ENERGY PERFORMANCE

ESTIMATION OF COOLING TOWERS

by

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A THESIS

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ABSTRACT

The goal of this project is to investigate and compare the performance of cooling towers using Effectiveness-NTU model and empirical model. The processes to achieve the goal include: Developed Effectiveness-NTU and empirical models of the cooling tower, and predicted the performance of the design and off-design conditions; Stated experimental protocols and gathered data on the HVAC cooling tower on the campus of the University of Alabama; Used collected data to validate the models; Compared results from models with real measurements and found the limitation of each model; Applied known annual weather data to estimate the performance and energy consumption of the cooling tower; Recommended the approach for the best energy and heat performance.

All data and specifications were gathered and measured from the experiment on campus. In fact, the mass flow rate of air and the temperature of leaving air are not always possible to gather from different cooling towers, especially industry cooling towers. Therefore, both models were designed to predict mass flow rate of air, and the temperature of leaving air by applying the temperature and the relative humidity of entering air. Fan status and energy consumption of cooling tower were predicted according to the mass flow rate of air. Total annual energy consumption of single-speed, 2-speeds, and variable-speed cooling towers were calculated.

Further testing is required to validate the accuracy of the models because there was a limited control over the running status of fans. More experimental data needs to be collected in wintertime with lower temperature of the entering air. Validations on other cooling towers are essential in the future. Additionally, the accuracy of the empirical model can be improved by resetting all coefficients.
DEDICATION

This thesis is dedicated to my Mom and Dad. No words to show my appreciation to both of you.
LIST OF ABBREVIATIONS AND SYMBOLS

\( m_{\text{water}} \) Mass flow rate of water
\( m_{\text{air}} \) Mass flow rate of air
\( h_{\text{water \_out}} \) Enthalpy of leaving water
\( h_{\text{water \_in}} \) Enthalpy of entering water
\( h_{\text{air \_out}} \) Enthalpy of leaving air
\( h_{\text{air \_in}} \) Enthalpy of entering air
\( h_{\text{sat \_water \_in}} \) Enthalpy of saturated air at entering water temperature
\( w_{\text{air \_out}} \) Specific humidity of leaving air
\( w_{\text{air \_in}} \) Specific humidity of entering air
\( T_{w} \) Wet bulb temperature
\( T_{d} \) Dry bulb temperature
\( RH\% \) Relative humidity
\( e_{\text{sw}} \) Pressure of saturated vapor
\( e_{\text{air}} \) Effectiveness of air
\( c_{p\_\text{water}} \) Specific heat of water
\( c_{p\_\text{air}} \) Specific heat of air
\( P \) Power
ACKNOWLEDGMENTS

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Thanks my dear Danika to take good care of me. We shared many wonderful memories. Always feel happy to stay with you and hope everything goes well in the future. The guys in Hardaway 285 provided great support on my research, and you made the office fulfill of cheers. Finally, thanks to Binbo, the King of Husky, nothing bothers when I see your face.
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CHAPTER 1
INTRODUCTION

There are only a few available energy performance models for air-conditioning cooling towers: American Society of Heating, Refrigerating and Air-Conditioning Engineer (ASHRAE) HVAC Toolkit, the Effectiveness-NTU Model, and the empirical model using performance curves. All of them have issues limiting the application of these models directly to performance analysis of cooling towers: Too many parameters are required and hard to find for many existing cooling towers; all are limited to air conditioning cooling towers, and may not be appropriate for industrial cooling towers.

The theories and applications of Effectiveness-NTU model were introduced by Stout and Leach (2002). They provided the capacity control of cooling towers. Capacity control can be achieved by changing the air velocity, and be accomplished by modifying the fan speeds (full, half, one-third, and variable). The fans of a cooling tower do not need to run at their maximum speed the whole time because of the change of entering air temperature and humidity. Therefore, a capacity control fan can save energy for a cooling tower. Their study shows energy variable speeds fan can save maximum 70% of energy compare to single speed fan. The energy saving not only varies on different cooling tower capacity but also varies with the locations because of the climates. Their tests were done in Raleigh, Columbus, Denver, Houston, and Los Angeles.

According to Benton, Bowman, Hydeman and Miller (2002), Variable Speed Cooling Towers Empirical Models provided the basic knowledge of empirical models applied in the EnergyPlus. YorkCalc and CoolTools are the empirical simulating models released by different manufacturers. EnergyPlus is a whole building energy simulation program used by
engineers, architects, and researchers (EnergyPlus, 2014). It was developed by the U.S. Department of Energy’s Building Technologies Office. Both of YorkCalc and CoolTools models have numbers of pre-set coefficients. The users of EnergyPlus can also define their coefficients based on different cooling towers.

The goal of this project is to investigate and compare the performance of cooling towers using Effectiveness-NTU model and empirical model. The processes to achieve the goal include: Developed Effectiveness-NTU and empirical models of the cooling tower, and predicted the performance of the design and off-design conditions; Stated experimental protocols and gathered data on the HVAC cooling tower on the campus of the University of Alabama; Used collected data to validate the models; Compared results from models with real measurements and found the limitation of each model; Applied known annual weather data to estimate the performance and energy consumption of the cooling tower; Recommended the approach for the best energy and heat performance.
CHAPTER 2
COOLING TOWER

The cooling tower is a heat-exchange device that rejects heat to surrounding air. The evaporation of water removes heat to cool the water. Based on the study of Viska Mulyandasari (2011), several important factors dominate the performance of a cooling tower: Dry-bulb and wet-bulb temperature of the air; Temperature of the water; Air pressure; Efficiency of contact between air and water. The design condition is used to estimate cooling tower capacity. The temperature of entering water at design condition is 95 °F; The temperature of leaving water at design condition is 85 °F; Wet-bulb temperature of the entering air at design condition is 78 °F. Figure 2.1 from Mulyandasari’s work (2011) shows how a cooling tower reduces the temperature of water using air flow. Cooling towers have two important parameters: range temperature and approach temperature. The range temperature of a cooling tower is the difference between entering and leaving water temperature. The approach temperature of a cooling tower defines the temperature difference between leaving water and wet-bulb of entering air.

Figure 2.1 Cooling Tower Working Process (Mulyandasari, 2011)
A cooling tower can be used in HVAC systems or industrial plants. An HVAC cooling tower disposes heat from a chiller and reduces the water temperature near to wet-bulb temperature. A ton of air conditioning is defined to remove 3,500W (12,000 Btu/hr.) of heat. Industrial cooling towers are much larger than HVAC cooling towers and are used in power plants, petroleum refineries, and other industrial facilities. For example, an industrial cooling tower can process 80,000 cubic meters of water per hour in petroleum refineries (Stout and Leach, 2002). More classifications of cooling towers were shown in Figure 2.2 based on Mulyandasari’s (2011) study.

Figure 2.2. Classification of Cooling Towers (Mulyandasari, 2011)

Cooling towers have two different airflow methods: natural draft and mechanical draught. Natural draft cooling tower uses a tall chimney. Hotter and moister air naturally rises...
to meet with cooler and drier air, and remove the heat of the fluid by evaporation. Mechanical
draught cooling towers use fans to increase the flow rate of air. A fan can set at either the air
inlet or outlet. Most fans are installed at the air outlet of the cooling towers as draw-through
because of better efficiency. Two types of air-to-water flow cooling towers are cross-flow
and counter-flow. The cross-flow cooling tower has the airflow perpendicular to the water
flow. The counter-flow cooling tower has the airflow goes opposite to the water flow.
Figure 2.3 shows how cross-flow and counter-flow cooling towers works. The benefit of the
cross-flow cooling tower is low cost. The counter-flow cooling tower is better at anti-freeze.

Figure 2.4. Energy Balance of Cooling Towers (Lu and Cai, 2002)

Figure 2.4 shows how heat transfers in a cooling tower according to the work of Lu
and Cai (2002), and an energy conservation equation of the cooling tower can be expressed
as:

\[ m_{\text{air}} \left[ (h_{\text{airout}} - h_{\text{airin}}) - (w_{\text{airout}} - w_{\text{airin}}) \times h_{\text{waterout}} \right] = m_{\text{water}} \times (h_{\text{waterout}} - h_{\text{waterin}}) \]

Eq. 2.1

where \( m_{\text{water}} \) and \( m_{\text{air}} \) are the mass flow rate of water and air with units of kilograms per
second. \( h_{\text{airout}} \) is the enthalpy of leaving air; \( h_{\text{airin}} \) is the enthalpy of entering air; \( h_{\text{waterout}} \)
is the enthalpy of leaving water; \( h_{\text{waterin}} \) is the enthalpy of entering watering. \( w_{\text{airout}} \) and
\( w_{\text{airin}} \) are the specific humidity of leaving and entering air.
CHAPTER 3
PSYCHROMETRIC EQUATIONS

In cooling towers, heat and energy transfer from water to air by lowering the
temperature of the water, and rising the temperature and humidity of air. Therefore, the
energy balance of a cooling tower depends on the study of moist air. The psychrometric
equations helped to establish the models to predict the performance of a cooling tower.

Both temperature and humidity of air affect the performance of transferring energy
from water to air. Lower temperature and less humidity air can absorb more heat from water.
Both of the temperature and humidity of leaving air increase compare to entering air. Wet-
bulb temperature can express both temperature and humidity of air. According to
MacPherson’s work (1993), wet bulb temperature, \( T_w \) at sea-level air pressure can be
determined using the following equation:

\[
T_w = T_d \times \frac{\tan\left(0.151977 + 8.313659 \cdot \frac{RH}{100}\right)^{1/2}}{\left(0.00391838 \times (RH)^{3/2} \times \tan\left(0.023101 \times RH\right) - 4.686035\right)^2} + \tan(T_d + RH\%) - \tan(RH\% - 1.676331)
\]

Eq. 3.1

where \( T_d \) is the dry-bulb temperature and \( RH \) is the relative humidity of the air. If the
wet-bulb temperature is known, all the other psychrometric parameters can be calculated.
According to MacPherson’s work (1993), Equation 3.2 to Equation 3.6 were listed below:
The pressure of saturated vapor, \( e_{sw} \) can be calculated using:

\[
e_{sw} = 610.18 \times \exp\left[\frac{17.27 \times t_w}{237.3 + t_w}\right]
\]

Eq. 3.2
The range of the wet-bulb temperature applies in this equation is from 0 °C to 60 °C, and the unit of the pressure is Pascal, Pa. Specific humidity of saturated air, $X_s$ with a unit of kg/kg of dry air can be determined using:

$$X_s = 0.622 \times \frac{e_{sw}}{P - e_{sw}}, \quad \text{Eq. 3.3}$$

where $P$ is the air pressure, and sea-level pressure of 101kPa was applied in the thesis. Latent heat, $L_w$ can be determined using following equation, and the unit is kJ/kg.

$$L_w = (2502.5 - 2.386 \times t_w) \times 1000, \quad \text{Eq. 3.4}$$

The enthalpy of moist air, $h$ with a unit of J/kg can be determined using:

$$h = L_w \times X_s + 1005 \times t_w, \quad \text{Eq. 3.5}$$

The specific humidity of moist air with a unit of kg/kg equals:

$$w = \frac{h - 1005 \times T_d}{L_w + 1884 + (T_d - t_w)}, \quad \text{Eq. 3.6}$$

where $T_d$ is the dry-bulb temperature and $L_w$ is the latent heat.
CHAPTER 4

EFFECTIVENESS-NTU MODEL

According to Incropera, DeWitt, Bergaman and Lavine’s work (2007), Number of Transfer Units (NTU) method was used to estimate the rate of heat transfer. The performance of heat exchanger is usually estimated by either the Logarithmic Mean Temperature Difference (LMTD) or the Effectiveness-NTU methods. The LMTD method calculates the overall heat transfer coefficient based on the measured temperature of entering and leaving fluid. The Effectiveness-NTU method is easier for predicting leaving fluid temperatures if the heat transfer coefficient and the entering fluid temperature are known.

According to Stout and Leach (2002), Effectiveness-NTU method can predict the leaving temperature of a cooling tower without solving nonlinear equations. The equation of Effectiveness-NTU method provides:

\[ NTU = \frac{h_c A}{m_a C_{\text{moistair}}} \],

Eq.4.1

where \( h_c \) is the heat transfer coefficient; \( m_a \) is the mass flow rate of air; \( C_{\text{moistair}} \) is the effective specific heat of moist air; \( A \) is the heat transferring surface area. According to Stout and Leach (2002), a cooling tower with large heat transfer surface and small fans may have the nominal NTU values of 4.5, and a low initial cost towers with small heat transfer surfaces and large fans may have the nominal NTU values of 1.5 or less. Since \( h_c \) depends on the mass flow rate of air and water, the NTU value at off design conditions is different than at design condition. Equation. 4.2 (Stout and Leach, 2002) shows how to predict the performance of cooling towers at off design condition:

\[ NTU = A * m_{\text{air}}^m * m_{\text{water}}^n \],

Eq.4.2
In most cases, a constant value of $A$, $m$ and $n$ cannot be found. Therefore, a less accurate formula for NTU is given below (Stout and Leach, 2002):

$$NTU = c \cdot \left(\frac{m_{\text{water}}}{m_{\text{air}}}\right)^n,$$

Eq.4.3

In this equation, $m_{\text{water}}$ and $m_{\text{air}}$ is the mass flow rate of the water and air through the cooling tower. Usually, the range of $n$ is between 0.4 and 0.6. Once the $n$ value was assumed, the $c$ value can be determined at design condition. For example, the nominal capacity of 200 tons to 1000 tons cooling tower has $c = 2.66$ and $n = 0.634$. As a cooling tower with capacity of 68 tons has $c = 3$ and $n = 0.4$, $c = 1.33$ and $n = 0.4$ for a 16 tons cooling tower. (Stout and Leach, 2002)

The effectiveness of the air in cooling tower can represent as (Stout and Leach, 2002):

$$e_{\text{air}} = \frac{h_{\text{airout}} - h_{\text{airin}}}{h_{\text{satwaterin}} - h_{\text{airin}}},$$

Eq.4.4

where $e_{\text{air}}$ is the effectiveness of air; $h_{\text{satwaterin}}$ is the enthalpy of saturated air at the temperature of entering water; $h_{\text{airout}}$ and $h_{\text{airin}}$ are the enthalpy of the entering and leaving air. When a cooling tower reaches 100 percent efficiency, the approach temperature equals to 0, which means the temperature of leaving saturated air from the cooling tower should equal to the temperature of entering water. However, the temperature of air leaving the cooling tower should be less than the entering water in real condition. Other equations in Stout and Leach’s (2002) study include:

$$c_{\text{moistair}} = \frac{h_{\text{satwaterin}} - h_{\text{satwaterout}}}{T_{\text{waterin}} - T_{\text{waterout}}},$$

Eq.4.5

where $c_{\text{moistair}}$ is the relative specific heat of moist air, $T$ is temperature and $h$ is enthalpy.

For cross-flow cooling towers, the effectiveness of air related to $NTU$ and $m$ equals to (Stout and Leach, 2002):

$$e_{\text{air}} = \left[1 - \exp(m \cdot (\exp(-NTU) - 1))\right]/m,$$

Eq.4.6
For counter-flow cooling towers, the equation becomes to (Stout and Leach, 2002):

\[ e_{\text{air}} = \frac{1-\exp(NTU(m-1))}{1-m\exp(NTU(m-1))} , \]

Eq.4.7

where \( m \) is a dimensional variable of the ratio between moist air and water, and the equation shows below:

\[ m = \frac{m_{\text{air}}c_{\text{moistair}}}{m_{\text{water}}c_{\text{water}}} , \]

Eq.4.8

where \( c_{\text{water}} \) is the specific heat of water. When the fan turns off, air continues to flow into the cooling tower because of natural convection. Once the fans are turned off, the air flow rate into the cooling tower can be determined using (Stout and Leach, 2002):

\[ cfm_0 = C_0 * cfm * \left( \frac{\Delta h}{\Delta h_n} \right)^{0.2} , \]

Eq.4.9

where \( C_0 \) is a constant value; \( cfm \) is the capacity of the fan; \( \Delta h \), and \( \Delta h_n \) are the mean enthalpy difference. Following equations indicate how to calculate \( \Delta h \) and \( \Delta h_n \) (Stout and Leach, 2002):

\[ \Delta h = \frac{\Delta h_o - \Delta h_i}{\log(\frac{\Delta h_o}{\Delta h_i})} , \]

Eq.4.10

\[ \Delta h_o = h_{\text{satwaterin}} - h_{\text{airout}} , \]

Eq.4.11

\[ \Delta h_i = h_{\text{satwaterout}} - h_{\text{airin}} , \]

Eq.4.12

where \( \Delta h_n \) is the enthalpy difference when the fan is running in design condition. The constant value \( C_0 \) is the ratio of the airflow with natural convection to the airflow with the fan capacity. Larger heat transferring areas and smaller fans will have higher \( C_0 \). The calculation of natural convection of cooling towers can also be simplified by using the manufacturing specifications. The capacity with natural convection of their products is about 10 percent of the rated capacity (EVAPCO Inc., 2016).

According to Stout and Leach (2002), the performance of cooling tower can be expressed by comparing the temperature of leaving water and the wet-bulb temperature of entering air. The predicted performance of the cooling towers is shown in Figure 4.1. NTU
can be determined once the \( c \) and \( n \) values, the design conditions, and the flow rate of water and air were known. Then, the effectiveness of air can be calculated using Equation 4.6 or Equation 4.7 depends on the type of cooling tower. Based on Equation 3.1 and Equation 4.4, if the enthalpy of entering air and the temperature of entering water are known, the enthalpy of leaving air can be determined. The temperature of leaving water can be determined using Equation 4.13:

\[
T_{\text{water out}} = \frac{c_p \text{water} \cdot m_{\text{water in}} \cdot (T_{\text{water in}} - T_{\text{water out}}) + c_p \text{water} \cdot m_{\text{water out}}}{c_p \text{water} \cdot m_{\text{water in}}} \tag{Eq. 4.13}
\]

Figure 4.1 Performance Predictions (Stout and Leach, 2002)

According to Stout and Leach (2002), the NTU was found first via Equation 4.3 only to lead directly to the effectiveness of air through Equation 4.6 and Equation 4.7. If the effectiveness of air can be found without calculating NTU, the NTU will be no longer needed. The idea is explained in Section 9.1 of this thesis.
CHAPTER 5
EMPIRICAL MODEL

The performance of a cooling tower can be estimated via empirical curve fits. An empirical model requires the wet-bulb temperature of entering air, range temperature, and mass flow rate of air and water at design condition. The objective of the empirical model is to estimate the approach temperature at different fan operating conditions.

The empirical model was built based on CoolTools correlation and YorkCalc correlation that applied in EnergyPlus (EnergyPlus, 2014). These empirical models estimate the approach temperature using a polynomial curve fit with the amount of coefficients and either three or four independent variables. CoolTools is software released by Pacific Gas and Electric Company (PG&E), and the objective is to develop, disseminate, and promote a tool for design and operation of chilled water plants. CoolTools products are Internet-based, public domain resources, and are targeted to building owners, design professionals, and operators involved in both new construction and retrofits. Unfortunately, the CoolTools is no longer available for users and was removed from the website of PG&E for an unknown reason. YorkCalc model was developed by York International Corporation simulating the performance of their produced chiller systems. Johnson Controls used $3.2 billion to acquire York International Corporation in 2005, and the information of YorkCalc was no longer shown on their website since that day.

Figure 5.1 shows the CoolTools correlation has 35 coefficients and requires four independent variables (Benton, Bowman, Hydeman & Miller, 2002):
Figure 5.1 CoolTools Equations (EnergyPlus, 2014)

\[
\text{Approach} = \text{Coeff}(1) + \text{Coeff}(2)\cdot FR\text{air} + \text{Coeff}(3)\cdot (FR\text{air})^2 + \text{Coeff}(4)\cdot (FR\text{air})^3 + \text{Coeff}(5)\cdot FR\text{water} + \text{Coeff}(6)\cdot FR\text{air}\cdot FR\text{water} + \text{Coeff}(7)\cdot (FR\text{air})^2\cdot FR\text{water} + \text{Coeff}(8)\cdot (FR\text{water})^2 + \text{Coeff}(9)\cdot FR\text{air}\cdot (FR\text{water})^2 + \text{Coeff}(10)\cdot (FR\text{water})^3 + \text{Coeff}(11)\cdot TWb + \text{Coeff}(12)\cdot FR\text{air}\cdot TWb + \text{Coeff}(13)\cdot (FR\text{air})^2\cdot TWb + \text{Coeff}(14)\cdot FR\text{water}\cdot TWb + \text{Coeff}(15)\cdot FR\text{air}\cdot FR\text{water}\cdot TWb + \text{Coeff}(16)\cdot (FR\text{water})^2\cdot TWb + \text{Coeff}(17)\cdot (TWb)^2 + \text{Coeff}(18)\cdot FR\text{air}\cdot (TWb)^2 + \text{Coeff}(19)\cdot FR\text{water}\cdot (TWb)^2 + \text{Coeff}(20)\cdot (TWb)^3 + \text{Coeff}(21)\cdot TWb + \text{Coeff}(22)\cdot FR\text{air}\cdot TWb + \text{Coeff}(23)\cdot (FR\text{air})^2\cdot TWb + \text{Coeff}(24)\cdot FR\text{water}\cdot TWb + \text{Coeff}(25)\cdot FR\text{air}\cdot FR\text{water}\cdot TWb + \text{Coeff}(26)\cdot (FR\text{water})^2\cdot TWb + \text{Coeff}(27)\cdot (TWb)^2 + \text{Coeff}(28)\cdot FR\text{air}\cdot TWb + \text{Coeff}(29)\cdot FR\text{water}\cdot TWb + \text{Coeff}(30)\cdot (TWb)^2\cdot TWb + \text{Coeff}(31)\cdot (TWb)^3 + \text{Coeff}(32)\cdot FR\text{air}\cdot (TWb)^3 + \text{Coeff}(33)\cdot FR\text{water}\cdot (TWb)^3 + \text{Coeff}(34)\cdot (TWb)^4 + \text{Coeff}(35)\cdot (TWb)^5
\]

where:

Approach = approach temperature (°C) = outlet water temperature minus inlet air wet-bulb temperature

FRair = air flow rate ratio (actual air flow rate divided by design air flow rate)

FRwater = water flow rate ratio (actual water flow rate divided by design water flow rate)

Tr = range temperature (°C) = inlet water temperature minus outlet water temperature

Twb = inlet air wet-bulb temperature (°C)

Coeff(#) = correlation coefficients

Figure 5.2 YorkCalc Equations (EnergyPlus, 2014)

\[
\text{Approach} = \text{Coeff}(1) + \text{Coeff}(2)\cdot TWb + \text{Coeff}(3)\cdot TWb^2 + \text{Coeff}(4)\cdot Tr + \text{Coeff}(5)\cdot TWb\cdot Tr + \text{Coeff}(6)\cdot TWb^2\cdot Tr + \text{Coeff}(7)\cdot Tr^2 + \text{Coeff}(8)\cdot TWb\cdot LGRatio + \text{Coeff}(9)\cdot TWb^2\cdot LGRatio + \text{Coeff}(10)\cdot LGRatio + \text{Coeff}(11)\cdot TWb\cdot LGRatio + \text{Coeff}(12)\cdot TWb^2\cdot LGRatio + \text{Coeff}(13)\cdot Tr\cdot LGRatio + \text{Coeff}(14)\cdot TWb\cdot Tr\cdot LGRatio + \text{Coeff}(15)\cdot TWb^2\cdot Tr\cdot LGRatio + \text{Coeff}(16)\cdot Tr^2\cdot LGRatio + \text{Coeff}(17)\cdot TWb\cdot Tr^2\cdot LGRatio + \text{Coeff}(18)\cdot TWb^2\cdot Tr^2\cdot LGRatio + \text{Coeff}(19)\cdot LGRatio^2 + \text{Coeff}(20)\cdot TWb\cdot LGRatio^2 + \text{Coeff}(21)\cdot TWb^2\cdot LGRatio^2 + \text{Coeff}(22)\cdot Tr\cdot LGRatio^2 + \text{Coeff}(23)\cdot TWb\cdot Tr\cdot LGRatio^2 + \text{Coeff}(24)\cdot TWb^2\cdot Tr\cdot LGRatio^2 + \text{Coeff}(25)\cdot Tr^2\cdot LGRatio^2 + \text{Coeff}(26)\cdot TWb\cdot Tr^2\cdot LGRatio^2 + \text{Coeff}(27)\cdot TWb^2\cdot Tr^2\cdot LGRatio^2
\]

where:

Approach = approach temperature (°C) = outlet water temperature minus inlet air wet-bulb temperature

Tr = range temperature (°C) = inlet water temperature minus outlet water temperature

Twb = inlet air wet-bulb temperature (°C)

LGRatio = liquid-to-gas ratio = ratio of water flow rate ratio (FRwater) to air flow rate ratio (FRair)

Coeff(#) = correlation coefficients
Figure 5.2 shows the YorkCalc correlation has 27 terms with four independent variables (York Co., 2002). The user needs to enter all coefficients to the equations to finish simulation. The coefficients for both CoolTools and YorkCalc are shown below (EnergyPlus, 2014).

<table>
<thead>
<tr>
<th>Coefficient Number</th>
<th>Coefficient Value: CoolTools</th>
<th>Coefficient Value: YorkCalc</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coeff(1)</td>
<td>0.52049709893241</td>
<td>-0.359741205</td>
</tr>
<tr>
<td>Coeff(2)</td>
<td>-10.617046395344</td>
<td>-0.05053608</td>
</tr>
<tr>
<td>Coeff(3)</td>
<td>10.7296274722538</td>
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<td>Coeff(35)</td>
<td>0.000807370864480284</td>
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</tr>
</tbody>
</table>

Figure 5.3 Coefficients of CoolTools and YorkCalc (EnergyPlus, 2014)

Both of CoolTools and YorkCalc models have their limitations and were shown in Figure 5.4. YorkCalc is more flexible than CoolTools because YorkCalc can accept larger range and approach temperatures. Since CoolTools and YorkCalc are developed based on specific types of cooling towers and chiller systems, different coefficients and constraints need to be determined to use on another system.
Figure 5.4 Limitations of CoolTools and YorkCalc (EnergyPlus, 2014)

<table>
<thead>
<tr>
<th>Independent Variable Limit</th>
<th>CoolTools</th>
<th>YorkCalc</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum Inlet Air Wet-Bulb Temperature</td>
<td>-1.0°C</td>
<td>-34.4°C</td>
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<tr>
<td>Maximum Inlet Air Wet-Bulb Temperature</td>
<td>26.7°C</td>
<td>26.7°C</td>
</tr>
<tr>
<td>Minimum Tower Range Temperature</td>
<td>1.1°C</td>
<td>1.1°C</td>
</tr>
<tr>
<td>Maximum Tower Range Temperature</td>
<td>11.1°C</td>
<td>22.2°C</td>
</tr>
<tr>
<td>Minimum Tower Approach Temperature</td>
<td>1.1°C</td>
<td>1.1°C</td>
</tr>
<tr>
<td>Maximum Tower Approach Temperature</td>
<td>11.1°C</td>
<td>40°C</td>
</tr>
<tr>
<td>Minimum Water Flow Rate Ratio</td>
<td>0.75</td>
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</tr>
<tr>
<td>Maximum Water Flow Rate Ratio</td>
<td>1.25</td>
<td>1.25</td>
</tr>
<tr>
<td>Maximum Liquid-to-Gas Ratio</td>
<td>N/A</td>
<td>8.0</td>
</tr>
</tbody>
</table>
CHAPTER 6
COOLING TOWER SPECIFICATIONS

Once the Effectiveness–NTU model and the empirical model were made using MATLAB, a field test was prepared to validate both models. The cooling tower used in the validation located behind AIME building at the University of Alabama. All the measuring equipment were ordered at Onset Computer Corporation. Experimental protocols were written and ready to apply to any other field tests. All collected data were analyzed and used to verify the models.

Basic specifications and properties of the cooling tower were gathered from online research. The cooling tower used in the experiment was made by Marley Cooling Tower Corporation in 1999, and the series number is NC4222GS. Marley Co. was acquired by SPX Cooling Co., and no longer makes these series of cooling towers. However, there is a current model called NC8403 has similar dimensions and specifications with NC4222GS. A sketch of the cooling tower is shown in Figure 6.1, and it is a dual-entering cooling tower that has two cells and connected by the pipes of entering water. The vents of entering air located on both sides of the cooling tower and the dimension of each vent is 11.5 feet by 7.5 feet. The vents of leaving air are located on the top of the cooling towers with diameters of 7 feet. Other dimensions and sketches of NC8403 can be found in the handbook in NC Steel Cooling Tower Engineering Data on their website (SPX Cooling Technologies, Inc., 2011). According to the classifications of cooling towers, the cooling tower used in the experiment is wet, cross-flow and mechanical draught with fans.
Each of these fans was operated by a 25 horsepower motor made by Marathon Electric. The motors were designed with two speeds by serving cooling towers. The RPM of the motors is 1755, and the nominal effectiveness is 87.5%. Since each fan has three speeds includes natural convection with fans turned off, there are six running statuses of the cooling tower.

The entering water comes from two chillers and flows into the cooling tower through the pipes of entering water, and the leaving water flows out to a water basin. Two pumps made by Goulds Pumps Inc. pump the leaving water from the basin to the chiller. The capacity of both pumps is 1020 GPM, and the head pressure is 72 feet. Each pump is operated by a 25 horsepower motor made by GE Motors. According to the operation procedure, each pump and chiller switching their operation every 15 days, and rarely turn on at the same time.
7.1 Procedure and Devices

An experiment was designed to collect data, estimate and validate the Effectiveness–NTU model and empirical model. A graph of the designed experiment is shown in Figure 7.1. The measurements include the air temperature and relative humidity, the velocity of air, the pump pressure, and the power of the motors.

The Facilities Department of the University provided the temperature of entering and leaving water, the flow rate of makeup water and the running statuses of fans and pumps. The sensors and data loggers were installed as shown in Figure 7.1. All the data was collected using RX3002 and UX90 data loggers made by Onset Computer Co. The RX3002 data
logger was powered by a 6W solar panel shown in Figure 7.2 (a), and the data was uploaded to the cloud through wireless.

![Figure 7.2 Experimental Equipment](image)

(a) Data Logger RX 3002 (b) Pressure Sensor T-ASH-G2-500 (c) Hood for S-THB-M008

Air temperature and humidity were measured using S-THB-M008 at both air inlet and outlet. The sensors record dry-bulb temperature and relative humidity of the air. Wet-bulb temperature can be calculated using Equation 3.1. Pump pressure was collected using T-ASH-G2-500, which is shown in Figure 7.2 (b). The pump curve can be found by looking for the series number of the pumps in Eprism released by Gouldspump Inc. (Gouldspump Inc., 2016). The flow rate of leaving water can be found using the pump curve. Figure 7.3 shows the pump curve of VIC/VIT pumps with a motor of 25 horsepower and 1770 RPM. In Figure 7.2 (c), a hood was assembled over the sensor on the top of the cooling tower to avoid solar radiation.
The velocity of entering water was measured using GE Parametric Transport PT878 Ultrasonic Flowmeter System shown in Figure 7.4 (a). This device was installed on the pipe of entering water and recorded the water velocity. In this project, clamp-on #402 (Shear) transducer was used, and wedge temperature was 20 °C as measured. The pipe is made of Steel (Stainless 304), and the outer diameter of the pipe is 254 mm with a thickness of 5.08 mm. There is no lining material around the pipe, and the fluid temperature was set as 30 °C of water. According to all these pre-setting above, the device determined the distance between two transducers should be 233.15 mm.
Fan and pump power were measured using T-WNB-3D-480 transducer and SCT-0750 sensors shown in Figure 7.4 (b). The current sensed by transducers generates a proportional series of pulses, and the count of pulses was recorded by data loggers. The power of the motor can be determined using the equation from the user manual of the device. According to the user manual (Continental Control System, 2016), the equations applied in the experimental cooling tower is:

\[
Power = WHpP \times 3600 \times \text{Pulse Frequency}
\]

Eq. 6.1

where \( WHpP \) equals 5.7708, and the pulse frequency equals measured counts divided by 300.

Air velocity was measured using T-DCI-F99-L-P at the vent of entering air. The unit of the velocity is foot per minute. The volumetric flow rate can be determined by multiplying the velocity by the area of the vents. The mass flow rate can be calculated by multiplying volumetric flow rate by air density.
7.2 Experiment Protocol

The frequency of recording data was every 5 minutes. Handheld devices are recommended to use to double-check the measurements read by sensors. Figure 7.5 shows handheld devices of air velocity and pump pressure.

![Handheld Devices](image)

**Figure 7.5 Handheld Devices**

Because the area of the air vent is large, the air speed varies at different locations on the vent. At least ten measurements should be done at each location of the vent using handheld air velocity device to reduce the variance of measurements. The locations are shown in Figure 7.7 (a). Comparing the average velocity with the center point yields the ratio between air velocity at a center point and average velocity. The ratio of this cooling tower is 1.24 when the fan was running at high speed. Additionally, measurements of air velocity at low speed and stopped running statuses were needed to be captured and recorded. The airspeed sensor should locate in the center of the vent, and the average airspeed can be calculated by multiply the ratio by the measurements.

There are smart connections and analog connections between sensors and loggers. Smart connections can easily plug into the data logger, but analog connections require additional power startup. The frequency of power startup must be defined before the analog-connecting sensors can start reading or recording data.
GE Parametric Transport PT878 Ultrasonic Flow Meter System is complicated to use and readout. The material, diameter, flow direction and thickness of the pipe must be known. The flow meter can only transfer data through infrared radiation, which is an outdated technology and lost connections all the time. The flow meter is not waterproof, and the battery can only last 2 hours. Therefore, the flow meter must be used indoors and connected to a charger.

One must be extremely careful when installing power sensors and transducers. The high voltage in the electrical cabinet can kill people. Make sure the switch was turned off, and assure safety for oneself and any assistants.

![Figure 7.7](image)

(a) Points of Measurements (b) Analog Connection
CHAPTER 8

MEASURED DATA ANALYSIS

The purpose of data analysis is testing if the collected data is accurate enough to be used to validate the models. The verification processes include collecting data, calculating variables, and validating results. The period of the data collection is from August 25 to September 1, 2016, and the frequency is every 5 minutes.

Data was collected from three sources include data logger, ultrasonic flow meter, and the Facilities Department of the University of Alabama. The data logger records the relative humidity and the dry-bulb temperature of entering and leaving air, the pump pressure, the velocity of entering air and the power of fan and pump motors. The facility provides the running status of pumps and fans, the flow rate of makeup water, and the temperature of entering and leaving water. The ultrasonic flow meter records the flow rate and velocity of entering water. However, the result cannot apply in the analysis because of numerous errors due to the operation failure. Therefore, the mass flow rate of entering water was calculated by the sum of the mass flow rate of leaving water and make-up water.

The wet-bulb temperature was calculated using relative humidity and dry-bulb temperature based on Equation 3.1. Velocity and flow rate of the leaving water were determined using pump pressure and the pump curve in Figure 7.3. Figure 7.7 (a) shows the measured areas of vent, which is used to calculate velocity and the flow rate of air. Enthalpy of the entering and leaving air was computed based on psychrometric equations in Chapter 3. The temperature of leaving water can be found using Equations 4.13.
Since the temperature of leaving water was provided by the Facilities Department of the University, a comparison was made to validate the experimental measurements. Figure 8.1 shows both the measured and the calculated temperature of leaving water.

![Compare Calculated Value and Measured Value](image)

**Figure 8.1 Compare Calculated Value and Measured Value**

The calculated temperature of leaving water has larger variances than the measured temperature because the measurements of air velocity vary a lot. The ideal result of the air velocity is constant. However, the range of measured air velocity is from 600 foot per minute to 900 foot per minute when the fan was running at same speed. Although the ratio between the average speed and the center velocity was calculated earlier, the experiment was still limited by the accuracy of air velocity. Installing more air velocity sensors should reduce the variances, but the data logger does not have enough slots for more than two analog connection sensors.
Table 8.1 shows the statistical result of regression analysis between the two sets of data.

### Table 8.1 Regression Analyses

<table>
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<th>Regression Statistics</th>
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<td>Multiple R</td>
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<tr>
<td>R Square</td>
<td>0.74</td>
</tr>
<tr>
<td>Adjusted R Square</td>
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<tr>
<td>Standard Error</td>
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<td>Observations</td>
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### Table 8.2 Data Analysis

<table>
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<th>Measured Value</th>
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<tr>
<td>Mean</td>
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<td>Mean</td>
<td>77.81</td>
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<tr>
<td>Standard Error</td>
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<td>Standard Error</td>
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<tr>
<td>Median</td>
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<td>Median</td>
<td>77.90</td>
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<td>Mode</td>
<td>79.1</td>
</tr>
<tr>
<td>Standard Deviation</td>
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<td>Standard Deviation</td>
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</tr>
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<td>Sample Variance</td>
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<td>Sample Variance</td>
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<td>Kurtosis</td>
<td>-0.41</td>
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</tr>
<tr>
<td>Count</td>
<td>2017</td>
<td>Count</td>
<td>2017</td>
</tr>
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</table>

Another statistical result shows root mean square error (RMSE) of these two data is 1.71; The coefficient of variation mean square error (CV-RMSE) is 2.2%; The normalized mean bias error (NMBE) is 3.8%; The normalized root mean square error (NRMSE) is 12.8%; The R-squared value between measured data and calculated data equals to 0.7402. As a result, the experimental data is accurate enough to use in both Effectiveness–NTU and empirical models because the CV-RMSE and NMBE of the calculated data meet the requirements from ASHRAE Guideline 14. The frequency of measured data was 5 minutes.
and it was modified to 1 hour to verify the accuracy with ASHRAE’s requirements. The recommended value of ASHRAE Guideline 14 is shown in Figure 8.2 (ASHRAE, 2014).

<table>
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<tr>
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<th>Daily</th>
<th>Hourly</th>
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<tr>
<td>CV-RMSE</td>
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<td>22%</td>
<td>30%</td>
</tr>
<tr>
<td>NMBE</td>
<td>5%</td>
<td>7%</td>
<td>10%</td>
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</table>

Figure 8.2 Recommended Value for Baseline Model by ASHRAE (2014)
CHAPTER 9
MODEL VALIDATION

Experimental data was used to validate both Effectiveness–NTU and empirical models. The outputs of Effectiveness–NTU model are NTU and the effectiveness of air; the output of the empirical model is approach temperature. Both models were validated by comparing experimental measurements and the outputs from the models. Since the mass flow rate of air is difficult to measure in most circumstances, all models were modified to use the mass flow rate of air as an output, and a comparison was made between both models. The wet-bulb temperature was calculated using experimental measurements from HOBO data logger and the leaving water temperature was provided by Facilities Department. Since the sources of data are different, the approach temperature of the cooling tower shows negative values sometimes. The range of the negative values is from 0°F to -1°F, but the approach temperature should be greater than 0. Therefore, modifications of the data were made on negative approach temperature. Once the approach temperature shows less than 0, it was modified to 0. Then the leaving temperature of water at that time was changed equal to the wet-bulb temperature of the entering air.

9.0 Psychrometric Equations

Measuring the temperature and relative humidity of leaving air is limited by safety, accessibility, and accuracy. Instead of measuring the temperature of leaving air, a model was developed using psychrometric equations to calculate the temperature of leaving air. A validation process was done between the measured dry-bulb temperature and the calculated dry-bulb temperature of leaving air. Since the relative humidity of leaving air is almost 100%
from the measurements, the wet-bulb and dry-bulb temperature of leaving air can be determined if the relative humidity was assumed to stay constant at 100%. Figure 9.0.1 shows R-squared value is 0.9354 between measured and calculated dry-bulb temperature of the leaving air. Therefore, the model can be applied to calculate the temperature of leaving air if the relative humidity is 100%. The measured values were slightly higher than calculated value at daytimes because the relative humidity of leaving air seldom reaches 100% at daytimes in real life.

![Dry-bulb Temperature of Leaving Air](image)

**Figure 9.0.1 Dry-Bulb Temperature of Leaving Air**

9.1 Effectiveness-NTU Model

9.1.1 Both Fans Run at Full Speed.

According to Equation. 4.3, if NTU, mass flow rate of air and water were known, $c$ and $n$ value can be found using regression analysis. Since the variance of measured air velocity, the ratio between the mass flow rate of air and water varies from 1.2 to 2.8. However, the ratio should be constant because the mass flow rate of water and air is constant. Figure 9.1.1 shows the predicted $c$ and $n$ value based on the experimental data, and the correlation is less than 0.01. The regression analysis provided a horizontal trend line, which
cannot be used for prediction. Figure 9.1.2 shows the prediction based on \(c=0.5038\) and \(n=0.365\), and the result shows the NTU values are almost constant. Therefore, the Equation 4.3 is not recommended to use in this case because of lacking experimental measurements. Since the mass flow rate of air and water were constant during the experiment, one way to improve the estimation is taking more measurements from various ratios between the mass flow rate of air and water. Another way to improve the accuracy is minimum the variance of the measured airspeed.

![Figure 9.1.1 Prediction of c\&n value](image1)

![Figure 9.1.2 Compare Measured NTU and Predicted NTU](image2)

Since the trend line in Figure 9.1.1 cannot use to predict \(c\) and \(n\) value, other equations regarding the effectiveness of air were found. The effectiveness of air and approach
temperature can both be determined using experimental measurements. Figure 9.1.3 indicates the correlation between the approach temperature and the effectiveness of air.

\[
y = 0.4354x^{-0.267} \\
R^2 = 0.4703
\]

Figure 9.1.3 Compare Approach Temperatures and Effectiveness of Air 1

Figure 9.1.3 shows the R-squared value of the red trend line is 0.4703 between approach temperature and effectiveness of air. As the approach temperature decreases, the effectiveness of air rises. Therefore, a power equation was found at the condition of both fans running at full speeds:

\[
e_{air} = 0.4354 \times \text{Approach}^{-0.267}
\]

Eq. 9.1

In Figure 9.1.3, a power trend line of Equation. 9.1 is shown in red; an exponential trend line is shown in green; another linear trend line is shown in yellow. The linear equation is not applied in this case because the linear equation generates negative effectiveness of air once the approach temperature exceeds 8 °F. The exponential equation is not applied because the effectiveness of air should equal to 1 instead of 0.6 when the approach temperature is 0 °F. However, the exponential equation shows a reasonable trend when approach temperature is greater than 8 °F. The power trend line indicates the effectiveness of air nearly constant when the approach temperature is greater than 8 °F, but the effectiveness of air
should go to zero as approach temperature increases. A better prediction requires more data in winter when the approach temperature is larger, and the effectiveness of air is smaller.

The R-squared value between the effectiveness of air that calculated by Equation. 4.4 and Equation. 9.1 is 0.8434. The data implemented in this test was taken from August 25 to August 31, 2016, with a frequency of 5 minutes, and both of fans ran at high speed during this period. Another set of data with different fan running status was imported due to another running status in next section.

9.1.2 One Run in Full, One Stops.

This group of data was imported to represent another fan running status. The measurements were taken from June 21 to July 25, 2016, at a frequency of 5 minutes with one of the fans ran at full speed, and another stopped. The data lacks the temperature and the relative humidity of leaving air, so calculations based on Section. 9.0 were applied to determine the temperature of the leaving air at first. Then, the correlation between approach temperature and effectiveness of air were calculated and plotted.

![Approach and Effectiveness of Air](image_url)

**Figure 9.1.4 Compare Approach Temperatures and Effectiveness of Air 2**
Figure 9.1.4 shows the R-squared value is 0.7589 between approach temperature and effectiveness of air. A power equation was found when one fan runs at full speed, and another stops:

\[ e_{air} = 0.5061 \times \text{Approach}^{-0.193} \quad \text{Eq. 9.2} \]

The R-squared value between the effectiveness of air that calculated by Equation 4.4 and Equation 9.2 is 0.8109. The coefficients in the Equations 9.2 are slightly different from Equation 9.1 because the equations were predicted based on the different running statuses of fans. Therefore, a combination of both groups of data was made in next section.

9.1.3 Combination.

Both groups of data were imported to predict a general equation between approach temperature and effectiveness of air. The new power equation shows the R-squared value between the effectiveness of air and approach temperature is 0.4868 in Figure 9.1.5.

The power equation of combined data was found for the cooling tower:

\[ e_{air} = 0.4691 \times \text{Approach}^{-0.184} \quad \text{Eq. 9.3} \]

The power equation of the effectiveness of air above has the limitation because the effectiveness of air cannot exceed 100% theoretically. Therefore, the minimum approach
temperature can apply to this equation is 0.018 °F. Since the effectiveness of air increase while the approach temperature raises, the approach temperature less than 0.018 °F gives the effectiveness of air equals to 100% in this case.

Additionally, the relationship between approach temperature and relative humidity of entering air was found. The correlation of R-squared value was as weak as 0.13, but Figure 9.1.6 shows the approach temperature increases while the relative humidity rises. It means dryer entering air can lower the approach temperature and provide better performance to cooling towers.

![Figure 9.1.6 Relative Humidity and Approach Temperature](image)

9.1.4 Validation.

Data from May 24 to June 1, 2016, was imported to validate Equation 9.3. Since this data does not have measured temperature and humidity of leaving air either, a calculation according to Section 9.0 was applied to determine the dry-bulb temperature of leaving air and assumed the relative humidity is 100%. The R-squared value of effectiveness of air calculated using Equation 4.4 and Equation 9.2 is 0.8025. Figure 9.1.7 shows the result of comparisons. Since the correlation is strong enough, Equation 9.3 is ready to use in future estimations.
9.2 Empirical Model

According to the equation on Figure 5.2, approach temperature can be determined using empirical model. The input variables include the temperature of entering and leaving water, the temperature and the relative humidity of entering air, the YorkCalc coefficients, and the flow rate of air and water at design condition. The flow rate of air at design condition was set as 168 kg/s, which is the amount of mass flow rate of two fans running at full speed. The flow rate of water at design condition is 128 kg/s that is the sum of two pumps operating at capacity. The YorkCalc model was selected instead of CoolTools model because the YorkCalc model shows in Figure 5.3 has a larger range of limitation. The data was applied from August 25 to August 31, 2016, with a frequency of 5 minutes, and both of fans ran at high speed during this period.

Figure 9.2.1 shows a comparison between measured and calculated approach temperature using the equation in Figure 5.2. The R-squared value between two approach temperatures is 0.28. The R-squared value between the effectiveness of air from Effectiveness–NTU model is 0.8025, so the results from YorkCalc model shows less correlation than Effectiveness–NTU model. The reason is the YorkCalc model was designed
based on York’s cooling towers instead of the cooling tower tested at the campus. However, the YorkCalc model can still be used to predict mass flow rate of air as a reference. One way to improve the accuracy of the empirical model is to reset the 27 coefficients based on a specific cooling tower.

9.3 Mass Flow Rate of Air

Since the mass flow rate of air is dominated by the speed of fans, it has to be predicted before estimating energy consumption. If the entering and leaving water temperature, the entering air temperature, the relative humidity, and the entering and leaving water flow rate are known, the mass flow rate of air can be predicted. The equations used in the Effectiveness–NTU model are shown below. Based on the combination of Equation 4.4 and Equation 9.3, the difference between the enthalpy of leaving air can be calculated.

\[ h_{\text{air out}} = (0.4691 \times \text{Approach}^{-0.184}) \times (h_{\text{sat water in}} - h_{\text{air in}}) + h_{\text{air in}} \]  \hspace{1cm} \text{Eq. 9.4}

\[ m_{\text{air}} = \frac{cP_{\text{water}}(m_{\text{water in}}t_{\text{water in}} - m_{\text{water out}}t_{\text{water out}})}{h_{\text{air out}} - h_{\text{air in}}} \]  \hspace{1cm} \text{Eq. 9.5}

The temperature of entering air, entering water and leaving water were known from experimental measurements. The enthalpy of entering air and saturated air at water entering temperature can be calculated using Equation 3.5. Since the enthalpy of leaving air can be
calculated using Equation. 9.4, the mass flow rate of air based on energy balance can be determined using Equation. 9.5 shows above.

The equation used in Empirical-YorkCalc model is according to the theory of Empirical-YorkCalc model based on Figure 5.2:

\[ m_{\text{air}} = \frac{f_{\text{air}}}{m_{\text{air design}}} \]  
Eq. 9.6

where \( f_{\text{air}} \) is the airflow rate ratio and \( m_{\text{air design}} \) is the mass flow rate of the air at design condition.

Since both models were modified to calculate mass flow air as the outputs, Figure 9.3.1 shows the measured mass flow rate of air and the calculated mass flow rate of air using both models. The experimental measurements implemented in this comparison are from August 25 to August 31, 2016, with a frequency of 5 minutes. The R-squared value between the results of Effectiveness–NTU model and Empirical-YorkCalc model is only 0.12, but two groups of results have same trends. Both of the calculated results are the mass flow rate of air that the cooling tower needs to cool the water from the measured temperature of entering water to leaving water. The predicted results from both models show less mass flow rate of air than the measurements. The difference between measured and calculated results can be seen as the wasted mass flow rate of air.

![Figure 9.3.1 Compare Mass Flow Rate of Air](image)

Figure 9.3.1 Compare Mass Flow Rate of Air
10.1 Range Prediction

Range temperature represents the temperature between the entering and leaving water. From the previous measurements, the correlation was found between the dry-bulb temperature of the entering air and the range temperature of the cooling tower. The correlation shows when the dry-bulb temperature of air drops, the range of water temperature drops as well. Since all measurements from the experiment were taken at summertime, the range of dry-bulb temperatures only varies from 70 °F to 85 °F. The dry-bulb temperature was divided into groups by every 5 °F, and the average range temperature was found for each group. Table 10.1.1 shows the result of the dry-bulb temperature of the entering air and the average range temperature.

Table 10.1.1 Average Range Temperature from Measurement

<table>
<thead>
<tr>
<th>Variations of Dry-Bulb Temperature of Entering Air (°F)</th>
<th>Average Range (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>70 75</td>
<td>5.14</td>
</tr>
<tr>
<td>75 80</td>
<td>5.75</td>
</tr>
<tr>
<td>80 85</td>
<td>6.48</td>
</tr>
</tbody>
</table>

Once the average range temperature at different variations of entering air temperature was found, the relationship can be determined. Figure 10.1.1 shows a power equation between the lower bound of entering air dry-bulb temperature and the average range temperature. The R-squared value is 0.9968, and the equation of the trend line is:

\[ T_{range} = 0.0033 \times T_{lowerbound}^{1.7324} \]

Eq. 10.1.1

where \( T_{range} \) is the range temperature and \( T_{lowerbound} \) is the lower bound of entering air dry-bulb temperature.
Figure 10.1.1 Correlation of Average Range Temperature

In Figure 10.1.1, a linear trend line is shown in yellow, and the range temperature shows less than zero once the lower bound of entering air dry-bulb temperature below 30 °F. Since cooling towers can operate in cold temperature as low as 5 °F (SPX Inc., 2016) and the range temperature should greater than 0, the linear trend line cannot apply in this case. From the Equation. 10.1.1, the other average range temperature of the cooling tower can be estimated based on different variations of the dry-bulb temperature of entering air, and the result is shown in Table 10.1.2.
Table 10.1.2 Prediction of Range Temperature

<table>
<thead>
<tr>
<th>Variations of Dry-Bulb Temperature of Entering Air (°F)</th>
<th>Average Range (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>0.05</td>
</tr>
<tr>
<td>10</td>
<td>0.18</td>
</tr>
<tr>
<td>15</td>
<td>0.36</td>
</tr>
<tr>
<td>20</td>
<td>0.59</td>
</tr>
<tr>
<td>25</td>
<td>0.87</td>
</tr>
<tr>
<td>30</td>
<td>1.20</td>
</tr>
<tr>
<td>35</td>
<td>1.56</td>
</tr>
<tr>
<td>40</td>
<td>1.97</td>
</tr>
<tr>
<td>45</td>
<td>2.41</td>
</tr>
<tr>
<td>50</td>
<td>2.90</td>
</tr>
<tr>
<td>55</td>
<td>3.42</td>
</tr>
<tr>
<td>60</td>
<td>3.97</td>
</tr>
<tr>
<td>65</td>
<td>4.56</td>
</tr>
<tr>
<td>70</td>
<td>5.19</td>
</tr>
<tr>
<td>75</td>
<td>5.85</td>
</tr>
<tr>
<td>80</td>
<td>6.54</td>
</tr>
<tr>
<td>85</td>
<td>7.26</td>
</tr>
<tr>
<td>90</td>
<td>8.02</td>
</tr>
<tr>
<td>95</td>
<td>8.80</td>
</tr>
<tr>
<td>100</td>
<td>9.62</td>
</tr>
</tbody>
</table>

10.2 Predict Leaving Temperature

The relationship between the temperature of the entering air and leaving water is shown in the Figure 10.2.1. The experimental measurements indicate the temperature of leaving water increases when the temperature of the entering air rises. The linear equation of the trend line is:

\[ T_{\text{waterout}} = 0.1763 \times T_{\text{airin}} + 63.619 \]  
Eq. 10.2.1

where \( T_{\text{waterout}} \) is the temperature of leaving water and \( T_{\text{airin}} \) is the dry-bulb temperature of entering air. The reason to use linear function is the minimum leaving water of a cooling tower cannot be less than 55 °F (Johnson Control, 2015). A power trend line is shown in green and it exceeds the limitation of low-temperature condition.
10.3 Annual Weather Data

The weather data in Tuscaloosa, AL area in 2014 were provided by the weather station located on the roof of Hardaway Hall, and the measurements include both dry-bulb temperature and relative humidity. The frequency of the data is one hour, so there is a total of 8760 sets of data for a year. Wet-bulb temperature can be calculated using Equation. 3.1 based on the psychrometric equations.

10.4 Prediction

10.4.1 Effectiveness-NTU Model.

Since the annual wet-bulb temperature of the entering air can be calculated, the range of water temperature and the temperature of leaving water can be determined using Equation. 10.1.1 and Equation. 10.2.1. The range of water temperature and the temperature of leaving water were calculated and plotted in Figure 10.4.1.
Figure 10.4.1 Temperature of Leaving Water and Range Temperature

Since the temperature of leaving water and range temperature were known, the temperature of entering water of the cooling tower equals to:

\[ T_{\text{water in}} = T_{\text{water out}} + \text{Range} \quad \text{Eq. 10.4.1} \]

where \( T_{\text{water in}} \) is the temperature of entering water, and \( T_{\text{water out}} \) is the temperature of leaving water. The approach temperature can be calculated using:

\[ \text{Approach} = T_{\text{water out}} - T_{\text{wbin}} \quad \text{Eq. 10.4.2} \]

where \( T_{\text{wbin}} \) is the wet-bulb temperature of entering air. The effectiveness of air can be determined using Equation 9.3; the enthalpy of leaving air can be calculated using Equation 9.4. Since the enthalpy of leaving air of the cooling tower were found, the mass airflow rate can be calculated using following equation:

\[ m_{\text{air}} = \frac{C_{p_{\text{water}}}(m_{\text{water in}}T_{\text{water in}} - m_{\text{water out}}T_{\text{water out}})}{h_{\text{air out}} - h_{\text{air in}}} \quad \text{Eq. 10.4.3} \]

where \( C_{p_{\text{water}}} \) is the constant specific heat of water, and it equals to 4.178kJ/kg; \( m_{\text{water out}} \) is the constant mass flow rate of leaving water, which equals to 64.35kg/s. The constant makeup water flow rate equals to 0.03 kg/s, and the constant mass flow rate of entering water equals to 64.38 kg/s. The temperature of entering and leaving water was calculated earlier using Equation 10.2.1 and Equation 10.4.1. The enthalpy of entering air can be determined using
Equation 3.4. Figure 10.4.2 shows the predicted annual mass flow rate of air required by the cooling tower at the University of Alabama in 2014.

![Figure 10.4.2 Prediction of Mass Flow Rate of Air (Effectiveness)](image)

10.4.2 Empirical Model.

According to the equations in Figure 5.2, the mass flow rate of air can be predicted using the empirical model as well. The result calculated by Equation 9.4 is shown in Figure 10.4.3. The mass flow rate of water at design condition equals to 128 kg/s, and mass flow rate of air at design condition equals to 168 kg/s. The accuracy of the empirical model was discussed in Section 9.2. The following result cannot accurately predict the annual mass flow rate of air required by the cooling tower at the University of Alabama in 2014.

![Figure 10.4.3 Prediction of Mass Flow Rate of Air (Empirical)](image)
10.5 Different Fan Conditions

The cooling tower used in this project has two of 2-speeds fans. Each of the fans can operate in three statuses: high, low, and stop. Air velocity of each fan was measured during the experiment, and the mass flow rate was calculated. Since the power of the fans at capacity was recorded, power of the fans in different conditions can be found using mass flow rate of air based on the Affinity Law (Whitesides, 2012):

$$\left(\frac{Q_1}{Q_2}\right)^3 = \frac{P_1}{P_2}$$  \hspace{1cm} \text{Eq. 10.5.1}

where $Q$ is the mass flow rate, and $P$ is the power. The power of the fans was calculated for each status, and the result is shown in Table 10.5.1. Since two fans equipped on the cooling tower, there are six combinations of running statuses, and each of them has a different mass flow rate of air and power. Table 10.5.2 shows the mass flow rate of air and power of each running status.

Table 10.5.1 Single Fan Conditions

<table>
<thead>
<tr>
<th>Fan Condition</th>
<th>Mass Air Flow Rate (kg/s)</th>
<th>Fan Power (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>High</td>
<td>80.00</td>
<td>18.00</td>
</tr>
<tr>
<td>Low</td>
<td>32.00</td>
<td>1.15</td>
</tr>
<tr>
<td>Stop</td>
<td>8.00</td>
<td>0.00</td>
</tr>
</tbody>
</table>

Table 10.5.2 Dual Fans Conditions

<table>
<thead>
<tr>
<th>Fan 1-Fan 2</th>
<th>Mass Air Flow Rate (kg/s)</th>
<th>Fan Power (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>High-High</td>
<td>160.00</td>
<td>36.00</td>
</tr>
<tr>
<td>High-Low</td>
<td>112.00</td>
<td>19.15</td>
</tr>
<tr>
<td>High-Stop</td>
<td>88.00</td>
<td>18.00</td>
</tr>
<tr>
<td>Low-Low</td>
<td>64.00</td>
<td>2.30</td>
</tr>
<tr>
<td>Low-Stop</td>
<td>40.00</td>
<td>1.15</td>
</tr>
<tr>
<td>Stop-Stop</td>
<td>16.00</td>
<td>0.00</td>
</tr>
</tbody>
</table>

Because the annual mass flow rate and dual fans running statuses were both predicted, an annual prediction of the fan running statuses can be found in Figure 10.5.1 based on the predicted mass flow rate of air. For example, if the predicted mass flow rate of air is between 160 kg/s and 112 kg/s, the cooling tower requires both fans run at high speed. If the predicted
mass flow rate of air is less than 16 kg/s, the cooling tower can use natural convection by shutting down both fans.

Figure 10.5.1 Annual Fan Running Status

According to Affinity Law, the annual energy consumption of current 2-speeds fans is 55,265 kWh, and the annual energy of variable speed fans is 33,205 kWh. The energy consumption will be 302,328 kWh if both of the fans run at high speed for the whole year. A comparison is shown in Table 10.5.3. As a result, variable speeds fans can save nearly 20,000 kWh for a year. Compared to single speed fans, 2-speeds fans can save close to 100,000 kWh for a year.

Table 10.5.3 Compare Energy Consumption

<table>
<thead>
<tr>
<th></th>
<th>Single-Speed Fans</th>
<th>2-Speeds Fans</th>
<th>Variable Speeds Fans</th>
</tr>
</thead>
<tbody>
<tr>
<td>Annual Energy Consumption (kWh)</td>
<td>203,742</td>
<td>106,514</td>
<td>87,502</td>
</tr>
</tbody>
</table>

As a comparison, the energy consumption of a similar cooling tower was found. The data was generated by the software called UPDATE (SPX Cooling Technologies, Inc., 2016). The specifications and energy consumption were listed in Figure 10.5.2 (SPX Cooling Technologies, Inc., 2016).
The difference in energy consumption between single-speed fans and 2-speeds fans are huge, and 2-speeds fans with 6 stages of capacity controls are good enough for most of the cooling towers. Although the result shows the variable speeds fans can save nearly 20,000 kWh, a benchmarking needs to be done to find out if it is worth to switch from 2-speeds fans to variable speed fans because variable-speed motors cost much more.
CHAPTER 11

CONCLUSION

The goal of this project is to investigate and compare the performance of cooling towers using Effectiveness-NTU model and empirical model. The processes to achieve the goal include: Developed Effectiveness-NTU and empirical models of the cooling tower, and predicted the performance of the design and off-design conditions; Stated experimental protocols and gathered data on the HVAC cooling tower on the campus of the University of Alabama; Used collected data to validate the models; Compared results from models with real measurements and found the limitation of each model; Applied known annual weather data to estimate the performance and energy consumption of the cooling tower; Recommended the approach for the best energy and heat performance.

All data and specifications were gathered and measured from the experiment on campus. In fact, the mass flow rate of air and the temperature of leaving air are not always possible to gather from different cooling towers, especially industry cooling towers. Therefore, both models were designed to predict mass flow rate of air, and the temperature of leaving air by applying the temperature and the relative humidity of entering air. Fan status and energy consumption of cooling tower were predicted according to the mass flow rate of air. Total annual energy consumption of single-speed, 2-speeds, and variable-speed cooling towers were calculated.

As a result, the NTU equation in the Effectiveness-NTU model is not applied in this case, and another model regards effectiveness of air was built and tested. The effectiveness model was built based on a regression analysis of the experimental data. The R-squared value was 0.8 between the measured data and the predicted data. More data from lower outside air
temperature is needed to improve the accuracy. The empirical model built based on YorkCalc correlations provided approach temperature, and the result has larger variances compared to measured data. Another set of correlations relates to the testing cooling tower needs to be developed.

A dual-speed cooling tower can save 50% of the energy, and a variable-speed cooling tower can save 60% compared to a single-speed cooling tower. This result only applies to the cooling tower used in the experiment, and the results of other cooling towers may vary. The recommendation on choosing fan speeds for a cooling tower should consider: Cooling tower’s life; Cost of energy; Cost of equipment; Payback period.

Further testing is required to validate the accuracy of the models because there was a limited control over the running status of fans. More experimental data needs to be collected in wintertime with the lower temperature of entering air. Validations on other cooling towers are essential in the future. Additionally, the accuracy of the empirical model can be improved by resetting all coefficients.
REFERENCES


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