systems. The annual average evaporation rate was 23.5 cfs which indicates that approximately 70% of the tower cooling was by evaporation.

The blowdown from the wet tower is needed to calculate the treatment costs and is evaluated following Equation (4.18) as

$$Q_B = \frac{Q_E\ C_M}{C_B - C_M} \tag{5.19}$$

The maximum concentration C_B, allowable for condenser and cooling tower operation was assumed to be 350 ppm according to Croley (1975) while the make-up water at the site was assumed to be 100 ppm. The resulting blowdown flow for the mechanical draft towers is 9.4 cfs. The make-up water flow to be costed is then 32.9 cfs.

5.5 Results

As discussed in Section 5.4, mechanical draft wet towers are optimized according to the tower length (number of tower modules) and the natural draft towers are optimized by tower shell height. Tables 5.2 through 5.5 show the total power production cost for each type of tower as the sum of capital, operating and penalty costs for the nuclear and fossil plants using varying designs of natural and mechanical draft towers. For the mechanical draft towers the optimal tower lengths are found to be 1200 ft and 500 ft for the nuclear and fossil plants, respectively. The optimal vertical shell height for the natural draft towers is 375 ft and 300 ft for the nuclear and fossil plants, respectively.

A sensitivity study of power production cost to tower size was made for all the economic parameters discussed in Chapter II. Sensitivity to the cooling tower multiplier is shown in Figures 5.6 and 5.7 for the nuclear plant using natural and mechanical draft cooling towers. Sensitivity to the other factors is not shown graphically since their behavior is similar to that of the once-through system; however the sensitivity is included in the summary tables.

In Figures 5.6 and 5.7 the optimal cooling tower size decreases as the capital cost of the cooling tower increases. The shift to smaller sizes is consistent with the other cooling systems, but a greater shift is evidenced for the towers due to their larger capital cost.

Since, the cost of water may be significant for wet towers, sensitivity studies were made using prices for make-up water ranging from

Table 5.2 Power Production Cost Versus Tower Height-1200 MW Nuclear Plant

Cost Component (mills/KWH)	Natural Draft Tower Height (ft)				
	300	325	350	375	400
Plant Construction	15.125	15.117	15.113	15.059	15.069
Cooling System	0.586	0.640	0.693	0.743	0.796
Replacement Capacity	0.216	0.181	0.153	0.132	0.113
Fuel	4.879	4.878	4.878	4.871	4.874
Replacement Energy	0.177	0.099	0.068	0.099	0.078
Maintenance	0.703	0.703	0.704	0.703	0.705
Water Treatment	0.021	0.022	0.023	0.024	0.025
Total Power Production Cost	21.708	21.640	21.632	21.631	21.659

Table 5.3 Power Production Cost Versus Tower Height-800 MW Fossil Plant

Cost Component (mills/KWH)	Natural Draft Tower Height (ft)				
	250	275	300	325	350
Plant Construction	12.920	12.860	12.844	12.824	12.815
Cooling System	0.416	0.455	0.496	0.536	0.578
Replacement Capacity	0.212	0.091	0.086	0.076	0.072
Fuel	7.950	7.937	7.921	7.912	7.908
Replacement Energy	0.064	0.086	0.079	0.084	0.080
Maintenance	0.598	0.591	0.592	0.593	0.594
Water Treatment	0.015	0.015	0.015	0.016	0.016
Total Power Production Cost	22.174	22.035	22.033	22.039	22.055

Table 5.4 Power Production Cost Versus Tower Length-1200 MW Nuclear Plant

Cost Component (mills/KWH)	Mechanical Draft Tower Length (ft)				
	900	1000	1100	1200	1300
Plant Construction	15.176	15.165	15.165	15.169	15.177
Cooling System	0.482	0.521	0.556	0.593	0.629
Replacement Capacity	0.212	0.168	0.135	0.113	0.093
Fuel	4.900	4.897	4.899	4.900	4.903
Replacement Energy	0.238	0.151	0.095	0.064	0.044
Maintenance	0.700	0.699	0.700	0.700	0.701
Water Treatment	0.020	0.020	0.020	0.020	0.020
Total Power Production Cost	21.730	21.621	21.570	21.560	21.568

Table 5.5 Power Production Cost Versus Tower Length-800 MW Fossil Plant

Cost Component (mills/KWH)	Mechanical Draft Tower Length (ft)				
	400	500	600	700	800
Plant Construction	12.945	12.895	12.876	12.870	12.874
Cooling System	0.393	0.448	0.502	0.555	0.607
Replacement Capacity	0.078	0.067	0.060	0.056	0.053
Fuel	7.985	7.956	7.946	7.944	7.947
Replacement Energy	0.087	0.075	0.067	0.062	0.059
Maintenance	0.592	0.592	0.593	0.595	0.597
Water Treatment	0.014	0.014	0.014	0.015	0.015
Total Power Production Cost	22.092	22.045	22.059	22.096	22.152

Figure 5.6 Sensitivity Study for Tower Multiplier
Natural Draft Wet Tower
Nuclear Plant

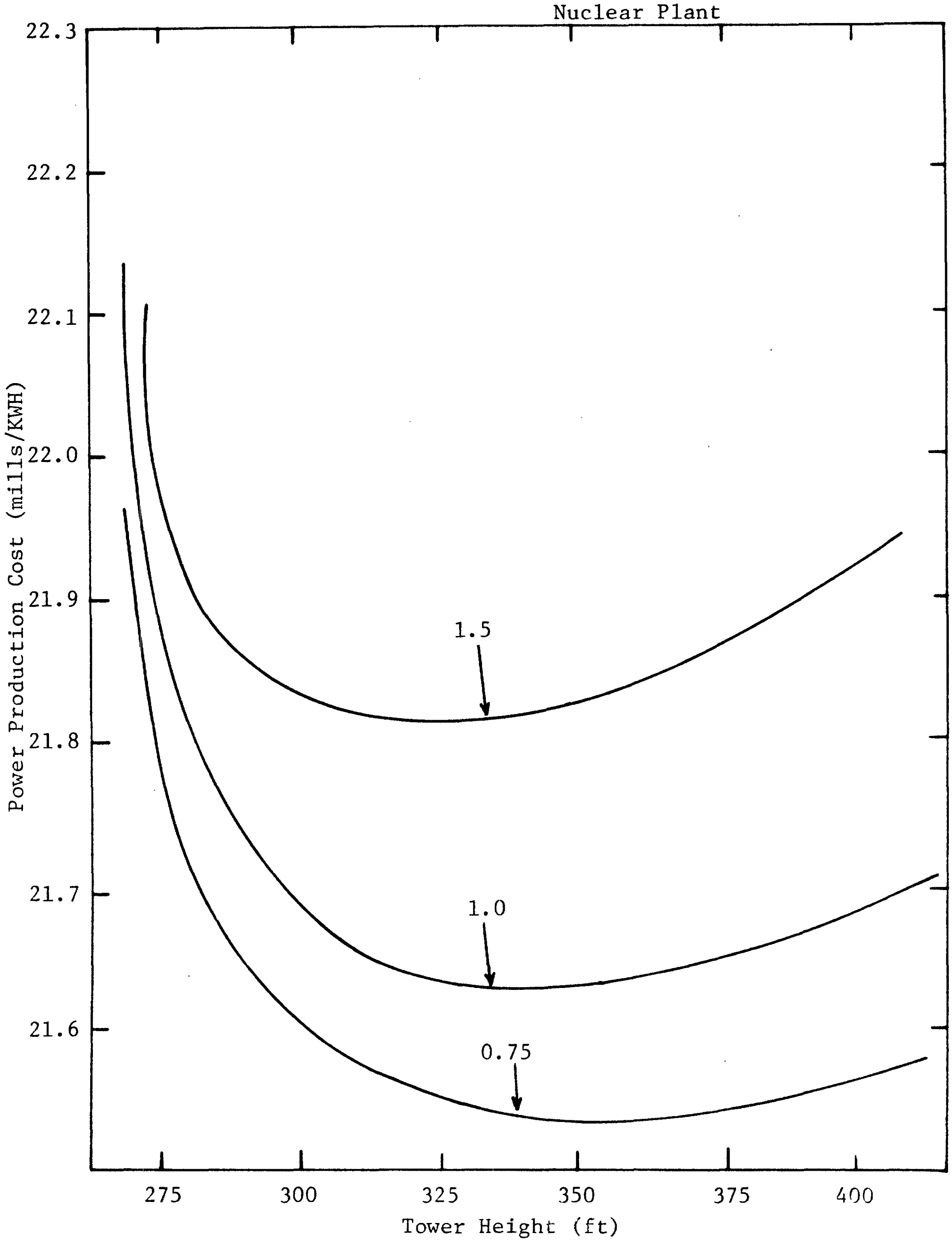
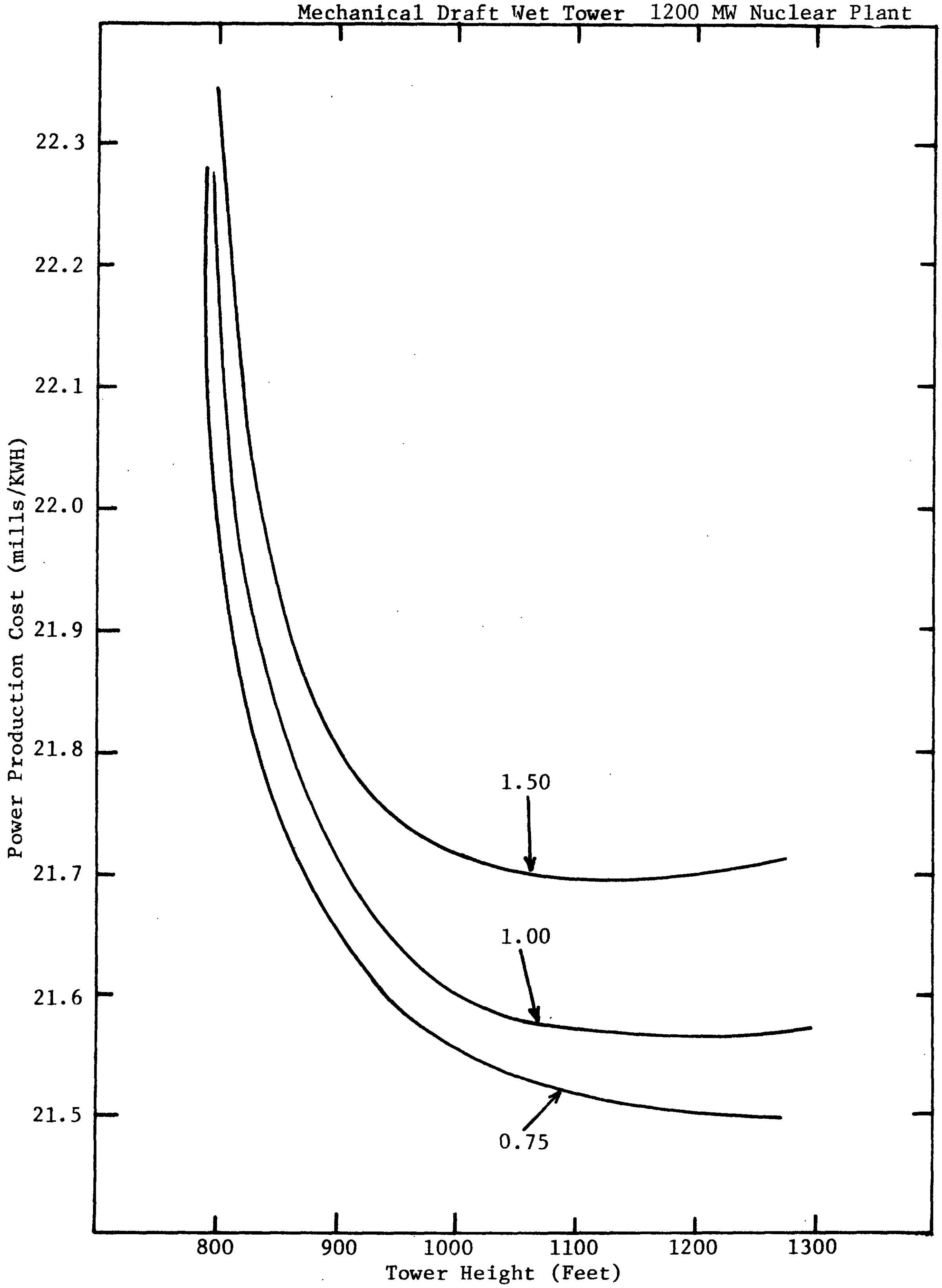


Figure 5.7 Sensitivity Study for Cooling System Multiplier
Mechanical Draft Wet Tower 1200 MW Nuclear Plant



\$0.00/1000 gal to \$1.00/1000 gal. This range of costs could include construction of storage ponds for wet tower operation in water scarce regions. The sensitivity study, however, showed little shift in the size of optimal tower size associated with increases in make-up water prices.

Sensitivity to cost of blowdown treatment was studied for values from \$0.05/1000 gal to \$0.50/1000 gal. In addition to treatment, these values could include the cost of a blowdown diffuser. Such diffusers are frequently being designed for stations on relatively small rivers to supplement treatment by diluting the waste heat, chemical constituents, or low-level radioactive concentrations associated with the blowdown discharge. As with the cost of water, variation in power production cost is small for increases in treatment costs and there is negligible shift of optimal tower sizes in response to this parameter.

Tables 5.6 to 5.9 summarize the design and sensitivity study for all the wet tower systems. The values in the tables follow the same format as in the other cooling system and coincide with the transferred system of each system using the optimal tower sizes from the base case parameters.

Table 5.6 Cost Sensitivity for Natural Draft Wet Cooling Towers - 1200 MW Nuclear Plant

	Plant Construction ($\$10^6$)	Cooling System ($\$10^6$)	Replacement Capacity ($\$10^6$)	Replacement Energy (Mills/KWH)	Fuel Cost (Mills/KWH)	Maintenance (Mills/KWH)	Water Treatment (Mills/KWH)	Water Cost (Mills/KWH)	Total Production Cost (Mills/KWH)
* Base Case:	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.000	21.631
	15.059 Mills/KWH	0.743 Mills/KWH	0.132 Mills/KWH						
Sensitivity:									
Plant Cost									
$450/KW	525.506	34.577	5.516	0.051	4.879	0.538	0.024	0.000	17.687
* $600/KW	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.000	21.631
$900/KW	1047.557	34.464	6.129	0.099	4.871	1.035	0.024	0.000	29.492
Fuel Cost									
$0.0012/KWH	700.674	34.577	5.516	0.051	3.659	0.705	0.024	0.000	20.411
* $0.0016/KWH	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.000	21.631
$0.0024/KWH	698.372	34.464	6.129	0.099	7.337	0.703	0.024	0.000	24.097
$0.0032/KWH	698.372	34.464	6.129	0.099	9.773	0.703	0.024	0.000	26.532
Fixed Charge Rate									
15%	700.674	34.577	5.516	0.051	4.879	0.705	0.024	0.000	19.752
* 17%	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.000	21.631
20%	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.000	24.443
Capacity Factor									
0.50	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.000	29.598
* 0.75	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.000	21.631
Cooling System Multiplier									
0.75	698.372	29.865	6.129	0.099	4.981	0.700	0.024	0.000	21.527
* 1.00	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.000	21.631
1.50	698.372	43.660	6.129	0.099	4.871	0.706	0.024	0.000	21.838
Replac. Capac.									
$120/KW	698.372	34.464	4.596	0.099	4.871	0.702	0.024	0.000	21.596
* $160/KW	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.000	21.631
$240/KW	700.674	34.577	8.274	0.051	4.879	0.705	0.024	0.000	21.693
Replac. Energy									
$0.0225/KWH	698.372	34.464	6.129	0.074	4.871	0.703	0.024	0.000	21.606
* $0.0300/KWH	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.000	21.631
$0.0375/KWH	700.674	34.577	5.516	0.063	4.879	0.705	0.024	0.000	21.644
$0.0450/KWH	700.674	34.577	5.516	0.076	4.879	0.705	0.024	0.000	21.656
$0.0600/KWH	700.674	34.577	5.516	0.101	4.879	0.705	0.024	0.000	21.682
Water Cost									
* $0.0/1000 gal.	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.000	21.631
$0.1/1000 gal.	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.084	21.715
$0.5/1000 gal.	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.422	22.052
$1.0/1000 gal.	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.844	22.474
Water Treatment									
$0.05/1000 gal.	698.372	34.464	6.129	0.099	4.871	0.703	0.012	0.000	21.619
* $0.10/1000 gal.	698.372	34.464	6.129	0.099	4.871	0.703	0.024	0.000	21.631
$0.25/1000 gal.	698.372	34.464	6.129	0.099	4.871	0.703	0.059	0.000	21.666
$0.50/1000 gal.	698.372	34.464	6.129	0.099	4.871	0.703	0.118	0.000	21.725

Table 5.7 Cost Sensitivity for Natural Draft Wet Cooling Towers - 800 MW Fossil Plant

	Plant Construction ($\$10^6$)	Cooling System ($\$10^6$)	Replacement Capacity ($\$10^6$)	Replacement Energy ($\frac{Mills}{KWH}$)	Fuel Cost ($\frac{Mills}{KWH}$)	Maintenance ($\frac{Mills}{KWH}$)	Water Treatment ($\frac{Mills}{KWH}$)	Water Cost ($\frac{Mills}{KWH}$)	Total Production Cost ($\frac{Mills}{KWH}$)
* **Base Case:**	397.099	15.326	2.668	0.079	7.921	0.592	0.015	0.000	22.033
	12.844 $\frac{Mills}{KWH}$	0.496 $\frac{Mills}{KWH}$	0.086 $\frac{Mills}{KWH}$						
Sensitivity:									
Plant Cost									
$375/KW	298.730	15.372	2.287	0.044	7.929	0.451	0.015	0.000	18.673
* $500/KW	397.099	15.326	2.668	0.079	7.921	0.592	0.015	0.000	22.033
$750/KW	592.026	15.232	3.431	0.192	7.892	0.871	0.015	0.000	28.723
Fuel Cost									
$0.0023/KWH	397.099	15.326	2.668	0.079	5.880	0.592	0.015	0.000	19.969
* $0.0031/KWH	397.099	15.326	2.668	0.079	7.921	0.592	0.015	0.000	22.033
$0.0046/KWH	397.099	15.326	2.668	0.079	11.760	0.592	0.015	0.000	25.872
$0.0061/KWH	395.892	15.279	3.049	0.128	15.574	0.591	0.015	0.000	29.706
Fixed Charge Rate									
15%	398.307	15.372	2.287	0.044	7.929	0.594	0.015	0.000	20.453
* 17%	397.099	15.326	2.668	0.079	7.921	0.592	0.015	0.000	22.033
20%	397.099	15.326	2.668	0.079	7.921	0.592	0.015	0.000	24.102
Capacity Factor									
0.50	395.892	15.279	3.049	0.128	7.908	0.592	0.015	0.000	28.739
* 0.75	397.099	15.326	2.668	0.079	7.921	0.592	0.015	0.000	22.033
Cooling System Multiplier									
0.75	397.099	13.691	2.668	0.079	7.921	0.591	0.015	0.000	21.978
* 1.00	397.099	15.326	2.668	0.079	7.921	0.592	0.015	0.000	22.033
1.50	397.099	18.594	2.668	0.079	7.921	0.594	0.015	0.000	22.143
Replac. Capac.									
$120/KW	397.099	15.326	2.001	0.079	7.921	0.591	0.015	0.000	22.010
* $160/KW	397.099	15.326	2.668	0.079	7.921	0.592	0.015	0.000	22.033
$240/KW	398.307	15.372	3.431	0.044	7.929	0.594	0.015	0.000	22.074
Replac. Energy									
$0.0225/KWH	395.892	15.279	3.050	0.096	7.908	0.591	0.015	0.000	22.008
* $0.0300/KWH	397.099	15.326	2.668	0.079	7.921	0.592	0.015	0.000	22.033
$0.0375/KWH	398.307	15.372	2.287	0.055	7.929	0.594	0.015	0.000	22.047
$0.0450/KWH	399.515	15.419	1.906	0.030	7.936	0.595	0.015	0.000	22.058
$0.0600/KWH	399.515	15.419	1.906	0.039	7.936	0.595	0.015	0.000	22.067
Water Cost									
* $0.000/1000 gal.	397.099	15.376	2.668	0.079	7.921	0.592	0.015	0.000	22.033
$0.100/1000 gal.	397.099	15.326	2.668	0.079	7.921	0.592	0.015	0.054	22.087
$0.500/1000 gal.	397.099	15.326	2.668	0.079	7.921	0.592	0.015	0.269	22.302
$1.000/1000 gal.	397.099	15.326	2.668	0.079	7.921	0.592	0.015	0.529	22.472
Water Treatment									
$0.05/1000 gal.	397.099	15.326	2.668	0.079	7.921	0.592	0.008	0.000	22.025
* $0.10/1000 gal.	397.099	15.326	2.668	0.079	7.921	0.592	0.015	0.000	22.033
$0.25/1000 gal.	397.099	15.326	2.868	0.079	7.921	0.592	0.038	0.000	22.056
$0.50/1000 gal.	397.099	15.326	2.668	0.079	7.921	0.592	0.077	0.000	22.094

Table 5.8 Cost Sensitivity for Mechanical Draft Wet Cooling Towers - 1200 MW Nuclear Plant

	Plant Construction ($\$10^6$)	Cooling System ($\$10^6$)	Replacement Capacity ($\$10^6$)	Replacement Energy ($\frac{Mills}{KWH}$)	Fuel Cost ($\frac{Mills}{KWH}$)	Maintenance ($\frac{Mills}{KWH}$)	Water Treatment ($\frac{Mills}{KWH}$)	Water Cost ($\frac{Mills}{KWH}$)	Total Production Cost ($\frac{Mills}{KWH}$)
* **Base Case:**	703.493	27.520	5.218	0.064	4.900	0.700	0.020	0.000	21.560
	15.169 $\frac{Mills}{KWH}$	0.593 $\frac{Mills}{KWH}$	0.113 $\frac{Mills}{KWH}$						
Sensitivity:									
Plant Cost									
\$450/KW	527.619	27.520	5.218	0.064	4.900	0.533	0.020	0.000	17.600
* \$600/KW	703.493	27.520	5.218	0.064	4.900	0.700	0.020	0.000	21.560
\$900/KW	1051.964	27.435	5.797	0.114	4.892	0.699	0.020	0.000	29.458
Fuel Cost									
\$0.0012/MWH	703.493	27.520	5.218	0.064	3.675	0.700	0.020	0.000	20.335
* \$0.0016/MWH	703.493	27.520	5.218	0.064	4.900	0.700	0.020	0.000	21.560
\$0.0024/KWH	701.309	27.435	5.797	0.114	7.368	0.699	0.020	0.000	21.040
\$0.0032/KWH	701.309	27.435	5.797	0.114	9.814	0.699	0.020	0.000	26.486
Fixed Charge Rate									
15%	703.493	27.520	5.218	0.064	4.900	0.700	0.020	0.000	19.692
* 17%	703.493	27.520	5.218	0.064	4.900	0.700	0.020	0.000	21.560
20%	701.309	27.435	5.797	0.114	4.892	0.699	0.020	0.000	24.358
Capacity Factor									
0.50	703.309	27.435	5.797	0.114	4.892	0.699	0.020	0.000	29.483
* 0.75	703.493	27.520	5.218	0.064	4.900	0.700	0.020	0.000	21.560
Cooling System Multiplier									
0.75	703.493	24.722	5.218	0.064	4.900	0.699	0.020	0.000	21.497
* 1.00	703.493	27.520	5.218	0.064	4.900	0.700	0.020	0.000	21.560
1.50	703.493	33.117	5.218	0.064	4.900	0.702	0.020	0.000	21.686
Replac. Capac.									
\$120/KWH	703.493	27.520	3.913	0.064	4.900	0.698	0.020	0.000	21.530
* \$160/KWH	703.493	27.520	5.218	0.064	4.900	0.700	0.020	0.000	21.560
\$240/KWH	703.493	27.520	7.827	0.064	4.900	0.702	0.020	0.000	21.618
Replac. Energy									
\$0.0225/KWH	701.309	27.435	5.797	0.085	4.892	0.699	0.020	0.000	21.535
* \$0.0300/KWH	703.493	27.520	5.218	0.064	4.900	0.700	0.020	0.000	21.560
\$0.0375/KWH	703.493	27.520	5.218	0.080	4.900	0.700	0.020	0.000	21.576
\$0.0450/KWH	703.493	27.520	5.218	0.096	4.900	0.700	0.020	0.000	21.592
\$0.0600/KWH	705.676	27.606	4.638	0.078	4.904	0.702	0.020	0.000	21.616
Water Cost									
* \$0.0/1000 gal.	703.493	27.520	5.218	0.064	4.900	0.700	0.020	0.000	21.560
\$0.1/1000 gal.	703.493	27.520	5.218	0.064	4.900	0.700	0.020	0.072	21.632
\$0.5/1000 gal.	703.493	27.520	5.218	0.064	4.900	0.700	0.020	0.359	21.919
\$1.0/1000 gal.	703.493	27.520	5.218	0.064	4.900	0.700	0.020	0.718	22.278
Water Treatment									
\$0.05/1000 gal.	703.493	27.520	5.218	0.064	4.900	0.700	0.010	0.000	21.550
* \$0.10/1000 gal.	703.493	27.520	5.218	0.064	4.900	0.700	0.020	0.000	21.560
\$0.25/1000 gal.	703.493	27.520	5.218	0.064	4.900	0.700	0.050	0.000	21.590
\$0.50/1000 gal.	703.493	27.520	5.218	0.064	4.900	0.700	0.100	0.000	21.640

Table 5.9 Cost Sensitivity for Mechanical Draft Wet Cooling Towers - 800 MW Fossil Plant

	Plant Construction ($\$10^6$)	Cooling System ($\$10^6$)	Replacement Capacity ($\$10^6$)	Replacement Energy ($\frac{Mills}{KWH}$)	Fuel Cost ($\frac{Mills}{KWH}$)	Maintenance ($\frac{Mills}{KWH}$)	Water Treatment ($\frac{Mills}{KWH}$)	Water Cost ($\frac{Mills}{KWH}$)	Total Production Cost ($\frac{Mills}{KWH}$)
* **Base Case:**	398.668	13.843	2.068	0.075	7.956	0.592	0.014	0.000	22.045
	12.895 Mills/KWH	0.448 Mills/KWH	0.067 Mills/KWH						
Sensitivity:									
Plant Cost									
$375/KW	300.404	13.908	1.477	0.023	7.969	0.451	0.014	0.000	18.670
* $500/KW	398.668	13.843	2.068	0.075	7.956	0.592	0.014	0.000	22.045
$750/KW	595.196	13.778	2.659	0.165	7.933	0.873	0.014	0.000	28.768
Fuel Cost									
$0.0023/KWH	399.603	13.876	1.773	0.052	5.912	0.591	0.014	0.000	19.994
* $0.0031/KWH	398.668	13.843	2.068	0.075	7.956	0.592	0.014	0.000	22.045
$0.0046/KWH	398.668	13.843	2.068	0.075	11.212	0.592	0.014	0.000	25.901
$0.0061/KWH	397.733	13.811	2.364	0.114	15.648	0.591	0.014	0.000	29.754
Fixed Charge Rate									
15%	399.603	13.876	1.773	0.045	7.963	0.592	0.014	0.000	20.466
* 17%	398.668	13.843	2.068	0.075	7.956	0.592	0.014	0.000	22.045
20%	398.668	13.843	2.068	0.075	7.956	0.592	0.014	0.000	24.412
Capacity Factor									
0.50	397.733	13.811	2.364	0.114	7.946	0.591	0.014	0.000	28.746
* 0.75	398.668	13.843	2.068	0.075	7.956	0.592	0.014	0.000	22.045
Cooling System Multiplier									
0.75	398.668	12.654	2.068	0.075	7.956	0.591	0.014	0.000	22.005
* 1.00	398.668	13.843	2.068	0.075	7.956	0.592	0.014	0.000	22.045
1.50	398.668	16.222	2.068	0.075	7.956	0.594	0.014	0.000	22.126
Replac. Capac.									
$120/KW	398.668	13.843	1.551	0.075	7.956	0.591	0.014	0.000	22.028
* $160/KW	398.668	13.843	2.068	0.075	7.956	0.592	0.014	0.000	22.045
$240/KW	399.603	13.876	2.659	0.045	7.963	0.593	0.014	0.000	22.076
Replac. Energy									
$0.0225/KWH	397.733	13.811	2.364	0.085	7.946	0.590	0.014	0.000	22.023
* $0.0300/KWH	398.668	13.843	2.068	0.075	7.956	0.592	0.014	0.000	22.045
$0.0375/KWH	399.603	13.876	1.773	0.056	7.963	0.592	0.014	0.000	22.057
$0.0450/KWH	400.539	13.908	1.477	0.034	7.969	0.593	0.014	0.000	22.063
$0.0500/KWH	401.474	13.941	1.182	0.018	7.973	0.594	0.014	0.000	22.074
Water Cost									
* $0.0/1000 gal.	398.668	13.843	2.068	0.075	7.956	0.592	0.014	0.000	22.045
$0.1/1000 gal.	398.668	13.843	2.068	0.075	7.956	0.592	0.014	0.050	22.095
$0.5/1000 gal.	398.668	13.843	2.068	0.075	7.956	0.592	0.014	0.249	22.294
$1.0/1000 gal.	398.668	13.843	2.068	0.075	7.956	0.592	0.014	0.497	22.542
Water Treatment									
$0.05/1000 gal.	398.668	13.843	2.068	0.075	7.956	0.592	0.007	0.000	22.038
* $0.10/1000 gal.	398.668	13.843	2.068	0.075	7.956	0.592	0.014	0.000	22.045
$0.25/1000 gal.	398.668	13.843	2.068	0.075	7.956	0.592	0.035	0.000	22.066
$0/50/1000 gal.	398.668	13.843	2.068	0.075	7.956	0.592	0.070	0.000	22.101

Chapter VI

DRY COOLING TOWERS

6.1 Introduction

Dry cooling towers are similar to evaporative towers in many respects, with the major exception that the cooling water never comes in direct contact with the circulating water. Rather, the condenser water passes through heat exchanges (such as finned tubes) where heat is transferred to the ambient air by conduction. Although dry cooling towers offer many advantages environmentally, they have not seen much use in the U.S. due to the very high capital costs and poor performance during extreme meteorologic conditions resulting in high operating and replacement expenses. Furthermore, efficient use of dry towers will require use of special high backpressure turbines discussed in Chapter II. These are presently not available for nuclear plants.

To make dry cooling towers more practical, it is important to optimize the detailed design parameters for the turbine, condenser, piping and dry cooling tower. The optimization is not limited to the use of predesigned dry cooling equipment. The characteristic of the present MIT computer code on dry cooling (Choi and Glicksman, 1978) is to perform a totally computerized optimization. The design parameters used for the optimization will be discussed in Subsection 6.4.1. This study will optimize and then compare dry cooling towers using the two turbine types discussed in Subsection 2.2.1: (a) conventional and (b) high backpressure turbines.

Many of the same distinctions between types of wet towers are found in dry towers (e.g., mechanical versus natural draft, forced versus induced draft, etc.). In addition to these, dry cooling systems can be direct — turbine exhaust steam condenses directly in the tower coils — or indirect — condensers are utilized to transfer heat from the exhaust steam to the cooling water. The dry cooling system considered in this study is taken to be an indirect type with the draft mechanically induced. The cooling process is shown in Figure 6.1. Heat is rejected from the cooling water by the heat exchanger to the cooling air. Only a metal finned tube heat exchanger was considered in this study and admiralty was chosen as the optimum material for the finned tubes.

6.2 Environmental Factors

Since there are no thermal discharges associated with dry cooling towers, they are an effective technological control of thermal pollution. However, they would not be selected for this purpose only. The high capital and operating costs of dry cooling, discussed previously, by far exceed those of the wet towers or ponds which are also effective in controlling thermal pollution. Yet dry towers offer other advantages which may serve to offset the higher power production cost.

Since the circulating water in the dry tower does not come into contact with the atmosphere there is no evaporative water loss. This is important in terms of overall water conservation and added flexibility for siting in arid regions. Also, problems of icing, fogging and visible vapor plumes (of particular concern in urban areas) are eliminated. There is no blowdown associated with dry cooling and thus no water treatment is required. Except for relatively large land requirements (compared to wet towers), and possible aesthetic considerations, the environmental behavior

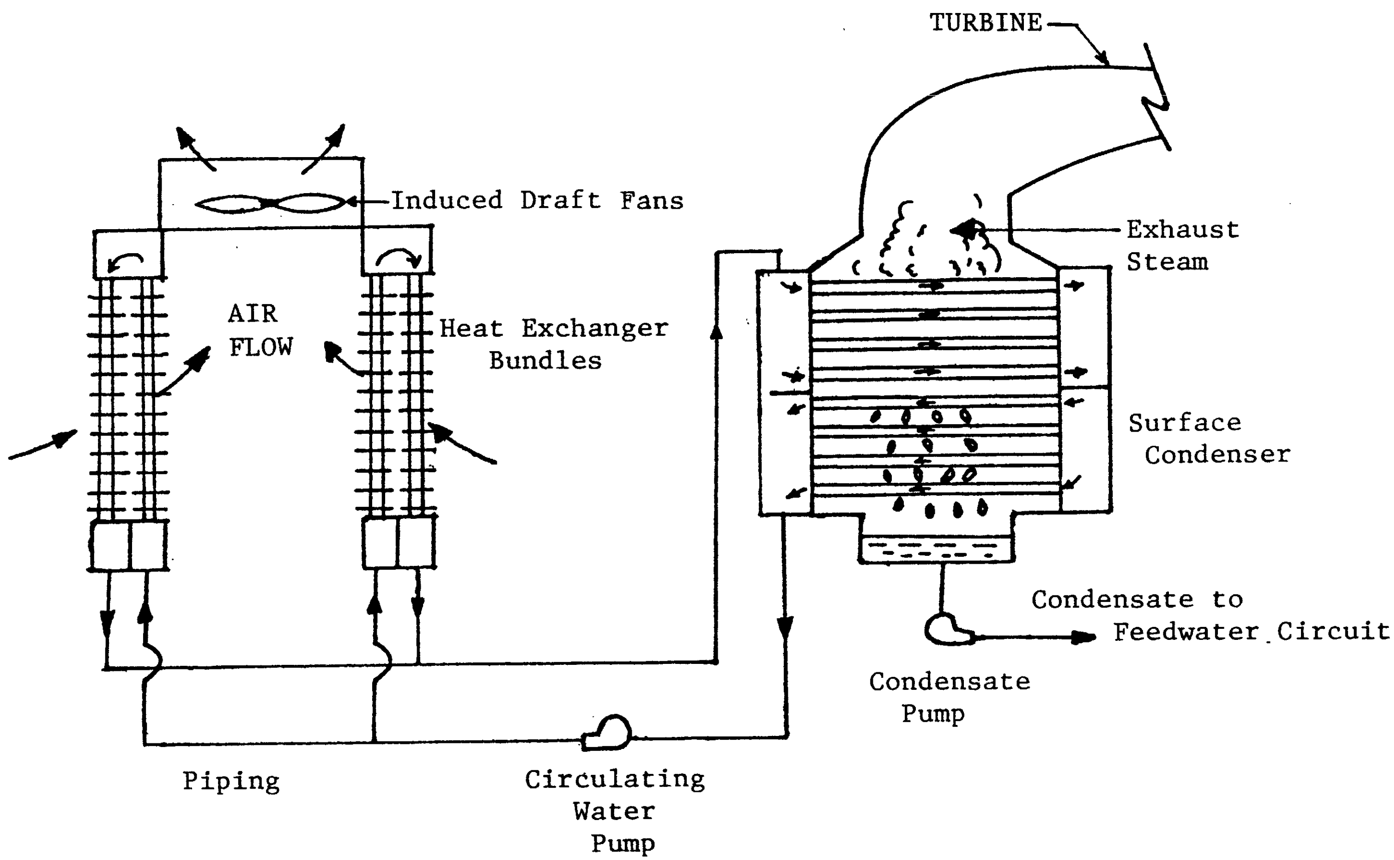


Figure 6.1 Indirect Type of Dry Cooling Tower with Mechanical Induced Draft

of dry towers improves their overall usefulness and also allows more flexibility for power plant siting which makes possible savings in electrical transmission and fuel transportation costs.

The concept of wet/dry cooling towers deserves mention at this time. This type of cooling provides for control of environmental factors related to fogging, plume abatement and water conservation usually associated with wet cooling towers. Wet/dry towers operate by use of the heat exchangers of the dry section to allow for the removal of part of the heat load via sensible heat transfer while the remaining heat load is removed via latent heat transfer in the evaporative section. This increase in the ratio of sensible to latent heat transfer reduces the relative humidity or moisture content of the tower effluent, thus reducing or eliminating the formation of visible plumes.

The amount of heat transfer by evaporation can be controlled by adjusting this ratio to some permissible level of water consumption. However, a reduction in water loss while maintaining plant performance entails higher costs in design. For example, in their study of wet/dry towers, Choi and Glicksman (1978) found that a reduction in make-up water requirements from 30% of that of a fully wet tower to 15% of that of a fully wet tower involved an average additional production cost of about 0.25 mills/kwh. Becasue of the possibility of combining the characteristics of wet and dry towers in some optimal manner, it can be expected that wet/dry towers will see significant employment before single-mode dry towers.

6.3 Design and Performance

In the dry tower, the transfer of heat from the inside fluid to the air depends primarily on the temperature difference between the fluid and the air, the design and surface arrangement of the finned tubes, the velocity and character of air flow across the tubes, and the velocity and physical properties of the fluid inside the tubes.

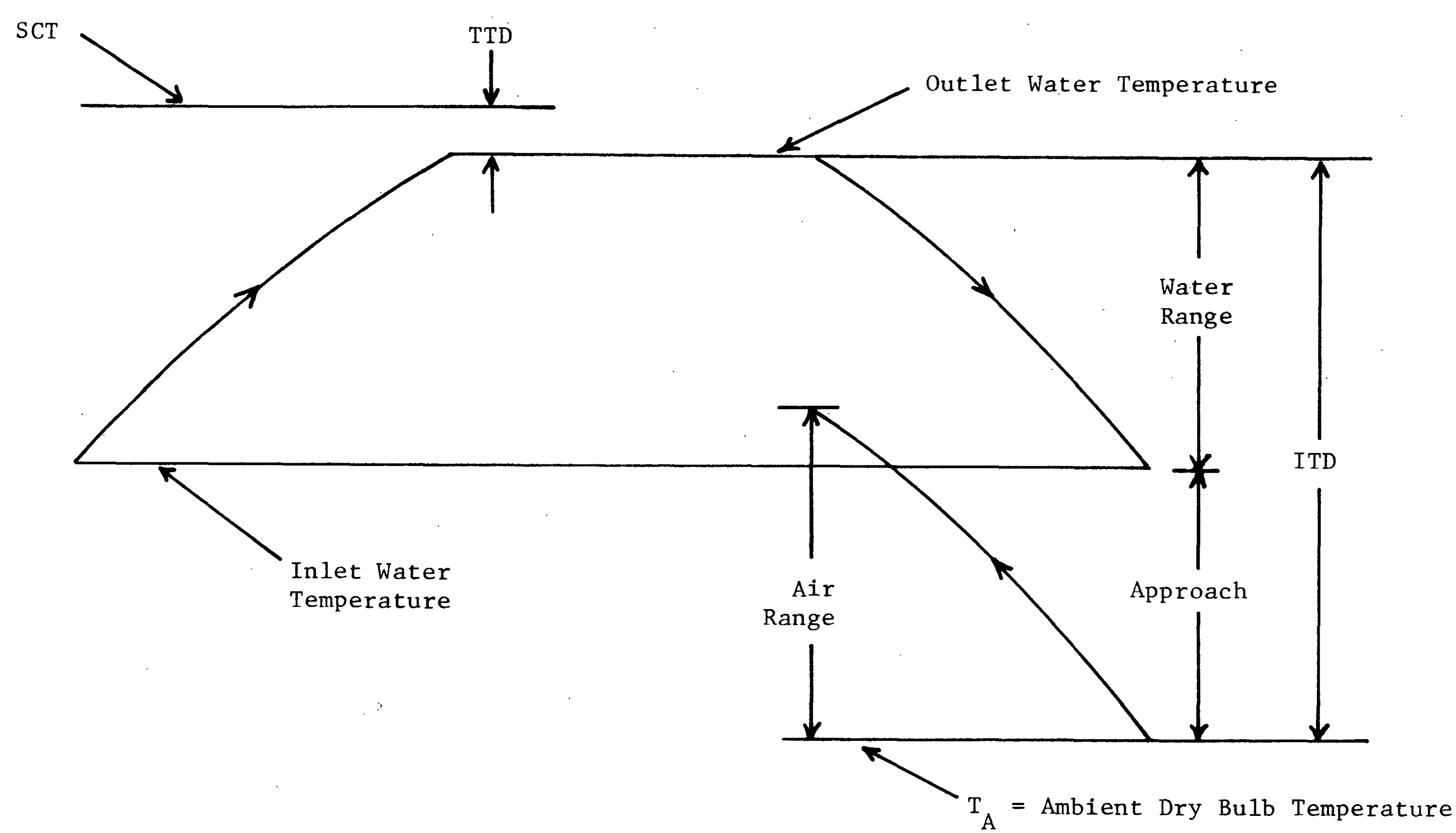
The temperature relationships in the indirect dry cooling tower are illustrated in Figure 6.2. Two variables which appear in this figure but are not usually used in figures for wet towers are the air range and the initial temperature difference (ITD). The former is used to describe the tower heat exchange by sensible heat which balances the heat rejected in the condenser. The ITD is the difference between the temperature of the water entering the tower and the inlet dry bulb temperature of the air entering the heat exchanger (T_A), and is the sum of the (water) range ΔT_o and the approach (ΔT_{APP}) used in the other systems. As with the other systems, in terms of plant performance, the turbine backpressure increases as the environmental temperature (T_A, in this case) increases, resulting in higher heat rates and poorer thermal efficiency.

6.4 Method of Optimization

6.4.1 Optimization Procedure

There are a number of design parameters relevent to the optimization of the dry cooling system. The initial temperature difference (ITD) described in Section 6.3 is the major variable and is an inverse measure of the cooling system (dry tower) size. Thus, a high ITD implies a small

Figure 6.2 Temperature Relationships in Indirect Type of Dry Cooling System



dry tower which, in turn, results in a relatively low thermodynamic efficiency. The logic in the dry tower design is somewhat different from that of the other systems in that the ITD is an independent design variable, i.e. the performance of the power plant is fixed for a given ITD. The heat exchanger is then sized to meet this design performance. This is contrasted to the approach in the other three cooling systems where the cooling system size is the design parameter and the performance is subsequently calculated.

The design dry bulb temperature (T_D) is a secondary design parameter which refers to the ambient dry bulb temperature at which the power plant is sized to produce the given net electrical output. At actual ambient temperatures above (below) the T_D, the turbine heat rate increases (decreases) and net power output decreases (increases) relative to the target demand.

For a given design temperature, the ITD is a function of: the range (ΔT_o), the water to air heat capacity ratio, the heat exchanger air-side frontal area, and the width-to-length ratio of the heat exchanger. The last two variables determine the dimension of the heat exchanger while the water to air heat ratio determines the air loading on the heat exchanger.

Thus, the optimization of the dry cooling system involves a selection of an optimum from a set of optima. The basic procedure is as follows:

(a) Select an ambient design temperature (T_D).

(b) Select a design ITD.

(c) Find the combination of water range (ΔT_o), capacity ratio, heat exchanger frontal area, width-to-length ratio, which gives the minimum power production cost.

(d) Using the optimized cooling system, determine the power plant performance over the year using the dry bulb temperature distribution shown in Figure 2.7; the costs of replacement capacity and replacement energy are thus found. Obtain the power production cost.

(e) Repeat steps (b) through (d) for a new design ITD.

(f) Repeat steps (a) through (e) for a new design temperature.

(g) Select the design temperature and design ITD which gives the minimum power production cost for the actual plant operation by comparing the results at each ambient design temperature.

6.4.2 Dry Cooling Costs

The costs of the mechanical draft dry tower were evaluated using the optimization procedure described above for both the high backpressure and the conventional turbines. For both the fossil and nuclear plants the dry tower capital cost using the conventional turbine was found to be over 60% higher than the cost using the high backpressure turbine which, on the other hand, are several times larger than the capital cost for the once-through system. The major cost of the dry tower is the heat exchanger whose cost is arrived at by considering a number of components (tubing bundles, spacers, headers, framing, etc.). No simple unit cost estimation can be applied to the heat exchanger since there is no marketed standard design; however, a full detailed breakdown of heat exchanger components can be found in Choi and Glicksman (1978). As a summary, the heat exchanger is found to comprise about 35% of the capital cost. The other components of the dry tower cost are the condenser (10%), piping (15%), pumps (5%), fans and equipment (15%). The remainder of the cost for the dry tower is in the tower structure and foundation and certain indirect capital costs. The sensitivities studies examine the cost of the dry tower using multipliers of 0.75 and 1.5 of all tower capital costs.

6.5 Results

In evaluating the optimum dry cooling systems we have considered two primary design variables and two turbine types for both the nuclear and fossil plants. The design variables (initial temperature difference (ITD) and design temperature (TD)) are plotted against the power production cost in Figures 6.3 and 6.4 for the conventional and high back pressure turbines, respectively for the nuclear unit. Comparison between turbine types shows that minimum cost for both nuclear and fossil (not shown) plants is associated with the high back pressure turbine. For both plants this optimum corresponds to a TD of 50°F and an ITD of 65°F. Since production cost appear to be more sensitive to ITD, as evidence in Figure 6.3 and 6.4, the remaining result in this section will consider a constant design temperature (50°F) and a single turbine type (high back pressure) thus allowing only ITD to vary.

Tables 6.1 and 6.2 show the total power production cost versus design ITD for the nuclear and fossil plants, respectively. Recalling from Section 6.4 that ITD is inversely proportional to the tower size we can identify some basic trends in the tables. First, the cooling system cost is considerably larger than that for other systems but it

Figure 6.3 Power Production Cost Versus Design ITD
Dry Cooling Tower
1200 MW Nuclear Plant
Conventional Turbine

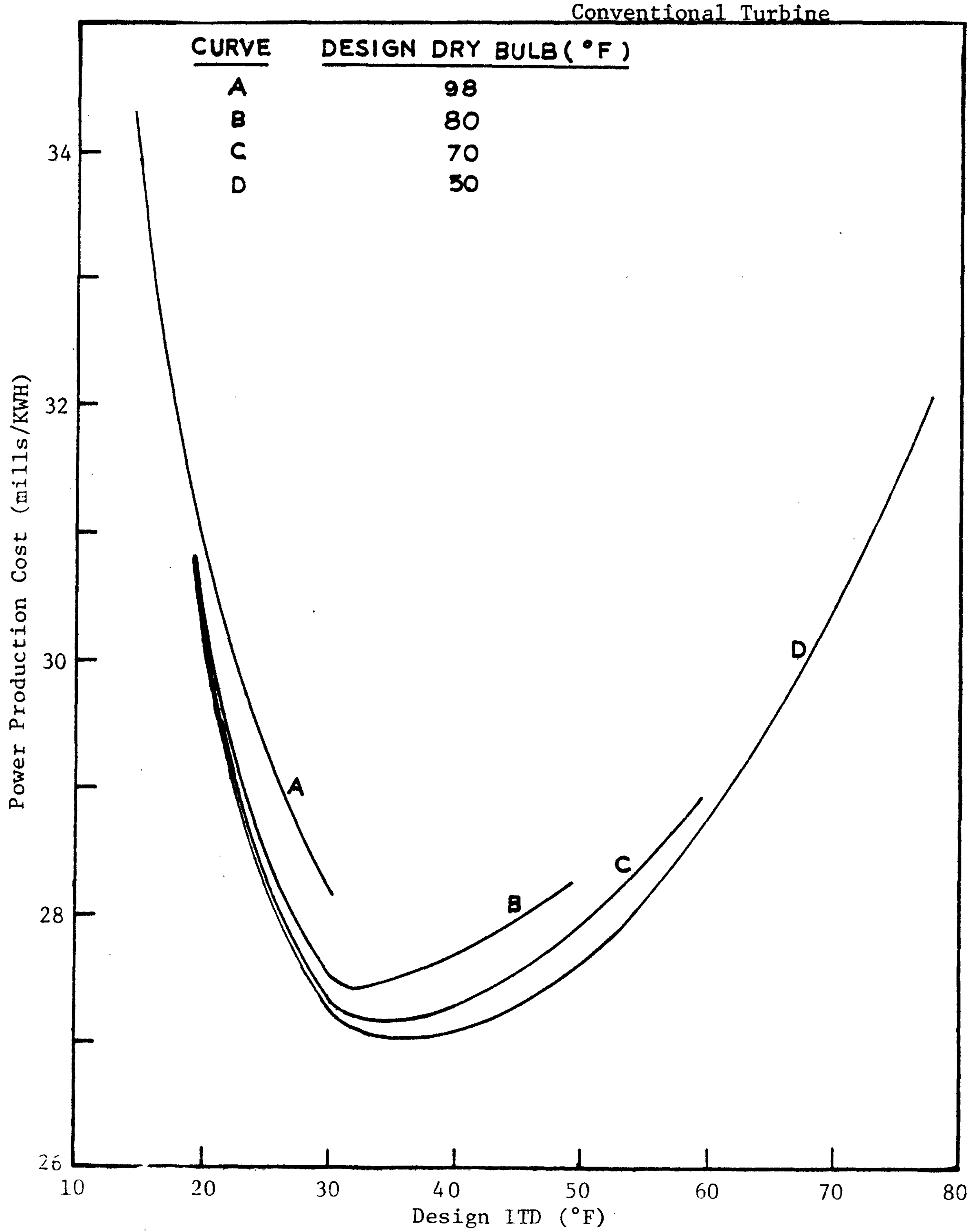


Figure 6.4 Power Production Cost Versus Design ITD
Dry Cooling Tower
1200 MW Nuclear Plant
High Back Pressure Turbine

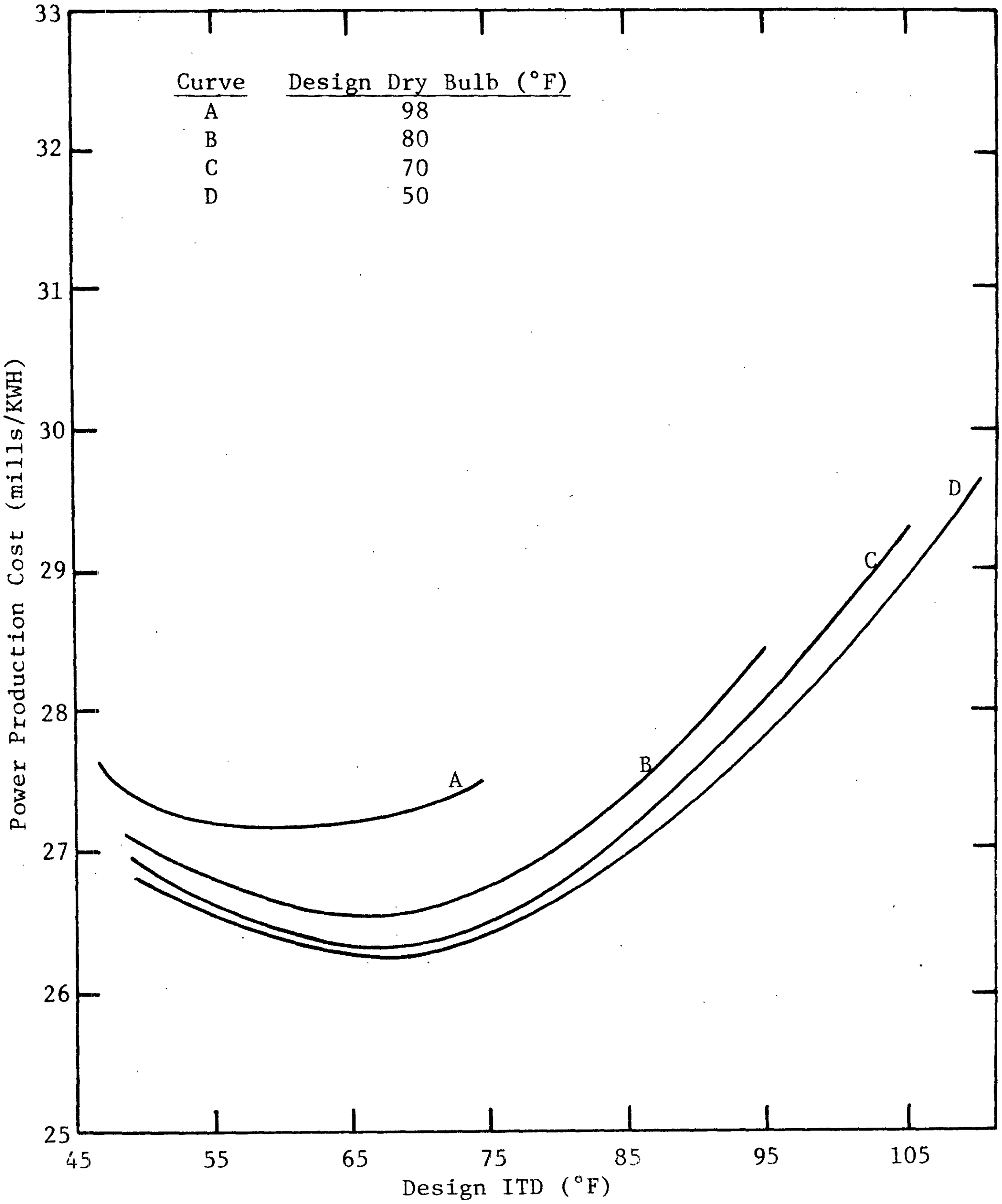


Table 6.1 Power Production Cost Versus Design ITD-1200 MW Nuclear Plant

Cost Component (mills/KWH)	Design ITD (°F) 55	65	75	85
Plant Construction	16.895	16.814	16.805	16.891
Cooling System	2.802	2.341	2.141	1.883
Replacement Capacity	0.127	0.270	0.535	1.026
Fuel	5.400	0.268	5.365	5.372
Replacement Energy	0.054	5.378	0.324	0.652
Maintenance	1.170	1.143	1.146	1.165
Total Power Production Cost	26.448	26.214	26.316	26.989

Table 6.2 Power Production Cost Versus Design ITD-800 MW Fossil Plant

Cost Component (mills/KWH)	Design ITD (°F) 55	65	75	85
Plant Construction	13.239	13.240	3.159	13.180
Cooling System	1.867	1.555	1.506	1.289
Replacement Capacity	0.116	0.224	0.627	1.073
Fuel	8.396	8.396	8.337	8.315
Replacement Energy	0.054	0.133	0.279	0.610
Maintenance	0.895	0.884	0.890	0.914
Total Power Production Cost	24.567	24.432	24.798	25.381

decreases rapidly as the ITD is increased. Replacement energy and capacity costs are also relatively large and increase rapidly with increasing ITD reflecting the high inefficiencies of the smaller dry towers. The combination is a well-defined optimum ITD.

Since sensitivity to the economic factors discussed in Chapter II reveals similar behavior to those discussed with the other systems only the sensitivity to the cooling system multiplier will be shown graphically in this part. Further graphs can be found in Choi and Glicksman (1978) and all results are summarized in Tables 6.3 and 6.4. Figure 6.5 shows that an increase in the dry tower cost causes a shift in the optimum to a smaller tower size. This shift is substantial considering the high replacement costs associated with a high ITD.

Tables 6.3 and 6.4 summarize the dry tower design and sensitivity study for the nuclear and fossil plants, respectively. The format is the same as for the other system except that no sensitivity was run for water and blowdown treatment costs which are negligible for the dry cooling system. As before, these tables summarize costs for the optimum dry tower design (initial temperature difference, design temperature and turbine type) obtained from the base case analysis.

Figure 6.5 Sensitivity Study for Cooling System Multiplier
Dry Cooling Tower
1200 MW Nuclear Plant

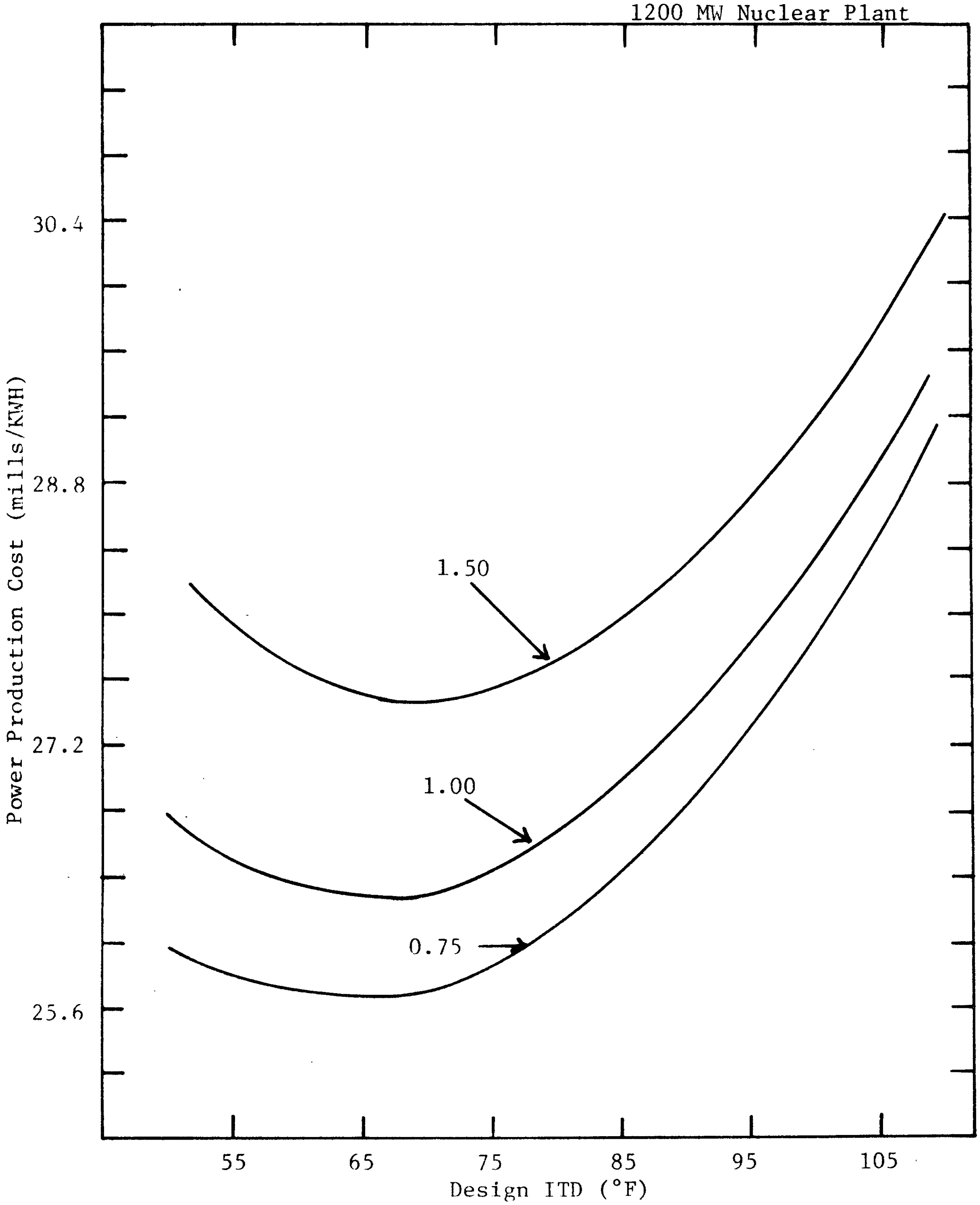


Table 6.3 Cost Sensitivity Study for Dry Cooling Towers - 1200 MW Nuclear Plant

	Plant Construction ($\$10^6$)	Cooling System ($\$10^6$)	Replacement Energy ($\$10^6$)	Replacement Energy ($\frac{Mills}{KWH}$)	Fuel Cost ($\frac{Mills}{KWH}$)	Maintenance ($\frac{Mills}{KWH}$)	Water Treatment ($\frac{Mills}{KWH}$)	Water Cost ($\frac{Mills}{KWH}$)	Total Production Cost ($\frac{Mills}{KWH}$)
* **Base Case:**	779.790	108.590	12.540	0.268	5.378	1.143			26.214
	16.814 $\frac{Mills}{KWH}$	2.341 $\frac{Mills}{KWH}$	0.270 $\frac{Mills}{KWH}$						
Sensitivity:									
Plant Cost									
$450/KW	582.952	108.590	12.540	0.268	5.378	0.898			21.726
* $600/KW	779.790	108.590	12.540	0.268	5.378	1.143			26.214
$900/KW	1,165.951	108.590	12.540	0.268	5.378	1.638			35.037
Fuel Cost									
$0.0012/KWH	779.790	108.590	12.540	0.268	4.020	1.143			24.857
* $0.0016/KWH	779.790	108.590	12.540	0.268	5.378	1.143			26.214
$0.0024/KWH	779.790	108.590	12.540	0.268	8.040	1.143			28.877
$0.0032/KWH	779.790	108.590	12.540	0.268	10.720	1.143			31.557
Fixed Charge Rate									
15%	691.149	96.370	14.887	0.268	5.378	1.018			23.966
* 17%	779.790	108.590	12.540	0.268	5.378	1.143			26.214
20%	907.124	126.561	19.293	0.268	5.378	1.813			30.164
Capacity Factor									
0.50	1,165.951	162.921	24.951	0.268	5.378	1.718			36.556
* 0.75	779.790	108.590	12.540	0.268	5.378	1.143			26.214
Cooling System Multiplier									
0.75	774.116	85.425	13.125	0.268	5.340	1.133			25.558
* 1.00	779.790	108.590	12.540	0.268	5.378	1.143			26.214
1.50	779.790	159.860	17.067	0.380	5.320	1.209			27.538
Replac. Capac.									
$120/KW	779.790	106.434	12.707	0.380	5.320	1.135			26.218
* $160/KW	779.790	108.590	12.540	0.268	5.378	1.143			26.214
$240/KW	779.635	116.498	19.710	0.200	5.350	1.162			26.460
Replac. Energy									
$0.0225/KWH	779.790	108.590	12.540	0.190	5.378	1.143			26.137
* $0.0300/KWH	779.790	108.590	12.540	0.268	5.378	1.143			26.214
$0.0375/KWH	779.790	111.675	15.768	0.360	5.378	1.146			26.446
$0.0450/KWH	779.790	111.675	15.304	0.400	5.378	1.156			26.486
$0.0600/KWH	779.635	113.901	13.125	0.440	5.380	1.150			26.520

Table 6.4 Cost Sensitivity Study for Dry Cooling Towers - 800 MW Fossil Plant

	Plant Construction ($\$10^6$)	Cooling System ($\$10^6$)	Replacement Capacity ($\$10^6$)	Replacement Energy ($\frac{Mills}{KWH}$)	Fuel Cost ($\frac{Mills}{KWH}$)	Maintenance ($\frac{Mills}{KWH}$)	Water Treatment ($\frac{Mills}{KWH}$)	Water Cost ($\frac{Mills}{KWH}$)	Total Production Cost ($\frac{Mills}{KWH}$)
* Base Case:	409.360	48.080	6.930	0.133	8.396	0.884			24.432
	13.240 $\frac{Mills}{KWH}$	1.555 $\frac{Mills}{KWH}$	0.224 $\frac{Mills}{KWH}$						
Sensitivity:									
Plant Cost									
$375/KW	306.023	48.080	6.930	0.133	8.396	0.689			20.895
* $500/KW	409.360	48.080	6.930	0.133	8.396	0.884			24.432
$750/KW	612.324	48.080	6.930	0.133	8.396	1.272			31.385
Fuel Cost									
$0.0023/KWH	409.360	48.080	6.930	0.133	6.320	0.884			22.356
* $0.0031/KWH	409.360	48.080	6.930	0.133	8.396	0.884			24.432
$0.0046/KWH	409.360	48.080	6.930	0.133	12.650	0.884			28.686
$0.0061/KWH	409.360	48.080	6.930	0.133	16.860	0.884			32.896
Fixed Charge Rate									
15%	362.942	44.831	4.978	0.133	8.396	0.079			21.958
* 17%	409.360	48.080	6.930	0.133	8.396	0.884			24.432
20%	476.255	58.991	6.431	0.133	8.396	1.030			27.079
Capacity Factor									
0.50	612.324	75.625	8.162	0.133	8.396	1.325			32.369
* 0.75	409.360	48.080	6.930	0.133	8.396	0.884			24.432
Cooling System Multiplier									
0.75	408.793	39.420	4.669	0.130	8.410	0.862			24.050
* 1.00	409.360	48.080	6.930	0.133	8.396	0.884			24.432
1.50	409.360	75.346	5.843	0.150	8.370	0.931			25.280
Replac. Capac.									
$120/KW	409.360	50.210	4.669	0.150	8.370	0.882			24.417
* $160/KW	409.360	48.080	6.930	0.133	8.396	0.884			24.432
$240/KW	408.793	52.560	6.122	0.120	8.390	0.890			24.520
Replac. Energy									
$0.0225/KWH	409.360	48.080	6.930	0.110	8.396	0.884			24.410
* $0.0300/KWH	409.360	48.080	6.930	0.133	8.396	0.884			24.432
$0.0375/KWH	409.360	48.080	6.930	0.170	8.396	0.884			24.470
$0.0450/KWH	408.484	51.385	5.843	0.200	8.410	0.886			24.560
$0.0600/KWH	408.793	52.560	6.122	0.240	8.390	0.890			24.640

Chapter VII

COMPARISONS AND CONCLUSIONS

7.1 Summary of Performance and Economics

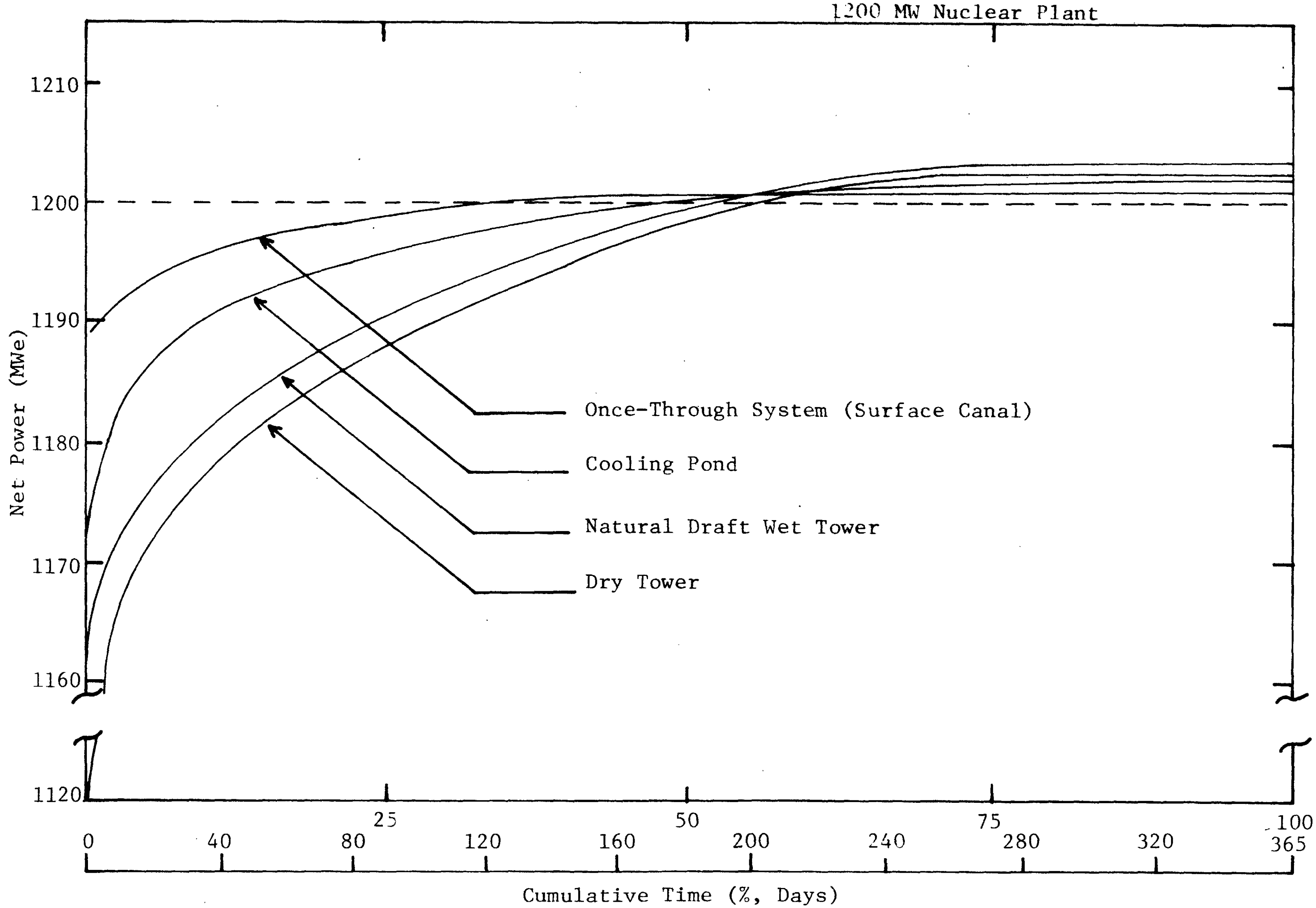
This thesis has examined the effect of cooling system choice on the issues of cost, energy and water consumption and environmental impact for a power plant at a single hypothetical river site. Once-through cooling systems (using both surface discharge and submerged multi-port diffusers), cooling ponds, mechanical and natural draft evaporative towers and mechanical draft dry towers were considered for both a 1200 MWe nuclear plant and an 800 MWe coal plant. A set of models was developed to optimize the components of each cooling system based on the local meteorological and hydrological conditions at the site in accordance with a fixed demand, scalable plant concept. This concept allows one to compare the costs of producing the same net power from each plant/cooling system.

7.1.1 System Performance

The fixed demand, scalable source approach is illustrated in Figure 7.1 for the optimal cooling systems on a graph of power versus cumulative time for the 1200 MWe nuclear plant. These curves represent the appropriate adjustment of base-load power produced relative to the target demand when the various cooling systems are optimally scaled. Remember from the discussion in Section 2.3 that, associated with each cooling system, there is sufficient replacement capability to meet the target demand during periods of high environmental temperatures and that the plant is given a fuel cost credit for excess power which could be produced during

Figure 7.1 Power vs Cumulative Time

Optimal Cooling Systems
1200 MW Nuclear Plant



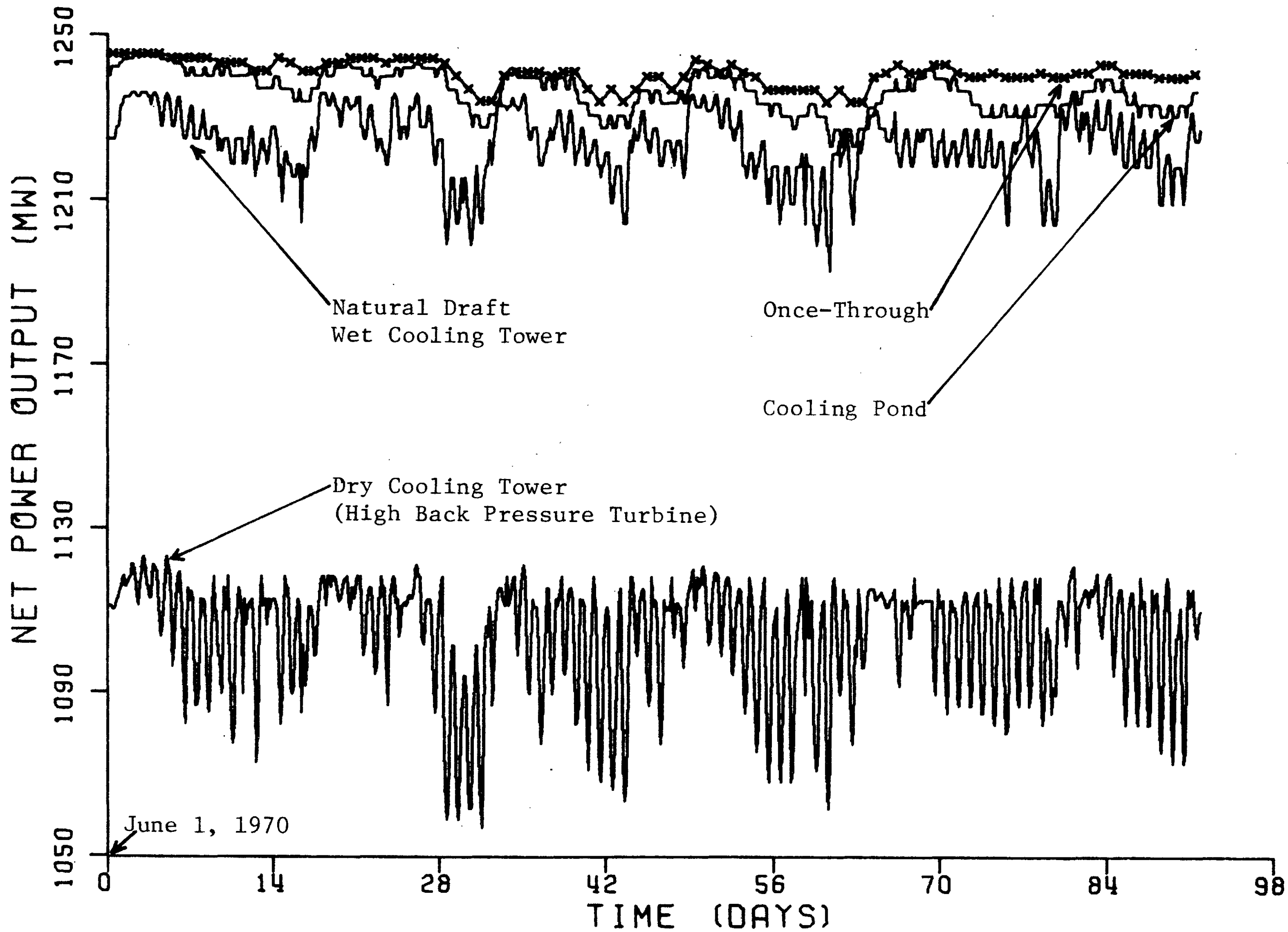
periods of low environmental temperatures. Note that the range of net power produced from the base-load plant, and hence the amount of replacement capacity which is needed, reflects the range in the appropriate environmental temperature which governs the cooling system performance (See Figure 2.7). Thus the dry tower, which responds to dry bulb temperature, undergoes wider variation than the once-through system which responds to river temperature.

The points on Figure 7.1 at which the net base-load output crosses the demand line represent the total amount of time that replacement power is utilized. This time is proportional to the difference in capital cost (plant plus cooling system) between base-load and replacement power and inversely proportional to the difference in operating (energy) cost between peaking and base-load operation. The once-through system, having the lowest capital cost is found to require replacement power for the shortest time while the dry tower is at the other extreme. It should be noted that predicted times are only approximate since they are very sensitive to small adjustments in the scaling of the base-load power. This is due to the relatively constant net power production at all but the higher environmental temperatures. Small errors in these predicted times, however, result in negligible changes in the computed replacement capacity or energy which are used to evaluate the systems.

7.1.2 Transient Simulation

For purposes of comparison with the cumulative plots of Figure 7.1, Figure 7.2 illustrates the transient output for the same power plant using a once-through system (surface canal), a cooling pond, a natural draft

Figure 7.2 Cooling System Transient Simulation (summer 1970)



evaporative tower and a mechanical draft dry tower (with high back pressure turbine). As with the cumulative plots, the optimal system has been chosen based on the results of the previous four chapters, but in this case the output has not been scaled. This allows one to see the differences in net output which would be produced from a fixed plant and steam supply. The simulation is based on the three months of June - August 1970 using three hourly meteorological data for the closed systems and on daily hydrological data (river temperatures) for the open cycle system. For the cooling pond, the transient model described in Section 4.3 (rather then the quasi-steady model) was used.

The figure shows that the greatest net power is produced by the once-through system followed by the cooling pond, the wet tower and the dry tower. Furthermore, the ordering of the systems in terms of consistency of power produced (i.e., absense of fluctuation) is similar. While the cumulative plots illustrate primarily seasonal variations, the transient plots highlight fluctuations on diurnal and synoptic (order of days) scales. On these scales, the once-through system and the cooling ponds show only small fluctuation due to the thermal inertia of the water body while the output from the wet, and especially the dry, tower indicates significant variability due to fluctuations in (the wet or dry bulb) air temperature.

Fluctuation in the net power are of interest because they reduce the supply of firm power upon which a utility may rely. In particular, firm power can be sold to other utilities or transferred within the utilities service area, thereby allowing the utility to cut back on generation on its less efficient units. To the extent that these day to day fluctuations show up in the cumulative environmental temperature distributions used to evaluate the cooling systems, they have been accounted for in this study. However, the tails of the cumulative temperature distribution reflect seasonal variability (which one can anticipate) as well as day to day variability (which one cannot anticipate). No attempt has been made in this study to assign a special cost to this short term variability.

7.1.3 Economic Comparison

Tables 7.1 and 7.2 present a comparative summary of cooling system operation for the base case scenarios. Shown in each table are a once-through system with a surface canal, a cooling pond, mechanical and natural draft wet towers and a mechanical draft dry tower. In addition to these systems a once-through system using multi-port diffusers and a wet/dry cooling tower are included. For the nuclear plant a multi-port diffuser design with a length of 1500 ft and nozzle exit velocity of 10 fps was chosen since it was the most effective in meeting the temperature standards at the site (see Section 3.2). A 1000-ft diffuser with an exit velocity of 10 fps has a comparable dilution when employed with the fossil plant and thus is used for comparison in Table 7.2. A wet/dry cooling tower having a design make-up water withdrawal of 30% (of a fully wet tower) was included following Choi and Glicksman (1978) for comparison.

The costs in the tables are the same as those found in the results section for each system and are presented here for comparison. The maximum minus minimum power production represents the difference in net power output between the extreme environmental conditions encountered during a year as was shown in Figure 7.1. The water withdrawal rate and water consumption have bearing in the costing for water and as environmental effects (see Section 7.2).

Before comparing systems with respect to cost, it should be emphasized that this study has not considered electrical transmission costs between the plant and the primary location of demand or any differences in cost of fuel as a function of the cooling system under study. As one's interests move away from once-through systems and towards evaporative and dry towers, one becomes increasingly less constrained by the requirement that the site have a sufficient supply of water available for cooling on hand. Thus, both evaporative and dry towers offer the utility greater flexibility in locating the plant near either the fuel source or the primary location of demand, an economic incentive which cannot be assessed by this study.

Under our assumptions, once-through cooling with a surface canal is always preferred to alternative cooling systems when the sole objective is that of minimizing costs. Once-through cooling with diffusers is the next best alternative followed by cooling ponds, evaporative towers, wet/dry towers and dry towers. As an approximate indication of cost one can examine the rankings as various costs are varied. For example, ponds and wet towers have many similar characteristics. Using our base case economic parameters for nuclear plants, mechanical draft evaporative towers would be preferred to ponds if the cost of land purchase and preparation exceeded approximately \$10,500/acre while natural draft

Table 7.1 Cooling System Comparison - 1200 MW Nuclear Plant

	Once-through: Surface Canal	Once-through: Multiport Diffuser *	Cooling Pond	Natural Draft Wet Tower	Mechanical Draft Wet Tower	Mechanical Draft Wet/Dry Tower **	Mechanical Draft Dry Tower
Power Production Cost (mills/KWH)	20.93	21.13	21.26	21.63	21.56	24.45	26.21
Cooling System Capital Cost ($ x 10^6)	18.48	27.31	27.53	34.46	27.52	73.49	108.59
Total Fuel Cost *** (mills/KWH)	4.87	4.88	4.91	4.97	4.96	5.29	5.64
Maximum - Minimum Power Production (MW)	13.3	13.5	28.8	44.6	37.2	60.0	81.1
Water Withdrawal Rate (cfs)	1500.	1500.	41.2	38.7	33.7	10.1	$\sim$
Water Consumption (cfs)	16.8	16.8	27.7	27.8	23.5	7.1	$\sim$

* diffuser design $L = 1500$ ft., $u_o = 10$ fps

** design make-up water requirement = 30% (of fully wet tower)

*** sum of nuclear fuel cost and replacement fuel cost

Table 7.2 Cooling System Comparison - 800 MW Fossil Plant

	Once-through: Surface Canal	Once-through: Multiport Diffusers *	Cooling Pond	Natural Draft Wet Tower	Mechanical Draft Wet Tower	Mechanical Draft Wet/Dry Tower **	Mechanical Draft Dry Tower
Power Production Cost (mills/KWH)	21.62	21.76	21.84	22.03	22.05	23.81	24.43
Cooling System Capital Cost ($ x 10^6)	11.89	16.20	14.57	15.33	13.84	36.92	48.08
Total Fuel Cost *** (mills/KWH)	7.92	7.93	7.94	8.00	8.03	8.11	8.53
Maximum - Minimum Power Production (MW)	12.8	12.9	20.4	23.8	18.6	26.0	43.7
Water Withdrawal Rate (cfs)	840	840	18.9	16.1	14.8	4.4	∿
Water Consumption (cfs)	7.7	7.7	12.5	11.6	10.6	3.2	∿

* diffuser design: L = 1000 ft., u_o = 10 fps

** design make-up water requirement of 30% (of fully wet tower)

*** sum of fossil fuel cost and replacement fuel cost evaporative towers would be competitive if these costs exceeded about \$12,400/acre. For fossil plants the break even land costs would be \$10,200/acre and \$11,000/acre for natural and mechanical draft towers respectively.

The cost of installing dry towers is staggering but they may eventually be competitive with wet cooling systems in regions of the country where make-up water is in short supply, land is extremely expensive (precluding ponds) or environmental constraints on fogging or visible plume might prevail. Using our base case economic parameters for nuclear plants, dry towers will be preferred to mechanical draft evaporative and natural draft evaporative towers when plant lifetime levelized prices for make-up water exceed about \$6.50/1000 gal and \$5.25/1000 gal, respectively. However, these are factors many times greater than those considered in the sensitivity analysis and there is good reason to believe that before water becomes that expensive, utilities would install the necessary treatment facilities to recover substantial amounts of blowdown water. If all blowdown were recovered for recirculation, the levelized water prices at which dry towers would be preferred to mechanical draft wet and natural draft wet towers must exceed \$10.80/1000 gal and \$8.75/1000 gal, respectively. Because of these high costs, it is easy to see why the development of wet/dry systems is so promising!

The comparable advantages of the cooling system alternatives which have been described are applicable only under our base case assumptions. The general order of preferrability, where cost minimization is the sole objective, places once-through cooling as the most preferred, followed, in order, by artificial ponds, evaporative towers and, finally, dry towers. However, as we have illustrated, extreme deviations from the base case assumption where water and/or land costs become significant can rearrange this order in favor of evaporative or even dry towers. On the other hand, deviations from the base case where fuel and plant construction costs rise will reinforce the order of system preference defined for the base case (See Tables 7.3 and 7.4). This is due to the fact that the rankings under the base scenario already reflect the efficiencies with which fuel and installed base-load capacity are used to meet a fixed power demand.

Tables 7.3 and 7.4 compare cooling system operation for the base case scenario and for the sensitivity to fuel and plant construction costs. In these tables, the cost of electrical generation is represented by annualized costs and present valued costs (discussed in Section 2.3). The present valued costs are based on a 10% discount rate over a 35 year plant lifetime.

It is worth noting from Tables 7.1 through 7.4 and from the results of the previous chapters the sensitivity to the major cost parameters. Consider fuel cost. Variation in fuel cost (and therefore fuel consumption) is relatively small for design changes within a particular cooling system (Sections 3.5, 4.5, 5.5, 6.5). The variation is larger between systems as the intrinsic differences in net efficiency between systems is more distinct. However, these changes are still relatively small when compared to the variation in the total production cost when the price of fuel changes over the range assumed in the sensitivity study.

Table 7.3 Annualized and Present Valued* Cooling System Costs
($ millions) 1200 MW Nuclear Plant

	Annual Cost	Present Value Cost	Incremental Present Value Cost
Base Scenario			
Once-Through (Surface Canal)	165.04	1,591.61	0.00
Once-Through (Diffusers)	166.59	1,606.58	14.98
Cooling Pond	167.58	1,616.16	24.56
Natural Draft Wet Tower	170.54	1,644.68	53.07
Mechanical Draft Wet Tower	169.98	1,639.28	47.67
Wet/Dry Tower	192.76	1,859.01	267.41
Dry Tower	206.80	1,994.35	402.75
2 x Fuel Cost			
Once-Through (Surface Canal)	203.13	1,959.00	0.00
Once-Through (Diffusers)	204.68	1,973.98	14.98
Cooling Pond	205.98	1,986.45	27.45
Natural Draft Wet Tower	209.18	2,017.32	58.33
Mechanical Draft Wet Tower	208.82	2,013.82	54.82
Wet/Dry Tower	233.21	2,249.07	290.07
Dry Tower	249.06	2,401.89	442.89
1.5 x Plant Cost			
Once-Through (Surface Canal)	226.54	2,184.74	0.00
Once-Through (Diffusers)	228.08	2,199.64	14.98
Cooling Pond	229.15	2,209.91	25.17
Natural Draft Wet Tower	232.52	2,242.37	57.63
Mechanical Draft Wet Tower	232.25	2,239.79	55.05
Wet/Dry Tower	260.01	2,507.58	322.84
Dry Tower	276.73	2,668.77	484.03
Water Cost ($1.00/1000 gal)			
Once-Through (Surface Canal)	165.04	1,591.61	0.00
Once-Through (Diffusers)	166.59	1,606.58	14.98
Cooling Pond	174.92	1,686.95	95.34
Natural Draft Wet Tower	177.19	1,708.77	117.17
Mechanical Draft Wet Tower	175.64	1,693.87	102.27
Wet/Dry Tower	194.55	1,876.27	284.67
Dry Tower	206.80	1,994.35	402.75

*Present value costs were calculated using a 10% discount rate over a 35-year plant lifetime.

Table 7.4 Annualized and Present Valued* Cooling System Costs
($ millions) 800 MW Fossil Plant

	Annual Cost	Present Value Cost	Incremental Present Value Cost
Base Scenario			
Once-Through (Surface Canal)	113.62	1,095.79	0.00
Once-Through (Diffusers)	114.37	1,102.99	7.50
Cooling Pond	114.79	1,107.05	11.25
Natural Draft Wet Tower	115.81	1,116.83	21.04
Mechanical Draft Wet Tower	115.87	1,117.44	21.64
Wet/Dry Tower	125.15	1,206.90	111.11
Dry Tower	128.42	1,238.43	142.64
2 x Fuel Cost			
Once-Through (Surface Canal)	153.51	1,480.42	0.00
Once-Through (Diffusers)	154.29	1,487.92	7.50
Cooling Pond	154.89	1,493.75	13.33
Natural Draft Wet Tower	156.14	1,505.76	25.34
Mechanical Draft Wet Tower	156.39	1,508.20	27.78
Wet/Dry Tower	165.98	1,600.75	120.34
Dry Tower	172.92	1,667.66	187.25
1.5 x Plant Cost			
Once-Through (Surface Canal)	148.36	1,430.74	0.00
Once-Through (Diffusers)	149.13	1,438.25	7.50
Cooling Pond	149.65	1,443.26	12.52
Natural Draft Wet Tower	150.97	1,445.94	25.19
Mechanical Draft Wet Tower	151.21	1,458.22	27.45
Wet/Dry Tower	162.04	1,562.74	131.99
Dry Tower	164.96	1,590.09	160.14
Water Cost ($1.00/1000 gal)			
Once-Through (Surface Canal)	113.62	1,095.79	0.00
Once-Through (Diffusers)	114.37	1,102.99	7.50
Cooling Pond	118.13	1,139.23	43.44
Natural Draft Wet Tower	118.11	1,139.08	43.29
Mechanical Draft Wet Tower	118.48	1,142.63	46.84
Wet/ Dry Tower	125.67	1,211.97	116.18
Dry Tower	128.42	1,238.43	142.64

*Present value costs were calculated using a 10% discount rate over a 35-year plant lifetime.

7.1.4 Comparison to Previous Studies

There has been a number of studies over the past 10 years that deal with the economic comparision of power plant cooling systems. Many of them have been directed toward dry tower systems where optimization of the dry cooled plant and application of site related costs (e.g. transmission costs, water costs, etc.) attempt to make that system more attractive. United Engineers and Constructors (UE&C) (1974) is the one major study that attempts optimization and comparison of the major alternative cooling systems. In this section, the results of some of these previous works will be compared with the results of this thesis.

Table 7.5 shows a comparison of this study with the results of United Engineers (1974), expressed in terms of incremental present value costs. These studies show reasonable agreement for the wet towers while vast differences are noticed for the cooling pond and the dry tower. The UE&C report does not identify the performance model for the cooling pond. The results for the pond suggest that high land or land preparation costs were used in the optimization. The simplistic design of the dry tower with a high back pressure turbine suggests insufficient optimization and costing of the components of the dry cooling system.

Teknekron (1976) considered only capital cost of unoptimized wet cooling systems and found considerable size variability with unit costs from 8 to 36 $/KW. The incremented capital costs (over once-through) for the mechanical and natural draft wet towers were found to be 9.1 $/KW and 14.4 $/KW, respectively. This can be compared with 7.5 $/KW and 13.3 $/KW for the mechanical and natural draft wet towers, respectively,

Table 7.5 Comparison of Results with United Engineers and Constructors (1974)

	Incremented Present Valued* Cost ($/KW)			
	Nuclear Plant		Fossil Plant	
	UE&C (1973 dollars)	This Study (1977 dollars)	UE&C (1973 dollars)	This Study (1977 dollars)
Once-Through	0.0	0.0	0.0	0.0
Cooling Pond	61.6	20.5	50.4	14.1
Natural Draft Wet Tower	30.9	44.2	31.8	26.3
Mechanical Draft Wet Tower	31.4	39.7	26.9	27.1
Dry Tower	180.0	335.6	125.2	178.3

* Present value costs for UE&C are evaluated at a discount rate of 8% over a 40-year lifetime while 10% over a 35 year lifetime was used in this thesis.

evaluated in this thesis.

In an attempt to examine the effect of water and water treatment costs on cooling system design, Gold (1976) compares evaporative cooling systems with dry cooling towers (using high backpressure turbines) to arrive at a difference in power production cost of 1.3 mills/KWH (excluding the cost of water) for nuclear plants. This is contrasted with 4.6 mills/KWH found in this thesis. The difference seems to lie in the treatment costs applied to the wet cooling systems and in the approach to replacement of lost performance.

Sebald (1976) and Rossie _et al_. (1972) both apply optimization of dry cooling towers and quantification of site specific characteristics to compare power production costs with wet towers. Sebald found the power production cost with dry towers to be 24% greater than that with wet towers for nuclear plants. While this is consistent with the results of the present study, Rossie's finding of a 1 mill/KWH difference between the wet and dry systems is not.

7.2 Comparison of Environmental Effects

The environmental effects associated with waste heat rejection systems have been detailed for each particular cooling system in Chapters III through VI. Table 7.6 is a qualitative summary of these effects which shows comparison between systems for each environmental impact. In general, as one moves from left to right on Table 7.6, the impacts decrease while, as observed in the previous section, the power production costs increase. Of course, site specific consideration of individual

Table 7.6 Comparison of Environmental Effects

Effect / Cooling System	Once-Through	Cooling Ponds	Mechanical Draft Wet Towers	Natural Draft Wet Towers	Dry Towers
Thermal Impact	Potentially large depending on site. May be minimized through outfall design.	Generally small. Restricted to blowdown heat. Could be significant on small streams but blowdown can be stored during critical periods	Generally small. Restricted to blowdown heat. Could be significant on small streams.	Generally small. Restricted to blowdown heat. Could be significant on small streams.	None
Intake Impacts	Potentially large depending on site.	Generally small due to modest make-up requirements. Could be significant on small streams.	Generally small due to modest make-up requirements. Could be significant on small streams.	Generally small due to modest make-up requirements. Could be significant on small streams.	None
Groundwater Impacts	Negligible	Potential contamination from seepage. Potential water table changes.	Usually negligible	Usually negligible	None
Blowdown - Chemical and Low Level Radioactive Wastes	Generally small. Large water bodies and no recycling imply low concentrations.	Potentially significant. May require low cycles of concentration or treatment. Ponds can store blowdown during low flows.	Potentially significant. May require low cycles of concentration or treatment.	Potentially significant. May require low cycles of concentration or treatment.	None

Table 7.6 Comparison of Environmental Effects (cont'd)

Effect \ Cooling System	Once-Through	Cooling Ponds	Mechanical Draft Wet Towers	Natural Draft Wet Towers	Dry Towers
Water Consumption	Considerable - About 0.4 gallons per KWH through forced evaporation. However, generally small fraction of water withdrawal.	Considerable - About 0.6 gallons per KWH through natural and forced evaporation. May vary considerably due to addition of seepage and subtraction of precipitation depending on site and accounting scheme.	Considerable - About 0.5 gallons per KWH through forced evaporation and drift.	Considerable - About 0.6 gallons per KWH through forced evaporation and drift.	None
Fogging, Icing and Visible Plume	Usually negligible due to relatively small water temperature rise.	Potentially significant depending on site. However large area (diffuse vapor source) and remote siting minimize impact.	More significant than ponds due to concentrated low level vapor source. May prevent siting in certain areas.	Higher release point minimizes ground level fog and ice but may affect visibility.	None
Drift	None	Negligible	Potential contamination to local soil and vegetation and corrosion of nearby structures.	Similar to mechanical draft but less significant due to higher release point.	None

Table 7.6 Comparison of Environmental Effects (cont'd)

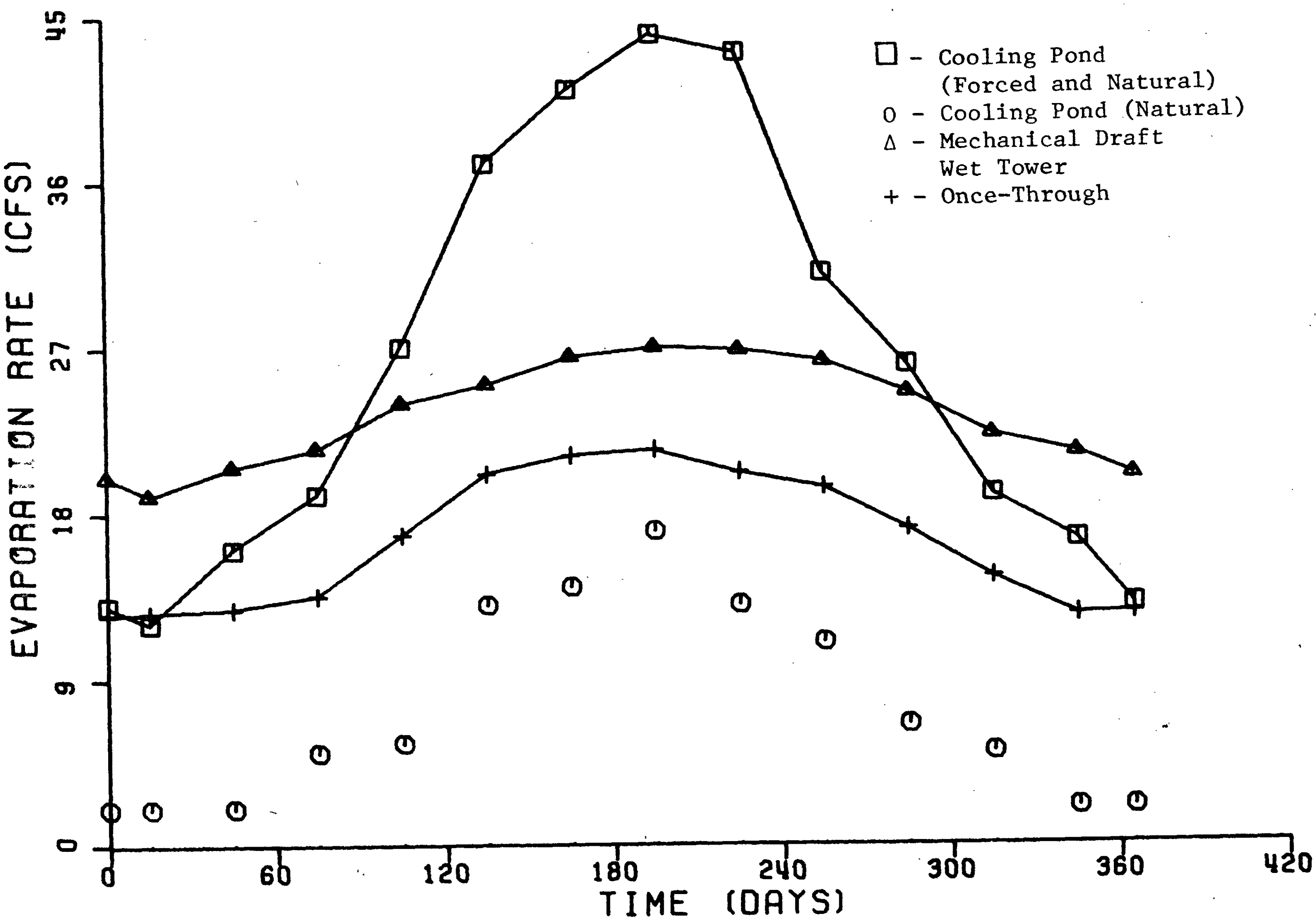
Effect / Cooling System	Once-Through	Cooling Ponds	Mechanical Draft Wet Towers	Natural Draft Wet Towers	Dry Towers
Land Use	Relatively small area required but land may be highly desireable for other uses due to its proximity to water.	Very large area required - typically 1-2 acres per MWe. Spray modules may reduce requirement by about a factor of 10.	Considerable area required though considerably less than ponds. Typically .012 acres per MWe.	Considerable area required. Typically .007 acres per MWe.	Considerable area required. Typically .020 acres per MWe.
Siting	Not very flexible. Must be located near large water supply.	Moderately flexible. Requires large open land area with suitable soil conditions and make-up water supply.	Moderately flexible. Requires reliable make-up source. (May require storage pond on unregulated rivers) Fogging, and icing may preclude some sites.	Moderately flexible. Requires reliable make-up source. Not suitable in hot arid climates or regions with hurricane potential.	Highly flexible. Negligible water requirement allows siting near fuel source or load center.
Noise	Generally small	Generally small	Potential annoyance from splashing on fill and fan operation.	Potential annoyance from splashing on fill.	Potential annoyance from fan operation.
Aesthetics	Cooling system least noticeable but plant may be highly visible due to nature of site.	Multiple use of pond may enhance aesthetics.	Small impact from visible plume.	More visible than mechanical draft due to tower height and size.	Relatively large structure but no visible plume.

impacts is necessary to accurately compare the various cooling systems.

The issue of water conservation was given particular attention in the previous chapters. It was found that while water withdrawal rates of once-through systems are higher than for both ponds and wet towers, consumptive water use is less. Ponds were found to consume slightly more water than wet towers because one must consider natural evaporation and seepage from the ponds. Since both natural and forced evaporation show seasonal variation, it is worth analyzing the transient evaporation behavior. Figure 7.3 plots the monthly evaporation rates for each cooling system at the study site for 1970. Natural and forced evaporation for the ponds was computed using the transient mathematical mold discussed in Section 4.3, while forced evaporation for the wet towers was computed using the procedure discussed in Section 5.4. In both cases three-hour time steps were used. The evaporation for the once-through system was based on monthly averaged meteorology using the procedure discussed in Section 3.4.

It is obvious from the plot that the peak evaporation for all the cooling systems is during the summer months. It is unfortunate that this period is usually correlated with lowest river flows and highest water demands by other sectors (e.g. residential and agricultural users). It should be mentioned that while ponds show the highest peak evaporation (44 cfs in July for this example) their intrinsic storage capability can be used to reduce their make-up requirements during critical times of the year. Wet towers by contrast must have a continuous make-up water supply which may necessitate construction of an adjacent storage pond if the primary water supply is not reliable.

Figure 7.3 Cooling System Transient Evaporation Rates
(monthly average for 1970)



7.3 Conclusions

This thesis has compared alternative cooling systems for steam-electric power plants. Differences between cooling systems were summarized in this chapter on the basis of performance, economics, fuel and water consumption, and environmental effects. The results attained in this study are clearly dependent on the approach, the site characteristics and the assumptions used.

While the true cost of each power plant/cooling system is a problem for the architect-engineer and utility planner, the consistent comparison performed in this study allows one to clearly identify differences in costs for various cooling system alternatives. The detailed design procedures used in this study, especially in the optimization of the cooling ponds and dry tower, are believed to represent the best state of the art performance models. Finally, although only a single site was considered in this study, one can identify the general behavior of each cooling system technology with regard to the issues presented.

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