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Optimization of Variable Speed Chiller Plants:

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The Green Proving Ground (GPG) program leverages GSA's real estate portfolio to evaluate innovative sustainable building technologies and practices. Findings are used to support the development of GSA performance specifications and inform decision-making within GSA, other federal agencies, and the real estate industry. The program aims to drive innovation in environmental performance in federal buildings and help lead market transformation through deployment of new technologies.

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I. Executive Summary

This report is divided into five sections. The first section describes the background and opportunity for the Control Optimization System for Chiller Plant technology to reduce space cooling energy consumption at U.S. General Services Administration (GSA) facilities with centrifugal chiller plants containing multiple water-cooled chillers. The second section discusses the new technology, how it may reduce energy consumption, and introduces the demonstration location. The third section provides a detailed description of the demonstration facility and the configuration of the technology at the demonstration facility. The third section also provides a detailed overview of the approach used to assess the performance of the technology and how the chiller plant was monitored. The fourth section presents the results of the monitoring activity, documents performance and resulting energy savings, and presents the results of a life-cycle cost analysis. The fourth section also presents additional opportunities to further improve the performance of the chiller plant with control optimization, based on observations and lessons learned. The final section draws conclusions from the demonstration results and projects how GSA may best benefit from the technology's targeted deployment.

BACKGROUND

In the U.S., space cooling accounts for 7.4% of energy consumption in buildings (9.6% in office buildings) (EIA 2003b). However, because space cooling is primarily driven by electricity—a higher cost energy source—space cooling may account for more than 7.4% of a facility's annual energy bill. Within U.S. office buildings, chillers provide space cooling in only 2.3% of buildings (by number of buildings), but because chillers are used more frequently in larger facilities, 18% of building floor space is cooled using chillers (32% of office building floor space) (EIA 2003a). Therefore, a more efficient chiller plant offers significant opportunity to reduce annual energy costs for GSA, as well as reducing annual energy consumption.

OVERVIEW OF THE DEMONSTRATION TECHNOLOGY

Control Optimization System for Chiller Plants technology claims to optimize centrifugal chiller plants to minimize total power requirements. This technology does not require variable frequency drives (VFDs) for chiller compressor motors, which can be expensive. However, the control strategy requires the use of VFDs on all ancillary components such as chilled-water primary and secondary pump motors, condenser-water pump motors, and cooling tower fan motors. For constant-speed chiller plants with a primary-secondary configuration, the chiller plant will often be converted from a constant-volume primary system operation to a variable-volume primary system operation when control optimization is implemented.

The optimization technology claims to optimize pressure and temperature setpoints for chilled water and condenser water, while controlling pump and fan speeds to maximize the chiller plant efficiency.

Per documentation from the vendor, the control optimization system is expected to produce the following outcomes:

1. Control the chilled-water system to operate at or near original design intent throughout the entire system load requirements at all times. Condenser and chilled-water pumps, along with cooling towers, will require VFDs for this technology to work successfully.

2. Manage chiller lift for stable refrigeration performance at virtually all tonnage loads, without the need to install VFDs on the chillers themselves. The technology will work with chillers that have VFDs, but it is not necessary for a chiller VFD to be in place.
3. Reset pressure and temperature setpoints on chilled and condenser water loops based on current system dynamics to increase chiller plant deliverable tonnage while reducing chilled-water and condenser-water pumping energy, thereby optimizing total chiller plant kW per ton.
4. The technology is customized for each site to create a variable water loop flow that reduces pumping energy for all pumping systems (condenser-water and chilled-water).

This technology can be applied to chiller plants that serve a single building or a campus environment. The technology extends down to individual air handler cooling coils in an attempt to create a wider temperature difference (Delta T) between temperatures entering the coil and leaving the coil; this positively impacts the chiller efficiency. Realization of this technology's full benefit may require valve changes at cooling coils (remove three-way control valves and replace with two-way control valves) and other end of line system piping changes where bypass may exist. Therefore, some engineering of end of line piping and control valve changes may be required, depending upon the existing designs. Isolation valves (if not existing) may also be added to minimize flow through chillers and cooling towers that are not running. All of these potential modifications have a net system effect of reduced pumping power when coupled to the technology.

Control system optimization is not unique among control technologies that optimize the entire chiller plant. Another system, a variable-speed loop control logic, establishes performance algorithms for all chiller system components and attempts to determine the most efficient operational configuration, based on load and ambient conditions. To take advantage of part-load chiller efficiencies, compressor motors, condenser pumps, cooling tower fans, and secondary pumps all require VFDs.

In 2012, GSA's Green Proving Ground (GPG) program worked with researchers from the Pacific Northwest National Laboratory (PNNL) to assess an all variable-speed (AVS) loop control technology at the Thomas F. Eagleton U.S. Courthouse in St. Louis, MO. Researchers encountered problems with the study design and with the technology itself, so findings were not released. Researchers did, however, find that while the variable-speed loop control logic demonstrated energy savings, those savings did not justify installed costs. Variable-speed loop control technology requires significant integration, and staff anticipated information technology (IT) difficulties incorporating the technology into current GSA practices. Also, after the assessment, but while the loop control logic was still operational, building staff reported frequent chiller cycling, which stopped only when the control logic was turned off and the chiller was operated through the building automation system (BAS). Because of concern that frequent cycling would damage the chiller and/or shorten its life, use of the variable-speed loop control logic was suspended. It also is noteworthy that the AVS loop system requires an annual service fee of approximately $20,000 for remote monitoring and support.

STUDY DESIGN AND OBJECTIVES

GSA identified the Frank M. Johnson Jr. Federal Building and U.S. Courthouse as the demonstration location for the chilled-water (CHW) system optimization technology assessment. The facility consists of two primary buildings: the Frank M. Johnson Jr. U.S. Courthouse and the Frank M. Johnson Jr. Federal Building (also known as the "Annex").

The original courthouse was completed in 1933 and is listed in the National Register of Historic Places. The courthouse is five stories tall, with over 135,000 gross square feet and a U-shaped footprint with an interior light well and a red tiled roof. The most significant interior space is the U.S. District Courtroom, where Judge Johnson presided, on the second floor. Limestone arches surround the windows. Terrazzo and marble floors, marble wainscot, bronze elevator doors with bas-relief panels, and bronze radiator grilles are found throughout the building. Between 2002 and 2006, the original building was renovated and the interior spaces reconfigured to accommodate the needs of the court. Even though the courthouse was renovated, apparently it does not have any means to economize on conditioning the building during winter and cool spring and fall months. This drives the need to run the chiller system almost all year, but, at times, during significantly low loads.

The five-story Annex was constructed in 2002. With over 325,000 gross square feet, it has a radial (arc) design that is stylistically compatible with the original building. The Annex contains judges' chambers, District Courtrooms, and bankruptcy courts. The Annex was constructed to meet current ventilation code requirements, including economizer functionality that provides for cooling during winter, spring, and fall months. In 2004, the two facilities were linked together.

Space cooling is provided by a central chiller plant consisting of three, 400-ton centrifugal chillers (Carrier model number 19XR series chillers, installed circa 1998). The chillers have constant-speed compressors. Each chiller has its own chilled-water pump and condenser-water pump. Chilled water is distributed through the two buildings in a primary-secondary chilled-water distribution system. Along with the installation of the control optimization system, each of the pump motors and tower fan motors was equipped with a VFD. Differential-pressure transmitters were installed on all the pumps. Additional instrumentation was also installed or replaced to accurately measure loop temperatures (supply/return) for each primary chilled-water loop and the two secondary chilled-water loops, and for cross-over (bridge) temperature between the primary and secondary piping. New current transducers were installed on all of the chiller compressor motors and several of the pump motors to measure individual power loads. In some cases, existing VFD-derived signals (pump and tower fan motor kW loads) were used to save on instrumentation costs. These will be detailed in the instrumentation section.

The purpose of this demonstration was to determine the energy and cost savings associated with control optimization in the central chiller plant serving the Frank M. Johnson Jr. Federal Building and U.S. Courthouse.

Ideally, such energy and cost assessments include developing a pre-installation baseline (before the technology is installed) and post-installation (after the technology has been installed and operating) performance profile of the chiller plant, establishing a cooling load profile for the building, and comparing the results using a weather-normalizing approach. Ideally, at least one year's worth of baseline data is desired to provide an accurate comparison to the post-installation technology performance, as reflected by the data and weather-normalized. In this case, the technology was installed without GSA collecting a monitored baseline. However, an estimated baseline was simulated over the course of one month, after the optimization technology was installed but before it was implementated.

GSA used PNNL's Manual Mode approach to develop an estimated baseline, whereby the chiller plant could be operated in Manual Mode for two separate two-week periods in an attempt to simulate the baseline behavior of the chiller plant (prior to the implementation of the optimization technology). PNNL suggested that the monitored data provided by this approach is preferable to modeled information. Also, the site should be able to confirm that Manual Mode reasonably represents the original chiller plant operation. Finally, Manual Mode would use the original building automation system (BAS) operating code, which provides better confidence than simply disabling the control optimization system algorithms.

There are many assumptions regarding whether Manual Mode accurately represents the original chiller plant operations. This includes the pump speeds, chiller setpoints, chilled-water distribution pressure, and actual air handler economizer performance. The short time period (20–24 days total) will not capture the full load range of the chiller plant load profiles, which will require extrapolation of the Manual Mode data that results in uncertainty in the energy savings results.

Additional changes may have been made to piping systems (e.g., three-way valves converted to two-way valves, isolation valves added, etc.) which cannot be returned to original piping configuration (therefore not simulated). Further, it is not ideal to collect monitored data after the new technology has been installed. The site will be influenced by the control optimization system's operating strategy and may not operate the chiller plant or the loads served (air handling unit (AHU) cooling coils, etc.) exactly in the same manner as they used to be operated.

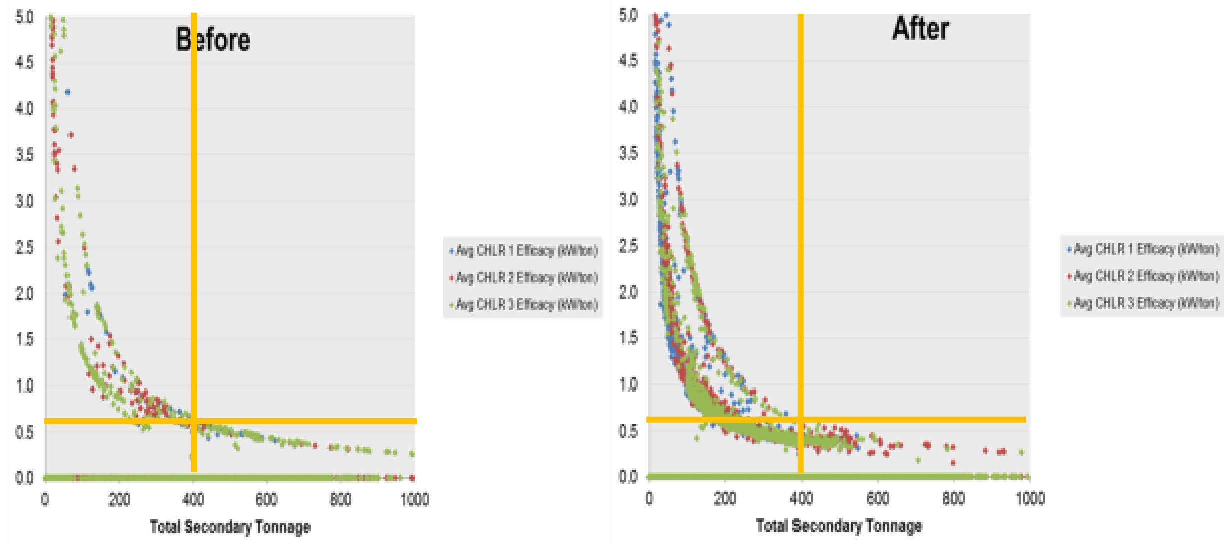
PNNL cannot validate that the Manual Mode data reasonably represents the chiller plant performance throughout the year. Therefore, results will be presented as energy and cost savings ranges, not specific values.

The monitoring plan targeted the recording of total chilled-water plant input power (kW), chilled-water flow rate and corresponding chilled-water supply and return temperatures, and the electric energy consumed by the chillers, chilled- and condenser-water pumps, and cooling tower fans. Parameters required to quantify the thermal cooling load provided by the chiller plant were also monitored. The chiller plant was monitored from January 13, 2013 through August 31, 2013 (peak summer and cooler fall and spring load periods). The monitored data was used to determine the operational performance of the chiller plant, as well as the thermal cooling load profile of the building as it relates to occupancy and weather conditions.

PROJECT RESULTS/FINDINGS

Analysis of the monitored data shows that the chiller plant with control optimization is more efficient than the baseline chiller plant. Figure 1 shows the performance profile of the chiller plant with the new technology compared to the Manual Mode (estimated baseline). There is a noticeable difference between the baseline chiller plant operations and the chiller plant operations with control optimization.

Figure 1. Comparison of constant-speed chilled-water pump estimated baseline (Before) and chiller efficacy with control optimization systems, post-installation (After) kW/ton performance.



The yellow lines on each chart represent chiller plant performance relative to 0.6 kW/ton efficacy and 400 Ton secondary (building loads). The optimization control technology performance is showing lower efficacy values than the pre-installation (simulated) performance values.

On a weather-normalized basis, the resulting analysis estimates that control optimization reduces the chiller plant energy consumption from a projected baseline value of 1,495.96 MWh/yr to a projected value of 972.51 MWh/yr, with an energy savings of 523.44 MWh/yr or 35% savings of the projected baseline.

Due to the limited data set from the brief Manual Mode period, the projected savings has a confidence level of 80%, resulting in a total uncertainty of (+/-) 10.8%. This means the projected savings could be as low as 24.2% or as high as 46.8%. The annualized cost savings are projected to be $42,485, assuming a utility cost of $0.08/kWh. With the level of uncertainty already noted, the annualized cost savings varies from $28,962 to $56,009.

Assuming that the Manual Mode simulation is a fair representation of previous chiller plant operations, a simple payback does occur during the 10-year life of the control technology[1] at all projected cost saving ranges, except for the very lowest savings range (<$31,000/year).

With a discount rate of 4%, the net present value of this demonstration is $324,436, and indicates this technology, as demonstrated, may be cost-effective because the energy savings justify the installed costs of $310,000[2], even at the highest levels of uncertainty.

[1] A 15-year equipment life for the VFDs was assumed, resulting in a 25% residual value at the end of the project period.

SPACE GAP

CONCLUSIONS

The application of control optimization system technology offers the potential for reducing energy consumption in space cooling applications through installation and optimization of variable-speed pumping systems and variable-speed airflow systems (cooling tower fans) that serve a water-cooled, centrifugal chiller plant.

The technology vendor claims that its technology can reduce annual chiller plant energy by between 20% and 50%, with an average predicted efficiency of 0.65 kW/ton. The chiller plant efficiency throughout this demonstration averaged 0.64 kW/ton, with an expected average annual energy savings of 35%. Therefore, the control system optimization technology is projected to be capable of achieving the claimed energy savings performance at the site being evaluated.

This technology is conditionally recommended for targeted deployment for locations that meet the following recommended guidelines[3]:

- The proposed site's electricity rates ($/kWh) are in the top 50% of GSA's portfolio.
- The proposed site's cooling season (or process loads) require chillers to operate >8 months/year (or longer).
- Buildings having cooling loads greater than 500 ton-hours/day, 3 million ton-hours per year.
- The proposed site will undertake new construction of facilities that meet the above criteria.

The technology cost-effectiveness is related to facilities annual ton-hrs and energy costs. In general, a larger cooling load will improve the ROI in areas of the country with lower energy costs. Guidelines for targeted deployment are best expressed by combining ton-hrs and energy costs.

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[2] This number is provided by the vendor and cannot be itemized. During the report investigation process, PNNL was told that at least 50% of the cost noted was for labor (engineering, design, management, startup, and site customization). Most of these technology upgrade efforts have significant labor costs built into them as they are unique, customized installations. There is an assumed labor component for engineering as a team of subject matter experts is attached to the site during design, installation, and startup. Also assumed to be part of the project costs are additional efforts to integrate the existing BAS with the vendor's technology, to customize the software, and start up/validate the new plant configuration with VFD-driven pumps and tower fans over several weeks, with additional site warranty requirements and validation of different seasonal loading that cannot be accounted for during the initial installation. There may be other costs of which PNNL was unaware. These costs (if true) would make the simple payback even longer, therefore not cost-effective.

[3] Prior to giving this technology consideration, a detailed site-specific engineering analysis is recommended which should include a monitored baseline for the existing chiller plant. This is a critical first-step to determine technology applicability.

Cost-effectiveness guidelines

Energy Cost	Facilities Cooling Load
At or above National avg $.11 blended KW/h	3 million ton-hr or greater
Below National avg $.11 blended KW/h	4 million ton-hr or greater

Variable-speed chiller plants already have variable frequency drives (VFDs), thus reducing installed costs, and should be evaluated for energy and costs savings with the control optimization system technology. However, the energy savings for variable-speed chiller plants is likely to be lower than the constant-speed chiller plant evaluated in this assessment. Therefore, it is recommended that a detailed engineering analysis, including a monitored baseline, be used to evaluate the possible application of control optimization for variable-speed chiller plants.

II. Introduction

The U.S. General Services Administration (GSA) is a leader among federal agencies in aggressively pursuing energy-efficiency opportunities for its facilities and installing renewable energy systems to provide heating, cooling, and power to these facilities. GSA's Public Building Services has jurisdiction, custody, or control over more than 9,600 assets and is responsible for managing a diverse inventory of federal buildings totaling more than 354 million square feet. This includes approximately 400 buildings listed in, or eligible for listing in, the National Register of Historic Places and over 800 buildings that are more than 50 years old.

GSA has an abiding interest in examining the technical performance and cost-effectiveness of different energy-efficient technologies in its existing building portfolio, as well as those buildings currently proposed for construction. Given that a large majority of GSA buildings include office space, identifying appropriate energy-efficient solutions has been a high priority for GSA, as well as for other United States federal agencies. Since the enactment of the Energy Policy Act of 2005 and Executive Order 13423, "Strengthening Federal Environmental, Energy, and Transportation Management" in 2007, other federal agencies are looking to GSA for strategies to meet the energy efficiency and renewable energy goals mandated by statute and Administration policy. Based on the sheer size of the building portfolio, there exists a huge opportunity for potential energy savings.

A. PROBLEM STATEMENT

Energy consumption for space cooling accounts for 9.6% of the total energy consumption in office buildings in the United States, according to the U.S. Energy Information Administration (EIA 2003b). This makes space cooling the fourth largest end-use energy consumer in office buildings. Further, space cooling in office buildings is provided by chillers in 32% of the total cooled floor space in office buildings (EIA 2003a). This makes chillers the second largest provider of space cooling in conditioned office buildings by total floor space. The largest provider of space cooling is packaged air conditioning units, which account for over 51% of total floor space.

Control optimization system technology is expected to offer higher efficiency compared to conventional constant-speed centrifugal chiller plants[4], as well as reduced energy consumption, thereby assisting GSA in achieving the energy-use intensity reduction requirements identified in the Energy Independence and Security Act of 2007 (EISA 2007). This report will assess the energy performance from one installation of control optimization system technology in a central centrifugal chiller plant and draw conclusions on how the application of this technology may contribute to further reductions in space cooling energy at GSA facilities.

The vendor-provided documentation for this technology claims that optimization control "resets pressure and temperature setpoints of chilled water and condenser water systems based on current system dynamics to increase chiller plant deliverable tonnage while reducing chilled water and condenser water pumping

[4] This report assumes, unless stated otherwise, that any reference to a chiller plant or the chiller plant equipment refers to a centrifugal chiller plant with centrifugal equipment.

energy and reducing chiller kW per ton." The technology utilizes variable frequency drives (VFDs) on all major chiller plant equipment to reduce flow rates so they are commensurate with the current system demand loads.

B. OPPORTUNITY

There are several alternatives for providing space cooling to commercial spaces. These include central chillers (air-cooled, water-cooled, centrifugal, reciprocating, rotary-screw, scroll, and absorption); direct expansion packaged air conditioning units (water-cooled and air-cooled); heat pumps (air-source, water-source, ground-source, and ground-water-source); plus a few other configurations. The Commercial Buildings Energy Consumption Survey (CBECS) estimates that chillers are used in 3.7% of commercial office buildings (by number of buildings), but in 31.9% of commercial office building floor space (EIA 2003a). This illustrates that chillers are predominantly used in larger (>200,000 ft^2) commercial office buildings.

What makes the control optimization system technology unique is its ability to calculate the chilled-water system's "dynamic" Variable System Pressure Curve to maintain the most efficient loop differential pressure for the systems (condenser water and chilled water) to operate, based on the current calculated system load. The technology application also reviews existing piping and equipment designs and may (if determined to be beneficial) add automatic isolation valves and convert piping loops with inherent losses to be more efficient. This is accomplished by converting three-way valves to two-way valves and removing bypass valves where water is flowing for no immediate benefit to the system load.

The control technology primarily reduces energy consumption at the VFD-driven pumps due to derived-benefits from the affinity laws. The affinity laws state that power consumed by a centrifugal device (e.g., pump motor, fan motor, etc.) is proportional to the cube of the device's speed (ASHRAE 2012), as shown in Table 1.

Table 1. Affinity laws applied to a centrifugal pump and fan motor.

Relationship	Affinity Law	Pump Motor	Fan Motor
Flow rate is proportional to speed	(Flow) α (Speed)		
Power is proportional to the cube of the speed	(Power) α $(\text{Speed})^3$	Since (Flow) α (Speed):	Since (Flow) α (Speed):

GPM: Pump flow rate, gallons per minute
RPM: Pump speed, revolutions per minute
Hp = Pump power, horsepower

To illustrate how this operating strategy can reduce power, assume a chiller plant has two 50-hp chilled-water pumps and operates one pump at full load. The shaft power demand for the one pump at full load is 50 hp. However, if both pumps operate at 50% load to achieve the same combined flow rate as one pump at full load, the shaft power demand is only 12.5 hp, representing a 75% reduction in shaft power. The same principle is applied to the cooling tower fan motors. However, in pumping systems, this is an idealized relationship because hydronic flow losses, due to piping frictional losses and other system effects, contribute to the power relationship being less than a cubed relationship.

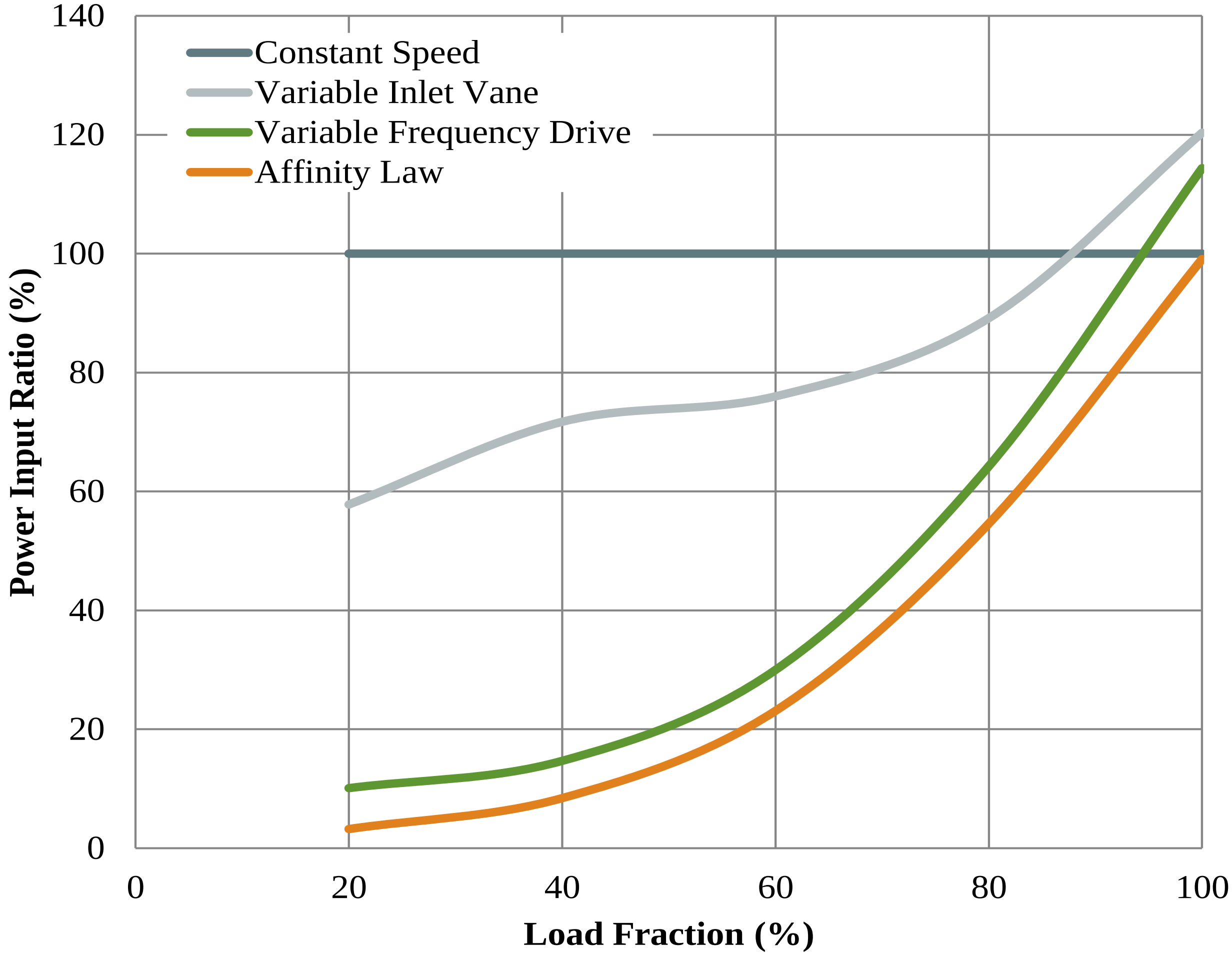
Individually, VFDs provide significant energy savings because the speed of the equipment can be decreased when the load allows. The benefit of VFDs is illustrated graphically in Figure 2, where the power input ratio, relative to constant-speed equipment, is compared to the centrifugal affinity law.

Control optimization may also reduce energy consumption by lowering the entering condenser water temperature (ECWT) [5] to the chiller while managing chiller lift, thus avoiding chiller surging. In general, the efficiency of a centrifugal chiller can be increased by 0.4% per 1°F reduction in the ECWT (Thumann 1991)[6]. Control optimization system technology optimizes cooling tower fan speeds in conjunction with optimized condenser pump speeds to maintain the required condenser water temperature and flow parameters. This operational approach lowers the ECWT and increases the chiller plant efficiency. However, many other building automation system (BAS) control algorithms are capable of achieving reduced condenser water temperatures, even with constant-speed equipment.

[5] The manufacturer cites condenser temperatures in the 50 degree range, though colder condenser water temperatures are limited by what the chiller manufacturer will allow.

[6] Based on constant condenser pumping. Variable condenser pumping can increase chiller efficacy 1-2%.

Figure 2. Power input ratio (relative to constant volume) for centrifugal equipment as compared to the affinity law (adapted from Doty and Turner, 2009).



III. Methodology

The methodology section is divided into four subsections. First, a detailed description of the technology is provided. Second, the controls behind the technology are described. Third, the desired technical objectives are discussed. Finally, the demonstration location is introduced.

A. TECHNOLOGY DESCRIPTION

This report refers to the subject technology as the Control Optimization System for Chiller Plants . In this technology, all chiller plant equipment that impacts flow of water and air (chilled water pumps, condenser water pumps, and cooling tower fans) is required to be variable-speed. This technology is applicable only to centrifugal chiller plants and does not apply to other chiller types (for example, positive displacement). The control technology is referred to as “demand flow” because the control of mass flow rates through the different parts of the chiller plant is required to meet the real-time calculated demand for heat transfer (chilled water flow, condenser water flow and cooling tower air flow) .

The specific technology evaluated is a network-based control strategy designed to minimize flow requirements in the chiller plant (towers and chillers) and at the connected loads (air handling units, AHUs). Typically, the technology can eliminate decoupling or distribution system bypasses (such as primary-secondary distribution systems and three-way valves) and replace them with primary/boost systems and variable-flow via variable frequency devices (VFDs). The technology will work with VFDs on chiller compressors, but VFDs on chiller compressor motors are not required.

Control optimization system technology optimizes the chiller plant efficiency by staging all the chillers, chilled water pumps, condenser water pumps, and cooling tower fans such that the combination of chillers and their related ancillary systems are operated closest to their original design curves throughout the system loading. Specifically, control optimization seeks to operate the minimum number of pumps and cooling tower fans required at lower part-load capacity to reduce overall chiller plant power consumption, while maintaining the required chiller lift for stable refrigeration performance at all tonnage loads. The following technical descriptions provide some additional insight:

- The technology requires a customized operating configuration (unique to each installation) to ensure the chiller plant is always operating at its peak efficiency. The technology achieves peak efficiency by optimizing pump speeds on the condenser, evaporator, and secondary building loops, along with optimizing the cooling tower fan speeds.
- The technology seeks to optimize flows by isolating equipment from each other (e.g., towers and chillers) and removes or eliminates the need for bypass valves and three-way valves (at pumps), ensuring pumping energy is used solely for moving fluid through coils and equipment that are needed for heat transfer. Isolation of equipment (e.g., towers and chillers) may require the addition of isolation valves (or replacement, if failed, or if check valves used for isolation are unreliable).
- While no two chilled water plants are the same, if similarly sized and designed chilled water plants existed, the cost savings of not having to create customized, control-coded sequences might exist. This would, hopefully, provide lower design and installation costs for the prospective site. However,

given other constraints (e.g., location, climate zone, building loads, mission of the site, AHU cooling coil designs, and level of maintenance) that exist from site to site, this may not be possible.

- The optimization technology's Variable Pressure Curve Logic algorithm is used in conjunction with a control panel to calculate the chilled water system's dynamic Variable System Pressure Curve. Calculated Proportional, Integral, Derivative (PID) loop signals are delivered to the condenser and chilled water VFD-driven pumps as well as the cooling tower VFD-driven fans. The technology determines the speed (hertz) at which the VFDs should be dynamically operating for the calculated system loads. The calculated Variable System Pressure Curve is continually resetting the loop differential pressure setpoints to maintain the most efficient evaporator and condenser water-side pumping pressures (matched to the current system load). The goal is to reduce chiller kW/ton performance while optimizing system pumping pressures so they are matched to system loads. The intent is to reduce or eliminate mixing of chilled water supply and chilled water return (without any effective end-use cooling). When this unwanted mixing occurs, the results include lower Delta T values (the difference in temperature) between leaving and entering water loops at the chiller machine(s) than the system design intended, which can result in reduction in chiller delivered capacity and degrading of chiller efficiency. The results often reflect movement of water with no real benefit to the end-use cooling load(s) in the building.

- System loads are calculated using chilled water plant instrumentation (temperature and flow sensors), as well as integrating into the building automation system (BAS) for the purpose of obtaining real-time load performance data at individual air handling units and their respective cooling coils. Real-time load performance at individual AHUs is generally determined from cooling coil valve positions, entering and leaving air temperatures for the cooling coils and individual space temperature and humidity sensors (as selected). To further optimize the technology's performance, if a significant number of cooling coils are designed with three-way control valves, they may be modified to be two-way control valves to ensure VFD-driven pumps can benefit from reduced valve opening (part-load) conditions.

- Based upon the AHU cooling load calculations, the primary chilled-water pumps (and secondary chilled-water pumps, if designed as a primary/secondary pumping system) can be varied in speed to reduce flow rates at both ends of the chiller plant (the chiller machines the AHU cooling coil loads). This is done while still maintaining minimum evaporator flow rates through individual chiller machine(s) and while still maintaining AHU cooling coil leaving temperatures. This translates to higher Delta T. As a rule of thumb, this difference in leaving and entering water temperature works to the benefit of the chiller machine life and operating efficiency when the Delta T is greater than 8^{o}F-12^{o}F.

- Since temperature measurement is also critical to the technology application, it would not be uncommon for existing chilled water plants with existing BAS infrastructure to have their existing temperature sensors and flow sensors on chilled water and condenser water loops replaced with higher accuracy (tighter precision) temperature sensors and flow meters. This sensor "optimization" may also include relocation of sensors (if poorly located) based on the vendor's engineering analysis.

- Since water flow rates are being reduced at the chiller machines, existing safety flow switches in the evaporator and condenser loops most likely will need to be replaced with switches that are sensitive to lower flow rates (to avoid nuisance chiller tripping/shutdown actions).

B. CONTROLS DESCRIPTION

The control optimization system technology for chiller plants provides lead-lag operation for the three chillers. On a demand for mechanical cooling, the lead chiller will be turned on by the control system. This chiller will operate and maintain the return chilled water temperature as required by the controls algorithm. If the desired return chilled water temperature cannot be maintained (after an adjustable time delay – 30 minutes), the lag chiller will be turned on. The lag chiller will run for a minimum of 2.5 hours and is adjustable. If the lag chiller is not able to maintain the return chilled water temperature required by the controls algorithm (Delta T), the third chiller will be turned on. When the Delta T is satisfied, the system will index the lag chiller(s) off in reverse order (last chiller turned on is the first chiller turned off, after the adjustable timer has expired).

Primary variable volume chilled water pumps and condenser water pumps will be started and stopped by the control optimization system. When a chiller sequence is initiated, the respective chilled water pump will be turned on after a time delay. After an additional time delay, the condenser water pump will be turned on. After a further time delay and when water flow has been established (by the flow switches in the respective condenser and chilled water loops), the respective chiller will be started. Since flow rates are often reduced during the chiller operations (to optimize flow with demand), the existing flow switches are removed and replaced with flow switches that are more sensitive to low flow rates (another project upgrade cost). When mechanical cooling is no longer required from the chiller, the chiller will be turned off first. After a time delay, the condenser water pump will be turned off. After a further time delay, the chilled water pump will be turned off. If a pump failure is detected, the lag chiller system will be turned on.

The systems technology determines when to stage equipment up or down. After an adjustable time delay (30 minutes), the lag chiller will be turned on. The lag chiller will run for a minimum of 2.5 hours (adjustable). When a chiller sequence is initiated, the respective chilled water pump will be turned on after a time delay. After an additional time delay, the condenser water pump will be turned on. After a further time delay and when water flow has been established (by the flow switches in the respective condenser and chilled water loops), the respective chiller will be started. When mechanical cooling is no longer required from the chiller, the chiller will be turned off first. After a time delay the condenser water pump will be turned off. After a further time delay the chilled water pump will be turned off. These delays prevent short cycling of chiller plant equipment.

Secondary variable flow chilled water pumps will be started and stopped by the control system whenever mechanical cooling is needed. The speed of the pumps will be controlled to satisfy the chilled water demand, as determined by the building's chilled water control valves (located at each AHU).

On initial call for mechanical cooling, one tower fan and one condenser water pump and one chilled water pump will be energized, along with the lead chiller. Additional cooling towers will be brought on line (tower isolation valves will open accordingly) as needed, based upon the number of chillers in operation and the demand for condenser water. The condenser water temperature will be maintained by modulating the individual cooling tower fan speeds. The volume of condenser water required at each chiller will be maintained by modulating the condenser water pump VFD speed.

Control optimization integrates with any BACnet-enabled BAS, though if the existing version of BACnet is older than five years it will require additional programming. Also, some GSA regions are refusing to allow any BAS-integration product to be connected to their buildings unless it meets certain design criteria. In some cases, this means the product must be a "Tridium" BAS-compatible system/component.

C. TECHNICAL OBJECTIVES

The purpose of this demonstration is to determine the energy and cost savings associated with control optimization in a central chiller plant serving a federal building and federal courthouse, as compared to the baseline constant-speed chiller plant. The demonstration will monitor the operational performance of the chiller plant with the installed technology over the full range of normal operation. The cooling-load profile for the demonstration building also will be determined from the monitored data. The measured performance of a chiller plant with the new optimization technology will be compared, using a weather-normalized analysis, to assess the resulting energy savings delivered by the technology. The findings and conclusions also will include a life-cycle cost analysis. To the extent possible, the demonstration also will assess the technology's broader application within the GSA portfolio.

The loop control optimization (along with the chilled- and condenser-water pump and cooling tower fan VFDs) were installed prior to this demonstration[7]. Therefore, the performance of the baseline constant-speed chiller plant could not be monitored as part of this assessment. The control optimization system vendor provided model performance data for the baseline constant-speed chiller plant. This occurred prior to installing the technology, installing isolation valves, modifying bypass valves, and installing VFDs for different pumps and cooling tower fans. While the vendor modeling of proposed energy savings may be based upon multiple installed configurations and multiple sites in various climate zones, this is not adequate for a true measurement and verification (M&V) effort. Instead, a baseline was derived from the monitored peak power of the chiller plant equipment when operated in a mode (referred to as Manual Mode) that was configured to "simulate" pump operations prior to implementation of the technology.

In an attempt to develop retroactively a constant-speed chiller plant baseline performance, the control optimization system was disabled for two separate time periods: March 18, 2013 through March 31, 2013 (winter weather demonstration), and again from August 19, 2013 through August 31, 2013 (summer weather demonstration). It was assumed that while the control technology was disabled, the VFDs on all the chiller plant equipment would be disabled (VFDs would be bypassed or placed at 100% speed) and the performance of the baseline constant-speed chiller plant would be replicated. In most cases, this was achieved without too much effort, but the control vendor had to make some modifications to the control technology to simulate the original control responses (e.g., equipment staging), especially during low-load periods (the March 2013 demonstration period).

[7] The control optimization system, condenser water pump and primary chilled water pump VFDs and cooling tower fan VFDs were installed in 2012 as part of an American Recovery and Reinvestment Act (ARRA)-funded project. The secondary chilled water pump and cooling tower fan VFDs are assumed to be installed circa 1998 and have been replaced/upgraded only as VFDs have failed or become obsolete.

In some cases, three-way valve and bypass valve conversions, or lack of isolation valves, could not be simulated without a significant level of effort by the on-site GSA Operations & Maintenance (O&M) contractor staff. Simulating previous conditions that no longer exist adds a certain level of uncertainty (which cannot be avoided), when verification of equipment and plant performance is attempted without properly measuring the baseline system performance.

Typically, different technologies are compared on the basis of efficiency (or output energy divided by input energy). However, refrigeration equipment does not generate output energy, but rather transfers thermal energy from the building to the outside environment; therefore, the conventional definition of efficiency (i.e., output divided by input) is irrelevant. An appropriate comparison is better defined as cooling energy provided (or useful thermal energy) divided by energy consumed. There are several methods to convey chiller efficiency. The most general is to use the coefficient of performance (COP), which is determined as the ratio of useful thermal energy (in Btu) to energy consumed (also in Btu). The COP also can be calculated using the rate of useful thermal energy (in Btu/h) divided by power demand (also in Btu/h). COP is a unitless measure of efficiency and provides a direct comparison of how much energy is required by the chiller to transfer thermal energy out of the building.

For industrial chilling equipment, chiller performance also may be expressed as power input per unit of capacity. As defined by the Air Conditioning, Heating and Refrigeration Institute (AHRI), power input per capacity is a ratio of the power input supplied to the chiller (in kilowatts, kW) to the net refrigerating capacity (in tons of refrigeration) at any given set of rating conditions, expressed in kW/ton (AHRI Standard 550/590-2011). Also, the rated efficiency level for commercial chillers used by the Federal Energy Management Program (FEMP)-designated products and ASHRAE Standard 90.1-2010 (ASHRAE 2010) are expressed using the power input per capacity ratio (kW/ton). This is an inverse efficacy measure because a smaller number implies a greater efficiency (or higher COP).

The primary objectives are to optimize flow rates by measuring the supply/return temperatures of various flow loops (condenser water and primary and secondary chilled water loops). When the difference (Delta) between the supply and return loop temperatures is lower than the optimum design value, this can initiate a reduction in flow rates that results in pump horsepower (hp) savings while returning the Delta to design levels.

Other benefits can include more reliable operations during low-load periods (cool spring and fall months, winter months). In some climate zones (1A, 2A and 3A), ASHRAE design guidance (at the time of the demonstration) allowed for minimum ventilation designs to be applied to buildings in such a way that they can never optimize the use of outdoor air for cooling, even when optimal. When these designs are applied, the greatest amount of outdoor air that can ever be introduced to the supply fan systems is often less than 20% of the total supply air. If there are minimal hours per year (most likely the winter months of November–February) when a full economizer design would allow the building to be cooled without having to operate the chillers, that cannot happen when the building has an economizer designed to provide only the minimum required ventilation. In locations in the southern U.S., this is often the case. The alternative for many of these locations is to provide some means to economize via water-side (cooling tower) economizer systems that bypass the chiller plant, while using the cooling tower during low ambient wet-bulb conditions.

While this analysis is not focused on other means to economize, it may be prudent to evaluate these systems because many locations may find additional benefit from water-side economizer technology. In some cases, it has been our observation that water-side economizers have been abandoned in place due to poor design, poor maintenance, poor control, and other operational issues. Other low-load operations exist as follows:

1. Systems in which full air-side economizers exist on some AHUs, but not on all AHUs. This can result in the need to still mechanically cool (chiller operations) in cool outdoor conditions (but with lower loads than would typically be found with working air-side economizers).
2. Systems in which all AHUs can utilize a full economizer, but the building has a year-round process load (often 24/7) that is running inside the building. Often, this can be a data center process load, but it may be other process loads (e.g., medical or manufacturing).

When the only option for the building O&M staff is to provide mechanical cooling, this technology has significant advantage for optimizing chiller plants that operate during low-load conditions.

Data from power meters and sensors (i.e., temperature and flow) was collected through the existing Siemens building automation system (BAS). Data from the meters and sensors was collected at 1-minute intervals and averaged before being stored in 15-minute interval tables. The stored data was used both to determine the performance profile for the chiller plant and to determine a thermal cooling load profile for the building. Data was collected from January 13, 2013 through August 31, 2013.

This period of monitored data includes peak summer and cool fall and spring season loads and represents the full range of normal operation. The chiller plant performance, building thermal load profile and typical meteorological year (TMY) data were used to estimate weather-normalized energy savings associated with the control technology. The results were used to develop an economic assessment of the technology.

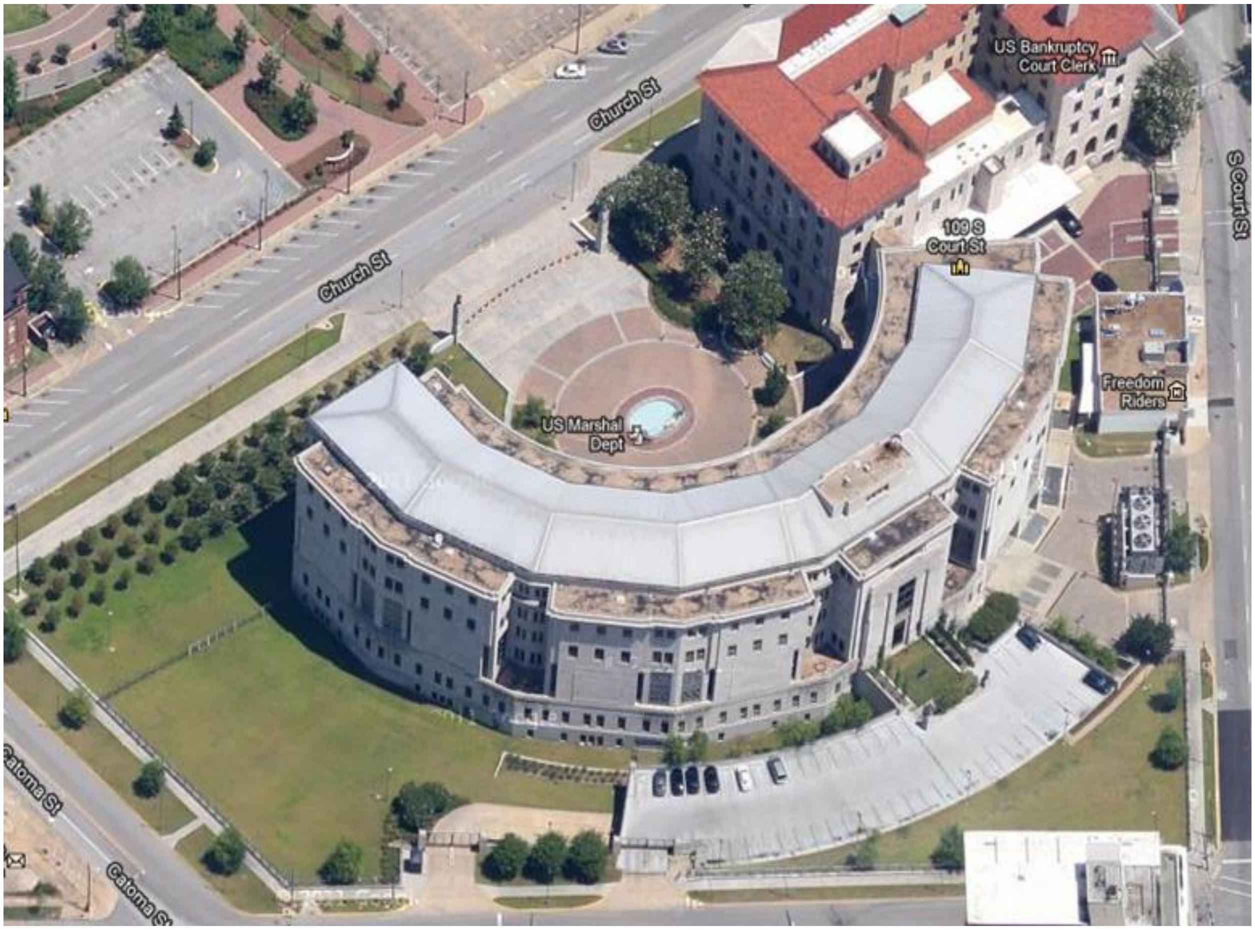
SPACE GAP

D. DEMONSTRATION PROJECT LOCATION AND DESCRIPTION

GSA identified the Frank M. Johnson Jr. Federal Building and U.S. Courthouse as the demonstration location for the control optimization system for chiller plants. The five-story U.S. Courthouse is a major landmark in the city of Montgomery, Alabama. Completed in 1933 and later listed in the National Historical Register, the Frank M. Johnson Courthouse has more than 135,000 gross square feet of space. The Federal Building (known as the "Annex") is a five-story plus basement building that was constructed in 2002 in the shape of an arc and is located adjacent to the existing U.S. Courthouse. The Annex has more than 325,000 gross square feet of space. The central chiller plant is located in a basement-level mechanical room and the cooling towers are located outside on the street level (S. Court Street). A photograph of the building is shown in Figure 3.

The optimization technology was installed as part of the multimillion dollar American Recovery and Reinvestment Act of 2009 (ARRA) project. The project included advanced metering installation, BAS graphical user interface upgrades, installation of VFDs on all primary chilled- and condenser- water pumps, and installation of the optimization technology. See Table 3 for additional instrumentation used by the technology.

Figure 3. Photograph of the Frank M. Johnson Jr. Federal Building and U.S. Courthouse, located in Montgomery, Alabama. The Courthouse is the upper right building (red roof) and the Federal Building (Annex) is the lower (arc/semi-circle) building. The cooling towers are located outside, just to the right of the Annex next to S. Court Street. Photo courtesy of Google Maps.



IV. M&V Evaluation Plan

The measurement and verification plan section includes four subsections. The first section provides a detailed description of the demonstration facility, including the baseline constant-speed chiller plant operational strategy. The second subsection provides a description of the planned operation of the technology. The third subsection identifies the analytical method that will be used to evaluate the performance of the chiller plant with the subject technology. The fourth section identifies the instrumentation that was installed to collect the necessary data used to evaluate the technology.

A. DETAILED CHILLER PLANT EQUIPMENT DESCRIPTION AND HISTORICAL OPERATION

The Frank M. Johnson Jr. Federal Building and U.S. Courthouse central chiller plant consists of three, 400-ton Carrier centrifugal water-cooled chillers. Three cooling towers provide heat rejection for the chillers. Each of the three cooling towers is served by a 50-hp fan. Each chiller has its own chilled-water pump (15 hp) and condenser-water pump (30 hp). The nameplate specifications for the chiller plant equipment are listed in Table 2, where all the equipment is rated at three-phase, 480 volts. A schematic of the chiller plant is shown in Figure 4.

Table 2. Chiller plant equipment nameplate information.

Chiller Plant Equipment	Nominal Capacity	Rated Electrical Power (kW)	Run Load Current (A)	Rated Efficiency
Chillers 1, 2, 3	400 tons	291	350	N/A
Primary Chilled-water Pumps 1, 2, 3	15 hp	11	19	90%
Secondary Chilled-water Pumps 4, 5	60/25 hp	40/20	72/30	92%
Condenser-water Pumps 1, 2, 3	30 hp	24	35.5	92-94%
Cooling Tower Fans 1, 2, 3	50 hp	40	N/D[1]	N/D

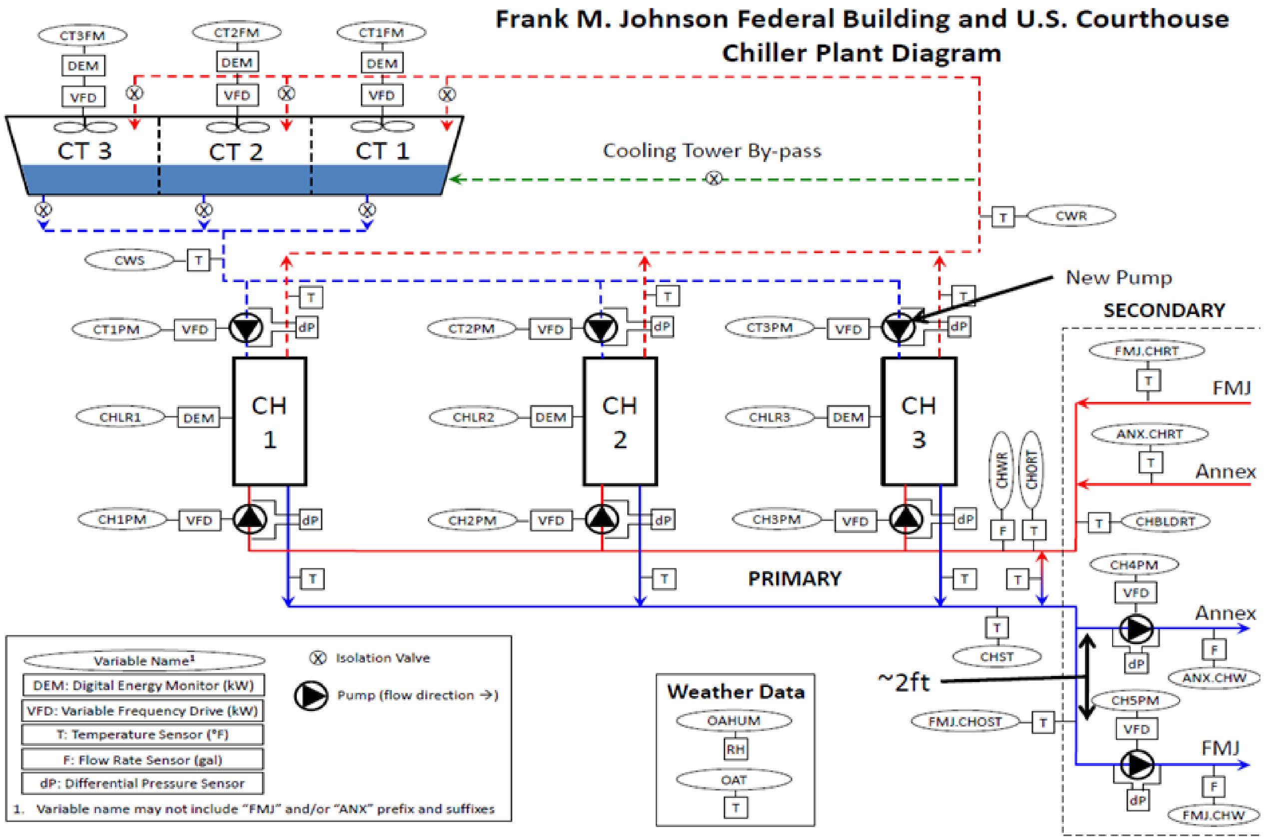
1. N/D: Not documented

Chilled water from each chiller feeds into a central header that leads to the main chilled-water supply pipe for the two buildings (Courthouse and Annex). Individual secondary chilled water pumps circulate chilled water to the Annex (Pump 4) and to the Courthouse (Pump 5) buildings to provide chilled water to the various AHUs.

Condenser water from each chiller feeds a common pipe that travels to the cooling towers located outside the building at the S. Court Street level (see Figure 3 for cooling tower location). There is a bypass valve and line between the condenser leaving water and the tower basin that bypasses condenser water flow around the tower fans when the leaving water is too cool (because of cold weather conditions). Isolation valves are provided on the entering and leaving side of each cooling tower.

SPACE GAP

Figure 4. Schematic of the Frank M. Johnson Federal Building and U.S. Courthouse central chiller plant.



The chilled- and condenser-water distribution piping remained unchanged as a result of the control technology installation, except for the automatic isolation control valves that were installed on the entering and leaving sides of each cooling tower, as illustrated in Figure 4. It is unclear from interviews and documents if any field-level cooling coil control valves (AHU cooling coils) or their associated piping were replaced or modified (converted from three-way bypass piping to two-way). Field-level modifications can be part of the control technology upgrades that are performed as part of the optimization. Figure 4 also shows locations of new and existing VFDs and power metering, differential pressure, temperature, and flow instrumentation.

Installation of the control technology was started in February 2012 and completed in June 2012. In addition to the control technology, new VFDs were installed on each chiller’s primary chilled water and condenser water pumps. VFDs already existed on the secondary chilled water pumps and the cooling tower fans (assumed to be installed in 1998). Ultrasonic flow meters also were installed on the two secondary chilled-water supply pipe loops that feed the Courthouse and the Annex buildings and on the primary chilled water return loop. Additional instrumentation (i.e., temperature sensors, flow switches, and differential pressure transducers) and isolation valves were installed as noted in Table 3. In addition, the existing Siemens BAS software and hardware upgrades were performed to permit BAS integration with the new control technology.

Discussions with site's O&M operating staff and observations from site visits, mechanical print and BAS sequence documents all highlighted the fact that the chiller plant was a constant-speed plant prior to the installation of the control technology.

Previously, the chiller plant was controlled through the BAS to operate the number of chillers and pumps required to satisfy the cooling loads in the two buildings. Due to legacy Heating, Ventilating, and Air Conditioning (HVAC) design issues in the 1933-era Courthouse, the ability to economize (bring in cool outside air) to satisfy comfort cooling requirements fully does not exist during winter and cool fall and spring conditions (outside air temperatures < 55°F-65°F). Therefore, it is common to need mechanical cooling in the Courthouse even during times when outdoor air temperatures are below 40°F. During cool outdoor conditions like this, the need for chilled water is not as great or consistent, and return chilled water temperatures can drop quickly (due to satisfied loads and cooling valves closing down). The setpoint for supply water leaving the chiller does not always adjust automatically. As a result, the chiller(s) would short-cycle and, in some cases, fail on any number of chiller safety trip events (typically low pressure).

The chiller plant is designed to provide a constant chilled-water supply temperature of 44°F. The secondary chilled-water pump VFDs were operated to maintain a minimum loop differential pressure setpoint of 8–12 PSI. The VFD-driven cooling tower fans were controlled to deliver a constant Entering Condenser Water Temperature (ECWT) of around 80°F–85°F. While this may be the expected summer operations when outdoor wet-bulb temperatures are higher, this does not take advantage of lower wet-bulb temperatures that often exist in the winter and cool fall and spring months. The cooling towers are equipped with basin heaters (rated at 24 kW total capacity) to prevent freezing and allow for year-round operations.

B. INSTRUMENTATION PLAN

The existing Siemens BAS was used to monitor and record data. Data collection included sensors that are directly used in the various control loops for the chiller plant performance. A few additional sensors were installed to validate chiller operation. Sensor data was measured every minute. Average sensor data was recorded in 15-minute intervals. Table 3 identifies the sensors monitored and recorded by the BAS.

The primary sensors included 480-volt, three-phase Siemens digital energy monitor (DEM) power meters on each chiller and cooling tower fan. Ultrasonic flow meters were installed on the two secondary chilled-water supply loops to the Annex and Courthouse buildings, along with an ultrasonic flow meter in the primary return water line. Differential pressure sensors were installed cross each condenser water pump and each chilled water pump. In addition, newer high-accuracy temperature sensors were installed on the individual leaving chilled water and condenser water lines from each chiller, a common return water line from the chiller plant to the towers, a common supply water line from the towers to the chiller plant, a common secondary loop supply temperature sensor, individual secondary return water temperature sensors, and a secondary loop bypass temperature sensor.

Though not related to the primary instrumentation monitoring sensors and plan, new flow switches were installed on each chiller's condenser and evaporator piping loops. Standard flow switches are designed for constant flow, but the addition of VFDs converted the chiller machines from constant flow to variable flow.

At lower flow rates, the existing flow switches can become problematic. The vendor has found that the optimum action is to replace with newer flow switches, which are more reliable at lower flow rates.

Table 3. Sensors monitored by the Siemens building automation system (BAS).

Parameter	Sensor	Units	Signal Type
Chilled-water supply flow rate	Siemens FST020 ultrasonic flow meter	gpm	Pulse
Chilled-water supply and return temperatures	Siemens high-accuracy, surface-mounted thermocouples	°F	Voltage Differential
Condenser-water supply and return temperatures	Siemens high-accuracy, surface-mounted thermocouples	°F	Voltage Differential
Power, Chiller #1	Siemens DEM power meter, 500-amp current transformer	kW	Pulse
Power, Chiller #2	Siemens DEM power meter, 500-amp current transformer	kW	Pulse
Power, Chiller #3	Siemens DEM power meter, 500-amp current transformer	kW	Pulse
Power, Primary Chilled-water pump motor #1	VFD-derived	kW	BAS-Digital
Power, Primary Chilled-water pump motor #2	VFD-derived	kW	BAS-Digital
Power, Primary Chilled-water pump motor #3	VFD-derived	kW	BAS-Digital
Power, Secondary Chilled-water pump motor #4	VFD-derived	kW	BAS-Digital
Power, Secondary Chilled-water pump motor #5	VFD-derived	kW	BAS-Digital
Power, Condenser-water pump motor #1	VFD-derived	kW	BAS-Digital
Power, Condenser-water pump motor #2	VFD-derived	kW	BAS-Digital
Power, Condenser-water pump motor #3	VFD-derived	kW	BAS-Digital
Power, Cooling tower fan #1	Siemens DEM power meter, 100-amp current transformer	kW	Pulse
Power, Cooling tower fan #2	Siemens DEM power meter, 100-amp current transformer	kW	Pulse
Power, Cooling tower fan #3	Siemens DEM power meter, 100-amp current transformer	kW	Pulse
Outside dry-bulb temperature and relative humidity	Siemens high-accuracy, sensor and radiation shield	°F %RH	Voltage Differential

SPACE GAP

C. TEST PLAN

A monitoring plan was created to collect data on the chillers, chilled- and condenser-water pumps and cooling tower fans. The cooling plant performance data was collected from January 2013 through August 2013. The data were used to determine the performance of the chiller plant under various load conditions. The data for determining thermal performance were monitored and stored using the site's BAS provided by Siemens. This data was uploaded periodically for independent review to ensure completeness (no holes, no "stale" data). Once the data was validated, it was stored for future analysis. Anomalies in the data were reviewed with the site staff to ensure completeness or make provisions to account for operational issues.

Further, since the site did not provide any data for the chiller plant prior to the ARRA-funded improvements, there was no way to validate the actual pre-technology improvement period. Therefore, it was determined that GSA would configure the chiller plant to simulate a "winter" season and a "summer" season (to the extent possible) by configuring equipment and controls to operate in a manner that simulated the pre-technology operations. This simulation period was performed in March (winter operations) and August (summer operations) of 2013. The simulation period was supposed to be 2 weeks (14 days), but, in both cases, the simulation period was closer to 12 days. The results of this simulation will be discussed later.

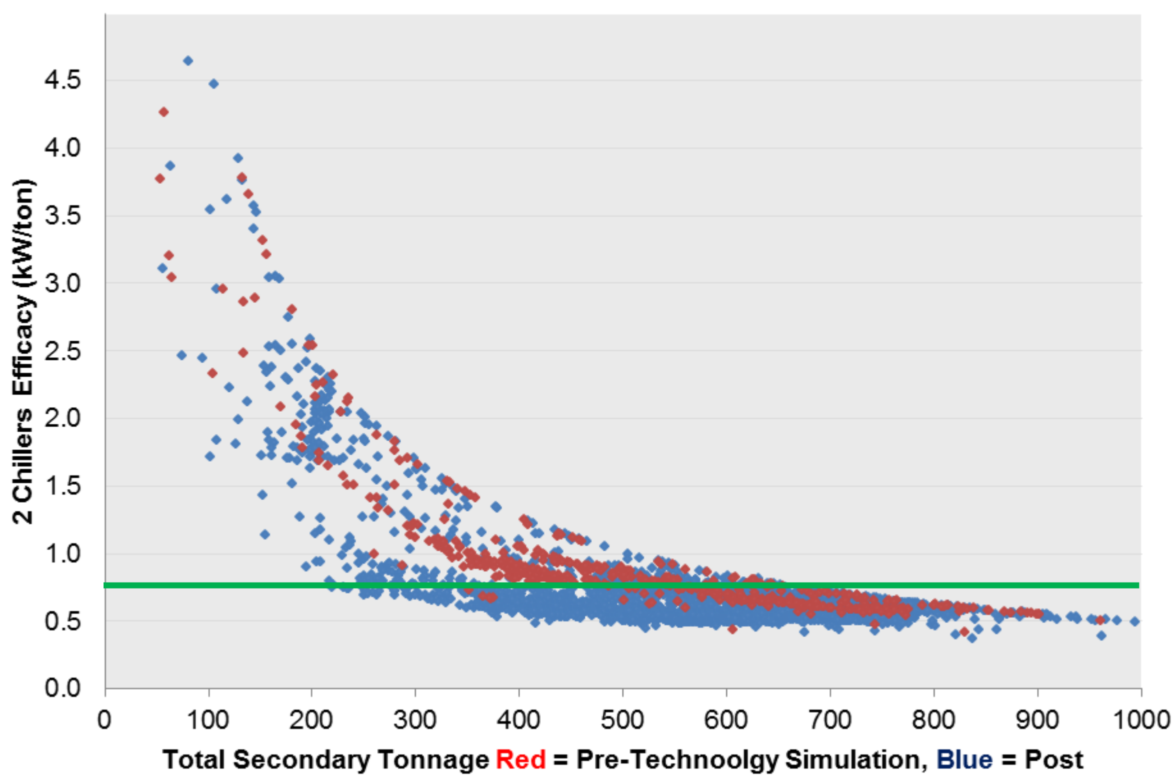
V. Results

The results section is divided into four sections. The first section highlights the impacts of the control optimization system technology. The second and third sections describe the baseline (before) and post-installation (after) performance profiles of the chiller plant resulting from the monitored data, respectively. The fourth section extrapolates from the observed data and findings to identify additional opportunities for improving the performance of the chiller plant both for the monitored location, as well as for potential other applications in GSA's portfolio.

A. PRE- AND POST-INSTALLATION CHILLER PLANT PERFORMANCE

Figure 5 highlights the improved performance (lower kW/ton) when building cooling loads are less than 400 tons.

Figure 5. Pre- and post-installation power and load comparison for 2 chillers.

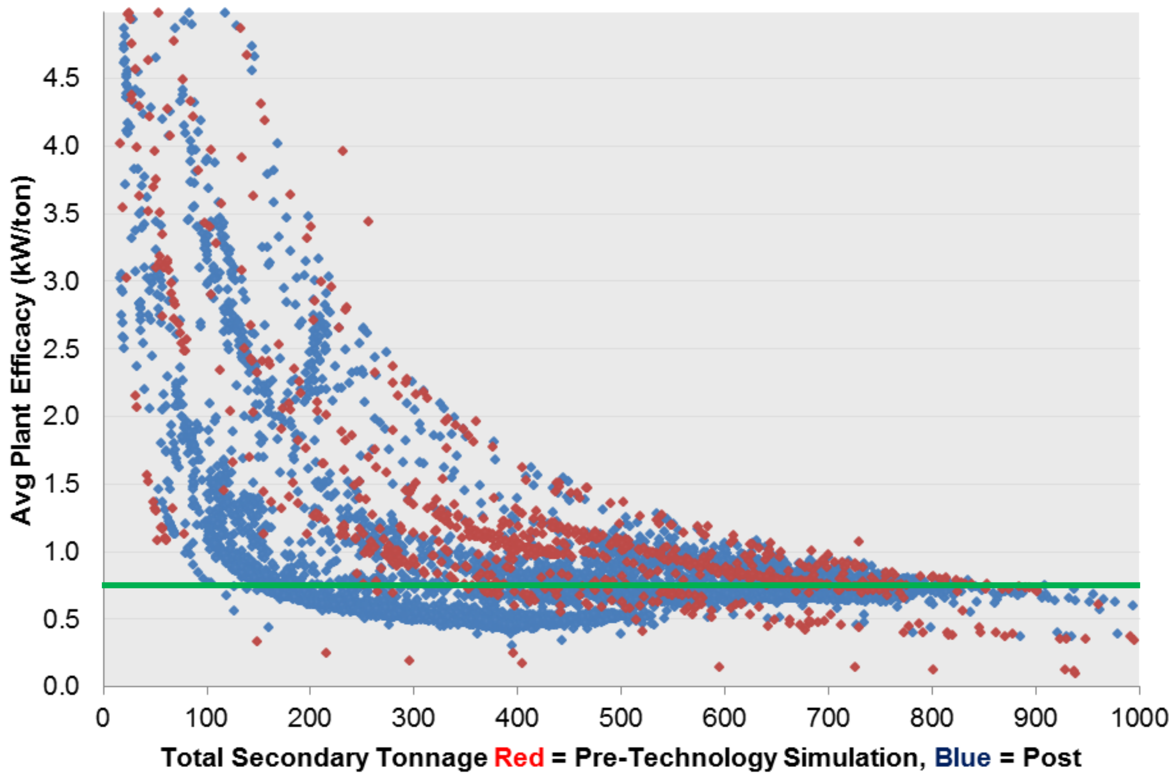


In Figure 5, the green line shows performance relative to an efficacy of 0.75 kW/ton. As loads increase above 500 tons, the kW/ton efficiency of the pre-technology simulation and post-technology operations begin to converge.

Figure 6 highlights the improved performance (lower kW/ton) when building cooling loads are less than 400 tons.

SPACE GAP

Figure 6. Pre- and post-installation power and load comparison for the entire chiller plant.



In Figure 6, the green line shows performance relative to an efficacy of 0.75 kW/ton. It is of particular interest to see that loads less than 500 tons show the greatest improvement (reduced kW/ton).

While the data suggests some separation, things start to converge at loads greater than 500 tons. Higher loads demand that the energy to move more chilled water and condenser-water also increases. Tower fan speeds also increase due to higher wet-bulb temperatures when these higher-load conditions are seen. Thus the convergence.

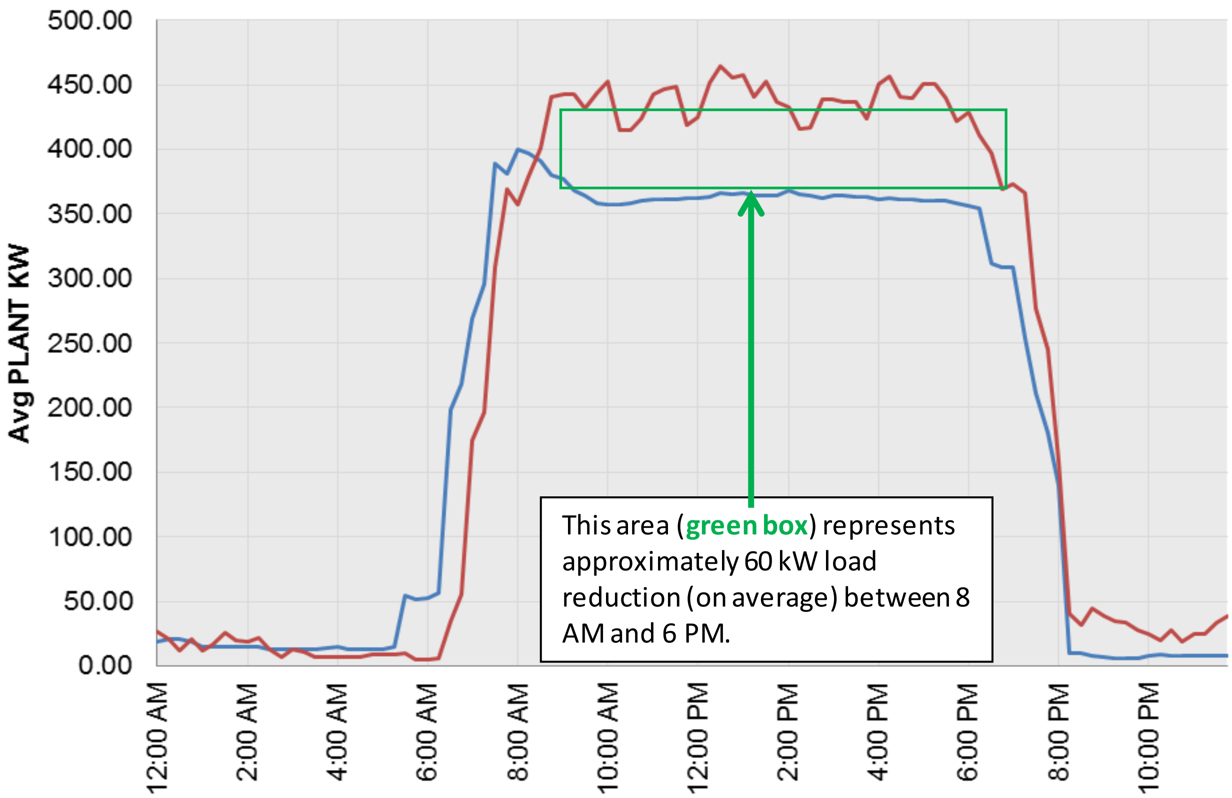
However, during lower-load conditions, the ability to vary pump speeds (speed reduction) is evident. This highlights the technology's benefit when lower-load periods prevail.

SPACE GAP

B. CHILLER PLANT POST-INSTALLATION AVERAGE PERFORMANCE PROFILE

As Figure 7 shows, between 9 AM until almost 6 PM over the 8-month monitoring period, a conservative decrease in energy of 60 kW can be seen once the control optimization system technology was installed.

Figure 7. Pre- and post-installation power comparison for the chiller plant time averaged over the 8-month monitoring period (Jan. 2013 through Sept. 2013).

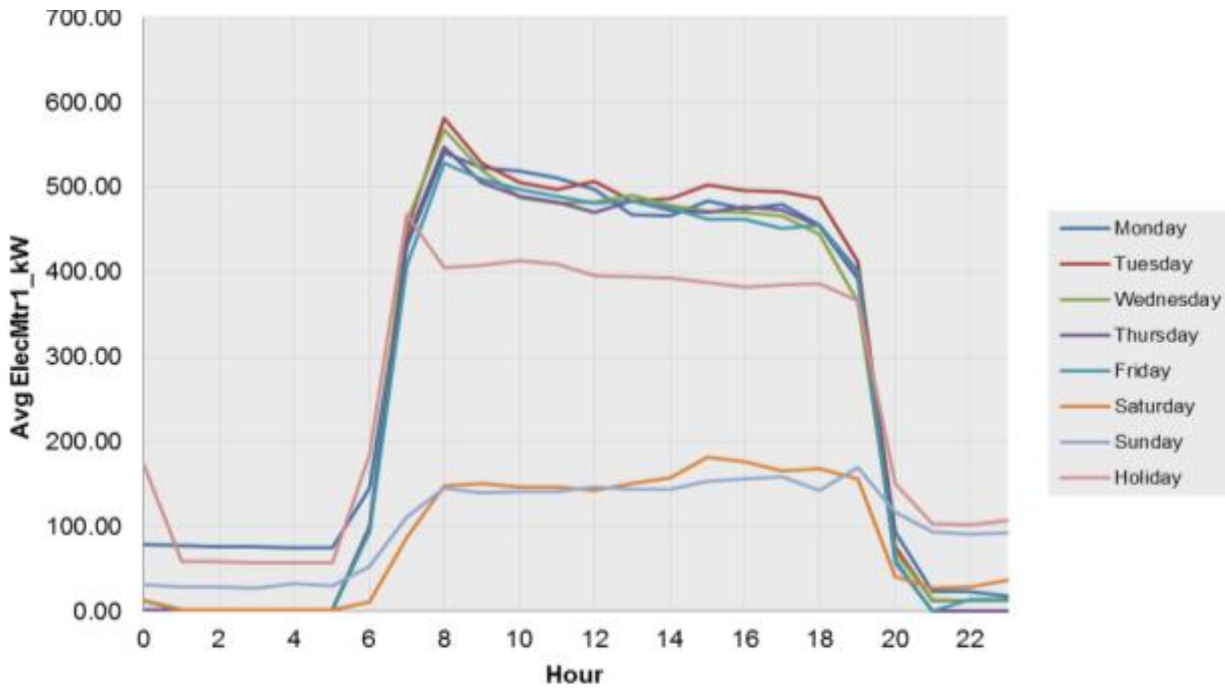


*Chiller Plant average kW **Red** = Pre-Simulation, **Blue** = Post*

C. CHILLER PLANT POST-INSTALLATION DAILY PERFORMANCE PROFILE

Figure 8 shows the monitored hourly chiller plant performance post-installation for each day of the week, including holidays.

Figure 8. Daily performance of the chiller plant post-installation.



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The daily profiles shown in Figure 8 should match the profile shown in Figure 7 (post-installation blue line). When averaged together (including weekends and holidays), the results closely match Figure 7.

Figure 8 data suggests potential additional opportunities for improvement, either in the site operations or in the control optimization system (or both).

- Federal holidays show power consumption at levels approaching occupied weekday periods. The holiday period closely represents the average for all occupied time periods (Figure 7).
- Sunday nights and holiday nights indicate that energy is being consumed after 8 PM (all night), and Monday mornings show significant energy consumption from midnight until 5 AM. This is assumed to be related to O&M staff overrides through the BAS controls.
- The holiday and weekday morning cool down period from 6 AM until 8 AM shows a higher demand spike than the rest of the 24-hour period. This may be due to the higher demand required for cooling down a warm loop or building, or it may be some other anomaly in the controls, etc. A morning spike as shown is not abnormal, followed by a decline through the mid-morning period. However, an afternoon spike also would be expected for a building with a high afternoon cooling load (like Montgomery, Alabama), but the data does now show this.
- The weekend period shows what would be expected for a typical morning cool down with an afternoon spike, before shutting down in the early evening. These data plots may indicate extreme

over-cooling of the building during weekday morning periods. This should be evaluated for further improvement potential if this operation is occurring without merit.

D. WAYS TO FURTHER IMPROVE CHILLER PLANT PERFORMANCE

Methods to improve chiller plan performance, discussed below, are relative to observations made of the Frank M. Johnson Jr. Federal Building and U.S. Courthouse chiller plant. These methods also should be considered relative to other chilled water plant operations with similar design (water-cooled chillers using cooling towers) applications. The efficiency of a chiller plant is a relative function of three primary variables:

- ***Cooling tower leaving water temperature setpoint reset***: The optimal cooling tower water temperature to the chillers is a function of the outdoor air wet-bulb temperature (OAWBT), which is often overlooked in the automatic control sequences. As OAWBT increases, the cooling towers must work harder to maintain the desired entering condenser water temperature (ECWT) setpoint. Due to chiller designs and the need to maintain a maximum compressor lift (delta between entering condenser water and leaving chilled water temperatures), the cooling tower fans will run at maximum capacity, when required. This is seen in the warmer/humid summer days. However, during cooler/dryer conditions, as the OAWBT values decrease, the cooling towers can be automatically managed (by resetting the ECWT setpoint) to run more efficiently. In some cases, the OAWBT is sufficiently low to allow for an automatic setpoint reset that drives the ECWT even lower, without incurring higher costs (increased tower fan power penalties) that exceed the savings at the chiller compressor. Unfortunately, this improvement opportunity is often overlooked.

- ***Chilled-water supply temperature setpoint reset***: When values are set too low for the actual load requirements, the over-cooling of chilled water impacts the chilled water plant (increased tower loading, greater refrigerant lift requirements for the compressor motors, and greater potential for zone/space over-cooling), resulting in excess reheating and occupant complaints. Automated resets should take into account true load conditions that reflect demand and help optimize chiller lift. It also is imperative that humidity sensors are accurately calibrated and maintained. If not done correctly, this strategy can transfer energy to AHU fans across the building.

- ***Improved Optimal Equipment Runtime***: Equipment run-time is often based upon automatic scheduling for daily load requirements, automatic override for setback requirements, and finally, operator override based upon perceived need or load, to which the BAS control did not adequately respond or was not allowed the opportunity to respond. When chiller plants are run beyond their "expected" hours of operations, based on "perceived" need or often for "just-in-case" scenarios, this also results in system inefficiencies and additional energy consumption, increased wear-and-tear, and shorter lifespans of critical and expensive equipment. Control optimization system's control sequences are designed to mitigate chiller and pump short-cycling. The minimum run-time values provided in the updated control sequences may be helping to decrease equipment short cycling, but they also could be creating added efficiency losses by running pumps longer than needed or required (pumping systems were observed to be running when no load is apparent before the building occupancy period begins or long after it ends). All the pumps and tower fans are VFD-driven and VFDs offer soft-start capabilities, which should lessen impacts due to frequent cycling of pumps. Minimum run-time values should be evaluated for possible reduction.

COOLING TOWER LEAVING WATER TEMPERATURE SETPOINT RESET

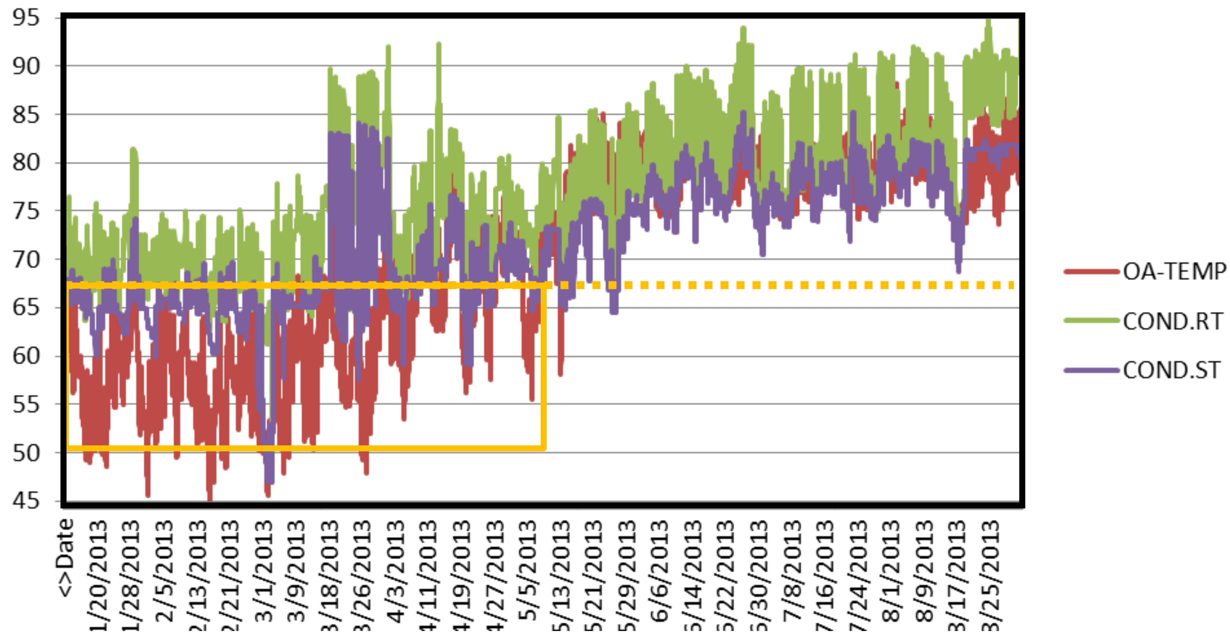
In general, lowering the ECWT will reduce the refrigerant head pressure, thereby reducing the load on the compressor and raising the COP (Thumann 1991). The potential improvement in COP is dependent on the type of refrigeration compressor. Further, the type of refrigeration compressor and type of load control mechanism will limit the extent that this strategy can be employed. For conventional centrifugal chillers, Thumann (1991, p 153[8]) estimates the increase in COP to be around 0.4% per 1°F reduction in the ECWT.

Cooling tower equipment designs and level of maintenance also will impact the strategy that can be used to achieve an optimal ECWT. The cooling tower's design approach will impact the ability to optimize the ECWT seen by the chillers. The design approach is the difference between the OAWBT and the cooling tower's evaporative cooling capacity, which can be as much as 8°F–10°F. This means that OAWBT values of 70°F will result in ECWT values of 78°F to 80°F. The design approach assumes that the tower cells, spray nozzles, and other flow devices are properly maintained along with good water treatment and filtration.

During periods of low OAWBT conditions (< 55°F), the controls should be configured to take advantage automatically of the opportunity to provide lower ECWT. Manufacturer specifications should be reviewed before employing this strategy. Lowering the ECWT too much may have adverse effects on the compressor operation with regard to head pressure and surging in certain makes and models of centrifugal compressors. The chiller manufacturer can assist in determining the lower (and upper) limit for the ECWT. Trying to maintain a lower condensing water temperature can be achieved (or at least attempted), but this may come at the expense of higher tower fan speeds and increased tower fan energy consumption. This needs to be balanced with savings that can be achieved at the chiller equipment, based upon consultation with the chiller vendor's recommendations.

[8] 8 Based on constant condenser pumping. Variable condenser pumping can increase chiller efficacy 1-2%.

Figure 9. Post-installation monitored condenser-water temperature versus outdoor air temperature.

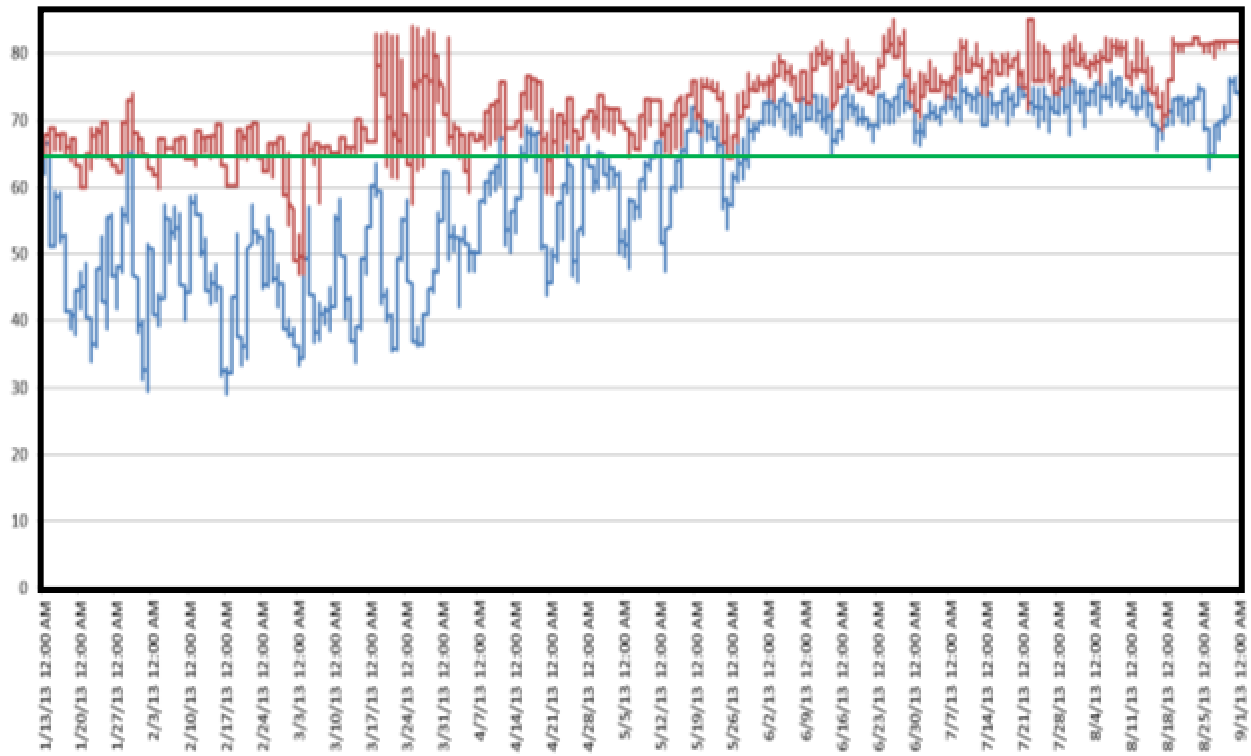


The **orange dashed line** shown in Figure 9 indicates the minimum cooling tower leaving water temperature setpoint (assumed) as the condenser water leaving temperature (**COND-ST/purple line**) hovers near this value, even during cooler OA dry-bulb (**OA-TEMP/red line**) conditions.

The orange box in Figure 9 indicates the optimum outdoor air temperature (OAT) value that would allow for minimum temperature setpoint for the cooling tower leaving water (if a reset were implemented). This would typically be during the months of October–April. Conversely, if the ECWT is not reset, based upon the dynamic OAWBT values, the tower fans may be forced to operate at full speeds, with no benefit relative to the actual achievable setpoint (not being reset upwards as the OAWBT value concurrently increases, as seen in the May-September time frame).

The Frank M. Johnson buildings do not have an OAWBT sensor, but the value can be calculated from their outdoor temperature and outdoor humidity sensors (as shown in Figure 10).

Figure 10. Post-installation monitored condenser water supply temperature versus outdoor air wet bulb temperature.



Presently, the chiller plant with control optimization is achieving a 65°F minimum temperature value, as illustrated in Figure 10. Occasionally, the condenser water temperature (**red line**) can be seen dropping below 65°F (as low as 60°F).

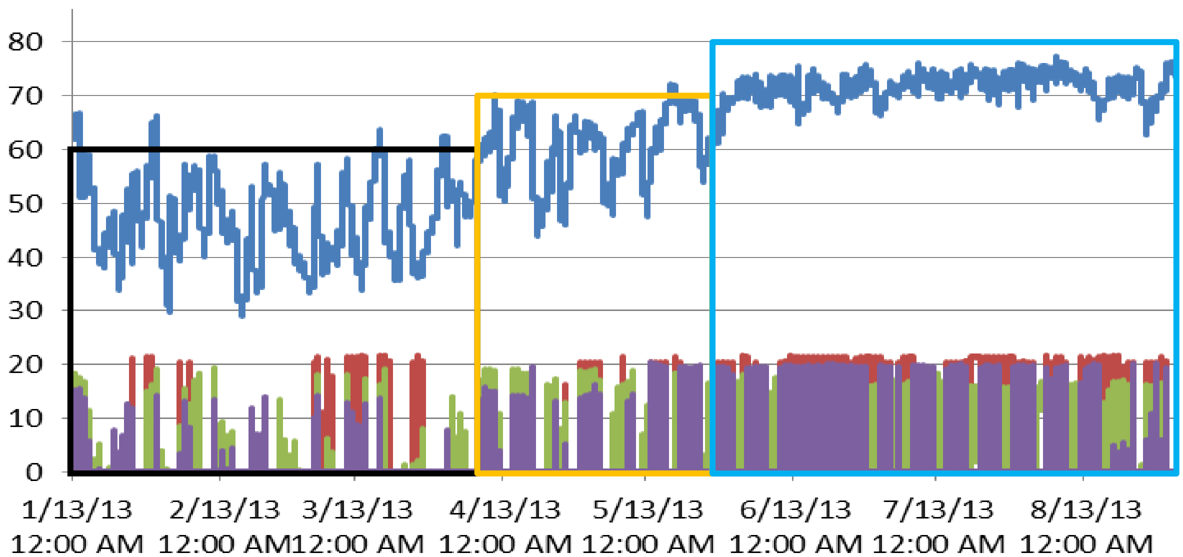
However, at lower wet-bulb temperatures and lower cooling loads, the ECWT exhibits a large variance and deviates from the 65°F temperature, as indicated by the calculated OAWBT (**blue line**) compared to the assumed cooling tower leaving water temperature setpoint (**solid green**) in Figure 10. It may be possible to further reduce the ECWT. As mentioned previously, the chiller manufacturer should be consulted to determine the allowable low limit before this possible improvement is pursued.

Based on the trended condenser water temperature performance, the setpoint over the eight months of monitored operations is assumed to be near 65°F. Only during the March pre-installation simulation testing did the condenser ECWT drop below 60°F. This was assumed to occur because of the simulated 100% tower fan speeds, but may have been caused by other configuration parameters that existed in the controls prior to the optimization technology installation.

Over-cooled condenser water temperatures can lead to chiller surging and other unwanted anomalies. One of the benefits of control optimization is the ability to run cold condenser water temperatures while maintaining chiller lift, thus avoiding chiller surging. During warmer weather, the delta between the OAWBT and the condenser supply temperature is between 5°F–8°F, which indicates an appropriate cooling tower system approach design.

Figure 11 shows the fan power for the three cooling tower VFD-driven fans and peak power is noted as approximately 20 kW per cooling tower fan. When OAWBT values are predominantly less than 60°F (**Black Box**), the cooling tower fan kW consumption (**red**, **green** and **purple** lines) indicates lower values and less frequent in duration. When OAWBT values are predominantly between 60°F and 70°F (**yellow box**), the values increase, as well as the duration. Once the OAWBT values are consistently greater than 70°F (Light Blue Box, June through August), the cooling tower fans all appear to be running near-simultaneously and near their maximum load values.

Figure 11. Post-installation monitored cooling tower fan power demand versus OAWBT.



It is assumed that if the cooling tower leaving water temperature setpoint were configured to increase automatically according to the OAWBT, this would better match the tower fan energy consumption to the tower's design capabilities (versus running tower fans at full speed with no benefit for the energy being expended to meet a setpoint that is physically not attainable).

With VFD-operated tower fans and staged towers (via individual tower isolation valves), the ability to match part-load tower capacity should be evaluated (cooling tower leaving water temperature setpoint reset). The O&M staff on site should evaluate the cooling tower BAS or control optimization system controls for an automatic reset of the leaving water temperature setpoint.

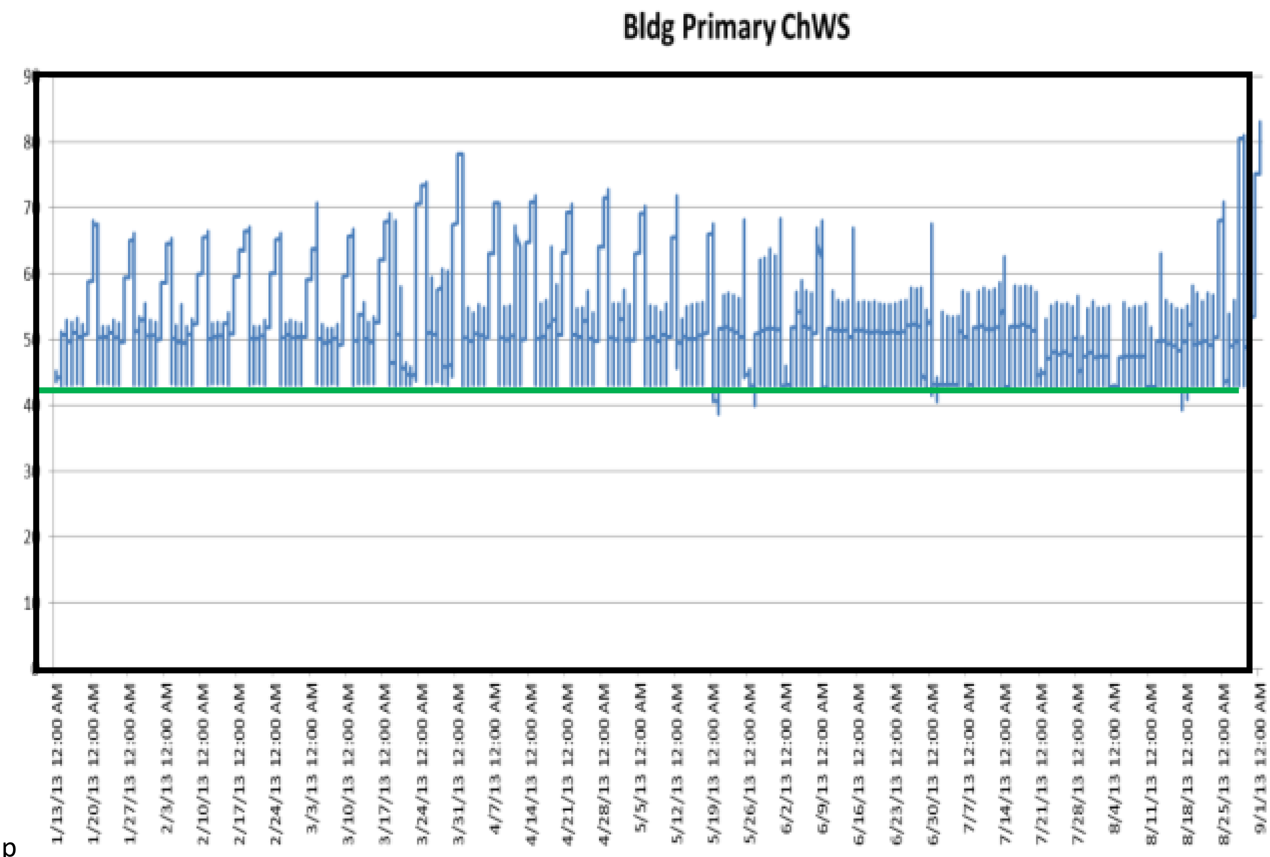
CHILLED-WATER SUPPLY TEMPERATURE SETPOINT RESET

Another method to improve chiller plant performance is to increase the chilled-water supply temperature (chWST) during low-load conditions. Raising the chilled-water supply temperature will allow the evaporator refrigerant pressure to be increased, thereby reducing the load on the compressor and raising the COP (Thumann 1991). The potential improvement in COP is dependent on the type of refrigeration compressor. Further, the type of refrigeration compressor and load control will limit the extent this strategy can be employed. For conventional centrifugal chillers, Thumann (1991, p 155) estimates the increase in COP to be around 2% per 1°F increase in the chilled-water supply temperature.

The primary concern with increasing the chilled-water temperature is the potential reduction in the ability to dehumidify. Under design conditions, chilled-water supply temperatures are kept excessively low to allow the cooling coil also to act as a dehumidifier by lowering the air temperature to the point where moisture condenses out of the air stream. However, when air does not need to be dehumidified, there should be a lower demand for excessively low chilled-water temperatures. These lower demand conditions also should exist during cool spring and fall months (winter/heating).

As can be seen in Figure 12, the monitored period of chilled water supply temperatures indicates that the chiller plant was producing approximately 42°F –43°F chilled water temperatures. As already noted, these temperatures observed in the January to the April or May time frame appear to be too cold. An automatic chilled water supply temperature setpoint reset is highly encouraged.

Figure 12. Post-installation monitored chilled water supply temperature.



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VI. Summary Findings and Conclusions

The summary findings and conclusions are discussed below, including information on best practices to assist in optimizing chiller plants. The barriers to technology adoption and market potential also are discussed. Finally, this section provides recommendations regarding the future installation of the technology and the importance of a measured baseline.

A. OVERALL TECHNOLOGY ASSESSMENT AT THE DEMONSTRATION FACILITY

The application of the control optimization system technology offers the potential for reducing energy consumption in space cooling applications by optimizing variable-speed chiller plant equipment (primarily pumps), leading to increased chiller plant efficiency. In general, the operating efficiency of a typical water-cooled chiller plant with the new control technology is estimated by the vendor to be around 0.6 kW/ton, resulting in an average annual energy reduction between 25% and 50%. Factors impacting the actual performance include climate zone, building type, hours of occupancy, and other part-load impacts.

The average chiller plant efficiency throughout this demonstration was 0.64 kW/ton, with a projected annual energy savings of 35%. Due to the limited data set from the brief Manual Mode period, the projected savings has a confidence level of 80%, resulting in a total uncertainty of (+/-) 10.8%. This means the projected savings could be as low as 24.2% or as high as 46.8%.

On a weather-normalized basis, the resulting analysis estimates the data point will reduce the chiller plant energy consumption from a projected baseline value of 1,495.96 MWh/yr to a projected value of 972.51 MWh/yr, with an energy savings of 523.44 MWh/yr, or 35%, over the projected baseline.

The annualized cost savings are projected to be $42,485. With the level of uncertainty already noted, the annualized cost savings varies from $28,962 to $56,009. Based on the performance data collected and evaluated from the Frank M. Johnson Jr. Federal Building and U.S. Courthouse demonstration, control optimization is projected to be capable of achieving its performance claims.

It is recommended that the energy and cost savings associated with the technology be evaluated for variable-speed chiller plants that already have the required VFDs installed, thereby reducing the total installed costs (and becoming economically more viable). However, the potential energy savings for variable-speed chiller plants is likely to be lower than for the constant-speed chiller plant evaluated in this demonstration because it is expected that most BAS configurations applied to chiller plants will have some energy-saving attributes (e.g., temperature and pressure resets) already embedded.

Based on the economic assessment for this demonstration, the targeted deployment of the control optimization is recommended only where a site-specific engineering analysis determines that the technology is likely to be life-cycle cost effective. Specific guidance for targeted deployment of this technology is provided in Section VI.D.

This demonstration's projected energy savings is in the mid-range of the 25% to 50% reported savings range, so it is unlikely that additional energy savings alone will make this technology cost-effective. However, control optimization should be evaluated for other locations where energy rates are high and installed costs can be reduced, or where utility rebates can be obtained to help further reduce the overall installed costs. In addition, the deployment of the technology for variable-speed chiller plants, which would not require the costly VFD upgrades, should be evaluated for life-cycle cost-effectiveness (with the further assumption that the variable speed chiller plants are already enjoying significant energy and cost-savings, thereby impacting the economics).

B. BEST PRACTICES

IMPROVED LOOP DELTA-T

A chiller plant needs to be capable of meeting the building's peak demand, as well as operating efficiently under part-load conditions. A chiller plant with control optimization offers excellent part-load efficiency because of the integration of VFDs into the optimized pump and cooling tower fan operations. One means of measuring part-load efficiency is to evaluate the different pumping loop Delta T values.

Delta T has been the subject of much discussion in the HVAC industry. "Low Delta T" syndrome, in particular, is an undesirable condition that afflicts many chilled water plants, for many reasons.

The chiller plant using control optimization in this study showed Delta T values that ranged from 8°F –18°F. The chilled water loop Delta T values (primary and secondary) were consistently found to be operating above 10°F, even during part-load (and low-load) conditions. This is significant because it indicates the chiller equipment is seeing a good load and operating in a range that most likely offers the greatest efficiency performance for the chiller(s). It is not uncommon to find chiller plants across the country operating with Delta T values that are less than 5°F.

In addition, while the chiller plant with the new control optimization technology was highly efficient at part-load, there is still room for additional improvement, and some of the best practices discussed below may apply.

DASHBOARD FEEDBACK

In general, BASs provide an optional dashboard that is capable of providing considerable feedback to the operator on the equipment's operating conditions. GSA's *Facilities Standards for the Public Buildings Service* (P100) specifies that new chillers must provide input/output (kW/ton) information, as well as monitoring energy consumption (GSA 2010).

The BAS system in the chiller plant evaluated in this study provides such an interface for displaying chiller plant performance in kW/ton. However, it is recommended that the dashboard be programmed to include trending the kW/ton versus plant load to provide long-term performance feedback. This will require minimal effort, and no hardware procurements are required. Trending the kW/ton of the chiller plant's chilled-water pumps, condenser-water pumps, and cooling tower fans would provide useful insight regarding the performance of the equipment relative to the entire chiller plant. Trending (and alarming) loop Delta T values that persist below lower limit values (<3°F–5°F) will alert operations staff to unwanted and harmful anomalies. This information will allow operators to better optimize the chiller plant operation, configure alarms for "out-of-normal" operations (such as the low chilled water Delta T issues), or be alerted to unwanted equipment configurations where a VFD-driven pump is placed in Manual Mode operations at the VFD, resulting in lack of control from the BAS. This actually occurred at the demonstration site and ran the pump 24/7 for over three weeks. The historical data was being reviewed when this condition was found and restored to normal.

GSA has invested in its own dashboard of sorts. This technology is called GSALink and is part of the Intelligent Buildings initiative. Sites like the Frank M. Johnson Federal Building and U.S. Courthouse should be evaluated for possible inclusion into the GSALink system (if not already included). This would provide for automatic fault detection and analysis of not only the chilled water plant (and the control optimization system technology), but also of all the other major building HVAC systems.

C. BARRIERS AND ENABLERS TO ADOPTION

The most recognized barriers to this technology include cost, familiarity, training, and procurement specification issues. The control optimization system requires VFDs on all chiller plant equipment for upgrading constant-speed chiller plants, a requirement that may be costly. Some older motors that serve pumps or tower fans may not be able to accept VFD technology, which may limit the application of the new optimization technology when procuring newer motors is not possible. Because the technology is modular, it can be installed on parts of the chiller plant, such as the condenser water distribution system.

Conventional constant-speed chiller plant operators are likely to be resistant to the “variable flow” operating approach associated with this new technology and, therefore, training will be crucial to the success of this technology’s adoption. Operators should be trained on how control optimization saves energy, so that they fundamentally understand why operating variable-speed pumps and variable-speed tower fans at part load is more efficient than at full load.

Procurement and specification issues may still be a potential barrier to widespread deployment. The optimization technology is offered by only one vendor, who also happens to own and distribute its own BAS system. It is not clear if more competition from other vendors will be forthcoming in the future. The vendor whose technology was used in this study claims that its control optimization system will interface with almost any other BAS vendor system. In many cases, an integration package may be required to allow the optimization technology to communicate with (integrate to) non-Siemens BAS systems. To date, control optimization has been installed at 27 federal sites—with two sites using Tridium controls and one site using JCI controls.

D. MARKET POTENTIAL WITHIN THE GSA PORTFOLIO

The control optimization technology claims to be capable of integrating into almost any centrifugal chiller plant that utilizes multiple chillers. While this analysis did not concern itself with this claim (see BARRIERS, above), this technology is suitable for new construction and retrofit applications alike. This technology lends itself particularly well to retrofit applications because upgrading an existing chiller plant to include the control software and VFDs can be accomplished without waiting for the current equipment to fail or be replaced.

GSA sites with chiller plant operating efficiency problems, low-load problems or low Delta-T syndrome problems should all be screened initially for possible use of this technology.

This technology is recommended for potential deployment for locations that meet the following guidelines[9]:

- Electricity rates ($/kWh) in the top 50% of GSA's portfolio (>$0.10/kWh blended rate)
- BACnet-enabled BAS (versions more than five years old will require additional programming)
- Cooling season or process loads that require chillers to operate 8–10 months a year or longer
- Buildings having cooling loads greater than 500 ton-hours/day, 3 million tons per year
- New construction facilities that meet above criteria[10]
- Existing central plants with three or more constant-speed centrifugal chiller flow systems (primary/secondary pumps, condenser pumps and cooling tower fans)

In addition, deployment of control optimization at locations with the following characteristics are not likely to be cost-effective:

- Electricity rates ($/kWh) in the bottom 50% of GSA's portfolio
- The cooling season requires chiller to operate six months or less based on a typical meteorological year (TMY)

Insufficient resources are available to identify the number of cost-effective candidates throughout GSA's portfolio of buildings. It is recommended that control optimization be part of a recommended suite of possible chiller technologies for consideration during facility energy assessments, which are required by EISA for every covered GSA facility every four years[11] (EISA 2007).

It is estimated that the market potential within GSA for the application of chiller plants with the new optimization technology could reduce annual energy consumption by around 281,275 MMBTU. Based on GSA's average utility pricing, this equates to annual savings of $9,008,688. This estimate is based on the following assumptions:

[9] Prior to installing the technology, a detailed site-specific engineering analysis is recommended, which should include a monitored baseline for the existing chiller plant, as well as ease-of-integration into the site's BAS.

[10] This assessment was not intended to determine if the studied optimization technology would be cost-effective for new variable-speed chiller plants. It is recommended that variable-speed chiller plants be evaluated using a detailed engineering analysis, including a monitored baseline.

[11] EISA section 432 requires agencies to identify their facilities that constitute at least 75% of the agency's facility energy use, also referred to as "covered facilities," and complete comprehensive energy and water evaluations of 25% of covered facilities each year, so that an evaluation of each facility is completed at least once every four years (Reference: FEMP web site at http://www1.eere.energy.gov/femp/regulations/eisa.html).

- GSA Chilled Water Consumption – 803,643 MMBTU (10,045,532 total GSA energy use (EUAS, 02/16); 10% for Cooling (CBECS 2012) 80% for Chillers (chillers provide majority of conditioning for buildings > 200,000 ft^2, EIA 2003).
- Chiller plants using the demonstrated control optimization system technology can reduce space cooling energy consumption by an estimated 35% compared to constant-speed chiller plants (consistent with the findings in this report).
- GSA average electricity price, $0.1092 kWh, EAUS FY2015.

E. RECOMMENDATIONS FOR INSTALLATION, COMMISSIONING, TRAINING, AND CHANGE MANAGEMENT

As for any new control optimization technology, it is recommended that measurement and verification of performance be tracked to ensure energy savings and operational claims are met and maintained continually. Attention should be paid to ensure the appropriate sequencing of equipment is occurring and that the sequence is optimized based on the particular profile of the chiller plant. A convenient dashboard is typically provided in conjunction with this technology; the dashboard is capable of long-term trending and performance tracking to meet this need. Since this control technology relies on sensor measurements as input for controlling the operation of the chiller plant, it is recommended that sensors be checked for calibration annually.

The capabilities of the technology (including chilled-water reset, condenser-water reset and maximized part-load efficiency) need to be utilized to their full extent at each site to ensure optimum energy savings are attained.

Generally speaking, equipment life tends to be related more to load hours than operating hours. In other words, equipment life depends more on how hard the equipment works, as compared to how long it runs. Therefore, although the application of the optimization technology will increase the operating hours for the chiller plant equipment, the load hours on the individual equipment may be decreased due to reduced speed operation of pumps and tower fans. Control optimization technology requires VFDs, which in general, reduce wear associated with equipment start-up.

Therefore, the O&M costs associated with deploying the new optimization technology are not expected to increase. However, equipment should be monitored periodically for unexpected degradation in performance. Also, ongoing O&M costs should be tracked and periodically compared to historic records.

F. IMPORTANCE OF BASELINE MEASUREMENT AND DOCUMENTATION

The control optimization technology was installed prior to this demonstration. Therefore, a monitored baseline was not available. Prior to the installation of the technology, baseline estimated data for the constant-speed chiller plant was estimated by the vendor, based upon a snapshot of the plant's performance and the vendor's internal modeling software.

The lack of a reliable and accurate monitored baseline required the development of a baseline that may vary from the real-world baseline performance, as a result of the multiple technical assumptions, which may or may not be completely accurate or valid. It is recommended that future evaluations include a monitored baseline, sufficient in duration to characterize properly and fully the energy performance profile of the system under examination, thus providing a more reliable basis for comparison and more accurately determining energy and cost savings estimates.

Investing in a more accurate baseline (preferably developed by an independent source) likely would have supported a better conclusion regarding the potential cost-effectiveness of this technology. The simulated peak-power baseline developed for the assessment of the optimization technology does not likely accurately represent the true baseline performance profile of the chiller plant prior to the installation of the technology. The error associated with this baseline cannot be accurately determined.

VII. Appendices

A. LIST OF ABBREVIATIONS AND SYMBOLS

The following is a list of abbreviations and symbols used throughout this report.

Term	Description
AHRI	Air Conditioning, Heating and Refrigeration Institute
A, amp	ampere (electric current)
ANSI	American National Standards Institute
ASHRAE	American Society of Heating, Refrigerating, and Air Conditioning Engineers
ARRA	American Recovery and Reinvestment Act of 2009
BAS	building automation system
Btu	British thermal unit
Btu/h	British thermal units per hour
CBECS	Commercial Buildings Energy Consumption Survey
CFM	cubic feet per minute
CH	Chiller
CHW	Chilled water
CHP	chilled-water pump
ChWST	chilled water supply temperature
COP	coefficient of performance, a unitless measure for efficiency
C_p	specific volume, Btu/lbm-°F
CWP	condenser-water pump
DEER	Database for Energy-efficient Resources
DOD	Department of Defense
DOE	U.S. Department of Energy
ECIP	Energy Conservation Investment Program
ECWT	entering condenser-water temperature
EIA	U.S. Energy Information Agency
EISA	Energy Independence and Security Act of 2007
FEMP	U.S. DOE Federal Energy Management Program
gal	gallon
GPG	Green Proving Ground
gpm, GPM	gallons per minute
GSA	U.S. General Services Administration
h, hr	hour
hp, Hp	horsepower

Term	Description
HVAC	heating, ventilating, and air conditioning
lb, lbs	pounds
lbm	pound mass
kW	kilowatt
kWh	kilowatt-hour
kW/ton	power input per capacity, an inverse efficacy measure
LCC	life-cycle cost
MWh	megawatt-hour
M&V	measurement and verification
N/A	not applicable
NO	normally-open
NPV	net-present value
OAWBT	outdoor air wet-bulb temperature
O&M	operations and maintenance
PNNL	Pacific Northwest National Laboratory
$\dot{q}$	volumetric flow rate, gpm
Q_{in}	input power, in Btu/h
Q_{cool}	cooling energy rate, in Btu/h
Q_{tons}	cooling energy rate, in tons
RPM	revolution per minute
SIR	savings-to-investment ratio
TMY	typical meteorological year
TMY3	a third update of TMY data for 1020 locations based on data from 1991 to 2005
T_{ret}, T_{return}	return temperature
T_{sup}, T_{supply}	supply temperature
VFD	variable-frequency drive
yr	year
°	degree

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C. GLOSSARY

The following is a glossary of advanced terminology in support of this report.

Terminology	Description
Affinity laws	A set of three mathematical equations that define the relationship between rotational speed, flow rate, pressure rise, and brake horsepower for centrifugal systems.
Air-cooled chiller	A refrigerating machine in which heat removal is accomplished entirely by heat absorption by air flowing over condensing heat exchanger surfaces (condenser is air cooled).
Air-handling unit	A device used to condition and circulate air as part of a heating, ventilating, and air-conditioning (HVAC) system
Annual load profile	A measure of the time distribution of thermal or other energy loads, such as Btu/h, for a system over the course of a year.
Baseline	Typically referring to an energy profile, model and/or characteristics before changes have been made to a system for the purpose of modeling the original operating condition, derived from measurements are taken over a period of time and used as a basis for comparison to one of more options or alternatives.
Bin data	A data pre-processing technique used to reduce observation errors.
Building automation system (BAS)	A computerized network of electronic devices designed to monitor and control the mechanical, electronics, and lighting systems in a building.
Capacity	Intended technical full-load (a.k.a., maximum) sustained output of a facility, system, or device. Typically quantified as a power rating or rate of energy transfer. May also be known as nameplate capacity, rated capacity, nominal capacity, or installed capacity.
Central chiller plant	A single or common chilled-water facility consisting of one or more chillers used to serve one or more facilities.
Centrifugal compressor	A dynamic (or non-positive-displacement) compressor that uses rotational (centrifugal) forces to raise pressure in the refrigerant system.
Chilled-water pumps	Pump and motor system used to circulate chilled water for space or process cooling distribution systems.
Chilled-water return temperature	The temperature of the chilled water entering the evaporator or returning from the facility distribution system.
Chilled-water supply temperature	The temperature of the chilled water leaving the evaporator or being supplied to the facility distribution system.

Terminology	Description
Chilled-water temperature reset	An operating strategy in which the chilled water supply temperature setpoint is varied in response to a control signal or varying sensor input.
Chiller	A machine that removes heat from a liquid via a vapor-compression or absorption refrigeration cycle.
Climate zone	A region with similar weather characteristics.
Coefficient of Performance (COP)	The ratio of the heating or cooling provided divided by the electrical energy consumed, as a unitless measure. The COP provides a metric of performance for heat pumps that is analogous to thermal efficiency for power cycles.
Commission (commissioning process)	A quality focused process for enhancing the delivery of a project. The process focuses upon verifying and documenting that the facility and all of its systems and assemblies are planned, designed, installed, tested, operated, and maintained to meet the Owner's Project Requirements.
Condenser	The part of a refrigerant system where refrigerant is liquefied by removal of heat through use of a heat sink.
Condenser-water temperature	The temperature of the water departing the cooling tower and/or entering the water-cooled condenser.
Condenser-water temperature reset	An operating strategy in which the condenser-water temperature setpoint is varied in response to a control signal or varying sensor input.
Condensing coil	A heat exchanger in which the refrigerant rejects heat to the point where the refrigerant condenses from vapor to liquid.
Condenser-water pump	Motor-driven pump used to circulate water through the condensing system
Constant speed	A motor system in which the motor's rotational speed remains (relatively) constant--may be applied to a pump, fan or chiller.
Constant volume (pumping system)	A pumping system in which the pump speed remains (relatively) constant.
Cooling-degree day	The difference in temperature between the outdoor mean temperature over a 24-hour period and a given base temperature. For cooling degree-days, includes data when the mean daily temperature is above the given base temperature. Unless otherwise noted, the base temperature is typically 65°F.
Cooling tower	A heat removal device used to transfer (reject) heat to the atmosphere via the evaporation (mass transfer) of water.

Terminology	Description
Cooling-tower-water temperature reset	An operating strategy in which the condenser-water temperature (water temperature returning from the cooling tower) setpoint is varied in response to a control signal or varying sensor input.
Current transformer	Sensor used to measure the electric current in a wire
Dashboard	A display capable of providing operational data or information on a system or unit of equipment either in real time or near real time.
Design conditions	Specified environmental conditions, such as temperature and humidity, required to be produced and maintained by a system.
Design load	The thermal load expected to occur under design conditions.
Dew point	The temperature below which the water vapor in a volume of humid air at a constant barometric pressure will condense into liquid water.
Dry-bulb temperature	The temperature of air measured by a thermometer freely exposed to the air but shielded from radiation and moisture. Dry-bulb temperature is the temperature that is usually thought of as air temperature.
Economizer (HVAC)	A duct and damper arrangement and automatic control system that together allow a cooling system to supply outdoor air to reduce or eliminate the need for mechanical cooling during mild or cold weather.
Efficiency	Typically energy output divided by energy input, but may also be defined as useful energy output divided by consumed energy input.
Energy-efficiency ratio (EER)	A ratio of the net refrigerating capacity in Btu/h to the power input value in Watts at a given set of rating conditions, expressed in Btu/(h · W).
Energy-use intensity (EUI)	A metric determined as energy consumed within an facility divided by the gross square foot of the facility, expressed as Btu/ft2-yr.
Entering water temperature	The temperature of water, or heat transfer fluid, entering the device or equipment.
Fan coil unit	A small air-handling unit consisting of a heating and/or cooling coil and a fan used to serve a single space (control zone) without a ducted air distribution system. The coils are typically designed for use with heated or chilled water and used to condition air recirculated within the space.
Full load	The nominal peak load that a piece of equipment is designed to carry under design conditions.
Grain (of moisture)	A unit of measure of water vapor; 7,000 grains of moisture per pound of water.

Terminology	Description
Heating-degree day	The difference in temperature between the outdoor mean temperature over a 24-hour period and a given base temperature. For heating degree-days, includes data when the mean daily temperature is below the given base temperature. Unless otherwise noted, the base temperature is typically 65°F.
Heat-transfer fluid	A fluid, such as water or water-glycol solution) used to transfer heat from the heat source (equipment) to the heat sink (load).
Humidity ratio	A measure of the absolute humidity in moist air, expressed as the mass of water contained per mass unit of dry air.
Impeller	The device inside a pump or fan used to increase pressure, and thereby induce flow, when in rotation.
Integrated energy-efficiency ratio (IEER)	A partial-load efficiency measure, calculated with the sum of weighting factors applied to tested efficiencies at four part load conditions: (EER at 25%) X 0.125 + (EER at 50%) X 0.238 + (EER at 75%) X 0.617 + EER at 100%) X 0.02.
Integrated part-load value (IPLV)	A single-number metric based on part-load EER, COP, or kW/ton expressing part-load efficiency for air conditioning and heat pump equipment on the basis of weighted operation at specific increments of load capacities for the equipment. Typically used for ARI rating purposes
Interval data	Data collected with a uniform time period.
Kilowatt	An electric power term, meaning a rate of electric energy consumption.
Kilowatt-hour	An electric energy term, equal to 3412 Btu and 1,000 watt-hours.
latent heat energy	Heat exchanged by a thermodynamic body based on a phase change, such as water condensing into liquid or liquid evaporating into vapor.
Least-squares regression	A statistical model that relates the dependent (y-axis) to the independent (x-axis) variable by minimizing the summed square of the difference between the observed x-y data and the regression.
Leaving water temperature	The temperature of the water, or heat transfer fluid, leaving the device or equipment.
Life-cycle cost	The total discounted dollar costs of owning, operating, maintaining, and disposing of a building or building system over the study period.
Load factor	Actual load or average load divided by full load, typically expressed as a percentage of full or design-full load.
Meter	A sensor used for measurement.

Terminology	Description
Mixed-humid climate	As defined by the U.S. DOE, a region that receives more than 20 inches (50 cm) of annual precipitation, has approximately 5,400 heating degree days (65°F basis) or fewer, and where the average monthly outdoor temperature drops below 45°F (7°C) during the winter months.
Monitoring system	A computer- or electronic-based device that observes or displays, but does not necessarily record, the results of input signals from meters and/or sensors.
Net-present value	The difference in life-cycle cost analysis of two project alternatives.
Nominal (capacity)	The capacity reported by the manufacturer for a specified device under general conditions or recorded and reported by a given test.
Non-standard part-load value (NPLV)	A seasonal average efficiency measure using kW/ton with measurements taken at non-standard conditions outside air temperature, also outdoor air temperature.
Occupied (period)	A period in which a facility is typically occupied as a result of the normal business process.
Original Equipment Manufacturer (OEM)	Refers to the company that originally manufactured the finished product. Also refers to a company that may purchase for use in its own products a component made by a second company.
Plenum	According to ASHRAE, a compartment or chamber to which one or more ducts are connected, that forms a part of the air distribution system, and that is not used for occupancy or storage. A plenum often is formed in part or in total by portions of the building. An air compartment that is attached to, or is an integral part of, a forced-air furnace, which is designed to either distribute the heated air after it leaves the heat exchanger in the case of a supply plenum or collects air that enters the return inlet in the case of a return plenum. A component forming an interface between a ductwork and one or more air terminal devices; by virtue of its design or by the inclusion of accessories, it can also be used to equalize the pressure/velocity across the air terminal device.
Positive-displacement compressor	A compressor type that raises the pressure of a gas by trapping a fixed amount (volume) and forcing (displacing) that trapped volume into the discharge.
Post installation (period)	A period of time after a change to the system, typically compared or in contrast to a baseline period.
Power	The rate at which energy is transferred, used, or transformed; also the rate at which this work is performed. Power is determined as energy divided by time.

Terminology	Description
Power Input per Capacity	The ratio of the power input supplied to the unit in kilowatts [kW], to the net refrigerating capacity in tons refrigeration at any given set of rating conditions, expressed in kW/ton.
Primary distribution system	A chilled- or hot-water distribution system in which the circulation flow through the facility is coupled with the circulation flow through the chilled-water or hot-water generators.
Primary-boost distribution system	A (water) distribution system in which multiple circulation pumps may be installed in series.
Primary-secondary distribution system	A chilled- or hot-water distribution system in which the circulation flow through the facility is de-coupled from the circulation flow through the chilled-water or hot-water generators.
Qualified engineer	A sufficiently trained and experienced engineer; may also include certification from an accredited professional organization or registered as a licensed professional engineer through a state regulatory agency.
Rate tariff	The schedule of charges and fees charged by a provider of energy services, such as a utility.
Reciprocating compressor	A positive-displacement compressor that uses pistons to raise the pressure of a gas or vapor.
Refrigerant head pressure	The pressure of a refrigerant system, typically at the discharge of the compressor.
Replace-at-end-of-life	Used in life-cycle cost analysis as a replacement alternative when the existing equipment is at the end of its useful life, so replacement in not optional. This option may also be called replace-on-failure. This type of life-cycle cost analysis is equivalent to the new construction analysis.
Retrofit	Used in life-cycle cost analysis as a replacement alternative when the existing equipment has considerable useful life remaining, so replacement is optional.
Roof-top unit	A packaged HVAC system designed to be mounted on the roof.
Rotary-screw compressor	A positive-displacement compressor type that uses a rotating mechanism of one or more screws to raise pressure of a vapor or gas.

Terminology	Description
Scroll compressor	A positive-displacement compressor type that uses two interleaving scrolls to raise pressure of a vapor or gas. One of the scrolls may be fixed, while the other orbits eccentrically without rotating, thereby trapping and compressing pockets of fluid between the scrolls. Another method for producing the compression motion is co-rotating the scrolls, in synchronous motion, but with offset centers of rotation. The relative motion is the same as if one were orbiting.
Sensible heat energy	Heat exchanged by a thermodynamic system based solely on a change in temperature.
Sensor	A device that measures a physical quantity and converts it into a signal which can be read by an observer or electronic instrument or system.
Sequencing	The strategy that defines how equipment will operate, such as loading and unloading and when a piece of equipment is placed on line and taken off line.
Service life	The expected usable or economic life (years) expected from a piece of equipment or system. The period of time over which a system continues to generate benefits.
Simple payback	The time in which an investment is recovered, or repaid, through the accumulation of savings, determined as installed cost divided by savings; however, the result must be less than or equal to the service life of the project.
Soft start	A method used with electric motors to temporarily reduce the load and torque in the powertrain of the motor during startup. Normal start up process is extended from less than a second to several seconds for the purpose of reducing the stress than may occur during the starting process.
Specific heat (of water)	Heat capacity per unit of mass. For water, the specific heat is 1 Btu/lbm-°F.
Split system	A packaged HVAC system consisting of two primary components; and indoor system for delivering heating, cooling and ventilation to the control zone; and an outdoor system for heat rejection.
Temperature bin	An interval range of temperature data used for energy analysis.
Testing standard	Common and repeated use of rules, conditions, guidelines or characteristics for products, equipment or systems; methods of measuring capacity, or other aspects of operation, of a specific unit or system of a given class of equipment, together with a specification of instrumentation, procedure, and calculations, typically set forth by a professional technical association.

Terminology	Description
Thermal cooling load	The heat energy needed to be removed from a space to maintain the desired space temperature setpoint.
Thermal efficiency	A dimensionless performance metric for a device, determined by useful thermal energy output divided by added thermal energy input.
Thermal shock	The expansion and contraction that occurs within equipment when exposed to rapid or large temperature differentials. The term is a misnomer in that the event is not sudden (as the word shock would imply) but rather it is the result of long term stresses from being exposed to repeated expansion and contraction or uneven temperature changes.
Ton (cooling)	A unit of measure equal to 12,000 Btu/h, the equivalent energy absorbed by melting 1-ton of ice from solid to liquid at 32°F in a 24-hour period.
Trend line	An approach to modeling the relationship between a scalar dependent variable, Y, and one or more explanatory variables denoted, X. Typically the result of a best fit regression analysis.
Turn-down ratio	The ratio of maximum to minimum operating load factor for a modulating piece of equipment.
Typical meteorological year (TMY)	A typical meteorological year (TMY) is a collation of selected weather data for a specific location, generated from a data bank much longer than a year in duration. It is specially selected so that it presents the range of weather phenomena for the location in question, while still giving annual averages that are consistent with the long-term averages for the location in question.
Ultrasonic flow meter	A transit-time ultrasonic flow meter is a device that utilizes two transducers, which function as both ultrasonic transmitters and receivers, to measure the velocity of a fluid (in a pipe) by measuring the difference in time response of sonic signals through the fluid in both up-stream and down-stream measurements.
Unoccupied (period)	A period outside of normal business-occupancy time. Although the facility may still have occupancy, the occupancy level is significantly below what is considered normal during the business process.

Terminology	Description
Utility	An energy-service provider.
Utility program	An incentive or support program sponsored by a serving utility company.
Variable speed	A system in which the (rotational) speed can vary as a result of changing control parameters.
Variable-air volume	A type of HVAC system in which the supply air flow rate to the conditioned space varies to meet variable thermal loads.
Variable-frequency drive	A type of adjustable-speed drive used in electro-mechanical drive systems to control AC motor speed and torque by varying motor input frequency and voltage.
Water-cooled chiller	A chiller in which water is used in the refrigerant condensing process.
Water-side economizer (HVAC)	A heat exchanger that uses the condenser water side of the system for cooling without requiring the operation of the chiller when conditions are favorable.
Weather normalized	A statistical analytical technique which allows the comparison of corresponding normalized values for different datasets in a way that eliminates the effects of certain gross influences, in this case weather conditions.
Wet-bulb temperature	The (thermodynamic) wet-bulb temperature, also known as the adiabatic saturation temperature, is the temperature a volume of air would have if it were cooled adiabatically to saturation (100% relative humidity) by the evaporation of water into it, with the latent heat being supplied by the volume of air. The wet-bulb temperature is the lowest temperature that can be reached under current ambient conditions by the evaporation of water only.
Zone (thermal control zone, HVAC)	A space (or group of spaces) within a building with heating or cooling requirements that are sufficiently similar so that desired conditions can be maintained throughout using a single controlling device.