Cost Analysis of Power Plant Cooling Using Aquifer Thermal Energy Storage

May 1989

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**Pacific Northwest Laboratory
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COST ANALYSIS OF POWER PLANT COOLING
USING AQUIFER THERMAL ENERGY STORAGE

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SUMMARY

Most utilities in the U.S. experience their peak demand for electric power during periods with high ambient temperature. Unfortunately, the performance of many power plants decreases with high ambient temperature. The use of aquifer thermal energy storage (ATES) for seasonal storage of chill can be an alternative method for heat rejection. Cold water produced during the previous winter is stored in the aquifer and can be used to provide augmented cooling during peak demand periods increasing the output of many Rankine cycle power plants. This report documents an investigation of the technical and economic feasibility of using aquifer thermal energy storage for peak cooling of power plants.

The investigation consists of developing models that describe the performance of key power plant components such as the cooling tower, condenser, heat exchangers, and the aquifer system. The performance models are combined with cost models to calculate unit capital costs and energy costs for this concept.

In the study, seven cases are examined. The first is a "base" case to which the six others are compared. The six cases are variations on geographic location, desired condenser pressure, operation strategy, capital costs, and aquifer recovery temperature. Within each case, 15 combinations of cooling tower effectiveness and circulating water temperature range are considered. An optimum (minimum) cost is obtained for each combination. In general, it was found that costs decrease with increasing range and/or decreasing cooling tower effectiveness. These conditions would be encountered when the installed cooling tower was undersized or not performing up to design specifications.

The ATES power plant cooling augmentation system increases the output of the plant when the ATES system is in operation. The cost of the increased power production depends on the cost of the ATES system and the amount of increase in power production. For plants with a large cooling tower range and a cooling tower with poor performance (cooling tower effectiveness between 0.4 and 0.5), the ATES system produces relatively large energy savings. This

results in a levelized energy cost, for the increased power production, of between 50 and 1000 mills/kWh. Corresponding unit capital costs range between 100 and 1100 \$/kWe. For cooling tower effectiveness in excess of 0.6 or cooling tower ranges less than 11°C (implying an appropriately sized or oversized cooling tower), levelized energy costs exceed 1000 mills/kWh and capital costs exceed 1100 \$/kWe.

While it appears that the ATEs system for peak power plant cooling can be economic for some situations, the applications will be limited to power plants with either undersized cooling towers or cooling towers that are not operating as designed. While not the subject of this study, dry cooled plants may be a more attractive application because of the large performance penalty associated with operating an all dry cooling tower at high ambient temperatures.

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NOMENCLATURE

A	Heat exchanger heat transfer area [m^2]
CC	Capital cost [\$]
C_p	Specific heat of water [$\text{MJ}/(\text{kg}\cdot^\circ\text{C})$]
CTTD	Condenser terminal temperature difference [$^\circ\text{C}$]
E	Electrical energy produced without ATES [MWh]
EFFECT	Cooling tower effectiveness [p.u.]
EFFIC	Gross cycle efficiency [p.u.]
ELEC	Annual cost of electricity for pumps and fans [\$/year]
FCR	Fixed charge rate
H	Number of hours at a wet-bulb temperature [hr]
HEAD	Pressure drop that pump must overcome [$\text{kJ}/(\text{kg}\cdot\text{sec})$]
HOURL	Number of hours of operation per day [hr/day]
i	Index
LEC	Levelized energy cost [\$/MWh]
$\dot{m}_{cw}$	Mass flow rate of circulating water [kg/sec]
$\dot{m}_{ad}$	Mass flow rate from the aquifer during discharge [kg/hr]
$\dot{m}_{rdt}$	Mass flow rate through heat exchanger [kg/hr]
$\dot{m}_w$	Mass flow rate per well [kg/hr]
n	Maximum value of index
N_{WELL}	Number of wells
O&M	Annual operation and maintenance cost [\$/year]
P_E	Gross electricity power output [MW]

P_{EATES}	Electric power output with ATES for selected condenser pressure [MW]
$PEFF$	Pump and motor efficiency [p.u.]
P_{EP}	Electric power output of plant without ATES at maximum wet-bulb temperature [MW]
P_{ES}	Gross electric power output after load shedding [MW]
POW	Pumping power [kW]
$\dot{Q}_{ATES}$	Heat rejection rate to aquifer [MW]
$\dot{Q}_{CT}$	Heat rejection rate by cooling tower [MW]
$\dot{Q}_{CTDES}$	Desired heat rejection rate [MW]
$\dot{Q}_{IN}$	Cycle heat input rate [MW]
$\dot{Q}_{RJ}$	Cycle heat rejection rate [MW]
$\dot{Q}_{RJS}$	Cycle heat rejection rate during load shedding [MW]
$\dot{Q}_{RJMAX}$	Cycle heat rejection rate at maximum condenser pressure [MW]
$RANGE$	Cooling tower range [°C]
RHO	Density of water [kg/m ³]
T_{adh}	Temperature at aquifer inlet during summer [°C]
T_{ad2}	Temperature at aquifer outlet during summer [°C]
T_{AWB}	Ambient wet-bulb temperature [°C]
T_{CI}	Temperature of circulating water at condenser inlet [°C]
T_{CTI}	Temperature of circulating water at cooling tower inlet [°C]
T_{CTO}	Temperature of circulating water at cooling tower outlet [°C]
T_{EXH}	Temperature at turbine exhaust [°C]
$TIME$	Total summer operating time [hr]
T_{2m}	Log mean temperature difference [°C]
U	Overall heat transfer coefficient of heat exchanger [MJ/(m ² -sec-°C)]

UCC	Unit capital cost [\$/kWe]
V_{DT}	Volume of day tank [m ³]
WPPSS	Washington Public Power Supply System

COST ANALYSIS OF POWER PLANT COOLING USING AQUIFER THERMAL ENERGY STORAGE

1.0 INTRODUCTION

Most utilities in the United States experience their peak demand for electric power during periods with high ambient temperature. Unfortunately, the performance of many power plants decreases with high ambient temperature. A review of Licensed Operative Reactors, NUREG-0020 (U.S. Nuclear Regulatory Commission 1987) showed that, on average, nuclear power plant output is lower than rated capacity by 2% or 17.9 MWe per plant. (See Appendix A.) This results in a total deficit of 1668 MWe for 93 plants. The worst case is the Grand Gulf nuclear power plant, which produces 142 MWe or 11% under its rated capacity. If we assume that oil and coal-fired plants suffer the same capacity reduction and recognize that nuclear power is approximately 10% of the nation's generating capacity, there is a 16,680 MWe capacity reduction nationwide.

It is reasonable to assume that oil and coal plants can be included in this estimate because NUREG-0020 attributes the capacity reduction to seasonal limitations. The seasonal limitation is experienced in the summertime when the heat sink (atmosphere, river, lake, ocean) temperature is high. The high heat sink temperature causes the plant to operate at a lower thermodynamic efficiency, which reduces output. For example, a 650-MWe plant with evaporative cooling can experience a 5.1-MWe capacity reduction for a 5°C increase in wet-bulb temperature. Coincidentally, the reduced output occurs during the high-demand period for utilities, that is, during summer afternoons.

The aquifer thermal energy storage (ATES) concept offers the potential of rejecting heat at a lower temperature, which would increase plant capacity during high demand periods without adding additional generating capacity. This is accomplished by storing water, cooled during winter, for use in power plant cooling during times with high ambient temperature.

The benefit of storing or rejecting energy to underground water tables, also known as aquifers, has been recognized and exploited. For example, the

Shippingport nuclear power station mixed cold well water with river water to obtain additional cooling capacity and avoided the cost of replacing heat exchangers. The technology associated with ATES is developing rapidly with a number of field tests in progress in the United States and overseas.

The objective of this activity is to perform a scoping study that will investigate the feasibility of using ATES for power plant cooling augmentation. The scoping study is limited to retrofit application of ATES systems to power plants with evaporative cooling towers. Application of ATES to dry cooling towers or to river, lake, or ocean nonevaporative cooling towers may be viable alternatives but are not included. Use of ATES systems in new construction may allow smaller cooling towers, but this alternative is beyond the scope of this study.

The concept of applying ATES to power plant cooling is described in Section 2.0, followed by descriptions of thermal and cost models in Section 3.0. The cases selected for analysis are shown in Section 4.0, with results presented in Section 5.0. Finally, conclusions are presented in Section 6.0.

2.0 CONCEPT DESCRIPTION

The ATEs power plant cooling augmentation system stores winter-cooled water in natural aquifers for use during the summer, when ambient temperatures are higher. During the winter, water is drawn from wells into a cooling tower, where it is chilled by the cool ambient temperatures, then injected into the aquifer again, which sustains the cool temperatures until the water is needed for summertime use. Then, it is recovered from the aquifer in the summer and placed in a day tank, from which it is drawn into a heat exchanger to reduce the temperature of cooling water coming from the power plant's condenser. The routing of water from the wells to the cooling tower, day tank, heat exchanger, and condenser varies depending on the phase of operation, i.e., whether the system is in the cooling augmentation phase (summer) diagrammed in Figure 2.1, or the heat rejection phase (winter) in Figure 2.2.

During the cooling augmentation phase in summer, winter-chilled water is continuously withdrawn (24 hours per day) from the aquifer to a day tank. When peak demand coincides with hot weather, the day tank water is circulated through a heat exchanger that cools the condenser cooling water, allowing the condenser to operate at a lower pressure. The purpose of the day tank is to allow a reduced number of wells. The rate of flow from the aquifer determines the number of wells. Without a day tank, six times more wells would be required to meet the peak flow. The day tank has a capacity sufficient to provide cooling for a 4- to 6-hour period. The warmed day tank water is also pumped continuously back to the injection wells. The day tank is arranged to form a thermocline with cold water on the bottom and heated water on the top. Thus, the day tank always has the same amount of water but with a varying ratio of cold to heated water. A heat exchanger is included in the design to prevent mixing of chemically balanced circulating water with untreated aquifer water and to avoid putting treated water back into the aquifer. Thus, the only environmental impact will be caused by heating and cooling the aquifer.

In winter (Figure 2.2), the day tank and heat exchanger are in standby. Because plant cooling tower capacity exceeds plant cooling requirements in the winter, it is assumed that one or more cells of the cooling tower could be

FIGURE 2.1. ATEs Power Plant Cooling Augmentation System, Cooling Augmentation Phase

FIGURE 2.2. ATE Power Plant Cooling Augmentation System, Heat Rejection Phase

isolated from the circulating water and be used to reject heat from the summer-warmed aquifer water. This avoids the cost of an additional aquifer cooling tower. The cell(s) of the cooling tower used to cool aquifer water may require cleaning before continued use for circulating water cooling.

The aquifer layout and piping arrangement are shown in Figure 2.3. Two identical well fields are used, one for chill storage and one for reject heat storage. The wells are arranged in a hexagonal close-pack arrangement, and piping connecting the wells to the manifold is orthogonal and parallel. This layout minimizes the required land area and piping length. The well flow rate and spacing is very site-specific. In this study, a well flow rate of 1000 gpm with a spacing of 15 acres has been assumed.

FIGURE 2.3. Well Layout Design and Routing of Fluid Transmission Lines

3.0 MODEL DEVELOPMENT

The objective of this study is to conduct a feasibility analysis; therefore, simple models were developed. Models were developed only for summer operation. Winter conditions were not modeled because summer operation determined the key design parameters. The equations and definitions are introduced, then the derivations of relationships used for determining energy output, component sizes, and component costs are presented. The energy output is based on the Middletown plant (Drost and Johnson 1980) and costs are from Brown (1983).

3.1 EQUATIONS AND DEFINITIONS

The equations used are taken from heat exchanger theory and basic energy balance equations. The equations are presented without derivation.

3.1.1 Cooling Tower Model

Cooling tower effectiveness is the ratio of the amount of heat transferred to the amount of heat that could be transferred with infinite surface area, and can be expressed as a ratio of temperature differences (Welty, Wicks, and Wilson 1976; Kays and London 1964). For an evaporative cooling tower (Figures 2.1 and 2.2), the effectiveness is

$$\text{EFFECT} = (T_{\text{CTI}} - T_{\text{CTO}}) / (T_{\text{CTI}} - T_{\text{AWB}}) \quad [\text{p.u.}] \quad (3.1)$$

where EFFECT = cooling tower effectiveness [p.u.]

T_{CTI} = circulating water temperature entering cooling tower [°C]

T_{CTO} = circulating water temperature leaving cooling tower [°C]

T_{AWB} = ambient wet-bulb temperature [°C].

The range is defined as the temperature range of the circulating water

$$\text{RANGE} = T_{\text{CTI}} - T_{\text{CI}} \quad [^{\circ}\text{C}] \quad (3.2)$$

where T_{CI} equals circulating water temperature entering the condenser [°C].

Heat exchanger terminal temperature difference is the temperature difference between the two streams leaving the heat exchanger. For the condenser (Figures 2.1 and 2.2), this parameter is

$$CTTD = T_{EXH} - T_{CTI} \quad [^{\circ}\text{C}] \quad (3.3)$$

where CTTD is condenser terminal temperature difference [$^{\circ}\text{C}$] and T_{EXH} is turbine exhaust or condenser temperature [$^{\circ}\text{C}$]. Heat losses from the circulating water pipe and condenser are neglected as well as subcooling effects.

3.1.2 Power Plant Model

The power plant efficiency used in this study is the gross cycle efficiency, which ignores cycle electrical parasitics.

$$EFFIC = P_E / \dot{Q}_{IN} \quad [\text{p.u.}] \quad (3.4)$$

where $EFFIC$ = gross cycle efficiency [p.u.]

P_E = gross electricity power output [MW]

$\dot{Q}_{IN}$ = cycle heat input rate [MW]

and

$$\dot{Q}_{IN} = P_E + \dot{Q}_{RJ} \quad [\text{MW}] \quad (3.5)$$

where $\dot{Q}_{RJ}$ is cycle heat rejection rate [MW].

Substituting Equation (3.5) into (3.4) and solving for $\dot{Q}_{RJ}$ yields

$$\dot{Q}_{RJ} = P_E (1/EFFIC - 1.0) \quad [\text{MW}] \quad (3.6)$$

3.1.3 Heat Exchanger Model

The ATES heat exchanger (Figures 2.1 and 2.2) is sized using the log mean temperature difference approach:

$$\dot{Q}_{ATES} = U(A) T_{lm} \quad [MW] \quad (3.7)$$

where U = overall heat transfer coefficient $[MJ/(m^2 \cdot sec \cdot ^\circ C)]$

A = heat transfer area $[m^2]$

$\dot{Q}_{ATES}$ = heat rejected to aquifer $[MW]$

and

$$T_{lm} = \frac{(T_{CTO} - T_{adh}) - (T_{CI} - T_{adl})}{\ln[(T_{CTO} - T_{adh}) / (T_{CI} - T_{adl})]} \quad [^\circ C] \quad (3.8)$$

where T_{lm} = log mean temperature difference $[^\circ C]$

T_{adh} = temperature of heated water to aquifer $[^\circ C]$

T_{adl} = temperature of cool water from aquifer $[^\circ C]$.

3.2 DETERMINATION OF ENERGY OUTPUT WITH AND WITHOUT ATES

Plant output with ATES is constant because the ATES system maintains a selected condenser temperature during the period that the ATES system is in use. Plant output without ATES will vary as the ambient wet-bulb temperature varies. Therefore, a relationship that provides the condenser temperature as a function of ambient wet-bulb temperature is needed.

Without ATES, the temperature of the circulating water leaving the cooling tower equals the temperature of the circulating water entering the condenser, or

$$T_{CTO} = T_{CI} \quad [^\circ C] \quad (3.9)$$

Therefore, the numerator of Equation (3.1) can be replaced with the range from Equation (3.2) and solved for T_{CTI} . Substituting that expression for T_{CTI} into Equation (3.3) and solving yields the desired relationship for the condenser temperature.

$$T_{EXH} = (RANGE/EFFECT) + CTTD + T_{AWB} \quad [^{\circ}C] \quad (3.10)$$

If conditions result in a condenser temperature in excess of the maximum allowable for the given turbine design, electricity production is reduced (load shedding to prevent turbine damage) to maintain the condenser at the maximum allowable temperature. The determination of power output after load shedding is given by Equation (3.11) where Equation (3.6) has been solved for P_E , and the variables subscripted with an "S" to denote load shedding conditions:

$$P_{ES} = \dot{Q}_{RJS} / (1/EFFIC_S - 1.0) \quad [MW] \quad (3.11)$$

Determination of P_{ES} (power output after load shedding) requires information on the reject heat and efficiency. The efficiency is assumed to be the cycle efficiency at the maximum temperature. The actual efficiency is somewhat less due to part-load characteristics of the turbine, but this is neglected in this study. The reject heat is found from a heat balance assuming constant circulating water flow rate and specific heat:

$$\dot{Q}_{RJS} = \dot{Q}_{RJMAX} [(T_{CTI} - T_{CTO})/RANGE] \quad [MW] \quad (3.12)$$

where $\dot{Q}_{RJMAX}$ equals heat rejection rate [MW] from Table 3.1, and T_{CTI} and T_{CTO} are determined from Equations (3.3) and (3.1), respectively.

The electrical energy output is obtained by multiplying the power level (P_E and P_{ES}) by H, the number of hours at the corresponding ambient temperature

TABLE 3.1. Data for Middletown Plant

Condenser Pressure, in. Hg	Condenser Temperature, °C (T_{EXH})	Gross Electric Output, MWe (P_E)	Cycle Heat Input Rate, MWth ($\dot{Q}_{IN}$)	Cycle Heat Rejection Rate, MWth ($\dot{Q}_{RJ}$)
2.5	42.5	657	1534	876
3.0	46.1	653.8	1534	880
3.5	49.2	650	1534	884
4.0	51.9	645.3	1534	888
4.5	54.4	640.1	1534	893.6
5.0	56.7	634.4	1534	899.3

oil. The energy is summed for all ambient temperature increments up to the number of hours of operation assumed for ATES:

$$E = \sum_{i=1}^n P_{Ei} * H_i + \sum_{j=n+1}^m P_{ESj} * H_j \quad [\text{MWh}] \quad (3.13)$$

The hours corresponding to wet-bulb temperatures [temperature duration curves from Drost and Johnson (1980) and Pacific Northwest Laboratory (1984)] are tabulated in Appendix B for the three locations considered in this study.

Table 3.1 provides the relationship of plant parameters to condenser temperature for a 650 MW, Middletown, U.S.A., nuclear power plant (Drost and Johnson 1980). To avoid turbine damage, the turbine is not allowed to exceed backpressure of 5 in. Hg.

3.3 SIZING ATES EQUIPMENT

Equipment sizes were required to develop concept cost estimates.

3.3.1 Heat Exchanger Sizing

Heat exchanger area is obtained from Equation (3.7), which requires determination of the maximum heat rejection rate to the aquifer and T_{2m} . The maximum heat rejection rate to the aquifer ($\dot{Q}_{ATES}$) is the difference between

the heat rejection rate required to maintain the selected condenser conditions and the heat rejection rate by the cooling tower at maximum wet-bulb temperature, or

$$\dot{Q}_{ATES} = \dot{Q}_{CTDES} - \dot{Q}_{CT} \quad [MW] \quad (3.14)$$

where $\dot{Q}_{CTDES}$ is the desired heat rejection rate and $\dot{Q}_{CT}$ is the heat rejection rate by the cooling tower.

The amount of heat rejected by the cooling tower is

$$\dot{Q}_{CT} = \dot{m}_{cw} C_p (T_{CTI} - T_{CTO}) \quad [MW] \quad (3.15)$$

where $\dot{m}_{cw}$ equals the mass flow rate of circulating water and C_p is the specific heat of circulating water [MJ/(kg-°C)].

The amount of heat rejected to maintain desired condenser conditions is

$$\dot{Q}_{CTDES} = \dot{m}_{cw} C_p (T_{CTI} - T_{CI}) \quad [MW] \quad (3.16)$$

Equation (3.17) is obtained by combining Equations (3.15) and (3.16) to eliminate $\dot{m}_{cw} C_p$ and obtain $\dot{Q}_{CT}$ in terms of $\dot{Q}_{CTDES}$, then substituting for $\dot{Q}_{CT}$ in Equation (3.14) and using Equation (3.2).

$$\dot{Q}_{ATES} = \dot{Q}_{CTDES} [1.0 - (T_{CTI} - T_{CTO}) / \text{RANGE}] \quad [MW] \quad (3.17)$$

where $\dot{Q}_{CTDES}$ is taken from Table 3.1 for the desired condenser conditions.

For combinations of range, effectiveness, CTID, and T_{adh} , heat exchanger area and cost will depend on the choice of heated water return temperature (T_{adh}). The value of T_{adh} can result in small heat exchanger and high flow or large heat exchanger and low flow, with flow rate determining pumping costs and day tank costs. Thus, a range of values of T_{adh} was considered to find the lowest cost system for each range/effectiveness combination.

The temperatures needed to obtain the log mean temperature difference in Equation (3.8) are obtained as follows. When the ATES system is operating,

the condenser temperature is held at its selected value. Thus, using an assumed value for CTDT, Equation (3.3) yields T_{CTI} . Equation (3.1) yields T_{CTO} for assumed values of cooling tower effectiveness and maximum wet-bulb temperature. The last unknown is the aquifer water temperatures. Results were calculated for Middletown and Jacksonville, Florida. Therefore, average aquifer recovery temperatures were assumed for both sites (3.00°C for Middletown and 6.67°C for Jacksonville).

3.3.2 Day Tank Sizing

The volume of the day tank depends on the flow rate through the heat exchanger and the number of hours per day of operation:

$$V_{DT} = (1.1)\dot{m}_{rdt}(24-HOURA)/RHO \quad [m^3] \quad (3.18)$$

where 1.1 = 10% oversizing

HOURA = number of hours per day of operation [hr]

RHO = density of water [kg/m³]

24 = number of hours per day

$\dot{m}_{rdt}$ = mass flow rate through heat exchanger [kg/hr]

V_{DT} = volume of day tank [m³].

The mass flow is

$$\dot{m}_{rdt} = 3600 \dot{Q}_{ATES} / [C_p(T_{ad2} - T_{adh})] \quad [kg/h] \quad (3.19)$$

3.3.3 Number of Wells

The number of wells is determined from the required flow rate to fill the day tank and the unit production per well. Because the day tank is filled over a 24-hour period, the flow rate to fill it is

$$\dot{m}_{ad} = \dot{m}_{rdt}(HOURA)/24 \quad [kg/h] \quad (3.20)$$

where $\dot{m}_{ad}$ is the mass flow rate from aquifer to the day tank during discharge.

The number of wells required is then

$$N_{\text{WELL}} = \dot{m}_{\text{ad}} / \dot{m}_{\text{w}} \quad [\text{p.u.}] \quad (3.21)$$

where $\dot{m}_{\text{w}}$ is unit production (kg/h) per well. (Value assumed for this study is 227,060 kg/h or 1000 gpm.) The ratio N_{WELL} is always rounded to the next highest integer, then doubled to account for both injection and recovery well fields.

3.4 ATES SYSTEM COSTS

The ATES system costs consist of capital costs and operation and maintenance (O&M) costs, as summarized in Table 3.2.

3.4.1 Capital Costs

The total well cost, heat exchanger cost, and day tank costs are obtained by multiplying assumed unit costs to the size parameters obtained in Section 3.3. The interconnecting piping and O&M depend on many factors. From Brown (1983), the costs per well of these items were obtained for 2.0, 5.0, and 20.0 MWth aquifer chill storage systems. It was desired to have costs for 1000-gpm wells with a 15-acre spacing (Figure 2.3). However, Brown (1983) reported results for wells with a flow rate of 300 gpm at 15- and 25-acre spacing, and 1000 gpm with 25-acre spacing. By using ratios of costs, estimates were obtained for the 1000-gpm, 15-acre combination. It was assumed that, for systems with more than 16 wells, the unit costs would be given by the results for the 16-well field. Costs were not included for a separate ATES cooling tower because it is assumed that the excess cooling tower capacity of the plant could be used in winter. Land costs were not included because it is assumed that the utility owns the land close to the plant. The cost range for the wells is obtained from a water well cost survey (National Water Well Association 1979) for wells producing 1000 gpm or more with depths from 65 to 2000 ft. The cost range for the heat exchanger is taken from Meyer and Hauz (1980) with the lower cost corresponding to carbon steel construction and the higher cost for stainless steel. Heat exchanger unit cost includes installation, fittings, and valves.

TABLE 3.2. Cost Summary^(a), 1987 Dollars

Fixed capital costs		
Literature search		430
Reconnaissance		36,600
Aquifer characterization		
Pump test		1,200
Tracer test		3,700
Anisotropy test		6,000
Geophysical log		18,400
Microbiology/geochemistry		12,300
Exploration drilling		49,000
Variable capital costs		
Environmental impact statement		14,100 /well
Equipment		
Well and pump (1000 gpm) ^(b)	61,250 - 147,000	/well
Heat exchanger ^(c)	123 - 245	/m ²
Piping		
Transmission		
(from wells to manifold)	2 wells	367,500 /well
	4 wells	240,960 /well
	16 wells	96,530 /well
Manifold (from field to day tank)	2 wells	92,400 /well
	4 wells	95,200 /well
	16 wells	139,000 /well
Day tank ^(d)		24 /m ³
Chill cooling tower (assume use of plant tower excess winter capacity)		0.0 /well
Contingency		8% of equipment
Operation and maintenance costs, such as maintenance and supplies, well abandonment general and administrative (G&A), repair and replacement, (function of number of wells):		
	2 wells	63,200 /well-yr
	4 wells	54,086 /well-yr
	16 wells	37,100 /well-yr
Electricity		
Recovery and injection pumping		0.041 /kW-h
Chilled water production source and injection		
Chilled water production cooling tower fan power		

(a) All costs from Brown (1983) except as noted.

(b) National Water Well Association (1979)

(c) Meyer and Hauz (1980)

(d) Petri, Ong, and Olszewski (1984)

3.4.2 O & M Costs

Operation and maintenance costs also depended on the number of wells and were taken from Brown (1983). Electricity costs are separate from other O&M cost components in Table 3.2. This was done to allow for different well depth than was used by Brown (1983). The pumping power is determined from

$$POW = [HEAD(\dot{m}_{ad})TIME/PEFF] \times 2 \quad [kW] \quad (3.22)$$

where POW = pumping power [kW]

HEAD = pressure drop that the pump must overcome [kJ/(kg-sec)]

TIME = total summer operating time [hr]

PEFF = pump and motor efficiency [p.u.]

$\dot{m}_{ad}$ = mass flow rate from aquifer to the day tank during discharge
[kg/hr]

2 = doubling to include winter cooling pumping.

Fan power associated with winter cooling tower operation to chill the aquifer water was estimated by assuming that 1/3 of the cooling tower capacity would be available for winter chilling. Installed fan power for the Middletown plant was estimated by scaling the installed fan power for a typical 1100-MW nuclear power plant. This resulted in an estimated fan power of 2106 kW. Multiplying this value by the number of hours of winter chilling gave the total energy consumption.

The sum of pumping and fan energy consumption was multiplied by the unit electricity cost to obtain the annual electricity cost for the ATEs system.

3.4.3 Levelized Energy Costs and Unit Capital Costs

Two costs are obtained: levelized energy cost and unit capital cost. Levelized energy cost is given by Equation (3.23).

$$LEC = \frac{CC \times FCR + (O\&M + ELEC)}{(P_{ATES} \times TIME) - E} \quad [\$ / MWh] \quad (3.23)$$

where LEC = levelized energy cost [\$/MWh]

CC = capital cost [\$]

FCR = fixed charge rate (assumed to be 0.18)

O&M = annual operations and maintenance cost [\$/year]

ELEC = cost of electricity for pumping and for fan power of the cooling tower during winter cooling [\$/year]

E = electrical energy for operation without ATES Equation [MWh]

P_{EATES} = electric power output with ATES for selected condenser pressure [MW] (P_E from Table 3.1).

Unit capital cost is given by Equation (3.24)

$$UCC = \frac{CC}{(P_{EATES} - P_{EP}) \times 1000} \quad (3.24)$$

where UCC is the unit capital cost and P_{EP} is the electric power output of the plant without ATES at maximum wet-bulb temperature [P_{ES} is either taken from Equation (3.11) or is given by P_E in Table 3.1].

4.0 SELECTION OF CASES

The study was limited to retrofit application to power plants with evaporative cooling towers. Within the scope, the major variables are location, cooling tower effectiveness, circulating water temperature range, desired condenser conditions, number of hours of desired operation, aquifer recovery temperature, and component unit costs. The location determines the meteorology and characteristics of the aquifer. Aquifer characteristics affect the cost of drilling wells and the well spacing, but for this study, it was assumed that well spacing and well costs were similar for the two locations considered.

It was decided to perform cost calculations to obtain the optimum cost system at various combinations of the following variables of range and effectiveness:

- two locations: Middletown, U.S.A. (Washington, D.C.) and Jacksonville, Florida
- two condenser pressures: 2.5 and 3.5 in. Hg
- two operational scenarios
- two capital cost variations [high and low cost wells and heat exchanger (Hx)]
- two aquifer recovery temperatures.

Middletown and Jacksonville were selected to give a range of environmental conditions. Middletown, U.S.A., is an average U.S. site with relatively mild summer climatic conditions for heat rejection. Jacksonville, Florida, is a site with extreme conditions for summer heat rejection. Seven cases are summarized in Table 4.1. In practice, cooling tower effectiveness ranges from 0.4 to 0.6 and circulating water temperature ranges have been observed as low as 8°C and as high as 16°C. Selecting 3 effectiveness values (0.4, 0.5, and 0.6), along with 5 values of range (8, 10, 12, 14, and 16°C), provides 15 combinations for each of the 7 cases. Some of the combinations would not benefit from an ATEs system because there are no hours during the year that

TABLE 4.1. Variations Studied

	<u>Base</u> <u>Case</u>	<u>Case</u> <u>2</u>	<u>Case</u> <u>3</u>	<u>Case</u> <u>5</u>	<u>Case</u> <u>6</u>	<u>Case</u> <u>7</u>	<u>Case</u> <u>8</u>
Location							
Middletown	X	X	X		X		X
Jacksonville				X		X	
Aquifer recovery temperature							
6.67°C (44°F)				X		X	X
4.5°C (40°F)	X	X	X		X		
Condenser pressure							
3.5 in. Hg	X	X	X	X			X
2.5 in. Hg					X	X	
Operation							
4 hr/day, 135 days	X		X	X	X	X	X
6 hr/day, 180 days		X					
Capital costs							
Lowest cost wells and Hx			X	X		X	
Highest cost wells and Hx	X	X			X		X

cause the condenser pressure to exceed its design conditions. These combinations are indicated by a zero in Tables 4.2 and 4.3. Table 4.2 reports results for a condenser pressure of 3.5 in. Hg, and Table 4.3 reports results for 2.5 in. Hg. Thus, optimized costs are obtained only for the non-zero combinations.

TABLE 4.2. Hours Above 3.5 in. Hg in Middletown

<u>Effectiveness</u>	<u>Range, °C</u>				
	<u>8</u>	<u>10</u>	<u>12</u>	<u>14</u>	<u>16</u>
0.4	0	288	1597	3358	4673
0.5	0	0	211	1138	2277
0.6	0	0	0	133	797

TABLE 4.3. Hours Above 2.5 in. Hg in Middletown

<u>Effectiveness</u>	<u>Range, °C</u>				
	<u>8</u>	<u>10</u>	<u>12</u>	<u>14</u>	<u>16</u>
0.4	797	2279	4014	5332	7122
0.5	48	797	1865	3358	4336
0.6	0	133	797	1865	3011

5.0 RESULTS

Results are reported in levelized energy costs (mills/kWh) and unit capital costs (\$/kWe). However, these figures-of-merit are based on gross energy and gross power, respectively. Gross energy and power do not include the impact of parasitic power consumption, which results in the costs being lower than if net energy and power were used. On the other hand, cooling tower fans may not be needed for winter chill; without the fans, levelized costs would be lower because cooling tower fan consumption is 50% of annual cost and about 20% of levelized energy costs. Nevertheless, useful comparisons can be made between successive cases because consistent procedures were used for all cases.

Results for cases 2 through 8 are compared against the base case in Figures 5.1 to 5.12. All of the curves have the same general shape: decreasing cost with increasing range and/or decreasing effectiveness. A plateau appears for cases of lower condenser pressure.

Figures 5.1 and 5.2 show that costs are slightly higher for a 6 hr/day, 180-day operation (case 2) than for a 4 hr/day, 135-day operation (base case). The reason is that for the 6 hr/day case, more wells, larger heat exchanger, larger day tank, and more pumping are required to provide the required cooling. If cooling for 4 hr/day was required for more than 135 days, the levelized energy cost would be reduced.

The base case is compared to higher aquifer recovery temperature (case 8) in Figures 5.3 and 5.4. For a 2°C aquifer recovery temperature increase, levelized and capital costs rise from 20 to 100 \$/MWh and from 10 to 50 \$/kWe. This demonstrates the importance of maintaining the aquifer at as low a temperature as possible.

The base case is compared to reduced unit costs for wells and heat exchanger (case 3) in Figures 5.5 and 5.6. For an estimated well cost of 1/3 the base-case cost and heat exchanger cost of 1/2 of the base-case cost, system costs are reduced by about 30%.

Costs drop dramatically for case 5 (Figures 5.7 and 5.8), which simulates a plant located at Jacksonville, Florida, with low-cost wells and heat exchanger. The lower costs are caused by the many hours of high wet-bulb temperature. (See Appendix B.) The extreme summer high temperatures more than compensated for the warmer winter aquifer temperature (6.67°C), compared to Middletown (4.5°C).

In Figures 5.1 through 5.8, ATES is beneficial for condenser ranges from 10°C or 11°C to 16°C . None of the variations so far show reasonable costs for the condenser ranges between 8°C and 10°C because the condenser pressure was held at 3.5 in. Hg for these cases.

The condenser pressure was reduced to 2.5 in. Hg in Figures 5.9 through 5.12. In 5.9 and 5.10 (case 6), it can be seen that ATES costs, though high, are orders of magnitude less than for the 3.5-in.-Hg condenser pressure in the 8°C to 10°C ranges. The lines cross, however, making the 2.5-in.-Hg ATES system more expensive for large ranges.

Picking parameters to give the lowest possible costs for ranges 8°C to 10°C resulted in Figures 5.11 and 5.12 (case 7). Again, the costs are high.

The horizontal plateaus on Figures 5.10 through 5.12 are not fully understood. It may be a real effect or a combination of coarse-binned temperature data and the use of discrete points in optimization. It is possible that the "knee" (right side) of the plateau is not the minimum for that point. We did observe some cases of two minima and picked the smaller. In the right-side plateau points, it may be that the optimization window was too small to catch the second minima. This would require further work beyond the scope of this study.

Thus, the lowest-cost ATES systems for augmenting evaporative cooling towers will be in the southern U.S. (Jacksonville, Florida), operating 4 hr/day for 135 days. For plants with circulating water temperature ranges from 8°C to 10°C , the levelized cost will be from 600 to 800 mills/kWh and capital costs from 500 to 1300 \$/kWe, assuming condensers are cooled to 2.5 in. Hg. For ranges higher than 10°C , 3.5-in.-Hg condenser pressure produces the lowest costs of 50 to 800 mills/kWh and 100 to 1100 \$/kWe.

FIGURE 5.1. Comparison of Levelized Energy Cost of Base Case and Case 2 (6 hr/day, 180-day operation)

FIGURE 5.2. Comparison of Unit Capital Cost of Base Case and Case 2 (6 hr/day, 180-day operation)

CONCLUSION: Extended ATES cooling system operation period slightly increases cost.

FIGURE 5.3. Comparison of Levelized Energy Cost of Base Case and Case 8 [Aquifer recovery temperature of 6.67°C (44°F)]

FIGURE 5.4. Comparison of Unit Capital Cost of Base Case and Case 8 [Aquifer recovery temperature of 6.67°C (44°F)]

CONCLUSION: A 2°C increase in the temperature of cooling water in the aquifer slightly increases cost.

FIGURE 5.5. Comparison of Levelized Energy Cost of Base Case and Case 3 (Low cost wells and Hx)

FIGURE 5.6. Comparison of Unit Capital Cost of Base Case and Case 3 (Low cost wells and Hx)

CONCLUSION: Low cost heat exchangers and wells substantially decrease cost.

FIGURE 5.7. Comparison of Levelized Energy Cost of Base Case and Case 5 [Jacksonville, Florida, $T_{AQO} = 6.67^{\circ}\text{C}$ (44°F), low cost wells and Hx]

FIGURE 5.8. Comparison of Unit Capital Cost of Base Case and Case 5 [Jacksonville, Florida, $T_{AQO} = 6.67^{\circ}\text{C}$ (44°F), low cost wells]

CONCLUSION: The costs for a plant installed at Jacksonville, Florida are substantially below the costs associated with a plant located in New England.

FIGURE 5.9. Comparison of Levelized Energy Cost of Base Case and Case 6 (condenser pressure at 2.5 in. Hg)

FIGURE 5.10. Comparison of Unit Capital Cost of Base Case and Case 6 (condenser pressure at 2.5 in. Hg)

CONCLUSION: Decreasing target turbine backpressure to 2.5 in. Hg substantially increases the cost of the system.

FIGURE 5.11. Comparison of Levelized Energy Cost of Base Case and Case 7 [Jacksonville, Florida, $T_{AQ0} = 6.67^{\circ}\text{C}$ (44°F), low cost wells and Hx, condenser pressure = 2.5 in. Hg]

FIGURE 5.12. Comparison of Unit Capital Cost of Base Case and Case 7 [Jacksonville, Florida, $T_{AQ0} = 6.67^{\circ}\text{C}$ (44°F), low cost wells and Hx, condenser pressure = 2.5 in. Hg]

CONCLUSION: Using a combination of low cost assumptions, the ATEs system is not competitive when cooling tower range is less than 10°C .

6.0 CONCLUSIONS

The results of this study show the economic attractiveness of the peak cooling of power plants using ATES as a function of plant cooling system performance characteristics and ambient conditions. Based on these results, a number of conclusions can be drawn including:

- Cooling System Characteristics - For a cooling system range between 10 and 16°C and a cooling tower effectiveness less than 0.5, the ATES system can be competitive with other sources of peak power. In these situations, the levelized cost of power can be as low as 50 mills/kWh for a capacity factor of between 0.06 and 0.12. For very low capacity factors, any source of electric power will be expensive. As cooling system range or cooling tower effectiveness increases, the ATES system rapidly becomes uneconomical. This suggests that there may be a market niche where the ATES system for power plant peak cooling may be attractive. Applications would consist of power plants with undersized cooling systems or cooling towers that do not meet design performance requirements.
- Ambient Conditions - The ATES system is more attractive in hot, humid climates where the performance penalty associated with high ambient temperature is largest.
- Other Applications - The study only considered retrofit application with evaporative cooling systems. In new construction, the ATES system would allow the intentional undersizing of the cooling towers because peak cooling loads would be met with the ATES system. Dry cooling applications are attractive because of the large performance penalty associated with operating a completely dry cooling system at high ambient temperatures.

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APPENDIX A

POWER PLANT CAPACITIES FROM LICENSED OPERATING REACTORS, NUREG-0020

APPENDIX A

POWER PLANT CAPACITIES FROM LICENSED OPERATING REACTORS, NUREG-0020

List of Power Plants with Their Maximum Reliable Capacity and Type of Cooling

	<u>Rates MWe</u>	<u>Maximum Reliable Capacity</u>	<u>Delta</u>	<u>Type of Cooling</u>
ANO-1	850	836	14	Reservoir, Ark R
ANO-2	912	858	54	Tower (natural)
Beaver Valley-1	835	810	25	Towers (natural)
Browns Ferry 1	1065	1065	0	Combined cycle
Browns Ferry 2	1065	1065	0	Combined cycle
Browns Ferry 3	1065	1065	0	Combined cycle
Brunswick 1	821	790	31	Atl O outfall
Brunswick 2	821	790	31	Atl O outfall
Byron	1120	1129	-9	Towers (natural)
Callaway 1	1171	1120	51	Tower (natural)
Calvert Cliffs-1	845	825	20	Chesapeake Bay
Calvert Cliffs-2	845	825	20	Chesapeake Bay
Catawba-1	1145	1145	0	Towers (mech)
Catawba-2	1145	1145	0	Towers (mech)
Cook-1	1030	1020	10	Lake Michigan
Cook-2	1100	1060	40	Lake Michigan
Cooper Station	778	764	14	Missouri R
Crystal River-3	825	821	4	Gulf of Mexico
Davis Besse-1	906	860	46	Tower (natural)
Diablo Canyon-1	1086	1073	13	Pac O
Diablo Canyon-2	1119	1079	40	Pac O
Dresden-2	794	772	22	Cooling lake
Dresden-3	794	773	21	Cooling lake
Duane Arnold	538	515	23	Tower (mech)
Farley-1	829	827	2	Tower (mech)
Farley-2	829	829	0	Tower (mech)
Fitzpatrick	816	796	20	Lake Ontario
Fort Calhoun	478	478	0	Missouri R
Ginna	470	470	0	Lake Ontario
Grand Gulf	1250	1108	142	Towers (natural)
Haddam Neck	582	569	13	Connecticut R
Hatch-1	777	768	9	Towers (mech)
Hatch-2	784	777	7	Towers (mech)
Hope Creek	1067	1067	0	Towers (mech)

	<u>Rates MWe</u>	<u>Maximum Reliable Capacity</u>	<u>Delta</u>	<u>Type of Cooling</u>
Indian Pt-2	873	864	9	Hudson R
Indian Pt-3	965	965	0	Hudson R
Kewaunee	535	503	32	Lake Michigan
LaSalle-1	1078	1036	42	Reservoir
LaSalle-2	1078	1036	42	Reservoir
Limerick-1	1055	1055	0	Tower (natural)
Main Yankee	825	810	15	Alt 0
McGuire-1	1150	1150	0	Reservoir
McGuire-2	1150	1150	0	Reservoir
Millstone-1	660	654	6	L I Sound
Millstone-2	870	857	13	L I Sound
Millstone-3	1154	1149	5	L I Sound
Monticello	545	536	9	Towers (mech)
Nine Mile Pt-1	620	610	10	Lake Ontario
North Anna-1	907	893	14	Reservoir
North Anna-2	907	893	14	Reservoir
Oconee-1	887	860	27	Reservoir
Oconee-2	887	860	27	Reservoir
Oconee-3	887	860	27	Reservoir
Oyster Creek	650	620	30	Barnegat Bay
Palisades	805	730	75	Towers (mech)
Palo Verde-1	1221	1221	0	Towers (mech)
Palo Verde-2	1221	1221	0	Towers (mech)
Peach Bottom-2	1065	1051	14	Towers (mech H)
Peach Bottom-3	1065	1035	30	Towers (mech H)
Pilgrim-1	655	670	-15	Cape Cod Bay
Point Beach-1	497	485	12	Lake Michigan
Point Beach-2	497	485	12	Lake Michigan
Prairie Island-1	530	503	27	Towers (mech)
Prairie Island-2	530	500	30	Towers (mech)
Quad Cities-1	789	769	20	Spray Canal
Quad Cities-2	789	769	20	Spray Canal
Rancho Seco	918	873	45	Towers (natural)
Robinson-2	700	665	35	Reservoir
Salem-1	1090	1079	11	Delaware R
Salem-2	1115	1106	9	Delaware R
San Onofre-1	436	436	0	Pac 0
San Onofre-2	1070	1070	0	Pac 0
San Onofre-3	1080	1080	0	Pac 0
Sequoyah-1	1148	1148	0	Combined cycle
Sequoyah-2	1148	1148	0	Combined cycle

	<u>Rates MWe</u>	<u>Maximum Reliable Capacity</u>	<u>Delta</u>	<u>Type of Cooling</u>
St Lucie-1	830	827	3	Alt 0
St Lucie-2	830	837	-7	Alt 0
Summer	900	885	15	Reservoir
Surry-1	788	781	7	James R
Surry-2	788	781	7	James R
Susquehanna-1	1065	1032	33	Tower (natural)
Susquehanna-2	1065	1032	33	Tower (natural)
TMI-1	819	776	43	Towers (nat & mech)
Trojan	1130	1075	55	Tower (natural)
Turkey Pt-3	693	666	27	Canals
Turkey Pt-4	693	666	27	Canals
Vermont Yankee	514	504	10	Towers & Conn R
Washington Nuclear-2	1100	1095	5	Towers (mech)
Waterford-3	1104	1104	0	Mississippi R
Wolf Creek-1	1170	1128	42	Cooling lake
Yankee Rowe	175	167	8	Deerfield R
Zion-1	1040	1040	0	Lake Michigan
Zion-2	1040	1040	0	Lake Michigan

APPENDIX B

WET-BULB TEMPERATURE DURATION DATA FOR TWO SITES

APPENDIX B

WET-BULB TEMPERATURE DURATION DATA FOR TWO SITES

Middletown, U.S.A. (a) (Washington, D.C.)		Jacksonville, Florida (b)	
Temperature, °F	Hours	Temperature, °F	Hours
81.5	0	80	17
79.5	1.1	77	280
77.5	10.3	74	1264
75.5	36.8	71	1421
73.5	238.0	68	866
71.5	274.8	65	843
69.5	318.8	62	783
67.5	331.3	59	566
65.5	341.2	56	490
63.3	327.9	53	454
61.5	321.0	50	476
59.5	319.8	≤47	1300 (c)
57.5	313.2		
55.5	308.3		
53.5	299.5		
51.5	292.9		
49.5	305.5		
≤47.5	4726.4		
≤37.4	2687.1 (c)		

(a) Taken from Drost and Johnson (1980).

(b) Taken from PNL-5143, Vol. 1, Weather Data for Simplified Energy Calculation Methods, Volume 1 - Eastern United States TMY Data, August 1984.

(c) Number of hours greater than hours of summer operation.

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