Optimization of Condenser Water Loop Control in Hot and Humid Climates

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ABSTRACT
A chilled water plant optimization program has been implemented on central plants at Texas A&M University, College Station, Texas. Due to the conflict conclusions drawn by other researchers, there was a debate on the advantage of applying variable condenser water flow control sequence. A calibrated component-based model is used to evaluate the energy savings potential with variable condenser water flow control sequence. The simulation result indicates that maximum achievable energy savings at the load profile of case study plant is only 1.03%. If counting the requirement of cooling tower minimum water flow, the energy saving will be negative which means the annual power consumption may increase by 3.4%. Considering the increased complexity of control sequence, it is not recommended to implement variable condenser water flow control sequence for case study project.

INTRODUCTION
A condenser water loop consists of condensers, condenser water piping, pumps, and cooling tower. Condenser water transfers cooling load and heat generated by compressors into cooling tower, which rejects heat to environment through heat transfer and evaporation to the ambient air. Condenser water loop operation has significant effect on the overall system performance. The main issues for efficient operation of condenser water loop are the performance of cooling towers as well as the interactions between cooling tower, condenser water pumps and chillers.

The cooling tower leaving water temperature and the condenser water flow rate are the main inputs that are directly related to the optimization the condenser side. However, only the condenser water supply temperature is considered for most chiller plant optimization application because usually the effects of variable condenser water flow are not well known.

Research has shown (Taylor S. 2012) that variable condenser water flow control logic caused energy use to increase vs. constant condenser water flow in humid climates. Therefore, it is recommended that a variable condenser water flow control sequence is implemented only on plants in dry climates.

Zhang Z. and Liu J. (2012) stated the optimal condenser water flow is highly dependent on the design conditions and the sensitivity of chiller performance to the condenser water flow. The location and climate have a minor effect on the optimal condenser water flow. Compared to the operation with a constant optimal condenser water flow, the savings of applying variable condenser water flow are negligible.

Yu F.W. (2008) proposed a load based speed control for condenser water loop. The results showed that for a chiller operating at above half load, it is possible to adjust the condenser water flow rate linearly with the part load

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ratio. A case study showed that the optimum control could bring about a drop of 5.3% in the operating cost.

Hartman T. (2001) suggested using all variable speed equipments on the heat rejection side and modulating their speed based on load conditions, which resulted in the highest overall plant performance at part load conditions.

A chilled water plant optimization program was implemented on a central plant at Texas A&M University, College Station, Texas. Because of the conflict conclusions above drawn by these researchers, there was a debate on the advantage of applying variable condenser water flow control sequence on this project. To solve this issue, a systematic simulation and analysis are required. In this paper, authors developed a component-based model and calibrated it using measured plant data collected from the building automation system. An example of chiller system is used and the total power of chiller compressor, condenser water pump, chilled water pump and fan motors are minimized by optimizing cooling tower leaving temperature set point for both constant and variable condenser water flow scenarios.

**DESIGN INFORMATION**

Figure 1 shows the diagram of an example of condenser water loop. Which considering calculating annual electricity energy and costing savings with the application of variable condenser water control strategies in hot and humidity climate zone. The design information of chiller and cooling towers are presented in Table 1 and Table 2, respectively.

![Diagram of condenser water loop](image)

**Figure 1** The diagram of an example of condenser water loop

<table>
<thead>
<tr>
<th>CHLRs</th>
<th>Capacity</th>
<th>CHW Flow</th>
<th>CW Flow</th>
<th>Efficiency</th>
<th>Power Input</th>
<th>CW</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Ton</td>
<td>kW</td>
<td>GPM</td>
<td>L/S</td>
<td>kW/ton</td>
<td>COP</td>
</tr>
<tr>
<td>CHLR301</td>
<td>2500</td>
<td>8790</td>
<td>5000</td>
<td>315</td>
<td>7500</td>
<td>473</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>CTs</th>
<th>Design water flow</th>
<th>Min. water Flow ratio</th>
<th>Entering water temp.</th>
<th>Leaving water temp.</th>
<th>Fan Motor</th>
<th>WB</th>
<th>L/G</th>
<th>KAV/L</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>GPM</td>
<td>L/S</td>
<td>%</td>
<td>°F</td>
<td>°C</td>
<td>°F</td>
<td>°C</td>
<td>kW</td>
</tr>
<tr>
<td>CT301</td>
<td>4800</td>
<td>302</td>
<td>60</td>
<td>97.6</td>
<td>36.4</td>
<td>87.6</td>
<td>30.9</td>
<td>45</td>
</tr>
<tr>
<td>CT302</td>
<td>4800</td>
<td>302</td>
<td>60</td>
<td>97.6</td>
<td>36.4</td>
<td>87.6</td>
<td>30.9</td>
<td>45</td>
</tr>
</tbody>
</table>

It is an all-variable speed centrifugal chiller plant that consists of one variable-speed water-cooled centrifugal chiller, one variable-speed condenser water pump, one variable speed chilled water pump and two VSD-equipped induced draft type cooling towers. The two towers are staged on at the same time, and the condenser water flow is equally split by the towers. The fan speed is modulated to maintain the tower cooling water leaving temperature.

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(CWLT) set point. The condenser water (CW) flow rate can be adjusted by changing CW pump speed. The chiller is controlled to maintain a constant chilled water leaving temperature. The minimum entering condenser water for the chiller is 59°F (15°C).

The total objective function is to minimize the instant total power of the chiller compressor, condenser water pump and cooling tower fan motors.

\[ P_{\text{tot}} = P_{\text{CHLR}} + P_{\text{CWP}} + P_{\text{fan}} \]  

CHILLER MODELING

A variable speed chiller model is developed based on manufacture performance data.

\[ \frac{P_{\text{output}}}{P_{\text{design}}} = (0.083 \ 650 \ 18x - 0.072 \ 432 \ 79x^2 + 0.214 \ 727 \ 93y - 0.137 \ 789 \ 58y^2 + \ 0.293 \ 738 \ 30xy + 0.648 \ 840 \ 70) \]  

Where:  
X is the chiller partial load ratio \( \frac{\text{Load}_{\text{actual}}}{\text{Capacity}} \)  
Y is the chiller lift ratio \( \frac{\text{CWLT} - \text{CHWLT}}{\text{Design Lift}} \)

The chiller model is calibrated by measured compressor electricity input from 4/30/2018 through 5/30/2018. The comparison of measured power and predicated power by model are shown in Figure 2.

![Figure 2 The measured and predicated chiller power consumption](image)

COOLING TOWER MODELING

The mass and heat transfer process in a cooling tower is complicated. The effectiveness-NTU model is the most popular model in cooling tower simulations, but iterations are required to obtain a converged solution (Braun, 1989). The overall number of transfer units (NTU) can be correlated with the following form:

\[ \text{NTU} = c \left( \frac{\dot{m}_w}{\dot{m}_a} \right)^{1+n} \]  

Where \( \dot{m}_w \) is the water mass flow; and \( \dot{m}_a \) is the air mass flow.

The value of \( c \) is between 1.0 and 3.0 for towers, and \( n \) ranges between -0.4 and -0.8 (Kreider et al., 2002). The cooling tower CWLT at the given airflow rate can be calculated by following equation.

\[ T_{w,o} = T_{\text{ref}} + \frac{\dot{m}_w(T_{w,i}-T_{\text{ref}})c_{pw} - \dot{m}_a(b_{a,o}-h_{a,i})}{\dot{m}_w c_{pw}} \]  

The cooling tower coefficients \( c \) and \( n \) are identified with the design manufacture data. The cooling tower
model is applied for trending data from 10/10/2017 through 12/01/2017. Figure 3 presents the comparison of the measured and predicated cooling tower leaving temperature.

**Figure 3** Comparison of measured and predicated cooling tower leaving temperature

**PUMP AND FAN MODELING**

The pump power is calculated with the following formula:

CW Pump power model:

\[ P_{\text{pump}} = 125 \times \text{Speed}^{1.4} \times 0.745 \]  

(5)

The comparison of measured and predicated CW pump power is shown in Figure 4.

**Figure 4** Comparison of measured and predicated CW pump power.

Because the CW pipe is an open loop system, the CW flow is not proportion to pump speed. According to trending data, a linear regression model is developed to estimate pump speed for different CW flow as shown in Figure 5.

Pump speed =0.7179*CW flow +0.2819

(6)

**Figure 5** The relationship between CW pump speed and flow
Cooling tower fan model:

$$ P_{\text{fan}} = 60 \times R_{\text{Airflow}}^3 \times 0.745 $$

(7)

Where $R_{\text{Airflow}} = \frac{\text{Airflow}}{\text{Design Airflow}}$ and $R_{\text{Airflow}} \geq 20$

Weather and load profile analysis

Weather and chiller partial load ratio play an important role on the operation of a water-cool chiller plant. The operation hours in each wet bulb and partial load ratio bin for one-year data are shown in Table 3.

<table>
<thead>
<tr>
<th>WB</th>
<th>Partial Load Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>°F</td>
<td>30%</td>
</tr>
<tr>
<td>35</td>
<td>2</td>
</tr>
<tr>
<td>40</td>
<td>4</td>
</tr>
<tr>
<td>45</td>
<td>7</td>
</tr>
<tr>
<td>50</td>
<td>10</td>
</tr>
<tr>
<td>55</td>
<td>13</td>
</tr>
<tr>
<td>60</td>
<td>16</td>
</tr>
<tr>
<td>65</td>
<td>18</td>
</tr>
<tr>
<td>70</td>
<td>21</td>
</tr>
<tr>
<td>75</td>
<td>24</td>
</tr>
<tr>
<td>80</td>
<td>27</td>
</tr>
<tr>
<td>85</td>
<td>29</td>
</tr>
<tr>
<td>90</td>
<td>32</td>
</tr>
</tbody>
</table>

RESULTS AND DISCUSSION

When the chiller is operated at constant CW flow mode, the two towers are staged on at the same time, and the condenser water flow is equally split by the towers. Because the minimum water flow ratio of cooling tower only accounts for 60% of design flow, when a variable CW flow control strategy is implemented, the two cooling towers cannot stage on at same time if the chiller condenser water flow is lower than 5760 GPM (363 L/S). In other words, when the chiller partial load ratio is lower than 76%, only one cooling tower can be running to serve the chiller. At this scenario, the energy savings result can not represent the saving potential of variable CW flow control strategy because the operation scheme of cooling tower is different. Therefore, the following two scenarios are studied in this paper to illustrate the general application and this specific case study respectively.

**Scenario 1** is for general application. Assume no minimum CW flow limitation for each cooling tower, and the minimum chiller CW flow ratio is 60% of design.

**Scenario 2** is for specific case study. The minimum limit of tower water flow is 60%, and the minimum chiller CW flow ratio is 60% of design.

The energy saving percentage of Scenario 1 with difference partial load ratio at different wet bulb is shown in Figure 6.
The energy saving percentage increases when the partial load ratio reduces. For each given partial load ratio, the maximum energy saving achieved when outside wet bulb temperature reached around 50°F (10°C). Because the low limit of CW entering temperature is 59°F (15°C), a bypass control sequence will take effect when wet bulb temperature is below 50°F (10°C). Table 4 compared the distribution of energy savings percentage with different load ratio at different wet bulb for the case study plant.

### Table 4: Distribution of energy savings percentage with different load ratio at different wet bulb

<table>
<thead>
<tr>
<th>WB</th>
<th>0°F</th>
<th>1°C</th>
<th>30%</th>
<th>40%</th>
<th>50%</th>
<th>60%</th>
<th>70%</th>
<th>80%</th>
<th>90%</th>
<th>100%</th>
</tr>
</thead>
<tbody>
<tr>
<td>35</td>
<td>2</td>
<td>15.2%</td>
<td>10.3%</td>
<td>6.3%</td>
<td>3.3%</td>
<td>0.9%</td>
<td>0.5%</td>
<td>0.1%</td>
<td>0.0%</td>
<td></td>
</tr>
<tr>
<td>40</td>
<td>4</td>
<td>15.3%</td>
<td>10.6%</td>
<td>7.0%</td>
<td>4.1%</td>
<td>1.9%</td>
<td>1.1%</td>
<td>0.4%</td>
<td>0.0%</td>
<td></td>
</tr>
<tr>
<td>45</td>
<td>7</td>
<td>15.5%</td>
<td>11.2%</td>
<td>8.5%</td>
<td>6.3%</td>
<td>3.4%</td>
<td>2.1%</td>
<td>0.9%</td>
<td>0.0%</td>
<td></td>
</tr>
<tr>
<td>50</td>
<td>10</td>
<td>16.5%</td>
<td>13.3%</td>
<td>10.1%</td>
<td>7.6%</td>
<td>4.0%</td>
<td>1.9%</td>
<td>0.8%</td>
<td>0.0%</td>
<td></td>
</tr>
<tr>
<td>55</td>
<td>13</td>
<td>15.0%</td>
<td>11.8%</td>
<td>8.9%</td>
<td>6.6%</td>
<td>3.4%</td>
<td>2.1%</td>
<td>1.1%</td>
<td>0.0%</td>
<td></td>
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<tr>
<td>60</td>
<td>16</td>
<td>13.3%</td>
<td>10.4%</td>
<td>7.7%</td>
<td>5.6%</td>
<td>2.8%</td>
<td>1.7%</td>
<td>0.9%</td>
<td>0.0%</td>
<td></td>
</tr>
<tr>
<td>65</td>
<td>18</td>
<td>11.8%</td>
<td>8.8%</td>
<td>6.6%</td>
<td>4.6%</td>
<td>2.3%</td>
<td>1.0%</td>
<td>0.4%</td>
<td>0.0%</td>
<td></td>
</tr>
<tr>
<td>70</td>
<td>21</td>
<td>10.4%</td>
<td>7.6%</td>
<td>5.3%</td>
<td>3.4%</td>
<td>1.4%</td>
<td>0.8%</td>
<td>0.4%</td>
<td>0.0%</td>
<td></td>
</tr>
<tr>
<td>75</td>
<td>24</td>
<td>9.2%</td>
<td>6.7%</td>
<td>4.7%</td>
<td>3.0%</td>
<td>1.3%</td>
<td>0.7%</td>
<td>0.2%</td>
<td>0.0%</td>
<td></td>
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<tr>
<td>80</td>
<td>27</td>
<td>8.1%</td>
<td>5.5%</td>
<td>3.6%</td>
<td>1.9%</td>
<td>0.9%</td>
<td>0.4%</td>
<td>0.2%</td>
<td>0.0%</td>
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<tr>
<td>85</td>
<td>29</td>
<td>6.6%</td>
<td>4.4%</td>
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<td>1.5%</td>
<td>0.4%</td>
<td>0.2%</td>
<td>0.1%</td>
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</tr>
<tr>
<td>90</td>
<td>32</td>
<td>5.5%</td>
<td>3.5%</td>
<td>2.2%</td>
<td>0.8%</td>
<td>0.0%</td>
<td>0.0%</td>
<td>0.0%</td>
<td>0.0%</td>
<td></td>
</tr>
</tbody>
</table>

Table 3 and Table 4 clearly illustrate opposite color maps of operation hours and energy savings percentage. There are only few hours that the case study plant operated at high energy savings percentage bins. It can explain why the annual energy savings are only 1.23%, which is 124,318 kWh. The annual cost avoid is only $5400 if the average electricity rate of FY17, $0.043/kWh, is applied.

For scenario 2, the chart of energy savings percentage is significantly different with scenario 2, because only a
cooling tower can be staged on when the partial load ratio is lower than 76%. At this scenario, the annual power consumption increases by 3.4%, 343,726 kWh. The increase cost is $14,780. The energy saving percentage of scenario 2 with difference partial load ratio at different wet bulb is shown in Figure 7.

![Figure 7 The energy saving percentage of scenario 2 for difference partial load ratio and wet bulb](image)

**CONCLUSION**

This paper focuses specifically on energy savings potential of variable condenser water flow in hot and humidity climate zone. A calibrated component-based model is implemented to calculate total power of chiller compressor, condenser water pump, chilled water pump and fan motors for both constant and variable condenser water flow scenarios. Because the specific low flow constraint of cooling tower, two scenarios are simulated to evaluate the energy savings potential of variable condenser water flow for general application and specific case study plant.

For general application (scenario 1), which is assumed that there is no minimum water flow requirement of cooling tower and the two cooling towers can always run at same time, reducing condenser water flow can enhance chiller plant performance. The more energy savings will be achieved when both of partial load ratio and wet bulb temperature are lower. Therefore, variable condenser water flow is an energy reduction measure for a candidate located in lower wet bulb climate zone and operated at low partial load ratio for majority of hours. Because the load ratio and wet bulb temperature are high for case study plant, the energy savings potential is small when implementing condenser water flow control strategy. The annual energy reduction is only 1.03%, which is 124,318 kWh. When the average electricity rates of FY17, $0.043/kWh, is used, the annual cost avoid is only $5,400.

For this specific case application (scenario 2), the annual power consumption increases 3.4 %, which is 343,726 kWh. The cost avoid is -$14,780.

For both scenarios, there is no significant energy reduction when reducing condenser water flow during partial load for the case study plant. Considering that the increased complex of control sequence, it is not recommended to implement variable condenser water flow control sequence for case study project.

**NOMENCLATURE**

- CWLT = condenser water leaving temperature
- CHWLT = chilled water leaving temperature
- CW = condenser water
- CHW = chilled water
CT = cooling tower
CWP = condenser water pump
CHLR = chiller
PPMP = primary pump
DB = dry bulb temperature, °F (°C)
WB = wet bulb temperature, °F (°C)
P = power, kW
NTU = number of transfer units
\(c\) = cooling tower model coefficients
\(c_{pw}\) = water heat capacity, Btu/lbm °F (W/kg °C)
\(m\) = mass, lbm (kg)
\(n\) = cooling tower model index
\(T\) = temperature, °F (°C)
GPM = gallons per minute
PLR = part load ratio
VSD = variable speed drive
\(x\) = independent variables
\(y\) = independent variables
\(V\) = flow rate, gpm

**Greek Symbols**

\(\rho\) = density, lbm/ft (kg/m³)

**Subscripts**

\(a\) = air
\(w\) = water
\(o\) = outlet
\(i\) = inlet
\(ref\) = reference

**REFERENCES**


