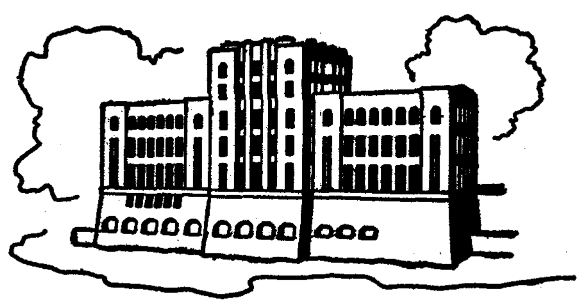
ISWRRI Completion Report
No. ISWRRI–98

OPTIMUM MECHANICAL DRAFT WET COOLING TOWERS TO SUPPLEMENT ONCE-THROUGH COOLING AT SELECTED MISSOURI RIVER SITES

by

A. R. Giaquinta and T. E. Croley II

Project supported in part by a grant from the U.S. Department of the Interior, Office of Water Research and Technology (Project No. A-061-IA) made available through the Iowa State Water Resources Research Institute.



IIHR Report No. 224

Iowa Institute of Hydraulic Research
The University of Iowa
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PROJECT COMPLETION REPORT

OWRT PROJECT NO. A-061-IA

Optimum mechanical draft wet cooling tower
to supplement once-through cooling at
selected Missouri River sites.

Agreement No. 14-34-0001-8017

Project dates: July 1976 - September 1979

Principal Investigators

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Institute of Hydraulic Research
The University of Iowa
Iowa City, Iowa 52242
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NOTATION

The following symbols are used in this paper:

A_c	=	Surface area of the condenser heat exchanger
F_c	=	Crossflow correction factor for the condenser
k_1	=	Conversion factor = 2.011×10^6 (gal/min)°F/Btu/hr = 1.196×10^{-4} (m^3/sec)°K/W
L_c	=	Water flow rate through the condenser
L_o	=	Amount of cooled water discharged to river
L_r	=	River flow rate
L_T	=	Total water flow rate through the cooling tower
LMTD	=	Logarithmic mean temperature difference for the condenser
Q_T	=	Heat rejection rate from condenser
Q_w	=	Cooling tower heat rejection rate
RHAC	=	River heat assimilation capacity
S_{max}	=	Maximum allowable turbine back pressure
T_{db}	=	Dry-bulb temperature
T_i	=	Temperature of hot water entering the cooling tower
T_{ic}	=	Condenser inlet water temperature
T_m	=	Maximum allowable river temperature
T_o	=	Temperature of cold water leaving the cooling tower
T_{oc}	=	Condenser outlet water temperature
T_r	=	River water temperature
T_s	=	Turbine exhaust steam temperature
T_{wb}	=	Wet-bulb temperature
U_c	=	Heat transfer coefficient for condenser
W_w	=	Evaporative water loss

Δt = Condenser temperature rise

δ = Allowable river temperature rise

τ_s = Turbine throttle opening level

OPTIMUM MECHANICAL DRAFT WET COOLING TOWERS TO SUPPLEMENT ONCE-THROUGH COOLING AT SELECTED MISSOURI RIVER SITES

By Arthur R. Giaquinta and Thomas E. Croley II

INTRODUCTION

Low thermal efficiencies of steam-electric power plants result in rejection of large amounts of waste heat to the surroundings. Waste heat is dissipated basically in two ways: once-through cooling and auxiliary system cooling. In once-through cooling, waste heat from the condenser is rejected to a natural water body such as a river. Heat eventually is transferred through evaporation, conduction, and convection from the receiving water body to the atmosphere. Auxiliary cooling involves the employment of cooling devices such as cooling towers or cooling ponds through which heat is transferred directly to the atmosphere. From the economic and water conservation points of view, once-through cooling is the best method for managing power plant waste heat in those situations where it can be demonstrated that the addition of thermal loads to a natural water body will not cause any harmful ecological effects and also will not violate environmental thermal standards.

Once-through cooling thus should be utilized to the maximum permissible extent, determined by the available river heat assimilation capacity which is defined as the product of the allowable river temperature rise and river flow rate. The allowable river temperature rise is set by federal and state water pollution control agencies for prevention of ecological damage. By using all of the permissible heat assimilation capacity with once-through cooling and by employing auxiliary cooling devices as "helper" systems to reject waste heat above the permissible capacity, a combination cooling system can be designed to achieve the required total heat dissipation. A combination system consisting of once-through cooling and a "helper" cooling-tower system is referred to herein as a hybrid cooling system. Compared with other cooling devices, it appears that mechanical draft wet cooling towers offer reasonable supplemental cooling capability in terms of capital expenses and land requirements for power plants located along rivers. Use of a helper system is also governed by its water consumption (evaporation) related to water costs and availability.

The wet tower/once-through hybrid cooling-system concept is not entirely new to the power industry. In 1970, Kadel (8) made a study of arrangements of cooling towers in conjunction with once-through cooling to reduce evaporative losses and to increase plant site potential. The description of several possible cooling system arrangements was addressed. Other studies in this area (7, 11, 14) were not concerned with compliance with thermal regulations or full utilization of permissible heat assimilation capacity of the water body. Recently, more attention has been given to hybrid cooling systems at river sites by Su (16) who found that a substantial reduction of cooling expenses resulted from the use of a hybrid cooling system, instead of a simple once-through system, at Fort Calhoun, Nebraska. However, his analysis was not a complete study of hybrid cooling system arrangements.

The aim of this study is to present a complete, detailed analysis of hybrid cooling systems and their applications. The specific objectives are:

1. To construct computer based, numerical models for determining the most economical arrangement of hybrid cooling systems and the corresponding

tower size for power plants along major rivers;

2. to analyze typical, but specific, situations on the Missouri River illustrating the use of hybrid cooling systems;

3. to compare the thermodynamic and economic performance of hybrid cooling systems with that of once-through and closed-cycle wet tower cooling systems;

4. to evaluate the effects of unit water cost, water consumption constraints, and seasonal variations of meteorological and hydrothermal conditions on the optimum design and utilization of hybrid cooling systems.

The method of identifying the optimum design of hybrid cooling systems is based on the "least cost" rule. For each type of hybrid cooling system, the total cooling-related costs are computed for an entire range of design parameters such as tower length, height, and condenser flow rate. The optimum cooling system is identified by selecting the tower size and condenser flow rate corresponding to the minimum total cost.

BACKGROUND

In once-through cooling systems, the amount of waste heat that can be discharged to the surroundings depends on the allowable heat assimilation capacity of the ambient water. The river heat assimilation capacity is a function of the physical characteristics of the receiving water, such as the flow rate and the temperature distribution at various points downstream from the discharge structure. Detailed descriptions of river-temperature prediction and river heat assimilation capacity have been developed by Paily et al. (13).

The various types of wet towers in current use are described elsewhere (10, 12). The type considered herein is the mechanical draft crossflow wet cooling tower. The heat balance equations for this cooling system are given by Croley et al. (5) in terms of air flow rate through the cooling tower; temperatures of the air entering and leaving the tower; water flow rate loading of the tower; tower fill volume; and various physical properties of the water and air. The details of the procedure for solving the equations for the cooling tower exit cold water temperature also are described by Croley et al. (5). For a hybrid cooling system the cooling tower heat rejection rate plus the river heat assimilation capacity, RHAC, should be greater than or equal to the total heat rejection rate to the cooling water through the condenser, Q_T; that is,

$$Q_T = L_c(T_{oc} - T_{ic})/k_1 \leq Q_w + \mathrm{RHAC} \quad \ldots\ldots\ldots\ldots (1)$$

or

$$Q_T \leq L_T(T_i - T_o)/k_1 + \mathrm{RHAC} \quad \ldots\ldots\ldots\ldots (2)$$

$$\begin{aligned} \mathrm{RHAC} &= \delta L_r/k_1;\ \text{if } T_r \leq T_m - \delta \\ &= (T_m - T_r)L_r/k_1;\ \text{if } T_m - \delta < T_r < T_m \\ &= 0;\ \text{if } T_r > T_m \quad \ldots\ldots\ldots\ldots (3) \end{aligned}$$

in which L_c = condenser flow rate; T_{oc} = condenser outlet water temperature; T_{ic} = condenser inlet water temperature; $k_1 = 2.011 \times 10^6$ (gal/min) °F/Btu/hr = 1.196×10^{-4} (m³/sec)°K/W; Q_w = cooling tower heat rejection rate; L_T = total water flow rate through the cooling tower, T_i and T_o = water temperatures entering and leaving the cooling tower, respectively; δ = allowable temperature rise; L_r = river water flow rate; T_r = river temperature; and T_m = maximum allowable river temperature. Furthermore,

$$Q_T = L_c(T_{oc} - T_{ic})/k_1 = F_c U_c A_c(\mathrm{LMTD}) \quad \ldots\ldots\ldots\ldots (4)$$

in which F_c = crossflow correction factor for the condenser; U_c = heat transfer coefficient for the condenser; A_c = the surface area of the condenser heat exchanger; and LMTD = the logarithmic mean temperature difference for the condenser. LMTD may be computed as shown elsewhere (5) where the relationship for the condenser outlet water temperature in terms of the turbine exhaust steam temperature, T_s, and the condenser heat rejection rate also is given;

$$T_s - T_{oc} = cQ_T \quad (5)$$

where

$$c = \frac{k_1}{L_c[\exp(\frac{F_c U_c A_c k_1}{L_c})-1]} \quad (6)$$

It is noted that regardless of the type and size of the hybrid cooling system used, the turbine must operate such that the turbine back pressure will not be higher than the maximum allowable back pressure, S_{max}. Otherwise, the turbine must be operated at a partial throttle opening level, τ_s, such that the turbine back pressure is S_{max}. Then, the nameplate power output cannot be produced, resulting in a loss of power capacity. The loss in power capacity and the associated energy loss depend upon meterological and hydrothermal conditions. The procedure for calculating the heat rejection rate, Q_T, and τ_s for the integrated system is presented by Hsu (6).

HYBRID COOLING SYSTEM THERMODYNAMICS

Three different wet tower/once-through hybrid cooling system arrangements are shown in Fig. 1. Type 1 is an open-cycle

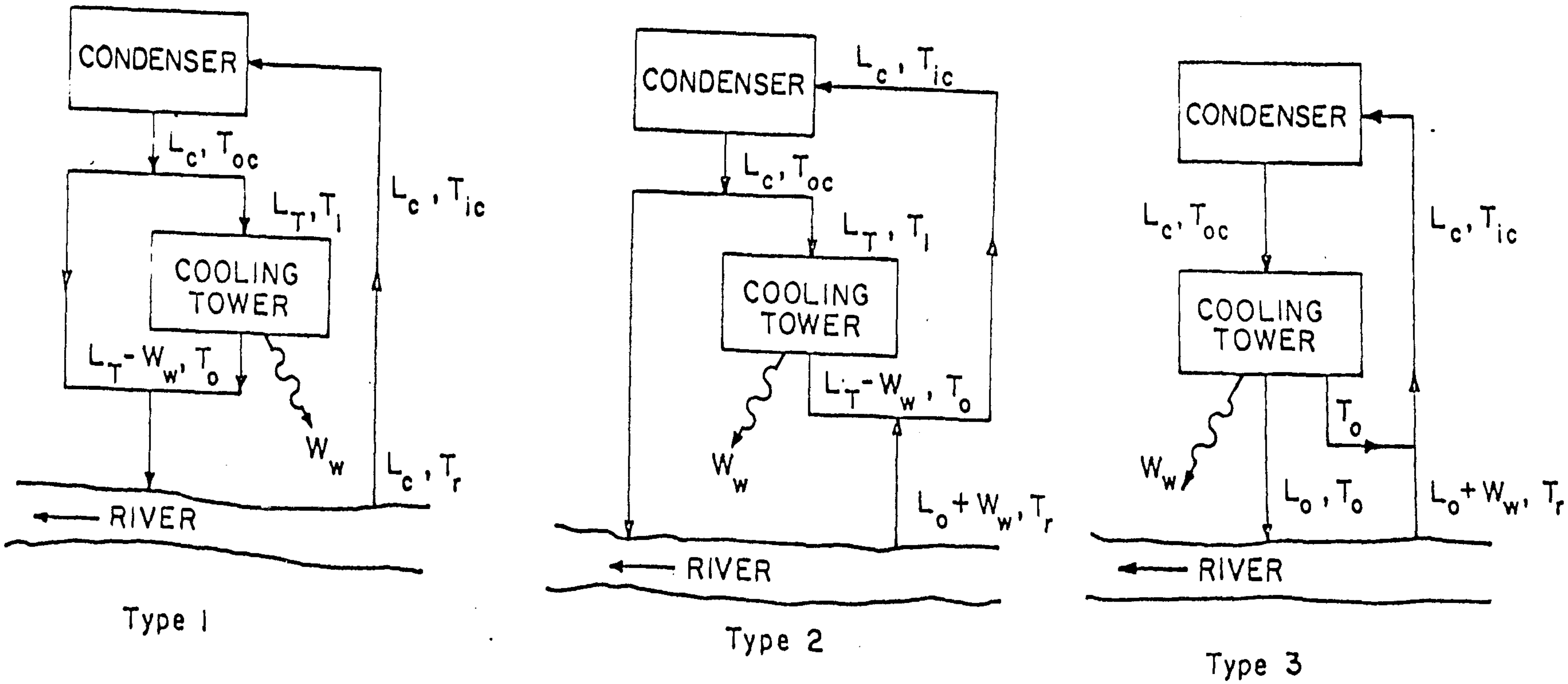

Figure 1. Wet Tower/Once-Through Hybrid Cooling Systems

arrangement in which part of the heated condenser effluent is discharged directly into the river, while the rest passes through the cooling tower and then is discharged into the river. Type 2 contains a closed-cycle loop in which part of the condenser effluent is not discharged directly to the river but is passed through the cooling tower and recirculated through the condenser. In the Type 3 arrangement all of the condenser effluent passes through the cooling tower. All of the cooling water from the tower either can be recirculated through the condenser (referred to as closed-cycle operation), discharged into the river (referred to as open-cycle operation), or divided so that part of the cooling water is recirculated through the condenser and the rest discharged into the river.

It should be noted that for these hybrid cooling systems the amount of waste heat discharged into the river depends on the allowable river heat assimilation capacity, and the cooling water flow rate through the condenser does not necessarily equal the flow rate through the cooling tower. Also, the condenser inlet temperature does not equal the tower outlet temperature. The performance equations and the solution procedures for each of the hybrid cooling systems are discussed next.

Type 1 - The heat rejection rate from the condenser, Q_T, with a specified condenser flow rate, L_c, is given by

$$Q_T = L_c(T_{oc} - T_{ic})/k_1 = L_c \, \Delta t/k_1 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (7)$$

It can be seen from Fig. 1 that the temperature of the cold water entering the condenser is the same as the ambient river temperature. The heat rejection rate from the condenser equals the heat rejected from the turbine. For a given turbine exhaust steam temperature, T_s, the condenser outlet water temperature, T_{oc}, is obtained from Eq. 5, and the heat rejection rate from the turbine can be found from the turbine-characteristics curve. By comparing the heat rejection rate obtained from Eq. 7 with that from the curve and by adjusting the value of exhaust temperature until they agree, the heat rejection rate and the turbine exhaust steam temperature can be obtained. Once these values are known, the tower outlet cold water temperature, T_o, can be calculated from the wet cooling tower heat balance.

Type 2 - For this arrangement, the heat rejection rate through the condenser also is given by Eq. 7. Referring to Fig. 1, the heat balance equation for this cooling system can be expressed as

$$L_c T_{ic} = (L_o + W_w)T_r + (L_T - W_w)T_o \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \quad (8)$$

or

$$T_{ic} = (L_T - W_w)(T_o - T_r)/L_c + T_r \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \quad (9)$$

in which L_o = flow rate of water discharged into the river and W_w = evaporative water loss from the cooling tower.

The procedure for calculating the values of Q_T, T_{ic}, T_o, and T_{oc} follows: 1) Given L_c, L_T, and T_r, obtain the heat rejection rate, Q_T, from the turbine characteristics curve for an assumed T_s. 2) Calculate T_{oc} and T_{ic} from Eqs. 5 and 7 using the given T_s and Q_T obtained from step 1. 3) Calculate T_o and Q_T from the wet cooling tower equation (5). 4) Calculate T_{ic} from Eq. 9 and compare with T_{ic} obtained in step 2. 5) Adjust T_s and repeat steps 1 through 4 until the two values of T_{ic} are approximately the same (an absolute difference of 0.1°F ≅ 0.06°C was used herein).

Type 3 - The heat rejection rate for the Type 3 system also is given by Eq. 7 since

$$T_{ic}L_c = (L_c - W_w - L_o)T_o + (L_o + W_w)T_r \quad \ldots\ldots (10)$$

and

$$L_o = \text{RHAC}\ (k_1)/(T_o - T_r) \quad \ldots\ldots (11)$$

The cold water temperature entering the condenser is

$$T_{ic} = T_o - W_w(T_o - T_r)/L_c - (\text{RHAC})\ k_1/L_c \quad \ldots\ldots (11)$$

It is seen from Fig. 1 that the quantity of cooling water discharged into the river depends upon the allowable river heat assimilation capacity. If the allowable river heat assimilation capacity is large enough that all the cooled water can be discharged into the river, Eq. 12 will no longer hold. In other words, if RHAC $\geq (L_c - W_w)(T_o - T_r)/k_1$, then $T_{ic} = T_r$.

The procedure for calculating the values of Q_T, T_{ic}, T_{oc}, and T_o is the same as for the Type 2 cooling system except for the following modifications: 1) Calculate T_{ic} using Eq. 12 instead of Eq. 9. 2) Check the inequality, RHAC $\geq (L_c - W_w)(T_o - T_r)/k_1$; if it is true, then set T_{ic} equal to T_r and repeat the steps.

It should be noted that the arrangement of a completely closed-cycle cooling system is a special case of the Type 2 and Type 3 systems wherein the river heat assimilation capacity (RHAC) is equal to zero, so no water is discharged to the river, and the water flow rate through the cooling tower, L_T, is equal to the condenser flow rate, L_c. This equivalence can be verified from Eqs. 9 and 12. If L_T equals L_c, Eq. 9 becomes

$$T_{ic} = (L_c - W_w)(T_o - T_r)/L_c + T_r$$

or

$$T_{ic} = T_o - W_w(T_o - T_r)/L_c \quad \ldots\ldots (13)$$

which is equivalent to Eq. 12 if RHAC equals zero.

OPTIMIZATION METHODOLOGY AND ASSUMPTIONS

The optimization problem can be stated as a minimization of the total cost of the power plant including the cooling system. Since the turbine and power plant costs are considered fixed for the cooling system optimization, the problem becomes the minimization of the toal cost of the cooling system only. In this study, the term "total cost" refers to the cooling system total cost. Total cost is normally broken down into capital costs and operating costs. Capital costs consist of all the expenses incurred at the time of construction of the power plant, such as the costs of land, cooling system equipment (wet tower structure, once-through system, condenser, pump and pipe systems), and replacement capacity which is defined as the capacity required to replace the energy loss (examined subsequently) occurring at extreme meteorological conditions. The operating costs consist of the costs of fuel, water, water treatment, maintenance, and replacement energy (the difference between the design output and the actual output plus the internal requirement for fans and pumps). Once the capital and operating costs have been determined for a given cooling system, the total annual costs can be obtained by adding the annual operating cost to the levelized annual equivalent capital cost (capital cost times the fixed charge rate). The fixed charge rate is based on a utility's experience and is a measure of the cost of money including the estimated effects of inflation, taxation, and depreciation.

In order to retain a certain amount of flexibility for broad application of this model, the use of available cost information is as general as possible; "unit costs" are introduced for calculating cost items. Procedures for estimating capital costs, operating costs, and total annual costs (excluding land and once-through cooling system costs) are available in other publications (1, 4). The cost of once-through cooling mainly depends on the water flow rate and the length of the cooling system intake and discharge structure. The empirical cost function for once-through cooling is adapted from private utility company experience (8). The additional land area required for the cooling tower depends upon the plan area of the system and upon considerations of interference with adjacent structures, plume recirculation, and fan noise. In the present study, additional land area is computed on the basis of a noise level limit, and the specific land area required is a 0.1 acre/MW (404.7 m^2/MW).

To establish the design, performance, and cost of the optimum cooling system, a computer program was developed by modifying the Iowa Institute of Hydraulic Research (IIHR) Model of Wet Cooling Tower Economics developed by Croley et al. (4). The computer program for wet tower/once-through hybrid cooling system analysis is documented elsewhere (3). The frequency-weighted operating cost is calculated for each month over selected meteorological and hydrothermal conditions: dry-bulb temperature, T_{db}, wet-bulb temperature, T_{wb}, river temperature, T_r, and river heat assimilation capacity, RHAC. The river temperature is calculated by using the Steady-State Iowa Thermal Regime Model (ITRM) (13). Temperature distributions are computed under the assumption of complete mixing of heat loads with river water. The annual operating cost is then obtained by summing all the weighted monthly operating costs.

In addition to the meteorological and hydrothermal conditions, operating costs and capital costs also depend on the design parameters for the condenser and cooling tower. The design parameters for sizing the condenser and wet tower include condenser flow rate, tower length, height, fill width, and water and air flow rate loadings. For the purposes of this investigation, the values of water and air flow rate loadings and fill width are taken as constants, representing popular design values. Thus, an optimum hybrid cooling system design in this study is characterized by the condenser flow rate, tower height, and tower length. During the computation procedure, a single parameter is varied over a practical/feasible range, while the two remaining variables are fixed. The search for the most economical arrangement of the hybrid cooling system is carried out for a range of condenser flow rates, tower lengths, and tower heights.

Since the characteristics of hybrid cooling systems are related to so many parameters, the following restrictions are made throughout this study. 1) The unit being considered is assumed to operate at full throttle except during the period of low permissible river heat assimilation capacity and high ambient air or water temperatures. 2) Tower lengths of the Type 1 and Type 2 cooling systems cannot be greater than the length which is consistent with the tower water loading (condenser flow rate divided by the product of the water loading and the fill width), and the tower length of the Type 3 system is fixed at that length. 3) The comparison of the total costs among the hybrid cooling systems and the closed-cycle wet tower cooling system is based on a zero unit water cost since the power plant is located along a river from which adequate make-up water is available. The importance of water usage is considered in a parametric analysis of the effect of unit water cost on total cost and optimum tower design for each hydbrid system, and in a trade-off between total costs and water consumption. 4) Evaporative water loss of the hybrid cooling system does not include river water evaporation resulting from river temperature increases. 5) Power plant cooling characteristics are evaluated from meteorological and hydrothermal conditions on a month-to-month basis; thus "annual" operating costs are calculated for each month (cost models are similar to those described in ref. 4), and then the weighted average is taken as the annual operating cost. 6) For the optimization study, the cooling towers are assumed to operate continuously.

Based on the thermodynamic models and the cost functions described previously, the optimization procedure is as follows: 1) find the heat rejection rate and related water temperatures of the cooling system by using the thermodynamics model for all sets of meteorological and hydrothermal conditions for each month; 2) calculate all capital costs; 3) calculate the frequency-weighted operating costs for all the meteorological and hydrothermal conditions for each month; 4) obtain the annual operating cost by summing the weighted monthly operating costs; 5) calculate the unit total cost using the fixed-charge-rate method; 6) plot total costs versus tower length for all sets of design parameters such as condenser flow rate and tower height; and 7) identify the optimum hybrid cooling system design by selecting the tower length, tower height, and condenser flow rate corresponding to the minimum total cost.

APPLICATION

The thermodynamic and economic models of wet tower/once-through hybrid cooling systems are applied to a hypothetical nuclear power plant located at Sioux City, Iowa, on the Missouri River where the heat assimilation capacity of the river is not able to accomodate once-through cooling when all the future plants at upstream sites are put into full-load operation. Case studies also were made at two other sites, namely, Council Bluffs, Iowa, and Fort Calhoun, Nebraska. Results obtained at the three sites are similar; therefore, the details presented herein refer only to Sioux City, Iowa. A comparison of results at the three sites is given by Hsu (6).

Since power plant and cooling system operation fluctuates daily, the frequencies of occurrence of the meteorological and hydrothermal parameters must be considered. The distributions of T_{db} and T_{wb} at the study sites are obtained from available climatic data. T_r and RHAC are calculated from river discharges and allowable temperature rises by using the steady-state ITRM. Data at three-day intervals over a twenty-year period (1955 - 1974), form a data base of 200 sets per month for each of the four parameters (6). Temperature design values (selected as 5 percent, exceedance values) are taken as T_{db} = 82.5°F, (28.1°C), T_{wb} = 68.1°F (20.1°C), T_r = 76.4°F (24.6°C). Since cooling system designs are based on low magnitudes of RHAC, the design value is set at the value exceeded 95 percent of the time during the year which is 2.7 x 10^9 Btu/hr (8.0 x 10^8 W). The extreme values of T_{db}, T_{wb}, and T_r are the maximum values of the given data sets. These values are 89.0 °F (31.7°C), 80.1°F (27.1°C), and 83.4°F (28.4°C), respectively. The extreme value of RHAC is the minimum value of the given data set which is 0.35 Btu/hr (1.0 x 10^8 W). The allowable river temperature rise at the site is 5°F (2.8°C). Other input data are: plant nameplate capacity

= 1,150 MW; maximum turbine back pressure = 4.5 in Hg (15.2 kpa); plant efficiency = 32 percent; water flow rate loading per unit plan area of tower fill = 12.5 gal/min/ft^2, (8.49 x 10^{-3} m^3/sec/m^2); air flow rate loading = 1,800 lb/hr/ft^2 (2.44 kg/sec/m^2); fill width = 36 ft (11 m); condenser heat transfer coefficient = 630 Btu/hr/ft^2/°F (3,580 W/m^2/°K).

In addition to the meteorological and hydrothermal conditions, the performance of the power plant and the cooling system also depends on the condenser flow rate, tower length, and tower height. These parameters are varied over a reasonable range to analyze each hybrid cooling system. The condenser flow rate ranges from 360,000 gal/min (22.7 m^3/sec) to 900,000 gal/min (56.8 m^3/sec) and the tower length ranges from 200 ft to 2,000 ft (61 m to 610 m). Three different tower heights, namely 35 ft, 45 ft, and 55 ft (10.7 m, 13.7 m, and 16.8 m) are considered. It should be noted that since the water flow rate loading in the wet tower is maintained at a constant value (12.5 gal/min/ft^2 or 8.49 x 10^{-3} m^3/sec/m^2), representative of current design practices, the tower lengths of the hybrid cooling systems are controlled as mentioned earlier.

The effects of meterological and hydrothermal conditions on hybrid cooling system performance are illustrated by the variation of energy loss shown in Fig. 2. It is seen that for the Type 1 and Type 2 hybrid coolings systems, energy losses are very high in the winter season (from October to March) due to the low river heat assimilation capacity associated with the low river flow rates. Low river heat assimilation capacity means less waste heat can be discharged into the river. Therefore, for a given tower size, power plant output levels must be reduced to cut down the heat rejection; hence, the energy loss is increased due to this reduced

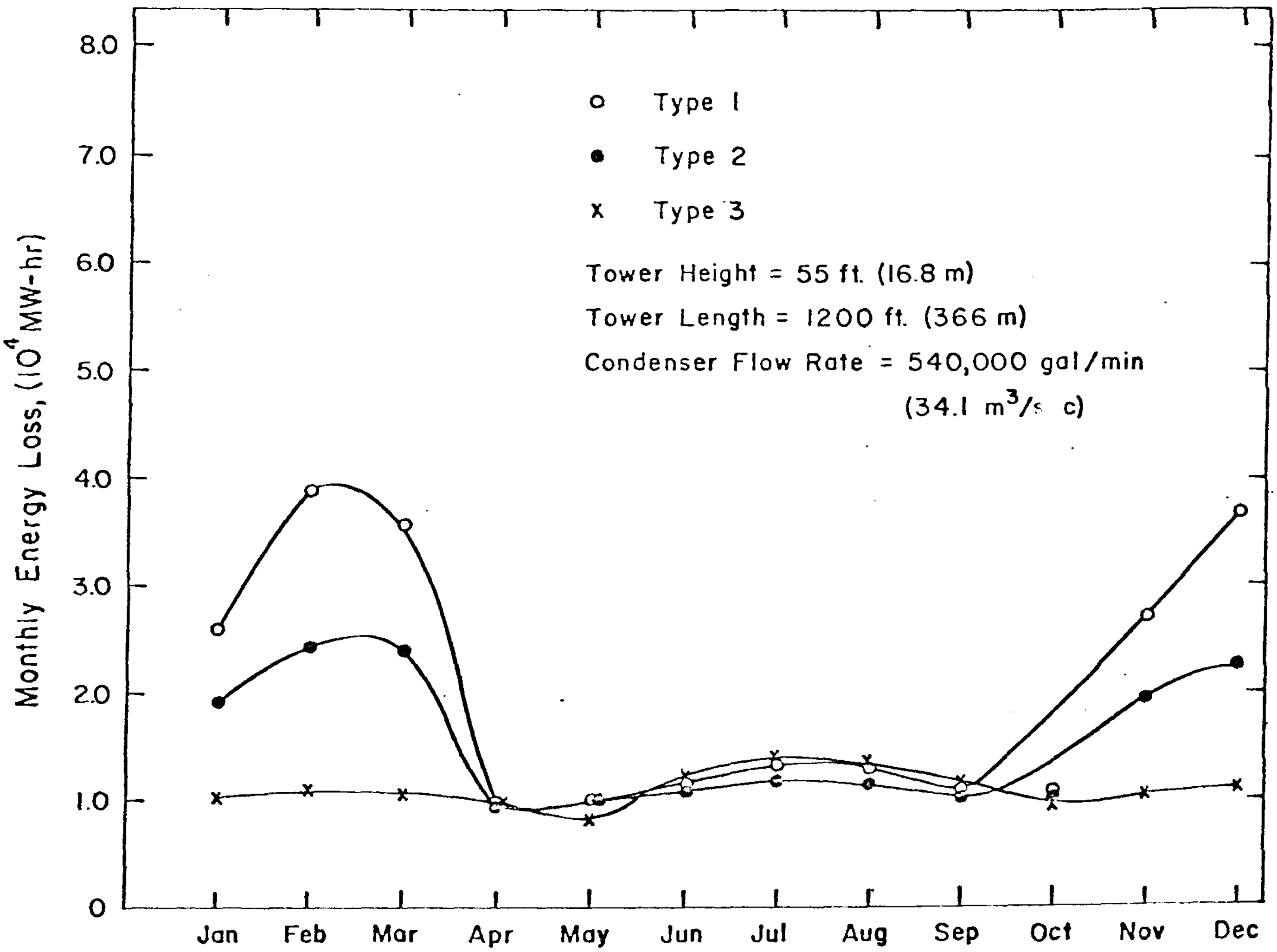

Figure 2. Monthly Energy Loss for Different Hybrid Cooling Systems

power output level. For the Type 3 hybrid cooling system, most of the heated condenser effluent is cooled by the tower and recirculated; only a small amount of waste heat is discharged to the river. This small amount of waste heat usually is less than the allowable river heat assimilation capacity except for very extreme conditions. Hence, energy loss and fuel consumption are small for this system as compared with other cooling systems.

As also noted earlier by Su (16), when the ambient air and water temperatures are low, e.g., in the winter, the low temperature of the water entering the condenser for the Type 1 cooling system results in a relatively low hot water temperature out of the condenser. This low hot water temperature leads to a low cooling range; hence, the amount of heat which is rejected through the cooling tower is low. The type 1 cooling system also suffers from a practical lower limit on the temperature of the water entering the condenser. If this temperature were too low, cooling system efficiency would decrease because of possible tower icing.

For the Type 2 hybrid cooling system, on the other hand, the condenser inlet and outlet water temperatures are driven up by the contribution of the tower exit water which is normally at a higher temperature than the ambient river water. In this case, the cooling range is larger than for the Type 1 system. More waste heat can be rejected; hence, the energy loss and fuel consumption are reduced. Therefore, the energy loss and fuel consumption for the Type 2 hybrid cooling system are less than for the Type 1 cooling system during the winter period, but both are still less energy efficient than the Type 3 system.

In the summer season (from April to September), more waste heat is rejected from the condenser due to the high turbine back pressure and cooling inefficiency which occur when the river temperature and the dry-bulb and wet-bulb temperatures are high. However, the river heat assimilation capacity is large enough to absorb most of the waste heat which reduces the energy loss and fuel consumption. For this period the energy loss of the Type 3 system is slightly greater than that of the Type 1 and Type 2 systems. The conclusions made from Fig. 2 are also valid for other tower sizes and condenser flow rates. From the above discussion, the Type 3 hybrid cooling system appears more attractive than the other two cooling systems for the site investigated herein.

Additional information on thermal performance of the hybrid cooling systems is illustrated in Fig. 3. It is seen that the Type 3 cooling system has the lowest energy losses for tower lengths equal to or greater than 800 ft (244 m). The energy loss of the Type 1 system is greater than that of the Type 2 cooling system because of the cooling inefficiency of the Type 1 system during the winter time as discussed previously. Therefore, the Type 3 hybrid cooling system is better than the Type 1 and Type 2 systems except for tower lengths less than 800 ft (244 m). Comparisons similar to Fig. 3 for other tower heights lead to the same general conclusions.

Other parameters which represent the thermal performance of hybrid cooling systems are turbine back pressure and cooling range. Turbine back pressure determines the plant efficiency; a high turbine back pressure implies low turbine efficiency and high capacity loss. Variations of these two parameters versus condenser flow rate at design temperatures are very similar and are shown for each hybrid cooling system in Fig. 4. It is seen

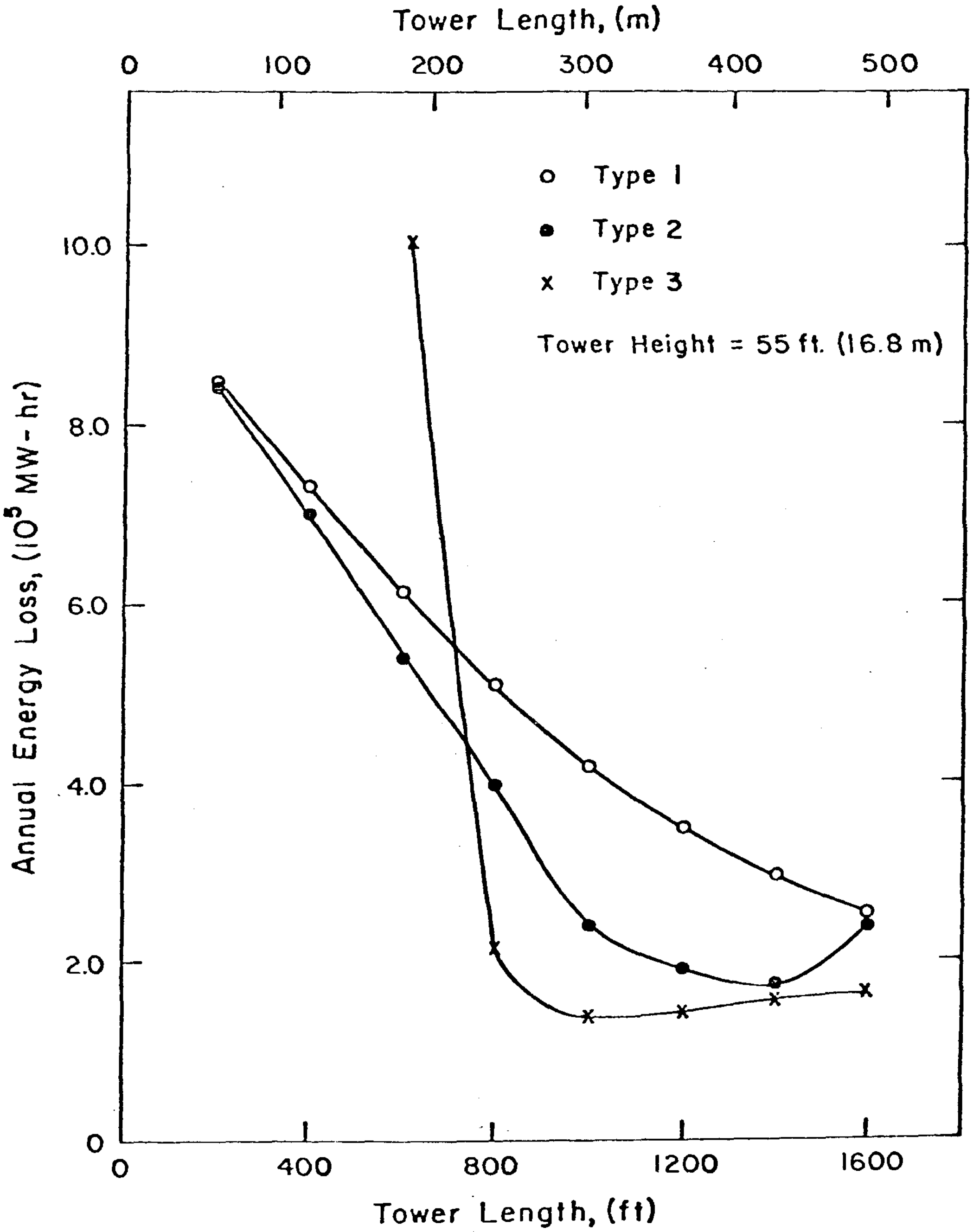

Figure 3. Annual Energy Loss for Different Hybrid Cooling Systems

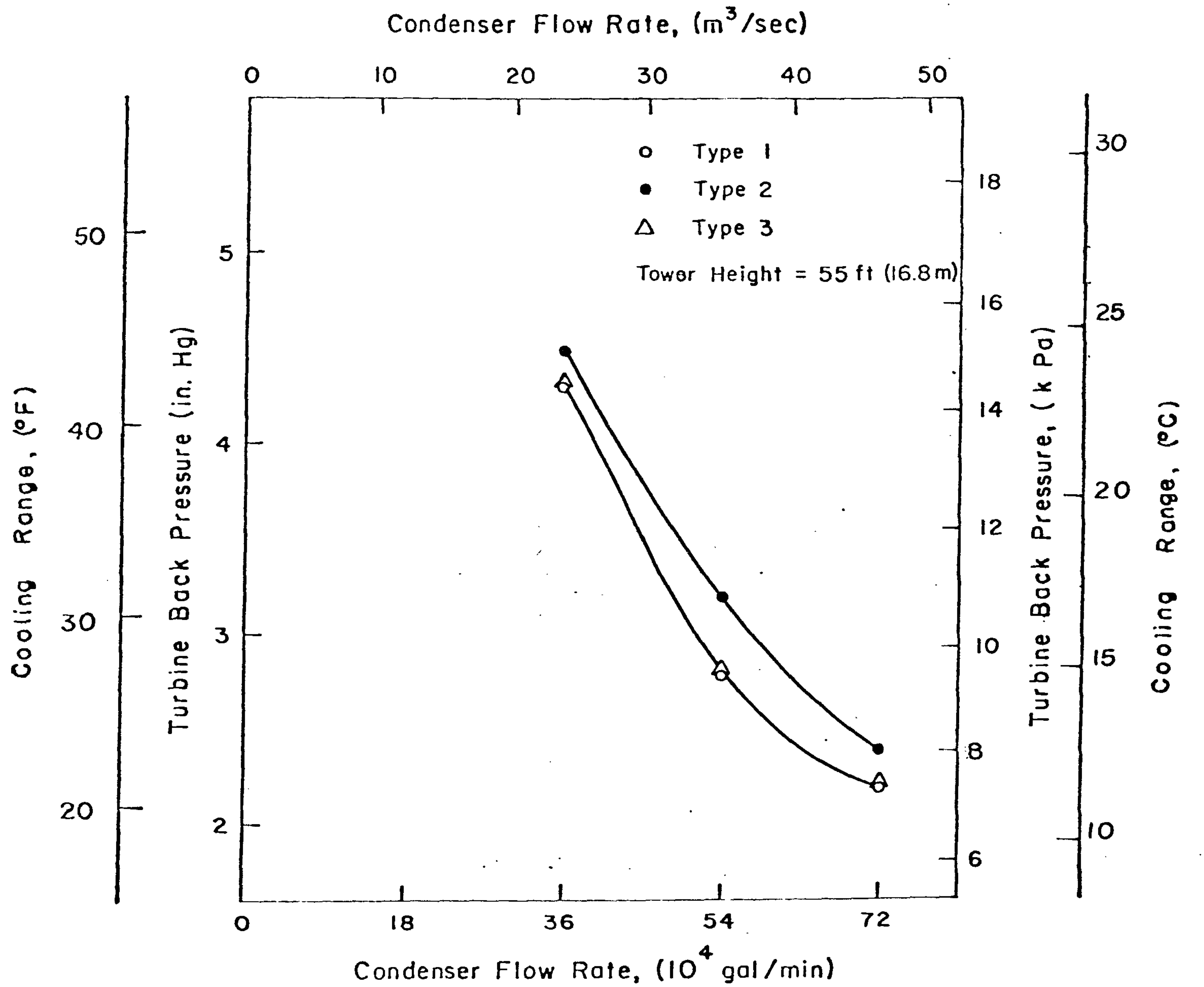

Figure 4. Turbine Back Pressure and Cooling Range at Design Temperatures for Different Hybrid Cooling Systems

that the turbine back pressure and cooling range decrease as condenser flow rate is increased. The condenser temperature rise and the turbine exhaust steam temperature decrease as the condenser flow rate increases, which lead to a lowering of turbine back pressure. For a fixed condenser flow rate, the Type 2 hybrid cooling system has a higher turbine back pressure and also a larger cooling range than the other hybrid systems. These properties tend to counteract each other since the first leads to a decrease of turbine efficiency but the second leads to an efficiency increase. The reduction of energy loss due to a larger cooling range may be greater than the increase of energy loss due to the high turbine back pressure.

It is very difficult to say which cooling system is best based on only the behavior of the turbine back pressure and cooling range at design temperatures. The economics of the system must be considered to determine system superiority.

From a practical point of view, a large cooling range is expected to decrease the total power plant cooling cost. However, if a large cooling range is obtained by reducing the condenser flow rate, the turbine back pressure will be increased which will reduce the plant efficiency, thereby increasing energy loss and fuel consumption. Hence, the choice of condenser water flow rate is very important in the design of hybrid cooling systems. The optimum design of condenser flow rate and tower size of the hybrid cooling systems can be obtained by an economic evaluation of the cooling systems. The models described previously can be used with auxiliary calculations to investigate the economics of different types of hybrid cooling systems and to determine the optimum wet tower design. Such an investigation is carried out in the following section.

ECONOMIC ANALYSIS AND OPTIMIZATION

The choice of the best hybrid cooling system arrangement at the study site depends on the results of a comparison of the total cooling system costs of each alternative. The assumed unit cost values for total cost computations are as follows: unit cooling tower cost = \$7.5/TU (TU = "tower units"); unit condenser cost = \$10.0/ft^2 (\$107.6/m^2); unit capacity loss cost = \$154,000/MW; unit fuel cost = \$0.000751/kW-hr; unit replacement energy cost = \$0.027/kW-hr; annual maintenance cost = \$200/tower cell; unit land cost = \$2,000/acre (\$0.49/m^2); and fixed charge rate = 0.147 for a new power plant operating with an expected life of 35 years.

Plots of total cost versus tower length for the different hybrid cooling systems are given in Figs. 5 through 7. The total costs of the Type 1 and Type 2 cooling systems decrease with increasing tower height because more heated water can be cooled with the increased tower height reducing the replacement energy cost. However, for a given tower length of the Type 3 cooling system, Fig. 7, the effect of cooling tower height is not significant, since the river absorbs most of the rejected heat.

The total costs of the hybrid cooling systems also depend on the tower length. For the Type 1 system, Fig. 5, the total cost decreases continuously as the tower length is increased up to the limiting value of tower length (determined by the tower water loading) since more heated water can be cooled with an increased tower length, and the energy loss is reduced.

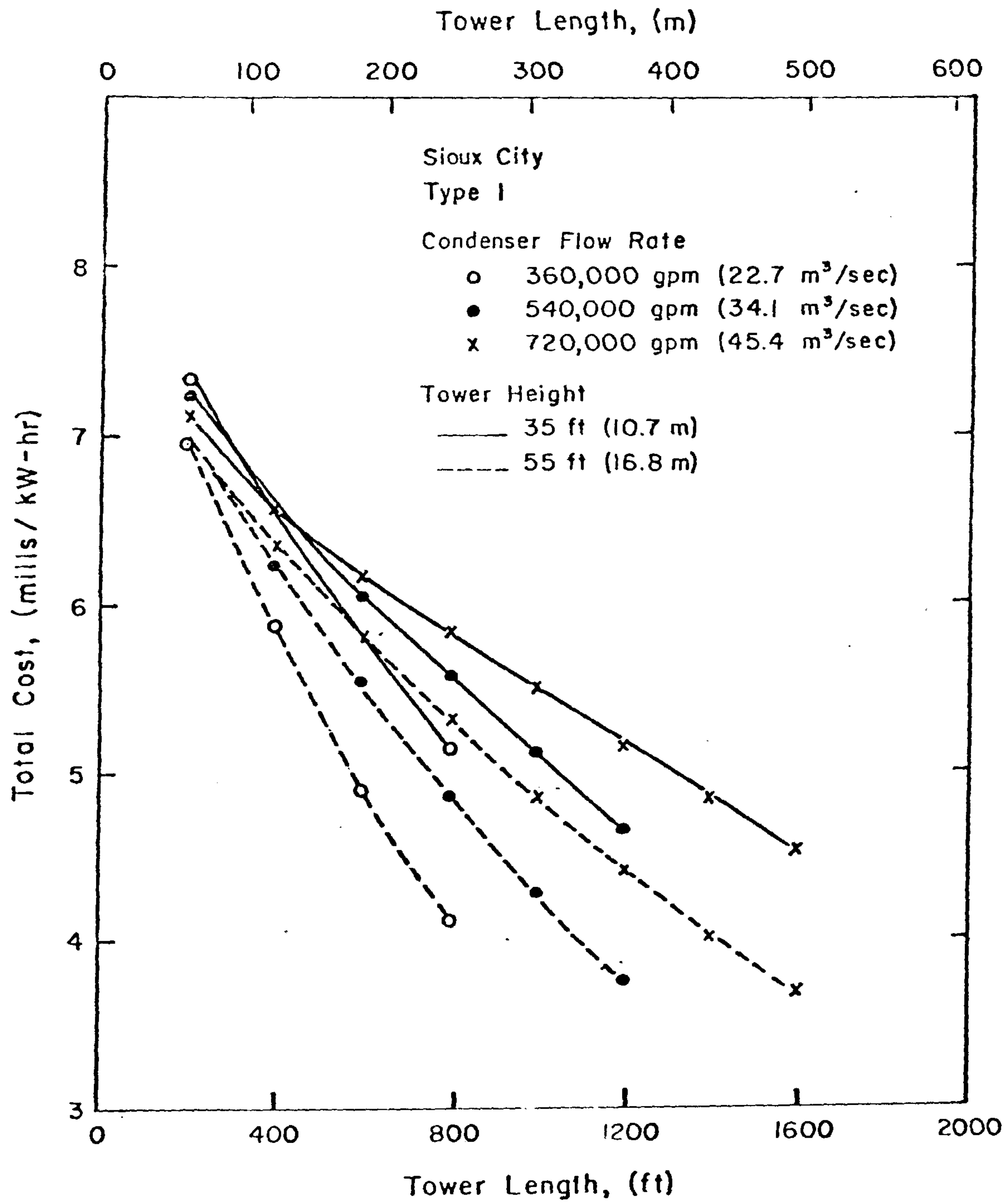

Figure 5. Total Cost of Type 1 Hybrid Cooling System

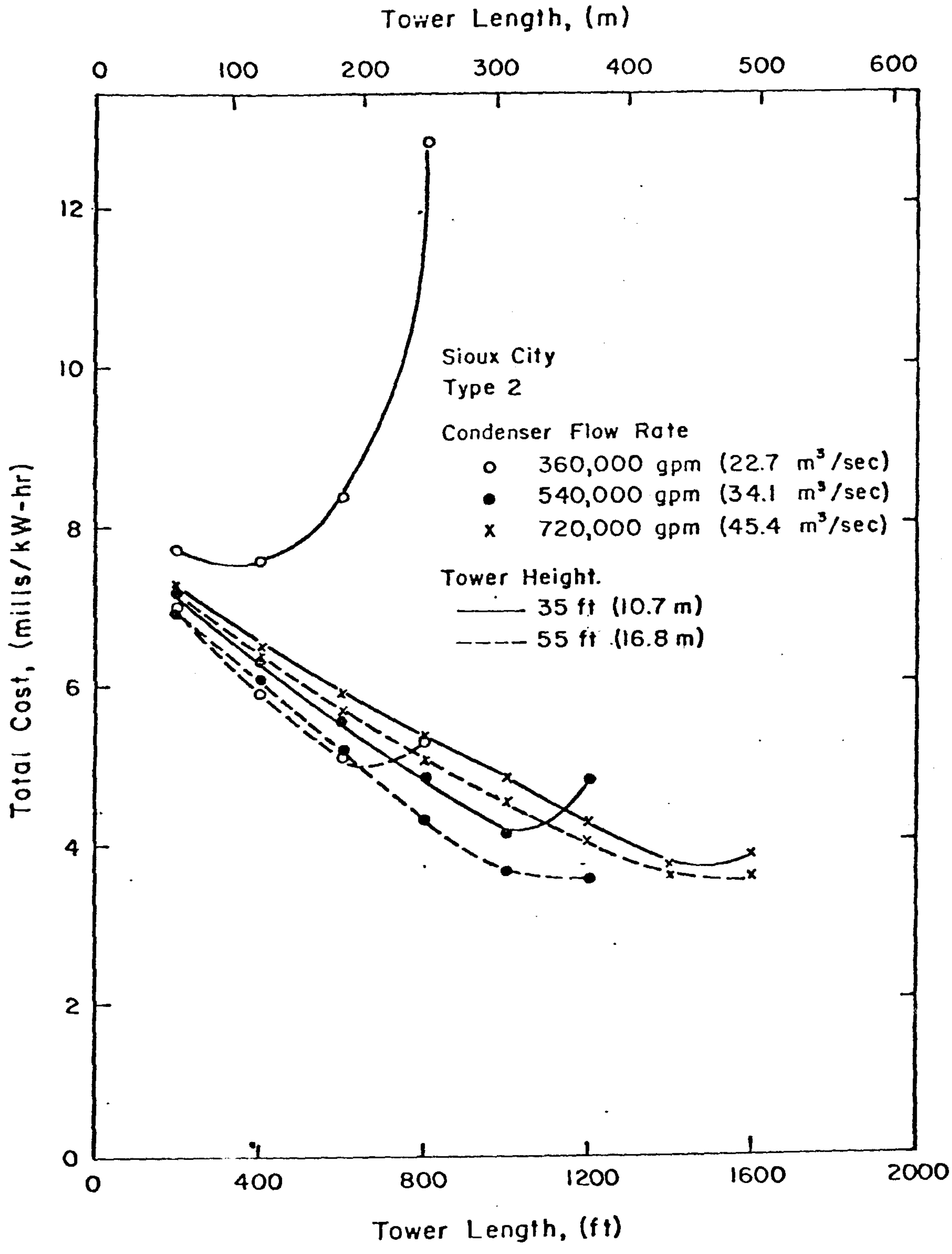

Figure 6. Total Cost of Type 2 Hybrid Cooling System

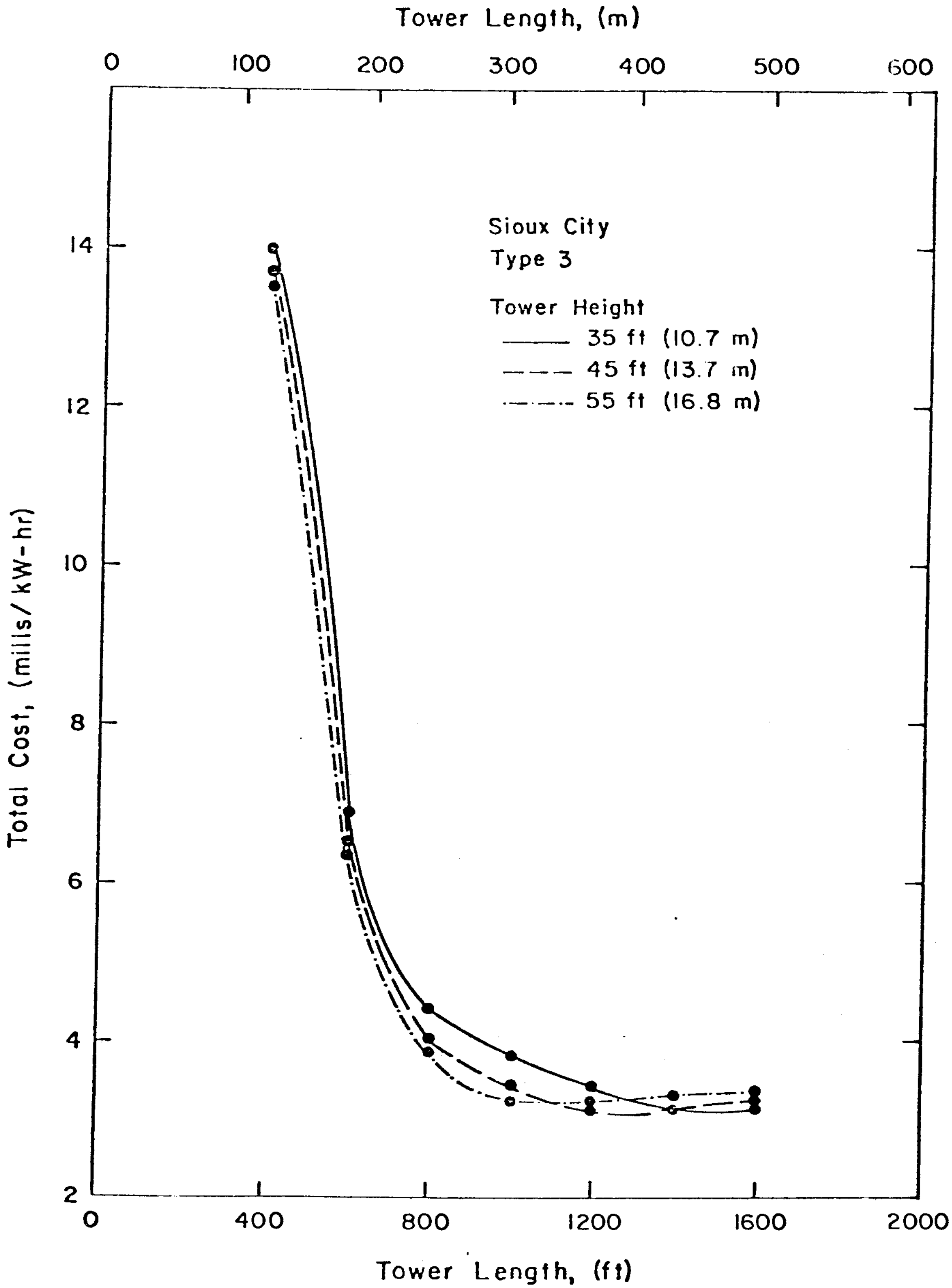

Figure 7. Total Cost of Type 3 Hybrid Cooling System

However, Fig. 5 seems to indicate that if the water loading were increased, allowing longer towers, the total cooling system costs could be further decreased, which is not necessarily true. For the Type 2 cooling system, Fig. 6, the total cost first decreases, reaching a minimum, then it increases as the tower size continues to increase because the reduction of replacement energy cost does not offset the incremental tower cost for large tower lengths. The total cost of the Type 3 system levels off at a tower length of about 800 ft (244 m) because of the significant contribution of the cost of replacement energy. This point is illustrated by reference to Fig. 3.

The optimum tower lengths for different condenser flow rates and tower heights and the corresponding minimum total costs are listed in table 1. The optimum design of the Type 1 cooling system is the case in which the design tower length is such that the toal water flow rate through the cooling tower equals the condenser flow rate. In this case, all the condenser effluent is first cooled by the cooling tower and then discharged into the river. This cooling system is the open-cycle cooling system described previously. The performance of this system is highly dependent on the river heat assimilation capacity. Because the tower length is limited by the water loading, the power plant must operate at a partial throttle opening during the winter to reduce the rejected heat when the river is not able to absorb the waste heat. Table 1 also shows that the minimum costs of the Type 1 and Type 2 systems for different condenser flow rates decrease as condenser flow rates increase. An increase of condenser flow rate results in a decreased turbine exhaust temperature and turbine back pressure. The lower turbine back pressure yields more efficient operation

Table 1

Minimum Total Cost, in mills/kW-hr, and Optimum
Tower Length* for Each Type of Hybrid Cooling System

Condenser Flow Rate	Type 1		Type 2	
	Tower Height			
(gal/min)	35 ft	55 ft	35 ft	55 ft
360,000	5.14 (800)	4.11 (800)	7.57 (400)	5.10 (600)
540,000	4.66 (1,200)	3.77 (1,200)	4.12 (1,000)	3.52 (1,200)
720,000	4.52 (1,600)	3.69 (1,600)	3.67 (1,400)	3.55 (1,600)
	Type 3			
	h = 35 ft	45 ft	55 ft	
540,000		3.17 (1,200)	3.22 (1,200)	
630,000	3.18 (1,400)			

*values within the parentheses are optimum tower length in feet

NOTE: 10,000 gal/min = 0.63 m^3/sec; 1 ft = 0.3048 m with less energy loss and a lower replacement energy cost. Since the minimum total cost is a function of condenser flow rate, the optimum design of the Type 1 and Type 2 systems are obtained by investigating all condenser flow rates. For the cases studied, the optimum tower length of the Type 1 system is about 1,600 ft (488 m) with a tower height of 55 ft (16.8 m) and a condenser flow rate of 720,000 gal/min (45.4 m^3/sec); the optimum tower length of the Type 2 system is about 1,200 ft (366 m) with a tower height of 55 ft (16.8 m) and a condenser flow rate of 540,000 gal/min (34.1 m^3/sec). The optimum tower length of the Type 3 cooling system is 1,200 ft (366 m) with a tower height of 45 ft (13.7 m) and a condenser flow rate of 540,000 cfs (34.1 m^3/sec).

Cost Comparisons - The main reason for using a hybrid cooling system is to take advantage of the available river heat assimilation capacity to minimize the total cooling system cost. Comparisons of total costs of hybrid cooling with closed-cycle and once-through cooling are necessary to judge the cost-effectiveness of hybrid systems. Also, the best arrangement of the once-through/wet tower hybrid system for various condenser flow rates is needed. Fig. 8 shows that the minimum total costs of the hybrid cooling systems are about 60 percent lower than a once-through cooling system. Hybrid cooling systems are more economical than closed-cycle cooling systems for condenser flow rates less than 450,000 gal/min (28.4 m^3/sec). The minimum total costs of the hybrid cooling systems are seen to approach each other as the condenser flow rate is increased. The figure also indicates that the Type 3 cooling system is the most economical one except for condenser flow rates less than 360,000 gal/min (22.7 m^3/sec). In that case, the cost of the Type 1 system is less.

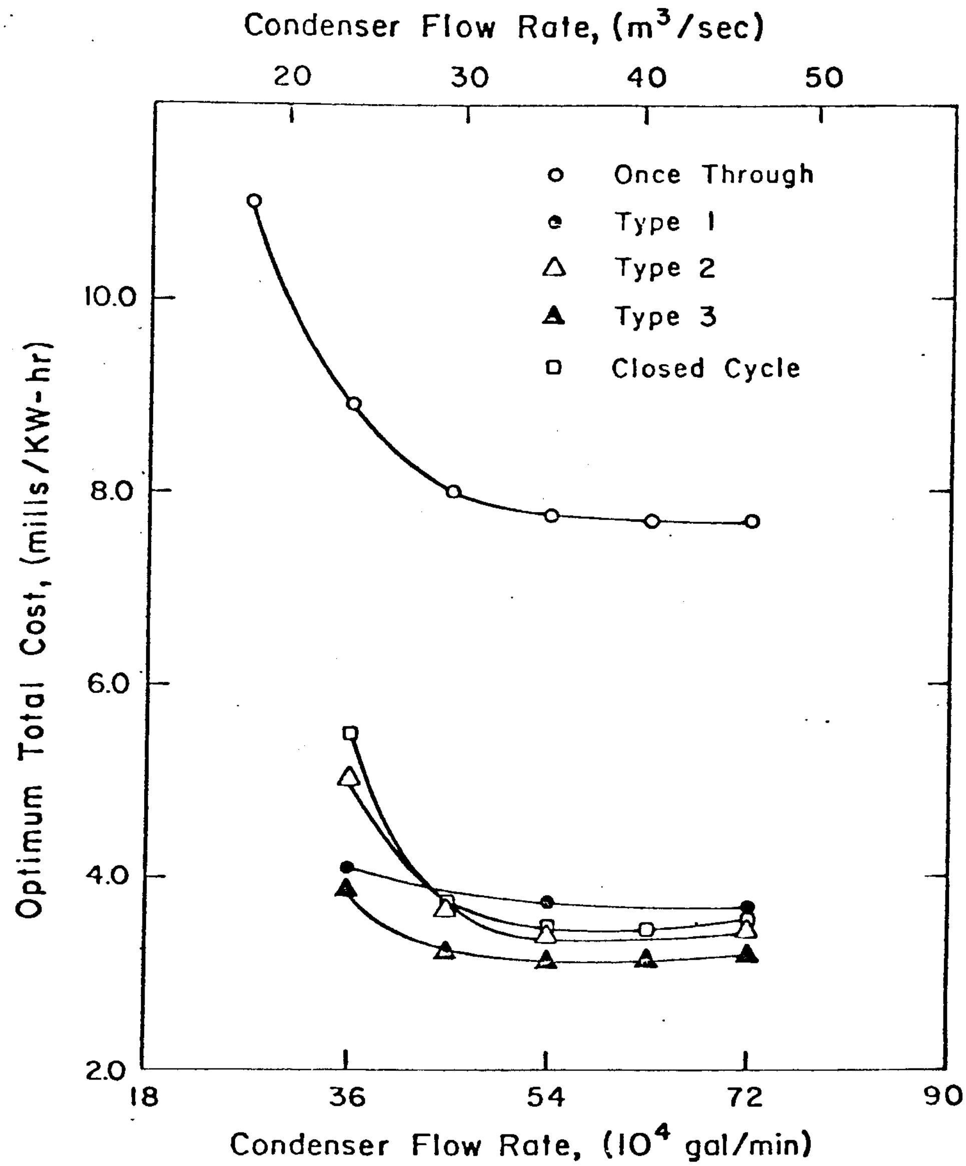

Figure 8. Optimum Total Cost of Hybrid Cooling Systems

Water Costs - It should be noted that the total costs of these hybrid cooling systems do not include the cost of make-up water because adequate water is assumed to be available from the river. However, the effect of the water cost can be very important in determining the optimum design. Table 2 illustrates the variation of total cost and optimum tower length for different unit water costs. It is indicated that for the Type 1 and Type 2 cooling systems with a condenser flow rate of 900,000 gal/min (56.8 m^3/sec), the optimum tower length will decrease as the unit water cost is increased because of the large evaporation rate at large tower lengths. Similar results are obtained for other condenser flow rates. For the Type 3 cooling system, the optimum tower length does not change because the water evaporation remains constant for tower lengths greater than about 800 ft (244 m); hence, the total cost will not be affected by the unit water cost, and the optimum tower length remains at 1,200 ft (366 m).

Water Consumption Trade-Off. Heretofore, it was assumed that there are no restrictions on water consumption. If the water consumption is limited by environmental regulations, optimum utilization of a hybrid cooling system will be different. The trade-off function between water evaporation and optimum total cost of the cooling system is shown in Fig. 9. This trade-off function was obtained by comparing evaporation and total cost of each type of cooling system as explained in refs 2 and 6. For water consumptions greater than about 10,000 acre-ft/yr ($1.23 \times 10^7 m^3$/yr), the Type 3 cooling system has the lowest cost. For water consumption less than 10,000 acre-ft/yr ($1.23 \times 10^7 m^3$/yr) the Type 1 cooling system is the most economical.

Table 2

Total Cost (mills/kW-hr) and Optimum Tower Length* (ft) for Different Unit Water Costs

Water Cost ($/100 gal)	Type 1 Condenser Flow Rate: 900,000 gal/min	Type 2 Condenser Flow Rate: 900,000 gal/min	Type 3
0.0	3.33 (2,000)	3.22 (1,800)	3.18 (1,200)
0.2	3.37 (2,000)	3.31 (1,800)	3.25 (1,200)
0.4	3.46 (2,000)	3.50 (1,800)	3.39 (1,200)
0.6	3.59 (2,000)	3.76 (1,600)	3.60 (1,200)
0.8	3.76 (2,000)	4.07 (1,600)	3.88 (1,200)
1.0	3.98 (2,000)	4.46 (1,600)	4.23 (1,200)
1.2	4.24 (2,000)	4.83 (1,100)	4.66 (1,200)
1.4	4.53 (1,800)	5.12 (800)	5.15 (1,200)
1.6	4.85 (1,800)	5.15 (200)	5.71 (1,200)
1.8	5.04 (200)	5.21 (200)	6.34 (1,200)
2.0	5.08 (200)	5.27 (200)	7.04 (1,200)

*values in parentheses are optimum tower length in feet.

NOTE: $1.0/1000 gal = $0.264/m^3; 10,000 gal/min = 0.63 m^3/sec; 1 ft = 0.3048 m.

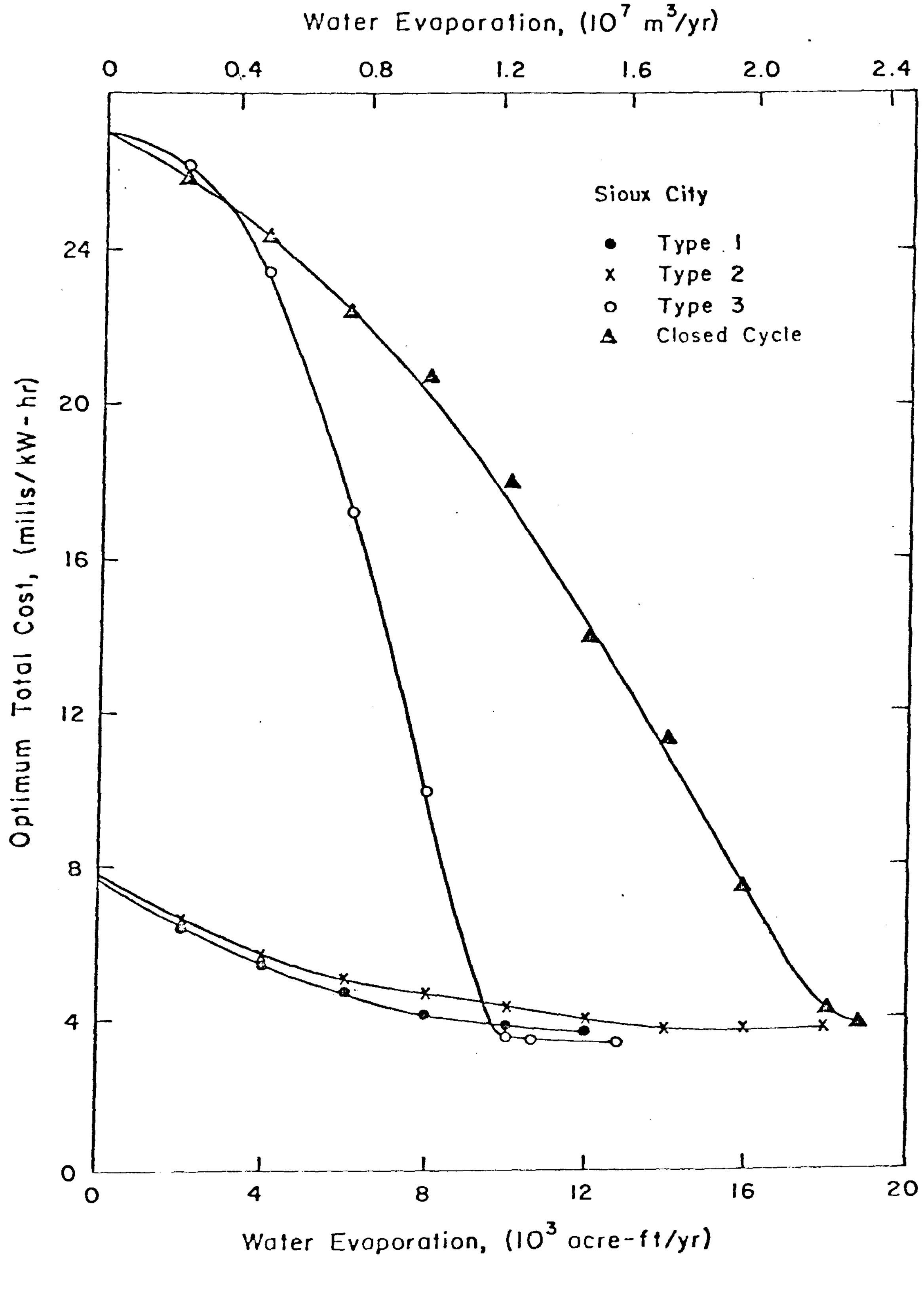

Figure 9. Trade-off Function between Optimum Total Cost and Water Evaporation

In addition to the case studies presented herein, other plant sites, including Fort Calhoun, Nebraska, and Council Bluffs, Iowa, also were investigated. The results from the three sites are compared by Hsu (6). All results are similar; therefore, conclusions and observations made previously are applicable for much of the Upper Missouri River downstream from Gavins Point Dam.

SUMMARY

Wet tower/once-through hybrid cooling systems are an acceptable method of managing waste heat from steam-electric power plants. The main idea is to use permissible river heat assimilation capacity to the extent possible with wet cooling towers supplementing once-through cooling at sites which have inadequate once-through cooling capacity.

A generalized computer model was developed to assess the thermodynamics and economics of hybrid cooling systems. Different hybrid cooling system arrangements, have been investigated. Application results indicate that performance of hybrid cooling systems depends upon the meteorological and hydrothermal conditions, especially the river heat assimilation capacity, the condenser flow rate, the size of the wet cooling tower, and the water consumption.

For different condenser flow rates, the optimum wet cooling tower size and the minimum total cost is given for the three arrangements of hybrid cooling systems studied. A comparison of once-through cooling with hybrid cooling systems indicates that the latter are economically superior for the specific sites investigated. This general conclusion applies for other meteorological and hydrothermal conditions where the permissible heat assimilation capacity of the ambient water body is inadequate for dissipating all of the power plant waste heat.

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The report reflects findings made by T.D. Hsu on research leading to his doctoral dissertation.

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PROJECT PUBLICATIONS

The following reports, theses, and papers have been published as a result of either partial or complete financial support from this project.

1. CROLEY, T.E. II, A.R. Giaquinta, R. M.-H. Lee, and T.-D. Hsu (1978), "Optimum combination of cooling alternatives for Steam-Electric Power Plants," IIHR Report No. 212, Iowa Institute of Hydraulic Research, The University of Iowa, Iowa City, Iowa.

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