Optimization of Water - Cooled Chiller – Cooling Tower Combinations

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Abstract

Water-cooled chiller systems have typically been designed around entering condenser water temperatures of 85°F with a nominal condenser water flow of 3.0 USGPM/ton and a 10°F range. In recent years, there has been considerable debate on the merits of designing around lower condenser water flow rates with a higher range in order to improve system lifecycle costs.

However, two other parameters must also be considered in any analysis - approach and design wet bulb. The question to be answered is:

What nominal condenser water flow rate and approach is best from a first cost standpoint as well as from a full load energy standpoint 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 water-cooled, centrifugal chiller system. This paper reports the findings from that study.

Introduction & Background

Many facilities employ chilled water systems for comfort cooling. Three distinct parts comprise every chiller system:

- The *chilled water loop* absorbs heat from the building and then rejects that heat to the chiller.
- The evaporator in the chiller absorbs heat from the chilled water loop and rejects that heat, along with the heat of compression, to the condenser using a *vapor compression cycle*.
- The condenser rejects that heat to the atmosphere.

To start, chilled water coils located in air-handling units throughout the building absorb heat from the building air, transferring that heat to the chilled water loop. The air-handling units distribute the conditioned air to the building. Why use a two-step chilled water process to remove heat rather than removing it directly, such as in a room air conditioner? As buildings become larger, it becomes less practical to pipe refrigerant and/or duct cool air to the extremities of the building. Chilled water, on the other hand, can be easily distributed by insulated pipe, even when piping runs are very long. The larger the space a comfort cooling system serves, the more likely it is to use a chilled water system for cooling.

The warm water leaving the chilled water coils is pumped to the evaporator of the chiller, where the unwanted heat from the building is transferred by the latent heat of vaporization of the refrigerant. The compressor of the chiller then compresses the refrigerant to a higher pressure, adding the heat of compression in the process. The high pressure refrigerant then moves to the condenser, where the unwanted heat is rejected by the latent heat of condensation of the refrigerant to the atmosphere. This system is illustrated in Figure 1 below:

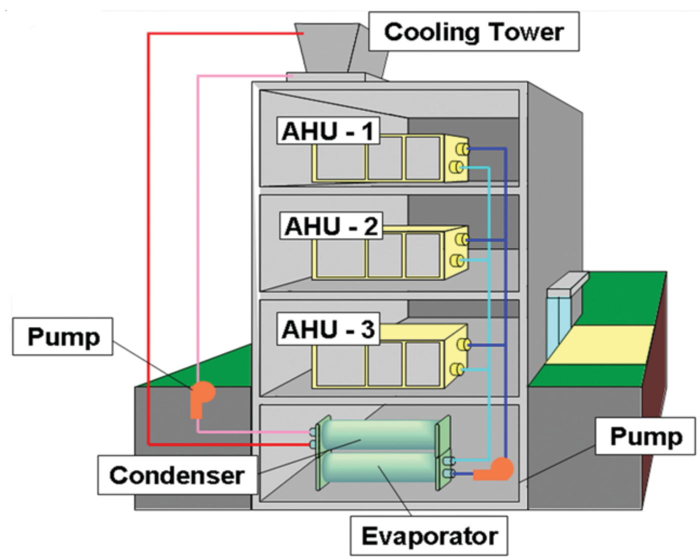

Figure 1 Chilled Water System Overview

Note: AHU-1, 2, & 3 are air-handling units, which house chilled water coils for heat transfer. Cool air is ducted from each air-handling unit to the different conditioned spaces that each unit serves.

Chillers can be classified as air-cooled or water-cooled depending on the method of heat rejection used. Air-cooled chillers reject heat to the atmosphere via a fan that draws air across the condenser coil, condensing the refrigerant within the tubes. Water-cooled chillers reject heat to the water that flows through the condenser tube bundle, typically condensing the refrigerant on the outside of the tubes. The lower process fluid temperatures (and corresponding refrigerant pressures) available through the use of water-cooled equipment lower system energy usage and reduce equipment size, sound levels, and cost. Although the cooling water for the condenser can come from a public water utility, a river or lake, a cooling tower is used in the majority of installations.

The advantages of evaporative cooling over air-cooled heat rejection stem from several key factors. First, cooling towers use the ambient wet-bulb temperature of the entering air as a heat sink, which is typically 10°F to 30°F lower than the dry-bulb temperature, depending on the local climate. The lower the temperature of the heat sink, the more efficient the process. Second, the evaporative cooling process involves both latent and sensible heat transfer (primarily latent), where a small portion of the recirculating water is evaporated to cool the remaining water. Air-cooled systems (which involve sensible cooling only) require a much greater volume of air to reject the same heat load, in turn requiring more fan horsepower to do so. Third, cooling towers allow direct contact between the air and water in the wet deck, or fill media, increasing the efficiency of heat transfer. Fourth, water is a much more efficient heat transfer medium than air which results in smaller equipment sizes for the same capacity both in terms of the condenser tube bundle and the atmospheric heat rejection device.

Because of these advantages, evaporative water-cooled systems consume approximately half the overall energy of comparably sized air-cooled systems, yielding substantial lifecycle cost savings. Low energy consumption is not only a financial victory for the building owner, but also helps to respect and preserve the environment by reducing greenhouse gas emissions from supporting power plants. Lower system temperatures and pressures also reduce system maintenance and extend the life of the mechanical equipment.

The focus of this paper is on water-cooled chiller/evaporative cooling tower combinations and seeks to promote a better understanding of the influence of condenser flowrates and cooling tower approaches on system lifecycle costs.

Condenser Water Flow

For well over half a century, electric motor driven water-cooled chiller systems for commercial cooling applications have more often than not been designed around entering condenser water temperatures of 85°F with a nominal condenser water flow of 3.0 USGPM/ton. Note that here we refer to a traditional cooling tower ton, which is equal to 15,000 Btu/h to account for both the cooling load (12,000 Btu/h = 1.0 ton of refrigeration) and the added heat of compression (approximately 3,000 Btu/h). For a cooling tower heat load of 15,000 Btu/h (1.0 cooling tower ton), the 3.0 USGPM/ton nominal flow rate yields a 10°F range, or temperature increase, across the condenser. This has long been the standard rating condition for water-cooled chillers as found in ARI 550/590-2003. Correspondingly, cooling towers are traditionally selected with the same 10°F range and 3.0 USGPM/ton at a 7°F approach, with the approach defined as the temperature difference between the water leaving the tower and the wet bulb temperature of the entering air. Using the most common wet bulb for most of the United States leads to the traditional tower conditions of 95°F inlet water, 85° F leaving water at a 78°F entering wet bulb temperature.

Conventional design practice calls for system components to be selected for the peak design heat rejection load in order to meet the building's comfort cooling requirements at all times. The efficiencies of each system component are typically specified independent of each other and the lowest first cost components meeting the full load specification are usually supplied for the project.

In recent years, there has been considerable debate as to the merits of designing around lower nominal condenser water flow rates in order to improve system lifecycle costs. Those favoring lower flow systems, typically 2.0 USGPM/ton, argue that there is a reduction in first (acquisition) costs. This claimed advantage is derived from lower acquisition costs associated with downsized pumps, pipe sizes, and cooling towers more than offsetting any increases in cost associated with the additional heat transfer surface being required in the chiller to meet the specified chilled efficiency. They also claim that the lower flow system can deliver improved operating costs as well as a reduction in energy usage by utilizing smaller pumps and cooling tower fan motors. They contend these savings more than offset the increase in power required by the chiller to overcome the greater pressure lift imposed by the higher condensing temperature.

Under this design scenario, a flow rate of 2.0 USGPM/ton increases the range to 15°F to maintain the same heat rejection rate. The "standard" cooling tower conditions become 100°F at the inlet, 85°F at the outlet when using the traditional 7°F approach and 78°F wet bulb. These higher cooling water temperatures lead to a higher condensing temperature and pressure in the chiller condenser. How much higher depends on what changes are made to the chiller, such as increased heat transfer surface (which increases chiller cost), to account for the increased temperature range across the condenser.

A discussion of condenser water flow rate optimization with respect to chiller system first cost, operating cost, or lifecycle cost cannot reasonably take place unless two additional parameters are taken into consideration: approach and design wet bulb temperature. Approach, the difference in temperature 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. This relationship can be seen in Figures 2, 3, and 4 below, which demonstrate how the cost, fan horsepower, and weight of a typical cooling tower selection vary with changes in the approach. In each of these graphs, the heat load is held constant (both flow and range) as was the entering wet bulb temperature. All data has been normalized to a typical selection for the nominal 7°F approach as well as smoothed to a single line for illustrated purposes. For comparison purposes, the lowest first cost selection was used for each thermal duty (approach) on all three graphs. Note on Figure 3 below that relatively small increases in the physical size of the tower (along with a corresponding increase in cost) can dramatically reduce the fan horsepower penalty of closer approach selections, often lowering the fan horsepower by 30% or more.

Note: Constant heat load (flow and range) and entering wet bulb were used for the cooling tower selection data used to develop the graphs shown in Figures 2, 3, and 4 below.

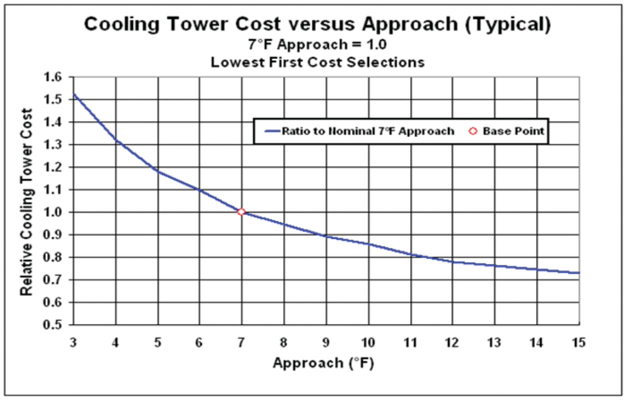

Figure 2

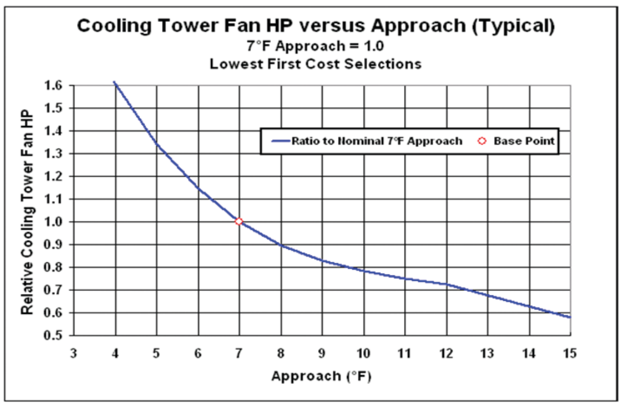

Figure 3

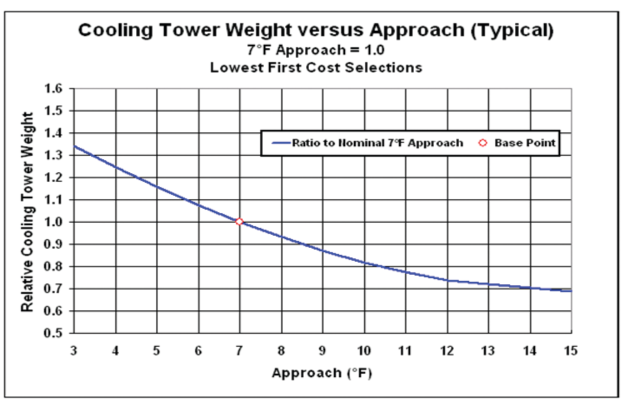

Figure 4

Traditionally, the HVAC industry has designed around approaches of 7°F or greater for two reasons. First, the majority of geographies are associated with design wet bulbs of 78°F and below, thus making the attainment of "standard" 85°F condenser water possible. Second, an industry mindset prevails which accepts 85°F 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 standpoint as well as from a full load energy standpoint at any given wet bulb?

This study was undertaken 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 water-cooled, centrifugal chiller system (note that equipment cost and performance data was obtained from Manufacturers in 2002). The manufacturers involved supplied one set of equipment selections and pricing data aimed at optimizing first cost at the expense of efficiency (a "Cheap but Inefficient" strategy, often mistaken for a "value engineering" strategy) and a separate set aimed at optimizing full load energy consumption at the expense of first cost (an "Efficient but Expensive" strategy). The following parameters and assumptions were used:

- Chiller evaporator conditions were fixed at 54°F entering and 44°F leaving temperatures with a maximum water-side pressure drop of 20 ft. w.c.
- Chiller condensers were all limited to a maximum water-side pressure drop of 20 ft. w.c.
- Condenser pumps were sized at a fixed system head of 60 ft. w.c.
- Piping costs and system pressure drops were not taken into consideration in this study (though we will revisit this assumption later in the paper).

First cost and energy data were then generated for a range of nominal condenser flows from 2.0 to 3.5 USGPM/ton in 0.5 USGPM/ton increments, cooling tower approaches from 3.0°F to 10.0°F in 1.0°F increments, and wet bulb conditions from 66°F to 84°F in 6°F increments. The cost and energy data used was the average for all manufacturers supplying data and are considered typical of both the market price and efficiency for this equipment. Lastly, ARI 550/590 and CTI certified performance ratings were used for the chillers and cooling towers respectively, which is a requirement of any study of this nature to ensure that all comparisons are based on truly verifiable and comparable performance data.

Optimized First Cost Selection Strategy

Under this scenario, equipment was selected on the basis of meeting full load capacity at the lowest possible first cost. Component equipment first costs for the condenser pumps, cooling tower, and chiller were plotted separately as a function of approach for all combinations of nominal flow rates and wet bulb temperatures. The sum of all component costs were then taken as the system cost as illustrated in Figures 5 and 6 below for the 78°F wet bulb case at 2.0 USGPM/ton and 3.0 USGPM/ton respectively:

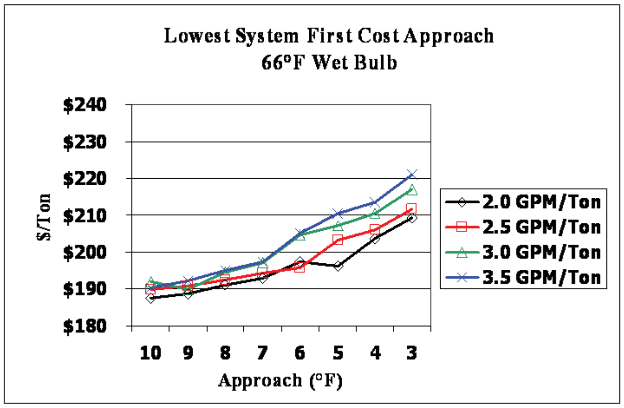

Figure 5

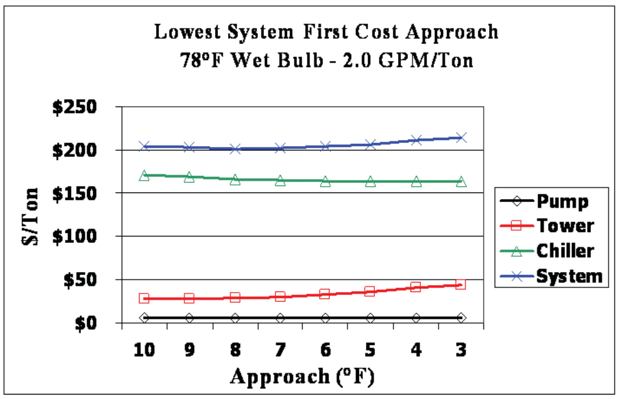

Figure 6

From the above graphs, it is evident that the chiller is the largest first cost component in the system. Note that the chiller cost/ton (shown in green on the graphs) is surprisingly independent of approach, unlike the cooling tower.

Removing the component detail allows the individual system first costs to be plotted as a function of nominal flow rate at a given wet bulb temperature. Plots of systems costs for 2.0 USGPM/ton through 3.5 USGPM/ton at 66°F, 72°F, 78°F, and 84°F entering wet bulbs are shown in Figures 7 through 10 below:

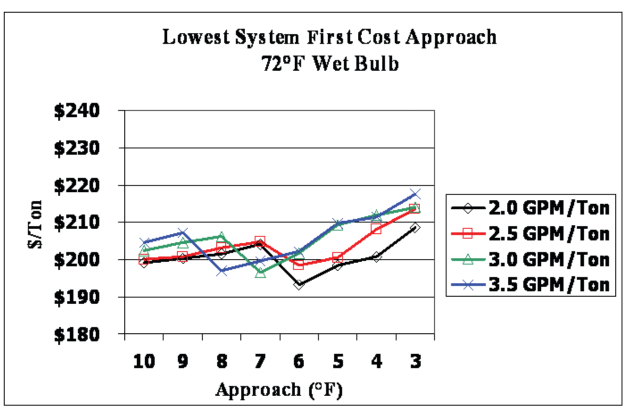

Figure 7

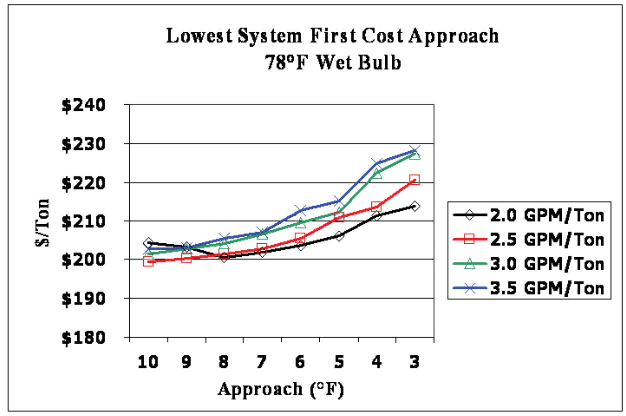

Figure 8

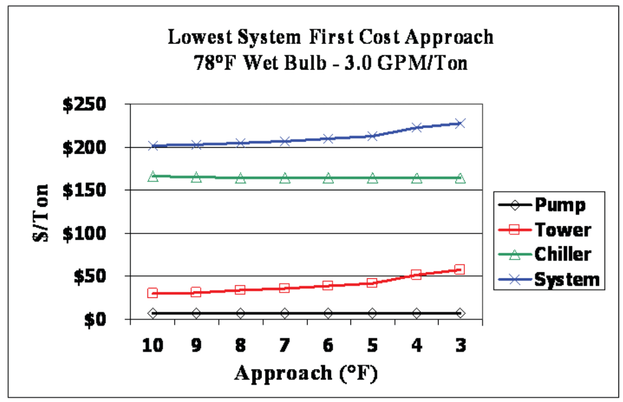

Figure 9

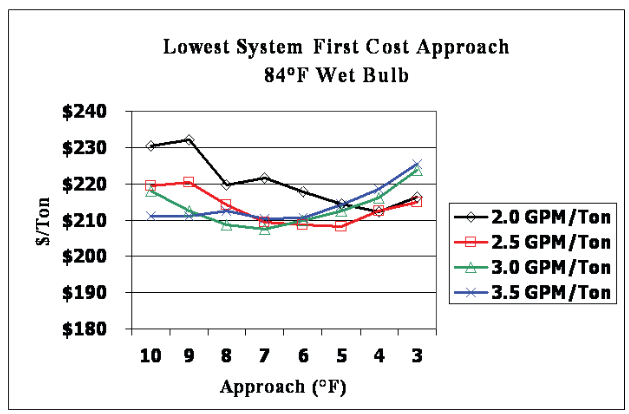

Figure 10

A review of these plots reveals that lower condenser water flows (2.0 to 2.5 USGPM/ton) and higher approaches (8°F to 10°F) are indeed viable means by which to optimize system first costs in all but the highest wet bulb environments, as shown on the graph for 84°F (Figure 10). The first cost advantage of the lower flow systems can be more pronounced when the material cost savings of smaller diameter condenser water piping is considered. However, for the 500-ton system that is the basis for this study, 8" pipe can be utilized for the 2.0 USGPM/ton (1,000 USGPM) through the 3.0 USGPM/ton (1,500 USGPM) cases. Therefore, piping costs would have been equal and thus have no effect on these particular results. The 3.5 USGPM/ton case would require the use of 10" piping, raising the first cost, but reducing pressure drop equal to or below that of the lower flow cases.

Photograph 1: Central Plant Equipment Room

Optimized Full Load Energy Selection Strategy

Under this scenario, equipment was selected on the basis of meeting full load capacity with the lowest possible energy consumption. In the case of both the chiller and cooling tower selections, this translated into equipment with significantly greater heat transfer surface than the equipment selected based on optimizing first costs. The full load energy consumption of the various system components were plotted as a function of approach for all combinations of nominal flow rates and wet bulb temperatures. Full load system energy consumption was taken as the sum of the full load energy consumption of all system components as illustrated in Figures 11 and 12 below for the 78°F wet bulb cases at 2.0 USGPM/ton and 3.0 USGPM/ton cases respectively:

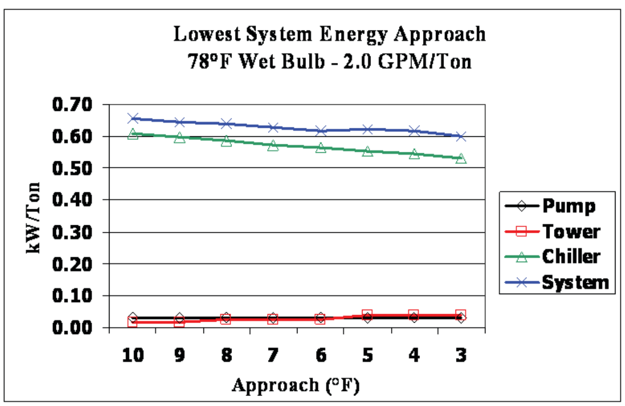

Figure 11

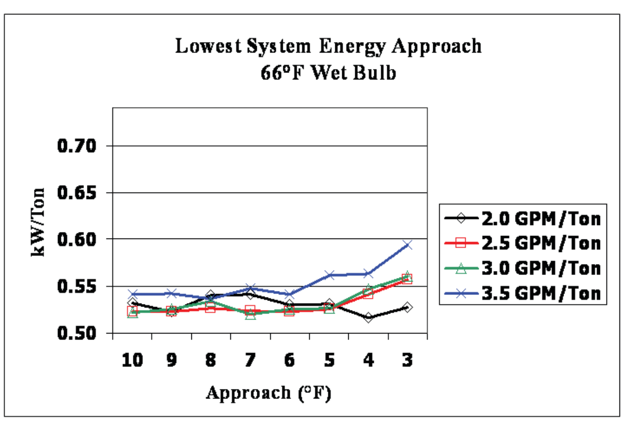

Figure 12

From the above plots, it is evident that the chiller consumes the largest amount of energy in the system by a wide margin. On average, the chiller accounts for 85% or more of the energy used by the system. However, unlike the first cost optimization case where chiller first cost was relatively flat versus condenser water temperatures, chiller energy consumption drops considerably with a decreasing approach. This results from the colder condenser water reducing the condensing pressure, which in turn reduces the lift, or work, that must be done by the compressor.

Plots of full load system energy consumption as a function of nominal condenser/cooling tower flow at 66°F, 72°F, 78°F, and 84°F are shown in Figures 13 through 16 below:

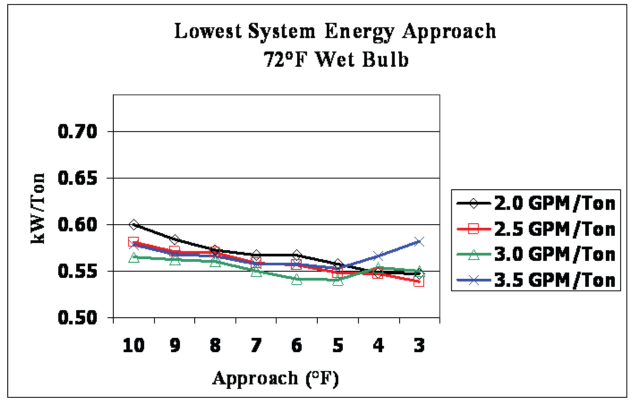

Figure 13

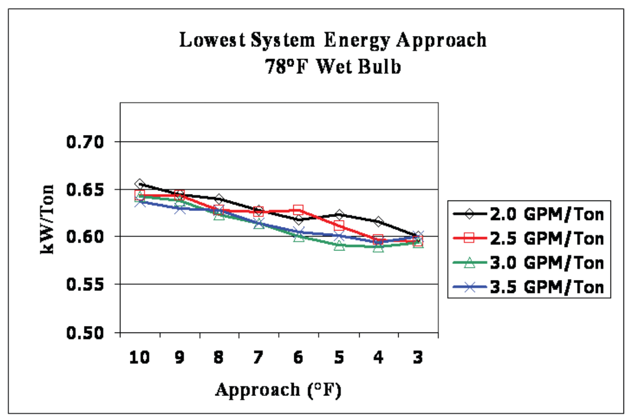

Figure 14

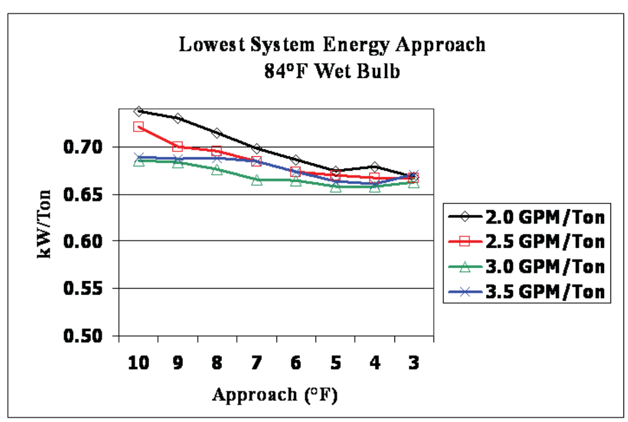

Figure 15

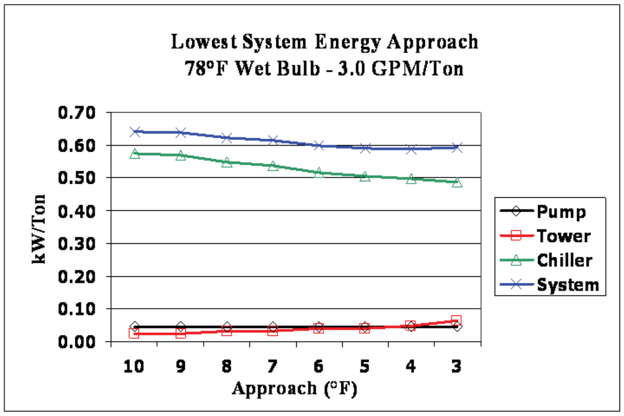

Figure 16

A summary of the optimized system energy (kW/ton) plots is revealing. The higher the entering wet bulb, the higher the required nominal condenser water flowrate and the tighter the approach needs to be in order to achieve optimum system energy usage. Only at the lowest wet bulb condition considered, 66°F (Figure 13), does a 2.0 or 2.5 USGPM/ton system show an energy advantage. At 72°F, the optimal energy balance occurs at 3.0 USGPM/ton with a 5°F approach. At 78°F wet bulb as well as at 84°F wet bulb, the optimal energy balance occurs at 3.0 USGPM/ton and an even tighter 4°F approach.

The advantage of a tighter approach in higher wet bulb environments is significant as illustrated in these plots. At 78°F wet bulb (Figure 15), the optimum system energy is achieved at 3.0 USGPM/ton with a 4°F approach as mentioned earlier. Had the "typical" 3.0 USGPM/ton, 7° F approach design been specified on this system, a system energy penalty of over 4% would have been realized. The additional fan energy required by the cooling tower is more than offset by chiller energy savings in all but the lowest wet bulb environments.

It is also interesting to note what little impact approach and condenser flow have on system energy in low wet bulb environments. At 66°F wet bulb, system energy is largely unchanged as the approach is tightened from 10°F to 5°F. The energy saved by the chiller when operating with 71°F versus 76°F entering condenser water temperature is being entirely offset by the additional energy consumed by the cooling tower fan in making the condenser water colder. In such a low wet bulb case, it would not be economically justified to invest in the larger cooling tower required to generate the 5°F approach.

As was the case with the "Optimized First Cost" scenario discussed earlier, it should be noted that the energy advantage of the higher flow systems would be reduced somewhat if the added condenser water pressure drop was taken into consideration. In our 500-ton example, 8" pipe is used for both the 2.0 and 3.0 USGPM/ton as both cases meet the ASHRAE friction guideline of 1 to 4 feet of pressure drop per 100 feet of pipe. The use of 8" pipe does result in a pressure drop increase for the higher flow system along with a consequent increase in system energy. However, 10" pipe could also have been used for the higher flow design as the velocity for 1,500 USGPM in an 8" pipe is close to the design limit (9.64 fps). This change would have raised the system first cost but lowered the condenser water pressure drop below that of the low flow system with 8" pipe, thus saving pump energy. The balance between piping costs and pump energy consumption, along with the length of the piping runs, should be taken into account in any actual system design.

Comparison between Low First Cost and Low Energy Approaches

Examining the two approaches together leads to the following question:

At what point is the added first cost of a more energy efficient system economically justified?

From the cost data in the study, the first cost premium and energy savings of the optimum low energy over the low cost solutions can be plotted versus wet bulb temperature. This is done by taking the optimum kW/ton and corresponding cost for the low energy solution minus the optimum kW/ton and cost for the low first cost solution to obtain both the energy usage and first cost difference at each of the four wet bulbs examined. Note that all these values are for a nominal 500 ton water-cooled, centrifugal chiller system, which is the basis for this study.

These values are shown in Figure 17 below:

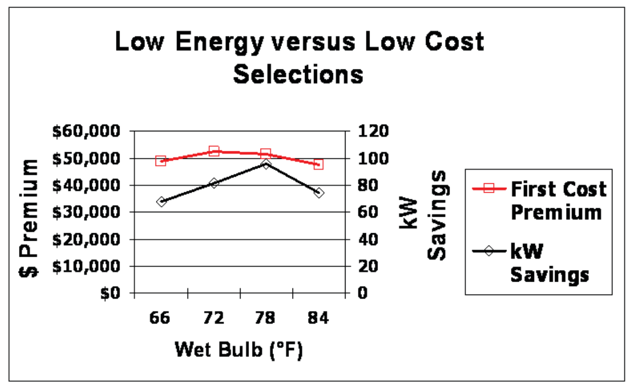

Figure 17

From Figure 17, we can calculate the first cost premium per saved kW for the lowest energy system, which is plotted in Figure 18 below:

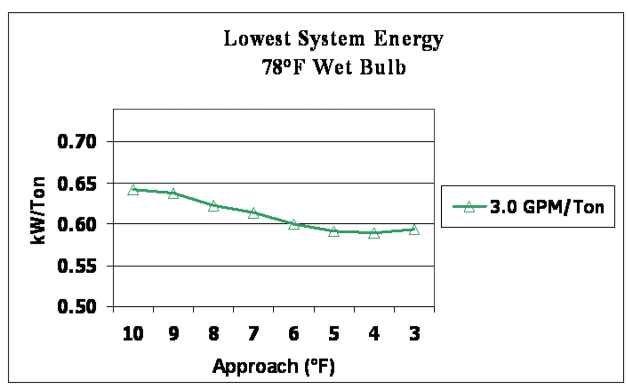

Figure 18

To establish the payback, the cost of electrical power must be determined. In this example, a typical demand charge of $12/kW-Month for six months and a power cost of $0.10/kWh was assumed, along with 3,000 equivalent full load hours (to account in part for off-peak operation):

Equivalent Full Load Hours Estimate

Demand: 1 kW x $12/kW-Mo x 6 Months =	$ 72/Year
Energy: 1 kW x 3,000 hours x $0.10/kWh =	$300/Year
Total Annual Value:	$372/Year

This would result in premium thresholds of $744 and $1,116 for a two and a three year simple payback respectively. From Figure 18, the premium for the lowest energy system at all wet bulb temperatures are below the two year threshold, which is an attractive payback for most projects.

The cost premium for the lowest energy system versus the conventional 7°F approach can also be examined. Using the 78°F wet bulb chart for lowest system energy (Figure 15), the flow rate that results in the lowest full load energy usage can be selected, which in this case is 3.0 USGPM/ton, as shown in Figure 19 below:

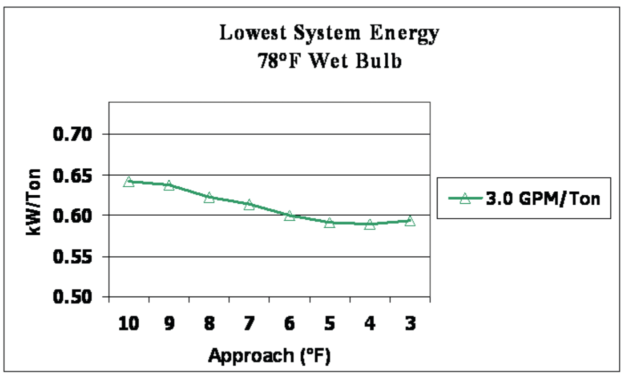

Figure 19

The lowest energy point occurs at an approach of 4°F as shown in graph above (Figure 19). The cost premium and energy savings of low approach selections versus the traditional 7°F approach for our 500-ton system are plotted in Figure 20 below:

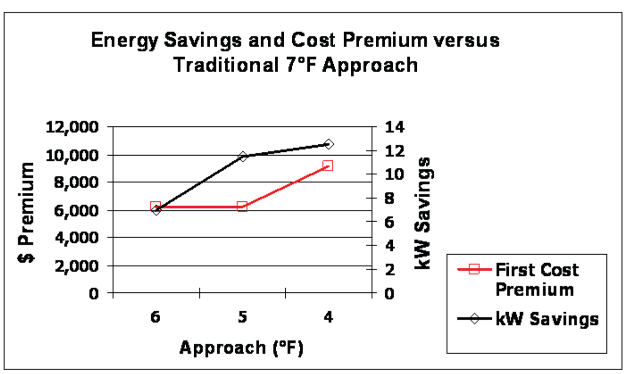

Figure 20

From Figure 20, the first cost premiums per kW can be calculated, which is plotted in Figure 21 below:

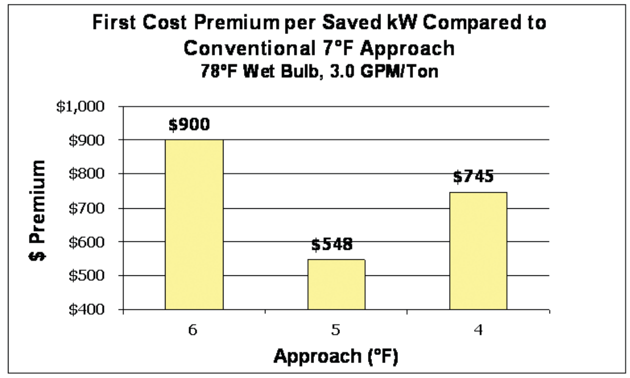

Figure 21

Based on the value of $372/kW established earlier, we can calculate simple paybacks of 2.4, 1.5, and 2.0 years respectively for 6°F, 5°F, and 4°F approaches as compared to designing for the traditional 7°F approach. These are attractive paybacks for the investment and would help to soften the impact of any increases in the price of electricity in the future.

Summary and Conclusions

A number of conclusions can be drawn from this study:

- Lower flow (2.0 USGPM/ton) condenser water systems generally have first cost advantages over higher flow (3.0 USGPM/ton) systems in all but the highest wet bulb environments.
- Higher flow (3.0 USGPM/ton) condenser water systems generally have full load energy advantages over lower flow (2.0 USGPM/ton) systems in all but the lowest wet bulb environments.
- Approaches in the range of 4°F to 5°F offer significant full load system energy advantages over “traditional” 7°F approaches in environments with higher wet bulbs (78°F and above).
- While low flow systems can reduce first cost, the energy savings derived from higher flow, close approach designs offer improved lifecycle costs for the building owner with attractive paybacks for the additional investment.

A side benefit of the higher flow, closer approach designs is that the cooling towers are larger, often with lower horsepower fan motors. This can lead to additional advantages for the designer and the building owner:

- More annual hours of waterside economizer operation, where system operators utilize the cooling tower to produce chilled water in lieu of running the chiller, enabling further energy savings.
- Lower tower sound levels in those instances where the fan motor horsepower is lower, which is an important consideration on many projects.

Based on the results of this study, system designers should not take a “one size fits all” strategy, but instead evaluate the merits of each system based on the design load, load profile, and local ambient conditions for the building. While system optimization can add time and cost to the design process, the resulting advantages for the building owner and society in general in terms of energy efficiency and reduced environmental impact are justified. The influence of other variables, such as compressor type, variable speed capabilities for both the compressor and tower, and off-peak loading should also be considered. Many chiller manufacturers offer system software programs that can assist in these evaluations. Lastly, control strategies to minimize system energy consumption under all weather and load conditions, not just at design conditions, should be considered in any system design.

Photograph 2: Crossflow Cooling Tower Installation

End Note:

The authors would like to thank the chiller, tower, and pump manufacturers who supplied the selection, cost, and energy data that made this study possible.

References

2 GPM/Ton Condenser Water Flow Rate – Does It Save Energy?, Kirsner Consulting Engineering, February 6, 2002

3 GPM/Ton Condenser Water Flow Rate: Does It Waste Energy?, Wayne Kirsner, P.E., ASHRAE Journal, February 1998, Pages 63 through 69

Can Improving Off-Design Chiller Performance Be as Easy as Redesigning a Chart?, HVAC&R Engineering Update, York International Corporation, 1999

Central Cooling and Heating (Chapter 4), Condenser Water Systems (Chapter 13), and Cooling Towers (Chapter 36), 2004 ASHRAE HVAC Systems and Equipment Handbook, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc., 2004

Chiller Plant Control, Michael Bitondo and Mark Tozzi, Carrier Corporation, August 1999

Chiller-Plant Energy Performance, HVAC&R Engineering Update, York International, downloaded July 20, 2004 from: http://www.york.com/products/esg/updates/eng-Updates/3.pdf

Evaluation of Low Condenser Water Flow in Chiller Systems, HVAC&R Engineering Update, York International Corporation, downloaded July 20, 2004 from: http://www.york.com/products/esg/updates/eng-Updates/22.pdf

Operating Chiller in the Max-Cap Range, Gil Avery, Engineered Systems, June 2003, Pages 66 through 72

Optimizing Chillers & Towers, Hugh Crowther and James Furlong, ASHRAE Journal, July 2004, Pages 34 through 40

Tower Water Temperature – How to Control It, Mick Schwedler and Brenda Bradley, Trane Corporation, downloaded July 21, 2004 from: http://www.trane.com/commercial/library/vol241/v24a1.asp

Why Change the Chilled Water Temperature Range?, Hugh Crowther, McQuay International, Engineering Systems Solutions, December 2002