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A MODELING TOOL TO ANALYZE THE PERFORMANCE OF INDUSTRIAL COOLING TOWERS

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ABSTRACT

Energy efficiency and energy savings have become an important factor as industries look for ways to save energy and minimize their energy consumption while reducing their carbon footprint. Cooling towers are utilized significantly in industries for either serving chillers or process cooling. Depending on the size of the cooling tower, it can use a surprising amount of energy and water, which is why it is crucial to make sure that the facility has optimized their cooling tower.

A modeling tool has been developed to perform a thorough analysis of a cooling tower and its various operations to ensure that the facility has optimized its cooling tower. This model analyzes an annual base case in comparison with the revised case of a cooling tower operation. This modeling tool simulates 8,760 hourly calculations for fan power for various fan controls, water consumption, and pumping energy consumption and demand based on the user's location and its corresponding Typical Meteorological Year 3 (TMY3) weather data. This model is capable to simulate up to five cooling tower cells as one large tower utilizing one pump or parallel pumping. The successful validated cooling tower model should assist industry to save energy at their facility through their cooling tower, whether they use it for process cooling or heating, ventilation and air conditioning (HVAC) applications. The entering/leaving water temperatures based on a monthly operating schedule or a wet-bulb temperature schedule will be used as inputs, and the model can calculate savings in energy for various conditions including variable-frequency drive (VFD) on the fan/pump, drift eliminators, number of cycles of concentration, reduced water flow rate, etc.

NOMENCLATURE

a	area of water interface (sq ft/cu ft)
AF	inlet mass air flow (lbm/min)
B	blowdown (gal/hr)
C.O.C.	cycles of concentration
c_p	specific heat of water (BTU/lbm*°F)
D	drift loss (gal/hr)
d_{q_s}	rate of sensible heat transfer, interface to air stream (BTU/hr)
d_{q_w}	rate of heat transfer, bulk water to interface (BTU/hr)
E	evaporation loss (gal/hr)
ECS	energy consumption savings (kWh)
EDS	energy demand savings (kWh)
f	cooling tower capacity factor
f_1	rating factor for cooling tower capacity, function of approach and wet-bulb temperature
f_2	rating factor for cooling tower capacity, function of range and wet-bulb temperature
f_3	rating factor for cooling tower capacity, function of fan air flow
$flow_{air}$	percent cooling tower air flow rate (%)
G	air flow rate (lb dry air/hr)
h	enthalpy of moist air (BTU/lb dry air)
h"	enthalpy of moist air at interface temperature (BTU/lb dry air)
H	absolute humidity (humidity ratio) of main air mass (lb vapor/lb dry air)
H'	absolute humidity saturated at water temperature
HR	absolute humidity ratio
K'	unit conductance, mass transfer, interface to main air stream (lb/(hr)(sq ft)(lb/lb))
K_G	overall unit conductance, sensible heat transfer between interface and main air stream (BTU/(hr)(sq ft)(°F))
K_L	unit conductance, heat transfer, bulk water to interface (BTU/(hr)(sq ft)(°F))
L	mass water rate (lb/hr)
L_E	mass evaporation loss (lb/hr)
L/G	liquid to gas ratio
m	mass-transfer rate, interface to air stream (lb/hr)

$\dot{m}$ mass flowrate (lbm/min)
N fan speed (RPM)
P power (kW)
P_{atm} atmospheric pressure (psia)
P_{one_speed} fan power of one-speed control (HP)
P_{two_speed} fan power of two-speed control (HP)
P_v water vapor partial pressure (psia)
P_{VFD} fan power of VFD control (HP)
Q flow (GPM)
r latent heat of evaporation, assumed constant in system
ratedPower total power of user fan power input (HP)
RH relative humidity (%)
SPP simple payback period (yrs)
t bulk water temperature (°F)
t_1 bulk water temperature at inlet (hot water) (°F)
t_2 bulk water temperature at outlet (cold water) (°F)
T dry-bulb temperature of air stream (°F)
T' dry-bulb temperature of air at interface (°F)
T_{app} approach temperature (°F)
T_{db} dry-bulb temperature (°F)
T_{dp} dew point temperature (°F)
T_{high} time fan speed is spent on high speed (%)
T_{low} time fan speed is spent on low speed (%)
T_{in} entering water temperature (°F)
T_{out} leaving water temperature (°F)
T_{rng} range temperature (°F)
T_{WB} wet-bulb temperature (°F)
TDC tower design capacity (MMBTU/hr)
V active cooling tower volume (cu ft/sq ft plan area)
$\%_{AF}$ percentage of hourly air flow (%)
$\%_{MWF}$ percentage of monthly water flow (%)

Greek Symbols

η_{motor} motor efficiency (%)
η_{pump} pump efficiency (%)
η_{VFD} VFD efficiency (%)
ρ_{air} air density (lb/ft^3)
ρ_{H2O} water density (lbm/gal)

	English Unit	**SI Unit**
Temperature	°F	°C
Length	ft	m
Pound	lbs	kg
Pressure	psi	kg/m^2
Enthalpy	BTU/lbm	J/kg
Density	lbm/ft^3	kg/m^3
Specific Volume	ft^3/lbm	m^3/kg
Air Flow	lbm/min	kg/min
Air Flow	CFM	m^3/hr
Water Flow	GPM	L/s
	gal/hr	L/hr
Power	hp	kW
	MMBTU/hr	MW
Co_2 Emission Rate	lbs/MWh	g/MJ

1. INTRODUCTION

The purpose of a cooling tower is to cool down water that has been heated up due to industrial processes or air conditioning. A cooling tower operates as a heat exchanger that allows water and air to interact with one another to lower the temperature of the hot water. This is achieved mostly through evaporation as the water circulates through the tower. Typically, the water is pumped to the top of the tower where it is sprayed across the cooling tower fill, usually a Polyvinyl chloride (PVC) medium, which allows the water to form thin flowing streams as it falls down the tower through the fill. Simultaneously, air is being pulled across the fill and out the top of the tower, thus allowing the interaction between the water and air which will lead to the water cooling to the desired set point. This creates a simple means for the user to cool their water to return back to the system.

Cooling towers range in size and kind depending on the cooling load for the building [1]. They can be found at various manufacturing facilities, power plants, hospitals, and universities [1]. The most common application of a cooling tower is in heating, ventilation and air conditioning (HVAC) systems [1]. It can also be used in process cooling in industrial facilities [1].

There are two main types of cooling towers: natural and mechanical draft. The main difference between them is that a natural draft cooling tower operates using natural convection to push hot air out and pull cool air in. A natural draft tower is typically used for a much higher flow rate. A mechanical draft tower will either be induced or forced draft. Induced draft will have a fan at the discharge to pull air into the system, and a forced draft will have a fan at the intake to push air into the system. A mechanical draft cooling tower can also be classified as either being a counterflow or a crossflow tower. A counterflow cooling tower will allow air to flow in the opposite direction of the water. The water will fall downward while the air flows vertically upward. A crossflow cooling tower will have air flow horizontally, or perpendicular, to the water as it flows downward [1].

Energy efficiency and energy savings have become an important factor as industries look for ways to save money and minimize their energy consumption while reducing their carbon footprint. Often times the main focus of industrial energy efficiency will be on lighting, compressed air, furnaces, or other large industrial components. Cooling towers often get overlooked when trying to find potential ways to make a facility more energy efficient [2]. This is due to the minimal ways available to quantify and project the anticipated savings from various energy savings projects. Depending on the size of the cooling tower, it can use a surprising amount of energy and water, which is why it is crucial to make sure that the facility has optimized their cooling tower(s) [2].

The cooling tower modeling tool developed in this study has been created to perform a detailed analysis of a mechanical draft cooling tower and its various operations, which will then allow the user to see potential ways to optimize their cooling tower. The model is a macro-enabled excel file that analyzes an annual base case in comparison with the revised case of cooling tower operation. This modeling tool simulates 8,760 hourly calculations for fan power for various fan controls, water consumption, and pumping energy consumption and demand based on the user's location and its corresponding Typical Meteorological Year 3 (TMY3) weather data. The model simulates up to five cooling tower cells as one large tower utilizing one pump or parallel pumping. Through a collaboration with Oak Ridge National Laboratory (ORNL), some eQuest curve fits in analyzing various fan controls in cooling towers were incorporated in the model [3]. In order to run complex scenarios and conditions, the model utilizes the Visual Basic for Applications programming language that Excel offers. This reduces the size of the program and allows for more complex scenarios and equations to generate every hourly condition of the year depending on its individual weather data. The program will then give a summary of the projected annual savings for fan power, water, and pumping results as well as carbon reduction based on the user's inputs for their base and revised cases.

The model could be extremely useful to the industry as they analyze ways to save energy and reduce cost at their facility through their cooling tower, whether they use it for process cooling or HVAC applications. Users will be able to input their entering/leaving water temperatures based on a monthly operating schedule or a wet-bulb temperature schedule. For the analysis, they will be able to tell the program whether or not the base and revised case has a variable-frequency drive (VFD) on the fan/pump, drift eliminators, number of cycles of concentration, reduced water flowrate, etc. This will allow the user to visualize how they could operate their cooling tower differently in order to save energy and reduce cost, and in turn, this will allow the user to be able to determine the payback period for various projects to optimize their cooling tower.

The biggest benefits to this program compared to that of other programs is its ease of use due to the model being an Excel-based program, which allows a flat learning curve for the average user. Also, this program is able to model cooling towers' energy and water performance for various operational scenarios in manufacturing settings without having to build the whole system, including air side equipment and chillers, which is common in other modeling tools such as EnergyPlus [4] and eQuest's CoolTools [5]. Thus, the model presented in this paper can allow for quick and detailed cooling tower calculations without extensive system modeling.

This particular paper will focus on the modeling and theory behind this modeling tool which is used to analyze the performance of industrial cooling towers. Additionally, it will analyze the results and potential savings for a cooling tower that has been simulated using this program, thus showing the simplicity of the model and its ease of use for the industry as they look to optimize their cooing tower. It will also express the novelty of this program due to the power of the program by its usage of unique hourly weather data calculations to model fan power consumption, pump power consumption and water consumption for various operation scenarios encountered in manufacturing settings.

2. LITERATURE REVIEW

The purpose of the literature review is to analyze previous works concerning cooling tower modeling and operation. Specifically, it will explore the technical equations of Merkel [6].

2.1 Merkel Theory

Cooling towers have been an intriguing topic for researchers for many years due to their complex nature. The early investigators of cooling tower theory grappled with the problem given by the dual transfer of energy and mass. Merkel [6] was able to combine these two into a single process based on enthalpy potential, which is known as the Merkel Equation [6]. The Merkel Equation was developed in 1925 and is widely accepted as a general concept of cooling tower performance. The Merkel equation is able to combine sensible and latent heat transfer terms into a system that focuses on an enthalpy change from the low temperature fluid to a high temperature fluid [6]. "A Comprehensive Approach to the Analysis of Cooling Tower Performance," illustrates this process in Fig. 1 [6].

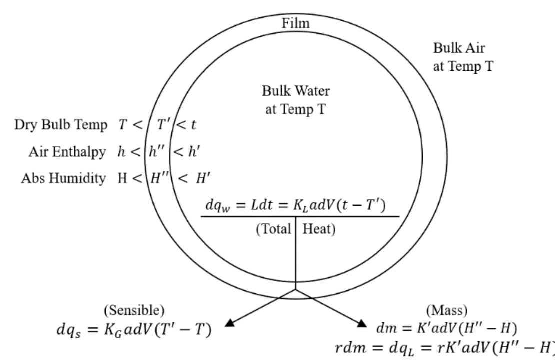

FIGURE 1: MASS AND HEAT TRANSFER BETWEEN WATER, FILM, AND AIR [4]

Referring to the schematic shown in Fig. 1, Baker and Shryock state that "each particle of the bulk water in the cooling tower is assumed to be surrounded by an interface to which heat is transferred from the water [6]. This heat is then transferred from the interface to the main air mass by a transfer of sensible heat, and by the latent heat equivalent of the mass transfer resulting from the evaporation of a portion of the bulk water." We can then show the process leading to the Merkel equation by first defining an equation that quantifies the rate of heat transfer (d_{q_w}) from the water to the film by Equation 1, by utilizing the mass water rate (L) times the temperature difference of the water (dt), unit conductance (K_L), area of water interface (a), active cooling tower volume (V), bulk water temperature (t), and the dry-bulb temperature of air at interface (T'). It should be noted that Baker and Shryock do not include the specific heat of water,

c_p, in the equations discussed in this section since it has been assumed to be 1.0

$$d_{q_w} = Ldt = K_L adV(t - T') \tag{1}$$

From there a portion of the heat can be transferred to the bulk air from the main air stream as sensible heat(d_{q_s}), which is then represented by Equation 2, using the overall unit conductance (K_G), and the dry-bulb temperature of air stream (T).

$$dq_s = K_G\ a\ dV(T' - T) \tag{2}$$

Any resistance to the mass transfer from the water to the interface (K') can be ignored, but mass transfer of vapor from the film to the air (dm) should be considered, which can be seen below by utilizing the absolute humidity ratio of main air mass (H) and the absolute humidity saturated at water temperature (H').

$$dm = K'a\ dV(H' - H) \tag{3}$$

The latent heat transfer of evaporation (r) is equivalent to mass transfer and is transferred as a result of evaporation of the bulk water to the bulk air mass. From Equation 1, the mass rate is converted to heat rate to produce Equation 4.

$$rdm = rK'adV(H' - H) \tag{4}$$

From there, Merkel claims the two heat rates can be set equal since the heat lost by water must be equal to the heat gained by the air, thus resulting in the Merkel equation seen below which utilizes the air flow rate (G), the enthalpy of moist air (h), and the enthalpy of moist air at the interface (h") [4].

$$Ldt = K^{'}adV(h'' - h) = G\ dh \tag{5}$$

Merkel's equation [6] assumes negligible water loss due to evaporation; however, in order to perform a proper heat balance on the system, evaporation must be considered. In steady state conditions, the rate of mass leaving the water by evaporation equals the rate of humidity increase of the air which can be seen in the equation below.

$$dm = G\ dH \tag{6}$$

The heat loss by the water equals that of the heat gained by the air. Therefore, the resultant heat balance can be seen in Equation 7 where L_E is the mass evaporation loss and t_1 and t_2 are the bulk water temperatures at the inlet and the outlet.

$$G\ dh = L\ (t_1 - 32) - (L - L_E)(t_2 - 32) \tag{7}$$

In Equation 7, the entering and leaving water temperatures, t_1 and t_2 respectively, are being subtracted by 32 in order to create an expression of enthalpy difference in the water flow.

$$G\ dh = Ldt - L_E\ (t_2 - 32) \tag{8}$$
$$L_E = G\ dH \tag{9}$$
$$G\ dh = L\ dt - G\ dH(t_2 - 32) \tag{10}$$

Equation 10 represents the heat required to raise the liquid water from the 32º F datum point to the cold-water temperature. This heat of liquid is usually ignored and the equation then becomes:

$$G\ dH = L\ dt \tag{11}$$

For a full breakdown of the Merkel equations see reference [6].

3. METHODOLOGY

One of the main goals of the presented model in this paper was to provide a simple program for an industry user to test the efficiency of their cooling towers and allow them to see the potential savings of different operating methods and their payback period. The program will dissect the fan power for various fan controls, i.e., VFD, two-speed, or single-speed control, water consumption, and the energy consumption and demand for the water pumps. In order for the program to run, the user will need to input the design conditions for their cooling tower along with the operational conditions. They will also have the option to change various inputs that the model will analyze for the revised case of their operational cooling tower. A flow diagram depicting the how the program functions can be seen in Fig. 2. This section will break down the methodology used in this program to generate the necessary calculations used to predict the energy savings for the user's cooling tower.

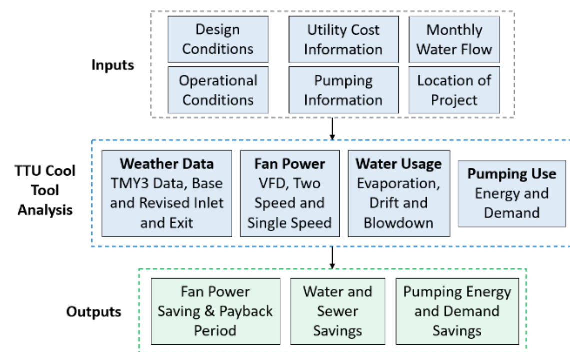

FIGURE 2: THE MODELING TOOL FLOW DIAGRAM

3.1 Typical Meteorological Year 3 (TMY3) Weather Data

A cornerstone of the model is its use of TMY3 weather data. TMY3 weather data "is a set of meteorological data with data values for every hour in a year for a given geographical location. The data are selected from hourly data in a longer time period (normally 10 years or more). For each month in the year the data have been selected from the year that was considered most 'typical' for that month. For instance, January might be from 2007, February from 2012 and so on" [7]. This data is for "1020 locations in the USA including Guam, Puerto Rico, and US Virgin Islands, derived from a 1991-2005 period of record" [8]. These locations can be seen in Fig. 3. The TMY3 weather data that has been implemented into this program was obtained from National Solar Radiation Database (NSRDB) [9]. This data allows for the program to simulate unique hourly conditions of a cooling tower for every hour of the year for a given location, thus providing precise results to the user.

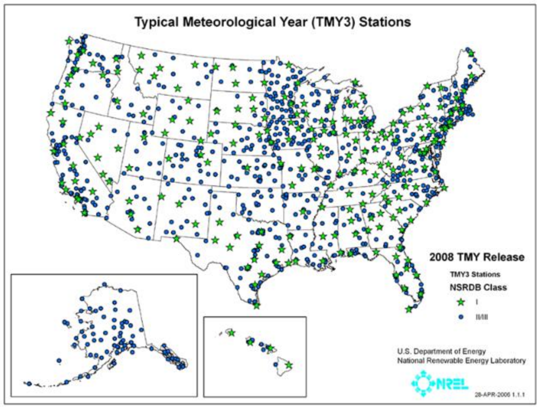

FIGURE 3: TMY3 GEOGRAPHIC COVERAGE [9]

3.2 Weather Calculations

After the user has inputted their conditions, the program will run through approximately 500,000 unique calculations in order to predict the savings for the user's cooling tower. The first thing that the program must do is to bring in the user's TMY3 weather data files. The TMY3 Excel file will contain 8,760 unique hourly conditions for the location's Dry-bulb Temperature (°F), Relative Humidity (%), Atmospheric Pressure (psia), Wet-bulb Temperature (°F), and Enthalpy (BTU/lbm). Wet-bulb temperature and enthalpy are not given from the TMY3 weather data; therefore, these conditions were calculated via psychrometric calculators, and their numerical results were stored in their individual excel files.

Using the TMY3 data, the program will calculate the inlet weather conditions of the user's cooling tower. First, the Water Vapor Partial Pressure (psia) is found through utilizing a curve fit equation. This curve fit is generated by plotting the saturated water pressures against their respective dry-bulb temperatures. There is a plot for temperatures greater than 0 °F and another plot for less than 0 °F. The program will view the TMY3 dry-bulb temperature in order to determine the appropriate curve. In order to calculate the appropriate water vapor partial pressure, the program implements the TMY3 relative humidity into the curve fit equation to generate the water vapor partial pressure.

From there the Absolute Humidity Ratio can be found for the inlet weather conditions. The absolute humidity ratio is calculated using the previously found water vapor partial pressure, P_v, and the TMY3 atmospheric pressure, P_{atm}. In order to calculate the absolute humidity ratio, the program uses the following equation:

$$Absolute\ Humidity\ Ratio = \frac{0.621945 \times P_v}{(P_{atm} - P_v)} \tag{12}$$

Following the absolute humidity ratio, the Air Density (lbm/ft^3) will be calculated by Equation 13 which will utilize the previously found absolute humidity ratio (HR).

$$HR = \frac{(P_{atm} - P_v) \times 144}{\left(53.35 \times (T_{db} + 459.67)\right) \times (1 + HR)} \tag{13}$$

Additionally, the program will determine the Specific Volume (ft^3/lbm) of the inlet weather condition by using the TMY3 air density, ρ_{air}, and the absolute humidity ratio which was defined above. This condition can be calculated via Equation 14.

$$Specific\ Volume = \frac{1}{\frac{\rho_{air}}{1 + HR}} \tag{14}$$

The last inlet weather condition that must be considered is the mass Air Flow (lbm/min), which can be seen below in Equation 15. In order to determine the mass air flow, the total design inlet air flow must be considered. This will be taken from the design conditions for all of the cooling tower cells combined, which is provided by the user on the input page.

$$Air\ Flow = \frac{Total\ Design\ Inlet\ Air\ Flow}{Specific\ Volume} \tag{15}$$

Following the inlet weather conditions, the program will then calculate the base exit tower conditions followed by the revised exit tower conditions based on the user's inputs. These two sections will have identical components with the only variance being the source of the variables used in their respective equations depending on whether it is for the base case or the revised case. The first condition that is needed for either case is the Exit Dry-bulb Temperature (°F). The complex nature and path of the air conditions in a cooling tower can be represented in Fig. 4 [10].

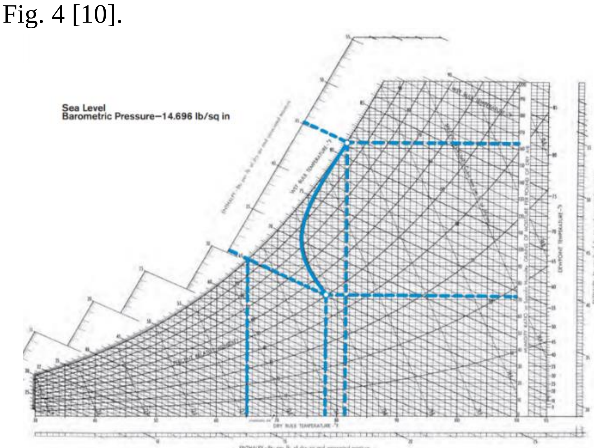

FIGURE 4: ESTIMATED AIR CONDITIONS THROUGH A COOLING TOWER [10]

Therefore, taking into consideration Marley's estimations [6], we are able to incorporate various assumptions in the calculations to predict the exit air temperature for a cooling tower. In order to calculate the exit dry-bulb temperature, there are multiple variables that must be calculated first. First, the Water Flow (GPM) must be obtained for each hourly condition. This water flow can be expressed by the percent of the monthly water flow, found in the user's inputs, multiplied by the total design flowrate of all the cooling tower cells. This relationship can be expressed in Equation 16.

$$Flow = Monthly\ Water\ Flow \times Total\ Design\ Flowrate \tag{16}$$

From there the program needs the mass Water Flow (lbm/min) through the tower which will be expressed in Equation 17. The program will use the previous water flow found in Equation 16 and the operational water density, ρ_{H2O}, which is taken from the operational water condition schedule of the user, to determine the mass water flow.

$$Flow = Water\ Flow \times \rho_{H2O} \tag{17}$$

Next, the program will determine the Liquid to Gas ratio, which can be defined as the ratio between the water and air mass flow rates which can be expressed as the following:

$$L/G\ Ratio = \frac{Water\ Flow}{Air\ Flow} \tag{18}$$

The L/G ratio can be further defined as the ratio of the change in enthalpy of the air over the change in the entering and leaving water temperatures. Therefore, in order to determine the exit air Enthalpy (BTU/lbm) we will utilize Equation 19, where h_1 is the entering enthalpy, t_{in} and t_{out} are the entering and leaving water temperatures, respectively, and the specific heat of water, c_p, which is assumed to be 1.0.

$$Enthalpy = h_1 + \frac{L}{G} \times c_p \times (t_{in} - t_{out}) \tag{19}$$

"It is assumed that exiting air leaves the tower saturated (100% RH)" [11]. Therefore, using the estimated exit enthalpy, the program uses a lookup table to find the corresponding exit dry-bulb temperature at saturation, i.e., 100% relative humidity, which has been generated in accordance to the average atmospheric pressure for the location of the user. After the exiting dry-bulb temperature has been generated, the program will continue with the water vapor partial pressure, absolute humidity ratio, air density, specific volume, and air flow for the exit conditions.

3.3 Fan Power Calculations

Following the inlet and exit weather calculations, the model will calculate the fan energy consumption results for various fan controls, i.e., it will be comparing the efficiency and savings of a variable frequency drive to a two-speed fan and to a single-speed fan. The eQuest fan power models were used for cooling tower fan power calculations [5]. These fan power models have been used by many engineers to model open cooling tower energy performance and some of these curve fits were studied by Benton and other researchers [5].

Initially, the first condition that must be calculated in order to determine the fan results would be to calculate the Approach (°F), or Equation 20.

$$Approach = T_{out} - T_{wb} \tag{20}$$

Following the approach, there are multiple rating factors that must be calculated in order to determine the necessary fan power results. The steps in order to find these rating factors is expressed in the following.

Equations/Calculations

Steps: (a) Find rating factor (f_1) for cooling tower capacity as a function of approach and wet-bulb temperature. The following multivariable polynomial function is used.

$$f_1 = C_0 + C_1 \times T_{app} + C_2 \times \left(T_{app}\right)^2 + C_3 \times T_{WB} + C_4 \times (T_{WB})^2 + C_5 \times T_{app} \times T_{WB} \tag{21}$$

Where,
T_{app} = Approach temperature
T_{WB} = Ambient wet-bulb temperature
C_0, C_1, C_2, C_3, C_4, C_5 are constants and as follows:

C_0	C_1	C_2	C_3	C_4	C_5
0.50061393	0.00588251	0.0002163	-0.01913189	0.0002236	0.00106108

(b) Find rating factor (f_2) for cooling tower capacity as a function of range and wet-bulb temperature. The following multivariable polynomial function is used.

$$f_2 = C_0 + C_1 \times T_{rng} + C_2 \times \left(T_{rng}\right)^2 + C_3 \times T_{WB} + C_4 \times (T_{WB})^2 + C_5 \times T_{rng} \times T_{WB} \tag{22}$$

Where,
T_{rng} = Range temperature
T_{WB} = Ambient wet-bulb temperature
C_0, C_1, C_2, C_3, C_4, C_5 are constants and as follows:

C_0	C_1	C_2	C_3	C_4	C_5
0.08352359	0.11247273	-0.00135847	0.00003417	0.00003125	-0.00034001

(c) Calculate cooling tower capacity factor (f) as follows:

$$f = \frac{f_1}{f_2} \tag{23}$$

Where,
f_1 = Rating factor for cooling tower capacity as a function of (T_{app}, T_{WB})
f_2= Rating factor for cooling tower capacity as a function of (T_{rng}, T_{WB})

For clarity on cooling tower capacity factor, consider the following statement. *"When cooling tower capacity, f = 2.0, the cooling tower can produce twice the cooling effect with the existing conditions"* [5]. In such conditions, the fan power can be reduced to obtain a cooling tower capacity factor of 1.0. This relationship can be mathematically formulated using the following steps.

After the model discussed in this paper calculates the above rating factors to generate the Cooling Tower Capacity Factor, f, the program will then divide the capacity factor by the monthly water flow percentage depending on if it is for the base or revised case. This will ensure that the appropriate capacity factor is calculated due to the monthly water flow percentage, which will then guarantee that the fan energy will be reduced if the monthly water flow is reduced. Following the capacity factor, the Tower Capacity (MMBTU/hr) for each hourly condition can be calculated, Equation 25, in relation to the capacity factor that was previously found by the eQuest curve fits and the tower design capacity (MMBTU/hr). The tower design capacity, TDC, is calculated from the input page based upon the design conditions inputted by the user, which is expressed by Equation 24. It can be assumed that the specific heat of water, c_p, is equal to 1.0. The water density (lbm/gal) is configured by using a curve fit equation of water densities at their respective temperatures and inputting the user's average water temperature to determine the appropriate water density (lbm/gal).

$$TDC = \frac{60\, Flowrate \times \rho_{H2O} \times Range \times c_p}{1{,}000{,}000} \tag{24}$$

The Tower Capacity (MMBTU/hr) is also adjusted by the monthly water flow percentage, $\%_{MWF}$, for the same reasons as the capacity factor, which is mentioned above.

$$Tower\ Capcity \left[\frac{MMBTU}{hr}\right] = \frac{TDC \times f}{\%_{MWF}/100} \tag{25}$$

From there, we are able to continue with eQuest equations to calculate fan energy for the various fan controls. This continuation is best described below:

Let f_3 be the rating factor for cooling tower capacity as a function of fan air flow.

$$f \times f_3 = 1.0 \tag{26}$$

Therefore,

$$f_3 = \frac{1}{f} \tag{27}$$

The polynomial function representing f_3 is denoted as follows:

$$f_3 = C_0 + C_1 \times flow_{air} + C_2 \times (flow_{air})^2 \tag{28}$$

Where,
$flow_{air}$ = % cooling tower air flow rate
C_0, C_1, C_2 are constants and take the following values.

C_0	C_1	C_2
0.049768250	1.04669762	-0.09646816

(d) Use quadratic formula to solve for % air flow rate as follows:

$$f_3 - \frac{1}{f} = 0 \tag{29}$$

$$0.049768250 + 1.04669762 \times flow_{air} - 0.09646816 * (flow_{air})^2 - \frac{1}{f} = 0 \tag{30}$$

One can use the the quadratic formula to solve the Eq. (30). When $ax^2 + bx + c = 0$, a, b, c are constants, and x is unknown.

Note that only the solution resulting from the positive discriminant is valid. Airflow and fan speed have been assumed to have a directly proportional relationship with a constant of proportionality equal to 1. The cooling tower performance calculator has the capability to estimate fan power at operating conditions for variable speed, two speed and single speed fans.

(e) Calculate fan energy consumption/savings as follows:

<u>Variable Speed</u>

For variable speed fans, fan power can be calculated using a cubic polynomial which defines fan power as a function of airflow. This function yields a correction factor for fan power at off-design airflow rates, which can be utilized in calculating the fan power requirement. The calculation procedure is as follows:

Let f_4 be the correction factor for fan power as a function of airflow. The polynomial function representing f_4 is denoted as follows:

$$f_4 = C_0 + C_1 \times flow_{air} + C_2 \times (flow_{air})^2 + C_3 \times (flow_{air})^3 \tag{31}$$

Where,
$flow_{air}$ = % cooling tower air flow rate
C_0, C_1, C_2, C_3 are constants and take the following values.

C_0	C_1	C_2	C_3
0.01055507	-0.05704023	0.14686301	0.92961746

The fan power at operating conditions can be calculated as:

$$P_{VFD} = f_4 * ratedPower \tag{32}$$

<u>Two-Speed</u>

For the purposes of the two-speed fan power calculation, it was assumed that the fan operates at 0% speed, 50% speed and 100% speed. In order to quantify the power consumption, the first step would be to calculate the time spent on each speed (0%, 50% and 100%.). For clarity, we can define two states for fan operation as: fan speed between 0% to 50% and fan speed between 50% to 100%. For each of these states, the time on high speed and low speed can be calculated as follows. Furthermore, the calculation of power requirement is also shown below.

<u>IF airflow < 50% :</u>

T_{high}= Airflow rate / 0.5, T_{low}= 1 - T_{high}, P_{two_speed} = ratedPower× [T_{high}×f_4(0.5) + T_{low}× f_4(0)]

Where, T_{high} = Time spent on high speed (50% speed), T_{low}= Time spent on low speed (0 % speed), P_{two_speed}=Fan power requirement

<u>IF airflow > 50%</u>

T_{high}= (Airflow rate – 0.5) / 0.5, T_{low}= 1 - T_{high}, P_{two_speed} = ratedPower× [T_{high}×f_4(1) + T_{low}× f_4(0.5)]

Where,

T_{high}= Time spent on high speed (100% speed), T_{low}= Time spent on low speed (50 % speed), P_{two_speed} = Fan power requirement

<u>One-Speed</u>

It has been assumed that the time spent on 100% speed is equivalent to the air flow rate. The remaining portion of time will be spent on idle. The following steps can be used to calculate the fan power requirement.

T_{high} = Airflow rate, T_{low}= 1 - T_{high},

P_{one_speed} = ratedPower× [T_{high}×f_4(1) + T_{low}× f_4(0)]

It should be noted that there is some variation in the names of the calculations when they are displayed in the program.

3.4 Water Calculations

"The water balance of a tower or cooling water system involves all of the water inputs and outputs associated with the operation of the system. Water outputs from a cooling tower include controlled losses such as evaporation, bleed, drift and pump gland leakage and uncontrolled losses including leaks, splash out, overflows and windage. All of these losses are replaced by makeup water from the system water supply." [12]. For this program, the evaporation, drift, and blowdown losses will be considered. The total water and sewer usages will also be generated for both the base and the revised conditions. This section will go into detail about what these losses are and how these losses have been calculated.

Evaporation loss is the water that will be lost in the evaporation process of the cooling tower. For the model, the calculation for evaporation loss (gal/hr) will incorporate the inlet mass air flow, AF, the air flow percentage found in the fan power calculations, $\%_{AF}$, and the density of water, ρ_{H2O}, which is calculated via the operational water schedule from the user's inputs. Additionally, the difference in the entering and leaving absolute humidity ratio, HR_1 and HR_2, will be included in the equation. The leaving humidity ratio will either be for the base case or the revised case, depending on which case is being calculated. The equation used to solve for evaporation loss, E, can be seen below.

$$E = \frac{AF \times 60 \times (HR_2 - HR_1) \times \frac{\%_{AF}}{100}}{\rho_{H2O}} \tag{33}$$

Following the evaporation loss, the drift loss, D, can be calculated. The drift loss will vary depending on if the user utilizes drift eliminators or not. A typical drift loss can be assumed to be 0.2% of the re-circulating water flow rate. Efficient drift eliminators will minimize drift loss to less than 0.001% of the re-circulating water. These respective equations for calculating drift loss, D, is expressed in Equations 34 and 35, where the unit will be expressed as gallon per hour (gal/hr) for this model.

If drift eliminators are present:

$$D = Water\ Flow \times 0.00001 \times 60 \tag{34}$$

If drift eliminators are not present:

$$D = Water\ Flow \times 0.002 \times 60 \tag{35}$$

The next calculation will be the blowdown loss, B, which is represented by Equation 36, where C.O.C. is the cycles of concentration taken from the user's inputs. The unit for blowdown will be expressed as gallon per hour (gal/hr) for this model.

$$B = \frac{Evaporation\ loss}{(C.O.C. - 1)} \tag{36}$$

In order to calculate the total water usage (gal/hr), the program will sum all of the water losses together, which is represented by Equation 37 below.

$$Total\ Water\ Usage = E + D + B \tag{37}$$

Following the total water usage, the sewer water usage (gal/hr) will be calculated. This calculation will be a percentage of the total water usage depending on the percentage chosen by the user. The sewer usage will only change if the user has a contract with their water provider to adjust their sewer charges, whether it be via a meter on their sewer line or by an agreed upon percentage of their total water usage. If the user does not have either of those, then their sewer usage will likely be the same as their total water usage.

3.5 Pumping Calculations

The last calculations that the program will need to simulate the estimated energy savings for a cooling tower is that of the pumping energy. These calculations simulate a base and revised case for the pump's water flow (GPM), pump head (ft), and pump power (kW). The water flow has already been calculated and discussed in Section 3.2 when it was used to estimate the exit dry-bulb temperature. Next, the pump head will be generated using Equation 38, with the flow, $Flow_2$, being the water flow found in the cell adjacent to the pump head column.

$$Pump\ Head = Static\ Head + \left[\frac{Pump\ Head - Static\ Head}{(Flow_1)^{1.9}}\right] * (Flow_2)^{1.9} \tag{38}$$

Lastly, the pump power (kW) will be estimated. The pump energy equations will vary depending on whether there is a VFD. If there is a VFD selected or the user has reduced their monthly flow, then the VFD efficiency will be included in the equation as seen in Equation 39. It should be noted that if the user is operating a VFD with full flow in their pump, then the user will have more efficiency losses in their pump due to using a VFD at one hundred percent speed, i.e., the user will not receive the monetary and energy saving benefits of having a VFD in the months where the flow is not reduced. Water specific gravity has been assumed to be 1.0.

If a VFD is present:

$$Power\ [kW] = \frac{Q[GPM] \times H[ft] \times 0.746}{3960 \times \eta_{motor} \times \eta_{pump} \times \eta_{VFD}} \tag{39}$$

If a VFD is not present:

$$Power\ [kW] = \frac{Q[GPM] \times H[ft] \times 0.746}{3960 \times \eta_{motor} \times \eta_{pump}} \tag{40}$$

4. RESULTS AND DISCUSSION

The last part of the model will be a summary of all the results from the annual calculations for the inputted cooling tower. For the discussion of this paper, a 400-ton, induced draft, crossflow, multi-cell type cooling tower has been simulated in the program. This particular cooling tower is used in HVAC applications located in Cookeville, TN, USA. Additionally, the design conditions of the two-cell cooling tower are as follows: a design flowrate of 4,200 GPM, 95°F entering water temperature, 85°F leaving water temperature, 40 horsepower fan, and an approximate inlet airflow of 161,000 CFM. The model will analyze the energy and water savings for converting this single-speed fan control to a VFD fan control, adding drift eliminators, and changing the cycles of concentration from two to five. The operational conditions will be maintained at 95°F entering water temperature and 85°F leaving water temperature for the entire year. Additionally, for the winter months, i.e. December to March, the cooling tower design flowrate has been reduced by 10% which will still allow the cooling tower to maintain the cooling load desired by the user.

The utility cost information should also be noted to fully illustrate the predicted savings for this cooling tower. That information is as follows: electrical consumption cost is 0.05 ($/kWh), electrical demand cost is 14.50 ($/kW), water cost is 5.00 ($/1000 gal), sewer cost is 5.50 ($/1000 gal). It should also be noted that for this particular cooling tower, it is estimated that the user is charged for approximately 70% of their water usage as sewer usage as well. This section will analyze the results from the model for this particular cooling tower.

For the energy savings, the model will break down the energy savings into the energy consumption and energy demand rather than using the blended rate due to the additional accuracy it provides. For the results, the energy consumption will be taken as the sum of each month and displayed by the program, and the energy demand will be taken as the maximum demand for each month. The maximum demand is used in order to determine the peak demand, or the maximum operating load, of the cooling tower fan/pump energy for that month. The program will then depict the savings in a graph. For the fan power results, the graph will depict the monthly monetary savings of switching to VFD fan control from either a two-speed or a single-speed fan control for the base case. For this cooling tower, we will analyze the savings from converting from a single-speed fan control to a VFD fan control, i.e., the orange bars in the graph. This graph can be seen in Fig 5.

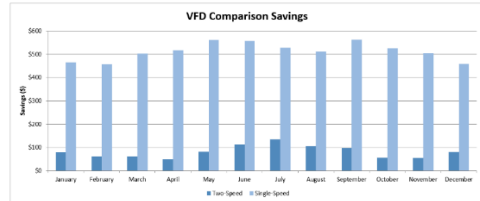

FIGURE 5: FAN POWER RESULTS GRAPH

According to the model, for this particular cooling tower, the user will save approximately $6,153 per year. The fan energy consumption savings is estimated to be 87,226.41 (kWh) and the fan energy demand is 123.54 (kW). For a fan of this size, it was estimated that a VFD project cost would be approximately $6,000, assuming that it costs roughly $150 per horsepower. Therefore, this would give a simple payback period of just under a year for this particular user.

Additionally, the program will estimate the Carbon Dioxide Reduction (lbs). The Carbon Dioxide (CO_2) emission rate can be calculated based off the energy consumption savings (ECS) multiplied by the user's inputted carbon dioxide emission rate for their selected state. If the user does not know their respective rate the program will use the United States national average carbon dioxide emission rate, 1,558.8 (lbs/MWh), which is taken from the United States Environmental Protection Agency average for 2018 [13]. However, for this run the carbon dioxide emission rate for Tennessee was used, which is 741.66 (lbs/MWh) [14]. This carbon dioxide reduction is represented by Equation 41.

$$CO_2\ Reduction\ [lbs] = \frac{ECS\ [kWh] \times 741.66 \left[\frac{lbs}{MWh}\right]}{1{,}000\ \left[\frac{kWh}{MWh}\right]} \tag{41}$$

Therefore, it is estimated that the annual carbon dioxide reduction is estimated to be 64,692 (lbs) based on the fan power energy consumption savings.

Likewise, a similar simulation has been done for the pumping results. The estimated annual energy consumption savings will be 90,429.58 (kWh) and the annual energy demand savings will be 124.98 (kW). Therefore, the estimated cost savings is approximately $6,334 per year. These savings from the base to the revised case are depicted in Fig 6. It is key to note that this example depicts an instance where a VFD is used on a pump that operates at one hundred percent flow; therefore, there will be a loss in savings due to the extra efficiency loss while operating a VFD at full speed. Additionally, the annual carbon dioxide reduction is estimated to be 67,068 (lbs).

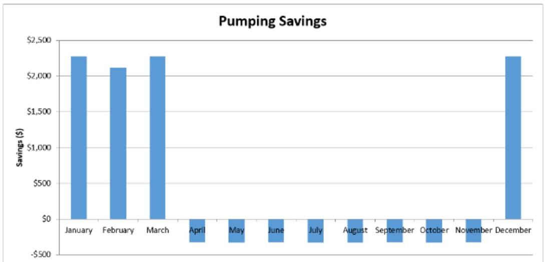

FIGURE 6: PUMPING RESULTS GRAPH

Additionally, the model will estimate the water and sewer savings for the cooling tower from the base case to the revised case. These savings are estimated from reducing the monthly flow for December to March, adding drift eliminators and increasing the cycles of concentration from two to five. The program estimates that for this cooling tower, the user will save 9,379,401 gallons per year for the water usage and 6,565,581 gallons per year for the sewer usage. Therefore, it was estimated that the annual water cost savings is $46,897 and the annual sewer cost savings is $36,111. These monthly savings are depicted by Fig. 7.

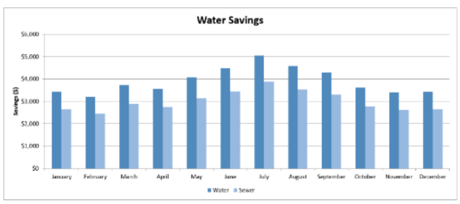

FIGURE 7: WATER RESULTS GRAPH

5. CONCLUSION

In this paper, a cooling tower modeling tool was developed and tested to study existing industrial cooling towers. The main goals of this model are to identify and quantify energy, water and carbon reduction opportunities. In this study, an operating industrial cooling tower located in Cookeville, Tennessee, USA was modeled and analyzed with the modeling tool to verify and understand the modeling results. This model estimates that the user will annually save 177,656 kWh in energy, 248.5 kW in energy demand, 131,760 lbs in CO_2 reduction, 9,379,401 gal in water, and 6,565,581 gal for the sewer usage. This results in approximately $95,495 savings per year if the user makes the operational changes discussed in this simulation. Therefore, the model has shown to have significant promise to help industry as they analyze ways to save energy and reduce their cost at their facility through their cooling tower.

Like any other modeling study, there are opportunities for future development of this model. Going forward, future researchers will improve the model in several areas including allowing the program to run individual results independently without having to calculate each resulting section, i.e. calculating the fan power results without the water results, increasing the speed of the modeling, and validating the results of the model with various operating industrial cooling towers as it will be tested in diverse industrial facilities by the U.S. Department of Energy's Industrial Assessment Centers.

The main goal of this research was to create a modeling tool to easily analyze the annual performance of an industrial cooling tower based on the respective location of the tower. Therefore, this paper has been written to illustrate how this model can help determine the estimated annual savings for an industrial cooling tower.

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