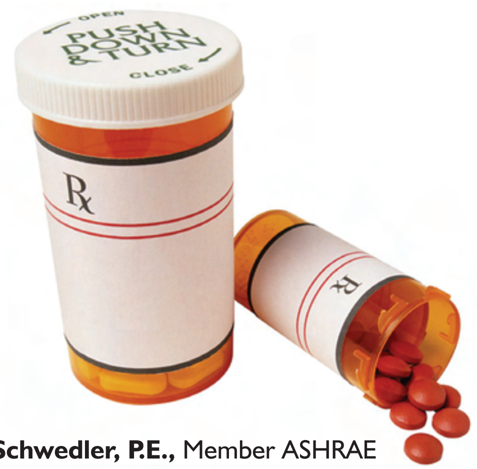
Prescription FOR Chiller Plants



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The growth of medical facilities has been accelerated by the move to managed care and outpatient services, increased competition, and the desire to create patient-friendly environments. Continually changing technology demands maximum flexibility in facility layouts and support systems. These pressures, coupled with limited time and resources, can lead to ad hoc expansion.

The staff at Winchester Medical Center, which is a subsidiary of Valley Health System (VHS) in Virginia, adopted a more deliberate approach. The campus consists of a large hospital building that includes patient and operating rooms, diagnostic facilities, administrative areas, and a clinic. The VHS Master Plan includes the addition of at least two separate medical office buildings by 2008.

With no forecasted increase in staffing, the expanded services must perform reliably and minimize operating costs, but also maximize efficiency and flexibility. It was clear that a new central chilled water plant would best meet these requirements.

This article traces the evolution of the plant's complex design, from definition of requirements through initial startup. It also highlights the challenges of system integration and the value of an effective project team.

Objectives for the New Plant

Together, the design engineer and Mark Baker, VHS director of facilities and construction, identified the following requirements for the new plant:

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- **42°F (5.5°C) supply, 58°F (14°C) return**. These temperatures provide 50°F (10°C) supply air for the operating rooms and sufficient cooling for the advanced medical equipment while accommodating the extended piping loop. Past "rule of thumb" flow rates and temperatures would have resulted in significantly larger, and more costly, piping, with accompanying higher pump power.
- **Interoperability**. It was important to provide an infrastructure that would not only permit information-sharing between the new chilled water plant and the existing facility but also accommodate future changes in technology. Specifying BACnet®, LonWorks®, and Modbus® protocols would allow the facility management system (FMS) to gather all of the necessary data to optimize chilled water generation. It also would allow the FMS to communicate with the heating system, engine generator, and other non-HVAC systems residing in the plant (*Figure 1*). As a result, operators can monitor and control all of this equipment from a single user interface.
- **Direct digital control**. To maximize its long-term value, the specifications for the FMS were patterned after ASHRAE Guideline 13-2000, *Specifying Direct Digital Control System*.
- **Refrigerant safety**. The plant includes provisions for monitoring, alarming, safety, and evacuation in accordance with ANSI/ASHRAE Standard 15-2001, *Safety Standard for Refrigeration Systems*.
- **Cost-effectiveness**. VHS committed to purchase "the most efficient chilled water plant possible."

Fleshing Out the Design

Using a spreadsheet created by Baker, which detailed the anticipated growth of the campus through 2008, the designer established plant requirements for each stage of the expansion. Total plant capacity is based on $N+1$, which provides a redundant chiller, cooling tower cell, chilled water pump, and condenser water pump.

The decision to site the plant in a remote part of the campus increased costs for underground piping but maximizes the area available for new facilities. The distance between the plant and the hospital also required special attention to ensure delivery of the desired supply-water temperature. The chilled water piping, constructed of ductile iron, passes through an 800 ft (240 m) thermal trench. Calcium-carbonate particles provide an insulative barrier no less than 4 in. (100 mm) thick. Strategically positioning the valve boxes within the thermal trench simplifies future expansion.

Condenser water for the chillers is provided by cooling towers, which are placed directly *behind* the plant to minimize noise transmission to the rest of the campus.

The designer also sized a 1400 kW generator and switchgear to operate two chillers, two chilled water pumps, two condenser water pumps, and two cooling tower cells. This addition provides the facility with the ability to shed electrical load and allows the hospital to operate almost normally during power outages. (Many of the air handlers and terminal units already are connected to an essential power source.)

Weighing Alternatives

The design team investigated various system level choices before settling on a final design. (All but the last option, condenser heat recovery, was incorporated.)

- **Variable primary flow** can reduce the costs of installing and operating the chilled water system because it requires fewer pumps.[1-15] Implementation requires the addition of a bypass line and two-way modulating valve. Interfacing with the hospital's existing pumps posed a control challenge (discussed later) that may resurface when buildings are added.
- **Water-side economizing** can provide inexpensive cooling when load and outdoor wet-bulb temperature are low. As with many health-care facilities, Winchester Medical Center often operates at these conditions, so a waterside heat exchanger was added to handle small loads. Two of the new chillers also were equipped for compressor-less free cooling, which uses refrigerant migration to produce chilled water. Leaving tower water temperature determines the cooling capacities of the economizer and refrigerant-migration cycles. At 30°F (–1°C) ambient wet-bulb temperature and below, with no compressors online, the plant produces chilled water at a total kW/ton of approximately 0.2 or less (0.7 kW/kW or less).
- **Variable-speed operation**, at the designer's discretion, was required for *all* of the plant's dynamic components. Therefore the chillers, cooling tower fans, and pumps were provided with premium efficiency, inverterduty motors and controlled with individual variable-frequency drives.

Two of the medical center's three existing chillers already included variable-speed drives (VSDs). Valley Health System decided to purchase two more chillers with VSDs and use the remaining constant-speed chiller as a backup.

- **Chiller selection** was based on the design parameters summarized in *Table 1,* with each machine selected to produce 750 tons (2638 kW) of cooling. At these conditions (42°F [5.5°C] chilled water, 84°F [29°C] entering tower water), ANSI/ASHRAE Standard 90.1-2001, *Energy Standard for Buildings Except Low-Rise Residential Buildings,* and Virginia require a minimum full-load efficiency of 0.590 kW/ton (2 kW/kW). Full-load performance is 0.594 kW/ton (21 kW/kW) for each of the three existing chillers and 0.571 kW/ton (2 kW/kW) for both of the new chillers.

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Performance			Evaporator				Condenser			
Chiller*	FLA	kW/ton	Flow Rate, gpm	Entering Water	Leaving Water	Pressure Difference ft H_2O	Flow Rate, gpm	Entering Water	Leaving Water**	Pressure Difference, ft H_2O
1	618	0.571	1,125	58°F	42°F	14	2,250	84°F	95°F	22
2	618	0.571	1,125	58°F	42°F	14	2,250	84°F	95°F	22
3	643	0.594	1,125	58°F	42°F	29.9	2,250	84°F	95°F	28
4	643	0.594	1,125	58°F	42°F	29.9	2,250	84°F	95°F	28
5	643	0.594	1,125	58°F	42°F	29.9	2,250	84°F	95°F	28

* Each chiller has a nominal capacity of 750 tons. Chiller 5 has a constant-speed drive; the others are equipped with variable-speed drives. Chillers 3–5 were part of the existing chilled water plant.
** At the given conditions the chiller condenser leaving water temperature would be 93.4°F. The design engineer specified the conditions shown. This gives a safety factor, for example, for those times when the ambient wet bulb is higher than the ASHRAE design conditions.

Table 1: Chiller design parameters.

Pump Application	Flow Rate, gpm	Head, ft H_2O	Power, hp	Efficiency	Speed, rpm
Chilled Water (5 Pumps)	1,125	120	50	81%	1,750
Condenser Water (5 Pumps)	2,250	75	60	85%	1,150

Table 2: Pump design parameters.

- **Chilled water pumps** were sized equally, equipped with VSDs, and manifolded together to allow use of any pump with any chiller. *Table 2* shows the design parameters.

Although the pumps are identical, the pressure difference across the evaporators is not. The resulting flow imbalance is discussed later in this article (see the chillers section).

- **Condenser water pumps** also are sized equally and equipped with VSDs. For normal operation, each pump is dedicated to a specific chiller. Manual valves were added so that each pump can operate with a different chiller, if needed. See *Table 2* for design parameters.
- **Cooling towers** were sized equally and a VSD was provided for each cooling tower fan. *Table 3* summarizes the design parameters.
- **Condenser heat recovery** was dismissed as an option due to the plant's remote location. With makeup water already available at the hospital for domestic hot water, it was impossible to justify the cost of piping, trenching, insulation, etc.

Climbing the Learning Curve

With the basic design of the plant established, the design engineer and plant owner worked closely with the providers to understand the interactive effect of variable-speed operation on all pumps, tower fans, and chillers. Hartman[6] discusses such plants but does not describe the specific method for plant control.

Flow Rate, gpm	Water Temperature		Outdoor Wet Bulb	Fan (5)	
	Entering	Leaving		hp	rpm
2,250	95°F	84°F	76°F	40	1,800

Table 3: Cooling tower design parameters.

Water Flow Rate	Range	Approach Temp.	Tower Water Temp.	
			Leaving	Entering
100%	6.0°F	9.0°F	64.0°F	70.0°F
75%	8.0°F	7.8°F	62.8°F	70.8°F
60%	10.0°F	6.7°F	61.7°F	71.7°F

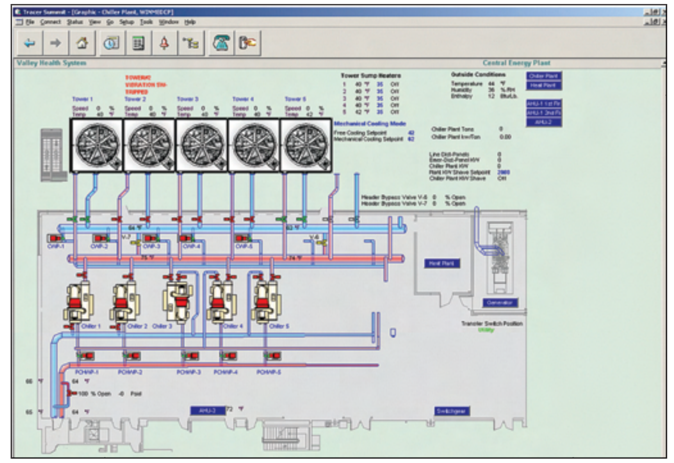
Table 4: Cooling tower performance at 50°F WB and 60% load.

Condenser Water Pumps

VSDs on condenser water pumps pose a particular optimization challenge. Pumping less condenser water obviously reduces pump power, but the potential savings depend on the available reduction in pressure drop. Although a lower flow rate reduces the pressure drop through the chiller's condenser bundle, piping, and valves, the cooling tower static lift remains constant. It's also important to balance pump savings with the effect of reduced condenser water flow on the performance of the chiller and cooling tower.

Cooling Towers: Performance With Variable Water Flow

Obtaining relevant performance data for the cooling tower was difficult. Unlike fan speed, variable water flow is uncommon enough that few selection programs model it. *Figures 2* and *3* show the data provided by the tower manufacturer. Note that the maximum range is 10°F (though the selection specified



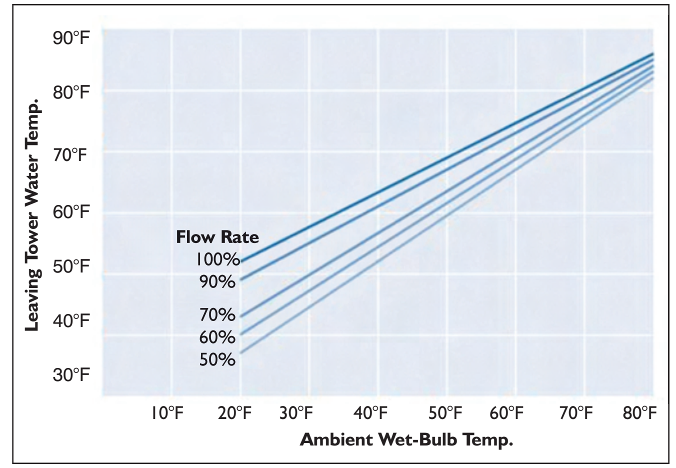


Figure 1 (left): FMS summary of the chilled water plant. Figure 2 (right): Cooling tower performance with variable flow and a 10°F design range.

11°F), and all three charts are based on full-speed operation of the tower fans. Would this data predict, with sufficient accuracy, the effect of flow rate on tower performance? The following example was devised to discover this.

Table 4 shows how flow rates of 100%, 75%, and 60% of design affect tower performance when the outdoor air is 50°F wet bulb and the heat-rejection load is 60%. (Remember that this data is based on a 10°F design range and *full-speed* fan operation.) Each reduction in flow rate ultimately lowers the cooling tower approach due to warmer entering water and the resulting increase in log mean temperature difference.

By itself, reducing the condenser water flow rate under these conditions is advantageous. But as discussed under the chillers section, that conclusion oversimplifies the chiller–tower–pump relationship. The accompanying increase in the cooling tower's entering water temperature causes the chiller to consume more energy. *Table 5* demonstrates how reduced flow rates affect tower performance if the heat-rejection load is 80% and the ambient air is 70°F (21°F) wet bulb. Once again, the result is a closer approach and a higher cooling tower entering water temperature.

The design team gleaned two lessons from their investigation. When the tower fans operate at full speed:

- Reducing the water flow rate improves *cooling tower* performance; and
- The amount of heat rejection determines the minimum range. (Larger heat-rejection loads require a higher percentage of the design flow rate.)

Cooling Towers: Performance With Variable Water Flow and Airflow

Evidence shows that when the condenser water flow rate is constant, there is an optimal tower-fan speed at which the reduction in fan power exceeds the accompanying increase in chiller power.[16–19] However, does such an operating point exist when both water flow rate *and* fan speed are reduced? By this point in the project, the tower manufacturer was no longer in business, making it impossible to obtain the information needed to arrive at a definitive answer.

Water Flow Rate	Range	Approach Temp.	Tower Water Temp.	
			Leaving	Entering
133%*	6.0°F	Not Available	Not Available	Not Available
100%	8.0°F	9.8°F	79.8°F	87.8°F
80%	10.0°F	8.0°F	78.0°F	88.0°F

*Mathematically, heat rejection for a 6°F range requires 133% of the design flow rate.

Table 5: Cooling tower performance at 70°F WB and 80% load.

Chillers

To reject the same amount of heat at lower flow rates, the range must increase—which means that warmer water must enter the tower. Since the water temperature entering the tower is identical to that leaving the condenser, reducing the condenser water flow rate raises the leaving condenser water temperature and causes the chiller to consume more power. Given this relationship, reducing the condenser water flow rate does not provide an economic benefit unless the savings in pump power exceeds the increase in chiller power.

As for the VSDs on the chillers (all but the backup, which is constant-speed), reduced speed does not provide an economic benefit unless the lift between the evaporator and condenser also is reduced. Operationally, unless the lift is reduced, the drive may need to remain at full speed to avoid surge. Leaving chilled water temperature sets the evaporator pressure, so any reduction of lift requires a lower leaving condenser water temperature. However,

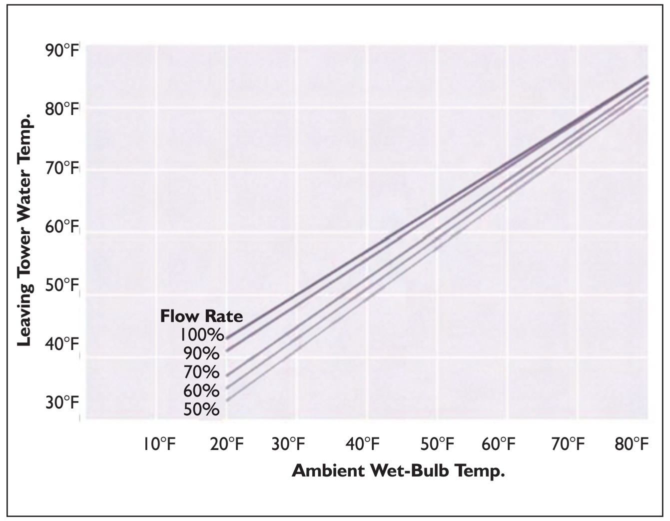


Figure 3: Cooling tower performance with variable flow and a 6°F design range.

reducing the speed of the cooling tower fans and/or condenser water pumps raises the leaving condenser water temperature.

There were operational considerations to address. For example, it was important to avoid creating an unstable operating condition, such as excessive condenser pressure at low load, that would cause the compressor to surge.

It was also necessary to contend with the unequal pressure differences of the two-pass and three-pass evaporators *(Table 1):* 14 ft w.g. (42 kPa) for Chillers 1 and 2 vs 29.9 ft w.g. (90 kPa) for Chillers 3, 4, and 5. Since chilled water pumps are dedicated, each pump could have been selected specifically for its chiller. To simplify pump interactions, identical pumps were chosen. Although the design engineer specified balancing valves to equalize the chilled water flow, the owner chose not to incur the pressure drop penalty at all load conditions and opened the valves.

Opening the balancing valves creates a flow imbalance that could make it difficult for the plant to supply water that is cold enough. Chillers 1 and 2 receive more flow than Chillers 3, 4, and 5. When flow rates equalize at 19.9 ft (60 kPa) pressure drop, 1,336 gpm (84 L/s) will pass through Chiller 1 and 914 gpm (58 L/s) through Chiller 4. (More-than-design flow through Chillers 1 and 2 may elevate the leaving water temperature.) Decreasing the chilled water setpoint for Chillers 3 and 4 would offset the imbalance by loading these machines comparably to their loads at reduced flow. However, this strategy was not used because it further complicated plant control.

Control: Chiller Sequencing

The control decisions made during the final design and at system startup established a chiller sequencing strategy that:

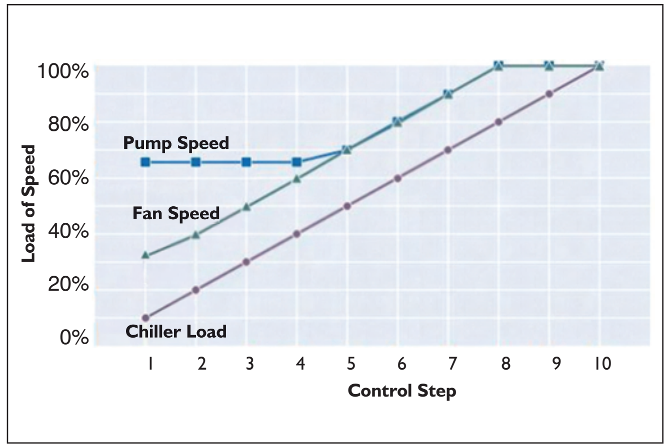


Figure 4: Comparison of chiller load to speed of chilled water pump and cooling tower fan.

- Adds a chiller when the system supply water temperature exceeds setpoint by 2°F (1°C) for 15 minutes;
- Starts a condenser water pump and a cooling tower cell whenever a chiller is added;
- Turns off a chiller if the resulting load on the chillers still operating is less than 85%; and
- Will not stop a chiller for at least 30 minutes after the last chiller startup to avoid cycling the machines on and off too rapidly.

The design pressure drop differences between chillers have not caused problems using this sequencing method.

Plant control for the condenser water pumps and cooling tower fans is based on chiller load. When the chiller load, calculated using chilled water flow and temperature difference, is less than 80%, the facility management system varies the speeds of the cooling tower fan and condenser-water pump in proportion to chiller load *(Figure 4)*. This strategy may not be ideal, but it balances the power of the chiller, tower fan, and condenser water pump while avoiding the points at which the chiller might surge. When the cooling tower water is extremely cold, the condenser water pump VFDs are controlled to maintain the minimum evaporator-to-condenser pressure differential required by the chiller manufacturer.

The speed of the chilled water pumps at the plant maintains a 5 psi (35.5 kPa) pressure differential (chilled water supply-to-return) at the existing hospital, where local pumps circulate water through the facility. This arrangement provides the campus with a variable primary/variable secondary control strategy. Differential pressure sensors monitor the chilled water flow rate across each chiller's evaporator. Using data from the chiller manufacturer, the differential pressure signal is converted to flow rate.

Data Acquisition

The electric energy consumption data were obtained from the variable speed drives. Some feel that more accurate measurement techniques may be required. ASHRAE Guideline 22, *Instrumentation for Monitoring Central Chilled Water Plant Efficiency,* is being written and likely will give information on this subject.

Initial Startup

The team encountered several challenges at initial startup. Most notable was the discrepancy between the two flow meters in the system. Although calibration brought their readings closer together, no one knows which (if either) of the meters is more accurate.

Also, it became apparent that chiller evaporator flow was miscalculated during design. This error resulted in a discrepancy between estimations of individual chiller loads (which are based on chiller flow rates as described earlier) and system load (which is based on flow meter readings). After the algorithm was corrected, chiller and system capacities closely paralleled each other.

Figure 5 graphs system operation on Aug. 8, 2003. Ambient wet-bulb temperatures ranged from 64°F to 72°F (18°C to 22°C) that day. Although the cooler-than-design wet-bulb temperatures helped reduce the plant's energy use, the 0.6 kW/ton (2.1 kW/kW) average is impressive.

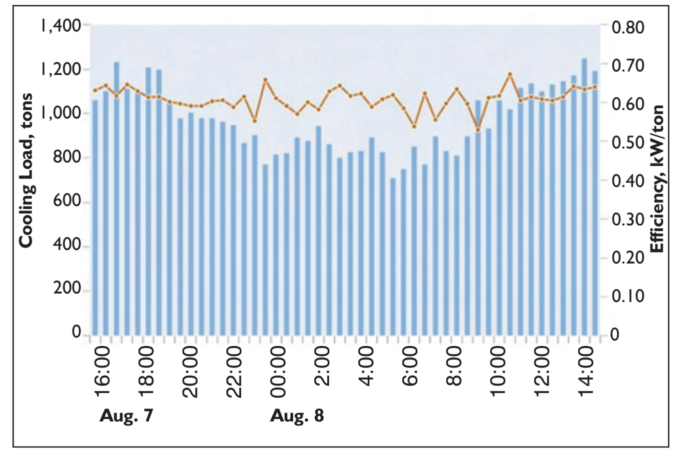


Figure 5: Summary of chilled water plant performance for Aug. 7–8, 2003.

Fine-Tuning Operation

Soon after initial startup, the building owner, plant operators, consulting engineer, and providers of the chilled water plant and controls met on-site to discuss ways to optimize system performance.

For example, not starting another chiller unless the chilled water temperature exceeded setpoint by 2°F (1°C) for 15 minutes resulted in a loss of cooling in critical areas of the hospital. Reducing the deadband to 1°F (0.6°C) for 15 minutes appeared to mitigate that problem.

Also, the number of operating chilled water pumps was increased as a means of reducing pump power (e.g., now three—not two—chilled water pumps operate in conjunction with two chillers). Some might argue that this strategy can't reduce pump power because power is proportional to flow and pressure difference. These parameters don't change whether two or three pumps operate. However, bringing an additional pump online *distributes flow through more fittings,* reducing operating pressure and, therefore, pump power.

Similarly, the team considered increasing the number of operating cooling tower cells from *N,* the number of operating chillers, to *N* +1. Based on examination of the plant data, it was estimated that increasing the available heat-exchange surface could decrease system power by 0.03 to 0.05 kW/ton (0.1 to 0.18 kW/kW). Of course, the challenge is to ensure that each cell receives its minimum flow. (This adjustment was not implemented during the 2003 cooling season.)

Figure 6 graphs system operation on Aug. 24–31, one week after implementing the agreed-upon adjustments. At first glance, it appears that the plant performed poorly on Aug. 25 (i.e., power use exceeded 2.5 kW/ton [8.8 kW/kW]). Closer examination of the data and system revealed the failure of a pump sensor. The faulty sensor registered almost 1400 kW when, in fact, the pump was off. Readings returned to normal when the sensor was replaced. Prompt diagnosis was made possible by the facility management system's data-rich, graphical environment. With the sensor working properly, plant performance ranged from 0.50 to 0.60 kW/ton (1.8 to 2.1 kW/kW)—efficient operation indeed.

Project Prognosis

The most significant hurdle in the Winchester Medical Center project was integrating the FMS with the heating and electrical generation systems, electrical switchgear and circuit breakers, and variable-speed drives. The challenge lay not in the capability of the FMS, but in obtaining information and support from the electrical equipment provider, and in establishing the communications interface between the FMS and the heating plant. In each case, the owner perceived an large gap between the integration capability he'd paid for and what he received.

As an example, although the specification required a communication interface between the heating plant and the FMS, it was necessary to hardwire the control points. Although the work was done well and provides the required data exchange, the owner questions whether he received the benefit he expected from the heating system provider.

To avoid such credibility gaps and facilitate system integration, the specification must explicitly define *what data* will be communicated, the *method of exchange,* and *who will supply* the necessary hardware, front-end graphics and system control software.

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A Clean Bill of Health

Winchester Medical Center's FMS gathers data from all plant components, including:

- Power use of all chiller compressors, water pumps, and cooling tower fans;
- Condenser-water temperature;
- Chilled water flow; and
- Speed from all variable-speed drives.

Programming allows the FMS to analyze the information based on outdoor conditions and to control the components as an efficient "team" so that the plant, as a whole, uses the least possible power to generate each ton of chilled water.

This approach not only minimizes the medical center's operating expense; it also benefits the environment. Minimizing the use of electrical energy lessens the amount of power that must be generated, which in turn decreases power plant emissions.

This project's successful outcome is largely attributable to the active involvement of the same team throughout planning, design, installation, and start-up. Team members invested considerable time in on-site visits and communication. Great care was taken to explain and understand all options under consideration, and to integrate the infrastructure, equipment, and controls into a cohesive system. Mutual trust and respect made team interactions more effective. The highly integrated, highly efficient plant that now serves Winchester Medical Center demonstrates the value of healthy, synergistic teamwork.

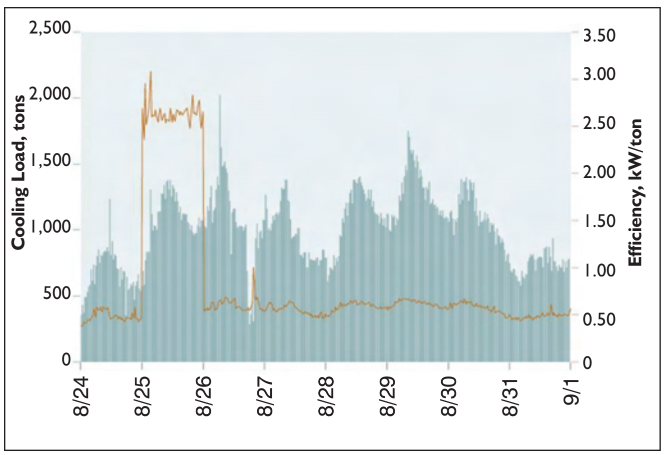

***Figure 6:** Summary of chilled water plant performance for Aug. 24–31, 2003.*

Acknowledgments

No project of this scope can succeed without operators and technicians who willingly perform the day-to-day work necessary to run and optimize the plant. The authors thank Paul Otworth and George Sloane, whose insights and willingness to innovate streamlined the operation of this cutting-edge plant. Ronnie Mitchell also deserves special recognition for his tireless programming and calibration efforts; his extensive knowledge of chilled water and automation systems benefited the entire team.

References

1. Avery, G. 2001. "Improving the efficiency of chilled water plants." *ASHRAE Journal* 43(5):14–18.

2. Bahnfleth, W.P. and E. Peyer. 2001. "Comparative analysis of variable and constant primary-flow chilled-water-plant performance." *Heating/Piping/Air Conditioning Engineering* 73(4):41–50.

3. Bynum, H. and E. Merwin. 1999. "Variable flow: A control engineer's perspective." *ASHRAE Journal* 41(1):26–30.

4. Coad, W.J. 1998. "A fundamental perspective on chilled water systems." *Heating/Piping/Air Conditioning Engineering* 70(8).

5. Groenke, S. and M. Schwedler. 2002. "Series–series counterflow for central chilled water plants." *ASHRAE Journal* 44(6):23–29.

6. Hartman, T. 2001. "All-variable-speed centrifugal chiller plants." *ASHRAE Journal* 43(9):43–51.

7. Houghton, D. 1996. "Know your flow: A market survey of liquid flow meters." *E SOURCE Tech Update* TU-96-3.

8. Kirsner, W. 1996. "The demise of the primary–secondary pumping paradigm for chilled water plant design." *Heating/Piping/Air Conditioning Engineering* 68(11).

9. Kreutzmann, J. 2002. "Campus cooling: Retrofitting systems." *Heating/Piping/Air Conditioning Engineering* 74(7):27–34.

10. Luther, K. 1998. "Applying variable volume pumping." *Heating/Piping/Air Conditioning Engineering* 70(10).

11. Schwedler, M. and B. Bradley. 2003. "Variable primary flow in chilled-water systems." *Heating/Piping/Air Conditioning Engineering* 75(3):37–45.

12. Schwedler, M. and B. Bradley. 2000. "Variable-primary-flow systems: An idea for chilled-water plants the time of which has come." *Heating/Piping/Air Conditioning Engineering* 72(4):41–44.

13. Taylor, S.T. 2002. "Degrading chilled water plant delta-T: Causes and mitigation." *ASHRAE Transactions* 108(1):641–653.

14. Taylor, S.T. 2002. "Primary-only vs. primary-secondary variable flow systems." *ASHRAE Journal* 44(2):25–29.

15. Waltz, J.P. 1997. "Don't ignore variable flow." *Contracting Business* 54(7).

16. Braun, J.E. and G.T. Diderrich. 1990. "Near-optimal control of cooling towers for chilled-water systems." *ASHRAE Transactions* 96(2):806–813.

17. Cascia, M.A. 2000. "Implementation of a near-optimal global set point control method in a DDC controller." *ASHRAE Transactions* 106(1):249–263.

18. Hydeman, M., K. Gillespie, and R. Kammerud. 1997. "PG&E's CoolTools Project: A toolkit to improve evaluation and operation of chilled water plants." Cool $ense National Forum on Integrated Chiller Retrofits, Lawrence Berkeley National Laboratory and Pacific Gas & Electric.

19. Schwedler, Mick. 1998. "Chiller/tower interaction: Take it to the limit—or just halfway?" *ASHRAE Journal* 40(7):32–39.