# FINAL REPORT

## Air Source Cold Climate Heat Pump

## ESTCP Project EW-201136

AUGUST 2013

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# REPORT DOCUMENTATION PAGE

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**14. ABSTRACT**

Executive Summary
A university/industry team, Purdue University, Trane, Emerson Climate Technologies, Danfoss, and Automated Logic Corporation of Indiana, demonstrated a new air-source heat pump technology that was optimized for colder climates. The technology has significant potential to reduce the primary energy used for heating small commercial or residential buildings and expand the range of air-source heat pumps to Department of Defense (DoD) facilities in the northern half of the U.S. Cold Climate Heat Pumps (CCHP) are less expensive to operate than an electric furnace and are cost competitive with fossil fuel sources of heat, even though the cost for natural gas is very low at this point in time. CCHP technology also has the potential for reducing greenhouse gas emissions because they are powered by electricity that could come from renewable energy.

The field demonstration was conducted at the Camp Atterbury Joint Maneuver Training Center in Edinburgh, IN, a small town located about 1 hour south of Indianapolis. Two barracks were selected for the test because they are typical for the small to medium size buildings encountered on military bases. Each building was approximately 6,000 ft2 and constructed of cinderblocks. Even though the barracks were roughly 50 years old they had recently been updated with insulation, a sheet metal roof, and a modern central HVAC system.

Both buildings had two zones for heating and cooling, which allowed for a direct comparison of CCHP technology to a modern natural gas furnace (NGF). The buildings were modified so that one zone used the cold climate heat pump and the other zone used its original modern central HVAC system. Both zones were instrumented so that energy consumption and comfort could be evaluated using a web-based control platform.

The key finding from this field demonstration was that the CCHP reduced the primary energy for heating by 19% as compared to the NGF. Although the energy savings was substantial, this did not meet the success criteria of 25% that was established at the start of the project. Cost savings and reductions in emissions can be computed directly from the energy savings. The operating cost of the CCHP was approximately the same as the NGF, which also did not meet the success criteria of 15% cost savings that was established at the start of the project. The field demonstration did meet the target for reductions in CO2 emissions by achieving a 19 % reduction as compared to the success criteria of 15%. The project was successful in terms of meeting other performance objectives for comfort, ease of installation, and maintenance.
This was the first full scale field demonstration of this CCHP technology and thus it is not surprising that the university/industry team encountered several challenges during testing. The implementation issues were mostly resolved during the course of the project, but included managing the flow of return oil to the compressors, flooding of the compressors by liquid refrigerant, and maintaining an appropriate level of subcooling at the condenser outlet. The refinement of control algorithms used to manage the operation of multiple compressors, a variable speed drive for the high stage compressor, and expansion valves for modulating refrigerant flow were essential for correcting problems and improving operation.

The field demonstration as a whole was very successful. The CCHP technology is being further developed and commercialized in partnership with Unico, Inc. of St. Louis, MO. The units will be partially decommissioned to allow for future use of the demonstration site. There is significant potential for further improvements in performance and ultimately delivering a new HVAC technology that will help the Department of Defense meet its energy reduction goals.

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**Acronyms**

| | |
|---|---|
| AFUE | Annual Fuel Utilization Efficiency |
| ASHRAE | American Society of Heating, Refrigeration, and Air Conditioning Engineers |
| CAJMTC | Camp Atterbury Joint Maneuver Training Center |
| CCHP | Cold Climate Heat Pump |
| COMP | Compressor |
| COP | Coefficient of Performance |
| DAQ | Data Acquisition |
| dB | Decibel |
| DoD | Department of Defense |
| DoE | Department of Energy |
| ECB | Engineering and Construction Bulletin |
| EPA | Environmental Protection Agency |
| ESTCP | Environmental Security Technology Certification Program |
| fpm | Feet per Minute |
| GWP | Global Warming Potential |
| HASP | Health and Safety Plan |
| HVAC | Heating, Cooling, and Air Conditioning |
| HX | Heat Exchanger |
| IDHX | Indoor Heat Exchanger |
| MSDS | Material Safety Data Sheet |
| ODP | Ozone Depletion Potential |
| ODHX | Outdoor Heat Exchanger |
| NGF | Natural Gas Furnace |
| PI | Principal Investigator |
| PM | Project Manager |
| POC | Point of Contact |
| RPM | Revolutions per Minute |
| SEMS | SERDP and ESTCP Management System |
| SERDP | Strategic Environmental and Development Program |
| UFC | Unified Facilities Criteria |
| VAC | Voltage in Alternating Current |

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# EXECUTIVE SUMMARY

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be partially decommissioned to allow for future use of the demonstration site. There is significant potential for further improvements in performance and ultimately delivering a new HVAC technology that will help the Department of Defense meet its energy reduction goals.

# 1.0 INTRODUCTION

A university/industry team, Purdue University, Trane, Emerson Climate Technologies, Danfoss, and Automated Logic Corporation of Indiana, demonstrated a new air-source heat pump technology that is optimized for colder climates. The technology has significant potential to reduce the primary energy used for heating small commercial or residential buildings and expand the range of air-source heat pumps to Department of Defense (DoD) facilities in the northern half of the U.S.

Table 1-1summarizes the two year project to demonstrate a cold climate heat pump (CCHP) using the tasks that have been established for the management of this project. Design work began in March of 2011 and continued for six months. The CCHP system was fabricated and tested in psychrometric chambers at Purdue University's Ray W. Herrick Laboratories over a 7 month period, ending in December of 2011. The field demonstration at Camp Atterbury in Edinburgh, Indiana is Task E (Monitoring & Refine) and lasted from January, 2012 through April of 2013. Since that time the emphasis has shifted towards commercializing the technology.

**Table 1-1: Gantt Chart of CCHP Field Demonstration.**

| | 2011 | | | | | | | | | | 2012 | | | | | | | | | | | | 2013 | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Tasks Month: | M | A | M | J | J | A | S | O | N | D | J | F | M | A | M | J | J | A | S | O | N | D | J | F | M | A | M | J | J | A |
| a. Design | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| b. Fabrication | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| c. Experimental Plan | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| d. Installation | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| e. Monitor & Refine | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| f. Commercialization | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| g. Reporting | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |

The field demonstration tested an integrated solution for the heating and cooling of buildings to:

- Reduce the primary energy used for heating small commercial or residential buildings, particularly as compared to traditional heating methods used in the northern states of the U.S.
- Provide a cost effective means of heating in cold climates, particularly where natural gas or propane are not readily available
- Reduce the combustion of gas, oil, and other fossil fuels for heating in cold climates
- Be easily deployed where geologic conditions, space constraints, or other environmental factors preclude the use of geothermal heat pumps

## 1.1 BACKGROUND

The building sector in the U.S. accounted for 41% of the primary energy consumption in 2010 and 40% of the country's carbon dioxide emission in 2009 (2011 Buildings Energy Data Book). The energy challenge is particularly acute for buildings in colder climates that have a longer

heating season. Heating is by far the biggest consumer of energy, accounting for as much as 60% of the energy used in buildings located in cold climates.

The problems with cold climate heating become even more significant when climate change considerations are factored in. The Alliance to Save Energy reports that 63% of households use fossil fuels to heat their homes [2]. All combustion-based heat sources (oil, gas, coal, wood, etc.) add $CO_2$ to the environment. A heat pump uses electricity that could be produced using renewable methods that eliminate or greatly reduce $CO_2$ emissions.

The heating energy can be reduced significantly by improving building efficiency with insulation and air sealing. However, improved technologies that improve heating performance are still needed. Heat pumps have advantages over other heating technologies. They can provide for example, three or more units of heating while using only one unit of power input. Heat pumps can also supply both heating and cooling to a building. This combination reduces the amount of equipment, installation, and maintenance required. Additionally, a heat pump only requires electricity. This enables the utilization of renewable methods to produce electricity that eliminate or greatly reduce CO2 emissions.

Both ground and air-source are two heat pump technologies available to residential and small commercial buildings. However, ground-source heat pumps are unattractive due to their high installation costs from drilling. Typical air-source heat pumps are also undesirable because they become increasingly ineffective as the outdoor temperature continues to drop below 25°F (-4°C). These significant drawbacks result in many military bases choosing other heating technologies over heat pumps despite heat pump advantages.

Current air-source heat pumps installed in DoD facilities often require a supplemental source of heat when ambient conditions are below 25°F (-4°C). The supplementary source of heat is generally supplied by electric resistance, which is highly inefficient at approximately 1/3 the efficiency of the standard air-source heat pump, or a natural gas/propane furnace. Specifically, some buildings in Fort McCoy (located near LaCrosse, WI) have air-source heat pumps installed with a back-up NGF. Michael J. Kelley, the Chief Engineer/Energy-Utilities Branch at Fort McCoy, stated that they operate air-source heat pumps until they cannot reach capacity and then operate the furnace. During the summer months the heat pump cycle is reversed to provide air conditioning. (M. Kelly, personal communication, March 16, 2011).

The need for supplementary heat is not attractive to facility managers because of increased cost and maintenance. Supplementary heating sources also diminish the energy, operating, and $CO_2$ emission savings from utilizing air-source heat pump technologies. Implementing a CCHP wouldn't require a back-up furnace, reducing energy consumption, operating, installation, and maintenance costs of current air-source heat pumps that require a furnace. Unlike traditional air source heat pumps, a CCHP would be able to provide heat during the winter for most hours where the system is called to heat due to its ability to have a modulating output, resulting is less electric resistance heating operation for defrost and maintaining temperature during extreme ambient conditions. The benefits include lower operating costs, energy consumption, $CO_2$ emissions and that the cycle can reverse to provide cooling.

## 1.2 OBJECTIVE OF THE DEMONSTRATION

The overall goal of the project was to demonstrate a CCHP that will reduce the energy and costs for heating buildings in the northern half of the United States. More specifically, the project tested whether a CCHP can function efficiently at temperatures below 25°F (-4°C), where typical heat pumps revert to supplemental heating. The new technology will reduce the need for fossil fuel heating and thus reduce the carbon footprint of military bases. Heating with a CCHP is substantially less expensive than an electric furnace and cost competitive with natural gas and other fossil fuel heating sources. The CCHP is expected to achieve these goals while providing thermal comfort for human occupancy and DoD activities.

Current DoD practices and standards may need to be changed, specifically in the northern U.S. where many DoD installations are only equipped with a NGF to condition a zone. A possible outcome of this demonstration would allow electric powered CCHPs to become general DoD practice when selecting heating/cooling equipment.

Unified Facilities Criteria (UFC) are building codes tailored to the DoD. Sections of the Mechanical Series of the UFC need to be updated to allow more energy efficient methods and technologies. For example, the current UFC recommends oversizing HVAC equipment to ensure adequate heating/cooling capacity but that approach may limit the efficiency of the equipment at off-peak conditions. During these conditions, the CCHP will need to cycle if its lowest capacity is larger than the required load. The cycling reduces the efficiency by increasing the amount of time with losses during start-up, however, if the system has the ability to vary its output with equipment staging, these cycling losses can be minimized. One key benefit for sizing the CCHP system to the peak load is the reduced need for electric resistance heat backup. Also the benefits of having an electric based heating system are it can rely on only renewable electric energy such as photovoltaic or wind turbines. Another possibility is to add a specific section in the Mechanical Series of the UFC that relates solely to CCHPs.

## 1.3 REGULATORY DRIVERS

The development and deployment of a CCHP supports a number of mandates to improve the efficiency of federal buildings, including buildings operated by the Department of Defense:

- Executive Order 13514 “Federal Leadership in Environmental, Energy, and Economic Performance” that was signed by President Obama in October of 2009. This is the latest of several Executive Orders that include mandatory energy reductions for federal buildings. One overarching goal is achieving net-zero-energy buildings by 2030. This effort is largely managed by GSA’s Office of Federal High-Performance Green Buildings.
- The CCHP supports goals of the Energy Independence and Security Act of 2007. Section 315 specifically discusses “Improved Energy Efficiency for Appliances and Buildings in Cold Climates”. This section calls for improved efficiency of mechanical systems as well as an increase of renewable resource usage. Current heating technologies in cold climates are challenged to operate from renewable resources. A CCHP is a versatile technology and would be able to operate off of electricity generated by wind, hydro, and solar renewable energy.

- The Board on Infrastructure and the Constructed Environment (BICE) of the National Academies studied the possibility of high performance federal facilities. Their report noted the abundance of opportunities to increase the performance of existing facilities. CCHPs would be one possible strategy for helping achieve high performance federal buildings.
- After federal energy legislation was passed in 2005 and 2007, the DoD developed its' own Energy Security Initiatives. One component of the strategic plan is to create more efficient facilities [3]. The Department of Defense Energy Security Initiatives stated that 25% of the energy used by the DoD is consumed by buildings. This mandates that DoD installations reduce their energy consumption by 3% per year through 2015. A fully commercialized and widely deployed CCHP could help contribute to that goal.

# 2.0 TECHNOLOGY DESCRIPTION

A technology assessment conducted in 2002 for the U.S. Department of Energy's Building Technologies Program identified "cold climate heat pumps" as one of the top technologies with the potential for significantly reducing primary energy use in buildings. The report acknowledges that CCHP's are not the cheapest option in all situations. Heating using fossil fuels can sometimes be less expensive. The argument for CCHPs is easy to make when fossil fuel alternatives are not readily available. In these situations, a CCHP is clearly superior to a system that uses electric resistance for heat. The case for CCHP's becomes even more compelling when the negative impacts of increased global warming emissions due to the combustion of fossil fuels for heating are considered.

## 2.1 TECHNOLOGY OVERVIEW

An air source heat pump uses electricity to drive a vapor compression cycle to provide efficient heating or cooling to a building. The limitation of conventional heat pump designs is that the operational efficiency (COP) of a conventional single stage heat pump declines at ambient temperatures below -4°C (25 °F). Reducing the heating COP below a value of 2 means the heat pump is producing less than two units of heating per one unit of electricity input. As a reference, electric resistance heaters have a COP of 1 at best, providing one unit of heat for every unit of electricity. In addition, at -4°C conventional heat pumps can't produce 10kW of heating and thus in many applications could not provide a comfortable environment. Therefore, there is great potential to expand the use of heat pumps if they can be designed for use in colder climates.

A CCHP differs from conventional heat pump designs with the addition of a compressor and a heat exchanger (economizer) as shown in Figure 2-1. It has two compressors connected in series to pressurize a low pressure superheated, SH, vapor refrigerant to a high pressure, and thus reaching a high temperature. The return air from the building absorbs energy in the form of heat from the highly pressurized refrigerant and then condenses the refrigerant to a subcooled liquid. This subcooled liquid is then separated into two streams. One stream of the liquid refrigerant enters the economizer to be subcooled further (explained later) and afterwards undergoes a rapid pressure drop via an expansion value causing a decrease in the temperature of the refrigerant.

After leaving the expansion valve the low temperature refrigerant enters the outdoor heat exchanger. The outdoor heat exchanger causes the refrigerant to absorb heat from the relatively warmer outdoor air and evaporate until reaching a superheated vapor. The low pressure refrigerant, superheated vapor is then sucked into the compressor to be compressed and the cycle repeats. The second stream of the liquid refrigerant previously noted, goes through a secondary expansion value before entering the other side of the economizer.

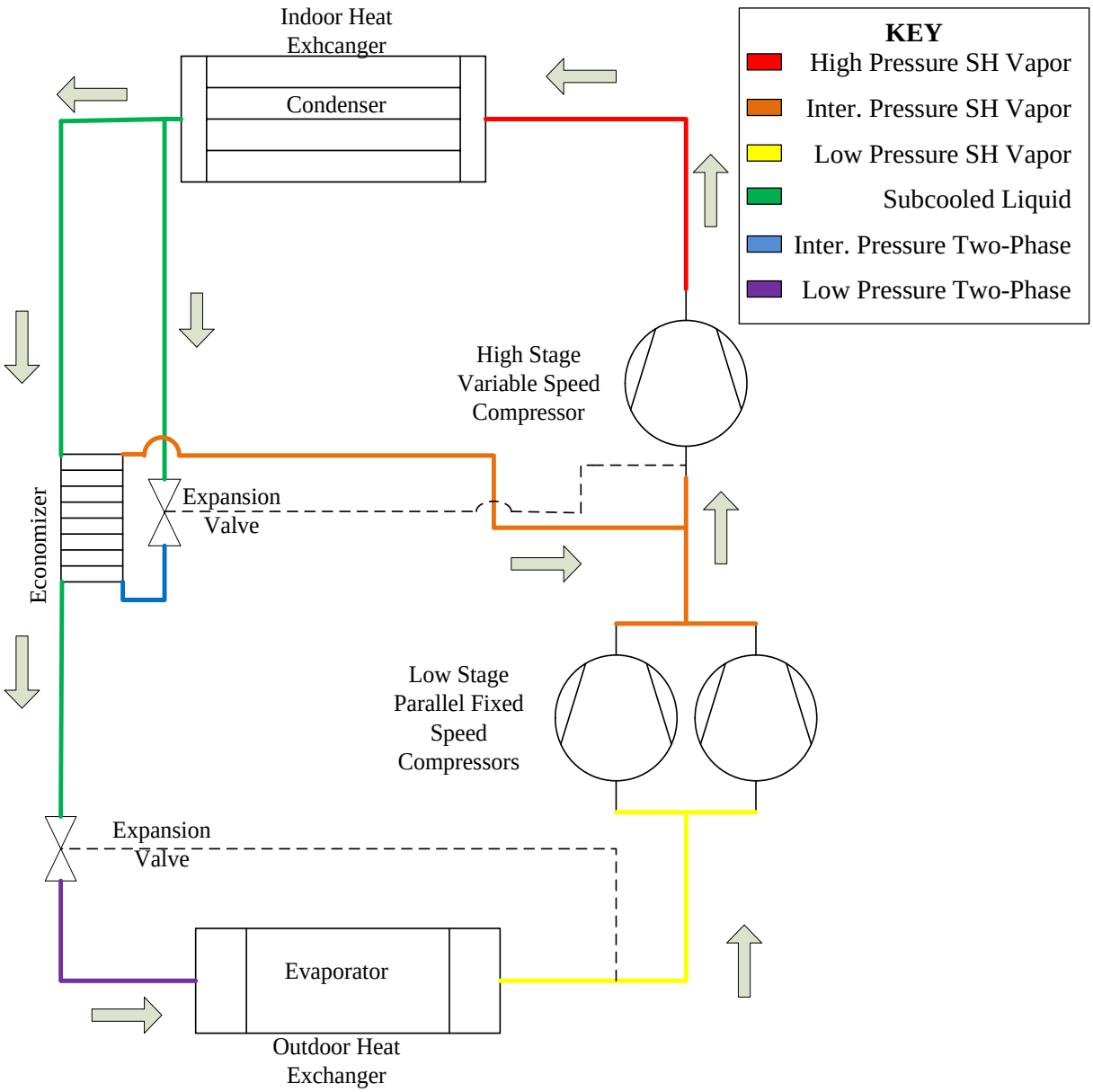


**Figure 2-1: 2-Stage Cold Climate Heat Pump with Economizing.**

The second stream is now able to provide additional subcooling due to its lower temperature relative to the refrigerant on the other side of the economizer. The cooling of the other stream leads to the heating of this refrigerant stream. The heated refrigerant is injected into a mixing chamber between the two compressors. The injection leads to a gain in performance of the system though the cooling of the discharge vapor of the low stage compressor.

Implementing a four-way valve simply switches the condensing and evaporation locations. Heat pumps are equipped with this valve so that the indoor heat exchanger can provide heating and air conditioning. During the heating mode as described above, the condensing heat exchanger is inside while evaporation occurs outside. The simple change of the condensing/evaporating locations allows the heat pump to provide air conditioning or heating to the building.

Purdue University has been involved in the development and testing of CCHP technology for approximately 10 years. A prototype air-source two-stage heat pump that was optimized for heating loads in colder climates was constructed and successfully tested in the Ray W. Herrick Laboratories at Purdue University in West Lafayette, IN starting in 2004. The equipment was tested in psychrometric chambers for ambient temperatures as low as -27 °C (-17 °F). In 2010 two U.S. patents (#7,654,104 and #7,810,353) were issued for the innovative control strategy and mechanical platform. This prior work was the basis for the 2010 proposal to the ESTCP.

## 2.2 TECHNOLOGY DEVELOPMENT

Early in the project a building energy model was created to establish the building's heating and cooling demand, and thus the overall required capacity of the CCHP. Several types of models with varying levels of sophistication were ultimately used, but the starting point was a relatively simple building energy model, eQUEST (QUick Energy Simulation Tool), based on DOE-2 from the U.S. Department of Energy. The program uses second generation Typical Meteorological Year (TMY2) weather data and details of the building construction to simulate heating and cooling loads on an hourly basis over an entire year.

The eQUEST model used the rough dimensions and orientation of the building as well as information about construction materials obtained from building plans. Reasonable assumptions were made on the details of the building envelope (infiltration) and building schedules (occupancy and lighting). Since the building in question was a barracks, a usage pattern for a hotel was assumed. Based on this information, the eQUEST model calculated that the maximum building heating load was roughly 65 kBTU/hr (19 kW) at an outdoor temperature of -4°F (-20°C). In order to eliminate the amount of electric resistance heating needed as a back-up, this maximum heating load was selected as the design point for sizing the CCHP.

The outdoor design temperature is not arbitrary, but is an estimate of the coldest weather conditions that can be reasonably expected at Camp Atterbury. This 65 kBTU/hr heating value is significantly smaller than the 100 kBTU/hr (29 kW) rating of the NGF that was already installed in the building, but this type of oversizing is a routine part of military specifications for HVAC equipment in buildings. Additional information on the building energy modeling can be obtained from Menzi et al, 2013.

The next phase of the design was the development of a heat pump model to evaluate and select individual components of the heat pump system. The heat pump simulation is necessary to verify that the CCHP has adequate capacity over the entire heating season for the field demonstration site. Detailed information about the heat transfer and flow characteristics of compressors and heat exchangers were the key elements of this model. This information was loaded into a computer program developed using EES (Engineering Equation Solver).

The governing equations for this computer program were taken from a research paper published by Bertsch and Groll, 2008. The compressors were modeled using the 10-coefficient map, provided by the manufacturer, which adheres to the ANSI/ARI Standard 540, 1999. Two sets of 10-coefficient maps are needed for each simulation run since two compressors are to be used in the CCHP system. A variety of commercially available heat exchangers and compressors were

evaluated in an iterative fashion to identify an optimal CCHP design. Additional information on the heat pump modeling can be obtained from Caskey et al., 2012a and Caskey et al., 2012b.

Figure 2-2 summarizes the performance of the optimal heat pump design during several different CCHP operating modes. Specific operating modes are to match how the building load varies over a range of ambient temperatures. The linearized building load uses two points; one is at the design heating load, the second assumes no heating load when the ambient temperature is equal to the heating set-point, 20°C (68°F).

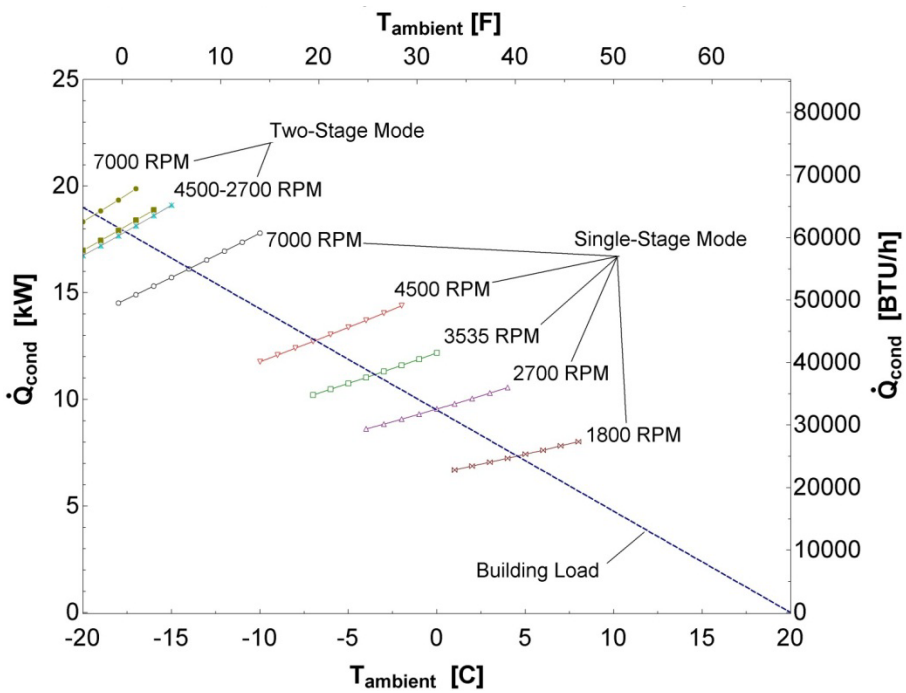


**Figure 2-2: EES Results of the CCHP Capacity.**

The different operating modes are achieved by varying the compressor speed and the number of compressors. Single stage (one compressor) mode is satisfactory for meeting the building heating load for a range of warmer outdoor temperatures by varying its speed. Two stage (two compressor) mode becomes necessary at ambient temperatures below about 7°F (-14°C).

Figure 2-3 estimates the COP of the CCHP for each operating mode and compares this with the total amount of hours at each outdoor temperature for the test site. The weather data used was from the third generation Typical Meteorological Year (TMY3) weather database. The results show that COP's greater than 2 are expected even in the coldest of outdoor conditions. A COP of 2.5 is predicted during the coldest weather expected at Camp Atterbury while meeting the full building load while using no supplemental electric resistance heating.

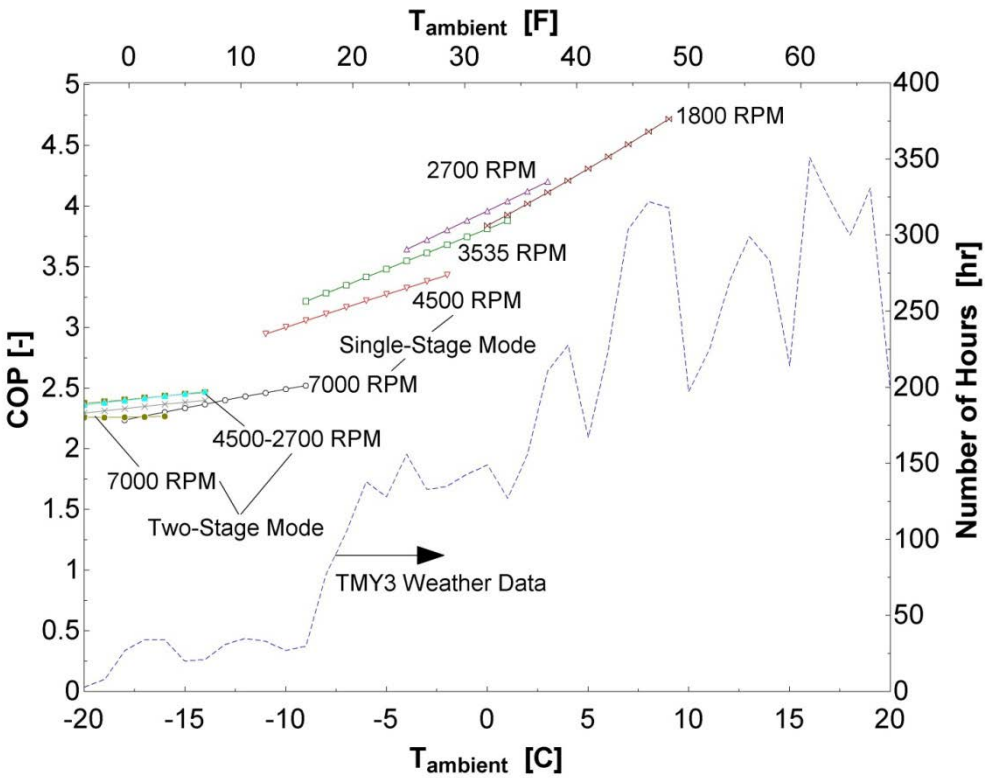


**Figure 2-3: EES Results of the CCHP COP Efficiency.**

## 2.3 ADVANTAGES AND LIMITATIONS OF THE TECHNOLOGY

The CCHP system can outperform all combustion-based heating sources in terms of energy efficiency. The best an oil, gas, or electric furnace can hope to achieve is something less than 100% energy conversion efficiency for the first and second law of thermodynamics. Basic thermodynamics shows that a heat pump cycle will deliver at least two or three times as much heat per unit of energy input. Using low stage and high stage compressors allows the CCHP to achieve an acceptable coefficient of performance of around 3 during air conditioning and mild winters. This technology is driven electrically, so it is also compatible with renewable sources such as solar or wind. Additionally, the CCHP is operational at ambient temperatures far below air-source heat pump technology.

The CCHP should reduce the primary energy use for heating a small commercial building or an individual home by at least 10%. The first cost of this system will be higher than a NGF, but is estimated to yield a payback on the order of 5 to 6 years. The payback would be even shorter in areas where incentives are in place to encourage more heat pump installations. These savings are scalable to many residential and small commercial buildings on military bases in colder climates. The biggest savings and shortest return on investment will occur in situations where other heating options are not feasible. This could occur in locations where combustion-based heating (gas, oil, or propane) is not widely available. There exist other situations where geologic conditions, space constraints, or other environmental factors preclude the use of geothermal heat pumps. In these situations, a cold climate air source heat pump is vastly superior to other alternatives (e.g. electric heating).

A CCHP has the potential for complications since it is a new technology. The possibility exists that one stage of the two stage heat pump can have premature compressor failure if the lubricating oil migrates over time to the other compressor; causing it to run dry and seize. Satisfactory operation can be achieved using additional equipment, such as an oil separator and additional electric valves. Other options include turning off the compressors after a certain runtime for a required oil equalization time. These methods lead either to increased investment costs or a slight decrease of performance.

Another potential concern is freezing up of the outdoor coil during cold weather. This can be accommodated during the design of the outdoor coil. Designing around the drainage of water can reduce the accumulation of frost by optimizing the heat exchanger geometry and the air flow over the evaporator. Once frosting of the coils occurs, silent defrosting could be implemented, closing one or two circuits of the heat exchanger at a time. Closing the circuits allows an increase in temperature on the surface of the respective circuit and in turn melts the accumulated frost. Use of this technique will increase performance of the CCHP. It should be noted that currently only one expansion valve on the market can perform silent defrosting. Electric resistance heating is also an option in extreme situations.

The CCHP utilizes multiple compressors, economizing, and a control strategy that is not found with conventional heat pumps. Due to the extra equipment required in the CCHP design the initial cost of the CCHP will be higher than most heating methods. The additional components create a potential to have higher maintenance costs through degradation and failure. One possible tradeoff is the elimination of oil management equipment to reduce the initial cost. The reduced initial cost would be a tradeoff by reducing performance resulting from the need of shut down to prevent compressor failure due to oil migration. Additional considerations would need to be made with this elimination. The projected savings from operating a CCHP over other heating equipment is expected to overcome these cost limitations.

New technologies have the potential to require an acceptance period for maintenance staff and facility managers. The CCHP controls should not present any significant barriers to personnel operating the new technology. Simple instructions on start-up and shut-down operations should eliminate this issue. The maintenance of the additional components and controls may require technicians who are performing maintenance and installation on the CCHP to attend training and education prior to corrective actions. The performance of the CCHP needs to be evaluated over many months to validate its' benefits. Management may desire concrete results of expected performance before wide deployment.

## 3.0 PERFORMANCE OBJECTIVES

Six performance objectives for the CCHP field demonstration are listed in Table 3-1. Both quantitative and qualitative performance objectives are listed in the far left column. The metrics, data requirements, and success criteria for each performance objective are listed left to right in subsequent columns. The far right column includes the results achieved during the field demonstration. The following sections discuss each performance objective in greater detail.

**Table 3-1: Performance Objectives.**

| Performance Objective | Metric | Data Requirements | Success Criteria | Results |
|---|---|---|---|---|
| **Quantitative Performance Objectives** | | | | |
| 1. Reduce primary energy for heating *(Energy)* | Natural Gas (SCF) or Electricity (kWh) | Electric and gas use metered | Reduce primary energy use by 25% | 19% |
| 2. Reduce costs *(Finances)* | Heating Energy Costs ($) | Base rates for electricity and fuel | 10% reduction in heating costs | (1%) |
| 3. Reduce emissions *(Environment)* | Metric ton $CO_2$ equivalent | Conversions for fuels | Reduce $CO_2$ emissions by 15% | 19% |
| 4. Comfort | Maintain temperature within comfort range of building occupants | Indoor temperature readings and survey of occupants | Compliance with ASHRAE 55 | Yes |
| **Qualitative Performance Objectives** | | | | |
| 5. Ease of installation | Ability of a technician-level individual to install the heat pump | Feedback from the technicians on installation time | A field technician team is able to install the system | Yes |
| 6. Maintenance | Ability of a technician-level individual to maintain the heat pump | Feedback from the technicians on maintenance calls | A field technician team is able to operate the system | Yes |

## 3.1 PRIMARY ENERGY

One goal of the CCHP demonstration was to demonstrate a new technology with the potential to reduce the energy required to heat a building. The energy performance objective compared the energy consumption of the CCHP to traditional NGF. The energy consumption of two similar zones was measured to achieve this comparison.

Each building was already divided into two zones for heating and cooling, which provided an ideal situation for this field demonstration. One building zone was the "control" and used traditional heating methods. The other building zone was the "experiment" used to test the CCHP. Two buildings were used to achieve the necessary redundancy.

The NGF consumed two forms of energy. Electricity was used for its supply fan and natural gas was used for heating. Both energy terms were measured independently and then added to represent the total energy for operating the NGF. An electric meter (Figure 3-1 left) measured the kW-hr consumed by supply fan. A gas meter (Figure 3-1 right) determined how many standard cubic feet (SCF) of natural gas were used. The same electric meter was used to measure the electricity consumption of the CCHP.

**Figure 3-1: Electric (left) and Gas Meter (right).**

Energy conversions were used to compare the performance of two heating technologies (NGF and electric air-source heat pump) on a consistent basis. All forms of energy were converted to kW-hr. As one example, the standard cubic feet (SCF) of natural gas in a NGF was converted into kW-hr as shown in Equation 3-1.

$$SCF\ Natural\ Gas\ to\ kWh = 1\ SCF * \left(\frac{1012\ BTUs}{1\ SCF}\right) * \frac{0.292\ kWh}{3412\ BTUs} \qquad \text{Equation 3-1.}$$

Equation 3-2 shows the basis of the energy reduction computation. The primary energy consumption of the CCHP ($kWh_{CCHP}$) was subtracted from the primary energy consumption of the NGF ($kWh_{NGF}$) then, this value was divided by the energy consumption of the NGF and multiplied by 100 to obtain the percent reduction.

$$CCHP\ Energy\ Reduction\ (\%) = \left(\frac{kWh_{NGF} - kWh_{CCHP}}{kWh_{NGF}}\right) * 100 \qquad \text{Equation 3-2.}$$

The experimental hypothesis was tested as shown below.

$$H_o: CCHP\ Energy\ Reduction \leq 25\%$$

$$H_a: CCHP\ Energy\ Reduction\ > 25\%$$

The null hypothesis is that the CCHP achieves an energy reduction of 25% (or less) when compared to a NGF. The alternative hypothesis is that the CCHP will achieve an energy reduction greater than 25% when compared to a NGF.

As reported in Table 3-1, the measured energy reduction was only 19%. A full description of the calculation procedure can be found in section 6. The null hypothesis was confirmed since energy savings did not achieve the 25% level. Despite not achieving the overall performance objective, the energy reduction was still substantial. It is expected that future versions of this technology will achieve the 25% goal.

## 3.2 COST

The financial performance objective tracked the change in heating costs for the CCHP system compared to a NGF. This objective specifically referred to operating costs and is separate from the maintenance and installation expenses. The purpose of the financial objective was to determine the financial savings from operating a CCHP compared to a furnace. The financial objective also helps determine the payback if a CCHP is selected over a furnace.

The operating costs for a CCHP and NGF were computed from their energy consumption. The CCHP uses only electricity so its costs were calculated by multiplying the real energy consumption (kWh) by the electrical utility rate. The NGF operational costs were calculated by using the same approach except that it had two components. The SCF consumed by the NGF was multiplied by the gas utility's rate and added to the cost of the electricity for operating its' supply fan. This provided the total operating cost of a NGF.

The goal of the financial performance objective was to show that the operating cost of the CCHP was 10% less than a NGF at the demonstration site. The financial performance objective was not met. Table 3-1 shows that energy costs were 1% higher with the CCHP compared to a NGF. There are two main reasons for this occurrence. First, the energy savings (from performance objective 1) were not as high as originally anticipated. Since the energy savings drive the cost savings it is not surprising that the second performance objective was not met. Second, natural gas is currently an inexpensive fuel when compared to electricity, and these results vary significantly depending on the natural gas fuel costs.

Even if a CCHP fails to achieve cost savings as compared to a NGF, there is still a substantial opportunity for cost savings from this technology. The operating cost of a NGF is difficult to beat right now since natural gas prices are at historic lows. The CCHP will fare better in comparison to other technologies, such as electric resistance heating. The CCHP is the best

option in locations where electricity is the only option; such as when fossil fuel sources for heating are not readily available.

## 3.3 $CO_2$ EMISSIONS

The environmental objective determined a reduction in the carbon dioxide ($CO_2$) emissions by deploying a CCHP. Installing an electric heat pump is environmentally beneficial compared to natural gas heating. Heat pumps can be powered by electricity through renewable methods and become emission free. A NGF can never be emission free because it cannot operate off renewable energy. Also, according to the EPA unburned natural gas that is released to the atmosphere has twenty-one times the warming effect of CO2. The purpose of the environmental objective was to determine if the operation of CCHPs are cleaner for the atmosphere than operating NGFs.

The units used in the environmental performance objective were kilograms (kg) of $CO_2$. The CCHP metric was kg of $CO_2$/kW-hr and the furnace used kg of $CO_2$/SCF as well as kg of $CO_2$/kW-hr. Data for the environmental performance objective is obtained from previously obtained data in the energy performance objective. Environmental performance data is obtained indirectly with a greenhouse gas equivalencies calculator. Using references from the Environmental Protection Agency, the energy consumed by each heating method was converted into kg of $CO_2$.

The greenhouse gas equivalency calculation is an estimate but assisted in determining if the environmental objective was achieved. Once this conversion is computed the environmental objective was to be analyzed statistically. Again the procedure had the same hypothesis test and confidence level as previously described. If the CCHP emits 15% less $CO_2$ than a NGF and data is statistically confident then the environmental performance objective was met.

The environmental performance objective was met. One observation is under the assumption all electricity is derived from a natural gas power plant; the CCHP must have reduced the primary energy consumption by 15% to meet the environmental performance objective. The results of the field demonstration show that this percentage was 19%, and therefore enabled the environmental performance objective to be met. Note, that if electric power is produced from “dirtier” fuels then the CCHP must have even greater energy savings.

## 3.4 COMFORT

The comfort performance objective determined a CCHP’s ability to achieve desired indoor conditions throughout the demonstration. The purpose of the comfort performance objective was to determine whether a CCHP can provide a zone that is comfortable for human occupancy. The comfort performance metrics are guidelines found in ASHRAE Standard 55 –“Thermal Environmental Conditions for Human Occupancy”. ASHRAE Standard 55 has been almost universally adopted by state and municipalities as the benchmark for quantifying occupant comfort.

Data was collected using temperature and relative humidity sensors placed outdoors and in the two buildings. This allowed a direct comparison of how well a CCHP is able to maintain the

temperature as compared to a NGF. This comparison made the reasonable assumption that the test and control zones within the same building are being used in a similar fashion.

The variables collected during the demonstration were evaluated according to ASHRAE standard 55. In addition to zone temperature and relative humidity, other variables involved with analysis such as soldier clothing insulation and air speed were estimated. On this basis it was found that the CCHP and the NGF were able to meet acceptable levels of comfort, 80% of CCHP average temperatures lie within and only 50% of high room temperatures are outside the comfort zone.

## 3.5 INSTALLATION

The installation performance objective evaluated the ability of an HVAC contractor's ability to install the CCHP. The CCHP team chose a qualified, Trane and NATE certified technician to perform the installation. The installation objective determined the ease of installation and corrected barriers to the install.

The metric used in the installation objective was feedback obtained from the technician performing the installation. The technician was able to successfully install two CCHPs. Therefore, the installation performance objective was met. This finding was based on two prototype installations, so more data is needed to fully evaluate this objective.

## 3.6 MAINTENANCE

The ability of an HVAC technician to maintain, repair, and operate the CCHPs was included in the maintenance performance objective. The contractor chosen for the installation also performed maintenance on the CCHP. The purpose of the maintenance objective was to determine the ability of a technician to maintain, repair, and operate a CCHP.

The metric for the maintenance objective is feedback given by the technician. The technician was able to successfully maintain each CCHP. Therefore, the maintenance performance objective was met. More installations are needed to fully evaluate this objective. This finding was based on two prototype installations, so more data is needed to fully evaluate this objective.

# 4.0 FACILITY/SITE DESCRIPTION

The CCHP demonstration site was the Camp Atterbury Joint Maneuver Training Center (CAJMTC). It is an Army National Guard base located near Edinburgh, IN, a rural town about 1 hour's drive south of Indianapolis. Constructed in 1941, the 33,000 acre military base has been used as a training center to support the Global War on Terror.

The red star in Figure 4-1shows the location of Camp Atterbury and the Purdue "Flying P" logo indicates the location of Purdue University's flagship. The demonstration site is 100 miles (approximately 2 hours) from Purdue University.

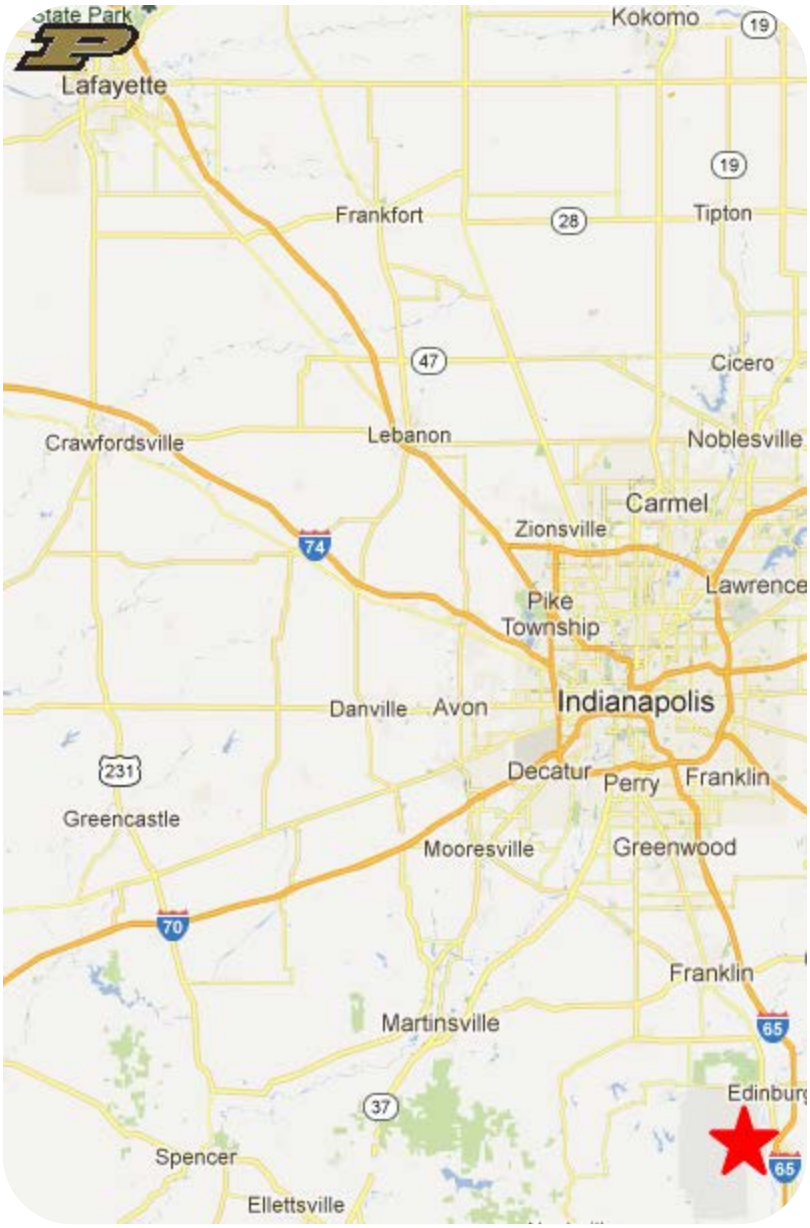


**Figure 4-1: Map between West Lafayette and Camp Atterbury.**

Camp Atterbury has a number of features that made it an ideal location for hosting the CCHP field demonstration:

- Camp Atterbury has many buildings of similar size and construction. These buildings are also similar to what is found on many other military bases.
- Temperatures at Camp Atterbury can be as low as -10°F and as high as 90°F. These varying weather conditions allowed the CCHP to operate through different stages of heating and cooling and provided insight to the robustness of the CCHP components.
- Camp Atterbury is a short 2 hour drive from the main campus of Purdue University in West Lafayette, IN. This was extremely helpful for installing, monitoring, and maintaining equipment used in the field demonstration.
- Camp Atterbury staff were responsive and helpful during visits from the CCHP team.

## 4.1 FACILITY/SITE LOCATION AND OPERATIONS

Camp Atterbury serves as a joint maneuver training center for the Army Reserve and National Guard. However, CAJMTC also provides a training site for Active Duty military personnel. CAJMTC is responsible for mobilizing the National Guard and Army Reserve. Other base operations include providing training to Public Service Organizations. The uncertainty of deployment means CAJMTC was occupied by hundreds or sometimes thousands of military personnel.

The physical facilities staff at Camp Atterbury have been a valuable partner on this CCHP project. They provided easy access to the test site buildings and provided an additional electric panel in the test buildings. This would be an excellent location for Purdue University or the ESTCP program to pursue additional field demonstrations in the future. There have been indications that CAJMTC would like to continue collaboration with Purdue University and the ESTCP program on other research projects.

The data collection for the field demonstration did not require access to the test buildings at Camp Atterbury on a daily basis. The CCHP team monitored the field demonstration from a distance using a web-based monitoring system. The Directorate of Information Management at CAJMTC provided a successful and secure internet connection that allowed remote access. In person visits were made for maintenance, equipment malfunctions, or tours.

Figure 4-2 is an aerial view of buildings 113 and 114 used during the field demonstration. They are two barracks located within 100 feet of each other and can be directly accessed from 1st Street. They both have a north south orientation. Collaboration from the CAJMTC billeting office ensured that the buildings underwent similar levels of occupancy and overall use during the field demonstration.

**Figure 4-2: Aerial View of Buildings 113 & 114.**

Figure 4-3 is an example of the multi-purpose buildings that were chosen for the field demonstration. They are single story buildings constructed of cinderblocks and are approximately 6,000 $ft^2$ . The windows are single-paned with an aluminum frame. The buildings were constructed more than 50 years ago, but have been recently upgraded and now feature modern HVAC systems, tankless water heaters, and sheet metal roofs.

**Figure 4-3: Barracks Multi-Purpose Building at CAJMTC.**

A blueprint of the buildings can be seen in Figure 4-4. The original mechanical rooms consisted of two separate furnaces each with a split DX air conditioner. These are the conventional means

of heating and cooling a building in northern climates. One conventional system supplies comfort to the northern occupied zone and the other to the southern occupied zone. The lavatory has a supply diffuser from each system so; each conventional system provides comfort to the lavatory.

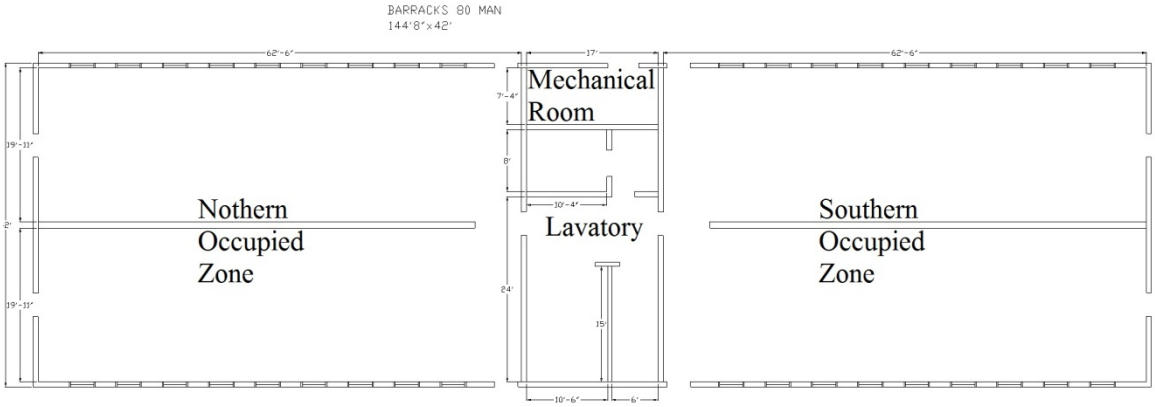


**Figure 4-4: CAJMTC Barrack Blueprint.**

One building with two zones provided a near ideal experiment for comparing two different HVAC systems. The experimental plan resulted in installing a CCHP in a building (Building 113) to provide comfort to the southern occupied zone and installing the second CCHP in another barracks (Building 114) to provide comfort to the northern occupied zone. This enabled the comparative analysis to be performed between two southern zones and two northern occupied zones of two near identical buildings as well as, comparing the north to the south in the same building.

Observations during the field demonstration indicated the military does not operate with energy efficiency as a priority. Multiple times doors were left open (Figure 4-5) and windows were not locked shut. Additionally, lights were left on and the HVAC set points were at the occupied level while the building was vacant for extended periods of time.

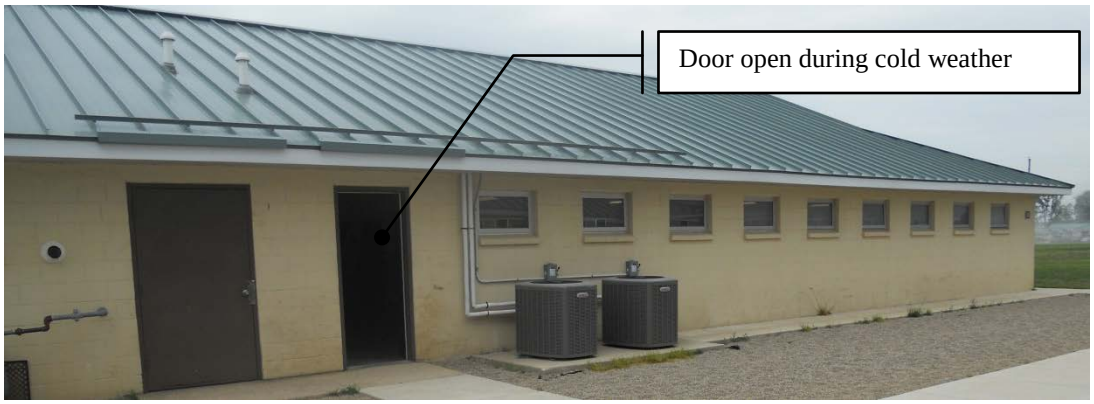


**Figure 4-5: Building 114 Operations.**

## 4.2 FACILITY/SITE CONDITIONS

The annual weather conditions in Indianapolis, IN are summarized in Figure 4-6, which is a graphical representation of the Typical Meteorological Year (TMY3) outdoor temperatures. The vertical axis is the outdoor temperature and the horizontal axis is the duration in hours. Figure 4-6 shows that CAJMTC can experience temperatures below 25°F (the threshold temperature for conventional heat pumps) for over 1100 hours in one year. The temperature versus accumulated hours shows how long a given location experiences weather conditions at and below the specified temperature. Even though it is not as cold as some locations, Edinburgh, IN (the location of CAJMTC) provided a sufficient amount of cold weather data to confirm the operation and advantages of a CCHP. Recorded temperatures from the Indianapolis airport for the past heating season show about 560 hours at temperatures below 25°F. Therefore the heating season during the demonstration was warmer than the TMY3 data base spanning 30 years.

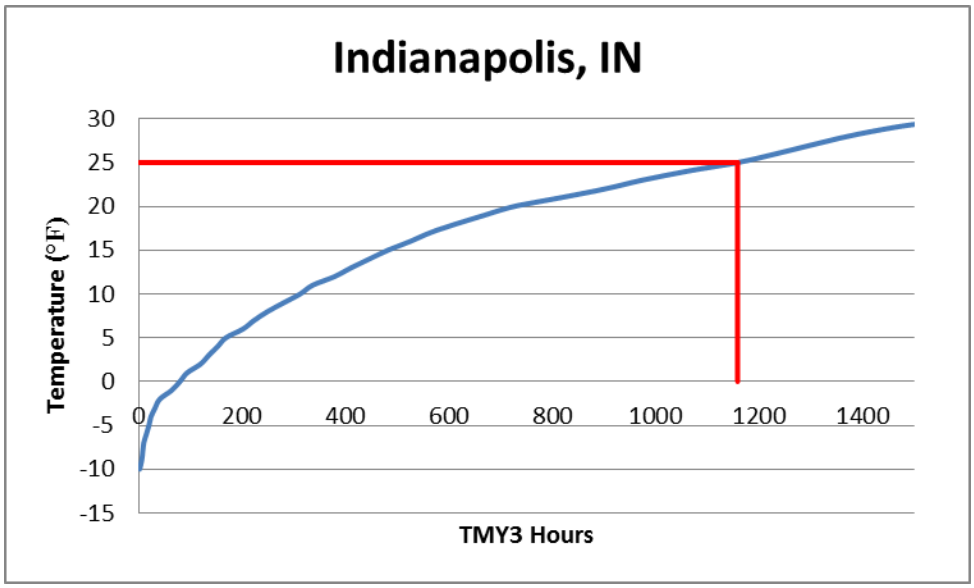


**Figure 4-6: Temperature Profile.**

Many temperature profiles were created to compare Indiana winters to other northern states. The locations chosen were based on the locations of existing DoD installations. A total of twenty cold climate locations were plotted. Only seven of the locations analyzed experienced colder climates than CAJMTC.

# 5.0 TEST DESIGN

This field demonstration evaluated the potential for energy and cost savings from the deployment of a CCHP. The experimental design allowed for a direct comparison of the performance of a NGF to the performance of a CCHP. The methods and materials for this project are explained so that similar results can be achieved from repeated and independent experimentation.

## 5.1 CONCEPTUAL TEST DESIGN

The multi-purpose buildings that normally serve as barracks for soldiers were selected as the buildings for the CCHP demonstration at Camp Atterbury. The two buildings (Buildings 113 & 114) selected are oriented north to south and are within 100 feet of each other. Figure 5-1shows that each building is split into two halves separated by a mechanical room and lavatory. The middle zones shown in green and blue are the mechanical rooms and lavatories, respectively.

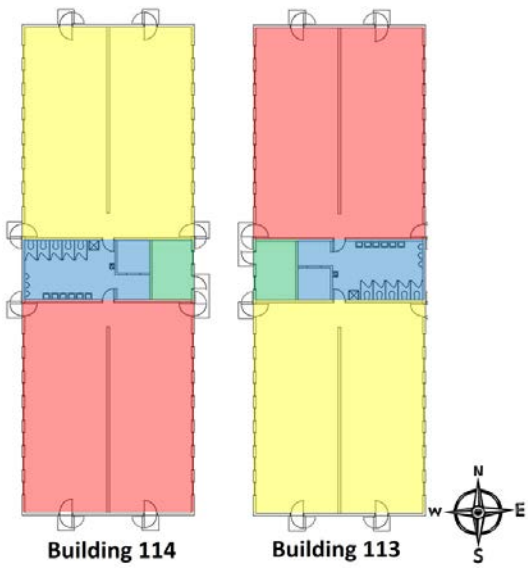


**Figure 5-1: General Experimentation Plan.**

The experiment was set up so that a direct comparison between the baseline and experimental HVAC systems could be made in one building. The zones highlighted in red in Figure 5-1 used the baseline heating, ventilation, and air conditioning (HVAC) system, which was a split-system air conditioner with a NGF. The yellow zones were conditioned by the CCHP. This experimental framework allowed the CCHP to be compared to traditional heating methods in the same building or the same half of a near identical building.

## 5.2 BASELINE CHARACTERIZATION

During the design phase of this field demonstration, Typical Meteorological Year data (TMY) was used for energy modeling and to put the indoor measurements in a proper context. TMY is normalized data used be researchers to predict expected weather conditions at specific locations all over the world. TMY3 is the third generation and most recent was used whenever possible. TMY2 is the second generation data set, but is also used in some situations to predict energy use.

These TMY data sets were one form of baseline characterization because they were needed for the CCHP design. The TMY data and information about the size and construction of buildings 113 & 114 was used to create an energy model of the building in eQUEST. The model was then used to predict monthly energy use. This model was also used to size the heating and cooling capacity required by the CCHP system.

Direct measurements of outdoor temperature conditions were taken as the field demonstration was conducted. Both outdoor temperature and outdoor relative humidity were tracked. This outdoor data is the basis for comparisons to energy modeling based on TMY data. Outdoor temperature is the key independent variable that dictates overall performance of the CCHP system, which is why many of the findings are expressed with respect to outdoor temperature.

The experimental design was planned so that direct measurements of baseline and experimental data could be collected at the same time. As described in Chapter 4, each building (113 & 114) had two independent but nearly identical zones. One zone had a NGF that provided baseline (reference) values for energy, indoor temperature, and indoor relative humidity. The other zone in the building zone had the CCHP that provided experimental values for energy, temperature, and relative humidity. With this approach a direct and valid comparison of baseline and experimental performance was made.

Energy was the core value for the demonstration; because the energy data drives the two other quantitative performance objectives. The energy objective is a direct measurement. The financial objective is computed from a comparison of the energy costs for electricity and natural gas. The environmental objective is also computed by converting the energy for electricity and natural gas into $CO_2$ equivalent emissions.

## 5.3 DESIGN AND LAYOUT OF TECHNOLOGY COMPONENTS

The field demonstration directly compared the performance of a CCHP to a conventional HVAC system in two zones of one building. This same test was conducted in two separate buildings. The first step was to install a new 200 Amp circuit breaker panel in the mechanical room of each barracks. The second step was to install the necessary supporting equipment, which included sensors and controls to support the experiment. Specific equipment consisted of a third party controller for remote monitoring and control, a 240/24VAC Transformer, a Silicon-Controlled Rectifier (SCR) for modulating electric reheat, electric meters, a natural gas meter, multiple in/outdoor temperature and relative humidity sensors, dampers, and a personal computer. Some of these electrical components can be seen in Figure 5-2.

**Figure 5-2: Supporting Equipment.**

New ductwork to integrate the CCHP air handler into the existing ducts (Figure 5-3) was completed by an HVAC contractor before the CCHP components were installed at CAJMTC. Dampers were installed with the new ductwork and were manually set to allow the conventional HVAC system to operate until CCHP testing was concluded at Purdue's Herrick Laboratories.

**Figure 5-3: Ductwork Integration.**

Lastly, the mechanical housing (Figure 5-4) and outdoor heat exchanger (Figure 5-5) were installed. If a third party programmable controller was not required then the only difference between installing a CCHP compared to a traditional heat pump would be the wiring and an additional line set. Line sets include a set of gas and liquid copper lines to attach the mechanical housing to the air handler and a set of gas and liquid copper lines to attach the mechanical housing to the outdoor heat exchanger. The additional wiring requirements include power to the

low stage compressor and control signal wiring (thermostat wire) from the air handler to the mechanical housing and then from the mechanical housing to the outdoor heat exchanger.

**Figure 5-4: Mechanical Housing Installed at CAJMTC.**

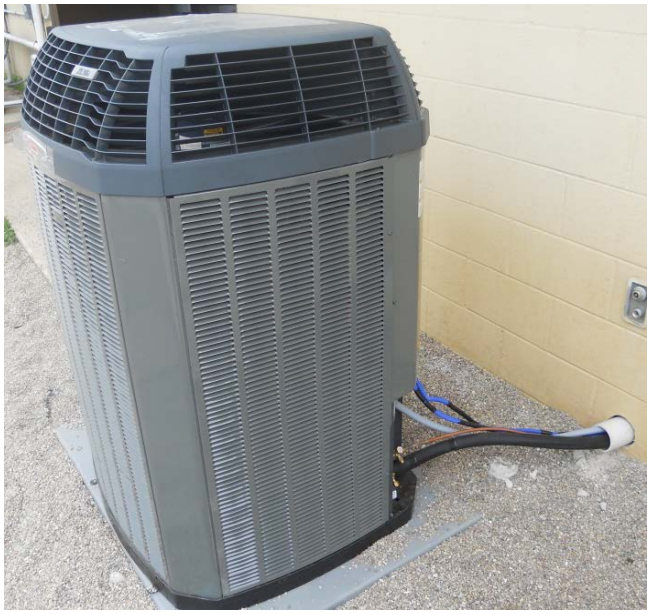

**Figure 5-5: Outdoor Heat Exchanger Installed at CAJMTC.**

Installing the CCHP components into an existing building without interfering in its operation was desirable while monitoring the performance of the CCHP and traditional HVAC equipment. The mechanical room provided ample space for most of the components and monitoring equipment. Additionally, the large attic allowed more ductwork to be installed which allowed for an easy transition between the CCHP and existing conventional system.

Due to the experimental nature of the CCHP, it was advantageous to keep the existing HVAC system as a backup. Figure 5-6 shows that this was accomplished by adding new ductwork in parallel to the existing ductwork to allow either system to operate.

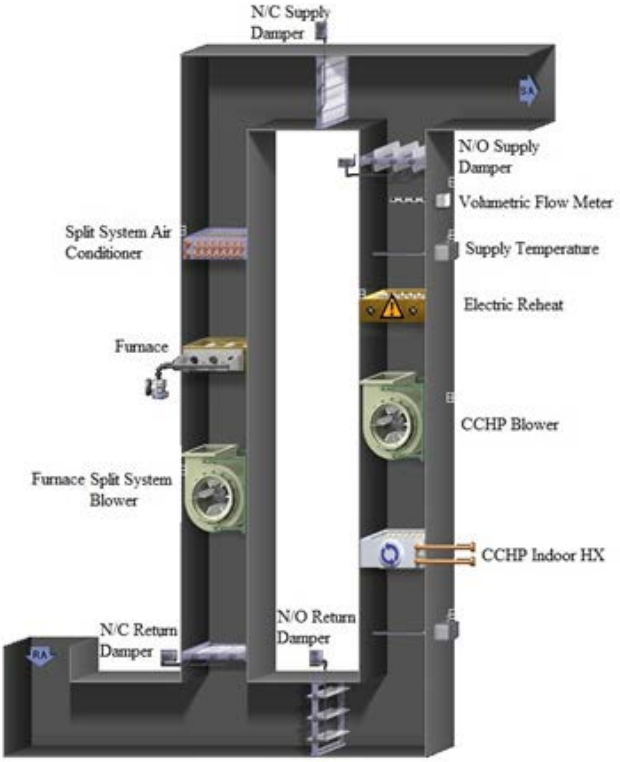


**Figure 5-6: Building Integration.**

The right column of ductwork represents the CCHP component integration into the existing (left column) ductwork containing the split-system air conditioner and furnace. During normal CCHP operation, there were two open dampers and two closed dampers to force air flow through the right column so that comfort is provided by the CCHP system. In the event of CCHP system failure, the dampers reverse their operation so the dampers that were once open are now closed and vice versa. This now forces the air to flow through the conventional system (left column) so that building conditions are not interrupted due to any experimental failure.

The travel time between Purdue University (West Lafayette, IN) and the test site (CAJMTC, Edinburgh, IN) is roughly 2 hours. Therefore, it was very desirable to have remote control using a third party controller. The third party controller communicated to a server (personal computer) that had access to an internet connection at CAJMTC (Figure 5-7).

**Figure 5-7: Third Party Controller (left) and Server (right).**

The CCHP has seven different modes of operation, which includes single stage cooling, single stage heating, two-stage heating, defrost, oil equalization, oil return, and free cooling. A control program uses a variety of temperature and pressure inputs to determine which mode is active at any given time. Figure 5-8 is a flow chart that provides an overview of how these elements interact to form the control strategy for the CCHP.

The sequence begins in the upper left hand corner at the "START" ellipse. The diamonds represent a conditional statement where, if the statement is false the path flows to the right and if the statement is true the path moves downward. The rectangles are high level program elements that contain detailed conditions and other program elements within them. Upon reaching the "END" ellipse, the sequence of operation repeats its' self as illustrated with the circular arrow at the bottom-center. A nomenclature list is provided in Figure 5-8 to describe the abbreviations.

Appendix B has the complete control code developed for this field demonstration as well as detailed discussions of the control program and logic.

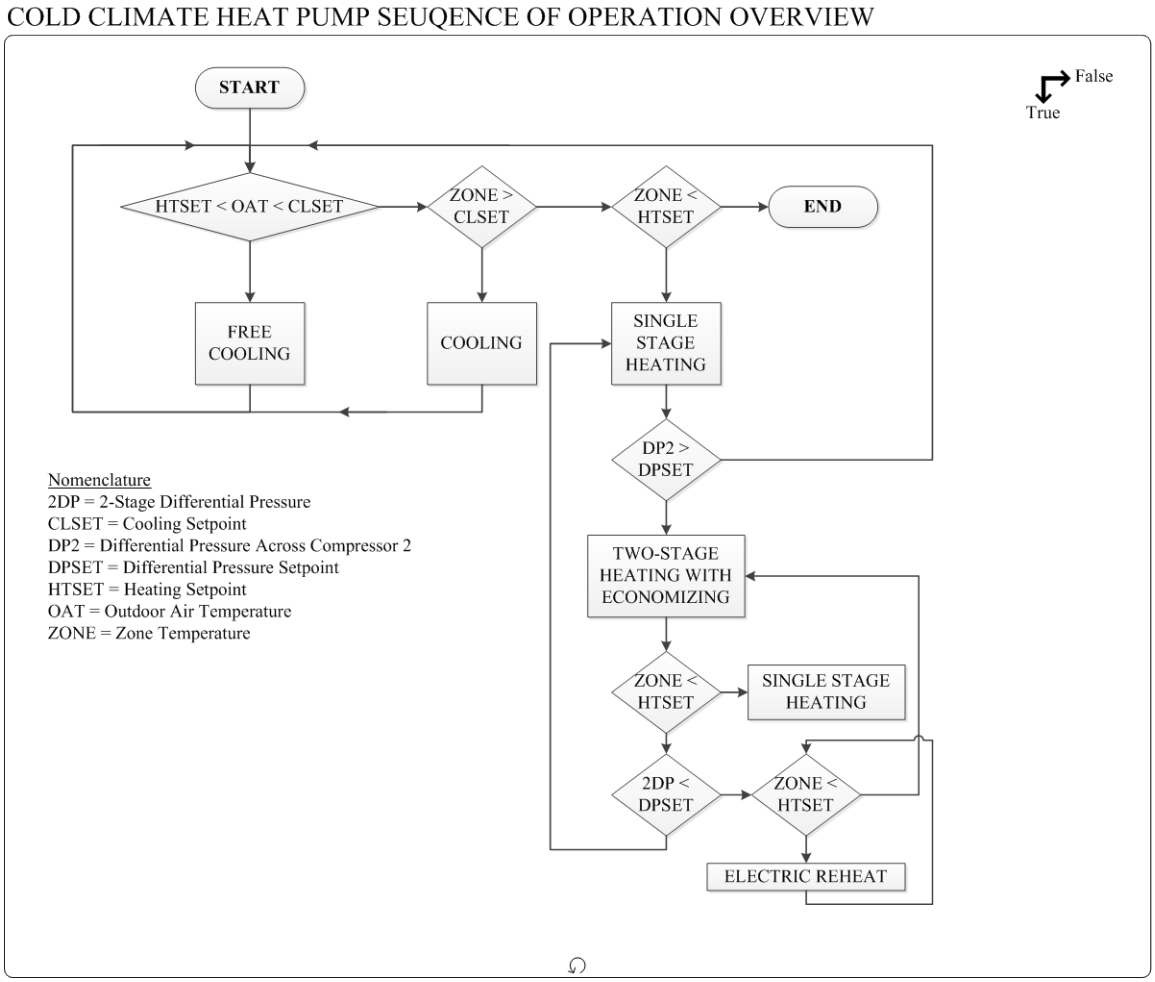


**Figure 5-8: Control Strategy Overview.**

## 5.4 OPERATIONAL TESTING

Before any field experimentation could begin, the CCHP was tested at Purdue University's Ray W. Herrick Laboratories. The CCHP was installed into two psychrometric chambers to simulate its operation. Pictures of the CCHP installed at Herrick Labs can be seen in Figure 5-9.

**Figure 5-9: Psychrometric Room Installation.**

The indoor psychrometric room (left image of Figure 5-9) simulated a building load for the CCHP to ensure it would operate in heating, cooling, and two-stage heating. The indoor heat exchanger, controller, and mechanical housing were placed in the indoor psychrometric room. The outdoor psychrometric room (right image of Figure 5-9) was brought down to a temperature of -6ºC (21ºF) to ensure it was able to operate at temperatures below the traditional heat pump cut off of -4°C (25°F).

Additionally, the outdoor psychrometric room operated at temperatures of 5ºC (41ºF) with high humidity to test the defrost cycle (see Figure 5-10). The defrost cycle is important to avoid ice buildup on the outdoor coil. The defrost cycle is accomplished by reversing the direction of refrigerant in the heat pump so that hot gas is diverted to the outdoor coil. This process continues until ice is melted from the coils and normal operation can commence again.

The primary purpose for installing the CCHP into the psychrometric chambers was to evaluate the control algorithms to ensure that the CCHP could operate autonomously. Once the control algorithms were tested, the CCHP components were removed from the psychrometric rooms and transported down to CAJMTC.

**Figure 5-10: Testing the Defrost Cycle.**

As shown earlier in Table 1-1, the field demonstration at Camp Atterbury commenced in January of 2012 and ended 16 months later in April of 2013. This work proceeded in three phases: 1) system start-up, 2) cold climate testing, and 3) shutdown and demobilization. Each phase is discussed in more detail in the paragraphs that follow.

***System start-up*** was for 9 months, from January of 2012 through September of 2012. Much of this work involved testing and improving the CCHP control algorithm to achieve stable long term operation. However, significant delays to CCHP testing were encountered from refrigerant leaks. The R410A used in the CCHP is a high pressure refrigerant and therefore, all leak tests involved pressuring the system with Nitrogen up to 3800 kPa (550 psi) and monitoring for over 24 hours. The air handler and outdoor heat exchanger held pressure very well, but the mechanical housing fabricated for this project was the primary cause of leaks. This was due to the mechanical housing having the most amount of instrumentation which penetrated the system (i.e. required direct contact with the refrigerant).

All temperature sensors used were thermistors and were simply secured to the outside wall of the piping. System penetrations instrumentation included pressure transducers, switches, and relief valves. There were a total of three pressure switches (low, intermediate, and high) to protect each compressor. Four pressure transducers were used (high, intermediate, low, and the outdoor liquid line). Installing these components resulted in a continuous loop of testing for leaks. Two pressure relief valves were installed to eliminate the possibility of bursting any piping. Due to the experimental nature of the system additional components were utilized as well.

For instance, a capillary tube was used on all of the pressure switches and sensors to remotely place them; making them easily replaceable and alleviate any vibrations experienced if they were directly attached to the piping. Additionally, all of the pressure transducers had a ball valve installed before them. This allowed a pressure transducer to be easily replaced and not require evacuating and recharging the system each time a pressure transducer failed.

The refrigerant leaks were finally overcome by the following corrective actions:

- Ensured all pressure transducers and switches had ¼ SAE threads.
- Installed a copper O-ring when applicable.
- Ensured all fittings for transducers and actuators were made with Schroder valves.
- Avoided the use of thread sealants; but applied a light film of compressor oil to the threads.

***Cold climate testing*** commenced in October of 2012, as soon as the outdoor temperature was cold enough so that heating was needed. This work continued for the entire 6 month heating season, which lasted through March of 2013. The goal during this time was to run the CCHP systems every day and all day as the primary source of heat for the buildings used in the field demonstration.

Unfortunately, the CCHP systems had to be shut down on a regular basis due to problems with oil migration, at which point the HVAC systems for the barracks buildings reverted back to the backup system, which was a traditional NGF. The most frequent cause of system shutdowns was due to oil migration from the compressor housing that caused lubrication problems that ultimately led to premature compressor failure. Chapter 8 of this final report discusses corrective actions that were taken to resolve this problem.

Table 5-1 summarizes the operating conditions for a total of eleven data sets representing over 400 hours of CCHP operation for both buildings at Camp Atterbury. The tests occurred in October 2012, and January, February of 2013. Each data set reflects a period when the CCHP operated for over ten consecutive hours. The analysis was separated into discrete time intervals because the CCHP was not running outside of these time intervals. As shown in Table 5-1, the outdoor temperatures fell to as low as -8.3°C (17°F) and rose up to 27.8°C (82°F) during the over 400 hours of data acquisition. The extreme temperatures predicted from the TMY databases were not recorded due to experiencing a mild winter. The design point for the CCHP used an ambient temperature of -4°F (-20°C) that was never reached during the field demonstration. Luckily some data sets had low enough outdoor temperatures to require two-stage operation of the CCHP. Otherwise very mild winter temperatures would eliminate the need for a CCHP at the testing site.

***Shut down and demobilization*** took place from May of 2013 and proceeded through the end of the contract. The CCHP test equipment is still installed at Camp Atterbury to support further commercialization work that is still underway. A letter from Camp Atterbury acknowledging this agreement is included in Appendix D of this final report.

**Table 5-1: Summary of CCHP Test Interval.**

| Building | Data Set No. | Data Collection Period | | | Hours: Minutes of Data | Low Outdoor Temperature [F/C] | Average Outdoor Temperature [F/C] | High Outdoor Temperature [F/C] | CCHP DoD Energy Cost $ | NGF DoD Energy Cost $ |
|---|---|---|---|---|---|---|---|---|---|---|
| 114 | 1 | 10/12/2012 9:40 PM | - | 10/13/2012 1:10 PM | 15:30 | 42 / 5.6 | 53 / 11.7 | 68 / 20 | $4.48 | $2.29 |
| | 2 | 10/15/2012 2:40 PM | - | 10/16/2012 1:05 PM | 22:25 | 39 / 3.9 | 52 / 11.1 | 65 / 18.3 | $4.80 | $2.85 |
| | 3 | 10/25/2012 8:22 AM | - | 10/26/2012 3:29 PM | 31:07 | 46 / 7.8 | 65 / 18.3 | 82 / 27.8 | $5.43 | $4.43 |
| | 4 | 10/27/2012 1:50 AM | - | 10/27/2012 3:45 PM | 13:55 | 38 / 3.3 | 45 / 7.2 | 55 / 12.8 | $4.61 | $5.71 |
| | 5 | 1/4/2013 11:00 AM | - | 1/5/2013 9:05 AM | 22:05 | 17 / -8.3 | 27 / -2.8 | 37 / 2.8 | $12.89 | $9.90 |
| | 6 | 1/10/2013 9:35 AM | - | 1/17/2013 1:55 AM | 160:20 | 19 / -7.2 | 41 / 5 | 67 / 19.4 | $96.76 | $46.00 |
| | 7 | 1/25/2013 1:25 AM | - | 1/25/2013 6:10 PM | 16:45 | 21 / -6.1 | 25 / -3.9 | 30 / -1.1 | $7.99 | $6.82 |
| | 8 | 2/14/2013 10:30 AM | - | 2/15/2013 7:25 PM | 32:55 | 30 / -1.1 | 39 / 3.9 | 53 / 11.7 | $13.98 | $11.13 |
| 113 | 9 | 2/1/2013 2:45 PM | - | 2/4/2013 4:45 AM | 62:00 | 17 / -8.3 | 27 / -2.8 | 42 / 5.6 | $36.38 | $35.10 |
| | 10 | 2/13/2013 3:00 PM | - | 2/14/2013 12:50 AM | 9:50 | 40 / 4.4 | 44 / 6.7 | 49 / 9.4 | $10.73 | $12.75 |
| | 11 | 2/14/2013 9:40 AM | - | 2/15/2013 9:10 PM | 35:30 | 29 / -2.2 | 40 / 4.4 | 59 / 15 | $3.48 | $3.20 |

## 5.5 SAMPLING PROTOCOL

Instrumentation was selected based on accuracy, resolution, and compatibility with the data acquisition system. A summary of the instrumentation used for evaluating the performance objectives is illustrated in Table 5-2. The appendices of this report have certificates of calibration for key pieces of data acquisition equipment and sensor to establish their accuracy.

**Table 5-2: Performance Objective Sampling.**

| Instrument | Measurement | Units | Accuracy | Frequency |
|---|---|---|---|---|
| CCHP Electric Meter | Energy | kWh | ±0.5% | 0.5 COV |
| Furnace Electric Meter | Energy | kWh | ±0.5% | 0.5 COV |
| Gas Meter | Energy | SCF | ±0.5% | 1 COV |
| Outdoor Conditions Sensor | Temperature | °F | ±0.36°F | 10 min |
| | Relative Humidity | % | ±2.0% | 10 min |
| Indoor Comfort Sensors | Temperature | °F | ±0.36°F | 10 min |
| | Relative Humidity | % | ±2.0% | 10 min |

Note that the frequency of the data collection varies between the outdoor conditions, gas meter, and electric meters. The gas and electric meter data is collected on a change of value (COV) basis. This is beneficial because during low heating loads the server will not be overloaded with insignificant data compared to sampling at set intervals. However, when there is a high heating load many data points are collected due to the systems consuming more energy. The outdoor conditions are collected at ten minute intervals. In a standard research setting this would be too long. However, during an in-situ demonstration the temperatures and humidity's do not fluctuate quickly, allowing lower data collection rates to become acceptable.

In addition to remote control, the third party controller also provided the ability to remotely acquire data. Figure 5-11 is a screen shot of the interface that provided real time data on the performance of the equipment. This screen was most useful when directly monitoring the CCHP operation during a specific test because it was able to show changes in the values of multiple variables at the same time. Figure 5-11shows multiple refrigerant pressures, refrigerant temperatures, control valve status, and air side set points all on one convenient screen.

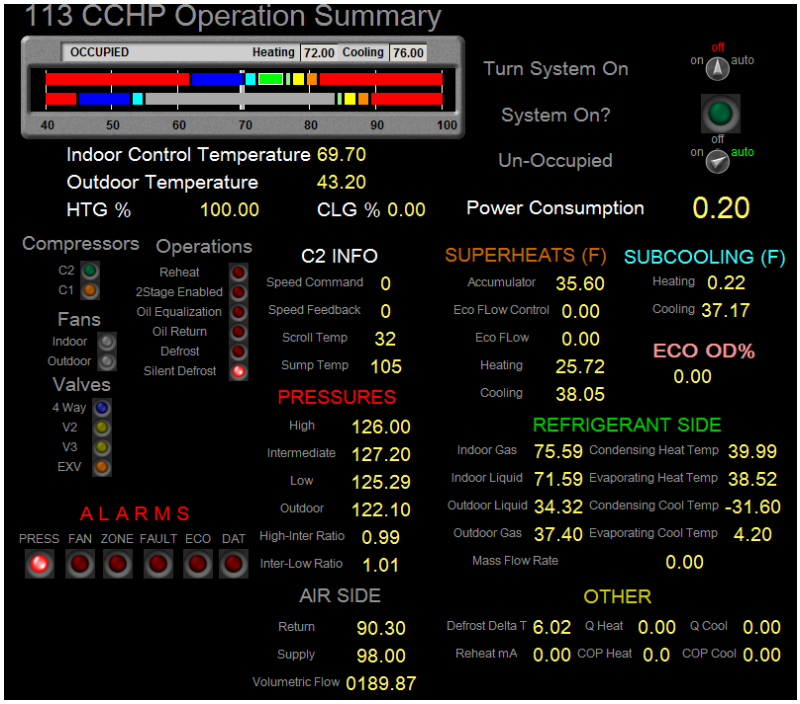


**Figure 5-11: Data Acquisition Interface.**

Figure 5-12 presents a compilation of specific variables from Figure 5-11 as a function of time. This data trend was useful during longer term testing that lasted hours or days. The data trend was most useful for confirming that stable operation of the CCHP had been achieved over a period of time. This data was collected at regular intervals and saved in multiple places to ensure nothing was lost.

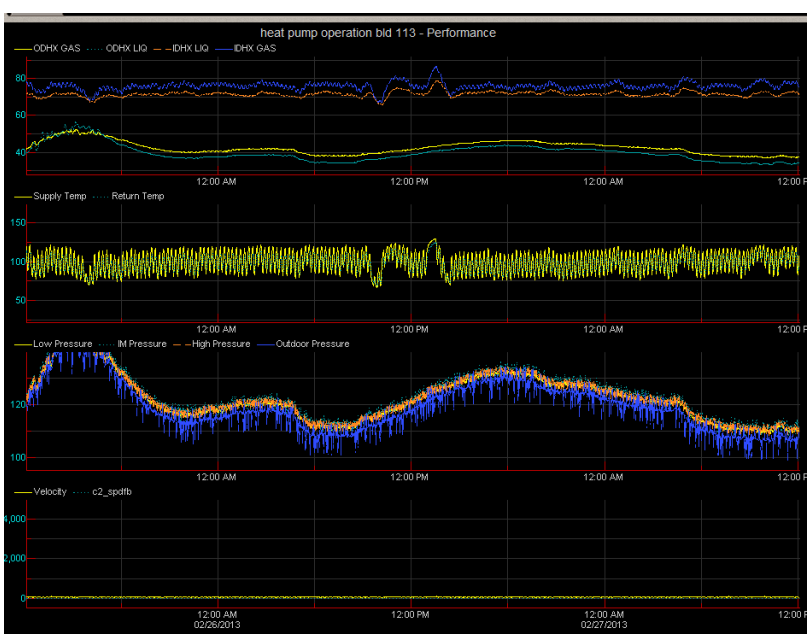


**Figure 5-12: Trending Interface.**

## 5.6 SAMPLING RESULTS

A significant amount of instrumentation was installed on the CCHP to record temperatures, pressures, and its overall electric consumption. Additional instrumentation provided verification on the status of individual components as well. Since a majority of these measurements were not used to generate the results needed for the performance objectives, only points of significance are used in this section. Also, due to the large amount of data collected from all eleven data sets, only one data set is selected and presented in this section. Therefore, only points of significance from the selected data set are shown within this section. An excerpt of the data points recorded for the selected data set can be seen in appendix E. For a complete set of all the raw data collected, an Excel file will be submitted as part of this final report.

The data set that was selected for discussion in this report included both two-stage and single-stage operation and also had a relatively small time span for ease of plotting. The 7$^{th}$ data set, January 25$^{th}$, for building 114 was found to meet these criteria. The measurement points of significance were any values that were needed to evaluate the performance objectives as well as additional points that provide significant insight into the operation of the CCHP.

The CCHP had four pressure sensors installed to monitor the compressors operation when in single or two-stage mode. Two of the sensors were located on the high side pressure of the CCHP. One sensor location was directly after the compressor, which gives the highest pressure reading of all sensors, and the second location was up-stream of the expansion valve used for heating. The highest pressure sensor is labeled “high” and the sensor located before the expansion valve is labeled “out” since its location is outside, on the outdoor heat exchanger. The two other pressure sensors are on the suction side of each compressor. When in single-stage mode, both sensors will read the same value since only the high-side compressor is running. Although when running in two-stage mode, both sensors will read different pressures since the suction port of each compressor is operating at a different pressure.

Figure 5-13 is a plot of the four different pressure readings during the entire 7$^{th}$ data set versus time. The heat pump operates in two-stage mode for most of this data set and can be identified easily between the hours of 2:00 am and 10:00 am. Then, between the hours of 4:00 pm and 6:00 pm, the heat pump switches into single-stage operation. A difference between the intermediate and low pressure readings is an indicator of when the heat pump is operating in two-stage mode.

The CCHP had seven temperature sensors installed to monitor refrigerant conditions throughout different locations. Two additional temperature sensors were equipped on the high-stage compressor; discharge or scroll temperature and the motor temperature. These readings provide insight into the performance of the overall heat pump, expansion valves, and individual compressors. A summary of most of these temperature readings and the outdoor air temperature plotted versus time can be seen in Figure 5-14.

The temperature drop across the indoor heat exchanger or air handler is shown by comparing the entering, gas temperature of the IDHX versus the exiting, liquid temperature of the IDHX. For the selected data set, this difference is approximately 100°F. The operation of the outdoor heat exchanger is captured by comparing the outdoor air temperature versus the gas temperature leaving the coil, ODHX gas. The difference between these temperatures is more than 20°F for

the data set investigated. The smaller this difference is, the higher the efficiency of the outdoor heat exchanger.

The highest temperatures recorded are at the discharge of the high-stage compressor, also known as the scroll temperature. The measurement is influenced by any two-stage operation. In Figure 5-14 between the hours of 2:00 am and 10:00 am, the scroll temperature is lower than the reading between the hours of 4:00 pm and 6:00 pm. The difference is due to the mixing chamber cooling the discharge gasses leaving the low-stage compressor before entering the high-stage compressor. By cooling these low-stage discharge gasses, the entering refrigerant is closer to a saturated vapor state and results in reduced gas temperatures leaving the high-stage compressor. Additionally, this effect improves the efficiency of the compressor.

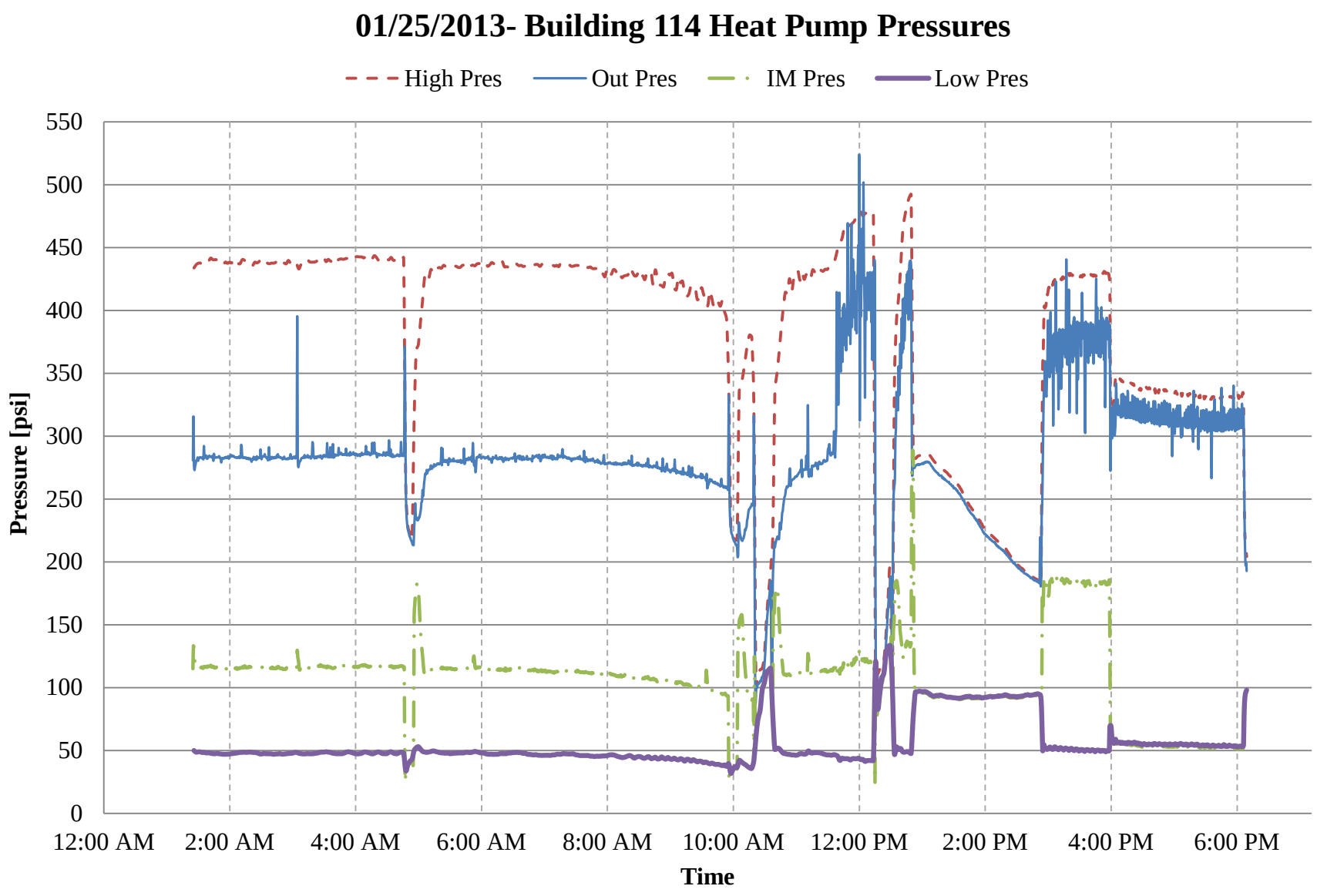


**Figure 5-13: Data Set 7 – Heat Pump Refrigerant Pressures.**

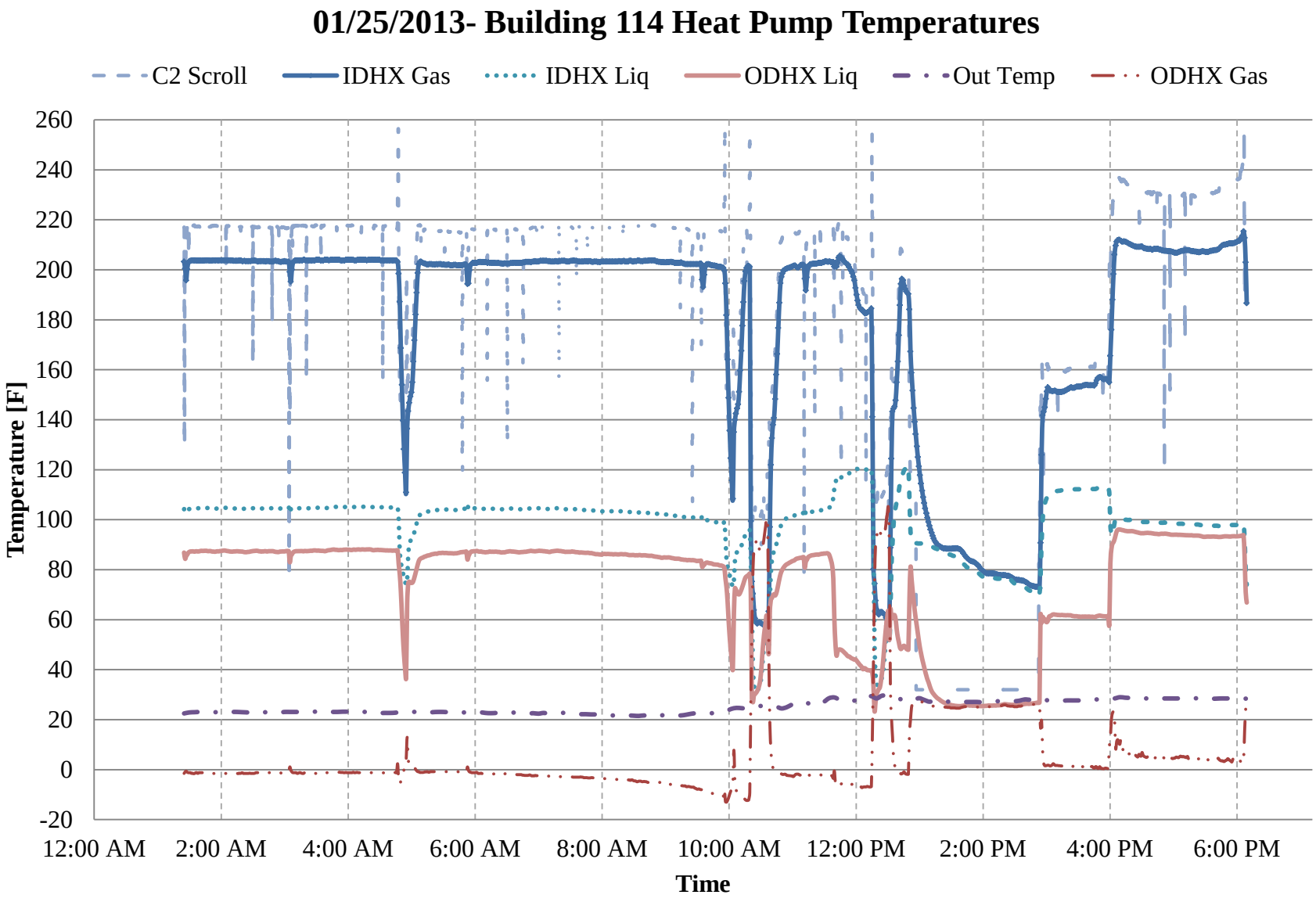


**Figure 5-14: Data Set 7 – Heat Pump Refrigerant Temperatures.**

To monitor the air side performance of the heat pump, air temperature sensors were located before and after the indoor heat exchanger. Also, the temperature sensors located within the conditioned zones of the building were used to evaluate the effects of the CCHP and furnace operation on the space. Figure 5-15 provides a summary of all the zone temperatures, CCHP supply and return temperatures, and the CCHP control temperature. Also, as a reminder, please note that for data set 7, the heat pump provides heating to the northern half of building 114 and the furnace for the southern half.

The first point to note from Figure 5-15 is the high temperatures of the supply air relative to the other readings. These high supply temperatures are dependent on the number of stages the CCHP is utilizing. When in two-stage mode, higher supply temperatures are observed due to the CCHP running at higher pressures. The difference in temperatures can be observed in Figure 5-15 when comparing hours 2:00 to 10:00 am with hours 4:00 to 6:00 pm and the difference of pressures in Figure 5-13 over the same hour ranges. The larger amount of heat available when using two-staged compression becomes evident with this comparison to single-stage operation.

The northern zone has different temperature trends than the southern zone due to the difference in the operation of the CCHP from a NGF. In Figure 5-15, the southern zone temperatures show averages around 85°F while the northern zone has lower averages closer to 70°F. The northern zone has a larger temperature difference between measurement points than between points for the southern zone. It is believed that the larger temperature differences between points for the CCHP are due to the ductwork modifications made to allow for integration into the existing duct work.

The comparison between the CCHP and NGF energy consumption is plotted and shown as Figure 5-16. The energy consumption for the NGF is shown as a rate, standard cubic feet of natural gas per hour. Note that data set 7 does not have a complete record of the energy rate but the totalized amount of natural gas is available to make primary energy comparisons. The NGF shows some electricity consumption due to use of the blower. The CCHP operates using only electricity as shown in the figure. The power consumption of the CCHP increases by more than double for this data set when switching from single-stage to two-stage mode. Between the hours of 2:00 to –10:00 am the CCHP uses roughly 13 kW of electricity, but during single-stage mode, between the hours for 4:00 to 6:00 pm, the CCHP uses only 6 kW of electricity. The doubling of the CCHP electricity consumption is still able to provide efficient heating due to a corresponding multiplier increase of the CCHP heating capacity.

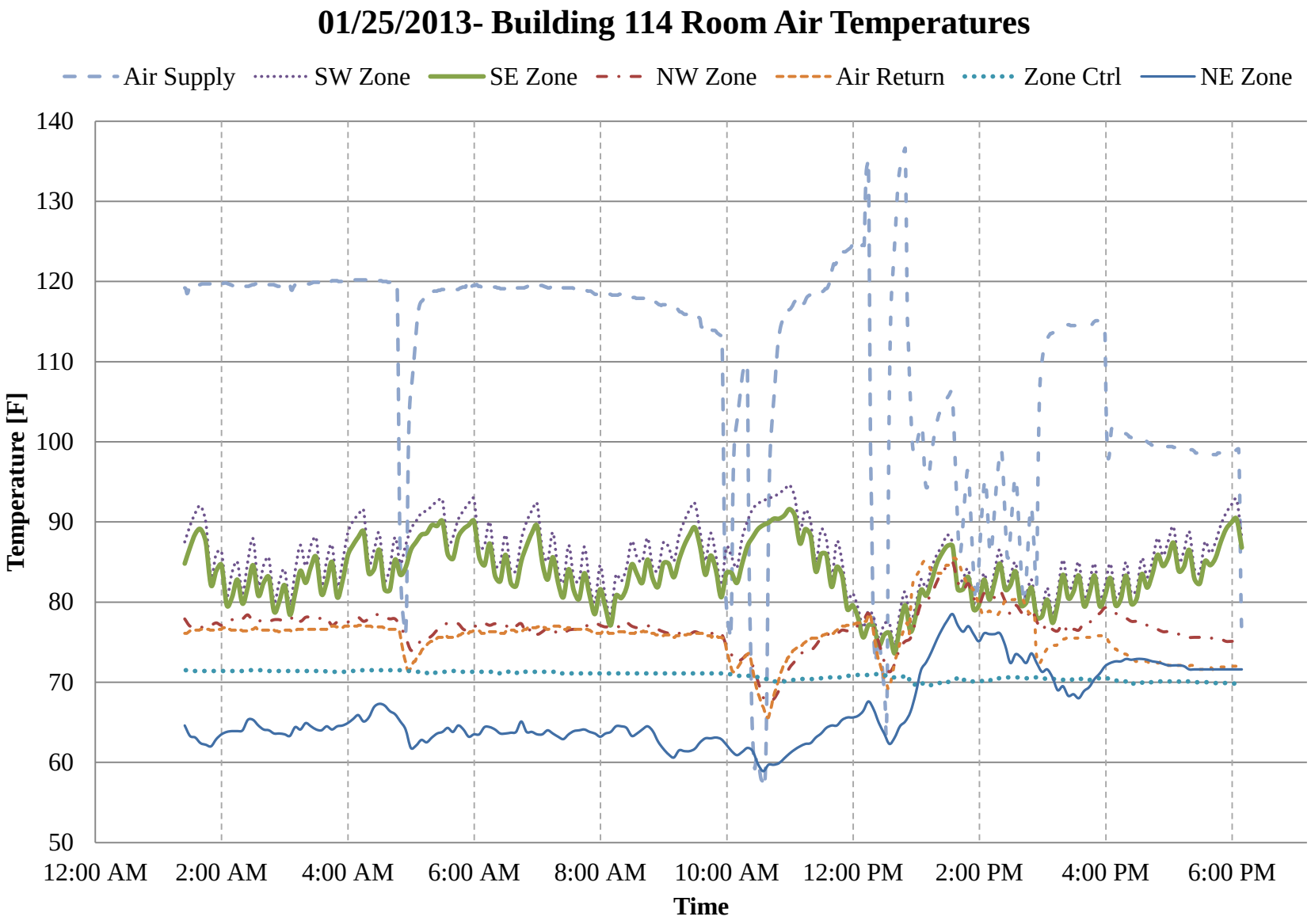


**Figure 5-15: Data Set 7 – Room Air Temperatures.**

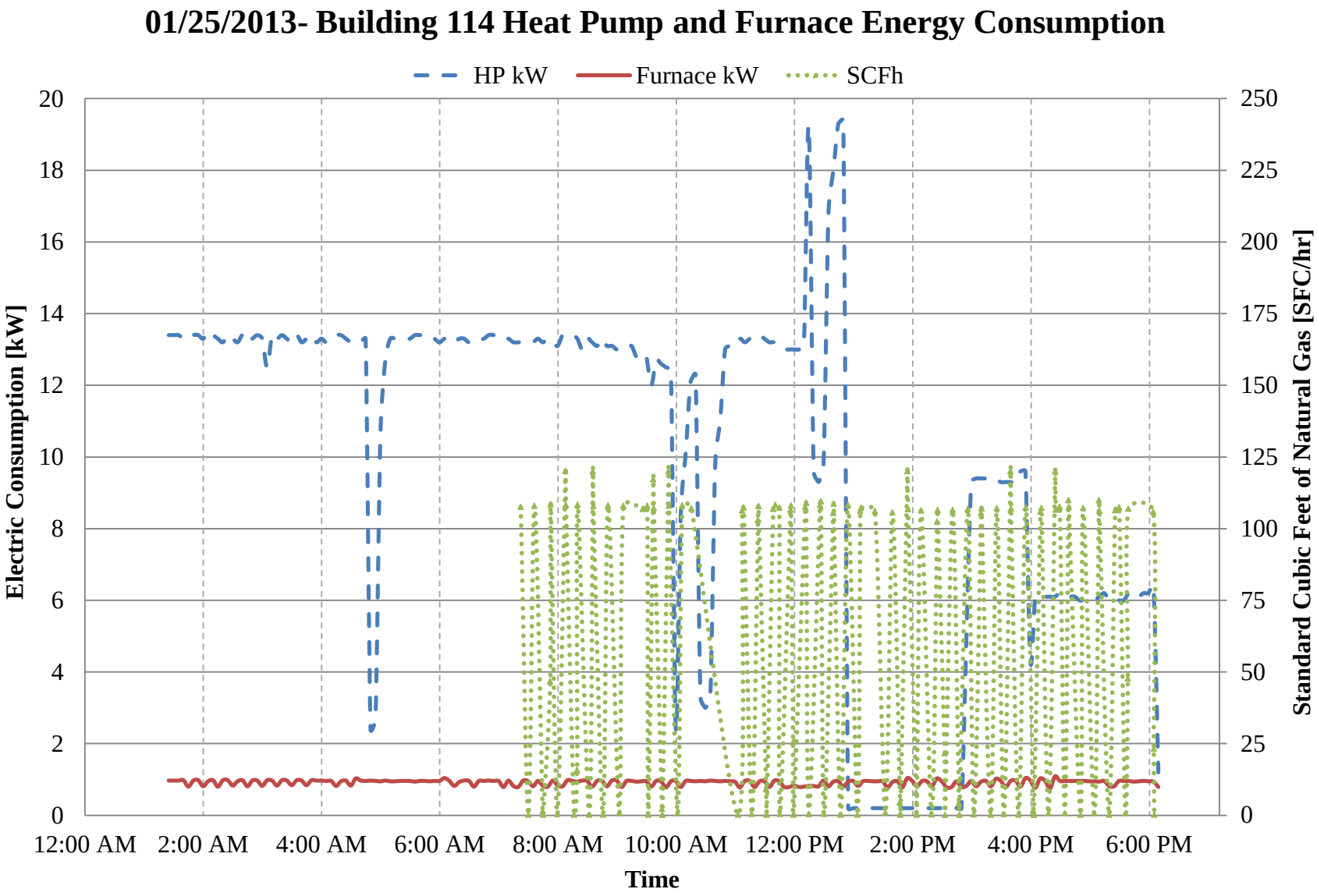


**Figure 5-16: Data Set 7 – Energy Consumption of Heat Pump and Furnace.**

# 6.0 PERFORMANCE ASSESSMENT

A laboratory environment (e.g. psychrometric room testing) is useful for testing components of and HVAC system and establishing equipment performance for rating purposes. However, a field test is the ultimate step for evaluating a new technology prior to full commercialization. Variables that cannot be controlled by the experimenters are certainly introduced which creates an imperfect, but real world environment. During the CCHP test, there were instances where doors and windows were left open, which were costly in terms of achieving optimal energy performance but are also part of a real world operation.

The energy consumption was monitored over time to provide a direct comparison of primary energy. Cost and emissions are computed directly from this energy consumption data. Table 6-1 and Table 6-2 summarize the results achieved for performance objectives 1, 2, and 3.

Table 6-1 is the energy, cost, and emission data for building 114 (CCHP and NGF supplying comfort to the northern and southern zones of building 114, respectively) and Table 6-2 has the same information for building 113 (CCHP and NGF supplying comfort to the southern and northern zones, respectively). The results are presented as a direct comparison between the CCHP and NGF on a source (primary) energy basis in kilowatt-hours (kWh). In Table 6-1 the first row in the CCHP section are the values obtained directly from the CCHP electric meter. These values were used to calculate the results shown in the last three rows of the CCHP section. Similarly, the first two rows of the NGF section are the values obtained directly from the electric and gas meter. The values from the gas meter were converted to kWh and added to the NGF electric meter. Again, these values were used to obtain the NGFs performance which can be found in the last three rows of the NGF section.

**Table 6-1: Building 114 Results for Energy, Cost, and Emissions.**

| | Data Set No. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
|---|---|---|---|---|---|---|---|---|---|
| | Average Outdoor Temperature [F/C] | 53 / 11.7 | 52 / 11.1 | 65 / 18.3 | 45 / 7.2 | 27 / -2.8 | 41 / 5 | 25 / -3.9 | 39 / 3.9 |
| CCHP (Bld 114) | Electric Meter Real Energy (kWh) | 56.0 | 60.0 | 67.9 | 57.6 | 161.1 | 1209.6 | 99.9 | 174.7 |
| | Primary Energy (kWh) | 143.3 | 153.5 | 173.7 | 147.3 | 412.0 | 3093.9 | 255.4 | 446.9 |
| | DoD Operational Cost ($) | $4.48 | $4.80 | $5.43 | $4.61 | $12.89 | $96.76 | $7.99 | $13.98 |
| | Residential Operational Cost ($) | $5.88 | $6.30 | $7.13 | $6.05 | $16.91 | $127.00 | $10.49 | $18.34 |
| | Carbon Dioxide Emissions (kg) | 26.3 | 28.2 | 31.9 | 27.0 | 75.6 | 567.8 | 46.9 | 82.0 |
| Furnace (Bld 114) | Electric Meter Real Energy (kWh) | 12.9 | 18.1 | 26.6 | 12.0 | 20.3 | 141.0 | 15.4 | 23.9 |
| | Standard Cubic Feet (CF) of Nat. Gas | 266.5 | 298.5 | 489.5 | 1010.0 | 1762.1 | 7386.3 | 1188.9 | 1960.8 |
| | Primary Energy (kWh) | 112.1 | 134.9 | 213.2 | 330.2 | 574.4 | 2551.5 | 392.0 | 642.6 |
| | DoD Operational Cost ($) | $2.29 | $2.85 | $4.43 | $5.71 | $9.90 | $46.00 | $6.82 | $11.13 |
| | Residential Operational Cost ($) | $3.88 | $4.73 | $7.42 | $10.81 | $18.80 | $84.68 | $12.86 | $21.06 |
| | Carbon Dioxide Emissions (kg) | 20.6 | 24.8 | 39.1 | 60.6 | 105.4 | 468.3 | 71.9 | 117.9 |

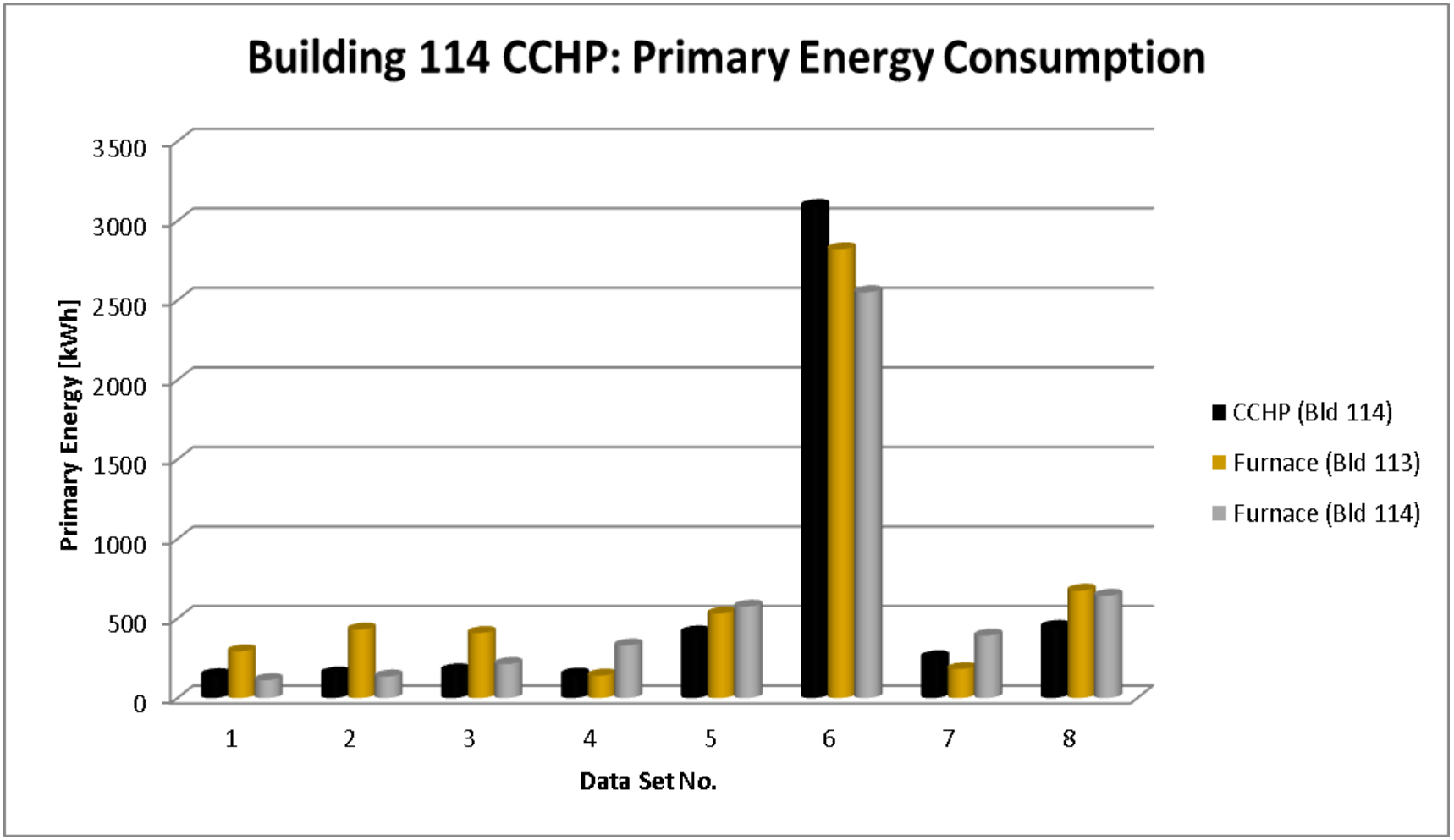


**Figure 6-1: Primary Energy Consumption Summary for CCHP in Building 114.**

**Table 6-2: Building 113 Results for Energy, Cost, and Emissions.**

| | Data Set No. | 9 | 10 | 11 |
|---|---|---|---|---|
| | Average Outdoor Temperature [F/C] | 27 / -2.8 | 44 / 6.7 | 40 / 4.4 |
| CCHP (Bld 113) | Electric Meter Real Energy (kWh) | 454.7 | 134.2 | 43.6 |
| | Primary Energy (kWh) | 1163.2 | 343.1 | 111.4 |
| | DoD Operational Cost ($) | $36.38 | $10.73 | $3.48 |
| | Residential Operational Cost ($) | $47.75 | $14.09 | $4.57 |
| | Carbon Dioxide Emissions (kg) | 213.5 | 63.0 | 20.4 |
| Furnace (Bld 113) | Electric Meter Real Energy (kWh) | 52.2 | 26.7 | 6.0 |
| | Standard Cubic Feet (CF) of Nat. Gas | 6579.7 | 2258.4 | 579.8 |
| | Primary Energy (kWh) | 2085.0 | 738.0 | 187.3 |
| | DoD Operational Cost ($) | $35.10 | $12.75 | $3.20 |
| | Residential Operational Cost ($) | $67.73 | $24.16 | $6.11 |
| | Carbon Dioxide Emissions (kg) | 382.7 | 135.4 | 34.4 |

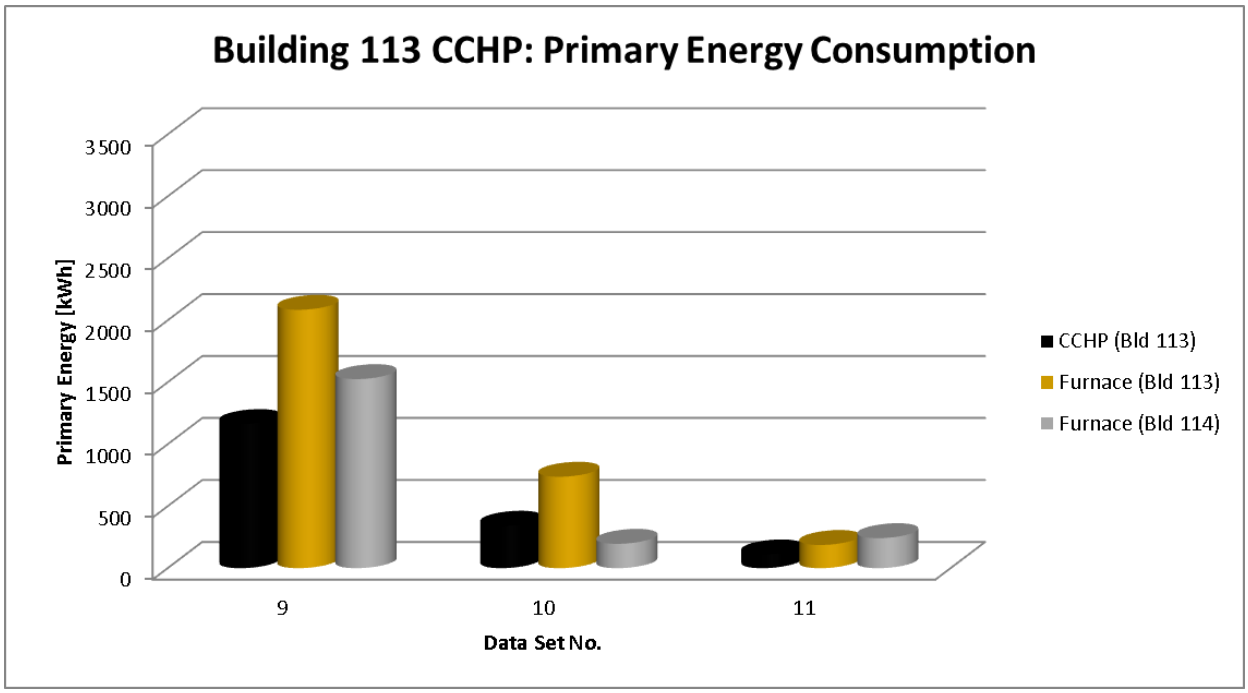


**Figure 6-2: Primary Energy Consumption Summary for CCHP in Building 113.**

## 6.1 PRIMARY ENERGY

***Calculation Procedure*** Table 6-1 and Table 6-2 illustrate the importance of comparing each method on a primary energy basis. The CCHP uses only electricity, while the NGF uses electricity for the fan and burns natural gas to generate the heat. Taking both the electricity and the natural gas back to primary energy requires making energy conversions but allows for both systems to be accurately compared. For the NGF, the SCF is converted using Equation 6-1. Two factors are used for the conversion. The first is the heat content of natural gas referenced from the 2011 EIA data base for Indiana. The value is also listed in Table 6-3 under the NGF category. The second factor is a simple conversion between BTUs and kWh.

$$SCF\ Natural\ Gas\ to\ kWh = 1\ SCF * \frac{1012\ BTUs}{SCF} * \frac{1\ kWh}{3412\ BTU} \qquad \text{Equation 6-1.}$$

The next energy conversion required taking the kWh of electricity back to the primary energy consumed by the power plant. In order to convert the electric meter values ($kWh_{EM}$) to primary energy, a power plant ($\eta_{PP}$) and transmission line ($\eta_{TL}$) efficiency were assumed. The efficiency values assumed are listed in Table 6-3 under the natural gas power plant category. It is also assumed all the electricity is produced by a natural gas power plant to provide a fair comparison by having both technologies using natural gas, one burning the natural gas directly, and the other burning natural gas to generate electricity. The electric meter values are then divided by the product of both the power plant and transmission line efficiencies. The electric meter energy to primary energy conversion can be seen in Equation 6-2.

$$Electric\ Meter\ Energy\ to\ Primary\ Energy = \frac{kWh_{EM}}{\eta_{PP} * \eta_{TL}} \qquad \text{Equation 6-2}$$

**Table 6-3: Assumptions for Primary Energy, Energy Costs and Emissions.**

| **Natural Gas Power Plant** | **Camp Atterbury Rates** |
|---|---|
| Equivalent BTUs for a kWh: 3412 BTU/kWh | Electricity: 0.08 $/kWh |
| 2011 Natural Gas Heat Rate: 8152 BTU/kWh | Natural Gas: 0.47 $/therm |
| Power Plant Efficiency: 0.42 | Natural Gas: 4.85 $/1000 CF |
| 2009 T&D Losses + Unaccounted: 260,580,117 kWh | **IN Residential Rates** |
| 2009 Total Net Elec. Generation: 3,950,330,928 kWh | Electricity: 0.105 $/kWh |
| Transmission Efficiency: 0.93 | Natural Gas: 9.46 $/1000CF |
| **Natural Gas Furnace** | **EPA Emission Factor** |
| 2011 Heat Content of Natural Gas: 1012 BTU/CF | 120 lbs/1000 CF |
| Convert Natural Gas CF to kWh: 0.30 kWh/CF | |

***Results*** To reach the 19% primary energy reduction quoted in section 3, first calculate the sum of the primary energy of the CCHP for both buildings across all data sets, 6444 kWh (1618 kWh for Building 113 and 4826 kWh for Building 114). Next calculate the sum of the primary energy consumption for the NGF in both buildings across all data sets, 7961 kWh (3010 kWh for

Building 113 and 4951 kWh for Building 114). Then calculating the total primary energy savings by using the CCHP, 1517 kWh, and dividing by the total primary energy consumption of the NGF, 7961 kWh, arrives at 0.19 or 19% reduction of primary energy.

Table 6-1 illustrates that the CCHP in building 114 used less primary energy in five of the eight sets and Table 6-2 shows the CCHP in building 113 used less primary energy in all three data sets. Data sets 1 and 2 show more primary energy was consumed but the difference between the two systems is not very large, roughly 20-30 kWh. The outdoor condition ranges for both sets are comparable, 42°F to 68°F for the first set and 39°F to 65°F for the second set. Outdoor conditions for all data sets can be referenced by looking at Table 5-1. Both sets had extremes within a couple degrees of each other and therefore, had almost equivalent average outdoor temperatures.

Data set 6 also identifies the CCHP having more primary energy consumption than the NGF by roughly 500 kWh. This data set stands out as the longest amount of hours logged before any complications occurred, roughly 160 hours or almost 7 days. The outdoor conditions also were over a large range spanning a low of 19°F to a high of 67°F producing an average outdoor temperature of 41°F. If these three data sets were able to show equivalent primary energy consumption as the NGF, then the primary energy savings for all data sets would be at approximately 26%. The system controls play an important part in primary energy consumption of the CCHP. While for a NGF, the controls are more straightforward since cycling for a NGF does not create a large amount of inefficiencies. Data sets 1 and 2 are the first two data sets collected when the CCHP was installed. The significance of being the first sets collected is the CCHP controls were seeing the field demonstration conditions for the first time.

As problems were encountered, modifications were made to the controls which could result in a positive influence on the performance of the CCHP. The impact of the controls on the 6$^{th}$ data set could also explain the higher primary energy consumption seen of the CCHP. Over this long period of time the CCHP experienced outdoor conditions that were more extreme than what was observed during the first two data sets. With colder temperatures comes a higher building load and results in increased capacity demand. Colder temperatures also create conditions for frost build up on the outdoor coil and require very inefficient defrost cycles. This is due to all of the heating from the CCHP is done by electric resisters that have at best COPs of 1. The lower COPs of the CCHP mean more primary energy consumption to meet the same load. The colder temperatures also require the CCHP to run in two-stage mode more often to meet the larger building load. It has to switch between stages more frequently during this week period to conduct defrost cycles as well as handling the changing heating load throughout a typical day. Therefore in the 6$^{th}$ data set, the two-stage and defrost controls were utilized extensively having a large impact on the overall operation of the CCHP.

At first glance these results seem concerning for the future of CCHPs. However, as explained earlier; NGFs operate cyclically and a heat pumps optimal operation is to run consistently. The different operation designs can make it difficult to compare. This is further magnified by the graph below (Figure 6-3). Essentially, when a zone temperature is below its' set point, a NGF is designed to provide high temperature air to a zone until the temperature set point is met (cyclical operation) whereas, a CCHP is designed to provide low temperature air to consistently maintain a zones' temperature set point.

These differences in operation also affect the zone temperature values. Since, the NGF has higher discharge temperatures than that of a CCHP, its' zone temperature was generally higher than the CCHP's zone. Although the CCHP zone temperature was usually lower than the NGF, both zone temperatures were within the range of acceptable indoor conditions.

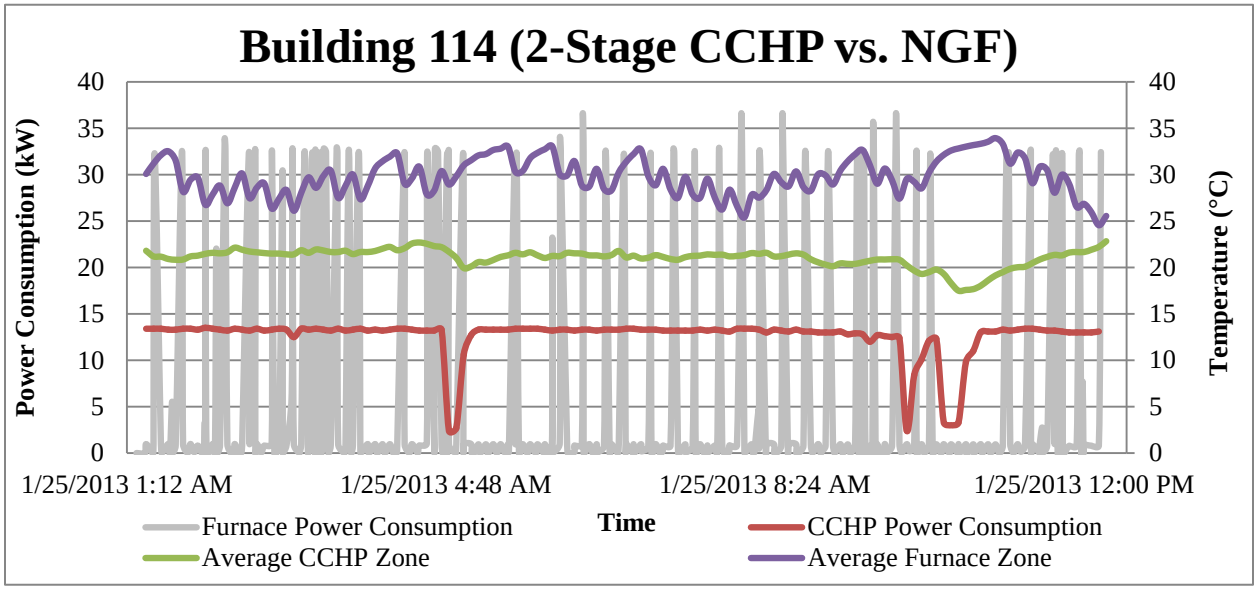


**Figure 6-3: Operation of a CCHP Compared to NGF.**

The gray line in Figure 6-3 represents the instantaneous power consumption of the NGF and the red illustrates the instantaneous power consumption of the CCHP. The green line that shows the average CCHP zone temperature illustrates the impact of the CCHP operation on the zone temperature, and the purple line shows the average NGF zone temperature. As expected the NGF burns natural gas in a very sporadic fashion (primarily caused by the high discharge temperatures of a NGF) and the heat pump remains steady except for three "dips" in power consumption. The first two dips are due to the oil equalization control strategy. If the CCHP operates continuously in two-stage for a long enough period it will shut down the compressors and open up a pressure and oil equalization valve. This allows the oil levels in the crankcases of the compressors to equalize to ensure the premature failure of a compressor does not occur.

Also, notice during each "dip" while the heat pump is not providing heat, the CCHPs' average zone temperature (green line) drops a few degrees (outdoor temperature had little fluctuation during this time period). Normally, the electric heat would come on during this time. However, during this test, it was manually disabled to see the sole effectiveness of the CCHP. It is expected that the indoor zone temperature will drop even more during defrost as the indoor heat exchanger is now acting as an evaporator. The furnace zone temperature is much higher than the CCHP. This is primarily due to the higher discharge temperatures of the NGF compared to the CCHP and the location of the temperature sensors are on the ceiling, where the supply vents are for both systems.

Table 6-1, Table 6-2, and Figure 6-3 do not provide the results of a completely fair comparison. Table 6-1, Table 6-2, and Figure 6-3 neglect the effects the orientation of a building can have on a zones' heating and cooling load. Therefore, further analysis was conducted and is presented in

Figure 6-1and Figure 6-2. In both figures, the CCHP located in building 113 and building 114 is compared to the NGF from both buildings. As a reminder, the CCHP in building 113 supplies the southern half and the CCHP in building 114 supplies the northern half. This comparison allows for insight on the influence of building orientation on the CCHP operation. In Figure 6-1, the primary energy savings changes depending on which NGF the CCHP is compared to. When compared to building 113 NGF, the CCHP has more primary energy consumption for data sets 4, 6 and 7, while comparing the CCHP to the NGF in building 114 shows more primary energy consumption for data sets 1, 2 and 6. In Figure 6-2, the CCHP always has a primary energy savings for data sets 9 and 11, but for data set 10, the primary energy savings only exists when compared to the NGF in building 113. These inconsistences could be due to the building orientation or if other influences are considered, could be due to differences in the occupant behavior in both buildings, the level of occupancy in both building, or an imbalance of occupancy between the individual buildings and/or their northern and southern halves. Overall, it is positive to note that the CCHP more often consumes less primary energy than either building's NGF.

## 6.2 COST

The financial performance objective was evaluated by taking the energy consumption of each system, the CCHP and NGF, and using utility rates to calculate the individual operating costs. The CCHP system required only the electricity rate while the NGF needed both a natural gas rate and an electricity rate. Two sets of utility rates, CAJMTC and residential, were used to generate two sets of cost comparisons that can be seen in Table 6-3. The rates for Camp Atterbury were obtained from CAJMTC staff and the residential rates were referenced from the 2011 EIA data book for Indiana. The historical trends of natural gas prices can be seen in Figure 6-4 which can be compared to the Camp Atterbury rate of 4.85 $/1000 CF and the Indiana 2012 rate of 9.46 $/1000 CF used in this study. This rate has dropped over the last couple years. Two different sets of rates were used to provide a comparison between the energy costs for a military base and for a typical residence. The electricity rates obtained were not the ideal rates due to the fuels used for power generation. In Indiana, electricity is predominately generated by burning coal while the primary energy conversion used in Section 0 for electricity assumes natural gas. Due to the inaccuracy in assuming a different utility rate, the actual costs for the energy consumed was used.

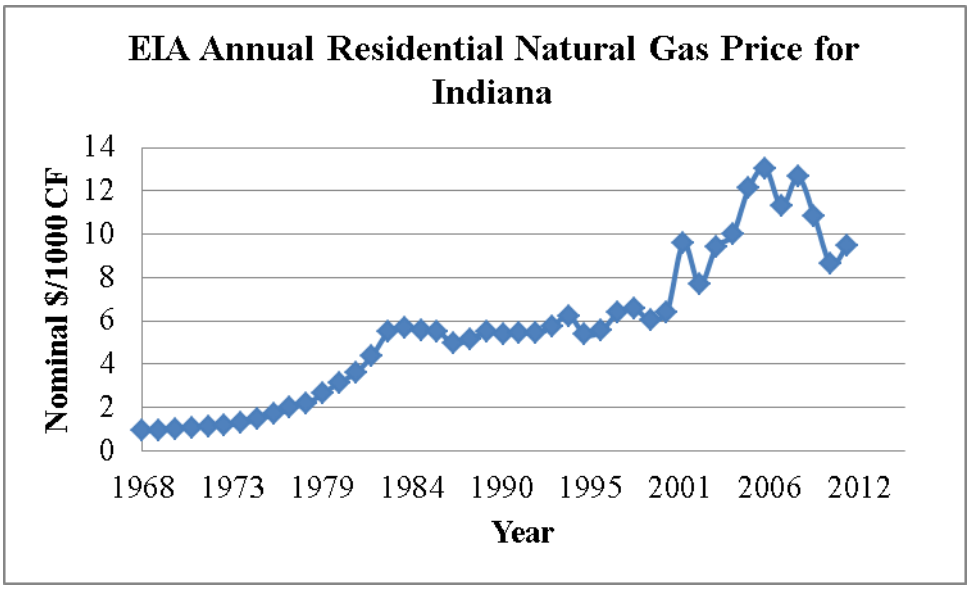


**Figure 6-4: EIA Database-Historical Trend of Indiana Residential Natural Gas Prices.**

The operating cost for the CCHP was approximately the same as the one of the NGF, 1% more, when using the residential utility rates. This is in spite of the CCHP consuming less primary energy, but is due to the low cost of natural gas. Residential customers currently are paying record low prices for natural gas. The operating cost for the CCHP at CAJMTC was about 44% more than the NGF. Commercial users like CAJMTC are paying even less for utilities because long term purchase agreements that are negotiated before the natural gas is used. Ultimately, the CCHP technology will be most competitive in locations where fossil fuel sources are not available. Although a heat pump will have significant energy savings as compared to an electric furnace.

## 6.3 $CO_2$ EMISSIONS

Previous studies have shown that heat pumps are only environmentally friendly from a $CO_2$ perspective when the electricity is produced by renewable means. However, for the field demonstration, the electricity used is assumed to be produced by a natural gas power plant. If the CCHP were to consume less primary energy than a NGF then, a CCHP would actually be better for the environment on a $CO_2$ basis. Please note that the previous statement merely refers to $CO_2$ production while the system is operating and does not imply the cradle to grave $CO_2$ production of a CCHP will be less. Additionally, further examination of different power plant fuels should be analyzed to provide a full representation of the CCHP emissions.

To calculate the $CO_2$ emissions produced by both systems required assuming the amount of $CO_2$ per volume of natural gas. An EPA report, AP-42, Compilation of Air Pollutant Emission Factors was referenced for this factor. The value can be seen listed in Table 6-3. The amount of primary energy consumed by each system is converted to a volume of natural gas. Both a kWh to BTU conversion and with the 2011 EIA heat content of natural gas for Indiana are used for the conversion. The emissions factor can now be used with the volume of natural gas consumed to generate the amount of $CO_2$ emitted as shown by Equation 6-3. By using the primary energy

consumption, both percent savings for primary energy and emissions are the same at 19%. For different locations, the electrical grid may be powered by a larger percentage of renewable power that lends to a smaller amount of $CO_2$ production per kWh of electricity.

$$Primary\ Energy\ kWh\ to\ CO_2 emissions = 1\ kWh * \frac{3412\ BTUs}{kWh} * \frac{1\ SCF}{1012\ BTUs} * \frac{120{,}000\ lbs\ CO_2}{1E6\ SCF} \quad \text{Equation 6-3.}$$

## 6.4 COMFORT

To evaluate the comfort performance objective, ASHRAE Standard 55, Thermal Environmental Conditions for Human Occupancy, was applied to quantitatively evaluate the room conditions. Compliance with the standard is estimated to predict at least 80% of the occupants are comfortable. The standard considers six primary factors that must be addressed when defining conditions for thermal comfort: metabolic rate, clothing insulation, air temperature, radiant temperature, air speed and the humidity.

The standard also offers different methods to evaluate these six primary factors for determining if compliance has occurred. The “Graphic Comfort Zone Method for Typical Indoor Environments” is selected to provide a visible indictor of the comfort conditions. In

Figure 6-5, a psychrometric chart from the standard is shown identifying the comfort zone with reference to an operative temperature and humidity ratio ranges. The graphical method can only be applied when the occupants have activity levels that result in metabolic rates between 1.0 and 1.3 met and when clothing is worn that provides between 0.5 and 1.0 clo of thermal insulation. The Appendices A and B of the standard offer estimates of these parameters. A sleeping person has a metabolic rate of 0.7 met and a seated person has a rate of 1.0 met. Therefore, this method will only be applicable to persons that are sleeping or at rest. Trousers, and a short-sleeved shirt has a rated clothing insulation value of 0.57, while a person with trousers, a long-sleeved shirt, long sleeve sweater, and undershirt has an insulation value of 1.01. Sweat pants and a long-sleeve sweatshirt combined have an insulation value of 0.74. For persons using a military barracks, typical activity and clothing is expected to adhere to the required ranges set by the graphical method.

The operative temperature is calculated by combining the effects of the air temperature and the radiant temperature. The radiant temperature is taken as the measured temperature of the surrounding walls and surfaces within a conditioned space and their position with respect to the person, Chapter 9: ASHRAE 2009 Handbook Fundamentals. Appendix C of the standard explains how the operative temperature is to be calculated. If the relative air speeds in the space are small (<0.2 m/s, 40 fpm), or the temperature difference between the mean radiant and air temperature is small (<4°C, 7°F), the operative temperature can be calculated from the average value of the air and mean radiant temperature. Alternatively, the operative temperature can be set equal to the air temperature if 4 conditions are met; no radiant and/or radiant panel heating or cooling system is used, the average U-factor of the outside window/wall is below a calculated value, the window solar heat gain coefficients are less than 0.48, and there is no major heat generating equipment in the space. Two of the four criteria are met automatically without any

calculations due to the known conditions of the barracks. The other two factors require calculations of the building materials that involve a significant level of uncertainty. For the scope of this analysis, it will be assumed the remaining two factors are satisfied and thus, the measured air temperature will be used as the operative temperature.

The air speed in the conditioned space was not measured. In spite of the absence of this data, the impact of air speed can still be considered.

Figure 6-5 identifies the impact of air speed on the allowable range of operative temperatures. The right boundary of the 0.5 clo zone corresponds to a 20 fpm air speed. As the air speed in the space is increased, the zone extends further to the right to include warmer operative temperatures. This increase reaches a limit at an air speed of 236 fpm shown as a dashed line in the figure. Further details on the air speed impacts are explained in the standard, section 5.2.3.

Before the air temperature and humidity measurements are considered, the raw data is processed for ease of interpretation by reducing the amount of points. Each building half, north and south, has temperature and relative humidity measurements resulting in four temperature and four relative humidity measurements. Seven of the original data sets have complete measurements of the indoor conditions. The data for each set is reduced to the maximum, average, and minimum values. With these values, the comfort range of each zone can be determined. If the average value lies within the comfort conditions, then the zone on average was adequately providing comfort to at least 80% of the occupants. The maximum and minimum values provide indication if the average value is affected by an extreme measurement that could be considered an outlier. Table C-1 in Appendix C provides a summary of these values for comparison.

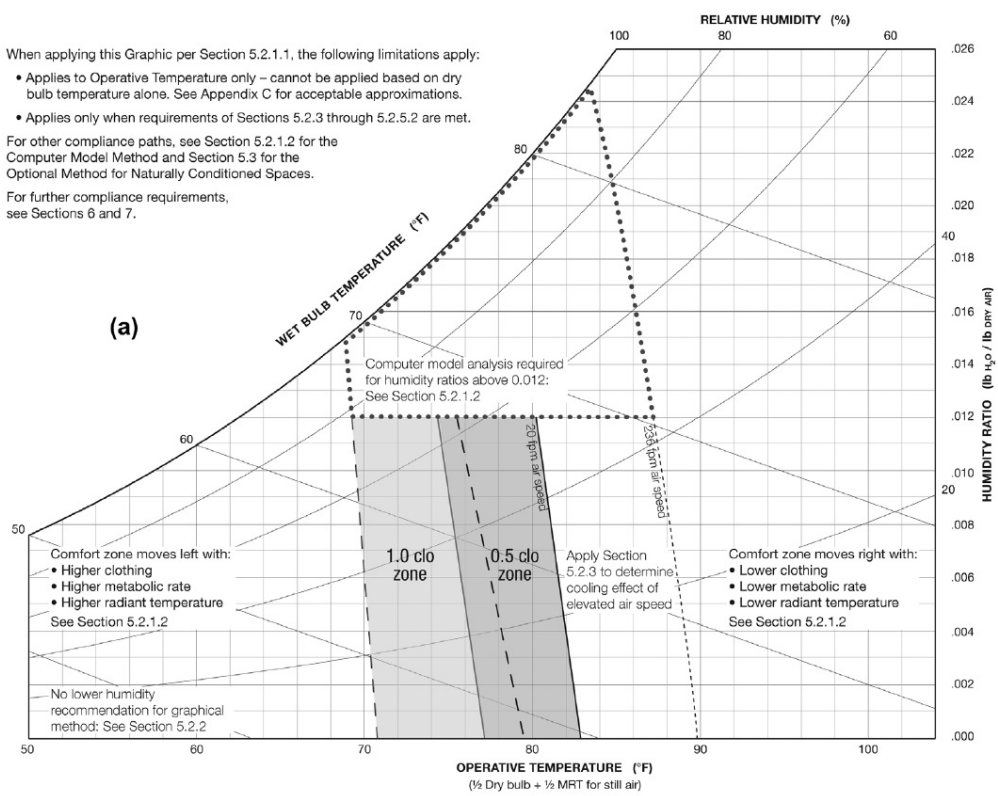

**Figure 6-5: Graphic Comfort Zone Method: Acceptable Range of Operative Temperature and Humidity for Spaces (I-P).**

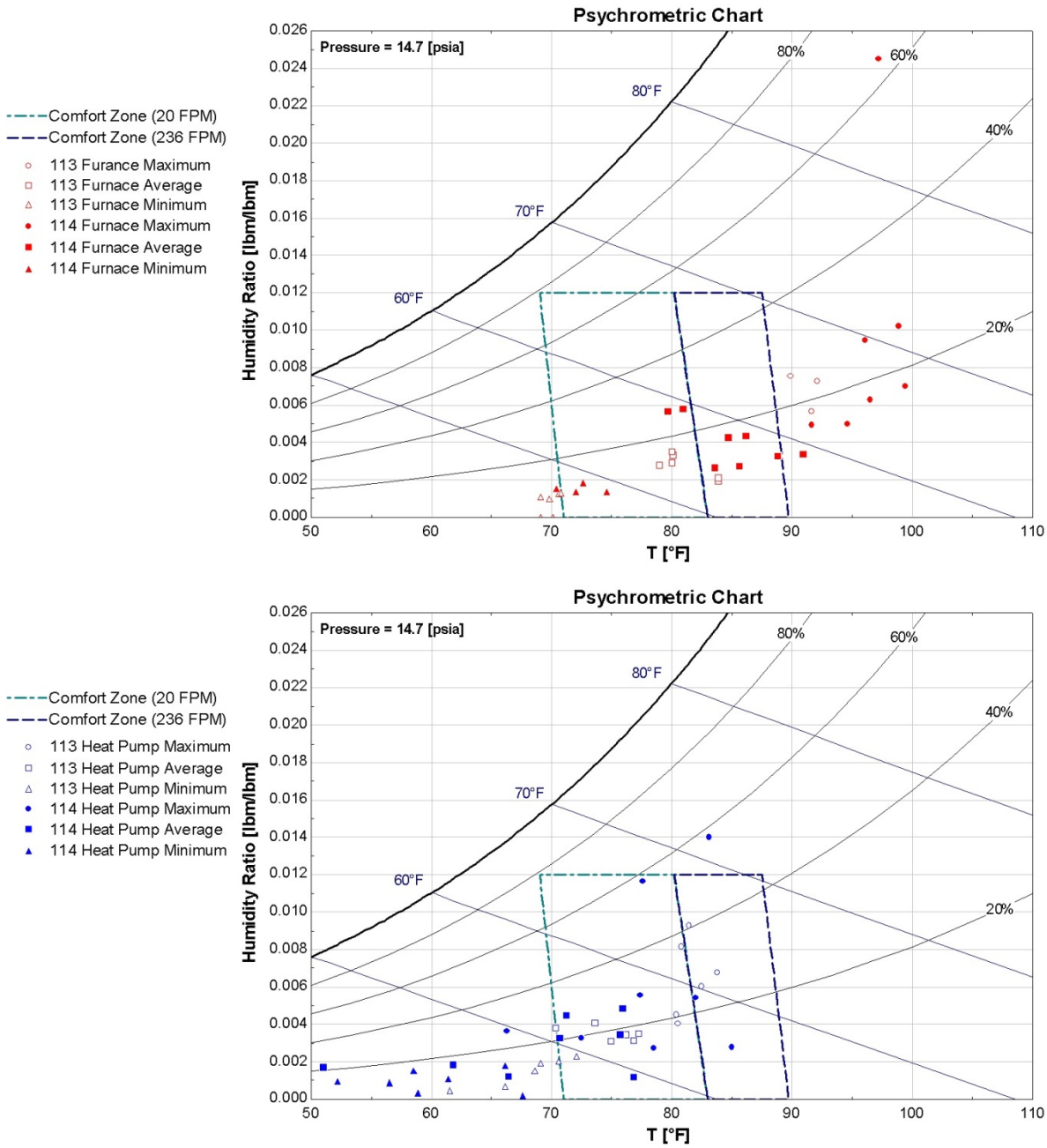


**Figure 6-6: Thermal Comfort Zone for Furnace (top) and Heat Pump (bottom) Operation.**

After obtaining the filtered, raw data, the humidity ratio is calculated using the air temperature and relative humidity. The humidity ratio with its respective air temperature is plotted onto a psychrometric chart. This chart can be seen in Figure 6-6. The two comfort zones are also plotted on the same chart to identify points that are outside the comfort zone. The points for building 113 are open symbols while the points for building 114 are filled in. The color of the point indicates if the point corresponds to the furnace operation, red, or for heat pump operation, blue. Lastly the

symbols for each point either refer to the maximum value, a circle, the average value, a square, or the minimum value, a triangle.

Overall the heat pump has lower temperatures than the furnace but 11 of the 14 heat pump averages lie within the comfort zone. These 11 average heat pump temperatures are still promising in spite of only 2 of the 14 heat pump low values satisfy the comfort requirements. The furnace has all the high values outside the comfort zone with approximately 50% of its averages within the faster air velocity requirement. As expected all points except two high values, one for the furnace, one for the heat pump, satisfy the humidity requirement. The reason being is only the heating operation is considered and the outside air during the winter can hold only a small amount of water vapor. This leads to lower indoor relative humidity during heating season. Another factor of the results to consider is the air temperature was measured on the ceiling of the conditioned space. Ideally, all measurements would be collected at specified heights to capture the building conditions within the occupied zone. Due to the duration of the field test and typical usage of the building, the collection of temperature and humidity from the ceiling proved to be the most dependable measurement location. The impact of the different discharge temperatures between the two systems could influence these results. A furnace using combustion as a heat source will have higher air supply temperatures than what a heat pump will provide. This can explain why there is a significant shift on the operative temperature axis between the two systems.

ASHRAE Standard 55 was satisfied for the majority of the maximum and average points for the heat pump. Almost all of the minimum values of the heat pump were outside the standard. It should also be noted that the furnace operation requires a higher air velocity to ensure adherence with the standard. All of the maximum points and roughly half of the average points for the furnace lie outside of the low air speed zone. And about half of the low points for the furnace were outside the low air speed zone completely. The lower supply temperatures of heat pumps are known and can cause user discomfort if the air distribution system is designed improperly. From the results, it can be concluded that the heat pump will satisfy 80% of the occupants a majority of the time. Potentially during start-up, the room conditions fall outside this comfort zone for a short period of time. One area that would need to be investigated further is the control of the system as its start-up procedure could eliminate this system characteristic.

The barracks which hosted the field demonstration were split into halves. Referring back to Figure 4-4, each half was partially separated by a wall and thus, the barracks were essentially split into quarters. Temperature and relative humidity sensors were installed in the center of each quarter of the barracks (as shown in Figure 6-7).

**Figure 6-7: Temperature and Relative Humidity Sensor Installed in One a Quarter.**

The red circle in Figure 6-7 reveals that these sensors are actually installed on the ceiling. This is not an ideal measurement location due to the temperature gradient from the floor to the ceiling. However, for reliability purposes it was the ideal location as it had the least chance of getting damaged.

## 6.5 INSTALLATION

The installation of the CCHP was performed by a HVAC contractor, who was familiar with installing heat pumps. The main difference for an installation between a CCHP and a traditional air-source heat pump was an additional component. Both systems utilize an indoor and outdoor unit but the CCHP had a compressor housing unit that contained all the compressors and auxiliary equipment needed. This required four extra piping connections compared to the traditional system. The connections were no different than standard piping connections used on any refrigeration system, including heat pumps. Also, the compressor housing required standard electrical connections which are additional compared to a traditional heat pump. The HVAC contractor had no difficulty handling these additional piping and electrical connections.

The indoor and outdoor units for the CCHP were off the shelf components that are currently used in a pre-packed air-source heat pump. More space is needed than traditionally available in outdoor units for the CCHP. Ideally the outdoor unit would be slightly redesigned creating enough space to contain all the essential equipment needed for the CCHP. With this small change, the installation of the CCHP becomes almost identical to a traditional air-source heat pump.

## 6.6 MAINTENANCE

Most maintenance of air-source heat pumps involves charging refrigerant, replacing fan motors or blades, replacing a compressor or replacing a bad sensor. All of these tasks could be completed by a HVAC technician on the CCHP with minimal difficulty. The redesign of the outdoor coil would need to take into account allowing the ability for the HAVC technician to conduct any maintenance on the CCHP.

Tasks that would be considered additional maintenance compared to a traditional heat pump were not identified during the field demonstration. More installations would be needed to determine if different maintenance tasks are required by the CCHP. Repairs that were conducted as a result of the complications with the CCHP were not considered for the maintenance performance objectives. Section 8.0 outlines the implementation issues of the CCHP and the repairs needed to solve and prevent these problems.

## 6.7 SEASONAL PERFORMANCE RATINGS

For air-source heat pumps, there are three different ratings for measuring its seasonal performance; the seasonal energy efficiency ratio, SEER, determines the cooling performance over the entire cooling season, the heating seasonal performance factor, HSPF, determines the heating performance over the entire heating season, and the seasonal coefficient of performance, seasonal COP, provides the heating or cooling performance for either the entire heating or cooling season. All three require the amount of energy that was removed from the building for cooling or the amount of energy delivered for heating the building and the amount of electricity consumed to achieve said cooling or heating. The difference the SEER or HSPF and the seasonal COP are units used for energy and electricity. The seasonal COP uses the same units for both energy and electricity while SEER or HSPF use BTU for energy and W-h for electricity. A SEER rating cannot be calculated for the CCHP because the field demonstration was conducted during the heating season.

ANSI/AHRI Standard 210/240 sets the requirements to calculate the HSPF for a new air-source heat pump without running the new equipment for an entire heating season. Various outdoor conditions are to be tested while at steady-state and if necessary with variable outputs, during different system capacities. For the field demonstration, testing the CCHP at these rating points is not required because the actual performance is calculated. Data was to be collected over an entire heating season and used to calculate a value similar to the HSPF. This is done by summing all the energy delivered to the building, in BTUs, during the heating season and dividing this sum by the total amount of electricity consumed, W-h, during the same period. These units are the same used for the HSPF to have a rough baseline comparison. The value calculated using experimental data cannot be labeled as a HSPF in order to avoid confusion with HSPF ratings of off-the-shelf equipment. Due to the complications encountered during the project, an entire heating season worth of data could not collected. In spite of this limitation, the data collected can be used to update the inputs of a building modeling program to generate an experimentally adjusted, simulation heating seasonal performance.

***TRNSYS*** Taking the EES results from section 2.2 further, a complex building model was developed within TRNSYS, Transient System Simulation Tool. TRNSYS provides the

capabilities to simulate systems over an entire year on an hourly basis or any other time step desired. With a weather data file, a complex building model, and the EES simulation results, a TRNSYS simulation can model a field demonstration. The location of the building can be changed by referencing a different weather file. A blower that is operated by a programmable controller can be evaluated by changing the controller logic. Thermal losses of supply and return ductwork can be quantified. All these factors can be connected together and modeled within TRNSYS. Figure 6-8 presents the overall structure of the TRNSYS model used to simulate the field demonstration. The same load schedules that were generated by the eQUEST model were used for the occupant, lighting, and infiltration building loads. The building was modeled as a multi-zone space and is defined by a preprocessor program, TRNBuild that is available in the TRNSYS software package. All the building details such as dimensions and materials are defined within this preprocessor program. The blower, supply and return air ducts were obtained from built-in functions already available in TRNSYS.

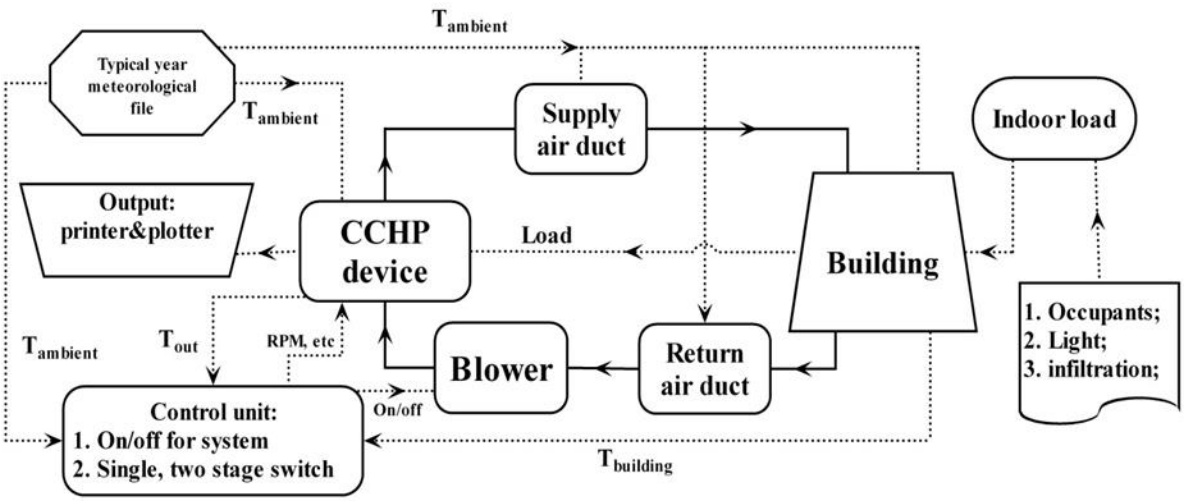


**Figure 6-8: TRNSYS Model Layout of CCHP Field Demonstration.**

***CCHP Device*** The CCHP device is a custom function that was developed in FORTAN and compiled to be called by the TRNSYS model. The custom function is preprogrammed using curve-fits that are generated using the EES model results shown previously in Figure 2-2 and Figure 2-3. Seven curve-fits that provide the CCHP capacity as a function of temperature are obtained. Five are for different compressor speeds during single-stage operation, and the remaining two are for two different speeds during two-stage operation. The same number and conditions of curve-fits are also obtained for the COP of the system as a function of the outdoor temperature.

The building load is used an input for the CCHP function and is compared against the calculated, available capacities of the system. If the building load is between two available capacities, it is determined that the CCHP output can match the building load. If the building load is between two capacities that involve both two-stage and single-stage operation, the higher capacity is selected as the CCHP output. The set compressor speed is calculated by interpolating between the two speeds that correspond to the capacities that surround the building load. If the load is between capacities of different modes, then the speed is set to the mode corresponding to the higher capacity.

The selected COP is calculated the same way. If the building load is between the lowest two-stage capacity and the highest single-stage capacity, then the CCHP output capacity is selected as the two-stage capacity. This decision will lead to the CCHP producing more heat than is necessary, but ensures the building load is always met or exceeded. In typical operation, the CCHP cycles between single and two-stage operation during these conditions. The calculated COP is used with the calculated CCHP capacity to back-out the electric consumption of the CCHP.

The TRNSYS simulation was run for the entire year to observe fluctuations of the heating and cooling loads compared to the outdoor temperature. The heating set point was 68°F (20°C) and the cooling set point was 74°F (23°C) for the building. The simulation results of the building load can be seen in Figure 6-9.

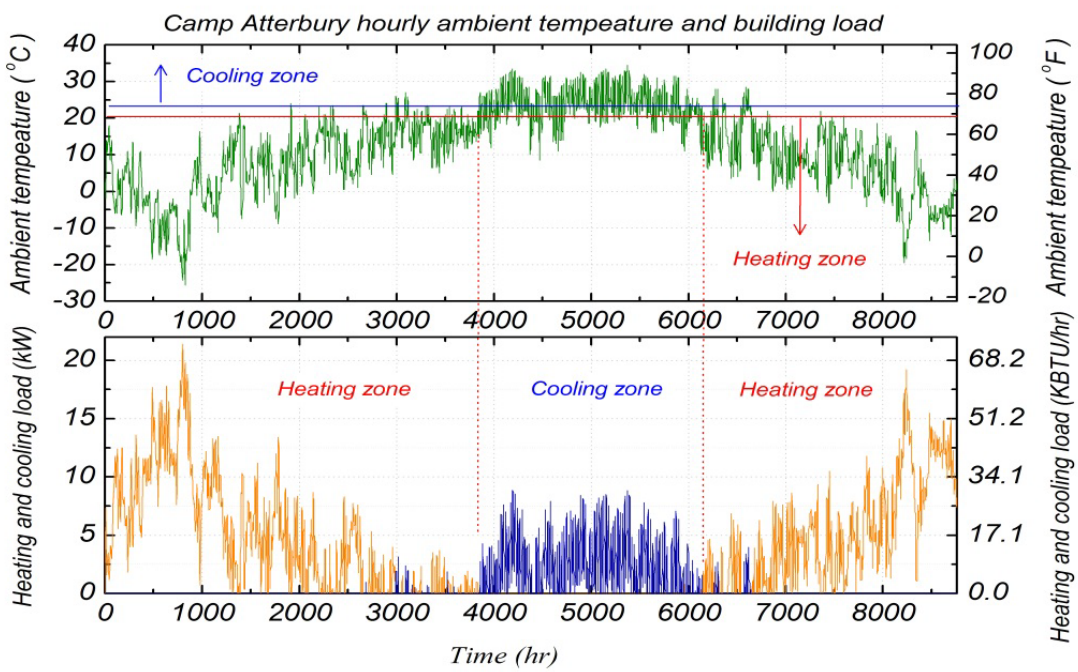


**Figure 6-9: TRNSYS Model Building Simulation – Heating and Cooling Loads.**

***Simulation Results*** The heating season for the field demonstration is shown to be from September to April. The entire TRNSYS model, with the CCHP device, is run during only the heating season to predict the monthly and overall heating season performance of the CCHP. The monthly heating COPs and electric consumption of the CCHP can be seen in Figure 6-10. The overall heating seasonal COP from these results was calculated to be 3.75 which corresponds to a heating seasonal performance of 12.8 BTU/W-hr. A conference paper was published that presents the use of the TRNSYS model to predict the operation of the CCHP during the field demonstration, Caskey et al., 2012a.

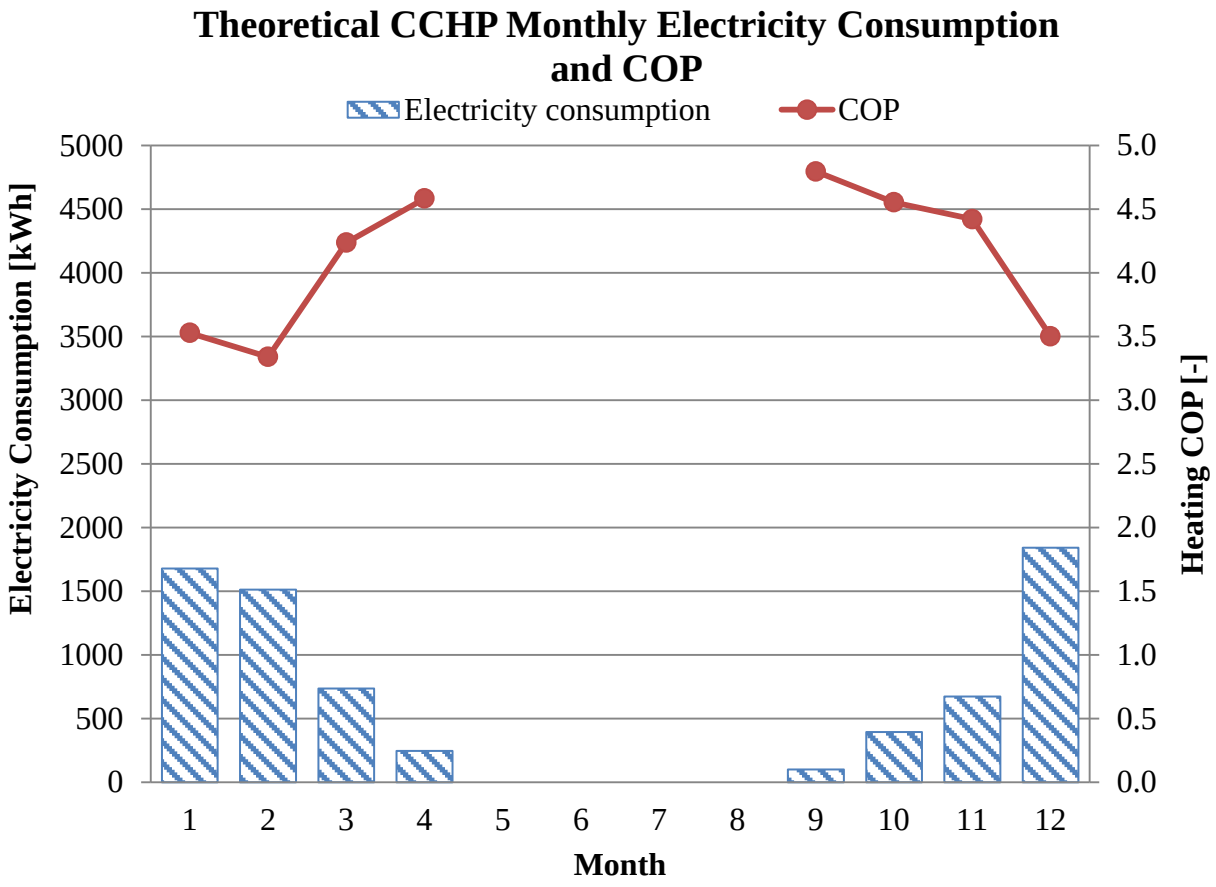


**Figure 6-10: TRNSYS Model – Theoretical Monthly CCHP Electric Consumption and Heating COP.**

***Experimental Improvement*** With a complete building model, improvements on the simulation results could be made by modifying references used by the CCHP device to better match the experimental data. The CCHP device references equations from curve fits that were generated using points from the EES simulation results. These original fits were made for both the CCHP heating capacity and the COP as a function of ambient temperature. This process can be seen in Appendix D.

The new curve fits are loaded into the CCHP device in the TRNSYS simulation program. The monthly electricity consumption and the monthly seasonal heating COP are plotted only for the months during the heating season. An updated version of the plot in Figure 6-10 can be seen in Figure 6-11. Overall the newly predicted electricity consumption is more than double the original prediction. This is due to the newly predicted COPs being less than the original simulation prediction. The new simulation predicts a seasonal heating COP of 2.25 or a heating seasonal performance of 7.7 BTU/W-h. The HSPF requirement for the Department of Energy's appliance Energy Star rating is a level of 8.2 or above. To accurately compare the CCHP to existing HSPF ratings, the system would need to be tested in a laboratory to find its HSPF rating or a conventional heat pump would be tested over an entire heating season at the testing site to provide a measured heating seasonal performance.

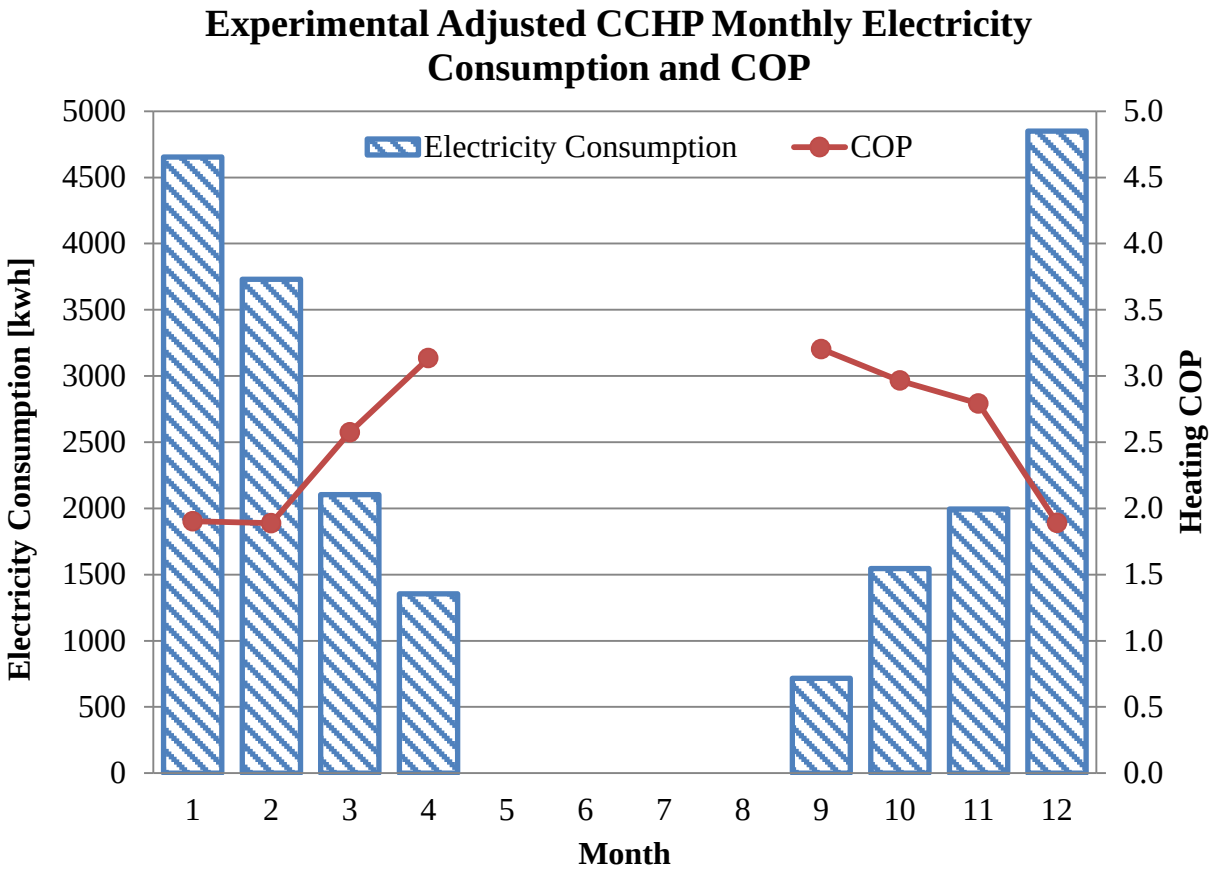


**Figure 6-11: Experimentally Adjusted TRNSYS Model – Monthly CCHP Electric Consumption and Heating COP.**

# 7.0 COST ASSESSMENT

While operating costs are important, that information, by itself, fails to give a complete picture of the savings potential because other factors must be considered. A fully developed cost model must also include the first cost of the technology, installation costs, ongoing maintenance, and other factors.

One challenge that the university/industry team faces right now is that information for the life cycle cost assessment is incomplete. What has been built and tested so far is a functional prototype of a CCHP. The equipment is heavily instrumented and is not optimized in terms of the components or functional layout. The cost information will change considerably as the device is value engineered to minimize parts, minimize assembly time, and create a strategy for long term maintenance.

Pre-commercialization work with the CCHP is going on now as part of a new partnership with Unico, Inc. of St. Louis, Missouri. Unico was recently awarded a two year contract from the U.S. Department of Energy for the commercialization of a CCHP. Unico has licensed the two CCHP patents owned by Purdue University and has signed a research agreement with Purdue's Herrick Labs to support ongoing work towards full commercialization. With this information on the current status of the heat pump in mind, this cost assessment will look into the future, when Unico has fully commercialized Purdue's heat pump technology. The projection will be based on Unico's expectations for performance and cost.

Unico is a small, privately-owned business that makes a variety of unique HVAC products including innovative small duct, high velocity air handlers and ducts along with chillers and heat pumps. Unico has metal manufacturing equipment and the ability to turn raw sheets of galvanized and stainless steel into our blower and coil cabinets. Unico also has a culture of continuous improvement. In 2007, Unico vertically integrated the manufacturing of sheet metal cabinets, building another production facility in Arnold, MO, to set up with the latest metal fabrication machines including a Bystronic laser. They now control the delivery and quality of critical components and in June 2013, Unico, Inc. received ISO 9001:2008 certification.

## 7.1 COST MODEL

The market for CCHP consists includes both residential and light commercial customers. The market includes both new construction and retrofitting existing buildings. The market focus for the Unico CCHP would primarily be in the U.S. northern climatic zones five, six, seven, and eight, but the CCHP has the potential to be sold as a heat pump anywhere throughout the U.S. and Canada.

Table 7-1 has the 2011 Air Conditioning, Heating and Refrigeration Institute (AHRI) data showing the number of outdoor condensing units shipped and sold. Based on this data Unico will produce sizes 3, 4, and 5 Ton CCHP with capacities ranging from 36,000 BTU to 65,000 BTU.

**Table 7-1: Condenser Sales for 2011.**

| Condensing Units Sold | 3 ton | 4 ton | 5 ton |
|---|---|---|---|
| | 1,175,000 | 637,000 | 572,000 |

AHRI data shows that there has been a continual increase in the number of heat pumps sold each year. Unico predicts that this trend will continue into the future as homeowners adopt the use of heat pump technology as the main source of heating and cooling for their homes. Unico also believes that as the 30% tax rebate for ground source heat pumps expires in 2016 the sales of air source heat pumps will increase even higher due to the higher cost associated with ground source heat pumps and the discontinuation of the rebates.

## 7.2 COST DRIVERS

A key to success of Unico's CCHP product is to develop a relationship with the National Rural Electric Cooperative Association - Cooperative Research Network (NRECA-CRN), and other electric utility companies across the country, in an effort to partner with them on testing equipment at different locations throughout the United States and Canada. It will also be important to work with these particular organizations in an effort to secure rebates and incentives to offset the cost of Unico's CCHP. These rebates will allow a homeowner to realize a quicker ROI and shorter cost/payback on the CCHP product.

The total estimated US market for residential and light commercial units between 3 and 5 tons is approximately $212 billion and 2,384,000 units per year. Once the CCHP product is ready for full commercialization Unico will utilize its existing channels of distribution of 219 HVAC wholesale distribution partners with over 900 locations in the US and Canada and Unico's HVAC manufacturers' representatives (28 manufacturers' representatives covering the entire US market and parts of Canada), to sell and stock the product out into the market space. Unico will develop an American Institute of Architecture (AIA) continuing education units program to educate architects on the effectiveness of CCHP in colder climates. Unico will also develop a Lunch and Learn program for the engineering community to educate engineers on the technical aspects and benefits of a CCHP.

In order for air-source heat pumps to become serious contenders for use in colder climates, significant changes must be made for them to realize their true potential. Theoretical equations of efficiency, such as Carnot and Lorenz, show very high COPs. However, a practical system must include the temperature difference between the refrigerant and air, fan penalty, and compressor efficiency. These are compared in the Figure 7-1 and show that the proposed level (blue solid line) of COP is within reach from a practical point of view.

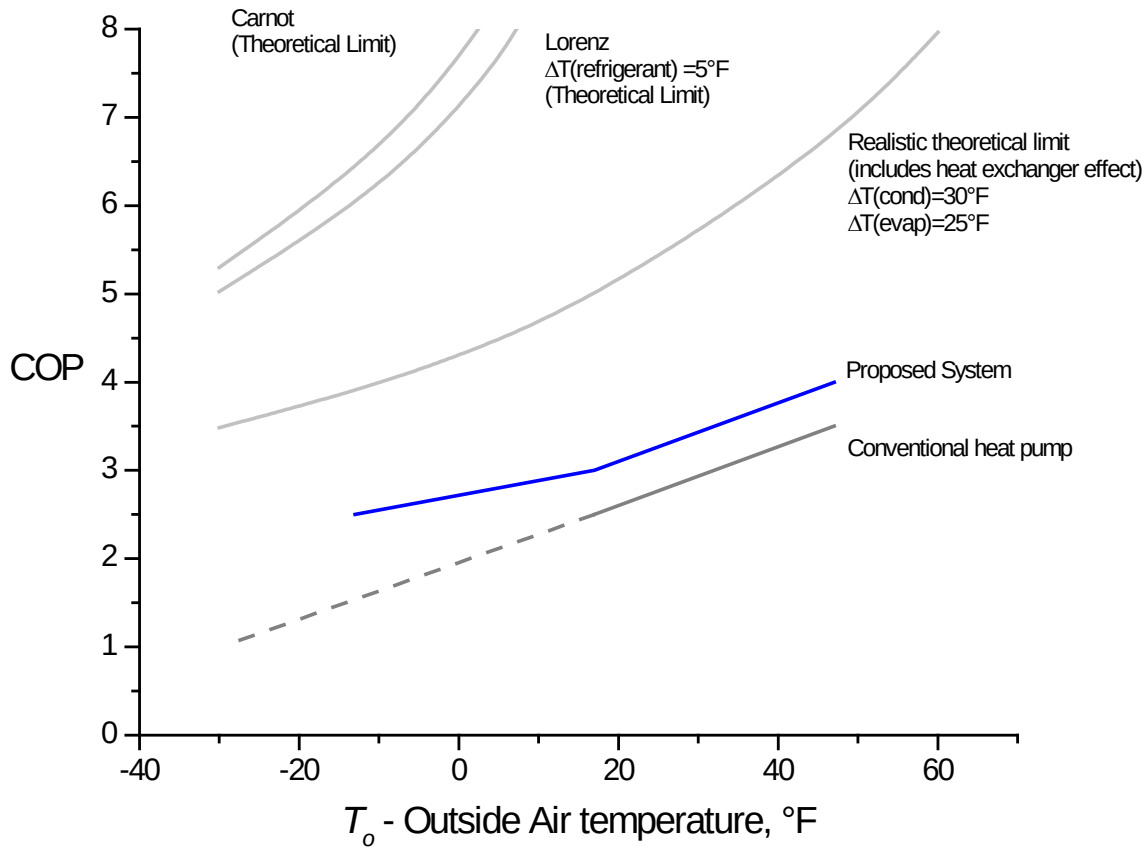


**Figure 7-1: Unico Projections for Performance of Commercialized CCHP System.**

In real systems, the time averaged COP (seasonal) is less than steady state because of cycling and defrost degradation. Unico's projections for a seasonal COP of approximately 3.0 from Figure 7-1 is higher than the COP of 2.5 that was achieved in this field demonstration. However, we believe that Unico's COP projections are achievable once improvements are made to the components and controls of the prototype CCHP systems that were tested at Camp Atterbury. For purposes of comparison, the COP for a conventional heat pump averages 1.0 below freezing.

## 7.3 COST ANALYSIS AND COMPARISON

Table 7-2 shows the different types of energy costs listed by NRECA, National Rural Electric Cooperative Association, "Heating Fuel Cost Per Million Btu". Electricity is compared against natural gas, propane, and fuel oil. The cost projections for the Unico CCHP are included in Table 7-2 at two levels of performance, low and high, which illustrate the COP ranges summarized in Figure 7-1.

With the two exceptions of geothermal heat pumps and extremely low priced natural gas, the Unico residential CCHP solution will outperform every other energy category. To further enhance the cost/payback model Unico plans to work with the NRECA and other national utilities to secure rebates and cash incentives for the homeowner to purchase and install a Unico CCHP.

Estimated balance of materials cost show that Unico will have the ability to sell a 3 ton CCHP at a market competitive price that will yield a short pay back estimated at less than 5 years. With an average fuel oil heating bill for a 2500 square foot building in a severe winter climate zone of $1200, compared to the Unico CCHP (High) of $562, the annual average savings from the Unico CCHP (High) would be $638/year.

**Table 7-2: Heat Cost Comparison.**

| Heating Fuel Cost per Million Btu | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| **Fuel** | **Cost** | **Unit** | **Energy Content** | **Units** | **Fuel Cost/ Million Btu** | **Heating System** | **Heating System Efficiency** | **Heating Cost / Million Btu Delivered** |
| **Electricity** | $0.07 | kWh | 3,413 | Btu/ kWh | $20.51 | Ducted Resistance | 100% | $20.51 |
| | | | | | | Heat Pump (normal) | 178% | $11.52 |
| | | | | | | Unico CCHP (Low)* | 262% | $7.83 |
| | | | | | | Unico CCHP (High)* | 315% | $6.51 |
| | | | | | | Heat Pump (Geo) | 320% | $6.41 |
| | $0.10 | kWh | 3,413 | Btu/ kWh | $29.30 | Ducted Resistance | 100% | $29.30 |
| | | | | | | Heat Pump (normal) | 178% | $16.46 |
| | | | | | | Unico CCHP (Low)* | 262% | $11.18 |
| | | | | | | Unico CCHP (High)* | 315% | $9.30 |
| | | | | | | Heat Pump (Geo) | 320% | $9.16 |
| | $0.16 | kWh | 3,413 | Btu/ kWh | $46.88 | Ducted Resistance | 100% | $46.88 |
| | | | | | | Heat Pump (normal) | 178% | $26.34 |
| | | | | | | Unico CCHP (Low)* | 262% | $17.89 |
| | | | | | | Unico CCHP (High)* | 315% | $14.88 |
| | | | | | | Heat Pump (Geo) | 320% | $14.65 |
| **Natural Gas**** | $1.00 | therm | 100,000 | Btu/ therm | $10.00 | Force Air, Typical | 70% | $14.29 |
| | | | | | | Force Air, Condensing | 90% | $11.11 |
| | $1.50 | therm | 100,000 | Btu/ therm | $15.00 | Force Air, Typical | 70% | $21.43 |
| | | | | | | Force Air, Condensing | 90% | $16.67 |
| **Propane **** | $5.30 (about $1.85 /gallon) | ccf | 252,400 | Btu/ cff | $20.99 | Force Air, Typical | 70% | $29.99 |
| | | | | | | Force Air, Condensing | 90% | $23.32 |
| **Fuel Oil**** | $2.60 | gal | 130,000 | Btu/ gal | $20.00 | Force Air, Typical | 70% | $28.57 |

# 8.0 IMPLEMENTATION ISSUES

Current CCHP technology continues to transition from a successful laboratory experiment to a commercially viable product. This chapter describes the most significant issues that were encountered during the field demonstration but were not fully resolved. These key issues include returning oil to the compressor, liquid flooding, subcooling and most importantly system control.

## 8.1 RETURNING OIL

Modern scroll compressors discharge a very small percentage of their oil at the refrigerant outlet. However, if the discharged oil is not returned to the compressor it will eventually result in the compressor seizing due to a lack of lubrication. To combat this, an oil separator was installed at the discharge of each compressor. An oil return flow schematic can be seen in Figure 8-1. When enough oil accumulates in the oil separator a float valve allows oil to flow through a filter and then back to the suction of the compressor.

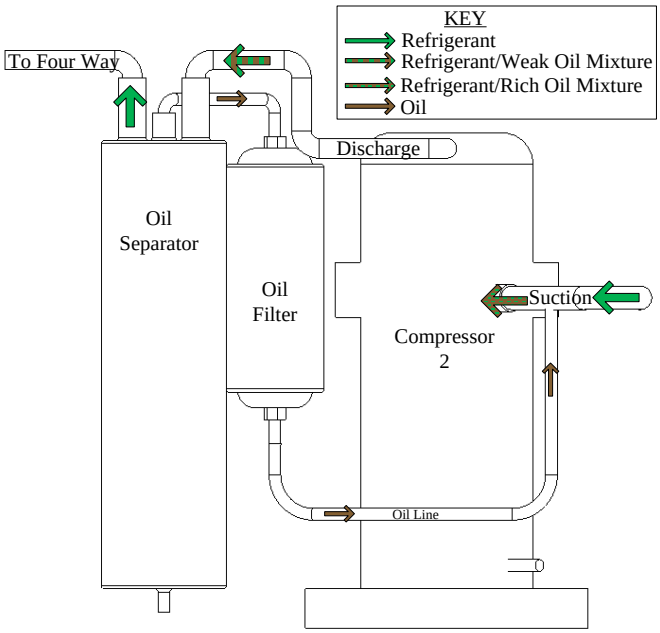


**Figure 8-1: Oil Return Flow Schematic.**

Compressor 2 was a variable speed compressor that allowed precise heat delivery and increased system efficiency. Compressor 2 can operate between 1800 and 7000 RPM. The variable speed compressor had issues when operating at 1800 to 3600 RPM, speeds too low to capture oil from the oil line. Operating at these low speeds for a long duration will eventually lead to the same result as if an oil separator were not present (i.e. the compressor will be starved of oil and seize). In addition to low operating speeds the accumulator utilized was very large. The large accumulator was required due to the additional refrigerant charge required however; large accumulators have large pipe diameters. The large suction pipe diameter resulted in an even lower refrigerant velocity, amplifying the effect.

The solution to the oil return problem required control logic and physical modification. Both solutions proved effective and were quite simple. A simplified version of the logic to solve the oil return problem can be seen in Figure 8-2.

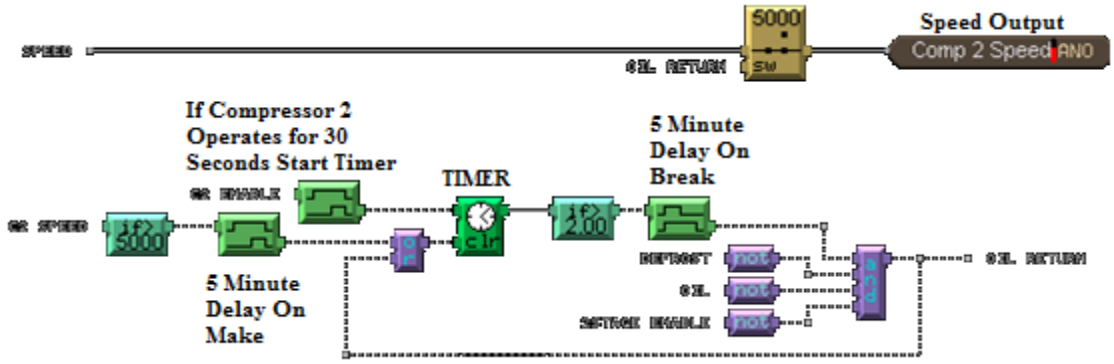


**Figure 8-2: Simplified Oil Return Logic.**

Once, the compressor has operated for thirty seconds, the oil return program records the amount of time that the compressor has operated. If it operates lower than 5000 RPM for more than two hours then it enables the oil return operation unless the system is in two-stage operation, undergoing a defrost cycle, or is in the oil equalization sequence. When the oil return operation is enabled the compressor speed is set to 5000 RPM regardless of what the rest of the control program sets the speed to. When oil return is enabled it clears the timer which would disable the oil return operation but, there is a five minute delay on break. This delay on break allows the timer to be cleared while still operating compressor 2 at oil return speeds for five minutes. If the traditional control program commands compressor 2 to operate above 5000 RPM for at least five minutes then the oil return timer is cleared, eliminating unnecessary oil return operation.

The oil return problem was not only a cause of low refrigerant velocities but the layout of the suction line plumbing also caused oil to be “trapped” in the system. The original fabrication of the suction line can be seen in Figure 8-3.

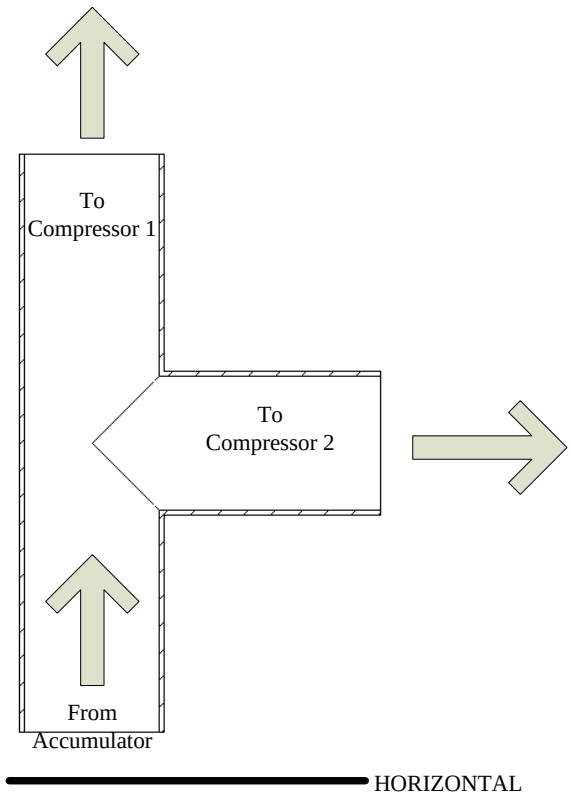


**Figure 8-3: Suction Line Piping (Before).**

During single stage operation, compressor 2 pulls a refrigerant/oil mixture from the accumulator. However, the density of the oil is much greater than the vapor refrigerant which tends to cause the oil to "overshoot" the suction line to the compressor. Switching compressor 2's suction line to the vertical component of the tee fitting would solve the problem but the oil "overshooting" problem would then be present during two-stage operation.

A solution that was successfully implemented was to rearrange the tee fitting. This change is illustrated in Figure 8-4.

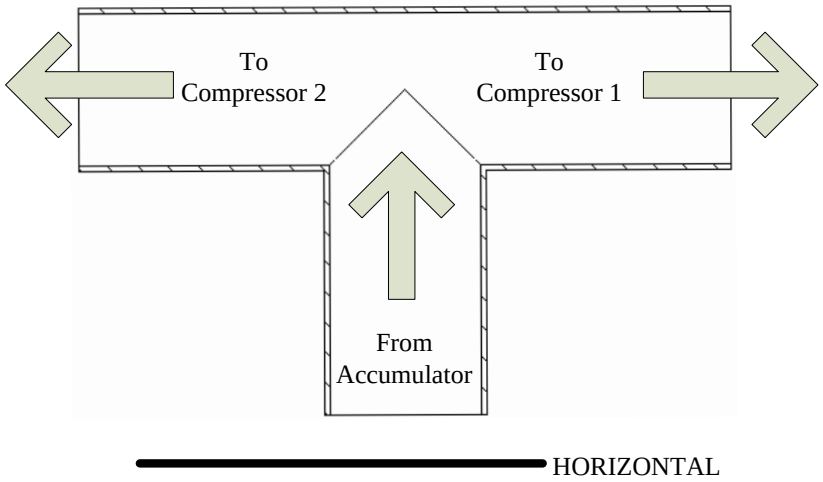


**Figure 8-4: Suction Line Piping Changes.**

The layout above allows the high density and high velocity oil to hit the wall of the tee fitting. This obviously slows down the oil allowing it to be drawn into the suction line of the compressor and overshooting the suction line is near impossible. This solution solves the oil return piping problem in single stage as well as two-stage operation. Please note that in reality the tee fitting is perpendicular to the horizontal plane so the oil will not fall back down into the accumulator via gravity.

## 8.2 LIQUID FLOODING

As the name suggests compressors are designed to intake a vapor and discharge a vapor at a higher pressure. However, R410A is a refrigerant that can exist as a vapor/liquid mixture at ambient temperatures. This can result in "refrigerant migration" while the system is idle. Refrigerant migration is when liquid refrigerant travels to the lowest temperature part of the system while it is not operating. Refrigerant migration can cause slugs of liquid to enter the compressor upon start-up and damage it. Due to the compressors residing in a fairly well conditioned zone, refrigerant migration was not an initial concern. However, refrigerant migration started to become present further into the field demonstration.

The refrigerant migration obstacle was also overcome with a physical and programming change. The physical change required installing crankcase heaters which are commonly found on commercially available heat pumps and the programming change was a "pump down" strategy. Crankcase heaters are small electric resistance heaters (under 100W) that wrap around the outside of compressors crankcase to drive out liquid refrigerant during compressor inactivity.

A "pump down" operation involves closing the primary expansion valve while letting the compressor operate for a short time during CCHP shut down. Also, when starting up the CCHP the primary expansion valve remains closed while the compressor starts up. After a short duration of the compressor running the expansion valve opens. The "pump down" strategy traps the liquid refrigerant on the high pressure side of the system to reduce refrigerant migration while the system is idle.

Liquid flooding can also be present during heat pump operation and is primarily caused by an incorrect amount of refrigerant charge. According to Mark Stansbury, owner of a HVAC service company; charging the correct amount of refrigerant is the most significant problem with heat pump and air conditioning installations (M. Stansbury, personal communication, January 7, 2013).

The issues encountered with charging a heat pump is trying to achieve sufficient subcooling (5°K) and superheat (5°K). It's the responsibility of the expansion valves to maintain the proper superheat however; if the system is undercharged it may be impossible to do so. Generally to ensure the desired subcooling, charge is added until the liquid line sight glass is fairly clear of bubbles. The problems encountered when charging the CCHP was the terrible subcooling seen at the liquid line sight glass. Therefore, additional refrigerant charge was added. The addition of charge resulted in flooding the accumulator with liquid. This caused liquid refrigerant to enter the suction of the compressor. Eventually, the liquid refrigerant flushed away oil in the compressors' crankcase which led to premature compressor failure.

A solution that has not yet been implemented is to have only one oil separator and connect it to the discharge of compressor 2. Then the oil line would return to the inlet of the suction line accumulator. The accumulator would need to be pre-charged with oil to a level above the oil orifice. This would ensure any oil discharged by compressor 2 is returned to the accumulator to be sent to the suction of either compressor 2 or 1. An example is illustrated in Figure 8-5.

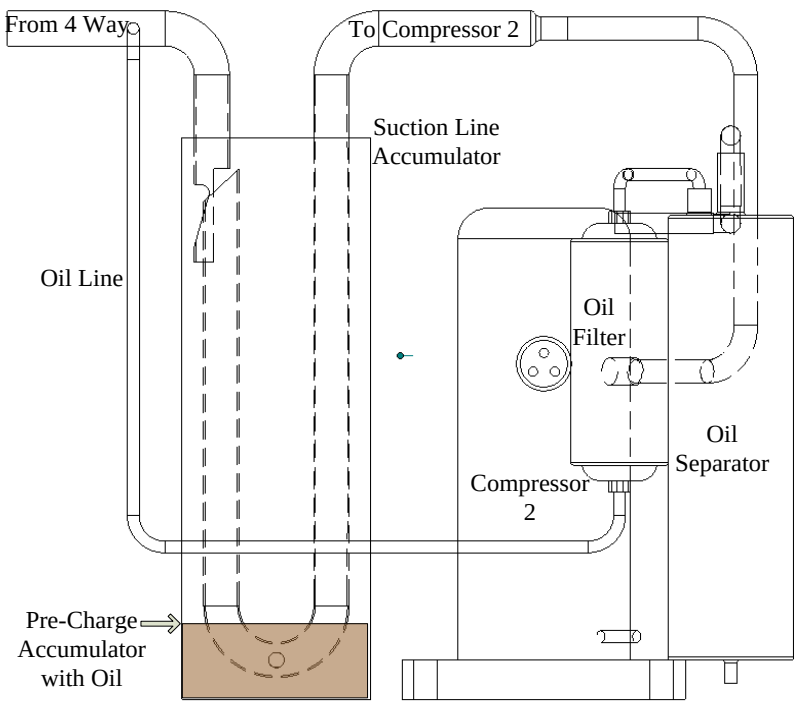


**Figure 8-5: Liquid Flooding Solution.**

Research performed by Yun, K. W. in 1998 presented that the prior solution did extend compressor life. Returning oil to a pre-charged accumulator ensures that if any liquid refrigerant does enter the suction line of the compressor that it would be saturated with oil. Please note that during two-stage operation there must be an oil equalization procedure to balance out the oil levels between the two compressors if this solution were to be implemented.

## 8.3 SUBCOOLING

Ideally, a heat pump has at least 5 Kelvin of subcooling at the inlet to an expansion valve. The CCHP consistently lacked sufficient subcooling. Without sufficient subcooling the primary expansion valve was unable to properly control the amount of superheat at the inlet of the compressor. This in turn resulted in too high of discharge temperatures, resulting in internal compressor safeties shutting down the compressor before damage occurred.

The absence of subcooling was primarily caused by having a receiver installed. A refrigerant receiver helps to manage the amount of working charge in a vapor compression cycle. Liquid refrigerant is stored inside a fixed capacity chamber that acts as an inventory management device. The receiver can at best send saturated liquid unless it is flooded. So, an attempt to remove the receiver was tested. The removal of the receiver resulted in very high condensing

pressures in two-stage operation. The pressures reached critical points in a matter of minutes. Pre-set pressure safeties in the programming logic prevented any serious damage. Therefore, a receiver or some other method of managing refrigerant inventory is mandatory.

## 8.4 SYSTEM CONTROLS

Due to the novel technology employed by the CCHP, most of the system controls had to be developed in house. The psychrometric testing described in section 5.5 ensured the controls were able to meet the basic requirements to operate the system; all autonomous operation, heating in single and two-stage, single-stage cooling, defrost, oil management, etc. Once the CCHP was installed, these controls had to be modified using several iterations to improve the system operation and response to the building conditions. Performing these changes while the system was both in the field and operational was challenging. The CCHP had to be brought off-line before a new control program could be uploaded. If a mistake with the logic existed or the new program did not perform as desired, the CCHP would be taken off-line again. Sometimes an issue with the logic would not present itself until a certain operation was encountered. For example, the compressor speed control during two-stage operation would cause a fault and shut down the CCHP. Since the system was already installed, testing a new program for two-stage operation was not possible because the outdoor temperature was too warm.

A second challenge with controlling the CCHP was determining when to increase or decrease the heat output to the building. The zone temperature would go below the set point and a call for heat would be given. The CCHP would kick on in single-stage mode with a fixed compressor speed. If the zone temperature stayed below the set point, the speed would increase to increase the heat output of the CCHP to the building. At some point the top speed is reached and switching into two-stage is required for additional heating capacity. The decision for how long to continue in single stage before switching into two-stage was done with a PID controller. The details of all the control logic can be seen in appendix B. The gains were estimated at first and modified during testing to reach reasonable operation. Robust testing of all the CCHP controls, single-stage and two-stage heating, defrost, oil management, etc. are needed to reach an optimum performance. These improvements are expected to have a strong impact on the overall efficiency of the system.

## 8.5 SUMMARY

The field demonstration encountered unexpected obstacles. However, the field testing made great strides to identifying key problems and implementing solutions. Below are several recommendations that are being considered for new development that is already underway.

- Select a suction pipe diameter that provides a high enough velocity to return oil back to the compressor or implement a similar programming strategy in which the compressor periodically speeds up to provide a sufficient velocity to return oil back the compressor.
- Always use crankcase heaters regardless of the compressor’s location.
- Utilize a single oil separator and have the oil line return to the inlet to the accumulator.
- Pre-charge accumulator with compressor oil until the oil lever is higher than the oil orifice.
- Always utilize a refrigerant inventory management device/method.

- Use a commercially available and robust electronic expansion valve that can tolerate saturated-liquid or slightly two-phase refrigerant.
- Optimize the CCHP controls by running extensive tests on modulating the heat output.

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# APPENDICES

## Appendix A: Points of Contact

| POINT OF CONTACT<br>Name | ORGANIZATION<br>Name<br>Address | Phone<br>Fax<br>E-mail | Role in Project |
|---|---|---|---|
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# Appendix B: Control Algorithms

| Building 113 Input/Output Points (Note Building 114 Differs Slightly) | | | |
|---|---|---|---|
| Name | Description | Type | Signal |
| IDHX GAS | Indoor Heat Exchagner Gas Line Refrigerant Temperature | AI | Thermistor |
| IDHX LIQ | Indoor Heat Exchanger Liquid Line Refrigerant Temperature | AI | Thermistor |
| Supply Temp | Supply Air Temperature | AI | Thermistor |
| Velocity | Supply Air Velocity | AI | 0-10V |
| Return Temp | Return Air Temperature | AI | Thermistor |
| Ambient Temperature | Outdoor Temperature | AI | Thermistor |
| Ambient RH | Outdoor Relative Humidity | AI | 0-20mA |
| ODHX GAS | Outdoor Heat Exchanger Gas Line Refrigerant Temperature | AI | Thermistor |
| Indoor Temperature | Thermostat Temperature | AI | Thermistor |
| High Pressure | Pressure discharged from Compressor 2 | AI | 0-20 mA |
| Low Pressure | Pressure at the Accumulator | AI | 0-20 mA |
| IM Pressure | Pressure discharged from Compressor 1 | AI | 0-20 mA |
| N Temp East | Northern Half East Side Temperature Inside the Barrack | AI | Thermistor |
| N RH East | Northern Half East Side Relative Humidity Inside the Barrack | AI | 0-20 mA |
| N Temp West | Northern Half West Side Temperature Inside the Barrack | AI | Thermistor |
| N RH West | Northern Half West Side Relative Humidity Inside the Barrack | AI | 0-20 mA |
| S Temp East | Southern Half East Side Temperature Inside the Barrack | AI | Thermistor |
| S RH East | Southern Half East Side Relative Humidity Inside the Barrack | AI | 0-20 mA |
| S Temp West | Southern Half West Side Temperature Inside the Barrack | AI | Thermistor |
| S RH West | Southern Half West Side Relative Humidity Inside the Barrack | AI | 0-20 mA |
| Fan Status | Current Switch on the Indoor Fan Power Cable | BI | Dry Contact |
| Outdoor Fan Status | Current Switch on the Outdoor Fan Power Cable | BI | Dry Contact |
| Outdoor Pressure | Liquid Line Pressure on the Outdoor Heat Exchanger | AI | 0-20 mA |
| ODHX LIQ | Outdoor Heat Exchanger Liquid Line Refrigerant Temperature | AI | Thermistor |
| Sump Temp | Temperature at the Bottom of the Compressor Shell on C2 | AI | Thermistor |
| Accum Temperature | Refrigerant Temperature at the Outlet of the Accumulator | AI | Thermistor |
| C2 Suc Temp | Refrigerant Temperature at the Suction Line to C2 | AI | Thermistor |
| Return Dampers | Power Damper Actuators on the Return Duct | BO | Relay |
| Supply Dampers | Power Damper Actuators on the Supply Duct | BO | Relay |
| V2 | Opens Pressure Equalization Valve (V2) | BO | Relay |
| V3 | Opens Oil Equalization Valve (V3) | BO | Relay |
| Fan Start/Stop | Starts Indoor Fan | BO | Relay |
| Outdoor Fan Start/Stop | Starts Outdoor Fan | BO | Relay |
| Comp Stage 1 S/S | Starts Compressor 1 | BO | Relay |
| C2 Driver | Starts the Variable Speed Driver for C2 | BO | Relay |
| 4-Way Valve | Powers the 4-Way Valve (Heating = OFF, Cooling = ON) | BO | Relay |
| ECO POWER | Turns on the Primary Expansion Valve (XV1) | BO | Relay |
| SCR | Adjusts the Heating Output of the Electric Resistance Heater | AO | 0-20 mA |
| Conventional | Disables the Conventional HVAC System | BO | Relay |
| Conv Unocc | Sets the Conventional System Setpoints to Unoccupied | BO | Relay |
| C2 - Sump Heat | Turns on the Crankcase Heater on C2 | BO | Relay |
| C1 - Sump Heat | Turns on the Crankcase Heater on C1 | BO | Relay |
| EXV POW | Turns on the Economizing Expansion Valve (XV2) | BO | Relay |
| EXV Close | Closes Economizing Expansion Valve (XV2) | BO | Relay |
| Reheat Relay | Closes the Relay to Allow Power to the Electric Heater | BO | Relay |

***Automated Logic Corporation Controller*** The ALC controller is a programmable logic controller, PLC, with a main control module having a fixed number of inputs and outputs. For increased capacity, the main module allows for connections with expanders having additional input and output ports. The main module is under the first expander located in the upper left side of the image and the second expander is below both units. A terminal strip connects the ALC inputs and output with the hardware or sensors of the system. Below the terminal strip are ice cube relays that control 24 VAC supply to motor contactors or hardware directly such as solenoid valves.

WebCTRL is a browser-based building automation system that is integrated with the ALC controller. This platform allows for an internet connection to be made using Internet Explorer to view the execution of the program in real time. EIKON is the programming tool used by the WebCTRL platform to develop the logic and sequence of operations of the control program. EIKON is coded using a graphical method and is similar to LabVIEW in having prebuilt function blocks connected together with the inputs and outputs via a virtual wire. All programs described in the following sections can be seen in this appendix.

***Automatic Mode*** Before any actions were performed by the controller, the system needed to be set into the automated mode. This mode ensured the back-up system was taken off-line and the heat pump was in a position to provide the main heating source for the space. If no alarms were tripped after the last shut-down, the automatic mode would allow the system to be enabled for heating.

***Enable Heating*** A reverse acting PI, proportional integrator, controller was used to determine the heating percentage required by comparing the set point to the indoor temperature sensor. The individual gains were manually adjusted until a reasonable response was obtained that engaged heating in a short amount of time but would not increase the system capacity too fast. Before the heating percentage was sent to enable the heating operation, the conditions for heating were verified first. A manual switch in the program had to be set to on and the measured outdoor temperature needed to be below 20°C (68°F). Both these conditions prevented the system operating if a manual shut-down was engaged and if the outdoor conditions were warm enough to dictate the building should not need heating.

***Engage Compressors*** Once the PID percentage was larger than 2%, heating was activated and the system was ready to engage the compressor. The high stage compressor was powered to operate in single stage mode, but the program first required that all component controls, such as fans, and the 4-way valve, were in the proper position. The requirements to activate the respective operation were different for defrost, oil return, cooling and heating modes. For the heating mode, both fans needed to be powered, the four-way valve had to be off, and the system should not be going into defrost or oil equalization. Once these conditions were met, the compressor speed was set off the PI percentage from the room conditions. A linear scale was assumed to convert a heating percent between 0% to 90%, to reach a speed command scaled to the range between 1800 RPM and 7000 RPM.
To engage the low stage compressor, the controller would need to call for two-stage heating. If the pressure ratio across the high stage compressor reaches a level higher than 4, then two-stage operation is enabled. The compressor speed is reduced to a switching speed of 3600 RPM and the tandem compressor is powered up. The speed control of the high stage compressor during

two-stage mode uses two PID controllers where one is reverse acting and the other is forward acting. The overall goal of the PID controllers is to keep the pressure ratios across each compressor equal. For example, if the ratio across the low stage is too high, the intermediate pressure needs to be reduced and the compressor speed is increased.

***Electric Reheat*** The bay of electric resistors that serves as back-up heating or as the heat source during defrost is controlled by a silicon-controlled rectifier, SCR, controller. The SCR takes a 4-20 mA signal and converts the voltage output by a percentage assuming a linear scale where 0% corresponds to a 4 mA signal and 100% corresponds to a 20 mA signal. If the heating percentage reaches 100%, then 100% of the electric reheat is engaged due to the capacity required by the building. A heating percentage of 100% is also used for defrost and oil equalization mode. Additional verifications are required before the electric reheat is energized.

***Defrost*** Traditional defrost of the outdoor coil is done by shutting the system down, and starting back up in cooling mode to melt the ice build-up. Defrost is engaged when the difference between the outdoor temperature and the evaporation temperature is above 19.4°C (35°F). The heat pump has to be in heating mode with the high stage compressor in operation and an outdoor temperature colder than 10°C (50°F) before this difference is evaluated. After defrost is engaged, the system runs in cooling mode until the outdoor coil temperature reaches a specified level that depends on the outdoor temperature. The conditions for termination were copied from the defrost logic described by Trane for the packaged heat pump used. It was assumed that the manufacturer has thoroughly tested the defrost termination conditions for the specific coil.

***Oil Management*** Two levels of oil management were designed into the control program. The first is an oil return strategy during single-stage heating and oil equalization between compressors after two-stage operation. For oil return, the compressor speed is increased to 5000 RPM for a set period of time. The increase in refrigerant velocity allows for oil to be returned to the compressor by capturing oil trapped in parts of the system from low refrigerant velocities. The second level of oil management is engaged after 5 hours of two-stage operation. The amount of time is larger than previously recommended for two-stage heat pumps because this system has two oil separators while the previous system utilized only one. It is expected the imbalance of oil level between compressors will take a longer period of time due to the extra oil separator. When oil equalization is engaged, the heat pump is shut down and a bypasses solenoid valve is opened to equalize the low and intermediate pressures. By doing this, the oil sumps can be connected right away by opening a second solenoid valve to equalize the oil levels. Without equalizing the pressures, the second valve cannot be opened immediately because the pressure imbalance between the compressor shells will result in pressure driven oil flow that cannot be controlled.

***Safeties*** Several safety checks were established in the program to protect the heat pump as well as to prevent critical temperatures in the zone. The major alarms were pressure, zone temperatures, compressor sump and superheat temperatures, discharge air temperature and any fan failures. Alarm states were also read in from the variable speed controller and the EcoFlow controller. If any of these alarms were activated, the system went into shut-down, activated the back-up system, and sent an email out notifying what fault occurred.

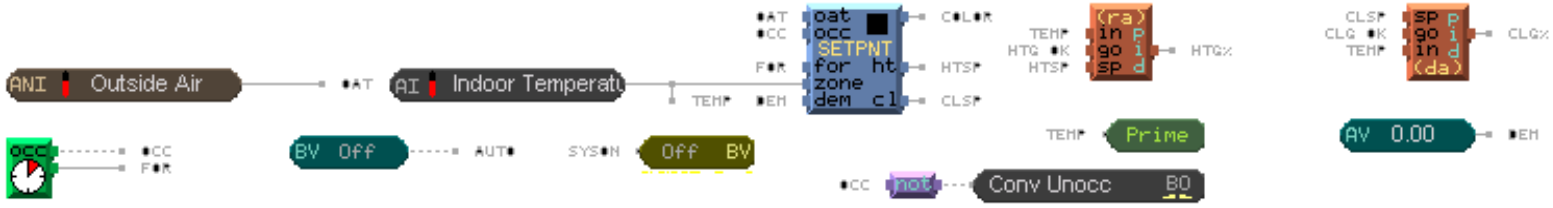


**Figure B-1: Thermostat Control.**

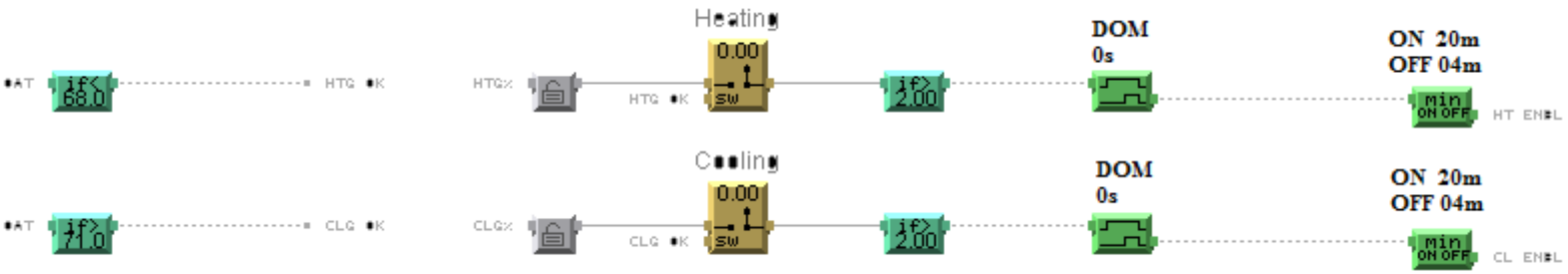


**Figure B-2: Enabling Heating or Cooling.**

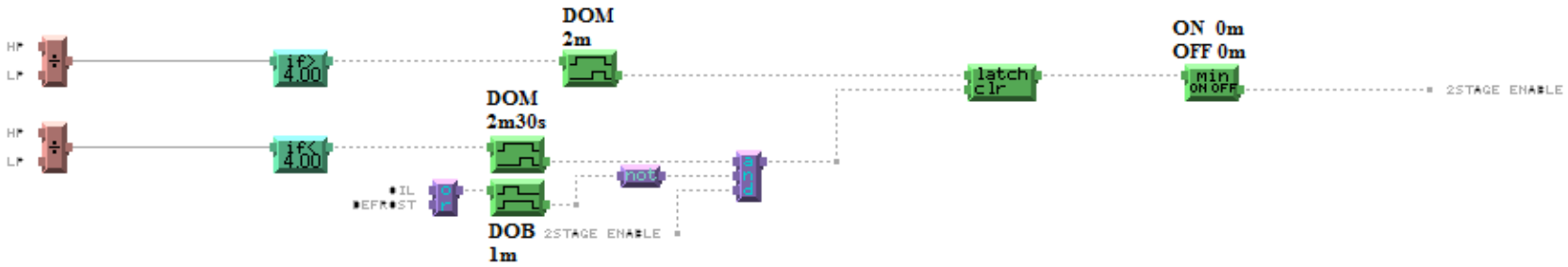


**Figure B-3: Enabling 2-Stage Heating.**

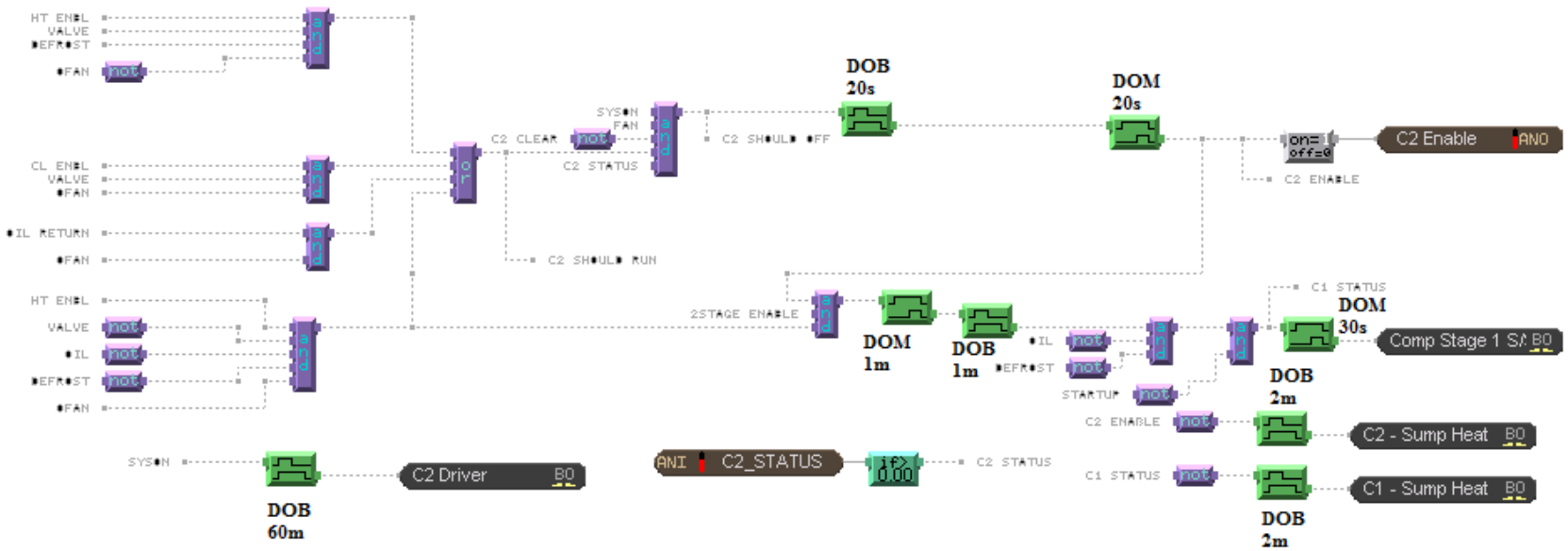


**Figure B-4: Start/Stop Compressors.**

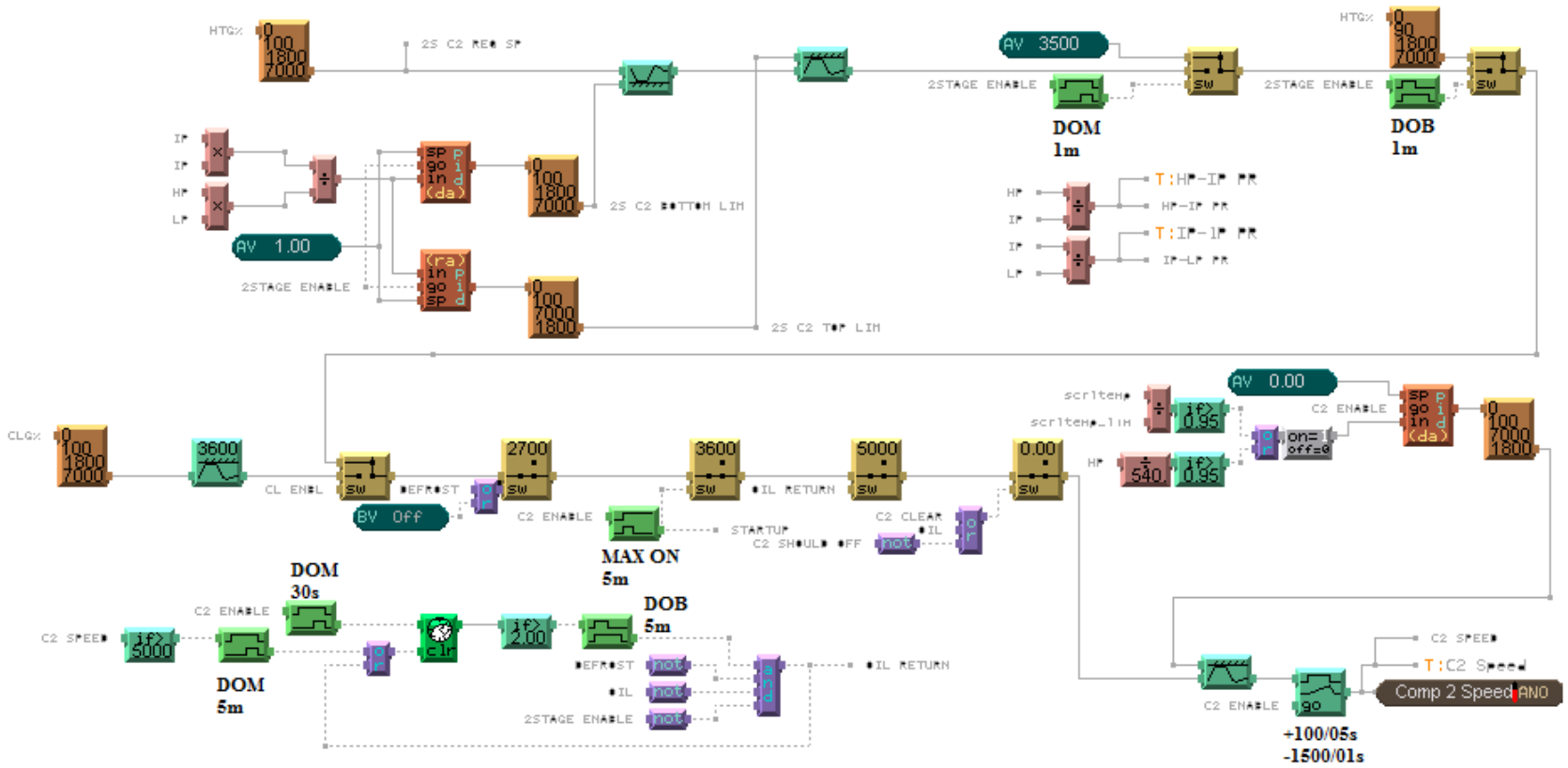


**Figure B-5: Compressor 2 Speed.**

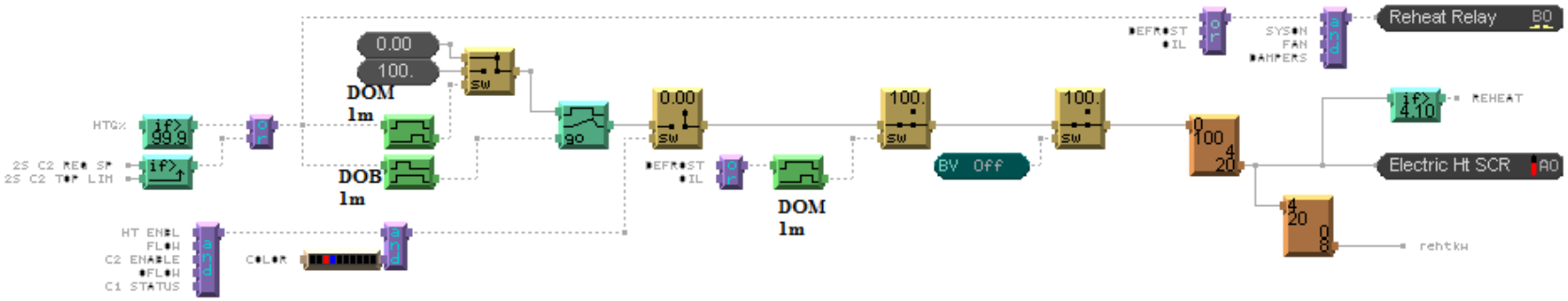


**Figure B-6: Reheat.**

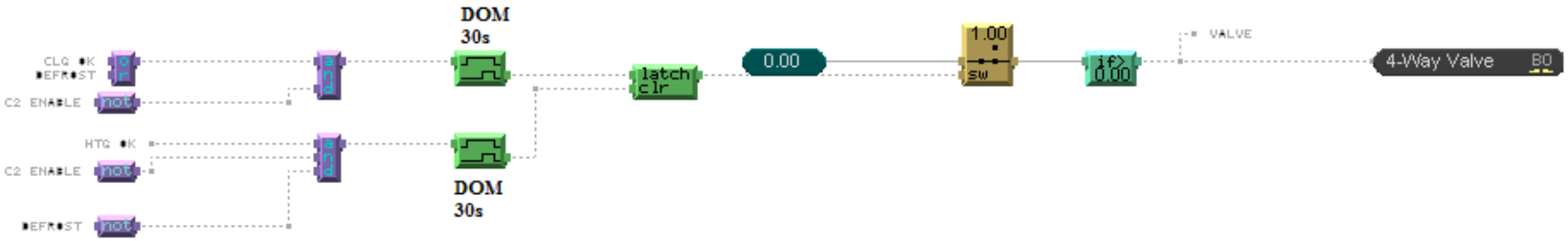


**Figure B-7: 4-Way Valve.**

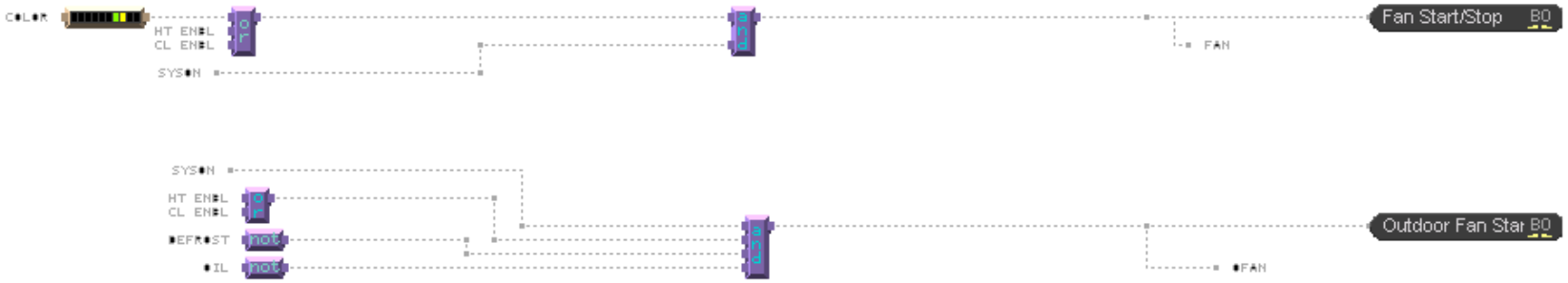


**Figure B-8: Fans.**

**Figure B-9: Economizing.**

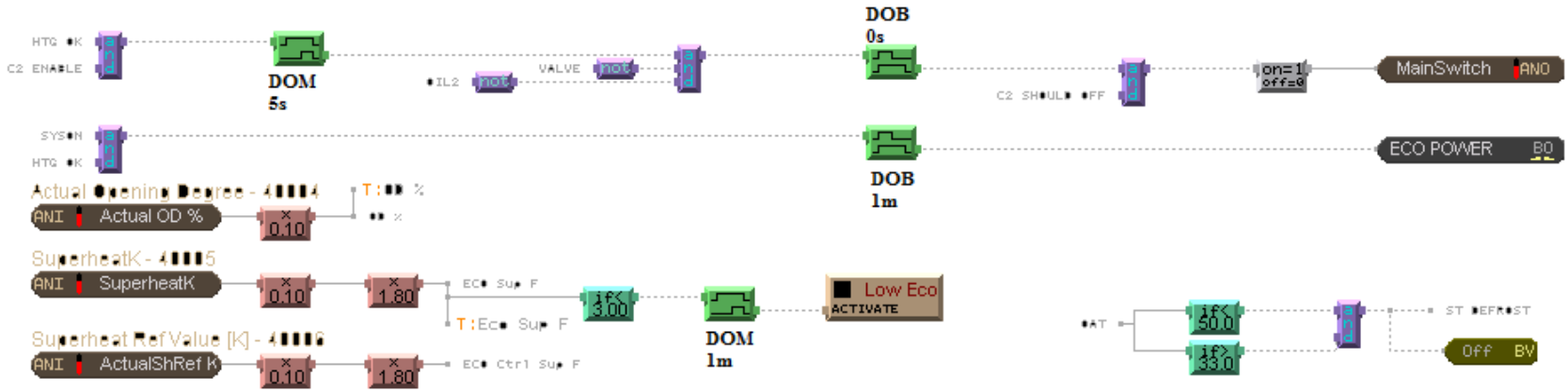


**Figure B-10: Primary Expansion Valve.**

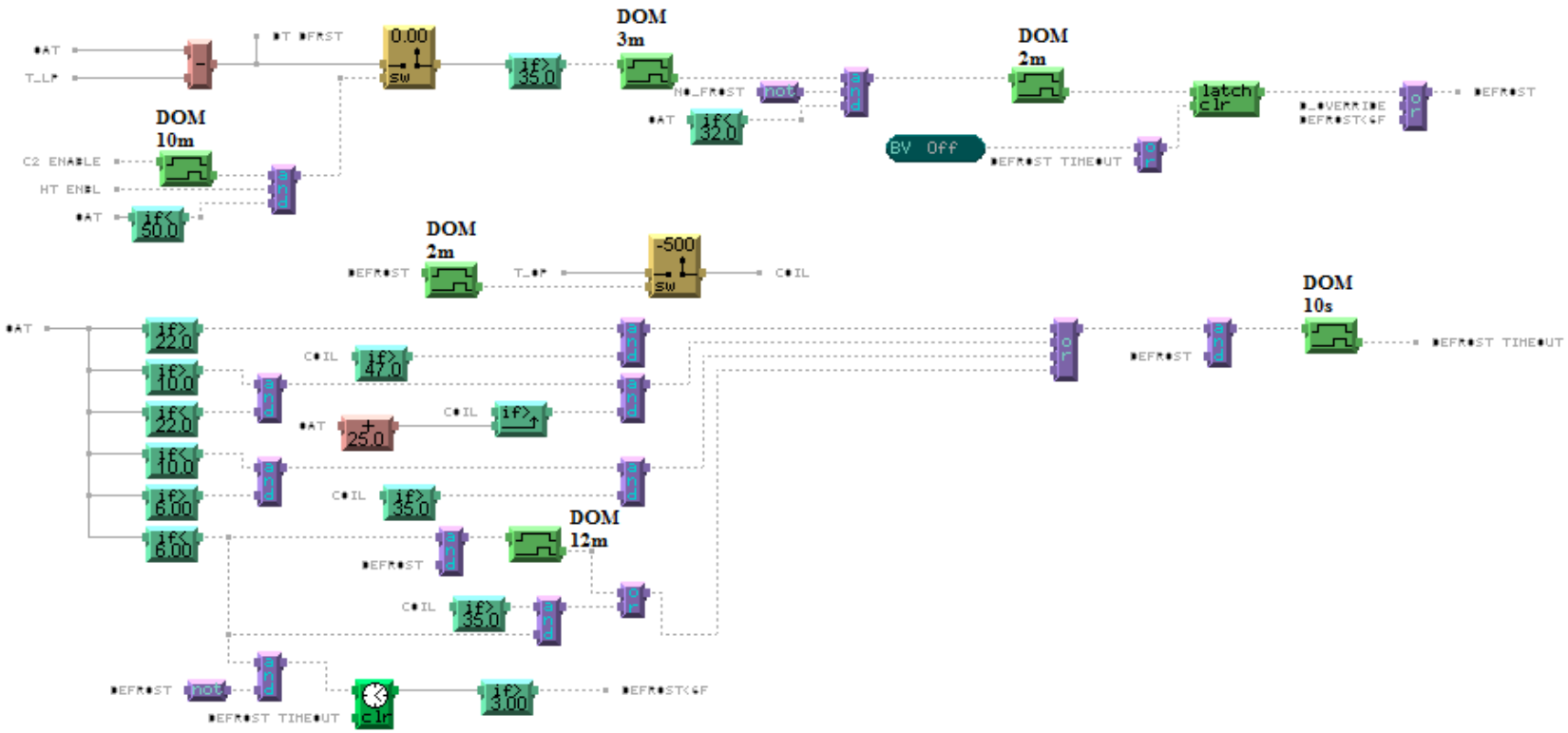


**Figure B-11: Defrost.**

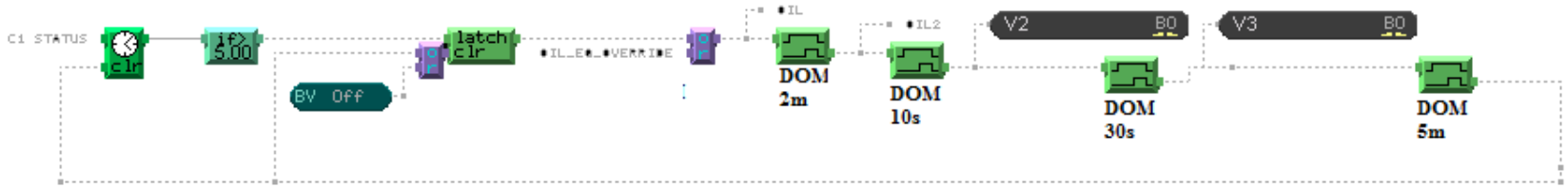


**Figure B-12: Oil Equalization.**

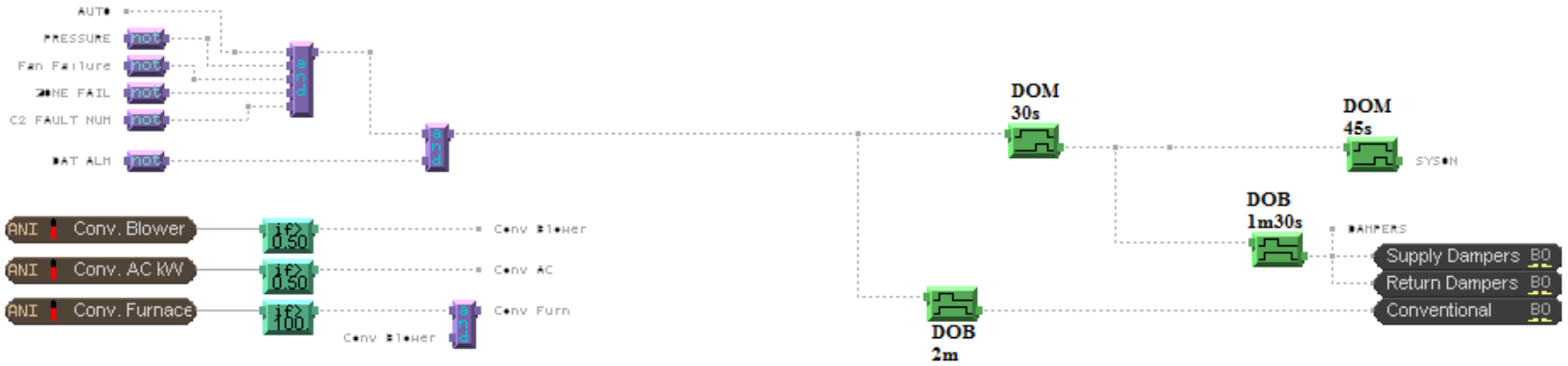


**Figure B-13: Dampers and Conventional System.**

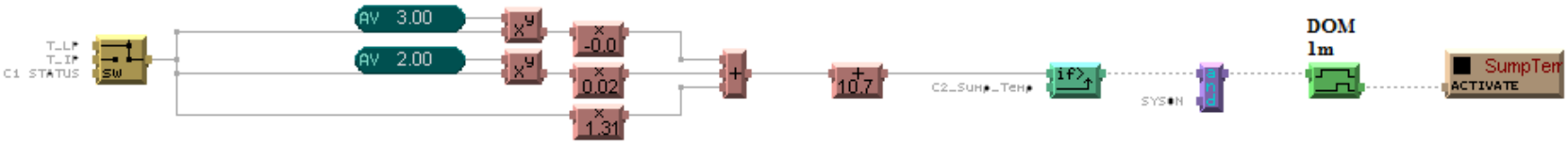


**Figure B-14: Crankcase Safety.**

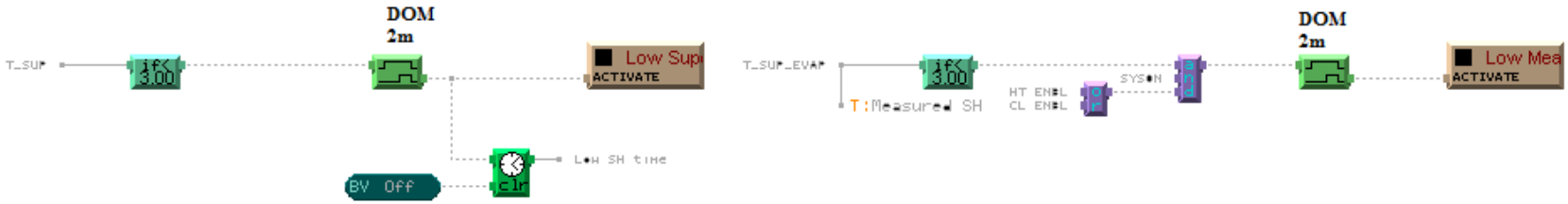


**Figure B-15: Superheat Safety.**

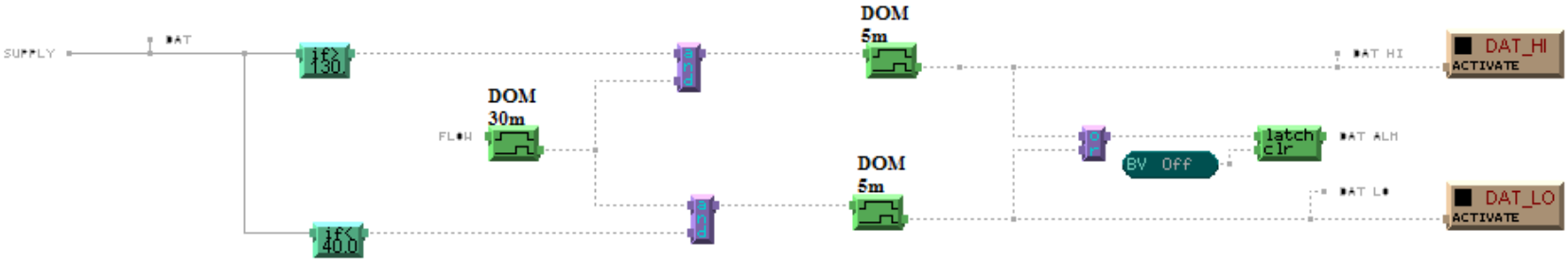


**Figure B-16: Discharge Temperature Safety.**

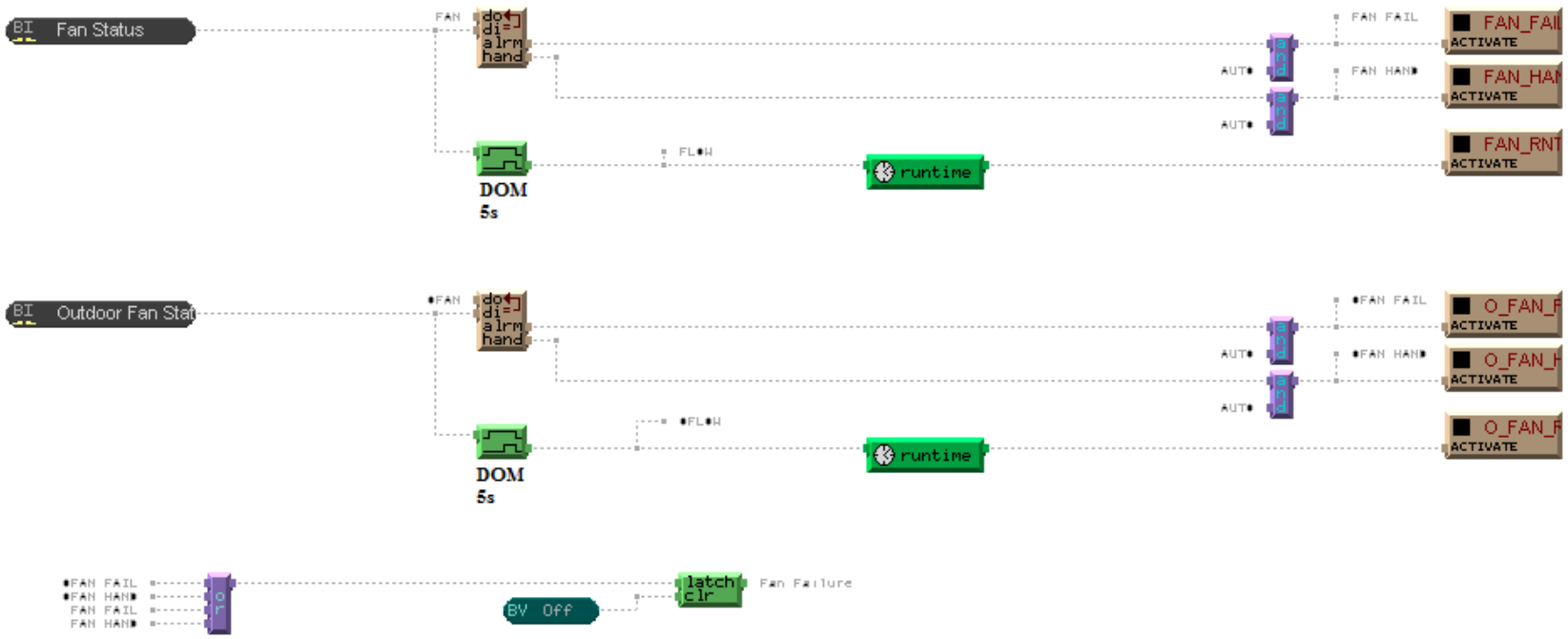


**Figure B-17: Fan Proof.**

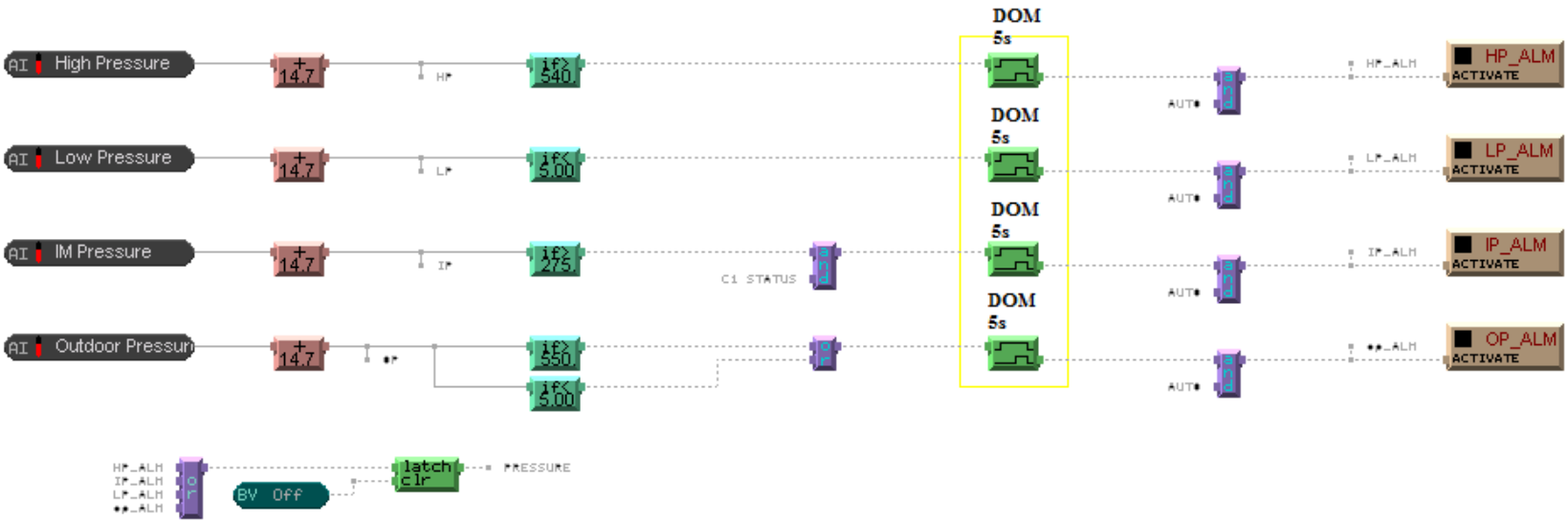


**Figure B-18: Pressure Safety.**

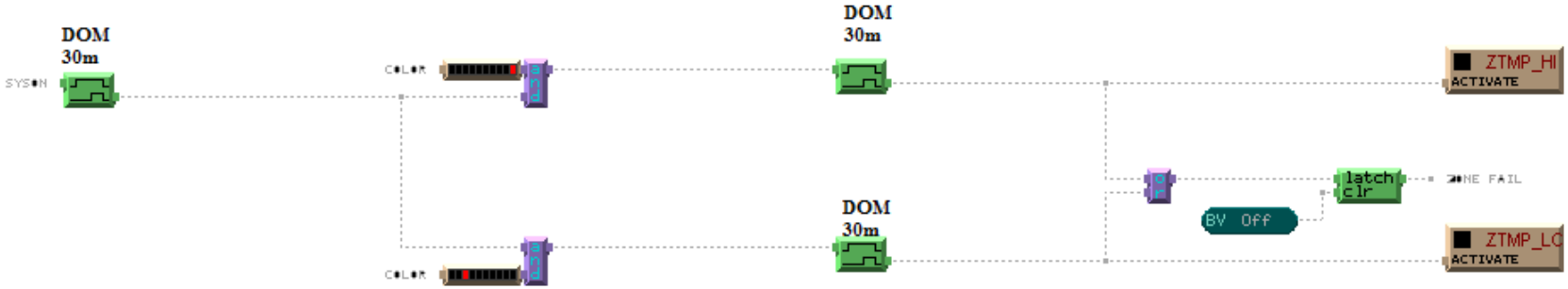


**Figure B-19: Zone Temperature Safety.**

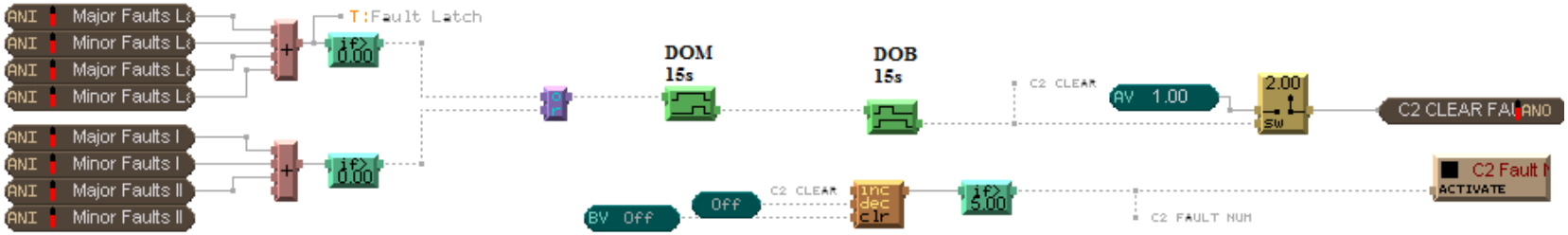


**Figure B-20: Compressor Safety.**

# Appendix C: Comfort Calculation Raw Data

## Table C 1: Maximum, Average and Minimum Temperatures and Relative Humidities to Determine Thermal Comfort.

| Building 113 | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Data Set | | Heat Pump | | | | Furnace | | | |
| | | T_SW [F] | RH_SW [%] | T_SE [F] | RH_SE [%] | T_NE [F] | RH_NE [%] | T_NW [F] | RH_NW [%] |
| 02/1-02/4 | Max. | 81 | 37 | 81 | 41 | 91 | 22 | 92 | 23 |
| | Avg. | 74 | 23 | 70 | 24 | 84 | 8 | 84 | 9 |
| | Min. | 66 | 5 | 62 | 4 | 69 | 0 | 70 | 0 |
| 02/13-02/4 | Max. | 81 | 18 | 80 | 21 | 90 | 18 | 92 | 18 |
| | Avg. | 77 | 16 | 75 | 17 | 79 | 13 | 80 | 14 |
| | Min. | 72 | 14 | 69 | 13 | 69 | 7 | 71 | 8 |
| 02/14-02/15 | Max. | 83 | 26 | 84 | 28 | 90 | 25 | 90 | 25 |
| | Avg. | 77 | 18 | 76 | 18 | 80 | 15 | 80 | 16 |
| | Min. | 71 | 13 | 69 | 11 | 70 | 7 | 71 | 8 |
| Building 114 | | | | | | | | | |
| Data Set | | Furnace | | | | Heat Pump | | | |
| | | T_SW [F] | RH_SW [%] | T_SE [F] | RH_SE [%] | T_NE [F] | RH_NE [%] | T_NW [F] | RH_NW [%] |
| 01/4 - 01/5 | Max. | 99 | 18 | 97 | 17 | 66 | 27 | 73 | 20 |
| | Avg. | 91 | 11 | 89 | 11 | 51 | 22 | 62 | 16 |
| | Min. | 72 | 8 | 72 | 9 | 45 | 10 | 57 | 9 |
| 01/10 - 01/17 | Max. | 100 | 62 | 97 | 64 | 78 | 58 | 83 | 58 |
| | Avg. | 81 | 26 | 80 | 26 | 71 | 28 | 76 | 26 |
| | Min. | 70 | 10 | 70 | 10 | 52 | 12 | 61 | 10 |
| 01/25 | Max. | 95 | 15 | 92 | 16 | 79 | 13 | 85 | 11 |
| | Avg. | 86 | 11 | 84 | 11 | 66 | 9 | 77 | 6 |
| | Min. | 75 | 8 | 74 | 8 | 59 | 3 | 68 | 1 |
| 02/14 - 02/15 | Max. | 99 | 26 | 96 | 26 | 77 | 28 | 82 | 24 |
| | Avg. | 86 | 16 | 85 | 17 | 71 | 21 | 76 | 18 |
| | Min. | 73 | 11 | 72 | 12 | 59 | 15 | 66 | 13 |

## Appendix D: EES Heat Pump Model Improved with Experimental Results

To generate each fit, the lowest order polynomial is selected that produces a coefficient of simple determination, r-squared, value of 1. The data collected from the field demonstration is used to generate experimental points for the CCHP heat output and COP that are plotted as a function of outdoor temperature. Using the EES simulation curve fits and the experimental points, a visual comparison can be made to adjust the EES curve fits to match the experimental results. Figure D-1shows a plot of this comparison. The simulation results seem to over predict the CCHP capacity as a function of outdoor temperature. Therefore, to adjust the EES curve fits, the linear term of each equation was adjusted to allow the entire curve to be shifted along the y-axis. Altering only the linear term of the curve fit equation maintains the original curvature while allowing the fit to be adjusted to the experimental data. Each curve seen in Figure D-1 is shifted until a maximum r-squared value is achieved to indicate the best match possible. Once all the new curve fits have been created for both the CCHP capacity and COP, the TRNSYS model is updated.

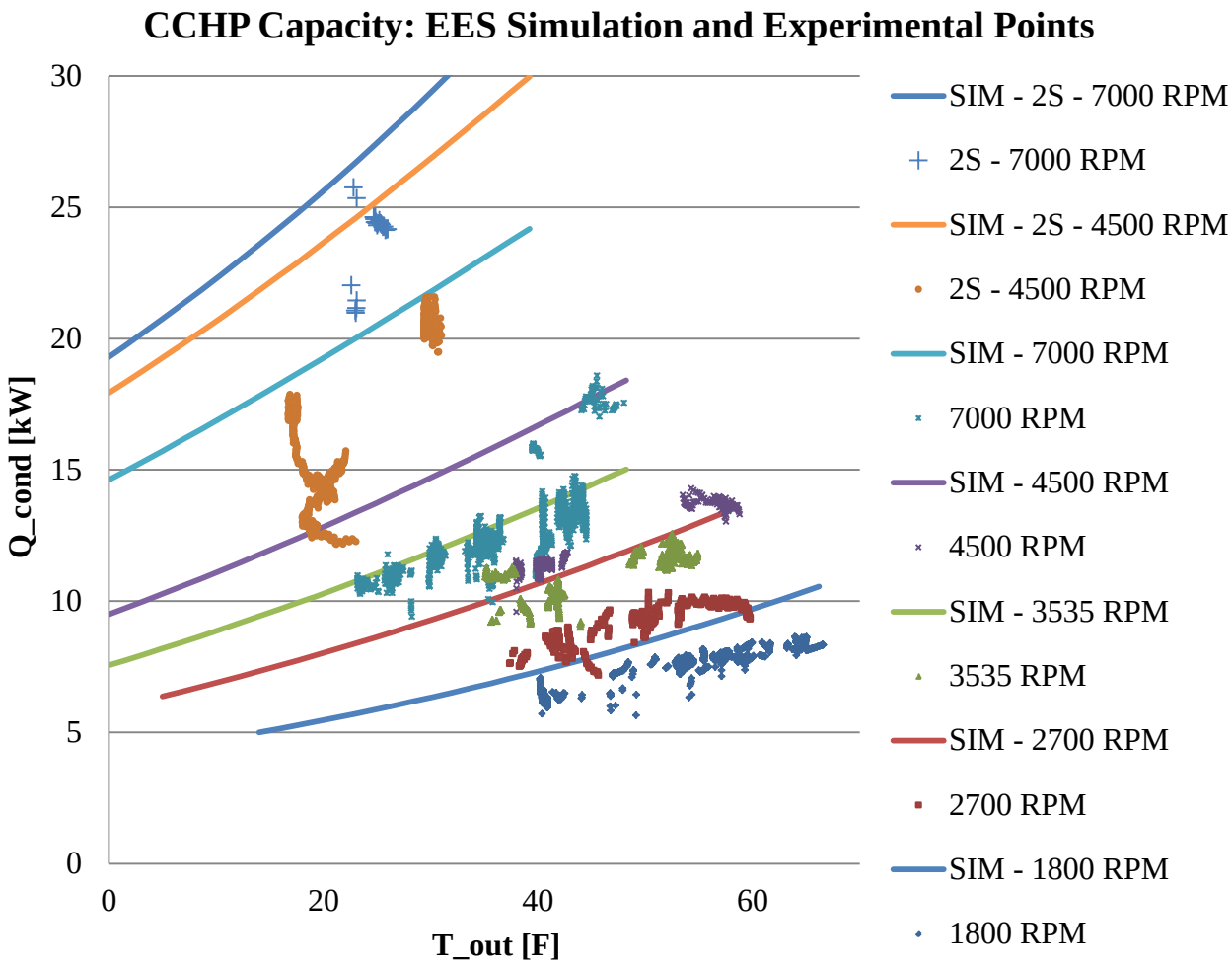


**Figure D-1: Comparison CCHP of Capacity – EES Simulation and Experimental Results.**

# Appendix E: Equipment Calibration

## Certificate of Calibration

**Model:** E50C2
**Serial Number:** 4E3515CB
**Manufacture date code:** 201214

All equipment used for product calibration is inspected, calibrated, and certified NIST-traceable on an annual basis. All power and energy metering products undergo a complete functional test prior to shipment, this certificate certifies that the product listed above meets its published specifications.

Bob Smith

**Veris Industries Quality Manager**

ZP00066-0A


## Certificate of Calibration

**Model:** E50C2
**Serial Number:** 4E3515CB
**Manufacture date code:** 201214

All equipment used for product calibration is inspected, calibrated, and certified NIST-traceable on an annual basis. All power and energy metering products undergo a complete functional test prior to shipment, this certificate certifies that the product listed above meets its published specifications.

Bob Smith

**Veris Industries Quality Manager**

ZP00066-0A

SAGE
METERING INC.

# PRODUCT QUALITY CERTIFICATE OF CONFORMANCE

## Product Inspection & Quality Statement

**All individual parts and components which make up the product being provided have been inspected and approved for manufacture. In addition, subassemblies have been inspected, tested, and accepted for final assembly. Each completed assembly has been final tested and approved for shipment.**

## Conformance Statement

**SAGE Metering Incorporated certifies this instrument was tested in compliance with ANSI/NCSL Z540 and ISO/IEC 17025 requirements. SAGE Metering, Inc. calibration services are derived from MIL-STD-45662A. The tests are performed using measuring & test equipment with certified NIST traceability. (Applicable NIST numbers are available upon request). Reproduction of the complete certificate is allowed. Parts of the certificate may only be reproduced after written permission is granted by SAGE Metering, Inc.**

| | | |
|---|---|---|
| **CUSTOMER:** | Automated Logic - Indiana | |
| **PURCHASE ORDER:** | CC1144-0007 | |
| **SAGE SALES ORDER:** | 13683 | |
| **MODEL:** | SIP-050-AC115-NG | |
| **POWER REQUIREMENT:** | AC115 | |
| **OPTIONAL OUTPUT:** | Flow, 4 - 20mA | 10 SCF/PULSE |
| **SAGE UNIT/SENSOR SERIAL NUMBERS:** | 78172-40689 | Slave ID = 32 HEX |
| **TAG:** | | 50 dec |
| **PRIME BAUD RATE / PRIME PARITY** | 19200.00 | EVEN |
| **SUGGESTED CALIB/VALIDATION INTERVAL:** | 12 months after Calibration | |
| **CALIBRATION DATE:** | 4/24/2012 | |
| **OPERATING PRESSURE RANGE:** | (14.7 PSIA + PSIG) ± 20% | |
| **MAXIMUM PRESSURE RATING:** | 500 PSIG | |
| **SENSOR TEMPERATURE RANGE:** | STD: -40 to 200 F | |
| **ELECTRONICS TEMPERATURE RANGE:** | 0° to +150°F (-18° to +65.56°C) | |
| **ACCURACY REFERENCED TO 70°F (21°C):** | +/- 1% Rdg + 0.5% FS | |
| **CALIBRATION REFERENCE CONDITIONS:** | 70°F and 29.92" Hg | |
| **PROCESS GAS:** | NG | |
| **PROCESS FLOW (FS, 4-20 mA)/LowFlowCutoff** | 0 - 88 SCFH | |
| **CALIBRATED FLOW (Incl Over Rg)** | 1.47 SCFM | |
| **PROCESS LINE SIZE** | 0.5 in sch 40 | |
| **PROCESS TEMPERATURE:** | 70 F | |
| **PROCESS PRESSURE:** | 1 PSIG | |
| **CALIBRATION TECHNICIANS:** | mv/mk | |
| **TURBINES** | 0 0 0 | |
| **DMM's** | DMM #1 & #2 | |
| **FLOW CALIBRATION PROCEDURE USED:** | 100-0164 | |
| **TEMP CALIBRATION PROCEDURE USED:** | 100-0089 | |
| **SPECIAL NOTES:** | | |
| **SOFTWARE REV#** | 2.09 | |
| **AMBIENT AIR ZERO in mW/GAS FLOW ZERO in mW** | 105 | 122 |

***Authorization:*** **Date:** **April 24, 2012**

100-0059

Sage Metering, Inc / 8 Harris Ct / Building D1 / Monterey, CA 93940 / 866-677-SAGE / 831-242-2030 / Fax 831-655-4965 / www.sagem

# Calibration Report

Date/Time: 5/3/2012 9:05:15 AM
Operator Name: Bonnie Stroud

Unit Serial Number: 460218

Sensor Series: ELF/F00

Barometric Pressure 30.00 In Hg

Calibration Flows

| Flow | AFPM |
|---|---|
| 1 | 0.0 |
| 2 | 128.1 |
| 3 | 359.0 |
| 4 | 521.9 |
| 5 | 782.0 |
| 6 | 1557.1 |
| 7 | 2855.1 |

Output Comparison

| Test | LabTemp | AFPM | UUT FPM | Difference | % Difference |
|---|---|---|---|---|---|
| 1 | 73.83 | 780.5 | 787.9 | 7.39 | 0.95 |
| 2 | 74.34 | 2855.1 | 2857.7 | 2.64 | 0.09 |

***EBTRON, Inc.*** certifies that all materials, components, and workmanship used in the manufacture of this equipment are in strict accordance with published specifications. All test and calibration standards are traceable either to the National Institute of Standards and Technology (NIST) or have been verified by instruments whose accuracy are traceable to NIST standards, or are derived from accepted values of physical constants.

Airflow calibration for the ELF model identified in this document, was found to be within EBTRON's published performance tolerance. Specific examples from the results of this calibration are included with this report.

Rick Rivers
Production Manager

Casey Lewis
Calibration Engineer

**Building Automation Products, Inc.**
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13364_Agency01.pdf

# CE Declaration of Conformity

The manufacturer hereby declares that the product:

| | |
|---|---|
| **Product Name:** | Humidity or Humidity/Temperature Combination Units |
| **Product Number:** | BA/H#-R#, BA/1.8K-H#-R#, BA/2.2K-H#-R#, BA/3K-H#-R#, BA/3.3K-H#-R#, BA/10K-2-H#-R#, BA/10K-3-H#-R# and BA/10K-3(11K)-H#-R# (# indicates any combination of numbers or letters) |
| **Product Options:** | 2% or 3% humidity range with or without display |
| **Date of Issue:** | 11/04/2003 |

Conforms to the following standards or other normative documents:

| | | |
|---|---|---|
| **Electronic Emission:** (by Council Directive 89/336/EEC) | EN50081-1:1998<br>Radiated EN55022: 1998 Class A limit<br>Conducted EN55022: 1998 Class A limit | |
| **Electromagnetic Immunity** (by Council Directive 89/336/EEC) | EN61000-6-2<br>EN61000-4-2 | ±4 kV contact discharge<br>±8 kV air discharge |
| | EN61000-4-3 | 3 V/M |
| | EN61000-4-4 | ±1 kV Power<br>±0.5kV Signal |
| | EN61000-4-5 | ±1kV |
| | EN61000-4-6 | 3 V |
| | EN61000-4-8 | 1 A/m |

This document certifies that the aforementioned products comply with the requirements of Council Directives 89/336/EEC and carry the "CE" mark accordingly.

Signature of Manufacturer

Mark Eric Nurczyk
Product Engineer
Building Automation Products, Inc
750 North Royal Ave.
Gays Mills, WI 54631
USA

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13363_Agency01.pdf

# CE Declaration of Conformity

The manufacturer hereby declares that the product:

| | |
|---|---|
| **Product Name:** | 1.8K, 2.2K, 3K, 3.3K, 10K-2,10K-3 and 10K-3(11K) Thermistor Product Family |
| **Product Number:** | BA/1.8K-#, BA/2.2K-#, BA/3K-#, BA/3.3K-# BA/10K-2-#, BA/10K-3-#, BA10K-3(11K)-# (# indicates any combination of numbers or letters) |
| **Product Options:** | All products and options using the 1.8K, 2.2K, 3K, 3.3K, 10K-2, 10K-3 and 10K-3(11K) Thermistors that do not have an LCD display or transmitter |
| **Date of Issue:** | 07/15/2002 |

Conforms to the following standards or other normative documents:

| | | |
|---|---|---|
| **Electromagnetic Immunity:** (by Council Directive 89/336/EEC) | EN61000-6-2 | |
| | EN61000-4-2 | ±4 kV contact discharge<br>±8 kV air discharge |
| | EN61000-4-3 | 10 V/m |
| | EN61000-4-4 | ±1 kV |
| | EN61000-4-6 | 10 V |
| | EN61000-4-8 | 30 A/m |

This document certifies that the aforementioned products comply with the requirements of Council Directives 89/336/EEC and carry the "CE" mark accordingly.

Signature of Manufacturer

Mark Eric Nurczyk
Product Engineer
Building Automation Products, Inc
750 North Royal Ave.
Gays Mills, WI 54631
USA

# Appendix F: Test Site Demobilization

**DEPARTMENT OF THE ARMY**
**CAMP ATTERBURY – MUSCATATUCK**
**CENTER FOR COMPLEX OPERATIONS**
**Building 4A, PO Box 5000**
**Edinburgh, Indiana 46124-5000**

JFHQ-IN-MCCO 1 August 2013

MEMORANDUM FOR: To Whom It May Concern

SUBJECT: Acknowledgement of Decommissioning – Heat Pump Test

1. Purdue University has been a partner with Camp Atterbury and the Indiana National Guard on a variety of initiatives. The most recent one, the "cold climate heat pump" demonstration project for the Environmental Security Technology Certification Program (ESTCP) of the Department of Defense, has not only served to further cement past partnerships but also to promote future joint opportunities.

2. We believe the recent demonstration was a very successful venture for the Purdue School of Mechanical Engineering Technology and for Camp Atterbury. Faculty and students contributed a significant amount of time - both here and on campus - to prepare for the demonstration, and then to actively monitor results. We met jointly on a number of occasions to share information on the current project, and to look at future partnering opportunities. In the final analysis, we found the field demonstration project well executed and of no inconvenience to our daily operations.

3. As we closeout the heat pump project, we understand Purdue would like to use our facilities as a test site for future work on a "new" cold climate heat pump that is much closer to full commercialization. In support of this initiative, we acknowledge that the demonstration units installed in buildings 113/114 will only be partially decommissioned. The decommissioning will be limited to removal of the housing surrounding the indoor compressors and the outdoor exchange unit. Major items remaining include the fan coil unit and the extra ductwork - to host future demonstrations. Removal of some items of equipment and leaving some will not impact the HVAC equipment in either building.

4. We have enjoyed our time working with Purdue faculty and students on this project over the last 2 years. We also look forward to assisting with their future ventures into other energy saving possibilities. Please contact Mr. R. Joseph Rathz, Assistant Master Planner, Cml (812) 526-1499/ext 2518, for any clarification or additional information.

JACK E FOWLER
LTC, USA (Ret)
Chief Operations Officer

# Appendix G: Excerpt of Data Set Used for 5.6 SAMPLING RESULTS

| Heat Pump | | Furnace | | Outdoor Temperature | |
|---|---|---|---|---|---|
| Time | kW | Time | kW | Time | F |
| 01/25/13 1:25 AM | 13.40 | 01/25/13 1:25 AM | 0.97 | 01/25/13 1:25:00 AM | 22.5 |
| 01/25/13 1:30 AM | 13.40 | 01/25/13 1:30 AM | 0.97 | 01/25/13 1:30:00 AM | 22.9 |
| 01/25/13 1:35 AM | 13.40 | 01/25/13 1:35 AM | 0.97 | 01/25/13 1:35:00 AM | 23 |
| 01/25/13 1:40 AM | 13.30 | 01/25/13 1:40 AM | 0.98 | 01/25/13 1:40:00 AM | 23.1 |
| 01/25/13 1:45 AM | 13.30 | 01/25/13 1:45 AM | 0.80 | 01/25/13 1:45:00 AM | 23 |
| 01/25/13 1:50 AM | 13.40 | 01/25/13 1:50 AM | 0.97 | 01/25/13 1:50:00 AM | 23.1 |
| 01/25/13 1:55 AM | 13.40 | 01/25/13 1:55 AM | 0.98 | 01/25/13 1:55:00 AM | 23 |
| 01/25/13 2:00 AM | 13.30 | 01/25/13 2:00 AM | 0.81 | 01/25/13 2:00:00 AM | 22.9 |
| 01/25/13 2:05 AM | 13.50 | 01/25/13 2:05 AM | 0.95 | 01/25/13 2:05:00 AM | 23 |
| 01/25/13 2:10 AM | 13.40 | 01/25/13 2:10 AM | 0.98 | 01/25/13 2:10:00 AM | 23.1 |
| 01/25/13 2:15 AM | 13.30 | 01/25/13 2:15 AM | 0.80 | 01/25/13 2:15:00 AM | 23.1 |
| 01/25/13 2:20 AM | 13.20 | 01/25/13 2:20 AM | 0.98 | 01/25/13 2:20:00 AM | 23 |
| 01/25/13 2:25 AM | 13.40 | 01/25/13 2:25 AM | 0.98 | 01/25/13 2:25:00 AM | 22.9 |
| 01/25/13 2:30 AM | 13.30 | 01/25/13 2:30 AM | 0.83 | 01/25/13 2:30:00 AM | 22.7 |
| 01/25/13 2:35 AM | 13.20 | 01/25/13 2:35 AM | 0.95 | 01/25/13 2:35:00 AM | 22.6 |
| 01/25/13 2:40 AM | 13.40 | 01/25/13 2:40 AM | 0.98 | 01/25/13 2:40:00 AM | 23 |
| 01/25/13 2:45 AM | 13.20 | 01/25/13 2:45 AM | 0.81 | 01/25/13 2:45:00 AM | 22.9 |
| 01/25/13 2:50 AM | 13.30 | 01/25/13 2:50 AM | 0.97 | 01/25/13 2:50:00 AM | 23 |
| 01/25/13 2:55 AM | 13.40 | 01/25/13 2:55 AM | 0.97 | 01/25/13 2:55:00 AM | 22.9 |
| 01/25/13 3:00 AM | 13.30 | 01/25/13 3:00 AM | 0.82 | 01/25/13 3:00:00 AM | 23.1 |
| 01/25/13 3:05 AM | 12.50 | 01/25/13 3:05 AM | 0.98 | 01/25/13 3:05:00 AM | 23.1 |
| 01/25/13 3:10 AM | 13.40 | 01/25/13 3:10 AM | 0.97 | 01/25/13 3:10:00 AM | 23.1 |
| 01/25/13 3:15 AM | 13.30 | 01/25/13 3:15 AM | 0.83 | 01/25/13 3:15:00 AM | 23.1 |
| 01/25/13 3:20 AM | 13.40 | 01/25/13 3:20 AM | 0.98 | 01/25/13 3:20:00 AM | 23.1 |
| 01/25/13 3:25 AM | 13.30 | 01/25/13 3:25 AM | 0.97 | 01/25/13 3:25:00 AM | 23.2 |
| 01/25/13 3:30 AM | 13.20 | 01/25/13 3:30 AM | 0.84 | 01/25/13 3:30:00 AM | 23.2 |
| 01/25/13 3:35 AM | 13.40 | 01/25/13 3:35 AM | 0.98 | 01/25/13 3:35:00 AM | 23.2 |
| 01/25/13 3:40 AM | 13.20 | 01/25/13 3:40 AM | 0.97 | 01/25/13 3:40:00 AM | 23 |
| 01/25/13 3:45 AM | 13.30 | 01/25/13 3:45 AM | 0.84 | 01/25/13 3:45:00 AM | 23.1 |
| 01/25/13 3:50 AM | 13.40 | 01/25/13 3:50 AM | 0.98 | 01/25/13 3:50:00 AM | 23.1 |
| 01/25/13 3:55 AM | 13.20 | 01/25/13 3:55 AM | 0.97 | 01/25/13 3:55:00 AM | 23.2 |
| 01/25/13 4:00 AM | 13.30 | 01/25/13 4:00 AM | 0.97 | 01/25/13 4:00:00 AM | 23.2 |
| 01/25/13 4:05 AM | 13.20 | 01/25/13 4:05 AM | 0.96 | 01/25/13 4:05:00 AM | 23.2 |
| 01/25/13 4:10 AM | 13.30 | 01/25/13 4:10 AM | 0.96 | 01/25/13 4:10:00 AM | 23.2 |
| 01/25/13 4:15 AM | 13.40 | 01/25/13 4:15 AM | 0.82 | 01/25/13 4:15:00 AM | 23.1 |
| 01/25/13 4:20 AM | 13.40 | 01/25/13 4:20 AM | 0.95 | 01/25/13 4:20:00 AM | 23.2 |
| 01/25/13 4:25 AM | 13.30 | 01/25/13 4:25 AM | 0.97 | 01/25/13 4:25:00 AM | 23 |

| | | | | | |
|---|---|---|---|---|---|
| 01/25/13 4:30 AM | 13.20 | 01/25/13 4:30 AM | 0.83 | 01/25/13 4:30:00 AM | 22.7 |
| 01/25/13 4:35 AM | 13.20 | 01/25/13 4:35 AM | 1.03 | 01/25/13 4:35:00 AM | 22.7 |
| 01/25/13 4:40 AM | 13.20 | 01/25/13 4:40 AM | 0.97 | 01/25/13 4:40:00 AM | 22.7 |
| 01/25/13 4:45 AM | 13.30 | 01/25/13 4:45 AM | 0.96 | 01/25/13 4:45:00 AM | 22.8 |
| 01/25/13 4:50 AM | 2.40 | 01/25/13 4:50 AM | 0.97 | 01/25/13 4:50:00 AM | 23.1 |
| 01/25/13 4:55 AM | 2.70 | 01/25/13 4:55 AM | 0.96 | 01/25/13 4:55:00 AM | 23 |
| 01/25/13 5:00 AM | 10.60 | 01/25/13 5:00 AM | 0.95 | 01/25/13 5:00:00 AM | 23.1 |
| 01/25/13 5:05 AM | 12.70 | 01/25/13 5:05 AM | 0.97 | 01/25/13 5:05:00 AM | 23.1 |
| 01/25/13 5:10 AM | 13.30 | 01/25/13 5:10 AM | 0.95 | 01/25/13 5:10:00 AM | 22.9 |
| 01/25/13 5:15 AM | 13.30 | 01/25/13 5:15 AM | 0.95 | 01/25/13 5:15:00 AM | 22.7 |
| 01/25/13 5:20 AM | 13.30 | 01/25/13 5:20 AM | 0.96 | 01/25/13 5:20:00 AM | 23 |
| 01/25/13 5:25 AM | 13.30 | 01/25/13 5:25 AM | 0.96 | 01/25/13 5:25:00 AM | 23.1 |
| 01/25/13 5:30 AM | 13.30 | 01/25/13 5:30 AM | 0.96 | 01/25/13 5:30:00 AM | 23.1 |
| 01/25/13 5:35 AM | 13.40 | 01/25/13 5:35 AM | 0.94 | 01/25/13 5:35:00 AM | 23 |
| 01/25/13 5:40 AM | 13.40 | 01/25/13 5:40 AM | 0.96 | 01/25/13 5:40:00 AM | 23 |
| 01/25/13 5:45 AM | 13.40 | 01/25/13 5:45 AM | 0.96 | 01/25/13 5:45:00 AM | 22.9 |
| 01/25/13 5:50 AM | 13.40 | 01/25/13 5:50 AM | 0.95 | 01/25/13 5:50:00 AM | 22.9 |
| 01/25/13 5:55 AM | 13.30 | 01/25/13 5:55 AM | 0.96 | 01/25/13 5:55:00 AM | 22.9 |
| 01/25/13 6:00 AM | 13.20 | 01/25/13 6:00 AM | 0.96 | 01/25/13 6:00:00 AM | 23.1 |
| 01/25/13 6:05 AM | 13.30 | 01/25/13 6:05 AM | 1.04 | 01/25/13 6:05:00 AM | 23 |
| 01/25/13 6:10 AM | 13.30 | 01/25/13 6:10 AM | 0.96 | 01/25/13 6:10:00 AM | 22.7 |
| 01/25/13 6:15 AM | 13.20 | 01/25/13 6:15 AM | 0.82 | 01/25/13 6:15:00 AM | 22.6 |
| 01/25/13 6:20 AM | 13.30 | 01/25/13 6:20 AM | 0.93 | 01/25/13 6:20:00 AM | 22.7 |
| 01/25/13 6:25 AM | 13.30 | 01/25/13 6:25 AM | 0.97 | 01/25/13 6:25:00 AM | 22.9 |
| 01/25/13 6:30 AM | 13.20 | 01/25/13 6:30 AM | 0.96 | 01/25/13 6:30:00 AM | 22.9 |
| 01/25/13 6:35 AM | 13.30 | 01/25/13 6:35 AM | 0.80 | 01/25/13 6:35:00 AM | 22.9 |
| 01/25/13 6:40 AM | 13.30 | 01/25/13 6:40 AM | 0.96 | 01/25/13 6:40:00 AM | 22.7 |
| 01/25/13 6:45 AM | 13.30 | 01/25/13 6:45 AM | 0.96 | 01/25/13 6:45:00 AM | 22.5 |
| 01/25/13 6:50 AM | 13.40 | 01/25/13 6:50 AM | 0.97 | 01/25/13 6:50:00 AM | 22.4 |
| 01/25/13 6:55 AM | 13.40 | 01/25/13 6:55 AM | 0.96 | 01/25/13 6:55:00 AM | 22.6 |
| 01/25/13 7:00 AM | 13.30 | 01/25/13 7:00 AM | 0.96 | 01/25/13 7:00:00 AM | 22.4 |
| 01/25/13 7:05 AM | 13.30 | 01/25/13 7:05 AM | 0.79 | 01/25/13 7:05:00 AM | 22.7 |
| 01/25/13 7:10 AM | 13.30 | 01/25/13 7:10 AM | 0.97 | 01/25/13 7:10:00 AM | 22.5 |
| 01/25/13 7:15 AM | 13.20 | 01/25/13 7:15 AM | 0.82 | 01/25/13 7:15:00 AM | 22.7 |
| 01/25/13 7:20 AM | 13.20 | 01/25/13 7:20 AM | 0.79 | 01/25/13 7:20:00 AM | 22.8 |
| 01/25/13 7:25 AM | 13.20 | 01/25/13 7:25 AM | 0.97 | 01/25/13 7:25:00 AM | 22.7 |
| 01/25/13 7:30 AM | 13.20 | 01/25/13 7:30 AM | 0.95 | 01/25/13 7:30:00 AM | 22.6 |
| 01/25/13 7:35 AM | 13.20 | 01/25/13 7:35 AM | 0.81 | 01/25/13 7:35:00 AM | 22.4 |
| 01/25/13 7:40 AM | 13.30 | 01/25/13 7:40 AM | 0.96 | 01/25/13 7:40:00 AM | 22.2 |
| 01/25/13 7:45 AM | 13.20 | 01/25/13 7:45 AM | 0.83 | 01/25/13 7:45:00 AM | 22.2 |
| 01/25/13 7:50 AM | 13.30 | 01/25/13 7:50 AM | 0.80 | 01/25/13 7:50:00 AM | 22.1 |

| | | | | | |
|---|---|---|---|---|---|
| 01/25/13 7:55 AM | 13.20 | 01/25/13 7:55 AM | 0.97 | 01/25/13 7:55:00 AM | 22.1 |
| 01/25/13 8:00 AM | 13.10 | 01/25/13 8:00 AM | 0.84 | 01/25/13 8:00:00 AM | 22 |
| 01/25/13 8:05 AM | 13.40 | 01/25/13 8:05 AM | 0.81 | 01/25/13 8:05:00 AM | 21.7 |
| 01/25/13 8:10 AM | 13.40 | 01/25/13 8:10 AM | 0.98 | 01/25/13 8:10:00 AM | 21.7 |
| 01/25/13 8:15 AM | 13.40 | 01/25/13 8:15 AM | 0.96 | 01/25/13 8:15:00 AM | 21.7 |
| 01/25/13 8:20 AM | 13.30 | 01/25/13 8:20 AM | 0.94 | 01/25/13 8:20:00 AM | 21.6 |
| 01/25/13 8:25 AM | 13.00 | 01/25/13 8:25 AM | 0.97 | 01/25/13 8:25:00 AM | 21.7 |
| 01/25/13 8:30 AM | 13.30 | 01/25/13 8:30 AM | 0.96 | 01/25/13 8:30:00 AM | 21.6 |
| 01/25/13 8:35 AM | 13.20 | 01/25/13 8:35 AM | 0.80 | 01/25/13 8:35:00 AM | 21.5 |
| 01/25/13 8:40 AM | 13.10 | 01/25/13 8:40 AM | 0.96 | 01/25/13 8:40:00 AM | 21.7 |
| 01/25/13 8:45 AM | 13.30 | 01/25/13 8:45 AM | 0.96 | 01/25/13 8:45:00 AM | 21.4 |
| 01/25/13 8:50 AM | 13.10 | 01/25/13 8:50 AM | 0.81 | 01/25/13 8:50:00 AM | 21.2 |
| 01/25/13 8:55 AM | 13.10 | 01/25/13 8:55 AM | 0.97 | 01/25/13 8:55:00 AM | 21.6 |
| 01/25/13 9:00 AM | 13.00 | 01/25/13 9:00 AM | 0.97 | 01/25/13 9:00:00 AM | 21.9 |
| 01/25/13 9:05 AM | 13.00 | 01/25/13 9:05 AM | 0.79 | 01/25/13 9:05:00 AM | 21.9 |
| 01/25/13 9:10 AM | 13.00 | 01/25/13 9:10 AM | 0.96 | 01/25/13 9:10:00 AM | 21.7 |
| 01/25/13 9:15 AM | 13.10 | 01/25/13 9:15 AM | 0.96 | 01/25/13 9:15:00 AM | 21.7 |
| 01/25/13 9:20 AM | 12.80 | 01/25/13 9:20 AM | 0.94 | 01/25/13 9:20:00 AM | 22 |
| 01/25/13 9:25 AM | 12.90 | 01/25/13 9:25 AM | 0.95 | 01/25/13 9:25:00 AM | 22.4 |
| 01/25/13 9:30 AM | 12.80 | 01/25/13 9:30 AM | 0.95 | 01/25/13 9:30:00 AM | 22.6 |
| 01/25/13 9:35 AM | 12.00 | 01/25/13 9:35 AM | 0.80 | 01/25/13 9:35:00 AM | 22.6 |
| 01/25/13 9:40 AM | 12.70 | 01/25/13 9:40 AM | 0.96 | 01/25/13 9:40:00 AM | 22.6 |
| 01/25/13 9:45 AM | 12.60 | 01/25/13 9:45 AM | 0.95 | 01/25/13 9:45:00 AM | 22.6 |
| 01/25/13 9:50 AM | 12.50 | 01/25/13 9:50 AM | 0.78 | 01/25/13 9:50:00 AM | 22.7 |
| 01/25/13 9:55 AM | 12.40 | 01/25/13 9:55 AM | 0.95 | 01/25/13 9:55:00 AM | 23.1 |
| 01/25/13 10:00 AM | 2.40 | 01/25/13 10:00 AM | 0.96 | 01/25/13 10:00:00 AM | 23.9 |
| 01/25/13 10:05 AM | 8.40 | 01/25/13 10:05 AM | 0.79 | 01/25/13 10:05:00 AM | 24.6 |
| 01/25/13 10:10 AM | 10.10 | 01/25/13 10:10 AM | 0.96 | 01/25/13 10:10:00 AM | 24.7 |
| 01/25/13 10:15 AM | 12.10 | 01/25/13 10:15 AM | 0.96 | 01/25/13 10:15:00 AM | 24.5 |
| 01/25/13 10:20 AM | 12.30 | 01/25/13 10:20 AM | 0.95 | 01/25/13 10:20:00 AM | 24.7 |
| 01/25/13 10:25 AM | 3.30 | 01/25/13 10:25 AM | 0.96 | 01/25/13 10:25:00 AM | 25.2 |
| 01/25/13 10:30 AM | 3.00 | 01/25/13 10:30 AM | 0.95 | 01/25/13 10:30:00 AM | 25.5 |
| 01/25/13 10:35 AM | 3.30 | 01/25/13 10:35 AM | 0.97 | 01/25/13 10:35:00 AM | 25.2 |
| 01/25/13 10:40 AM | 9.80 | 01/25/13 10:40 AM | 0.96 | 01/25/13 10:40:00 AM | 25 |
| 01/25/13 10:45 AM | 11.00 | 01/25/13 10:45 AM | 0.95 | 01/25/13 10:45:00 AM | 25.2 |
| 01/25/13 10:50 AM | 13.00 | 01/25/13 10:50 AM | 0.96 | 01/25/13 10:50:00 AM | 24.5 |
| 01/25/13 10:55 AM | 13.10 | 01/25/13 10:55 AM | 0.95 | 01/25/13 10:55:00 AM | 25 |
| 01/25/13 11:00 AM | 13.10 | 01/25/13 11:00 AM | 0.94 | 01/25/13 11:00:00 AM | 26.1 |
| 01/25/13 11:05 AM | 13.30 | 01/25/13 11:05 AM | 0.78 | 01/25/13 11:05:00 AM | 27 |
| 01/25/13 11:10 AM | 13.20 | 01/25/13 11:10 AM | 0.96 | 01/25/13 11:10:00 AM | 27.3 |
| 01/25/13 11:15 AM | 13.30 | 01/25/13 11:15 AM | 0.96 | 01/25/13 11:15:00 AM | 26.6 |

| | | | | | |
|---|---|---|---|---|---|
| 01/25/13 11:20 AM | 13.40 | 01/25/13 11:20 AM | 0.79 | 01/25/13 11:20:00 AM | 25.6 |
| 01/25/13 11:25 AM | 13.40 | 01/25/13 11:25 AM | 0.95 | 01/25/13 11:25:00 AM | 26.1 |
| 01/25/13 11:30 AM | 13.30 | 01/25/13 11:30 AM | 0.95 | 01/25/13 11:30:00 AM | 27.1 |
| 01/25/13 11:35 AM | 13.20 | 01/25/13 11:35 AM | 0.79 | 01/25/13 11:35:00 AM | 28.6 |
| 01/25/13 11:40 AM | 13.20 | 01/25/13 11:40 AM | 0.97 | 01/25/13 11:40:00 AM | 28.9 |
| 01/25/13 11:45 AM | 13.10 | 01/25/13 11:45 AM | 0.95 | 01/25/13 11:45:00 AM | 28 |
| 01/25/13 11:50 AM | 13.00 | 01/25/13 11:50 AM | 0.79 | 01/25/13 11:50:00 AM | 27.9 |
| 01/25/13 11:55 AM | 13.00 | 01/25/13 11:55 AM | 0.79 | 01/25/13 11:55:00 AM | 28.2 |
| 01/25/13 12:00 PM | 13.00 | 01/25/13 12:00 PM | 0.83 | 01/25/13 12:00:00 PM | 27.5 |
| 01/25/13 12:05 PM | 13.00 | 01/25/13 12:05 PM | 0.79 | 01/25/13 12:05:00 PM | 27.5 |
| 01/25/13 12:10 PM | 13.10 | 01/25/13 12:10 PM | 0.80 | 01/25/13 12:10:00 PM | 29.1 |
| 01/25/13 12:15 PM | 19.20 | 01/25/13 12:15 PM | 0.82 | 01/25/13 12:15:00 PM | 29.4 |
| 01/25/13 12:20 PM | 9.60 | 01/25/13 12:20 PM | 0.82 | 01/25/13 12:20:00 PM | 28.5 |
| 01/25/13 12:25 PM | 9.30 | 01/25/13 12:25 PM | 0.81 | 01/25/13 12:25:00 PM | 29.6 |
| 01/25/13 12:30 PM | 9.80 | 01/25/13 12:30 PM | 0.97 | 01/25/13 12:30:00 PM | 29.5 |
| 01/25/13 12:35 PM | 16.80 | 01/25/13 12:35 PM | 0.81 | 01/25/13 12:35:00 PM | 28.7 |
| 01/25/13 12:40 PM | 18.00 | 01/25/13 12:40 PM | 0.94 | 01/25/13 12:40:00 PM | 28.8 |
| 01/25/13 12:45 PM | 19.30 | 01/25/13 12:45 PM | 0.95 | 01/25/13 12:45:00 PM | 27.6 |
| 01/25/13 12:50 PM | 19.40 | 01/25/13 12:50 PM | 0.80 | 01/25/13 12:50:00 PM | 27.4 |
| 01/25/13 12:55 PM | 0.20 | 01/25/13 12:55 PM | 0.94 | 01/25/13 12:55:00 PM | 27.8 |
| 01/25/13 1:00 PM | 0.20 | 01/25/13 1:00 PM | 0.96 | 01/25/13 1:00:00 PM | 28.6 |
| 01/25/13 1:05 PM | 0.20 | 01/25/13 1:05 PM | 0.82 | 01/25/13 1:05:00 PM | 27.9 |
| 01/25/13 1:10 PM | 0.20 | 01/25/13 1:10 PM | 0.95 | 01/25/13 1:10:00 PM | 27.2 |
| 01/25/13 1:15 PM | 0.20 | 01/25/13 1:15 PM | 0.96 | 01/25/13 1:15:00 PM | 27.7 |
| 01/25/13 1:20 PM | 0.20 | 01/25/13 1:20 PM | 0.95 | 01/25/13 1:20:00 PM | 27.9 |
| 01/25/13 1:25 PM | 0.20 | 01/25/13 1:25 PM | 0.95 | 01/25/13 1:25:00 PM | 27.3 |
| 01/25/13 1:30 PM | 0.20 | 01/25/13 1:30 PM | 0.95 | 01/25/13 1:30:00 PM | 27.2 |
| 01/25/13 1:35 PM | 0.20 | 01/25/13 1:35 PM | 0.78 | 01/25/13 1:35:00 PM | 26.9 |
| 01/25/13 1:40 PM | 0.20 | 01/25/13 1:40 PM | 0.93 | 01/25/13 1:40:00 PM | 27.1 |
| 01/25/13 1:45 PM | 0.20 | 01/25/13 1:45 PM | 0.95 | 01/25/13 1:45:00 PM | 27.1 |
| 01/25/13 1:50 PM | 0.20 | 01/25/13 1:50 PM | 0.78 | 01/25/13 1:50:00 PM | 27.1 |
| 01/25/13 1:55 PM | 0.20 | 01/25/13 1:55 PM | 1.04 | 01/25/13 1:55:00 PM | 27.1 |
| 01/25/13 2:00 PM | 0.20 | 01/25/13 2:00 PM | 0.96 | 01/25/13 2:00:00 PM | 27 |
| 01/25/13 2:05 PM | 0.20 | 01/25/13 2:05 PM | 0.81 | 01/25/13 2:05:00 PM | 27.4 |
| 01/25/13 2:10 PM | 0.20 | 01/25/13 2:10 PM | 0.96 | 01/25/13 2:10:00 PM | 27.4 |
| 01/25/13 2:15 PM | 0.20 | 01/25/13 2:15 PM | 0.95 | 01/25/13 2:15:00 PM | 27.4 |
| 01/25/13 2:20 PM | 0.20 | 01/25/13 2:20 PM | 0.81 | 01/25/13 2:20:00 PM | 27.5 |
| 01/25/13 2:25 PM | 0.20 | 01/25/13 2:25 PM | 1.03 | 01/25/13 2:25:00 PM | 27.4 |
| 01/25/13 2:30 PM | 0.20 | 01/25/13 2:30 PM | 0.95 | 01/25/13 2:30:00 PM | 27.4 |
| 01/25/13 2:35 PM | 0.20 | 01/25/13 2:35 PM | 0.78 | 01/25/13 2:35:00 PM | 27.6 |
| 01/25/13 2:40 PM | 0.20 | 01/25/13 2:40 PM | 0.79 | 01/25/13 2:40:00 PM | 28.1 |

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| 01/25/13 2:45 PM | 0.20 | 01/25/13 2:45 PM | 0.95 | 01/25/13 2:45:00 PM | 28 |
| 01/25/13 2:50 PM | 0.20 | 01/25/13 2:50 PM | 0.79 | 01/25/13 2:50:00 PM | 28.5 |
| 01/25/13 2:55 PM | 4.90 | 01/25/13 2:55 PM | 0.82 | 01/25/13 2:55:00 PM | 28.2 |
| 01/25/13 3:00 PM | 9.30 | 01/25/13 3:00 PM | 0.96 | 01/25/13 3:00:00 PM | 27.8 |
| 01/25/13 3:05 PM | 9.40 | 01/25/13 3:05 PM | 0.81 | 01/25/13 3:05:00 PM | 27.8 |
| 01/25/13 3:10 PM | 9.40 | 01/25/13 3:10 PM | 0.94 | 01/25/13 3:10:00 PM | 27.7 |
| 01/25/13 3:15 PM | 9.40 | 01/25/13 3:15 PM | 0.96 | 01/25/13 3:15:00 PM | 27.6 |
| 01/25/13 3:20 PM | 9.40 | 01/25/13 3:20 PM | 0.81 | 01/25/13 3:20:00 PM | 27.7 |
| 01/25/13 3:25 PM | 9.40 | 01/25/13 3:25 PM | 1.02 | 01/25/13 3:25:00 PM | 27.7 |
| 01/25/13 3:30 PM | 9.30 | 01/25/13 3:30 PM | 0.96 | 01/25/13 3:30:00 PM | 27.7 |
| 01/25/13 3:35 PM | 9.30 | 01/25/13 3:35 PM | 0.79 | 01/25/13 3:35:00 PM | 27.7 |
| 01/25/13 3:40 PM | 9.30 | 01/25/13 3:40 PM | 0.96 | 01/25/13 3:40:00 PM | 27.8 |
| 01/25/13 3:45 PM | 9.30 | 01/25/13 3:45 PM | 0.97 | 01/25/13 3:45:00 PM | 28.1 |
| 01/25/13 3:50 PM | 9.60 | 01/25/13 3:50 PM | 0.80 | 01/25/13 3:50:00 PM | 28.1 |
| 01/25/13 3:55 PM | 9.60 | 01/25/13 3:55 PM | 1.04 | 01/25/13 3:55:00 PM | 28.3 |
| 01/25/13 4:00 PM | 4.30 | 01/25/13 4:00 PM | 0.97 | 01/25/13 4:00:00 PM | 28.3 |
| 01/25/13 4:05 PM | 6.10 | 01/25/13 4:05 PM | 0.77 | 01/25/13 4:05:00 PM | 28.5 |
| 01/25/13 4:10 PM | 6.10 | 01/25/13 4:10 PM | 1.03 | 01/25/13 4:10:00 PM | 29 |
| 01/25/13 4:15 PM | 6.10 | 01/25/13 4:15 PM | 0.96 | 01/25/13 4:15:00 PM | 28.8 |
| 01/25/13 4:20 PM | 6.10 | 01/25/13 4:20 PM | 0.75 | 01/25/13 4:20:00 PM | 28.7 |
| 01/25/13 4:25 PM | 6.10 | 01/25/13 4:25 PM | 1.09 | 01/25/13 4:25:00 PM | 28.6 |
| 01/25/13 4:30 PM | 6.20 | 01/25/13 4:30 PM | 0.96 | 01/25/13 4:30:00 PM | 28.5 |
| 01/25/13 4:35 PM | 6.10 | 01/25/13 4:35 PM | 0.96 | 01/25/13 4:35:00 PM | 28.5 |
| 01/25/13 4:40 PM | 6.10 | 01/25/13 4:40 PM | 0.96 | 01/25/13 4:40:00 PM | 28.4 |
| 01/25/13 4:45 PM | 6.10 | 01/25/13 4:45 PM | 0.96 | 01/25/13 4:45:00 PM | 28.5 |
| 01/25/13 4:50 PM | 6.00 | 01/25/13 4:50 PM | 0.96 | 01/25/13 4:50:00 PM | 28.5 |
| 01/25/13 4:55 PM | 6.00 | 01/25/13 4:55 PM | 0.96 | 01/25/13 4:55:00 PM | 28.5 |
| 01/25/13 5:00 PM | 6.00 | 01/25/13 5:00 PM | 0.95 | 01/25/13 5:00:00 PM | 28.5 |
| 01/25/13 5:05 PM | 6.00 | 01/25/13 5:05 PM | 0.95 | 01/25/13 5:05:00 PM | 28.5 |
| 01/25/13 5:10 PM | 6.10 | 01/25/13 5:10 PM | 0.94 | 01/25/13 5:10:00 PM | 28.6 |
| 01/25/13 5:15 PM | 6.20 | 01/25/13 5:15 PM | 0.96 | 01/25/13 5:15:00 PM | 28.7 |
| 01/25/13 5:20 PM | 6.00 | 01/25/13 5:20 PM | 0.82 | 01/25/13 5:20:00 PM | 28.6 |
| 01/25/13 5:25 PM | 6.00 | 01/25/13 5:25 PM | 0.81 | 01/25/13 5:25:00 PM | 28.5 |
| 01/25/13 5:30 PM | 6.00 | 01/25/13 5:30 PM | 0.96 | 01/25/13 5:30:00 PM | 28.4 |
| 01/25/13 5:35 PM | 6.00 | 01/25/13 5:35 PM | 0.96 | 01/25/13 5:35:00 PM | 28.4 |
| 01/25/13 5:40 PM | 6.20 | 01/25/13 5:40 PM | 0.96 | 01/25/13 5:40:00 PM | 28.4 |
| 01/25/13 5:45 PM | 6.10 | 01/25/13 5:45 PM | 0.94 | 01/25/13 5:45:00 PM | 28.5 |
| 01/25/13 5:50 PM | 6.10 | 01/25/13 5:50 PM | 0.95 | 01/25/13 5:50:00 PM | 28.5 |
| 01/25/13 5:55 PM | 6.20 | 01/25/13 5:55 PM | 0.96 | 01/25/13 5:55:00 PM | 28.5 |
| 01/25/13 6:00 PM | 6.20 | 01/25/13 6:00 PM | 0.95 | 01/25/13 6:00:00 PM | 28.5 |
| 01/25/13 6:05 PM | 6.40 | 01/25/13 6:05 PM | 0.94 | 01/25/13 6:05:00 PM | 28.5 |

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| 01/25/13 6:10 PM | 1.20 | 01/25/13 6:10 PM | 0.79 | 01/25/13 6:10:00 PM | 28.4 |