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Chilled Water System Modeling and Optimization

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Abstract— The emergence of increasingly affordable variable speed drive technology has changed the approach for how chilled water systems equipped with variable speed drives should be controlled. The purpose of this research was to estimate the potential energy savings that can be achieved through optimization of a single chiller system equipped with Variable Frequency Drives (VFDs) on all pieces of equipment in the condenser water system. Data for a case study was collected from a local museum's chilled water system. 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 buildings 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 overall 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 ascertain a simple control logic strategy for the building automation system to operate the system. The saving potential of the optimized system was found to be 12-15%.

Keywords—chilled water system, variable speed drive, condenser water system, cooling tower, HVAC optimization

I. 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 around 20% of the total commercial building floor space [2]. Considering the impact 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 the 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 consistent method to determine the energy saving potential of implementing a variable flow condenser water system.

II. BUILDING AND CHILLER SYSTEM DESCRIPTION

The Eiteljorg Museum in Indianapolis, IN is a 120,000 ft^2 building that houses a variety of western and Native American arts. As a museum, the facility has strict climate control requirements to maintain the integrity of the exhibits housed inside the building. 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 [3]. 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 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 building's three air handling unit's cooling coils.

A. *System Description*

The museum utilizes a 300-ton Carrier 19XRV variable speed driven chiller to produce the buildings 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 fan is driven by a 30 HP motor connected to a variable speed drive. The cooling tower fan is controlled to reach and maintain an exit tower water temperature of 18.3°C. Once the condenser water temperature setpoint is reached, the building automation system will cycle the cooling tower fan speed between 25% and 100% to maintain the condenser water at the setpoint temperature. The condenser and evaporator water pumps are Bell & Gossett series 1510 driven by 15 HP Baldor Reliance SuperE motors. The pump motors are also equipped with variable frequency drives; however, the building automation system (BAS) does not possess control logic for how to operate these pumps. Consequently, the BAS currently operates these pump motors at 100% speed constantly.

B. *Data Collection*

Data for the building's chilled water system were collected in 15-minute intervals from July 10^{th} to October 31st, 2019. Several different time periods within the data collection phase had to be erased due to either data corruption or a lack the complete set of required data. The water flow rate of the condenser line was found to be a relatively constant value of 42.3 (L/s). Due to the chilled water system's importance in maintaining the integrity of the museum's exhibits, the Eiteljorg's HVAC system operators were 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 for entering condenser water temperatures lower than the setpoint temperature of 18.3°C. Table 1 gives an overview of the data that was collected from the Eiteljorg's system.

TABLE I. DATA COLLECTION OVERVIEW

Data Collected	
Cooling Tower Fan Amps	Cooling Tower Fan Power
Chiller Currrent	Chiller Power
Condenser Pump Power	Condenser Water Flow Rate
Condenser Entering & Exiting Water Temperature	Evaporator Entering & Exiting Water Temperature
Weather Data	

III. MODELING

A. *Overview*

To model the overall system, component models of the individual pieces of equipment need to be developed and linked together. The components included models for the chilled water system's condenser water pump, cooling tower and chiller. The respective outputs from each model feed into the other models 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 solved iteratively with respect to the inlet and outlet condenser water temperature. The relationship between the component models can be seen in Figure 1.

B. *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 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 discreet load.

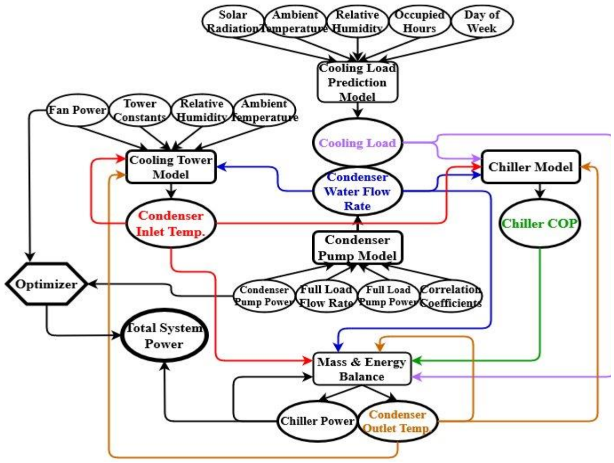

Fig. 1. Measured vs. Predicted Building Cooling Load

A multiple non-linear regression algorithm was chosen for the load prediction model because it can achieve good correlation for various building types with low computation requirements and without exceedingly detailed building information. 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 [5] 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 researchers 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 [5]. 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 model. Eq. 1 shows the final regression model used after statistically insignificant variables were removed and table 2 gives the value of the coefficients. Figure 2 shows a scatter plot of the measured vs. predicted cooling load for the entire dataset.

$$Q_L = a_1 \times Occupied + a_2 \times SolarRad. + a_3 \times (Temp.)^2 + a_4 \times (Temp. \times RH\%) + a_5 \times Q_{L,2hr} + a_6 \times Sat. + b \quad (1)$$

TABLE II. COOLING LOAD REGRESSION COEFFIECIENTS

Coeff.	a_1	a_2	a_3	a_4	a_5	a_6	b
English Units	Ton	$\frac{\text{Ton}}{\frac{\text{BTU}}{\text{ft}^2 \cdot \text{hr}}}$	$\frac{\text{Ton}}{°\text{F}^2}$	$\frac{\text{Ton}}{°\text{F} \cdot \%}$	$\frac{\text{Ton}}{\text{Ton}}$	$\frac{1}{\text{Ton}}$	Ton
Values	-1.44	0.026	0.19	0.002	0.79	1.04	22
SI Units	kW	$\frac{\text{kW}}{\frac{\text{W}}{\text{m}^2}}$	$\frac{\text{kW}}{°\text{C}^2}$	$\frac{\text{kW}}{°\text{C} \cdot \%}$	$\frac{\text{kW}}{\text{kW}}$	$\frac{1}{\text{kW}}$	kW
Values	-5.39	0.029	2.13	0.026	0.79	3.74	105

C. *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

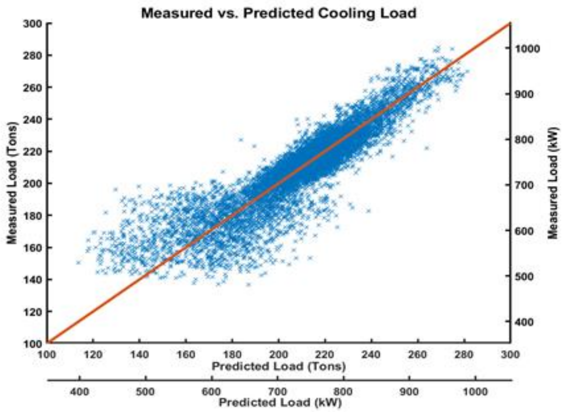

Fig. 2. Measured vs. Predicted Building Cooling Load

determined. However, since the condenser water pump only operates 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 to estimate the relationship between water flowrate and pump power for a broad range of system configurations known as the default curve method. The method uses a polynomial expression and predetermined correlation coefficients to the pump's power consumption to the condenser water flow rate [6]. 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 3 shows pump power as a function of flow rate.

$$V_F = 0.21976 - 0.87478 \times P_P + 1.6526 \times P_P \qquad (2)$$

Where,

V_F = condenser water flow rate (% of max)

P_P = pump power (% of max)

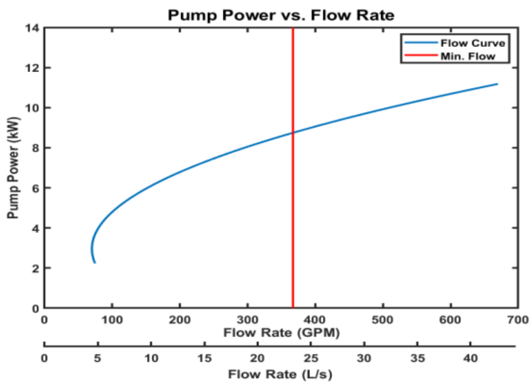

Fig. 3. Condenser Pump Power Input vs. Flow Rate

D. *Cooling Tower Model*

The cooling tower fan control strategy is an important factor in developing an accurate model. The building automation system controlling the cooling tower fan operates the fan speed at 100% until the tower supply water temperature reaches a setpoint of 18.3°C. Once the temperature setpoint has been reached the cooling tower cycles between 25% and 100% fan speed to maintain the tower outlet water temperature near the setpoint. The transient nature of the cycling cooling tower fan would undermine a steady state model, leading to the removal of the data from 9/27-10/30 for training the cooling tower model. As a result, the entire cooling tower dataset is comprised of only one value for both the air and water volumetric flow rates. The NTU-effectiveness model [7] was selected because it can achieve accurate estimations for outlet water temperatures without the extensive data requirements and without complicated tower information. The NTU-effectiveness model is a physical model which is derived from performing a mass and energy balance on counter-flow air and water streams. The model also uses empirically determined constants to estimate the number of heat transfer units in a cooling tower over a range of air and water flow rates. Eqs. 3-5 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 estimate the tower's outlet water temperature.

$$NTU = \frac{h_D A_V V_T}{\dot{m}_a}, \qquad \frac{h_D A_V V_T}{\dot{m}_w} = c\left(\frac{\dot{m}_w}{\dot{m}_a}\right)^n \qquad (3,4)$$

$$NTU = c\left(\frac{\dot{m}_w}{\dot{m}_a}\right)^{1+n} \qquad (5)$$

Where,

h_D = mass transfer coefficient, A_V =surface area of water droplets per unit tower volume, V_T =total tower volume, $\dot{m}_w$ = mass flow rate of water, $\dot{m}_a$ =mass flow rate of air

The constants c and n are specific to a cooling tower that relate the NTU to the mass flow ratio between water and air. The values of c and n 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 where the slope 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 [8]. The constants determined from using only performance data was found to over-predict the cooling tower's effectiveness. The performance dataset was combined with an equal number of data points from the data 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 4. The empirical constants, the R^2 and, the Root-mean-square (RSME) deviation of the performance dataset and the mixed data set are given in Table 3. Figure 5 shows the measured vs. predicted outlet water temperature for the entire dataset after removing the period from 9/27- 10/31 which was used for validation.

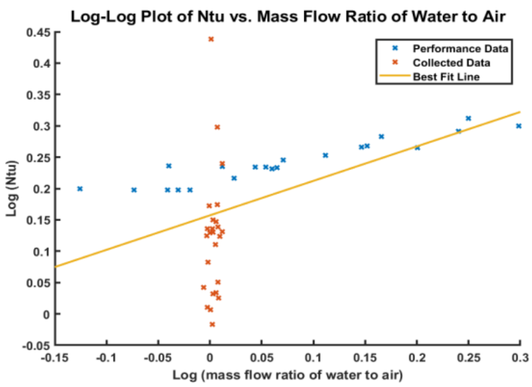

Fig. 4. Log-Log Plot of NTU vs. Mass Flow Ratio of Water to Air

TABLE III. COOLING TOWER CONSTANTS & STATISTICS

Dataset	C	n	RSME	R^2
Performance	1.66379	-0.7017	0.917	1.343
Mixed	1.43714	-0.4378	0.929	1.186

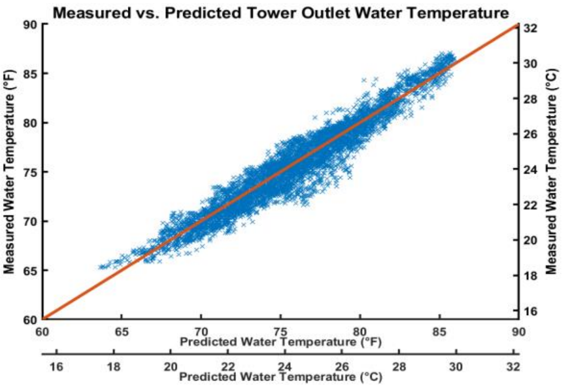

Fig. 5. Measured vs. Predicted Tower Outlet Temperature

E. Chiller Model

The primary consideration when analyzing potential chiller models was the model's ability to incorporate the condenser water flow rate as an optimization variable. 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 [9]. The thermodynamic equation governing the original chiller model can be seen in Eq. 6.

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

Where,
$T_{e,o}$ =Evaporator Outlet Temperature, Q_L =Cooling Load, R =Heat Exhanger Themal Resistance, $T_{c,i}$ =Condenser Inlet Temperature, ΔS_{int} = Internal Entropy Change, L =Rate of Heat Leaks to/from Environment

Additionally, two modifications to the original model have been made over the years that can help improve the model's flexibility. Jiang and Reddy [10] proposed a modification to the model that incorporates a term to make the rate of internal entropy generation linear with respect to the max cooling load. The modified thermodynamic equation can be seen in Eq. 7.

$$\left(\frac{T_{e,o} - Q_L R}{T_{c,i}}\right)\left[1 + \frac{1}{COP}\right] = 1 + \frac{T_{e,o}\left(\Delta S_{int,1} + \Delta S_{int,2}\frac{Q_L}{Q_{max}}\right)}{Q_L} + \frac{L}{Q_L}\left[\frac{T_{c,i} - T_{e,o}}{T_{c,i}}\right] \tag{7}$$

Where,
Q_{max} =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. Gordon et.al. [11] 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 representing the thermal resistance of the evaporator and condenser separately. The full variable condenser flow Gordon model can be seen in Eq. 8. After making a simplifying assumption that the heat exchanger constant K is approximately unity, the thermodynamic relationship can be solved with multiple linear regression of Eq. 9.

$$\left[1 + \frac{1}{COP}\right] = 1 + \frac{T_{e,o}\Delta S_{int}}{Q_L} + \frac{L}{Q_L}\left[\frac{T_{c,i} - T_{e,o}}{T_{c,i}}\right] + \frac{Q_L}{T_{c,i}}\left[1 + \frac{1}{COP}\right] \times \left[\frac{1}{\dot{V}_w \rho C\left[1 - \exp\left(-\frac{K}{\dot{V}_w}\right)\right]} + R_e\right] \tag{8}$$

Where,
$\dot{V}_w$ =Volumetric Flow Rate of Water, ρ =Density, C =Heat Capacity, R_e =Evaporator Thermal Resistance, K =Heat Exchanger Constant

$$\left[1 + \frac{1}{COP}\right] = 1 + \frac{T_{e,o}\Delta S_{int}}{Q_L} + \frac{L}{Q_L}\left[\frac{T_{c,i} - T_{e,o}}{T_{c,i}}\right] + \frac{Q_L}{T_{c,i}}\left[1 + \frac{1}{COP}\right] \times \left[\frac{1}{\dot{V}_w \rho C} + R_e\right] \tag{9}$$

Jiang and Reddy [10] and Gordon et. al [11] 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 used 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 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.

$$\left[1 + \frac{1}{COP}\right] = 1 + \frac{T_{e,o}(\Delta S_{int,1} + \Delta S_{int,2})}{Q_L} + \frac{L}{Q_L}\left[\frac{T_{c,a} - T_{e,o}}{T_{c,a}}\right] + \frac{Q_L}{T_{c,a}}\left[1 + \frac{1}{COP}\right] \times \left[\frac{1}{\dot{V}_w \rho C} + R_e\right] \tag{10}$$

$$y = b_1 x_1 + b_2 x_2 + b_3 x_3 + b_4 x_4 \tag{11}$$

Where,

$$y = \frac{T_{e,o}}{T_{c,a}}\left[1 + \frac{1}{COP}\right] - 1 - \frac{Q_L}{T_{c,a}}\left[1 + \frac{1}{COP}\right] \times \frac{1}{\dot{V}_w \rho C}$$

$$x_1 = \frac{T_{e,o}}{Q_L}, \qquad x_2 = \frac{T_{e,o}}{Q_{max}},$$

$$x_3 = \frac{T_{c,a} - T_{e,o}}{T_{c,a} Q_L}, \qquad x_4 = \frac{Q_L}{T_{c,a}}\left[1 + \frac{1}{COP}\right]$$

$$COP = \frac{\frac{T_{e,o}}{T_{c,a}} - \frac{Q_L}{T_{c,a}} \times \frac{1}{\dot{V}_w \rho C} - b_4 \frac{Q_L}{T_{c,a}}}{1 + b_1 \frac{T_{e,o}}{Q_L} + b_2 \frac{T_{e,o}}{Q_{max}} + b_3 \frac{T_{c,a} - T_{e,o}}{T_{c,a} Q_L} + b_4 \frac{Q_L}{T_{c,a}} + \frac{Q_L}{T_{c,a}} \frac{1}{\dot{V}_w \rho C} - \frac{T_{e,o}}{T_{c,a}}} \tag{12}$$

A constant chilled water temperature of 4.4°C was used as the input for the chiller's outlet evaporator water temperature. Figure 7 shows the measured vs. predicted chiller efficiency for the entire dataset. Figure 8 shows 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.

F. Combined Model

The chiller's projected energy consumption is determined using the estimated coefficient of performance from the

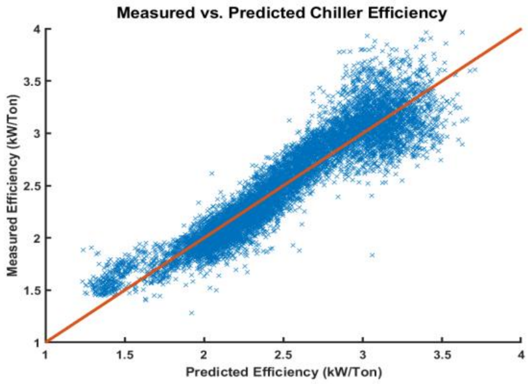

Fig. 6. Measured vs. Predicted Chiller Efficiency

model and the building's predicted cooling load as shown in Eq. 13. Additionally, 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.

$$P_{ch} = \frac{Q_L}{COP} \tag{13}$$

$$T_{c,o} = \frac{P_{ch} + Q_L}{\dot{V}_w \rho C} + T_{c,i} \tag{14}$$

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 7 shows a plot of the modeled and actual inlet and outlet condenser water temperatures.

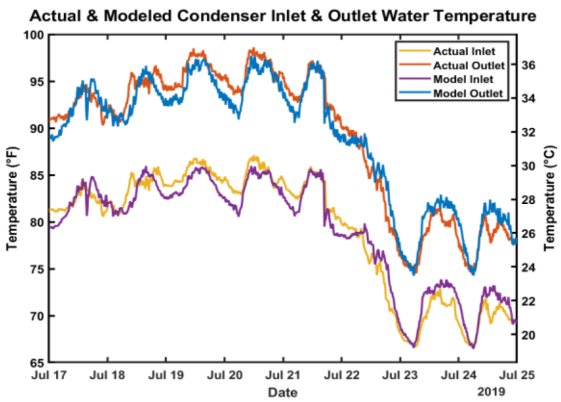

Fig. 7. Actual & Modeled Condenser Inlet & Outlet Water Temperature

G. 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 speed 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 buildings cooling season. The algorithm chosen to minimize the condenser water system's energy consumption was the Frank-Wolfe algorithm [12]. The method is used to optimize constrained smooth nonlinear programs where the derivative of the objective function is not directly available.

IV. Results and Discussion

A. 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 IV. Scenario Control Strategies

Control Scenario	Control Strategy	
	Fan	***Pump***
1	Full Load	Full Load
2	65°F Condenser Water	Full Load
3	Optimized	Full Load
4	Optimized	Optimized

The optimization procedures for scenario 3 & 4 were run from July 11th to July 19th, September 1st to September 9th and from October 21st to October 29th. The 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 results of the different scenarios over a range of external conditions. The results of the optimization scenarios for the first time period can be seen in Figures 8. The figures for the other two periods have been ommited but show a similar pattern between the optimization scenarios.

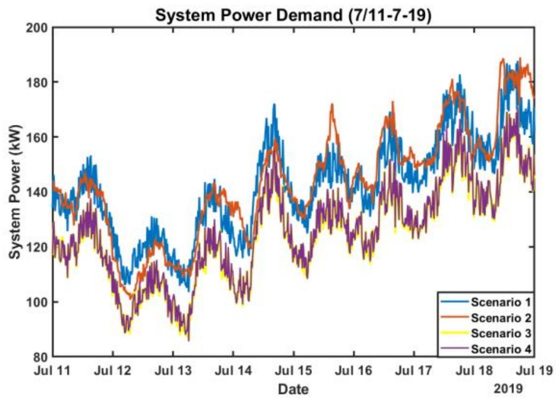

Fig. 8. System Power Demand for Scenarios (1-4) from (7/11-7/19)

Integrating the condenser water system's power demand over the three optimization periods offers an estimate for each scenario's energy consumption during high, medium and low load conditions. Table 5 shows the percent change in energy consumption relative to scenario 2 as the baseline.

TABLE V. SCENARIO PERCENT SAVINGS VS. BASELINE

Control Scenario	% Savings (7/11-7/19)	% Savings (9/1 – 9/9)	% Savings (10/21-10/29)
1	1.12%	-2.42%	-9.73%
2	-	-	-
3	12.96%	12.41%	14.93%
4	12.58%	12.23%	15.01%

B. Discussion

The results of the optimization procedure suggest that for a condenser water system serving a single chiller there is almost no energy saving potential to controlling the condenser water flow rate and cooling tower fan with VFDs, but significant potential to save energy through optimizing the cooling tower fan alone. The findings are indicative only to the specific system analyzed.

The results for scenario 3 were further analyzed to find if there exists 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 9 shows the optimized cooling tower fan speed vs. the ambient wet-bulb temperature.

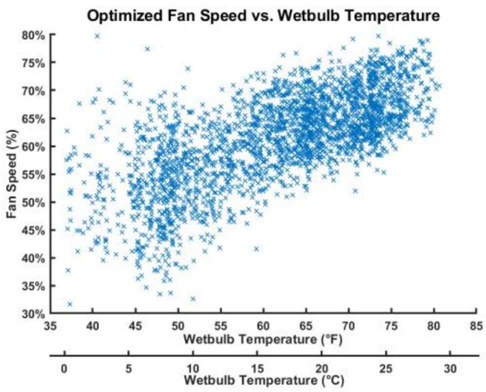

Fig. 9. Optimized Cooling Tower Fan Speed vs. Ambient Wet-bulb

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. Liu and Chuah suggested resetting the condenser water temperature setpoint based on the optimal approach temperature for a cooling tower [13]. The approach is defined as the temperature difference between the tower outlet water temperature and the ambient wet-bulb temperature. Following the control strategy proposed by Liu and Chuah, the approach temperature determined from the optimized cooling tower fan speed was compared to the ambient wet-bulb. Figure 10 shows the optimized cooling tower's approach vs. the ambient wet-bulb temperature.

The correlation between the variables would suggest that resetting the condenser water temperature setpoint based on the optimal approach would be an advantageous control strategy. The difficulty with the indirect control technique is in determining the appropriate cooling tower fan speed to achieve the optimal tower approach. The model would need to operate the cooling tower fan speed with the object of attaining a specific tower outlet water temperature, which may or may not be an improvement over optimizing the system in real-time.

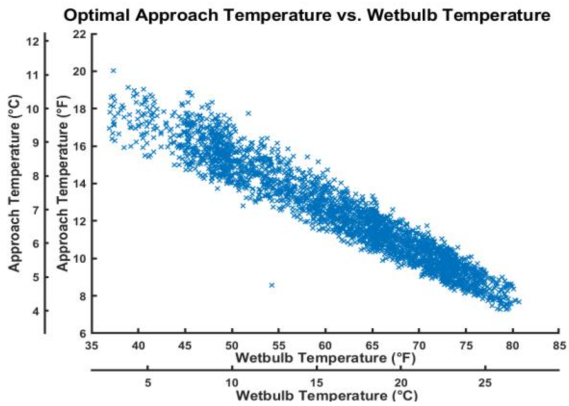

Fig. 10. Optimal Tower Approach vs. Ambient Wet-bulb

REFERENCES

[1] Annual energy outlook (2018). US Energy Information Administration. [Online]. Available: www.eia.gov/outlooks/aeo/pdf/aeo2019.pdf. (accessed: 01.24.2019)

[2] Commercial building energy consumption survey (2012): Table (30b.) cooling energy sources. US Energy Information Administration. [Online]. Available: https://www.eia.gov/consumption/commercial/data/2012/bc/cfm/b30.php (accessed: 01.29.2019)

[3] G. Thomson, The museum environment, 2nd ed. Butterworth-Heinemann, 2013.

[4] H. Diamond, T. Karl, M. Palecki, C. Baker, J. Bell, R. Leeper, D. Easterling, J. Lawrimore, T. Meyers, M. Helfert, G. Goodge, and P. Thorne. (2013) U.s. climate reference network after one decade of operations: status and assessment. NOAA.

[5] C. Fan and Y. Ding, “Cooling load prediction and optimal operation of hvac systems using a multiple nonlinear regression model,” Energy and Buildings, vol. 197, 2019.

[6] J. Romberger, Chapter 18: Variable Frequency Drive Evaluation Protocol. The Uniform Methods Project. Methods for Determining energy efficiency savings for specific measures. National Renewable Energy Laboratory, 2017.

[7] J. E. Braun, “Methodologies for the design and control of central cooling plants,” Ph.D. dissertation, 1988.

[8] Series V cooling towers. Baltimore Aircoil Company. [Online]. Available: http://www.refrisistemas.com/docs/504edcf4Broshure.pdf (accessed: 03.08.2019)

[9] J. Gordon and K. Ng, “Thermodynamic modeling of reciprocating chillers,” Journal of Applied Physics, 1994.

[10] W. Jiang and T. A. Reddy, “Reevaluation of the gordon-ng performance models for water-cooled chillers,” ASHRAE Transactions 109, 2003.

[11] J. Gordon, K. Ng, H. Chua, and C. Lim, “How varying condenser coolant flow rate affects chiller performance:thermodynamic modeling and experimental confirmation,” Applied Thermal Engineering, 2000.

[12] M. Frank and P. Wolfe. (1956) An algorithm for quadratic programming.

[13] C.-W. Liu and Y.-K. Chuah, “A study on an optimal approach temperature control strategy of condensing water temperature for energy saving,” International journal of refrigeration, vol. 34, no. 3, pp. 816–82.