COLD CLIMATE FIELD TEST OF AN AIR-SOURCE HEAT PUMP WITH TWO-STAGE COMPRESSION AND ECONOMIZING

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Abstract: Within a university/industry partnership, a new air-source heat pump technology optimized for cold climates was designed, fabricated and tested in southern Indiana. The goal was to demonstrate that a cold climate heat pump (CCHP) uses less primary energy than traditional cold climate heating methods. For a field demonstration, two CCHPs were tested in two separate, nearly identical buildings located at a National Guard base during the 2012-2013 heating season. Each building is split into halves; allowing one half to be heated by conventional methods (natural gas furnace) and the other half to be heated by the CCHP. The energy consumption of each CCHP and furnace was monitored and recorded along with its operating conditions.

The CCHP utilized two-stage compression with an economizing refrigerant loop. The CCHP was designed for a heating capacity of 19 kW (~65 kBTU/h) at an outdoor temperature of -20°C (-4°F). The measured coefficient of performances (COPs) was in a range of 1 to 3 for outdoor temperatures from -8°C (18°F) to 18°C (64°F). A primary energy savings of 19% was determined when comparing the two-stage heat pump to a natural gas-fired furnace.

Key Words: two-stage, field demonstration, economizing, air-source heat pump

1 INTRODUCTION

For heating, a majority of HVAC systems utilize combustion of fossil fuels to maintain the space conditions; while in moderate climates of the United States, heat pumps powered by electricity are utilized. In climates with more extreme winter temperatures, below 0°C (32°F), heat pumps using the ambient air as the heat source have reduced capacities and require inefficient electric resistors as supplemental or back-up heat. As a result, home in these colder climates rely heavily on combustion based heating sources. Air-source heat pumps that are designed specifically for cold climates can operate efficiently at the extreme temperatures while satisfying the building load.

Within a previous research project by Bertsch et al., 2008, the use of two-stage compression with economizing was investigated for an air-source heat pump as a cold climate heat pump. The system was tested in a laboratory with ambient temperatures as low as -30°C (-22°F) and had second law efficiencies up to 45%. The United States Department of Defense, DoD, mandates for reducing energy consumption of military installations has presented an opportunity to demonstrate the CCHP's ability to help reach the reductions in energy consumption. Conducting a field test of a two-stage compression air-source heat pump with economizing moves the technology closer to commercialization and potential deployment for the DoD.

A DoD program, the Energy Security Technology Certification Program or ESTCP, acts as an environmental technology demonstration and validation program. The goal of the program is to identify and demonstrate promising technologies that address the DoD's environmental requirements. A heat pump designed for cold climates was demonstrated in the field at a military installation. A contract was awarded for a two year project, testing two air-source heat pumps designed for cold climates, at a military base in southern Indiana during an entire heating season. The heat pumps were to be compared to the existing heating systems, natural gas furnaces, using 6 performance objectives. The objective of the project was to reach or surpass the success criteria listed for each performance objective (ESTCP Final Report 2013). The focus of this paper will be on the reduction of primary energy.

2 SYSTEM DESIGN AND FIELD DEMONSTRATION

To determine the required capacity of the heat pump and to establish the experimental design, the building on the military base had to be known. A standard building that was not custom to Camp Atterbury was desired to allow for a straightforward comparison to buildings on other military bases. Also, since two heat pumps were being tested, two buildings were needed that had a roughly identical layout and usage type. The building type selected was an 80 person barracks with a north-south rectangular orientation, totaling roughly 560 m^2 (6,000 ft^2). The layout was considered to be standard for barracks. Two living quarters situated on opposite ends with a shared lavatory in the center. Each living quarters had a dividing wall separating ¾ of the space. A mechanical room located in next to the shared lavatory housed all the HVAC equipment and on-demand water heaters. Due to the upgrades that eliminated the storage tanks for hot water, the mechanical room had plenty of space to house the equipment needed for the field demonstration.

Two heat pump systems were fabricated and installed into two separate barracks buildings. The heat pump in Building 114 was connected to providing heating for the northern half while the system in Building 113 was connected to the southern half. Installing the heat pump into different halves allows for two comparisons; one within individual buildings comparing the north to south performance and the other between buildings on a north to north, or south to south basis. If any differences occur due to orientation, the two comparisons should capture this. Figure 1 shows a diagram of both buildings where the colors indicate the heat pump (yellow), conventional system (red), mechanical room (green), and the lavatory (blue). To maintain a fair comparison, on-base personnel were asked to maintain an equal level of occupancy between both buildings.

Due to the system being novel and having a long testing period, the conventional system being replaced by the heat pump had to be left connected to the ductwork. The idea was, by leaving the existing system in place there would be the ability to switch between the conventional and heat pump system. Whenever the heat pump was taken off-line, the existing HVAC system would be brought on-line as an emergency back-up.

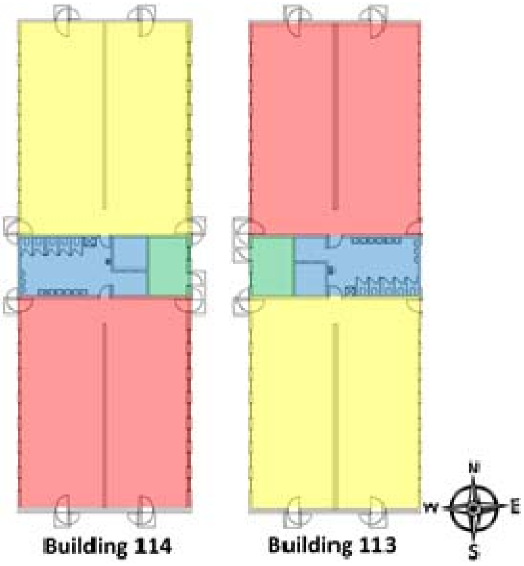

Figure 1: Experimental Design for Heat Pump Installation

The monthly building loads for both cooling and heating were calculated using a simple building modeling software. The maximum heating load is 18.9 kW (64.4 kBTU/hr) at a dry bulb temperature of -20.6°C (-5°F). The design point for the heat pump selected to be 19 kW (64.8 kBTU/hr) at an ambient temperature of -20°C (-4°F). Using the design point and setting the heating load to zero at an ambient temperature of 20°C (68°F), a linear heat load of the building as a function of the ambient temperature was made. The weather data to determine the amount of hours was used from TMY3, typical meteorological year 3, for Indiana. By designing the heat pump capacity to meet the building load at 20°C (-4°F), the need for supplemental or back-up heat is greatly reduced.

The schematic of a bread board two-stage heat pump with economizing from Bertsch 2005 was modified to reduce the complexity of the system and increase the robustness of its operation. The schematic of the system constructed can be seen in Figure 2. Oil and refrigerant inventory management systems were added per recommendations from laboratory testing. Isolation valves for ease of component replacement and sight glasses used during the laboratory testing were eliminated to have a system ready for commercialization. The loop allowing for the heating of water was also eliminated as the heat pump only needed to provide heating by air. The system maintained the use of a “low-stage bypass line” that allows for compressor 2 to run without the operation of compressor 1. This will be referred to as single-stage mode. Compressor 1 cannot run without compressor 2 operating and when both compressors are running; this will be referred to as two-stage mode.

An EES model was used to calculate the heating COP of the heat pump for the different operating modes and outdoor temperatures. Even at the extreme outdoor temperatures below -10°C (14°F), the predicted heating COP is above 2.0 based on the EES results (Caskey et. al 2012).

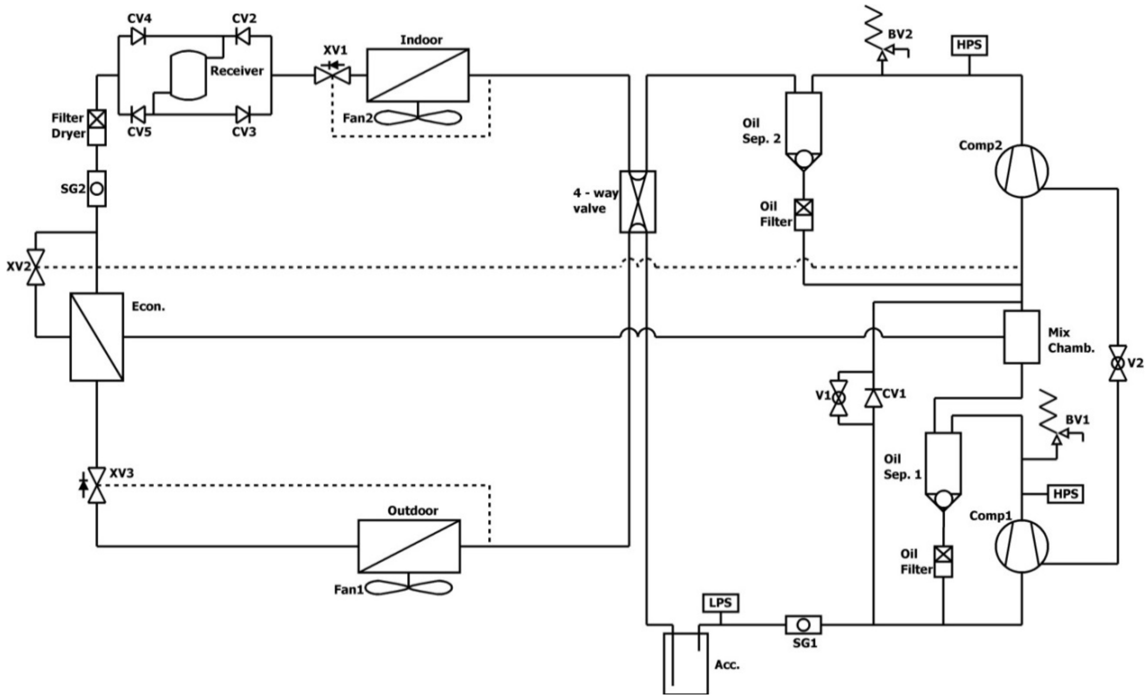

Figure 2: Schematic of Two-Stage Heat Pump with Economizing

3 SAMPLE OPERATION

The system 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. One 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 during heating mode. 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. When running in two-stage mode, the sensors will read different pressures since the suction port of each compressor is operating at a different pressure.

Figure 3: Heat Pump Refrigerant Pressures during Single and Two-Stage Figure **3** is a plot of the four different pressure readings during a selected data set versus time. The heat pump operates in two-stage mode for most of this data set as can be identified 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. The difference between the intermediate and low pressure readings is an indicator of when the heat pump is operating in two-stage mode.

The heat pump also had seven temperature sensors installed to monitor refrigerant conditions throughout different locations. Two additional temperature sensors were equipped by the manufacturer on the high-stage compressor; including the discharge temperature and the motor temperature. These readings provided 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

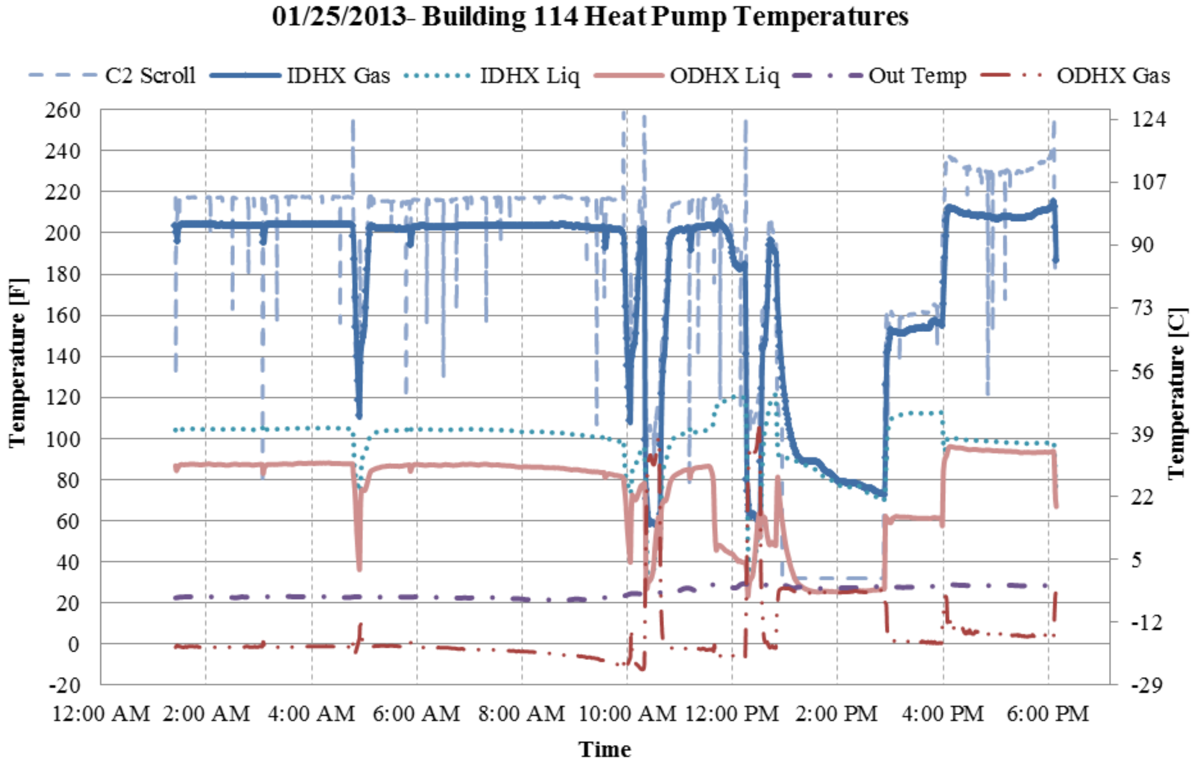

Figure 4.

The temperature drop across the indoor heat exchanger, IDHX, 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 55.6°C (100°F). The operation of the outdoor heat exchanger, ODHX, 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 11.1°C (20°F) for the data set investigated. The smaller this difference is, the higher the efficiency is of the outdoor heat exchanger.

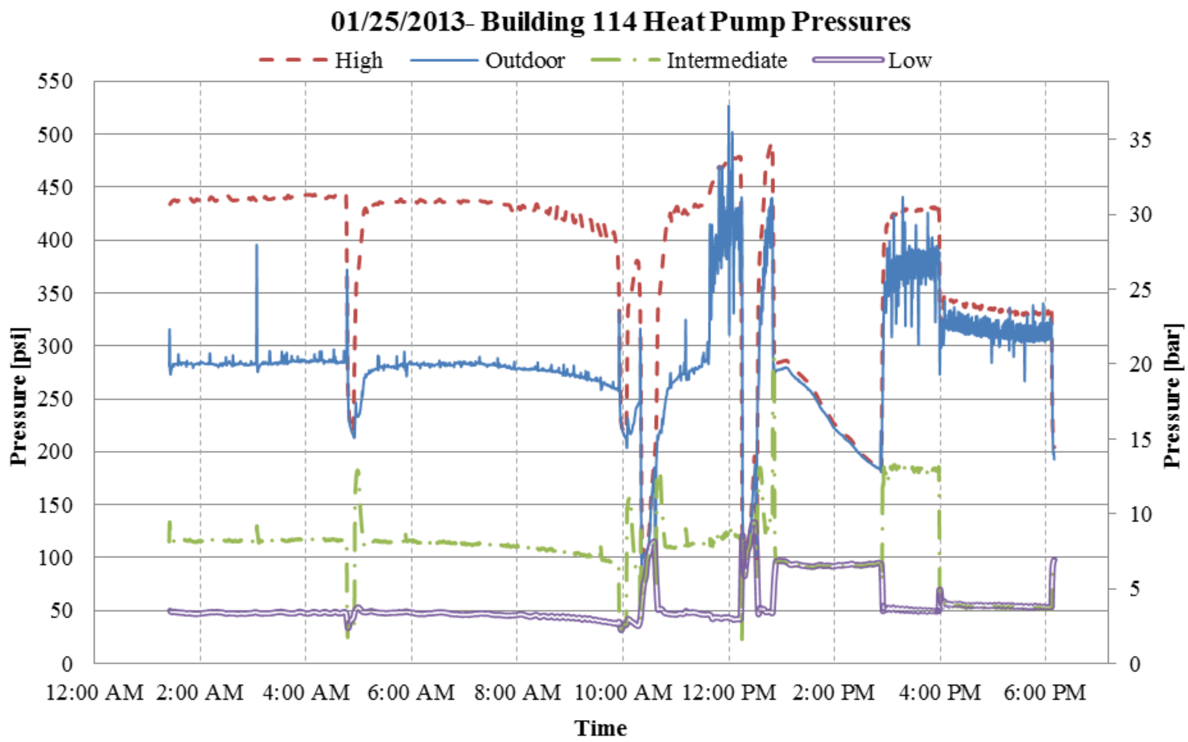

Figure 3: Heat Pump Refrigerant Pressures during Single and Two-Stage

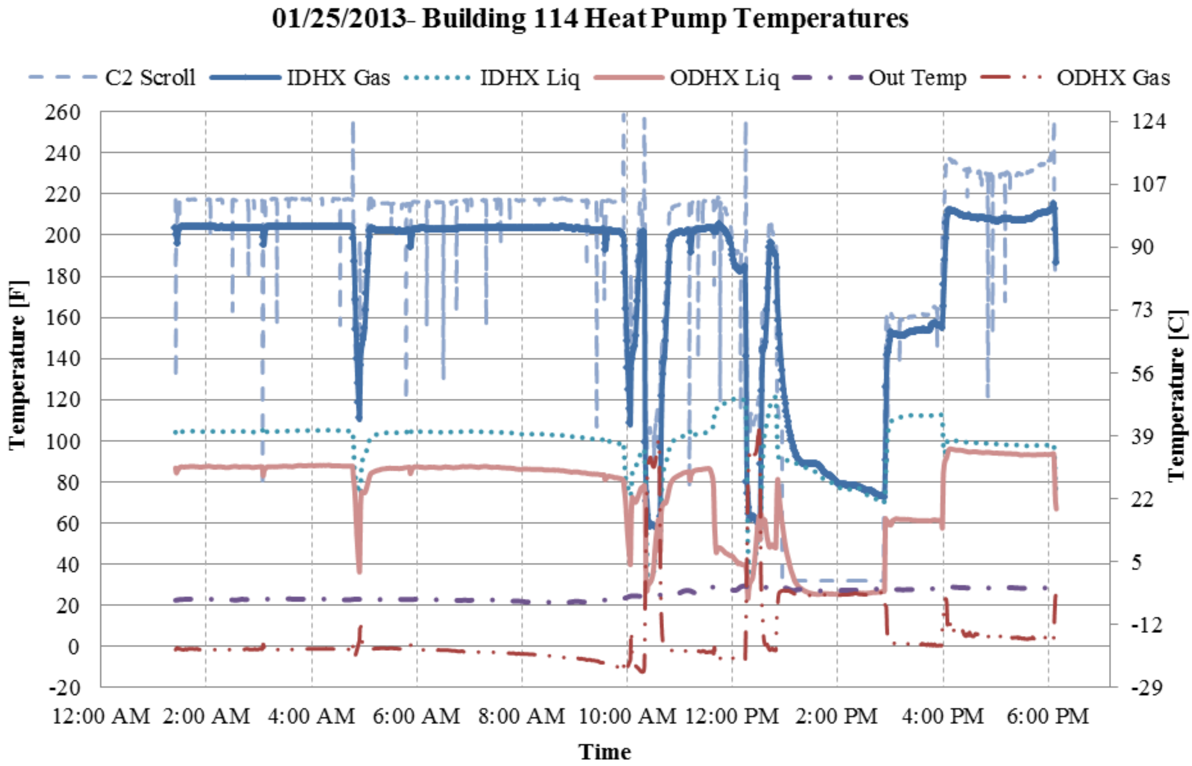

Figure 4: Heat Pump Refrigerant Temperatures during both Single and Two-Stage Operation

The highest temperatures recorded are at the discharge of the high-stage compressor, also known as the scroll temperature. The measured temperature is depending on single versus two-stage operation. Figure 4 indicates that the scroll temperature is lower between the hours of 2:00 am and 10:00 am than the reading between the hours of 4:00 pm and 6:00 pm. This difference is due to the mixing chamber cooling the discharge gasses leaving the low stage compressor before entering the high stage compressor. By cooling the 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 volumetric efficiency of the compressor leading to higher refrigerant flow rates.

4 FIELD RESULTS

Due to complications during the field test, an entire heating season of data could not be collected. Only a grouping of data periods were identified between long periods of system shut-down. The primary energy calculations of the measured heat pump electrical consumption are compared to the natural gas furnace consumption.

The heat pump in Building 114 recorded eight data sets covering about 315 hours and the unit in Building 113 recorded only three data sets with a little over 100 hours of data providing a total of eleven data sets with over 400 hours of data. The field demonstration covered a large range of temperatures. The highest instantaneous temperature recorded for a data set was 27.8°C (82°F) and the highest average temperature was 18.3°C (65°F). The lowest recorded instantaneous temperature was -8.3°C (17°F) and the lowest average temperature was -2.8°C (27°F). The main details of each data set, such as the collection period and temperature ranges, are shown in Table 1.

The heat pump uses only electricity, while the natural gas furnace 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 natural gas furnace, the standard cubic feet, SCF, is converted using Equation (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 2 under the natural gas furnace category. The second factor is a simple conversion between BTUs and kWh.

$$\text{SCF Natural Gas to kWh} = 1\ \text{SCF} * \frac{1012\ \text{BTUs}}{\text{SCF}} * \frac{1\ \text{kWh}}{3412\ \text{BTU}} \tag{1}$$

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 (η_{PP}) and transmission line (η_{TL}) efficiency were assumed. The efficiency values assumed are listed in Table 2 under the natural gas power plant category. It should be mentioned that it is also assumed all the electricity is produced by a natural gas power plant. This provides 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 (2).

$$\text{Meter Energy to Primary Energy} = \frac{\text{kWh}_{\text{EM}}}{\eta_{\text{PP}} * \eta_{\text{TL}}} \tag{2}$$

Table 1: Summary of Data Sets Collected During Field Demonstration

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]
114	1	10/12/2012 9:40 PM	-	10/13/2012 1:10 PM	15:30	42 / 5.6	53 / 11.7	68 / 20
	2	10/15/2012 2:40 PM	-	10/16/2012 1:05 PM	22:25	39 / 3.9	52 / 11.1	65 / 18.3
	3	10/25/2012 8:22 AM	-	10/26/2012 3:29 PM	31:07	46 / 7.8	65 / 18.3	82 / 27.8
	4	10/27/2012 1:50 AM	-	10/27/2012 3:45 PM	13:55	38 / 3.3	45 / 7.2	55 / 12.8
	5	1/4/2013 11:00 AM	-	1/5/2013 9:05 AM	22:05	17 / -8.3	27 / -2.8	37 / 2.8
	6	1/10/2013 9:35 AM	-	1/17/2013 1:55 AM	160:20	19 / -7.2	41 / 5	67 / 19.4
	7	1/25/2013 1:25 AM	-	1/25/2013 6:10 PM	16:45	21 / -6.1	25 / -3.9	30 / -1.1
	8	2/14/2013 10:30 AM	-	2/15/2013 7:25 PM	32:55	30 / -1.1	39 / 3.9	53 / 11.7
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
	10	2/13/2013 3:00 PM	-	2/14/2013 12:50 AM	9:50	40 / 4.4	44 / 6.7	49 / 9.4
	11	2/14/2013 9:40 AM	-	2/15/2013 9:10 PM	35:30	29 / -2.2	40 / 4.4	59 / 15

From this conversion method, it can also be determined what the performance requirements are of the heat pump to achieve a primary energy reduction. By taking the product of the efficiencies, a total efficiency of primary energy to electrical kWh is at 0.391. Then using the definition of a heating COP, the amount of 1 unit of heat output by the heat pump for 1 unit of primary energy consumed determines the COP goal of a heat pump. The COP goal is calculated by taking the inverse of the total efficiency assumed, 0.391 to arrive at a heating COP of 2.56. Heat pump heating COP values above this requirement will reduce the amount of primary energy consumed compared to a system that converts 1 unit of primary energy into 1 unit of heat output.

Table 2: Assumptions for Primary Energy, Energy Costs and Emissions

Natural Gas Power Plant	**Camp Atterbury Rates**
Equivalent BTUs for a kWh	Electricity
3412 BTU/kWh	0.08 $/kWh
2011 Natural Gas Heat Rate	Natural Gas
8152 BTU/kWh	0.47 $/therm
Power Plant Efficiency	
0.42	**IN Residential Rates**
2009 T&D Losses and Unaccounted	Electricity
260,580,117 kWh	0.105 $/kWh
2009 Total Net Elec. Generation	2012 Natural Gas
3,950,330,928 kWh	9.46 $/1000 CF
Transmission Efficiency	**EPA Emission Factor**
0.93	120000 lbs/1E6 CF
Natural Gas Furnace	
2011 Heat Content of Natural Gas – IN	
1012 BTU/CF	
Convert Natural Gas CF to kWh	
0.30 kWh/CF	

The results are presented in Figure 5 and Figure 6. In these figures, the primary energy consumptions of the heat pumps located in Building 113 and Building 114 are compared to the primary energy consumptions of the furnaces in both buildings, respectively. In addition, the heat pump heating COPs are shown. This comparison allows for insight on the influence of building orientation on the heat pump operation as well as the operating performance of the heat pump. In Figure 5, the primary energy savings changes depending on which furnace the heat pump is compared to. When compared to the furnace in Building 113, the heat pump has more primary energy consumption for data sets 4, 6 and 7, and while comparing the heat pump to the furnace in Building 114, it shows more primary energy consumption for data sets 1, 2 and 6.

In Figure 6, the heat pump 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 furnace in Building 113. These inconsistences could be due to the building orientation or if other influences are considered, could be due to 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 heat pump more often consumes less primary energy than either building's furnace. The second consideration is the fluctuation of the system heating COP. To reach a primary energy savings, the heat pump should have a heating COP above 2.56 as described in the paragraph after Table 1. Only data sets 5, 8 and 11 show COPs above this minimum limit. Therefore, data sets that show a reduction in primary energy consumption without a COP above this limit are influenced by the previously described factors where the furnace and heat pump did not provide the same amount of heat to the space. In summary, when considering total primary energy consumed by each system, across all data sets, the heat pump shows a primary energy savings of 19% relative to a natural gas furnace.

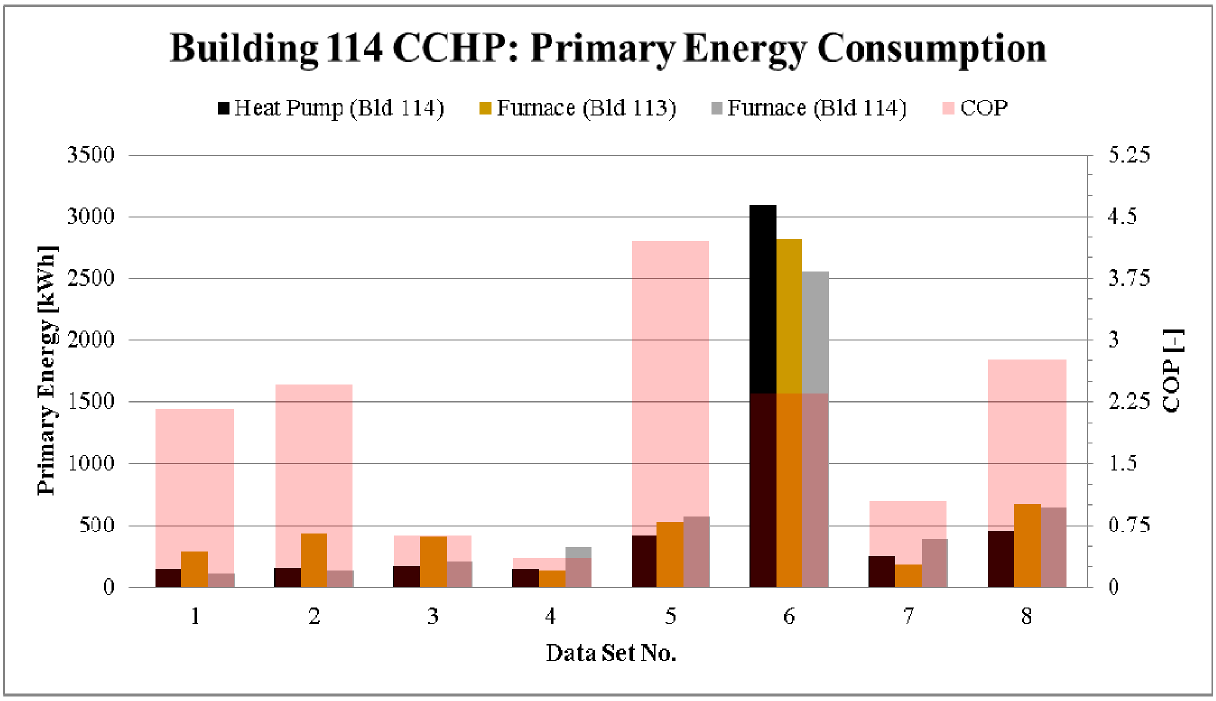

Figure 5: Primary Energy Consumption and COP Summary for Heat Pump in Building 114

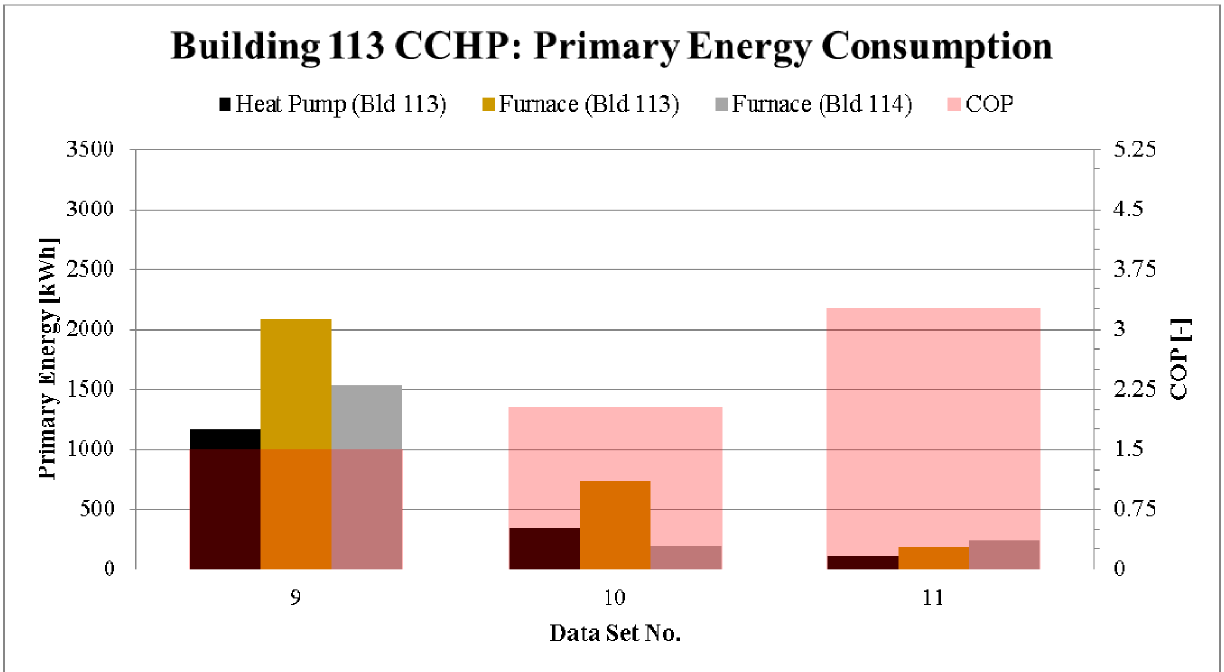

Figure 6: Primary Energy Consumption Summary and COP for Heat Pump in Building 113

5 DISCUSSION AND CONCLUSION

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 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 heat pump having more primary energy consumption than the furnace 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 furnace, 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 heat pump. While for a furnace, the controls are more straightforward since cycling for a furnace does not create a large amount of inefficiencies.

Data sets 1 and 2 are the first two data sets collected when the heat pump was installed. The significance of being the first sets collected is the designed heat pump controls experienced 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 heat pump. The impact of the controls on the 6th data set could also explain the higher primary energy consumption seen of the heat pump. Over this long period of time the heat pump 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 produced conditions for frost build up on the outdoor coil and thus required very inefficient defrost cycles. This is due to all of the heating from the heat pump is done by electric resisters that have at best COPs of 1. The lower COPs of the heat pump mean more primary energy consumption to satisfy the same load. The colder temperatures also require the heat pump 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 varying heating load throughout a typical day. Therefore in the 6th data set, the two-stage and defrost controls were utilized extensively having a large impact on the overall performance of the heat pump.

During the field demonstration several complications were encountered that both affected the quantity of data collection and the system performance. The design of the heat pump controls resulted in a slow response to the heating demand. Adjustments in the control program for selecting the system capacity are needed. The compressor would flood with liquid refrigerant at start-up and during operation. Changes in both the control strategy and the selection of the system components would prevent liquid flooding of the compressor. The primary heating expansion valve experienced complications due to a low subcooling level. Improving the refrigerant inventory system of the heat pump is one area that could improve the amount of subcooling. In spite of these complications, the heat pump primary energy consumption is calculated to provide a 19% reduction when compared to a natural gas furnace.

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