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A complete model of wet cooling towers with fouling in fills

Bilal A. Qureshi, Syed M. Zubair *

Mechanical Engineering Department, King Fahd University of Petroleum and Minerals (KFUPM), Mail Box 1474, Dhahran 31261, Saudi Arabia

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Abstract

A cooling tower basically consists of three zones; namely, spray zone, packing and rain zones. In cooling towers, a significant portion of the total heat rejected may occur in the spray and rain zones. These zones are modeled and solved simultaneously using engineering equation solver (EES) software. The developed models of these zones are validated against experimental data. For the case study under consideration, the error in calculation of the tower volume is 6.5% when the spray and rain zones are neglected. This error is reduced to 3.15% and 2.65% as the spray and rain zones are incorporated in the model, respectively. Furthermore, fouling in cooling tower fills as well as its modeling strategy is explained and incorporated in the cooling tower model to study performance evaluation problems. The fouling model is presented in terms of normalized fill performance index ($\eta_{F,norm}$) as a function of weight gain due to fouling. It is demonstrated that the model is asymptotic, which is similar to typical asymptotic fouling model used in conventional heat exchangers.

Keywords: Cooling tower; Spray zone; Rain zone; Fouling

1. Introduction and background

Cooling towers are commonly used in large thermal systems, such as most industrial power generation units, refrigeration and air conditioning plants, chemical and petrochemical industries, to reject waste heat. The towers are designed to cool a warm water stream through evaporation of some of the water into an air stream. There are several types of cooling towers; of which the mechanical draft tower is probably the most common wherein the water enters at the top of the tower as a spray and flows downward through the tower. Ambient air is drawn into the tower using fans, and flows in a counter or crosscurrent direction to the water stream. If the fans are at the bottom of the tower and blow the air upward past the water stream, the tower is called a forced draft tower, whereas if the fans are at the top, it is an induced draft tower. However, large-size atmospheric towers do not use a fan and rely on the buoyancy effect of the heated air that naturally moves from bottom to the top of the tower due to hyperbolic shape. The cooling towers at conventional or nuclear power plants are of this type.

The inside of the tower is packed with fill or packing material providing a large surface area for both heat and mass transfer to take place from water droplets to air, as shown in Fig. 1. Wooden slats are a common fill material but fiberglass reinforced plastics are becoming very popular. A basic theory of cooling tower operation was originally proposed by Walker et al. [1] but the practical use of basic differential equations, however, was first presented by Merkel [2], in which he combined the equations governing heat and mass transfer between water droplets and air in the tower. Webb [3] presented a unified theoretical treatment for thermal analysis of cooling towers, evaporative condensers and evaporative fluid coolers. For a detailed procedure for thermal design of wet cooling towers is described by Mohiuddin and Kant [4,5]. Performance analysis for the steady-state counter-flow wet cooling tower with new definitions of tower effectiveness and number of transfer units is discussed by Dessouky et al. [6]. They considered

* Corresponding author. Tel.: +966 3 860 3135; fax: +966 3 860 2949.
E-mail address: smzubair@kfupm.edu.sa (S.M. Zubair).

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Nomenclature

A_V	surface area of water droplets per unit volume of the tower ($m^2\ m^{-3}$)
$b_{1\ldots4}$	dimensional coefficients
c_p	specific heat at constant pressure ($kJ\ kg^{-1}\ K^{-1}$)
C_D	drop drag coefficient
C_1	constant used in Eq. (22)
C_2	constant used in Eq. (22) ($kg\ m^{-3}$)
d	diameter (m)
D	diffusion coefficient ($m^2\ s^{-1}$)
G	mass velocity ($kg\ m^{-2}\ s^{-1}$)
g	acceleration due to gravity ($m\ s^{-2}$)
h	enthalpy of moist air ($kJ\ kg_a^{-1}$)
h_c	convective heat-transfer coefficient of air ($kW\ m^{-2}\ K^{-1}$)
h_D	convective mass transfer coefficient ($kg_w\ m^{-2}\ s^{-1}$)
h_f	specific enthalpy of saturated liquid water ($kJ\ kg_w^{-1}$)
$h_{f,w}$	specific enthalpy of water evaluated at t_w ($kJ\ kg_w^{-1}$)
h_g	specific enthalpy of saturated water vapor ($kJ\ kg_w^{-1}$)
h_g^0	specific enthalpy of saturated water vapor evaluated at 0 °C ($kJ\ kg_w^{-1}$)
$h_{fg,w}$	change-of-phase enthalpy ($h_{fg,w} = h_{g,w} - h_{f,w}$) ($kJ\ kg_w^{-1}$)
H	height (m)
k	thermal conductivity ($W\ m^{-1}\ K^{-1}$)
Le	Lewis factor ($Le = h_c/h_D c_{p,a}$)
m	mass (kg)
$\dot{m}$	mass flow rate ($kg\ s^{-1}$)
m_{ratio}	water-to-air mass flow rate ratio ($m_{ratio} = \dot{m}_w/\dot{m}_{air}$)
M	median weight to reach critical level of fouling ($kg\ m^{-3}$)
n	number of drops
Nu	Nusselt number $= h_c(2r_{d,eff})/k$
ODEs	ordinary differential equations
p	risk level
P	pressure (kPa)
r	radial direction (m)
Re	Reynolds number
Sc	Schmidt number $= \mu_a/\rho_a D$
t	dry-bulb temperature of moist air (°C)
t_w	water temperature (°C)
v	velocity ($m\ s^{-1}$)
V	volume of tower (m^3)
w	weight density of fouling material ($kg\ m^{-3}$)
W	humidity ratio of moist air ($kg_w\ kg_a^{-1}$)
z	axial direction (m)
η_F	fill performance index ($\eta_F = h_D A_V V/\dot{m}_w$)
$\alpha^{1/2}$	scatter in time
Φ^{-1}	inverse of normal distribution function
ε	effectiveness
μ	viscosity ($kg\ m^{-1}\ s^{-1}$)
ρ	density ($kg\ m^{-3}$)
σ	surface tension ($N\ m^{-1}$)

Subscripts

a	air
c	clean conditions
cal	calculated
cr	critical fouled conditions
d	drop
db	dry-bulb
eff	effective
em	empirical
f	fouled conditions
g, w	vapor at water temperature
hor	horizontal component (of velocity)
in	inlet
norm	normalized
out	outlet
rz	rain zone
sz	spray zone
s, w	saturated moist air at water temperature
v	vapor or volume
w	water
wb	wet-bulb
0	denotes the initial value

the effect of interface temperature and Lewis number but the effect of water evaporation on the air process states, along the vertical length, was not considered. It is important to note that all the above studies basically deal with heat and mass transfer analysis in the fill zone of the tower, which is considered to be the main part of the tower.

In a counter-flow cooling tower, the hot process water to be cooled is sprayed into an upward flowing air stream using a number of nozzles. The nozzles are arranged in such a way that the water is uniformly distributed over the fill material. Due to heat and mass transfer, the water temperature is reduced while the air enthalpy is increased because the air is heated and saturated by the water as it moves up. Kroger [7] indicated that up to 15% of the cooling might actually occur in the spray zone of large cooling towers. The typical height from the nozzle to top of the fill is about 18 in. regardless of a tower's capacity [8]. This is the height normally required for the spray pattern to develop. Using a greater number of smaller nozzles could lessen the distance but the cost and tendency to plug up increases.

The rain zone is required in a conventional tower to permit uniform airflow into the fill; however, from a thermal perspective, this is a very inefficient portion of the cooling

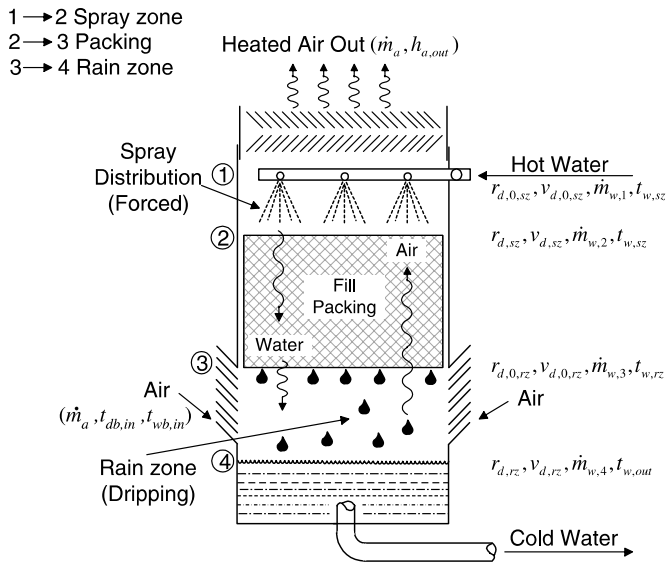

Fig. 1. Schematic of a counter-flow wet cooling-tower.

tower. It is only as large as necessary to allow even airflow. Observations show [8] that the droplets and jets in the rain zone are formed due to dripping of water from the sheets of the fill. Therefore, the radius of the droplets is quite large compared to the spray zone. Davis [8] indicated that cooling achieved in one foot of fill can be more than the cooling in ten feet of free-fall of water and, as a consequence, is an ineffective use of pump energy. For blow through towers, it tends to be bigger to make room for the fans. Though a significant part (10–20%) of the total heat and mass transfer in large towers occurs in the rain zone below the packing; however, this is not the case for small-sized towers. For a typical 100 ton blow-through tower, the rain zone may be 36 in., the fill 36 in. and the spray zone 18 in. As the overall size of a tower increases, the fill would be as much as 54 in. and the rain zone height would increase proportional to air flow [8].

In any detailed performance analysis of the wet cooling tower, the transfer processes in the spray or rain zone may not be ignored. Earlier studies considered these transfer processes either too complex or relatively unimportant to analyze. As discussed above, in large counter flow wet-cooling towers, these zones make an important contribution to the overall performance, therefore, knowledge of reliable models for prediction of the total performance are important to fully understand the contribution of these regions. The objective of this paper is to present mathematical modeling of three parts i.e., spray zone, packing and rain zones. Furthermore, the importance of incorporating fouling model is also highlighted. In this regard, a case study is presented to show the validity of using spray and rain zone models in conjunction with the packing model for accurate sizing and performance evaluation purpose.

2. Fill zone model

The control volume of a counter-flow cooling tower is presented in Fig. 2. The major assumptions that are used to derive the basic modeling equations are summarized in [9,10].

From steady-state energy balance between air and water where evaporation is considered, we get

$$\dot{m}_a \mathrm{d}h = (\dot{m}_{w,out} - \dot{m}_a(W_{in} - W))\mathrm{d}h_{f,w} + \dot{m}_a \mathrm{d}W\ h_{f,w} \tag{1}$$

The water energy balance can be written in terms of the heat- and mass transfer coefficients, h_c and h_D, respectively, as

$$\dot{m}_w \mathrm{d}h_{f,w} + \dot{m}_a \mathrm{d}W\ h_{f,w} = h_c A_V (t_w - t)\mathrm{d}V + h_D A_V\ h_{fg,w}(W_{s,w} - W)\mathrm{d}V \tag{2}$$

and the air side water–vapor mass balance as

$$\dot{m}_a \mathrm{d}W = h_D A_V \mathrm{d}V (W_{s,w} - W) \tag{3}$$

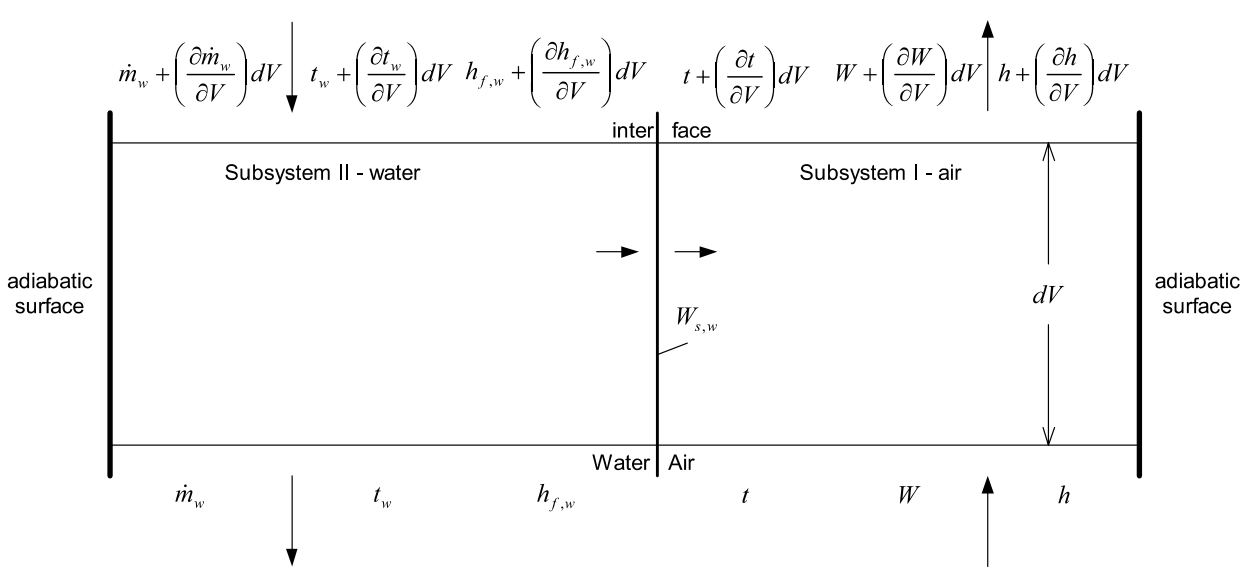

Fig. 2. Control volume of a counter-flow wet cooling-tower.

Now, by substituting Lewis factor as $Le = h_c/h_D c_{p,a}$ in Eq. (2), we get after some simplification

$$\dot{m}_w dh_{f,w} + \dot{m}_a dW\ h_{f,w} = h_D A_V dV[Le\ c_{p,a}(t_w - t_{db}) + (W_{s,w} - W)h_{fg,w}] \quad (4)$$

It should be noted that the Lewis factor is defined similar to what is used by Kuehn et al. [9] and Braun et al. [11]. Combining Eqs. (1)–(4), we get after simplification

$$\frac{dh}{dW} = Le\frac{(h_{s,w} - h)}{(W_{s,w} - W)} + (h_{fg,w} - h_g^0 Le) \quad (5)$$

Eq. (5) describes the condition line on the psychrometric chart for the changes in state for moist air passing through the tower. For given water temperatures ($t_{w,in}$, $t_{w,out}$), Lewis factor (Le), inlet condition of air and mass flow rates, Eqs. (1),(3) and (5) may be solved numerically for exit conditions of both the air and water streams.

A computer program is written in Engineering Equation Solver (EES) for solving the above equations. In this program, properties of air–water vapor mixture are needed at each step of the numerical calculation, which are obtained from the built-in functions provided in EES. The program gives the dry-bulb temperature, wet-bulb temperature and humidity ratio of air as well as water temperature at each step of the calculation starting from air-inlet to air-outlet values.

The correlations for heat and mass transfer of cooling towers in terms of physical parameters are not easily available. The mass transfer coefficient is unknown but it is often correlated in the form [12]

$$\frac{h_D A_V V}{\dot{m}_{w,in}} = c\left(\frac{\dot{m}_{w,in}}{\dot{m}_a}\right)^n \quad (6)$$

where c and n are empirical constants specific to a particular tower design. Multiplying both sides of the above equation by $(\dot{m}_{w,in}/\dot{m}_a)$ and considering the definition for number of transfer unit (NTU) gives the empirical value of NTU as

$$\mathrm{NTU}_{em} = \left.\frac{h_D A_V V}{\dot{m}_a}\right|_{em} = c\left(\frac{\dot{m}_{w,in}}{\dot{m}_a}\right)^{n+1} \quad (7)$$

Braun et al. [11] fitted the coefficients c and n in the above equation to the measurements of Simpson and Sherwood [13] for four different tower designs over a range of performance conditions.

The definition for the cooling tower effectiveness used is given below [9], where it is described as the ratio of actual energy to maximum possible energy transfer. Also, we note that the dimensionless temperature difference described in the cooling tower literature is defined as the ratio of actual to maximum water temperature drop.

$$\varepsilon = \frac{h_{out} - h_{in}}{h_{s,w} - h_{in}} \quad (8)$$

Notice that the system of three differential equations describing fill zone operation is given by Eqs. (1), (3) and (5) with Eq. (7) used to calculate the mass transfer coefficient. These will be solved numerically for the case under consideration to ascertain the importance of the spray and rain zones in design of cooling towers.

2.1. Fouling in the packing material

Fouling, as defined for cooling towers, is the process of deposition of foreign matter, including bio-growth, on the plastic water film flow area. It inhibits the cooling process or allows excessive weight to build up on the packing. In more severe circumstances; however, fouling can result in a reduction in the overall effectiveness of the tower—a symptom of fill fouling interfering with air and water flow through the tower. It is important to note that plastic fills are more prone to fouling than traditional splash bars.

Fouling of cooling tower fills is one of the most important factors affecting its thermal performance, which reduces cooling tower effectiveness and tower capability with time. In this paper, the fouling model presented in an earlier paper [14], using experimental data [15] on fill fouling, is discussed, which can be used to investigate the risk based thermal performance of the cooling tower. The data showed that the tower characteristics reduced to 18% that of clean condition and then stabilized even though weight continued to increase due to fouling. The developed model shows a strong correlation between normalized fill performance index due to fouling ($\eta_{F,norm}$) as a function of weight gain (w). The model is of the form [14]:

$$\eta_{F,norm} = \left(\frac{h_D A_V V}{\dot{m}_w}\right)_{norm} = \frac{\left(\left(\frac{h_D A_V V}{\dot{m}_w}\right)_c - \left(\frac{h_D A_V V}{\dot{m}_w}\right)_f\right)}{\left(\frac{h_D A_V V}{\dot{m}_w}\right)_c} = C_1(1 - \exp(-w/C_2)) \quad (9)$$

where C_1 and C_2 are constants depending on the fouling characteristics of the tower. C_1 represents the increase in value of $\eta_{F,norm}$ when the fouling reaches its asymptotic value, and C_2 represents the weight gain constant indicating that the fill performance index has decreased to 63.2% of the asymptotic value of weight gain due to fouling.

It is important to note that Eq. (9) can be expressed as

$$\ln\left(\frac{1}{1 - \eta_{F,norm}/C_1}\right) = w/C_2 \quad (10)$$

The above equation may be expressed as a linear representation of the asymptotic fouling growth model, where the constant C_2 is expressed in terms of the critical acceptable value of fouling resistance $\eta_{F,norm,cr}$ and the fouling weight to reach this critical value w_{cr}, is given by

$$C_2 = w_{cr}/\ln\left[1/(1 - \eta_{F,norm,cr}/C_1)\right] \quad (11)$$

where

$$w_{cr} = M/\left[1 - \sqrt{\alpha}\Phi^{-1}(p)\right] \quad (12)$$

Substituting the value of C_2 into Eq. (9) and rearranging, we get

$$\eta_{\mathrm{F,norm}}(w,p;\sqrt{\alpha}) = C_1[1-\exp\{-\ln[1/(1-\eta_{\mathrm{F,norm,cr}}/C_1)] \times [1-\sqrt{\alpha}\Phi^{-1}(p)](w/M)\}] \quad (13)$$

The parameters M and $\alpha^{1/2}$ in the above equation represent the median weight and the scatter parameter in a transformed coordinate system. The value of the critical normalized fill performance index divided by the asymptotic value (i.e., $\eta_{\mathrm{F,norm,cr}}/C_1$) can be calculated from the experimental data [14]. A packing model in conjunction with the above fouling model can be used to study the effect of fouling on tower effectiveness and water outlet temperatures for cooling tower operating under different conditions. Fig. 3 shows the plot of normalized fill performance index as a function of reduced weight (w/M) and the risk level (p), representing the probability of fill surface being fouled up to a critical level after which a cleaning is needed. In plotting this figure, we have taken the critical normalized fill performance index as 84% that of the asymptotic value since Khan and Zubair [14] carried out a systematic regression analysis of the experimental data and found that this value of ($\eta_{\mathrm{F,norm,cr}}/C_1$) gave the best representation. As expected, this figure also shows that, for a low risk level, the fouling weight required to reach the critical level is faster compared with the deterministic case (i.e., $p = 0.50$).

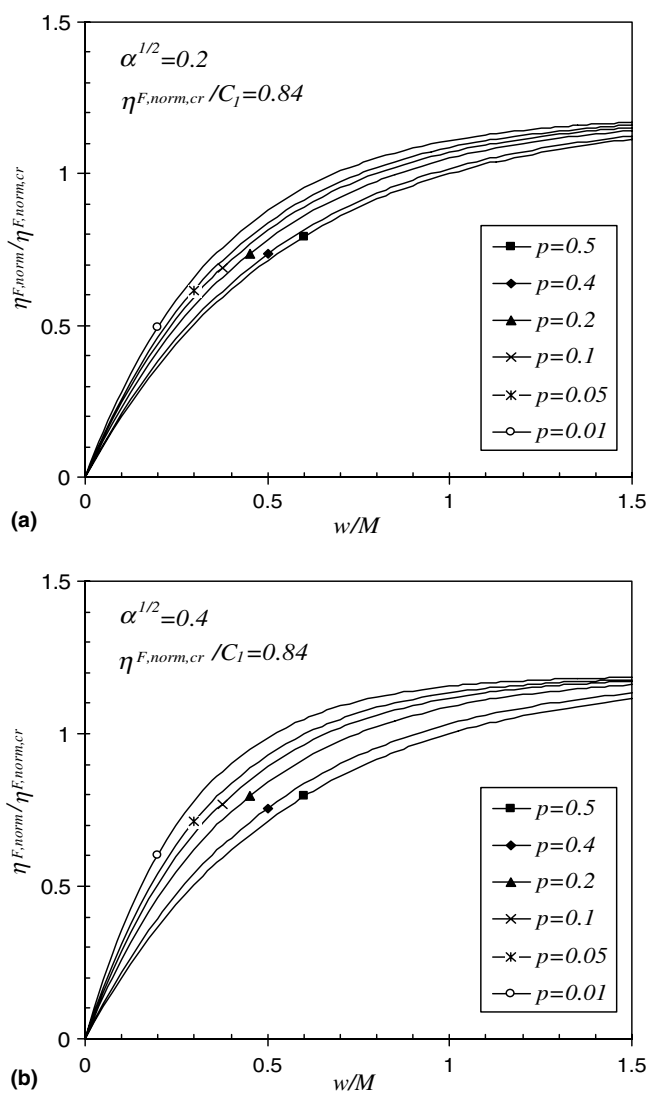

Fig. 3. Normalized fill performance index versus reduced weight (w/M); (a) for $\sqrt{\alpha} = 0.2$, and (b) $\sqrt{\alpha} = 0.4$.

2.2. Spray and rain zone model

Fisenko et al. [16] explained that the contribution of droplets cooling to heat balance of the tower mainly depended on their radius; however, it is obvious that the radius of the droplets in the spray zone depends on the water flow rate: the higher the water flow rate, the smaller is the droplet size due to larger pressure drop on sprinklers. The dependence of the radius of the droplets in the spray zone on the hydraulic load (water flux) is attributable to the design of the sprinkler nozzle and is not associated with the breaking phenomenon of the droplets. Furthermore, at maximum hydraulic load, the droplets' velocity leaving the sprinkler is not sufficient for breaking. Therefore, the maximum radius of the droplet falling with the velocity v_{d} was calculated from the balance of contributions of aerodynamic drag force and the surface tension with the air ascending flow velocity v_{a}. This balance of forces helps us to determine the minimal size of the droplets taking part in the process of evaporative cooling. If the force of aerodynamic resistance exceeds that of the gravity, which is true for rather small droplets, the droplets are carried away by the ascending airflow.

The influence of number of droplets per unit volume n_{V} on the moist air parameters is taken into account wherein it is defined by the water flow rate and droplet size by [16]

$$n_{\mathrm{V}} = \frac{3G_{\mathrm{w}}}{4\pi\rho_{\mathrm{w}} r_{\mathrm{d,eff}}^3 v_{\mathrm{d}}} \quad (14)$$

The Reynolds number and Nusselt number are defined by using the relations [17]

$$Re_{\mathrm{d}} = \frac{2\rho_{\mathrm{a}} r_{\mathrm{d,eff}}[(v_{\mathrm{d}}-v_{\mathrm{a}})^2 + v_{\mathrm{d,hor}}^2]^{0.5}}{\mu_{\mathrm{a}}} \quad (15)$$

$$Nu_{\mathrm{d}} = 2 + 0.5Re_{\mathrm{d}}^{0.5} \quad (16)$$

where $v_{\mathrm{d,hor}}$ is the horizontal component of the drop velocity.

Using the analogy between the heat and mass transfer processes, for a droplet falling in an ascending airflow, the mass transfer coefficient $h_{\mathrm{D,d}}$ can be expressed as

$$h_{\mathrm{D,d}} = \rho_{\mathrm{a}} \frac{DNu_{\mathrm{d}}}{2r_{\mathrm{d,eff}}(z)} \quad (17)$$

where D is the diffusion coefficient of water vapor.

The aerodynamic drag force of a droplet was calculated by [18]:

$$C_{\mathrm{D}} = \frac{24}{Re_{\mathrm{d}}}\left(1 + \frac{1}{6}Re_{\mathrm{d}}^{2/3}\right) \quad (18)$$

The system of differential equations that are used to calculate the processes of heat and mass transfer between the falling droplet and the ascending moist air can be summarized in terms of the following differential equations describing [16]:

$$\frac{\mathrm{d}r_{\mathrm{d,eff}}(z)}{\mathrm{d}z} = -\frac{h_{\mathrm{D,d}}[W_{\mathrm{s,w}}(t_{\mathrm{d}}(z)) - W(z)]}{\rho_{\mathrm{w}} v_{\mathrm{d}}(z)} \tag{19}$$

$$\frac{\mathrm{d}v_{\mathrm{d}}(z)}{\mathrm{d}z} = \frac{g}{v_{\mathrm{d}}(z)} - C_{\mathrm{D}} \frac{\rho_{\mathrm{a}}[v_{\mathrm{d}}(z) - v_{\mathrm{a}}]^2}{2v_{\mathrm{d}}(z)} \frac{\pi r_{\mathrm{d,eff}}^2}{m_{\mathrm{d}}} \tag{20}$$

$$\frac{\mathrm{d}t_{\mathrm{d}}(z)}{\mathrm{d}z} = \frac{3\left[h_{\mathrm{c,d}}\{t(z) - t_{\mathrm{d}}(z)\} - h_{\mathrm{D,d}} h_{\mathrm{fg}}\{W_{\mathrm{s,w}}(t_{\mathrm{d}}(z)) - W(z)\}\right]}{c_{\mathrm{p,w}} \rho_{\mathrm{w}} r_{\mathrm{d,eff}}(z) v_{\mathrm{d}}(z)} \tag{21}$$

$$\frac{\mathrm{d}t(z)}{\mathrm{d}z} = \frac{4\pi r_{\mathrm{d,eff}}^2(z) n_{\mathrm{V}}}{c_{\mathrm{p,a}} \rho_{\mathrm{a}} (v_{\mathrm{d}}(z) - v_{\mathrm{a}})} [h_{\mathrm{c,d}}\{t(z) - t_{\mathrm{d}}(z)\}] \tag{22}$$

$$\frac{\mathrm{d}W(z)}{\mathrm{d}z} = -\frac{4\pi r_{\mathrm{d,eff}}^2(z) n_{\mathrm{V}}}{\rho_{\mathrm{a}} (v_{\mathrm{d}}(z) - v_{\mathrm{a}})} h_{\mathrm{D,d}}[W_{\mathrm{s,w}}(t_{\mathrm{d}}(z)) - W(z)] \tag{23}$$

The five boundary conditions needed for the above system of differential equations consist of the initial value of the droplet radius, temperature and velocity at the beginning of the droplet fall (refer to point # 3 in Fig. 1) while the temperature and humidity ratio of the air at the final point of the droplet fall (i.e., point # 4 in Fig. 1).

2.3. Determination of effective drop diameter

Fisenko et al. [16] explained a method to determine the effective drop radius for the spray zone. This required an experimental value of the temperature drop occurring in the spray zone. Then, using the model, water temperature drop was calculated against various effective drop radii. The correct radius is found where the experimental and calculated water temperature drops are the same. In the current situation, Simpson and Sherwood [13] did not provide such an experimental value and this was substituted with the temperature drop calculated from Dreyer's procedure [19] of evaluating the performance of the spray zone where the experimental value of the outlet enthalpy was used instead of assuming it. Fig. 4 illustrates the result

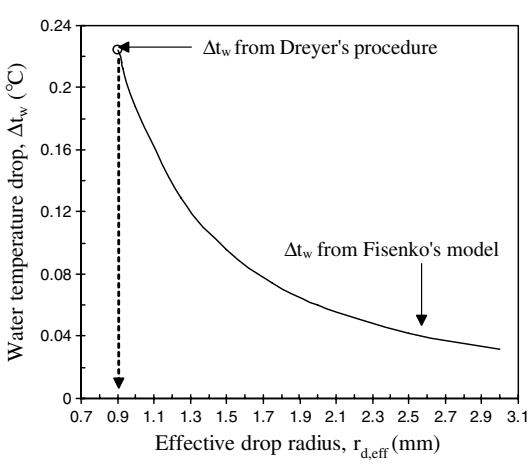

Fig. 4. Determination of effective drop radius for the spray zone.

of this method showing that an effective drop radius of 0.9 mm was obtained.

The model of De Villiers and Kroger [20] can be used to determine the effective drop diameter for the rain zone, which is summarized as

$$\begin{aligned}\frac{h_{\mathrm{D,rz}} A_{\mathrm{V,rz}} H_{\mathrm{rz}}}{G_{\mathrm{w}}} = {} & 3.6\left(\frac{P_{\mathrm{a}}}{R_{\mathrm{v}} t_{\mathrm{a}}} \Big/ \rho_{\mathrm{w}}\right)\left(\frac{D}{v_{\mathrm{a,in}} d_{\mathrm{d,eff}}}\right)\left(\frac{H_{\mathrm{rz}}}{d_{\mathrm{d,eff}}}\right) Sc^{0.33} \\ & \times \ln\left[\frac{W_{\mathrm{s}} + 0.622}{W + 0.622}\right] \Big/ (W_{\mathrm{s}} - W) \\ & \times \{5.01334 b_1 \rho_{\mathrm{a}} - 192121.7 b_2 \mu_{\mathrm{a}} \\ & - 2.57724 + 23.61842 \\ & \times [0.2539(b_3 v_{\mathrm{a,in}})^{1.67} + 0.18] \\ & \times [0.83666(b_4 H_{\mathrm{rz}})^{-0.5299} + 0.42] \\ & \times [43.0696(b_4 d_{\mathrm{d,eff}})^{0.7947} + 0.52]\}\end{aligned} \tag{24}$$

where the term on the left-hand-side is called the Merkel number and the 'b' coefficients represent combinations of g, ρ_{w}, σ_{w} and the values are given below.

$$\begin{aligned} & b_1 = 998/\rho_{\mathrm{w}}; \quad b_2 = 3.06 \times 10^{-6}\left[\rho_{\mathrm{w}}^4 g^9/\sigma_{\mathrm{w}}\right]^{0.25} \\ & b_3 = 73.298\left[g^5 \sigma_{\mathrm{w}}^3/\rho_{\mathrm{w}}^3\right]^{0.25}; \quad b_4 = 6.122[g\sigma_{\mathrm{w}}/\rho_{\mathrm{w}}]^{0.25}\end{aligned} \tag{25}$$

for the following restrictions:

$$0.927 \leqslant \rho_{\mathrm{a}} \leqslant 1.289, \mathrm{kg/m^3}; 1 \leqslant v_{\mathrm{a,in}} \leqslant 5\ \mathrm{m/s}$$
$$0.002 \leqslant d_{\mathrm{d}} \leqslant 0.008, \mathrm{m}; 1.717 \leqslant \mu_{\mathrm{a}} \leqslant 1.92 \times 10^{-5}, \mathrm{kg/m \cdot s}$$

It can be seen that the data necessary to solve the above equations includes the height of the rain zone, mass flow rate of water as well as the dry- and wet-bulb temperatures of the ambient air. In addition we note that Eq. (24) required the simultaneous solution of 18 equations and some constants like the diffusion coefficient of water vapor and mass flow rate of air. Besides calculating the effective drop diameter, the equation also calculates the mass transfer coefficient ($h_{\mathrm{D,rz}}$). The effective drop radius for the rain zone of the tower under consideration is calculated to be 6.284 mm, which is much larger than that of the spray zone.

2.4. Validation of packing model

Calculations regarding the packing or fill material of the cooling tower have been validated from the data provided by Simpson and Sherwood [13] as this offers the most comprehensive data in terms of experimental measurement as well as physical description of the tower used. Table 1 contains some experimental values that were compared. It can be seen that the experimental and predicted values are in excellent agreement and the errors associated with these predictions were found to be less than 1%. Also, there is an improvement in the calculated wet-bulb temperature of outlet air ($t_{\mathrm{wb,out}}$), as compared to the work of Khan

Table 1
Comparison of experimental and predicted values of outlet wet-bulb temperature

$t_{w,in}$ (°C)	$t_{w,out}$ (°C)	$t_{db,in}$ (°C)	$t_{wb,in}$ (°C)	$\dot{m}_a$ (kg/s)	$\dot{m}_{w,in}$ (kg/s)	$t^{exp}_{wb,out}$ (°C)	$t^{cal}_{wb,out}$ (°C)
31.22	23.88	37.05	21.11	1.158	0.754	26.05	*26.31*
41.44	26	34.11	21.11	1.158	0.754	30.72	*30.97*
28.72	24.22	29	21.11	1.187	1.259	26.17	*26.30*
34.5	26.22	30.5	21.11	1.187	1.259	29.94	*29.93*
38.78	29.33	35	26.67	1.265	1.008	32.89	*32.98*
38.78	29.33	35	26.67	1.250	1.008	32.89	*33.04*

and Zubair [10] that used an improved model to predict these parameters (without incorporating the spray and rain zones) but did not take into account the decrease in water flow rate due to evaporation. In light of the above discussion, the model used is providing reliable results for the fill zone.

2.5. Validation of spray and rain models

The spray and rain zone model, discussed earlier in Section 2.2, was validated separately using the data provided by Dreyer [19]. Dreyer indicated that the only good work available in the literature regarding determination of drop velocity was by Laws [21] that he used to compare with his own model. The results, shown in Fig. 5 at different heights, clearly demonstrate that the experimental and predicted values are in good agreement for the two drop-diameters tested. It is noted that Dreyer estimated the error in the experimental measurements of velocity to be less than 3% and we find that the current model predicts the drop velocities with an error of less than 2.5%.

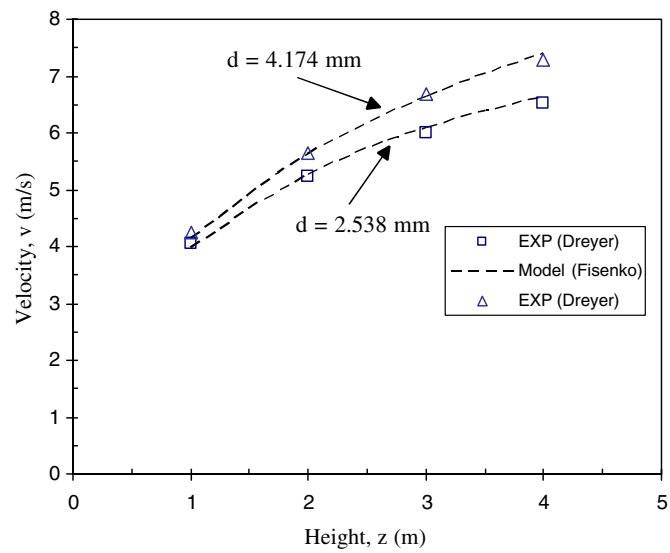

Fig. 5. Determination of effective drop radius for the spray zone.

2.6. The complete model

Fisenko et al. model [16] that we discussed above was coupled with the packing model presented earlier to study the combined performance of the spray zone and packing. It should be noted that the assumption of a negligible pressure drop is still employed. This combined model was verified using the experimental data provided by Simpson and Sherwood [13] that used a small-sized tower (refer to data presented in Table 2). It is to be noted that these results show an improvement in the prediction of the outlet air wet-bulb temperature as compared to the values in Table 1. As the outlet air is considered to be saturated, the dry-bulb temperatures are also compared and it was found that these predictions agree well with the experimental values with a maximum error of 3.6%. Furthermore, the complete model, i.e., spray zone plus fill and rain zones were coupled wherein a comparison of volume prediction was performed against the known volume of the tower used in the experiments of reference [13]. This was done in stages by first using the packing model only, then the spray zone plus packing models and finally, all three parts, i.e., spray zone plus packing plus rain zone were combined to ascertain the improvement in the calculated volume. The error in volume prediction for each stage, as detailed above, is found to be 6.5%, 3.15% and 2.65% which shows improvement as each zone is included.

Table 2
Comparison of experimental and predicted values of the outlet wet- and dry-bulb temperatures modeled with spray zone and packing coupled

$t_{w,in}$ (°C)	$t_{w,out}$ (°C)	$t_{db,in}$ (°C)	$t_{wb,in}$ (°C)	$\dot{m}_a$ (kg/s)	$\dot{m}_{w,in}$ (kg/s)	$t^{exp}_{wb,out}$ (°C)	$t^{cal}_{wb,out}$ (°C)	$t^{exp}_{db,out}$ (°C)	$t^{cal}_{db,out}$ (°C)
31.22	23.88	37.05	21.11	1.158	0.754	26.05	*26.19*	27.16	*26.19*
41.44	26	34.11	21.11	1.158	0.754	30.72	*30.76*	30.94	*30.76*
28.72	24.22	29	21.11	1.187	1.259	26.17	*26.22*	26.67	*26.22*
34.5	26.22	30.5	21.11	1.187	1.259	29.94	*29.80*	30.27	*29.80*
38.78	29.33	35	26.67	1.265	1.008	32.89	*32.86*	33.27	*32.86*
38.78	29.33	35	26.67	1.250	1.008	32.89	*32.92*	33.27	*32.92*

3. Concluding remarks

A complete cooling tower model is investigated by using engineering equation solver (EES) program, which is validated with the experimental data reported in the literature. It is demonstrated through a case study that it is important to include the spray and rain zones in analyses for a greater accuracy in design as well as rating calculations. This is mostly important in medium and large-size cooling towers. The results show a comparative improvement in the prediction of the wet-bulb temperature of the outlet air for the coupled three zone model. A comparison of the calculated volume is carried out against the known volume of the tower as each zone is included in the analysis. The error in volume prediction, with the addition of each zone, is calculated as 6.5%, 3.2% and 2.7%. The results indicate that these zones should be included in reliable analyses of cooling towers. Fouling is a major source of cooling tower performance deterioration and, therefore, a strategy to model fouling in cooling tower fills is also outlined to highlight the importance of fouling in rating calculations of cooling towers.

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References

[1] W.H. Walker, W.K. Lewis, W.H. McAdams, E.R. Gilliland, Principles of Chemical Engineering, third ed., McGraw-Hill Inc., New York, 1923.
[2] F. Merkel, Verdunstungshuhlung, Zeitschrift des Vereines Deutscher Ingenieure (VDI) 70 (1925) 123–128.
[3] R.L. Webb, A unified theoretical treatment for thermal analysis of cooling towers, evaporative condensers, and fluid coolers, ASHRAE Trans. 90 (2) (1984) 398–415.
[4] A.K.M. Mohiuddin, K. Kant, Knowledge base for the systematic design of wet cooling towers. Part I: Selection and tower characteristics, Int. J. Refrig. 19 (1) (1996) 43–51.
[5] A.K.M. Mohiuddin, K. Kant, Knowledge base for the systematic design of wet cooling towers. Part II: Fill and other design parameters, Int. J. Refrig. 19 (1) (1996) 52–60.
[6] H.T.A. El-Dessouky, A. Al-Haddad, F. Al-Juwayhel, A modified analysis of counter flow cooling towers, ASME J. Heat Transfer 119 (3) (1997) 617–626.
[7] D.G. Kroger, Air-Cooled Heat Exchangers and Cooling Towers, Begell House Inc., New York, 1999.
[8] D. Davis, Personal Correspondence, Cooling Tower Institute, (June 2003).
[9] T.H. Kuehn, J.W. Ramsey, J.L. Threlkeld, Thermal Environmental Engineering, third ed., Prentice-Hall Inc., New Jersey, 1998.
[10] J.R. Khan, S.M. Zubair, An improved design and rating analyses of counter flow wet cooling towers, ASME J. Heat Transfer 123 (4) (2001) 770–778.
[11] J.E. Braun, S.A. Klein, J.W. Mitchell, Effectiveness models for cooling towers and cooling coils, ASHRAE Trans. 95 (2) (1989) 164–174.
[12] ASHRAE, ASHRAE Equipment Guide, Chapter 3. American Society of Heating, Refrigerating, and Air-Conditioning Engineers Inc., Atlanta, GA, 1983.
[13] W.M. Simpson, T.K. Sherwood, Performance of small mechanical draft cooling towers, Refrig. Eng. 52 (6) (1946) 525–543, and 574–576.
[14] J.R. Khan, S.M. Zubair, A study of fouling and its effects on the performance of counter flow wet cooling towers, Proc. IMechE J. Process Mech. Eng. 218 (E1) (2004) 43–51.
[15] M. Michel, R. Noble, R.M. Gary. Research of fouling film fills. Paper # TP93-06. In: Proc. of Cooling Tower Institute Annual Meeting, New Orleans, LO, February 17–19, (1993).
[16] S.P. Fisenko, A.I. Petruchik, A.D. Solodukhin, Evaporative cooling of water in a natural draft cooling tower, Int. J. Heat Mass Transfer 45 (2002) 4683–4694.
[17] A.A. Zukauskas, High Performance Single-Phase Heat Exchangers, Hemisphere Publishing Corporation, New York, Washington, Philadelphia, London, 1989, Revised and Augmented Edition, p. 515.
[18] V.G. Levich, Physicochemical Hydrodynamics, Prentice-Hall, New Jersey, 1962.
[19] A.A. Dreyer, Modelling of Cooling Tower Splash Pack, Ph.D. Department of Mechanical Engineering, University of the Stellenbosch, South Africa, 1994.
[20] E. De Villiers, D.G. Kroger, Analysis of Heat, Mass and Momentum Transfer in the Rain Zone of Couterflow Cooling Towers, J. Eng. Gas Turbines Power 121 (4) (1999) 751–755.
[21] J.O. Laws, Measurement of the fall velocity of water drops and rain drops. In: Proc. of Trans. of the American Geophysical Union, Part III, 22nd Annual Meeting, (1941) 709–721.