

# Simulation of Heat and Mass Transfer in the Corrugated Packing of the Counter Flow Cooling Tower

Montri Pirunkaset\* and Santi Laksitanonta

## ABSTRACT

This paper presents the simulation results of the heat and mass transfer characteristics of corrugated packing in a counter-flow cooling tower. The numerical analysis has been partially validated by comparing the experimental data from a testing apparatus of a cooling tower under the same conditions at inlet dry bulb temperature of 32.3°C, inlet wet bulb temperature of 25.2°C and inlet water temperature of 40°C. Due to the complicated configuration of the packing surface and its installation, it was not possible to measure the temperature of the air and water in the intermediate section, but only the exit air temperature and exit water temperature. It was found that the errors between the calculated and measured temperature of the exit air were less than 2.5% for the exit dry bulb temperature, less than 2.2% for the exit wet bulb temperature and for the exit water were less than 1.3% (by testing at  $L/G=1.800, 2.171$  and  $2.820$ ). A comparison of the volumetric heat transfer coefficients derived from the experiment and the simulation using the Tchebycheff method found that all the errors were less than 4.8%. Consequently simulation can be used as a tool for studying the phenomenon of heat and mass transfer in corrugated packing.

**Key words:** simulation, corrugated packing, counter-flow cooling tower

## INTRODUCTION

To ensure high performance in cooling tower packing, it is essential that both the area of water film surface in contact with the air and the time of contact are as great as possible. In a cooling tower, the packing serves to increase the interfacial area, the fan creates the relative air-to-water velocity and the time of contact depends on the tower size. Therefore, all these three factors may be influenced by the cooling tower design and development. Practically, the outlet water temperature of the tower will approach the wet bulb temperature of the surrounding air. The magnitude of this approach depends on the design

of the cooling tower with regard to relative velocity and the contact time between the air and water, the surface area of packing, the water distribution over packing and the amount and size of the water droplets formed (Hill *et al.*, 1990).

Simulation is a tool for studying the operating variables of systems and their performance over wide ranges of conditions (Rieder and Busby, 1986). The models involved can always be constructed using basic principles and concepts to develop some equations to predict system behavior. The results of simulation can be generated using a computer program and proving the validity by a comparison with experimental data. It is obvious that classical simulation

Department of Mechanical Engineering, Faculty of Engineering, Kasetsart University, Bangkok 10900, Thailand.

\* Corresponding author, e-mail: fengmtp@ku.ac.th

techniques can be valuable for verifying the numerical methods used in engineering work.

The purpose of this study was to apply the basic principles of heat and mass transfer to develop the mathematical models to simulate the packing system. This study, predicted the operating variables, namely, dry bulb temperature, wet bulb temperature, water temperature, humidity ratio and the water flow rate at each divided cross section of the packing by using the fourth-order Runge-Kutta method. In addition, Merkel's method was used to calculate the volumetric heat transfer coefficient by the trapezoidal rule of integration (Mills, 1995). Finally, model equipment was set up in an experiment to compare with the mathematical models under the same initial conditions and given values of L/G. The Tchebycheff method was used to calculate the volumetric heat transfer coefficient from the experimental data (Cooling Tower Institute, 1982).

## MATERIALS AND METHODS

### Objectives

This study aimed to use computer-based models and the fourth-order Runge-Kutta technique to compare modelled results with experimental data. The following procedures were considered:

- 1) To study the behavior of heat and mass transfer between the air and a water film on the surface of packing in a counter-flow cooling tower;
- 2) To set up a mathematical model of the heat and mass transfer on the film surface of the packing;
- 3) To carry out the simulation by the fourth-order Runge-Kutta method for solving the operating variables at each divided cross section of packing under the given initial conditions;
- 4) To prepare the apparatus for testing mathematical models in accordance with CTI standards; and

- 5) To conclude and report on the results from the mathematical models and the on-site experimental data.

### Assumptions

Before deriving the relevant equations, the following assumptions were made:

- 1) Heat and mass transfer was a steady-state and steady-flow process;
- 2) The water film temperature was equal to the water temperature at the level of packing height under consideration;
- 3) The effect of water drift loss on the heat and mass transfer was negligible;
- 4) The areas of heat and mass transfer were the same;
- 5) The specific heat of dry air, water vapor and water were constants. For moist air, its specific heat was the sum of the specific heat of dry air and the product of the humidity ratio and the specific heat of water vapor;
- 6) At any intermediate horizontal cross section of the packing, temperatures of the air and water were uniformly distributed throughout that cross sectional area of packing; and
- 7) The Lewis number equalled one.

### Setup of model

To study the behavior of heat and mass transfer between the air and a water film on the surface of packing, the representative equations of the model were developed base on the concepts of heat and mass balance. It was considered important to understand the mechanism of this transport phenomenon beforehand. Therefore, energy and mass balance were the concepts used to determine the governing differential equations of this model (Mills, 1995).

Figure 1 shows the mass balance of water vapor in the air stream on the differential control volume with transfer area,  $\Delta A$  and height,  $\Delta z$ . For a differential control volume with the constant cross sectional area of packing,  $A_{fill}$  and transfer

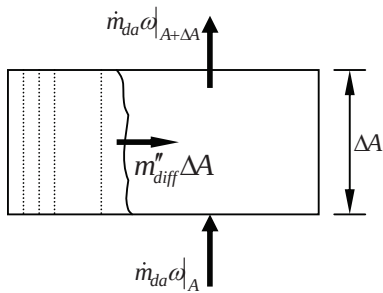


**Figure 1** Mass balance of water vapor in the air stream.

area,  $\Delta A = a_m A_{fill} \Delta z$ . Therefore, the mass conservation of water vapor in the upward moving air stream can be written as :

$$\dot{m}_{da} \omega|_{A+\Delta A} - \dot{m}_{da} \omega|_A = m''_{diff} \Delta A \quad (1)$$

Based on the low mass transfer rate theory, the rate of evaporation of water film on the packing can be approximated as  $m''_{diff} = G_m(\omega_s - \omega)/(1 + \omega_s)(1 + \omega)$ , and the Lewis number for the diluted water vapor and air mixture equals 1 (Mills, 1995). Dividing by  $\Delta z$  and letting  $\Delta z$  approach zero, the differential change of  $\omega$  can be written as:

$$\frac{d\omega}{dz} = \frac{A_{fill} a_m G_m (\omega_s - \omega)}{\dot{m}_{da} (1 + \omega_s)(1 + \omega)} \quad (2)$$

Figure 2 shows the mass balance of water vapor in the upward moving air stream and the downward moving water for the differential control volume that contains a transfer area,  $\Delta A$ . The mass conservation for water and vapor in the

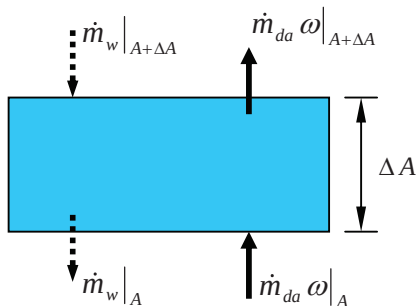


**Figure 2** Mass balance of water vapor in the upward moving air stream and the downward moving water.

air stream can be written as:

$$\dot{m}_{da} \omega|_{A+\Delta A} - \dot{m}_{da} \omega|_A + \dot{m}_w|_A - \dot{m}_w|_{A+\Delta A} = 0 \quad (3)$$

By rearranging and dividing Eq.(3) by  $\Delta z$  and letting  $\Delta z$  approach zero, the differential change in the water mass flow rate can be written as:

$$\frac{d\dot{m}_w}{dz} = \dot{m}_{da} \frac{d\omega}{dz} = \frac{A_{fill} a_m G_m (\omega_s - \omega)}{(1 + \omega_s)(1 + \omega)} \quad (4)$$

In Figure 3, the energy balance of the heat transfer from the water film and the enthalpy difference of the moist air stream in the differential control volume of packing can be considered as the transfer mechanism for the sensible heat transfer,  $q''_{conv} \Delta A$  and the latent heat due to mass transfer,  $m''_{diff} \Delta A h_{flm}$ . The energy conservation of the control volume can be written as:

$$\dot{m}_{da} h_a|_{A+\Delta A} - \dot{m}_{da} h_a|_A = q''_{conv} \Delta A + (m''_{diff} \Delta A) h_{flm} \quad (5)$$

Dividing Eq.(5) by  $\Delta z$  and letting  $\Delta z$  approach zero, the differential of air enthalpy can be written as:

$$\frac{dh_a}{dz} = \frac{A_{fill} a_m G_m}{\dot{m}_{da}} (q''_{conv} + m''_{diff} h_{flm}) \quad (6)$$

Substituting  $G_m = G_h$ , the convective heat transfer rate on the water film can be written as  $q''_{conv} = \alpha(T_f - T_a) = G_m C'_{pa} (T_f - T_a)$ . The moist air specific heat is  $C'_{pa} = (1.005 + 1.88\omega)/(1 + \omega)$ , (Mills, 1995). Hence, the air enthalpy can be determined in the form of  $h_a = 1.005T_a + \omega(2501 + 1.88T_a)$  and the differential equation of

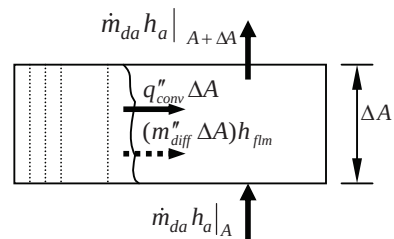


**Figure 3** Heat and mass transfer from the water film to the upward moving air stream.

$T_a$  can be expressed as:

$$\frac{dT_a}{dz} = \frac{A_{fill} a_m G_m}{\dot{m}_{da} (1.005 + 1.88\omega)} \left[ \frac{(1.005 + 1.88\omega)}{1 + \omega} (T_f - T_a) + \frac{(\omega_s - \omega) h_{flm}}{(1 + \omega_s)(1 + \omega)} - \frac{(2501 + 1.88T_a)(\omega_s - \omega)}{(1 + \omega_s)(1 + \omega)} \right] \quad (7)$$

In Figure 4, if the differential control volume contains a transfer area  $\Delta A$ , then the energy conservation for downward moving water and upward moving air passing through the control volume can be written as:

$$(\dot{m}_w h_w)|_{A+\Delta A} - (\dot{m}_w h_w)|_A = \dot{m}_{da} (h_a|_{A+\Delta A} - h_a|_A) \quad (8)$$

Dividing Eq.(8) by  $\Delta z$  and letting  $\Delta z$  approach zero, the differential change of water enthalpy can be expressed as:

$$\frac{dh_w}{dz} = \frac{A_{fill} a_m G_m}{\dot{m}_w} \left[ \frac{(1.005 + 1.88\omega)}{1 + \omega} (T_f - T_a) + \frac{(\omega_s - \omega) h_{flm}}{(1 + \omega_s)(1 + \omega)} - \frac{(\omega_s - \omega) h_w}{(1 + \omega_s)(1 + \omega)} \right] \quad (9)$$

To fit the representative equation for water enthalpy using steam table data with a range in water temperature from 25 to 49°C, the relation of  $h_w$  and  $T_f$  can be written as  $h_w = 4.179T_f + 0.4164$ . The differential equation of  $T_f$  can be obtained in the form of:

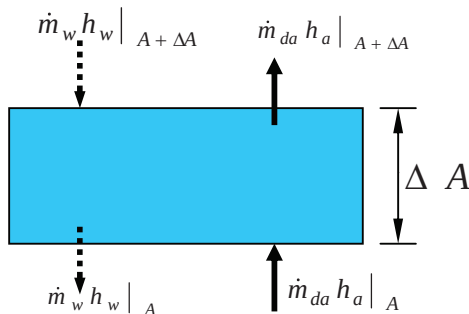


**Figure 4** Energy balance of downward moving water and upward moving air.

$$\frac{dT_f}{dz} = \frac{A_{fill} a_m G_m}{4.179 \dot{m}_w} \left[ \frac{(1.005 + 1.88\omega)}{1 + \omega} (T_f - T_a) + \frac{(\omega_s - \omega) h_{flm}}{(1 + \omega_s)(1 + \omega)} - \frac{(\omega_s - \omega)(4.179T_f + 0.4164)}{(1 + \omega_s)(1 + \omega)} \right] \quad (10)$$

Finally, the four governing differential equations for this model are completed, which are Eq.(2), (4), (7) and (10) respectively.

Before proceeding with the simulation of these four governing equations, we must define the various terms in these equations. The relationships required for solving the four governing equations are as follows (Mills, 1995):

$$h_{flm} = 2501 + 1.789T_f + 5.337 \times 10^{-5} [T_f + 273.15]^2 - 273.16^2 + 1.952 \times 10^{-7} [(T_f + 273.15)^3 - 273.16^3] - 5 \times 10^{-11} [T_f + 273.15]^4 - 273.16^4 \quad (11a)$$

For  $6^\circ C \leq T_{wb} \leq 28^\circ C$ ;

$$P_s = 0.0034T_{wb}^2 + 0.0108T_{wb} + 0.7645 \quad (11b)$$

For  $28^\circ C \leq T_{wb} \leq 60^\circ C$ ;

$$P_s = 0.0109T_{wb}^2 - 0.4754T_{wb} + 8.7967 \quad (11c)$$

$$P_{sf} = 0.0109T_f^2 - 0.4754T_f + 8.7967 \quad (11d)$$

For  $5^\circ C \leq T_{wb} \leq 60^\circ C$ ;  $h_{fg} = 2501.8 - 2.387T_{wb}$ ,

$$h_f = 4.182T_{wb} + 0.2293 \quad (11e)$$

$$\omega = \frac{\left[ 1.005(T_{wb} - T_a) + \left( \frac{0.622P_s}{101.325 - P_s} \right) h_{fg} \right]}{2501.5 + 1.806T_a - h_f} \quad (11f)$$

$$\omega_s = \frac{0.622P_{sf}}{101.325 - P_{sf}} \quad (11g)$$

$$C_{P,a} = C_{P,da} + \omega C_{P,v}, C'_{p,a} = \frac{1.005 + 1.88\omega}{1 + \omega} \quad (11h)$$

$$\rho_f = 2.116235 \times 10^{-10} (T_f + 273.15)^6 - 4.0524 \times 10^{-7} (T_f + 273.15)^5 + 3.221635 \times 10^{-4} (T_f + 273.15)^4 - 1.36077 \times 10^{-1} (T_f + 273.15)^3 + 32.1989 (T_f + 273.15)^2 - 4045.674 (T_f + 273.15) + 21181.62 \quad (11i)$$

$$\dot{m}_w = \frac{\rho_f (\dot{V}_w \times 10^{-3})}{60} \quad (11j)$$

$$v_a = \frac{(0.082T_a + 22.4135)(\frac{1}{29} + \frac{\omega}{18})}{1 + \omega} \quad (11k)$$

$$\dot{m}_a = \frac{\dot{V}_a}{v_a}, \quad \dot{m}_{da} = \frac{\dot{m}_a}{(1 + \omega)} \quad (11l)$$

The calculation of the wet bulb temperature at each divided cross section of the packing can be carried out using Eq.(11f) with the known values of dry bulb temperature and the humidity ratio of air passing through the cross section. The wet bulb temperature can be solved by the technique of trial and error.

The value of  $a_m G_m$  can be determined from the equation of packing and be considered in the dimensionless term of  $Ha_m G_m / L' = KaV / L$  and  $KaV / L = C(L/G)^{-n}$ . So,  $a_m G_m$  can be expressed as:

$$a_m G_m = \frac{CL}{H} \left( \frac{L}{G} \right)^{-n} \quad (11m)$$

A commercial cooling tower with 3 tons of refrigeration was used to teste the model according to CTI standards.

### Solution method

The model of four governing equations and four operating variables namely,  $T_a$ ,  $T_f$ ,  $\omega$ , and  $\dot{m}_w$  is given by  $f_1 = dT_a/dz$ ,  $f_2 = dT_f/dz$ ,  $f_3 = d\omega/dz$ , and  $f_4 = d\dot{m}_w/dz$  for the four governing equations. The recursion equations are given by relationship as:

$$T_{a,i+1} = T_{a,i} + \frac{a_1 + 2a_2 + 2a_3 + a_4}{6}$$

$$T_{f,i+1} = T_{f,i} + \frac{b_1 + 2b_2 + 2b_3 + b_4}{6}$$

$$\omega_{i+1} = \omega_i + \frac{c_1 + 2c_2 + 2c_3 + c_4}{6}$$

$$\dot{m}_{w,i+1} = \dot{m}_{w,i} + \frac{d_1 + 2d_2 + 2d_3 + d_4}{6}$$

where

$$a_1 = (\Delta z)f_1(T_{a,i}, T_{f,i}, \omega_i)$$

$$b_1 = (\Delta z)f_2(T_{a,i}, T_{f,i}, \omega_i, \dot{m}_{w,i})$$

$$c_1 = (\Delta z)f_3(\omega_i)$$

$$d_1 = (\Delta z)f_4(\omega_i)$$

$$a_2 = (\Delta z)f_1(T_{a,i} + \frac{a_1}{2}, T_{f,i} + \frac{b_1}{2}, \omega_i + \frac{c_1}{2})$$

$$b_2 = (\Delta z)f_2(T_{a,i} + \frac{a_1}{2}, T_{f,i} + \frac{b_1}{2}, \omega_i + \frac{c_1}{2}, \dot{m}_{w,i} + \frac{d_1}{2})$$

$$c_2 = (\Delta z)f_3(\omega_i + \frac{c_1}{2})$$

$$d_2 = (\Delta z)f_4(\omega_i + \frac{c_1}{2})$$

$$a_3 = (\Delta z)f_1(T_{a,i} + \frac{a_2}{2}, T_{f,i} + \frac{b_2}{2}, \omega_i + \frac{c_2}{2})$$

$$b_3 = (\Delta z)f_2(T_{a,i} + \frac{a_2}{2}, T_{f,i} + \frac{b_2}{2}, \omega_i + \frac{c_2}{2}, \dot{m}_{w,i} + \frac{d_2}{2})$$

$$c_3 = (\Delta z)f_3(\omega_i + \frac{c_2}{2})$$

$$d_3 = (\Delta z)f_4(\omega_i + \frac{c_2}{2})$$

$$a_4 = (\Delta z)f_1(T_{a,i} + a_3, T_{f,i} + b_3, \omega_i + c_3)$$

$$b_4 = (\Delta z)f_2(T_{a,i} + a_3, T_{f,i} + b_3, \omega_i + c_3, \dot{m}_{w,i} + d_3)$$

$$c_4 = (\Delta z)f_3(\omega_i + c_3)$$

$$d_4 = (\Delta z)f_4(\omega_i + c_3)$$

The conservation of heat and mass between the upward moving air and the downward moving water in the counter-flow cooling tower can be solved by the fourth-order Runge –Kutta method and simulated by developing a computer program. The initial values of the operating variables and the increment of packing height of the cooling tower were given before running the program.

The initial given values of the three operating variables (inlet air temperature,  $T_a = 32.3^\circ\text{C}$ ,  $T_{wb} = 25.2^\circ\text{C}$  and inlet water temperature  $T_f = 40^\circ\text{C}$ ) in the simulation of this mathematical model were used in conjunction with three values for the water-to-air-flow ratio of 1.800, 2.171 and 2.820. This simulation was carried out by the computer program and the operating variables at any intermediate horizontal cross section of the packing could be calculated.

## RESULTS AND DISCUSSION

The computer program developed using the previous governing equations, was used to calculate dry bulb temperatures ( $T_a$ , °C), wet bulb temperatures ( $T_{wb}$ , °C), water temperatures ( $T_f$ , °C), humidity ratio ( $\omega$ , kg/kg-dry air) and mass flow rate of water ( $\dot{m}_w$ , kg/s) at each divided cross section of the packing. These calculated results can be illustrated graphically to indicate the variance of air and water states throughout the packing as shown in Figures 7, 8, 9 and 10.

In Figures 7 and 8, it can be seen that as the water descends from the top to the bottom of the packing, its temperature will be decreased while maintaining inlet water temperature at 40°C. For a large air flow (corresponding low L/G), the effect of the water on the air temperature and humidity will be less. Therefore, the change in air temperature (the change of humidity ratio) for a low L/G will be less than that for a high L/G under the constant inlet air temperature of 32.3°C and the change of water temperature (change of

saturated humidity ratio) for a low L/G will be larger than that for a high L/G under the constant inlet water temperature of 40°C. For a low L/G, it was found that the temperature gradient ( $T_f - T_a$ ) increased with the vertical position of packing. Therefore, the sensible heat in the upper zone was larger than that in the lower zone of packing. For a high L/G, it was found that the temperature gradient decreased with the vertical position of packing. Therefore, the sensible heat in the upper zone was less than that in the lower zone of packing.

Furthermore, it was found that the humidity gradient ( $\omega_s - \omega$ ) decreased with the vertical position of packing for both a low and a high L/G. Therefore, the latent heat transfer in the upper zone was less than that in the lower zone of packing.

Figure 9 illustrates the principles of the heating and humidifying process for the upward-moving air. It can also be noted in this figure that the humidity ratio at each divided cross section of

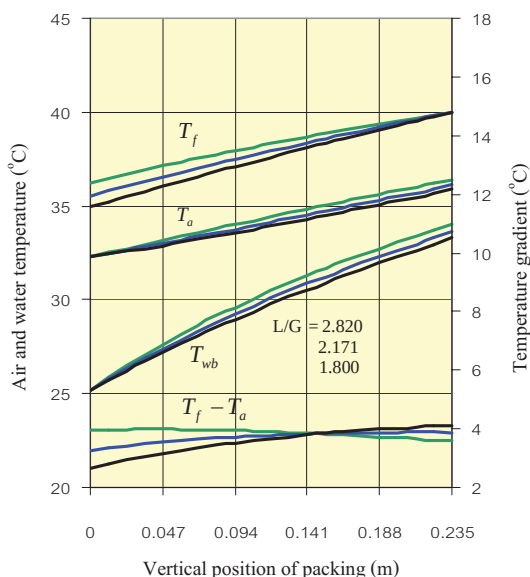


**Figure 7** The calculated air and water temperature at each divided cross section of packing in the cooling tower.

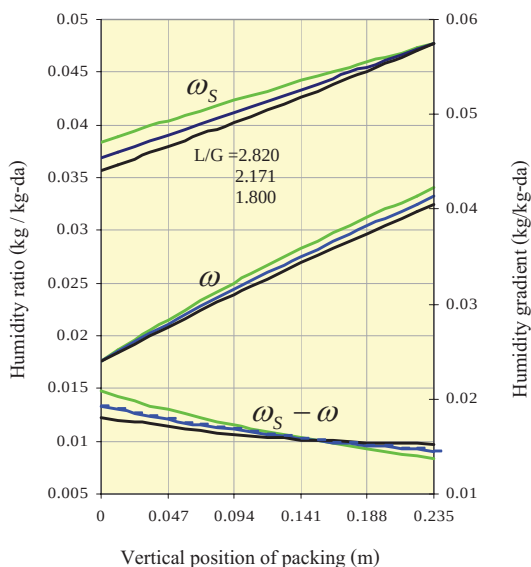


**Figure 8** The calculated air humidity ratio, saturated air humidity ratio, and humidity gradient at each divided cross section of packing in the cooling tower.

packing is increased with dry bulb temperature and is not dependent on the value of  $L/G$ . The process line appears as a theoretical curved line.

Figure 10 presents the relationship between the volumetric heat transfer coefficient,  $KaV/L$  and the vertical position for three values of the ratio  $L/G$  of 1.800, 2.171 and 2.820, respectively. Running the model under these conditions indicated that operating at a low  $L/G$  would enhance heat and mass transfer from the water film to the air stream more than for a high  $L/G$ . It can be seen that the  $KaV/L$ -values of this simulation model were close to those of the Tchebycheff method as shown in Table 1. The  $KaV/L$  differences of this simulation model and the Tchebycheff method were less than 4.8%, which were considered acceptable.

Table 2 shows a comparison of the calculated and measured exit air and water temperatures under the given initial values of the three operating variables (inlet air temperature at 32.3°C, inlet wet bulb temperature at 25.2°C and inlet water temperature at 40°C) and operating with the three different three values of the water to air flow ratio of 1.800, 2.171 and 2.820. It can be seen that the calculated air and water temperatures are approximately equal to those of the on-site experimental data. All errors in the comparison of calculated and measured

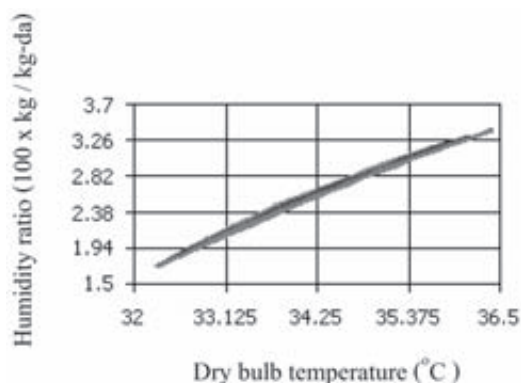


**Figure 9** The increase in the humidity ratio with dry bulb temperature.

temperatures of exit air and water were less than 2.5%, which was considered acceptable.

## CONCLUSIONS

The objective of this research was to develop an analytical method for evaluating the heat and mass transfer between the air and water in the corrugated packing of a small counter flow cooling tower. Due to the complicated configuration of packing and without accessible space for measuring air and water temperatures at intermediate sections, the numerical analysis has been partially validated. The simulated results for the exit air and water temperature compared favorably well with the measured values. However, it was not possible to compare simulated with measured air and water temperatures at the intermediate sections. Future work under controlled laboratory conditions to achieve full validation of the numerical analysis will require measuring the air and water temperature at the intermediate sections in the corrugated packing.

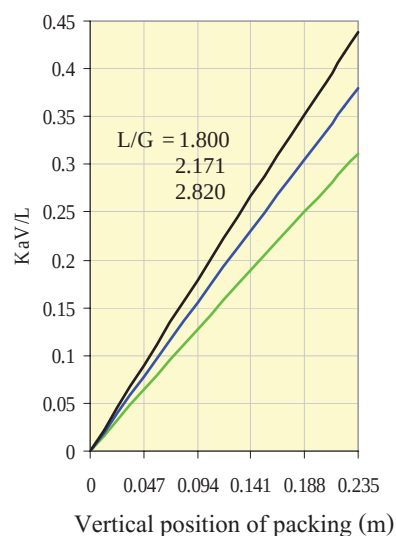


**Figure 10** The calculated volumetric heat transfer coefficient of packing in the counter-flow cooling tower.



|                |                                                                                      |                                                                                                                                                                                          |                                                                  |
|----------------|--------------------------------------------------------------------------------------|------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|------------------------------------------------------------------|
| CWT            | = Cold water temperature (°C)                                                        | $v_a$                                                                                                                                                                                    | = Specific volume of moist air (m <sup>3</sup> /kg)              |
| $G_m$          | = Mass transfer conductance (kg/m <sup>2</sup> .s)                                   | $V_a$                                                                                                                                                                                    | = Velocity of moist air (m/s)                                    |
| $G_h$          | = Heat transfer conductance (kg /m <sup>2</sup> .s)<br>and $G_h = \alpha / C'_{p,a}$ | WBT                                                                                                                                                                                      | = Inlet wet bulb temperature (°C)                                |
| $H$            | = Packing height (m)                                                                 | $\rho_a$                                                                                                                                                                                 | = Density of moist air (kg/ m <sup>3</sup> )                     |
| HWT            | = Hot water temperature (°C)                                                         | $\rho_f$                                                                                                                                                                                 | = Density of water (kg/ m <sup>3</sup> )                         |
| $h_a$          | = Enthalpy of moist air (kJ/kg-da)                                                   | $\omega$                                                                                                                                                                                 | = Humidity ratio of moist air (kg/kg-da)                         |
| $h_w$          | = Enthalpy of water stream (kJ /kg)                                                  | $\omega_s$                                                                                                                                                                               | = Humidity ratio of saturated air (kg/kg-da)                     |
| $h_{film}$     | = Enthalpy of water film (kJ /kg)                                                    | $\alpha$                                                                                                                                                                                 | = Convective heat transfer on film surface (W/m <sup>2</sup> .K) |
| $h_{fg}$       | = Latent heat of water vaporization (kJ/kg)                                          | $\Delta Z$                                                                                                                                                                               | = Increment of packing height (m)                                |
| $h_f$          | = Enthalpy of saturated liquid of water (kJ/kg)                                      | $\Delta A$                                                                                                                                                                               | = Mass or heat transfer area (m <sup>2</sup> )                   |
| $KaV/L$        | = Volumetric heat transfer coefficient of packing (dimensionless)                    | <b>Subscripts</b>                                                                                                                                                                        |                                                                  |
| $L/G$          | = Water to air flow ratio (kg-water flow/kg-dry air flow)                            | a                                                                                                                                                                                        | = Moist air                                                      |
| $L'$           | = Water loading (kg/m <sup>2</sup> .s)                                               | da                                                                                                                                                                                       | = Dry air                                                        |
| $\dot{m}_{da}$ | = Mass flow rate of dry air (kg/s)                                                   | <b>LITULATURE CITED</b>                                                                                                                                                                  |                                                                  |
| $\dot{m}_a$    | = Mass flow rate of moist air (kg/s)                                                 | Cooling Tower Institute. 1982. <b>Acceptance Test Code for Water Cooling Towers</b> . CTI Code ATC-105. 8 p.                                                                             |                                                                  |
| $m''_{diff}$   | = Diffusivity of mass transfer (kg/s.m <sup>2</sup> )                                | Hill, G.B., E.J. Pring and P.D. Osborn. 1990. <b>Cooling Towers Principles and Practice</b> . 3 <sup>rd</sup> ed. Butterworth-Heinemann Ltd., London. 191 p.                             |                                                                  |
| $P_s$          | = Saturated pressure of vapor in the atmospheric air (kPa)                           | Mills, A. F. 1995. <b>Heat and Mass Transfer</b> . Richard D. Irwin Inc., Chicago. 1239 p.                                                                                               |                                                                  |
| $P_{sf}$       | = Saturated pressure of vapor at the water film temperature ( kPa)                   | Rieder, G.W. and H.R. Busby. 1986. <b>Introductory Engineering Modeling Emphasizing Differential Models and Computer Simulations</b> . John Willey & Sons Company Inc., New York. 346 p. |                                                                  |
| $P$            | = Perimeter of cross-sectional area of packing (m)                                   |                                                                                                                                                                                          |                                                                  |
| $q''_{conv}$   | = Convective heat transfer rate (kW/m <sup>2</sup> )                                 |                                                                                                                                                                                          |                                                                  |
| $T_a$          | = Dry bulb temperature (°C)                                                          |                                                                                                                                                                                          |                                                                  |
| $T_{wb}$       | = Wet bulb temperature (°C)                                                          |                                                                                                                                                                                          |                                                                  |
| $T_f$          | = Water film temperature (°C)                                                        |                                                                                                                                                                                          |                                                                  |
| $\dot{V}_a$    | = Volume flow rate of air (m <sup>3</sup> /min)                                      |                                                                                                                                                                                          |                                                                  |
| $\dot{V}_w$    | = Volume flow rate of water (L/min)                                                  |                                                                                                                                                                                          |                                                                  |