Optimizing Design & Control Of Chilled Water Plants

Part 5: Optimized Control Sequences

By Steven T. Taylor, P.E., Fellow ASHRAE

This is the last of a series of articles discussing how to optimize the design and control of chilled water plants. The series summarizes ASHRAE's Self Directed Learning (SDL) course called *Fundamentals of Design and Control of Central Chilled Water Plants* and the research that was performed to support its development. The articles, and the SDL course upon which it is based, are intended to provide techniques for plant design and control that require little or no added engineering time compared to standard practice but at the same time result in significantly reduced plant life-cycle costs.

A procedure was developed to provide near-optimum plant design for most chiller plants including the following steps:

1. Select chilled water distribution system.
2. Select chilled water temperatures, flow rate, and primary pipe sizes.
3. Select condenser water distribution system.
4. Select condenser water temperatures, flow rate, and primary pipe sizes.
5. Select cooling tower type, speed control option, efficiency, approach temperature, and make cooling tower selection.
6. Select chillers.
7. Finalize piping system design, calculate pump head, and select pumps.
8. Develop and optimize control sequences.

Each of these steps is discussed in this series of five articles. This article discusses step 8.

Typical Chiller Plant

Figure 1 is a typical primary-only variable flow chilled water plant. The plant has two of each major component (chillers, towers, condenser water pumps, and chilled water pumps) each sized for 50% of the load. This plant design is very common and was used as the basis of the simulations and optimization for this series of articles and the SDL course upon which it is based.

Note that the condenser water (CW) pumps in *Figure 1* do not have variable speed drives (VSDs). Sequences for variable speed CW pumps are also addressed in this article but, as discussed in Part 2[1] of this series and in more detail below, VSDs on condenser water

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Modeling the Plant

The plant in *Figure 1,* serving a typical office building, was modeled with all permutations of the following design variables:

Weather Oakland, Calif., Albuquerque, N.M., Chicago, Atlanta, Miami, Las Vegas

CHWST Reset by valve position from 42°F to 57°F

Chillers
- Two styles (two stage and open-drive)
- Efficiency at 0.35, 0.5, and 0.65 kW/ton at AHRI conditions

Towers
- Approach: 3°F, 6°F, 9°F, and 12°F
- Tower Range: 9°F, 12°F, and 15°F
- Efficiency: 50, 70, and 90 gpm/hp

Condenser water pumps With and without VSDs

The control equation coefficients were determined from each run, then these coefficients were themselves regressed against various design parameters and weather indicators. The results are shown below. The development of these regressions is ongoing to include more weather sites and chiller variations.

1. Condenser water temperature control. Control CW return temperature to the setpoint determined from Equation 1a:

$$CWRT = CHWST + A \times PLR + B \quad (1a)$$

A = –63 + 0.0053 × CDD65 – 0.0087 × WBDD55 + 1.67 × WB + 0.52 × APPROACH – 0.029 × GPM/HP

B = 18 – 0.0033 × CDD65 + 0.0053 × WBDD55 – 0.26 × WB + 0.15 × APPROACH – 0.014 × GPM/HP

2. Variable speed condenser water pumps. Control CW flow ratio to the setpoint determined from Equation 2:

$$CWFR = C \times PLR + D \quad (2)$$

C = 1.35 - 1.27E–05 × CDD65 + 1.36 × NPLV – 0.0212 × WB – 0.012 × APPROACH + 0.0765 × RANGE

D = –0.147 + 7.04E–06 × CDD65 – 0.124 × NPLV + 0.0038 × WB + 0.00133 × APPROACH + 0.00217 × RANGE

3. Chiller Staging. Use one chiller when PLR is less than SPLR determined from Equation 3:

$$SPLR = E \times (CWRT - CHWST) + F \quad (3)$$

E = 0.057 – 0.000569 × WB – 0.0645 × IPLV – 0.000233 × APPROACH – 0.000402 × RANGE + 0.0399 × KW/TON

F = –1.06 + 0.0145 × WB + 2.16 × IPLV + 0.0068 × APPROACH + 0.0117 × RANGE – 1.33 × KW/TON

These control sequences strictly apply to primary-only plants with centrifugal chillers serving air handlers with outdoor air economizers in a typical office building. It is not known how well they apply to other applications.

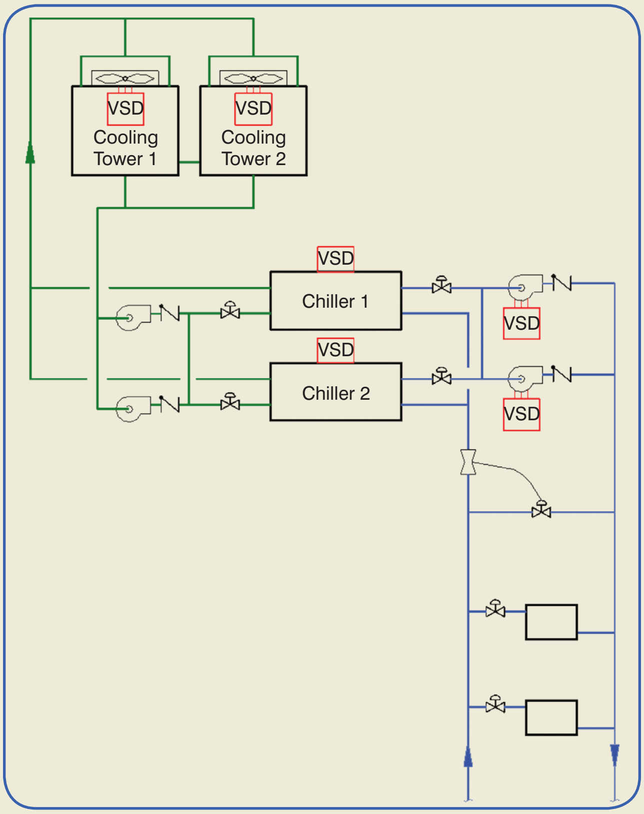

Figure 1: Typical chilled water plant schematic.

Variables

APPROACH	Design tower leaving water temperature minus WB, °F
CHWFR	Chilled water flow ratio, actual flow divided by total plant design flow
CHWST	Chilled water supply temperature (leaving evaporator temperature), °F
CWFR	Condenser water flow ratio, actual flow divided by total plant design flow
CWRT	Condenser water return temperature (leaving condenser water temperature), °F
CDD65	Cooling degree-days base 65°F
DP	Differential pressure, feet H_2O
KW/TON	Chiller efficiency at AHRI conditions, kW/ton
ΔT	Temperature difference, °F
GPM/HP	Tower efficiency per ASHRAE Standard 90.1
IPLV	Integrated part load value per AHRI 550/590, kW/ton
NPLV	Non-standard part load value per AHRI 550/590, kW/ton
RANGE	Design tower entering minus leaving water temperature, °F
PLR	Plant part load ratio, current load divided by total plant design capacity
TOPP	Theoretical optimum plant performance
WB	Design wet-bulb temperature, ASHRAE 1%, °F
WBDD55	Wet-bulb cooling degree-days base 55°F

pumps are usually not life-cycle cost effective for plants serving office building type loads.

Also note in *Figure 1* that the cooling towers do not include any isolation valves to shut off flow to allow one tower to operate alone. As discussed in Part 4[2] of this series, towers generally can be selected with nozzles and dams that allow half flow from one CW pump while still providing full coverage of fill and it is always most efficient to run as many tower cells as possible. So whether one or two CW pumps are operating, both tower cells are enabled and fans are controlled to the same speed.

Determining Optimal Control Sequences

Chilled water plants have many characteristics that make each plant unique so that generalized sequences of control that maximize plant efficiency are not readily determined. Equipment and system variables that affect performance include:

Chillers. Each chiller has unique characteristics that affect full-load and part-load efficiency such as compressor design, evaporator and condenser heat transfer characteristics, unloading devices (such as variable speed drives, slide valves, and inlet guide vanes), oil management systems, and internal control logic.

Cooling towers. Tower efficiency (gpm/hp) varies significantly by almost an order of magnitude between a compact centrifugal fan tower to an oversized propeller fan tower. Towers can also be selected for a wide range of approach temperatures.

Chilled and condenser water pumps. Pumps and piping systems can be selected for a broad range of ΔTs and may or may not include variable speed drives. Pump efficiency also varies by pump type and size and pump head varies significantly depending on physical arrangement and pipe sizing standards.

Chilled water distribution systems. Distribution system arrangements, such as primary-secondary vs. primary-only variable flow, significantly affect plant control logic.

Weather. Changes in outdoor air conditions affect loads and the ability of cooling towers to reject energy.

Load profile. The size and consistency of loads will affect optimum sequences. For instance, control sequences that are optimum for an office building served by air-handling systems with airside economizers may not be optimum for a data center served by systems without economizers.

With so many variables, no single control sequence will maximize the plant efficiency of all plants in all climates for all building types.

There are a number of papers[3,4] on techniques to optimize control sequences for chilled water plants. Almost all require some level of computer modeling of the system and system components, and the associated amount of engineering time that most plant designers do not have. In writing this series of articles and the SDL upon which it is based, significant modeling was performed in an effort to determine generalized control sequences that account for most of the variation in plant design parameters summarized above. The technique used to determine optimized performance is described in a June 2007 ASHRAE Journal article.[4]

In brief, the technique involves developing calibrated simulation models of the plant and plant equipment that are run against an annual hourly chilled water load profile with coincident weather data while parametrically modeling virtually all of the potential modes of operation at each hour. The operating mode requiring the least amount of energy for each hour is determined. The minimum hourly energy use summed for the year is called the theoretical optimum plant performance (TOPP). Since all modes of operation were simulated, the plant performance cannot be better than the TOPP within the accuracy of the component models.

TOPP modeling was performed for the chilled water plant shown in *Figure 1* for a wide range of plant design options for tower range, approach, and efficiency; different chiller types and chiller efficiency; and varying climates (see "Modeling the Plant," Page 57). The operating modes (e.g., number of chillers, condenser water flow and pump speed, tower fan speed and related condenser water temperatures) that result in the TOPP for each plant design scenario were studied to see how they relate to independent variables such as plant load and weather (e.g., wet-bulb temperature) to find trends that can be used to control the plant in real applications through the direct digital control (DDC) system.

Ideally, equipment should be controlled as simply as possible; complex sequences are less likely to be sustained since operators are more likely to disable them at the first sign of perceived improper operation. The remainder of this article discusses the TOPP modeling and the generalized sequences that were developed from the analysis for the chilled water plant shown in *Figure 1* serving a typical office building.

CHW Pump Control

Chilled water pump speed is typically controlled to maintain supply-to-return differential pressure (DP) at setpoint. Standard 90.1[5] requires that the DP sensor(s) be located at the most remote coil(s). This is because the lower the DP setpoint, the lower the pump energy, as shown in *Figure 2*.* If the DP setpoint is reset by valve position, as discussed further below, pump energy can be close to the ideal curve in *Figure 2* for "DP setpoint = 0."

Figure 3 shows the optimum number of CHW pumps as a function of CHW flow ratio and as a function of pump speed for the chilled water plant shown in *Figure 1* based on TOPP modeling. Unlike cooling towers, the optimum sequence is not to run as many pumps as possible. This is because the pumps all pump through the same circuit (other than the pipes into and out of the each pump between headers) so there are not "cube-law" energy benefits for each pump individually.

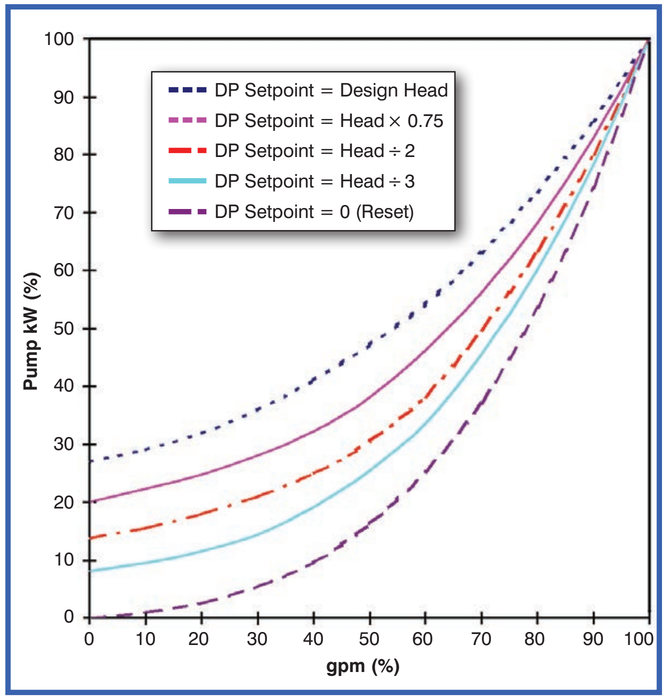

Figure 2: Variable speed performance at varying DP setpoint.

Figure 3 clearly indicates that staging pumps off of flow provides better optimization than staging off of pump speed.

As suggested by *Figure 3*, CHW pumps should be staged as a function of CHW flow ratio (CHWFR = actual flow divided by total plant design flow) at a staging point of 47%, i.e., one pump should operate when the CHWFR is below 47% and two pumps should operate when CHWFR is above 47%, with a time delay to prevent short cycling. The 47% optimum staging point assumes DP setpoint is reset by valve position; it will be somewhat higher at higher DP setpoints.

For very large pumps (>~100 hp [75 kW]), it may be worth the effort to determine the actual pump operating point (flow vs. head) and optimize staging based on pump efficiency determined by flow and pressure drop readings mapped to pump curves duplicated mathematically in the DDC system.[6] This can allow pumps to operate closer to their design efficiency as the system operating curve varies from the ideal parabolic curve due to modulating valves and minimum differential pressure setpoint. But the small potential energy savings is not worth the effort for most chilled water plants.

Chilled Water Temperature and DP Setpoint Reset

Chillers are more efficient at higher leaving water temperatures so, in general, optimum efficiency is achieved when the chilled water supply temperature (CHWST) setpoint is as high as possible. (The impact of CHWST on CHW pump energy is discussed below.) Where all zones are controlled by the DDC system, the best reset strategy is based on valve

* The curves in this figure assume pressure drop varies with flow to the 1.8 power since flow is typically in the transitional region between turbulent and laminar flow. They do not account for the impact of opening and closing control valves, which change system geometry and hence the system flow characteristics. The curves do include reductions to the efficiency of motors and VSDs at low load.

position where the CHWST setpoint is reset upwards until the valve controlling the coil that requires the coldest water is wide open. This strategy ensures that no coil is starved; all are able to maintain their desired supply air or space temperature setpoints.†

Valve position can also be used to reset the DP setpoint used to control pump speed. In fact, this is required by Standard 90.1. The logic is similar to CHWST setpoint reset: the DP setpoint is reset upwards until the valve controlling the coil that requires the highest DP is wide open.

So we have a dilemma: Valve position can be used to reset either CHWST setpoint or DP setpoint, but not both independently; it is not possible to know if the valve is starved for lack of pressure or from lack of cold enough water. We must decide which of the two setpoints to favor.

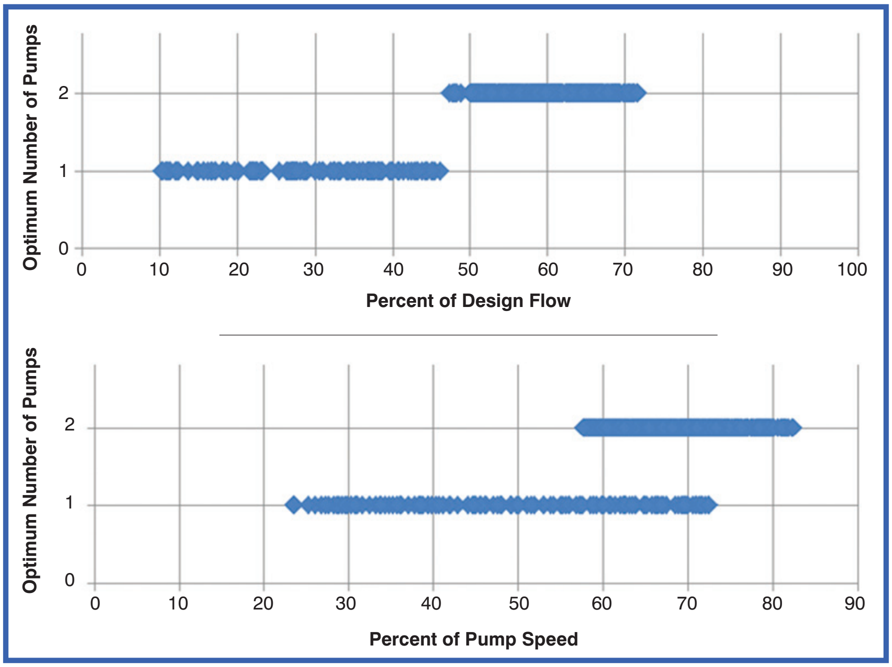

Figure 3: TOPP CHW pump staging vs. CHW flow ratio and pump speed.

While resetting CHWST setpoint upward reduces chiller energy use, it will increase pump energy use in variable flow systems. Higher chilled water temperature will cause coils to require more chilled water for the same load, degrading CHW ΔT and increasing flow and pump energy requirements. Degrading ΔT can also affect optimum chiller staging; however, this is not generally an issue in primary-only plants with variable speed chillers.[7] Furthermore, our simulations have shown that the positive impact of resetting chilled water temperature on chiller efficiency is much greater than the negative impact on pump energy even when distribution losses are high.

Figure 4 shows a DOE2.2 simulation of a primary-only plant with variable speed chillers and CHW pumps with high pump head (150 ft [450 kPa]) using three reset strategies based on valve position: reset of chilled water temperature alone; reset of differential pressure setpoint alone; and a combination of the two that first resets chilled water temperature then resets DP setpoint. The simulations were done in several climate zones (Houston and Oakland results are shown in the figure) and in all cases, resetting chilled water temperature was a more efficient strategy than resetting DP setpoint. Sequencing the two (resetting chilled water temperature first then DP setpoint) was the best approach, although only slightly better than resetting chilled water temperature alone.

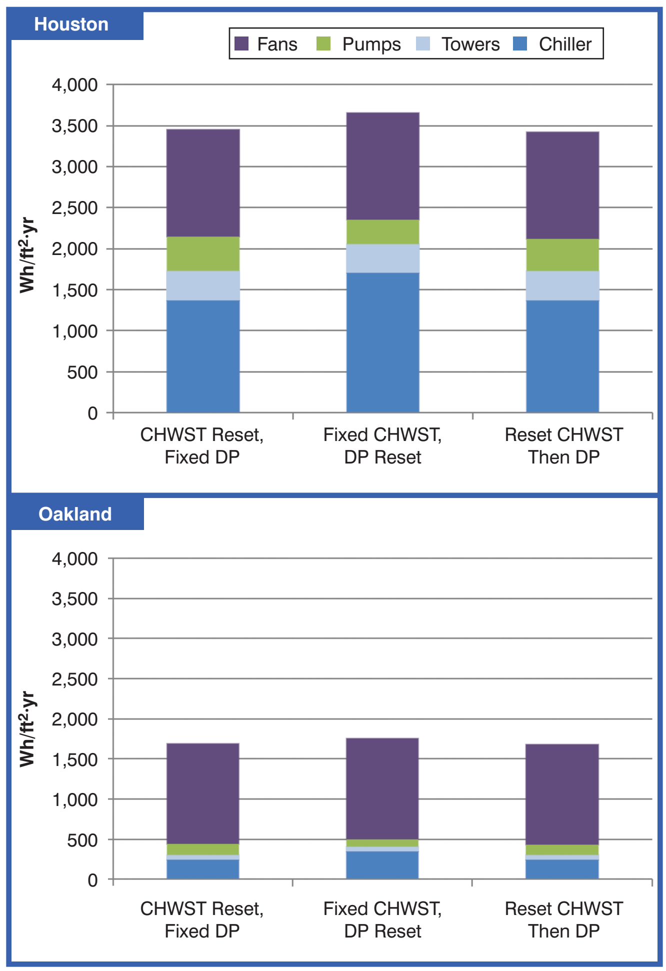

Figure 4: Plant energy with CHWST Setpoint Reset, CW DP Setpoint Reset and a combination of the two.

† Contrary to conventional wisdom, the impact of reset on the dehumidification capability of air handlers is quite small and should not be a concern. Space humidity is a function of the supply air humidity ratio, which in turn, is a function of the coil leaving dry-bulb temperature setpoint. Regardless of CHWST, the air leaving a wet cooling coil is nearly saturated; lowering CHWST only slightly reduces supply air humidity ratio. As long as the supply air temperature can be maintained at the desired setpoint, resetting CHWST will not impact space humidity.

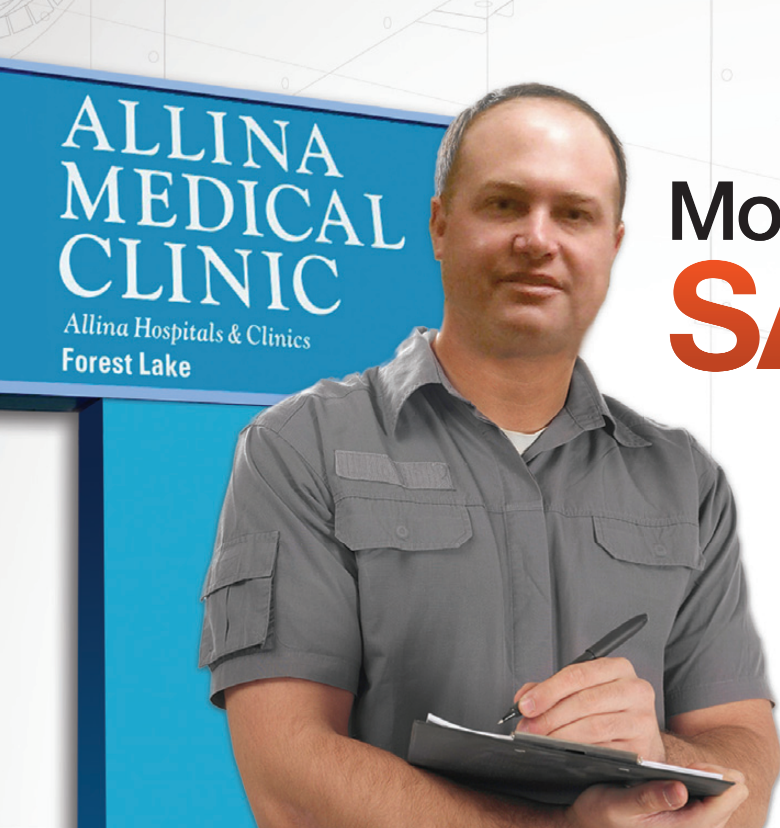

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Figure 5 shows how this sequenced reset strategy can be implemented. The x-axis is a software point called "CHW Plant Reset" that varies from 0% to 100% using "trim and respond" logic.[8] The coil valve controllers generate "requests" for colder chilled water temperature or higher pump pressure when the valve is full open. When valves are generating "requests," CHW Plant Reset increases; when they are not, CHW Plant Reset steadily decreases.

When CHW Plant Reset is 100%, the CHWST setpoint is at T_{min} (the design chilled water temperature) and the DP setpoint is at DP_{max} (the design DP setpoint). As the load backs off, the trim and respond logic reduces the CHW Plant Reset point. As it does, chilled water temperature is increased first up to a maximum T_{max} (equal to the lowest air handler supply air temperature setpoint less 2°F [1°C]), then DP setpoint is reduced down to a minimum value DP_{min} (such as 3 psi [21 kPa]).

In practice, this logic seldom results in much reset of the DP setpoint—the CHWST reset is aggressive enough to starve the coils first—so it is important to locate the DP sensor(s) at the most remote coil(s) so that DP_{max} can be as low as possible to minimize pump energy (*Figure 2*).

‡ For plants with more consistent loads that do not vary with weather, such as those serving data centers and those located in consistently humid climates such as Miami, correlation of load with CWRT/CHWST temperature difference is poor. For these plants, optimum CWST vs. wet-bulb temperature was found to have better correlation. But for office buildings in general, the correlations in *Figure 7* were more consistent.

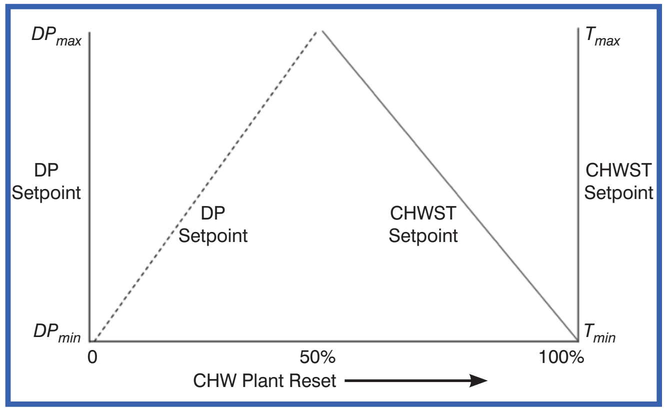

Figure 5: CHWST Setpoint and CW DP Setpoint Reset sequenced off of CHW valves.

Tower Fan Speed Control

A common approach to controlling cooling towers is to reset condenser water supply temperature based on outdoor air wet-bulb temperature. But our simulations seldom indicated a good fit; as shown in *Figure 6*, the correlation was fairly good in Miami but not in Oakland and most other climates.

For plants serving typical office buildings,‡ good correlations were found in all TOPP simulations between plant part load ratio (PLR, actual plant load divided by total plant design

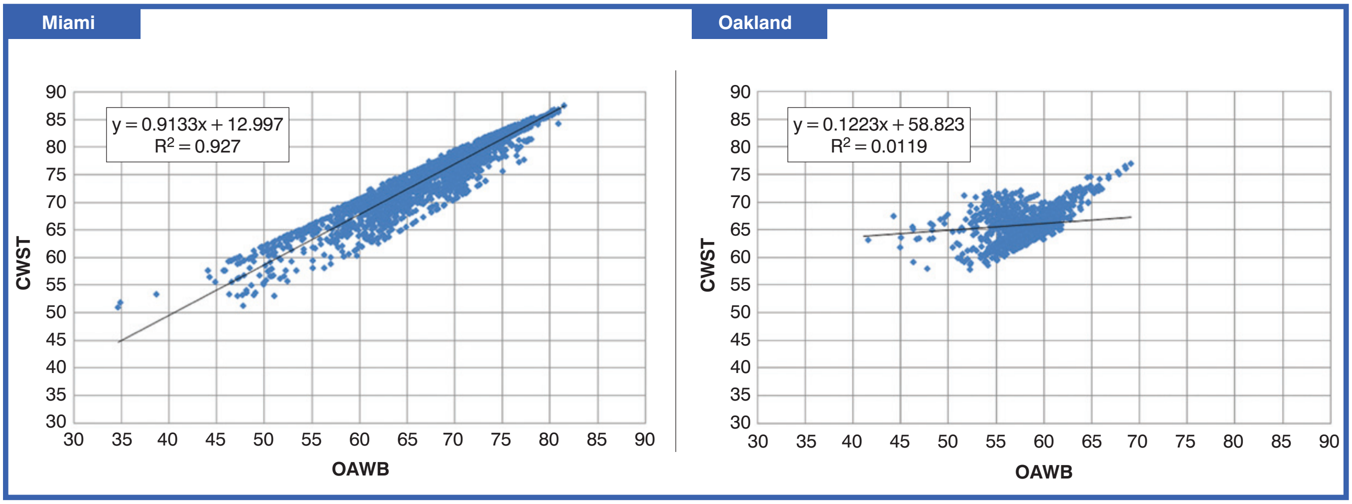

Figure 6: TOPP optimum condenser water supply temperature vs. wet-bulb temperature.

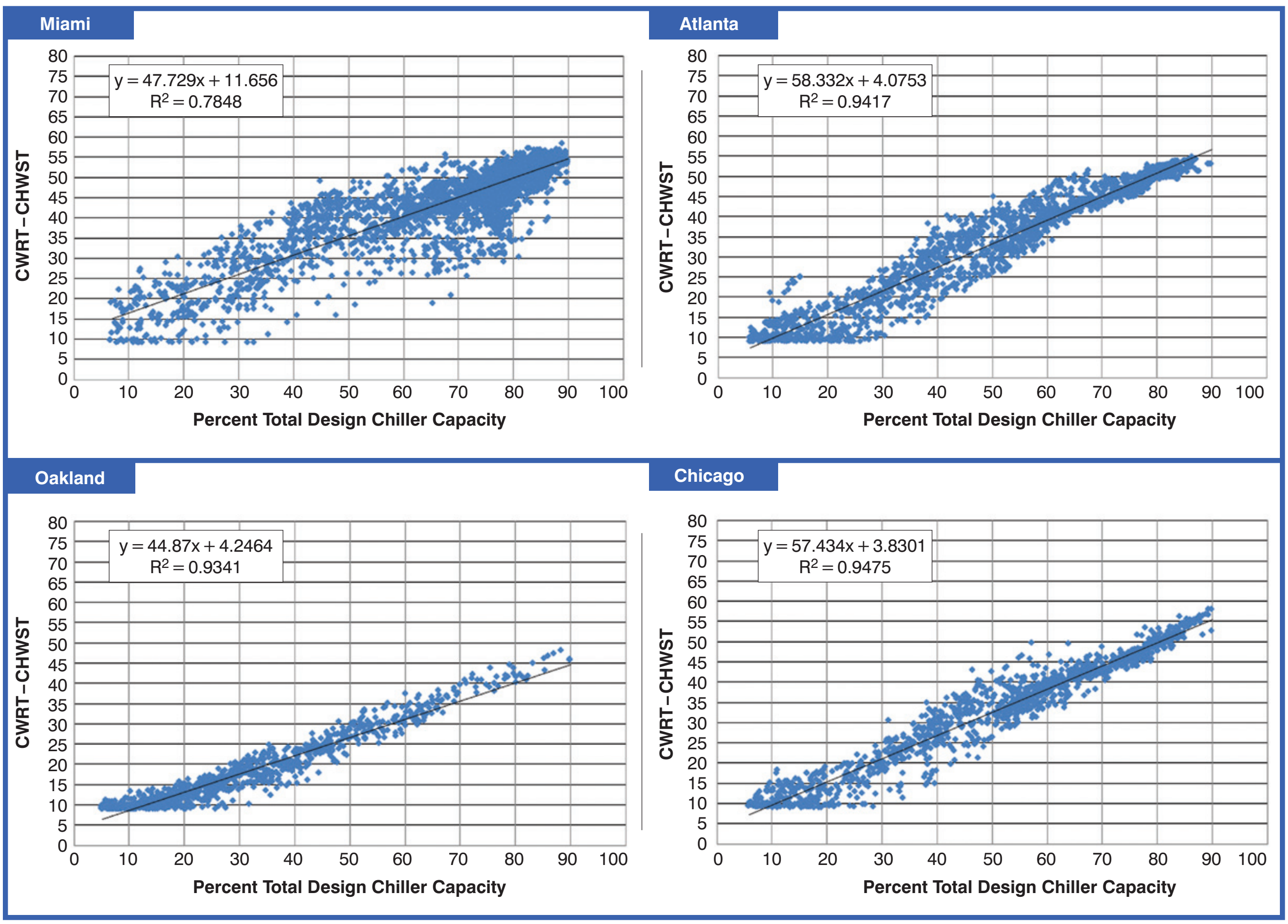

Figure 7: TOPP [CWRT-CHWST] vs. plant load ratio.

capacity) and the difference between the condenser temperature return temperature (CWRT, leaving the condenser) and the CHWST. Examples are shown in *Figure 7*. The CWRT-CHWST difference is a direct indicator of the refrigerant lift (the condenser and evaporator leaving water temperatures are determined by the condenser and evaporator temperatures), which drives chiller efficiency.

The data in *Figure 7* can be fit to a straight line:

$$\text{CWRT} - \text{CHWST} = \text{A} \times \text{PLR} + \text{B} \qquad (1)$$

A and B are coefficients that vary with climate and plant design (see "Modeling the Plant," Page 57). Equation 1 can be solved for the optimum CWRT setpoint given the current CHWST:

$$\text{CWRT} = \text{CHWST} + \text{A} \times \text{PLR} + \text{B} \qquad (1a)$$

This setpoint must be bounded by the minimum CWRT-CHWST difference at low load prescribed by the chiller manufacturer. This minimum (9°F [5°C] for the chiller in *Figure 7*) is a function primarily of the chiller's oil management design and can range from only a few degrees for oil-free chillers (e.g., those with magnetic or ceramic bearings) to as high as about 20°F [11°C]. The lower this minimum is, the lower annual chiller plant energy will be, particularly in mild climates.

So near-optimum tower performance can be achieved by controlling tower fan speed based on condenser water *return* temperature to the setpoint determined from Equation 1a. Controlling tower fan speed based on return temperature rather than supply temperature is non-conventional but it makes sense because it is the temperature leaving the condenser that determines chiller lift, not the entering (supply) water temperature. Chiller efficiency is not sensitive to entering chilled or condenser water temperature.

Condenser Water Pump Control

No good correlations were found for control of VSDs on condenser water pumps. Optimum condenser water pump speed and flow were plotted against various parameters such as PLR, wet-bulb temperature, chiller percent power, and lift with no consistent relationships. The best correlation was flow vs. PLR as shown in *Figure 8*, but the correlations were seldom strong (R^2 typically less than 0.85 and some as low as 0.5). The correlations were significantly weaker for pump speed than for flow so a condenser water flow meter should be added if one is not already part of the design.

The curve fit can be expressed as follows

$$\text{CWFR} = \text{C} \times \text{PLR} + \text{D} \qquad (2)$$

where CWFR is the ratio of desired CW flow setpoint to the design CW flow. The CW flow setpoint is then calculated as:

$$\text{CWFSP} = \text{CWFR} \times \text{CWDF} \qquad (2a)$$

where CWDF is the design CW flow rate for the plant (both pumps). This setpoint must be bounded by the minimum required CW flow rate obtained from the chiller manufacturer. The minimum flow from most manufacturers cor-


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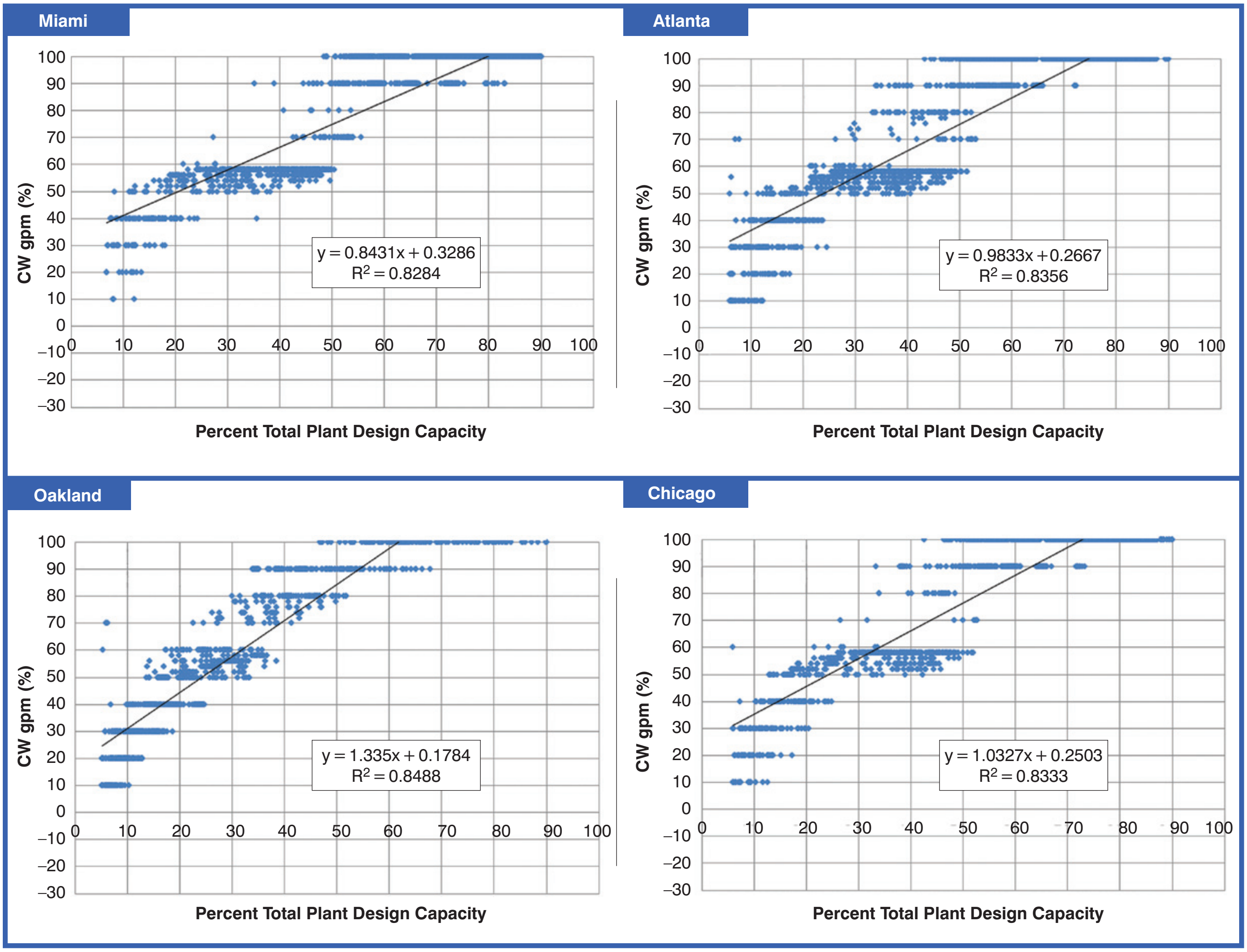

Figure 8: TOPP CW flow vs. plant load ratio.

relates to the onset of laminar flow and will be on the order of 40% to 70% of design flow depending on the number of tubes, number of passes, and tube design (e.g., smooth vs. enhanced). Higher rates are reputed to discourage fouling of condenser tubes but to the author's knowledge, no studies have been done to support that notion.[9] Once the flow rate is determined, CW pump speed is modulated to maintain the CW flow at setpoint.

When C and D coefficients determined for specific plants were fed back into the energy model, actual performance ranged from 101% to 110% of the TOPP. With this less than optimum performance, VSDs were found to be marginally life-cycle cost effective in dry climates (Albuquerque, N.M.) and not cost effective elsewhere. This performance gets worse when C and D are determined from the regression equations based on plant design (see "Modeling the Plant"), rather than from actual plant performance modeling (e.g., *Figure 8*). In some cases, particularly in humid climates, the CW pump control logic caused energy use to *increase* vs. constant speed CW pumps. Therefore, VSDs on CW pumps are recommended only on plants in dry climates and only if

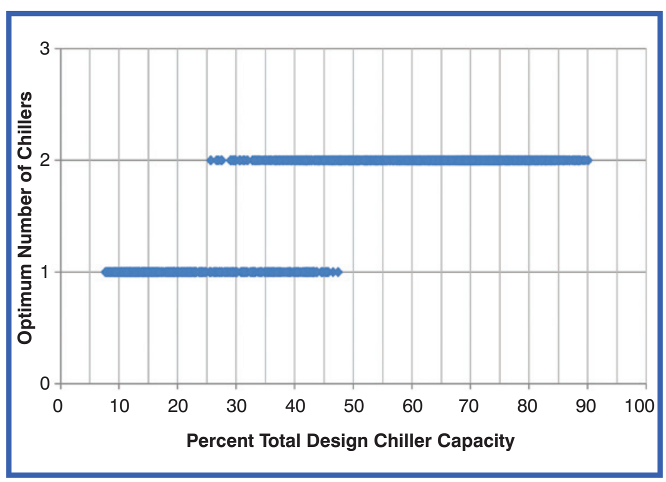

Figure 9: TOPP variable speed chiller staging vs. plant load ratio (Albuquerque).

C and D coefficients are based on TOPP simulations of the actual plant, not from the equations list in "Modeling the Plant."

Optimum staging for variable speed CW pumps was found to correlate very well to CW flow with 60% of the total design flow as the staging point, i.e., one pump should operate when the CWFR is below 60% and two pumps should operate when CWFR is above 60%, with a time delay to prevent short cycling.

Optimum staging for constant speed pumps was found to vary with both CWRT-CHWST difference and with PLR, but with fairly weak correlations and relatively small energy impact regardless of logic. For simplicity, constant speed CW pumps should simply be staged with the chillers.

Chiller Staging

Figure 9 (Page 68) shows the optimum number of chillers that should be run plotted against plant load for variable speed centrifugal chillers. The graph shows that it is often optimum to operate two chillers as low as 25% of overall plant load. This result may seem somewhat counterintuitive; conventional wisdom is to run as few chillers as possible. That is true for fixed speed chillers, but not for variable speed chillers, which are more efficient at low loads when condenser water temperatures are low.

Figure 9 shows that staging chillers based on load alone will not optimize performance since there is a fairly wide range where either one or two chillers should be operated.

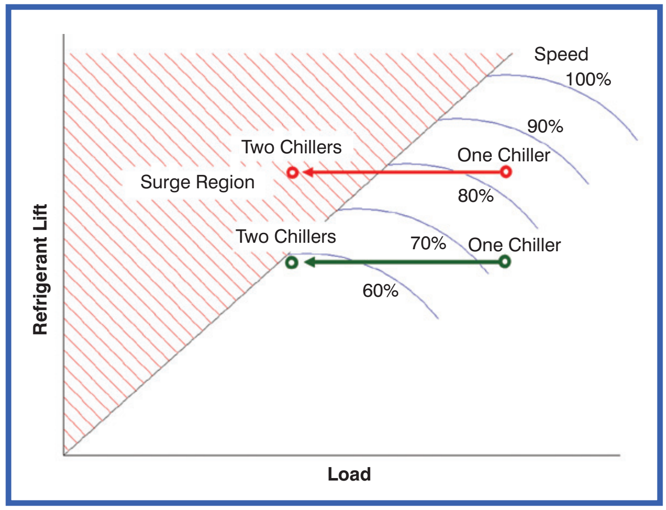

Figure 10: Possible surge problem staging by load only.

There is also another problem with staging based on load alone: it can cause the chillers to operate in surge. This can be seen in *Figure 10*, which schematically shows centrifugal chiller load vs. lift, defined as the difference between condenser and evaporator refrigerant temperature. If two chillers are operated when the refrigerant lift is high (red line), the chillers will operate in the surge region. To avoid surge, the chiller controllers will speed up the compressors and throttle inlet guide vanes to control capacity. This reduces chiller efficiency so that it would then be more efficient to operate one chiller rather than two. But if the lift is low (green line in *Figure 10*), the chillers would not be in surge so operating two chillers would be more efficient than operating one. So in addition to load, chiller staging must take chiller lift into account. (This consideration applies only to centrifugal chillers; surge does not occur with positive displacement chillers such as those with screw compressors.)

Figure 11 shows the optimum number of operating chillers (blue dots indicate one chiller while red dots indicate two chillers) for example TOPP simulations. For all plant design options and for all climate zones, good correlations were found for the optimum staging point described by a straight line:

$$SPLR = E \times (CWRT - CHWST) + F \qquad (3)$$

where SPLR is the staging PLR and E and F are coefficients that vary with climate and plant design (see "Modeling the Plant"). If the actual measured PLR is less than SPLR, one chiller should operate; if the PLR is larger than SPLR then two chillers should operate, with a time delay to prevent short cycling.

Note that the number of operating chilled water pumps and the number of operating chillers may not match. The pumps must respond to the flow and pressure requirements of the system, not to the load, and thus are staged independently from chillers.

Primary-only variable flow plants like this also will require "soft staging" and minimum flow control. These sequences and why they are needed are discussed in more detail in the SDL and in Reference 10.

Example

The TOPP model results for an Oakland plant were plotted per *Figures 7, 8,* and *11* and the following slopes and intercepts were determined from curve-fits:

A = 47, B = 5.2
C = 1.3, D = 0.13
E = 0.009, F = 0.21

Figure 12 shows the theoretical optimum performance for both variable speed (VS) constant speed (CS) CW pumps compared to our proposed "real" sequences using the coefficients listed above. Despite their simplicity, our sequences resulted in only about 1% higher energy use than the TOPP. Variable speed drives on the CW pumps saved 3% vs. constant speed pumps, but this was not enough savings to make them cost effective at a 15 scalar ratio (simple payback period) for this plant. Also shown in the figure for comparison is plant performance using the AHRI 550/590 condenser water relief curve to reset condenser water temperature (4% higher energy use than our sequences) and performance assuming CWST

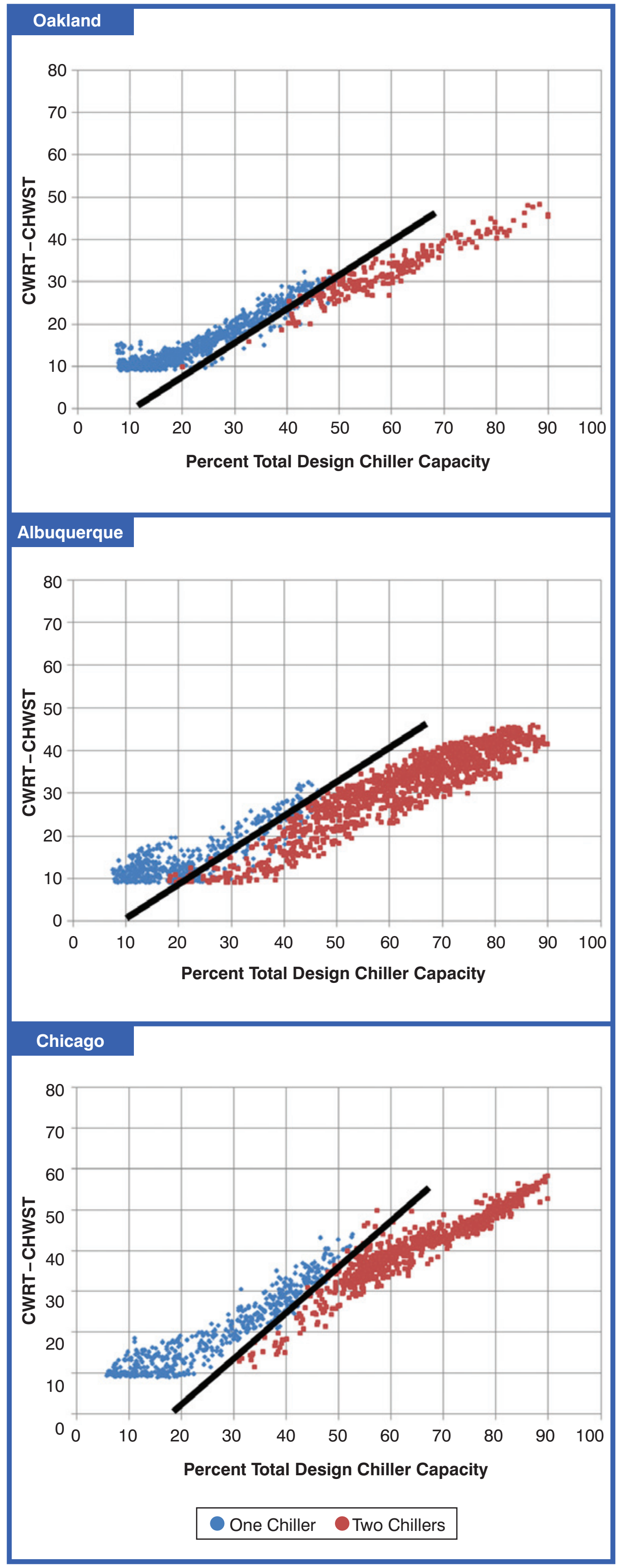

Figure 11: Optimum staging vs. (CWRT-CHWST) and plant part load ratio.

setpoint is fixed at the design temperature (16% higher than our sequences).

Summary

This article is the last in a series of five that summarize chilled water plant design techniques intended to help engineers optimize plant design and control with little or no added engineering effort. In this article, optimized control logic was addressed. The logic is very simple and easily programmed into any DDC system controlling the plant. With these sequences properly implemented, chiller plants can perform within a few percent of their theoretical optimum.

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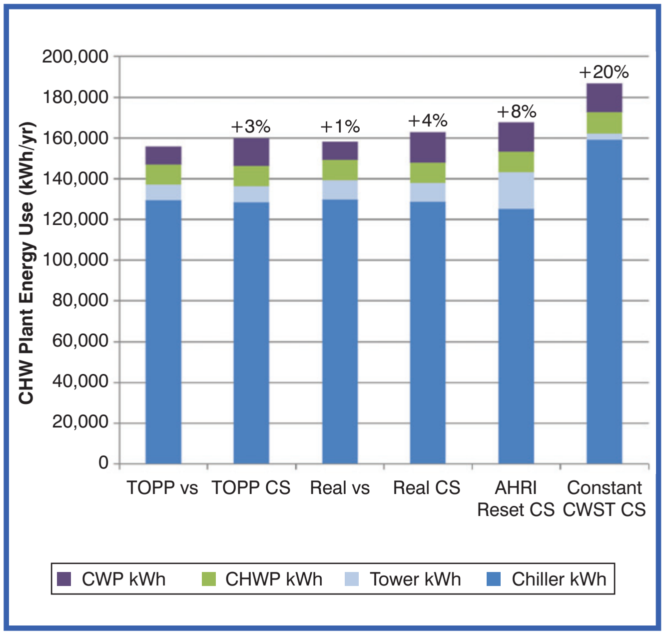

Figure 12: TOPP vs. real sequences for both constant speed and variable speed CW pumps.


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(And that’s a good thing.)
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Central Chilled Water Plants Series

This series of articles summarizes the upcoming Self Directed Learning (SDL) course Fundamentals of Design and Control of Central Chilled Water Plants and the research that was performed to support its development. The series includes five segments.

Part 1: "Chilled Water Distribution System Selection" (July 2011), Part 2: "Condenser Water System Design" (September 2011), Part 3: "Pipe Sizing and Optimizing ΔT" (December 2012), and Part 4: "Chiller & Cooling Tower Selection" (March 2012).

Optimized control sequences. The series concluded with a discussion of how to optimally control chilled water plants, focusing on all-variable speed plants.

The intent of the SDL (and these articles) is to provide simple yet accurate advice to help designers and operators of chilled water plants to optimize life-cycle costs without having to perform rigorous and expensive life-cycle cost analyses for every plant.

In preparing the SDL, a significant amount of simulation, cost estimating, and life-cycle cost analysis was performed on the most common water-cooled plant configurations to determine how best to design and control them. The result is a set of improved design parameters and techniques that will provide much higher performing chilled water plants than common rules-of-thumb and standard practice.

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