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# Computer Simulation of Transport Phenomena in Evaporative Cooling Towers

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*A computer model of the simultaneous heat, mass, and momentum transfer processes occurring throughout an entire cooling tower is described in this paper. The model includes the flexibility to analyze the several configurations, fill arrangements, and flow distributions commonly used by the power industry. The fundamental governing equations are solved using a finite-integral technique to provide a quasi-two-dimensional description of the flow and cooling process within the tower. The model has been successfully compared with field data from cooling towers at three TVA power plants as well as data from other utilities. Each of these towers was significantly different in design, thereby demonstrating the versatility of the model for correctly predicting the cooling performance of mechanical and natural draft towers, as well as crossflow and counterflow orientations, for a range of meteorological and plant operating conditions.*

## Introduction

To comply with environmental regulations, many power plants constructed since the early 1970s use cooling towers in the recirculating loop of the condenser cooling water. The thermal efficiency of these power plants is directly dependent upon the efficiency of the cooling towers. Inefficiency in the cooling process of these towers will result in a continuous loss of power generation. Even the loss of a few megawatts, representing a fraction of a precent of the total plant generation, may amount to millions of dollars per year. This continuous power loss may be insignificant in comparison to load reductions which may be required to achieve an internal temperature limit during extremely hot meteorological conditions. A recent survey of utilities by Boroughs (1983) revealed that most evaporative cooling towers do not deliver the design cooling capacity. Consequently, there is a heightened interest in cooling tower performance throughout the power industry.

Evaporative cooling towers are based upon a very simple principle—energy is removed from the hot water by facilitating contact with relatively cool, dry air. The two key factors in this transer of energy are interfacial area and contact time between the air and water, the product of which is defined as the activity. The numerous configurations of cooling towers represent different approaches to increase the activity with minimum resistance to airflow. In addition, this must be accomplished for the lowest possible capital and operating cost.

Although evaporative cooling towers appear quite diverse in design, they generally can be described with a few classifications. The direction of the airflow relative to the waterflow distinguishes whether the tower is of the crossflow or counterflow type. A fill material is used to increase the contact time and interfacial area. There are two major classifications of fill—splash-bar and film. Splash-bar fill is designed to break the waterflow into droplets as it cascades and is often used in crossflow towers. Film fill breaks the waterflow into thin sheets. Because of its typically vertical orientation, it is used predominantly in counterflow towers. Towers are also classified by the mechanism used to induce airflow. Mechanical draft towers use fans to force airflow, whereas natural draft towers rely upon the buoyancy of the moist air to draw through tall chimneys. Regardless of the classifications or the combinations thereof, the physical phenomena within the tower are similar.

The simultaneous heat, mass, and momentum transfer processes occurring throughout a cooling tower are difficult to analyze. Scale modeling of these processes throughout an entire tower is infeasible because the necessary similitude, which includes two-phase flow, cannot be achieved. However, physical modeling of segments of individual components of the tower, such as a block of fill or a cluster of spray nozzles, has produced empirical heat and mass transfer coefficients for the various components of a tower (e.g., Lowe and Christie, 1961).

Perhaps the first attempt at modeling the processes taking place in an evaporative cooling tower was made by Merkel in 1962. Merkel made several simplifying assumptions which reduce the governing relationships for a counterflow tower to a single separable ordinary differential equation; this Merkel integrated numerically using the four-point Chebyshev method. Lowe and Christie (1961) performed laboratory studies on several types of counterflow fill, employing Merkels' method in the data reduction. The Cooling Tower Institute (CTI) still employs the method developed by Merkel

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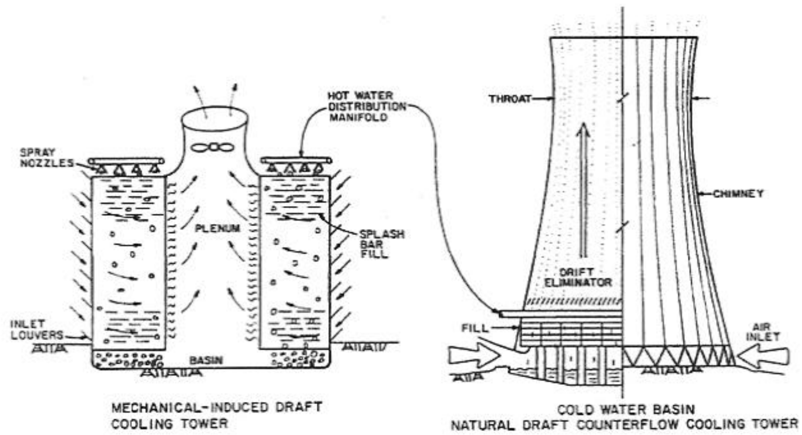


**Fig. 1 Evaporative cooling towers**

(CTI, 1977). Zivi and Brand (1956) extended the analysis of Merkel to crossflow towers. Kelly (1976) used the model of Zivi and Brand along with laboratory data to produce a volume of crossflow cooling tower characteristic curves to be used in graphical solutions and design calculations. Cross et al. (1976) and Bourillot (1983) also developed models with transfer equations similar to Zivi and Brand. None of these models included any variation in airflow throughout the fill. Penney and Spalding (1979) introduced a model for natural draft cross- and counterflow towers that attempted to solve for the air velocity field within the tower using a finite-difference method. This model was extended to mechanical draft towers by Majumdar and Singhal (1981). Some difficulty was experienced with this model in attempting to fit field data.

A computer model of the simultaneous heat, mass, and momentum transfer processing occurring throughout a cooling tower has been developed based on the finite-integral method (Benton, 1984a, 1984b). This model uses heat and mass transfer, and pressure loss coefficients available in the literature (e.g., Lowe and Christie, 1961; Kelly, 1976; CTI, 1977). Consequently, model calibration is necessary only for cases where these coefficients are unavailable.

The model was verified by comparing predicted results with test data from Tennessee Valley Authority (TVA) cooling towers as well as data from other utilities. These test data not only cover a range of operational and ambient meteorological conditions, but also represent a wide spectrum of fundamentally different tower and fill configurations. A mathematical description of the physical phenomena, the numerical scheme for solution of the equations, and the model verification are presented.

## Equations Describing the Physical Processes

In an evaporative cooling tower, the water acts as both the heat transporting medium and the source of evaporative mass. The evaporation process which occurs in a cooling tower when hot water contacts cooler air has two effects: Heat is extracted from the water, and the density of the air is decreased, which makes the cooling tower effluent buoyant relative to the ambient air. Since the latent heat of vaporization of water at atmospheric pressure is three orders of magnitude greater than the specific heat, evaporation of even a small fraction of the water produces significant cooling. The buoyancy of the air can assist the flow of air through the tower in the case of mechanical draft towers which employ fans; or drive the flow of air, as in the case of natural draft towers which rely solely upon a chimney to utilize this buoyant force (see Fig. 1).

The analysis of cooling towers is complicated by the interrelationship of the heat transfer, mass transfer, and two-phase

## Nomenclature

- $a$ = interfacial area per unit volume 1/m
- $A$ = outward normal area vector, $m^2$
- $A_i$ = interfacial area, $m^2$
- $B$ = mass transfer driving potential
- $C_{pa}$ = constant pressure specific heat of air, J/kg/°C
- $C_{pw}$ = constant pressure specific heat of water, J/kg/°C
- $D$ = air/water vapor diffusion coefficient, $m^2/s$
- $E$ = evaporative mass flux, $kg/m^2/s$
- $g$ = acceleration of gravity, $m/s^2$
- $g_c$ = Newton's constant, $Nt\text{-}m/kg/s^2$
- $G$ = dry airflow rate, kg/s
- $h_a$ = enthalpy of moist air (per unit mass dry air), J/kg dry air
- $h_g$ = enthalpy of saturated water vapor, J/kg
- $H$ = sensible heat tranfer coefficient, $W/m^2/°C$
- $K$ = mass transfer coefficient, $kg/m^2/s$
- $L$ = waterflow rate, kg/s
- Le = Lewis number
- $p$ = absolute pressure, $Nt/m^2$
- $Q_e$ = evaporative heat flux, W
- $Q_s$ = sensible heat flux, W
- $Q_t$ = total heat flux, W
- $s$ = distance along a streamline, m
- $T$ = water temperature, °C
- $T_{db}$ = dry-bulb temperature, °C
- $T_{dbo}$ = ambient dry-bulb temperature, °C
- $T_h$ = inlet water temperature, °C
- $T_{wb}$ = wet-bulb temperature, °C
- $T_{wbo}$ = ambient wet-bulb temperature, °C
- $v$ = air velocity, m/s
- $V$ = vector velocity, m/s
- $W_{fan}$ = fan power input, W
- $x$ = horizontal distance, m
- $y$ = vertical distance, m
- $z$ = lateral distance, m
- $\eta_{fan}$ = fan efficiency
- $\kappa$ = thermal conductivity of air, W/m/°C
- $\lambda$ = headloss coefficient (velocity heads per unit length of air travel), 1/m
- $\rho$ = density of moist air, $kg/m^3$
- $\omega$ = absolute humidity
- $\omega_i$ = absolute humidity at interface

flow, the two phases being liquid water and moist air. As in most analysis of transport phenomena, the processes are described by the conservation of mass, momentum, and energy. Because the evaporative process changes the phase of a portion of the water passing through the tower, the conservation of mass of the dry air and water vapor, and the conservation of energy of the gas and water phases are considered individually. These equatons are applied in their steady-state, steady-flow form.

The independent variables for the analysis are: $x$, $y$, $z$, $L$, $T_h$, $T_{wbo}$, and $T_{dbo}$. The dependent variables in the conservation equations are: $v$, $\omega$, $h_a$, $T$, and $p$.

The auxiliary quantities $(T_{db})$, $(T_{wb})$, and $(\rho)$ are computed throughout the tower from thermodynamic relationships for air–water vapor mixtures from computed values of $\omega$, $h_a$, and $p$. These relationships are presented in Van Wylen and Sonntag (1973) and ASHRAE (1977), and the properties are tabulated in Keenan et al. (1969) and ASHRAE (1977).

Conservation of momentum of the gas-phase flow is approximated by the Bernoulli equation. For two points (1) and (2) along a streamline of length $\Delta s$, this is expressed as (Streeter and Wylie, 1975)

$$P_1+\frac{\rho_1 v_1^2}{2g_c}(1+\omega_1)=P_2+\frac{\rho_2 v_2^2}{2g_c}(1+\omega_2)+\frac{\rho_1 v_1^2+\rho_2 v_2^2}{4g_c}\lambda\Delta s+\left(\frac{\rho_1+\rho_2}{2}\right)\frac{g}{g_c}(y_2-y_1) \tag{1}$$

where $\lambda$ is the headloss coefficient having units of velocity heads per unit length along the streamline. Using headloss coefficients $\lambda$ for the various components of the tower (i.e., fill, spray, rain zones), the pressure distribution may be computed throughout. The flowrate between streamlines is controlled by the exit boundary condition, which requires the head at a common point (such as the plenum) to be equal across streamlines. This approach is therefore somewhat analogous to a branched pipe flow analysis.

The total heat transfer may be expressed in terms of sensible and evaporative (or latent) heat transfer. The differential sensible heat transfer rate $dQ_s$ from the water to the air is expressed as the product of the local heat transfer coefficient $H$; the temperature difference between the local water temperature $T$; the local air dry-bulb temperature $T_{db}$; and the differential interfacial area $dA_i$

$$dQ_s=H(T-T_{db})dA_i \tag{2}$$

To establish an equivalent heat transfer rate for evaporation $dQ_e$, it is convenient first to develop an expression for the mass transfer rate. The differential evaporative mass transfer rate $dE$ is defined in terms of the driving potential $B$, and the mass transfer coefficient $K$, by (Kays, 1966)

$$dE=B\,K\,dA_i \tag{3}$$

In the case of mixtures of air and water vapor, Kays defines the mass transfer driving potential $B$ in terms of the absolute humidity $\omega$ as

$$B=\frac{\omega_i-\omega}{1+\omega} \tag{4}$$

where $\omega_i$ is the absolute humidity at the interface and is assumed to be at saturation. The differential evaporative mass transfer rate $dE$ may then be expressed in terms of the absolute humidity as

$$dE=K\frac{(\omega_i-\omega)}{(1+\omega)}dA_i \tag{5}$$

The differential mass transfer rate is related to the differential latent heat transfer rate $dQ_e$ through the enthalpy of saturated water vapor $h_g$

$$dQ_e=h_g\,dE \tag{6}$$

The differential interfacial area $dA_i$ within a differential volume $dx\,dy\,dz$ of fill is expressed as

$$dA_i=a\,dx\,dy\,dz \tag{7}$$

where $a$ is the interfacial area per unit volume.

The three transfer equations of interest are then

$$dQ_s=H\,a(T-T_{db})dx\,dy\,dz \tag{8}$$

$$dE=K\,a\frac{(\omega_i-\omega)}{(1+\omega)}dx\,dy\,dz \tag{9}$$

$$dQ_e=h_g K\,a\frac{(\omega_i-\omega)}{(1+\omega)}dx\,dy\,dz \tag{10}$$

For cylindrical coordinates, $dx\,dy\,dz$ is replaced with $r\,dr\,d\theta\,dy$.

The differential total heat transfer rate $dQ_t$ is the sum of the sensible and evaporative transfers

$$dQ_t=dQ_s+dQ_e \tag{11}$$

These three transfer equations can now be used to express the conservation of mass and energy within a control volume by including the advective terms. The *conservation of mass of the water vapor* can be expressed as

$$\iiint K\,a\frac{(\omega_i-\omega)}{(1+\omega)}dx\,dy\,dz=\iint\frac{\omega}{(1+\omega)}\rho V\cdot dA \tag{12}$$

where $V\cdot dA$ is the dot product of the vector velocity $V$ and the outward normal area vector $dA$. This simply says that the net efflux of water vapor from each cell through advection must be equal to the evaporaton within the cell.

A *conservation of mass of the air* within each cell is formulated upon the concept that the net efflux of dry air from each cell is zero, or

$$\iint\left(\frac{1}{1+\omega}\right)\rho\,V\cdot dA=0 \tag{13}$$

The *conservation of energy for the air* within a control volume similarly includes the addition of energy from the sensible and latent heat transfer. This can be expressed as

$$\iiint\left[h_g K\,a\frac{(\omega_i-\omega)}{(1+\omega)}+H\,a(T-T_{db})\right]dxdydz=\iint\frac{1}{(1+\omega)}h_a\,\rho V\cdot dA \tag{14}$$

Note that by convention $h_a$ is the energy of the moist air per unit mass dry air.

The *conservation of energy for the water* within a control volume uses a slightly different formulation for the advective component. The total heat transfer rate $dQ_t$ may be expressed in terms of the change in the product of the mass flowrate of water $L$; specific heat $C_{pw}$; and the temperature $T$; or

$$dQ_t=-d(L\,C_{pw}\,T) \tag{15}$$

Combining equations (8), (10), (11), and (15) yields

$$d(L\,C_{pw}\,T)=-\left[h_g\,K\,a\frac{(\omega_i-\omega)}{(1+\omega)}+H\,a(T-T_{db})\right]dx\,dy\,dz \tag{16}$$

A sensible heat transfer coefficient $H$ and a mass transfer coefficient $K$ are required for this analysis. In the event that only one is available, the Lewis analogy, expressed as

$$H=C_{pa}\mathrm{Le}\,K \tag{17}$$

is applied (Lewis, 1922). The local Lewis number Le is deter-

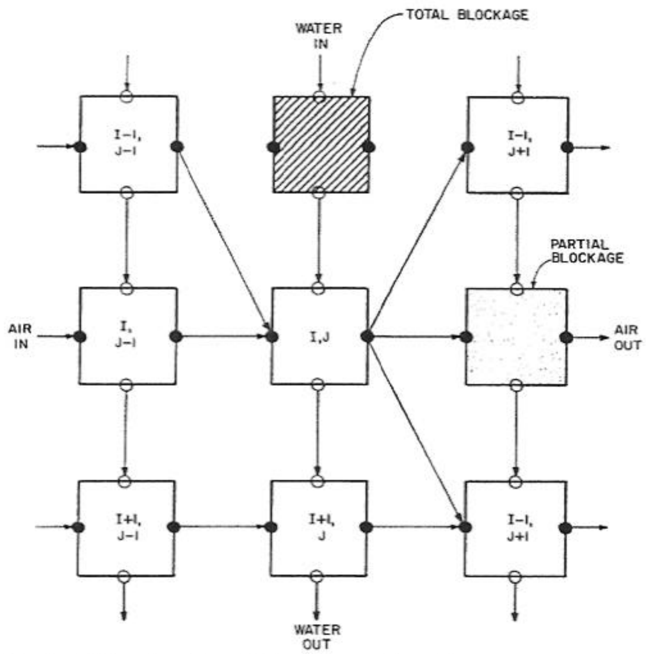


Fig. 2 Cell and node notation in crossflow fill and flowpaths in crossflow fill with blockage

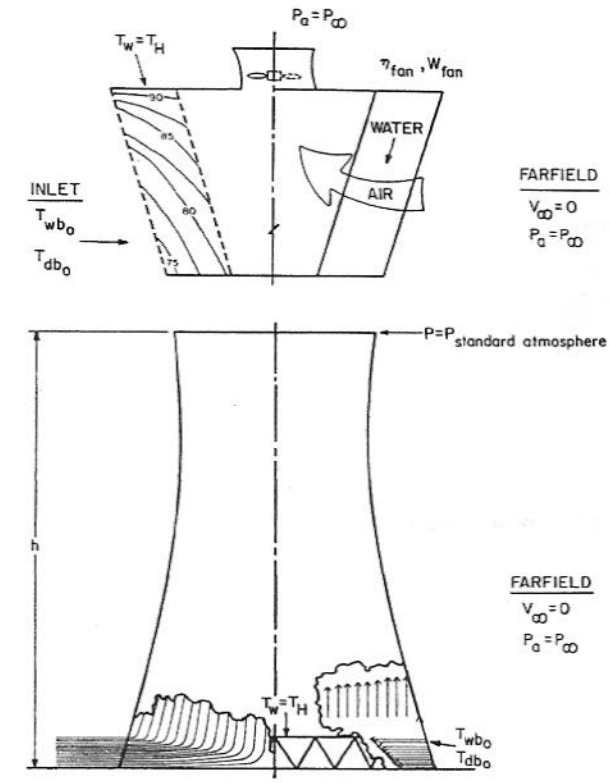


Fig. 3 Boundary conditions, streamlines, and internal distributions

mined from the local values of molecular thermal conductivity $\kappa$, density $\rho$, diffusion coefficient $D$, and specific heat $C_{pa}$

$$\mathrm{Le} = \frac{\kappa}{\rho D C_{pa}} \tag{18}$$

The local values of $\kappa$, $\rho$, and $C_{pa}$ are functions of temperature and absolute humidity; the diffusion coefficient $D$ is a function of temperature.

Certain assumptions are implicit in the simplified form of the governing equations being used. The steady-state, steady-flow assumption precludes the analysis of transients associated with plant operations or meteorology. The wet-bulb temperature is assumed equal to the adiabatic saturation temperature, the difference being negligible for these computations. Two-dimensional symmetry is also assumed. This is expressed as either lateral symmetry in Cartesian coordinates or circumferential symmetry in cylindrical coordinates, depending upon the tower geometry. Therefore, the model in its present form cannot address asymmetric airflow resulting from winds or multiple tower interference.

## Solution of the Governing Equations

Simulation of the mass, momentum, and heat transfer processes in the cooling tower requires that the tower be discretized, or divided into computational cells. Each cell is treated as a control volume, and the governing equations are applied to each. At each cell the computed dependent variables from adjacent upstream cells are utilized. These variables (e.g., enthalpy, velocity, water temperature, absolute humidity, and pressure) are defined at nodes located at the midpoints of the cell boundaries. The use of boundary nodes assures conservation of mass and energy from cell to cell (e.g., the mass leaving the east face of one cell enters the west face of the adjacent cell by virtue of common storage of the variables, see Fig. 2). Applying the Bernoulli equation and conservation equations to each cell results in a set of nonlinear simultaneous equations (five for each cell) implicitly relating the dependent variables $h_a$, $v$, $T$, $\omega$, and $p$. These implicit nonlinear simultaneous equations are solved using the Gauss-Seidel method (i.e., point-by-point successive substitution). A finite-integral formulation of the conservation equations (1), (12), (13), (14), and (16) was used. This method is described in detail by Benton (1984b).

Although natural and mechanical draft towers are quite different in appearance, much of the solution procedure is similar. For natural draft towers, flow through the chimney must be computed based upon the buoyancy of the moist air. For mechanical draft towers, the flow is computed by replacing the chimney with a fan. The pressure rise across the fan $\Delta p_{\mathrm{fan}}$ is computed using the following relationship:

$$\Delta p_{\mathrm{fan}} = \frac{\eta_{\mathrm{fan}} W_{\mathrm{fan}} \rho_{\mathrm{fan}}}{G(1+\omega_{\mathrm{fan}})} \tag{19}$$

The fan efficiency is an implicit function of airflow, air density, input power, and the pressure drop across the fill. The functional relationship for fan efficiency must be determined from field or laboratory measurements and supplied to the model. The model uses a cubic iteration to solve the pressure balance at the fan for the dry airflow $G$ and the fan efficiency $\eta_{\mathrm{fan}}$.

The differences between counterflow and crossflow towers are more significant to the computation than the differences between mechanical and natural draft towers. The differences in the boundary conditions (illustrated in Fig. 3) make the formulation and solution of the finite-integral equations somewhat different. For this reason, these two cases will be discussed separately.

## Crossflow Towers

Computation of the transport phenomena within crossflow towers is the simpler of the two cases. The finite-integral formulation of the conservation equations for this case, expressed in rectangular coordinates, becomes

*Conservation of Mass for the Dry Air*

$$\frac{\rho_w v_w A_w}{1+\omega_w} = \frac{\rho_e v_e A_e}{1+\omega_e} \tag{20}$$

*Conservation of Mass for the Water Vapor*

$$\frac{\omega_e \rho_e v_e A_e}{1+\omega_e} - \frac{\omega_w \rho_w v_w A_w}{1+\omega_w} = 1/2\ Ka\Big(\frac{\omega_{in}-\omega_e}{1+\omega_e} + \frac{\omega_{is}-\omega_w}{1+\omega_w}\Big)\Delta x \Delta y \Delta z \qquad (21)$$

where $n$, $s$, $e$, and $w$ refer to north, south, east, and west, respectively (see Fig. 2); and $\omega_{in}$ and $\omega_{is}$ are the saturated absolute humidities evaluated at the north and south water temperatures, respectively.

*Conservation of Energy for the Air*

$$\frac{\rho_e v_e A_e h_{ae}}{1+\omega_e} - \frac{\rho_w v_w A_w h_{aw}}{1+\omega_w} = \Big(1/2\ Ka\Big(\frac{\omega_{in}-\omega_e}{1+\omega_e} + \frac{\omega_{is}-\omega_w}{1+\omega_w}\Big)h_{ge} + 1/2\ Ha((T_n - T_{dbe}) + (T_s - T_{dbw}))\Big)\Delta x \Delta y \Delta z \qquad (22)$$

where $T_n$ and $T_s$ are the water temperatures at the north and south nodes, respectively; and $T_{dbe}$ and $T_{dbw}$ are the dry-bulb temperatures at the east and west nodes, respectively.

*Conservation of Energy for the Water*

$$C_{pw}(L_n T_n - L_s T_s) = \frac{\rho_e v_e A_e h_{ae}}{1+\omega_e} - \frac{\rho_w v_w A_w h_{aw}}{1+\omega_w} \qquad (23)$$

*Bernoulli's Equation*

$$P_w + \frac{\rho_w v_w^2}{2g_c}(1+\omega_w) = P_e + \frac{\rho_e v_e^2}{2g_c}(1+\omega_e) + \frac{\rho_w v_w^2 + \rho_e v_e^2}{4g_c}\lambda\,\Delta x \qquad (24)$$

These five equations are solved throughout a two-dimensional network of cells representing the fill region. The properties of the fill do not have to be homogeneous; consequently, the model will accommodate obstructions, voids, and hybrid fill as demonstrated by Benton (1984a). The entering waterflow distribution at the top row of cells and a distribution of inlet wet- and dry-bulb temperatures can also be arbitrarily specified.

The iterative solution process is initiated by solving a simplified point model to obtain initial values for the airflow rate and wet- and dry-bulb temperature at the exit of the cooling tower. The results are used to provide an initial distribution of the absolute humidity, enthalpy, and wet- and dry-bulb temperatures throughout the tower. Linear interpolation from entrance to exit is used to obtain initial values at intermediate locations.

The computation of sensible and latent heat transfer rates starts in the *upper lefthand corner* cell as shown in Fig. 3, and proceeds one column at a time. The computation for each cell is an iterative process since the driving potential for the transfer utilizes the conditions at the exit. After the energy transfer in the cells located in the fill has been computed, the air is assumed to be thoroughly mixed as it flows through the chimney (or fan) without further heat or mass transfer. Exit conditions of the two-dimensional analysis of the rain, fill, and spray zones are numerically integrated to establish initial conditions for the one-dimensional computation through the chimney. In the absence of heat or mass transfer, a one-dimensional distribution of cells is assumed to be sufficient in the chimney region of natural draft towers and in the fan and stack region of mechanical draft towers.

With the temperatures and absolute humidities established based on a trial airflow estimate, an evaluation of the airflow distribution is undertaken. The Bernoulli equation and the conservation of mass of dry air is applied to each cell. The airflow distribution is solved in the same manner as a branched pipe network having interconnecting paths which permit crossflow. This solution of the Bernoulli and the conservation of mass of air equations provides values of velocity and pressure at the exit face of each cell within the fill. An average velocity is computed for the plenum, and the Bernoulli equation is used through the chimney or fan. The fan and recovery stack in mechanical draft towers are each modeled as a single cell. In natural draft towers the chimney is discretized into a number of cells.

Based on the new flow distribution, revised values of temperature, enthalpy, and humidity ratio are computed. These values reflect vertical mixing between adjacent cells in the fill as indicated in Fig. 3. The sensible and latent heat tranfer rates in the various cells are recomputed using the temperatures, densities, and velocities from the preceding step.

The iterative process is considered to have converged when the computed pressure at the exit plane corresponds to the ambient pressure.

## Counterflow Towers

Computations for counterflow towers are more complicated than those for crossflow towers because of the nature of the boundary conditions. Also the transport phenomena must be calculated in four separate zones: the rain, fill, spray, and chimney (or fan) zones. Each zone is treated separately, but obviously the dependent variables at the boundaries of the zones must match. The finite-integral approximation to the governing equations is slightly different from that of the crossflow towers because the direction of flow is different. For counterflow towers, the finite-integral equations expressed in cylindrical coordinates are:

*Conservation of Mass for the Dry Air*

$$\frac{\rho_s v_s A_s}{1+\omega_s} = \frac{\rho_n v_n A_n}{1+\omega_n} \qquad (25)$$

*Conservation of Mass for the Water Vapor*

$$\frac{\omega_n \rho_n v_n A_n}{1+\omega_n} - \frac{\omega_s \rho_s v_s A_s}{1+\omega_s} = 1/2\ Ka\Big(\frac{\omega_{in}-\omega_n}{1+\omega_n} + \frac{\omega_{is}-\omega_s}{1+\omega_s}\Big)2\pi r \Delta r \Delta y \qquad (26)$$

*Conservation of Energy for the Air*

$$\frac{\rho_n v_n A_n h_{an}}{1+\omega_n} - \frac{\rho_s v_s A_s h_{as}}{1+\omega_s} = \Big(1/2\ Ka\Big(\frac{\omega_{in}-\omega_n}{1+\omega_n} + \frac{\omega_{is}-\omega_s}{1+\omega_s}\Big)h_{gn} + 1/2\ Ha((T_n - T_{dbn}) + (T_s - T_{dbs}))\Big)2\pi r \Delta r \Delta y \qquad (27)$$

*Conservation of Energy for the Water*

$$C_{pw}(L_n T_n - L_s T_s) = \frac{\rho_n v_n A_n h_{an}}{1+\omega_n} - \frac{\rho_s v_s A_s h_{as}}{1+\omega_s} \qquad (28)$$

*Bernoulli's Equation*

$$P_s + \frac{\rho_s v_s^2}{2g_c}(1+\omega_s) = P_n + \frac{\rho_n v_n^2}{2g_c}(1+\omega_n) + \frac{\rho_s v_s^2 + \rho_n v_n^2}{4g_c}\lambda\,\Delta y + \frac{(\rho_n + \rho_s)}{2}\frac{g\Delta y}{g_c} \qquad (29)$$

Boundary conditions for this case (shown in Fig. 3) require an iterative procedure. The iterative solution process is initiated by solving a simplified point model to obtain initial

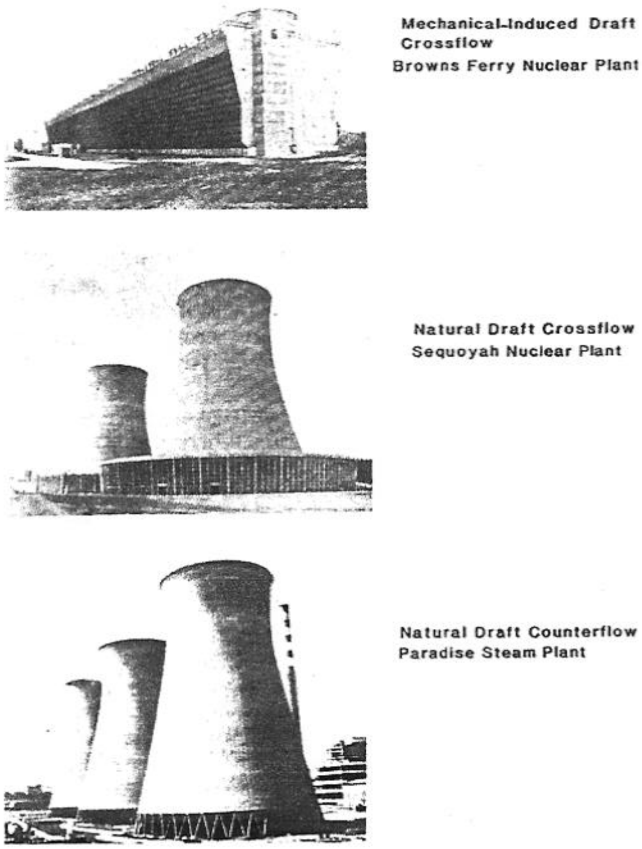


Fig. 4 TVA evaporative cooling towers used for model verification

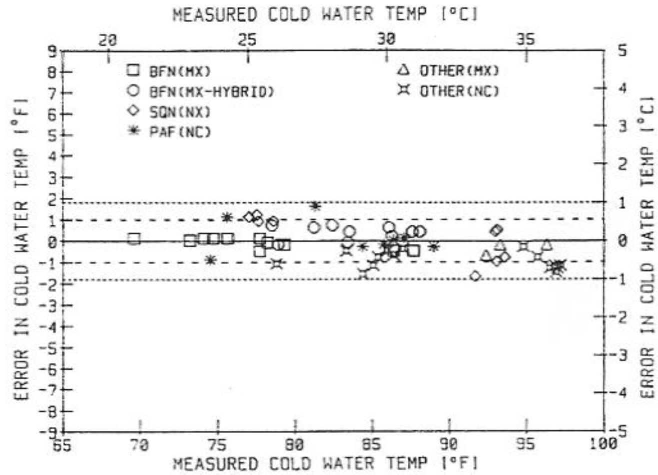


Fig. 5 Comparison of measured and computed cold water temperature

values for the airflow rate and wet- and dry-bulb temperature at the exit of the cooling tower. The results are used to provide an initial distribution of the absolute humidity, enthalpy, and wet- and dry-bulb temperatures throughout the tower. Linear interpolation from entrance to exit is used to obtain initial values at intermediate locations. An initial estimate of the airflow distribution is computed based on uniform radial velocity at the inlet.

The solution procedure is initiated in the rain zone and is based on an assumed cold water temperature so that heat transfer computations can proceed. Flows through the fill zone are confined to the vertical direction, which is a logical assumption for sheet fill. When the computations are advanced to the spray nozzles, a check is made to determine whether the computed hot water temperature corresponds to the specified hot water temperature. If the computed and specified hot water temperatures do not agree, the cold water temperature at the base of the tower is adjusted and the heat transfer is recomputed. When the computed and specified hot water temperatures agree, the pressure computations are performed. The airflow within the rain and fill zones is redistributed among the assumed pathlines (Fig. 3) according to the Bernoulli equation such that dynamic pressure (i.e., $p+(1+\omega)\rho v^2/2g_c$) at the top of the fill is uniform in the radial direction. The computed pressure at the exit plane is then compared to the specified ambient pressure. If the pressures are not in agreement, the airflow is appropriately adjusted and the heat transfer computations restarted. This series of computations continues until the computed hot water temperature and exit pressure correspond with the values prescribed as boundary conditions.

The entering waterflow can be arbitrarily distributed for these computations. Likewise, an arbitrary distribution of inlet wet- and dry-bulb temperatures can be specified. The model can also analyze nonhomogeneous or unevenly distributed fills so long as they retain circumferential symmetry.

## Comparison of Model Results With Field Data

The validity of the model has been tested by comparing model results with field data collected on cooling towers at three TVA power plants as well as data from other utilities. These towers are each fundamentally different in design as shown in Fig. 4, providing data on natural draft counterflow, natural draft crossflow, and rectangular mechanical draft crossflow towers. In addition, field data are also available on one of the mechanical draft towers modified by adding hybrid fill (Gidley, 1980; Benton, 1984a).

Each application was initiated by specifying the geometry of the tower, the fill characteristics, meteorology, and plant operating data (i.e., water temperature and flowrate). The fill characteristics used in the model for these comparisons were taken from Majumdar and Singhal (1981). The results of the comparison are presented in Fig. 5. It is important to note that no calibration of the model was used in making the predictions. Although only the global results of these computations are shown in Fig. 5, the model also provides distributions of dependent variables throughout the tower for each case as illustrated in Fig. 3. Such distributions are important in providing insight into the internal transport phenomena, which is essential to correcting deficiencies.

## Model Statistics

The present model has been developed into a FORTRAN coded computer program. The program is comprised of a main program and 38 subprograms in approximately 7000 statements. The computer program requires 109K bytes (1 byte = 8 binary bits) of memory. However, only 33K bytes are required for any one geometry (i.e., mechanical-crossflow). The execution time is approximately 30 s on an HP-1000F minicomputer, 90 s on an IBM-XT, or 1 s on an IBM-3031.

The computer program models mechanical draft crossflow (rectangular), mechanical draft counterflow (round or rectangular), natural draft crossflow (round), and natural draft counterflow (round) towers.

## Summary

Complex and complete solution of heat, mass, and momentum processes cannot be described in their most basic form. However, a cooling tower can be subdivided into components, with heat, mass, and momentum described using empirical coefficients.

A mathematical model, which can be run economically on a minicomputer or a microcomputer, has been developed to solve equations describing these processes. The solutions to these equations provide a two-dimensional description of the flow and cooling processes throughout the tower. Results from the mathematical model compare favorably with data from field tests of TVA cooling towers as well as those of other utilities under a variety of operating conditions.

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