EPA-660/2-73-004
September 1973

Environmental Protection Technology Series

Nomographs For Thermal Pollution Control Systems

Office of Research and Development
U.S. Environmental Protection Agency
Washington, D.C. 20460

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NOMOGRAPHS FOR THERMAL POLLUTION CONTROL SYSTEMS

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ABSTRACT

Nomographs are presented and described which permit the estimation of heat rejection system performance, tower or pond capital costs and the perturbations to power plant efficiency and costs which result from the incorporation and operation of any one of the following thermal pollution control systems within a power plant as a substitute for once-through cooling:

- natural draft wet towers
- mechanical draft wet towers
- spray ponds
- cooling ponds
- natural and mechanical draft dry towers

The base case plant for cost comparisons is chosen as having a nominal turbine back pressure of 2 in. Hg absolute. The total heat rejection system with its associated costs is defined to extend outward from the turbine exhaust flange, a common boundary for each of the systems mentioned above.

Performance and capital costs for the thermal pollution control systems were compared with data from existing facilities and theoretical estimates from various sources.

This report was submitted in fulfillment of Contract No. 68-01-0171 by Hittman Associates, Inc. under the sponsorship of the Environmental Protection Agency. The principal author is Mr. Charles Jedlicka, Hittman Associates, Inc., Columbia, Maryland 21045.

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SECTION I

CONCLUSIONS

The nomographs developed on this project and presented in this report provide for the estimation of heat rejection system performance, tower or pond capital costs and the perturbations to power plant efficiency and costs due to the presence of a thermal pollution control system at a power plant site. The nomographs are intended primarily for the use of regulatory organizations, such as the Environmental Protection Agency (EPA) for making rapid checks on design cost data generated by contractors or utilities. They are also intended to provide engineers and managers concerned with the performance and costs of proposed cooling systems with a tool for identification of critical factors and for rapid estimation of their impact on operating efficiencies and costs, capital investment requirements, and the site environment. The nomographs are not intended to be used for design optimization purposes.

As a result of developing the nomographs, specific comments can be made regarding the performance and capital costs of pertinent thermal pollution control systems. The performance nomograph for natural draft wet towers is satisfactory for the general application of making performance estimates following comparisons with several sources. For mechanical draft wet towers, the capital costs predicted by the nomographs show good correlation with those from other sources. Due to limits on availabile data the scope of the spray pond nomographs is confined to conventional spray ponds only, excluding powered spray modules (PSM). The cooling pond performance nomographs (open and closed cycle operation) provide reasonable estimates of cooling pond size based on comparison of results from various other sources.

It was assumed in the nomographs that at a given plant site, the rated output would be required under all site conditions. In so doing, it is assumed that the generator can take a certain overload steam rate. However, for existing plants the overload ability may be limiting for certain thermal pollution control systems. The nomographs are readily applicable to new plants where the additional heat capacity required can be made available.

The turbine performance nomographs are based on a conventional steam turbine generator modified for operation at higher back pressure. A new plant employing dry towers will be designed to utilize a high back pressure turbine, when they become commercially available, whose performance and costs are different from those of conventional units. However, the nomographs are satisfactory in treating the utilization of dry towers in new plants. The use of dry towers on existing plants will almost always be economically unfeasible.

The nomographs yield performance, water requirements and costs for heat rejection systems operating under design meteorological conditions and full load plant operation. Examples are presented to show how average annual water requirements from evaporation and average annual operating costs can be estimated.

It is important to evaluate water requirements and the annual operating costs under the off-design conditions actually occurring during the year at a given site.

SECTION II

RECOMMENDATIONS

Cascaded system designs considering combinations of cooling systems such as evaporative towers and dry towers, designated wet/dry cooling towers, could feasibly be nomographed. New spray cooling systems have been developed such as the powered spray systems (Ceramic, Rockford, Ashbrook) and the Cherne fixed thermal rotor system. Therefore, it is recommended that the scope of the nomographs be enlarged to include these and other new systems.

A technical handbook or data book could be prepared for thermal pollution control systems. It would include background, detailed descriptions, performance and equations based on the current sources for these systems. The newest technology on powered spray systems and dry towers would be included. Actual operating data would be included for existing thermal pollution control systems from sources such as the power plant and Federal Power Commission Form 67. It is recommended that this handbook be prepared.

There are more recent studies available that consider the application of dry type towers in new plants. This is also true for the other thermal pollution control systems. Thus, it is recommended that the nomographs be updated periodically in order to incorporate new data on performance and costs.

SECTION III

METHODOLOGY OF NOMOGRAPH PREPARATION

Introduction

The selection of a type of heat rejection system, or a system for thermal pollution control, for use at an electric utility site depends upon the comparisons of many parameters. The purpose of the nomographs developed and presented in this report is to take input data such as site weather data, power plant data, and economic conditions in the locale and allow one to estimate significant output parameters. These are heat rejection system performance, cooling tower or pond capital costs, and the perturbations to power plant efficiency and costs. Thus, the key factors and parameters from the various alternative systems can be quantified and analysed at a given utility plant site following application of the nomographs.

In the next section, the general procedure to be followed when using the nomographs is schematically presented. In succeeding major sections, each type of thermal pollution control system is described. The nomographs for each system are discussed. The development of the turbine performance, water requirements, and cost nomographs is discussed. In general, output parameters were determined from the given input parameters based on either direct functional relationships or from available analytical or experimental data.

Section XII contains illustrative examples for each of the thermal pollution control systems. All of the nomographs are then presented together in the Appendix. They are easily referred to in conjunction with the instructive sample cases preceeding them.

Nomograph Utilization Flow

Utilization of the nomographs for thermal pollution control systems follows a flow presented schematically in Figure 1. The nomographs

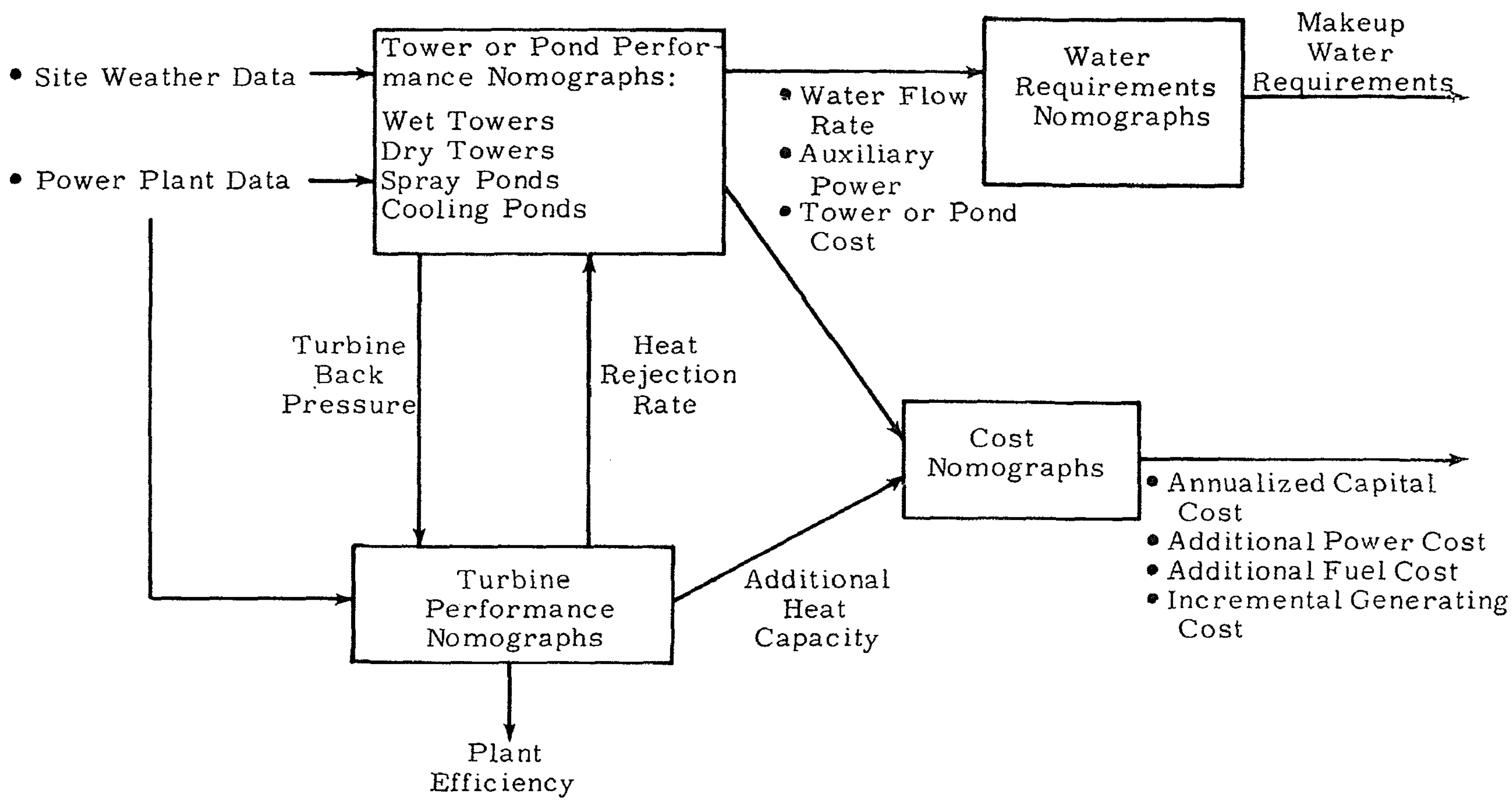


Figure 1. Nomograph Utilization Flow

can be organized into the four basic classifications enclosed in the boxes:

Tower or pond performance nomographs

Turbine performance nomographs

Water requirements nomographs

Cost nomographs

The general procedure one follows to obtain output parameters is explained in the following paragraphs with the aid of Figure 1.

Performance nomographs were developed for the following heat rejection systems:

Natural draft wet towers - closed cycle operation

Mechanical draft wet towers - closed and open cycle operation

Spray ponds - closed and open cycle operation

Cooling ponds - closed and open cycle operation

Mechanical draft and natural draft dry towers - closed cycle operation

Tower or pond performance is dependent upon the design weather conditions chosen at the site such as wet bulb temperature, dry bulb temperature, and relative humidity. Steam turbine back pressure is then determined from the resulting cold water temperature, the range and the terminal temperature difference (see Section XV for nomenclature).

The turbine performance nomographs take as input the turbine back pressure and power plant data such as net generating capacity or size, type (fossil or nuclear), and base plant heat rate. Light water reactor plants have plant efficiencies ranging from approximately 31 to 33 percent while high temperature gas-cooled reactor plants have plant efficiencies similar to those of fossil-fueled plants. Thus, in the subsequent discussions, the term nuclear will mean light water reactor plants. For the purposes of heat dissipation, HTGR plants are similar to fossil-fueled plants. The heat rejection rate for the cooling system is determined.

The presence of the cooling tower or pond necessitates the requirement of an additional heat capacity in order to maintain the rated plant output. This imposes a cost penalty determined later in the cost nomographs. The presence of the thermal pollution control system usually results in an increase of turbine back pressure and thus a lower plant efficiency for the power plant when compared to the base plant operating at 2 in. Hg absolute turbine back pressure.

The tower or pond performance nomographs are returned to, with the heat rejection rate. The water flow rate, auxiliary power requirements, and the tower or pond capital cost are determined. All costs correspond to the year 1971 (ENR Construction Cost Index 1570).

The water requirements nomographs estimate the makeup water quantity required for each type of heat rejection system. The components of makeup water are attributed to drift, evaporative, and blowdown losses.

In the cost nomographs, "incremental costs" are determined as being the various additional capital and operating costs required to incorporate and operate the thermal pollution control system within the power plant. The base case plant for cost comparisons is chosen as having a nominal turbine back pressure of 2 in. Hg absolute. Thus, additional heat capacity would be required for operation at turbine back pressures above this level. It was assumed that the turbine generator can take a certain overload steam rate. In computing the cost of the loss of efficiency from operation above 2 in. Hg absolute, one needs to distinguish between added fuel costs and capability losses. In the nomographs, added fuel costs are computed as a result of the operation of the plant above the base level of 2 in. Hg absolute. In contrast, capability losses and their associated cost penalty are considered only when the turbine back pressure exceeds some maximum value such as 3.5 in. Hg absolute. Also capability loss can be made up with base load (assumed in the nomographs) or peaking units. If it is made up with base load, additional revenue at turbine back pressures less than 3.5 in. Hg absolute should be assessed since it reduces the average annual cost of the capability loss. Thus an accurate assessment of all of the various "incremental costs" is a complex problem which exceeds the scope of the nomographs.

The total heat rejection system with its associated costs is defined to extend from the turbine exhaust flange, a common boundary for all of the thermal pollution control systems. Incremental plant costs are determined separately in the nomographs for the steam supply (either fossil or nuclear plants) and for the turbine-generator plant. Installed capital costs for condensers of different tube materials are determined. The cost of the water circulation system is estimated including pumps, motors, piping, and installation. After summing these capital cost components with the tower or pond cost, the total incremental capital cost is obtained.

The fixed charge rate depends upon the economic conditions at the site locale such as interest rate, amortization period, interim replacement, insurance and various taxes. Then the annualized capital cost is determined from a nomograph using the fixed charge rate and the total incremental capital cost.

Additional power cost and additional fuel cost are two "annual" operating costs determined in the cost nomographs. However, they are based only on design conditions and are thus not true annual costs. Operation under off-design conditions must be evaluated in order to determine true annual costs. These two "annual" costs are summed with the annualized capital cost yielding the total incremental "annual" cost. Finally, the incremental generating cost in mills/Kw-hr is obtained.

Each type of heat rejection system utilizes the turbine performance nomographs and the water requirements nomographs which are common to all of the systems. Also, the output of all tower or pond performance nomographs feeds into the set of cost nomographs which are common to all systems.

SECTION IV

NATURAL DRAFT WET TOWERS

System Description

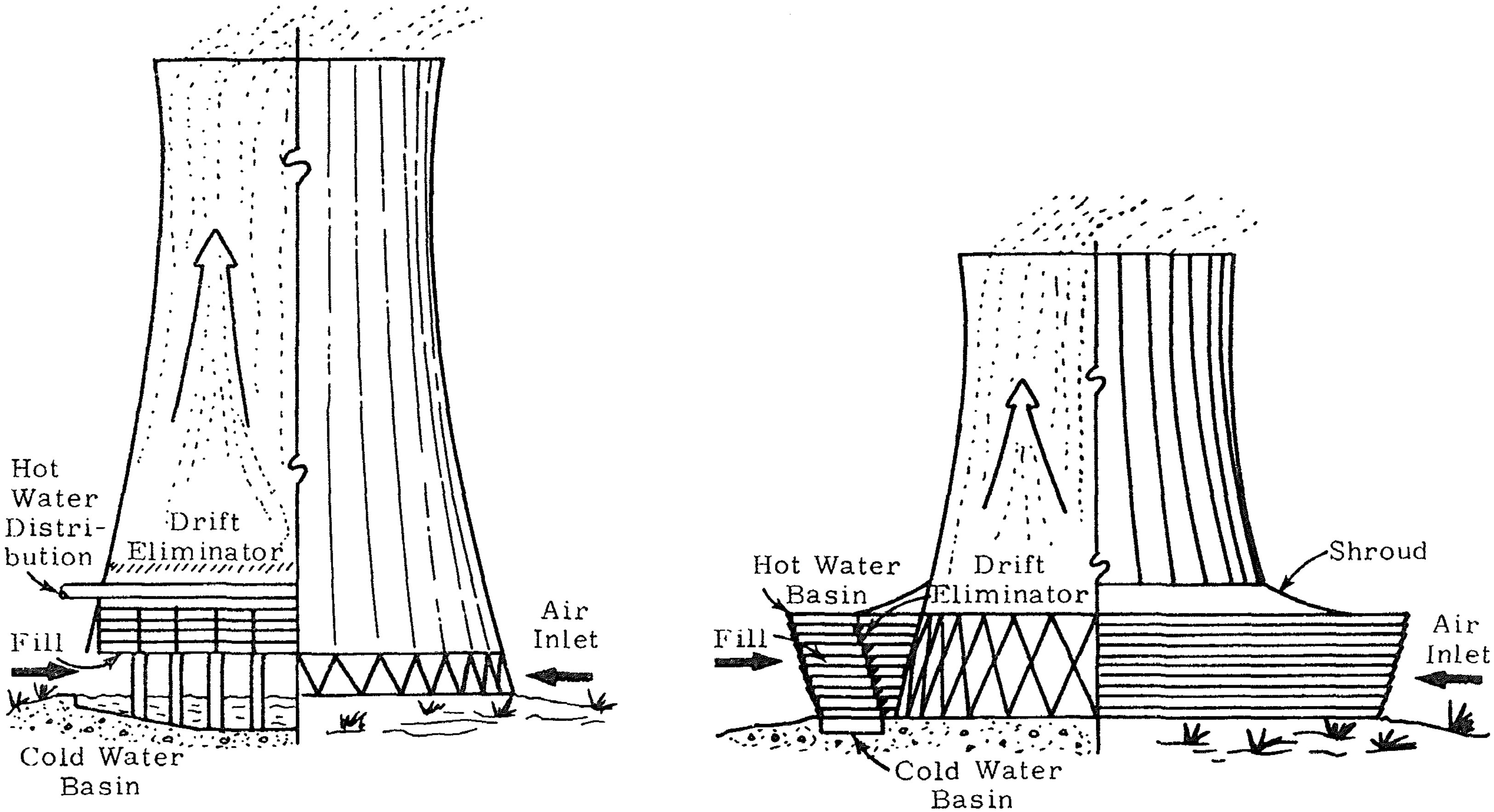
The first natural draft wet tower in the United States was built in late 1962 at the Big Sandy Plant of the Kentucky Power Company. The tower is 245 feet in diameter and 320 feet high. Currently natural draft wet towers range in size from 250 feet to 400 feet in diameter with heights from 320 to nearly 500 feet. They are basically a large chimney that provides a draft to pull air over a large surface of water.

Natural draft towers are usually constructed from reinforced concrete. The hyperbolic shape distributed across the great height is optimum for aerodynamic and structural reasons rather than thermodynamic. The basic components of counterflow and crossflow tower types are shown in Figure 2. The tower and packing can be designed to operate with air flowing upward through the packing (counterflow) or horizontally across the packing (crossflow).

Among the advantages of natural draft wet towers are long term maintenance free operation, smaller amounts of ground space required for multiple towers, reduced piping costs when towers can be located adjacent to plant, no electricity required for operating fans, fewer electrical controls and less mechanical equipment. Disadvantages include a decreased ability to design as precisely as for mechanical draft towers, and inability to control outlet temperatures as well as with mechanical draft towers. Also, because of their large size natural draft towers tend to dominate the landscape.

Discussion of the Nomographs

In the Appendix, the nomographs for natural draft wet towers are found on pages N-1 through N-6. The development of these nomographs and their underlying assumptions are discussed in the following paragraphs.



Counterflow

Crossflow

Figure 2. Types of Natural Draft Wet Towers

The Tower Performance nomograph, page N-1, is based on a parametric study performed by Research-Cottrell(Ref. 1). Cold water temperature (CWT) is obtained as a function of dry bulb temperature (DBT), relative humidity and variations in tower height. A design condition assumed to be fixed in the performance study was a base diameter of 300 feet.

The performance data from Research-Cottrell were compared to data from other sources in order to determine the degree of general applicability of the performance nomograph. In Figure 3, the performance data for the base diameter of 300 feet and a height of 350 feet is compared to the performance of the Keystone plant towers (Ref. 2) of 247 feet in diameter and 325 feet high. At 100 percent relative humidity, the curves compare very closely. The curves for 50 percent relative humidity are close at lower dry bulb temperatures and then diverge from one another at the higher temperatures. The curves at 20 percent relative humidity deviate by 1.5°F at 50°F dry bulb and by 3°F at 80°F dry bulb.

Data from Research-Cottrell for the base diameter of 300 feet and a height of 400 feet are compared in Figure 4 to the corresponding data reported by the Pacific Northwest Water Laboratory (PNWL) in Reference 3. Generally, a comparison between two curves of equal relative humidity shows a single intersection and then divergence from one another. In the dry bulb temperature range between 70°F and 90°F the comparison between curves of the same relative humidity is very good.

A comparison of nomograph performance predictions to the design conditions at existing plants is presented in Table 1. An error is defined for comparison purposes to be:

$$\text{error} = \left\{\text{CWT value determined from N-1}\right\} - \left\{\text{design approach + design wet bulb temperature}\right\} \qquad (1)$$

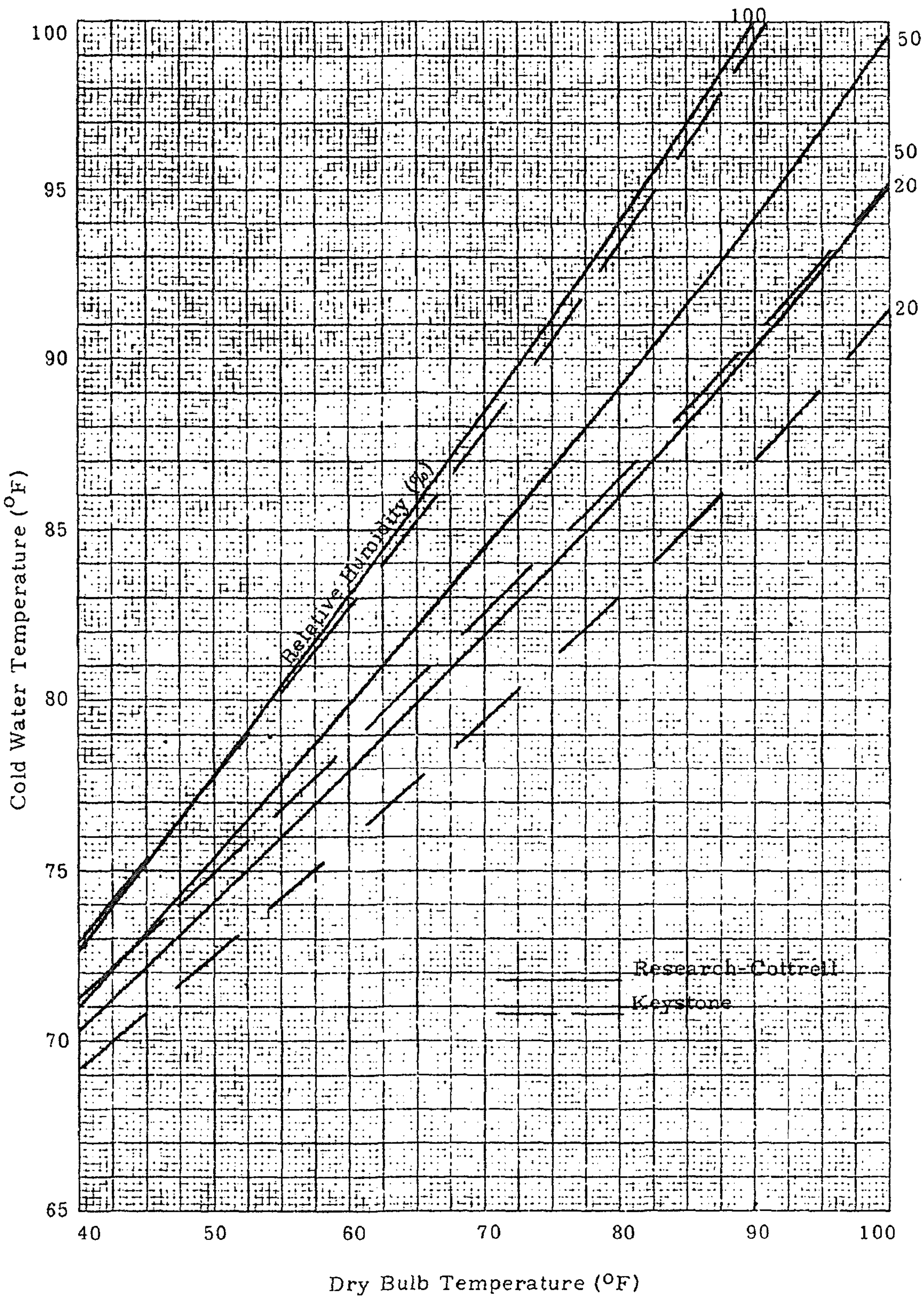


Figure 3. Comparison of Tower Performance
(Research-Cottrell vs. Keystone)

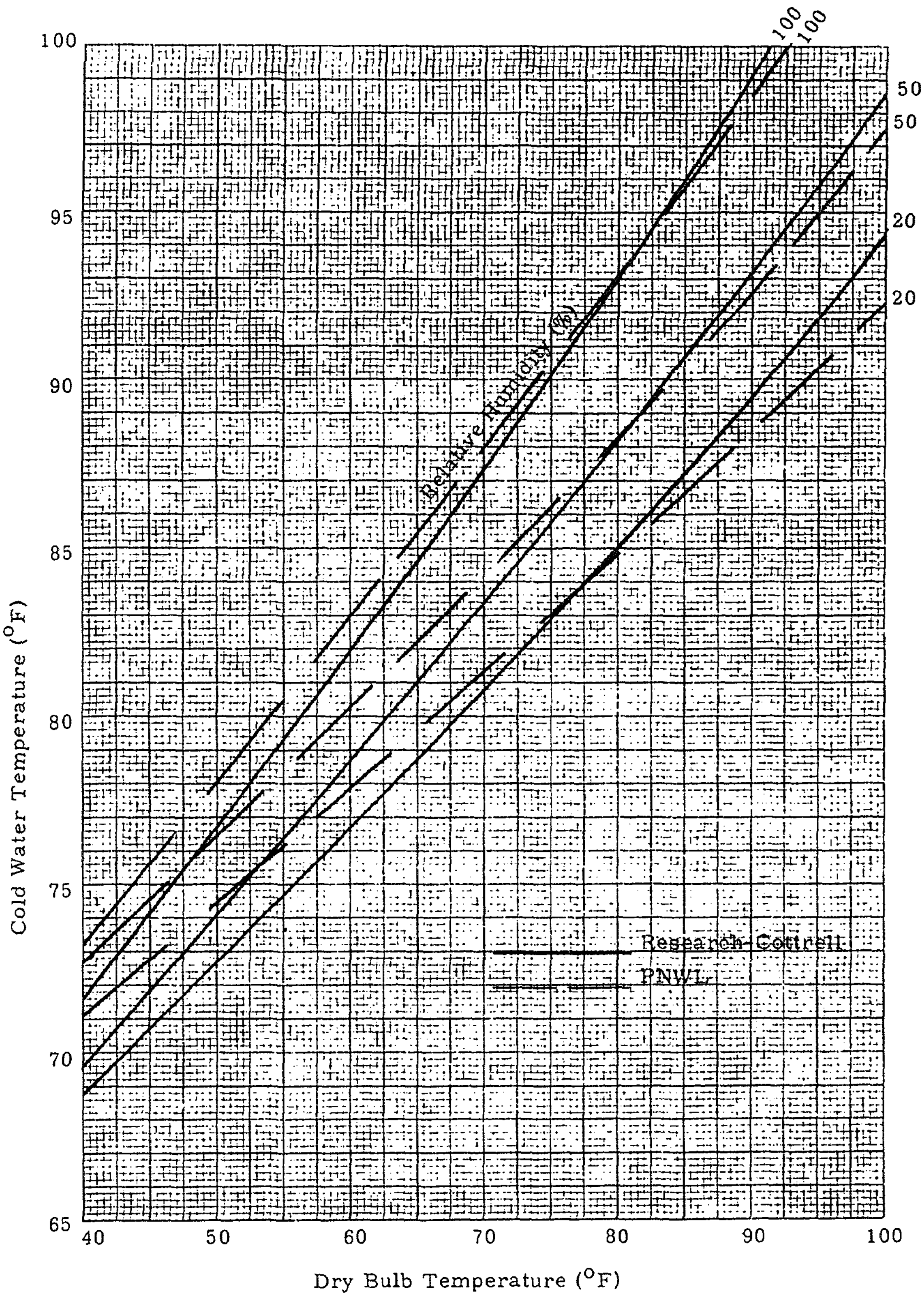


Figure 4. Comparison of Tower Performance (Research-Cottrell vs PNWL)

TABLE 1. COMPARISON OF NOMOGRAPH PERFORMANCE TO EXISTING PLANT DESIGN CONDITIONS

Plant	Tower Height Ft	Tower Base Diameter Ft	Total Flow Rate 10^6 gpm	Tower Design Basis: Range °F	Approach °F	WBT °F	DBT °F	Relative Humidity %	WBT + Approach °F	CWT From N-1 °F	Error °F
Big Sandy No. 1	322	244	0.12	23	15	72	79.5	70	87	91.5	4.5
Keystone Nos. 1 & 2	325	247	0.56	28	18	72	86.5	50	90	92	2
Ft. Martin Nos. 1 & 2	371	377	0.50	24	18	72	86.5	50	90	91	1
Paradise Nos. 1, 2, & 3	436	320	0.26	27.5	22.6	72.6	78	78	95.2	92	-3.2
Muskingum River	371	394	0.22	24	16	70	77	71	86	88	2
Big Sandy No. 2	371	397	0.25	28.7	16	70	77	71	86	88.5	2.5
Conemaugh No. 1	371	288	0.28	28	18	72	86.5	50	90	92.3	2.3

The Tower Performance nomograph predicts the design performance from these plants to within from 1°F to 4.5°F. In conclusion, as a result of the various comparisons presented, the Performance Nomograph is assumed to be satisfactory for the general application of making performance estimates. However, it is realized that a unique set of performance curves exists for towers at each plant site.

The Turbine Back Pressure nomograph, page N-2, allows one to determine the steam turbine back pressure through the following relationships:

turbine back pressure, in. Hg absolute = f (steam condensing temperature or saturation temperature) (2)

steam saturation temperature = cold water temperature + condenser rise + terminal temperature difference (TTD) (3)

The cold water temperature is an input obtained from the first nomograph. The range is the temperature decrease in the tower. The range of the natural draft tower is equal to the condenser rise because only closed cycle operation is assumed for the system. Ranges from 24°F to 32°F cover those for existing towers. The terminal temperature difference (TTD) is assumed constant at 6°F throughout. It can range from 5°F to 10°F. Reference 4 indicates that TTD must exceed 5°F at design conditions. This is a minimum level established by the Heat Exchange Institute. Steam tables were then employed to obtain the absolute turbine back pressure corresponding to a steam saturation temperature.

The Tower Unit Cost nomograph, page N-3, is based on the cost performance curves for budget estimates presented by the Marley Company (Ref. 4). These data were reduced so that the nomograph yields the unit tower cost in dollars per thousand Btu/hr versus wet bulb temperature (WBT), range and approach. The relative humidity was eliminated as a variable since unit tower costs at 50 percent relative humidity best represent the design conditions from existing towers. Reference 4 also presented data for 25 percent and 100 percent relative humidities.

The unit costs from Reference 4 were corrected prior to plotting in order to represent 1971 costs. An annual factor based on the Engineering News Record was employed. The input wet bulb temperature is determined from a psychrometric chart and corresponds to the dry bulb temperature and relative humidity chosen as entry design conditions to N-1. The approach parameter is the difference between the cold water temperature determined from N-1 and the wet bulb temperature.

The Tower Unit Cost nomograph is based on cost data which will be shown to estimate capital costs for several existing towers better than a cost equation presented in Reference 5. This equation is based on the calculation of capital costs for British Towers:

$$\text{tower capital cost} = \$3.4 \times 10^5 \times (\text{height of tower in ft times diameter of tower in ft})^{.17} \tag{4}$$

The capital cost comparison for towers at four existing plant sites is presented in Table 2. The design conditions are shown. The installed cost for each plant in the year of installation was determined from FPC Form 67 (Ref. 6). The installed cost excludes only the cost of the condenser. Each cost was then corrected to 1971 using factors from the Engineering News Record.

From the design conditions for each tower, the capital cost was determined from the Marley cost performance curves. Capital costs from equation (4) were computed. Before a direct comparison could be made between the Marley and Reference 5 costs and those of Form 67, an incremental cost for the water circulation system was added to the former. This amounted to $\$0.23 \times 10^6$ for the Keystone Tower and $\$0.41 \times 10^6$ for the other towers. Water circulation system cost includes capital costs for pumps, motors, piping and installation. In all cases, excluding Conemaugh, the costs based on the Marley data were closer to those obtained from Form 67. It was observed, in passing, that for Big Sandy No. 2 the tower unit cost at 50 percent relative humidity was only 7 percent higher than the one at the design condition of 71 percent relative humidity.

TABLE 2. COMPARISON OF CAPITAL COSTS FOR EXISTING NATURAL DRAFT TOWERS

Plant	Tower Design Basis				Form 67 Installed Cost Corrected to 1971 - $\$10^6$	1971 Capital Cost Based on Marley Data - $\$10^6$	1971 Capital Cost From Ref. 5 Eqn. - $\$10^6$
	WBT °F	Percent RH	Range °F	Approach °F			
Keystone No. 2a	72	50	28	18	4.04	2.98	2,87
Ft. Martin No. 2	72	50	24	18	6.88	5.13	3.33
Big Sandy No. 2	70	71	28.7	16	5.47	5.96	3.22
Conemaugh No. 1	72	50	28	18	3.64	5.87	3.37

The Tower Cost nomograph, page N-4, is used to obtain the natural draft tower cost in millions of dollars. This capital cost is obtained from the multiplication of the heat rejection rate by the tower unit cost. The heat rejection rate is an input from the turbine performance nomographs. The tower unit cost is the output of N-3.

The Flow Rate nomograph, page N-5, yields the water flow rate in millions of gpm. The following equation is employed:

$$\text{water flow rate, gpm} = \frac{\text{heat rejection rate, Btu/hr}}{\text{condenser rise, } {}^{\circ}\text{F} \times 500} \qquad (5)$$

In this equation, the specific heat of water is 1 Btu/lb°F, and 500 is a conversion factor. Heat rejection rate is an input from the turbine performance nomographs. Under closed cycle operation the condenser rise is equal to the range of the natural draft tower. A detailed explanation of the effects of closed cycle operation versus open cycle operation is given in Section V.

The Auxiliary Power nomograph, page N-6, yields the auxiliary power requirements in Mw. The power requirements for the natural draft tower are due to circulating water pumps only. The pump power requirements are obtained from the following equation:

$$\text{pump power, Mw} = \frac{\text{water flow rate, gpm} \times \text{total head, ft}}{\text{pump efficiency}} \times 1.88 \times 10^{-7} \qquad (6)$$

The water flow rate is an input from the previous nomograph. The pump efficiency is assumed to be 0.85. Total head consists of pumping head plus head losses such as friction. The range of total head shown in the nomograph is justified from three sources. In Reference 1, a data sheet summarizing potential natural draft tower designs at one large plant site indicated a range of pump heads from 40 to 55 ft. At the Keystone Plant (Ref. 2), water enters the natural draft towers approximately 42 ft above the basin. A total head of 75 ft was assumed in the cost cases analysed in Reference 4 considering usage of natural draft towers at a large nuclear plant site.

SECTION V

MECHANICAL DRAFT WET TOWERS

System Description

Mechanical draft wet towers are divided into two categories, forced air flow and induced air flow. Induced draft towers are further subdivided into counterflow and crossflow towers. Induced draft towers are favored over forced draft towers (Ref. 7). Crossflow induced draft towers have advantages over counterflow induced draft towers and are most commonly used. Figure 5 shows the basic arrangement of a crossflow induced draft tower.

Crossflow induced draft towers can usually attain better thermal performance than counterflow towers, i. e., heavier water loadings, longer ranges and closer approaches. For a specified applied horsepower a greater airflow is possible in crossflow towers because of lower static losses, and thus they are more efficient. Other advantages of the crossflow induced draft tower include low pumping head, convenient arrangement of the distribution system, the fill height approximately equal to the tower height, more air flow per fan horsepower, and the ability to use fans of larger diameter so that fewer cells are required for a given capacity. Disadvantages are an insufficient pressure head on the distribution pans to keep the orifices from becoming clogged and the entire water feed exposed to the air which favors growth of algae. Also a substantial crossflow correction factor needs to be applied to the driving force, particularly where long range and close approach performance are required.

Discussion of the Nomographs

The nomographs for the mechanical draft wet towers are found on pages N-7 through N-12 of the Appendix. The development of each of these nomographs is discussed in the following paragraphs.

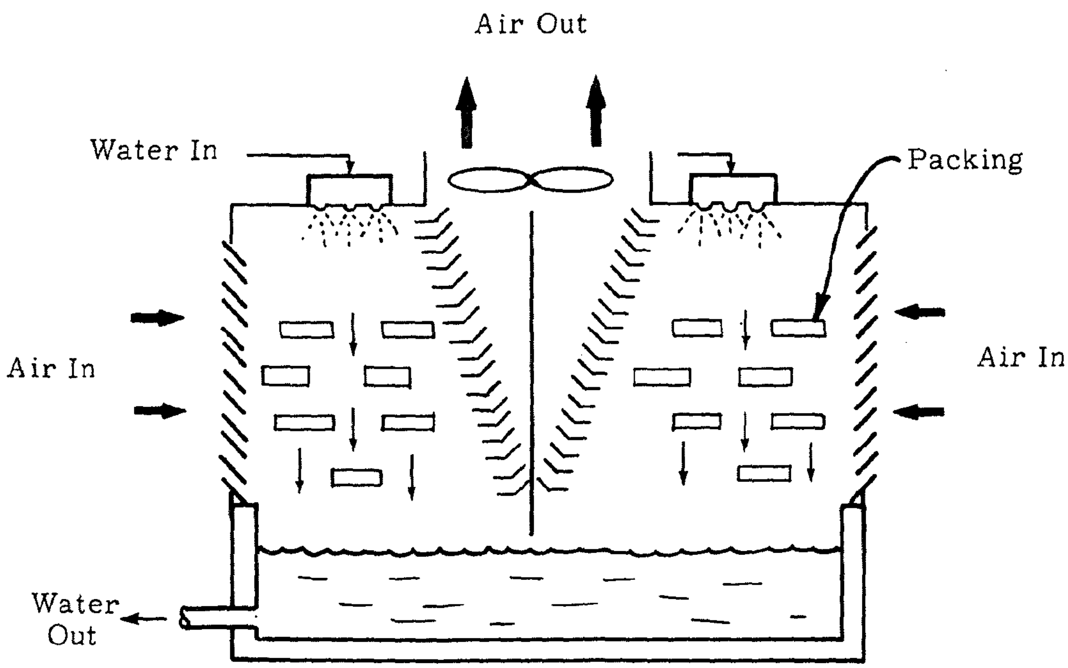


Figure 5. Crossflow Induced Draft Tower

The Tower Correction Factor nomograph, page N-7, is based on the rating factor curves for large plants published by the Marley Company in Reference 4. The tower correction factor (synonymous to Marley's rating factor) is a function of wet bulb temperature, range and approach. The limits of these input parameters cover all of the most logical design conditions relative to the electric utility industry.

The Turbine Back Pressure nomograph, page N-8, yields the condenser inlet temperature and the steam turbine back pressure. The temperature of the cold water leaving the tower, or tower effluent temperature, equals the sum of the wet bulb temperature and the approach. Under closed cycle operation, the tower effluent temperature is equal to the condenser inlet temperature. The turbine back pressure is a function of the steam saturation temperature which is determined from equation (3).

The Flow Rate nomograph, page N-9, yields the water flow rate from equation (5). Under closed cycle operation the condenser rise is equal to the range of the mechanical draft tower. The types of operation for a heat rejection system are defined with the aid of Figure 6. In the figure, the closed cycle case shows an example where both the range and condenser rise are equal to 25^{o}F. The cooling water is totally recirculated except for a small quantity of makeup water necessary to replace evaporative, drift and blowdown losses.

Open cycle cooling is also illustrated by an example in Figure 6. Hot water from the condenser is passed through a supplementary heat rejection system (wet tower or pond) for cooling prior to discharge into a natural receiving body (stream, river, bay or ocean). An allowable temperature difference is defined as being the temperature of the hot discharge water from the heat rejection system into the stream minus the temperature of the water available from the stream. The temperature of the water available from the stream is equal to the condenser inlet temperature, an input to N-8 for the open cycle case. The allowable temperature difference usually ranges from 0^{o}F to 5^{o}F maximum and depends upon site characteristics and legal requirements.

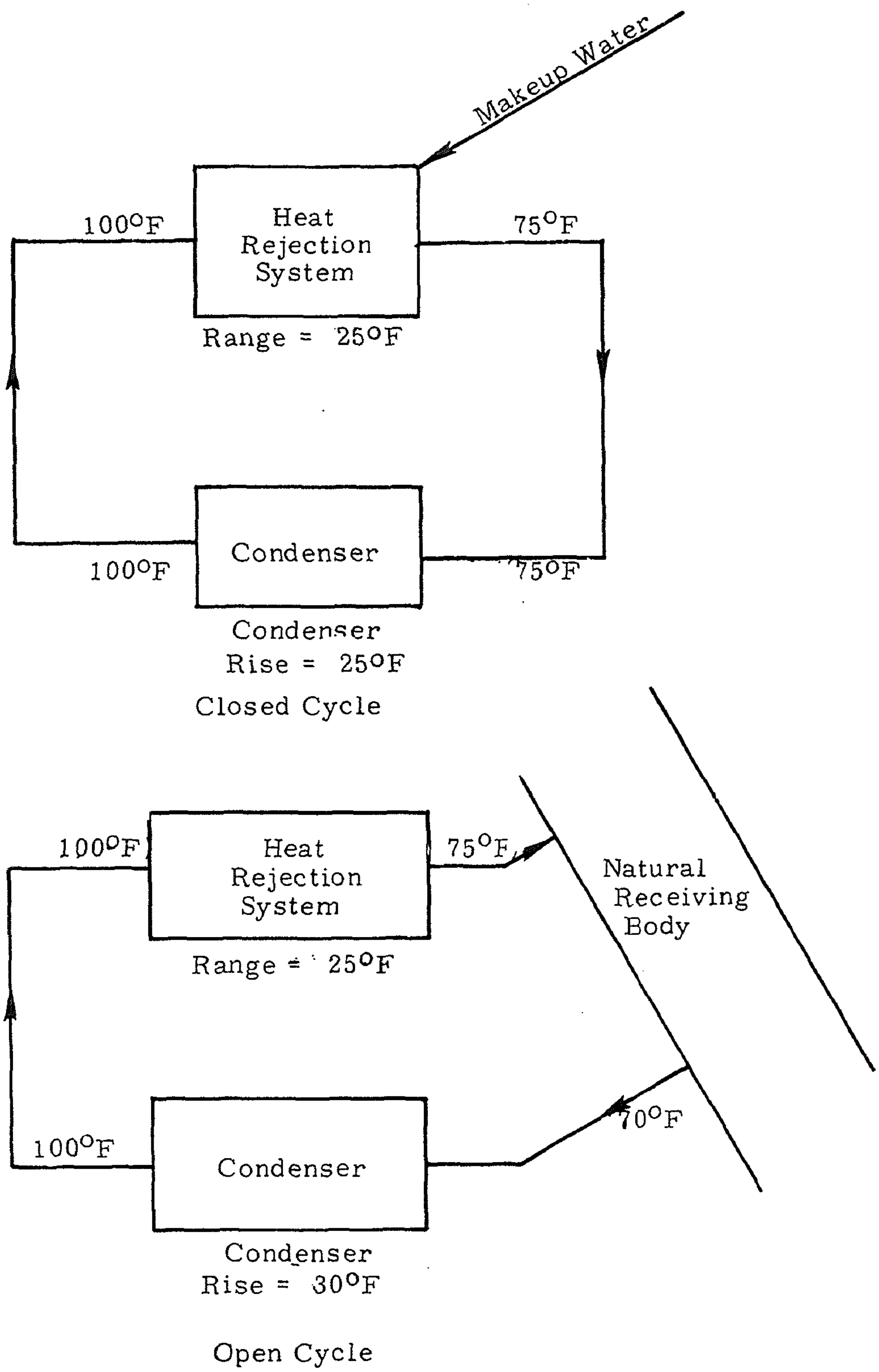


Figure 6. Types of Heat Rejection System Operation

The allowable temperature range is discussed further in Reference 8 with respect to streams with cold or warm water fisheries, lakes and reservoirs. In the figure the discharged water temperature to the stream ($75^{o}F$) minus the available water temperature from the stream ($70^{o}F$) is equal to $5^{o}F$.

Open cycle cooling has its limitations caused by changing meteorological conditions. It may not work year round. For example, if the wet bulb temperature is $75^{o}F$ and the approach is $10^{o}F$, then the discharge water temperature from the heat rejection system would be $85^{o}F$. Water at this temperature would not be allowed to be discharged into the stream if the available water temperature from the stream was $77^{o}F$.

Some power plants can utilize a combination closed cycle and open cycle system. The plant would try to maximize the time during the year where open cycle cooling would take place. However, under adverse meteorological conditions, it would need to close the open cycle loop when the discharge temperature exceeded that allowed under local water quality standards. Restriction to closed cycle operation might mean that the plant would have to cut back on output (partial load operation to reduce heat rejection).

The Tower Cost nomograph, Page N-10, provides for the determination of the number of tower units and the corresponding mechanical draft tower cost. The number of tower units (in millions) results from the product of the tower correction factor from N-7 and the water flow rate determined in the previous nomograph. The tower unit will be recognized as a synthetic square foot of heat transfer hardware used in a similar manner as a square foot of surface condenser. Thus, the power plant designer can accumulate a library of known costs per tower unit and costs per square foot of condenser surface in order to estimate capital costs. In Reference 4 a library of costs per tower unit was presented for nine cases representing actual mechanical draft cooling tower selections for the electric utility industry. The costs per tower unit ranged from $4 to nearly $5.5 and the average of the data was $4.75

per tower unit. Therefore, this range of costs per tower unit was incorporated into the Tower Cost nomograph. Then mechanical draft tower cost is the product of the number of tower units and the costs per tower unit.

The nomographs developed up to this point were employed to check, as an example, the tower cost result for case No. 8 of Reference 4. Design conditions for this case were:

WBT = 75°F

Range = 25.8°F

Approach = 18°F

Water flow rate = 0.305×10^6 gpm

Output results from the nomographs were:

Tower correction factor = 0.76

Tower effluent or condenser inlet temperature = 93°F

Turbine back pressure = 3.95 in.Hg abs

Heat rejection rate = 3.9×10^9 Btu/hr

Number of tower units = 0.23×10^6

Mechanical draft tower cost = $\$1.2 \times 10^6$

A cost per tower unit equal to $5.26 was used in the Tower Cost nomograph. The mechanical draft tower cost agreed with that of Reference 4. This capital cost does not include cold water basins, fan motor starters, controls, and wiring. Comparison of costs predicted by the nomographs with those from other sources is done in Section XII after introducing the nomograph example. Good correlation between the costs is shown. Based on these costs, an approximate 15% increment should be added to the value from the Tower Cost nomograph in order to include the basins and electrical apparatus.

In the Fan Power nomograph, page N-11, fan power requirements are directly proportional to the number of tower units (TU). In Reference 4 similar to cost per tower unit, the fan horsepower per tower unit was presented for nine different cases considering applications of mechanical draft cooling towers. The average mechanical draft fan power required was 0.011 $\pm$ 25% hp per tower unit. Therefore, the fan power requirements in Mw is obtained from the following:

$$\text{fan power, Mw} = 0.011 \frac{\text{hp}}{\text{TU}} \times 0.746 \times 10^{-3} \frac{\text{Mw}}{\text{hp}} \times \text{TU}$$
$$= 0.82 \times 10^{-5} \times \text{TU} \qquad (8)$$

The Pump Power nomograph, page N-12, estimates the pump power requirements for mechanical draft towers in either the closed cycle or open cycle modes of operation. Equation (6) is employed for either mode of operation. The pump efficiency is assumed to be 0.85. The difference in pump power arises in the total head parameter corresponding to either mode of operation.

Pumping heads for mechanical draft towers alone in large power station units range from 50 to 60 ft according to Reference 9. Therefore a 60 ft total pumping head was assumed as a constant in the pump power equation. Consequently, for the tower operating in the closed cycle mode, a single straight line relates pump power requirements to the water flow rate.

Under open cycle operation, water is transported from the source and then returned to the source after passing through the condenser and the cooling tower. The pumping head and losses are very dependent upon the plant site and source of water. Therefore, in the nomograph the total pumping head is represented by a family of lines. Total pumping head includes the 60 ft pumping ahead ahead attributed to the tower alone plus a variable pumping head component whose choice is site dependent.

Finally, the auxiliary power required to operate the mechanical draft tower equals the fan power determined in the preceding nomograph plus the pump power resulting from either closed or open cycle operation.

SECTION VI

SPRAY PONDS

System Description

Spray ponds for utility sites currently are available in two different configurations, conventional spray ponds and powered spray systems, including those produced by Ceramic Cooling Tower Co., Richards of Rockford and Ashbrook. Also, a spinning disk spray system is being developed by Cherne. In conventional spray ponds, warm water is pumped through pipes from the condenser and then out of the spray nozzles. The nozzles atomize the warm water into fine droplets in the neighborhood of 3/16-inch diameter. The powered spray systems employ quantities of floating spray modules, with one or more sprays, independently deployed in the warm water discharge canal or reservoir. The modules require mooring and electrical connections to shore. An attempt was made to include the performance and costs of the powered spray systems in the nomographs. However, due to limits on available data, the scope of the nomographs is confined to conventional spray ponds only.

The basic arrangement of a conventional spray pond is shown in Figure 7. The spray nozzles are usually located five to 10 ft above the surface of the pond. Height of the sprays is approximately seven ft. The drops move irregularly relative to the air medium and are more or less deformed, while circulation occurs within the droplets. Heat transfer between the droplets and the medium occurs by the three mechanisms of conduction, convection and radiation.

Performance of the spray pond is a strong function of the design of the nozzles. Poor designs require high pressure and power, do not mix the air and water efficiently, and have excessive water losses. However, water losses can be decreased with the use of louvered fences. Performance is limited by the relatively short contact time between the air and water spray.

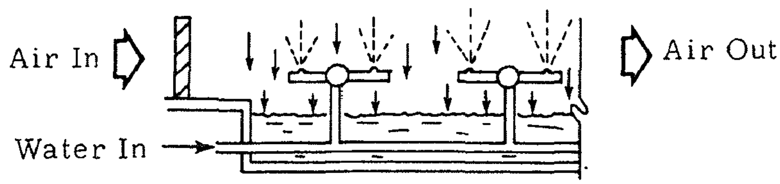


Figure 7. Spray Pond

An advantage of spray ponds is the requirement for very little maintenance other than occasional cleaning of the pipes and nozzles and routine maintenance on the pumps. However, impurities can easily enter the open water area and be carried into the condenser, increasing maintenance requirements. The disadvantages of the danger of freezing and/or poor heat transfer due to climatic conditions may also occur.

Discussion of the Nomographs

The nomographs for spray ponds are found on pages N-13 through N-16 of the Appendix. Their development is discussed in the following paragraphs.

The Spray Pond Performance nomograph, page N-13, is based on data published by Elgawhary and Rowe (Ref. 10) and Perry's Handbook (Ref. 11). Pond water temperature after spray is a function of inlet water temperature, wet bulb temperature, and wind velocity. In Reference 10, the spray model consisted of an elevated nozzle spraying hot water above a large pond surface.

In order to utilize the N-13 nomograph, the selected inlet (to the spray system) water temperature must be greater than the wet bulb temperature plus condenser rise for a closed cycle system. For example, entry with an inlet water temperature of $80^{o}F$ and a WBT of $75^{o}F$ would be an invalid set of conditions for a closed cycle system. This is true because the condenser rise and the approach together are bound to exceed the $5^{o}F$ required in this impossible case.

The Turbine Back Pressure nomograph, page N-14, yields the steam turbine back pressure which is a function of the steam saturation temperature. For a closed cycle system, first, the condenser rise is determined by subtracting the pond water temperature determined from N-13 from the inlet water temperature. The pond water temperature is equal to the condenser inlet temperature (not to be confused with the inlet water temperature to the spray pond). Then one enters the Turbine Back Pressure nomograph with the condenser inlet temperature and the condenser rise in order to obtain the turbine back pressure. For this system, the steam saturation temperature results from equation (3). For an open cycle system, the turbine back pressure is determined from N-14 after entry with a given condenser rise and a condenser inlet temperature due to the available water from the natural receiving body.

The Flow Rate nomograph, page N-15, yields the water flow rate from equation (5). Under closed cycle operation the condenser rise is equal to the range of the spray pond. Figure 6 can also be employed for spray ponds to illustrate the magnitude of condenser rise for both closed and open cycle modes of operation of this heat rejection system.

On page N-16 are found nomographs for Spray Pond Cost and Auxiliary Power. The spray pond capital cost in dollars is defined as being the product of the water flow rate in gpm and the pond unit cost in dollars per gpm. The nominal unit cost value of \$9 per gpm is based on the installed cost for the spray pond at the Canadys Station of the South Carolina Electric and Gas Company. From Reference 12 data, it was derived as follows:

$$\text{pond unit cost, \$/gpm} = \frac{\text{spray pond capital cost, \$/Kw x plant size, Kw}}{\text{water flow rate, gpm}}$$

$$= \frac{\$3.65/\text{Kw x } 450,000 \text{ Kw}}{180,000 \text{ gpm}} = \$9/\text{gpm}$$

The spray pond capital cost in \$/Kw includes site preparation, material and construction labor costs on a 1971 basis.

Auxiliary power requirements in Mw are defined as being the product of the water flow rate in gpm and the power consumption in Kw per gpm. The range of values for power consumption are based on Reference 13. They represent the pumping power requirements of conventional spray ponds under both closed and open cycle modes of operation. The pumping power requirements in Mw are due to the spray and to the distribution piping losses.

SECTION VII

COOLING PONDS

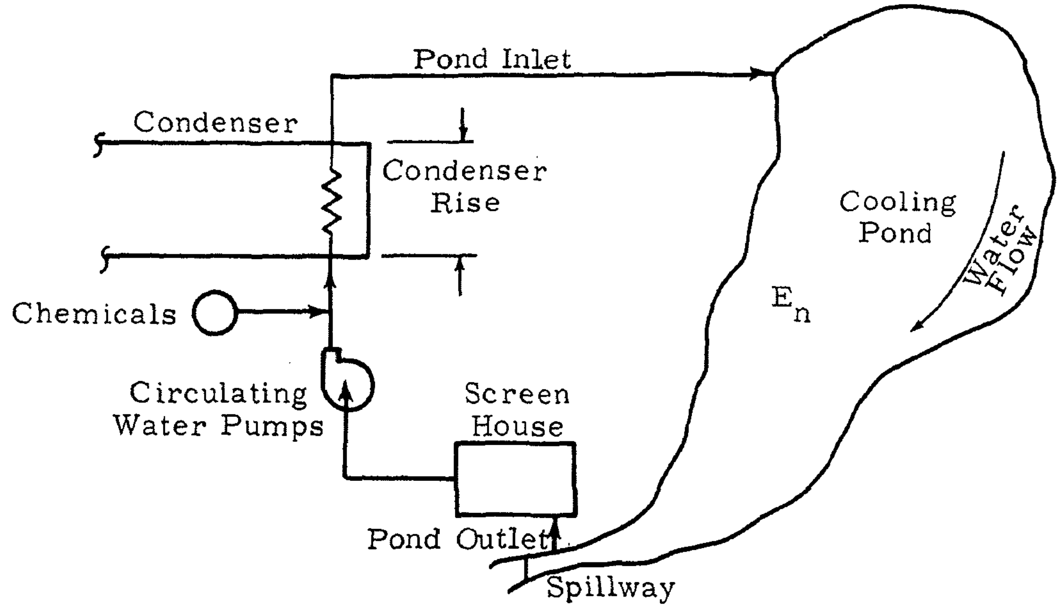
System Description

Heat rejection from power plants through the use of cooling ponds is highly dependent upon site weather conditions. The heat is rejected from the pond surface by the natural effects of conduction, convection, radiation, and evaporation. However, the cooling pond also absorbs heat through the processes of solar radiation and atmospheric radiation to the pond, as well as waste heat from the power plant.

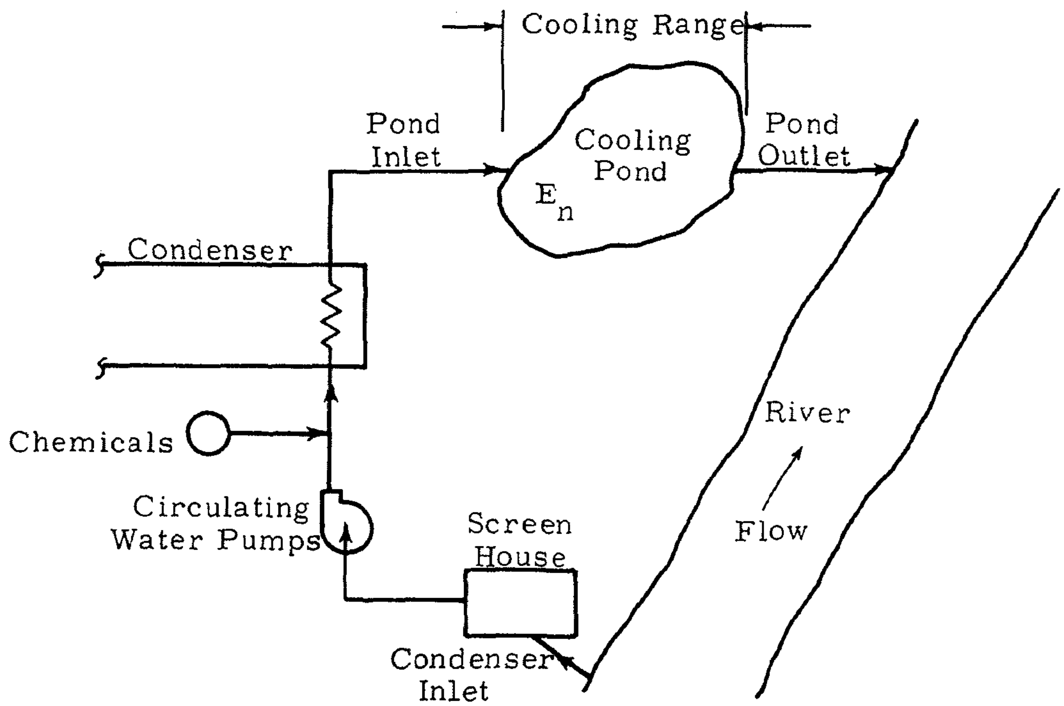
Cooling pond systems in the closed cycle and open cycle modes of operation are shown schematically in Figure 8. The closed cycle system resembles the once-through cooling system with the exception that the cooling water is recirculated in a cooling pond. If the pond size is in excess of 2 acres/Mw for a fossil plant or 3 acres/Mw for a nuclear plant, the surface water temperatures will closely coincide with once-through cooling systems. However, many cooling lakes are much smaller and therefore, operate at much higher surface water temperatures.

Advantages of using cooling pond systems are: reasonable construction costs where soil conditions permit, their service as a settling basin for suspended solids, potential operation for extended periods without makeup water, and benefits from other purposes such as recreation. Disadvantages of cooling pond systems are: the requirement for large land area and the requirement for a soil basin of low permeability.

Cooling ponds can be classified, on the basis of circulation pattern and temperature distribution, as completely mixed ponds and flow-through ponds. In a completely mixed pond, the flow between the inlet and outlet locations of the pond combined with wind mixing tend to keep the pond at a nearly uniform temperature. The drop in the temperature takes place by mixing in a small portion of the pond. This



Closed Cycle



Open Cycle

Figure 8. Cooling Pond Systems

condition seems to exist in ponds where the surface area is large compared to the water flow rate from the plant and the pond temperature is uniformly 1 to 2°F above the equilibrium temperature. The equilibrium temperature is defined as the pond surface temperature for which the heat leaving the surface will exactly equal the heat entering the surface of a body of water.

In a flow-through pond the temperatures decrease continually along the pond. The pond effluent can either be returned to the plant intake (closed cycle) or discharged to a natural receiving body (open cycle). The amount of temperature drop from one end of the pond to the other depends upon the initial temperature rise, water flow rate, pond surface area, and meteorological conditions. For a given pond volume and similar meteorological conditions, a pond with greater surface area will provide a greater temperature drop. The rate at which heat is lost decreases from the pond inlet to the pond outlet because the temperature excess gradually decreases along the pond. Flow-through ponds are more common.

Discussion of the Nomographs

The nomographs for cooling ponds are presented on pages N-17 through N-21 of the Appendix. Their development and underlying assumptions are discussed in the following paragraphs.

For a flow-through pond, the one-dimensional, exponential temperature decay equation is:

$$T_{out} = (T_{in} - E_n) \exp\left(- \frac{k A_p}{\rho C_p WFR}\right) + E_n \qquad (9)$$

where:

T_{out} = pond outlet temperature, °F

T_{in} = pond inlet temperature, °F

E_n = design equilibrium temperature, °F

k	=	heat exchange coefficient, Btu/ft^2 day ^oF
A_p	=	pond area
ρ, C_p	=	density and specific heat for water
WFR	=	water flow rate

Under closed cycle operation (see Figure 8), ΔTC is defined to be the condenser rise which is equal to $T_{in} - T_{out}$. The residual temperature rise, ΔT, is defined as:

$$\Delta T = T_{out} - E_n \tag{10}$$

It is similar to the "approach" parameter specified for the wet cooling towers and spray ponds. Following the substitution of these definitions and the rearrangement of equation (9), one obtains the following equation:

$$\frac{A_p}{WFR} = - \frac{\rho C_p}{k} \ln \left(\frac{\Delta T}{\Delta T + \Delta TC}\right) \tag{11}$$

The water flow rate in equation (11) and the plant size are related through equation (5) and equation (17), for the heat rejection rate presented in Section IX. In the Cooling Pond Performance nomograph, page N-17, a family of curves, ΔT vs. Mw/acre (pond loading) was developed for each ΔTC using k as a variable. The heat exchange coefficient, k, describes the rate of heat lost across the air-water interface per unit area per unit temperature increase and depends upon the local site meteorology. It was assumed in the computations that the fossil and nuclear plant efficiencies remained constant at their respective values, 35. 8 percent and 31. 2 percent, corresponding to a turbine back pressure of 2 in. Hg absolute. This is a reasonable assumption for the range of back pressure operation applicable to cooling ponds, 2 to 4 in. Hg absolute.

For the open cycle cooling pond system (see Figure 8), equation (9) again applies for the flow-through pond. However, cooling pond performance also depends upon the temperature of the water available from the river or the condenser inlet temperature in addition to condenser rise, ΔT and k. The following parameter is defined:

$$\Delta TC' = \text{condenser rise} - \text{pond outlet temperature} + \text{condenser inlet temperature} \qquad (12)$$

Then equation (9) becomes for the open cycle case:

$$\frac{A_p}{WFR} = - \frac{\rho C_p}{k} \ln \left(\frac{\Delta T}{\Delta T + \Delta TC'}\right) \qquad (13)$$

Thus, it follows that the same curves can be employed on N-17 for the open cycle case except for the substitution of $\Delta TC'$ for ΔTC when one enters the nomograph.

Pond sizes predicted from the curves of N-17 (following interpolation) are compared in Table 3 with those from several plant cases described in References 14, 15, and 16. The k values for each site were estimated from the design chart presented by Brady (Ref. 17). He presents k as a function of windspeed and an average temperature which is based on the dewpoint temperature, T_d, and the water surface temperature, T_s. For each case, the T_d was assumed to be the maximum dewpoint temperature in July for the site location. T_s was assumed to be equal to the design equilibrium temperature, E_n, at each site. The design windspeed for each site was assumed to be equal to the July mean windspeed except for the case from Reference 15 where this parameter was given. The comparison of the pond size shows excellent agreement between the nomograph curves and the cases from References 14 and 15. The last case in the table is overestimated by the performance curves. It is concluded that the performance nomograph will provide good estimates of cooling pond sizes.

The Turbine Back Pressure nomograph, page N-18, yields the steam turbine back pressure and is based on equation (3).

TABLE 3. COOLING POND PERFORMANCE

Site Location	Plant Size Mw	Cooling Pond Cycle	Water Flow Rate cfs	Condenser Rise or Range °F	k Btu/day-ft2-°F	E_n °F	ΔT °F	Pond Size: Acres (Reference)	Pond Size: Acres (From Curves of N-17)
Southeast -Miami	385-Fossil	Closed	320	24	153	84.6	1	1000 (Ref. 14 Sta. 2)	895
Michigan	1000-Fossil	Closed (Flow-through)	950	20	162	86	2	1740 (Ref. 15 Case V)	1890
Southwest -Yuma	1000-Nuclear	Closed (Flow-through)	1300	22	177	83.5	6.5	1140 (Ref. 16)	1695

The Cooling Pond Cost nomograph, page N-19. yields the land and excavation costs. First the pond size in acres is obtained by dividing the plant size in Mw by the pond loading parameter, Mw/acre, determined in the Cooling Pond Performance nomograph. The cooling pond land and excavation cost in dollars is obtained from the product of the pond size and the unit cost in dollars per acre.

The range of this unit cost appearing on the nomographs is justified in several references. In Reference 9, it was pointed out that cooling pond costs are a very strong function of the amount of land required. A unit cost of $1000 per acre for land and excavation appears to be reasonable. However, in the future the cooling pond may prove to be more expensive as larger amounts of more expensive land are required for big nuclear plants. Also, in the series of cooling pond cases considered in Reference 15, land costs of $500 per acre and $1000 per acre were employed in the Lake Michigan area.

The Flow Rate nomograph, page N-20, yields the water flow rate directly from equation (5).

The Auxiliary Power nomograph, Page N-21, estimates the auxiliary power due to the pump power requirements for cooling lakes. Equation (6) is employed for either mode of operation. Pump efficiency, as before, is assumed to be 0. 85. The difference in pump power arises in the total head parameter corresponding to either mode of operation. The pumping head and losses are also very dependent upon the plant site. Therefore, in the nomograph the total pumping head is represented by a family of lines whose range is assumed to span cooling pond applications under the two modes of operation.

SECTION VIII

MECHANICAL AND NATURAL DRAFT DRY TOWERS

System Description

In the mechanical and natural draft dry tower heat rejection systems, the circulating water never comes into direct contact with the cooling air. Condensation by air cooling has been used in small industrial power plants for over 50 years. However, the application of air cooling to relatively large generating units has been limited. There are two basic types of air-cooled condensing systems, the indirect system and the direct system. The indirect system uses a condenser at the turbine to condense the exhaust steam. It is often referred to as the Heller system since the concept to use the indirect system of condensation by air cooling with a steam turbine generator was presented by Dr. László Heller in 1956. In the direct system, steam is condensed in the tower cooling coils without the use of a condenser or circulating water.

Indirect, dry-type heat rejection systems with a natural draft tower and with a mechanical draft tower are shown schematically in Figures 9 and 10, respectively. These diagrams present the principal components for the indirect systems according to Reference 18. Water from the condenser is pumped to the dry-type tower for cooling. Water from the tower cooling coils is sprayed into the direct contact steam condenser and mixes directly with the exhaust steam from the turbine. The water of condensate purity falls to the bottom where it is removed by circulating and condensate pumps. The greater part of the water flows through the pipes back to the cooling coils, and an amount equal to the exhaust steam from the turbine is directed back to the boiler feedwater circuit.

In the Heller system, the cooling coils are mounted vertically, and the warm circulating water enters the bottom of the coils, flows upward in the inner rows of coils to the top water boxes, and then is directed downward through the outer rows of coils. The outer rows of coils come into contact with the entering air, thereby providing the greatest cooling range in water temperature.

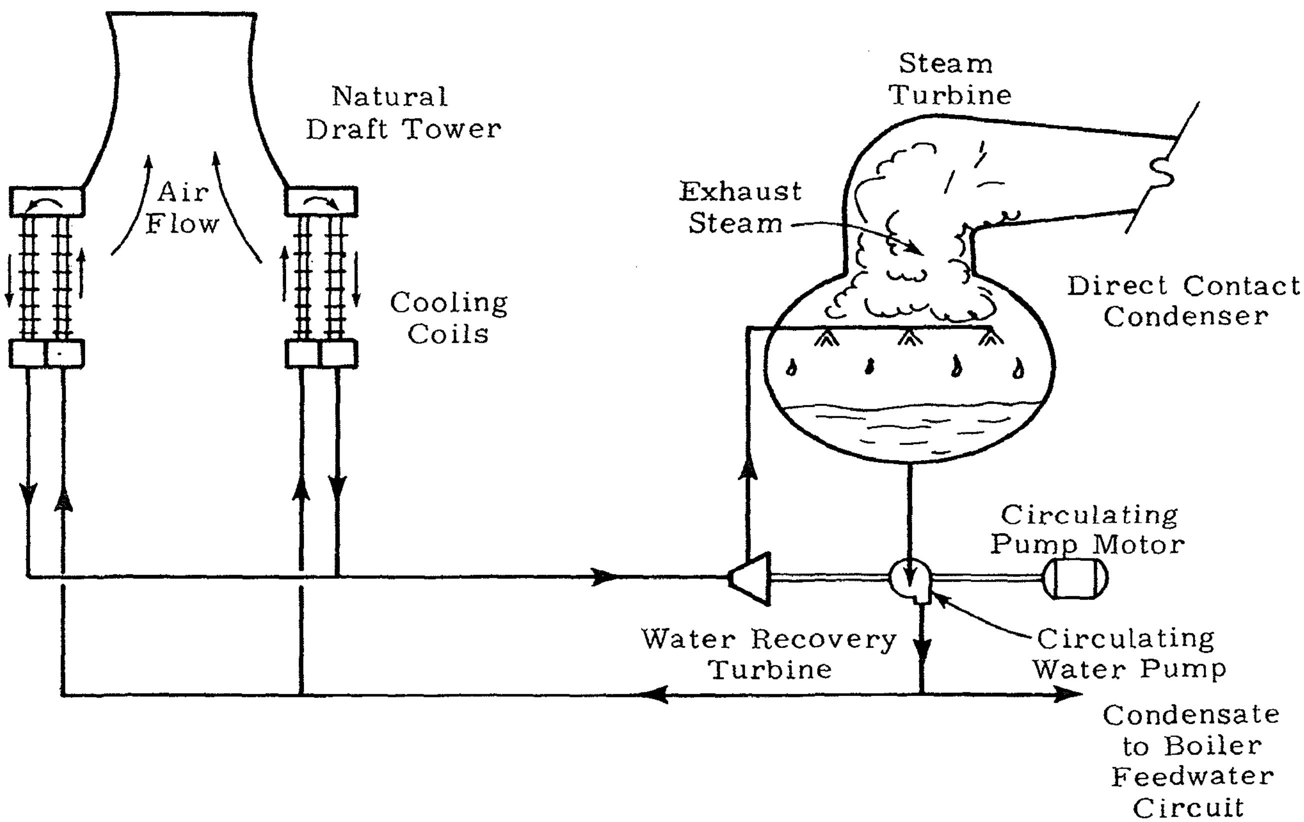


Figure 9. Indirect, Dry-Type Heat Rejection
System with Natural Draft Tower

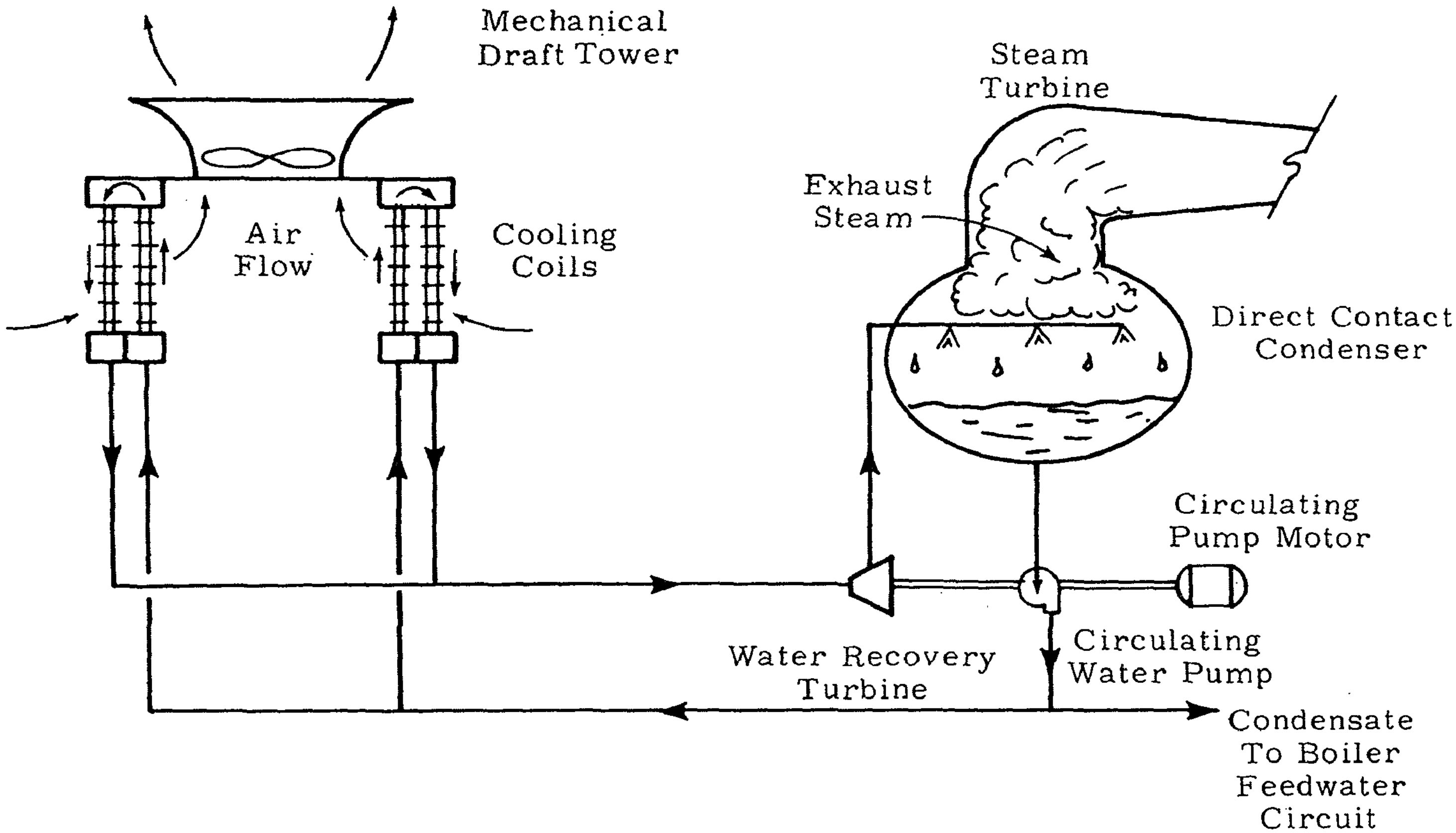


Figure 10. Indirect, Dry-Type Heat Rejection System with Mechanical Draft Tower

In order to prevent drawing air into the system in case of leaks in the cooling coils, a positive pressure head of approximately three feet is imposed at the top of the coils. This is accomplished by means of either a throttling valve in the circulating water discharge from the tower, or, if a water recovery turbine is used, by varying the position of the adjustable turbine vanes. In order to recover some of the pressure head between the cooling coils and the condenser, in some installations water recovery turbines are coupled to the drive shaft of the circulating water pump to recover the available energy.

After passing through the recovery turbine, the circulating water is again sprayed into the direct contact condenser and recycled through the cooling system. Since the circulating water does not come into direct contact with the cooling air, there is no evaporative loss of water as with the wet-type towers.

Discussion of the Nomographs

The nomographs for the mechanical draft and natural draft dry towers are presented on pages N-22 through N-26 of the Appendix. The development of each of these nomographs is discussed in the following paragraphs.

The nomograph on page N-22 entitled Combined Performance - Natural Draft shows the performance curves for a large natural draft dry tower superimposed on steam turbine performance curves for power plants representing several sizes. The performance for this tower was presented in Reference 18. The tower was dual rated, 4×10^9 Btu/hr heat rejection rate at 50°F ITD (initial temperature difference) and 6×10^9 Btu/hr at 67.8°F ITD, corresponding to fossil and nuclear plant application, respectively. This natural draft tower represents one of many applicable tower designs. The derivation of natural draft dry tower performance and the use of the nomograph is discussed in succeeding paragraphs.

The performance of dry-type towers is governed by the following equation (Ref. 18):

$$\text{ITD} = AQ^b \tag{14}$$

where,

ITD	=	initial temperature difference, ^{o}F
	=	steam saturation temperature - dry bulb temperature
A	=	tower constant
Q	=	heat rejection rate, 10^6 Btu/hr
b	=	exponent dependent upon natural or mechanical draft towers

For the natural draft dry tower, an average value of b is about 0.75 (Ref. 18). Therefore, the performance equation for the natural draft tower rated 4×10^9 Btu/hr heat rejection rate and 50^o ITD is fully determined after solving for A in equation (14):

$$\text{ITD} = AQ^b$$

$$50 = A \times (4000)^{0.75} \text{ or}$$

$$A = 0.1$$

This performance equation is:

$$\text{ITD} = 0.1\,(Q)^{0.75} \tag{15}$$

With equation (15) the family of performance curves of turbine back pressure versus heat rejection rate for various dry bulb temperatures was developed. First equation (15) was used to compute ITD's corresponding to Q's or heat rejection rates other than 4×10^9 Btu/hr. Then a dry bulb temperature was selected, e.g. 90^oF. The addition of 90^oF to each ITD yields a corresponding steam saturation temperature. The steam tables are employed to obtain turbine back pressures in in. Hg absolute. Thus, a curve of turbine back pressure versus heat rejection rate at constant dry bulb temperature can be plotted. Following the same procedure, performance curves for other constant dry bulb temperatures are determined and appear in nomograph N-22. Thus, the turbine back pressure performance of this natural draft dry tower is shown for various ambient air temperatures and heat rejection loads.

Turbine performance curves for fossil and nuclear plants of several net generating capacities are overlaid on the nomograph. The development and discussion of these curves, some of which appear in the turbine performance nomographs, are in Section IX. The curves of heat rejection rate versus turbine back pressure are based on the assumption that the specified net generating capacity is maintained constant as back pressure increases.

The intersection of the turbine performance curve for a plant operating at a given rated capacity with the tower performance curve at a given dry bulb temperature yields operating conditions for the given tower. Thus, the output of the nomograph is turbine back pressure and heat rejection rate. The range of net generating capacities displayed is assumed to correspond to the range of applicability for this tower design. This tower design is not the optimum one for the range of plant capacities shown. Reference 18 pointed out that this particular tower design was dual rated for application to both 800 Mw fossil and nuclear plants. The optimum tower design results only after the performance and costs are computed for dry towers having various combinations of Q and ITD (ranging from 30 to 80^{o}F) in equation (14). Therefore, N-22 is only applicable for making first estimates of turbine back pressure and heat rejection rate for the natural draft dry tower with a given plant.

The nomograph on page N-23 entitled Combined Performance - Mechanical Draft shows the superposition of mechanical draft dry tower performance curves onto steam turbine performance curves of various power plants. This mechanical draft tower has a dual rating of 4×10^9 Btu/hr at 50^{o}F ITD and 6×10^9 Btu/hr at 72^{o} ITD corresponding to fossil and nuclear plant applications, respectively. It represents one of many applicable tower designs. The Q equal to 4000 and the ITD equal to 50^{o}F are substituted into the dry tower performance equation (14). Also substituted is b equal to 0.91, the exponent for mechanical draft dry towers (Ref. 18). The solution of equation (14) based on these values yields a tower constant, A, equal to .0263. Therefore, the performance equation for this mechanical draft dry tower is:

$$\mathrm{ITD} = .0263\,(Q)^{0.91} \tag{16}$$

With equation (16) the family of performance curves was developed for various dry bulb temperatures by following the same procedure outlined for the natural draft dry tower nomograph. Thus, in N-23 the turbine back pressure performance of the mechanical draft dry tower is shown versus heat rejection rate and dry bulb temperature.

Turbine performance curves for several fossil and nuclear plant capacities are overlaid onto N-23. Incorporated into these curves is the assumption that the specified net generating capacity is maintained constant as back pressure increases.

The intersection of the turbine performance curve for a plant operating at a given rated output with the tower performance curve at a given dry bulb temperature yields the outputs of turbine back pressure and heat rejection rate for the given tower. This mechanical draft dry tower design is assumed to be applicable over the range of net generating capacities displayed. For the same reasons stated in the natural draft dry tower discussion, this tower design is also not the optimum one for the range of plant capacities shown. Therefore, N-23 is only applicable for making first estimates of turbine back pressure and heat rejection rate for the mechanical draft dry tower with given plant sizes confined to a limited size range.

The Initial Temperature Difference nomograph, page N-24, allows the ITD to be determined, as defined in equation (14), from the turbine back pressure and the dry bulb temperature. For an input turbine back pressure from the dry tower performance nomographs, a corresponding steam saturation temperature is obtained from steam tables. A saturation temperature scale is also displayed for approximate estimates on the nomograph. The inital temperature difference is the design approach parameter for dry towers.

The Capital Cost nomograph, page N-25, yields costs of heat rejection systems employing dry towers versus ITD and plant capacity and type. The heat rejection system (Figure 9 and 10) is defined as including all equipment and installation from the turbine flange outward: condenser, cooling system piping, water storage facilities, pumps, valves, controls, recovery turbine if used, and the cooling tower with its heat exchanger equipment. Reference 18 presented data showing that the capital costs of the natural draft and mechanical draft dry systems (and the physical size) decrease with increasing ITD for 800 Mw fossil and nuclear-fueled plants based on 1970 cost levels. These capital cost data were escalated to 1971 cost levels using a factor of 1.14 based on the Engineering News Record. Then the 1971 capital cost levels for 800 Mw power plants were linearly extrapolated to yield capital costs for the plant sizes displayed in the nomograph. The extrapolation is justified based on Reference 18. It was stated that the results of their cost analysis, as evaluated on a cost per Kw basis, should be generally applicable to generating plants in the size range of 600 Mw to 1000 Mw, or over a somewhat larger range of sizes.

The Auxiliary Power nomograph, page N-26, yields auxiliary power requirements in Mw for mechanical draft and natural draft dry towers versus ITD and the plant capacity and type. These data were tabulated in the Beck report, Part II (Ref. 18) for 800 Mw fossil and nuclear fueled plants. The auxiliary power requirements for pumps and fans decrease with increasing ITD. The data were linearly extrapolated to yield auxiliary power for the plant sizes displayed in the nomographs.

SECTION IX

STEAM TURBINE GENERATOR PERFORMANCE

Background

In order to obtain performance data on large steam turbine generators, contacts were made with the Power Generation Sales Division of the General Electric Company, the Large Turbine Division of Westinghouse Electric Corporation, Allis-Chalmers Power Systems, Inc., and the Brown-Boveri Corporation. Requests were made specifically for turbine heat rate and throttle steam rate versus turbine generator output and high turbine back pressure. It would have been desirable to obtain these specific data for standard modified turbine designs and high back pressure turbine designs for both fossil and nuclear plant applications. A large quantity of publications was received from Westinghouse and General Electric covering the heat rate performance of large steam turbine generators in nuclear and fossil-fueled plant applications. These papers have been published over the years in the Proceedings of the American Power Conference and the ASME Journal of Engineering for Power.

Turbine performance at increasing back pressure will be different for the two cases:

(1) Maintain constant generator output by increasing steam rate.

(2) Allow generator output to decrease while maintaining constant steam flow rate (derated plant).

It was assumed in the nomographs that at a given plant site, the rated output would be required under all site conditions, Case 1. In so doing, it is assumed that the generator can take a certain overload steam rate.

Discussion of the Nomographs

The Turbine Performance nomographs are presented on pages N-27 through N-32 of the Appendix. Their development and underlying assumptions are discussed in the following paragraphs.

In the Heat Rejection Rate nomograph, page N-27, the heat rate rejection rate is found as a function of the turbine back pressure, the net generating capacity for a fossil-fueled plant and the base load heat rate. Tower or pond heat rejection rates can be related to the in-plant and plant efficiencies by the following equation (Ref. 9):

$$\text{Heat Rejection Rate, Btu/hr} = 3413 \times (\text{plant size, Kw}) \times \left[\frac{\eta_I}{\eta_P} - 1\right] \qquad (17)$$

where:

η_I = in-plant or steam supply efficiency, which accounts for in-plant heat losses. This efficiency is 85 percent (15 percent in-plant and stack losses) for fossil plants and 95 percent (5 percent in-plant losses) for nuclear plants (also Refs. 8 and 15).

η_P = plant efficiency which depends upon the turbine back pressure. (It is the thermal efficiency of the total plant.)

The heat rejection rate varies as the turbine back pressure changes. This is due to varying plant efficiency and more heat addition to the plant in order to keep the electrical output constant.

Plant efficiencies versus turbine back pressures, required in equation (17), were based on turbine performance data tabulated by Beck Associates in Reference 18, Part II. They presented plant heat rate versus turbine back pressure for 800 Mw fossil and nuclear plants which assumed in-plant efficiencies of 90 percent and 100 percent, respectively. These plant heat rates were corrected to account for fossil and nuclear in-plant efficiencies of 85 percent and 95 percent, respectively. Therefore, at the base back pressure of 2 in. Hg absolute, there resulted plant heat rates equal to 9541 and 10,937 Btu/Kw-hr, respectively, for fossil and nuclear plants. The performance data used from Reference 18 were for a conventional turbine modified for operation at high exhaust pressure. Their source was the General Electric Company.

The family of lines, identified by plant heat rate values ranging from 8000 to 12,000 Btu/Kw-hr, allow one to determine heat rejection rate versus increasing turbine back pressure for plants whose base plant heat rates at 2 in. Hg absolute differ from 9541 Btu/Kw-hr. It is emphasized that the plant heat rate values indicated on the nomograph are only for plant operation at a back pressure of 2 in. Hg absolute. Plant heat rates and efficiency change as back pressure increases. As an example, consider a plant whose plant heat rate at 2 in. Hg absolute is 9000 Btu/Kw-hr. If this plant is 800 Mw in capacity and is operating at 4 in. Hg absolute back pressure, a heat rejection rate of 3.5×10^9 Btu/hr results from nomograph N-27. Now, after substitution into equation (17), the plant efficiency is determined:

$$3.5 \times 10^9 \text{ Btu/hr} = 3413 \times 8 \times 10^5 \text{ Kw} \times \left[\frac{.85}{\eta_P} - 1\right]$$

or,

$$\eta_P = 37.3 \text{ percent.}$$

Then the plant heat rate $= \dfrac{3413 \text{ Btu/Kw-hr}}{.373} = 9160$ Btu/Kw-hr at 4 in. Hg absolute instead of 9000 Btu/Kw-hr at 2 in. Hg.

The range of plant heat rates displayed on N-27 is justified from statistics presented in Reference 19. In 1969 the best annual plant heat rates from coal fired plants were just above 8700 Btu/Kw-hr exhibited by the Muskingum River plant and the Marshall plant. The best plant heat rates for gas-fired plants exceeded 9500 Btu/Kw-hr. The national average of the plant heat rates for FPC regions was 10,447 Btu/Kw-hr in 1969. Two regional averages exceeded 11,000 Btu/Kw-hr.

The range of plant net generating capacities up to 10,000 Mw is displayed in order to address anticipated future large plant complexes.

In the Heat Rejection Rate nomograph on page N-28, heat rejection rates for nuclear plants are determined as a function of turbine back pressure, net generating capacity, and the base load heat rate. The development of this nomograph is also based on equation (17). The same procedure was followed here as was done in developing the fossil plant nomograph, N-27. For the nuclear plant, in-plant losses were assumed to be 5 percent. The base plant heat rate for the nuclear plant is 10,937 Btu/Kw-hr at 2 in. Hg absolute based on the Beck data (Ref. 18). The family of plant heat rates ranging from 9000 to 14,000 Btu/Kw-hr allow one to determine heat rejection rate versus increasing turbine back pressure for plants whose base plant heat rates differ from 10,937 Btu/Kw-hr at 2in. Hg absolute.

The Additional Heat Capacity nomograph, page N-29, for fossil plants allows the determination of additional heat capacity versus turbine back pressure and the net generating capacity. As back pressure increases the plant efficiency decreases. The power output (Mwe) decreases if the heat input is kept constant. An alternative, which was chosen in this effort, is to increase the heat input and maintain the power output at the rated value. In this case, a penalty must be paid for the additional heat input or additional heat capacity. The derivation for additional heat capacity follows.

The baseline is at 2 in. Hg absolute which corresponds to plant efficiencies for fossil and nuclear plants of 35.8 percent and 31.2 percent, respectively. At the baseline of 2 in. Hg absolute and efficiency, η_P, the heat input is:

$$Q_{in} = \frac{W}{\eta_P} \tag{18}$$

where:

Q_{in} = heat input, Btu/hr, at 2 in. Hg

W = heat equivalent of the rated output, Btu/hr

η_P = plant efficiency = 0.358 (fossil plant) or 0.312 (nuclear plant)

At a certain back pressure other than 2 in. Hg absolute, the plant efficiency is η_P' and the heat input is:

$$Q_{in}' = \frac{W}{\eta_P'} \tag{19}$$

where:

Q_{in}' = heat input, Btu/hr, at a certain back pressure

η_P' = plant efficiency at a certain back pressure

Finally, the additional heat capacity is:

$$\Delta Q_{in} = Q_{in}' - Q_{in} = W\left(\frac{1}{\eta_P'} - \frac{1}{\eta_P}\right) . \tag{20}$$

The Additional Heat Capacity nomograph, page N-30, for nuclear plants allows the determination of additional heat capacity versus turbine back pressure and the net generating capacity. The additional heat capacities also result from the use of equation (20) where nuclear plant heat rates versus back pressures are substituted.

The Plant Efficiency nomograph, page N-31, illustrates the change of plant efficiency for fossil and nuclear plants as the turbine back pressure increases. These curves apply only for plants whose base plant heat rates are 9541 Btu/Kw-hr (fossil)and 10,937 Btu/Kw-hr (nuclear) at 2 in. Hg absolute pressure. The plant efficiency was derived from plant heat rate data presented in Reference 18.

The Turbine Heat Rate nomograph, page N-32, shows the variation of turbine heat rate with increasing turbine back pressure. These data are from Reference 18 and apply to a conventional steam turbine generator modified for operation at high exhaust pressure. Turbine heat rate is the product of the plant heat rate and the in-plant efficiency. Nomographs N-31 and N-32 are not generally applicable to all plants.

SECTION X

MAKEUP WATER REQUIREMENTS

The nomographs for determining makeup water requirements for the various thermal pollution control systems are presented on pages N-33 through N-38 of the Appendix. Their development is discussed in the following paragraphs.

Evaporative Losses Under Design Conditions

The Cooling Pond nomograph, page N-33, yields the evaporative loss in cfs/Mw as a function of wet bulb temperature and the pond loading parameter (acre/Mw). Evaporative loss in gal/net Kw-hr was presented in Reference 20 for nuclear plants only. Thus in order to obtain this loss for fossil plants under design conditions, it can be assumed that they reject 50 percent less heat to the cooling water than the nuclear plants. Therefore the evaporative data given by Hauser and Oleson was divided by 1. 5 to obtain fossil plant evaporative losses. This assumes that the proportion of heat rejected by evaporation remains constant. Following a straightforward conversion of units, the evaporative loss in cfs/Mw results and is shown in the nomograph.

The evaporative losses from N-33 are good for rough estimates only because they are based on a special set of assumptions made in Reference 20. These assumptions are a 60 percent relative humidity, a 70 percent cloud cover, a windspeed of 8 mph and a cooling range of 20°F.

The input wet bulb temperature to N-33 can be estimated based on the design meteorological conditions for the cooling pond at the site. If dry bulb temperatures are not available, an approximation can be made by assuming DBT is equal to the design equilibrium temperature, E_n.

Then following the selection of the design relative humidity, WBT can be established. The pond loading input to N-33 is obtained from the Cooling Pond Performance nomograph, N-17.

The output of the nomograph, based on Reference 20, was compared to the evaporative losses predicted using the data of Reference 14. Evaporative losses were compared at 1 and 2 acres/Mw at a New York site location. The comparison shown in Table 4 is very good.

TABLE 4. COOLING POND EVAPORATIVE LOSS

Site Location	Pond Loading Nuclear Acre/Mw	Cooling Pond Cycle	Range ^oF	E_n ^oF	WBT ^oF	Evaporative Loss-Ref. 14 10^{-5} cfs/Kw	Evaporative Loss-Ref. 20 10^{-5} cfs/Kw
NE-New York	1	Closed	20	76	66. 3	2. 86	3. 23
NE-New York	2	Closed	20	76	66. 3	3. 7	4. 1

The input wet bulb temperature, 66. 3^oF, to N-33 was determined from a psychrometric chart after assuming 1) a 60 percent design relative humidity and 2) the air temperature or DBT was equal to E_n.

The nomograph for Evaporative Loss for Wet Systems, Page N-34, provides for the determination of evaporative loss for natural draft and mechanical draft wet towers and for spray ponds. Evaporative loss in gal/net Kw-hr was presented in Reference 20 for these wet systems as a function of range. Evaporative losses for the wet towers were presented versus approach also, in addition to range.

N-34 is based on the same assumptions for meteorological conditions as N-33. Thus it is good for rough estimates only.

Hauser and Oleson performed a sensitivity analysis whereby individual design parameters were varied to determine the effect on evaporative losses from cooling towers, spray ponds, and cooling ponds. Evaporative rates for mechanical draft and natural draft wet towers were very sensitive to the range and approach parameters. Spray ponds showed a large increase in evaporative losses for low ranges. Evaporative losses for cooling ponds were most sensitive to changes in wet bulb temperature.

Average Annual Evaporative Losses

The rate of overall water usage in actual plant applications will generally be much less than those calculated for design conditions for several reasons. Design conditions are generally selected to permit the plant to produce full power 90 to 100 percent of the time during the year. During the majority of the year, conditions will be more favorable than the design meteorological conditions. This was illustrated in a study of a 1000 Mwe fossil-fuel power plant to be located near Lake Michigan, (Ref. 15). In this case, the basic design conditions discussed above were used for the plant and the following adjustments were made to compute evaporative water loss on an average annual basis.

1. An 82 percent plant capacity factor was assumed
2. Evaporative losses for wet towers and spray ponds were reduced 20 percent to account for more favorable conditions over the year.
3. Evaporative losses from cooling ponds were reduced to account for the annual average incoming short wave radiation of 1600 Btu per square foot per day compared to a design value of about 2800 Btu per square foot per day with the difference being equivalent to 10 cfs per 1000 acres pond surface averaged over the year when adjusted for reflective losses. It should be noted again that the evaporation data provided in Reference 20 were based upon a cloud cover of 70 percent. Thus, the above "correction" for short wave radiation would not be suitable.

Therefore, the data in N-33 are assumed to be based on average rather than design short wave radiation.

In addition to the adjustments noted above, the natural evapotranspiration losses should also be considered in the case of cooling ponds. Natural evapotranspiration is the difference between the average annual precipitation rate and the average annual runoff rate in a given region. For the Lake Michigan area, the annual average natural evapotranspiration rate is about 20 inches per year. This is equivalent to a reduction of 3.5 cfs for 1500 acre ponds and 4.6 cfs for 2000 acre ponds. These data apply only to the Lake Michigan area and should not be directly applied to other regions.

In the nomograph examples for water requirements found in Section XII, appropriate factors are used to estimate the average annual evaporative losses from the design values obtained from the nomographs.

Percent Evaporative Loss, Blowdown and Makeup Water Nomographs

In the Percent Evaporative Loss nomograph, page N-35, the evaporative loss is determined as a percentage of the water flow rate. Initially, the nomograph is entered with the evaporative loss input in cfs/Mw from either N-34 or N-35. Then this evaporative loss parameter is multiplied by the net generating capacity (Mw) for a given plant in order to yield evaporative loss in cfs. The input water flow rate, which may be in gpm, is taken. Before entering the nomograph it is converted to cfs by the multiplication factor, 2.23×10^{-3} cfs per gpm. Finally, the evaporative loss is:

$$\text{evaporative loss, \%} = \frac{\text{evaporative loss, cfs}}{\text{water flow rate, cfs}} \times 100 \qquad (21)$$

Nomographs on pages N-36 and N-37 are identified as Percent Blowdown. Blowdown is determined as a percentage of the water flow rate. As water is lost by evaporation from the cooling water supply of wet cooling systems, nonevaporating substances are concentrated in the cooling water remaining. There is a practical limit of concentration of the substances if corrosion, fouling and general deterioration of the cooling structures are to be prevented. To avoid such problems, a certain amount of the cooling water customarily is drained off from the system for disposal. This water, termed blowdown, is replaced by fresh makeup water.

The equationfor concentration factor or number of concentrations may be given as:

$$C = \frac{E + D + B}{D + B} \tag{22}$$

where:

C	=	factor of concentration in the evaporative system
E	=	evaporative loss or evaporation, percent
D	=	drift loss, percent
B	=	blowdown, percent

The variables, E, D, and B, are all percentages of the water flow rate. This equation was rearranged to yield the governing equation for nomographs N-36 and N-37:

$$B = \frac{E - D(C - 1)}{C - 1} \tag{23}$$

Nomographs N-36 and N-37 apply only to natural draft and mechanical draft wet towers and the spray ponds. In the first nomograph, a constant drift loss of 0.005 percent is assumed. In N-37, blowdown is determined at two other constant levels of drift, 0.001 percent and 0.01 percent. Drift is the loss of entrained water that is carried out of the top of a wet tower or from a spray pond. The typical drift guarantee of 0.2 percent of the water flow rate is far in excess of current engineering capability and practicality for large towers. Drift loss can be almost completely eliminated by control of air velocity and use of drift eliminators. Mechanical draft wet towers can be currently purchased with certification of drift elimination at the 0.02 percent level.

Natural draft towers can be certified at the 0.002 percent level. Also drift measurements on operating towers show as low as 0.005 percent for mechanical draft and 0.001 percent for natural draft towers. The percent evaporation is an input from N-35. The number (or cycles) of concentrations and the drift loss are chosen for a given application. The blowdown parameter then follows from the nomographs. Note that the blowdown values from the nomographs for the three drift levels are essentially equal for cycles of concentration less than about 10.

The Makeup Water nomograph, page N-38, yields the makeup water requirements for the wet systems in either cfs or gpm. They are assumed to be negligible for the dry cooling towers. The input parameter to this nomograph is the makeup water requirements as a percentage of water flow rate. For cooling ponds this input is assumed to be the percent evaporative loss, E, from Nomograph N-35. However, it should also include precipitation and runoff as well as blowdown, if applicable. For the wet towers and spray ponds, makeup water requirements are:

$$\text{makeup water requirements, percent of water flow rate} = \text{evaporation} + \text{drift} + \text{blowdown} = E + D + B \qquad (24)$$

Then, multiplication of the input parameter by the water flow rate in cfs yields the makeup water requirements in cfs.

Makeup water requirements are determined for two illustrative examples representing a natural draft wet tower and a cooling pond in Section XII.

SECTION XI

COSTS FOR THE HEAT REJECTION SYSTEMS

Discussion of the Nomographs

In the cost nomographs, incremental costs are determined as being the various additional capital and operating costs required to incorporate and operate the given heat rejection system within the power plant. These nomographs are presented on pages N-39 through N-49 of the Appendix. Their development is discussed in the following paragraphs.

The Steam Supply Incremental Cost nomograph, page N-39, estimates the incremental cost for the steam supply system of either fossil or nuclear plants due to the presence of a given heat rejection system. This cost is based on the following approximate equation:

steam supply system incremental cost (SSIC), dollars = (25)

$$\frac{\Delta Q}{3413 \text{ Btu/Kw-hr}} \times K_1 \times K_2 \times \text{plant unit costs, \$/Kw}$$

where:

ΔQ = additional heat capacity in Btu/hr. This is an input from the turbine performance nomographs corresponding to a given cooling system in either a fossil or nuclear plant

K_1 = 0.358, efficiency of a fossil plant or 0.312, efficiency of a nuclear plant (in reality, this varies with turbine back pressure.)

K_2 = Ratio of steam supply system cost to total plant construction cost

The breakdown of costs for power plants shows that approximately 37 percent of the total construction cost is for the steam supply system (References 21, 22, and 23). Therefore, K_2 is assumed equal to 0.37. The ranges of unit costs, shown on N-39, for fossil and nuclear plants are from Reference 24, one of many potential sources.

The Turbine Generator Incremental Cost nomograph, page N-40, estimates the incremental cost for the turbine generator plant (TGIC) of either fossil or nuclear plants. This cost is also determined from equation (25) except that K_2 is replaced by K_3. K_3 is defined as being the ratio of turbine generator plant cost to the total plant construction cost. In Reference 23, the cost breakdown for the power plant showed that approximately 21 percent of the total cost was for the turbine generator plant (condenser system excluded). Therefore, K_3 is assumed a constant equal to 0. 21.

These nomographs have limited application. The wet towers, spray ponds and cooling ponds are assumed to incur no cost penalty for capability losses unless the turbine back pressures exceed values beyond the range of roughly 3. 5 in. Hg absolute. However, the nomographs are used to determine SSIC and TGIC for dry towers operating at high back pressures. The cost penalty would be computed and apply to the range of back pressure from 3. 5 in. Hg absolute to the high back pressure value of the plant using a dry tower. The procedure is shown in the dry tower nomograph examples in Section XII. It should also be pointed out that the capability loss can be possibly made up with gas turbine peaking units at a cost of \$100/Kw, as an alternative to increasing the base load. Nomographs N-39 and N-40 can also serve as auxiliary nomographs to determine costs of capability loss for derated plant cases. There will be a certain time of year when a plant won't have a loss of capability. Thus, the capability loss under design conditions is conservative.

The Condenser Surface Area nomograph, page N-41, yields the area of the heat transfer surface for the condenser. Two governing equations for condensers are:

$$q = WFR \times C_p \, (T_o - T_i) \qquad (26)$$

and

$$\frac{q}{UA_c} = \frac{T_o - T_i}{\ln \left\{ \frac{T_s - T_i}{T_s - T_o} \right\}} \qquad (27)$$

where:

q = total heat transferred in the condenser, Btu/hr

WFR = water flow rate, lb/hr

U = over-all heat transfer coefficient, Btu/hr-ft^2-^oF

A_c = area of heat transfer surface, ft^2

T_o = water outlet temperature from condenser, ^oF

T_i = water inlet temperature to condenser, ^oF

T_s = steam saturation temperature, ^oF

C_p = specific heat for water = 1 Btu/lb-^oF

Following the combination of equations (26) and (27) the following equation for A_c is obtained:

$$A_c = \left(\frac{WFR}{U}\right) \ell n \left\{\frac{T_s - T_i}{T_s - T_o}\right\} = \left(\frac{WFR}{U}\right) \ell n \left\{\frac{\text{condenser rise} + TTD}{TTD}\right\} \quad (28)$$

If the terminal temperature difference is assumed to be 6^oF and the water flow rate is in gpm, the final equation for N-41 is:

$$A_c = 500\left(\frac{WFR}{U}\right) \ell n \left\{\frac{\text{condenser rise} + 6}{6}\right\} \quad (29)$$

The range of U displayed in the nomograph is partially based on the use of 630 Btu/hr-ft^2-^oF in Reference 4.

The Condenser Cost nomograph, page N-42, estimates capital cost of condensers employing several types of tube materials, including 90–10 copper-nickel and admiralty. Based on Westinghouse price lists (Ref. 25), it was determined that the equipment (or material) unit cost per surface condensers was \$7.70/ft^2 for 90-10 copper-nickel. The cost includes surface condensers, auxiliaries, and air ejectors. From Reference 26, the multiplication factor for obtaining field installation cost is found to be approximately 3. Thus, the condenser capital cost including field installation is \$23.10/ft^2 for 90-10 copper-nickel. The corresponding cost for condensers with admiralty tube materials isn't too far different at $\$22/ft^2$. Since admiralty is the most popular tube material a cost line for it appears in N-42. Two other cost lines, $\$15/ft^2$ and $\$27/ft^2$, are displayed in order to cover the range of condenser tube materials.

The Water Circulation System Cost nomograph, page N-43, estimates the capital costs for pumps, motors, piping and installation. A cost equation was derived comprised of contributions from each component on a cost per gpm basis. The pump cost is estimated as \$1.05/gpm at 100 ft totalhead based on Reference 27. In the estimate, the capacity per pump was 200,000 gpm. The motor cost, based on the same reference, is estimated to be \$32/hp. The horsepower required per gpm was computed to be 0.0301 from an equation similar to equation (6) assuming 100 ft total head. Cost for piping was estimated as \$0.25/gpm (Ref. 27 and 28) based on a length of piping of 1000 ft. Cost of installation was estimated to be \$0.4/gpm based on one million gpm. Then the governing cost equation for N-43 takes the following form:

$$\text{water circulation system cost (WCSC), dollars} = \text{water flow rate, gpm} \times \left(1.05 \times \frac{\text{total head, ft}}{100} + 32 \times 0.0301 \times \frac{\text{total head, ft}}{100} + 0.25 + 0.4\right) \qquad (30)$$

The nomograph applies only to the wet towers, spray ponds, and cooling ponds.

The Cost for Once-Through Cooling nomograph, page N-44, estimates the intake and outfall structure cost based on FPC Form 67 (Ref. 6) data for 13 existing power plants. The average cost for intake and outfall structure was \$5.80/gpm/1000 ft for both once-through fresh water and salt water systems. As defined in Reference 6, the costs for these once-through systems include pumps, piping, canals, ducts, intake and discharge structures. Therefore, the cost equation for N-44 is:

cost for once-through cooling (OTCC), dollars = 5.8 x

$$\text{water flow rate, gpm} \times \frac{\text{total structural length, ft}}{1000} \qquad (31)$$

The total structural length is from and to the condenser.

In the Capital Recovery Factor nomograph, page N-45, the capital recovery factor is a function of interest rate and years amortized. The curves in the nomograph are from data in interest tables from Reference 29. Then the fixed charge rate in percent per year can be determined by adding the capital recovery factor in percent from the nomograph to the recurring cost percentage. The recurring cost percentage includes insurance, interim replacements, and various taxes such as property, state and federal. In Reference 30, representative values for recurring cost percentage are given as 5.13 percent for fossil-fueled plants and 5.68 percent for nuclear plants. Thus, the fixed charge rate depends upon the economic conditions at a particular site locale. Private financing is assumed. Federal financing results in lower interest rates and no taxes.

In the Annualized Capital Cost nomograph, page N-46, annualized, incremental capital cost in dollars per year is obtained from the product of the fixed charge rate and the total incremental capital cost (TICC). The input to this nomograph, TICC, will be defined since it depends upon the type of thermal pollution control system. In general, the total incremental capital cost, TICC, is equal to the heat rejection system cost (HRSC) for a given method, plus the component costs from the cost nomographs.

For wet towers, spray ponds, and cooling ponds:

$$\text{TICC} = \text{HRSC} + \text{CC} + \text{WCSC} \qquad (32)$$

For these systems, HRSC is the capital cost of a natural draft wet tower, a mechanical draft wet tower, a spray pond, or a cooling pond from previous nomographs. The CC and WCSC are input capital costs defined on the respective cost nomographs in this section.

For the dry towers:

$$\text{TICC} = \text{HRSC} + \text{SSIC} + \text{TGIC} \tag{33}$$

The heat rejection system cost, HRSC, for the dry towers already includes cost for the water circulation system and condenser. The procedure for obtaining SSIC and TGIC is given in the dry tower examples of Section XII.

For once-through cooling:

$$\text{TICC} = \text{OTCC} + \text{CC} \tag{34}$$

The cost for once-through cooling, OTCC, includes the water circulation system.

The Additional Annual Power Cost Nomograph, page N-47, determines the cost which results from the auxiliary power requirements. First, the additional annual energy consumption is obtained from the following equation:

$$\text{additional annual energy consumption, Kw-hr/yr} = \text{auxiliary power requirements, Mw} \times \text{plant load factor} \times 8.76 \times 10^6 \tag{35}$$

Finally, the additional annual power cost, in dollars per year, is obtained from the product of the additional annual energy consumption and the unit cost for electrical power.

The Additional Annual Fuel Cost Nomograph, page N-48, is based on the equation:

$$\text{additional annual fuel cost, \$/yr} = \text{additional heat capacity, } 10^6 \text{ Btu/hr} \times \text{plant load factor} \times \text{fuel cost, \$/}10^6 \text{ Btu} \times 8760 \tag{36}$$

Fossil and nuclear fuel cost variations are covered by the range of fuel costs presented in the nomograph. The cost is for the additional fuel required from the increase of steam rate to maintain rated output. Additional fuel cost is incurred for plant operation above 2 in. Hg absolute.

The last nomograph, page N-49, is entitled Incremental Generating Cost. The input to this nomograph is the resulting summation of costs from the last three cost nomographs:

$$\text{total incremental annual cost, \$/yr} = \text{annualized capital cost} + \text{additional annual power cost} + \text{additional annual fuel cost} \tag{37}$$

Then the incremental generating cost results from the equation:

$$\text{incremental generating cost, mills/Kw-hr} = \frac{\text{total incremental annual cost, \$/yr}}{8760 \times \text{plant load factor} \times \text{net generating capacity, Mw}} \tag{38}$$

It should be emphasized that the costs obtained from the last three nomographs, N-47, N-48, and N-49, are based on design conditions. The true annual costs for additional power and additional fuel based on off-design conditions would be less than those costs obtained from N-47 and N-48.

SECTION XII

NOMOGRAPH EXAMPLES

General

Illustrative cases are presented which consider an 800 Mw fossil-fueled plant utilizing each of the types of thermal pollution control systems. The purpose of the examples is an instructive one with respect to their use. These results should not be used alone to evaluate the merits of one system versus another. In some cases, results of interest are pointed out and compared with other sources. Each example is solved through as far as application of the turbine performance nomographs. Only the example for the natural draft wet tower is carried through completely and displayed by the example lines in the turbine performance, water requirements, and cost nomographs.

Natural Draft Wet Tower

In Table 5, most of the input conditions correspond to the design conditions of Big Sandy, Unit No. 2. The tower cost, calculated in the table to be $4.9 million, compares well with that estimated by Woodson in Reference 31, $4.2 million (based on 1970 prices). However, no other assumptions from Reference 31 are known except for the application to an 800 Mw fossil-fueled plant.

Mechanical Draft Wet Tower

In Tables 6 and 7, examples are shown for the mechanical draft wet tower assuming closed cycle and open cycle modes of operation, respectively. Nomographs N-7 through N-12 show only the example lines for the closed cycle case.

The mechanical draft tower cost of Table 6 is $\$1.55 \times 10^6$ and excludes cold water basins, fan motor starters, controls, and wiring. It can be compared to capital costs from other sources. For example, Hauser,

TABLE 5. NOMOGRAPH EXAMPLE - NATURAL DRAFT WET TOWER

(sheet 1 of 2)

Net generating capacity, Mw		800
Type of plant		Fossil
Plant heat rate, Btu/Kw-hr		9050
Plant load factor, percent		60
Dry bulb temperature, ^{o}F		77
Wet bulb temperature, ^{o}F		70
Relative humidity, percent		71
Range, ^{o}F		28.7
A. Tower Performance (N-1)		
Base diameter, ft		300
Dry bulb temperature, ^{o}F	(enter)	77
Relative humidity, percent	(enter)	71
Tower height, ft	(enter)	400
Cold water temperature, ^{o}F	(read)	88.5
B. Turbine Back Pressure (N-2)		
Cold water temperature, ^{o}F	(enter)	88.5
Condenser rise, ^{o}F	(enter)	28.7
Terminal temperature difference, ^{o}F		6
Turbine back pressure, in.Hg abs	(read)	3.8
C. Tower Unit Cost (N-3)		
Wet bulb temperature, ^{o}F	(enter)	70
Range, ^{o}F	(enter)	28.7
Approach, ^{o}F	(enter)	18.5
Tower unit cost, \$/1000 Btu/hr	(read)	1.4
D. Tower Cost (N-4)		
Heat rejection rate, 10^9 Btu/hr (from Turbine Performance Nomographs at above plant capacity and type, heat rate, and turbine back pressure)	(enter)	3.5
Tower unit cost, \$/1000 Btu/hr	(enter)	1.4
Natural draft tower cost, $\$10^6$	(read)	4.9
E. Flow Rate (N-5)		
Heat rejection rate, 10^9 Btu/hr	(enter)	3.5
Condenser rise, ^{o}F	(enter)	28.7
Water flow rate, 10^6 gpm	(read)	0.25

F. Auxiliary Power (N-6)

Water flow rate, 10^6 gpm	(enter)	0.25
Total head, ft	(enter)	50
Pump power requirements, Mw	(read)	2.8

G. Plant Efficiency (N-31)

Turbine back pressure, in.Hg abs	(enter)	3.8
Plant type	(enter)	Fossil
Plant efficiency, percent	(read)	35.2

H. Additional Heat Capacity (N-29)

Turbine back pressure, in.Hg abs	(enter)	3.8
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Additional heat capacity, 10^9 Btu/hr	(read)	0.115

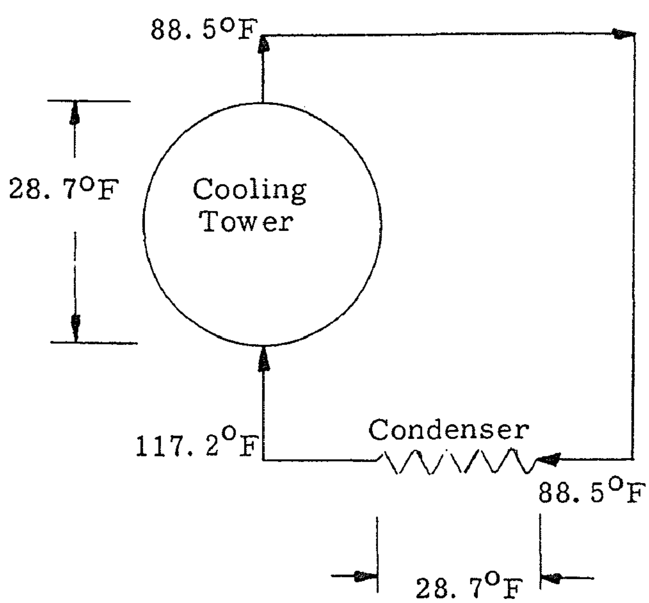


TABLE 6. NOMOGRAPH EXAMPLE - MECHANICAL DRAFT WET TOWER (CLOSED CYCLE)

Net generating capacity, Mw		800
Type of plant		Fossil
Plant heat rate, Btu/Kw-hr		9540
Plant load factor, percent		60
Dry bulb temperature, ^oF		
Wet bulb temperature, ^oF		70
Relative humidity, percent		
Range, ^oF		25
Approach, ^oF		15
A. Tower Correction Factor (N-7)		
Wet bulb temperature, ^oF	(enter)	70
Range, ^oF	(enter)	25
Approach, ^oF	(enter)	15
Tower correction factor	(read)	1.04
B. Turbine Back Pressure (N-8)		
Wet bulb temperature, ^oF	(enter)	70
Approach, ^oF	(enter)	15
Condenser Inlet temperature, ^oF	(read)	85
Condenser Rise, ^oF	(enter)	25
Terminal temperature difference, ^oF		6
Turbine back pressure, in. Hg abs	(read)	3.1
C. Flow Rate (N-9)		
Heat rejection rate, 10^9 Btu/hr (from Turbine Performance Nomographs at above plant capacity and type, heat rate and turbine back pressure)	(enter)	3.9
Condenser rise, ^oF	(enter)	25
Water flow rate, 10^6 gpm	(read)	0.32
D. Tower Cost (N-10)		
Water flow rate, 10^6 gpm	(enter)	0.32
Tower correction factor	(enter)	1.04
Number of tower units, 10^6 TU	(read)	0.33
Cost per tower unit, \$/TU	(enter)	4.75
Mechanical draft tower cost, \$$10^6$	(read)	1.55

E. Fan Power (N-11)

Number of tower units, 10^6 TU	(enter)	0.33
Fan power requirements, Mw	(read)	2.7

F. Pump Power (N-12)

Type of operation	(enter)	Closed
Water flow rate, 10^6 gpm	(enter)	0.32
Total pumping head, ft	(enter)	60 (constant)
Pump power requirement, Mw	(read)	4.2
Auxiliary power, Mw		6.9

G. Plant Efficiency (N-31)

Turbine back pressure, in. Hg abs	(enter)	3.1
Plant type	(enter)	Fossil
Plant efficiency, percent	(read)	35.5

H. Additional Heat Capacity (N-29)

Turbine back pressure, in. Hg abs	(enter)	3.1
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Additional heat capacity, 10^9 Btu/hr	(read)	0.058

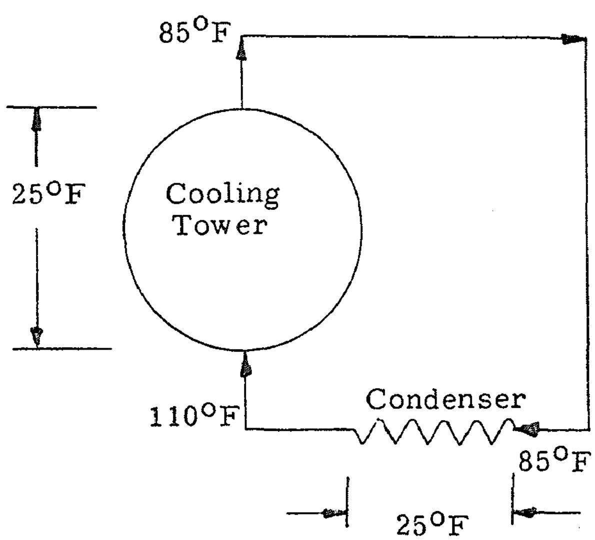


TABLE 7. NOMOGRAPH EXAMPLE - (Sheet 1 of 2)
MECHANICAL DRAFT WET TOWER
(OPEN CYCLE)

Net generating capacity, Mw		800
Type of plant		Fossil
Plant heat rate, Btu/Kw-hr		9540
Plant load factor, percent		60
Condenser inlet temperature, ^{o}F		80
Wet bulb temperature, ^{o}F		70
Relative humidity, percent		
Range, ^{o}F		25
Approach, ^{o}F		15
A. Tower Correction Factor (N-7)		
Wet bulb temperature, ^{o}F	(enter)	70
Range, ^{o}F	(enter)	25
Approach, ^{o}F	(enter)	15
Tower correction factor	(read)	1.04
B. Turbine Back Pressure (N-8)		
Condenser inlet temperature, ^{o}F	(enter)	80
Condenser rise, ^{o}F	(enter)	30
Terminal temperature difference, ^{o}F		6
Turbine back pressure, in. Hg abs	(read)	3.1
C. Flow Rate (N-9)		
Heat rejection rate, 10^9 Btu/hr (from Turbine Performance Nomographs at above plant capacity and type, heat rate and turbine back pressure)	(enter)	3.9
Condenser Rise (Range $+\Delta$ temp. of $5^{o}F$), ^{o}F	(enter)	30
Water flow rate, 10^6 gpm	(read)	0.26
D. Tower Cost (N-10)		
Water flow rate, 10^6 gpm	(enter)	0.26
Tower correction factor	(enter)	1.04
Number of tower units, 10^6 TU	(read)	0.27
Cost per tower unit, \$/TU	(enter)	4.75
Mechanical draft tower cost, \$$10^6$	(read)	1.3

E. Fan Power (N-11)

Number of tower units, 10^6 TU	(enter)	0.27
Fan power requirements, Mw	(read)	2.25

F. Pump Power (N-12)

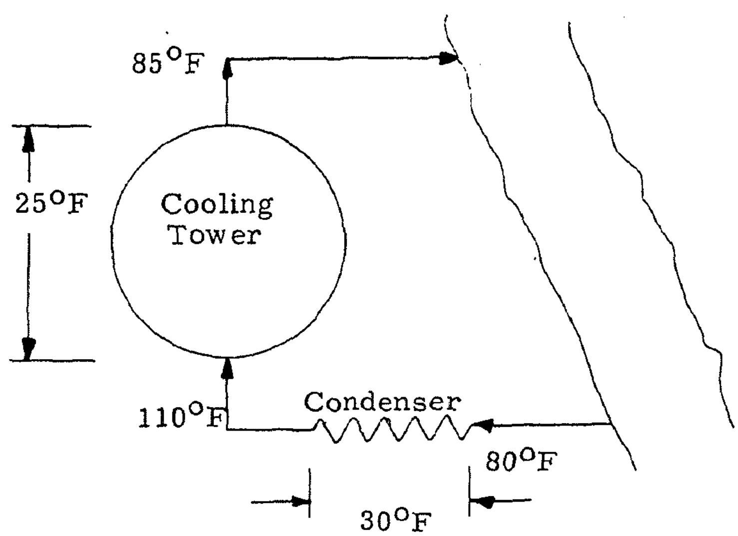
Type of operation	(enter)	Open
Water flow rate, 10^6 gpm	(enter)	0.26
Total pumping head, ft	(enter)	120
Pump power requirements, Mw	(read)	6.9
Auxiliary power, Mw		9.15

G. Plant Efficiency (N-31)

Turbine back pressure, in. Hg abs	(enter)	3.1
Plant type	(enter)	Fossil
Plant efficiency, percent	(read)	35.5

H. Additional Heat Capacity (N-29)

Turbine back pressure, in. Hg abs	(enter)	3.1
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Additional heat capacity, 10^9 Btu/hr	(read)	0.058



Oleson, and Budenholzer (Ref. 32) present curves which yield an estimated cost of $\$1.9 \times 10^6$ for the range, approach, and plant size assumed in our example. Their tower cost should be higher since it includes the basin and electrical apparatus. Following the procedure in a series of curves presented by Dynatech Co. (Ref. 9), one obtains a cooling tower cost of $\$2.5 \times 10^6$ for the conditions in Table 6. However, they pointed out that the costs from their curves appear to be 50 percent high when compared to other data. Accounting for this fact and an escalation factor from 1969 to 1971, a corrected cost from Reference 9 of $\$2 \times 10^6$ is obtained. Woodson (Ref. 31) indicates a capital cost of $\$1.7 \times 10^6$ for mechanical draft wet towers and basin in an 800 Mw fossil-fueled plant. Other design conditions employed to obtain this result are unknown. As a result of these comparisons, it is concluded that the nomographs yield good capital cost estimates for mechanical draft towers. Based on these costs, an approximate 15 percent cost increment should be added to the value from N-10 in order to include the basins and electrical apparatus.

Spray Pond

Closed cycle and open cycle examples for the spray pond are presented in Tables 8 and 9, respectively. Nomographs N-13 through N-16 show only the example lines for the closed cycle case.

Cooling Pond

Closed cycle and open cycle examples for the cooling pond are presented in Tables 10 and 11. Nomographs N-17 through N-21 show only the example lines for the closed cycle case.

In Reference 31, Woodson reported a cooling lake cost of \$2.6 million for an 800 Mw fossil-fueled plant. This shows reasonable comparison with the \$2.2 million value for the closed cycle case in Table 10. The value from the table is based on a land and excavation unit cost of \$2000 per acre.

TABLE 8. NOMOGRAPH EXAMPLE - SPRAY POND (CLOSED CYCLE)

Net generating capacity, Mw		800
Type of plant		Fossil
Plant heat rate, Btu/Kw-hr		9540
Plant load factor, percent		60
Wet bulb temperature, ^{o}F		70
Condenser rise, ^{o}F		20
Inlet water temperature, ^{o}F		110
Wind velocity, mph		5
A. Spray Pond Performance (N-13)		
Inlet water temperature, ^{o}F	(enter)	110
Wet bulb temperature, ^{o}F	(enter)	70
Wind velocity, mph	(enter)	5
Pond water temperature, ^{o}F	(read)	90
B. Turbine Back Pressure (N-14)		
Condenser inlet temperature, ^{o}F	(enter)	90
Condenser rise, ^{o}F	(enter)	20
Terminal temperature difference, ^{o}F		6
Turbine back pressure, in. Hg abs	(read)	3. 1
C. Flow Rate (N-15)		
Heat rejection rate, 10^9 Btu/hr (from Turbine Performance Nomographs at above plant capacity and type, heat rate and turbine back pressure)	(enter)	4
Condenser rise, ^{o}F	(enter)	20
Water flow rate, 10^6 gpm	(read)	0. 4
D. Spray Pond Cost (N-16)		
Water flow rate, 10^6 gpm	(enter)	0. 4
Pond unit cost, $/gpm	(enter)	9
Spray pond capital cost, $\$10^6$	(read)	3. 6
E. Auxiliary Power (N-16)		
Water flow rate, 10^6 gpm	(enter)	0. 4
Power consumption, Kw/gpm	(enter)	0. 025
Auxiliary power requirements, Mw	(read)	10

TABLE 8 (Continued) (sheet 2 of 2)

F. Plant Efficiency (N-31)

Turbine back pressure, in. Hg abs	(enter)	3.1
Plant type	(enter)	Fossil
Plant efficiency, percent	(read)	35.5

G. Additional Heat Capacity (N-29)

Turbine back pressure, in. Hg abs	(enter)	3.1
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Additional heat capacity, 10^9 Btu/hr	(read)	0.059

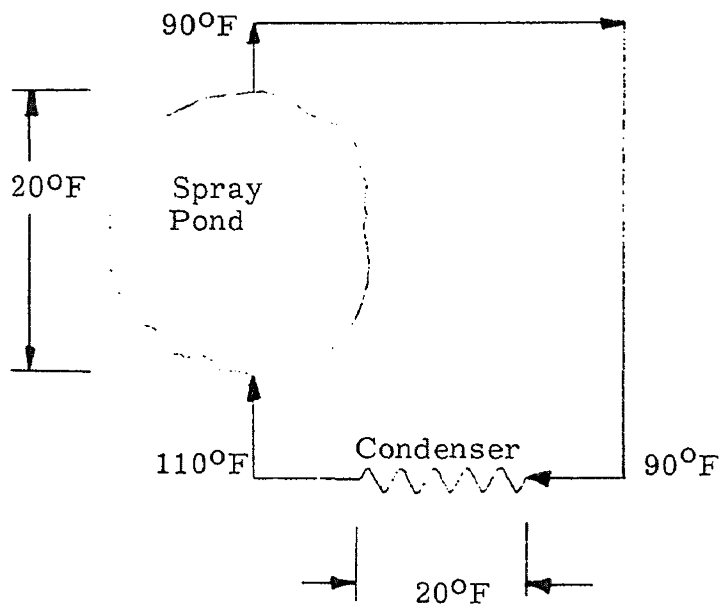


TABLE 9. NOMOGRAPH EXAMPLE - SPRAY POND (OPEN CYCLE)

Net generating capacity, Mw		800
Type of plant		Fossil
Plant heat rate, Btu/Kw-hr		9540
Plant load factor, percent		60
Wet bulb temperature, ^{o}F		70
Range, ^{o}F		20
Inlet water temperature, ^{o}F		110
Wind velocity, mph		5
A. <u>Spray Pond Performance</u> (N-13)		
Inlet water temperature, ^{o}F	(enter)	110
Wet bulb temperature, ^{o}F	(enter)	70
Wind velocity, mph	(enter)	5
Pond water temperature, ^{o}F	(read)	90
B. <u>Turbine Back Pressure</u> (N-14)		
Condenser inlet temperature, ^{o}F	(enter)	85
Condenser rise, ^{o}F	(enter)	25
Terminal temperature difference, ^{o}F		6
Turbine back pressure, in. Hg abs	(read)	3.1
C. <u>Flow Rate</u> (N-15)		
Heat rejection rate, 10^9 Btu/hr (from Turbine Performance Nomographs at above plant capacity and type, heat rate and turbine back pressure)	(enter)	4
Condenser rise (range + Δ temp. of $5^{o}F$), ^{o}F	(enter)	25
Water flow rate, 10^6 gpm	(read)	0.32
D. <u>Spray Pond Cost</u> (N-16)		
Water flow rate, 10^6 gpm	(enter)	0.32
Pond unit cost, $/gpm	(enter)	9
Spray pond capital cost, $\$10^6$	(read)	2.9
E. <u>Auxiliary Power</u> (N-16)		
Water flow rate, 10^6 gpm	(enter)	0.32
Power consumption, Kw/gpm	(enter)	0.025
Auxiliary power requirements, Mw	(read)	8

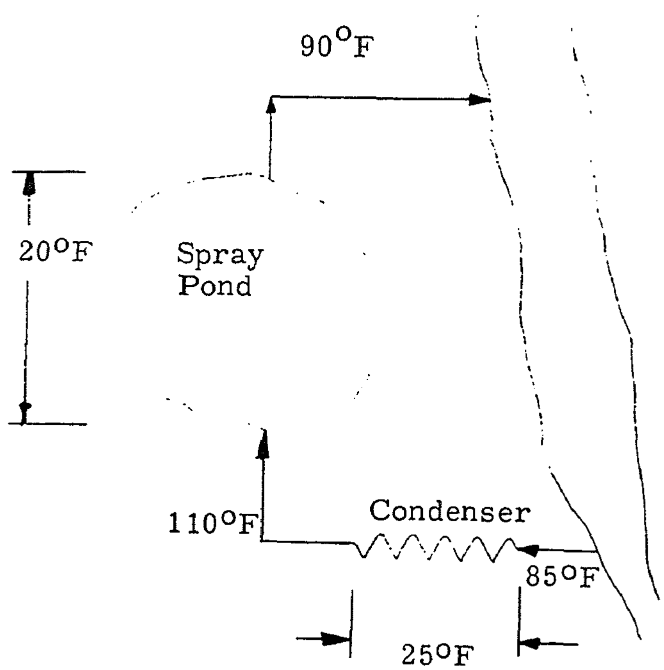
TABLE 9 (Continued)

F. Plant Efficiency (N-31)

Turbine back pressure, in. Hg abs	(enter)	3.1
Plant type	(enter)	Fossil
Plant efficiency, percent	(read)	35.5

G. Additional Heat Capacity (N-29)

Turbine back pressure, in.Hg abs	(enter)	3.1
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Additional heat capacity, 10^9 Btu/hr	(read)	0.059



(sheet 1 of 2)

TABLE 10. NOMOGRAPH EXAMPLE - COOLING POND (CLOSED CYCLE)

Net generating capacity, Mw	800
Type of plant	Fossil
Plant heat rate, Btu/Kw-hr	9540
Plant load factor, percent	60
Heat exchange coefficient, Btu/day-ft^2-^{o}F	150
Condenser rise, ^{o}F	20
Residual temperature rise, ^{o}F	5
Design equilibrium temperature, ^{o}F	80.6

A. Pond Performance (N-17)

Condenser rise, ^{o}F	(enter)	20
Residual temperature rise, ^{o}F	(enter)	5
Heat exchange coefficient, Btu/day-ft^2-^{o}F	(enter)	150
Plant type	(enter)	Fossil
Pond loading, Mw/acre	(read)	0.73

B. Turbine Back Pressure (N-18)

Condenser inlet temperature, ^{o}F	(enter)	85.6
Condenser rise, ^{o}F	(enter)	20
Terminal temperature difference, ^{o}F		6
Turbine back pressure, in. Hg abs	(read)	2.72

C. Cooling Pond Cost (N-19)

Pond loading, Mw/acre	(enter)	0.73
Plant capacity, Mw	(enter)	800
Pond size, acres	(read)	1100
Land and excavation unit cost, \$/acre	(enter)	2000
Pond land and excavation cost, \$$10^6$	(read)	2.2

D. Flow Rate (N-20)

Heat rejection rate, 10^9 Btu/hr (from Turbine Performance Nomographs at above plant capacity and type, heat rate and turbine back pressure)	(enter)	3.9
Condenser rise, ^{o}F	(enter)	20
Water flow rate, 10^6 gpm	(read)	0.39

TABLE 10(Continued) (Sheet 2 of 2)

E. Auxiliary Power (N-21)

Water flow rate, 10^6 gpm	(enter)	0.39
Total head, ft	(enter)	20
Pump power requirements, Mw	(read)	1.75

F. Plant Efficiency (N-31)

Turbine back pressure, in. Hg abs	(enter)	2.72
Plant type	(enter)	Fossil
Plant efficiency, percent	(read)	35.6

G. Additional Heat Capacity (N-29)

Turbine back pressure, in.Hg abs	(enter)	2.72
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Additional heat capacity, 10^9 Btu/hr	(read)	0.031

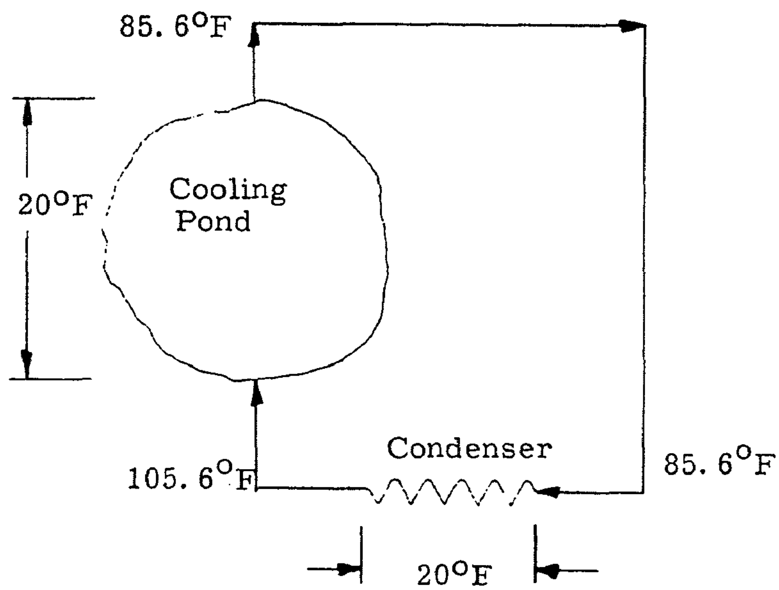


TABLE 11. NOMOGRAPH EXAMPLE - COOLING POND (OPEN CYCLE)

Net generating capacity, Mw	800
Type of plant	Fossil
Plant heat rate, Btu/Kw-hr	9540
Plant load factor, percent	60
Heat exchange coefficient, Btu/day-ft^2-^{o}F	150
Condenser rise, ^{o}F	25
Residual temperature rise, ^{o}F	4
Design equilibrium temperature, ^{o}F	84.6
Condenser inlet temperature, ^{o}F	83.6

A. Pond Performance (N-17)

ΔT_c, ^{o}F	(enter)	20
Residual temperature rise, ^{o}F	(enter)	4
Heat exchange coefficient, Btu/day-ft^2-^{o}F	(enter)	150
Plant type	(enter)	Fossil
Pond loading, Mw/acre	(read)	0.64

B. Turbine Back Pressure (N-18)

Condenser inlet temperature, ^{o}F	(enter)	83.6
Condenser rise, ^{o}F	(enter)	25
Terminal temperature difference, ^{o}F		6
Turbine back pressure, in. Hg abs	(read)	2.96

C. Cooling Pond Cost (N-19)

Pond loading, Mw/acre	(enter)	0.64
Plant capacity, Mw	(enter)	800
Pond size, acres	(read)	1250
Land and excavation unit cost, $/acre	(enter)	2000
Pond land and excavation cost, $10^6	(read)	2.5

D. Flow Rate (N-20)

Heat rejection rate, 10^9 Btu/hr (from Turbine Performance Nomographs at above plant capacity and type, heat rate and turbine back pressure)	(enter)	3.9
Condenser rise, ^{o}F	(enter)	25
Water flow rate, 10^6 gpm	(read)	0.31

TABLE 11 (Continued)

E. Auxiliary Power (N-21)

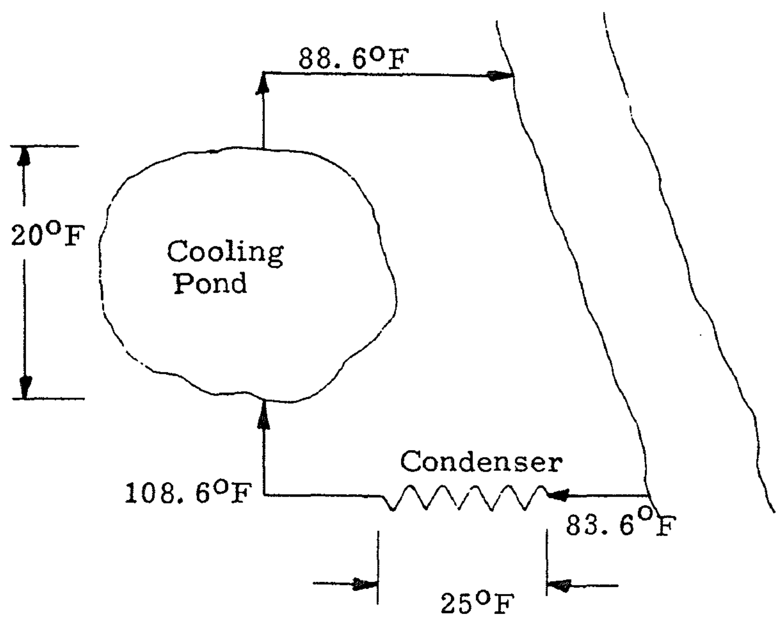
Water flow rate, 10^6 gpm	(enter)	0.31
Total head, ft	(enter)	50
Pump power requirements, Mw	(read)	3.5

F. Plant Efficiency (N-31)

Turbine back pressure, in.Hg abs	(enter)	2.96
Plant type	(enter)	Fossil
Plant efficiency, percent	(read)	35.6

G. Additional Heat Capacity (N-29)

Turbine back pressure, in.Hg abs	(enter)	2.96
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Additional heat capacity, 10^9 Btu/hr	(read)	0.048



Natural and Mechanical Draft Dry Towers

The illustrative examples for the natural and mechanical draft dry owers are presented in Tables 12 and 13, respectively. The example ines for each of these cases appear on Nomographs N-22 and N-26.

The presence of the dry towers causes this plant to operate at a turbine back pressure of 6 in. Hg absolute. The cost penalty for capability oss is computed for each case to be approximately \$3.3 x 10^6. This s based on the requirement for supplementary additional heat capacity between 3.5 in. Hg absolute and 6 in. Hg absolute. It should be observed that this cost penalty could be reduced if peaking units were available.

n the nomographs, it is assumed that these plants utilize a conventional turbine modified for operation at high back pressure. In the uture, plants employing dry towers will be designed to utilize a high back pressure turbine whose performance and costs are different. However, for the present the nomographs may be used for dry towers n new plants.

TABLE 12. NOMOGRAPH EXAMPLE - NATURAL DRAFT DRY TOWER

Net generating capacity, Mw	800
Type of plant	Fossil
Plant heat rate, Btu/Kw-hr	9540
Plant load factor, percent	60
Dry bulb temperature, $^{\circ}F$	90

A. Combined Performance (N-22)

Dry bulb temperature, $^{\circ}F$	(enter)	90
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Heat rejection, 10^9 Btu/hr	(read)	4.05
Turbine back pressure, in. Hg abs	(read)	6

B. Initial Temperature Difference (N-24)

Turbine back pressure, in. Hg abs	(enter)	6
Dry bulb temperature, $^{\circ}F$	(enter)	90
Initial temperature difference, $^{\circ}F$	(read)	50

C. Capital Cost (N-25)

Initial temperature difference, $^{\circ}F$	(enter)	50
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Heat rejection system cost, $\$10^6$	(read)	19.5

D. Auxiliary Power (N-26)

Initial temperature difference, $^{\circ}F$	(enter)	50
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Auxiliary power requirements, Mw	(read)	7.1

E. Plant Efficiency (N-31)

Turbine back pressure, in. Hg abs	(enter)	6
Plant efficiency, percent	(read)	34.1

F. Additional Heat Capacity (N-29)

Turbine back pressure, in. Hg abs	(enter)	6
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Additional heat capacity, 10^9 Btu/hr	(read)	0.37

TABLE 12. (Continued)

Steam Supply Incremental Cost (N-29 and N-39)

Additional heat capacity at 3.5 in. Hg, 10^6 Btu/hr		89
Additional heat capacity at 6 in. Hg, 10^6 Btu/hr		365
SSIC at 89 x 10^6 Btu/hr, $\$10^6$	at \$200/Kw	0.67
SSIC at 365 x 10^6 Btu/hr, $\$10^6$	at \$200/Kw	2.8

$\therefore \Delta$ SSIC = (2.8 - 0.67) x $\$10^6$ = $\$2.1 \times 10^6$

Turbine Generator Incremental Cost (N-40)

Same input additional heat capacities as in G.

TGIC at 89 x 10^6 Btu/hr, $\$10^6$	at \$200/Kw	0.38
TGIC at 365 x 10^6 Btu/hr, $\$10^6$	at \$200/Kw	1.6

$\therefore \Delta$TGIC = (1.6 - 0.38) x $\$10^6$ = $\$1.2 \times 10^6$

The cost penalty for capability loss for this plant =
(2.1 + 1.2) x $\$10^6$ = $\$3.3 \times 10^6$

TABLE 13. NOMOGRAPH EXAMPLE - MECHANICAL DRAFT DRY TOWER

Net generating capacity, Mw	800
Type of plant	Fossil
Plant heat rate, Btu/Kw-hr	9540
Plant load factor, percent	60
Dry bulb temperature, ^{o}F	90

A. Combined Performance (N-23)

Dry bulb temperature, ^{o}F	(enter)	90
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Heat rejection, 10^9 Btu/hr	(read)	4.1
Turbine back pressure, in. Hg abs	(read)	6

B. Initial Temperature Difference (N-24)

Turbine back pressure, in. Hg abs	(enter)	6
Dry bulb temperature, ^{o}F	(enter)	90
Initial temperature difference, ^{o}F	(read)	50

C. Capital Cost (N-25)

Initial temperature difference, ^{o}F	(enter)	50
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Heat rejection system cost, $\$10^6$	(read)	18.5

D. Auxiliary Power (N-26)

Initial temperature difference, ^{o}F	(enter)	50
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Auxiliary power requirements, Mw	(read)	21

E. Plant Efficiency (N-31)

Turbine back pressure, in.Hg abs	(enter)	6
Plant efficiency, percent	(read)	34.1

F. Additional Heat Capacity (N-29)

Turbine back pressure, in. Hg abs	(enter)	6
Plant capacity, Mw	(enter)	800
Plant type	(enter)	Fossil
Additional heat capacity, 10^9 Btu/hr	(read)	0.37

The same procedure is followed as in Table 12 yielding the same cost penalty for capability loss of $\$3.3 \times 10^6$ for this plant.

Makeup Water Requirements

Two examples are presented that utilize the Water Requirements nomographs. In the first case, the natural draft wet tower introduced in Table 5 is solved to yield the makeup water requirements. Components result from N-34 through N-38 for evaporative loss, drift loss, and blowdown and are shown in Table 14. In Case 2, the makeup water requirements for the cooling pond (Ref. Table 10) operating in a closed cycle mode are presented. Table 15 shows the results due to evaporative losses alone following application of N-33 and N-35.

In Table 14, the average annual evaporative loss is estimated to be 6.2 cfs following application of the factors introduced in Section X. This can be compared to the design evaporative loss of 13 cfs determined from the nomographs.

In Table 15, the average annual evaporative loss for the cooling pond was computed to be 9.3 cfs after accounting for the evapotranspiration correction assuming characteristics similar to the Lake Michigan area. This estimate is much less than the design evaporative loss of 19.8 cfs, yet higher than the 6.2 cfs average water loss from a cooling tower.

TABLE 14. NOMOGRAPH EXAMPLE - WATER REQUIREMENTS (CASE 1)

Net generating capacity, Mw		800
Type of plant		Fossil
Type of heat rejection system		Natural Draft Wet
Range, °F		28.7
Approach, °F		18.5
Water flow rate, 10^6 gpm		0.25
Drift loss, percent		0.005
Number of concentrations		3
A. Evaporative Loss (N-34)		
Range, °F	(enter)	28.7
Type of cooling system	(enter)	Natural Draft Wet
Approach, °F	(enter)	18.5
Evaporative loss, cfs/Mw	(read)	0.0162
B. Percent Evaporative Loss (N-35)		
Evaporative loss, cfs/Mw	(enter)	0.0162
Plant capacity, Mw	(enter)	800
Evaporative loss, cfs	(read)	13.0
Water flow rate, cfs	(enter)	560
Evaporative loss, percent of flow rate	(read)	2.32
C. Percent Blowdown (N-36)		
Drift loss, percent	(enter)	0.005
Number of concentrations	(enter)	3
Evaporative loss, percent	(enter)	2.32
Blowdown, percent	(read)	1.16
D. Makeup Water (N-38)		
Makeup water requirements, percent	(enter)	3.48
Water flow rate, cfs	(enter)	560
Makeup water requirements, cfs	(read)	19.5
Makeup water requirements, 10^6 gpm	(read)	0.0087

TABLE 14. (Continued)

E. Average Annual Evaporative Loss

Design evaporative loss, cfs	13.0
x plant capacity factor	(.6)
x off-design conditions factor	(.8)
= Average annual evaporative loss, cfs	6.2

TABLE 15. NOMOGRAPH EXAMPLE - WATER REQUIREMENTS (CASE 2)

Net generating capacity, Mw		800
Type of plant		Fossil
Type of heat rejection system		Cooling Pond (Closed Cycle)
Design equilibrium temperature, ^{o}F		80.6
Wet bulb temperature, ^{o}F		70
Condenser rise, ^{o}F		20
Water flow rate, 10^6 gpm		0.39
Pond loading, fossil Mw/acre		0.73
A. Cooling Pond Evaporation (N-33)		
Wet bulb temperature, ^{o}F	(enter)	70
Pond loading, fossil acre/Mw	(enter)	1.37
Evaporative loss, cfs/Mw	(read)	0.0247
B. Percent Evaporative Loss (N-35)		
Evaporative loss, cfs/Mw	(enter)	0.0247
Plant capacity, Mw	(enter)	800
Evaporative loss, cfs	(read)	19.8
Water flow rate, cfs	(enter)	870
Evaporative loss, percent of flow rate	(read)	2.28
Makeup water requirements, cfs		19.8
Makeup water requirements, 10^6 gpm		0.0089
C. Average Annual Evaporative Loss		
Design evaporative loss, cfs		19.8
Plant capacity factor (0.6) x 19.8 =		11.9
Evapotranspiration correction* (at 1100 acres)		- 2.6
Average annual evaporative loss, cfs		9.3

* Assumes characteristics similar to the Lake Michigan region.

Costs

Two examples are presented which utilize the cost nomographs. In Case 1, the incremental generating cost is determined for the natural draft wet tower introduced in Table 5. Example lines for this case appear on Nomographs N-41 through N-49, excluding N-44. Results for this case are in Table 16 and 17. Table 18 presents the development of the incremental generating cost for a once-through cooling system. In the cost estimation for once-through cooling systems, only Nomographs N-41, N-42, N-44, N-45, N-46, and N-49 are employed.

Some of the input quantities selected in the two examples are discussed in the following paragraphs. The plant load factor was assumed to be 60 percent. This magnitude represents an upper bound of the approximate average annual plant factors for selected conventional, fossil-fueled steam-electric generating plants as reported in Reference 19. Modern, large, fossil-fueled and nuclear-fueled power plants will be expected to achieve 70 percent and 80 percent, respectively, in future years.

The unit power cost was assumed at 0. 8 cents/Kw-hr. This value is a little higher than the reported (Ref. 33) total power plant busbar energy cost of 0. 72 cents/Kw-hr for stations put into service after 1967. In Table 16, the additional annual power cost has the least contribution to the total incremental annual cost.

The fuel cost for the fossil plant example was assumed to be $0. 30/ 10^6 Btu. This quantity is representative (at the low end) of the cost range for the various fossil fuels. In Reference 34, fuel costs in 1971 for the total United States averaged 29. 1 cents/10^6 Btu for gas, 36. 3 cents/10^6 Btu for coal and 55. 5 cents/10^6 Btu for oil consumed. According to Reference 35, projected nuclear fuel costs up through 1975 will remain nearly constant at 20 cents/10^6 Btu. Nuclear fuel costs are also listed for specific power plants in Reference 19. Fuel costs for four of these nuclear plants average out to 25 cents/10^6 Btu.

TABLE 16. NOMOGRAPH EXAMPLE - COSTS (CASE 1)

Net generating capacity, Mw	800
Type of plant	Fossil
Plant load factor, percent	60
Type of heat rejection system	Natural Draft Wet
Turbine back pressure, in. Hg abs	3. 8
Additional heat capacity, 10^6 Btu/hr	115
Water flow rate, 10^6 gpm	0. 25
Auxiliary Power requirements, Mw	2. 8
Natural draft tower cost, $\$10^6$	4. 9

A. Condenser Surface Area (N-41)

Water flow rate, 10^6 gpm	(enter)	0. 25
Condenser rise, ^oF	(enter)	28. 7
Overall heat transfer coefficient, Btu/hr-ft^2-^oF	(enter)	600
Area of heat transfer surface, 10^6 ft^2	(read)	0. 37

B. Condenser Cost (N-42)

Area of heat transfer surface, 10^6 ft^2	(enter)	0. 37
Condenser tube material	(enter)	Admiralty
Condenser capital cost, $\$10^6$ (including field installation)	(read)	8. 2

C. Water Circulation System Cost (N-43)

Water flow rate, 10^6 gpm	(enter)	0. 25
Natural draft tower at 50 ft head	(enter)	50
Water circulation system capital cost, $\$10^6$	(read)	0. 41

∴ The total incremental capital cost =

heat rejection system cost for a given method + costs from B and C above =

$(4.9 + 8.2 + 0.41) \times \$10^6 = \$13.51 \times 10^6$

D. Capital Recovery Factor (N-45)

Interest rate, percent	(enter)	8
Years amortized	(enter)	30
Capital recovery factor, percent	(read)	8. 9
Recurring costs, %/year		5. 2
Fixed charge rate, %/year		14. 1

(sheet 2 of 2)

TABLE 16. (Continued)

E. Annualized Capital Cost (N-46)

Total incremental capital cost, $\$10^6$	(enter)	13.5
Fixed charge rate, %/year	(enter)	14.1
Annualized incremental capital cost, $\$10^6$/year	(read)	1.9

F. Additional Annual Power Cost (N-47)

Auxiliary power requirements, Mw	(enter)	2.8
Plant load factor, percent	(enter)	60
Additional annual energy consumption, 10^6 Kw-hr/yr	(read)	15
Unit power cost, cents/Kw-hr	(enter)	0.8
Additional annual power cost, $\$10^6$/year	(read)	0.12

G. Additional Annual Fuel Cost (N-48)

Additional heat capacity, 10^6 Btu/hr	(enter)	115
Plant load factor, percent	(enter)	60
Fuel cost, $\$/10^6$ Btu	(enter)	0.30
Additional annual fuel cost, $\$10^6$/year	(read)	0.19

∴ Total incremental annual cost = annual costs from E, F, and G above = $(1.9 + 0.12 + 0.19) \times \$10^6$/year = 2.21×10^6/year

H. Incremental Generation Cost (N-49)

Total incremental annual cost, $\$10^6$/year	(enter)	2.21
Plant load factor, percent	(enter)	60
Plant capacity, Mw	(enter)	800
Incremental generating cost, mills/Kw-hr	(read)	0.52

TABLE 17. AVERAGE ANNUAL OPERATING COSTS FOR CASE 1*

Season	Average DBT °F	Average Relative Humidity %	CWT °F	TBP in. Hg abs	Auxiliary Power Mw	ΔQ 10^6 Btu/hr	Additional Annual Fuel Cost $/yr
Winter	24	80	-	~ 2	2.7	0	0
Spring	43.5	70	72.5	2.39	2.7	14	5,500
Summer	68	70	84.2	3.35	2.8	77	30,400
Fall	50.5	75	75.8	2.64	2.8	27	10,700
Total							46,600
Value Under Design Conditions	77	71	88.5	3.8	2.8	115	190,000

* Meteorological data for Lake Michigan region.

TABLE 18. NOMOGRAPH EXAMPLE - COST OF ONCE-THROUGH COOLING (CASE 2)

Net generating capacity, Mw	800
Type of plant	Fossil
Plant load factor, percent	60
Type of heat rejection system	Once-Through
Turbine back pressure, in. Hg abs	~ 2
Range, ^oF	20
Water flow rate, 10^6 gpm	0.38

A. Condenser Surface Area (N-41)

Water flow rate, 10^6 gpm	(enter)	0.38
Condenser rise, ^oF	(enter)	20
Overall heat transfer coefficient, Btu/hr-ft^2-^oF	(enter)	600
Area of heat transfer surface, 10^6 ft^2	(read)	0.48

B. Condenser Cost (N-42)

Area of heat transfer surface, 10^6 ft^2	(enter)	0.48
Condenser tube material	(enter)	Admiralty
Condenser capital cost, \$$10^6$ (including field installation)	(read)	10.6

C. Cost for Once-Through Cooling (N-44)

Water flow rate, 10^6 gpm	(enter)	0.38
Total structural length, ft	(enter)	1000
Intake and outfall structure cost, \$$10^6$	(read)	2.2

∴ The total incremental capital cost = costs from B and C = (10.6 + 2.2) x \$$10^6$ = \$12.8 x 10^6

D. Capital Recovery Factor (N-45)

Interest rate, percent	(enter)	8
Years amortized	(enter)	30
Capital recovery factor, percent	(read)	8.9
Recurring costs, %/year		5.2
Fixed charge rate, %/year		14.1

TABLE 18. (Continued)

E. Annualized Capital Cost (N-46)

Total incremental capital cost, $\$10^6$	(enter)	12.8
Fixed charge rate, %/year	(enter)	14.1
Annualized incremental capital cost, $\$10^6$/year	(read)	1.8

F. Incremental Generating Cost (N-49)

Total incremental annual cost, $\$10^6$/year	(enter)	1.8
Plant load factor, percent	(enter)	60
Plant capacity, Mw	(enter)	800
Incremental generating cost, mills/Kw-hr	(read)	0.42

Based on the end results found in Tables 16 and 18, the differential busbar cost due to closed cycle cooling is:

$$0.52 - 0.42 = 0.10 \text{ mills/Kw-hr}$$

These values are valid assuming operation under design conditions.

The additional annual power cost and the additional annual fuel cost will be less than the design values presented in Table 16 when the plant operates under off-design conditions through the year. Average annual operating costs were estimated for the natural draft wet tower of Case 1. The results of Table 17 follow after applying several of the nomographs using off-design meteorological input conditions. In this table, discrete quantities appear for the four seasons of the year. The seasonal values can then be compared to the corresponding one under design conditions presented at the bottom of the table.

The dry bulb temperatures and relative humidities are average, seasonal, meteorological conditions for the Lake Michigan region from Reference 15. The data from this reference provided an excellent example. This reference also discussed and stressed the importance of annual operating costs under the more favorable off-design conditions.

Cold water temperatures (CWT) were determined from the Tower Performance nomograph, N-1. Then turbine back pressure was determined from N-2 for each season. The design condenser rise for this natural draft wet tower was 28.7°F. The TBP for the summer season is the value closest to the design value.

The values in Table 17 for auxiliary power depend upon water flow rate which in turn depends upon heat rejection rate (see Nomographs N-5 and N-6). Heat rejection rate as determined from the Turbine Performance nomographs, assumes performance of the turbine-generator at full load. In reality, the turbine-generator operates during the year at part-load (i. e., at capacities less than 100 percent). Under part-load operation, the turbine heat rate at a given turbine back pressure is higher than the corresponding value under full-load operation.

Thus, turbine performance under part-load operation is reduced, and there is a loss of power plant efficiency under part-load operation. The values of auxiliary power only reflect the effect of off-design meteorological conditions and differ little from the design value. Lower auxiliary power requirements and the corresponding costs will result under actual part-load operation over the year. Accurate determination of auxiliary power requirements under part-load operation requires the generation of additional turbine performance nomographs which are beyond the scope of this report.

Additional heat capacities (ΔQ) are also shown only for the effect of varying meteorological conditions. Full load turbine performance was assumed. Additional annual fuel costs were determined for each season after employing an equation similar to equation (36). A plant load factor of 60 percent was assumed in order to be consistent with Table 16. The total additional annual fuel cost, \$46,600/year, under off-design meteorological conditions is significantly less than the value under design conditions, \$190,000/year. The additional annual cost contributions due to auxiliary power and fuel are small compared to the annualized capital cost contribution in Table 16. Thus the incremental generating cost in mills/Kw-hr is primarily made up by the annualized capital cost contribution.

SECTION XIII

ACKNOWLEDGMENTS

The support of the Environmental Protection Agency, and Dr. Bruce A. Tichenor, Project Officer, National Environmental Research Center - Corvallis, Pacific Northwest Environmental Research Laboratory, is gratefully appreciated.

We are also grateful for the cooperation from individuals and companies, some cited in the references, who, in response to our contacts, supplied invaluable data and information.

SECTION XIV

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SECTION XV

NOMENCLATURE

The following parameters are defined in this report:

range	the water temperature decrease in the tower or pond
approach	the difference between the cold water temperature out of the tower and the wet bulb temperature
terminal temperature difference	the difference between the steam saturation temperature and the hot water leaving the condenser
cold water temperature	temperature of the water being supplied to the power plant
completely mixed pond	the flow between the inlet and outlet locations of the pond combined with wind mixing tend to keep the pond at a nearly uniform temperature
flow-through pond	a pond in which the temperatures decrease continually along the pond length
equilibrium temperature	the pond surface temperature for which the heat leaving the surface will exactly equal the heat entering the surface of a body of water
initial temperature difference	the steam saturation temperature minus the dry bulb temperature (dry tower only)
plant heat rate	heat energy required for production of electricity expressed in Btu/Kw-hr
in-plant efficiency	the steam supply efficiency for fossil or nuclear plants

turbine heat rate	the product of the plant heat rate and the in-plant efficiency
drift	the loss of entrained water that is carried out of the top of a wet tower or from a spray pond
blowdown	the amount of cooling water drained off for disposal and replaced by fresh makeup water

The following symbols are used in this report:

A	=	tower constant for dry towers
A_c	=	area of heat transfer surface, ft^2
A_p	=	pond area
B	=	blowdown, percent
b	=	exponent for dry towers
C	=	number of concentrations of the makeup water
CC	=	condenser cost, dollars
CWT	=	cold water temperature, ^oF
C_p	=	specific heat for water, Btu/lb-^oF
D	=	drift loss, percent
DBT	=	dry bulb temperature, ^oF
E	=	evaporative loss, percent
E_n	=	design equilibrium temperature, ^oF
f, f_1, f_2	=	nonlinear functions of parameters
HRSC	=	heat rejection system cost, dollars
ITD	=	initial temperature difference, ^oF
k	=	heat exchange coefficient, Btu/ft^2-day-^oF
K_1, K_2	=	SSIC equation constants
OTCC	=	once-through cooling cost, dollars
Q	=	heat rejection rate, 10^6 Btu/hr
Q_{in}	=	heat input to plant, Btu/hr
ΔQ_{in}, ΔQ	=	additional heat capacity, Btu/hr
q	=	total heat transferred in the condenser, Btu/hr

SSIC	=	steam supply system incremental costs, dollars
TBP	=	turbine back pressure, in. Hg absolute
TGIC	=	turbine generator incremental cost, dollars
TICC	=	total incremental capital cost, dollars
TTD	=	terminal temperature difference, ^oF
TU	=	mechanical draft wet tower units
T_d	=	dewpoint temperature, ^oF
T_i	=	water inlet temperature to condenser, ^oF
T_{in}	=	pond inlet temperature, ^oF
T_o	=	water outlet temperature from condenser, ^oF
T_{out}	=	pond outlet temperature, ^oF
T_s	=	steam saturation temperature or water surface temperature, ^oF
ΔT	=	residual temperature rise, ^oF
ΔT_c	=	condenser rise, ^oF (closed cycle)
U	=	overall heat transfer coefficient, Btu/hr-ft^2-^oF
W	=	heat equivalent of the rated output, Btu/hr
WBT	=	wet bulb temperature, ^oF
WCSC	=	water circulation system cost, dollars
WFR	=	water flow rate, lb/hr
η_I	=	in-plant or steam supply efficiency
η_P	=	thermal efficiency of the total plant
ρ	=	density of water

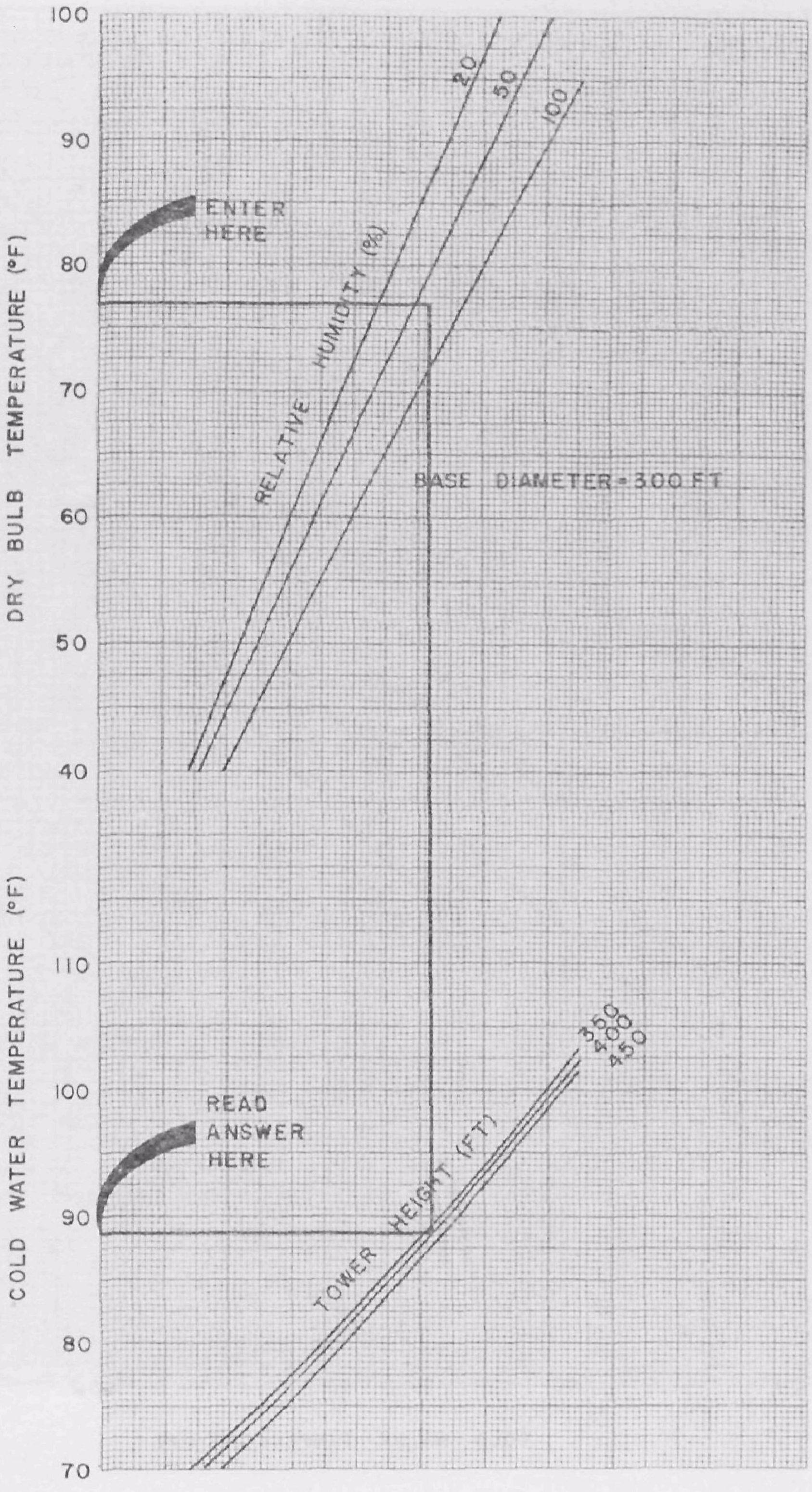
SECTION XVI

APPENDIX

The nomographs follow on the pages numbered N-1 through N-49.. If desired the nomographs could be made a separate entity themselves. In either case, they can be identified in sequence, for example, as page N-1, or as Nomograph -1 or as simply the abbreviated form, N-1.

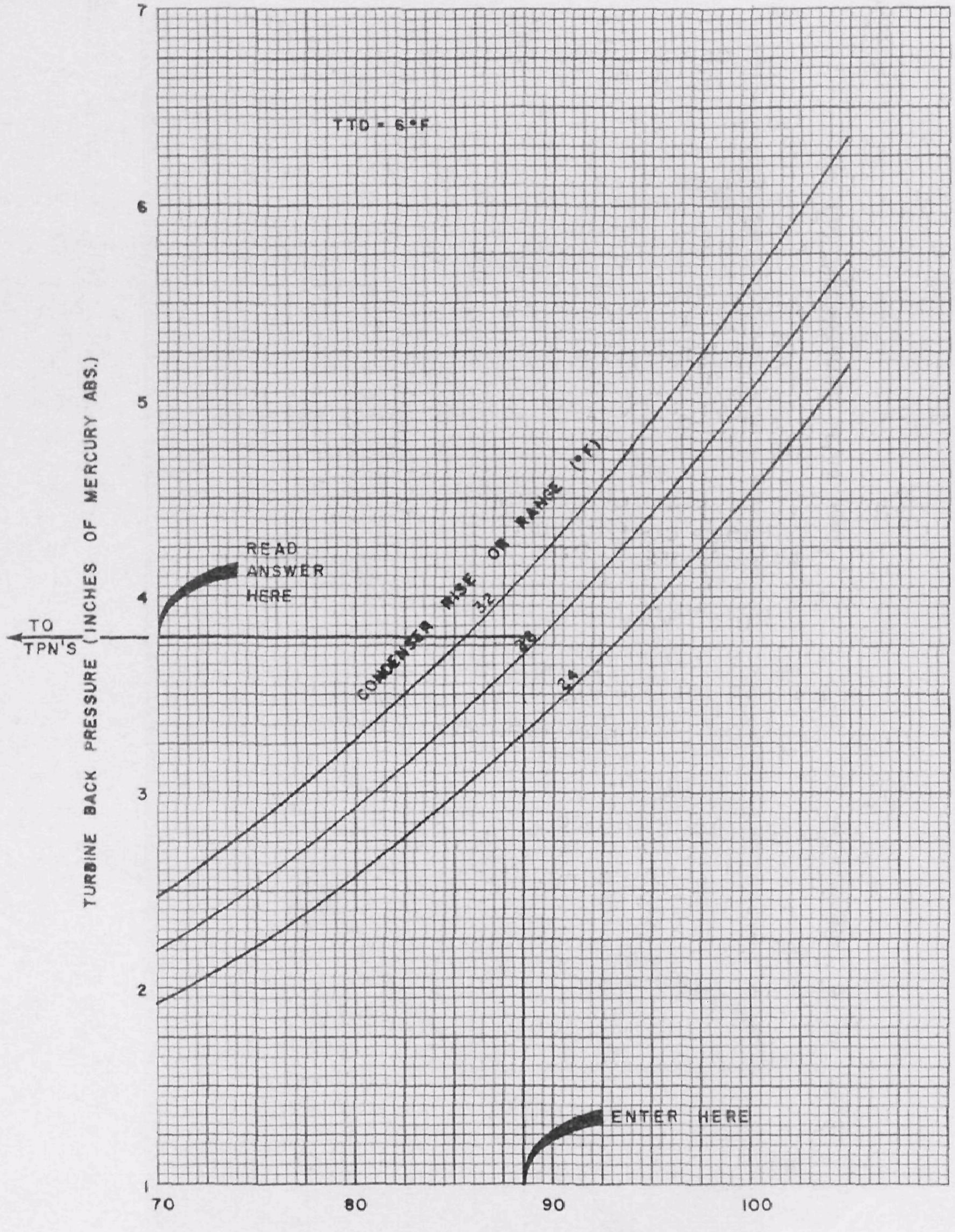
Following the nomographs is a series of figures indicating how the individual nomographs should be combined to incorporate the entire series for a given thermal pollution control system into a single oversize page if desired.

TOWER PERFORMANCE



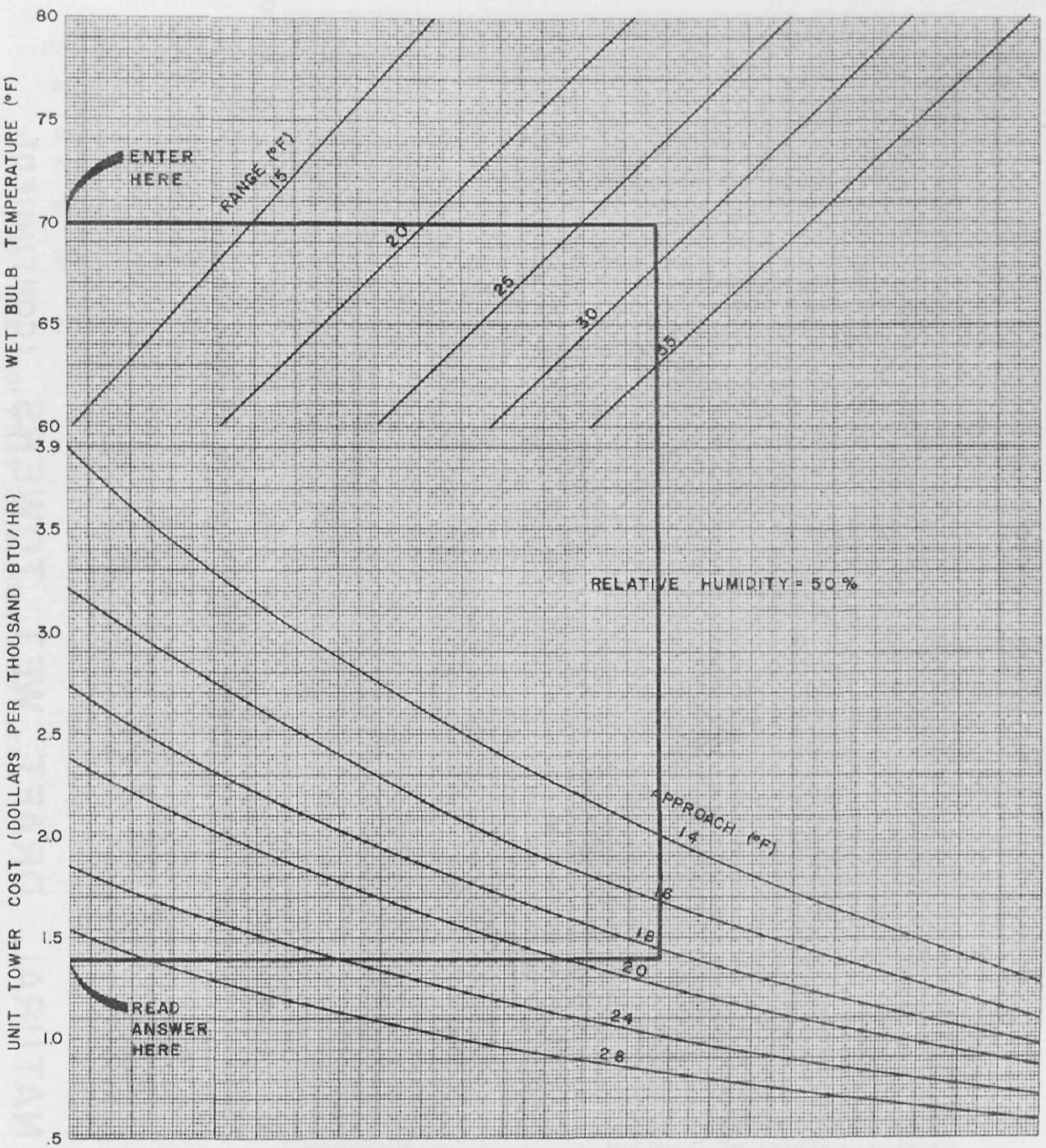
NATURAL DRAFT WET TOWERS

TURBINE BACK PRESSURE



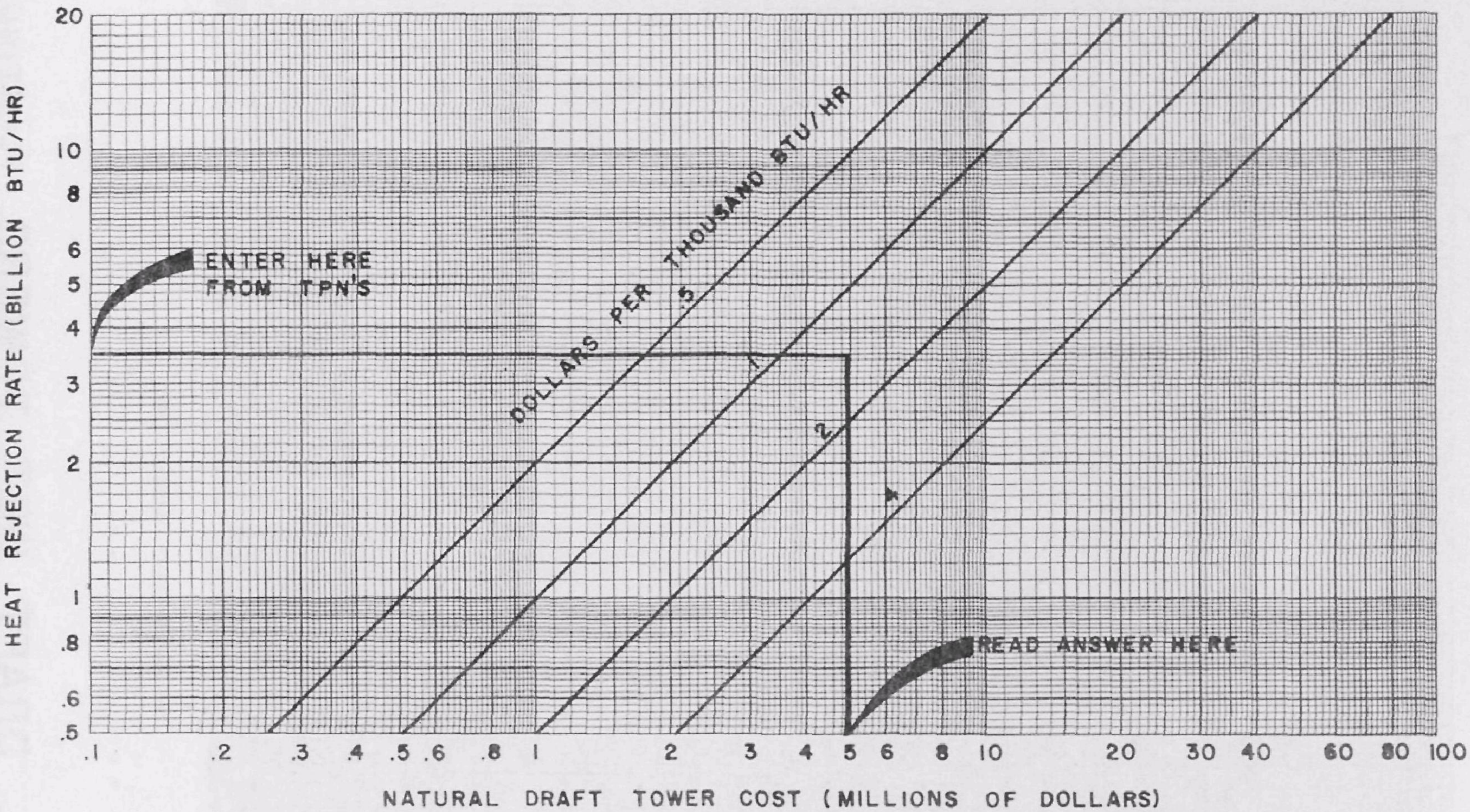
NATURAL DRAFT WET TOWERS (CONTINUED)

TOWER UNIT COST



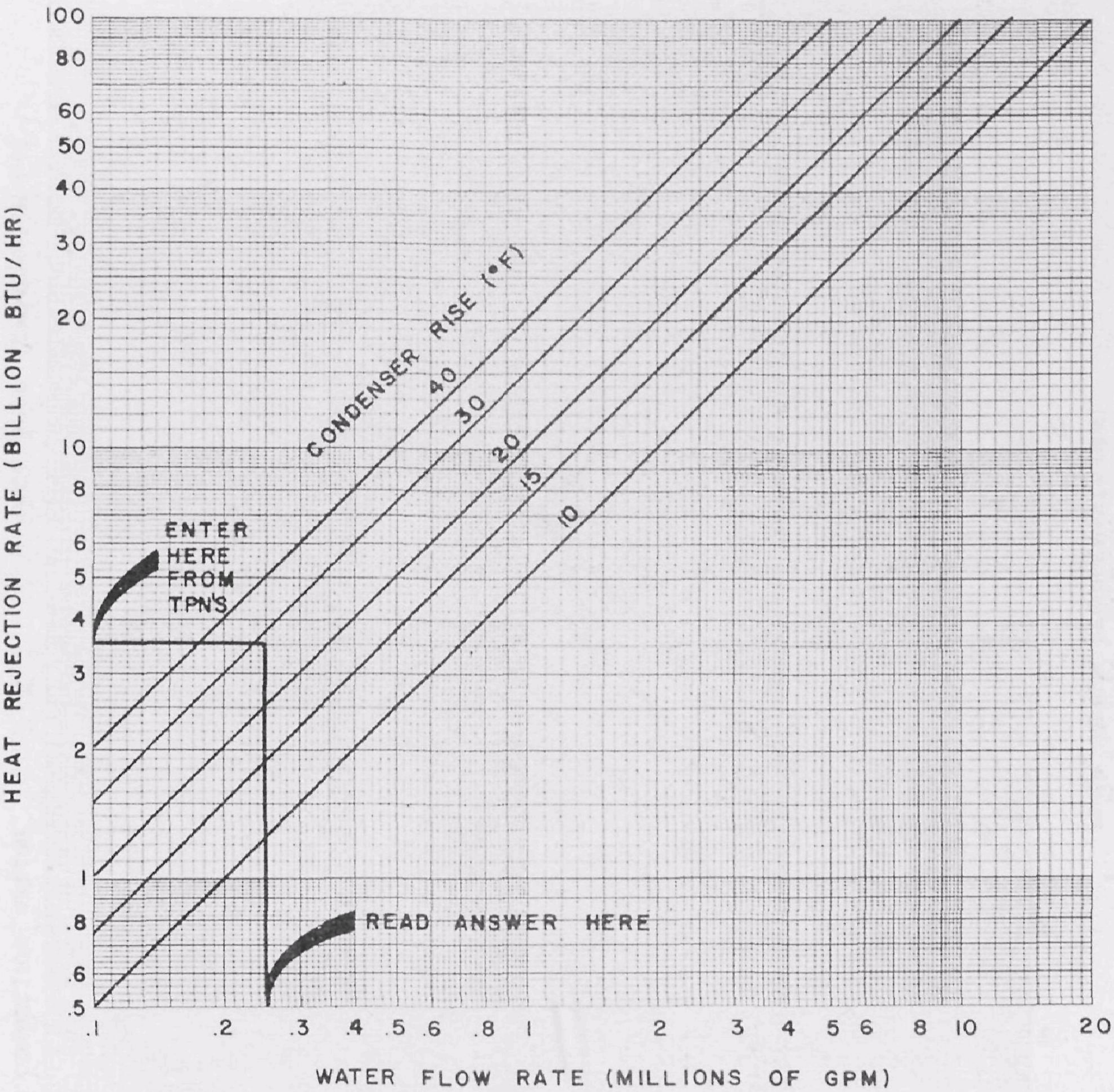
NATURAL DRAFT WET TOWERS (CONTINUED)

TOWER COST



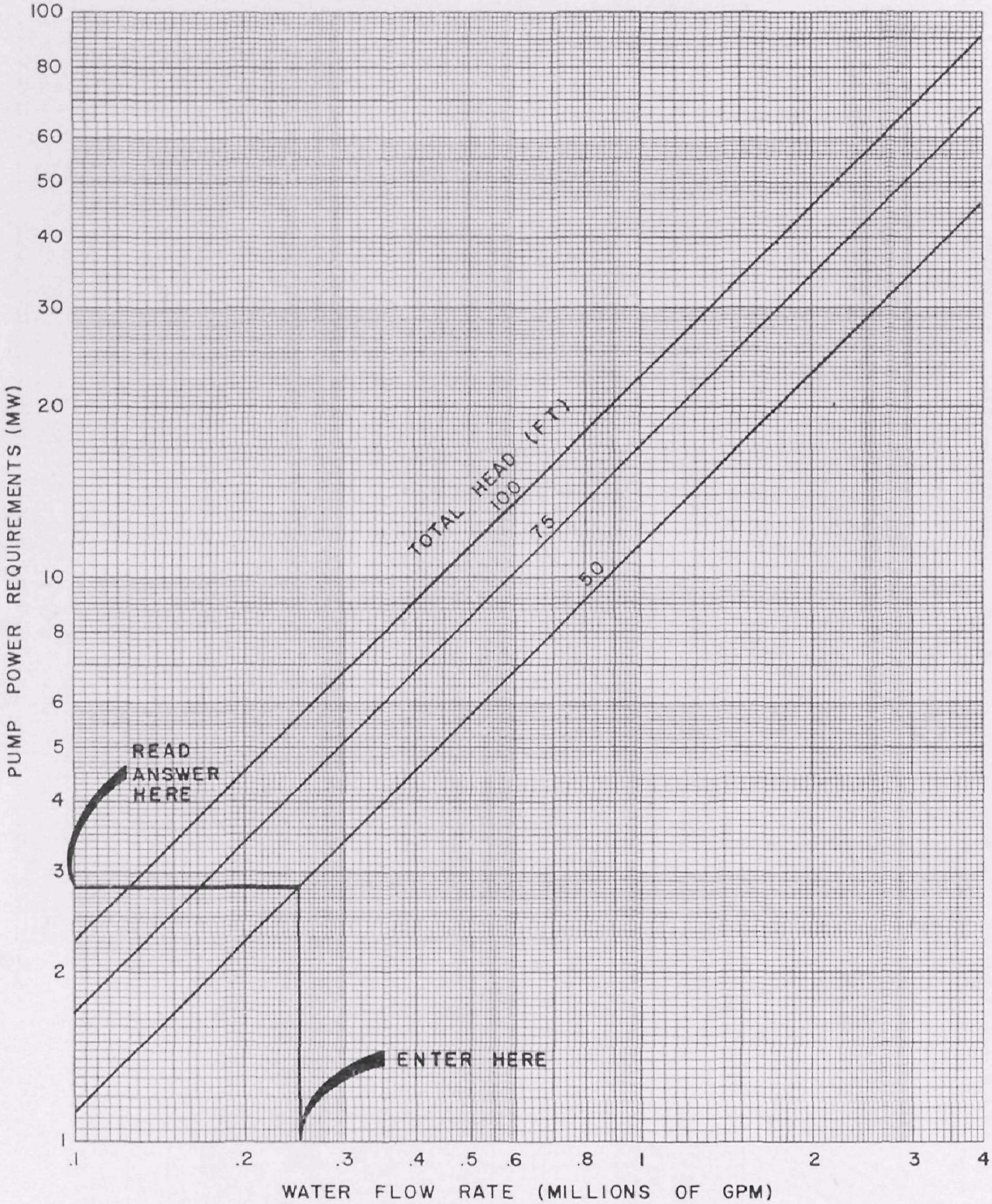
NATURAL DRAFT WET TOWERS (CONTINUED)

FLOW RATE



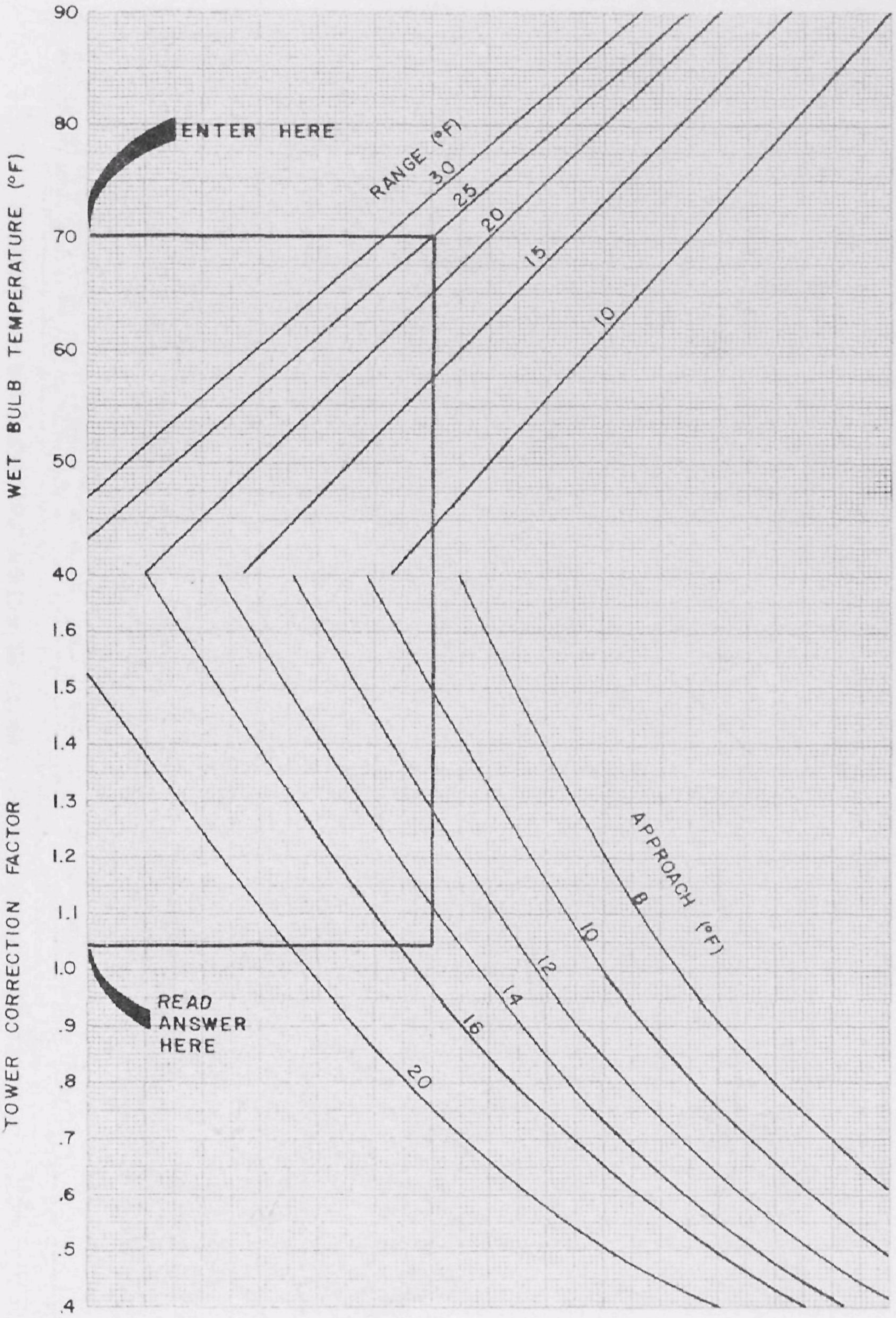
NATURAL DRAFT WET TOWERS (CONTINUED)

AUXILIARY POWER



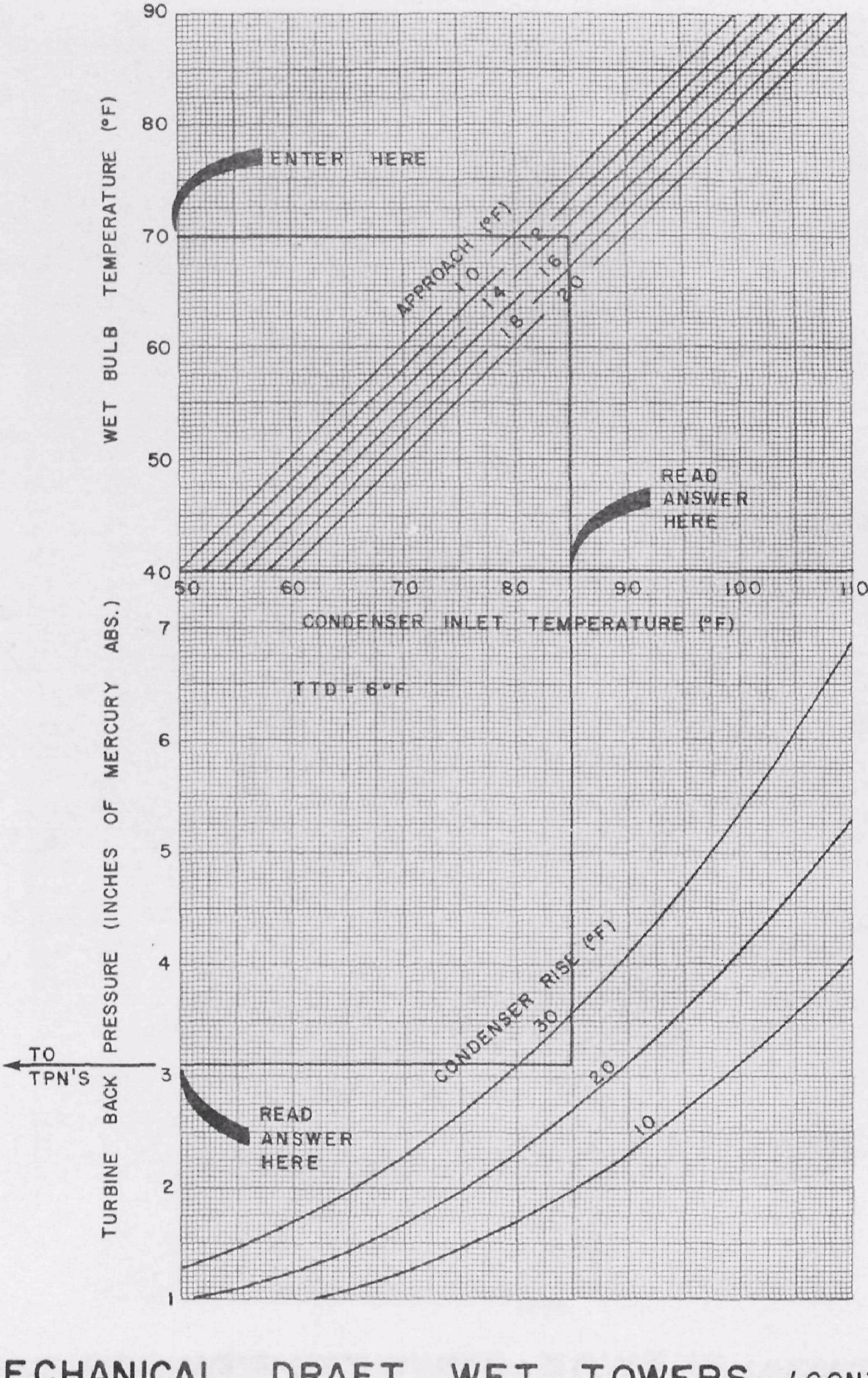
NATURAL DRAFT WET TOWERS (CONTINUED)

TOWER CORRECTION FACTOR



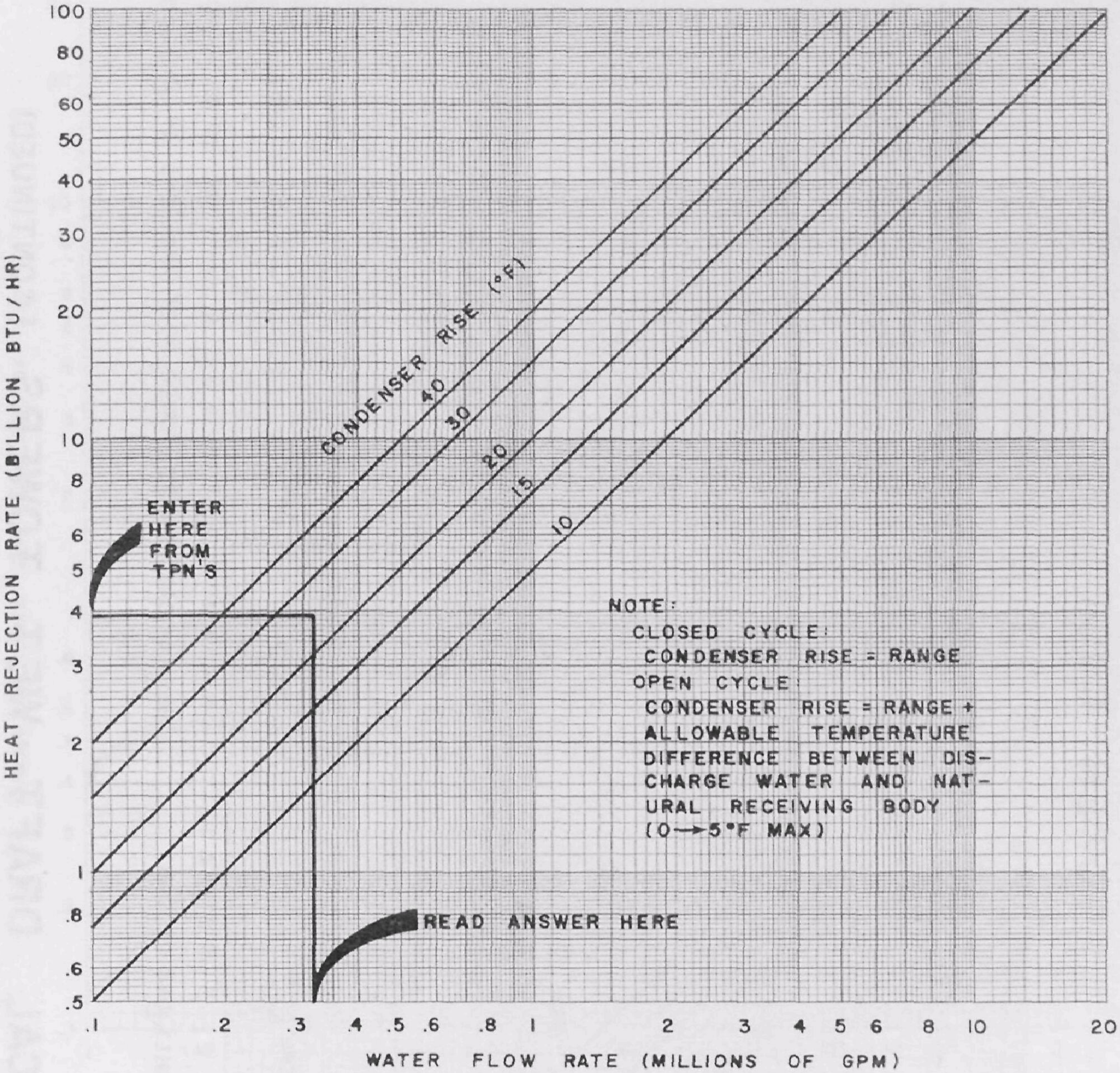
MECHANICAL DRAFT WET TOWERS

TURBINE BACK PRESSURE



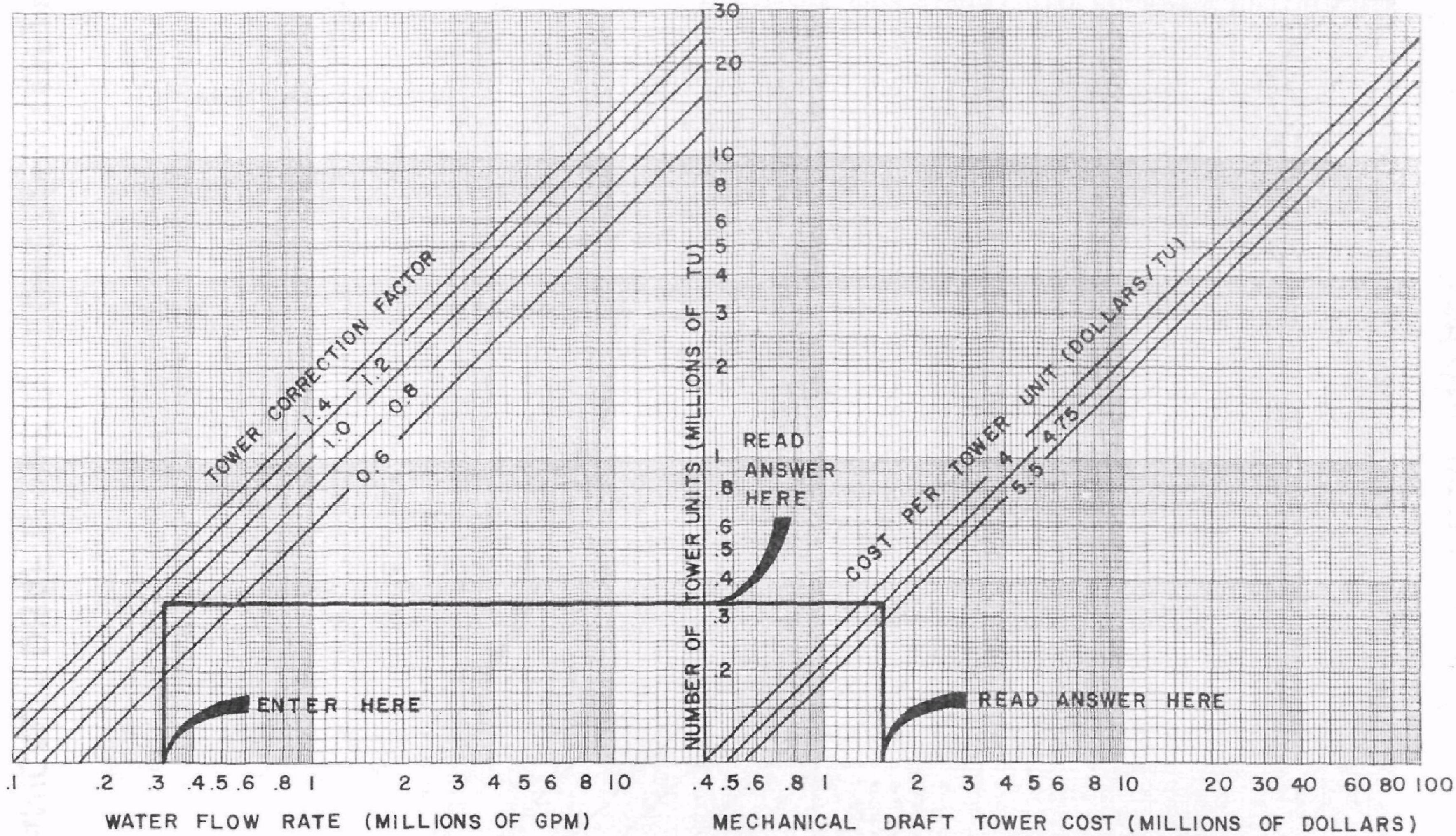
MECHANICAL DRAFT WET TOWERS (CONTINUED)

FLOW RATE



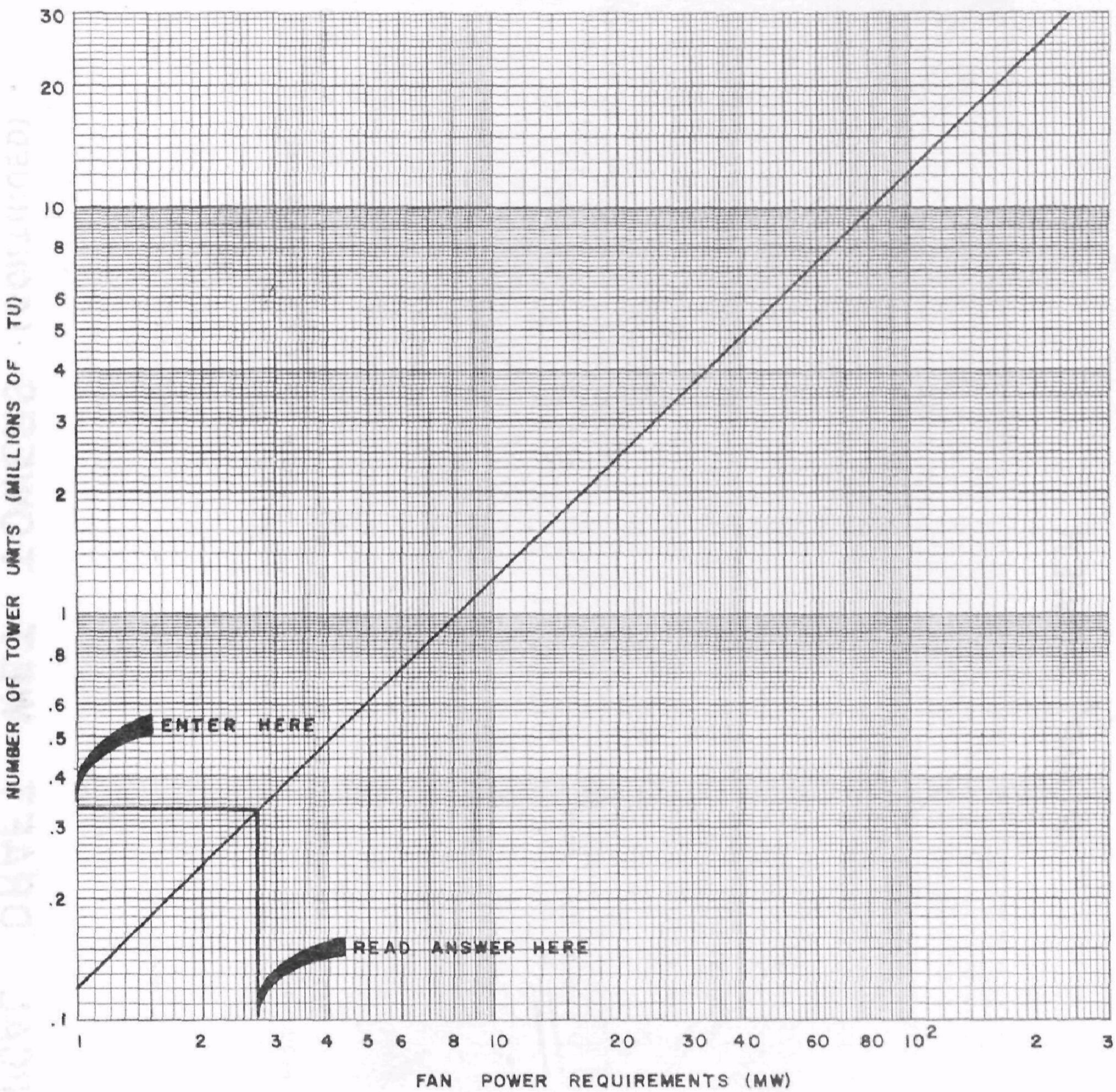
MECHANICAL DRAFT WET TOWERS (CONTINUED)

TOWER COST



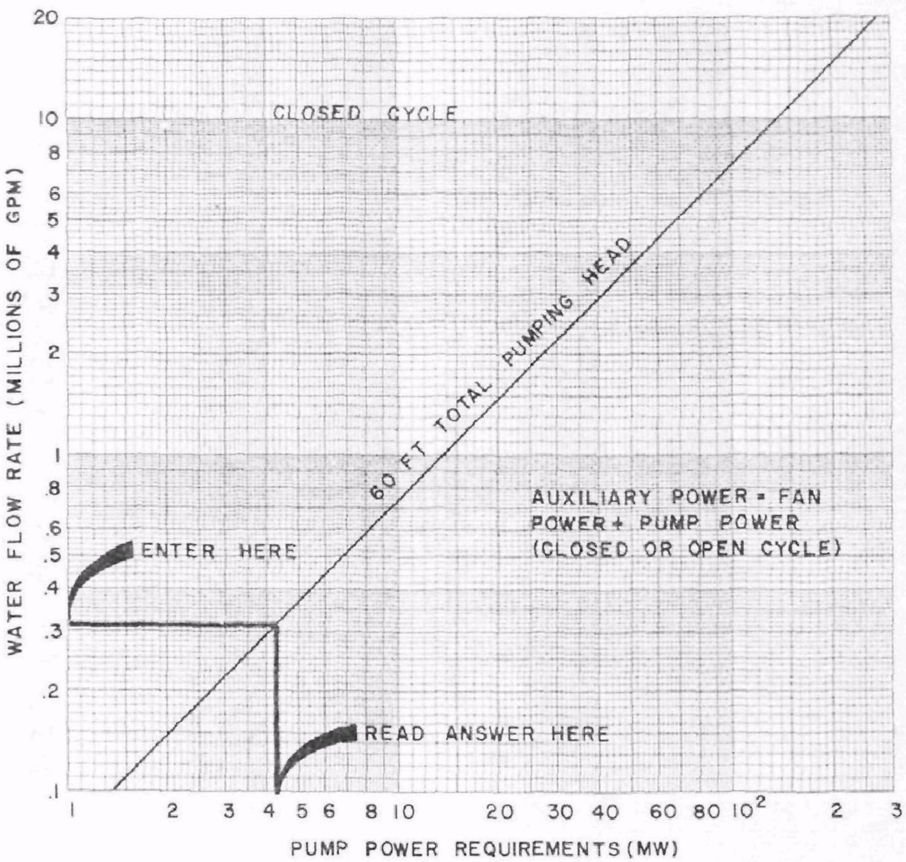
MECHANICAL DRAFT WET TOWERS (CONTINUED)

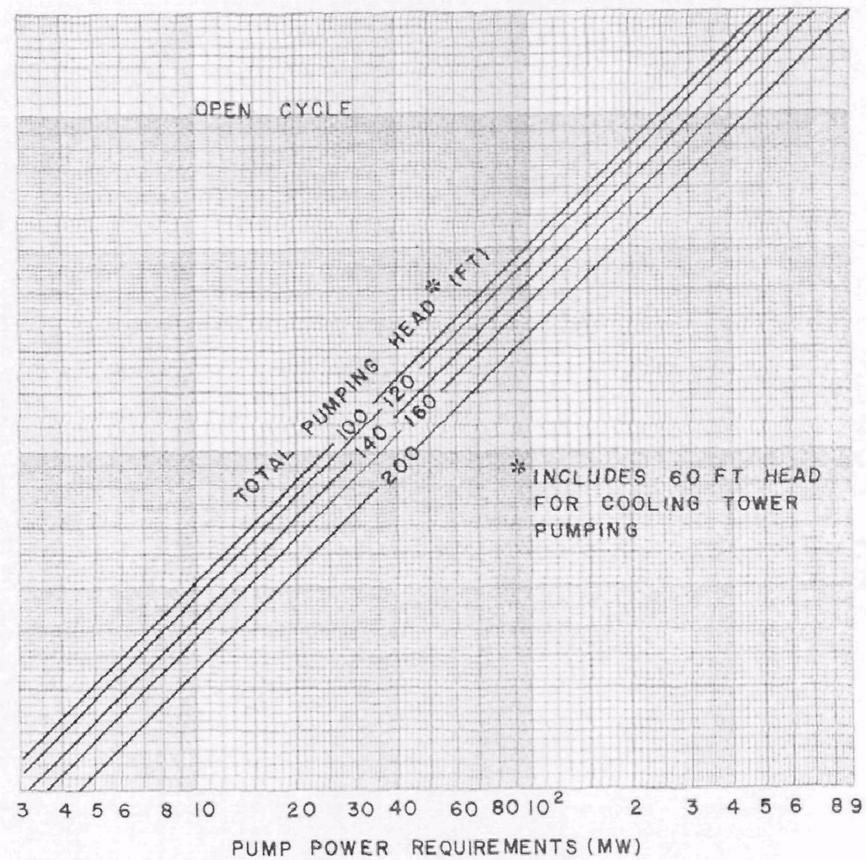
FAN POWER



MECHANICAL DRAFT WET TOWERS (CONTINUED)

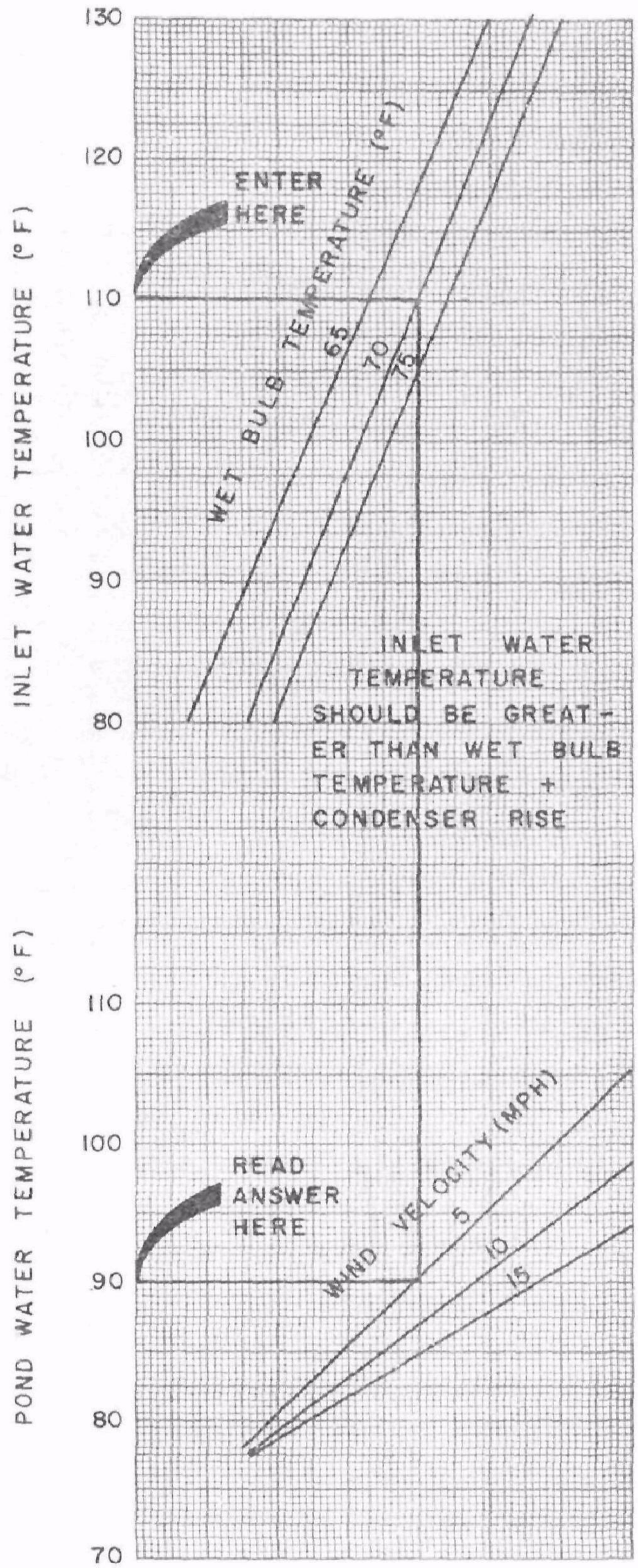
PUMP POWER





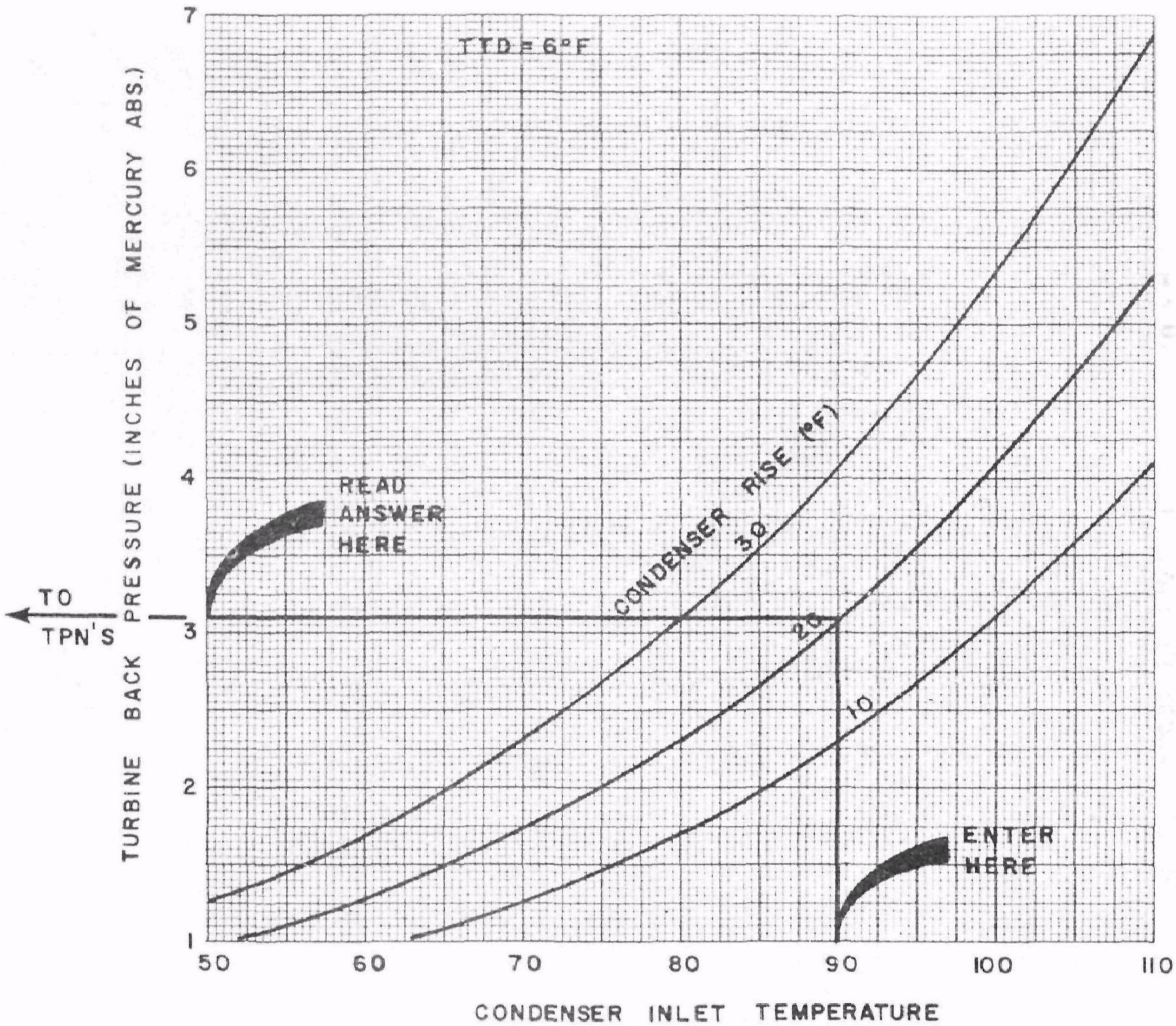
MECHANICAL DRAFT WET TOWERS (CONTINUED)

SPRAY POND PERFORMANCE



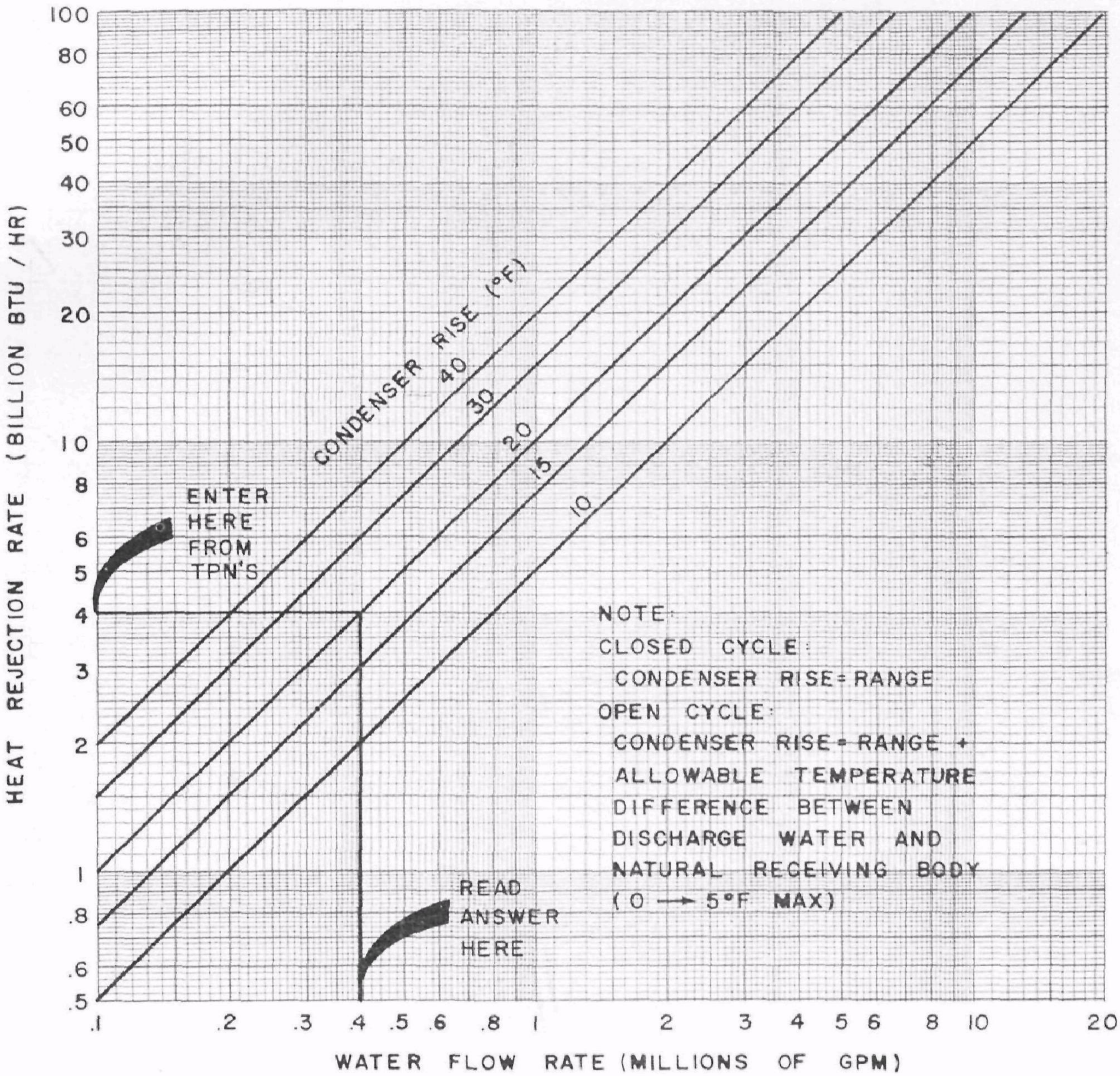
SPRAY PONDS

TURBINE BACK PRESSURE



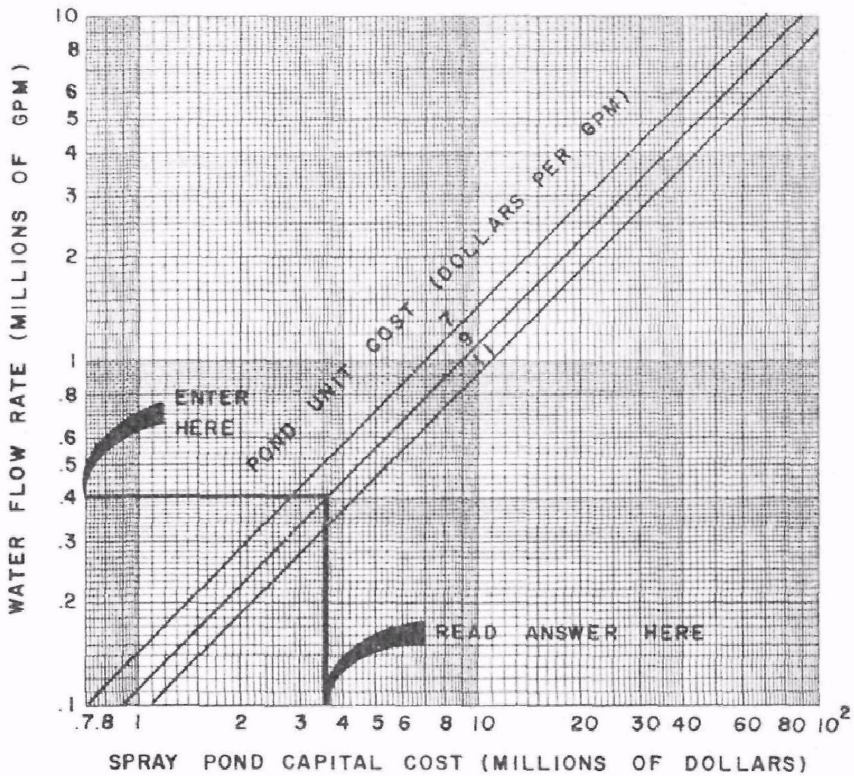
SPRAY PONDS (CONTINUED)

FLOW RATE

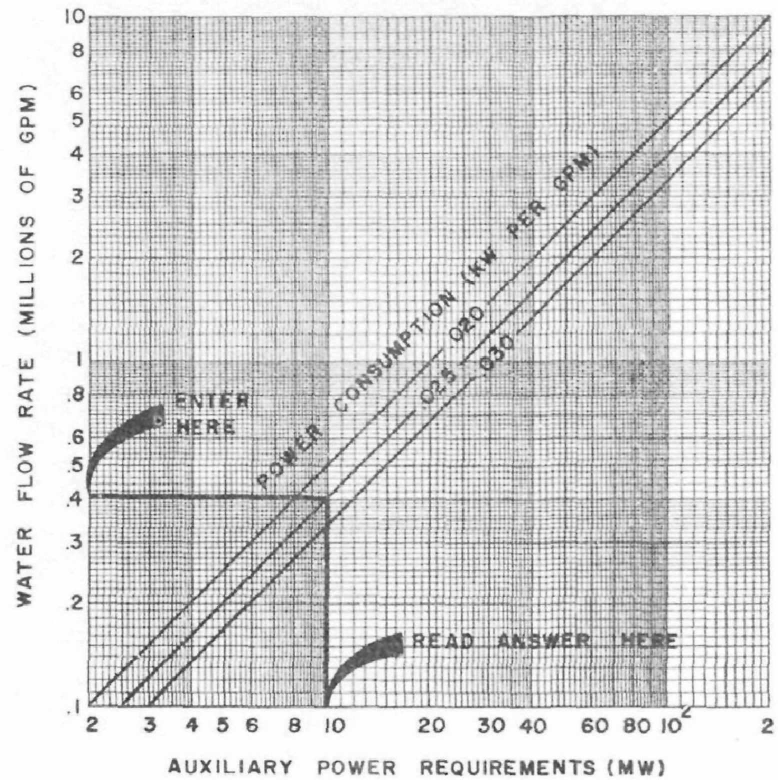


SPRAY PONDS (CONTINUED)

SPRAY POND COST



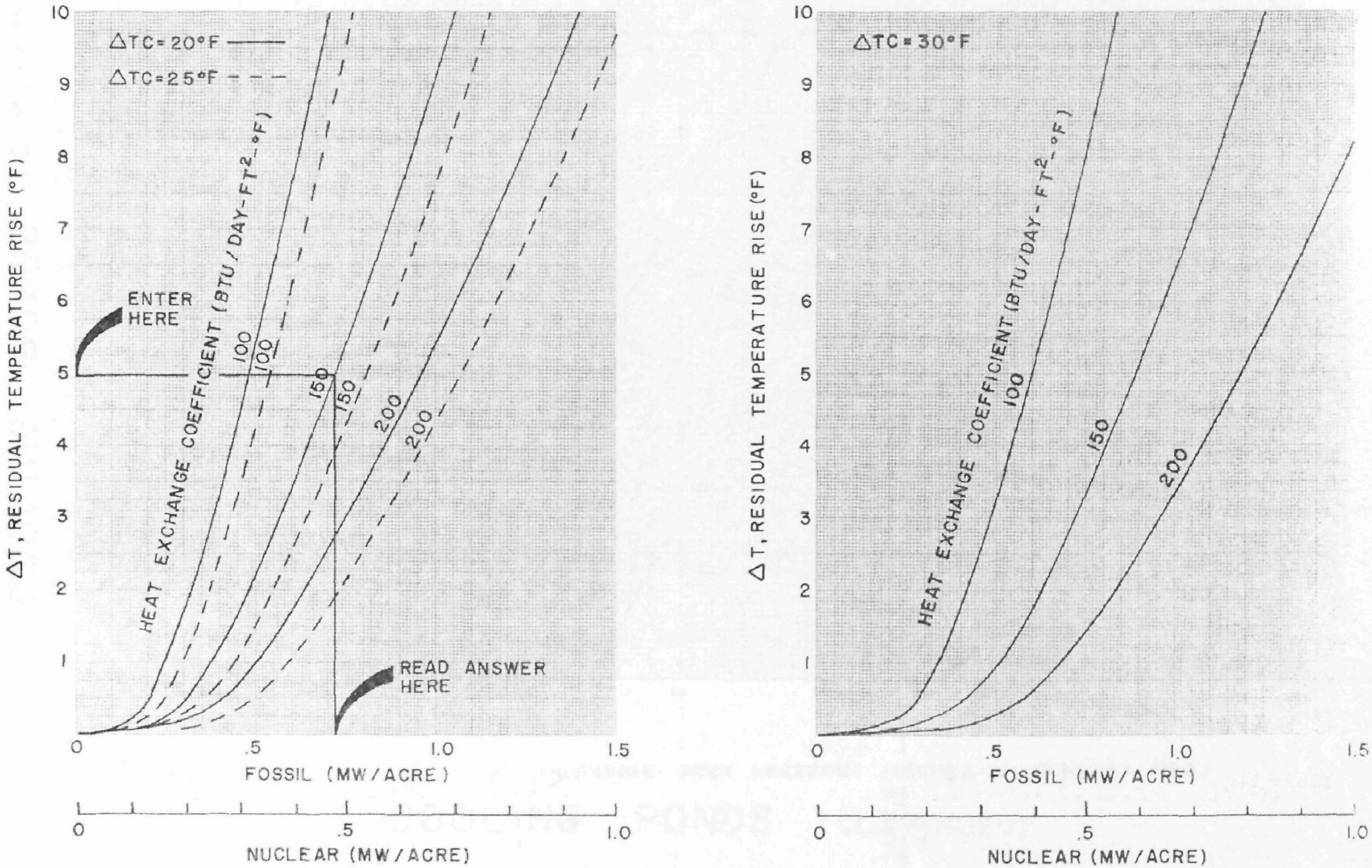
AUXILIARY POWER



SPRAY PONDS (CONTINUED)

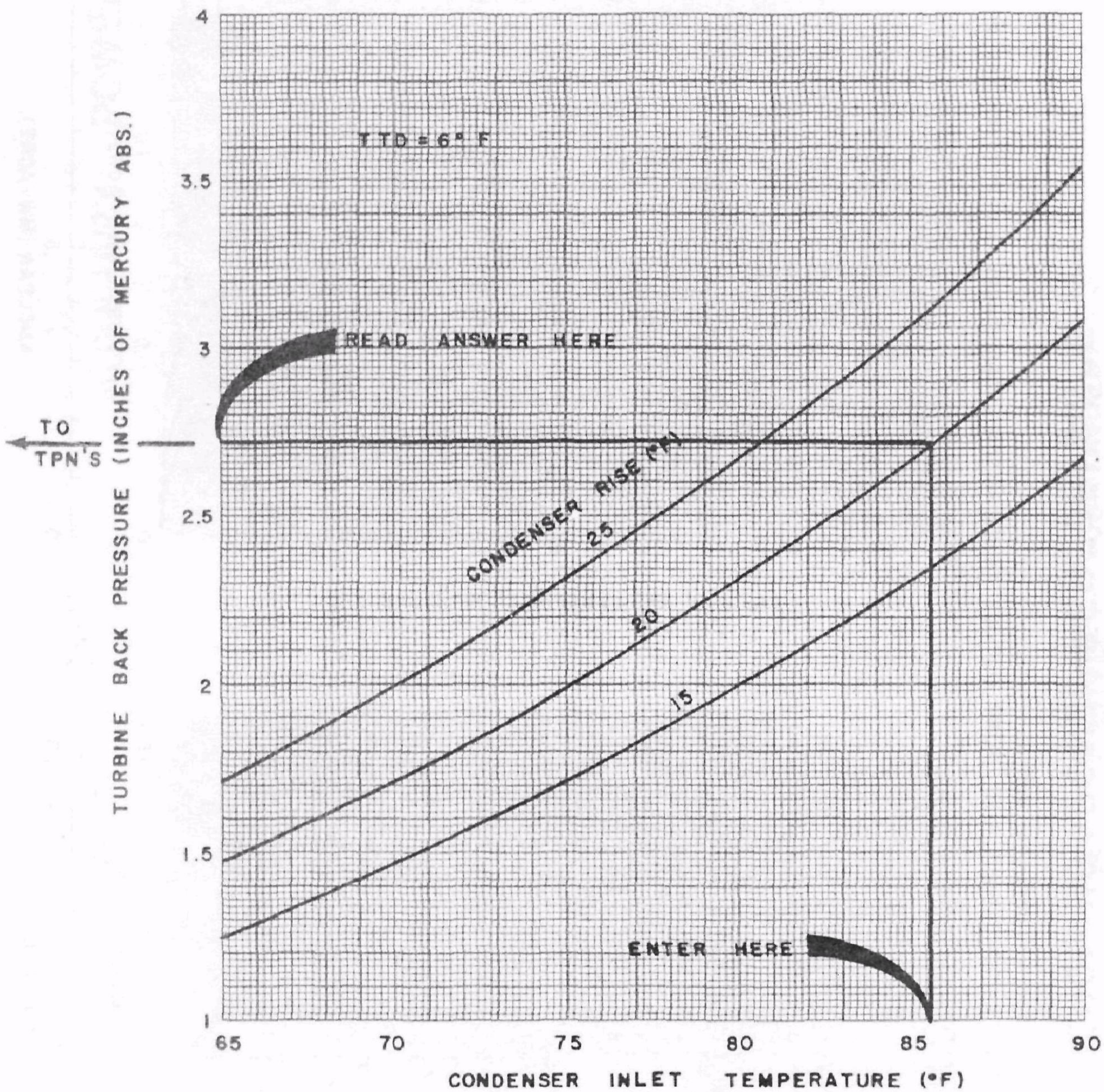
COOLING POND PERFORMANCE

CLOSED CYCLE: ΔTC = CONDENSER RISE
OPEN CYCLE: ΔTC = CONDENSER RISE - POND OUTLET TEMPERATURE + CONDENSER INLET TEMPERATURE



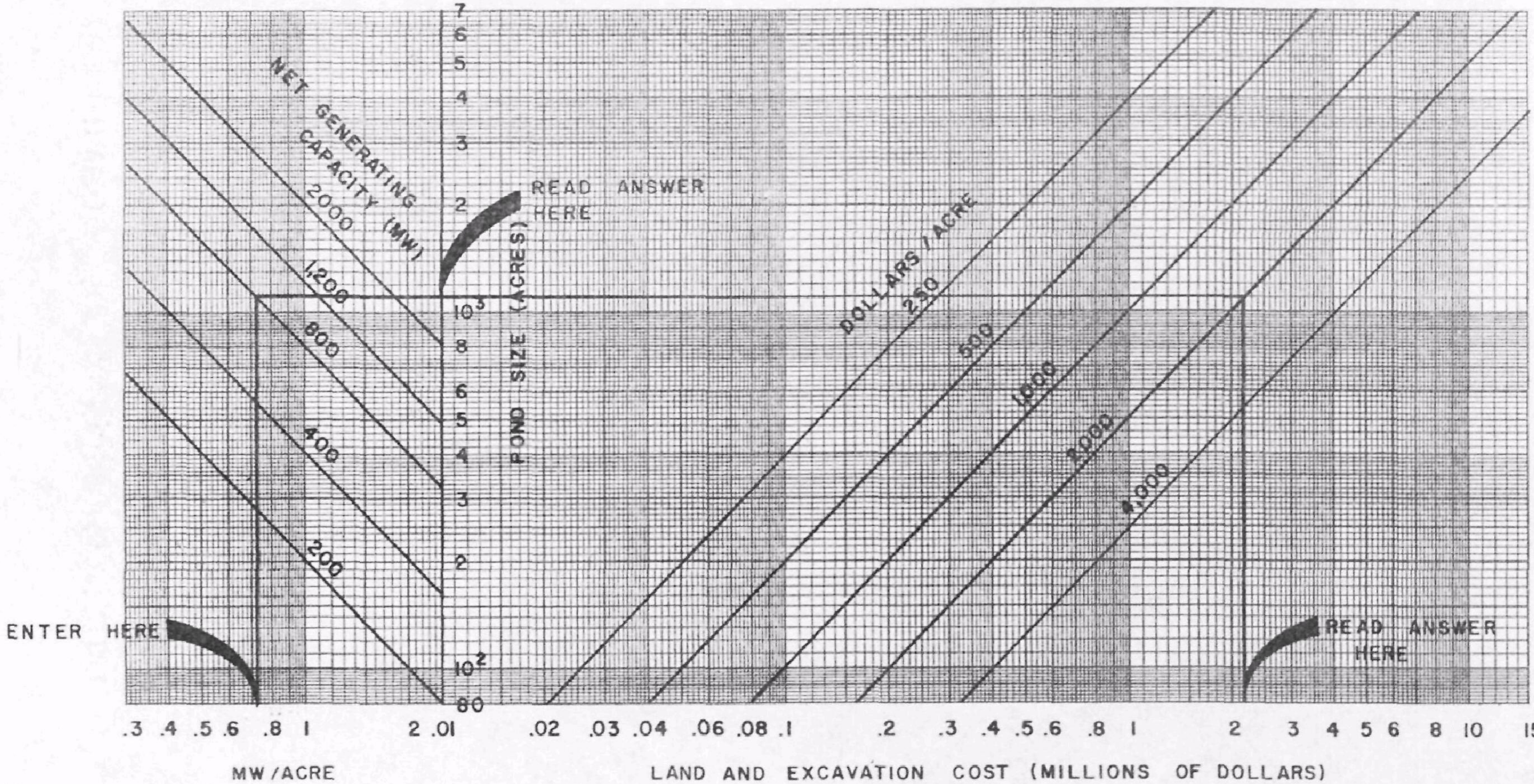
COOLING PONDS

TURBINE BACK PRESSURE



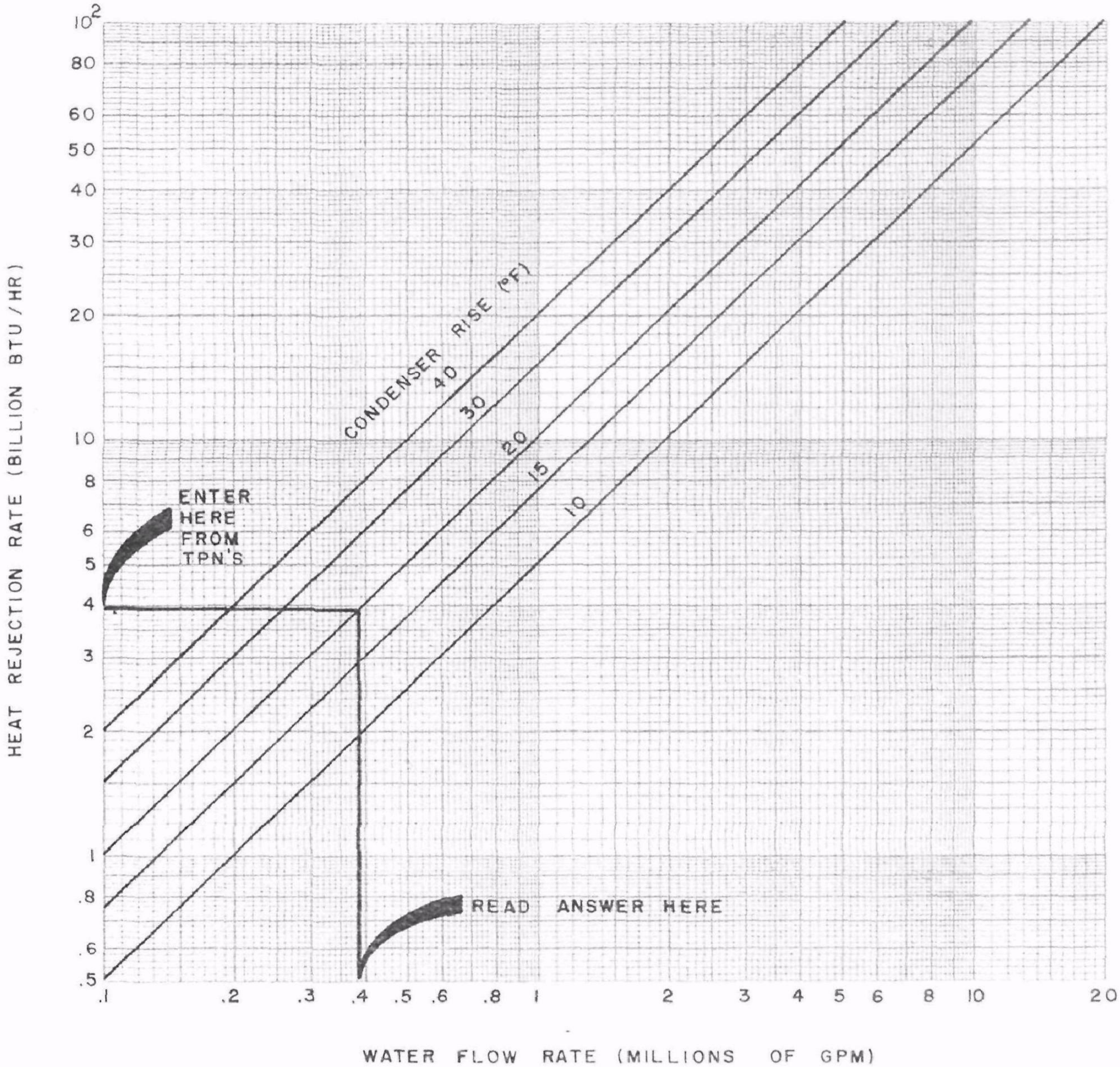
COOLING PONDS (CONTINUED)

COOLING POND COST



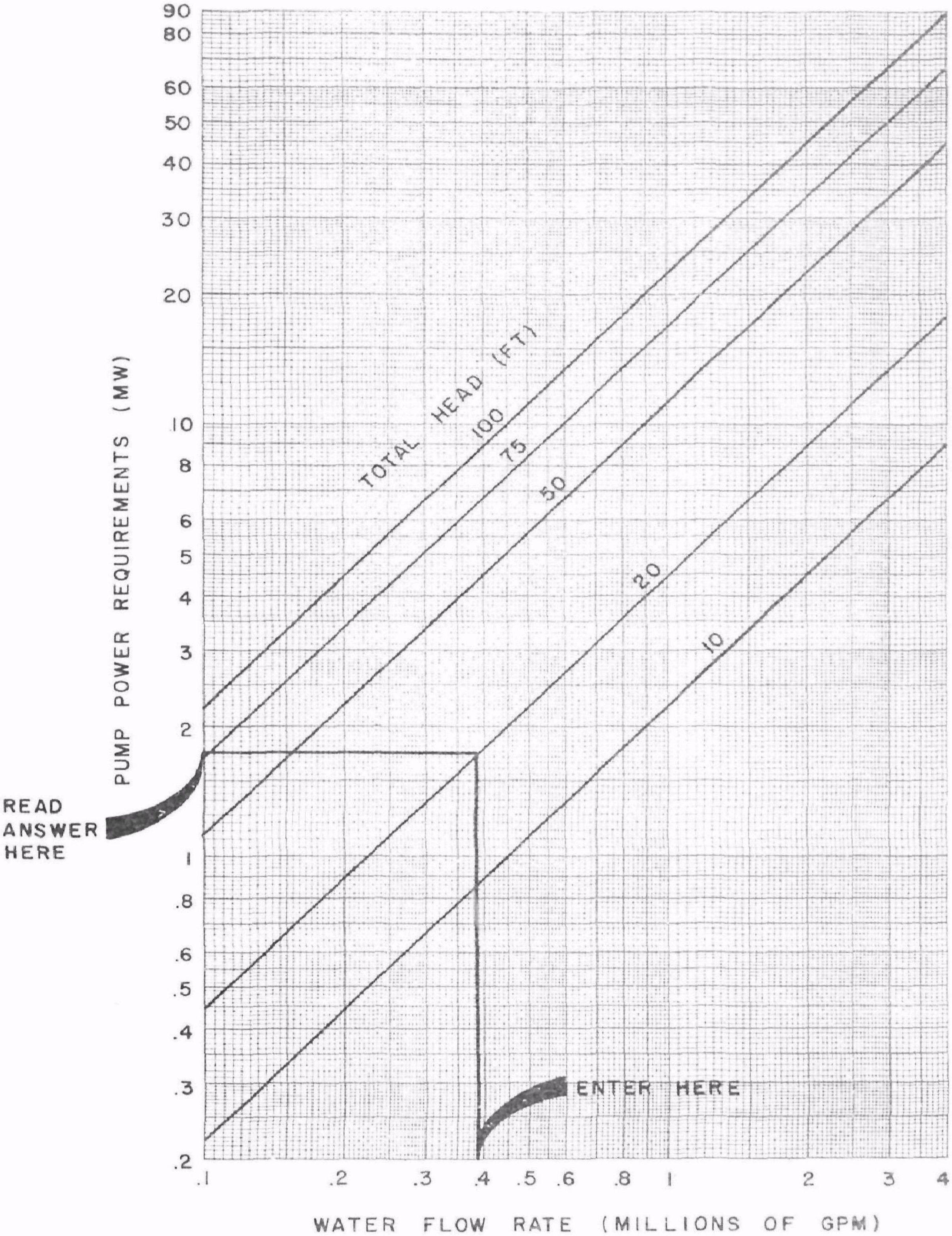
COOLING PONDS (CONTINUED)

FLOW RATE



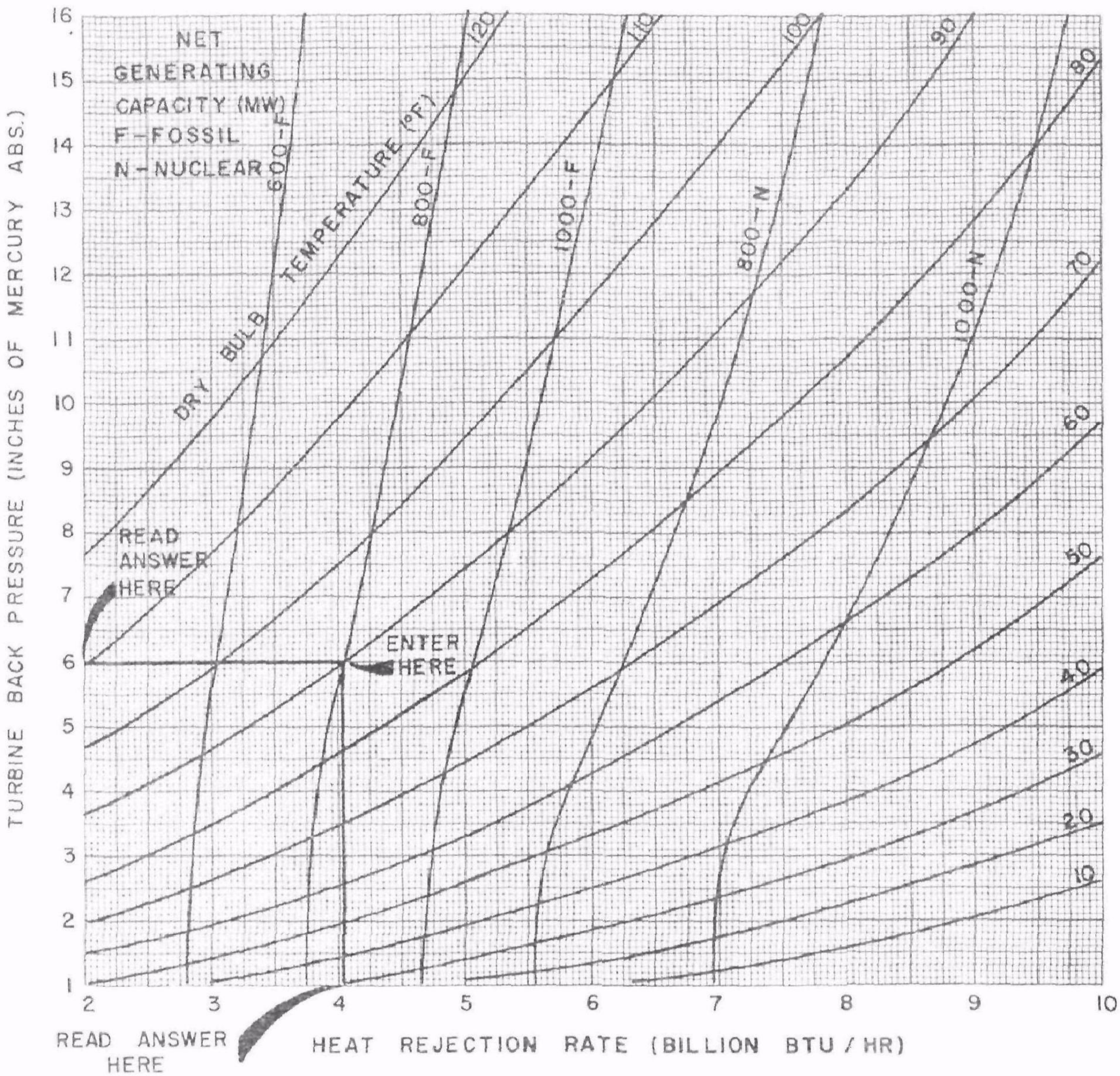
COOLING PONDS (CONTINUED)

AUXILIARY POWER



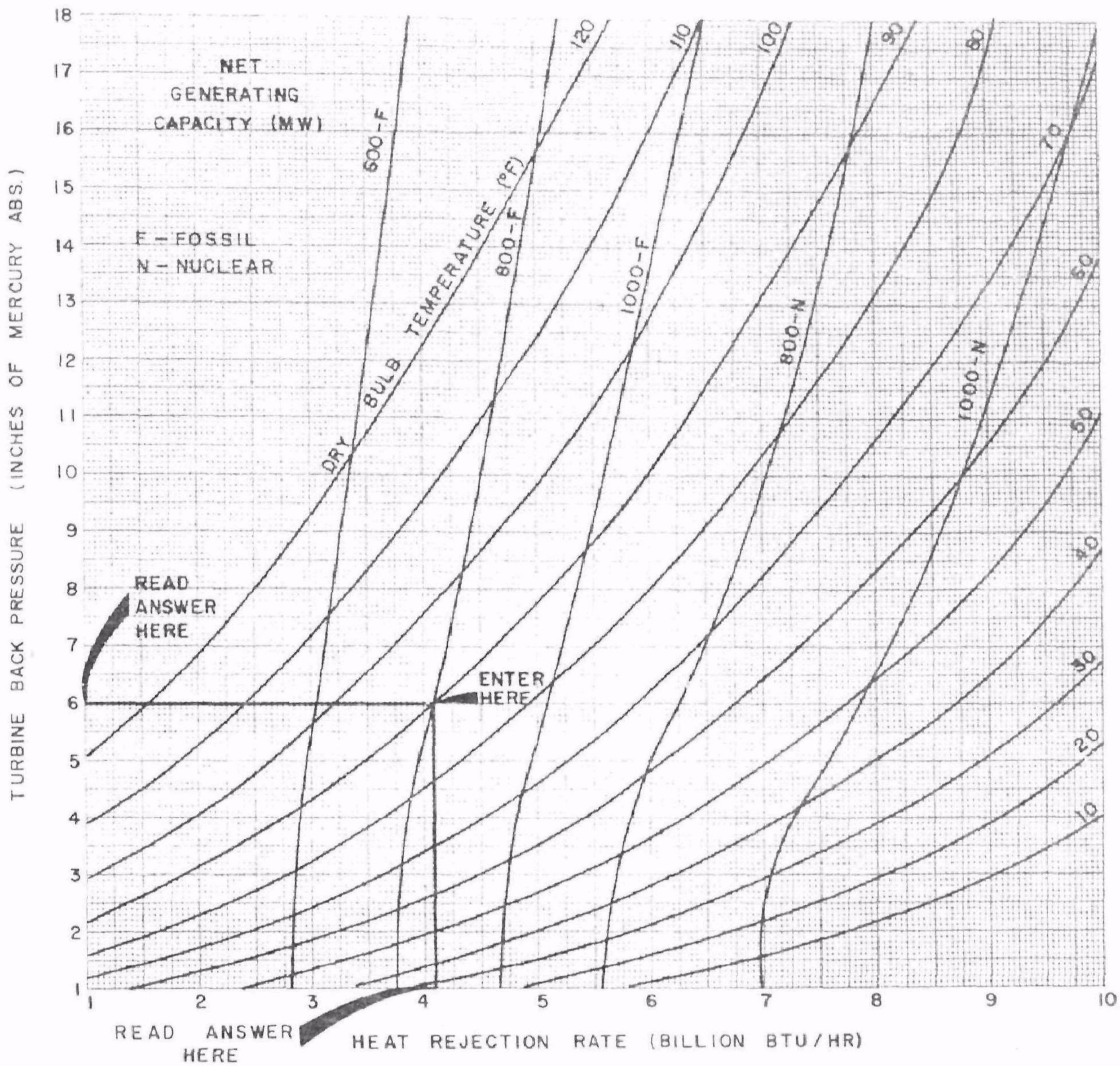
COOLING PONDS (CONTINUED)

COMBINED PERFORMANCE - NATURAL DRAFT



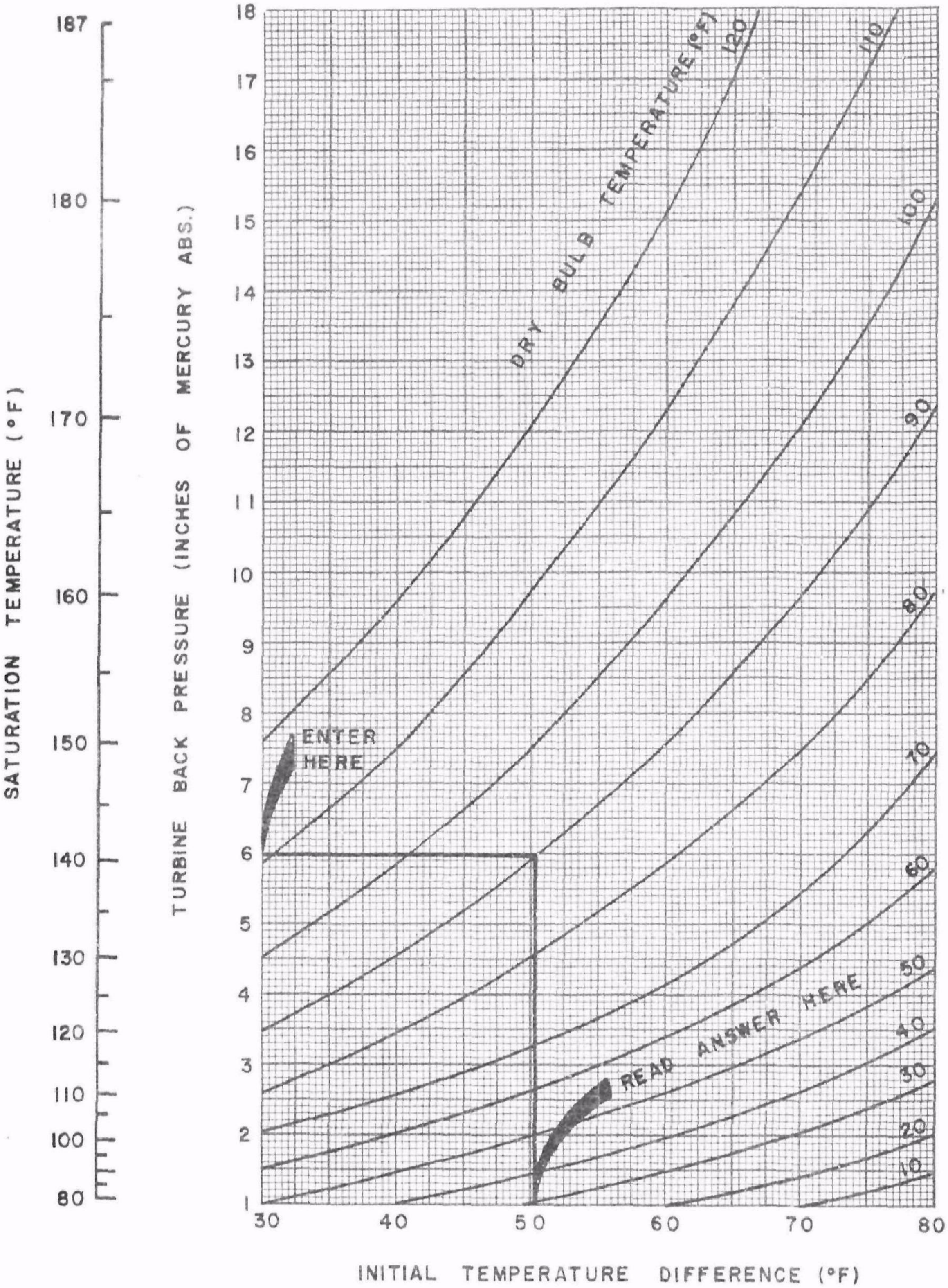
DRY TOWERS

COMBINED PERFORMANCE - MECHANICAL DRAFT



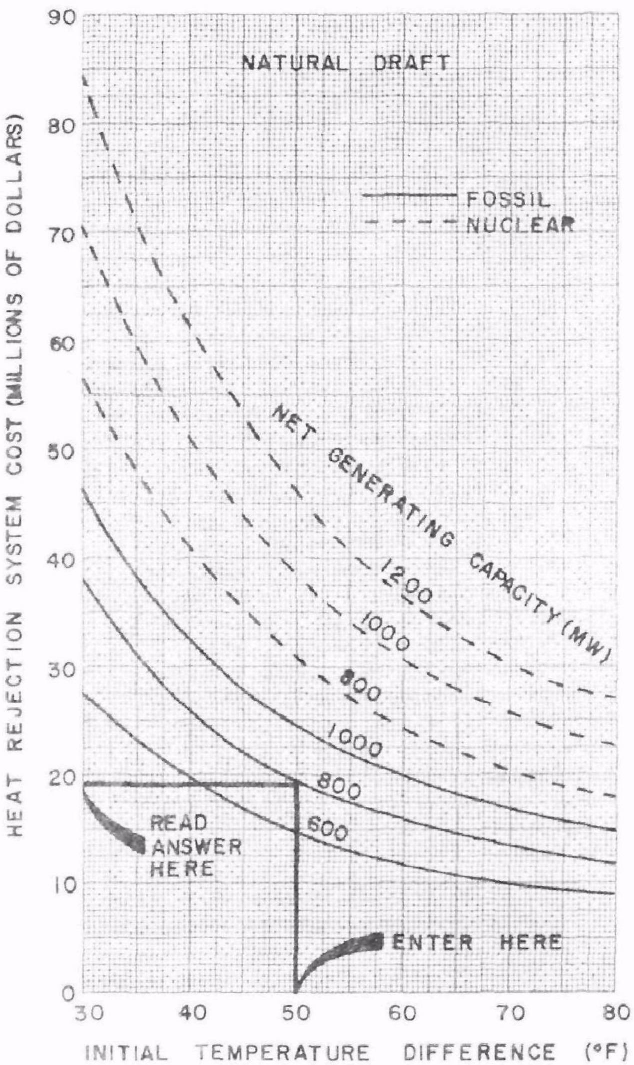
DRY TOWERS (CONTINUED)

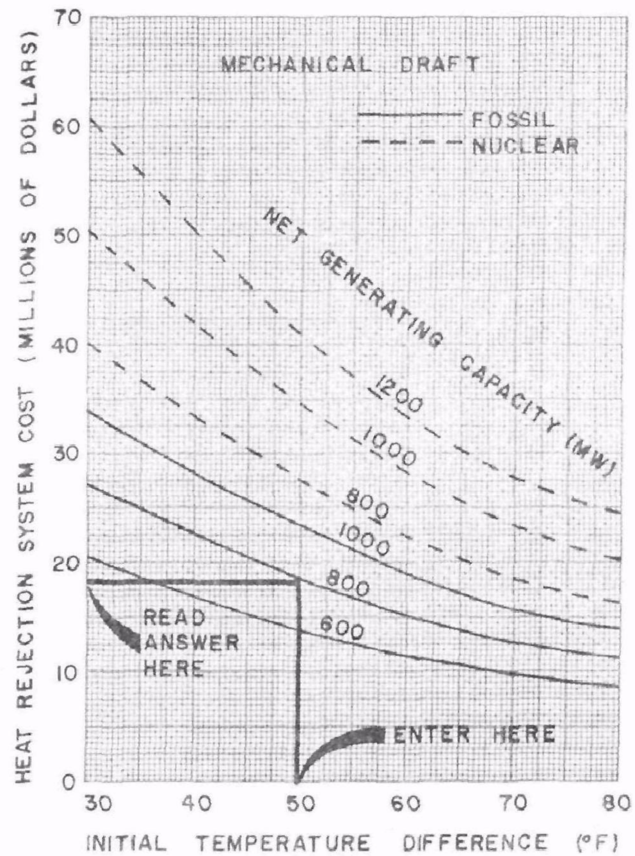
INITIAL TEMPERATURE DIFFERENCE



DRY TOWERS (CONTINUED)

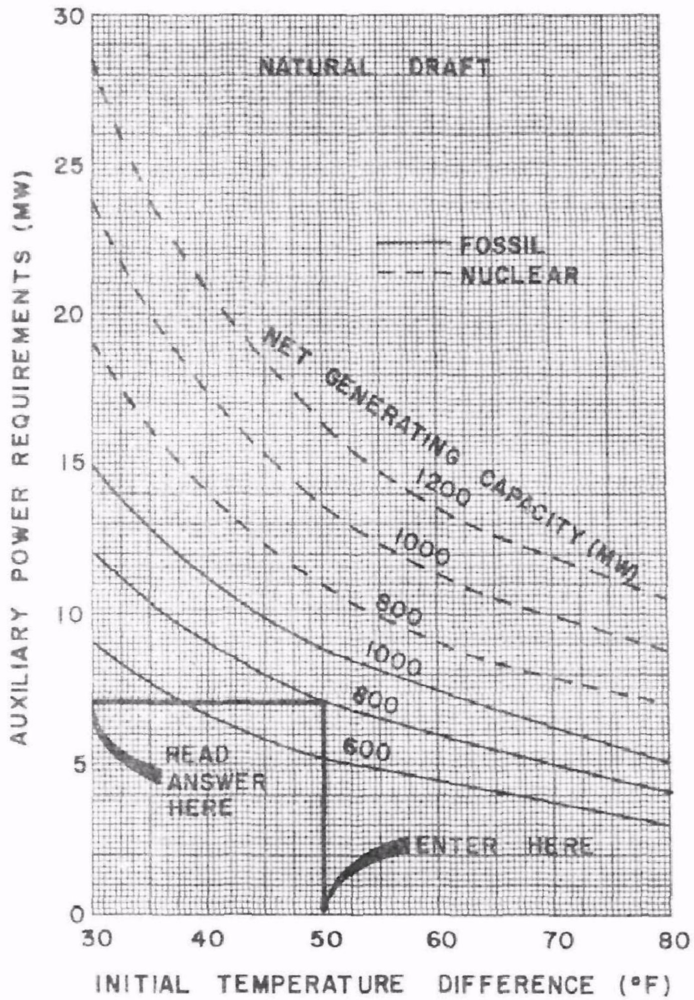
CAPITAL COST

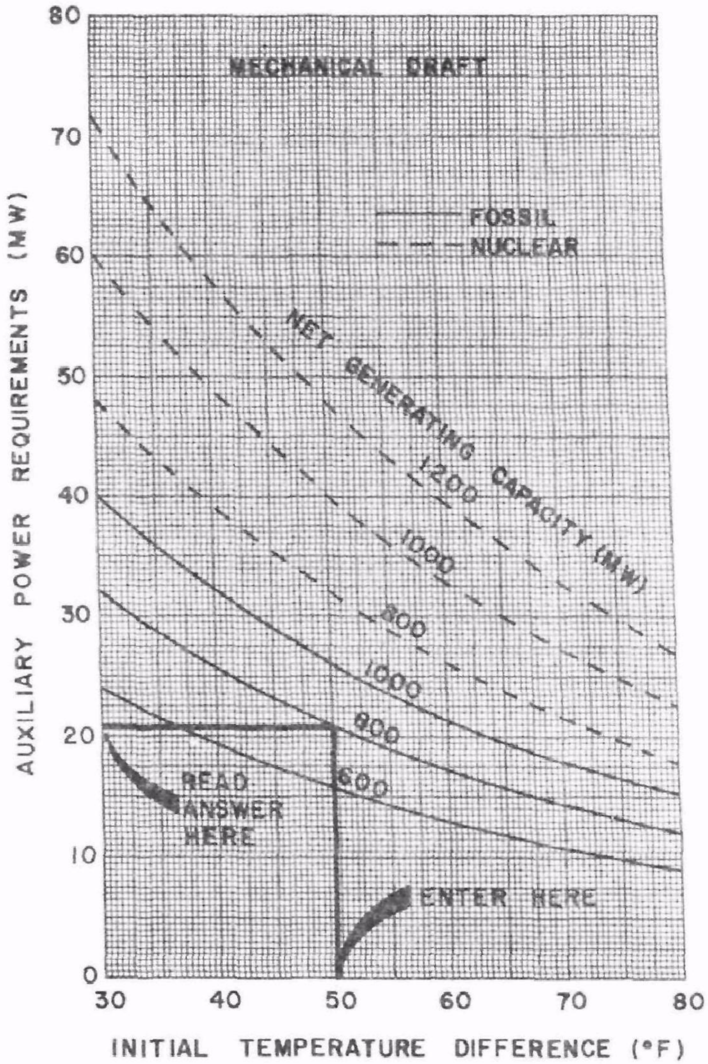




DRY TOWERS (CONTINUED)

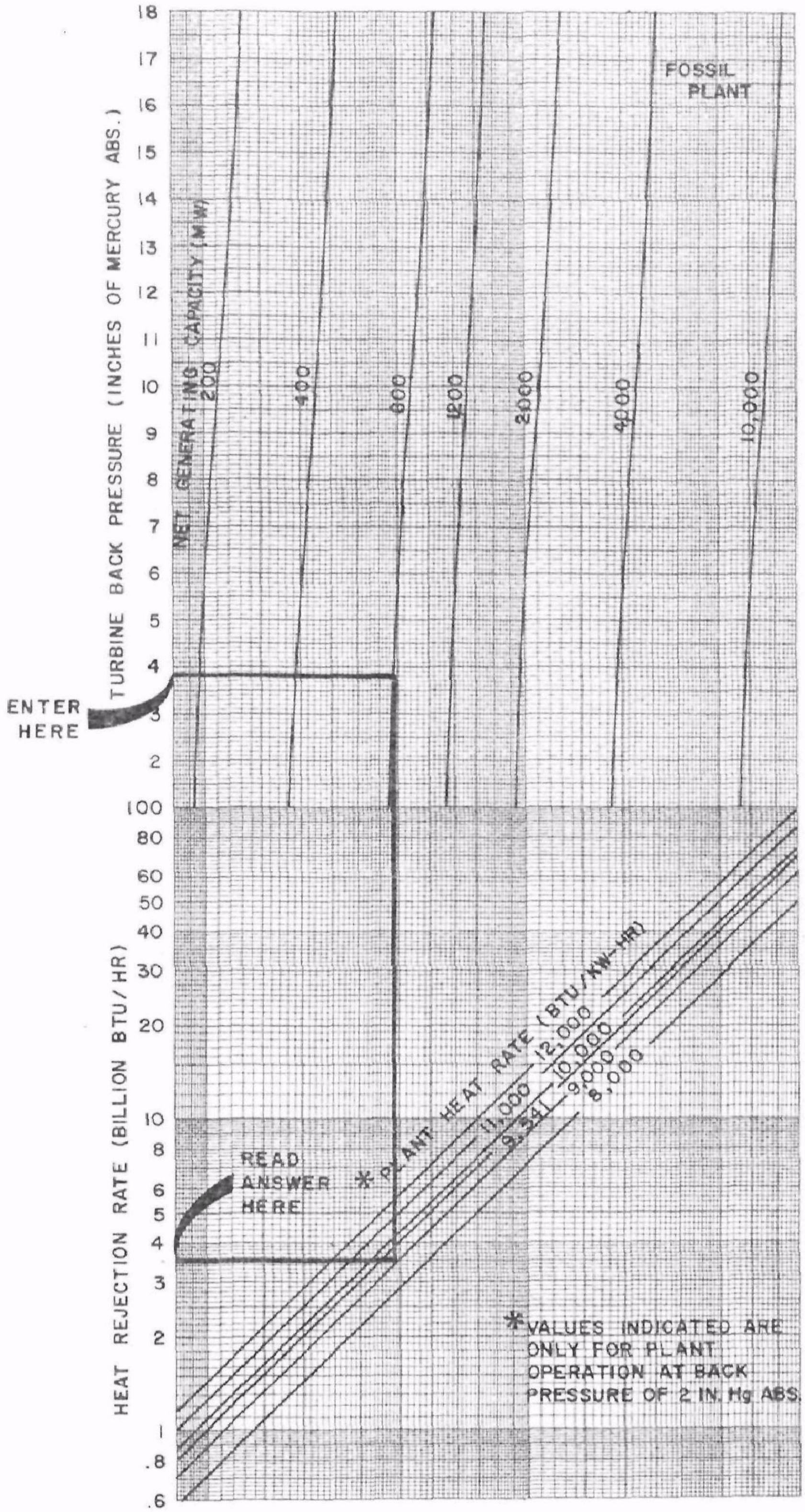
AUXILIARY POWER





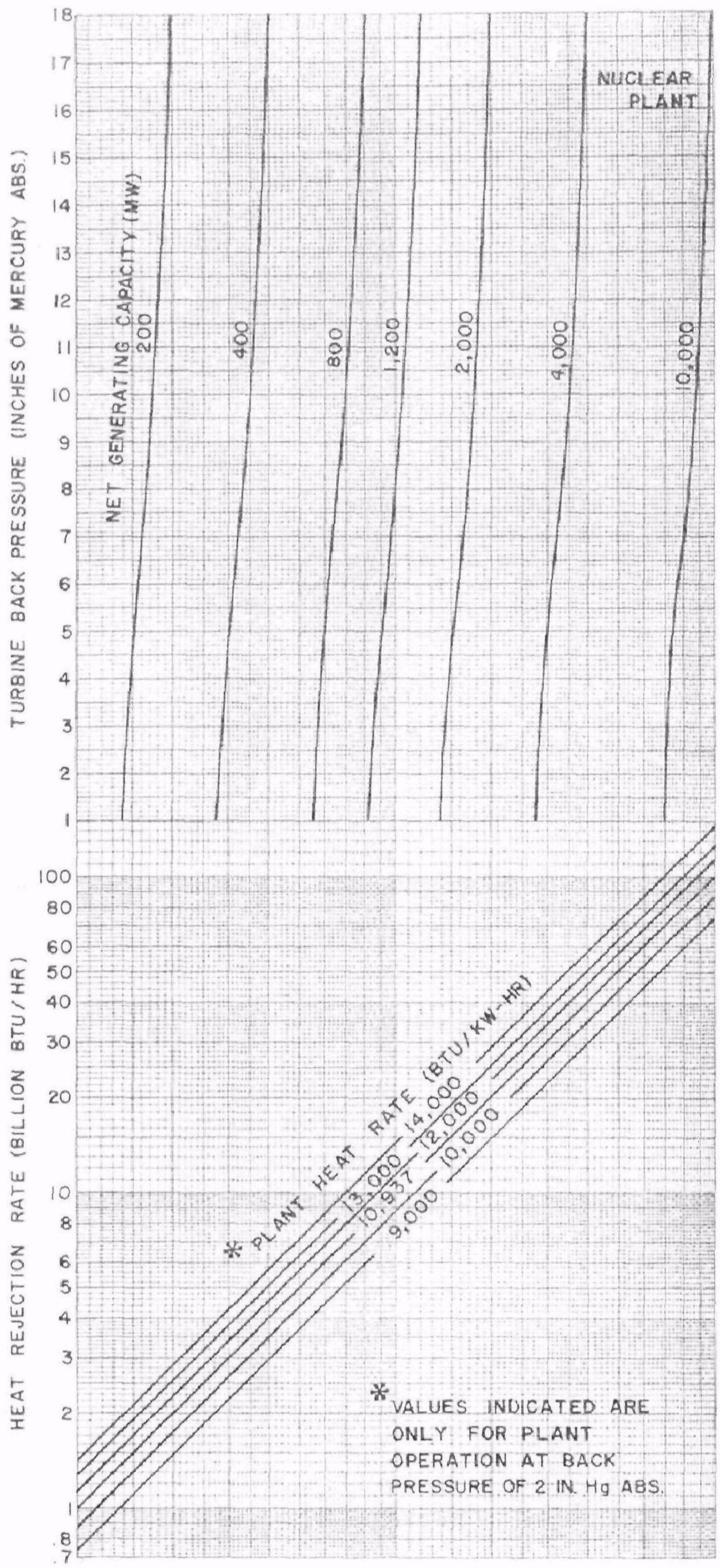
DRY TOWERS (CONTINUED)

HEAT REJECTION RATE



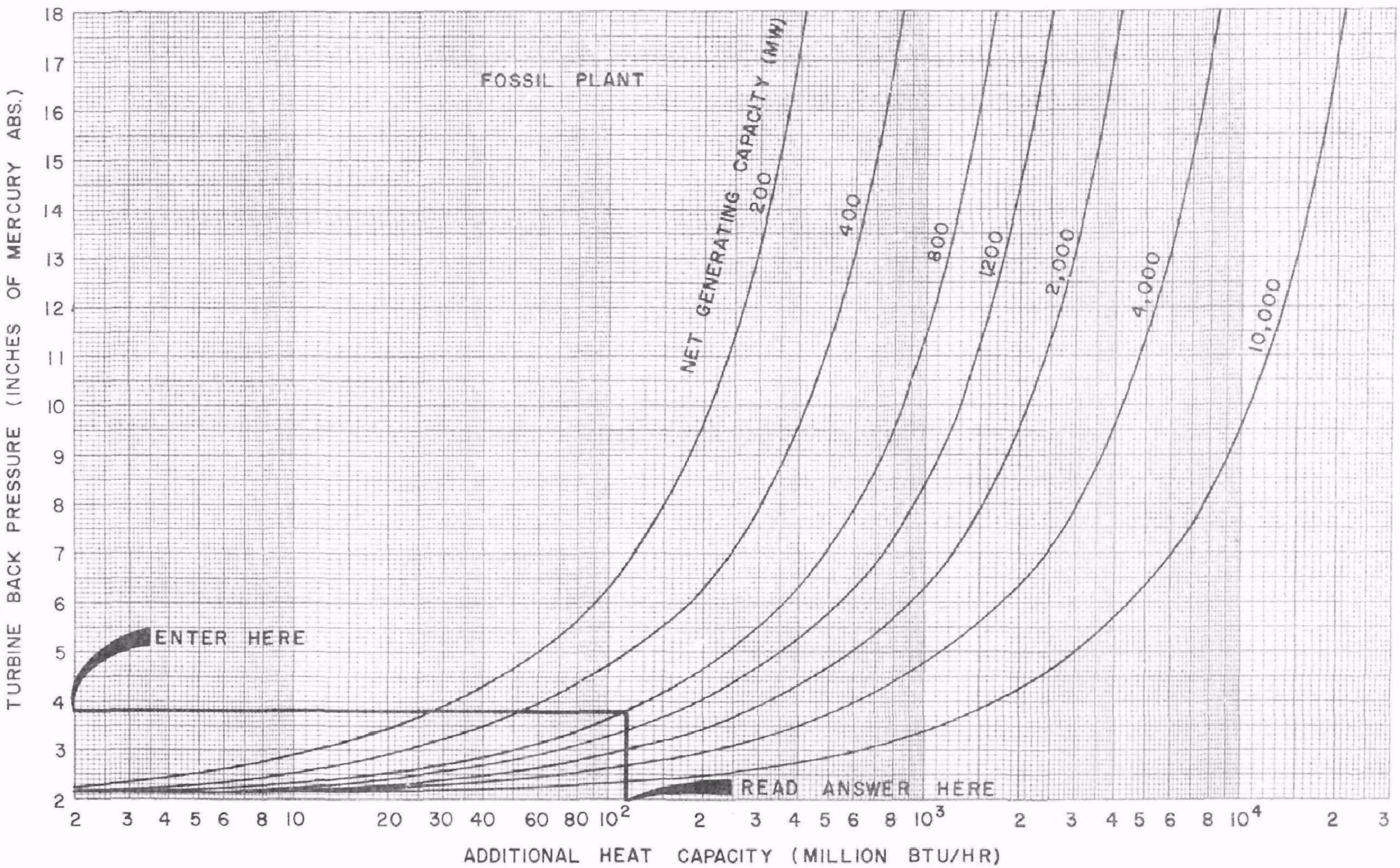
TURBINE PERFORMANCE

HEAT REJECTION RATE



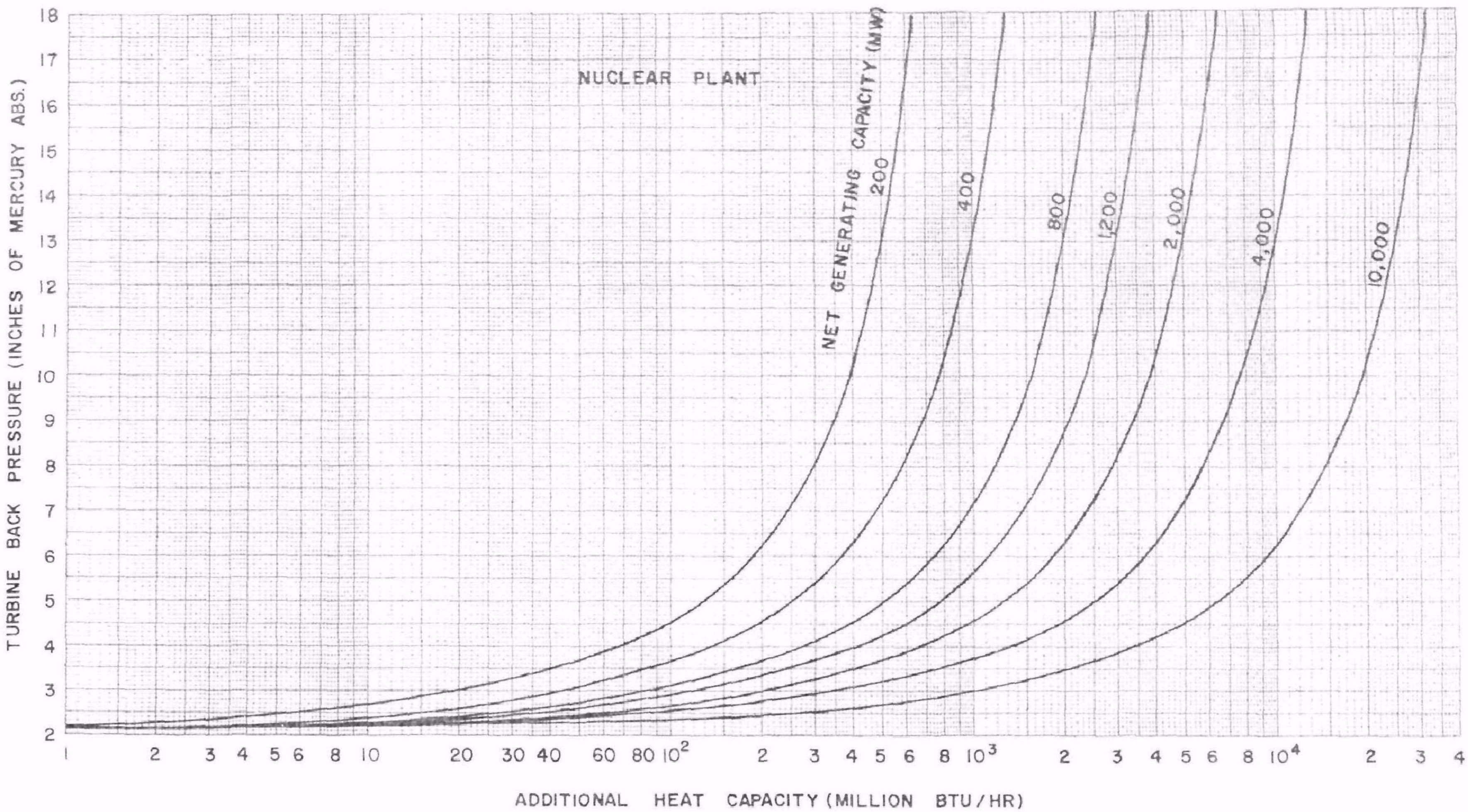
TURBINE PERFORMANCE (CONTINUED)

ADDITIONAL HEAT CAPACITY



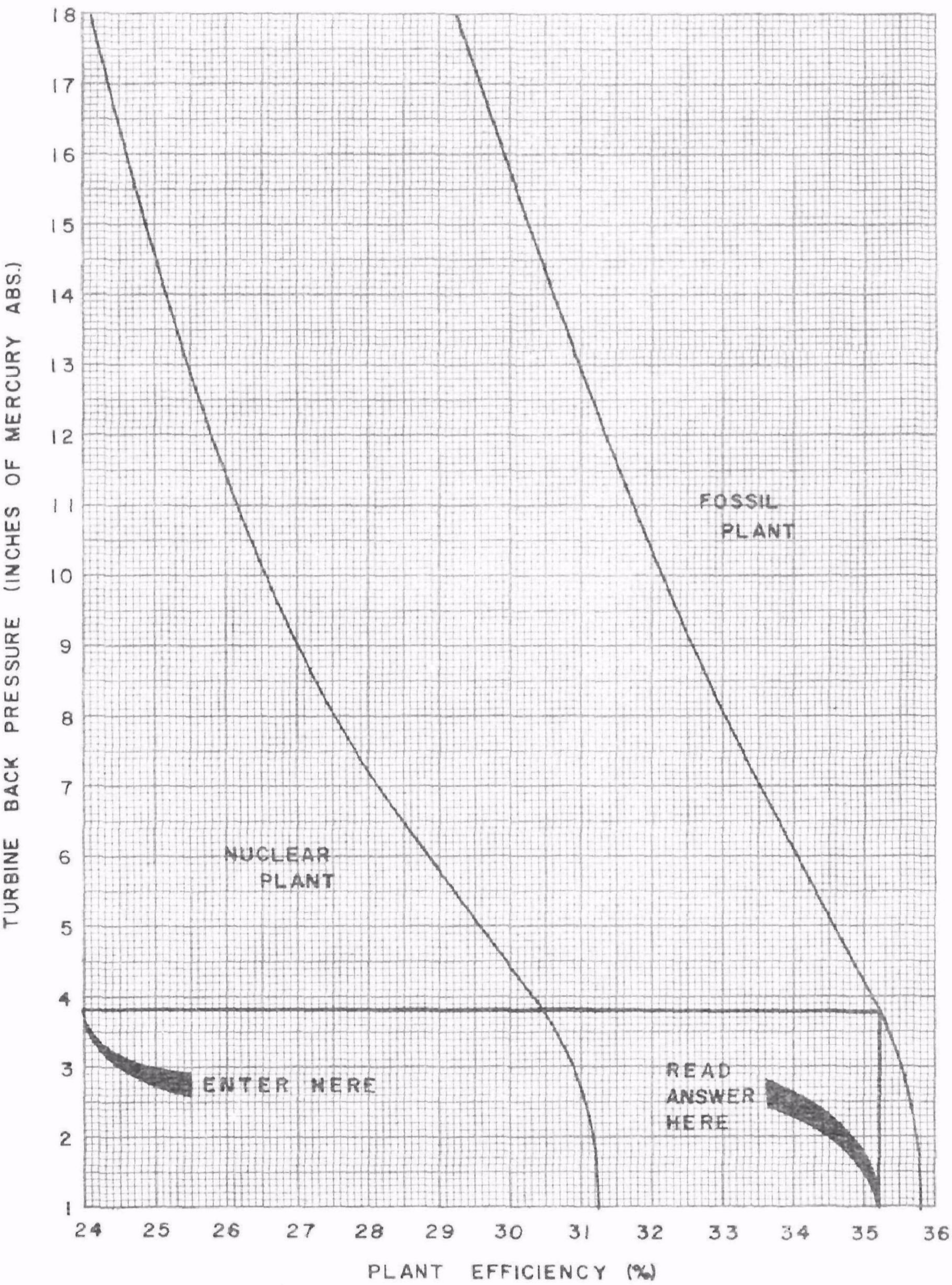
TURBINE PERFORMANCE (CONTINUED)

ADDITIONAL HEAT CAPACITY

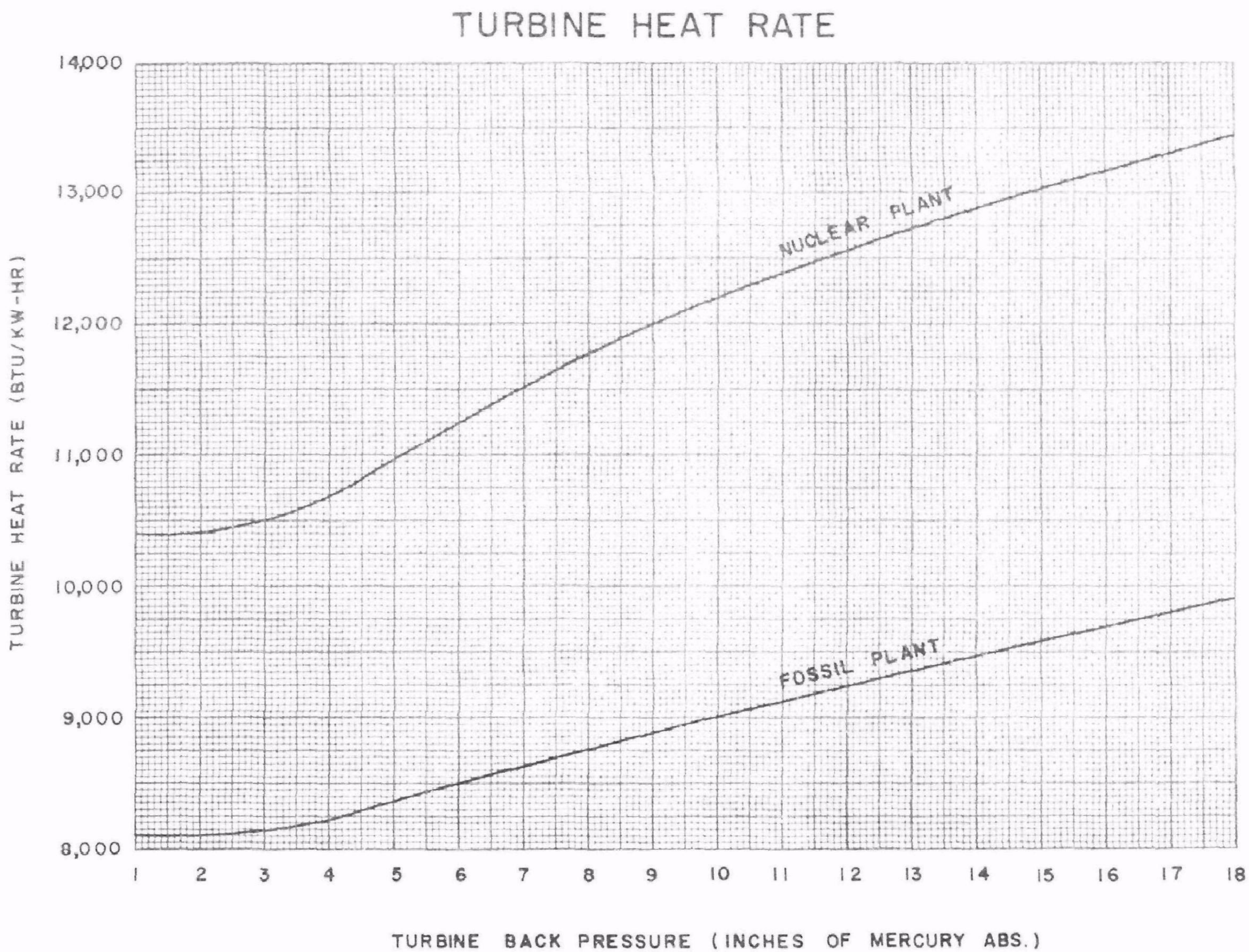


TURBINE PERFORMANCE (CONTINUED)

PLANT EFFICIENCY

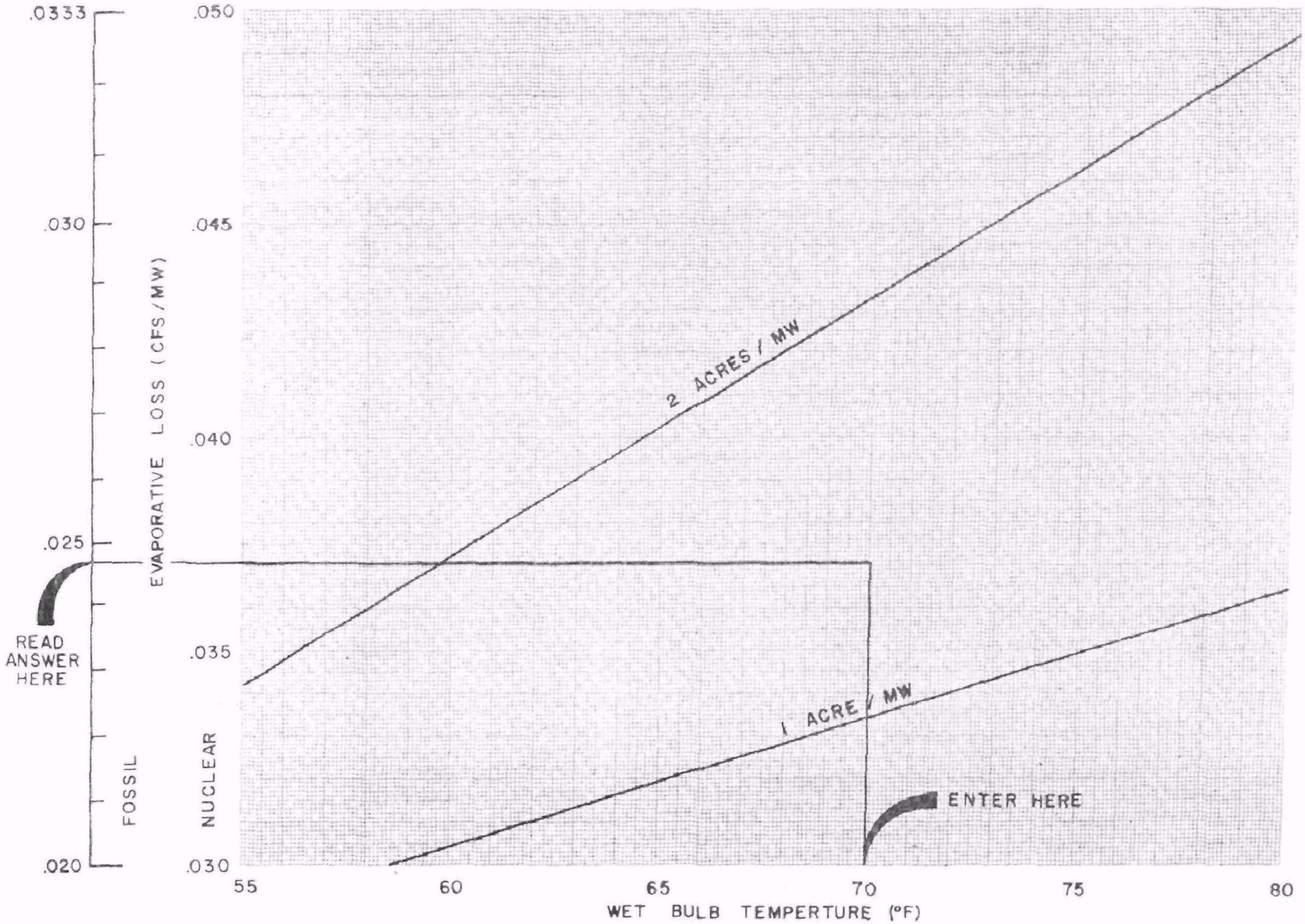


TURBINE PERFORMANCE (CONTINUED)



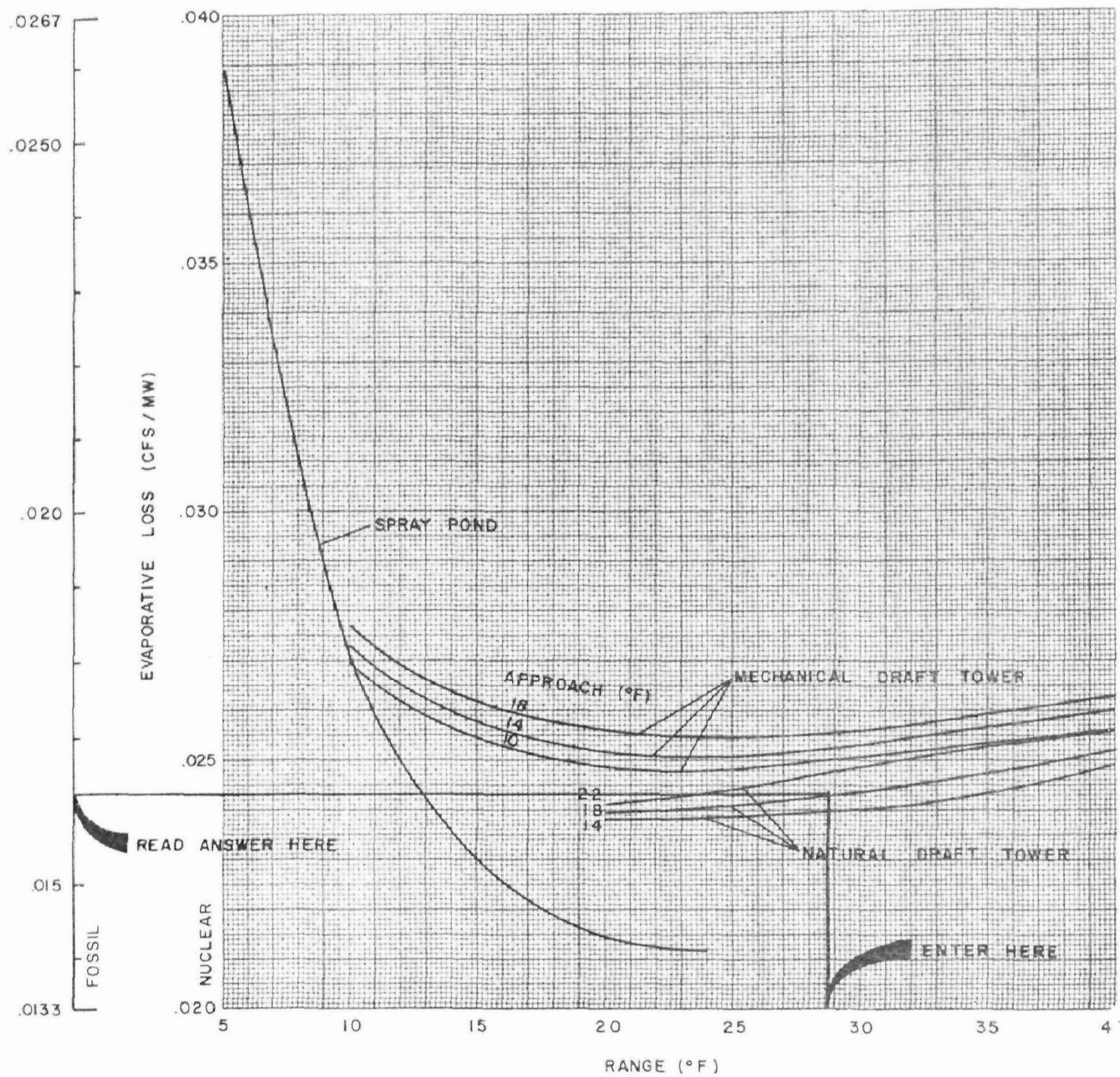
TURBINE PERFORMANCE (CONTINUED)

COOLING POND EVAPORATION



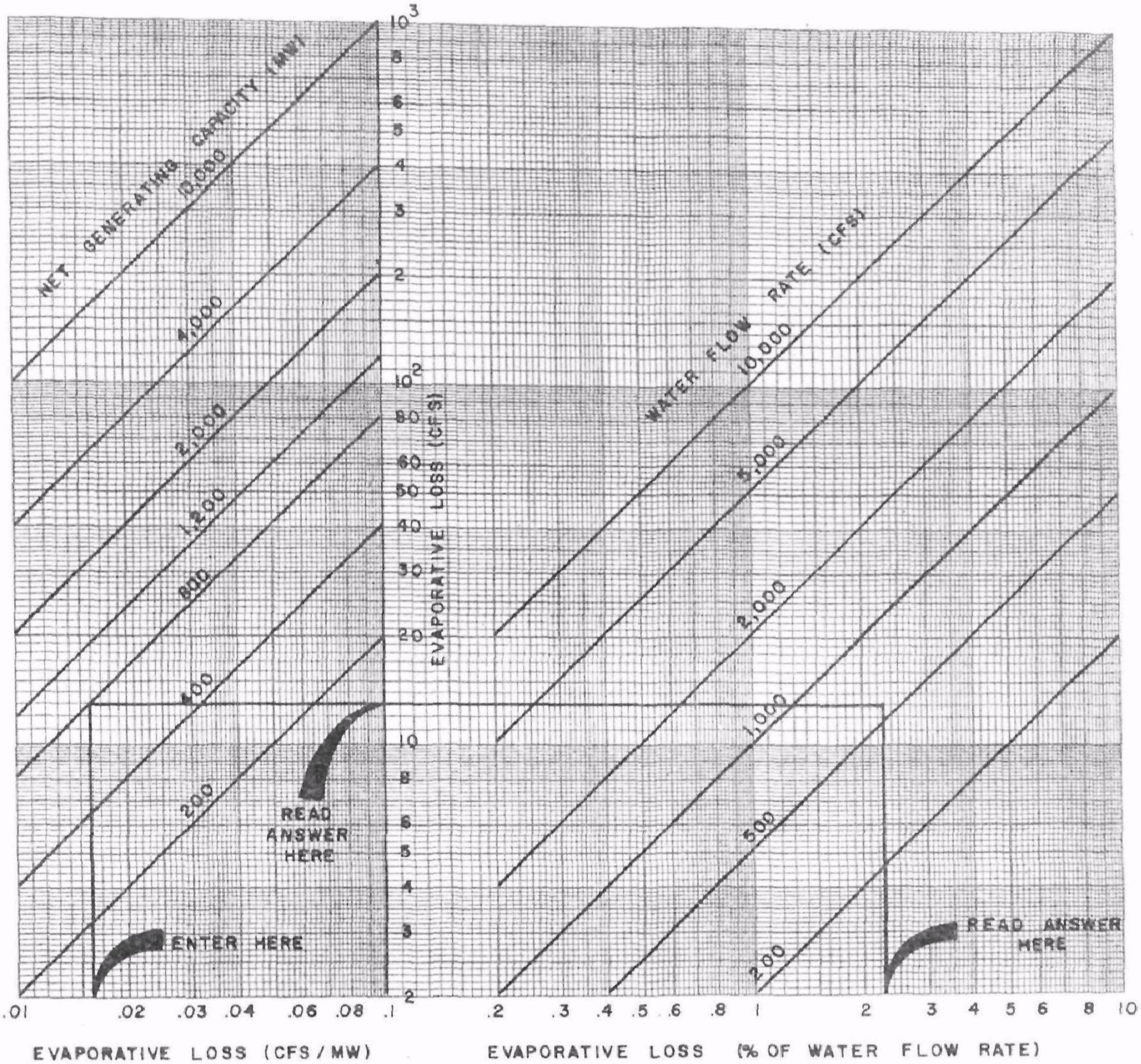
WATER REQUIREMENTS

EVAPORATIVE LOSS FOR WET SYSTEMS



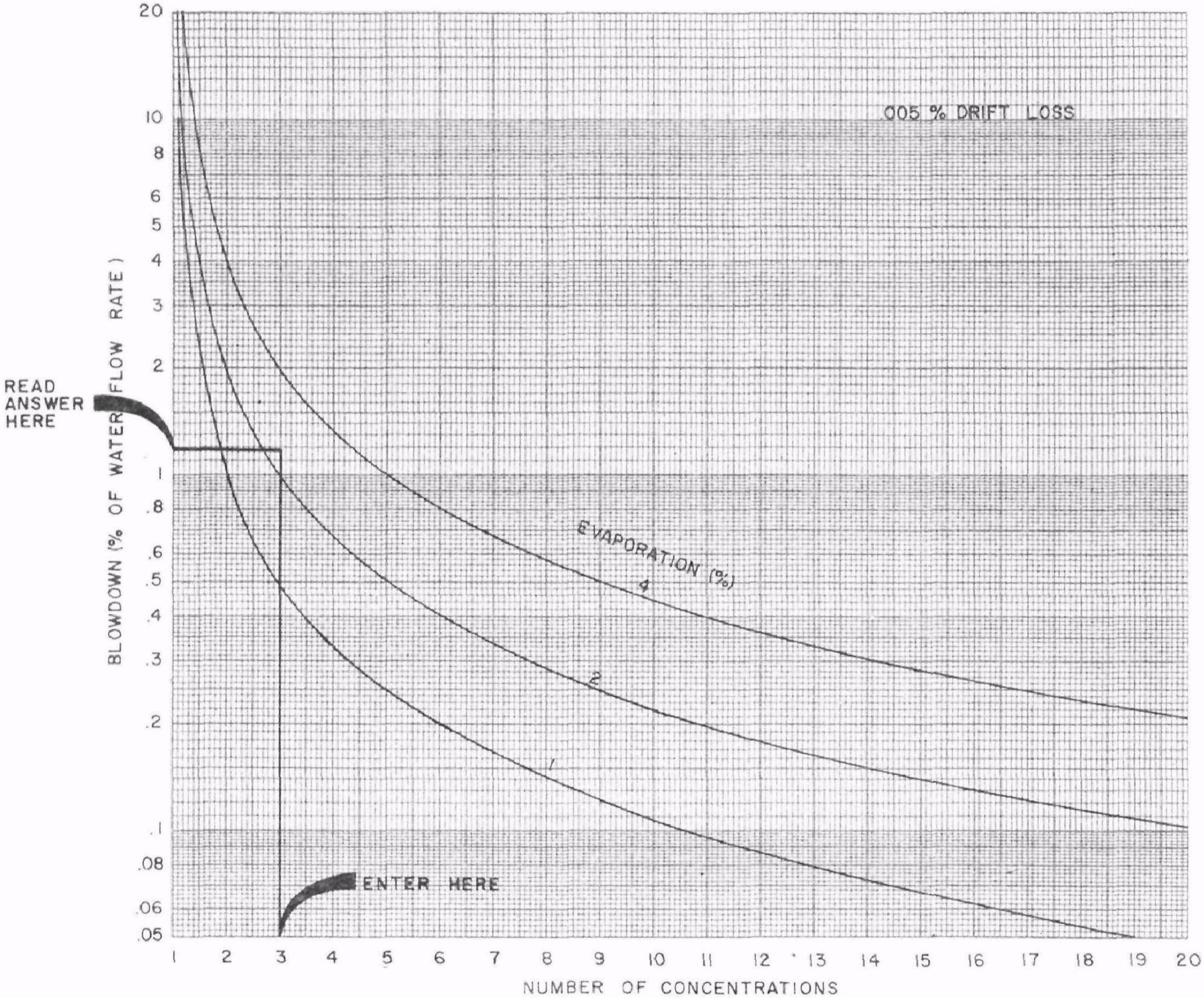
WATER REQUIREMENTS (CONTINUED)

% EVAPORATIVE LOSS



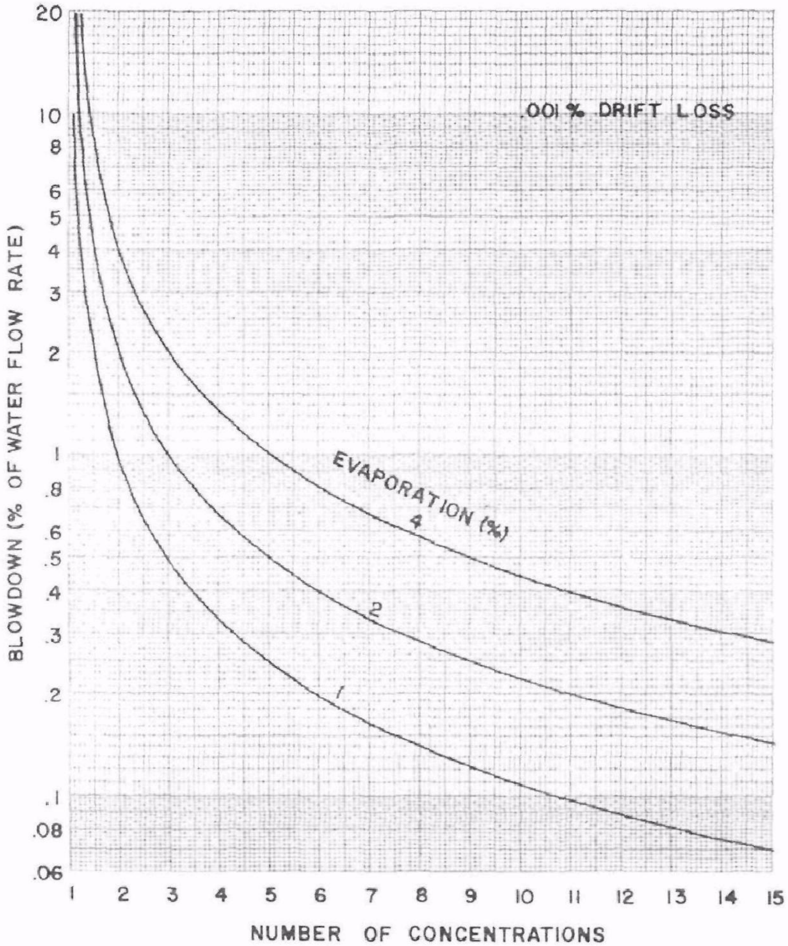
WATER REQUIREMENTS (CONTINUED)

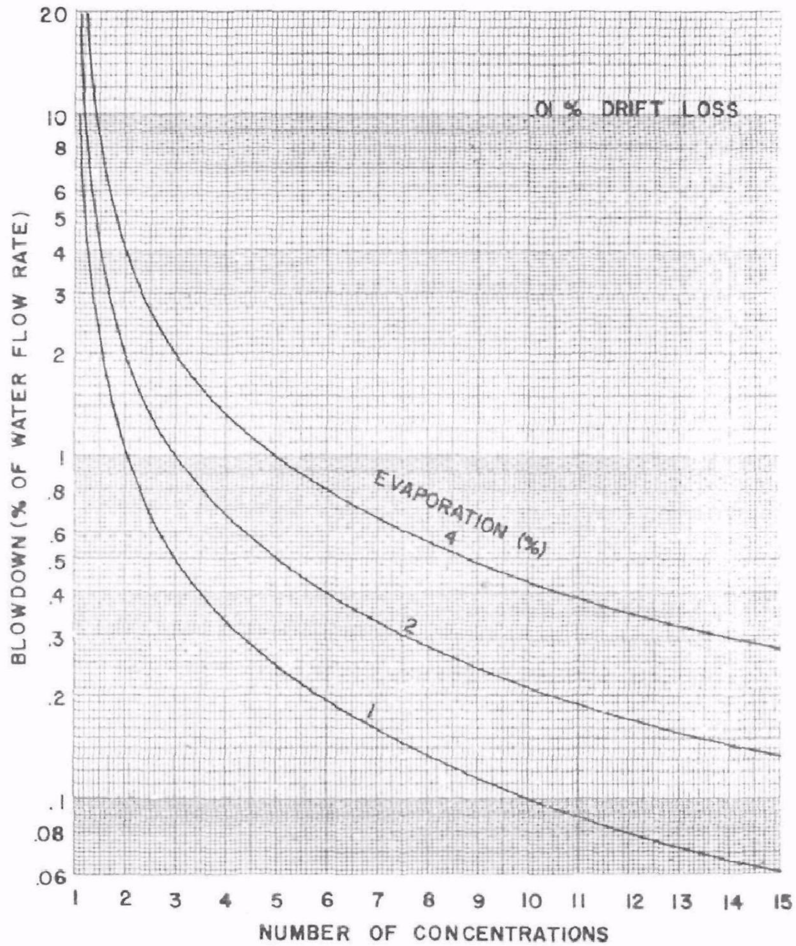
% BLOWDOWN



WATER REQUIREMENTS (CONTINUED)

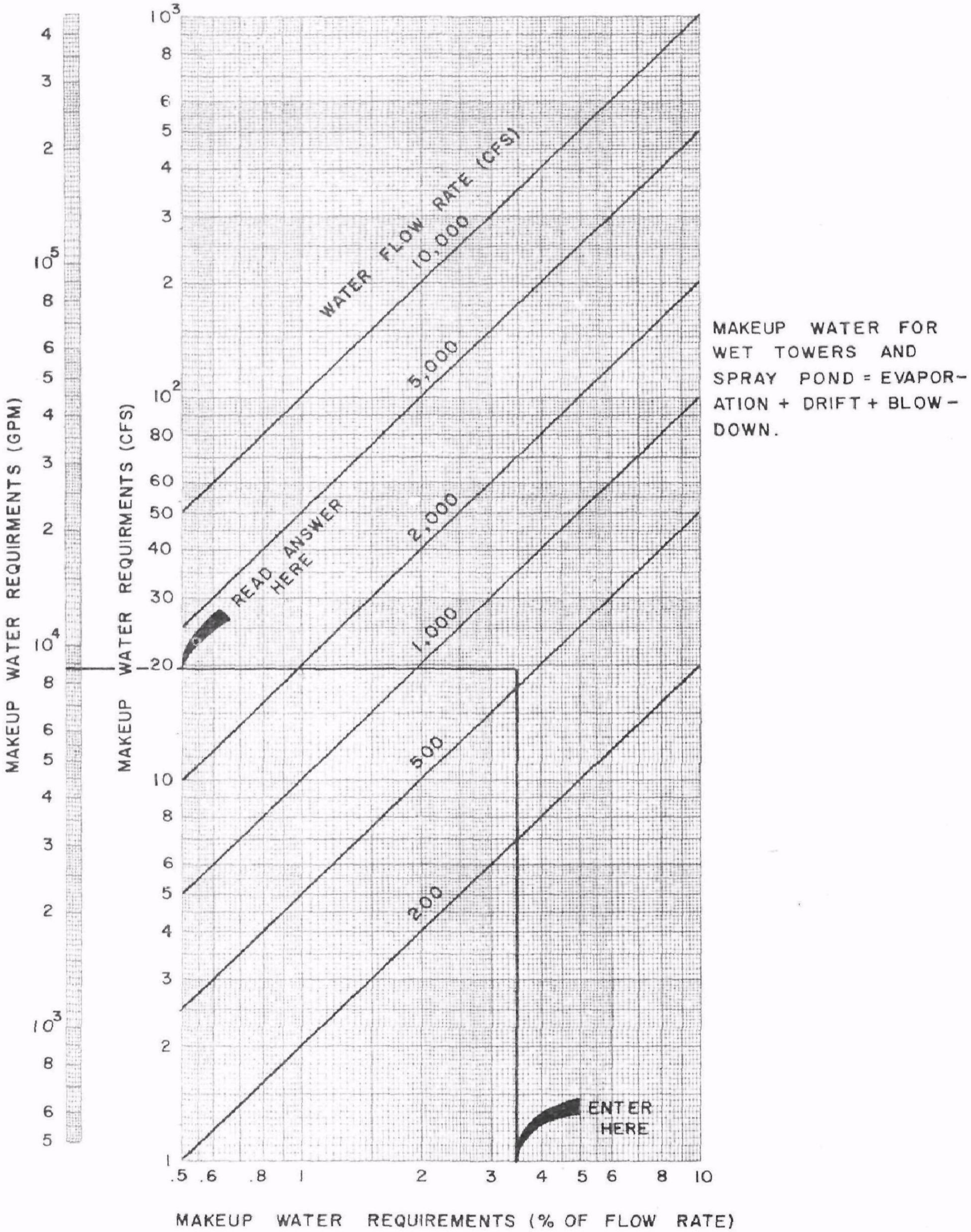
% BLOWDOWN





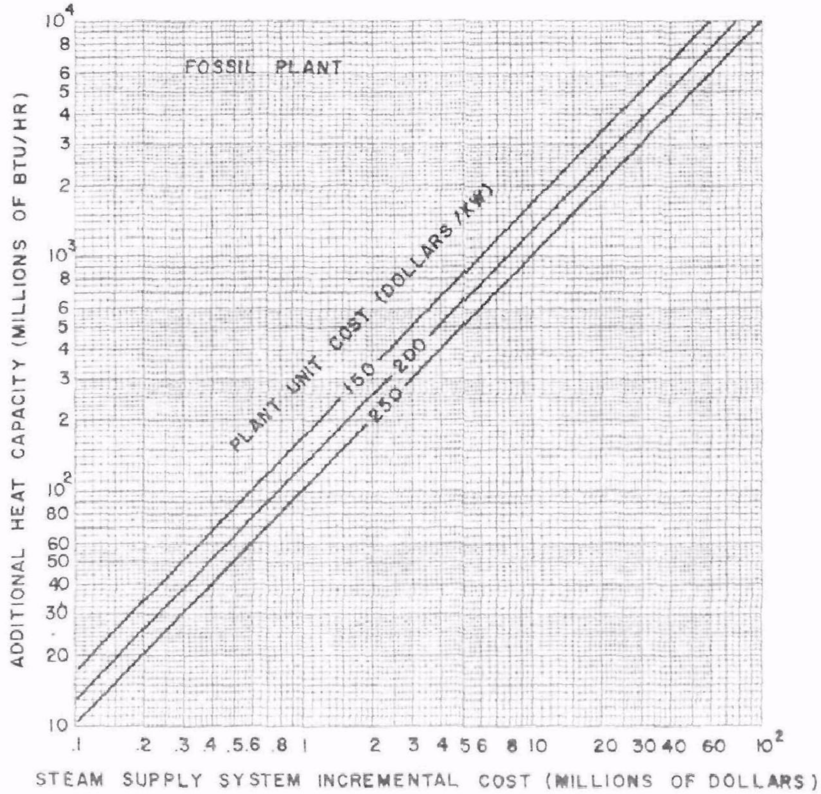
WATER REQUIREMENTS (CONTINUED)

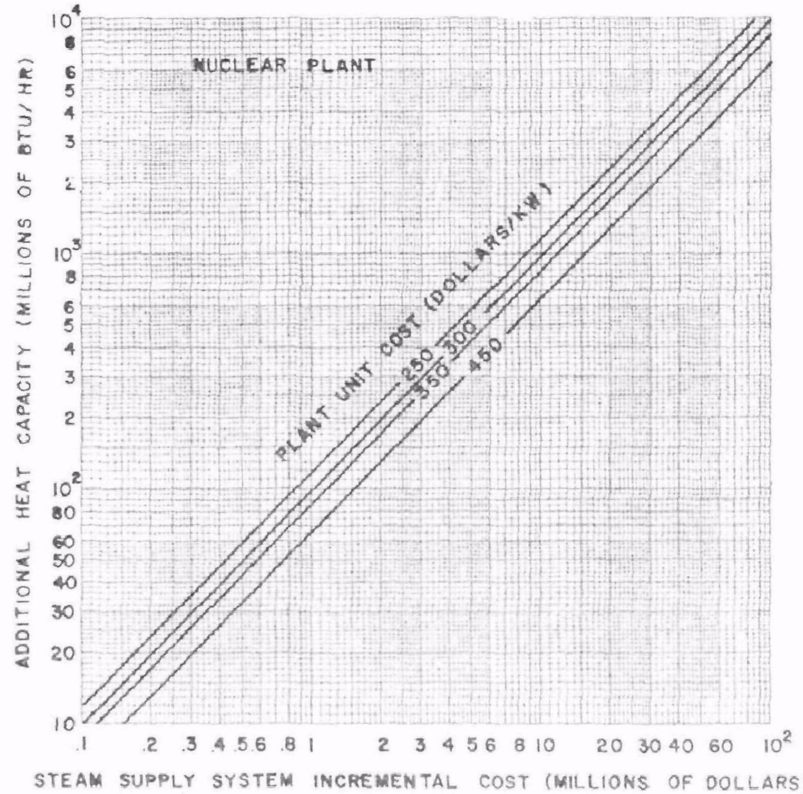
MAKEUP WATER



WATER REQUIREMENTS (CONTINUED)

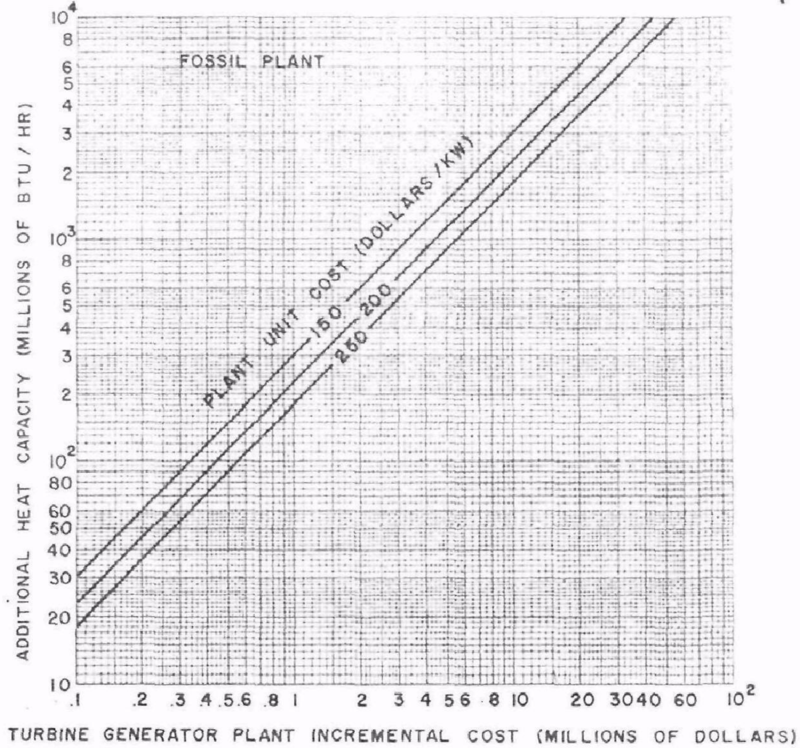
STEAM SUPPLY INCREMENTAL COST
(SSIC)

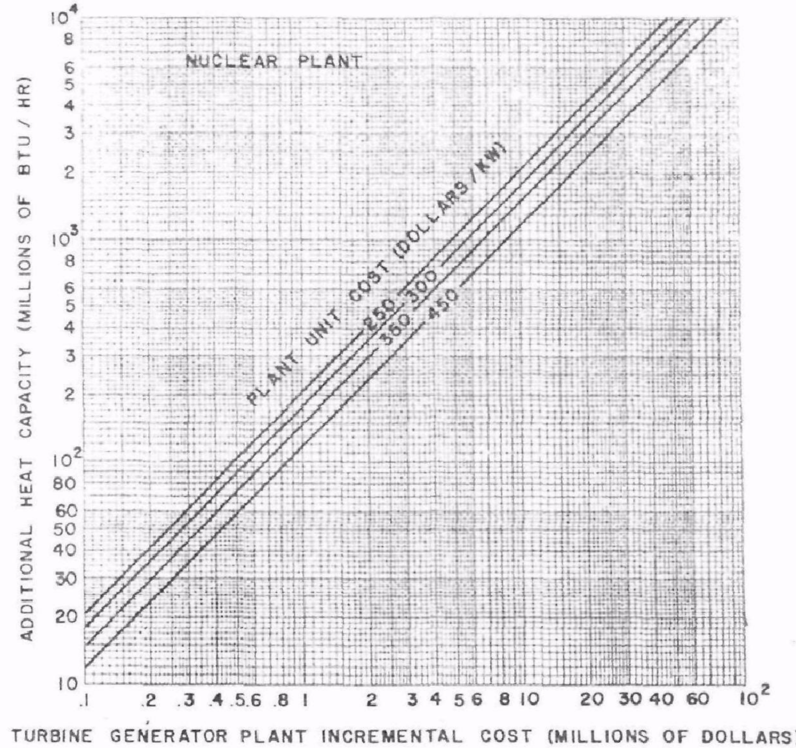




COSTS

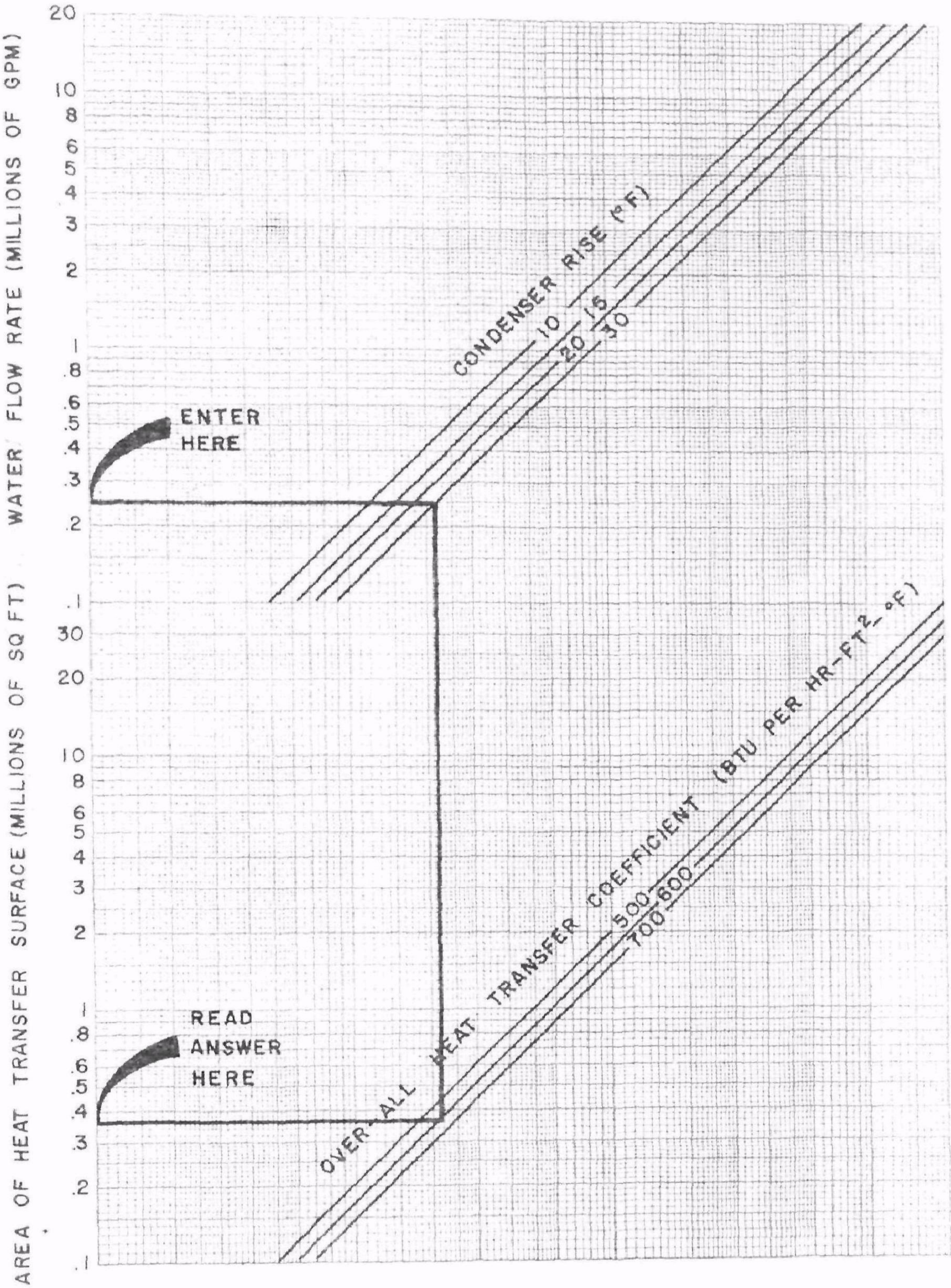
TURBINE GENERATOR INCREMENTAL COST
(TGIC)





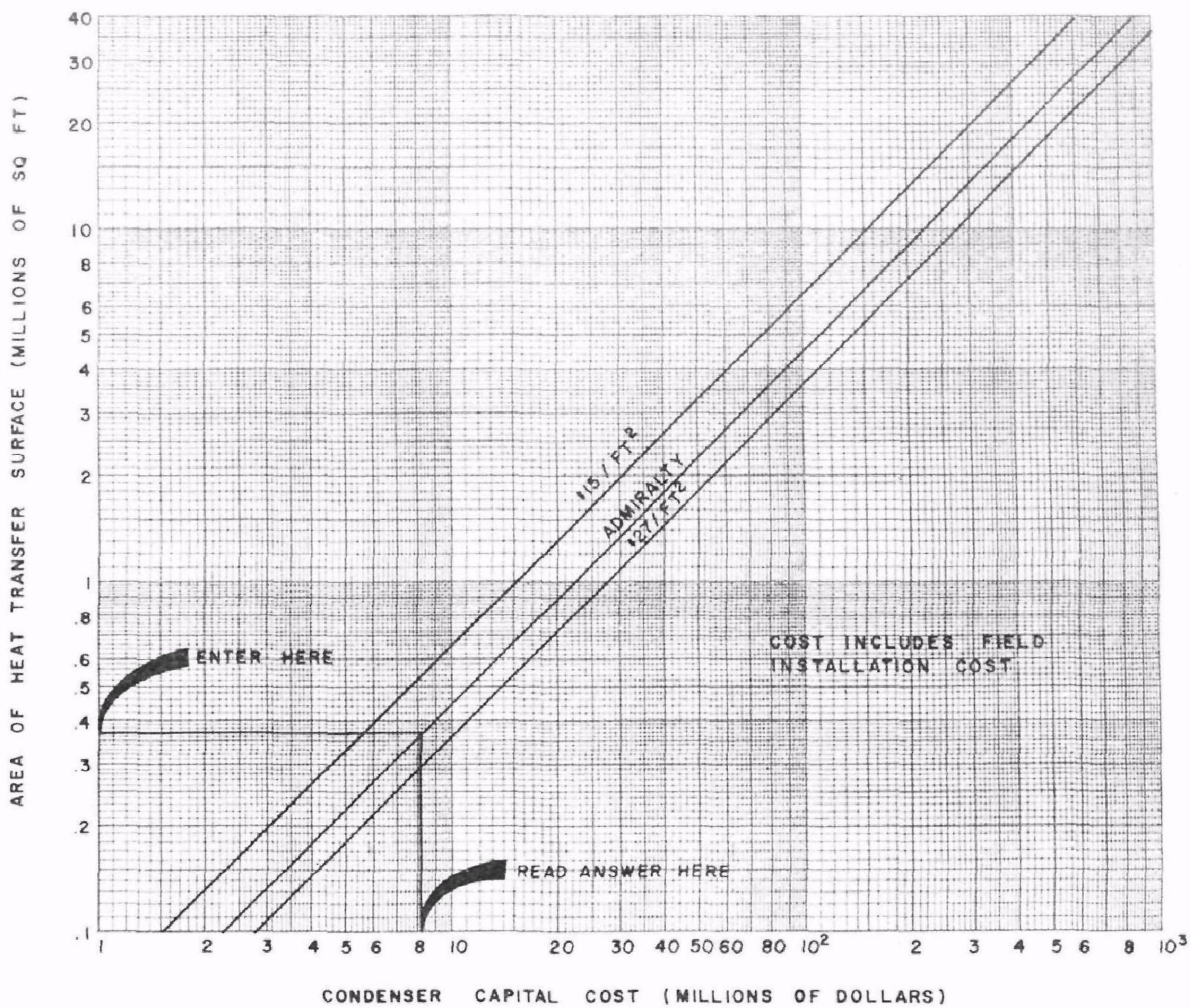
COSTS (CONTINUED)

CONDENSER SURFACE AREA



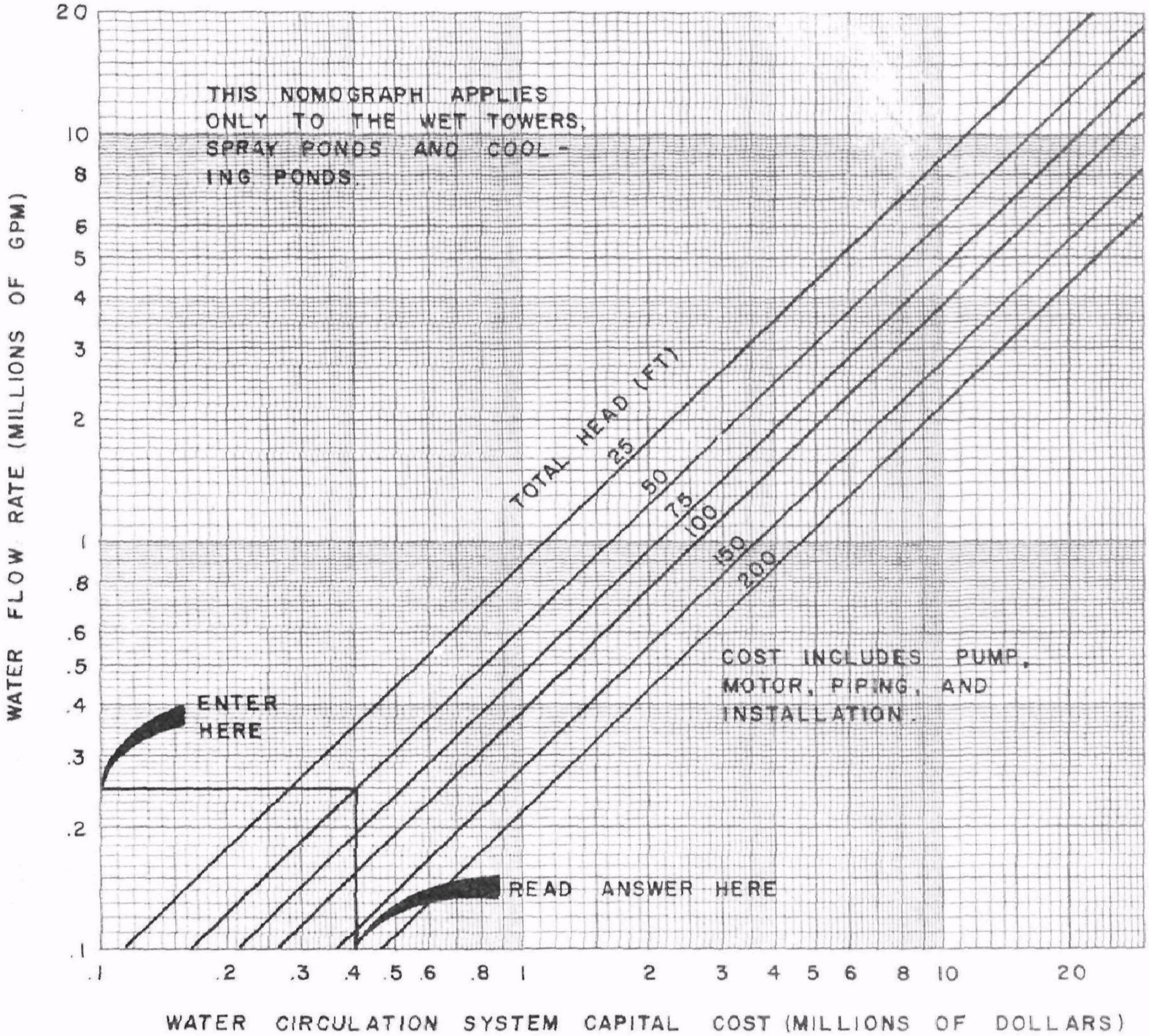
COSTS (CONTINUED)

CONDENSER COST
(CC)



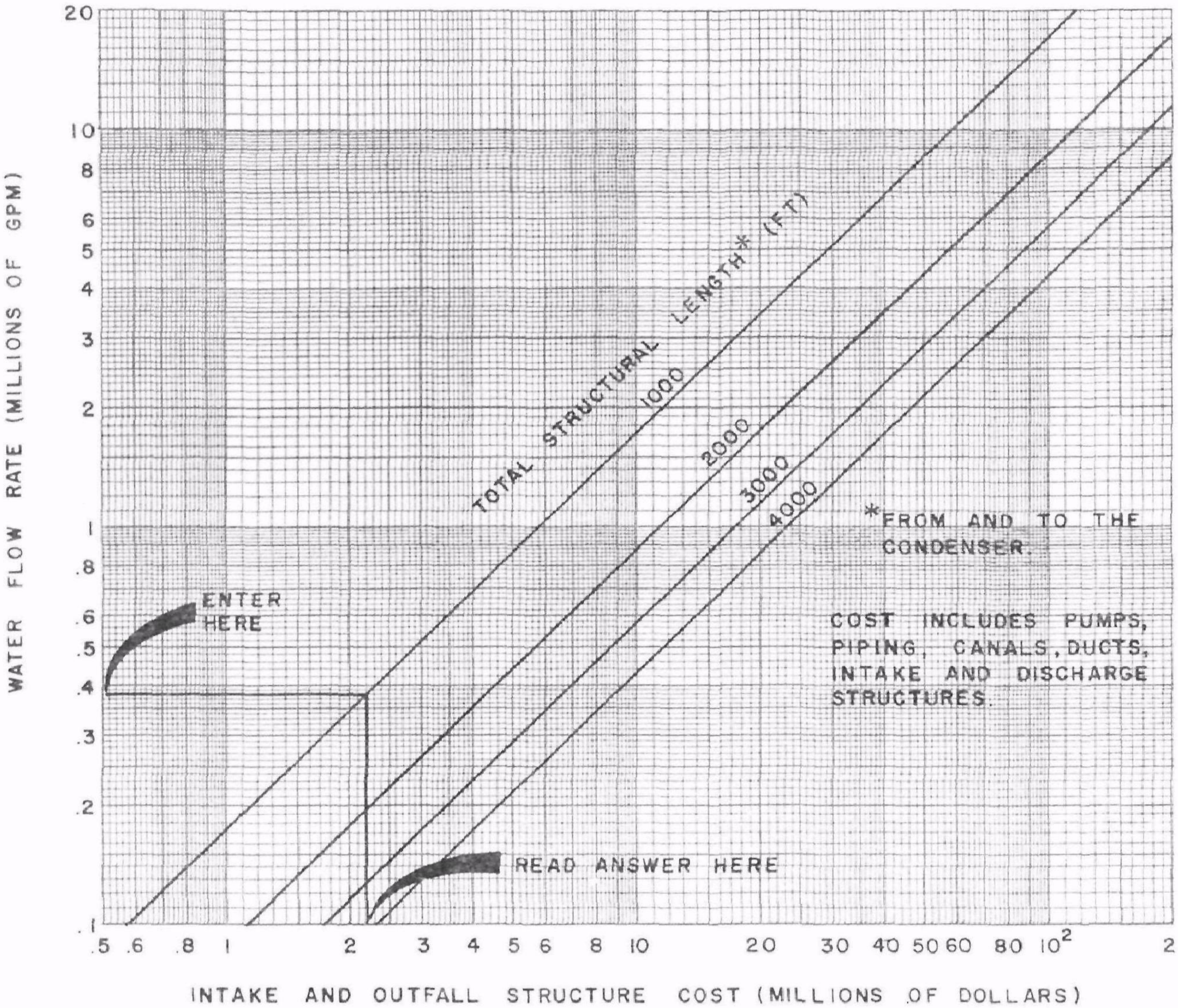
COSTS (CONTINUED)

WATER CIRCULATION SYSTEM COST (WCSC)



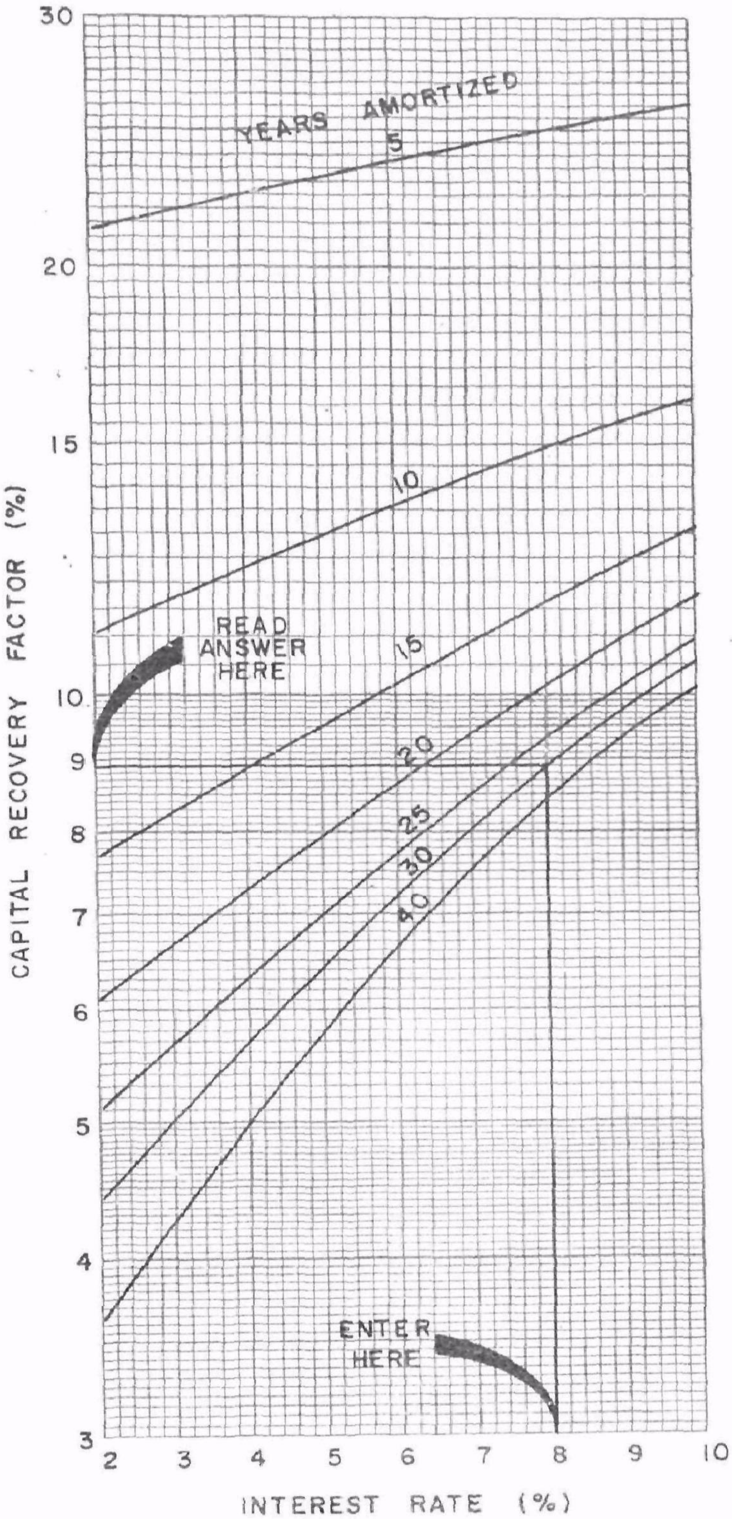
COSTS (CONTINUED)

COST FOR ONCE THROUGH COOLING (OTCC)



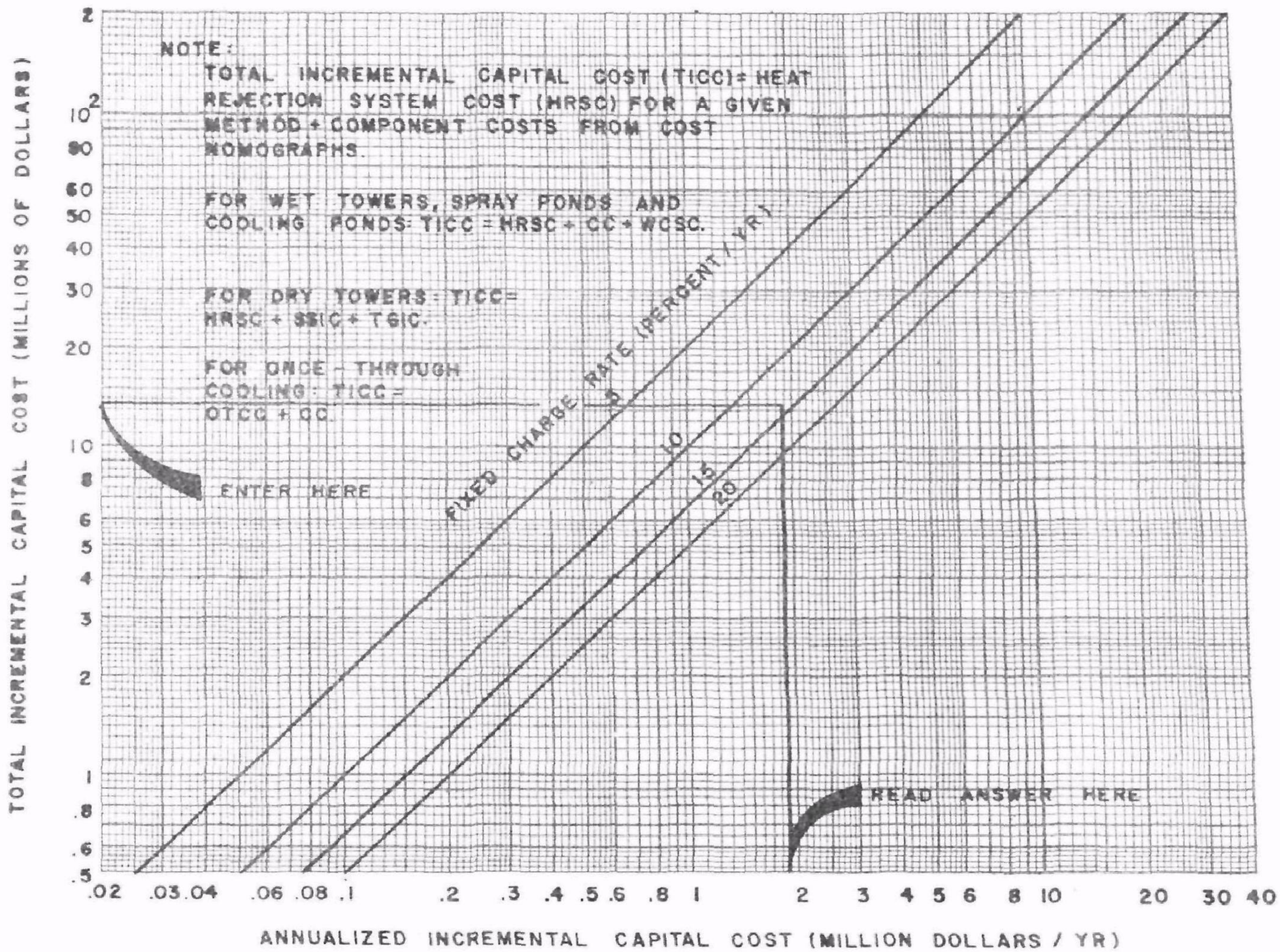
COSTS (CONTINUED)

CAPITAL RECOVERY FACTOR



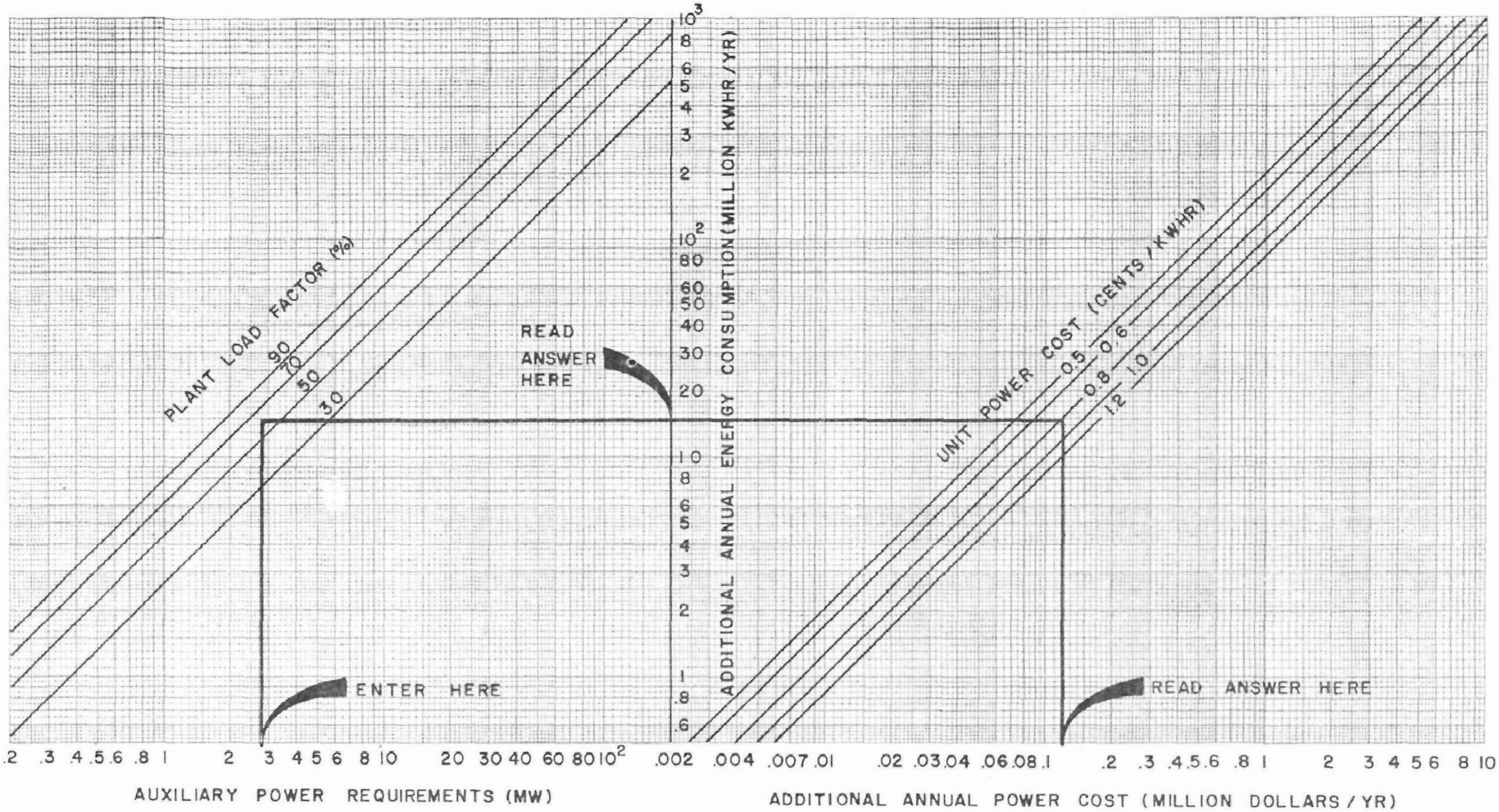
COSTS (CONTINUED)

ANNUALIZED CAPITAL COST



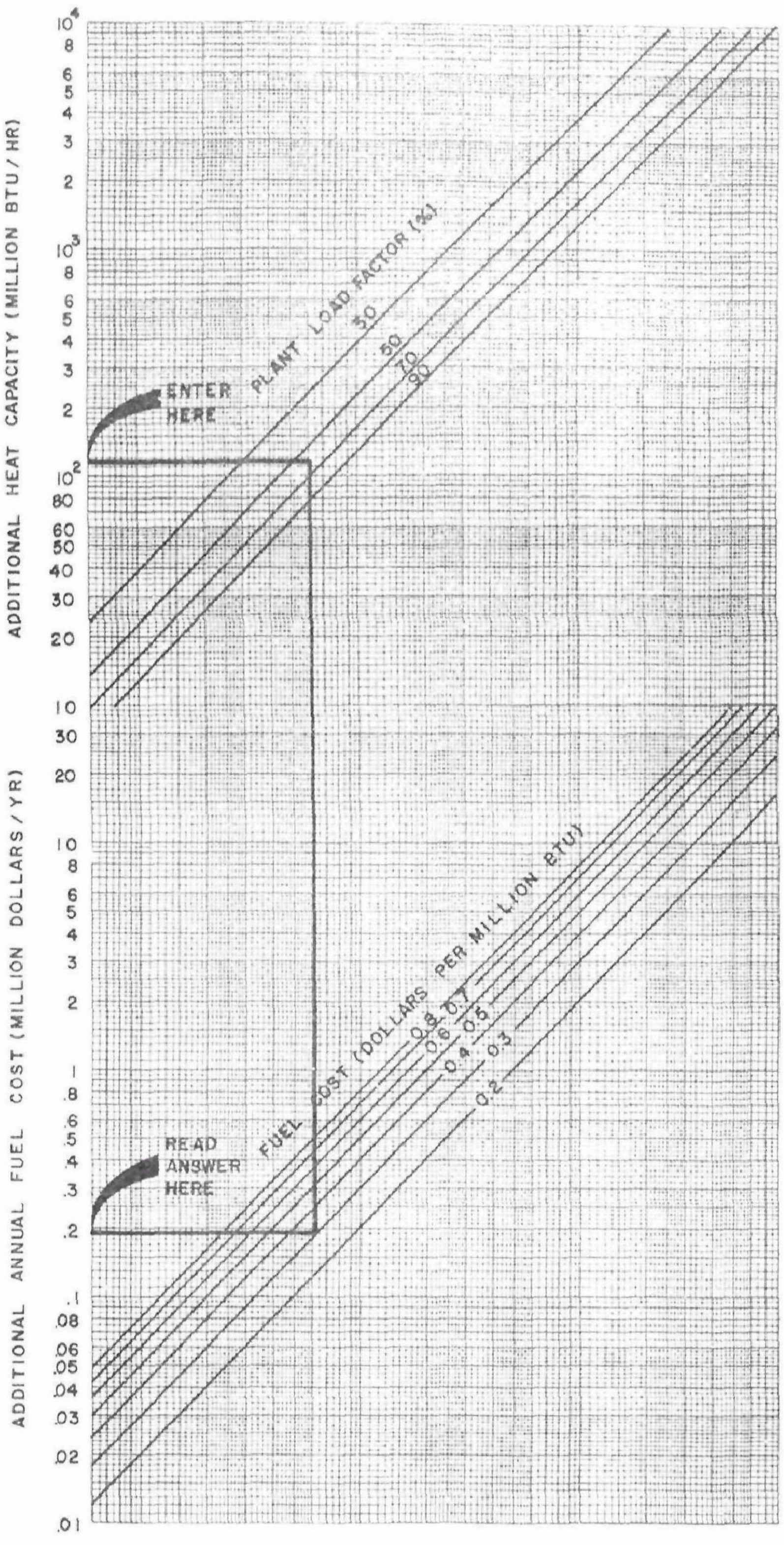
COSTS (CONTINUED)

ADDITIONAL ANNUAL POWER COST



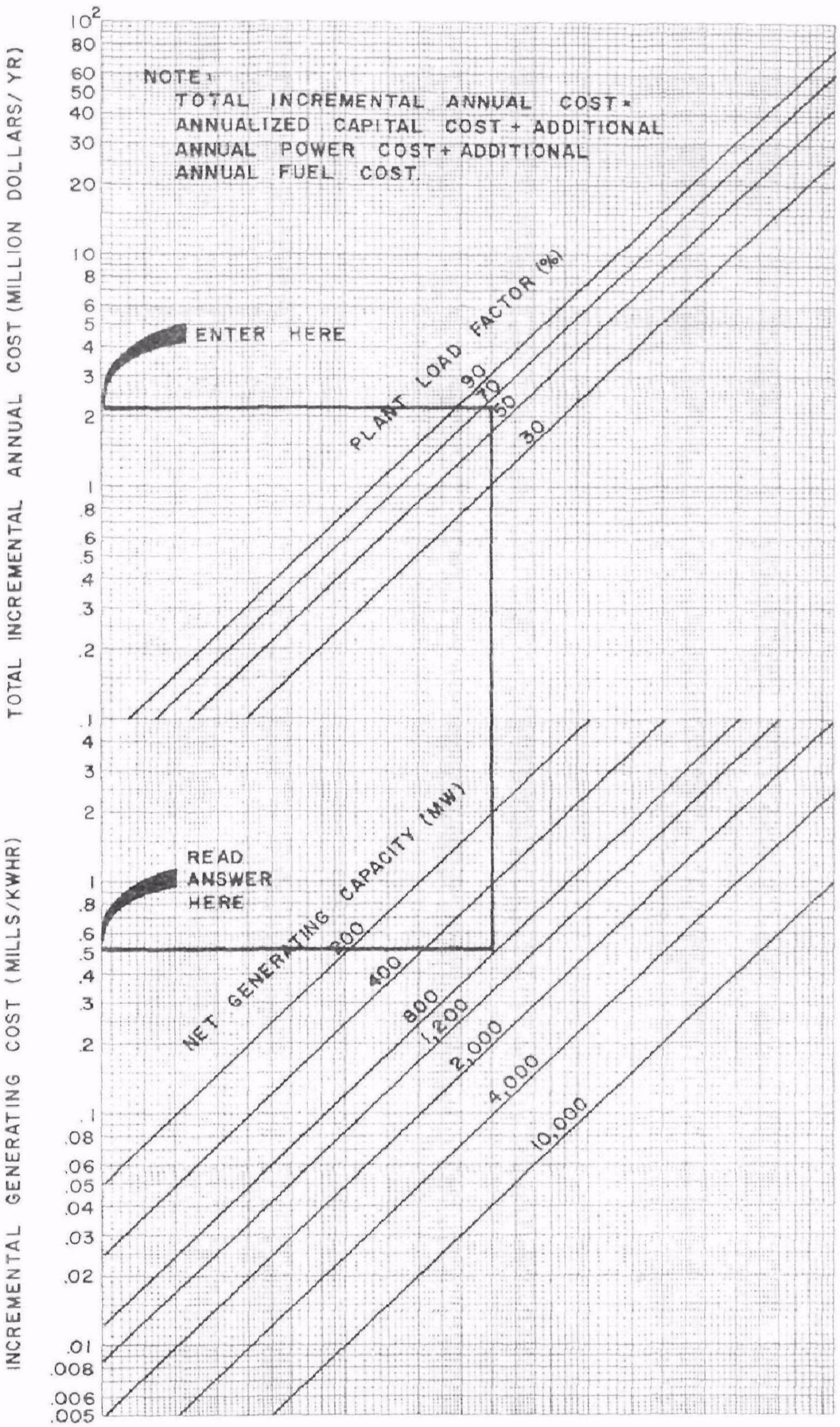
COSTS (CONTINUED)

ADDITIONAL ANNUAL FUEL COST



COSTS (CONTINUED)

INCREMENTAL GENERATING COST



COSTS (CONTINUED)

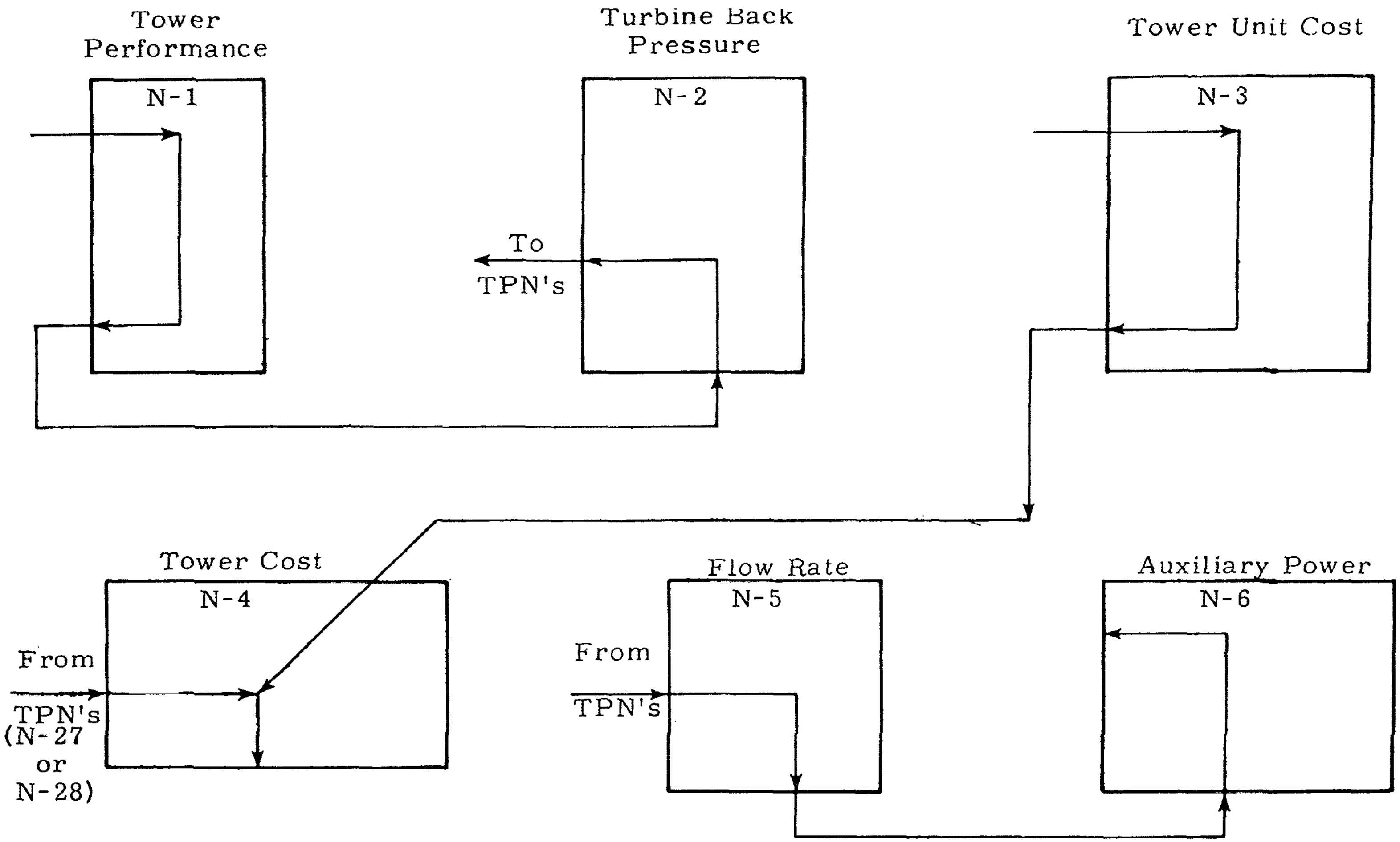


Figure 11. Natural Draft Wet Towers

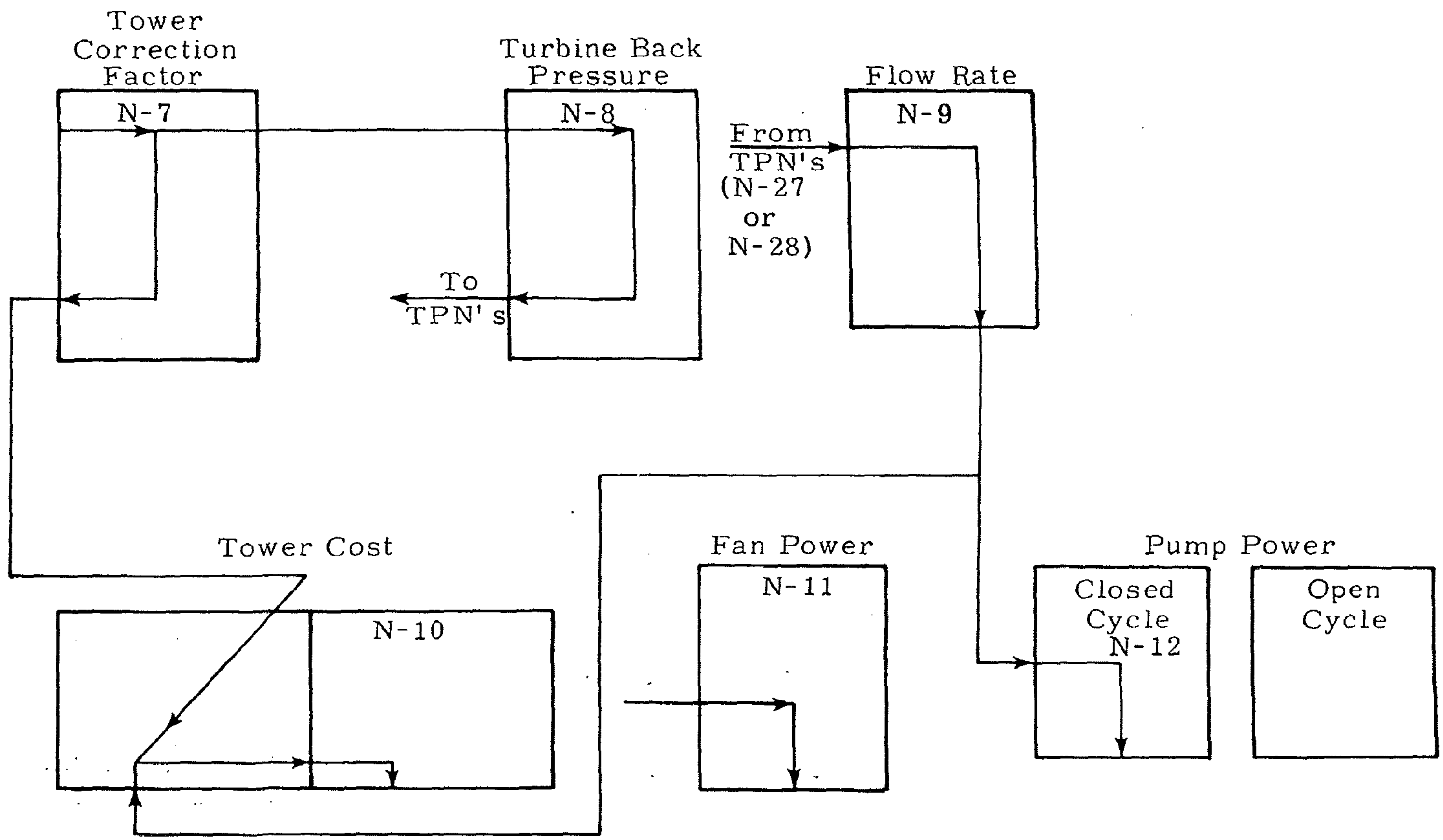


Figure 12. Mechanical Draft Wet Towers

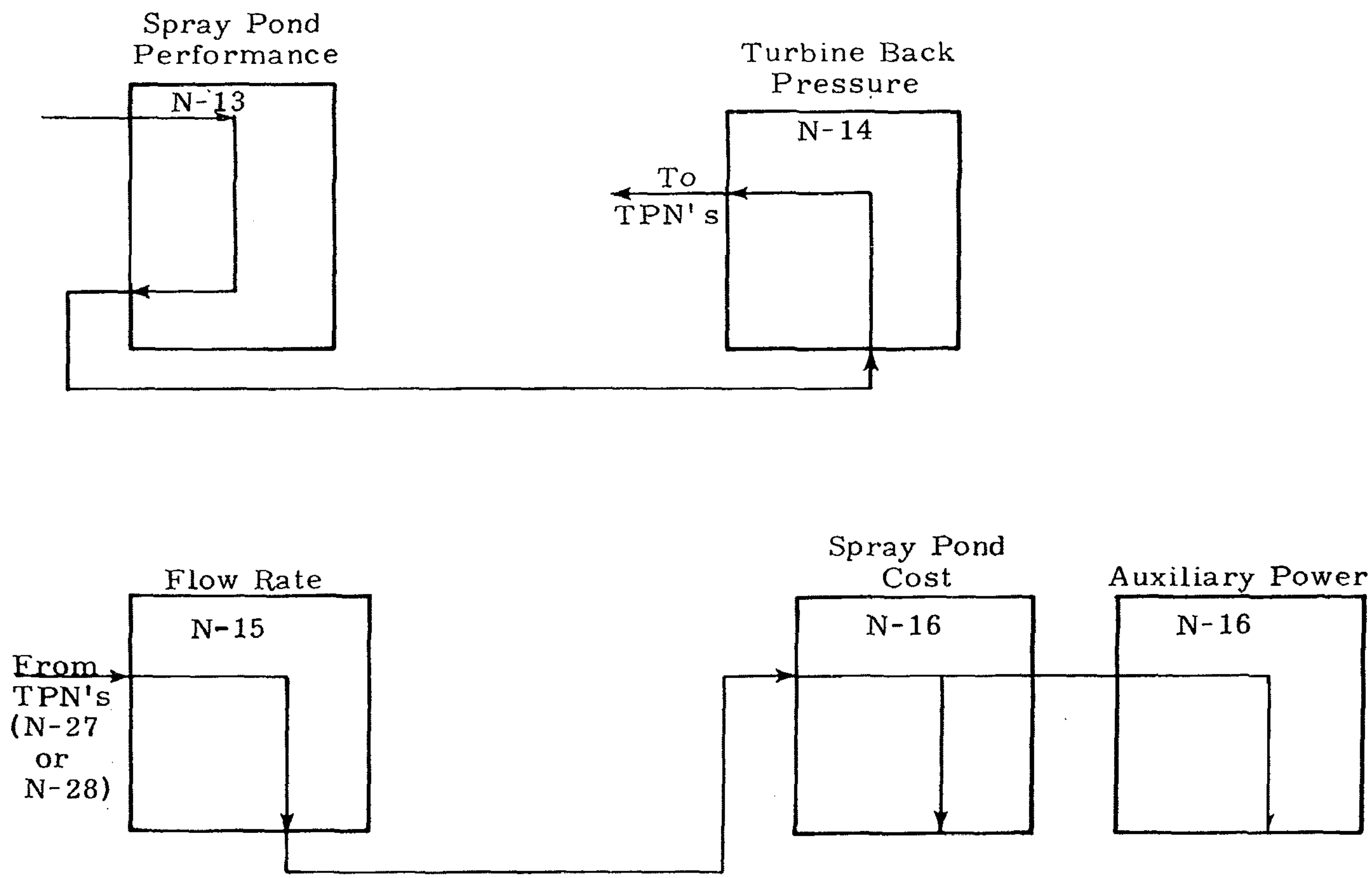


Figure 13. Spray Ponds

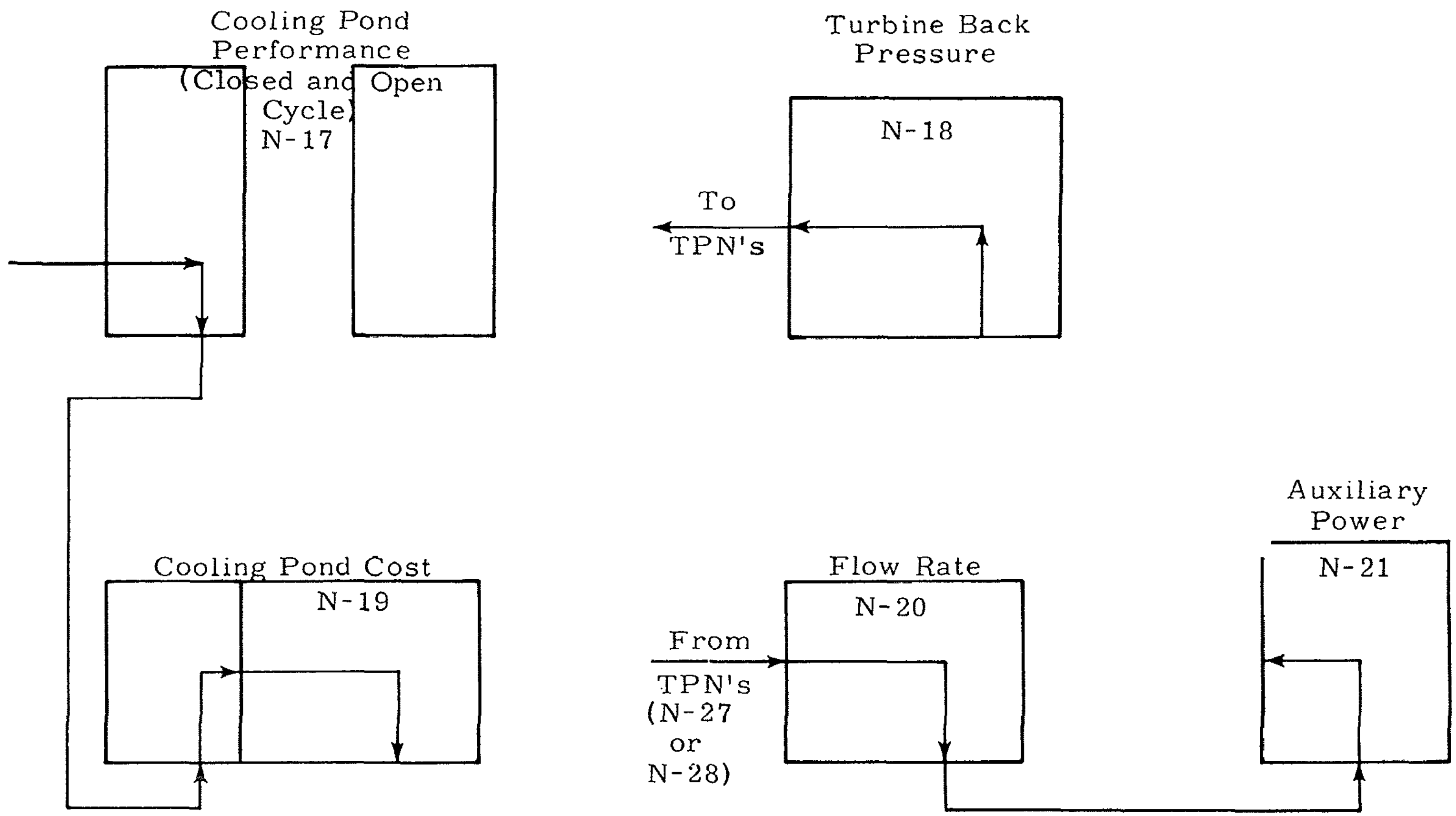


Figure 14. Cooling Ponds

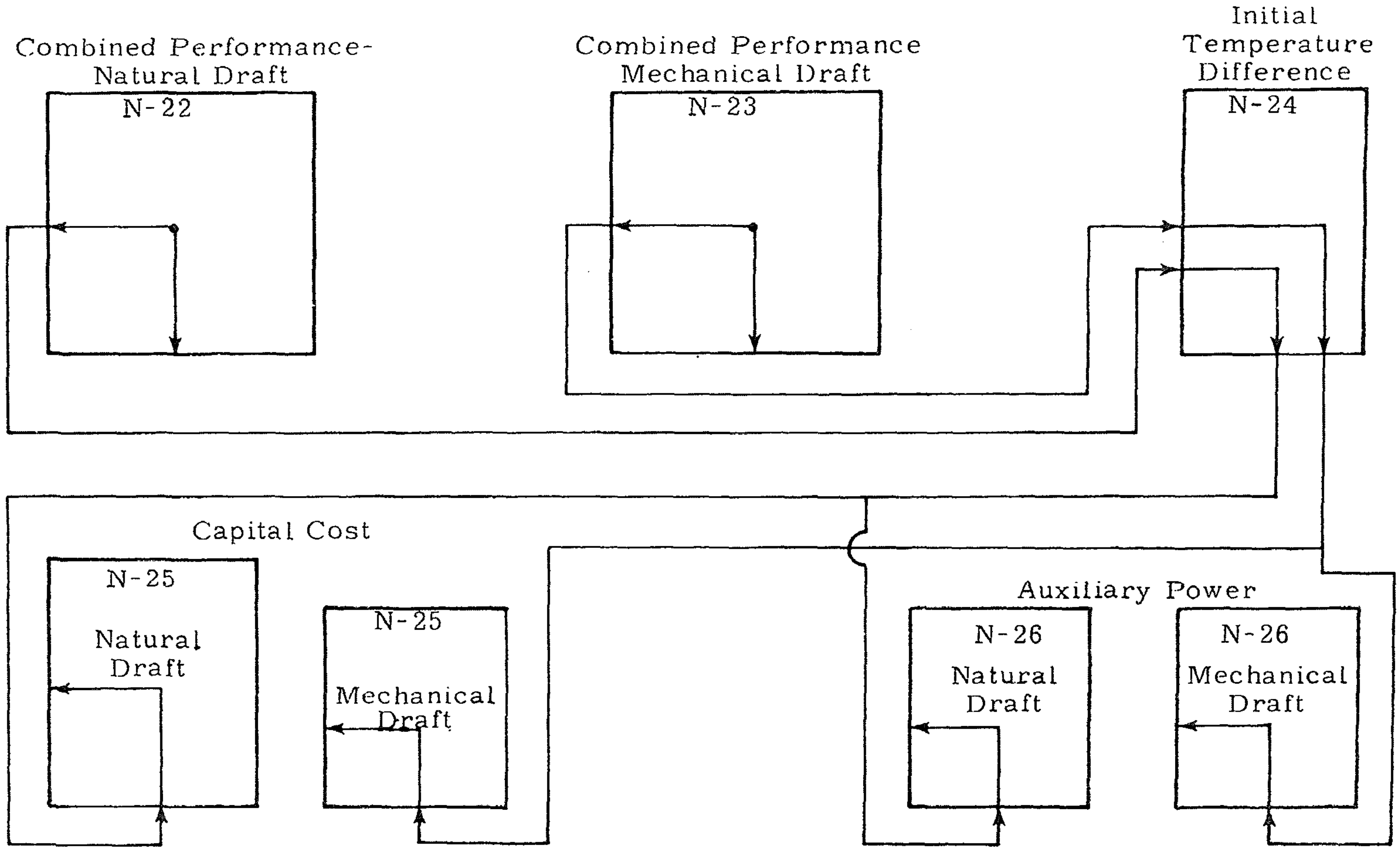


Figure 15. Dry Towers

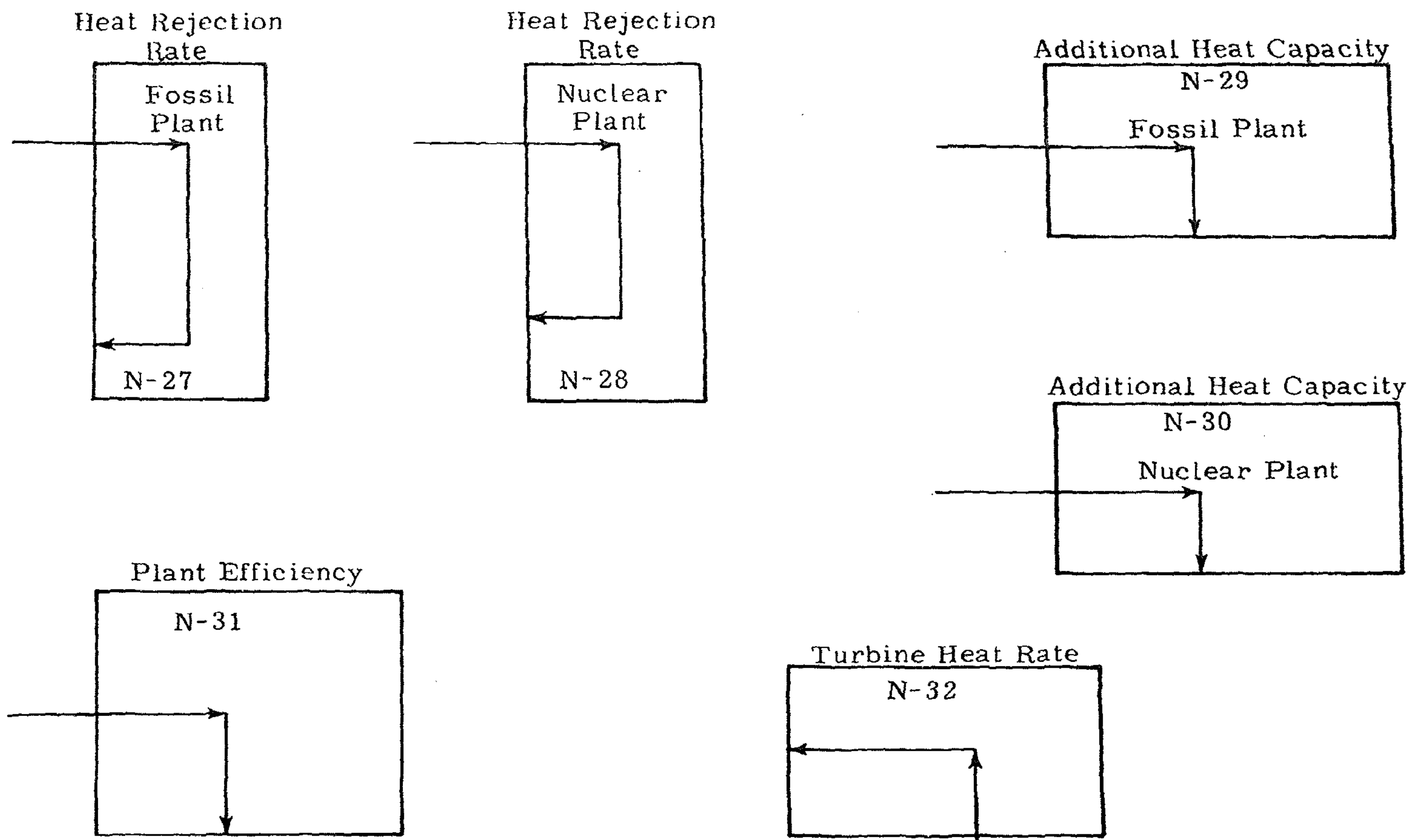


Figure 16. Turbine Performance

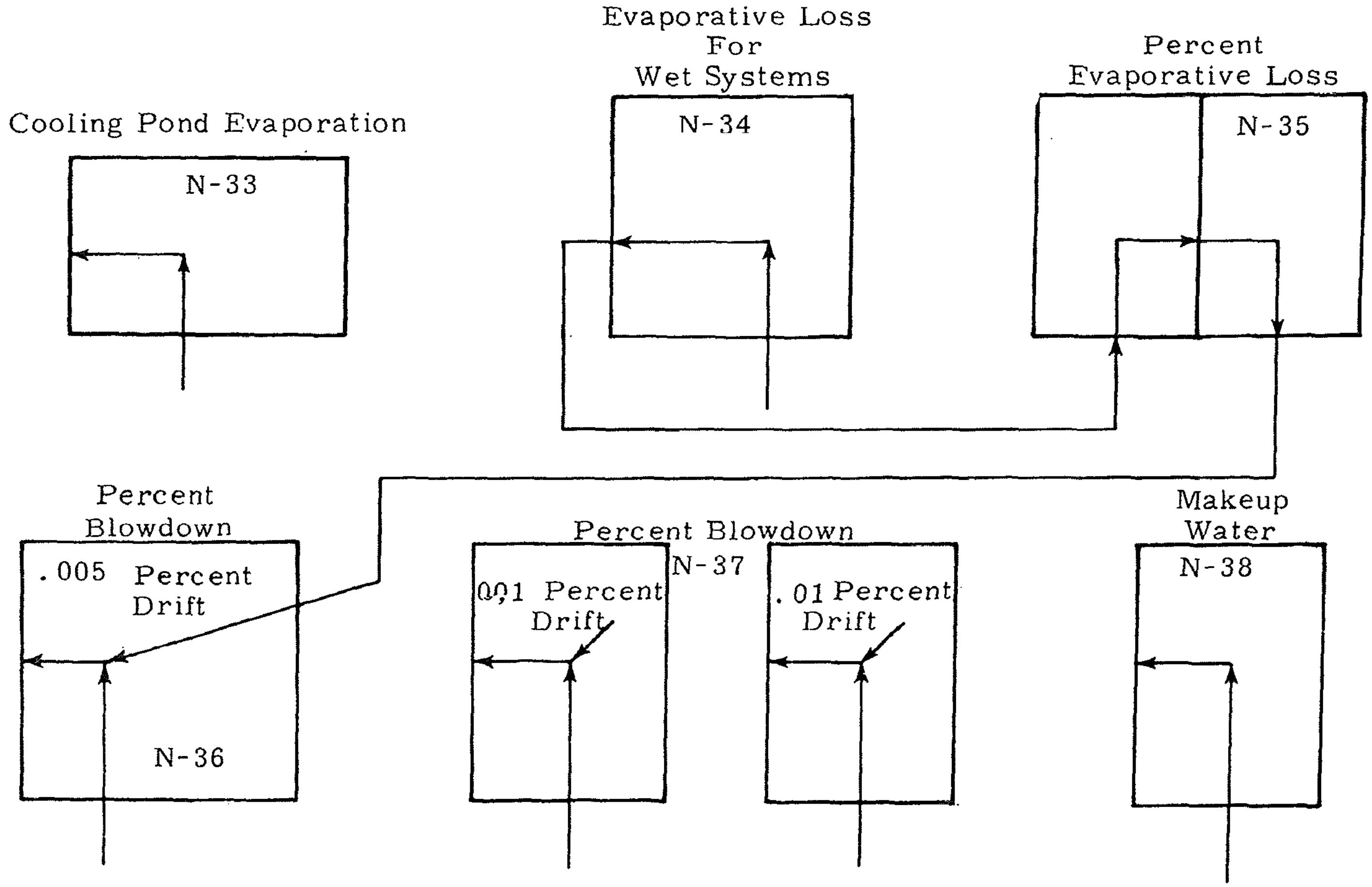


Figure 17, Water Requirements

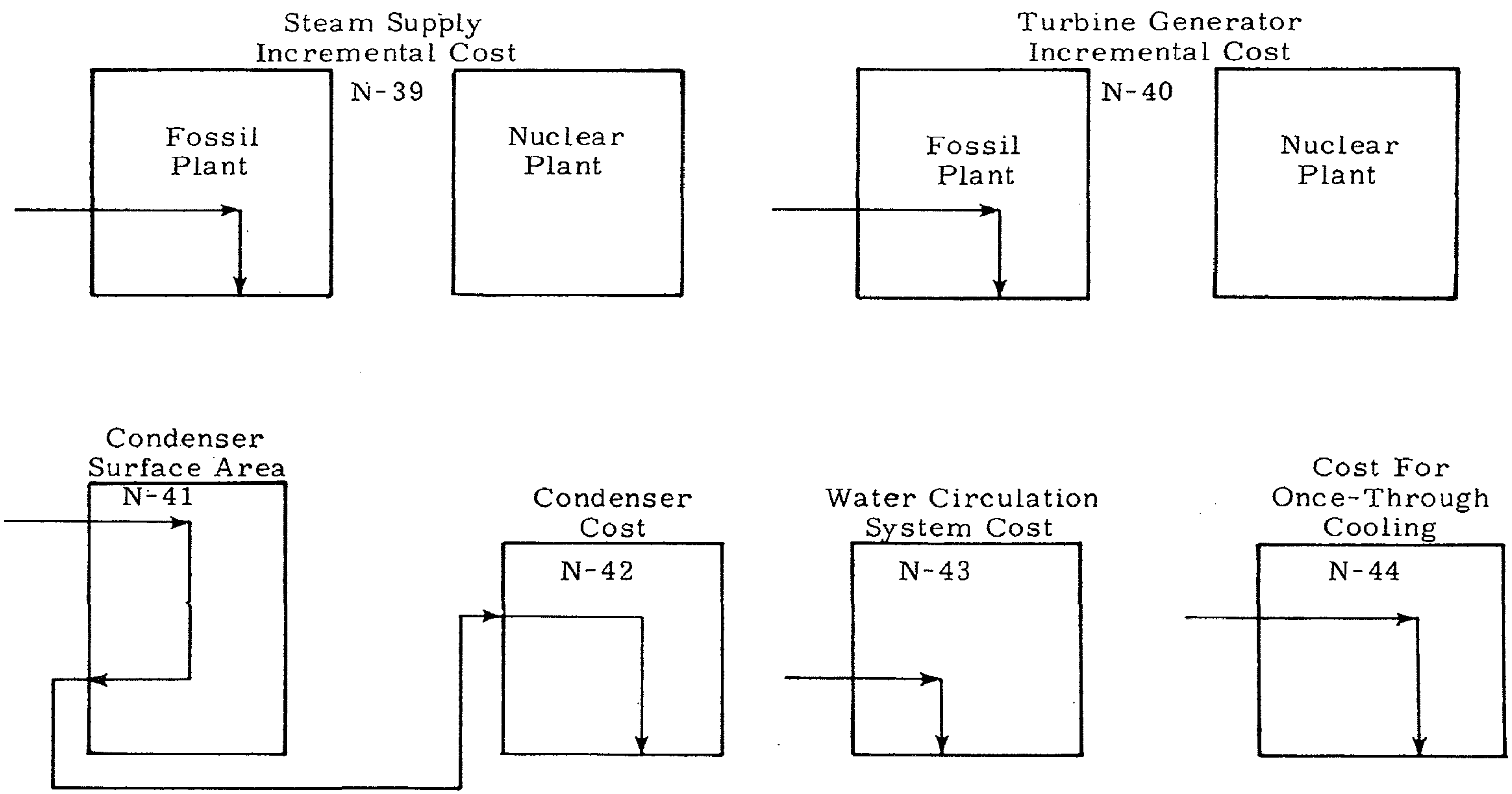


Figure 18. Costs

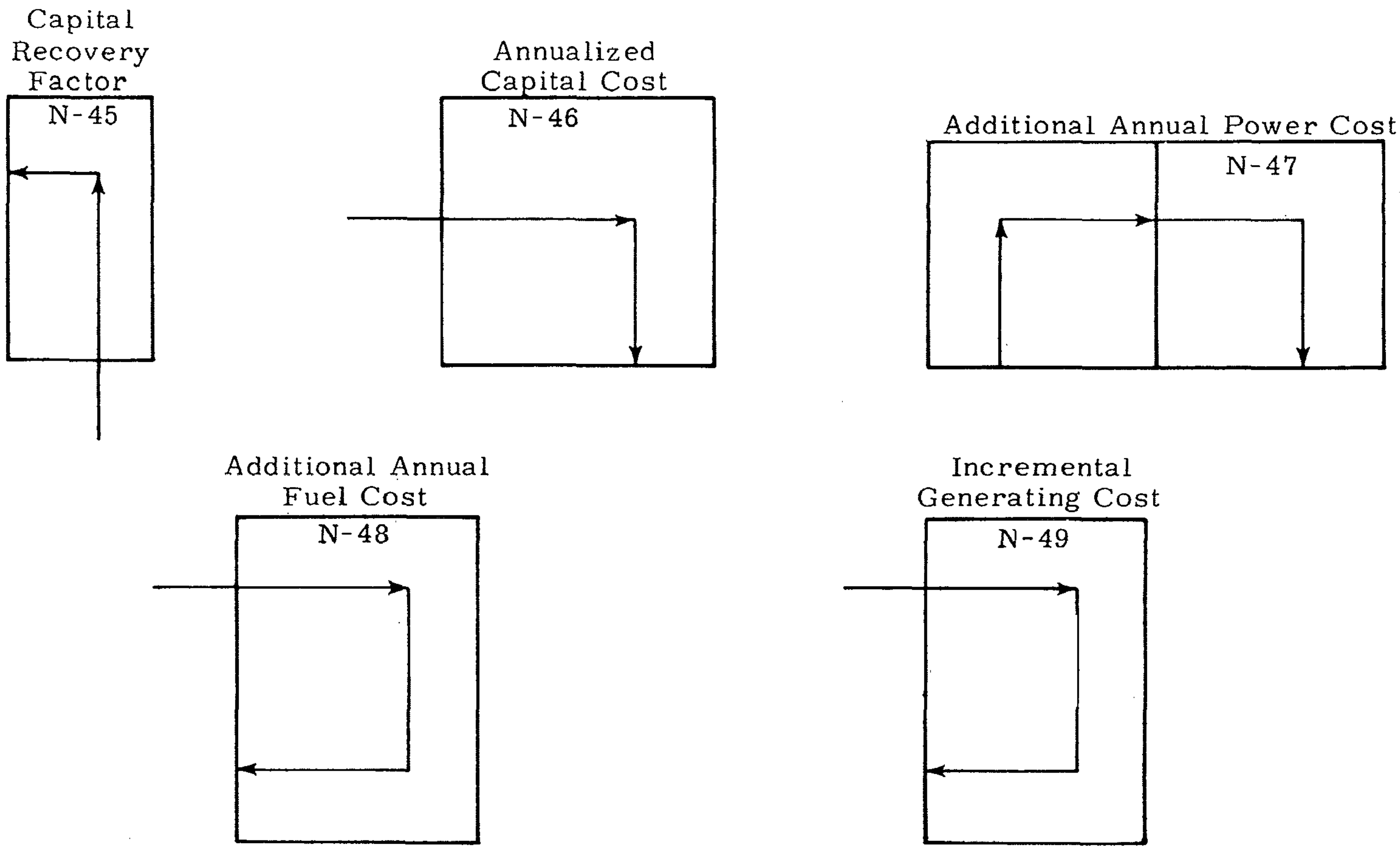


Figure 19. Costs (Continued)

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SELECTED WATER RESOURCES ABSTRACTS

INPUT TRANSACTION FORM

1. Report No.	2.	3. Accession No.
		W

4. Title

Nomographs for Thermal Pollution Control Systems

5. Report Date

6.

8. Performing Organization Report No.

7. Author(s)

10. Project No.

16130 HKK

9. Organization

Hittman Associates, Inc.
9190 Red Branch Road
Columbia, MD 21045

11. Contract/Grant No.

68-01-0171

13. Type of Report and Period Covered

12. Sponsoring Organization U.S. Environmental Protection Agency

15. Supplementary Notes

U.S. Environmental Protection Agency
Report No. EPA-660/2-73-004, September 1973.

16. Abstract

Nomographs are presented and described which permit the estimation of heat rejection system performance, tower or pond capital costs and the perturbations to power plant efficiency and costs which result from the incorporation and operation of any one of the following thermal pollution control systems within a power plant as a substitute for once-through cooling: natural draft wet towers, mechanical draft wet towers, spray ponds, cooling ponds, and natural and mechanical draft dry towers. The base case plant for cost comparisons is chosen as having a nominal turbine back pressure of 2 in. Hg absolute. The total heat rejection system with its associated costs is defined to extend outward from the turbine exhaust flange, a common boundary for each of the systems mentioned above.

Performance and capital costs for the thermal pollution control systems were compared with data from exisitng facilities and theoretical estimates from various sources. The nomographs yield performance, water requirements and costs for heat rejection systems operating under design meteorological conditions and full load plant operation. Examples are presented to show how average annual water requirements from evaporation and average annual operating costs can be estimated.

17a. Descriptors

Thermal pollution,* thermal power plants, economics,* water cooling,* cooling towers,* cooling water, costs, annual costs, capital costs, cost analysis, electric power costs, water consumption, water loss, graphical methods,* curves, mathematical studies.

17b. Identifiers

Cooling systems, heat rejection systems, closed cycle cooling,* thermal pollution control costs,* cooling tower performance.*

17c. COWRR Field & Group 05D

18. Availability	19. Security Class. (Report)	21. No. of Pages	Send To:
	20. Security Class. (Page)	22. Price	WATER RESOURCES SCIENTIFIC INFORMATION CENTER U.S. DEPARTMENT OF THE INTERIOR WASHINGTON, D. C. 20240

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