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Xiao Li
University of Wisconsin Milwaukee

Yaoyu Li
University of Wisconsin Milwaukee

John E. Seem
Building Efficiency Research Group

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Xiao Li[1]*, Yaoyu Li[2], John E. Seem[3]

[1]Department of Mechanical Engineering, University of Wisconsin-Milwaukee, Milwaukee, WI, USA
xiaoli@uwm.edu
[2]Department of Mechanical Engineering, University of Wisconsin-Milwaukee, Milwaukee, WI, USA
yyli@uwm.edu
[3]Building Efficiency Research Group, Johnson Controls, Inc., Milwaukee, WI, USA
john.seem@gmail.com

*Corresponding Author

ABSTRACT

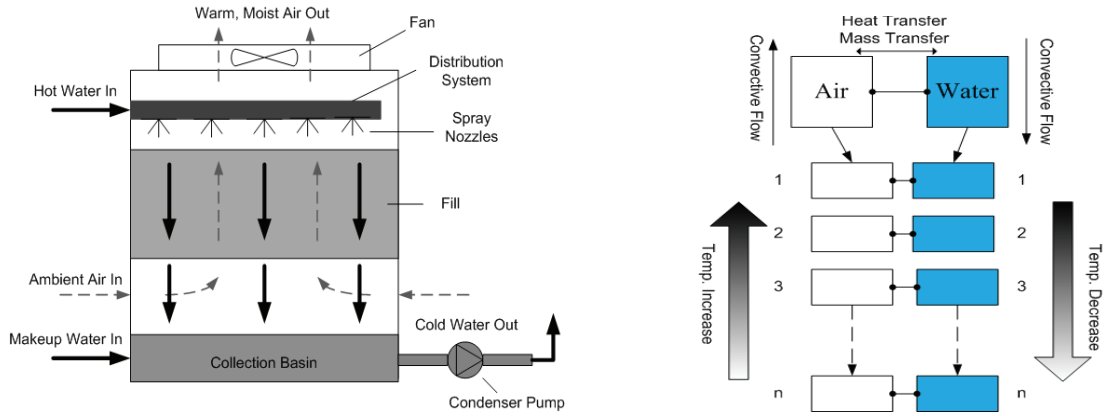
Cooling towers are important equipments for the HVAC systems in commercial buildings, rejecting the process heat generation to the atmosphere. Dynamic modeling of cooling towers is beneficial for control design and fault detection and diagnostics of the chilled-water systems. This paper proposes a simple and yet effective dynamic model for a typical mechanical draft counter-flow wet cooling tower. The finite volume method is applied to the one-dimensional heat and mass transfer analysis. With control volumes defined separately for the water and air sides, the dynamic equations are constructed with the mass and energy balances. The steady-state performance of the proposed model is evaluated with experimental data from literature. The transient behavior is also simulated under the changes of tower inlet conditions, with the performance to be evaluated in the future with field test data.

1. INTRODUCTION

Cooling towers are commonly used to reject heat from power generation units, water-cooled refrigeration and air conditioning for commercial buildings (ASHRAE, 2008). For cooling tower operation, heat rejection is accomplished via the heat and mass transfer occurring at the direct contact between hot water droplets and ambient air. Figure 1(a) shows the schematic of a mechanical draft counter-flow wet cooling tower that is typically used for chilled water system in commercial buildings. The cooling tower includes the fan, the distribution system, the spray nozzles, the fill (packing), the collection basin and the condenser pump. The warm water from the chiller is sprayed downward through the pressurized nozzles and then flows through the fill, and evaporation cooling occurs as the air flow is pulled upward by the tower fan through the fill. The fill is used to increase both the surface area and contact time between the air and water flows. For relatively dry air, the warm water can be cooled to a temperature below the ambient dry-bulb temperature. During the process, some water is evaporated into the air while some water is lost by misting effect (drift). Therefore, an external source of water, called makeup water, is needed to compensate for the water loss due to evaporation and drift. The condenser pump drives the water back to the chiller.

A lot of work has been done for modeling cooling towers in the past century. Walker *et al.* (1923) proposed a basic theory of cooling tower operation. Merkel (1925) developed the first practical theory including the differential equations of heat and mass transfer, which has been well received as the basis for most work on cooling tower modeling and analysis (Khan *et al.*, 2003; Elsarrag, 2006; Qureshi and Zubair, 2006; ASHRAE, 2008; Lucas *et al.*, 2009). In Merkel's model, in order to simplify the analysis, the water loss of evaporation is neglected, and the Lewis relation is assumed as unity. These assumptions may cause Merkel's model to underestimates the effective tower volume by 5-15% (Sutherland, 1983). Jaber and Webb (1989) introduced the effectiveness-NTU (number of transfer units) design method for counter-flow cooling towers using Merkel's simplified theory. Sutherland (1983) gave a more rigorous analysis of cooling tower including water loss by evaporation. Braun (1988) and Braun *et al.* (1989) gave a detailed analysis and developed effectiveness models for cooling tower by assuming a linearized air saturation enthalpy and a modified definition of effectiveness using the constant saturation specific heat C_s. A modeling framework was developed for estimating the water loss and then validated over a wide range of operating conditions. Bernier (1994,1995) presented a one-dimensional (1D) analysis of an idealized spray-type tower, which

showed how the cooling tower performance is affected by the fill height, the water retention time, and the air and water mass flow rates. Fisenko *et al.* (2004) developed a mathematical model of mechanical draft cooling tower, and took into account the radii distribution of the water droplets. Wetter (2009) proposed a cooling tower model by using static mapping to the performance curve of a York cooling tower. Most existing models for cooling towers are steady-state or effectiveness models. Dynamic modeling of cooling tower is needed for control design and fault detection and diagnostics, and to the authors' best knowledge, no work has been reported on the dynamic model.



(a) (b)

Figure 1: (a) Schematic Diagram for Mechanical Draft Counter-Flow Wet Cooling Tower and (b) Illustration of Control Volumes for Cooling Tower Modeling

This study presents a dynamic model for a mechanical draft counter-flow wet cooling tower based on 1D heat and mass balance dynamic equations. The assumptions from Braun's work (Braun *et al.*, 1989) were followed to simplify the analysis. Heat and mass transfer is only considered in a direction normal to the flows, while the heat and mass transfer through the tower walls to the environment is neglected. The mass fraction of water vapor in the moist air is approximated equal to the humidity ratio. Several distinctive treatments are carried out in this study. First, the mutative water and air specific heats are used to relax the constraints, with the help of the property calculation capability available in the TIL Media Library (Richter, 2008). Second, instead of considering the Lewis relation as unity, the formulation in Bosnjakovic (1965) is followed. Thirdly, the finite volume (FV) method is applied in order to achieve more robust performance for start-up and all load-change transients (Bendapudi *et al.*, 2008). The control volumes of water and moist air are defined separately, with opposite flow directions. Dynamic mass and energy balances are evaluated for each control volume, and the heat and mass transfer are considered between each pair of water and moist-air control volumes. The proposed model includes both sensible and latent heat transfer effects on the tower performance. The balance between the water loss and the humidity increase in the moist air is reinforced through all the control volumes. The water loss is determined by the mass transfer coefficient based on the geometry and performance map of specific cooling tower.

In this study, the simulation model is implemented in Modelica with Dymola Version 6.1 (developed by Dynasim (Dynasim, 2007)) and the TLK/IfT Library (TIL) (Richter, 2008) developed by TLK-Thermo. Modelica is an acausal equation based object-oriented language for multi-physical modeling (Modelica, 2007), which has demonstrated its advantages in various engineering applications, especially for large, complex, and hybrid systems. Modeling of thermofluid system components can be directly represented by differential algebraic equations (DAE). Dymola is an integrated development environment for Modelica based modeling. It has a Modelica translator to perform symbolic transformations, index reduction algorithms for reducing the degrees-of-freedom caused by constraints, and can better handle algebraic loops. TIL is a Modelica library developed by TLK-Thermo GmbH (Richter, 2008) for steady-state and transient simulation of thermofluid systems. The library featured a simple inheritance structure that makes it easy to extend to a variety of applications. In addition to the primary evaporation cooling process, other related components, including fan, pump and collection basin, are also modeled.

The reminder of this paper is organized as follows. Section 2 presents the dynamic model of the cooling tower; Section 3 presents the models for fan, pump and collection basin; The developed model is evaluated with the experimental values from Simpson and Sherwood (1946) in Section 4, in terms of the steady-state value of the outlet

water temperature. The dynamic behavior is also simulated under the change of inlet condition; the performance will be evaluated in the future with field test data. The paper is concluded in Section 5.

2. DYNAMIC COOLING TOWER MODEL

2.1 Cooling Tower Dynamic Model

The evaporation cooling process of the mechanical draft counter-flow cooling tower in Fig. 1(a) is modeled with the FV method. As shown in Fig. 1(b), the two kinds of control volumes, for water and moist air, are shown respectively. The water and moist air flow are in opposite directions. The modeling process follows the similar assumptions as in Braun *et al.* (1989):

1) Heat and mass transfer in the direction normal to the flows only.
2) Negligible heat and mass transfer through the tower walls to the environment.
3) Negligible heat transfer from the tower fans to the air or water streams.
4) The mass fraction of water vapor in the moist air is approximately equal to the humidity ratio.
5) Uniform temperature throughout the water stream at any cross section.
6) Uniform cross-sectional area of the tower.

Dynamic mass and energy balances are established for both water- and air-sides, with the control volumes shown in Fig. 2(a) and Fig. 2(b), and the heat and mass transfer are considered between each pair of the water and moist air control volumes. The transient mass and energy storage is considered at the water side but neglected at the air side.

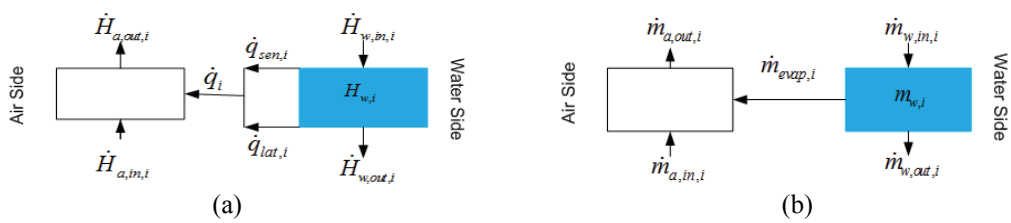


Figure 2: (a) Energy Balance between Neighbored Water and Air Control Volumes and (b) Mass Balance between Neighbored Water and Air Control Volumes

For the ith water-side control volume in Fig. 2(a), the energy balance leads to

$$\Delta H_{w,i} = \dot{H}_{w,in,i} - \dot{H}_{w,out,i} - \dot{q}_i \tag{1}$$

Where $\Delta H_{w,i}$ is the enthalpy change for the cell, $H_{w,in,i}$ is the inlet water enthalpy, $H_{w,out,i}$ is the outlet water enthalpy, $\dot{q}_i$ is the heat flow transferred to the neighbored (also the ith) moist-air cell which include both the sensible heat flow and the latent heat flow due to evaporation. Equation (1) can be expanded into

$$m_{w,i} \cdot c_{p,w,i} \cdot \frac{dT_{w,i}}{dt} = \dot{m}_{w,in,i}\left(h_{w,in,i} - h_{w,i}\right) - \dot{m}_{w,out,i}\left(h_{w,out,i} - h_{w,i}\right) - \dot{q}_i \tag{2}$$

where $m_{w,i}$ is the mass of water stored in the cell, $c_{p,w,i}$ is the specific heat of water (which can be determined by the local water temperature $T_{w,i}$), $\dot{m}_{w,in,i}$ and $\dot{m}_{w,out,i}$ are the mass flow rates for the inlet and outlet water flow, respectively, $h_{w,in,i}$ and $h_{w,out,i}$ are the specific enthalpy of the inlet and outlet water flow, respectively, and $h_{w,i}$ is the specific enthalpy of water in the cell.

For the mass balance of the same water-side control volume as shown in Fig. 2(b), the volume of cell V_{cell} is considered constant, while water density $\rho_{w,i}$ may change with evaporation and temperature change in the cell. The following differential equation may be written

$$\frac{dm_{w,i}}{dt} = \dot{m}_{w,in,i} - \dot{m}_{w,out,i} - \dot{m}_{evap,i} \tag{3}$$

$$m_{w,i} = V_{effective} \cdot \rho_{w,i} \tag{4}$$

where $\dot{m}_{evap,i}$ is the vapor mass transfer flow rate into the moist air. $V_{effective}$ is the water droplet volume in the cell, the ratio of water droplet volume per unit volume of the tower is around the level of 0.001(Bernier, 1994). Substituting Eq. (4) into Eq. (3) yields

$$V_{effective} \cdot \frac{d\rho_{w,i}}{dt} = \dot{m}_{w,in,i} - \dot{m}_{w,out,i} - \dot{m}_{evap,i} \tag{5}$$

The time derivative of density can be formulated as (Richter, 2008)

$$\frac{d\rho}{dt} = \left(\frac{\partial\rho}{\partial P}\right)_h \frac{dP}{dt} + \left(\frac{\partial\rho}{\partial h}\right)_P \frac{dh}{dt} \tag{6}$$

where pressure P, specific enthalpy h, and density ρ are selected as the three differential variables for property calculation in each control volume. As the cell pressure is approximately constant for the cooling tower operation, Eq. (6) can be simplified as

$$\frac{d\rho}{dt} = -\frac{\beta\rho}{c_{pw}}\frac{dh}{dt} = -\beta\rho\frac{dT}{dt} \tag{7}$$

where $\beta = -\frac{1}{\rho}\left(\frac{\partial\rho}{\partial T}\right)_P$ is the isobaric coefficient of expansion and c_{pw} is the specific heat capacity at constant pressure. Substituting Eq. (7) into Eq. (5) leads to the mass balance of the ith water cell,

$$\dot{m}_{w,in,i} - \dot{m}_{w,out,i} - \dot{m}_{evap,i} = -V_{effective}\beta_{w,i}\rho_{w,i}\frac{dT_{w,i}}{dt} \tag{8}$$

where $\beta_{w,i}$ and $\rho_{w,i}$ can be determined by the local water temperature.

On the air side, the transient mass and energy storage is neglected. The steady-state relations were derived following Braun's detailed analysis model (Braun *et al.*, 1989). The energy balance results in

$$\dot{H}_{a,in,i} - \dot{H}_{a,out,i} + \dot{q}_i = 0 \tag{9}$$

$$\dot{q}_i = \dot{q}_{sen,i} + \dot{q}_{lat,i} \tag{10}$$

The sensible and latent heat flow rates can be determined by

$$\dot{q}_{sen,i} = h_{C,i}A_V V_{cell}\left(T_{w,i} - T_{a,i}\right) \tag{11}$$

$$\dot{q}_{lat,i} = h_{f,g,i}\cdot\dot{m}_{evap,i} = h_{f,g,i}\cdot h_{D,i}A_V V_{cell}\left(\omega_{s,w,i} - \omega_{a,i}\right) \tag{12}$$

where $h_{C,i}$ is the local heat transfer coefficient, A_V is the surface area of water droplets per volume of cooling tower, $T_{a,i}$ is the local air temperature, $h_{f,g,i}$ is the latent heat of vaporization depending on the local water temperature. $h_{D,i}$ is the local mass transfer coefficient, $\omega_{s,w,i}$ is the saturated air humidity ratio at the local water temperature, and $\omega_{a,i}$ is the local humidity ratio of moist air.

The mass transfer coefficient can be derived by using the overall NTU for mass transfer, i.e.

$$NTU = \frac{h_D A_V V_T}{\dot{m}_{a,in}} \tag{13}$$

where V_T is the total tower volume and $\dot{m}_{a,in}$ is the air inlet flow rate of the cooling tower. The mass transfer coefficient can thus be determined with

$$h_D A_V = \frac{NTU\cdot\dot{m}_{a,in}}{V_T} \tag{14}$$

which varies with the tower geometry, NTU and air inlet flow rate. The heat transfer coefficient is determined by

$$h_{C,i}A_V = \frac{Le_f\cdot NTU\cdot c_{pm,i}\cdot\dot{m}_{a,in}}{V_T} \tag{15}$$

where the Lewis relation $Le_f = h_C/(h_D c_{pm,i})$ and the local specific heat of moist air $c_{pm,i}$ is determined by

$$c_{pm,i} = c_{pa,i} + \omega_{a,i}c_{pv,i} \tag{16}$$

where $c_{pa,i}$ is the local specific heat of dry air and $c_{pv,i}$ is the local specific heat of water vapor (Braun, 1988). $h_{C,i}$ may change due to the local value of Le_f and $c_{pm,i}$.

The NTU can be determined from experimental data using empirical equations of thermal properties (ASHRAE, 1983; Braun *et al.*, 1989; Kröger, 2004). Kloppers and Kröger (2005) stated that the variation of the Lewis relation has little influence on the water outlet temperature and heat rejected from the cooling tower for very humid ambient air; while for dry conditions, the variation of the Lewis relation can lead to significantly different results. It was also

suggested the equation by Bosnjakovic (1965) should be used, and a numerical value of 0.92 be preferred when the fill performance test data are insufficient to accurately predict the Lewis relation of a particular fill.

3. MODEL OF RELATED COMPONENTS

3.1 Fan

The related fan model follows the model of TIL.MoistAirComponents.Fans.Fan2ndOrder in the TIL Library. From the fan affinity law, the volume flow rate, pressure increase and rotational speed are related by

$$Q_{fan,affinity,0} = Q_{fan,0} \cdot \frac{n_{fan}}{n_{fan,0}} \tag{20}$$

$$\Delta p_{fan,affinity,0} = \Delta p_{fan,0} \cdot \left(\frac{n_{fan}}{n_{fan,0}} \right)^2 \tag{21}$$

where $n_{fan,0}$ is the nominal speed, n_{fan} is the rotational speed, $Q_{fan,0}$ is the volume flow rate for zero pressure increase, and $Q_{fan,affinity,0}$ is the volume flow rate for zero pressure increase following the fan affinity law. $\Delta p_{fan,0}$ is the pressure increase at volume flow rate $Q_{fan,0}$=0, $\Delta p_{fan,affinity,0}$ is the pressure increase at $Q_{fan,0}$=0 following the fan affinity law (Richter, 2008). The actual pressure increase can be determined with

$$\Delta p_{fan} = \Delta p_{fan,affinity,0} \cdot \left(1 - \frac{Q_{fan}}{Q_{fan,affinity,0}} \right)^2 \tag{22}$$

Then the fan power can be given by

$$\dot{W}_{fan} = \frac{\Delta p_{fan} \cdot Q_{fan}}{\eta_{fan} \eta_{fan,m}} \tag{23}$$

where η_{fan} is the fan efficiency and $\eta_{fan,m}$ is the motor efficiency. η_{fan} can be determined by a polynomial regression of the manufacture's data (Clark, 1985).

3.2 Pump

The pump model aims to predict the power consumption by pump. The modeling followed TIL.LiquidComponents. Pumps.Pump2ndOrder in the TIL Library (Richter, 2008), with the pump affinity law defined similarly to that for the fan modeling. The equations are listed as follow:

$$Q_{pump,affinity,0} = Q_{pump,0} \cdot \frac{n_{pump}}{n_{pump,0}} \tag{24a}$$

$$\Delta p_{pump,affinity,0} = \Delta p_{pump,0} \cdot \left(\frac{n_{pump}}{n_{pump,0}} \right)^2 \tag{24b}$$

where $n_{pump,0}$ is the nominal speed, n_{pump} is the rotational speed, $Q_{pump,0}$ is the volume flow rate for zero pressure increase, and $Q_{pump,affinity,0}$ is the volume flow rate for zero pressure increase following the fan affinity law. $\Delta p_{pump,0}$ is the pressure increase at volume flow rate $Q_{pump,0}$=0, $\Delta p_{pump,affinity,0}$ is the pressure increase at $Q_{pump,0}$=0 following the fan affinity law (Richter, 2008). The actual pressure increase can be determined with

$$\Delta p_{pump} = \Delta p_{pump,affinity,0} \cdot \left(1 - \frac{Q_{pump}}{Q_{pump,affinity,0}} \right)^2 \tag{24c}$$

The power loss and the shaft power of the pump can then be determined by (Richter, 2008)

$$\dot{W}_{pump,loss,0} = \left(\frac{1}{\eta_{pump,0}} - 1 \right) \cdot \Delta p_{pump,0} \cdot Q_{pump,0} \cdot \frac{2}{3^{1.5}} \tag{24d}$$

$$\dot{W}_{pump,loss} = \dot{W}_{pump,loss,0} \left(\frac{n_{pump}}{n_{pump,0}} \right)^{e_{pump,loss}} \tag{24e}$$

$$\dot{W}_{pump,shaft} = \dot{W}_{pump,loss} + \Delta p_{pump} \cdot Q_{pump} \tag{24f}$$

where $\dot{W}_{pump,loss,0}$ is the power loss at nominal speed, $\eta_{pump,0}$ is the nominal efficiency, $\dot{W}_{pump,loss}$ is the actual power loss at rotational speed n_{pump}, and $e_{pump,loss}$ is the exponent for power loss calculation, which is a constant.

The mass and energy balances for the pump are

$$\dot{m}_{in} - \dot{m}_{out} = -\beta \rho_w V_{pump} \frac{dT_w}{dt} \tag{25a}$$

$$c_{pw} \frac{dT_w}{dt} = \frac{\dot{m}_{in}\left(h_{in} - h_{pump}\right) - \dot{m}_{out}\left(h_{out} - h_{pump}\right) + \dot{W}_{shaft}}{\rho_w V_{pump}} \tag{25b}$$

where $\dot{m}_{in}$ are $\dot{m}_{out}$ are the water inlet and outlet flow rates, respectively. V_{pump} is the volume of water in the pump, which is generally treated as a constant. h_{in} and h_{out} are the specific enthalpies for the inlet and outlet water, respectively. h_{pump} is the specific enthalpy of water in the pump, and $\dot{W}_{shaft}$ is the pump shaft power.

3.3 Collection Basin

The balance equations of collection basin are derived as

$$\dot{m}_{in} - \dot{m}_{out} + \dot{m}_{makeup} = -\beta \rho_w V_{cb} \frac{dT_w}{dt} \tag{26}$$

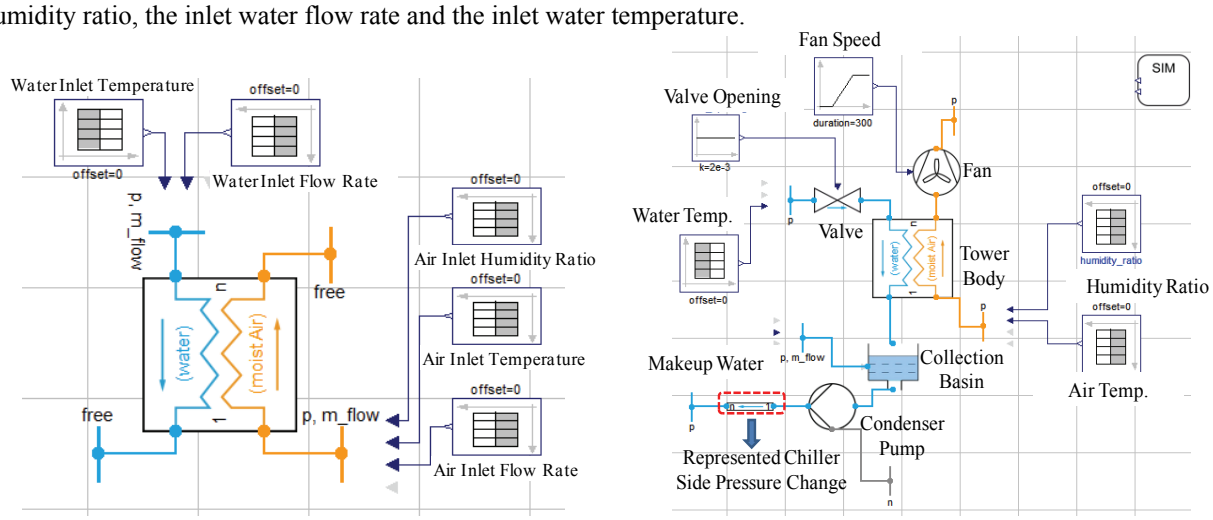
$$c_{pw} \frac{dT_w}{dt} = \frac{\dot{m}_{in}\left(h_{in} - h_{cb}\right) - \dot{m}_{out}\left(h_{out} - h_{cb}\right) + \dot{m}_{makeup}\left(h_{makeup} - h_{cb}\right)}{\rho_w V_{cb}} \tag{27}$$

where $\dot{m}_{makeup}$ is the water flow rate from some source of make-up water. h_{cb} is the specific enthalpy of water in the collection basin. The volume of water V_{cb} in the collection basin is assumed to be constant for now. So the flow of water make-up is equal to the total water loss from evaporation.

4. SIMULATION STUDY

4.1 Steady-State Simulation

Simulation study was conducted to study the behavior and performance of the cooling tower. Figure 3(a) shows the Dymola layout of the model of evaporation cooling process for the cooling tower, developed with TIL. There are five inputs in the cooling tower model, i.e. the inlet moist-air flow rate, inlet moist-air temperature, inlet moist-air humidity ratio, the inlet water flow rate and the inlet water temperature.



(a) Evaporation Cooling Process of Cooling Tower (b) Whole System of Cooling Tower

Figure 3: Dymola Layout for the Cooling Tower Simulation Model

The steady-state performance of the proposed model is evaluated with the experimental data from Simpson and Sherwood (1946), with five cases compared in Table 1. $T_{w,out,cal}$ is the model predicted water outlet temperature, respectively. Figure 4 plots all the experimental data of the outlet water temperature and those predicted with the proposed model. The prediction error has the mean of 0.344K and the standard derivation of the 0.428K, which is comparable to the results in Braun *et al.* (1989). The Le_f calculated by the equation from Bosnjakovic (1965) is around 0.915, which is compatible with the recommended numerical value of 0.92 in Kloppers and Kröger (2005).

Table 1: Comparison of Model Prediction and Experimental Data

Case	$T_{w,in}$ (°C)	$T_{w,out}$ (°C)	$T_{db,in}$ (°C)	$T_{wb,in}$ (°C)	$T_{db,out}$ (°C)	$\dot{m}_{a,in}$ (kg/s)	$\dot{m}_{w,in}$ (kg/s)	$T_{w,out,cal}$ (°C)
1	33.22	25.50	28.83	21.11	28.44	1.1871	1.0088	25.46
2	34.39	29.0	31.78	26.67	31.22	1.1653	1.0088	28.78
3	43.61	27.89	35.0	23.89	32.78	1.1584	0.7548	28.12
4	38.78	29.33	35.0	26.67	33.28	1.2653	1.0088	29.87
5	43.06	29.72	35.72	26.67	33.89	1.1566	0.7548	29.94

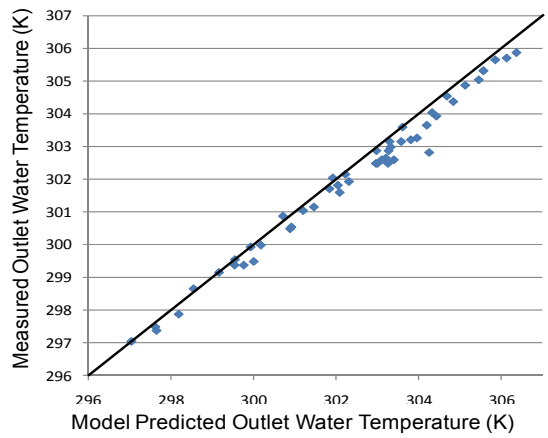
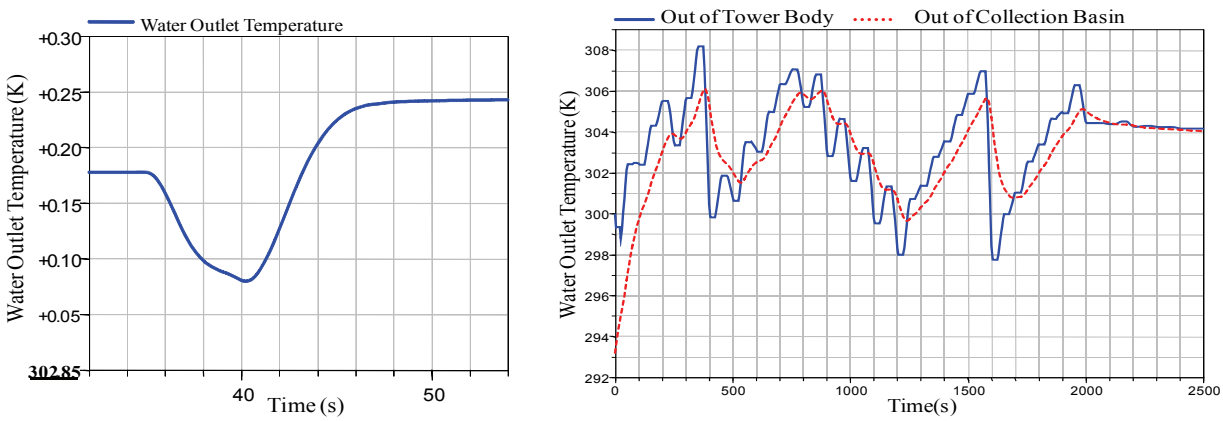


Figure 4: Comparison for Outlet Water Temperature between Model Prediction and Measured Data

4.2 Transient Simulation

The transient performance of the proposed model is evaluated via benchmarking against the case studies in Bernier (1995). The profile of outlet water temperature is observed under the changes of the inlet water temperature, the inlet air temperature, the inlet air humidity ratio, the inlet water and the air flow rate. Figure 5(a) shows the transient performance from case 4 to case 5 in Table 1. The water inlet temperature and the air inlet temperature increase, which may cause an increase of the water outlet temperature. Meanwhile, the increase of the difference between the dry-bulb temperature and the wet-bulb temperature indicates a decrease of the relative humidity of the inlet air, which may cause a decrease of the water outlet temperature. Therefore, the transient behavior demonstrates a significant undershoot instead of a smooth transient.



(a) Stand-alone Cooling Tower (b) Comparison between Tower Body and Collection Basin Outputs

Figure 5: Transient Performance of Water Outlet Temperature for Cooling Tower Simulation

By the purpose of further control design for combined system of chiller and cooling tower, a whole system of cooling tower with connections of tower body, collection basin, valve, fan, condenser pump, and a makeup water source is modeled as shown in Fig. 3(b), Fig. 5(b) shows an additional transient on the water outlet temperature caused by collection basin. In near future, the proposed whole system model of cooling tower will also be evaluated with field test data.

5. CONCLUSIONS

This paper presents a simple and yet effective dynamic model for a typical mechanical draft counter-flow cooling tower. The finite volume method is applied to the 1D heat and mass transfer analysis based on the assumptions given by Braun's earlier work. With control volumes defined separately for the water and air sides, respectively, the dynamic equations are established with the mass and energy balances. The steady-state performance of the proposed model is evaluated with the experimental data from Simpson and Sherwood (1946). The performance seems comparable with the existing steady-state models for the cooling tower. The transient behavior is also simulated under the changes of tower inlet conditions, with the performance to be evaluated in the future with field test data.

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