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# Variable-speed Oil-free Centrifugal Chiller with Magnetic Bearings Assessment: George Howard, Jr. Federal Building and U.S. Courthouse, Pine Bluff, Arkansas

SA Parker

J Blanchard



The Green Proving Ground 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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Pacific Northwest National Laboratory (PNNL): Kristen Alexander, Susan Arey, Nathan Bauman, Jeremy Blanchard, Jeromy Jenks, Steven Parker, Marcus De La Rosa, Lorena Ruiz, and William Sandusky.

For more information contact:

Kevin Powell  
Program Manager  
Green Proving Ground  
Office of the Commissioner  
Public Buildings Service  
U.S. General Services Administration  
555 Battery Street, Room 518  
San Francisco, CA 94708  
Email: kevin.powell@gsa.gov

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## I. Executive Summary

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 31.9% 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%.)

Because the variable-speed oil-free centrifugal chiller with magnetic bearings is expected to offer higher efficiency compared to rotary-screw chillers, the use of this new technology is expected to reduce energy consumption, thereby assisting the U.S. General Services Administration (GSA) in achieving the energy-use intensity reduction requirements as identified in the Energy Independence and Security Act of 2007 (EISA 2007). This report will assess the energy performance from one installation of a variable-speed oil-free centrifugal chiller with magnetic bearings in a central chiller plant and draw conclusions on how the application of this technology may contribute to further reductions in space cooling energy at GSA facilities.

This report is divided into five sections:

- The first section describes the background and opportunity for the variable-speed oil-free centrifugal chiller with magnetic bearings to reduce space cooling energy consumption at GSA facilities.
- The second section discusses the new technology and how it may reduce energy consumption.
- The third section introduces a demonstration of the technology.
- The fourth section provides a more detailed overview of the demonstration, how the chiller plant operates, and how the two chiller technologies were monitored.
- The fifth section presents the results of the monitoring activity, documents the performance and resulting energy savings, and presents the results of a life-cycle cost analysis. This section also presents additional opportunities to further improve the performance of the variable-speed oil-free centrifugal chiller with magnetic bearings based on observations and lessons learned. The final section draws conclusions from the demonstration results and projects how the GSA may best benefit from its application.

### BACKGROUND

In the U.S., space cooling only accounts for 7.4% of energy consumption in commercial buildings and 9.6% in office buildings (EIA 2003b). However, because space cooling is primarily driven by electricity—a higher cost energy source—space cooling can account for a larger percentage of a facility’s annual energy bill. Within U.S. commercial 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, chillers provide space cooling in 18% of commercial building floor space and 31% of office building floor space (EIA 2003a). Therefore, a more efficient chiller technology offers significant opportunity to reduce annual energy costs for GSA, as well as reducing annual energy consumption.

## TECHNOLOGY OVERVIEW

The Variable-speed Oil-free Centrifugal Chiller with Magnetic Bearings offers several features that make it unique and efficient:

- The compressor uses two impellers on a single high-speed rotating shaft, which is the only moving part in the compressor. The two impellers provide two-stages of compression, which provides high efficiency in a compact design.
- The compressor has a fully integrated variable-speed drive. This feature allows the compressor to be highly efficient, especially at partial load. The shaft with impellers is levitated during rotation using digitally-controlled magnetic bearings. This unique feature reduces the friction and heat caused by conventional bearings, adding to the overall high efficiency, but it also eliminates the need for lubricating oil and the ancillary components required to support the lubricating oil system, which further improves efficiency. The oil-free feature is also credited for the reduced maintenance requirements.
- The compressor is digitally controlled, with microprocessors that manage compressor operation and provide self diagnostics and correction.

Large chillers with centrifugal compressors are known for high efficiency and reasonably high turn-down ratios. However, as the capacity of centrifugal chillers decreases, so does the advantage of centrifugal chillers. Historically, centrifugal chillers were infrequently offered below 300 to 500 tons. Positive-displacement compressors, such as the rotary-screw, tended to dominate applications in the 100 to 300-ton range. This compressor-selection strategy may be changing with the impact of variable-speed drives, which allow for much more efficient centrifugal applications in smaller capacities.

Chillers using the variable-speed oil-free centrifugal compressor with magnetic bearings offer a significant efficiency benefit over chillers with positive-displacement compressors, such as the rotary-screw, in capacities as low as 60 tons—a capacity previously unseen in centrifugal chillers. The chiller using the variable-speed oil-free centrifugal compressor with magnetic bearings offers a rated efficiency in the range of 0.33 to 0.37 kW/ton (integrated part-load value, IPLV), whereas rotary-screw chillers meeting the high-efficiency requirements of Federal Energy Management Program (FEMP) designated products only require an efficiency rating of 0.51 kW/ton (IPLV). For larger equipment capacities, such as for chillers greater than 500 tons, the potential efficiency benefit of this new compressor design is much smaller. Large capacity centrifugal chillers, available from several manufacturers, may be just as efficient and not come with such a cost premium. The FEMP designated product efficiency requirements for large centrifugal chillers is 0.36 kW/ton (IPLV)—a range similar to the variable-speed oil-free centrifugal compressor with magnetic bearings. IPLV, however, is a weighted average efficiency at specific partial load factors under rated conditions. Under real world operating conditions, chillers equipped with the variable-speed oil-free centrifugal compressor with magnetic bearings may still save energy compared to larger conventional centrifugal chillers. This study focuses on the efficiency benefit of chillers using the variable-speed oil-free centrifugal compressor with magnetic bearings over chillers with rotary-screw compressors.

## STUDY DESIGN AND OBJECTIVES

With assistance from the Pacific Northwest National Laboratory (PNNL), GSA identified the George Howard, Jr. Federal Building and U.S. Courthouse as the demonstration location for the variable-speed oil-free

centrifugal compressor with magnetic bearings. Built in 1966, the four-story federal office building is located in the heart of Pine Bluff, Arkansas. Space cooling is provided by a central chiller plant located in the basement. The previous chiller plant consisted of two 150-ton rotary-screw chillers, which were installed in 1993. During 2011, as part of an energy-savings performance contract (ESPC), one of the rotary-screw chillers was replaced with one new 150-ton chiller configured with two variable-speed oil-free centrifugal compressors with magnetic bearings. GSA hired PNNL to monitor and assess the performance of the variable-speed oil-free centrifugal chiller with magnetic bearings at the facility for GSA's Green Proving Ground program.

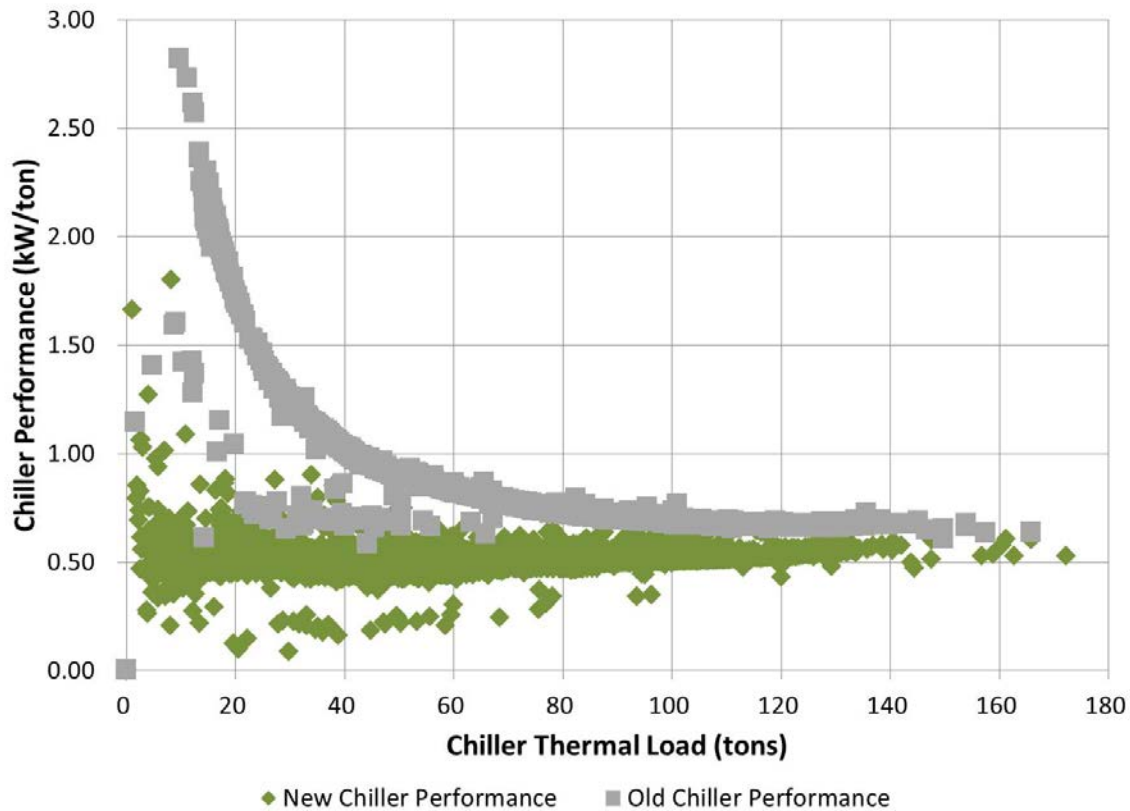
The purpose of the study was to assess operation of the variable-speed oil-free centrifugal chiller with magnetic bearings, quantify the operational performance of the chiller, quantify the load profile for the building application, apply the measured performance to estimate the normalized energy savings compared to the original (remaining) rotary-screw chiller, assess (to the extent possible) the technology's broader application within the GSA portfolio, and report the findings and conclusions. The monitoring plan included recording the electric energy consumed by the individual chillers, which were separately submetered, as well as monitoring and recording the parameters required to quantify the thermal cooling load provided by the individual chillers. The chillers were monitored for six months, from March 1, 2012, through September 12, 2012. The monitored data allowed for determining the operational performance of both chillers, as well as determining the thermal cooling load profile of the building as it relates to occupancy and weather conditions. The development of these profiles supported the weather-normalized assessment of annual energy savings for the new variable-speed oil-free centrifugal chiller with magnetic bearings compared to the original rotary-screw chillers at the George Howard, Jr. Federal Building in Pine Bluff, Arkansas.

## PROJECT RESULTS AND FINDINGS

Analysis of the monitored data shows the new chiller with the variable-speed oil-free centrifugal compressor with magnetic bearings is more efficient than the original chiller with the rotary-screw compressor, especially at lower loads. Figure ES-1 shows the performance profile for each chiller resulting from the monitored data. At high load factors (*i.e.*, greater than 110 tons) the efficiency benefit is measurable, but is not as significant. However as the load factors decrease, the efficiency benefit becomes more pronounced. The George Howard Jr., Federal Building chiller plant operates with a single chiller at less than 33% load for more than 50% of the cooling season. It operates a single chiller less than 50% load for more than 80% of the cooling season. Partial load operation is when the new chiller provides the greatest benefit.

On a weather-normalized basis, the resulting analysis estimates the new chiller with the variable-speed oil-free centrifugal compressor with magnetic bearings has reduced electricity consumption from 294,794 kWh/yr to 170,181 kWh/yr, for a savings of 124,613 kWh/yr, or 42.3% (See Table 5). The results of the life-cycle cost analysis indicate that the new chiller with the variable-speed oil-free centrifugal compressor with magnetic bearings can be cost effective for new construction and replace-at-end-of-life projects. However, because of the high capital cost, retrofit projects are unlikely to be cost effective when the existing chiller has significant remaining economic life, except for locations with high electricity and high demand rate tariffs.

Figure ES-1. Power Input per Capacity (kW/ton) versus Chiller Thermal Load (tons) for the Variable-speed Oil-free Centrifugal Chiller Compared to a Rotary-Screw Chiller



## CONCLUSIONS

Chillers equipped with the variable-speed oil-free centrifugal compressor with magnetic bearings can deliver high efficiency compared to chillers equipped with a rotary-screw compressor. Compared to rotary-screw chillers, the efficiency benefit becomes greater as the equipment load decreases. Because chillers spend considerable time operating at partial load, annualized energy savings can be significant. In the demonstration application, with a single chiller of appropriate capacity to meet design load requirements, the operating chiller spends more than 50% of the time operating with a partial-load factor less than 33% and more than 80% of the time operating with a partial-load factor less than 50%. This is particularly advantageous given the efficiency of the new chiller technology peaks in this load range. Application of the new chiller technology also opens the potential to deploy additional energy-savings measures, such as condenser water temperature reset as well as improved chiller staging controls (which are still under review by GSA's Green Proving Ground program).



The assessment in this report focused on the comparison between a chiller equipped with the variable-speed oil-free centrifugal compressor with magnetic bearings and a chiller equipped with a rotary-screw compressor. Additional studies should be performed to compare this new technology to other centrifugal chillers equipped with variable-speed drives in capacities greater than 500 tons. Larger chillers (*i.e.*, with capacities between 500 and 1000 tons) account for a larger percentage of chiller energy within the GSA portfolio and, thus, could be a significant source of additional energy savings if the technology proves to be as beneficial in the larger capacity range. For the time being, this report recommends targeted deployment of this technology as an end-of-life replacement to chillers with positive displacement compressors, such as rotary-screw and reciprocating, where they are found to be cost-effective based on careful engineering analysis and life-cycle costing.

## II. Background

### A. INTRODUCTION

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 31.9% 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%.)

Because the variable-speed oil-free centrifugal chiller with magnetic bearings is expected to offer higher efficiency compared to rotary-screw chillers, the use of this new technology is expected to reduce energy consumption, thereby assisting the U.S. General Services Administration (GSA) in achieving the energy-use intensity reduction requirements as identified in the Energy Independence and Security Act of 2007 (EISA 2007). This report will assess the energy performance from one installation of a variable-speed oil-free centrifugal chiller with magnetic bearings in a central chiller plant and draw conclusions on how the application of this technology may contribute to further reductions in space cooling energy at GSA facilities.

This technology has the potential to reduce space cooling energy because the efficiency is greater than that of many rotary-screw chiller options. Depending on the annual load profile for the application, the variable-speed oil-free centrifugal chiller with magnetic bearings may also offer efficiency benefits over more conventional centrifugal chillers because the new technology offers extremely high efficiency while operating at lower partial load.

As a chiller, the method of testing and reporting efficiency is covered by the Air Conditioning, Heating and Refrigeration Institute (AHRI) testing standard, Standard for Performance Rating Of Water-Chilling and Heat Pump Water-Heating Packages Using the Vapor Compression Cycle, ANSI/AHRI Standard 550/590 (AHRI 2011). Minimum efficiency requirements (for chillers in commercial buildings) for both full load and integrated part-load value (IPLV) conditions are identified in the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) Standard 90.1-2010, Table 6.8.1C.

For use in Federal facilities, chillers are covered as a U.S. Department of Energy (DOE) Federal Energy Management Program (FEMP) designated product.<sup>1</sup> Minimum efficiency ratings for water-cooled chillers depend on both the compressor type (centrifugal versus positive displacement), as well as rated capacity. Centrifugal compressors are inherently more efficient than positive displacement compressors. For a 150-ton positive displacement chiller, such as a chiller with a rotary-screw compressor, the rated efficiency must be 0.51 kW/ton (IPLV) or 0.61 kW/ton (full load) or better.<sup>2,3</sup> For a 150-ton centrifugal chiller, such as this

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<sup>1</sup> Federal agencies are required by the National Energy Conservation Policy Act (P.L. 95-619), Executive Order 13423, and Federal Acquisition Regulation (FAR) Subpart 23.2 and 53.223 to specify and buy ENERGY STAR® qualified products or, in categories not included in the ENERGY STAR program, FEMP designated products. FEMP-designated products include the highest 25% of equivalent products for energy efficiency. Chillers are covered as an FEMP-designated product.

<sup>2</sup> To meet FEMP-designated product requirements for chillers, the chiller must meet or exceed either the full load or IPLV efficiency rating.

<sup>3</sup> For efficiency listed in units of kW/ton, lower numbers are better.

new technology, the rated efficiency must be 0.37 kW/ton (IPLV) or 0.59 kW/ton (full load) or better. By comparison, the line of chillers using this new technology reportedly has IPLV efficiencies in the range of 0.33 kW/ton to 0.37 kW/ton. Assuming this new chiller technology can meet expectations, this technology could reduce space cooling energy by around 35% compared to using high-efficiency FEMP-designated chillers equipped with rotary-screw compressors.

## 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 31.9% of commercial office building floor space (EIA 2003a). This illustrates that chillers are predominantly used in larger (*i.e.*, greater than 200,000 ft<sup>2</sup>) commercial office buildings.

What makes the variable-speed oil-free centrifugal chiller with magnetic bearings unique is the compressor. The compressor evolved from research that began in 1993 (<http://www.turbocor.com>). The first field test of the compressor occurred in California in 2000. Initially, a 60-ton unit was manufactured which led to developing larger compressors. In 2007, the the developer of these compressors opened its first manufacturing facility in the United States. Today, the Turbocor™ compressor by Danfoss, a leading commercial provider, is available in four different capacities, ranging from 60 up to 200 tons, and in both air-cooled and water-cooled models. In addition, several original equipment manufacturers (OEM) now offer the unique compressors integrated into their line of chillers. The Navy Technology Validation (Techval) program began demonstrating the technology in 2005, first as a compressor retrofit and later using OEM chillers.<sup>4</sup> The technology is still emerging into the federal sector. While the Navy has installed more than 100 units, the Army has yet to install its first unit. GSA recently started installing the variable-speed oil-free centrifugal chiller with magnetic bearings in a small, but growing, number of locations across the country.

The technology has the potential to become an agencywide standard, at least as an alternative to rotary-screw chillers. While the compressors are currently available in capacities from 60- to 200-tons cooling, the compressors can be stacked on common tube bundles to create much larger capacity chillers. One OEM offers a single chiller that employs six compressors for a rated capacity totaling 1,200-tons cooling. Further, the modular chillers can be staged to form even larger chiller plants. Also, given that the resulting chillers can require smaller footprints than conventional chillers, large chiller plants configured with the new variable-speed oil-free centrifugal chiller with magnetic bearings may require less floor space (*i.e.*, smaller mechanical rooms) than conventional rotary-screw or centrifugal chiller systems.

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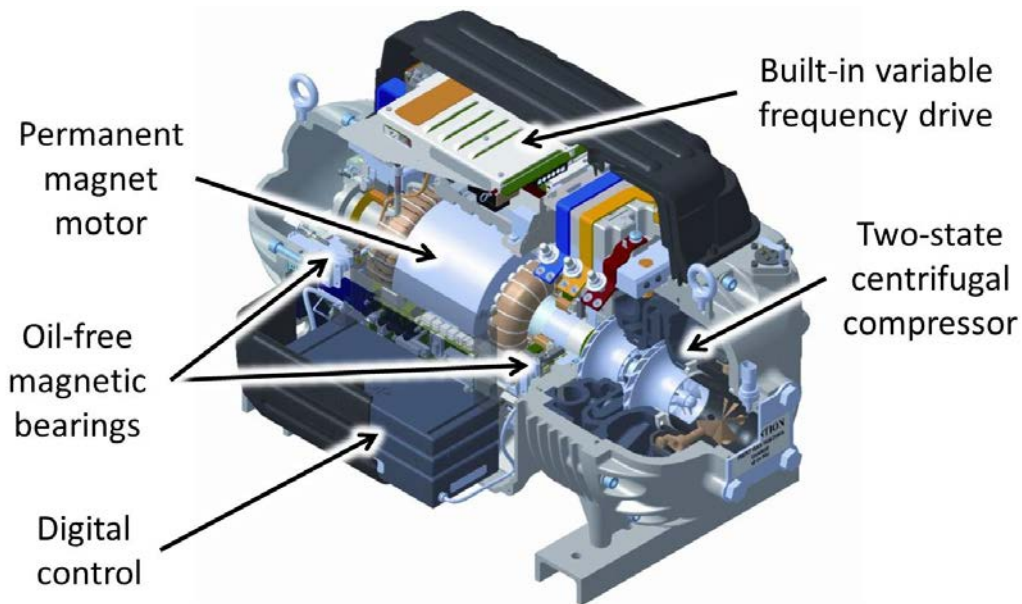
<sup>4</sup> Four demonstrations sponsored by the Navy Techval program reported savings from 40% to 65% with simple payback ranging from 4 to 8 years. The Navy Techval Program does not publish public reports, but federal energy staff may request results by contacting the Navy Techval program manager at [paul.kistler@navy.mil](mailto:paul.kistler@navy.mil).

### III. Methodology

#### A. TECHNOLOGY DESCRIPTION

This report refers to the subject technology as a variable-speed oil-free centrifugal chiller with magnetic bearings because there is not an easy name that fully describes what makes the technology unique without referring to a trademark or brand name. As noted earlier, the primary technology, and that which makes the product highly efficient and unique, is the compressor. The Turbocor™ compressor is manufactured by Danfoss Turbocor Compressors, Inc. A cut-away view of the compressor is shown in Figure 1. While the compressor is available as a chiller retrofit, several OEM providers have fully integrated the unique compressor into their chiller product line.

**Figure 1. Cut-away view of the Turbocor™ compressor**  
(image courtesy of Danfoss Turbocor Compressors, Inc., used with permission)



The manufacturer identifies several features that make the compressor unique and efficient:

- The compressor uses two impellers on a single, high-speed, rotating shaft. The rotating shaft and impeller assembly is the only moving part in the compressor. The high-speed rotating shaft with two-stage compressor design offers high efficiency in a small package.
- The compressor has a fully integrated variable-speed drive with soft start. Chiller plants are sized to meet peak load under design conditions, which means most chillers operate much of the time significantly under loaded. The variable-speed drive allows the compressor to be highly efficient, especially at partial load. Conventional chillers are designed to achieve peak efficiency at high load (typically between 80% and full load), and efficiency can drop off significantly as the load decreases. This new chiller technology may only be slightly more efficient at full load, but its efficiency will increase as the load decreases, due to the variable-speed drive. Efficiency can peak between 30%

and 50% of rated capacity, which tends to coincide with the most frequent operating load range. The soft-start feature reduces start-up stress, which is expected to reduce long-term maintenance.

- The single shaft rotates within magnetic bearings. This unique feature reduces friction, adding to the overall high efficiency, and eliminates the metal-on-metal contact of conventional bearings. This, in turn, allows the chiller to operate without the need for lubricating oil. Eliminating lubricating oil eliminates the need for several ancillary components required to support the oil system (e.g., oil pumps, oil heaters, oil separators, and oil filters) and is credited with providing much of the reduced maintenance benefit associated with the technology.
- The compressor is digitally controlled. Microprocessors manage compressor operation and provide self diagnostics and correction, so compressors can be integrated into building automation systems (BAS) and web-enabled for monitoring. The overall design allows for very quiet chiller operation compared to conventional chiller designs, which allows chiller plants to be located closer to public spaces.

While the cost of the compressor may be as much as twice that of a conventional chiller compressor, the new technology offers much higher operating efficiency at partial load (*i.e.*, where the chiller operates most frequently) and the potential for lower maintenance costs.

Many case studies are available on the application of the new chiller compressor.<sup>5</sup> Chillers with the new compressor have been studied by utility programs, as well as regional energy offices. In addition, the compressor manufacturer and OEM providers have published case studies. As noted earlier, the Navy Techval program has studied the new technology and presented results.<sup>6</sup> The range of findings is large (from energy reduction of 20% to as high as 65%). The four demonstrations sponsored by the Navy Techval program reported savings from 40% to 65% with simple payback ranging from 4 to 8 years.

## B. TECHNICAL OBJECTIVES

The purpose of this study is to assess the operation of an OEM chiller using the variable-speed oil-free compressor with magnetic bearings in a central chiller plant serving a federal building. The study will monitor the operational performance of a baseline chiller and the new variable-speed chiller over the full range of normal operation. The cooling-load profile for the building application will also be determined from the monitored data. The measured performance for the two chillers will be compared in a weather-normalized analysis to assess the resulting energy savings delivered by the new chiller. The findings and conclusions will also include a life-cycle cost analysis. To the extent possible, the study will also assess the technology's broader application within the GSA portfolio.

Refrigeration equipment does not create energy, but rather it concentrates the energy and moves it from one location to another. For this reason, the efficiency of refrigeration equipment is defined differently than for most equipment. The conventional definition of efficiency (*i.e.*, output/input) becomes meaningless.

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<sup>5</sup> A quick internet search will result a listing of several case studies, which may be useful in identifying a broad number of applications. However, the case studies tend to be very low on useful technical information regarding documented performance and cost.

<sup>6</sup> See GovEnergy 2009 presentation by Paul Kistler at <http://www.govenergy.com/2009/presentations.html#tech>.

Therefore, the efficiency of refrigeration equipment is better defined as “useful thermal energy” (or cooling energy provided) 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 “useful thermal energy” (in Btu) divided by “energy consumed” (also in Btu) or “rate of useful thermal energy” (in Btu/h) divided by “power” (also in Btu/h). COP is a unitless measure. For industrial capacity equipment, chiller efficacy may also be expressed as “power input per 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 unit in kilowatts [kW], to the net refrigerating capacity in tons refrigeration at any given set of rating conditions, expressed in kW/ton (AHRI Standard 550/590-2011). This is an inverse measure because a smaller number implies a greater efficiency. The rated efficiency level for commercial chillers used by FEMP-designated products and ASHRAE Standard 90.1-2010 is expressed using the *Power Input per Capacity* ratio (kW/ton).

Data from meters and sensors were collected through an independent data acquisition system (DAS). Data from the meters and sensors were scanned every second with data averages stored in 15-minute interval tables. In addition to monitoring the cooling output of each of the two chillers, integrated power demand for each chiller was also measured and recorded. The stored data were used to determine the performance profile for each of the two chillers, as well as to determine a thermal (cooling) load profile model for the building. Data were collected from March 2012 through early September 2012. The two chiller performance profiles, building thermal load model and typical meteorological year (TMY) data, were used to estimate normalized energy savings associated with the variable-speed oil-free centrifugal chiller. The results were used to develop an assessment of the installation economics.

### C. DEMONSTRATION PROJECT LOCATION

GSA identified the George Howard, Jr. Federal Building and U.S. Courthouse located in Pine Bluff, Arkansas for monitoring and analysis of the variable-speed oil-free centrifugal compressor with magnetic bearings. The George Howard, Jr. Federal Building and U.S. Courthouse, is located in the heart of downtown Pine Bluff, Arkansas. Built in 1966, the building has four floors, 108,000 ft<sup>2</sup> and currently houses the U.S. Postal Service, U.S. Courts, Social Security Administration, Department of Agriculture, Department of Labor, and the Internal Revenue Service. A photo of the building is shown in Figure 2.

**Figure 2. View of the George Howard, Jr. Federal Building and U.S. Courthouse, located in Pine Bluff, Arkansas**



The central chiller plant, located in a basement mechanical room, originally consisted of two 150-ton chillers, each with a rotary-screw compressor. Based on the serial number of the remaining chiller, the original rotary-screw chillers were manufactured in 1993. A single cooling tower, common to both chillers, is located on the roof of the building. A single chiller typically operates to meet the facility cooling load. According to the site staff, the control sequence is programmed to operate both chillers when the outside air temperature (OAT) exceeds 90°F. As part of an energy-savings performance contract (ESPC), one of the chillers had recently been replaced with a new chiller using the variable-speed oil-free centrifugal compressor with magnetic bearings. One of the previous rotary-screw chillers remained in place and was used to establish a baseline performance profile.

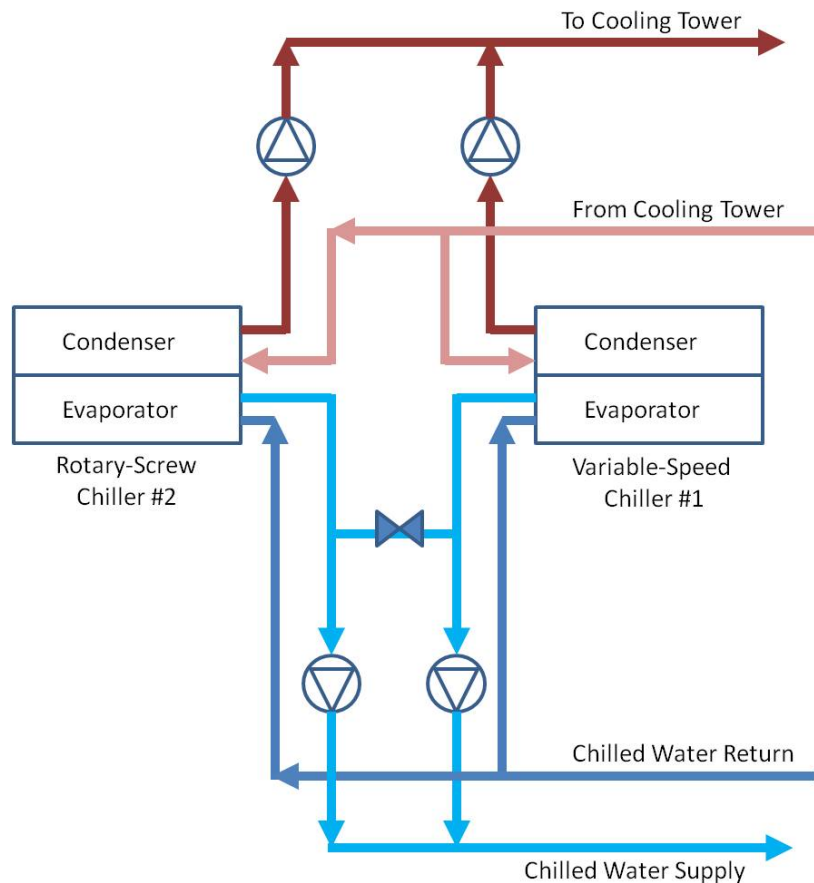
The new chiller is equipped with dual variable-speed oil-free centrifugal compressors with magnetic bearings and has a nominal capacity of 150-tons cooling. The two chillers operate in parallel, and even with the record temperatures during the 2012 summer, the facility only ran both chillers during 68 hours over the 190-day study period.

## IV. Measurement and Validation Plan

### A. FACILITY DESCRIPTION

The George Howard, Jr. Federal Building and U.S. Courthouse is a four-story federal office building, including a courthouse and post office, located in the heart of Pine Bluff, Arkansas. The chiller room is located on the basement level with two chillers piped in parallel. The chillers are served by a common cooling tower located on the roof. Figure 3 illustrates the cooling system for the building.

**Figure 3. One-Line Diagram of the George Howard, Jr. Federal Building Cooling System**



The new variable-speed oil-free centrifugal chiller with magnetic bearings was installed as a replacement for one of the original chillers before this location was identified as a Green Proving Ground program demonstration project. However, because one of the original chillers remained, a side-by-side comparison of performance is possible.

### B. TECHNOLOGY SPECIFICATION

Space cooling requirements are served from two chillers piped in parallel with a primary-only chilled water distribution system. Constant-volume pumps circulate chilled water through the building, serving the various air-handling units. A common cooling tower provides water to the chiller's condensing coils. Each



chiller is served by its own chilled water and condenser water pumps. The original chiller is a rotary-screw chiller rated at 150 tons. The new chiller is a variable-speed oil-free centrifugal chiller with magnetic bearings. The new chiller has dual compressors rated at a nominal 150 tons. Nameplate specifications for the two chillers are listed in Table 1. The primary chiller sequence operates the new chiller as the lead chiller with the older chiller in the lag position. The older chiller is mostly used as a back-up chiller since the new chiller was installed.

**Table 1. Chiller Nameplate Information**

Description	Original Chiller	New Chiller
Nominal Capacity	150-tons cooling	150-tons cooling
Minimum Circuit Rating	460-volt, 185-amp, 3-phase	460-volt, 166-amp, 3-phase
Maximum Circuit Rating	480-volt, 300-amp, 3-phase	480-volt, 225-amp, 3-phase
Compressor Rating	1 unit 460 volt 148 run-load amps (RLA)	2 units 460 volt 72 RLA (each)
Oil Tank Heater	2 units 115 volts 2 RLA	Not applicable
Refrigerant	R-22 330 pounds	R-134A 531 pounds
Oil	35 pints	Not applicable

**C. TEST PLAN**

While the new chiller had already been installed at the start of this activity, one of the original chillers remained installed to be used as a baseline. The monitoring system was installed to collect data on both chillers independently. While the new variable-speed oil-free centrifugal chiller operated most of the time, the operators did switch the chiller sequence to allow some run-time on the older chiller to serve as a baseline for comparison. Chiller power, chilled water flow rate, and supply and return chilled water temperatures were monitored from March 1, 2012, through July 12, 2012. The new chiller was monitored for more than 2,890 hours of operation, while the older chiller was monitored for more than 1,200 hours of operation.<sup>7</sup> Both chillers were monitored from minimum to full load with sufficient data to characterize

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<sup>7</sup> Monitoring of the older rotary-screw chiller was limited to minimize potential negative impact on the ESPC.

operating performance properly. The data were used to determine the performance of each chiller under various load conditions. The data for determining thermal performance were monitored and stored using a central data acquisition system (DAS), which provided data on input power, supply and return temperatures, and other information necessary to monitor cooling performance and power consumption. Thermal performance was calculated using the equations presented below. Because testing standards are based on steady-state controlled conditions, the method of testing used in this demonstration deviated from AHRI Standard 550/590 as required to assimilate data under dynamic field conditions. The purpose of this monitoring activity was to capture performance under actual operating conditions.

Data were collected using a central DAS located in the main chiller room. The interval data were transferred to PNNL through the use of a Campbell Scientific SC115, flash memory drive. The flash drive was periodically shipped between the GSA site point-of-contact and PNNL. The data transfer device allows data to be transferred with no loss of data during the transfer.

The thermal performance of the chillers is determined by the ratio of the rate of thermal cooling provided to power input. The power input,  $P_{in}$ , was measured directly at each chiller in kilowatts (kW). Power input may be converted into Btu/h (the rate of energy input) by multiplying by 3412 Btu/kWh, as shown in equation 1.

$$Q_{in} = (P_{in})(3412 \text{ Btu/kWh}) \quad (\text{equation 1})$$

where  $Q_{in}$  is the input power, in Btu/h, and  $P_{in}$  is the recorded input power for the chiller. The rate of thermal cooling provided,  $Q_{cool}$ , in Btu/h, is defined as

$$Q_{cool} = (\dot{q})(\rho)(C_p)(T_{return} - T_{supply})(8.345 \text{ lb}_m/\text{gal})(60 \text{ min/h}) \quad (\text{equation 2})$$

where  $\dot{q}$  is the chilled water flow rate, in gal/min;  $\rho$  is the density of the chilled water, in  $\text{lb}_m/\text{ft}^3$ ;  $C_p$  is the specific heat of the chilled water, in  $\text{Btu}/\text{lb}_m\text{-}^\circ\text{F}$ ; and  $T_{supply}$  and  $T_{return}$  are the supply and return chilled water temperatures, in  $^\circ\text{F}$ , respectively. The additional constants shown in equation 2 are used for unit conversion to achieve Btu/h. Tons cooling,  $Q_{tons}$ , can be achieved by taking the result of equation 2 and dividing by 12,000 Btu/ton-h, as illustrated in equation 3.

$$Q_{tons} = (Q_{cool})/(12,000 \text{ Btu/ton-h}) \quad (\text{equation 3})$$

As previously noted, there are various measures of chiller efficiency used by the industry. COP can be determined by dividing the result from equation 2 by the result from equation 1 as shown in equation 4.

$$\text{COP} = (Q_{cool})/(Q_{in}) \quad (\text{equation 4})$$

“Power Input per Capacity,” as defined by AHRI, can be determined using equation 5.

$$\text{Power Input per Capacity} = (P_{in})/(Q_{tons}) \quad (\text{equation 5})$$

#### D. INSTRUMENTATION PLAN

A Campbell Scientific CR1000 data acquisition system was located in the basement chiller room. Data collection included sensors that are directly used to determine chiller performance. A few additional sensors

were installed to validate chiller operation. The primary sensors included WattNode™ power meters on each chiller. An additional power meter was installed on the secondary compressor of the variable-speed oil-free centrifugal chiller to assess the potential impact of compressor sequencing. Surface-mounted ultrasonic flow meters were installed on the chilled water supply pipe of each of the two chillers. In addition, thermocouples were installed on the supply and return lines of each chiller.<sup>8</sup> Power meters were also installed on the two chilled water supply pump motors and the condenser water pump motors. These power meters were to assist in validating which chiller, or both, was in operation.

The DAS was mounted on the wall in the chiller room and supplied power from an existing 120-volt wall outlet. Power supply to the ultrasonic flow meters was also from the wall outlet, but it was transformed to 24-volt (dc) and hard wired along with the communication signal. The ultrasonic flow meters were installed on the chilled water supply lines from each chiller, which offered the longest straight run of pipe. A small section of pipe insulation was removed to accommodate the surface-mounted flow meter, and an insulated thermal blanket was installed around the flow meter to prevent condensation. The surface-mounted thermocouples were installed on the chilled water supply and return pipe lines for the individual chillers. A small section of insulation was removed to allow the thermocouples to be installed, and thermal compound was used to ensure a reasonable temperature reading from them. The original insulation was then replaced over the thermocouples, and an insulated thermal blanket was installed around the insulated pipe.

Sensor data were measured every second. Average sensor data were recorded in 15-minute intervals. Table 2 identifies the systems monitored by the DAS.

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<sup>8</sup> Insertion temperature sensors provide more accurate absolute temperature readings than surface-mounted temperature sensors but because temperature difference is the desired product, surface-mounted temperature sensors provide sufficient accuracy and are better suited to temporary monitoring activities.

**Table 2. Sensors used with the Data Acquisition System**

Parameter	Sensor	Units	Signal
Power, chiller 1 (new chiller)	WattNode pulse power meter, 193-Hz, 460-volt, 3-phase, 250-amp current transformer	kW	Pulse
Power, chiller 2 (original chiller)	WattNode pulse power meter, 100-Hz, 460-volt, 3-phase, 250-amp current transformer	kW	Pulse
Flow rate, chiller 1	Spirax/Sarco transit-time ultrasonic flow meter	gpm	Pulse
Flow rate, chiller 2	Spirax/Sarco transit-time ultrasonic flow meter	gpm	Pulse
Chilled water supply temperature, chiller 1	Type T, fast-response, surface-mounted thermocouples (redundant sensors installed)	°C	Voltage differential
Chilled water return temperature, chiller 1	Type T, fast-response, surface-mounted thermocouples (redundant sensors installed)	°C	Voltage differential
Chilled water supply temperature, chiller 2	Type T, fast-response, surface-mounted thermocouple	°C	Voltage differential
Chilled water return temperature, chiller 2	Type T, fast-response, surface-mounted thermocouple	°C	Voltage differential
Condenser water supply temperature (from common tower)	Type T, fast-response, surface-mounted thermocouple	°C	Voltage differential
Outside air dry-bulb temperature and relative humidity sensor	Campbell Scientific, HC2S3, with solar radiation shield	°C RH	Voltage differential
Power, pump 1, chiller 1 chilled water	WattNode pulse power meter, 193-Hz, 460-volt, 3-phase, 50-amp current transformer	kW	Pulse
Power, pump 3, chiller 2 chilled water	WattNode pulse power meter, 4-Hz, 460-volt, 3-phase, 50-amp current transformer	kW	Pulse
Power, chiller 1, compressor 2	WattNode pulse power meter, 100-Hz, 460-volt, 3-phase, 70-amp current transformer	kW	Pulse

## V. Results

The results section is divided into four subsections:

- The performance profile of the two chillers resulting from the monitored data.
- The building thermal load profile resulting from the monitored data.
- This section brings the profiles together to determine the resulting benefit of the new variable-speed oil-free centrifugal chiller compared to the original rotary-screw chiller for a normalized TMY. The results from the third section are also used to determine the economic assessment of the new variable-speed oil-free centrifugal chiller with magnetic bearings.
- This section extrapolates from the observed data and findings to identify additional opportunities for further improving the performance of the chiller plant both for the monitored location, as well as potential other applications in the GSA portfolio.

### A. CHILLER PLANT PERFORMANCE PROFILE

The data acquisition system collected data from March 1, 2012, through September 12, 2012. During this time, the baseline rotary-screw chiller accumulated 1,206 operating hours and the variable-speed oil-free centrifugal chiller accumulated 2,891 operating hours. Both chillers operated sufficient hours from minimum to full load to establish a meaningful performance profile. Applying the collected data to equation 3 (thermal cooling provided in tons) and corresponding measured power ( $P_{in}$ ) result in the thermal performance for each chiller, which is illustrated in Figure 4 for both chillers. The two performance profiles shown in Figure 4 illustrate a clear reduction in power required by the variable-speed oil-free centrifugal chiller with magnetic bearings across the entire load profile of the chillers. Further, the reduction in power becomes greater at lower cooling loads. To further document the comparison, Figure 5 illustrates the *Power Input per Capacity* (kW/ton) resulting from equation 5 versus chiller thermal load (tons) for both chillers. In addition, Figure 6 illustrates the COP resulting from equation 4 versus chiller thermal load (tons) for both chillers. Each figure illustrates the efficiency advantage of the new, variable-speed, chiller over the previous chiller from minimum to full load. However, Figure 6 is the best at illustrating how the efficiency of the older rotary-screw chiller continuously decreases as the chiller load is reduced (*i.e.*, the peak efficiency occurs at full load); whereas the efficiency of the new, variable-speed, chiller increases as the load is reduced, peaking around 40 to 50 tons (27 to 33% of nominal full load). Additional findings related to how the variable-speed oil-free centrifugal chiller can operate more efficiently will be discussed later in this report.

Figure 4. Chiller Performance Profile Resulting from the Measured Data

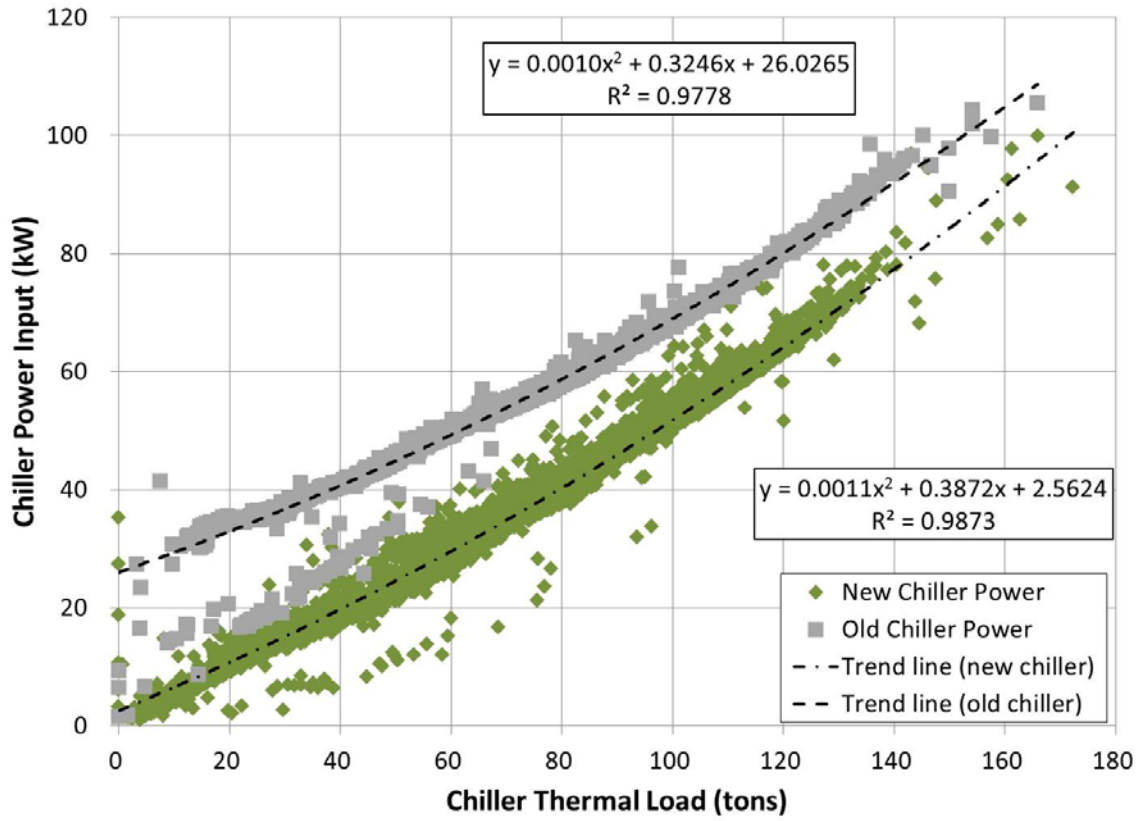


Figure 5. Comparing Power Input per Capacity (kW/ton) versus Chiller Thermal Load (tons)

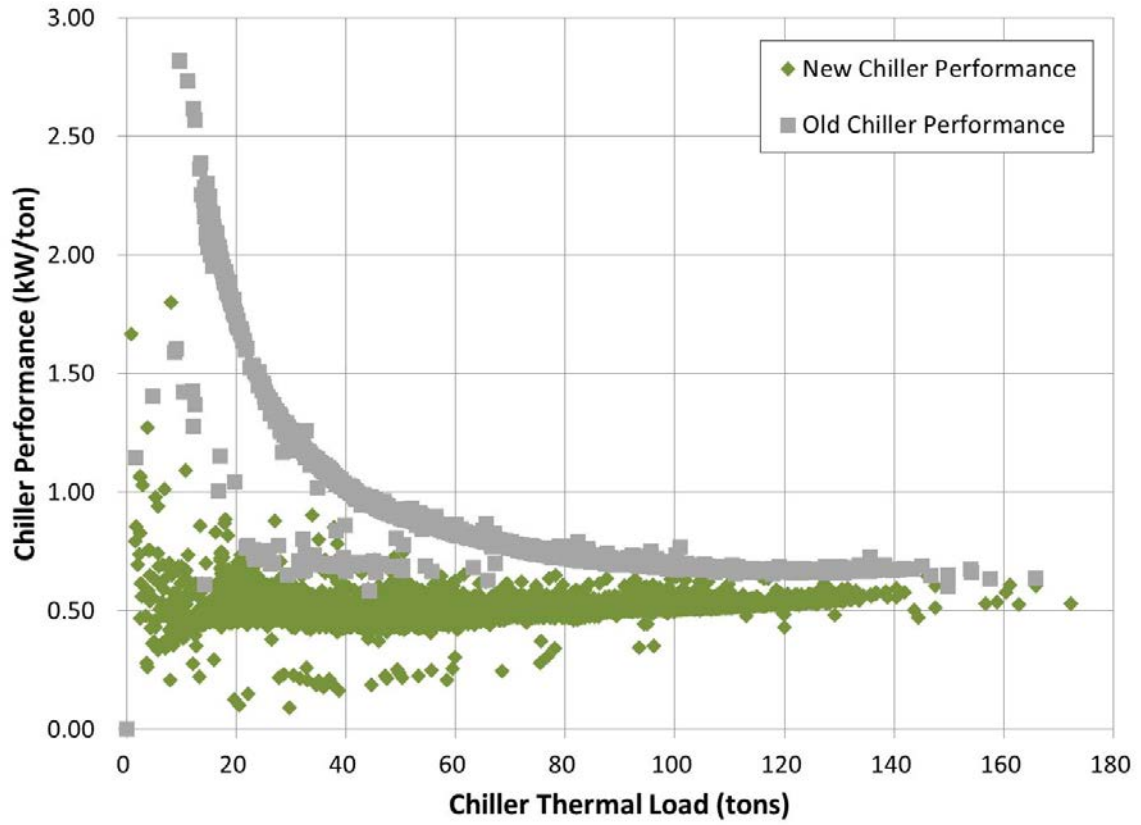
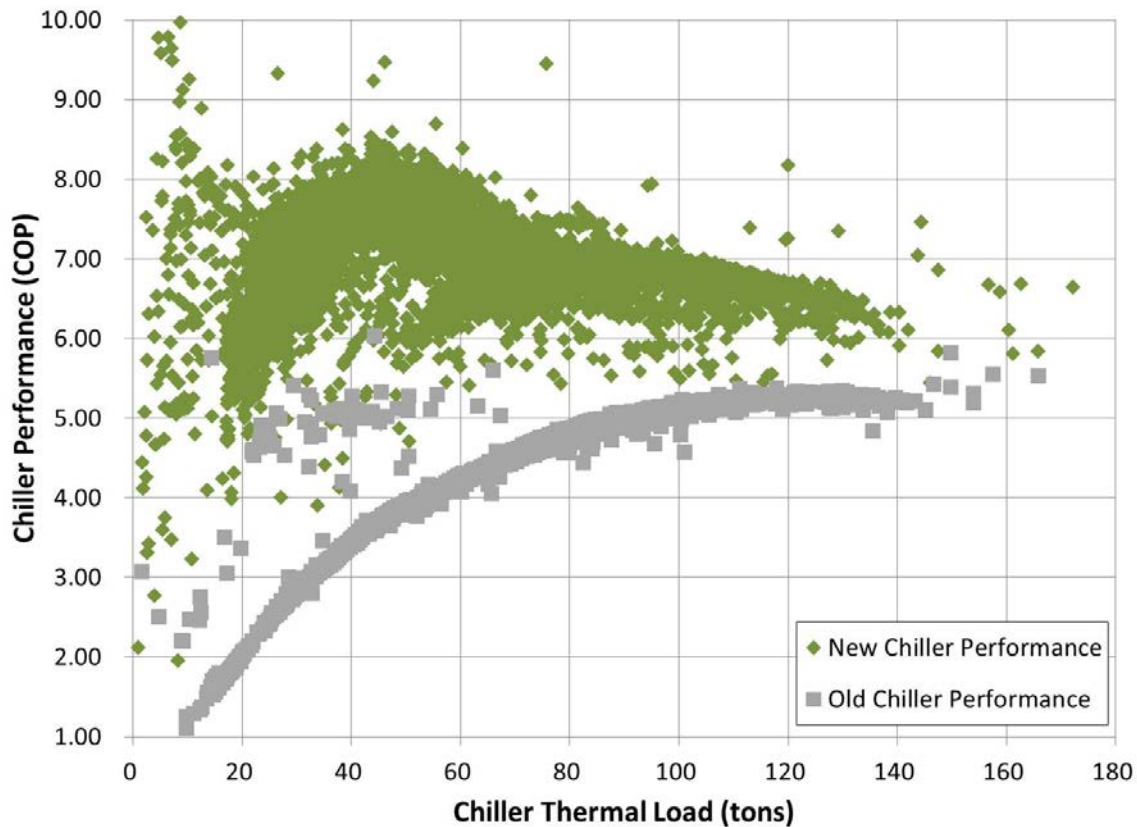


Figure 6. Comparing Coefficient of Performance (COP) versus Chiller Thermal Load (tons)



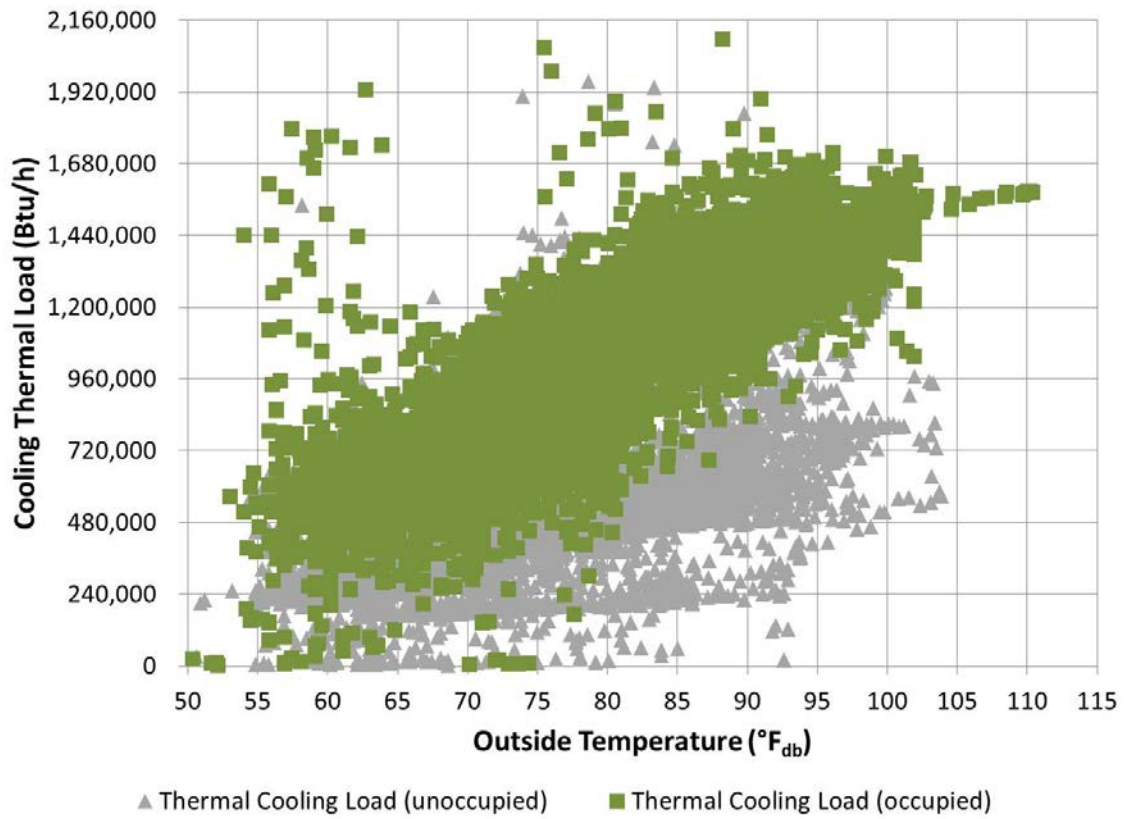
## B. BUILDING THERMAL LOAD PROFILE

The data from equation 2 (thermal cooling load) plotted versus outside air temperature can also be used to determine the thermal cooling load profile for the application—space cooling for the George Howard, Jr. Federal Building and U.S. Courthouse. Figure 7 illustrates the building thermal cooling load for the measured parameters. Close examination of the thermal cooling load shows that there are two operating profiles for the building, which is illustrated in Figure 7 using different markers. Transition between the two periods occurs around 4:45 AM and 5:30 PM, Monday through Friday. The various “non-occupied” periods (*e.g.*, workday unoccupied period, weekends (both daytime and nighttime), and holidays (both daytime and nighttime)) are combined into one primary “unoccupied” operating profile.

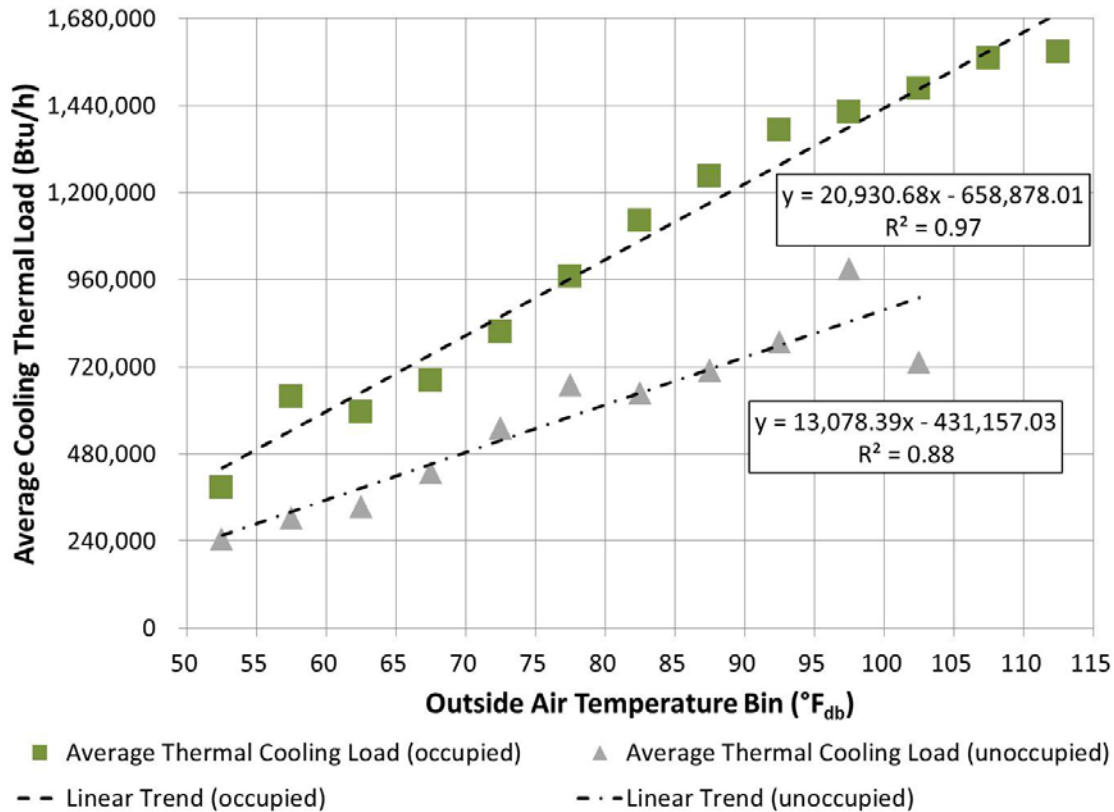
The monitoring period was not a typical cooling season. The cooling season started early and exceeded the range of typical temperatures. This allowed development of the two chiller profiles and building of the thermal cooling load profile before mid-season. The monitored data were used to develop a typical building thermal load profile, which can then be applied to TMY data to achieve a normalized cooling season. The data from Figure 7 for each of the operational periods were divided into conventional 5°F temperature bins and averaged for the corresponding bin and occupancy status. The result is illustrated in Figure 8. The resulting bin data from Figure 8 are used to develop trend line equations, which result in a building thermal cooling load model based on occupancy status and outside air temperature.



Figure 7. Building Thermal Cooling Load versus Outside Air Temperature



**Figure 8. Average Building Thermal Cooling Load versus Outside Air Temperature for Bin Data Analysis**



**C. NORMALIZED ANNUAL ENERGY AND ECONOMIC ASSESSMENT**

A weather-normalized estimate of annual cooling energy consumption for the chiller boiler plant can be determined using the chiller performance profiles in conjunction with the building thermal cooling load profiles and applying TMY data. TMY3<sup>9</sup> data for Pine Bluff, Arkansas, are used in this analysis (Wilcox and Marion 2008). Further, based on the observed data, it is assumed the chiller plant operates anytime the outside air temperature rises above 50°F. The baseline profile is based on the observed performance for the original rotary-screw chiller. The post-installation profile is based on the observed performance of the variable-speed oil-free centrifugal chiller with magnetic bearings. The facility prefers to operate the new variable-speed chiller to maximize savings. The comparison, shown in Table 3, assumes the new chiller operates continuously and the old rotary-screw chiller is only used for back-up. Normalized for TMY data, the variable-speed oil-free centrifugal chiller with magnetic bearings is estimated to consume 170,181 kWh/yr in electricity, while the previous rotary-screw chillers are estimated to consume 294,794 kWh/yr in electricity. Based on this analysis, the variable-speed oil-free centrifugal chiller with magnetic bearings is estimated to reduce chiller electricity consumption by 124,613 kWh/yr, or 42.3%.

<sup>9</sup> TMY3 refers to a third update of TMY data for 1020 locations based on data from 1991 to 2005.

Because the condenser water pumps are constant volume, condenser pump energy is the same for both the new and old chiller. The reduction in chiller compressor energy can reduce the amount of thermal energy that needs to be rejected into the atmosphere through the cooling tower. Depending on the cooling tower fan control system and its set points, it may be reasonable to assume that there may be some reduction in cooling tower fan motor energy resulting from the application of the variable-speed oil-free centrifugal chiller. However, the cooling tower energy system was excluded from the monitoring and analysis in this report.

The local electricity costs for the location are \$0.073/kWh.<sup>10</sup> This results in an annualized energy cost reduction of \$9,097. Installation of the variable-speed oil-free centrifugal chiller with magnetic bearings cost \$351,056, as reported by GSA, which results in a 39-year simple payback. It is noted that only one of the two existing chillers was replaced with the new variable-speed oil-free centrifugal chiller with magnetic bearings. The remaining rotary-screw chiller remains to serve as back up during down time, as well as supporting peak load conditions. The installation cost of the new chiller at the George Howard Jr., Federal Building appears to be higher than expectations. E Source provides a general cost of a new chiller installation at between \$1,500 and \$1,800/ton for a conventional rotary-screw or centrifugal chiller (Criscione and Zuboy 2012). In addition, E Source estimates the cost of a chiller equipped with the new variable-speed oil-free centrifugal compressor with magnetic bearings comes with a cost premium of 30 to 35% over an equivalent capacity rotary-screw or centrifugal chiller (Criscione 2006). A quick survey of GSA facilities was performed to collect installed cost data for other installations of new chillers equipped with variable-speed oil-free centrifugal compressors with magnetic bearings. As shown in Table 4, the cost (on a per ton basis) was higher at the George Howard Jr., Federal Building. The George Howard, Jr. Federal Building installation was part of a turnkey ESPC project. ESPC projects include a performance guarantee, annual measurement and validation reporting requirements and an assumption of financial and technical risks by the energy-service company (ESCO). Therefore, we assume that the higher than normal installed cost is partially attributed to the fact this project was funded through an ESPC.

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<sup>10</sup> The rate tariff for the local energy service provider, Entergy, includes both on-peak and off-peak energy charges, on-peak demand charges, excess demand charges, and several additional fees. A more detailed analysis of the rate tariff versus the site's whole-building consumption and demand patterns resulted in a similar on-peak and off-peak energy rate. Therefore, an average electricity cost is used in this analysis.

**Table 3. Weather-Normalized Energy Assessment of Variable-Speed Oil-Free Centrifugal Chiller with Magnetic Bearings for the George Howard, Jr. Federal Building and U.S. Courthouse**

Temperature Bin	Average Bin Temperature	TMY3 Total Observations	TMY3 Total Observations	Average Facility Thermal Cooling Load <sup>(1)</sup>	Average Facility Thermal Cooling Load <sup>(2)</sup>	Baseline Chiller Power <sup>(3)</sup>	Baseline Chiller Power <sup>(4)</sup>	Post-Installation Chiller Power <sup>(5)</sup>	Post-Installation Chiller Power <sup>(6)</sup>	Baseline Electricity Consumption <sup>(7)</sup>	Post-Installation Electricity Consumption <sup>(8)</sup>
A	B	C	D	E	F	G	H	I	J	K	L
		Occupied	Unoccupied	Occupied	Unoccupied	Occupied	Unoccupied	Occupied	Unoccupied		
(°F)	(°F)	(h/yr)	(h/yr)	(tons)	(tons)	(kW)	(kW)	(kW)	(kW)	(kWh/yr)	(kWh/yr)
105 to 109°F	107.5	5	0	132.60	81.23	87.5	59.3	72.4	41.0	438	362
100 to 104°F	102.5	50	12	123.88	75.78	82.3	56.7	66.7	37.9	4,797	3,788
95 to 99°F	97.5	95	45	115.16	70.33	77.3	54.0	61.1	35.0	9,777	7,379
90 to 94°F	92.5	129	93	106.43	64.88	72.5	51.5	55.7	32.1	14,137	10,170
85 to 89°F	87.5	273	235	97.71	59.43	67.8	49.0	50.4	29.3	30,020	20,652
80 to 84°F	82.5	322	464	88.99	53.98	63.2	46.6	45.3	26.5	41,984	26,911
75 to 79°F	77.5	313	671	80.27	48.53	58.8	44.3	40.4	23.8	48,112	28,642
70 to 74°F	72.5	162	637	71.55	43.09	54.6	42.0	35.6	21.2	40,985	22,807
65 to 69°F	67.5	272	462	62.83	37.64	50.6	39.7	31.0	18.6	32,108	17,047
60 to 64°F	62.5	291	470	54.11	32.19	46.7	37.6	26.6	16.1	31,233	15,312
55 to 59°F	57.5	198	320	45.39	26.74	42.9	35.5	22.3	13.7	19,844	8,789
50 to 54°F	52.5	227	372	36.67	21.29	39.3	33.4	18.2	11.3	21,359	8,322
Total		2,436	3,781							294,794	170,181

- 1) Determined using the trend line equation from Figure 8, (column E) = [(20,930.68) \* (column B) - 658,878.01]/(12,000 Btu/ton-h)
- 2) Determined using the trend line equation from Figure 8, (column F) = [(13,078.39) \* (column B) - 431,157.03]/(12,000 Btu/ton-h)
- 3) Determined using the trend line equation from Figure 4, (column G) = [(0.00104915) \* (column E)<sup>2</sup> + (0.32461065) \* (column E) + (26.02653868)]
- 4) Determined using the trend line equation from Figure 4, (column H) = [(0.00104915) \* (column F)<sup>2</sup> + (0.32461065) \* (column F) + (26.02653868)]
- 5) Determined using the trend line equation from Figure 4, (column I) = [(0.00105153) \* (column E)<sup>2</sup> + (0.38719325) \* (column E) + (2.56242283)]
- 6) Determined using the trend line equation from Figure 4, (column J) = [(0.00105153) \* (column F)<sup>2</sup> + (0.38719325) \* (column F) + (2.56242283)]
- 7) Determined using (column K) = [(column G) \* (column C)] + [(column H) \* (column D)]
- 8) Determined using (column L) = [(column I) \* (column C)] + [(column J) \* (column D)]

**Table 4. Installation Cost for New Variable-Speed Oil-Free Centrifugal Chillers with Magnetic Bearings at GSA Installations**

Location	City, ST	No. of Chillers	Unit Chiller Capacity	Total Installed Cost	Average Cost (\$/Ton)	ESPC (Y/N)	Additional Notes
George Howard, Jr. Federal Building	Pine Bluff, AR	1	150	\$351,056	\$2,340	Y	
Goodfellow Federal Center	St. Louis, MO	2	200	\$425,000	\$1,063	N	
Hammond Federal Courthouse	Hammond, IN	2	290	\$727,090	\$1,254	N	
Neal Smith Federal Building	Des Moines, IA	4	225	\$1,609,864	\$1,789	N	Reconfigured plant from 2 to 4 chillers, new pumps, motors, tower fans, and variable-frequency drives
Robert J. Dole U.S. Courthouse	Kansas City, KS	2	250	\$1,283,972	\$2,568	N	New pumps, controls, system rebalance

An assessment of the economics for the variable-speed oil-free centrifugal chiller with magnetic bearings at the George Howard, Jr. Federal Building and U.S. Courthouse was performed. Table 5 includes a summary of the economic assessment, including the resulting net present value (NPV), savings-to-investment ratio, and simple payback.<sup>11</sup> Two strategies were included in the assessment: retrofit and replace-at-end-of-life. Retrofit is defined as when the existing (rotary-screw) chiller has considerable useful life remaining, so replacement is optional. The economics of this option are driven by the total cost of replacing the existing chiller with a new variable-speed oil-free centrifugal chiller with magnetic bearings. The resulting energy cost reduction is used to justify the total installation cost of the retrofit. Replace-at-end-of-life is defined as when the existing (rotary-screw) chiller is at the end of its useful economic life and needs to be replaced. This option is sometimes called replace-on-failure. The economics of this option are driven by the additional cost of the new type of chiller over the cost of replacing with a similar rotary-screw chiller because the chiller needs to be replaced in any event. Therefore, the resulting energy cost reduction is used to justify the added differential cost of the new, more efficient chiller.

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<sup>11</sup> Net present values and savings-to-investment ratios are calculated according to the National Institute of Science and Technology (NIST) Handbook 135. Chiller life is assumed to be 23 years consistent with the DOE FEMP Covered Product Category for Water-cooled Electric Chillers (FEMP 2012).

As noted earlier, the variable-speed oil-free centrifugal chiller with magnetic bearings may result in lower maintenance costs, primarily as a result of the elimination of the compressor’s oil system. While we do not doubt the maintenance is simpler, there is no documentation that quantifies this potential savings in either labor or materials. For this reason, the economic analysis does not include a change in real maintenance costs.

**Table 5. Economic Assessment Results for the George Howard, Jr. Federal Building**

Description	Rotary-screw chiller <sup>(1)</sup> (original)	Rotary-screw chiller <sup>(2)</sup> (new)	Variable-speed oil-free centrifugal chiller <sup>(3)</sup> (new)
Energy consumption	294,794 kWh/yr	234,574 kWh/yr	170,181 kWh/yr
Energy cost	\$21,520/yr	\$17,124/yr	\$12,423/yr
<b>Retrofit option</b>			
Installed cost	Baseline	\$291,000	\$351,056
Energy cost reduction	Baseline	\$4,396/yr	\$9,097/yr
Simple payback	Baseline	66.2 yr	38.6 yr
Net-present value	Baseline	\$-218,302	\$-200,622
Savings-to-investment ratio	Baseline	0.25	0.43
<b>Replace-at-end-of-life option</b>			
Additional installed cost	Not applicable	Baseline	\$60,056
Energy cost reduction	Not applicable	Baseline	\$4,701/yr
Simple payback	Not applicable	Baseline	12.8 yr
Net-present value	Not applicable	Baseline	\$17,680
Savings-to-investment ratio	Not applicable	Baseline	1.29

Table footnotes:

- (1) Annual energy estimate based on measured performance
- (2) Annual energy estimate based on FEMP-designated product minimum performance requirement
- (3) Annual energy estimate based on measured performance

#### D. WAYS TO FURTHER IMPROVE CHILLER PERFORMANCE

There are ways to improve the performance of chillers. These methods are discussed relative to observations made of the George Howard, Jr. Federal Building and U.S. Courthouse chiller plant, but should be considered relative to any chiller application. The efficiency of a chiller plant is a relative function of three primary variables: entering condenser water temperature<sup>12</sup>, chilled water supply temperature, and the partial load factor of the equipment. The chiller plant at the George Howard, Jr. Federal Building and U.S. Courthouse operates with a fairly constant chilled water supply temperature, a fairly constant cooling tower return temperature, and only requires a single chiller to meet peak load.

In general, lowering the entering condenser water temperature 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 this strategy can be employed. For a rotary-screw compressor, Thumann (1991, p 153) estimates the increase in COP to be around 2.2% per 1°F reduction in the entering condensing water temperature. Manufacturer specifications should be reviewed before employing this strategy. Lowering the condenser water temperature too much may have adverse effects on the compressor operation. The chiller manufacturer can assist in determining a lower limit for the condenser water supply temperature. Similar rules-of-thumb for the variable-speed oil-free centrifugal chiller with magnetic bearings are not readily available; however, for conventional centrifugal chillers, Thumann estimates the increase in COP to be around 0.4% per 1°F reduction in the entering condensing water temperature. The reduced potential improvement level results from the already improved COP for the centrifugal design. On the bright side, the variable-speed oil-free centrifugal chiller with magnetic bearings is capable of operating with extremely low condenser water temperatures.

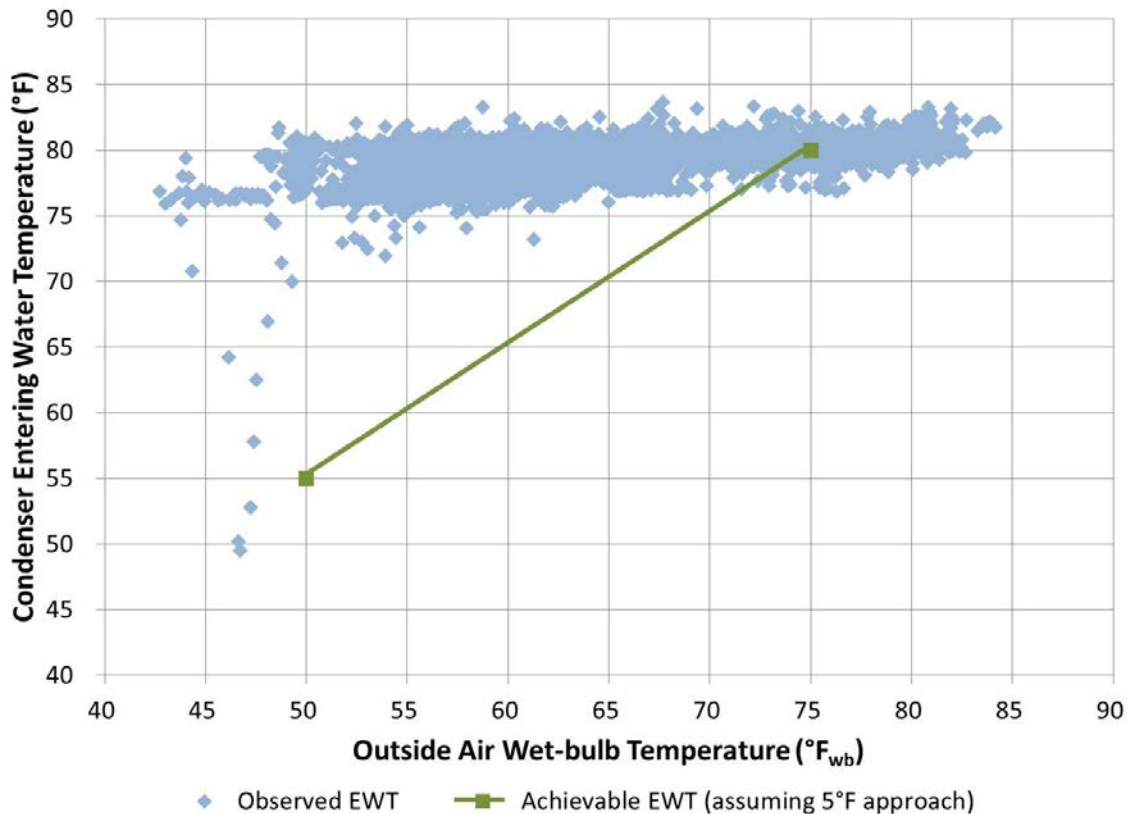
Figure 9 illustrates the entering condenser water temperature versus the coincident outside air wet-bulb temperature during the monitoring period.<sup>13</sup> Based on the cooling tower's approach temperature at peak conditions, the solid line in Figure 9 illustrates the condenser water temperature the cooling tower is capable of delivering to the chiller plant.

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<sup>12</sup> The entering condenser water temperature is the temperature of the water returning from the cooling tower.

<sup>13</sup> While wet-bulb temperature was not measured directly, it can be determined by knowing the dry-bulb temperature and relative humidity. For a definition of dry-bulb and wet-bulb temperature, see Appendix A.

**Figure 9. Entering Condenser Water Temperature versus Outside Air Web-bulb Temperature during the Monitoring Period**



Another method to improve chiller plant performance is by increasing the chilled water supply temperature. 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 a rotary-screw compressor, Thumann (1991, p 155) estimates the increase in COP to be around 2.3% per 1°F increase in the chilled water supply temperature. Similar rules-of-thumb for the variable-speed oil-free centrifugal chiller with magnetic bearings are not readily available; however, for conventional centrifugal chillers, Thumann estimates the increase in COP to be around 2% per 1°F increase in the chilled water supply temperature. The reduced potential improvement level results from the already improved COP for the centrifugal design. The primary concern with this strategy is the potential reduction in the ability to dehumidify. Under design conditions, chilled water supply temperatures are kept low to allow the cooling coil to also 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 is less reason for the low chilled water temperature.

Pine Bluff, Arkansas is located in climate zone #3, as defined by the DOE and the International Energy Conservation Code (IECC 2012). Zone #3 is classified as a mixed-humid climate zone. It is acknowledged that



dehumidification is a respectable portion of the design peak cooling load. However, dehumidification is not always required. Figure 10 illustrates the coincident outside air wet-bulb and dry-bulb temperatures during the monitoring period. The solid line, shown in Figure 10, illustrates a dew point of 55°F (equal to a humidity ratio of 0.0092 pounds of water per pound of dry air).<sup>14</sup> Data points occurring below the solid line (14% of the observed 3200 hours) indicate the opportunity to reduce the chilled water temperature without undermining the need for dehumidification.

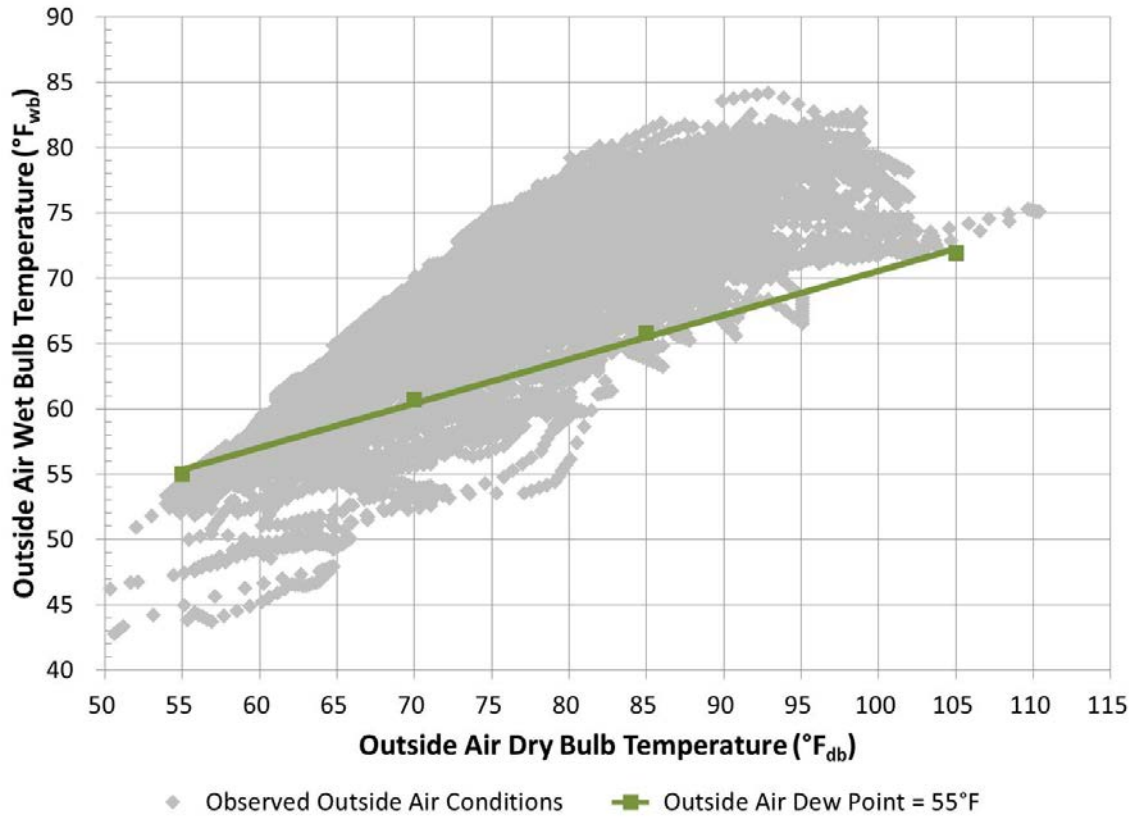
Chiller plant efficiency is also a function of the chiller equipment's partial load factor. For chiller plants with multiple chillers, the opportunity is to maximize overall system efficiency by optimizing the number of chillers operating under certain load conditions (*e.g.*, which is best—two chillers at 75% load or three chillers at 50% load?). In the case of the George Howard, Jr. Federal Building and U.S. Courthouse, a single chiller is capable of meeting the cooling load in all but the most extreme weather conditions. Therefore, there is little opportunity at this location to optimize chiller staging except when the location is forced to operate both chillers to meet the facility cooling load.

In the case of the original rotary-screw chiller, Figure 6 clearly shows the chiller's efficiency peaks near full load conditions. Even if multiple chillers of this type were available, overall chiller plant power is minimized by operating the fewest number of chillers required to meet load requirements. However, in the case of the variable-speed oil-free centrifugal chiller with magnetic bearings, Figure 6 shows the chiller's efficiency peaks around 40% of full load. If multiple chillers were available, it may reduce total chiller plant power by operating two chillers at half load rather than a single chiller at full load.

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<sup>14</sup> Maximum dew point temperature specified in the P100.

Figure 10. Coincident Outside Air Wet-bulb and Dry-bulb Temperatures Observed During the Monitoring Period



## VI. Summary Findings and Conclusions

The assessment in this report focused on the comparison between a chiller equipped with the variable-speed oil-free centrifugal compressor with magnetic bearings and a chiller equipped with a rotary-screw compressor. Additional studies should be performed to compare this new technology to other centrifugal chillers equipped with variable-speed drives in capacities greater than 500 tons. Larger chillers (capacities between 500 and 1000 tons or larger) account for a larger percentage of chiller energy within the GSA portfolio, and, therefore, could be a significant source of additional energy savings if the technology proves to be as beneficial for chillers in the larger capacity range.

This report recommends targeted deployment of this technology as an end-of-life replacement to chillers with positive displacement compressors, such as rotary-screw and reciprocating compressors, where they are found to be cost-effective based on careful engineering and life-cycle cost analysis.

### A. OVERALL TECHNOLOGY ASSESSMENT AT DEMONSTRATION FACILITY

The application of the variable-speed oil-free centrifugal chiller with magnetic bearings offers the potential for reducing energy consumption in space cooling applications through improved efficiency. The ASHRAE Standard 90.1-2010 offers multiple paths to meeting minimum efficiency requirements for space cooling equipment. Based on measured performance, the original rotary-screw chiller at the Pine Bluff facility does not meet the current ASHRAE Standard 90.1-2010 or the FEMP-designated products requirements. This is not surprising given the 19-year age of the chiller and the latest efficiency standard. New rotary-screw chillers are available that meet the current efficiency requirements. As shown in the field monitoring, the rotary-screw chiller operated with a full-load efficiency of 0.655 kW/ton and a non-standard part-load value (NPLV) efficiency of 0.704 kW/ton. By comparison, the variable-speed oil-free centrifugal chiller with magnetic bearings operated with a full-load efficiency of 0.562 kW/ton and an NPLV efficiency of 0.519 kW/ton. Both the ASHRAE Standard 90.1-2010 and the FEMP-designated products offer two pathways to meet the minimum efficiency requirements, and they are both dependent on the type and capacity of the chiller. Table 6 lists the minimum efficiency requirements for nominal 150-ton chillers excerpted from ASHRAE Standard 90.1-2010 (Table 6.8.1C) and the FEMP-designated products. For comparative purposes, Table 6 also includes the efficiency results from the two chillers monitored.

Based on the performance data collected and evaluated from the George Howard, Jr. Federal Building and U.S. Courthouse application, the variable-speed oil-free centrifugal chiller with magnetic bearings is capable of achieving manufacturer performance claims. The variable-speed oil-free centrifugal chiller with magnetic bearings offers higher operating efficiency than most positive displacement chillers (*i.e.*, reciprocating, rotary-screw, and scroll) and many alternative centrifugal chillers up to mid-range capacities.

**Table 6. Efficiency Requirements for 150-ton Water-Cooled Chillers and Field Test Results**

Equipment Type	ASHRAE Standard 90.1-2010 (Table 6.8.1C) Path A <sup>(1)</sup>	ASHRAE Standard 90.1-2010 (Table 6.8.1C) Path B <sup>(2)</sup>	FEMP Designated Product <sup>(3)</sup>	Field Test Results <sup>(4)</sup>
Rating	Full Load/IPLV (kW/ton)	Full Load/IPLV (kW/ton)	Full Load/IPLV (kW/ton)	Full Load/IPLV <sup>(5)</sup> (kW/ton)
Rotary-screw	0.775/0.615	0.790/0.586	0.61/0.51	0.655/0.704
Centrifugal	0.634/0.596	0.639/0.450	0.59/0.37	0.562/0.519

Table Footnotes:

- (1) Compliance to ASHRAE Standard 90.1-2010 can be obtained by meeting the minimum requirements of Path A or Path B. However, both the full load and IPLV must be met to fulfill the requirements for Path A or Path B. Path A allows a less efficient IPLV, but requires a more efficient full load efficiency than Path B.
- (2) Compliance to ASHRAE Standard 90.1-2010 can be obtained by meeting the minimum requirements of Path A or Path B. However, both the full-load and IPLV must be met to fulfill the requirements for Path A or Path B. Path B allows a less efficient full-load efficiency but requires a more efficient IPLV than Path A.
- (3) Compliance with FEMP-designated product can be obtained by meeting the minimum requirements of either the full-load or IPLV requirements (meeting both requirements is not required).
- (4) Test results for the original 150-ton rotary-screw chiller and 150-ton variable-speed oil-free centrifugal chiller with magnetic bearings are presented for illustrative purposes only. The field conditions for the monitored demonstration are not equivalent to the AHRI Standard 550/590 test conditions.
- (5) IPLV and NPLV are not equivalent. The IPLV is tested according to standard rating conditions (ANSI/AHRI Standard 550/590-2011), which includes controlled condenser water temperatures that are progressively lower for lower partial loads—conditions that do not exist at the demonstration location. NPLV is for non-standard rating conditions.

The installation costs of the variable-speed oil-free centrifugal chiller at the George Howard, Jr. Federal Building and U.S. Courthouse appears to be high compared to similar installations. Discussions of costs with the Navy, which has installed over 100 units, indicate the OEM chiller (equipment only) may cost as much as twice the cost of a conventional chiller of equivalent capacity. E-Source has reported the variable-speed oil-free centrifugal chiller with magnetic bearings comes with a cost premium of 30 to 35% over an equivalent capacity rotary-screw chiller (Criscione 2006). Based on observations, the cost to install the variable-speed oil-free centrifugal chiller should be equivalent to other chillers. The reported cost of the GSA chiller was \$351,056, or about \$2,340/ton. However, the cost was mostly associated with the installation rather than the cost of the chiller. According to GSA, the subcontractor’s cost to the ESCO for the chiller and installation (including demolition, design, chiller, piping modifications, insulation, controls, and programming) was \$228,663, or \$1,524/ton. The cost of the chiller alone, as reported by the chiller manufacturer, was around \$110,000, or \$733/ton. Installation of the chiller did not require extensive modification to the piping configuration. While the chiller plant is located in a basement, the location has a freight elevator for easy access. The George Howard, Jr. Federal Building installation was part of a turnkey ESPC project. ESPC projects include a performance guarantee, annual measurement and validation reporting requirements and an assumption of financial and technical risks by the ESCO. Therefore, we assume that the higher than

normal installed cost is partially attributed to the fact this project was funded through an ESPC. Regardless of the financing approach, replace-at-end-of-life and new construction are the most cost effective installation strategies; the added cost of the more efficient chiller option may be recoverable through reduced energy operating costs. Because of the high capital cost of equipment and installation costs, it will be more difficult to replace a functional chiller that still has significant remaining useful life.

## B. BEST PRACTICE

A chiller plant needs to be capable of meeting the building's peak demand, as well as operate efficiently under part-load conditions. The variable-speed oil-free centrifugal chiller with magnetic bearings accomplishes this task very well. The chiller is designed to be most effective in a modular configuration.

While the variable-speed oil-free centrifugal chiller with magnetic bearings monitored as part of this Green Proving Ground program activity was highly efficient compared to the previous rotary-screw chiller, there is still room for improvement by integrating additional best practices. The following best practices are identified to assist in optimizing chiller plant operations.

**Dashboard**—The optional dashboard (available from most OEM providers) is capable of providing considerable information 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 monitor energy consumption (GSA 2010). The dashboard system for the variable-speed chiller is capable of providing this information and communicating the data to the BAS. However, in touring multiple GSA installations, it was noted that no installation had fully installed this capability. Equipping the dashboard to monitor and communicate current cooling load (tons), current power (kW), and resulting efficiency (kW/ton), as well as trending this data, will allow operators to better optimize chiller plant operations.

**Condenser water temperature reset**—Also known as (cooling) tower water temperature reset, this best practice recommends running the cooling tower to deliver cooler water to the chiller's condenser coils when outside conditions allow. Using outside air sensors for temperature and humidity, most BAS software can determine when the proper conditions exist for the cooling tower to provide cooler condenser water to the chillers. Lowering the entering condenser water temperature will reduce the load on the compressor and raise the chiller's COP (Thumann 1991). As noted in the previous section, the potential for improvement in COP is dependent on the type of compressor, as well as the compressor load controllers. Further, some chillers, particularly older chillers, have limitations with regard to entering condenser water temperatures. New chillers, and in particular, the variable-speed oil-free centrifugal chiller with magnetic bearings, can accept very low entering condenser water temperatures. While this strategy does increase the energy consumed by the cooling tower fans, the reduction in compressor power is greater than the increase in cooling tower fan power.

**Chilled water temperature reset**—This best practice recommends resetting the chiller water temperature based on the outside air dry-bulb and wet-bulb temperatures, which can also be determined by most BAS. When the cooling load is reduced as a result of cooler outside air temperatures, the lower cooling load can be met with a higher chilled water temperature. Raising the chilled water temperature will reduce the load on the compressor and raise the COP (Thumann 1991). As noted in the previous section, the potential for improvement in COP is dependent on the type of compressor, as well as the compressor load controllers. Also, the primary concern with this strategy is the potential reduction in the ability to dehumidify when the

chilled water temperature is raised. Under design conditions, chilled water supply temperatures are kept low to allow the cooling coil to also 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 is less reason for the low chilled water temperature. Monitoring the dew point, or wet-bulb temperature, allows the control system to determine when this strategy can be engaged (only when current conditions allow).

**Smaller and quieter**—The variable-speed oil-free centrifugal compressor with magnetic bearings is smaller and lighter than other compressor types of similar capacity. In some applications, this can be a benefit. Although the compressor is smaller, when installed on a frame including the evaporator and condenser coils, the chiller’s footprint appears to be similar to conventional chillers. Therefore, physical size may or may not be a benefit to the facility. The variable-speed oil-free centrifugal compressor with magnetic bearings does run quieter than other chiller options. Whether it is a result of the single-shaft compressor design, the magnetic bearings, or the oil-free operation, the new chiller generates less noise than alternative chillers. The chiller is rated between 70 and 73 dBA at 5 feet (below the OSHA noise threshold [29 CFR 1910]). By comparison, rotary-screw chillers may be rated as high as 85 dBA at 5 feet.

### C. BARRIERS AND ENABLERS TO ADOPTION

The most recognized barriers to this technology include cost, familiarity, and procurement specification issues. The variable-speed oil-free centrifugal chiller with magnetic bearings is more expensive than conventional chiller options. In general, the OEM chiller equipment cost can be expected to be about twice the cost of conventional chiller options.

Conventional chiller options, including reciprocating, rotary-screw, and scroll compressors, have been around for ages. These compressor options have been the preferred design choices for smaller chiller capacities ranging up to 200 to 300 tons (individual compressor capacity). Centrifugal chillers were relegated to larger capacities, becoming more efficient as equipment capacities exceeded 300 to 500 tons. The new variable-speed oil-free centrifugal compressor is a radically new design concept. The individual compressor capacity is small, starting at 60-nominal tons. Larger capacity units are frequently configured using multiple compressors rather than a single larger compressor. Designers and specifiers are becoming more familiar with the new compressor design, which is helping to overcome this barrier. After several years of testing and hosting demonstrations of the new technology, the Navy is starting to consider the variable-speed oil-free centrifugal chiller with magnetic bearings as the emerging standard for modular chiller plants. Conversely, the Department of the Army has yet to install its first unit, even as a demonstration. Lack of familiarity and initial first cost still restrict the application of this potential energy saver.

Procurement and specifications issues are still a potential barrier to widespread deployment. The variable-speed oil-free centrifugal chiller with magnetic bearings is more commonly known by the model/brand name of the compressor. Initially, this raised sole-source issues, making the equipment difficult to procure, even after the company started manufacturing units in the United States.<sup>15</sup> Over the past few years, however, several large OEMs have started offering chillers with the new compressor. Currently, there are eight OEMs in the United States. Internationally, there are over two dozen OEMs that offer chillers with the variable-speed oil-free centrifugal compressor with magnetic bearings. Additional effort may still be

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<sup>15</sup> Opening a manufacturing facility in the United States assisted in overcoming Buy American Act (41 U.S.C.10a–10d) concerns.

necessary to develop a performance specification to assist federal facilities in buying the highly efficient chiller.

#### **D. MARKET POTENTIAL WITHIN THE GSA PORTFOLIO**

The variable-speed oil-free centrifugal chiller with magnetic bearings can be designed and integrated into almost any chilled-water plant, but the savings are likely to be greatest as an alternative to rotary-screw and reciprocating chillers. While the compressors are small, with nominal capacities of 60 to 200 tons, the modular compressors can be stacked to create larger integrated chillers. OEM water-cooled chillers are available with capacities ranging from 60 to 1200 tons. Larger chiller plants can be designed using multiple chillers. Air-cooled versions are also available in the lower capacity models, although the P100 only allows air-cooled chillers with capacities below 65 tons (GSA 2010). This makes chillers with the variable-speed oil-free centrifugal compressor with magnetic bearings applicable to many current water-cooled chiller applications. Based on the economic performance, the variable-speed oil-free centrifugal chiller with magnetic bearings should be considered for any new construction, as well as for end-of-life chiller replacement. On a retrofit basis, the variable-speed oil-free centrifugal chiller with magnetic bearings should be carefully assessed to determine if the run-time, partial load factor, and local electricity and demand rates will justify the replacement cost.

Insufficient resources are available to identify the number of cost-effective candidates throughout the GSA portfolio of buildings. It is recommended that the variable-speed oil-free centrifugal chillers with magnetic bearings be a required technology assessment during facility energy audits, which are required in 25% of GSA facilities each year (EISA 2007). It is estimated that the market potential within GSA for the application of variable-speed oil-free centrifugal chillers with magnetic bearings could reduce annual energy consumption by around 22,943 MWh/yr. This equates to potential energy savings of 0.6% of GSA's FY2007 total energy consumption. These estimates are based on the following assumptions:

- space cooling accounts for 9.6% of GSA's annual energy consumption (consistent with CBECS (EIA 2003a));
- annual space cooling energy broken down by source equipment type for GSA facilities is similar to estimates for U.S. facilities in general; 3% by rotary-screw chillers, 12% by reciprocating chillers, and 14% by centrifugal chillers (Westphalen et al. 2001);
- Chillers using the variable-speed oil-free centrifugal compressor with magnetic bearings can reduce space cooling energy consumption an estimated 40% compared to chillers using rotary-screw and reciprocating compressors (consistent with the findings in this report); and
- based on GSA's reported energy consumption during FY2007 (FEMP 2010).

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

As for any chiller replacement, it is recommended that thermal load calculations be performed by a qualified engineer to ensure the cooling plant is designed to meet the thermal load without excessive overcapacity. Relying on the size of the previous chiller plant capacity is not an acceptable sizing strategy. Given the energy reduction achievements by GSA, as well as other improvements in modern design techniques, previous and existing equipment may have excess capacity and be oversized to meet current loads. While the new chiller technology is still young, chiller service life is expected to be similar to other chiller equipment. With elimination of the oil systems, maintenance is expected to be simpler and require less time. The new chillers are different, and as such, operators should receive updated training from the manufacturer or other qualified source. Responses from local operators have been positive.

## VII. Appendices

### A. ABBREVIATIONS AND ACRONYMS

The following is a list of abbreviations, acronyms, and terms as used in this report.

Term	Description
AHRI	Air Conditioning, Heating and Refrigeration Institute
Amp	ampere (electric current)
ARRA	American Recovery and Reinvestment Act of 2009
ASHRAE	American Society of Heating, Refrigerating, and Air Conditioning Engineers
BAS	building automation system
Btu	British thermal unit
Btu/h	British thermal units per hour
CB ECS	Commercial Buildings Energy Consumption Survey
CFR	U.S. Code of Federal Regulations
COP	coefficient of performance, a unitless measure for efficiency
$C_p$	specific volume, $\text{Btu}/\text{lb}_m\text{-}^\circ\text{F}$
DAS	data acquisition system
dc	direct current
DOE	U.S. Department of Energy
EIA	U.S. Energy Information Agency
EISA	Energy Independence and Security Act of 2007
ESCO	energy-service company
ESPC	energy savings performance contract
EWT	entering water temperature
FEMP	U.S. DOE Federal Energy Management Program
gal	gallons
GPG	Green Proving Ground
gpm	gallons per minute
GSA	U.S. General Services Administration
H	hour
hp	horsepower
HVAC	heating, ventilating, and air conditioning
Hz	hertz (frequency)
IECC	International Energy Conservation Code
IPLV	integrated part-load value, a seasonal average efficiency measure using kW/ton
lb, lbs	pounds
lbm	pound mass
kW	kilowatt
kWh	kilowatt-hour



Term	Description
kW/ton	power input per capacity, an inverse efficacy measure
min	minute
MWh	megawatt-hour
NPLV	non-standard part-load value, a seasonal average efficiency measure using kW/ton with measurements taken at non-standard conditions
OAT	outside air temperature, also outdoor air temperature
OEM	original equipment manufacturer
O&M	operations and maintenance
P100	GSA Facilities Standards for the Public Buildings Service
Ph	phase (electric)
$P_i$	power input, in kW
PNNL	Pacific Northwest National Laboratory
psig	pounds per square inch gage pressure
$\dot{q}$	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
RH	relative humidity
RLA	run load amps (same as full load amps)
Techval	U.S. Department of the Navy's Technology Validation Program
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
USC	United States Code
V	voltage (electric potential)
VFD	variable-frequency drive
$\rho$	density, lb <sub>m</sub> /ft <sup>3</sup>
$\Delta T$	temperature difference
$^{\circ}F_{db}$	degree Fahrenheit, dry-bulb temperature
$^{\circ}F_{wb}$	degree Fahrenheit, wet-bulb temperature

## B. REFERENCES

- AHRI (2011), 2011 Standard for Performance Rating Of Water-Chilling and Heat Pump Water-Heating Packages Using the Vapor Compression Cycle. ANSI/AHRI Standard 550/590, Air Conditioning, Heating and Refrigeration Institute, Arlington, VA.
- ARRA (2009), American Recovery and Reinvestment Act. Public Law 111-5.
- ASHRAE (2010), Energy Standard for Buildings Except Low-Rise Residential Buildings, ASHRAE Standard 90.1-2010, American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, GA.
- Buy American Act (1933), 41 USC 10.
- CFR (2011), 29 CFR 1919.95 Occupational Health and Environmental Control, Occupational noise exposure. U.S. Code of Federal Regulations, Washington, DC.
- Criscione, P. (2006), Look! Up in the Chiller Room! It's a Screw Compressor, It's a Centrifugal... No, It's Turbocor! New Technology Files into Town with Superbenefits, ER-06-8, E Source, Boulder, CO.
- Criscione, P. and Zuboy, J. (2012), Water-Cooled Electric Chillers, Resource Guide. CEMC-RG-7, E Source, Boulder, CO.
- EISA (2007), Energy Independence and Security Act. Public Law 110-140.
- EIA (2003a), Commercial Buildings Energy Consumption Survey. Table 8. Types of Heating Equipment Used in Office Buildings, 2003, U.S. Energy Information Administration, Washington, DC, <http://www.eia.gov/emeu/cbecs/cbecs2003/officereport/office3.html>, last accessed 07/05/2012.
- EIA (2003b), Commercial Buildings Energy Consumption Survey: Energy End-Use Consumption Tables, Table E.1. Major Fuel Consumption (Btu) by End Use for Non-Mall Buildings, 2003, U.S. Energy Information Administration, Washington, DC, [http://www.eia.gov/emeu/cbecs/cbecs2003/detailed\\_tables\\_2003/2003set19/2003pdf/e1-e11.pdf](http://www.eia.gov/emeu/cbecs/cbecs2003/detailed_tables_2003/2003set19/2003pdf/e1-e11.pdf), last accessed 07/05/2012.
- FEMP (2010), Annual Report to Congress on Federal Government Energy Management and Conservation Programs Fiscal Year 2007, U.S. Department of Energy, Washington DC, <http://www1.eere.energy.gov/femp/pdfs/annrep07.pdf>, last accessed on 07/10/2012.
- FEMP (2012), FEMP Designated Product: Water-Cooled Electric Chillers, U.S. Department of Energy, Washington, DC, [https://www1.eere.energy.gov/femp/technologies/eep\\_wc\\_chillers.html](https://www1.eere.energy.gov/femp/technologies/eep_wc_chillers.html), last accessed 09/12/2012.
- GSA (2010), Facilities Standards for the Public Buildings Service, P100, U.S. General Services Administration, Washington DC, [www.epa.gov/P100](http://www.epa.gov/P100), last accessed 7/12/2012.

IECC (2012), 2012 International Energy Conservation Code. International Code Council, Inc. Washington, D.C.

Thumann, A. (1991), Energy Conservation in Existing Buildings Deskbook. A. Thumann, Editor. The Fairmont Press, Atlanta, GA.

Westphalen, D. and Koszalinski, S. (2001), Energy Consumption Characteristics of Commercial Building HVAC Systems, Volume 1: Chillers, Refrigerant Compressors, and Heating Systems. Aurthur D. Little, Inc. Cambridge, MA.

Wilcox, S. and Marion, W. (2008), User's Manual for TMY3 Data Sets, NREL/TP-581-43156. National Renewable Energy Laboratory. Golden, CO, [http://rredc.nrel.gov/solar/old\\_data/nsrdb/1991-2005/tmy3/](http://rredc.nrel.gov/solar/old_data/nsrdb/1991-2005/tmy3/), last accessed 6/14/2012.