Optimizing Chilled Water Plant Control

By Mark Hydeman, P.E., Fellow ASHRAE; **and Guo Zhou,** Associate Member ASHRAE

In typical commercial buildings, water-cooled chilled water plants use a significant amount of energy. They account for between 10% and 20% of the overall facilities usage and serve roughly one third of the commercial floor space.[1] From literature and experience the annual average "wire-to-water" efficiency* of these plants ranges from as low as 0.3 kW/ton (12 COP) up to 1.2 kW/ton (2.9 COP) or more. This efficiency is strongly influenced by three factors: the equipment efficiency (chillers, towers and pumps); the system configuration (e.g., the pumping scheme and types of control valves); and control sequences (e.g., equipment staging and setpoints). This article presents a parametric analysis technique to optimize the control sequences of chilled water plants that has been successfully applied to dozens of projects. This optimization technique is based on peer reviewed research and validated through measurement and verification techniques.[2,3] Unfortunately, the techniques used are not presently available in standard off-the-shelf simulation tools.

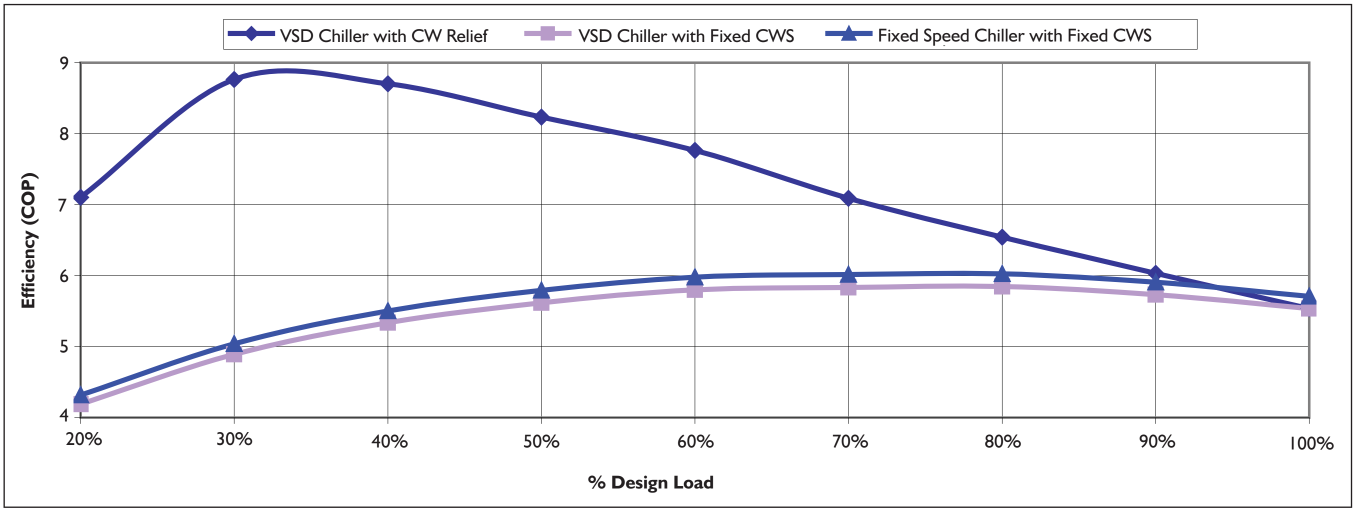

Figure 1: Part-load chiller efficiency of a water-cooled chiller with variable speed drive as a function of operating conditions.

The Quest for Efficiency

How can I make my chilled water plant efficient? This question is surprisingly complex to evaluate. We can measure a plant's efficiency and through experience or comparison with other plants determine when a plant is efficient, but we cannot easily determine the rules that make the plant efficient. To illustrate this complexity, consider the narrower and often asked question, "if I add a variable speed drive (VSD) to a chiller, will it make my plant more efficient?" The answer to this question can be found in *Figure 1,* which presents "zero-tolerance"[†] performance data of a water-cooled centrifugal chiller with a VSD.

In *Figure 1*, one line depicts the chiller operating with condenser water relief per ARI Standard 550/590[4] and other (less efficient) lines are with the chiller operating at a fixed condenser water supply (CWS) temperature. The second line, "VSD Chiller with Fixed CWS Temperature," represents the chiller modulating primarily on its inlet vanes with the variable speed drive at or near 100%. The internal controls do this to prevent the chiller from operating in a surge condition. The third line, "Fixed-speed Chiller with Fixed CWS Temperature," is extrapolated performance assuming a constant 3% power loss for the variable speed drive. By looking at these three curves, we can clearly see that the answer to the question, "will the VSD make the chiller and plant more efficient," is, "it depends." If the chiller operates on the line "VSD Chiller with CW Relief," it will likely be more efficient than the same chiller without the variable speed drive. However, if the facility has a limitation on the condenser water system that prevents the plant from operating with condenser water relief (such as tenant water-cooled AC units that don't have head pressure control), the VSD will decrease the plant efficiency and increase the first cost.

The situation becomes more complex as you add other variables such as:

- The number, size, and efficiency of the cooling towers;
- The number, size, and efficiency of the chillers;
- The type of pumping system;
- The unloading mechanisms for the chillers, towers and pumps; and
- The logic used to control these systems and equipment.

Breaking the Problem Down

Before we jump into the process of evaluating chilled water plants, a few guiding principles for plant efficiency are generally (but not always) true. By addressing these items first, we can reduce the number of parametric runs that we have to evaluate through simulation. This list is by no means comprehensive; it includes only those items that are relevant to the case study:

- It is always more efficient to run as many tower cells as possible as long as you stay within the flow limits of the towers.[5]
- Propeller fan towers are always more efficient (by approximately a factor of two) than centrifugal fan towers designed for the same duty.[5]
- Variable speed drives are cost effective on all cooling tower fans and variable flow pumps.[5] They should be considered on chillers as well, but are not always cost effective.

These principles have been established in the literature (e.g., the CoolTools™ Design Guide[2] and several papers cited in this article). Although they do reduce the number of design and control parameters that we have to evaluate, they are not sufficient to fully optimize the equipment selection and control strategies of a complex chilled water plant. We must rely on simulation for the rest. For plants with constant condenser water

*This term represents the total plant efficiency in units of input power to plant output capacity including all pumps, towers and chillers (see Guideline Project Committee 22P, Instrumentation for Monitoring Central Chilled Water Plant Efficiency).

†"Zero tolerance" data is manufacturer performance data supplied, often supplied at the time of the chiller bids, which is guaranteed by the manufacturer to meet or exceed the submitted capacities and efficiencies. These data can be subsequently verified with certified factory witness tests and enforced through a liquidated damage clause.[2]

About the Authors

Mark Hydeman, P.E., Fellow ASHRAE, is a principal at Taylor Engineering in Alameda, Calif. He is the incoming vice chair of Standing Standard Project Committee 90.1 and member of ASHRAE Technical Committee 9.9, Mission Critical Facilities. **Guo Zhou** is a mechanical designer at Taylor Engineering.

Advertisement formerly in this space.

flow we typically use a commercial simulation program (e.g., DOE-2) using parametric runs as described in this paper. For plants with variable condenser water flow, we use Visual Basic code resident in databases or spreadsheets that can use advanced chiller models and parametric simulation techniques.

There are two advanced chiller models are in the open literature that we have found, which respond well to variable condenser water flow: the modified DOE-2 model[6] and the modified Gordon and Ng model.[7] Although both do a fine job of predicting energy usage, the advantage of the modified DOE-2 model is that it explicitly models the limit of the chiller's capacity at all operating conditions. This is required for evaluation of the staging logic.

We use a parametric model to run the plant through the full range of possible plant control modes saving the hourly results of each run in a database. We then seek out the combination of control sequences that provide either the lowest energy or energy cost operation. To illustrate this process we'll describe the results from a recent project. In this case study, we used our model twice in the design process: first to select the chillers in a competitive bid (see Reference 2); and again to determine the optimum control sequences after the chillers were selected.

The process used to determine optimum control sequences consists of the following steps and is described in the following paragraphs:

1. Collect a baseline load profile including chilled water flow, chilled water supply temperature, chilled water return temperature and outside air wet-bulb temperature.
2. Calibrate the models of the chilled water plant equipment.
3. Run the model across the full range of acceptable operating conditions.
4. Calculate the sum of the lowest energy cost (or energy usage) for each hour. We call this the theoretical optimum plant performance (TOPP) value.
5. Analyze the results to determine realistic control sequences that approach the TOPP value.

Step 1: Collect or create a baseline load profile. In this example we used a commercially available simulation tool to model the facility loads. In other projects with existing buildings, we have used historic measured data. In either case, it is important to separate the temperature flow and load data so that you can explicitly model the pumping energy and the chiller performance. Simultaneous wet-bulb temperatures also are required for analysis of the cooling towers (or dry-bulb temperatures for air-cooled plants).

Step 2: Calibrate the models of the chilled water plant equipment. The calibration of chiller and cooling tower component models is covered in References 8 and 9. As addressed in Reference 2, it is important to specify zero tolerance data for an accurate model of the chilled water plant. In this example project, we received a full set of zero tolerance data from the chiller manufacturers during the performance-based bid process. The cooling tower models were calibrated to the CTI-certified data on the submitted towers, and the pumps were calibrated to the submitted pump performance data.

Step 3: Run the model across the full range of acceptable operating conditions. For this project, we ran the hourly loads with each of the following parameters:

- The number of chillers (one to three) with the load equally divided among the operating chillers.
- The speed of the condenser water pumps from 100% down to 50% in 5% increments. The lowest speed was based on the minimum flow recommended by the chiller manufacturer plus a small margin of safety.
- The speed of the cooling tower fans from 0% to 100% fan speed in 5% increments.

The number of cooling tower cells is directly calculated from the current condenser water flow (a function of the number of chillers and the speed of the CW pumps) and the maximum number of cells that can run at that flow from the tower manufacturer's submitted flow limits.

To simplify the model, we fixed the number of chilled water and condenser water pumps to match the number of operating chillers at each hour. We also modeled the pumps and motors as having a fixed efficiency. The fixed efficiency model is generally appropriate for chilled water pumps that are reset by demand from the coils. For chilled water pumps that run on a fixed system pressure setpoint and for condenser water pumps where there is a fixed static head in the system (the lifting of water to the cooling towers), this simplifying assumption is less accurate as the pump efficiency changes at lower flows.

A lot of buzz in the industry is over variable flow condenser water pumps on chilled water plants (e.g., Reference 10). Our analysis for this and other plants has shown a large potential energy savings exists, but you must carefully evaluate the energy used by the chillers, condenser water pumps and cooling tower fans at each stage of operation. If improperly controlled, reduced condenser water flow can increase plant energy use or cause the chillers to trip on high head.

With these three parameters, 673 parametric runs (three chillers, 11 CW pump speeds and 21 tower fan speeds) are generated. Since each of these runs is evaluated at 2,871 hours of unique load and weather conditions, we have around 2 million plant conditions to review.

For each hour simulated, we discarded those results where one or more components cannot satisfy the current load under the present parametric conditions or where operational limits (like minimum flows through cooling towers or chillers) were exceeded. Both are issues for the chillers and cooling towers. This process reduces the number of data records to manage. In this project it reduced the count from approximately 2 million down to 1.2 million records (a 38% reduction).

The chiller available capacity varies with leaving chilled and

Advertisement formerly in this space.

condenser water temperatures (which in turn vary the chiller lift). So the available chiller capacity is a function of the number of cooling tower cells, the cooling tower fan speed and the condenser water pump speed. For each parametric option at each hour of load, you must explicitly calculate and check the available chiller capacity against the load.

Due to the interactive effects, these calculations require some iteration. For a given hour, you must guess a leaving condenser water return temperature, which allows you to calculate the chiller efficiency and available capacity. This impacts the tower capacity. Through iteration between reasonable boundaries, the models converge.

Step 4: Calculate the TOPP value. Once the simulations are run and the results are stored in a database, it is a simple task to take either the minimum hourly energy usage (source energy for dual-fuel plants) or hourly energy costs and sum them. Energy usage is selected if the goal is to minimize carbon production or other environmental metrics; hourly energy costs are used if the target is to minimize operating costs. In either case, this TOPP value becomes a target for the proposed control scheme.

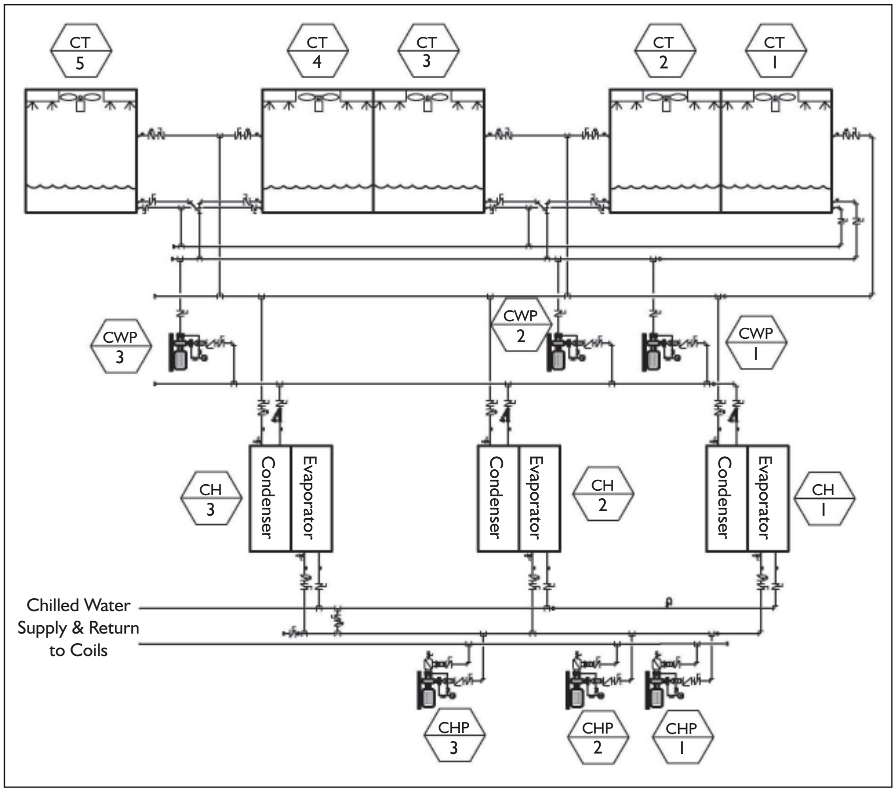

Figure 2: Piping schematic of the example plant.

Step 5: Analyze the results to determine realistic control sequences that approach the TOPP value. Upon achieving a theoretical optimum control strategy, we now have to find a practical control strategy that performs close to the optimum. We do this by analyzing patterns uncovered in the TOPP model that was developed in the previous step. We compare the energy usage (or cost) of each control strategy to the TOPP value to see how close we are. The goal is to find realistic control sequences that result in energy usage or cost that is within ~5% of the TOPP value.

This step is necessary since the TOPP value may be based on unrealistic staging of equipment due to minor changes in either load, flow or wet-bulb temperature. As demonstrated in our later example, we can reduce the cycling of equipment at a small penalty in energy usage or cost. This step is iterative and requires both intuition and graphical techniques. We look at the optimum controls as a function of different variables until a practical near-optimum strategy emerges.

The Example Plant

The piping schematic of the example plant is depicted in *Figure 2*. This plant serves a large office building in Southern California. It features three equally sized water-cooled centrifugal chillers with variable speed drives (nominal 775 tons [2720 kW] each). As previously mentioned, these chillers were selected using a performance-based bid following the process that is outlined in Reference 2. The plant has five draw-through cooling tower cells that are equally sized (designed for approximately 1,200 gpm [76 L/s] each cell). Each cell has a propeller fan with variable speed drive. The cells were designed to have a large flow turndown ratio. There are three equally sized condenser water pumps with variable speed drives (designed for 1,760 gpm [111 L/s] with 25 hp [19 kW] motors) that are piped together in a headered arrangement so that each pump can serve any of the chillers or tower cells. Similarly, three equally sized chilled water pumps with variable speed drives (designed for 900 gpm [57 L/s] with 40 hp [30 kW] motors) are also headered and configured for primary-only variable flow distribution.

A histogram of the plant load is depicted in *Figure 3*. As is evident in this figure, the loads ranged between 120 (420 kW) and 2,000 tons (7000 kW).

Modeling Results

Table 1 presents the base case TOPP model run and four alternative sequences of controls. Each of the alternatives is compared in annual energy usage to the results for the TOPP model. The TOPP value for this plant (Run 0) was calculated at 1.21 GWh/yr. The load was 2.14 × 106 ton·hrs/yr (7.52 GWh/yr) yielding an average annual wire-to-water efficiency of 0.57 kW/ton (6.21 COP). The proposed control strategy (Run 1) is projected to use 1.21 GWh/yr, only 0.39% more than the TOPP value. The proposed control strategy is described in the following paragraphs.

Equipment Staging. *Figure 4* presents the TOPP model re-

	Annual Energy Usage (kWh/yr, % of TOPP)									
	Chillers		Cooling Towers		CW Pumps		CHW Pumps		Total	
Run 0	980,000		40,900		76,000		113,000		1,210,000	
Run 1	980,000	–0.16%	38,800	–5.14%	84,000	10.93%	113,000	0%	1,210,000	0.39%
Run 2	1,010,000	3.85%	38,800	–5.13%	63,000	–17.33%	113,000	0%	1,230,000	1.85%
Run 3	990,000	0.81%	58,000	41.88%	175,000	130.90%	113,000	0%	1,330,000	10.31%
Run 4	1,140,000	16.42%	4,600	–88.71%	335,000	340.74%	113,000	0%	1,590,000	31.73%

Run Descriptions:

Run 0: TOPP model; Run 1: Recommended control sequences; Run 2: Run 1 with the cooling tower CWS temperature controlled by wet-bulb reset; Run 3: "Standard control sequence" with ARI 550/590 CW reset; Run 4: "Standard control sequence" with CW temperature fixed at design.

Table 1: Run results for base case TOPP model run and four alternative sequences of controls.

sults for optimum staging of the chillers and cooling towers as a function of plant load. From these trends, the staging strategy of chillers and cooling tower cells summarized in *Table 2* was developed. The 100 ton (350 kW) difference between the stage up and stage down setpoints in *Table 2* is provided to prevent equipment short-cycling.

We plan to stage the number of chilled and condenser water pumps to match the number of chillers. With a simplified pump model (using constant pump efficiency), the number of operating pumps has no impact on the analysis. In a more detailed analysis, the number of pumps would fall out from the variations of pump efficiency that resulted from operating pumps at different points on their pump curves. We have found from experience that this increased complexity of analysis and control is generally not warranted by energy savings until you get to large pump motor sizes (100 hp and up). (For an excellent discussion on pump staging and analysis, see Reference 11.)

Control of CW Pump Speeds. *Figure 5* is a plot of optimized condenser water pump speed against the plant load. The y-axis of *Figure 5* is the product of the percent condenser water pump speed times the number of operating condenser water pumps (which in turn is equal to the number of chillers). *Figure 5* was just one of many correlations that we examined in searching for component control schemes that could approach the TOPP model. In our control system the condenser water pump speeds will be directly controlled from the total plant load using the equation in *Figure 5*. This sequence can be implemented in the control system without a control loop, avoiding the issues of control

Stage	Stage Down	Stage Up	Number of Chillers	Number of Tower Cells
1		340	1	3
2	240	550	1	4
3	450	750	1	5
4	650	1600	2	5
5	1500		3	5

Table 2: Staging results.

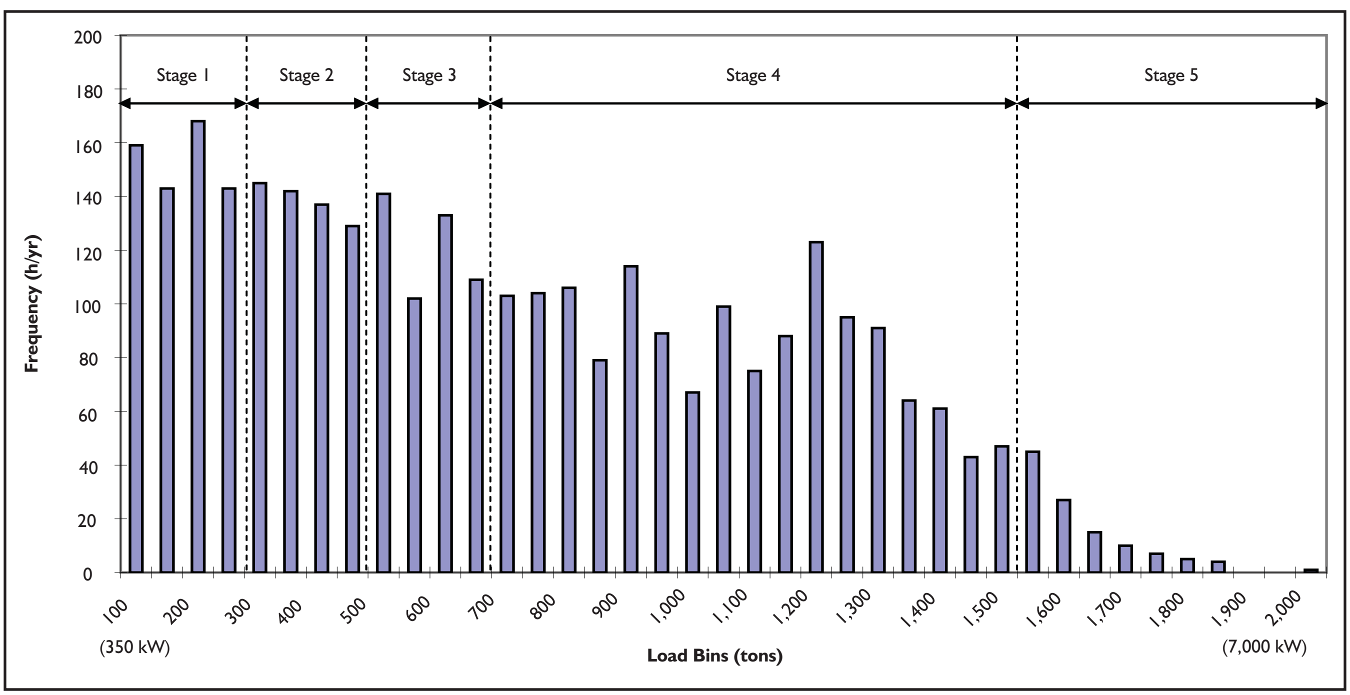

Figure 3: Load profile (histogram).

Advertisement formerly in this space.

loop tuning.

In the control system, we will reset this speed upwards as required to keep the chillers out of the surge region.‡ In the simulations we maintained the condenser water pump speed to achieve or exceed the chiller manufacturer's recommended minimum condenser water flow.

Control of Cooling Tower Fan Speeds. Similar to condenser water pumps, we found a direct control scheme for the cooling towers by examining the TOPP model results. *Figure 6* presents the correlation of the cooling tower fan speed as a function of plant load. In this sequence, all operating cooling tower fans will be running at the same speed. In our control system, the cooling tower fan speeds will be controlled directly from the total plant load using the equation in *Figure 6*.

We examined resetting of condenser water supply temperature as a function of outdoor wet-bulb temperature. The results of this alternate control sequence are shown in *Table 1* as Run 2. This common sequence uses approximately the same energy as our proposed run (within the inherent inaccuracy of the simulations) and has the drawbacks of having to tune a control loop and maintain humidity sensors, which require frequent recalibration.

Runs 3 and 4 in *Table 1* show the performance of two more conventional control sequences. In both of these runs, the condenser water pumps were modeled as constant speed and the cooling tower fans were provided with variable speed drives but controlled conventionally: in Run 3 they were controlled to reset by load following the ARI Standard 550/590[4] condenser water reset scheme; in Run 4 they were controlled to a fixed condenser water temperature at the design condition (80°F [27°C]). Run 3 used ~11% more energy than our recommended scheme. Run 4 used ~32% more energy.

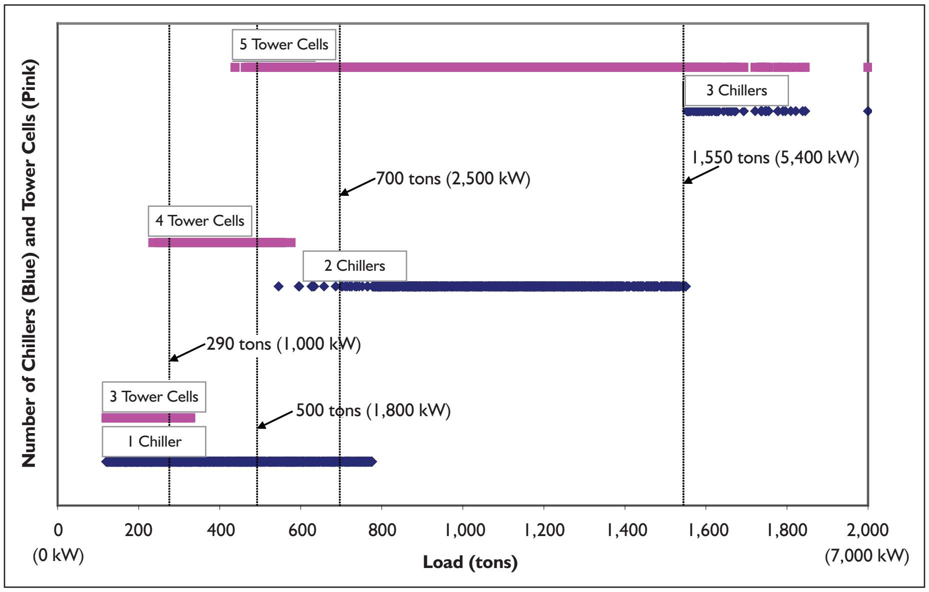

Figure 4: TOPP staging of towers and chillers.

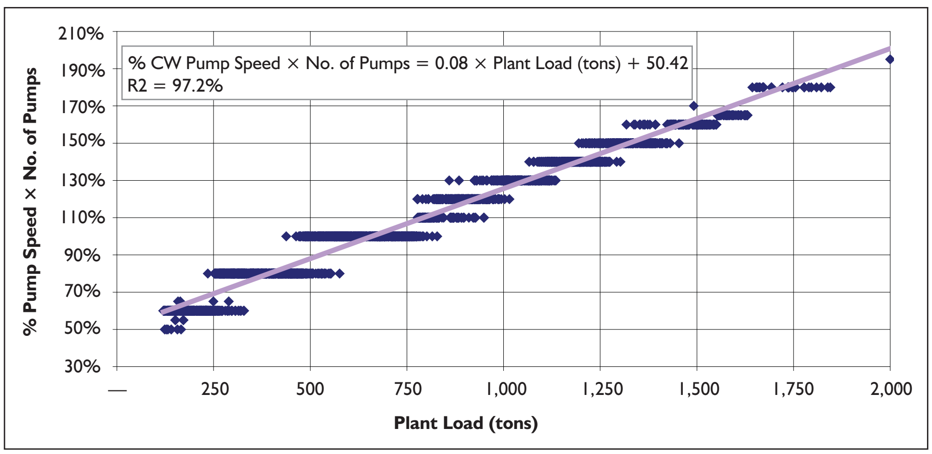

Figure 5: Condenser water pump control.

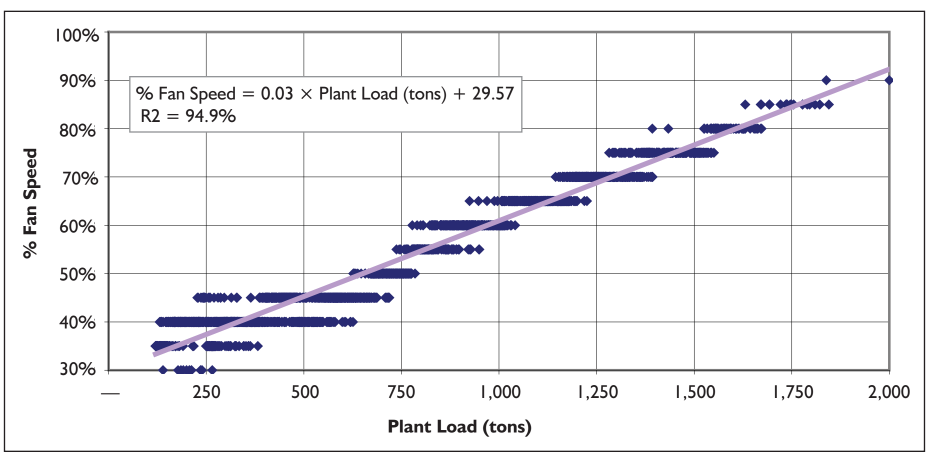

Figure 6: Cooling tower fan control.

Conclusions

Each chilled water plant is unique. In nearly a decade of performing optimizations of chilled water plant equipment and controls, we have found a wide variety in the optimum operating sequences. Some of the sequences can be generalized and applied to other plants, but to truly optimize performance, we

‡ We infer safe (non-surge) operation using the maximum difference between the condenser water return and chilled water supply temperatures as a function of chiller loads as reported in the manufacturer's submitted load data.

Advertisement formerly in this space.

have found that a detailed analysis like the one described in this paper is required. These modeling techniques take between 20 and 60 hours to perform but as documented in *Table 1*. can prove cost effective through energy savings.

If you compare Run 3 (conventional practice with condenser water reset) to Run 1 (our recommended sequence) we achieve an estimated reduction in energy usage of 120,000 kWh/yr. At current energy costs this could yield savings of $12,000 to $15,000 per year. In addition, we developed a direct control sequence for the cooling tower fans that will be easier to set up and maintain than a conventional control loop with reset. We would never have discovered this direct control scheme without the use of simulation. The extra engineering effort is offset in the first year's savings alone. In addition to the cost savings, this analysis helps us to better understand the operating characteristics of our plants, and provides calibrated models that can be used for commissioning and then integrated into the control system for automated fault diagnostics.

References

1. Pacific Gas and Electric Company. 1997. "Commercial Building Survey, Summary Report to the California Energy Commission."
2. Taylor, S. 1999. *The CoolTools™ Chilled Water Plant Design and Performance Specification Guide.* PG&E Pacific Energy Center. www.taylor-engineering.com/downloads/cooltools/CT-016_Design_Guide.pdf.
3. Hydeman, M. 1999. "Commissioning tools & techniques used in a large chilled water plant optimization project." *Proceedings of the 7th National Conference on Building Commissioning*. Portland Energy Conservation Inc. (PECI).
4. Air-Conditioning and Refrigeration Institute. ARI Standard 550/590-2003, *Performance Rating of Water-Chilling Packages Using the Vapor Compression Cycle*.
5. Hydeman, M., S. Taylor and J. Stein. 2002. "Codes and Standards Enhancement Report, 2005 Title 24 Building Energy Efficiency Standards Update, Code Change Proposal for Cooling Towers." April 8. This report was presented to the California Energy Commission in support for three prescriptive code changes: a limitation on centrifugal fan cooling towers, a requirement for minimum flow turndown for cooling tower cells and a limitation of air-cooled chillers.
6. Hydeman, M. et al. "Development and testing of a reformulated regression based electric chiller model." *ASHRAE Transactions* 108(2):1118–27.
7. Gordon, J.M., et al. 2000. "How varying condenser coolant flow rate affects chiller performance: thermodynamic modeling and experimental confirmation." *Applied Thermal Engineering* 20:1149–99.
8. Hydeman, M. and K. Gillespie. 2002. "Tools and techniques to calibrate electric chiller component models." *ASHRAE Transactions* 108(1):733–41.
9. Benton, D., et al. 2002. "An improved cooling tower algorithm for the CoolTools™ simulation model." *ASHRAE Transactions* 108(1):760–68.
10. Hartman, T. 2001. "All-variable speed centrifugal chiller plants." *ASHRAE Journal* 43(9):43–51.
11. Rishel, J.B. 1996. *HVAC Pump Handbook*. N.Y.: McGraw-Hill Professional.

Advertisement formerly in this space.