DYNAMIC MODELING OF MECHANICAL DRAFT COUNTER-FLOW WET COOLING TOWER WITH MODELICA

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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 steady-state performance of the proposed model is evaluated with experimental data from literature. The transient behavior is simulated under the changes of tower inlet conditions, with the performance to be evaluated in the future with field test data.

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 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 valve, 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, known as "makeup water", is supplied to replenish the basin against the water loss due to evaporation and drift. The condenser pump drives the water back to the chiller.

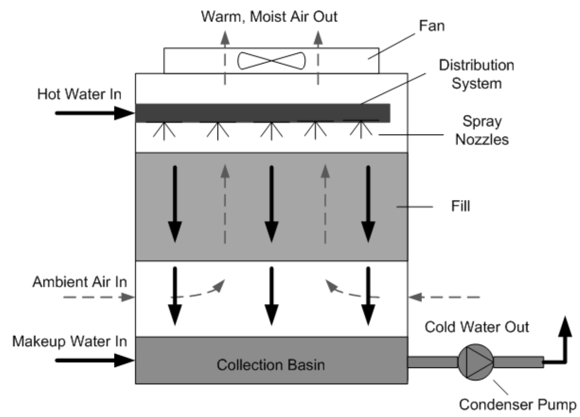

Figure 1: Schematic Diagram for Mechanical Draft Counter-Flow Wet Cooling Tower

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, as a simplified 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 (Braun 1988; 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) built a static cooling tower model in Modelica (Modelica Association 2010) with an empirical relationship fitted with the performance data 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 developing dynamic model for cooling towers yet.

This study presents a dynamic model for a mechanical draft counter-flow wet cooling tower based on the dynamic equations for 1D heat and mass balance. The assumptions from Braun's work (Braun et al. 1989) were followed to simplify the analysis. Heat and mass transfer is considered only in the direction normal to the water flow, 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 the 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 (Dassault Systemes 2010) and the TLK/IfT Library (TIL) (Richter 2008) developed by TLK-Thermo, GmbH. Modelica is an acausal, equation-based, object-oriented language for multi-physical modeling (Modelica Association 2010), which has demonstrated its advantages in various engineering applications, especially for large, complex, and hybrid systems. The thermofluid system models can be constructed directly by differential algebraic equations (DAE). Dymola is an integrated development environment for Modelica based modeling, featured with a Modelica translator to perform symbolic transformations and index reduction algorithms for reducing the degrees-of-freedom caused by constraints. Thus it can better handle algebraic loops. TIL is a Modelica library (Richter 2008) for steady-state and transient simulation of thermofluid systems. The library featured a simple and efficient inheritance structure that makes it easy to develop new component models. In addition to the primary evaporation cooling process, other related components, including fan, pump and collection basin, are also modeled in this study.

DYNAMIC COOLING TOWER MODEL

Cooling Tower Dynamic Model

The evaporation cooling process of the mechanical draft counter-flow cooling tower in Fig. 1 is modeled with the FV method. Two kinds of control volumes are shown in Fig. 2, for water and moist air, 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 water/air flow 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. 3 and Fig. 4, 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.

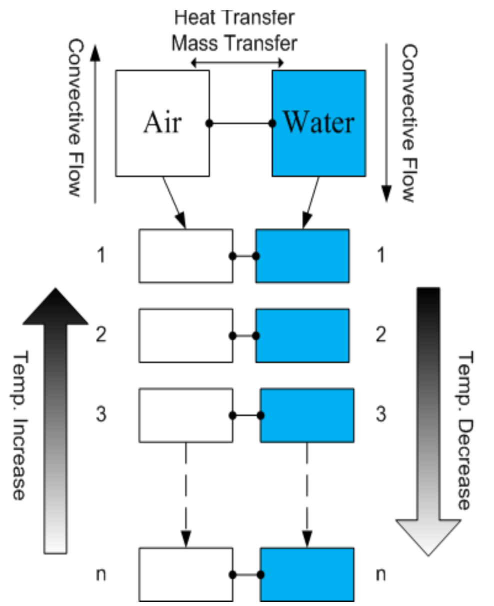

Figure 2: Illustration of Control Volumes for Cooling Tower Modeling

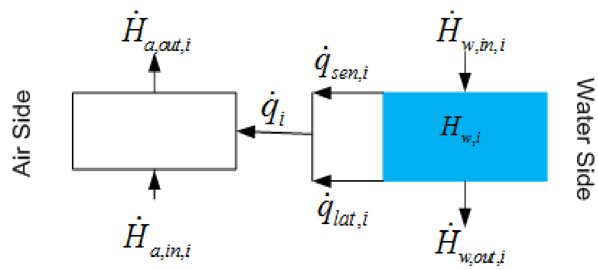

Figure 3: Energy Balance between Neighbored Water and Air Control Volumes

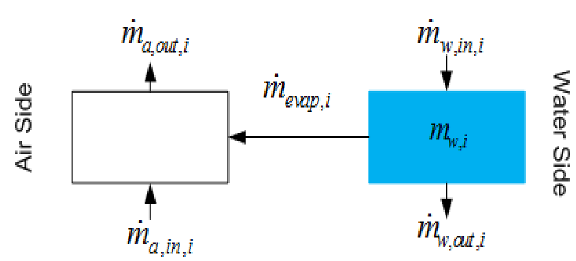

Figure 4: Mass balance between Neighbored Water and Air Control Volumes

For the i^{th} water-side control volume in Fig. 3, 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 i^{th}) 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. 4, 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, and $V_{effective}$ 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 i^{th} water cell,

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

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

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

$$\dot{q}_i = \dot{q}_{sen,i} + \dot{q}_{lat,i} \quad (10)$$

The sensible and latent heat flow rates can be determined by

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

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

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.

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

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

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

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

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

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

$$c_{pm,i} = c_{pa,i} + \omega_{a,i}c_{pv,i} \quad (16)$$

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 $c_{pm,i}$.

Determination of NTU and Le_f

The fill is used in most cooling towers, while it is usually hard to predict the heat rejection performance analytically because of the difficulty in evaluating the contact time and the surface area between the air and the water (Bernier 1994). The fouling in the packing materials may result in a reduction in the overall effectiveness of the tower and make it even harder to evaluate the fill geometry accurately. Due to the difficulty in getting a general correlations for heat and mass transfer in cooling tower in terms of the physical tower characteristics, the NTU and Le_f have been used to characterize the heat and mass transfer coefficients for specific tower designs (Braun et al. 1989). The Merkel's Number Me_M can be related to the mass transfer coefficient by (ASHRAE 1983)

$$Me_M = \frac{h_D A_V V_T}{\dot{m}_{w,in}} = c\left(\frac{\dot{m}_{w,in}}{\dot{m}_{a,in}}\right)^n \quad (17)$$

where $\dot{m}_{w,in}$ is the water inlet flow rate of cooling tower, c and n are empirical constants specific to a particular tower design. Multiplying both sides of Eq. (17) by $\dot{m}_{w,in}$ / $\dot{m}_{a,in}$ leads to

$$NTU = c\left(\frac{\dot{m}_{w,in}}{\dot{m}_{a,in}}\right)^{n+1} \quad (18)$$

where coefficients c and n can be from the performance measurements of a specific tower. According to Eq. (18), a linear regression of NTU vs. the flow rate ratio can be performed on a log-log plot. The slope and intercept of the fitted line are (n+1) and log(c), respectively (Braun et al. 1989). Kröger (2004) gave the methodology to obtain the Merkel's number from experimental data using empirical equations of thermal properties.

The Lewis relation has been discussed in literature. Poppe and Rogener (1991) cited the definition of Lewis relation based on Bosnjakovic (1965)

$$Le_f = Le^{2/3}\left[\left(\frac{\omega_{s,w}+d}{\omega_a+d} - 1\right)/\ln\left(\frac{\omega_{s,w}+d}{\omega_a+d}\right)\right] \quad (19)$$

where Le is the Lewis number, assumed as constant at 0.865. d is the ratio of molecular weight of water and molecular weight of air, which is a constant of 0.622. Grange (1994) and Bourillot (1983) claimed that for a wet cooling tower, Eq. (19) is approximately valued to be 0.92. 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.

MODEL OF RELATED COMPONENTS

Fan

The related cooling tower 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).

Pump

The condenser pump model is intended to predict the power consumption by the pump. A pump model from *TIL.LiquidComponents.Pumps.Pump2ndOrder*, has been adopted (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, and $\Delta p_{pump,affinity,0}$ 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

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

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

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

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 a constant exponent for power loss calculation.

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.

Collection Basin

The balance equations of collection basin are derived as

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

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

where $\dot{m}_{makeup}$ is the water flow rate from some source of make-up water and 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 rate of makeup water is equal to the total water loss rate from evaporation.

SIMULATION STUDY

Steady-State Simulation

Simulation was conducted to study the behavior and performance of the cooling tower. Figure 5 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.

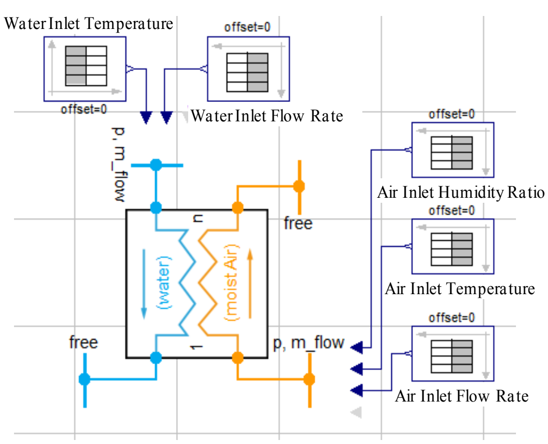

Figure 5: Dymola Layout for Evaporation Cooling Process of Cooling Tower

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. $T_{w,in}$ is the measured water inlet temperature, $T_{w,out}$ is the measured water outlet temperature, $T_{db,in}$ is the measured air inlet dry-bulb temperature, $T_{wb,in}$ is the measured air inlet wet-bulb temperature, $T_{db,out}$ is the measured air outlet dry-bulb temperature, $\dot{m}_{w,in}$ is the measured water inlet flow rate, $\dot{m}_{a,in}$ is the measured air inlet flow rate, $T_{w,out,cal}$ is the model predicted water outlet temperature, respectively. Figure 6 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 Eq. (19) is around 0.915, which is quite close to the value of 0.92 recommended by Kloppers and Kroger (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	1.187	1.009	25.46
2	34.39	29.0	31.78	26.67	31.22	1.165	1.009	28.78
3	43.61	27.89	35.0	23.89	32.78	1.158	0.755	28.12
4	38.78	29.33	35.0	26.67	33.28	1.265	1.009	29.87
5	43.06	29.72	35.72	26.67	33.89	1.157	0.755	29.94

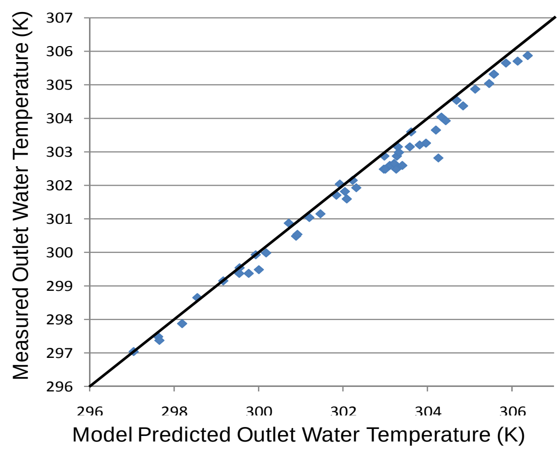

Figure 6: Comparison for Outlet Water Temperature between Model Prediction and Measured Data

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 7 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.

To facilitate the future 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. 8. Figure 9 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.

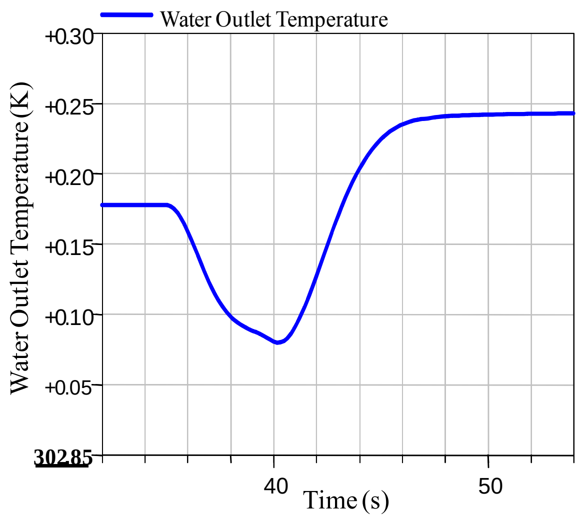

Figure 7: Transient Performance of Water Outlet Temperature - Stand-alone Cooling Tower Model

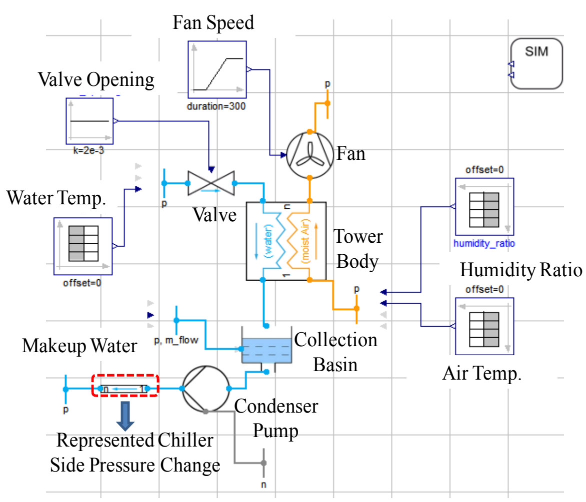

Figure 8: Dymola Layout for the Whole System of Cooling Tower Model

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.

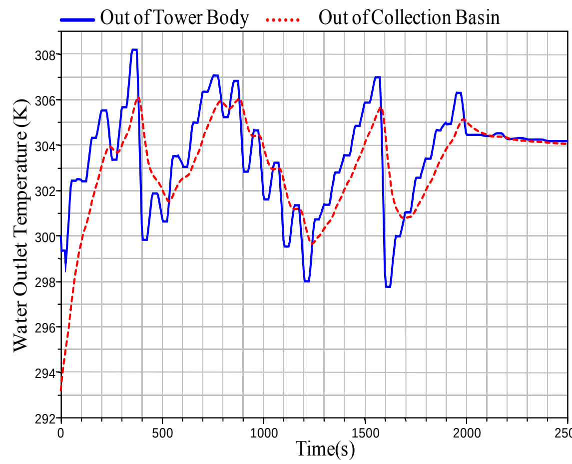

Figure 9: Transient Performance of Water Outlet Temperature - Comparison between Tower Body and Collection Basin Outputs

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