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CALCULATION OF EVAPORATIVE LOSS COEFFICIENTS FOR THERMAL POWER PLANTS

MASTER

Hanford Engineering Development Laboratory

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ABSTRACT

Evaporative loss coefficients are calculated for cooling the condensers of steam electric power plants. Models which simulate the cooling of such plants using cooling ponds, cooling towers, and once-through cooling are developed and applied at 230 locations within the contiguous United States. Evaporative loss coefficients are calculated for twenty-four cooling configurations using mean monthly weather parameters. These coefficients are expressed in terms of ft^3 per sec per 1000 MWt. Results indicate that the evaporative loss coefficient is dependent upon the location, type of cooling, and the specific design of the cooling system.

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SUMMARY

A model has been developed for the purpose of calculating the evaporative loss associated with cooling the condensers of steam electric power plants. The model simulates the evaporation loss for condenser cooling when using wet cooling towers, cooling ponds, and once-through cooling.

In all, a total of 24 different cooling configurations: 8 cooling towers, 12 cooling ponds, and 4 once-through systems, were investigated in an attempt to realistically bracket the evaporative loss associated with operating each type of cooling facility. Cooling tower calculations were performed over the following range of tower design characteristics; cooling range 20-40^{o}F, effective heat transfer coefficient 0.5 - 1.5, and a water to air flow ratio 0.75 - 1.50. Cooling pond calculations were performed for inverse pond loadings of 1 and 3 acres/megawatt thermal (MWt) for both slug flow and completely mixed flow regimes. For the slug flow cooling pond configuration, cooling ranges of 10^{o}F and 30^{o}F were assigned. Similarly four cooling configurations were analyzed for once-through cooling. Cooling ranges of 10^{o}F and 30^{o}F with instantaneous mixing and no mixing were examined.

Calculations for 230 locations within the contiguous United States are presented in this report. Mean monthly weather data for each location was used to calculate the evaporative loss coefficients. The evaporative loss coefficients have units of ft^3 per sec per 1000 MWt. Assuming an overall thermal power plant efficiency of 33% and that all waste is dissipated through the process of evaporation, the evaporative loss coefficient in terms of ft^3 per sec per 1000 MWt is equivalent to .054 gal per kW-hr.

The calculated evaporative loss coefficient typically varies by 20 to 30 percent over the range in cooling tower design characteristics. The seasonal variation in the weather parameters can cause the evaporative loss coefficient to vary by 10 to 15 percent. For natural ponds, the evaporative loss coefficient was found to vary by 20 to 30 percent over the range of cooling pond design characteristics. In this case the seasonal variation

resulting from changes in the weather parameter can cause the evaporative loss coefficient to vary by a factor of 2. For man-made cooling ponds the evaporative loss coefficient, i.e., water requirement, is dependent upon the makeup to the cooling pond in the form of precipitation. In some locations in the United States which receive considerable precipitation it was found that the cooling ponds are self-sufficient, requiring no makeup water.

A listing of the program used to make the calculations and a discussion of the equilibrium temperature concept used in simulating the temperature distribution in cooling ponds and rivers are provided in two Appendixes.

1.0 INTRODUCTION

During the 1960s both the size and number of installed steam electric power plants increased considerably. Generally speaking both before and during this period, condenser cooling was accomplished by utilizing once-through cooling. Examination of this trend caused considerable concern over the ability of our various water bodies to assimilate the required and projected water withdrawal rates, the associated evaporation rates and projected heat loads. Obviously with fewer and smaller units it was relatively easy to find suitable sites for once-through cooling, however, the problem became much more acute as the size and number of thermal power plants began to increase. As a means of protecting our waterways, thermal pollution control standards emerged during the late 1960s and early 1970s. Since then, the combination of economy of scale, growing demand, and environmental protection has had a significant effect upon condenser cooling design. For example, in 1967 once-through cooling systems accounted for over 65% of the added generation capacity, whereas today 80% of the added generation capacity is cooled using man-made cooling ponds and evaporative towers.(1)

Unfortunately, in the long run this change in trend appears to trade one type of problem for another. Whereas in the past primary concern centered on potential degradation of the resource through thermal pollution, and damage to life forms passed through power plants with large once-through cooling systems, the current trend places a tremendous burden on the resource in terms of evaporation. How successfully this trade-off can be accomplished is of course dependent upon the size of the resource in question and its various competing uses. In some water-scarce parts of this country this limit of water consumption has already been reached, and undoubtedly in future years water deficits will appear elsewhere. The use of large-scale evaporative cooling systems will obviously play a significant role in determining what areas are affected. In addition, new cooling concepts such as dry cooling and dry/wet cooling may have to be brought into play in water scarce regions much sooner than currently realized.(2)

The purposes of this study were first, to develop a methodology whereby the effect of using current evaporative-type condenser coolers could be examined, and secondly, to apply this methodology using appropriate power forecasts to examine specifically how these types of condenser coolers reduce the surface water resource. In this report the model (WATER) which was developed to simulate the physical models of wet towers, cooling ponds, and once-through cooling configurations will be reviewed. The result of using this model or algorithm is a matrix of unit consumption or evaporative loss coefficients which for a given geographical location is a function of the type of cooling system employed and the ambient monthly mean meteorology. These coefficients when combined with future power projections and load factors can be used to calculate a thermal power plant consumptive demand. The consumptive demand can then be compared with other competing demands and the impact on the available resource assessed. The results of this analysis were recently issued under separate cover.[2]

2.0 MATHEMATICAL MODELS

The analytical models used in this study to simulate the evaporative loss associated with operating wet evaporative cooling towers, cooling ponds, and once-through cooling systems are discussed in this section. The models were incorporated into an algorithm which was used to calculate evaporative loss coefficients associated with operating such units in various areas of the contiguous United States.

2.1 Evaporative Cooling of Condenser Water

With the exception of dry cooling towers, all cooling of steam electric condenser water relies to some extent upon the process of evaporation. The models discussed in this section were derived primarily for analytically assessing this evaporative loss.

2.1.1 Cooling Towers

An upper bound on the evaporative loss from the operation of a wet evaporative cooling tower can be established by assuming that all heat is exchanged through the process of evaporation. Since the latent heat of vaporization requires that approximately one pound of cooling water be consumed for every 1000 Btu's of cooling, a 1% loss in total coolant flow results for every 10^{o}F of cooling. In general, however, such assumptions result in overestimating evaporative losses by as much as 35%. On the other hand, quantitative treatment of cooling tower performance involving the use of both heat and mass transfer is extremely complex. To simplify the analysis, Merkel developed the total heat theory[3] which today is used almost universally for analyzing wet cooling tower performance. Simply stated, Merkel's theory assumes that energy is conserved between the air and water mass; i.e., energy lost or added to the water equals the change in total heat gained or lost by the air. Mathematically this can be stated as follows:

$$\frac{hg}{Cp} = \left(i_\theta - i\right) \quad dA = LCp_L \; d\theta = Gdi \qquad (1)$$

where: hg = convection coefficient of heat transfer for air (Btu/hr - ft^2-^oF)

Cp = specific heat of air vapor mixture (Btu/lb-^oF)

$\frac{hg}{cp}$ = coefficient of mass transfer

θ = water temperature (^oF)

i_θ = total heat of saturated air and water vapor at θ (Btu/lb)

i = total heat of air at air temperature (Btu/lb)

dA = heat transfer surface area (ft^2/ft^2 of cross section)

L = water flow rate per ft^2 of cross section (lb/hr-ft^2)

Cp_L = specific heat of water

$d\theta$ = change in water temperature as it flows over dA (^oF)

G = air flow rate per ft^2 of cross section (lb/hr-ft^2)

di = change in total heat of air passing over dA (Btu/lb)

Rearranging Equation (1), the change in water temperature can be written:

$$d\theta = \left(\frac{i_\theta - i}{LC_{PL}}\right) \frac{hg}{CP} \; dA \qquad (2)$$

Similarly, the change in total heat (enthalpy) of the air can be expressed:

$$di = \left(\frac{i_\theta - i}{G}\right) \frac{hg}{CP} \; dA \qquad (3)$$

To solve Equations (2) and (3), a computer model has been developed by Winiarski et al.[4] Using finite element techniques, routines for both counterflow and cross flow cooling tower configurations have been developed.

For the counterflow configuration water enters from the top and the air from the bottom. Equations (2) and (3) are used to calculate the changes that occur as the air and water mix while flowing over a differential area (dA). To update values for θ and i, the following relationships are used:

$$\theta_{A+dA} = \theta_A + d\theta \tag{4}$$

$$i_{A+dA} = i_A + di \tag{5}$$

The change in air temperature is assumed to equal the sensible heat transferred between the water and the air described by the following relationship:

$$dt = hg/GC_p\ (\theta - t)dA \tag{6}$$

The change in air temperature is computed similarly to Equations (4) and (5):

$$t_{A+dA} = t_A + dt \tag{7}$$

Winiarski's procedure for calculating the counterflow configuration employs a trial and error iterative process. First an estimate of the outlet water temperature is made. Then beginning at the outlet, calculations are performed back through the tower packing to determine the inlet conditions. The temperature of the water is then compared with the desirable duty for an assigned cooling range and if it falls within the convergence criterion the solution is terminated, if not the process is continued. Basically this same computational procedure was used in this study.

For the cross flow configuration the process is slightly different. The tower packing is assumed to consist of a lattice of horizontal rows and vertical columns. Water enters from the top and leaves through the bottom. Air enters from one side and leaves through the other. The flow paths of the air and water particles are idealized; water is assumed to remain in a given column, and air in a given row. Mixing of the water or air between adjacent elements is not permitted.

By using the appropriate subscripts denoting rows and columns, Equations (2), (3) and (7) can be rewritten to describe the differential changes in θ, i, and t within cross flow packing as follows:[4]

$$d\theta = \frac{\left(i_{\theta_{I,J}} - i_{I,J}\right)}{LjCpL} \left(\frac{hg}{Cp}\right) dA_{I,J} \tag{8}$$

$$di = \frac{\left(i_{\theta_{I,J}} - i_{I,J}\right)}{GI} \left(\frac{hg}{Cp}\right) dA_{I,J} \tag{9}$$

$$dt = \frac{\theta_{I,J} - t_{I,J}}{GI} \left(\frac{hg}{Cp}\right) dA_{I,J} \tag{10}$$

where I is the subscript for rows, and J is the subscript for columns. To solve Equations (8-10) a finite element integration scheme is applied similar to the one used to solve the counterflow configuration. However, differential changes in air temperature and air total heat apply along a row, while changes to the water temperature apply along a column. Consequently, to update the parameters (θ, i, t), the following adjustments are made for subsequent iterations:

$$\begin{aligned} \theta_{I+1,j} &= \theta_{I,J} - d\theta \\ i_{I,J+1} &= i_{I,J} + di \\ t_{I,J+1} &= t_{I,J} + dt \end{aligned} \tag{11}$$

Referring to Figure 1, the calculations are based on the following. Water temperatures for all elements of the top row are set equal to the inlet water temperature. Air temperature and total heat for all elements of the first column are set equal to that of the incoming air. Starting with

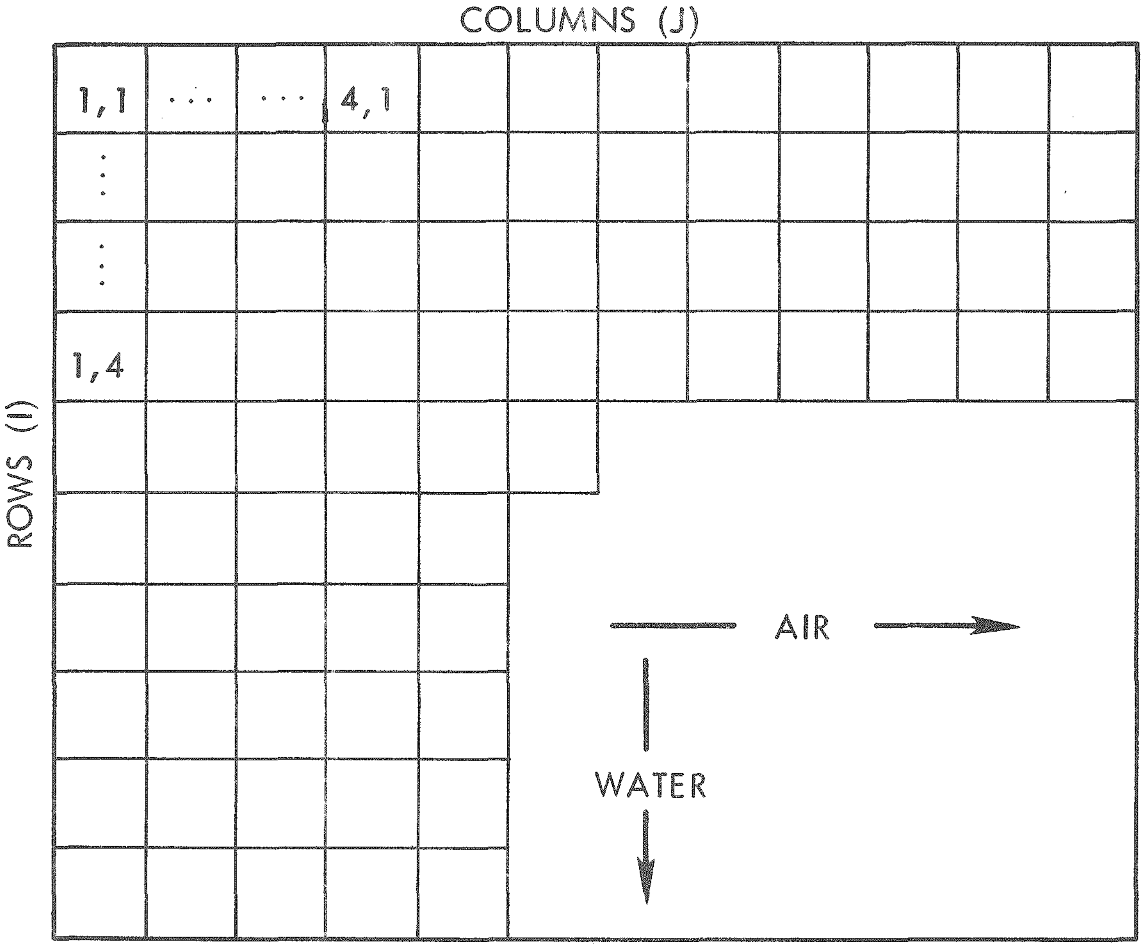

FIGURE 1. Cross Flow Tower Configuration. HEDL 7803-354.1

the element (1,1), the numerical integration scheme is solved for water temperature in each element of the first column until the outlet water temperature for the column has been evaluated. Similarly, water and air temperatures for the remainder of the matrix are determined. Outlet air and water conditions are then determined by averaging the outlet states for all the columns and rows. The outlet water temperature is then compared to the inlet water temperature to see if the proper predescribed tower duty has been satisfied:

$$LCp_L(\theta_{IN}) = LCp_L[\theta_{OUT} + \text{Cooling Range } (\Delta\theta)].$$

When this condition is satisfied, the iteration process is terminated, otherwise a temperature adjustment to the inlet water is made and the process is continued.

To evaluate the water lost through evaporation, a straightforward calculation is undertaken using the mixing ratio or absolute humidity of the air at the inlet and outlet of the tower, and the mass flow rate of the air.

The psychrometry for the air stream within the tower packing is described through the following relationship:[5]

$$j = 0.24t + a(1061.8 + .44t) \tag{12}$$

where:

a = absolute humidity (#water/#air)
j = air enthalpy (Btu's/#)
t = temperature (^oF)

the constants:

0.24 = the specific heat of the air (Btu's/#-^oF)
1061.8 = latent heat of vaporization of water (Btu's/#water)
0.44 = specific heat of water vapor between (0-120^oF)

Since j and t are known, the absolute humidity can be calculated directly. To assure that the absolute humidity calculated does not exceed saturated conditions, the saturated vapor pressure is calculated using the following expression:[5]

$$PSAT = 0.252\ e^{18.1585(t-60./t+460)}(Psia) \quad (13)$$

Similarly the atmospheric vapor pressure is calculated as follows:[6]

$$P_A = (1.0 - 0.0065*E/3.28*288)^{5.2566}(Psia) \quad (14)$$

where E = elevation above mean sea level (ft)

Using Equations (13) and (14) the maximum allowable absolute humidity of the air related through the partial pressure of saturated water vapor and total partial pressure is calculated as follows:[5]

$$X = \frac{PSAT}{PA-PSAT}\ .622$$

where .622 = molecular weight of water/molecular weight of air

If the calculated absolute humidity is greater than the maximum allowable absolute humidity, a condition of super saturation exists and slight adjustments are made to both the air and water temperatures conserving energy at all times.

As proposed by Leung and Moore[6] and implied previously, the tower makeup requirements from the evaporative loss can be determined by examining the state of the air entering and leaving the tower. The following relationship is applied:

$$L_M = G(W_o - W_i) \quad (15)$$

where: W_o = mixing ratio (absolute humidity) of air leaving fill (lb water/lb air)

W_i = mixing ratio (absolute humidity) of air entering tower.

As indicated, the appropriate mixing ratios are determined directly from the enthalpy or total heat flux and temperature calculation of the air mass as it moves through the packing. Consequently, since both the inlet and outlet air states are known for a specified gas flow rate, Equation (15) can be calculated directly.

2.1.2 Cooling Pond Models

A large number of parameters can and generally do influence the performance of a specific cooling pond. An attempt to include all of these parameters is unrealistic in a study such as this. Consequently, a generalized approach was attempted to bracket evaporative loss resulting from the operation of a cooling pond.

Two models were used. Both employ the conservation of thermal energy and an assumed circulation pattern or flow regime. Indeed the two, commonly referred to as the completely mixed and slug flow cooling pond models, derive their names from the assumed circulation pattern. As indicated in the previous paragraph, the rationale behind using these two models stems from the belief that cooling pond performance should be bracketed. In terms of cooling pond performance, it can be shown that for the cold or return leg, 1) application of the mixed pond model results in a maximum temperature; and 2) application of the slug flow model results in a minimum temperature. The heat transfer characteristics associated with the slug flow type of cooling pond configuration result in a more efficient way to dissipate heat. In this section both types of cooling ponds will be discussed. The techniques discussed are basically the same as those presented by Sonnichsen.[7]

Completely Mixed Pond

By definition, a completely mixed pond has a nearly uniform temperature distribution throughout. The precise mechanisms by which a uniform temperature distribution can be created will not be reviewed. The steady-state representation for predicting temperatures within the pond are summarized by the basic conservation of thermal energy relationship:

Heat in - Heat out = Heat dissipated

expressed mathematically:

$$\rho CpQpT_i - \rho CpQpT_o = KA(T_o - E) \quad (16)$$

where: A = surface area (ft^2)

Cp = specific heat of water (Btu/#-^oF)

E = equilibrium temperature (^oF)

Qp = pumping rate (ft^3/day)

K = exchange coefficient (Btu/ft^2-^oF-day)

T_i = inlet temperature (^oF)

T_o = outlet temperature (^oF)

ρ = weight of water (#/ft^3)

As indicated in Equation (16), the concept of equilibrium temperature was used in this study. By definition, the equilibrium temperature is that temperature (water) at which the net exchange of energy between the air and water is zero for a given set of meteorological conditions. Although the concept has been recognized for years, Edinger, et al.[8] are primarily responsible for reintroducing the concept into contemporary literature and for quantifying the exchange coefficient. There are several existing techniques for evaluating the exchange coefficient and equilibrium temperature for a given set of meteorological conditions. In computing this temperature and coefficient, the standard meteorological parameters (which include wet and dry bulb temperatures, windspeed, cloud cover, and incoming solar radiation) are accounted for by using the approach suggested by Brady.[9]

Appendix A includes a description of the technique used to calculate the equilibrium temperature and the exchange coefficient.

The addition of heat to the pond by the operation of the power plant can be written as follows:

$$Hp = (\Delta T)\rho CpQp \tag{17}$$

where Hp = Heat added (Btu/day)

$\Delta T = T_i - T_o$ temperature rise across the cooling condenser (^oF)

Substituting Equation (12) into Equation (16) and simplifying results in the following expression for the outlet or, in this case, the mixed pond temperature, yields

$$T_o = \frac{Hp}{KA} + E \tag{18}$$

Equation (18) is a very convenient way of solving for the average pond temperature based upon a given plant load and pond surface area once the equilibrium temperature and exchange coefficients have been determined. In terms of inverse pond loading (acre/MWt), Equation (18) can be written:

for the case of 1 acre/MWt loading:

$$T_o = \frac{1889.0}{K} + E\ (^oF) \tag{19}$$

for the case of 3 acre/MWt loading:

$$T_o = \frac{629.7}{K} + E\ (^oF) \tag{20}$$

Equations (19) and (20) were used in this study to calculate the mixed pond temperature. Note that the mixed pond temperature is independent of cooling range. The selection of 1 and 3 acre/MWt ponds is to some extent arbitrary;

however, it is typical of past pond designs and is used consistently throughout this study.

Slug Flow

Possibly a more realistic representation of cooling pond behavior is depicted by the slug flow or flow through cooling pond model. As such, the temperature of a water parcel as it passes through the pond will decrease in an exponential manner. Since the temperature differential is greatest at the inlet, higher rates of heat rejection occur near the inlet than occur in the vicinity of the outlet. The precise mechanisms by which a slug flow region can be created will not be reviewed.

Using the approach suggested by John[10] for an element of length dx and surface area dA (assuming no longitudinal mixing and complete lateral mixing), the flux of energy dissipated from the surface is equal to the decrease in the energy of the mass flowing through the element. Performing an energy balance and employing the concept of equilibrium temperature, the slug flow model can be summarized mathematically as:

heat in - heat out = heat dissipated

$$\rho QpCpdt = - KA(T-E) \tag{21}$$

where the sign convention is the same as that proposed by John.[10]

Integrating over the surface area of the ponds yields

$$T_{(A)} - E = (T_i - E)\ e^{-KA/\rho QpCp} \tag{22}$$

As performed previously for the case of completely mixed ponds, substituting Equation (2) into Equation (22) and simplifying will result in the following expression:

$$T_{(A)} = \frac{Te^{-\gamma}}{1 - e^{-\gamma}} + E \tag{23}$$

where

$$\gamma = \frac{KA\Delta T}{H_p}$$

In terms of inverse pond loading (acre/MWt), Equation (23) can be written in the following ways:

for the case of 1 acre/MWt pond loadings:

$$\gamma = -0.000532K\Delta T \quad \text{for temperatures } (^{\circ}F) \tag{24}$$

for the case of 3 acre/MWt pond loading

$$\gamma = -0.00159K\Delta T \quad \text{for temperatures } (^{\circ}F) \tag{25}$$

Similar to Equations (19) and (20), Equations (24) and (25) are used in the analysis to evaluate outlet pond temperature for the various size cooling ponds. Note that the pond temperature is dependent upon the cooling range.

The heat lost from a body of water to the atmosphere through the process of evaporation is dependent upon both the water temperature and the overlying or local ambient meteorology. Heat removed through the process of evaporation varies from a few hundred to several thousand Btu/ft^2-day. A number of semi-empirical models have been proposed for computing the evaporative loss. In terms of forced convection, wind-speed-dominated mechanisms, the expression generally takes the form:

$$Q_e = f(u)\left[e_w - e_a\right] \text{ Btu/ft}^2\text{-day} \tag{26}$$

e_a = vapor pressure of the ambient air (mm-Hg)

e_w = saturated vapor pressure at the temperature of the surface water (mm-Hg)

$f(u)$ = wind speed function

In Expression (26) the vapor pressure difference $(e_w - e_a)$ is commonly refered to as the "Dalton Difference". The wind speed function f(u) has been described in various ways, however in this study the expression developed by Brady[10] was employed. Brady's proposed model based on standard meteorological and cooling pond data expresses evaporation in the form:

$$Qe = (70 + .7u^2)(e_w - e_a) \tag{27}$$

To verify the use of Equation (27), the mean annual natural evaporation in terms of (in./yr) was computed for a number of locations in the U.S. and the results compared to the estimated mean annual evaporation from lake surfaces found in Reference (11). Generally the results compared favorably to within 10 percent. In computing natural evaporation, the temperature of the surface water was assumed to be at its equilibrium value. The seasonal variation of the equilibrium temperature (assumed surface water temperature) was based upon monthly averaged meteorology.

Cooling ponds artificially heated operate at temperatures higher than those termed natural. As suggested in the previous paragraph, it was assumed in this study that the natural temperature of the surface water when averaged over the studied duration is approximately the same as the equilibrium temperature averaged over the same time span. The time lag or phase shift normally associated with water temperature was not considered important in this study for these reasons: (1) evaporation is averaged over an entire year, and (2) the relative size of cooling ponds compared to other sized water bodies is small, resulting in a shorter time lag and smaller phase shift.

Using an analogous "Dalton Difference" relationship for artificially heated ponds, the total evaporation (natural plus forced) from the surface of a cooling pond can be computed as:

$$Q_e = f(u)\ (e_H - e_a) \tag{28}$$

where

e_H = saturated vapor pressure associated with the temperature of the heated water surface (mm - Hg)

To determine the total heat transferred through evaporative cooling, the "Dalton Difference" relationship is multiplied by the effective pond surface area. This quantity, in turn, is equated to the mass lost through the latent heat of evaporation, which varies slightly with temperature. In this study, the latent heat was approximated as:

$$H_e = 1076 - 0.56714\ (T_w - 32) \quad \frac{\text{Btu}}{\text{lb}} \tag{29}$$

where

T_w = temperature of water surface (^{o}F)

By combining, the total evaporation from a completely mixed pond can be computed in a straightforward manner using Equations (19), (20), (28), and (29). Due to the exponential nature of the slug flow model, the computation scheme employed to compute total evaporation from the slug flow ponds is slightly more complex. Before Equations (28) and (29) can be used, the weighted average temperature of the pond must be computed. This was accomplished as follows: The incremental surface area of the pond was first divided into one hundred equal-sized cells (0.01 A_T), where A_T equals the total surface area of the ponds. Temperatures (T_A) and, subsequently, saturated vapor pressures were determined as a function of surface area using Equation (21) for the two sized ponds. From this, the weighted average saturated vapor pressure over the entire pond surface area can be calculated. Equations (28) and (29) are then used to compute the total evaporative loss from slug flow cooling ponds. The selection of one hundred cells is once again arbitrary, although it was found that selecting smaller subdivisions did not appreciably change the results.

Tiechenor and Christianson[12] state that the total loss through evaporation from a cooling pond can be reduced by decreasing the size of the pond. In terms of total evaporation from the surface of the cooling pond this conclusion is correct. However, as will be argued in the following paragraphs, total evaporation is not necessarily the rate of evaporation that should be assigned against operating a cooling pond.

As indicated previously, the evaporation from operating a cooling pond consists of both natural and forced evaporation. Natural evaporation is that which occurs due to the natural vapor pressure difference between the ambient water and air. Forced evaporation is the difference between natural evaporation and total evaporation associated with artificially increasing the temperature of the pond through addition of waste heat. Distinguishing between natural and forced evaporation is particularly relevant when specifying a difference between using a natural or man-made pond. (See Reference 7.)

When using a natural or existing body of water for cooling, evaporation is defined in this study as the water required to replace the loss resulting from only forced evaporation. This reasoning is based on the premise that the natural flux from a natural water body would occur regardless of use and consequently should not be assessed against the operation of a power plant. It was assumed that the natural background temperature of the water is equal to the equilibrium temperature. Thus the water loss from forced evaporation assigned to a natural pond is restricted to the evaporative loss associated with operating the pond at temperature above the equilibrium temperature. Consequently, the heat loss resulting from what might be termed "net evaporation" is defined as:

$$Q_e = fu(e_H - e_E) \tag{30}$$

where

e_E = saturated vapor pressure at the equilibrium temperature

The computational scheme mentioned in the previous paragraph was modified appropriately to reflect this reasoning when computing the net evaporation from a natural cooling pond.

For the case of a created body of water it would seem that a mass balance should be performed on the entire system, in order to specify the net water lost. To simplify the analysis in this study, it was assumed that no water is exchanged or transported between the pond and adjacent surface and ground water supplies. In other words, the pond is lined, either naturally or artificially. Under these conditions, performing a mass balance on the system simply at the free surface suggests that the net water loss equals the net evaporative loss. This can be equated to total evaporation minus makeup through all forms of precipitation. As such, makeup in the form of precipitation can be applied as a credit. Water losses for a man-made pond are therefore defined as

$$\text{water loss} = \text{total evaporation} - \text{precipitation}$$

Similarly, the computational scheme was modified appropriately when considering the net evaporation from a man-made cooling pond.

2.1.3 Once-Through Cooling Models

Generally speaking, simplicity in design and operation, and low condensing pressures have in the past been sufficient to entice utilities into using once-through cooling systems. However, as indicated previously, present restrictions both in terms of water quality and quantity coupled with an increase in unit size will undoubtedly hinder its acceptability in future designs.

In terms of consumption, aside from dry cooling, once-through cooling systems are generally looked upon as imposing the least demand on the water resource. This results from the following: 1) As the size of the water body increases the ambient or background temperature generally becomes slightly lower during the critical warm period of the year, i.e., temperature of a natural water body lags behind the equilibrium temperature, and 2) normally the capacity and turbulence of larger water bodies causes significant dilution to take place thereby lowering the peak temperatures within the plume.

Simplified once-through cooling pond models generally assume a non-stratified, one-dimensional temperature distribution. The temperature distribution varies longitudinally, decaying in the downstream direction and approaching the equilibrium temperature. Similar to Equation (21) the steady-state description consideration longitudinal advection to the dominant transport mechanism is written as follows

$$\rho Cpdu\frac{\partial T}{\partial X} = -K\ (T - E) \tag{31}$$

The solution to Equation (29) takes the form

$$T_{(A)} = T\ e^{-r} + E \tag{32}$$

where $r = \dfrac{Kx}{\rho Cpd(u)}$

letting Q_{RIVER} = velocity (u) * depth (d) * width (w) or $\dfrac{Q_{RIVER}}{w} = d*u$ (33)

and substituting Equation (33) into (32) results in the following expression for stream temperature as a function of surface area:

$$T_A = \Delta T\ e^{\frac{-KA}{\rho C_p Q_{RIVER}}} + E \tag{34}$$

Once again, to bracket the evaporative loss from operating once--through cooling systems, two conditions were assumed. The first assumes that the entire river flow is used for purposes of cooling, i.e., the entire flow of the river is used for condenser cooling. Combining Equation (17) with (34) results in the following relationship:

$$T_A = \Delta T\ e^{\frac{-KA\Delta T}{Hp}} + E \tag{35}$$

Consequently the formulation for the once-through cooling pond model is virtually the same as used for the slug flow cooling pond model with the exception that the outlet temperature equals the ambient sink temperature. Assigning an ambient sink temperature equal to the equilibrium temperature and an inverse pond loading of 50 acres/MWt, the following model results:

$$T(A) = \Delta T\ e^{-.0266\frac{KA\Delta T}{Hp}} + E \tag{36}$$

Specifying an inverse loading allows for a tractable solution consistent with the previously discussed slug flow cooling pond model.

The second condition assumes an instantaneous dilution of 80 percent. In effect, it assumes that sufficient jet or discharge momentum and sink cooling water are physically present to create this situation. Assuming the same inverse loading as discussed above, analogous to Equation (36), the following relationship can be written for the case involving dilution:

$$T_A = .2\Delta T \ e^{-.0053\frac{KA\Delta T}{Hp}} + E \qquad (37)$$

The computations used to arrive at the evaporative loss parallel the procedure for calculating the evaporative losses for the slug flow cooling pond configuration. Once again it was assumed throughout this study that the ambient sink temperature equals the equilibrium temperature. In reality, the overall thermal inertia related to the volume of the water mass causes the actual temperature of the water body to lag below the assumed equilibrium temperature. As a result the evaporative loss is underestimated during the spring and overestimated during the fall. Averaged over the year the net result should be near zero. In accounting for evaporative loss only forced evaporation is calculated.

2.2 Consumptive Use Algorithm (WATER)

A discussion of the computer program developed to simulate the cooling models discussed in the previous section is presented in the following paragraphs. A listing of the program is provided in Appendix B.

2.2.1 Overall Program Organization

Basically the model is structured to include the models presented in Section 2.0. A flow diagram depicting the overall structure of the model is shown in Figure 2. The program computes consumption or evaporative loss in terms of $\left(\frac{ft^3}{sec\text{-}1000\ MWt}\right)$. By comparison, assuming an overall thermal

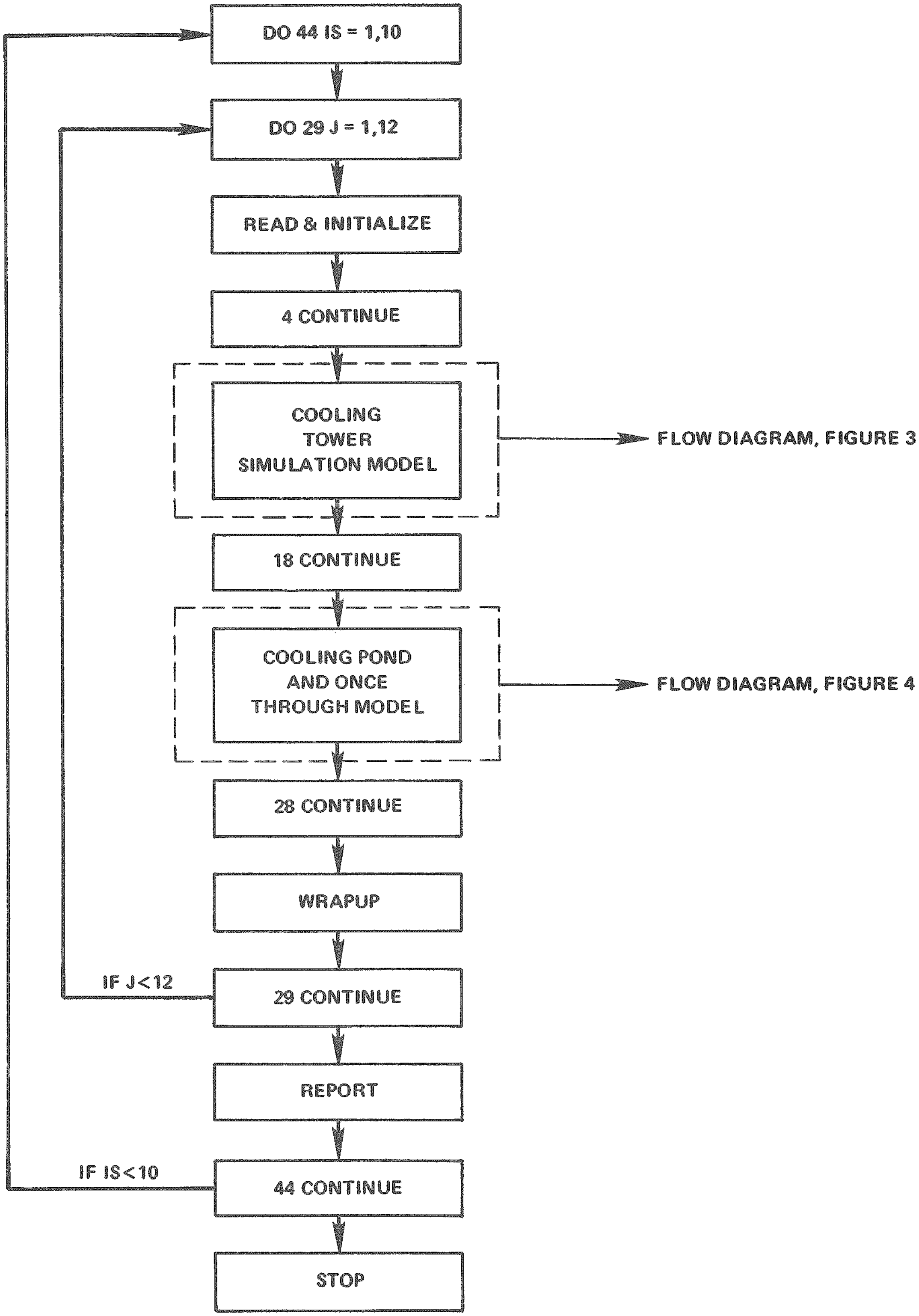

FIGURE 2. Flow Diagram for Consumptive Use Program (WATER). HEDL 7803-354.4

plant efficiency of 33% with all of the waste heat dissipated through the process of evaporation, the evaporation is equivalent to approximately .054 gal/kW-hr.

The program uses mean monthly weather data. Consequently twelve (12) passes are made consecutively through the program for each specific site examined. In turn the two major subsections (cooling towers, cooling ponds and once-through cooling) are further subdivided reflecting the specific features discussed in the previous section. In all, eight types of cooling towers, twelve types of cooling ponds, and four types of once-through cooling configurations are condensed in the analysis.

2.2.2 Cooling Tower Calculations

Eight types of wet evaporative cooling towers were examined. The eight types can be described in terms of a 2 x 2 x 2 matrix. Originally this matrix consisted of tower type (counter or cross), cooling range, and heat transfer coefficient. However, after some preliminary examination it was decided that the parameter liquid/gas flow ratio was more important than tower type and consequently this substitution was made. This observation is consistent with the fact that as long as the calculated approach was greater than approximately seven degrees the two models gave virtually the same results. Further comments on this subject are provided in References 13 and 14.

A heat transfer coefficient (Hg), air water interface area per unit volume, and water flow rate were selected to give an integral tower performance characteristic $\left(\frac{KAV}{L}\right)$ range from .43-1.3. The liquid/gas flow ratio was set at .75-1.50, and the cooling range ranged from 20-40^{o}F. A flow diagram for calculating cooling tower evaporation loss is shown in Figure 3.

2.2.3 Calculational Procedures for Cooling Ponds and Once-Through Cooling

The flow diagram for calculating consumptive demands for cooling ponds and once-through cooling systems is shown in Figure 4. Heat transfer

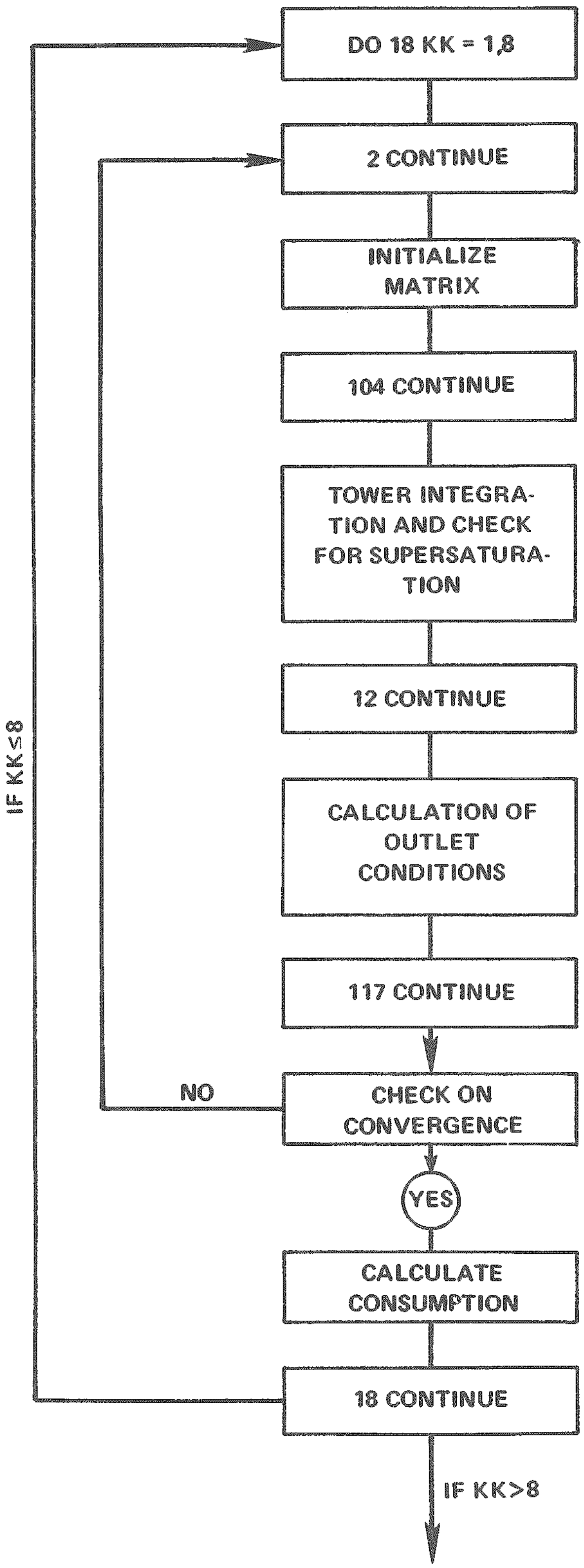

FIGURE 3. Flow Diagram for Cooling Tower Calculations. HEDL 7803-354.5

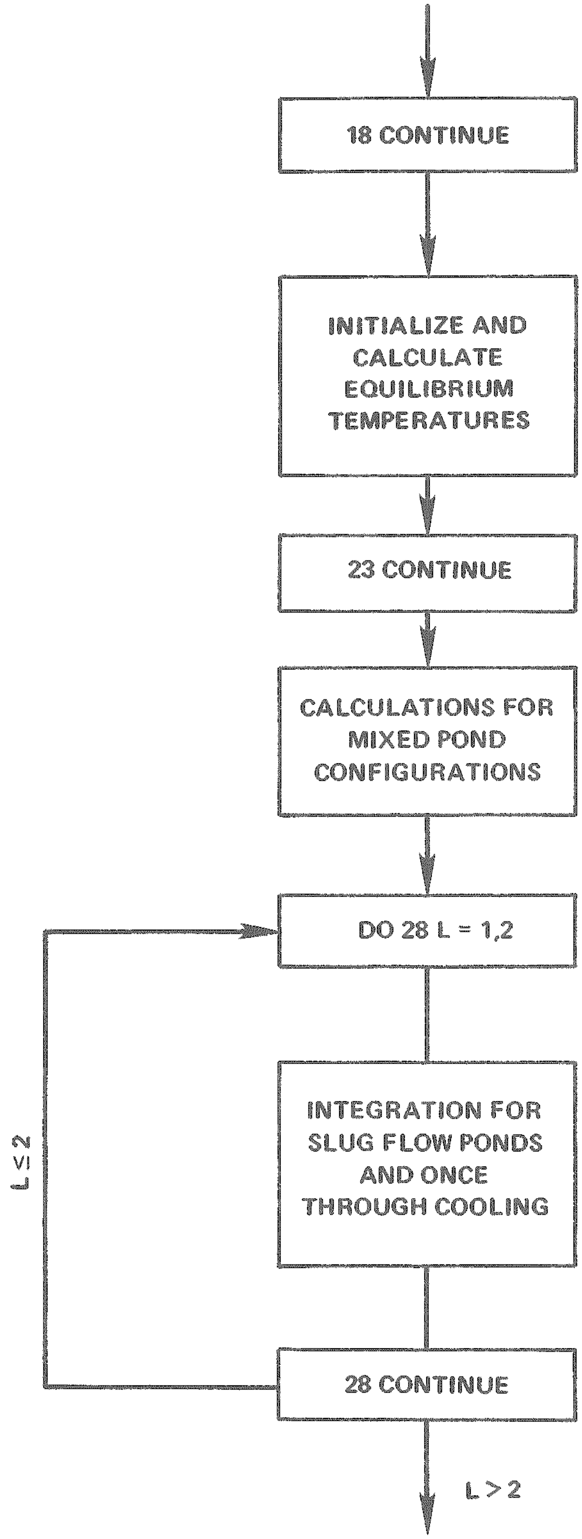

FIGURE 4. Flow Diagram for Cooling Ponds and Once-Through Cooling Calculations.

HEDL 7803-354.3

based upon the concepts of equilibrium temperature was used throughout the analysis. As indicated previously, two types of flow regimes were selected: the "mixed" pond, in which the surface water temperature is uniform, and the "slug flow" pond, in which the water gradually cools as it traverses the pond. Similar to the procedures used for calculating cooling tower consumption, three parameters: type (man-made or natural), size (1 acre/MWt and 3 acre/MWt), and cooling range (10^{0}F and 30^{0}F) were selected as a means of computing makeup or consumptive requirements. For the case of the mixed cooling pond the analysis is independent of cooling range and consequently only four cases (2 x 2 matrix) were needed for the analysis. With respect to the listing, these calculations are performed at statement 206-209. For the case of the slug flow ponds eight cases (2 x 2 x 2 matrix) were examined. A DO loop was included (DO 129) to account for the cooling range.

3.0 RESULTS

Consumption use calculations were performed for 24 cooling configurations on a monthly basis for 230 locations in the contiguous United States. The results of these calculations are shown in Appendix C. Mean monthly meteorological data were used for all stations.

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APPENDIX A

COMMENTS ON THE CONCEPT OF EQUILIBRIUM TEMPERATURE AND EXCHANGE COEFFICIENT

The energy budget has been developed over the years as a means of calculating water temperature. Basically an energy balance is performed on a parcel of water allowing heat transfer to occur at the free surface.

In this study the heat transfer was modeled using the concept of equilibrium temperature and an overall or average exchange coefficient. By definition, the equilibrium temperature is that temperature (water) at which the next exchange of energy between the water and air is zero for a given set of meteorological conditions. Although the concept has been accepted for years, it has only recently been accepted as a means of calculating the heat flux over a water body. The equilibrium temperature concept proposed by Edinger, et al.,[1] provides a convenient way of representing the complex heat transfer process and is subsequently reviewed.

The heat exchange can be expressed as:

$$Q_T = Q_s + Q_a - 1801 \left[\frac{T_w}{460}\right] + 1^{4} \quad -f(u)\ (e_w - e_a) - 0.26f(u)(T_w - T_a) \qquad (1)$$

Defining the exchange of energy in terms of an average exchange coefficient yields

$$\Delta Q_T = -K(T_w - E) \qquad (2)$$

where E = equilibrium temperature (^{o}F)

K = exchange coefficient (Btu/ft^2-day-^{o}F)

Letting $T_w = E$ with the corresponding saturation vapor pressure on any equilibrium temperature $e_w = e_E$, Equation (1) becomes

$$Q_s + Q_a = 1801 \left[\frac{E}{460} + 1\right]^4 \quad +f(u)(e_E = e_a) + 0.26f(u) \quad (E - T_a) \qquad (3)$$

Equation (3) equates the incoming total radiant energy flux to the equilibrium conditions.

Edinger[1] proposed the following linearizing assumption:

$$(e_W - e_E) = \beta_1(T_W - E) \tag{4}$$

where β_1 = the proportionality factor (mm Hg/^oF)

Brady[2] has proposed the following functional relationship for β_1:

$$\beta_1 = 0.255 - .0085\ T_W = .000204\ T_W^{\ 2} \tag{5}$$

Subtracting Equation (3) from (2), substituting Equation (4) into the remainder, and truncating the binomial expansion of the back radiation term (retaining only the linear portions) results in:

$$\Delta Q = -\left[15.71(T_W-E) + f(u)\beta_1(T_W-E) + 0.26\ .\ f(u)\right](T_W-E) \tag{6}$$

Simplifying Equation (6) results in the following:

$$\Delta Q = \left[15.7 + (0.26+\beta_1)\ .\ f(u)\right](T_W-E)$$

and the exchange coefficient is defined as

$$J = \left[15.7 + (0.26 + \beta_1)\ .\ f(u)\right] \tag{7}$$

An expression for the equilibrium temperature in terms of the exchange coefficient can then be obtained by substituting

$$f(u) = \frac{K - 15.7}{(0.26+\beta_1)}$$

into Equation (3) using the following linear approximation for the vapor pressure term:

$$e_E = (\beta_1)E + C(\beta)$$

and retaining only the linear and quadratic terms of E in the binomial expansion, i.e.,

$$\left[\frac{E}{460} + 1\right]^4 = 6\left(\frac{E}{460}\right)^2 + \frac{4E}{460} + 1$$

and simplifying results in the following equation

$$E + \frac{0.051E^2}{K} = \frac{(Q_s+Q_a) - 1801}{K} + \frac{K - 15.7}{K} \quad \frac{e_a - C(\beta)}{0.26+\beta_1} = \frac{0.26\ T\ a}{0.26+\beta_1} \qquad (8)$$

Solutions for K and E are then obtained by trial and error. For a complete discussion of the development and method of solution, the reader is referred to Reference 1. Detailed descriptions of other solution techniques are presented in References 2 and 3.

The equilibrium temperature concept is useful in providing a single "sink" temperature for effective simulation of the several natural sink temperatures (e.g., wet bulb for evaporation, dry bulb for convection, etc.). Similarly, the use of the exchange coefficient permits a considerable simplification of complex problems.

Discussion of Energy Budget Concept

Employing the energy budget approach, the net or total heat exchange from the surface of a body of water can be expressed as:

$$\Delta Q_T = Q_s + Q_a - Q_b - Q_e \pm Q_c \qquad (9)$$

where Q_s = the absorbed short wave solar radiation

Q_a = the absorbed long wave atmospheric radiation

Q_b = the long wave back radiation from the water body

Q_e = the heat loss by evaporation

Q_c = the heat gained or lost by conduction

The specific representation for each of these terms has been developed over the years. The expressions are semi-empirical in nature. The primary credit for the development of this methodology has been attributed to the Lake Hefner and Lake Mead studies[4,5] conducted by the U.S. Geological Survey. Since that time, a number of researchers have elaborated on this technique. In the following discussion of each of these terms, it is worthwhile recalling that the first two terms on the right side of Equation (9) are independent of the water temperature, whereas the final three are not. For a more complete discussion of the terms, the reader is referred to Reference 6 which provides an excellent treatise on the subject of heat transfer from the surface of a cooling pond.

1. Short-wave Solar Radiation

Incoming short-wave solar radiation (wave length 0.30 to 3.0 microns) is normally measured by the use of a pyroheliometer. Such data are collected by the U.S. Weather Bureau at a number of locations throughout the United States. This term can vary from 400 to 2800 Btu/ft^2-day, depending on location and meteorological conditions.

Empirical formulations have also been developed to estimate this source. The formulation considered most representative was proposed by Laevasta.[7]

$$Q_s = 0.014 S_{an} T_d \, (1 - 0.0006 C_L^{\,3}) \text{ langleys/day} \qquad (10)$$

where S_{an} = noon sun angle

t_d = length of the day

C_L = the average cloud cover in tenths of the total sky

Whenever possible, the measured insolation should be employed. The actual net insolation used by Jaske[8] is:

$$Q'_s = \alpha(Qs)$$

The absorbtivity coefficient, α, was obtained from the material presented by Budyko.(9) The coefficient, summarized in Table 1A, is a function of time and location.

2. Long-wave Atmospheric Radiation

The atmospheric radiation term, which is generally between 2400 and 3200 Btu/ft^2-day, is a function of a number of variables. There are two empirical formulations commonly used. Brunt(10) has proposed the following formulation:

$$Q_a = 4.5 \times 10^{-8} (T_a + 460)(C_1 + 0.31 e_a) \text{ Btu/ft}^2\text{-day} \quad (11)$$

where T_a = air-temperature in °F measured about six ft above the water surface

e_a = air-vapor pressure in mm-Hg measured about six ft above the water surface

C_1 = a coefficient determined from the air-temperature and ratio of the measured solar radiation to the clear-sky solar radiation.

A procedure for computing Q_a suggested by Edinger, et al.(1) is first to compare the measured short-wave radiation to the clear-sky radiation (Figure 1A). Using this ratio and the air temperature, enter Figure 2A to obtain the Brunt coefficient (C). The long-wave atmospheric radiation can then be calculated using Equation (11).

A second procedure was first suggested by Raphael,(11) who proposed the following empirical representation:

$$\frac{Q_a}{\sigma T_a^4} = a + be_a = \beta \quad (12)$$

TABLE 1A

VALUES FOR THE ABSORBTIVITY COEFFICIENT

AS A FUNCTION OF CALENDAR TIME AND LATITUDE

LAT.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	JULY	AUG.	SEPT.	OCT.	NOV.	DEC.
70 N	----	0.77	0.84	0.89	0.91	0.91	0.91	0.90	0.87	0.85	----	----
60	0.80	0.84	0.89	0.92	0.92	0.93	0.92	0.91	0.90	0.86	0.81	0.79
50	0.84	0.88	0.91	0.93	0.93	0.94	0.93	0.93	0.92	0.89	0.86	0.84
40	0.89	0.91	0.92	0.93	0.94	0.94	0.94	0.94	0.93	0.92	0.89	0.88
30	0.91	0.92	0.93	0.94	0.94	0.94	0.94	0.94	0.94	0.93	0.92	0.91
20	0.93	0.93	0.94	0.94	0.94	0.94	0.94	0.94	0.94	0.94	0.93	0.93
10	0.94	0.94	0.94	0.94	0.94	0.94	0.94	0.94	0.94	0.94	0.94	0.93
0	0.94	0.94	0.94	0.94	0.94	0.94	0.94	0.94	0.94	0.94	0.94	0.94

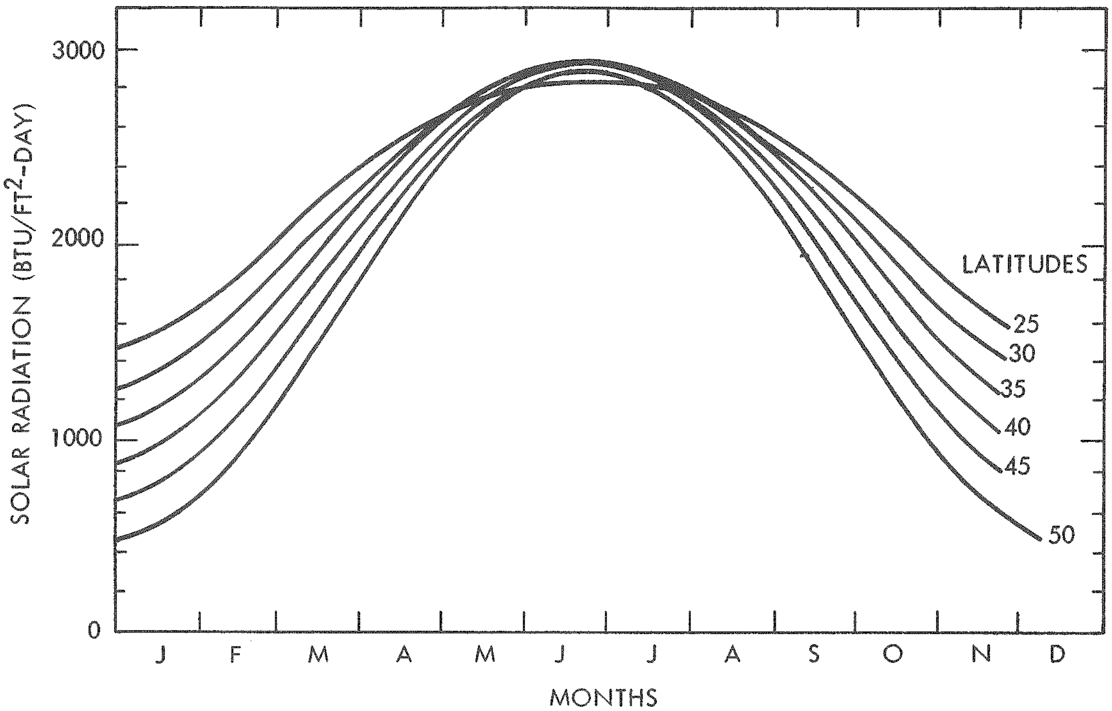

FIGURE 1A. Clear Sky Solar Radiation.

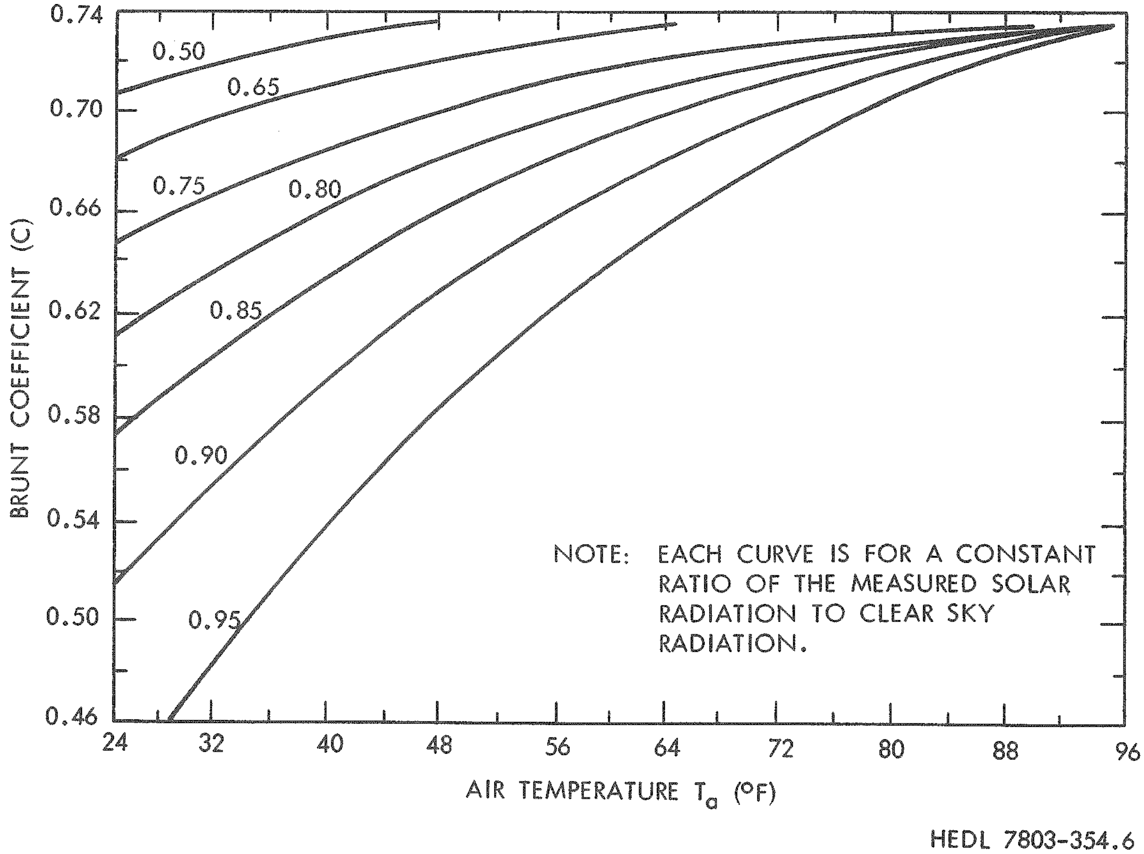

FIGURE 2A. Brunt Coefficient (C) from Air Temperature (T_a) and Ratio of Measured Solar Radiation to Clear Sky Radiation.

where $a = 0.740 + 0.025\ C_L\varepsilon - 0.0584\ Z_C$

$b = 0.00490 - 0.00054\ C_L\varepsilon - 0.060\ Z_C$

e_a = vapor pressure of air (mb)

β = radiation factor

Z_C = cloud height (ft)

σ = Stefan-Boltzmann Constant 4.5×10^{-8} Btu/ft^2-day-$^oF^4$

ε = Naperian base.

Since most weather observations available to engineers give the amount of cloud cover in tenths of sky obscured without distinguishing cloud heights, Raphael concluded that he could reduce the analysis by proposing that the radiation factor (β) could be related to the amount of cloud cover and the associated vapor pressure similar to the Brunt Analysis. This information is shown in Figure 3A. In Reference (27) the information shown in Figure 3A was placed in functional form to facilitate its use in a digital computer program.

3. Back Radiation

Water radiates energy in the form of long wave radiation as an almost perfect black body at a rate on the order of 2400-3600 Btu/ft^2-day. The rate at which heat is lost by this mechanism can be computed from the Stephan-Boltzmann fourth power law:

$$Q_b = \gamma_w'\sigma(T_w + 460)^4 \text{ Btu/ft}^2\text{-day} \tag{13}$$

where γ_w' = emissivity of water (0.97)

T_w = temperature of surface water

4. Evaporation

Heat loss from a body of water to the atmosphere through the evaporation of water occurs at a rate of a few hundred to several thousand Btu/ft^2-day,

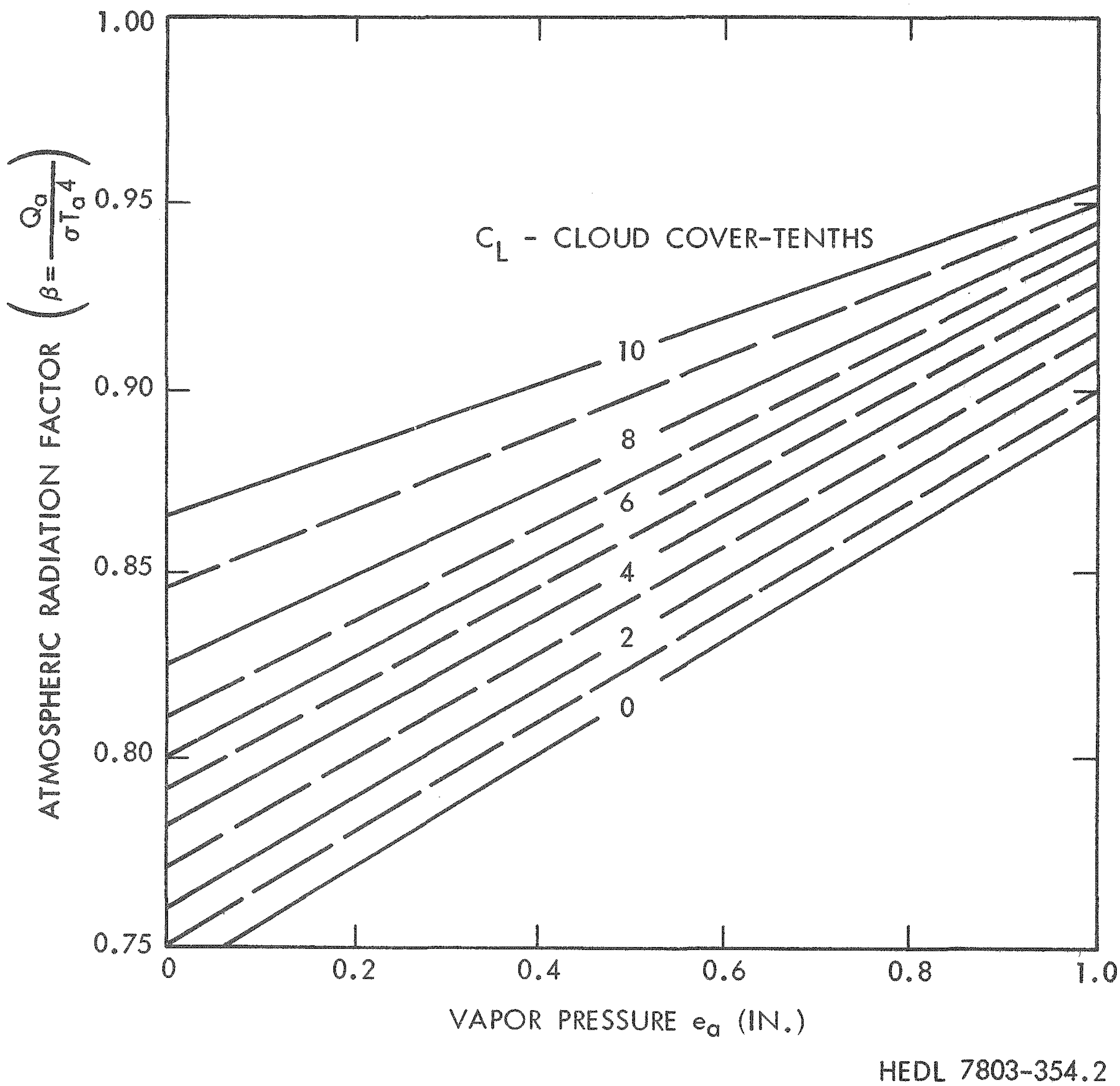

FIGURE 3A. Atmospheric Radiation Factor, β.

depending upon water temperature and local meteorology. There have been a number of semi-empirical models proposed for the evaporative term. These expressions take the form:

$$Q_e = f(u)\ e_w - e_a \text{ Btu/ft}^2\text{-day} \qquad (14)$$

where e_w = saturated vapor pressure at the temperature of surface water. In this expression, the term $(e_w - e_a)$ is the driving force commonly referred to as the "Dalton Difference". The wind speed function f(u) is described through a polynomial:

$$f(u) = a' + b'u + c'u^2$$

where u = wind speed (mph)

a', b', c' = empirically derived coefficients

Four of the more common models are:

Meyer formula(1)

$f(u) = 73 + 7.3\ u$

Lake Colorado City formula(1)

$f(u) = 16.8\ u$

Lake Hefner formula as expressed in(1)

$f(u) - 11.4\ u$

Brady formula(17)

$f(u) = 70 - .7\ u^2$

5. Conduction

Heat can either enter or leave water by conduction, depending upon whether the air temperature is greater or less than the water temperature. Generally this transfer is small, less than 500 Btu/ft^2-day. The rate at which heat is conducted between two media through molecular motion is expressed by Fourier's Law. Similarity arguments relating heat transfer and mass transfer have been used by Bowen to arrive at a proportionality between heat conduction and heat loss by evaporation.

$$B = \frac{Q_c}{Q_e}$$

where B = the Bowen ratio given by:

$$B = C_2 \left(\frac{T_w - T_a}{e_w - e_a}\right) \left(\frac{P_a}{760}\right)$$

where P_a = barometric pressure (mm hg)

C_2 = experimentally determined coefficient = 0.26

Using the general form of the evaporation model, the rate at which heat is conducted to or from the surface of the water can be written:

$$Q_c = 0.26\ f(u)(T_w - T_a)\ \text{Btu/ft}^2\text{-day.} \qquad (15)$$

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