Water-cooled centrifugal chiller plant.

Optimizing Chillers & Towers

By **Hugh Crowther, P.Eng.,** Member ASHRAE and **James Furlong, P.E.,** Member ASHRAE

How a system operates at design conditions is not always a good indicator of its overall annual performance. This article emphasizes controls logic and design conditions and how they can be used to optimize chiller, cooling tower and condenser pump system performance throughout the year. A software model of hour-by-hour energy use for a typical office building in three cities demonstrates the effect of different cooling seasons and wet-bulb profiles. (Results are presented in energy use [kWh]. Actual dollar savings can be estimated by assuming a blended energy rate.)

The model is based on a 160,000 ft² (14 864 m²), eight-story office building with a variable air volume (VAV) HVAC system that includes two centrifugal chillers in a variable primary flow configuration (*Figure 2*). The system uses induced draft-type cooling towers and constant flow condenser pumps that are dedicated to each chiller. Minimum turndown for the chillers is 50% and the bypass line opens below 25% of chiller plant capacity. Condenser pumps and cooling towers operate only when a chiller is on-line. The specific equipment size changes slightly with each location and is listed in the tables. *Figure 1* shows the annual cooling load profile for the three locations—Chicago, Las Vegas and Miami.

Design conditions are based around the Air Conditioning and Refrigeration Institute Standard 550/590 – *Water Chilling Packages Using the Vapor Compression Cycle* (54°F /44°F [12.2°C/6.7°C] chilled water, 85°F/95°F [29.4°C/35°C] condenser water) and the Cooling Tower Institute's (78°F [25.6°C] wet-bulb) test conditions, where possible, with exceptions noted. Changing these design con-

About the Authors

Hugh Crowther, P.Eng., is the vice president of product management & technology at McQuay International in Plymouth, Minn. He is a member of the ASHRAE Standards Committee. **James Furlong, P.E.,** is vice president comfort cooling marketing at Baltimore Aircoil Company in Baltimore.

ditions can have a noticeable effect on annual energy use as well as installed cost. The fixed design conditions allow comparisons of operating logic for each location.

Control Algorithms

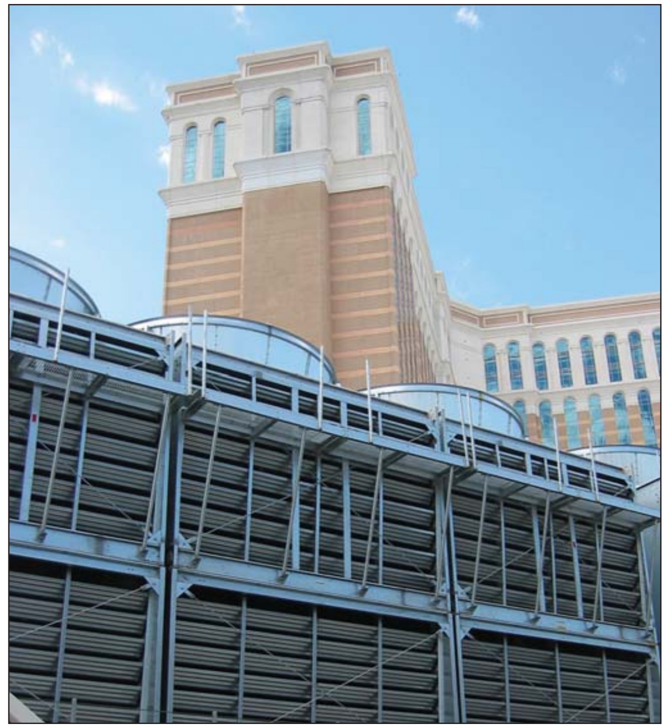
The first analysis is based on optimizing the chiller–condenser pump–tower. *Figure 3* shows the optimal point at which the chiller and tower's combined energy use is at the lowest. It is expected that the chiller plant operates at full load and at design parameters only at design conditions. During other periods (which is 99% of the time), we expect the chiller plant to operate at less than design load, which provides an opportunity for condenser water relief. Condenser water relief means that the condenser water temperature can be lowered because the cooling tower is no longer at 100% capacity and the ambient wet bulb is not at design condition. Lowering the condenser water temperature can provide significant chiller energy savings, as shown in *Figure 4.*

How the system will respond to non-design conditions can be analyzed by looking at the cooling tower fan modulation method and the control logic for condenser water supply temperature.

Figure 5 shows the fan power requirement for three fan modulation methods: on-off fans, two-speed (or pony) fan motors and variable frequency drives (VFD). In addition, three condenser water control methods have been considered: a fixed setpoint of 65°F (18.3°C), a mixed wet-bulb approach of 7°F (4°C) and optimized.

Fixed setpoint is an aquastat in the supply condenser water line that is set at the coldest water temperature you are willing to operate the chillers. Using 65°F (18.3°C) is a good choice because most chillers can operate at light loads with condenser water at this temperature. Colder water temperatures are possible, but the chiller type and load will become more important (e.g., as the chiller load is decreased, the water temperature will need to be raised). Fixed setpoint control logic effectively runs the cooling tower at full airflow until the minimum temperature is reached, and then modulates the fan. This optimizes the chiller, but not the system.

Fixed approach requires monitoring the ambient wet bulb. The cooling tower design approach is added to the current wet-bulb temperature to derive the setpoint. This logic assumes a correlation exists between reduced ambient wet bulb and building load. The condenser water temperature is lowered as the ambient wet bulb drops. As the chiller load drops (heat sent to the tower), the cooling tower fans will modulate and offer fan power savings. The process is complicated by how towers behave in reduced wet-bulb situations (see sidebar on cooling towers following this article).



Crossflow cooling tower installation.

The **optimized method** requires auto-adaptive controls. This control logic constantly adjusts the condenser water supply temperature to the value that uses the least amount of power. The controller measures the power requirement for the chiller and cooling tower. The condenser water temperature setpoint then is altered and the power consumption is checked again. If the total power consumption goes down, a similar adjustment is made and the total power is checked again.

Table 1 shows the specific HVAC system parameters at design conditions.

Table 2 shows the results of each control sequence for the Chicago office example.[2] Using the fixed setpoint and on-off fan control as a baseline, the analysis shows a 6.26% improvement when using VFD-controlled fans and optimized controls. This is substantial because the system components are the same for each example. Only the operation of those components at part load was changed. Referring to *Figure 3,* it is noted that the savings in tower fan work with optimized control far exceeded the increase in chiller work.

The next example is based on the same model building in Las Vegas — a dry hot climate with significantly more chiller operating hours. *Table 3* shows the system parameters. The design condenser water temperature was maintained at 85°F (29.4°C) to allow direct comparisons. The same control sequences were modeled, and the results are shown in *Table 4*.

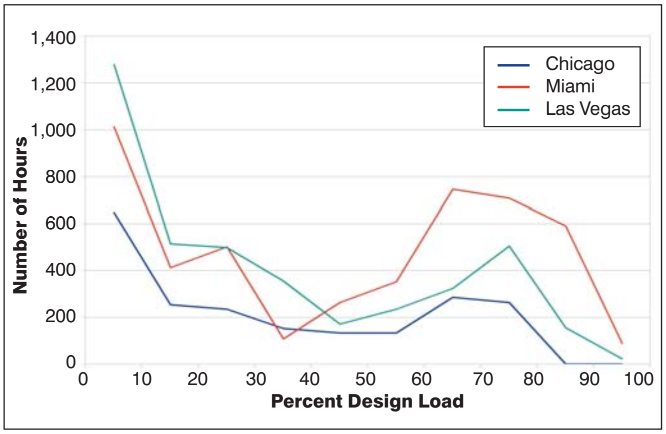


Figure 1: Annual load profiles.

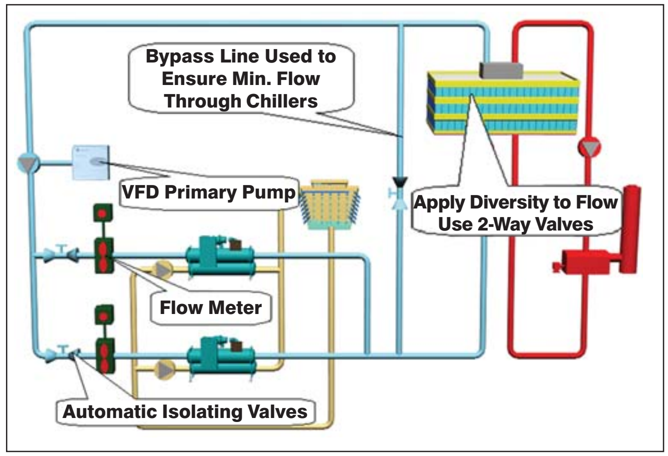


Figure 2: Chiller plant layout.

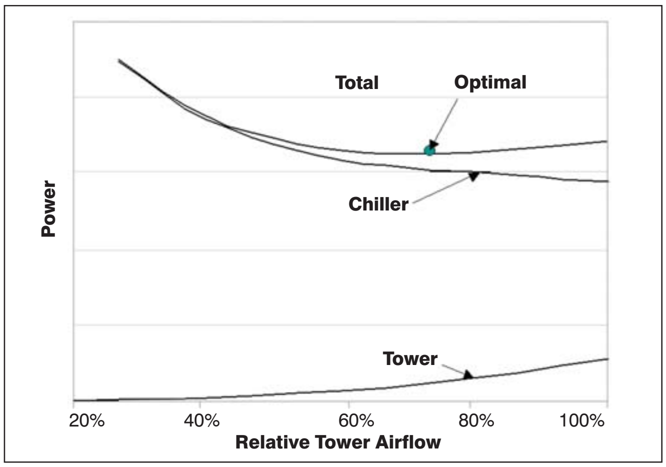


Figure 3: Chiller power vs. tower power.[1]

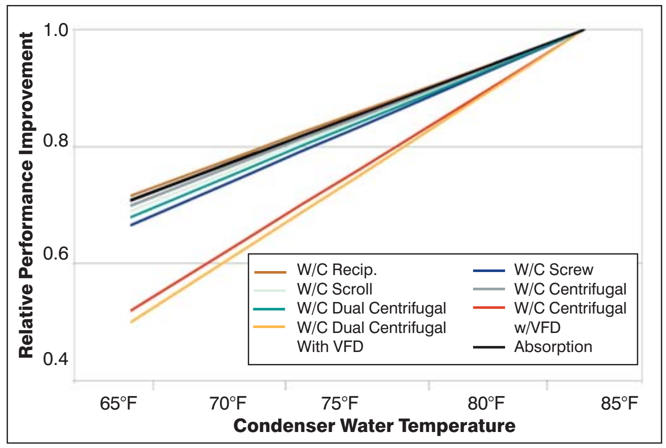


Figure 4: Chiller efficiency vs. condenser relief.

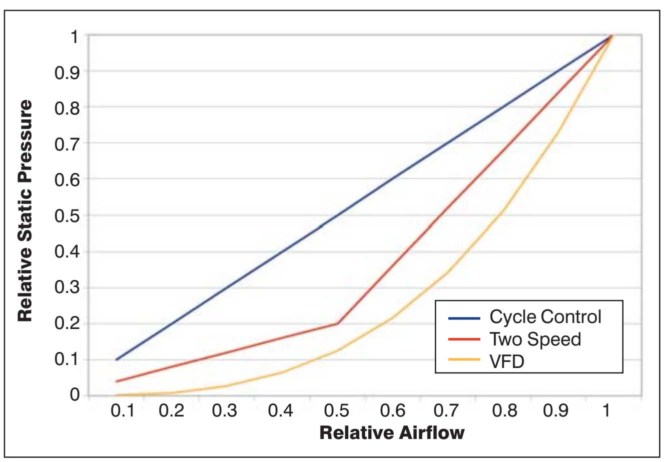


Figure 5: Cooling tower fan modulation.

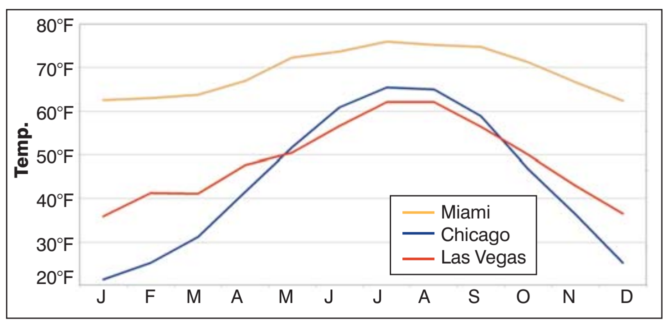


Figure 6: Annual wet-bulb profiles.

For the drier climate, the maximum relative savings from the baseline were only 4.66%. However, the actual energy savings went from 8,819 kWh/year for Chicago to 11,433 kWh/year for Las Vegas. The reasons for this difference are a combination of how cooling towers behave in dry climates (see cooling towers sidebar) and the difference in wet-bulb profiles between Chicago and Las Vegas.

The final example is based on the same building in Miami. Again, the location offers substantial chiller operating hours, but it also has a high wet-bulb with a flatter annual profile (*Figure 6*). In this case, the cooling towers were sized for a 5°F (2.8°C) approach due to the high design wet-bulb. *Table 5* shows the system parameters and *Table 6* shows the performance.

It is interesting that the location with the most humid climate enjoys the best enhancement by optimizing the chiller–tower relationship. From the baseline to the maximum savings, the performance improvement was more than 8%. In addi-

Example 1 – Chicago

Building Type	Office
HVAC System	VAV with Reheat
HVAC System Configuration	Variable Primary Flow Chiller Plant
Floor Area	160,000 ft² (14,864 m²)
Floors	8
Location	Chicago
Summer Design DB/WB	91°F/74°F (32.8°C/23.3°C)
Cooling Tower Rating – WB	78°F (25.6°C)
Design Cooling Load	400 tons (1400 kW)
Supply Fan Power	75.6 kW
Return Fan Power	22.7 kW
Number and Size of Chillers	2 at 200 tons (700 kW)
Chiller Performance	0.55 kW/ton
Chilled Water Temperatures	54°F/44°F (12.2°C/6.7°C)
Condenser Water Temperatures	85°F/95°F (29.4°C/35°C)
Number and Size of Primary Pumps	1 at 23 kW
Number and Size of Condenser Pumps	2 at 8 kW
Number and Size of Tower Fans	2 at 9 kW

Table 1: Chicago system parameters.

	Tower (kWh/yr)	Chiller (kWh/yr)	Total (kWh/yr)	Change From Baseline
65°F (18.3 °C) Setpoint				
On – Off	20,000	129,679	149,679	Baseline
Two Speed	19,271	129,679	148,950	0.49%
VFD	18,754	129,679	148,433	0.84%
Fixed Approach				
On – Off	17,537	130,475	148,012	1.13%
Two Speed	16,326	130,475	146,801	1.96%
VFD	15,317	130,475	145,792	2.67%
Optimized				
On – Off	15,649	132,176	147,825	1.25%
Two Speed	7,936	136,487	144,423	3.64%
VFD	5,892	134,968	140,860	6.26%

Table 2: Chicago chiller-tower performance.

Example 2 – Las Vegas

Building Type	Office
HVAC System	VAV with Reheat
HVAC System Configuration	Variable Primary Flow Chiller Plant
Floor Area	160,000 ft² (14 864 m²)
Floors	8
Location	Las Vegas
Summer Design DB/WB	108°F/66°F (42.2°C/18.9°C)
Cooling Tower Rating – WB	78°F (25.6°C)
Design Cooling Load	394 tons (1386 kW)
Supply Fan Power	79.1 kW
Return Fan Power	23.7 kW
Number and Size of Chillers	2 at 200 tons (700 kW)
Chiller Performance	0.55 kW/ton
Chilled Water Temperatures	54°F/44°F (12.2°C/6.7°C)
Condenser Water Temperatures	85°F/95°F (29.4°C/35°C)
Number and Size of Primary Pumps	1 at 22 kW
Number and Size of Condenser Pumps	2 at 8 kW
Number and Size of Tower Fans	2 at 9 kW

Table 3: Las Vegas system parameters.

	Tower (kWh/yr)	Chiller (kWh/yr)	Total (kWh/yr)	Change From Baseline
65°F (18.3 °C) Setpoint				
On – Off	30,212	226,514	256,726	Baseline
Two Speed	27,207	226,514	253,721	1.18%
VFD	25,021	226,514	251,535	2.06%
Fixed Approach				
On – Off	29,382	226,960	256,342	0.15%
Two Speed	26,090	226,960	253,050	1.45%
VFD	23,634	226,960	250,594	2.45%
Optimized				
On – Off	28,443	222,814	251,257	2.18%
Two Speed	16,831	234,103	250,934	2.31%
VFD	11,384	233,909	245,293	4.66%

Table 4: Las Vegas chiller-tower performance.

tion, the absolute savings (more than 40,000 kWh/yr) were the greatest due to the extended cooling season. The fixed setpoint systems all performed the same, regardless of fan control logic, because the high local wet-bulb never allows the fan to modulate. The control systems that allow fan modulation offer very good savings (remember 80% airflow results in 50% power reduction).

What about Design Conditions?

The previous three examples were based on the typical operating conditions specified in ARI 550/590 and the CTI Standard 201-2002, *Cooling Tower Test*. Moving away from these criteria can result in further system improvement.

One consideration is to "oversize" the cooling tower and lower the design condenser water temperature. This will increase the size and cost of the cooling tower, but it will have an advantageous effect on the chiller. A good place to start is to lower the design cooling tower approach by 2°F (1.1°C). The Cooling Tower Institute (CTI) Standard 201 program will certify cooling towers down to 5°F (2.8°C) approach.

Table 7 shows the Chicago example but with an oversized cooling tower capable of producing 83°F (28°C) water with a 5°F (2.8°C) approach. The tower fan motors increased from 8 to 11 kW. It is important to note that an oversized cooling tower actually uses more power annually with poor tower fan control. However, the same oversized cooling tower can produce about a 3% improvement with good tower fan control.

What About the Condenser Pump?

In the previous examples, we held the condenser pump constant and sized at 3 gpm/ton (0.0538 L/s per kW). Varying the flow of the condenser pump during part-load conditions also can lower the operating cost of the system. However, varying the flow based on load presents challenges. First, the cooling towers must be selected to operate with reduced flow. Also, the control sequence can be difficult.

Optimizing the chiller and tower to use the least amount of power and holding the condenser pump constant is easier to manage because it offers two degrees of freedom (the chiller and tower power) and one parameter (the condenser water supply temperature). Adding the third parameter (condenser flow) can be a difficult problem to solve. Thomas Hartman, P.E., Member ASHRAE,[3] has done some interesting work in this area. The author has experience using a stepped flow reduction approach, and for these projects, the condenser flow was reduced to 80% (a 50% improvement in condenser pump work) with little adverse affect on the cooling tower. The deciding factor in lowering the flow in each case was based on circumstances where one chiller was off in a series chiller plant, or where one compressor was off in a dual compressor chiller.

Changing the design conditions can also affect the system performance. Increasing the temperature range should be given very careful consideration. While this will reduce the flow and pump power, it will adversely affect the chiller work.[4,5] In many applications, the overall operating cost will rise, even with a 1°F (0.8°C) increase in range, depending on the relative size of the condenser pumps to the chillers. In applications with high pump head, it may make sense to increase the range.

Conclusions

Optimizing the chiller–tower condenser–pump system can be managed in two steps. First, optimize the design conditions while taking into account your local design weather and project specific details. Second, optimize the control logic to take full advantage on the capital equipment in the chiller plant. Changing from very basic to advanced controls can make a 5% to 8% improvement in chiller–tower condenser pump performance without changing the capital equipment.

The payback for integrating more effective controls will depend on the actual plant size and the number of operating hours. The cost of the additional controls remains relatively constant regardless of equipment size, which implies that larger plants will enjoy faster paybacks. Additionally, more operating hours (health care vs. office space) will improve the payback. As a benchmarking tool, designers can use the percent savings times the estimated operating cost of a basic system to estimate the annual savings. The cost of the additional controls can be used to estimate the payback for the project.

This article did not investigate the affect of changing the condenser water temperature range on the savings with more advanced cooling tower controls or different climatic conditions. Changing the temperature range affects many things and this would be a good future article.

Bibliography

Braun, J.E. and G.T. Diderrich. 1990. "Near-Optimal Control of Cooling Towers for Chilled Water Systems." *ASHRAE Transactions* SL-90-13-3.

Notes

1. Braun, J.E., and G.T. Diderrich. 1990. *Near-Optimal Control of Cooling Towers For Chilled Water Systems. ASHRAE Transactions*.
2. Annual energy analysis was performed using McQuay Energy Analyzer 4.0.
3. Hartman, T. 2001. "All-variable speed centrifugal chiller plants." *ASHRAE Journal* 43(9).
4. Kirsner, W. 1996. "Three gpm/ton condenser water rate: does it waste energy?" *ASHRAE Journal* 38(2).
5. Crowther, H. 2002. "Why change the chilled water range?" Minneapolis: McQuay Engineering Solutions.

Example 3 – Miami

Building Type	Office
HVAC System	VAV with Reheat
HVAC System Configuration	Variable Primary Flow Chiller Plant
Floor Area	160,000 ft² (14,864 m²)
Floors	8
Location	Miami
Summer Design DB/WB	91°F/77°F (32.8°C/25°C)
Cooling Tower Rating – WB	78°F (25.6°C)
Design Cooling Load	426 tons (1498 kW)
Supply Fan Power	76.4 kW
Return Fan Power	23.7 kW
Number and Size of Chillers	2 at 213.7 tons (751.6 kW)
Chiller Performance	0.55 kW/ton
Chilled Water Temperatures	54°F/44°F (12.2°C/6.7°C)
Condenser Water Temperatures	85°F/95°F (29.4°C/35°C)
Number and Size of Primary Pumps	1 at 24 kW
Number and Size of Condenser Pumps	2 at 9 kW
Number and Size of Tower Fans	2 at 10 kW

Table 5: Miami system parameters.

	Tower (kWh/yr)	Chiller (kWh/yr)	Total (kWh/yr)	Change From Baseline
65°F (18.3 °C) Setpoint				
On – Off	75,354	440,145	515,499	Baseline
Two Speed	75,130	440,145	515,275	0.04%
VFD	74,972	440,145	515,117	0.07%
Fixed Approach				
On – Off	63,370	442,873	506,243	1.83%
Two Speed	60,055	442,873	502,928	2.50%
VFD	56,841	442,873	499,714	3.16%
Optimized				
On – Off	43,026	457,956	500,982	2.90%
Two Speed	16,564	466,681	483,245	6.67%
VFD	14,815	460,094	474,909	8.55%

Table 6: Miami chiller-tower performance.

	Tower (kWh/yr)	Chiller (kWh/yr)	Total (kWh/yr)	Change From Baseline
65°F (18.3 °C) Setpoint				
On – Off	25,268	125,990	151,258	Baseline
Two Speed	24,050	125,990	150,040	0.81%
VFD	23,294	125,990	149,284	1.32%
Fixed Approach				
On – Off	22,491	126,408	148,899	1.58%
Two Speed	20,839	126,408	147,247	2.72%
VFD	19,606	126,408	146,014	3.59%
Optimized				
On – Off	14,725	131,590	146,315	3.38%
Two Speed	5,608	133,824	139,432	8.48%
VFD	4,503	132,079	136,582	10.75%

Table 7: Chicago chiller-tower performance with oversized tower.

"Cooling Towers: Lower Flow Systems" sidebar follows.

Cooling Towers: Lower Flow Systems

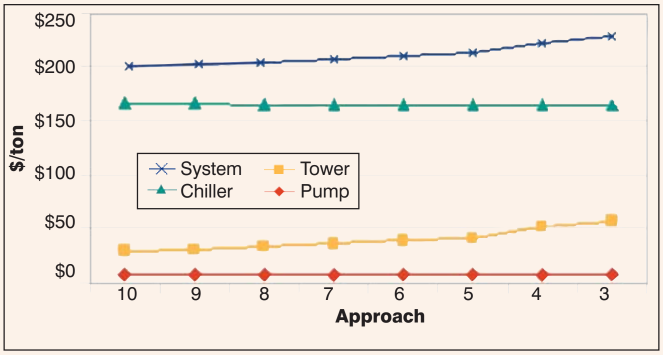


Figure 1: Lowest system first cost 78 wet bulb, 3 gpm/ton.

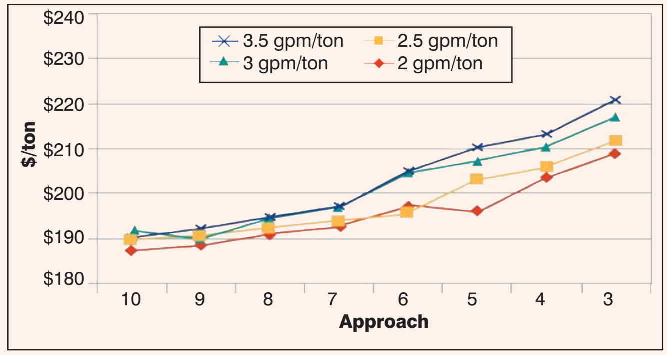


Figure 2: Lowest system first cost, 66°F wet bulb.

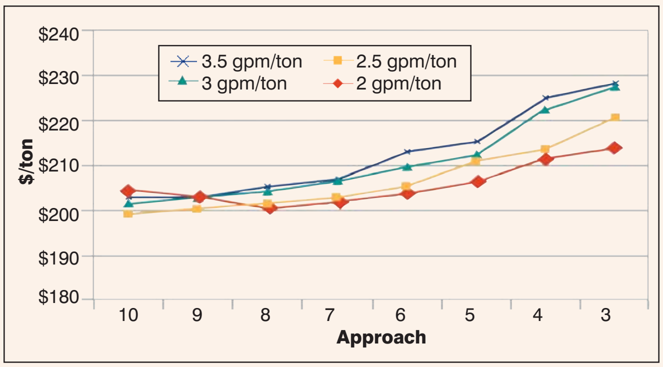


Figure 3: Lowest system first cost, 78°F wet bulb, variable gpm/ton.

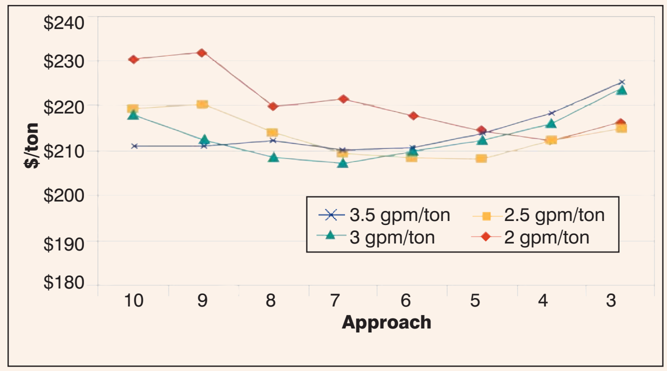


Figure 4: Lowest system first cost, 84°F wet bulb.

For more than 50 years, electric motor-driven, water-cooled chiller systems commonly have been designed around entering condenser water temperatures of 85°F (29.4°C) and a nominal condenser water flow of 3 gpm/ton (0.0538 L/s per kW). For reciprocating-type water-cooled chillers rated at 0.90 kW/ton, the 3 gpm/ton (0.0538 L/s per kW) nominal flow rate yields a 10°F (5.5°C) range, which has long been the ARI standard rating condition for water-cooled chillers.

In recent years, considerable debate has occurred as to the merits of designing around lower nominal condenser water flow rates to improve system life-cycle costs. Those who believe in using 2 gpm/ton (0.0359 L/s per kW) claim that there is a first-cost advantage to the lower flow system. This is derived from the lower costs associated with smaller pumps, smaller pipe, and a smaller cooling tower that more than offset any increases in cost associated with the additional heat transfer surface that may be required in the chiller. They also claim that the lower flow system can deliver improved operating costs because the reduced kW required by smaller pumps and cooling tower fan motors more than offsets the increased power required by the chiller to overcome the greater lift imposed by higher condensing temperatures.

A discussion of condenser water flow-rate optimization with respect to chiller system first cost, operating cost, or life-cycle cost cannot reasonably occur unless two additional parameters are taken into consideration: approach and design wet bulb. Approach, or the temperature difference between the water leaving the cooling tower and the ambient wet bulb, has a more significant influence on cooling tower size and energy consumption than any other parameter affecting the cooling tower. Traditionally, the HVAC industry has designed around approaches of 7°F (3.8°C) or greater for two reasons. First, the majority of geographies are associated with design wet bulbs of 78°F (25.6°C) and below, making the attainment of "standard" 85°F (29.4°C) condenser water possible. Second, an industry mindset prevails that accepts 85°F (29.4°C) condenser water temperatures as "ideal" from an overall system energy standpoint. Whether this mindset is based on reality gives rise to the following question: what nominal condenser water flow rate and approach is best from a first-cost and full-load energy perspective at any given wet bulb?

A study was recently completed in an effort to answer this question, using actual first cost and full load performance data from a variety of chiller, cooling tower, and pump manufacturers for a nominal 500 ton (1759 kW) water-cooled, centrifugal chiller system. The manufacturers involved supplied one set of equipment selections and pricing data aimed at optimizing first cost at the expense of efficiency (low cost but also inefficient) and a separate set aimed at optimizing full load energy consumption at the expense of first cost (efficient but expensive). Chiller evaporator conditions were fixed at 54°F (12.2°C) entering and 44°F (6.7°C) leaving temperatures with a maximum pressure drop of 20 ft w.c. (59.8 kPa). Chiller condensers were all limited to a maximum pressure drop of 20 ft w.c. (59.8 kPa) and condenser pumps were sized at a fixed system head of 60 ft w.c. (179.4 kPa). Piping costs and system pressure drops were not taken into consideration.

The study considered a range of nominal condenser flows from 2.0 to 3.5 gpm/ton (0.036 L/s per kW to 0.063 L/s per kW) in 0.5 gpm/ton (0.009 L/s per kW) increments, cooling tower approaches from 3°F (1.6°C) to 10°F (5.5°C) in 1°F (0.5°C) increments, and wet-bulb conditions from 66°F (18.9°C) to 84°F (28.9°C) in 6°F (3.3°C) increments.

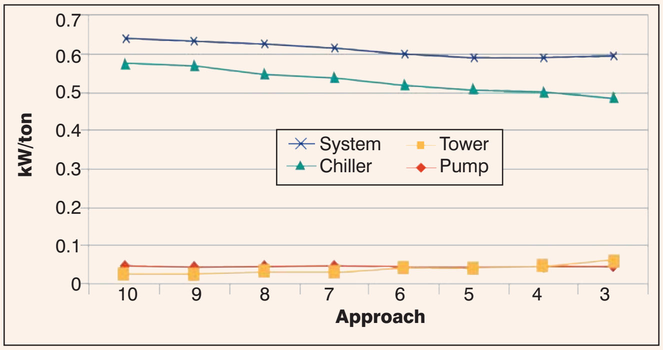


Figure 5: Lowest system energy, 78°F wet bulb, 3 gpm/ton.

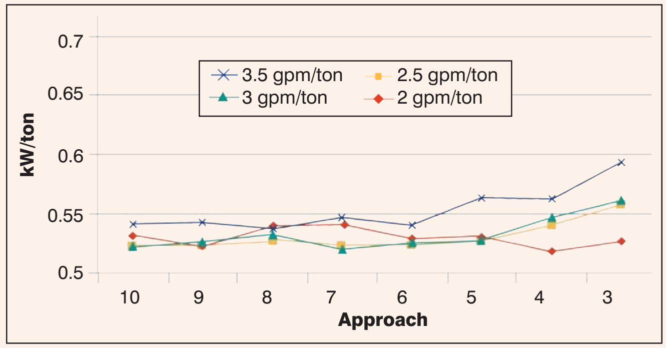


Figure 6: Lowest system energy, 66°F wet bulb.

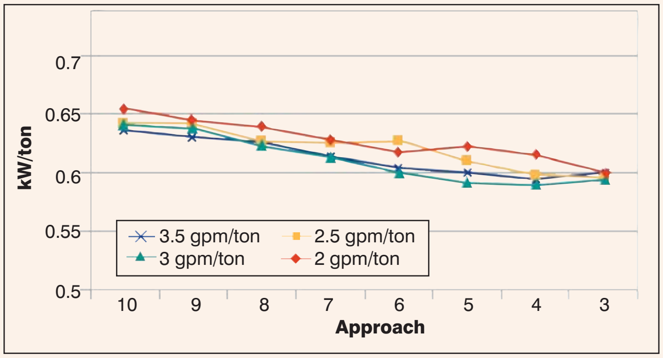


Figure 7: Lowest system energy, 78°F wet bulb, variable gpm/ton.

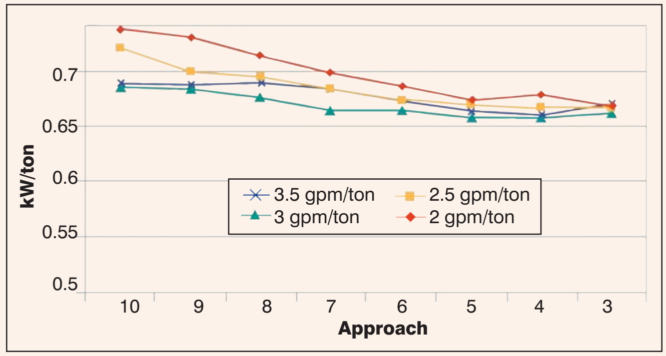


Figure 8: Lowest system energy, 84°F wet bulb.

Optimizing First Costs

Equipment was selected on the basis of meeting full load capacity at lowest possible first cost. Component equipment first costs were plotted as a function of approach for all combinations of nominal flow rate and wet-bulb temperature. Sum of all component costs was taken as the system cost (*Figure 1*).

Removing the component detail allows the system costs to be plotted as a function of nominal flow rate at a given wet bulb. Plots at 66°F (18.9°C), 78°F (25.6°C), and 84°F (28.9°C) wet bulb are shown in *Figures 2* through *4.* A review of these plots will reveal that lower condenser water flows (2.0 to 2.5 gpm/ton [0.0359 to 0.0449 L/s per kW]) and higher approaches (8°F to 10°F [4.4°C to 5.5°C]) are a viable means to optimize system first costs at all but the highest wet-bulb environments (84°F [28.9°C]). The first cost advantage of the lower flow systems would likely have been more pronounced if the material cost savings of smaller diameter condenser water piping had been considered.

Optimizing Full Load Energy Consumption

Equipment was selected on the basis of meeting full load capacity at the lowest possible energy consumption. In the case of the chiller selections and cooling tower selections, this translated to equipment with significantly greater heat transfer surface than the equipment selected based on optimizing first costs. The full load energy consumption of component equipment was plotted as a function of approach for all combinations of nominal flow rates and wet-bulb temperatures. Full-load system energy consumption is taken as the sum of the full load energy consumption of all system components (*Figure 5*). Plots of full-load energy consumption as a function of nominal flow at 66°F (18.9°C), 78°F (25.6°C), and 84°F (28.9°C) are shown in *Figures 6* through *8.*

A summary of the optimized energy cost plots is revealing. The higher the ambient wet bulb, the higher the required nominal condenser water flow rate and the tighter the approach needs to be to optimize system energy cost. Only at the lowest wet-bulb condition considered, 66°F (18.9°C), does a 2.0 or 2.5 gpm/ton (0.0359 or 0.0449 L/s per kW) system show an energy advantage. At 78°F (25.6°C) and 84°F (28.9°C) wet bulb, the optimal energy balance occurs at 3 gpm/ton (0.0538 L/s per kW) with a 4°F (2.2°C) approach.

The advantage of a tighter approach in the higher wet-bulb environments is significant. At 78°F (25.6°C) wet bulb, the optimum system energy is achieved at 3 gpm/ton (0.0538 L/s per kW) with a 4°F (2.2°C) approach. Had a "typical" 3 gpm/ton (0.0538 L/s per kW), 7°F (3.8°C) approach been specified on this system, a system energy penalty of more than 4% would have occurred. It is also interesting to note that approach and condenser flow have little impact on system energy in low wet-bulb environments. At 66°F (18.9°C) wet bulb, system energy is largely unchanged as the approach is tightened from 10°F (5.5°C) to 5°F (2.7°C). The energy saved by the chiller when operating with 71°F (21.7°C) vs. 76°F (24.4°C) entering condenser water is entirely offset by the additional energy consumed by the cooling tower fan to produce the colder condenser water. In this case, it would make no sense to invest in larger size cooling tower required to generate the 5°F (2.7°C) approach.

A number of conclusions can be drawn from the study:

• Low-flow (2 gpm/ton [0.0359 L/s per kW]) condenser water systems generally have first cost advantages over higher flow (3 gpm/ton [0.0538 L/s per kW]) systems in all but the highest wet-bulb environments.

• High-flow (3 gpm/ton [0.0538 L/s per kW]) condenser water systems generally have full load energy advantages over lower flow (2 gpm/ton [0.0359 L/s per kW]) systems in all but the lowest wet-bulb environments

• Approaches in the range of 4°F (2.2°C) to 5°F (2.7°C) offer significant full load system energy advantages over "traditional" 7°F (3.8°C) approaches in environments with higher wet bulbs (78°F to 84°F [25.6°C to 28.9°C]).