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Exam Preview:

1. According to the reference material, with the deployment of HEDS, peak day peak cooling loads can be cut by approximately 20%.
   a. True
   b. False
2. Based on DoD ESTCP test results, the HEDS unit will cut the average summertime need for cooling and reheat energy by ap-proximately ___ or more.
   a. 30%
   b. 40%
   c. 50%
   d. 60%
3. The background design work for the HEDS technology dates back to ____. The experience gained designing, implementing and testing cooling coils with extremely high chilled water system temperature differentials between ____ and 2007 was critical to the development of the HEDS technology
   a. 1980
   b. 1985
   c. 1990
   d. 1995
4. According to the reference material, the lowest reasonable dewpoint temperature that a HEDS unit can provide without requiring defrost cycles is approximately 55 ° F, so many industrial processes can use HEDS.
   a. True
   b. False

5. Which of the following most common dehumidification designs applicable to chilled water systems matches the description: capture energy from an exhaust or return air stream and transfer it directly to the supply air stream downstream of the cooling coil, providing the reheat to raise the temperature of the subcooled air off the cooling coil.
    a. Air-to-air heat exchangers
    b. Run-around coils
    c. Heat pipe coils
    d. Rotary Wheel heat exchanges
6. Considering the boiler efficiency, cycling losses, and distribution losses, typical delivered boiler system efficiencies can range from 30 to 85% de-pending on the system design, controls, delivery medium (steam, high temp hot water, hot water), and load factors.
    a. True
    b. False
7. Which of the following most common dehumidification designs applicable to chilled water systems has the disadvantage of: Higher air pressure drop due to additional air coils requiring more fan energy.
    a. Air-to-air heat exchangers
    b. Run-around coils
    c. Heat pipe coils
    d. Rotary Wheel heat exchanges
8. A top-level analysis was performed to determine potential Savings to In-vestment Ratios (SIRs) for various HEDS applications. The SIRs ranged from a low of 2 to a high of over ___.
    a. 100
    b. 300
    c. 50
    d. 150
9. One key difference is that the HEDS unit will typically cost 4 to 5 times that of a typical AHU, given the large coil sections, low face velocities, and enhanced controls.
    a. True
    b. False
10. Since HEDS is so similar to a typical AHU, lifetimes are expected to be similar to any other chilled water AHU. According to the reference material, what is the average equipment lifetimes for a Severe Duty or 100% Outdoor Air Units?
    a. 35 years
    b. 30 years
    c. 20 years
    d. 25 years

Abstract

The current “industry standard” method to control relative humidity (RH) and biological growth involves sub-cooling air to condense moisture out of the air, then reheating the same air that was just sub-cooled to reduce the RH of the air before it enters the space. However, the heating, ventilating, and air-conditioning (HVAC) systems at many Federal Facilities are not equipped with (or do not use) the required reheat function, so high indoor RH and the growth of mold are often inevitable occurrences. The High Efficiency Dehumidification System (HEDS) is a patent-protected, proprietary energy recovery method designed to save more than 50% of the dehumidification-related cooling and heating plant energy in RH controlled environments. This work validated the performance of a new HVAC dehumidification technology and investigated performance claims, installation costs, and maintenance impacts through the installation of two test units at Tinker Air Force Base (AFB), OK and Fort Bragg, NC. Based on the results of the ESTCP tests from Fort Bragg, NC and Tinker AFB, OK, HEDS significantly exceeded the energy savings targets, providing HVAC system savings related to the cooling, dehumidification and reheat process of 50% to well over 70%. HEDS appears to be a viable, low maintenance, effective alternative to current RH control technologies, and can be a significant contributor to meeting energy savings Policies, Mandates, and Executive Orders.

Executive Summary

Proper relative humidity (RH) control is critical to maintaining healthy and productive indoor environments in buildings. It is estimated that U.S. companies waste as much as $48 billion annually in medical costs and $160 billion annually in lost productivity as a result of sick building syndrome (Mumma 2006). Mold remediation costs associated with poor RH control have been observed to top $1 million annually on some military bases. Proper RH control minimizes the potential for indoor air quality problems and related sick-building illnesses while improving thermal comfort and productivity (Vavrin 2006).

The current “industry standard” method to control RH and biological growth involves sub-cooling air to condense moisture out of the air, then reheating the same air that was just sub-cooled to reduce the RH of the air before it enters the space. This method has been used for over 100 years, and is known to be very energy intensive due to the need for reheat. However, the reheat process is extremely important in dehumidification applications. The cold, 100% RH air leaving the air-handling units (AHUs) needs to be warmed up to eliminate the potential for surface condensation to occur in the space and to eliminate condensation in the space, which is critical to the control of mold and biological growth.

The heating, ventilating, and air-conditioning (HVAC) systems at many Federal Facilities are not even equipped with the required reheat function, so the growth of mold is an often inevitable occurrence. Many more of the facilities do not use the installed reheat function, as the energy expense is very high, and “common sense” tells people that you should not be running boilers to produce 180 °F hot water in the middle of the summer in humid environments, even though it is needed to perform the required reheat function. As a result, many Federal facilities have the compounded problems of excessive energy use and excessive biological growth, coupled with an HVAC system design or operation that actually promotes mold growth.

The High Efficiency Dehumidification System (HEDS) is a patent-protected, proprietary energy recovery method designed to save more than 50% of the dehumidification-related cooling and heating plant energy in RH controlled environments while also eliminating the health, wellness, product and productivity loss risks caused by poor RH control. By design, the HEDS system is simple and easily maintained; it requires knowledge of only basic

HVAC system operations. HEDS is designed to be scalable, from the smallest room level equipment to the largest central system equipment.

The basic concept underlying HEDS is very simple, and the need for the system is global. The HEDS process recovers 20 to 40% of the low-quality heat generated in the cooling and dehumidification process and uses that reclaimed heat for two purposes: (1) to eliminate the need for new reheat energy for RH control, and (2) to reduce the cooling load sent to the chiller plant from the HEDS AHU by the exact same amount of energy as is recovered to provide the reheat energy. The combined energy savings can exceed 60% during non-peak load conditions. The actual chiller plant and boiler plant energy savings related to the cooling, dehumidification and reheat process can exceed 80% for certain loads in humid environments.

Objective of the demonstration

The objective of this project was to validate the performance of a new HVAC dehumidification technology designed to significantly reduce energy use associated with dehumidification, while improving indoor air quality and reducing potential for mold growth. Performance claims, installation costs, and maintenance impacts were investigated through the installation of two test units at Tinker Air Force Base (AFB), OK and Fort Bragg, NC.

Technology description

The HEDS technology is very simple; a standard AHU is built with a pair of deep, low face velocity heat transfer coils: a cooling coil and a cooling recovery coil. The first coil does the cooling and dehumidifying, the second coil uses the warm water leaving the cooling coil to do the reheating for RH control and cuts the loads on the chiller and boiler plants by using the low quality recovered cooling energy to meet reheat loads. The result is a dehumidification system that is energy efficient, maintainable and resilient.

Demonstration results

Two test units were installed, a Variable Air Volume (VAV) system at Tinker AFB, OK and a Constant Air Volume (CAV) system at Fort Bragg NC. This report summarizes the observed field performance results from more than 6 months of real world testing for both sites. Performance tests were conducted across a range of supply air dew point temperatures to emulate the needs of various building types in the U.S. Department of Defense (DoD), General Services Administration (GSA), Veterans Administration (VA), and Federal building portfolios.

For the constant volume system at Fort Bragg, the peak day cooling load savings was 18%, while the average cooling load reduction was 25%. For the VAV system at Tinker AFB, the peak day cooling load savings was 29%, while the average cooling load reduction was 28%. The peak load reductions effectively expand the capacity of the existing chilled water systems, enabling the chiller plants to serve more cooling loads with the installed capacity, or to be downsized in the future for use in new construction projects. Both these benefits can help reduce capital costs.

Based on the results of the ESTCP HEDS tests from Fort Bragg, NC and Tinker AFB, OK, the energy reclamation function of HEDS is able to significantly reduce the cooling load associated with dehumidification while completely eliminating the need for additional reheat energy to provide RH control in a variety of facility types. Cooling load savings range from 20 to 37% depending on the application, and the dehumidification-related heating energy savings associated with the reheat function at the AHU is 100% in all cases.

Note that the actual cooling energy percentage savings that will show up at the utility meter can be a much greater figure than the cooling load savings percentage. This is due to the non-linear relationship between energy use and load on modern variable speed equipment such as pumps, fans and chillers. For example, reducing the cooling load on chilled water pumps with variable speed drives by 20% typically results in electricity savings of around 40%.

The results from the two ESTCP test sites indicate that HEDS exceeded the energy savings targets by a significant amount. Chiller plant energy savings related to the dehumidification process varied between 32% for hospital-type applications with 24/7 cooling loads, to 64% for administrative type VAV cooling loads that only need conditioning 12/5, but that are typically run 24/7 during the dehumidification season in humid climates. Reheat energy savings related to the dehumidification process were 100% for the test sites.

Implementation issues

Both demonstration sites had issues with failing chillers that led to high chilled water supply temperatures from the chiller plants. Even as chilled water supply temperatures rose as high as 60 °F, both HEDS units were able to continue to provide dehumidification while reducing cooling loads by 16 to 30%. The cooling load saved by the HEDS unit was used by the other AHUs on the chilled water system to provide added cooling to those spaces, which will lead to improved comfort, productivity, health, and wellness, even when chiller performance was sub-optimal. A HEDS installation can improve resiliency by doing more with less.

Throughout the demonstration, HEDS was shown to have the same, or slightly lower, maintenance needs as a normal AHU. In other words, the system's needs are significantly lower than the needs of other commercial dehumidification technologies. Technology transition is occurring through ongoing presentations, white papers, and direct project analysis with Federal energy managers and vendors, all of which combine to demonstrate performance results, and to illustrate implementation strategies for HEDS.

Conclusions

The installed HEDS units met or exceeded each Performance Objective target outlined in the original study plan. HEDS was able to deliver average dehumidification season cooling load savings ranging from 25 to 29%, while eliminating the need for additional reheat energy sources.

The HEDS units were able to maintain internal temperature and RH conditions 96 to 98% of the time. Internal conditions were maintained within RH conditions that typically do not allow biological growth to occur. When Federal facilities are required to comply with American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) 90.1 energy codes, it is likely that HEDS will be the least first cost option and the lowest lifecycle cost option compared to currently available alternatives. HEDS has the same, or slightly lower, maintenance needs as a normal AHU, thus the needs are significantly lower than many of the alternatives.

HEDS appears to be a viable, effective alternative to current RH control technologies, and can be a significant contributor to meeting energy savings Policies, Mandates, and Executive Orders. In addition to working on land-based assets, the technology can be applied to both combatant and non-combatant ships with similar effects (ship-based applications are currently being investigated under award N00167-17-BAA-01 with Naval Surface Warfare Center [NSWC] Carderock Division).

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Preface

Funding for this demonstration was provided by the Environmental Security Technology Certification Program (ESTCP) under FY13 Energy and Water Project EW-201344, “High Efficiency Dehumidification System (HEDS)” via Military Interdepartmental Purchase Requests (MIPRs) No. W74RDV30312782 (31 January 2013), W74RDV41828281 (01 July 2014), W74RDV50785262 (24 March 2015), W74RDV61477740 (27 May 2016), W74RDV30312785 (31 January 2013), W74RDV50785263 (24 March 2015), W74RDV61477741 (27 May 2016).The ESTCP technical monitor was Sarah Medepalli.

The work was performed by the Energy Branch (CFE) of the Facilities Division (CF), of the Facilities Division (CF), of the U.S. Army Engineer Research and Development Center, Construction Engineering Research Laboratory (ERDC-CERL). At the time of publication, Giselle Rodriguez was Chief, CEERD-CFE. The Deputy Director of ERDC-CERL was Dr. Kirankumar V. Topudurti and the Director was Dr. Lance D. Hansen.

COL Ivan P. Beckman was Commander of ERDC, and Dr. David W. Pittman was the Director.

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1 Introduction

1.1 Background

The objective of this project is to validate the performance of a new heating, ventilating, and air-conditioning (HVAC) dehumidification technology designed to significantly reduce energy use associated with dehumidification, while improving indoor air quality and reducing mold growth. This work was undertaken to investigate performance claims, installation costs, and maintenance impacts through the installation of two test units, at Tinker AFB, OK and Fort Bragg, NC.

1.2 Background

Proper relative humidity (RH) control is critical to maintaining healthy and productive indoor environments in buildings. It is estimated that U.S. companies waste as much as $48 billion annually in medical costs and $160 billion annually in lost productivity as a result of sick building syndrome (Mumma 2006). Mold remediation costs associated with poor RH control have been observed to exceed $1 million annually on military bases. Proper RH control minimizes the potential for indoor air quality problems and related sick-building illnesses while improving thermal comfort and productivity (Vavrin 2006).

The current "industry standard" method to control RH and biological growth involves sub-cooling air to condense moisture out of the air, then reheating the same air that was just sub-cooled to reduce the RH of the air before it enters the space. This method has been used for over 100 years, and is known to be very energy intensive due to the need for reheat. However, the reheat process is extremely important in dehumidification applications. The cold, 100% RH air leaving the air-handling units (AHUs) needs to be warmed up to eliminate the potential for surface condensation to occur in the space and to eliminate condensation in the space, which is critical to the control of mold and biological growth.

Unfortunately, the HVAC systems at many Federal Facilities are not equipped with the required reheat function. Many more of the remaining facilities do not use the installed reheat function because the energy expense is very high and because "common sense" tells people that you should not be running boilers to produce 180 °F hot water in the middle of the summer in humid environments, even though it is needed to perform the required reheat function. As a result, many Federal facilities have the compounded problems of excessive energy use and excessive biological growth, coupled with an HVAC system design or operation that actually promotes mold growth.

To combat those problems, in 2006, Retrofit Originality Incorporated was approached by the U.S. Army Corps of Engineers (USACE) to develop a cost effective, energy efficient, maintainable, sustainable and scalable dehumidification and RH control solution. The solution had to work in retrofit applications as well as new construction. After substantial research, development and computer modelling, the High Efficiency Dehumidification System (HEDS) was born. The HEDS is a patent-protected, proprietary energy recovery method designed to save more than 35% of the cooling and heating energy in RH controlled environments while also eliminating the health, wellness, product and productivity loss risks caused by poor RH control. It is essentially a standard AHU, equipped with a very large face area and depth cooling coil designed to deliver very warm chilled water (CHW) return temperatures, and a "Cooling Recovery Coil" (CRC) designed to reclaim 20 to 40% of the wasted low quality heat that was generated in the cooling and dehumidification process. This reclaimed waste heat is used for two purposes: (1) to completely eliminate the need for new reheat energy for RH control, and (2) to reduce the cooling load sent to the chiller plant from the HEDS AHU by the exact same amount of energy as is recovered to provide the reheat energy. The combined energy savings can exceed 60% during non-peak load conditions. The actual chiller and boiler plant energy savings related to the cooling, dehumidification and reheat process can exceed 70% for certain loads in humid environments.

The HEDS system offers many potential benefits that will impact a number of missions throughout DoD. These include saving energy; reducing condensation in AHUs, ducts, and occupied spaces; reducing lifecycle costs; and improving the health, comfort and productivity of employees—all of which are extremely important to DoD. The development of HEDS makes it possible to resolve the problems described above. Peak day peak cooling loads can be cut by approximately 20% and the reheat energy required for proper RH control on peak load days can be eliminated completely.

1.3 Objective of the demonstration

The main technical objective of this project is to evaluate the HEDS unit design in two real world buildings to determine if there are technical issues that must be addressed before full scale commercialization. Additional objectives are to quantify the extent to which systems are able to be downsized to assess the level of improved efficiency of the HVAC systems, and to determine the extent to which upgrade costs can be reduced. All previous development has been undertaken via computer analysis using cooling coil and heating coil rating programs and differing design conditions varying from recirculated air type systems in barracks in the Midwest to 100% dedicated outside air systems (DOAS) in the tropics. The analysis has shown that peak day peak loads can have substantial reductions for all of the test conditions that were evaluated, and if variable volume air distribution systems are used, the part load savings can also be very substantial. This project will verify the actual performance of the HEDS.

- Validate: The ESTCP demonstration project will validate the performance, costs, and benefits of the technology in the following manner. Performance will be validated by measuring and calculating the energy saved by the HEDS units at two separate locations and facilities types. Expected energy savings will occur at the chiller plant due to reduced cooling loads and pump energy savings due to higher CHW system temperature differentials, and at the boiler plant or power plant due to reduced/eliminated need for reheat energy. Costs will be validated by using the actual costs of the equipment and installation process that would normally be required for a unit replacement, i.e., excluding the research and development (R&D) costs and excluding instrumentation and controls costs associated with the demonstration process. Benefits will be calculated and determined based on the savings and results of the demonstration process.
- Findings and Guidelines: Once the HEDS technology has been proven to perform in hot and humid climates and any limitations have been discovered and rectified through the ESTCP demonstration process, it will be much easier to make the case for widespread adoption of the technology. The results of the ESTCP demonstration process will prove the levels of savings that the design can potentially make available. Proposed recommendations to DoD policies and standards may include: mandating proper HVAC and humidity control designs for high RH locations; mandating that no new energy be used for the reheat

portion associated with RH control; mandating that maintenance requirements for dehumidification systems be no greater than those of a "normal" AHU; and mandating that the loads served by the cooling plant be less than the sum of the loads associated with the cooling and dehumidification process when RH control is occurring.

- Technology Transfer: After the completion of the project, the results will be published in the ASHRAE Journal and other outlets such as the U.S. Green Building Council (USGBC), Austrian Energy & Environment (AEE), and Building Owners and Managers Association International (BOMA). The findings can also be presented at USACE, the Energy Exchange, Resource Efficiency Manager (REM) and Energy Service Co. (ESCO) conferences to educate them on how they can improve energy efficiency at their client facilities. Recommendations will be developed for revisions to relevant Unified Facilities Criteria (UFC) and Unified Facilities Guide Specifications (UFGS).
- Additionally, the participation of Trane as a team member can provide rapid and scaled deployments of the proposed technology for DoD. With over 400 offices in 100 countries worldwide, Trane has the resources to transfer the technology and to rapidly deploy HEDS at DoD facilities around the globe.
- Acceptance: The implementation of the HEDS ESTCP project at the two demonstration sites is intended to demonstrate that this new, simple to understand and operate technology will save energy, reduce capital costs to control moisture condensation in AHUs' ducts and occupied spaces repair expenditures, reduce lifecycle costs, and improve the comfort of the buildings' occupants, and that the system will have the same or fewer operational and maintenance requirements as the conventional systems it is replacing.

1.4 Regulatory drivers

The regulatory drivers listed below are intended reduce the energy utilization intensity (EUI) of Federal buildings on an annual basis. This requirement is dictated primarily by the Energy Policy Act of 2005. This project will directly support the cost effective attainment of these goals by reducing the amount of energy used in the HVAC system for dehumidification, heating, and cooling. In a typical DoD building, the HVAC energy is about 30 to 40% of the total energy. The component for cooling, dehumidification, and reheat for RH control expends up to 40% (even more in very humid climates) of the total energy in humid climates. Proper applications of this technology should reduce that amount by an average in the range of

30 to 40%, thereby reducing the energy total by about 5% for the total building energy use. The application of this one technology breakthrough can help DoD meet almost 2 years' worth of energy intensity reduction goals that are a 3% energy intensity annual reduction. There is also a potential water savings for systems that use hydronic cooling towers for heat rejection, but this will not be validated as part of this demonstration. The HEDS solution has a further effect of reducing capital costs for new central plant installations, and of reducing equipment maintenance costs due to reduced run time of the chiller, pumps, and boiler.

The regulatory drivers underlying this demonstration are:

- Executive Orders: Executive Order (EO) 13423, EO 13514, EO 13693: http://www.whitehouse.gov/administration/eop/ceq/sustainability; https://www.federalregister.gov/articles/2015/03/25/2015-07016/planning-for-federal-sustainability-in-the-next-decade
- Legislative Mandates: Energy Policy Act of 2005, Energy Independence and Security Act of 2007, National Defense Authorization Act (NDAA) for Fiscal Year 2015: http://www.armed-services.senate.gov/press-releases/senate-committee-on-armed-services-reach-agreement-with-house-counterparts-regarding-the-national-defense-authorization-act-for-fiscal-year-2015
- Federal Policy: Federal Leadership in High Performance and Sustainable Buildings Memorandum of Understanding (MOU) 2006
- DoD Policy: Strategic Sustainability Performance Plan, Energy Security MOU with U.S. Department of Energy (DOE)
- Service Policy: Army Sustainable Design and Development Policy Update, 16 December 2013: http://www.usace.army.mil/Portals/2/docs/Sustainability/Hydrology_LID/ASAIEE_SDD_Policy_Update_2013-12-16.pdf, Secretary of the Navy Energy Goals: http://www.navy.mil/features/Navy_EnergySecurity.pdf, Air Force Sustainable Design and Development Implementing Guidance: http://www.wbdg.org/ccb/AF/POLICY/af_sdd_impl_guidance.pdf; http://www.safie.hq.af.mil/shared/media/document/AFD-091208-027.pdf
- Guides: Whole Building Design Guide (http://www.wbdg.org/). See specifically: http://www.wbdg.org/pdfs/usace_dg_epact2005.pdf; http://www.wbdg.org/references/mou_ee.php
- Specifications: ASHRAE Standards 62.1 and 90.1, Leadership in Energy and Environmental Design [LEED], Institute of Electrical and Electronics Engineers [IEEE], International Code Council (ICC) Codes (International Mechanical Code [IMC], International Plumbing Code [IPC], International Energy Conservation Code [IECC,] etc.).

2 Technology Description

2.1 Technology overview

The High Efficiency Dehumidification System (HEDS) reclaims some of the very low quality heat generated during the cooling and dehumidification process in the chilled water stream, and uses it to provide the reheat energy used to lower the RH of the air supplied to buildings, which reduces the potential for condensation to occur and reduces reheat requirements to ensure that spaces are not overcooled due to dehumidification processes. The energy that is reclaimed for reheat, has a compounding benefit; every British Thermal Unit (BTU) of energy that is used for reheat also reduces the cooling load on the chiller plant by the exact same amount.

2.1.1 System description

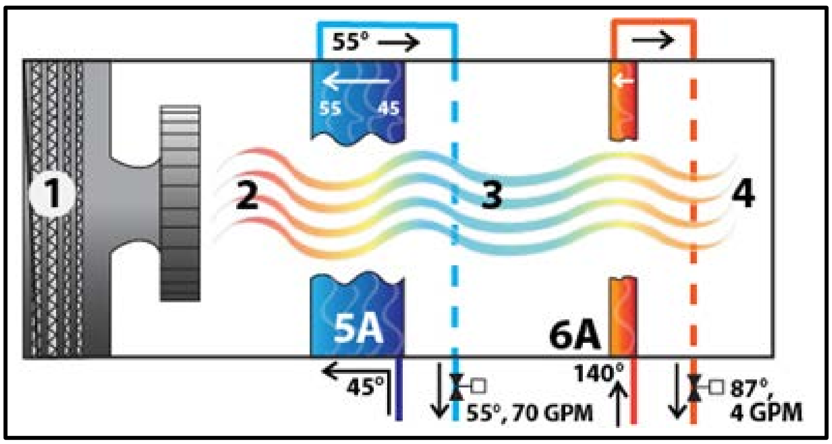
HEDS is a "Cooling Recovery System" designed to save lives and substantially reduce energy waste, reduce space RH and improve occupant safety, comfort and productivity. In hospitals, laboratories, and manufacturing facilities, the improved temperature control and RH stability can lead to better patient outcomes, and improved product quality. In administrative and other facilities, HEDS can reduce energy waste; eliminate biological growth; and improve occupant health, wellness, and productivity.

Based on DoD ESTCP test results, the HEDS unit will:

- Recover between 18 and 29% of the heat generated in the chilled water stream from the cooling and dehumidification process to maintain RH control during peak cooling load periods.
- Reduce total cooling loads between 25 and 37% by recovering heat generated during the cooling and dehumidification process to maintain RH control.
- Eliminate the need for new reheat energy (for example from reheat coils or electric strip elements) for RH control for all dehumidification loads encountered at both test sites.
- Cut the average summertime need for cooling and reheat energy by approximately 50% or more, while simultaneously reducing potable water usage in the cooling and heat rejection process for systems that use water cooled chiller equipment.
- Cut dehumidification-related energy use by 50 to over 80% for the chiller plant and boiler plant.

By far the most common solution used in dehumidification AHUs is to sub-cool the air to remove moisture by condensation, then reheat the sub-cooled and dehumidified air to lower the RH of that air and provide temperature control for the spaces (Figure 1). The reheat energy can be provided by hot water coils fed from a central boiler system, on-board furnace, or electric strip heating elements.

Figure 1. Typical reheat dehumidification design.

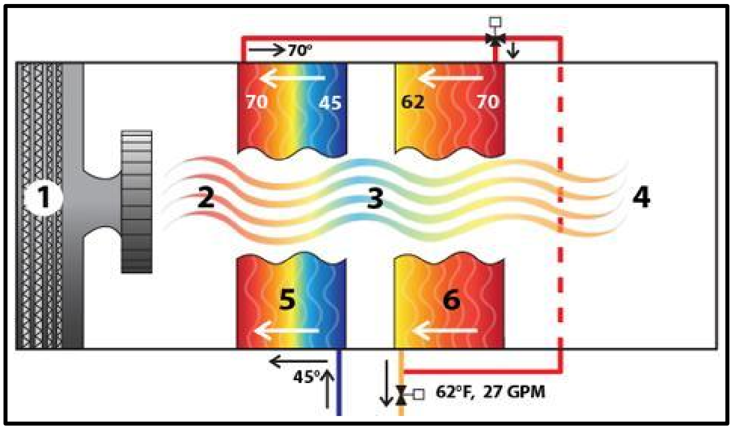


Data Points 1 through 4 in Figure 1 denote: [1] 10,000 CFM airflow [2] 78 °F dry bulb temp, 65 °F wet bulb temp [3] 55 °F dry bulb, 55 °F dew-point, essentially 100% RH [4] 65.3 °F dry bulb, 55 °F dewpoint, 55% RH. In the diagram, the air is moving through the system from the left to the right.

Typical AHUs providing dehumidification and reheat use relatively small, high air velocity cooling and reheat coils, high CHW flow rates, low CHW temperature differentials, and high AHU air pressure drops. In the example above, 45 °F CHW enters the cooling coil (5A) at 70 gallons per minute (GPM) and leaves the cooling coil at 55 °F. A new source of 140 °F water enters the reheat coil (6A) at 4 GPM and leaves the reheat coil at 87 °F. The unit requires 479,319 BTUs per hour to cool, dehumidify and reheat 10,000 CFM of air at the design conditions in this example.

Figure 2 shows how HEDS eliminates the need for new reheat energy and reduces the total cooling load of the unit, using the same design conditions shown in Figure 1.

Figure 2. High efficiency dehumidification system.



The HEDS units use very large, low air face velocity cooling and cooling recovery coils, low CHW flow rates, high CHW temperature differential, and low AHU air pressure drops. In this example, which matches the base case air conditions above, 45 °F CHW enters the cooling coil (5) at 27 GPM and leaves the cooling coil at 70 °F. This 70 °F water then enters the CRC coil (6) at 27 GPM and leaves the CRC coil at 62 °F while heating the air from 55 °F up to 65 °F. The HEDS unit requires 226,187 BTU per hour to cool, dehumidify and reheat 10,000 CFM of air at the same conditions, a total British Thermal Unit per Hour (BTUH) savings of 53% and a CHW flow reduction of 62% in this example.

Attributes of the HEDS unit include:

- Very large face area and depth cooling and cooling recovery coils
- Low CHW flow rates and High CHW temp differential
- Increased cooling capacity at lower CHW flows
- Elimination of "Low Delta T syndrome"
- Low AHU air pressure drops due to large coil face area and low face velocity
- Ability to reduce equipment run time by thousands of hours per year on non-8,760 loads
- Delivery of cool, dry air in an energy efficient manner
- Reduction of infrastructure and operations and maintenance (O&M) costs
- Reduction of pumping and chiller energy use
- Configuration that allows chillers to be piped in series to further improve chiller capacity and energy efficiency
- Increased CHW system infrastructure delivery capacity, saves infrastructure $$$.

- Reduction of water consumption/can generate water from condensation
- Support for ASHRAE 90.1 Prescriptive Energy Code Compliance.

A number of existing technologies currently on the market for dehumidification have significant limitations when compared with the HEDS system design, including:

- Increased maintenance costs due to complexity of additional fluid stream, pumps, heat recovery wheels, heat exchangers, motors, belts and other components
- Potentially decreased CHW system temperature differential due to smaller coils and reduced inlet air temperatures to the cooling coil, leading to the "Low Delta T Syndrome," which can increase central plant energy use and reduce cooling system usable capacity
- Poor temperature control due to uncontrolled inlet temperatures from heat recovery coils
- Added regeneration heat energy and post-wheel cooling associated with some desiccant designs
- Much longer, taller or heavier AHUs
- Higher air pressure drop and fan energy due to additional upstream and downstream coils and wheels requiring more fan energy
- Condensate re-evaporation when water is blown off the cooling coil into the fan or ductwork
- Designs that are not scalable to room or fan coil unit sizes, where many of the problems are found.

The following is an example of how the HEDS unit may operate in the field.

> To serve a specific load at a specific time, the mixed air may need to be sub-cooled to 52 °F using the Cooling Coil (CC) to condense and remove moisture, and then the air must be reheated back up to 62 °F to control RH and prevent space over cooling. To do this, the CRC is used to warm up the supply air leaving the cooling coil with the warmer water leaving the cooling coil. This will lower the RH of the supply air entering the space to prevent overcooling of the space and reduce the potential for condensation to occur in the space.
>
> Two hours later, occupancy may have reduced substantially, the HEDS AHU may have adequately removed moisture from the space, and the outside air (OSA) may have a dewpoint below 53 °F. Therefore, the unit

> only needs to cool the air to 60 °F to meet the load. Supply air can be supplied without any sub-cooling or reheat, because the space or return air dewpoint is at 52 °F, so the supply air RH is around 75% and it is going into a relatively dry space.
>
> If the space is very hot and muggy, the AHU may need to provide 48 °F supply air off the cooling coil to remove moisture, and provide 58 °F dry-bulb temperature coming off the CRC to dry the space out quickly, and reduce the potential for condensation to occur.

The cooling/ dehumidification/ reheat loads change constantly, and the control strategies will take the changing loads into account on a continuous basis. The HEDS unit is equipped with a standalone control system capable of performing all required functions (described below).

2.1.2 HEDS standalone unit controller description

The HEDS units are equipped with factory programmed standalone controllers. The HEDS controller hardware for these two test sites consists of the Trane UC 600 hardware platform, which is configured to receive inputs from all of the HEDS sensors and alarms, including water temperatures, water differential pressures, Belimo Energy Valve data, airside dry-bulb and dewpoint temperature sensors, airside RH sensors, airside airflow rates, filter alarms, low pressure cutout alarms, and low temperature cutout alarms. In addition, the HEDS controller will accept data from the variable speed drive network connection.

The HEDS standalone controller is configured to control the valve positions for the preheat coil, the cooling coil, the cooling recovery coil, and the reheat coil (RHC is at Tinker AFB only) to maintain space conditions within Unified Facilities Criteria (UFC) requirements. The outside air, mixed air, and exhaust air dampers are controlled in addition to the speed of the fan motor in response to logic commands contained in the HEDS controller software. The HEDS controller receives inputs from the Building Automation System (BAS) to start and stop the HEDS unit; it also sends requests to start the chiller plant and boiler plant if the HEDS unit determines the need for after-hours RH or temperature control.

The HEDS controller feeds operational data to the HEDS trending system, the Tracer SC system. The HEDS trending system stored data for the monitored and calculated variables at 5-minute intervals for retrieval and evaluation.

There are three basic operational modes for the HEDS unit contained in the HEDS standalone controller, with underlying mode specific sequences that describe the detail of how the system will be operated when in those modes

1. The first and simplest main mode is the heating mode. In this mode, the unit will operate the supply fan and the heating coil(s) as required to maintain the space conditions as needed.
2. The second main mode is the cooling-dehumidification-reheat mode of operation. This mode is where the majority of the energy savings occur. When in this mode, the cooling coil is operated to cool the air and to remove moisture from the air to reduce the RH in the space. In this mode, the cooling recovery coil is operated to increase the supply air temperature of the supply air, which will lower the RH of the supply air and the space conditions. Operation of the CRC will also reduce the potential for overcooling of the spaces and reduce the potential for condensation to occur in the HVAC ductwork and the occupied spaces.
3. The last main mode is the cooling-only mode. When the space temperatures and dewpoint are under control, and the outside air and mixed air temperature dewpoint temperatures are low enough to reduce the potential for condensation to occur, the CRC logic will be disabled and the cooling coil will still provide cooling and possibly dehumidification, but with no need to operate the CRC.

Other operational modes include:

- Startup mode. During initial system startup the system is enabled with a 5- to 10-minute delay to reduce the potential for unneeded system spikes.
- Overnight batch dehumidification mode. The overnight batch dehumidification mode can be selected by the operating staff if they feel the need to operate the HEDS unit to keep the facility dried out during hot muggy conditions when the facility is normally shut down. The batch mode would be used when conditions are bad but not terrible, and the HEDS unit may need to be started once or twice a night to prevent the facility from becoming humidity saturated. Due to chiller plant operational issues at both sites, it was not possible to test this sequence.
- Overnight continuous dehumidification mode. The overnight continuous dehumidification mode can be selected by the operating staff if they feel the need to operate the HEDS unit to keep the facility dried out during hot muggy conditions when the facility is normally shut

down. The continuous mode would be used when conditions are very moist, and the HEDS unit may need to be run continuously overnight to prevent the facility from becoming humidity saturated. Due to chiller plant operational issues at both sites, this sequence was unable to be tested.

- DOAS Mode. To simulate being a 100% outside air Dedicated Outdoor Air System unit. Due to chiller plant operational issues at both sites, this sequence could not be tested.
- Economizer mode. The economizer mode would be enabled when conditions show that use of the economizer would provide benefits to the facility. The economizer mode would have three triggers that would keep it off, or shut it off if it is operational: comparative enthalpy, outdoor dewpoint temperature and outdoor drybulb can all be used to limit the use of the economizer when it may impact humidity control.

These sequences should be operated seamlessly; facility occupants should feel no discernible changes. All these operating modes can occur daily during transitional weather patterns.

2.2 Technology development

The background design work for the HEDS technology dates back to 1985. The experience gained designing, implementing and testing cooling coils with extremely high chilled water system temperature differentials between 1985 and 2007 was critical to the development of the HEDS technology; therefore, the technology development timeline below includes representative work in those time frames. Some of the high points of the experience that has supported the technology development are:

- 1985 – Large Temperature Differential (LTD) cooling system designs, with systems designed to deliver 76 °F chilled water return temperatures when the cooling coils were provided with 39 °F chilled water from a chilled water thermal energy storage system.
- 1985 to 2007 – Installation of hundreds of LTD cooling coils at dozens of facilities prove the ability to reliably obtain cooling coil leaving chilled water temperatures in excess of 70 °F in the summer.
- 1992 to 2005 – University of Southern California (USC) campus converts the majority of their cooling coils to LTD design. Campus chilled water system temperature differential (TD) increases from 8 °F to 9 °F in the summer (“Low Delta T Syndrome”) to 25 °F to 27 °F. This enables a 300% increase in cooling capacity through the existing CHW

piping distribution system, saving millions of dollars for USC. This also enables a proposed 9MMG Thermal Energy Storage (TES) tank to be downsized to a 3MMG TES tank, saving additional millions of dollars.

- 2006/7 – Site visits to multiple DoD facilities to evaluate hundreds of buildings shows biological growth in human-occupied spaces is still a large problem. This further proves the need for a dehumidification system that is cost effective, efficient, reliable, maintainable, and sustainable, and that can be used with either two-pipe or four-pipe water distribution systems.
- 2007 – The USACE Challenge to Principal Investigator while on a base in a room full of biological growth was stated as a directive to "figure out a way to solve the biological growth problem with a system that our guys can understand and that can be maintained with very low maintenance budgets." The HEDS unit was developed to address this challenge.
- 2007 - Development of HEDS design, which reclaims very low quality cooling energy as a reheat energy source to eliminate the need for new reheat energy for RH control, and to save chiller plant energy at the same time.
- 2007 - 2011 – HEDS patents applied for and awarded.
- Technology Maturity for Commercialization: The results from the two ESTCP test sites indicates that the savings potential is on par with the estimated, modeled savings potential. Additionally, the HEDS units' maintenance is no different than that of a standard chilled water AHU.

2.3 Advantages and limitations of the technology

2.3.1 Comparative technologies

HEDS AHUs have a number of advantages over existing dehumidification systems solutions. Some of the most common dehumidification designs applicable to chilled water systems are:

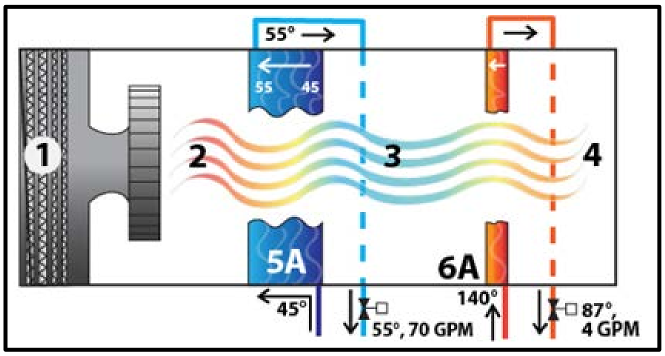
- Chilled water/ direct expansion coils with gas or electric reheat
- Run-around coils
- Heat pipe coils
- Rotary wheel heat exchangers
- Air-to-air heat exchangers
- Desiccant dehumidification wheels.

The following sections briefly discuss each system.

2.3.2 Chilled water/ direct expansion coils with gas or electric reheat

By far the most common and energy intensive solution used in dehumidification AHUs is the sub-cooling of the air to remove moisture by condensation, then to add heat to reheat the sub cooled and dehumidified air back up to lower the RH of that air and provide temperature control for the spaces (Figure 3). The reheat energy can be provided by hot water coils fed from a central boiler system, on-board furnace, or electric strip heating elements. Cooling energy can be provided by chilled water coils or direct expansion (DX) coils.

Figure 3. Typical reheat dehumidification design.

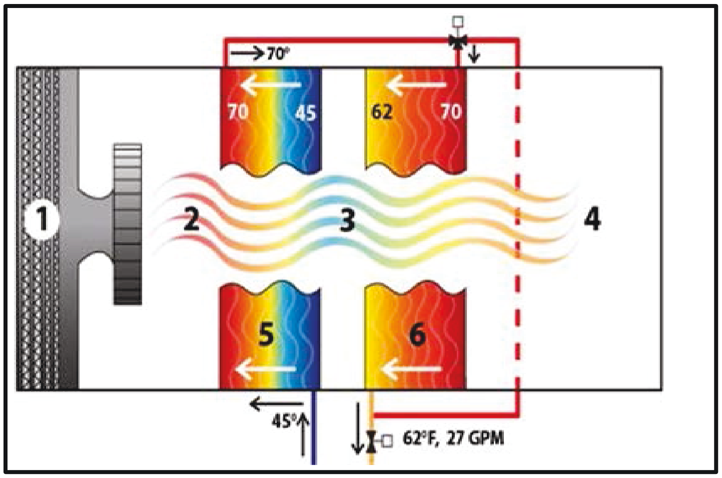


Data Points 1 through 4 in Figure 3 denote: [1] 10,000 CFM airflow [2] 78 °F dry bulb temp, 65 °F wet bulb temp [3] 55 °F dry bulb, 55 °F dewpoint, essentially 100% RH [4] 65.3 °F dry bulb, 55 °F dewpoint, 55% RH

Typical AHUs providing dehumidification and reheat use relatively small, high air face velocity cooling and reheat coils, high CHW flow rates, low CHW temperature differential and high AHU air pressure drops. In this example, 45 °F CHW enters the cooling coil (5A) at 70 GPM and leaves the cooling coil at 55 °F. A new source of 140 °F water enters the reheat coil (6A) at 4 GPM and leaves the reheat coil at 87 °F. The unit requires 479,319 BTUs per hour to cool, dehumidify and reheat 10,000 CFM of air at the design conditions in this example.

Figure 4 shows how HEDS eliminates the need for new reheat energy and reduces the total cooling load of the unit, using the same airside design conditions as shown in Figure 3 above.

Figure 4. HEDS unit depiction highlighting the mechanism for eliminating reheat and reducing cooling load.

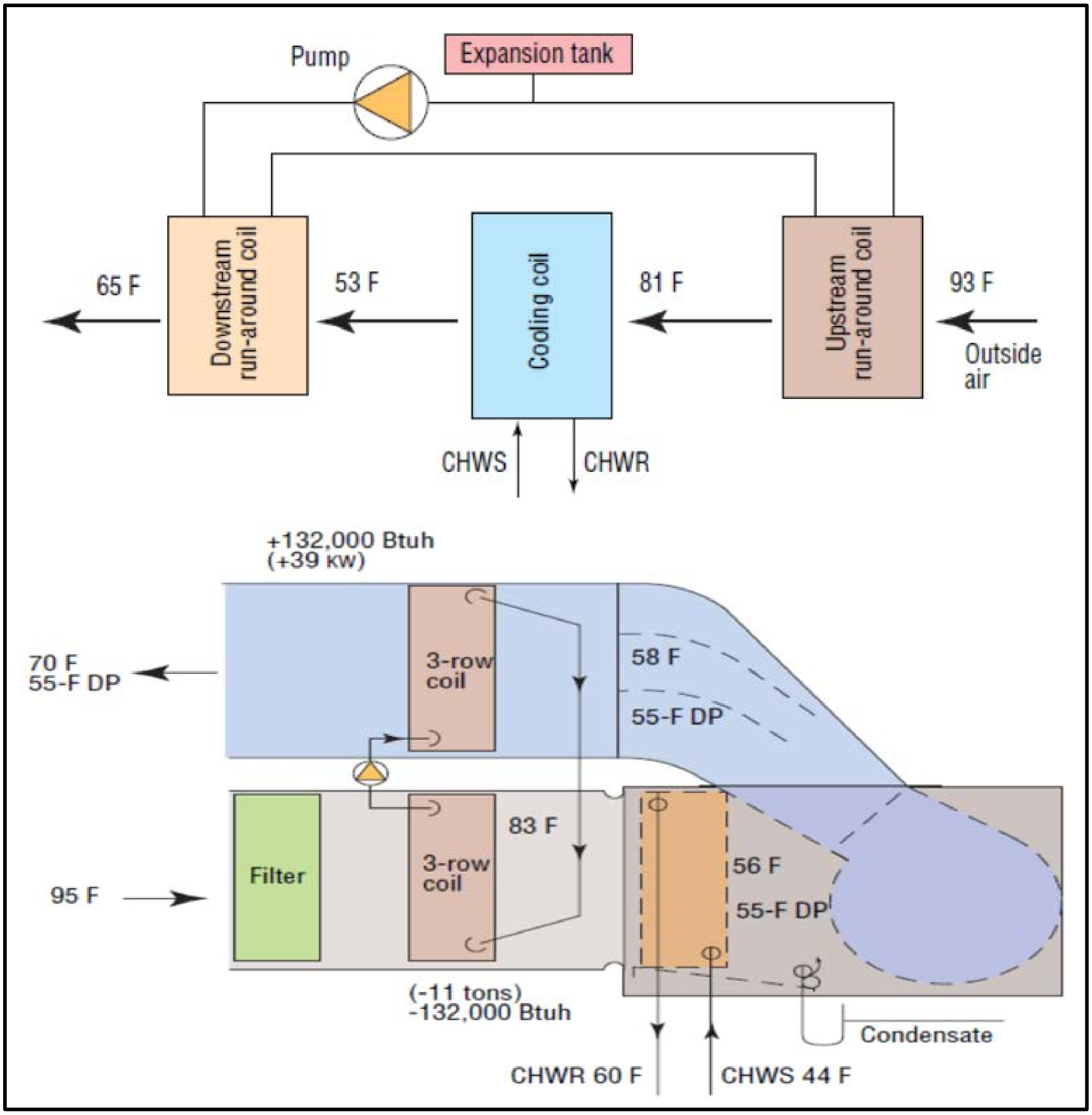


The HEDS units use very large face area and depth cooling and cooling recovery coils, low CHW flow rates, high CHW temperature differential, and low AHU air pressure drops. In this example, which matches the conditions above, 45 °F CHW enters the cooling coil (5) at 27 GPM and leaves the cooling coil at 70 °F. This 70 °F water then enters the CRC coil (6) at 27 GPM and leaves the CRC coil at 62 °F while heating the air to 65 °F. The HEDS unit requires 226,187 BTU per hour to cool, dehumidify and reheat 10,000 CFM of air at the same conditions, a BTUH savings of 53% and a CHW flow reduction of 62% in this example.

2.3.3 Run-around coils

Run-around coil designs use a set of coils to accomplish reheat with reduced energy consumption (Figure 5). The coils can be placed in the return/ exhaust air streams or in the upstream outside air stream or mixed air plenum before the cooling coil to provide the reheat energy. The run-around coil examples below show the pre-cool coil upstream of the cooling coil to act as a heat source for the downstream reheat coil to provide some heat for reheating in dehumidification units.

Figure 5. Run-around coil system example layouts.



Source: Donald P. Gatley, P.E. President, Gatley and Associates, for HPAC Engineering Magazine in 2000

Compared with HEDS technology, run-around coils have a number of disadvantages, including:

- Increased maintenance costs due to complexity of additional fluid stream, pumps, and components.
- Potentially decreased CHW system Delta T due to smaller coils and reduced inlet air temperatures to the cooling coil, leading to the "Low

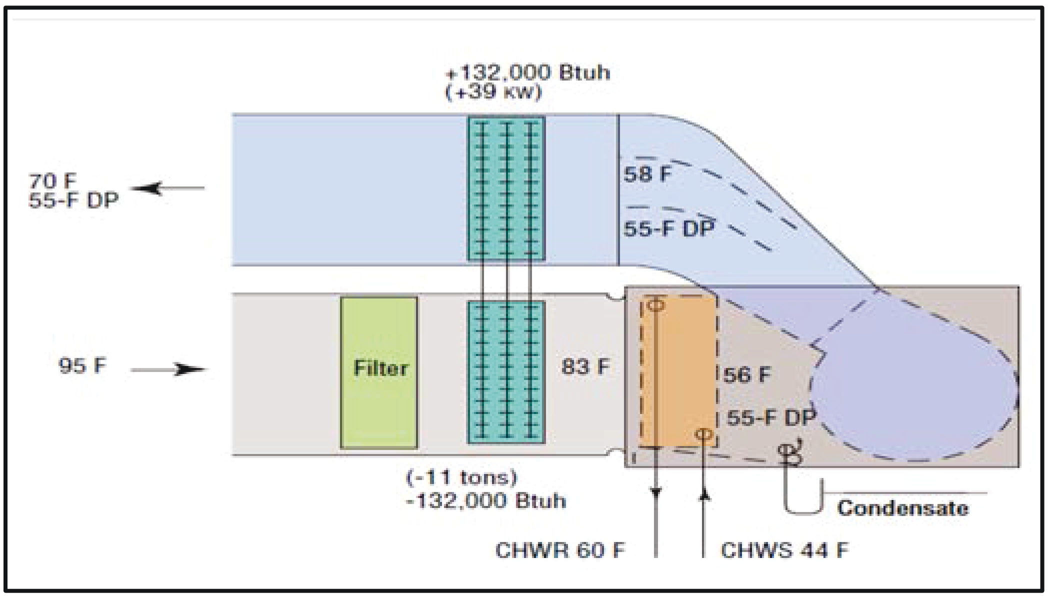
Delta T Syndrome." This can increase central plant energy use and reduce cooling system capacity.

- Poor temperature control due to uncontrolled inlet temperatures of the run-around coil.
- Much longer or taller AHU.
- Higher air pressure drop due to additional upstream and downstream coils requiring more fan energy.
- Condensate re-evaporation when blown off cooling coil.
- A design that is not scalable to Fan-Coil Unit (FCU) sizes.

2.3.4 Heat pipe coils

Heat pipes are very similar to run-around coils, but are refrigerant based, which eliminates the need for additional pumps, expansion tanks, and other ancillary equipment (Figure 6).

Figure 6. Heat pipe coil design example.



Source: Donald P. Gatley, P.E. President, Gatley and Associates, for HPAC Engineering Magazine in 2000

Compared with HEDS technology, heat pipe coils have a number of disadvantages, including:

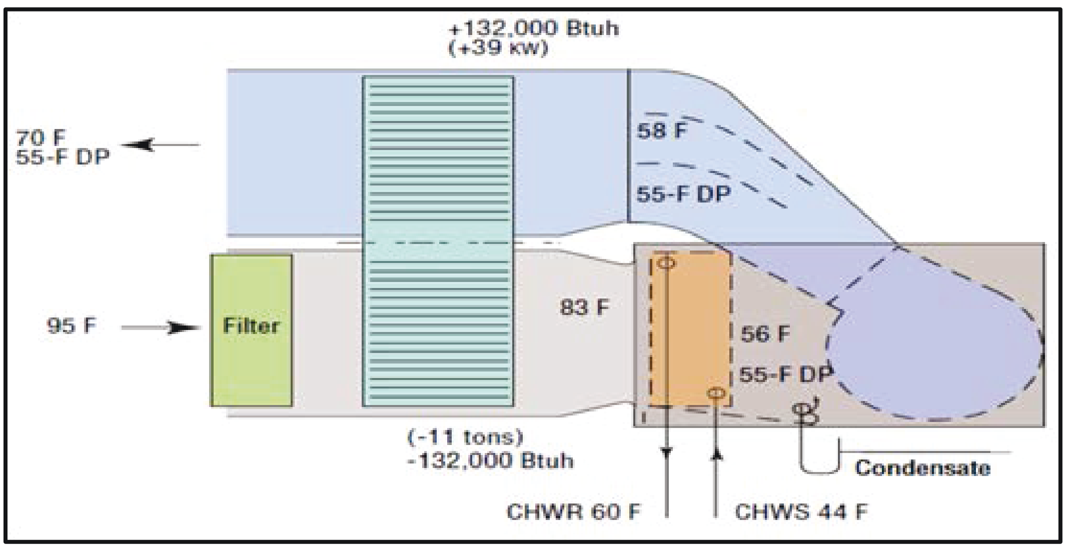
- Increased maintenance costs due to complexity of refrigerant transfer coils.
- Potentially decreased CHW system Delta T due to smaller coils and reduced inlet air temperatures to the cooling coil, leading to the "Low

Delta T Syndrome." This can increase central plant energy use and reduce cooling system capacity.
- Poor temperature control due to uncontrolled or minimally controlled inlet temperatures of the heat pipe coil.
- Much longer or taller AHU.
- Higher air pressure drop due to additional upstream/downstream coils requiring more fan energy.
- Condensate re-evaporation when blown off cooling coil.
- Design is not scalable to FCU sizes.

2.3.5 Rotary wheel heat exchangers

Rotary wheel heat exchangers such as sensible energy recovery wheels can also be used to provide the reheat energy associated with dehumidification (Figure 7). Enthalpy (or total energy) wheels can also be used to reduce the humidity of incoming air streams. Rotary wheels capture energy from an exhaust or return air stream and transfer it directly to the supply air stream downstream of the cooling coil, providing the reheat to raise the temperature of the subcooled air off the cooling coil.

Figure 7. Rotary heat exchanger design example.



Source: Donald P. Gatley, P.E. President, Gatley and Associates, for HPAC Engineering Magazine in 2000

Compared with HEDS technology, rotary heat exchangers have a number of disadvantages, including:

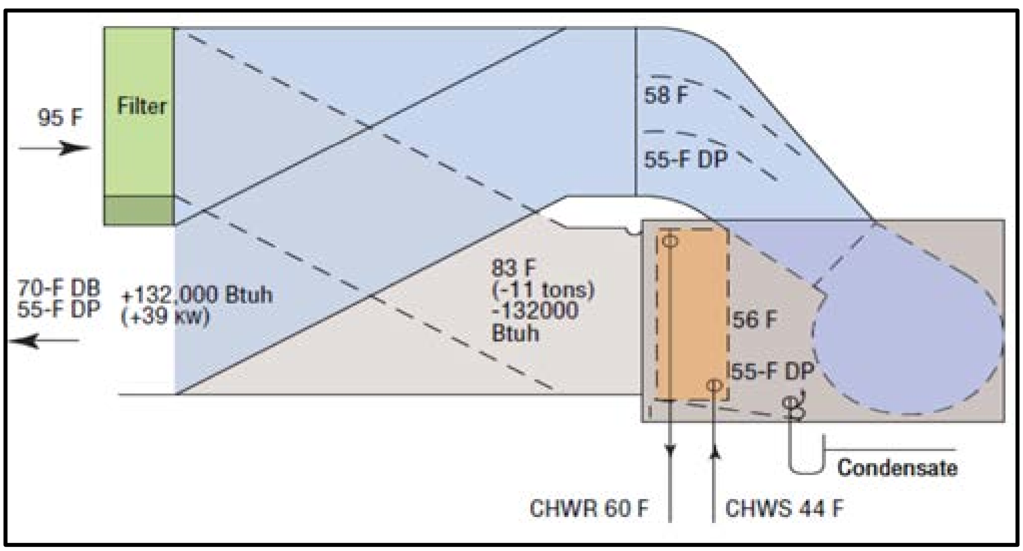
- Increased maintenance costs due to complexity of heat transfer wheels and motors.

- Plugging or contamination of heat exchanger wheels caused by imperfect air filtration, which can significantly decrease performance.
- Potentially decreased CHW system Delta T due to smaller coils and reduced inlet air temperatures to the cooling coil, leading to the "Low Delta T Syndrome." This can increase central plant energy use and reduce cooling system capacity.
- Much longer or taller AHU.
- Higher air pressure drop due to added losses in both supply and exhaust streams for the rotary wheel requiring more fan energy.
- Condensate re-evaporation when blown off cooling coil.
- A design that is not scalable to FCU sizes.

2.3.6 Air-to-air heat exchangers

Air-to-air heat exchangers use a set of plate heat exchangers to accomplish reheat without additional energy by recovering energy from the outside air or return air streams to provide reheat (Figure 8).

Figure 8. Air-to-air heat exchanger system example layouts.



Source: Donald P. Gatley, P.E. President, Gatley and Associates, for HPAC Engineering Magazine in 2000

Compared with HEDS technology, air-to-air heat exchangers have a number of disadvantages, including:

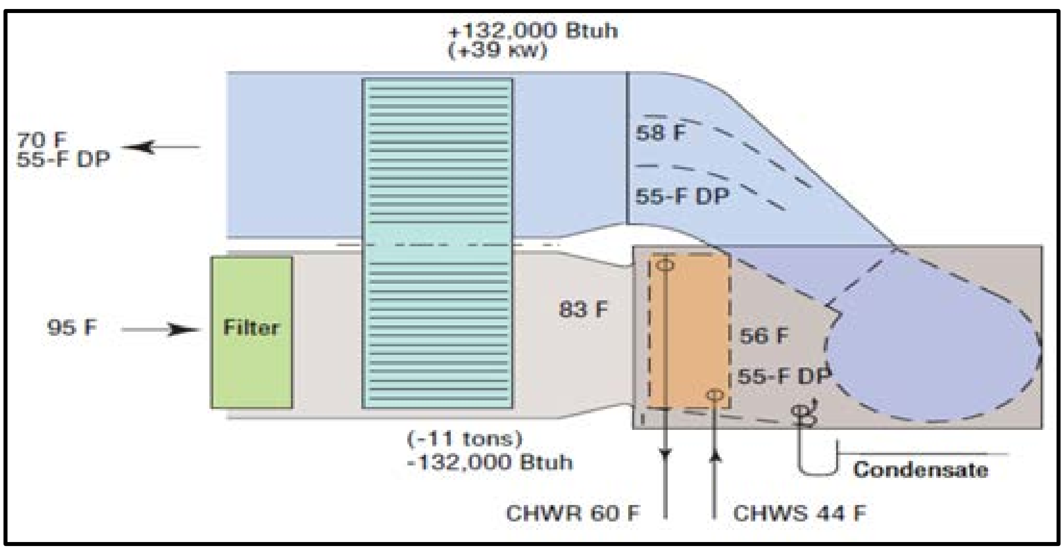
- A potentially decreased CHW system Delta T due to smaller coils and reduced inlet air temperatures to the cooling coil, leading to the "Low Delta T Syndrome." This can increase central plant energy use and reduce cooling system capacity.

- Poor temperature control due to uncontrolled inlet temperatures of the air-to-air heat exchanger.
- Much longer or taller AHU.
- Significantly higher ductwork costs.
- Higher air pressure drop due to additional air coils requiring more fan energy.
- Condensation may form inside the Heat Exchanger (HX).
- Condensate re-evaporation when blown off cooling coil.
- Design is not scalable to FCU sizes.

2.3.7 Desiccant dehumidification

At this time, a correct thermodynamic or physical diagram for the desiccant wheel design is not available, but it is vaguely similar to the rotary wheel diagram shown in Figure 9 if the temperature points are ignored. Desiccant-based systems use a regenerative desiccant wheel to accomplish dehumidification while eliminating required reheat energy, but adding post-unit cooling energy to reduce the supply air temperature to a reasonable level. The desiccant wheel is placed in the return/exhaust air streams or in the upstream outside air stream before the cooling coil to provide the required dehumidification while reducing cooling and reheat energy. Many desiccant wheel designs require an additional heat source of 200 °F or higher air to regenerate the desiccant to provide sufficient dehumidification.

Figure 9. Desiccant wheel system placeholder layout. (Ignore airside temperatures, rough physical layout only intended as a proxy for the desiccant wheel design)



Source: Donald P. Gatley, P.E. President, Gatley and Associates, for HPAC Engineering Magazine in 2000

Compared with HEDS technology, desiccant wheels have a number of disadvantages, including:

- Much longer or taller AHU.
- Significantly higher air pressure drop, such that desiccant wheel requires more fan energy.
- Added regeneration heat energy with some desiccant designs.
- A need for post wheel cooling (for most desiccants)
- Typically much larger and heavier units.
- A need for post cooling coil to drop supply air temperature off some wheel types from over 90 °F to a usable level.
- Significant increased complexity and maintenance costs.
- A design that is not scalable to FCU sizes.

2.3.8 Performance advantages of the HEDS technology

The HEDS unit uses the very low quality heat that is generated during the cooling and dehumidification process to provide the necessary reheat energy to lower the RH of the supply air. This in turn reduces the potential for moisture condensation in AHUs' ducts and occupied spaces, and reduces the cooling load on the chiller plant. This cooling load reduction in BTUs is equal to the amount of recovered energy that is used to provide reheat. For RH control processes, this eliminates the need for a supplemental reheat source in many climates, thereby substantially reducing both chiller plant and heating plant energy consumption. HEDS can work with two-pipe water distribution systems where other systems may not work.

The HEDS unit is designed exactly as a "normal" AHU would be built, but with two major changes

1. The cooling coil in a HEDS unit has approximately 300% more heat transfer surface area than a normal cooling coil to obtain a very warm water temperature leaving the coil. As a result, this warm water can be used as a reheating energy source by the CRC.
2. The use of a CRC, which has more than 1000% greater heat transfer surface area than a normal reheat coil, which enables the use of the low quality heat leaving the cooling coil. This enables the warm chilled water leaving the cooling coil to raise the temperature of the chilled air leaving the cooling coil so that the building spaces are kept comfortable and the supply air is delivered at a much lower RH.

The act of warming the sub-cooled air leaving the cooling coil using the CRC draws heat from the chilled water and reduces the temperature of the return chilled water, minimizing the load on the chiller. The CRC "looks" like an upstream series chiller to the chilled water system in that it reduces the water temperature returning to the chiller plant by 6 °F to 12 °F. The CRC also looks like a heating coil to the airstream, in that it raises the supply air temperature by 6 °F to 15 °F.

The large face area coil design reduces the air velocity through the AHU, which provides a >50% reduction in air filter and coil air pressure drop; the lower required cooling water flow rate results in >70% pumping energy reduction for the loads served by HEDS units. The combined air pressure drop of the CC and CRC is approximately 50% lower than that of a typical cooling coil and reheat coil combination due to the very large face area of the HEDS coils and the associated very low air velocities through the coils.

2.3.9 Cost advantages of the HEDS technology

A benefit of HEDS is that the chilled water flow rate required to meet peak day cooling/dehumidification needs will be reduced by approximately 50 to 60% compared to typically installed AHU systems. This results from a combination of reduced cooling plant loads and increased chilled water system temperature differentials provided by the very large cooling coils and the CRC. On sites that may be at the capacity limits of their piping infrastructure, the ability to meet the same cooling loads with a 50 to 60% reduction in the chilled water flow rate can mean that the avoided costs from not having to replace or augment the piping infrastructure can cover most or all of the costs of HEDS retrofit projects.

For many installations, if HEDS is not used, to provide code mandated RH control to facilities equipped with two-pipe water distribution systems, reheat energy for RH control must be provided by electric strip heaters, which, ironically, will not comply with new energy codes. The electrical infrastructure of most facilities is inadequate to provide this added power requirement to the buildings and down to the AHU level, so the facilities that need RH control typically go without RH control. This has led to the current situation with widespread biological growth and high biological remediation costs. Health, wellness, productivity and morale all suffer in a facility affected by biological growth.

Since the HEDS unit is a normal AHU built with very large cooling and cooling recovery coils, it is no more complex or costly to operate and maintain than a conventional AHU. A basis of its design is that it is intended to be maintained by HVAC mechanics with a basic level of maintenance training to reduce the lifecycle cost on the unit. Other systems, such as desiccant driven or direct expansion (DX) type dehumidifiers, require operators to have specialized maintenance and operations knowledge, and require additional energy use to perform RH and temperature control in comparison to the HEDS unit.

2.3.10 Performance limitations of the HEDS technology

Adequate physical space will need to be allocated for the HEDS units, which cannot be located in very tight mechanical spaces as they can be physically larger than a "normal" AHU. Where there is a lack of space in a very tight mechanical room, it may be possible to locate the HEDS unit next to the loads and cut it into the required point of connection. Note however that HEDS units will typically be smaller than a desiccant wheel-based system that delivers the same conditions.

The lowest reasonable dewpoint temperature that a HEDS unit can provide without requiring defrost cycles is approximately 35 °F, so many industrial processes can use HEDS. For process loads that need to be provided with ultra-low dewpoint air, some form of a desiccant-based system will most likely be the most effective option, as long as there are adequately trained mechanics and an appropriate maintenance budget allocated for this system.

Because a major intent of the HEDS technology is to cool and dehumidify conditioned air in a more efficient and lifecycle cost effective way, the HEDS units will work best at geographic locations that have a hot and humid climate for at least 4 months of the year, or that are in milder climates, but need to provide 48 to 50 °F dewpoint air to their cooling loads, such as hospitals and clean room environments located in the U.S. Southwest or precision semiconductor and pharmaceutical manufacturing facilities across much of the United States.

2.3.11 Cost limitations of the HEDS technology

The cost effectiveness of the HEDS units may be very site specific. On standalone implementations of an HVAC unit that is not currently equipped to provide the "reheat" part of the dehumidification-reheat process, it may be less costly to employ the HEDS unit than to try to create and implement a new reheat energy source, or to convert the unit to one of the other dehumidification/reheat strategies, especially if the installation must comply with energy codes that forbid simultaneous heating and cooling for RH control.

Conversely, if a facility has a 4 pipe water distribution system, and runs the boilers all summer long in addition to running the chillers, and the AHUs are already equipped with reheat coils (not just pre-heat coils), the HEDS unit may have a higher first cost, but the lower operating expenses or other cost offsets, such as reduced capital expenditures for chillers, pipes, pumps, cooling towers, and chiller plant physical room expansions may make it cost effective.

For facilities operating under a mandate to reduce energy and water consumption, the efficiency benefits of the HEDS unit may make it a lifecycle cost effective solution, even if it has a higher first cost.

A top-level analysis was performed to determine potential Savings to Investment Ratios (SIRs) for various HEDS applications. The SIRs ranged from a low of 2 to a high of over 300. Some implementations would have an infinitely high SIR, as the first cost of the HEDS may be lower than the base case alternative system, so there is no "investment"; the system evaluation starts with a cost reduction vs. a cost.

Non-energy benefits such as improved health and wellness, energy resiliency, improved use of renewable energy, and saving lives through the reduction of Healthcare Acquired Infections (HAIs) are non-trivial, and may be the driving forces behind HEDS implementations. The energy benefits are important, but may not be the main reason for the implementations.

3 Performance Objectives

Table 1. Summary of quantitative performance objectives.

Performance Objective	Metric	Data Requirements	Success Criteria
1. Peak Cooling Load Reduction %	Thermal Energy (Tons Refrigeration, kW, mmBTU)	Refrigeration tonnage, CC load, CRC load, supply and return water temperatures, chilled water flow rate through CC and CRC	Reduce 15-minute peak cooling load by 15% on a peak cooling load day during the demonstration period
2. Greatest Cooling Load Reduction %	Thermal Energy (Tons Refrigeration, kW, mmBTU)	Refrigeration tonnage, CC load, CRC load, supply and return water temperatures, chilled water flow rate through CC and CRC	Highest cooling load % reduction exceeds 20% during the demonstration period
3. Dehumidification /Reheat Coil Energy Reduction	Thermal Energy (Tons Refrigeration, kW, mmBTU)	CC load, CRC load, chilled water supply and return temperatures, chilled water flow rate through CC and CRC, reheat coil (RHC) load, hot water supply and return temperatures, RHC flow	CRC coil eliminates the need for at least 90% of the RH-control -related reheat energy required from the reheat coil when the system is in dehumidification-reheat mode during the demonstration period
4. Enhance Space Comfort Conditions	Space and return air conditions compared to UFC comfort zone for summer	Space drybulb and dewpoint temperatures, space RH%, return air drybulb and dewpoint temperatures, return air RH%	Space conditions fall within UFC comfort guidelines more than 90% of the time during occupied hours
5. Reduce Cooling Ton-Hours Consumption	Thermal Energy (Tons Refrigeration, kW, mmBTU)	CC load, CRC load, supply and return water temperatures, chilled water flow rate through CC and CRC	Cooling ton-hours associated with the HEDS unit are reduced by the cooling recovery coil by 7.5% compared to the ton-hours consumed by the cooling coil during the time that the HEDS is in dehumidification-reheat mode during the demonstration period
6. Improve "Low Delta T" Syndrome	Temperature and flow measurements and/or calculations	HEDS CC CHW TD and flow, HEDS CRC CHW TD and flow, HEDS unit CHW TD and flow	HEDS average CHW system TD exceeds 14F during the time that the HEDS is in the cooling or dehumidification-reheat modes during the demonstration period
7. Reduce Greenhouse Gas (GHG) Emissions	Fossil fuel GHG emissions (metric tons)	Information in #8 and estimated source energy GHG production for cooling and reheat energy sources	GHG emission reductions exceed 3% (annual comparison)

Performance Objective	Metric	Data Requirements	Success Criteria
8. Reduce Energy cost of Dehumidification/ Reheat process	%, $	HEDS estimated kWh/ton-hour for chiller plant, HEDS cooling ton-hours for dehumidification, HEDS CRC tons/MMBTU, calculated RHC energy use, estimated chiller and boiler plant system efficiency, kWh and therms Average cost/kWh and cost/therm for Natural Gas (NG)	Cost of Dehumidification and reheat with HEDS vs. CV subcool/ terminal reheat is reduced by 10% during dehumidification-reheat modes of operation.
9. System Economics Reduce Lifecycle cost of Dehumidification/ Reheat process	%, $, years	Estimated and calculated dollar costs and savings, discount rate, usable life	5% reduction in lifecycle costs.

3.1 Performance objective results

Tables 2 and 3 summarize the results of each objective for the test sites. All success criteria were met, and often substantially exceeded, across all objectives at both test sites.

Table 2. Quantitative performance objective results summary for the Fort Bragg Test Site.

Performance Objective	Success Criterion	Results (CHWST<46 °F)
1. Peak Cooling Load Reduction %	Reduce 15-minute cooling load by 15% on a peak cooling load day during the demonstration period	18.3%
2.Greatest Cooling Load Reduction %	Highest average cooling load % reduction exceeds 20% during the demonstration period	37.4%
3. Dehumidification /Reheat Coil Energy Reduction	CRC coil eliminates the need for at least 90% of the RH-control-related reheat energy required from the reheat coil during the time the system is in dehumidification-reheat mode during the demonstration period	100.0%
4. Enhance Space Comfort Conditions	Space conditions fall within UFC comfort guidelines more than 90% of the time during operating hours	96.0%
5. Reduce Cooling Ton-Hours Consumption	Cooling ton-hours associated with the HEDS unit are reduced by the CRC by 7.5% compared to the ton-hours consumed by the cooling coil during the time that the HEDS is in dehumidification-reheat modes during the demonstration period	24.7%
6. Improve "Low Delta T" Syndrome	HEDS average cooling coil CHW system TD exceeds 14 °F during the time that the HEDS is in dehumidification-reheat modes during the demonstration period	17.1%
7. Reduce Greenhouse Gas (GHG) Emissions	GHG emission reductions associated with the dehumidification/reheat process exceed 3% (annual comparison)	45–79%

Performance Objective	Success Criterion	Results (CHWST<46 °F)
8. Reduce Energy cost of Dehumidification/ Reheat process	Cost of dehumidification and reheat with HEDS vs. CV subcool/terminal reheat is reduced by 10% during dehumidification –reheat modes of operation	41–51%+
9. System Economics Reduce Lifecycle cost of Dehumidification/ Reheat process	5% reduction in lifecycle costs	Retrofit: 26–29%+ New construction/ end of useful life (EUL): 38–44%+
10. Savings vary based on central plant chiller and heating system efficiencies; uses eGrid national average electricity emissions factors.		

Table 3. Performance objective summary for the Tinker AFB test site.

Performance Objective	Success Criterion	Results (CHWST<46 °F)
1. Peak Cooling Load Reduction %	Reduce 15-minute cooling load by 15% on a peak cooling load day during the demonstration period	28.9%
2.Greatest Cooling Load Reduction %	Highest average cooling load % reduction exceeds 20% during the demonstration period	28.7%
3. Dehumidification /Reheat Coil Energy Reduction	CRC coil eliminates the need for at least 90% of the RH-control-related reheat energy required from the reheat coil during the time the system is in dehumidification-reheat mode during the demonstration period	100.0%
4. Enhance Space Comfort Conditions	Space conditions fall within UFC comfort guidelines more than 90% of the time during operating hours	98.0%
5. Reduce Cooling Ton-Hours Consumption	Cooling ton-hours associated with the HEDS unit are reduced by the CRC by 7.5% compared to the ton-hours consumed by the cooling coil during the time that the HEDS is in dehumidification-reheat modes during the demonstration period	27.6%
6. Improve “Low Delta T” Syndrome	HEDS average cooling coil CHW system TD exceeds 14 °F during the time that the HEDS is in dehumidification-reheat modes during the demonstration period	24%
7. Reduce Greenhouse Gas (GHG) Emissions	GHG emission reductions associated with the dehumidification/reheat process exceed 3% (annual comparison)	70–86%
8. Reduce Energy cost of Dehumidification/ Reheat process	Cost of dehumidification and reheat with HEDS vs. CV subcool/terminal reheat is reduced by 10% during dehumidification –reheat modes of operation	68–75%+
9. System Economics Reduce Lifecycle cost of Dehumidification/ Reheat process	5% reduction in lifecycle costs	Retrofit: 13–41%+ New construction/ EUL: 43–61%+
12. Savings vary based on central plant chiller and heating system efficiencies; uses eGrid national average electricity emissions factors. 13. Savings vary based on central plant chiller and heating system efficiencies, as well as electricity and gas commodity rates. Average potential savings over a range of potential cost, efficiency, and reheat source scenarios is shown.		

3.2 Performance objectives (POs) descriptions

3.2.1 Determine the peak cooling load reduction percent that occurs as a result of the energy recovered via the CRC during the dehumidification/reheat process.

3.2.1.1 Purpose

The purpose of this PO is to determine the amount of the peak cooling load that can be reduced by using internally reclaimed heating energy for RH control, vs. introducing externally generated heat into the airstream/facility for RH control.

A substantial amount of cooling and heating energy is used in the dehumidification/reheat process. During the cooling season, most HVAC systems use a substantial amount of recirculated air with a small fraction of fresh outside air brought into the facility, usually 10 to 15%. Since up to 90% of the air is being recirculated, up to 90% of the newly introduced reheat energy that is being used for RH control ends up being recirculated by the air distribution system and brought back to the cooling coil in the form of higher return air temperatures and higher cooling loads.

The ability to reduce the peak cooling load can lead to substantially lower operating costs, and can potentially help eliminate the need to expand chiller plants and increase the size of chilled water distribution piping infrastructure.

3.2.1.2 Metric

Tons of cooling required to cool/dry the air via the cooling coil, minus tons of cooling energy reclaimed by the CRC. Tons of cooling required, and subtract the cooling loads that are reduced due to the use of the cooling recovery coil, i.e., if it takes 50 tons to condition the supply air to a 55 °F dew point temperature, and 120,000 BTUs (10 tons) of reheat energy provided by the cooling recovery coil to raise the supply air temperature to lower the RH, a reduction in the cooling load at the chiller plant of 10 tons would be seen, so the peak load reduction in this case would be 10/50 = 20%. The expected range of peak cooling load reduction is between 10% and 20%.

The peak load will be determined as follows:

Determine Peak Day 15-minute Peak Load Reduction % at the Chiller Plant. During dehumidification-reheat operation, reduce the peak-day 15-minute peak cooling load served by the chiller plant for the HEDS AHU by

15% compared to the cooling load served by the cooling coil at the HEDS AHU. This assumes that the chiller plant is delivering the design chilled water supply temperature of approximately 44 to 45 °F.

Determine peak day cooling load by summing up the individual day cooling coil tons measurements from 4 AM to 10 PM and comparing all the days that the system is in the dehumidification-reheat mode. The day with the highest cooling coil ton-hours will be considered the peak load day. Use data from this day for the calculations associated with this PO.

3.2.1.3 Data

The following data are required to evaluate the metric:

CC load, CRC load, chilled water supply temperature.

3.2.1.4 Analytical Methodology

Please refer to Section 6.1, “Savings analysis methodology.”

3.2.1.5 Success criteria

Reduce 15-minute peak cooling load by 15% on a peak cooling load day during the demonstration period.

3.2.2 Determine the greatest cooling load reduction percent that occurs as a result of the energy recovered via the CRC during the dehumidification/reheat process.

3.2.2.1 Purpose

The purpose of this PO is to determine the greatest cooling load reduction percent that can be obtained by recovering cooling energy via the CRC.

The CRC performs two main functions, the first is to provide a recovered source of heat energy to perform RH control and temperature control for the supply air and occupied spaces, and the second function is to reduce the cooling load on the chiller plant by the amount of reheat energy that was used for RH control duties.

Being able to determine the greatest cooling load reduction percent delivers another metric to help determine cost effectiveness and the extent to which cooling plant sizes may be able to be reduced.

3.2.2.2 Metric

Tons of cooling required to cool/dry the air via the cooling coil, minus tons of cooling energy reclaimed by the Cooling Recovery Coil. Tons of cooling required, then subtract the cooling loads that are reduced due to the use of the cooling recovery coil. This is very similar to the first PO, but the project is looking to determine the maximum percent reduction, not the maximum peak load percent reduction. The expected range of greatest cooling load reduction percent is between 15 and 40%.

The greatest load percent reduction will be determined as follows:

Determine greatest load percent reduction at the Chiller Plant by comparing the cooling load served by the cooling coil, to the cooling energy recovered by the CRC. Calculate the percent load reduction for each 5-minute period that the system is in the dehumidification-reheat mode. From this data, calculate the maximum percent load reduction at the chiller plant. Does the maximum percent cooling load reduction load served by the chiller plant for the HEDS AHU exceed 20%? This assumes that the chiller plant is delivering the design chilled water supply temperature of approximately 44 to 45 °F.

3.2.2.3 Data

The following data are required to evaluate the metric:

CC load, CRC load, chilled water supply temperature.

3.2.2.4 Analytical Methodology

Please refer to the Section 6.1, “Savings analysis methodology.”

3.2.2.5 Success criteria

Reduce the cooling load by 20% or more for any 5-minute period that the system is in the dehumidification-reheat mode.

3.2.3 Dehumidification /reheat coil energy reduction. Does the HEDS unit reduce reheat energy required by the downstream reheat coil at Tinker AFB by more than 90% during the dehumidification-reheat process? Fort Bragg AHU does not have a downstream reheat coil, so this PO only applies to Tinker AFB.

3.2.3.1 Purpose

The purpose of this PO is to determine if the CRC can eliminate more than 90% of the reheat-related energy needed for RH control during the dehumidification-reheat process. Ideally, the HEDS CRC would eliminate all reheat energy required for the RH control process.

Does the HEDS unit eliminate the need for more than 90% of the supplemental reheat energy from the downstream reheating coil for RH control at the Tinker AFB test site? This assumes that the chiller plant is delivering the design chilled water supply temperature of approximately 44 to 45 °F during the cooling season.

3.2.3.2 Metric

Total MBTU of heat/cool energy recovered by the CRC and MBTU of heating thermal energy used for reheat in the reheat coil during the dehumidification/reheat process.

The total amount of reheat energy added by the downstream reheat coil during the dehumidification/reheat coil will be compared to the total amount of reheat energy added by the CRC.

The expected range of avoided reheat coil provided reheat energy during the dehumidification/reheat process is between 90 and 100%.

3.2.3.3 Data

The following data are required to evaluate the metric: CRC load, downstream Reheat Coil (RHC) load, chilled water supply temperature.

3.2.3.4 Analytical methodology

Please refer to the Section 6.1, “Savings analysis methodology.”

3.2.3.5 Success criteria

Reduce the amount of reheat energy required to be provided by the downstream RHC at Tinker AFB by 90% or more when the system is in dehumidification-reheat mode. (CRC BTUH/(CRC BTUH + (RHC BTUH)) > 0.9

3.2.4 Enhance space comfort conditions

3.2.4.1 Purpose

The purpose of this PO is to determine if the HEDS unit can deliver comfort conditions that fall within UFC comfort guidelines more than 90% of the time during occupied hours, on an annual basis.

3.2.4.2 Metric:

Space conditions and return air conditions will be compared to UFC comfort guidelines during occupied hours. Because it may be difficult to find one sensor location in the space that represents the entire space, sensor have been included in the return air in the projects. These will be the sensors that are more representative of the entire space being served. The expected annual range of compliance with UFC comfort guidelines is between 90 and 95% during occupied hours.

3.2.4.3 Data

Space conditions and return air conditions – dry bulb temperature, dewpoint temperature and RH will be used to determine whether the system is operating within UFC guidelines during occupied hours. This assumes that the chiller plant is delivering the design chilled water supply temperature of approximately 44 to 45 °F during the cooling season.

3.2.4.4 Analytical methodology

The total hours of space conditions within the UFC guidelines will be determined, compared to the total hours of operations (with chilled water systems delivering approximately design chilled water to the AHUs).

3.2.4.5 Success criteria

Does the HEDS unit deliver comfort conditions that fall within UFC comfort guidelines more than 90% of the time during occupied hours, on an annual basis?

3.2.5 Reduce cooling ton-hours consumption. Determine the cooling load ton-hours savings percent that occurs as a result of the cooling energy recovered via the CRC during the dehumidification/reheat process.

3.2.5.1 Purpose

The purpose of this PO is to determine the amount of the cooling load in ton-hours that can be reduced by using internally reclaimed heating energy via the CRC for RH control, vs. introducing externally generated heat into the airstream/facility for RH control during the cooling season.

A substantial amount of cooling and heating energy is used in the dehumidification/reheat process. The cooling recovery coil has the ability to reduce the annual cooling load that must be served, since it uses reclaimed heat from the cooling process for RH control of the supply air and spaces, rather than introducing a new source of heat into the facility/airstream. The amount of reheat energy saved in BTUH translates exactly into the amount of cooling energy saved (BTUH converted to ton-hours).

Being able to reduce the annual cooling load (ton-hours) can lead to substantially lower operating costs.

3.2.5.2 Metric

Tons of cooling required to cool/dry the air via the cooling coil, compared to tons of cooling energy reclaimed by the CRC.

Tons of cooling delivered by the cooling coil when the system is in the dehumidification-reheat mode of operation, and subtract the cooling loads that are reduced due to the use of the cooling recovery coil. This assumes that the chiller plant is delivering the design chilled water supply temperature of approximately 44 to 45 °F during the cooling season.

The expected range of cooling load reduction during the dehumidification-reheat mode of operation is between 5 and 10% for the systems being tested.

3.2.5.3 Data

The following data are required to evaluate the metric:

CC load, CRC load, chilled water supply temperature.

3.2.5.4 Analytical methodology

Please refer to the Section 6.1, “Savings analysis methodology.”

3.2.5.5 Success criteria

Reduce cooling load ton-hours by 7.5% when the system is in a dehumidification-reheat mode of operation.

3.2.6 Determine if the HEDS unit can provide a chilled water system temperature differential that is higher than typical, to help solve the “Low Delta T” syndrome

3.2.6.1 Purpose

The purpose of this PO is to determine if the HEDS unit can provide a chilled water system temperature differential that is higher than typical, to help solve the “Low Delta T” syndrome.

A substantial amount of chiller plant energy waste occurs due to the “Low Delta T Syndrome.” This problem occurs due to undersized/improperly sized cooling coils that require substantially greater CHW flow at a much lower chilled water system temperature differential to meet the typically occurring cooling loads.

This problem creates the need to move a substantial amount of flow through the system at relatively low loads, which can mean that a facility needs to run multiple chillers, when the load may only equate to 50% of one chiller.

Being able to increase the chilled water system temperature differential can lead to substantially lower operating costs, and can potentially help eliminate the need to expand chiller plants and increase the size of chilled water distribution piping infrastructure.

3.2.6.2 Metric

HEDS unit chilled water temperature differential when in the cooling mode.

The expected range of the HEDS unit chilled water temperature differential for the cooling coil when in the cooling mode is between 15 and 30 °F.

When the system is in the dehumidification-reheat mode of operation, the expected range is between 10 and 20 °F. This assumes that the chiller plant is delivering the design chilled water supply temperature of approximately 44 to 45 °F during the cooling season.

3.2.6.3 Data

The following data are required to evaluate the metric:

CC chilled water temperature differential, CRC chilled water temperature differential, overall HEDS unit chilled water temperature differential, chilled water supply temperature entering the cooling coil.

The cooling coil TD and flow are being taken such that the tons of cooling required for the cooling/dehumidification process can be calculated. The cooling recovery coil TD and flow are being taken such that the BTUs of heating required for the reheat process can be calculated. This data enables simulation of a “normal” cooling/reheat AHU for comparative purposes.

3.2.6.4 Analytical methodology

Please refer to the Section 6.1, “Savings analysis methodology.”

3.2.6.5 Success criteria

HEDS average CHW cooling coil TD exceeds 14 °F during the time that the HEDS is in the cooling or dehumidification-reheat modes during the demonstration period.

3.2.7 Determine the level of GHG emissions that the HEDS unit contributes to

3.2.7.1 Purpose

The purpose of this PO is to determine the level of reduction of Green House Gases (GHG) that the HEDS unit contributes to.

3.2.7.2 Metric

Tons or lbs. of GHG reduction made possible by the HEDS system converted to a percent of the baseline GHG emissions.

The expected range of GHG reductions made possible by the HEDS system is between 1 and 5%.

3.2.7.3 Data

Energy saved at the AHU, chiller plant, boiler plant as calculated in other POs. Conversion factors to electrical generation sources.

3.2.7.4 Analytical methodology

Please refer to the Section 6.1, “Savings analysis methodology,” along with detailed GHG emissions factors.

3.2.7.5 Success criteria

Are GHG reductions equivalent to an approximate 3% savings comparing HEDS operation to Non-HEDS operation?

3.2.8 Reduce energy cost of dehumidification/reheat process

3.2.8.1 Purpose

To determine if the savings cost and percent cost savings are greater than 10% for the dehumidification-reheat process due to the energy savings associated with the HEDS system.

3.2.8.2 Metric

Fan, chiller plant and boiler plant energy savings associated with the energy for reheat in the dehumidification-reheat process being provided by the CRC vs. being provided by a new source of reheat energy, i.e., boilers/thermal or electric strip reheat.

The expected range of the dehumidification-reheat process cost savings is between 5 and 20% for the systems being tested.

3.2.8.3 Data

Thermal and electrical energy savings as calculated in other POs. Average cost/kWh and cost/therm for Natural Gas (NG) at each of the facilities – seasonal if available

3.2.8.4 Analytical methodology

Please refer to the Section 6.1, “Savings analysis methodology.”

3.2.8.5 Success criteria

Does the savings cost and percent cost savings exceed 10% for the dehumidification-reheat process due to the energy savings associated with the HEDS system?

3.2.9 Reduce lifecycle cost of dehumidification/ reheat process

3.2.9.1 Purpose

To determine if the HEDS system can reduce lifecycle costs of the HVAC process for facilities that need RH control by at least 5%.

3.2.9.2 Metric

Energy costs as determined in other POs. Equipment expected useful life. Lifecycle cost analysis tools.

The expected range of the lifecycle cost savings is between 0 (zero) and 10% for the systems being tested.

3.2.9.3 Data

Energy cost savings as determined in other POs. Equipment expected lifecycles as determined by DoD or ASHRAE publications in the absence of DoD lifecycles.

3.2.9.4 Analytical methodology

Please refer to the Section 6.1, “Savings analysis methodology.”

3.2.9.5 Success criteria

To determine if the HEDS system can reduce lifecycle costs of the HVAC process for facilities that need RH control by at least 5%.

4 Facility/Site Description

4.1 Facility/site location and operations

Two locations were selected for the pilot demonstration. The first is at Tinker AFB, OK in Bldg. 3, an administrative building, and the second is at Fort Bragg, NC in Bldg. A-3556, a Dining Facility (DFAC). Both Fort Bragg and Tinker AFB are in Department of Energy Climate Zone 3A, which is defined as Warm – Humid.

The selected installations are representative of those located in climates that have at least 4 months each year of dehumidification. This represents a large percentage of military installations worldwide. The two types of buildings chosen were an administrative building and a dining facility, which represent a large portion of building stock across DoD and General Services Administration (GSA). For the demonstration, the intent was to select facilities that would have stable occupancies during the test period. A barracks facility was originally targeted, but with deployments and empty barracks being a possibility, the two installations worked to select facilities that were felt to provide the greatest potential for stability over the test period. The administrative/office facility is supposed to operate with a 5 day per week, 12 hour per day HVAC schedule, and is representative of office types of facilities. The DFAC has variable occupancy and relatively long operating hours, and is representative of many other DoD facilities, even barracks to some extent. Both sites have ongoing problems with biological growth, odors and a "musty feel" associated with high internal RH and biological growth.

Tinker AFB's Bldg. 3 had a Variable Air Volume AHU that serves office spaces. The unit was supposed to run to match the occupancy schedule of the building. As with many facilities encountered in humid climates, the HVAC system in this facility was undersized. The AHUs did not have adequate cooling capacity and the chiller as unable to allow the building to shut down at night or over the weekends. When the HVAC system was shut down at night, it was unable to regain control of the facility until late in the following day. If the HVAC system was shut down over the weekend and at night, the facility would be hot and muggy inside until Tuesday or Wednesday. For this reason, many administrative type facilities that are supposed to have 5 day per week, 12 hour per day (5/12) operating schedules run their HVAC systems continuously, 24/7, during the peak summer cooling months, and also during the peak winter heating months. Bldg. A-3556, the DFAC at Fort Bragg, had a Constant Air Volume (CAV) AHU that serves a kitchen. The unit ran continuously due to the existing AHUs that serve the building being undersized, for similar reasons as discussed above for Tinker AFB.

Neither of the demonstration buildings contain or are located near critical military operations that would have impacted the demonstration. The occupants, however, are subject to discomfort if the respective buildings' HVAC does not function properly and is unable to maintain reasonable interior environmental conditions. If the HEDS were not functioning properly in Bldg. 3 of Tinker AFB, the people in that area would have decreased productivity due to poor comfort levels until the HEDS unit was restored to proper operation. If the HEDS unit were not functioning properly in Bldg. A-3556 of Fort Bragg, the people dining would be uncomfortable until the HEDS unit was restored to proper operation. Figures 10 to 13 show the respective AHUs that were replaced with new HEDS units. Figures 14 and 15 show site maps for Tinker AFB, and Figures 16 and 17 show site maps for Fort Bragg.

Figure 10. Existing AHU that was replaced on Bldg. 3 at Tinker AFB.

Figure 11. New HEDS AHU on Bldg. 3 at Tinker AFB.

Figure 12. Existing AHU that was replaced in Bldg. A-3556 at Fort Bragg.

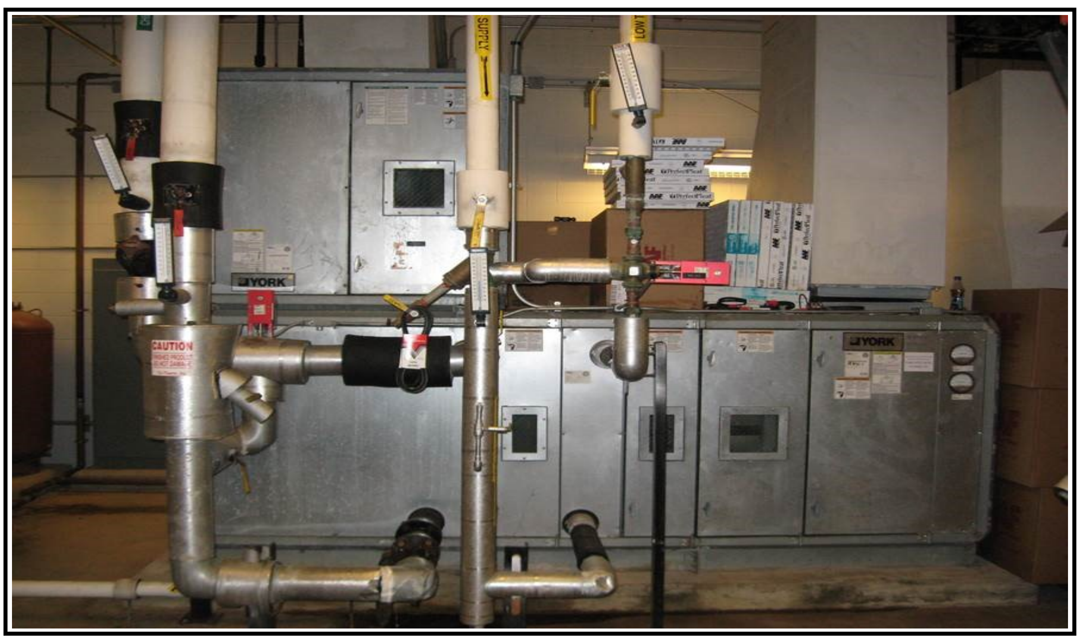


Figure 13. New HEDS AHU in Bldg. A-3556 at Fort Bragg.

Figure 14. Location of Bldg. 3, the Administrative Building for the demonstration on Tinker AFB, OK.

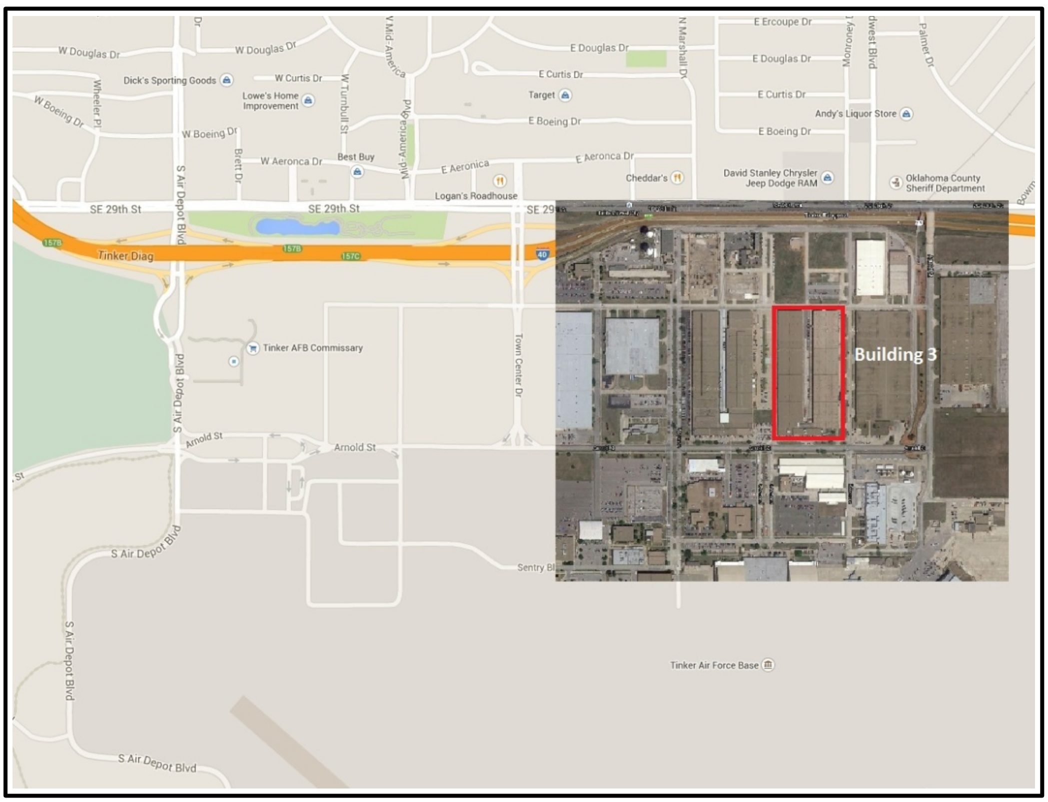


Figure 15. Close-up view of Bldg. 3.

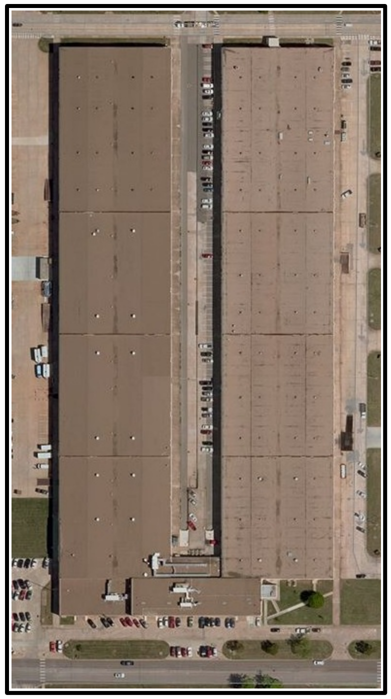


Figure 16. Location of Dining Facility (DFAC) A-3556 on Fort Bragg, NC.

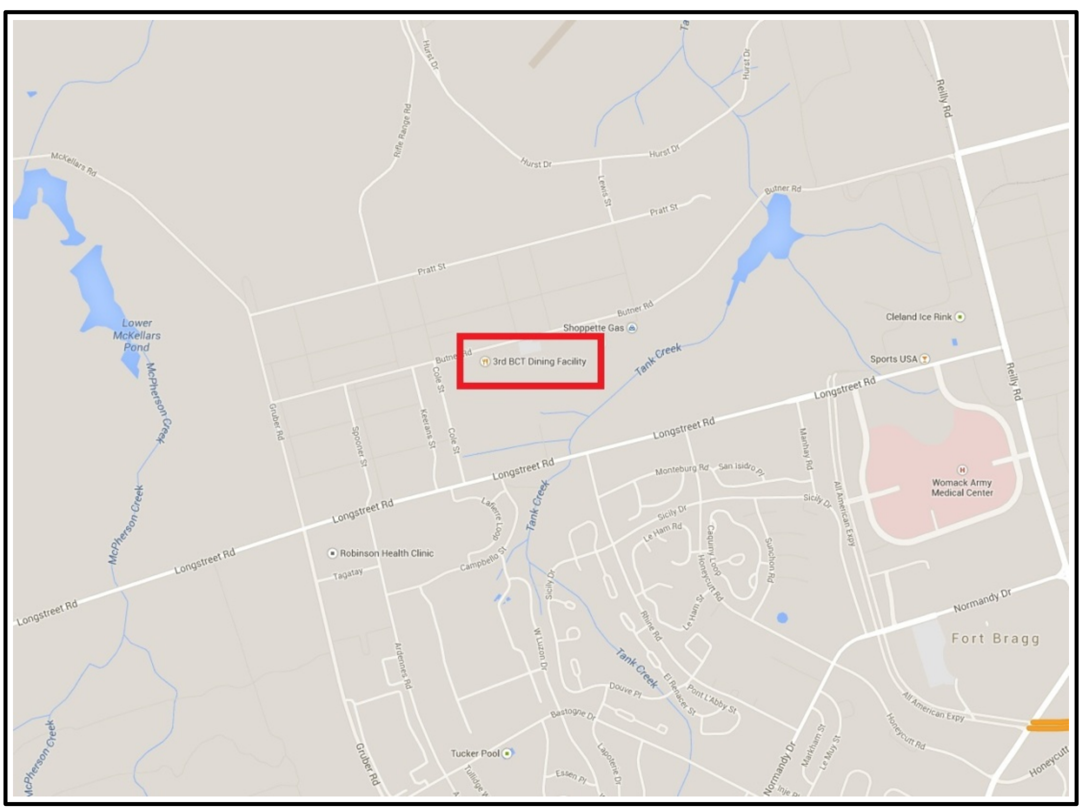
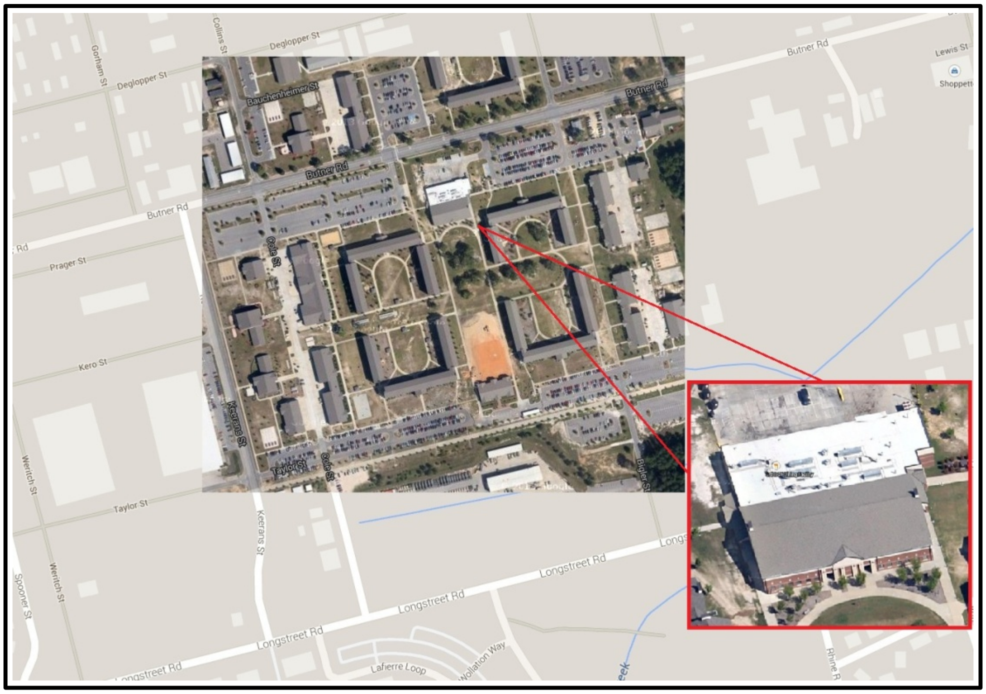


Figure 17. Close-up views of DFAC location and building.



4.2 Facility/site conditions

As described above, both project locations currently have significant operating constraints due to undersized equipment and failing chilled water systems. Before the project, even when the chiller systems were operating properly, they had to be run 24/7 during the summer to try (unsuccessfully) to maintain temperature and RH control.

The existing HVAC unit on Tinker AFB's Bldg. 3 was a Variable Air Volume (VAV) AHU that serves office spaces, which ran continuously due to lack of adequate cooling capacity and undersized AHUs. Throughout the majority of the monitoring period, the building's chiller had heavily fouled condenser tubes and nearly plugged fill in the cooling towers due to sediment in the condenser system. These issues combined to reduce the chiller output to approximately 60% of rated capacity. To keep the chiller operational, the HEDS onsite team instituted a 60% demand limit setpoint reset routine. The chiller would only allow the demand limit to remain functional for a 4-hour period. If the reset time were missed, the chiller would load up, then fail on high surge count in very short order. This resulted in pervasive, higher than design chilled water supply temperatures. In fact, the chilled water supply temperature was below 46 °F, or "in control," less than 10% of the monitoring period. Due to the severity of the issue, a temporary chiller was installed at the base to ensure chilled water temperature control in the fall of 2016. Due to the issues observed in the chilled water systems at both locations, the performance data had to be “binned” into chilled water supply temperature ranges to explore the impact of HEDS across a range of chilled water supply conditions, from “in control” design conditions (Chilled Water Supply [CHWS] temperature less than 46 °F) to out of control failing systems with chilled water supply temperatures above 62 °F.

At Fort Bragg, there are a total of four AHUs serving the facility, three existing AHUs in addition to the new HEDS AHU that replaced the original CAV AHU serving the kitchen. All are fed from the same chilled water plant. The existing AHUs serving the facility are currently undersized, requiring that they be run 24/7 in an attempt to maintain comfortable and dehumidified operations for the kitchen, serving area, and the two dining halls. Additionally, throughout the monitoring period, the chiller experienced significant capacity constraints due to failed sensors that limited the compressor staging, effectively limiting the chiller output to roughly 50% of rated capacity. As a result of this decreased capacity, chilled water supply temperatures from the plant were routinely above the design range of 42 to 46 °F. In fact, the chilled water supply temperature was below 46 °F, or “in control,” less than 50% of the monitoring period. Increased chilled water supply temperatures from the plant lead to decreased cooling and dehumidification capacities in the AHUs, and to reduced potential energy for the HEDS unit to use for RH control.

5 Test Design

This chapter provides the detailed description of the system design and testing conducted during the demonstration.

5.1 Conceptual test design

5.1.1 Hypothesis

The HEDS system will reduce the energy consumed by an HVAC system while simultaneously providing RH and space dewpoint control without the use of additional equipment.

5.1.2 Independent variable

The independent variable in this case is the installation of the HEDS AHU and the related controls, instrumentation, piping, valves, and control valves.

5.1.3 Dependent variable(s)

The dependent variables are: AHU cooling coil load in tons and ton-hours, avoided cooling coil load in tons and ton-hours, avoided reheat energy in tons and BTU/BTUH, interior space temperature, interior space RH, interior space dewpoint temperature.

5.1.4 Controlled variable(s)

The controlled variables are: the cooling coil leaving dewpoint temperature, the AHU leaving drybulb temperature after the cooling recovery coil, the size of the building, the area being served by the HEDS AHUs. They also include the mission of the building, and occupancy hours and levels (to the extent practical), and the other central plant equipment serving the buildings.

5.1.5 Test design

The test design is to serve the cooling/dehumidification loads of a facility with the HEDS unit, and to compare the HEDS unit energy consumption and performance to a hypothetical "normal" AHU that performs cooling, dehumidification, and reheat duties, which serve the identical loads.

In the test case, the loads and other variables are being measured at 5-minute intervals. The space temperature, dewpoint and RH conditions must be maintained within the UFC comfort zone, so cooling needs to be provided to dry the air out and re-heating to lower the RH of the supply air so that 100% saturated, cold wet air is not being delivered into a space. The total load being served by the AHU —associated with fresh air, solar and internal loads— will not be altered as part of this demonstration, only the method of serving the loads changes.

Since “the load is the load,” the amount of BTUs required to cool the air down and how many BTUs are required to reheat the air back up to meet comfort conditions and control space RH will be measured. From this data, the avoided BTUs of reheat energy that the CRC airside temperature increase provides will be measured, and will be equated to the decreased cooling load that has to be served by the chiller plant. The cooling load on the chiller plant is reduced due to the reduced CHW return temperature associated with the cold supply air coming off the CC and entering the CRC. The cold air coming off the CC being blown through the CRC cools the CHW return going back to the chiller plant, removing the same amount of load from the chiller plant that was added to the supply air in the reheat process by the CRC. By measuring only the chilled water flow rate entering CHW temp to the cooling coil, leaving CHW temp from the cooling coil, and leaving CHW temp from the HEDS unit, the baseline case cooling load and reheat loads can be calculated.

In this manner, the HEDS unit serves as both the baseline and the test case. Since the cooling load is known (and is not impacted by the installation of the HEDS unit), and the reheat loads needed to meet comfort and RH control conditions in the space are known, a separate AHU is not needed for baseline comparison.

Test phases included:

- HEDS Unit commissioning and startup
- Initial data collection and evaluation
- HEDS unit operation and ongoing data collection and evaluation
- Required report development.

5.2 Baseline characterization

The baseline energy use will be measured continually as a part of the test as described elsewhere in the demonstration plan. The baseline consists of the cooling energy in ton-hours that is consumed in the cooling and dehumidification process, and the reheat energy that is used to warm up the supply air to reduce the RH of the supply air and the spaces being conditioned. This is the same load as for the HEDS case. The fan energy (kW) is being monitored and trended. However, fan energy savings as a PO have been removed. Fan energy savings may occur, but presently, saving fan energy is not considered to be a critical part of the lifecycle cost reductions. This opinion may change during the testing process, and the data will be available to calculate any potential savings that occur.

One cooling season of data was collected, as were shoulder season data from one fall and one spring. More details on the precise data ranges and amount of data collected can be found further in this report.

5.2.1 Reference conditions

Although more than 300 physical and calculated variables are being monitored at each test site, the six main data points that will be obtained to determine compliance with comfort and RH control conditions, which drive all other variables, are: Space and return air dry bulb temperatures, space and return air dewpoint temperatures, and space and return air RH conditions are being measured. Maintaining these comfort conditions drives the cooling load and reheat loads will be measured and used for the basis of the savings calculations.

5.2.2 Existing baseline data

The loads that are being served on a continuous basis are the baseline data and the HEDS data. As noted elsewhere, the main difference will be the calculations for the reheat energy source, be they CRC-sourced or new energy-sourced.

5.2.3 Baseline estimation

Since all critical data points are being measured, baseline estimation is not required.

5.2.4 Data collection equipment

Section 5.5 describes the data collection and calculated points. Appendices D and E include the instrumentation diagrams for Tinker AFB and Fort Bragg. From a data collection perspective, the two systems are very similar to each other.

5.3 Design and layout of technology components

5.3.1 System Design

- Other sections of the Report (especially Section 2.1) and the diagrams in the Appendix D describe the HEDS unit.
- Appendices D and E contain the Trane HEDS unit drawings for Fort Bragg and Tinker AFB, including the physical layouts for the instrumentation within the HEDS units.
- The units will be connected into the existing ductwork and piping systems in close proximity to the current piping and ductwork points of contact (POCs). The piping and ductwork sizes are not being changed past the POCs. There are no changes to any interior ductwork.
- For the Fort Bragg DFAC unit, the motor horsepower (HP) was upgraded from a 10- to a 15-HP system, and now include a variable frequency drive (VFD) instead of using a two-speed motor for volume control when the exhaust fan 39 is on or off. The Fort Bragg HVAC system was described as problematic in terms of total capacity; consequently, a lack of CFM as a driver of the success or failure of the test was eliminated.
- For the Tinker AFB Roof Top Unit 4 (RTU-4), the motor HP was upgraded from 7.5- to a 10-HP system, and now includes a VFD for static pressure control. As with the DFAC unit at Fort Bragg, the RTU-4 HVAC system was described as problematic in terms of total capacity; consequently, a lack of CFM as a driver of the success or failure of the test was eliminated.
- The Tinker AFB HEDS unit is roof mounted and exposed to the weather, so extra precautions are being taken to make the unit reliable in the OK environment.
- The HEDS units were delivered complete with all instrumentation, valves, unit controls and trending equipment required for the test and to operate the systems. Factory testing was conducted to ensure proper operation of the components before shipment to the site.
- Each unit is being equipped with a full airside economizer damper section.
- For the two test sites, the HEDS units were built with a preheat coil to match existing construction of the AHUs that were replaced as a part of this project. The pre-heat coils are used in cold weather climates to reduce the potential for cooling coil freeze-ups. Downstream from the preheat coil are the cooling coil and the cooling recovery coil.

- The AHU replaced at Tinker AFB had a ductwork-mounted reheat coil located downstream from the cooling recovery coil that was added to the discharge ductwork after the initial project construction. It is not known why this reheat coil was installed, but it was left in place and incorporated as a part of the system. Note that this coil was never required for use during the demonstration period.
- The main portion of the HEDS that is being tested is the relationship between the cooling coil and the cooling recovery coil that is located downstream of the cooling coil.
- There are no "custom" components used in the system. The HEDS unit uses standard Trane AHU construction, fans, VFDs and coils. The instrumentation is off the shelf, but high accuracy equipment from Vaisala or Setra; the coil control valves are the Energy valve and ball valves from Belimo.

5.3.2 Components of the System

- The major system components include the Trane HEDS unit, the air-side and waterside instrumentation and controls and the Belimo Energy Valves.
- Appendices D and E contain data sheets for the major equipment, including the sensor locations on the Trane shop drawings.

5.3.3 System Depiction

- Appendix D includes schematics of the HEDS units.

5.3.4 System Integration

- The HEDS units replaced the existing under-performing AHUs. Everything between the supply and return ductwork points of connections was replaced. At Tinker AFB, the downstream ductwork mounted reheat coil was left in place.
- The failure modes are the same for the HEDS unit as for a typical AHU, and the recovery processes are also the same. The main difference is that, if there is a failure of some sort when the unit is brought back on line, the larger cooling coils and larger heating coils will allow the spaces to come back into comfort conditions more rapidly than the baseline units with smaller coils could accomplish.

5.3.5 System Controls

- Appendix E contains schematics of the HEDS units, including the controls hardware diagrams. Section 2.1 of this report describes detailed controls sequences.

5.4 Operational testing

5.4.1 Operational Testing of Cost and Performance

- The data collection process is the same for all modes of operation and all times of the day. Physical data and calculated values, as described elsewhere in this document, will be trended at 5-minute intervals, continuously throughout the process.
- Standalone trending equipment is provided for both facilities.
- At Tinker AFB, several control points are hardwired between the HEDS controller and the base BAS system. These points include: HEDS start/stop, space temperature drybulb reading, space temperature dewpoint temperature reading, fan speed command, unit discharge static pressure, and unit supply air temperature.
- At Fort Bragg, at the request of the base personnel, the HEDS controller is controlling the unit and trending data locally.
- During the commissioning process, data were downloaded at 7- to 10-day intervals, to validate the data and address any potential operational or data discrepancies before operations.
- The systems were started up and commissioned by trained Trane personnel in the field, with data validation throughout the start-up and commissioning process by ROI.

Modeling and Simulation: Given the testing setup and available data, no simulations were required to support the analysis. Modeling has been conducted in Microsoft Excel to annualize the results to typical meteorological year (TMY3) weather.

Timeline: Table 4 lists the project overall timeline for the test sites.

Table 4. Project timeline for each HEDS installation and data collection period.

Fort Bragg	2Q 2015	3Q 2015	4Q 2015	1Q 2016	2Q 2016	3Q 2016	4Q 2016	1Q 2017
HEDS Unit Ordered from Trane		X						
HEDS Installation			X					
HEDS Start-up and Commissioning				XXX	X			
HEDS Data Collection for Dehumidification Performance Data					XXX	XXX	X	
Data Analysis and Report Development							X	XXX
The number of “Xs” in the column indicate the number of months in that quarter that the work was conducted								

Tinker AFB	2Q 2015	3Q 2015	4Q 2015	1Q 2016	2Q 2016	3Q 2016	4Q 2016	1Q 2017
HEDS Unit Ordered from Trane		X						
HEDS Installation			X					
HEDS Start-up and Commissioning				XXX	X			
HEDS Data Collection for Dehumidification Performance Data					XXX	XXX	X	
Data Analysis and Report Development							X	XXX
The number of “Xs” in the column indicate the number of months in that quarter that the work was conducted								

5.5 Sampling protocol

Data Collector(s) and Data Recording: The data are being automatically collected by the HEDS unit control system at 5-minute intervals. The HEDS unit archives the data locally so it can be retrieved periodically by Trane staff for archive and analysis.

Data Description: There are approximately 300 points that are being trended or calculated as described in this document for each site. The trend interval is 5-minutes per point, and data were collected from April to November 2016.

Appendix C contains the entire list of calculated points that are being trended, numbering approximately 200 additional points. The points that are either physical points or commanded points (approximately 100 points total) in trend are:

- Measured or Commanded Values Trended
  i. Differential pressure (DP) across each coil:
     1. Preheat coil (PHC) DP
     2. CC DP
     3. cooling recovery coil (CRC) DP
     4. reheat coil (RHC) DP.
  ii. Water flow as provided by the Belimo Energy Valves:
     1. Preheat coil (PHC) flow
     2. cooling coil (CC) flow
     3. reheat coil (RHC) hot water flow.
  iii. Coil tons or BTU as provided by the Belimo Energy Valves:
     1. Preheat coil (PHC) Btu
     2. CC Tons
     3. reheat coil (RHC) Btu.
  iv. Supply fan CFM
  v. Supply fan speed command
  vi. Supply fan kW
  vii. Preheat coil control valve command percent open
  viii. Cooling coil control valve command percent open
  ix. Cooling recovery coil control valve command percent open
  x. Reheat coil control valve command percent open
  xi. Return air dewpoint temperature @ return air temperature (RAT) sensors
  xii. Space dewpoint temperature @ space sensors
  xiii. Mixed air dewpoint temperature upstream of supply fan
  xiv. Mixed air dewpoint temperature downstream of supply fan
  xv. Outside air dewpoint temperature
  xvi. PHC leaving dewpoint temperature
  xvii. CC leaving dewpoint temperature
  xviii. CRC leaving dewpoint temperature
  xix. RHC leaving dewpoint temperature
  xx. Return air drybulb temperature @ RAT sensors
  xxi. Space drybulb temperature @ space sensors
  xxii. Mixed air drybulb temperature upstream of supply fan
  xxiii. Mixed air drybulb temperature downstream of supply fan
  xxiv. Outside air drybulb temperature
  xxv. PHC leaving drybulb temperature

xxvi. CC leaving drybulb temperature
xxvii. CRC leaving drybulb temperature
xxviii. RHC leaving drybulb temperature
xxix. Return air RH% @ RAT sensors
xxx. Space RH% @ space sensors
xxxi. Mixed air RH% upstream of supply fan
xxxii. Mixed air RH% downstream of supply fan
xxxiii. Outside air RH%
xxxiv. PHC leaving RH%
xxxv. CC leaving RH%
xxxvi. CRC leaving RH%
xxxvii. RHC leaving RH %
xxxviii. OSA economizer damper position command percent open
xxxix. Mixed and Return air economizer damper position command percent open
xl. Air filter, air pressure drop
xli. HEDS discharge static pressure, actual
xlii. HEDS discharge static pressure, setpoint
xliii. Coil and HEDS water temperatures

1. PHC inlet hot water temperature, DDC
2. PHC leaving hot water temperature, DDC
3. CC inlet chilled water temperature, DDC (HEDS Inlet CHWS temp)
4. CC leaving chilled water temperature, DDC
5. CRC leaving chilled water temperature, DDC
6. HEDS leaving CHWR temp chilled water temperature, DDC (Mixed - common HEDS CHWR temp)
7. RHC inlet hot water temperature, DDC
8. RHC leaving hot water temperature, DDC
9. HEDS unit inlet hot water temperature, DDC
10. HEDS unit leaving hot water temperature, DDC
11. PHC inlet hot water temperature, Belimo Energy Valve
12. PHC leaving hot water temperature, Belimo Energy Valve
13. CC inlet chilled water temperature, Belimo Energy Valve
14. CC leaving chilled water temperature, Belimo Energy Valve
15. RHC inlet hot water temperature, Belimo Energy Valve
16. RHC leaving hot water temperature, Belimo Energy Valve

xliv. Space drybulb temperature setpoint, cooling and dehumidification modes

xlv. Space dewpoint temperature setpoint, cooling and dehumidification modes
xlvi. Space drybulb temperature setpoint, heating mode
xlvii. Return air drybulb temperature setpoint, cooling and dehumidification modes
xlviii. Return air dewpoint temperature setpoint, cooling and dehumidification modes
xlix. Return air drybulb temperature setpoint, heating mode
l. Economizer enable setpoint, drybulb logic
li. Economizer disable setpoint, drybulb logic
lii. Economizer enable setpoint, enthalpy logic
liii. Economizer disable setpoint, enthalpy logic
liv. The status of each mode of operation will be trended both in COV (Change of Value) and at 5-minute intervals. Samples of modes are:
    1. Startup mode
    2. Heating mode
    3. Cooling mode
    4. Dehumidification-reheat mode
    5. Overnight batch dehumidification mode
    6. Overnight continuous dehumidification mode
    7. Economizer mode, enthalpy on/off command
    8. Economizer mode, drybulb on/off command
    9. Economizer mode, dewpoint on/off command
    10. Economizer mode, actual on/off command
    11. AHU Start command
    12. Facility occupied mode
    13. Other distinct operating modes, to be determined (TBD)
    14. etc.
lv. The status of each alarm will be trended both in COV (Change of Value) and at 5-minute intervals. Samples of alarms are:
    1. Economizer failure
    2. PHC enabled when should be off
    3. RHC enabled when should be off
    4. Low pressure cutout
    5. Freeze stat
    6. Fire/life safety alarms
    7. Air filter, air pressure drop
    8. Other alarms TBD.

Data Storage and Backup: Trend data are stored locally in the HEDS unit controller and retrieved monthly via jobsite visits. The data are sent to ROI, Trane, and CERL, and are archived in at least three locations for redundancy.

Data Collection Diagram: Appendices D and E includes the instrumentation diagrams for the HEDS units. The points list described above also describes calculated values used in the test.

Non-standard Data: All data are collected and retrieved in a standard format.

Data Binning for Analysis: As mentioned earlier in the report, the data analysis results where binned into chilled water supply temperature ranges to examine the HEDS performance across a range of plant operating conditions as shown below. These bin descriptions should be used for reference when reviewing results figures throughout the report, as needed.

1. All Data (average performance across all chilled water supply temperature ranges).
2. Chilled Water Supply Temperature (CHWST) less than (LT) 46 °F, (which indicates a chilled water plant in control and providing adequate capacity).
3. CHWST greater than (GT) 46 and less than 50 °F (which indicates of a chilled water plant with mild chiller and capacity issues).
4. CHWST GT 50 and LT 54 °F (which indicates of a chilled water plant with moderate chiller and capacity issues).
5. CHWST GT 54 and LT 58 °F (which indicates of a chilled water plant with significant chiller and capacity issues).
6. CHWST GT 58 and LT 62 °F (which indicates of a chilled water plant with significant chiller and capacity issues).
7. CHWST GT 62 and LT 66 °F (which indicates a chilled water plant with significant chiller and capacity issues).
8. CHWST GT 66 °F (which indicates a chilled water plant that has failed and is completely out of control).

Additionally, the results were binned into supply air temperature ranges within each chilled water supply temperature bin to analyze the impact of varying levels of dehumidification as evidenced by the cooling coil supply air temperature. The bins are described as:

1. CC Supply Air Temperature (SAT) less than (LT) 50 °F (which indicates a very high dehumidification requirements associated with critical environ-

ments such as hospital operating rooms, clean rooms, and precision manufacturing or indicative of leaky envelopes in humid areas).
2. CC SAT greater than 50 and LT 52 °F (which indicates moderate dehumidification requirements associated with general hospital areas, pharmaceutical manufacturing, general manufacturing, and Navy ships or Military Sealift Command ships or indicative of lower loads in the critical environments referenced above).
3. CC SAT GT 52 and LT 56 °F (which indicates lower dehumidification requirements associated with office areas, barracks, DFACs, etc. or indicative of lower loads in the environments referenced above).
4. CC SAT GT 56 and LT 62 °F (which indicates light dehumidification loads at any of the above referenced environments).
5. CC SAT GT 62 °F (which indicates very light dehumidification loads at any of the above referenced environments).

For the presentation of the results, bins with less than 20 total hours of operation were removed as these low hour bins represent primarily dynamic conditions that are not representative of steady state operating performance. These dynamic conditions were included in the savings results reported for each overall CHWS temperature bin.

5.6 Sampling results

The following sections detail the results for each test site.

5.6.1 Fort Bragg

For a large portion of the analysis period, the chiller plant at Fort Bragg was experiencing significant performance issues associated with a failed pressure sensor on the chiller that limited capacity to around 50% of rated output. As a result, there are significant periods of time where the plant was unable to meet the leaving chilled water temperature setpoint due to inadequate capacity, resulting in chilled water supply temperatures to the AHU ranging from 42 °F to over 65 °F (compared with a design leaving temperature of 45 °F). Table 5 lists the resulting run hours in each temperature bin for the analysis period. Note that the rows highlighted in gray represent transient conditions for which there is insufficient data to draw significant conclusions, and therefore have been removed from presentation of savings in subsequent analyses.

Table 5. Summary of hours at each temperature bin analyzed for Fort Bragg.

	Hours at Condition	% of Hours at Condition
All Included (Filtered) Data	**4,454.3**	100.0%
CHWST LT 46F (All Data)	**2,143.3**	**48.1%**
CC SAT LT 50F	1,251.1	28.1%
CC SAT GT 50F and LT 52F	160.2	3.6%
CC SAT GT 52F and LT 56F	717.6	16.1%
CC SAT GT 56F and LT 62F	8.8	0.2%
CC SAT GT 62F	5.6	0.1%
CHWST GT 46F and LT 50F	**1,345.8**	**30.2%**
CC SAT LT 50F	185.4	4.2%
CC SAT GT 50F and LT 52F	592.3	13.3%
CC SAT GT 52F and LT 56F	567.2	12.7%
CC SAT GT 56F and LT 62F	0.5	0.0%
CC SAT GT 62F	0.4	0.0%
CHWST GT 50F and LT 54F	**524.3**	**11.8%**
CC SAT GT 50F and LT 52F	1.0	0.0%
CC SAT GT 52F and LT 56F	377.3	8.5%
CC SAT GT 56F and LT 62F	145.8	3.3%
CC SAT GT 62F	0.3	0.0%
CHWST GT 54F and LT 58F	**169.0**	**3.8%**
CC SAT GT 52F and LT 56F	1.0	0.0%
CC SAT GT 56F and LT 62F	167.8	3.8%
CC SAT GT 62F	0.2	0.0%
CHWST GT 58F and LT 62F	**90.8**	**2.0%**
CC SAT GT 56F and LT 62F	49.3	1.1%
CC SAT GT 62F	41.5	0.9%
CHWST GT 62F and LT 66F	**105.0**	**2.4%**
CC SAT GT 62F	104.9	2.4%
CHWST GT 66F	**76.1**	**1.7%**
CC SAT GT 62F	76.1	1.7%

The results of the analysis show that the HEDS AHU significantly reduces the cooling loads and reheat energy associated with dehumidification across all operating conditions, even considering the failing chiller conditions. Additionally, due to the large cooling coil design, HEDS was able to provide dehumidification and reheat even when the chilled water supply temperature from the failing plants approached 60 °F, as indicated in the table below by the reduced CC dewpoint temperatures compared with outdoor and return air dewpoints. Visual HVAC system inspections of all of the AHUs during these warm CHWS water conditions showed that the only AHU that was producing meaningful levels of condensate into the drains was the HEDS unit. The other units were producing zero to insignificant amounts of condensate, while the HEDS unit was pouring condensate down the drain.

The data in Table 6 confirm that the HEDS system demonstrated a net 21.5% cooling load reduction across all operating conditions, including times when the chiller was in a failure or capacity constraint mode of some sort.

Table 6. Fort Bragg HEDS savings summary with differential temperatures across all temperature bins. Note that gray rows represent transient conditions and may not be representative of steady state results.

	Total Cooling Coil Ton-hrs	Net AHU Total Ton-hrs DDC	Total Ton-hrs Savings from HEDS	Total HEDS Savings % (saved ton-hrs / CC ton-hrs)	Cooling Coil CHW Delta T	Net AHU CHW Delta T
All Included (Filtered) Data	**150,081**	**117,874**	**32,207**	**21.5%**	**14.38**	**10.61**
CHWST LT 46F (All Data)	**68,276**	**51,410**	**16,866**	**24.7%**	**17.08**	**11.80**
CC SAT LT 50F	47,189	37,929	9,260	19.6%	13.47	10.63
CC SAT GT 50F and LT 52F	4,899	3,354	1,545	31.5%	20.99	13.22
CC SAT GT 52F and LT 56F	15,725	9,843	5,882	37.4%	22.10	13.29
CC SAT GT 56F and LT 62F	293	178	116	39.5%	37.01	21.94
CC SAT GT 62F	170	106	63	37.4%	38.40	26.04
CHWST GT 46F and LT 50F	**51,626**	**41,645**	**9,981**	**19.3%**	**13.15**	**10.41**
CC SAT LT 50F	6,676	5,413	1,263	18.9%	11.93	9.57
CC SAT GT 50F and LT 52F	23,382	18,988	4,394	18.8%	12.93	10.45
CC SAT GT 52F and LT 56F	21,545	17,230	4,315	20.0%	13.77	10.64
CC SAT GT 56F and LT 62F	14	10	4	28.5%	18.72	13.23
CC SAT GT 62F	9	4	5	58.4%	20.55	13.10
CHWST GT 50F and LT 54F	**19,854**	**16,266**	**3,587**	**18.1%**	12.18	9.95
CC SAT GT 50F and LT 52F	18	15	4	19.7%	7.73	5.87
CC SAT GT 52F and LT 56F	14,130	11,590	2,540	18.0%	12.07	9.87
CC SAT GT 56F and LT 62F	5,693	4,656	1,037	18.2%	12.49	10.20
CC SAT GT 62F	12	5	6	53.8%	21.63	12.15
CHWST GT 54F and LT 58F	**5,678**	**4,658**	**1,020**	**18.0%**	10.81	8.84
CC SAT GT 52F and LT 56F	16	13	3	19.8%	5.01	3.98
CC SAT GT 56F and LT 62F	5,656	4,640	1,015	18.0%	10.84	8.87
CC SAT GT 62F	6	5	1	22.0%	12.82	10.64
CHWST GT 58F and LT 62F	**1,610**	**1,356**	**254**	**15.8%**	6.09	5.03
CC SAT GT 56F and LT 62F	847	736	111	13.1%	5.62	4.83
CC SAT GT 62F	762	619	143	18.7%	6.65	5.27
CHWST GT 62F and LT 66F	**1,798**	**1,495**	**303**	**16.9%**	5.53	4.59
CC SAT GT 62F	1,797	1,494	303	16.9%	5.53	4.59
CHWST GT 66F	**1,239**	**1,044**	**196**	**15.8%**	5.22	4.37
CC SAT GT 62F	1,239	1,044	196	15.8%	5.22	4.37

When considering only those times when the chiller plant was meeting setpoint (chilled water supply temperatures were less than 46 °F), the net cooling load reduction is 24.7%.

The peak observed savings for the operating bins shown below is 37.4% when bins with less than 20 hours of operation were removed. Fortunately, this highest savings range corresponded to the most typical conditions that will be found in DoD and GSA facilities.

Extrapolating these results to VA Hospital type loads, with cooling coil supply air temperatures of less than 50 °F, the savings of 19.6% when multiplied by the 24 hour per day operating periods and significant cooling loads equate to a significant financial benefit, in addition to the life-saving aspects of the technology when it comes to RH control and the reduction of Hospital Acquired Infections (HAIs).

The chilled water differential temperatures for each performance bin are also shown. Under the heaviest dehumidification loads with cooling coil supply air temperatures less than 50 °F, the cooling coil chilled water system temperature differential (TD) is still above 13 °F, with a net AHU waterside TD of nearly 11 °F. Under lighter loads with cooling coil supply air temperature between 52 to 56 °F, the cooling coil chilled water system TD increases to over 22 °F, with a net AHU waterside TD of over 13 °F.

Figure 18 shows the results of a psychometric analysis that highlights the impact of the cooling recovery coil on the system performance. As shown, the subcooled air leaving the cooling coil at 52.5 °F is heated to 61.5 °F using recovered energy from the chilled water in the cooling recovery coil, completely eliminating the need for additional reheat energy.

Figures 19 and 20 highlight the dynamic impact of the cooling recovery coil on the net AHU cooling load for a peak cooling week and peak cooling load day in August followed by part load days in October. The impact on total load, AHU cooling coil and net differential CHW temperatures, and total energy savings percentage are shown.

For the peak load week in August shown, the cooling coil supply air temperature is less than 50 °F due to the high dehumidification loads required. At this condition, the average savings from HEDS is approximately 18%. The peak cooling load day, August 5, 2016, is also shown, highlighting the savings at the time of coincident peak cooling load of 18.3%.

Figures 21 and 22 highlight the impact of increased cooling coil supply air temperatures from part load conditions on overall HEDS system performance. The observed CHW temperature differentials (TD) are higher than the peak load data presented for August (where the cooling coil SAT was less than 50 °F). TD ranges from 15 to 22 °F for the cooling coil and 10 to 15 °F for the entire AHU net of the cooling recovery coil (compared with 12 to 17 °F for the cooling coil and 10 to 14 °F for the entire AHU net of the cooling recovery coil in the August peak data). This increased temperature differential results in an average cooling load reduction of over 27% for the data presented.

As expected, the results shown in Figure 23 indicate a significant dependency of the savings on the cooling coil supply air temperature when the chiller plant is meeting the leaving setpoint (less than 46 °F). As the cooling coil supply air temperature increases due to lower dehumidification loads, the leaving water temperature of the coil increases, enabling more heat recovery in the cooling recovery coil.

Figure 18. Psychrometric chart highlighting the impact of the cooling recovery coil on HEDS system performance at Fort Bragg.

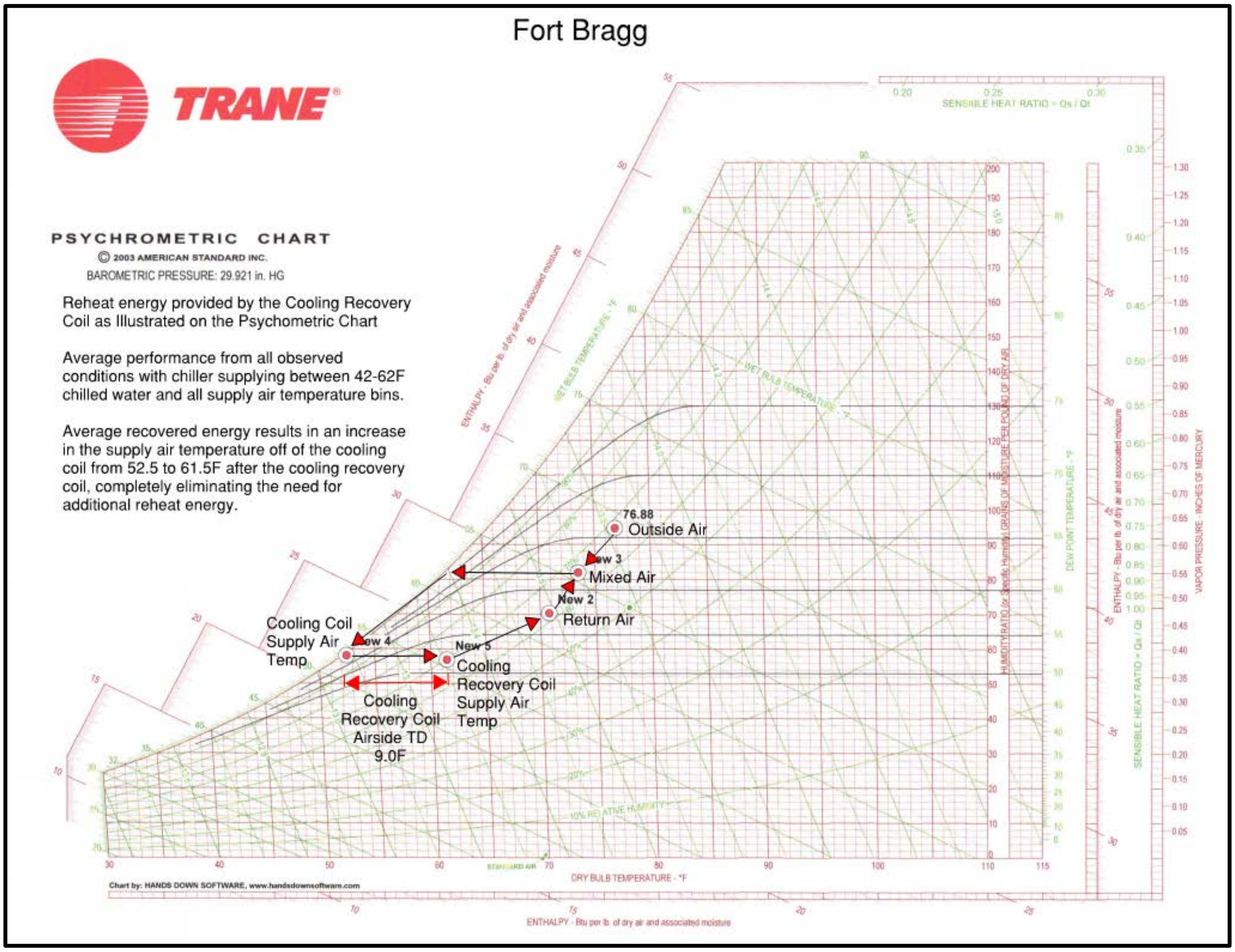


Figure 19. Dynamic HEDS savings and impact analysis for the peak cooling load week of August 1, 2016.

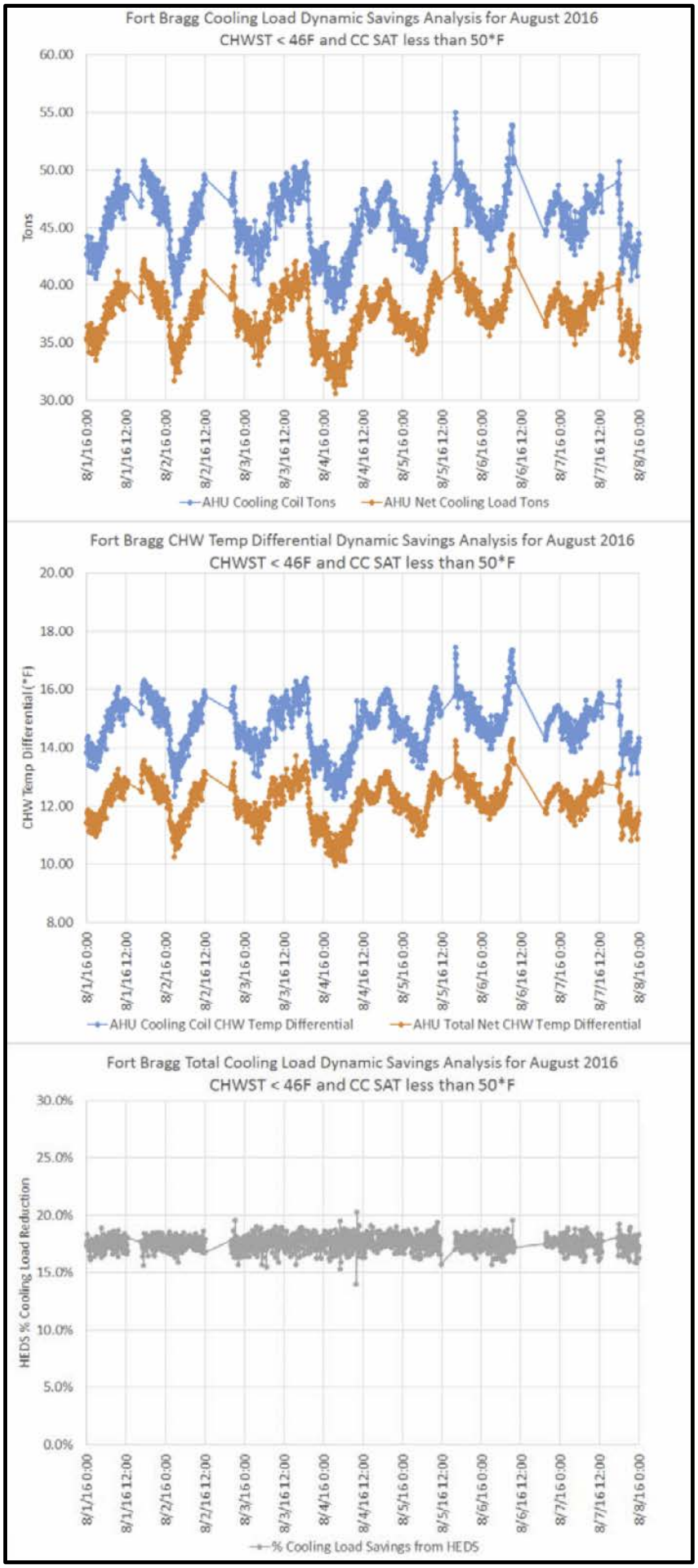


Figure 20. Dynamic HEDS savings and impact analysis for the peak cooling load day of August 5, 2016.

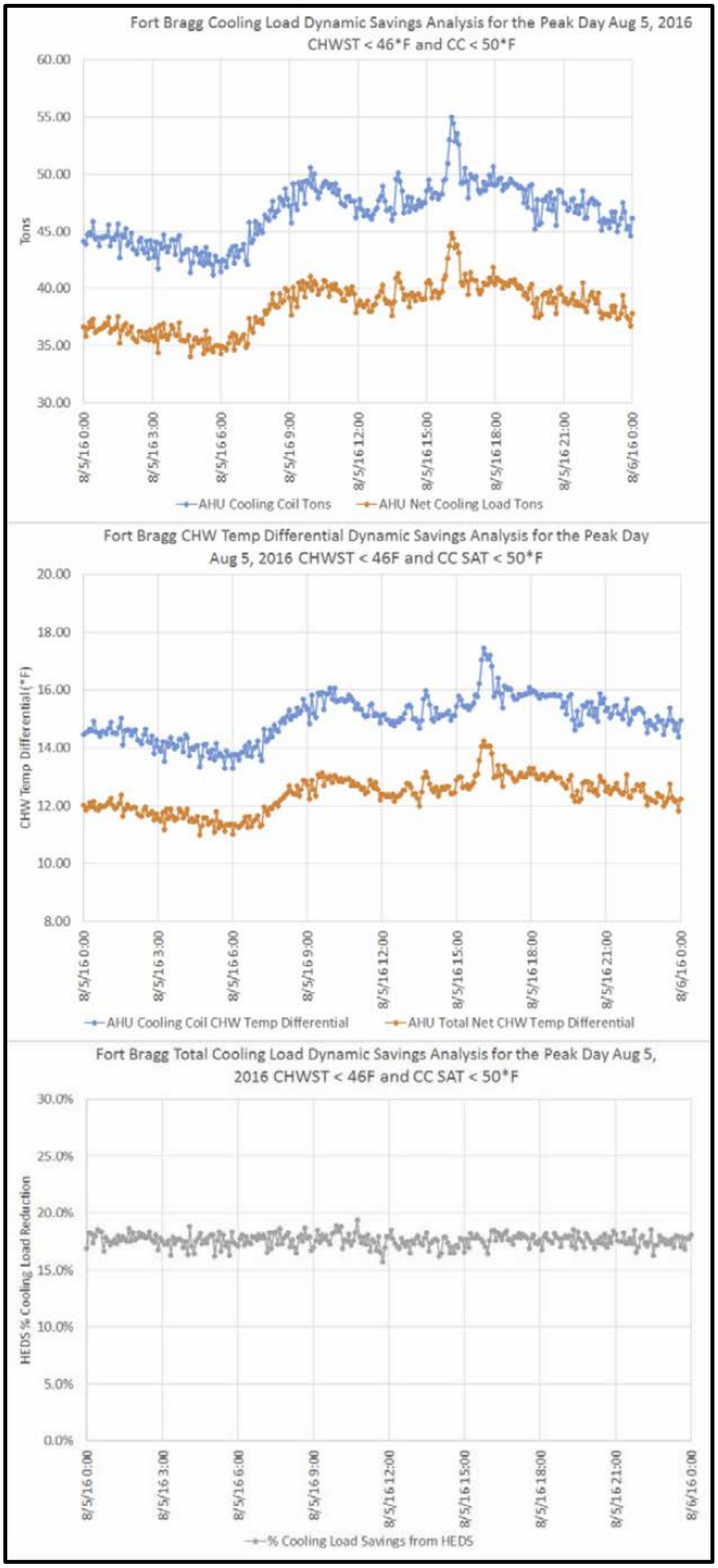


Figure 21. HEDS savings analysis for fall part load days where the CHWST is less than 46 °F and the CC SAT is between 52 to 56°F.

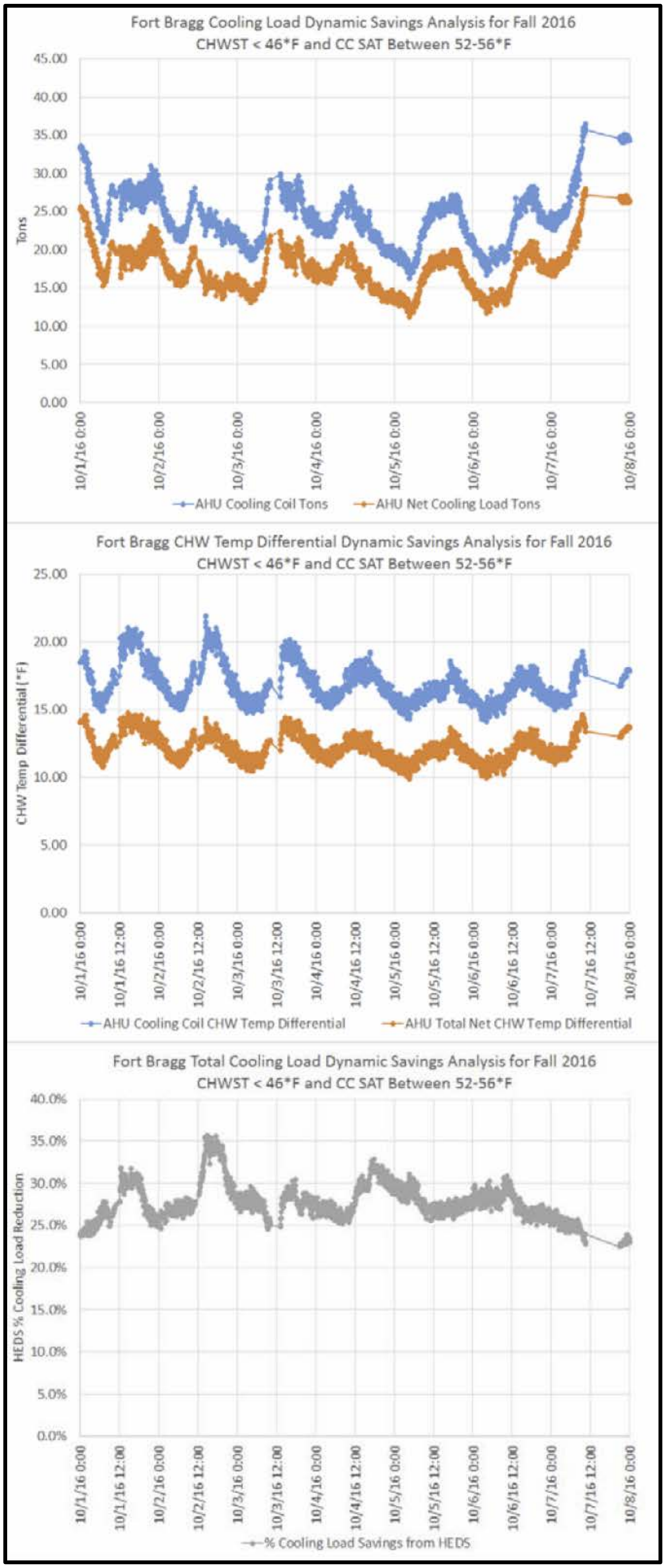


Figure 22. HEDS savings analysis for fall part load peak day October 2, 2016 where the CHST is less than 46 °F and the CC SAT is between 52 to 56 °F.

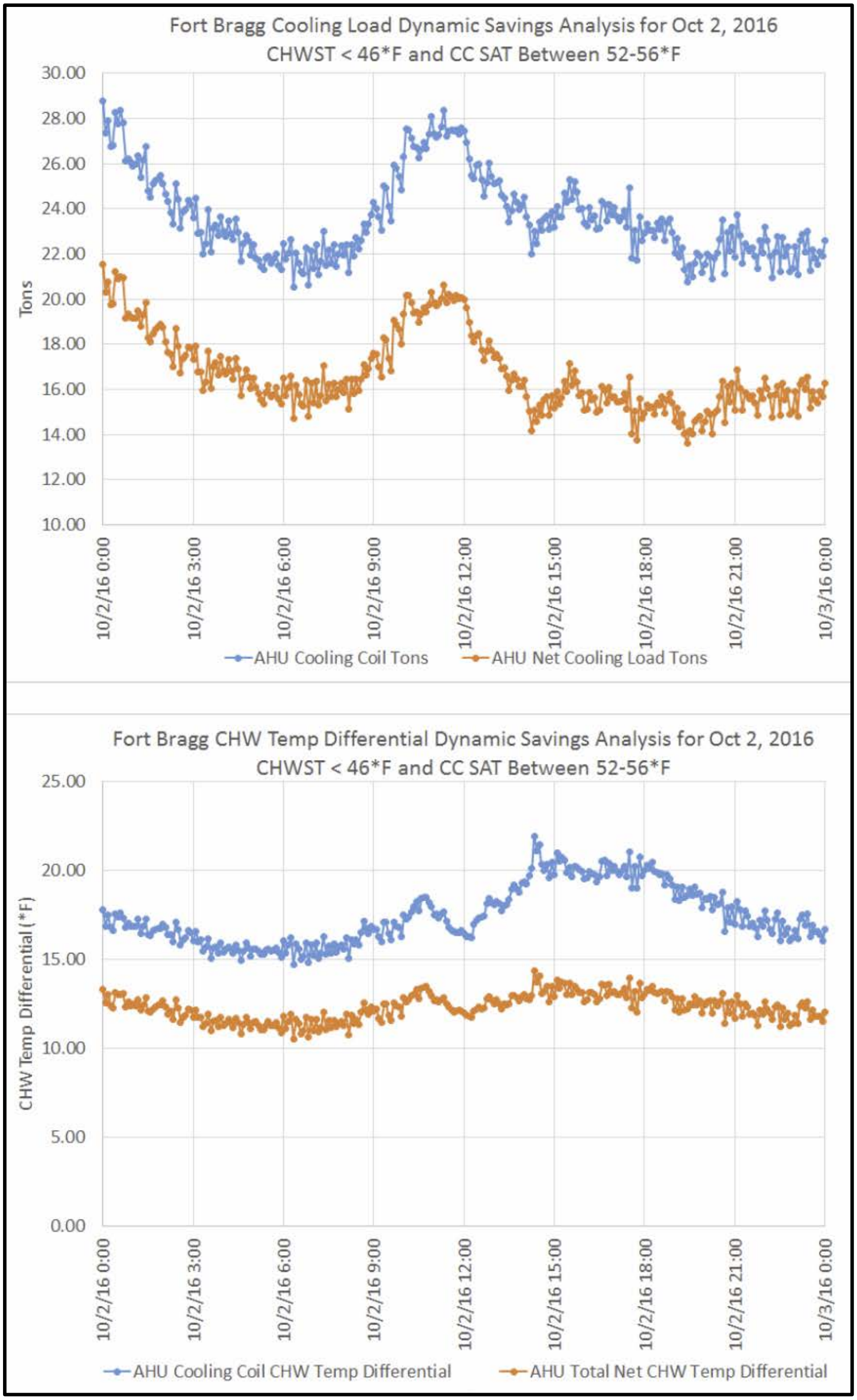
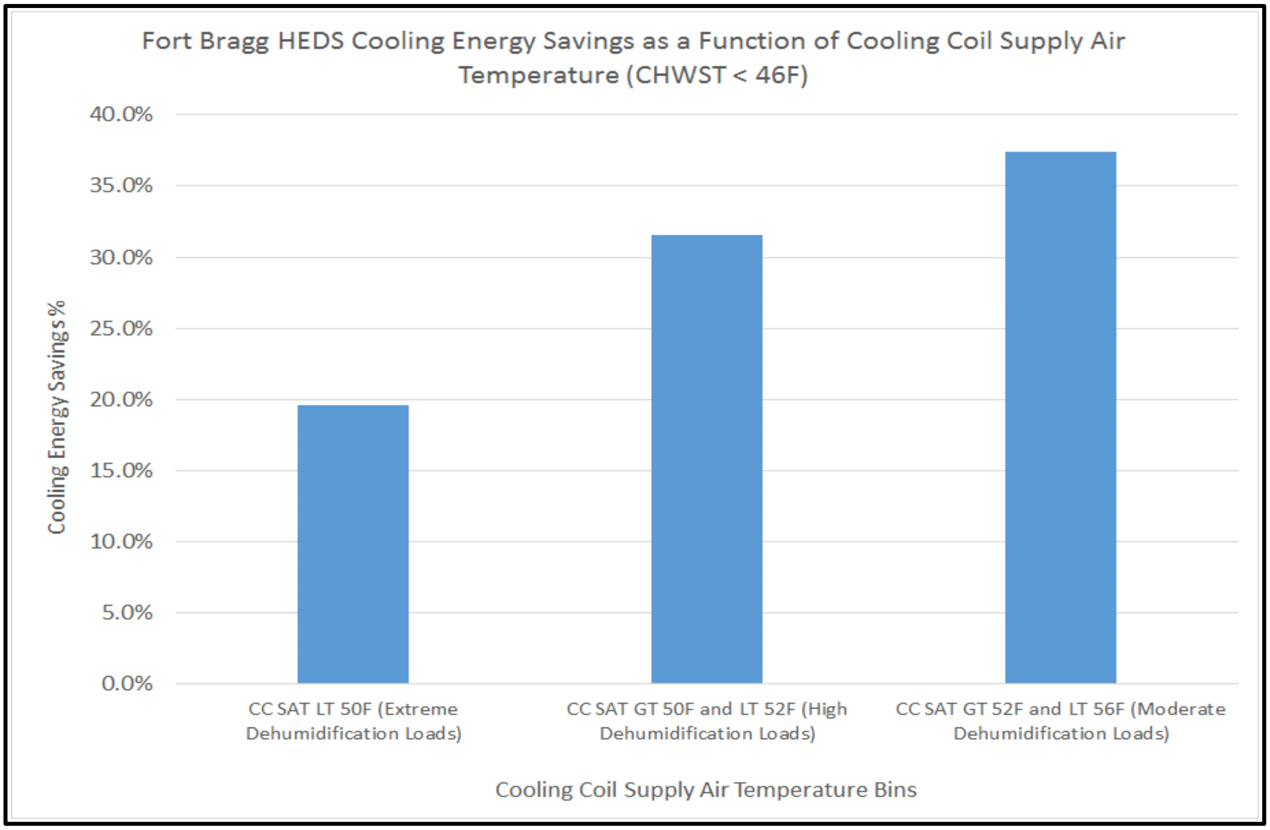


Figure 23. HEDS cooling load reduction as a function of CC supply air temp (SAT) when chilled water supply temp is in control (less than 46 °F). Note that the higher SAT ranges have been removed due to the limited hours of operation within the bins.



HEDS performance was tested across a range of conditions to determine a savings envelope. Very low supply air temperatures (and thus dewpoint temperatures) of less than 50 °F off the cooling coil indicate of extreme dehumidification conditions typical of hospital operating rooms and clean room conditions (or extremely leaky envelopes in humid environments), while supply air temperatures off the cooling coil in the low 50s (°F) are more typical of actual field conditions.

Table 7 lists the temperature conditions across the AHU for all temperature bins. When the CHWST is in control (less than 46 °F), the results show that, even at an average cooling coil leaving dewpoint of 52 °F, which provides significant dehumidification to the space, HEDS cooling load savings increase to over 37%. In the most extreme dehumidification conditions, delivering 48 °F dewpoint air off the cooling coil, HEDS still reduced the cooling load by nearly 20%. This points to the importance of cooling coil supply air temperature resets based on space dewpoint conditions to achieve maximum energy savings.

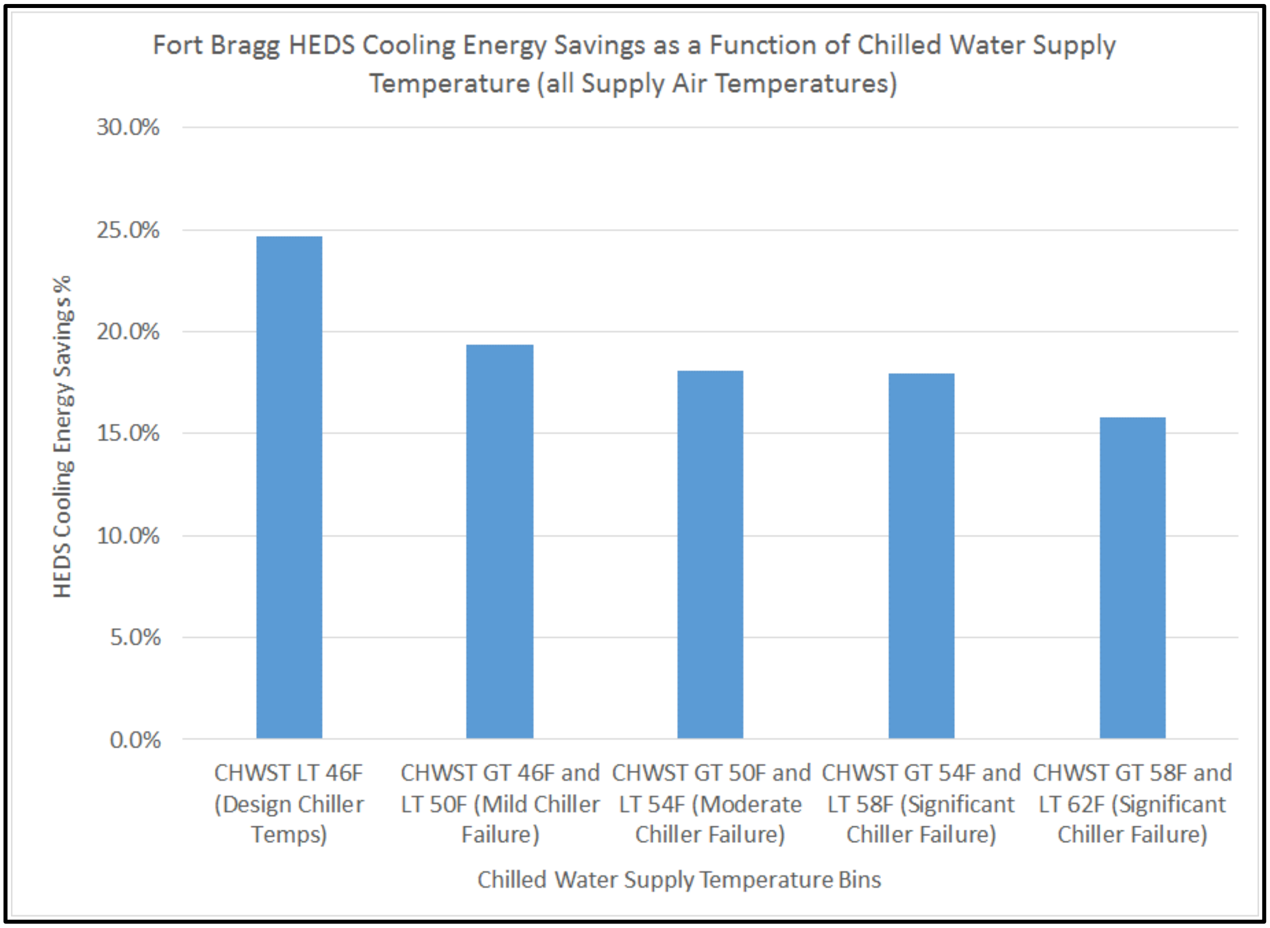
Table 7. Fort Bragg AHU temperatures across the range of operating conditions.

	Avg Outside Air Dewpoint Temp	Avg Space Dewpoint Temp	Avg Return Air Dewpoint Temp	Avg Mixed Air Dewpoint Temp	Avg CC Supply Dewpoint Temp	Avg CC Supply Drybulb Temp	AHU CRC Supply Drybulb Temp	AHU CRC Supply Dewpoint Temp
All Included (Filtered) Data	**65.56**	**58.44**	**57.27**	**61.49**	**52.13**	**52.49**	**61.46**	**51.56**
CHWST LT 46F (All Data)	**61.51**	**54.87**	**54.19**	**57.88**	**49.67**	**50.18**	**59.97**	**49.06**
CC SAT LT 50F	65.82	56.27	55.27	60.14	48.16	48.35	57.81	47.72
CC SAT GT 50F and LT 52F	58.39	53.39	53.35	56.35	50.66	50.99	62.83	49.77
CC SAT GT 52F and LT 56F	55.07	52.98	52.67	54.57	52.21	52.99	62.89	51.33
CC SAT GT 56F and LT 62F	43.17	44.92	45.43	45.36	44.82	58.72	69.98	44.61
CC SAT GT 62F	41.06	42.27	42.69	42.62	40.80	64.74	72.07	42.26
CHWST GT 46F and LT 50F	**68.75**	**60.07**	**58.27**	**63.32**	**51.45**	**51.60**	**60.76**	**50.94**
CC SAT LT 50F	66.33	58.13	56.11	61.31	49.20	49.36	58.32	48.96
CC SAT GT 50F and LT 52F	69.20	59.92	58.19	63.35	50.99	51.05	60.20	50.53
CC SAT GT 52F and LT 56F	69.07	60.87	59.07	63.96	52.68	52.89	62.14	52.02
CC SAT GT 56F and LT 62F	62.78	57.33	56.24	60.57	51.19	57.17	63.98	52.05
CC SAT GT 62F	58.67	55.24	54.47	52.27	52.62	64.08	64.79	49.68
CHWST GT 50F and LT 54F	71.29	63.76	61.46	66.65	55.21	55.23	63.58	54.69
CC SAT GT 50F and LT 52F	56.51	54.66	54.41	55.97	53.33	53.47	58.89	52.28
CC SAT GT 52F and LT 56F	70.46	63.15	60.79	65.82	54.55	54.58	62.91	54.07
CC SAT GT 56F and LT 62F	73.56	65.41	63.26	68.89	56.93	56.92	65.33	56.31
CC SAT GT 62F	48.73	49.42	50.61	52.09	51.22	53.37	65.94	46.81
CHWST GT 54F and LT 58F	71.21	64.79	63.39	68.17	58.87	58.93	66.20	58.37
CC SAT GT 52F and LT 56F	50.28	58.77	53.05	52.77	55.62	56.33	60.40	53.12
CC SAT GT 56F and LT 62F	71.33	64.83	63.45	68.26	58.89	58.95	66.24	58.40
CC SAT GT 62F	69.93	57.87	60.15	67.85	54.57	54.72	63.43	58.47
CHWST GT 58F and LT 62F	62.85	60.47	61.12	62.66	60.44	61.78	65.67	60.11
CC SAT GT 56F and LT 62F	61.52	59.18	59.79	61.51	59.21	60.82	64.05	59.01
CC SAT GT 62F	64.43	62.00	62.71	64.02	61.91	62.92	67.60	61.42
CHWST GT 62F and LT 66F	68.15	64.72	64.90	67.13	64.87	66.08	69.97	64.39
CC SAT GT 62F	68.15	64.72	64.90	67.15	64.87	66.08	69.97	64.40
CHWST GT 66F	71.56	68.37	68.72	71.08	68.92	69.56	73.04	68.58
CC SAT GT 62F	71.56	68.37	68.72	71.08	68.92	69.56	73.04	68.58

The elimination of the need for new reheat energy can be seen in the AHU CRC drybulb temperatures listed in Table 7. When the cooling coil leaving dewpoint temp averages 48 °F, the HEDS units is able to deliver 58 °F dry-bulb air to the space, reducing mold potential from surface condensation and ensuring a comfortable indoor environment. As loads decrease, HEDS is able to provide additional reheat to the supply air stream to both eliminate condensation potential and track the reduced cooling loads to prevent overcooling in the space. At an average cooling coil dewpoint temperature of 52 °F and drybulb temperature of 53 °F, HEDS is able to provide 63 °F drybulb air to the space.

Figure 24 shows that, even under the failing chiller conditions, the HEDS system is able to reduce cooling loads significantly, reserving capacity for other equipment in the system. Even with the chiller plant delivering chilled water above 60 °F, HEDS is able to reduce the cooling load required by the unit by nearly 16%, delivering added capacity to the other AHUs when the chiller is not performing.

Figure 24. Fort Bragg HEDS cooling load reduction as a function of chilled water supply temperature across all supply air temperature ranges.



Peak cooling load is also reduced across the operating conditions. Peak loads were only analyzed for the steady state conditions where the chilled water supply temperature from the chiller plant was in control. The results (listed in Table 8) show an 18% peak demand reduction across all operating conditions. The peak is reduced by nearly 24% when the cooling coil supply air temperature is above 52 °F.

Table 8. Fort Bragg HEDS peak load reduction results when CHWST is in control (less than 46 °F).

	Peak Cooling Coil Load (tons)	Peak Net AHU Load (tons)	Peak Load Savings (tons)	Peak Load Savings %
All Included (Filtered) Data	**55.52**	**45.37**	**10.14**	**18.3%**
CHWST LT 46F (All Data)	**55.52**	**45.37**	**10.14**	**18.3%**
CC SAT LT 50F	55.52	45.37	10.14	18.3%
CC SAT GT 50F and LT 52F	53.52	44.21	9.31	17.4%
CC SAT GT 52F and LT 56F	36.25	27.64	8.61	23.7%
CC SAT GT 56F and LT 62F	no steady state operations			

5.6.2 Tinker AFB

For a large portion of the analysis period, the chiller plant at Tinker AFB was experiencing significant performance issues associated with plugged condenser tubes in the chiller and plugged fill in the cooling tower, effectively limiting chiller output to around 60% of rated capacity when the chiller was not failed due to high surge count, which was a significant portion of the time. As a result, there were long periods of time where the plant was unable to meet the leaving chilled water temperature setpoint due to inadequate capacity or the chiller being failed off due to high surge count. This resulted in chilled water supply temperatures to the AHU ranging from 42 °F to over 80 °F. Table 9 lists the resulting run hours in each temperature bin for the analysis period. Note that the rows highlighted in gray represent transient conditions for which there was insufficient data to draw significant conclusions; these were therefore removed from discussion of savings in subsequent analyses.

Table 9. Summary of hours at each temperature bin analyzed for Tinker AFB.

	Hours at Condition	% of Hours at Condition
All Included (Filtered) Data	**987.4**	100.0%
CHWST LT 46F (All Data)	**326.5**	**33.1%**
CC SAT LT 50F	5.9	0.6%
CC SAT GT 50F and LT 52F	54.8	5.5%
CC SAT GT 52F and LT 56F	262.3	26.6%
CC SAT GT 56F and LT 62F	1.9	0.2%
CC SAT GT 62F	1.6	0.2%
CHWST GT 46F and LT 50F	**280.0**	**28.4%**
CC SAT LT 50F	4.6	0.5%
CC SAT GT 50F and LT 52F	79.1	8.0%
CC SAT GT 52F and LT 56F	187.0	18.9%
CC SAT GT 56F and LT 62F	8.2	0.8%
CC SAT GT 62F	1.2	0.1%
CHWST GT 50F and LT 54F	**227.0**	**23.0%**
CC SAT GT 50F and LT 52F	3.5	0.4%
CC SAT GT 52F and LT 56F	153.7	15.6%
CC SAT GT 56F and LT 62F	67.6	6.8%
CC SAT GT 62F	1.9	0.2%
CHWST GT 54F and LT 58F	**95.3**	**9.7%**
CC SAT GT 52F and LT 56F	2.9	0.3%
CC SAT GT 56F and LT 62F	85.8	8.7%
CC SAT GT 62F	6.3	0.6%
CHWST GT 58F and LT 62F	**42.2**	**4.3%**
CC SAT GT 56F and LT 62F	11.0	1.1%
CC SAT GT 62F	31.2	3.2%
CHWST GT 62F and LT 66F	**11.6**	**1.2%**
CC SAT GT 62F	11.2	1.1%
CHWST GT 66F	**4.8**	**0.5%**
CC SAT GT 62F	4.8	0.5%

The results of the analysis show that the HEDS AHU significantly reduces the cooling loads and reheat energy associated with dehumidification across all operating conditions, even considering the failing chiller conditions. Additionally, HEDS was able to provide dehumidification and reheat even when the chilled water supply temperature from the failing plants approached 60 °F.

The data in Table 10 confirm that the HEDS system demonstrated a net 29.0% cooling load reduction across all operating conditions. When considering only those times when the chiller plant was meeting setpoint (when chilled water supply temperatures were less than 46 °F), the net cooling load reduction is 27.6%. The peak observed savings for the operating bins shown below is 33.7% when bins with less than 20 hours of total operation are removed.

The differential temperatures for each performance bin are also shown. Under the heaviest dehumidification loads with cooling coil supply air temperatures between 50 and 52 °F and the chilled water supply temperature less than 46 °F, the cooling coil temperature differential (TD) is more than 24 °F, with a net AHU TD of nearly 18 °F. Under lighter loads with cooling coil supply air temperature between 52 to 56 °F and the chilled water supply temperature less than 46 °F, the cooling coil TD remains over 24 °F, with a net AHU TD of nearly 17 °F.

Figure 25 shows the results of a psychometric analysis that highlights the impact of the cooling recovery coil on the system performance. As shown, the subcooled air leaving the cooling coil at 54.8 °F is heated to 66.4 °F using recovered energy from the chilled water in the cooling recovery coil, completely eliminating the need for additional reheat energy.

The impact of the variable air volume system can be seen in the data compared to the Fort Bragg constant air volume results. The cooling coil and AHU net TD is higher across all ranges due to the lower volume of air across the coils. This results in relatively flat cooling load savings (~30%) across all ranges of chilled water supply and supply air temperatures.

Figure 26 highlights the impact of the cooling recovery coil on the net AHU cooling load for a peak cooling day in June followed by a part load week in the Fall. The impact on total load, AHU cooling coil and net differential CHW temperatures, and total energy savings percentage are shown.

During the peak load day in June, the average savings from HEDS is approximately 30%, with a savings at the time of coincident peak cooling load of 28.9%.

Figures 27 and 28 highlight the impact of the variable air volume control with the HEDS system. The resulting savings in the part load conditions stay relatively consistent with the peak load days, with average cooling load savings of just over 30%.

Table 10. Tinker AFB HEDS savings summary with differential temperatures across all temperature bins. Note that gray rows represent transient conditions and may not be representative of steady state results.

	Total Cooling Coil Ton-hrs	Net AHU Total Ton-hrs DDC	Total Ton-hrs Savings from HEDS	Total HEDS Savings % (saved ton-hrs / CC ton-hrs)	Cooling Coil CHW Delta T	Net AHU CHW Delta T
All Included (Filtered) Data	**11,658**	**8,282**	**3,376**	**29.0%**	**19.91**	**13.84**
CHWST LT 46F (All Data)	**2,837**	**2,053**	**783**	**27.6%**	**24.37**	**17.07**
CC SAT LT 50F	80	64	16	20.5%	13.31	9.55
CC SAT GT 50F and LT 52F	689	505	184	26.7%	24.69	18.07
CC SAT GT 52F and LT 56F	1,978	1,408	570	28.8%	24.51	16.96
CC SAT GT 56F and LT 62F	34	27	7	20.9%	25.34	18.58
CC SAT GT 62F	55	49	6	10.9%	32.75	28.25
CHWST GT 46F and LT 50F	**3,908**	**2,725**	**1,183**	**30.3%**	**19.30**	**13.24**
CC SAT LT 50F	65	51	14	21.1%	11.31	8.24
CC SAT GT 50F and LT 52F	1,497	993	504	33.7%	19.76	13.08
CC SAT GT 52F and LT 56F	2,190	1,557	633	28.9%	19.20	13.34
CC SAT GT 56F and LT 62F	108	78	30	27.7%	20.64	13.57
CC SAT GT 62F	49	46	3	5.9%	26.30	24.40
CHWST GT 50F and LT 54F	**2,853**	**2,078**	**775**	**27.2%**	19.01	13.39
CC SAT GT 50F and LT 52F	32	23	9	28.5%	14.43	9.33
CC SAT GT 52F and LT 56F	1,955	1,430	526	26.9%	19.78	14.11
CC SAT GT 56F and LT 62F	807	572	235	29.1%	17.54	11.98
CC SAT GT 62F	55	50	4	7.9%	16.52	10.79
CHWST GT 54F and LT 58F	**1,361**	**926**	**435**	**31.9%**	17.02	11.42
CC SAT GT 52F and LT 56F	38	29	9	23.3%	13.14	9.52
CC SAT GT 56F and LT 62F	1,234	830	404	32.8%	17.14	11.44
CC SAT GT 62F	86	66	20	23.5%	16.82	11.89
CHWST GT 58F and LT 62F	**509**	**355**	**154**	**30.3%**	14.96	9.76
CC SAT GT 56F and LT 62F	113	82	31	27.7%	10.25	6.87
CC SAT GT 62F	396	273	123	31.0%	16.27	10.56
CHWST GT 62F and LT 66F	**133**	**97**	**36**	**27.0%**	12.96	8.05
CC SAT GT 62F	130	95	35	26.9%	12.90	8.03
CHWST GT 66F	**57**	**48**	**9**	**16.3%**	12.76	8.10
CC SAT GT 62F	57	48	9	16.3%	12.76	8.10

Figure 25. Psychrometric analysis for Tinker AFB Highlighting the impact of the cooling recovery coil on HEDS performance.

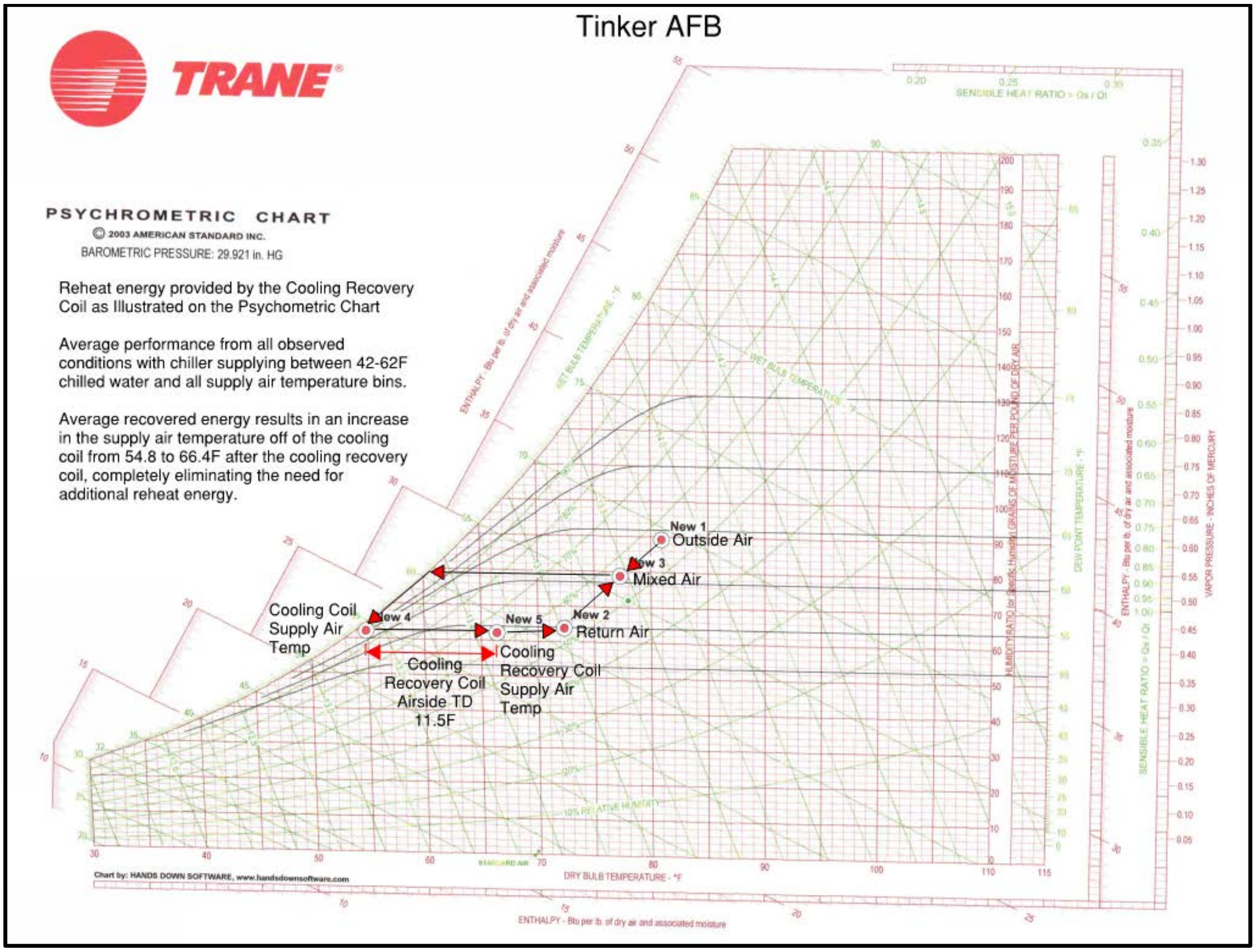


Figure 26. Tinker AFB dynamic HEDS savings and impact analysis for the peak cooling load on June 3, 2016.

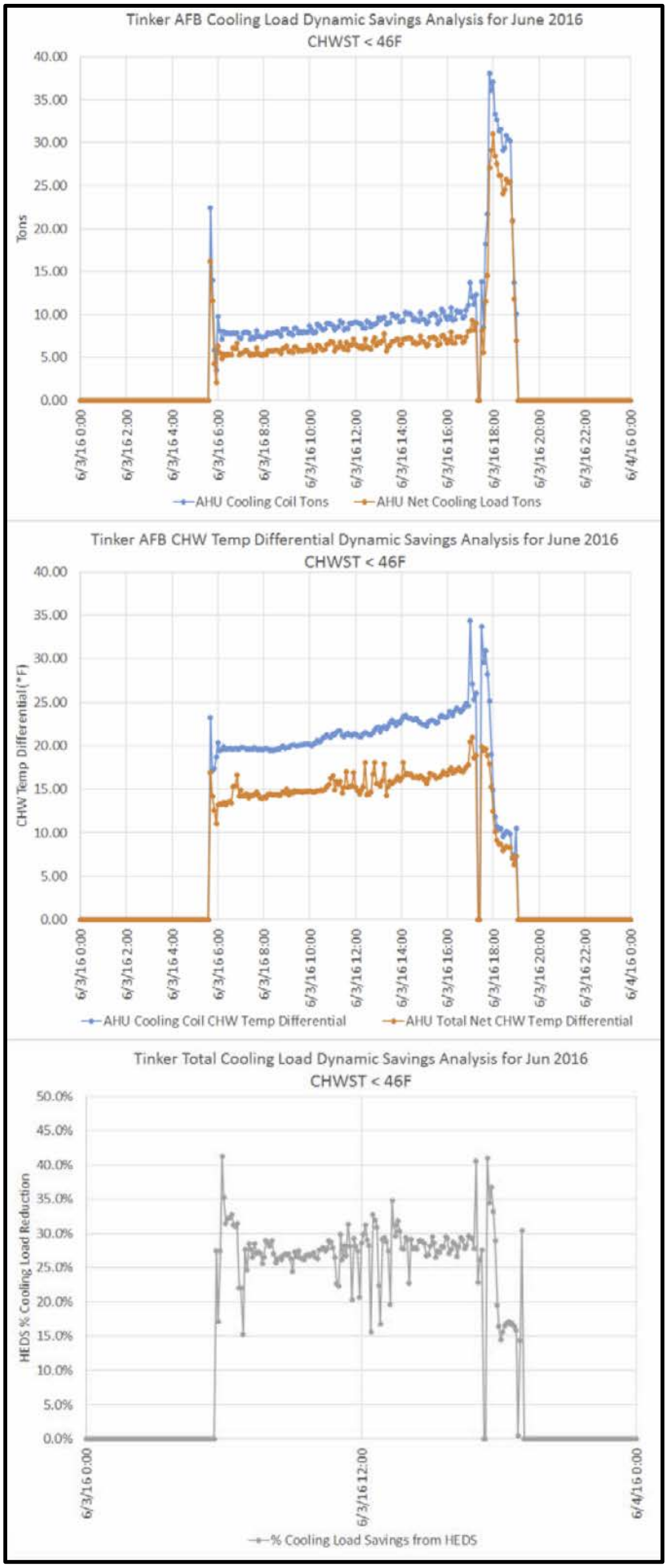


Figure 27. Tinker AFB HEDS dynamic savings analysis for part load days in the fall.

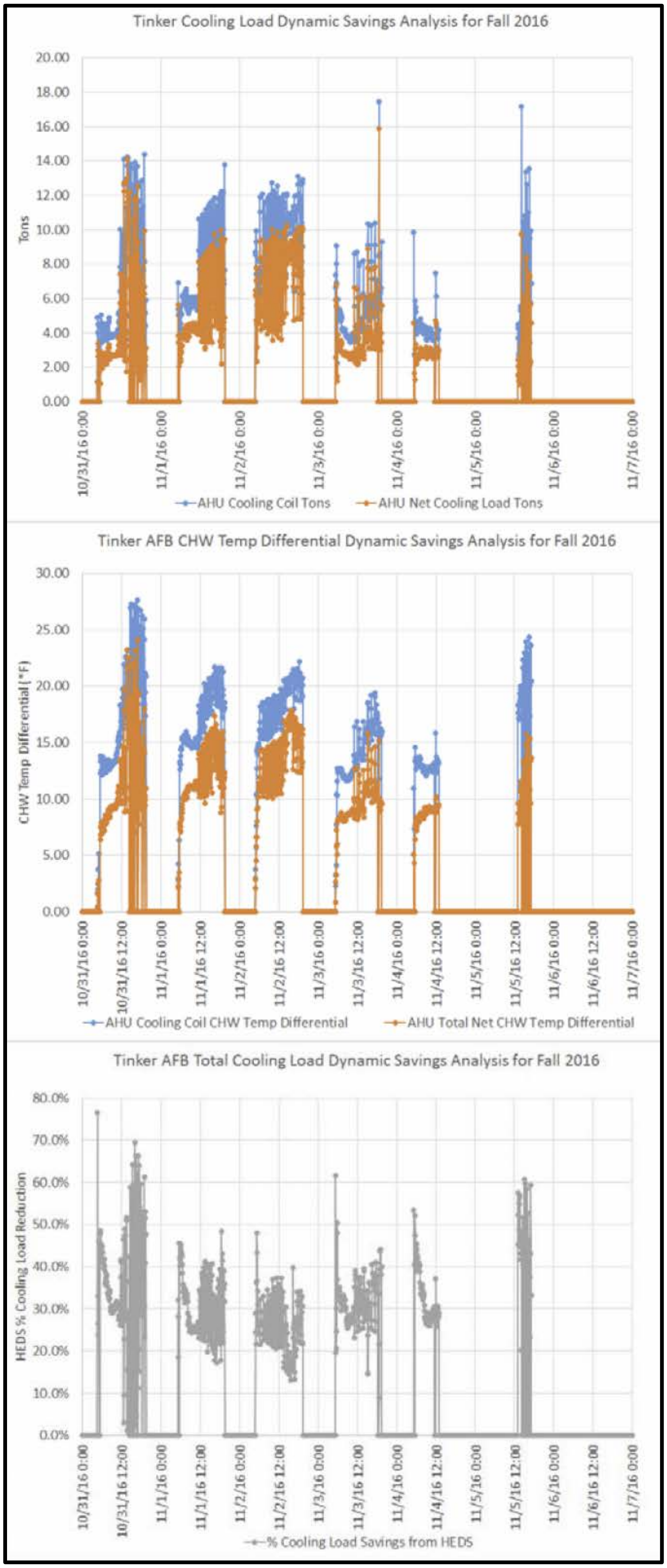
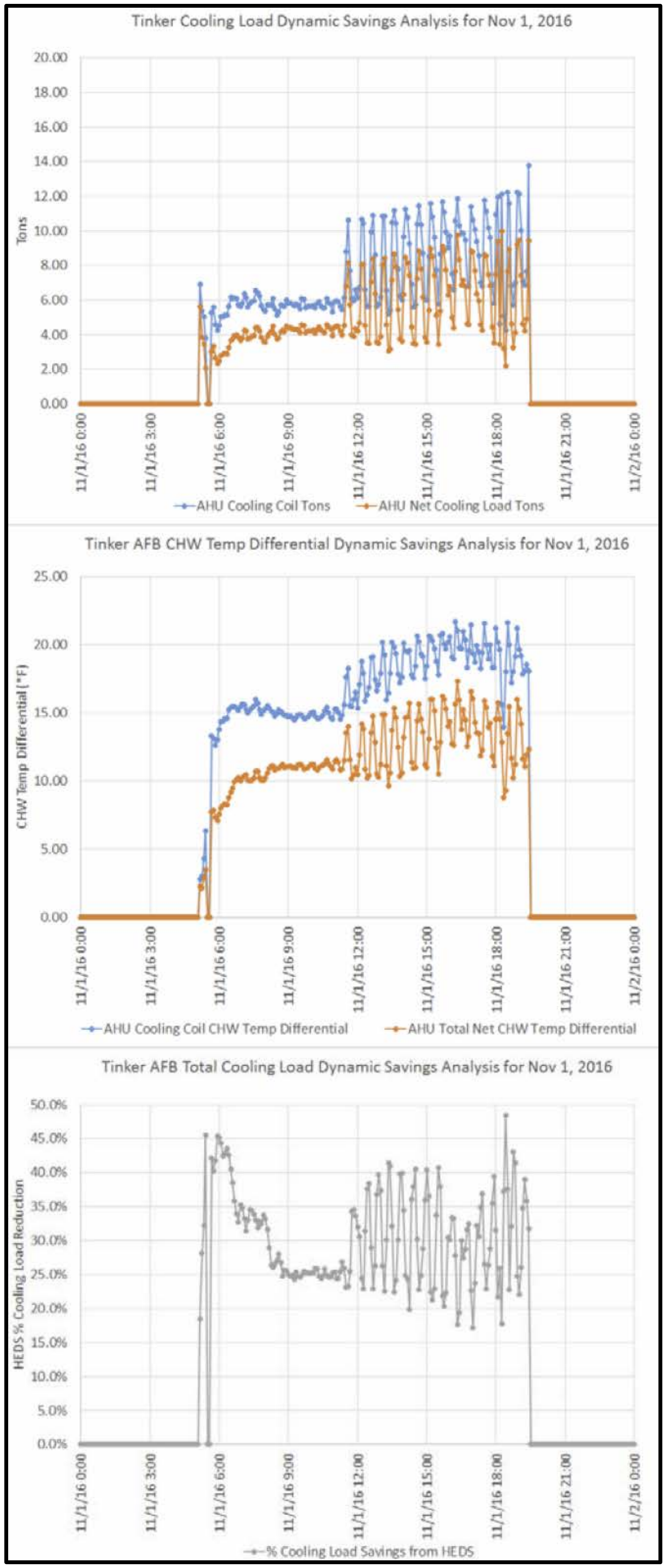
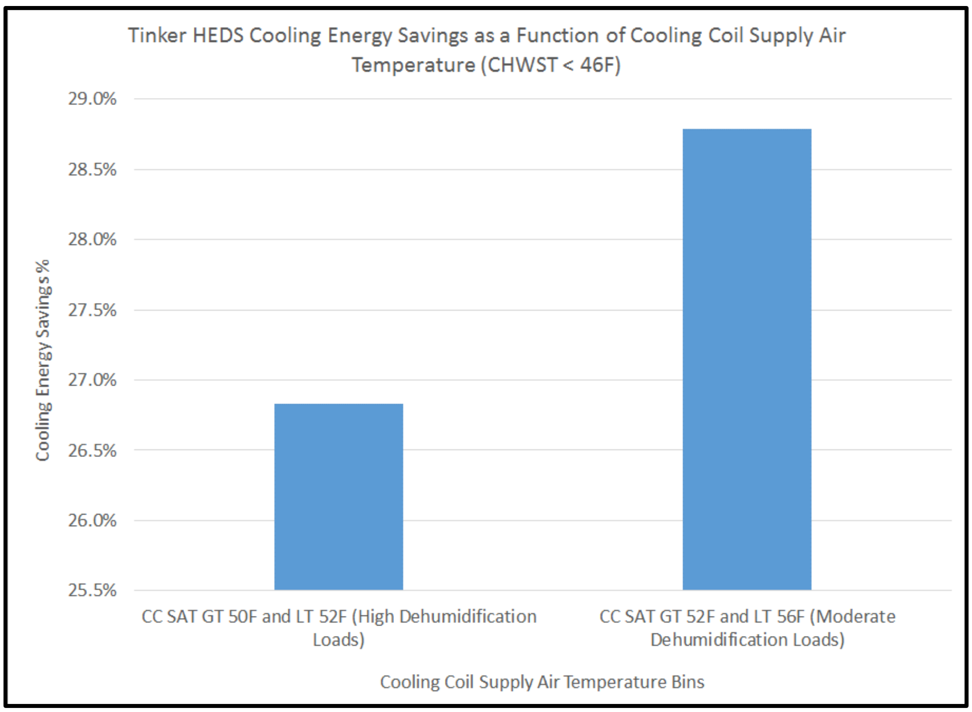


Figure 28. Tinker AFB HEDS dynamic savings analysis for part load day November 1, 2016.



As discussed previously (and shown in Figure 29), the HEDS-VAV results also indicate only a modest dependency of the savings on the cooling coil supply air temperature when the chiller plant is meeting the leaving chilled water supply temperature setpoint (less than 46 °F). As the cooling coil supply air temperature increases due to lower dehumidification loads, the leaving water temperature of the coil increases, enabling more heat recovery in the cooling recovery coil. The warmer supply air temperatures are well matched with the reduced cooling loads.

Figure 29. Tinker AFB HEDS cooling load reduction as a function of CC supply air temp when the chilled water temperature is in control (less than 46 °F).



HEDS performance was tested across a range of conditions to determine a savings envelope. Very low supply air temperatures (and thus dewpoint temperatures) less than 52 °F off the cooling coil are indicative of high dehumidification conditions typical of critical environments, while supply air temperatures off the cooling coil in the low to mid 50s (°F) are more typical of actual field conditions for office spaces like those served at Tinker AFB with the HEDS unit.

Table 11 lists the temperatures across the AHU for all temperature bins. With the chilled water supply temperature in control (less than 46 °F), the results show that, even at an average cooling coil leaving dewpoint of 52 °F, which provides significant dehumidification to the space, HEDS cooling load savings increase to nearly 29%, very similar to the Fort Bragg savings at the same conditions. In the higher dehumidification conditions, which delivered 50 °F dewpoint air off the cooling coil, HEDS still reduced the cooling load by nearly 27%.

Table 11. Tinker AFB average AHU temperatures across the range of operating conditions.

	Total HEDS Savings % (saved ton-hrs / CC ton-hrs)	Avg Outside Air Dewpoint Temp	Avg Return Air Dewpoint Temp	Avg Mixed Air Dewpoint Temp	Avg CC Supply Drybulb Temp	Avg CC Supply Dewpoint Temp	AHU CRC Supply Drybulb Temp	AHU CRC Supply Dewpoint Temp
All Included (Filtered) Data	**29.0%**	**63.88**	**54.62**	**60.41**	**54.85**	**53.87**	**66.39**	**53.91**
CHWST LT 46F (All Data)	**27.6%**	**61.33**	**51.57**	**56.88**	**52.65**	**51.76**	**63.73**	**51.76**
CC SAT LT 50F	20.5%	61.80	51.03	58.79	47.19	44.71	56.16	44.74
CC SAT GT 50F and LT 52F	26.7%	62.47	52.10	59.86	51.32	50.48	64.76	50.61
CC SAT GT 52F and LT 56F	28.8%	61.15	51.44	56.23	52.90	52.15	63.64	52.12
CC SAT GT 56F and LT 62F	20.9%	57.55	53.56	56.86	57.52	52.41	63.39	52.43
CC SAT GT 62F	10.9%	56.96	54.37	57.32	71.12	55.77	73.37	55.49
CHWST GT 46F and LT 50F	**30.3%**	**63.74**	**53.28**	**60.00**	**52.75**	**52.09**	**65.59**	**52.00**
CC SAT LT 50F	21.1%	65.15	53.62	60.32	49.41	51.88	57.93	48.13
CC SAT GT 50F and LT 52F	33.7%	65.46	52.83	60.33	51.44	51.26	65.49	50.97
CC SAT GT 52F and LT 56F	28.9%	63.13	53.43	59.89	53.13	52.37	65.75	52.35
CC SAT GT 56F and LT 62F	27.7%	59.17	54.10	58.12	56.88	53.95	66.12	55.52
CC SAT GT 62F	5.9%	66.36	54.48	65.30	68.48	52.82	72.98	59.47
CHWST GT 50F and LT 54F	**27.2%**	65.30	55.79	62.99	55.62	54.83	68.58	54.90
CC SAT GT 50F and LT 52F	28.5%	61.05	55.71	59.08	55.26	54.85	63.34	50.66
CC SAT GT 52F and LT 56F	26.9%	65.60	55.53	63.32	55.37	54.54	69.11	54.35
CC SAT GT 56F and LT 62F	29.1%	64.84	56.41	62.41	56.21	55.49	67.68	56.27
CC SAT GT 62F	7.9%	64.98	56.42	64.44	56.09	55.52	66.70	61.92
CHWST GT 54F and LT 58F	**31.9%**	67.73	59.55	64.61	59.38	58.64	70.19	59.19
CC SAT GT 52F and LT 56F	23.3%	68.50	58.74	64.52	56.45	55.93	65.60	54.68
CC SAT GT 56F and LT 62F	32.8%	67.75	59.54	64.62	59.49	58.74	70.42	59.25
CC SAT GT 62F	23.5%	67.04	60.15	64.70	58.95	58.23	68.54	61.31
CHWST GT 58F and LT 62F	**30.3%**	66.53	61.54	64.13	61.98	60.77	70.31	61.33
CC SAT GT 56F and LT 62F	27.7%	63.82	59.33	61.94	61.04	58.82	67.25	58.60
CC SAT GT 62F	31.0%	67.55	62.15	64.96	62.24	61.31	71.16	62.37
CHWST GT 62F and LT 66F	**27.0%**	66.49	61.96	64.91	62.90	61.14	71.13	63.22
CC SAT GT 62F	26.9%	66.46	62.01	65.04	62.96	61.19	71.15	63.30
CHWST GT 66F	**16.3%**	66.44	61.55	66.74	60.83	59.44	69.25	66.33
CC SAT GT 62F	16.3%	66.44	61.55	66.74	60.83	59.44	69.25	66.33

The AHU CRC drybulb temperatures at Tinker AFB listed in Table 11 indicate that the need for new reheat energy has been eliminated, which is similar to the Fort Bragg results.

When the cooling coil leaving dewpoint temp averages 50 °F, the HEDS units is able to deliver 65 °F drybulb air to the space, reducing mold potential from surface condensation and ensuring a comfortable indoor environment. At an average cooling coil dewpoint temperature of 52 °F and drybulb temperature of 53 °F, HEDS is able to provide 64 °F drybulb air to the space.

Even under the failing chiller conditions, the HEDS system is able to reduce cooling loads significantly, reserving capacity for other equipment in the system (Figure 30). Even when the chiller plant delivers chilled water above 55 °F, HEDS is able to provide significant dehumidification, and reduce the cooling load of the unit by more than 30%, effectively increasing the capacity of the system and delivering added capacity to the other AHUs when the chiller is not performing, thereby enhancing system resiliency.

Peak cooling load is also reduced across the operating conditions. Peak loads were only analyzed for the steady state conditions where the chilled water supply temperature from the chiller plant was in control. The results (listed in Table 12) show a 29% peak demand reduction during the peak cooling period on June 3, 2016.

Figure 30. Tinker AFB HEDS cooling load reduction as a function of chilled water supply temperature from the plant.

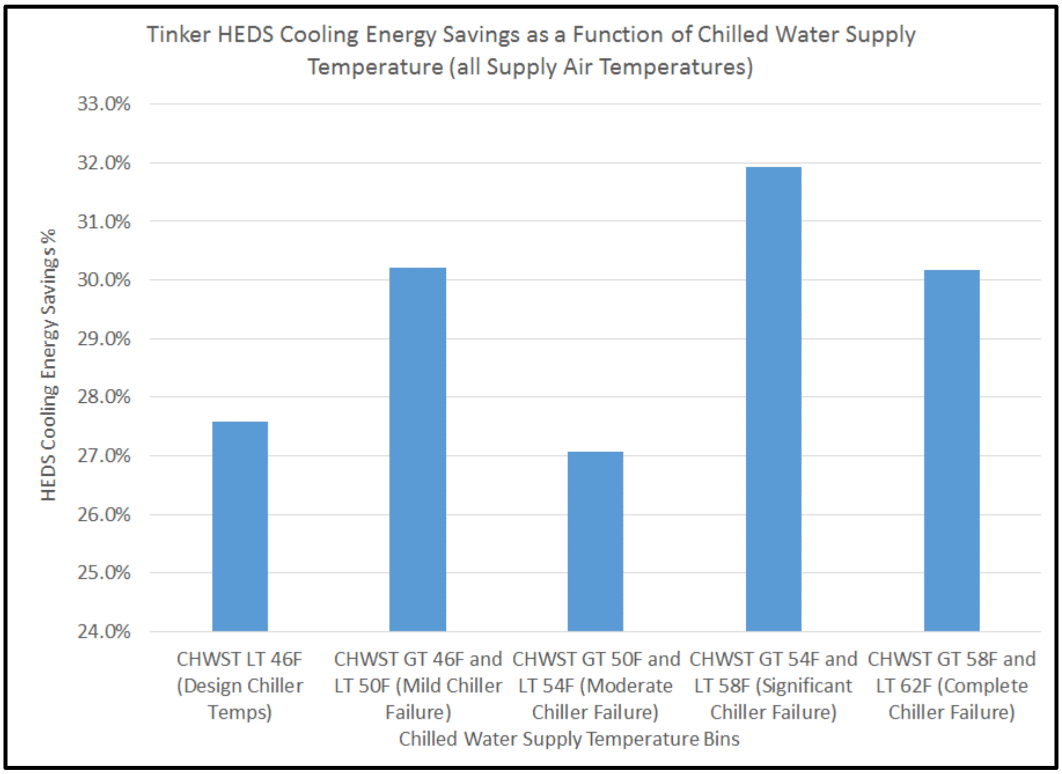


Table 12. Tinker AFB HEDS peak load reduction results when CHWST is in control (less than 46 °F).

	Peak Cooling Coil Load (tons)	Peak Net AHU Load (tons)	Peak Load Savings (tons)	Peak Load Savings %
All Included (Filtered) Data	**38.13**	**27.11**	**11.02**	**28.90%**
CHWST LT 46F (All Data)	**38.13**	**27.11**	**11.02**	**28.90%**
CC SAT LT 50F	no steady state operations			
CC SAT GT 50F and LT 52F	no steady state operations			
CC SAT GT 52F and LT 56F	38.13	27.11	11.02	28.90%
CC SAT GT 56F and LT 62F	no steady state operations			
CC SAT GT 62F	no steady state operations			

6 Performance Assessment

6.1 Savings analysis methodology

Although extensive performance monitoring equipment for both airside and waterside were installed on the AHUs, due to the nature of the HEDS system very few data points are required to determine the performance of the system compared to a typical AHU with reheat or "baseline" condition. This is a result of the fact that the cooling load is an independent, uncontrolled variable in the analysis since the cooling load is determined by the space and ambient conditions. Specifically, the leaving AHU cooling coil temperature is controlled to maintain primarily the space dewpoint and secondarily the space temperature within setpoints. (The control system resets the cooling coil leaving temperature based on the dewpoint temperature in the space. As the space dewpoint temperature rises, the cooling coil leaving temperature setpoint drops to provide increased dehumidification. As the space dewpoint falls, the cooling coil leaving temperature is increased to reduce unnecessary dehumidification.) Therefore, the baseline cooling load can be considered to be the load on the cooling coil in the HEDS unit, and the savings due to the cooling recovery coil are determined by the cooling coil load minus the net delivered AHU cooling load to the chiller plant

Baseline Cooling Load (tons) = (CC CHWST – CC CHWRT) * BEV CHW Flow (GPM) / 24
Net AHU Cooling Load (tons) = (CC CHWST – Common CHWRT) * BEV CHW Flow (GPM) / 24
HEDS Cooling Load Savings (tons) = Baseline Cooling Load – Net AHU Cooling Load

Given the design of the HEDS unit, all of the cooling energy recovered in the cooling recovery coil (by warming the airstream coming off the cooling coil) results in a direct decrease in energy required for reheat associated with dehumidification. Therefore, the cooling load savings in BTUs is exactly equal to the reheat energy savings, as the reheat requirement is displaced by the cooling recovery coil. For both project sites, additional reheat energy for RH control from other sources such as natural gas hot water systems or electric strip heating was completely eliminated, resulting in a 100% reduction in reheat energy.

Note that for the purposes of this demonstration, fan energy savings associated the HEDS low pressure drop coils have not been considered. To determine net electricity savings from the cooling load savings, the efficiency of the central plant must be considered

HEDS Electricity Savings (kW) = HEDS Cooling Load Savings (tons) * Chiller Plant kW/ton

Chiller plant efficiencies, considering all parasitic components, can typically range from 0.5 – 2.5 kW/ton, depending on the equipment, configuration, controls, and geographic location. Reheat energy savings calculations depend on the reheat energy source. For electric reheat, the cooling load savings can be converted directly to electricity with no losses

HEDS Electric Reheat Savings (kW) = HEDS Cooling Load Savings (tons) * 3.517 kW/ton

For boiler systems, the delivered efficiency of the hot water or steam must be considered

HEDS Boiler Reheat Savings (therms) = HEDS Cooling Load Savings (tons) * 0.12 therms/ton / Boiler System Efficiency

Considering the boiler efficiency, cycling losses, and distribution losses, typical delivered boiler system efficiencies can range from 30 to 85% depending on the system design, controls, delivery medium (steam, high temp hot water, hot water), and load factors.

6.2 Cooling load savings vs. cooling energy savings

One important distinction to make is that the percent of cooling load savings is a much smaller figure than the actual cooling energy percent savings that will show up at the utility meter.

For 24/7 loads, most chiller plants are equipped with VFDs on pumps and cooling towers, and many chiller plants that are 15 years old (or newer) also have one or more VFD high efficiency chillers. Due to Affinity Laws, equipment that is controlled by a VFD saves energy in a non-linear fashion, meaning a small speed reduction due to a load reduction equates to a much higher energy percent reduction. For example, chilled water pumps with VFDs operate with an Affinity Law power relationship of approximately 2.5, so if the cooling load goes down by 20%, the pump energy requirement equals (1-0.2)^2.5 = 57%, a savings of 43%. Condenser water pumps and cooling tower fan motors typically unload to the power of 2, while a VFD chiller may unload to the power of 1.5, so it can be seen that cooling energy savings can be significantly greater than cooling load percent savings.

Additionally, for administrative use facilities that are supposed to have 5 day per week, 12-hour per day HVAC occupancy schedules, but that may be running 24/7 due to inadequate and undersized HVAC AHUs, the switch to a HEDS-based system can reduce AHU and chiller plant runtime by 90+ hours per week, in addition to the cooling load reduction. This equates to an approximate 50% run time reduction, which provides not only substantial energy savings, but significant maintenance savings as well.

6.3 Data collection methodology and quality analysis

Extensive monitoring equipment has been installed on the HEDS units at both sites to facilitate detailed performance analyses. The following sections give an overview of the sensing equipment and data collection procedures. Appendices D and E include more detailed information and cut sheets.

6.3.1 Data description

The data spans from April 1 through November 9, 2016 for the analysis included in this report. During this range, trend data points at Fort Bragg were collected for 98.8% of the available time period, as some data gaps exist due to equipment outages and upgrades. During this range, trend data points at Tinker AFB were collected for 86.6% of the available time period, as some data gaps exist due to trend collection issues, equipment outages and upgrades.

Note that Tinker AFB has a Reheat Coil (RHC), and Fort Bragg does not have a RHC, so there are slight differences between the two sites. Appendix C includes a full list of monitoring points for each site. Key monitoring points used for the calculations are:

- Trend Points Used for Energy Savings Calculations:
- CC Chilled Water Flow from the Belimo Energy Valve (GPM)
- DDC CC Chilled Water Supply Temperature (°F)
- DDC CC Chilled Water Return Temperature (°F)
- DDC Common Chilled Water Return Temperature (°F).
- Trend Points Used for Data Filtering, Data Quality Analyses, and Reporting:
- CC Chilled Water Flow from the Belimo Energy Valve (GPM)

- DDC CC Chilled Water Supply Temperature (°F)
- DDC CC Chilled Water Return Temperature (°F)
- DDC Common Chilled Water Return Temperature (°F)
- DDC Cooling Recovery Coil Chilled Water Return Temperature (°F)
- Supply Air Flow (CFM) (Fort Bragg only)
- Return Air Flow (CFM) (Tinker AFB only)
- Supply Fan Power (kW)
- CC CHW Valve Position (%)
- CRC CHW Valve Position (%)
- Mixed Air Dewpoint Temperature (°F)
- Mixed Air Drybulb Temperature (°F)
- CC Supply Air Dewpoint Temperature (°F)
- CC Supply Air Drybulb Temperature (°F)
- Space Dewpoint Temperature (°F)
- Return Air Dewpoint Temperature (°F)
- Outside Air Dewpoint Temperature (°F)
- Cooling Recovery Coil Supply Air Dewpoint Temperature (°F)
- Cooling Recovery Coil Supply Air Drybulb Temperature (°F)
- Belimo Energy Valve CC CHW Delta T (°F).

6.3.2 Data quality

Given the savings validation approach for HEDS that focuses on cooling coil load and net AHU cooling load, that considers the effect of the cooling recovery coil, five sensors are required to accomplish the performance validation and data quality analysis. The sensors included in the performance analysis, along with a description and accuracy ratings are:

> CC CHW Entering Temperature (DDC). Minco 4-wire Platinum Resistance Temperature (PRT) sensors with the highest rated accuracy from Minco, at 0.1%. Including the wiring and transducers, the matched system accuracy is 0.75%.
> CC CHW Leaving Temperature (DDC). Minco 4-wire Platinum Resistance Temperature (PRT) sensors with the highest rated accuracy from Minco, at 0.1%. Including the wiring and transducers, the matched system accuracy is 0.75%.
> Cooling Recovery Coil CHW Leaving Temperature (DDC). Minco 4-wire Platinum Resistance Temperature (PRT) sensors with the highest rated accuracy from Minco, at 0.1%. Including the wiring and transducers, the matched system accuracy is 0.75%.

Common CHW Leaving Temperature (DDC). Minco 4-wire Platinum Resistance Temperature (PRT) sensors with the highest rated accuracy from Minco, at 0.1%. Including the wiring and transducers, the matched system accuracy is 0.75%.
Belimo Energy Valve (BEV) CHW Flow Rate. Belimo Energy Valve internal flowmeter, with a rated accuracy of ±2% of flow rate, but more importantly, it is rated at 0.50% for repeatability.

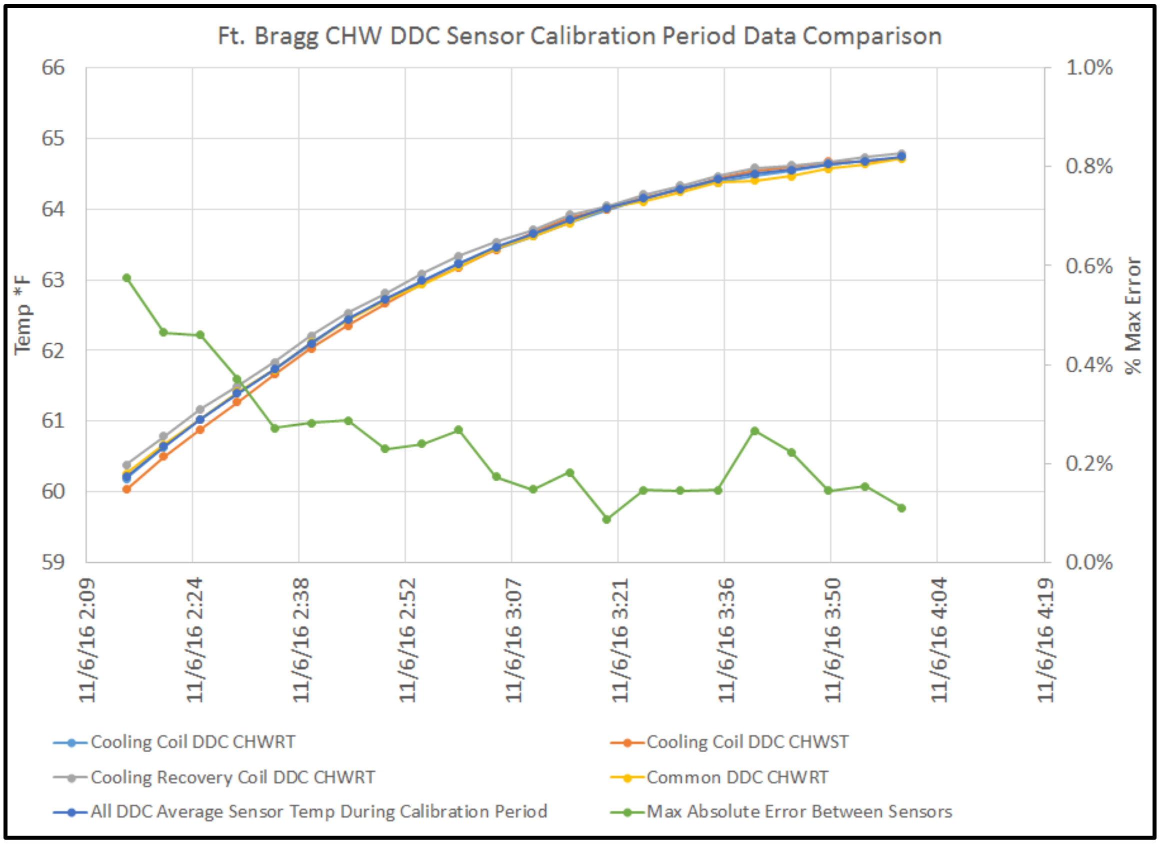
To ensure the quality of the waterside temperature data, sensor validations were performed upon initial installation in the field by Trane personnel. Additionally, in the fall, a sensor calibration procedure was implemented in the controls sequence of each AHU to enable ongoing validation. This was accomplished by turning the AHUs off while opening the cooling coil and cooling recovery coil valves while leaving the chilled water pumps still running, which allows real time comparison of the temperature sensors described above between themselves. Under steady state conditions, each of the sensors in series will see the same chilled water flow, with no air flow so the waterside temperature readings should be nearly identical. To eliminate periods of cooling coil loss even with the fan off, the calibration data were filtered to include only data where the mixed air dewpoint temperature before the cool coil and leaving dewpoint temperature after the cooling coil are within 3 °F, and where chilled water temperature dynamics were low (less than 0.6 °F change in CHWS temperature in 5 minutes).

For the calibration analysis, the data from the four temperature sensors in series were averaged, then the deviation from the average as calculated for each individual sensor. This deviation was then averaged for the calibration period to determine an adjustment factor to calibrate the sensors against. This will eliminate any persistent bias in the data that may alter the Delta T results for the cooling coil and AHU net of the cooling recovery coil.

6.3.3 Fort Bragg data calibration analysis

Figure 31 shows the data that inform a calibration analysis for the Fort Bragg DDC chilled water sensors that shows that there was less than a 1% error between the sensors throughout the calibration period when considering the maximum observed range (from highest to lowest sensor) at each data point. On average, the sensors were within 0.2% of each other during the calibration period.

Figure 31. Calibration period results highlighting the low error between sensors for Fort Bragg.



Using these data, calibration offsets were derived for each sensor by comparing the deviation from the average sensor temperature at each point, then averaging this deviation within each sensor. Table 13 lists the resulting offsets, which, although almost insignificant, have been included in the performance analysis, back cast for all data points in the analysis period.

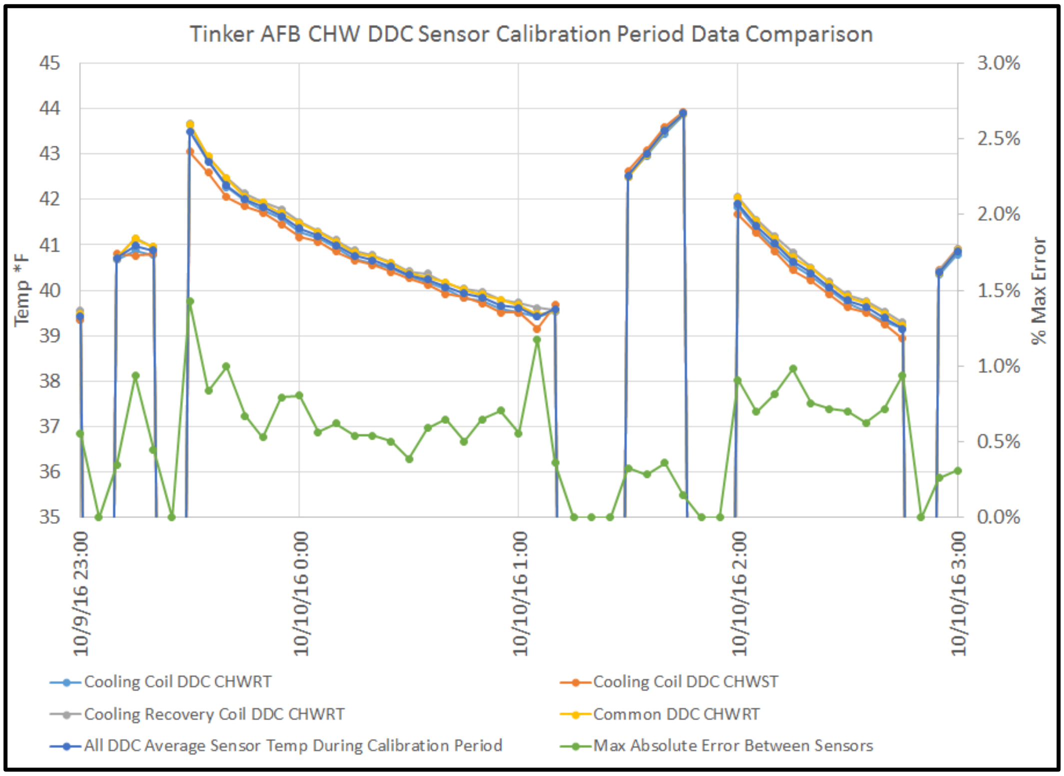
Table 13. Calibration period results highlighting the low error between sensors for Fort Bragg.

HEDS, AHU_FTBragg, CC-CHWS Temperature (Units: °F) DEVIATION FROM ALL SENSOR CAL AVG	HEDS, AHU_FTBragg, CC-CHWR Temperature (Units: °F) DEVIATION FROM ALL SENSOR CAL AVG	HEDS, AHU_FTBragg, CRC-CHWR Temperature (Units: °F) DEVIATION FROM ALL SENSOR CAL AVG	HEDS, AHU_FTBragg, Common-CHWR Temperature (Units: °F) DEVIATION FROM ALL SENSOR CAL AVG
0.039	-0.015	0.080	-0.026

6.3.4 Tinker AFB calibration analysis

By focusing in on a period where the chilled water supply temperature from the plant was stable, the calibration analysis for the Tinker AFB DDC chilled water sensors showed that there was less than 1.5% error between the sensors throughout the calibration period when considering the maximum observed range (from highest to lowest sensor) at each data point (Figure 32). On average, the sensors were within 0.6% of each other.

Figure 32. Calibration period results highlighting the low error between sensors for Tinker AFB.



Using these data, calibration offsets were derived for each sensor by comparing the deviation from the average sensor temperature at each point, then averaging this deviation within each sensor. Table 14 lists the resulting offsets, which, although almost insignificant, have been included in the performance analysis, back cast for all data points in the analysis period.

Table 14. Calibration period results highlighting the low error between sensors for Tinker AFB.

HEDS Tinker, VAV_Tinker, CC-CHWS Temperature (Units: °F) DEVIATION FROM ALL SENSOR CAL AVG	HEDS Tinker, VAV_Tinker, CC-CHWR Temperature (Units: °F) DEVIATION FROM ALL SENSOR CAL AVG	HEDS Tinker, VAV_Tinker, CRC-CHWR Temperature, (Units: °F) DEVIATION FROM ALL SENSOR CAL AVG	HEDS Tinker, VAV_Tinker, Common-CHWR Temperature (Units: °F) DEVIATION FROM ALL SENSOR CAL AVG
-0.160	-0.096	-0.021	-0.058

6.3.5 Data filtering

To ensure proper evaluation of HEDS performance before calculating the new cooling load savings and reductions from baseline, data quality filters where applied to remove out of range data and anomalous/ transient conditions that may skew the results. The following conditions were enforced for data to be included in the analysis

- Supply Fan On
  - Fort Bragg: measured supply airflow > 100CFM and fan power > 0.5kW (Typical supply air flow while the unit is running is a constant 8000CFM and typical fan power is 4kW) Typical return airflow while the unit is running is greater than 1800CFM even during low load conditions)
- Cooling being provided (BEV chilled water flow > 3 GPM and calculated tons cooling > 2 tons)
  - Fort Bragg: typical chilled water flow is between 5 and 70 GPM with CC CHW valve open
  - Tinker AFB: typical chilled water flow is between 5 and 80 GPM with CC CHW valve open
- Chilled water supply temperature from the chiller plant between 30 and 70 °F
  - Fort Bragg: design chilled water supply temperature is 45 °F; typical chilled water supply temperature from the plant is between 42 and 65 °F
  - Tinker AFB: design chilled water supply temperature is 45 °F; typical chilled water supply temperature from the plant is between 40 and 70 °F
- All DDC chilled water temperatures between 30 and 100 °F
  - Typical temperatures at both locations range between 40 and 80 °F

When applied, these data quality filters resulted in the removal of approximately 820 hours of data from the Fort Bragg dataset, or roughly 15% of the collected dataset. For Tinker, due to the much lower runtimes from the AHU shutting down at night and the poor plant chilled water supply temperature control with multiple chiller failures that allowed supply temperatures to get as high as 80+ °F, the filtering resulted in the removal of 79% of the data, leaving 21% of the data points collected for performance analysis. While this is a significant reduction in the available data points, nearly

1000 hours (over 40 days) of filtered performance data remain, ensuring significance of the performance results.

6.3.6 Performance objectives results summary

Tables 15 and 16 summarize the results of each objective for the test sites. All success criteria were met, and were often substantially exceeded, across all objectives at both test sites.

Table 15. Quantitative performance objective results summary for the Fort Bragg Test Site.

Performance Objective	Success Criterion	Results (CHWST<46 °F)
1. Peak Cooling Load Reduction %	Reduce 15-minute cooling load by 15% on a peak cooling load day during the demonstration period	18.3%
2. Greatest Cooling Load Reduction %	Highest average cooling load % reduction exceeds 20% during the demonstration period	37.4%
3. Dehumidification /Reheat Coil Energy Reduction	CRC coil eliminates the need for at least 90% of the RH-control-related reheat energy required from the reheat coil during the time the system is in dehumidification-reheat mode during the demonstration period	100.0%
4. Enhance Space Comfort Conditions	Space conditions fall within UFC comfort guidelines more than 90% of the time during operating hours	96.0%
5. Reduce Cooling Ton-Hours Consumption	Cooling ton-hours associated with the HEDS unit are reduced by the CRC by 7.5% compared to the ton-hours consumed by the cooling coil during the time that the HEDS is in dehumidification-reheat modes during the demonstration period	24.7%
6. Improve "Low Delta T" Syndrome	HEDS average cooling coil CHW system TD exceeds 14 °F during the time that the HEDS is in dehumidification-reheat modes during the demonstration period	17.1%
7. Reduce Greenhouse Gas (GHG) Emissions	GHG emission reductions associated with the dehumidification/reheat process exceed 3% (annual comparison)	45–79%
8. Reduce Energy cost of Dehumidification/ Reheat process	Cost of dehumidification and reheat with HEDS vs. CV subcool/terminal reheat is reduced by 10% during dehumidification –reheat modes of operation	41–51%+
9. System Economics Reduce Lifecycle cost of Dehumidification/ Reheat process	5% reduction in lifecycle costs	Retrofit: 26–29%+ New construction/ EUL: 38–44%+
10. Savings vary based on central plant chiller and heating system efficiencies; uses eGrid national average electricity emissions factors.		

Table 16. Performance objective summary for the Tinker AFB Test Site.

Performance Objective	Success Criterion	Results (CHWST<46 °F)
1. Peak Cooling Load Reduction %	Reduce 15-minute cooling load by 15% on a peak cooling load day during the demonstration period	28.9%
2.Greatest Cooling Load Reduction %	Highest average cooling load % reduction exceeds 20% during the demonstration period	28.7%
3. Dehumidification /Reheat Coil Energy Reduction	CRC coil eliminates the need for at least 90% of the RH-control-related reheat energy required from the reheat coil during the time the system is in dehumidification-reheat mode during the demonstration period	100.0%
4. Enhance Space Comfort Conditions	Space conditions fall within UFC comfort guidelines more than 90% of the time during operating hours	98.0%
5. Reduce Cooling Ton-Hours Consumption	Cooling ton-hours associated with the HEDS unit are reduced by the CRC by 7.5% compared to the ton-hours consumed by the cooling coil during the time that the HEDS is in dehumidification-reheat modes during the demonstration period	27.6%
6. Improve "Low Delta T" Syndrome	HEDS average cooling coil CHW system TD exceeds 14 °F during the time that the HEDS is in dehumidification-reheat modes during the demonstration period	24.0%
7. Reduce Greenhouse Gas (GHG) Emissions	GHG emission reductions associated with the dehumidification/reheat process exceed 3% (annual comparison)	70–86%
8. Reduce Energy cost of Dehumidification/ Reheat process	Cost of dehumidification and reheat with HEDS vs. CV subcool/terminal reheat is reduced by 10% during dehumidification –reheat modes of operation	68–75%+
9. System Economics Reduce Lifecycle cost of Dehumidification/ Reheat process	5% reduction in lifecycle costs	Retrofit: 13–41%+ New construction/ EUL: 43–61%+
10. Savings vary based on central plant chiller and heating system efficiencies; uses eGrid national average electricity emissions factors. 11. Savings vary based on central plant chiller and heating system efficiencies, as well as electricity and gas commodity rates. Average potential savings over a range of potential cost, efficiency, and reheat source scenarios is shown.		

7 Cost Assessment

This chapter describes the cost components of the HEDS system, along with cost-benefit assessments and lifecycle cost analysis.

7.1 Cost model

HEDS components are very similar to components in a typical AHU designed for dehumidification duty. As such, cost estimating for the HEDS system is very similar to cost estimating for typical dehumidification-duty AHU deployments in either new construction or retrofit scenarios. One key difference is that the HEDS unit will typically cost 2 to 3 times that of a typical AHU, given the large coil sections, low face velocities, and enhanced controls.

Table 17 lists the elements of a simple cost model for cost estimating support for HEDS projects. Note that the actual equipment and installation costs will vary significantly due to a number of factors, including:

- Requirements for marine environments and coil coatings
- Requirements for special duty such as low dewpoint applications, redundancy, etc.
- Location of the unit (roof, mechanical room, etc.) and whether the unit will be installed as separate components or broken up to fit the unit into an existing space
- Existing available of chilled water distribution.

Table 17. Cost elements of HEDS.

Cost Element	Data Tracked During the Demonstration	Estimated Costs
Equipment capital costs	Estimates made based on component costs for demonstration, includes all HEDS AHU components and controls	$10/CFM
Installation costs	Labor and material required to install, including curbs/ pads, electrical connections, etc.	$6/CFM
Consumables	Air Filters are the only consumables, typical of a normal AHU; data not tracked	Same as typical AHU, no additional cost
Facility operational costs	Reduction in energy required vs. baseline data	See Performance Results

Cost Element	Data Tracked During the Demonstration	Estimated Costs
Maintenance	Frequency of required maintenance Labor and material per maintenance action Data not tracked	Same as typical AHU, no additional cost
Hardware lifetime	Estimate based on components degradation during demonstration	25 years, based on typical AHU lifetimes
Operator training	Estimate of training costs	Included in equipment costs, no additional cost

The following sections describe each cost element.

7.1.1 Equipment capital costs

The capital cost for the HEDS AHU is typically the largest cost element of a HEDS project. This cost includes all of the elements typical of a dehumidification-duty AHU, including supply fan(s), return or exhaust fan(s) (if required) preheat coil (if required), cooling coil, cooling recovery coil, control valves, reheat coil or strip heating (if required), instrumentation, controls, filter and access sections, and dampers. The cost data presented are based on multiple cost estimates from Trane for units of similar size to those deployed at the test sites. The test site unit costs could not be used directly due to the enhanced instrumentation, which significantly increase first costs. Future equipment costs can be explored with approved HEDS AHU providers on a case-by-case basis as is typical of the AHU industry.

7.1.2 Installation costs

Installation costs are the other key cost element of a HEDS installation. These costs typically include all elements of the physical installation including demolition of existing units (as needed), new pads or curbs to accept the new HEDS unit, electrical and controls connections, ductwork connections, start-up, balancing, and commissioning. Cost estimates were developed based on typical AHU installations on-grade. Note that many factors will influence installation costs in the field, and can be estimated based on typical AHU estimating techniques (assuming a HEDS AHU is roughly twice the size of a standard AHU with 500-ft-per-minute face velocity).

7.1.3 Consumables

The only consumables for the HEDS units are air filters, as is typical of any AHU. Typically, air filters in HEDS units will last longer than those in traditional AHUs due to lower face velocities, which reduce the total pressure drops experienced by the filters for a given loading. However, there may be more filters in a HEDS unit so it is expected that the net costs of filters will be neutral. At the time of the report, the filters had not yet been changed, but it is expected that the filter cost may be neutral, or slightly less expensive overall than a normal AHU.

7.1.4 Facility operational costs

As discussed throughout this report, energy costs will be significantly reduced for HEDS units compared to almost any other dehumidification technology. Energy cost savings will vary based on a number of factors, including utility rates, chilled water plant efficiencies, and reheat plant type and efficiency. The total energy savings will also depend on the baseline system with which HEDS is being compared. This work compared HEDS against a typical dehumidification reheat AHU with a cooling coil and reheat coil provided by hot water or electric strip heat.

7.1.5 Maintenance

HEDS AHUs have components typical of any chilled water AHU (as described above). Therefore, maintenance requirements are similar to those of any other AHU. At the time of this writing, the HEDS AHU has required no maintenance. The HEDS AHUs use direct drive, rather than belt drive, fans and motors so the annual maintenance costs should be slightly lower than a normal AHU.

7.1.6 Hardware lifetime

Again, since HEDS is so similar to a typical AHU, lifetimes are expected to be similar to any other chilled water AHU. The BOMA Preventative Maintenance Guidebook (Schoen 2010) uses the following average equipment lifetimes for AHU equipment, which will be similar for HEDS units based on application

- Severe Duty or 100% Outdoor Air Units: 20 years
- Packaged Medium Duty: 25 years
- Built-up Heavy Duty: 30 years.

7.1.7 Operator training

Operator training is an important component of any robust operating and maintenance program. Although HEDS units are very similar to traditional chilled water AHUs in components and layout, specialized control functions require training to support ongoing performance. This training is included with the purchase of the HEDS units based on existing manufacturer licensing agreements, so currently has no upfront cost.

7.2 Cost drivers

Many of the key cost drivers for HEDS deployment were discussed in the Sections 7.1.1 to 7.1.7 . There are infinite combinations of requirements for any AHU selection that will affect total system cost. Two key elements are crucial to how the net cost of a HEDS project should be considered:

1. First, it is critical to consider what the baseline equipment selection is or would have been, and consider the incremental cost (or savings) associated with the HEDS unit deployment. For example, for a building project with a simple chilled water reheat AHU as the baseline, the cost of the baseline system may be $3/CFM compared with a HEDS unit at $10/CFM. Installation costs between the two units are likely similar, when the larger footprint and weight of the HEDS installation compared with this baseline is offset by the smaller pipe diameters and pump flow rates required to serve the HEDS unit. Assuming a $1/CFM higher installation cost, the net cost of the HEDS installation would be $8/CFM, typically resulting in a very fast payback when the energy savings are considered compared with a traditional reheat AHU baseline.

   Now consider the case where new sources of reheat are not allowable by code (such as governed by ASHRAE 90.1). In this case, the baseline system may be a direct expansion (DX) unit with hot gas reheat, which eliminates the need for new reheat sources. The first cost for this DX unit may be $9/CFM, compared with a HEDS unit at $10/CFM. Installation costs may be slightly higher for the HEDS unit for chilled water piping, but this may be offset by the increased electrical distribution costs and higher weight of the DX unit. Even assuming the HEDS unit installation has a net cost $2/CFM higher, the net cost of the HEDS unit is only $3/CFM. In this case, HEDS does not provide reheat energy savings compared with the baseline, but does provide significant cooling energy savings given the higher efficiency of a chilled water plant compared to air-cooled DX equipment and the reduced load of the HEDS unit from the cooling recovery process for the reheat.

2. Another key element to consider in HEDS installations is that of the space constraints for the unit. In certain applications, particularly retrofit scenarios, there simply may not be adequate space for a HEDS unit to fit properly, given its size compared to tradition dehumidification-reheat solutions. However, in situations where a reheat AHU is not allowed (as discussed above, for example where ASHRAE 90.1 energy code is in effect), a like-for-like retrofit would not be possible anyway, in that such a retrofit would require additional engineering to develop solutions to enable other technologies like HEDS to be deployed to eliminate the use of new energy for reheat. These scenarios can greatly increase the overall installation cost, but this cost would be seen in both the baseline and HEDS scenarios.

7.3 Cost analysis and comparison

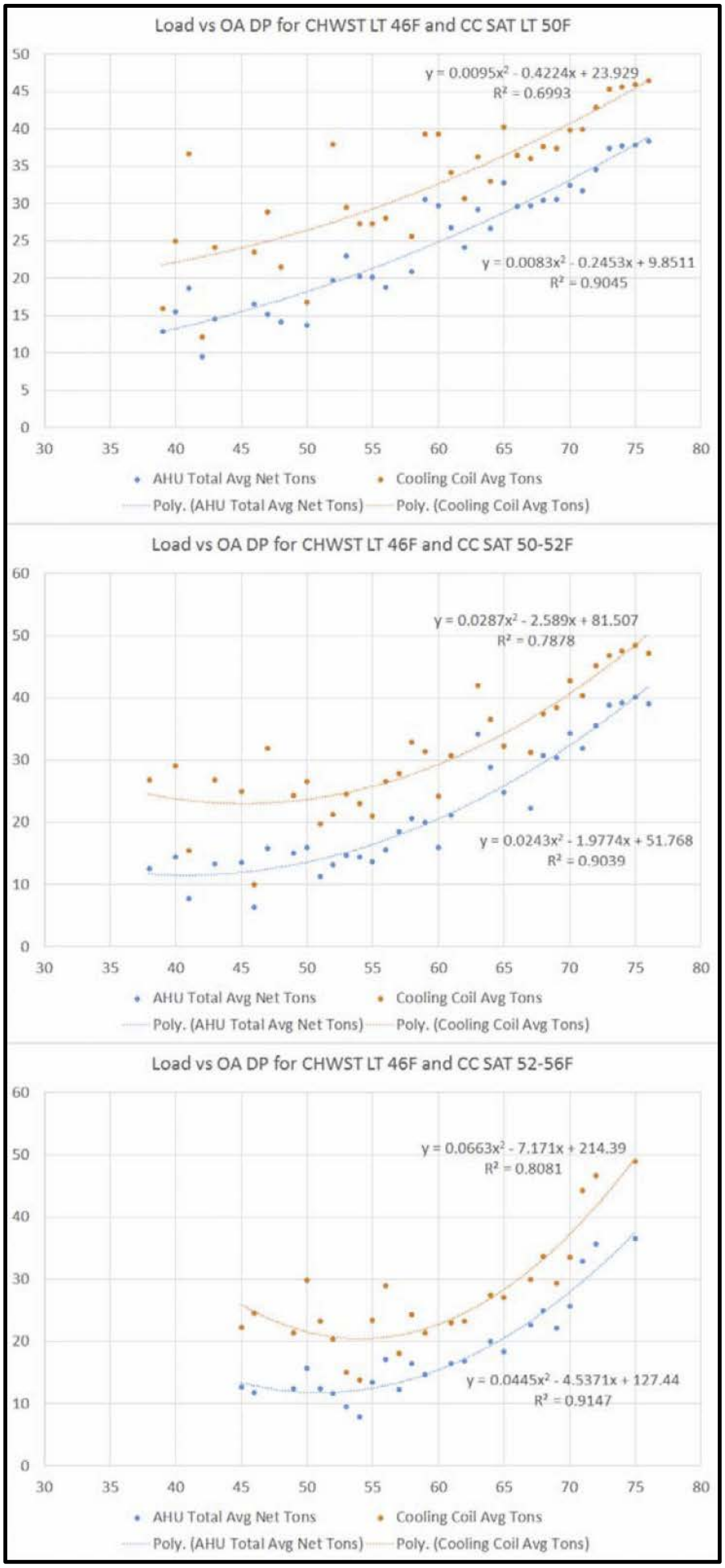
The following sections give an overview of the cost comparisons and lifecycle analyses for both test sites.

7.3.1 Fort Bragg

Translating the cooling and dehumidification-related reheat load savings to energy savings requires an analysis of the central plant operations serving the HEDS units. Given that load reductions have nonlinear impacts on energy use for most chilled water systems, a spreadsheet model was developed to explore potential HEDS energy, GHG, and lifecycle cost savings across a range of chilled water and heating plant efficiencies and types.

To accomplish this, a model of the baseline load and AHU net load with HEDS was developed as a function of average daily outdoor air dewpoint temperatures. Figure 33 shows the resulting modeled performance for each cooling coil supply air temperature bin.

Figure 33. Characteristic load curves for the baseline and net HEDS cooling loads as a function of average daily outdoor air temperature for Fort Bragg.



The modeled curves were then applied to typical meteorological year (TMY3) dewpoint data to develop annualized savings estimates for each cooling coil temperature range. An annual schedule was applied to chiller plant operations such that the savings analysis only occurs during the dehumidification season from May through October, reflecting the annual wintertime shutdown from approximately November through April. Also, to appropriately determine total plant energy impacts, it was assumed that HEDS units would be used on the entire load for each chiller plant (i.e., all AHUs would be replaced with HEDS units). This simplifies the analysis of total energy impacts compared with partial HEDS implementations.

As discussed, energy, GHG, and lifecycle cost impacts were calculated across a range of chiller and boiler plant efficiency scenarios

Chilled Water Plant Scenarios:

- High Efficiency Water Cooled Chillers: IPLV 0.45
- Moderate Efficiency Water Cooled Chillers: IPLV 0.56
- Poor Efficiency Water Cooled / High Efficiency Air-Cooled Chillers: IPLV 0.79
- Moderate Efficiency Air-Cooled Chillers: IPLV 1.01
- Poor Efficiency Air-Cooled Chillers: IPLV 1.35.

Heating Plant Scenarios (net efficiencies include boiler, cycling, and distribution losses):

- Electric Resistance: 100% net delivery efficiency
- High Efficiency Condensing Boilers: 80% net delivery efficiency
- High Efficiency Non-Condensing Boilers: 70% net delivery efficiency
- Moderate Efficiency Non-Condensing Boilers: 60% net delivery efficiency
- Poor Efficiency Non-Condensing Boilers: 50% net delivery efficiency.

Tables 18 to 20 list the resulting energy savings, both in total energy and savings percent, for each cooling coil supply air temperature bin. Note that total energy savings associated with cooling and dehumidification-related reheat are greater than 50% in all cases, and reach as high as 87% with the combination of a high efficiency chilled water plant and very low efficiency boiler system. Absolute savings are highest at the lower cooling coil supply air temperatures indicative of higher dehumidification loads, given the higher total system load for that condition.

Table 18. Energy savings results for CC supply air temperatures below 50 °F (very high dehumidification loads).

	Savings	Plant Electricity Only Savings (kWh)	Total Energy Savings by Reheat System Type and Efficiency (MMBTU)				
			Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	108,674	1,777	2,185	2,436	2,771	3,240
	Moderate Efficiency Water Cooled	131,083	1,854	2,262	2,513	2,848	3,317
	Poor Efficiency Water Cooled / High Eff Air Cooled	175,915	2,007	2,415	2,666	3,001	3,470
	Moderate Efficiency Air Cooled	220,734	2,160	2,568	2,819	3,154	3,623
	Poor Efficiency Air Cooled	287,975	2,389	2,797	3,048	3,383	3,852
	Savings %	Plant Electricity Only Savings	Total Energy Savings by Reheat System Type and Efficiency				
			Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	32.7%	70.0%	74.1%	76.1%	78.4%	80.9%
	Moderate Efficiency Water Cooled	32.4%	66.5%	70.8%	72.9%	75.3%	78.0%
	Poor Efficiency Water Cooled / High Eff Air Cooled	32.1%	61.2%	65.5%	67.7%	70.2%	73.2%
	Moderate Efficiency Air Cooled	31.8%	57.3%	61.4%	63.6%	66.2%	69.2%
	Poor Efficiency Air Cooled	31.7%	53.0%	56.9%	59.0%	61.5%	64.5%

Table 19. Energy savings results for CC supply air temperatures between 50 and 52 °F (moderate dehumidification loads).

	Savings	Plant Electricity Only Savings (kWh)	Total Energy Savings by Reheat System Type and Efficiency (MMBTU)				
			Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	99,982	1,854	2,288	2,558	2,919	3,423
	Moderate Efficiency Water Cooled	120,824	1,925	2,359	2,630	2,990	3,494
	Poor Efficiency Water Cooled / High Eff Air Cooled	162,521	2,067	2,502	2,772	3,132	3,636
	Moderate Efficiency Air Cooled	204,205	2,210	2,644	2,914	3,274	3,778
	Poor Efficiency Air Cooled	266,744	2,423	2,857	3,127	3,488	3,992
	Savings %	Plant Electricity Only Savings	Total Energy Savings by Reheat System Type and Efficiency				
			Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	35.8%	75.2%	78.9%	80.7%	82.6%	84.8%
	Moderate Efficiency Water Cooled	35.5%	71.9%	75.9%	77.8%	79.9%	82.3%
	Poor Efficiency Water Cooled / High Eff Air Cooled	35.1%	66.8%	70.9%	73.0%	75.3%	78.0%
	Moderate Efficiency Air Cooled	34.9%	62.9%	67.0%	69.1%	71.5%	74.4%
	Poor Efficiency Air Cooled	34.7%	58.5%	62.5%	64.6%	67.0%	69.9%

Table 20. Energy savings results for CC supply air temperatures between 52 and 56 °F (low dehumidification loads).

	Savings	Plant Electricity Only Savings (kWh)	Total Energy Savings by Reheat System Type and Efficiency (MMBTU)				
			Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	84,383	1,712	2,125	2,379	2,718	3,193
	Moderate Efficiency Water Cooled	102,164	1,773	2,185	2,440	2,779	3,254
	Poor Efficiency Water Cooled / High Eff Air Cooled	137,737	1,894	2,307	2,561	2,900	3,375
	Moderate Efficiency Air Cooled	173,298	2,016	2,428	2,682	3,022	3,496
	Poor Efficiency Air Cooled	226,652	2,198	2,610	2,864	3,204	3,678
	Savings %	Plant Electricity Only Savings	Total Energy Savings by Reheat System Type and Efficiency				
			Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	38.3%	78.7%	82.1%	83.7%	85.4%	87.3%
	Moderate Efficiency Water Cooled	37.9%	75.6%	79.3%	81.0%	83.0%	85.1%
	Poor Efficiency Water Cooled / High Eff Air Cooled	37.5%	70.7%	74.6%	76.6%	78.7%	81.2%
	Moderate Efficiency Air Cooled	37.2%	66.9%	70.9%	72.9%	75.2%	77.8%
	Poor Efficiency Air Cooled	37.0%	62.5%	66.5%	68.5%	70.9%	73.6%

GHG emissions impacts were calculated using the results of the energy savings analysis (Tables 21 to 23). The electric grid GHG emissions rate was applied from the U.S. Environmental Protection Agency (USEPA) eGrid for the 2014 national average (1150.322 lb. CO_2e/MWh). The natural gas GHG emissions rate is fixed at 53.2 kg CO_2e/MMBtu.

The data show total GHG emissions savings ranging from 290 to 330 tons per year, a reduction of approximately 50 to 80% from the baseline emissions.

Table 21. GHG emissions savings results for CC supply air temperatures below 50 °F (very high dehumidification loads).

			Total GHG Savings by Reheat System Type and Efficiency (tons)				
	Savings	Plant Electricity Only Savings	Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	63	300	169	183	203	230
	Moderate Efficiency Water Cooled	75	313	182	196	216	243
	Poor Efficiency Water Cooled / High Eff Air Cooled	101	338	207	222	242	269
	Moderate Efficiency Air Cooled	127	364	233	248	267	295
	Poor Efficiency Air Cooled	166	403	272	287	306	334
			Total GHG Savings by Reheat System Type and Efficiency				
	Savings %	Plant Electricity Only Savings	Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	32.7%	70.0%	56.7%	58.8%	61.2%	64.2%
	Moderate Efficiency Water Cooled	32.4%	66.5%	53.6%	55.5%	57.8%	60.7%
	Poor Efficiency Water Cooled / High Eff Air Cooled	32.1%	61.2%	49.2%	50.9%	53.0%	55.6%
	Moderate Efficiency Air Cooled	31.8%	57.3%	46.2%	47.7%	49.6%	52.0%
	Poor Efficiency Air Cooled	31.7%	53.0%	43.2%	44.5%	46.1%	48.3%

Table 22. GHG emissions savings results for CC supply air temperatures between 50 and 52 °f (moderate dehumidification loads).

			Total GHG Savings by Reheat System Type and Efficiency (tons)				
	Savings	Plant Electricity Only Savings	Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	58	313	171	187	208	238
	Moderate Efficiency Water Cooled	69	324	183	199	220	250
	Poor Efficiency Water Cooled / High Eff Air Cooled	93	348	207	223	244	274
	Moderate Efficiency Air Cooled	117	372	231	247	268	298
	Poor Efficiency Air Cooled	153	408	267	283	304	334
			Total GHG Savings by Reheat System Type and Efficiency				
	Savings %	Plant Electricity Only Savings	Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	35.8%	75.2%	62.4%	64.5%	66.9%	69.7%
	Moderate Efficiency Water Cooled	35.5%	71.9%	59.2%	61.2%	63.5%	66.4%
	Poor Efficiency Water Cooled / High Eff Air Cooled	35.1%	66.8%	54.5%	56.3%	58.5%	61.3%
	Moderate Efficiency Air Cooled	34.9%	62.9%	51.3%	53.0%	55.0%	57.6%
	Poor Efficiency Air Cooled	34.7%	58.5%	48.0%	49.5%	51.3%	53.6%

Table 23. GHG emissions savings results for CC supply air temperatures between 52 and 56 °f (low dehumidification loads).

			Total GHG Savings by Reheat System Type and Efficiency (tons)				
	Savings	Plant Electricity Only Savings	Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	49	289	156	171	191	219
	Moderate Efficiency Water Cooled	59	299	166	181	201	229
	Poor Efficiency Water Cooled / High Eff Air Cooled	79	319	187	202	221	249
	Moderate Efficiency Air Cooled	100	340	207	222	242	270
	Poor Efficiency Air Cooled	130	370	238	253	273	300
			Total GHG Savings by Reheat System Type and Efficiency				
	Savings %	Plant Electricity Only Savings	Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	38.3%	78.7%	66.6%	68.6%	70.9%	73.6%
	Moderate Efficiency Water Cooled	37.9%	75.6%	63.3%	65.3%	67.6%	70.4%
	Poor Efficiency Water Cooled / High Eff Air Cooled	37.5%	70.7%	58.6%	60.4%	62.6%	65.4%
	Moderate Efficiency Air Cooled	37.2%	66.9%	55.2%	56.9%	59.0%	61.6%
	Poor Efficiency Air Cooled	37.0%	62.5%	51.7%	53.3%	55.1%	57.5%

Finally, a lifecycle cost analysis was performed across a selected combination of scenarios representing the most common field conditions and electricity rates for DoD sites. Table 24 lists the low, mid, and high scenarios analyzed for electricity and natural gas rates.

Table 24. Low, mid, and high scenarios.

Scenario	Electricity Rate $/kWh	Natural Gas Rate $/therm
Low	0.08	0.5
Mid	0.14	0.8
High	0.20	1.1

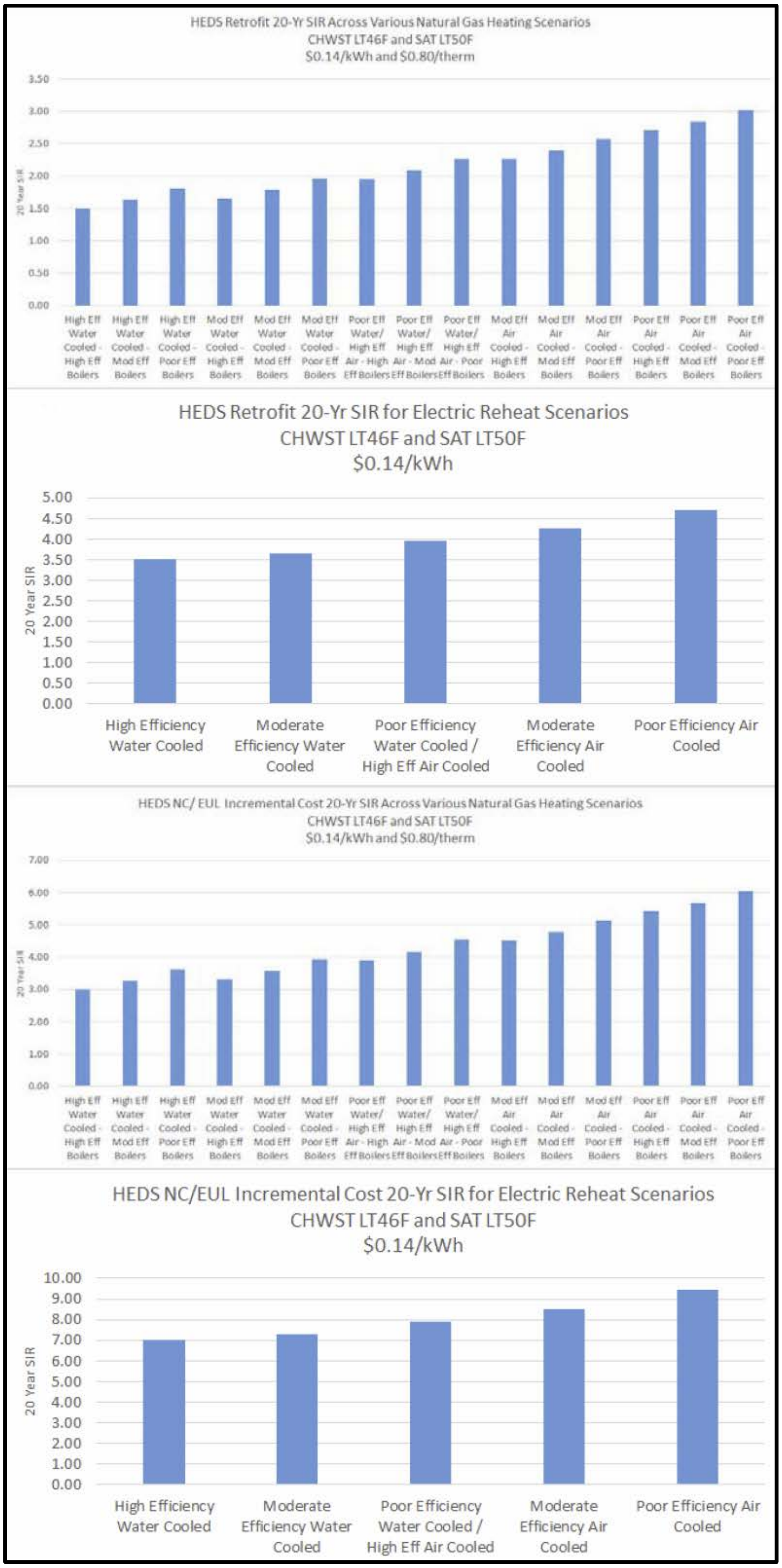
Additionally, two capital cost values were used in the analysis. A retrofit cost value of $16/CFM was used; a retrofit scenario assumes the project bears the entire cost of the HEDS unit and installation, such as equipment replacement before end of useful life (EUL). An incremental cost value of $8/CFM was also used. Incremental cost values would apply in new construction, major renovation, and equipment EUL situations; incremental cost values also represent retrofit cases where the AHU can be rebuilt in place without entire unit replacement. Note that in new construction applications, HEDS can significantly reduce other infrastructure costs due to chiller and piping downsizing, cooling tower downsizing, etc. due to the cooling load reductions; these cost savings were not included in the lifecycle cost analysis presented here, but should be considered where possible.

Note that, where energy codes require that simultaneous heating and cooling cannot be used for RH control (such as ASHRAE 90.1), the incremental cost of HEDS would approach $0/CFM, and could even have significant cost savings, depending on the comparative technology used as baseline.

The charts shown in Figure 34 highlight the lifecycle performance across a range of scenarios. For the purposes of this work, savings for the temperature bin for high dehumidification loads (cooling coil supply air temperature less than 50 °F), and savings for mid-level electricity and natural gas are presented. Other temperatures bins have very similar results. Results are shown as a 20-year savings to investment ratio (SIR), where the total savings over 20 years are divided by the project costs.

The results show SIRs above 1 across all scenarios, reaching over 4 for the retrofit applications and over 9 for new construction, EUL, and major renovation applications. Even at the low-level electricity and natural gas rates, SIRs are above 1 in almost all scenarios and reach as high as 2.7.

Figure 34. Lifecycle performance across a range of scenarios.

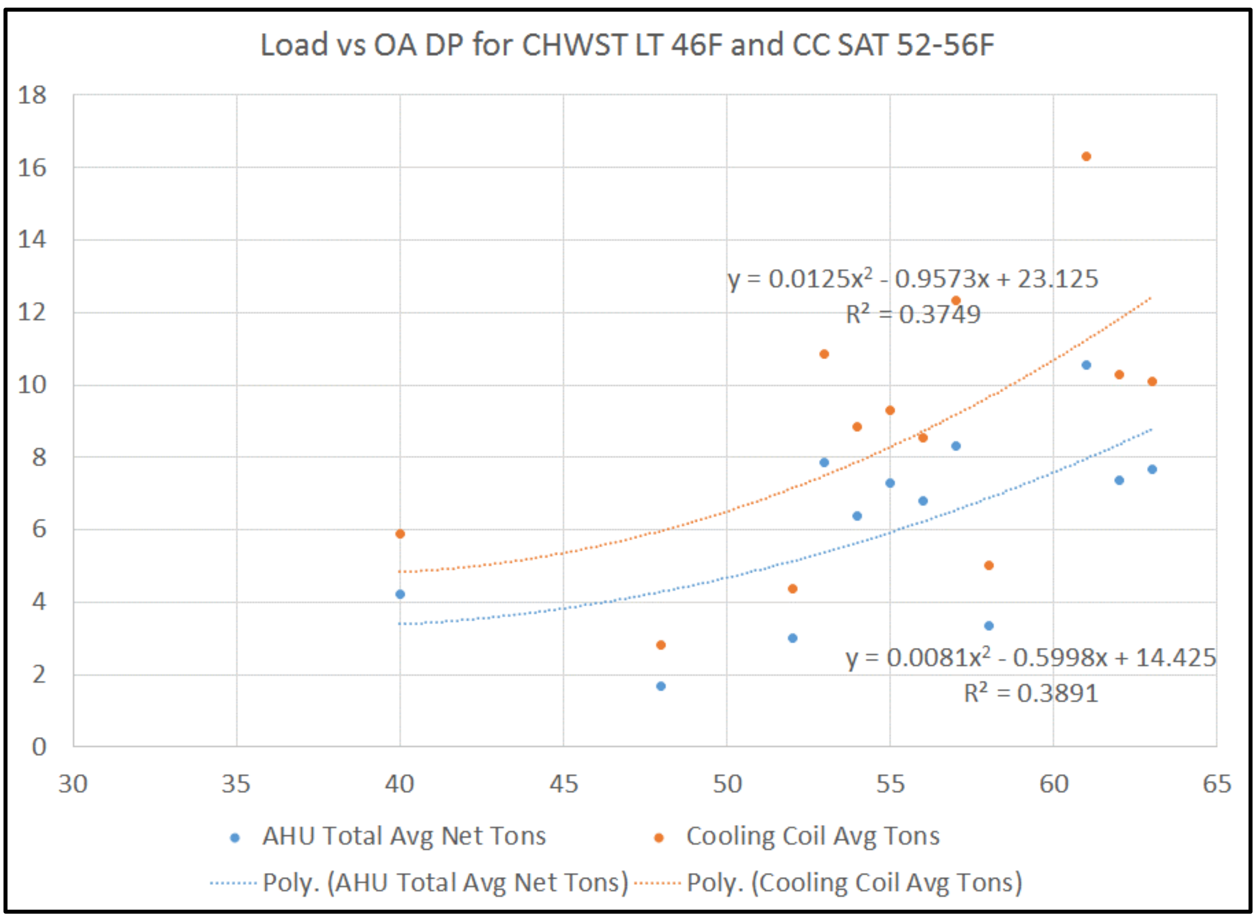


7.3.2 Tinker AFB

Translating the cooling and dehumidification-related reheat load savings to energy savings requires analysis of the central plant operations serving the HEDS units. Given that load reductions have nonlinear impacts on energy use for most chilled water systems, a spreadsheet model was developed to explore potential HEDS energy, GHG, and lifecycle cost savings across a range of chilled water and heating plant efficiencies and types.

First, a model of the baseline load and AHU net load with HEDS was developed as a function of average daily outdoor air dewpoint temperatures. Figure 35 shows the resulting modeled performance for the 52 to 56 °F cooling coil supply air temperature bin. Note that this bin was used for the energy, GHG and lifecycle savings analysis due to limited data sets in other temperature ranges (due to chiller system issues as noted earlier in the report).

Figure 35. Characteristic load curves for the baseline and net HEDS cooling loads as a function of average daily outdoor air temperature for Tinker AFB.



Note that the load model correlations are much weaker for the Tinker AFB data sets. This is due to the lower amount of data available, and to the more dynamic nature of the VAV system, which acts to decouple the observed loads from the outside air dewpoint temperature.

The modeled curves were then applied to typical meteorological year (TMY3) dewpoint data to develop annualized savings estimates for each cooling coil temperature range. An annual schedule was applied to chiller plant operations such that the savings analysis only occurs from May through October, reflecting the annual wintertime shutdown from approximately November through April. Also, to appropriately determine total plant energy impacts, it was assumed that HEDS units would be used on the entire load for each chiller plant (i.e., all AHUs would be replaced with HEDS units). This simplifies the analysis of total energy impacts compared with partial HEDS implementations.

Note that runtime schedule savings are a significant component of the Tinker deployment for HEDS. Due to limitations in dehumidification capacity associated with the existing unit that was replaced, the baseline system operated 24x7 throughout the dehumidification season. Because of the increased cooling and dehumidification capacity associated with the HEDS unit, the schedule was able to be reduced by approximately 10 hours per day during the week and by over 40 hours on the weekend, resulting in an average weekly runtime reduction of nearly 90 hours.

As discussed, energy, GHG, and lifecycle cost impacts were calculated across a range of chiller and boiler plant efficiency scenarios:

<u>Chilled Water Plant Scenarios:</u>

- High Efficiency Water Cooled Chillers: IPLV 0.45
- Moderate Efficiency Water Cooled Chillers: IPLV 0.56
- Poor Efficiency Water Cooled / High Efficiency Air-Cooled Chillers: IPLV 0.79
- Moderate Efficiency Air-Cooled Chillers: IPLV 1.01
- Poor Efficiency Air-Cooled Chillers: IPLV 1.35.

<u>Heating Plant Scenarios (net efficiencies include boiler, cycling, and distribution losses):</u>

- Electric Resistance: 100% net delivery efficiency
- High Efficiency Condensing Boilers: 80% net delivery efficiency

- High Efficiency Non-Condensing Boilers: 70% net delivery efficiency
- Moderate Efficiency Non-Condensing Boilers: 60% net delivery efficiency
- Poor Efficiency Non-Condensing Boilers: 50% net delivery efficiency.

Table 25 lists the resulting energy savings, both in total energy and savings percent, for each cooling coil supply air temperature bin. Note that total energy savings associated with cooling and dehumidification-related re-heat are greater than 70% in all cases, and reach as high as 91% with the combination of a high efficiency chilled water plant and very low efficiency boiler system.

Table 25. Energy savings results for CC supply air temperatures between 52 to 56 °f (office dehumidification loads).

			Total Energy Savings by Reheat System Type and Efficiency (MMBTU)				
	Savings	Plant Electricity Only Savings (kWh)	Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	98,101	1,087	1,331	1,465	1,644	1,895
	Moderate Efficiency Water Cooled	116,475	1,149	1,393	1,528	1,707	1,957
	Poor Efficiency Water Cooled / High Eff Air Cooled	153,237	1,275	1,519	1,653	1,832	2,083
	Moderate Efficiency Air Cooled	189,986	1,400	1,644	1,778	1,957	2,208
	Poor Efficiency Air Cooled	245,122	1,588	1,832	1,967	2,146	2,396

			Total Energy Savings by Reheat System Type and Efficiency				
	Savings %	Plant Electricity Only Savings	Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	64.9%	85.7%	88.0%	89.0%	90.1%	91.3%
	Moderate Efficiency Water Cooled	64.4%	84.0%	86.4%	87.4%	88.6%	89.9%
	Poor Efficiency Water Cooled / High Eff Air Cooled	63.9%	81.2%	83.7%	84.8%	86.1%	87.6%
	Moderate Efficiency Air Cooled	63.6%	79.0%	81.6%	82.7%	84.1%	85.6%
	Poor Efficiency Air Cooled	63.3%	76.6%	79.1%	80.2%	81.6%	83.2%

Using the results of the energy savings analysis, GHG emissions impacts were calculated (Table 26). The electric grid GHG emissions rate was applied from the USEPA eGrid for the 2014 national average (1150.322 lb. CO_2e/MWh). The natural gas GHG emissions rate is fixed at 53.2 kg CO_2e/MMBtu.

Table 26. GHG emissions savings results for CC supply air temperatures between 52 to 56 °f (low dehumidification loads).

			Total GHG Savings by Reheat System Type and Efficiency (tons)				
	Savings	Plant Electricity Only Savings	Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	56	183	115	123	133	148
	Moderate Efficiency Water Cooled	67	194	125	133	144	158
	Poor Efficiency Water Cooled / High Eff Air Cooled	88	215	146	154	165	179
	Moderate Efficiency Air Cooled	109	236	168	175	186	201
	Poor Efficiency Air Cooled	141	268	199	207	218	232

			Total GHG Savings by Reheat System Type and Efficiency				
	Savings %	Plant Electricity Only Savings	Electric	High Efficiency Condensing Boilers	High Efficiency Non-Condensing Boilers	Moderate Efficiency Non-Condensing Boilers	Poor Efficiency Non-Condensing Boilers
Chiller System Type and Efficiency	High Efficiency Water Cooled	64.9%	85.7%	79.0%	80.0%	81.3%	82.9%
	Moderate Efficiency Water Cooled	64.4%	84.0%	77.2%	78.3%	79.5%	81.1%
	Poor Efficiency Water Cooled / High Eff Air Cooled	63.9%	81.2%	74.6%	75.6%	76.8%	78.3%
	Moderate Efficiency Air Cooled	63.6%	79.0%	72.8%	73.7%	74.8%	76.2%
	Poor Efficiency Air Cooled	63.3%	76.6%	70.9%	71.7%	72.7%	74.0%

The data show total GHG emissions savings ranging from 115 to 270 tons per year, a reduction of approximately 70 to 85% from the baseline emissions.

Finally, lifecycle cost analysis was performed across a selected combination of scenarios representing the most common field conditions for DoD sites. Low, mid, and high scenarios for electricity and natural gas rates were analyzed (Table 27).

Table 27. Lifecycle cost analysis

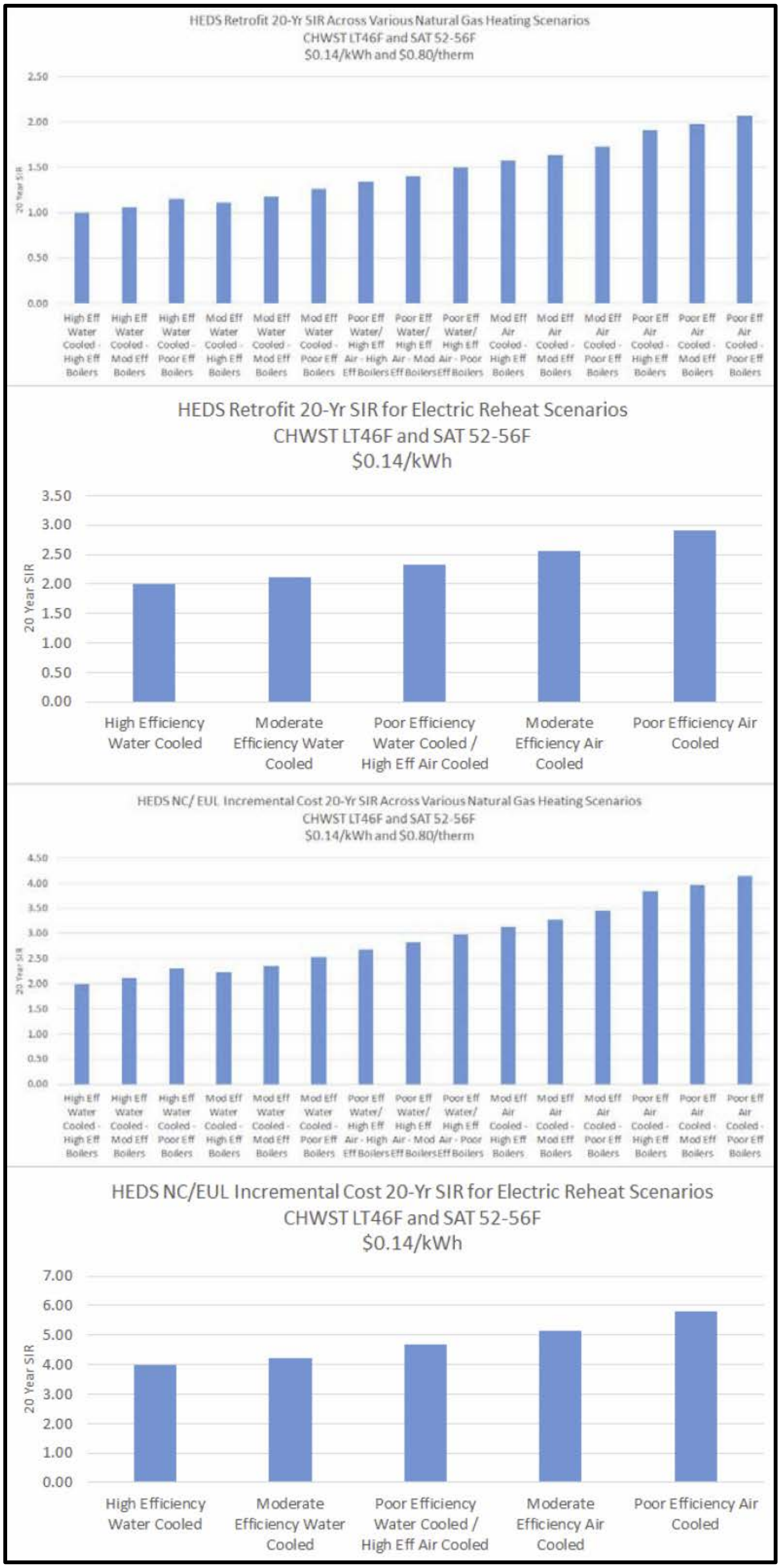
Scenario	Electricity Rate $/kWh	Natural Gas Rate $/therm
Low	0.08	0.5
Mid	0.14	0.8
High	0.20	1.1

Additionally, two capital cost values were used in the analysis. A retrofit cost value of $16/CFM was used; a retrofit scenario assumes the project bears the entire cost of the HEDS unit and installation, such as equipment replacement before EUL. An incremental cost value of $8/CFM was also used. Incremental cost values would apply in new construction, major renovation, and equipment EUL situations; incremental cost values also represent retrofit cases where the AHU can be rebuilt in place without entire unit replacement. Note that in new construction applications, HEDS can significantly reduce other infrastructure costs due to chiller and piping downsizing, cooling tower downsizing, etc. due to the cooling load reductions; these cost savings were not included in the lifecycle cost analysis presented here, but should be considered where possible.

Note that, since energy codes require that simultaneous heating and cooling cannot be used for RH control (such as ASHRAE 90.1), the incremental cost of HEDS would approach $0/CFM, and could even have significant cost savings, depending on the technology used as baseline.

The charts shown in Figure 36 highlight the lifecycle performance across a range of scenarios. Results are shown as a 20-year savings to investment ratio (SIR), where the total savings over 20 years are divided by the project costs. The results show that SIRs are above 1 across all scenarios, nearly 3 for the retrofit applications, and nearly 6 for new construction, EUL, and major renovation applications. Generally, lifecycle cost performance is lower for the Tinker installation due to lower overall cooling loads associated with the VAV system in an administrative building versus the constant volume system in a kitchen at Fort Bragg.

Figure 36. Lifecycle performance across a range of scenarios.



8 Implementation Issues

8.1 Procurement issues

Currently, HEDS units are only available under license with one manufacturer, which can limit procurement options. The current plan is to evaluate several different manufacturers to determine their ability to meet the expected quality and support levels, and to license the technology to at least two more manufacturers. Given that few AHUs are commercial off-the-shelf items (they are mostly built to order), HEDS units will need to use the same market channels of mechanical product vendors, installers, and AHU manufacturers to achieve market scale. This will require deep engineering support from the vendor networks, which requires training, education, and experience with HEDS systems.

8.2 Potential barriers to acceptance

One major barrier to acceptance is market skepticism with new technologies that claim such high savings levels. It is common to encounter situations where there is a great potential project, or projects, but the engineer is able to stop the projects by asking “where have you done HEDS in a facility similar to mine, in a climate similar to mine?” More technology demonstration projects in different applications and third party validation are needed to substantiate the savings claims. Additionally, there is often significant pushback within the industry that requires demonstrated performance in similar applications, which slows the adoption of HEDS and any new and potentially market-disruptive technologies.

Insufficient resources to properly operate and maintain HVAC systems on DoD installations is an ongoing concern for public works staff. With limited funding and/or understaffed personnel available to accommodate their existing building stock and associated equipment, installation directorates of public works are often reluctant, unwilling, or unable to work with new technologies that they are unfamiliar with.

The simplicity of the HEDS design and operating strategy should help to overcome public works staff reluctance to embrace a new technology. Documenting and publicizing the implementation of the HEDS ESTCP project at the two demonstration sites should help to encourage further adoption of the technology. Users of the new technology must be confident that it will consistently and reliably save energy, reduce biological growth, reduce lifecycle costs, and improve the comfort of the buildings' occupants over the long term, without increasing their manpower and funding requirements.

8.3 ASHRAE 90.1 Prescriptive energy code requirements

The latest version of ASHRAE 90.1 prescriptive energy codes explicitly disallows any form of simultaneous cooling and heating or reheating of air for RH control, if the heat or reheat is not from a reclaimed or solar-thermal source. HEDS is one of the few HVAC system designs that is compliant with ASHRAE 90.1 prescriptive energy code regarding RH control, as it uses reclaimed energy for the reheat energy source.

The vast majority of HVAC systems in Federal facilities do not comply with the latest versions of ASHRAE 90.1 (90.1-2007, -2010, -2013 and -2016) with respect to RH Control. HEDS may be the most cost effective solution for DoD and also for the tens of thousands of Federal office buildings, embassies and consulates in humid climates to reduce energy and water waste; improve comfort, health and wellness; and to comply with ASHRAE.

When HVAC systems must be replaced, repaired or upgraded, HEDS may be the only cost effective solution to provide ASHRAE 90.1 compliance across a broad range of HVAC system sizes and types, given some of the following attributes

- HEDS can be a cost effective ASHRAE 90.1 RH control solution that can be applied for systems ranging from 100 CFM (i.e., barracks), to 1,000,000 + CFM (i.e., aircraft paint hangars, known as corrosion control facilities) and all sizes in between.
- HEDS is the only ASHRAE 90.1 compliant solution that will physically fit in many of the existing AHU, DOAS, RTU, Packaged Terminal Air Conditioner (PTAC) and FCU locations.
- HEDS maintenance requirements are lower than any other RH control option.

8.4 Lessons learned

In addition to validating the key performance objectives, several key lessons were gleaned from the demonstration project, as described below.

- Chilled water plant performance can significantly impact HEDS performance. Both test sites experienced chiller plant failures and capacity limitations that resulted in very high and unstable chilled water temperature control. Since dehumidification is limited by the chilled water temperature entering the AHU, this impacts any AHU's ability to provide dehumidification. However, even under these conditions HEDS was able to provide more dehumidification and reduce total cooling loads compared with a traditional dehumidification-reheat AHU.
- Accessibility of operating staff and maintenance data on DoD sites can be challenging. Given the significant turnover and lack of documentation of maintenance practices, quantifying non-energy impacts of systems demonstrations can be difficult on military bases.
- Other system operating constraints may limit overall HEDS impact. At Fort Bragg, the three other AHUs serving the DFAC had capacity limitations and operating issues that limited the ability to realize additional savings from reducing the runtimes of the building. If all units had been replaced with HEDS units, it is expected that the operating times of the equipment could be better aligned with the actual occupied hours of the facility, instead of running 24x7 as is currently required. For the Tinker AFB demonstration site, the runtimes were able to be reduced for the HEDS unit compared to the previous unit operation.

8.5 Future potential HEDS applications for DoD

The overall objective is to position the HEDS technology for immediate and widespread commercialization and adoption in DoD facilities and floating assets. Target facilities would have some combination of the following conditions

- Facilities that are mandated to reduce energy and water use
- Facilities that must reduce thermal and or electrical loads and costs
- Facilities that must comply with ASHRAE 90.1 prescriptive energy codes that do not allow simultaneous heating and cooling for RH control
- Facilities that use chilled water from a chiller plant as their source of cooling

- Facilities that have large Direct Expansion (DX) RTUs can be candidates if they are converted to chilled water coil systems
- Facilities where the climate is humid at least 4 months per year, or facilities in many milder climates that need 48 to 50 °F dewpoint supply air conditions, (e.g., hospitals and semiconductor fabrication facilities in Southern California)
- Facilities that must comply with the UFC indoor dewpoint temperature requirement of no greater than 55 °F dewpoint
- Facilities that do not have a cost effective source of reheat thermal energy for use in the cooling-dehumidification-reheat process
- Facilities that are currently experiencing, or have previously experienced unwanted biological growth
- Facilities where cooling loads may have met or exceeded the available chiller plant cooling capacity
- Facilities where chilled water distribution system may be at, or above, its capacity limit for current or planned loads
- Facilities where heating hot water distribution system may be at its capacity limit for current or planned loads
- Facilities that operate outside of UFC comfort guidelines on a regular basis
- Facilities with water-cooled chiller plants that must reduce cooling tower water use
- Facilities with two-pipe switchover water distribution systems – chilled water in the pipes in the summer, hot water in the pipes in the winter
- Healthcare facilities that are mandated to reduce energy and water use
- Healthcare facilities that have high rates of Hospital Acquired Infections (HAIs)
- Healthcare facilities that struggle to maintain proper temperature and RH setpoints in occupied areas
- Military Sealift Command (MSC) ships of all sorts that operate in hot/humid climates (ship-based applications, which are currently being investigated under award N00167-17-BAA-01 with NSWC Carderock Division)
- Combat vessels that must not have thermal stress on Sailors during engagements.

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