2014

Application of Near-Optimal Tower Control and Free Cooling on the Condenser Water Side for Optimization of Central Cooling Systems

Rita Cristina Jaramillo
Purdue University, West Lafayette, IN., rjaramil@purdue.edu

James E. Braun
Purdue University, West Lafayette, IN., jbraun@purdue.edu

W. Travis Horton
Purdue University, West Lafayette, IN., wthorton@purdue.edu

Application of Near-Optimal Tower Control and Free Cooling on the Condenser Water Side for Optimization of Central Cooling Systems

Rita JARAMILLO[1*], James BRAUN[2], Travis HORTON[3]

[1]*School of Mechanical Engineering, Purdue University, West Lafayette, IN
E-mail: rjaramil@purdue.edu

[2]Professor of Engineering, School of Mechanical Engineering
Purdue University, West Lafayette, IN
E-mail: jbraun@purdue.edu

[3] Assistant Professor of Civil Engineering with Courtesy Appointment in Mechanical Engineering
Purdue University, West Lafayette, IN
E-mail: wthorton@purdue.edu

* Corresponding Author

ABSTRACT

This paper presents an application of tower fan control for optimization of the performance of chiller plants combined with free cooling on the condenser water side. Mathematical models including all the main components of an existing cooling plant were developed and implemented in MATLAB. Simulation results include a mapping of the performance of the plant working in free cooling mode which was used to select control parameters for free cooling operation. Then a mapping of the plant operating with chillers was developed to find the correlation between load and near-optimal air flow, which is the basis of the near-optimal tower control (NOTC) strategy. Finally, simulations were carried out using three consecutive years of historical data to predict the performance of the plant under three different control strategies: 1) tower fan control aiming to keep the temperature of the water supplied to chiller condensers at a constant set point (current control strategy), 2) NOTC and 3) NOTC and free cooling combined. Comparison of the performance of the plant with the baseline (constant condenser water temperature) shows that significant savings can be achieved through the implementation of NOTC along with free cooling. It is expected that the methodology and results of this study provide a useful framework for optimization of cooling plants.

INTRODUCTION

The increasing concern on the ecological impact of end-energy production and use, which is reflected in more stringent environmental regulations worldwide, brings more interest in the development and application of technologies leading to both a reduction on the energy demand and a higher efficiency in end-use energy. Currently, one of the largest end-use energy shares in the U.S. is the energy consumed by HVAC systems in residential and commercial buildings. A relatively low-cost approach for realizing energy savings is to improve controls.

This paper documents development and application of a near-optimal tower control strategy (NOTC) combined with free cooling that can achieve significant energy savings in large chiller plants. A description of NOTC is provided by Braun (1989) and ASHRAE (2011). A simplified approach for NOTC is also presented by Braun (1990) and a more recent extension of NOTC for hybrid cooling plants (Braun, 2007). Free cooling takes advantage of low-ambient temperatures to provide cooling without using the energy-consuming chillers. Two basic approaches to do this are air-side economizers and water-side free cooling. Water-side free cooling, the option considered in this study, has been extensively investigated and some studies provide useful guidelines and recommendations for the design and implementation of these systems (Goswami and Reveliotty, 1987) and (Vallabhaneni, 2006).

The cooling system of Purdue University Campus at West Lafayette was used in the current study to illustrate the application of the free cooling and NOTC. For this purpose, mathematical models of the main components of the cooling system (chillers, cooling towers, and pumps) were developed based on performance data and assembled in MATLAB to predict the performance of the cooling plant under different operating conditions. First, computational simulation was carried out to map the performance of the plant operating in free cooling mode in order to determine the maximum wetbulb temperature suitable for free cooling operation. Further, a map of the chillers performance was developed to find the optimal air flow. Regression of the results was used to obtain a correlation between load and near-optimal air flow, which is the basis of the NOTC strategy. Finally, simulations were carried out using three consecutive years of historical data to predict the integrated performance of the plant under three different control strategies: 1) tower fan control to keep the temperature of the water supplied to chiller condensers constant (current control strategy), 2) NOTC w/o free cooling and 3) NOTC and free cooling combined. An estimation of the average annual energy and CO_2 savings obtained taking the current control strategy as the baseline is presented. Results show that significant savings can be achieved through the implementation of NOTC along with free cooling.

1. BACKGROUND

1.1. Cooling System Overview

The case-study considered in this work is the Purdue Campus at West Lafayette, IN, which has more than 150 buildings in operation, the largest of them requiring cooling even during the winter. Currently the campus cooling demand is satisfied by a chilled water production system, consisting of two physically separated plants: the Wade Power Plant and the Northwest Chiller Plant (NWCP) with nominal cooling capacities of 79,000 kW and 49,600 kW respectively. These two plants are on opposite sides of the campus and deliver chilled water to all the buildings through 37 km of underground piping. The Wade Power Plant, which also supplies part of the electricity to the campus through steam-driven generators, uses a combination of electric and steam-driven chillers. Given that the cooling towers at the Wade Plant are continuously used to cool the generators this prevents them from being used for free cooling. Consequently, the study will focus on the NWCP.

The NWCP equipment consists mainly of 6 dual centrifugal chillers, 4 cooling towers, 6 chiller pumps, 6 condenser water pumps and 6 system pumps as shown in Figure 1. The water returning from campus (chilled water loop) is circulated through the chillers by the chiller pumps, and then returned to campus through the high pressure-system pumps. The number of chillers and pumps in operation is controlled to meet the campus demand. The chiller condensers' water circulates through the cooling towers and is stored in the cold well, a common reservoir with capacity of 340m^3, from which it is pumped again to the chiller condensers by the condenser pumps. Specifications of the chillers and the cooling tower cells are provided in Tables 1 and 2, respectively.

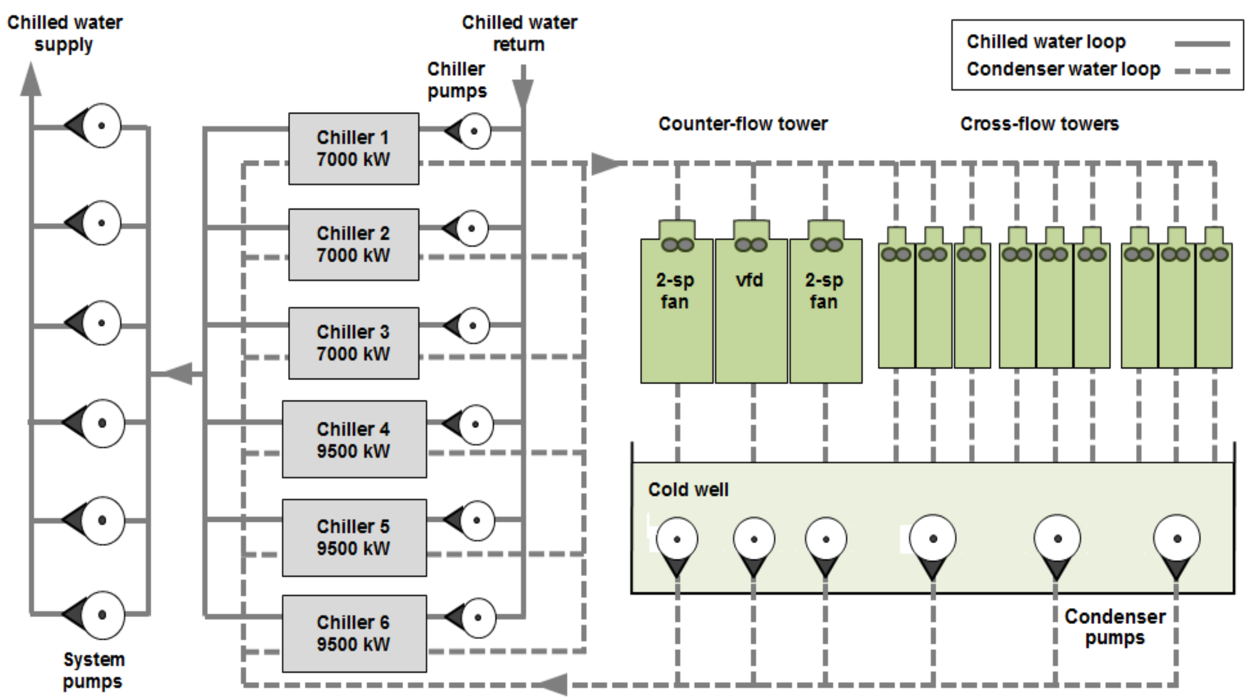

Figure 1. Northwest Plant Schematics

Table 1. Northwest Plant chillers

Description	Units Nr.	Capacity kW	Evaporator flow lt/s	Condenser flow lt/s	Nominal power kW
Centrifugal chiller duplex	3	7000	202	379	1205
Centrifugal chiller duplex	2	9500	271	486	1633
Centrifugal chiller duplex	1	9500	273	511	1540

Table 2. Northwest Plant cooling towers

Description	Units Nr.	Water flow per cell m^3/s	Fan Type	Air flow per cell m^3/s	Unit power kW
3-cell counter flow tower	1	379	vfd and 2-speed	226	55.9
3-cell cross-flow tower	3	162	Single speed	118	44.7

1.2. Free Cooling Approach

The free cooling approach considered in this study is called indirect free cooling, and consists in using a plate-frame heat exchanger piped in a parallel by-pass circuit with the chillers to allow the cooling tower water to cool the campus chilled water when conditions are appropriate as shown in Figure 2.

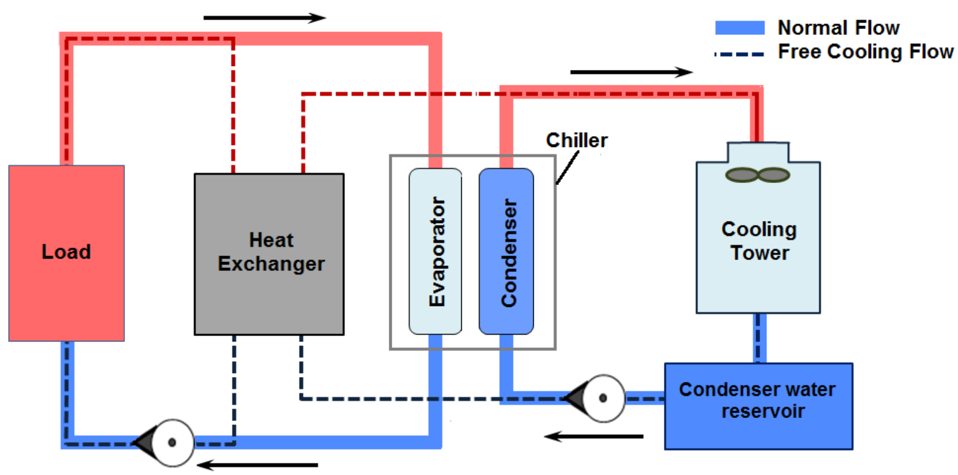

Figure 2. Indirect free cooling scheme

1.3. Near-Optimal Tower Control (NOTC)

With the current operation strategy at the NWCP, the condenser water pumps are sequenced with chillers such that \ each chiller's condenser operates approximately with the nominal design flow. In addition, there is feedback control of the tower fans to maintain a constant condenser water supply temperature of 23°C, even though the minimum inlet water temperature for each chiller condenser is 13°C. It can be expected, therefore, that significant energy savings are attainable through the implementation of NOTC. The strategy described by ASHRAE (2011) comprises the determination of the optimal tower air flow (the one that maximizes the plant COP) followed by the tower sequencing to provide the required air flow with minimum power consumption. Even though the chiller plant power consumption is a function of ambient wet bulb temperature, Braun et al. (1989) demonstrated that the air flow that minimizes the power consumption is nearly a linear function of the cooling load and the dependency on wetbulb is small compared with the load effect. Consequently, it is possible to find a correlation that gives a near-optimal air flow for any cooling plant and then apply sequencing rules for bringing tower fans online until the required air flow is achieved with minimum fan power consumption. In the case of NWCP, the 2-speed fans should be activated first at low speed when adding tower capacity and the variable-speed fan adjusted to meet the requirements. Multiple or single-speed fans are added when the variable-speed fan speed matches the speed associated with the next fan increment to be added.

2. SYSTEM MODELING

Previous work developed by Jaramillo et al. (2014) presents the development of a mathematical model of the NWCP plant in free cooling mode, consisting of the cooling towers, condenser water pumps and a plate-frame heat exchanger modeled as a constant effectiveness heat exchanger. Here the model is extended by incorporating performance models for the chillers and chiller pumps along with psychrometric functions developed from the correlations given in ASHRAE (2011), to simulate the transient operation of the NWCP.

In order to model the chillers, performance data was used to correlate the power consumption with load ratio and the temperature lift (the difference between the condenser flow and the evaporator flow leaving temperatures), using a biquadratic fitting of performance data (Braun, 1987).

$$Z = a_0 + a_1X + a_2X^2 + a_3Y + a_4Y^2 + a_5XY \tag{1}$$

In the above expression, Z is the power ratio (Equation 2), X is the partial load ratio (Equation 3), Y is the non-dimensional temperature lift (Equation 4) and the coefficients a_0 to a_5 are determined by regression using chiller performance data. Figure 3 shows a comparison between the power ratio obtained from performance data and the power ratio calculated with the correlation for the 7000 kW chillers. It can be observed that the correlation provides a close fit of the performance data. Similar results were obtained for the other two types of chillers. The root mean square error obtained for the three types of chillers was less than 0.005.

$$Z = \frac{Power}{Power_{des}} \tag{2}$$

$$X = \frac{\dot{Q}_{ev}}{\dot{Q}_{ev,des}} \tag{3}$$

$$Y = \frac{T_{co,o} - T_{ev,o}}{(T_{co,o} - T_{ev,o})_{des}} \tag{4}$$

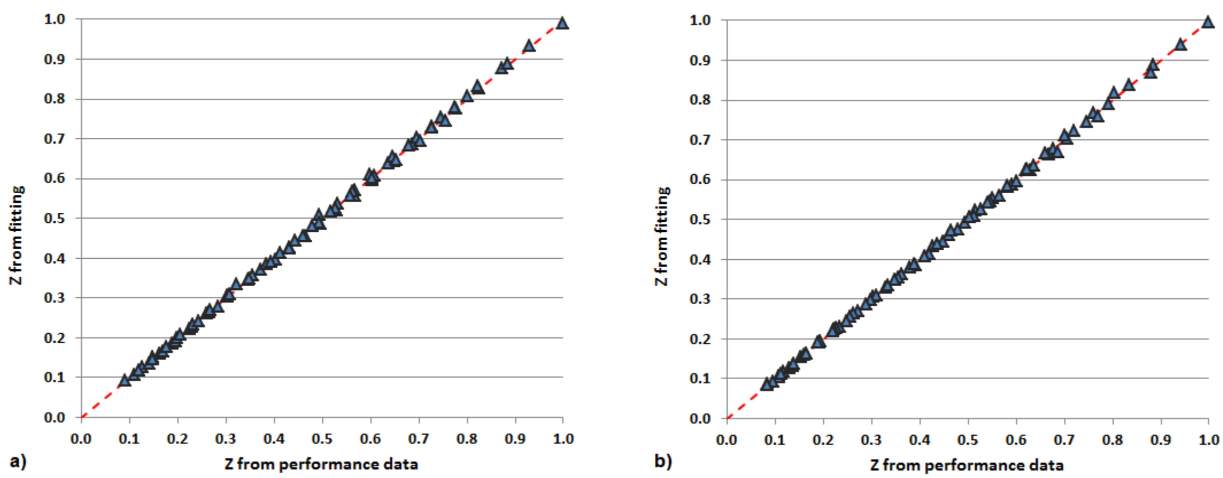

Figure 3. Power Ratio (Z) from fitting vs performance data for a) Chillers 1-3 and b) Chillers 4-5

The condenser heat rejection is related to the chiller load ($\dot{Q}_{ev}$) and the power consumed by the compressor ($Power_m$) by Equation 5, where $\dot{Q}_{co}$ is condenser heat rejection and η_m is the compressor motor efficiency. The energy balances on the chiller evaporator side and the condenser side results in equations 6 and 7, respectively.

$$\dot{Q}_{co} = \dot{Q}_{ev} + \eta_m Power_m \tag{5}$$

$$\dot{Q}_{ev} = \dot{m}_{ev} C_{p,ev} (T_{ev,i} - T_{ev,o}) \tag{6}$$

$$\dot{Q}_{co} = \dot{m}_{co} C_{p,co} (T_{co,o} - T_{co,i}) \tag{7}$$

Substitution of the expression for power given by equations 1 and 2, and the expressions for $\dot{Q}_{ev}$ and $\dot{Q}_{co}$ given by equations 6 and 7 into Equation 5, gives a quadratic expression in Y that may be solved explicitly. Finally, An overall energy balance on the cold well (condenser water reservoir) results in Equation 8, which gives the variation of the cold well temperature (same as condenser water inlet temperature, $T_{co,i}$) in time, assuming that the water in the cold well is fully mixed and neglecting thermal losses.

$$M_{cw}\frac{dT_{co,i}}{dt} = \sum_{j=1}^{12}(T_{tc,o,j}\dot{m}_{tc,o,j}) - T_{co,i}\sum_{i=1}^{12}\dot{m}_{tc,i,j} + T_{mains}\sum_{j=1}^{12}(\dot{m}_{tc,i,j} - \dot{m}_{tc,o,j}) \tag{8}$$

In the above expression M_{cw} is the mass of water stored in the cold well (assumed constant), $T_{tc,o,j}$ is the temperature of the water leaving the j-th tower cell, T_{mains} is the make-up water temperature, $\dot{m}_{tc,i,j}$ and $\dot{m}_{tc,o,j}$ are the water flows entering and leaving the j-th tower cell, respectively (See Figure 1).

The chiller model, based on equations 1 to 7 was integrated with the other components. The transient model programed in MATLAB solves the energy balance (Equation 8) using the Crank Nicolson method in conjunction with the mathematical models of all the components in order to determine the cooling delivered by the plant and also the power consumed by the chillers and auxiliary equipment. The power consumption of tower fans is estimated based on the power at nominal speed using fan affinity laws (power consumption is proportional to the cube of shaft speed). Regarding the pumps, they currently operate very close to design conditions and the power consumed is basically the nominal. When free cooling mode is activated, however, the chiller pumps are turned off and the fluid circulates through the heat exchangers, causing the remaining pump's operating conditions to change. The system head loss is modeled as a function of the square of the flow. Then this equation is solved in conjunction with the correlations of head and power as a function of flow obtained from cubic fitting of pump's performance data in order to find the pump's new operating point and the corresponding power consumption.

Input data for the model consists of hourly campus cooling load, chilled water flow, and ambient conditions (wetbulb and dry bulb temperatures). Four modes of operation are available:

- Free Cooling operation (FC)
- Chiller operation with constant condenser water inlet temperature (CCWT)
- Chiller operation with NOTC w/o Free Cooling.
- Automatic selection: the model switches between chiller operation (either CCWT or NOTC) and free cooling depending on the control setting (maximum wetbulb temperature for free cooling operation)

For chiller operation it is assumed that the chillers are sequenced to meet the campus load up to the plant nominal capacity (49,600 kW). Once the maximum capacity is reached, the remaining load is met by the chillers at Wade Plant. The chiller's evaporator water leaving temperature setting is adjusted between 6 and 9°C according to ambient temperature, the lowest temperature set point corresponding to summer time operation and the highest to winter operation. When free cooling is activated, only the concrete counter-flow tower is used. Tower fans are controlled to maintain the campus chilled water supply setting of 9°C. Once the load exceeds the free cooling capacity, the flow is limited to the amount that can be cooled to the set point. The rest is diverted to the Wade plant.

3. RESULTS

3.1. Free Cooling Mapping

The model previously described was used to estimate the capacity and COP of the NWCP in free cooling mode as a function of the relative total air flow (the ratio of the current air flow to the air flow that would be obtained with all the NWCP towers working at 100% percent capacity) for different wet bulb temperatures. For this purpose, the campus water return temperature was fixed at 12.8 °C, which is the average historical value. Then for each wet bulb temperature considered, the concrete tower air flow was allowed to vary by sequencing the fans from minimum to 100% capacity. Steady state conditions were assumed in each case, so the left-hand side of Equation 8 is zero. The results are presented in Figure 2. As expected the cooling capacity increases almost linearly with air flow and drops as wet bulb increases, being negligible for 8°C. The maximum wetbulb temperature suitable for free cooling operation occurs when the free cooling COP equals the average COP of 6.2 for chiller operation with CCWT during the winter time, and is approximately 6°C. Therefore, the free cooling mode is activated when the ambient wetbulb temperature is below 6°C. Discontinuities are presented in the plots when additional condenser pumps are activated.

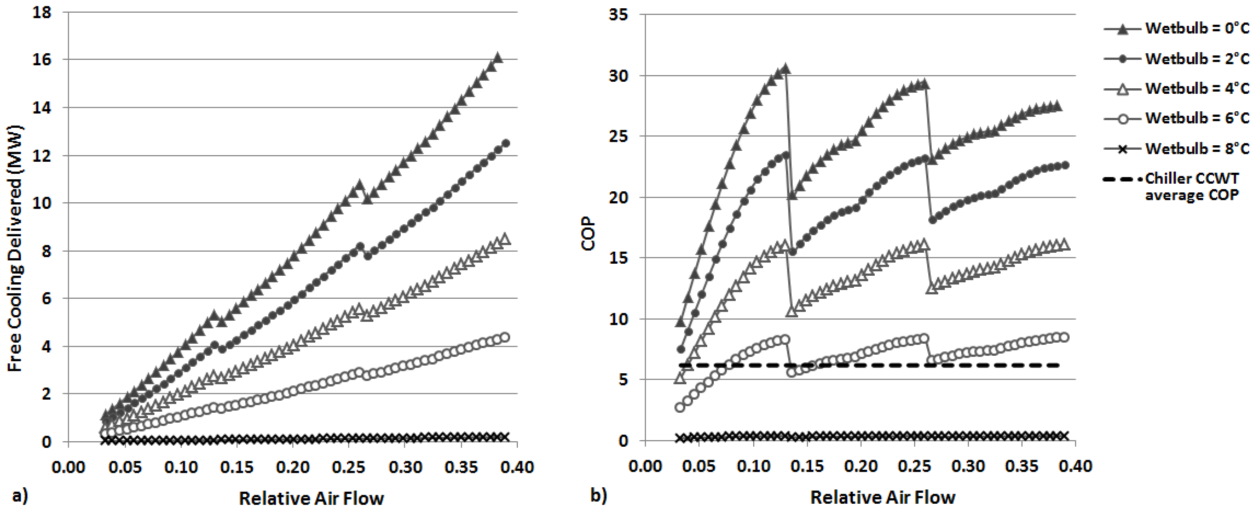

Figure 4. Free cooling map vs. relative air flow a) Free cooling capacity, b) COP

3.2. Chiller Operation Mapping and Near-Optimal Tower Air Flow

The total power consumed by a cooling plant to meet a specific cooling load is a function of ambient wet bulb, tower total air flow and tower fan sequencing. Therefore, in order to determine the air flow that minimizes the plant power consumption (i.e. maximizes the plant COP) the maximum COP was mapped with cooling load according to the following procedure: 1) For a fixed wet bulb temperature, the cooling load was varied from minimum to the plant nominal capacity; 2) For each cooling load, the chiller sequencing necessary to meet the load was determined; 3) The tower fans were sequenced from minimum to maximum air flow capacity and the plant COP was calculated for each value of air flow; and 4) The tower fan sequence that maximized the plant COP (optimal tower air flow) for the given load was selected. The COP and the relative air flow associated with optimal tower air flow (OTAF) are presented in Figures 5 and 6 as a function of load for different wet bulb temperatures. The COP obtained applying the current control strategy (CCWT with condenser water set point of 23°C) is also included in Figure 5 for comparison. It can be seen that the OTAF is more advantageous at low ambient wet bulb temperatures. As the wet bulb temperature approaches the current condenser water set point, both strategies show quite similar efficiency.

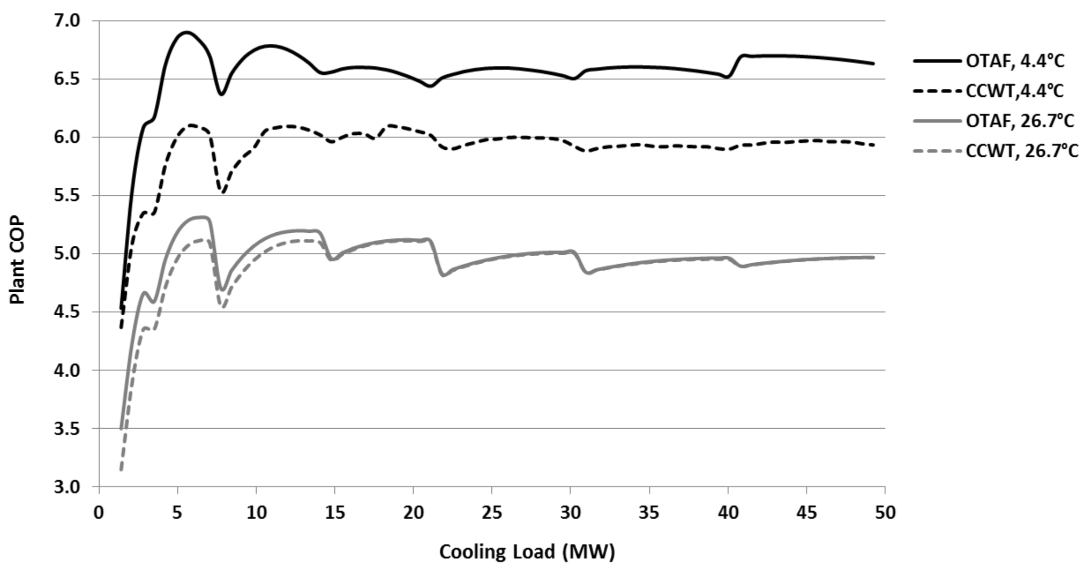

Figure 5. Optimal chiller plant COP vs. COP with constant condenser water temperature

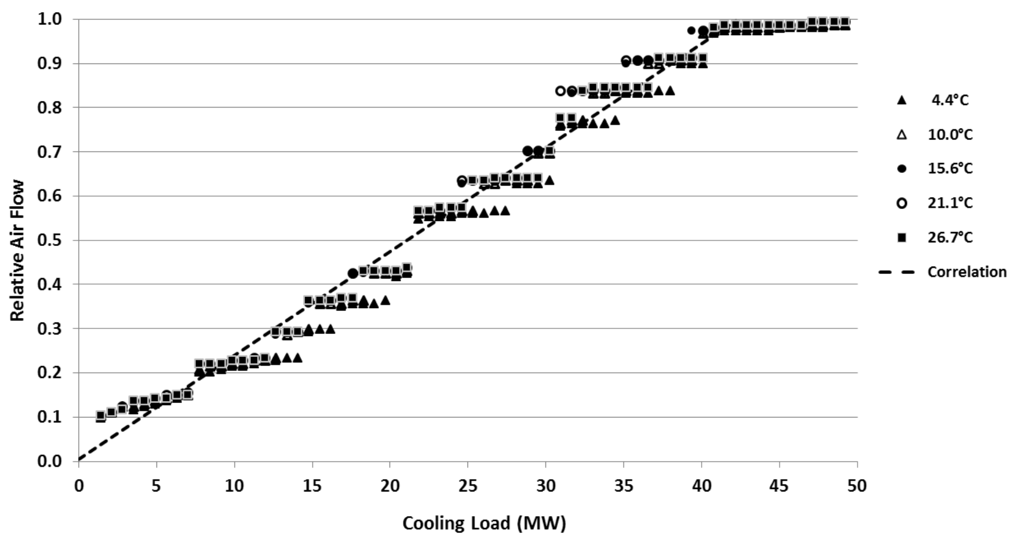

Figure 6. Optimum relative air flow vs. cooling load for different wet bulb temperatures

It can be seen that the plant COP is highly dependent on wetbulb. As expected, the optimum air flow provides more benefit as wetbulb temperature decreases. Figure 6 presents the optimum relative air flow for various wetbulb temperatures. Although the plant COP is highly dependent on wetbulb, it can be seen that the air flow that maximizes the COP is nearly a linear function of the cooling load. This agrees with the results obtained by Braun (1989). Regression of simulation results gave a correlation for near-optimal relative air flow (RAF) that is expressed in Equation 7, where the plant cooling load is given in MW.

$$RAF = min(0.02349311*Load + 0.0048810, 1) \tag{9}$$

3.3. Northwest Plant Performance Simulation

Free cooling performance is highly dependent on ambient conditions and these conditions may vary considerably from one year to the next. Therefore, simulations were carried out using three years of hourly historical data (2011 to 2013) consisting of campus cooling demand, ambient wet bulb temperature and dry bulb temperature. Simulation was carried out for three modes of operation of the NWCP:

- Alternative 1: CCWT with condenser water set point of 23°C. This is the current control strategy of the NWCP and the baseline for comparisons.
- Alternative 2: NOTC w/o Free Cooling.
- Alternative 3: NOTC + Free Cooling

The simulation results are consistent over the three years considered; so for space limitations only monthly results for 2013 are presented in Figures 7 and 8. Figure 7 shows the campus cooling load compared to the cooling delivered by the NWCP for the three alternatives of operation described. The second bar on the graph represents the cooling supplied by the NWCP chillers, which is the same regardless the control strategy (CCWT or NOTC), while the third bar represents the energy delivered by the NWCP plant operating in alternative 3. Given that in this mode the plant automatically switches between chiller operation (NOTC) and free cooling depending on ambient wetbulb temperature, the bar is stacked to show separately the contributions of the chillers and the free cooling to the total energy delivered. This reveals that a substantial part of the cooling demand can be met by free cooling during the coldest months. Finally, the difference between either the second or the third bar and the first one (campus load) corresponds to the amount of cooling that might be supplied by the Wade Plant. Figure 8 compares the plant average COP obtained with the three modes of operation. As expected, the improvement in performance obtained with the implementation of NOTC for chiller operation compared with the current control strategy is more evident during the winter months. Addition of free cooling results in a dramatic improvement in the plant average COP during this period.

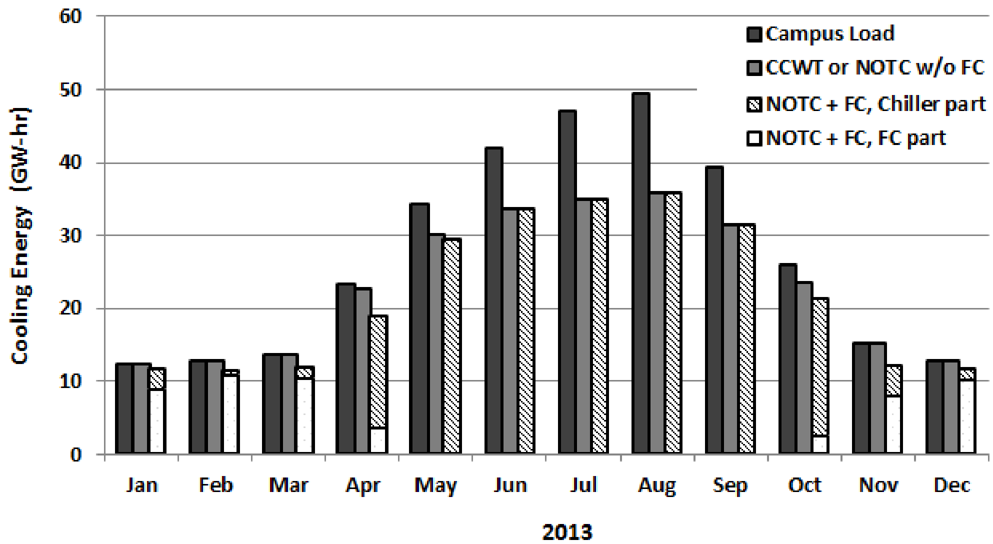

Figure 7. Campus load vs. cooling delivered by NWCP with different alternatives

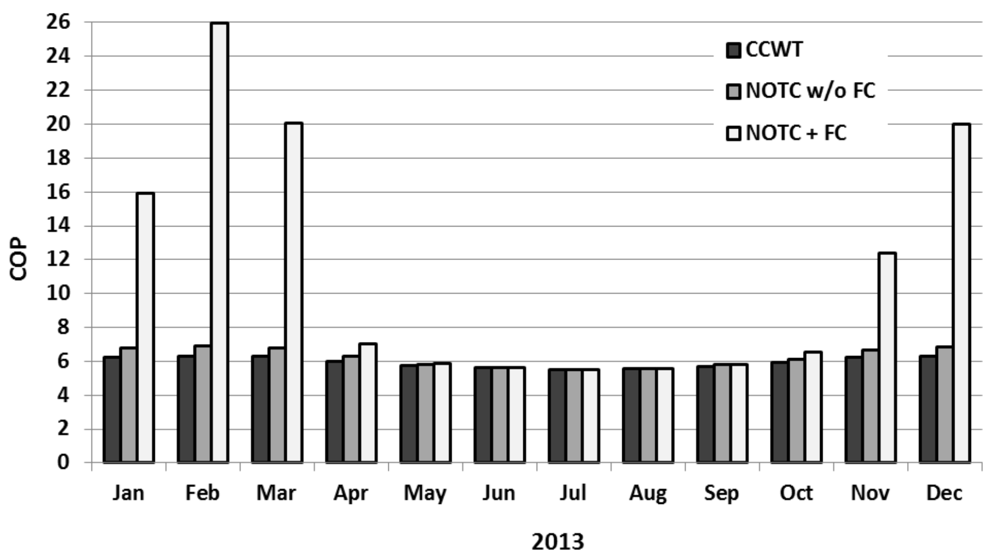

Figure 8. Northwest monthly average COP

A summary of the annual simulation results obtained for the three alternatives of operation of the NWCP is shown Tables 3, 4 and 5. According to these results the NWCP can supply 86% of the campus cooling in normal operation, and 5% less when free cooling is added. The average annual plant COP increases in 3% with the NOTC and an additional increment of 13% is achieved by the inclusion of free cooling.

Table 3. Annual estimates for chiller operation with CCWT

Description	2011	2012	2013	Average
Campus cooling load, GW-hr	328.48	311.21	328.51	322.73
Chiller cooling, GW-hr	282.86	269.44	278.71	277.00
Energy consumption, GW-hr	48.59	46.38	47.84	47.61
Cooling Fraction	0.86	0.87	0.85	0.86
Average Plant COP	5.82	5.81	5.83	5.82

Table 4. Annual estimates for chiller operation with NOTC w/o Free Cooling

Description	2011	2012	2013	Average
Campus cooling load, GW-hr	328.48	311.21	328.51	322.73
Chiller cooling, GW-hr	282.86	269.44	278.71	277.00
Energy consumption, GW-hr	47.13	45.16	46.48	46.25
Cooling Fraction	0.86	0.87	0.85	0.86
Average Plant COP	6.00	5.97	6.00	5.99

Table 5. Annual estimates for chiller operation with NOTC and Free Cooling

Description	2011	2012	2013	Average
Campus cooling load, GW-hr	328.48	311.21	328.51	322.73
Chiller cooling, GW-hr	209.82	213.59	209.46	210.96
Free cooling, GW-hr	51.68	41.31	54.64	49.21
Energy consumption, GW-hr	38.42	38.73	38.36	38.50
Cooling Fraction	0.80	0.82	0.80	0.81
Average Plant Combined COP	6.81	6.58	6.88	6.76

The average of the annual estimates presented above were used to estimate the energy savings attainable through the implementation of NOTC and free cooling taking the current plant operation strategy as the baseline for comparison. These results are presented in Table 6. The total energy consumed to meet the campus load was calculated assuming that the part of the load that could not be met by the NWCP was met by the Wade Plant with an average COP of 5.83 (same as the NWCP for current operating mode). The energy cost savings are based on an electricity price of \$0.05 per kW-hr. To estimate the savings in CO_2 emissions, the electricity savings were converted to primary energy using an average electricity efficiency of 32.87% (EIA, 2011), and the CO_2 emissions from electricity generation (kg CO_2/GW-hr) were calculated using the energy mix for electricity generation in Indiana (U.S. Department of Energy, 2013) and the CO_2 emissions coefficients by fuel (EIA, 2013).

Table 6. Alternatives Comparison Based on Average Annual Estimates

Alternative	Cooling Load GW-hr	Cooling supplied NWCP GW-hr	Cooling supplied Wade GW-hr	Energy consumption NWCP GW-hr	Energy consumption Wade GW-hr	Energy consumption Total GW-hr	Savings** Energy GW-hr	Savings** Costs \$	Savings** CO2 Kg*10^6
1. CCWT	322.73	277.00	45.73	47.61	7.86	55.46	0.00	\$0	0.00
2. NOTC	322.73	277.00	45.73	46.25	7.86	54.11	1.35	\$67,577	1.25
3. NOTC & FC	322.73	260.17	62.56	38.50	10.75	49.25	6.21	\$310,427	5.74

CONCLUSIONS

In this paper, near-optimum tower control (NOTC) and free cooling on the condenser water side were applied to optimize the performance of a large-scale chiller plant. A mathematical model of the plant was developed and simulations were carried out with three consecutive years of historical data to predict the performance of the plant under different control strategies. Comparison with the current strategy shows that an average of 1.35 GW-hr in annual energy savings is attainable through the implementation of NOTC. The addition of free cooling increases the savings up to 6.21 GW-hr, which represents a reduction of \$310,000 in electricity budget. These savings are based on very low electricity costs of \$0.05 per kW-hr that are available for this site. Substantially greater savings would be possible for most locations where electricity rates are double or triple these rates.

Further work will incorporate chiller sequencing and condenser water flow as variables to be optimized. Regarding the free cooling operation, it can be seen that most of the campus demand can be met by free cooling during the coldest months (December, January and February). Further work will evaluate the possibility of including thermal energy storage (TES) along with real time pricing to increase savings during the shoulder months.

NOMENCLATURE

C_p	specific heat	(kJ/Kg°K)
M	mass	(Kg)
$\dot{m}$	flow rate	(Kg/s)
$\dot{Q}$	heat transfer rate	(kW)
RAF	relative air flow	(-)
T	temperature	(°C)
X	partial load ratio	(-)
Y	non-dimensional temperature lift	(-)
Z	power ratio	(-)
η	efficiency	(-)
Subscripts		
co	chiller's condenser	
cw	cold well (condenser water reservoir)	
des	design conditions	
ev	chiller's evaporator	
i	inlet conditions	
m	motor	
mains	make-up water	
o	outlet conditions	
tc	cooling tower cell	

REFERENCES

ASHRAE, 2009. 2009 ASHRAE Handbook- Fundamentals (S.I). Atlanta: American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc., Chap. 1.

ASHRAE, 2011. 2011 ASHRAE Handbook - HVAC Applications. Atlanta: American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc., Chap. 42.

Braun, J.E., 1987. Performance and control Characteristics of Large Central Cooling Systems, ASHRAE Transactions, Vol. 93, Part 1.

Braun, J.E., Klein, S.A., Mitchell, J.W., and Beckman, W.A., 1989. Applications of Optimal Control to Chilled Water Systems without Storage, ASHRAE Transactions, Vol. 95, Part 1, pp. 663-674, 1989.

Braun, J.E. and Diderrich, G.T., 1990. Near-Optimal Control of Cooling Towers for Chilled-Water Systems. ASHRAE Transactions, Vol. 96, Part 2, pp. 806-813.

Braun, J.E., 2007. A General Control Algorithm for Cooling Towers in Cooling Plants with Electric and/or Gas-Driven Chillers. HVAC&R Research, Vol. 13, No. 4, Pages 581-598.

Goswami, D., and C.N. Reveliotty, 1987. Free Cooling by Cooling Tower Water: the Hydraulics and "Hidden Problems" of Water-Side Applications Should Be Factored into the Design from the Start of the Project. ASHRAE Journal. January 1987: 32-37.

Jaramillo, R., Jeon, B., Braun, J.E., Horton, T., and Schuster D., 2013. Simulation Assessment of Free Cooling Technology for a Large Campus. Submitted to ASHRAE Transactions, 2014.

MathWorks, Inc., 2013. MATLAB. Version R2013a (8.1.0.604). Natick, MA.

U.S. Department of Energy, 2013. Indiana Electricity Generation. http://apps1.eere.energy.gov/states/electricity_generation.cfm/state=in?print

U.S. Energy Information Administration, 2012. Annual Energy Review 2011, p. 219. http://www.eia.gov/totalenergy/data/annual/pdf/aer.pdf

U.S. Energy Information Administration, 2013. Carbon Dioxide Emissions Coefficients. http://www.eia.gov/environment/emissions/co2_vol_mass.cfm

Vallabhaneni, K, 2006. Minimizing Energy Costs with Free Cooling. HPAC Engineering. October 2006: 34-41.

ACKNOWLEDGMENTS

The authors would like to acknowledge the collaboration and financial support provided by Purdue Physical Facilities, West Lafayette, IN.