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MATHEMATICAL MODELS FOR PREDICTING THE THERMAL PERFORMANCE OF CLOSED-CYCLE WASTE HEAT DISSIPATION SYSTEMS

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ABSTRACT

The literature concerning the mathematical modelling of the thermal performance of closed-cycle waste heat dissipation systems for the steam-electric plant is critically examined. Models suitable for survey analysis of waste heat systems are recommended. The specific models discussed are those for a mechanical draft evaporative cooling tower, a natural draft evaporative cooling tower, a spray canal, a plug-flow cooling pond, and a mechanical or natural draft dry cooling tower. FORTRAN computer programs of these models are included to facilitate their application.

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CHAPTER 1

INTRODUCTION

A study of mixed-mode waste heat dissipation systems for steam-electric plants has been performed in the Nuclear Engineering Department [28]. Mixed-mode waste heat dissipation systems are defined as those waste heat dissipation systems composed of combination of two or more different heat rejection devices or those waste heat dissipation systems operated with variable cycles. This report summarizes the mathematical thermal-performance models of the various component waste heat systems which were developed in the completion of this study. The thermal-performance models presented in this report are those for a mechanical-draft evaporative cooling tower, a natural-draft evaporative cooling tower, a spray canal, a plug-flow cooling pond, and a natural or mechanical draft dry cooling tower. These models are recommended for survey-type analyses of waste heat rejection systems.

Chapter 2 includes a review of literature pertaining to the mathematical modeling of each of these heat rejection devices and a discussion of the assumptions inherent to the recommended models. Appendix A contains FORTRAN program listings of the recommended models and a description of the required input data for each of the models.

CHAPTER 2

MATHEMATICAL MODELS FOR PREDICTING THE THERMAL PERFORMANCE OF CLOSED-CYCLE WASTE HEAT DISSIPATION SYSTEMS

2.1 Introduction

The literature concerning the dissipation of waste heat from central power stations has grown rapidly in the last decade. All areas within the general category - from biological effects to heat transfer developments - have been the subject of an increasing number of technical reports, journal articles, and trade magazine articles.

The two fundamental reasons for the rapid growth of this literature are the imposition of environmentally-motivated governmental regulations on the traditional "once-through" cooling system and the increasing unavailability of adequate sources of "once-through" cooling water at otherwise attractive central power station sites.

However, there is as yet no definitive source of information from which one can independently construct reliable thermal behavior and economic models of waste heat dissipation systems. The few studies which have addressed the general problem of developing the independent capability of evaluating the thermal performance of alternative waste heat dissipation systems are either out-of-date [9] or lacking in the details [3] [10] and thus can not be directly applied to the present task. Thus, considerable effort was required to review the available information and compile it into a useful tool for evaluating the costs/benefits of various alternative waste heat dissipation schemes.

The available literature concerning the mathematical modeling of the economics and thermal behavior of waste heat systems has been authored primarily by 1) the vendors of waste heat dissipation equipment, 2) the electric utility industry, and 3) various research institutes and universities. In view of the present task of developing accurate mathematical models of conventional waste heat rejection devices some general comments can be made about the literature with regard to its authorship.

Although there has been a tremendous increase in the waste heat dissipation equipment vendor sector in both size and diversity, the publications of these vendors are generally qualitative in nature. With a few notable exceptions, the literature published does not deal quantitatively with thermal behavior analysis, but, rather, describes qualitatively the particular vendors present capabilities and highlights the economic advantages of the particular vendors devices. Little of this information is of value to those interested in developing an independent analysis capability.

The dearth of substantial information published by equipment vendors is, of course, understandable since their proprietary interests are not well served by the free-flow of their costly research and development results.

The literature on this topic authored by the electric utility industry has come from the electric utilities themselves as well as their consultants - mainly the large architectural engineering firms. As is the case above, little substantive information has been published with regard to the mathematical modeling of the thermal behavior of various heat rejection systems by this sector. However, valuable government-sponsored information has been reported by architectural engineering forms. Many trade journal articles which review the waste heat dissipation solutions applied to specific sites have been authored by utility system engineers, but these findings are usually of little value to the present task.

Much useful information concerning the mathematical modeling of the thermal performance of heat rejection systems has been authored by various research institutes and universities under the sponsorship of federal and state agencies and electric utilities. In applying some of this information, however, difficulty is encountered in attempting to relate the published results to the actual thermal performance of modern, well-designed waste heat dissipation systems.

2.2 Mechanical Draft Evaporative Cooling Towers

2.2.1 Literature Review

Croley et al. [2] have recently addressed the problem of developing an accurate thermal and economic model of conventional cross-flow mechanical draft evaporative cooling towers. Their review of the literature led then to the use of a thermal analysis model based on a simple straightforward finite-difference solution of the well-known Merkel [11] evaporative heat transfer differential equation.

The Merkel formulation of evaporative heat transfer combines the mass transfer (evaporation) and the sensible heat transfer coefficient into a single coefficient. The approximate net energy transfer is then a product of the coefficient and the enthalpy potential difference between the water and the air streams. The standard "Merkel" equation is as follows:

$$\frac{KaV}{L} = \int_{T_2}^{T_1} \frac{dT}{(h''-h)} \tag{2.1}$$

where K = overall transfer coefficient, lb/(hr)(ft^2 of interface)(lb of water/lb of dry air)

a = interfacial contact area (ft^2/ft^3 of tower fill)

V = planar volume (ft^3/ft^2 of plan area,)

L = water flow rate ($lb/hr\text{-}ft^2$ of plan area),

T_1 = inlet water temperature, and

T_2 = exit water temperature.

dt = water temperature differential

h'' = enthalpy of saturated air at the water temperature

h = enthalpy of the main air stream (BTU/lb of dry air)

Derivation of this relationship may be found in several references [26] [27]. Physically the quantity KaV/L in the above equation represents an effective heat transfer ability or "number of transfer units" for a particular cooling tower. This coefficient is dependent on the relative amounts of water and air flow in the tower and must be determined experimentally.

Croley et al. [2] have applied this differential equation in finite-difference form to solve the two-dimensional heat exchange problem of the widely-utilized induced draft crossflow evaporative cooling tower for known inlet air and water boundary conditions. The finite-difference approximation to the Merkel equation consists basically of the division of the energy transfer volume into a number of equal sized blocks over which the energy transfer potential (enthalpy) is averaged.

The conclusions of Croley et al. concerning the utility of the basic Merkel formulation for the predicting of the energy transfer in a cooling tower has since been substantiated by the recommendation of Hallet [12]. Hallet, representing a leading cooling tower vendor, has suggested that the best approach (for a non-vendor) to the problem of evaluating the thermal performance of wet cross-flow towers is a finite-difference solution of the basic Merkel equation. This author also points out that, although many improvements in the theory of simultaneous heat and mass transfer at water/air interfaces have been suggested, the basic Merkel formulation is the only widely accepted and proven theory.

The analysis technique suggested by Hallet is essentially identical to that of Croley et al. except that Hallet recommends the inclusion of a temperature dependence in the expression for the tower fill energy transfer coefficient:

$$Ka = f(T_1) \tag{2.2}$$

where T_1 is the tower inlet water temperature. It is interesting to note that no physical justification is given by Hallet for this "temperature effect". Consideration of recent works which address the errors inherent to the Merkel equation suggest that this "temperature effect" fixup is necessary because of errors in the Merkel approximate formulation for evaporative heat transfer.

The investigations of Nahavandi [13] and Yadigaroglu [14] have been concerned with an evaluation of the errors inherent to Merkel equation. The results of Yadigaroglu are based on a comparison of the predictions of the Merkel theory and a more exact and complete theory which treats the mass and sensible heat processes separately. This investigator found that the effect of the various approximations of the Merkel theory tends to be small since the different approximations of the Merkel theory result in partially cancelling positive and negative errors. The conclusion is that, given the other errors associated with cooling tower performance predictions (uniform air and water flow rates, for example) and performance verifications (experimental uncertainties), the added complexity of performing the more exact energy transfer analysis is not justified. Nevertheless, it is of interest to note that Yadigaroglu found that the net positive error in predicting the cooling range increased with increasing air inlet temperature and humidity. This error could be corrected by arbitrarily decreasing the value of KaV/L by the appropriate amount as the water inlet temperature increased. Indeed, this is the same, but unjustified, approach recommended by Hallet. Examining the magnitude of the over-prediction resulting from the use of Merkel theory (on the order of 5%), it is found that the Merkel theory error is consistent with the suggested "temperature effect" correction of KaV/L (about 5% per 10 °F rise in inlet water temperature for inlet water temperatures in excess of 90 °F).

2.2.2 Selection of a Model

The mathematical model to be used in the prediction of the thermal performance of mechanical draft evaporative cooling towers is the finite-difference approximation of the Merkel equation. The finite-difference approximation to the Merkel equation can be stated as [2]

$$G(h_o - h_i) = \frac{KaV}{N}\left[\frac{h_i' - h_i + h_o' - h_o}{2}\right] \tag{2.3}$$

where h_i' and h_o' = saturated air enthalpies at the inlet and outlet of an incremental element,

h_i and h_o = saturated air enthalpies at the temperature of the water entering and leaving the incremental element,

G = air flow rate per incremental element,

N = square root of the number of incremental elements, and

KaV = transfer coefficient.

The application of this equation to the cross-flow problem of a conventional induced draft cross-flow cooling tower is shown in Fig. 2.1.

In addition to the above equation, the energy balance equation

$$G(h_o-h_i) = c_p L(t_i-t_o) = L(h_i^{'}-h_o^{'}) \qquad (2.4)$$

is needed to completely describe the temperature history of the air and water as it passes through the tower fill. In the above equation:

L = water loading per incremental element,

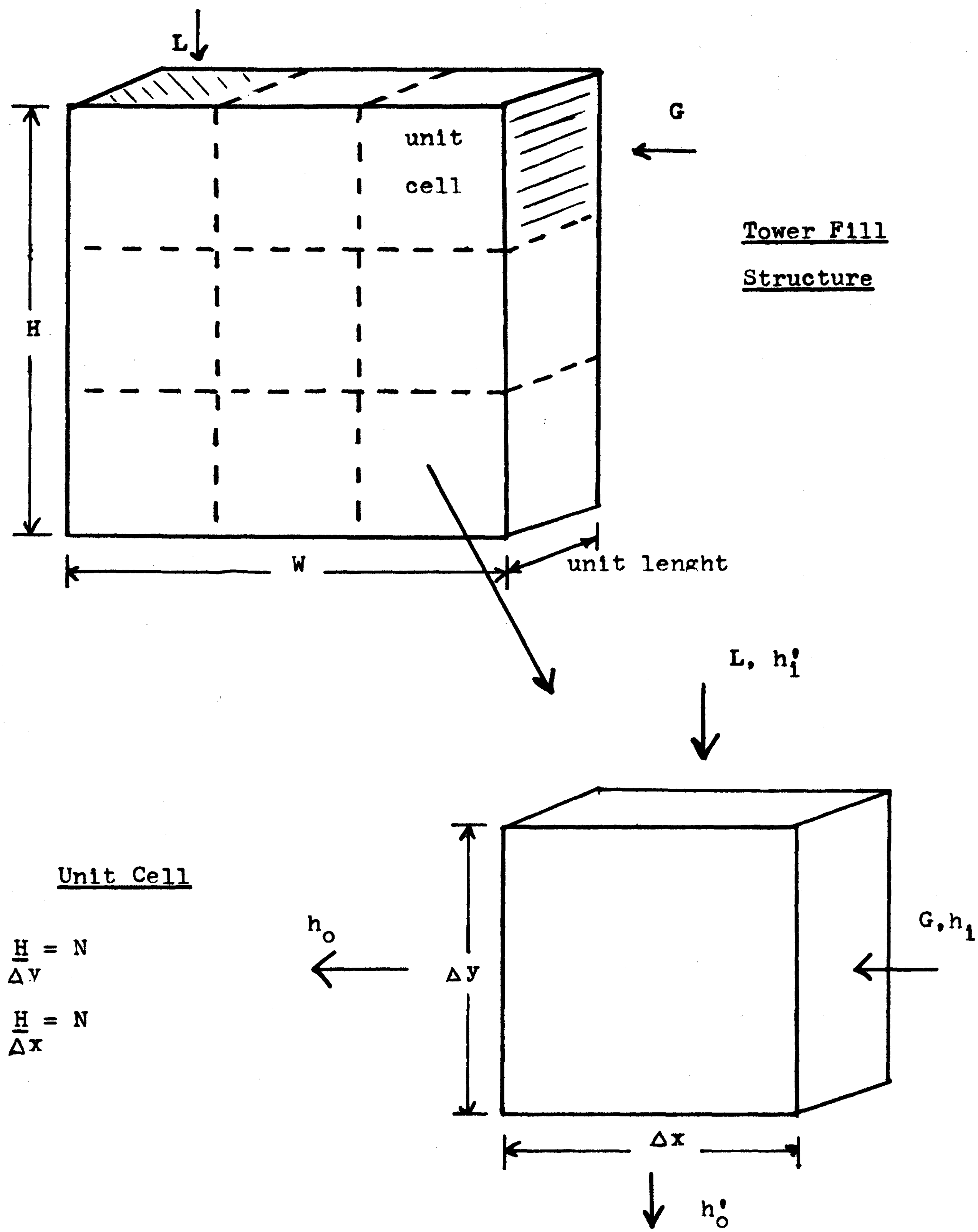
t_i and t_o = inlet and outlet water temperatures for an incremental element, and

c_p = specific heat capacity of water.

Equations 2.3 and 2.4 form a set of coupled equations with unknown variables h_o and $h_o^{'}$ which must be solved for iteratively. The algorithm for calculating the average outlet water temperature and average outlet air temperature is given in Fig. 2.2. Note thta, for practical purposes, the water and air flow rates are fixed by the tower design and to a good approximation can be assumed to be uniform and constant throughout the tower. Note, also, that the algorithm is for calculating the performance of a given tower design. If we wish to find the size of the tower needed to meet a

Fig. 2.1

Illustration of Tower Fill Finite-Difference Calculation



specific cooling requirement, a trial and error calculation may be performed.

The saturated air enthalpy used as the driving potential in the Merkel equation depends on both the dry bulb temperature and the humidity of the air. However, a good approximation to the enthalpy which depends solely on the thermodynamic wet bulb temperature may be derived. From Marks [15] we have the relationship,

$$E = 0.24Td + W(1062.0 + 0.44Td) \tag{2.5}$$

and

$$W = \frac{W^* - (0.24 + 0.44W^*)(Td - Twb)}{(1094 + 0.44Td - Twb)} \tag{2.6}$$

where E = enthalpy of moist air,

Td = dry bulb temperature,

W = specific humidity,

Twb = wet bulb temperature, and

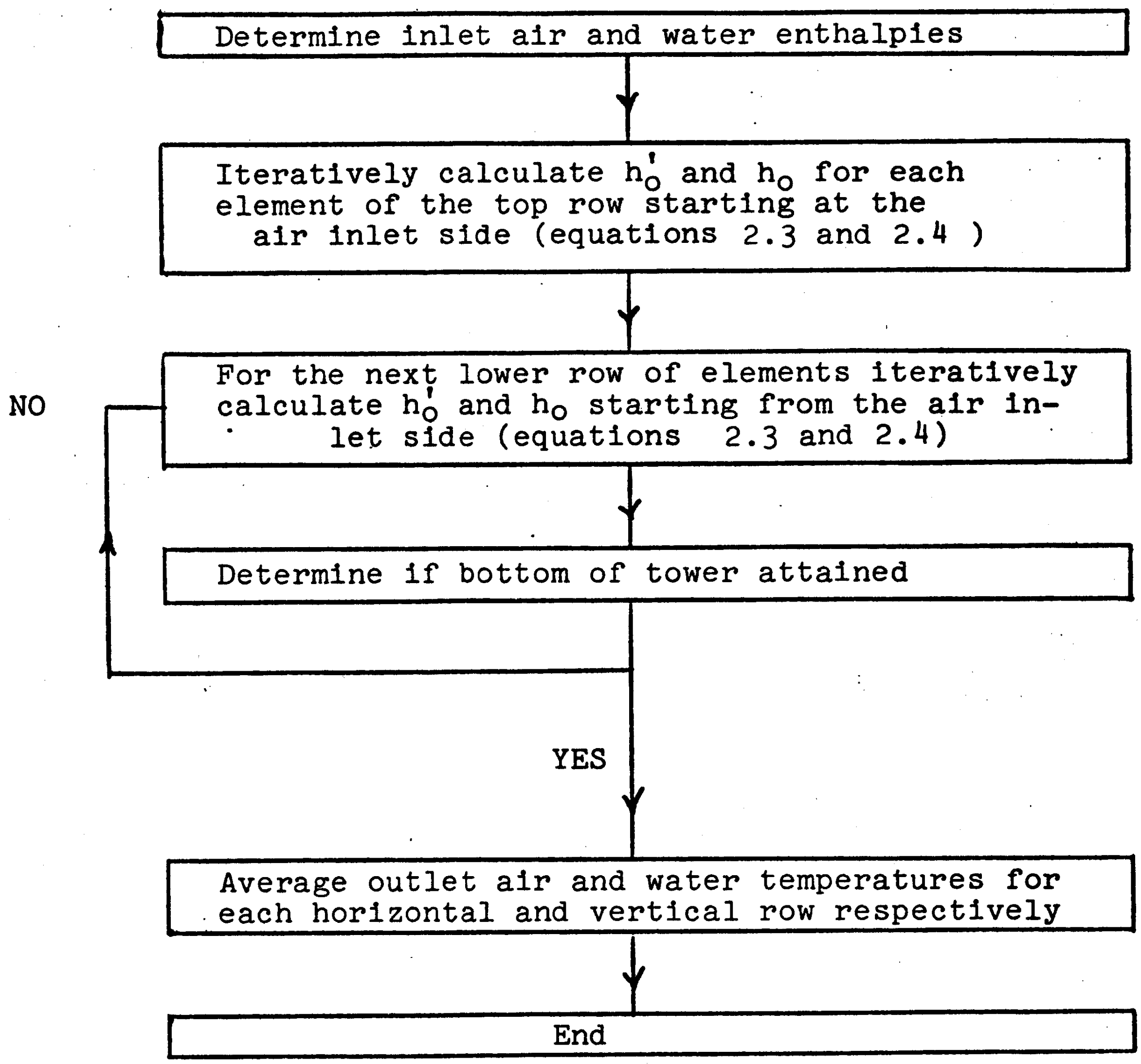
W^* = specific humidity for saturation at Twb.

Substituting the latter into the former we have

$$E = 0.24Td + W^*(1062 + .44Td) - \frac{(0.24 + 0.44W^*)(Td-Twb)(1062 + 0.44Td)}{(1094 + 0.44Td-Twb)} \tag{2.7}$$

Fig. 2.2

Calculational Algorithm for Predicting the Performance of Mechanical Draft Cross-flow Evaporative Cooling Tower (MECDRAFT Program)



Now assuming that in the deminator we can make the approximation

$$32 - Twb \approx 0 \qquad (2.8)$$

and expressing the saturation himidity in terms of saturation pressure we have

$$E \approx 0.24Twb + \frac{0.622P_{sa}}{P_{atm}-P_{sa}}(1062.0 + 0.44Twb) \qquad (2.9)$$

where P_{atm} = total atmospheric pressure, and

P_{sa} = saturation pressure of water vapor at Twb.

The above assumption is a good one in this particular circumstance since the error affects the ratio of large numbers. An error of 50 °F in magnitude in the denominator would be typical with the total resultant error being about 5%. However, in all applications of the approximate enthalpy equation the equation is ultimately used to find the difference of two enthalpies and thus the resultant error in the difference is minimal.

2.2.3 Application of Model

To achieve the goal of obtaining an accurate thermal performance model of a conventional cross-flow induced draft evaporative cooling tower module the physical dimensions and empirical heat transfer and air friction data for a typical module must be acquired. Croley et al. [2] have modeled the thermal behavior of such modules and reported the results. From the published information the physical dimensions of the tower fill are readily obtainable. They are

height = 60 feet

width = 36 feet, and

length = 32.

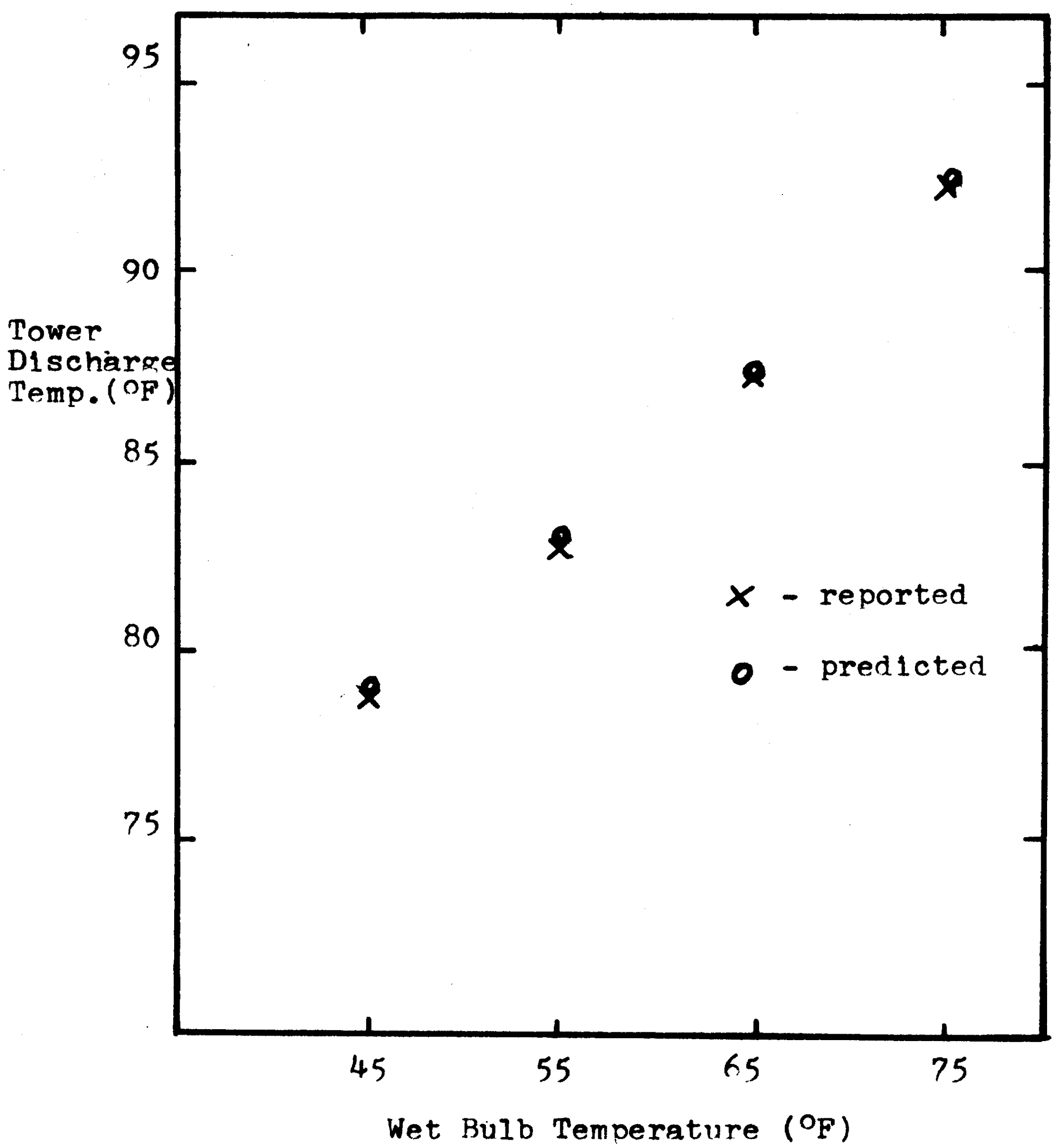
However, the air friction factors for this fill is not directly obtainable from the published results. Nevertheless, an energy balance on the modeled tower based on the published information indicates an average air flow rate of 2.4×10^3 lb_m/hr-ft^2. It will suffice for the purposes of this study to assume the air flow is constant and equal to this value. Croley et al. do not report the values of the energy transfer coefficient used in their study since empirical proprietary information was used in evaluating the energy transfer coefficient. However, sufficient calculational results using this proprietary information are reported to allow a regression of the required information.

The Cooling Tower Institute [16] states that the dependency of the energy transfer coefficient Ka on the air and water flow rates in a tower can be well expressed by a relationship of the form

Fig. 2.3

Comparison of Reported and Predicted Mechanical Draft Cooling Tower Performance

Water Loading = 6200 lbm/hr - ft^2
Air Loading = 1692 lbm/hr - ft^2



$$\dot{K}a = \alpha G^{\beta} L^{1-\beta} \quad (2.10)$$

where α depends on the fill configuration and β is, to a good approximation, equal to 0.6. Using the following expression

$$\alpha = 0.065 - (T_1 - 110.0)*(0.000335) \quad T_1 > 90°F$$

and (2.11)

$$\alpha = 0.0715 \quad T_1 \leq 90°F$$

where T_1 is the inlet water temperature, the performance predictions of Croley et al. based on proprietary data can be closely matched as shown in Fig. 2.3. This value of α is consistent with the type of fill used in modern towers and the values of α experimentally determined by Lowe and Christie [23].

2.3 Spray Systems

2.3.1 Literature Review

Spray cooling systems for the dissipation of waste heat at large central power stations are a relatively new concept [17]. As a consequence, the development of thermal analysis techniques for these systems is presently incomplete. The development of reliable mathematical prediction models has not been achieved and has been hindered by the complexity of the problem.

As opposed to cooling towers, the water-air interfacial area and relative air to water flow rates are not well defined for spray systems. Open to the atmosphere, variations in the ambient wind result in different spray patterns, different air flows through the sprays both in magnitude and direction, and different interference effects between the individual sprays. The spray canal system also presents a channel hydraulics problem in that the behavior of the water in the canal must be understood to insure optimum spray system performance.

Porter et al.[18] [19] have authored the only two presently available detailed works on the thermal performance of spray canals. The two papers represent two different approaches to the problem, one analytical and one numerical. Both models, however, are based on the same limited data which according to the authors result in optimistic predictions [20].

Richards of Rockford [4] have published some limited information concerning the application of their spray modules. They indicate that an empirical "NTU" approach is used in the basic heat transfer calculation. Most interesting, however, is their description of the flow requirements of the channel in which the spray modules are utilized since this description indicates their recognition of the importance of the channel thermal-hydraulics in the overall performance of the system.

2.3.2 Selection of Model

For the purposes of survey-type analyses, the numerical prediction of the thermal performance of spray canals as suggested by Porter et al. [19] is most advantageous. In this model the heat transfer ability of each spray module is defined by an empirical "NTU" or number of transfer units which is dependent on the ambient wind speed. The effects of air interference between individual sprays is considered through the use of an empirical air humidification coefficient. Given the ambient meterological conditions and inlet water temperature and flow rate, the calculational procedure is to march down the canal taking into account the cooling effect of each spray module as it is encountered. The basic calculational algorithm is given in Fig. 2.4.

The heat transfer equation used in the model is

$$\mathrm{NTU} = \frac{C_p(T_n - T_s)}{\frac{(h(T_s) + h(T_n))}{2} - h(Twb)} \qquad (2.12)$$

where C_p = specific heat capacity of liquid water,

T_n = temperature of water exiting spray nozzle,

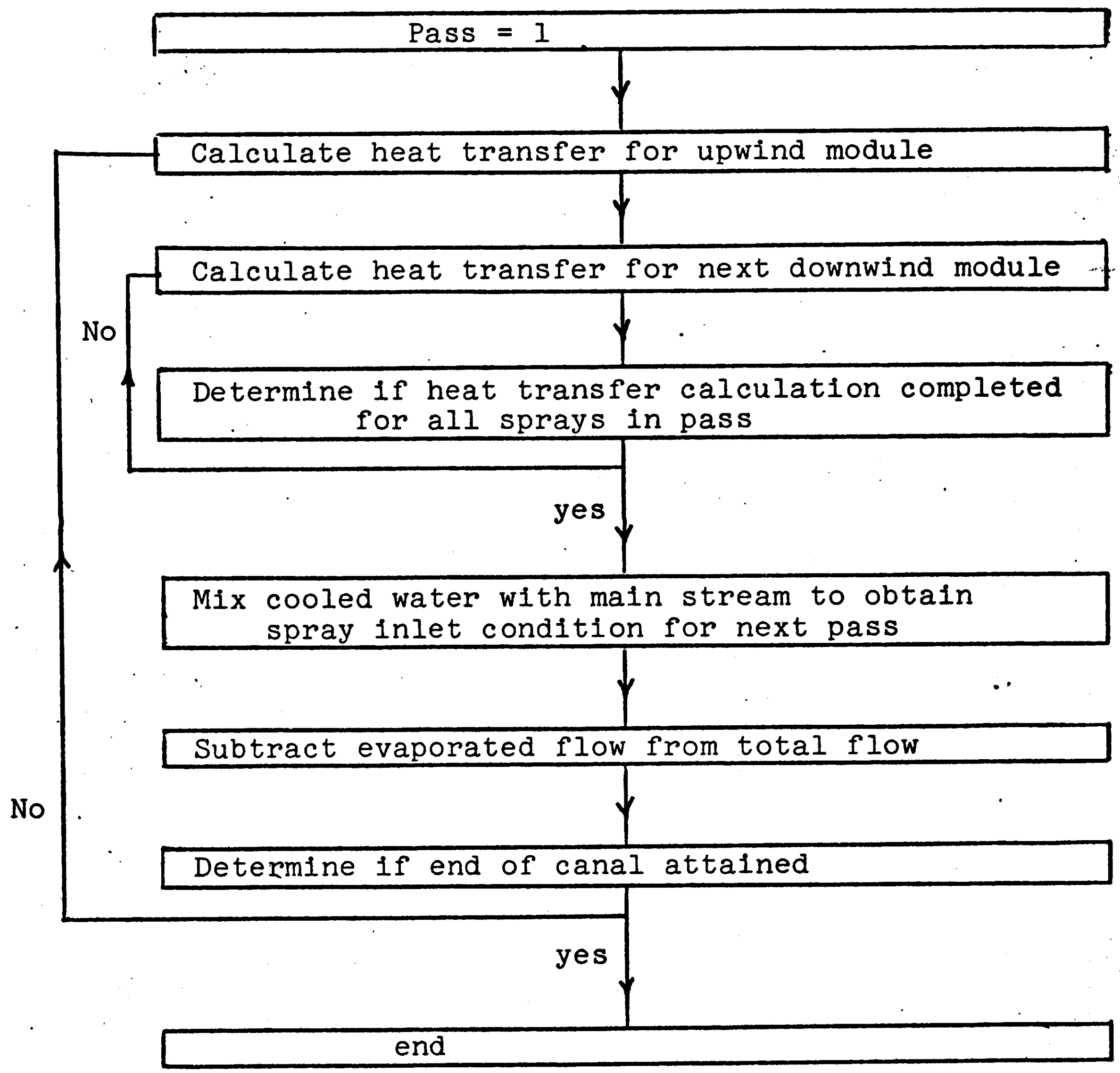
T_s = final spray temperature,

$h(T)$ = total heat or sigma function as defined by Marks [15],

Twb = local wet bulb temperature, and

Fig. 2.4

Computational Algorithm for Spray Canal Thermal Performance Model (SPRANAL Program)



NTU = number of transfer units of an individual module.

The total heat or sigma function used as the driving potential for the energy transfer is defined by Marks [15] as

$$\Sigma = h_m^* - W^* h_f^* \tag{2.13}$$

where h_m^* = enthalpy of moist air at the wet bulb temperature,

W^* = specific humidity for saturation at the wet bulb temperature, and

h_f^* = enthalpy of liquid water at the wet bulb temperature.

However, comparison of the sigma function and the enthalpy indicates that, for the temperature range and temperature differences of interest the following is a good approximation;

$$\Delta\Sigma(twb) \approx \Delta h(Twb) \tag{2.14}$$

where h is the enthalpy of saturated air at temperature Twb.

Since we are attempting to determine T_s by using Eq.(2.12) and T_s is a term in the same equation an iterative solution is necessary. The evaporated water loss is calculated using the expression of Porter [19]. It is

$$\alpha = C_p(T_n - T_s)/i_{fg}(1 + B) \tag{2.15}$$

where α = fraction of water evaporated in each spray,

i_{fg} = specific heat of vaporization of water, and

B = so-called Bowen ratio of sensible to evaporative heat transfer.

In the application of the above equations, the Bowen ratio can be conservatively set equal to zero, since, in any case, the effect of water evaporation on the spray canal thermal performance is minimal.

From the data given by Porter the relationship between the NTU and windspeed has been deduced to be approximated by

$$NTU = 0.16 + 0.053 * V \tag{2.16}$$

where V is the windspeed in miles per hour.

In this model no direct account is made of the thermal-hydraulic behavior of the water in the channel. However, Porter has made some simple arguments in favor of assuming that the channel is vertically fully-mixed between successive passes of sprays.

2.4 Natural Draft Evaporative Cooling Towers

2.4.1 Literature Review

Conceptually, the thermal analysis of natural draft evaporative cooling towers is a straightforward extension of the mechanical draft cooling tower analysis developed in this chapter. However, from a practical standpoint the problem is considerably more complex since the heat transfer characteristics and the air flow in the tower are dynamically coupled. Also, in addition to needing to know the empirical heat transfer coefficient of the fill, one also needs to know the empirical air friction factors for the tower structures and the fill. Further, a more exact determination of the psychrometric condition of the air exiting the fill is desirable since this condition ultimately determines the overall performance of the tower.

In the past, attempts have been made, notably by Chilton[21] to simplify the performance prediction for natural draft evaporative cooling towers by applying an empirical relationship for the overall thermal behavior. These efforts, however, were not well received and presently the suggested approach to the thermal analysis problem is based on a detailed evaluation of the important physical phenomena.

Keyes [22] has outlined the necessary steps for the construction of a thermal behavior model of natural draft cooling towers. Essentially, the mathematical modeling of a natural draft tower requires the solution of three coupled equations. The equations are 1) an energy balance between the air and water streams, 2) an energy transfer equation for the combined evaporative and sensible heat transfer, and 3) an energy equation for the density induced air flow through the tower. Keyes only reviews the general problem and discusses the empirical information which is available for accomplishing the modeling task.

Winiarski et al.[24] have developed a computer model of the thermal behavior of a natural draft cooling tower based on the three equations mentioned above. The author notes, however, that the model presented awaits final verification based on reliable test data from actual towers.

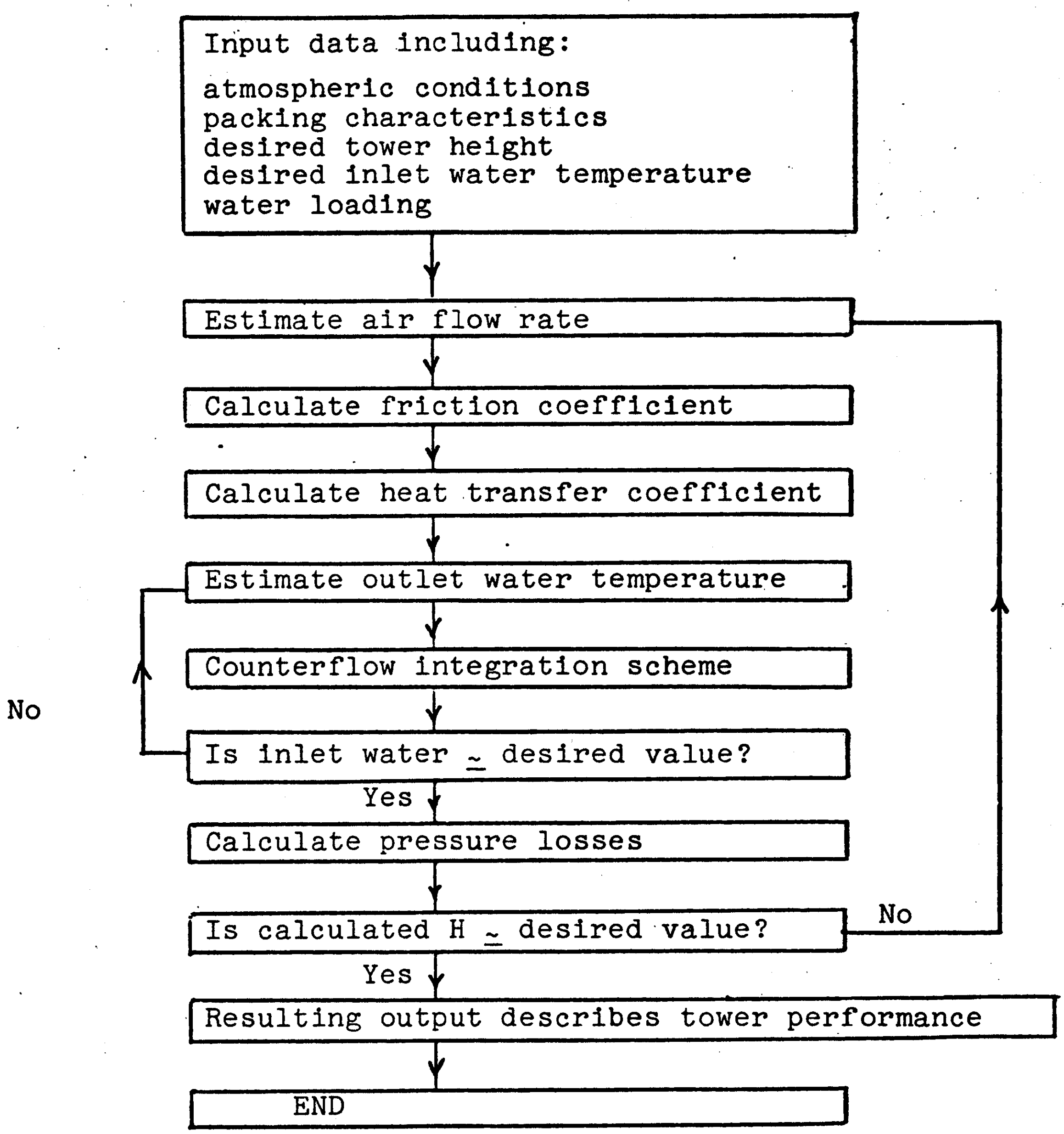
2.4.2 Selection of Model

The model of Winiarski et al. [24] has been chosen as the basis for the development of a thermal behavior model of natural draft evaporative cooling towers. The thermal analysis calculational procedure is reported in the form of a computer program. The basic computational algorithm is given in Fig. 2.5. The major remaining task in the model development was, thus, the acquisition of the necessary empirical information which would enable the computer program application. In this regard all domestic vendors of natural draft evaporative cooling towers were contacted and sufficient information was obtained.

The data obtained was not typical heat transfer coefficients and air flow friction factors for a modern natural draft tower but instead consisted of a set of typical perfor-

Fig. 2.5

Calculational Algorithm for Natural Draft Evaporative Cooling Tower Performance Model (NATDRAFT Program)



mance curves and tower and fill structural dimensions. Thus, is was required to fit the computer model to the performance curves by a trial and error selection of appropriate heat transfer coefficients and friction factors. The performanoe data are known to be based on roughened-surface parallel-plate-type tower fill with counter air/water flow. Rish [25] has reported an empirical relationship for the heat transfer coefficient and friction factors for smooth parallel plate packing. They are;

$$C_f = 0.0192(L/G)^{0.5} , \tag{2.17}$$

and

$$h = \frac{C_p C_f G}{2 + 71.6 C_f (\frac{L}{G})^{-0.25}} \tag{2.18}$$

where C_f = friction factor,

C_p = specific heat capacity of liquid water,

G = air flow rate lbm/ft^2-hr,

L = water flow rate lbm/ft^2-hr, and

h = heat transfer coefficient for evaporative and sensible heat transfer based on enthalpy difference potential.

It was assumed that the effect of the roughened surface of the parallel plates could be simply accounted for by a friction factor multiplier F_m. That is;

$$C_{fa} = F_m * C_f \qquad (2.19)$$

where C_{fa} is the actual friction factor. The relationship between the heat transfer coefficient and the friction factor was assumed to remain the same.

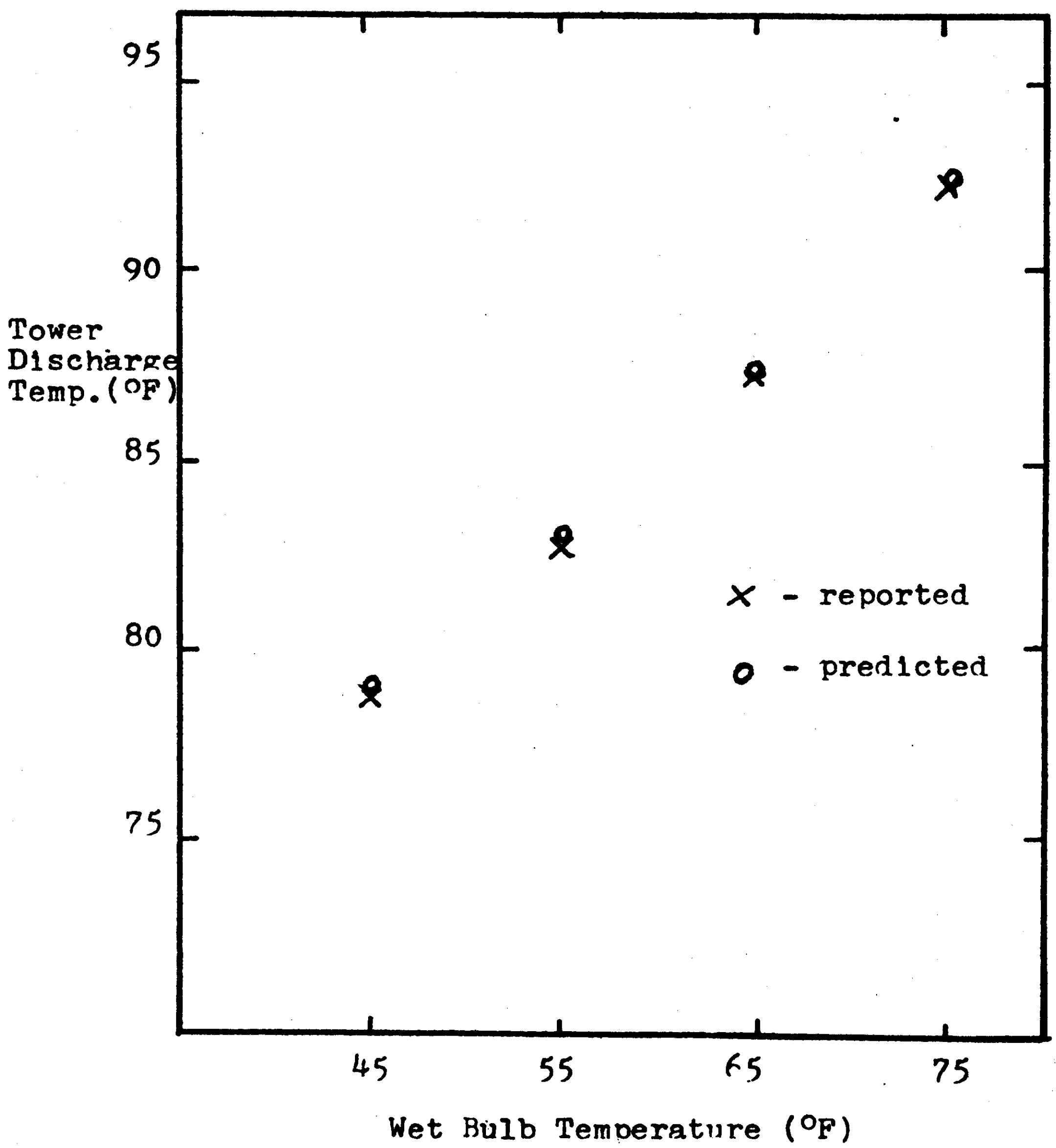
A trial and error approach to determining F_m was used and, as Fig. 2.6 indicates, a value of F_m of 3.2 gives excellent results over a representative range of operating temperatures and flow rates. In the determination of F_m all other air friction effects other than that of the fill were neglected.

All the details of the computer model will not be discussed here, but may be found in the original report. Nevertheless, some important points are worth mentioning. In this model, water vapor saturation of the air stream is not a basic assumption as was the case for the heat transfer model developed for the mechanical draft tower. Instead, the sensible heat transfer is calculated in addition to the total heat transfer due to both evaporation and sensible heat transfer. As in the mechanical draft tower model the transfer calculation is based on a finite-difference approximation to the Merkel Equation, but in this case the counter-flow of the air and water streams necessitates only a one-dimensional calculation.

Fig. 2.3

Comparison of Reported and Predicted Mechanical Draft Cooling Tower Performance

Water Loading = 6200 lbm/hr - ft^2
Air Loading = 1692 lbm/hr - ft^2



The calculation of both the total energy transfer and the sensible heat transfer allows the determination of the exact psychrometric condition (both dry bulb and humidity) of the air stream leaving each "cell" of the finite difference integration. The assumption of water vapor saturation, if it in fact did not exist, would result in an underestimate of the fill air exhaust dry bulb temperature and hence an underestimate of the induced draft.

To complete the thermal model of a natural draft tower a relationship between the tower height and the tower base diameter needed to be established for different sized towers. This was necessary because while a mechanical draft tower may be sized to a particular cooling duty by varying the number of tower modules, a natural draft tower is sized by varying the tower size. Flangan [7] has published data concerning the ratio of height to diameter for 16 large natural draft towers which indicates an average ratio of 1.248.

2.5 Cooling Ponds

2.5.1 Literature Review

The task of mathematically modeling the thermal-hydraulic behavior of a cooling pond is a problem which is substantially different from the problem of modeling cooling towers. This is because actual cooling ponds are not physically well-defined in the sense that the important parameters which determine their thermal behavior can not be assigned values which are representative of all, or even most. cooling ponds. In fact different cooling ponds may exhibit completely different types of thermal-hydraulic behavior each of which require different analysis approaches and techniques.

There are two idealized cases of pond thermal-hydraulic behavior which yield themselves to very simple analytical treatment [8]. These are termed the plug-flow and fully-mixed models. In plug flow there is no mixing between the discharge into the pond and the receiving water and the surface temperature, for steady-state conditions, decreases exponentially from the pond inlet to the pond outlet. The fully-mixed pond represents an extremely high degree of mixing of the discharge and the receiving water. Thus a uniform temperature over the entire pond results. In reality, the behavior of most ponds would fall between these two extreme cases. The plug flow pond represents the best possible heat dissipation situation since the temperature of the discharge is kept as high as possible. Conversely, the fully-mixed pond represents a lower bound on the heat transfer performance of the pond. The "worst case" performance, however, is a short-circuited pond. For either the plug-flow or fully-mixed model both steady-state and transient behavior can be readily calculated.

Ryan [5] reported the development of a transient cooling pond thermal-hydraulic model which was the first attempt to realistically mathematically model the actual physical process occurring in a cooling pond. Watanabe [6] extended the model and reported criteria for its applicability. This model is recommended for use as a design tool or means of evaluating the performance of cooling ponds relative to alternative waste heat disposal systems. However, since the model is not fully developed into a documented computer program its application appears difficult. Also, for the purposes of most surveys the computational time is excessive.

2.5.2 Selection of a Model

The task of formulating a representative thermal-hydraulic model of a cooling pond can be considered to be different from the task of formulating a model of a cooling pond which is to be used for design purposes. The present interest is in mathematically representing the approximate thermal-hydraulic behavior of a representative cooling pond. It is perceived that this limited goal can be accomplished through the use of a plug-flow, vertically-mixed pond model capable of accounting for variable meterological conditions, variable inlet temperatures, and variable flow rates. For a given cooling requirement such a model would tend to predict pond sizes which are smaller than would be normally required. Thus, if the model were to be used in a detailed economic comparison of alternative waste heat disposal systems the pond economics would be unduly favored.

The vertically-mixed, plug-flow model predicts the transient pond behavior by following a slug of water of uniform temperature through the pond and calculating the average heat loss for each successive day of residence in the pond. The heat transfer correlations used in this model are those recommended by Ryan [5] . The basic equation of the net energy flux from a water surface exposed to the environment is

$$\phi_n = \phi_r - \left[4.0\text{x}10^{-8}(T_s+460) + FW[(es-ea) + 0.25(T_s-T_a)]\right] \tag{2.20}$$

where $FW = 17{*}W$ for an unheated water surface,

$FW = 22.4(\Delta\theta)^{1/3} + 14{*}W$,

$\Delta\theta = T_{sv} - T_{av}$ (°F),

W = wind speed at 2 meters (MPH),

T_{sv} = virtual temperature of a thin vapor layer in contact with the water surface,

$= (T_s + 460)/(1 - .378\ es/P)$,

T_{av} = virtual air temperature,

$= (T_a + 460)/(1 - .378\ ea/P)$,

es = saturated vapor pressure at T_s (mm Hg),

ea = saturated vapor pressure at T_a (mm Hg),

P = atmospheric pressure (mm Hg),

T_s = bulk water surface temperature (°F),

T_a = air dry bulb temperature (°F),

ϕ_n = net heat from pond surface (BTU/day-ft^2)

$\phi_r = \phi_{sn} + \phi_{an}$ = net absorbed radiative energy,

ϕ_{sn} = net absorbed solar radiation,

$= .94(\phi_{sC})(1 - 0.64C^2)$

ϕ_{sC} = incident solar radiation,

C = fraction of sky covered by clouds,

ϕ_{an} = net absorbed longwave radiation, and

$= 1.16\text{x}10^{-13}(460 + T_a)^6(1 + 0.17C^2)$.

The computational algorithm for the plug-flow model is given in Fig. 2.7. Note that the model is not a perfect plug-flow model in that each plug of water entering the pond is assumed to be mixed with the slug immediately preceeding it. This mixing qualitatively accounts for the effect of entrance mixing.

2.6 Dry Cooling Towers

In relation to the other waste heat dissipation systems, the development of a reliable performance model of dry cooling towers is simple. The amount of heat rejected by a mechanical draft dry tower can be shown to be directly proportional to

Fig. 2.7

Cooling Pond Model Computational Algorithm

Assign to each discharge slug an identification number = day of discharge

initialize slugs in pond at time zero; temperature, volume, fraction of volume in pond

begin calculation for day J

calculate temperature change for all slugs in pond

add discharge volume of day J to pond volume, set fraction of volume of slug J in pond equal to 1

mix slugs J and J-1

determine which slugs (and fractions thereof) remain in pond by summing up slug volumes for day J, J-1, J-2, ...; until pond volume is equaled or exceeded

determine which slugs (and fractions thereof) exhausted from pond during day J by comparing new pond inventory (day J) and old pond inventory (day J-1)

mix all exhausted slugs to find withdrawal temperature for day J

end day J calculation

the difference between the inlet water temperature and inlet air temperature for a fixed dry tower design. With reference to Fig. 2.8

$$Q = UA\Delta T_{lm}F_g \tag{2.21}$$

where Q = heat rejection rate,

A = heat transfer surface area,

U = effective heat transfer coefficient,

F_g = cross-flow correction factor, and

ΔT_{lm} = log mean temperature difference.

$$\Delta T_{lm} = \frac{(T_o-T_i') - (T_i-T_o')}{\ln\frac{(T_o-T_i')}{(T_i-T_o')}} \tag{2.22}$$

where $(T_o-T_i') > (T_i-T_o')$

T_i = water inlet temperature,

T_o = water outlet temperature,

T_i' = air inlet temperature, and

T_o' = air outlet temperature.

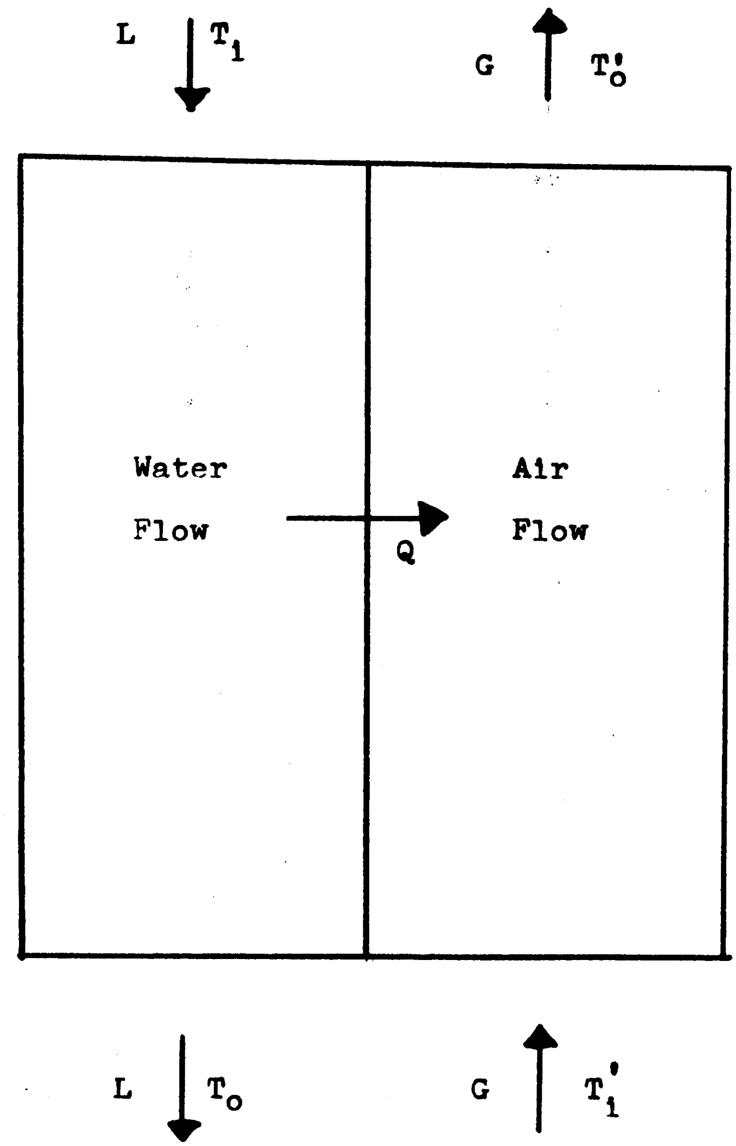
A heat balance on the tower gives

$$LC_w(T_i-T_o) = GC_a(T_o'-T_i') \tag{2.23}$$

Equations (2.21), (2.22) and (2.23) may be combined to yield

Fig. 2.8

Dry Tower Schematic Drawing



$$Q = \frac{ITD(e^x - 1)}{\frac{e^x}{GC_a} - \frac{1}{LC_w}} \tag{2.24}$$

where $\quad ITD = T_i - T_i'$,

and

$$x = F_g UA \left[\frac{1}{GC_a} - \frac{1}{LC_w} \right] .$$

Now note that, for fixed values of the parameters U, A, F_g, G and L,

$$Q \ \alpha \ ITD \tag{2.26}$$

This result has been found by Rossie [1] to be experimentally verified. Further, Rossie has found that the thermal performance of natural draft dry cooling towers may be reasonably expressed by a relationship of the form

$$Q \ \alpha \ ITD^b \tag{2.27}$$

where b is a constant for a given tower. A typical value of b is 1.33.

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APPENDIX A

COMPUTER PROGRAMS FOR WASTE HEAT REJECTION SYSTEM THERMAL PERFORMANCE CALCULATIONS

This Appendix contains FORTRAN computer programs for predicting the thermal performance of a mechanical-drfat evaporative cooling tower (MECDRAFT), a natural-draft evaporative cooling tower (NATDRAFT), and a spray canal (SPRANAL). Programming of the cooling pond and dry cooling tower models discussed in this report may be easily done by the user. Tables A.1 to A.3 list the required input variables for the three programs. The FORMAT of the required input may be easily obtained by examining the appropriate program listing.

Table A.1

Required Data Input for Natural Draft Evaporative Cooling Tower Model (NATDRAFT)

Definitions of Input Variables in Order of Occurrence

WTRF = Water flow rate (lbm/ft^2-hr)

HUM = Relative humidity of ambient air (%/100)

AIRTI = Inlet air temperature (°F)

WTRTI = Inlet water temperature (°F)

WTRTOA = Initial geuss at water outlet temperature (°F)

Other Variables Defined Internally in the Porgram Which May Be Adjusted

AIRF = Initial geuss at air flow rate (lbm/hr-ft^2)

PPP = Type of Tower Packing

HPACK = Height of Tower Packing (ft)

HAIRIN = Height of Tower Packing Above Ground (ft)

ATOTAL = Total Packing Surface Area per unit flow area

ADPK = Surface Area per unit flow area for computing pressure loss in packing due to skin friction (ft^2)

AFPK = Fraction of Tower cross-section which is unobstructed by packing

HTOWER = Height of Tower Chimney (ft)

DTOWER = Diameter of Tower at Base (ft)

Table A.2

Required Data Input for Spray Canal Model
(SPRANAL)

All Required Data to be Defined Internally in Program

R = Fraction of total water flow sprayed by each spray device

TEMDIS = Canal inlet water temperature (°F)

TWB = Ambient wet-bulb temperature (°F)

WSPEED = Ambient wind speed (MPH)

PASSES = Number of spray passes marching down canal

NROW = Number of rows of spray devices across canal

Table A.3

Required Data Input for Mechanical-Draft Evaporative Cooling Tower Model (MECDRAFT)

Program presented as a complete SUBROUTINE TOWER(J) which determines the tower output temperature CTWOUT on Day J with an ambient wet bulb temperature TWBXX(J). All data input through COMMON statements.

TTOWIN = tower inlet water temperature (°F)

PSA = vapor pressure of water at 1 °F increments from 0 to 150 °F (psia)

WATERL = water flow per square foot of tower cross-section (lbm/hr-ft^2)

WL = total water flow to tower (lbm/hr)

Other variables defined internally in program which may be adjusted.

ACELLW = calculational cell water loading area (ft^2)

ACELLA = calculational cell air loading area (ft^2)

N = square root of number of calculational cells

HEIGHT = height of tower fill (ft)

```
C   PROGRAM FOR PREDICTING THE THERMAL PERFORMANCE OF A NATURAL DRAFT
C      COOLING TOWER — NATDRAFT —
C   ADIN=NORMALIZED CROSS-SECTIONAL DRAG AREA AT AIR INLET
C   ADOT=NORMALIZED CROSS-SECTIONAL DRAG AREA AT AIR OUTLET
C   ADPK=SURFACE AREA PER UNIT FLOW AREA FOR COMPUTING PRESSSUE LOSS IN
C      PACKING DUE TO SKIN FRICTION LOSS
C   ADSL=NORMALIZED CROSS-SECTIONAL AREA FOR DRAG IN SHELL
C   AFIN=NORMALIZED CROSS-SECTIONAL FLOW THROUGH AREA AT THE AIR INLET
C   AFPK=PORTION OF TOWER-SECTION WHICH IS UNOBSTRUCTED BY PACKING
C   AFOT=NORMALIZED CROSS-SECTIONAL FLOW AREA THROUGH  OUTLET OF PACKING
C   AFSL=NORMALIZED CROSS-SECTIONAL FLOW THROUGH AREA IN THE SHELL
C   AIRF=INITIAL GEUSS FOR THE AIR FLOW RATE
C   AIRTI=INLET AIR DRY BULB TEMPERATURE
C   ATMOS=ATMOSPHERIC PRESSURE
C   ATOTAL=TOTAL PACKING SURFACE AREA IN ONE SQUARE FOOT OF TOWER X-SECTION
C   CDIN=DRAG COEFFICEINT FOR INLET STRUCTURE
C   CDOT=DRAG COEFFICIENT FOR OUTLET STRUCTURE
C   CDSL=DRAG COEFFICIENT FOR THE SHELL
C   CP=SPECIFIC HEAT OF AIR
C   DTOWER=TOWER DIAMETER AT PACKING
C   HAIRIN=HEIGHT OF PACKING AIR INLET
C   HPACK=HEIGHT OF THE PACKING
C   HTOWER=TOWER HEIGHT
C   HUM=RELATIVE HUMIDITY OF INLET AIR
C   LAMBDA=EMPIRICAL COEFFICIENT FOR SPLASH PACKING
C   N=EMPIRICAL COEFFICIENT FOR SPLASH PACKING
C   P1,,P16,P23,P26=EMPIRICAL PRESSURE DROP DATA---LOWE AND CHRISTIE
C   SPACE=CENTER TO CENTER SPACING OF PARALLEL PLATES
C   THICK=THICKNESS OF PARALLEL PLATE PACKING
C   TOLERH=CONVERGENCE TOLERANCE FOR TOWER HEIGHT
C   TOLERT=CONVERGENCE TOLERANCE FOR INLET WATER TEMP
C   WTRF=NORMALIZED WATER FLOW RATE
C   WTRTI=INLET WATER TEMPERATURE
C   WTRFT=TOTAL WATER FLOW RATE
C   WTRTO=INITIAL GEUSS FOR OUTLET WATER TEMP
C   A=INTEGRATION SEGMENT AREA
```

```
C    AIRFL=CURRENT VALUE OF AIR FLOW RATE
C    AIRT=CURRENT AIR TEMPERATURE
C    C=TEMPORARY VARIALBE
C    CF=FRICTION COEFFICIENT
C    CONWTR=WEIGTH OF CONDENSED WATER
C    DA=AREA SEGMENT
C    DAIRT=CHANGE IN AIR TEMPERATURE DURING ONE INTEGRATION STEP
C    DNSARI=DENSITY OF INLET AIR
C    DNSAVG=AVERAGE AIR DENSITY
C    DNSARO= DENSITY OF OUTLET AIR
C    DTODTI=RATE OF OULET WATER TEMP CHANGE VERSUS INLET WATER TEMP CHANGE
C    DWTRT=CHANGE IN WATER TEMPERATURE DURING ONE INTEGRATION STEP
C    ENT=AIR ENTHALPY AS INTEGRATION PROCEEDS
C    ENTI=ENTHALPY OF INLET AIR
C    ENTSA=ENTHALPY OF AIR DURING THE SATURATION ADJUSTMENT LOOP
C    ENTSAT=ENTHALPY OF A POUND OF SATURATED AIR-WATER MIXTURE
C    H=CALCULATED TOWER HEIGHT
C    H1,H2=HOLDING VALUES OF TOWER HEIGHT
C    HENT=ADJUSTED ENTHALPY OF AIR-WATER DROPLET MIXTURE IN SATURATION
C       ADJUSTMENT LOOP
C    HG=HEAT TRANSFER COEFFICIENT
C    HUMI=RELATIVE HUMIDITY AS INTEGRATION PROCEEDS
C    LBW=POUNDS OF WATER PER POUND OF AIR AT ANY POINT IN PACKING
C    LBVI=POUNDS OF VAPOR PER POUND OF AIR
C    LBVLBA=POUNDS OF VAPOR PER POUND OF AIR AT ANY POINT IN THE PACKING
C    NOITER=NUMBER OF ITERATIONS COMPLETED
C    PRLIN=PRESSURE LOSS AT THE INLET
C    PRLPK=PRESSURE LOSS IN PACKING
C    PRLPR=PRESSURE LOSS DUE TO PROFILE
C    PRLOT=PRESSURE LOSS AT OUTLET
C    PRLSL=PRESSURE LOSS IN SHELL
C    PRLSP=PRESSURE LOSS DUE TO SPRAY
C    PSA=SATURATION VAPOR PRESSURE AT THE AIR TEMPERATURE
C    PSAH=SATURATION VAPOR IN SATURATION ADJUSTMENT LOOP
C    PSAT()=SATURATION VAPOR PRESSURE
C    PSW=SATURATION VAPOR PRESSURE AT THE WATER TEMPERATURE
```

```
C   VHSP=VELOCITY HEADS LOST DUE TO SPRAY
C   VHVC=VELOCITY HEADS LOST DUE TO VENA-CONTRACTA IN THE TOWER
C   VIN=AIR INLET VELOCITY
C   VNOM=NOMINAL VELOCITY IN PACKING
C   VPEN ENTHALPY OF MOISTURE IN AIR, USED IN SATURATION ADJUSTMENT LOOP
C   VPENT=ENTHALPY OF VAPOR IN AIR
C   VPRES=VAPOR PRESSURE OF AIR
C   VPK=AIR VELOCITY IN PACKING
C   VOT=AIR VELOCITY AT OUTLET
C   VSL=AIR VELOCITY IN THE SHELL
C   WTRLT=WATER WHICH CONDENSES OUT DURING AN INTEGRATION STEP
C   WTRT1,WTRT2=HOLDS WATER INLET TEMPERATURE FOR EXTRAPOLATION
C      LOGICAL VARIABLES
C   ENDFLG= TRUE IF PROGRAM HAS REACHED NORMAL TERMINATION
C   EXTAFL=TRUE IF ITERATION IS BEING MADE TO EXTRAPOLATE AIRFLOW
C   EXTWTO=TRUE IF ITERATION IS BEING MADE TO EXTRAPOLATE OUTLET WATER TEMP
C   PPP=TRUE IF TOWER HAS PARALLEL PLATE PACKING
C   PRIN=TRUE IF SPLASH PACKING
C   PRITER=TRUE IF RESULTS OF EACH ITERATION ARE TO BE PRINTED
C   PRSTEP=TRUE IF EACH STEP IN ITERATION IS TO BE PRINTED
        LOGICAL ENDFLG, PRITER, PRSTEP, EXTWTO, EXTAFL
        LOGICAL PPP
        REAL LBVLBA,LBW,LAMBDA,N,LBVLBS,LBVI,KAL
1111    CONTINUE
        READ(5,999)WTRF,HUM,AIRTI,WTRTI,WTRTOA
999     FORMAT(5F10.3)
        FACTOR=3.2
1001    CONTINUE
        WTRTO=WTRTOA
        AIRF=1264.0
C*************************************************************************
C       IMPORTANT###  SET PPP = TRUE IF PARRALLEL PLATE PACKING IS USED
C     IF PPPP IS TRUE  RISH'S HT TRANSFER AND PRESSURE DROP REALATIONS
C          ARE USED
C*************************************************************************
        PPP=.TRUE.
```

```
      HPACK=5.33
      HAIRIN=35.6
C*******************************************************************************
C     SKIP THE NEXT INPUT PARAMETERS IF PARRALLEL PLATE PACKING IS USED
      IF(PPP) GO TO 2
      LAMBDA=0.065
      N=.6
      P13=1.2
      P16=0.9
      P23=2.0
      P26=1.3
      ATOTAL=HPACK
C*******************************************************************************
2     CONTINUE
C*******************************************************************************
C     IF PPP IS TRUE INPUT ATCWER AND AFPK AS REQUIRED
C*******************************************************************************
      IF(.NOT.PPP) GO TO 7
      ATOTAL=204.6
      ADPK=204.0
      AFPK=.70
7     CONTINUE
      CP=0.24
      ATMOS=14.4
      AFIN=1.0
      AFOT=1.0
      AFSL=1.0
      ADIN=0.0
      ADOT=0.0
      ADSL=0.0
      CDOT=0.0
      CDSL=0.0
      CDIN=0.0
      TOLERT=0.3
      TOLERH=10.0
      HTOWER=514.0
```

```
      DTOWER=372.0
      STEPS=20.0
      LSTEP=50
      ENDFLG=.FALSE.
      PRITER=.TRUE.
      PRSTEP=.FALSE.
      EXTWTC=.FALSE.
      EXTAFL=.FALSE.
      LITER=52
      AIRT=AIRTI
      NOITER=0
      VHVC=0.167*(DTOWER/HAIRIN)**2
      VPRES=HUM*PSAT(AIRT)
      LBVLBA=0.622*VPRES/(ATMOS-VPRES)
      VPENT=1061.0+.444*AIRT
      ENTI=CP*(AIRT-32.0)+VPENT*LBVLBA
      VPRESI=VPRES
      LBVI=LBVLBA
      DNSARI=((ATMOS-VPRES)/53.3+VPRES/85.7)*144.0/(460.0+AIRT)
      DA=ATOTAL/STEPS
      AIRFL=0.0
C*****************************************************************************
C
C     END INPUT AND INITIALIZATION
C     START ITERATION
C
C*****************************************************************************
95    VNOM=AIRF/(DNSARI*3600.0)
      VHSP=0.16*HAIRIN*(WTRF/AIRF)**1.32
      IF(PPP) GO TO 16
      KAL=HPACK*LAMBDA*(AIRF/WTRF)**N
      HG=CP*WTRF*KAL/HPACK
      HGOUT=0.0
      T1=VNOM/3.0-1.0
      P1=(P16-P13)*T1+P13
      P2=(P26-P23)*T1+P23
```

```
      VHLPK=((P2-P1)*(WTRF-1000.0)/1000.0+P1)*HPACK
      CF=0.0
      GO TO 15
16    CF=0.0192*(WTRF/AIRF)**0.5
      CF=FACTOR*CF
      HG=CP*AIRF*CF/(2.0+CF*71.6*(AIRF/WTRF)**0.25)
      KAL=HG*ATOTAL/(CP*WTRF)
      HGOUT=HG
15    WTRT=WTRTO
      ENT=ENTI
      HUMI=HUM
      A=0.0
      LBVLBA=LBVI
      VPRES=VPRESI
      CONWTR=0.0
      AIRT=AIRTI
C*****************************************************************************
C     INTEGRATION LOOP BEGINS WITH STATEMENT 6
C*****************************************************************************
6     PSW=PSAT(WTRT)
      IF(PSW.EQ.0.0) GO TO 110
      ENTSAT=CP*(WTRT-32.0)+(1061.0+.444*WTRT)*0.622*PSW/(ATMOS-PSW)
      C=HG*DA*(ENTSAT-ENT)/CP
      IF(.NOT.PRSTEP.OR.EXTWTO.OR.EXTAFL) GO TO 35
      IF(LSTEP.LT.47) GO TO 36
      WRITE(6,37)
37    FORMAT(52H1COOLING TOWER PROGRAM - STEP BY STEP RESULTS OF ONE,
     1 10H ITERATION/57H0        WATER    AIR  SATUR  ACTUAL REL PNDS W
     2TR/  VAPOR/56H   AREA   TEMP   TEMP ENTHAL  ENTHAL HUM  PNDS AIR
     3PRES)
      LSTEP=0
      LITER=52
36    LSTEP=LSTEP+1
      WRITE(6,38) A,WTRT,AIRT,ENTSAT,ENT,HUMI,LBVLBA,VPRES
38    FORMAT(5F7.1,F6.3,F9.5,F7.4)
35    DWTRT=C/WTRF
```

```
      DENT=C/AIRF
      DAIRT=HG*DA*(WTRT-AIRT)/(AIRF*CP)
      WTRT=WTRT+DWTRT
      ENT=ENT+DENT
      AIRT=AIRT+DAIRT
      A=A+DA
      VPENT=1061.0+0.444*AIRT
      LBVLBA=(ENT-CP*(AIRT-32.0))/VPENT
      PSA=PSAT(AIRT)
      IF(PSA.EQ.0.0) GO TO 110
      LBVLBS=0.622*PSA/(ATMOS-PSA)
      HUMI=LBVLBA*(0.622+LBVLBS)/(LBVLBS*(.622+LBVLBA))
      VPRES=HUMI*PSA
      IF(HUMI.LE.1.0) GO TO 99
C**********************************************************************
C     IF MIXTURE IS SUPER-SATURATED, FIX-UP
C**********************************************************************
      T=AIRT
97    T=T+0.1
      PSAH=PSAT(T)
      IF(PSAH.EQ.0.0) GO TO 110
      VPEN=1061.0+.444*T
      LBW=0.622*PSAH/(ATMOS-PSAH)
      ENTSA=CP*(T-32.0)+VPEN*LBW
      HENT=(LBVLBA-LBW+CONWTR)*(T-32.0)+ENTSA
      IF(ENT.GT.HENT) GO TO 97
      CONWTR=LBVLBA-LBW+CONWTR
      ENT=ENTSA
      AIRT=T
99    IF(A.LT.ATOTAL) GO TO 6
C**********************************************************************
C     END INTEGRATION SECTION
C
C     COMPUTE PRESSURE LOSSES FOR THIS ITERATION
C**********************************************************************
100   IF(EXTWTO) GO TO 24
```

```
      VPENT=1061.0+0.444*AIRT
      LBVLBA=(ENT-CP*(AIRT-32.0))/VPENT
      WTRLT=AIRF*(LBVLBA+CONWTR-LBVI)
      VPRES=LBVLBA*ATMOS/(0.622+LBVLBA)
      DNSARO=((ATMOS-VPRES)/53.3+VPRES/85.7)*144.0/(460.0+AIRT)
      DNSARO=DNSARO*(1.0+CONWTR)/(1.0+CONWTR*DNSARO/62.4)
      DNSAVG=(DNSARI+DNSARO)/2.0
      VIN=VNOM/AFIN
      VOT=AIRF/(DNSARO*AFOT*3600.0)
      VSL=AIRF/(DNSARO*AFSL*3600.0)
      PRLIN=CDIN*DNSARI*0.016126*ADIN*VIN**2
      IF(.NOT.PPP) GO TO 102
      VPK=AIRF/(DNSAVG*AFPK*3600.0)
      PRLPK=CP*DNSAVG*0.016126*ADPK*VPK**2
      GO TO 103
102   PRLPK=DNSARI*0.016126*VHLPK*VNOM**2
      VPK=VNOM
103   PRLOT=CDOT*DNSARO*0.016126*ADOT*VOT**2
      PRLSL=CDSL*DNSARO*0.016126*ADSL*VSL**2
      PRLVC=VHVC*DNSARI*0.016126*VNOM**2
      PRLSP=VHSP*DNSARI*0.016126*VNOM*VNOM
      PRLPR=PRLOT+PRLIN+PRLSL
      H=(PRLPR+PRLPK+PRLSP+PRLVC)/(DNSARI-DNSARO)
      IF(ENDFLG) GO TO 40
      NOITER=NOITER+1
      IF(.NOT.PRITER.OR.EXTAFL) GO TO 21
40    IF(LITER.LT.52) GO TO 30
      LSTEP=50
      LITER=0
      WRITE(6,31)
31    FORMAT(46H1COOLING TOWER PROGRAM - RESULTS OF ITERATIONS/
     1 22X,17HAIR  CALC   TOWER/
     263H             OUTLET  VELOTY HEAT  CHARAC-   SKIN          INLET
     3, 50H OUTLET OUTLET PROFILE  PACKING    SPRAY  VENA CON/
     463H ITER  WATER   AIR     IN   TRANS TERISTIC FRICTION RELAT WATER
     5, 56H  AIR    AIR   PRESSURE PRESSURE PRESSURE PRESSURE TOWER/
```

```
     663H  NO   LOSS  DENSITY PAKING COEFF (K*A/L)   COEFF   HUMID TEMP
     7,  57H  TEMP  ENTHAL   LOSS     LOSS     LOSS     LOSS   HEIGHT)
30      WRITE(6,32)NOITER,WTRLT,DNSARO,VPK,HGOUT,KAL,CF,HUMI,WTRT,AIRT,
     1   ENT,PRLPR,PRLPK,PRLSP,PRLVC,H
32      FORMAT(1H0,I4,F7.2,F8.6,F7.3,F6.3,F8.4,F9.5,F7.3,F6.1,
     1  F6.1,F7.1,F10.6,3F9.6,F7.0)
        LITER=LITER+2
        IF(ENDFLG) GO TO 33
C***********************************************************************************
C     END PRINTING RESULTS OF ONE ITERATION
C***********************************************************************************
21      IF(NOITER.LE.100) GO TO 39
        WRITE(6,98)
98      FORMAT(47H MORE THAN 100 ITERATIONS. EXECUTION TERMINATED)
        GO TO 39
C  ITERATION STOP BYPASSED
        STOP
C***********************************************************************************
C     NOW FIND IF SPECIFIED TOLERANCES ARE MET
C***********************************************************************************
39      IF(ABS(WTRT-WTRTI).LE.TOLERT) GO TO 27
        IF(.NOT.PRITER) GO TO 46
        IF(.NOT.EXTAFL) GO TO 48
        WRITE(6,42)WTRTO
42      FORMAT(30H     (EXTRAPOLATING FROM WTRTO=,F6.1,1H))
        LITER=LITER+1
        GO TO 46
48      WRITE(6,43) WTRTO
        LITER=LITER+2
43      FORMAT(31H      (EXTRAPOLATING FROM WTRTO=,F6.1,1H))
46      WTRT1=WTRT
        WTRTO=WTRTO+0.1
        EXTWTO=.TRUE.
        GO TO 15
27      IF(EXTAFL) GO TO 50
        IF(ABS(H-HTOWER).LE.TOLERH) GO TO 29
```

```
      IF(.NOT.PRITER) GO TO 44
      WRITE(6,41) AIRF
      LITER=LITER+2
41    FORMAT(26H0(EXTRAPOLATING FROM AIRF=,F7.1,1H))
44    AIRFL=AIRF
      H1=H
      AIRF=AIRF+10.0
      EXTAFL=.TRUE.
      GO TO 95
C*****************************************************************************
C     A SAMPLE ITERATION HAS BEEN MADE TO ADJUST AIRF OR WTRTO
C*****************************************************************************
50    H2=H
      DAFDH=10.0/(H2-H1)
      EXTAFL=.FALSE.
      AIRF=AIRF+DAFDH*(HTOWER-H)
      IF(.NOT.PRITER) GO TO 95
      WRITE(6,55) AIRF
      LITER=LITER+1
55    FORMAT(20H (MODIFYING AIRF TO ,F7.1,1H))
      GO TO 95
24    WTRT2=WTRT
      DTODTI=0.1/(WTRT2-WTRT1)
      EXTWTO=.FALSE.
      WTRTO1=WTRTO
      WTRTO=WTRTO+DTODTI*(WTRTI-WTRT)
      IF(.NOT.PRITER) GO TO 15
      IF(.NOT.EXTAFL) GO TO 62
      WRITE(6,61) WTRTO
61    FORMAT(25H     (MODIFYING WTRTO TO ,F6.1,1H))
      LITER=LITER+1
      GO TO 15
62    WRITE(6,60) WTRTO
      LITER=LITER+2
60    FORMAT(21H (MODIFYING WTRTO TO ,F6.1,1H))
      GO TO 15
```

```
29      IF(PRITER) GO TO 33
        ENDFLG=.TRUE.
        LITER=52
        GO TO 100
33      WRITE(6,96)WTRTO, H
96      FORMAT(26H END COOLING TOWER PROGRAM/34H0FINAL OUTLET WATER TEMP
       1ERATURE IS,F6.1/22H0FINAL TOWER HEIGHT IS,F7.0)
        WRITE(6,1002) FACTOR
1002    FORMAT(8H FACTOR=,F10.4)
        RANGE=WTRTI-WTRTO
        WRITE(6,998)
998     FORMAT(102H WATER LOADING       HUMIDITY    INLET AIR TEMP   INLE
       1T WATER TEMP     ACTUAL OUTLET WATER TEMP   RANGE)
        WRITE(6,997)WTRF,HUM,AIRTI,WTRTI,WTRTOA,RANGE
997     FORMAT(2X,F10.3,6X,F10.3,6X,F10.3,8X,F10.3,13X,F10.3,10X,F10.3)
        GO TO 1111
        STOP
110     AIRF=(AIRF-AIRFL)/2.0+AIRFL
        IF(.NOT.PRITER) GO TO 95
        WRITE(6,111) AIRF
        LITER=LITER+2
111     FORMAT(19H0(ADJUSTING AIRF TO,F7.1,15H FOR STABILITY))
        GO TO 95
        END
        FUNCTION PSAT(T)
        DIMENSION V(181)
        DATA M/0/
        DATAV/.08854,.09223,.09603,.09995,.10401,.10821,.11256,.11705,.121
       170,.12652,.13150,.13665,.14199,.14752,.15323,.15914,.16525,.17157,
       2.17811,.18486,.19182,.19900,.20642,.2141,.2220,.2302,.2386,.2473,.
       32563,.2655,.2751,.2850,.2951,.3056,.3164,.3276,.3390,.3509,.3631,.
       43756,.3886,.4019,.4156,.4298,.4443,.4593,.4747,.4906,.5069,.5237,.
       55410,.5588,.5771,.5959,.6152,.6351,.6556,.6766,.6982,.7204,.7432,.
       67666,.7906,.8153,.8407,.8668,.8935,.9210,.9492,.9781,1.0078,1.0382
       7,1.0695,1.1016,1.1345,1.1683,1.2029,1.2384,1.2748,1.3121,1.3504,1.
       83896,1.4298,1.4709,1.5130,1.5563,1.6006,1.6459,1.6924,1.7400,1.788
```

```
     98,1.8387,1.8897,1.9420,1.9955,2.0503,2.1064,2.1638,2.2225,2.2826,2
     *.3440,2.4069,2.4712,2.5370,2.6042,2.6729,2.7432,2.8151,2.8886,2.96
     *37,3.0404,3.1188,3.1990,3.281,3.365,3.450,3.537,3.627,3.718,3.811,
     *3.906,4.003,4.102,4.203,4.306,4.411,4.519,4.629,4.741,4.855,4.971,
     *5.090,5.212,5.335,5.461,5.590,5.721,5.855,5.992,6.131,6.273,6.471,
     *6.565,6.715,6.868,7.024,7.183,7.345,7.510,7.678,7.850,8.024,8.202,
     *8.383,8.567,8.755,8.946,9.141,9.339,9.541,9.746,9.955,10.168,10.38
     *5,10.605,10.830,11.058,11.290,11.526,11.769,12.011,12.262,12.512,1
     *.771,13.031,13.300,13.568,13.845,14.123,14.410,14.696/
      NT=T
      PSAT=0.0
      IF(NT.GT.31) GO TO 5
      PSAT=V(1)
      WRITE(6,2) T
    2 FORMAT(36H0ERROR IN PSAT:  TABLE EXCEEDED.  T=,F8.2)
    4 M=M+1
      IF(M.LE.50) RETURN
      WRITE(6,3)
    3 FORMAT(53H0 MORE THAN 50 ERRORS IN PSAT -- EXECUTION TERMINATED)
      STOP
    5 IF(NT.GE.212) GO TO 4
    1 PSAT=V(NT-31)+(V(NT-30)-V(NT-31))*(T-NT)
      RETURN
      END
C     PROGRAM FOR PREDICTING THE THERMAL PERFORMANCE OF A SPRAY CANAL   — SPRANAL —
C     E=TOTAL WATER EVAPORATED
C     ALPHA=FRACTION OF WATER EVAPORATED
C     F=AIR INTERFERENCE FACTORS
C     PSA=SATURATION VAPOR PRESSURE
C     TMIX=MIXED CANAL TEMPERATURE
C     TST=SPRAY TEMPERATURE
C     TWBL=LOCAL WET BULB TEMPERATURE
C     PASSES=NUMBER OF PASSES MARCHING DOWN CANAL
C     NROW=NUMBER OF SPRAYS ACROSS CANAL
C     R=FRACTION OF WATER SPRAYED BY EACH SPRAY
C     TEMDIS= CANAL INLET TEMPERATURE
```

```
C    TWB=WET BULB TEMPERATURE
C    WSPEED=WIND SPEED
C    B=BOWEN RATIO
C    I=PASS NUMBER
C    J=ROW NUMBER
C    NTU=NUMBER OF TRANSFER  UNITS PER SPRAY
C    HTN=ENTHALPY AT TEMPERAURE TN
C    IRATE=NUMBER OF ITERATIONS
C    TN=TEMPERATURE OF SPRAY AT NOZZLE
C    TCC=CANAL DISCHARGE TEMPERATURE
         DIMENSION E(200),ALPHA(10,200),F(10)
         DIMENSION PSA(150)
         DIMENSION TMIX(200), TST(10,200),TWBL(10,200),T(10,200)
         INTEGER PASSES
         REAL NMPR,NTU
         READ(5,5) PSA
5        FORMAT(8F10.2)
C    F VALUES AS REPORTED BY PORTER
         F(1)=0.0
         F(2)=0.18
         F(3)=0.44
         F(4)=0.70
         F(5)=0.96
         R=*****INPUT VALUE****
         TEMDIS=*****INPUT VALUE*****
         TWB=*****INPUT VALUE*****
         WSPEED=*****INPUT VALUE*****
         PASSES=*****INPUT VALUE*****
         NROW=*****INPUT VALUE*****
         TMIX(1)=TEMDIS
         B=0.0
         PATM=14.7
         RNROW=NROW
         I=0
C    BEGIN CALCULATION FOR EACH PASS
8        J=0
```

```
         I=I+1
C   BEGING PASS CALCULATION WITH UPWIND NODULE
10       J=J+1
         IF(I.EQ.1)  TWBL(J,I)=F(J)*(TMIX(I)-TWB)+TWB
         IF(I.GT.1)  TWBL(J,I)=F(J)*(TMIX(I-1)-TWB)+TWB
         NTU=0.16+(0.053)*WSPEED
         IF(I.EQ.1)  TN=TMIX(I)
         IF(I.GT.1)  TN=TMIX(I-1)
         TS=TN-30.0
         ITWBL=TWBL(J,I)
         ITWBL1=ITWBL+1
         PSATB1=PSA(ITWBL)
         PSATB2=PSA(ITWBL1)
         ITN=TN
         ITN1=ITN+1
         PSATN1=PSA(ITN)
         PSATN2=PSA(ITN1)
         TITN=ITN
         TITWBL=ITWBL
         PSATN=PSATN1+(TN-TITN)*(PSATN2-PSATN1)
         PSATWB=PSATB1+(TWBL(J,I)-TITWBL)*(PSATB2-PSATB1)
         HTN=0.935*(0.24*TN+(0.622*PSATN/(14.7-PSATN))*(1061.8+0.44*TN))
         HTWBL=0.935*(0.24*TWBL(J,I)+(0.622*PSATWB/(14.7-PSATWB))
       1 *(1061.8+0.44*TWBL(J,I)))
         ITRATE=0
C   BEGIN ITERATION FOR HEAT TRANSFER CCALCULATION
30       TSAVE=TS
         ITRATE=ITRATE+1
         ITS=TS
         ITS1=ITS+1
         PSATS1=PSA(ITS)
         PSATS2=PSA(ITS1)
         TITS=ITS
         PSATS=PSATS1+(TS-TITS)*(PSATS2-PSATS1)
         HTS=0.935*(0.24*TS+(0.622*PSATS/(14.7-PSATS))*(1061.8+0.44*TS))
         TS=TN-NTU*((HTS+HTN)/2.0-HTWBL)
```

```
      DIFTEM=ABS(TSAVE-TS)
      IF(DIFTEM.LE.0.01) GO TO 50
      TS=(TSAVE+TS)/2.0
      GO TO 30
50    TST(J,I)=TS
      ALPHA(J,I)=(TW-TS)/(1061.8*(1.0+B))
      WRITE(6,45) J,I,TS
45    FORMAT(28H THE SPRAY COLD TEMP AT ROW ,I3,10H AND PASS ,
     1 I3,4H IS ,F15.3)
      IF(J.EQ.NROW) GO TO 60
      GO TO 10
C   SUMMUP EVAPORATIVE LOSSES FOR EACH PASS
60    CONTINUE
      TALPHA=0.0
      DO 70  L=1,NROW
      TALPHA=TALPHA+ALPHA(L,I)
70    CONTINUE
      IET=I-1
      IF(IET.EQ.0) E(1)=0.0
      IF(IET.EQ.0) IET=1
      E(I)=R*TALPHA+E(IET)
      ETOTAL=E(I)
      IMINS=I-1
      IF(IMINS.EQ.0) IMINS=1
      TMINUS=TMIX(IMINS)
      WRITE(6,72) TMINUS
72    FORMAT(34H THE PRESENT MIXED TEMPERATURE IS ,F15.3)
      TMIX(I)=0.0
      DO 100  L=1,NROW
      T(L,I)=((1.0-ETOTAL-RNROW*R)*TMINUS+RNROW*R*(1.0-ALPHA(L,I))*
     1 TST(L,I))/(1.0-ETOTAL-RNROW*R*ALPHA(L,I))
      TMIX(I)=TMIX(I)+T(L,I)/RNROW
100   CONTINUE
      IF(I.EQ.PASSES) GO TO 150
      GO TO 8
150   CONTINUE
```

```
      TCC=TMIX(I)
160   CONTINUE
      WRITE(6,175) TCC
175   FORMAT(36H CANAL DISCHARGE TEMPERATURE EQUALS ,F15.3)
      STOP
      END
C   THIS PROGRAM PREDICTS THE THERMAL PERFORMANCE OF EVAPORATIVE
C      MECHANICAL DRAFT COOLING TOWERS -MECDRAFT-
C   KAL=KA/L
C   JKEEP=RETAINS VALUE OF J
C   TWI= TOWER INLET TEMPERATURE
C   ALPHA=PACKING EMPIRICAL COEFFICIENT
C   BETA=EMPIRICAL PACKING COEFFICIENT
C   AIRG=AIR LOADING  LB/HR-FT**2
C   ACELLW=NORMALIZED CALCULATIONAL CELL WATER LOADING AREA
C   ACELLA=NORMALIZED CALCULATIONAL CELL AIR LOADING AREA
C   RLG  ==RATIO OF WATER FLOW TO AIR FLOW IN EACH CELL
C   NOITT=NUMBER OF ITERATIONS
C   N=SQUARE ROOT OF NUMBER OF CELLS
C   HEIGHT=HEIGHT OF PACKING
C   GNTU=KAV/L
C   FNTU=KAV/AIRG
C   DNTU=NTU PER CELL
C   TWO=OUTLET WATER TEMPERATURE
C   PS=SATURATED VAPOR PRESSURE AT TOWER WATER INLET TEMPERATURE
C   H=ENTHALPY OF MIOST AIR AT WATER TEMPERATURE
C  TS=SATURATED VAPOR PRESSURE AT SPECIFIED METR. CONDITION
C   HA=ENTHALPY OF MOIST AIR AT SPECIFIED METR.CONDITION
C   J=NUMBER OF ROW ACROSS,1=TOP ROW
C   I=NUMBER OF ROW ACROSS, 1=AIR INLET SIDE
C   TW=WATER TEMPERATURE
C   HW=MOIST AIR ENTHALPY
C   KC=ITERATION NUMBER CHECK
C   TW2=TEMPERATURE OF WATER AT THE EXIT OF AN INCREMENT(CELL)
C   TWBAL=TEMPERATURE OF AIR AT TOWER EXIT
C   DH=ENTHALPY DIFFERENCE OF THE AIR BETWEEN THE INLET AND OUTLET OF A CELL
```

```
C   DH1=ENTHALPY DIFFERENCE BETWEEN WATER AND AIR ENTERING A CELL
C   HA2=ENTHALPY OF MOIST AIR AT THE EXIT OF A CELL
C   THIS PROGRAM HAS INCLUDED THE VARIOUS PARAMETER VALUES FOR THE
C      FOLLOWING TOWER
C              FILL WIDTH=36 FEET(INCLUDES BOTH SIDES)
C              FILL HEIGHT=60 FEET
C              UNIT FILL LENTH=32 FEET
C              FAN DIAMETER=28 FEET
        SUBROUTINE TOWER(J)
        DIMENSION TWBXX(380),CTWOUT(380)
        DIMENSION HW(30), TW(30), PSA(150)
        COMMON/TOWTEM/TTOWIN
        COMMON/WET/TWBXX
        COMMON/EFF/CTWOUT
        COMMON/ATMOS/PSA
        COMMON/WATER/WATERL,WI
        REAL KAL
        JKEEP=J
        TWB=TWBXX(J)
        TWI=TTOWIN
53      CONTINUE
        ALPHA=0.065-(TWI-110.0)*(0.000325)
        IF(TWI.LE.90.0) ALPHA=0.0715
        BETA=.6
        AIRG=1692.0
        KAL=ALPHA*(WATERL/AIRG)**(-BETA)
        HEIGHT=60.0
        GNTU=KAL*HEIGHT
        ACELLW=32.0
        ACELLA=53.3
        ROCELA=ACELLW/ACELLA
        RLG=ROCELA*WATERL/AIRG
        NOITT=5
        CONST=7.481/60.0/62.0*10.0**9
        CONST1=0.124683/62.0
        PATM=14.7
```

```
      N=4
      FNTU=GNTU*RLG
      CELLN=N
      DNTU=FNTU/CELLN
      IT=TWI
      PS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TWI-IT)
      H=0.24*TWI+0.622*PS/(PATM-PS)*(1061.8+0.44*TWI)
      IT=TWB
      TS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TWB-IT)
      HA=0.24*TWB+0.622*TS/(PATM-TS)*(1061.8+0.44*TWB)
      DO 100 I=1,N
      TW(I)=TWI
  100 HW(I)=H
      TWBAL=0.0
C   BEGIN ROW-DOWN CALCULATIONS
      DO 104 J=1,N
      H=HA
C   BEGIN ROW-ACROSS CALCULATIONS
      DO 101 I=1,N
      KC=0
      DH1=HW(I)-H
      DH=DH1/1.2*DNTU
C   BEGIN HEAT TRANSFER ITERATION
  102 KC=KC+1
      TW2=TW(I)-DH/RLG
      IT=TW2
      PS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TW2-IT)
      HW2=0.24*TW2+0.622*PS/(PATM-PS)*(1061.8+0.44*TW2)
      DHH=(DH1+HW2-H-DH)/2.0*DNTU
      DHH=(DHH+DH)/2.0
      IF(KC.GE.NOITT) GO TO 106
      DH=DHH
      GO TO 102
C   END HEAT TRANSFER ITERATION
  106 TW(I)=TW(I)-DHH/RLG
      IT=TW(I)
```

```
      PS=PSA(IT)+(PSA(IT+1)-PSA(IT))*(TW(I)-IT)
      HW(I)=0.24*TW(I)+0.622*PS/(PATM-PS)*(1061.8+0.44*TW(I))
  101 H=H+DHH
C     DETERMINE AIR EXHAUST WET BULB TEMPERATURE FOR EACH ROW ACROSS
      TWB2=TWB
   20 ITWB2=TWB2
      PS=PSA(ITWB2)+(PSA(ITWB2+1)-PSA(ITWB2))*(TWB2-ITWB2)
      HA2=0.24*TWB2+0.622*PS/(PATM-PS)*(1061.8+0.44*TWB2)
      IF(HA2.GE.H) GO TO 10
      TWB2=TWB2+5.0
      HA22=HA2
      GO TO 20
   10 TWB2=TWB2-4.0
   40 ITWB2=TWB2
      PS=PSA(ITWB2)+(PSA(ITWB2+1)-PSA(ITWB2))*(TWB2-ITWB2)
      HA2=0.24*TWB2+0.622*PS/(PATM-PS)*(1061.8+0.44*TWB2)
      IF(HA2.GE.H) GO TO 30
      TWB2=TWB2+1.0
      HA22=HA2
      GO TO 40
   30 TWB2=TWB2-(HA2-H)/(HA2-HA22)
  104 TWBAL=TWBAL+TWB2
C     DETERMINE AIR AND WATER OUTLET AVERAGE TEMPERATURES
      TWBAL=TWBAL/CELLN
      TWO=0.0
      DO 103 I=1,N
  103 TWO=TWO+TW(I)
      TWO=TWO/CELLN
      CTWOUT(JKEEP)=TWO
      RETURN
      END
```