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ECONOMIC ASSESSMENT OF BACKFITTING POWER PLANTS WITH CLOSED-CYCLE COOLING SYSTEMS

Industrial Environmental Research Laboratory
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CONTENTS

CONTENTS (continued)

FIGURES

FIGURES (continued)

FIGURES (continued)

FIGURES (continued)

FIGURES (continued)

FIGURES (continued)

FIGURES (continued)

TABLES

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SECTION I

CONCLUSIONS

The general methodology for the economic evaluation of backfitting power plants with closed-cycle cooling systems has been presented. The following major conclusions can be drawn from the study.

1. The computer programs developed here can be used to assess the total differential cost of backfitting power plants with any of the following closed-cycle systems:

   Mechanical-draft crossflow wet cooling towers,
   Natural-draft crossflow wet cooling towers,
   Cooling ponds, and
   Spray canals.

   The programs accept as input data turbine size and characteristics, size of cooling system, fluctuations in power demand, variations in site meteorological conditions, and economic parameters. The thermodynamic and performance models used to evaluate the operating consequences such as capacity and energy losses, excess fuel consumption and water requirements are representative of those in current use. The overall accuracy of the economic predictions therefore depends largely on accurate projection of the capital cost of the cooling system and on the unit costs associated with capacity replacement, make-up energy and fuel. While an effort has been made to incorporate the most recent estimates for capital costs of cooling systems, it is also recognized that these costs vary over a wide margin, particularly in the cases of cooling ponds and spray canals, due to unforeseen problems associated with construction at particular sites.

2. The overall complexity of the backfit analysis is such that it does not allow the use of simplified models and construction of simple nomographs for general use. Nevertheless, it has been possible to obtain a comprehensive set of results from selected model applications which will enable a rapid evaluation of the total cost of backfitting when some of the basic characteristics of the power plant, the cooling system, and the site are known. A considerable amount of generality has been achieved by (a) assuming that the power demand remains constant at the full-throttle value of the turbine; (b) presenting in a graphical form only the basic quantities such as capacity loss, energy loss, excess fuel consumption and water evaporation as functions of cooling-system size in a convenient format, so that these quantities can be evaluated for any given nameplate capacity of the turbine and size of the cooling system; (c) estimating the capital cost of the cooling system in terms of its physical size and not in terms of thermodynamic properties of the system; and (d) performing the calculations for three different types of turbines and for four climatically different sites. The presentation of the results in this form is particularly useful since the influence of using different values of the economic parameters, such as capital cost of cooling system, unit costs of replacement capacity and energy, etc., can be investigated with ease. The use of these results has been illustrated by means of a hypothetical example for each of the four closed-cycle cooling systems.

3. The actual performance of the affected power plant or unit in open-cycle operation can be investigated by modifying the basic procedures to include the variations in the water-body temperature. Such an alteration was not, however, considered in the present study since it would have resulted in a loss of generality which was considered essential for the presentation of the results in

simple graphical form.

4. The present study indicates that the total cost of backfitting, in mills/kW-hr of energy delivered, depends mainly on the capital cost of the cooling system, the capital cost associated with the replacement of lost capacity, the operating costs of peaking plants built to replace the energy losses, and the excess fuel consumption of the affected units due to higher back-pressure operation resulting from the backfit. Under the assumption of full-throttle power demand, however, the excess fuel consumption is much smaller than would actually result from a variable power demand, but there is a corresponding increase in the energy losses. Comparison between the detailed calculations using the computer programs with realistic power demand variations and those performed using the graphical results with full-throttle power suggest that the total costs calculated by the two methods may differ only by a few percent. This small difference gives added confidence in the use of the results presented graphically.

5. The sensitivity of the total cost of backfitting to the factors listed above suggests that great care must be taken in estimating the following for each application:

   Capital cost of the cooling system,
   Unit cost of replacement capacity,
   Unit cost of replacement energy,
   Unit cost of fuel, and
   Fixed charge rate.

   As indicated earlier, the first of these quantities can be found readily from the data presented in this report, but an on-going check must be maintained in order to ascertain the impact of inflation within the industry. Needless to say, the most reliable and up-to-date information can best be obtained directly from the manufacturers and construction engineers.

A considerable amount of variation in these costs can be expected from site to site, but for cooling towers the estimates presented here appear to be reliable within about 15 percent for applications up to 1980. The remaining factors listed above depend upon the particular utility situation. However, if capacity and energy losses are to be made up by means of gas-turbine peaking units, the unit costs will be of the order of \$100/kW for capacity replacement and 10 mills/kW-hr for energy replacement based upon 1975 estimates.

6. In the hypothetical examples analyzed in detail in the text, where the power plant characteristics as well as the site meteorlogical conditions were approximately the same for all four closed-cycle cooling systems, the total excess unit costs of backfitting (in mills/kW-hr) were found to be

Mechanical-draft crossflow wet cooling towers	:	0.582
Natural-draft crossflow wet cooling towers	:	0.916
Cooling ponds	:	0.666
Spray canals	:	0.694

These costs can not, of course, be compared on an absolute scale since the size of each cooling system was chosen arbitrarily, and no attempt was made to verify an "optimum" size. Nevertheless, since each size is realistic and since the same values of the unit costs of replacement capacity, replacement energy, fuel, water, and fixed charge rate have been used, the total costs listed above give a good general indication for each type of cooling system.

7. The computer programs as well as the graphical results of the present study can be used to make an independent assessment of the cost of backfitting a given power plant or unit at a known site with a range of sizes of the four different types of closed-cycle cooling systems.

8. Finally, the assessment of backfitting costs can best be made for any particular situation by using the computer models given in the appendices. No interpolation error will be involved, and inclusion of the actual design power loading for the specific situation may be made (instead of the assumed "full-throttle" design loading implicit in the figures of this report). The programs are designed to accomodate a design loading composed of two power levels (one of which is the maximum) and can be easily extended to more.

SECTION II

RECOMMENDATIONS

Four major recommendations are made on the basis of the present study:

1. As far as possible, the computer programs presented in this report should be used to independently assess the economic consequences of backfitting power plants with closed-cycle cooling systems and to compare the estimated total costs with those evaluated by other methods. Rapid estimates of backfitting costs can also be made by using the representative set of graphical results, which will, however, involve a certain amount of approximation.

2. The utilization of the computer programs allows the economic analysis of backfitting to be based upon the actual design power loading instead of the full-throttle loading assumed in the graphical results. Design power loadings with more than two, defined levels of power output can be analyzed with slight modifications of the program. The actual performance of the affected unit during open-cycle operation can also be modeled by incorporating expected variations in the water-body temperature in the program.

3. The validity of any economic analysis of cooling systems will depend upon the proper selection of individual cost factors and constraints. Among the more important site-specific considerations is the requirement for land and its availability. While the example problems for cooling towers in this report have land

requirements based on noise attenuation, other criteria such as construction area and plume recirculation may be more applicable.

4. An on-going survey should be maintained to determine the prevailing and expected costs of cooling systems, unit costs of replacement capacity, replacement energy, fuel, and water, as well as the fixed charge rate, so that the general methodology developed for this study can be updated periodically.

SECTION III

INTRODUCTION

Following the recent enactment of "environmental" legislation (The Federal Water Pollution Control Act Amendments of 1972), the Environmental Protection Agency has been charged with the task of developing guidelines and standards of performance for steam electric power plants. Originally, the EPA's proposed §304 guidelines and §306 standards [1,2,3,4] suggested that, except for the power plants receiving exemption under §316(a), all plants operating with open-cycle cooling systems should be backfitted with closed-cycle systems by the year 1983. However, these guidelines soon met with much opposition, and in ensuing adversary hearings, a set of revised guidelines were constructed [5]. In accordance with these new EPA guidelines, the thermal discharges are to be limited according to the following schedule [6:§423.13 ℓ(1)-(6),m]:

A. There shall be no discharge of heat from the main condensers except

(1) "Heat may be discharged in blowdown from recirculated cooling water systems provided the temperature at which the blowdown is discharged does not exceed at any time the lowest temperature of recirculating cooling water prior to the addition of the make-up water.

(2) "Heat may be discharged in blowdown from recirculated cooling water systems which have been designed to discharge blowdown water at a temperature above the lowest temperature of recirculated cooling water prior to the addition of make-up water providing such recirculating cooling systems have been placed in operation or are under construction prior to the effective date of this regulation (July 1, 1981).

(3) "Heat may be discharged where the owner or operator of a unit otherwise subject to this limitation can demonstrate

that a cooling pond or cooling lake is used or is under construction as of the effective date of this regulation to cool recirculated cooling water before it is recirculated to the main condensers.

(4) "Heat may be discharged where the owner or operator of a unit otherwise subject to this limitation can demonstrate that sufficient land for the construction and operation of mechanical draft evaporative cooling towers is not available (after consideration of alternate land use assignments) on the premises or on adjoining property under the ownership or control of the owner or operator as of March 4, 1974, and that no alternate recirculating cooling system is practicable.

(5) "Heat may be discharged where the owner or operator of a unit otherwise subject to this limitation can demonstrate that the total dissolved solids concentration in blowdown exceeds 30,000 mg/l and land not owned or controlled by the owner or operator as of March 4, 1974, is located within 150 meters (500 feet) in the prevailing downwind direction of every practicable location for mechanical draft cooling towers and that no alternate recirculating cooling system is practicable.

(6) "Heat may be discharged where the owner or operator of a unit otherwise subject to this limitation can demonstrate to the regional administrator or State, if the State has NPDES permit issuing authority, that the plume which must necessarily emit from a cooling tower would cause a substantial hazard to commercial aviation and that no alternate recirculated cooling water system is practicable. In making such demonstration to the regional administrator or State the owner or operator of such unit must include a finding by the Federal Aviation Administration that the visible plume emitted from a well-operated cooling tower would in fact cause a substantial hazard to commercial aviation in the vicinity of a major commercial airport.

(m) "The limitation of paragraph (1) of this section shall become effective on July 1, 1981.

B. These new guidelines shall have both the exclusions implicit in the above paragraphs and the additional exclusions outlined here:

units on line before January 1, 1970, are excluded;
units of 500MW or less on line before January 1, 1974, are excluded;
units of less than 25MW are excluded; and
units in a system of 150MW or less are excluded.

The above guidelines are to be effective July 1, 1981, as indicated in paragraph (m) above; however, there are provisions of deferral of compliance until July 1, 1983, if system reliability would be seriously affected.

The present study is concerned with the development of a detailed methodology for the evaluation of the cost of backfitting a plant or unit currently operating on open-cycle with a closed-cycle cooling system. Four different closed-cycle systems are considered:

Mechanical-Draft Crossflow Wet Cooling Towers,
Natural-Draft Crossflow Wet Cooling Towers,
Cooling Ponds, and
Spray Canals.

It is recognized that a large number of conflicting factors enter into the estimation of the cost of backfitting. Since many of these are highly site-dependent, it is not possible to arrive at general conclusions applicable to all utility situations. However, the purpose here has been to develop a method which is flexible enough to take these factors into consideration so that when they are prescribed or determined the cost can be estimated.

The evaluation of the additional costs against the power generated is important to the utility since it provides a basis for determining the necessary rate increases. Of major concern in the backfitting operation is the fact that the capacity of the unit will be reduced by the amount of power consumed within the closed-cycle system and by penalties that may be incurred by requiring adjustments in the operating characteristics of the unit, the main factor being the increase in the turbine exhaust pressure. This lost capacity must be replaced either by adding new capacity at the same site or elsewhere, or by operating other units at higher levels.

The major factors to be considered in the economic assessment of backfitting an existing unit are:

1. The cost of installing the closed-cycle system, including materials, labor, site acquisition and preparation;
2. The plant downtime for hook-up and testing;
3. The provision of additional generating capacity to replace the lost capacity;

4. Operation and maintenance costs of the cooling system;
5. Operation and maintenance costs of replacement capacity; and
6. Additional cost of power generation due to decrease in plant efficiency or limitations occasioned by the use of the closed-cycle system.

It will be clear that the first three of these are capital costs incurred at the time of backfitting while the last three are costs recurring over the remaining period of plant life. When these factors have been determined and the cost of borrowing the required capital expenditure are known, it is a simple matter to find the total cost, in mills per kilowatt-hour, to be charged against the actual power delivered after the backfit operation. The work described herein is concerned primarily with the evaluation of the various factors listed above. It is of course possible to design a closed-cycle cooling system, regardless of whether it is a cooling tower, pond, or spray canal, which is sufficiently large to reproduce, very nearly, the performance of the once-through system being used at present. Such a system will obviously be expensive, at least from a first-cost point of view, and the various factors enumerated above will undoubtedly intervene and dictate a somewhat smaller closed-cycle system, requiring less operating and maintenance expenses. If the cost of backfitting is to be assessed in a realistic manner, it is then obvious that a range of sizes must be considered.

SECTION IV

GENERAL CONSIDERATIONS AND ASSUMPTIONS

Perhaps the most important question that needs to be considered immediately is: What are the characteristics of the power plant or unit which should be known before a backfitting study can be undertaken? Most of these are obvious: the nameplate capacity of the affected unit, in megawatts; the type of unit i.e., fossil or nuclear; the thermodynamic characteristics of the existing turbine and condenser system; the variations in the stream or water-body temperatures in open-cycle operation; the power demand history; and the general economic situation of the particular utility operating the unit. Since the intent here is to develop a general approach, it is necessary to make certain simplifications and assumptions concerning some of these variables and, at the same time, incorporate some flexibility which allows adjustments to be made in order to consider particular units. The following restrictions are therefore made throughout this work:

(1) It is assumed that the power plant or unit operates at full throttle, when possible, throughout the year, to satisfy a constant demand for nameplate capacity, except during scheduled or unscheduled outages. (Although this loading pattern is rarely realized in practice, it is used herein in lieu of consideration of all possible loading patterns, which is impractical. Consequences of this assumed loading pattern are examined in qualitative terms throughout this report).

(2) With the existing open-cycle cooling system, the plant or unit is considered to operate with an "equivalent" constant,

relatively low, turbine back pressure and that the corresponding heat rejection rate is known.

(3) The existing condensers may be retained without modification or new condensers (compatible with the new cooling system) may be installed, but their performance is similar to currently available equipment.

(4) The detailed thermodynamic characteristics of the affected turbine-generator units are known.

(5) The net power available for sale must be the nameplate capacity both before and after backfitting; any losses will have to made-up in some manner.

The first assumption implies that a base-loaded unit is being considered. The actual fluctuations in the power demand, which vary widely from utility to utility, are therefore neglected in the first instance. It would be difficult to correct results, obtained under the full-throttle assumption, to represent output demand loadings of less than nameplate capacity. However, it is expected that relative comparison of cooling systems (both open-cycle and the various closed-cycle systems), when made for the same loading pattern, are generally relevant. Therefore, the general results presented herein apply only for the full-throttle loading pattern for constant demand of nameplate capacity. Any other loading pattern may be evaluated directly with the computer models and an example of a variable loading pattern is presented in Section V. J.

The second assumption enables the establishment of a reference base in the cost estimation of closed-cycle cooling systems and their performance in comparison with the existing open-cycle system. In reality, the power plant will operate under a variety of conditions depending upon the daily as well as seasonal variations in the temperature of the stream or water body, but here it is assumed that the original open-cycle system is designed such that the turbine back pressure can be maintained at the relatively low levels where the turbine heat rates are nearly independent of back pressure. This particular assumption is not unduly restrictive. Again, both the variability of actual closed-cycle system performance and the intent to give some generally applicable results require that this assumption be made. To consider the effects of actual open-cycle cooling system performance, it would be necessary to incorporate actual data for the particular unit under consideration into the analysis. The differential costs presented here would have to be modified accordingly to allow this inclusion.

The third assumption reflects the practicality of the backfit situation since it may not be possible to consider major modifications in the condenser system. Herein lies a difficulty since closed-cycle cooling systems are usually designed and optimized in conjunction with the condenser design. Thus, some allowances must be made if the existing condensers are to be retained. In order to ensure compatibility between the new closed-cycle cooling system and the old condensers, it may therefore be necessary to impose certain restrictions on the design of the new equipment. These restrictions may take the form of constraints on the allowable temperature rise (cooling range) across the condenser and the allowable water flow rates through the system. The alternative of designing new condensers and salvaging the old ones may also be considered, without the aforementioned constraints.

The fourth assumption listed above is, of course, essential if the thermodynamic and economic consequences of backfitting an existing unit are to be evaluated realistically. The fifth assumption is made to conform with the first and with most practical situations. It may simply not be possible to deliver any capacity loss power, thus incurring a "loss cost." The economic loss incurred could then come from loss of revenue, cost of contract renegotiation and consequent penalties, etc. These losses are difficult to determine from a general point of view. However, the cost of making up lost power from other sources or of building auxiliary capacity can be estimated. Furthermore, making up capacity losses are expected to be the course of action most utilities will be required to follow. The capacity losses are assumed to be made up from other sources or from building the required capacity as just mentioned. They could also be made up by increasing the capacity of the baseload additions for the <u>system</u>. In this case, the additional revenue from operation at off peak periods would also have to be taken into account. Such possibilities are not explicitly considered in the present study although the economics could be easily adjusted to account for them.

Some of the essential elements of the basic methodology adopted throughout this work are described in the remainder of this section. Included here are all aspects of backfitting which are common to all types of closed-cycle cooling systems. The subsequent sections deal with the four different closed-cycle systems which constitute some of the available alternatives for backfitting.

A. TURBINE CHARACTERISTICS

Turbine performance is usually described in the form of a plot of the relative fractional change Δ [$= \Delta T_{HR}/T^*_{HR}$], which is the change (ΔT_{HR}, increase or decrease) in the turbine heat rate over some fixed reference heat rate T^*_{HR}, versus the back pressure, p. A typical set of turbine characteristics is shown in Figure 1 for a low back pressure loaded turbine. This figure and some basic concepts will be used to define a number of quantities which assume great importance, particularly in backfit considerations.

First of all, the point labelled R corresponds to the fixed reference heat rate T^*_{HR} and the reference back pressure p^* ($\Delta T_{HR} = 0$) at full-throttle, or valve-wide-open (VWO) operation. At this point (for VWO) power output is equal to the NAMEPLATE CAPACITY P^* of the turbine. The corresponding heat rejection rate (from the turbine) Q^* may then be obtained from the definition:

$$\text{Turbine Heat Rate} = \frac{\text{Heat Input (to the turbine)}}{\text{Power Output}} \qquad (1)$$

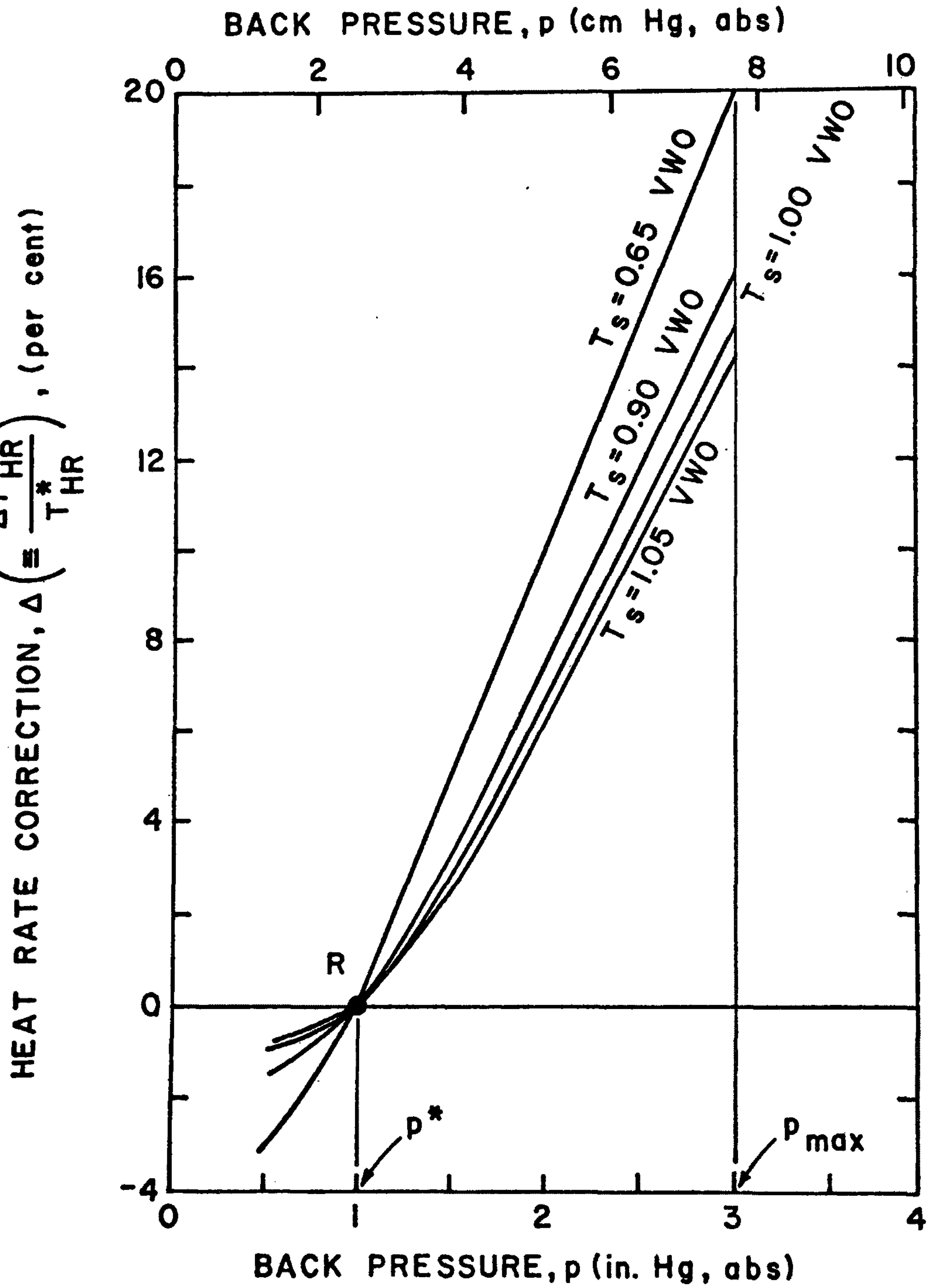

Figure 1. Typical turbine heat rate correction curves

i.e.,

$$T_{HR}^* = \frac{cP^* + Q^*}{P^*}, \qquad \text{(VWO)}$$

or

$$cP^* + Q^* = T_{HR}^* P^*, \qquad \text{(VWO)} \tag{2}$$

where P^* = nameplate capacity, kW,

T_{HR}^* = reference turbine heat rate, Btu/kW-hr or kJ/kW-hr,

c = a conversion factor = 3.413×10^3 Btu/kW-hr = 3.601×10^3 kJ/kW-hr,

Q^* = reference heat rejection rate, Btu/hr or kJ/hr.

Secondly, each line in Figure 1 represents a constant steam throttle setting T_s(%VWO) expressed as a fraction of the full-throttle condition, and therefore each line corresponds to a constant rate of heat input to the turbine. Since the rate of turbine heat input is the sum of the power output and the heat rejection rate, we have in general:

$$cP + Q = \text{constant} = (cP^* + Q^*)T_s \tag{3}$$

where P = power output, kW,

Q = heat rejection rate, Btu/hr or kJ/hr,

T_s = throttle setting (T_s = 1 corresponding to VWO).

Thus, from equations (1), (2), and (3) we have:

$$T_{HR} P = cP + Q = (cP^* + Q^*)T_s = T_{HR}^* P^* T_s \tag{4}$$

Now T_{HR} can be calculated from the turbine characteristics in Figure 1 as:

$$T_{HR} = T_{HR}^*(1 + \Delta)$$

Using this definition, equation (4) gives

$$P = \frac{P^* T_s}{1 + \Delta} \tag{5}$$

and

$$Q = T_{HR}^* P^* T_s - cP = T_{HR}^* P T_s - \frac{cP^* T_s}{1 + \Delta}$$

$$= Q^* T_s \left\{1 + \frac{cP^*}{Q^*} \frac{\Delta}{1 + \Delta}\right\} \tag{6}$$

These equations, together with Figure 1, show that as the back pressure rises above the reference value p^* the power output decreases and the heat rejection rate increases. Since there is usually a range of back pressures over which the excess heat rate is negligible or small, for simplicity it will be assumed that in open-cycle cooling the stream or water-body temperatures are such that the turbine back pressure is maintained in this range. In order to quantify the performance of the plant or unit to be backfitted and establish a reference point for the subsequent analyses of closed-cycle cooling systems, it will be assumed that the value of Q^*, hereafter referred to as the REFERENCE HEAT REJECTION RATE, and P^*, hereafter referred to as the NAMEPLATE POWER OUTPUT, are known. The knowledge of Q^* and P^* and the heat rate correction curves of the form shown in Figure 1 then enable the determination of the power output P and the heat rejection rate Q at any steam throttle setting (rate of heat input to the turbine) and the back pressure via equations (5) and (6). In particular, equation (6) can be used to construct the detailed heat rejection characteristics in the form shown in Figure 2. Since the heat rejection rate is an important parameter in the detailed analysis of the performance of closed-cycle cooling systems, the construction of such characteristics is an essential preliminary to the evaluation of the consequence of backfitting.

It will be noticed that the lines of constant throttle setting in Figures 1 and 2 terminate abruptly at some high value of the back pressure. For turbines of conventional design this upper limit is usually less than 5 inches (12.7 cm) Hg absolute. This value, denoted by p_{max}, is assumed to be the maximum allowable back pressure which, if exceeded, will result in some damage to the turbine or a catastrophic loss in performance. The upper horizontal line EDC in Figure 2 therefore constitutes one of the boundaries of possible operation of the turbine. The other boundary corresponds to the full-throttle line A'ABC, although most turbines will tolerate a certain amount of overload.

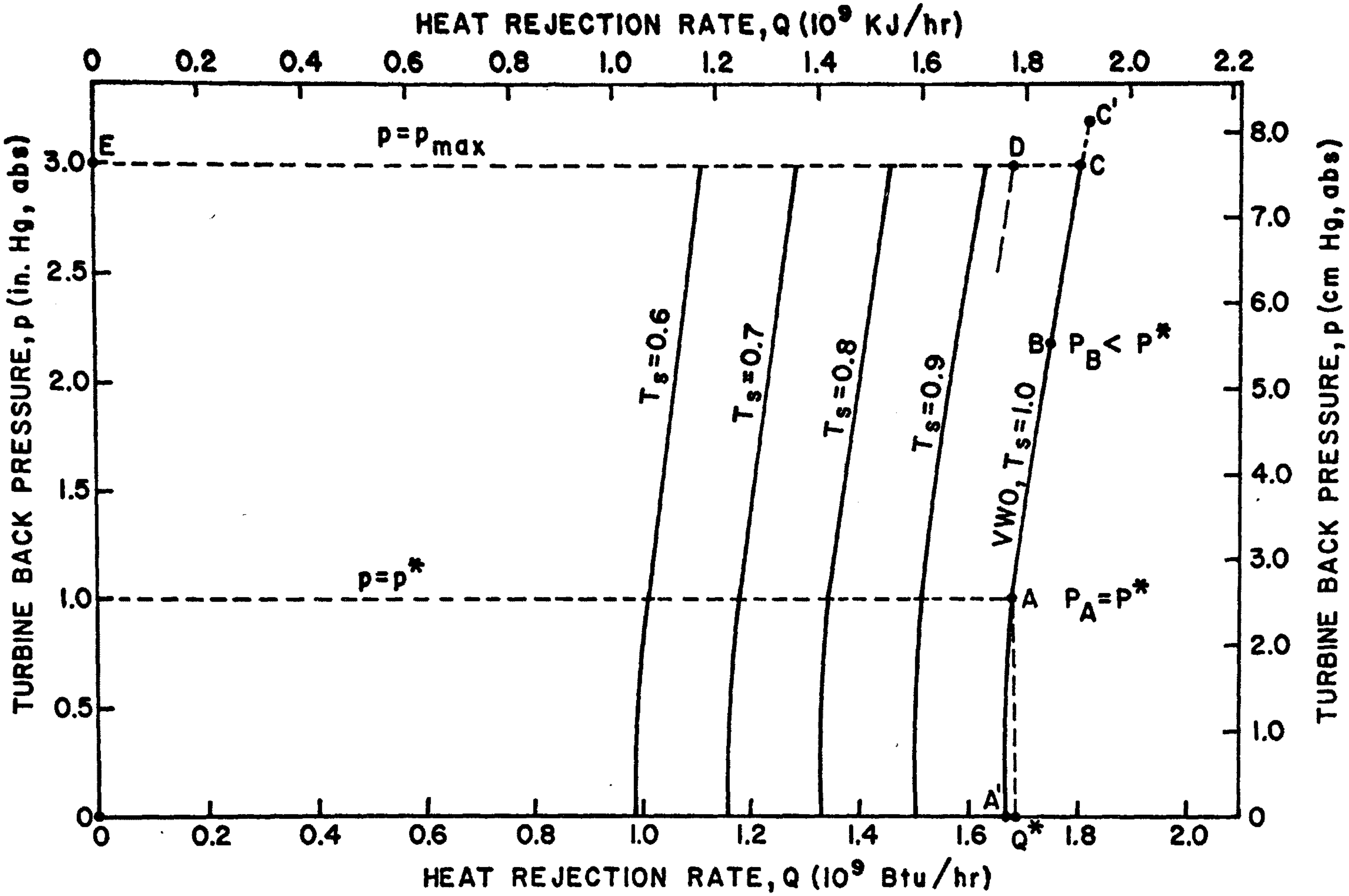

Figure 2. Typical turbine heat rejection rate characteristics

The reference heat rejection rate Q^* defined above can, of course, be found from equation (2) when the nameplate capacity and the reference turbine heat rate are known. Alternatively, it can be estimated from the overall plant heat rate in the following manner: The rate of heat input to the boiler (i.e., the heat equivalent of the fuel consumption) Q_T is given by

$$Q_T = cP + Q_{IP} + Q \tag{7}$$

and the thermal efficiency η_p of the plant (or plant efficiency) is defined by

$$\eta_p = \frac{cP}{Q_T} \equiv \frac{c}{P_{HR}} \tag{8}$$

where $P_{HR} \equiv Q_T/P \equiv c/\eta_p$ = plant heat rate, Btu/kW-hr or kJ/kW-hr.

Q_{IP} = in-plant and stack losses.

The in-plant losses are usually accounted for by an in-plant or steam supply efficiency η_I defined by

$$Q_{IP} = (1 - \eta_I)Q_T \tag{9}$$

η_I is usually 0.85 (0.15 Q_T in-plant and stack losses) for fossil units and 0.95 (0.05 Q_T in-plant losses) for nuclear units. From equations (7), (8), and (9), we have:

$$Q = cP\left(\frac{\eta_I}{\eta_p} - 1\right) = P(\eta_I P_{HR} - c) \tag{10}$$

which shows that the rate at which heat must be rejected in the cooling system depends upon the type (fossil or nuclear) of unit, the power level and the plant heat rate. The plant heat rate, of course, depends upon a number of factors, including the age, size and the detailed design features of the various components as well as the turbine back pressure. Figure 3 shows the variation of η_p (= c/P_{HR}) with back pressure for a large (800 MW) turbine of contemporary design modified to operate up to high back pressures. Such a curve cannot, however, be used in the backfit situation since the affected units will vary widely in age and size, and will have generally higher heat rates. The basic parameter recommended earlier, namely the heat rejection rate Q^*, can

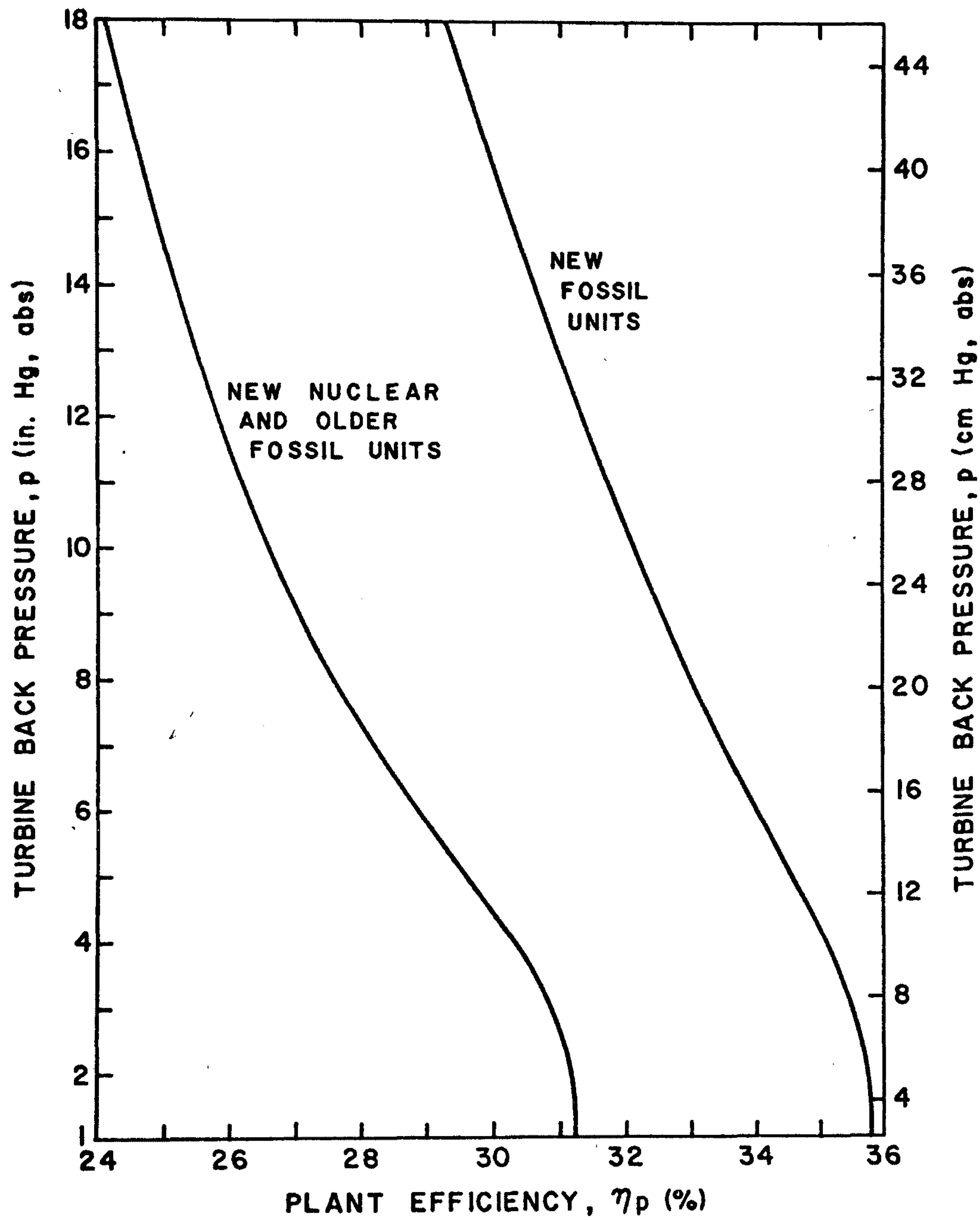

Figure 3. Typical variation of plant efficiency with turbine back pressure

nevertheless be defined as

$$Q^* = cP^* \left(\frac{\eta_I}{\eta_p^*} - 1 \right) = P^*(\eta_I P_{HR}^* - c) \qquad (11)$$

where η_p^* = plant efficiency at back pressure p^*,

P_{HR}^* = plant heat rate at back pressure p^*.

Q^* can therefore be found if the relevant quantities are known. The use of Q^* as a basic parameter characterizing the "size" of the cooling problem, however, avoids the need to distinguish between different ages, sizes and types of units to be considered in the detailed analysis of the performance of closed-cycle cooling systems. These factors can readily be taken into account in the determination of Q^* using either equation (2) or (11).

Returning to the turbine characteristics shown in Figure 2, a number of quantities of prime importance in the backfit situation can now be defined. It has already been mentioned that this particular turbine cannot be operated to generate more than the full-throttle power corresponding to the line ABC and that the back pressure cannot exceed the maximum value p_{max} corresponding to the line EDC. Thus, regardless of the cooling system used, the turbine must operate within the area bounded by OA'CE. Full-throttle operation with the open-cycle cooling system corresponds to the point A and to small deviations from it along ABC associated with the variations in the stream or water-body temperature. The corresponding heat rejection rates are nearly constant and equal to Q^*.

If this unit is to be backfitted with a closed-cycle cooling system, it is necessary to recognize that the performance of all such systems (mechanical or natural-draft cooling towers, cooling ponds, or spray canals) depends upon the physical size as well as the prevailing meteorological conditions. The first observation that can be made, however, is that it is possible to design a closed-cycle system that is large enough to operate at point A under a specified set of fixed meteorological conditions. In this case, the turbine back pressure remains the same as in open-cycle operation and consequently the turbine delivers nameplate power with the same heat rejection rate Q^*. The plant heat rate also remains the same. Such a size of the closed-cycle system represents a useful reference and is discussed in greater detail later. Now, if either the size of the cooling system is smaller, or if the meteorological conditions become more adverse than the specified set, the full-throttle operation point will shift upward along the line ABC, to a point B, say. Smaller sizes and/or more severe meteorological conditions will lead to the operating point C which corresponds to the maximum back pressure that can be tolerated. If the size of the cooling system is still smaller and full-throttle operation is to be maintained, the operating point C' will fall along the extrapolated portion of the full-throttle line ABC. However, since this implies a back pressure greater than p_{max}, the turbine must be throttled back to operate at a point such as D where the back pressure is p_{max}. It is clear that the exact location of the operating point will depend, among other things, on (a) the size and type of the closed-cycle cooling system considered, (b) the detailed cooling properties of the system, and (c) the meteorological parameters which affect the cooling system performance. The above considerations, however, enable us to identify a number of factors that have an important bearing on the economics of backfitting.

(1) The net power available for sale is equal to the gross power generated P, minus any power P_{cs} that is consumed internally in order to operate the closed-cycle cooling system, e.g., the pump and fan power requirements in the case of mechanical-draft cooling towers. The power output at any operating point (A, B, C, or D) is, of course, given by equation (5) and depends upon the turbine back pressure and the throttle setting T_s. In comparison with open-cycle operation where the power output is the nameplate capacity, there is now a <u>CAPACITY LOSS</u>, C_L:

$$C_L = P^* - P + P_{cs} \tag{12}$$

The evaluation of the maximum capacity loss is therefore important.

(2) Since the capacity loss occurs continuously, its magnitude depending upon the meteorological conditions, there is an associated <u>ENERGY LOSS</u>, E_L:

$$E_L = \sum (P^* - P + P_{cs})\Delta t \tag{13}$$

where Δt is the duration of any set of meteorological conditions and the summation is taken over all such sets of conditions occurring during the given period of time, e.g., one year. Again, it is useful to remember that this definition and succeeding definitions are made in the context of an assumed full-throttle loading and a constant demand for the nameplate capacity. For the general case of demanded power, P_D, capacity loss and energy loss would be defined respectively in terms of the gross turbine output, P as :

$$C_L = P_D - P + P_{cs}$$

$$E_L = \sum (P_D - P + P_{cs})\Delta t$$

where P = the maximum possible gross turbine output at full throttle or at the back pressure limitation if $P_D + P_{cs} \geq$ this max, and $P = P_D + P_{cs}$ if $P_D + P_{cs} <$ this max. Since the assumptions of constant demand for nameplate capacity ($P_D = P^*$) and the implied consequence of full-throttle loading (if possible) (P = max) are made, then capacity loss and energy loss are defined as in equations (12) and (13).

(3) The fuel consumption for full-throttle operation with the open-cycle cooling system can be deduced from the reference plant heat rate, P^*_{HR}. If the closed-cycle system and the prevailing meteorological conditions are such that the turbine

always operates at full-throttle (i.e., along line ABC), then the fuel consumption with the closed-cycle system will be the same as that with the open-cycle system (since Q_T is constant). However, if the operation is required at less than full-throttle (i.e., at points such as D) due to the back pressure limitation, the fuel consumption will be smaller than with the open-cycle system. Following the usual terminology, the difference between the fuel consumptions with closed- and open-cycle systems will be called the <u>EXCESS FUEL CONSUMPTION</u>, F_E, although, as indicated above, it will be either zero or negative. This peculiarity is easily seen to be the consequence of our basic assumption that the plant or unit to be backfitted is operated continuously at full throttle.

$$F_E = 0 \ , \quad \text{for } T_s = 1$$

$$= \frac{1}{\eta_I} \ [cP + Q - cP^* - Q^*]\Delta t \ , \ \text{for } T_s < 1 \qquad (14)$$

Positive excess fuel consumption (i.e., a fuel penalty) may result from backfitting if this assumption is relaxed and a specified power demand curve is used. Consider, for example, the case where the power demand is constant and lower than the nameplate capacity for part of the year. Then during that part of the year the demand can be met even with a closed-cycle system by suitable adjustment of the throttle setting, but the increased back pressure (compared to open-cycle cooling) would imply higher turbine and plant heat rates than those associated with open-cycle cooling, and therefore higher rates of fuel consumption. However, during this period the capacity loss and the energy loss will not be equal to those given by equations (12) and (13). These losses can indeed be taken to be zero since the power level can be adjusted to equal the demand plus that consumed internally by the cooling system (P_{cs}).

From the foregoing discussion it will be evident that the capacity loss, the energy loss, and the excess fuel consumption, which are all of great importance in the economics of backfitting, depend upon the size and type of the closed-cycle cooling system as well as the prevailing meteorological conditions. In particular, to evaluate the maximum capacity loss it is necessary to specify the "most severe" meteorological conditions a priori. The determination of the various factors mentioned above is considered in greater detail for each of the four types of closed-cycle cooling systems in later sections. The general discussion presented here, however, emphasizes the importance of the turbine characteristics in the overall economic analysis of backfitting.

Any survey of power plants now operating with open-cycle cooling will indicate that a wide variety of turbines will be encountered in practice. Some of these are considerably older than others, and the nameplate capacities vary over a wide range. In a study such as this, it is obviously impossible to consider each particular situation in detail and some generalizations and simplifications must be made. An effort must, however, be made to make the results as widely applicable as possible and retain a certain amount of flexibility in the methodology so that some of the peculiarities of particular units can be incorporated. To this end, three basic types of turbines which are representative of those currently in use are considered in the example calculations. The characteristics of these turbines and their nameplate capacities have been taken from a recent report prepared by Sargent and Lundy [Ref. 7 , Vol. I] and are shown in Figures 4, 5 and 6. Turbine A shown in Figure 4 is a high back-end loaded unit of contemporary design; turbine B shown in Figure 5 is a low back-end loaded unit representing some of the older plants, while turbine C shown in Figure 6 is a low back-end loaded unit whose performance is only marginally poorer than that of a contemporary unit. It is expected that most existing turbines can be classified in one of these three categories.

As indicated earlier, for the detailed evaluation of the capacity and

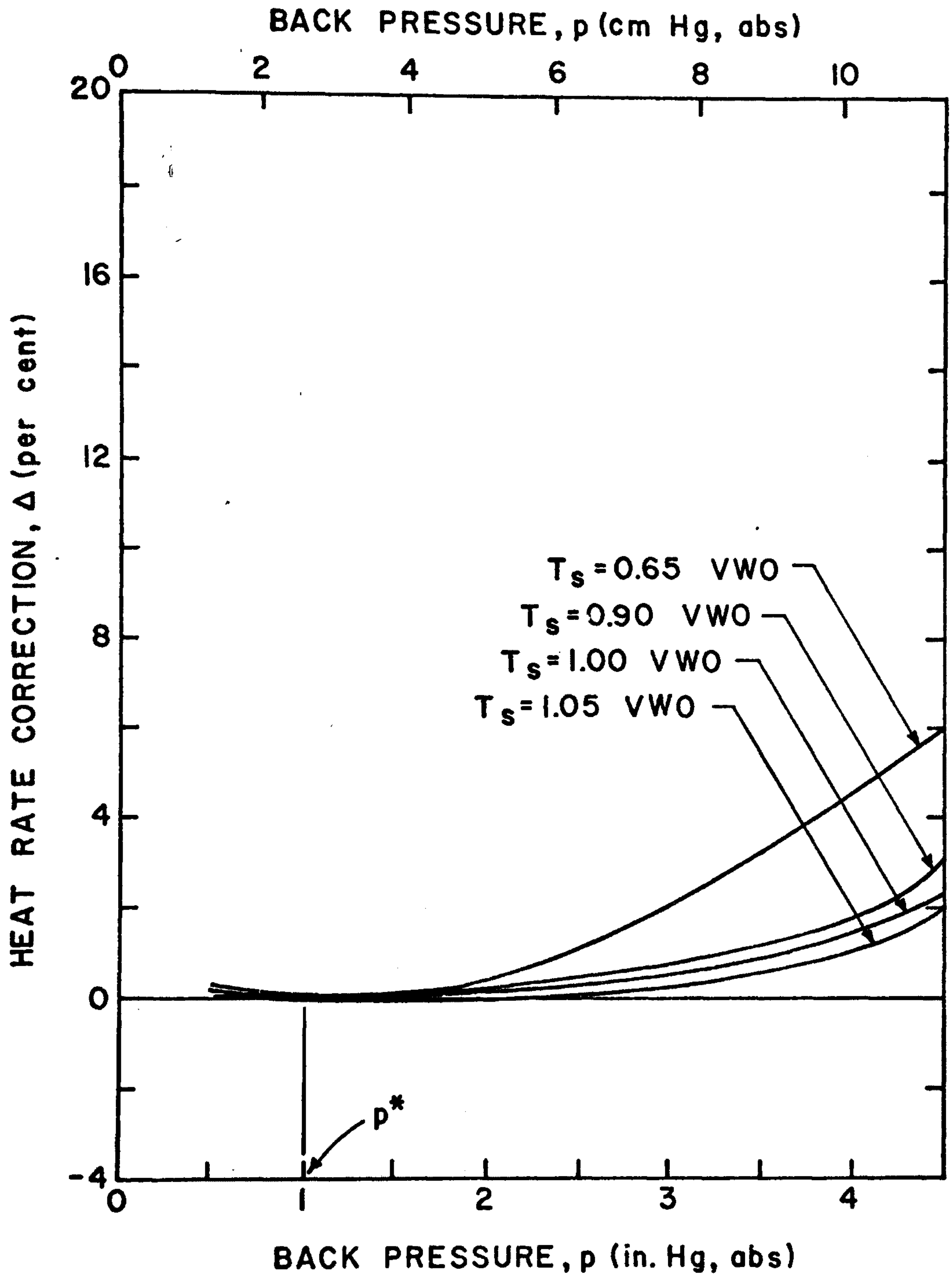

Figure 4. Heat rate characteristics of turbine A

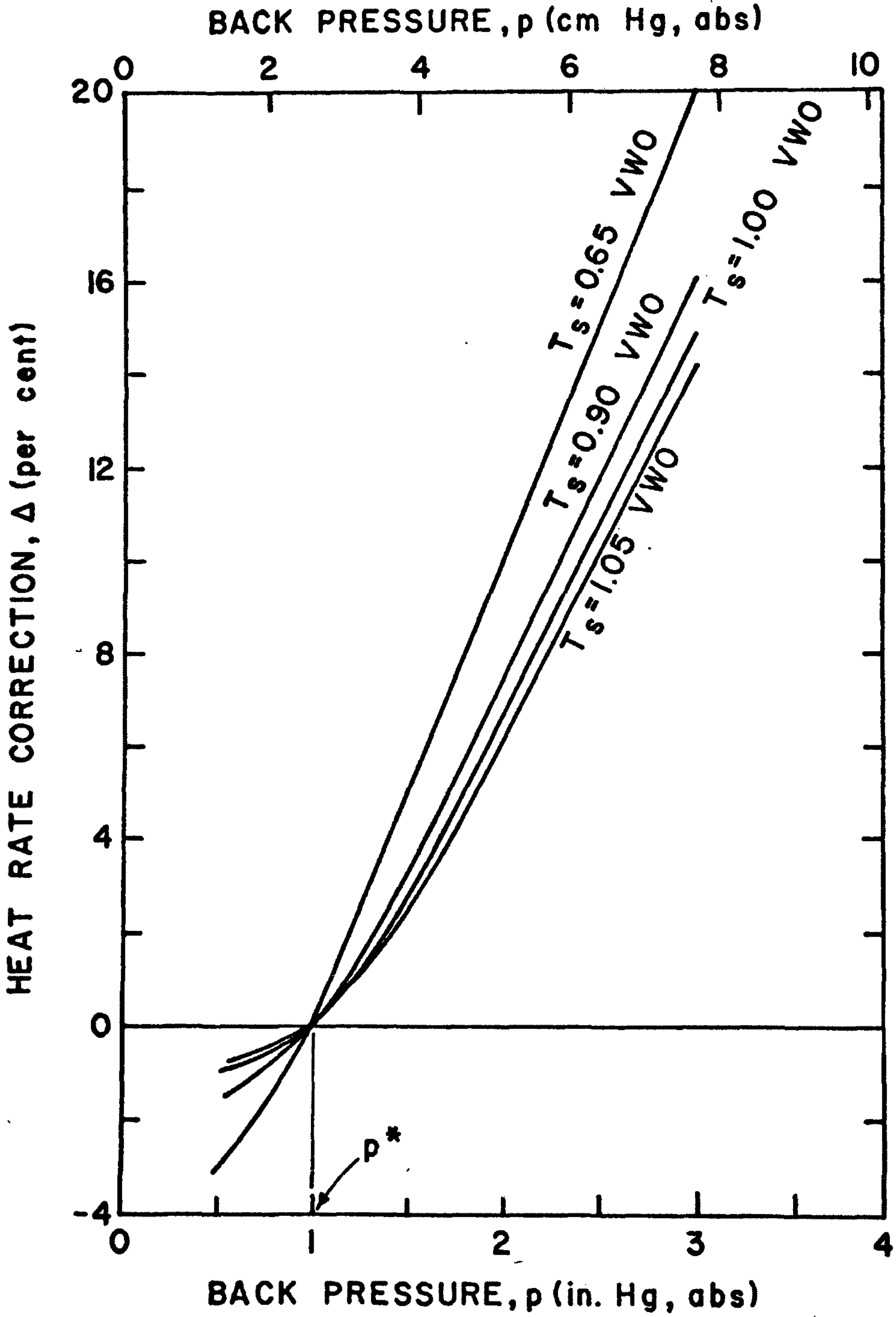

Figure 5. Heat rate characteristics of turbine B

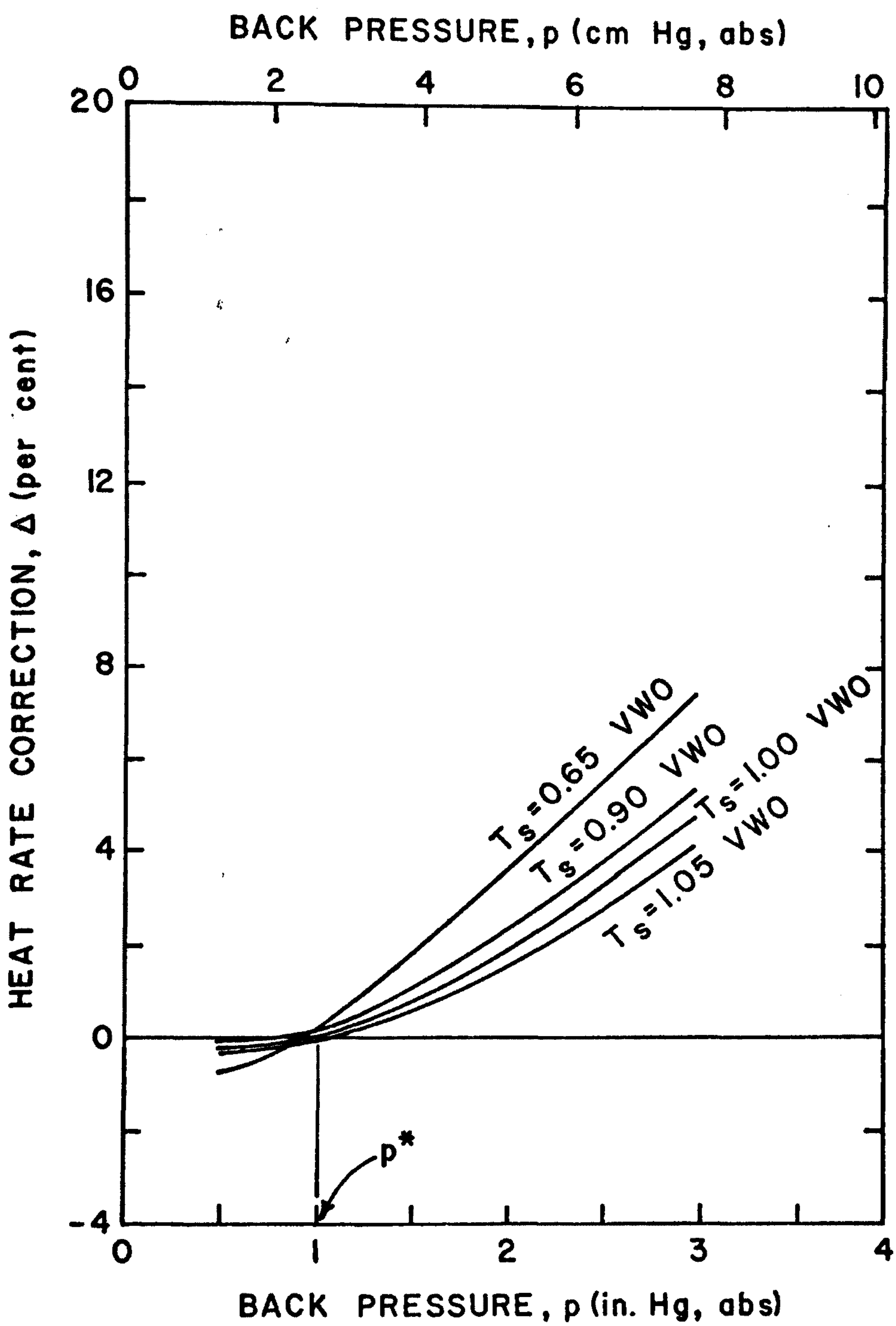

Figure 6. Heat rate characteristics of turbine C

energy losses and excess fuel consumption resulting from backfitting, it is necessary to obtain the turbine characteristics in the form shown in Figure 2. This requires a knowledge of the reference heat rejection rate Q^* defined earlier, which, in turn, depends upon the nameplate capacity, the overall thermal efficiency and on whether the unit is fossil or nuclear. For the purposes of the subsequent example calculations, however, it is assumed that the characteristic curves shown in Figures 4 through 6 can be applied to fossil as well as nuclear units, and that the plant efficiency η_p^* in open-cycle operation is 0.302 (30.2%), representative of older fossil fueled units and newer nuclear units. Then equation (11) leads to the reference heat rejection rates given in Table 1. At first sight, it would appear that the detailed calculations must be performed for both fossil as well as nuclear units (using differing values of η_I or η_p^*) and repeated for a range of values of the nameplate capacity. The foregoing assumptions imply, however, that this may not be necessary since the influence of changing the type (fossil or nuclear) of the unit is simply to change η_I and η_p^*. However, the calculation of capacity loss, energy loss, and excess fuel consumption depends only upon knowing the turbine characteristics, the nameplate capacity, and Q^* (and not on η_I or η_p^*) for the assumed full-loading pattern. Thus, it is possible to use the results obtained from a particular value of Q^* (associated with a particular value of the nameplate capacity and type of unit) to predict the performance of nuclear or fossil units (different η_p^*, but with turbine heat rejection = Q^*) with different nameplate capacities. This is best achieved by presenting the results in a suitable nondimensional form.

From equation (8) it will be noted that the thermal efficiency assumed here (0.302) leads to a reference (open-cycle) plant heat rate P_{HR}^* of about 11300 Btu/kW-hr (11920 kJ/kW-hr) for both fossil and nuclear units. While it is recognized that the historical data collected by the Federal Power Commission and analyzed in the EPA Development Document [2, see Figures IV-10 to IV-12 on pp. 76-78] indicate a wide

Table 1. REFERENCE CONDITIONS FOR TURBINES A, B AND C FOR EXAMPLE CALCULATIONS

TURBINE	NAMEPLATE CAPACITY P^*, MW	REFERENCE BACK PRESSURE p^*, inch Hg abs (cm Hg abs)	REFERENCE HEAT REJECTION RATES @ $\eta_p^* = 0.302$ Q^*, 10^9 Btu/hr (10^9 kJ/hr)	
			Old Fossil	New Nuclear
A	411	1.00 (2.54)	2.545 (2.686)	3.010 (3.176
B	275	1.00 (2.54)	1.703 (1.797)	2.014 (2.125)
C	535	1.00 (2.54)	3.313 (3.496)	3.918 (4.134)

variation of plant heat rates with the age and size of the units (and thus a wide variation in η_p^*), the above values are representative, as already mentioned, of open-cycle operation of the older fossil units and the relatively new nuclear units. The detailed example results will need correction when they are used to study the consequences of backfitting units whose reference heat rates deviate substantially from those used here. In any case, as will be emphasized repeatedly in this study, the general methodology adopted here, and particularly the various computer programs which have been developed, can be used in conjunction with any set of specified inputs i.e., type and size of turbine, reference heat rejection rate and therefore the plant heat rate.

Reference has already been made to the possibility of extending the validity of the results from a relatively small number of specific calculations to treat a much wider variety of cases by the use of suitable dimensionless plots. Therefore, it is useful to define the REFERENCE SIZE OF A CLOSED-CYCLE COOLING SYSTEM: L^* = reference length of mechanical-draft cooling tower, S^* = reference shell height of natural-draft cooling tower, A^* = reference area of cooling pond, and N^* = reference number of module groups along a spray canal. Formally, the reference size is defined as the size required to reject Q^*, at some specified meteorological conditions while maintaining the turbine back pressure at some selected value, p'. It should be noted that because of physical limitations, p' is not necessarily equal to p^*, nor is it necessary to use the same value of p' for all systems since the final economic evaluations are independent of the reference sizes. Thus, for example, the capacity loss for a given turbine system using the mechanical-draft cooling tower can be plotted nondimensionally as (C_L/P^*, kW/kW) vs. (L/L^*), where L denotes the size (length of tower) of the closed-cycle system. It will be seen later that such plots enable the presentation of the results in a compact manner.

B. GEOGRAPHICAL LOCATION AND METEOROLOGICAL CONDITIONS

The various factors that influence the economics of backfitting depend to a large extent on the size and the performance of the closed-cycle cooling system being considered. The day-to-day performance of a system of given size, in turn, depends upon the expected variations in the meteorological conditions at the site. Thus, for example, the frequencies of occurrence of various wet- and dry-bulb temperatures must be considered in the analysis of mechanical-draft evaporative cooling towers, while the performance of cooling ponds is influenced by the variations in wet- and dry-bulb temperatures as well as wind velocities and cloud-cover. It is, of course, not possible to make a detailed evaluation of each site in a study such as the present one. A few specific sites have therefore been chosen as being representative of the areas in which there is a large concentration of open-cycle operations. In particular, the performance of the three basic types of turbines mentioned earlier will be investigated in detail in conjunction with the meteorological data from four different sites, namely Chicago, Los Angeles, Miami and St. Louis. These four stations also fall in the major climatological regions of continental U.S. [see, for example, Ref. 7, Vol. II. Appendix to Section IV.A] and are therefore expected to give a reasonable representation of other sites in their respective regions.

The types of meteorological data usually compiled by the U.S. Weather Bureau [8] are shown in Table 2. Most of the required information can be obtained from such records. Thus, for example, it is possible to obtain the frequency of occurrence of given values of a particular parameter or combinations of parameters by simple analysis. The frequencies of occurrence $f(T_{db}, T_{wb})$ of various values of the dry- and wet-bulb temperatures at Miami are shown in Table 3. Note that the sum of all the frequencies for all the combinations is equal to unity. Similar frequency distributions can be generated for other combinations of meteorological parameters. It is also possible to calculate the dry- and wet-bulb temperatures which are not exceeded more than a given number of hours in a year. The dry- and wet-bulb temperatures not exceeded more than 10 hours per year at the four sites listed above are given in Table 4.

In the design of closed-cycle cooling systems it is customary to quote what are known as "design meteorological conditions." Thus, for example, in the case of mechanical-draft wet cooling towers, a "design wet-bulb temperature" is generally specified and defined as the value which is not exceeded by more than a certain percentage (usually between 2% and 5%) of time during the warmest consecutive four months. In the United States that period is taken to be June through September. Cooling tower manufacturers have available a list of the design conditions appropriate for various sites in the United States. Table 5 shows the relevant values for the four sites to be considered here. While the design parameters give a good indication of the relative sizes of cooling systems required for identical duty at different locations, it will be evident from the considerations of the previous

Table 2. ANNUAL SUMMARY OF WEATHER DATA AT MIAMI
(REPRODUCED FROM U. S. WEATHER BUREAU RECORDS [8])

TEMPERATURE AND WIND SPEED-RELATIVE HUMIDITY OCCURRENCES

WIND	0-4 M.P.H.						5-14 M.P.H.						15-24 M.P.H.						25 M.P.H. AND OVER						TOTAL OBS
REL. HUMID. TEMP (°F)	< 30	30-49%	50-69%	70-79%	80-89%	90 100%	< 30	30-49%	50-69%	70-79%	80-89%	90 100%	< 30	30-49%	50-69%	70-79%	80-89%	90 100%	< 30%	30-49%	50-69%	70-79%	80-89%	90-100%	
99/ 95								1	+						+										2
94/ 90		2	4				1	16	82				+	3	17					+					125
89/ 85		2	33	7	+		1	31	541	104	1		+	10	137	19	+			1	1	+			888
84/ 80	+	5	26	121	134	6	2	46	379	539	239	8	+	16	175	74	17	1		+	4	2	1		1795
79/ 75	+	5	24	55	329	265	3	58	340	348	465	239	+	26	162	85	43	10		1	4	2	1	2	2463
74/ 70		4	28	55	130	213	4	59	261	211	297	267	1	26	71	30	30	16		1	1	+	1	1	1708
69/ 65	1	4	28	41	75	106	3	36	138	84	109	124	2	10	27	8	7	7			+	+	+	+	810
64/ 60	+	2	11	26	50	50	2	19	65	56	83	56	1	9	11	4	5	3		+			+	+	277
59/ 55		1	6	11	30	16	2	12	47	45	52	29	1	5	11	4	4	1						+	147
54/ 50			4	7	12	5	1	7	31	28	28	7	+	2	7	4	2	2						+	71
49/ 45			2	3	4	1	+	3	15	17	17	3	+	1	2	2	1	+							26
44/ 40			1	1	2	1	+	1	6	6	4	2	+		1	1	+	1							4
39/ 35			+	+	1	+		+	1	1	+	+		+			+	+							
TOTAL	1	25	168	327	765	664	19	289	1906	1439	1294	731	6	107	621	230	109	41		2	10	4	4	3	8767

Occurrences are for the average year (10-year total divided by 10). Values are rounded to the nearest whole, but not adjusted to make their sums exactly equal to column or row totals. "+" indicates more than 0 but less than 0.5.

Table 2 (continued). ANNUAL SUMMARY OF WEATHER DATA AT MIAMI
PERCENTAGE FREQUENCIES OF WIND DIRECTION AND SPEED

Direction	HOURLY OBSERVATIONS OF WIND SPEED (IN MILES PER HOUR)										AV SPEED
	0-3	4-7	8-12	13-18	19-24	25-31	32-38	39-46	47 OVER	TOTAL	
N	1	4	3	1	+	+				9	7.8
NNE	1	3	2	1	+	+				6	7.3
NE	1	2	2	1	+	+				6	8.7
ENE	1	2	3	3	+	+	+	+		9	10.4
E	1	2	4	2	+	+		+	+	9	9.6
ESE	1	3	5	3	+	+			+	11	9.8
SE	1	3	4	2	+	+	+	+		10	9.7
SSE	+	2	3	2	+	+	+	+		7	10.8
S	+	1	2	1	+	+	+			5	10.0
SSW	+	1	1	1	+	+	+			3	9.4
SW	1	2	1	+	+	+				4	7.9
WSW	+	1	1	1	+	+	+			3	9.0
W	+	1	1	+	+	+				2	8.4
WNW	+	1	1	+	+	+	+			2	8.2
NW	1	1	1	+	+	+				3	7.8
NNW	1	2	2	1	+	+				6	8.3
CALM	4									4	
TOTAL	14	30	34	20	2	+	+	+	+	100	8.8

Table 2 (continued). ANNUAL SUMMARY OF WEATHER DATA AT MIAMI
PERCENTAGE FREQUENCIES OF SKY COVER, WIND, AND RELATIVE HUMIDITY

HOUR OF DAY	CLOUDS SCALE 0-10			WIND SPEED (M.P.H.)				RELATIVE HUMIDITY (%)					
	0-3	4-4	8-10	0-3	4-12	13-24	25- & OVER	0-29	30-49	50-69	70-79	80-89	90-100
00	55	23	23	20	70	10	+		+	10	25	43	22
01	55	23	23	24	67	9	+		+	8	20	43	28
02	58	20	22	27	64	8	+		+	7	18	42	33
03	57	21	22	28	63	8	+		+	6	16	40	38
04	56	22	22	30	62	8	+		+	6	15	37	42
05	53	24	23	28	65	8	+		+	6	13	37	45
06	43	25	31	28	65	7	+		+	6	14	37	44
07	39	26	35	23	67	9	+		+	7	20	42	32
08	37	29	34	14	67	18	+		1	20	35	31	13
09	32	33	35	7	64	29	+	+	3	44	33	15	5
10	25	37	37	3	60	37	+	+	7	60	22	8	3
11	23	40	37	3	55	41	1	1	11	66	14	5	3
12	22	40	38	2	52	45	1	1	15	64	12	5	2
13	22	38	40	1	50	49	1	1	16	62	12	5	3
14	23	37	40	1	50	48	1	1	17	61	13	5	3
15	25	35	40	1	51	46	1	1	15	59	15	6	3
16	26	31	43	1	56	42	1	1	12	58	18	8	4
17	29	29	42	2	65	33	+	1	8	54	24	10	4
18	32	27	41	5	75	20	+	+	5	43	32	15	6
19	36	26	38	9	78	12	+	+	3	31	39	20	8
20	43	25	32	14	76	10	+		1	22	39	28	10
21	49	24	27	17	73	10	+		1	17	36	33	12
22	52	23	25	19	72	9	+		1	14	33	38	15
23	54	23	23	20	70	10	+		+	12	29	41	18
AVG	39	28	32	14	64	22	+	+	5	31	23	25	16

Table 3. FREQUENCY OF OCCURRENCE OF T_{wb}, T_{db} AT MIAMI

Dry Bulb Temp, T_{db} °F (°C) \ Wet Bulb Temp, T_{wb} °F (°C)	20 - 30 ([-6.7]-[-1.1])	30 - 40 ([-1.1] - 4.4)	40 - 50 (4.4 - 10.0)	50 - 60 (10.0 - 15.6)	60 - 70 (15.6 - 21.1)	70 - 80 (21.1 - 26.7)	80 - 90 (26.7 - 32.2)	90 - 100 (32.2 - 37.8)
20-30 ([-6.7]-[-1.1])	0.0							
30-40 ([-1.1] - 4.4)		0.0003						
40-50 (4.4-10.0)		0.0027	0.0084					
50-60 (10.0-15.6)			0.0200	0.0283				
60-70 (15.6-21.1)			0.0033	0.0570	0.0838			
70-80 (21.1-26.7)				0.0146	0.1945	0.2667		
80-90 (26.7-32.2)				0.0002	0.0333	0.2632	0.0092	
90-100 (32.2-37.8)					0.0001	0.0064	0.0078	
100-110 (37.8-43.3)								0.0

Table 4. TEMPERATURES EQUALLED OR EXCEEDED LESS THAN 10 Hrs/Yr

Site	$\hat{T}_{wb}$ °F	(°C)	$\hat{T}_{db}$ °F	(°C)
Chicago	82	(27.8)	96	(35.6)
Los Angeles	73	(22.8)	93	(33.9)
Miami	83	(28.3)	97	(36.1)
St. Louis	83	(28.3)	103	(39.4)

Table 5. "DESIGN" TEMPERATURES FOR COOLING TOWERS

SITE	WET-BULB TEMPERATURE, °F (°C)				DRY-BULB TEMPERATURE, °F (°C)			
	1%	2.5%	5%	10%	1%	2.5%	5%	10%
Chicago	78 (25.6)	76 (24.4)	75 (23.9)	73 (22.8)	94 (34.4)	92 (33.3)	89 (31.7)	85 (29.4)
Los Angeles	71 (21.7)	69 (20.6)	68 (20.0)	67 (19.4)	84 (28.9)	81 (27.2)	78 (25.6)	75 (23.9)
Miami	79 (26.1)	79 (26.1)	78 (25.6)	78 (25.6)	91 (32.8)	90 (32.2)	89 (31.7)	88 (31.1)
St. Louis	79 (26.1)	78 (25.6)	77 (25.0)	75 (23.9)	98 (36.7)	95 (35.0)	93 (33.9)	89 (31.7)

section that a realistic evaluation of the performance of the power plant or unit must consider the detailed variations in the meteorological conditions from their design values as well as load variations (if considered). This is particularly so in a backfit situation where quantities such as capacity and energy losses and excess fuel consumption are of prime importance and must be predicted accurately.

In the example calculations described here, the MAXIMUM CAPACITY LOSS, against which the capital cost of replacement is assessed, is evaluated at the meteorological conditions which are not exceeded more than

10 hours during the year.

It is important to realize that other definitions have been and are being used for determining the maximum capacity loss for use in sizing the additional required capacity. Most definitions involve a specification of the sort made here; i.e., maximum capacity loss is the capacity loss at meteorological conditions which are not exceeded more than some number of hours during the year. Any other definition can be easily incorporated into the computer models, but the example calculations are based on the 10 hours per year figures of Table 4. The total energy loss and the excess fuel consumption are calculated by summation, with respect to time, of the capacity loss and the excess rates of fuel consumption, respectively, over all possible combinations of meteorological conditions.

C. ECONOMIC CONSIDERATIONS

For the purposes of this section, the method outlined earlier is used to determine the maximum capacity loss (C_L in kW), the annual energy loss (E_L in kW-hr), and the excess fuel consumption (F_E in kW-hr) for the situation in which a particular power plant or unit (of known nameplate capacity at a specific site) is to be backfitted with a closed-cycle cooling system of known type and size. The problem to be considered is then the determination of the total extra cost, in mills per kilowatt-hour, of backfitting.

The TOTAL DIFFERENTIAL CAPITAL COST, CC in dollars, to be charged against the project will involve the following:

(a) The differential capital cost of the closed-cycle cooling system minus salvage values, CC_S. This cost, depends upon the type and size of the system, and it includes the cost of site acquisition and preparation, the purchase and installation of the cooling equipment and associated auxiliaries, as well as the start-up and testing costs. For most closed-cycle systems the capital costs can be estimated by recourse to the experience of the industry concerned, and although a number of site-dependent factors need to be considered, reasonable estimates can be made using standard procedures. The methods used here for the different types of cooling systems will be described in detail in subsequent sections. The differential capital costs of the closed-cycle cooling system can be estimated as follows. The salvage worth of the old condenser (which equals the estimated sale price if sold or the replacement cost if salvaged for new construction) is subtracted from the capital cost of the new condenser, if a new condenser is indicated, to calculate the differential condenser cost. The differential capital cost of the pumps and piping system is calculated either by subtracting the estimated sale price of the old system from that of the new, or by estimating capital cost of an additional system to make-up additional pumping capacity. The differential cost of the cooling system (excluding pumps, piping and condenser) is estimated by subtracting the salvage worth of all old cooling system components, excluding pumps, piping and condenser, (which again equals estimated sale price if sold or replacement cost if salvaged for new construction), from the capital cost of the new system. Land requirements are limited to consideration of only the additional land required for the new system. Any hook-up and testing costs (exclusive of lost revenue) which would be incurred in the backfit are also differential capital costs to be considered. Adding all of these differential capital costs results in the determination of the differential capital cost of the closed-cycle cooling system, CC_S.

(b) The differential cost associated with the plant or unit shut down at the time of the changeover from the open-cycle to the closed-cycle cooling system, CC_{DT}. It is obvious that this will depend upon the affected capacity and the duration of the outage. The time required for the changeover will depend on the layout and accessibility of the existing system. The EPA Development Document [Ref. 2, p. 598] estimates that the time required for this purpose will vary from 2 to 5 months, depending upon the site conditions, with an average time of 3 months. This time generally depends upon the cooling system being used, the unit being backfitted, and many site-specific factors. However, this estimate appears reasonable, and since more definitive estimates could not be obtained during the course of this study, it was decided to make downtime a variable parameter whose influence on the overall economics of backfitting could be examined. It is obviously beneficial to schedule the backfit operation such that the changeover coincides with periods of low power demand and with the annual maintenance period (of the order of one month) during which the plant is down in any case. Perhaps the most logical way in which the downtime cost can be evaluated is to equate it to the cost of energy lost during the outage, i.e., the product of the downtime, the affected capacity, the overall capacity factor during the outage and the unit differential cost of energy loss, e'_ℓ (the purchase price minus the usual generating cost). This is basically the procedure adopted here. It should, however, be mentioned that the recent Sargent and Lundy study [Ref. 7, Vol. I, p. II.28] incorporated an outage capital cost of $4.00 per kilowatt for fossil capacity and $7.00 per kilowatt for nuclear capacity (1970 dollars) for the installation of cooling towers on a retrofit basis, although it was suggested that outage costs can easily range from $1-$21 per kilowatt and cannot be assigned on an a priori basis. The procedure suggested here would therefore appear to be more satisfactory.

(c) The capital cost of installing additional generation capacity to replace the lost capacity, CC_R. Once the maximum capacity loss C_L has been determined, the assessment of the capital cost depends largely on the choice of an appropriate unit cost in dollars per kilowatt. It has generally been assumed that the lost capacity and the energy loss resulting from backfitting will be replaced by installing gas-turbine peaking units. In this connection, two additional factors need to be recognized. First, the anticipated demand of gas turbines may exceed the available production capacity, resulting in an escalation of prices over present estimates. Secondly, it is likely that many of the larger utilities might consider building additional fossil-fuel or nuclear power plants to replace their cummulative capacity losses occasioned by backfitting. In this case, the increased revenue from operation at noncritical periods would also have to be taken into account. Thus, it is possible that the capital as well as operating costs of the replacement capacity will vary over a wide range. Table 6, taken from Ref. 7 (Vol. I, p. II-33), shows the cost estimates for various types of replacement methods in 1980 dollars. In the present study, the unit costs associated with the replacement capacity are treated as basic variables since they can exert a significant influence on the economics of backfitting, and it is suggested that their inclusion in the final economic analysis be based on the particular circumstances of the affected utility. Since the amount of capacity loss, in megawatts, can be calculated by the procedure described earlier, it is a simple matter to study the influence of varying the unit costs of replacement capacity. As previously mentioned, if capacity losses are to be made up using increases of the base load additions for the system then the economic procedure outlined in this report does not strictly apply because of the necessity of including the increased revenue that would accrue at off-peak conditions. Therefore, for this situation, it is advised that the cost of the new cooling system be calculated including these increased revenues and the differential costs be evaluated in a more appropriate way.

The TOTAL DIFFERENTIAL CAPITAL COST can therefore be written in the following form:

Table 6. CAPITAL COST OF REPLACEMENT CAPACITY AND FUEL (1980 estimates)
[Ref. 7, Vol. I, p. II-33]

TYPE OF REPLACEMENT POWER	UNIT CAPITAL COST OF REPLACEMENT POWER $ per kW	UNIT COST OF FUEL $ per 10^6 Btu	
		A*	B*
Coal	376	0.73	1.04
Oil	292	1.56	1.91
Nuclear	457	0.30	0.30
Gas or Combustion Turbines	154	2.90	2.06

*The two sets of values correspond to different assumptions concerning the price of oil (depending on the market conditions) and the price of coal (reflecting possible effect on environmental regulations). These may therefore be regarded as the upper and lower limits.

$$CC = CC_S + CC_{DT} + CC_R, \quad \text{in } \$ \qquad (15)$$

The DIFFERENTIAL OPERATING COSTS, OC in dollars per year, to be assessed against backfitting, consist of the following:

(a) The operating and maintenance costs of the replacement capacity, OC_R. As explained earlier, it is assumed that a peaking unit needs to be installed such that its peak power is equal to the maximum capacity loss (C_L), and the energy supplied by it is equal to the energy loss (E_L), sustained by the basic unit as a result of backfitting. (Again the definitions of capacity loss and energy loss are relevant only for the assumptions of constant power demand for nameplate capacity and the implied consequence of full-throttle loading). More generally, installed replacement capacity would be required to have its peak power equal to the maximum (10 hour exceedance, say) value of

$$C_L = P_D - P + P_{cs}$$

and supply energy equal to

$$E_L = \sum (P_D - P + P_{cs}) \Delta t$$

Both of these expressions are zero (i.e., $P = P_D + P_{cs}$) if $P_D + P_{cs}$ is smaller than the possible full-throttle output, P. Since $P_D = P^*$ and P = the maximum possible gross turbine output at full throttle or at the back pressure limitation, equations (12) and (13) are used.

It is important to note that the capacity and energy losses can be made up in other manners, besides supplying a peak unit, as mentioned earlier. In the case of increasing the base load additions to the system, the operating and maintenance cost of the "increased system capacity" will be included in the operation and maintenance cost of the new system and must be considered. Such evaluations are not attempted in the present study.

While the annual operation and maintenance of such a peaking plant will depend upon a number of complex factors, it is certain that the cost of energy produced by it will be substantially greater than that produced by the basic, base-loaded unit. It is usually assumed that these costs can be taken into account by assigning a single unit cost of replacement energy produced. For the case of gas-turbine peaking units, a value of 10 mills per kilowatt-hour appears to be a reasonable figure. In the present study, however, this unit cost is again left as a variable parameter since it has a significant influence on the final economic assessment of backfitting.

(b) The cost of excess fuel consumption, OC_{EF}. It has already been mentioned that the rate of fuel consumption with a closed-cycle cooling system will be different from that with the open-cycle system. The annual cost associated with the difference is easily found by multiplying the excess fuel consumption, F_E (in kW-hr thermal or Btu) by the unit cost of fuel (in \$ per kW-hr or \$ per Btu). Table 6, taken from Ref. 7 (Vol. I, p. II-33), includes the unit costs expected to prevail in 1980 for various types of fuel. These cost estimates are included here simply as a guide. Better estimates can, of course, be obtained from the affected utility for any particular unit.

(c) The differential cost of operation and maintenance of the new closed-cycle cooling system over the existing open-cycle cooling system, OC_S. This will obviously depend on the type of closed-cycle system that is considered. Taking the specific example of mechanical-draft wet cooling towers, the operating costs will include the cost of make-up water (evaporation, drift and blowdown), the cost of blowdown treatment, and the maintenance of the tower structures and related equipment such as fans, pumps and controls. The differential operating and maintenance costs will then be these costs minus those associated with the present system. It will be clear, however, that the cost of the power consumed by the fans and pumps need not be considered since that has already been taken into account in the evaluation of the costs of capacity and energy replacement. The assessment of the operating and maintenance costs of the different closed-cycle cooling systems will be considered in greater detail in later sections.

From the foregoing, the TOTAL DIFFERENTIAL OPERATING COST to be assessed against backfitting can be written as follows:

$$OC = OC_R + OC_{EF} + OC_S, \qquad \$ \text{ per year} \tag{16}$$

Once the differential capital cost CC (in dollars) and the differential operating cost OC (in dollars per year) have been determined for a specific power plant or unit, the problem reduces to that of assessing the total differential cost, in mills per kilowatt-hour, to be charged against the NET energy delivered. The manner in which the capital and operating costs are combined to obtain the total cost depends primarily upon the general economic situation of the utility and the age of the affected unit. Adopting the levelized annual cost method of accounting, the total differential cost, in dollars per year, can be written

$$TC = OC + CC \times FCR, \qquad \$ \text{ per year} \tag{17}$$

where FCR is the "fixed charge rate" which reflects the annual cost of raising the required capital and includes such factors as interest on debt, required return on the stockholders' equity, depreciation of the equipment, and salvage value (useful life of the plant or unit), property taxes, property and income tax rates, etc. Although these factors vary from utility to utility, the value of the fixed charge rate to be used in the backfit analysis is determined mainly by the remaining life of the plant or unit to be backfitted. The rates recommended and utilized in the EPA Development Document [Ref. 2, p. 597] and in the Sargent and Lundy study [Ref. 7, Vol. I, p. II-32] are compared in Figure 7. The two separable projections made in Ref. 7 reflect the influence of making different assumptions concerning the rates of return on the capital investiment. For the purposes of the present work, it is recognized from equation (17) that the precise value that is chosen for the fixed charge rate will greatly influence the total cost assessed against backfitting, and consequently it is retained as a basic variable that needs to be ascertained with some care by a detailed examination of the financial structure of the utility concerned.

The total cost, in dollars per year, obtained from equation (17) can now be prorated over the rated net energy output of the affected unit or over the actual net energy output of the affected unit. The rated net energy output for one year is simply

$$ER = 8760 \sum [P_D \times f(T_{wb}, T_{db}, P_D)] \tag{18}$$

Likewise, the actual net energy generated by the affected unit for one

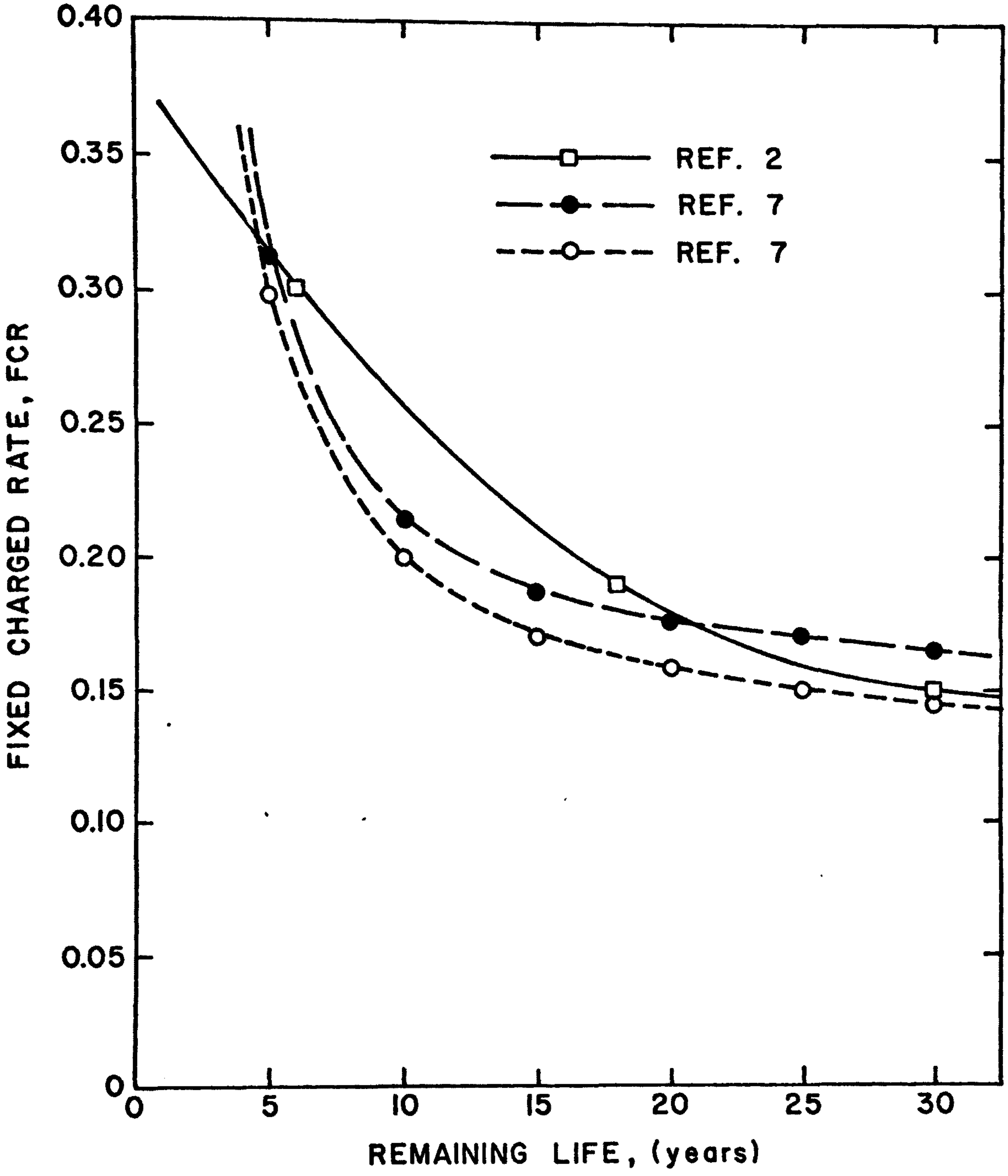

Figure 7. Variation of the fixed charge rate with remaining life of plant or unit [2,7]

year is given by

$$EA = 8760 \sum [P_D \times f(T_{wb}, T_{db}, P_D)] - E_L$$

$$= 8760 \sum [(P-P_{cs}) \times f(T_{wb}, T_{db}, P-P_{cs})] \tag{19}$$

where the frequency, f, depends upon both meteorological conditions and power, accounting for fluctuations in output (or demand, as the case may be) as well as scheduled or unscheduled outages for repairs and maintenance.

There is some question as to what basis to use in prorating the total cost: actual or rated net energy output. The answer depends upon the purpose for which prorated costs are calculated. For purposes of comparing total costs of a cooling system for different nameplate capacity power plants, the rated net energy basis would prove more useful, allowing cost comparisons which do not penalize twice for energy losses. For purposes of estimating costs of power to consumers, the actual net energy output basis would be more useful, allowing a more realistic expression of real costs.

It will be recalled from the introduction to this section that in order to maintain a certain amount of generality the analysis has been restricted to the idealized situation in which the power plant or unit delivers maximum power possible throughout the year, for which $P_D = P^*$ and $\sum f(T_{wb}, T_{db}, P^*)$ equals unity. The rated net energy output for one year then becomes $ER = 8760\ P^*$ and the actual net energy output for one year, $EA = 8760\ P^* - E_L$. The prorations are arbitrarily made in terms of the rated net energy output in the various examples which follow. Conversion to the basis of actual net energy output can be easily accomplished using the above two equations. The UNIT EXCESS COST OF ENERGY PRODUCTION resulting from backfitting, tc, is then

$$tc = \frac{OC + CC \times FCR}{8760 \sum [P_D \times f(T_{wb}, T_{db}, P_D)]} \quad \text{(for any loading pattern)}$$

(20)

or

$$tc = \frac{OC + CC \times FCR}{8760\ P^*} \quad \text{(for idealized full-throttle loading pattern)}$$

The various relations proposed in this section on economic analysis are summarized in Appendix I. A discussion of the treatment of variable loading conditions follows in part D of this section, and a numerical example is presented in Section V.J.

D. TREATMENT OF A VARIABLE LOADING PATTERN

Although it has been assumed in the present study that the power plant or unit operates at full throttle throughout the year and satisfies a constant demand for nameplate capacity, a discussion of variable loading patterns is included for completeness. It should also be mentioned that the computer programs used in the analysis of the closed-cycle cooling systems are written in general terms and can accept any variation in power demand and meteorological conditions; i.e., input data for the programs include the relative frequency of occurrence of various meteorological conditions and corresponding power demands. It is, therefore, quite straightforward to analyze variable loading patterns by the programs given in the present study.

For the case of a variable loading pattern, the rated net energy generated in a year is given by equation (18), and the unit excess cost of energy production resulting from backfitting may be calculated by the first of equations (20). For use in that equation, the differential operating cost, OC, is calculated in the computer program with the proper accounting for the variable operating schedule. However, if the graphical results are used, it must be remembered that the value of OC will correspond to the idealized full-throttle case and will be over-estimated for the variable loading application.

Another term commonly employed in the power industry called capacity factor, CF, should now be mentioned. Use of the capacity factor offers a simplified empirical method to account for fluctuations in power demand as well as scheduled or unscheduled power outages. However, in the present study, it is not necessary to apply CF in the computation of annual energy output because of the use of the general expression in equations (18), (19), and (20).

The major factors that influence the economics of backfitting a power plant with a closed-cycle cooling system have been identified in this section. It should be emphasized that the general method of approach described here is common to all types of closed-cycle systems. In order to evaluate the total cost of backfitting any particular type and size of cooling system to a plant or unit with given characteristics, however, it is obvious that it is necessary to perform the detailed calculations described in parts A and B in conjunction with a knowledge of the thermodynamic characteristics of the cooling system. These computations form the subject matter of the next four sections. In each case, the application of the general methodology is illustrated by a hypothetical example.

An important aspect of the work described here is the development of major computer programs which are flexible enough to allow the analysis of the economics of backfitting given any set of site and utility dependent inputs. It goes without saying that the proper identification of these inputs is a significant part of the problem and the results are no better or worse than the inputs themselves. Although results have been obtained using the best available information, and presented in graphical form wherever possible, it is important to note that for any particular situation it is preferable to use the computer programs. The example results nevertheless give a quick estimate of the cost of backfitting.

SECTION V

MECHANICAL-DRAFT WET COOLING TOWERS

It is well known that the amount of cooling performed by a wet (or evaporative) cooling tower depends primarily upon the ambient wet-bulb temperature, the temperature of the hot water entering the tower, and the size and thermodynamic characteristics of the "wet pile" inside the tower. Although the basic theory of evaporative cooling has been presented in considerable detail in the literature, the actual performance of towers designed and built by various manufacturers will differ due to the differences in the internal construction of the wet piles and in the air and water loadings recommended. Much of the empirical information on the heat transfer properties of particular designs and the criteria used to determine the air and water loadings, which are required to complete the theoretical models, are, however, regarded as proprietary by the manufacturers for obvious reasons. In what follows, an attempt has been made to develop a methodology that is capable of accepting any set of design parameters so that the performance of towers of different designs can be analyzed. Detailed example results are then presented for a particular set of input parameters which were obtained through the cooperation of a leading manufacturer of cross-flow cooling towers. These results therefore apply to CROSSFLOW, MECHANICAL-DRAFT WET TOWERS. It is hoped, however, that the different designs of such equipment are not so radically different so as to limit the applicability of the results to the product of a single manufacturer.

For the purposes of backfitting a power plant or unit with crossflow mechanical-draft cooling towers, it is first necessary to ask the question: how large a tower is being built? Although the answer to this is not simple, it is obvious that owing to the peculiarities of the backfit situation, the actual size will be different from that which will be recommended for a new plant or unit of identical design. Throughout this section, therefore, the physical size of the tower is regarded as a primary variable so that the various quantities of interest, such as the capital costs, maintenance costs, capacity and energy losses, and excess fuel consumption, can be calculated for a range of sizes. These quantities can then be used, in conjunction with the economic considerations outlined in the previous section, to identify the project costs.

A typical mechanical-draft, crossflow, evaporative cooling tower is shown in Figures 8 and 9. From Figure 8 it will be seen that the overall tower structure consists of a number of distinct "cells", each with its own fan. The physical size of a tower is specified if the number of cells and the dimensions of the fill in each cell are known. Alternatively, the size can be specified by the height H, the width W and the total length L of the fill, the length of each cell being L/N, where N is the number of cells. The quantities H, W and L will be used as the primary indicators of the size of the cooling tower. It will be clear that H is the length of the water path and is also a measure of the pumping height required. W is a measure of the length of the air path and therefore will influence the size and the horsepower of the fans required to maintain the desired air flow rates. Finally, L determines the number of fans, the length of the piping required, the total water flow rate and therefore the total pumping power needed to circulate the cooling water. When the dimensions of the fill, the air and water flow rates, the empirical heat-exchange characteristics of the fill, and the temperature of the hot water at the tower inlet are specified, the basic theory of Merkel [9,10,11,12,13,14,15] can be used to calculate the temperature of the cold water

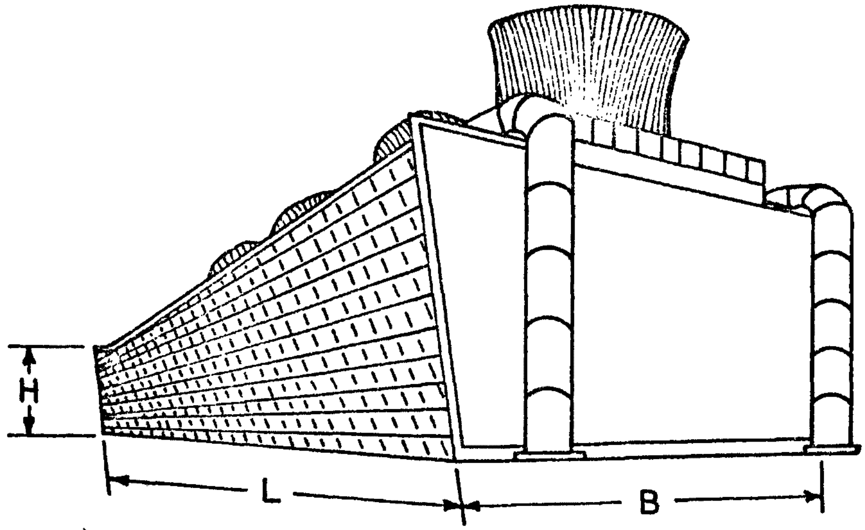

Figure 8. Overall view of typical mechanical-draft tower

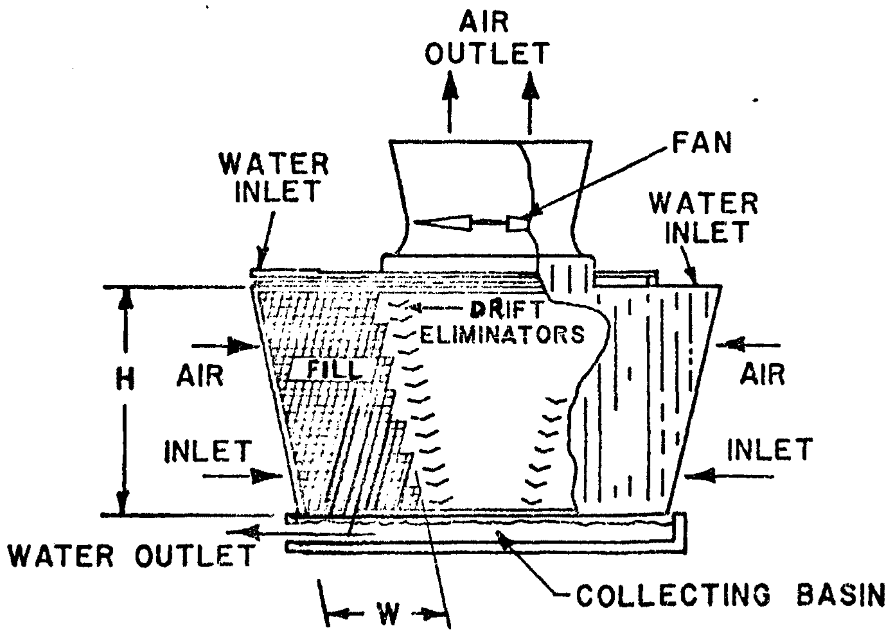

Figure 9. Mechanical-draft crossflow tower

at the tower outlet, the temperature and humidity of the exit air, and the rate of heat rejection from the water to the air. The manner in which such calculations are used to determine the overall performance of a power plant or unit fitted with cooling towers will be discussed in the subsequent sections. For the present, however, it may be noted that the basic calculation scheme is described in detail by Croley, Patel and Cheng [15], and that reference will be made to that work from time to time.

A. CAPITAL COST OF TOWERS (C_{cs}) AND AUXILIARY EQUIPMENT

From the preliminary considerations outlined above it would appear that the capital cost of a mechanical-draft cooling tower will be determined primarily by its size since the fill dimensions H, W and L fix not only the cost of the tower structure itself but also the cost of site acquisition and preparation, water basin, and auxiliary equipment such as fans, motors, pumps, controls and pipes. While this is so, manufacturers of such cooling towers recommend sizing and pricing procedures which bear no direct relation to the physical size of the tower. Instead, the cost of the tower is linked to the "design" meteorological conditions (here, the design wet-bulb temperature) and parameters describing the overall performance of the tower at these design conditions, notably the RANGE and APPROACH. The rating-factor tower-unit method [16] and the K-factor method [17] are examples of such procedures. In the former method, which is the most well publicized, the manufacturers present charts, such as those shown in Figure 10, from which a rating factor can be found for any given range, approach and wet-bulb temperature. The rating factor may be interpreted as the relative degree of difficulty of heat rejection. The product of the rating factor and the water flow rate (GPM, gallons per minute) then gives the "required tower units," i.e.,

$$TU = RF \times GPM \tag{21}$$

The capital cost of the tower, C_{cs}, can be found simply by multiplying

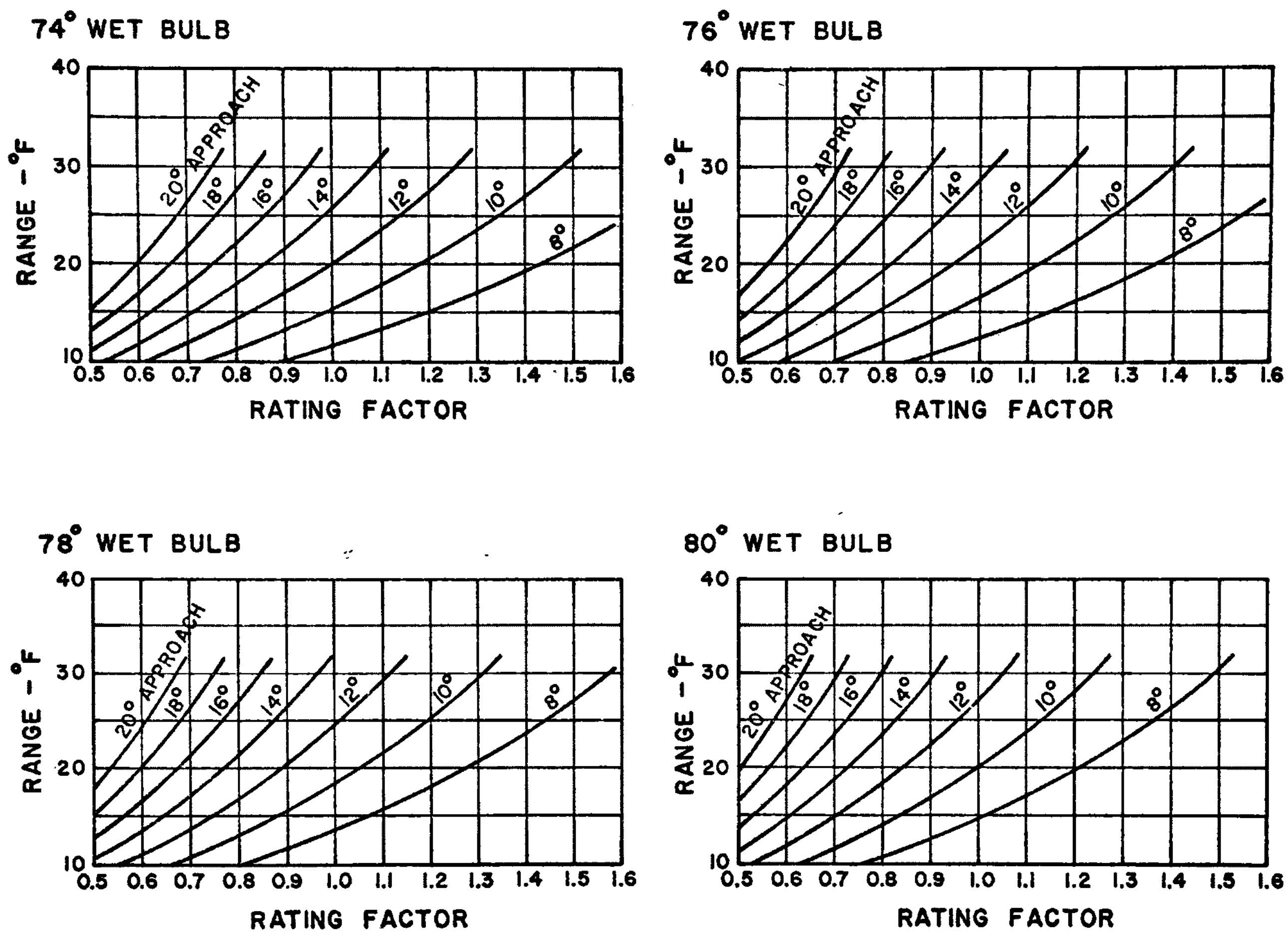

Figure 10. Typical rating factor charts for mechanical-draft crossflow towers [Ref. 16]

the tower units by a unit cost c_t in \$/TU, i.e.,

$$C_{cs} = TU \times c_t \tag{22}$$

From an analysis of previous experience, Dickey and Cates [16] have found that the installed cost per tower unit is of the order of \$7.50 with a scatter of ± 12 percent for 1976 erection. This figure includes the structure, fans and motors, concrete basin with sump, the construction costs, and the necessary electrical components and controls. It is assumed that this cost also includes the cost of hook-up and testing. The scatter of 12 percent in the unit cost was observed for the best 85 percent of data obtained from 22 generating units and represents the influence of site-dependent conditions. It will be noticed that this procedure for the estimation of the capital cost of cooling towers is very simple, and since the unit cost is based on past experience of the industry, it yields realistic results. The rating factor charts can also be used to predict the performance at specific off-design conditions but their use is restricted to the particular class of towers for which they were constructed. This procedure gives no indication of the physical size of the tower structure nor does it indicate the performance of towers which are radically different in internal design.

In order to proceed further and establish a capability for handling towers of different designs it is necessary to return to a more basic approach in which the Merkel theory is used to predict the amount of cooling delivered by a tower fill of given type and dimensions. Such a procedure is described in detail in Ref. 15 and will not, therefore, be repeated here. There it is shown that when the dimensions (L, W, H) and the heat transfer coefficients of the fill are specified it is possible to calculate the cold-water temperature, and therefore the heat rejection rate, range and approach, for any given set of values of the hot-water temperature, air- and water-flow rates and ambient wet-bulb temperature. When the calculations are performed for the design wet-bulb temperature over a range of values of the design heat rejection rates and tower dimensions, and use is made of the rating-factor tower-unit method, it is possible to express the tower units as a function of the tower dimensions as shown in Figure 11. These results were obtained using a known set of heat transfer coefficients, air- and water-loadings on the pile, pile resistance and fan characteristics. A fixed one-side fill width, W = 18 ft (5.49 m), had to be used since the available pile resistance and fan performance data were restricted to that particular value. It should be emphasized that Figure 11 results from a large number of calculations performed using a range of values of heat rejection rate, fill height and fill width, and a number of values of the design wet-bulb temperature. For each set of conditions (Q, L, H, T_{wb_d}), the thermodynamic model of the evaporative pile, described in detail in [15], was used to calculate the corresponding range and approach. This, in turn, was used to find the corresponding rating factor from the charts shown in Figure 10. The total water flow rate, computed from the water loading and the plan area of the fill, was then used in equation (21) to find the tower units corresponding to the specified set of input conditions. For each design wet-bulb temperature and pile height, the number of tower units was found to be a linear function of the pile length, irrespective of the heat rejection rate. A small scatter was observed between the results obtained with different design wet-bulb temperatures which is shown by the shaded area in Figure 11. While the scatter is somewhat consistent, insofar as smaller tower units correspond to lower design wet-bulb temperatures, its origin lies mainly in the fact that a highly complex phenomenon is being represented in a relatively simple form. In any case, the scatter is small and well within the accuracy expected from the various assumptions made in the thermodynamic model of evaporative cooling. The most remarkable feature of Figure 11 is that the number of tower units, and therefore the cost of the tower, is primarily a function of of the dimensions of the fill, as was conjectured earlier.

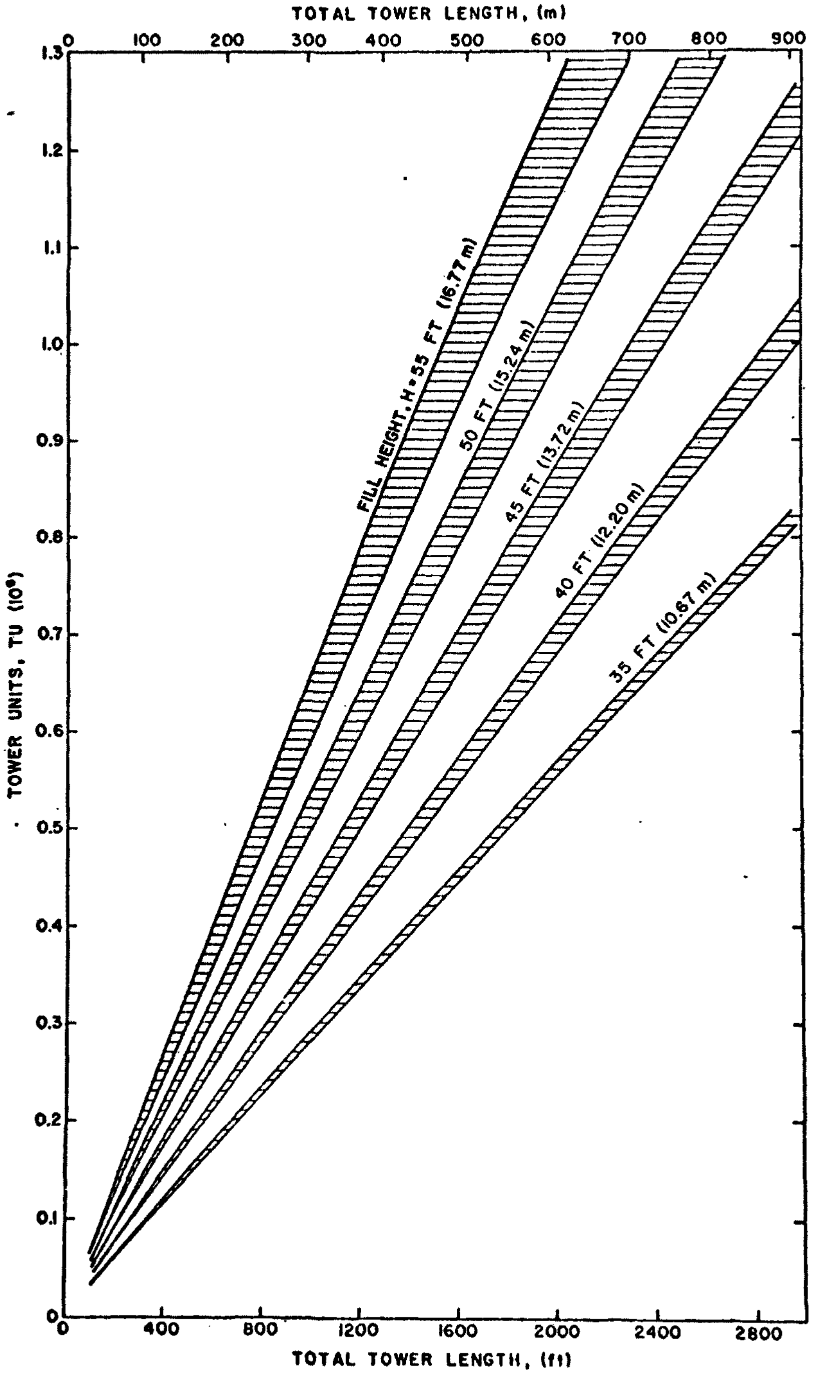

Figure 11. Relation between tower units and physical dimensions of the tower

Thus, for the estimation of the capital cost of mechanical-draft cooling towers, either Figure 10 or Figure 11 can be used, depending upon the information that is known. The range and approach which were calculated as an intermediate step in the development of Figure 11 are shown in Figure 12 for three different design wet-bulb temperatures.

The capital cost of pump and pipe system, C_{pp}, depends primarily upon the total water flow rate (GPM), although some variations will result due to different pumping heights, structure length, distance between the power plant and the towers, and other site-related factors. Figure 13 which is based on the estimates made in Ref. 16, shows the dependence upon the water flow rate. If the water loading on the fill, in gpm or m^3/min per unit plan area, is known, then of course the total water flow rate can be related to the length and width of the tower. Figure 13 also shows the cost of pump and pipe system plotted against the tower length for a water loading of 12.5 gpm/ft^2 (0.509 $m^3/min/m^2$) and a fill width of 18 ft (5.49 m) per side.

When cooling towers are designed for a new power plant, the design is usually optimized in conjunction with the condenser design. The new condenser area A_c required for compatibility with cooling towers is shown in Figure 14 as a function of the tower size and the reference heat rejection rate (for a heat transfer coefficient, U_c = 630 Btu/hr/ft^2/°F). The capital cost C_c of new condensers can then be found from

$$C_c = A_c\, c_c \tag{23}$$

where c_c is a unit cost. In an example, Dickey and Cates [16] use an installed value of \$4.00/ft^2 (\$43.00/m^2) for c_c for 1976 construction, and that figure is utilized in the present study even though it appears low compared to other sources. The information presented in Figure 14 can be used if new condensers are considered in the retrofit situation. In that case, the differential cost to be charged against the project will be the difference between C_c and any salvage

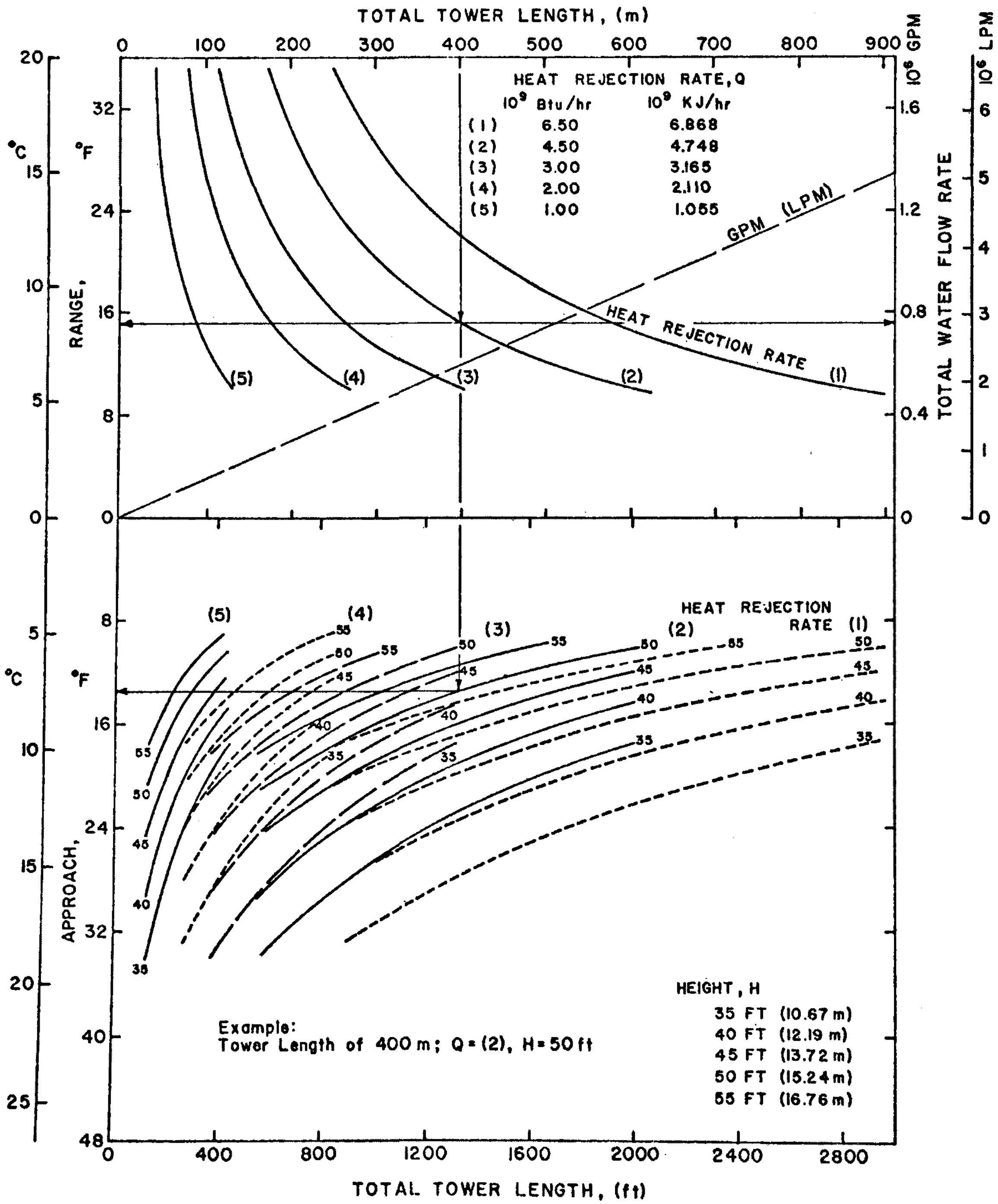

Figure 12(a). Range, approach and water flow rate as functions of tower size, $T_{wb_d} = 60^oF$ (15.56 °C)

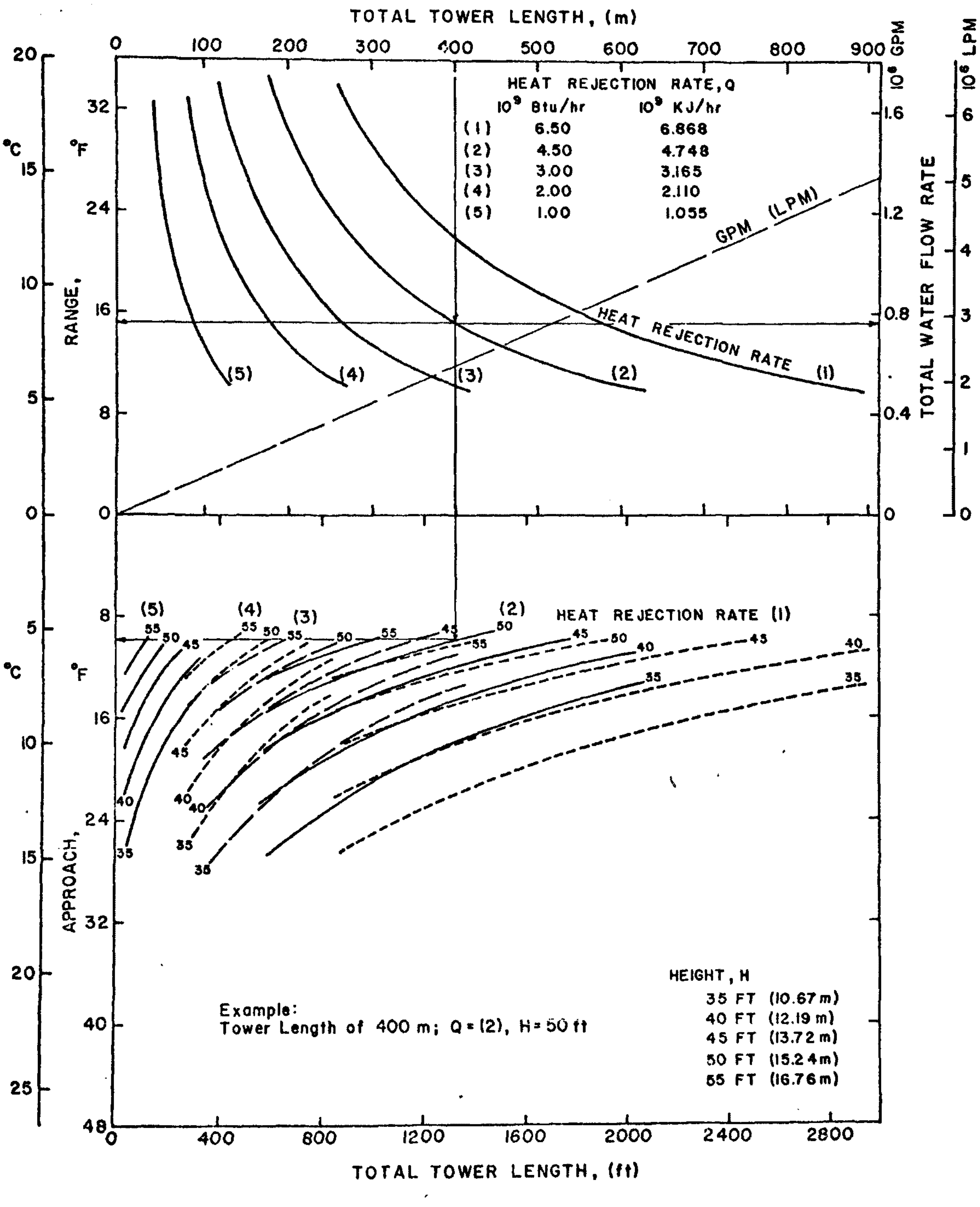

Figure 12(b). Range, approach and water flow rate as functions of tower size, $T_{wb_d} = 70^{o}F$ (21.11 °C)

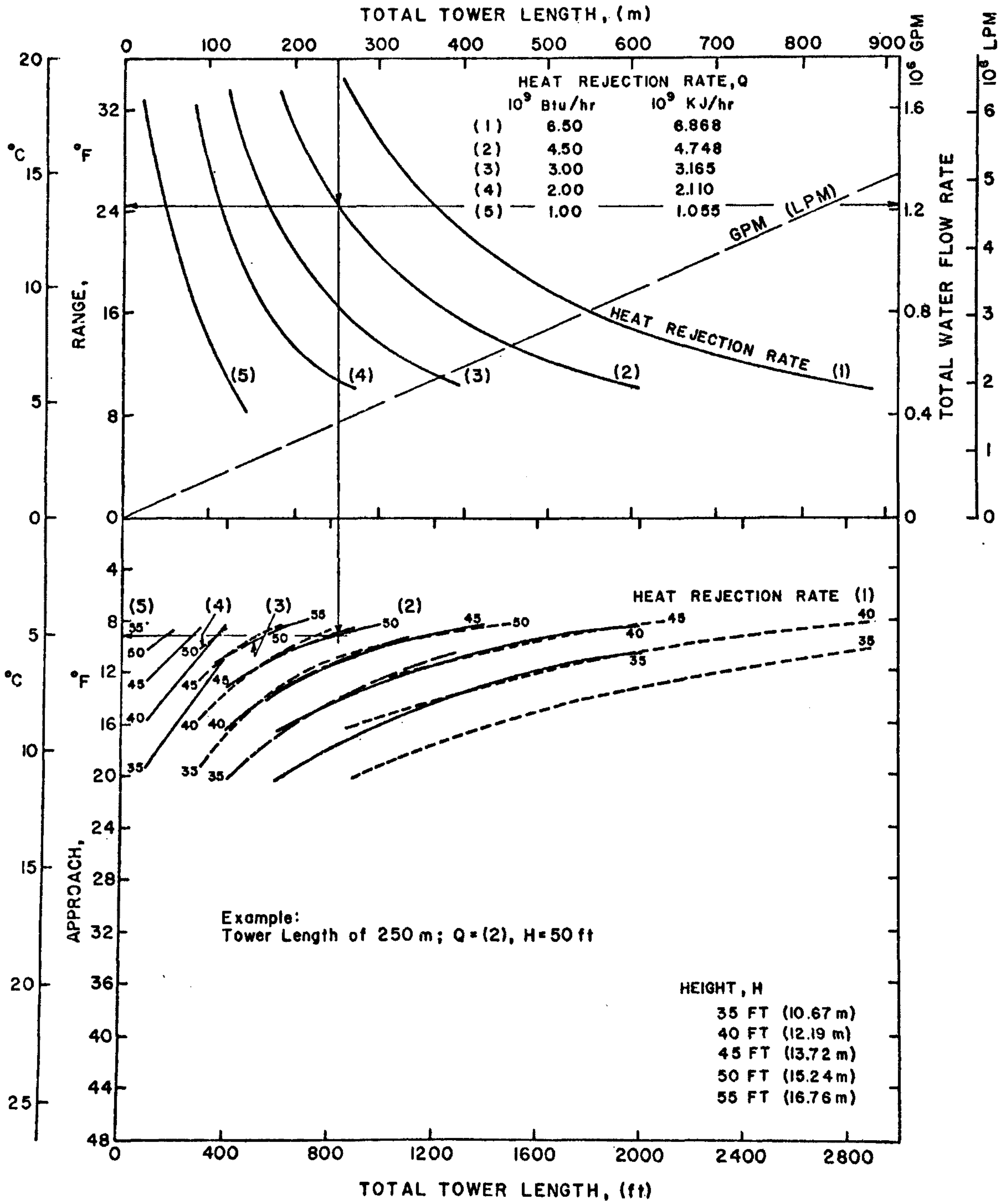

Figure 12(c). Range, approach and water flow rate as functions of tower size, $T_{wb_d} = 80^{o}F$ (26.67 °C)

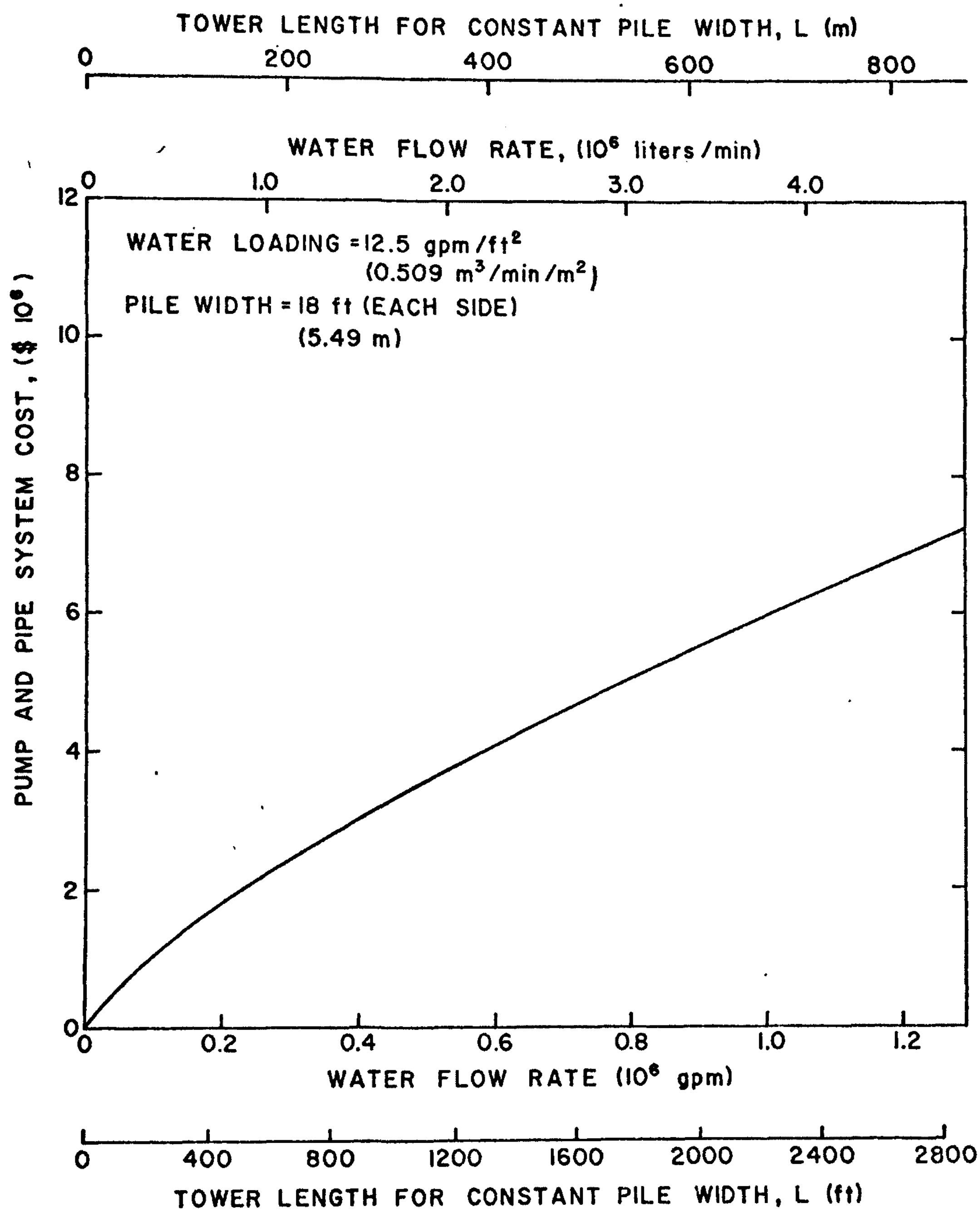

Figure 13. Capital cost of pump and pipe system

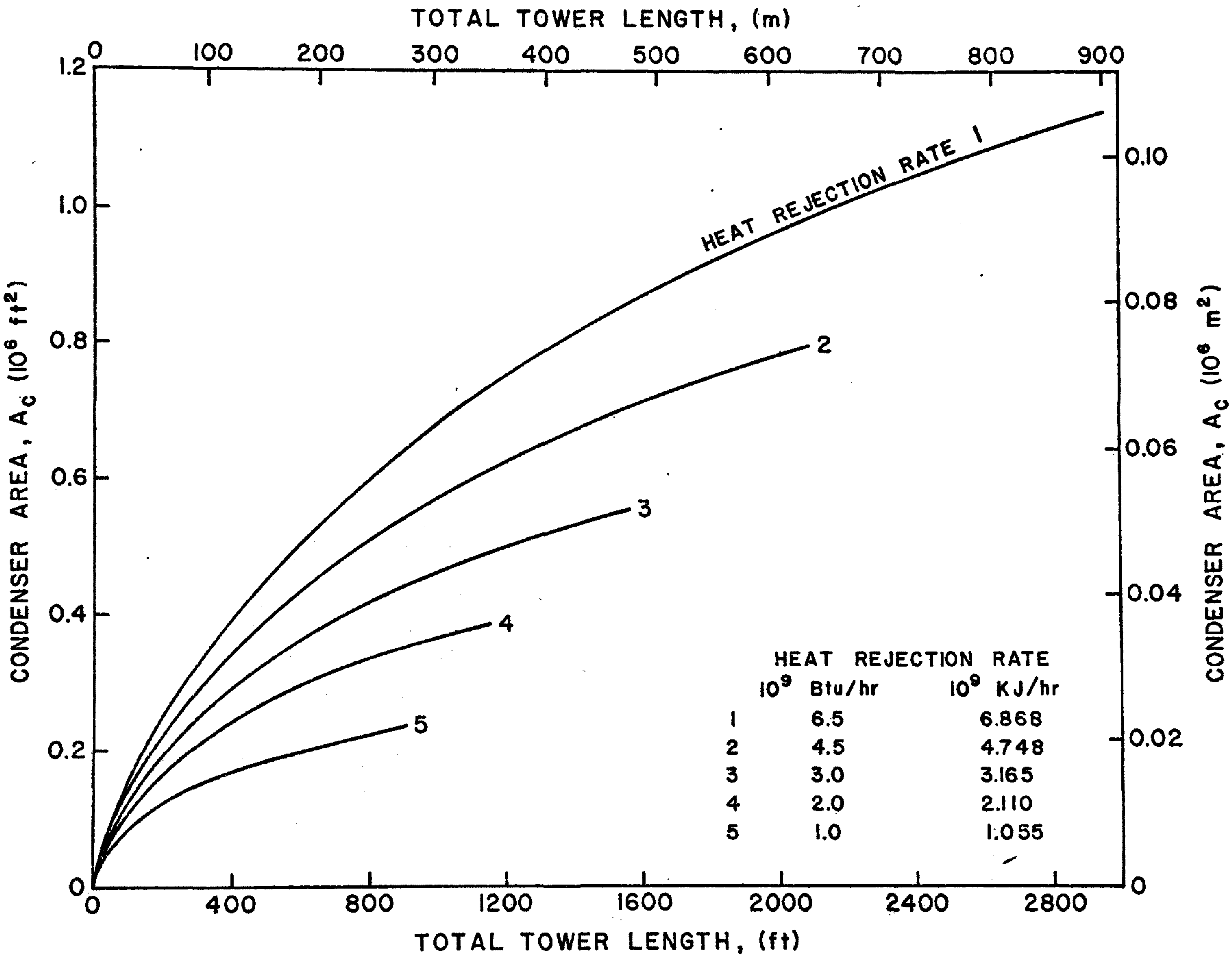

Figure 14. Required surface area of new condensers

value C_c' of the existing equipment. If the old condensers are to be retained, however, the tower sizes which can be employed in any particular application will be constrained by the temperature rise and water flow rate that can be tolerated by the old condensers.

The additional land area required for backfitting with a mechanical-draft cooling tower depends upon the plan area of the tower and upon considerations of interference with adjacent towers and neighboring structures, plume recirculation, and fan noise. The problem of recirculation is primarily dependent upon meteorological conditions and tower length [2, p. 630]. Minimizing recirculation, therefore, depends more upon tower orientation with respect to wind direction and tower design than upon the land area. One exception is the case when a long tower is split into multiple smaller units. In that case sufficient land should be available for adequate spacing of the towers to avoid interference.

The EPA Development Document [2, p. 628] suggests that from 100 to 200 ft (30.5 to 61 m) of clearance is required around a single mechanical-draft cooling tower to avoid interference. If two or more towers are needed, tower separation should range from 400 to 600 ft (122-183 m). Based on this criterion, the required land area, A_L, for a single mechanical-draft wet cooling tower of length, L, and breadth, B, may be expressed as

$$A_L = BL + 2D(B+L) + 4D^2 \tag{24}$$

where D is the width of the clear area around the tower. Other land requirement critera are given in Ref. 2 (p. 631) based upon power plant size. In a Federal Power Commission survey, a land requirement of 1000 to 1200 sq. ft (93 to 112 sq. m) per megawatt, including area for spacing, is mentioned.

Additional land area determined from the standpoint of acceptable noise levels may be necessary, particularly in populated regions. A detailed study of this criterion may be found in Ref. 7 (Vol. I, Appendix G). In the hypothetical test case presented in part H of this Section, additional land area is computed on the basis of a noise level limit of 60 dBA; the specific land area thus required is 0.1 acre/MW (0.04 hectare/MW). It is readily seen that land area requirement based on this noise level is approximately four times larger than that based on equation (24) with D = 200 ft (61 m) and that problems of interference and recirculation can be handled adequately. The estimates based on noise apply only to situations where the availability of land is not a major problem. In many backfit cases, however, limitations of available space may dictate the use of noise suppression devices.

Since criteria for the determination of additional land area needed for backfitting differ so much, the specific land area requirement, in acres/MW or hectares/MW, is left as a variable parameter in the present study.

The total differential capital cost of the closed-cycle cooling system, CC_S, can now be expressed in the form

$$CC_S = C_c - C'_c + C_{pp} - C'_{pp} + C_{cs} - C'_o + A_L a_\ell + C_{HT} \qquad (25)$$

where CC_S = differential capital cost of cooling towers

C_c = capital cost of new condensers (see equation (23) and Figure 14)

C'_c = salvage value of old condensers ($C_c - C'_c = 0$ if old condensers are retained)

C_{pp} = capital cost of pumps and piping (see Figure 13)

C'_{pp} = salvage value of pumps and piping used in open-cycle cooling

C_{cs} = capital cost of towers, including tower structure, fans, motors, controls, basin; installed cost (see Figure 10 or Figure 11)

C'_o = salvage worth of old system components, excluding condensers, pumps and piping

A_L = land area required for towers [see equation (24) or Ref. 7 (Vol. I, Appendix G)]

a_ℓ = unit land cost

C_{HT} = cost of hook-up and testing of towers

All cost figures used in this study are based on 1974 values unless otherwise specifically stated. If estimates of inflation, labor costs, construction costs, material costs, etc. can be made, standard methods of proration may be used to project costs to a future date.

As explained in the previous section, the total capital cost to be assessed against backfitting consists of the cost of the cooling system considered above, the cost of downtime CC_{DT} which has already been discussed, and the capital cost of replacement capacity CC_R. The evaluation of the last quantity is considered in later sections.

B. REFERENCE LENGTH OF COOLING TOWERS, L*

A number of quantities which characterize the operation of an existing power plant or unit using open-cycle cooling were defined in the previous section. For a specific turbine (A, B, or C; Table 1), P* is the rated or nameplate capacity which is obtained at the reference back pressure p^* where the excess turbine heat rate Δ is zero, and the corresponding heat rejection rate is Q^*. Here, it is useful to define a reference size of the cooling towers. For any given pile height H and pile width W, the reference length of tower L^* can be defined as the length required to remove Q^* at some reference ambient wet-bulb temperature while maintaining the back pressure at p' which can be selected arbitrarily without loss of generality. For the example calculations, L^* is determined holding the pile width, W, constant at 18 ft (5.49 m) for reasons discussed earlier; the reference wet-bulb temperature is set equal to 60°F (15.6°C), and p' is taken as 2 in. Hg abs (5.08 cm Hg abs).

It is clear that L^* can be found for any given set of values of Q^*

and H using the theory of Merkel in conjunction with the known heat transfer properties of the condenser and the evaporative pile, and the air- and water-loadings recommended by tower manufacturers. A computation construction which has proved useful in calculating the reference length L^* is as follows. In the thermodynamic model, the turbine characteristics curves (as in Figure 2) are replaced with a vertical line (corresponding to the heat rejection rate Q^*) as in Figure 15. The operation of the cooling system (with condenser) then corresponds strictly to rejection of Q^* (regardless of what turbine it is rejected from or what power level or throttle opening is being used). Rejection of Q^* from any specified cooling system occurs at a unique set of values of hot-water temperature and cold-water temperature (and thus corresponding steam temperature and pressure) for a specified design wet-bulb temperature. By repeating the calculations for a given value of Q^* and several different cooling system sizes, at a specified wet-bulb temperature, the system size that corresponds to the specified back pressure, p', for the given heat rejection rate, Q^*, can be determined. This cooling system size is the reference system size (L^* for mechanical-draft crossflow wet cooling towers) used to nondimensionalize succeeding example operation results. With an air-loading of 1800 lb/hr/ft^2-face area (8790 kg/hr/m^2-face area), water-loading of 12.5 gpm/ft^2-plan area (0.509 m^3/min/m^2-plan area), and typical (proprietary) information concerning the heat transfer properties, the dependence of L^* on Q^* and H is shown in Figure 16. It will be seen that the reference length is a nearly linear function of Q^* which decreases with increasing H for a constant Q^*. In what follows, the reference length L^* will be used to normalize the tower size so that the example results can be used to assess the performance of power plants or units with different nameplate capacities and heat rejection rates. It is perhaps useful to emphasize that the foregoing considerations apply regardless of the type of turbine that is employed since the definition and evaluation of L^* is independent of the source of Q^*. The definition of L^* thus depends on the values of wet-bulb temperature and p' used in the definition. It makes no difference

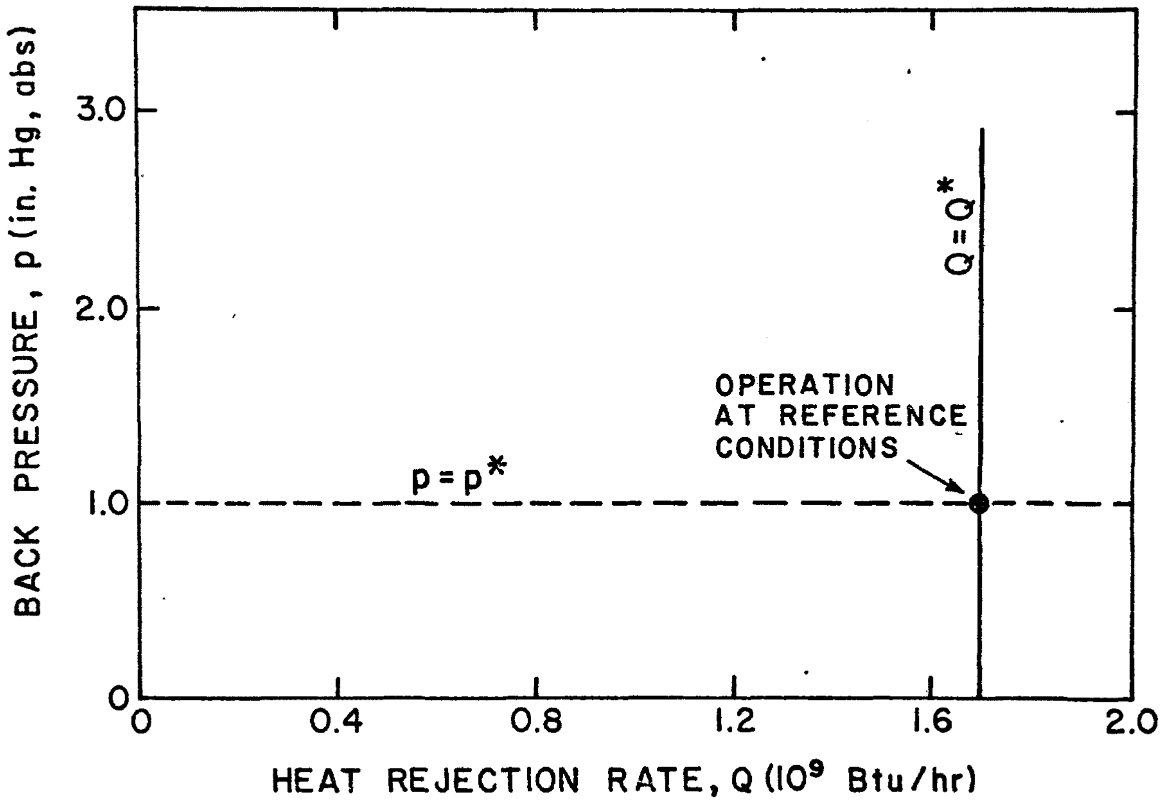

Figure 15. Construction for the definition of reference tower length, L^*

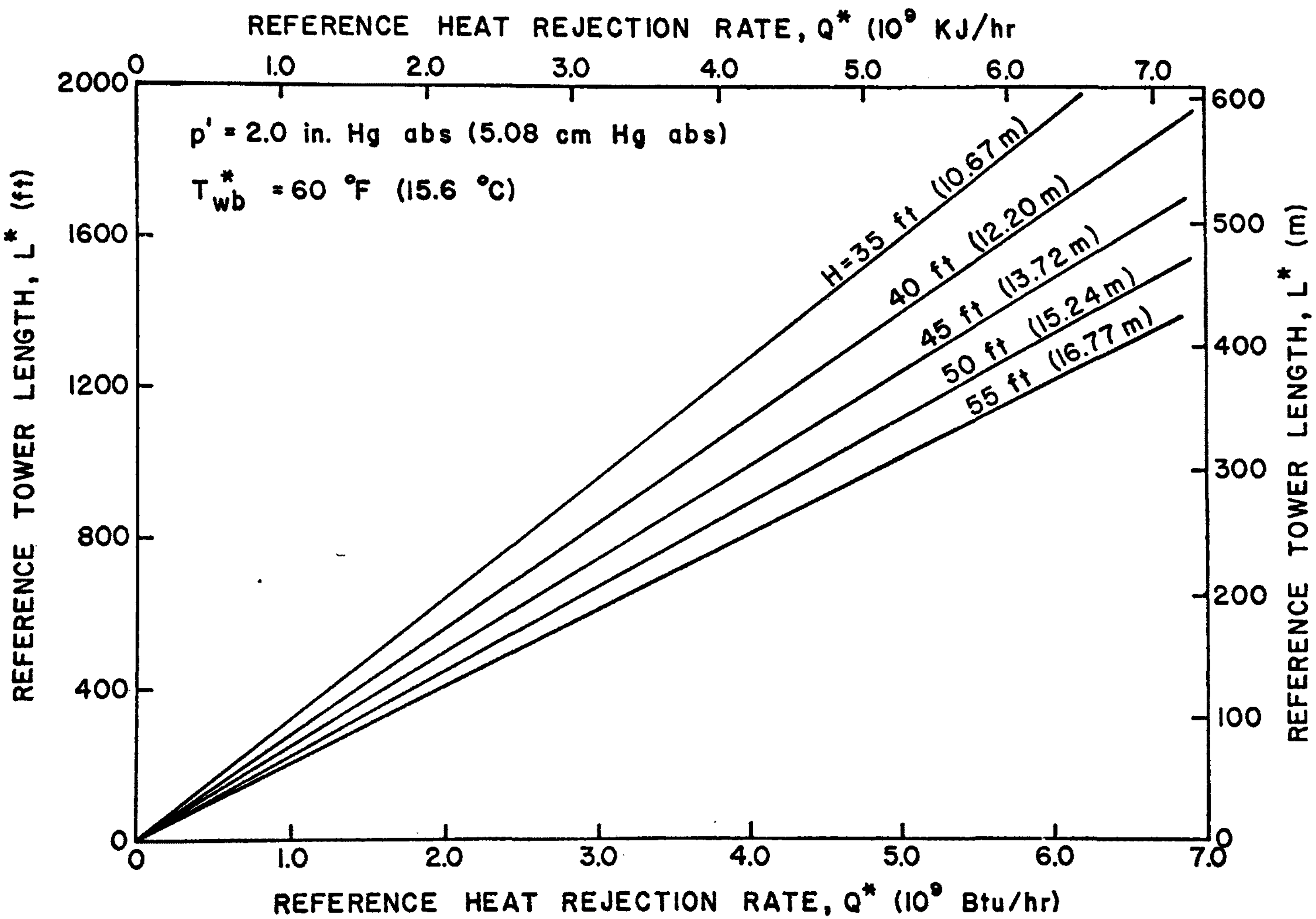

Figure 16. Reference length of towers

which reference temperature or back pressure is used in defining L^* as long as all such references are consistent. As mentioned, the reference length is defined for a 60°F (15.6°C) reference wet-bulb temperature and p'=2 in. Hg abs (5.08 cm Hg abs). All calculations of L^* consistently use these same references. Thus, an L^* from Figure 16 is suitable for "dimensionalizing" results obtained from succeeding nondimensional plots.

C. OPERATION OF A TOWER OF GIVEN SIZE (L, H)

When the width of the pile W is fixed, the physical size of the cooling tower is characterized by only two parameters, namely the length L and the height H. In turn, the length determines the number of fans required, while the plan area (2WL) and the face area (2HL) determine the total water and air flow rates when the appropriate water and air loads per unit area are prescribed. For the example calculations presented herein, the air- and water-loadings mentioned in the previous paragraph and proprietary heat transfer properties of the pile are used. For different loadings the results can be expected to change; however, these data are characteristic of currently manufactured cooling towers and are expected to be representative of available units.

In this part, consideration is given to the operation of a cooling tower of a given size (L and H) in conjunction with a turbine whose performance characteristics are known. Thus, it is assumed that the following quantities are prescribed:

1. Nameplate capacity, P^* (kW), (see Sections IV.D and V.J for variations in loading pattern)
2. Reference heat rejection rate, Q^* (Btu/hr or kJ/hr),
3. Turbine heat rate correction curves, Δ vs. p (as in Figure 1),
4. Frequency of occurrence of dry- and wet-bulb temperatures (as in Table 3),
5. The size of the cooling tower, L and H (ft or m).

Then, items 2 and 3 can be combined to obtain the heat rejection rate characteristics of the turbine in the form shown in Figure 2 using the procedure described in the previous section. The information in item 4 can be used to find the various design temperatures and also the extreme temperatures which are not exceeded or equalled more than 10 hours per year. The basic theory of Merkel can then be used to find the turbine back pressure p which will occur at each set of values of the dry- and wet-bulb temperatures. These calculations again involve a certain amount of iteration since the rate of heat rejection from the turbine must be balanced by the cooling capacity of the towers. It is also necessary to assume the performance characteristics of the condensers so that the temperature of the hot water entering the tower can be related to the steam condensing temperature corresponding to the back pressure p. The detailed procedures adopted and the computer programs developed to accomplish such calculations are described in Ref. 15 (see Sections III.B, E, and F) and will not be presented here. It should be noted, however, that these steps are included in the major program listed and described later for the analysis of the backfit situation.

For each set of dry- and wet-bulb temperatures, these calculations identify a corresponding operation point on the turbine characteristics curves shown in Figure 2. Consequently, it is possible to determine all quantities of interest including the back pressure p, heat rejection rate Q, power output P, the rate of evaporation of water from the tower, the hot-water temperature, the cold-water temperature, the range and approach, and the power required by the fans and pumps. When such calculations are performed for all possible combinations of the dry- and wet-bulb temperatures occurring at the site, it becomes possible to evaluate the following (these definitions apply for the assumed "full-throttle" power loading; for consideration of variations in power loading related to meteorological conditions, see Section V.J):

(a) The maximum capacity loss, C_L is given by

$$C_L = P^* - P_{min} + P_{cs} \quad , \quad kW \tag{26}$$

where P_{min} is the gross output from the turbine at the extreme temperatures $\hat{T}_{db}$, $\hat{T}_{wb}$ which are not exceeded or equalled more than 10 hours per year, and P_{cs} is the power required to operate the pumps and fans.

(b) If the time duration (in hours) of each set of meteorological conditions (T_{db}, T_{wb}) is Δt, then the annual loss of energy, E_L is given by

$$E_L = \sum (P^* - P + P_{cs})\Delta t \quad , \quad kW\text{-}hr \tag{27}$$

where the summation is carried out over all sets of T_{db}, T_{wb}. Note that $\Delta t = f(T_{db}, T_{wb}) \times 8760$ hrs, where f is the frequency of joint occurrence of the temperatures T_{db}, T_{wb}.

(c) Similarly, the difference between the annual fuel consumption using cooling towers and that with open-cycle cooling F_E is given by

$$\eta_I F_E = \sum (cP + Q - cP^* - Q^*)\Delta t \quad , \quad kW\text{-}hr \tag{28}$$

As explained in section IV, the contribution to this quantity will be zero during periods of full-throttle operation, $T_s = 1$, and negative when the turbine is throttled back, $T_s < 1$.

(d) The annual water loss due to evaporation from the towers is given by

$$W_L = \sum c_1 (w_o - w_i) G \, \Delta t \quad , \quad m^3/\text{year} \tag{29}$$

where w_i and w_o are, respectively, the absolute humidities of the air entering and leaving the tower (in kg water/kg dry air), G is the total air flow rate through the tower (in kg/hr) and c_1 is a numerical conversion factor (= 0.001 m^3 water/kg water). The theoretical development of equation (29) and the method used to calculate w_i and w_o are described in Ref. 15 (Section III.G).

D. PARAMETRIC STUDIES

Detailed calculations of the type described above can of course be performed for a range of values of the tower length and height. Figures 17 through 20 show the variations, with tower size, in the maximum capacity loss C_L, the annual energy loss E_L, the annual fuel penalty F_E, and the annual water loss due to evaporation W_L, for the particular case of turbine A (whose $P^* = 411$ MW, $p^* = 1.00$ in. Hg abs = 2.54 cm Hg abs, and $Q^* = 2.545 \times 10^9$ Btu/hr = 2.686×10^9 kJ/hr for fossil-fuel operation, see Table 1) with the meteorological conditions at Los Angeles (Tables 4 and 5) for the assumed "full-throttle" loading. Also shown in these figures are the results obtained for a hypothetical turbine whose nameplate capacity and reference heat rejection rate are twice those of turbine A (i.e., $P^* = 822$ MW, $Q^* = 5.090 \times 10^9$ Btu/hr = 5.372×10^9 kJ/hr) but whose basic heat rate characteristics are the same as those of turbine A (Figure 4). For consideration of variations in power loading, see Section V.J. The following important observations can be made from these results:

(a) The range of values of tower heights and lengths considered here were dictated by the guidelines on practicable configurations suggested by the manufacturers of conventional equipment.

(b) From Figure 17 it will be seen that the maximum capacity loss C_L varies markedly with tower size and Q^*. For a given tower height and Q^*, C_L decreases rapidly with increasing length, reaches a minimum and then increases slowly. The high values of C_L at the smaller lengths arise primarily due to the maximum back pressure limitation, requiring the

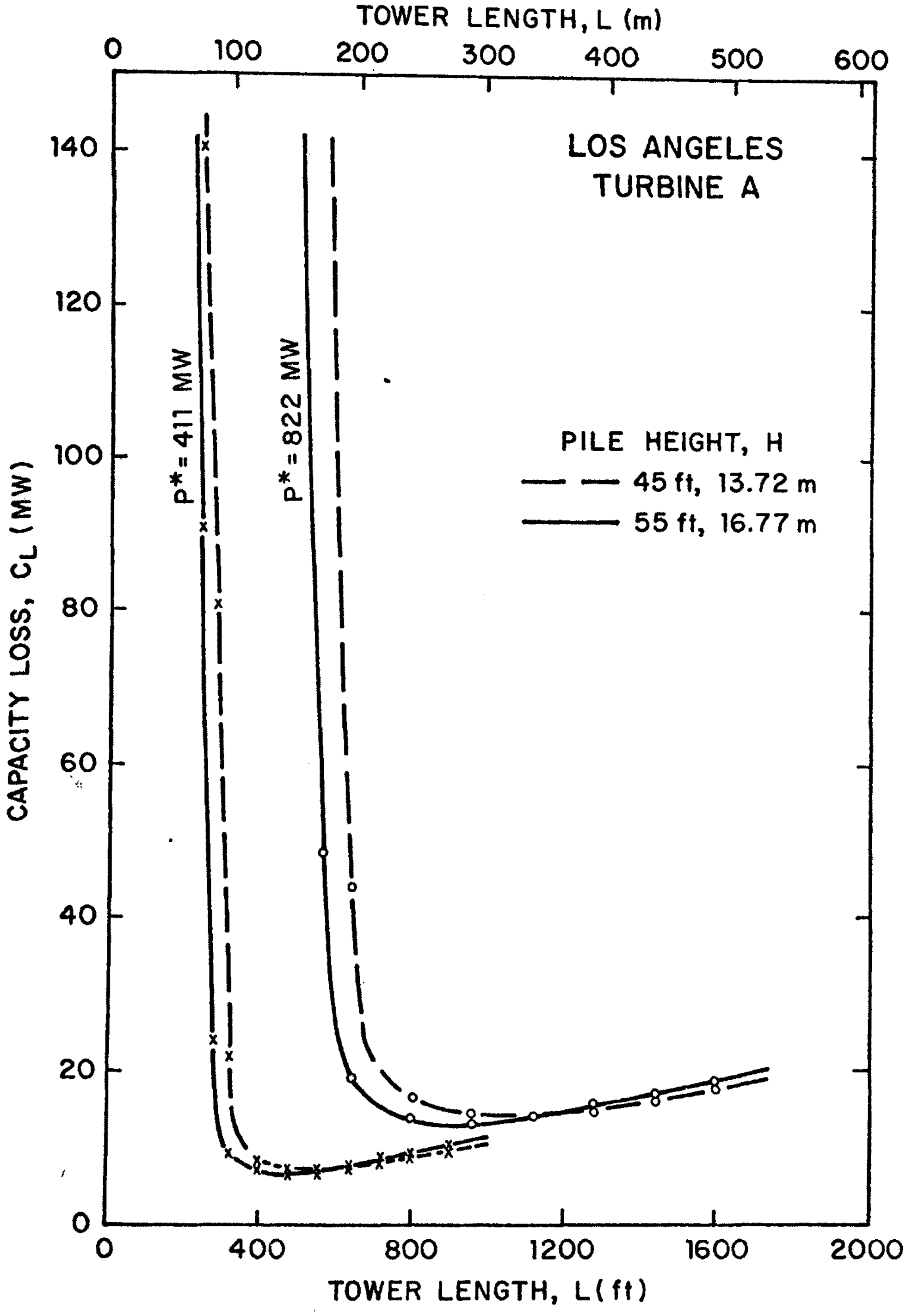

Figure 17. Variation of capacity loss with tower size and plant capacity

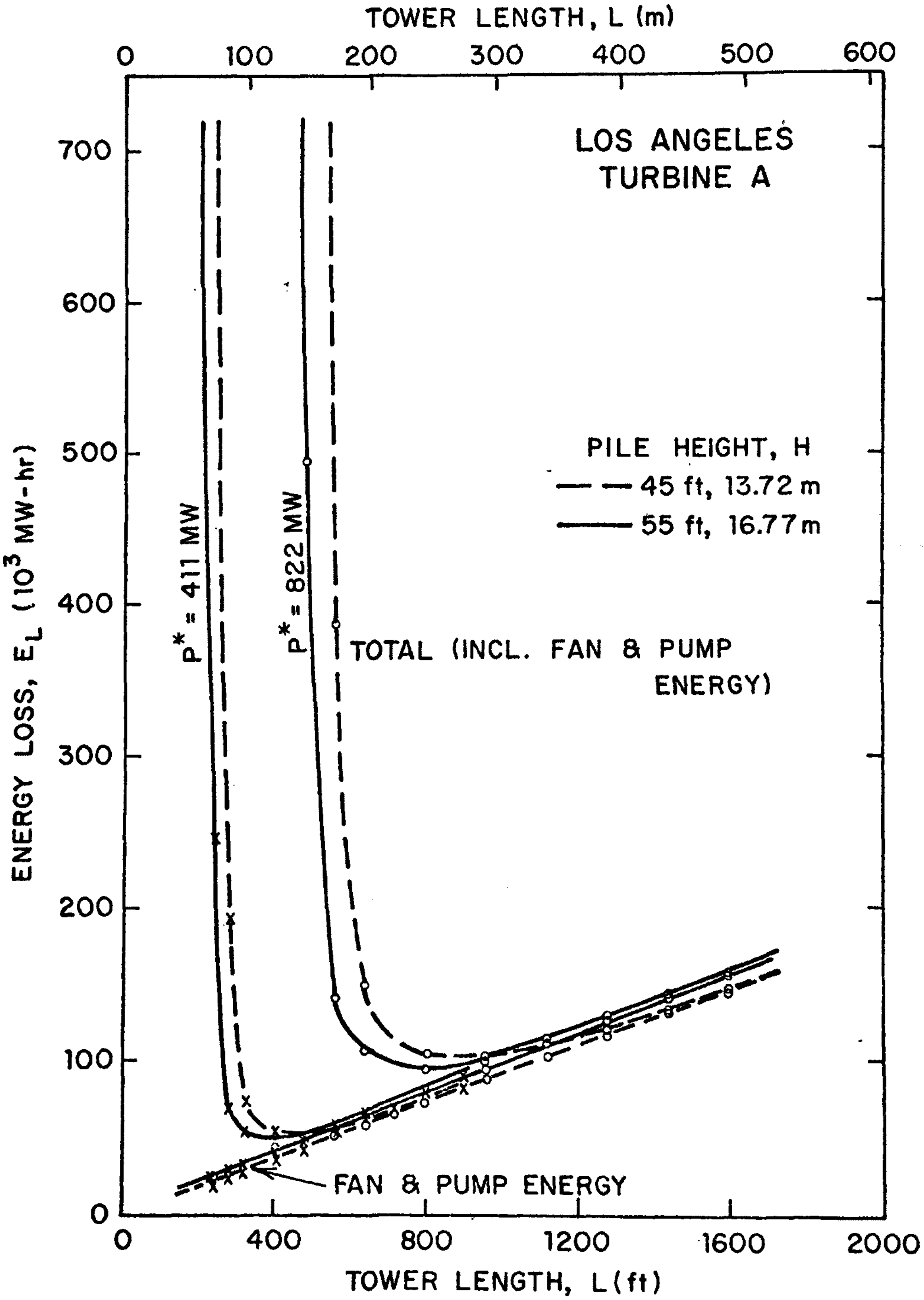

Figure 18. Variation of energy loss with tower size and plant capacity

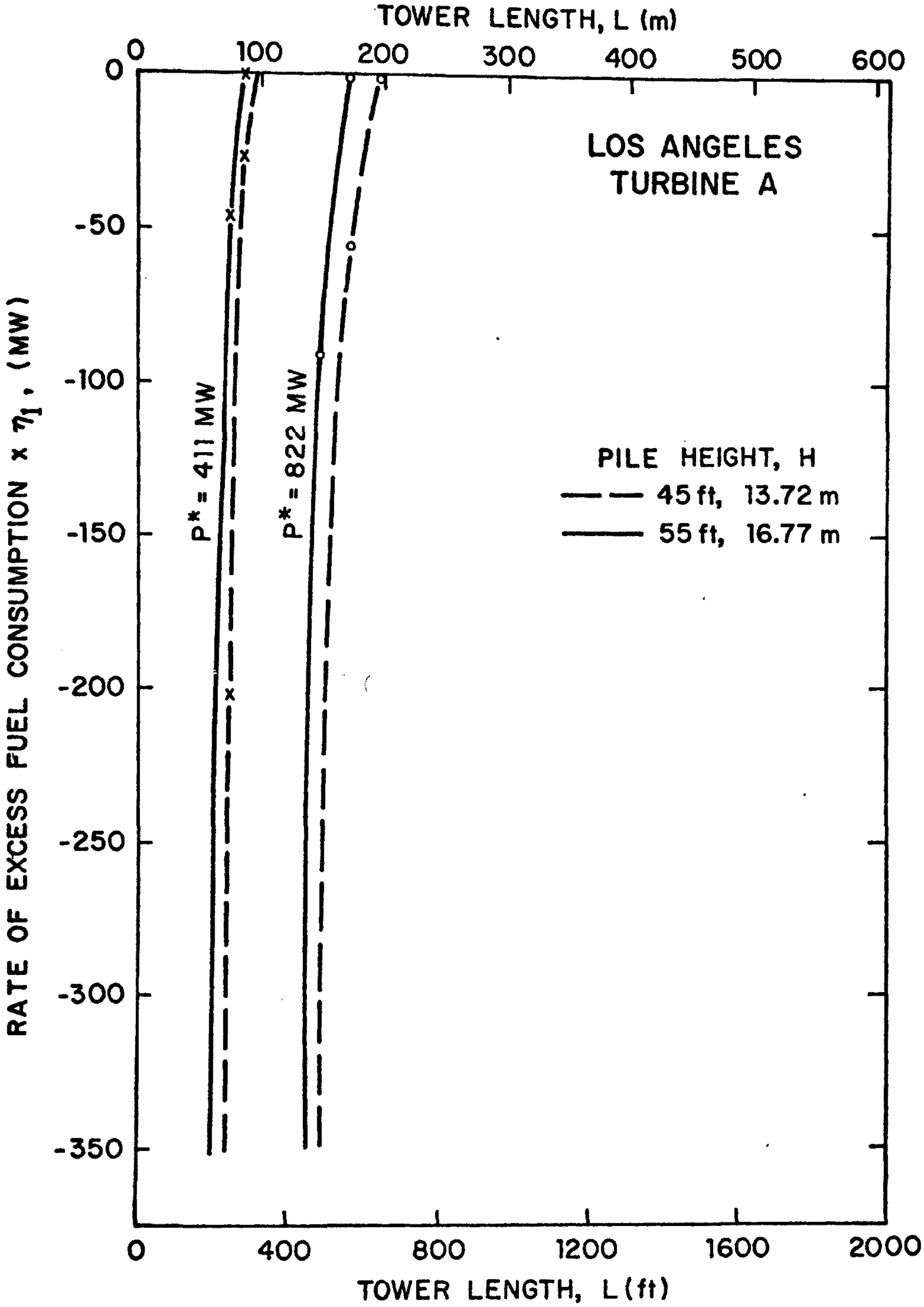

Figure 19. Variation of excess fuel consumption rate with tower size and plant capacity

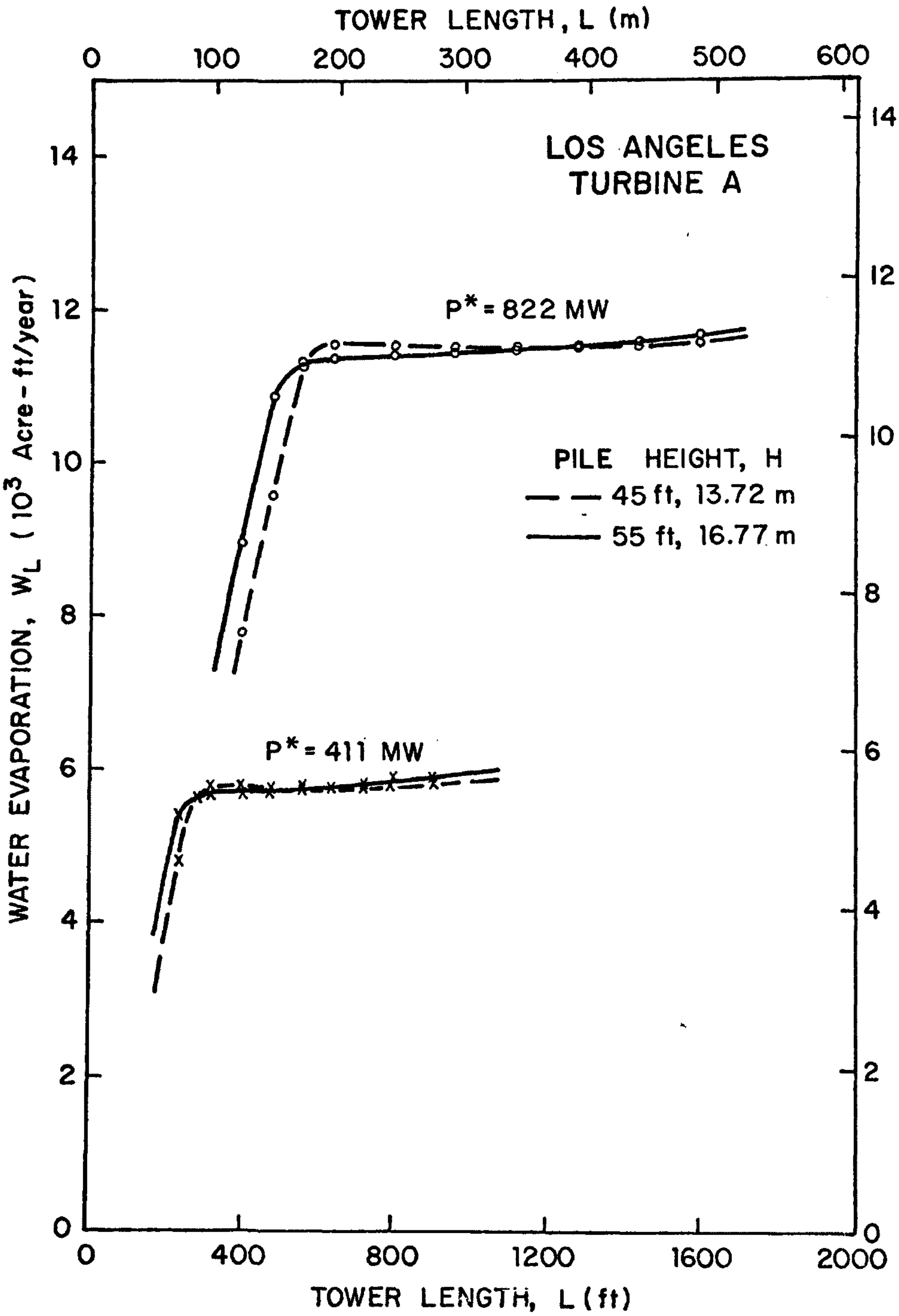

Figure 20. Variation of evaporation with tower size and plant capacity

turbine to operate at less than full-throttle to maintain $p = p_{max}$ during periods of severe meteorological conditions. The increase in C_L at larger values of tower length, on the other hand, results primarily from the increase in the pump and fan power required to operate the larger towers.

(c) Figure 18 shows that the behavior of the annual energy loss, E_L, is similar to that of the maximum capacity loss. The reason for this is obvious from equations (26) and (27).

(d) Figure 19 indicates that, for given P^* and Q^*, there is a range of tower sizes over which the turbine operates at full throttle at all sets of meteorological conditions so that the annual fuel consumption is the same as that in open-cycle operation. For smaller towers, however, the reduced throttle operation during periods of severe temperatures implies that the fuel consumption will be smaller than with the open-cycle cooling system.

(e) Figure 20 shows that, for fixed P^* and Q^*, the annual water evaporation increases with tower size. This is mainly due to the larger water flow rates and smaller cooling ranges associated with the larger towers.

(f) Comparison between the results obtained with the two values of the nameplate capacities, and corresponding reference heat rejection rates indicates that the precise values of the capacity and energy losses, excess fuel consumption, and water loss due to evaporation are dependent upon the SIZE AND TYPE of the power plant or unit that is considered, even though the distribution of the meteorological conditions and the turbine heat rate characteristics may be identical. In other words, calculations of the type shown in Figures 17 through 20 need to be repeated for any specified values of P^* and Q^*. These calculations can of course be accomplished by means of the computer program which was developed and used to obtain the example results. Fortunately, however, it turns out that the usefulness of these results can be greatly enhanced if they are rendered "nondimensional" by employing suitable "scaling parameters." Thus, if the tower length is normalized with respect to the reference length L^* (which, as shown in Figure 16 is a function of the pile height and Q^*), the capacity loss is normalized with respect to the nameplate capacity, and the energy loss and excess fuel consumption are normalized using the maximum energy (PE^*) that can be produced in a year, i.e. (8760 hr/year) $\times$ P^* (kW), then the two sets of results shown in Figures 17 through 20 can be plotted as shown in Figures 21 through 24. The problem of the water evaporation is somewhat difficult since there is no suitable reference value that can be used. Various alternatives, such as evaporation per unit flow rate, or evaporation per unit nameplate capacity, were attempted, but it was found that best results were obtained by defining a "specific rate of water evaporation" by W_L/Q^*, i.e. evaporation per unit reference heat rejection rate. This quantity is of course not dimensionless. However, Figure 24 shows a plot of W_L/Q^* vs. L/L^* for the two sets of results given in Figure 20. It will be seen from Figures 21 through 24 that there is a remarkable coincidence between the results obtained with the two different sets of values of P^* and Q^*. The major implication of this is that for a given set of turbine heat rate characteristics and meteorological data there is no economy of scale in the detailed operation of a particular type of turbine. A similar collapse of the results was also observed when the calculations were repeated with the same value of P^* (411 MW) but a different, higher value of Q^* (3.010×10^9 Btu/hr = 3.176×10^9 kJ/hr) corresponding to a nuclear unit.

It will be recalled that the foregoing discussion applies to the re-

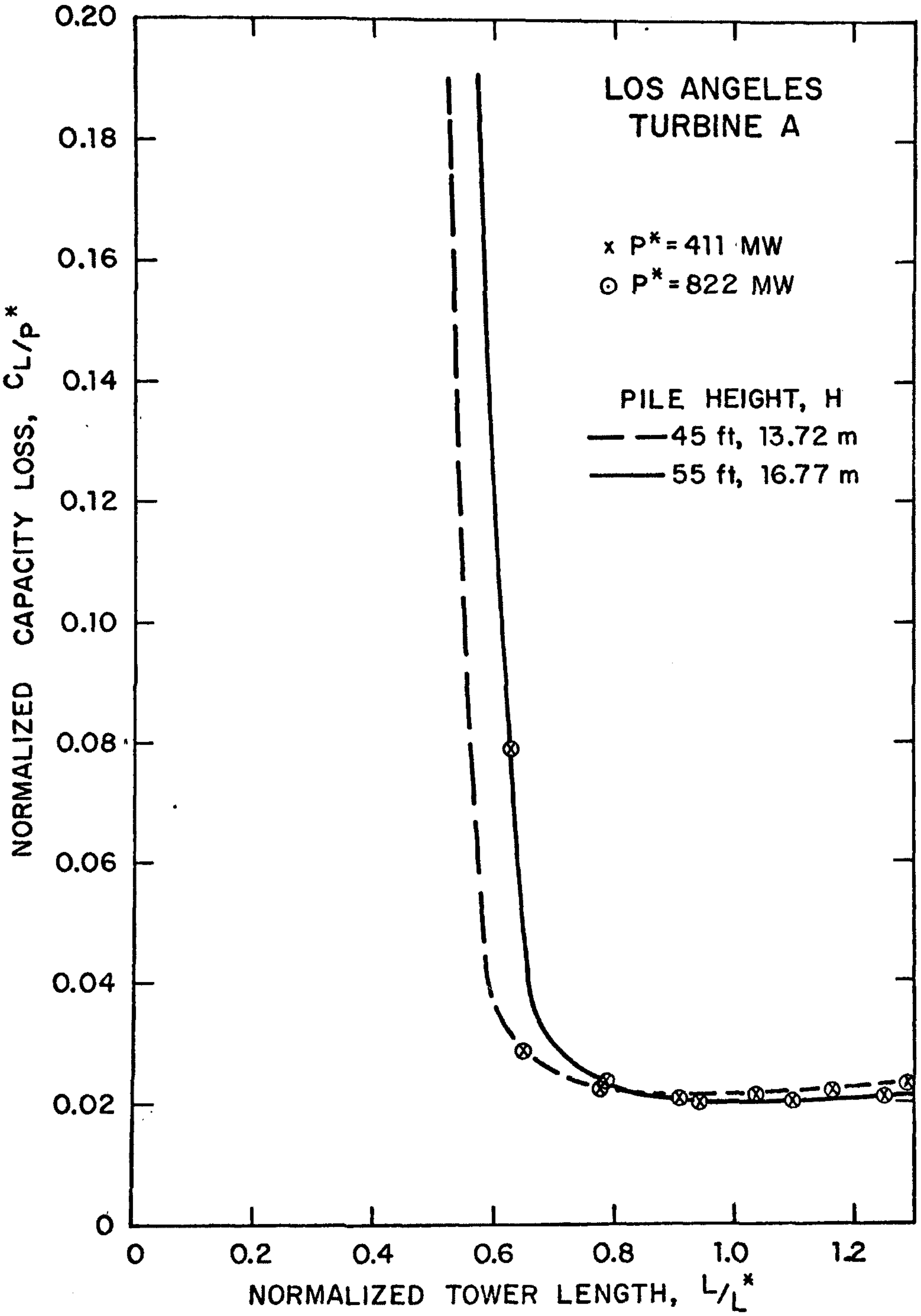

Figure 21. Normalized capacity loss

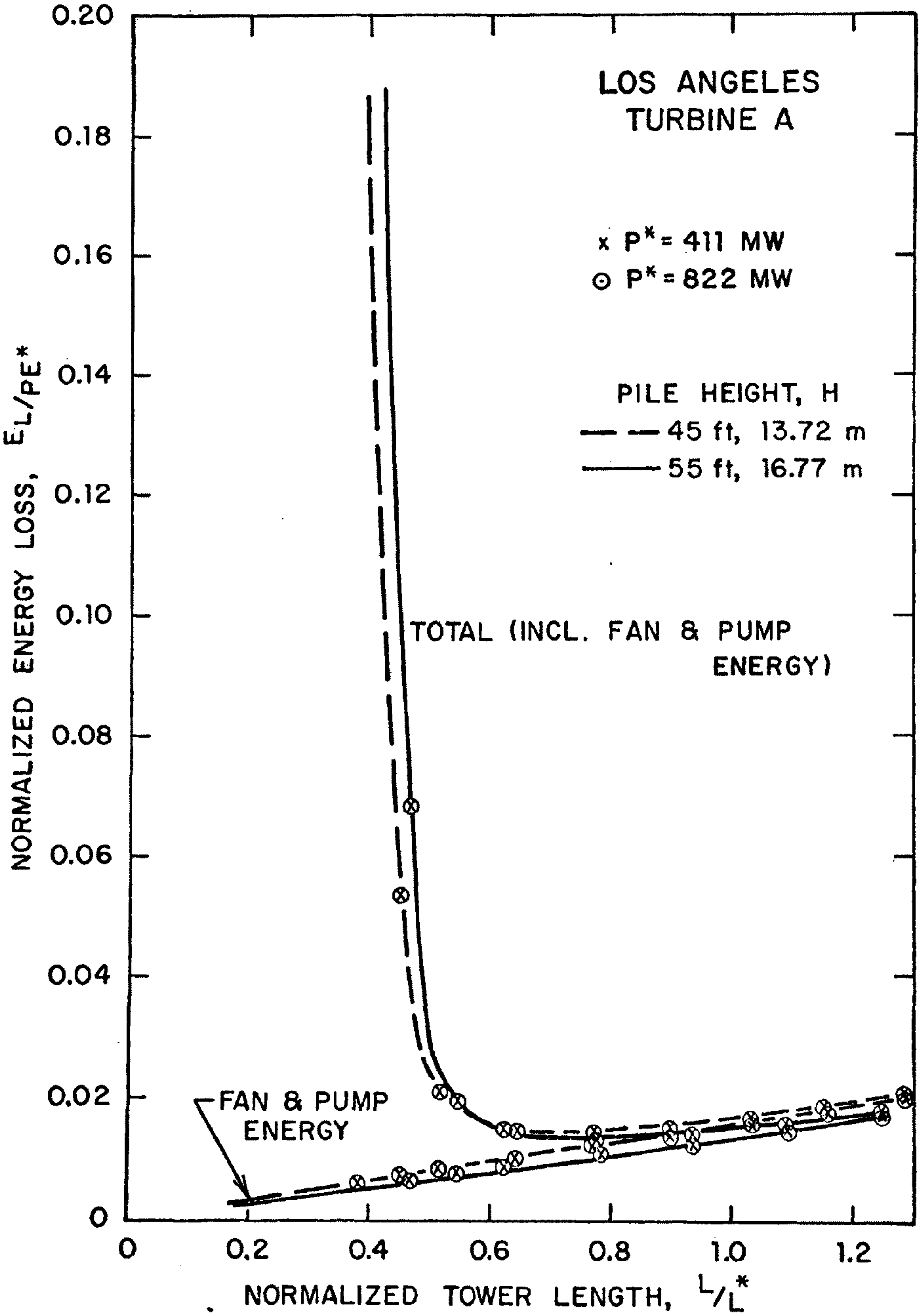

Figure 22. Normalized energy loss

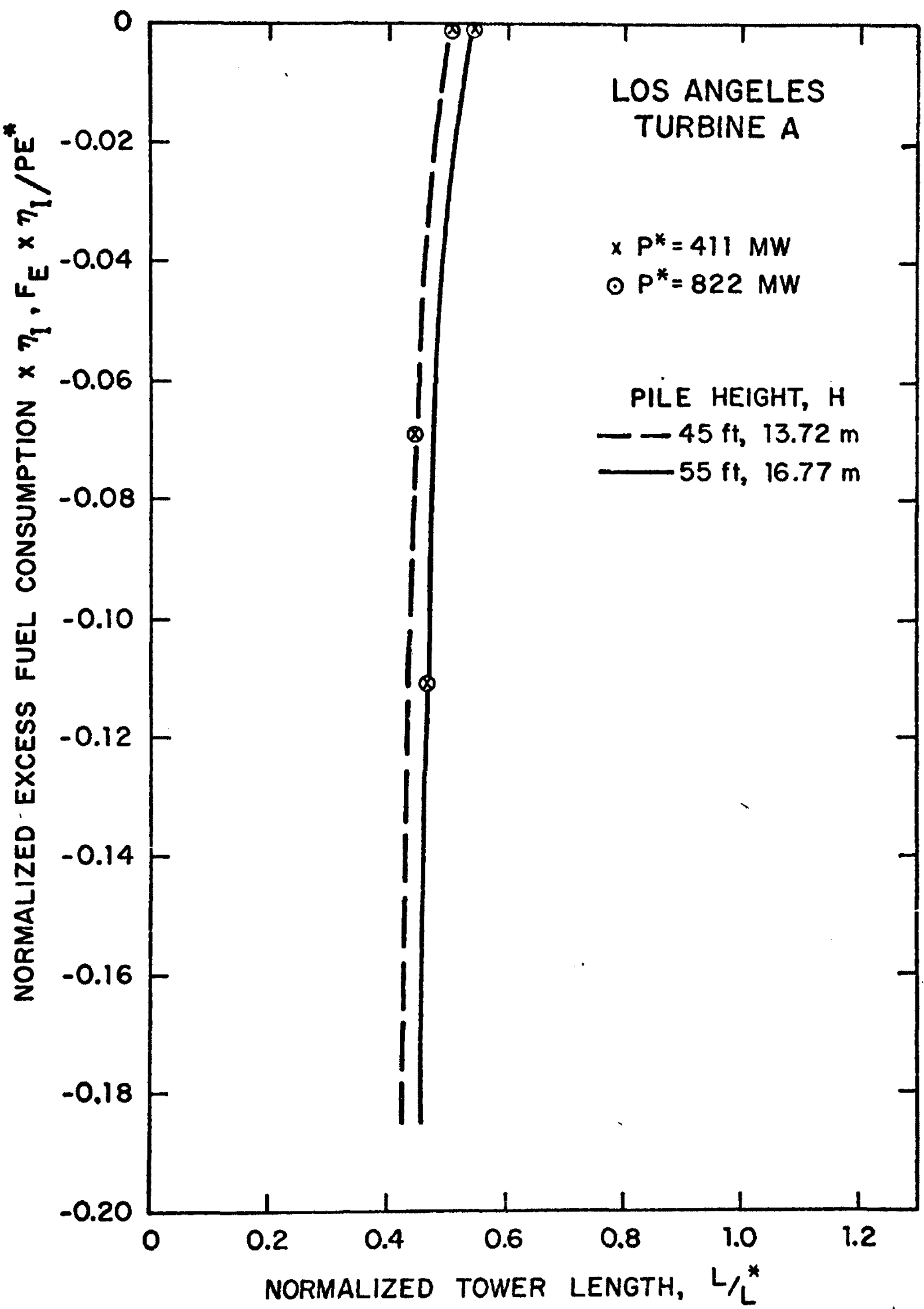

Figure 23. Normalized excess fuel consumption

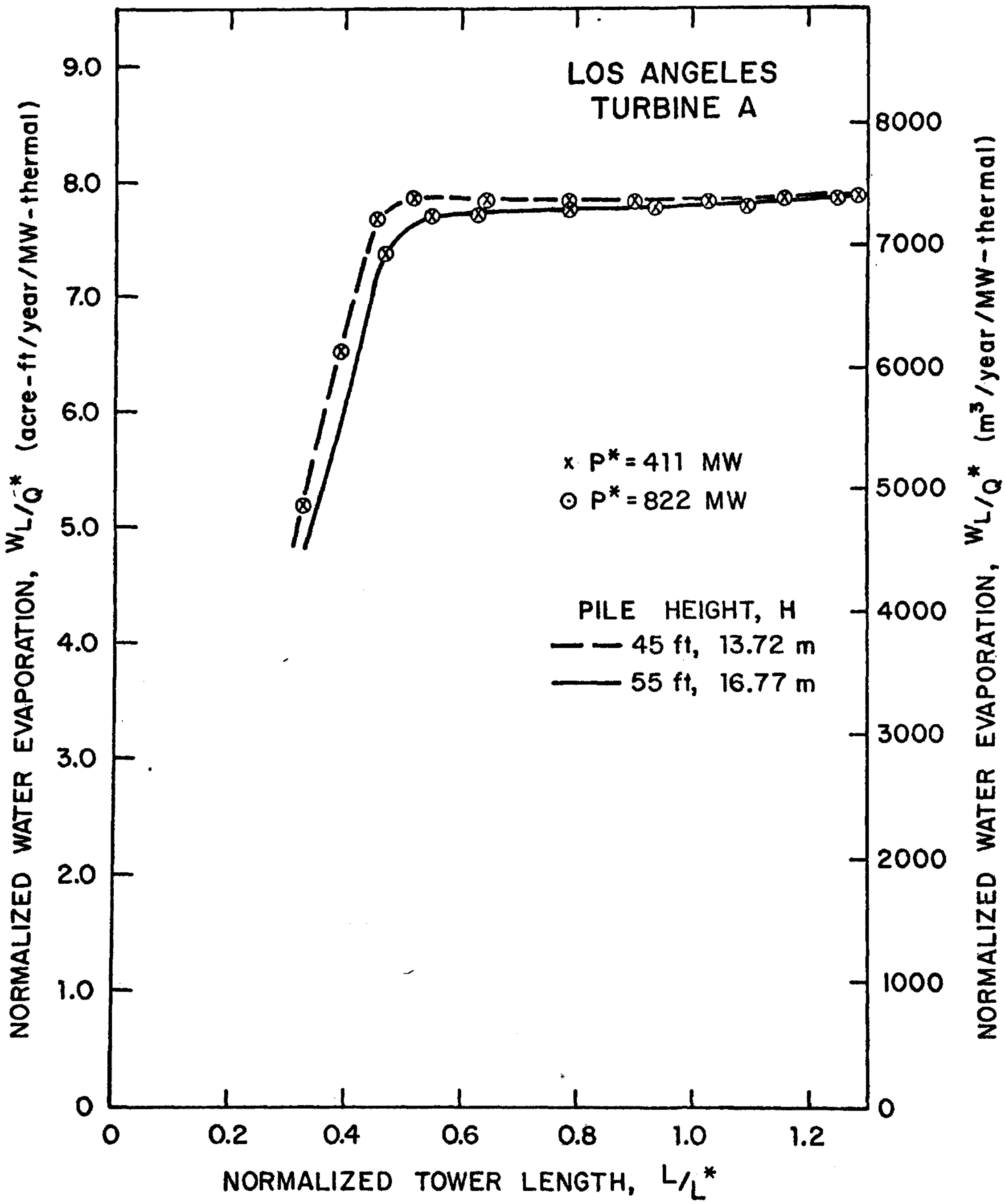

Figure 24. Normalized evaporation

sults obtained with the heat rate characteristics of turbine A (Figure 4) and the meteorological data corresponding to Los Angeles. It is obvious, however, that even the normalized quantities shown in Figures 21 through 24 will change if either the heat rate characteristics or the meteorological conditions are different. In order to build up a representative library of the operating characteristics of cooling towers, therefore, a parametric study was conducted using the following:

- Heat rate characteristics of turbines A, B, C. (Figures 4, 5, 6)
- Meteorological data at Chicago, Los Angeles, Miami, and St. Louis.

As indicated in Section IV, these studies of full-throttle loadings, are expected to represent a majority of the situations which will be encountered in the considerations of backfitting in this country; for discussion of variations in loading, see Section V.J. The final results are presented in Figures 25 through 28.

E. OPERATING COSTS WITH COOLING TOWERS

As discussed in Section IV.C, in the consideration of the costs of backfitting a power plant or unit with cooling towers, the maximum capacity loss (Figure 25) contributes to the capital cost of the project, while the energy loss, the excess fuel consumption, and the loss of water due to evaporation all contribute to the operating costs after backfitting. From Section IV.C, it will be recalled that the total operating cost resulting directly from backfitting can be written as

$$OC = OC_R + OC_{EF} + OC_S \tag{16}$$

where OC_R is the cost of replacing the energy loss E_L, OC_{EF} is the cost resulting from the excess fuel consumption F_E and OC_S is the differential operating cost of the cooling towers. The first two of these can be found from Figures 26 and 27, respectively, when the tower size, nameplate capacity, reference heat rejection rate, and

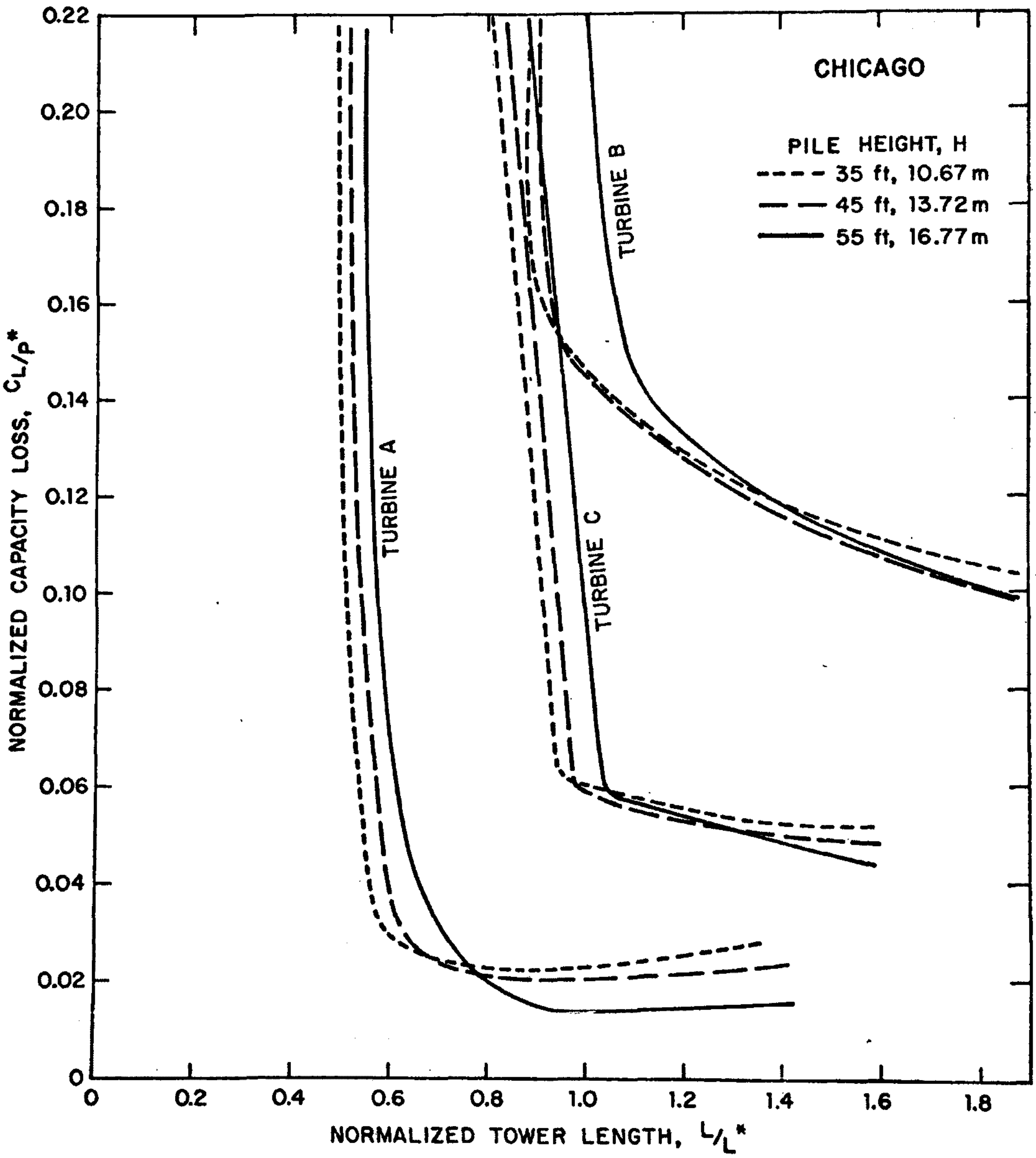

Figure 25(a). Normalized capacity loss, Chicago

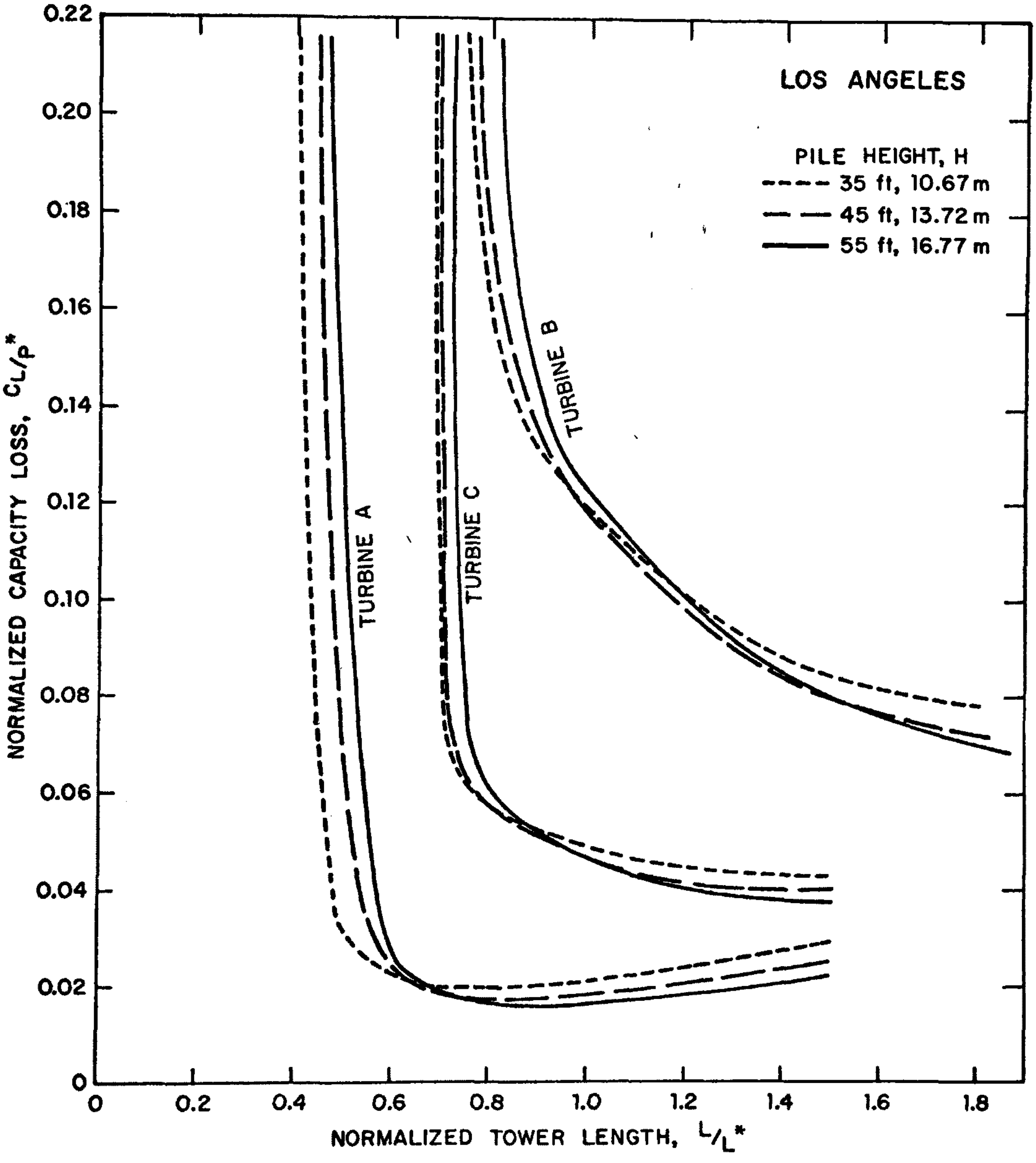

Figure 25(b). Normalized capacity loss, Los Angeles

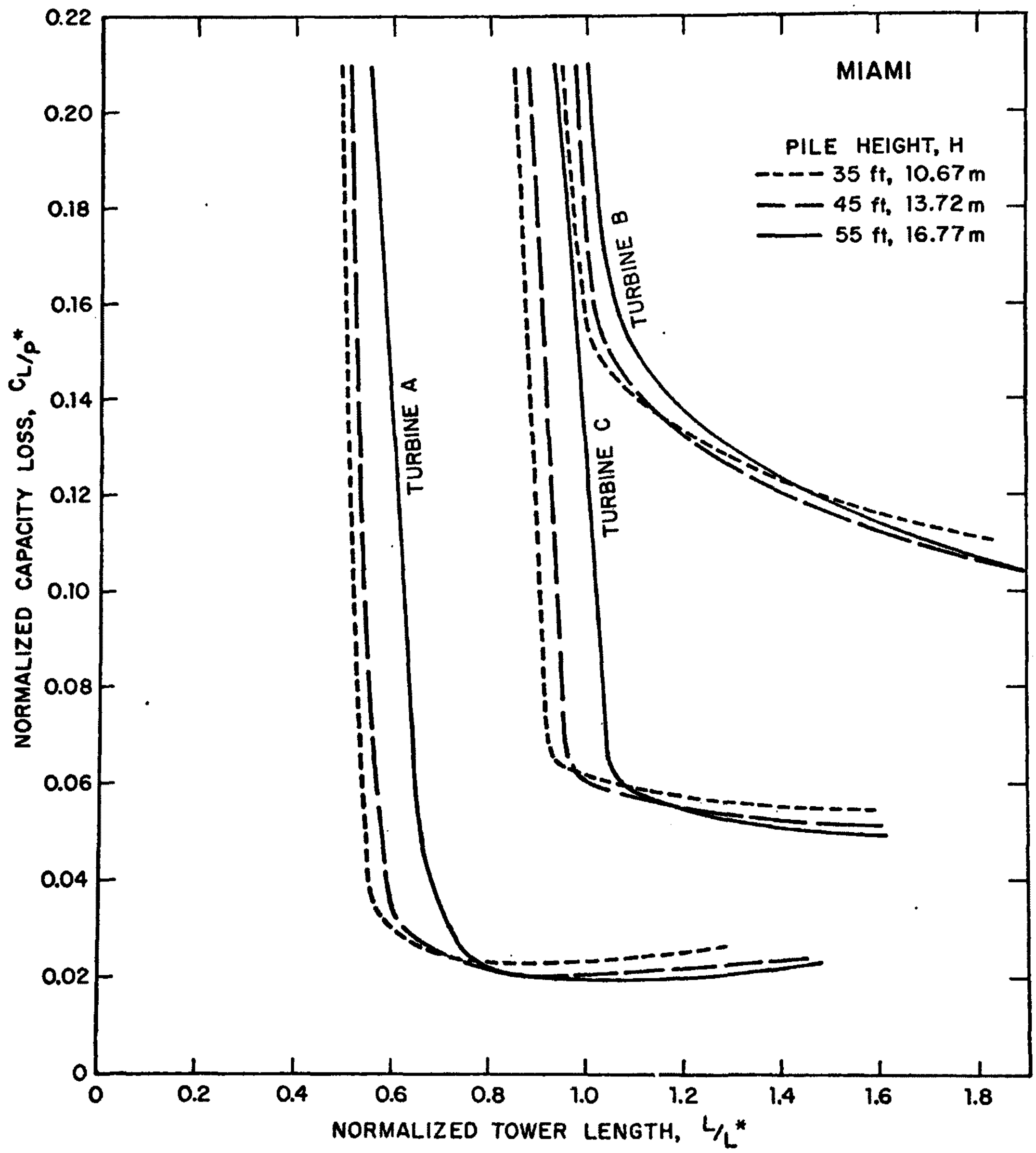

Figure 25(c). Normalized capacity loss, Miami

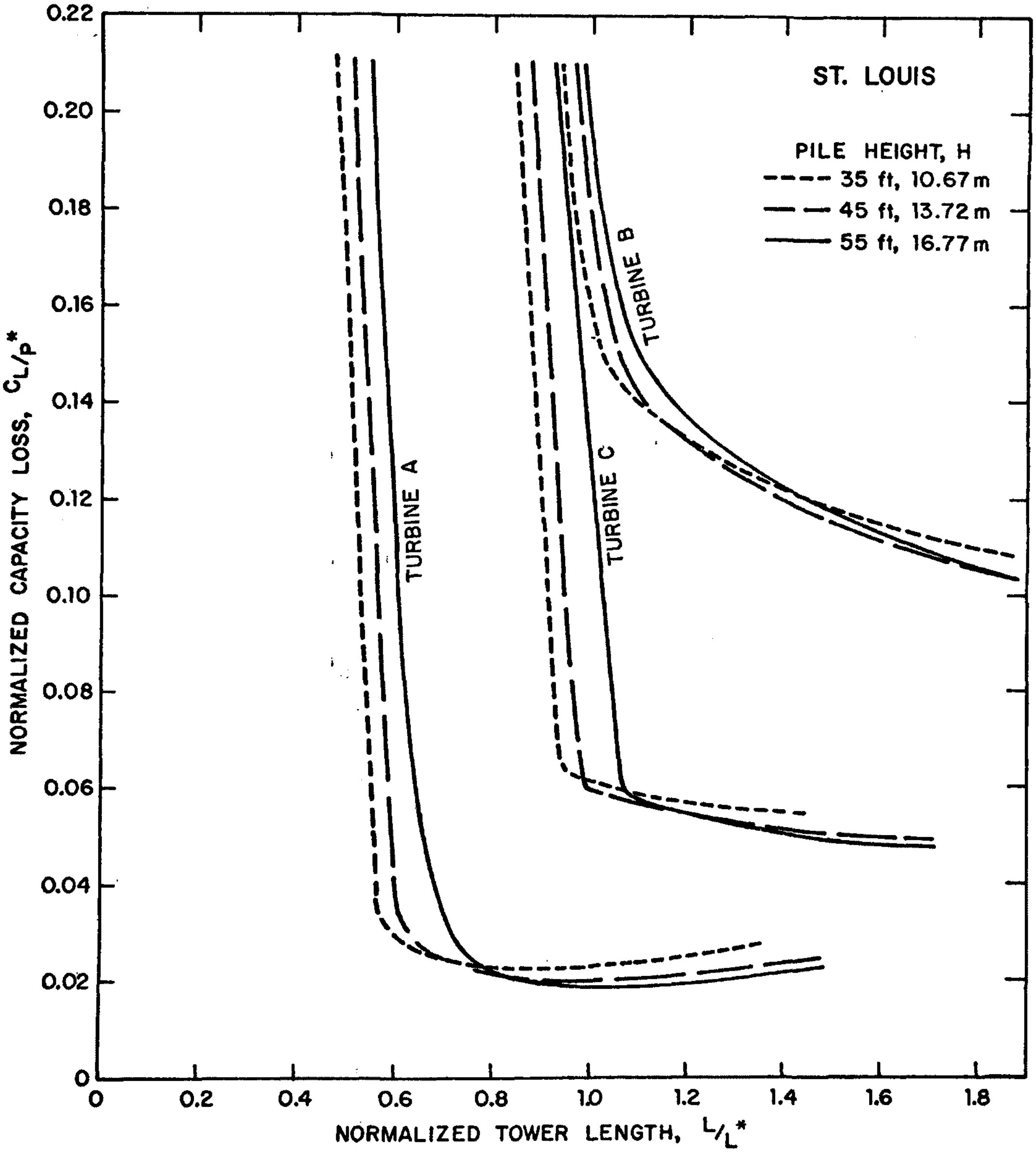

Figure 25(d). Normalized capacity loss, St. Louis

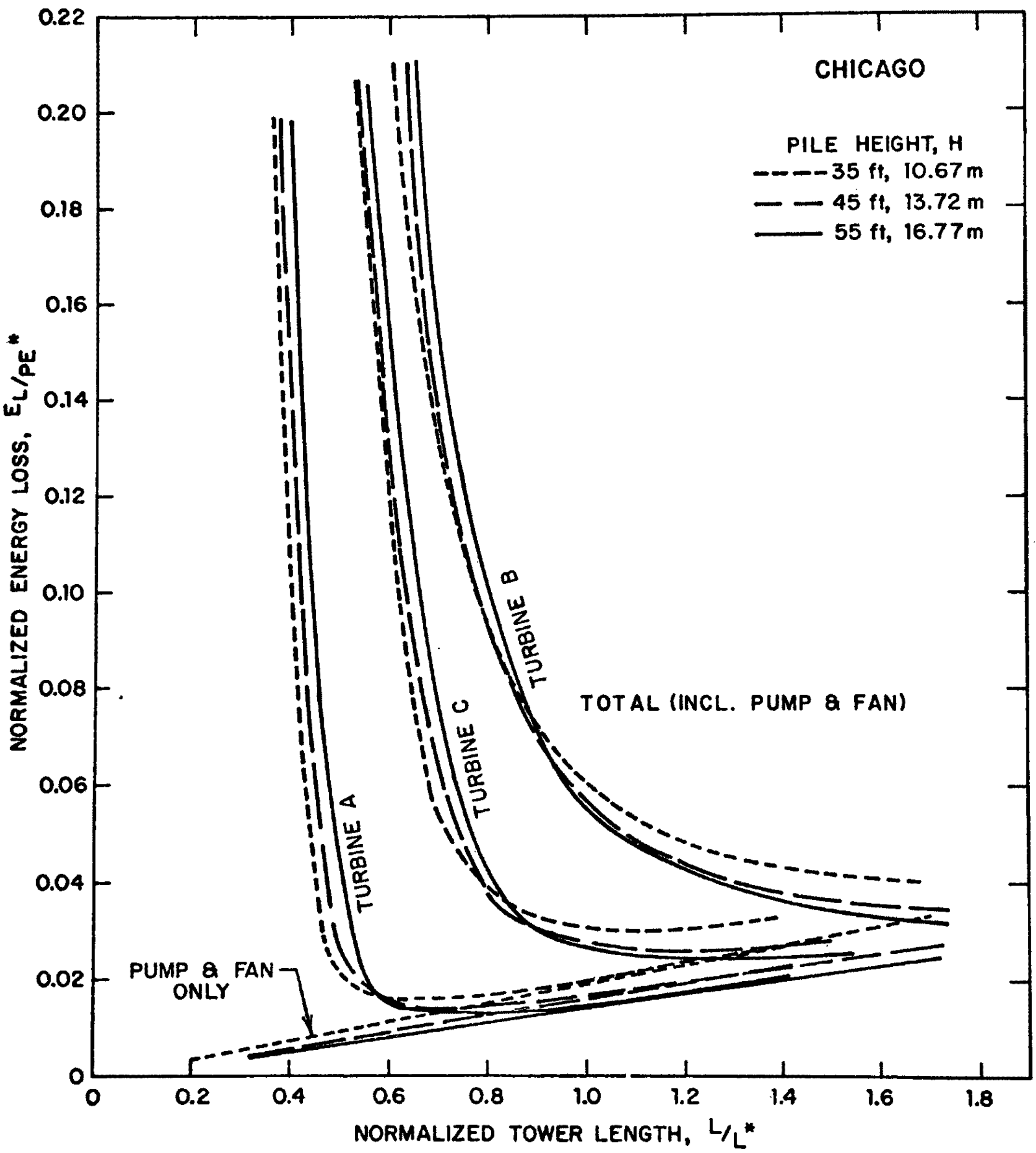

Figure 26(a). Normalized energy loss, Chicago

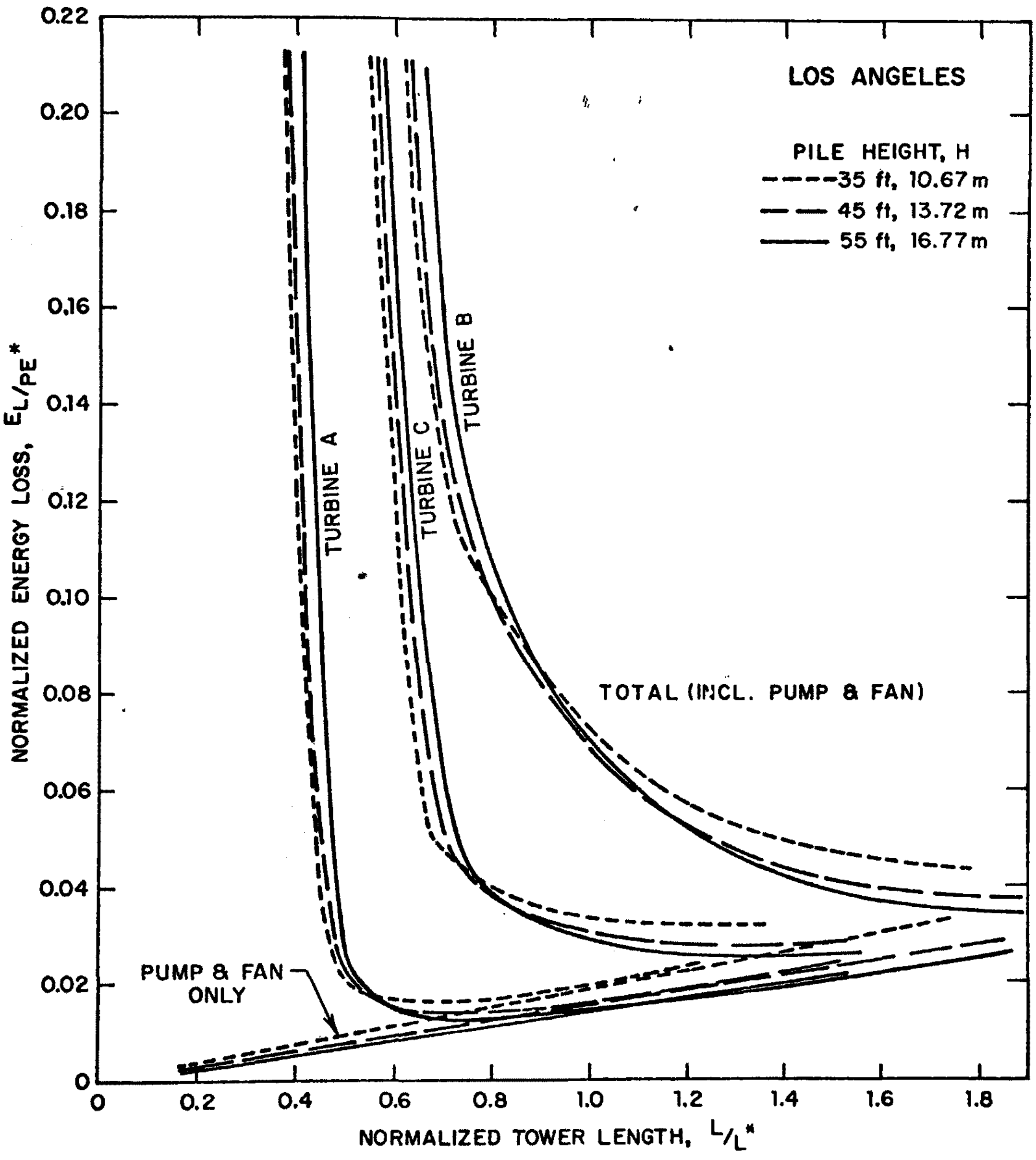

Figure 26 (b). Normalized energy loss, Los Angeles

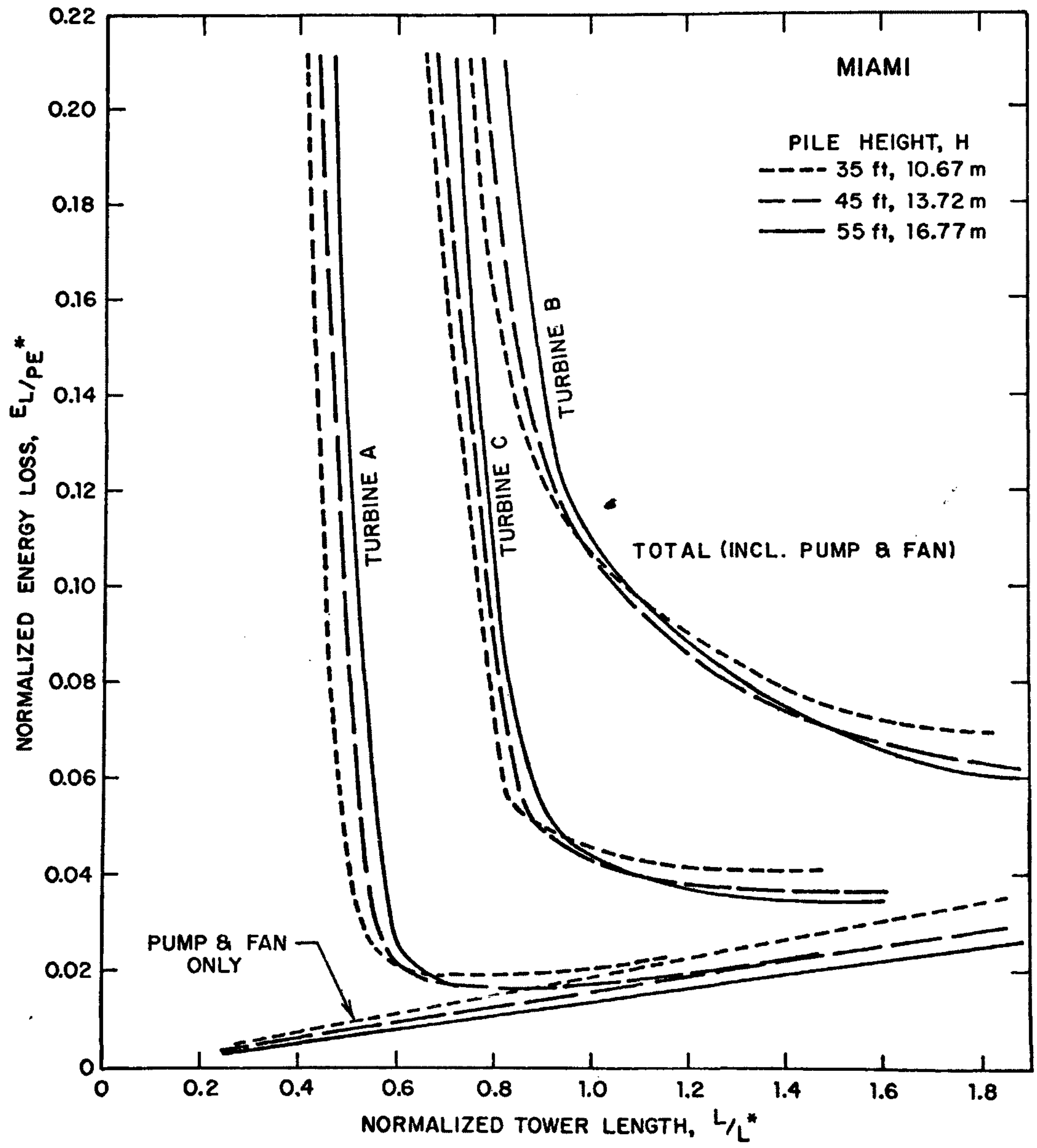

Figure 26 (c). Normalized energy loss, Miami

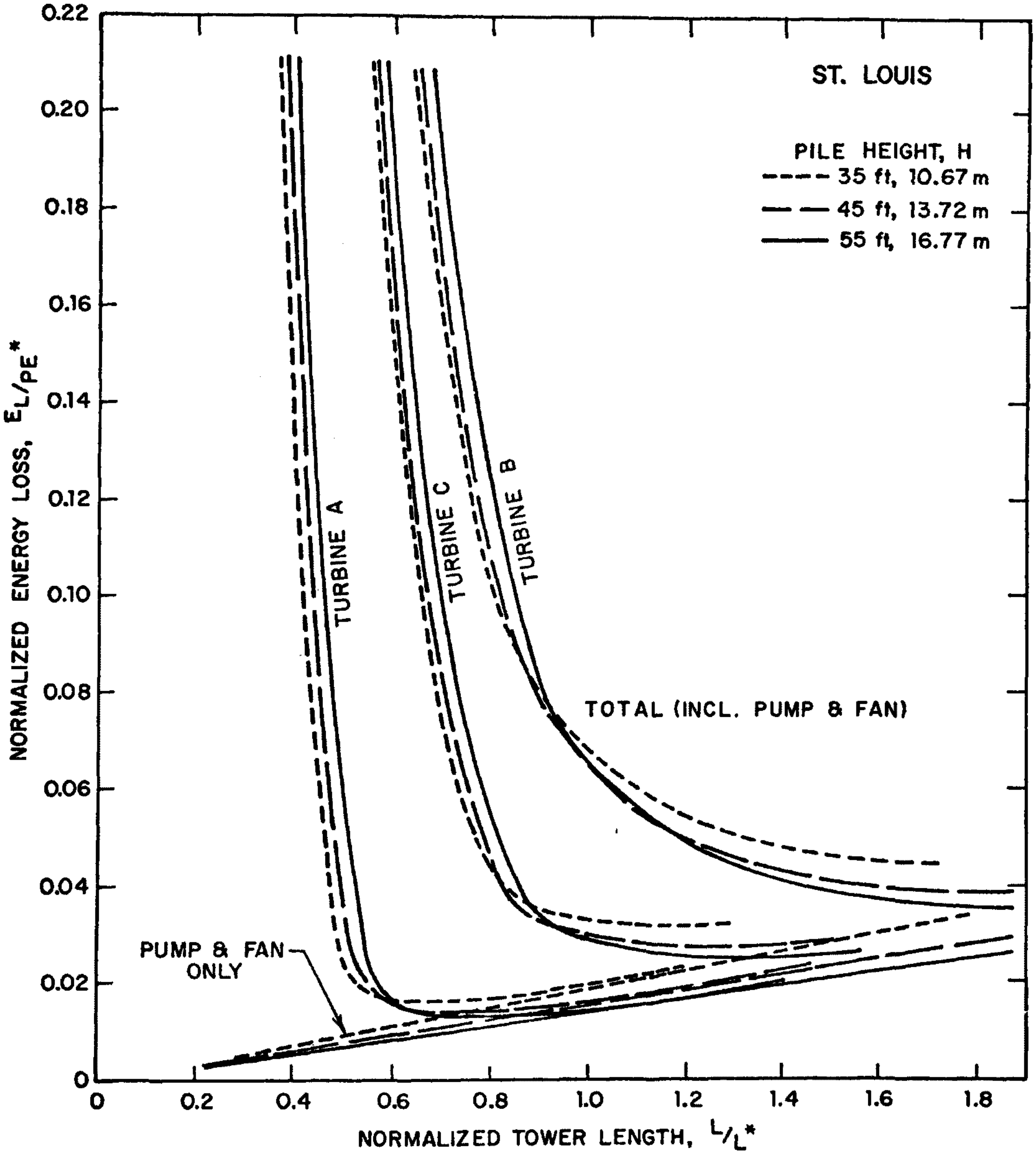

Figure 26(d). Normalized energy loss, St. Louis

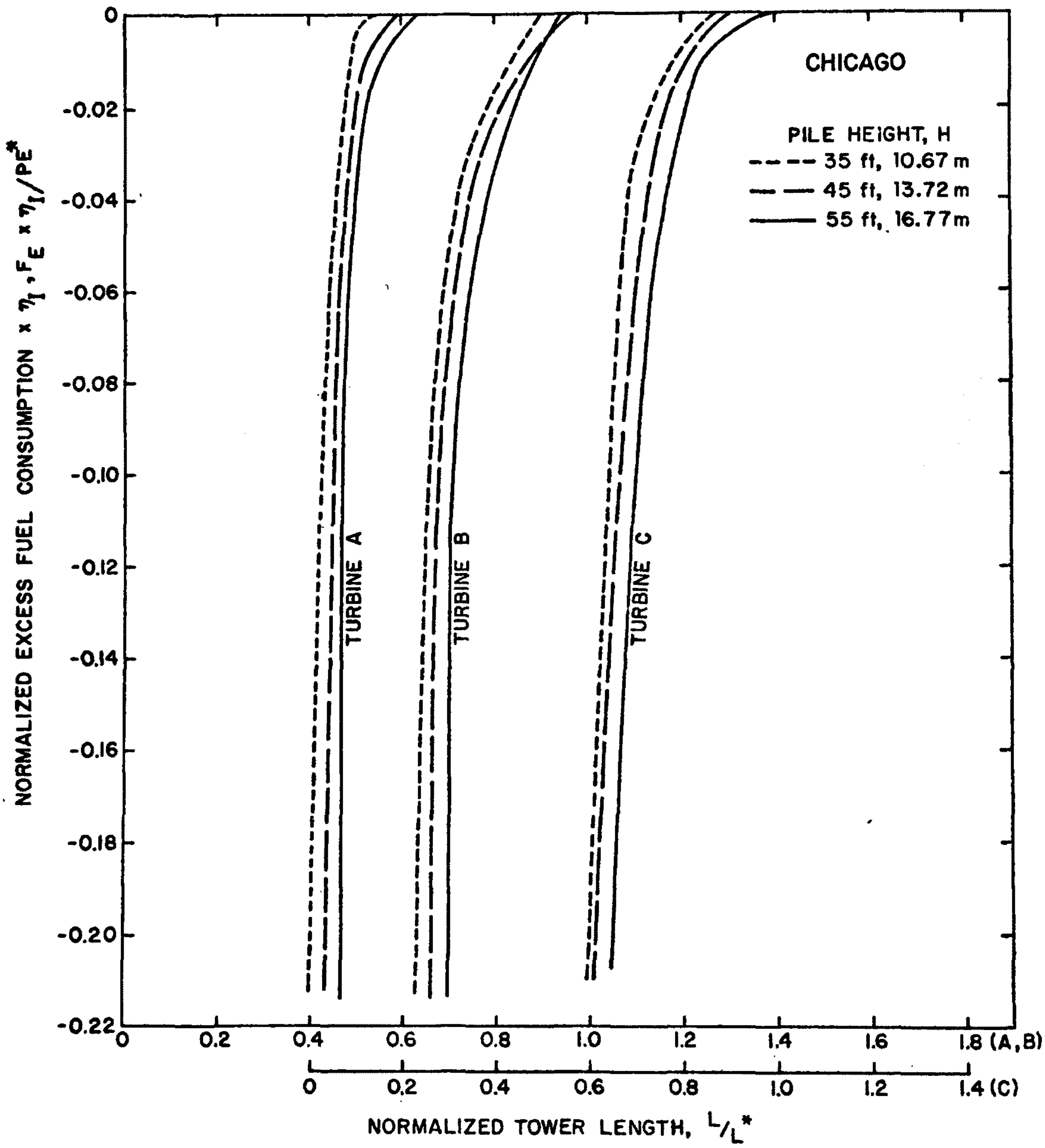

Figure 27(a).. Normalized excess fuel consumption, Chicago

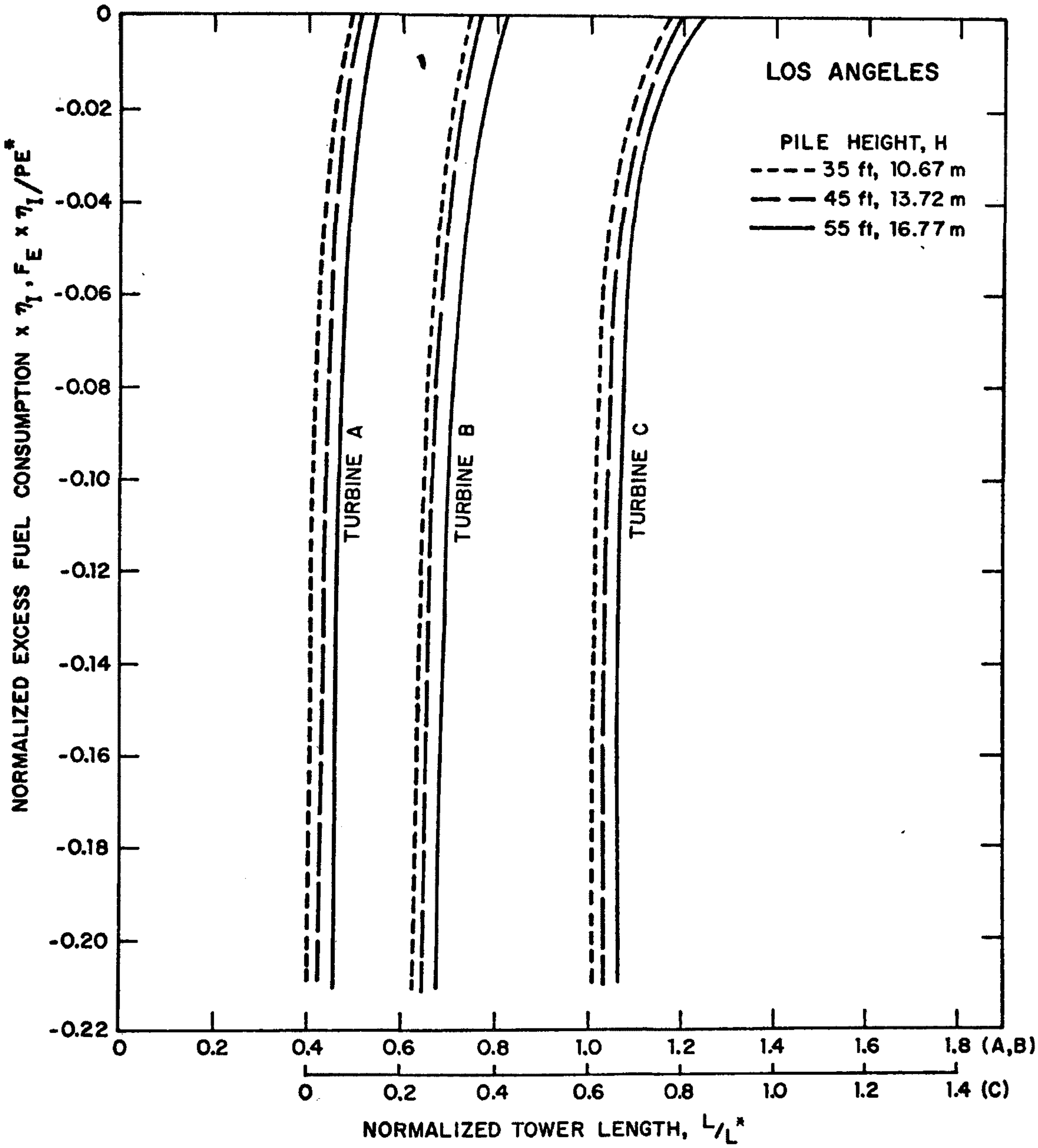

Figure 27(b). Normalized excess fuel consumption, Los Angeles

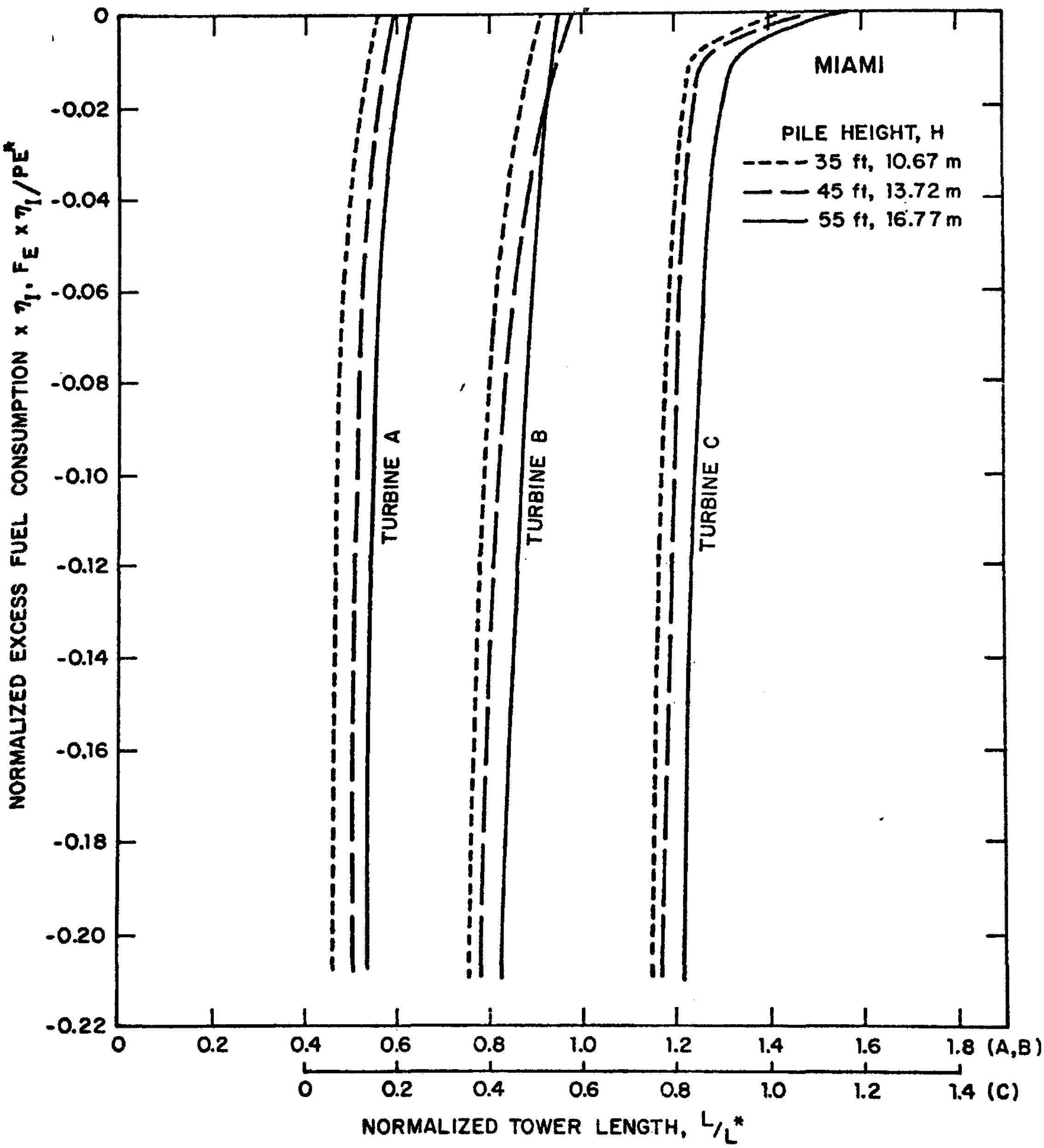

Figure 27(c). Normalized excess fuel consumption, Miami

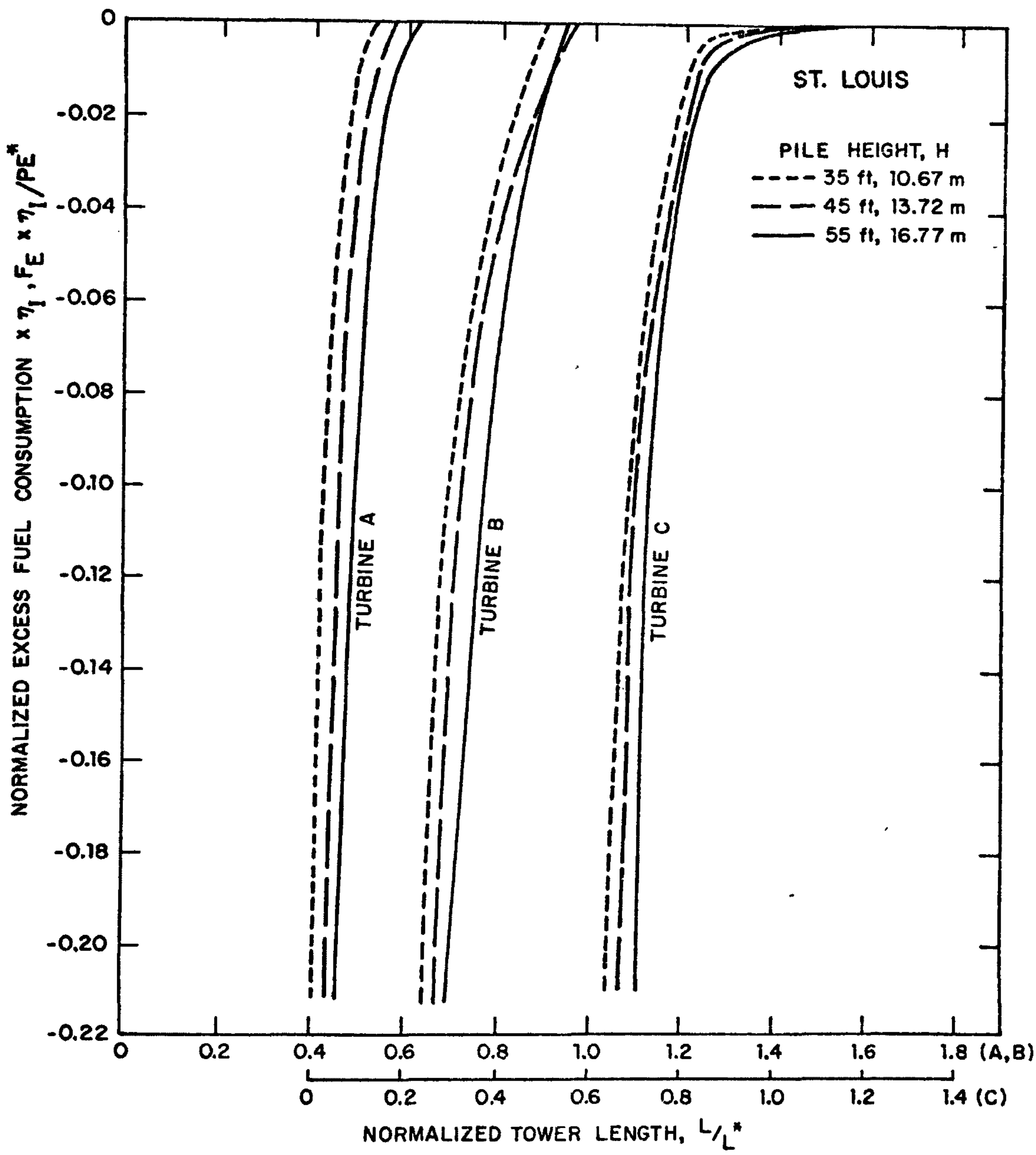

Figure 27(d). Normalized excess fuel consumption, St. Louis

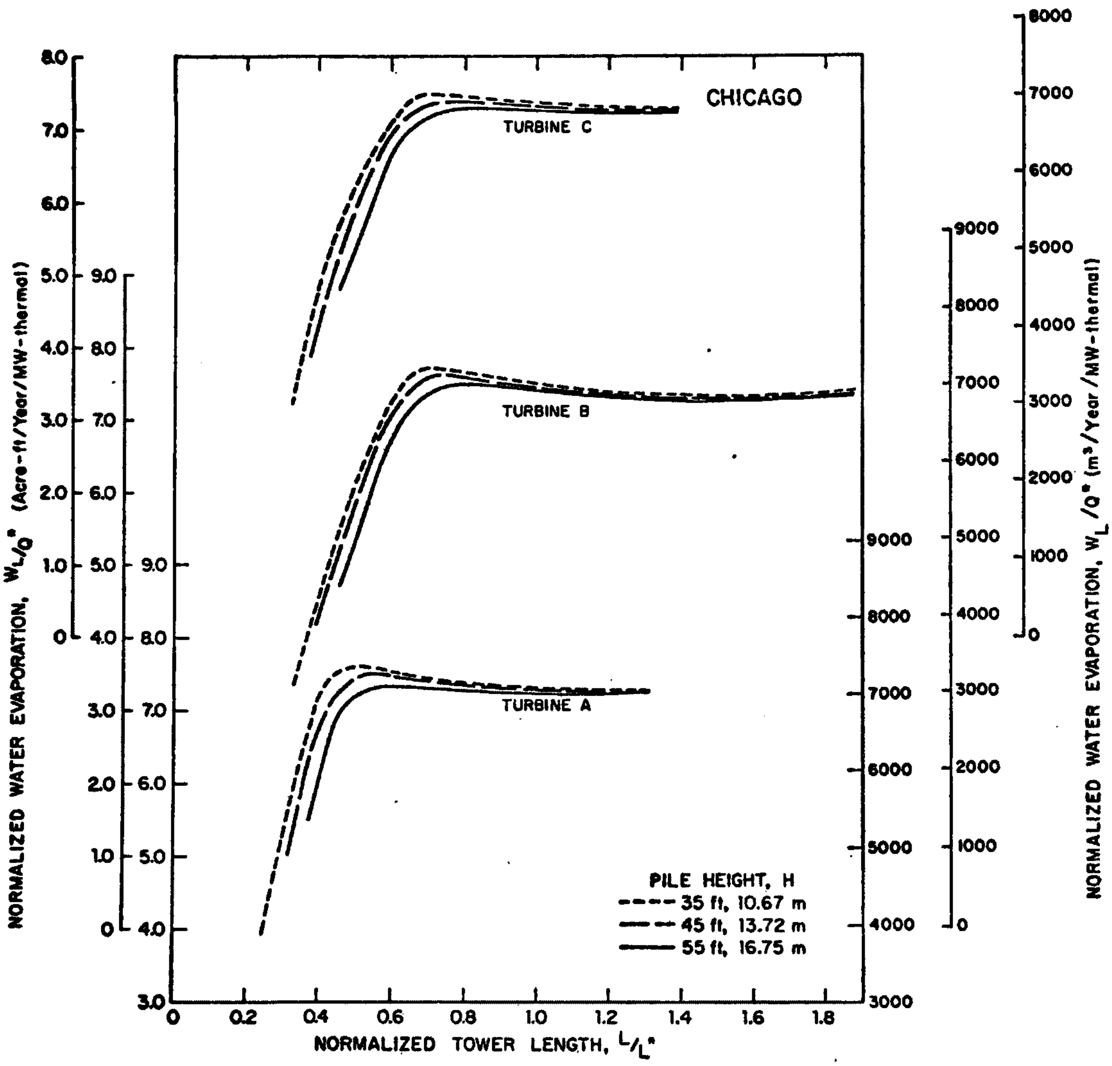

Figure 28(a). Normalized evaporation, Chicago

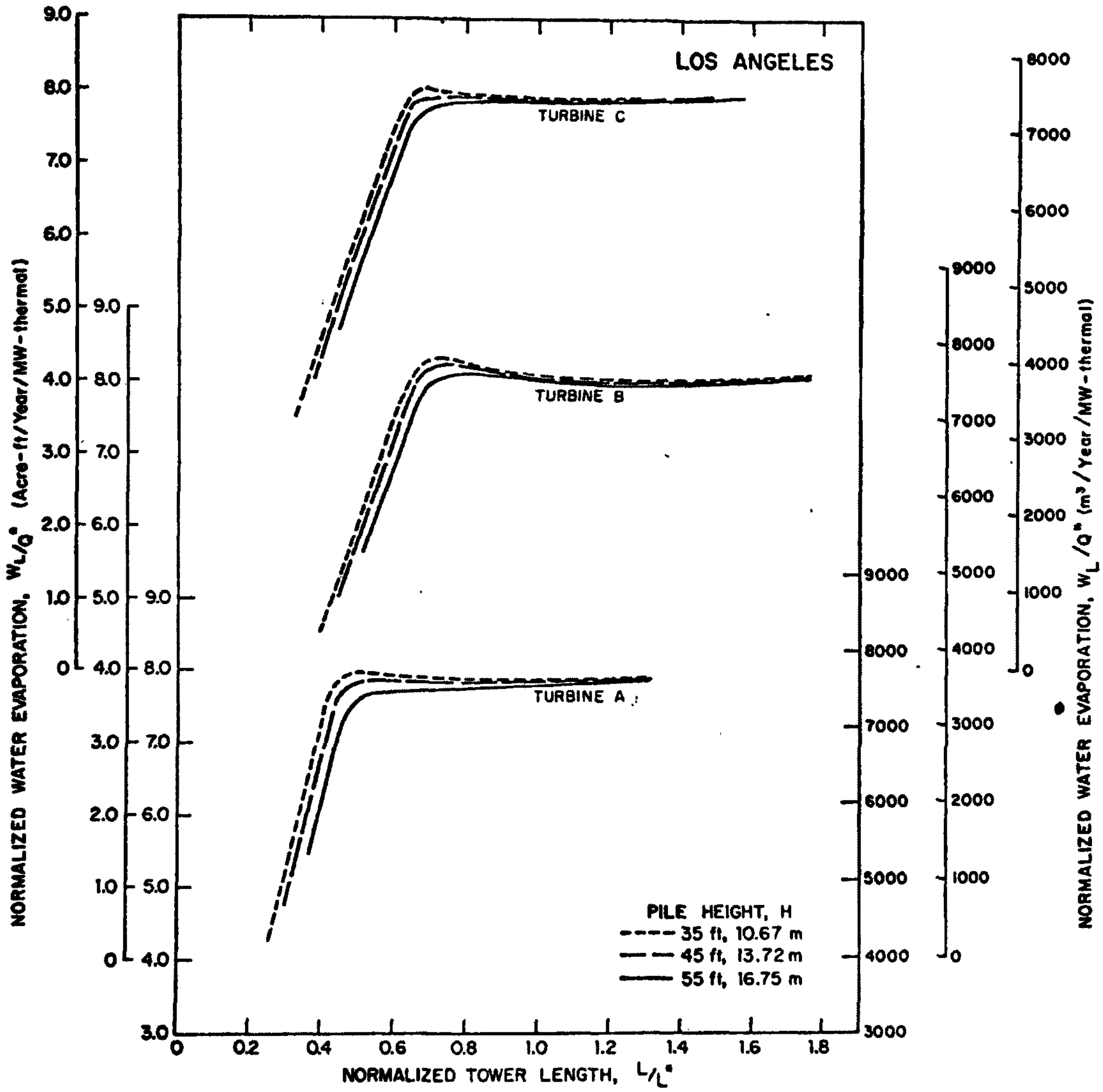

Figure 28(b). Normalized evaporation, Los Angeles

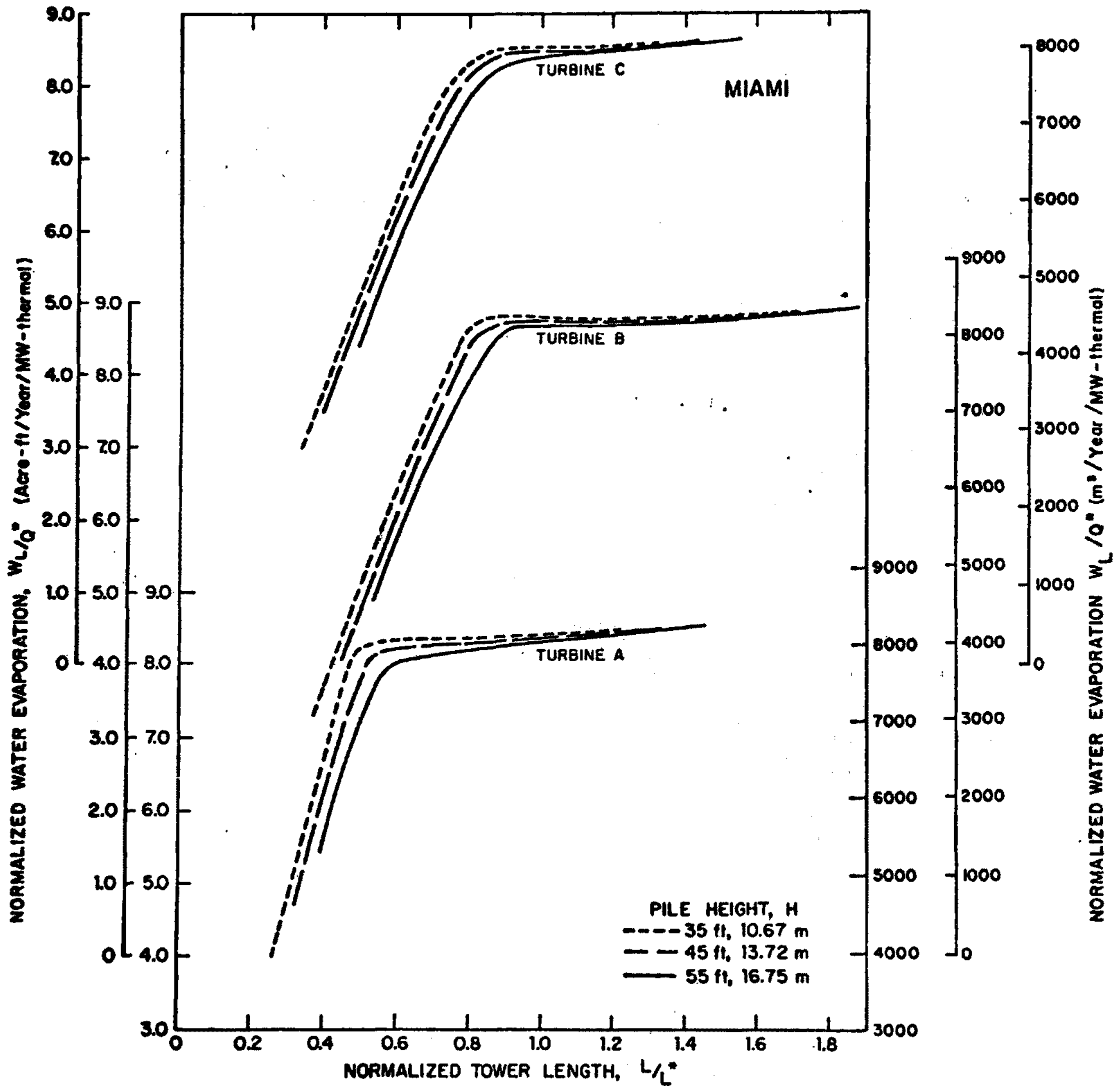

Figure 28(c). Normalized evaporation, Miami

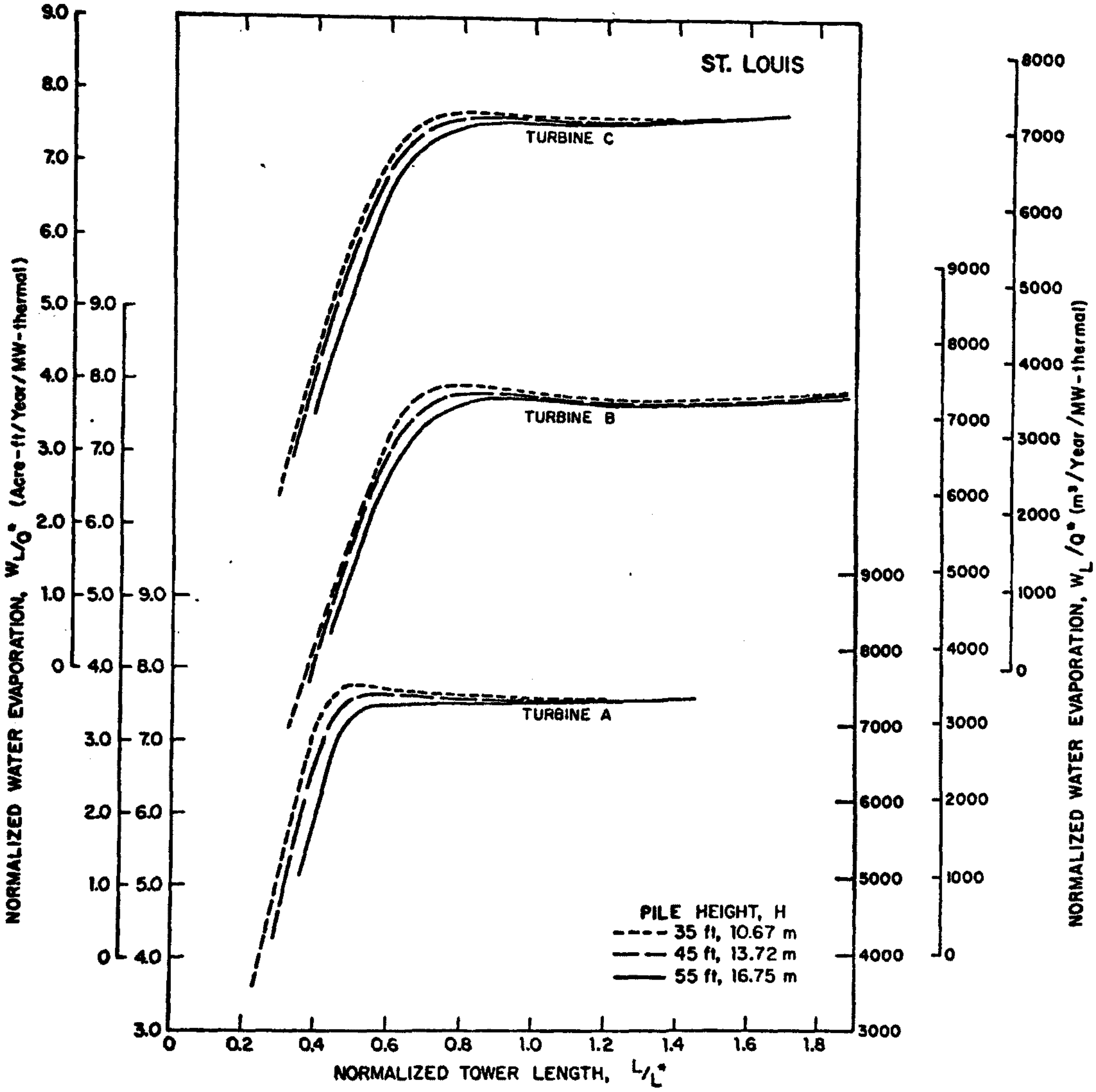

Figure 28(d). Normalized evaporation, St. Louis

unit costs of replacement energy and fuel are known. Before describing the general procedure for the detailed economic assessment of backfitting, however, it is necessary to examine the operating costs of the cooling towers, OC_S, in some detail.

First of all, it will be noted that the power required to run the fans and pumps has already been accounted for in the evaluation of the energy loss (and also in the calculation of the capacity loss). The tower operating costs can therefore be further due only to the additional quantities:

- The cost of makeup water (evaporation + blowdown; drift is neglected),
- The cost of blowdown treatment,
- Maintenance of the towers and associated equipment.

The water loss due to evaporation W_L can be found directly from Figure 28. The makeup water required, W_m, is then the sum of the evaporation W_L and the blowdown W_b:

$$W_m = W_L + W_b \tag{30}$$

The amount of blowdown will depend upon the concentration k (in ppm) of undesirable constituents in the makeup water and the maximum concentration k_m permitted in the cooling tower. Then

$$W_m\ k = W_b\ k_m \tag{31}$$

From equations (30) and (31), the blowdown and makeup are given by

$$W_b = \frac{1}{k^* - 1} W_L \tag{32}$$

and
$$W_m = \frac{k^*}{k^* - 1} W_L \tag{33}$$

where $k^* = k_m/k$

Now, the annual cost of makeup water can be found simply by multiplying W_m by the unit cost of water c_w (\$ per 1000 gal. or \$ per m^3). The cost of water varies widely from region to region in this country. In the backfit situation, however, it is likely that the water body used in the open-cycle operation can be relied upon as a readily available source. In any case, c_w is left as a basic variable, like all other costs, so that its influence on the overall economics can be evaluated at will. The cost of treating the blowdown prior to discharge into the environment can be found in a similar manner by multiplying W_b by a unit treatment cost c_b (\$ per 1000 gal. or \$ per m^3).

The maintenance cost of mechanical-draft cooling towers includes the annual overhaul labor and parts, and associated overhead. Both the fans and pumps are, however, low maintenance items and tower manufacturers usually suggest a unit cost, in dollars per year per tower cell, to account for all tower related maintenance costs. The maintenance cost can be found by using a unit cost of the order of \$200 per cell per year [Ref. 2, p. 568]. In the overall economics of backfitting, the tower maintenance cost is rather insignificant compared with the other penalties, and therefore small variations in this unit cost are unlikely to affect the total cost picture.

The differential operating and maintenance cost of cooling towers can now be written as

$$OC_S = \left(\frac{k^*}{k^* - 1}\right) W_L\, c_w - M' + \left(\frac{1}{k^* - 1}\right) W_L\, c_b - B' + C_m - C_m'$$

$$= Q^* \left\{ \frac{k^* c_w + c_b}{k^* - 1} \right\} \frac{W_L}{Q^*} - M' - B' + C_m - C_m' \qquad (34)$$

where OC_S = differential operation and maintenance cost of cooling towers, \$/year,

Q^* = reference heat rejection rate,

k^* = ratio of maximum permissible concentration of undesirable constituents in the circulating water to the concentration in the makeup water,

W_L = annual water evaporation, m^3/year

c_w = unit cost of supply water, $\$/m^3$,

c_b = unit cost of blowdown treatment, $\$/m^3$,

M' = makeup water cost with open-cycle system, \$/year,

B' = blowdown treatment cost with open-cycle system, \$/year,

C_m = annual maintenance cost of cooling towers, \$/year,

C'_m = annual maintenance cost of open-cycle system, \$/year.

F. PROCEDURE FOR THE ECONOMIC EVALUATION OF BACKFITTING

The various items which must be considered in the evaluation of the cost of backfitting an existing power plant or unit with mechanical-draft wet cooling towers have been described individually in the preceding paragraph. The manner in which these items are to be combined in order to calculate the total cost of backfitting will be considered next, followed by a general description of the computer program that has been developed for this purpose. Subsequently, in part H of this section, a hypothetical test case is considered in order to illustrate the general methodology presented below. In Section V.J, a variation in the loading pattern (from "full-throttle") with meteorological conditions will be considered.

As indicated earlier, it is necessary to have available a certain amount of information concerning (a) the characteristics of the power plant and site, (b) the size of cooling towers which are to be used, and (c) the various unit costs and economic parameters which apply to the particular plant or utility situation, before a detailed economic analysis can be undertaken. In particular, the methodology suggested here requires that the following quantities be known a priori:

(a) Power plant and site data:

1. Nameplate capacity, P^* (kW);
2. Reference heat rejection rate, Q^* (kJ/hr); this can be found from the reference turbine heat rate, T^*_{HR}, or from the reference plant heat rate, P^*_{HR}, and plant efficiencies η_I, η_p (see Section IV.A);
3. Turbine heat rate correction curves $\Delta[p,T_s]$, as in Figure 1;
4. Remaining useful life of the plant or unit (years);
5. Characteristics of the existing condensers if they are to be retained (limitations of temperature rise and water flow rate), or their salvage value C'_c if new condensers are to be fitted;
6. Salvage value C'_{pp} of pumps and piping associated with the open-cycle system, and the salvage value C'_o of system components other than pumps, piping and condensers;
7. Annual makeup water cost, M', blowdown cost, B', and maintenance cost C'_m associated with open-cycle cooling;
8. Meteorological data for the site (as in Table 2). These can be used to determine the design temperatures, T_{wb_d}, T_{db_d}, the extreme 10-hour exceedance temperatures, $\hat{T}_{wb}$, $\hat{T}_{db}$, and the frequencies of occurrence of T_{wb}, T_{db} as explained in Section IV.B.

(b) Cooling Towers:

1. The SIZE of cooling towers, EITHER explicitly in terms of the length L and the height H of the evaporative pile,* OR implicitly in terms of the design range, approach and water flow rate corresponding to a specified design wet-bulb temperature;

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*It will be recalled that L and H are sufficient to describe the physical size of the towers since the width of the pile (W) has been fixed, and since all detailed calculations are based upon a representative set of empirical data concerning the heat transfer properties, and air- and water-loadings in the pile.

2. Unit cost of towers, c_t (\$/TU);
3. Unit maintenance cost, c_m (\$/tower cell or \$/fan, per year);
4. Concentration ratio, k^*, and unit cost of blowdown treatment, c_b (\$/m^3);
5. Capital cost, C_{HT}, and downtime, DT, required for hook-up and testing of towers.

(c) Economic parameters:
1. Fixed charge rate, FCR (see Section IV.C and Figure 7);
2. Unit capital cost of replacement capacity, c_ℓ (\$/kW);
3. Unit cost of replacement energy, e_ℓ' (\$/kW-hr) during outage due to hook-up and testing;
4. Unit cost of replacement energy, e_ℓ (\$/kW-hr) after backfitting, and
5. Unit cost of fuel, f_c (\$/kW-hr of consumed fuel), water, c_w (\$/m^3) and land, a_ℓ (\$/m^2).

Once this information has been gathered, the calculation of the total differential cost of backfitting can be carried out either by using the computer program or by referring to the results presented graphically in the preceding sections. Since the latter have been obtained for a representative number of turbine types and meteorological conditions, and presented in a normalized format, they can be used to analyze a wide variety of power plants or units. The general procedure to be followed is described below:

(a) Preliminary considerations: The heat rate correction curves of the affected turbine should be examined to determine which one of the three model turbines (A, B or C) will best represent the affected unit. Similarly, the site meteorological data should be studied to establish which one of the four model sites (Chicago, Los Angeles, Miami or St. Louis) will best describe the affected site.

(b) Cooling tower data: The procedure for the economic evaluation of backfitting becomes particularly simple when the physical size of the tower is prescribed in the form of the pile length L and the pile height H, the pile width W being fixed at the standard value of 18 ft (5.49 m), since the various quantities of interest can then be determined directly from the example results. As remarked upon earlier, however, cooling tower manufacturers do not usually specify the physical size of the towers. Instead, the size is implied by specifying the range and approach occurring with a specified water flow rate at a design wet-bulb temperature. In order to make use of the example results it is then necessary to determine the corresponding physical dimensions of the towers of the type used in the example calculations. This can be accomplished either by requesting the relevant information from the manufacturers or by inferring the physical dimensions from Figure 12 (which assumes a fixed width, water loading, air loading, and thermal performance characteristics). In the latter case, the length, height and the total water flow rate can be determined when the range, approach, reference heat rejection rate and design wet-bulb are given.

(c) Capital costs and performance data: The example results can now be used to find capital cost of the cooling towers and associated equipment, and also the capacity loss, energy loss, excess fuel consumption and water evaporation as follows:

- Given L and H, read Figure 11 to find the number of tower units, TU. Alternatively, given the range, approach, Q^* and T_{wb}, read Figure 12 to find L and H, and then read Figure 11 to find TU. (Note that, in this case, the rating factor can be found from the manufacturer's charts, such as those shown in Figure 10, and TU determined from equation (21) using the specified water flow rate in gpm. Figure 12 should nevertheless be used to find L and H since this information is required for the subsequent analysis.)

- Determine the capital cost of the towers, C_{cs} from equation (22) using the appropriate unit cost c_t.
- Given L, determine the total water flow rate from Figure 12. Hence, read Figure 13 to find the pump and pipe system cost, C_{pp}.
- If new condensers are to be used, determine the required surface area A_c from Figure 14 and the cost C_c from equation (23) using the appropriate unit cost c_c.
- Determine additional land area requirement based on desired criterion. If noise level is important, see Ref. 7 (Vol. I, Appendix G). (Alternatively, use equation (24) or other site-dependent criterion).
- Given Q^* and H, read Figure 16 to determine the reference length L^* of towers. Calculate the normalized length L/L^*.
- With L/L^* and H known, determine the normalized capacity loss (C_L/P^*) from Figure 25, the normalized energy loss (E_L/PE^*) from Figure 26, the normalized excess fuel consumption (F_E/PE^*) from Figure 27 and the normalized water evaporation (W_L/Q^*) from Figure 28. Hence find C_L, E_L, F_E and W_L.

(d) Final economic evaluation: The above information, along with the quantities specified initially, can now be used in the equation given in Appendix I to evaluate the total cost of backfitting the power plant with a cooling tower of dimensions L and H.

The procedure outlined here is further demonstrated by taking a hypothetical test case in Section V.H.

G. THE COMPUTER PROGRAM

The computer program which accepts any set of numerical values for the various parameters and performs the calculations outlined in the previous sections is listed in Appendix III. The thermodynamic models used to simulate the performance of cooling towers are basically the same as those developed by Croley, Patel and Cheng [15] for the wet portion of dry-wet combination towers, but there are a number of important differences in other respects. In particular, the economic considerations are formulated specifically for the analysis of backfitting an existing power plant or unit with mechanical-draft wet cooling towers and cannot be used, without modification, to study the design of towers for new plant or units.

The computer program consists of the MAIN program and seven subroutines, namely OPECOS, MODELW, NTUCAL, RATFAT, FAN, FOGSEN, and POWERS. The MAIN program reads all inputs, calculates the overall capital and total costs, and controls the printout of these quantities. The inputs, along with the symbols and units used, are listed in Appendix II, and a typical output is shown at the end of the program listing in Appendix III. The primary functions of the various subroutines are as follows:

OPECOS: This subroutine evaluates the annual operating costs by summing the various costs associated with each set of meteorological conditions (see equations (27) through (29).

MODELW: This subroutine determines the turbine operating point (p,Q) on the heat rejection rate characteristics by matching the heat rejected from the turbine with the cooling capacity of the towers. These calculations are performed for each set of meteorological conditions.

NTUCAL: This subroutine contains the basic thermodynamic model of evaporative cooling. Given the ambient meteorological conditions, the heat transfer coefficients for the pile, the air- and water-loading used, the hot-water temperature, the output is the cold-water temperature and consequently the rate at which heat is rejected from the towers. This calculation is nested in an iterative cycle, controlled by MODELW, in which the cooling tower performance is matched with the turbine performance.

RATFAT: Here the rating factor charts (Figure 10) are used to find the rating factor corresponding to a given set of range, approach and design wet-bulb temperature. This subroutine is used only once, in the evaluation of the capital cost via the tower-unit method.

FAN: Here, the specified fan characteristics are used to find the fan horsepower corresponding to the given air flow rate (determined by the air-loading and the face area of the towers) and a prescribed pressure drop.

FOGSEN: This subroutine can be used to calculate the "amount of fogging" that may result at each set of meteorological conditions. Three different fog-sensitivity parameters are calculated. This particular feature of the program has not been used in the present study but has been retained in the listing for future reference. Further details are given in the report of Croley, Patel and Cheng [15].

POWERS: This subroutine calculates the turbine throttle setting T_S corresponding to the operation point (p,Q) on the turbine heat rejection rate characteristics. Equation (3) is then used in the MAIN program to calculate the power P from the heat rejection rate Q and the throttle setting T_S.

From the program outline given above, it will be clear that the MAIN program and the subroutines OPECOS, MODELW and POWERS do not contain any information concerning the type of cooling system that is considered. They relate primarily to the economic analysis and the operating characteristics of the turbine. The fact that mechanical-draft cooling towers are being analyzed is reflected only in the thermodynamic model used in subroutine NTUCAL and by the presence of subroutines RATFAT, FAN and FOGSEN. (In fact, subroutines NTUCAL and FOGSEN refer only to a particular crossflow evaporative pile and tower exhaust, respectively, whether used in mechanical- or natural-draft towers. Thus, these two subroutines are used essentially unchanged for natural-draft calculations also.) This particular arrangement was developed since it greatly facilitates the adaptation of the program to study other closed-cycle cooling systems considered later. In subsequent sections, therefore, only the changes in the basic program will be documented.

H. A HYPOTHETICAL TEST CASE

1. Consider a power plant with the following characteristics:

Nameplate capacity, P^*	= 312.5 MW
Reference heat rejection rate, Q^*	= 1.912×10^9 Btu/hr (2.017×10^9 kJ/hr)
Turbine type	= A
Remaining life of plant	= 20 years
Existing condensers are to be retained so that salvage value of old condensers (C_c') and cost of new condensers (C_c) are both	= 0
Salvage value of pumps and pipes associated with open-cycle system (assumed to be 20% of new pumps and pipes), C_{pp}'	= $0.20\ C_{pp}$
Salvage value of other open-cycle system components, C_o'	= 0
Annual cost of makeup water with open-cycle, M'	= 0
Annual cost of blowdown treatment with open-cycle, B'	= 0
Site meteorological data similar to	= MIAMI
Design dry-bulb temperature, T_{db_d}	= 89°F (31.7°C)
Design wet-bulb temperature, T_{wb_d}	= 78°F (25.6°C)

Extreme wet-bulb temperature, $\hat{T}_{wb}$	= 83°F (28.3°C)
Frequency of occurrence of T_{db}, T_{wb}	= As in Table 3

2. Assume that this plant is to be backfitted with cooling towers whose characteristics are:

†Pile length, L	= 400 ft (121.9 m)
†Pile height, H	= 45 ft (13.7 m)
A two-sided pile with one-side pile width, W	= 18 ft (5.49m)
Water loading, per unit plan area of pile	= 12.5 gpm/ft^2 (0.509 m^3/min/m^2)
Air loading, per unit face area of pile	= 1800 lb/hr/ft^2 (8790 Kg/hr/m^2)
†Total water flow rate, GPM (=12.5 × 2 × L ×W)	= 180,000 gpm (681.3 m^3/min)
Fan diameter	= 28 ft (8.53 m)
Distance between fan centers, approx.	= 32 ft (9.75 m)
Number of cells or fans, N(= INTEGER [400/32])	= 12
Unit cost of towers, c_t	= $7.50/TU
Unit maintenance cost, c_m	= $200/cell/year
Concentration ratio (supply water: 100 ppm; maximum permissible: 330 ppm), k^*	= 3.3
Unit blowdown treatment cost, c_b	= $0.05/1000 gal ($0.0132/m^3)
Cost of hook-up and testing, C_{HT}	= Assumed to be included in cost of towers
Downtime, DT	= 720 hrs (30 days)

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†Alternatively, Range = 21.4°F (11.9°C); Approach = 11.4°F (6.3°C); T_{wb_d} = 78°F (25.6°C). Then, read Figure 12 to obtain L, H and water flow rate given above

3. It is assumed that the following economic parameters apply to the affected utility:

Fixed charge rate (20 years remaining life), FCR (from Figure 7)	= 0.179
Unit cost of replacement capacity (gas turbines), c_{ℓ}	= \$90/kW
Unit cost of replacement energy during downtime (difference between purchase price and usual production costs), e_{ℓ}'	= \$0.007/kW-hr
Unit cost of replacement energy after backfit (gas turbine; capital, operation, maintenance, etc.), e_{ℓ}	= \$0.01/kW-hr
Unit cost of fuel (fossil), f_c	= \$0.000751/kW-hr-th
Unit cost of water, c_w	= \$0.10/1000 gal (\$0.0264/m^3)
Unit cost of land, a_{ℓ}	= \$3000/acre (\$7412.9/hectare)

4. Use of example results:

(a) Use Figure 11 to find the number of tower units (alternatively, given range, approach and design wet bulb, use Figure 10 to obtain the rating factor, multiply by GPM to find TU), TU	= 0.1770×10^6
(b) Refer to Ref. 7 to find the specific land area corresponding to a desired noise level (60 dB, say)	= 0.1 acres/MW
Thus, land area required, $A_L = 0.1 \times 312.5$	= 31.25 acres (12.65 hectares)
(c) Read Figure 16 to find L^*	= 485 ft (147.8 m)
Determine normalized length, L/L^*	= 0.825
(d) Read Figure 25 (turbine A, Miami) to find normalized capacity loss, C_L/P^*	= 0.0212
Thus, capacity loss $C_L = 0.0212 \times 312.5 \times 1000$	= 6625 kW
(e) Read Figure 26 (turbine A, Miami) to find normalized energy loss, E_L/PE^*	= 0.0168

Thus, energy loss,

$E_L = 0.0168 \times 312.5 \times 1000 \times 8760 \quad = 46.00 \times 10^6$ kW-hr/year

(f) Read Figure 27 (turbine A, Miami) to find normalized excess fuel consumption, $\eta_I F_E/PE^* \quad = 0$

Thus, excess fuel,

$F_E = 0 \times 312.5 \times 1000 \times 8760/0.85 \quad = 0$ kW-hr/year

(g) Read Figure 28 (turbine A, Miami) to find normalized water evaporation, $W_L/Q^* \quad = 8.33$ acre-ft/yr/MW-th ($1.03 \times 10^4 m^3$/yr/MW-th)

Thus, evaporation,

$W_L = 8.33 \times 1.912 \times 10^9/3.413 \times 10^6 \quad = 4667$ acre-ft/year ($5.757 \times 10^6 m^3$/year)

Also, blowdown, $W_b = W_L \dfrac{1}{k^* - 1} \quad = 2030$ acre-ft/year ($2.504 \times 10^6 m^3$/year

And, makeup,

$W_m = W_L \dfrac{k^*}{k^* - 1} = W_b + W_L \quad = 6697$ acre-ft/year ($8.261 \times 10^6 m^3$/year)

5. Cost determination:

<u>Capital costs</u>

Cooling towers, $C_{cs} = TU \times c_t = 0.1770 \times 10^6 \times 7.50$	= \$1,327,500
Pump and pipe system (Figure 13 with known GPM), C_{pp}	= \$1,656,000
Pump and pipe system salvage, $C'_{pp} = 0.2\ C_{pp}$	= (\$ 331,200)
New condensers, C_c	= \$ 0
Salvage value of old condensers, C'_c	= (\$ 0)
Salvage value of other open-cycle components, C'_o	= (\$ 0)
Hook-up and testing cost, C_{HT}	= included in tower cost
Additional land, $A_L a_\ell = 31.25 \times 3000$	= \$ 93,750
Replacement capacity, $CC_R = C_L c_\ell = 6625 \times 90$	= \$ 596,250

Downtime,

$CC_{DT} = DT \times P^* \times e'_\ell$

$= 720 \times 312.5 \times 1000 \times 0.007$ = \$1,575,000

TOTAL CAPITAL COST, CC = \$4,917,300

Operating costs/year

Excess fuel cost,

$OC_{EF} = F_E f_c$

$= 0 \times 0.000751$ = \$ 0

Replacement energy cost,

$OC_R = E_L e_\ell$

$= 46.00 \times 10^6 \times 0.01$ = \$ 460,000

Supply water cost,

$W_m c_w = 6697 \times (3.259 \times 10^5) \times 0.1/10^3$ = \$ 218,255

Cost of blowdown treatment,

$W_b c_b = 2030 \times (3.259 \times 10^5) \times 0.05/10^3$ = \$ 33,079

Maintenance of towers,

$C_m = Nc_m = 12 \times 200$ = \$ 2,400

Makeup water cost with open-cycle system, M' = (\$ 0)

Blowdown treatment cost with open-cycle system, B' = (\$ 0)

Maintenance cost of open-cycle system, C'_m = (\$ 0)

TOTAL ANNUAL OPERATING COST, OC = \$ 713,734

Total costs

From equation (20), the total excess unit cost due to backfitting, tc, is given by

$$tc = \frac{OC + CC \times FCR}{8760 \times P^*}$$

$$= \frac{713{,}734 + (4{,}917{,}300 \times 0.179)}{8760 \times 312.5 \times 1000}$$

$$\boxed{tc = 0.5822 \text{ mills/kW-hr}}$$

The costs in the above equation are seen to be close to the results given by the computer calculations included in Appendix III (OC = \$714,691/yr, CC = \$4,916,361 ; tc = 0.5825 mills/kW-hr).

J. EXAMPLE OF A VARIABLE LOADING PATTERN

A general discussion of the treatment of a variable loading pattern is presented in Section IV.D. A hypothetical example for the purpose of illustrating differences with the idealized full-throttle loading pattern is now given. The mechanical-draft wet cooling tower problem of the preceding section was rerun, employing the computer model in Appendix III, with a variable loading pattern. To summarize the features of this pattern, the full-throttle was maintained for about 55% of the meteorological conditions (when possible) and a 0.7 throttle opening was maintained for the rest of the meteorological conditions (when possible). The actual loading pattern considered is given in Table 7. It is not implied that the variable loading pattern is practical or realistic, and it is considered merely for illustration.

The summary results of these calculations appear in Appendix III, following those corresponding to the example calculations for the full-loading pattern. Several interesting differences in the results are worthy of comment here and are summarized in Table 8. The values presented in the table are from the computer calculations.

The excess fuel consumption is nearly zero for the full-throttle case while definitely nonzero for the reduced loading pattern. This difference is due to the change in the open-cycle fuel consumption with the

Table 7. VARIABLE LOADING PATTERN FOR MECHANICAL-DRAFT WET COOLING TOWER EXAMPLE

fraction of full loading (frequency of occurrence)

Dry Bulb Temp, T_{db} °F (°C) \ Wet Bulb Temp, T_{wb} °F (°C)	20-30 ([-6.7]-[-1.1])	30-40 ([-1.1]-4.4)	40-50 (4.4-10.0)	50-60 (10.0-15.6)	60-70 (15.6-21.1)	70-80 (21.1-26.7)	80-90 (26.7-32.2)	90-100 (32.2-37.8)
20-30 ([-6.7]-[-1.1])	0.0							
30-40 ([-1.1]-4.4)		0.7 (0.0003)						
40-50 (4.4-10.0)		0.7 (0.0027)	0.7 (0.0084)					
50-60 (10.0-15.6)			0.7 (0.0200)	0.7 (0.0283)				
60-70 (15.6-21.1)			0.7 (0.0033)	0.7 (0.0570)	0.7 (0.0838)			
70-80 (21.1-26.7)				0.7 (0.0146)	0.7 (0.1945)	1.0 (0.2667)		
80-90 (26.7-32.2)				0.7 (0.0002)	0.7 (0.0333)	1.0 (0.2632)	1.0 (0.0092)	
90-100 (32.2-37.8)					0.7 (0.0001)	1.0 (0.0064)	1.0 (0.0078)	
100-110 (37.8-43.3)								0.0

Table 8. COMPARISON OF SELECTED RESULTS FROM THE MECHANICAL-DRAFT WET COOLING TOWER EXAMPLES FOR DIFFERENT LOADING PATTERNS

	Full-throttle loading	Variable loading (Table 7)
excess fuel consumption	0	6.829 MW
energy loss	46086 MW-hr	27388 MW-hr
water evaporation	4668 acre-ft /yr (5.758×10^6 m^3/yr)	4230 acre-ft /yr (5.218×10^6 m^3/yr)
blowdown	2030 acre-ft /yr (2.504×10^6 m^3/yr)	1839 acre-ft /yr (2.268×10^6 m^3/yr)
total capital cost	$ 4,916,361	$ 4,443,861
total differential annual operating cost	$ 714,691 /yr	$ 549,083 /yr
total differential unit cost	0.5825 mills/kW-hr	0.5671 mills/kW-hr

reduced loading, resulting in an increase in the excess fuel consumption. The energy loss, water evaporation, blowdown, and total differential annual operating costs are all greater for the full-throttle operation than for the reduced loading, as expected. More power is produced under full loading, which is expected to generally increase all of these absolute quantities (as compared to the relative quantity of excess fuel consumption). The decrease in the total capital cost for the variable loading pattern reflects the difference in the energy loss during downtime because of operation at a lower power level. It is interesting to note that in this comparative example, the variable loading pattern exhibits a 23% decrease in differential operating costs and a 9.6% decrease in capital costs consequent with lower turbine output. However, the decrease in the total differential unit cost is only 2.6% because it is prorated with respect to a larger annual energy output.

Even though it is not used in the present study, the capacity factor may be computed from the variable loading pattern given in Table 7. The capacity factor, CF, which is the ratio of the annual design power output (power demand) to the maximum possible annual power production, is computed as the sum of the products of the fraction of full loading multiplied by the corresponding frequency of occurence over all meteorological conditions. For the variable loading pattern under consideration, CF = 0.834.

SECTION VI

NATURAL-DRAFT WET COOLING TOWERS

As is true for the mechanical-draft wet cooling tower, already discussed in Section V, the amount of cooling obtained with an evaporative, natural-draft cooling tower depends primarily upon the ambient wet-bulb temperature, the temperature of hot water entering the tower, and the size and thermodynamic characteristics of the "wet pile" inside the tower. Furthermore, since the air flow is generated by the difference in air densities inside and outside the tower shell, and not by a fan, the air-flow rate and hence, the tower performance is also dependent on the ambient dry-bulb temperature of the air. As is true for mechanical-draft towers, much of the empirical information on the design and heat transfer properties of natural-draft towers is regarded as proprietary by the manufacturers for obvious reasons. An attempt is made in the present study to develop a methodology that is capable of accepting any set of design parameters so that the performance of towers of different designs can be analyzed. Detailed example results are then presented for a particular set of input parameters which were obtained through the cooperation of a leading manufacturer of crossflow cooling towers. These example results therefore apply to CROSSFLOW, NATURAL-DRAFT WET TOWERS. It is believed, however, that equipment designs are not so radically different that the applicability of the example cost information is limited to the product of a single manufacturer.

A significant number of comments which are applicable to cooling towers or to closed-cycle cooling systems in general have already been listed in Section V. Therefore, this and the two succeeding sections will follow the format of Section V closely with reference to relevant comments made therein. As is true for the other closed-cycle cooling systems, the physical size of the natural-draft tower will be different in a backfit situation than for a new plant due to the economic peculiarities of the backfit situation. Throughout this section (as with other closed-cycle cooling systems), the physical size of the tower is regarded as a primary variable so that the various quantities of interest (as outlined in Section V) can be calculated for a range of sizes. These quantities are then to be used in conjunction with the economic considerations outlined in Section IV, to identify the project costs.

A typical natural-draft, crossflow, evaporative cooling tower is shown in Figure 29 where it is seen that the overall tower structure consists of an annular evaporative pile about the bottom circumference of a tower shell. For structural reasons, the shape of the tower shell is prescribed by the equation of a hyperboloid. Thus, the physical size of a tower is specified by the width, W, and height, H, of the evaporative pile; by the shell height, S; the height of the "throat" section of the shell, T, and by the diameters of the shell at the bottom, D_1, and throat, D_2. These six parameters can be used as primary indicators of the physical size of the cooling tower. However, so many variables make example calculations intractible. Therefore, a "standard shell shape" is assumed (Figure 29) which is used by one major cooling tower manufacturer and is believed to be representative of most shell shapes employed in the United States. The ratios of the shell dimensions portrayed in Figure 29 (r_1 = T/S, $r_2 = D_2/D_1$ and $r_3 = S/D_1$) as actually used in tower construction are proprietary information. However, if these ratios are known, the physical size of a tower can be specified by three variables W, H, and S together with the equation of the hyperboloid. These quantities are used as the primary variables in the example calculations which follow. It will be clear that H is the length of the water path and is a measure

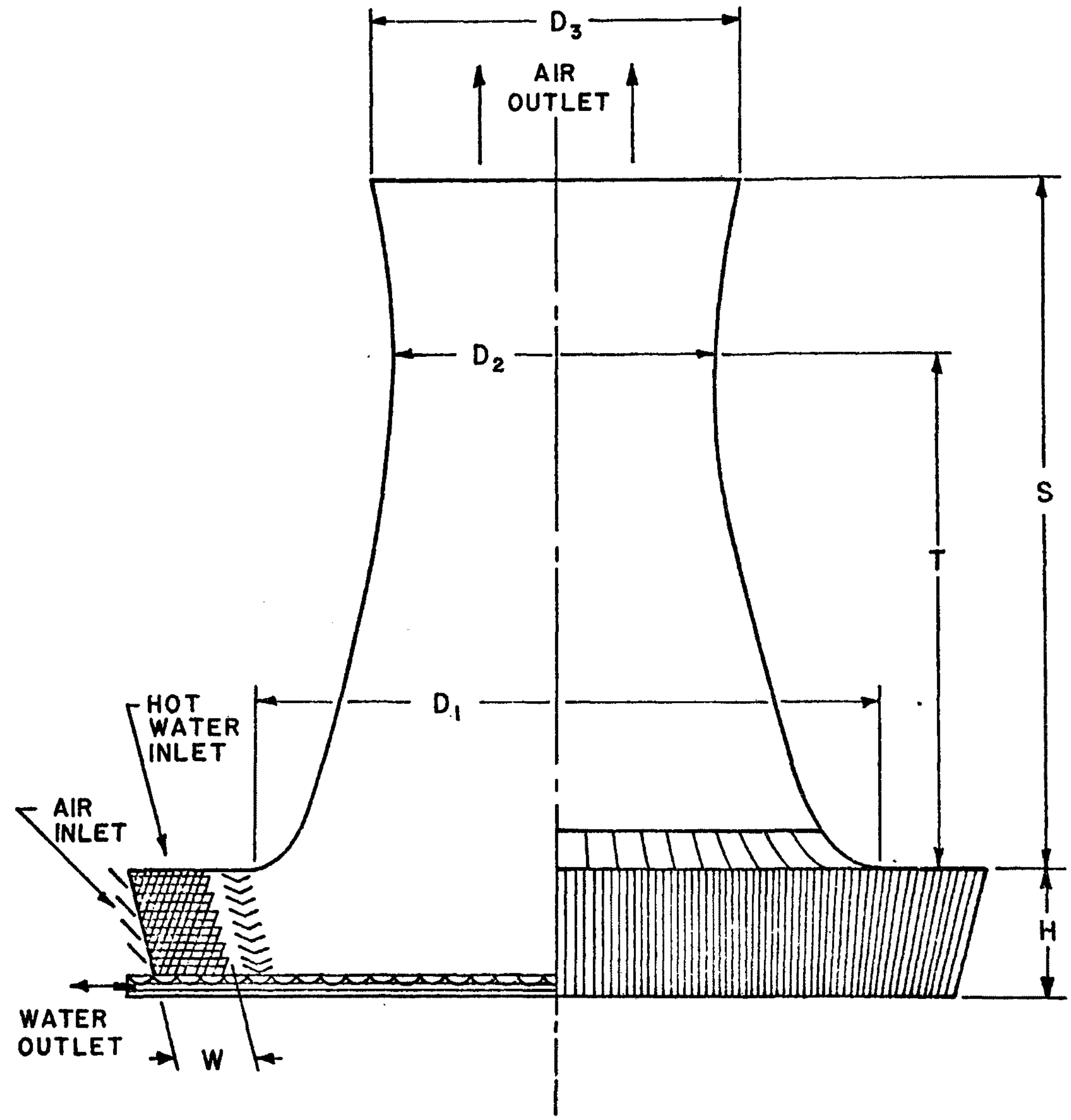

Figure 29. Hyperbolic natural-draft crossflow, wet cooling tower

of the pumping height required. Also, W is a measure of the length of the air path and therefore will influence the air-flow resistance. Furthermore, the shell height, S, determines the overall shell dimensions and will influence air flow both by determining reference air density at the top and by its "pipe flow" resistance. It is important to remember that the air flow rate is a complex function of shell geometry and the thermodynamic properties of the evaporative pile.

For a given air-flow rate, water-flow rate, dimensions of the fill, empirical heat-exchange coefficients of the fill, and hot-water temperature, the basic theory of Merkel [9,10,11,12,13,14,15] can be used to calculate the temperature of the cold water, the temperature and humidity of the exit air, and the heat rejection rate from the cooling water through the pile. In fact, these calculations exactly parallel those for the evaporative pile of the mechanical-draft crossflow tower as already described by Croley, Patel and Cheng [15] for a given air-flow rate. Of course, the dimensions and heat exchange coefficients of the pile are different, but the calculations are the same. However, an additional complication arises due to the wide fluctuation in the air-flow rate with air temperatures, water temperatures, and heat rejection rate for a given tower design. In actuality, the air-flow rate determines the heat rejection rate, cold-water temperature, and pile and shell flow resistance. In turn, air-flow rate is determined by the inside air temperature and humidity, outside air temperature and humidity, and pile and shell friction losses. Therefore, the joint determination of air-flow rate and heat rejection rate are necessary to determine operation characteristics of a given tower design at specified values of the air dry- and wet-bulb temperatures and hot-water temperature. This joint determination is described shortly. First, several basic models are described which are necessary for the joint determination. Then, the basic calculation of heat rejection rate, cold-water temperature, and air-flow rate for a given tower design and specified meteorological conditions are described.

A. OPERATION MODELS FOR NATURAL-DRAFT, CROSSFLOW, WET COOLING TOWERS

The operation of the natural-draft cooling tower depends heavily, as already mentioned, on the air-flow rate, which in turn depends upon ambient dry- and wet-bulb air temperatures, the hot-water temperature and the heat rejection rate for a given tower design. As the air passes through the tower, it experiences frictional head losses in flowing through the pile and through the shell. Plots of the pressure drop of the air flow through the pile as a function of the air-flow rate loading are made by manufacturers; see e.g., Figure 30. Such proprietary information establishes the flow resistance through the pile. The hyperbolic shell can be considered approximately as a large circular cylinder of the same height with some mean diameter. This diameter is calculated as that which yields the same cylindrical volume as contained in the hyperbolic shell. Such expressions have been used elsewhere [18] for simplification of geometries. Air flow through this equivalent cylindrical shell can be approximated as incompressible pipe flow. Therefore, the Darcy-Weisbach friction factor can be found from standard hydraulic charts as a function of the Reynolds number and the relative roughness of the pipe (assumed to be zero, representing a smooth pipe since the diameters are large). Both of these frictional head losses can then be combined to give an overall head loss coefficient, K as follows:

$$K = (\ell + \ell_p) \frac{2g}{V^2} \tag{35}$$

in which K = overall head loss coefficient, ℓ = frictional head loss in the shell (ft of air), ℓ_p = frictional head loss in the evaporative pile (ft of air) and V = average velocity of air in the tower cylinder (ft/sec).

The model for calculation of the air-flow rate for given values of the ambient dry- and wet-bulb temperatures, the hot-water temperature, the tower resistance coefficient, K, and a given design can be described

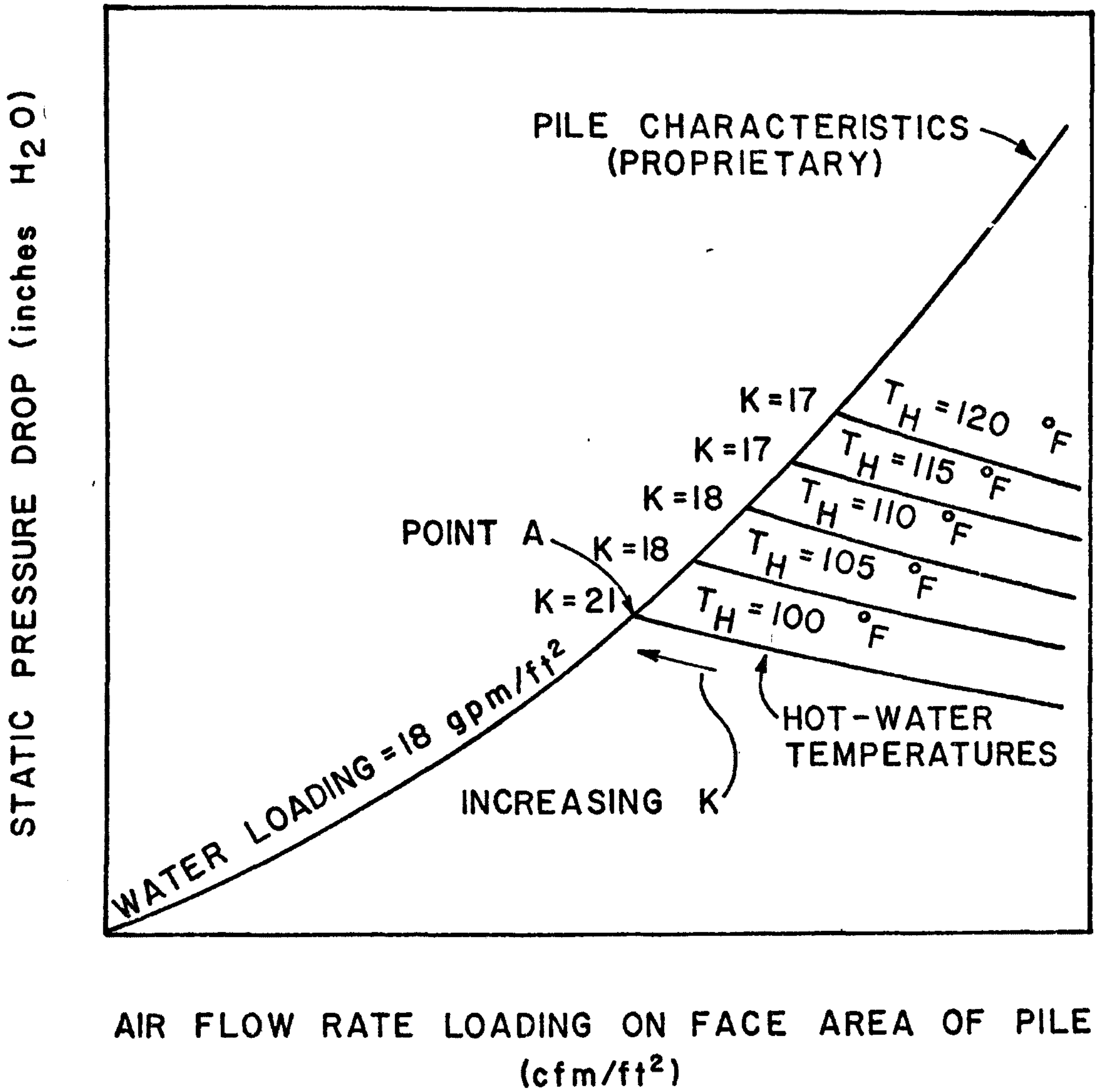

Figure 30. Pile characteristics curve and air flow rate calculations

as follows:

1. Calculate the air pressure at the top and bottom of the shell as a function of air temperature, assuming a "standard atmosphere" [19];
2. Calculate the humidity and density of the incoming ambient air;
3. Assume that the exit air temperature is equal to the ambient dry-bulb temperature as a first approximation; assume air-flow rate is zero as a first approximation;
4. Calculate the exit air density at the exit air temperature assuming saturation;
5. Calculate the air-flow rate through the tower using the Bernoulli equation for incompressible flow with no energy inputs [20] and K;
6. If the air-flow rate of step 5 is sufficiently close to the previous value, stop, otherwise proceed;
7. Calculate the cold-water temperature and heat rejection rate from the pile using this air-flow rate and standard thermodynamic models [15];
8. Assume that the exit air temperature is equal to the average of the hot- and cold-water temperatures [21,22]; and
9. Go to step 4.

Although the use of the Bernoulli equation in step 5 and the assumption of step 8 are simplifications, they were made in the interest of brevity and have been used elsewhere in design applications [20,21,22]. More complete thermodynamic balances are presently under research.

The procedure just identified above is referred to as subroutine AIRFLR in the computer model listings in Appendix IV. The use of this model can be made for any value of K for a given set of meteorological conditions, hot-water temperature and a given tower design. However, the values of K for different tower designs are not readily available information. Thus, a second model for determination of the tower resistance parameter, K, for given values of dry- and wet-bulb temperatures and hot-water temperature can be described as follows:

1. Specify dry- and wet-bulb temperatures and hot-water temperature;
2. Arbitrarily pick a "K" value;
3. Solve for air-flow rate using subroutine AIRFLR;
4. Calculate corresponding pile losses, ℓ_p, using K, ℓ and the air-flow rate (see equation 35);
5. Plot ℓ_p on pile characteristics chart as in Figure 30 and
6. Repeat steps 3 through 5 for selected values of "K" until a point is found (in Figure 30) corresponding to the flow characteristics of the pile (point A).

In general, it is found from this procedure that points corresponding to these calculations have associated K values which are small for high air-flow rates and large for low air-flow rates. Thus, the lines cross the pile characteristics curve somewhere, and the associated K value indicates the equivalent tower resistance parameter for this pile and shell. This calculation can be repeated for other values of hot-water temperature and air dry- and wet-bulb temperatures; see Figure 30. However, in the preliminary studies conducted under this research, it has been observed that the selected value of K does not change greatly. Thus, the pile characteristics curve in Figure 30 represents a nearly constant K value, as is expected. Furthermore, K values of the order of 20 to 30 are observed. The resulting air-flow rate, pressure drop, and tower cooling rate fluctuate very narrowly for variations of the K values in this range. Thus, it is deemed sufficient to perform the calculations (in the procedure just presented) for a few air and water temperatures, selected to cover the range of the pile characteristics curve in Figure 30, and then take the average selected K value as the best estimate of the overall tower resistance parameter for use at any conditions. This entire procedure is represented as subroutine BESTK in the computer model listings in Appendix IV.

In the operation models for this tower configuration, the subroutine BESTK is used to initially find the best "K" value for use in all subsequent calculations, for a given tower design. Then for any specific set of meteorologic conditions and hot-water temperature, subroutine AIRFLR is used to jointly determine the resulting air-flow rate, cold-water temperature and heat rejection rate, using the best "K" value. Other thermodynamic models and calculations are similar to those already described [15]. The economics models are also presented in Sections IV and V.

B. CAPITAL COST OF TOWERS AND AUXILIARY EQUIPMENT

As with the case of mechanical-draft crossflow cooling towers, the capital cost of a natural-draft crossflow cooling tower will be determined primarily as a function of its size as represented by the parameters H, W, and S. Again, however, manufacturers of such cooling towers recommend sizing and pricing procedures which bear no direct relation to the physical size of the tower. Instead, the cost of a tower is linked to the "design" meteorological conditions (here the relative humidity is determined from the design values of dry- and wet-bulb temperature) and parameters describing the overall performance of the tower at these design conditions, notably the RANGE and APPROACH. As mentioned earlier the rating-factor tower-unit method [16] and the K-factor method [17] are examples of such procedures for the mechanical-draft tower. Also contained in Ref. 16 is a similar procedure for natural-draft crossflow towers. The manufacturer presents charts, such as those in Figure 31, from which a unit cost (presumably in 1970 dollars per thousand Btu/hr) can be found for any given range, approach and relative humidity. The total capital cost can then be determined by multiplying the unit cost times the heat rejection rate of the turbine (in thousand of Btu/hr). From an analysis of previous experience, Dickey and Cates [16] have found that the scatter associated with use of the curves may be ±9%. Recent correspondence with the manufacturer places this scatter at about ±15%.

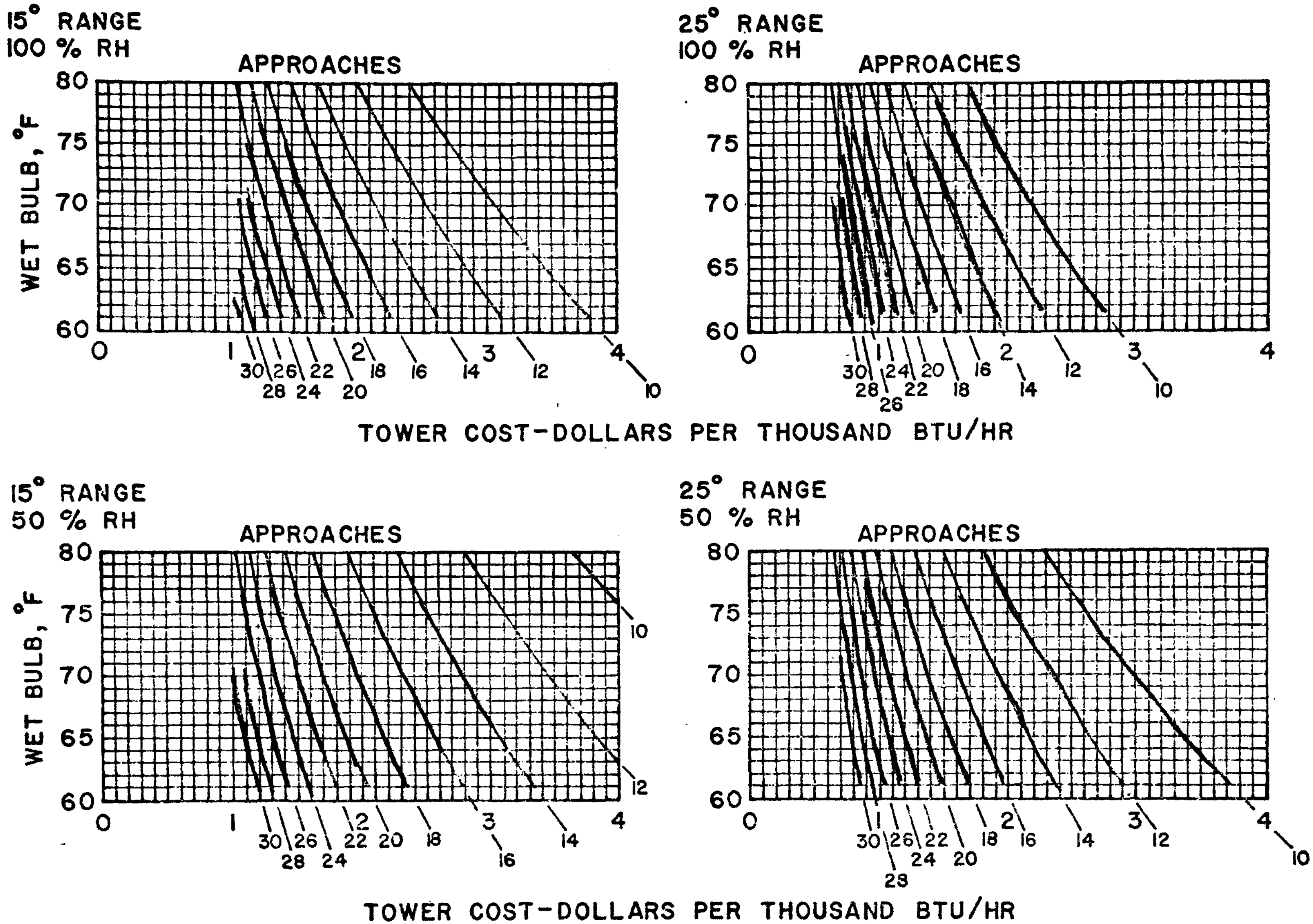

Figure 31. Typical cost-performance curves for budget estimates for the natural-draft crossflow cooling tower (1970 dollars) [16]

As discussed by the manufacturer [16], these curves were intended only for preliminary budget estimates. Economic factors related to each geographical location must be considered, including escalations for lead time, allowance for wind loading requirements, and special site preparations. The curves are suggested for relative evaluations, but also serve as initial estimates when updated appropriately.

In order to proceed further and establish a capability for handling towers of different designs, it is necessary to return to a more basic approach in which the Merkel theory is used to predict the amount of cooling delivered by a tower fill of given type and dimensions. Such a procedure is similar to that described in detail in Ref. 15 and will not be repeated. There it is shown that when the dimensions (W, H, and length, $L = \pi(D_1 + W)$) and heat transfer coefficients of the fill are specified, it is possible to calculate the cold-water temperature, and therefore the heat rejection rate, range, and approach, for any given set of values of the hot-water temperature, air- and water-flow rates and ambient wet-bulb temperature. When the calculations are performed, using the models for air-flow rate described in the preceding section, for the design dry- and wet-bulb temperatures over a range of values of the design heat rejection rates and tower dimensions, and use is made of the unit cost procedures already described, it is possible to calculate capital costs as a function of the tower dimensions as shown in Figure 32.

These results were obtained using a known (proprietary) set of heat transfer coefficients, air- and water-loadings on the pile, and pile resistance. A fixed fill width, W = 21 ft = 6.40 m, had to be used since the available pile resistance data were restricted to that particular value. It should be emphasized that Figure 32 results from a large number of calculations performed using a range of values of heat rejection rate, fill height and fill length, and a number of values of the design dry- and wet-bulb temperatures.

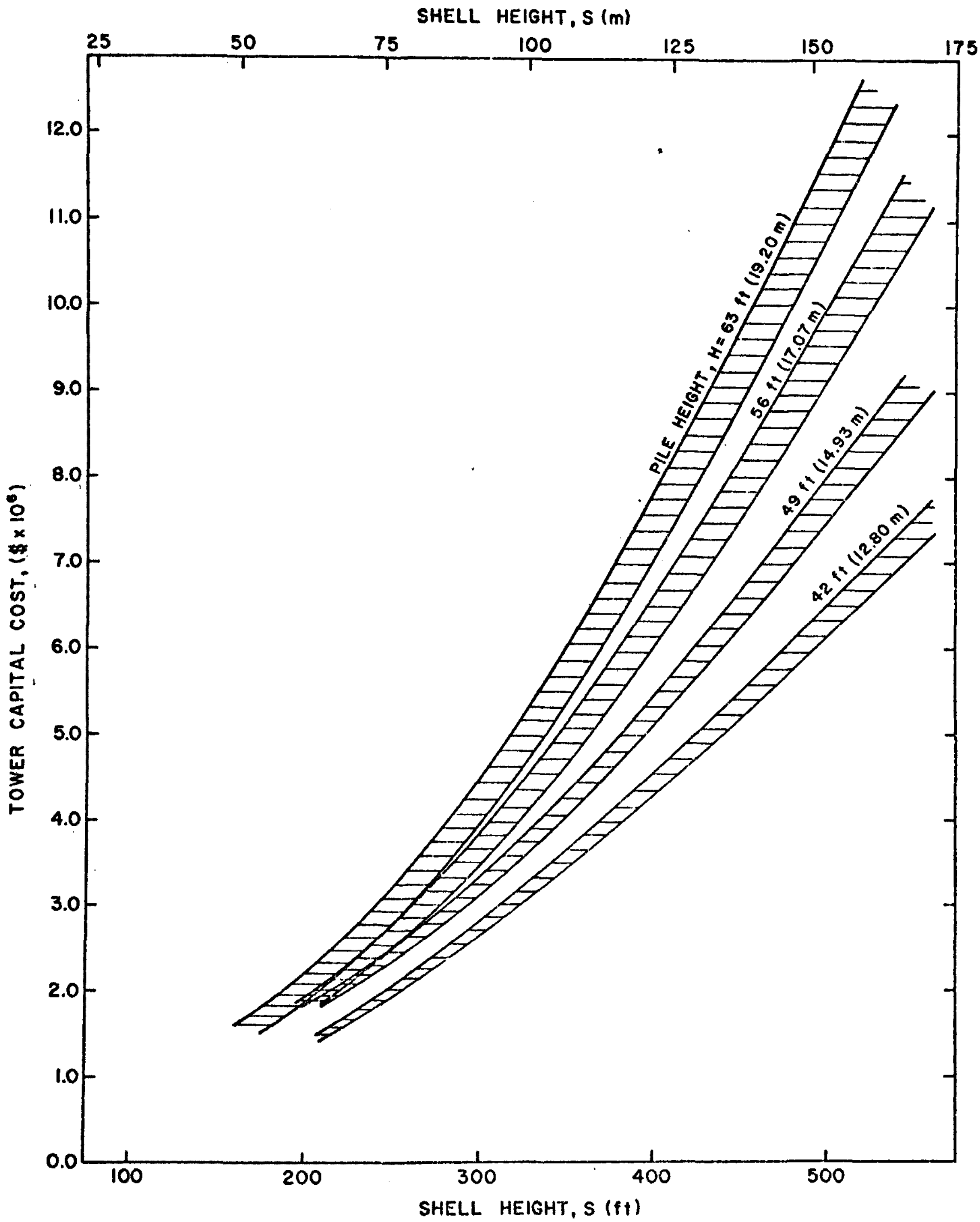

Figure 32. Capital cost estimates for the natural-draft crossflow, wet cooling tower

For each set of conditions (Q, L, H, T_{db_d}, T_{wb_d}), the thermodynamic model of the evaporative pile, described in detail in Ref. 15, was used to calculate the corresponding range and approach. These values, in turn, were used to find the corresponding unit costs from the charts shown in Figure 31. The total heat rejection rate was then used to find the cost corresponding to the specified set of input conditions. For each set of design dry- and wet-bulb temperatures and pile height, the cost was found to be a function of the pile length (and hence the shell height using the shape of Figure 29), irrespective of the heat rejection rate. A small scatter was observed between the results obtained with different design dry- and wet-bulb temperatures, and this is shown by the shaded area in Figure 32. While the scatter is somewhat consistent, insofar as smaller costs correspond to lower design temperatures, its origin lies mainly in the fact that a highly complex phenomenon is being represented in a relatively simple form. In any case, the scatter is small and well within the accuracy expected from the various assumptions made in the thermodynamic model of evaporative cooling. The most remarkable feature of Figure 32 is that the cost of the tower is primarily a function of the dimensions of the fill, and hence the shell height, as was conjectured earlier. Thus, for the estimation of the capital cost of natural-draft cooling towers, either Figure 31 or Figure 32 can be used, depending upon the information that is known.

Estimates of the capital cost of the pump and piping system are made as a function of the water-flow rate, and Figure 13 is also used for that calculation. Comments on the condenser design are similar to the mechanical-draft discussion, and equation (23) also applies for natural-draft towers.

The additional land area required for backfitting with a natural-draft evaporative cooling tower depends mainly upon the plan area of the tower and possibly upon the consideration of an acceptable noise level and other site-dependent conditions. Unlike the criteria for mechanical-draft towers, however, there are no problems of interference and plume recirculation due to the height of the natural-draft towers.

The EPA Development Document [2, p. 631] suggests the allowance of a clear area 100 ft (30.5 m) wide around the natural-draft tower. The required land area, A_L, for a natural-draft wet cooling tower of bottom diameter D_1 is therefore

$$A_L = \frac{\pi}{4} (D_1 + 2D)^2 \tag{36}$$

where D, the width of the clear area around the tower, may be 100 ft (30.5 m) according to the above criterion or some other value. Other land requirement standards for natural-draft towers given in Ref. 2 (p. 631) include the specification of 350 to 400 sq. ft (32 to 37 sq. m) per megawatt.

As for the mechanical-draft towers, land requirements for natural-draft towers based on acceptable noise levels have also been studied in Ref. 7 (Vol. I, Appendix G). In the hypothetical test case presented in part H of this Section, the land area requirement is computed on the basis of a noise level limit of 60 dBA; the width of the clear area around the tower is thus found to be D = 200 ft (61 m). The additional land area requirement for this example may easily be found from equation (36).

C. REFERENCE SIZE OF COOLING TOWERS, S^*

A number of quantities which characterize the operation of an existing power plant or unit using open-cycle cooling was defined in Section IV and part B of Section V. As in Section V.B, it is convenient to define a reference size of the natural-draft cooling towers for the purpose of nondimensionalization. For any given pile height, H, and pile width, W, the reference size of a tower, S^*, can be defined as the shell height required to reject Q^* while maintaining the back pressure at p' and delivering P^* at some reference ambient dry- and wet-bulb temperatures. The reference dry- and wet-bulb temperatures and p' can be selected arbitrarily without loss of generality. In the example calculations, the pile width, W, is held constant at 21 ft (6.40 m), p'=1 in. Hg abs (2.54 cm Hg abs) and the reference dry- and wet-bulb temperatures are set equal to 78°F and 68°F (25.6°C, 20.0°C) respectively. It should be noted that these reference conditions are not necessarily the same as those adopted in defining the reference sizes of the other closed-cycle cooling systems; see e.g., Section V.B. The reference size of each cooling system (L^*, S^*, A^*, or N^*) is used only for nondimensionalizing the cooling system size, and, therefore, as long as the reference size is known (Figures 16, 33, 54, and 63), the proper economic assessment of the prototype cooling system can be made. The reason for employing different sets of reference meteorological conditions is related to the peculiarities of each cooling system. For example, practical experience with the operating characteristics of natural-draft cooling towers suggests the use of more extreme reference meteorological conditions.

It is clear that S^* can be found for any given set of values of Q^* and H using the theory of Merkel in conjunction with the known heat transfer properties of the condenser and the evaporative pile, and the water-loadings recommended by the tower manufacturers. The computation construction outlined in Section V.B was also used with the natural-draft models to calculate the reference shell height, S^*. With a water loading of 18 gpm/ft^2-plan area (0.733 m^3/min/m^2-plan area), and appropriate information concerning the heat transfer properties of the fill, the dependence of S^* on Q^* and H is shown in Figure 33. It will be seen that the reference shell height is a nonlinear function of Q^* and decreases with increasing H for a constant Q^*. As in mechanical-draft towers, an attempt was made to use the reference shell height, S^*, to normalize the tower size so that the example results could be employed to assess the performance of power plants or units with different nameplate capacities and heat rejection rates. It is again important to emphasize that the foregoing considerations apply regardless of the type of turbine that is employed

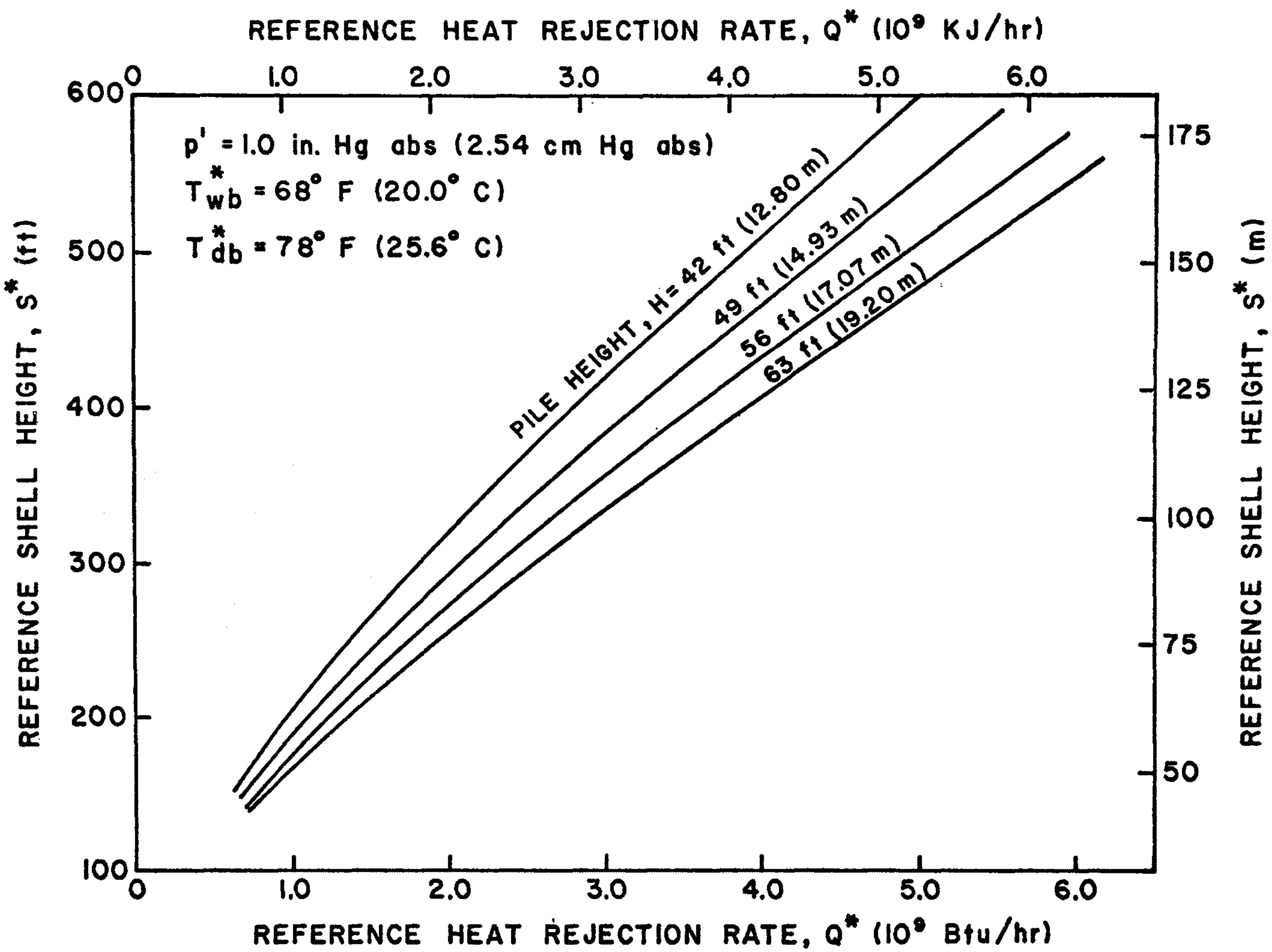

Figure 33. Determination of reference shell height

since the definition and evaluation of S^* is independent of the source of Q^*.

D. OPERATION OF A TOWER OF GIVEN SIZE (S,H)

When the width of the pile, W, is fixed, the physical size of the cooling tower is characterized by only two parameters, namely the shell height, S, and the pile height, H. In turn, all other dimensions may be determined from these two, using the simplifications presented at the beginning of this section. For the example calculations illustrating natural-draft towers, representative values recommended by leading manufacturers will be used for the various quantities. Following the procedure outlined in Section V.C and described in Ref. 15 (see Sections III.B, E, and F), the detailed operation of a turbine-condenser cooling system may be found for any and all meteorological conditions. As with the mechanical-draft cooling tower, these models were used with the natural-draft cooling tower to evaluate maximum capacity loss, C_L, annual energy loss, E_L, annual fuel "penalty", F_E, and annual evaporative water loss, W_L, as described in Section V.C.

E. PARAMETRIC STUDIES

Detailed calculations of the type mentioned above were performed for a range of values of the tower shell height and pile height. The results were nondimensionalized by again employing suitable scaling parameters. The tower shell height, S, was normalized with respect to the reference shell height, S^*; the capacity loss was normalized with respect to the nameplate capacity; the energy loss and excess fuel consumption were normalized using the maximum energy that can be produced in a year, PE^*, and the water evaporation was normalized with respect to the reference heat rejection rate, Q^*. Figures 34 through 37 show the variations of the normalized maximum capacity loss, the normalized annual energy loss, the normalized annual fuel penalty, and the normalized annual evaporative water loss with normalized tower size for the particular case of turbine A (i.e., $P^* = 411$ MW,

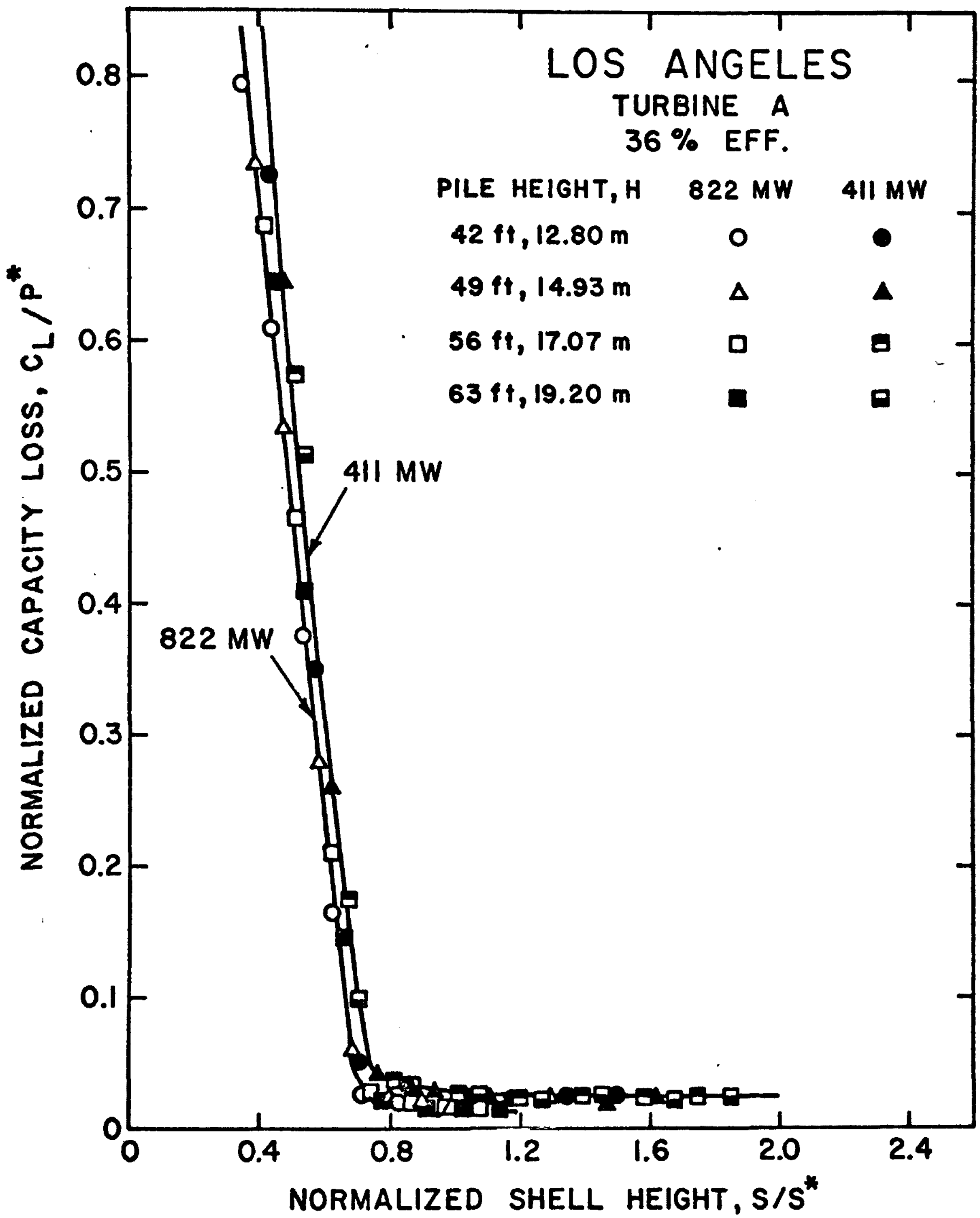

Figure 34. Normalized capacity loss

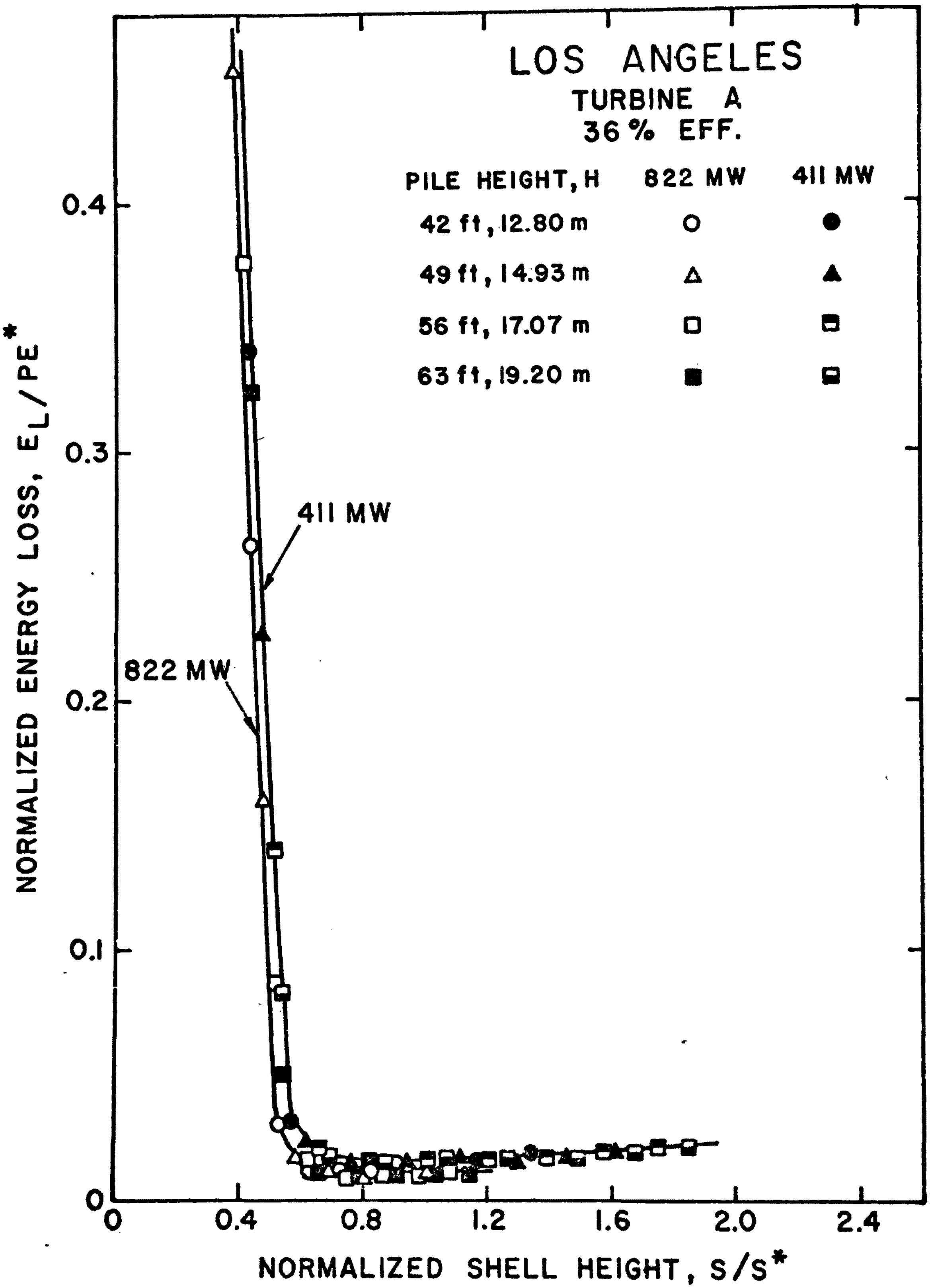

Figure 35. Normalized energy loss

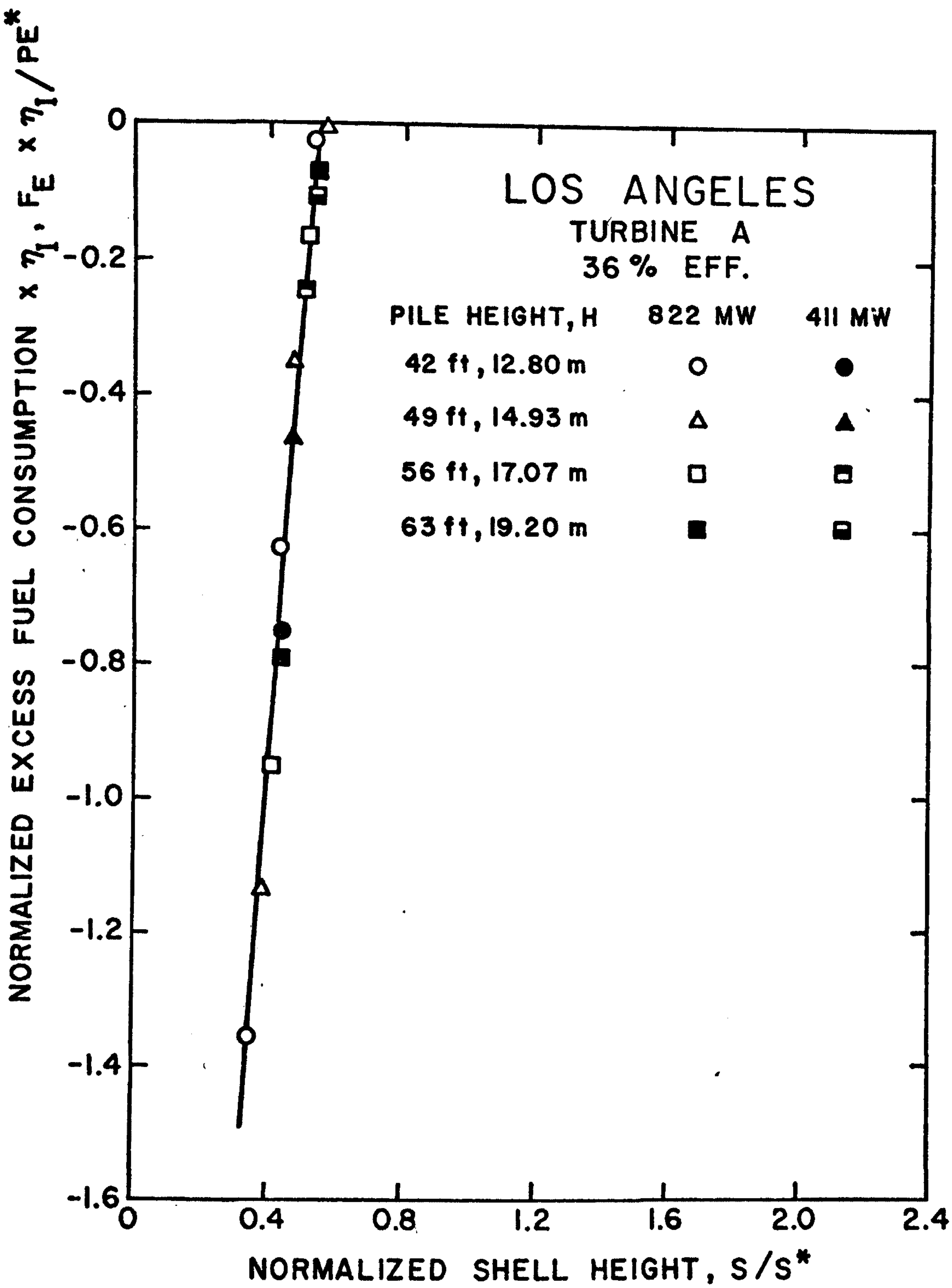

Figure 36. Normalized excess fuel consumption

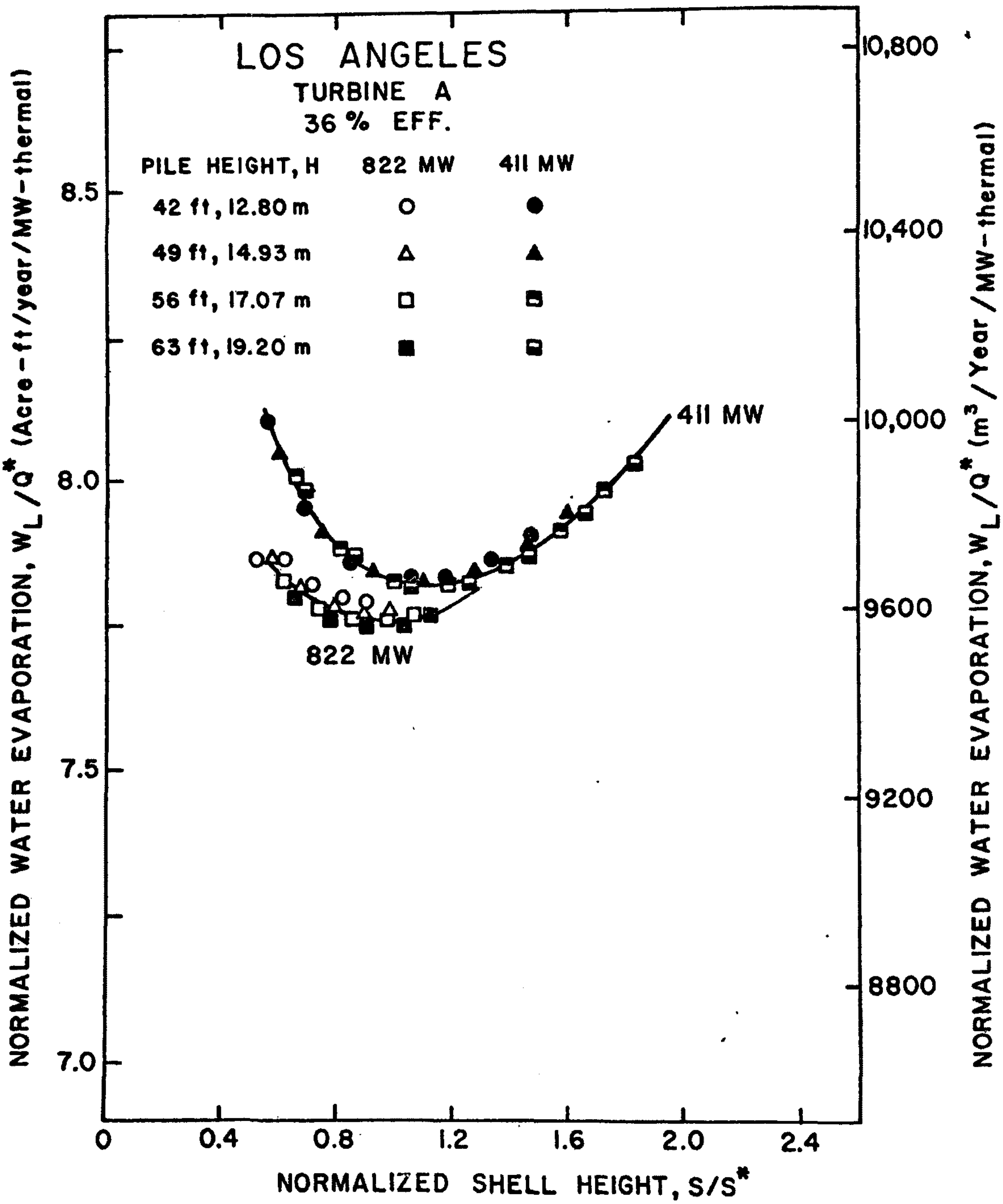

Figure 37. Normalized evaporation

$p^* = 1.00$ in. Hg abs = 2.54 cm Hg abs, and $Q^* = 2.545 \times 10^9$ Btu/hr = 2.686×10^9 kJ/hr for fossil-fuel operation, see Table 1) for the meteorological conditions at Los Angeles (Tables 4 and 5). Also shown in these figures are the same results for a hypothetical turbine whose nameplate capacity and reference heat rejection rate are twice those of turbine A (i.e., $P^* = 822$ MW, $Q^* = 5.090 \times 10^9$ Btu/hr = 5.372×10^9 kJ/hr) but whose basic heat rate characteristics are the same as those of turbine A (Figure 4). The observations presented in Section V.D as comments (a) through (e) can also be made for Figures 34 through 37 regarding natural-draft evaporative cooling towers. Furthermore, the following two comments are also in order.

(a) The annual energy loss, E_L, increases at a slightly higher rate than the maximum capacity loss, C_L, as compared to the mechanical-draft results (Figures 21 and 22). This difference is mainly due to the wide fluctuation of the air-flow rate in the natural-draft tower as compared to the constant air-flow rate of the mechanical-draft tower. At extreme meteorological conditions, energy losses might be realized with the large natural-draft towers which may not occur with the large mechanical-draft towers.

(b) The nondimensionalizing process did not result in a complete collapse of all results into single curves (for both the 411 MW and 822 MW outputs) as they did for the mechanical-draft calculations. Furthermore, the problem cannot be resolved by redefining the reference size, S^*, used in nondimensionalizing the shell height, S. All that can be accomplished with a redefinition of the reference size is to stretch and/or move the sets of curves in the horizontal sense only. There would still be similar differences between the two sets of curves in the vertical direction. The major implication of this difference is that for natural-draft cooling towers, there is an "economy of scale" operating. It is clear that the costs of capacity loss, energy loss, excess fuel consumption, and water loss are directly related to these quantities, and, therefore, the vertical axes of Figures 34 through 37 may be interpreted as costs. It is seen from the figures that the unit costs decrease as the turbine size increases. Since the unit costs are dependent upon the reference heat rejection rate, one would not observe a collapse in the results for calculations repeated with the same value of P^* but different Q^* reflecting type of unit (fossil or nuclear) as with the mechanical-draft towers (see Section V.D). Thus, the operating characteristic curves for natural-draft crossflow evaporative towers are dependent upon the turbine efficiency at reference conditions (P^*, Q^*, p^*).

In view of the discrepancies encountered in the calculations for natural-draft towers, as compared to mechanical-draft towers, the following two departures from procedures established with the mechanical-draft example presentations are made. The first procedural deviation is that all operating characteristics plots are based upon a dimensional tower size. There is no advantage to be gained in nondimensionalizing tower size as just discussed. The second deviation is that all calculations are repeated for a second base turbine efficiency, η_T = 28% (in addition to the assumed 36%). The example results can then be used by applying two corrections, described in detail in Sections VI.F and VI.H. Briefly, the corrections involve making coarse adjustments by means of interpolation or extrapolation to graphical presentations of capacity loss, energy loss, fuel penalty, and water loss, with regard to observed deviations due to differences in nameplate capacity and turbine efficiency. Because of the shape of these curves, a logarithmic-linear interpolation/extrapolation procedure is employed.

In order to build up a representative library of the operating characteristics of natural-draft cooling towers, a parametric study was conducted using the following information:

- Heat rate characteristics of turbines A, B, C (Figures 4, 5, 6);
- Meteorological data at Chicago, Los Angeles, Miami, and St. Louis;
- Base turbine efficiences, η_T = 36% and 28%;
- Power levels of 822 MW and 411 MW for turbine A at Los Angeles.

As indicated in Sections IV and V, and above, these conditions are expected to represent a majority of the situations which will be encountered in the consideration of backfitting across this country. The final results are presented in Figures 38 through 53.

Comments on the operating costs associated with the natural-draft cooling towers are the same as in Section V.E with the exclusion of items pertaining to the fans and fan requirements. Also, the unit maintenance costs for natural-draft towers are of the order of $1,000 to $3,000 per tower per year. This figure was estimated by the writers from considerations of mechanical-draft maintenance costs. The actual figure will depend upon the particular tower design and its size. Maintenance costs found in the literature [23] appear to be too large and probably include other items such as pump operating costs.

F. PROCEDURE FOR THE ECONOMIC EVALUATION OF BACKFITTING

The various items which must be considered in the evaluation of the cost of backfitting an existing power plant or unit with natural-draft wet cooling towers have been described individually in the preceding sections. The manner in which these items are to be combined in order to calculate the total cost of backfitting is presented in this part followed by a brief description of the computer program that has been developed for this purpose. A hypothetical test case is presented in part H to illustrate the general methodology and use of the graphical results. Reference will be made to related portions of the preceding Sections where indicated.

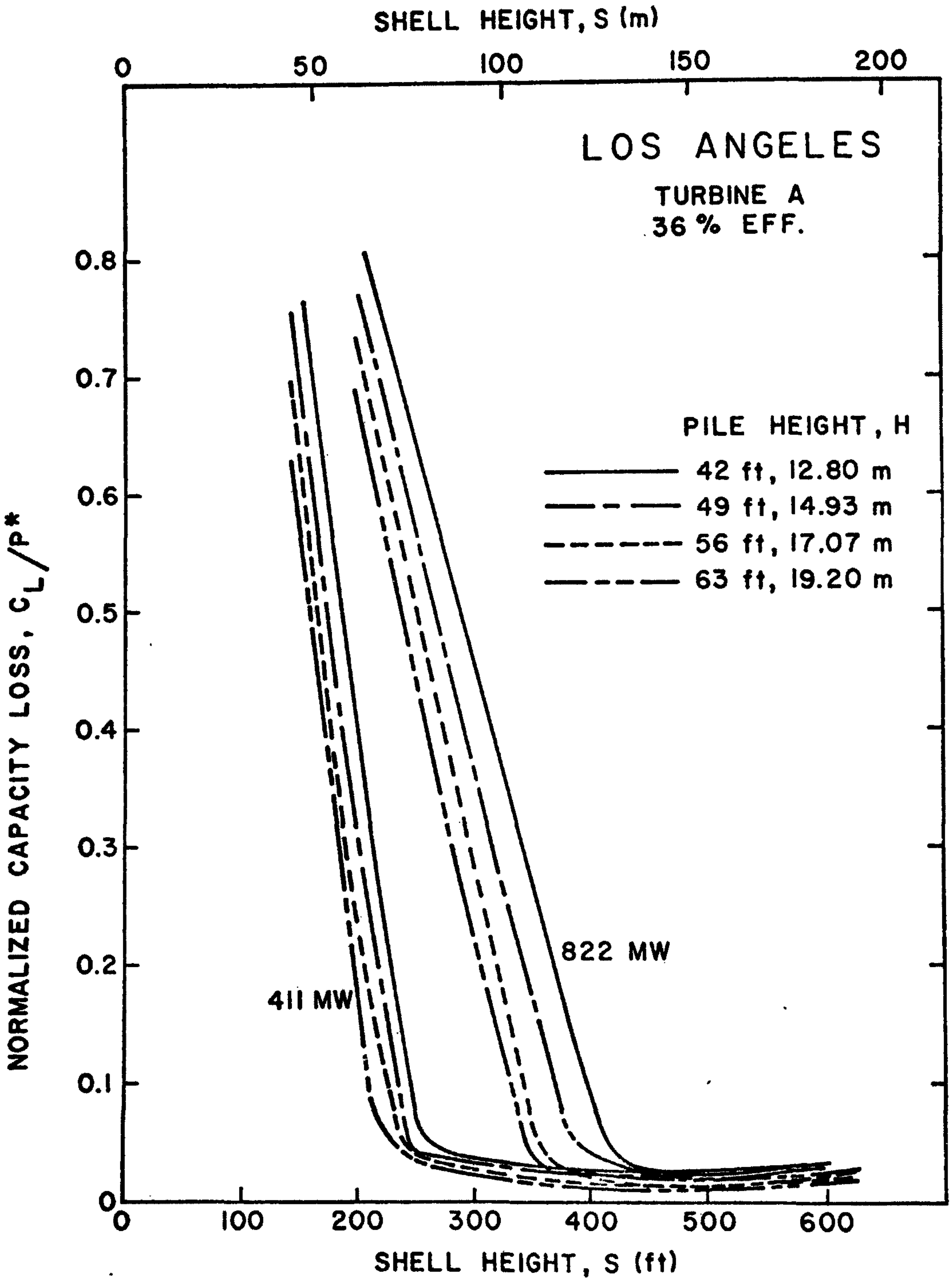

Figure 38. Variation of normalized capacity loss with shell height and plant capacity, 36% turbine efficiency

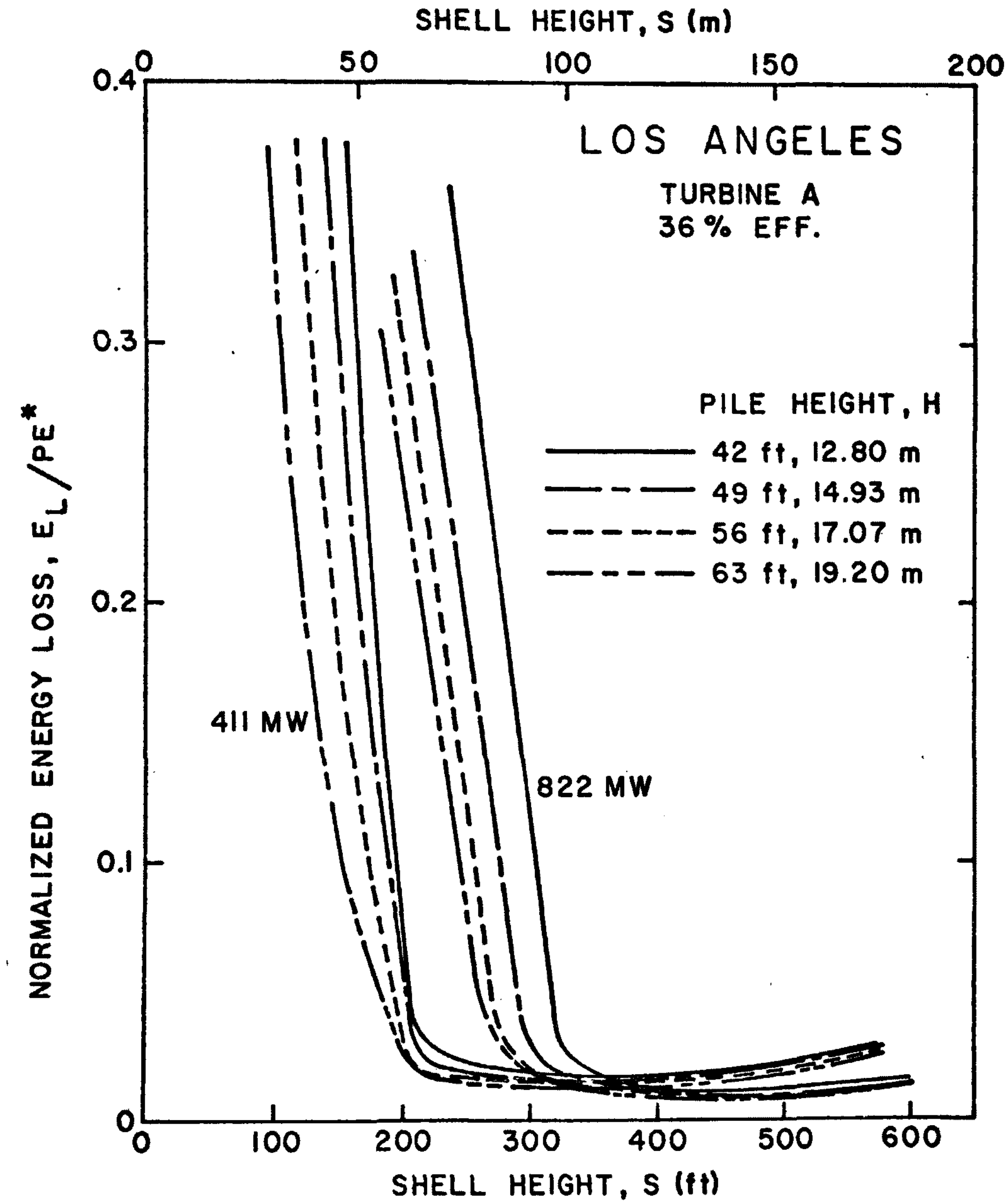

Figure 39. Variation of normalized energy loss with shell height and plant capacity, 36% turbine efficiency

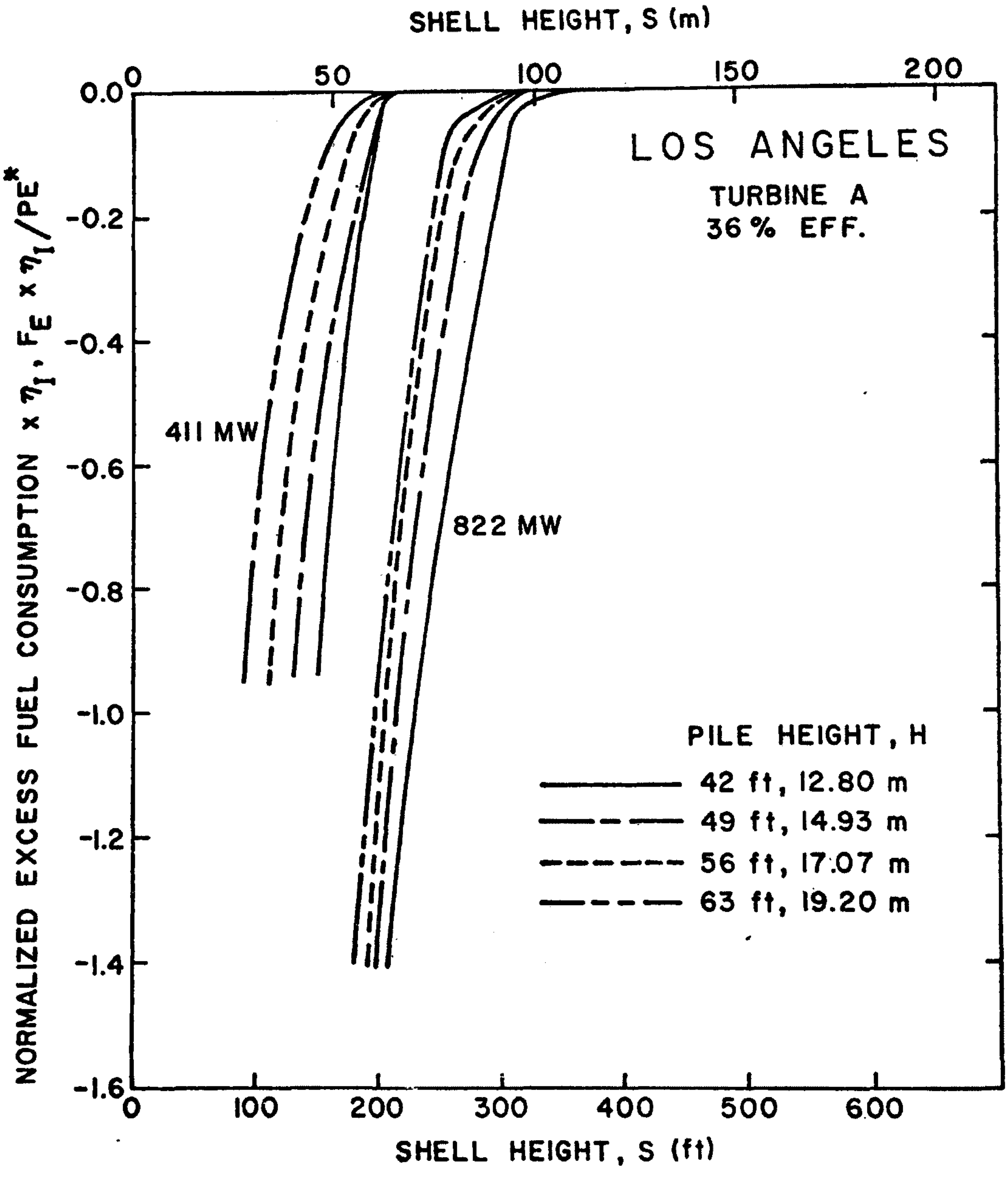

Figure 40. Variation of normalized excess fuel consumption with shell height and plant capacity, 36% turbine efficiency

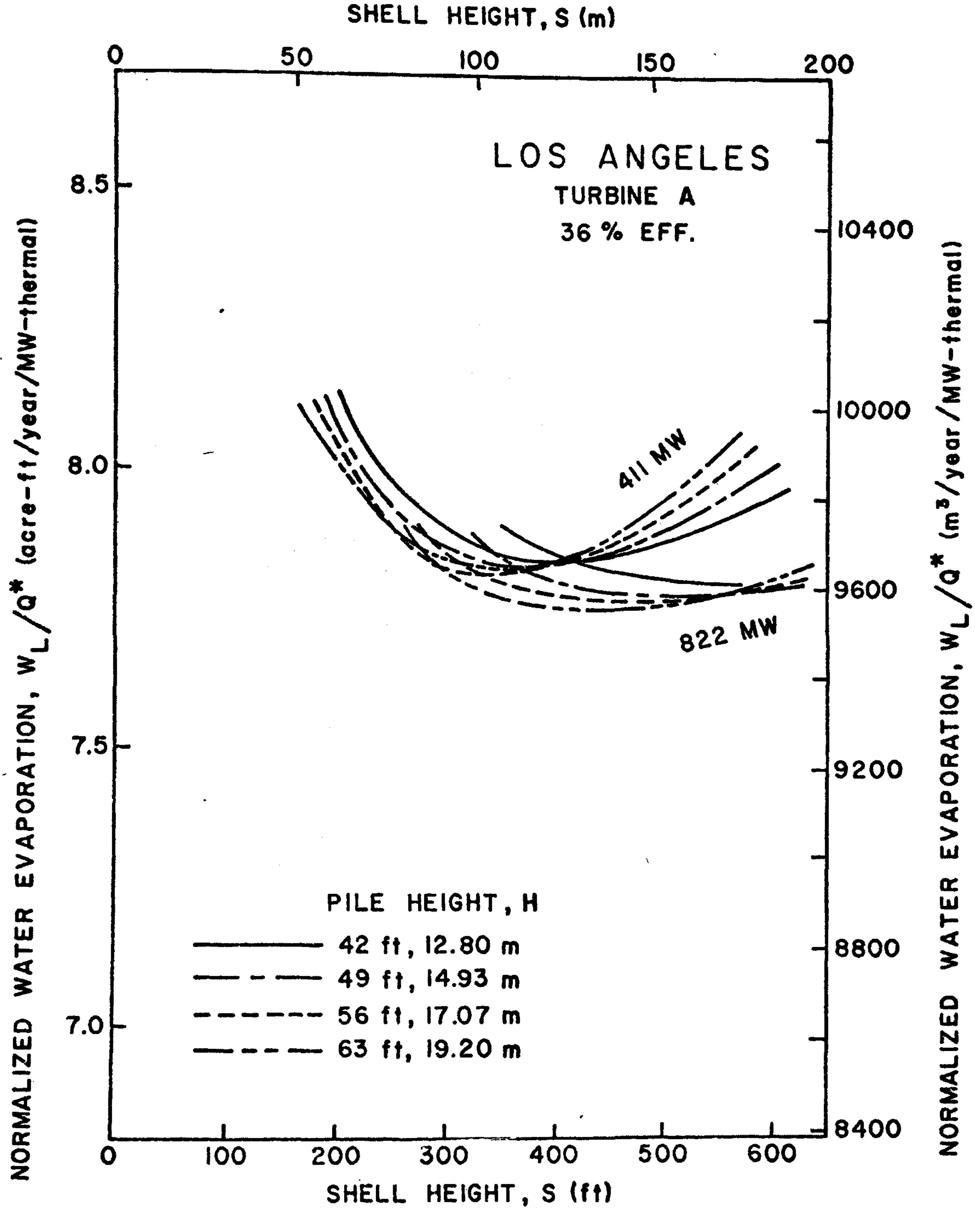

Figure 41. Variation of normalized evaporation with shell height and plant capacity, 36% turbine efficiency

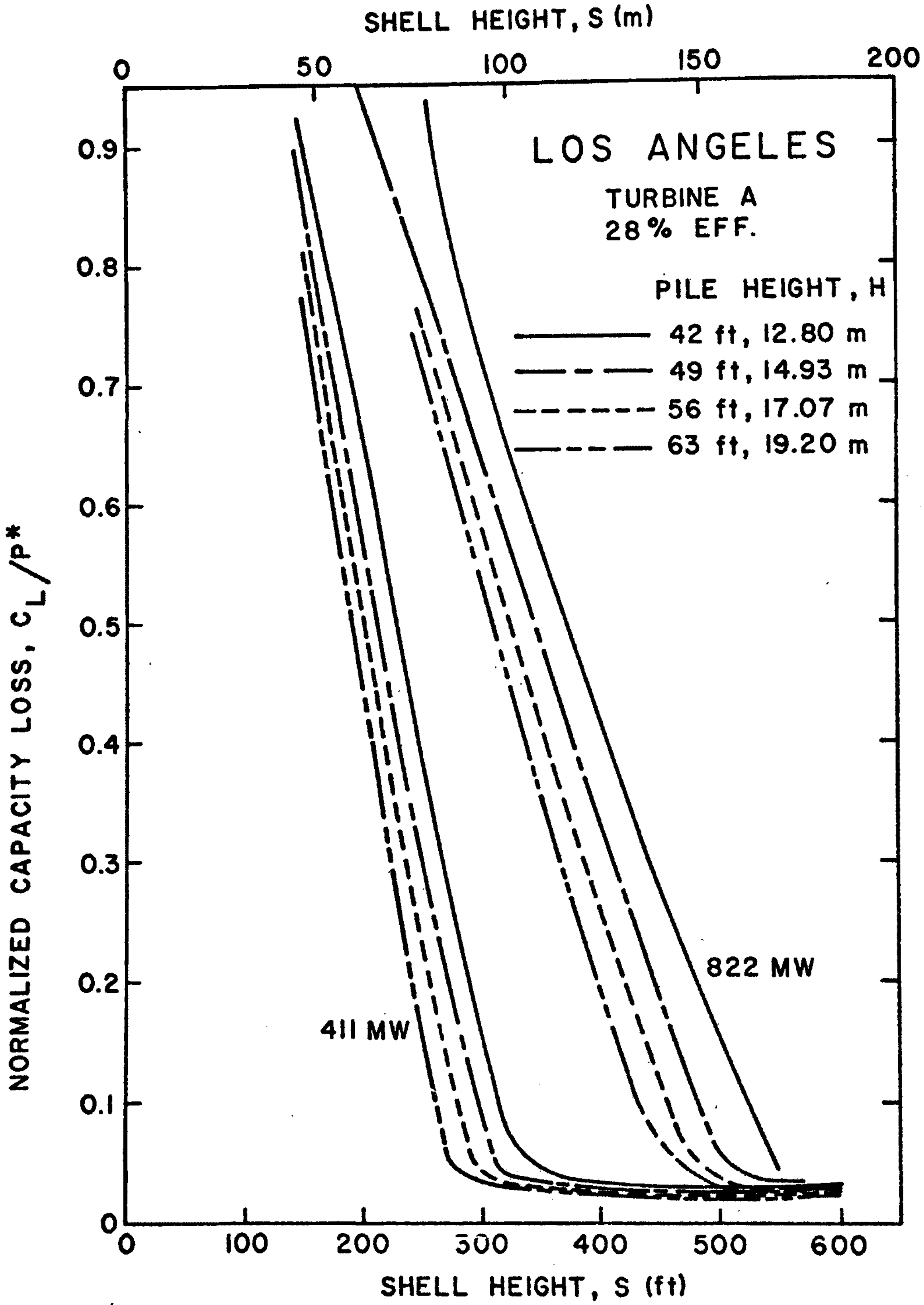

Figure 42. Variation of normalized capacity loss with shell height and plant capacity, 28% turbine efficiency

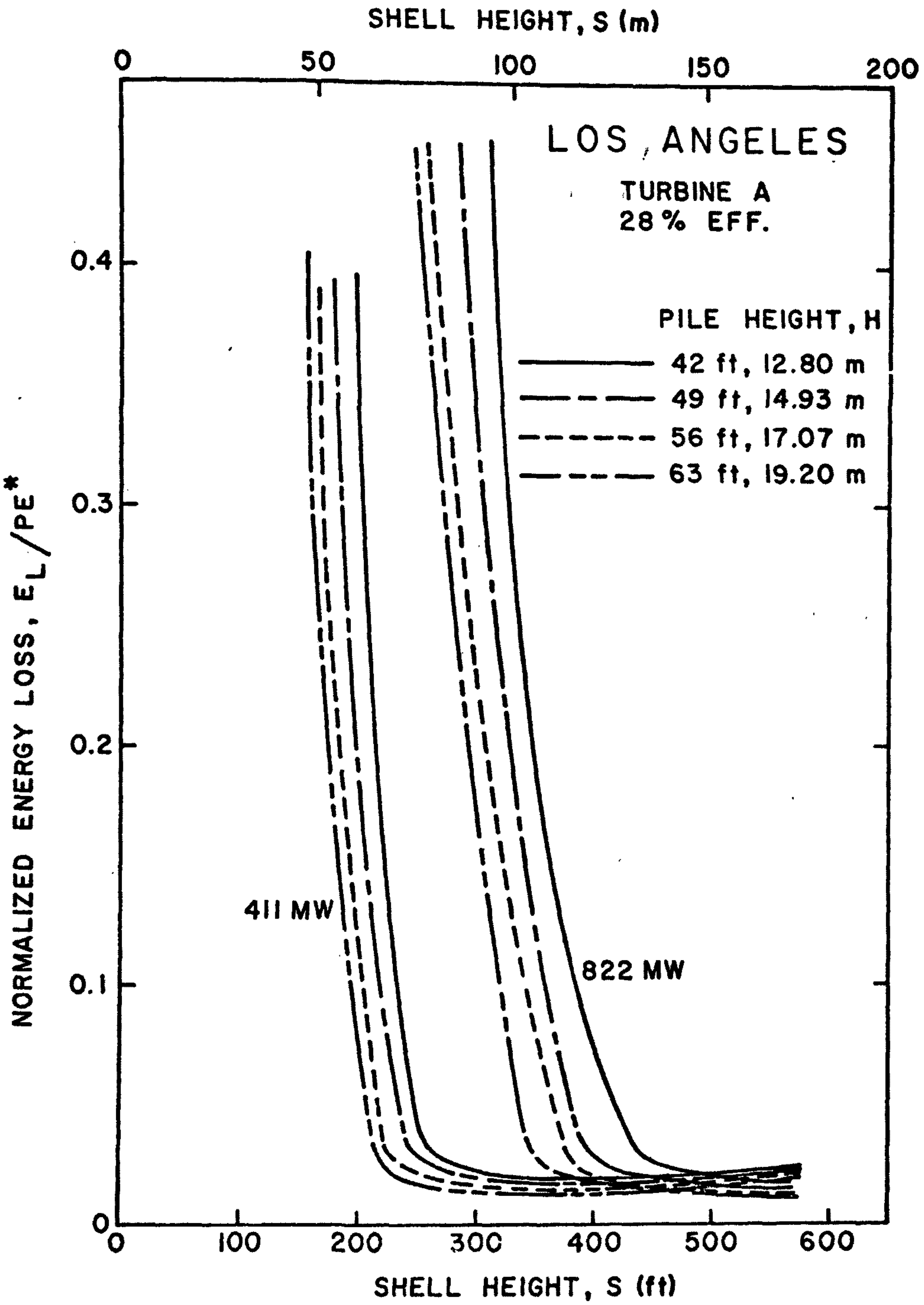

Figure 43. Variation of normalized energy loss with shell height and plant capacity, 28% turbine efficiency

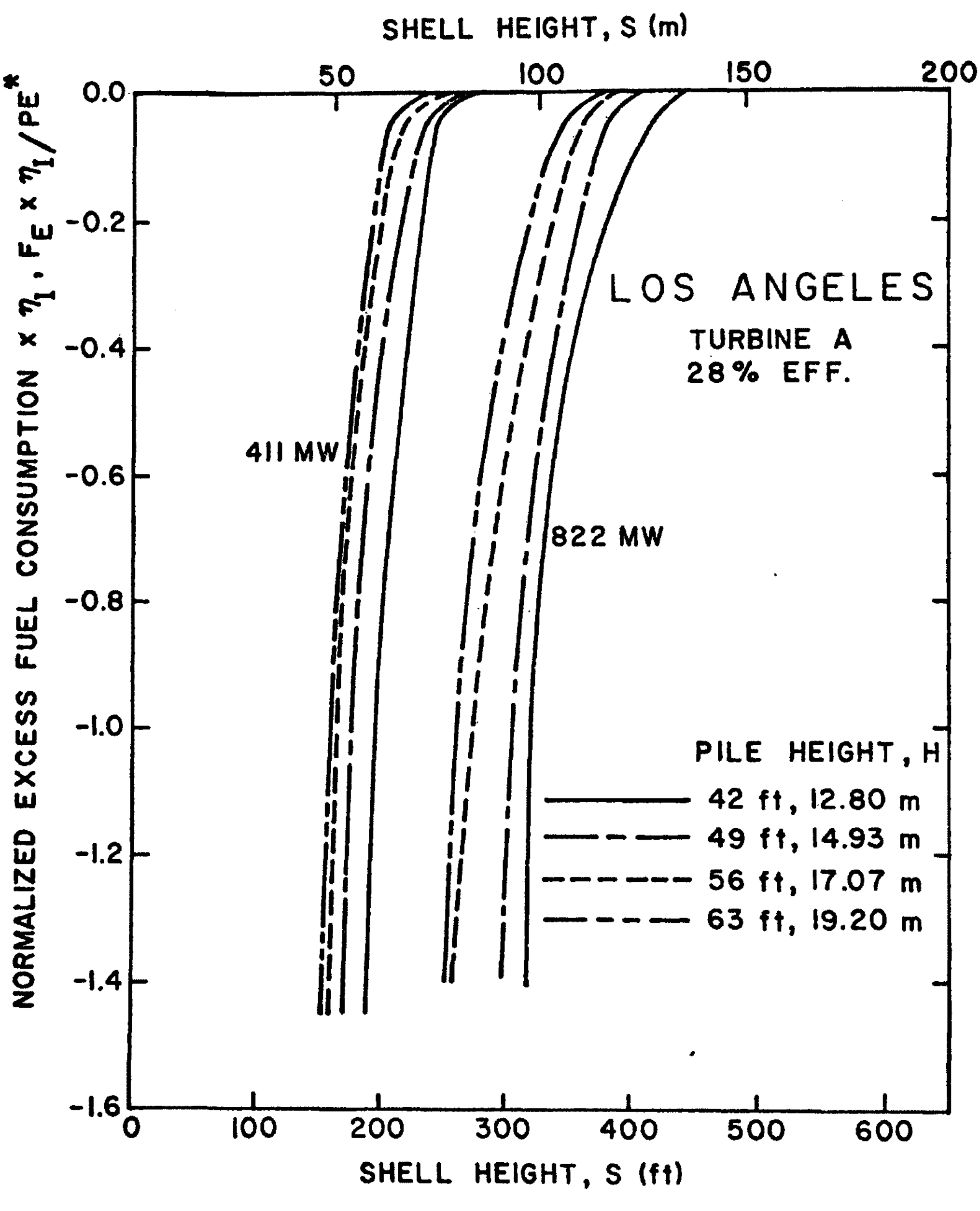

Figure 44. Variation of normalized excess fuel consumption with shell height and plant capacity, 28% turbine efficiency

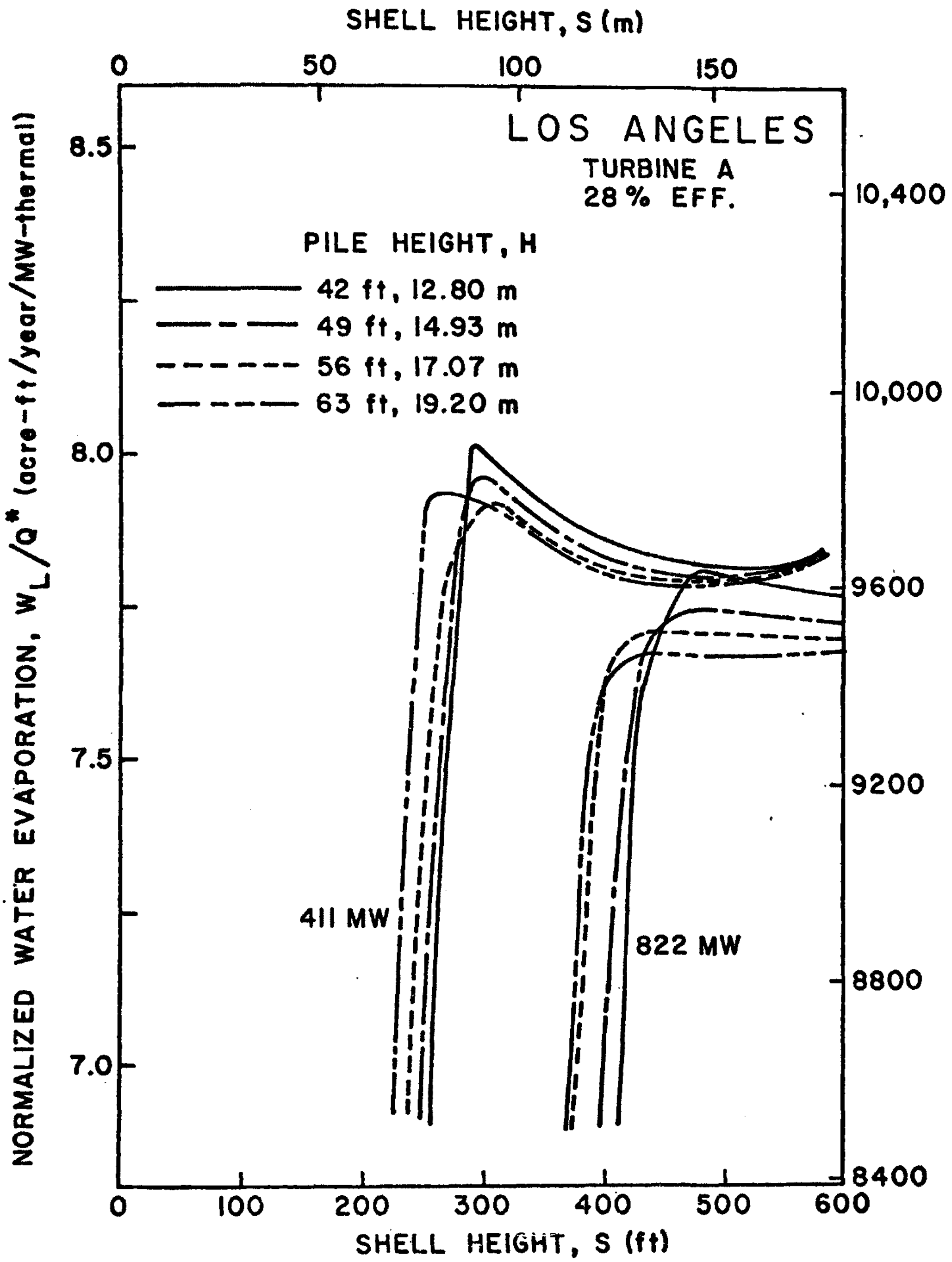

Figure 45. Variation of normalized evaporation with shell height and plant capacity, 28% turbine efficiency

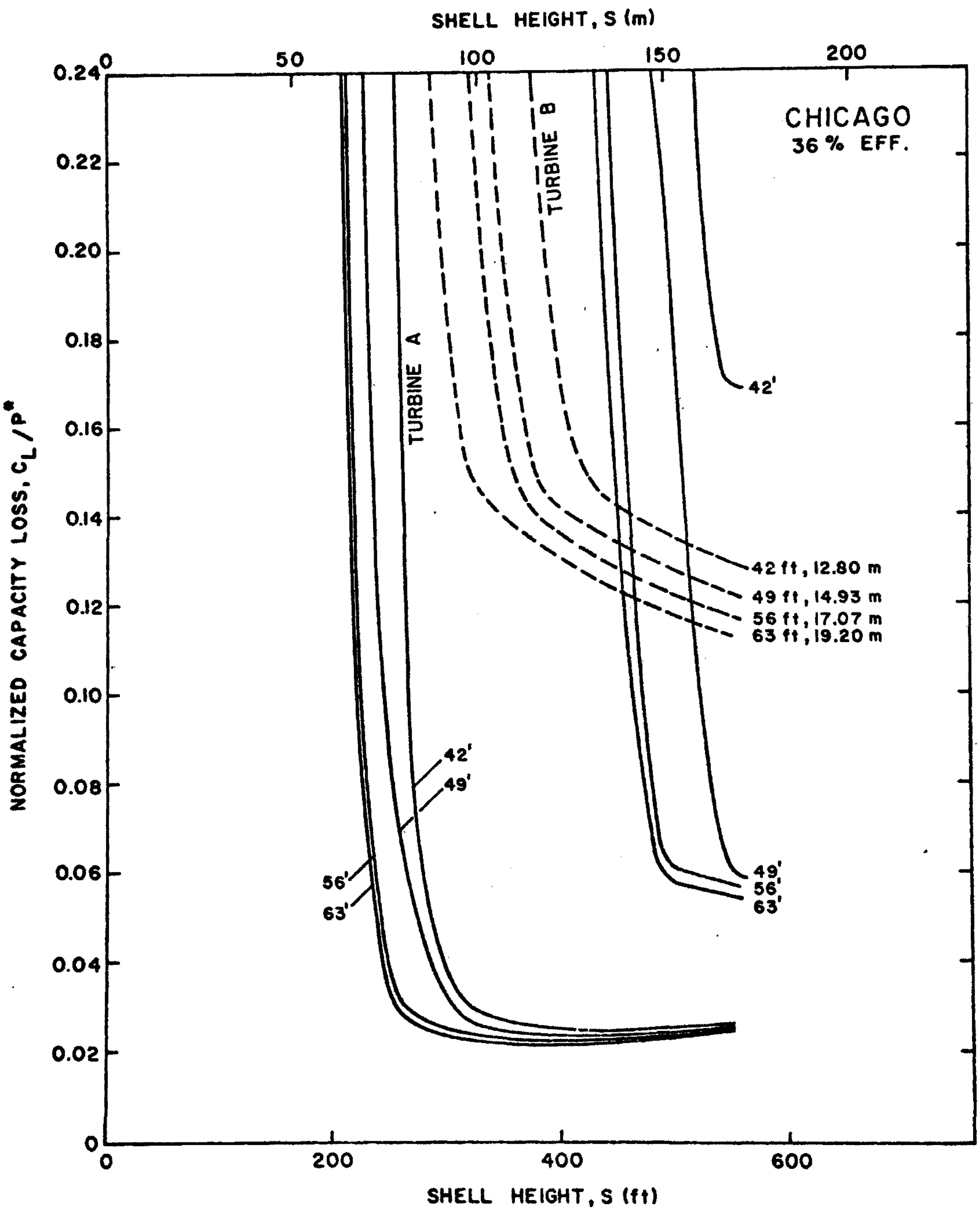

Figure 46(a). Normalized capacity loss,
36% turbine efficiency, Chicago

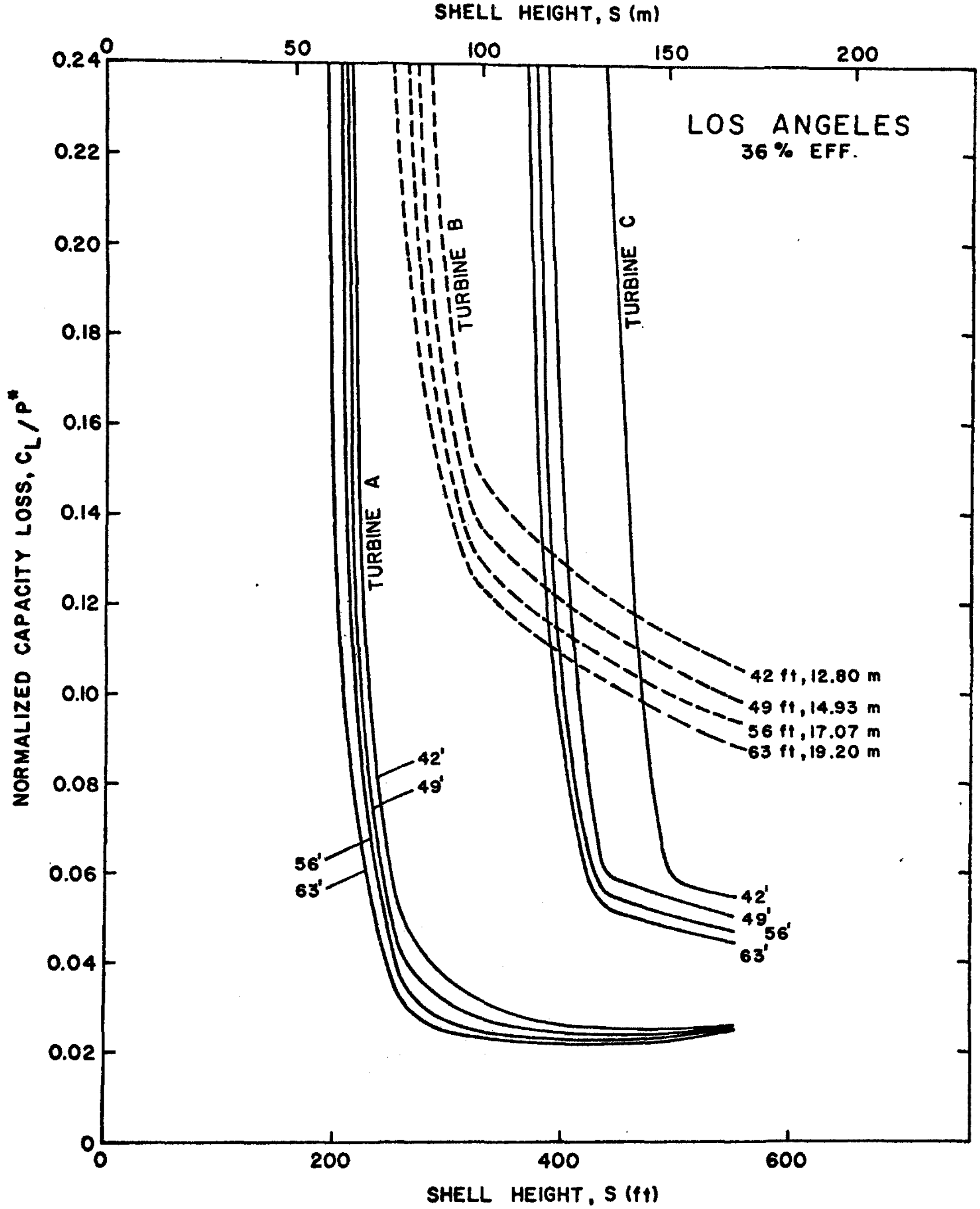

Figure 46(b). Normalized capacity loss, 36% turbine efficiency, Los Angeles

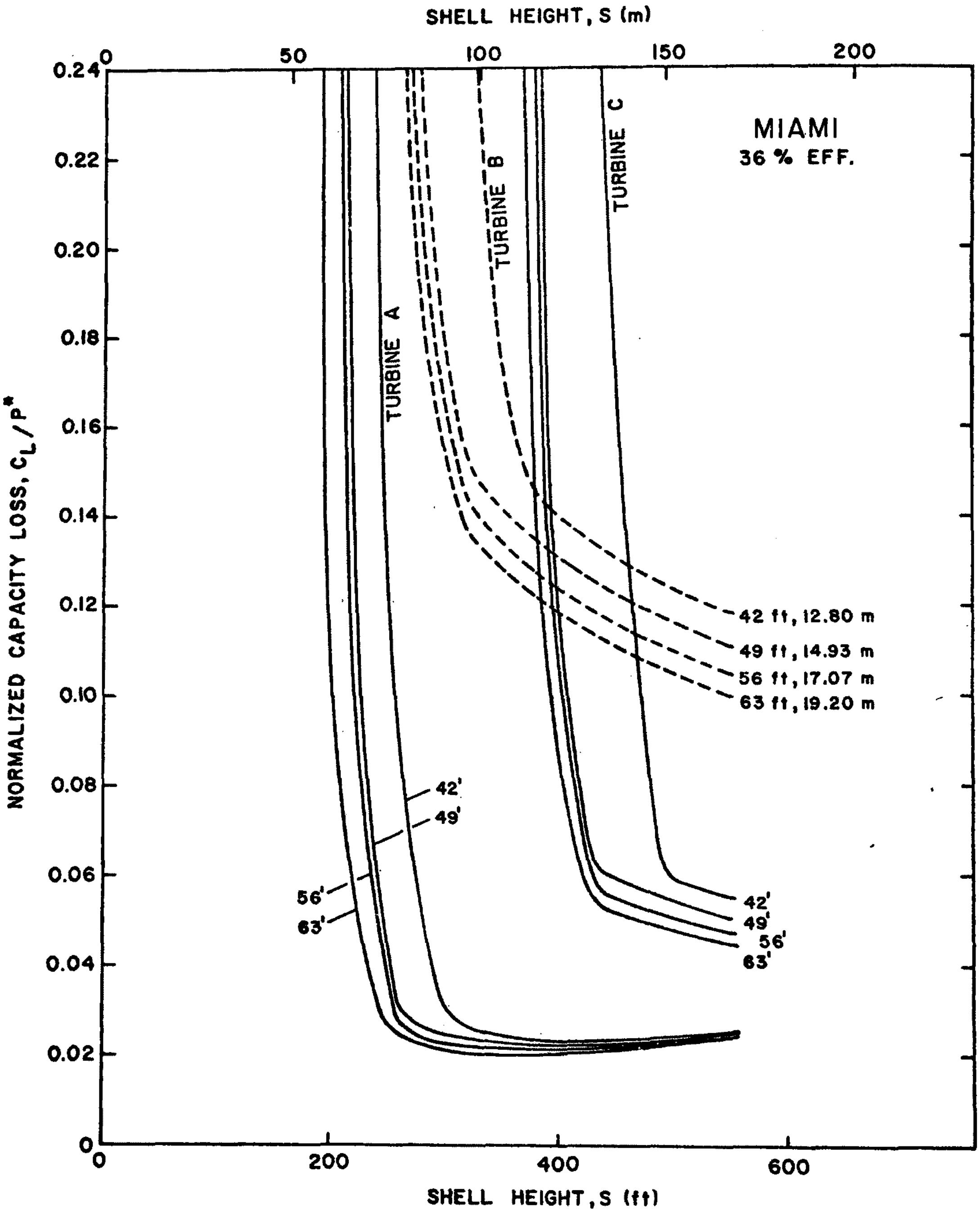

Figure 46(c). Normalized capacity loss, 36% turbine efficiency, Miami

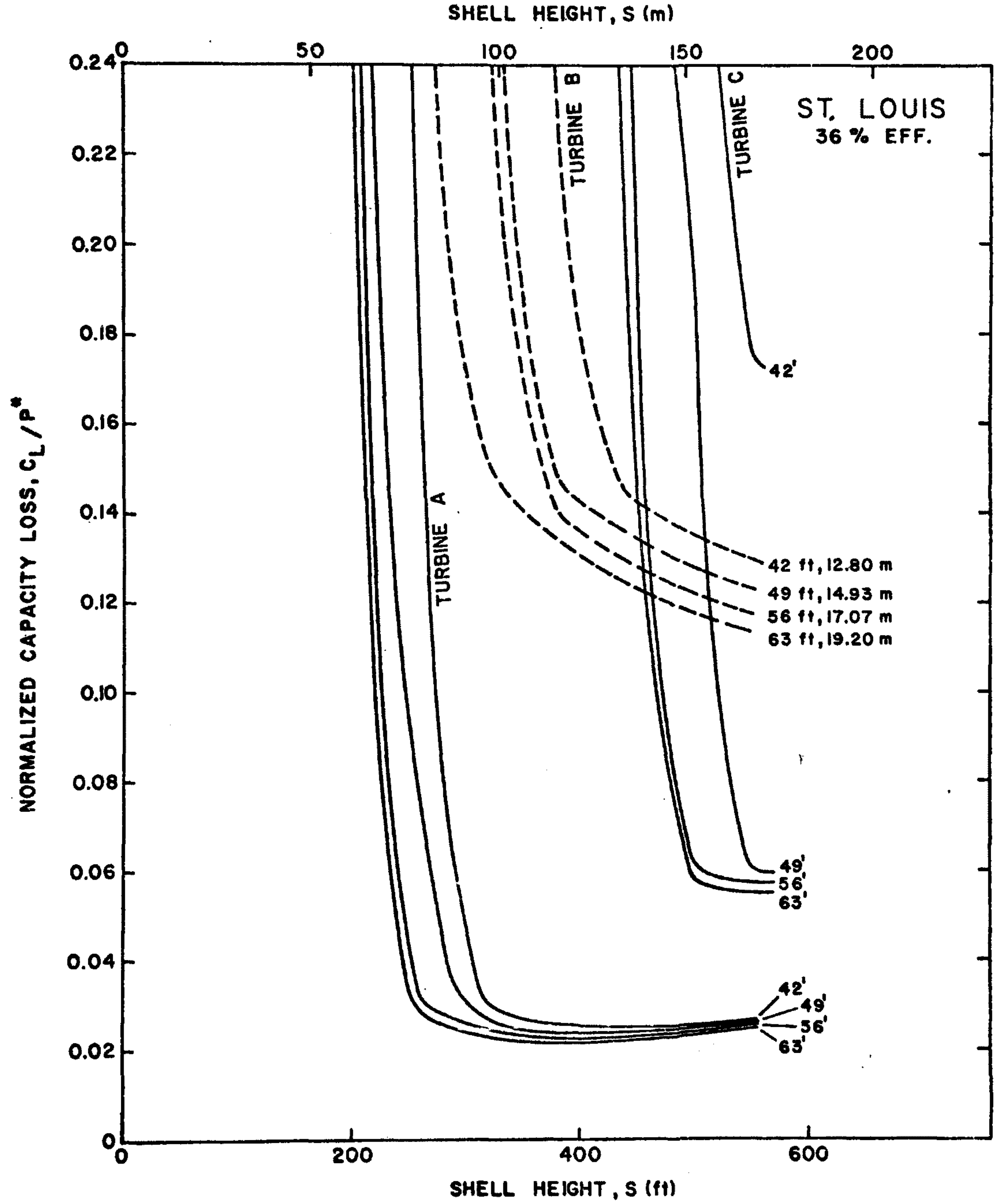

Figure 46(d). Normalized capacity loss, 36% turbine efficiency, St. Louis

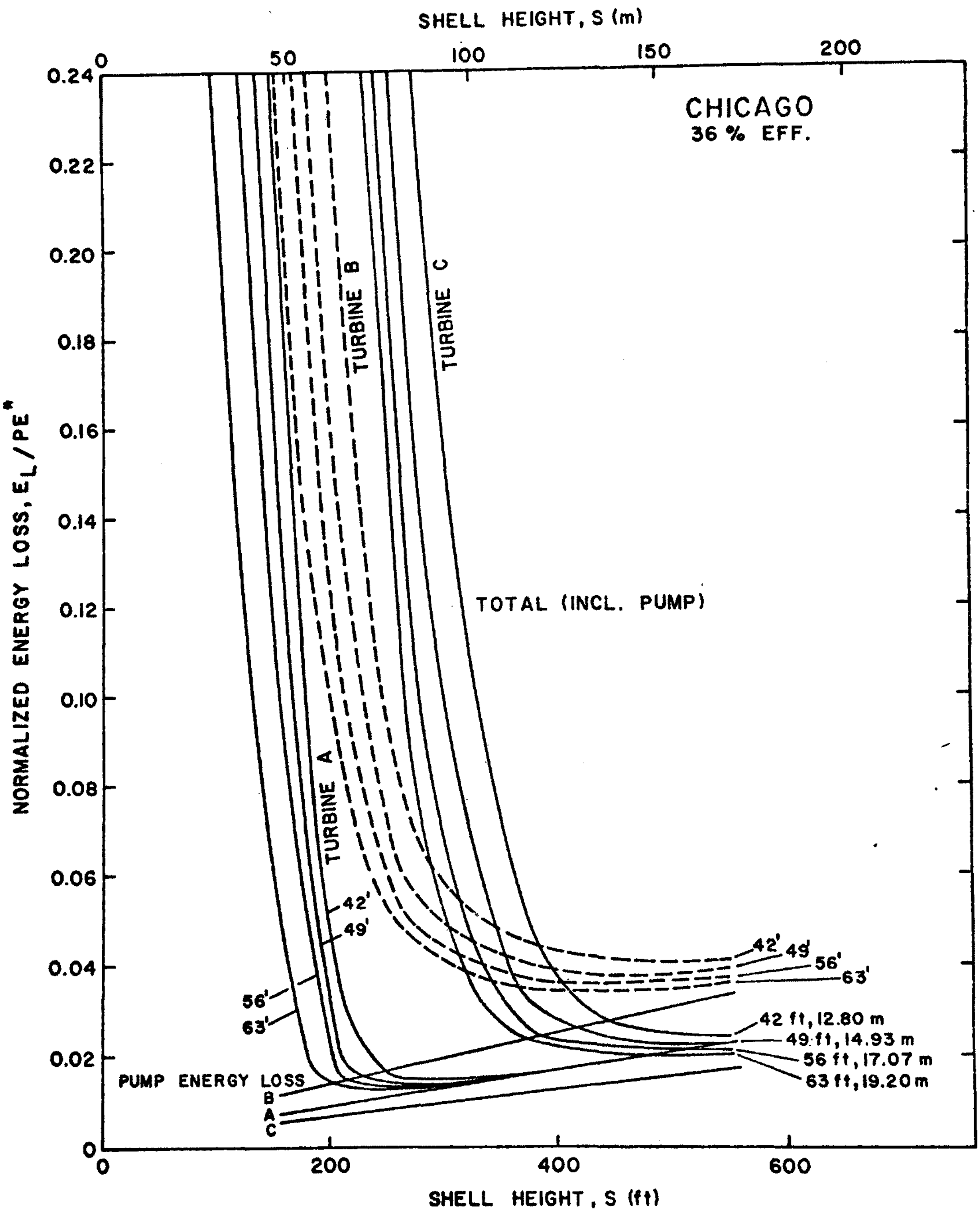

Figure 47(a). Normalized energy loss, 36% turbine efficiency, Chicago

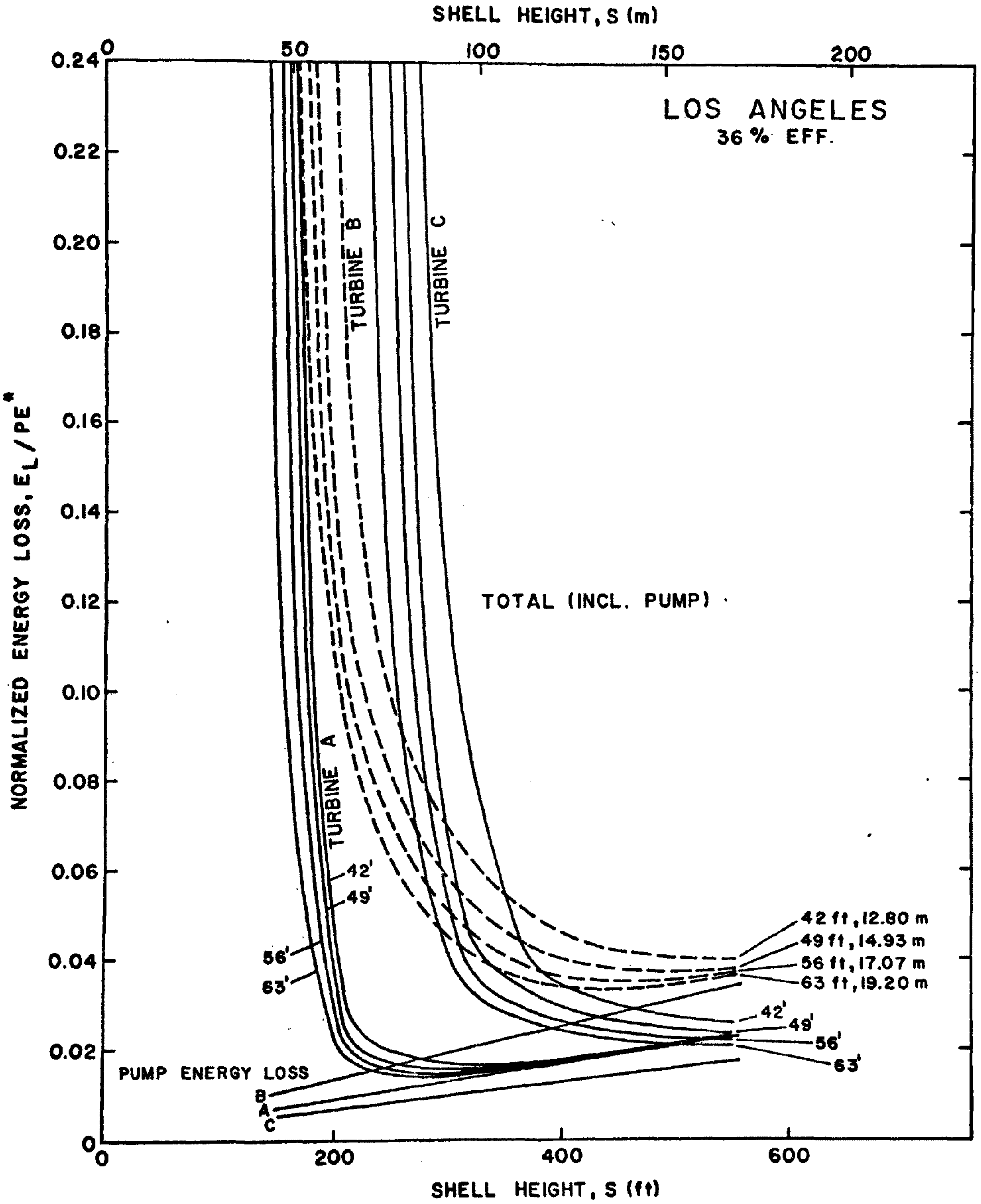

Figure 47(b). Normalized energy loss, 36% turbine efficiency, Los Angeles

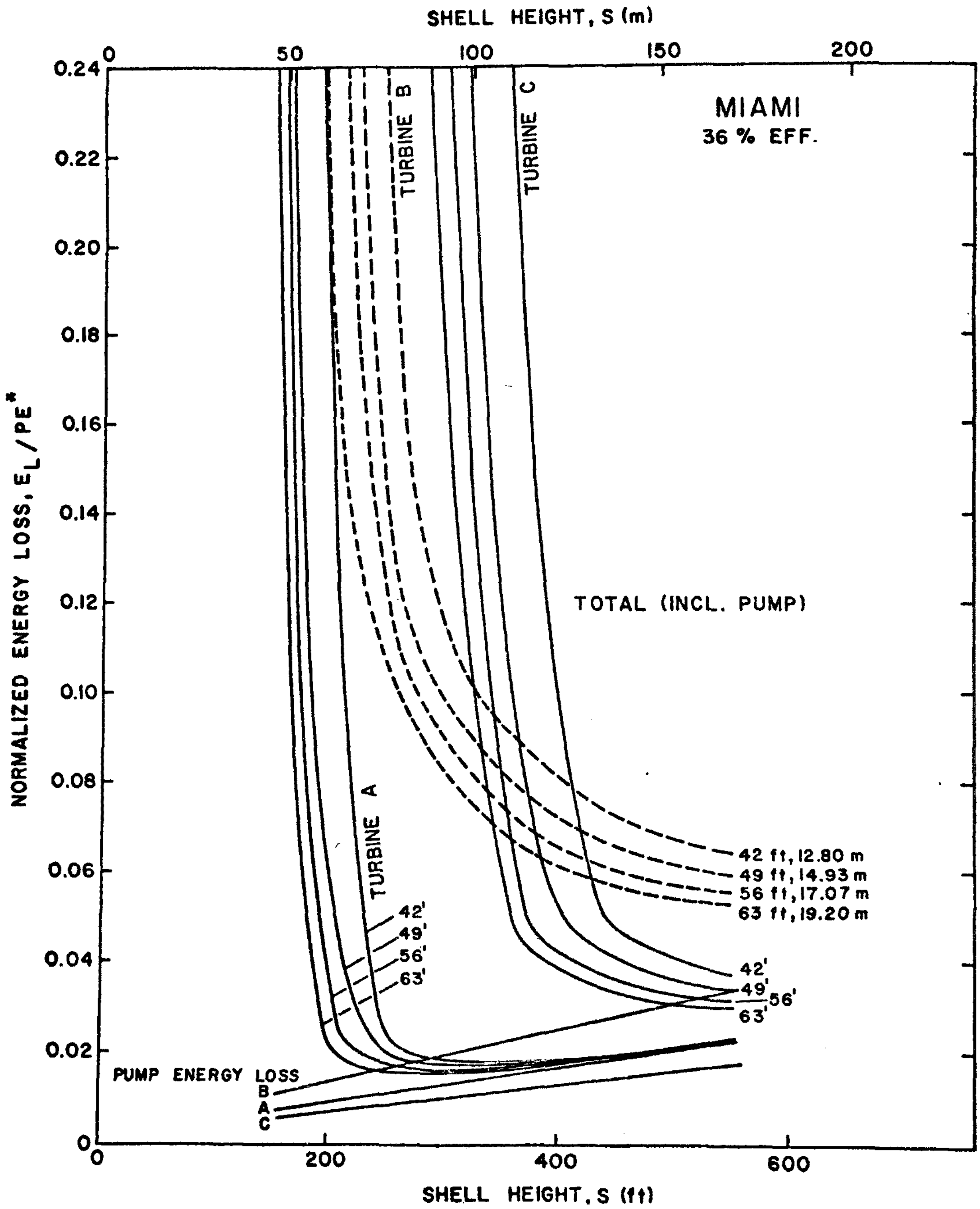

Figure 47(c). Normalized energy loss, 36% turbine efficiency, Miami

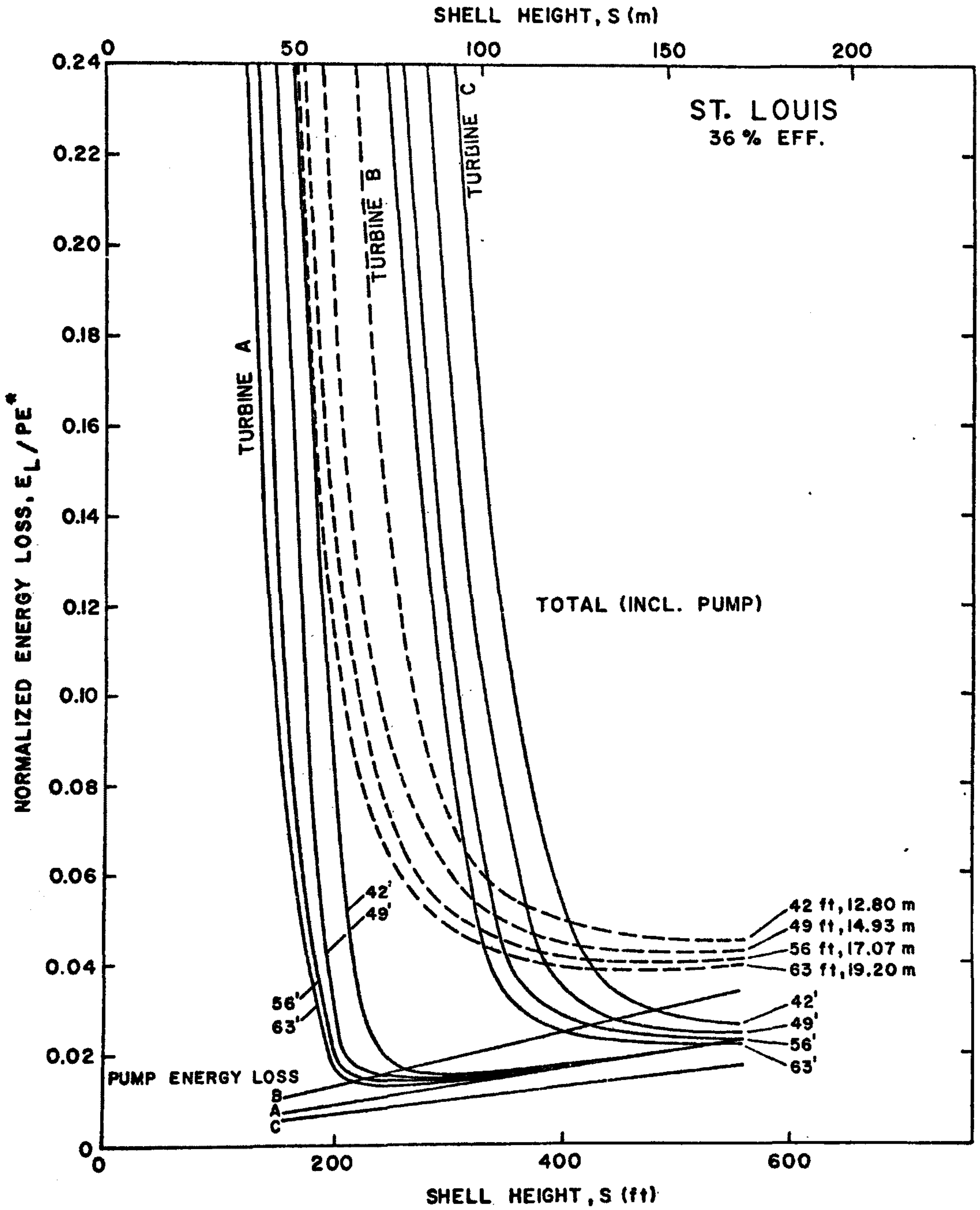

Figure 47(d). Normalized energy loss, 36% turbine efficiency, St. Louis

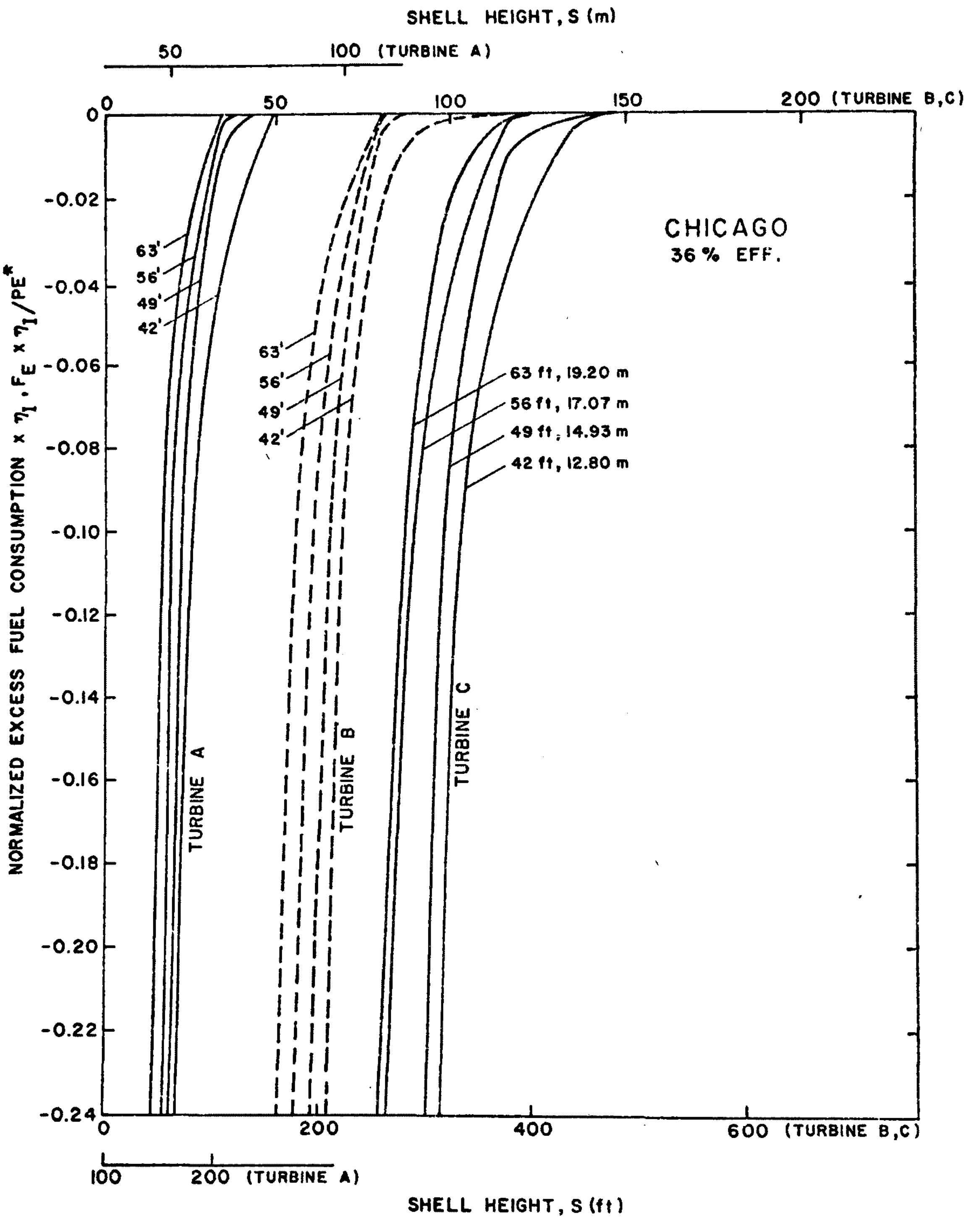

Figure 48(a). Normalized excess fuel consumption 36% turbine efficiency, Chicago

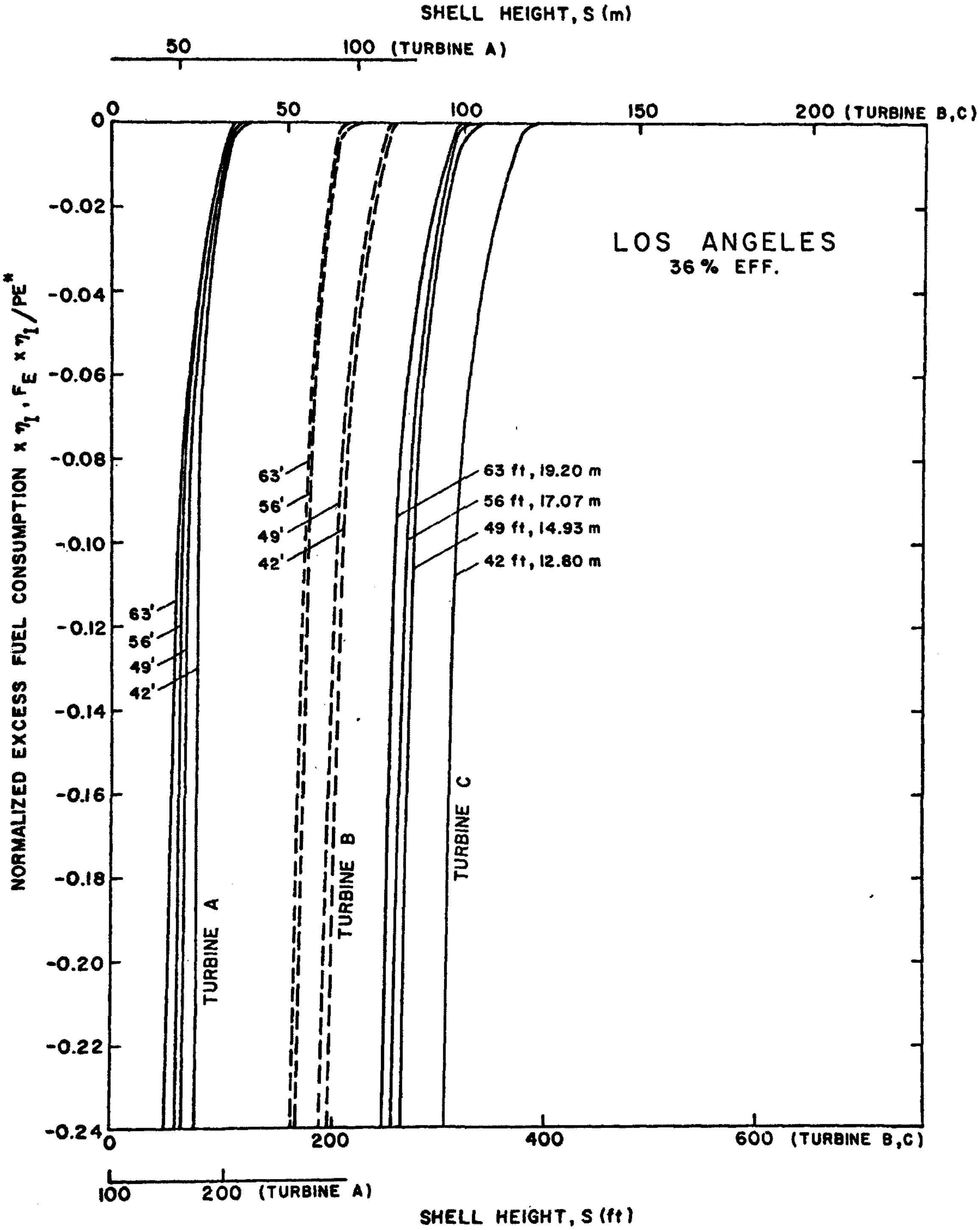

Figure 48(b). Normalized excess fuel consumption 36% turbine efficiency, Los Angeles

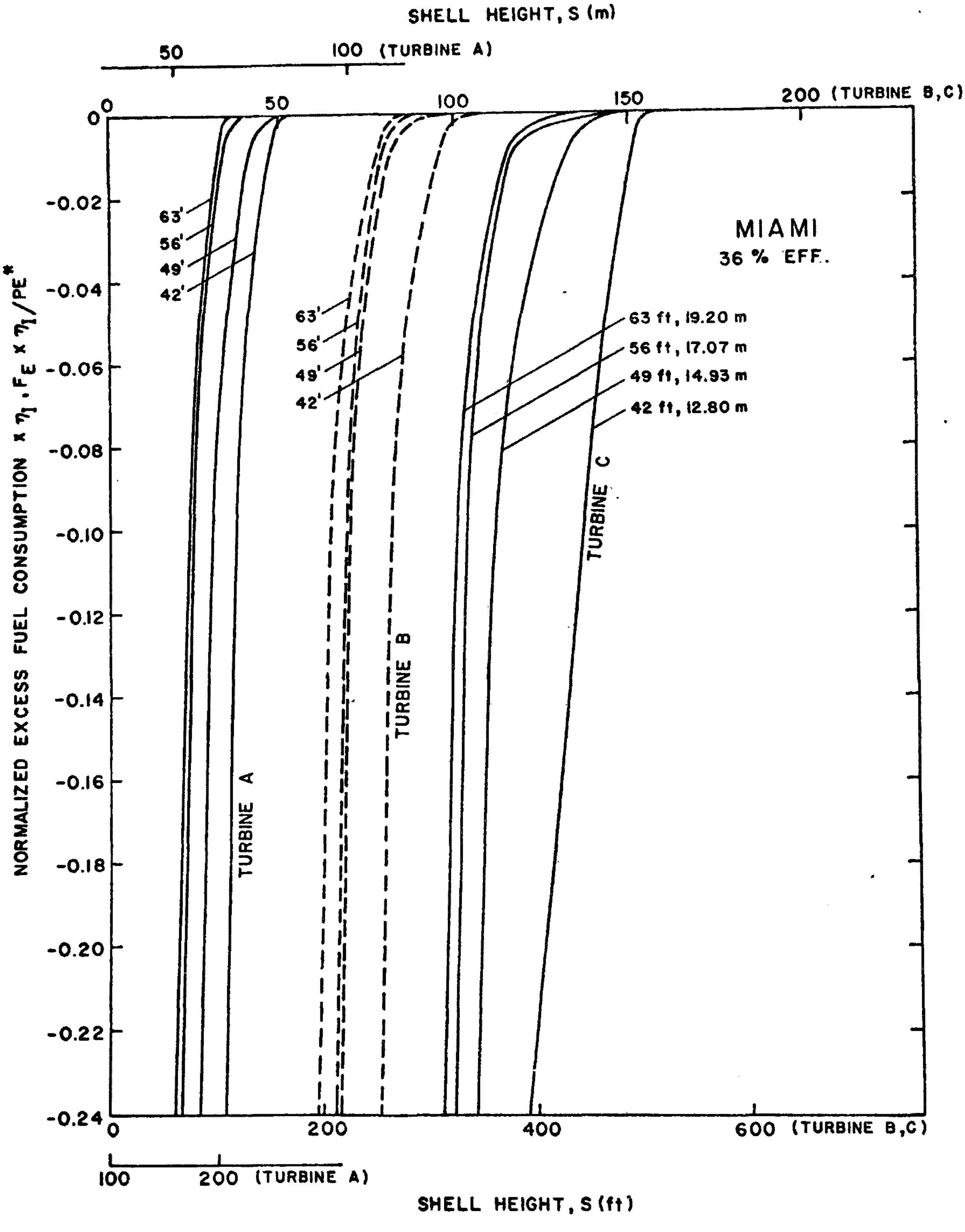

Figure 48(c). Normalized excess fuel consumption, 36% turbine efficiency, Miami

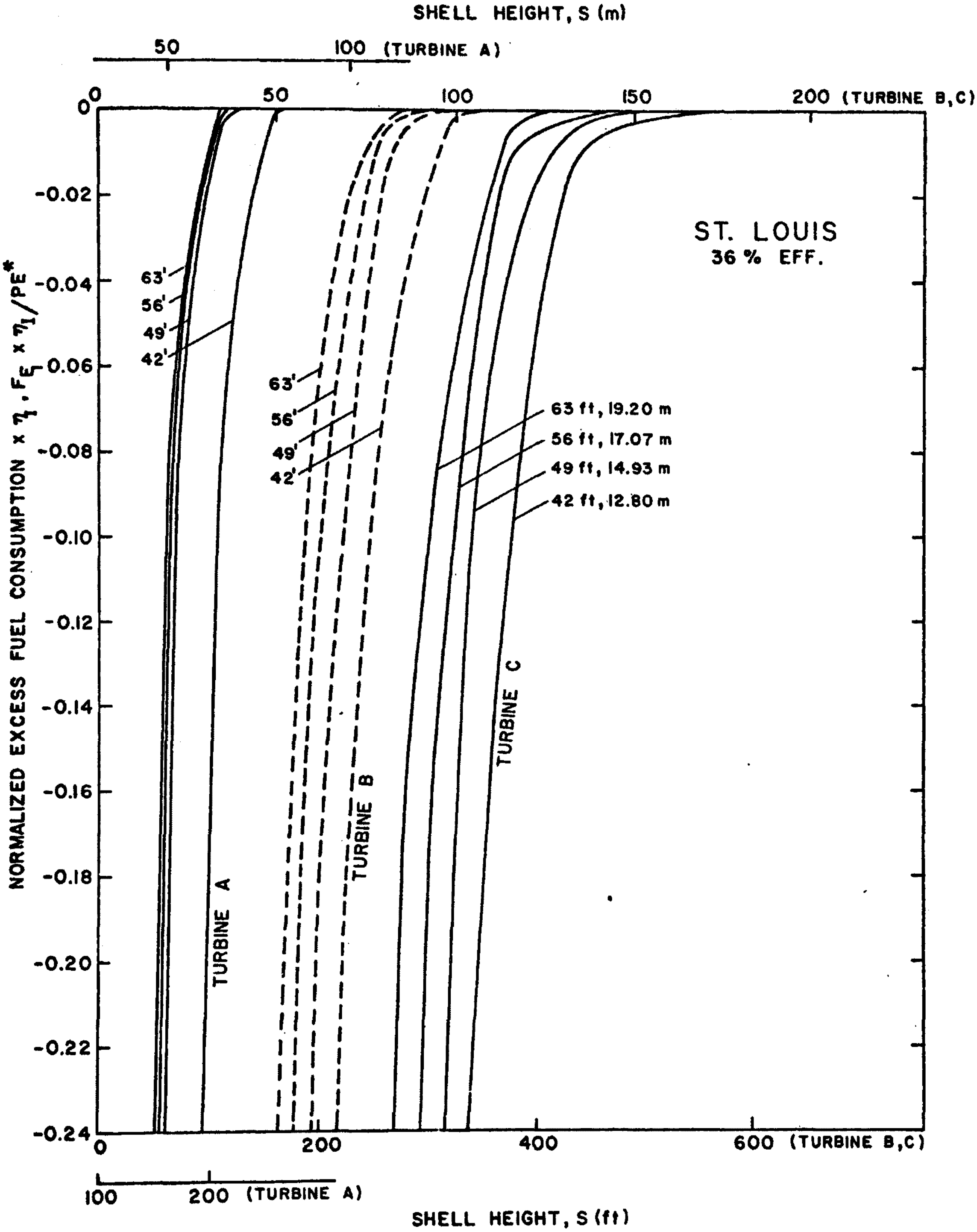

Figure 48(d). Normalized excess fuel consumption, 36% turbine efficiency, St. Louis

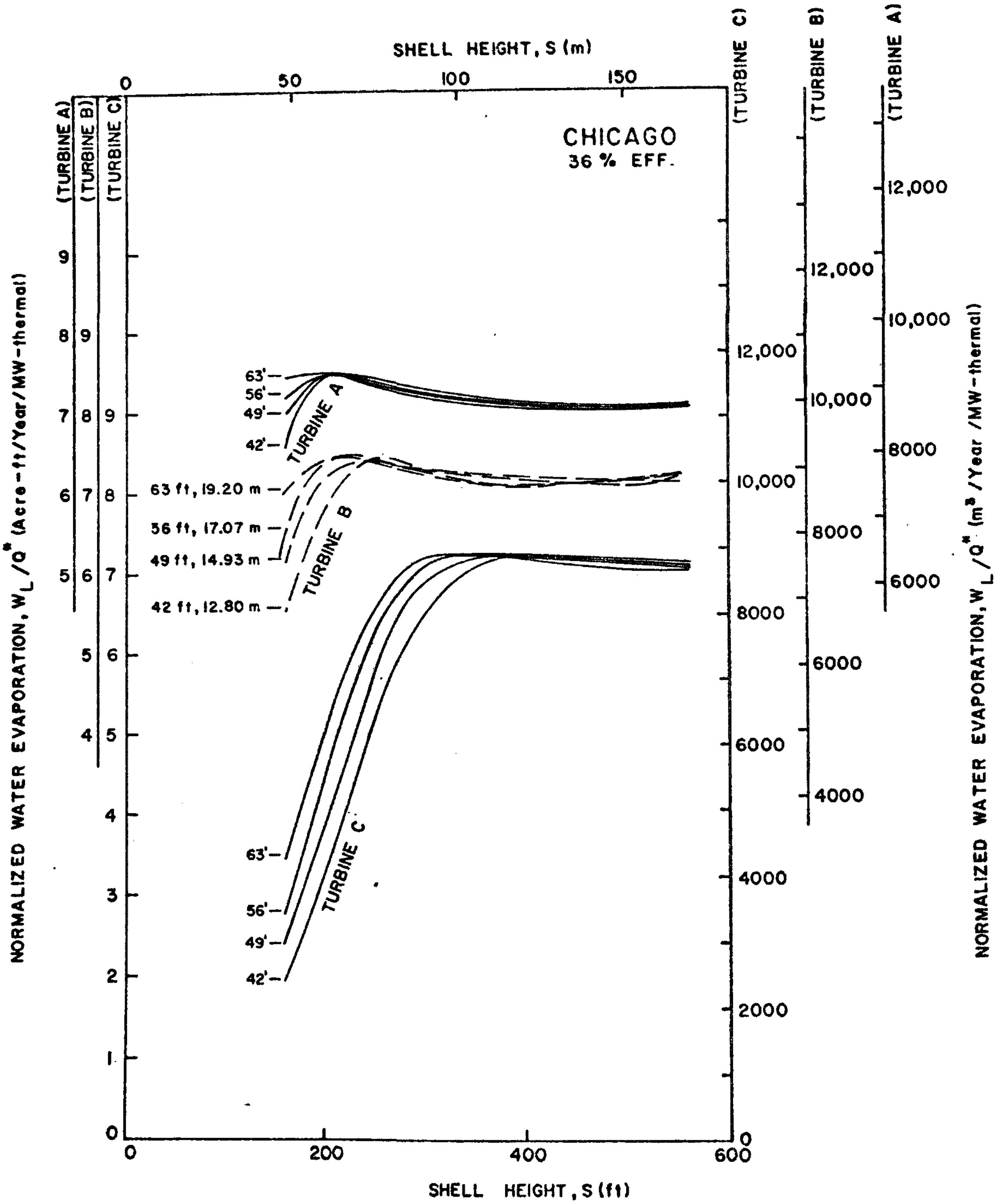

Figure 49(a). Normalized evaporation, 36% turbine efficiency, Chicago

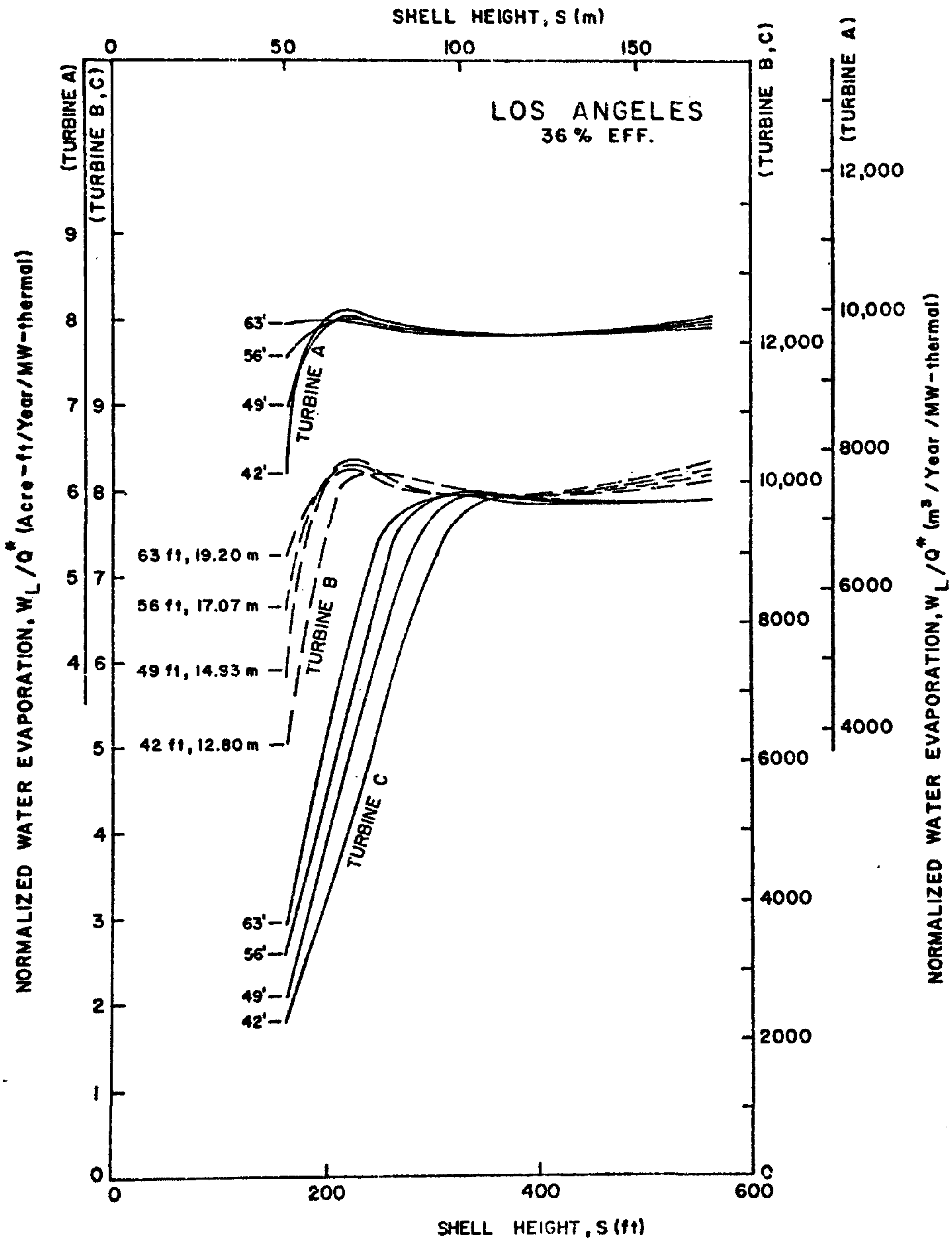

Figure 49(b). Normalized evaporation, 36% turbine efficiency, Los Angeles

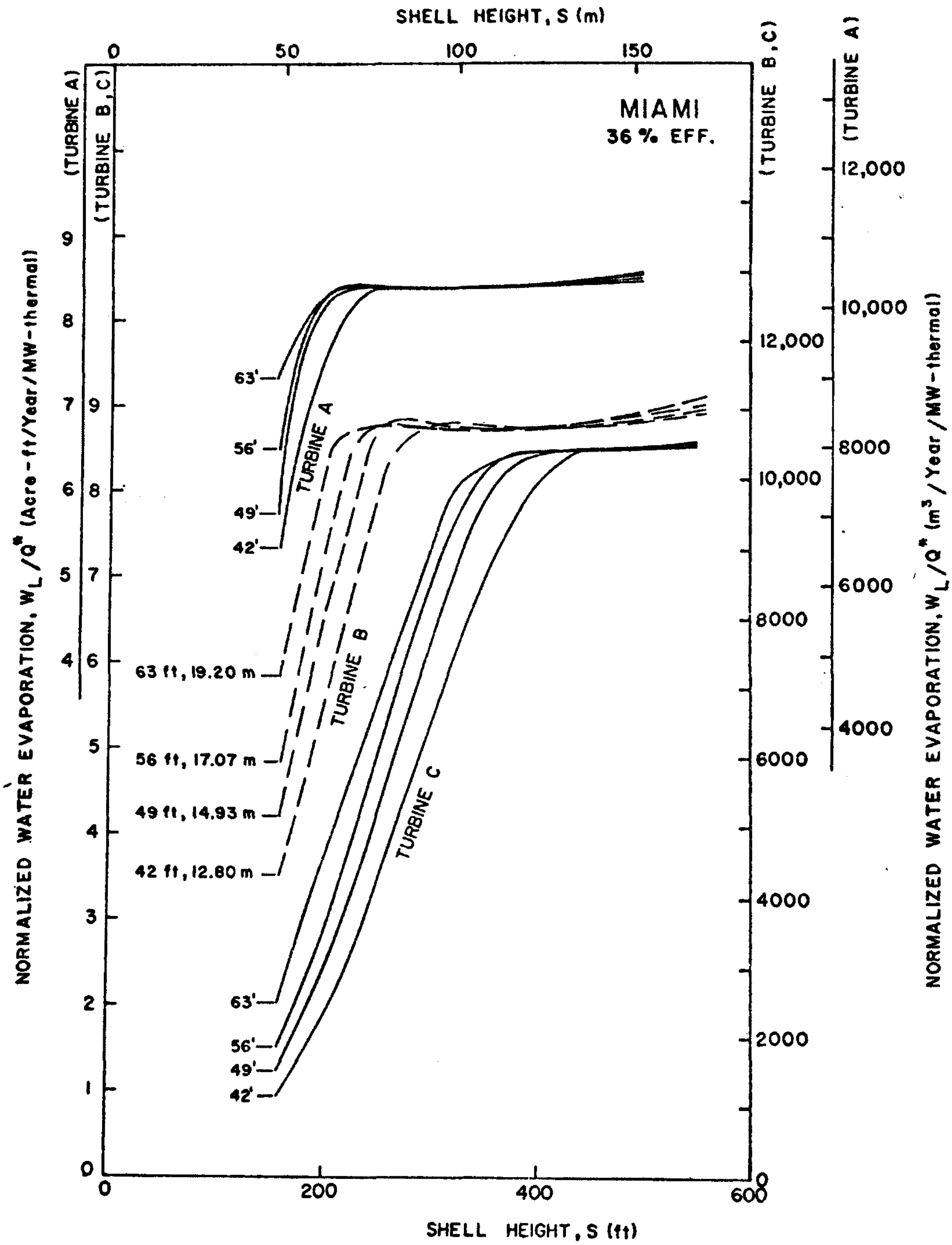

Figure 49(c). Normalized evaporation, 36% turbine efficiency, Miami

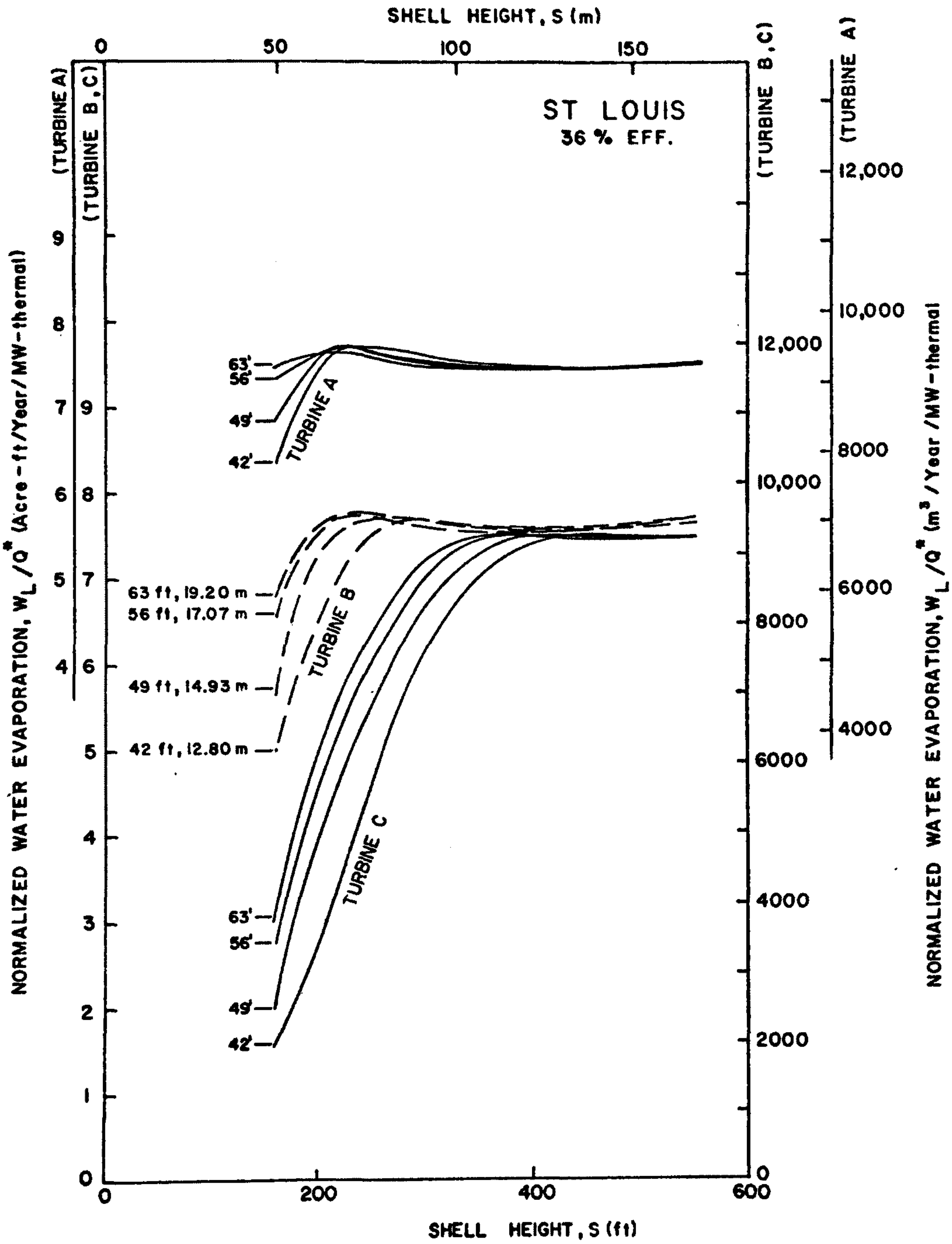

Figure 49(d). Normalized evaporation, 36% turbine efficiency, St. Louis

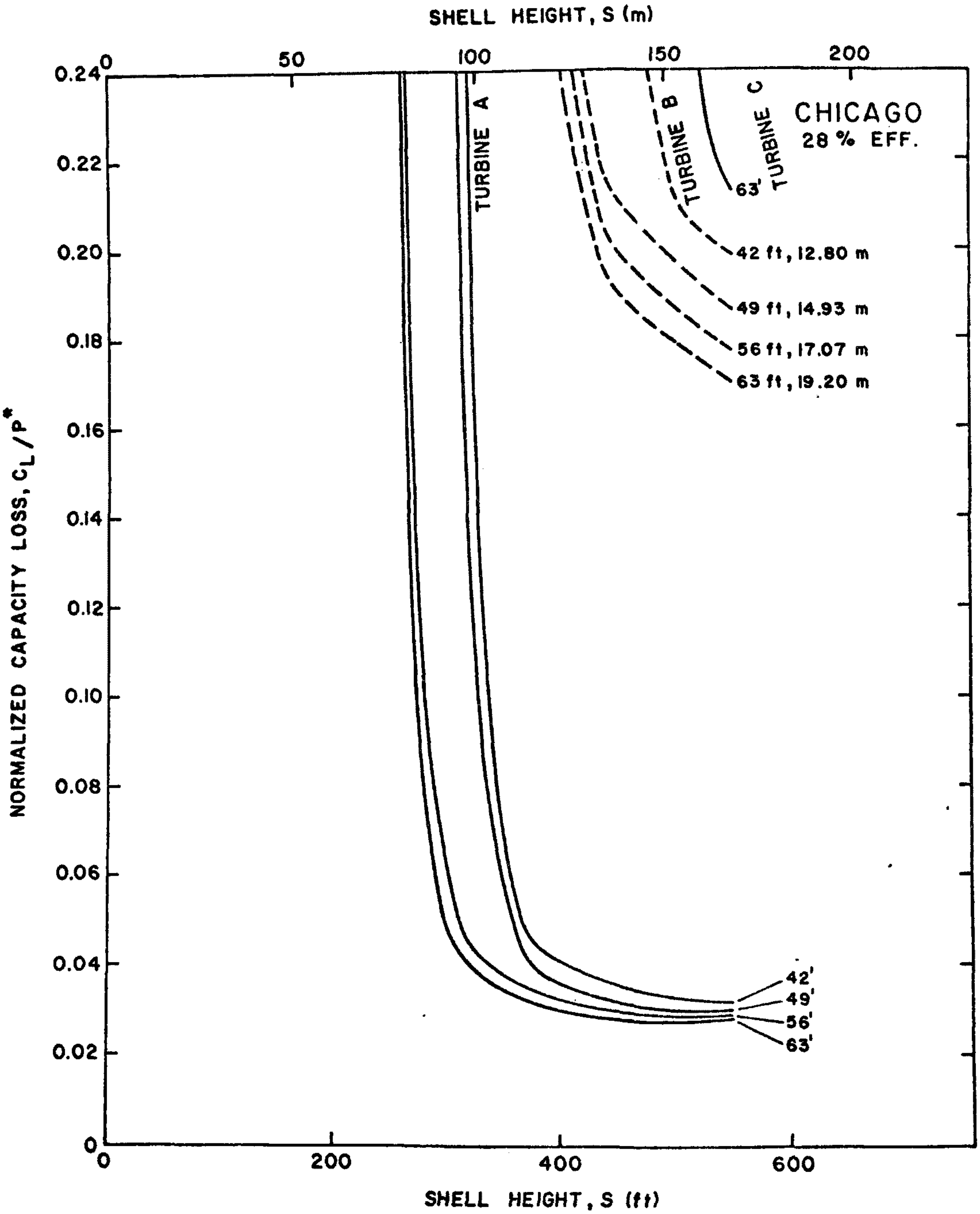

Figure 50(a). Normalized capacity loss, 28% turbine efficiency, Chicago

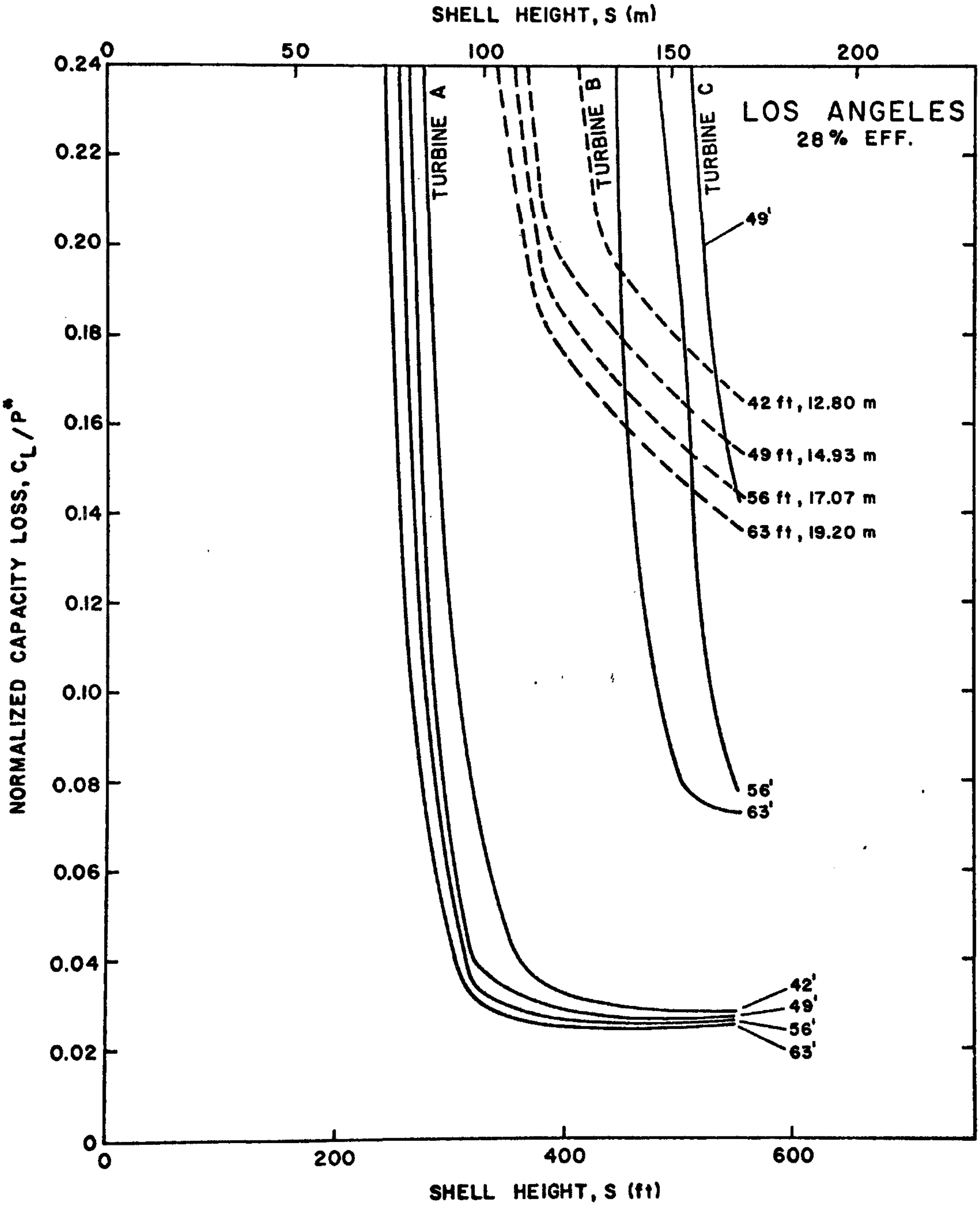

Figure 50(b). Normalized capacity loss, 28% turbine efficiency, Los Angeles

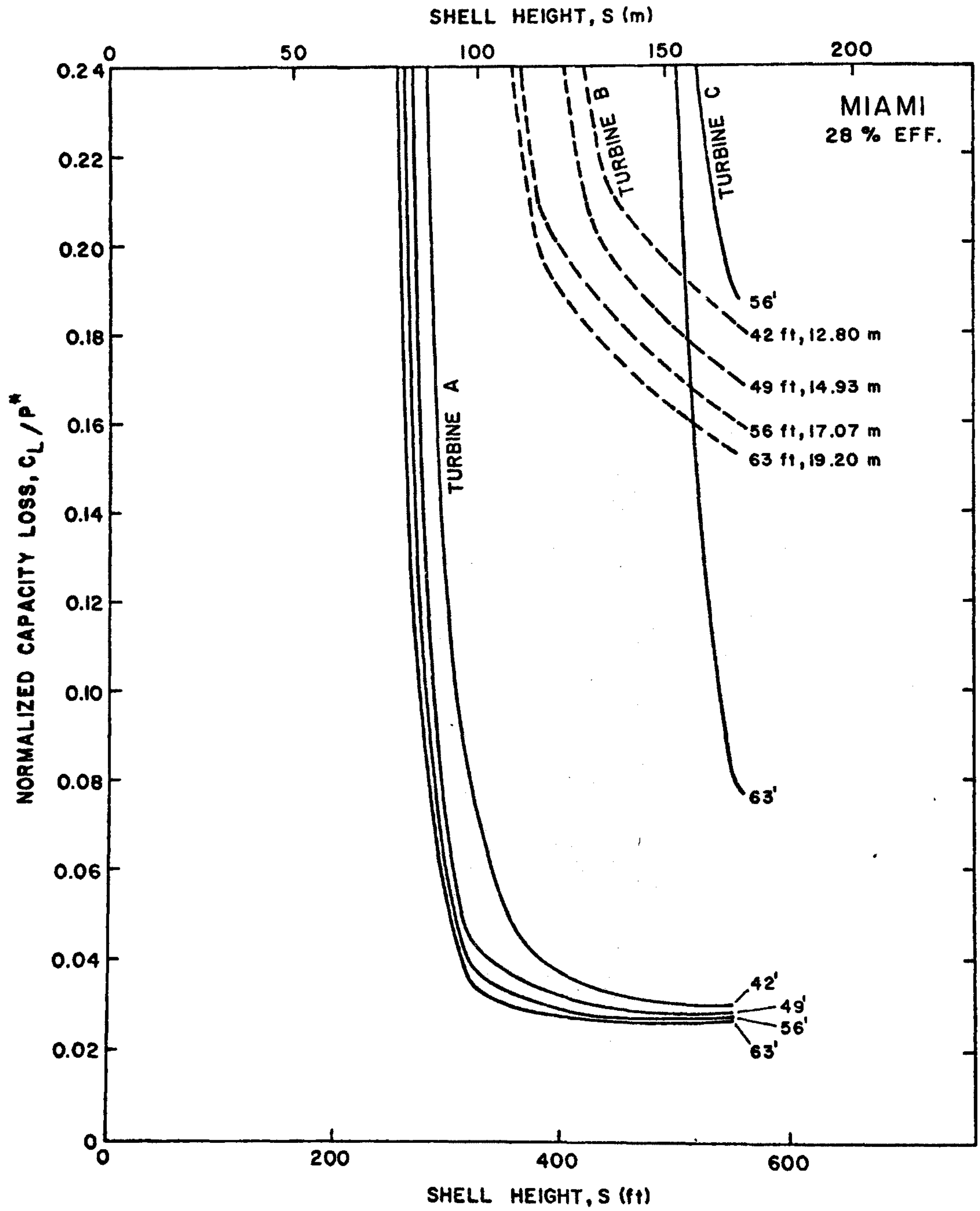

Figure 50(c). Normalized capacity loss, 28% turbine efficiency, Miami

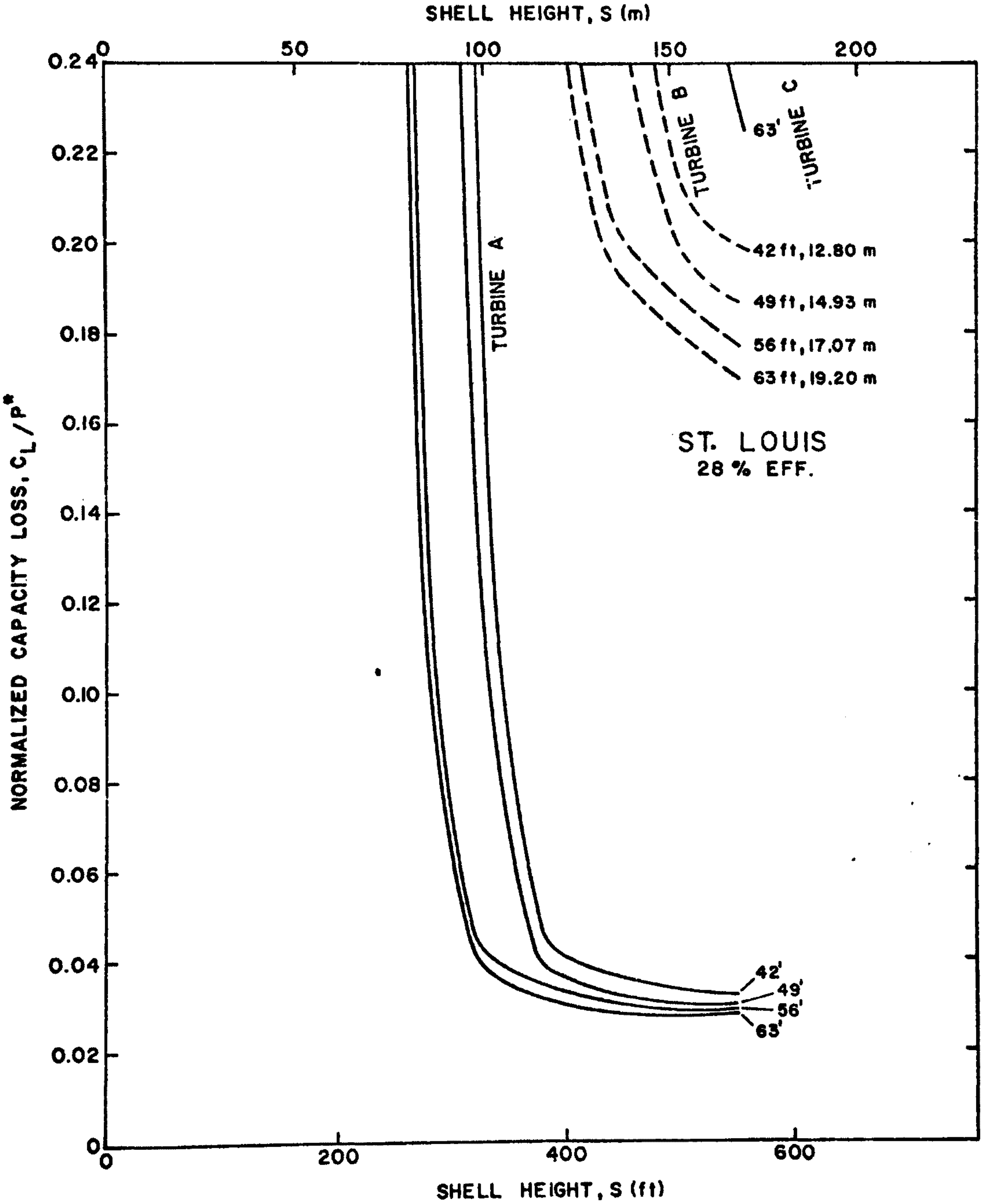

Figure 50(d). Normalized capacity loss, 28% turbine efficiency, St. Louis

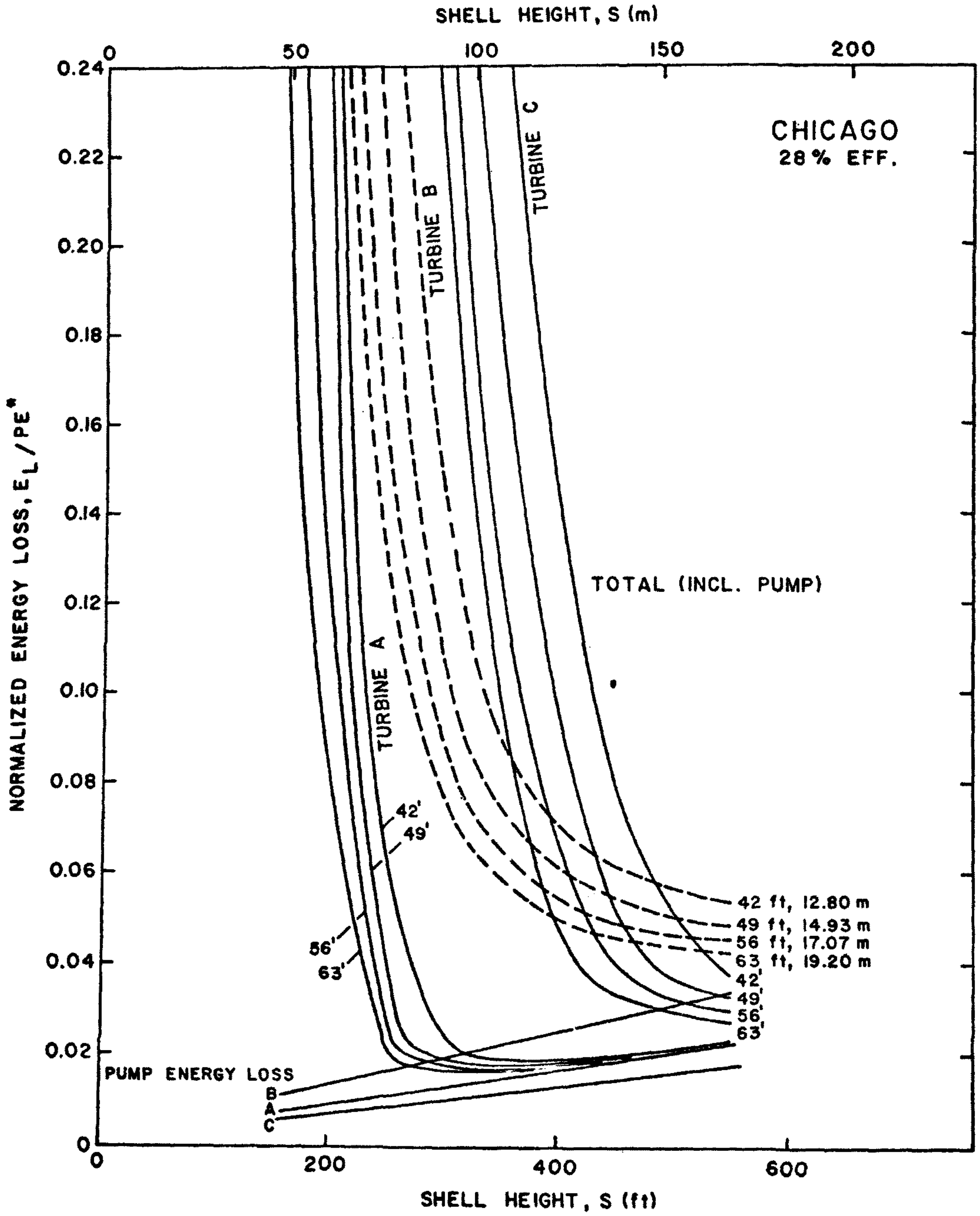

Figure 51(a). Normalized energy loss, 28% turbine efficiency, Chicago

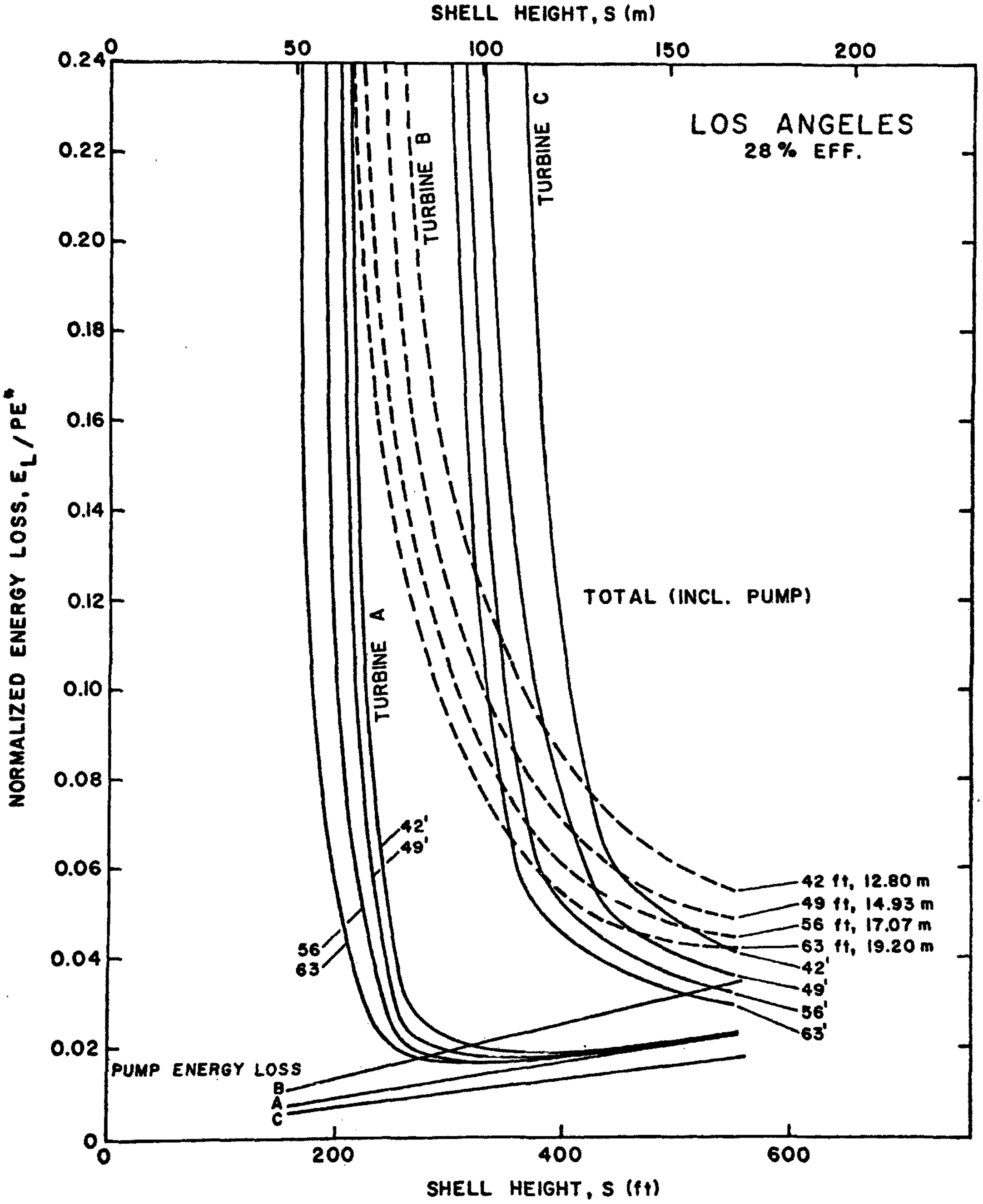

Figure 51(b). Normalized energy loss, 28% turbine efficiency, Los Angeles

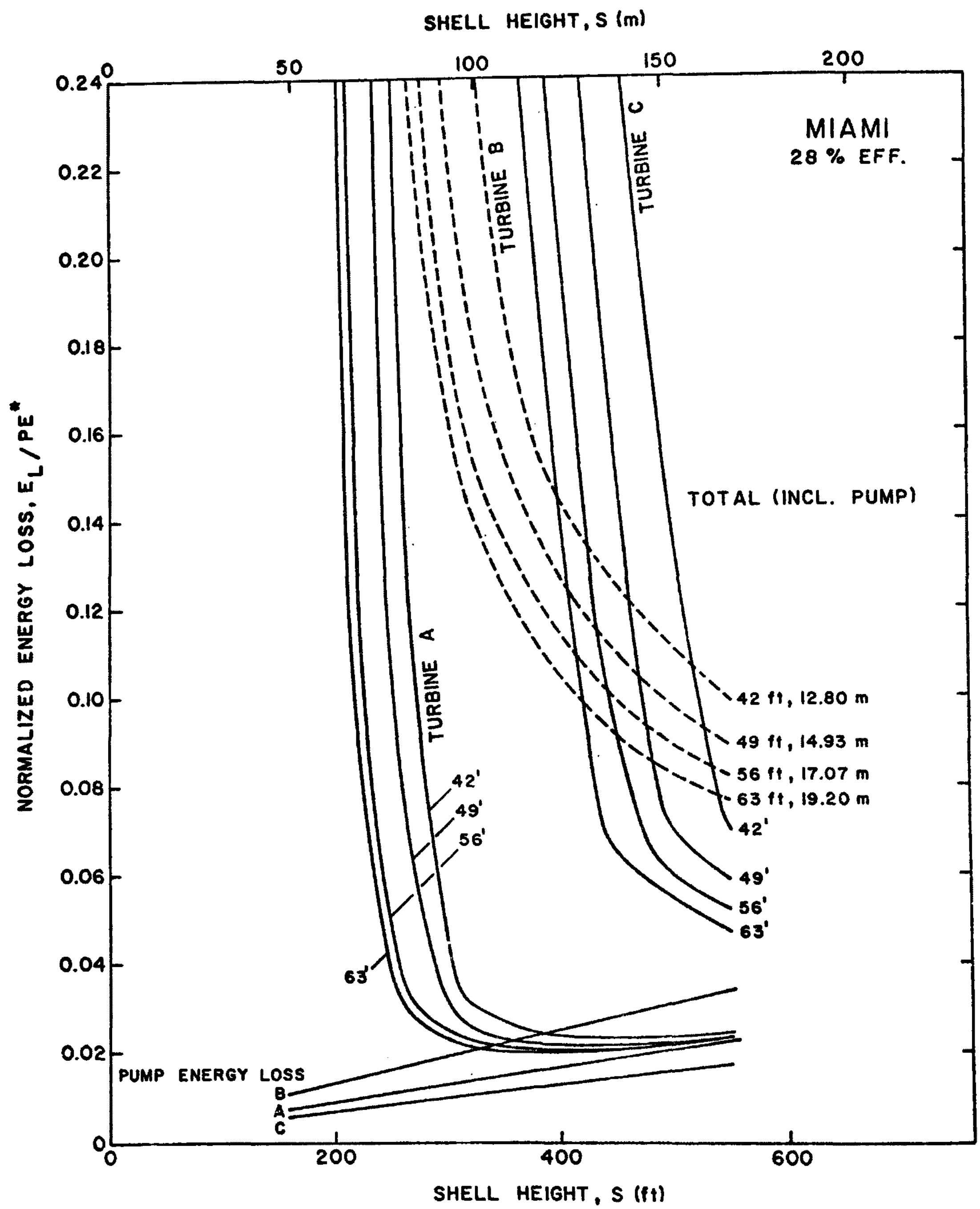

Figure 51(c). Normalized energy loss, 28% turbine efficiency, Miami

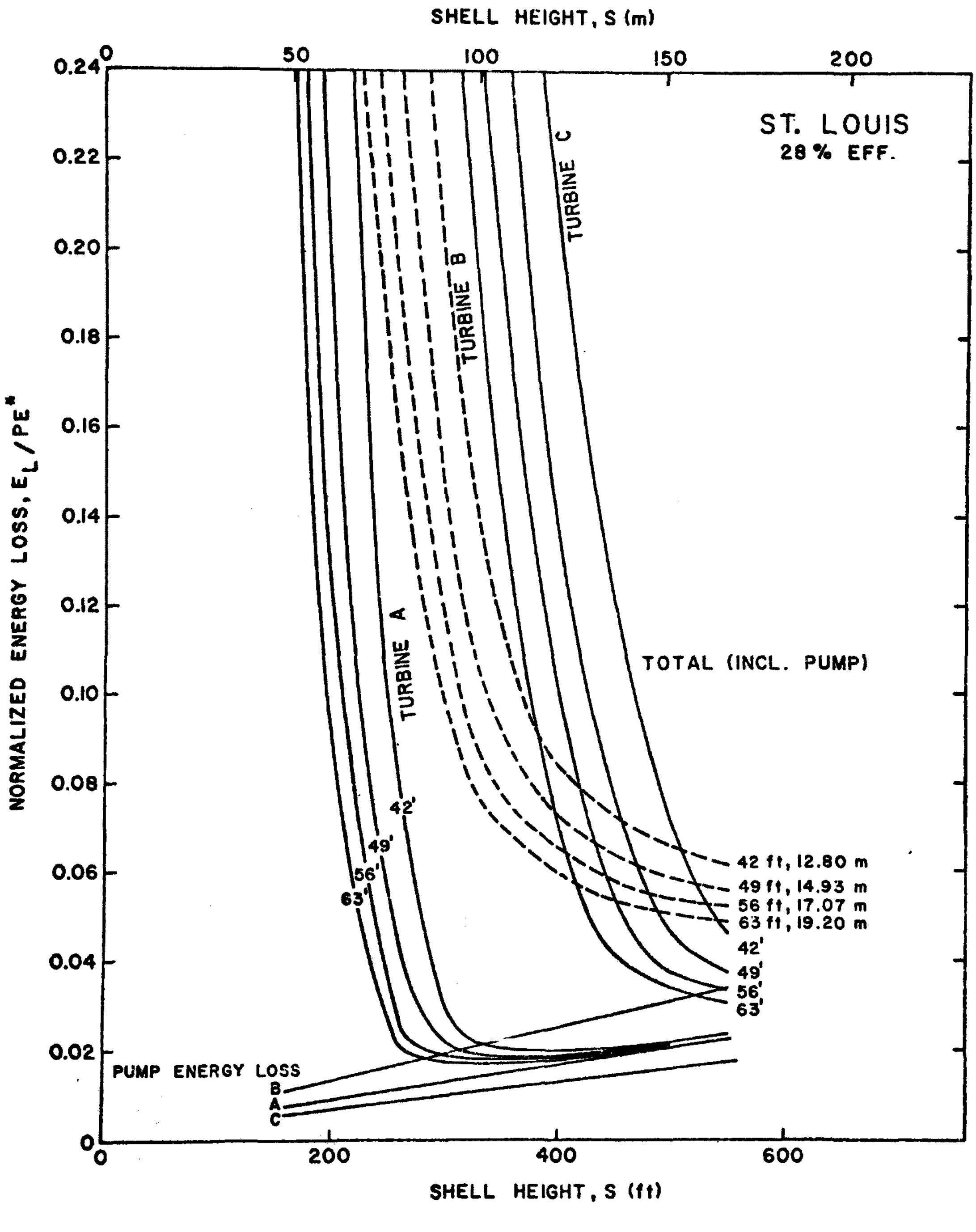

Figure 51(d). Normalized energy loss, 28% turbine efficiency, St. Louis

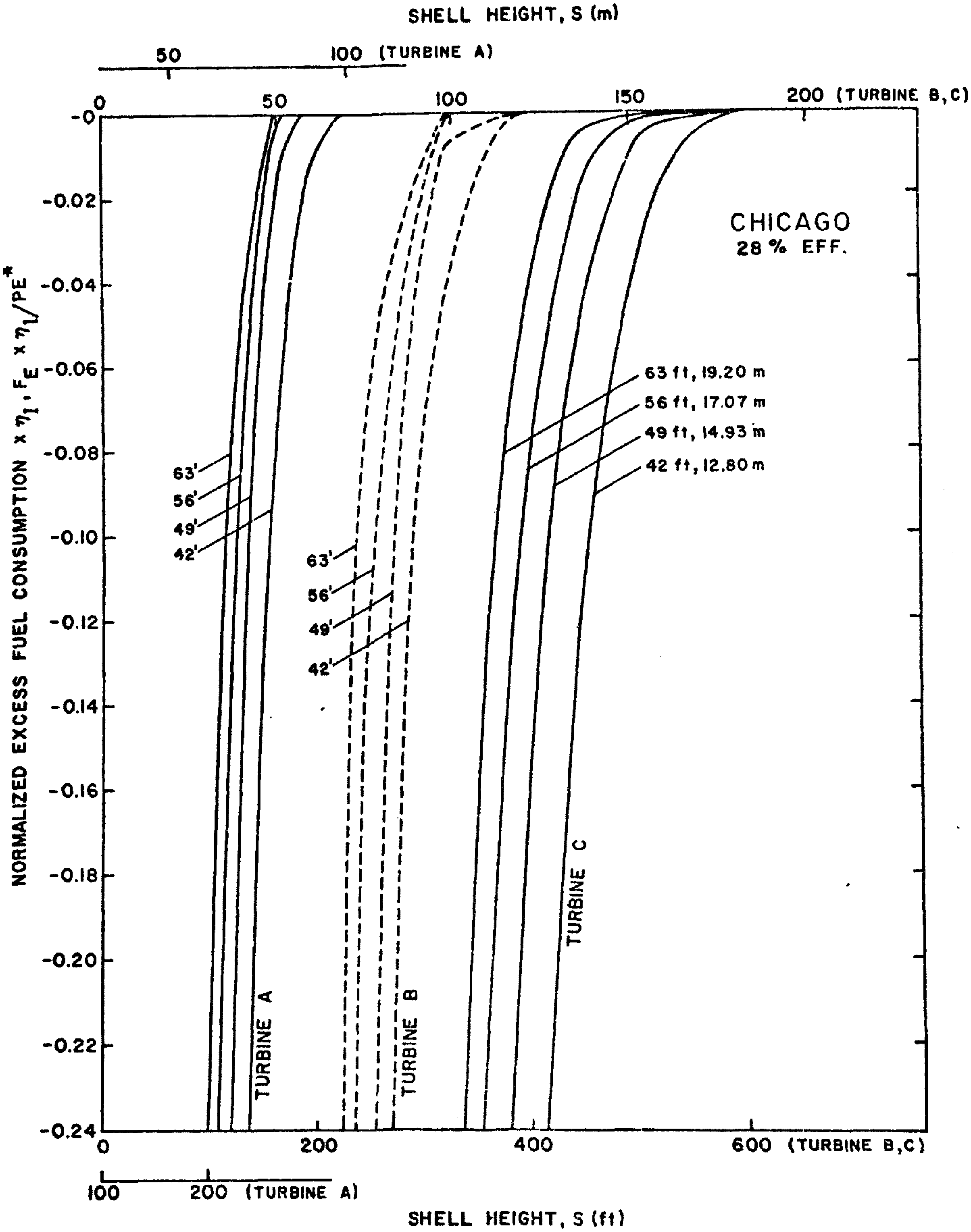

Figure 52(a). Normalized excess fuel consumption, 28% turbine efficiency, Chicago

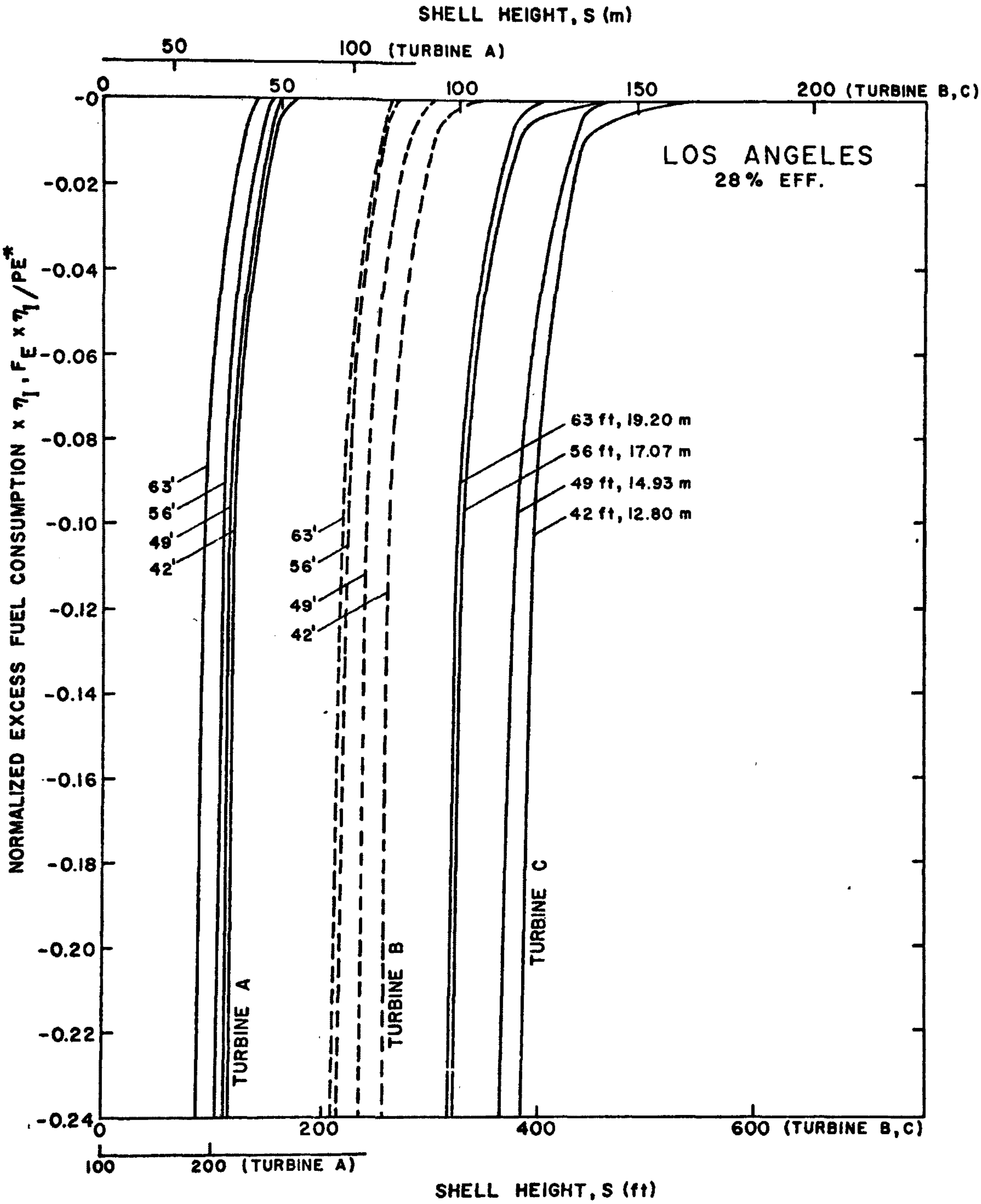

Figure 52(b). Normalized excess fuel consumption, 28% turbine efficiency, Los Angeles

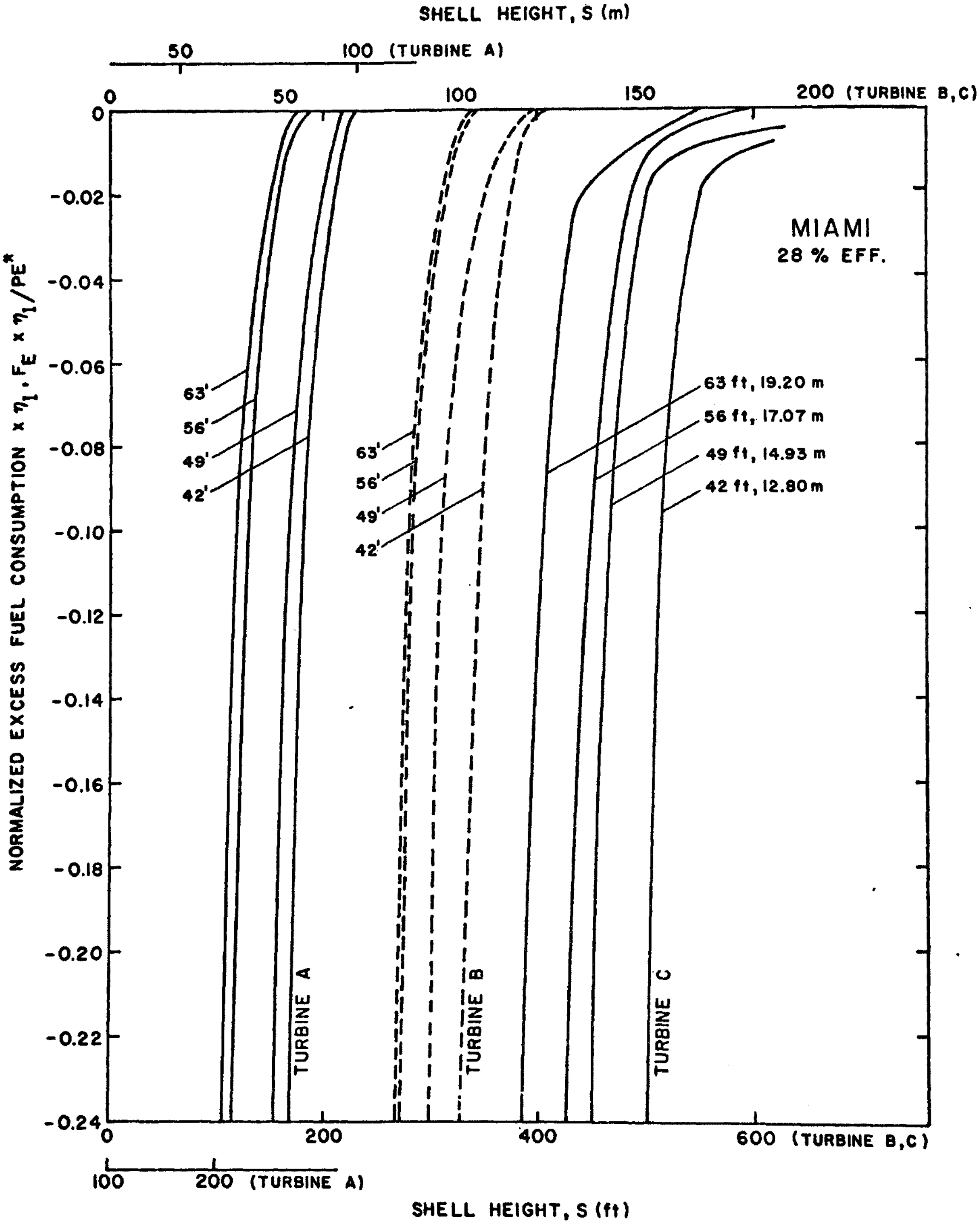

Figure 52(c). Normalized excess fuel consumption, 28% turbine efficiency, Miami

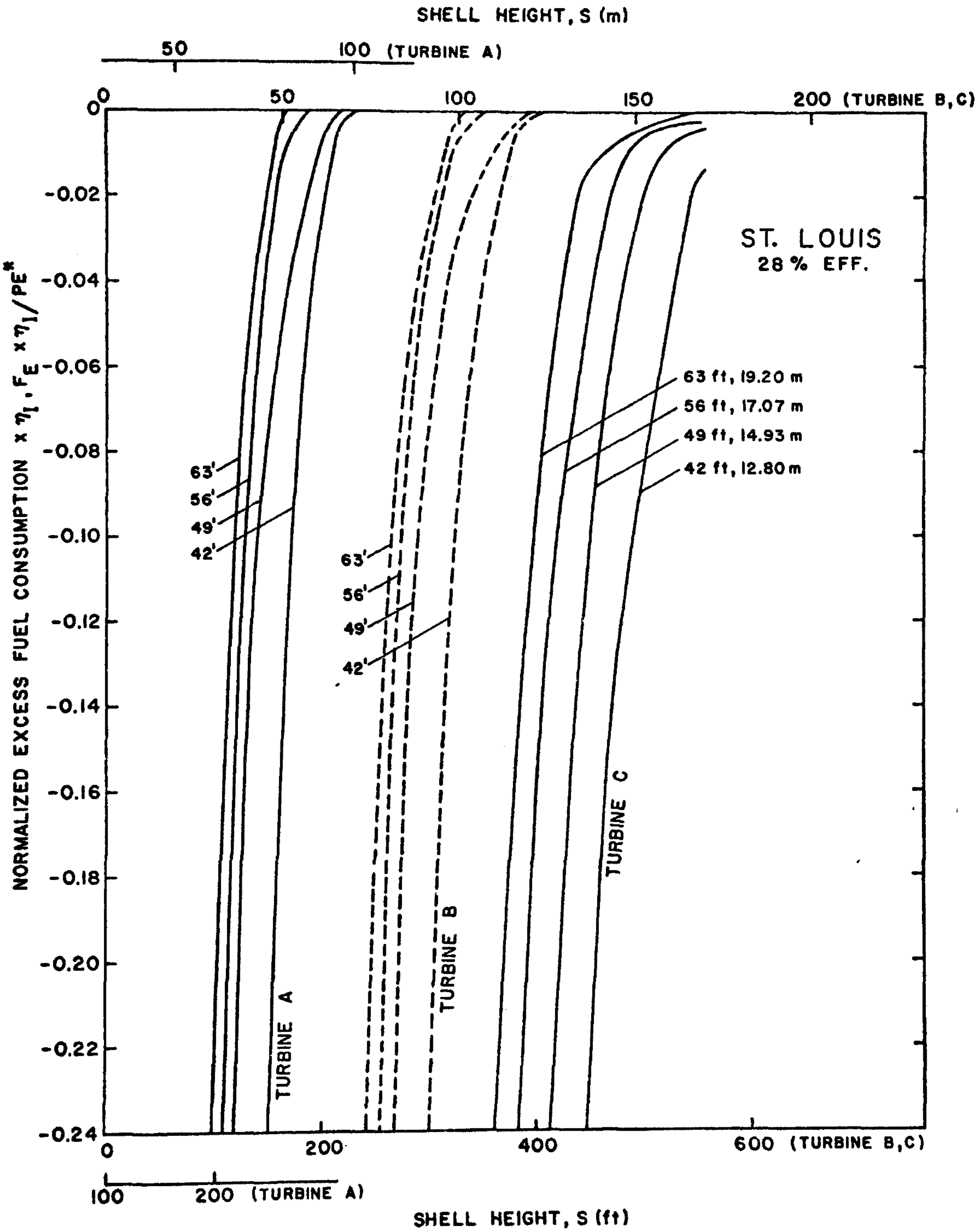

Figure 52(d). Normalized excess fuel consumption, 28% turbine efficiency, St. Louis

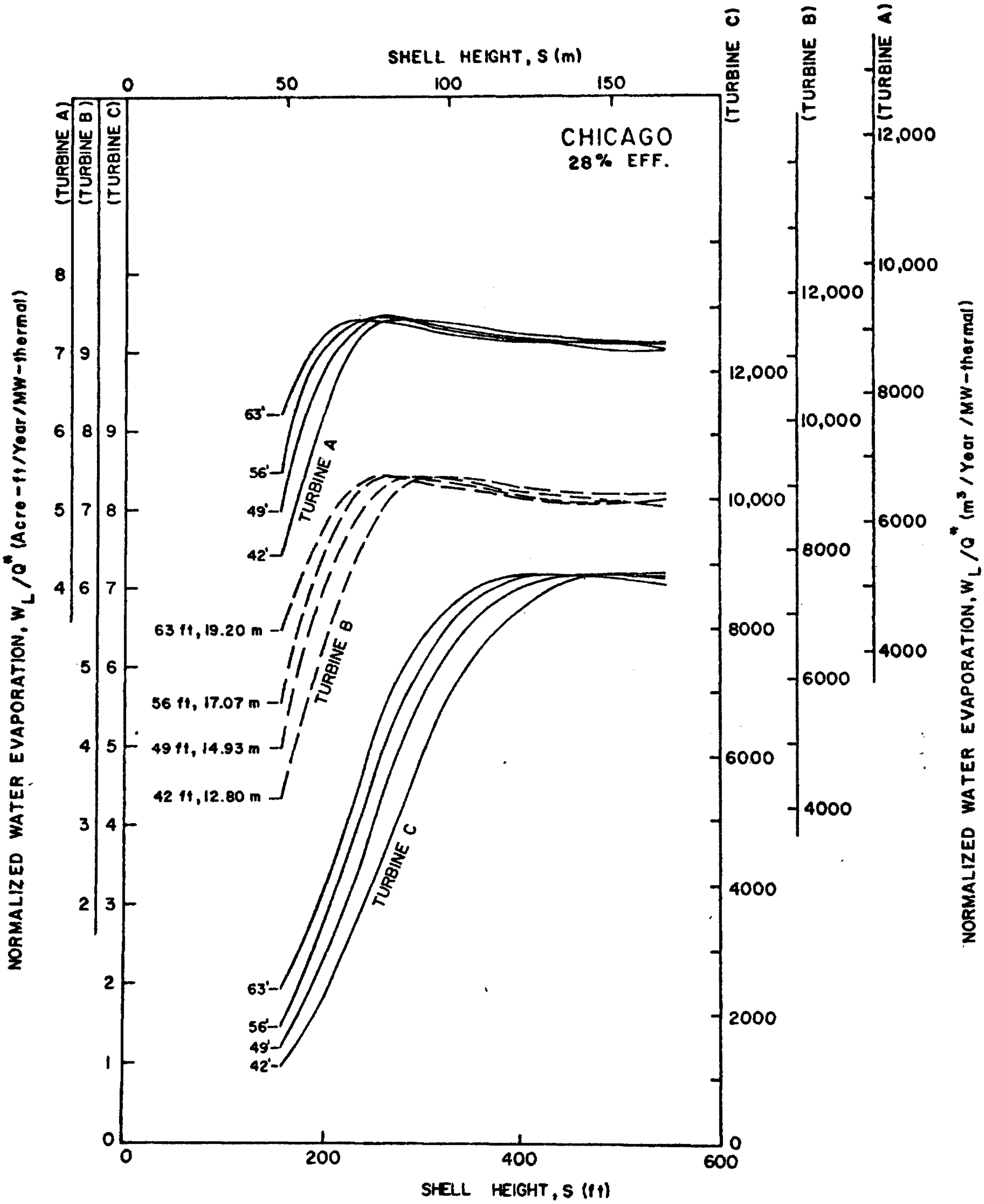

Figure 53(a). Normalized evaporation, 28% turbine efficiency, Chicago

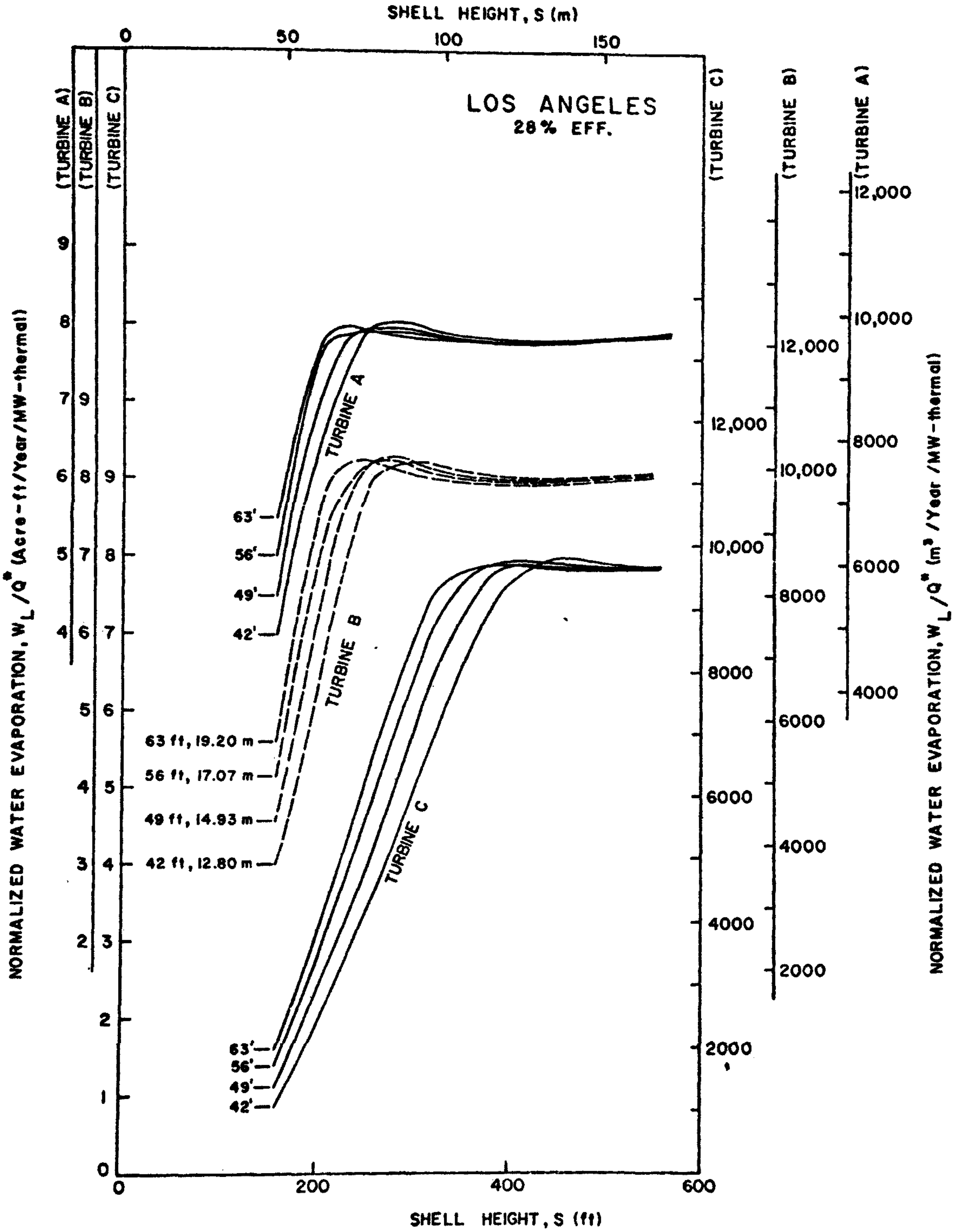

Figure 53(b). Normalized evaporation, 28% turbine efficiency, Los Angeles

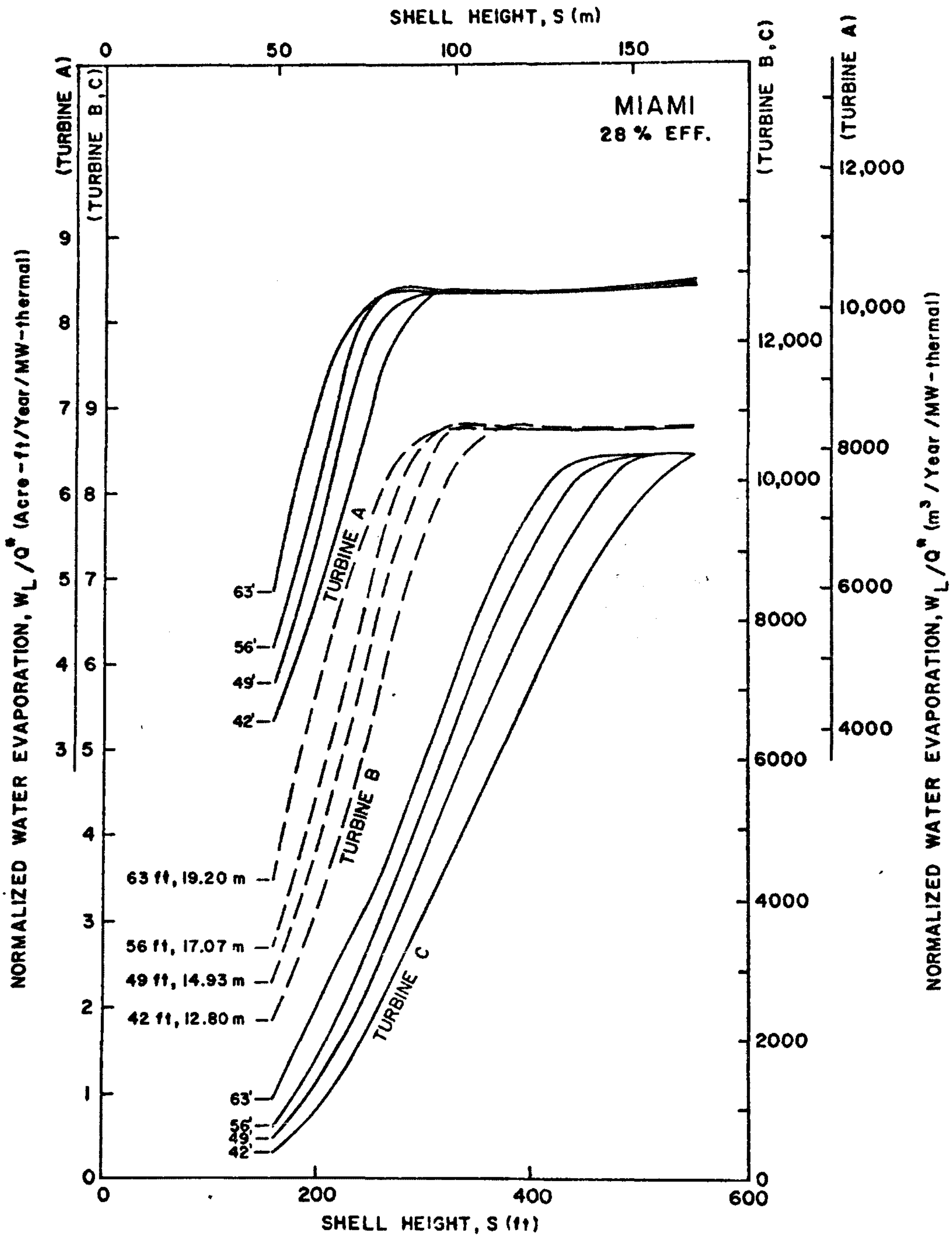

Figure 53(c). Normalized evaporation, 28% turbine efficiency, Miami

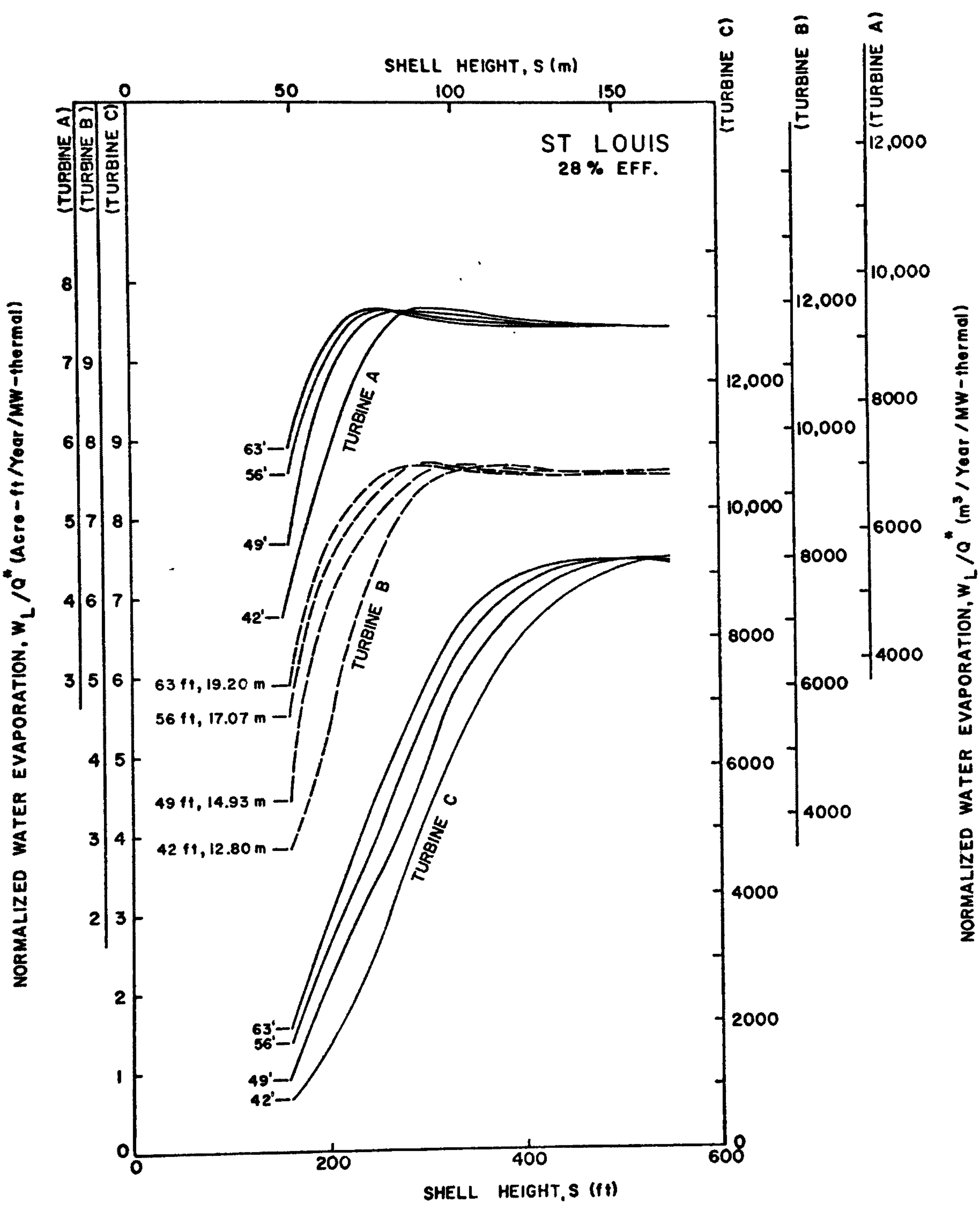

Figure 53(d). Normalized evaporation, 28% turbine efficiency, St. Louis

The economic evaluation of backfitting with natural-draft evaporative towers is very similar to the procedure already identified in Section V.F for mechanical-draft towers. In particular, the quantities which must be identified prior to the evaluation are the same as those listed in that section, except for items b-1,2. Instead the following quantities must be identified for natural-draft towers:

(b) Cooling Tower

1. The size of the cooling tower in terms of the shell height, S, and the height, H, of the evaporative pile*;
2. The capital cost of the natural-draft tower, C_{cs}, from Figure 32.

Once this information has been gathered, the calculation of the total differential cost of backfitting can be carried out either by using the computer program or by referring to the results presented graphically, if applicable. The general procedure to be followed is similar to that already identified with mechanical-draft towers in Section V.F, and it is further illustrated with the example presented in Section VI.H.

G. THE COMPUTER PROGRAM

The computer program which accepts any set of numerical values for the various parameters and performs the calculations outlined in the previous sections is listed in Appendix IV. The thermodynamic models used to simulate the performance of cooling towers are basically the same as those developed by Croley, Patel and Cheng [15] for the wet portion of dry-wet combination towers, but there are a number of important

*It will be recalled that S and H are sufficient to describe the physical size of the towers since the width of the pile (W) and the shape of the hyperbolic shell have been fixed, and since all detailed calculations are based upon a representative set of empirical data concerning the heat transfer properties, and water-loading in the pile.

differences in other respects. In particular, the economic considerations are formulated specifically for the analysis of backfitting an existing power plant or unit with natural-draft wet cooling towers and cannot be used, without modification, to study the design of towers for new plants or units.

The computer program consists of the MAIN program and eight subroutines, namely OPECOS, MODELW, NTUCAL, AIRFLR, BESTK, CAPCO, FOGSEN, and POWERS. The MAIN program reads all inputs, calculates the overall capital and total costs, and controls the printout of these quantities. The inputs, along with the symbols and units used, are listed in Appendix II. The primary functions of the various subroutines not previously identified are as follows:

AIRFLR: This subroutine is described in Section VI.A. It calculates, through iteration, the proper air flow rate and evaporative cooling. This subroutine calculates the buoyant air flow rise and outlet water temperature, given ambient dry- and wet-bulb temperatures, height of tower, tower friction factor, water temperature, and flow rate.

BESTK: This subroutine is also described in Section VI.A. It calculates, through iteration, the value of the tower friction factor which is appropriate for air flow rate calculations for a given tower specification.

CAPCO: Computation of the capital cost of the natural-draft cooling tower is made in this subroutine. Capital cost is determined as a function of the wet-bulb temperature, relative humidity, cooling range, approach, and heat rejection rate [16].

The overall program logic is similar to that already described for the mechanical-draft economic calculations. Minor changes have been made in the other subroutines and the main program to accommodate the different cooling model. These changes are included in the program listing in Appendix IV.

H. A HYPOTHETICAL TEST CASE

1. Consider a power plant with the characteristics identified in Section V.H.1, which also implies that the extreme dry-bulb temperature, $\hat{T}_{db}$ = 97.0°F (36.1°C) for Miami.

2. Assume that this plant is to be backfitted with a natural-draft cooling tower whose characteristics are:

Shell Height, S	= 400 ft (121.9 m)
Base Diameter, D_1	= 305 ft (93.1 m)
Pile Height, H	= 49 ft (14.9 m)
Pile Width, W	= 21 ft (6.4 m)
Water Loading, per unit plan area of pile	= 18 gpm/ft^2 (0.733 m^3/min/m^2)
Total water flow rate, GPM(= $18 \times \pi(D_1 + W) \times W$)	= 387,132 gpm (1465.3 m^3/min)
Concentration ratio, k^*	= 3.3
Unit blowdown treatment cost, c_b	= \$0.05/1000 gal. (\$0.0132/m^3)
Cost of hook-up and testing, C_{HT}	= included in cost of towers
Maintenance cost, C_m	= \$2,000/yr
Downtime, DT	= 720 hrs (30 days)

3. Assume that the various economic parameters are as identified in Section V.H.3.

4. Use of example results:

(a) Use Figure 32 to find the capital cost of the natural-draft evaporative cooling tower = \$5,160,000

(b) Refer to Ref. 7 to find the clearance width around tower corresponding to a desired noise level (e.g. 60dB) = 200 ft (61.0 m)

Thus, land area required, A_L from equation 36 is = 390,000 sq ft = 9 acres (3.6 hectares)

(c) Calculate the base efficiency of the turbine

$$\eta_T = \frac{cP^*}{cP^* + Q^*} = \frac{3.413 \times 10^6 \times 312.5}{3.413 \times 10^6 \times 312.5 \times 1.912 \times 10^9} \cong 0.36$$

(d) Read Figure 38 to find the normalized maximum capacity loss (for Los Angeles) for 411 MW, 36% efficiency (A), and 822 MW, 36% (B). Read Figure 42 to find the normalized maximum capacity loss for 411 MW, 28% (C), and 822 MW, 28% (D)

A = 0.020
B = 0.040
C = 0.026
D = 0.336

(e) Interpolate (log) "economy of scale" correction factor for 312.5 MW unit for normalized maximum capacity loss, i.e.

$$E = \ln A + \frac{(\eta_T - 0.36)}{(0.28 - 0.36)} (\ln C - \ln A) \qquad = -3.9120$$

$$F = \ln B + \frac{(\eta_T - 0.36)}{(0.28 - 0.36)} (\ln D - \ln B) \qquad = -3.2189$$

$$\text{correction factor} = \frac{F - E}{822 - 411} \qquad = 1.6865 \times 10^{-3}$$

(f) Read Figure 46(c) and 50(c) (Turbine A, Miami) to find the normalized maximum capacity loss, C_L/P^* for efficiencies of 36% (G), and 28% (H), respectively

G = 0.021
H =0.032

(g) Calculate the normalized maximum capacity loss corresponding to the given efficiency (I)

$$\ln I = \ln G + \frac{(\eta_T - 0.36)}{(0.28 - 0.36)} (\ln H - \ln G) \qquad = -3.8632$$

(h) Correct the normalized maximum capacity loss for economy of scale:

$\exp[\ln I + \text{correction factor} \times (P^* - 411)] = 0.01779$

Thus, the maximum capacity loss,
$C_L = 0.01779 \times 312.5 \times 1000 = 5560$ kW

(i) Repeat steps (d) through (h) using Figures 39, 43, 47(c), and 51(c) respectively, to determine the corrected value of the normalized energy loss, E_L/PE^*

A = 0.015
B = 0.011
C = 0.017
D = 0.029

$$E = \ln(0.015) + \frac{(0.36 - 0.36)}{-0.08} [\ln(0.017 - \ln(0.015)] = -4.1997$$

$$F = \ln(0.011) + \frac{(0.36 - 0.36)}{-0.08} [\ln(0.029) - \ln(0.011)] = -4.5099$$

correction factor

$$= \frac{-4.5099 + 4.1997}{822 - 411} = -7.5463 \times 10^{-4}$$

G = 0.018
H = 0.022

$$\ln I = \ln(0.018) + \frac{(0.36 - 0.36)}{-0.08} [\ln(0.022) - \ln(0.018)] = -4.0174$$

$\exp[\ln I + \text{correction factor} \times (P^* - 411)] = 0.01939$

Thus, the energy loss,

$E_L = 0.01939 \times 312.5 \times 1000 \times 8760 = 53.1 \times 10^6$ kW-hr/yr

(j) Repeat steps (d) through (h) using Figures 40, 44, 48(c), and 52(c) respectively, to determine the corrected value of the normalized excess fuel consumption, $\eta_I F_E/PE^*$ (use absolute values for log-linear interpolations)

A = 0
B = 0

C = 0

D = -0.01

but interpolation does not really apply since A, B, C, D, E, and $F \leq 0$; therefore set correction factor = 0

G = 0

H = 0

and using simple linear interpolation:

$$I = 0 + \frac{(0.36 - 0.36)}{-0.08} = 0$$

$$I + \text{correction factor} \times (P^{*} - 411) = 0$$

Thus, the excess fuel consumption

$$F_E = 0 \times 312.5 \times 1000 \times 8760/0.85 = 0 \text{ kW-hr/yr}$$

(k) Repeat steps (d) through (h) using Figures 41, 45, 49(c), and 53(c) respectively, to determine the corrected value of the normalized water evaporation, W_L/Q^*

A = 7.83

B = 7.79

C = 7.83

D = 7.04

$$E = \ln(7.83) + \frac{(0.36 - 0.36)}{-0.08}[\ln(7.83) - \ln(7.83)] = 2.0580$$

$$F = \ln(7.79) + \frac{(0.36 - 0.36)}{-0.08}[\ln(7.04) - \ln(7.79)] = 2.0528$$

$$\text{correction factor} = \frac{2.0528 - 2.0580}{822 - 411} = -1.2652 \times 10^{-5}$$

G = 8.40

H = 8.35

$$\ln I = \ln(8.40 + \frac{(0.36 - 0.36)}{-0.08}[\ln(8.35) - \ln(8.40)] = 2.1282$$

$$\exp[\ln I + \text{correction factor} \times (P^{*} - 411)] = 8.4102$$

Thus, the water evaporation,

$$W_L = 8.4102 \times 1.912 \times 10^{9}/3.413 \times 10^{6} = 4711 \text{ acre-ft/yr} \; (5.811 \times 10^{6} \text{ m}^3\text{/yr})$$

Also, blowdown, $W_b = W_L \dfrac{1}{k^* - 1}$ $= 2048$ acre-ft/yr (2.526×10^6 m^3/yr)

and makeup, $W_m = W_L \dfrac{k^*}{k^* - 1} = W_b + W_L$ $= 6759$ acre-ft/yr) (8.337×10^6 m^3/yr)

5. Cost Determination

Capital Costs

Cooling tower, C_{cs}	= \$5,160,000
Pump and pipe system (Figure 13 with total water flow rate = 387,132 gpm (1465.3 m^3/min), C_{pp}	= \$2,950,000
Pump and pipe system salvage, $C'_{pp} = 0.2\ C_{pp}$	= (\$590,000)
New condensers, C_c	= 0
Salvage value of old condensers, C'_c	= (0)
Salvage value of other open-cycle components, C'_o	= (0)
Hook-up and testing cost, C_{HT}	= included in tower cost
Additional land, $A_L a_\ell = 9 \times 3000$	= \$ 27,000
Replacement capacity, $CC_R = C_L c_\ell = 5560 \times 90$	= \$ 500,400
Downtime, $CC_{DT} = DT \times P^* \times e'_\ell$ $= 720 \times 312.5 \times 1000 \times 0.007$	= \$1,575,000
TOTAL CAPITAL COST, CC	\$9,622,400

Operating Costs/year

Excess fuel cost, $OC_{EF} = F_E f_c = 0 \times 0.000751$	= 0
Replacement energy cost, $OC_R = E_L e_\ell = 53.1 \times 10^6 \times 0.01$	= \$ 531,000
Supply water cost, $W_m c_w = 6759 \times (3.259 \times 10^5) \times 0.1/10^3$	= \$ 220,276

Cost of blowdown treatment,
$W_b c_b = 2048 \times (3.259 \times 10^5) \times 0.05/10^3$ = \$ 33,372

Maintenance of towers, C_m = \$ 2,000

Makeup water cost with open-cycle system, M' = (0)

Blowdown treatment cost with open-cycle system, B' = (0)

Maintenance cost of open-cycle system, C'_m = (0)

TOTAL ANNUAL OPERATING COST, OC = \$ 786,648

Total costs

From equation (20), the toal excess unit cost due to backfitting, tc, is given by

$$tc = \frac{OC + CC \times FCR}{8760 \times P^*}$$

$$= \frac{786{,}648 + (9{,}622{,}400 \times 0.179)}{312.5 \times 1000 \times 8760}$$

$$\boxed{tc = 0.9165 \text{ mills/kW-hr}}$$

The effectiveness of the logarithmic-linear interpolation/extrapolation scheme to correct the graphical results for economy of scale can be noted by comparing this solution with the results of the computer calculations. The total capital cost, annual operating cost, and excess unit cost given by the computer program are respectively: CC = \$ 9,974,271, OC = \$ 892,084, and tc = 0.9781 mills/kW-hr. The difference between the graphical result and the computer result for the total excess unit cost is seen to be approximately 6.4%. This small difference indicates that the graphical method with logarithmic-linear interpolation/extrapolation yields a good approximation for the given problem.

SECTION VII

COOLING PONDS

Man-made cooling ponds are a possible heat rejection method for backfitting needs. The economics of a cooling pond are dependent on topography, available land, and the ease of construction at a given site. The physical factors which determine the cooling capacity of these ponds have long been of fundamental interest in the fields of oceanography, limnology, hydrology, and meteorology. Various compendiums of information [24,25,26] have presented thermodynamic and economic methods and data for cooling ponds. Heat is rejected from cooling ponds by natural effects of conduction, evaporation, convection, and long-wave radiation. Ponds also absorb heat through solar and atmospheric radiation, plus waste heat from the power plant.

In most areas, the required size of a cooling pond is about 1 or 2 acres/MW (0.4 or 0.8 hectares/MW) [25], but some very efficient ponds require as little as 0.75 acres/MW (0.3 hectares/MW). At 4 acres/MW (1.6 hectare/MW) it is often possible to obtain cooling water temperatures within 5 degrees F (2.8 degrees C) of the equilibrium temperatures of open cycle cooling [24]. Generally, the overall water consumption is about 1 to 3% of the flow rate, comparable to cooling tower operations.

The economics of cooling ponds are strongly site dependent since they require large land areas and basins of low permeability. Like other evaporative systems, there may be problems of evaporative water loss, fogging, icing, and blowdown. Advantages of cooling ponds are simplicity of operation, low maintenance costs, low power requirements, aid in settlement of suspended solids, high thermal inertia, and they may also serve recreational purposes.

The thermodynamics of cooling ponds is also strongly site dependent due to meteorological and topographical variables. Analytical models have been developed to include transient effects, vertical temperature gradients, complex boundaries, and lateral and longitudinal temperature gradients [24,25,27,28,29]. Evaluation of these models and of various components of the heat balance equation are current topics of study [30,31,32,33]. The completely mixed, steady state, shallow cooling pond model is used in this report in an effort to make general economic evaluations of backfitting with a minimum of input parameters. The thermodynamics of this model are well known and restated in the following paragraphs.

A. OPERATION MODEL FOR THE FULLY-MIXED POND

To mesh with models describing condenser and power plant behavior, the cooling pond model was formulated to accept input parameters of two categories. One set of input data consists of parameters assumed to be fixed for a specified geographical location. For example, these parameters include atmospheric pressure, wind velocity, month of the year, fluid and thermal properties of water, cloudiness ratio, clear sky solar radiation values, and reflection percentages. The second set of input data consists of variables needed in the operation of the MODELW subroutine which performs the economic analyses. These variables are the temperature of the hot water entering the pond, the water flow rate, the area of the cooling pond, the dry-bulb air temperature, and the wet-bulb air temperature. The outputs of the model are: 1. the cold-water temperature (for a fully-mixed pond model, this temperature is the surface temperature of the pond), and 2. water loss due to evaporation from the pond.

In this section, the symbols for the input variables of the cooling pond model are:

T_H = temperature of hot water entering the cooling system

GPM = water flow rate

A = area of cooling pond

T_{db} = dry-bulb temperature of air

T_{wb} = wet-bulb temperature of air

The symbols for the output variables of the model are:

T_C,TS = cold water temperature, surface temperature of pond

W_L = water loss due to evaporation from the cooling pond

The outputs of the model are a result of solving the heat balance equation which is described next. The technique used is an iterative procedure which seeks out the surface pond temperature that satisfies the heat balance equation.

Significant terms in the heat balance equation are now reviewed. Complete discussions [24,25,33] are available in the literature. The general heat balance equation can be written:

$$QP + QR = QW + QE + QC \tag{37}$$

where QP = heat supplied to pond from power plant

QR = heat supplied to pond due to solar radiation and atmospheric radiation

The terms on the right-hand side of equation (37) are heat loss terms defined as

QW = long-wave radiation from pond

QE = evaporative heat loss

QC = conductive heat loss

The dimensions of these terms are heat/unit area/unit time. All of the terms in equation (37) except QR, depend on the surface water temperature of the pond. The purpose of the model is to calculate the surface temperature TS, by a simple iterative procedure.

The residual of equation (37), RES, is defined as the difference between the heat input and the heat loss terms, i.e.,

$$RES = QP + QR - (QW + QE + QC) \tag{38}$$

where RES is calculated for different estimates of the surface temperature. When RES = 0, the heat balance, equation (37) is satisfied. The iterative method is initiated by arbitrarily choosing two temperatures which span TS, and it advances by sequentially bisecting the temperature interval and correcting the limits of the temperature span as RES→0. When RES becomes sufficiently small, equation (37) is assumed to be satisfied.

In order to calculate the residual defined in equation (38) each of the heat transfer terms must be computed independently. The complete description of these heat transfer terms is found in the work by Ryan and Stolzenbach [34]. In the following equations, all of the heat transfer terms are daily averages given in Btu/ft^2/day. QR is the total radiative heat transfer expressed as

$$QR = QA - QAR + QS - QSR \tag{39}$$

where QA = atmospheric radiation to water surface
QAR = reflected atmospheric radiation from surface
QS = solar radiation to surface
QSR = reflected solar radiation from surface

An approximation [34, p. 1-23] to the atmospheric radiation term is

$$QA - QAR = 800 + 28T_{db} \tag{40}$$

This linear equation is applicable for $40 \leq T_{db} \leq 90^{\circ}F$. The solar radiation is approximated [34] by

$$QS = QSC(1.0 - 0.65\ R_c^2) \tag{41}$$

where QSC = the clear sky solar radiation, and R_c is the cloudiness ratio. The QSC depends on time of year and the latitude [24, p. 12, Fig. 4]. The worst condition (i.e., when solar radiation to a pond will be greatest) can be approximated as:

$$QSC = 2800 \text{ Btu/ft}^2\text{/day} \tag{42}$$

This value is used in the present calculations, and an average value of $R_c = 0.5$ is also used.

The long-wave radiation from the water surface is usually the largest item in the heat balance equation, and it is expressed as [34, p. 1-23]

$$QW = 4.10 \times 10^{-8} \ (TS + 460)^4 \tag{43}$$

The evaporative heat loss term is still subject to considerable research. For evaporation from a heated surface the MIT formula, based on field data, [34, p. 1-34, Eq. 44] is used.

$$QE = [22.4 \ (\Delta\theta_V)^{1/3} + 14.0V_2](e_S - e_A) \tag{44}$$

where V_2 = wind speed (mph) at a height of 2m

e_S = air saturation vapor pressure at water surface temperature (mm Hg)

e_A = air vapor pressure at T_{db} (mm Hg)

$\Delta\theta_V$ = virtual temperature difference $= TS_V - TA_V$

where $TS_V = TS/(1.0 - 0.378e_S/p_A)$

$TA_V = T_{db}/(1.0 - 0.378e_A/p_A)$

p_A = atmospheric pressure = 760 mm Hg

Some further comment is necessary for the evaporative heat loss term. If the wind speed is given at some height other than 2 m, then the logarithmic velocity profile can be used to approximate V_2.

$$V_2 = V_z \frac{\ln(2.0/z_o)}{\ln(z/z_o)} \tag{45}$$

where $z_o = 0.005$ m

z = arbitrary height

V_z = wind speed at height z.

For all the calculations, an average wind speed is chosen particular to the geographic location of the cooling pond. As noted in Ref. 34 (p. 1-44), equation (44) is only valid when $e_S > e_A$, or when the evaporative heat loss is out of water. In each iteration, the evaporative heat loss, QE, is calculated by equation (44) with $\Delta\theta_V$ replaced by $|\Delta\theta_V|$, if $TS \geq T_{db}$, and if $TS < T_{db}$, QE is set equal to zero.

The conductive heat loss term is discussed in Ref. 34 (p. 1-42). The relationship given there is written as

$$QC = c_2(e_S - e_A)\left|\frac{TS - T_{db}}{e_S - e_A}\right|[22.4(\Delta\theta_V)^{1/3} + 14.0V_2] \qquad (46)$$

where $c_2 = 0.255$ mm Hg/^oF

It should be noted that QC can be negative. The water loss term which is closely related to the evaporative heat loss is discussed next.

The water loss by evaporation can be calculated after the evaporative heat loss is known. The latent heat of vaporization is given by [35, p. 60]

$$H_v = 1087 - 0.54(TS) \text{ (Btu/lb)}$$

The evaporative water loss per unit area is then

$$w_\ell = QE/\gamma H_v \text{ (ft}^3\text{/ft}^2\text{/day)} \qquad (47)$$

where $\gamma = 62.4$ lb/ft^3

B. CAPITAL COSTS OF COOLING PONDS

The capital costs given below should be accepted as rough estimates based on the source information which is stated. In particular, the date bases for the costs are approximated from the vague source material which is available. Because of the large land area requirements, land cost is the most important economic factor for cooling ponds. The additional land necessary for backfitting with a cooling pond, A_L, is split into two categories, viz., A, the land required for the cooling pond itself and, A_a, additional land needed for access roads, placement of service facilities, landscaping, and other miscellaneous uses. ($A_L = A + A_a$) Land costs are highly variable and typical values given in the literature range from \$500 to \$5,000 per acre (\$1235 to \$12,355 per hectare). In the hypothetical example presented in Section VII.G, unit cost for the pond itself, c_p, including land, pond preparation and construction is taken as \$5,000/acre (\$12,355/hectare), and the cost of the access land, c_a, is taken as \$3000/acre (\$7,413/hectare). The amount of land needed for access roads, etc. is estimated as ten percent of the pond area; i.e., $A_a = 0.1 \times A$, and the total capital cost of the cooling pond and access land is given by the sum, $c_pA + c_aA_a$.

The capital cost for pumps and pipe systems is taken from the report by Jedlicka [26]. In the nomographs of Ref. 26 (p. N-43), the pump and pipe system costs are shown (assumed to be in 1970 dollars) as functions of water flow rate (gpm) and total head (ft). In the example which follows, a water flow rate of 0.54×10^6 gpm (2.04×10^3 m^3/min) and an assumed head of 40 ft (12.2 m) results in a total cost of \$$0.80\times10^6$. This is approximately \$1.50/gpm (\$396/m^3/min) which is considerably higher than the figure of \$0.50/gpm (\$132/m^3/min) stated in an earlier work [24, p. 81]. Of course, total head is strongly site dependent and contributes significantly when the pond is located a large distance from the plant. Usually, an advantage of cooling ponds, is that maintenance costs for the pond itself are low. For the present example, \$2.00/acre/year (\$4.94/hectare/year) [24, p. 84] (1970 dollars, assumed) is used for maintenace costs.

C. REFERENCE AREA OF COOLING POND, A*

As discussed in Section V.B., P* is the rated or nameplate capacity obtained at the reference back pressure, p*, which occurs when the excess turbine heat rate, Δ, is zero. The corresponding heat rejection rate is Q*. Then, for any given water loading, the reference area of the pond, A*, can be defined as the area required for the reference heat rejection rate, Q*, while maintaining the turbine back pressure at a specified value, p', at the reference meteorological conditions. As explained in Sections V and VI, all of the reference values chosen for computing A* may be selected arbitrarily. For this study, the reference dry-bulb temperature is set at $70^{\circ}F$ ($21.1^{\circ}C$), and the reference wet-bulb temperature at $60^{\circ}F$ ($15.6^{\circ}C$). The reference wind speed is set at 8 mph (3.57 m/sec) and the clear sky solar radiation to the pond is taken as 2800 Btu/ft^2/day (3.18×10^4 kJ/m^2/day).

The reference area A*, can be found by considering the heat transfer characteristics of the condenser and the performance of the cooling pond at the reference meteorological conditions. The reference area then depends on the values of Q*, water loading, the specified turbine back pressure, and the fixed meteorological conditions. The details for calculating A* are the same as for calculating the reference length of the mechanical-draft cooling tower, Section V.B. The results are shown in Figure 54 for a specified turbine back pressure of 2.0 in. Hg (5.08 cm Hg). The reference area, A*, is a nearly linear function of the reference heat rejection rate, Q*, and A* increases with increasing values of water loading. The reference area found from Figure 54 is used to nondimensionalize the pond area.

D. PARAMETRIC STUDIES

Detailed calculations can be made using the cooling pond as the closed-cycle cooling system to be backfitted to an existing power plant or unit. The calculations are made for a range of values of pond surface area and water loading. The pond area is normalized with respect to

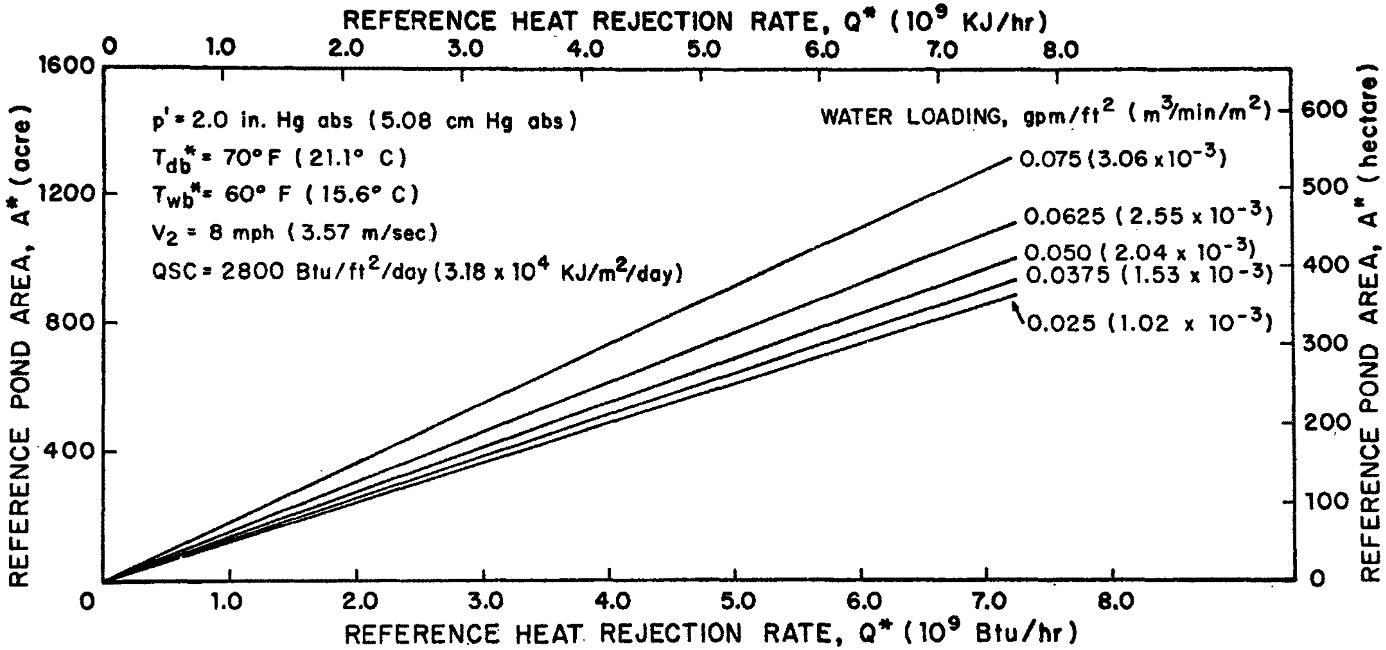

Figure 54. Reference area of cooling ponds

the reference area, A*; the maximum capacity loss, annual energy loss, annual excess fuel consumption, and annual evaporative water loss are all normalized as in Sections V and VI. Meteorological conditions are chosen as representative of four geographical locations: Chicago, Los Angeles, Miami, and St. Louis, as discussed in part B of Section III. In addition to the meteorological parameters of the ambient dry- and wet-bulb temperatures, the wind speed, clear sky solar radiation, and cloud cover must be considered in the study of cooling ponds. Values of the last three are chosen as fixed for all geographical locations in this parametric study. Because of the thermal inertia of cooling ponds, the use of average values for these parameters is probably more realistic than for the other cooling systems considered in this report.

As for the other cooling systems, it is necessary to carry out studies of the 411 MW and the 822 MW power plants, under identical meteorological conditions, in order to check for scale effects on the computed results. Variations of the normalized quantities (maximum capacity loss, annual energy loss, annual excess fuel consumption, and annual water loss due to evaporation) with the normalized pond area for the 411 MW plant (turbine A, $Q^* = 2.545 \times 10^9$ Btu/hr $= 2.686 \times 10^9$ kJ/hr) and the 822 MW plant (turbine A, $Q^* = 5.090 \times 10^9$ Btu/hr $= 5.372 \times 10^9$ kJ/hr) for meteorological conditions at Los Angeles (Table 4 and 5) and one water loading (0.025 gpm/ft^2 $= 1.02 \times 10^{-3}$ m^3/min/m^2) are shown in Figures 55 through 58. Since the curves shown in these figures do not completely collapse, it is seen that an economy of scale exists for the fully-mixed cooling pond (similar to the natural-draft cooling tower, as discussed in Section VI.E.).

Parametric studies of variations of the normalized quantities mentioned above with normalized pond area for the four geographical locations are shown in Figures 59 through 62. Since the study on scale effects, Figures 55 through 58, indicates differences in the normalized

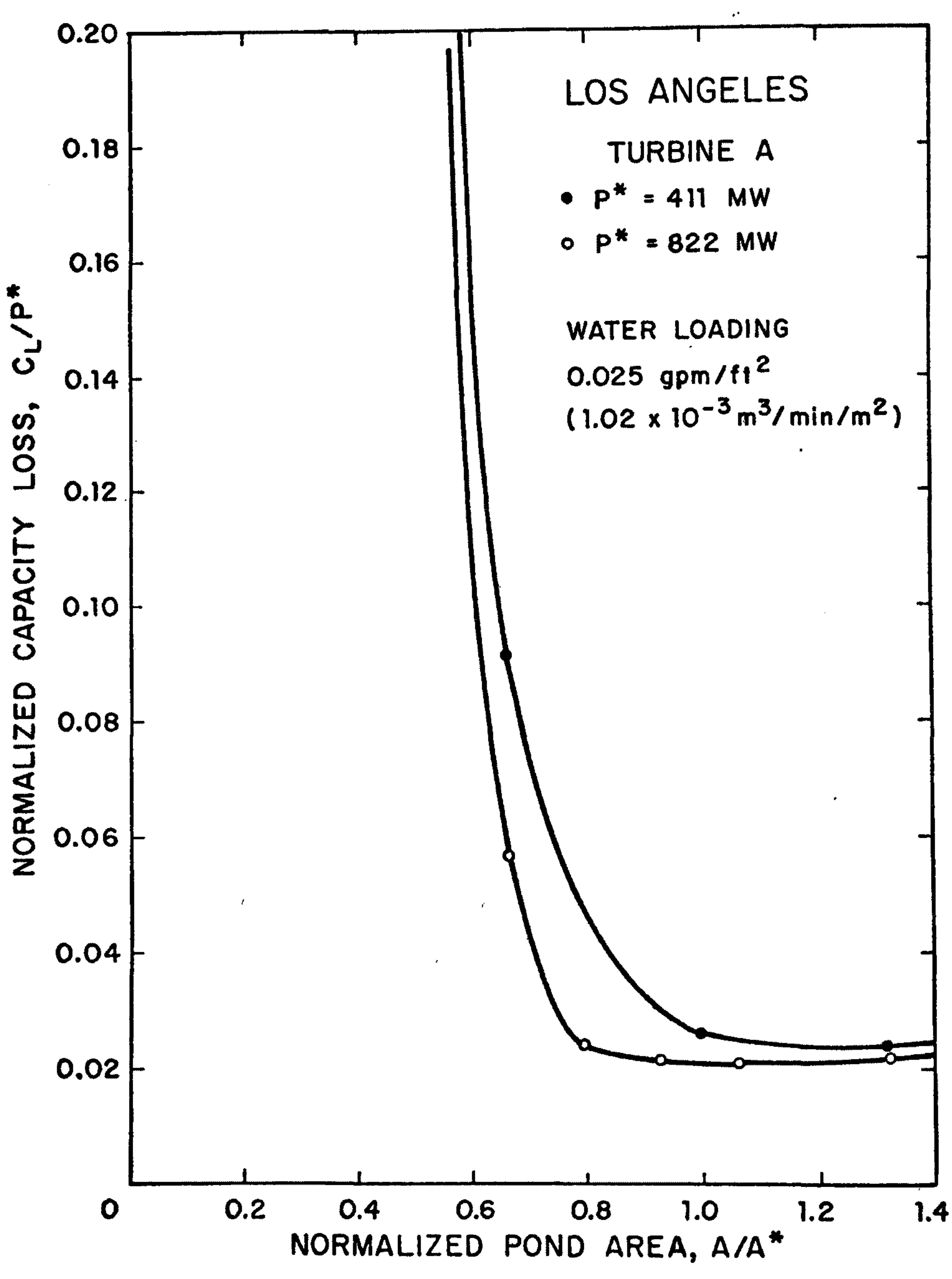

Figure 55. Normalized capacity loss

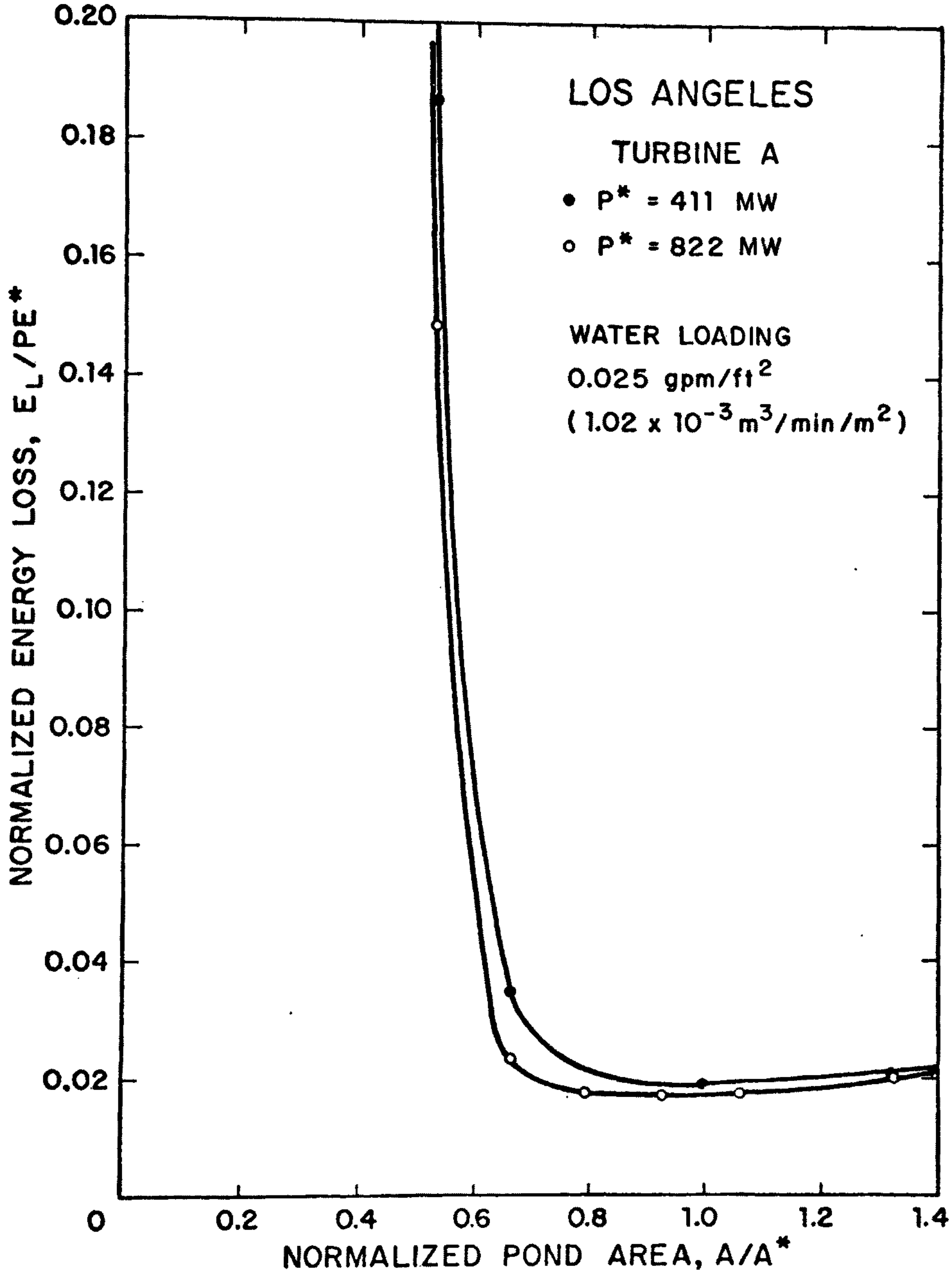

Figure 56. Normalized energy loss

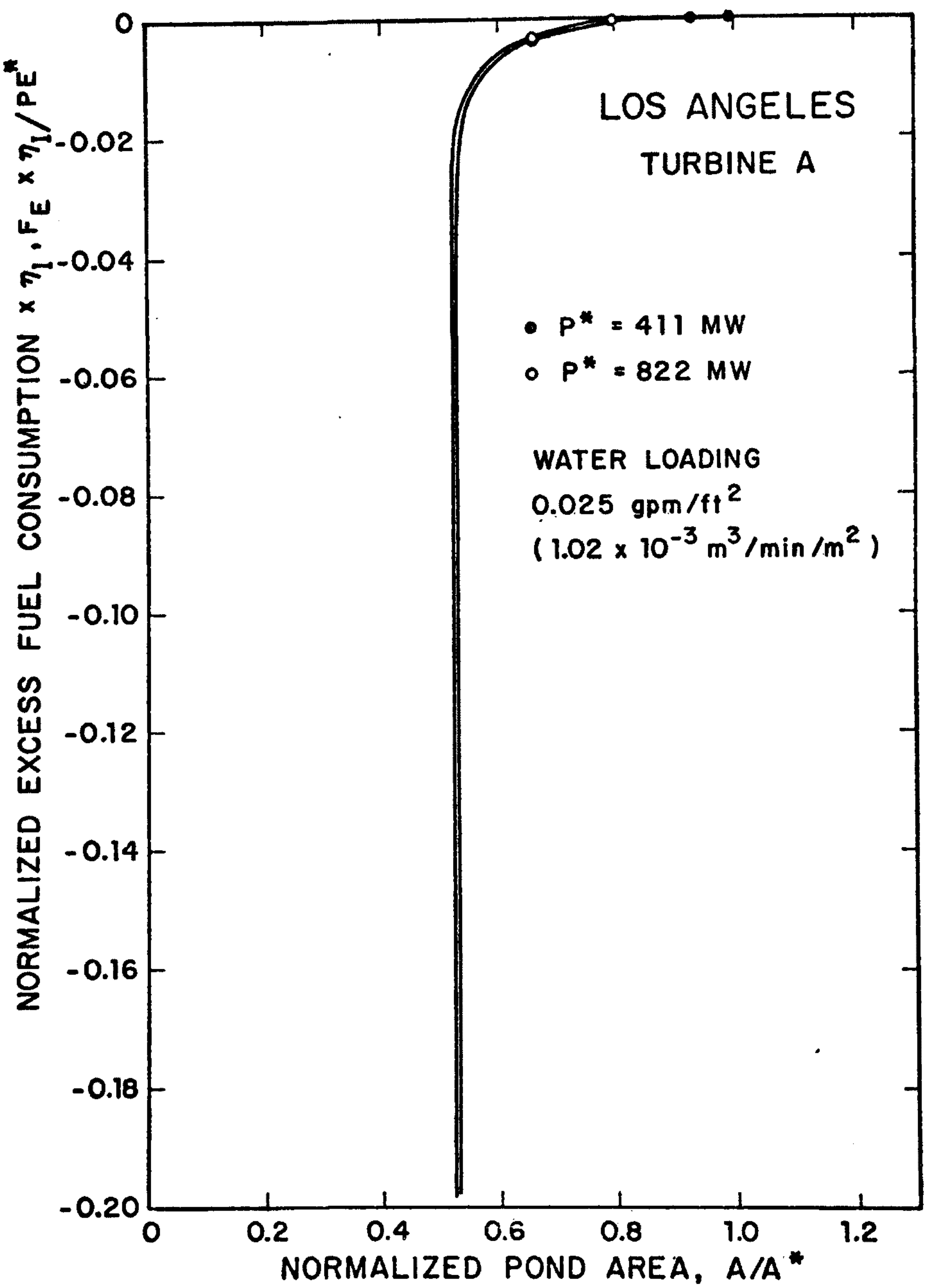

Figure 57. Normalized excess fuel consumption

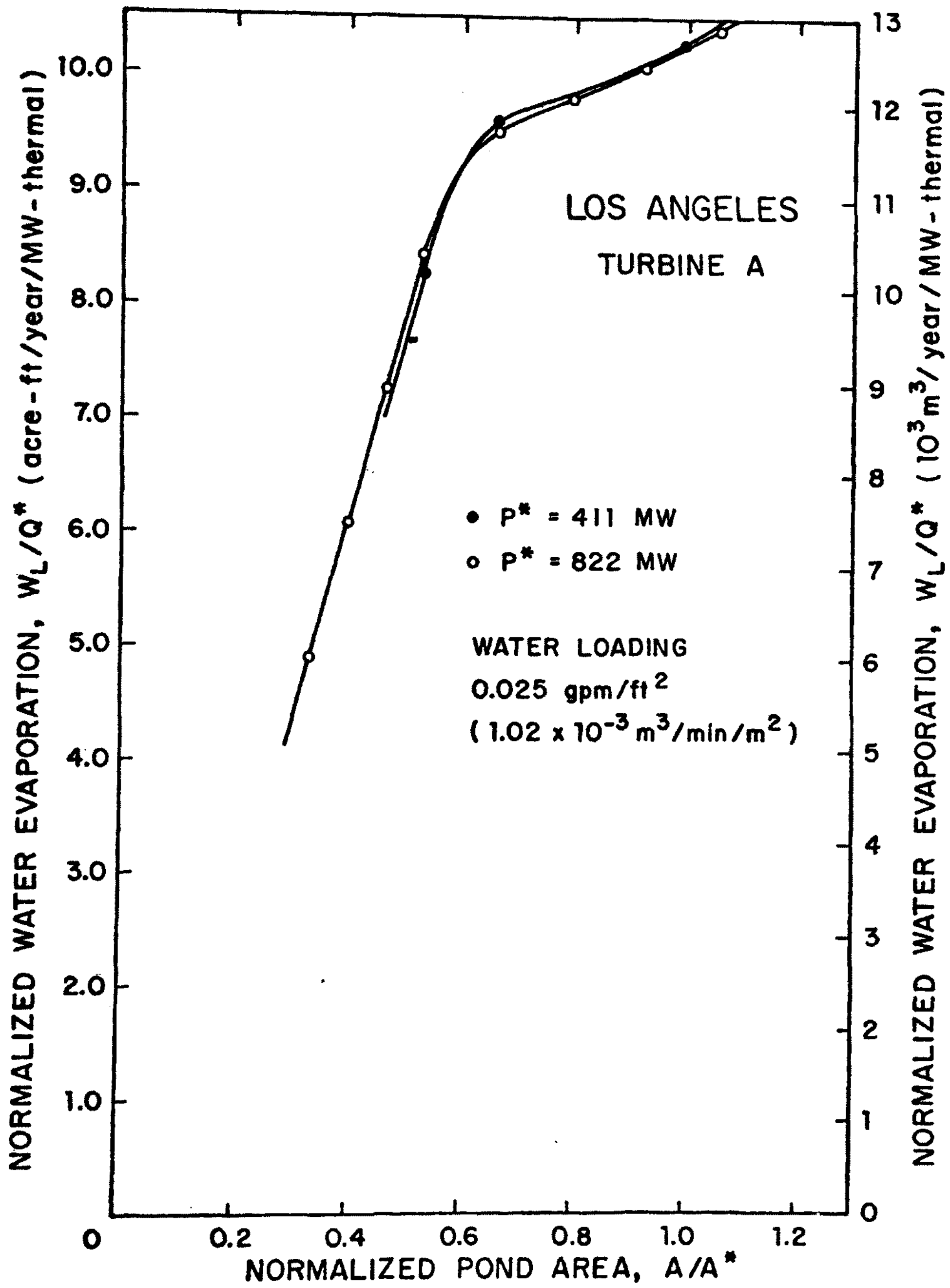

Figure 58. Normalized evaporation

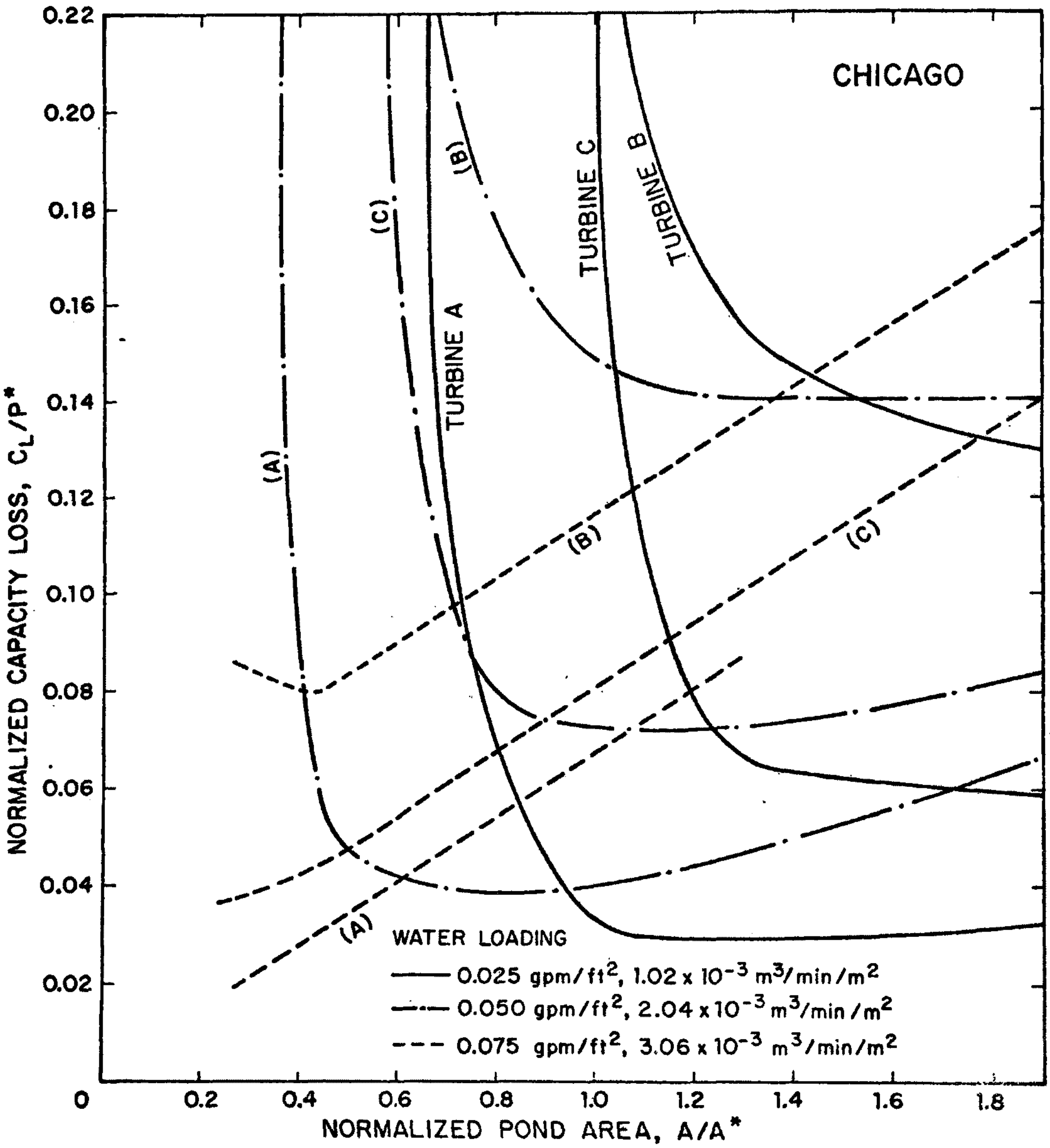

Figure 59(a). Normalized capacity loss, Chicago

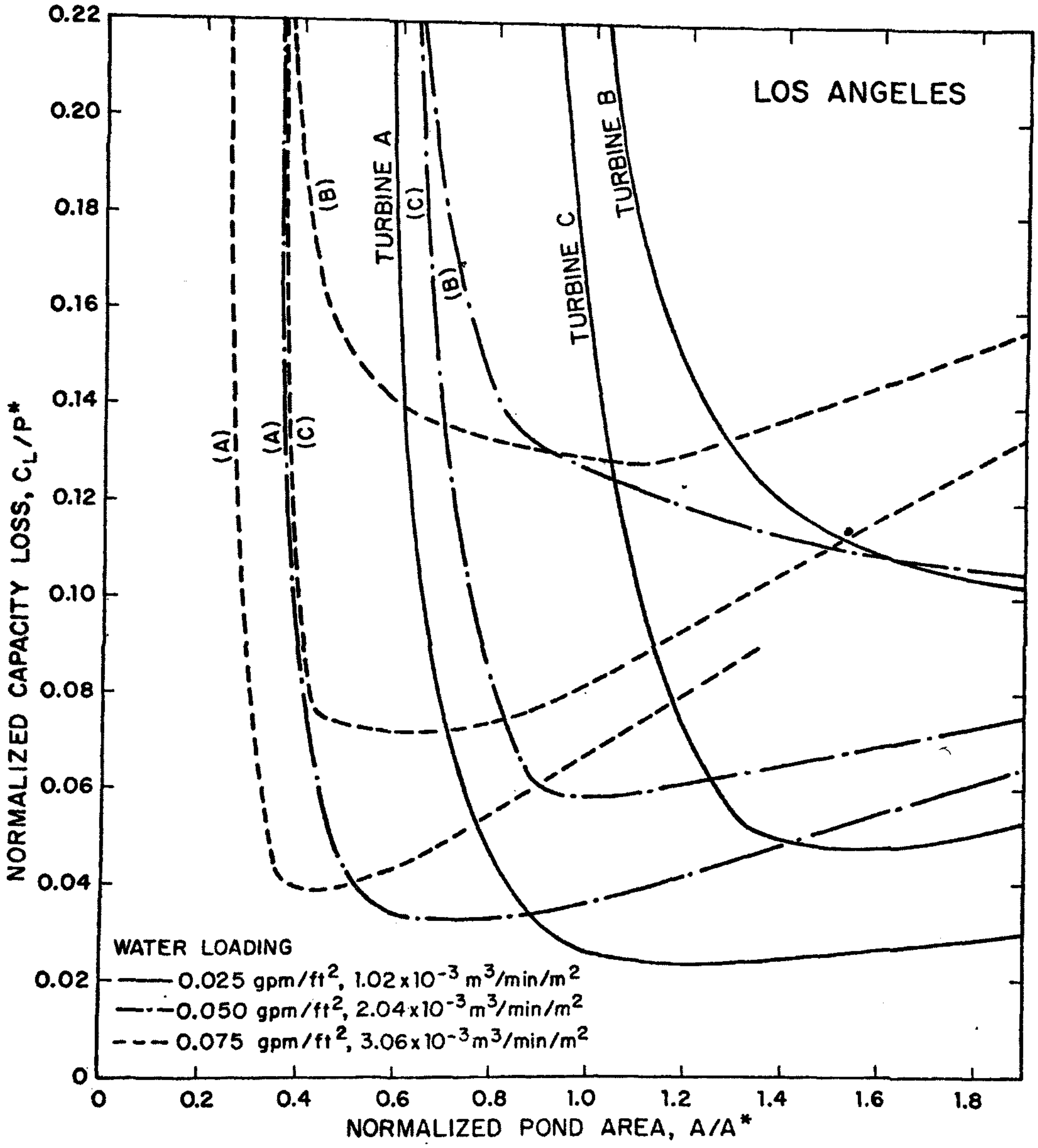

Figure 59(b). Normalized capacity loss, Los Angeles

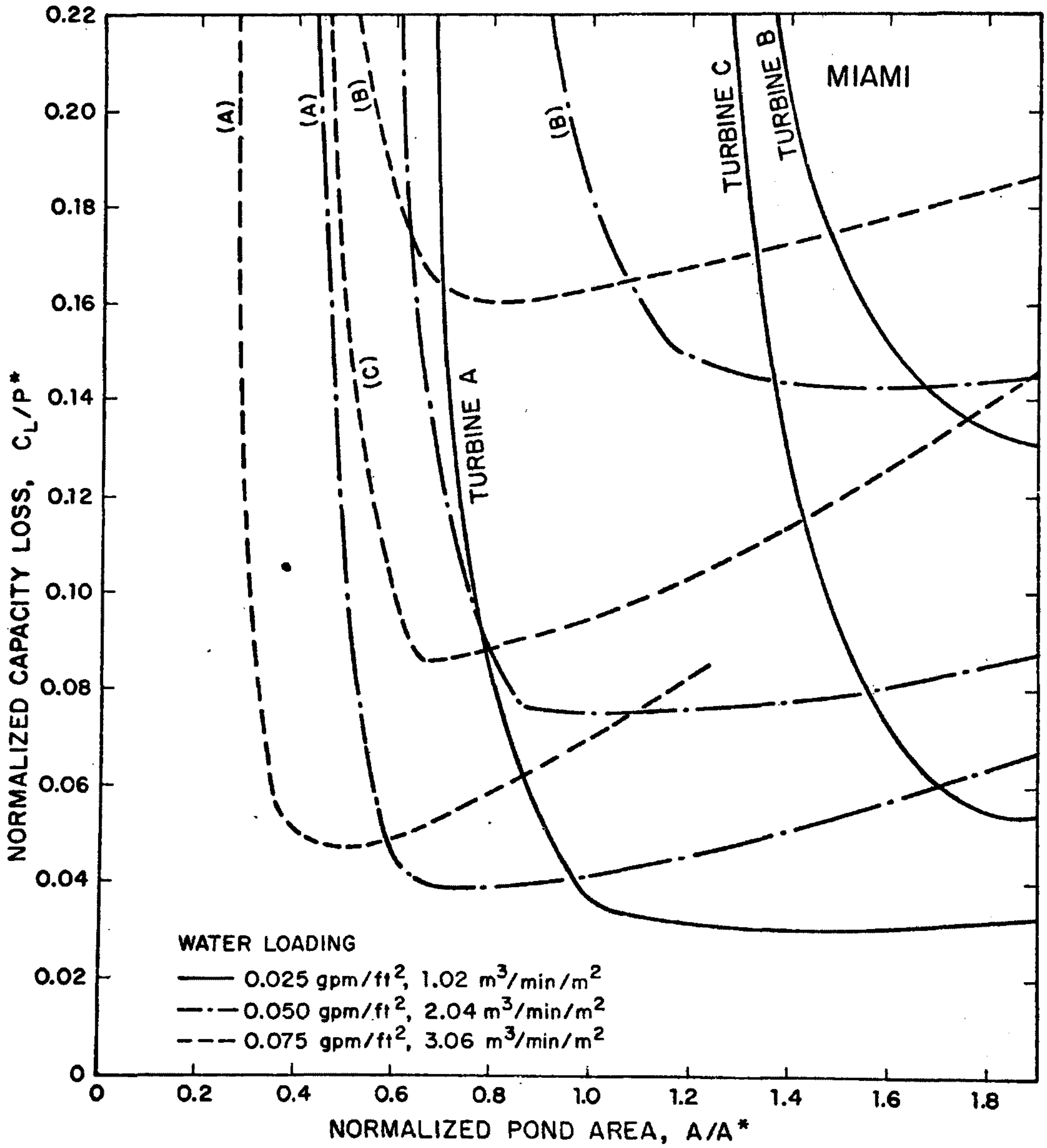

Figure 59(c). Normalized capacity loss, Miami

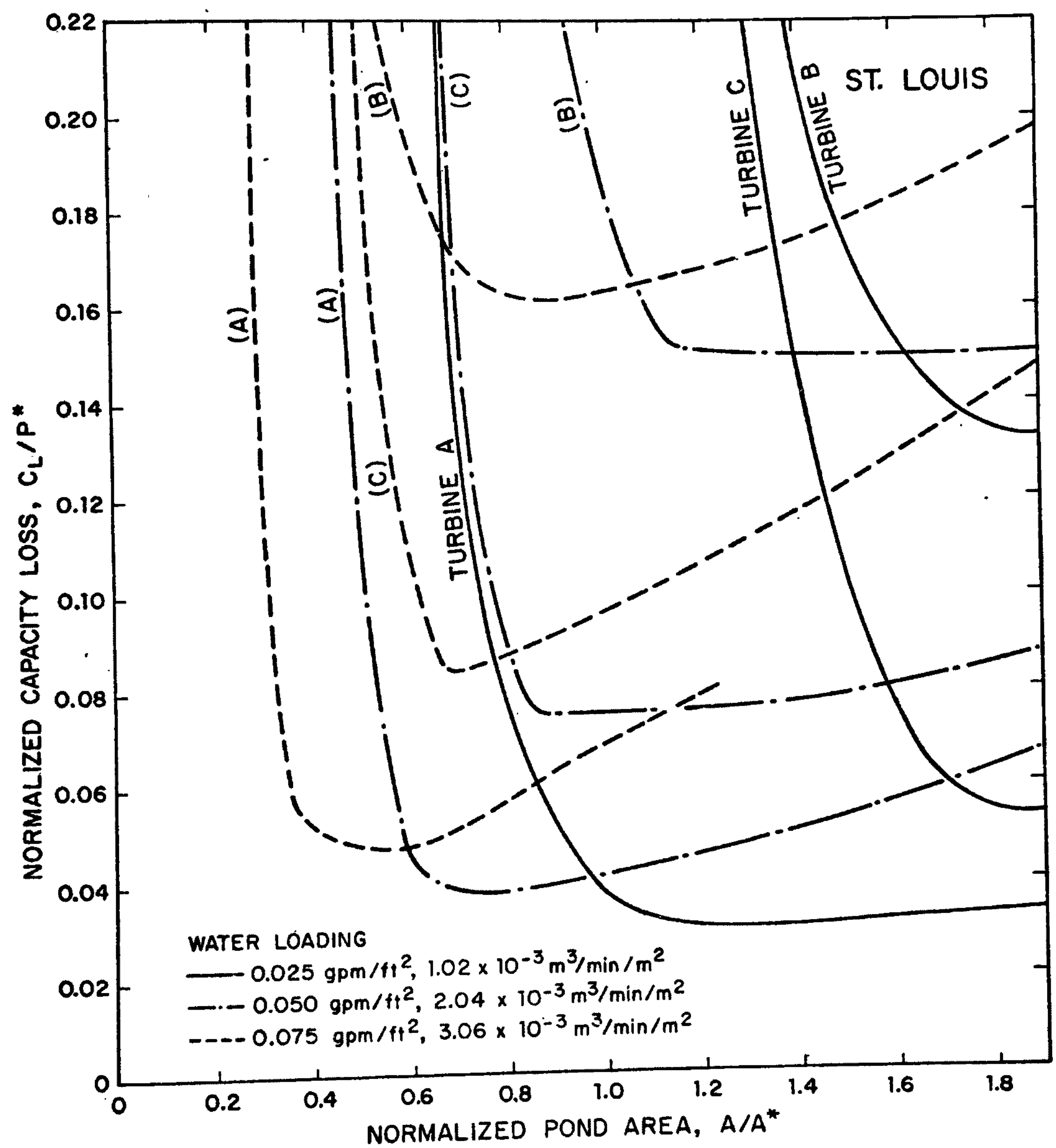

Figure 59(d). Normalized capacity loss,
St. Louis

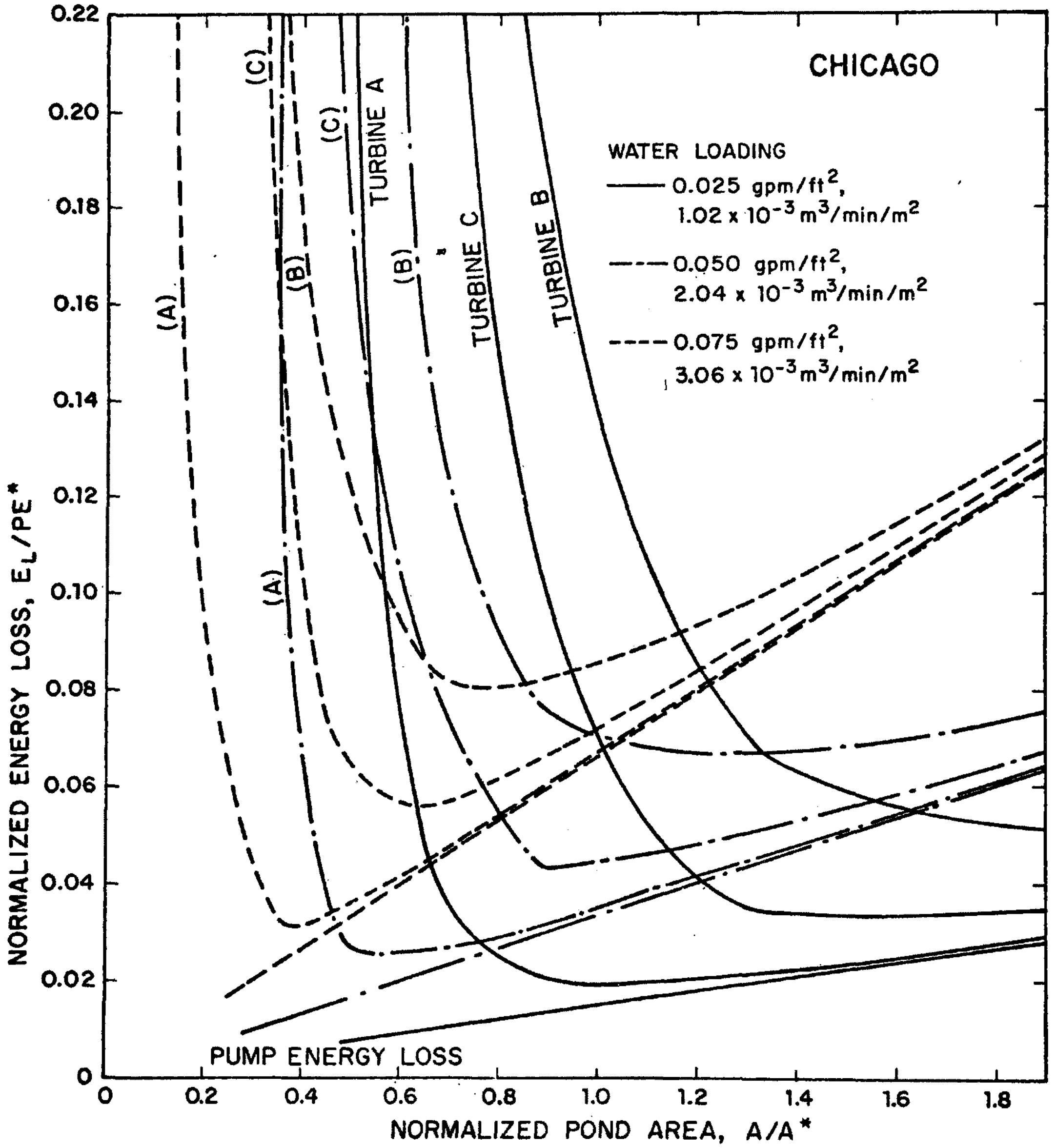

Figure 60(a). Normalized energy loss, Chicago

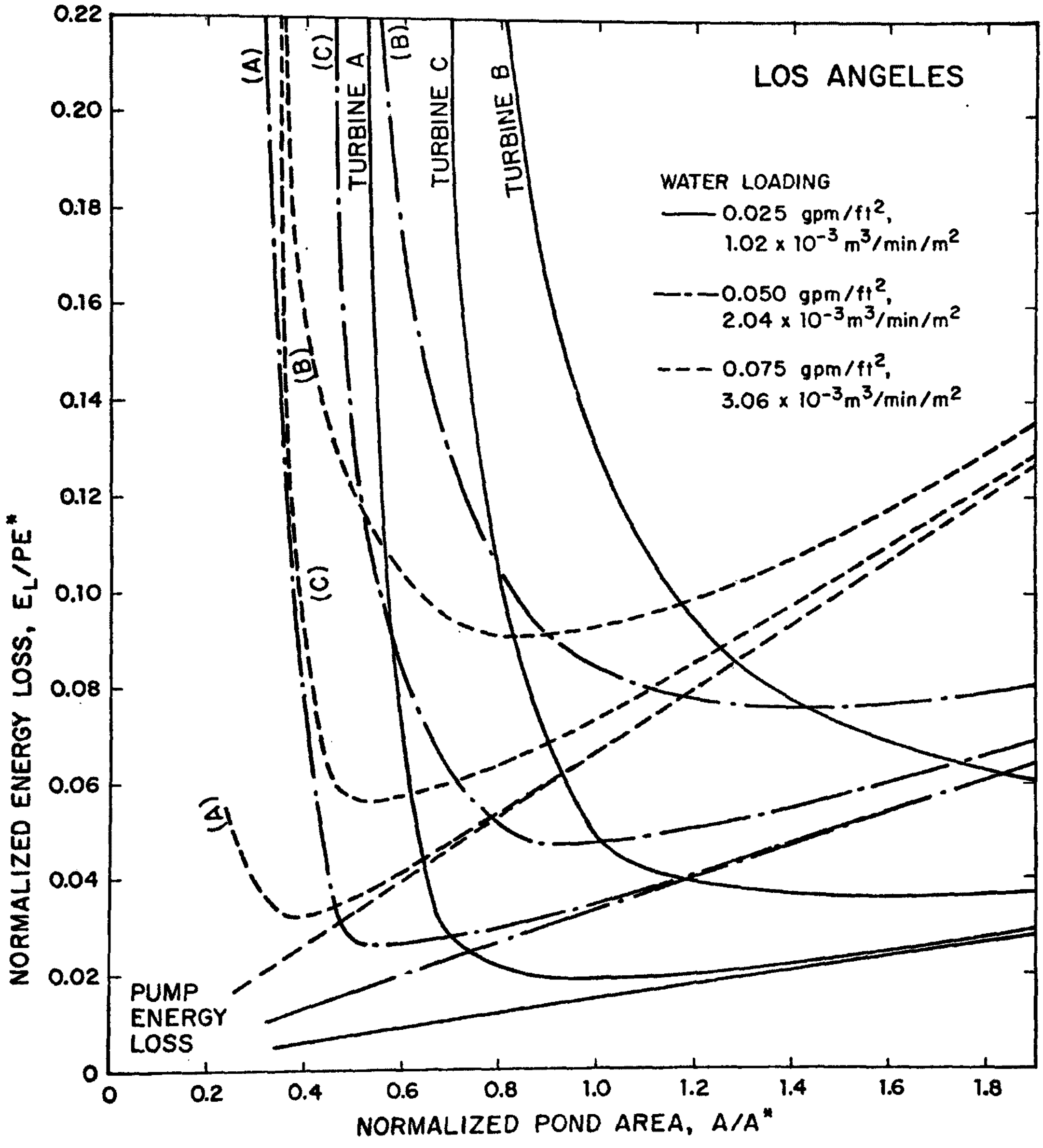

Figure 60(b). Normalized energy loss, Los Angeles

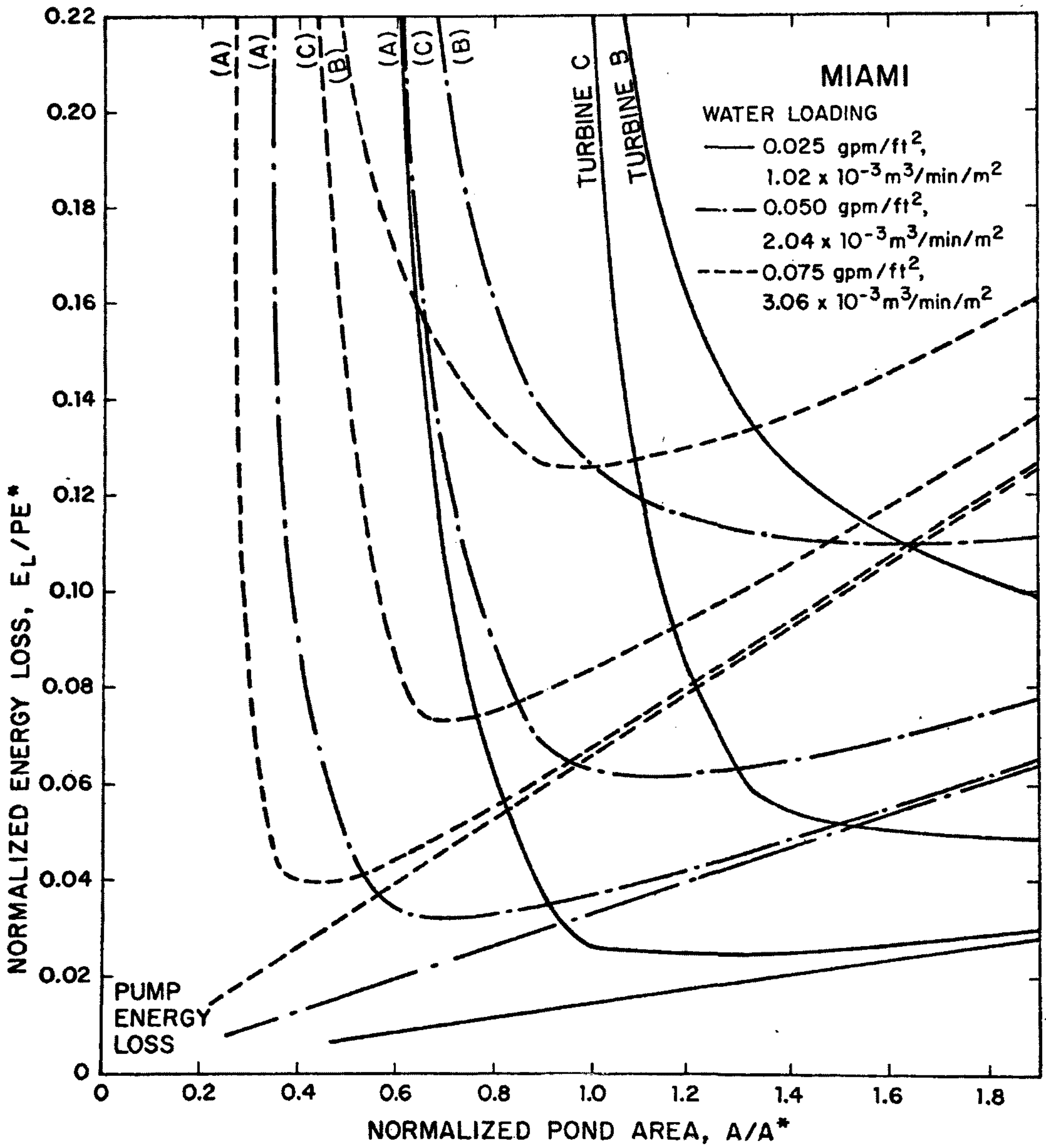

Figure 60(c). Normalized energy loss, Miami

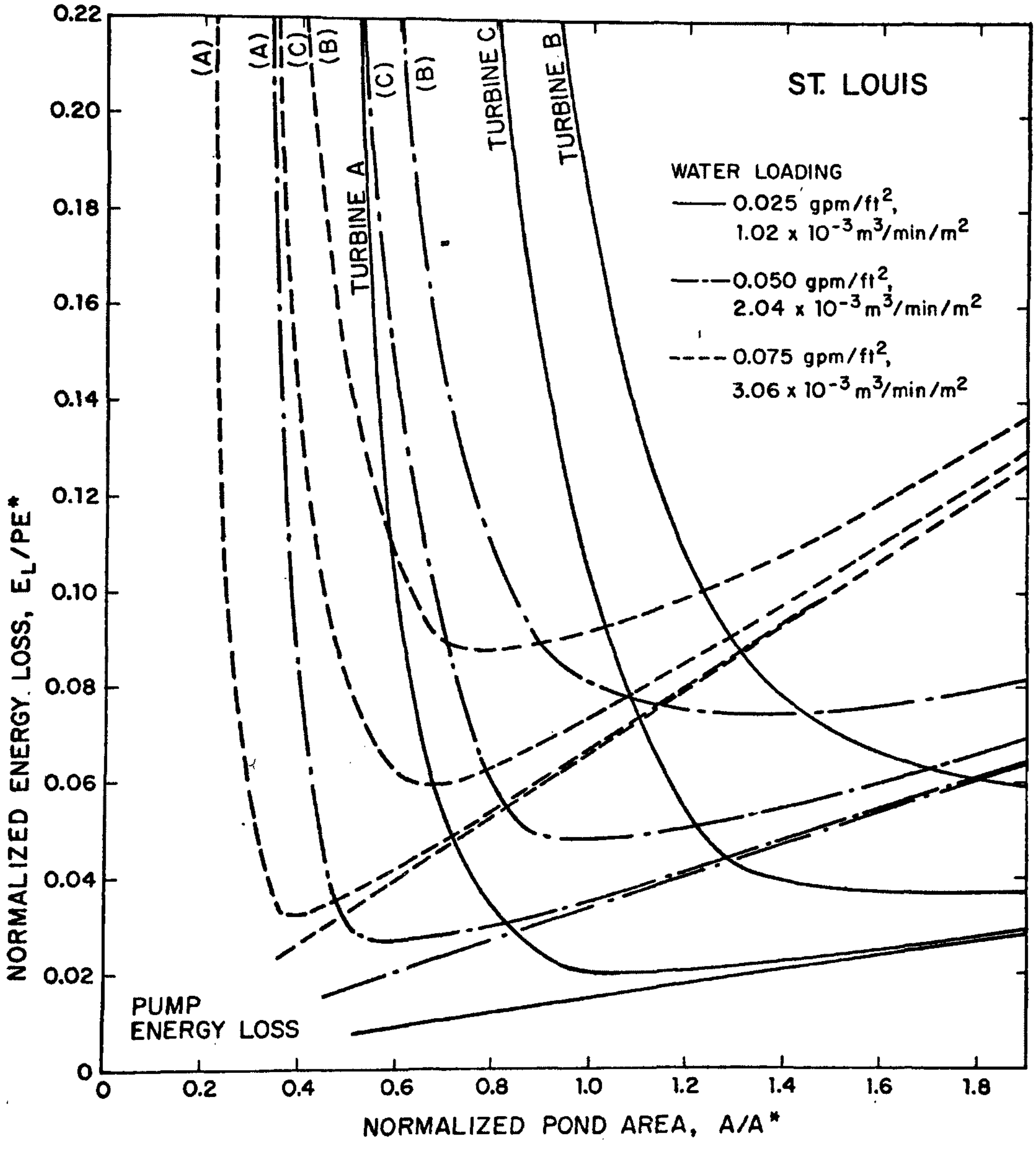

Figure 60(d). Normalized energy loss, St. Louis

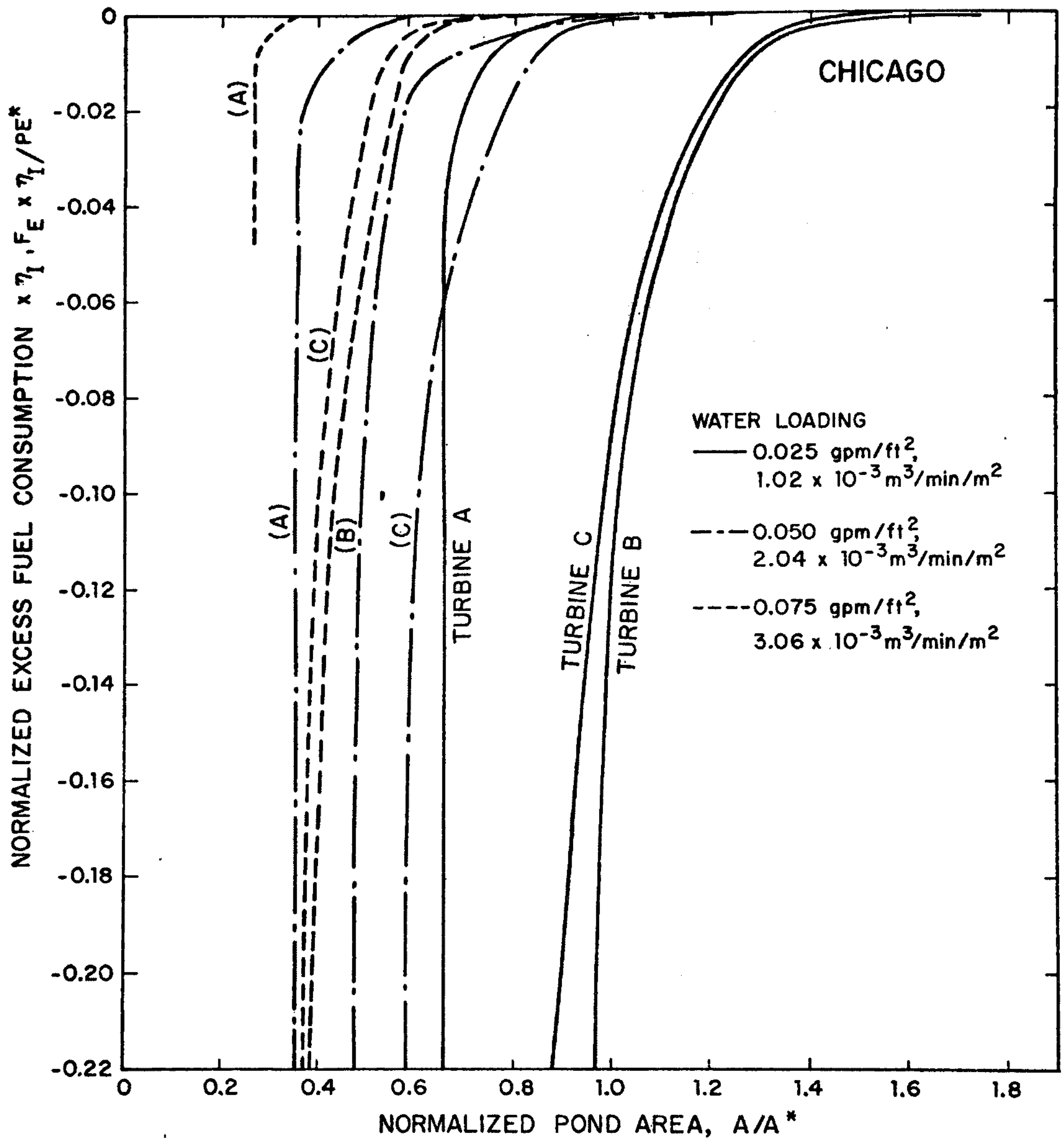

Figure 61(a). Normalized excess fuel consumption, Chicago

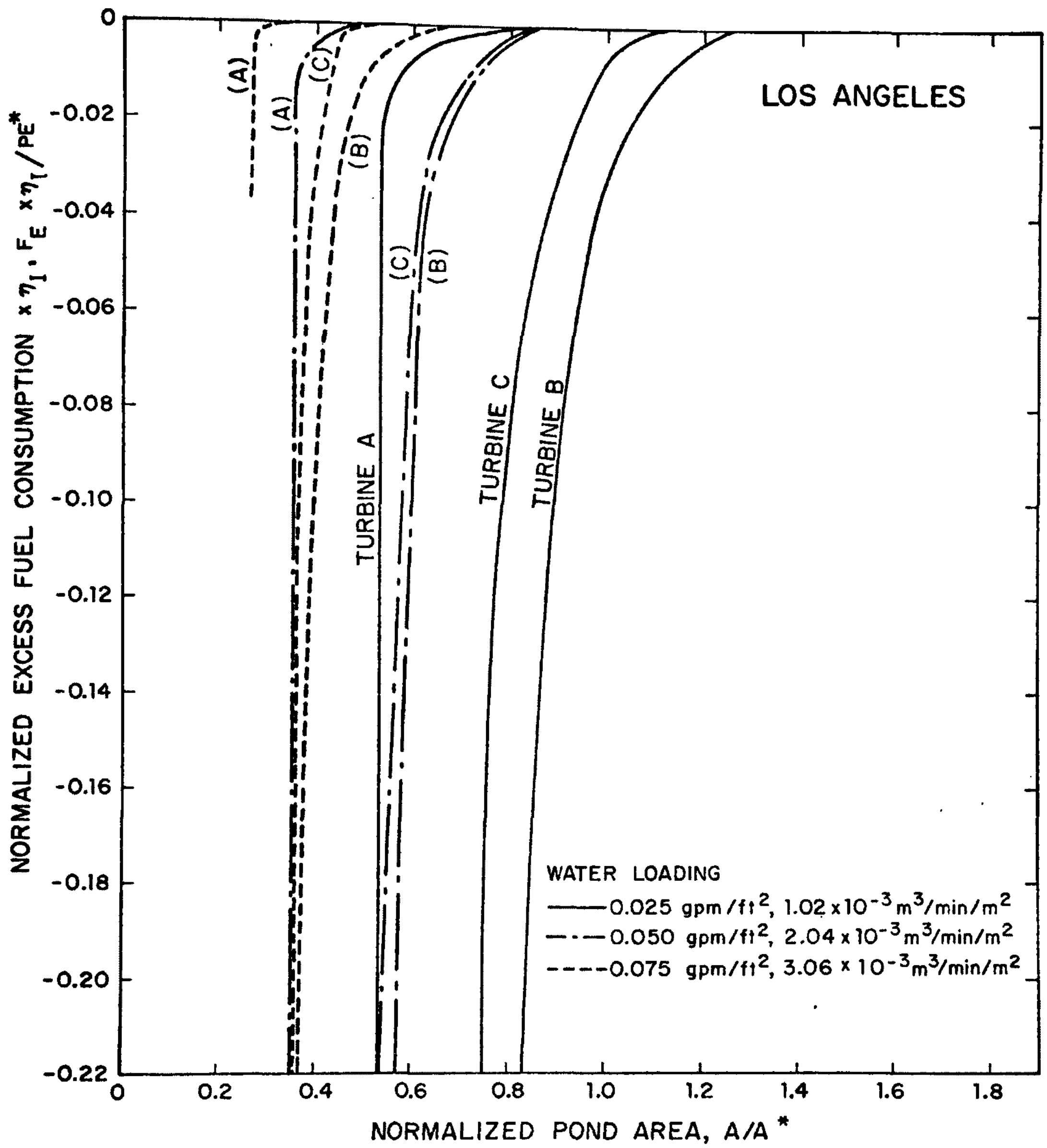

Figure 61(b). Normalized excess fuel consumption, Los Angeles

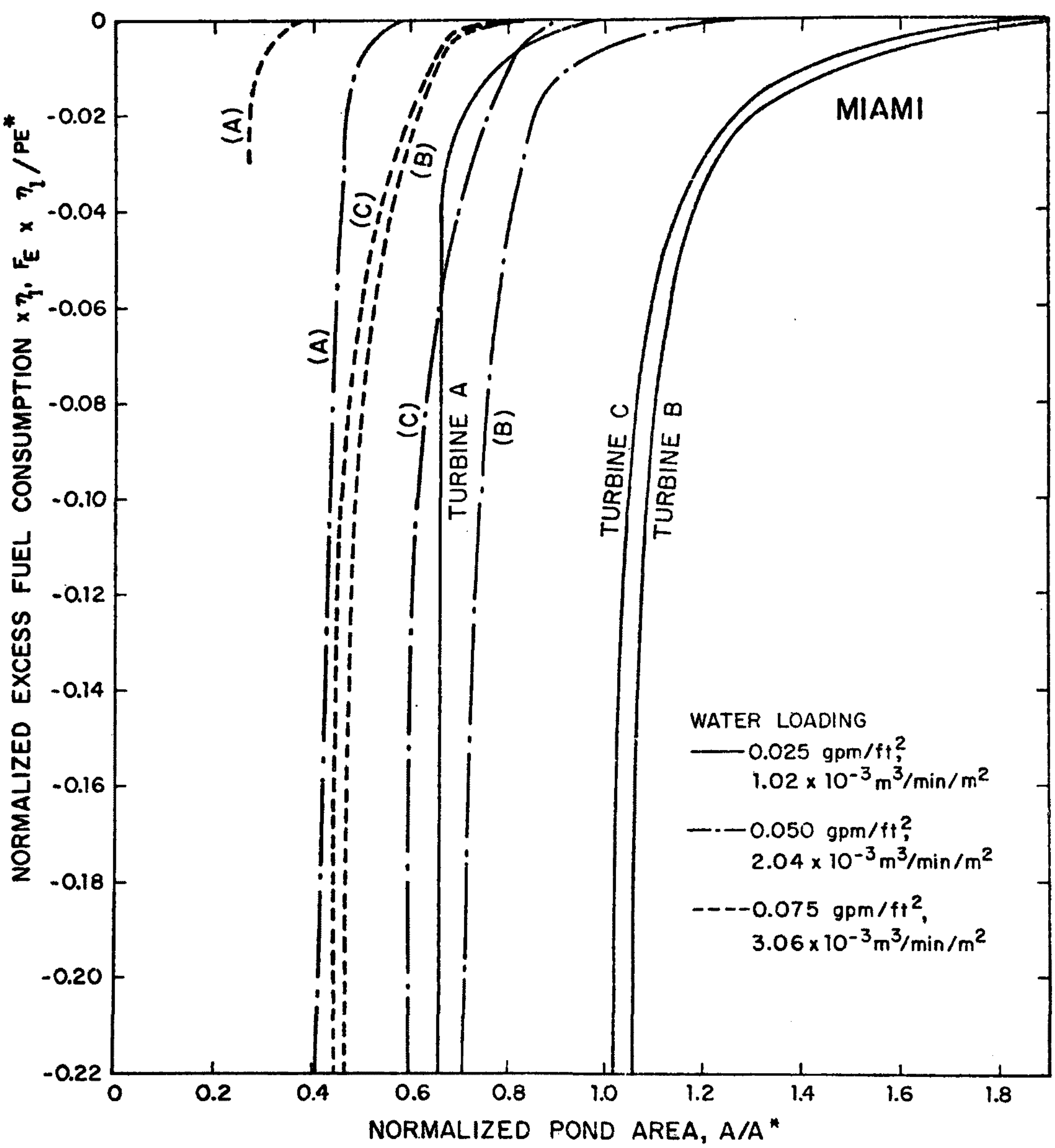

Figure 61(c). Normalized excess fuel consumption, Miami

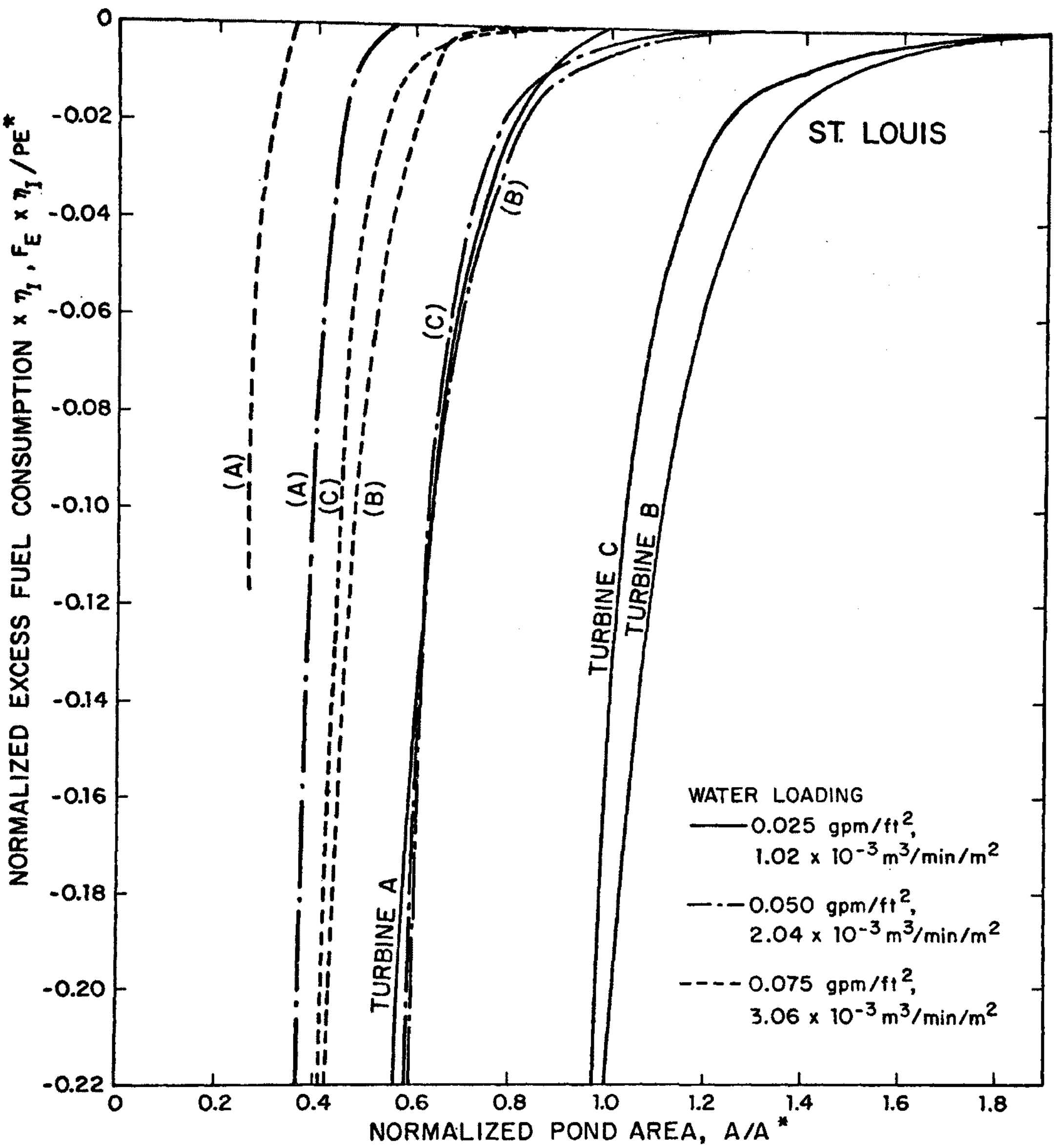

Figure 61(d). Normalized excess fuel consumption, St. Louis

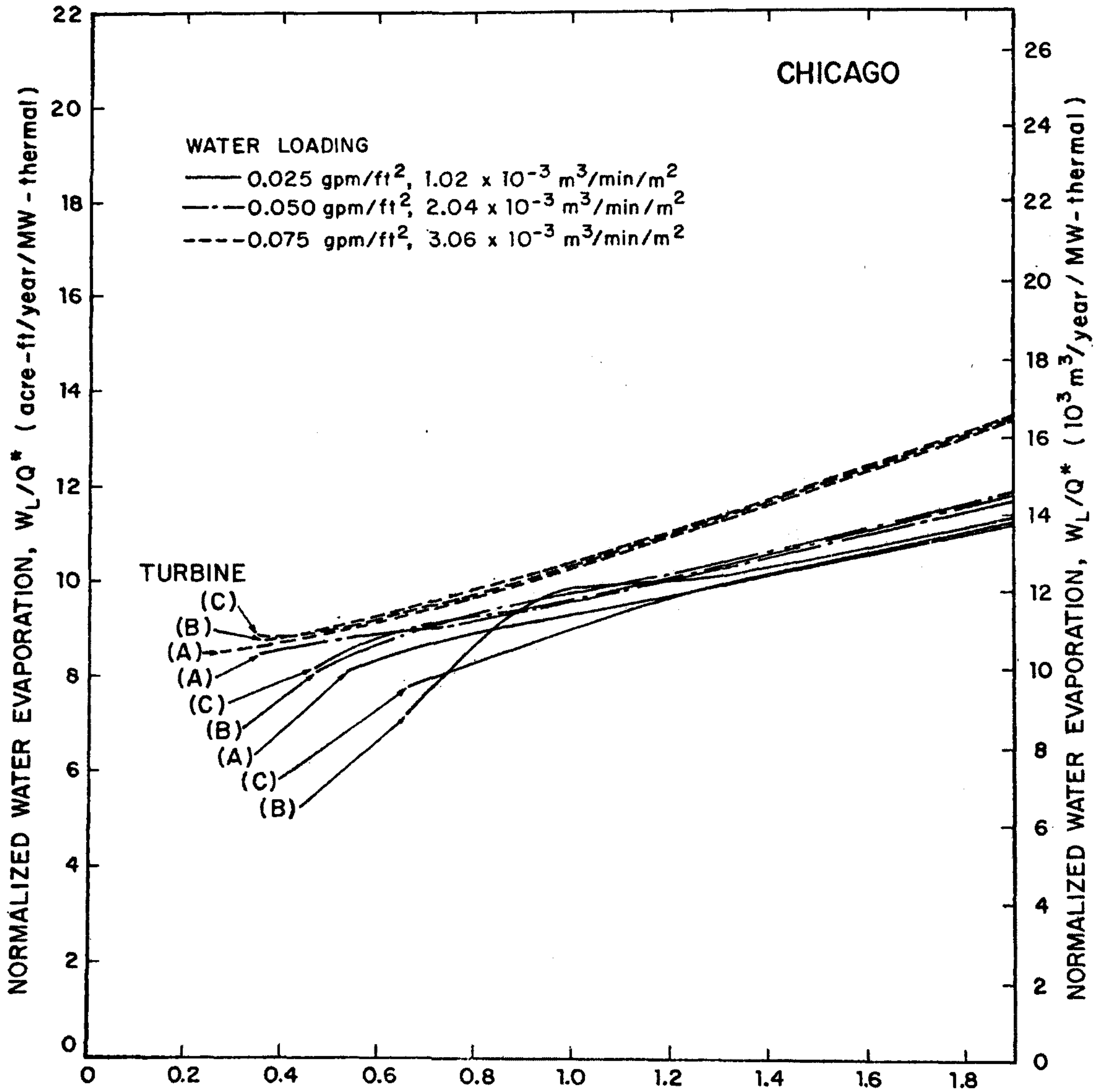

Figure 62(a). Normalized evaporation, Chicago

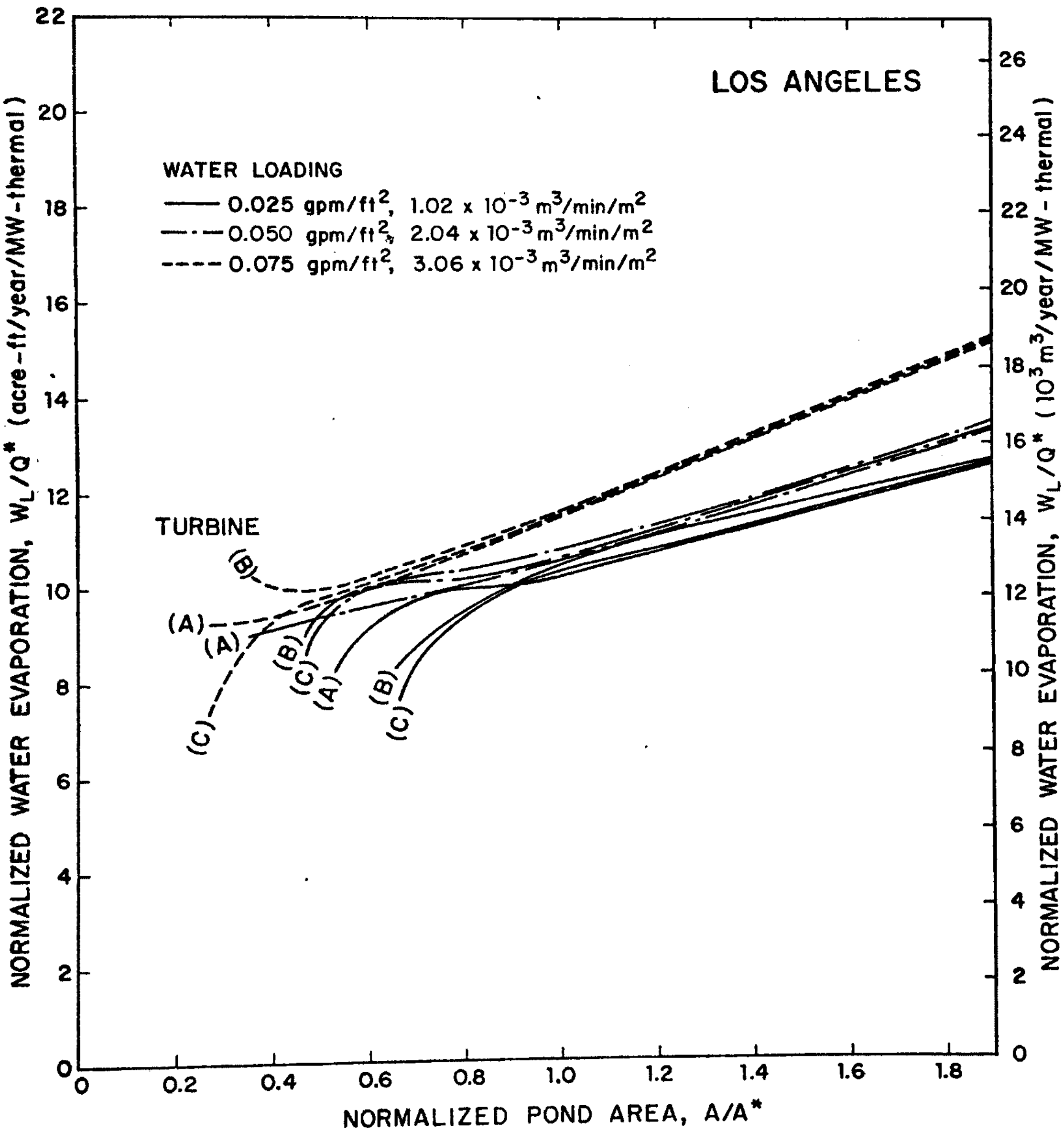

Figure 62(b). Normalized evaporation, Los Angeles

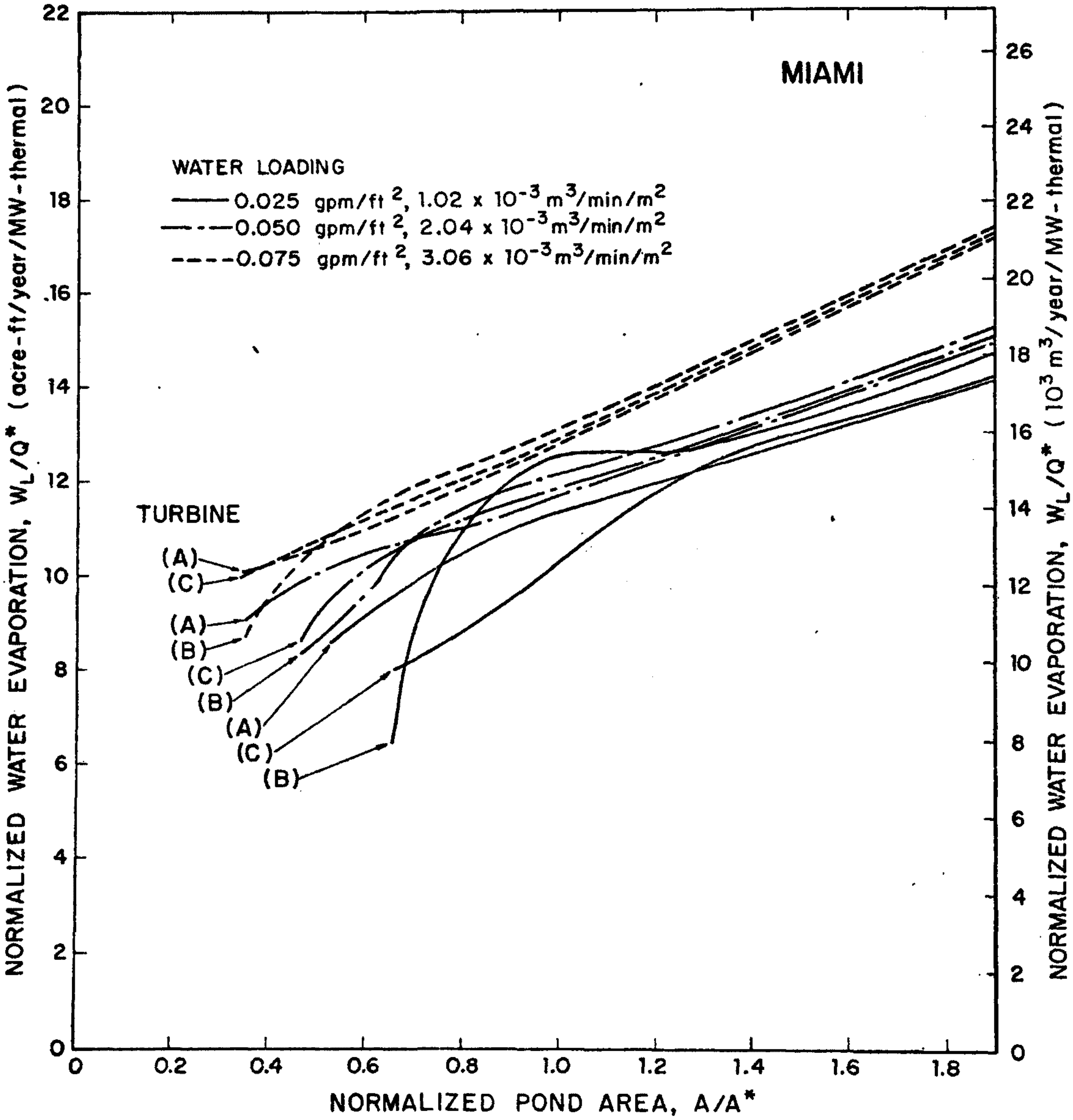

Figure 62(c). Normalized evaporation, Miami

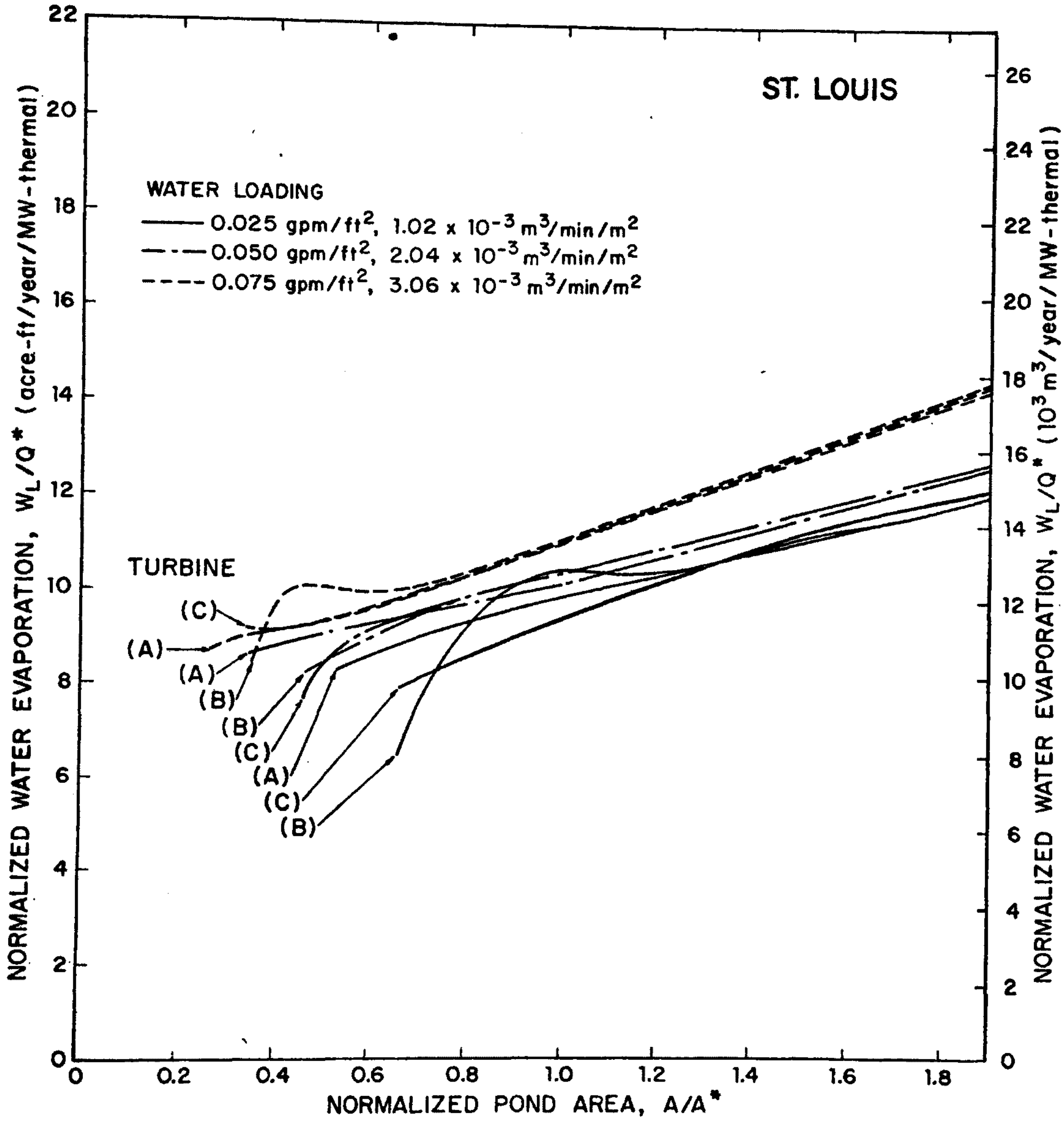

Figure 62(d). Normalized evaporation, St. Louis

quantities with power plant size, the final results for the cooling pond, Figures 59 through 62 are valid for a 411 MW plant only. Therefore, if the graphical results are to be used for other size plants or units, the interpolation technique described for the natural-draft towers (Sections VI.E and VI.H) should be employed. However, the differences in the results for the 411 MW and the 822 MW plants (Figures 55 through 58) are seen to be quite small.

In the sample calculations presented in Section VII.G for a 312.5 MW plant, the curves are read directly, assuming that any errors which are introduced are small. This assumption is checked by comparing the results from the graphical computation with results obtained using the computer program. The comparison is quite acceptable as seen at the close of Section VII.G. It must be emphasized that if the size of the power plant being studied varies greatly from 411 MW, the computer program must be used to obtain accurate results.

E. PROCEDURE FOR THE ECONOMIC EVALUATION OF BACKFITTING

Particular items which must be considered in the economic evaluation of backfitting with a cooling pond have been previously described. Any specific differences in the technique for computing the total cost of backfitting with a cooling pond are presented next, followed by a brief description of the computer program that has been developed. A hypothetical test case is presented in part G to illustrate the use of the graphical results.

The procedure for the economic evaluation of backfitting with a cooling pond is very similar to that already presented for the mechanical-draft cooling tower (Section V.F) and the natural-draft cooling tower (Section VI.F) with the following exceptions. In items (b) and (c) of those Sections, the following quantities must be identified for cooling ponds:

(b) Cooling pond

1. The size of the cooling pond in terms of its area, A;

2. Unit cost of the cooling pond including land, pond preparation and construction, c_p (\$/acre or \$/hectare);

(c) Economic parameters:

6. Unit cost of pump and pipe system [26], c_{pp}

Once this information has been gathered, the total differential cost of backfitting with a cooling pond can be calculated by the computer program or estimated by the graphical results presented in Section VII.D. Use of the graphical results is subject to the limitation discussed at the end of Section VII.D. Illustration of the use of the graphical results is presented in Section VII.G.

F. THE COMPUTER PROGRAM

The computer program which accepts any set of numerical values for the various parameters and performs the calculations previously described is listed in Appendix V.

The computer program consists of the MAIN program and five subroutines, namely OPECOS, MODELW, COOL, MIX, and POWERS. The main program reads all inputs, calculates the overall capital and total costs, and controls the printout of these quantities. The inputs along with the symbols and units used are listed in Appendix II. The primary functions of the various subroutines not previously defined are as follows:

COOL: This subroutine contains the iterative method for computing the cold-water (surface) temperature of the cooling pond from the heat balance, equation (37). The subroutine is independent of the particular mathematical model which is employed to predict the cooling performance of the pond, and it accepts information which is transferred from subroutine, FIX.

FIX: The thermodynamic (mathematical) model of the cooling pond is found in this subroutine. In the present study, the completely mixed, steady state, shallow cooling pond model is used, as described in Section VII.A.

The overall program logic is similar to that already explained for the cooling tower economic calculations. Minor changes have been made in the main program and subroutines to accomodate the cooling pond model. These changes are shown in the program listing included in Appendix V.

G. A HYPOTHETICAL TEST CASE

1. Consider a power plant with the characteristics identified in Section V.H.1, which also implies that the extreme dry-bulb temperature, $\hat{T}_{db}$ = 97.0°F (36.1°C) for Miami.

2. Assume that this plant is to be backfitted with a cooling pond whose characteristics are:

Pond area, A	= 250 Acres (101.2 hectares)
Water loading, per pond area	= 0.05 gpm/ft^2 (0.002 m^3/min/m^2)
Wind speed, V_2	= 8.0 mph (3.57 m/sec)
Solar Radiation, QSC	= 2800 Btu/ft^2/day (3.8×10^4 kJ/m^2/day)
Total water flow rate, GPM (=0.05 × 250 × 43560)	= 544,500 gpm (2061 m^3/min)
Unit cost of cooling pond (including land cost), c_p	= \$5000/acre (\$12,355/hectare)

Unit maintenance cost, c_m	= \$2.00/acre/year (\$4.94/hectare/year)
Concentration ratio, k*	= 3.3
Unit blowdown treatment cost, c_b	= \$0.00
Cost of hook-up and testing, C_{HT}	= included in cost of pond
Downtime, DT	= 720 hrs (30 days)
Unit cost of pump and pipe system, c_{pp}	= \$1.50/gpm (\$396./m^3/min)

3. It is assumed that the following economic parameters apply to the affected utility. All other characteristics are the same as in Section V.H.3.

Unit cost of water, c_w	= \$0.00
Unit cost of access land, c_a	= \$3000/acre (\$7413/hectare)

4. Use of example results:

(a) Read Figure 54 to find A*	= 260 acres (105.2 hectares)
Determine normalized area $A/A^* = 250/260$	= 0.961
(b) Read Figure 59(c) (turbine A, Miami) to find normalized capacity loss, C_L/P^*	= 0.0402
Thus, capacity loss $C_L = 0.0402 \times 312.5 \times 1000$	= 12562 kW
(c) Read Figure 60(c) (turbine A, Miami) to find normalized energy loss, E_L/PE^*	= 0.0360
Thus, energy loss, $E_L = 0.0360 \times 312.5 \times 1000 \times 8760$	= 98.55×10^6 kW-hr/year
(d) Read Figure 61(c) (turbine A, Miami) to find normalized excess fuel consumption, $\eta_I F_E/PE^*$	= 0.0

Thus, excess fuel

$F_E = 0 \times 312.5 \times 1000 \times 8760/0.85$ = 0 kW-hr/year

(e) Read Figure 62(c) (turbine A, Miami) to find normalized water evaporation, W_L/Q^* = 11.55 acre-ft/yr/MW-th (1.42×10^4 m^3/yr/MW-th)

Thus, evaporation,

$W_L = 11.55 \times 1.912 \times 10^9/3.413 \times 10^6$ = 6470 acre-ft/year (7.98×10^6 m^3/year)

Also, blowdown, $W_b = W_L \frac{1}{k^*-1}$ = 2813 acre-ft/year (3.47×10^6 m^3/year)

and makeup,

$W_m = W_L \frac{k^*}{k^*-1} = W_b + W_L$ = 9283 acre-ft/year (11.45×10^6 m^3/year)

5. Cost determination

<u>Capital costs</u>

Pond cost, $C_{cs} \doteq A \times c_p = 250 \times 5000$	= \$1,250,000
Access land cost $= A_a \times c_a = 25 \times 3000$	= \$75,000
Pump & pipe system cost, $c_{pp} \times GPM = 1.50 \times 544{,}500$	= \$816,750
Pump & pipe system salvage, $C'_{pp} = 0.2\ C_{pp}$	= (\$163,350)
New condensers, C_c	= \$ 0
Salvage value of old condensers, C'_c	= (\$ 0)
Salvage value of other open-cycle components, C'_o	= (\$ 0)
Hook-up and testing cost, C_{HT}	= included in pond cost
Replacement capacity, $CC_R = C_L c_\ell = 12562 \times 90$	= \$1,130,580

Downtime, $CC_{DT} = DT \times P^* \times e'_\ell$
$= 720 \times 312.5 \times 1000 \times 0.007$ = $\underline{\$1,575,000}$

TOTAL CAPITAL COST, CC = $\underline{\$4,683,980}$

Operating costs/year

Excess fuel cost, $OC_{EF} = F_E \, f_c$ = \$ 0

Replacement energy cost,
$OC_R = E_L e_\ell = 98.55 \times 10^6 \times 0.01$ = \$985,500

Supply water cost,
$W_m c_w = 9283 \times 3.259 \times 10^5 \times 0$ = \$ 0

Cost of blowdown treatment,
$W_b c_b = 2813 \times 3.259 \times 10^5 \times 0$ = 0

Maintenance of ponds,
$C_m = c_m \times A = 2.0 \times 250$ = \$500

Makeup water cost with open-cycle system, M' = (\$ 0)

Blowdown treatment cost with open-cycle system, B' = (\$ 0)

Maintenance cost of open-cycle system, C'_m = (\$ 0)

TOTAL ANNUAL OPERATING COST, OC = \$986,000

Total costs

From equation (20), the total excess unit cost due to backfitting, tc, is given by

$$tc = \frac{OC + CC \times FCR}{8760 \times P^*}$$

$$= \frac{986{,}000 + (4{,}683{,}980 \times 0.179)}{312.5 \times 1000 \times 8760}$$

$$\boxed{tc = 0.6665 \text{ mills/kW-hr}}$$

To check the assumption made at the end of Section VII.D, i.e., that the errors introduced by reading Figures 59 through 62 directly for the 312.5 MW plant are small, this example was also analyzed using the computer program. The computer results are CC = \$4,583,027 ; OC = \$927,333/yr; tc = 0.6384 mills/kW-hr. The difference between the graphical result and the computer result for the total excess unit cost is seen to be approximately 4.4%, indicating that the graphical method does indeed yield a good approximation for the given problem.

SECTION VIII

SPRAY CANALS AND PONDS

For applications where cooling towers are not desired and where land is not available for cooling ponds, another alternative which could be considered for a closed-cycle cooling system is the use of spray cooling. Such a system, simply described, consists of an array of nozzles or other devices which spray the cooling water directly into the air where both evaporative and sensible heat transfer take place. The cooled water is then collected for recirculation through the power plant condensing system.

The use of spray cooling may increase the heat transfer per unit surface area by twenty times that of a cooling pond resulting in a significant decrease in the land requirement [39]. The low profile of a spray cooling system generally presents little or no aesthetic disadvantages.

Spray cooling systems are usually arranged in one of several different ways. One method which has been employed for several years is referred to as "conventional" spray cooling [12, 23, 26, 32, 40, 41]. Such a system consists of a fixed array of pipes and spray nozzles located in a small pond which serves mainly as a collecting basin. The hot water is taken directly from the condenser, sprayed once, and the cooler pond water is returned to the condenser. Conventional spray ponds have been employed for relatively small scale applications, and they are usually designed for a 10°-15°F (5.6°-8.3°C) cooling range and for about 10°F (5.6°C) approach to the wet bulb temperature [12].

A second technique, employing what may be referred to as a "parallel pass" concept, is also in current use [32, 42, 43]. In this type of cooling system, the water is sprayed from a hot-water delivery canal to a cold-water receiving canal. Such a system may also be designed for multiple parallel passes by including one or more intermediate canals between the hot-water and the cold-water canals [39]. For this type of cooling system, devices other than spray nozzles are often used to propel the water. One manufacturer supplies a spray cooling system which utilizes a series of rotating discs mounted on a common shaft to spray the water [43].

A third possibility for spray cooling is the use of the "series concept" in which spray devices are arranged along a canal [39, 42, 44, 45, 46]. The condenser discharges hot water into the canal, and the canal water is sprayed into the air many times as it moves through the canal. The amount of heat which is dissipated to the atmosphere depends upon the number of times the water is sprayed, and the length of the canal. This technique has the capability of yielding a closer approach to the ambient wet-bulb temperature than the other methods [42].

Spray canals utilizing the series concept appear to be the most popular method of spray cooling in new installations and particularly for large power plants [39, 42]. A recent innovation in spray cooling is the floating (powered) spray module [46, 47, 48]. Spray modules usually consist of from one to twelve nozzles arranged as a floating, self-contained system with a pump. Dependent upon the particular design, the pump power may range from 20 hp (14.9 kW) to 100 hp (74.6 kW), and the pump discharge may vary from 2600 gpm (9.84 m^3/min) to 12000 gpm (45.4 m^3/min) [42, 47, 48, 49]. Each module is serviced by its own pump and is independently deployed along the canal, being moored and electrically connected to shore.

The floating spray modules take in water from just below the float level to a depth of approximately 3 ft (0.91 m), and the spray pattern produced may vary from 20 to 50 ft (6.1 to 15.2 m) in diameter and from 10 to 20 ft (3.05 to 6.1 m) high. These ranges are, again, dependent upon the particular design of the module [47, 48, 49].

One set of module characteristics is chosen for the present analysis. This set of characteristics reflects units commonly in use and conditions most likely to be faced in backfitting applications. The three major manufacturers of powered spray modules can each supply units matching the size which is considered in this study [47, 48, 49].

A. OPERATION MODELS OF SPRAY COOLING

The thermodynamic analysis of a spray canal for a particular application includes many site-dependent variables. It is not possible to include all of these variables in the present study, and certain features of the spray cooling system are held fixed with the resultant limitations on the analysis. The particular spray modules chosen for the current investigation utilize a 75 hp (55.9 kW) pump and have a discharge rate of 10000 gpm (37.8 m^3/min). In all cases, the spray canal is assumed to be straight and oriented perpendicularly to the direction of the prevailing summer wind. The canal cross-section is trapezoidal with side slopes of 3-horizontal to 1-vertical, and the depth is 10 ft (3.05 m). The canal top-width depends upon the number of rows of spray modules mounted across the canal, i.e., the number of modules per group. Likewise, the length of the canal depends upon the number of module groups deployed along it. The depth of water flowing in the canal is held constant at 8 ft (2.44 m).

The spacing between adjacent modules in both the streamwise and transverse directions is held constant according to the following plan. The modules are considered to be arranged along the canal on lines which are parallel to the canal sides. The centerline of the first module is located at a distance of 100 ft (30.5 m) from the upstream end of the canal. Adjacent modules along the canal are separated by a distance of 100 ft (30.5 m), center-to-center, and the downstream end of the canal is also 100 ft (30.5 m) from the center of the last module. For modules arranged in multiple rows across the canal, the center-to-center distance between rows is 75 ft (22.9 m), and the canal shoreline on either side of the modules is located at a distance of 50 ft (15.2 m) from the center of the outboard rows. The rows of modules are along lines which are perpendicular to the canal.

As mentioned, the canal geometry and the module layout are fixed for the present analysis. However, there is enough built-in flexibility for this set-up to handle many different module arrangements. See, for example [47, 48]. It should be mentioned that the canal geometry and module spacing as described determine, mainly, land requirements and canal construction costs which are considered in detail in Section VIII.B.

With the spray module characteristics and the canal geometry chosen, the thermodynamic analysis of spray canal cooling depends upon the mathematical simulation which is employed. There are three basic types of models for predicting the thermal performance of spray cooling systems. These models are listed by Ryan [39] as the manufacturer's model, the NTU model, and the cellular model.

The manufacturer's models are usually based on the measured performance of a single nozzle. Various performance curves and correction factors, all proprietary information, have been developed and are used by the manufacturers to size and fit given cooling applications. The information on system performance which is available to the public may be useful for simplified performance checks [47, 48].

The cellular model, which assumes that the spray field is made up of a number of identical droplets each surrounded by a cell of air, was originally developed for the analysis of conventional spray cooling [50]. This model may not be useful for modular spray cooling due to large droplet size in the sprays, but certain variations of the basic model are being considered [39].

The NTU (number of transfer units) model is similar to the theory based on the Merkel equation which is commonly used to describe the heat transfer occurring in evaporative cooling towers [44, 46]. The basis of this model is carefully explained by Chen [51] and summarized by Porter [52] and Porter and Chen [53]. The NTU method is used in the current investigation following their presentation.

The Merkel equation may be written as

$$\xi \int_{T_C}^{T_H} \frac{dT}{h(T)-h_a} = \overline{(K_c\ A_d/m_d)t_d} \tag{48}$$

in which ξ is the specific heat of liquid water at constant pressure; T_C and T_H are the temperatures of the cold and hot water, respectively; $h(T)$ and h_a are the total heats (sigma functions) of the water at temperature T, and of the air-vapor mixture, respectively; K_c is the effective droplet convective heat transfer coefficient; A_d is the droplet surface area; m_d is the mass of the droplet, and t_d is the time of flight of the water droplets. The overbar indicates average values over the time of flight of the spray which integrates the complex dynamical effect into a single parameter called the number of transfer units (NTU) [52].

If it is assumed that h_a may be approximated by its average value (constant) at the local wet-bulb temperature, $T_{wb\ell}$, and that $dh/dT=b$ is constant over the integration, NTU may be replaced by an average value, $\overline{NTU}$, which may be approximated by

$$\overline{NTU} \simeq (\xi/b_f)\ \ln\left[\frac{T_H - T_{wb\ell}}{T_C - T_{wb\ell}}\right] \tag{49}$$

Herein, b_f is the constant, b, evaluated at the film temperature, T_f, which is estimated as the average of the hot-water temperature and the local wet-bulb temperature [52]; i.e.,

$$T_f = (T_H + T_{wb\ell})/2 \tag{50}$$

Equation (49) may be inverted to yield a relationship for the cooling range of a single module,

$$T_H - T_C = (T_H - T_{wb\ell})\ (1-\exp[-\overline{NTU}(b_f/\xi)]) \tag{51}$$

For a known hot-water temperature entering a module, the temperature of the cold water being returned to the canal can be computed from this equation if the local wet-bulb temperature, $T_{wb\ell}$, the constant, b_f, and the average number of transfer units, $\overline{NTU}$, are known. The determination of these quantities is discussed in the following paragraphs.

For the proper assessment of spray cooling, it is quite important to consider the effects of the sprays upon the local psychrometric conditions. The presence of hot-water sprays will increase the air temperature and humidity in their vicinity. Therefore, if spray modules are placed close together, these interference effects upon the downwind units which cause a decrease in cooling performance must be considered. Since the wet-bulb temperature of the air is driven toward the local canal temperature, Porter [52] and Porter and Chen [53] suggest a correction in the local wet-bulb temperature of the form

$$T_{wb\ell} = T_{wb} - F_w(T_H - T_{wb}) \tag{52}$$

where T_{wb} is the ambient wet-bulb temperature and F_w is the wet-bulb correction factor. The correction factor, which varies from zero to one, is an experimentally determined function of distance downwind from the center of individual spray modules. A set of average values of F_w is given by Porter [52] for different numbers of rows of modules across the canal and for various row separation distances. These values were determined from field measurements of two types of modules operating at different flow rates and module configurations. It is noted by Porter [52] that these correction factors should be regarded as tentative due to the limited amount of verification. The maximum row separation distance reported in Porter's table is 60 ft (18.3 m), and those values of the wet-bulb correction factor are used in the present study for a row separation distance of 75 ft (22.9 m) resulting in a slightly conservative correction. The local wet-bulb temperature needed in equation (51) is then computed from equation (52) using the wet-bulb correction factor from Porter's table [52].

The constant, b_f, which is used in equation (51) is the rate of change of the total heat of the air-vapor mixture with respect to wet-bulb temperature evaluated at the film temperature, T_f. The dependence of the total heat (sigma function) of air-water vapor mixtures upon wet-bulb temperature is given by Berry [54] as

$$h_a = 0.240\ T_{db} + w_s (h_v - T_{wb} + 32) \tag{53}$$

where w_s is the specific humidity, and h_v, the enthalpy of the vapor is expressed in terms of the dry-bulb temperature as

$$h_v = 1061.8 + 0.44 T_{db} \tag{54}$$

Equations (53) and (54) may be combined yielding

$$h_a = (0.240 + 0.44 w_s) T_{db} + w_s (1093.8 - T_{wb}) \tag{55}$$

The specific humidity, w_s, is given in terms of the atmosphereic pressure, p_A, and the vapor pressure of the air, e_A, by [54]

$$w_s = e_A/1.608(p_A - e_A) \tag{56}$$

and the vapor pressure is expressed in terms of its saturation value, e_s, by [15]

$$e_A = e_s - 0.000367\ p_A[T_{db} - T_{wb}][1 + (T_{wb} - 32)/1571] \tag{57}$$

The saturation value of the vapor pressure is taken from the saturation curve as discussed in [15].

The total heat of the air-vapor mixture at different values of wet-bulb temperature can be computed from equation (55), and b_f can be found from a finite-difference approximation of its definition; i.e.,

$$b_f = \left(\frac{dh_a}{dT_{wb}}\right)_f \approx \left(\frac{\Delta h_a}{\Delta T_{wb}}\right)_f \tag{58}$$

in which the ratio is evaluated at the film temperature.

The average number of transfer units, $\overline{NTU}$, is also needed in equation (51) to compute the cooling range of a single module. It is generally accepted that $\overline{NTU}$ depends primarily upon wind speed [46, 47, 52, 53] although this concept is subject to some controversy [39]. Porter [52] and Porter and Chen [53] have presented an approximate correlation of module $\overline{NTU}$ with wind speed. The values of $\overline{NTU}$ were determined from tests of entire canals by matching their observed performance to the theory [53]. However, Porter [55] has reported that this curve results in values of $\overline{NTU}$ which are too high. In his discussion of the relationship between wind speed and $\overline{NTU}$, Ryan [39] compares the curve of Porter and Chen (as it first appeared in a technical report prior to the publication of [53]) with one constructed from some unpublished data of Hoffman. Hoffman's curve is somewhat lower than that of Porter and Chen. Another curve of $\overline{NTU}$ vs. wind speed presented by a spray module manufacturer [47] is also lower than Porter and Chen's curve. Therefore, for the present study, a relationship between $\overline{NTU}$ and wind speed was constructed by considering the average of the two curves presented by Ryan [39]. The equation of this straight line is

$$\overline{NTU} = 0.036\ V_2 + 0.156 \tag{59}$$

where V_2 is the wind speed in mph. This curve is useful for approximating values of $\overline{NTU}$ to be used in equation (51); however, a better estimate of $\overline{NTU}$ would be necessary for a more accurate assessment of a given situation.

Equation (51) and the material just presented allow the thermodynamic analysis of a single module. This information for a typical module must be included with the flow properties of the canal water in order to investigate the cooling performance of the entire spray canal system. Assuming complete mixing of the canal water with the spray water between passes of modules along the canal, Porter [52] gives the equation to determine the canal cooling performance which is used in the present study, viz.,

$$(T_C - T_{wb})/(T_H - T_{wb}) = \exp\{-N_{tot}\ r_F (1-F_w)(1-\exp[-\overline{NTU}\ b_f/\xi])\} \tag{60}$$

In this equation, N_{tot} is the total number of modules in the canal, and r_F is the ratio of the flow rate per module to the total canal flow rate. It should be noted that $N_{tot} = N \times m$ where N is the number of module groups deployed along the canal, and m is the number of modules per group. As mentioned earlier, each module may contain from one to twelve nozzles. The number of nozzles is not important in the present study, and it should not be confused with N or m.

The amount of spray water evaporated must also be calculated to determine make-up water requirements for the spray cooling system. The fraction of canal flow evaporated is directly proportional to the cooling range and is approximated by Porter [52] as

$$r_E = \xi(T_H - T_C)/[H_v(1+B_o)] \qquad (61)$$

where H_v is the latent heat of vaporization of water (at constant pressure) and B_o is the so-called Bowen ratio, which is the ratio of the sensible to the evaporative heat transfer. In the present study, the latent heat of vaporization is computed by [35]

$$H_v = 1087 - 0.54T_H \text{ (Btu/lb)} \qquad (62)$$

and the Bowen ratio is assumed to be zero, resulting in a conservative estimate of the water evaporated. The total evaporative water loss, W_L, is given by the product of $r_E \times$ GPM.

The thermodynamic model described in the preceding paragraphs of this Section is employed to estimate the cooling effectiveness and water evaporation rate of different size spray canal systems. The operational aspects of this model as it is applied to the backfit situation are described next.

The input variables for the spray canal model include the meteorological conditions, spray module layout, and hot-water flow conditions. The meteorological conditions include the dry-bulb temperature, T_{db}, wet-bulb temperature, T_{wb}, atmospheric pressure, p_A, and wind speed, V_2. The spray module layout is described in terms of the number of module groups (or passes) along the canal, N, the number of modules per group (or the number of rows of modules) across the canal, m, and the spacing between adjacent modules. The input variables related to the hot water at the canal inlet are the hot-water temperature, T_H, and the total canal flow rate, GPM. It should be noted that the canal flow rate is specified by the reference heat rejection rate, Q^*, if a particular design cooling range is predetermined; i.e.,

$$GPM = c_3 Q^*/\xi RC \tag{63}$$

where GPM is the canal flow rate in gpm; Q^* is the reference heat rejection rate in Btu/hr, ξ is the specific heat at constant pressure in Btu/lb-^{o}F, and RC is the cooling range in degree F. The constant $c_3 = 7.481/(60 \times 62)$ is the numerical factor converting lb/hr to gal/min.

The wet-bulb correction factor is chosen, based on the geometric information, and it is used together with the hot-water and ambient wet-bulb temperatures to compute the local wet-bulb temperature by equation (52). Next, the wind speed is used in equation (59) to estimate $\overline{NTU}$. The psychrometric variables are then combined according to equation (55) through equation (58) to obtain the value of b_f, and the cold-water temperature can be computed from equation (60) for the known ratio of module flow rate to canal flow rate for any number of modules. With the cold-water temperature thus computed, the fraction of water lost due to evaporation is found from equation (61), and the total evaporative water loss can be calculated.

The outputs from the spray cooling model are the temperature of the cold water being returned to the condenser and the total water loss due to evaporation. This information is utilized together with the condenser and other power plant parameters in performing the economic analysis.

B. CAPITAL COSTS OF SPRAY CANALS

The most significant components of the capital costs for a spray canal are the costs of the modules, the cost of the canal construction (excavation and canal lining), and the cost of the pump and piping system related to transporting the hot water from the condenser to the canal and returning the cold water to the condenser.

Capital costs of spray modules are, of course, dependent upon the particular model design and manufacturer which supplies the units. Unit prices may be quoted in terms of \$/hp or directly in \$/unit. On the basis of private communication with two manufacturers [56, 57] and a published cost estimate [49], a unit cost figure which lies between the highest and lowest cost estimates has been chosen. It is, however, not entirely clear what these costs include regarding auxiliary equipment; therefore, for the present study, a unit capital cost, c_s = \$22,500 per module (1974 estimate) including mooring and electrical equipment is used.

The cost of canal construction is another major component of the capital costs. The major items contributing to the cost of canal construction are the excavation and lining costs as well as the cost of the land itself. The length and top-width of the canal are based upon the number, layout, and spacing of the modules as described in the previous Section. The required land area is taken as 2.5 times the plan area of the canal. The additional land area is necessary for access roads, electrical service facilities, and mooring facilities.

The cost of land varies over a wide range, as mentioned earlier, and for this study, it is taken as a_ℓ = \$3000 / acre (\$7413/hectare). The cost of canal excavation is determined from the volume of earth removed in building the canal. In the present study a unit excavation cost, c_E = \$2.50/yd^3 (\$3.27/m^3) is used. The cross-sectional shape of the canal is held fixed as described in Section VIII.A, and the canal volume is easily computed when the length and top-width are known.

Canal lining costs depend upon the type of lining being used and vary over a wide range. Minimum lining costs are obviously incurred for an unlined canal, and the most expensive canal lining in common use is concrete [47]. There are several other possibilities for canal linings which offer a choice between higher capital cost and a decreased durability. The canal lining for the present study is taken as concrete, 6 in. (0.15 m) thick. The lining cost is based upon a unit concrete cost of $\$50/yd^3$ ($\$65.40/m^3$) which includes the canal construction costs. For a canal lining of constant thickness, unit costs are usually expressed in terms of the lining area. Expressing the concrete lining cost for the given cross-sectional shape in terms of the lining area results in a unit lining cost, $c_L = \$0.93/ft^2$ ($\$10/m^2$). These 1974 estimates for excavation and concrete were obtained from local contractors, and it is assumed that the total estimate of canal construction costs is representative of actual charges.

The capital cost of the pump and piping system which circulates the condenser cooling water is based upon the canal flow rate and the total pumping head. This cost may be determined from the chart or formula given by Jedlicka [26, p. N-43 or Eq. (30), p. 64]. Assuming a pumping head of 40 ft (12.2 m), the unit cost of the pump and piping system, c_{pp}, (including installation) is seen to be approximately \$1.50/gpm ($\$396/m^3$/min) based on 1970 estimates. Annual maintenance cost for the spray canal system is taken as 1% of the pump and module operating cost.

C. REFERENCE SIZE OF SPRAY CANALS, N*

The reference size for a spray canal, N*, is defined as the number of groups of modules along the canal, N, required to reject Q* while maintaining the turbine back pressure at p' and delivering P* at the reference meteorological conditions which may be arbitrarily selected. In the present study, spray canals are investigated with either one or four modules per group; therefore, for each value of Q*, two reference sizes are needed. The reference meteorological conditions for the spray canal study are: wet-bulb temperature, 68°F (20.0°C); dry-bulb temperature, 78°F (25.6°C), and wind speed, 8 mph (3.57 m/sec).

The reference spray canal size corresponding to a particular heat rejection rate (Figure 63) is found by first computing and plotting the turbine back pressure, p, versus N, for a range of heat rejection rates, Q, at the reference meteorological conditions. These computations are carried out for various values of m, the number of modules per group. The canal flow rate is determined from equation (63) (for each value of Q), and the thermodynamic model for spray cooling is combined with the turbine heat rejection rate characteristics to obtain the desired information, p vs. N.

Data for Figure 63 (Q vs. N) are read from these curves at the value of turbine back pressure selected for defining the reference size of the cooling system, viz., p' = 2.0 in. Hg (5.08 cm Hg). One curve is drawn for each of the two cases m = 1 and m = 4. It is seen that the curves are linear over the entire range which is considered. Figure 63 defines the reference spray canal size, N*, corresponding to the heat rejection rate, Q*, which is used to nondimensionalize the size of the spray canal.

D. PARAMETRIC STUDIES

A series of detailed parametric studies are made for the spray cooling system in a manner paralleling the studies of the other cooling systems presented in this report. The calculations are carried out for a wide range of canal sizes, defined by N, for the two cases, m = 1 and m = 4 module rows across the canal. The size of the spray cooling system depends only on N and m because the spray module size and capacity is held fixed as described in the introduction to this Section.

Results of detailed calculations of maximum capacity loss, C_L, annual energy loss, E_L, annual excess fuel consumption, F_E, and annual water loss by evaporation, W_L, are presented for a large range of canal sizes. The canal size is normalized by N* as described in Section VIII.C, and the other variables are normalized with the same reference

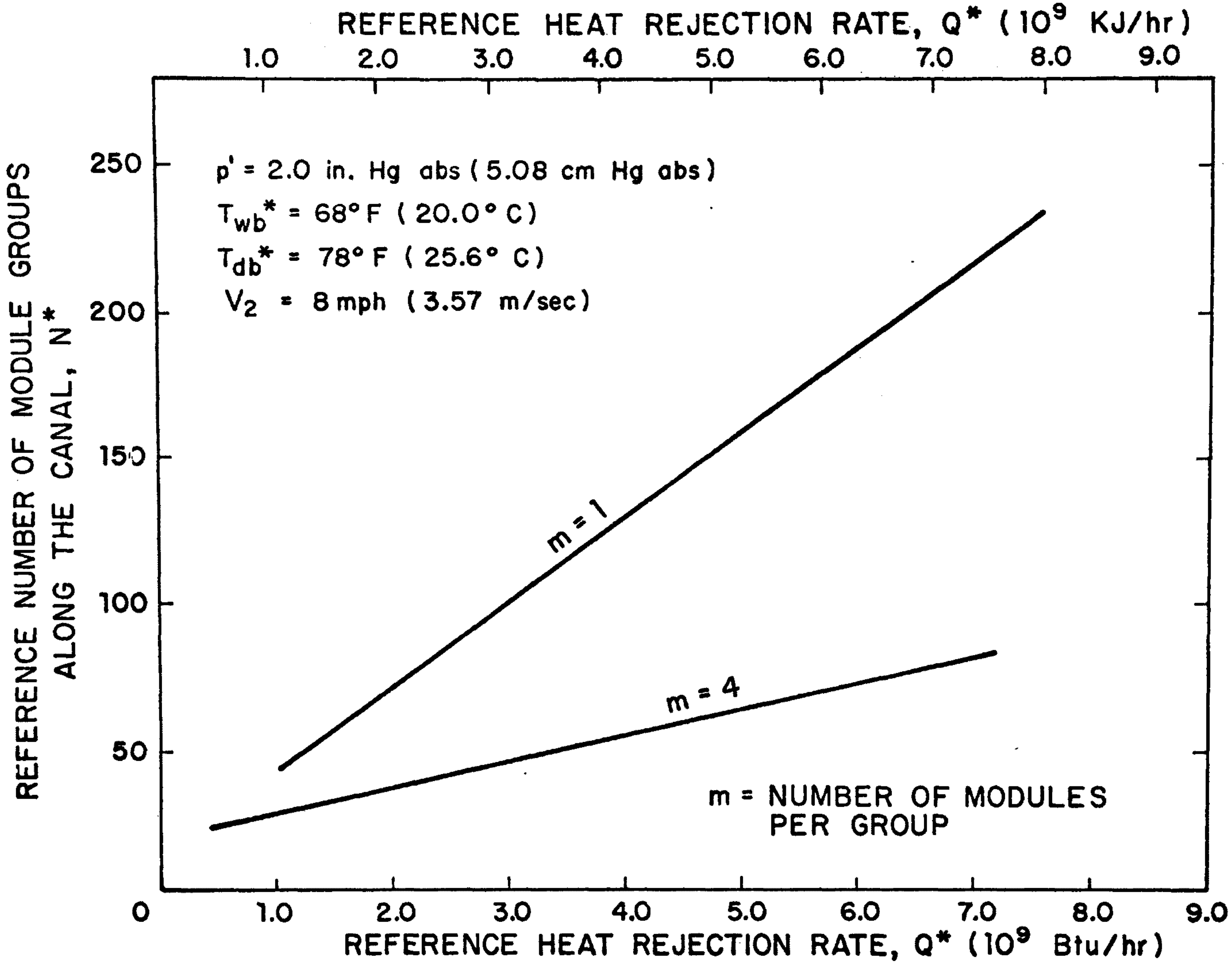

Figure 63. Reference size of spray canals

quantities as the three cooling systems described previously. Four sets of meteorological conditions are chosen as representative of four geographical locations in this country, viz., Chicago, Los Angeles, Miami, and St. Louis. All the studies of spray cooling are carried out for a wind speed of 8 mph (3.57 m/sec), and the other meteorological conditions at each location are described in Section IV.B.

Figures 64 through 67 are plots of normalized capacity loss, normalized energy loss, normalized excess fuel consumption, and normalized water loss by evaporation vs. normalized canal size for the particular case of turbine A (411 MW), at Los Angeles with $m = 1$ and $m = 4$. Also shown in these figures are the results for a hypothetical turbine which has a nameplate capacity and reference heat rejection rate twice those of turbine A, but whose basic heat rate characteristics are the same as those of turbine A. (See Section V.D or VI.E for details.) As for the natural-draft cooling tower and the cooling pond, the normalized results for the two turbine sizes did not collapse indicating a certain scale effect as discussed in Section VI.E.

Detailed studies of the dependence of the quantities listed above upon spray canal size for turbines A, B, and C (see Figures 4, 5, and 6 and Table 1) at each of the four geographical locations are presented in Figures 68 through 71. The turbine designs represent a wide range of practical applications for which the plotted results may be applied. But, because of the scale effects, the final results are valid for a 411 MW unit only. If the graphical results are to be used for other sizes of turbines, the interpolation technique described for the natural-draft towers (Sections VI.E and VI.H) should be employed. However, over most of the range of practical application, the differences in the results for the 411 MW and the 822 MW units (Figure 64 through 67) are seen to be small.

In the sample calculations presented in Section VIII.G for a 312.5 MW unit, the curves are read directly, assuming that any errors which are

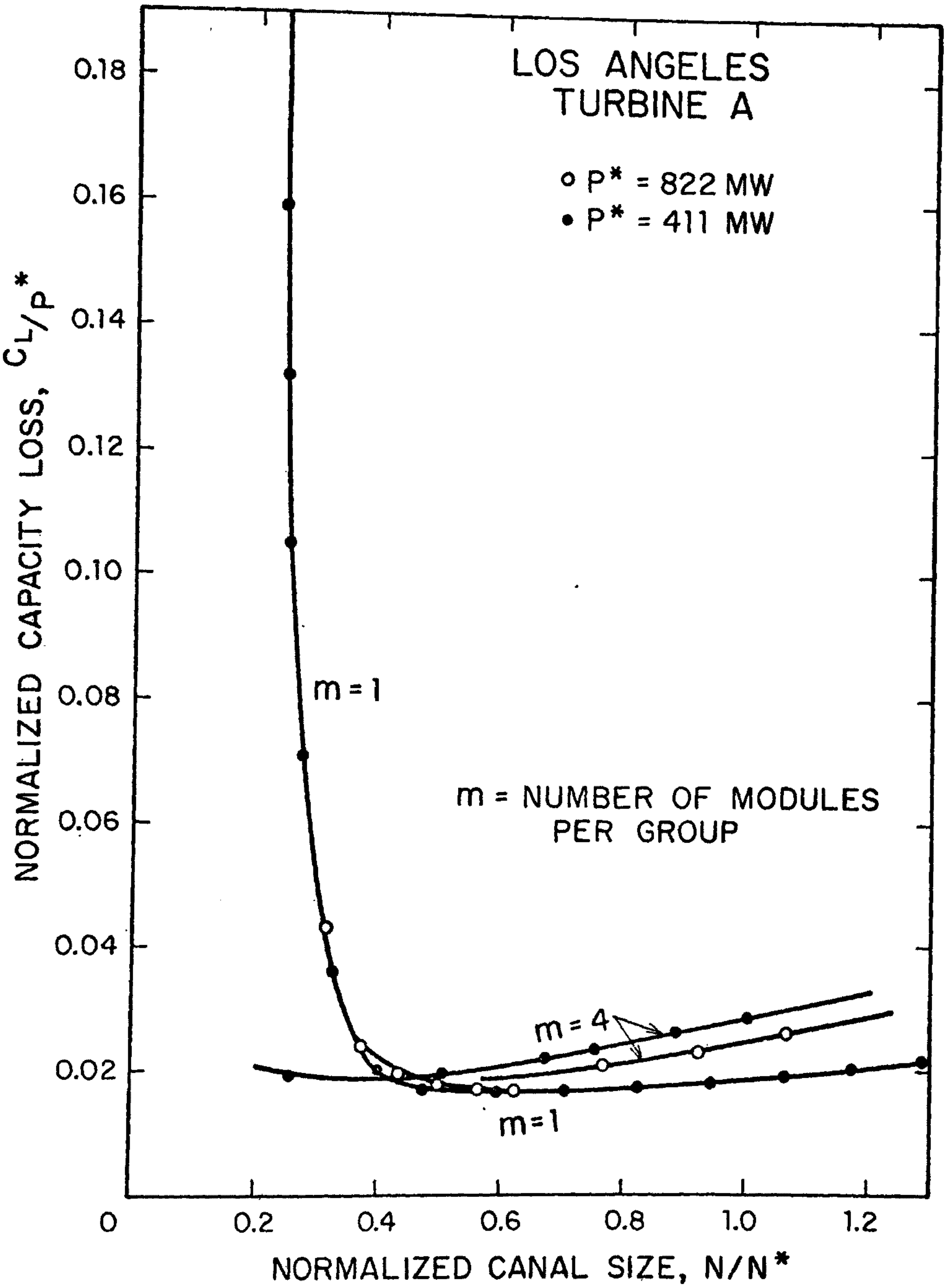

Figure 64. Normalized capacity loss

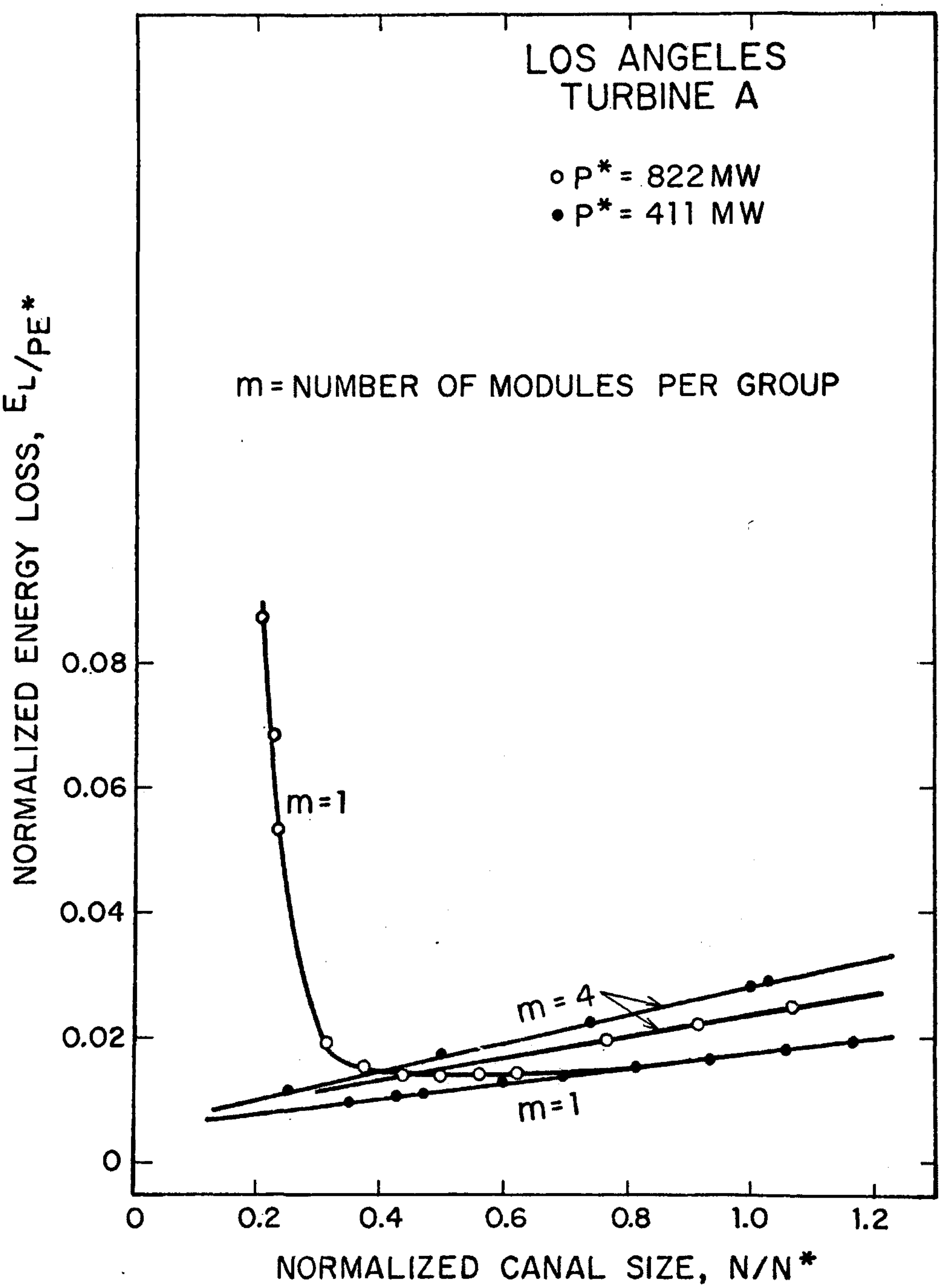

Figure 65. Normalized energy loss

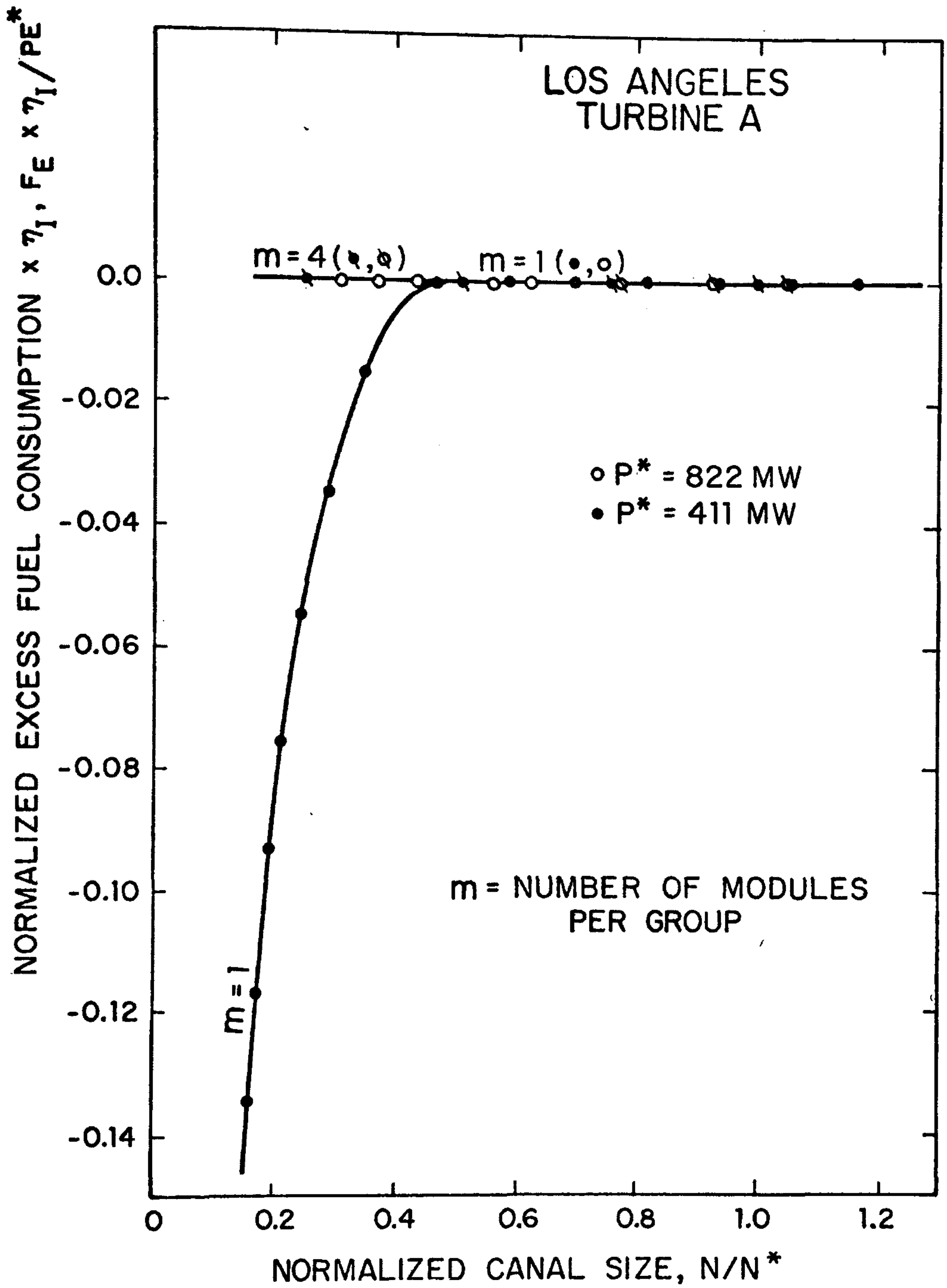

Figure 66. Normalized excess fuel consumption

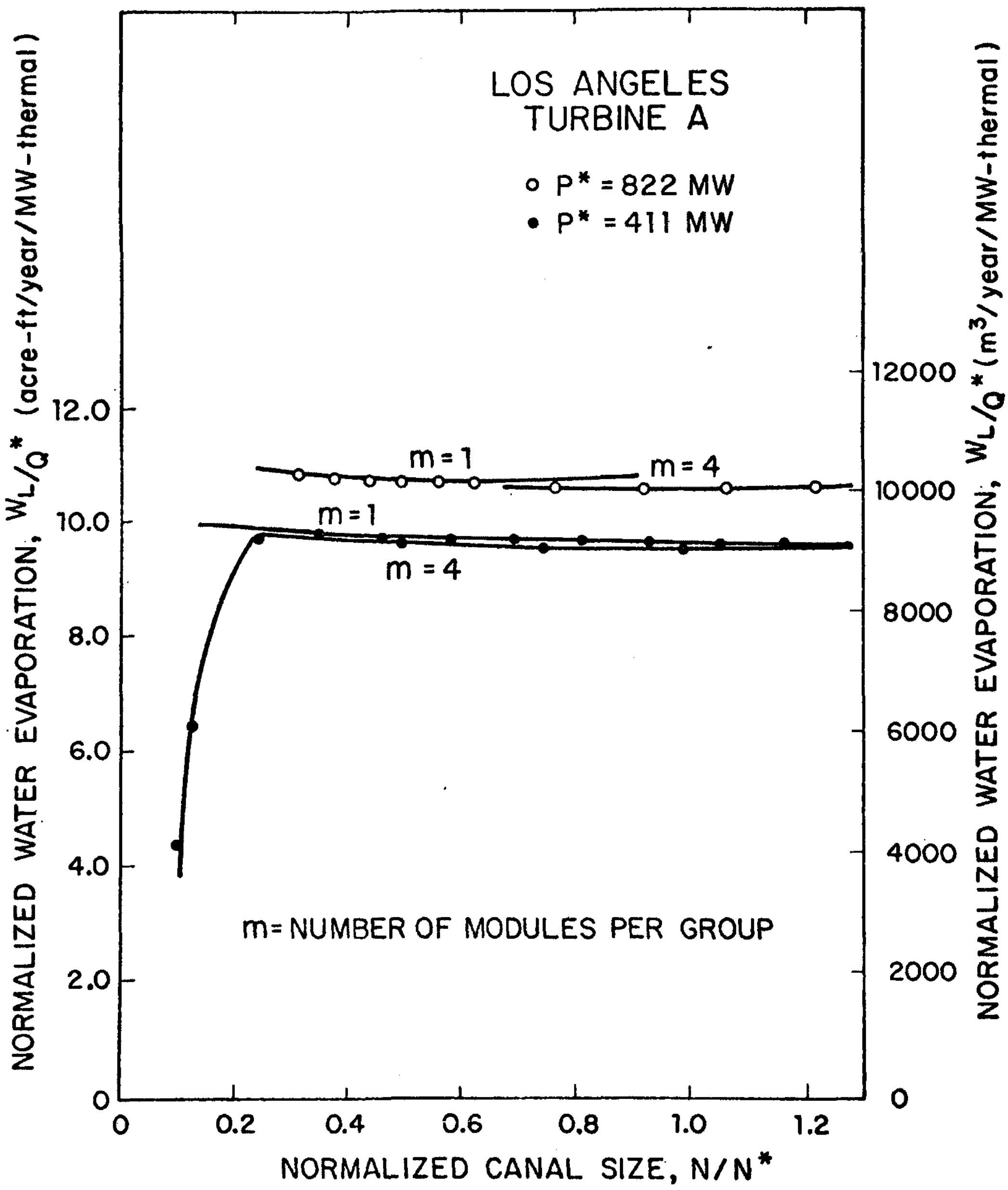

Figure 67. Normalized evaporation

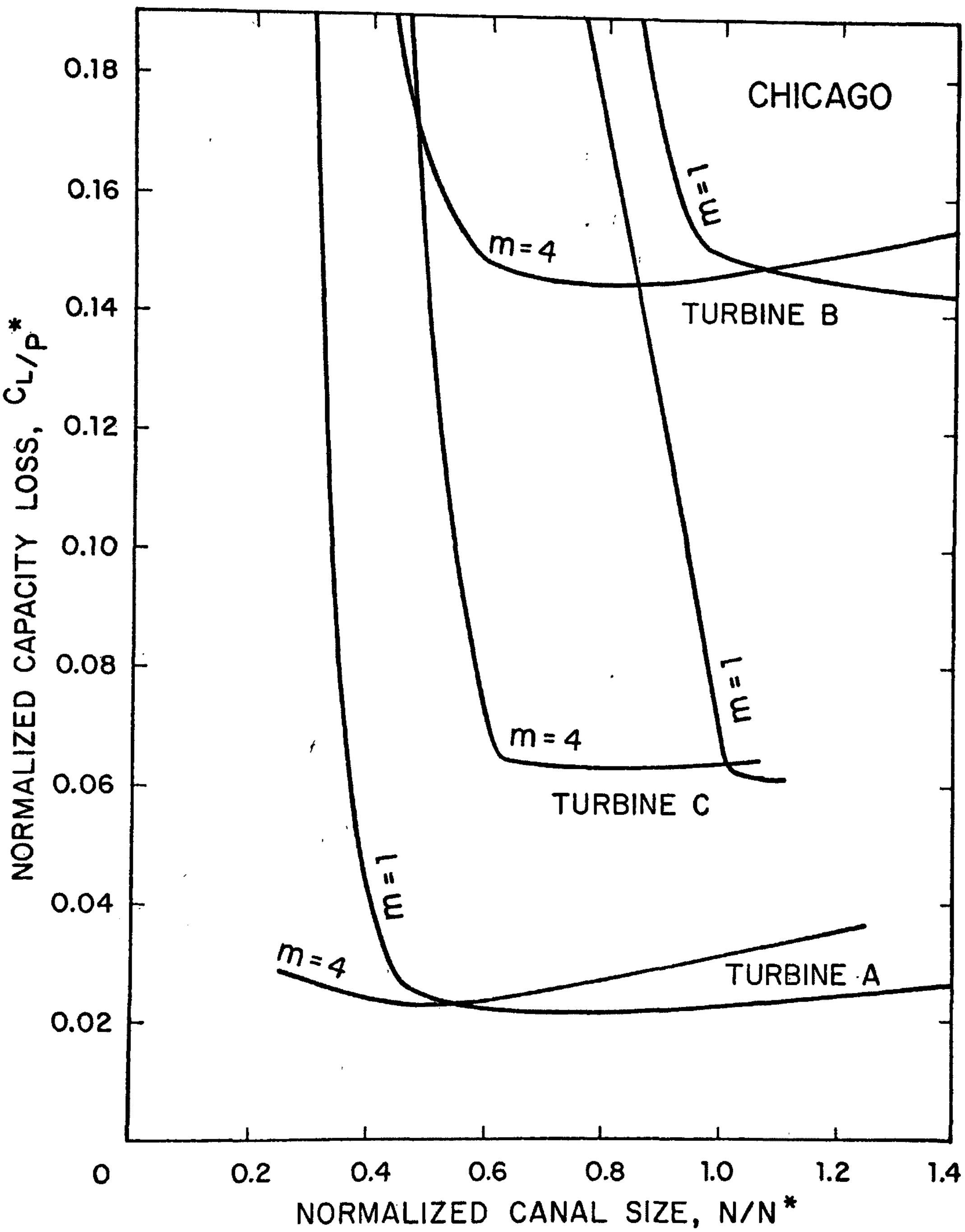

Figure 68(a). Normalized capacity loss, Chicago

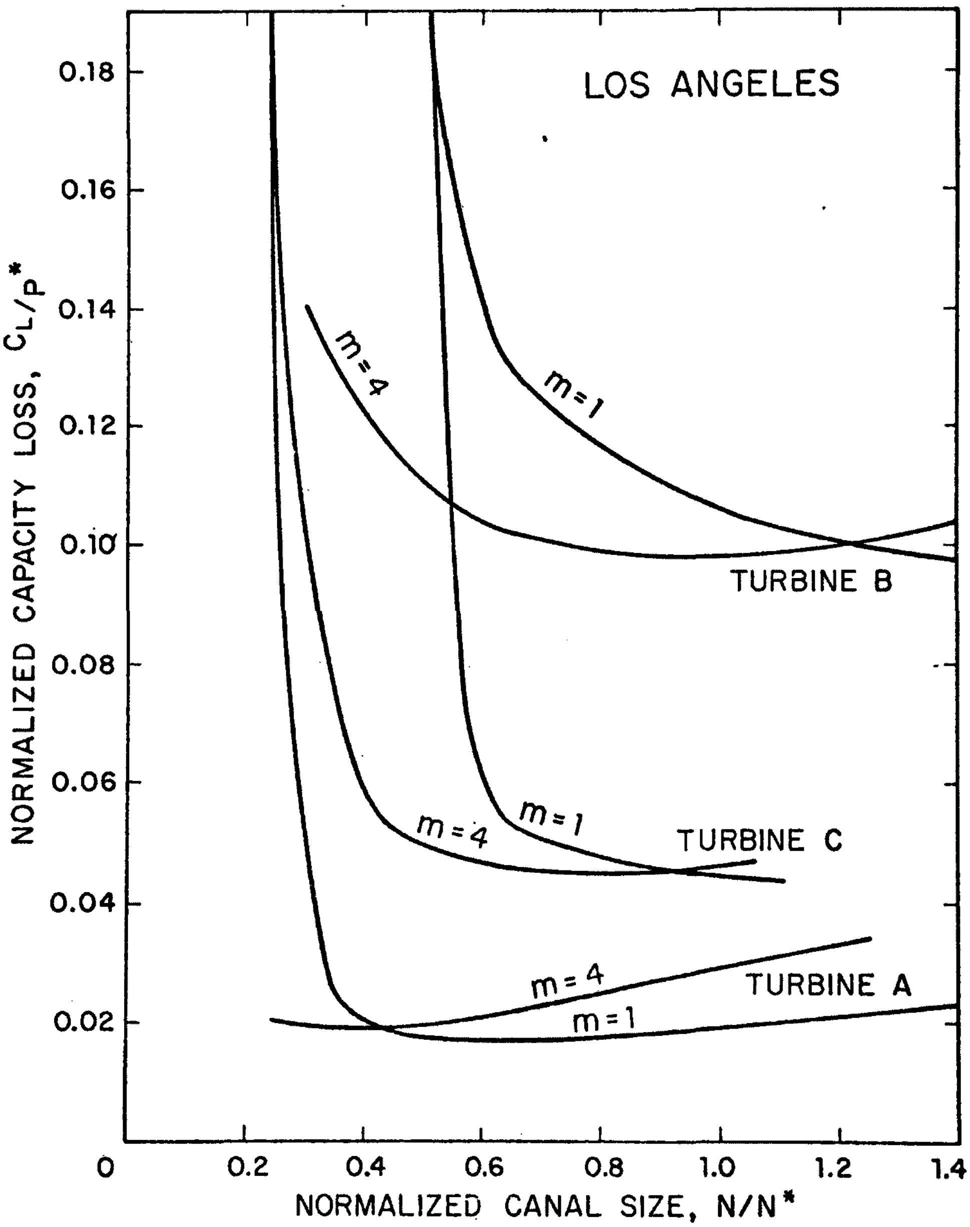

Figure 68(b). Normalized capacity loss, Los Angeles

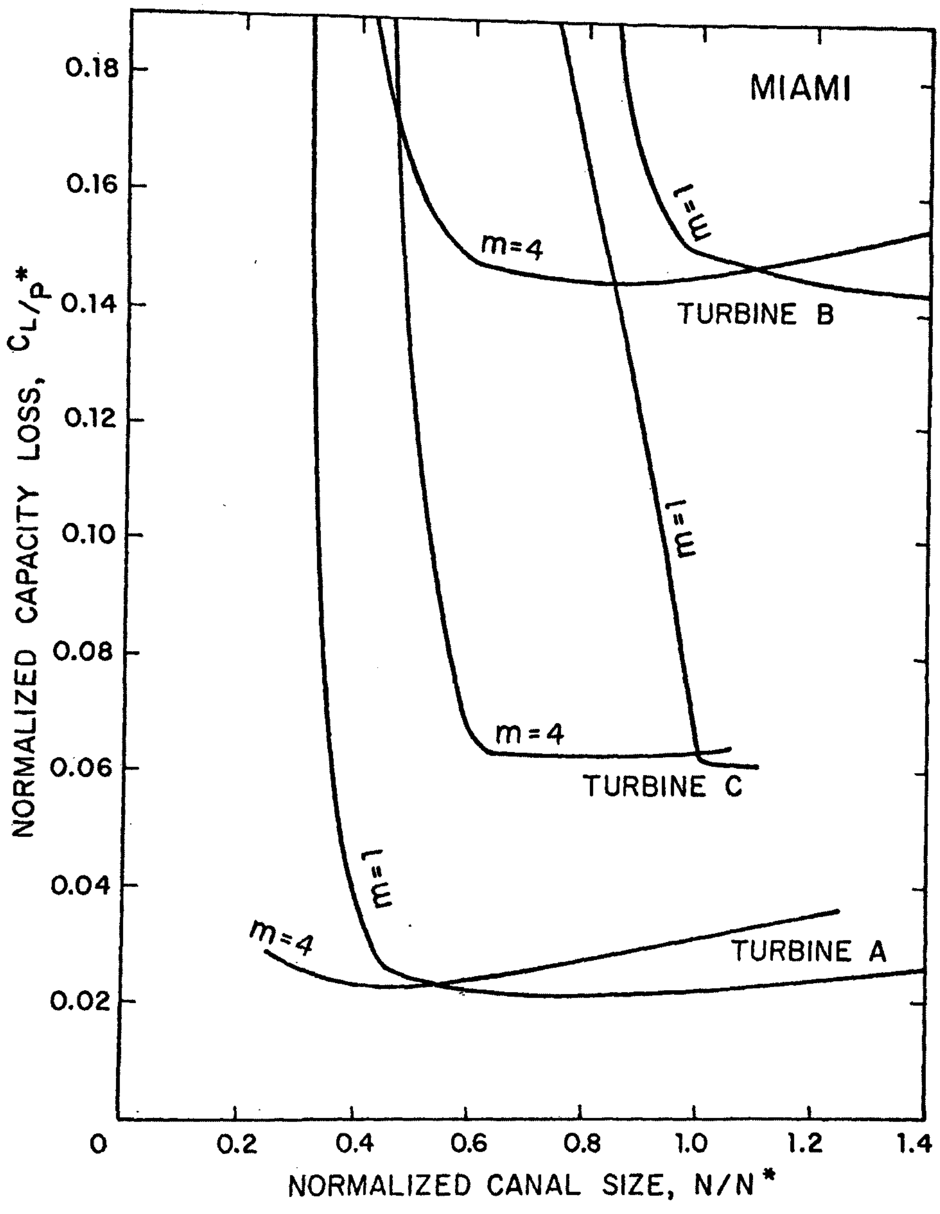

Figure 68(c). Normalized capacity loss, Miami

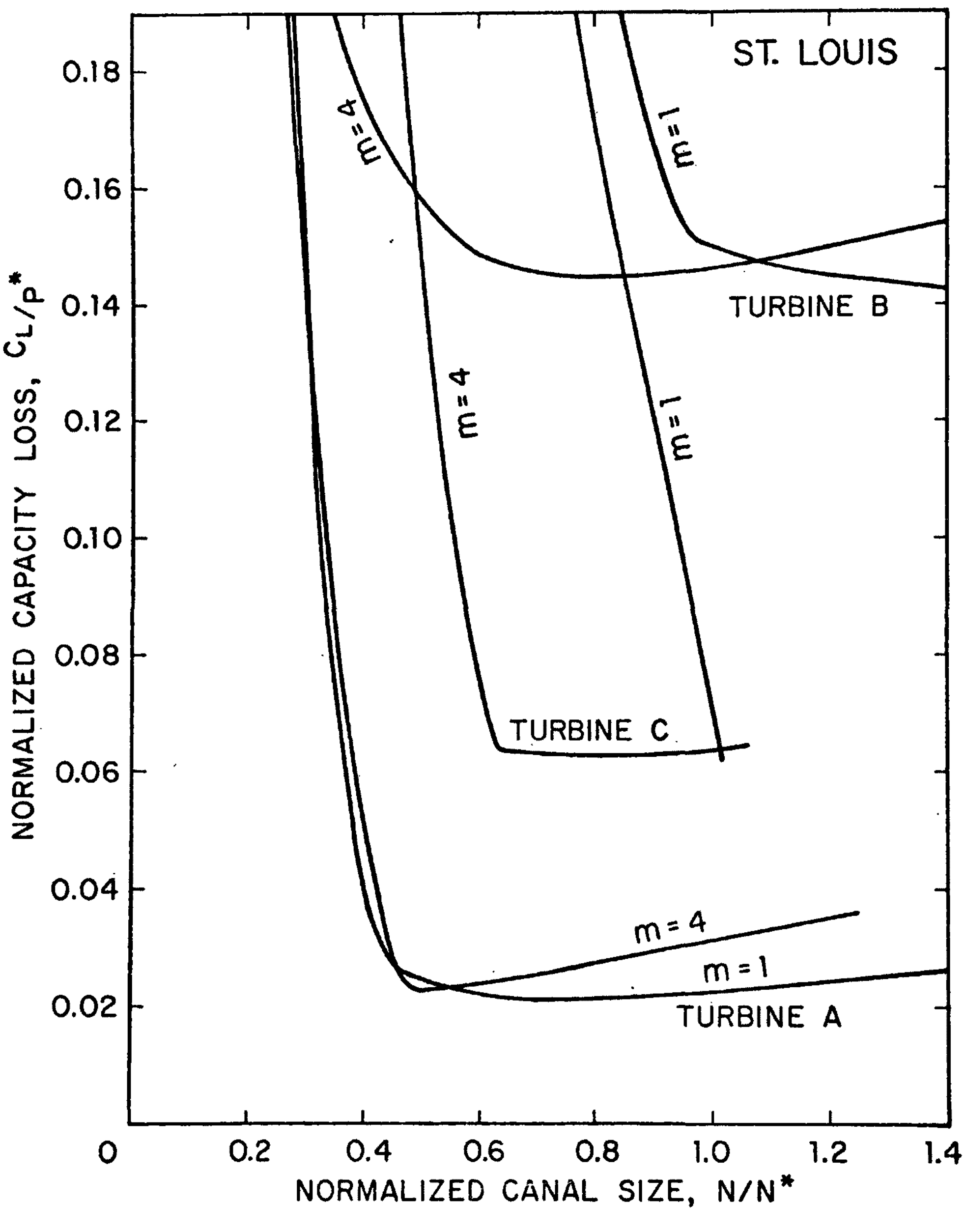

Figure 68(d). Normalized capacity loss, St. Louis

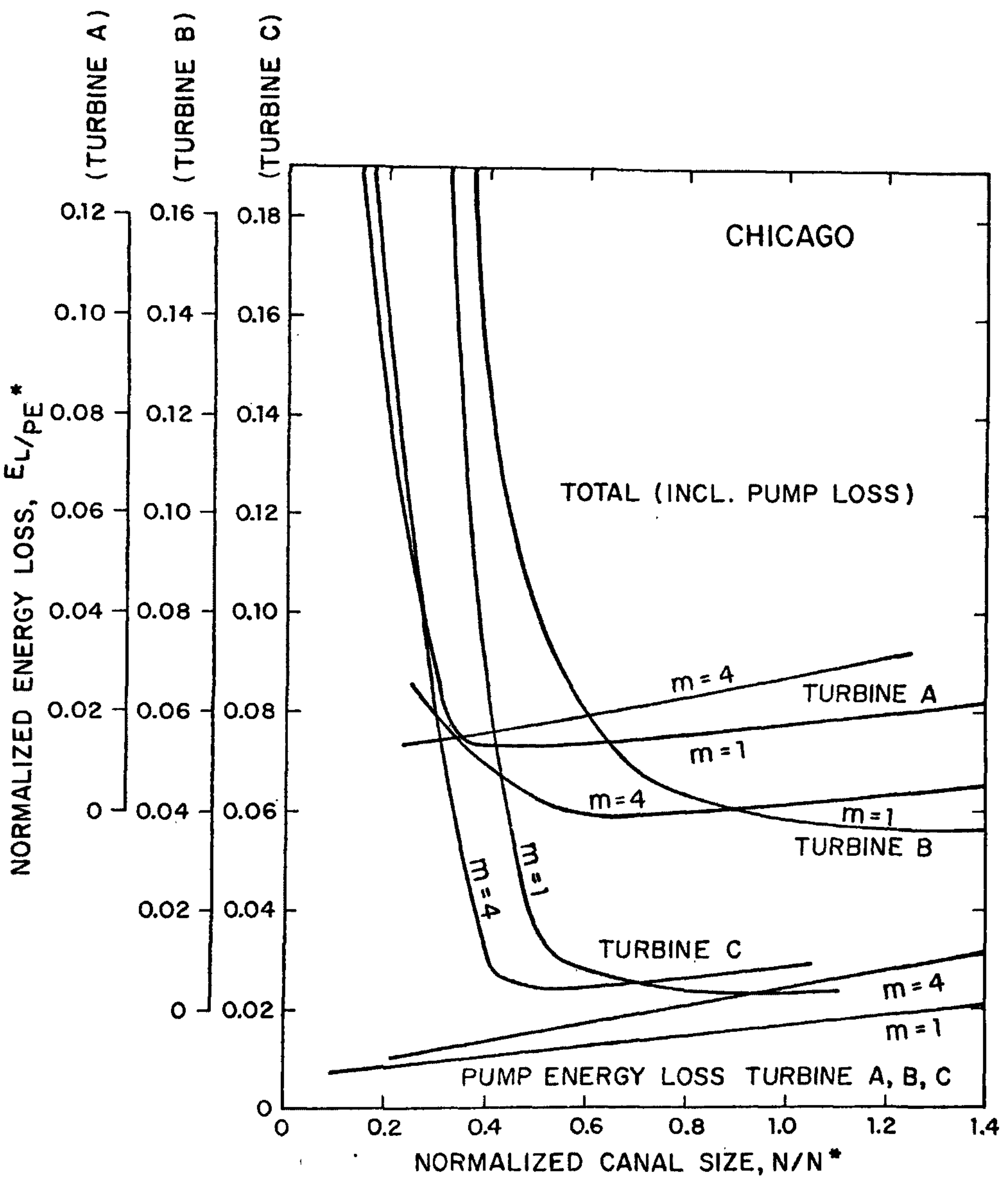

Figure 69(a). Normalized energy loss, Chicago

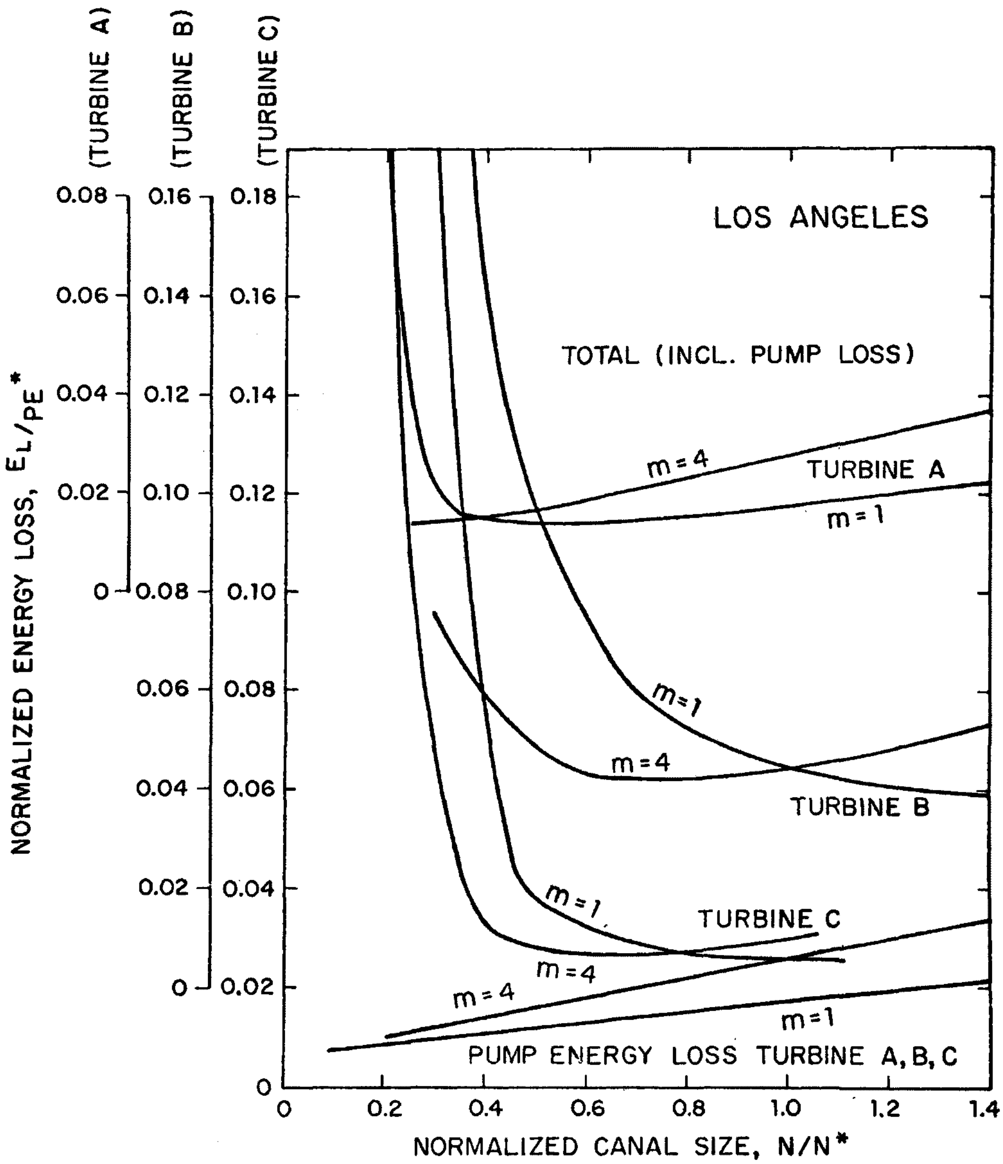

Figure 69(b). Normalized energy loss, Los Angeles

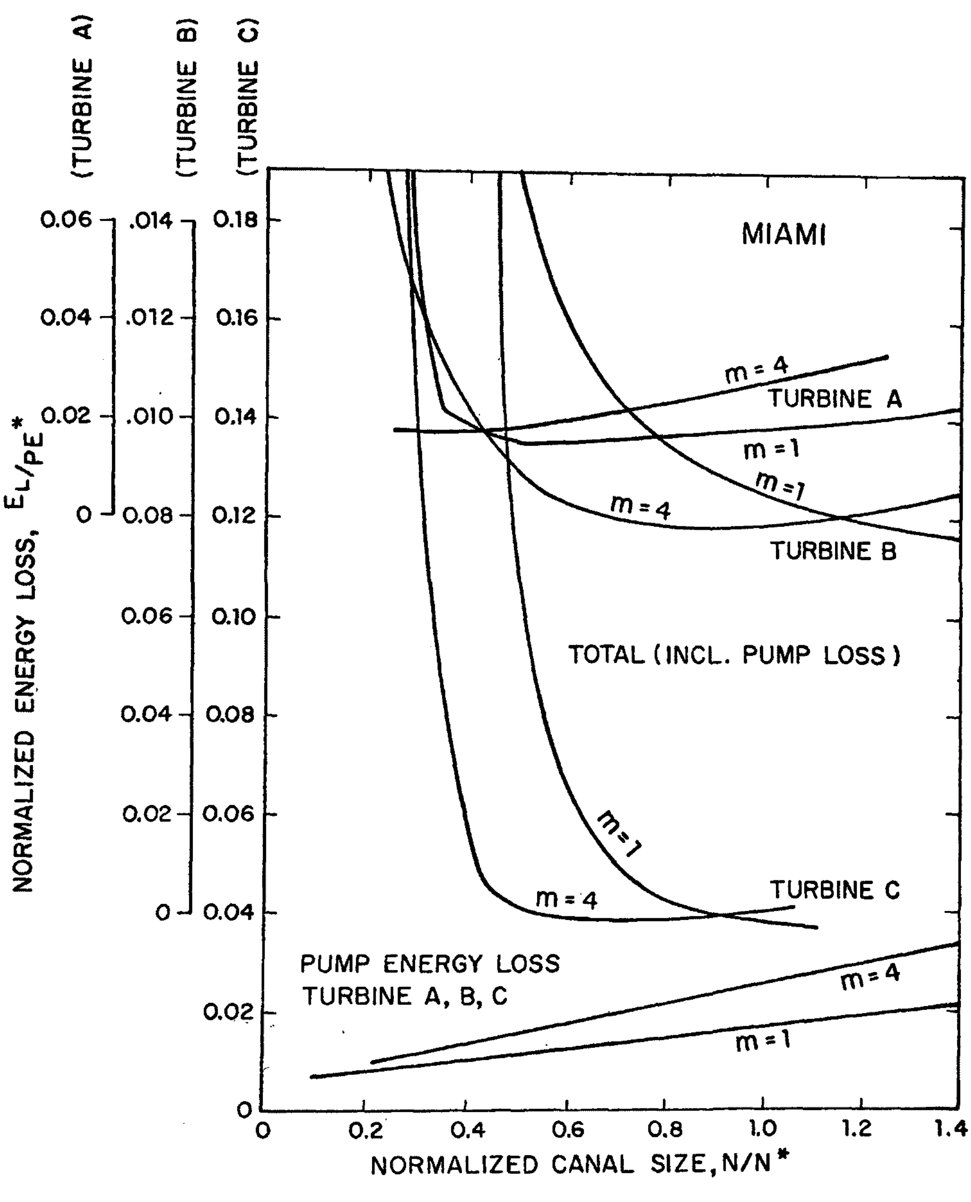

Figure 69(c). Normalized energy loss, Miami

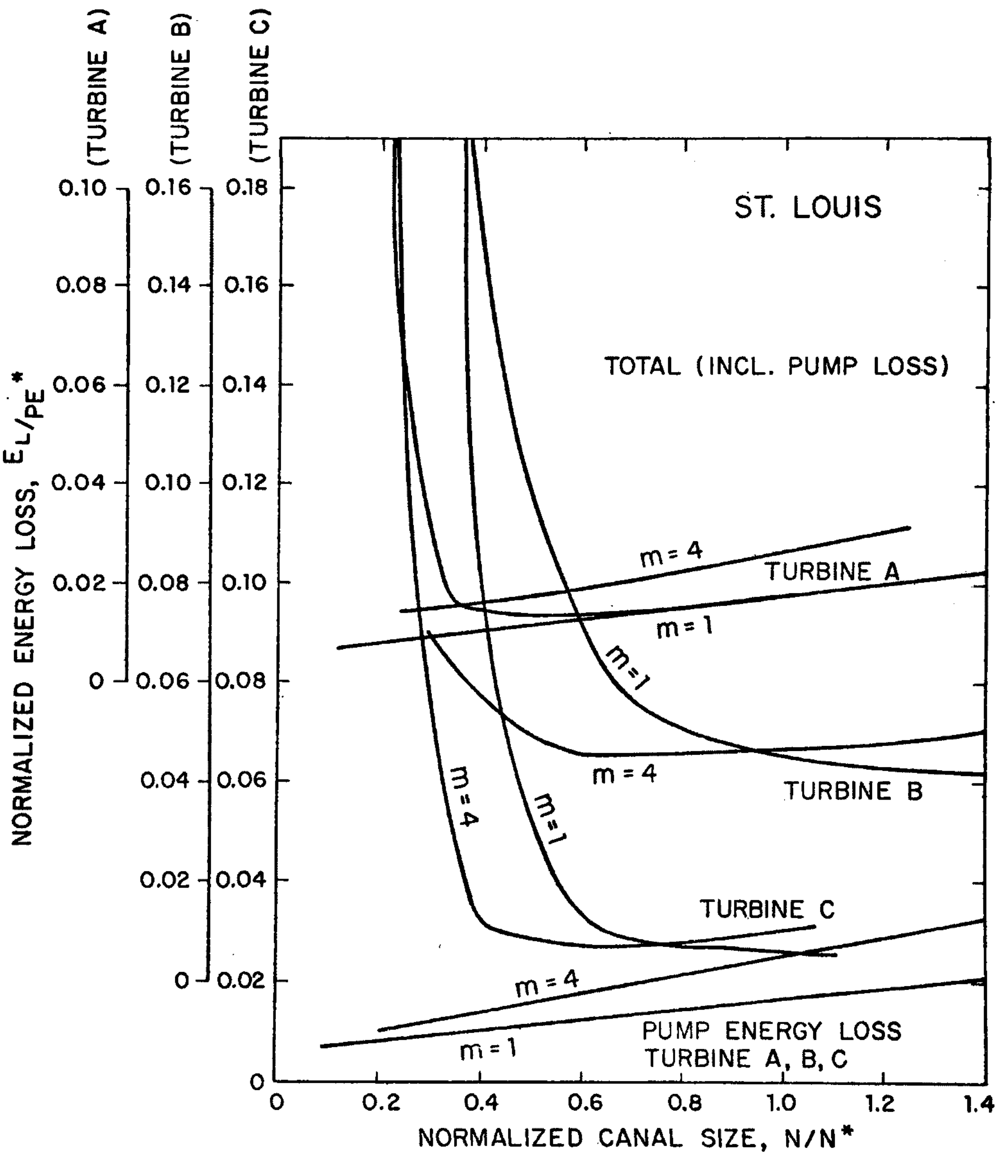

Figure 69(d). Normalized energy loss, St. Louis

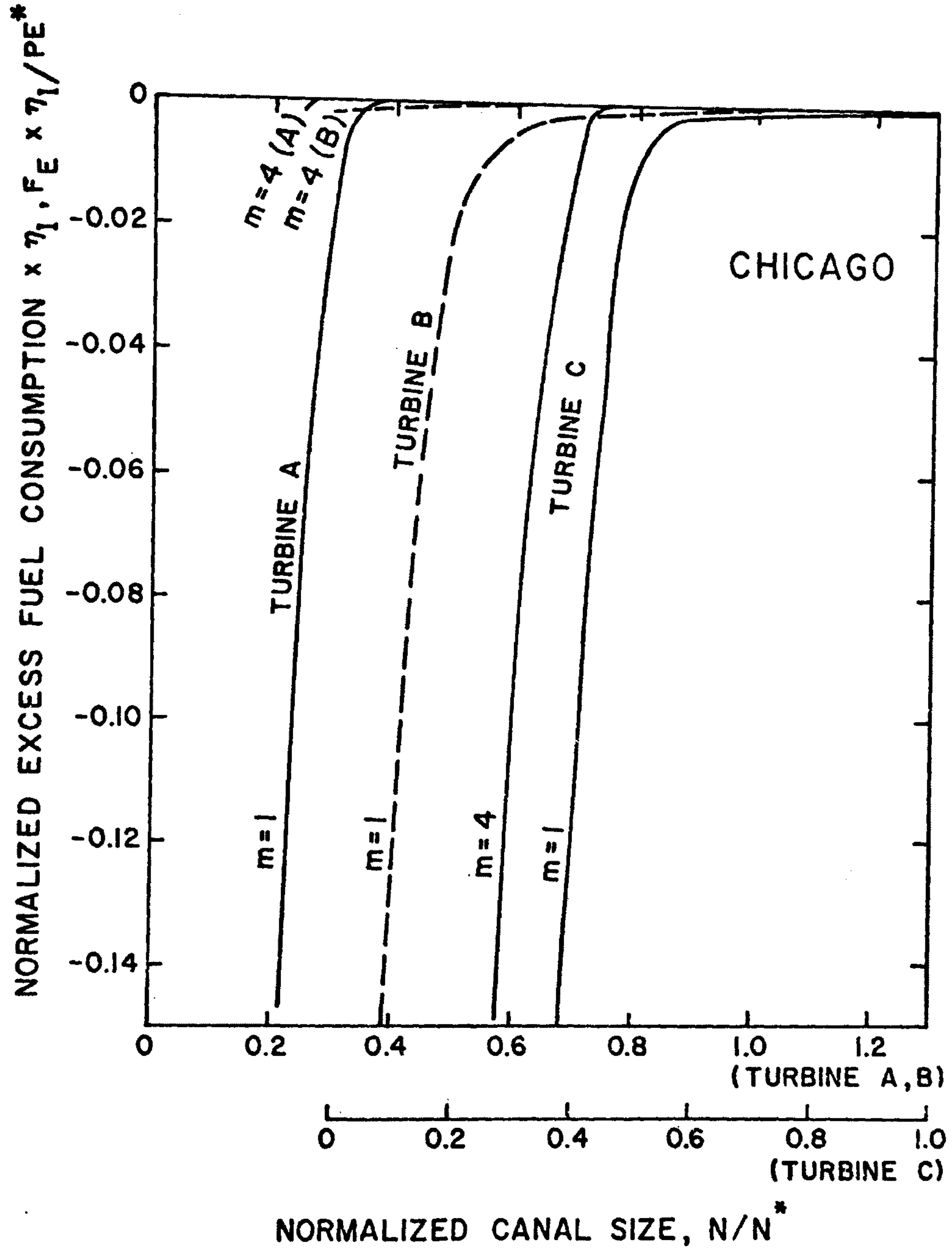

Figure 70(a). Normalized excess fuel consumption, Chicago

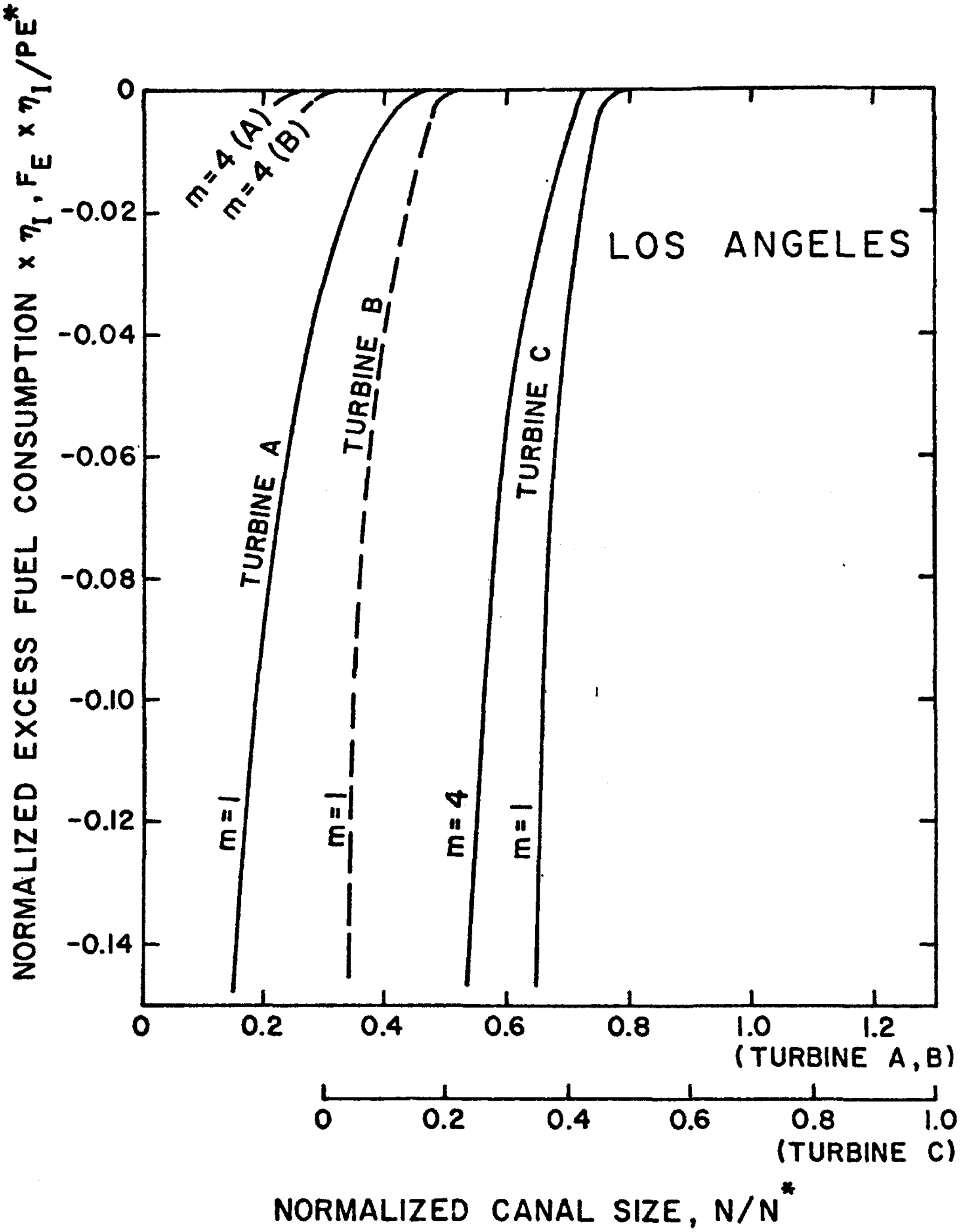

Figure 70(b). Normalized excess fuel consumption, Los Angeles

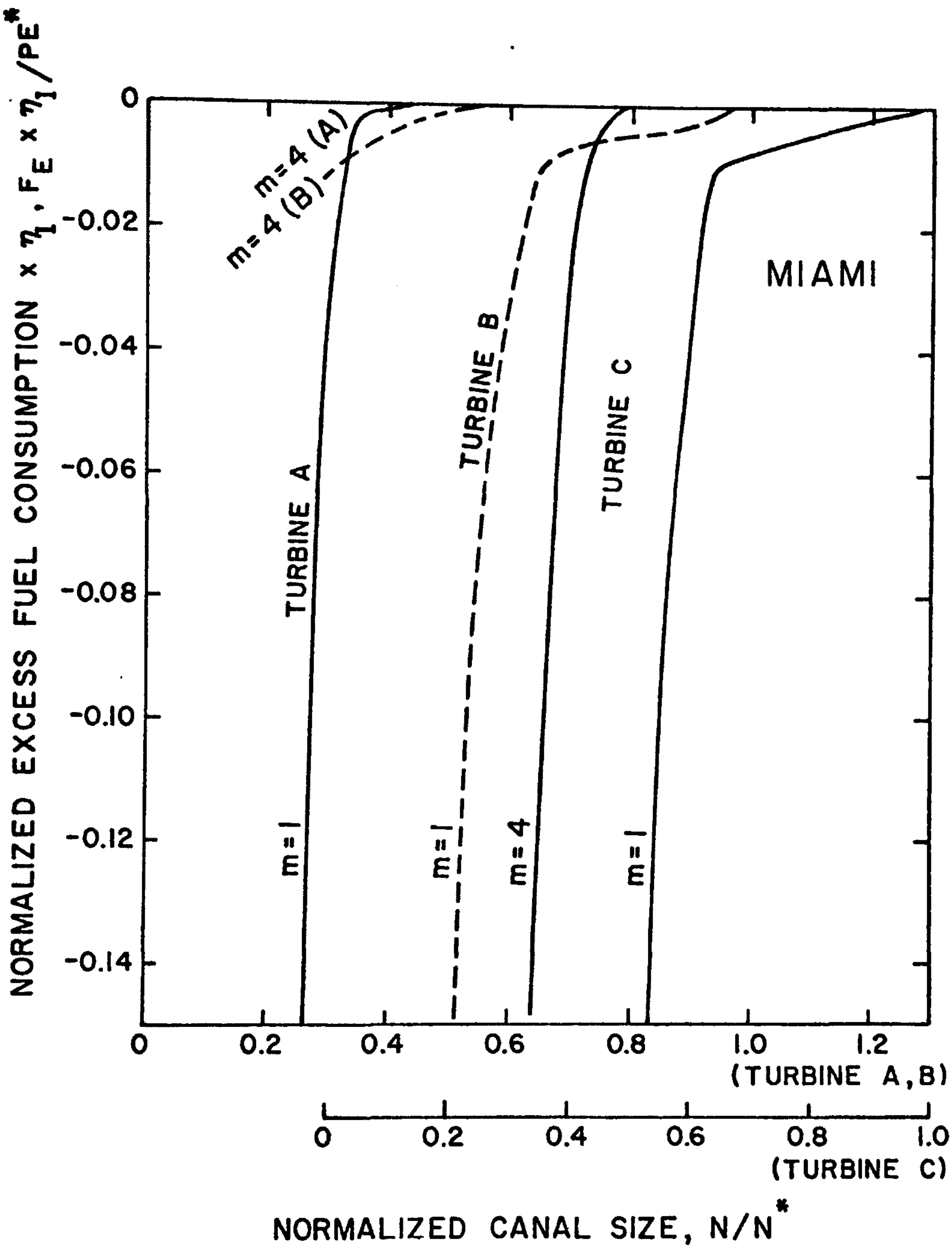

Figure 70(c). Normalized excess fuel consumption, Miami

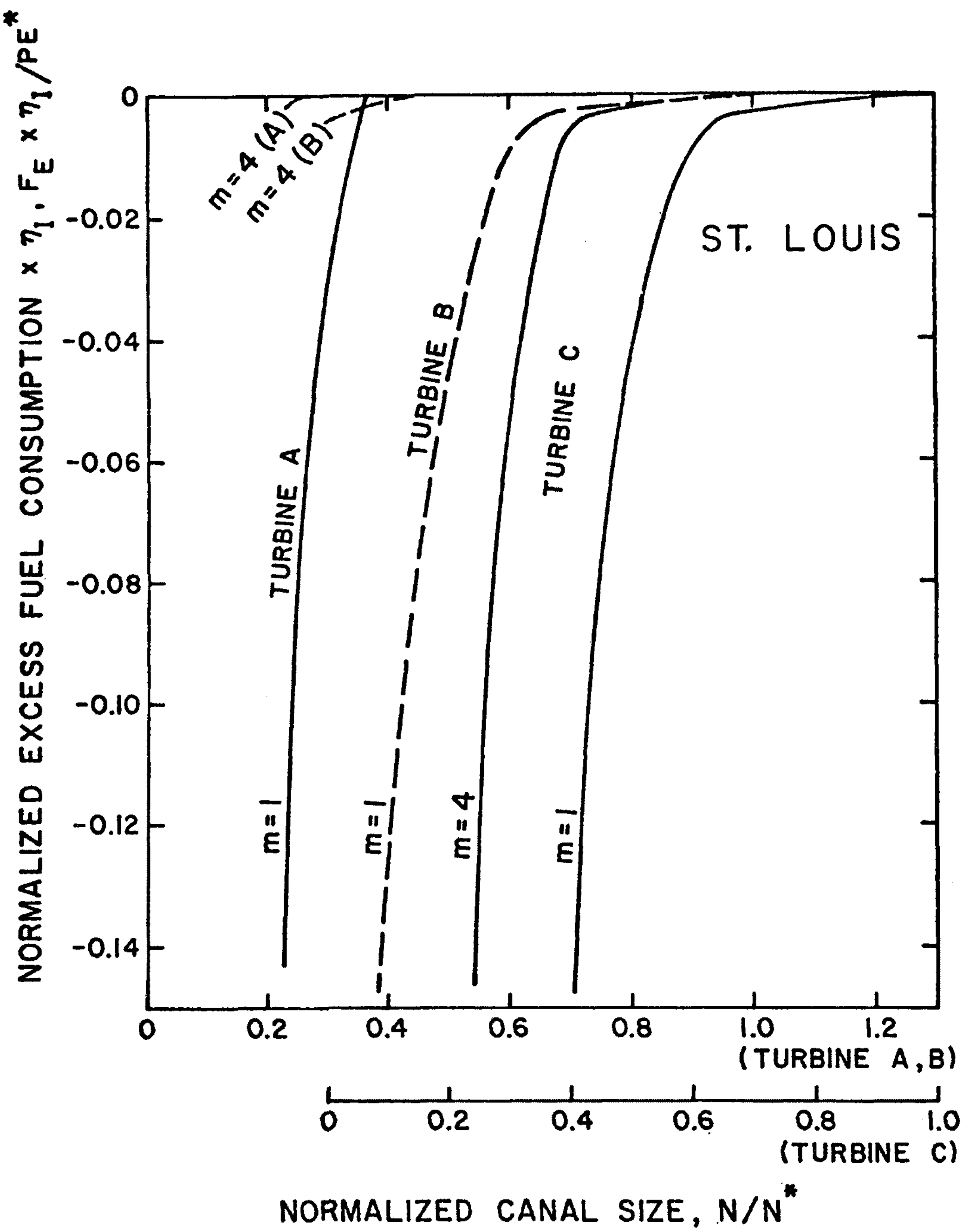

Figure 70(d). Normalized excess fuel consumption, St. Louis

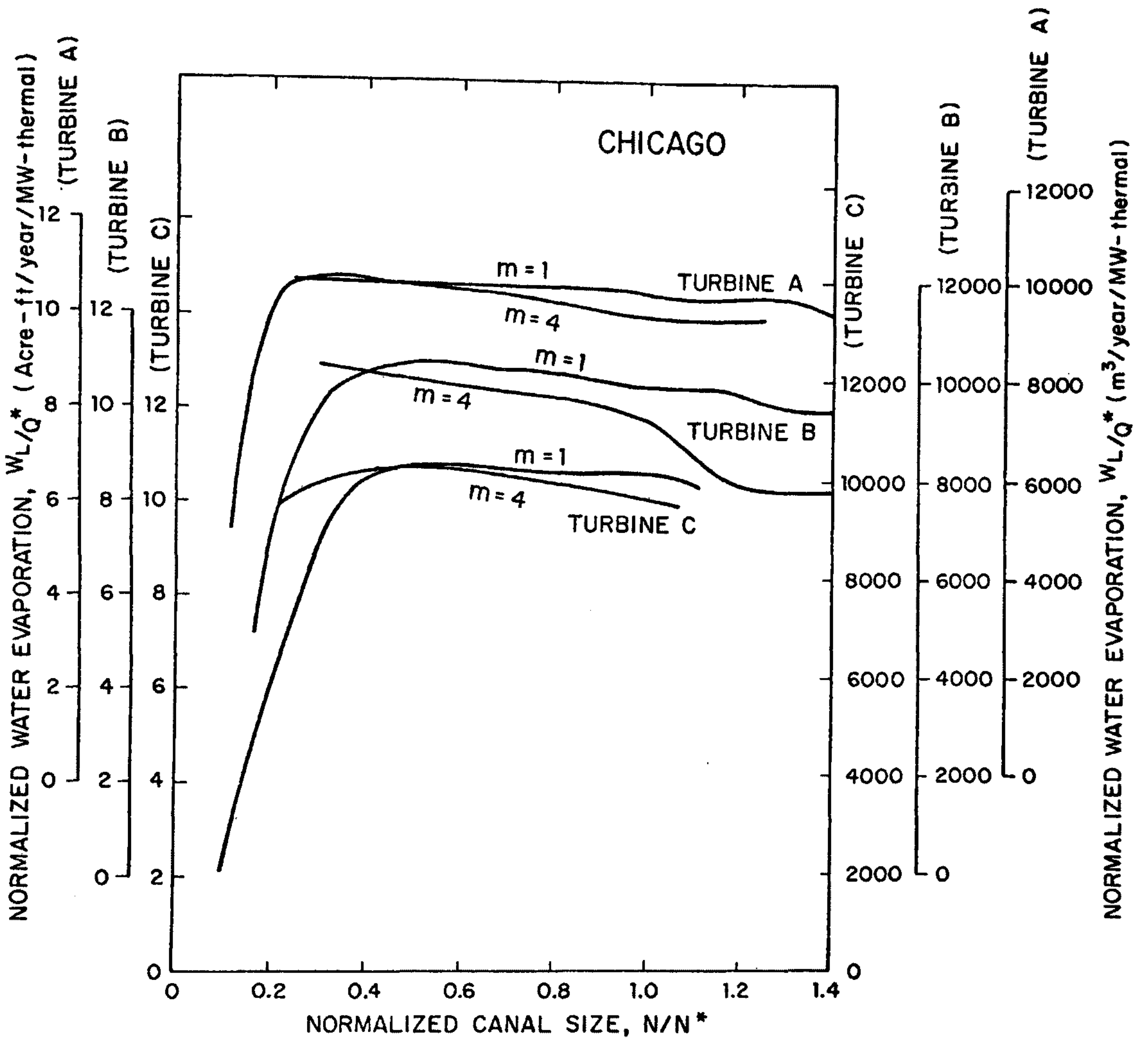

Figure 71(a). Normalized evaporation, Chicago

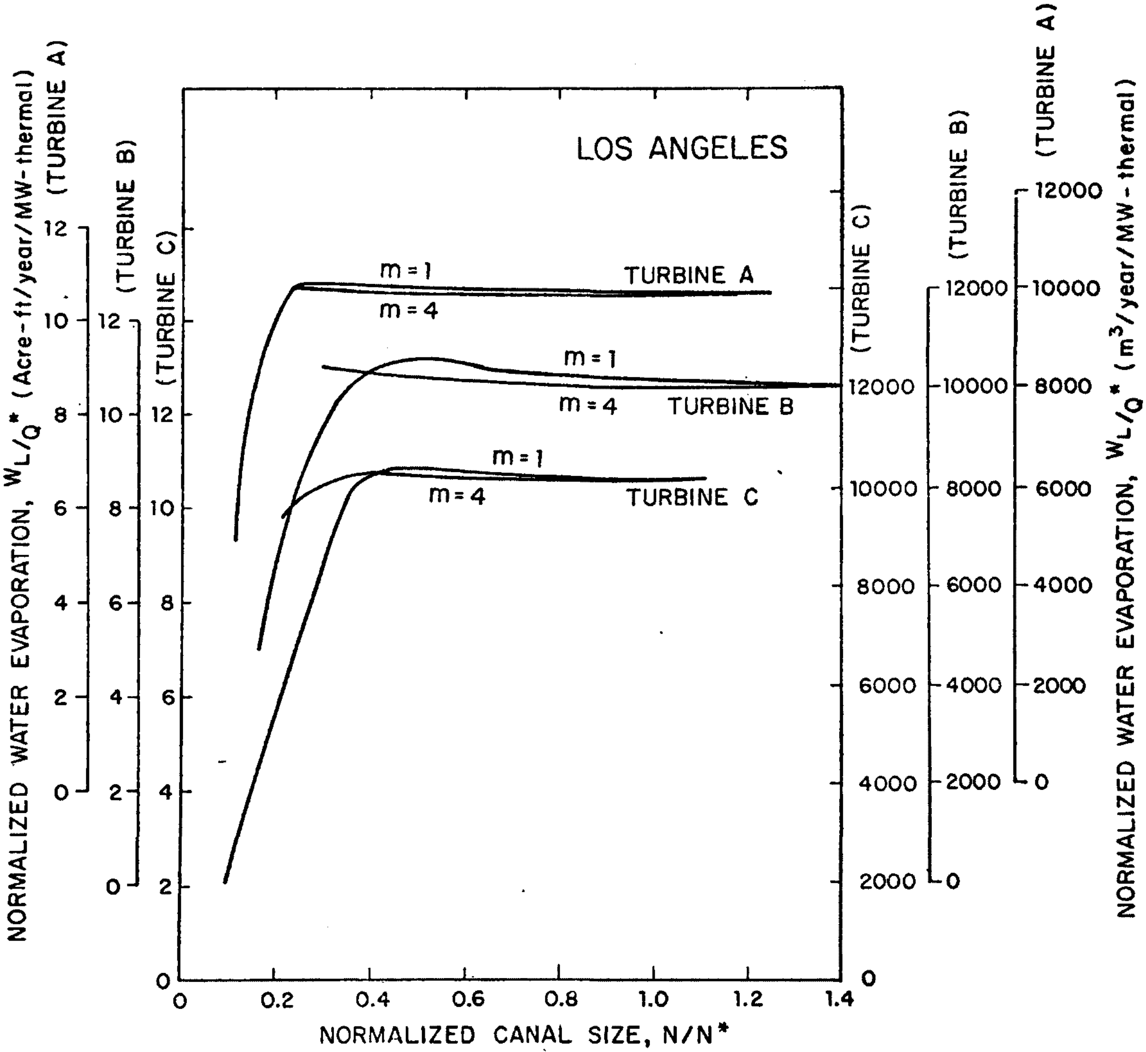

Figure 71(b). Normalized evaporation, Los Angeles

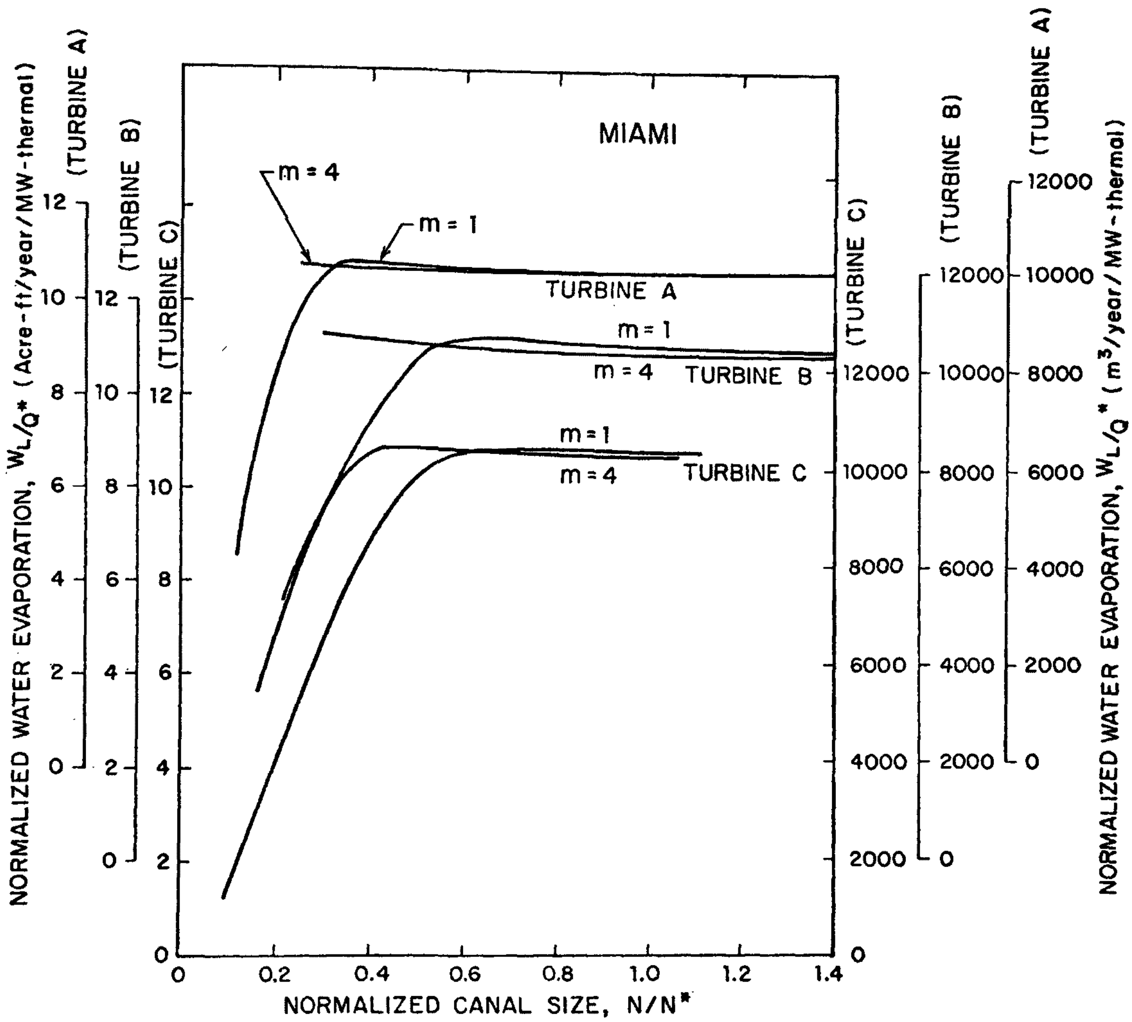

Figure 71(c). Normalized evaporation, Miami

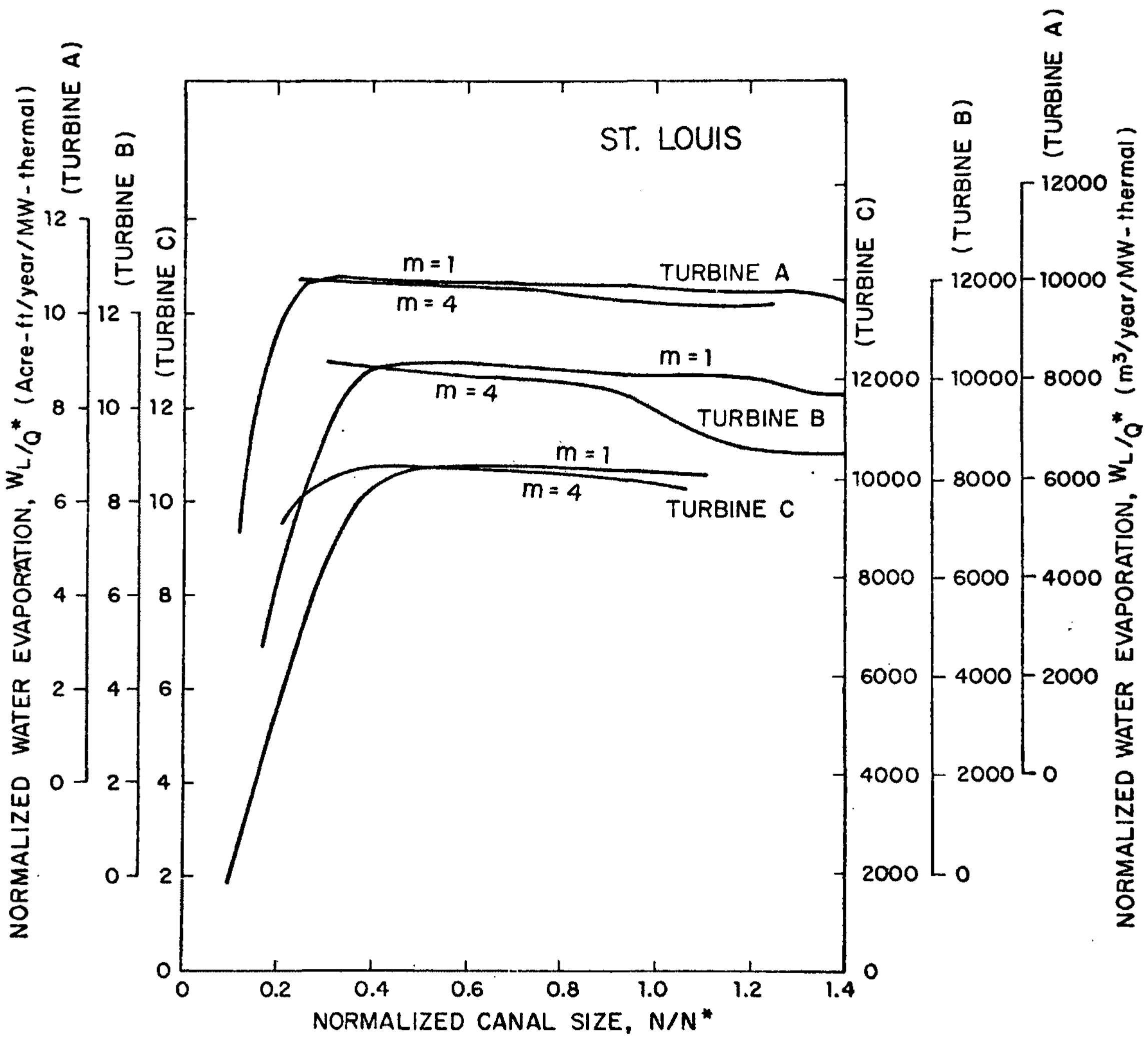

Figure 71(d). Normalized evaporation, St. Louis

introduced are small. This assumption is checked by comparing the results from the graphical computation with results obtained using the computer program. The comparison is quite acceptable as seen at the close of Section VIII.G. It must be emphasized that if the size of the unit being studied varies greatly from 411 MW, the computer program must be used to obtain accurate results.

E. PROCEDURE FOR THE ECONOMIC EVALUATION OF BACKFITTING

The various items which must be considered in the evaluation of the cost of backfitting an existing power plant or unit with a spray canal cooling system have been previously described. The manner in which these items are to be combined in order to calculate the total cost of backfitting is considered in this section which is followed by a brief description of the computer program that has been developed for this purpose.

The technique for the economic evaluation of backfitting with a spray canal cooling system is very similar to the procedure already discussed for the other cooling systems (Sections V.F, VI.F, and VII.E) with the following exceptions. In items (b) and (c) of those Sections, the following quantities must be substituted for the study of spray canals:

(b) Spray canals

1. The size of the spray cooling system in terms of the number of module groups along the canal, N, and the number of modules per group (rows) across the canal, m;
2. Unit cost of spray modules, c_s (\$ /module), unit canal construction costs (excavation, c_E, and lining, c_L), and physical dimensions of the canal (length, L_c, top-width, W_c, depth, D_c, water depth, D_{cw}, and cross-sectional shape).

(c) Economic parameters:

6. Unit cost of pump and pipe system [26], c_{pp}

Once this information has been gathered, the total differential cost of backfitting with a spray canal can be calculated by the computer program or estimated by the graphical results presented in Section VIII.D. Use of the graphical results is subject to the limitation discussed at the end of that Section. Illustration of the application of the graphical results is presented in Section VIII.G.

F. THE COMPUTER PROGRAM

The computer program which accepts any set of numerical values for the various parameters (except those which have been held fixed for the present study) and performs the calculations outlined in the previous sections is listed in Appendix VI. The thermodynamic model used to simulate the performance of spray canals is basically the same as that developed by Porter [52] and Porter and Chen [53], but there are a number of important differences in other respects. In particular, the economic considerations are formulated specifically for the analysis of backfitting in existing power plant or unit with a spray canal cooling system and cannot be used, without modification, to study the design of spray canals for new plants or units.

The computer program consists of the MAIN program and four subroutines, namely OPECOS, MODELW, SPRCOL, and POWERS. The MAIN program reads all inputs, calculates the overall capital and total costs, and controls the printout of these quantities. The inputs, along with the symbols and units used, are listed in Appendix II. The primary functions of the various subroutines not previously identified are as follows.

SPRCOL: This subroutine which computes the cold-water temperature and the water loss due to evaporation contains the thermodynamic model described in Section VIII.A.

The overall program logic is similar to that already described for the other cooling system calculations. Minor changes have been made in the main program and subroutines to accomodate the spray canal cooling model. These changes are included in the program listing in Appendix VI.

G. A HYPOTHETICAL TEST CASE

1. Consider a power plant with the characteristics identified in Section V.H.1, which also implies that the extreme dry-bulb temperature, $\hat{T}_{db}$ = 97.0°F (36.1°C) for Miami.
2. Assume that this plant is to be backfitted with a spray canal system whose characteristics are:

Number of module groups, N	= 80
Number of modules/group, m	= 1
Module size,	= 75 hp pump (55.9 kW)
Module pump flow rate	= 10,000 gpm (37.8 m^3/min)
Canal cross-section	= Trapezoidal (3-horizontal to 1-vertical side slope)
Canal length, L_c	= 8100 ft (2470 m)
Canal depth, D_c	= 10 ft (3.05 m)
Water depth, D_{cw}	= 8 ft (2.44 m)
Canal top-width, W_c	= 100 ft (30.5 m)
Wind speed, V_2	= 8 mph (3.57 m/sec)

Total water flow rate,

GPM (= $\frac{1.912 \times 10^9}{20} \times \frac{7.481}{60 \times 62}$, equation 63) = 192,254 gpm (727.7 m^3/min)

Unit module cost, c_s	= \$22,500/module
Unit cost of pump and pipe system, c_{pp}	= \$1.50/gpm ($396 /m^3/min)
Unit canal excavation cost, c_E	= $2.50/yd^3$ ($3.27/m^3$)
Unit canal lining cost, c_L (Concrete, 6 in. (0.15 m) thick	= $\$0.93/ft^2$ ($\$10/m^2$)
Concentration ratio, k*	= 3.3
Unit blowdown treatment cost, c_b	= \$0.05/1000 gal ($\$0.0132/m^3$)
Cost of hook-up and testing, C_{HT}	= included in unit module cost
Maintenance cost, C_m	= 1% of pump and module operating cost
Downtime, DT	= 720 hrs (30 days)

3. Assume that the various economic parameters are as identified in Section V.H.3.

4. Use of example results:

(a) Land Area required

$A_L = 2.5 \times L_c \times W_c$ = 46.5 acres (18.8 hectares)

(b) Excavation required,

excavation volume = $L_c D_c (W_c - 3D_c)$

$= 8100(10)(100-30)/27$ = $210,000\ yd^3$ ($160,566\ m^3$)

(c) Lining required

lining area = $L_c (W-6D_c+2\sqrt{10}\ D_c)$

$= 8100 \times (100-60+20\sqrt{10})$ = $836,290\ ft^2$ ($77,691\ m^2$)

(d) Read Figure 63 to find N* = 68

Determine normalized canal size, N/N* = 1.18

(e) Read Figure 68(c) (turbine A, Miami) to find normalized capacity loss, C_L/P^* = 0.024

Thus, capacity loss

$C_L = 0.024 \times 312.5 \times 1000$ = 7500 kW

(f) Read Figure 69(c) (turbine A, Miami) to find normalized energy loss, E_L/PE^* = 0.02

Thus, energy loss,

$E_L = 0.02 \times 312.5 \times 1000 \times 8760$ = 54.75×10^6 kW-hr/year

(g) Read Figure 70(c) (turbine A, Miami) to find normalized excess fuel consumption,

$\eta_I \; F_E/PE^*$ = 0

Thus, excess fuel,

$F_E = 0 \times 312.5 \times 1000 \times 8760/0.85$ = 0 kW-hr/year

(h) Read Figure 71(c) (turbine A, Miami) to find normalized water evaporation, W_L/Q^* = 10.7 acre-ft/yr-MW(th) (1.32×10^4 m^3/yr-MW(th)

Thus, evaporation

$W_L = 10.7 \times 1.912 \times 10^9/3.413 \times 10^6$ = 5994 acre-ft/year (7.39×10^6 m^3/year)

Also, blowdown, $W_b = W_L \frac{1}{k^*-1}$ = 2606 acre-ft/year (3.21×10^6 m^3/year)

And, makeup,

$W_m = W_L \frac{k^*}{k^*-1} = W_b + W_L$ = 8600 acre-ft/year (1.06×10^7 m^3/year)

(i) Read Figure 69(c) (turbine A, Miami) to find normalized pump and module energy loss,

$E_L/PE^* =$ = 0.019

Thus, pump and module energy loss,

$E_L(\text{pump}) = 0.019 \times 312.5 \times 1000 \times 8760$ = 52.01×10^6 kW-hr/year

5. Cost determination:

Capital costs

Spray module cost,

$N(m)c_s = 80 \times 1 \times 22{,}500$ = $1,800,000

Excavation cost,

(excavation volume)$\times c_E$ $= 210{,}000 \times 2.50$ = $ 525,000

Lining cost,

(lining area) $\times c_L$ $= 836{,}290 \times 0.93$ = $ 777,750

Pump and pipe system cost,

$c_{pp} \times GPM = 1.50 \times 192{,}254$ = $ 288,380

Pump and pipe system salvage,

$C'_{pp} = 0.2\ C_{pp}$ = ($ 57,680)

New Condensers, C_c = 0

Salvage value of old condensers,

C'_c = (0)

Salvage value of other open-cycle components, C'_o = (0)

Hook-up and testing cost, C_{HT} = included in module cost

Additional Land,

$A_L a_\ell = 46.5 \times 3000$ = \$139,500

Replacement capacity,

$CC_R = C_L c_\ell = 7500 \times 90$ = \$675,000

Downtime,

$CC_{DT} = DT \times P^* \times e'_\ell$

$= 720 \times 312.5 \times 1000 \times 0.007$ = \$1,575,000

TOTAL CAPITAL COST, CC = \$5,722,950

Operating costs/year

Excess fuel cost,

$OC_{EF} = F_E f_c$

$= 0 \times 0.000751$ = 0

Replacement energy cost,

$OC_R = E_L e_\ell$

$= 54.75 \times 10^6 \times 0.01$ = \$547,500

Supply water cost,

$W_m c_w = 8600 \times (3.259 \times 10^5) \times 0.1/10^3$ = \$280,274

Cost of blowdown treatment

$W_b c_b = 2606 \times (3.259 \times 10^5) \times 0.05/10^3$ = \$42,465

Maintenance cost,

$$C_m = 0.01 \times (52.01 \times 10^6) \times 0.01 \qquad = \$5,201$$

Makeup water cost with open-cycle system, M' = (0)

Blowdown treatment cost with open-cycle system, B' = (0)

Maintenance cost of open-cycle system, C'_M = (0)

TOTAL ANNUAL OPERATING COST, OC = $875,440

Total costs

From equation (20), the total excess unit cost due to backfitting, tc, is given by

$$tc = \frac{OC + CC \times FCR}{8760 \times P^*}$$

$$= \frac{875,440 + (5,722,950 \times .179)}{312.5 \times 1000 \times 8760}$$

$$\boxed{tc = 0.6940 \text{ mills/kW-hr}}$$

To check the assumption made at the end of Section VIII.D, i.e., that the errors introduced by reading Figures 68 through 71 directly for the 312.5 MW unit are small, this example was also analyzed using the computer program. The computer results are CC = $5,703,772; OC = $907,214/yr; tc = 0.7044 mills/kW-hr. The difference between the graphical result and the computer result for the total excess unit cost is seen to be approximately 1.5%, indicating that the graphical method does indeed yield a good approximation for the given problem.

SECTION IX

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SECTION X

GLOSSARY OF SYMBOLS

A	cooling pond area
A^*	reference cooling pond area
A_a	cooling pond access land area
A_c	condenser surface area
A_d	water droplet surface area
A_L	required land area
a_ℓ	unit land cost
B	tower breadth
B'	blowdown cost with open-cycle system
B_o	Bowen ratio
b_f	dh/dT evaluated at T_f
C_c	capital cost of new condenser
C'_c	salvage value of old condenser
C_{cs}	capital cost of closed-cycle cooling system
C_{HT}	cost of hook-up and testing
C_L	maximum capacity loss
C_m	annual maintenance cost of closed-cycle system
C'_m	annual maintenance cost of open-cycle system
C'_o	salvage value of old system components other than pumps, piping and condensers
C_{pp}	capital cost of new pump and pipe system
C'_{pp}	salvage value of old pumps and piping
CC	total differential capital cost
CC_{DT}	differential cost of unit downtime during changeover to closed-cycle cooling

CC_R	capital cost of replacement capacity
CC_S	differential capital cost of closed-cycle cooling system
CF	plant capacity factor
c, c_1, c_2, c_3	numerical conversion factors
c_a	unit cost of cooling pond access land
c_b	unit cost of water treatment
c_c	unit cost of new condenser
c_E	unit excavation cost for spray canal
c_L	unit canal lining cost
c_ℓ	unit capital cost of replacement capacity
c_m	unit maintenance cost
c_p	unit cost of cooling pond
c_{pp}	unit cost of pump and pipe system
c_s	unit cost of spray modules
c_t	unit cost of cooling towers
c_w	unit cost of water
D	width of clearance around cooling tower
D_1	bottom diameter of hyperbolic shell
D_2	throat diameter of hyperbolic shell
D_3	top diameter of hyperbolic shell
D_c	spray canal depth
D_{cw}	water depth in spray canal
DT	downtime during hook-up
E_L	annual energy loss
EA	actual net energy output for one year
ER	rated net energy output for one year
e_A	vapor pressure of air at T_{db}
e_ℓ	unit cost of replacement energy
e'_ℓ	unit differential cost of replacement energy during downtime
e_S	saturation vapor pressure of air at T_{db}
F_E	annual excess fuel consumption
F_w	wet-bulb correction factor
FCR	fixed charge rate

f	frequency function
f_c	unit cost of fuel
G	total air flow rate through cooling tower
GPM	water flow rate
H	fill (pile) height
H_v	latent heat of vaporization
$h(T)$	total heat of water at temperature T
h_a	total heat of air-vapor mixture
h_v	enthalpy of water vapor
K	overall head loss coefficient
K_c	effective droplet convective heat transfer coefficient
k	concentration of contaminants
k^*	concentration ratio
k_m	maximum allowable concentration of contaminants
L	fill (pile) length
L^*	reference length of mechanical-draft cooling tower
L_c	spray canal length
ℓ	frictional head loss in hyperbolic shell
ℓ_p	frictional head loss in evaporative pile
M'	makeup water cost with open-cycle system
m	number of spray modules per group
m_d	water droplet mass
N	number of module groups (passes) along spray canal
N^*	reference size of spray canal
N_{tot}	$N \times m$, total number of spray modules in canal
NTU	number of transfer units
$\overline{NTU}$	average number of transfer units
OC	differential operating cost
OC_{EF}	cost of excess fuel consumption
OC_R	operating and maintenance cost of replacement capacity
OC_S	differential operating and maintenace cost of closed-cycle cooling system
P	power output

P^*	nameplate power output
P_{cs}	power to operate closed-cycle cooling system
P_D	power demand
P_{HR}	plant heat rate
P_{min}	gross power output at extreme temperature
PE*	nameplate energy
p	turbine back pressure
p^*	reference turbine back pressure
p'	turbine back pressure for calculation of reference cooling system size
p_A	atmospheric pressure
p_{max}	maximum allowable turbine back pressure
Q	heat rejection rate
Q^*	reference heat rejection rate
Q_{IP}	in-plant and stack heat losses
Q_T	rate of heat input (heat equivalent of fuel consumption)
QA	atmospheric radiation to water surface
QAR	reflected atmospheric radiation from water surface
QC	conductive heat loss
QE	evaporative heat loss
QP	heat input to cooling system from plant
QR	heat input by solar and atmospheric radiation
QS	solar radiation to water surface
QSC	clear sky solar radiation
QSR	reflected solar radiation from water surface
QW	heat loss by long-wave radiation
R	nominal natural-draft tower radius
R_c	cloudiness ratio
RC	cooling range
RES	residual in iterative solution
r_1, r_2, r_3	ratios of hyperbolic shell dimensions
r_e	fraction of total spray canal flow rate lost by evaporation
r_F	ratio of flow rate per module to total canal flow rate

S	shell height of natural-draft cooling tower
S^*	reference shell height
T	throat height of hyperbolic shell
$\hat{T}$	extreme temperature, equalled or exceeded by 10 hours/year
T_C	cold-water temperature
T_d	design temperature
T_{db}	dry-bulb temperature
T_f	film temperature
T_H	hot-water temperature
T_{HR}	turbine heat rate
T^*_{HR}	reference turbine heat rate
T_s	throttle setting
T_{wb}	wet-bulb temperature
T_{wb_d}	design wet-bulb temperature
$T_{wb\ell}$	local wet-bulb temperature
TC	total differential cost
TS	cooling pond water surface temperature
TU	tower units
t_d	time of flight of water droplets
tc	unit excess cost of energy production
U_c	heat transfer coefficient
V_2	wind speed at height of 2m
v	average velocity
W	fill (pile) width
W_b	blowdown water volume
W_c	spray canal top-width
W_L	annual water loss due to evaporation
W_m	makeup water volume
w	absolute humidity of air
w_ℓ	evaporative water loss per unit area
w_s	specific humidity
z	height above cooling pond
z_o	reference height above cooling pond

γ	specific weight of water
Δ	heat rate correction
Δt	time duration
$\Delta\theta_V$	virtual temperature difference
η_I	in-plant efficiency
η_p	plant efficiency
η_t	turbine base efficiency
ξ	specific heat of liquid water at constant pressure

SECTION XI

APPENDICES

APPENDIX I

SUMMARY OF ECONOMIC ANALYSIS

Differential Capital Costs

$$CC = CC_S + CC_{DT} + CC_R \, , \qquad \$ \tag{15}$$

$$= \gamma_p (C_c - C'_c + C_{pp} - C'_{pp} + C_{cs} - C'_o + A_L a_\ell + C_{HT})$$

$$+ \gamma_p (DT \times P^* \times e'_\ell)$$

$$+ \gamma_p \left(\frac{C_L}{P^*} \times P^* \times c_\ell \right)$$

where

CC = total differential capital cost, \$

CC_S = differential capital cost of closed-cycle system, \$

CC_{DT} = cost due to outage at hook-up, \$

CC_R = capital cost of replacement capacity, \$

γ_p = price escalation factor for materials and labor

C_c = capital cost of new condenser ($= A_c c_c$), \$

A_c = surface area of new condenser

c_c = unit cost of new condenser, \$/surface area

C'_c = salvage value of old condenser, \$

C_{pp} = capital cost of new pumps and piping, \$

C'_{pp} = salvage value of old pumps and piping, \$

C_{cs} = capital cost of closed-cycle cooling system (less condenser, pumps and piping), \$

C'_o = salvage value of old cooling system components, excluding condensers, pumps and piping, \$

A_L = additional land requirement

a_ℓ = unit cost of additional land , \$/unit area

C_{HT} = hook-up and testing costs for new system, \$

DT = downtime during hook-up, hours

P^* = nameplate capacity, kW

e'_ℓ = unit cost of replacement energy during hook-up at \$/kW-hr (difference between purchase price and usual production cost with the affected unit)

C_L = maximum capacity loss (10-hr exceedance), kW

c_ℓ = unit capital cost of replacement capacity, \$/kW

<u>Differential Operating Costs</u>

$$OC = OC_S + OC_R + OC_{EF} \quad , \qquad \$/\text{year} \tag{16}$$

$$= \left\{ \frac{k^*}{k^* - 1} W_L c_w - M' + \frac{1}{k^* - 1} W_L c_b - B' + C_m - C'_m \right\}$$

$$+ E_L e_\ell + F_E f_c$$

$$= Q^* \left\{ \frac{k^* c_w + c_b}{k^* - 1} \right\} \frac{W_L}{Q^*} - M' - B' + C_m - C'_m + PE^* \left\{ \frac{E_L}{PE^*} e_\ell + \frac{F_E}{PE^*} f_c \right\}$$

where

OC = total differential operating and maintenance cost corresponding to maximum power output, \$/year

OC_S = differential operating and maintenance cost of new closed-cycle system, \$/year

OC_R = operating and maintenance cost of replacement energy, \$/year

OC_{EF} = cost of excess fuel consumption, \$/year

Q^* = reference heat rejection rate, kJ/hr

k^* = ratio of maximum permissible concentration of contaminants in the circulating water or blowdown to the concentration in the make-up water

W_L = total annual evaporation from the new system, m^3/year

c_w = unit cost of supply water, \$/m^3

c_b = unit cost of blowdown treatment, \$/m^3 (alternatively, unit cost of damage to the environment due to blowdown release)

M' = make-up water cost of open-cycle system, \$/year

B' = blowdown cost or damage for open-cycle system, \$/year

C_m = annual maintenance cost of closed-cycle system, \$/year

C'_m = annual maintenance cost of open-cycle system, \$/year

PE^* = nameplate energy = P^*(kW) × 8760 (hrs/year), kW-hr/year

E_L = annual energy loss due to backfit, kW-hr/year

e_ℓ = unit cost of replacement energy, \$/kW-hr

F_E = annual excess fuel consumption, kW-hr-th/year

f_c = unit cost of fuel, \$/kW-hr-th

Differential Total Costs

$$TC = OC + CC \times FCR, \quad \$/\text{year} \tag{17}$$

$$tc = \frac{TC}{PE^*} \tag{20}$$

where TC = levelized annual differential cost of backfitting, \$/year

FCR = fixed charge rate

tc = unit excess cost of energy production resulting from backfit, \$/kW-hr or mills/kW-hr

APPENDIX II

LIST OF INPUTS TO COMPUTER PROGRAMS

POWER PLANT CHARACTERISTICS

(1) Turbine Characteristics

(a) HR, Heat rejection rate matrix (10^9 Btu/hr)

(b) IHR and IPMAX, Number of rows of heat rejection rate matrix

(c) NPL, Number of columns of heat rejection rate matrix

(d) TBP, Design turbine back pressure (in. Hg abs)

(e) TPMAX, Maximum turbine back pressure (in. Hg abs)

(f) TLOW, Lowest turbine temperature in heat rejection rate matrix (^oF)

(g) FINC, Increment of temperature in heat rejection rate matrix, (^oF)

(h) EF, Base turbine efficiency for HR matrix

(i) EN, Alternative base turbine efficiency

(2) Plant Capacity

(a) PLMAX, Maximum plant capacity (MW)

(b) DPL, Power interval (=1/10 maximum plant capacity (MW)

(c) PLMIN, Minimum plant capacity (MW)

(d) LP, Power level

(e) TTDD, Design terminal temperature difference (^oF)

(f) IPLI, Initial power level index

(g) IPLF, Final power level index

(h) MM, Interval of power level index

(i) CF, Fraction of full load

(j) EFI, in-plant efficiency

(k) NYEAR, Remaining life of plant (years)

SITE CHARACTERISTICS

(1) TDBD, Design Dry-Bulb Temperature (^oF)

(2) TWBD, Design Wet-Bulb Temperature (^oF)

(3) TDB10, Extreme Dry-Bulb Temperature (^oF)

(4) TWB10, Extreme Wet-Bulb Temperature (^oF)

(5) PERCEN, Temperature Distribution (Fraction)

(6) ITWBI, Initial Wet-Bulb Temperature (^oF)

(7) ITWBF, Final Wet-Bulb Temperature (^oF)

(8) ITDBF, Final Dry-Bulb Temperature (^oF)

(9) ITBD, Temperature Interval (^oF)

(10) TWBREF, Reference Wet-Bulb Temperature (^oF)

(11) FOGL, Upper Limit of Medium Fogging (^oF-lb H_2O/lb air)

(12) FOGM, Upper Limit of Medium Fogging (^oF-lb H_2O/lb air)

ECONOMIC PARAMETERS

(1) Unit Costs
    (a) UNCOND, Condenser ($/ft^2)
    (b) FC, Fuel ($/kW-hr)
    (c) WC, Water ($/1000 gal)
    (d) WW, Waste water treatment ($/1000 gal)
    (e) UCAPAB, Capability (capacity) loss ($/MW)
    (f) UENER, Replacement energy loss ($/MW-hr)
    (g) ULAND, Land ($/acre)
    (h) UDOWN, Downtime cost ($/kW-hr)

(2) PAPCOS, Pump and Pipe System Cost ($)

(3) DAYS, Downtime (days)

(4) FCR, Fixed Charge Rate Array

(5) CCO, Salvage Value of Old Condenser ($)

(6) COO, Salvage Value of Other Open-Cycle System Components ($)

(7) CHT, Hook-up and Testing Cost ($)

(8) CWATEO, Makeup Water Cost with Open-Cycle System ($)

(9) CBLOWO,-Blowdown Treatment Cost with Open-Cycle System ($)

(10) CMAINO, Maintenance Cost of Open-Cycle System ($)

BASIC THERMODYNAMIC AND OTHER INPUTS

(1) Psychrometric Data
    (a) PSA, Saturated Vapor Pressure (psia)
    (b) DAIR, Density of Dry Air (lb_m/ft^3)
    (c) QHSUM, Cumulative area under saturation curve

(^{o}F-lb $H_2$0/lb air)

(d) REFSV, Specific volume of air (ft^3/lb)

(e) PATM, Atmosphere pressure (psia)

(f) CA, Specific heat of air (Btu/lb$_m$/^{o}F)

(g) CW, Specific heat of water (Btu/lb$_m$/^{o}F)

2. Other Information and Program Control Parameters

(a) CCNO, Concentration of solids in makeup water (ppm)

(b) CCN1, Maximum allowable limit of concentration of solids (ppm)

(c) LOCATI, LOCATF, Geographical location parameters

(d) IWRITE, Parameter controlling printout of information

(e) IPUNCH, Parameter controlling output on cards

(f) ITPMAX, Flag for maximum turbine back pressure limitation (=0, no; = 1, yes)

(g) IEXTRA, Flag for cooling system cost of backfit or new plant (=0, new; = 1, backfit)

(h) INUCAL, Parameter controlling base turbine efficiency (=0, base turbine eff. = EF; = 1, base turbine eff. = EN)

(i) NEWCON, Flag for use of new condenser (=0, no; = 1, yes)

COOLING SYSTEM DATA

(1) Mechanical-Draft Cooling Tower

(a) NNOTSI, NNOTS, Parameter for different tower heights

(b) NOWTSI, NOWTS, Parameter for different tower lengths

(c) RFT, Rating factor matrix

(d) DTWB, Wet-bulb temperature interval in rating factor matrix (^{o}F)

(e) NOITT, Number of iterations in NTU calculations

(f) Tower size

(i) ELENG, Total tower length (ft)

(ii) FT, Tower height (ft)

(iii) WIDTHW, Tower width (ft)

(iv) DIAM, Fan diameter (ft)

(v) DMIN, Distance between adjacent fans (ft)

(g) Tower parameters

(i) AW, Proprietary pile coefficient

(ii) BW, Proprietary pile coefficient

(h) EFFICW, Water pump efficiency

(i) HEIGHT, Pumping height (ft)

(j) HW, Tower operation parameter (ft)

(=0, tower not operating; = HEIGHT, tower operating)

(k) AFRL, Air flow rate loading (lb/hr/ft^2-face area)

(l) WFRL, Water flow rate loading (gpm/ft^2-plan area)

(m) SLANDA, Specific land area (acres/MW)

(n) FANPOW, Fan power (hp)

(o) WPDRO, Static pressure drop across pile (proprietary), (in. H_2O abs)

(p) UO, Overall heat transfer coefficient (Btu/hr/ft^2/°F)

(q) WTCOS, Unit cost of wet tower ($/tower unit)

(r) AMANT, Tower maintenance cost ($/tower cell)

(2) Natural-Draft Cooling Tower

(a) NNOTSI, NNOTS, Parameters for different shell heights

(b) NOWTSI, NOWTS, Parameter for different pile heights

(c) UC, Unit cost matrix ($/1000 Btu)

(d) DRH, Relative humidity in unit cost matrix

(e) NOITT, Number of iterations in NTU calculations

(f) FRIFAC, Smooth pipe curve from Moody chart

(g) Tower size

(i) WIDTHW, Pile width (ft)

(ii) BASEDI, Base diameter (ft)

(iii) HTND, Total tower height (ft)

(iv) FILLHT, Pile height (ft)

(h) ELEV, Elevation of site from sea level (ft)

(i) Tower parameters

(i) AW, Proprietary pile coefficient

(ii) BW, Proprietary pile coefficient

(j) EFFICW, Water pump efficiency

(k) HEIGHT, Pumping height (ft)

(l) AFRL, Air flow rate loading (lb/hr/ft^2-face area)

(m) WFRL, Water flow rate loading (gpm/ft^2-plan area)

(n) SLANDA, Specific land area (acres/MW)

(o) UO, Overall heat transfer coefficient (Btu/hr/ft^2/°F)

(3) Cooling Ponds

(a) NNOTSI, NNOTS, Parameter for different water loadings

(b) NOWTSI, NOWTS, Parameter for different pond areas

(c) RHO, Specific weight of water (lb/ft^3)

(d) C, Specific heat of water (Btu/lb_m/°F)

(e) MONTH, Index for month of year

(f) W2, Wind speed at 2 meters above ground (miles/hr)

(g) QSC, Clear sky solar radiation (Btu/ft^2/day)

(h) AMR, Average monthly reflection (fraction)

(i) UPOND, Unit cost of pond ($/acre)

(j) UPUMP, Unit cost of pump and pipe system ($/gpm)

(k) UMAINT, Maintenance cost ($/acre/year)

(l) CLD, Cloud cover ratio

(m) AREAL, Specific pond area (acre/MW)

(n) GPMLOD, Water flow rate loading (gpm/ft^2)

(4) Spray Canals

(a) UTCOST, Unit cost of modules ($/module)

(b) F, Interference matrix for modules

(c) RP, Design cooling range for water flow rate calculations (°F)

APPENDIX III

FORTRAN LISTING

MECHANICAL-DRAFT WET COOLING TOWER

```
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *   MECHANICAL DRAFT WET COOLING TOWER                                 *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
      DIMENSION HR(50,14),IHR(14),ARWT(10),PSA(250)
     1,PAPCOS(24),FT(10),WTCOST(10),AW(10),BW(10),FCRR(5)
      DIMENSION ELENG(10),WIDTHW(10),NYEAR(5),FCR(11)
      COMMON/DENSIT/ DAIR(250)
      COMMON/INPU/ PERCEN(12,15,2),HW(10),EFI
      COMMON/INPUT4/ WPDRO(13),FILLHT(10),PDROP,WIDTH,FANPOW(20,13)
      COMMON/CMWCT2/ RFT(10,20,6),DTWB(9)
      COMMON/TURBIN/ HR,IHR,TLOW,FINC
      COMMON/TOWERS/ ARWT,AW,BW,GPM,CONST,NOITT,AFR
     1,AIJK,BIJK,RLG,ENTU,N,FNTU,EGPM,GPMW
      COMMON/AIRF/ AFR1,FANW2,PUMOP1
      COMMON/NOT/ NUMTOW,ELENGT,AMANT
      COMMON/CONSTA/ CONST1,DCNST
      COMMON/TURB/ POWER,TTDO,TET,TTD,TTDKO,TTDO
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,QH1,CH2,HA
      COMMON/TOWERC/ NOTS(10),WTCOST
      COMMON/TEMP/ ITWBI,ITWBF,ITBD,ITDBF
      COMMON/ECONO/ NYEAR,FC,WC,WW,DR,CCNO,CCN1,ANPOWE,HEIGHT,EFFICW
      COMMON/POWERC/ IPLI,IPLF,MM,PL,LP
      COMMON/ATMOS/ PATM
      COMMON/FPL11/ FPLMAX
      COMMON/LOSS/ S,CWATEO,CBLOWO,CMAINO
      COMMON/CELL/ CELLTH,FAWET
      COMMON/WSS/ QHSUM(210),FOGL,FOGM
      COMMON/WITREF/ IWRITE,IPUNCH,REFSV,FAWETT,PAWETT,AFRL,TWBREF
      COMMON/PLEVEL/ PLMAX,UCAPAB,CAPCAP,UENER,ENERLS,TEI,TOTOP
      COMMON/TBPR/ ITPMAX,TETMAX,ICAP
      COMMON/TBPRE/ IPMAX,NPL,DPL,PLMIN
      COMMON/QSTAR/ TQSTAR,CF
C
      READ(5,109) IPMAX,NPL,DPL,PLMIN
  109 FORMAT(2I10,2F10.0)
      READ(5,110) ((HR(I,J),J=1,NPL),I=1,IPMAX)
  110 FORMAT(10F8.5)
      READ(5,101) (PSA(I),I=1,250)
  101 FORMAT(10F8.5)
      READ(5,101) (DAIR(I),I=1,250)
      READ(5,101) (QHSUM(I),I=1,210)
      READ(5,102) (IHR(I),I=1,NPL)
  102 FORMAT(14I3)
      READ(5,501) (((RFT(I,J,K),K=1,6),J=1,20),I=1,10)
  501 FORMAT(12F6.3)
      READ(5,512) (DTWB(I), I=1,9)
  512 FORMAT(9F8.3)
      READ(5,106) (FCR(I),I=1,11)
      READ(5,106) (PAPCOS(I),I=1,24)
  106 FORMAT(10F8.0)
      READ(5,103) ((FANPOW(I,J),J=1,13),I=1,20)
  103 FORMAT(13F6.1)
      READ(5,104) (WPDRO(I),I=1,13)
  104 FORMAT(13F6.4)
      READ(5,106) (FT(I),I=1,10)
      READ(5,106) (WIDTHW(I),I=1,10)
      READ(5,106) AFRL
      READ(5,106) GPMW
      READ(5,106) (AW(I),I=1,10)
      READ(5,106) (BW(I),I=1,10)
      READ(5,107) TLOW,FINC
  107 FORMAT(2F10.0)
      READ(5,504) NOITT
      READ(5,503) (ELENG(I),I=1,10)
  503 FORMAT(10F8.0)
      READ(5,555) DIAM,DMIN
  555 FORMAT(2F10.2)
      READ(5,504) NNOTSI,NNOTS,NOWTSI,NOWTS,LOCATI,LOCATF
  504 FORMAT(6I4)
      READ(5,106) WTCOS,CHT,SLANDA,CMAINO
      READ(5,502) CCNO,CCN1
  502 FORMAT(2F10.0)
      READ(5,506) (NYEAR(I),I=1,5),FC,WC,WW
  506 FORMAT(5I4,4F10.0)
      READ(5,106) HEIGHT,EFFICW,UNCOND,UO,AMANT
      READ(5,106) (HW(I),I=1,10)
      READ(5,507) ITWBI,ITWBF,ITBD,ITDBF
  507 FORMAT(6I4)
      READ(5,108) LP,TTDO,REFSV,TWBREF,PLMAX,UCAPAB
  108 FORMAT(I10,5F10.0)
      READ(5,106) UENER,ULAND,UDOWN,DAYS,CF,CCO,COC,CWATEO,CBLOWO
      READ(5,507) IPLI,IPLF,MM
```

```
      NTW=(ITWBF-ITWBI)/ITBD+1
      NTD=(ITDBF-ITWBI)/ITBD+1
      NKN=(IPLF-IPLI+0.01)/MM+1
      READ(5,509) PATM,TPMAX
  509 FORMAT(3F10.0)
      READ(5,513) FOGL,FOGM
  513 FORMAT(2F10.0)
      READ(5,505) CA,CW1
  505 FORMAT(2F10.2)
      READ(5,509) TBP
      READ(5,509) EF,EN,EFI
      READ(5,507) IWRITE,IPUNCH,ITPMAX,IEXTRA,INUCAL,NEWCON
C
      CONST=7.481/60./62.*10.**9
      DONST=CONST
      CONST1=0.124683/62.
C
C  CALCULATE CORRESPONDING FIXED CHARGE RATE
C
      DO100K=1,5
      Y=NYEAR(K)/4.+1.
      IY=Y
  100 FCRR(K)=FCR(IY)+(FCR(IY+1)-FCR(IY))*(Y-IY)
C
      IF(INUCAL.EQ.0)GOTO300
      PARAME=(1.-EN)*EF/(1.-EF)/EN
      DO400I=1,IPMAX
      DO400J=1,NPL
  400 HR(I,J)=HR(I,J)*PARAME
C
C  FIND FUEL CONSUMPTION WITH OPEN-CYCLE COOLING SYSTEM
C
  300 TBP1=TBP*62.4*13.6/1728.
      NP=(CF-0.49)*10.
      IT=TLOW
      IF(TBP1.GT.PSA(IT))GOTO710
      TQSTAR=HR(1,LP)
      TQST1=HR(1,NP)
      TQST2=HR(1,NP+1)
      GOTO716
  710 IT=IT+5
      IF(TBP1.GT.PSA(IT))GOTO710
  714 IT=IT-1
      IF(TBP1.LT.PSA(IT))GOTO714
      TTEMP=IT+(TBP1-PSA(IT))/(PSA(IT+1)-PSA(IT))
      TTEMP=(TTEMP-TLOW)/2.+1
      IT=TTEMP
      TQSTAR=HR(IT,LP)+(HR(IT+1,LP)-HR(IT,LP))*(TTEMP-IT)
      TQST1=HR(IT,NP)+(HR(IT+1,NP)-HR(IT,NP))*(TTEMP-IT)
      TQST2=HR(IT,NP+1)+(HR(IT+1,NP+1)-HR(IT,NP+1))*(TTEMP-IT)
  716 DQHR=(TQST1/CF-TQSTAR)/(PLMAX*3.6/1055.04-TQST1/CF+TQSTAR)
      FPL1=PLMAX*CF/(1.+DQHR)
      DQHR=(TQST2/(CF+0.1)-TQSTAR)/(PLMAX*3.6/1055.04-TQST2/(CF+0.1)
     1+TQSTAR)
      FPL2=PLMAX*(CF+0.1)/(1.+DQHR)
      TQST=TQST1+(TQST2-TQST1)*(PLMAX*CF-FPL1)/(FPL2-FPL1)
C
C  DETERMINE MAXIMUM ALLOWABLE TURBINE TEMPERATURE
C
      TETMAX=1000.
      IF(ITPMAX.EQ.0)GOTO717
      TBP2=TPMAX*62.4*13.6/1728.
  718 IT=IT+5
      IF(TBP2.GT.PSA(IT))GOTO718
  719 IT=IT-1
      IF(TBP2.LT.PSA(IT))GOTO719
      TETMAX=IT+(TBP2-PSA(IT))/(PSA(IT+1)-PSA(IT))
C
  717 DO6000LOCATE=LOCATI,LOCATF
      READ(5,509) TWBD,TDBD,TWB10
      READ(5,508) (((PERCEN(I,J,K),J=1,NTD),I=1,NTW),K=1,NKN)
  508 FORMAT(10F8.6)
C
      SS1=0.
      S=0.
      DO2011LP=IPLI,IPLF,MM
      LP1=(LP-IPLI)/MM+1
      PL=PLMAX
      IF(LP.NE.IPLI) PL=PLMAX*CF
      DO2012IJK=ITWBI,ITWBF,ITBD
      IW1=(IJK-ITWBI)/ITBD+1
      ITDBMA=PSA(IJK)/(0.000367*PATM*(1.+(IJK-32.)/1571.))+IJK
      IF(ITDBMA.GT.ITDBF) ITDBMA=ITDBF
      DO2012IIK=IJK,ITDBMA,ITBD
```

```
      ID1=(IIK-ITWBI)/ITBD+1.
 2012 SS1=SS1+PERCEN(IW1,ID1,LP1)
      S=S+SS1*PL
      SS2=SS1
 2011 SS1=0.
      IF(IPLI.EQ.IPLF) SS2=0.
      S=S*8760.
      TEI=PLMAX*(1.-SS2)+PLMAX*CF*SS2+(TQSTAR*(1.-SS2)+TQST*SS2)*293.067
      TEI=TEI/EFI
C
      WRITE(6,610) TWBD,TDBD,TWB10,PLMAX,CA,CW1,TTDD,PATM
  610 FORMAT(1H1///10X,'DESIGN WET-BULB TEMPERATURE OF AIR =',F5.1,' F'
     1/10X,'DESIGN DRY-BULB TEMPERATURE OF AIR =',F5.1,' F'/10X,
     2'EXTREME WET BULB TEMPERATURE =',F8.3,' DEG. F'/10X,
     3'POWER LEVEL =',F6.0,' MW'/10X,
     4'SPECIFIC HEAT OF AIR AT CONSTANT PRESSURE =',F6.2,' BTU/IB./F'/
     510X,'SPECIFIC HEAT OF WATER =',F6.2,' BTU/LB./F'/
     610X,'DESIGN TERMINAL TEMP. DIFFERENCE =',F5.1,' F'/10X,
     7'ATMOSPHERIC PRESSURE =',F7.2,' PSIA')
C
      WRITE(6,620) FC,WC,WW,AMANT,CCN0,CCN1
  620 FORMAT(1H ,9X,'UNIT FUEL COST =',F9.6,' $/KW-HR'/10X,
     1'UNIT SUPPLY WATER COST =',F7.4,' $/1000 GAL'/10X,
     2'UNIT WASTE WATER COST =',F7.4,' $/1000 GAL'/10X,
     3'ANNUAL MAINTENANCE COST =',F7.1,' $/CELL/YEAR'/10X,
     4'MAX. TOLERABLE CONCENTRATION OF PROCESS WATER =',F5.0,' PPM'/10X,
     5'SUPPLY WATER CONCENTRATION =',F5.0,' PPM')
C
      WRITE(6,630) DIAM,DMIN,WTCOS,PLMAX,UCAPAB
  630 FORMAT(1H ,9X,'DIAMETER OF WET TOWER FAN =',F6.2,' FT'/10X,
     1'SPACE BETWEEN TWO WET TOWER FANS =',F5.2,' FT'/10X,
     2'UNIT WET TOWER COST =',F6.2,' $/TOWER UNIT'/10X,
     3'MAXIMUM POWER OUTPUT =',F8.2,' MW'/10X,
     4'UNIT CAPACITY LOSS COST =',F10.2,' $/MW')
C
      WRITE(6,631) UENER,ULAND,SLANDA,UDOWN,DAYS
  631 FORMAT(1H ,9X,'UNIT ENERGY COST =',F7.3,' $/KW-HR'/10X,
     1'UNIT LAND COST =',F8.1,' $/ACRE'/10X,
     2'SPECIFIC LAND AREA =',F5.2,' ACRES/MW'/10X,
     3'REPLACEMENT ENERGY COST DURING DOWNTIME =',F7.4,' $/KW-HR'/10X,
     4'DOWNTIME FOR CONSTRUCTION =',F6.1,' DAYS')
C
      WRITE(6,640) HEIGHT,EFFICW,UNCOND,UO
  640 FORMAT(1H ,9X,'PUMPING HEIGHT OF WATER THROUGH TOWER =',F8.1,
     1' FEET'/10X,'PUMPING EFFICIENCY FOR WATER PUMP =',F7.3/10X,
     2'UNIT CONDENSER COST =',F6.2,' $/SQ. FT.'/10X,
     3'OVERALL CONDENSER COEFFICIENT, U =',F6.1,' BTU/HR/FT2/F')
C
      WRITE(6,662) ITWBI,ITWBF,ITBD,ITDBF
  662 FORMAT(1H ,9X,'INITIAL WET BULB TEMPERATURE =',I4,' DEG. F'/10X,
     1'FINAL WET BULB TEMPERATURE =',I5,' DEG. F'/10X,
     2'INCREMENT OF DRY AND WET BULB TEMPERATURE =',I4,' DEG. F'/10X,
     3'FINAL DRY BULB TEMPERATURE =',I5,' DEG. F')
C
      WRITE(6,650) TLOW,FINC,NOITT,REFSV,FOGL,FOGM
  650 FORMAT(1H ,9X,'LOWEST TEMP. IN TURBINE CHARAC. CHART =',F5.1,
     1' DEG. F'/10X,'TEMP. INCREMENT IN TURBINE CHARAC. MATRIX =',F4.1,
     2' DEG. F'/10X,'NUMBER OF ITERATIONS IN NTU CALCULATION =',I5
     3/10X,'REFERENCE SPECIFIC VOLUME OF AIR =',F7.3,'FT3/LB'/10X,
     4'LOWER BOUND OF LIGHT FOGGING =',F7.3,' LB H2O/LB AIR*F'
     5/10X,'LOWER BOUND OF MEDIUM FOGGING =',F7.3,' LB H2O/LB AIR*F')
C
      IT=TWBD
      TSS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TWBD-IT)
      HAA=0.24*TWBD+0.622*TSS/(PATM-TSS)*(1061.8+0.44*TWBD)
      DO1000II=NNOTSI,NNOTS
      DO1100III=1,10
 1100 FILLHT(III)=FT(II)
      DO1000IW=NOWTSI,NOWTS
      TS=TSS
      HA=HAA
C
C  DETERMINE WATER FLOW RATE FOR EACH WET TOWER
C  DETERMINE AIR-FLOWS FOR WET TOWERS, DETERMINE N.T.U. FOR WET TOWER
C
      NOTS(IW)=IFIX((ELENG(IW)-DIAM+0.1)/(DIAM+DMIN))+1
      ELENGT=ELENG(IW)
      WIDTH=WIDTHW(IW)
      NUMTOW=NOTS(IW)
      CELLTH=ELENGT/NUMTOW
      FAWET=FILLHT(IW)*CELLTH*2.
      FAWETT=FAWET*NUMTOW
      PAWETT=ELENGT*WIDTH
      GPM=GPMW*CELLTH*WIDTH
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      EGPM=GPM*NUMTOW
      AFR1=AFRL*FAWET/60.*REFSV
      AAFR=AFRL/60.*REFSV/50+1
      IAFR=AAFR
      PDROP=WPDRO(IAFR)+(WPDRO(IAFR+1)-WPDRO(IAFR))*(AAFR-IAFR)
C
      WRITE(6,660) TEI,TBP,SS2
  660 FORMAT(1H1,10X,'FUEL CONSUMPTION WITHOUT COOLING SYSTEM =',F9.3,
     1' MW  (TUR. BACK PRE. =',F5.2,' IN.HG)'/10X,
     2'***',F8.5,' OF THE TIME IS NOT OPERATED AT FULL LOADING ***')
C
      AFR=AFR1*60./REFSV
      RLG=GPM/CONST*10.**9/AFR
      FNTU=AW(IW)*GPMW**BW(IW)*FILLHT(IW)
      N=IFIX(FNTU/0.5)
      ENTU=FNTU/N
C
C  DETERMINE CAPITAL COST OF THE TOWER
C
      IIII=-5
      TTD=TTDD
C
      CALL MODELW (TDBD,TWBD,IW,NPL,TET,TQ,TWL,IIII,K,TT,IM)
C
      IF(TET.LT.0)GOTO999
      TTSTAR=1.
      IF(ICAP.EQ.0)GOTO679
C
      CALL POWERS (TET,TQ,TTSTAR)
C
  679 DQHR=(TQ/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TC/TTSTAR+TQSTAR)
      TQQ=(PLMAX*TTSTAR/(1.+DQHR)+TQ*1055.04/3.6)/EFI
      RP=TQ*CONST/EGPM
      AP=TET-TTD-RP-TWBD
      ITET=TET
      PDESI=(PSA(ITET)+(PSA(ITET+1)-PSA(ITET))*(TET-ITET))*1728./848.64
C
      CALL RATFAT (RP,AP,TWBD,RF)
C
      IF(RF.GT.0.01)GOTO677
      IF(AP.GT.15.0)GOTO678
      RF=1.7
      GOTO677
  678 RF=0.4
  677 WTCOST(IW)=WTCOS*RF*GPM
      CAPCO1=WTCOST(IW)*NUMTOW
C
C  DETERMINE CONDENSER COST, AND PUMP AND PIPE SYSTEM COST
C
      THW=TET-TTD
      TCW=THW-TQ*CONST/EGPM
      RANGE=THW-TCW
      TTDKO=TTD/RANGE*CONST/EGPM
      RLL=ALOG((RANGE+TTD)/TTD)
      CONCOS=UNCOND*EGPM/CONST/UO*RLL*10.**9
      IF(NEWCON.EQ.0) CONCOS=CCO
      IP=EGPM/10.**5
      PPCOST=PAPCOS(IP+1)+(PAPCOS(IP+2)-PAPCOS(IP+1))*(EGPM/10.**5-IP)
      PPCOSO=0.20*PPCOST
C
C  DETERMINE DOWNTINE, AND ADDITIONAL LAND COST
C
      DOWNCO=UDOWN*PL*24.*DAYS*1000.
      ALANDC=PLMAX*SLANDA*ULAND
C
C  DETERMINE REPLACEMENT  CAPABILITY LOSS
C
      CALL FAN (AFR1,PDROP,FANW2)
C
      PUMOP1=EGPM*HEIGHT*62.4/7.481/60./550./EFFICW*0.7457
      FANW1=FANW2*NUMTOW*0.7457
      IIII=0
C
      CALL MODELW (TWB10,TWB10,IW,NPL,TET,TQ,TWL,IIII,K,TT,IM)
C
      IF(TET.LT.0.)GOTO999
      TTSTAR=1.
      IF(ICAP.EQ.0)GOTO681
C
      CALL POWERS (TET,TQ,TTSTAR)
C
  681 DQHR=(TQ/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TQ/TTSTAR+TQSTAR)
      FPL=PLMAX*TTSTAR/(1.+DQHR)
      FPL1=PLMAX-FPL
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      FPL1=FPL1+(FANW1+PUMOP1)/1000.
      FPLMAX=FPL1
      CAPCAP=FPL1*UCAPAB
C
      WRITE(6,661) EGPM,AFR,GPM,GPMW,AFRL,PDROP,FNTU
  661 FORMAT(1H0,10X,
     3'TOTAL WATER FLOW RATE =',F11.0,' GPM'/11X,
     4'AIR FLOW RATE THROUGH EACH WET TOWER =',F11.0,' LB./HR'/11X,
     5'WATER FLOW RATE THROUGH EACH WET TOWER =',F11.0,' GPM'/11X,
     6'WATER LOADING =',F11.2,' GPM/SQ. FT. PLAN AREA'/11X,
     7'AIR LOADING =',F13.2,' LB./HR/SQ. FT. FACE AREA'/11X,
     8'PRESSURE DROP DUE TO FAN OPERATING =',F8.4,' IN. H2O'/11X,
     9'TOTAL NUMBER OF TRANSFER UNIT =',F9.4/)
      WRITE(6,665) ELENGT
  665 FORMAT(1H0,8X,'*** TOWER SIZE ***'/11X,
     1'LENGTH OF WET TOWERS =',F13.1,' FT')
      WRITE(6,664) FILLHT(IW),WIDTH,NUMTOW
  664 FORMAT(1H ,10X,'FILL HEIGHT FOR WET SECTION =',F6.1,' FT'/11X,
     2'FILL WIDTH FOR WET SECTION =',F7.1,' FT'/11X,
     3'NUMBER OF WET TOWER FANS =',I9)
      WRITE(6,663)
  663 FORMAT(1H0,8X,'*** DESIGN CONDITIONS ***')
      WRITE(6,666) THW,TCW,RP,AP,RF,PDESI,TQC
  666 FORMAT(1H ,10X,'DESIGN HOT WATER TEMPERATURE =',F8.3,' DEG. F'
     1/11X,'DESIGN COLD WATER TEMPERATURE =',F8.3,' DEG. F'/11X,
     2'DESIGN COOLING RANGE =',F8.3,' DEG. F'/11X,
     3'DESIGN APPROACH =',F8.3,' DEG. F'/11X,
     4'RATING FACTOR =',F8.4/11X,
     5'DESIGN TURBINE BACK PRESSURE =',F8.4,' IN. HG'/11X,
     6'FUEL CONSUMPTION AT DESIGN CONDITION =',F9.2,' MW')
      IF(ICAP.EQ.1) WRITE(6,667)
  667 FORMAT(1H ,12X,'NOTE ... CAPACITY LOSS AT DESIGN CONDITION')
C
C  COMPUTE OPERATION COST AND TOTAL COST
C
      CALL OPECOS (IW,TOTOPE)
C
      IF(TOTOPE.GT.10.**11)GOTO1001
      CAPCOS=CAPCO1+PPCOST-PPCOSO+CONCOS-CCO-COO+CHT+ALANDC+CAPCAP
     1+DOWNCO
C
      WRITE(6,604)
  604 FORMAT(1H0,7X,'*** CAPITAL COSTS ***')
      WRITE(6,602) CAPCO1,PPCCST,PPCOSO,CONCOS,CCO,COO,CHT,ALANDC,
     1CAPCAP,DOWNCO,CAPCOS
  602 FORMAT(1H ,9X,'CAPITAL COST OF TOWERS = $',F20.0/10X,
     1'PUMP AND PIPE SYSTEM COST = $',F17.0/10X,
     2'PUMP AND PIPE SYSTEM SALVAGE = ( $',F12.0,')'/10X,
     3'NEW CONDENSER COST = $',F24.0/10X,
     4'SALVAGE VALUE OF OLD CONDENSER = ( $',F10.0,')'/10X,
     5'OTHER OPEN-CYCLE COMPONENTS SALVAGE = ( $',F5.0,')'/10X,
     6'HOOKUP AND TESTING COST = $',F19.0/10X,
     7'ADDITIONAL LAND COST = $',F22.0/10X,
     8'REPLACEMENT CAPABILITY COST = $',F15.0/10X,
     9'DOWNTIME COST = $',F29.0/40X,'-----------------'/10X,
     1'TOTAL CAPITAL COST = $',F24.0)
C
      IF(IEXTRA.EQ.1) TOTOPE=TOTOP
      IF(IEXTRA.EQ.1) WRITE(6,614)
  614 FORMAT(1H0/12X,'NOTE   OPERATING COSTS ARE BASED ON  EXTRA  OPERAT
     1ING COST')
C
      WRITE(6,612)
  612 FORMAT(1H0,7X,'*** TOTAL COST   --- ANNUAL BASIS --- FIXED CHARGE
     1RATE ***'//10X,'NO. OF YRS',5X,'CAPITAL COST',5X,'ANNUAL OPERATING
     2 COST',5X,'TOTAL COST',5X,'FIXED CHARGE RATE'/26X,
     3'MILLS/KW-HR',10X,'MILLS/KW-HR',9X,'MILLS/KW-HR')
      DO5000KK=1,5
      CAPCO1=CAPCOS*FCRR(KK)/S
      TOTOP2=TOTOPE
      TOTCOS=CAPCO1+TOTOP2
 5000 WRITE(6,611) NYEAR(KK),CAPCC1,TOTOP2,TOTCOS,FCRR(KK)
  611 FORMAT(1H ,I15,F20.7,F21.7,F20.7,F19.6)
      GOTO1000
  999 WRITE(6,603) EGPM,NUMTOW,TET
  603 FORMAT(1H0//10X,'************'/12X,'TOTAL WATER FLOW RATE THROUGH
     1THESE COOLING TOWERS =',F11.0,' GPM'/12X,'NUMBER OF TOWERS =',
     2I5/12X,'TURBINE TEMPERATURE =',F10.4/12X,
     3'TOWER SIZE IS TOO LARGE')
      TOTCOS=10.**12
 1000 CONTINUE
      GOTO6000
 1001 TOTCOS=10.**12
 6000 CONTINUE
```

```
      STOP
      END



      SUBROUTINE OPECOS (IW,TCTOPE)
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *  PROGRAM TO DETERMINE TOTAL ANNUAL OPERATING CCST                     *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
      DIMENSION HR(50,14),IHR(14),ARWT(10),PSA(250),AW(10),BW(10)
     1,WTCOST(10),S1(15),S2(15),S3(15),NYEAR(5)
      COMMON/DENSIT/ DAIR(250)
      COMMON/INPU/ PERCEN(12,15,2),HW(10),EFI
      COMMON/INPUT4/ WPDRO(13),FILLHT(10),PDROP,WICTH,FANPOW(20,13)
      COMMON/TURBIN/ HR,IHR,TLOW,FINC
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,QH1,QH2,HA
      COMMON/TOWERS/ ARWT,AW,BW,GPM,CONST,NOITT,AFR
     1,AIJK,BIJK,RLG,ENTU,N,FNTU,FGPM,GPMW
      COMMON/AIRF/ AFR1,FANW2,PUMOP1
      COMMON/NOT/ NUMTOW,ELENGT,AMANT
      COMMON/TURB/ POWER,TBP,TET,TTD,TTDKO,TTDO
      COMMON/TOWERC/ NOTS(10),WTCCST
      COMMON/CONSTA/ CONST1,DCNST
      COMMON/TEMP/ ITWBI,ITWBF,ITBD,ITDBF
      COMMON/ECONO/ NYEAR,FC,WC,WW,DR,CCNO,CCN1,ANPOWE,HEIGHT,EFFICW
      COMMON/POWERC/ IPLI,IPLF,M,PL,LP
      COMMON/ATMOS/ PATM
      COMMON/LOSS/ S,CWATEO,CBLOWO,CMAINO
      COMMON/CELL/ CELLTH,FAWET
      COMMON/WSS/ QHSUM(210),FOGL,FOGM
      COMMON/WITREF/ IWRITE,IPUNCH,REFSV,FAWETT,PAWETT,AFRL,TWBREF
      COMMON/PLEVEL/ PLMAX,UCAPAB,CAPCAP,UENER,ENERLS,TEI,TOTOP
      COMMON/FPL11/ FPLMAX
      COMMON/TBPR/ ITPMAX,TETMAX,ICAP
      COMMON/TBPRE/ IPMAX,NPL,DPL,PLMIN
      COMMON/QSTAR/ TQSTAR,CF
C
C  OPERATION DUE TO WET COOLING TOWER
C
      IF(FANW2.LT.0.1)HW(IW)=0.0
      IF(HW(IW).LT.0.01 )GOTO1002
C
      IF(IWRITE.EQ.1)WRITE(6,899)
C
      IM=0
      TOTOPE=0.
      TOTBLD=0.
      TOTWL=0.
      TOTEI=0.
      TOTFUE=0.
      TOTWAT=0.
      TOTWAW=0.
      TOTMAN=0.
      TOTLOS=0.
      TOTPRO=0.
      SEN1=0.
      SEN2=0.
      SEN3=0.
      FOGOS=0.
      FOGLS=0.
      FOGMS=0.
      FOGHS=0.
      CAPLOS=0
      CAPRRO=0.
      ENERLS=0.
      FAPLS=0.
C
      AFR=AFR1*60./REFSV
      RLG=GPM/CONST*10.**9/AFR
      FNTU=AW(IW)*GPMW**BW(IW)*FILLHT(IW)
      N=IFIX(FNTU/0.5)
      ENTU=FNTU/N
      DO1000LP=IPLI,IPLF,M
      LP1=(LP-IPLI)/M+1
C
      TTTSAV=0.
      KKSAVE=0
      JJJJ=0
C
      DO901IIJ=ITWBI,ITWBF,ITBD
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      TWB=IIJ
      TS=PSA(IIJ)
      IW1=(IIJ-ITWBI)/ITBD+1
      KK=KKSAVE
      TTT=TTTSAV
      I2=0
      IJ=IIJ
      TDB=TWB
      ITDBMA=PSA(IIJ)/(0.C00367*PATM*(1.+(IIJ-32.)/1571.))+IIJ
      IF(ITDBMA.GT.ITDBF)ITDBMA=ITDBF
      ID1=(IJ-ITWBI)/ITBD+1
      AH=TS-0.000367*PATM*(TDB-TWB)*(1.+(TWB-32.)/1571.)
      WL1=0.622*AH/(PATM-AH)
      QH1=WL1
      FANW1=FANW2*DAIR(IJ)*(1.+WL1)*REFSV*NUMTOW*0.7457
      PLC=PLMAX*CF+(FANW1+PUMOP1)/1000.
      FAP=(FANW1+PUMOP1)/1000.
      NP=NPL
      IF(LP.NE.IPLI) NP=(CF-0.49)*10.
C
      CALL MODELW (TDB,TWB,IW,NP,TET,TQ,TWL,JJJJ,KK,TTT,IM)
C
      I2=I2+1
      IF(TET.LT.0)GOTO200
      TTSTAR=1.
      IF(LP.NE.IPLI) TTSTAR=CF
      IF(ICAP.EQ.0)GOTO9666
C
      CALL POWERS (TET,TQ,TTSTAR)
C
      CAPPRO=CAPPRO+PERCEN(IW1,ID1,LP1)
 9666 DQHR=(TQ/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TC/TTSTAR+TQSTAR)
      FPL=PLMAX*TTSTAR/(1.+DQHR)
      PL=FPL
      IF(LP.EQ.IPLI)GOTO96680
      IF(ICAP.EQ.0)GOTO96681
      FPL1=CF*PLMAX-FPL
      PL=FPL
      FAP=(FANW1+PUMOP1)/1000.
      GOTO96682
96681 TETST=TET
      TQST=TQ
      FPLST=FPL
      TWLST=TWL
      NP=NP+1
      KKST=KK
      TTTST=TTT
C
      CALL MODELW (TDB,TWB,IW,NP,TET1,TQ1,TWL1,JJJJ,KKST,TTTST,IM)
C
      TTSTAR=CF+0.1
      IF(ICAP.EQ.0)GOTO96683
C
      CALL POWERS (TET1,TQ1,TTSTAR)
C
96683 DQHR=(TQ1/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TQ1/TTSTAR+TQSTAR)
      FPL=PLMAX*TTSTAR/(1.+DQHR)
C
      TET=TETST+(TET1-TETST)/(FPL-FPLST)*(PLC-FPLST)
      IF(TET.LT.(TETMAX+0.05))GOTO96684
      FPL1=CF*PLMAX-FPL
      TQ=TQ1
      TET=TETMAX
      TWL=TWL1
      PL=FPL
      FAP=(FANW1+PUMOP1)/1000.
      GOTO96682
96684 TQ=TQST+(TQ1-TQST)/(FPL-FPLST)*(PLC-FPLST)
      TWL=TWLST+(TWL1-TWLST)/(FPL-FPLST)*(PLC-FPLST)
      FPL1=0.
      FAP=0.
      PL=PLMAX*CF+(FANW1+PUMOP1)/1000.
      GOTO96685
96680 FPL1=PLMAX-FPL
96682 FPL1=FPL1+(FANW1+PUMOP1)/1000.
96685 ENERLS=ENERLS+FPL1*PERCEN(IW1,ID1,LP1)*8760.
      FAPLS=FAPLS+FAP*PERCEN(IW1,ID1,LP1)*8760.
      EI=(PL+TQ*1.05504/3.6*1000.)*1000./EFI
      FUECOS=FC*EI*8760.
      BLDOWN=TWL*CCN1/(CCNO-CCN1)
      WATCOS=(TWL+BLDOWN)*WC*525.6
      WAWACO=BLDOWN*WW*525.6
      ANUCAP=ENERLS*UENER*1000.
      HOTWTT=TET-TTD
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      COLDWT=HOTWTT-TQ*CONST/FGPM
      ITET=TET
      P=(PSA(ITET)+(PSA(ITET+1)-PSA(ITET))*(TET-ITET))/0.491111
      TOTBLD=TOTBLD+BLDOWN*PERCEN(IW1,ID1,LP1)*60.*24.*365./326046
      TOTWL=TOTWL+TWL*PERCEN(IW1,ID1,LP1)*60.*24.*365./326046
      EI1=EI/1000.
      TOTEI=TOTEI+EI1*PERCEN(IW1,ID1,LP1)
      TOTPRO=TOTPRO+PERCEN(IW1,ID1,LP1)
      TOTFUE=TOTFUE+FUECOS*PERCEN(IW1,ID1,LP1)
      TOTWAT=TOTWAT+WATCOS*PERCEN(IW1,ID1,LP1)
      TOTWAW=TOTWAW+WAWACO*PERCEN(IW1,ID1,LP1)
      AMANT1=AMANT*NUMTOW*PERCEN(IW1,ID1,LP1)
      TOTMAN=TOTMAN+AMANT1
      FUECIS=FUECOS*PERCEN(IW1,ID1,LP1)
      WATCOS=WATCOS*PERCEN(IW1,ID1,LP1)
      WAWACO=WAWACO*PERCEN(IW1,ID1,LP1)
      OPCOS=(FUECIS+WATCOS+WAWACO+AMANT1)
     1+FPL1*PERCEN(IW1,ID1,LP1)*8760.*UENER*1000.
      TOTOPE=TOTOPE+OPCOS
      IF(PERCEN(IW1,ID1,LP1).LT.0.000001)GOTO312
C
      CALL FOGSEN (TDB,TWB,TWBAL,QH2,SENSI1,SENSI2,SENSI3)
C
      SEN1=SEN1+SENSI1*PERCEN(IW1,ID1,LP1)
      SEN2=SEN2+SENSI2*PERCEN(IW1,ID1,LP1)
      SEN3=SEN3+SENSI3*PERCEN(IW1,ID1,LP1)
      IF(SENSI3.LT.0.00005)GOTO317
      IF(SENSI3.LT.FOGL)GOTO315
      IF(SENSI3.LT.FOGM)GOTO316
      FOGHS=FOGHS+PERCEN(IW1,ID1,LP1)
      GOTO311
  315 FOGLS=FOGLS+PERCEN(IW1,ID1,LP1)
      GOTO311
  316 FOGMS=FOGMS+PERCEN(IW1,ID1,LP1)
      GOTO311
  312 SENSI1=0.
      SENSI2=0.
      SENSI3=0.
  317 FOGOS=FOGOS+PERCEN(IW1,ID1,LP1)
  311 S1(ID1)=SENSI1
      S2(ID1)=SENSI2
      S3(ID1)=SENSI3
      IF(IPUNCH.EQ.1) WRITE(7,701) FUECOS,ANUCAP,TWL,OPCOS,BLDOWN,FPL1,
     1SENSI1,SENSI2,SENSI3,EI,IM
  701 FORMAT(2F10.0,F6.0,F10.C,F5.0,F9.4,F7.5,F6.4,F8.5,F8.0,I1)
  200 IF(IM.LT.2)GOTO475
      IF(I2.NE.1)GO TO 902
      TTTSAV=TTT
      KKSAVE=KK
      GO TO 902
  475 DO9233IJK=IIJ,ITDBMA,ITBD
      IF(IWRITE.EQ.1)WRITE(6,666)
  666 FORMAT(5X,'WET COOLING TOWER IS TOO LARGE TO OPERATE')
      ID1=(IJK-ITWBI)/ITBD+1
      TOTLOS=TOTLOS+PLMAX*8760000.*UENER*PERCEN(IW1,ID1,LP1)
      TOTOPE=TOTOPE+PLMAX*8760000.*UENER*PERCEN(IW1,ID1,LP1)
      FOGOS=FOGOS+PERCEN(IW1,ID1,LP1)
      IM=0
      IF(IPUNCH.EQ.1)WRITE(7,702) IM
  702 FORMAT(I80)
 9233 CONTINUE
      GO TO 901
  902 IF(IWRITE.LT.1)GOTO90222
      WRITE(6,601) PL,TWB
  601 FORMAT(1H0,5X,'POWER =',F5.0,' MW',10X,'TWB =',F8.3,' DEG. F')
      WRITE(6,333)TET,HOTWTT,COLDWT,P,TQ
333   FORMAT(/6X,'TURB.TEMP. = ',F10.4,1X,'DEG.F. ',
     15X,'HOT WATER TEMP. = ',F10.4,1X,'DEG.F.',
     25X,'COLD WATER TEMP. = ',F10.4,1X,'DEG.F.',
     3//,6X,'PRESSURE = ',F8.5,1X,'IN.HG.',
     45X,'HEAT REJECTION = ',F8.5,1X,'BTU*10**9',/)
      WRITE(6,602)
  602 FORMAT(1H0,1X,'TDB',3X,'WATER EVA.',3X,'BLOWDOWN',3X
     1,'PROBABILITY',3X,'FUEL COST',3X,
     2'WATER COST',3X,'WASTE WATER COST',3X,'SUBS ENERGY LOSS',3X,
     3'OPERATING COST'/3X,'F',
     47X,'GPM',9X,'GPM',22X,'$/YEAR',6X,
     5'$/YEAR',10X,'$/YEAR',13X,'$/YEAR',12X,'$/YEAR'/)
      WRITE(6,607)IJ ,TWL,BLDOWN,PERCEN(IW1,ID1,LP1),FUECIS,WATCOS,WAWAC
     1O,ANUCAP,OPCOS,FPL1
90222 IJ=IIJ+ITBD
      IF(IJ.GT.ITDBMA)GO TO 911
      DO9231IJK=IJ,ITDBMA,ITBD
      TDB=IJK
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      ID1=(IJK-ITWBI)/ITBD+1
      AH=TS-0.000367*PATM*(TDB-TWB)*(1.+(TWB-32.)/1571.)
      WL=(0.622*TPS/(PATM-TPS)-0.622*AH/(PATM-AH))*AFR*CONST/10.**9
     1*NUMTOW
      BLDOWN=WL*CCN1/(CCNO-CCN1)
      WATCOS=(WL+BLDOWN)*WC*525.6
      WAWACO=BLDOWN*WW*525.6
      TOTBLD=TOTBLD+BLDOWN*PERCEN(IW1,ID1,LP1)*60.*24.*365./326046
      TOTWL=TOTWL+WL*PERCEN(IW1,ID1,LP1)*60.*24.*365./326046
      EI1=EI/1000.
      TOTEI=TOTEI+EI1*PERCEN(IW1,ID1,LP1)
      ENERLS=ENERLS+FPL1*PERCEN(IW1,ID1,LP1)*8760.
      FAPLS=FAPLS+FAP*PERCEN(IW1,ID1,LP1)*8760.
      IF(ICAP.EQ.1) CAPPRO=CAPPRO+PERCEN(IW1,ID1,LP1)
      ANUCAP=ENERLS*UENER*1000.
      TOTPRO=TOTPRO+PERCEN(IW1,ID1,LP1)
      TOTFUE=TOTFUE+FUECOS*PERCEN(IW1,ID1,LP1)
      TOTWAT=TOTWAT+WATCOS*PERCEN(IW1,ID1,LP1)
      TOTWAW=TOTWAW+WAWACO*PERCEN(IW1,ID1,LP1)
      AMANT1=AMANT*NUMTOW*PERCEN(IW1,ID1,LP1)
      TOTMAN=TOTMAN+AMANT1
      FUECIS=FUECOS*PERCEN(IW1,ID1,LP1)
      WATCOS=WATCOS*PERCEN(IW1,ID1,LP1)
      WAWACO=WAWACO*PERCEN(IW1,ID1,LP1)
      OPCOS=(FUECIS+WATCOS+WAWACO+AMANT1)
     1+FPL1*PERCEN(IW1,ID1,LP1)*8760.*UENER*1000.
      TOTOPE=TOTOPE+OPCOS
      IF(PERCEN(IW1,ID1,LP1).LT.0.000001)GOTO322
C
      CALL FOGSEN (TDB,TWB,TWBAL,QH2,SENSI1,SENSI2,SENSI3)
C
      SEN1=SEN1+SENSI1*PERCEN(IW1,ID1,LP1)
      SEN2=SEN2+SENSI2*PERCEN(IW1,ID1,LP1)
      SEN3=SEN3+SENSI3*PERCEN(IW1,ID1,LP1)
      IF(SENSI3.LT.0.00005)GOTO327
      IF(SENSI3.LT.FOGL)GOTO325
      IF(SENSI3.LT.FOGM)GOTO326
      FOGHS=FOGHS+PERCEN(IW1,ID1,LP1)
      GOTO321
  325 FOGLS=FOGLS+PERCEN(IW1,ID1,LP1)
      GOTO321
  326 FOGMS=FOGMS+PERCEN(IW1,ID1,LP1)
      GOTO321
  322 SENSI1=0.
      SENSI2=0.
      SENSI3=0.
  327 FOGOS=FOGOS+PERCEN(IW1,ID1,LP1)
  321 S1(ID1)=SENSI1
      S2(ID1)=SENSI2
      S3(ID1)=SENSI3
      IF(IPUNCH.EQ.1) WRITE(7,701) FUECOS,ANUCAP,WL,OPCOS,BLDOWN,FPL1,
     1SENSI1,SENSI2,SENSI3,EI,IM
      IF(IWRITE.LT.1)GOTO923
      WRITE(6,607)IJK,WL,BLDOWN,PERCEN(IW1,ID1,LP1),FUECIS,WATCOS,WAWACO
     1,ANUCAP,OPCOS,FPL1
  607 FORMAT(1H ,I3,F15.5,F11.4,F13.6,F13.0,F12.1,F16.1,F20.3,F18.1,
     1F10.3)
  923 CONTINUE
  911 IF(IWRITE.LT.1)GOTO901
      WRITE(6,69699)
69699 FORMAT(1H0//5X,'***** FOGGING PARAMETERS *****'//3X,
     1'TDB',3X,'TDB(EXIT)',3X,'SPE. HUMID.',6X,
     2'SENSIBILITY',3X,'FOGGING ANGLE',3X,'FOGGING MAG.'/4X,
     3'F',8X,'F',7X,'= H2O/= AIR',5X,'WESTINGHOUSE',3X,'RAD. (MARLEY)')
      DO310IJK=IIJ,ITDBMA,ITBD
      ID1=(IJK-ITWBI)/ITBD+1
  310 WRITE(6,69698) IJK,TWBAL,QH2,S1(ID1),S2(ID1),S3(ID1)
69698 FORMAT(1H ,I5,F11.3,F13.6,F17.7,F14.5,F17.5)
      WRITE(6,899)
  899 FORMAT(///1X,130('*')////)
  901 CONTINUE
 1000 CONTINUE
      FUELEX=TOTEI-TEI
      TOTFU1=FUELEX*FC*1000.*8760.
      TOTOPE=TOTOPE-CWATEO-CBLOWO-CMAINO
      TOTOP=TOTOPE-TEI*FC*1000.*8760.
      WRITE(6,66667)
66667 FORMAT(1H0,8X,'*** FOGGING PARAMETERS ***')
      WRITE(6,66666) SEN1,SEN2,SEN3,FOGOS,FOGLS,FOGMS,FOGHS
66666 FORMAT(1H ,/10X,'AVERAGE SENSIBILITY OF FOGGING, BASED ON WESTINGH
     1OUSE CALCULATION =',F8.5     /10X,'AVERAGE FOGGING ANGLE, BASED ON
     2MARLEY CALCULATION =',F9.4,' RAD.'/10X,
     3'AVERAGE FOGGING MAGNITUDE =',F10.5,' DEG. F*LB. H2O/LB. AIR'/10X,
     4'PROBABILITY OF NO FOGGING =',F8.5,5X,'LIGHT FOGGING =',F8.5/15X,
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     5'MEDIUM FOGGING =',F8.5,5X,'HEAVY FOGGING =',F8.5)
      TOTEI1=TOTEI/TOTPRO
      TOTWL1=TOTWL/(TOSTAK*1055.04/3.6)
      FUELE1=FUELEX/PLMAX
      FPLMA1=FPLMAX/PLMAX
      ENERL1=ENERLS/(PLMAX*8760.)
      FAPLS1=FAPLS/(PLMAX*8760.)
C
      WRITE(6,605) TOTBLO,TOTWL,TOTWL1,TOTEI,TOTEI1,FUELEX,FUELE1,
     1FPLMAX,FPLMA1,ENERLS,ENERL1,FAPLS,FAPLS1
  605 FORMAT(1H ,/10X,'TOTAL ANNUAL BLOWDOWN =',F15.0,' ACRE-FT/YEAR'/
     110X,'TOTAL ANNUAL WATER EVAP. =',F12.0,' ACRE-FT/YEAR'
     2,5X,'(',F10.5,' )'/10X,
     3'TOTAL ENERGY RATE IN =',F12.3,' MW'/10X,
     4'AVERAGE ENERGY RATE IN DURING ACTUAL POWER PRODUCTION ='
     5,F1..3,' MW'//8X,'*** CAPABILITY LOSSES ***'/10X,
     6'EXCESS FUEL CONSUMPTION =',F9.3,' MW',5X,'(',F9.6,' )'/10X,
     7'MAXIMUM CAPABILITY LOSS =',F9..,' MW',5X,'(',F9.6,' )'/10X,
     8'ENERGY LOSS =',F15.5,' MW-HR',5X,'(',F9.6,' )'/10X,
     9'FAN & PUMP ENERGY LOSS =',F11.4,' MW-HR',5X,'(',F9.6,' )')
      WRITE(6,606) TOTFUE,TOTFU1,ANUCAP,TOTWAT,TOTWAW,TOTMAN,CWATEO
     1,CBLOWO,CMAINO,TOTOPE,TOTOP
  606 FORMAT(1H0/8X,
     1'*** TOTAL ANNUAL COSTS ***'/10X,
     2'TOTAL ANNUAL FUEL COST =',F24.0,' $/YEAR',5X,'OR'/10X,
     3'EXCESS FUEL COST =',F30.0,' $/YEAR'/10X,
     4'TOTAL ANNUAL REPLACEMENT ENERGY LOSS =',F10.0,' $/YEAR'/10X,
     5'TOTAL ANNUAL WATER COST =',F23.0,' $/YEAR'/10X,
     6'TOTAL ANNUAL WASTE WATER COST =',F17.0,' $/YEAR'/10X,
     7'TOTAL ANNUAL MAINTAINANCE COST =',F16.0,' $/YEAR'/10X,
     8'MAKEUP WATER COST WITH OPEN-CYCLE =',F13.0,' $/YEAR'/10X,
     9'BLOWDOWN TREATMENT COST WITH OPEN-CYCLE =',F7.0,' $/YEAR'/10X,
     1'MAINTENANCE COST WITH OPEN-CYCLE =',F14.0,' $/YEAR'/40X,
     2'---------------------------'/10X,
     3'TOTAL ANNUAL OPERATING COST =',F19.0,' $/YEAR'/10X,
     4'EXTRA ANNUAL OPERATION COST =',F19.0,' $/YEAR')
C
      TOTFUE=TOTFUE/S
      TOTWAT=TOTWAT/S
      TOTWAW=TOTWAW/S
      TOTMAN=TOTMAN/S
      ANUCAP=ANUCAP/S
      TOTOP=TOTOP/S
      TOTOPE=TOTOPE/S
      WRITE(6,621) TOTFUE,TOTWAT,TOTWAW,TOTMAN,ANUCAP,TOTOPE,TOTOP
  621 FORMAT(1H0,7X,'*** AVERAGE OPERATING COSTS --- IN MILLS/KW-HR ***'
     1/10X,'AVERAGE FUEL COST =',F22.6,' MILLS/KW-HR'/10X,
     2'AVERAGE WATER COST =',F21.6,' MILLS/KW-HR'/10X,
     3'AVERAGE WASTE WATER COST =',F15.6,' MILLS/KW-HR'/10X,
     4'AVERAGE MAINTAINANCE COST =',F14.6,' MILLS/KW-HR'/10X,
     5'AVERAGE CAPACITY LOSS =',F18.6,' MILLS/KW/HR'/10X,
     6'AVERAGE TOTAL OPERATING COST =',F11.6,' MILLS/KW-HR'/10X,
     7'AVERAGE EXTRA OPERATING COST =',F11.6,' MILLS/KW-HR')
      RETURN
 1002 WRITE(6,623)
  623 FORMAT(1H0/10X,'WET COOLING TOWER IS NOT SUFFICIENT TO OPERATE')
      TOTOPE=10.**12
      RETURN
      END



      SUBROUTINE MODELW (TDB,TWB,IW,LP,TET,TQ,TWL,IIII,K,TT,TM)
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *  THIS SUBROUTINE CALCULATES THE MODELING RELATIONSHIPS FOR POWER   *
C *   PLANT AND COOLING TOWER .   GIVEN WET AND DRY BULB                *
C *   TEMPERATURES, AND WET TOWER SIZE,    THE RESULTS ARE TURBINE     *
C *   EXHAUST TEMPERATURE, AND HEAT REJECTION.                          *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
      DIMENSION HTRJIN(50,14),TENHTR(14),AW(10),BW(10),ARWT(10),PSA(250)
      COMMON/TURBIN/HTRJIN,TENHTR,TLOW,FINC
      COMMON/TOWERS/ ARWT,AW,BW,GPM,CONST,NOITT,AFR
     1,AIJK,BIJK,PLG,ENTU,N,FNTU,FGPM,GPMW
      COMMON/NCALA/ PSA,TPS,TS,TWAL,WH1,WH2,HA
      COMMON/NOT/ NUMTOW,ELENGT,AMANT
      COMMON/TURB/ BL1,BL2,BL3,TTD,TTDKO,TTDO
      COMMON/TBPR/ ITPMAX,TETMAX,ICAP
C
C   IF TWB IS HIGH ENOUGH, THEN COOLING CANNOT TAKE PLACE AT ALL UNTIL
C   TURBINE CONDENSER TEMPERATURE IS HIGHER. THUS, WILL SKIP TO
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C    HIGHER TURBIN  TEMPERATURE.
C
C    IF TWB IS LOW ENOUGH,  COOLING WATER FREEZES, WHICH
C    IS NEVER DESIRED.  THUS WHEN NEVER COOLING WATER WOULD HAVE BEEN
C    COOLED BELOW FREEZING ANYWHERE IN THE CYCLE, NO COOLING IS
C    PERFORMED (IMPLYING ALTERNATE SYSTEM USED IN PRACTICE).
C
C   ASSIGN MODEL PARAMETERS FOR TOWER SECTION
C
        ICAP=0.
        IM=2
        IFR=0
        IFRE=-1
        IF(IIII.GT.0.5)GOTO1000
        WL=0.
        TPS=0.
        TWBAL=0.
        QH2=0.
        IF(IIII.EQ.-5)GOTO991
        TTDO=HTRJIN(1,LP)*TTDKO
        GOTO992
  991   TTDO=TTD
C
C   ASSIGN INITIAL TRIAL TURBINE TEMPERATURE
C
  992   K=0
   99   IFRE=IFRE+1
  201   K=K+1
        IF(K.GT.IENHTR(LP))GOTO999
        TT=TLOW+(K-1)*FINC
        IF(IIII.EQ.-5)GOTO404
        TTD=HTRJIN(K,LP)*TTDKO
  404   TT1=TT-TTD
C
C   COOL THROUGH COOLING SYSTEM IF POSSIBLE TO GET TQ1
C
        IF(TWB.LT.TT1)GOTO403
        GOTO201
  403   WL2=WL
        TPS2=TPS
        TWBAL2=TWBAL
        QH22=QH2
        IF(TT.GT.(TETMAX+1.95))GOTO999
C
        CALL NTUCAL (TT1,TDB,TWB,RLG,ENTU,TT2,NOITT,AFR,WL,N)
C
        IF(TT2.GT.32.)GOTO503
        GOTO99
C
C   DETERMINE DIRECTION OF APPROACH TO INTERSECTION OF CURVES
C
  503   TQ1=(TT1-TT2)*FGPM/CONST
        IF(TQ1.LT.HTRJIN(K,LP))GOTO100
C
C   REACH INTERSECTION BY DECREASING TURBINE TEMPERATURE
C
        IF(IFRE.GT.0.)GOTO206
        IF(TWB.GT.(TLOW-FINC-TTDO))GOTO104
C
C   COOLING CURVE ENDS MORE THAN 1 DECREMENT BELOW TLOW
C
  206   TT=TT-FINC
        TT1=TT-TTDO
        IF(TT1.LT.32.)GOTO703
        IF(TWB.GT.TT1)GOTO304
C
C   COOLING THROUGH COOLING SYSTEM IF POSSIBLE TO GET TQ2
C
        WL2=WL
        TPS2=TPS
        TWBAL2=TWBAL
        QH22=QH2
        IF(TT.GT.(TETMAX+1.95))GOTO999
C
        CALL NTUCAL (TT1,TDB,TWB,RLG,ENTU,TT2,NOITT,AFR,WL,N)
C
        IF(TT2.GT.32.)GOTO505
        IFR=1
  505   TQ2=(TT1-TT2)*FGPM/CONST
        IF(TQ2.LT.HTRJIN(1,LP))GOTO105
        IF(IFR.EQ.1)GOTO703
        TQ1=TQ2
        GOTO206
C
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C  INTERPOLATE FOR TQ,TET,TWL
C
  105 TQ=HTRJIN(1,LP)
      HTDIF1=(TQ-TQ2)/(TQ1-TQ2)
      TET=TT+HTDIF1*FINC
      IF(IIII.EQ.-5)GOTO405
      TTD=TQ*TTDKO
  405 TWL=(WL+HTDIF1*(WL2-WL))*NUMTOW
      TPS=TPS+HTDIF1*(TPS2-TPS)
      TWBAL=TWBAL+HTDIF1*(TWBAL2-TWBAL)
      QH2=QH2+HTDIF1*(QH22-QH2)
      IF(IFR.EQ.0)GOTO210
C
C  (PREVIOUS TOWER COOLING INDICATES THAT THE OPERATING CHARACTERISTICS
C   CURVE FOR THE COOLING SYSTEM ENDS IN THE SAME TEMPERATURE INTERVAL
C   AS TET)
C  COOL THROUGH COOLING SYSTEM USING TET, TQ FOR CHECK
C
  211 TT1=TET-TTD
      IF(TT1.GT.(TSTMAX+1.95))GOTO999
C
      CALL NTUCAL (TT1,TDB,TWB,RLG,ENTU,TT2,NOITT,AFR,WL,N)
C
      IF(TT2.GT. 32.)GOTO210
      GOTO703
  210 CONTINUE
      IIII=1
      RETURN
  304 TT=TT+FINC
C
C  COOLING CURVES END JUST BELOW TT AND DECREMENTING TURBINE TEMPERATURE
C   WILL NOT INTERSECT IT
C
C  DETERMINE PROPER VALUE OF HTRJIN(1,LP)
C  DOUBLE INTERPOLATE FOR TQ,TET,TWL
C
  104 IF(K.GT.1)GOTO106
      HQ=HTRJIN(1,LP)
      GOTO107
  106 HQ=HTRJIN(K-1,LP)
  107 IF(IIII.GT.-5)GOTO407
      HQ=HQ+(TWB+TTD-TT+FINC)/FINC*(HTRJIN(K,LP)-HQ)
      GOTO408
  407 HQ=(HQ+(TWB-TT+FINC)/FINC*(HTRJIN(K,LP)-HQ))/(1.-TTDKO/FINC
     1*(HTRJIN(K,LP)-HQ))
  408 IF(IIII.EQ.-5)GOTO406
      TTD=HQ*TTDKO
  406 TQ=TQ1*HQ/(TQ1+HQ-HTRJIN(K,LP))
      TET=TWB+TTD+(TT-TWB-TTD)*TQ/TQ1
      IF(IIII.EQ.-5)GOTO409
      TTD=TQ*TTDKO
  409 TWL=WL/TQ1*TQ*NUMTOW
      TPS=TPS/TQ1*TQ
      TWBAL=TWBAL/TQ1*TQ
      QH2=QH2/TQ1*TQ
      TT=TT-FINC
      IF(K.GT.1)K=K-1
      GOTO211
C
C  REACH INTERSECTION BY INCREMENTING TURBINE TEMPERATURE
C
  100 IF(K.EQ.IENHTR(LP))GOTO999
  108 TT=TT+FINC
      K=K+1
      IF(IIII.EQ.-5)GOTO410
      TTD=HTRJIN(K,LP)*TTDKO
  410 TT1=TT-TTD
C
C  COOL THROUGH SYSTEM TO GET TQ2
C
      WL2=WL
      TPS2=TPS
      TWBAL2=TWBAL
      QH22=QH2
      IF(TT.GT.(TSTMAX+1.95))GOTO999
C
      CALL NTUCAL (TT1,TDB,TWB,RLG,ENTU,TT2,NOITT,AFR,WL,N)
C
      TQ2=(TT1-TT2)*FGPM/CONST
      IF(TQ2.GT.HTRJIN(K,LP))GOTO101
      IF(K.EQ.IENHTR(LP))GOTO999
      TQ1=TQ2
      GOTO108
C
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```
C   INTERPOLATE FOR TQ, TET, TWL
C
  101 HTDIF1=HTRJIN(K,LP)-HTRJIN(K-1,LP)
      HTDIF2=TQ2-TQ1
      TQ=(HTRJIN(K,LP)*HTDIF2-TQ2*HTDIF1)/(HTDIF2-HTDIF1)
      TET=TT-(TQ2-TQ)/HTDIF2*FINC
      IF(IIII.EQ.-5)GOTO411
      TTD=TQ*TTDKO
  411 TWL=(WL2+(WL-WL2)/HTDIF2*(TQ-TQ1))*NUMTOW
      TPS=TPS2+(TPS-TPS2)/HTDIF2*(TQ-TQ1)
      TWBAL=TWBAL2+(TWBAL-TWBAL2)/HTDIF2*(TQ-TQ1)
      QH2=QH22+(QH2-QH22)/HTDIF2*(TQ-TQ1)
      TT=TT-FINC
      K=K-1
      IIII=1
      RETURN
C
C   RETURN WITH MESSAGE
C
  703 TQ=-50
      TET=-50.
      TWL=-50.
      IIII=0
      IM=-50
      RETURN
C
C   FIND INTERSECTION WHEN WET-BULB TERPERATURE INCREASES
C
C   REACH INTERSECTION BY INCREMENTING TURBINE TEMPERATURE
C
 1000 TTD=HTRJIN(K,LP)*TTDKO
      TT1=TT-TTD
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      CALL NTUCAL (TT1,TDB,TWB,RLG,ENTU,TT2,NOITT,AFR,WL,N)
C
      TQ1=(TT1-TT2)*FGPM/CONST
 1004 TT=TT+FINC
      IF(TT.LT.(TLOW+0.001))GOTO1001
      IF(K.EQ.IENHTR(LP))GOTO999
      K=K+1
      HQ=HTRJIN(K,LP)
      HTDIF1=HQ-HTRJIN(K-1,LP)
      GOTO2070
 1001 HQ=HTRJIN(1,LP)
      HTDIF1=0.
 2070 TTD=HQ*TTDKO
      TT1=TT-TTD
      WL2=WL
      TPS2=TPS
      TWBAL2=TWBAL
      QH22=QH2
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      CALL NTUCAL (TT1,TDB,TWB,RLG,ENTU,TT2,NOITT,AFR,WL,N)
C
      TQ2=(TT1-TT2)*FGPM/CONST
      IF(TQ2.GT.HQ)GOTO1010
      TQ1=TQ2
      GOTO1004
C
C   INTERPOLATE FOR TQ, TET, TWL
C
 1010 IF(K.GT.1)GOTO412
      TTD=HTRJIN(1,LP)*TTDKO
      GOTO413
  412 TTD=HTRJIN(K-1,LP)*TTDKO
  413 IF(TWB.LT.(TT-FINC-TTD))GOTO1011
      TQ1=TQ2
      GOTO104
 1011 HTDIF2=TQ2-TQ1
      TQ=(HQ*HTDIF2-TQ2*HTDIF1)/(HTDIF2-HTDIF1)
      TET=TT-(TQ2-TQ)/HTDIF2*FINC
      TTD=TQ*TTDKO
      TWL=(WL2+(WL-WL2)/HTDIF2*(TQ-TQ1))*NUMTOW
      TPS=TPS2+(TPS-TPS2)/HTDIF2*(TQ-TQ1)
      TWBAL=TWBAL2+(TWBAL-TWBAL2)/HTDIF2*(TQ-TQ1)
      QH2=QH22+(QH2-QH22)/HTDIF2*(TQ-TQ1)
      IF(K.GT.1) K=K-1
      TT=TT-FINC
      RETURN
  999 IF(TT1.LT.32.0.OR.TT2.LT.32.)GOTO703
      ICAP=1
      TT1=TETMAX-TTD
```

```
C
      CALL NTUCAL (TT1,TDB,TWB,RLG,ENTU,TT2,NOITT,AFR,WL,N)
C
      TQ=(TT1-TT2)*FGPM/CONST
      TLI=TETMAX
      IF(ITLI.NE.-5)TTD=TQ*TTCKO
      TWL=WL*NUMTOW
      RETURN
      END



      SUBROUTINE NTUCAL (TWI,TDB,TWB,RLG,DNTU,TWO,NOITT,AFR,WL,N)
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *  CROSSFLOW CALCULATION OF COLD WATER TEMPERATURE FOR TOWER          *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
      DIMENSION HW(30),TW(30),PSA(250)
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,DH1,DH2,HA
      COMMON/CONSTA/ CONST1,CONST
      COMMON/ATMOS/PATM
C
      IT=TWI
      PS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TWI-IT)
      H=0.24*TWI+0.622*PS/(PATM-PS)*(1061.8+0.44*TWI)
      IT=TWB
      TS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TWB-IT)
      HA=0.24*TWB+0.622*TS/(PATM-TS)*(1061.8+0.44*TWB)
      AH=TS-0.000367*PATM*(TDB-TWB)*(1.+(TWB-32.)/1571.)
      WL=0.622*AH/(PATM-AH)
      DO 100 I=1,N
      TW(I)=TWI
  100 HW(I)=H
      TWBAL=0.
      DO 104 J=1,N
      H=HA
      DO 101 I=1,N
      KC=0
      DH1=HW(I)-H
      DH=DH1/1.2*DNTU
  102 KC=KC+1
      TW2=TW(I)-DH/RLG
      IT=TW2
      PS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TW2-IT)
      HW2=0.24*TW2+0.622*PS/(PATM-PS)*(1061.8+0.44*TW2)
      DHH=(DH1+HW2-H-DH)/2.*DNTU
      DHH=(DHH+DH)/2.
      IF(KC.GE.NOITT) GO TO 106
      DH=DHH
      GO TO 102
  106 TW(I)=TW(I)-DHH/RLG
      IT=TW(I)
      PS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TW(I)-IT)
      HW(I)=0.24*TW(I)+0.622*PS/(PATM-PS)*(1061.8+0.44*TW(I))
  101 H=H+DHH
      TWB2=TWB
   20 ITWB2=TWB2
      PS=PSA(ITWB2)+(PSA(ITWB2+1)-PSA(ITWB2))*(TWB2-ITWB2)
      HA2=0.24*TWB2+0.622*PS/(PATM-PS)*(1061.8+0.44*TWB2)
      IF (HA2.GE.H) GO TO 10
      TWB2=TWB2+5.
      HA22=HA2
      GO TO 20
   10 TWB2=TWB2-4.
   40 ITWB2=TWB2
      PS=PSA(ITWB2)+(PSA(ITWB2+1)-PSA(ITWB2))*(TWB2-ITWB2)
      HA2=0.24*TWB2+0.622*PS/(PATM-PS)*(1061.8+0.44*TWB2)
      IF(HA2.GE.H) GO TO 30
      TWB2=TWB2+1.
      HA22=HA2
      GO TO 40
   30 TWB2=TWB2-(HA2-H)/(HA2-HA22)
  104 TWBAL=TWBAL+TWB2
      TWBAL=TWBAL/N
      TWO=0.0
      DO103I=1,N
  103 TWO=TWO+TW(I)
      TWO=TWO/N
      IT=TWBAL
      TPS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TWBAL-IT)
      DH2=0.622*TPS/(PATM-TPS)
```

```
      WL=WH2-WL
      WL=WL*AFR*CONST/10.**7
      RETURN
      END



      SUBROUTINE RATFAT (RANGE,APPRO,TWB,RAFA)
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *   DETERMINE RATING FACTOR BY GIVING TWB, APPROACH, AND RANGE      *
C *   - APPLY MARLEY COMPANY'S METHOD -                               *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
      REAL LRFA1,LRFA2,LRF
      COMMON/CMWCI2/ RFT(10,20,6),DTWB(9)
C
      IF(TWB.LT.35.0 .OR. TWB.GT.80.0)GOTO999
      IF(APPRO.LT.8.0 .OR. APPRO.GT.46.0)GOTO999
      IF(RANGE.LT.10.0 .OR. RANGE.GT.35.0)GOTO999
C
C    DETERMINE THE LOWER AND UPPER BOUNDS OF WET-BULB TEMPERATURE
C
      I1=35
      DO 100 I=1,9
      I2=I1+DTWB(I)
      IF(TWB .LE. I2) GO TO 10
100   I1=I2
C
C    DETERMINE THE LOWER AND UPPER BOUNDS OF APPROACH
C
   10 AP=(APPRO-6.)/2.
      J=AP
C
C    DETERMINE THE LOWER AND UPPER BOUNDS OF RANGE
C
      RA=(RANGE-5.)/5.
      K=RA
C
C    INTERPOLATION BETWEEN  RANGE  FOR BOTH TABLES
C
      IF(RFT(I,J,K) .LT. 0.1 .OR. RFT(I,J,K+1) .LT. 0.1
     1.OR. RFT(I,J+1,K) .LT. 0.1 .OR. RFT(I,J+1,K+1) .LT. 0.1
     2.OR. RFT(I+1,J,K) .LT. 0.1 .OR. RFT(I+1,J,K+1) .LT. 0.1
     3.OR. RFT(I+1,J+1,K) .LT. 0.1 .OR. RFT(I+1,J+1,K+1) .LT. 0.1)
     4 GO TO 999
      RAT=RA-K
      LRFA1=RFT(I,J,K)+(RFT(I,J,K+1)-RFT(I,J,K))*RAT
      LRFA2=RFT(I,J+1,K)+(RFT(I,J+1,K+1)-RFT(I,J+1,K))*RAT
      URFA1=RFT(I+1,J,K)+(RFT(I+1,J,K+1)-RFT(I+1,J,K))*RAT
      URFA2=RFT(I+1,J+1,K)+(RFT(I+1,J+1,K+1)-RFT(I+1,J+1,K))*RAT
C
C    INTERPOLATIONS BETWEEN  APPROACH  FOR BOTH TABLES
C
      LRF=LRFA1+(LRFA2-LRFA1)*(AP-J)
      URF=URFA1+(URFA2-URFA1)*(AP-J)
C
C    INTERPOLATION BETWEEN  WET-BULB TEMPERATURE
C
      RAFA=LRF+(URF-LRF)/(I2-I1)*(TWB-I1)
      RETURN
  999 RAFA=0.
      RETURN
      END



      SUBROUTINE FAN (AFR,P,FAN1)
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *  FIND FAN HORSE POWER CORRESPONDING TO THE GIVEN AIR FLOW RATE     *
C *     AND STATIC PRESSURE                                            *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
      COMMON/INPUT4/ WPDRO(13),FILLHT(10),PDROP,WIDTH,FANPOW(20,13)
C
C  DETERMINE THE LOWER BOUNDS OF STATIC PRESSURE AND AIR FLOW RATE
C
      IP=P*20.+1
      A=AFR/100000.+1.
```

```
      IA=A
      IP1=IP+1
      IA1=IA+1
      IF(IP.GT.13)GOTO999
      IF(IA.GT.20)GOTO996
      IF(IP.EQ.13)GOTO999
      IF(IA.EQ.20)GOTO997
C
C  INTERPOLATION BETWEEN 'PRESSURE'
C
      A1=FANPOW(IA,IP)+(FANPOW(IA,IP1)-FANPOW(IA,IP))*(P*20.+1.-IP)
      A2=FANPOW(IA1,IP)+(FANPOW(IA1,IP1)-FANPOW(IA1,IP))*(P*20.+1.-IP)
C
C   INTERPOLATION BETWEEN 'AIR FLOW RATE'
C
      FAN1=A1+(A2-A1)*(A-IA)
      RETURN
  997 FAN1=FANPOW(20,IP)+(FANPOW(20,IP1)-FANPOW(20,IP))*(P*20.+1.-IP)
      RETURN
  998 FAN1=FANPOW(IA,13)+(FANPOW(IA1,13)-FANPOW(IA,13))*(A-IA)
      RETURN
  999 FAN1=0.0
      RETURN
  996 A1=FANPOW(20,IP)+30.*(AFR-1900000.)/100000.
      A2=FANPOW(20,IP1)+30.*(AFR-1900000.)/100000.
      FAN1=A1+(A2-A1)*(P*20.+1.-IP)
      RETURN
      END




      SUBROUTINE FOGSEN (TDB,TWB,TDA,SH2,SENSI1,SENSI2,SENSI3)
C
C* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C    CALCULATES FOGGING PARAMETERS FOR A KNOWN DRY & WET BULB TEMPERATURE      *
C* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
      DIMENSION PSA(250)
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,QH1,QH2,HA
      COMMON/ATMOS/ PATM
      COMMON/WSS/ QHSUM(210),FOGL,FOGM
C
      DT=2.0
      AH=1.-0.000367*PATM/TS*(TDB-TWB)*(1.+(TWB-32.)/1571.)
      SH1=0.622*AH*TS/(PATM-AH*TS)
      IT1=TDB
      PS1=PSA(IT1)+(PSA(IT1+1)-PSA(IT1))*(TDB-IT1)
      SHS1=0.622*PS1/(PATM-PS1)
      QHSUM1=QHSUM(IT1)+(QHSUM(IT1+1)-QHSUM(IT1))*(TDB-IT1)
C
      IT=TDA
      PS2=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TDA-IT)
      SHS2=0.622*PS2/(PATM-PS2)
      QHSUM2=QHSUM(IT)+(QHSUM(IT+1)-QHSUM(IT))*(TDA-IT)
      DSH=(SH2-SH1)/(TDA-TDB)*DT
      SENSI1=0.
      SENSI3=0.
      SHG=SH1-(TDB-IT1)*DSH/DT
      IF(SHS1.LE.SH1+0.0001)GOTO10
      IF(SHS2.LE.SH2+0.0001)GOTO20
C
C  FIND PLUME EVAPORATION AND CONDENSATION POINTS (IF POSSIBLE)
C
      IT2=IT1
   40 IT=IT2+DT
      IF(IT.GT.TDA)GOTO50
      SHS=0.622*PSA(IT)/(PATM-PSA(IT))
      SHG=SHG+DSH
      IF(SHG.GT.SHS)GOTO30
      IT2=IT
      DH=SHS-SHG
      GOTO40
   30 T4=IT2+DH/(DH+SHG-SHS)*DT
      IT4=T4
      PS4=PSA(IT4)+(PSA(IT4+1)-PSA(IT4))*(T4-IT4)
      QH4=0.622*PS4/(14.7-PS4)
      QHSUM4=QHSUM(IT4)+(QHSUM(IT4+1)-QHSUM(IT4))*(T4-IT4)
C
      DH=SHG-SHS
      IT2-IT
   70 IT=IT2+DT
      SHS=0.622*PSA(IT)/(PATM-PSA(IT))
```

```
      SHG=SHG+DSH
      IF(SHG.LT.SHS)GOTO60
      IT2=IT
      DH=SHG-SHS
      GOTO70
   60 T3=IT2+DH/(DH+SHS-SHG)*DT
      IT3=T3
      PS3=PSA(IT3)+(PSA(IT3+1)-PSA(IT3))*(T3-IT3)
      QH3=0.622*PS3/(PATM-PS3)
      QHSUM3=QHSUM(IT3)+(QHSUM(IT3+1)-QHSUM(IT3))*(T3-IT3)
      SENSI1=(T3-T4)/(TDA-TDB)
      SENSI3=0.5*(QH4+QH3)*(T3-T4)-(QHSUM3-QHSUM4)
      GOTO50
C
C  FIND PLUME EVAPORATION POINT ONLY (IF POSSIBLE)
C
   20 IT2=IT1
   90 IT=IT2+DT
      IF(IT.GT.TDA)GOTO50
      SHS=0.622*PSA(IT)/(PATM-PSA(IT))
      SHG=SHG+DSH
      IF(SHG.GT.SHS)GOTO80
      IT2=IT
      DH=SHS-SHG
      GOTO90
   80 T4=IT2+DH/(DH+SHG-SHS)*DT
      IT4=T4
      PS4=PSA(IT4)+(PSA(IT4+1)-PSA(IT4))*(T4-IT4)
      QH4=0.622*PS4/(14.7-PS4)
      QHSUM4=QHSUM(IT4)+(QHSUM(IT4+1)-QHSUM(IT4))*(T4-IT4)
      SENSI1=(TDA-T4)/(TDA-TDB)
      SENSI3=0.5*(QH4+SHS2)*(TDA-T4)-(QHSUM2-QHSUM4)
      GOTO50
C
C  AMBIENT AIR IS SATURATED
C
   10 IF(SHS2.LE.SH2+0.0001)GOTO100
C
C  FIND PLUME CONDENSATION POINT ONLY (IF POSSIBLE)
C
      IT2=IT1
      IT=IT2+DT
      SHS=0.622*PSA(IT)/(PATM-PSA(IT))
      SHG=SHG+DSH
      IF(SHS.GT.SHG)GOTO50
      IT2=IT
      DH=SHG-SHS
  120 IT=IT2+DT
      SHS=0.622*PSA(IT)/(PATM-PSA(IT))
      SHG=SHG+DSH
      IF(SHG.LT.SHS)GOTO110
      IT2=IT
      DH=SHG-SHS
      GOTO120
  110 T3=IT2+DH/(DH+SHS-SHG)*DT
      IT3=T3
      PS3=PSA(IT3)+(PSA(IT3+1)-PSA(IT3))*(T3-IT3)
      QH3=0.622*PS3/(PATM-PS3)
      QHSUM3=QHSUM(IT3)+(QHSUM(IT3+1)-QHSUM(IT3))*(T3-IT3)
      SENSI1=(T3-TDB)/(TDA-TDB)
      SENSI3=0.5*(SHS1+QH3)*(T3-TDB)-(QHSUM3-QHSUM1)
      GOTO50
  100 SENSI1=1.0
      SENSI3=0.5*(SHS1+SHS2)*(TDA-TDB)-(QHSUM2-QHSUM1)
C
C  CALCULATE PLUME ABATEMENT ANGLE (MARLEY'S MANNER)
C
   50 SENSI2=ATAN(DSH/DT)
      RETURN
      END



      SUBROUTINE POWERS (TEM,C,TTSTAR)
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C * DETERMINE THROTTLE LEVEL OF TURBINE FOR A GIVEN CONDITION         *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
      DIMENSION HR(50,14),IHR(14)
      COMMON/TURBIN/ HR,IHR,TLOW,FINC
      COMMON/TBPRE/ IPMAX,NPL,LPL,PLMIN
```

```
      COMMON/PLEVEL/ PLMAX,UCAPAB,CAPCAP,UENER,ENERLS,TEI,TOTCP
C
      IF(TEM.LT.TLOW)GOTO30
      TT=(TEM-TLOW+FINC)/FINC
      IT=TT
      Q11=HR(IT,1)+(HR(IT+1,1)-HR(IT,1))*(TT-IT)
      Q12=HR(IT,2)+(HR(IT+1,2)-HR(IT,2))*(TT-IT)
      IF(Q.LT.Q11)GOTO40
      IF(Q.LT.Q12)GOTO50
      Q1=Q12
      IP=2
   20 Q2=HR(IT,IP+1)+(HR(IT+1,IP+1)-HR(IT,IP+1))*(TT-IT)
      IF(Q2.GT.Q)GOTO10
      Q1=Q2
      IP=IP+1
      GOTO20
   10 PL=IP-(Q2-Q)/(Q2-Q1)+1
      TTSTAR=((PL-1.)*DPL+PLMIN)/PLMAX
      RETURN
   30 TTSTAR=((NPL-1.)*DPL+PLMIN)/PLMAX
      RETURN
   50 IP=1
      Q2=Q12
      Q1=Q11
      GOTO10
   40 DQ=(Q11-Q)/(Q12-Q11)
      TTSTAR=(PLMIN-DQ*DPL)/PLMAX
      RETURN
      END
```

Example Results

Mechanical-draft wet cooling tower

Full loading pattern

```
DESIGN WET-BULB TEMPERATURE OF AIR = 78.0 F
DESIGN DRY-BULB TEMPERATURE OF AIR = 89.0 F
EXTREME WET BULB TEMPERATURE = 83.430 DEG. F
POWER LEVEL = 313. MW
SPECIFIC HEAT OF AIR AT CONSTANT PRESSURE = 0.24 BTU/IB./F
SPECIFIC HEAT OF WATER = 1.00 BTU/LB./F
DESIGN TERMINAL TEMP. DIFFERENCE = 5.0 F
ATMOSPHERIC PRESSURE = 14.70 PSIA
UNIT FUEL COST = C.000751 $/KW-HR
UNIT SUPPLY WATER COST = 0.1000 $/1000 GAL
UNIT WASTE WATER COST = 0.0500 $/1000 GAL
ANNUAL MAINTENANCE COST = 200.0 $/CELL/YEAR
MAX. TOLERABLE CONCENTRATION CF PROCESS WATER = 330. PPM
SUPPLY WATER CONCENTRATION = 100. PPM
DIAMETER OF WET TOWER FAN = 28.00 FT
SPACE BETWEEN TWO WET TOWER FANS = 4.00 FT
UNIT WET TOWER COST = 7.50 $/TOWER UNIT
MAXIMUM POWER OUTPUT = 312.50 MW
UNIT CAPACITY LOSS COST = 90000.00 $/MW
UNIT ENERGY COST = 0.010 $/KW-HR
UNIT LAND COST = 3000.0 $/ACRE
SPECIFIC LAND AREA = 0.10 ACRES/MW
REPLACEMENT ENERGY COST DURING DOWNTIME = 0.0070 $/KW-HR
DOWNTIME FOR CONSTRUCTION = 30.0 DAYS
PUMPING HEIGHT OF WATER THROUGH TOWER = 75.0 FEET
PUMPING EFFICIENCY FOR WATER PUMP = 0.782
UNIT CONDENSER COST = 4.00 $/SQ. FT.
OVERALL CONDENSER COEFFICIENT, U = 630.0 BTU/HR/FT2/F
INITIAL WET BULB TEMPERATURE = 5 DEG. F
FINAL WET BULB TEMPERATURE = 100 DEG. F
INCREMENT CF DRY AND WET BULB TEMPERATURE = 10 DEG. F
FINAL DRY BULB TEMPERATURE = 110 DEG. F
LOWEST TEMP. IN TURBINE CHARAC. CHART = 60.0 DEG. F
TEMP. INCREMENT IN TURBINE CHARAC. MATRIX = 2.0 DEG. F
NUMBER CF ITERATIONS IN NTU CALCULATION = 2
REFERENCE SPECIFIC VOLUME OF AIR = 13.333FT3/LB
LOWER BOUND OF LIGHT FOGGING = 0.400 LB H2O/LB AIR*F
LOWER BOUND OF MEDIUM FOGGING = 1.350 LB H2O/LB AIR*F
```

```
 FUEL CONSUMPTION WITHOUT COOLING SYSTEM = 1026.947 MW  (TUR. BACK PRE. = 1.00 IN.HG)
*** 0.0      OF THE TIME IS NOT OPERATED AT FULL LOADING ***

 TOTAL WATER FLOW RATE =      180000. GPM
 AIR FLOW RATE THROUGH EACH WET TOWER =     5399994. LB./HR
 WATER FLOW RATE THROUGH EACH WET TOWER =      15000. GPM
 WATER LOADING =       12.50 GPM/SQ. FT. PLAN AREA
 AIR LOADING =       1800.00 LB./HR/SQ. FT. FACE AREA
 PRESSURE DROP DUE TO FAN OPERATING =  0.2625 IN. H2O
 TOTAL NUMBER OF TRANSFER UNIT =   2.4238


*** TOWER SIZE ***
 LENGTH OF WET TOWERS =           400.0 FT
 FILL HEIGHT FOR WET SECTION =   45.0 FT
 FILL WIDTH FOR WET SECTION =    36.0 FT
 NUMBER OF WET TOWER FANS =        12

*** DESIGN CONDITIONS ***
 DESIGN HOT WATER TEMPERATURE = 110.824 DEG. F
 DESIGN COLD WATER TEMPERATURE =  89.396 DEG. F
 DESIGN COOLING RANGE =  21.428 DEG. F
 DESIGN APPROACH =  11.396 DEG. F
 RATING FACTOR =  0.9834
 DESIGN TURBINE BACK PRESSURE =  3.0663 IN. HG
 FUEL CONSUMPTION AT DESIGN CONDITION =  1026.95 MW
```

```
*** FOGGING PARAMETERS ***

  AVERAGE SENSIBILITY OF FOGGING, BASED ON WESTINGHOUSE CALCULATION = 0.40328
  AVERAGE FOGGING ANGLE, BASED CN MARLEY CALCULATION =   0.0014 RAD.
  AVERAGE FCGGING MAGNITUDE =   0.01281 DEG. F*LB. H2O/LB. AIR
  PROBABILITY OF NO FOGGING = 0.58048      LIGHT FOGGING = 0.41952
       MEDIUM FOGGING = 0.0          HEAVY FCGGING = 0.0

  TOTAL ANNUAL BLCWDCWN =          2030. ACRE-FT/YEAR
  TOTAL ANNUAL WATER EVAP. =       4668. ACRE-FT/YEAR      (   8.32942 )
  TOTAL ENERGY RATE IN =     1026.945 MW
  AVERAGE ENERGY RATE IN DURING ACTUAL POWER PRODUCTION =  1026.945 MW

*** CAPABILITY LOSSES ***
  EXCESS FUEL CCNSUMPTION =   -0.003 MW     (-0.000009 )
  MAXIMUM CAPABILITY LCSS =    6.614 MW     ( 0.021166 )
  ENERGY LCSS =    46085.96484 MW-HR     ( 0.016835 )
  FAN & PUMP ENERGY LOSS = 36966.3047 MW-HR     ( 0.013504 )


*** TOTAL ANNUAL COSTS ***
  TOTAL ANNUAL FUEL COST =                     6756026. $/YEAR      OR
  EXCESS FUEL COST =                               -18. $/YEAR
  TOTAL ANNUAL REPLACEMENT ENERGY LCSS =        460859. $/YEAR
  TOTAL ANNUAL WATER COST =                     218364. $/YEAR
  TOTAL ANNUAL WASTE WATER CCST =                33086. $/YEAR
  TOTAL ANNUAL MAINTAINANCE CCST =                2400. $/YEAR
  MAKEUP WATER COST WITH OPEN-CYCLE =               0. $/YEAR
  BLOWDOWN TREATMENT COST WITH CPEN-CYCLE =         0. $/YEAR
  MAINTENANCE COST WITH OPEN-CYCLE =                0. $/YEAR
                                      -------------------------
  TOTAL ANNUAL OPERATING COST =                7470727. $/YEAR
  EXTRA ANNUAL OPERATION COST =                 714691. $/YEAR

*** AVERAGE OPERATING COSTS --- IN MILLS/KW-HR ***
  AVERAGE FUEL COST =                2.467957 MILLS/KW-HR
  AVERAGE WATER CCST =               0.079768 MILLS/KW-HR
  AVERAGE WASTE WATER COST =         0.012086 MILLS/KW-HR
  AVERAGE MAINTAINANCE COST =        0.000877 MILLS/KW-HR
  AVERAGE CAPACITY LCSS =            0.168351 MILLS/KW/HR
  AVERAGE TCTAL OPERATING COST =     2.729035 MILLS/KW-HR
  AVERAGE EXTRA OPERATING COST =     0.261075 MILLS/KW-HR

*** CAPITAL COSTS ***
  CAPITAL COST OF TOWERS = $              1327532.
  PUMP AND PIPE SYSTEM CCST = $           1655998.
  PUMP AND PIPE SYSTEM SALVAGE = ( $       331200.)
  NEW CCNCENSER COST = $                        0.
  SALVAGE VALUE OF CLD CONDENSER = ( $          0.)
  OTHER OPEN-CYCLE COMPONENTS SALVAGE = ( $     0.)
  HOOKUP AND TESTING COST = $                   0.
  ADDITIONAL LAND COST = $                  93750.
  REPLACEMENT CAPABILITY COST = $          595283.
  DOWNTIME COST = $                       1574999.
                                      ---------------
  TOTAL CAPITAL COST = $                  4916361.

    NOTE : OPERATING COSTS ARE BASED CN "EXTRA" OPERATING CCST

*** TOTAL COST    --- ANNUAL BASIS --- FIXED CHARGE RATE ***
```

NO. OF YRS	CAPITAL COST MILLS/KW-HR	ANNUAL CPERATING COST MILLS/KW-HR	TCTAL COST MILLS/KW-HR	FIXED CHARGE RATE
6	0.5459633	0.2610746	0.8070379	0.304000
10	0.4552688	0.2610746	0.7163434	0.253500
15	0.3780437	0.2610746	0.6391183	0.210500
20	0.3214719	0.2610746	0.5825465	0.179000
30	0.2693899	0.2610746	0.5304645	0.150000

Example Results

Mechanical-draft wet cooling tower

Variable loading pattern

```
DESIGN WET-BULB TEMPERATURE OF AIR = 78.0 F
DESIGN DRY-BULB TEMPERATURE OF AIR = 89.0 F
EXTREME WET BULB TEMPERATURE =  83.430 DEG. F
POWER LEVEL =   313. MW
SPECIFIC HEAT OF AIR AT CONSTANT PRESSURE =  0.24 BTU/IB./F
SPECIFIC HEAT OF WATER =  1.00 BTU/LB./F
DESIGN TERMINAL TEMP. DIFFERENCE =  5.0 F
ATMOSPHERIC PRESSURE =  14.70 PSIA
UNIT FUEL COST = 0.000751 $/KW-HR
UNIT SUPPLY WATER COST = 0.1000 $/1000 GAL
UNIT WASTE WATER COST = 0.0500 $/1000 GAL
ANNUAL MAINTENANCE COST =  200.0 $/CELL/YEAR
MAX. TOLERABLE CONCENTRATION OF PROCESS WATER = 330. PPM
SUPPLY WATER CONCENTRATION = 100. PPM
DIAMETER OF WET TOWER FAN = 28.00 FT
SPACE BETWEEN TWO WET TOWER FANS = 4.00 FT
UNIT WET TOWER COST =  7.50 $/TOWER UNIT
MAXIMUM POWER OUTPUT =  312.50 MW
UNIT CAPACITY LOSS COST =  90000.00 $/MW
UNIT ENERGY COST =  0.010 $/KW-HR
UNIT LAND COST =  3000.0 $/ACRE
SPECIFIC LAND AREA = 0.10 ACRES/MW
REPLACEMENT ENERGY COST DURING DOWNTIME = 0.0070 $/KW-HR
DOWNTIME FOR CONSTRUCTION =  30.0 DAYS
PUMPING HEIGHT OF WATER THROUGH TOWER =    75.0 FEET
PUMPING EFFICIENCY FOR WATER PUMP =  0.782
UNIT CONDENSER COST =  4.00 $/SQ. FT.
OVERALL CONDENSER COEFFICIENT, U = 630.0 BTU/HR/FT2/F
INITIAL WET BULB TEMPERATURE =   5 DEG. F
FINAL WET BULB TEMPERATURE =  100 DEG. F
INCREMENT OF DRY AND WET BULB TEMPERATURE =  10 DEG. F
FINAL DRY BULB TEMPERATURE =  110 DEG. F
LOWEST TEMP. IN TURBINE CHARAC. CHART = 60.0 DEG. F
TEMP. INCREMENT IN TURBINE CHARAC. MATRIX = 2.0 DEG. F
NUMBER OF ITERATIONS IN NTU CALCULATION =    2
REFERENCE SPECIFIC VOLUME OF AIR = 13.333FT3/LB
LOWER BOUND OF LIGHT FOGGING =  0.400 LB H2O/LB AIR*F
LOWER BOUND OF MEDIUM FOGGING =  1.350 LB H2O/LB AIR*F
```

```
 FUEL CONSUMPTION WITHOUT COOLING SYSTEM =  889.354 MW  (TUR. BACK PRE. = 1.00 IN.HG)
*** 0.44662 OF THE TIME IS NOT OPERATED AT FULL LOADING ***

 TOTAL WATER FLOW RATE =     180000. GPM
 AIR FLOW RATE THROUGH EACH WET TOWER =    5399994. LB./HR
 WATER FLOW RATE THROUGH EACH WET TOWER =      15000. GPM
 WATER LOADING =       12.50 GPM/SQ. FT. PLAN AREA
 AIR LOADING =       1800.00 LB./HR/SQ. FT. FACE AREA
 PRESSURE DROP DUE TO FAN OPERATING =  0.2625 IN. H2O
 TOTAL NUMBER OF TRANSFER UNIT =   2.4238


*** TOWER SIZE ***
  LENGTH OF WET TOWERS =         400.0 FT
  FILL HEIGHT FOR WET SECTION =  45.0 FT
  FILL WIDTH FOR WET SECTION =   36.0 FT
  NUMBER OF WET TOWER FANS =       12

*** DESIGN CONDITIONS ***
  DESIGN HOT WATER TEMPERATURE = 110.824 DEG. F
  DESIGN COLD WATER TEMPERATURE =  89.396 DEG. F
  DESIGN COOLING RANGE =  21.428 DEG. F
  DESIGN APPROACH =  11.396 DEG. F
  RATING FACTOR =  0.9834
  DESIGN TURBINE BACK PRESSURE =  3.0663 IN. HG
  FUEL CONSUMPTION AT DESIGN CONDITION =  1026.95 MW
```

*** FOGGING PARAMETERS ***

AVERAGE SENSIBILITY OF FOGGING, BASED ON WESTINGHOUSE CALCULATION = 0.39725
AVERAGE FOGGING ANGLE, BASED CN MARLEY CALCULATION = 0.0015 RAD.
AVERAGE FOGGING MAGNITUDE = 0.01054 DEG. F*LB. H2O/LB. AIR
PROBABILITY OF NO FOGGING = 0.60052 LIGHT FOGGING = 0.39948
MEDIUM FOGGING = 0.0 HEAVY FOGGING = 0.0

TOTAL ANNUAL BLOWDOWN = 1839. ACRE-FT/YEAR
TOTAL ANNUAL WATER EVAP. = 4230. ACRE-FT/YEAR (7.54851)
TOTAL ENERGY RATE IN = 896.183 MW
AVERAGE ENERGY RATE IN DURING ACTUAL POWER PRODUCTION = 896.184 MW

*** CAPABILITY LOSSES ***
EXCESS FUEL CONSUMPTION = 6.829 MW (0.021854)
MAXIMUM CAPABILITY LOSS = 6.614 MW (0.021166)
ENERGY LOSS = 27387.57422 MW-HR (0.010005)
FAN & PUMP ENERGY LOSS = 20414.7188 MW-HR (0.007457)

*** TOTAL ANNUAL COSTS ***

TOTAL ANNUAL FUEL COST =	5895782. $/YEAR	OR
EXCESS FUEL COST =	44929. $/YEAR	
TOTAL ANNUAL REPLACEMENT ENERGY LOSS =	273875. $/YEAR	
TOTAL ANNUAL WATER COST =	197892. $/YEAR	
TOTAL ANNUAL WASTE WATER COST =	29984. $/YEAR	
TOTAL ANNUAL MAINTAINANCE COST =	2400. $/YEAR	
MAKEUP WATER COST WITH OPEN-CYCLE =	0. $/YEAR	
BLOWDOWN TREATMENT COST WITH OPEN-CYCLE =	0. $/YEAR	
MAINTENANCE COST WITH OPEN-CYCLE =	0. $/YEAR	
TOTAL ANNUAL OPERATING COST =	6399926. $/YEAR	
EXTRA ANNUAL OPERATION COST =	549083. $/YEAR	

*** AVERAGE OPERATING COSTS --- IN MILLS/KW-HR ***

AVERAGE FUEL COST =	2.486929 MILLS/KW-HR
AVERAGE WATER COST =	0.083474 MILLS/KW-HR
AVERAGE WASTE WATER COST =	0.012648 MILLS/KW-HR
AVERAGE MAINTAINANCE COST =	0.001012 MILLS/KW-HR
AVERAGE CAPACITY LOSS =	0.115525 MILLS/KW/HR
AVERAGE TOTAL OPERATING COST =	2.699585 MILLS/KW-HR
AVERAGE EXTRA OPERATING COST =	0.231611 MILLS/KW-HR

*** CAPITAL COSTS ***

CAPITAL COST OF TOWERS = $	1327532.
PUMP AND PIPE SYSTEM COST = $	1655998.
PUMP AND PIPE SYSTEM SALVAGE = ($	331200.)
NEW CONDENSER COST = $	0.
SALVAGE VALUE OF OLD CONDENSER = ($	0.)
OTHER OPEN-CYCLE COMPONENTS SALVAGE = ($	0.)
HOOKUP AND TESTING COST = $	0.
ADDITIONAL LAND COST = $	93750.
REPLACEMENT CAPABILITY COST = $	595283.
DOWNTIME COST = $	1102499.
TOTAL CAPITAL COST = $	4443861.

NOTE : OPERATING COSTS ARE BASED ON "EXTRA" OPERATING COST

*** TOTAL COST --- ANNUAL BASIS --- FIXED CHARGE RATE ***

NO. OF YRS	CAPITAL COST MILLS/KW-HR	ANNUAL OPERATING COST MILLS/KW-HR	TOTAL COST MILLS/KW-HR	FIXED CHARGE RATE
6	0.5698439	0.2316115	0.8014554	0.304000
10	0.4751823	0.2316115	0.7067938	0.253500
15	0.3945795	0.2316115	0.6261910	0.210500
20	0.3355332	0.2316115	0.5671447	0.179000
30	0.2811731	0.2316115	0.5127845	0.150000

APPENDIX IV

FORTRAN LISTING

NATURAL-DRAFT WET COOLING TOWER

```
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *  PERFORMANCE OF NATURAL DRAFT WET COOLING TOWER                             *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
C
      DIMENSION HR(50,14),IHR(14),A(10),B(10),ARWT(10),PSA(250)
     1,PAPCOS(20),PERCED(15,2),AMANT(3,2)
      DIMENSION WTCOST(10),AW(10),BW(10),CW(10),FCRR(5)
      DIMENSION BASEDI(10),WIDTHW(10),NYEAR(5),FCR(11)
      REAL KM
      COMMON/DENSIT/ DAIR(250)
      COMMON/INPUT4/ WPDRU(13),FILLHT(10),PDROP,WICTH
      COMMON/TURBIN/ HR,IHR,TLOW,FINC
      COMMON/TOWERS/ A,B,ARWT,AW,BW,CW,GPM,CONST,NOITT,AFR
     1,AIJK,BIJK,RLG,ENTU,N,FNTU,EGPM,GPMW
      COMMON/NOT/ NUMTOW
      COMMON/UTCOST/UC(4,3,11,5),DRH(2)
      COMMON/NATDR/H,HT,KM,D1,D2,ELEV,R,PDWAF(15)
      COMMON/CONSTA/ CONST1,DCNST
      COMMON/TURB/ POWER,TTDD,TET,TTD,TTDKO,TTDO
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,QH1,QH2,HA
      COMMON/TOWERC/ NOTS(10),WTCOST
      COMMON/INPU/ PERCEN(12,15,2),HTND(10),PERCED
      COMMON/TEMP/ ITWBI,ITWBF,ITBD,ITDBF
      COMMON/ECONO/ NYEAR,FC,WC,WW,DR,CCNO,CCN1,ANPOWE,HEIGHT,EFFICW
      COMMON/POWERC/ IPLI,IPLF,MM,PL,LP
      COMMON/LOSS/S,CWATEO,CBLOWO,CMAINO
      COMMON/ATMOS/ PATM
      COMMON/MAINTA/ AMANT,CF,EFI
      COMMON/WSS/ QHSUM(210),FOGL,FOGM
      COMMON/WITREF/ IWRITE,IPUNCH,REFSV,FAWETT,PAWETT,AFRL,TWBREF
      COMMON/PLEVEL/ PLMAX,UCAPAB,CAPCAP,UENER,ENERLS,TEI,TOTOP
      COMMON/TBPR/ ITPMAX,TETMAX,ICAP
      COMMON/TBPRE/ IPMAX,NPL,DPL,PLMIN
      COMMON/AREAS/PIPLAR,PIFAAR,W,DBAR
      COMMON/TEMPE/TWBD,TDBD
      COMMON/FRIC/FRIFAC(18)
      COMMON/RENEW/TQSTAR
      COMMON/FPL11/FPLMAX
C
C  DRY-BULB AND WET-BULB TEMPERATURE INTERVAL, ITBD, MUST BE GREATER THAN 1
C
      READ(5,1) ((((UC(I,J,K,L),L=1,5),K=1,11),J=1,3),I=1,4)
1     FORMAT(5F10.4)
      READ(5,2)(DRH(I),I=1,2)
2     FORMAT(2F10.4)
      READ(5,109) IPMAX,NPL,DPL,PLMIN
  109 FORMAT(2I10,2F10.0)
      READ(5,110) ((HR(I,J),J=1,NPL),I=1,IPMAX)
  110 FORMAT(10F8.5)
      READ(5,101) (PSA(I),I=1,250)
  101 FORMAT(10F8.5)
      READ(5,101) (DAIR(I),I=1,250)
      READ(5,101) (QHSUM(I),I=1,210)
      READ(5,102) (IHR(I),I=1,NPL)
  102 FORMAT(14I3)
      READ(5,106) (FCR(I),I=1,11)
      READ(5,106) (PAPCOS(I),I=1,20)
  106 FORMAT(10F8.0)
      READ(5,106) AFRL
      READ(5,106) GPMW
      READ(5,106) (AW(I),I=1,10)
      READ(5,106) (BW(I),I=1,10)
      READ(5,107) TLOW,FINC
  107 FORMAT(2F10.0)
      READ(5,504) NOITT
      READ(5,504) NNOTSI,NNOTS,NOWTSI,NOWTS
  504 FORMAT(4I4)
      READ(5,106) WTCOS ,CHT,SLANDA,CMAINO
      READ(5,502) CCNO,CCN1
  502 FORMAT(2F10.0)
      READ(5,506) (NYEAR(I),I=1,5),FC,WC,WW
  506 FORMAT(5I4,4F10.0)
      READ(5,507) ITWBI,ITWBF,ITBD,ITDBF
  507 FORMAT(6I4)
      READ(5,108) TWBD,TDBD,LP,TTDD,REFSV,TWBREF,PLMAX,UCAPAB
  108 FORMAT(2F10.0,I10,5F10.0)
      READ(5,106) UENER,ULAND,UDOWN,DAYS,CF,    CCO,COO,CWATEO,CBLOWO
      READ(5,507) IPLI,IPLF,MM
      NTW=(ITWBF-ITWBI)/ITBD+1
      NTD=(ITDBF-ITWBI)/ITBD+1
      NKK=(IPLF-IPLI)/MM+1
      READ(5,508) (((PERCEN(I,J,K),J=1,NTD),I=1,NTW),K=1,NKK)
```

```
  508 FORMAT(10F8.6)
      READ(5,509) PATM,TPMAX
  509 FORMAT(3F10.0)
      READ(5,510) ((AMANT(I,J),I=1,3),J=1,2)
  510 FORMAT(6F10.0)
      READ(5,513) FOGL,FOGM
  513 FORMAT(2F10.0)
      READ(5,505) CA,CW1
  505 FORMAT(4F10.2)
      READ(5,509) TBP,TWB10,TDB10
      READ(5,509) EF,EN,EFI
      READ(5,507) IWRITE,IPUNCH,ITPMAX,IEXTRA,INUCAL,NEWCON
      READ(5,111) (FRIFAC(I),I=1,18)
111   FORMAT(10F7.5)
      READ(5,112) (POWAF(I),I=1,15)
112   FORMAT(15F5.3)
      READ(5,106) EFFICW,UNCOND,UO,HEIGHT
      READ(5,106) (WIDTHW(I),I=1,10)
      READ(5,595)ELEV
595   FORMAT(3F5.0)
      READ(5,596) (BASEDI(I),I=1,10)
596   FORMAT(10F5.0)
      READ(5,105) (HTND(I),I=1,10)
105   FORMAT(10F8.2)
      READ(5,106) (FILLHT(I),I=1,10)
C
      NUMTOW=1
      R=53.35
      CONST=7.481/60./62.*10.**9
      DONST=CONST
      CONST1=0.124683/62.
C
C  CALCULATE CORRESPONDING FIXED CHARGE RATE
C
      DO100K=1,5
      NEAR=NYEAR(K)
      Y=NYEAR(K)/4.+1.
      IY=Y
  100 FCRR(K)=FCR(IY)+(FCR(IY+1)-FCR(IY))*(Y-IY)
      IF(INUCAL.EQ.0)GOTO300
      PARAME=(1.-EN)*EF/(1.-EF)/EN
      DO400I=1,IPMAX
      DO400J=1,NPL
  400 HR(I,J)=HR(I,J)*PARAME
C
  300 SS1=0.
      S=0.
      DO2011LP=IPLI,IPLF,MM
      LP1=(LP-IPLI)/MM+1
      PL=PLMAX
      IF(LP.NE.IPLI) PL=PLMAX*CF
      DO2012IJK=ITWBI,ITWBF,ITBD
      IW1=(IJK-ITWBI)/ITBD+1
      ITDBMA=PSA(IJK)/(0.00367*PATM*(1.+(IJK-32.)/1571.))+IJK
      IF(ITDBMA.GT.ITDBF) ITDBMA=ITDBF
      DO2012IIK=IJK,ITDBMA,ITBD
      ID1=(IIK-ITWBI)/ITBD+1.
 2012 SS1=SS1+PERCEN(IW1,ID1,LP1)
      S=S+SS1*PL
      SS2=SS1
 2011 SS1=0.
      IF(IPLI.EQ.IPLF) SS2=0.
      S=S*8760.
C
C   FIND FUEL CONSUMPTION WITH OPEN-CYCLE COOLING SYSTEM
C
      TBP1=TBP*62.4*13.6/1728.
      LP=NPL
      NP=(CF-0.49)*10.
      IT=TLOW
      IF(TBP1.GT.PSA(IT))GOTO710
      TQSTAR=HR(1,LP)
      TQST1=HR(1,NP)
      TQST2=HR(1,NP+1)
      GOTO716
  710 IT=IT+5
      IF(TBP1.GT.PSA(IT))GOTO710
  714 IT=IT-1
      IF(TBP1.LT.PSA(IT))GOTO714
      TTEMP=IT+(TBP1-PSA(IT))/(PSA(IT+1)-PSA(IT))
      TTEMP=(TTEMP-TLOW)/2.+1
      IT=TTEMP
      TQSTAR=HR(IT,LP)+(HR(IT+1,LP)-HR(IT,LP))*(TTEMP-IT)
      TQST1=HR(IT,NP)+(HR(IT+1,NP)-HR(IT,NP))*(TTEMP-IT)
```

```
      TQST2=HR(IT,NP+1)+(HR(IT+1,NP+1)-HR(IT,NP+1))*(TTEMP-IT)
  716 DQHR=(TQST1/CF-TQSTAR)/(PLMAX*3.6/1055.04-TQST1/CF+TQSTAR)
      FPL1=PLMAX*CF/(1.+DQHR)
      DQHR=(TQST2/(CF+0.1)-TQSTAR)/(PLMAX*3.6/1055.04-TQST2/(CF+0.1)
     1+TQSTAR)
      FPL2=PLMAX*(CF+0.1)/(1.+DQHR)
      TQST=TQST1+(TQST2-TQST1)*(PLMAX*CF-FPL1)/(FPL2-FPL1)
      TEI=PLMAX*(1.-SS2)+PLMAX*CF*SS2+(TQSTAR*(1.-SS2)+TQST*SS2)*293.067
      TEI=TEI/EFI
C
C  DETERMINE MAXIMUM ALLOWABLE TURBINE TEMPERATURE
C
      TETMAX=1000.
      IF(ITPMAX.EQ.0)GOTO717
      TBP2=TPMAX*62.4*13.6/1728.
  718 IT=IT+5
      IF(TBP2.GT.PSA(IT))GOTO718
  719 IT=IT-1
      IF(TBP2.LT.PSA(IT))GOTO719
      TETMAX=IT+(TBP2-PSA(IT))/(PSA(IT+1)-PSA(IT))
C
  717 WRITE(6,610) TWBD,TDBD,TWB10,PLMAX,CA,CW1,TTDD,PATM
  610 FORMAT(1H1///10X,'DESIGN WET-BULB TEMPERATURE OF AIR =',F5.1,' F'
     1/10X,'DESIGN DRY-BULB TEMPERATURE OF AIR =',F5.1,' F'/10X,
     2'EXTREME WET BULB TEMPERATURE =',F8.3,' DEG.F'/10X,
     2'POWER LEVEL =',F6.0,' MW'/10X,
     5'SPECIFIC HEAT OF AIR AT CONSTANT PRESSURE =',F6.2,' BTU/IB./F'/
     610X,'SPECIFIC HEAT OF WATER =',F6.2,' BTU/LB./F'/
     710X,'DESIGN TERMINAL TEMP. DIFFERENCE =',F5.1,' F'/10X,
     8'ATMOSPHERIC PRESSURE =',F7.2,' PSIA')
C
      WRITE(6,620) FC,WC,WW,CCNO,CCN1
  620 FORMAT(1H ,9X,
     4'UNIT FUEL COST =',F9.6,' $/KW-HR'/10X,
     5'UNIT SUPPLY WATER COST =',F7.4,' $/1000 GAL'/10X,
     6'UNIT WASTE WATER COST =',F7.4,' $/1000 GAL'/10X,
     7'MAX. TOLERABLE CONCENTRATION OF PROCESS WATER =',F5.0,' PPM'/10X,
     8'SUPPLY WATER CONCENTRATION =',F5.0,' PPM')
C
      WRITE(6,630) PLMAX,UCAPAB
  630 FORMAT(1H ,9X,
     7'MAXIMUM POWER OUTPUT =',F8.2,' MW'/10X,
     8'UNIT CAPACITY LOSS COST =',F10.2,' $/MW')
C
      WRITE(6,631) UENER,ULAND,SLANDA,UDOWN,DAYS
  631 FORMAT(1H ,9X,'UNIT ENERGY COST =',F8.4,' $/KW-HR'/10X,
     1'UNIT LAND COST =',F8.1,' $/ACRE'/10X,
     1'SPECIFIC LAND AREA =',F7.4,' ACRES/MW'/10X,
     2'REPLACED ENERGY COST DURING DOWN TIME =',F7.4,' $/KW-HR'/10X,
     3'DOWNTIME FOR CONSTRUCTION =',F6.1,' DAYS')
C
      WRITE(6,640) HEIGHT,EFFICW,UNCOND,UO
  640 FORMAT(1H ,9X,'PUMPING HEIGHT OF WATER THROUGH TOWER =',F8.1,
     2' FEET'/10X,'PUMPING EFFICIENCY FOR WATER PUMP =',F7.3/10X,
     3'UNIT CONDENSER COST =',F6.2,' $/SQ. FT.'/10X,
     4'OVERALL CONDENSER COEFFICIENT, U =',F6.1,' BTU/HR/FT2/F')
C
      WRITE(6,662) ITWBI,ITWBF,ITBD,ITDBF
  662 FORMAT(1H ,9X,'INITIAL WET BULB TEMPERATURE =',I4,' DEG. F'/10X,
     1'FINAL WET BULB TEMPERATURE =',I5,' DEG. F'/10X,
     2'INCREMENT OF DRY AND WET BULB TEMPERATURE =',I4,' DEG. F'/10X,
     3'FINAL DRY BULB TEMPERATURE =',I5,' DEG. F')
C
      WRITE(6,650) TLOW,FINC,NOITT,REFSV,FOGL,FOGM
  650 FORMAT(1H ,9X,'LOWEST TEMP. IN TURBINE CHARAC. CHART =',F5.1,
     1' DEG. F'/10X,'TEMP. INCREMENT IN TURBINE CHARAC. MATRIX =',F4.1,
     2' DEG. F'/10X,'NUMBER OF ITTERATION IN NTU CALCULATION =',I5
     3/10X,'REFERENCE SPECIFIC VOLUME OF AIR =',F7.3,'FT3/LB'/10X,
     4'LOWER BOUND OF LIGHT FOGGING =',F7.3,' LB H2O/LB AIR*F'
     6/10X,'LOWER BOUND OF MEDIUM FOGGING =',F7.3,' LB H2O/LB AIR*F')
C
      IT=TWBD
      TSS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TWBD-IT)
      HAA=0.24*TWBD+0.622*TSS/(PATM-TSS)*(1061.8+0.44*TWBD)
      DO1000II=NNOTSI,NNOTS
      DO1000IW=NOWTSI,NOWTS
      IF(IW.GT.3) IWRITE=0
      D1=BASEDI(II)
      W=WIDTHW(IW)
      WIDTH=W
      H=FILLHT(IW)
      HT=HTND(II)+H
C
      CALL GEOMET(D1,D2,HT,H,DBAR)
```

```
C
      TS=TSS
      HA=HAA
C
C  DETERMINE WATER FLOW RATE FOR EACH WET TOWER
C  DETERMINE AIR-FLOWS FOR WET TOWERS, DETERMINE N.T.U. FOR WET TOWER
C
      PIPLAR=-3.1415926535*D1**2/4.+3.1415926535*(D1+2*W)**2/4.
      PIFAAR=3.1415926535*D1*H
      GPM=GPMW*PIPLAR
      EGPM=GPM
      FAWET=H*D1*3.1415926535
      FNTU=AW(IW)*GPMW**BW(IW)*H
      N=IFIX(FNTU/0.5)
      ENTU=FNTU/N
C
C
C* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C  CALCULATION OF BEST ''K'' FOR AIR FLOW RATE CALCULATIONS
C* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
C
      CALL BESTK(ENTU,NOITT,N,GPM,AFR,RLG,TCW,WL,KM)
C
      IIII=-5
      TTD=TTDD
C
      CALL MODELW (TDBD,TWBD,IW,NPL,TET,TQ,TWL,IIII,K,TT,IM)
C
      IF(TET.LT.0)GOTO999
      TTSTAR=1.
      IF(ICAP.EQ.0)GOTO679
C
      CALL POWERS (TET,TQ,TTSTAR)
C
679   DQHR=(TQ/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TQ/TTSTAR+TQSTAR)
      TQQ=(PLMAX*TTSTAR/(1.+DQHR)+TQ*1055.04/3.6)/EFI
      RP=TQ*CONST/EGPM
C
C  DETERMINE CONDENSER COST, AND PUMP AND PIPE SYSTEM COST
C
      THW=TET-TTD
      TCW=THW-TQ*CONST/EGPM
      RANGE=THW-TCW
      TTDKO=TTD/RANGE*CONST/EGPM
      RLL=ALOG((RANGE+TTD)/TTD)
      CONCOS=UNCOND*EGPM/CONST/UO*RLL*10.**9
      IF(NEWCON.EQ.1) CONCOS=CCO
      IP=EGPM/10.**5
      PPCOST=PAPCOS(IP+1)+(PAPCOS(IP+2)-PAPCOS(IP+1))*(EGPM/10.**5-IP)
      PPCOSO=0.20*PPCOST
C
C  DETERMINE CAPITAL COST OF THE TOWER
C
      AP=TET-TTD-RP-TWBD
      APPRO=AP
      ITET=TET
      PDESI=(PSA(ITET)+(PSA(ITET+1)-PSA(ITET))*(TET-ITET))*1728./848.64
      AH=TS-0.000367*14.7*(TDBD-TWBD)*(1.+(TWBD-32.)/1571.)
      WL1=0.622*AH/(14.7-AH)
      RH1=1.608*PATM/TS*WL1/(1+1.608*QH1)*100.
C
      CALL CAPCO (TWBD,RANGE,APPRO,RH1,TQ,COST)
C
      WTCOST(IW)=COST
      CAPCO1=COST
C  DETERMINE DOWNTIME, AND ADDITIONAL LAND COST
C
      DOWNCO=UDOWN*PL   *24.*DAYS*1000.
      ALANDC=ULAND*PLMAX*SLANDA
C
C  DETERMINE REPLACEMENT CAPABILITY COST
C
      PUMOP1=EGPM*HEIGHT*62.4/7.481/60./550./EFFICW*0.7457
      IIII=0
C
      CALL MODELW(TWB10,TWB10,IW,NPL,TET,TQ,TWL,IIII,K,TT,IM)
C
      IF(TET.LT.0) GO TO 999
      TTSTAR=1.
      IF(ICAP.EQ.0)GOTO681
      WRITE(6,55555)TWB10,TET,TQ
55555 FORMAT(2X,F17.5)
C
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```
      CALL POWERS (TET,TQ,TTSTAR)
C
681   DQHR=(TQ/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TQ/TTSTAR+TQSTAR)
      FPL=PLMAX*TTSTAR/(1.+DQHR)
      FPL1=PLMAX-FPL
      FPL1=FPL1+PUMOP1/1000.
      FPLMAX=FPL1
      CAPCAP=FPL1*UCAPAB
C
      WRITE(6,607) SS2
  607 FORMAT(1H0,10X,'****',F8.5,
     1' OF THE TIME IS NOT OPERATED AT FULL CONDITION ***')
      WRITE(6,661) EGPM,AFR,GPM,GPMW
  661 FORMAT(1H0,10X,
     3'TOTAL WATER FLOW RATE =',F11.0,' GPM'/11X,
     4'AIR FLOW RATE THROUGH EACH WET TOWER =',F11.0,' LB./HR'/11X,
     5'WATER FLOW RATE THROUGH EACH WET TOWER =',F11.0,' GPM'/11X,
     6'WATER LOADING =',F11.2,' GPM/SQ. FT. PLAN AREA')
      WRITE(6,665) HT
  665 FORMAT(1H0,8X,'*** TOWER SIZE ***'/10X,
     1' HEIGHT OF WET TOWER =',F6.2,' FT')
      WRITE(6,664) FILLHT(IW),WIDTH,NUMTOW,D1,D2
  664 FORMAT(1H ,10X,'FILL HEIGHT FOR WET SECTION =',F6.1,' FT'/11X,
     2'FILL WIDTH FOR WET SECTION =',F7.1,' FT'/11X,
     3'NUMBER OF WET TOWERS  =',I3/11X,
     4'BASE DIAMETER OF THE TOWER =',F6.2,' FT'/11X,
     5'EXIT DIAMETER OF THE TOWER =',F6.2,' FT')
      WRITE(6,663)
  663 FORMAT(1H0,8X,'*** DESIGN CONDITIONS ***')
      WRITE(6,666) THW,TCW,RP,AP,PDESI,TQ,RH1,TQC
  666 FORMAT(1H ,10X,'DESIGN HOT WATER TEMPERATURE =',F8.3,' DEG. F'
     1/11X,'DESIGN COLD WATER TEMPERATURE =',F8.3,' DEG. F'/11X,
     2'DESIGN COOLING RANGE =',F8.3,' DEG. F'/11X,
     3'DESIGN APPROACH =',F8.3,' DEG. F'/11X,
     5'DESIGN TURBINE BACK PRESSURE =',F8.4,' IN. HG'/11X,
     5'DESIGN HEAT REJECTION =',F8.4,'*10**9  BTU'/11X,
     5'DESIGN RELATIVE HUMIDITY =',F8.3/11X,
     5'FUEL CONSUMPTION AT DESIGN CONDITION =',F9.2,' MW')
      IF(ICAP.EQ.1) WRITE(6,667)
  667 FORMAT(1H ,12X,'NOTE ... CAPACITY LOSS AT DESIGN CONDITION')
C
C  COMPUTE OPERATION COST AND TOTAL COST
C
      CALL OPECOS (IW,TOTOPE)
C
      IF(TOTOPE.GT.10.**11)GOTO1001
      CAPCOS=CAPCO1+PPCOST+CAPCAP+ALANDC+DOWNCO -CCO-COO+CHT-PPCOSO
C
      WRITE(6,604)
  604 FORMAT(1H0,7X,'*** CAPITAL COSTS ***')
      WRITE(6,602) CAPCO1,PPCOST,PPCOSO,CONCOS,CCO,COO,CHT,ALANDC,
     1CAPCAP,DOWNCO,CAPCOS
  602 FORMAT(1H ,9X,'CAPITAL COST OF TOWERS = $',F20.0/10X,
     1'PUMP AND PIPE SYSTEM COST = $',F17.0/10X,
     2'PUMP AND PIPE SYSTEM SALVAGE = ( $',F12.0,')'/10X,
     3'NEW CONDENSER COST = $',F24.0/10X,
     4'SALVAGE VALUE OF OLD CONDENSER = ( $',F10.0,')'/10X,
     5'OTHER OPEN-CYCLE COMPONENTS SALVAGE = ( $',F5.0,')'/10X,
     6'HOOKUP AND TESTING COST = $',F19.0/10X,
     7'ADDITIONAL LAND COST = $',F22.0/10X,
     8'REPLACEMENT CAPABILITY COST = $',F15.0/10X,
     9'DOWNTIME COST = $',F29.0/40X,'-----------------'/10X,
     1'TOTAL CAPITAL COST = $',F24.0)
C
      IF(IEXTRA.EQ.1) TOTOPE=TOTOP
      IF(IEXTRA.EQ.1) WRITE(6,614)
  614 FORMAT(1H0/12X,'NOTE   OPERATING COSTS ARE BASED ON  EXTRA  OPERAT
     1ING COST')
C
      WRITE(6,612)
  612 FORMAT(1H0,8X,'*** TOTAL COST 1 --- ANNUAL BASIS --- FIXED CHARGE
     1RATE ***'//12X,'NO. OF YRS',5X,'CAPITAL COST',5X,'ANNUAL OPERATING
     2 COST',4X,'ANNUAL COST',5X,'FIXED CHARGE RATE'/28X,
     3'MILLS/KW-HR',10X,'MILLS/KW-HR',9X,'MILLS/KW-HR')
      DO5000KK=1,5
      CAPCO1=CAPCOS*FCRR(KK)/S
      TOTOP2=TOTOPE
      TOTCOS=CAPCO1+TOTOP2
 5000 WRITE(6,611) NYEAR(KK),CAPCO1,TOTOP2,TOTCOS,FCRR(KK)
  611 FORMAT(1H ,I17,F21.9,F22.9,F20.9,F17.6)
      GOTO1000
  999 WRITE(6,603) EGPM,NUMTOW,TET
  603 FORMAT(1H0//10X,'************'/12X,'TOTAL WATER FLOW RATE THROUGH
     1THESE COOLING TOWERS =',F11.0,' GPM'/12X,'NUMBER OF TOWERS =',
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     2I5/12X,'TURBINE TEMPERATURE =',F10.4)
      TOTCOS=10.**12
 1000 CONTINUE
      STOP
 1001 TOTCOS=10.**12
      STOP
      END



      SUBROUTINE OPECOS (IW,TOTOPE)
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *  PROGRAM TO DETERMINE TOTAL ANNUAL OPERATING COST                       *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
C
      DIMENSION HR(50,14),IHR(14),A(10),B(10),ARWT(10),PSA(250)
      DIMENSION AW(10),BW(10),CW(10),PERCED(15,2),AMANT(3,2)
     1,WTCOST(10),S1(15),S2(15),S3(15),NYEAR(5)
      COMMON/DENSIT/ DAIR(250)
      COMMON/INPU/ PERCEN(12,15,2),HTND(10),PERCED
      COMMON/INPUT4/ WPDRO(13),FILLHT(10),PDROP,WIDTH
      COMMON/TURBIN/ HR,IHR,TLOW,FINC
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,QH1,QH2,HA
      COMMON/TOWERS/ A,B,ARWT,AW,BW,CW,GPM,CONST,NOITT,AFR
     1,AIJK,BIJK,RLG,ENTU,N,FNTU,FGPM,GPMW
      COMMON/NOT/ NUMTOW
      COMMON/TURB/ POWER,TBP,TET,TTD,TTDKO,TTDC
      COMMON/TOWERC/ NOTS(10),WTCOST
      COMMON/CONSTA/ CONST1,DCONST
      COMMON/TEMP/ ITWBI,ITWBF,ITBD,ITDBF
      COMMON/ECONO/ NYEAR,FC,WC,WW,DR,CCNO,CCN1,ANPOWE,HEIGHT,EFFICW
      COMMON/POWERC/ IPLI,IPLF,M,PL,LP
      COMMON/ATMOS/ PATM
      COMMON/MAINTA/ AMANT,CF,EFI
      COMMON/LOSS/S,CWATEO,CBLOWO,CMAINO
      COMMON/WSS/ QHSUM(210),FOGL,FOGM
      COMMON/WITREF/ IWRITE,IPUNCH,REFSV,FAWETT,PAWETT,AFRL,TWBREF
      COMMON/PLEVEL/ PLMAX,UCAPAB,CAPCAP,UENER,ENERLS,TEI,TOTOP
      COMMON/TBPR/ ITPMAX,TETMAX,ICAP
      COMMON/TBPRE/ IPMAX,NPL,DPL,PLMIN
      COMMON/RENEW/TQSTAR
      COMMON/FPL11/FPLMAX
C
C  OPERATION DUE TO WET COOLING TOWER
C
C  DRY-BULB AND WET-BULB TEMPERATURE INTERVAL, ITBD, MUST BE GREATER THAN 1
C
      IF(HTND(IW).LT.0.01 )GOTO1002
      PUMOP1=FGPM*HEIGHT*62./7.481/60./550./EFFICW*0.7457
C
      IF(IWRITE.EQ.1)WRITE(6,899)
C
      NUMTOW=1
      IM=0
      TOTOPE=0.
      TOTBLD=0.
      TOTWL=0.
      TOTEI=0.
      TOTFUE=0.
      TOTWAT=0.
      TOTWAW=0.
      TOTMAN=0.
      TOTLOS=0.
      TOTPRO=0.
      SEN1=0.
      SEN2=0.
      SEN3=0.
      FOGOS=0.
      FOGLS=0.
      FOGMS=0.
      FOGHS=0.
      CAPLOS=0
      CAPPRO=0.
      FPLMAX=0.
      ENERLS=0.
      FAPLS=0.
C
      RLG=GPM/CONST*10.**9/AFR
      FNTU=AW(IW)*GPMW**BW(IW)*FILLHT(IW)
      N=IFIX(FNTU/0.5)
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      ENTU=FNTU/N
      PLC=PLMAX*CF+PUMOP1/1000.
      DO1000LP=IPLI,IPLF,M
      LP1=(LP-IPLI)/M+1
C
      TTTSAV=0.
      KKSAVE=0
      JJJJ=0
C
      DO901IIJ=ITWBI,ITWBF,ITBD
      TWB=IIJ
      TS=PSA(IIJ)
      IW1=(IIJ-ITWBI)/ITBD+1
      KK=KKSAVE
      TTT=TTTSAV
      I2=0
      ITDBMA=PSA(IIJ)/(0.000367*PATM*(1.+(IIJ-32.)/1571.))+IIJ
      IF(ITDBMA.GT.ITDBF)ITDBMA=ITDBF
      IF(IIJ.GT.ITDBMA)GO TO 901
      DO923IJK=IIJ,ITDBMA,ITBD
      TDB=IJK
      ID1=(IJK-ITWBI)/ITBD+1
      AH=TS-0.000367*PATM*(TDB-TWB)*(1.+(TWB-32.)/1571.)
      WL1=(0.622*TPS/(PATM-TPS)-0.622*AH/(PATM-AH))*AFR*CONST/10.**9
     1*NUMTOW
      QH1=WL1
      FAP=PUMOP1/1000.
      NP=NPL
      IF(LP.NE.IPLI) NP=(CF-0.49)*10.
C
      CALL MODELW (TDB,TWB,IW,NP,TET,TQ,TWL,JJJJ,KK,TTT,IM)
C
      I2=I2+1
      IF(TET.LT.0)GOTO200
      TTSTAR=1.
      IF(LP.NE.IPLI) TTSTAR=CF
      IF(ICAP.EQ.0)GOTO9666
C
      CALL POWERS (TET,TQ,TTSTAR)
C
      CAPPRO=CAPPRO+PERCEN(IW1,ID1,LP1)
 9666 DQHR=(TQ/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TQ/TTSTAR+TQSTAR)
      FPL=PLMAX*TTSTAR/(1.+DQHR)
      PL=FPL
      IF(LP.EQ.IPLI)GOTO96680
      IF(ICAP.EQ.0)GOTO96681
      FPL1=CF*PLMAX-FPL
      PL=FPL
      FAP=PUMOP1/1000.
      GOTO96682
96681 TETST=TET
      TQST=TQ
      FPLST=FPL
      TWLST=TWL
      NP=NP+1
      KKST=KK
      TTTST=TTT
C
      CALL MODELW (TDB,TWB,IW,NP,TET1,TQ1,TWL1,JJJJ,KKST,TTTST,IM)
C
      TTSTAR=CF+0.1
      IF(ICAP.EQ.0)GOTO96683
C
      CALL POWERS (TET1,TQ1,TTSTAR)
C
96683 DQHR=(TQ1/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TQ1/TTSTAR+TQSTAR)
      FPL=PLMAX*TTSTAR/(1.+DQHR)
C
      TET=TETST+(TET1-TETST)/(FPL-FPLST)*(PLC-FPLST)
      IF(TET.LT.(TETMAX+0.05))GOTO96684
      FPL1=CF*PLMAX-FPL
      TQ=TQ1
      TET=TETMAX
      TWL=TWL1
      PL=FPL
      FAP=PUMOP1/1000.
      GOTO96682
96684 TQ=TQST+(TQ1-TQST)/(FPL-FPLST)*(PLC-FPLST)
      TWL=TWLST+(TWL1-TWLST)/(FPL-FPLST)*(PLC-FPLST)
      FPL1=0.
      FAP=0.
      PL=PLC
      GOTO96685
96680 FPL1=PLMAX-FPL
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96682 IF(FPL1.LT.0.) FPL1=0.
      FPL1=FPL1+(PUMOP1)/1000.
      IF(FPL1.GT.FPLMAX.AND.PERCEN(IW1,ID1,LP1).GT.0.000001)FPLMAX=FPL1
96685 ENERLS=ENERLS+FPL1*PERCEN(IW1,ID1,LP1)*8760.
      FAPLS=FAPLS+FAP*PERCEN(IW1,ID1,LP1)*8760.
      EI=(PL+TQ*1.05504/3.6*1000.)*1000./EFI
      FUECOS=FC*EI*8760.
      BLDOWN=TWL*CCN1/(CCNO-CCN1)
      WATCOS=(TWL+BLDOWN)*WC*525.6
      WAWACO=BLDOWN*WW*525.6
      ANUCAP=ENERLS*UENER*1000.
      HOTWTT=TET-TTD
      COLDWT=HOTWTT-TQ*CONST/FGPM
      ITET=TET
      P=(PSA(ITET)+(PSA(ITET+1)-PSA(ITET))*(TET-ITET))/0.491111
      TOTBLD=TOTBLD+BLDOWN*PERCEN(IW1,ID1,LP1)*60.*24.*365./326046
      TOTWL=TOTWL+TWL*PERCEN(IW1,ID1,LP1)*60.*24.*365./326046
      EI1=EI/1000.
      TOTEI=TOTEI+EI1*PERCEN(IW1,ID1,LP1)
      TOTPRO=TOTPRO+PERCEN(IW1,ID1,LP1)
      TOTFUE=TOTFUE+FUECOS*PERCEN(IW1,ID1,LP1)
      TOTWAT=TOTWAT+WATCOS*PERCEN(IW1,ID1,LP1)
      TOTWAW=TOTWAW+WAWACO*PERCEN(IW1,ID1,LP1)
      TOTMAN=TOTMAN+AMANT(2,LP1)*PERCEN(IW1,ID1,LP1)*NUMTOW
      FUECIS=FUECOS*PERCEN(IW1,ID1,LP1)
      WATCOS=WATCOS*PERCEN(IW1,ID1,LP1)
      WAWACO=WAWACO*PERCEN(IW1,ID1,LP1)
      AMANT1=AMANT(2,LP1)*PERCEN(IW1,ID1,LP1)*NUMTOW
      OPCOS=(FUECIS+WATCOS+WAWACO+AMANT1)
     1+FPL1*PERCEN(IW1,ID1,LP1)*8760.*UENER*1000.
      TOTOPE=TOTOPE+OPCOS
      IF(PERCEN(IW1,ID1,LP1).LT.0.000001)GOTO312
C
      CALL FOGSEN (TDB,TWB,TWBAL,QH2,SENSI1,SENSI2,SENSI3)
C
      SEN1=SEN1+SENSI1*PERCEN(IW1,ID1,LP1)
      SEN2=SEN2+SENSI2*PERCEN(IW1,ID1,LP1)
      SEN3=SEN3+SENSI3*PERCEN(IW1,ID1,LP1)
      IF(SENSI3.LT.0.00005)GOTO317
      IF(SENSI3.LT.FOGL)GOTO315
      IF(SENSI3.LT.FOGM)GOTO316
      FOGHS=FOGHS+PERCEN(IW1,ID1,LP1)
      GOTO311
  315 FOGLS=FOGLS+PERCEN(IW1,ID1,LP1)
      GOTO311
  316 FOGMS=FOGMS+PERCEN(IW1,ID1,LP1)
      GOTO311
  312 SENSI1=0.
      SENSI2=0.
      SENSI3=0.
  317 FOGOS=FOGOS+PERCEN(IW1,ID1,LP1)
  311 S1(ID1)=SENSI1
      S2(ID1)=SENSI2
      S3(ID1)=SENSI3
      IF(IPUNCH.EQ.1) WRITE(7,701) FUECOS,ANUCAP,TWL,OPCOS,BLDOWN,FPL1,
     1SENSI1,SENSI2,SENSI3,EI,IM
  701 FORMAT(2F10.0,F6.0,F10.0,F5.0,F9.4,F7.5,F6.4,F8.5,F8.0,I1)
  200 IF(IM.LT.2)GOTO475
      IF(I2.NE.1)GO TO 902
      TTTSAV=TTT
      KKSAVE=KK
      GO TO 902
  475 DO9233IJL=IIJ,ITDBMA,ITBD
      IF(IWRITE.EQ.1)WRITE(6,666)
  666 FORMAT(5X,' WET COOLING TOWER IS TOO LARGE TO OPERATE')
      ID1=(IJL-ITWBI)/ITBD+1
      TOTLOS=TOTLOS+PLMAX*8760000.*UENER*PERCEN(IW1,ID1,LP1)
      TOTOPE=TOTOPE+PLMAX*8760000.*UENER*PERCEN(IW1,ID1,LP1)
      FOGOS=FOGOS+PERCEN(IW1,ID1,LP1)
      IM=0
      IF(IPUNCH.EQ.1)WRITE(7,702) IM
  702 FORMAT(I80)
 9233 CONTINUE
      GO TO 901
  902 IF(IWRITE.LT.1)GOTO923
      WRITE(6,601) PL,TWB
  601 FORMAT(1H0,5X,'POWER =',F5.0,' MW',10X,'TWB =',F8.3,' DEG. F')
      WRITE(6,333)TET,HOTWTT,COLDWT,P,TQ
333   FORMAT(/6X,'TURB.TEMP. = ',F10.4,1X,'DEG.F. ',
     15X,'HOT WATER TEMP. = ',F10.4,1X,'DEG.F.',
     25X,'COLD WATER TEMP. = ',F10.4,1X,'DEG.F.',
     3//,6X,'PRESSURE = ',F8.5,1X,'IN.HG.',
     45X,'HEAT REJECTION = ',F8.5,1X,'BTU*10**9',/)
      WRITE(6,602)
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  602 FORMAT(1H0,1X,'TDB',3X,'WATER EVA.',3X,'BLOWDOWN',3X
     1,'PROBABILITY',3X,'FUEL COST',3X,
     2'WATER COST',3X,'WASTE WATER COST',3X,'SUBS ENERGY LOSS',3X,
     3'OPERATING COST'/3X,'F',
     47X,'GPM',9X,'GPM',22X,'$/YEAR',6X,
     5'$/YEAR',10X,'$/YEAR',12X,'$/YEAR',12X,'$/YEAR'/)
      WRITE(6,607)IIJ,TWL,BLDOWN,PERCEN(IW1,ID1,LP1),FUECIS,WATCOS,WAWAC
     1O,ANUCAP,OPCOS,FPL1
  607 FORMAT(1H ,I3,F15.5,F11.4,F13.6,F13.0,F12.1,F16.1,F20.3,F18.1,
     1F10.3)
      WRITE(6,69699)
69699 FORMAT(1H0///5X,'***** FOGGING PARAMETERS *****'//3X,
     1'TDB',3X,'TDB(EXIT)',3X,'SPE. HUMID.',6X,
     2'SENSIBILITY',3X,'FOGGING ANGLE',3X,'FOGGING MAG.'/4X,
     3'F',8X,'F',7X,'# H2O/# AIR',5X,'WESTINGHOUSE',3X,'RAD. (MARLEY)')
      WRITE(6,69698) IJK,TWBAL,QH2,S1(ID1),S2(ID1),S3(ID1)
69698 FORMAT(1H ,I5,F11.3,F13.6,F17.7,F14.5,F17.5)
      WRITE(6,899)
  899 FORMAT(///1X,130('*')/////)
  923 CONTINUE
  901 CONTINUE
 1000 CONTINUE
      FUELEX=TOTEI-TEI
      TOTFU1=FUELEX*FC*1000.*8760.
      TOTOP=TOTOPE-TEI*FC*1000.*8760.
      WRITE(6,66667)
66667 FORMAT(1H0,8X,'*** FOGGING PARAMETERS ***')
      WRITE(6,66666) SEN1,SEN2,SEN3,FOGOS,FOGLS,FOGMS,FOGHS
66666 FORMAT(1H ,/10X,'AVERAGE SENSIBILITY OF TOWER PLUME, BASED ON WEST
     1INGHOUSE CALCULATION =',F8.5/10X,'AVERAGE FOGGING ANGLE, BASED ON
     2MARLEY CALCULATION =',F9.4,' RAD.'/10X,
     3'AVERAGE FOGGING MAGNITUDE =',F10.5,' DEG. F*LB. H2O/LB. AIR'/10X,
     4'PROBABILITY OF NO FOGGING =',F8.5,5X,'LIGHT FOGGING =',F8.5,5X,
     5'MEDIUM FOGGING =',F8.5,5X,'HEAVY FOGGING =',F8.5)
      TOTEI1=TOTEI/TOTPRO
      TOTWL1=TOTWL/(TQSTAR*1055.04/3.6)
      FUELE1=FUELEX/PLMAX
      FPLMA1=FPLMAX/PLMAX
      ENERL1=ENERLS/(PLMAX*8760.)
      FAPLS1=FAPLS/(PLMAX*8760.)
      WRITE(6,12121)
12121 FORMAT(//10X,'VALUES IN PARANTHESIS ARE,THE VALUES DIVIDED BY POWE
     1R OUTPUT EXCEPT THE LAST TWO WHICH ARE'/10X,'THE VALUES DIVIDED BY
     2 THE POWER OUTPUT PER YEAR'//)
      WRITE(6,605) TOTBLD,TOTWL,TOTWL1,TOTEI,FUELEX,FUELE1,
     1FPLMAX,FPLMA1,ENERLS,ENERL1,FAPLS,FAPLS1
     1,TOTFUE,TOTFU1,TOTWAT,TOTWAW,TOTMAN,ANUCAP,TOTOPE,TOTOP
  605 FORMAT(1H ,/10X,'TOTAL ANNUAL BLOWDOWN =',F15.0,' ACRE-FT/YEAR'/
     110X,'TOTAL ANNUAL WATER EVAP. =',F12.0,' ACRE-FT/YEAR'
     1,5X,'(',F10.5,' )'/10X,
     2'TOTAL ENERGY RATE IN =',F12.3,' MW'//8X,
     2'*** CAPABILITY LOSSES ***'/10X,
     3'EXCESS FUEL CONSUMPTION =',F9.3,' MW',5X,'(',F9.6,' )'/10X,
     3'MAXIMUM CAPABILITY LOSS =',F9.3,' MW',5X,'(',F9.6,' )'/10X,
     3'ENERGY LOSS =',F15.5,' MW-HR',5X,'(',F9.6,' )'/10X,
     3'PUMP ENERGY LOSS =',F11.4,' MW-HR',5X,'(',F9.6,' )'//8X,
     3'*** TOTAL ANNUAL COSTS ***'/10X,
     3'TOTAL ANNUAL FUEL COST =',F20.0,' $/YEAR'/10X,
     4'EXCESS FUEL COST =',F26.0,' $/YEAR'/10X,
     4'TOTAL ANNUAL WATER COST =',F19.0,' $/YEAR'/10X,
     5'TOTAL ANNUAL WASTE WATER COST =',F13.0,' $/YEAR'/10X,
     6'TOTAL ANNUAL MAINTAINANCE COST =',F12.0,' $/YEAR'/10X,
     6'TOTAL ANNUAL CAPACITY LOSS =',F16.0,' $/YEAR'/10X,
     7'TOTAL ANNUAL OPERATING COST =',F15.0,' $/YEAR'/10X,
     8'EXTRA ANNUAL OPERATION COST =',F15.0,' $/YEAR')
      TOTFUE=TOTFUE/S
      TOTWAT=TOTWAT/S
      TOTWAW=TOTWAW/S
      TOTMAN=TOTMAN/S
      ANUCAP=ANUCAP/S
      TOTOP=TOTOP/S
      TOTOPE=TOTOPE/S
      WRITE(6,621) TOTFUE,TOTWAT,TOTWAW,TOTMAN,ANUCAP,TOTOPE
     1,TOTOP
  621 FORMAT(1H0,7X,'*** AVERAGE OPERATING COSTS --- IN MILLS/KW-HR ***'
     1/10X,'AVERAGE FUEL COST =',F22.6,' MILLS/KW-HR'/10X,
     1'AVERAGE WATER COST =',F21.6,' MILLS/KW-HR'/10X,
     2'AVERAGE WASTE WATER COST =',F15.6,' MILLS/KW-HR'/10X,
     4'AVERAGE MAINTAINANCE COST =',F14.6,' MILLS/KW-HR'/10X,
     5'AVERAGE CAPACITY LOSS =',F18.6,' MILLS/KW/HR'/10X,
     6'AVERAGE TOTAL OPERATING COST =',F11.6,' MILLS/KW-HR'/10X,
     7'AVERAGE EXTRA OPERATING COST =',F11.6,' MILLS/KW-HR')
      RETURN
 1002 WRITE(6,623)
```

```
  623 FORMAT(1H0/10X,'WET COOLING TOWER IS NOT SUFFICIENT TO OPERATE')
      TOTOPE=10.**12
      RETURN
      END



      SUBROUTINE MODELW (TDB,TWB,IW,LP,TET,TQ,TWL,IIII,K,TT,IM)
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *  THIS SUBROUTINE CALCULATES THE MODELING RELATIONSHIPS FOR POWER    *
C *   PLANT AND COOLING TOWER .   GIVEN WET AND DRY BULB                *
C *   TEMPERATURES, OPERATION LEVELS OF WET TOWERS, POWER LEVEL OF      *
C *   TURBINE OUTPUT, THE RESULTS ARE TURBINE EXHAUST TEMPERATURE AND   *
C *   HEAT REJECTION.                                                   *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
C
      DIMENSION HTRJIN(50,14),IENHTR(14),A(10),B(10),ARWT(10)
     1,AW(10),BW(10),CW(10),PSA(250)
      COMMON/TURBIN/HTRJIN,IENHTR,TLOW,FINC
      COMMON/TOWERS/ A,B,ARWT,AW,BW,CW,GPM,CONST,NOITT,AFR
     1,AIJK,BIJK,RLG,ENTU,N,FNTU,FGPM,GPMW
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,QH1,QH2,HA
      COMMON/NOT/ NUMTOW
      COMMON/TURB/ BL1,BL2,BL3,TTD,TTDKO,TTDO
      COMMON/TBPR/ ITPMAX,TETMAX,ICAP
      COMMON/TEST/IFLAG
C
C   IF TWB IS HIGH ENOUGH, THEN COOLING CANNOT TAKE PLACE AT ALL UNTIL
C   TURBINE CONDENSER TEMPERATURE IS HIGHER. THUS, WILL SKIP TO
C   HIGHER TURBINE TEMPERATURE.
C
C   IF TWB IS LOW ENOUGH,  COOLING WATER FREEZES, WHICH
C   IS NEVER DESIRED.  THUS WHENEVER COOLING WATER WOULD HAVE BEEN
C   COOLED BELOW FREEZING ANYWHERE IN THE CYCLE, NO COOLING IS
C   PERFORMED (IMPLYING ALTERNATE SYSTEM USED IN PRACTICE).
C
C  ASSIGN MODEL PARAMETERS FOR TOWER SECTION
C
      ICAP=0.
      IM=2
      IFR=0
      IFRE=-1
      IF(IIII.GT.0.5)GOTO1000
      WL=0.
      TPS=0.
      TWBAL=0.
      QH2=0.
      IF(IIII.EQ.-5)GOTO991
      TTDO=HTRJIN(1,LP)*TTDKO
      GOTO992
  991 TTDO=TTD
C
C  ASSIGN INITIAL TRIAL TURBINE TEMPERATURE
C
  992 K=0
   99 IFRE=IFRE+1
  201 K=K+1
      TT=TLOW+(K-1)*FINC
      IF(K.GT.IENHTR(LP).AND.TT.GT.(TETMAX+1.95))GO TO 999
      IF(IIII.EQ.-5)GOTO404
      TTD=HTRJIN(K,LP)*TTDKO
  404 TT1=TT-TTD
C
C  COOL THROUGH COOLING SYSTEM IF POSSIBLE TO GET TQ1
C
      IF(TWB.LT.TT1)GOTO403
      GOTO201
  403 WL2=WL
      TPS2=TPS
      TWBAL2=TWBAL
      QH22=QH2
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      TWI=TT1
      CALL   AIRFLR(TWI,TDB,TWB,ENTU,NOITT,N,GPM,AFR,RLG,TT2,WL)
C
      IF(TT2.GT.32.AND.IFLAG.NE.1)GOTO503
      GOTO99
C
C  DETERMINE DIRECTION OF APPROACH TO INTERSECTION OF CURVES
```

```
C
  503 TQ1=(TT1-TT2)*FGPM/CONST
      IF(TQ1.LT.HTRJIN(K,LP))GOTO100
C
C  REACH INTERSECTION BY DECREASING TURBINE TEMPERATURE
C
      IF(IFRE.GT.0.)GOTO206
      IF(TWB.GT.(TLOW-FINC-TTDO))GOTO104
C
C  COOLING CURVE ENDS MORE THAN 1 DECREMENT BELOW TLOW
C
  206 TT=TT-FINC
      TT1=TT-TTDO
      IF(TT1.LT.32.)GOTO703
      IF(TWB.GT.TT1)GOTO304
C
C  COOLING THROUGH COOLING SYSTEM IF POSSIBLE TO GET TQ2
C
      WL2=WL
      TPS2=TPS
      TWBAL2=TWBAL
      QH22=QH2
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      TWI=TT1
      CALL  AIRFLR(TWI,TDB,TWB,ENTU,NOITT,N,GPM,AFR,RLG,TT2,WL)
C
      IF(TT2.GT.32.AND.IFLAG.NE.1)GOTO505
      IFR=1
  505 TQ2=(TT1-TT2)*FGPM/CONST
      IF(TQ2.LT.HTRJIN(1,LP))GOTO105
      IF(IFR.EQ.1)GOTO703
      TQ1=TQ2
      GOTO206
C
C  INTERPOLATE FOR TQ,TET,TWL
C
  105 TQ=HTRJIN(1,LP)
      HTDIF1=(TQ-TQ2)/(TQ1-TQ2)
      TET=TT+HTDIF1*FINC
      IF(IIII.EQ.-5)GOTO405
      TTD=TQ*TTDKO
  405 TWL=(WL+HTDIF1*(WL2-WL))*NUMTOW
      TPS=TPS+HTDIF1*(TPS2-TPS)
      TWBAL=TWBAL+HTDIF1*(TWBAL2-TWBAL)
      QH2=QH2+HTDIF1*(QH22-QH2)
      IF(IFR.EQ.0)GOTO210
C
C  (PREVIOUS TOWER COOLING INDICATES THAT THE OPERATING CHARACTERISTICS
C   CURVE FOR THE COOLING SYSTEM ENDS IN THE SAME TEMPERATURE INTERVAL
C   AS TET)
C  COOL THROUGH COOLING SYSTEM USING TET, TQ FOR CHECK
C
  211 TT1=TET-TTD
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      TWI=TT1
      CALL  AIRFLR(TWI,TDB,TWB,ENTU,NOITT,N,GPM,AFR,RLG,TT2,WL)
C
      IF(TT2.GT.32.AND.IFLAG.NE.1)GOTO210
      GOTO703
  210 CONTINUE
      IIII=1
      RETURN
  304 TT=TT+FINC
C
C  COOLING CURVES END JUST BELOW TT AND DECREMENTING TURBINE TEMPERATURE
C   WILL NOT INTERSECT IT
C
C  DETERMINE PROPER VALUE OF HTRJIN(1,LP)
C  DOUBLE INTERPOLATE FOR TQ,TET,TWL
C
  104 IF(K.GT.1)GOTO106
      HQ=HTRJIN(1,LP)
      GOTO107
  106 HQ=HTRJIN(K-1,LP)
  107 IF(IIII.GT.-5)GOTO407
      HQ=HQ+(TWB+TTD-TT+FINC)/FINC*(HTRJIN(K,LP)-HQ)
      GOTO408
  407 HQ=(HQ+(TWB-TT+FINC)/FINC*(HTRJIN(K,LP)-HQ))/(1.-TTDKO/FINC
     1*(HTRJIN(K,LP)-HQ))
  408 IF(IIII.EQ.-5)GOTO406
      TTD=HQ*TTDKO
  406 TQ=TQ1*HQ/(TQ1+HQ-HTRJIN(K,LP))
```

```
      TET=TWB+TTD+(TT-TWB-TTD)*TQ/TQ1
      IF(IIII.EQ.-5)GOTO409
      TTD=TQ*TTDKO
  409 TWL=WL/TQ1*TQ*NUMTOW
      TPS=TPS/TQ1*TQ
      TWBAL=TWBAL/TQ1*TQ
      QH2=QH2/TQ1*TQ
      TT=TT-FINC
      IF(K.GT.1)K=K-1
      GOTO211
C
C  REACH INTERSECTION BY INCREMENTING TURBINE TEMPERATURE
C
  100 IF(K.EQ.IENHTR(LP))GOTO999
  103 TT=TT+FINC
      K=K+1
      IF(IIII.EQ.-5)GOTO410
      TTD=HTRJIN(K,LP)*TTDKO
  410 TT1=TT-TTD
C
C  COOL THROUGH SYSTEM TO GET TQ2
C
      WL2=WL
      TPS2=TPS
      TWBAL2=TWBAL
      QH22=QH2
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      TWI=TT1
      CALL  AIRFLR(TWI,TDB,TWB,ENTU,NOITT,N,GPM,AFR,RLG,TT2,WL)
C
      TQ2=(TT1-TT2)*FGPM/CONST
      IF(TQ2.GT.HTRJIN(K,LP))GOTO101
      IF(K.EQ.IENHTR(LP))GOTO999
      TQ1=TQ2
      GOTO108
C
C  INTERPOLATE FOR TQ, TET, TWL
C
  101 HTDIF1=HTRJIN(K,LP)-HTRJIN(K-1,LP)
      HTDIF2=TQ2-TQ1
      TQ=(HTRJIN(K,LP)*HTDIF2-TQ2*HTDIF1)/(HTDIF2-HTDIF1)
      TET=TT-(TQ2-TQ)/HTDIF2*FINC
      IF(IIII.EQ.-5)GOTO411
      TTD=TQ*TTDKO
  411 TWL=(WL2+(WL-WL2)/HTDIF2*(TQ-TQ1))*NUMTOW
      TPS=TPS2+(TPS-TPS2)/HTDIF2*(TQ-TQ1)
      TWBAL=TWBAL2+(TWBAL-TWBAL2)/HTDIF2*(TQ-TQ1)
      QH2=QH22+(QH2-QH22)/HTDIF2*(TQ-TQ1)
      TT=TT-FINC
      K=K-1
      IIII=1
      RETURN
C
C  RETURN WITH MESSAGE
C
  703 TQ=-50
      TET=-50.
      TWL=-50.
      IIII=0
      IM=-50
      RETURN
C
C  FIND INTERSECTION WHEN WET-BULB TERPEPATURE INCREASES
C
C  REACH INTERSECTION BY INCREMENTING TURBINE TEMPERATURE
C
 1000 TTD=HTRJIN(K,LP)*TTDKO
      TT1=TT-TTD
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      TWI=TT1
      CALL  AIRFLR(TWI,TDB,TWB,ENTU,NOITT,N,GPM,AFR,RLG,TT2,WL)
C
      TQ1=(TT1-TT2)*FGPM/CONST
 1004 TT=TT+FINC
      IF(TT.LT.(TLOW+0.001))GOTO1001
      IF(K.EQ.IENHTR(LP))GOTO999
      K=K+1
      HQ=HTRJIN(K,LP)
      HTDIF1=HQ-HTRJIN(K-1,LP)
      GOTO2070
 1001 HQ=HTRJIN(1,LP)
      HTDIF1=0.
```

```
 2070 TTD=HQ*TTDKO
      TT1=TT-TTD
      WL2=WL
      TPS2=TPS
      TWBAL2=TWBAL
      QH22=QH2
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      TWI=TT1
      CALL  AIRFLR(TWI,TDB,TWB,ENTU,NOITT,N,GPM,AFR,RLG,TT2,WL)
C
      TQ2=(TT1-TT2)*FGPM/CONST
      IF(TQ2.GT.HQ)GOTO1010
      TQ1=TQ2
      GOTO1004
C
C  INTERPOLATE FOR TQ, TET, TWL
C
 1010 IF(K.GT.1)GOTO412
      TTD=HTRJIN(1,LP)*TTDKO
      GOTO413
  412 TTD=HTRJIN(K-1,LP)*TTDKO
  413 IF(TWB.LT.(TT-FINC-TTD))GOTO1011
      TQ1=TQ2
      GOTO104
 1011 HTDIF2=TQ2-TQ1
      TQ=(HQ*HTDIF2-TQ2*HTDIF1)/(HTDIF2-HTDIF1)
      TET=TT-(TQ2-TQ)/HTDIF2*FINC
      TTD=TQ*TTDKO
      TWL=(WL2+(WL-WL2)/HTDIF2*(TQ-TQ1))*NUMTOW
      TPS=TPS2+(TPS-TPS2)/HTDIF2*(TQ-TQ1)
      TWBAL=TWBAL2+(TWBAL-TWBAL2)/HTDIF2*(TQ-TQ1)
      QH2=QH22+(QH2-QH22)/HTDIF2*(TQ-TQ1)
      IF(K.GT.1) K=K-1
      TT=TT-FINC
      RETURN
999   IF(TT2.LT.32.)GOTO703
      ICAP=1
      TT1=TETMAX-TTD
C
      TWI=TT1
      CALL  AIRFLR(TWI,TDB,TWB,ENTU,NOITT,N,GPM,AFR,RLG,TT2,WL)
C
      TQ=(TT1-TT2)*FGPM/CONST
      TET=TETMAX
      IF(IIII.NE.-5)TTD=TQ*TTDKO
      TWL=WL*NUMTOW
      RETURN
      END




      SUBROUTINE GEOMET(D1,D2,HT,H,DBAR)
      HTT=HT-H
      ASQR=(0.5958)**2.*D1**2./4.
      BSQR=ASQR*(0.7409)**2.*HTT**2./(D1**2./4.*(1.-(0.5958)**2.))
      Y=0.2591*HTT
      YY=0.7409*HTT
      XSQR=(ASQR*BSQR+ASQR*Y)/BSQR
      X=SQRT(XSQR)
      D2=X*2.
      VOL1=3.1415926535*ASQR/BSQR/3.*Y*(3.*BSQR+Y**2.)
      VOL2=3.1415926535*ASQR/BSQR/3.*YY*(3.*BSQR+YY**2.)
      VOLT=VOL1+VOL2
      ABAR=VOLT/HTT
      ARGUM=4.*ABAR/3.1415926535
      DBAR=SQRT(ARGUM)
      RETURN
      END




      SUBROUTINE RESTK(DNTU,NOITT,N,GPM,AFR,RLG,TCW,WL,REALK)
      DIMENSION FRIFAC(18),PSA(250)
      REAL LHS,KK,KM
      COMMON/TEST/IFLAG
      COMMON/NCALA/PSA,TPS,TS,T4,W1,W2,HA
      COMMON/VEL/V4,V1,RHO1,RHO2
      COMMON/FRIC/FRIFAC
```

```
      COMMON/ATMOS/PATM
      COMMON/CONSTA/ CONST1,CONST
      COMMON/TEMPE/TWB,TDB
      COMMON/MATOR/H,HT,KM,D1,D2,ELEV,R,PDWAF(15)
      COMMON/AREAS/PIPLAR,PIFAAR,W,DBAR
      SUMK=0.
      DO 55556 II=100,120,10
      TWI=II
      DO 55555 J=1,30,4
      KK=J
      KM=KK
      CALL AIRFLR(TWI,TDB,TWB,DNTU,NOITT,N,GPM,AFR,RLG,TCW,WL)
      RN=(V4*D2**2./DBAR**2.)*DBAR*RHO2/4.2*10.**7
      RI=(ALOG(RN)-3.6)/0.2+1.
      I=RI
      IF(I.LE.0) GO TO 44
      IF(I.GT.17) GO TO 66
      FF=FRIFAC(I)+(FRIFAC(I+1)-FRIFAC(I))*(RI-I)
      GO TO 55
44    FF=64./RN
      GO TO 55
66    FF=0.008
67    LHS=1./SQRT(FF)
      ARGIM=RN*SQRT(FF)
      RHS=0.86*ALOG(ARGIM)-0.8
      IF((LHS/RHS).GE.0.99) GO TO 55
      FF=FF-0.0001
      GO TO 67
55    AHF=FF*(HT-H)/DBAR*(V4*D2**2./DBAR**2.)**2./64.348
      HLPILE=KK*RHO1*V1**2./2./32.174-AHF*RHO2
      CFM=V1*60.
      HLPLIW=HLPILE/62.4*12.
      IJ=CFM/50.
      IF(IJ.GT.14) GO TO 65555
      HLPILA=PDWAF(IJ)+(PDWAF(IJ+1)-PDWAF(IJ))*(CFM/50.-IJ)
      IF(HLPILA.GE.HLPLIW) GO TO 65556
      REALL=CFMSAV-CFM
      X1=HLSAV1-HLSAVE
      X2=HLPLIW-HLPILA
      X=X1*REALL/(X1+X2)
      CFMACT=CFMSAV-X
      REAKK=(CFMSAV-CFMACT)/(CFMSAV-CFM)*4.+(KK-4.)
      SUMK=SUMK+REAKK
      GO TO 55556
65556 HLSAVE=HLPLIW
      HLSAV1=HLPILA
      CFMSAV=CFM
65555 CONTINUE
55555 CONTINUE
55556 CONTINUE
      REALK=SUMK/3.
      KM=REALK
      WRITE(6,57)
57    FORMAT('1')
      WRITE(6,56)KM
56    FORMAT(5X,' K = ',F5.2)
      RETURN
      END



      SUBROUTINE AIRFLR(TWI,TDB,TWB,DNTU,NOITT,N,GPM,AFR,RLG,TCW,WL)
C
C*************************************************************************
C                                                                        *
C     THIS PROGRAM CALCULATES THE AIR FLOW RATE THRU THE                 *
C      ++++    NATURAL DRAFT WET COOLING TOWER   ++++                    *
C                                                                        *
C*************************************************************************
C
C
C
C
C
C
      DIMENSION PSA(250)
      REAL KM
      COMMON/MATOR/H,HT,KM,D1,D2,ELEV,R,PDWAF(15)
      COMMON/VEL/V4,V1,RHO1,RHO2
      COMMON/CONSTA/ CONST1,CONST
      COMMON/ATMOS/PATM
      COMMON/NCALA/PSA,TPS,TS,T4,W1,W2,HA
```

```
      COMMON/TEST/IFLAG
      IFLAG=0
      BAAR=3.1415926535*D1**2/4.
      PIFAAR=3.1415926535*D1*H
      AR4=D2**2/4.*3.1415926535
      TDBSL=TDB+0.003566*ELEV+459.67
      TDB5=TDBSL-0.003566*(ELEV+HT)-459.67
      P1=2116.224*((TDB+459.67)/TDBSL)**5.256
      P4=2116.224*((TDB5+459.67)/TDBSL)**5.256
      I=TWB
      PS1=PSA(I)+(PSA(I+1)-PSA(I))*(TWB-I)
      PS1=PS1*144.
      PS1=PS1-0.000367*P1*(TDB-TWB)*(1.+(TWB-32.)/1571.)
      RHO1=P1/R/(TDB+459.67)*(1.-0.378*PS1/P1)
      W1=0.622*PS1/(P1-PS1)
      T4=TDB
      INDEX=0
44    I=T4
      PS4=PSA(I)+(PSA(I+1)-PSA(I))*(T4-I)
      PS4=PS4*144.
      RHO2=P4/R/(T4+459.67)*(1.-0.378*PS4/P4)
      IF(RHO2.GE.RHO1) GO TO 124
      ARGUM=(2*32.174*HT*(RHO1-RHO2)/RHO2)/(1.+KM*RHO2/RHO1*(AR4/PIFAAR)
     1**2.)
      V4=SQRT(ARGUM)
      V1=(RHO2/RHO1)*V4*AR4/PIFAAR
      AFR=AR4*V4*(RHO1/(1.+W1))*3600.
      IF(INDEX.NE.1) GO TO 123
      AFR=1./2.*(AFR+AFRO)
      IF(ABS(AFRO-AFR)/AFR.LE.0.005 ) GO TO 55
123   RLG=GPM/CONST*10.**9/AFR
      CALL NTUCAL(TWI,TDB,TWB,RLG,DNTU,TCW,NOITT,AFR,WL,N)
      INDEX=1
      AFRO=AFR
      T4=(TCW+TWI)/2.
      GO TO 44
124   IFLAG=1
55    RETURN
      END
```

```
      SUBROUTINE CAPCO(TWB,RANGE,APPRO,RH,Q,COST)
C
C* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C*                                                                            *
C* DETERMINE THE NATURAL DRAFT WATER COOLING TOWER'S COST                     *
C*   - USE MARLEY COMPANY'S CHARTS -                                          *
C*                                                                            *
C* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
C
C
C
      COMMON/UTCOST/ UC(4,3,11,5),DRH(2)
C
      IF(RANGE.LT.15.0 .OR. RANGE.GT.45.0 .OR. RH.LT.25.0
     1.OR. TWB.LT.60.0 .OR. TWB.GT.80.0 .OR. APPRO.LT.10.0
     2.OR. APPRO.GT.30.0) GO TO 999
C
C DETERMINE THE LOWER BOUND OF RANGE
C
      RA=(RANGE-5.)/10.
       I=RA
      IA=I+1
C
C  DETERMINE THE LOWER BOUND OF RELATIVE HUMIDITY
C
      J1=25.0
      DO 100 J=1,2
      J2=J1+DRH(J)
      IF(RH.LT.J2) GO TO 10
100   J1=J2
10    JA=J+1
C
C  DETERMINE THE LOWER BOUND OF APPROACH
C
      AP=(APPRO-8.)/2.
      K=AP
      KA=K+1
C
C  DETERMINE THE LOWER BOUND OF WET BULB TEMPERATURE
```

```
      TW=(TWB-55.0)/5.0
      L=TW
      LA=L+1
C
      IF(I.EQ.4) IA=I
      IF(J.EQ.3) JA=J
      IF(K.EQ.11) KA=K
      IF(L.EQ.5) LA=L
C
      IF(UC(I,J,K,L).LT.0.1 .OR. UC(I,J,K,LA).LT.0.1
     1.OR. UC(I,J,KA,L).LT.0.1 .OR. UC(I,J,KA,LA).LT.0.1
     2.OR. UC(I,JA,K,L).LT.0.1 .OR. UC(I,JA,K,LA).LT.0.1
     3.OR. UC(I,JA,KA,L).LT.0.1 .OR. UC(I,JA,KA,LA).LT.0.1) GO TO 999
C
      IF(UC(IA,J,K,L).LT.0.1 .OR. UC(IA,J,K,LA).LT.0.1
     1.OR. UC(IA,J,KA,L).LT.0.1 .OR. UC(IA,J,KA,LA).LT.0.1
     2.OR. UC(IA,JA,K,L).LT.0.1 .OR. UC(IA,JA,K,LA).LT.0.1
     3.OR. UC(IA,JA,KA,L).LT.0.1 .OR. UC(IA,JA,KA,LA).LT.0.1) GO TO 999
C
C  INTERPOLATION BETWEEN " WET BULB TEMPERATURE "
      RAT=TW-L
C
      AUC11=UC(I,J,K,L)+(UC(I,J,K,LA)-UC(I,J,K,L))*RAT
      AUC12=UC(I,J,KA,L)+(UC(I,J,KA,LA)-UC(I,J,KA,L))*RAT
      BUC11=UC(I,JA,K,L)+(UC(I,JA,K,LA)-UC(I,JA,K,L))*RAT
      BUC12=UC(I,JA,KA,L)+(UC(I,JA,KA,LA)-UC(I,JA,KA,L))*RAT
      AUC21=UC(IA,J,K,L)+(UC(IA,J,K,LA)-UC(IA,J,K,L))*RAT
      AUC22=UC(IA,J,KA,L)+(UC(IA,J,KA,LA)-UC(IA,J,KA,L))*RAT
      BUC21=UC(IA,JA,K,L)+(UC(IA,JA,K,LA)-UC(IA,JA,K,L))*RAT
      BUC22=UC(IA,JA,KA,L)+(UC(IA,JA,KA,LA)-UC(IA,JA,KA,L))*RAT
C
C  INTERPOLATION BETWEEN " APPROACH "
C
      RAT=AP-K
      AUC1=AUC11+(AUC12-AUC11)*RAT
      BUC1=BUC11+(BUC12-BUC11)*RAT
      AUC2=AUC21+(AUC22-AUC21)*RAT
      BUC2=BUC21+(BUC22-BUC21)*RAT
C
C  INTERPOLATION BETWEEN " RELATIVE HUMIDITY "
      UC1=AUC1+(BUC1-AUC1)*(RH-J1)/(J2-J1)
      UC2=AUC2+(BUC2-AUC2)*(RH-J1)/(J2-J1)
C
C  INTERPOLATION BETWEEN " RANGE "
C
      UCOST=UC1+(UC2-UC1)*(RA-I)
C
C  CALCULATE TOWER COST
C
      COST=UCOST*Q*10.**6
      RETURN
999   COST=0.0
      RETURN
      END



      SUBROUTINE NTUCAL (TWI,TDB,TWB,RLG,DNTU,TWO,NOITT,AFR,WL,N)
C
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *  CROSSFLOW CALCULATION OF COOL WATER TEMPERATURE FOR TOWER          *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
C
C
C
      DIMENSION HW(30),TW(30),PSA(250)
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,QH1,QH2,HA
      COMMON/CONSTA/ CONST1,CONST
      COMMON/ATMOS/PATM
      IT=TWI
      PS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TWI-IT)
      H=0.24*TWI+0.622*PS/(PATM-PS)*(1061.8+0.44*TWI)
      IT=TWB
      TS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TWB-IT)
      HA=0.24*TWB+0.622*TS/(PATM-TS)*(1061.8+0.44*TWB)
      AH=TS-0.000367*PATM*(TDB-TWB)*(1.+(TWB-32.)/1571.)
      WL=0.622*AH/(PATM-AH)
      DO 100 I=1,N
      TW(I)=TWI
  100 HW(I)=H
```

```
      TWBAL=0.
      DO 104 J=1,N
      H=HA
      DO 101 I=1,N
      INDE=0
      KC=0
      DH1=HW(I)-H
      DH=DH1/1.2*DNTU
      GO TO 102
5566  KC=0
      DH=DH*5./6.
      INDE=INDE+1
      IF(INDE.EQ.5) GO TO 555
  102 KC=KC+1
      TW2=TW(I)-DH/RLG
      IF(TW2.LT.1.0) GO TO 5566
      IT=TW2
      PS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TW2-IT)
      HW2=0.24*TW2+0.622*PS/(PATM-PS)*(1061.8+0.44*TW2)
      DHH=(DH1+HW2-H-DH)/2.*DNTU
      DHH=(DHH+DH)/2.
      IF(KC.GE.NOITT) GO TO 106
      DH=DHH
      GO TO 102
  106 TW(I)=TW(I)-DHH/RLG
      IT=TW(I)
      PS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TW(I)-IT)
      HW(I)=0.24*TW(I)+0.622*PS/(PATM-PS)*(1061.8+0.44*TW(I))
  101 H=H+DHH
      TWB2=TWB
   20 ITWB2=TWB2
      PS=PSA(ITWB2)+(PSA(ITWB2+1)-PSA(ITWB2))*(TWB2-ITWB2)
      HA2=0.24*TWB2+0.622*PS/(PATM-PS)*(1061.8+0.44*TWB2)
      IF (HA2.GE.H) GO TO 10
      TWB2=TWB2+5.
      HA22=HA2
      GO TO 20
   10 TWB2=TWB2-4.
   40 ITWB2=TWB2
      PS=PSA(ITWB2)+(PSA(ITWB2+1)-PSA(ITWB2))*(TWB2-ITWB2)
      HA2=0.24*TWB2+0.622*PS/(PATM-PS)*(1061.8+0.44*TWB2)
      IF(HA2.GE.H) GO TO 30
      TWB2=TWB2+1.
      HA22=HA2
      GO TO 40
   30 TWB2=TWB2-(HA2-H)/(HA2-HA22)
  104 TWBAL=TWBAL+TWB2
      TWBAL=TWBAL/N
      TWO=0.0
      DO103I=1,N
  103 TWO=TWO+TW(I)
      TWO=TWO/N
      IT=TWBAL
      TPS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TWBAL-IT)
      QH2=0.622*TPS/(PATM-TPS)
      WL=QH2-WL
      WL=WL*AFR*CONST/10.**9
      RETURN
555   WRITE(6,556)
556   FORMAT(5X,' TOWER WILL NOT OPERATE FOR THIS COMBINATION OF TEMPERA
     1TURES')
      RETURN
      END


      SUBROUTINE POWERS (TEM,Q,TTSTAR)
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C * DETERMINE THROTTLE LEVEL OF TURBINE FOR A GIVEN CONDITION        *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
      DIMENSION HR(50,14),IHP(14)
      COMMON/TURBIN/ HR,IHR,TLOW,FINC
      COMMON/TBPRE/ IPMAX,NPL,DPL,PLMIN
      COMMON/PLEVEL/ PLMAX,UCAPAB,CAPCAP,UENER,ENERLS,TEI,TOTOP
C
      IF(TEM.LT.TLOW)GOTO30
      TT=(TEM-TLOW+FINC)/FINC
      IT=TT
      Q11=HR(IT,1)+(HR(IT+1,1)-HR(IT,1))*(TT-IT)
      Q12=HR(IT,2)+(HR(IT+1,2)-HR(IT,2))*(TT-IT)
```

```
      IF(Q.LT.Q11)GOTO40
      IF(Q.LT.Q12)GOTO50
      Q1=Q12
      IP=2
   20 Q2=HR(IT,IP+1)+(HR(IT+1,IP+1)-HR(IT,IP+1))*(TT-IT)
      IF(Q2.GT.Q)GOTO10
      Q1=Q2
      IP=IP+1
      GOTO20
   10 PL=IP-(Q2-Q)/(Q2-Q1)+1
      TTSTAR=((PL-1.)*DPL+PLMIN)/PLMAX
      RETURN
   30 TTSTAR=((NPL-1.)*DPL+PLMIN)/PLMAX
      RETURN
   50 IP=1
      Q2=Q12
      Q1=Q11
      GOTO10
   40 DQ=(Q11-Q)/(Q12-Q11)
      TTSTAR=(PLMIN-DQ*DPL)/PLMAX
      RETURN
      END

      SUBROUTINE FOGSEN (TDB,TWB,TDA,SH2,SENSI1,SENSI2,SENSI3)
```

See Appendix III for listing.

APPENDIX V

FORTRAN LISTING

COOLING POND

```
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *  DETERMINE THE PERFORMANCE OF COOLING PONDS                          *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
      DIMENSION HR(50,14),IHR(14),PSA(250),FCRR(5),AREAL(10)
      DIMENSION NYEAR(5),FCR(11),AMR(12),GPMLOD(10)
      COMMON/INPU/ PERCEN(12,15,2),HW(10)
      COMMON/TURBIN/ HR,IHR,TLOW,FINC
      COMMON/WFR/ GPML,EGPM,PUMOP1
      COMMON/CONSTA/ CONST1,DONST,CONST
      COMMON/TURB/ POWER,TTDD,TET,TTD,TTDKO,TTDO
      COMMON/CPOND1/ AMR,CLD,MONTH,W2,QSC
      COMMON/CPOND2/ RHO,C
      COMMON/PAREA/ AREA
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,QH1,QH2,HA
      COMMON/TEMP/ ITWBI,ITWBF,ITBD,ITDBF
      COMMON/ECONO/ NYEAR,FC,WC,WW,DR,CCNO,CCN1,ANPOWE,HEIGHT,EFFICW
      COMMON/POWERC/ IPLI,IPLF,MM,PL,LP
      COMMON/ATMOS/ PATM
      COMMON/FPL11/ FPLMAX
      COMMON/CLS/ S,CWATEQ,CBLOWO,CMAINO
      COMMON/CELL/ CELLTH,FAWET
      COMMON/WITREF/ IWRITE,IPUNCH,REFSV,TWBREF
      COMMON/PLEVEL/ PLMAX,UCAPAB,CAPCAP,UENER,ENERLS,TEI,TOTOP
      COMMON/TBPR/ ITPMAX,TETMAX,ICAP
      COMMON/TBPRE/ IPMAX,NPL,DPL,PLMIN
      COMMON/QSTAR/ TQSTAR
      COMMON/MAINTE/ UMAINT
      COMMON/CAPFAC/ CF,EFI
C
C  DRY-BULB AND WET-BULB TEMPERATURE INTERVAL, ITBD, MUST BE GREATER THAN 1
C
      READ(5,109) IPMAX,NPL,DPL,PLMIN
  109 FORMAT(2I10,2F10.0)
      READ(5,110) ((HR(I,J),J=1,NPL),I=1,IPMAX)
  110 FORMAT(10F8.5)
      READ(5,101) (PSA(I),I=1,250)
  101 FORMAT(10F8.5)
      READ(5,102) (IHR(I),I=1,NPL)
  102 FORMAT(14I3)
      READ(5,106) (FCR(I),I=1,11)
  106 FORMAT(10F8.0)
      READ(5,106) (GPMLOD(I),I=1,10)
      READ(5,107) TLOW,FINC
  107 FORMAT(2F10.0)
      READ(5,106) (AREAL(I),I=1,10)
      READ(5,504) NNOTSI,NNOTS,NOWTSI,NOWTS,LOCATI,LOCATF
  504 FORMAT(6I4)
      READ(5,502) CCNO,CCN1
  502 FORMAT(2F10.0)
      READ(5,506) (NYEAR(I),I=1,5),FC,WC,WW
  506 FORMAT(5I4,4F10.0)
      READ(5,106) HEIGHT,EFFICW,UNCOND,UO
      READ(5,106) (HW(I),I=1,10)
      READ(5,507) ITWBI,ITWBF,ITBD,ITDBF
  507 FORMAT(6I4)
      READ(5,108) LP,TTDD,REFSV,TWBREF,PLMAX,UCAPAB
  108 FORMAT(I10,5F10.0)
      READ(5,106) UENER,UDOWN,DAYS,CF,CCC,COO,CWATEQ,CBLOWO
      READ(5,507) IPLI,IPLF,MM
      NTW=(ITWBF-ITWBI)/ITBD+1
      NTD=(ITDBF-ITWBI)/ITBD+1
      NKN=(IPLF-IPLI+0.01)/MM+1
      READ(5,509) PATM,TPMAX
  509 FORMAT(4F10.0)
      READ(5,505) CA,CW1
  505 FORMAT(4F10.2)
      READ(5,509) TBP
      READ(5,509) EF,EN,EFI
      READ(5,801) RHO,C,CLD,MONTH,W2,QSC
  801 FORMAT(3F10.0,I10,2F10.0)
      READ(5,802) (AMR(I),I=1,12)
  802 FORMAT(12F5.2)
      READ(5,803) UFCND,UPUMP,UMAINT,CHT,CMAINO,ULAND
  803 FORMAT(6F10.0)
      READ(5,507) IWRITE,IPUNCH,ITPMAX,IEXTRA,INUCAL,NEWCON
C
      CONST=7.481/60./62.*10.**9
      DONST=CONST
      CONST1=0.124683/62.
C
C  CALCULATE CORRESPONDING FIXED CHARGE RATE
C
```

```
      DO100K=1,5
      Y=NYEAR(K)/4.+1.
      IY=Y
  100 FCRR(K)=FCR(IY)+(FCR(IY+1)-FCR(IY))*(Y-IY)
C
      IF(INUCAL.EQ.0)GOTO300
      PARAME=(1.-EN)*EF/(1.-EF)/EN
      DO400I=1,IPMAX
      DO400J=1,NPL
  400 HR(I,J)=HR(I,J)*PARAME
C
C  FIND FUEL CONSUMPTION WITH OPEN-CYCLE COOLING SYSTEM
C
  300 TBP1=TBP*62.4*13.6/1728.
      NP=(CF-0.49)*10.
      IT=TLOW
      IF(TBP1.GT.PSA(IT))GOTO710
      TQSTAR=HR(1,LP)
      TQST1=HR(1,NP)
      TQST2=HR(1,NP+1)
      GOTO716
  710 IT=IT+5
      IF(TBP1.GT.PSA(IT))GOTO710
  714 IT=IT-1
      IF(TBP1.LT.PSA(IT))GOTO714
      TTEMP=IT+(TBP1-PSA(IT))/(PSA(IT+1)-PSA(IT))
      TTEMP=(TTEMP-TLOW)/2.+1
      IT=TTEMP
      TQSTAR=HR(IT,LP)+(HR(IT+1,LP)-HR(IT,LP))*(TTEMP-IT)
      TQST1=HR(IT,NP)+(HR(IT+1,NP)-HR(IT,NP))*(TTEMP-IT)
      TQST2=HR(IT,NP+1)+(HR(IT+1,NP+1)-HR(IT,NP+1))*(TTEMP-IT)
  716 DQHR=(TQST1/CF-TQSTAR)/(PLMAX*3.6/1055.04-TQST1/CF+TQSTAR)
      FPL1=PLMAX*CF/(1.+DQHR)
      DQHR=(TQST2/(CF+0.1)-TQSTAR)/(PLMAX*3.6/1055.04-TQST2/(CF+0.1)
     1+TQSTAR)
      FPL2=PLMAX*(CF+0.1)/(1.+DQHR)
      TQST=TQST1+(TQST2-TQST1)*(PLMAX*CF-FPL1)/(FPL2-FPL1)
C
C  DETERMINE MAXIMUM ALLOWABLE TURBINE TEMPERATURE
C
      TETMAX=1000.
      IF(ITPMAX.EQ.0)GOTO717
      TBP2=TPMAX*62.4*13.6/1728.
  718 IT=IT+5
      IF(TBP2.GT.PSA(IT))GOTO718
  719 IT=IT-1
      IF(TBP2.LT.PSA(IT))GOTO719
      TETMAX=IT+(TBP2-PSA(IT))/(PSA(IT+1)-PSA(IT))
C
  717 DO6100LOCATE=LOCATI,LOCATF
      READ(5,509) TWBD,TDBD,TWB10,TDB10
      READ(5,508) (((PERCEN(I,J,K),J=1,NTD),I=1,NTW),K=1,NKN)
  508 FORMAT(10F8.6)
C
      SS1=0.
      S=0.
      DO2011LP=IPLI,IPLF,MM
      LP1=(LP-IPLI)/MM+1
      PL=PLMAX
      IF(LP.NE.IPLI) PL=PLMAX*CF
      DO2012IJK=ITWBI,ITWBF,ITBD
      IW1=(IJK-ITWBI)/ITBD+1
      ITDBMA=PSA(IJK)/(0.000367*PATM*(1.+(IJK-32.)/1571.))+IJK
      IF(ITDBMA.GT.ITDBF) ITDBMA=ITDBF
      DO2012IIK=IJK,ITDBMA,ITBD
      ID1=(IIK-ITWBI)/ITBD+1.
 2012 SS1=SS1+PERCEN(IW1,ID1,LP1)
      S=S+SS1*PL
      SS2=SS1
 2011 SS1=0.
      IF(IPLI.EQ.IPLF) SS2=0.
      S=S*8760.
      TEI=PLMAX*(1.-SS2)+PLMAX*CF*SS2+(TQSTAR*(1.-SS2)+TQST*SS2)*293.067
      TEI=TEI/EFI
C
      WRITE(6,610) TWBD,TDBD,TWB10,TDB10,PLMAX,CA,CW1,TTDD,PATM
  610 FORMAT(1H1//10X,'DESIGN WET-BULB TEMPERATURE OF AIR =',F5.1,' F'
     1/10X,'DESIGN DRY-BULB TEMPERATURE OF AIR =',F5.1,' F'/10X,
     2'EXTREME WET BULB TEMPERATURE =',F8.3,' DEG. F'/10X,
     3'EXTREME DRY BULB TEMPERATURE =',F8.3,' DEG. F'/10X,
     4'POWER LEVEL =',F6.0,' MW'/10X,
     5'SPECIFIC HEAT OF AIR AT CONSTANT PRESSURE =',F6.2,' BTU/IB./F'/
     610X,'SPECIFIC HEAT OF WATER =',F6.2,' BTU/LB./F'/
     710X,'DESIGN TERMINAL TEMP. DIFFERENCE =',F5.1,' F'/10X,
```

```
     8'ATMOSPHERIC PRESSURE =',F7.2,' PSIA')
C
      WRITE(6,620) FC,WC,WW,CCNO,CCN1
  620 FORMAT(1H ,9X,'UNIT FUEL COST =',F9.6,' $/KW-HR'/10X,
     1'UNIT SUPPLY WATER COST =',F7.4,' $/1000 GAL'/10X,
     2'UNIT WASTE WATER COST =',F7.4,' $/1000 GAL'/10X,
     3'MAX. TOLERABLE CONCENTRATION OF PROCESS WATER =',F5.0,' PPM'/10X,
     4'SUPPLY WATER CONCENTRATION =',F5.0,' PPM')
C
      WRITE(6,630) TWBREF,PLMAX,UCAPAB
  630 FORMAT(1H ,9X,
     1'CRITICAL WET BULB TEMPERATURE =',F7.2,' DEG. F'/10X,
     2'MAXIMUM POWER OUTPUT =',F8.2,' MW'/10X,
     3'UNIT CAPACITY LOSS COST =',F10.2,' $/MW')
C
      WRITE(6,632) UPOND,ULAND,UPUMP,UMAINT
  632 FORMAT(1H ,9X,'UNIT POND COST =',F8.1,' $/ACRE'/10X,
     1'UNIT ACCESS LAND COST =',F8.1,' $/ACRE'/10X,
     2'UNIT PUMP AND PIPE SYSTEM COST =',F7.2,' $/GPM'/10X,
     3'UNIT MAINTENANCE COST =',F7.2,' $/ACRE-YEAR')
C
      WRITE(6,631) UENER,UDOWN,DAYS
  631 FORMAT(1H ,9X,'REPLACEMENT ENERGY COST =',F8.4,' $/KW-HR'/10X,
     1'REPLACEMENT ENERGY COST DURING DOWNTIME =',F7.4,' $/KW-HR'/10X,
     2'DOWNTIME FOR CONSTRUCTION =',F6.1,' DAYS')
C
      WRITE(6,640) HEIGHT,EFFICW,UNCOND,UO
  640 FORMAT(1H ,9X,'PUMPING HEIGHT OF WATER THROUGH TOWER =',F8.1,
     2' FEET'/10X,'PUMPING EFFICIENCY FOR WATER PUMP =',F7.3/10X,
     3'UNIT CONDENSER COST =',F6.2,' $/SQ. FT.'/10X,
     4'OVERALL CONDENSER COEFFICIENT, U =',F6.1,' BTU/HR/FT2/F')
C
      WRITE(6,662) ITWBI,ITWBF,ITBD,ITDBF
  662 FORMAT(1H ,9X,'INITIAL WET BULB TEMPERATURE =',I4,' DEG. F'/10X,
     1'FINAL WET BULB TEMPERATURE =',I5,' DEG. F'/10X,
     2'INCREMENT OF DRY AND WET BULB TEMPERATURE =',I4,' DEG. F'/10X,
     3'FINAL DRY BULB TEMPERATURE =',I5,' DEG. F')
C
      WRITE(6,650) TLOW,FINC,REFSV
  650 FORMAT(1H ,9X,'LOWEST TEMP. IN TURBINE CHARAC. CHART =',F5.1,
     1' DEG. F'/10X,'TEMP. INCREMENT IN TURBINE CHARAC. MATRIX =',F4.1,
     2' DEG. F'
     3/10X,'REFERENCE SPECIFIC VOLUME OF AIR =',F7.3,'FT3/LB')
C
      IT=TWBD
      TSS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TWBD-IT)
      HAA=0.24*TWBD+0.622*TSS/(PATM-TSS)*(1061.8+0.44*TWBD)
      DO1000II=NNOTSI,NNOTS
      GPML=GPMLOD(II)
      DO1000IW=NOWTSI,NOWTS
      AREA=AREAL(IW)*PLMAX
      EGPM=GPML*AREA*43560.
      TS=TSS
      HA=HAA
C
      WRITE(6,660) TEI,TBP,SS2
  660 FORMAT(1H1,10X,'FUEL CONSUMPTION WITHOUT COOLING SYSTEM =',F9.3,
     1' MW  (TUR. BACK PRE. =',F5.2,' IN.HG)'/10X,
     2'***',F8.5,' OF THE TIME IS NOT OPERATED AT FULL LOADING ***')
C
C  DETERMINE CAPITAL COST OF COOLING PONDS
C
      IIII=-5
      TTD=TTDD
C
      CALL MODELW (TDBC,TWBD,IW,NPL,TET,TQ,TWL,IIII,K,TT,IM)
C
      IF(TET.LT.0)GOTO999
      TTSTAR=1.
      IF(ICAP.EQ.0)GOTO679
C
      CALL POWERS (TET,TQ,TTSTAR)
C
  679 DQHR=(TQ/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TQ/TTSTAR+TQSTAR)
      TQQ=(PLMAX*TTSTAR/(1.+DQHR)+TQ*1055.04/3.6)/EFI
      RP=TQ*CONST/EGPM
      AP=TET-TTD-RP-TWBD
      ITET=TET
      PDESI=(PSA(ITET)+(PSA(ITET+1)-PSA(ITET))*(TET-ITET))*1728./848.64
C
      CAPCO1=UPOND*AREA
      CAPCO2=0.1*AREA*ULAND
C
C  DETERMINE CONDENSER COST, AND PUMP AND PIPE SYSTEM COST
```

```
C
      THW=TET-TTD
      TCW=THW-TQ*CONST/EGPM
      RANGE=THW-TCW
      TTDKO=TTD/RANGE*CONST/EGPM
      RLL=ALOG((RANGE+TTD)/TTD)
      CONCOS=UNCOND*EGPM/CONST/UO*RLL*10.**9
      IF(NEWCON.EQ.0)CONCOS=CCO
      PPCOST=UPUMP*AREA*GPML*43560.
      PPCOSO=0.20*PPCOST
C
C  DETERMINE DOWNTIME COST, AND REPLACEMENT CAPABILITY LOSS
C
      DOWNCO=UDOWN*PL*24.*DAYS*1000.
      PUMOP1=EGPM*HEIGHT*62.4/7.481/60./550./EFFICW*0.7457
      IIII=0
C
      CALL MODELW (TDB10,TWB10,IW,NPL,TET,TQ,TWL,IIII,K,TT,IM)
C
      IF(TET.LT.0.)GOTO999
      TTSTAR=1.
      IF(ICAP.EQ.0)GOTO681
C
      CALL POWERS (TET,TQ,TTSTAR)
C
  681 DQHR=(TQ/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TQ/TTSTAR+TQSTAR)
      FPL=PLMAX*TTSTAR/(1.+DQHR)
      FPL1=PLMAX-FPL
      FPL1=FPL1+PUMOP1/1000.
      FPLMAX=FPL1
      CAPCAP=FPL1*UCAPAB
C
      WRITE(6,661) GPML,EGPM
  661 FORMAT(1H0,10X,
     2'WATER FLOW RATE LOADING =',F9.5,' GPM/SQ. FT.'/11X,
     3'TOTAL WATER FLOW RATE =',F11.0,' GPM')
      WRITE(6,665) AREA,AREAL(IW)
  665 FORMAT(1H0,8X,'*** POND SIZE ***'/11X,
     1'AREA OF COOLING POND =',F13.1,' ACRES'/16X,
     2'(POND AREA PER MW =',F7.3,' ACRE/MW)')
      WRITE(6,663)
  663 FORMAT(1H0,8X,'*** DESIGN CONDITIONS ***')
      WRITE(6,666) THW,TCW,RP,AP,PDESI,TQQ
  666 FORMAT(1H ,10X,'DESIGN HOT WATER TEMPERATURE =',F8.3,' DEG. F'
     1/11X,'DESIGN COLD WATER TEMPERATURE =',F8.3,' DEG. F'/11X,
     2'DESIGN COOLING RANGE =',F8.3,' DEG. F'/11X,
     3'DESIGN APPROACH =',F8.3,' DEG. F'/11X,
     5'DESIGN TURBINE BACK PRESSURE =',F8.4,' IN. HG'/11X,
     5'FUEL CONSUMPTION AT DESIGN CONDITION =',F9.2,' MW')
      IF(ICAP.EQ.1) WRITE(6,667)
  667 FORMAT(1H ,12X,'NOTE ... CAPACITY LOSS AT DESIGN CONDITION')
C
C  COMPUTE OPERATION COST AND TOTAL COST
C
      CALL OPECOS (IW,TOTOPE)
C
      IF(TOTOPE.GT.10.**11)GOTO1001
      CAPCOS=CAPCO1+PPCOST-PPCOSO+CONCOS-CCO-COO+CHT+CAPCAP+DOWNCO
     1+CAPCO2
C
      WRITE(6,604)
  604 FORMAT(1H0,7X,'*** CAPTIAL COSTS ***')
      WRITE(6,602) CAPCO1,CAPCO2,PPCOST,PPCOSO,CONCOS,CCO,COO,CHT,
     1CAPCAP,DOWNCO,CAPCOS
  602 FORMAT(1H ,9X,'CAPITAL COST OF PONDS = $',F21.0/10X,
     1'CAPITAL COST OF ACCESS LAND =',F15.0/10X,
     1'PUMP AND PIPE SYSTEM COST = $',F17.0/10X,
     2'PUMP AND PIPE SYSTEM SALVAGE = ( $',F12.0,')'/10X,
     3'NEW CONDENSER COST = $',F24.0/10X,
     4'SALVAGE VALUE OF OLD CONDENSER = ( $',F10.0,')'/10X,
     5'OTHER OPEN-CYCLE COMPONENTS SALVAGE = ( $',F5.0,')'/10X,
     6'HOOKUP AND TESTING COST = $',F19.0/10X,
     7'REPLACEMENT CAPABILITY COST = $',F15.0/10X,
     8'DOWNTIME COST = $',F29.0/40X,'------------------'/10X,
     9'TOTAL CAPITAL COST = $',F24.0)
C
      IF(IEXTRA.EQ.1) TOTOPE=TOTOP
      IF(IEXTRA.EQ.1) WRITE(6,614)
  614 FORMAT(1H0/12X,'NOTE   OPERATING COSTS ARE BASED ON  EXTRA  CPERAT
     1ING COST')
C
      WRITE(6,612)
  612 FORMAT(1H0,7X,'*** TOTAL COST   --- ANNUAL BASIS --- FIXED CHARGE
     1RATE ***'//10X,'NO. OF YRS',5X,'CAPITAL COST',5X,'ANNUAL OPERATING
```

```
     2 COST',5X,'TOTAL COST',5X,'FIXED CHARGE RATE'/26X,
     3'MILLS/KW-HR',10X,'MILLS/KW-HR',9X,'MILLS/KW-HR')
      DO5000KK=1,5
      CAPCO1=CAPCOS*FCRR(KK)/S
      TOTOP2=TOTOPE
      TOTCOS=CAPCO1+TOTOP2
 5000 WRITE(6,611) NYEAR(KK),CAPCO1,TOTOP2,TOTCOS,FCRR(KK)
  611 FORMAT(1H ,I15,F20.7,F21.7,F20.7,F19.6)
      GOTO1000
  999 WRITE(6,603) EGPM,TET
  603 FORMAT(1H0//10X,'*************'/12X,'TOTAL WATER FLOW RATE THROUGH
     1THESE COOLING PONDS =',F11.0,' GPM'
     2  /12X,'TURBINE TEMPERATURE =',F10.4/12X,
     3'PONDS SIZE IS TOO LARGE')
 1001 TOTCOS=10.**12
 1000 CONTINUE
 6100 CONTINUE
      STOP
      END



      SUBROUTINE OPECOS (IW,TOTOPE)
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *  PROGRAM TO DETERMINE TOTAL ANNUAL OPERATING COST                  *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
      DIMENSION HR(50,14),IHR(14),PSA(250),S1(15),S2(15),S3(15),NYEAR(5)
      COMMON/INPU/ PERCEN(12,15,2),HW(10)
      COMMON/TURBIN/ HR,IHR,TLOW,FINC
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,QH1,QH2,HA
      COMMON/WFR/ GPML,EGPM,PUMOP1
      COMMON/TURB/ POWER,TBP,TET,TTD,TTDKO,TTDO
      COMMON/CONSTA/ CONST1,DONST,CONST
      COMMON/TEMP/ ITWBI,ITWBF,ITBD,ITDBF
      COMMON/ECONO/ NYEAR,FC,WC,WW,DR,CCNO,CCN1,ANPOWE,HEIGHT,EFFICW
      COMMON/POWERC/ IPLI,IPLF,M,FL,LP
      COMMON/ATMOS/ PATM
      COMMON/CLS/ S,CWATEC,CBLOWC,CMAINO
      COMMON/PAREA/ AREA
      COMMON/WITREF/ IWRITE,IPUNCH,REFSV,TWBREF
      COMMON/PLEVEL/ PLMAX,UCAPAB,CAPCAP,UENER,ENERLS,TEI,TOTOP
      COMMON/FPL11/ FPLMAX
      COMMON/TBPR/ ITPMAX,TETMAX,ICAP
      COMMON/TBPRE/ IPMAX,NPL,DPL,PLMIN
      COMMON/QSTAR/ TQSTAR
      COMMON/MAINTE/ UMAINT
      COMMON/CAPFAC/ CF,EFI
C
C  OPERATION DUE TO COOLING PONDS
C
      IF(HW(IW).LT.0.01 )GOTO1002
C
      IF(IWRITE.EQ.1)WRITE(6,899)
C
      IM=0
      TOTOPE=0.
      TOTBLD=0.
      TOTWL=0.
      TOTEI=0.
      TOTFUE=0.
      TOTWAT=0.
      TOTWAW=0.
      TOTMAN=0.
      TOTLOS=0.
      TOTPRO=0.
      CAPLOS=0
      CAPPRO=0.
      ENERLS=0.
      FAPLS=0.
C
      PLC=PLMAX*CF+PUMOP1/1000.
      DO1000LP=IPLI,IPLF,M
      LP1=(LP-IPLI)/M+1
C
      TTTSAV=0.
      KKSAVE=0
      JJJJ=0
C
      DO901IIJ=ITWBI,ITWBF,ITBD
      TWB=IIJ
```

```
      TS=PSA(IIJ)
      IW1=(IIJ-ITWBI)/ITBD+1
      KK=KKSAVE
      TTT=TTTSAV
      I2=0
      ITDBMA=PSA(IIJ)/(0.000367*PATM*(1.+(IIJ-32.)/1571.))+IIJ
      IF(ITDBMA.GT.ITDBF)ITDBMA=ITDBF
      DO910IJ=IIJ,ITDBMA,ITBD
      TDB=IJ
      ID1=(IJ-ITWBI)/ITBD+1
      AH=TS-0.000367*PATM*(TDB-TWB)*(1.+(TWB-32.)/1571.)
      WL1=0.622*AH/(PATM-AH)
      QH1=WL1
      FAP=PUMOP1/1000.
      NP=NPL
      IF(LP.NE.IPLI) NP=(CF-0.49)*10.
C
      CALL MODELW (TDB,TWB,IW,NP,TET,TQ,TWL,JJJJ,KK,TTT,IM)
C
      I2=I2+1
      IF(TET.LT.0)GOTO200
      TTSTAR=1.
      IF(LP.NE.IPLI) TTSTAR=CF
      IF(ICAP.EQ.0)GOTO9666
C
      CALL POWERS (TET,TC,TTSTAR)
C
      CAPPRO=CAPPRC+PERCEN(IW1,ID1,LP1)
 9666 DQHR=(TQ/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TC/TTSTAR+TQSTAR)
      FPL=PLMAX*TTSTAR/(1.+DQHR)
      PL=FPL
      IF(LP.EQ.IPLI)GOTO96680
      IF(ICAP.EQ.0)GOTO96681
      FPL1=CF*PLMAX-FPL
      PL=FPL
      FAP=PUMOP1/1000.
      GOTO96682
96681 TETST=TET
      TQST=TQ
      FPLST=FPL
      TWLST=TWL
      NP=NP+1
      KKST=KK
      TTTST=TTT
C
      CALL MODELW (TDB,TWB,IW,NP,TET1,TQ1,TWL1,JJJJ,KKST,TTTST,IM)
C
      TTSTAR=CF+0.1
      IF(ICAP.EQ.0)GOTO96683
C
      CALL POWERS (TET1,TQ1,TTSTAR)
C
96683 DQHR=(TQ1/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TQ1/TTSTAR+TQSTAR)
      FPL=PLMAX*TTSTAR/(1.+DQHR)
C
      TET=TETST+(TET1-TETST)/(FPL-FPLST)*(PLC-FPLST)
      IF(TET.LT.(TETMAX+0.05))GOTO96684
      FPL1=CF*PLMAX-FPL
      TQ=TQ1
      TET=TETMAX
      TWL=TWL1
      PL=FPL
      FAP=PUMOP1/1000.
      GOTO96682
96684 TQ=TQST+(TQ1-TQST)/(FPL-FPLST)*(PLC-FPLST)
      TWL=TWLST+(TWL1-TWLST)/(FPL-FPLST)*(PLC-FPLST)
      FPL1=0.
      FAP=0.
      PL=PLC
      GOTO96685
96680 FPL1=PLMAX-FPL
96682 FPL1=FPL1+PUMOP1/1000.
96685 ENERLS=ENERLS+FPL1*PERCEN(IW1,ID1,LP1)*8760.
      FAPLS=FAPLS+FAP*PERCEN(IW1,ID1,LP1)*8760.
      EI=(PL+TQ*1.05504/3.6*1000.)*1000./EFI
      FUECOS=FC*EI*8760.
      BLDOWN=TWL*CCN1/(CCNC-CCN1)
      WATCOS=(TWL+BLDOWN)*WC*525.6
      WAWACO=BLDOWN*WW*525.6
      ANUCAP=ENERLS*UENER*1000.
      HOTWTT=TET-TTD
      COLDWT=HOTWTT-TQ*CCNST/EGPM
      ITET=TET
      P=(PSA(ITET)+(PSA(ITET+1)-PSA(ITET))*(TET-ITET))/0.491111
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      TOTBLD=TOTBLD+BLDOWN*PERCEN(IW1,ID1,LP1)*60.*24.*365./326046
      TOTWL=TOTWL+TWL*PERCEN(IW1,ID1,LP1)*60.*24.*365./326046
      EI1=EI/1000.
      TOTEI=TOTEI+EI1*PERCEN(IW1,ID1,LP1)
      TOTPRO=TOTPRO+PERCEN(IW1,ID1,LP1)
      TOTFUE=TOTFUE+FUECOS*PERCEN(IW1,ID1,LP1)
      TOTWAT=TOTWAT+WATCOS*PERCEN(IW1,ID1,LP1)
      TOTWAW=TOTWAW+WAWACO*PERCEN(IW1,ID1,LP1)
      AMANT1=UMAINT*AREA*PERCEN(IW1,ID1,LP1)
      TOTMAN=TOTMAN+AMANT1
      FUECIS=FUECOS*PERCEN(IW1,ID1,LP1)
      WATCOS=WATCOS*PERCEN(IW1,ID1,LP1)
      WAWACO=WAWACO*PERCEN(IW1,ID1,LP1)
      OPCOS=(FUECIS+WATCOS+WAWACO+AMANT1)
     1+FPL1*PERCEN(IW1,ID1,LP1)*8760.*UENER*1000.
      TOTOPE=TOTOPE+OPCOS
      IF(IPUNCH.EQ.1) WRITE(7,701) FUECOS,ANUCAP,TWL,OPCOS,BLDOWN,FPL1,
     1EI,IM
  701 FORMAT(2F10.0,F6.0,F10.0,F5.0,F9.4,F8.0,I1)
  200 IF(IM.LT.2)GOTO475
      IF(I2.NE.1)GO TO 902
      TTTSAV=TTT
      KKSAVE=KK
      GO TO 902
  475 DO9233IJK=IIJ,ITOBMA,ITBD
      IF(IWRITE.EQ.1)WRITE(6,666)
  666 FORMAT(5X,'WET COOLING TOWER IS TOO LARGE TO OPERATE')
      ID1=(IJK-ITWBI)/ITBD+1
      CFF=CF
      IF(LP.EQ.IPLI) CFF=1.0
      ENERLS=ENERLS+PLMAX*CFF*8760.*PERCEN(IW1,ID1,LP1)
      TOTLOS=TOTLOS+PLMAX*8760000.*UENER*PERCEN(IW1,ID1,LP1)*CFF
      TOTOPE=TOTOPE+PLMAX*8760000.*UENER*PERCEN(IW1,ID1,LP1)*CFF
      IM=0
      IF(IPUNCH.EQ.1)WRITE(7,702) IM
  702 FORMAT(I80)
 9233 CONTINUE
      GO TO 901
  902 IF(IWRITE.LT.1)GOTO910
      WRITE(6,601) PL,TWB
  601 FORMAT(1H0,//6X,'POWER =',F5.0,' MW',10X,'TWB =',F8.3,' DEG. F')
      WRITE(6,333)TET,HOTWTT,COLDWT,P,TQ
333   FORMAT(/6X,'TURB.TEMP. = ',F10.4,1X,'DEG.F. ',
     15X,'HOT WATER TEMP. = ',F10.4,1X,'DEG.F.',
     25X,'COLD WATER TEMP. = ',F10.4,1X,'DEG.F.',
     3//,6X,'PRESSURE = ',F8.5,1X,'IN.HG.',
     45X,'HEAT REJECTION = ',F8.5,1X,'BTU*10**9',/)
      WRITE(6,602).
  602 FORMAT(1H0,1X,'TDB',3X,'WATER EVA.',3X,'BLOWDOWN',3X
     1,'PROBABILITY',3X,'FUEL COST',3X,
     2'WATER COST',3X,'WASTE WATER COST',3X,'SUBS ENERGY LOSS',3X,
     3'OPERATING COST'/3X,'F',
     47X,'GPM',9X,'GPM',22X,'$/YEAR',6X,
     5'$/YEAR',10X,'$/YEAR',13X,'$/YEAR',12X,'$/YEAR'/)
      WRITE(6,607)IJ ,TWL,BLDCWN,PERCEN(IW1,ID1,LP1),FUECIS,WATCOS,WAWAC
     1O,ANUCAP,OPCOS,FPL1
  607 FORMAT(1H ,I3,F15.5,F11.4,F13.6,F13.0,F12.1,F16.1,F20.3,F18.1,
     1F10.3)
  910 CONTINUE
      IF(IWRITE.LT.1)GOTO901
      WRITE(6,899)
  899 FORMAT(///1X,130('*')//)
  901 CONTINUE
 1000 CONTINUE
      FUELEX=TOTEI-TEI
      TOTFU1=FUELEX*FC*1000.*8760.
      TOTOPE=TOTOPE-CBLOWO-CMAINO
      TOTOP=TOTOPE-TEI*FC*100C.*8760.
      TOTEI1=TOTEI/TOTPRO
      TOTWL1=TOTWL/(TQSTAR*1055.04/3.6)
      FUELE1=FUELEX/PLMAX
      FPLMA1=FPLMAX/PLMAX
      ENERL1=ENERLS/(PLMAX*8760.)
      FAPLS1=FAPLS/(PLMAX*8760.)
C
      WRITE(6,605) TOTBLD,TOTWL,TOTWL1,TOTEI,TOTEI1,FUELEX,FUELE1,
     1FPLMAX,FPLMA1,ENERLS,ENERL1,FAPLS,FAPLS1
  605 FORMAT(1H ,/10X,'TOTAL ANNUAL BLOWDOWN =',F15.0,' ACRE-FT/YEAR'/
     110X,'TOTAL ANNUAL WATER EVAP. =',F12.0,' ACRE-FT/YEAR'
     2,5X,'(',F10.5,' )'/10X,
     3'TOTAL ENERGY RATE IN =',F12.3,' MW'/10X,
     4'AVERAGE ENERGY RATE IN DURING ACTUAL POWER PRODUCTION ='
     5,F10.3,' MW'//8X,'*** CAPABILITY LOSSES ***'/10X,
     6'EXCESS FUEL CONSUMPTION =',F9.3,' MW',5X,'(',F9.6,' )'/10X,
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     7'MAXIMUM CAPABILITY LOSS =',F9.3,' MW',5X,'(',F9.6,' )'/10X,
     8'ENERGY LOSS =',F15.5,' MW-HR',5X,'(',F9.6,' )'/10X,
     9'WATER PUMP ENERGY LOSS =',F11.4,' MW-HR',5X,'(',F9.6,' )')
      WRITE(6,606) TOTFUE,TOTFU1,ANUCAP,TOTWAT,TOTWAW,TOTMAN,CWATEO
     1,CBLOWO,CMAINO,TOTOPE,TOTOP
  606 FORMAT(1HO/8X,
     1'*** TOTAL ANNUAL COSTS ***'/10X,
     2'TOTAL ANNUAL FUEL COST =',F24.0,' $/YEAR',5X,'OR'/10X,
     3'EXCESS FUEL COST =',F30.0,' $/YEAR'/10X,
     4'TOTAL ANNUAL REPLACEMENT ENERGY LOSS =',F10.0,' $/YEAR'/10X,
     5'TOTAL ANNUAL WATER COST =',F23.0,' $/YEAR'/10X,
     6'TOTAL ANNUAL WASTE WATER COST =',F17.0,' $/YEAR'/10X,
     7'TOTAL ANNUAL MAINTAINANCE COST =',F16.0,' $/YEAR'/10X,
     8'MAKEUP WATER COST WITH OPEN-CYCLE =',F13.0,' $/YEAR'/10X,
     9'BLOWDOWN TREATMENT COST WITH OPEN-CYCLE =',F7.0,' $/YEAR'/10X,
     1'MAINTENANCE COST WITH OPEN-CYCLE =',F14.0,' $/YEAR'/40X,
     2'------------------------'/10X,
     3'TOTAL ANNUAL OPERATING COST =',F19.0,' $/YEAR'/10X,
     4'EXTRA ANNUAL OPERATION COST =',F19.0,' $/YEAR')
C
      TOTFUE=TOTFUE/S
      TOTWAT=TOTWAT/S
      TOTWAW=TOTWAW/S
      TOTMAN=TOTMAN/S
      ANUCAP=ANUCAP/S
      TOTOP=TOTOP/S
      TOTOPE=TOTOPE/S
      WRITE(6,621) TOTFUE,TOTWAT,TOTWAW,TOTMAN,ANUCAP,TOTOPE,TOTOP
  621 FORMAT(1HO,7X,'*** AVERAGE OPERATING COSTS --- IN MILLS/KW-HR ***'
     1/10X,'AVERAGE FUEL COST =',F22.6,' MILLS/KW-HR'/10X,
     1'AVERAGE WATER COST =',F21.6,' MILLS/KW-HR'/10X,
     2'AVERAGE WASTE WATER COST =',F15.6,' MILLS/KW-HR'/10X,
     4'AVERAGE MAINTAINANCE COST =',F14.6,' MILLS/KW-HR'/10X,
     5'AVERAGE CAPACITY LOSS =',F18.6,' MILLS/KW/HR'/10X,
     6'AVERAGE TOTAL OPERATING COST =',F11.6,' MILLS/KW-HR'/10X,
     7'AVERAGE EXTRA OPERATING COST =',F11.6,' MILLS/KW-HR')
      RETURN
 1002 WRITE(6,623)
  623 FORMAT(1HO/10X,'WET COOLING TOWER IS NOT SUFFICIENT TO OPERATE')
      TOTOPE=10.**12
      RETURN
      END



      SUBROUTINE MODELW (TDB,TWB,IW,LP,TET,TQ,THL,IIII,K,TT,IM)
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *  THIS SUBROUTINE CALCULATES THE MODELING RELATIONSHIPS FOR POWER    *
C *   PLANT AND COOLING TOWER .   GIVEN WET AND DRY BULB                *
C *   TEMPERATURES, AND WET TOWER SIZE,    THE RESULTS ARE TURBINE      *
C *   EXHAUST TEMPERATURE, AND HEAT REJECTION.                          *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
      DIMENSION HTRJIN(50,14),IENHTR(14),PSA(250)
      COMMON/TURBIN/HTRJIN,IENHTR,TLOW,FINC
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,QH1,QH2,HA
      COMMON/PAREA/ AREA
      COMMON/TURB/ BL1,BL2,BL3,TTD,TTDKO,TTDO
      COMMON/TBPR/ ITPMAX,TETMAX,ICAP
      COMMON/WFR/ GPML,FGPM,PLMOP1
      COMMON/CONSTA/ CONST1,DCNST,CONST
C
C    IF TWB IS HIGH ENOUGH, THEN COOLING CANNOT TAKE PLACE AT ALL UNTIL
C    TURBINE CONDENSER TEMPERATURE IS HIGHER. THUS, WILL SKIP TO
C    HIGHER TURBINE TEMPERATURE.
C
C    IF TWB IS LOW ENOUGH,  COOLING WATER FREEZES, WHICH
C    IS NEVER DESIRED.  THUS WHENEVER COOLING WATER WOULD HAVE BEEN
C    COOLED BELOW FREEZING ANYWHERE IN THE CYCLE, NO COOLING IS
C    PERFORMED (IMPLYING ALTERNATE SYSTEM USED IN PRACTICE).
C
C   ASSIGN MODEL PARAMETERS FOR TOWER SECTION
C
      ICAP=0.
      IM=2
      IFR=0
      IFRE=-1
      IF(IIII.GT.0.5)GOTO1000
      WL=0.
      TPS=0.
      TWBAL=0.
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      QH2=0.
      IF(IIII.EQ.-5)GOTO991
      TTDO=HTRJIN(1,LP)*TTDKO
      GOTO992
  991 TTDO=TTD
C
C  ASSIGN INITIAL TRIAL TURBINE TEMPERATURE
C
  992 K=0
   99 IFRE=IFRE+1
  201 K=K+1
      IF(K.GT.IENHTR(LP))GOTO999
      TT=TLOW+(K-1)*FINC
      IF(IIII.EQ.-5)GOTO404
      TTD=HTRJIN(K,LP)*TTDKO
  404 TT1=TT-TTD
C
C  COOL THROUGH COOLING SYSTEM IF POSSIBLE TO GET TQ1
C
      IF(TWB.LT.TT1)GOTO403
      GOTO201
  403 WL2=WL
      TPS2=TPS
      TWBAL2=TWBAL
      QH22=QH2
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      CALL COOL (TT1,FGPM,AREA,TDB,TWB,TT2,WL)
C
      IF(TT2.GT.32.)GOTO503
      GOTO99
C
C  DETERMINE DIRECTION OF APPROACH TO INTERSECTION OF CURVES
C
  503 TQ1=(TT1-TT2)*FGPM/CONST
      IF(TQ1.LT.HTRJIN(K,LP))GOTO100
C
C  REACH INTERSECTION BY DECREASING TURBINE TEMPERATURE
C
      IF(IFRE.GT.0.)GOTO206
      IF(TWB.GT.(TLOW-FINC-TTDO))GOTO104
C
C  COOLING CURVE ENDS MORE THAN 1 DECREMENT BELOW TLOW
C
  206 TT=TT-FINC
      IF(K.GT.1) K=K-1
      TT1=TT-TTDO
      IF(TT1.LT.32.)GOTO703
      IF(TWB.GT.TT1)GOTO304
C
C  COOLING THROUGH COOLING SYSTEM IF POSSIBLE TO GET TQ2
C
      WL2=WL
      TPS2=TPS
      TWBAL2=TWBAL
      QH22=QH2
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      CALL COOL (TT1,FGPM,AREA,TDB,TWB,TT2,WL)
C
      IF(TT2.GT.32.)GOTO505
      IFR=1
  505 TQ2=(TT1-TT2)*FGPM/CONST
      IF(TQ2.LT.HTRJIN(K,LP))GOTO105
      IF(IFR.EQ.1)GOTO703
      TQ1=TQ2
      GOTO206
C
C  INTERPOLATE FOR TQ,TET,TWL
C
  105 TQ=HTRJIN(K,LP)
      HTDIF1=(TQ-TQ2)/(TQ1-TQ2)
      TET=TT+HTDIF1*FINC
      IF(IIII.EQ.-5)GOTO405
      TTD=TQ*TTDKO
  405 TWL= WL+HTDIF1*(WL2-WL)
      TPS=TPS+HTDIF1*(TPS2-TPS)
      TWBAL=TWBAL+HTDIF1*(TWBAL2-TWBAL)
      QH2=QH2+HTDIF1*(QH22-QH2)
      IF(IFR.EQ.0)GOTO210
C  (PREVIOUS TOWER COOLING INDICATES THAT THE OPERATING CHARACTERISTICS
C   CURVE FOR THE COOLING SYSTEM ENDS IN THE SAME TEMPERATURE INTERVAL
C   AS TET)
C  COOL THROUGH COOLING SYSTEM USING TET, TQ FOR CHECK
```

```
C
  211 TT1=TET-TTD
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      CALL COOL (TT1,FGPM,AREA,TDB,TWB,TT2,WL)
C
      IF(TT2.GT. 32.)GOTO210
      GOTO703
  210 CONTINUE
      IIII=1
      RETURN
  304 TT=TT+FINC
C
C  COOLING CURVES END JUST BELOW TT AND DECREMENTING TURBINE TEMPERATURE
C   WILL NOT INTERSECT IT
C
C  DETERMINE PROPER VALUE OF HTRJIN(1,LP)
C  DOUBLE INTERPOLATE FOR TQ,TET,TWL
C
  104 IF(K.GT.1)GOTO106
      HQ=HTRJIN(1,LP)
      GOTO107
  106 HQ=HTRJIN(K-1,LP)
  107 IF(IIII.GT.-5)GOTO407
      HQ=HQ+(TWB+TTD-TT+FINC)/FINC*(HTRJIN(K,LP)-HC)
      GOTO408
  407 HQ=(HQ+(TWB-TT+FINC)/FINC*(HTRJIN(K,LP)-HQ))/(1.-TTDKO/FINC
     1*(HTRJIN(K,LP)-HQ))
  408 IF(IIII.EQ.-5)GOTO406
      TTD=HQ*TTDKO
  406 TQ=TQ1*HQ/(TQ1+HQ-HTRJIN(K,LP))
      TET=TWB+TTD+(TT-TWB-TTD)*TQ/TQ1
      IF(IIII.EQ.-5)GOTO409
      TTD=TQ*TTDKO
  409 TWL=WL/TQ1*TQ
      TPS=TPS/TQ1*TQ
      TWBAL=TWBAL/TQ1*TQ
      QH2=QH2/TQ1*TQ
      TT=TT-FINC
      IF(K.GT.1)K=K-1
      GOTO211
C
C  REACH INTERSECTION BY INCREMENTING TURBINE TEMPERATURE
C
  100 IF(K.EQ.IENHTR(LP))GOTO999
  108 TT=TT+FINC
      K=K+1
      IF(IIII.EQ.-5)GOTO410
      TTD=HTRJIN(K,LP)*TTDKO
  410 TT1=TT-TTD
C
C  COOL THROUGH SYSTEM TO GET TQ2
C
      WL2=WL
      TPS2=TPS
      TWBAL2=TWBAL
      QH22=QH2
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      CALL COOL (TT1,FGPM,AREA,TDB,TWB,TT2,WL)
C
      TQ2=(TT1-TT2)*FGPM/CONST
      IF(TQ2.GT.HTRJIN(K,LP))GOTO101
      IF(K.EQ.IENHTR(LP))GOTO999
      TQ1=TQ2
      GOTO108
C
C  INTERPOLATE FOR TQ, TET, TWL
C
  101 HTDIF1=HTRJIN(K,LP)-HTRJIN(K-1,LP)
      HTDIF2=TQ2-TQ1
      TQ=(HTRJIN(K,LP)*HTDIF2-TQ2*HTDIF1)/(HTDIF2-HTDIF1)
      TET=TT-(TQ2-TQ)/HTDIF2*FINC
      IF(IIII.EQ.-5)GOTO411
      TTD=TQ*TTDKO
  411 TWL= WL2+(WL-WL2)/HTDIF2*(TQ-TQ1)
      TPS=TPS2+(TPS-TPS2)/HTDIF2*(TQ-TQ1)
      TWBAL=TWBAL2+(TWBAL-TWBAL2)/HTDIF2*(TQ-TQ1)
      QH2=QH22+(QH2-QH22)/HTDIF2*(TQ-TQ1)
      TT=TT-FINC
      K=K-1
      IIII=1
      RETURN
C
```

```
C  RETURN WITH MESSAGE
C
  703 TQ=-50
      TET=-50.
      TWL=-50.
      IIII=0
      IM=-50
      RETURN
C
C  FIND INTERSECTION WHEN WET-BULB TERPERATURE INCREASES
C
C  REACH INTERSECTION BY INCREMENTING TURBINE TEMPERATURE
C
 1000 TTD=HTRJIN(K,LP)*TTDKO
      TT1=TT-TTD
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      CALL COOL (TT1,FGPM,AREA,TDB,TWB,TT2,WL)
C
      TQ1=(TT1-TT2)*FGPM/CONST
 1004 TT=TT+FINC
      IF(TT.LT.(TLOW+0.001))GOTO1001
      IF(K.EQ.IENHTR(LP))GOTO999
      K=K+1
      HQ=HTRJIN(K,LP)
      HTDIF1=HQ-HTRJIN(K-1,LP)
      GOTO2070
 1001 HQ=HTRJIN(1,LP)
      HTDIF1=0.
 2070 TTD=HQ*TTDKO
      TT1=TT-TTD
      WL2=WL
      TPS2=TPS
      TWBAL2=TWBAL
      QH22=QH2
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      CALL COOL (TT1,FGPM,AREA,TDB,TWB,TT2,WL)
C
      TQ2=(TT1-TT2)*FGPM/CONST
      IF(TQ2.GT.HQ)GOTO1010
      TQ1=TQ2
      GOTO1004
C
C  INTERPOLATE FOR TQ, TET, TWL
C
 1010 IF(K.GT.1)GOTO412
      TTD=HTRJIN(1;LP)*TTDKO
      GOTO413
  412 TTD=HTRJIN(K-1,LP)*TTDKO
  413 IF(TWB.LT.(TT-FINC-TTD))GOTO1011
      TQ1=TQ2
      GOTO104
 1011 HTDIF2=TQ2-TQ1
      TQ=(HQ*HTDIF2-TQ2*HTDIF1)/(HTDIF2-HTDIF1)
      TET=TT-(TQ2-TQ)/HTDIF2*FINC
      TTD=TQ*TTDKO
      TWL= WL2+(WL-WL2)/HTDIF2*(TQ-TQ1)
      TPS=TPS2+(TPS-TPS2)/HTDIF2*(TQ-TQ1)
      TWBAL=TWBAL2+(TWBAL-TWBAL2)/HTDIF2*(TQ-TQ1)
      QH2=QH22+(QH2-QH22)/HTDIF2*(TQ-TQ1)
      IF(K.GT.1) K=K-1
      TT=TT-FINC
      RETURN
  999 IF(TT1.LT.32.0.OR.TT2.LT.32.)GOTO703
      ICAP=1
      TT1=TETMAX-TTD
C
      CALL COOL (TT1,FGPM,AREA,TDB,TWB,TT2,TWL)
C
      TQ=(TT1-TT2)*FGPM/CONST
      TET=TETMAX
      IF(IIII.NE.-5)TTD=TQ*TTDKO
      RETURN
      END




      SUBROUTINE COOL(THOT,W,A,TA,TWB,TCOLD,WLOSS)
C
      DIMENSION AMR(12),PSA(250),X(2),FX(2)
      COMMON/CPOND1/ AMR,CLD,MONTH,W2,QSC
```

```
      COMMON/CPOND2/ RHO,C
      COMMON/ATMOS/ PATM
      COMMON/NCALA/ PSA,TPS,TSK,TWBAL,QH1,QH2,HA
C
C  * * * *  COOLING POND  NO.1 * * * *
C  DEFINITION OF TERMS
C  A=POND AREA  FT**2  (1ACRE=43560 FT**2)
C  RHO=DENITY OF WATER
C  C=SPECIFIC HEAT OF WATER
C  DELTC=TEMPERATURE DIFFERENCE ACROSS CONDENSER   TIN-TOUT
C  W=WATER FLOW RATE   FT**3/DAY
C  HREJ=HEAT REJECTION RATE AT THE CONDENSER BTU/DAY   PER ACRE
C  X=DUMMY SURFACE TEMP.
C  FX=DUMMY RESISUAL OF HEAT BALANCE EQ.
C
C  WATER DATA FROM P.439 J.P.HOLMAN
C     *     *     *     *
C         BISECTION METHOD
C
C       CALCULATE SURFACE TEMPERATURE
C           COLD WATER TEMPERATURE
C
C  CONVERT 'ACRE' TO 'FT**2'
      A=A*43560.
C  CONVERT 'GPM' TO 'FT**3/DAY'
      W=W*192.4992
C
      X(1)=32.0
      X(2)=150.
      XBAR=(X(1)+X(2))/2.0
      CALL MIX(X(1),RES,A,TA,W,TWB,THOT,WLOSS)
      FX(1)=RES
      CALL MIX(X(2),RES,A,TA,W,TWB,THOT,WLOSS)
      FX(2)=RES
C  TEST TO BE SURE THERE IS XERO BWTWEEN X(1)&X(2)
      PROD=FX(1)*FX(2)
      IF(PROD) 21,22,22
22    CONTINUE
C   FLAG THAT PROD IS POSITIVE
      GO TO 10
21    CONTINUE
C
C  NBI=NO. OF BISECTIONS TO SEARCH FOR SURFACE TEMP.
      NBI=15
      DO 23 KK=1,NBI
      XBAR=(X(1)+X(2))/2.0
      CALL MIX(XBAR,RES,A,TA,W,TWB,THOT,WLOSS)
      FBAR=RES
C  TEST RESIDUAL AT MIDPOINT
      IF(FX(1)*FBAR) 24,25,26
24    CONTINUE
      X(2)=XBAR
      FX(2)=FBAR
      GO TO 23
26    CONTINUE
      X(1)=XBAR
      FX(1)=FBAR
23    CONTINUE
25    CONTINUE
      TS=XBAR
      TCOLD=TS
10    CONTINUE
      A=A/43560.
      W=W/192.4992
      TWBAL=TA+10.
      RETURN
      END



      SUBROUTINE MIX(TS,RES,A,TA,W,TWB,THOT,WLOSS)
C
      DIMENSION AMR(12),PSA(250)
      COMMON/NCALA/ PSA,TPS,BS,TWBAL,QH1,QH2,HA
      COMMON/CPOND1/ AMR,CLD,MONTH,W2,QSC
      COMMON/CPOND2/ RHO,C
      COMMON/ATMOS/ PATM
C
C   RADIATION HEAT TRANSFER QR (BTU/DAY-FT**2)  DAILY AVERAGES
C   QR=QA-QAR+QS-QSR
C   A...ATMOSPHERIC RAD. AND REFLECTED RAD.
C   S...SOLAR       RAD. AND REFLECTED RAD.
```

```
C    QAN=QA-QAR
C    AMR(I)=AVE. MONTHLY RFLECTION
C    QSC=CLEAR SKY SOLAR RAD. (FROM 100( CURVE R&S P. 1-14)
C    MONTH=INDEX OF MONTH UF YEAR
C    LINEAR APPROXIMATION (P.1-21 R&S)
C
C   CONVERT 'PSIA' TO 'MMHG'
        P=PATM*51.719
        QAN=800.+28.*TA
        QS=QSC*(1.0-.65*CLD**2)
        QSR=AMR(MONTH)*QS
        QR=QAN+QS-QSR
C
C    QW=BACK RAD. TERM-LARGEST SINGLE ITEM IN ENERGY BUDGET
C    TS=WATER SURFACE TEMPERATURE DEGREES F
C    A GOOD LINEAR APPROXIMATICN (P. 1-24 R&S)
        QW=1600.+23.*TS
C    THE EXACT EXPRESSION
        RAN=TS+460.
        QW=.41E-7*RAN*RAN*RAN*RAN
C
C   QEVAP=EVAPORATION HEAT FLOW/UNIT AREA
C
C   W2=WIND SPEED (MPH) AT 2 METERS
C    P=ATMOSPHERIC PRESSURE (MMHG)
C   EA=AIR VAPOR PRESSURE (MMHG)
C   TS=WATER SURFACE TEMPERATURE (DEGREES F)
C   ES=SATURATION VAPOR PRESSURE AT TS
C
C
C   CALCULATE THE  SAT. VAPOR PRESSURE OF AIR MUST CVER THE POND SURFACE
C   I.E. THE AIR TEMPERATURE IS EQUAL TH THE WATER SURFACE TEMP.
        ITS=TS
        T1=ITS
        T2=ITS+1
C   CF IS CONVERSION FACTOR (PSI TO MMHG)
        CF=51.6144
C   USE LINEAR INTERPOLATION TO APPROX ES FOR NON-INTERBER VALUES OF TS
        ES=PSA(ITS)+(PSA(ITS+1)-PSA(ITS))*(TS-T1)/(T2-T1)
C   CONVERT  (PSI TO MMHG)
        ES=ES*CF
C   NOW CALCULATE EA, THE VAPOR PRESSURE OF AIR AS FCN OF TDB ANDTWB
C   EA= V.P. OF AIR
        ITS=TWB
        T1=ITS
        T2=ITS+1
        ESWB=PSA(ITS)+(PSA(ITS+1)-(ITS))*(TWB-T1)/(T2-T1)
        ESWB=ESWB*CF
C   THE FOLLOWING IS AN EMPERICAL RELATIONSHIP (P.63 &AILY, MACAGNO,JFK)
        EA=ESWB-.000367*P*(TA-TWB)*(1.0+(TWB-32.0)/1571.0)
C
C
C   SUBSCRIPT V INDICATES VIRTUAL TEMP.
C   SUBSCRIPT R INDICVATES ABSLLUTE TEMPERATURE DEGREES RANKINE
        TSR=TS+460.
        TAR=TA+460.
        TSV=TSR/(1.0-.378*ES/P)
        TAV=TAR/(1.0-.378*ES/P)
C   DTHETA=VIRTUAL TEMP DIFFERENCE
        DTHETA=ABS(TSV-TAV)
C
        DELTE=ES-EA
        F=22.4*DTHETA**.33+14.0*W2
        QEVAP=F*DELTE
C    (P.1-44,R&S)
C    IF QEVAP.LT.ZERO, THEN SET QEVAP TO ZERO
C    QEC=THE EVAP. COEFFICIENT FOR THE CONDUCTIVE HEAT TRANSF. TERM
        QEC=QEVAP
        IF(QEVAP) 30,31,31
30      CONTINUE
        QEVAP=0.0
31      CONTINUE
C
C   CALCULATION OF WATER LOSS PER ACRE
C   XLHV=LATENT HEAT OF VAPORIZATION (BTU/LB)
        XLHV=1087.-.54*TS
        EVAP=QEVAP/(RHO*XLHV)
C   WLOSS=WATER LOSS/ACRE (CU. FT./DAY)
        WLOSS=EVAP*A
C   CONVERT 'FT3/DAY' TO 'GPM'
        WLOSS=WLOSS/192.513
C
C
C    *      *      *
```

```
C  CONDUCTION (SENSIBLE) HEAT LOSS.
C  USUALLY SMALL COMAARED TO QEVAP
C  REF.(P.1-42 RES) BEST METHOD AVAILABLE
C  R=BOWEN RATIO
C  C IS CONSTANT (.255 MMHG/DEG. F)
      R=.255*ABS((TS-TA)/DELTE)
      QC=QEC*R
C   *      *      *      *
C
C    *    *    *
C C  H=HEAT LOAD ON POND FROM PLANT BTU/DAY
C  HPA=HEAT LOAD ON POND FROM PLANT BTU/DAY-FT**2
C
      H=RHO*C*W*(THOT-TS)
      HPA=H/A
C     *     *     *     *
C              HEAT BALANCE   MIXED POND
C
C  RES=RESIDUAL OF HAEAT BALANCE EQUATION
C  RES=0 INDICATES THAT THE VALUE OF TS IS CORRECT
      RES=HPA+QR-(QW+QEVAP+QC)
C
      RETURN
      END


      SUBROUTINE POWERS (TEM,Q,TTSTAR)
```

See Appendix III for listing.

APPENDIX VI

FORTRAN LISTING

SPRAY CANAL

```
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *  SPRAY COOLING CANAL
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
      REAL NEWCON,LINCCS,LENGTH,KM
      DIMENSION HR(50,14),IHR(14),PSA(250),PAPCOS(20),
     1AMANT(3,2),FCRR(5),NYEAR(5),FCR(11)
      COMMON/DENSIT/ DAIR(250)
      COMMON/TURBIN/ HR,IHR,TLOW,FINC
      COMMON/CONSTA/ CONST1,DONST,CONST
      COMMON/TURB/ POWER,TTDD,TET,TTD,TTDKO,TTDO
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,QH1,QH2,HA
      COMMON/INPU/ PERCEN(12,15,2)
      COMMON/TEMP/ ITWBI,ITWBF,ITBD,ITDBF
      COMMON/ECONO/ NYEAR,FC,WC,WW,OR,CCNO,CCN1,ANFOWE,HEIGHT,EFFICW
      COMMON/POWERC/ IPLI,IPLF,MM,PL,LP
      COMMON/LOSS/S,CWATEO,CBLOWC,CMAINO
      COMMON/ATMOS/ PATM
      COMMON/MAINTA/ AMANT,CF,EFI
      COMMON/WITREF/ IWRITE,IPUNCH,REFSV,AFRL,TWBREF
      COMMON/PLEVEL/ PLMAX,UCAPAB,CAPCAP,UENER,ENERLS,TEI,TOTOP
      COMMON/TBPR/ ITPMAX,TETMAX,ICAP
      COMMON/TBPFE/ IPMAX,NPL,DPL,PLMIN,PORTIO
      COMMON/TEMPE/TWBD,TDBD
      COMMON/RENEW/TQSTAR
      COMMON/FPL11/FPLMAX
      COMMON/SPRAY/R,MK,N,WINDSP,FF,TFILM
      COMMON/SPRDIS/F(6,3),ROWDIS
      COMMON/PILIC/GPM
C
C  DRY-BULB AND WET-BULB TEMPERATURE INTERVAL, ITBD, MUST BE GREATER THAN 1
C
      READ(5,5555) ((F(I,J),I=1,6),J=1,3)
5555  FORMAT(6F5.2)
      READ(5,109) IPMAX,NPL,DPL,PLMIN
  109 FORMAT(2I10,2F10.0)
      READ(5,110) ((HR(I,J),J=1,NPL),I=1,IPMAX)
  110 FORMAT(10F8.5)
      READ(5,101) (PSA(I),I=1,250)
  101 FORMAT(10F8.5)
      READ(5,101) (DAIR(I),I=1,250)
      READ(5,102) (IHR(I),I=1,NPL)
  102 FORMAT(14I3)
      READ(5,106) (FCR(I),I=1,11)
      READ(5,106) (PAPCOS(I),I=1,20)
  106 FORMAT(10F8.0)
      READ(5,107) TLOW,FINC
  107 FORMAT(2F10.0)
      READ(5,106) UTCOST
      READ(5,502) CCNO,CCN1
  502 FORMAT(2F10.0)
      READ(5,506) (NYEAR(I),I=1,5),FC,WC,WW
  506 FORMAT(5I4,4F10.0)
      READ(5,507) ITWBI,ITWBF,ITBD,ITDBF
  507 FORMAT(6I4)
      READ(5,108) TWBD,TDBD,LP,TTDD,REFSV,TWBREF,PLMAX,UCAPAB
  108 FORMAT(2F10.0,I10,5F10.0)
      READ(5,106) UENER,ULAND,UDOWN,DAYS,CF,CCO,COC,CWATEO,CBLCWO
      READ(5,507) IPLI,IPLF,MM
      NTW=(ITWBF-ITWBI)/ITBD+1
      NTD=(ITDBF-ITWBI)/ITBD+1
      NKK=(IPLF-IPLI)/MM+1
      READ(5,508) (((PERCEN(I,J,K),J=1,NTD),I=1,NTW),K=1,NKK)
  508 FORMAT(10F8.6)
      READ(5,509) PATM,TPMAX
  509 FORMAT(3F10.0)
      READ(5,505) CA,CW1
  505 FORMAT(4F10.2)
      READ(5,509) TBP,TWB10,TDB10
      READ(5,509) EF,EN,EFI
      READ(5,507) IWRITE,IPUNCH,ITPMAX,IEXTRA,INUCAL
      READ(5,106) EFFICW,UNCOND,UD,EFFICA,HEIGHT
      READ(5,106)CHT,CMAINO
      READ(5,1212)NEWCCN
1212  FORMAT(F5.0)
      READ(5,106) RP
C
      WINDSP=8.
      CONST=7.481/60./62.*10.**9
      DONST=CONST
      CONST1=0.124683/62.
C
C  CALCULATE CORRESPONDING FIXED CHARGE RATE
```

```
C
      DO100K=1,5
      NEAR=NYEAR(K)
      Y=NYEAR(K)/4.+1.
      IY=Y
100   FCRR(K)=FCR(IY)+(FCR(IY+1)-FCR(IY))*(Y-IY)
C
      IF(INUCAL.EQ.0)GOTO300
      PARAME=(1.-EN)*EF/(1.-EF)/EN
      DO400I=1,IPMAX
      DO400J=1,NPL
  400 HR(I,J)=HR(I,J)*PARAME
C
  300 SS1=0.
      S=0.
      DO2011LP=IPLI,IPLF,MM
      LP1=(LP-IPLI)/MM+1
      PL=PLMAX
      IF(LP.NE.IPLI) PL=PLMAX*CF
      DO2012IJK=ITWBI,ITWBF,ITBD
      IW1=(IJK-ITWBI)/ITBD+1
      ITDBMA=PSA(IJK)/(0.000367*PATM*(1.+(IJK-32.)/1571.))+IJK
      IF(ITDBMA.GT.ITDBF) ITDBMA=ITDBF
      DO2012IIK=IJK,ITDBMA,ITBD
      ID1=(IIK-ITWBI)/ITBD+1.
 2012 SS1=SS1+PERCEN(IW1,ID1,LP1)
      S=S+SS1*PL
      SS2=SS1
 2011 SS1=0.
      IF(IPLI.EQ.IPLF) SS2=0.
      S=S*8760.
C
C  FIND FUEL CONSUMPTION WITH OPEN CYCLE COOLING SYSTEM
C
      TBP1=TBP*62.4*13.6/1728.
      NP=(CF-0.49)*10.
      LP=NPL
      IT=TLOW
      IF(TBP1.GT.PSA(IT))GOTO710
      TQSTAR=HR(1,LP)
      TQST1=HR(1,NP)
      TQST2=HR(1,NP+1)
      GOTO716
  710 IT=IT+5
      IF(TBP1.GT.PSA(IT))GOTO710
  714 IT=IT-1
      IF(TBP1.LT.PSA(IT))GOTO714
      TTEMP=IT+(TBP1-PSA(IT))/(PSA(IT+1)-PSA(IT))
      TTEMP=(TTEMP-TLOW)/2.+1
      IT=TTEMP
      TQSTAR=HR(IT,LP)+(HR(IT+1,LP)-HR(IT,LP))*(TTEMP-IT)
      TQST1=HR(IT,NP)+(HR(IT+1,NP)-HR(IT,NP))*(TTEMP-IT)
      TQST2=HR(IT,NP+1)+(HR(IT+1,NP+1)-HR(IT,NP+1))*(TTEMP-IT)
  716 DQHR=(TQST1/CF-TQSTAR)/(PLMAX*3.6/1055.04-TQST1/CF+TQSTAR)
      FPL1=PLMAX*CF/(1.+DQHR)
      DQHR=(TQST2/(CF+0.1)-TQSTAR)/(PLMAX*3.6/1055.04-TQST2/(CF+0.1)
     1+TQSTAR)
      FPL2=PLMAX*(CF+0.1)/(1.+DQHR)
      TQST=TQST1+(TQST2-TQST1)*(PLMAX*CF-FPL1)/(FPL2-FPL1)
      TEI=PLMAX*(1.-SS2)+PLMAX*CF*SS2+(TQSTAR*(1.-SS2)+TQST*SS2)*293.067
      TEI=TEI/EFI
      EGPM=TQSTAR*CONST/RP
      GPM=EGPM
C
C  DETERMINE MAXIMUM ALLOWABLE TURBINE TEMPERATURE
C
      TETMAX=1000.
      IF(ITPMAX.EQ.0)GOTO717
      TBP2=TPMAX*62.4*13.6/1728.
  718 IT=IT+5
      IF(TBP2.GT.PSA(IT))GOTO718
  719 IT=IT-1
      IF(TBP2.LT.PSA(IT))GOTO719
      TETMAX=IT+(TBP2-PSA(IT))/(PSA(IT+1)-PSA(IT))
C
  717 WRITE(6,610) TWBD,TDBD,TWB10,PLMAX,CA,CW1,TTDD,PATM
  610 FORMAT(1H1//10X,'DESIGN WET-BULB TEMPERATURE OF AIR =',F5.1,' F'
     1/10X,'DESIGN DRY-BULB TEMPERATURE OF AIR =',F5.1,' F'/10X,
     2'EXTREME WET BULB TEMPERATURE =',F8.5,' DEG.F'/10X,
     2'POWER LEVEL =',F6.0,' MW'/10X,
     5'SPECIFIC HEAT OF AIR AT CONSTANT PRESSURE =',F6.2,' BTU/LB./F'/
     610X,'SPECIFIC HEAT OF WATER =',F6.2,' BTU/LB./F'/
     710X,'DESIGN TERMINAL TEMP. DIFFERENCE =',F5.1,' F'/10X,
     8'ATMOSPHERIC PRESSURE =',F7.2,' PSIA')
```

```
C
      WRITE(6,620) FC,WC,WW,CCNO,CCN1
  620 FORMAT(1H ,9X,
     4'UNIT FUEL COST =',F10.6,' $/KW-HR'/10X,
     5'UNIT SUPPLY WATER COST =',F7.4,' $/1000 GAL'/10X,
     6'UNIT WASTE WATER COST =',F7.4,' $/1000 GAL'/10X,
     7'MAX. TOLERABLE CONCENTRATION OF PROCESS WATER =',F5.0,' PPM'/10X,
     8'SUPPLY WATER CONCENTRATION =',F5.0,' PPM')
C
      WRITE(6,630) TWBREF,PLMAX,UCAPAB
  630 FORMAT(1H ,9X,
     6'CRITICAL WET BULB TEMPERATURE =',F7.2,' DEG. F'/10X,
     7'MAXIMUM POWER OUTPUT =',F8.2,' MW'/10X,
     8'UNIT CAPACITY LOSS COST =',F10.2,' $/MW')
C
      WRITE(6,631) UENER,ULAND,UCOWN,DAYS
  631 FORMAT(1H ,9X,'REPLACED ENERGY COST =',F8.4,' $/KW-HR'/10X,
     1'UNIT LAND COST =',F8.3,' $/ACRE'/10X,
     2'REPLACED ENERGY COST DURING DOWNTIME =',F7.4,' $/KW-HR'/10X,
     3'DOWNTIME FOR CONSTRUCTION =',F6.1,' DAYS')
C
      WRITE(6,640) HEIGHT,EFFICW,UNCOND,UO
  640 FORMAT(1H ,9X,'PUMPING HEIGHT OF WATER THROUGH CANAL =',F8.1,
     2' FEET'/10X,'PUMPING EFFICIENCY FOR WATER PUMP =',F7.3/10X,
     3'UNIT CONDENSER COST =',F6.2,' $/SQ. FT.'/10X,
     4'OVERALL CONDENSER COEFFICIENT, U =',F6.1,' BTU/HR/FT2/F')
C
      WRITE(6,662) ITWBI,ITWBF,ITBD,ITDBF
  662 FORMAT(1H ,9X,'INITIAL WET BULB TEMPERATURE =',I4,' DEG. F'/10X,
     1'FINAL WET BULB TEMPERATURE =',I5,' DEG. F'/10X,
     2'INCREMENT OF DRY AND WET BULB TEMPERATURE =',I4,' DEG. F'/10X,
     3'FINAL DRY BULB TEMPERATURE =',I5,' DEG. F')
C
      WRITE(6,650) TLOW,FINC
  650 FORMAT(1H ,9X,'LOWEST TEMP. IN TURBINE CHARAC. CHART =',F5.1,
     1' DEG. F'/10X,'TEMP. INCREMENT IN TURBINE CHARAC. MATRIX =',F4.1,
     2' DEG. F')
C
      IT=TWBD
      TSS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TWBD-IT)
      HAA=0.24*TWBD+0.622*TSS/(PATM-TSS)*(1061.8+0.44*TWBD)
C
      DO 1000 MK=1,4,3
      IF(MK.EQ.1) NNN=170
      IF(MK.EQ.4) NNN=90
      DO 1000 N=10,NNN,10
      IF(MK.EQ.1) ROWDIS=40.
      IF(MK.EQ.4) ROWDIS=60.
      GPMMOD=10000.
      R=GPMMOD/GPM
C
      TS=TSS
      HA=HAA
C
      IIII=-5
      TTC=TTDD
C
      CALL MODELW (TDBD,TWBD,IW,NPL,TET,TQ,TWL,IIII,K,TT,IM)
C
      IF(TET.LT.0)GOTO999
      TTSTAR=1.
      IF(ICAP.EQ.0)GOTO679
C
      CALL POWERS (TET,TQ,TTSTAR)
C
679   DQHR=(TQ/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TQ/TTSTAR+TQSTAR)
      TQQ=(PLMAX*TTSTAR/(1.+DQHR)+TQ*1055.04/3.6)/EFI
      RP=TQ*CONST/EGPM
      AP=TET-TTD-RP-TWBD
      ITET=TET
      PDESI=(PSA(ITET)+(PSA(ITET+1)-PSA(ITET))*(TET-ITET))*1728./848.64
C
C  DETERMINE CONDENSER COST, AND PUMP AND PIPE SYSTEM COST
C
      THW=TET-TTD
      TCW=THW-TQ*CONST/EGPM
      RANGE=THW-TCW
      TTDKO=TTD/RANGE*CCNST/EGPM
      RLL=ALOG((RANGE+TTD)/TTD)
      CONCOS=UNCOND*EGPM/CCNST/UO*RLL*10.**9
      IF(NEWCON.EQ.0) CONCOS=CCO
      PMPHD=40.
      PPCOST=(1.05*PMPHD/100.+32.*0.0301*PMPHD/100.+0.65)*GPM
      PPCOSO=0.20*PPCOST
```

```
C
C  DETERMINE CAPITAL COST OF THE CANAL
C
      COST=UTCOST*MK*N
      CAPCO1=COST
C  DETERMINE DOWNTIME, LINING, EXCAVATION, AND ADDITIONAL LAND COSTS
C
      WIDTH=(MK-1)*75.+100.
      LENGTH=(N-1)*100.+200.
      DOWNCO=UDOWN*PL*24.*DAYS*1000.
      BASEAR=LENGTH*WIDTH /(4.35*10.**4)*2.5
      ALANDC=ULAND*BASEAR
      LINCOS=0.93*LENGTH*(WIDTH+3.245)
      EXCCOS=10./27.*LENGTH*(WIDTH-30.)*2.50
C
C  DETERMINE REPLACEMENT CAPABILITY LOSS
C
      PUMOP1=EGPM*HEIGHT*62.4/7.481/60./550./EFFICW*0.7457
      PUMOP2=MK*N*75.*0.7457
      PUMOP1=PUMOP1+PUMOP2
      IIII=0
C
      CALL MODELW(TDB10,TWB10,IW,NPL,TET,TQ,TWL,IIII,K,TT,IM)
C
C
      IF(TET.LT.0.0) GO TO 999
      IF(N .GE.120) GO TO 56565
      GO TO 65656
56565 CONTINUE
      ITET=TET
      PDESI=(PSA(ITET)+(PSA(ITET+1)-PSA(ITET))*(TET-ITET))*1728./848.64
      THW=TET-TTD
      TCW=THW-TQ*CONST/EGPM
      RANGE=THW-TCW
      WRITE(6,12345)
      WRITE(6,98765)THW,TCW,RANGE,PDESI
98765 FORMAT(5X,'HOT WATER TEMPERATURE =',F8.3,5X,'COLD WATER TEMPERATUR
     1E =',F8.3,/5X,'RANGE =',F8.3,5X,'TURBINE BACK PRESSURE =',F8.3)
65656 CONTINUE
C
      IF(TET.LT.0) GO TO 999
      TTSTAR=1.
      IF(ICAP.EQ.0)GOTO681
      WRITE(6,55555)TWB10,TET,TQ
55555 FORMAT(2X,F17.5)
C
      CALL POWERS (TET,TQ,TTSTAR)
C
681   DQHR=(TQ/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TQ/TTSTAR+TQSTAR)
      FPL=PLMAX*TTSTAR/(1.+DQHR)
      FPL1=PLMAX-FPL
      FPL1=FPL1+PUMOP1/1000.
      FPLMAX=FPL1
      CAPCAP=FPL1*UCAPAB
C
      WRITE(6,12345)
12345 FORMAT(///100('*'))
      WRITE(6,66666)N,MK
66666 FORMAT(///10X,'NUMBER ALONG THE CANAL =',I4,/10X,
     1'NUMBER ACROSS THE CANAL =',I4///)
      WRITE(6,676)LENGTH,WIDTH
676   FORMAT(/10X,'LENGTH OF THE CANAL =',F8.2,' FT'/10X,
     1'WIDTH OF THE CANAL =',F8.2,' FT')
      WRITE(6,661) EGPM
  661 FORMAT(1H0,10X,
     3'TOTAL WATER FLOW RATE =',F11.0,' GPM')
      WRITE(6,663)
  663 FORMAT(1H0,8X,'*** DESIGN CONDITIONS ***')
      WRITE(6,666) THW,TCW,RP,AP,PDESI,TQ,TQQ
  666 FORMAT(1H ,10X,'DESIGN HOT WATER TEMPERATURE =',F8.3,' DEG. F'
     1/11X,'DESIGN COLD WATER TEMPERATURE =',F8.3,' DEG. F'/11X,
     2'DESIGN COOLING RANGE =',F8.3,' DEG. F'/11X,
     3'DESIGN APPROACH =',F8.3,' DEG. F'/11X,
     5'DESIGN TURBINE BACK PRESSURE =',F8.4,' IN. HG'/11X,
     5'DESIGN HEAT REJECTION =',F8.4,'*10**9  BTU'/11X,
     5'FUEL CONSUMPTION AT DESIGN CONDITION =',F9.2,' MW')
      IF(ICAP.EQ.1) WRITE(6,667)
  667 FORMAT(1H ,12X,'NOTE ... CAPACITY LOSS AT DESIGN CONDITION')
C
C  COMPUTE OPERATION COST AND TOTAL COST
C
      CALL OPECOS (IW,TOTOPE)
C
      IF(TOTOPE.GT.10.**11)GOTO1001
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```
      CAPCOS=CAPCO1+PPCOST+CAPCAP+ALANDC-PPCOSO+DOWNCO-CCO-COO+CHT
     1+EXCCOS+LINCOS
C
      WRITE(6,601) COST
  601 FORMAT(1H ,/10X,'CAPITAL COST FOR THE MODULES     = $',F11.0)
      WRITE(6,604)
  604 FORMAT(1H0,7X,'*** CAPITAL COSTS ***')
      WRITE(6,602) CAPCO1,PPCOST,PPCOSO,CONCOS,CCO,COO,CHT,ALANDC,
     1LINCOS,EXCCOS,CAPCAP,DOWNCC,CAPCOS
  602 FORMAT(1H ,9X,'CAPITAL COST OF MODULES  = $',F18.0/10X,
     1'PUMP AND PIPE SYSTEM COST = $',F17.0/10X,
     2'PUMP AND PIPE SYSTEM SALVAGE = ( $',F12.0,')'/10X,
     3'NEW CONDENSER COST = $',F24.0/10X,
     4'SALVAGE VALUE OF OLD CONDENSER = ( $',F10.0,')'/10X,
     5'OTHER OPEN-CYCLE COMPONENTS SALVAGE = ( $',F5.0,')'/10X,
     6'HOOKUP AND TESTING COST = $',F19.0/10X,
     7'ADDITIONAL LAND COST = $',F22.0/10X,
     7'CANAL LINING COST = $',F25.0/10X,
     7'CANAL EXCAVATION COST = $',F21.0/10X,
     8'REPLACEMENT CAPABILITY COST = $',F15.0/10X,
     9'DOWNTIME COST = $',F29.0/40X,'-----------------'/10X,
     1'TOTAL CAPITAL COST = $',F24.0)
C
      IF(IEXTRA.EQ.1) TOTOPE=TOTOP
      IF(IEXTRA.EQ.1) WRITE(6,614)
  614 FORMAT(1H0/12X,'NOTE : OPERATING COSTS ARE BASED ON "EXTRA" OPERAT
     1ING COST')
C
      WRITE(6,612)
  612 FORMAT(1H0,8X,'*** TOTAL COST 1 --- ANNUAL BASIS --- FIXED CHARGE
     1RATE ***'//12X,'NO. OF YRS',5X,'CAPITAL COST',5X,'ANNUAL OPERATING
     2 COST',4X,'ANNUAL COST',5X,'FIXED CHARGE RATE'/28X,
     3'MILLS/KW-HR',10X,'MILLS/KW-HR',9X,'MILLS/KW-HR')
      DO5000KK=1,5
      CAPCO1=CAPCOS*FCRR(KK)/S
      TOTOP2=TOTOPE
      TOTCOS=CAPCO1+TOTOP2
 5000 WRITE(6,611) NYEAR(KK),CAPCO1,TOTOP2,TOTCOS,FCRR(KK)
  611 FORMAT(1H ,I17,F21.9,F22.9,F20.9,F17.6)
      GOTO1000
999   WRITE(6,603)EGPM,TET
603   FORMAT(//10X,'****************'/12X,'TOTAL WATER FLOW RATE THROUGH
     1 THE SPRAY CANAL =',F11.0,' GPM'/12X,'TURBINE TEMPERATURE =',F10.4
     1)
      WRITE(6,623)
623   FORMAT(10X,'SPRAY COOLING SYSTEM IS TOO LARGE TO OPERATE')
      TOTCOS=10.**12
 1000 CONTINUE
      STOP
 1001 TOTCOS=10.**12
      STOP
      END



      SUBROUTINE OPECOS (IW,TOTOPE)
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *  PROGRAM TO DETERMINE TOTAL ANNUAL OPERATING COST                     *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
C
      REAL NEWCON
      DIMENSION HR(50,14),IHR(14),A(10),B(10),ARWT(10),PSA(250)
      DIMENSION AW(10),BW(10),CW(10),AMANT(3,2)
     1,WTCOST(10),S1(15),S2(15),S3(15),NYEAR(5)
      COMMON/DENSIT/ DAIR(250)
      COMMON/INPU/ PERCEN(12,15,2)
      COMMON/TURBIN/ HR,IHR,TLOW,FINC
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,QH1,QH2,HA
      COMMON/TURB/ POWER,TBP,TET,TTD,TTDKO,TTDO
      COMMON/CONSTA/ CONST1,DONST,CONST
      COMMON/TEMP/ ITWBI,ITWBF,ITBO,ITDBF
      COMMON/ECONO/ NYEAR,FC,WC,WW,DR,CCNO,CCN1,ANPOWE,HEIGHT,EFFICW
      COMMON/POWERC/ IPLI,IPIF,M,PP,LP
      COMMON/ATMOS/ PATM
      COMMON/MAINTA/ AMANT,CF,EFI
      COMMON/LOSS/S,CWATEO,CBLOWC,CMAINO
      COMMON/WITREF/ IWRITE,IPUNCH,REFSV,AFRL,TWBREF
      COMMON/PLEVEL/ PLMAX,UCAPAB,CAPCAP,UENER,ENERLS,TEI,TOTOP
      COMMON/TBPR/ ITPMAX,TETMAX,ICAP
      COMMON/TBPRE/ IPMAX,NPL,DPL,PLMIN,PORTIO
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      COMMON/RENEW/TQSTAR
      COMMON/FPL11/FPLMAX
      COMMON/SPRAY/R,MK,N,WINDSP,FF,TFILM
      COMMON/SPRDIS/F(6,3),ROWDIS
      COMMON/PILIC/GPM
      FGPM=GPM
      EGPM=FGPM
C
C  OPERATION DUE TO SPRAY CANAL
C
C  DRY-BULB AND WET-BULB TEMPERATURE INTERVAL, ITBD, MUST BE GREATER THAN 1
C
      PUMOP1=FGPM*HEIGHT*62./7.481/60./550./EFFICW*0.7457
      PUMOP2=MK*N*75.*0.7457
      PUMOP1=PUMOP1+PUMOP2
C
      IF(IWRITE.EQ.1)WRITE(6,899)
C
      IM=0
      TOTOPE=0.
      TOTBLD=0.
      TOTWL=0.
      TOTEI=0.
      TOTFUE=0.
      TOTWAT=0.
      TOTWAW=0.
      TOTMAN=0.
      TOTLOS=0.
      TOTPRO=0.
      CAPLOS=0
      CAPPRO=0.
      FPLMAX=0.
      ENERLS=0.
      FAPLS=0.
C
      PLC=PLMAX*CF+PUMOP1/1000.
      DO1COOLP=IPLI,IPLF,M
      LP1=(LP-IPLI)/M+1
C
      TTTSAV=0.
      KKSAVE=0
      JJJJ=0
C
      DO901IIJ=ITWBI,ITWBF,ITBD
      TWB=IIJ
      TS=PSA(IIJ)
      IW1=(IIJ-ITWBI)/ITBD+1
      KK=KKSAVE  .
      TTT=TTTSAV
      I2=0
      ITDBMA=PSA(IIJ)/(0.000367*PATM*(1.+(IIJ-32.)/1571.))+IIJ
      IF(ITDBMA.GT.ITDBF)ITDBMA=ITDBF
      IF(IIJ.GT.ITDBMA)GO TO 901
      DO923IJK=IIJ,ITDBMA,ITBD
      TDB=IJK
      ID1=(IJK-ITWBI)/ITBD+1
      AH=TS-0.000367*PATM*(TDB-TWB)*(1.+(TWB-32.)/1571.)
C
      IF(PERCEN(IW1,ID1,LP1).LT.0.00000001) GO TO 923
      FAP=PUMOP1/1000.
      NP=NPL
      IF(LP.NE.IPLI) NP=(CF-0.49)*10.
C
      CALL MODELW (TDB,TWB,IW,NP,TET,TQ,TWL,JJJJ,KK,TTT,IM)
C
      IF(TET.LT.0.0) GO TO 1002
      IF(TWB.EQ.5.AND.TDB.EQ.5 ) GO TO 95135
      GO TO 84951
95135 ITET=TET
      PDESI=(PSA(ITET)+(PSA(ITET+1)-PSA(ITET))*(TET-ITET))*1728./848.64
      THW=TET-TTD
      TCW=THW-TQ*CONST/EGPM
      RANGE=THW-TCW
      WRITE(6,14725)
14725 FORMAT(5X,'LEAST CONDITIONS')
      WRITE(6,26159)THW,TCW,RANGE,PDESI
26159 FORMAT(5X,'HOT WATER TEMPERATURE =',F8.3,5X,'COLD WATER TEMPERATUR
     1E =',F8.3,/5X,'RANGE =',F8.3,5X,'TURBINE BACK PRESSURE =',F8.3)
84951 CONTINUE
      IF(TWB.EQ.95 .AND.TDB.EQ.95 ) GO TO 35724
      GO TO 25814
35724 ITET=TET
      PDESI=(PSA(ITET)+(PSA(ITET+1)-PSA(ITET))*(TET-ITET))*1728./848.64
      THW=TET-TTD
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      TCW=THW-TQ*CONST/EGPM
      RANGE=THW-TCW
      WRITE(6,35791)
35791 FORMAT(5X,'MAXIMUM CONDITIONS')
      WRITE(6,36925)THW,TCW,FANGE,PDESI
36925 FORMAT(5X,'HOT WATER TEMPERATURE =',F8.3,5X,'CCLD WATER TEMPERATUR
     1E =',F8.3,/5X,'RANGE =',F8.3,5X,'TURBINE BACK PRESSURE =',F8.3)
25814 CONTINUE
C
C
      I2=I2+1
      IF(TET.LT.0)GOTO200
      TTSTAR=1.
      IF(LP.NE.IPLI) TTSTAR=CF
      IF(ICAP.EQ.0)GOTO9666
C
      CALL POWERS (TET,TC,TTSTAR)
C
      CAPPRO=CAPPRO+PERCEN(IW1,ID1,LP1)
9666  DQHR=(TQ/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TC/TTSTAR+TQSTAR)
      FPL=PLMAX*TTSTAR/(1.+DQHR)
      PL=FPL
      IF(LP.EQ.IPLI)GOTO96680
      IF(ICAP.EQ.0)GOTO96681
      FPL1=CF*PLMAX-FPL
      PL=FPL
      FAP=PUMOP1/1000.
      GOTO96682
96681 TETST=TET
      TQST=TQ
      FPLST=FPL
      TWLST=TWL
      NP=NP+1
      KKST=KK
      TTTST=TTT
C
      CALL MODELW (TDB,TWB,IW,NP,TET1,TQ1,TWL1,JJJJ,KKST,TTTST,IM)
C
      TTSTAR=CF+0.1
      IF(ICAP.EQ.0)GOTO96683
C
      CALL POWERS (TET1,TQ1,TTSTAR)
C
96683 DQHR=(TQ1/TTSTAR-TQSTAR)/(PLMAX*3.6/1055.04-TQ1/TTSTAR+TQSTAR)
      FPL=PLMAX*TTSTAR/(1.+DQHR)
C
      TET=TETST+(TET1-TETST)/(FPL-FPLST)*(PLC-FPLST)
      IF(TET.LT.(TETMAX+0.05))GOTO96684
      FPL1=CF*PLMAX-FPL
      TQ=TQ1
      TET=TETMAX
      TWL=TWL1
      PL=FPL
      FAP=PUMOP1/1000.
      GOTO96682
96684 TQ=TQST+(TQ1-TQST)/(FPL-FPLST)*(PLC-FPLST)
      TWL=TWLST+(TWL1-TWLST)/(FPL-FPLST)*(PLC-FPLST)
      FPL1=0.
      FAP=0.
      PL=PLC
      GOTO96685
96680 FPL1=PLMAX-FPL
96682 IF(FPL1.LT.0) FPL1=0.
      FPL1=FPL1+(PUMOP1)/1000.
96685 IF(FPL1.GT.FPLMAX.AND.PERCEN(IW1,ID1,LP1).GT.0.000001)FPLMAX=FPL1
      ENERLS=ENERLS+FPL1*PERCEN(IW1,ID1,LP1)*8760.
      FAPLS=FAPLS+FAP*PERCEN(IW1,ID1,LP1)*8760.
      EI=(PL+TQ*1.05504/3.6*1000.)*1000./EFI
      FUECOS=FC*EI*8760.
      BLDOWN=TWL*CCN1/(CCNO-CCN1)
      WATCOS=(TWL+BLDOWN)*WC*525.6
      WAWACO=BLDOWN*WW*525.6
      ANUCAP=ENERLS*UENER*1000.
      HOTWTT=TET-TTD
      COLDWT=HOTWTT-TQ*CONST/FGPM
      ITET=TET
      P=(PSA(ITET)+(PSA(ITET+1)-PSA(ITET))*(TET-ITET))/0.491111
      TOTBLD=TOTBLD+BLDOWN*PERCEN(IW1,ID1,LP1)*60.*24.*365./326046
      TOTWL=TOTWL+TWL*PERCEN(IW1,ID1,LP1)*60.*24.*365./326046
      EI1=EI/1000.
      TOTEI=TOTEI+EI1*PERCEN(IW1,ID1,LP1)
      TOTPRO=TOTPRO+PERCEN(IW1,ID1,LP1)
      TOTFUE=TOTFUE+FUECOS*PERCEN(IW1,ID1,LP1)
      TOTWAT=TOTWAT+WATCOS*PERCEN(IW1,ID1,LP1)
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      TOTWAW=TOTWAW+WAWACO*PERCEN(IW1,ID1,LP1)
      AMANT1=PUMOP1*0.01*8760.*PERCEN(IW1,ID1,LP1)*UENER
      TOTMAN=TOTMAN+AMANT1
      FUECIS=FUECOS*PERCEN(IW1,ID1,LP1)
      WATCOS=WATCOS*PERCEN(IW1,ID1,LP1)
      WAWACO=WAWACO*PERCEN(IW1,ID1,LP1)
      OPCOS=(FUECIS+WATCOS+WAWACO+AMANT1)
     1+FPL1*PERCEN(IW1,ID1,LP1)*8760.*UENER*1000.
      TOTOPE=TOTOPE+OPCOS
      IF(IPUNCH.EQ.1) WRITE(7,701) FUECOS,ANUCAP,TWL,OPCOS,BLDOWN,FPL1,
     1EI,IM
  701 FORMAT(2F10.0,F6.0,F10.0,F5.0,F9.4,F7.5,F8.0,I1)
  200 IF(IM.LT.2)GOTO475
      IF(I2.NE.1)GO TO 902
      TTTSAV=TTT
      KKSAVE=KK
      GO TO 902
  475 DO9233IJL=IIJ,ITDBMA,ITBD
      IF(IWRITE.EQ.1)WRITE(6,666)
  666 FORMAT(5X,'COOLING CANAL IS TOO LARGE TO OPEARTS')
      ID1=(IJL-ITWBI)/ITBD+1
      CFF=CF
      IF(LP.EQ.IPLI) CFF=1.0
      ENERLS=ENERLS+PLMAX*CFF*8760.*PERCEN(IW1,ID1,LP1)
      TOTLOS=TOTLOS+PLMAX*8760000.*UENER*PERCEN(IW1,ID1,LP1)*CFF
      TOTOPE=TOTOPE+PLMAX*8760000.*UENER*PERCEN(IW1,ID1,LP1)*CFF
      IM=0
      IF(IPUNCH.EQ.1)WRITE(7,702) IM
  702 FORMAT(I80)
 9233 CONTINUE
      GO TO 901
  902 IF(IWRITE.LT.1)GOTO923
      WRITE(6,601) PL,TWB
  601 FORMAT(1H0,5X,'POWER =',F5.0,' MW',10X,'TWB =',F8.3,' DEG. F')
      WRITE(6,333)TET,HOTWTT,COLDWT,P,TQ
333   FORMAT(/6X,'TURB.TEMP. = ',F10.4,1X,'DEG.F. ',
     15X,'HOT WATER TEMP. = ',F10.4,1X,'DEG.F.',
     25X,'COLD WATER TEMP. = ',F10.4,1X,'DEG.F.',
     3//,6X,'PRESSURE = ',F8.5,1X,'IN.HG.',
     45X,'HEAT REJECTION = ',F8.5,1X,'BTU*10**9',/)
      WRITE(6,602)
  602 FORMAT(1H0,1X,'TDB',3X,'WATER EVA.',3X,'BLOWDOWN',3X
     1,'PROBABILITY',3X,'FUEL COST',3X,
     2'WATER COST',3X,'WASTE WATER COST',3X,'SUBS ENERGY LOSS',3X,
     3'OPERATING COST'/3X,'F',
     47X,'GPM',9X,'GPM',22X,'$/YEAR',6X,
     5'$/YEAR',10X,'$/YEAR',13X,'$/YEAR',12X,'$/YEAR'/)
      WRITE(6,607)IIJ,TWL,BLDOWN,PERCEN(IW1,ID1,LP1),FUECIS,WATCOS,WAWAC
     1O,ANUCAP,OPCOS,FPL1
  607 FORMAT(1H ,I3,F15.5,F11.4,F13.6,F13.0,F12.1,F16.1,F20.3,F18.1,
     1F10.3)
      WRITE(6,899)
  899 FORMAT(///1X,130('*')////)
  923 CONTINUE
  901 CONTINUE
 1000 CONTINUE
      FUELEX=TOTEI-TEI
      TOTFU1=FUELEX*FC*1000.*8760.
      TOTOPE=TOTOPE-CWATEO-CBLOWO-CMAINO
      TOTOP=TOTOPE-TEI*FC*1000.*8760.
      TOTEI1=TOTEI/TOTPRO
      TOTWL1=TOTWL/(TQSTAR*1055.04/3.6)
      FUELE1=FUELEX/PLMAX
      FPLMA1=FPLMAX/PLMAX
      ENERL1=ENERLS/(PLMAX*8760.)
      FAPLS1=FAPLS/(PLMAX*8760.)
      WRITE(6,12121)
12121 FORMAT(//10X,'VALUES IN PARENTHESIS ARE,THE VALUES DIVIDED BY POWE
     1R OUTPUT EXCEPT THE LAST TWO WHICH ARE'/10X,'THE VALUES DIVIDED BY
     2 THE POWER OUTPUT PER YEAR'//)
      WRITE(6,605) TOTBLD,TOTWL,TOTWL1,TOTEI,FUELEX,FUELE1,
     1FPLMAX,FPLMA1,ENERLS,ENERL1,FAPLS,FAPLS1
     1,TOTFUE,TOTFU1,TOTWAT,TOTWAW,TOTMAN,ANUCAP,TOTOPE,TOTOP
  605 FORMAT(1H ,/10X,'TOTAL ANNUAL BLOWDOWN =',F15.0,' ACRE-FT/YEAR'/
     110X,'TOTAL ANNUAL WATER EVAP. =',F12.0,' ACRE-FT/YEAR'
     1,5X,'(',F10.5,' )'/10X,
     2'TOTAL ENERGY RATE IN =',F12.3,' MW'//8X,
     2'*** CAPABILITY LOSSES ***'/10X,
     3'EXCESS FUEL CONSUMPTION =',F9.3,' MW',5X,'(',F9.6,' )'/10X,
     3'MAXIMUM CAPABILITY LOSS =',F9.3,' MW',5X,'(',F9.6,' )'/10X,
     3'ENERGY LOSS =',F15.5,' MW-HR',5X,'(',F9.6,' )'/10X,
     3'PUMP ENERGY LOSS =',F11.4,' MW-HR',5X,'(',F9.6,' )'//8X,
     3'*** TOTAL ANNUAL COSTS ***'/10X,
     3'TOTAL ANNUAL FUEL COST =',F20.0,' $/YEAR'/10X,
```

```
     4'EXCESS FUEL COST =',F26.0,' $/YEAR'/10X,
     4'TOTAL ANNUAL WATER COST =',F19.0,' $/YEAR'/10X,
     5'TOTAL ANNUAL WASTE WATER COST =',F13.0,' $/YEAR'/10X,
     6'TOTAL ANNUAL MAINTENANCE COST =',F13.0,' $/YEAR'/10X,
     6'TOTAL ANNUAL CAPACITY LOSS =',F16.0,' $/YEAR'/10X,
     7'TOTAL ANNUAL OPERATING COST =',F15.0,' $/YEAR'/10X,
     8'EXTRA ANNUAL OPERATION COST =',F15.0,' $/YEAR')
      TOTFUE=TOTFUE/S
      TOTWAT=TOTWAT/S
      TOTWAW=TOTWAW/S
      TOTMAN=TOTMAN/S
      ANUCAP=ANUCAP/S
      TOTOP=TOTOP/S
      TOTOPE=TOTOPE/S
      WRITE(6,621) TOTFUE,TOTWAT,TOTWAW,TOTMAN,ANUCAP,TOTOPE
     1,TOTOP
  621 FORMAT(1H0,7X,'*** AVERAGE OPERATING COSTS --- IN MILLS/KW-HR ***'
     1/10X,'AVERAGE FUEL COST =',F22.6,' MILLS/KW-HR'/10X,
     1'AVERAGE WATER COST =',F21.6,' MILLS/KW-HR'/10X,
     2'AVERAGE WASTE WATER COST =',F15.6,' MILLS/KW-HR'/10X,
     4'AVERAGE MAINTENANCE COST =',F15.6,' MILLS/KW HR'/10X,
     5'AVERAGE CAPACITY LOSS =',F18.6,' MILLS/KW/HR'/10X,
     6'AVERAGE TOTAL OPERATING COST =',F11.6,' MILLS/KW-HR'/10X,
     7'AVERAGE EXTRA OPERATING COST =',F11.6,' MILLS/KW-HR')
      RETURN
 1002 WRITE(6,623)
 623  FORMAT(10X,'SPRAY COOLING SYSTEM IS TOO LARGE TO OPERATE')
      TOTCOS=10.**12
      RETURN
      END



      SUBROUTINE MODELW (TDB,TWB,IW,LP,TET,TQ,TWL,IIII,K,TT,IM)
C
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C *  THIS SUBROUTINE CALCULATES THE MODELING RELATIONSHIPS FOR POWER    *
C *   PLANT AND SPRAY CANAL .     GIVEN WET AND DRY BULB                *
C *   TEMPERATURES, OPERATION LEVELS OF SPRAY CANAL,POWER LEVEL OF      *
C *   TURBINE OUTPUT, THE RESULTS ARE TURBINE EXHAUST TEMPERATURE AND   *
C *   HEAT REJECTION.                                                   *
C * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
C
C
      DIMENSION HTRJIN(50,14),IENHTR(14),PSA(250)
      REAL NTU
      COMMON/TURBIN/HTRJIN,IENHTR,TLOW,FINC
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,QH1,QH2,HA
      COMMON/TURB/ BL1,BL2,BL3,TTD,TTDKO,TTDO
      COMMON/TBPR/ ITPMAX,TETMAX,ICAP
      COMMON/SPRAY/R,MK,N,WINDSP,FF,TFILM
      COMMON/SPRDIS/F(6,3),ROWDIS
      COMMON/PILIC/GPM
      COMMON/CONSTA/ CONST1,DONST,CONST
      FGPM=GPM
C
C    IF TWB IS HIGH ENOUGH, THEN COOLING CANNOT TAKE PLACE AT ALL UNTIL
C    TURBINE CONDENSER TEMPERATURE IS HIGHER. THUS, WILL SKIP TO
C    HIGHER TURBINE TEMPERATURE.
C
C    IF TWB IS LOW ENOUGH,  COOLING WATER FREEZES, WHICH
C    IS NEVER DESIRED.  THUS WHENEVER COOLING WATER WOULD HAVE BEEN
C    COOLED BELOW FREEZING ANYWHERE IN THE CYCLE, NC COOLING IS
C    PERFORMED (IMPLYING ALTERNATE SYSTEM USED IN PRACTICE).
C
C  ASSIGN MODEL PARAMETERS FOR TOWER SECTION
C
      ICAP=0.
      IM=2
      IFR=0
      IFRE=-1
      IF(IIII.GT.0.5)GOTO1000
      WL=0.
      TPS=0.
      TWBAL=0.
      QH2=0.
      IF(IIII.EQ.-5)GOTO991
      TTDO=HTRJIN(1,LP)*TTDKO
      GOTO992
  991 TTDO=TTD
C
C  ASSIGN INITIAL TRIAL TURBINE TEMPERATURE
```

```
C
  992 K=0
   99 IFRE=IFRE+1
  201 K=K+1
      TT=TLOW+(K-1)*FINC
      IF(K.GT.IENHTR(LP).AND.TT.GT.(TETMAX+1.95))GO TO 999
      IF(IIII.EQ.-5)GOTO404
      TTD=HTRJIN(K,LP)*TTDKO
  404 TT1=TT-TTD
C
C  COOL THROUGH COOLING SYSTEM IF POSSIBLE TO GET TQ1
C
      IF(TWB.LT.TT1)GOTO403
      GOTO201
  403 WL2=WL
      TPS2=TPS
      TWBAL2=TWBAL
      QH22=QH2
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      TWI=TT1
      CALL SPRCOL(TWI,TDB,TWB,NTU,B,GPM,AFR,RLG,TT2,WL)
C
      IF(TT2.GT.32)GOTO503
      GOTO99
C
C  DETERMINE DIRECTION OF APPROACH TO INTERSECTION OF CURVES
C
  503 TQ1=(TT1-TT2)*FGPM/CONST
      IF(TQ1.LT.HTRJIN(K,LP))GOTO100
C
C  REACH INTERSECTION BY DECREASING TURBINE TEMPERATURE
C
      IF(IFRE.GT.0.)GOTO206
      IF(TWB.GT.(TLOW-FINC-TTDO))GOTO104
C
C  COOLING CURVE ENDS MORE THAN 1 DECREMENT BELOW TLOW
C
  206 TT=TT-FINC
      IF(K.GT.1) K=K-1
      TT1=TT-TTDO
      IF(TT1.LT.32.)GOTO703
      IF(TWB.GT.TT1)GOTO304
C
C  COOLING THROUGH COOLING SYSTEM IF POSSIBLE TO GET TQ2
C
      WL2=WL
      TPS2=TPS
      TWBAL2=TWBAL
      QH22=QH2
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      TWI=TT1
      CALL SPRCOL(TWI,TDB,TWB,NTU,B,GPM,AFR,RLG,TT2,WL)
C
      IF(TT2.GT.32)GOTO505
      IFR=1
  505 TQ2=(TT1-TT2)*FGPM/CONST
      IF(TQ2.LT.HTRJIN(K,LP))GOTO105
      IF(IFR.EQ.1)GOTO703
      TQ1=TQ2
      GOTO206
C
C  INTERPOLATE FOR TQ,TET,TWL
C
  105 TQ=HTRJIN(K,LP)
      HTDIF1=(TQ-TQ2)/(TQ1-TQ2)
      TET=TT+HTDIF1*FINC
      IF(IIII.EQ.-5)GOTO405
      TTD=TQ*TTDKO
  405 TWL=(WL+HTDIF1*(WL2-WL))
      TPS=TPS+HTDIF1*(TPS2-TPS)
      TWBAL=TWBAL+HTDIF1*(TWBAL2-TWBAL)
      QH2=QH2+HTDIF1*(QH22-QH2)
      IF(IFR.EQ.0)GOTO210
C
C  (PREVIOUS CANAL COOLING INDICATES THAT THE OPERATING CHARACTERISTICS
C   CURVE FOR THE COOLING SYSTEM ENDS IN THE SAME TEMPERATURE INTERVAL
C   AS TET)
C  COOL THROUGH COOLING SYSTEM USING TET, TQ FOR CHECK
C
  211 TT1=TET-TTD
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
```

```
      TWI=TT1
      CALL SPRCOL(TWI,TDB,TWB,NTU,B,GPM,AFR,RLG,TT2,WL)
C
      IF(TT2.GT.32)GOTO210
      GOTO703
  210 CONTINUE
      IIII=1
      RETURN
  304 TT=TT+FINC
C
C  COOLING CURVES END JUST BELOW TT AND DECREMENTING TURBINE TEMPERATURE
C   WILL NOT INTERSECT IT
C
C  DETERMINE PROPER VALUE OF HTRJIN(1,LP)
C  DOUBLE INTERPOLATE FOR TQ,TET,TWL
C
  104 IF(K.GT.1)GOTO106
      HQ=HTRJIN(1,LP)
      GOTO107
  106 HQ=HTRJIN(K-1,LP)
  107 IF(IIII.GT.-5)GOTO407
      HQ=HQ+(TWB+TTD-TT+FINC)/FINC*(HTRJIN(K,LP)-HC)
      GOTO408
  407 HQ=(HQ+(TWB-TT+FINC)/FINC*(HTRJIN(K,LP)-HQ))/(1.-TTDKO/FINC
     1*(HTRJIN(K,LP)-HQ))
  408 IF(IIII.EQ.-5)GOTO406
      TTD=HQ*TTDKO
  406 TQ=TQ1*HQ/(TQ1+HQ-HTRJIN(K,LP))
      TET=TWB+TTD+(TT-TWB-TTD)*TQ/TQ1
      IF(IIII.EQ.-5)GOTO409
      TTD=TQ*TTDKO
  409 TWL=WL/TQ1*TQ
      TPS=TPS/TQ1*TQ
      TWBAL=TWBAL/TQ1*TQ
      QH2=QH2/TQ1*TQ
      TT=TT-FINC
      IF(K.GT.1)K=K-1
      GOTO211
C
C  REACH INTERSECTION BY INCREMENTING TURBINE TEMPERATURE
C
  100 IF(K.EQ.IENHTR(LP))GOTO999
  108 TT=TT+FINC
      K=K+1
      IF(IIII.EQ.-5)GOTO410
      TTD=HTRJIN(K,LP)*TTDKO
  410 TT1=TT-TTD
C
C  COOL THROUGH SYSTEM TO GET TQ2
C
      WL2=WL
      TPS2=TPS
      TWBAL2=TWBAL
      QH22=QH2
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      TWI=TT1
      CALL SPRCOL(TWI,TDB,TWB,NTU,B,GPM,AFR,RLG,TT2,WL)
C
      TQ2=(TT1-TT2)*FGPM/CONST
      IF(TQ2.GT.HTRJIN(K,LP))GOTO101
      IF(K.EQ.IENHTR(LP))GOTO999
      TQ1=TQ2
      GOTO108
C
C  INTERPOLATE FOR TQ, TET, TWL
C
  101 HTDIF1=HTRJIN(K,LP)-HTRJIN(K-1,LP)
      HTDIF2=TQ2-TQ1
      TQ=(HTRJIN(K,LP)*HTDIF2-TQ2*HTDIF1)/(HTDIF2-HTDIF1)
      TET=TT-(TQ2-TQ)/HTDIF2*FINC
      IF(IIII.EQ.-5)GOTO411
      TTD=TQ*TTDKO
  411 TWL=(WL2+(WL-WL2)/HTDIF2*(TQ-TQ1))
      TPS=TPS2+(TPS-TPS2)/HTDIF2*(TQ-TQ1)
      TWBAL=TWBAL2+(TWBAL-TWBAL2)/HTDIF2*(TQ-TQ1)
      QH2=QH22+(QH2-QH22)/HTDIF2*(TQ-TQ1)
      TT=TT-FINC
      K=K-1
      IIII=1
      RETURN
C
C  RETURN WITH MESSAGE
C
```

```
  703 TQ=-50
      TET=-50.
      TWL=-50.
      IIII=0
      IM=-50
      RETURN
C
C  FIND INTERSECTION WHEN WET-BULB TEMPERATURE INCREASES
C
C  REACH INTERSECTION BY INCREMENTING TURBINE TEMPERATURE
C
 1000 TTD=HTRJIN(K,LP)*TTDKO
      TT1=TT-TTD
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      TWI=TT1
      CALL SPRCOL(TWI,TDB,TWB,NTU,B,GPM,AFR,RLG,TT2,WL)
C
      TQ1=(TT1-TT2)*FGPM/CONST
 1004 TT=TT+FINC
      IF(TT.LT.(TLOW+0.001))GCTO1001
      IF(K.EQ.IENHTR(LP))GOTO999
      K=K+1
      HQ=HTRJIN(K,LP)
      HTDIF1=HQ-HTRJIN(K-1,LP)
      GOTO2070
 1001 HQ=HTRJIN(1,LP)
      HTDIF1=0.
 2070 TTD=HQ*TTDKO
      TT1=TT-TTD
      WL2=WL
      TPS2=TPS
      TWBAL2=TWBAL
      QH22=QH2
      IF(TT.GT.(TETMAX+1.95))GOTO999
C
      TWI=TT1
      CALL SPRCOL(TWI,TDB,TWB,NTU,B,GPM,AFR,RLG,TT2,WL)
C
      TQ2=(TT1-TT2)*FGPM/CONST
      IF(TQ2.GT.HQ)GOTO1010
      TQ1=TQ2
      GOTO1004
C
C  INTERPOLATE FOR TQ, TET, TWL
C
 1010 IF(K.GT.1)GOTO412
      TTD=HTRJIN(1,LP)*TTDKO
      GOTO413
  412 TTD=HTRJIN(K-1,LP)*TTDKO
  413 IF(TWB.LT.(TT-FINC-TTD))GOTO1011
      TQ1=TQ2
      GOTO104
 1011 HTDIF2=TQ2-TQ1
      TQ=(HQ*HTDIF2-TQ2*HTDIF1)/(HTDIF2-HTDIF1)
      TET=TT-(TQ2-TQ)/HTDIF2*FINC
      TTD=TQ*TTDKO
      TWL=(WL2+(WL-WL2)/HTDIF2*(TQ-TQ1))
      TPS=TPS2+(TPS-TPS2)/HTDIF2*(TQ-TQ1)
      TWBAL=TWBAL2+(TWBAL-TWBAL2)/HTDIF2*(TQ-TQ1)
      QH2=QH22+(QH2-QH22)/HTDIF2*(TQ-TQ1)
      IF(K.GT.1) K=K-1
      TT=TT-FINC
      RETURN
999   IF(TT2.LT.32.)GOTO703
      ICAP=1
      TT1=TETMAX-TTD
C
      TWI=TT1
      CALL SPRCOL(TWI,TDB,TWB,NTU,B,GPM,AFR,RLG,TT2,WL)
C
      TQ=(TT1-TT2)*FGPM/CONST
      TET=TETMAX
      IF(IIII.NE.-5)TTD=TQ*TTDKO
      TWL=WL
      RETURN
      END



      SUBROUTINE SPRCOL(TWI,TDB,TWB,NTU,B,GPM,AFR,RLG,TT2,WL)
C* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
```

```
C
C     THIS SUBROUTINE CALCULATES THE COOLING BY A SPRAY CANAL
C
C* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
      DIMENSION PSA(250)
      REAL NTU
      COMMON/NCALA/ PSA,TPS,TS,TWBAL,QH1,QH2,HA
      COMMON/SPRAY/R,MK,N,WINDSP,FF,TFILM
      COMMON/SPRDIS/F(6,3),ROWDIS
      COMMON/ATMOS/ PATM
C
      IF(ROWDIS.LT.40.OR.ROWDIS.GT.60) FF=0.18
      KLM=IFIX((ROWDIS-40)/10+1)
      M=MK
      FF=F(M,KLM)
      CW=1.00
      NTU=0.036*WINDSP+0.156
      TWBA=TWB+FF*(TWI-TWB)
      HV=1087.-0.54*TWI
      IFLAG=1
      TFILM=0.5*(TWI+TWBA)
      TWFI=TFILM-0.5
111   TWF=TWFI
      IT=TWF
      PVSAT=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TWF-IT)
      PV=PVSAT-0.000367*PATM*(TDB-TWF)*(1.+((TWF-32.)/1571))
      S=PV/(1.608*(PATM-PV))
      SIGMA=(0.24+0.44*S)*TDB+S*(1093.8-TWF)
      IF(IFLAG.EQ.2) GO TO 121
      IFLAG=IFLAG+1
      TWFI=TWFI+1.
      SAVSIG=SIGMA
      GO TO 111
121   ARGUM=SIGMA-SAVSIG
      B=ABS(ARGUM)
      X=NTU*B
      TT21=TWI-(TWI-TWBA)*(1.-EXP(-X))
      NN=M*N
      TT2=TWB+(TWI-TWB)*EXP((-NN*R*(1.-FF))*(1.-EXP(-X)))
C*****  BOWEN RATIO IS ASSUMED  = 0  *****
      WL = (CW/HV)*(TWI-TT2)*GPM
      RETURN
      END

      SUBROUTINE POWERS (TEM,Q,TTSTAR)
```

See Appendix III for listing.

APPENDIX VII

RANGE OF VALUES OF VARIOUS ECONOMIC AND OTHER PARAMETERS

General

Unit cost of replacement capacity,	c_L = \$90-200/kW	[2]
Unit cost of "short term" replacement energy during downtime,	e_ℓ' = \$0.007/kW-hr	[2]
Unit cost of "long term" replacement energy, after backfitting	e_ℓ = \$0.01/kW-hr	[2]
Unit cost of fuel,	f_c = \$0.30-0.98/10^6Btu	[6]
Unit cost of water,	c_w = \$0.0-1.0/1000 gal (highly variable)	
Unit cost of land,	a_ℓ = \$500-5000/acre (highly variable)	
Unit cost of new condenser,	c_c = \$6.50-23.10/ft^2	[*]
	= \$4.00/ft^2	[16]
Unit cost of blowdown treatment	c_b = \$0.0-0.50/1000 gal (highly variable)	
Open-cycle maintenance cost	M' = 0	
Open-cycle blowdown cost	B' = 0	
Open-cycle water cost,	W' = 0	
Downtime for hook-up and testing	DT = 5-10 days	[**]
	= 30-90 days	[6]

Mechanical-Draft Cooling Towers

Unit cost of towers,	c_t = \$7.50/TU	[16]
Unit cost of maintenance,	c_m = \$200/cell/year	[2]

---

* Comment by J.P. Chasse (E.P.A.)

** Private communication, Commonwealth Edison & Iowa-Illinois, Quad Cities Nuclear Power Plant

Cost of pump and pipe system	C_{pp} = Figure 13	[16]

Natural-Draft Cooling Towers

Cost of towers,	C_{cs} = Figure 32	[16]
Maintenance cost,	C_{m} = \$1000-3000/tower/year	
Cost of pump and pipe system	C_{pp} = Figure 13	[16]

Cooling Ponds

Unit cost of ponds, (including land cost)	C_{cs} = \$500-5000/acre	[26]
Unit pump and pipe system cost,	c_{pp} = \$1.50/gpm	[26]
Unit maintenance cost	c_{m} = \$2.00/acre/year	[26]

Spray Canals

Unit cost of spray modules,	c_{s} = \$16,000-26,250 /module	[49,56,57]
Pump and pipe system cost,	c_{pp} = \$1.50/gpm	[26]
Maintenance cost,	c_{m} = 1% of pump and module operating cost	
Lining cost of canal,	c_{L} = \$0.93/ft^{2}	[*]
Excavation of canal,	c_{E} = \$2.50/yd^{3}	[*]

* Private communicaton with local industry

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16. ABSTRACT The report gives in detail a method for assessing the economic consequences of backfitting electric power plants (currently operating on open-cycle or once-through cooling systems) with conventional closed-cycle cooling systems. Four types of closed-cycle systems were investigated: mechanical- and natural-draft crossflow wet cooling towers, cooling ponds, and spray canals. To estimate operational penalties associated with backfitting, thermodynamic models were used to reproduce the operating characteristics of different types of turbines, condensers, and cooling systems. Capital and operating cost information was compiled and used, in conjunction with the levelized annual cost accounting method, to evaluate the total differential cost of power production resulting from the backfit. Computer programs were developed and are presented. Many representative calculations were performed and are presented graphically. The results for three types of conventional turbines and four geographical sites were obtained for a range of cooling system sizes: they are plotted visually. Once the various unit costs of replacement capacity, energy loss, fuel, and water are known, these results can be used to evaluate the cost to be assessed against backfitting. Representative unit cost values are included in the report.

17. KEY WORDS AND DOCUMENT ANALYSIS

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