Abstract—The emergence of increasingly affordable variable-speed drive technology has changed the approach used to control chilled water systems equipped with these drives. The purpose of this research was to develop an integrated chilled water modeling technique that can determine the optimal system setpoints and estimate the energy saving potential of chiller system. The chiller system equipped with Variable Frequency Drives (VFDs) on cooling tower fans and condenser water pumps. To accomplish the objective, physical component models of the centrifugal chiller, cooling tower and condenser water pump were established with the goal of incorporating the system’s condenser water flow rate and cooling tower fan speeds as optimization variables.

Furthermore, a cooling load prediction algorithm was developed using a multiple non-linear regression model to approximate the building’s cooling load subject to a range of environmental conditions. The inputs and outputs of the individual component models were linked to estimate how adjusting the cooling tower fan and condenser water pump speed would influence the system’s comprehensive performance. The overall system model was then optimized using a generalized reduced gradient optimization algorithm to determine the potential energy savings through speed control with VFDs and to ascertain a control logic strategy for the building automation system to operate the heating and cooling system. A case-study was performed on a single chiller system at a museum and the model was calibrated according to logged data collected over four months. Results showed that for the system analyzed, the energy saving of optimizing the cooling tower fan system was found to be 12-15%, while the energy saving potential of optimizing the condenser water pump with the cooling tower fan was negligible. Additionally, comparing different cooling tower fan control strategies showed that a wet-bulb approach-based cooling tower control strategy was shown to have the highest correlation to the optimized fan speed with an R² of 0.924.

Keywords—Chilled Water System, Variable Speed Drive, Condenser Water System, Cooling Tower, HVAC Optimization

1. INTRODUCTION

The U.S. Energy Information Administration estimated that in 2018, space cooling of commercial and residential buildings consumed 377 billion kWh of electricity, or approximately 9% of the total U.S. electricity consumption across all sectors [1]. In the United States, vapor compression and absorption chillers supply space cooling in approximately 2.9% of commercial buildings. However, since chillers frequently service large facilities with sizeable cooling demands, they provide cooling for an estimated 20% of the total commercial building floor space [2]. Considering the impact that chiller systems have on the energy consumption profile of large commercial and industrial facilities, measures to improve the efficiency of chiller cooling systems can reduce a significant amount of energy without compromising temperature and humidity requirements. Many authors have researched and validated the energy-saving potential of...
equipping variable-speed drives on cooling tower fans and evaporator pumps; however, there is not a clear consensus on the advantages of outfitting variable-speed drive on the condenser water pump. This research aims to find a reliable method to determine the energy-saving potential of implementing a variable-flow condenser water system.

2. LITERATURE REVIEW

The emergence of increasingly affordable variable-speed drives has changed conventional chilled water control strategies in order to achieve energy savings. An abundance of research has been done regarding the optimization of variable-speed-driven cooling tower fans, and a variety of control strategies have been proposed to optimize cooling tower fan operation [3],[4],[5],[6],[7]. These strategies generally focus on controlling the cooling tower fan based on cooling load and/or ambient weather conditions. Variable-speed-driven evaporator pumps have been examined extensively and there has been ample evidence to validate the energy-saving potential of variable-flow evaporator water pumps [8], [9], [10]. The two main types of variable-chilled-water-flow configurations are variable primary and constant-primary-variable-secondary. The consensus on these configurations is that variable-primary systems generally have lower install and operating costs than constant-primary-variable-secondary units. Regarding constant-speed versus variable-speed chillers, variable-speed-driven chillers have exhibited superior efficiency for partially-loaded operation and demonstrated a more drastic improvement to efficiency for lower condenser water temperatures [11],[12],[13]. What remains uncertain is whether or not operating one or more variable speed condenser water pumps to provide control of the condenser water flow rate will result in energy savings. If there is, the potential to save energy, a proper strategy to control the condenser water flow rate must be determined. Thomas Hartman has been a major proponent of all-variable-speed chiller plants. He proposed the Equal Marginal Performance Principle (EMPP) which optimizes chilled water systems by adjusting the power input of individual pieces of equipment until the marginal change in coefficient of performance is equal [14]. Taylor suggested that the energy-saving potential of variable condenser water flow applications does not justify either the additional cost of variable-speed drives or the additional complexity of controlling these drives [15]. Conversely, Hydeman used a parametric model to simulate a chilled water plant and found a large potential to save energy but stressed the importance of carefully evaluating and controlling the condenser water flow rate [16]. Lu et al. developed a model-based optimization strategy and used a genetic algorithm to minimize the energy consumption of the condenser water loop [17]. Yu and Chan proposed a load based speed control strategy for enhancing the energy performance of all-variable-speed chilled water systems and found that an optimally controlled system could increase the system COP 1.4-16.1% compared to an equivalent system with fixed temperature and flow rate control [18]. Zhang and Liu found that the energy-saving potential of an optimal variable condenser water flow rate is marginal, and that the optimal condenser water flow rate is dependent on the system’s configuration, design conditions, chiller sensitivity to variable condenser water flow rate and, to a lesser degree, the local climate in which the system operates [19]. Karami and Wang utilized a particle swarm algorithm to determine the optimal chilled water temperature setpoint, condenser water temperature setpoint and condenser sequencing for an all-variable speed chilled water system, however they did not explore the energy saving potential of optimizing the speed of chilled water or condenser water pumps [20]. Overall, the research that has been done regarding variable condenser water flow rate contains differing viewpoints concerning whether the additional cost and complexity of introducing a variable-flow condenser water is justified by the potential energy savings. The goal of this study is to develop a system model that can be routinely applied to various chilled water systems to determine the energy-saving potential of optimizing a system with variable-speed-driven condenser water pumps. Additionally, the study will aim to explore different proposed strategies for controlling both VFDs on condenser water equipment.

3. BUILDING AND CHILLER SYSTEM DESCRIPTION

Data for a case study was collected from a local museum’s chilled water system. The Eiteljorg Museum in Indianapolis, IN is a 11,150 m² building that houses a variety of western and Native American art as shown in Figure 1. As a museum, the facility has strict climate-control requirements to maintain the integrity of the exhibits housed inside. For mixed collections, a humidity level between 45-50% and temperature between 20-22.2°C is recommended to prevent chemical reactions and biodegradation in the art installations [21]. The Eiteljorg’s HVAC system operates to maintain an internal temperature of 21.1°C and relative humidity level of approximately 50%. Since dehumidification is also an important factor in maintaining the integrity of the museum’s exhibits, the chilled water system operates to provide a constant chilled water temperature of 4.4°C to the cooling coils of the building’s three air handling units.

3.1 System Description

The museum utilizes a 300-ton Carrier 19XRV variable-speed-driven chiller to produce the building’s chilled water. The chiller uses a centrifugal compressor to drive refrigerant R-134A to a high pressure and temperature on the shell side of the condenser. The refrigerant rejects heat into the condenser water running through the tube side of the heat exchanger. The condenser water is pumped through the condenser into one of the building’s two VT1-307-0 Baltimore Aircoil Company cooling towers. One cooling tower services the chiller while the other is used as a back-up in case the first tower requires maintenance. The cooling tower fans are driven by 30HP motors connected to variable-speed drives. The cooling tower fan is controlled to reach and maintain an exit tower water temperature of 18.3°C. Once the
condenser water temperature set point is reached, the building automation system will cycle the cooling tower fan speed between 25% and 100% to maintain the condenser water at the set point temperature. The condenser and evaporator water pumps are Bell & Gossett series 1510 driven by 15HP Baldor Reliance Super E motors. The pump motors are also equipped with variable-frequency drives; however, the building automation system (BAS) does not possess control logic for pump operation. Consequently, the BAS currently operates these pump motors at 100% speed constantly. Figure 2 shows a simple diagram of the chilled water and location of the data collection points.

![Chilled Water System Diagram & Data Collection Points](image)

3.2 Data Collection

Data for the building’s chilled water system were collected in 15-minute intervals from July 10, 2019 to October 31, 2019. Several different time periods within the data collection phase had to be erased due to either data corruption or incomplete data sets. The water flow rate of the condenser line was found to be a relatively constant value of 42.3 (L/s). Although the chilled water pump is equipped with a variable-speed drive, the BAS operates the pump at 100% and the flow rate on the evaporator side of the chiller is regulated with a bypass valve. The flow rate of the chilled water line could not be determined with the portable ultrasonic flowmeter because of the insulation that was covering the chilled water piping. Due to the chilled water system’s importance in maintaining the integrity of the museum’s exhibits, the Eiteljorg’s HVAC system operators opposed to allowing changes to the chilled water system’s current control strategies for data collection purposes. As a result, data could not be collected for various condenser water flow rates, cooling tower fan speeds or condenser water temperatures lower than the set point temperature of 18.3°C. The lack of data below 18.3°C introduces uncertainty when extrapolating values for the entering condenser water temperatures below 18.3°C. Table 1 provides an overview of the data that was collected from the Eiteljorg's system.

### Table 1. Data Collection Overview

<table>
<thead>
<tr>
<th>Data Collected</th>
<th>Equipment Used</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling Tower Fan Amps</td>
<td>Onset UM-120-005M Data Logger &amp; CTV-C Current Transducer</td>
</tr>
<tr>
<td>Cooling Tower Fan Power</td>
<td>Fluke 1732 Power Analyzer</td>
</tr>
<tr>
<td>Chiller Amps</td>
<td>Onset UM-120-005M Data Logger &amp; CTV-E Current Transducer</td>
</tr>
<tr>
<td>Chiller Power</td>
<td>Fluke 1732 Power Analyzer</td>
</tr>
<tr>
<td>Condenser Pump Power</td>
<td>Fluke 1732 Power Analyzer</td>
</tr>
<tr>
<td>Condenser Water Flow Rate</td>
<td>Fuji FCS Portable Ultrasonic Flowmeter</td>
</tr>
<tr>
<td>Condenser Entering &amp; Exiting Water Temperature</td>
<td>Building Automation System</td>
</tr>
<tr>
<td>Evaporator Entering &amp; Exiting Water Temperature</td>
<td>Building Automation System</td>
</tr>
<tr>
<td>Weather Data</td>
<td>NOAA Quality Controlled Datasets [22]</td>
</tr>
</tbody>
</table>

4. Modeling

4.1 Overview

To represent the overall system, component models of the individual pieces of equipment need to be developed and linked: the chilled water system’s condenser water pump, cooling tower and chiller. The respective outputs from each model component feed into the other model components to simulate the overall system. First the building’s cooling load is predicted to determine the load that must be met by the chiller. Second, a correlation between the condenser water pump input power and the resulting condenser water flow rate needs to be established. The condenser water flow rate can then be used as an input to the cooling tower model, chiller model and the mass and energy balance. The outputs of the chiller model, cooling tower model, and the mass and energy balance act as inputs to each other which requires the overall system to be assessed with respect to the inlet and outlet condenser water temperature. The relationship between the component models is illustrated in Figure 3.

4.2 Cooling Load Prediction

Without data for the water flow rate on the evaporator side of the chiller, the cooling load had to be determined using the temperature difference across the condenser and the condenser water flow rate by deducting the compressor’s power from the condenser water load. The calculated cooling load is subject to a large degree of variation with minimal change in temperature difference across the condenser. To reduce noise in the calculated cooling load, the one-hour moving average of the load was substituted for the 15-minute discrete load. Figure 4 shows the discrete and one-hour moving average of the cooling load during a week in July.
A multiple non-linear regression algorithm was chosen for the load prediction model because it can achieve excellent correlation for various building types with low computational requirements and without exceedingly detailed building information [23]. The environmental variables that have been shown to have the greatest influence over a building’s cooling load are the dry-bulb temperature, relative humidity and solar irradiance [23]. To account for occupancy-related loads, Boolean variables were added to incorporate each day of the week and to distinguish between occupied and unoccupied hours. Additionally, previous authors have shown that adding a term for the cooling load from two hours prior to the current time step can greatly improve the regression model’s accuracy [23]. After performing an analysis on the initial regression model, any variable found to have p-value of greater than 0.05 was determined to be statistically insignificant to the model and the variable was removed from the cooling load prediction. Eq. 1 shows the final regression model after statistically insignificant variables were removed and Table 2 gives the value of the coefficients. Figure 5 depicts a scatter plot of the measured vs. predicted cooling load for the entire dataset. The correlation exhibited a greater accuracy at predicting cooling loads in excess of 700kW. At lower loads, the buildings cooling load is likely more dependent on internal heat gain or another variable that was not incorporated in the regression variables.

\[ Q_L = a_1 \times \text{Occupied} + a_2 \times \text{SolarRad.} + a_3 \times (\text{Temp.})^2 + a_4 \times (\text{Temp.} \times \text{RH})\% + a_5 \times Q_{L,2h} + a_6 \times \text{Sat.} + b \]  

(1)

<table>
<thead>
<tr>
<th>Coeff.</th>
<th>(a_1)</th>
<th>(a_2)</th>
<th>(a_3)</th>
<th>(a_4)</th>
<th>(a_5)</th>
<th>(a_6)</th>
<th>(b)</th>
</tr>
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<tbody>
<tr>
<td>SI Units</td>
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<td>kW</td>
<td>(\text{C}^2)</td>
<td>kW</td>
<td>(\text{C} \times %)</td>
<td>kW</td>
</tr>
<tr>
<td>Values</td>
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<td>2.13</td>
<td>0.026</td>
<td>0.79</td>
<td>3.74</td>
<td>105</td>
</tr>
</tbody>
</table>

4.3 Condenser Pump Model

The condenser water flow rate is an important factor in optimizing the system’s overall energy consumption because it affects the performance of both the chiller and the cooling tower. A relationship between the pump's power consumption and the condenser water flow needed to be determined; however, since the condenser water pump only operated at full capacity, using empirical measurements of the relationship at multiple pump speeds could not be achieved. The National Renewable Energy Laboratory developed a simple method which uses predetermined polynomial coefficients to estimate the relationship between water flow rate and pump power for a broad range of system configurations [24]. The polynomial correlation can be seen in Eq. 2. The values for flow rate and pump power are input as a percentage of the variable to its maximum values. Figure 6 shows the condenser pump’s power demand versus the flow rate.

\[ Q_F = 0.21976 - 0.87478 \times P_P + 1.6526 \times P_P \]  

(2)

Where,

- \(Q_F\) = condenser water flow rate (% of max)
- \(P_P\) = pump power (% of max)
4.4 Cooling Tower Model

Long-term data for the cooling tower fan’s line amps were collected using an Onset UMI20-006M data logger and a CTV-C (10-100A) current transducer. The cooling tower fan power and line amps were measured using the Fluke 1732 three-phase power analyzer at fan speeds of 25%, 50%, 75% and 100%. The data collection process was designed to adhere to the recommendations from NREL’s VFD evaluation protocol of fans in lieu of long-term true power measurements [24]. The measurements were used to determine the relationship between the cooling tower’s fan speed, the motor’s line amps and power consumption. The correlations were used in lieu of true long-term power measurements because for online adaptive control system monitoring, the cooling tower’s line amps would be cheaper than using a device to constantly monitor the fan’s true power consumption. The correlation between motor line amps and the motor power consumption can be seen in Figure 7. Figure 8 not only indicates the relationship between the fan speed and the motor power consumption but also graphs the relationship in comparison to affinity laws.

The fan’s power consumption in relation to speed was found to closely mirror affinity. A second order polynomial correlation was used to relate fan power consumption and amps to the cooling tower's fan speed and, subsequently, to the cooling tower airflow rate. Actual airflow measurements of cooling tower fans are difficult to obtain so the cooling tower’s airflow rate was assumed to vary linearly in relation to fan speed according to affinity laws. The design air flow rate of the cooling tower operating at full fan speed is 35,090 L/s. The cooling tower fan control strategy is an important factor in optimizing the energy consumption of a chilled water system. The building automation system which controlled the cooling tower fan would operate the fan speed at 100% until the exit tower water temperature reached a set point of 18.3°C. Once the temperature set point had been reached, the cooling tower cycled between 25% and 100% fan speed to maintain the tower outlet water temperature near the set point. Figure 9 shows how the cooling tower fan operates at full load during the warmer months. Figure 10 illustrates how the fan begins to cycle as the ambient temperature drops and the tower set point temperature is reached.
The transient nature of the cycling cooling tower fan would undermine a steady state model; thus, the data from 9/27-10/30 was removed for training the cooling tower model. As a result, the entire cooling tower dataset was comprised of only one value for both the air and water volumetric flow rates. The NTU-effectiveness model [24] was selected to represent the cooling tower because it can achieve accurate estimations for outlet water temperatures without extensive data requirements and without complicated tower information. The NTU-effectiveness model is a physical model derived from performing a mass and energy balance on counter-flow air and water streams. The method assumes negligible heat transfer through tower walls, constant specific heat for water and dry air, and that the air saturation enthalpy with respect to temperature can be approximated as linear. With these assumptions and empirically determined constants, the NTU model is iteratively solved to determine the tower outlet water conditions over a range of air and water flow rates. Eqs. 3, 4, and 5 show how the NTU for a cooling tower is defined. The NTU-effectiveness model uses this definition to help solve the mass and energy balance on the streams and to estimate the tower’s outlet water temperature.

\[
NTU = \frac{h_D A_V V_T}{m_a}, \quad \frac{h_D A_V V_T}{m_w} = c \left( \frac{m_w}{m_a} \right)^n \quad (3,4)
\]

\[
NTU = c \left( \frac{m_w}{m_a} \right)^{\frac{1}{n+1}} \quad (5)
\]

The values for \( c \) and \( n \) are constants that are specific to a particular cooling tower and can be determined with a straight-line correlation of a log-log plot of NTU versus the mass flow rate ratio of water to air. The slope of the log-log plot is equal to \( n+1 \) and the intercept equal to \( \log(c) \). The dataset collected had no variation in the air or water mass flow rate ratio, so performance data published by the Baltimore Aircoil Company for the VT1-307-0 cooling tower was used to introduce data points with varying water flow rates [26]. The constants determined from using only performance data was found to slightly over predict the cooling tower’s effectiveness, so the performance dataset was combined with an equal number of data points from the information collected directly from the system. The log-log plot of NTU vs. the mass flow ratio for the mixed dataset can be seen in Figure 11. Mixing the datasets reduced the slope of the performance data on the log-log which leads to a more accurate prediction of the outlet water temperature because it incorporates how scale has degraded the tower heat rejection capacity from published performance data. The empirical constants, the \( R^2 \) and the Root-mean-square error (RSME) of the performance dataset and the mixed dataset are given in Table 3. Figure 12 shows the measured vs. predicted tower exit water temperature and Figure 13 illustrates how the outlet tower water temperature changes in response to varying condenser water pump and cooling tower fan speeds.

![Fig. 10. BAS Fan Speed with Wetbulb & Tower Outlet Water Temp. in Oct.](image)

![Fig. 11. Log-Log Plot of Tower NTU vs. Mass Flow Ratio of Water to Air](image)

![Fig. 12. Measured vs. Predicted Tower Outlet Water Temperature](image)

<table>
<thead>
<tr>
<th>Dataset</th>
<th>( C )</th>
<th>( n )</th>
<th>( R^2 )</th>
<th>RSME (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Performance</td>
<td>1.66379</td>
<td>-0.7017</td>
<td>0.917</td>
<td>0.746</td>
</tr>
<tr>
<td>Mixed</td>
<td>1.43714</td>
<td>-0.4378</td>
<td>0.929</td>
<td>0.659</td>
</tr>
</tbody>
</table>
4.5 Chiller Model

The primary consideration when analyzing potential chiller models was the ability to incorporate condenser water flow rate as an optimization variable. Most models with the ability to account for variable condenser water flow rate require large datasets that span the entire range of the chiller’s operating conditions. Without the freedom to manipulate the system’s flow rate, data could only be collected for the flow rate of 671 GPM (42.33 L/s) meaning that the selected model needed to be able to extrapolate the chiller’s operating character under different condenser flow rates. The Gordon and Ng thermodynamic chiller model was determined to be the most appropriate method due to the limited range of available data and the ability to regress physical parameters from the collected dataset. The thermodynamic equation governing the original chiller model can be seen in Eq. 6. [27]

\[
\frac{T_{e,\phi} - Q_s R}{T_{c,i}} \left[ 1 + \frac{1}{COP} \right] = 1 + \frac{T_{e,\phi} \Delta S_{int}}{Q_s} + \frac{L}{Q_s} \left[ T_{c,i} - T_{e,\phi} \right]
\] (6)

Over the years, two modifications to the original model have been made that can help improve the model’s flexibility. Jiang and Reddy [28] proposed a modification that incorporates a term to make the rate of internal entropy-generation linear with respect to the maximum cooling load. The addition of the linear term for the internal entropy-generation has been shown to improve the model’s accuracy for predicting the coefficient of performance, specifically for variable-speed-driven chillers. The modified thermodynamic equation can be seen in Eq. 7.

\[
\frac{T_{e,\phi} - Q_s R}{T_{c,i}} \left[ 1 + \frac{1}{COP} \right] = 1 + \frac{T_{e,\phi} \Delta S_{int,1} + \Delta S_{int,2}}{Q_s} + \frac{L}{Q_s} \left[ T_{c,i} - T_{e,\phi} \right]
\] (7)

Gordon et.al. [29] proposed a separate modification to the original model which manipulates the heat exchanger resistance term to incorporate condenser water flow rate as a control variable. The modification breaks the heat exchanger thermal resistance into two separate pieces and represents the thermal resistance of the evaporator and the condenser separately. The full variable condenser flow Gordon model can be seen in Eq. 8.

\[
\frac{Q_s}{T_{c,i}} \left[ 1 + \frac{1}{COP} \right] = 1 + \frac{T_{e,\phi} \Delta S_{int}}{Q_s} + \frac{L}{Q_s} \left[ T_{c,i} - T_{e,\phi} \right] + \frac{Q_s}{T_{e,\phi}} \left[ 1 + \frac{1}{COP} \right] \left( \frac{1}{\rho C_p \exp \left( \frac{T_{e,\phi} - T_{c,i}}{T_{e,\phi}} \right)} + R_s + R_e \right)
\] (8)

The unknown parameters can be determined from non-linear regression of the chiller’s coefficient of performance as a function of cooling loads and condenser water temperature at various condenser water flow rates. Gordon et al found that for the system analyzed, the heat exchanger constant K was large enough to be considered statistically insignificant. The assumption that K is approximately equal to one simplifies the thermodynamic equation to a function with three unknown parameters that can be determined through multiple linear regression. Since data could not be collected for multiple condenser water flow rates, the assumption that the coefficient of performance is relatively unaffected by K had to be made. Given that assumption, the relationship shown in Eq. 9 enables one to predict how the chiller’s coefficient of performance will respond to changes in condenser water flow rate.

\[
\frac{Q_s}{T_{c,i}} \left[ 1 + \frac{1}{COP} \right] = 1 + \frac{T_{e,\phi} \Delta S_{int}}{Q_s} + \frac{L}{Q_s} \left[ T_{c,i} - T_{e,\phi} \right] + \frac{Q_s}{T_{e,\phi}} \left[ 1 + \frac{1}{COP} \right] \left( \frac{1}{\rho C_p \exp \left( \frac{T_{e,\phi} - T_{c,i}}{T_{e,\phi}} \right)} + R_s \right)
\] (9)

Jiang and Reddy [28] and Gordon et al [29] both used the condenser inlet water temperature to represent the condition in the condenser; however, this assumption can lead the overall system model to preferential favor reducing the condenser water flow rate. Reducing the flow rate through the cooling tower increases the condenser’s outlet water temperature while the inlet condenser water temperature can remain relatively unchanged due to increase in cooling tower effectiveness. Instead, the average condenser water temperature was used to represent the condition in the condenser. The final model selected for use incorporated both modifications to the original Gordon model and employed the average condenser water temperature. The thermodynamic equation for the final chiller model can be seen in Eq. 10. The equation must be made linear with respect to the four unknown physical variables as shown in Eq. 11. Eq. 10 can then be algebraically rearranged to solve for the COP as shown in Eq. 12.

\[
\frac{Q_s}{T_{e,\phi}} \left[ 1 + \frac{1}{COP} \right] = 1 + \frac{T_{e,\phi} \Delta S_{int,1} + \Delta S_{int,2}}{Q_s} + \frac{L}{Q_s} \left[ T_{c,i} - T_{e,\phi} \right] + \frac{Q_s}{T_{e,\phi}} \left[ 1 + \frac{1}{COP} \right] \left( \frac{1}{\rho C_p \exp \left( \frac{T_{e,\phi} - T_{c,i}}{T_{e,\phi}} \right)} + R_s + R_e \right)
\] (10)
\[ y = b_1 x_1 + b_2 x_2 + b_3 x_3 + b_4 x_4 \] 

Where,
\[ y = \frac{T_{c,i}}{T_{c,o}} \left[ 1 + \frac{1}{\text{COP}} \right] - 1 - \frac{Q_i}{T_{c,o}} \left[ 1 + \frac{1}{\text{COP}} \right] \times \frac{1}{V_w \rho C} \]
\[ x_1 = \frac{T_{c,i}}{T_{c,o}} Q_i, \quad x_2 = \frac{T_{c,i}}{T_{c,o}} \frac{Q_i}{T_{c,o}} \]
\[ x_3 = T_{c,i} - T_{c,o}, \quad x_4 = T_{c,i} \left[ 1 + \frac{1}{\text{COP}} \right] \]
\[ \text{COP} = \frac{T_{c,o} - Q_i}{T_{c,o} \frac{1}{V_w \rho C} - b_4 Q_i} \]

(12)

Figure 14 depicts the measured chiller efficiency versus the predicted chiller efficiency. Figure 15 features a surface plot of the final model's relationship between the chiller's efficiency versus the building's cooling load and the average condenser water temperature.

© Integrated Model

The inputs and outputs of the individual models were conjoined to develop a model of the complete condenser water system. The chiller currently operates to provide a constant chilled water temperature of 4.4°C to maintain an acceptable level of humidity in the museum environment. Resetting the chilled water temperature set point could potentially reduce the chiller's energy consumption. The low chilled-water temperature is important for dehumidification and for the maintenance of the museum’s strict internal climate; therefore, resetting the system’s chilled water temperature set point will not be investigated. Hence, a constant chilled water temperature of 4.4°C was used as the input for the chiller’s outlet evaporator water temperature. The chiller's projected energy consumption is determined using both the estimated coefficient of performance from the model and the building's predicted cooling load as shown in Eq. 13. An energy balance is performed on the condenser water stream to determine the water temperature exiting the chiller. Using the chiller's predicted power consumption and the building’s cooling load, the water temperature exiting the chiller can be determined with Eq. 14.

\[ P_{ch} = \frac{Q_i}{\text{COP}} \]
\[ T_{c,o} = \frac{P_{ch}}{V_w \rho C} + T_{c,i} \]

(14)

The energy balance on the water stream provides the cooling tower and chiller model with the condenser outlet water temperature; however, the cooling tower model also must provide the chiller model with the condenser inlet water temperature necessary for determining the chiller’s COP and outlet water temperature. The circular reference requires that the model be solved iteratively with respect to both the condenser inlet and outlet water temperature. Figure 16 shows a plot of the actual and modeled inlet and outlet condenser water temperatures. Figure 17 shows a surface plot of the system’s power consumption as it varies with fan and pump speed.
4.7 Optimization

The objective of the optimization is to minimize the system’s overall power consumption by using the fan speed and pump motor power as optimization variables. The fan speed is constrained between the fan’s minimum and maximum speeds of 25% and 100%. The pump power is constrained between the minimum value of 8.8 kW and 11.2 kW. The minimum pump power is required to provide the minimum flow rate of 23.1 L/s to maintain turbulence in the chiller condenser. The optimization sequence was run for three separate time periods of data. The three weeks were meant to span the range of ambient conditions and cooling loads that would be experienced during the building’s cooling season. The generalized reduced gradient Frank-Wolfe algorithm was chosen to minimize the condenser water system’s energy consumption [30]. This method is used to optimize constrained smooth nonlinear programs when the derivative of the objective function is not directly available.

5. RESULTS AND DISCUSSION

5.1 Results

The combined system model was run for four different control scenarios to compare the energy consumption of the condenser system with various operational strategies. Table 4 describes the respective pump and fan control strategy for each scenario. The scenarios were selected to compare the individual benefits of optimizing different pieces of equipment in the chilled water system.

<table>
<thead>
<tr>
<th>Control Scenario</th>
<th>Control Strategy</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Fan Full Load</td>
</tr>
<tr>
<td></td>
<td>Pump Full Load</td>
</tr>
<tr>
<td>2</td>
<td>Fan 18.3°C Condenser Water</td>
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<tr>
<td>3</td>
<td>Fan Optimized</td>
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<td>4</td>
<td>Fan Optimized</td>
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</tbody>
</table>

For Scenario 1, the cooling tower fan and condenser water pump both constantly run at full load. Scenario 2 represents the Eiteljorg’s current control strategy in which the cooling tower fan cycles between 25-100% to achieve and maintain the condenser water set point of 18.3°C. Scenario 2 acts as the baseline for comparing the performance of the different scenarios. The data collected from the Eiteljorg's chilled water system was directly used to represent scenario 2. Scenario 3 operates the condenser water pump at full load and seeks to optimize the system's energy consumption by controlling the cooling tower fan speed. Scenario 4 seeks to optimize the system's power consumption by controlling both the cooling tower fan speed and the condenser water pump's power consumption. The optimization procedure for scenarios 3 & 4 was run from July 11 to July 19, from September 1 to September 9 and from October 21 to October 29. These periods were specifically chosen to determine the optimal system operating points for a range of building cooling loads and ambient wet-bulb temperatures in order to compare the various scenario results during a range of external conditions. Figures 18-20 compare the demand and energy consumption of each scenario throughout the different evaluation periods. Table 5 shows the percent change in energy consumption relative to scenario 2 as the baseline.
The results of the optimization procedure suggest that for the system analyzed there is almost no energy saving potential in controlling the condenser water flow rate with VFD but there is significant potential to save energy through optimizing the cooling tower fan.

5.2 Control Strategies

The results for scenario 3 were analyzed to find an emerging pattern that could offer a simple control strategy for optimizing the cooling tower fan. The simplest strategy would be to control the cooling tower fan speed directly from the ambient wet-bulb temperature. Figure 21 shows the optimized cooling tower fan speed vs. the ambient wet-bulb temperature. The poor correlation between the variables would suggest that controlling the cooling tower fan directly according to the ambient wet-bulb temperature is not an excellent control strategy.

Another method that has been proposed to control cooling tower fans is load-based speed control. This method suggests controlling the speed of the variable-frequency drives proportionally with the building’s cooling load. Figure 22 displays the optimized cooling tower percentage fan speed versus the building’s cooling load. The correlation shows a clear improvement when controlling the cooling tower fan in relation to the ambient wet-bulb temperature.

Some authors have suggested resetting the condenser water temperature set point to achieve an optimal approach temperature based on ambient conditions [4]. The approach is defined as the temperature difference between the tower outlet water temperature and the ambient wet-bulb temperature. Following the proposed control strategy, the approach temperature determined from the optimized cooling tower fan speed was compared to the ambient wet-bulb temperature. This control strategy is referred to as control strategy 3. Figure 23 shows the optimized cooling tower's approach vs. the ambient wet-bulb temperature.

The correlation would suggest that a linear wet-bulb approach condenser water reset strategy most closely resembles the results from the cooling tower fan optimization. The statistics from the linear regression of the control strategies can be seen in Table 6.
The ultimate goal of the research would be to implement the showed the greatest statistical correlation for optimal fan model into an online adaptive control algorithm that could speed control. A wet-bulb based approach control strategy algorithm and adjustments to the mass and energy balances, multiple chillers or cooling towers.

The results of the modeling showed that the composite model could accurately simulate the chilled water system's performance over a range of operating conditions. The results of the optimization procedure suggest that for the case study, the energy saving potential of optimizing the system's cooling tower fan could reduce approximately 12-15% of the overall energy consumption. Conversely, the energy-saving potential of optimizing the condenser water pump with the cooling tower fan was insignificant. The lack of savings from condenser water flow control could likely be attributed to the system's design condenser flow compared to the design chiller load. The case study system exhibited a design flow to load ratio of approximately 2.3 GPM/ton (0.1451 L/sec/ton). Studies have suggested that systems designed for ASHRAE green guide suggestion of (2.3-1.6) GPM/ton (0.1451-0.1009 L/sec/ton) show less potential to save energy through condenser water flow control compared to systems operating at AHARE standard of 0.1893 L/sec/ton (3 GPM/ton) [31].

Finally, the results of the cooling tower optimization scenario were analyzed to ascertain if there exists a method to control the cooling tower fan and achieve energy savings. It was found that controlling the cooling tower fan speed directly based on the ambient wet-bulb temperature resulted in a substandard correlation between the optimized fan speed and the wet-bulb temperature. Load-based speed control showed a marginally better correlation compared to wet-bulb based speed control. A wet-bulb based approach control strategy showed the greatest statistical correlation for optimal fan control. The ultimate goal of the research would be to implement the model into an online adaptive control algorithm that could adjust system setpoints to achieve energy savings in real-time. With the proper data, implementation of a staging algorithm and adjustments to the mass and energy balances, the composite model could be applied to estimate the energy saving potential from condenser fan and condenser pump optimization on a various types of chilled water systems with multiple chillers or cooling towers.

| Optimization Strategies | Units SI | Slope 0.988 45.0% 0.478 | Intercept 0.054 23.2% 0.504 | R² -0.2465 10.95 0.924 |

6. CONCLUSION

An integrated chilled water model was developed to incorporate condenser water flow and cooling tower fan speed as optimization variables. Modules comprising the cooling load prediction algorithm, chiller, cooling tower and condenser water pump were linked to iteratively solve for system performance. The system model was applied to a case study on a single chiller system at a local museum using a combination of rated performance data, logged data and local weather data.

The results of the modeling showed that the composite model could accurately simulate the chilled water system's performance over a range of operating conditions. The results of the optimization procedure suggest that for the case study, the energy saving potential of optimizing the system's cooling tower fan could reduce approximately 12-15% of the overall energy consumption. Conversely, the energy-saving potential of optimizing the condenser water pump with the cooling tower fan was insignificant. The lack of savings from condenser water flow control could likely be attributed to the system’s design condenser flow compared to the design chiller load. The case study system exhibited a design flow to load ratio of approximately 2.3 GPM/ton (0.1451 L/sec/ton). Studies have suggested that systems designed for ASHRAE green guide suggestion of (2.3-1.6) GPM/ton (0.1451-0.1009 L/sec/ton) show less potential to save energy through condenser water flow control compared to systems operating at AHARE standard of 0.1893 L/sec/ton (3 GPM/ton) [31]. Finally, the results of the cooling tower optimization scenario were analyzed to ascertain if there exists a method to control the cooling tower fan and achieve energy savings. It was found that controlling the cooling tower fan speed directly based on the ambient wet-bulb temperature resulted in a substandard correlation between the optimized fan speed and the wet-bulb temperature. Load-based speed control showed a marginally better correlation compared to wet-bulb based speed control. A wet-bulb based approach control strategy showed the greatest statistical correlation for optimal fan control.

The ultimate goal of the research would be to implement the model into an online adaptive control algorithm that could adjust system setpoints to achieve energy savings in real-time. With the proper data, implementation of a staging algorithm and adjustments to the mass and energy balances, the composite model could be applied to estimate the energy saving potential from condenser fan and condenser pump optimization on a various types of chilled water systems with multiple chillers or cooling towers.

**REFERENCES**


