Variable-Speed Heat Pumps for Energy Efficiency and Demand Response

Field Testing High-Efficiency Systems in a Simulated-Occupancy Home in Knoxville, Tennessee

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Technical Update, May 2014

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ABSTRACT

This report details an ongoing effort to study high-efficiency variable speed heat pump systems at the Campbell Creek Research Homes, a set of simulated-occupancy research homes in Knoxville, Tennessee. For this report, a research home was retrofitted with two high-efficiency variable speed heat pump systems, and data collected was compared with prior years' data from the same home for standard-efficiency, single-speed heat pump systems. The variable speed systems were also tested during the summer for their ability to provide demand response by either reducing the variable output of the heat pumps to a minimum capacity or turning off upon receiving a simulated signal. The report details the energy savings from the retrofit, contrasts the various operating modes of the variable speed systems, and quantifies the demand savings potential of each of the demand response modes.

Keywords

Air conditioners
Demand response
Energy efficiency
Heat pumps
Residential
Variable speed heat pumps

SUMMARY

This report highlights an ongoing study at the Campbell Creek Research Homes in Knoxville, TN in which high-efficiency, variable speed heat pump (VSHP) systems are being studied for energy efficiency and demand response capabilities. The Campbell Creek Research Homes, referred to as the CC Houses, are simulated-occupancy homes owned and operated by the Tennessee Valley Authority in collaboration with the Electric Power Research Institute (EPRI) and Oak Ridge National Laboratory (ORNL). The three homes (CC1 – builder grade, CC2 – retrofit efficient and CC3 – near zero energy) serve as test beds for various building technologies. This report addresses testing in CC House #1. This project is a collaboration between EPRI, TVA, the Bonneville Power Administration, and Southern Company.

Traditional HVAC practices size single-speed heat pumps (SSHP) based on cooling season loads and heating demands are accomplished with the compressor meeting a portion of the load and the remainder being supplied by resistance back-up heating. Electric resistance back-up heating is less efficient and contributes to Utility peak demands during periods of colder weather. In contrast, variable speed heat pumps like those investigated in this report can be sized to accommodate heating design conditions (reducing auxiliary heating needs) because variable speed system can still control humidity in part load operation during the cooling season. Minimizing strip heat and matching part load conditions by modulating system capacity can yield efficiency improvements and peak load reductions.

This study showed that when comparing data for like outdoor temperatures, the VSHP (2 ton downstairs unit and 3 ton upstairs unit) used 17-38% less power during heating and 34-38% less power during cooling conditions than the smaller SSHP (1.5- and 2.5 ton systems respectively). Though the weather was more extreme during the nearly two years of SSHP operation, comparing months with similar heating and cooling loads (Heating and Cooling Degree Days) showed a 34% energy savings in heating (Feb-11 vs. Dec-12) and a 44% energy savings in cooling (Jul-13 vs. Aug-12) with the high-efficiency VSHP systems. A brief overview of winter 2013-14 data which featured more extreme cold weather is shown in the addendum of this report and will be addressed in future reports.

The VSHP systems evaluated in this study have both Comfort and Efficiency operating modes. Comfort mode features lower airflow rates, higher supply air temperatures in heating, lower supply air temperatures in cooling and more latent cooling (dehumidification); while Efficiency mode sacrifices some of these space conditioning features in order to save energy. Testing of both Comfort and Efficiency modes showed a reduction in average power (5% in the higher temperature bins) and only slight improvements in efficiency in both heating and cooling mode using the Efficiency mode; however, in cooling operation savings were accompanied by an increase in the indoor humidity. The increase in humidity may be uncomfortable to some occupants and cause them to lower the thermostat set-point; therefore, comfort mode operation may be more desirable from the consumer's perspective.

During the relatively mild 2013 summer (continuing research will hopefully capture more extreme temperatures), the VSHP systems were tested for demand response capabilities utilizing two DR responses: "DR-Off" and "DR-Min". For "DR-Off" events, the heat pump shuts completely off and remains off until the signal (time) ends. In "DR-Min" events, the heat pump

is programmed not to exceed a minimum allowable compressor speed (although it may still cycle between minimum and off).

During an examined “DR: Min” event, the upstairs heat pump, which had been running at higher power, reduced to a steady minimum capacity; while the downstairs heat pump, which had been cycling, continued to cycle between minimum and off, but with increased frequency. This resulted in an approximate 0.5 kW load reduction for the total 5 ton system. During the event period the indoor dry-bulb and wet-bulb temperatures remained largely unchanged.

On the other hand, during an examined “DR: Off” event, both the upstairs and downstairs heat pumps turned off completely. Before the event, the upstairs unit had been running consistently at a high capacity and during the scheduled event reduced power by approximately 1.6 kW. Before the event, the downstairs unit was cycling at a low capacity and low duty cycle and therefore did not contribute to an appreciable, sustained demand reduction (although this did prevent some minimum capacity cycling). After the event, both units resumed operating at a greater-than-minimum capacity because the indoor dry-bulb and wet-bulb temperatures increased by over 5°F during the three-hour event period.

During DR testing, it was noticed that the zonal interactions of the two heat pumps (upstairs and downstairs) may have reduced the effectiveness of the DR signal especially under “DR: Min” operation. Since the cooling outside of DR events is provided predominantly by the upstairs heat pump (cool air falls into the space below), the downstairs heat pump typically ran on a low duty cycle; however, when the DR control caused the upstairs heat pump to reduce operation, the downstairs unit increased its’ duty cycle (staying within its’ “minimum output” programming). Therefore, the net result for “DR: Min” events was only a slight reduction in power compared to hours when there was no DR event. These findings suggest that for multi-system sites, a simple “DR: Min” control approach may not suffice and alternative schemes, such as combining one “DR: Min” command with one “DR: Off” command, may produce better demand reductions with occupant comfort. Strategies such as this will be tested in the continuation of this study.

Throughout this research effort various zonal interactions between the upstairs and downstairs zones during both the cooling and heating seasons were observed. These zonal interactions are not traditionally accounted for in the HVAC design process. In the cooling season the upstairs system carried a disproportionate amount of the cooling load since the chilled air fell into the downstairs zone. The trend is reversed in the heating season. Since hot air rises, the downstairs heat pump carries more of the whole-house heating load in the winter. Zonal bleed-over should have an impact on the sizing, performance and demand response reaction of multi-zone systems.

Continuing research on this project will seek to investigate these impacts and compare the traditional two unit (downstairs and upstairs) system to a single variable speed unit with zone control via air distribution to increase overall system efficiency and customer comfort.

EXECUTIVE SUMMARY

Introduction

This report highlights an ongoing study at the Campbell Creek Research Homes in Knoxville, TN in which high-efficiency, variable speed heat pump systems are being studied for energy efficiency and demand response capabilities. The Campbell Creek Research Homes, referred to as the CC Houses, are simulated-occupancy homes owned and operated by the Tennessee Valley Authority in collaboration with the Electric Power Research Institute (EPRI) and Oak Ridge National Laboratory (ORNL). The three homes (CC1 – builder grade, CC2 – retrofit efficient and CC3 – near zero energy) serve as test beds for various building technologies. This report addresses testing in CC House #1. This project is a collaboration between EPRI, TVA, the Bonneville Power Administration, and Southern Company.

For this study, the CC House #1 had two Carrier Greenspeed Variable Speed Heat Pumps (VSHPs) installed and instrumented to simulate a new-construction household with high-efficiency air conditioning. In addition, prior years' data in which a single-speed, standard-efficiency single-speed heat pump (SSHP) was analyzed and used to compare results and estimate energy and demand savings. The VSHPs had nominal capacities of 2 tons for the downstairs unit and 3 tons for the upstairs unit. The SSHPs were 1.5- and 2.5-ton systems, respectively. The objectives of the study include understanding the energy efficiency and demand response performance of the new VSHP systems, comparing its' capabilities with the baseline technology, as well as simulating demand response (DR) performance by simulating DR events over the course of the summer.

For baseline data, the prior years' data for CC House #1 is used. EPRI used data from July, 2010, through October, 2012 during which time two standard-efficiency (13 SEER, 7.7 HSPF) single-speed air source heat pump systems were installed in CC House #1. The thermostats were calibrated and set to 71°F. In late October and early November, 2012, two Carrier Greenspeed variable capacity heat pumps were installed in the home, with a 2-ton on the ground floor and 3-ton on the second floor. The GreenSpeed line has a nominal 20.5 SEER and 13 HSPF rating. The thermostats were tuned to match the previous systems' calibration.

The VSHP systems operate with two modes: Efficiency Mode and Comfort Mode. While the full difference between the modes is not disclosed by the manufacturer, some generalities are known: Efficiency Mode provides a higher airflow rate of warmer (cooling mode) or cooler (heating) supply air. The proposed benefit is better efficiency by lessening the pressure difference that the compressor must maintain, at the expense of a smaller increase in fan power. In cooling mode, one implication of this is reduced latent cooling (humidity removal).

Figure ES-1
VSHP Outdoor Units

For demand response testing, an external signal was supplied to the outdoor units and controlled on a set schedule. The VSHPs in this test have two modes of demand response: “Min Cooling” mode, in which the device will not exceed its’ minimum operating capacity, and “Off” mode, in which the system turns off. In order to understand both operating modes, all the while capturing valuable baseline (no DR event) data, a schedule was developed. The schedule mixed the different operating modes, and DR events of varying lengths and starting times. The project team agreed upon an equal balance of Comfort Mode and Efficiency Mode. The group also decided to use more “DR: Min” events than “DR: Off” events, since it was felt that the “DR: Off” response would be more predictable with limited data, while DR: Min is more complex. The schedule included baseline days with no events throughout, to attempt to capture all variation in weather with baseline operation.

Results - Efficiency

The larger-capacity and higher-efficiency VSHP systems used significantly less energy than the baseline SSHP systems. Figure ES-2 shows the average power totals for the SSHP and VSHP systems for outdoor temperature bins in which sufficient data was collected. As can be seen, the SSHP was exposed to a wider range of temperatures, as the test period for the VSHP had milder weather; a brief overview of winter 2013-14 data which featured much colder weather is shown in the addendum of this report and will be addressed in future reports. The results in Figure ES-2 show that, in each mode the VSHPs used significantly less power on average for like conditions than the SSHP. Comparing HP1 (downstairs heat pump) with HP2 (upstairs heat pump) for both pairs of systems, HP1 used more power on average than HP2 in heating mode, and HP2 used more than HP1 in cooling mode. The values and percentage reductions are also tabulated in Table 3-2 in the body of the report.

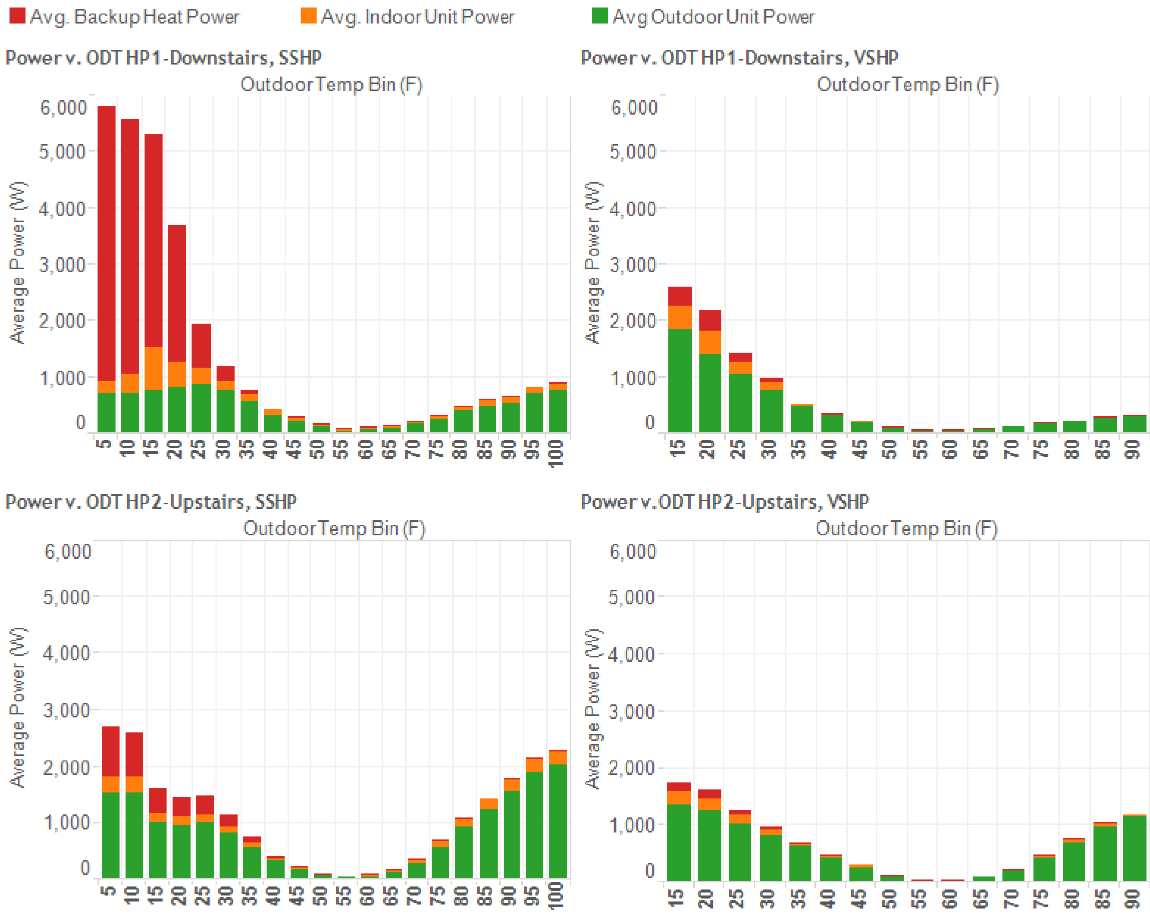

Figure ES-2
Power of SSHP and VSHP Systems vs. Outdoor Temperature Bin

The results shown in the figure above suggest that for similar weather, the VSHPs use less energy. The energy consumption over the entire heating or cooling seasons could not be directly compared due to significant differences in weather; however, using "degree days" as a basis, months with similar heating or cooling loads were compared. Degree days can be calculated for heating or cooling, and are a measure of the deviation of temperatures from the building's balance point over time. Degree days are calculated using outdoor temperature and a selected balance point; commonly, 65°F is used for non-specific comparison purposes. One heating degree day, for example, means that the outdoor temperature was below the designated temperature (65°F) for one day, by one degree.

The results supported energy savings from the VSHP systems. For instance, February, 2011 (580 Heating Degree Days) and December, 2012 (588 Heating Degree Days) can be compared as the heat load over the month was similar. In December, 2012, the VSHPs consumed 747 kWh; in February, 2011, the SSHPs consumed 1,131 kWh. This represents a 34% energy savings. Similar savings were seen in cooling mode operation: in July, 2013, there were 333 Cooling Degree Days and the VSHPs used 432 kWh, while in August, 2012 there were 336 Cooling Degree Days and the SSHPs used 778 kWh. This represents a 44% energy savings. While degree days are not a complete metric for definitive conclusions, these results strongly suggest energy savings due to the system change.

In addition to comparing performance of the 1.5- and 2.5-ton SSHPs with the 2- and 3-ton VSHPs, it is of interest to study the different operating modes of the VSHPs. In heating mode, the system was allowed to run in Comfort Mode and Efficiency Mode, and defrost could be performed with or without electric resistance heat running to prevent "cold blow" during defrost.

Figure ES-3 shows the average total power and indoor air temperature for both VSHPs for Comfort and Efficiency Modes. This graph shows a reduction in average power in Efficiency mode, and that in both modes the indoor temperature was maintained, though it was higher in Comfort Mode (likely relating to an increased use of resistance heat by HP1). It also shows that when HP1's power decreased in Efficiency Mode, HP2's power increased. This is likely representative of the shared zoning of the two heat pumps: when HP1 provides heat, some of that heat rises into the upstairs zone and HP2 is needed less; when HP1's output decreases, HP2's increases such that the upstairs zone's thermostat is still satisfied.

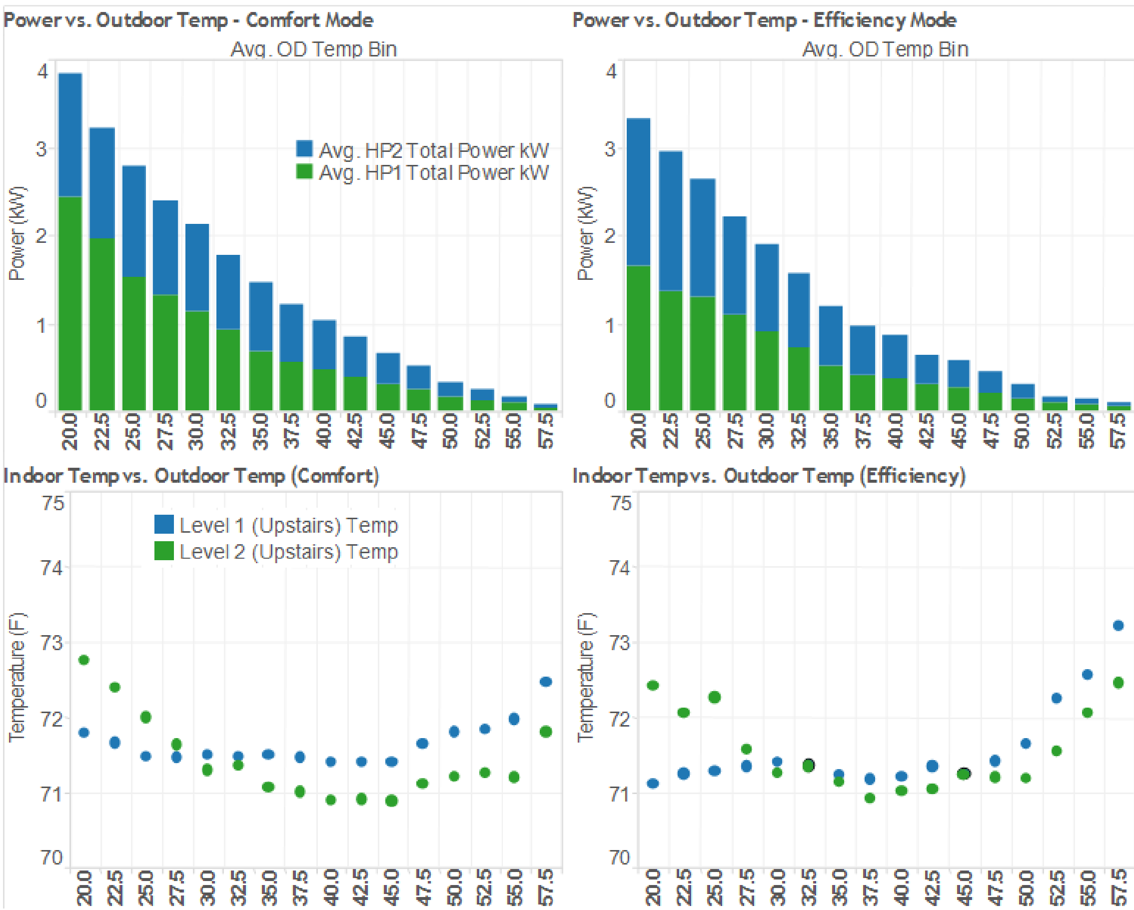

Figure ES-3
HP1 and HP2 Power, and Level 1 and Level 2 Indoor Temperature vs Outdoor Temperature for Comfort and Efficiency Mode in Heating Operation

As a general note of comparison, the coefficient of performance (COP, the ratio of heating capacity to power input in like units) and the supply air flow rate were higher, in Efficiency Mode than Comfort Mode. The supply air temperature was lower in Efficiency Mode than Comfort Mode.

In Cooling Mode, a similar comparison of Comfort Mode and Efficiency Mode can be made. Figure ES-4 shows the average power for each heat pump, as well as the average indoor dry-bulb and wet-bulb temperatures for each zone. The dry-bulb temperature is the temperature sensed by a thermometer. Wet-bulb temperature is a combined measure of temperature and humidity, and is always less than or equal to the dry-bulb temperature. The lower the wet-bulb temperature is, the less humidity is in the air.

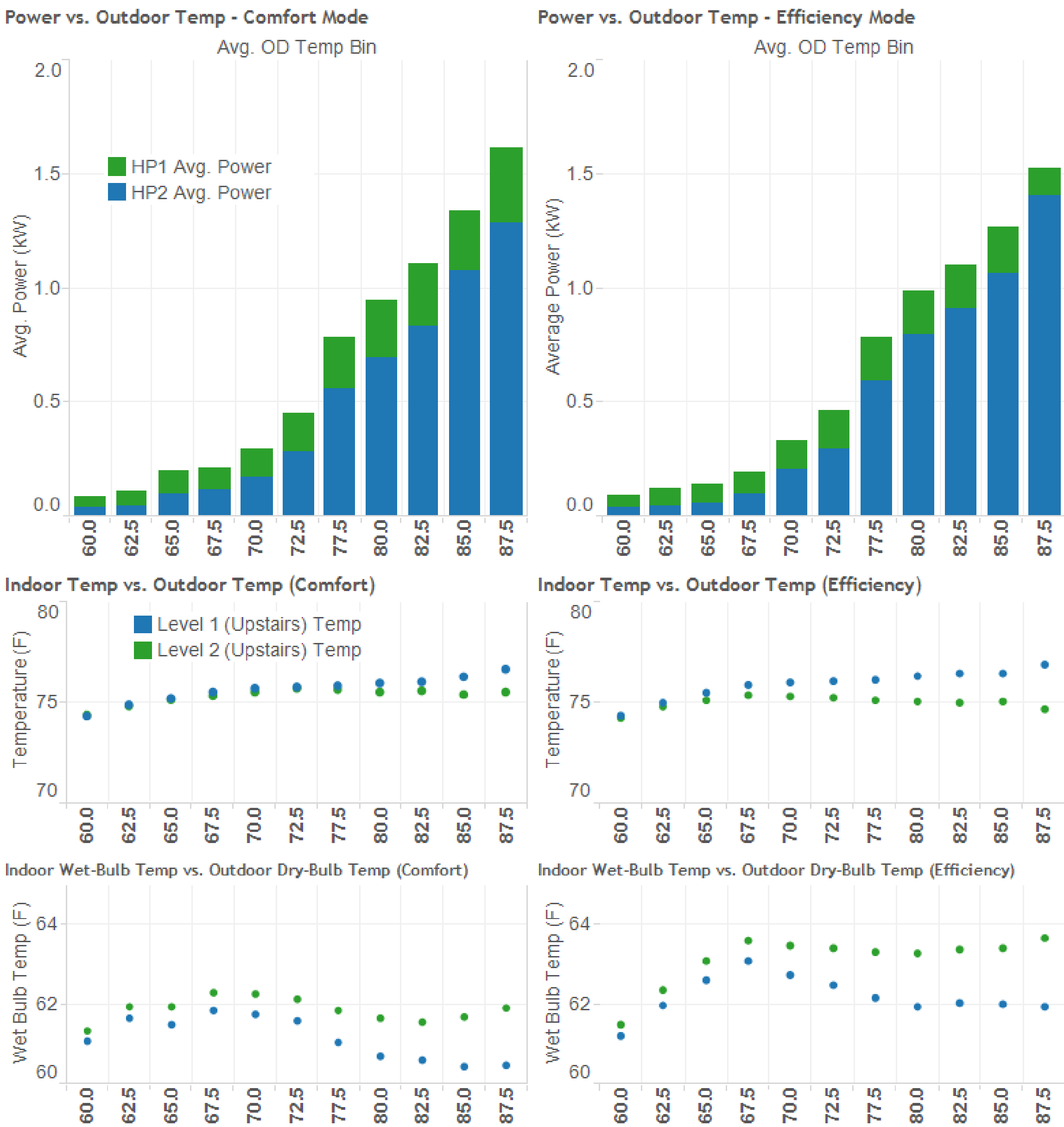

Figure ES-4
Average Power of HP1 and HP2 for Comfort Mode and Efficiency Mode, Indoor Dry-Bulb and Wet-Bulb Temperature, vs. Outdoor Temperature Bins for Cooling

Figure ES-4 shows that the average power of HP1 and HP2 combined was only slightly lower in Efficiency Mode. In the higher temperature bins, the average power was approximately 5% lower in Efficiency Mode than Comfort Mode. HP2 used more energy in Efficiency Mode than

Comfort Mode, and HP1 used more in Comfort Mode than Efficiency Mode. This is again indicative of inter-zonal effects. The slight reduction in power consumption comes with a trade-off: the average wet-bulb temperature in each zone was higher for Efficiency Mode, indicating that some occupants may feel uncomfortable due to higher humidity.

In Efficiency Mode, the Energy Efficiency Ratio (EER, the ratio of cooling in Btu/h to power in kW) was higher, the airflow was higher, the latent cooling (dehumidification) was lower, and the supply air temperature was higher than in Comfort Mode.

Results – Demand Response

During the summer, the systems were tested for demand response capabilities with two programmed responses: in "DR: Off" events, the heat pump is programmed to turn off and remain off until the signal ends. In "DR: Min" events, the heat pump is programmed not to exceed its' minimum allowable compressor speed (though it may still be off, or cycle on and off at minimum speed). These studies were performed in both Comfort and Efficiency Modes and with a pre-determined schedule of events which varied in duration and time-of-day, in an effort to capture a diverse range of weather conditions and possible DR scenarios.

Figure ES-5 shows an example of a DR: Min event on a hot summer day, which was from 3:00PM to 8:00 PM. HP2, which had been at higher power, reduced to a steady minimum capacity, but remained on. HP1, which had been cycling, continued to cycle between minimum and off, though the frequency of cycling increased. The lower portion of the figure shows that the indoor dry-bulb and wet-bulb temperature remained largely unchanged during the event.

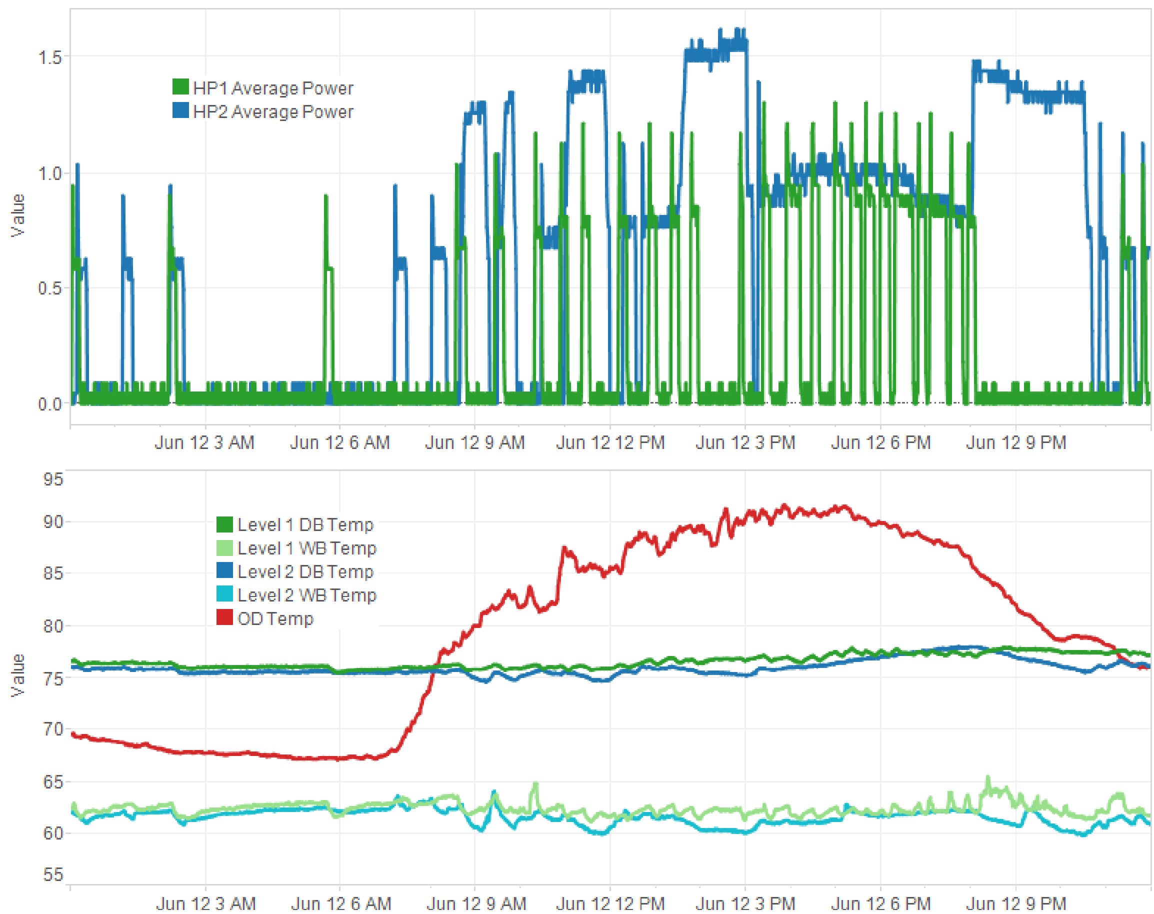

Figure ES-5
Heat Pump Power and Temperatures during a DR-Min Test Day

Figure ES-6 shows the systems during a “DR: Off” event where the outdoor temperature profile was similar to that in Figure ES-5. This event was from 5:00 PM to 8:00 PM (2 hours shorter than the previous example). Before the event HP2 was running at a higher capacity, and HP1 was cycling. Both systems turned off, and after the event both resumed operating, this time with HP1 operating at a greater-than-minimum capacity as well. The increased power after the event is because, as shown by the temperatures in the lower portion of the figure, the indoor dry-bulb and wet-bulb temperatures increased by over 5°F during the three-hour event.

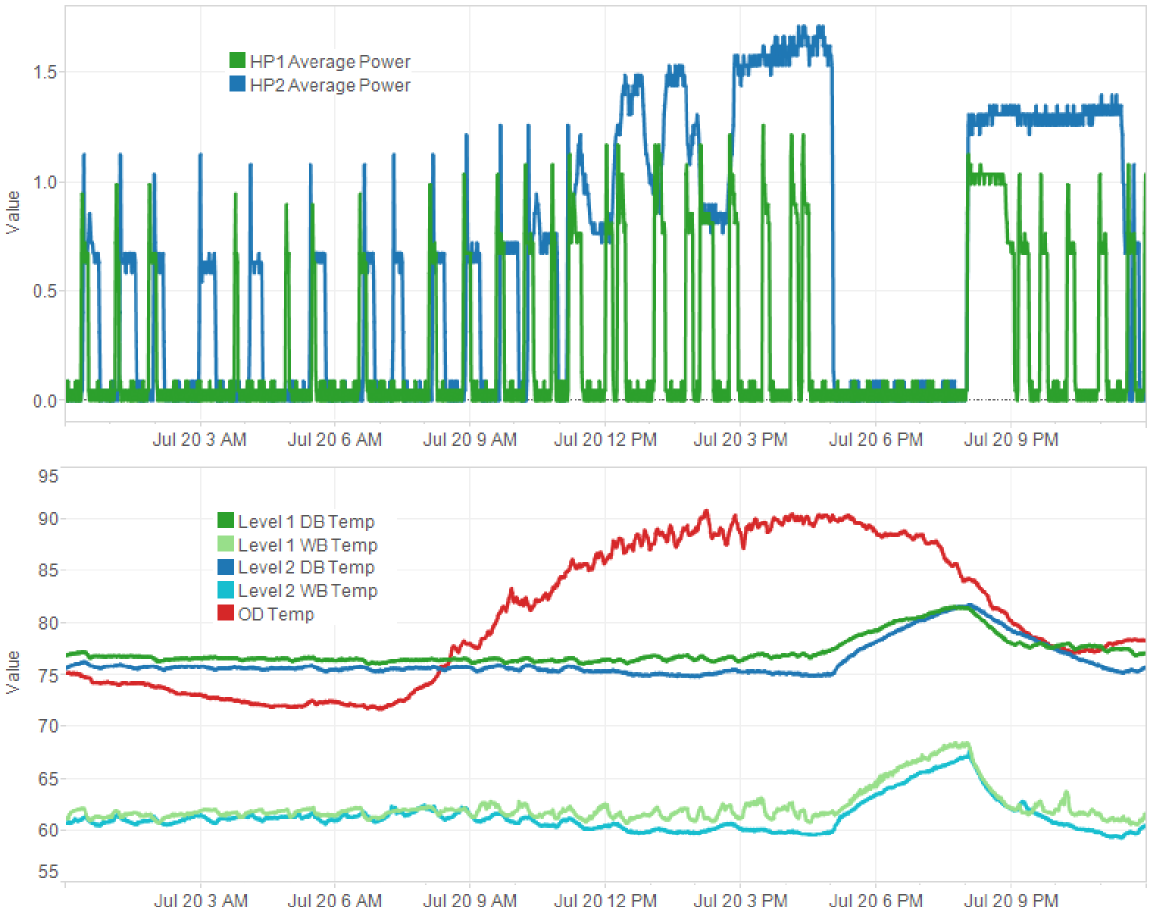

Figure ES-6
Heat Pump Power and Temperatures during a DR: Off Test Day

The above examples highlight the qualitative differences in performance between DR: Min and DR: Off events. The DR: Min event allows for some cooling which maintains zone temperatures or slows their increase, while in a DR: Off event temperatures increase freely such that occupants may become uncomfortable.

While the above may seem to indicate that DR: Min events are favorable, the zonal interactions of the heat pumps may reduce or eliminate the savings benefit of a DR: Min event. In the configuration tested, the cooling outside of DR events was provided predominantly by the upstairs heat pump, while the downstairs heat pump ran at a low duty cycle. When the DR control caused HP2 to reduce operation, HP1 increased its' duty cycle, still satisfying the condition of "minimum output." The result is shown in Figure ES-7 the power for a given outdoor temperature bin during DR: Min events was only very slightly lower than hours where there was no DR event; on the other hand, DR: Off events used only standby power during DR events, resulting in demand reductions increasing proportional to outdoor temperature. This finding suggests that for multi-system sites, a simple "DR: Min" control approach may not suffice. Alternative possibilities, such as combining one "DR: Min" system with a "DR: Off" system, may help provide demand savings while allowing some cooling. Such strategies will be tested in the continuation of this study.

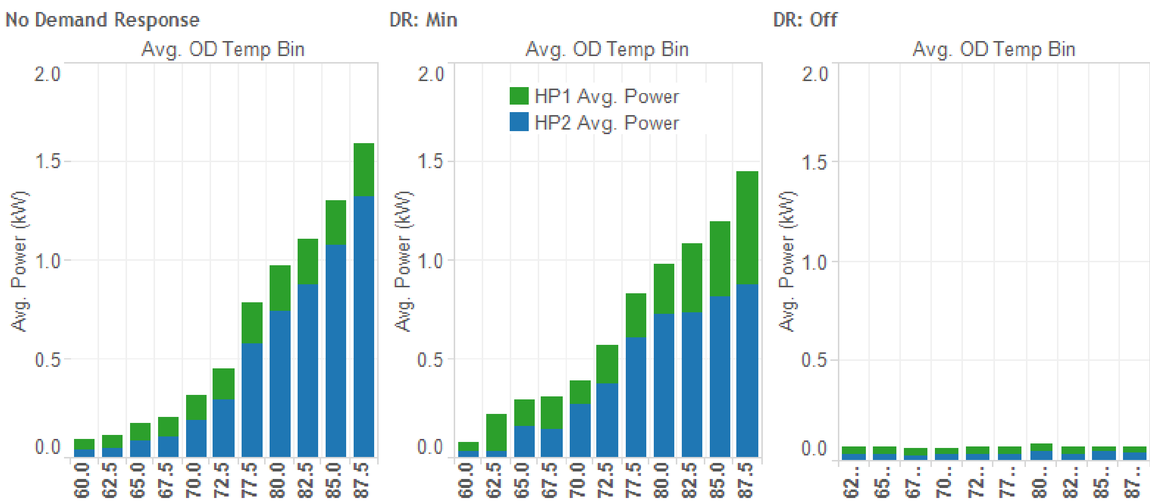

Figure ES-7
Heat Pump Power vs. Temperature for Periods of No DR Event, DR-Min Events, and DR-Off Events

Conclusions

Using the Campbell Creek Research Homes as a test bed, variable-speed heat pump systems are being tested for energy efficiency and demand response performance in Knoxville, TN. The systems have been tested since late 2012 in each of their operating modes and in cooling-mode (summer) Demand Response operation. Using single speed systems that were in the same house from 2010-2012, some assessment of the energy savings potential of the high-efficiency VSHPs can be made.

Comparing data for like outdoor temperatures, the VSHP was shown to use 17-38% less power during heating and 34-38% less power during cooling conditions. Though the weather was more extreme during the nearly two years of SSHP operation, comparing periods with similar heating and cooling loads shows a significant reduction in energy consumption with the high-efficiency VSHP systems.

The VSHP systems in this study can operate in Efficiency or Comfort Modes. While the full differences are not disclosed by the manufacturer, some differences are known. Comfort Mode features lower airflow rates and higher (in heating) or lower (in cooling) supply air temperatures; in cooling mode, Comfort Mode provides more latent cooling (dehumidification). A slight reduction in average power and improvement in efficiency was observed in both heating and cooling mode using Efficiency Mode. However, in cooling operation this savings was accompanied by an increase in the indoor humidity, which may be uncomfortable to some occupants and cause them to lower the thermostat set-point.

Since the nominal sizing of the systems changed from the SSHP to the VSHP, and the VSHP system adds both variable capacity components as well as improved components such as evaporator and condenser coils, a definitive statement on how much savings is attributable to variable capacity should not be made.

For Demand Response operation the VSHP was tested in Efficiency and Comfort Mode and with the heat pump's response set to either turn off, or limit capacity to the minimum operating speed. For testing, the systems were both set to the same configuration and run according to a pre-arranged schedule of events. In DR: Off events, the VSHPs both turned off but the temperature inside the house drifted according to the building load; in DR: Min events the upstairs VSHP, which carried more of the cooling load, reduced its output, while the downstairs system, already operating at a low duty cycle, tended to increase duty cycle while still satisfying the minimum capacity control requirement. This resulted in negligible demand reduction during DR: Min events. Continuing research will investigate using a combination of DR Min and DR Off commands for separate units in multi-unit systems to achieve greater demand savings and occupant comfort.

This research effort will be ongoing through September, 2014. Among the anticipated results of continuing research are a more extreme winter and summer periods for the VSHP, and testing of demand response control strategies in cooling mode which seek to make effective use of the "DR: Min" capability. In addition, data from the other two Campbell Creek Research homes (CC2 and CC3) will be incorporated which includes a single VSHP system that was also tested for DR events and another high-efficiency VSHP system made by another manufacturer. With the addition of a test house which utilizes a single, VSHP system with zone controls to condition the upstairs and downstairs spaces, future research will be able to compare the effectiveness and efficiency of the traditional two unit-two zone system and a single unit-dual zone approach.

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1 INTRODUCTION

This report details an ongoing study of high-efficiency variable capacity heat pump systems in the Campbell Creek Research Homes in Knoxville, Tennessee. The homes, referred to herein as the “CC Houses” or similar, are three simulated-occupancy houses owned and operated by the Tennessee Valley Authority in collaboration with the Electric Power Research Institute (EPRI) and Oak Ridge National Laboratory (ORNL). Additional collaborators for this project were the Bonneville Power Administration and Southern Company.

For this study, the CC House #1 had two Carrier Greenspeed Variable Speed Heat Pumps (VSHPs) installed and instrumented to simulate a new-construction household with high-efficiency air conditioning. In addition, prior years’ data in which a standard-efficiency single-speed heat pump (SSHP) was analyzed and used to compare results and estimate energy and demand savings. The objectives of the study include quantifying energy and demand savings from the VSHP, as well as simulating demand response (DR) performance by simulating DR events over the course of the summer.

Background

Space conditioning is one of the largest utility loads and a primary driver of both summer and winter peaks in most climates. In both heating and cooling operation, as outdoor temperatures grow more extreme, the need for conditioning increases and as such, loads fluctuate with outdoor temperatures. As a natural trait of the vapor compression cycle which drives air conditioning and heat pumps, as the temperature difference between indoor and outdoor increases, the efficiency of the cycle decreases, further increasing the load. In winter, many electric heating applications rely on electric resistance as a backup, which is a high-power, low-efficiency form of heating. Since electric resistance heat is inefficient and expensive to operate, natural gas is often considered a more attractive option. Further, by virtue of its high-power and intermittent operation, electric resistance heat contributes to high winter peaks and low utility load factors. It is therefore of considerable interest to investigate electric heat pump systems which both improve efficiency particularly in extreme conditions, and are capable of providing some active form of demand response, to preserve efficient and reliable grid operation.

A conventional single-speed heat pump system in cooling mode operates by turning on when the indoor temperature reaches a certain set point, operating until it reaches a second, lower set point, and turning off. Inherent to this process is some cycling loss as the energy that was built up in pressurizing refrigerant dissipates. Variable-capacity air-source heat pumps provide several potential advantages over single-speed systems for heating and cooling. Variable-speed systems are capable of modulating the speed of the compressor and fans to adjust capacity (and with it, power) in an effort to match the load of the building. In ideal operation, when cooling is needed the heat pump would operate always at the exact correct speed such that indoor temperature remains perfectly constant at the set point. Only when the load on the building is lower than the minimum capacity the heat pump can provide does the system cycle. This reduces cycling losses overall.

A more subtle benefit of variable capacity heat pumps is that, when operating at less-than-maximum capacity, the system efficiency tends to increase. This is because, with less refrigerant being driven through the system over time, the system components (the evaporator and condenser in particular) become effectively "over-sized" for the operating condition. This allows more heat transfer per unit of refrigerant in the cycle, improving efficiency at part-load operation.

Still another benefit of variable capacity systems exists in the capability to "over-speed" the compressor (run the compressor at higher-than-nominal capacity) during extreme loads, particularly in heating mode. The system may ramp up compressor power during the coldest conditions, allowing higher capacity and higher supply temperatures, reducing or eliminating the discomfort that comes from low supply air temperatures. While the overall efficiency of the heat pump is reduced by this, the efficiency can still be considerably better than electric resistance heating, and with a smoother power profile.

A more detailed description of how heat pump systems work may be found in Appendix A.

The objective of this report is to quantify the performance of high-efficiency VSHP systems in heating and cooling mode, as compared with standard-efficiency devices. Further, the report will examine cooling-mode demand response capability of the VSHP systems, which may modulate power down or cycle off in response to DR signals from a utility. Through simulated-occupancy field installations, the data analyzed in this report will help to bridge the gap between laboratory data and real-world field performance.

At some points in this report the concept of degree days will be used to compare data across different time periods. Degree days are a measure that is useful in comparing weather for heating and cooling seasons. Degree days can be calculated for heating or cooling, and are a measure of the deviation of temperatures from the building's balance point over time. Degree days are calculated using outdoor temperature and a selected balance point; commonly, 65°F is used for non-specific comparison purposes. One heating degree day, for example, means that the outdoor temperature was below the designated temperature (65°F) for one day, by one degree. Degree days are more accurately calculated with more precise data, for example using one hour interval temperature readings. The degree days would be calculated hourly (temperature difference times one twenty-fourth of a day) and added up, to give the total for the day. Degree days do not include negatives: for instance, if the outdoor temperature is above the baseline temperature, the HDD is not negative, it is zero. Degree days are not a perfect metric for estimating energy – the same total may be reached by many different temperature profiles, and loads do not necessarily scale linearly with outdoor temperature. However, they do provide a valuable high-level comparison of seasonal variations.

Another term which will be used is wet-bulb and dry-bulb temperature. Dry bulb temperature is the temperature that would be recorded by a common thermometer, sitting in relatively still air in a room. Wet bulb temperature is the temperature that would be recorded by a damp thermometer with a breeze moving over it. Wet-bulb temperature captures the evaporative cooling effect, and is in essence a measure of temperature and humidity combined. The wet-bulb temperature is equal to the dry-bulb temperature when relative humidity is 100%; at other conditions the dryer the air is, the lower the wet-bulb temperature will be. For a common reference, when wet-bulb temperatures are much lower than dry bulb temperatures, sweat or water evaporates quickly off of peoples' skin and the skin will feel cooler than when the wet-bulb and humidity are higher.

While relative humidity is more commonly used in everyday speech, wet-bulb temperature is typically more informative for HVAC applications.

2
TESTING OVERVIEW

Overview of Research Homes

The CC Homes are located in the Campbell Creek subdivision in the western end of Knoxville, Tennessee. A total of three homes are under study. CC House #1 was originally built as a standard "builder home" with appointments and equipment typical of houses in that neighborhood. CC House #2 was designed to be a high-efficiency retrofit home, with many of the efficient technologies that can be added as retrofits. CC House #3 was built as a near-zero energy home, with additional features not necessarily feasible as retrofit add-ons. This study took place in CC House #1, where single-speed, standard-efficiency air source heat pumps initially installed in the home were replaced by high-efficiency VSHPs. An additional modification to CC House #1 since its construction is the addition of an air-source heat pump water heater located in the garage, which may have a small impact on the space conditioning of the home. Details on the houses may be found in the ORNL annual report on the project [1].

Figure 2-1
Front of Campbell Creek House #1

CC House #1 has a conditioned area of 2,468 square feet. On April 30, 2013 a Home Energy Rating Standard assessment of the home was performed, and the house scored a 77 HERS Index, a nominal 5-Star rating. This rating included the high-efficiency air conditioner and water heater discussed. For reference, CC House #2 had a 54 HERS Index (5-Star Plus) and CC House #3 had a 35 Index (5-Star Plus). The HERS Index is a standard home efficiency metric, in which a score of 100 indicates a new, standard home; 0 indicates a zero-net-energy home, and existing homes are typically around 130 on average. A home with a 70 HERS is 30% more efficient than a standard new home, and a home with a 130 HERS would be 30% less efficient than a standard new home. The HERS assessment results for CC House #1 are included in Appendix B.

CC House #1 has two floors of nearly the same square footage. The lower floor has a kitchen, dining area and living room with a fairly open floor plan, with one half bathroom. The heat pump servicing the first floor will be referred to as Heat Pump 1 or HP1. The indoor unit serving the lower level is installed in the garage and ductwork runs between the floors to supply the zone. The thermostat for the lower level is located on the wall in the dining room area. The upstairs zone has 3 bedrooms and a laundry room, as well as two full bathrooms. The indoor unit serving this zone is located in the attic, along with the ductwork supplying the zone. The thermostat is located in the central hallway. For the purposes of the test the doors to all rooms were left open. The heat pump servicing the second floor will be referred to as Heat Pump 2 or HP2.

For baseline data, the prior years' data for CC House #1 is used. EPRI used data from July, 2010, through October, 2012 during which time two standard-efficiency (13 SEER, 7.7 HSPF) single-speed air source heat pump systems were installed in CC House #1. The first floor had a 1.5-ton system, and the second floor had a 2.5-ton system. The thermostats were calibrated and set to 71°F. In late October and early November, 2012, two Carrier Greenspeed variable capacity heat pumps were installed in the home, with a 2-ton on the ground floor and 3-ton on the second floor. The GreenSpeed line has a nominal 20.5 SEER and 13 HSPF rating. The thermostats were tuned to match the previous systems' calibration.

It is important to note that a number of changes occurred with the HVAC retrofit. First, the system capacity changed, with larger-capacity heat pumps installed. All other things being equal, this would still change performance as the house's balance point (the temperature at which full heat pump operation would exactly balance outdoor and internal loads) will be different for different capacity systems. Also, the systems changed from standard-efficiency, single-speed devices to high-efficiency, variable speed systems. The replacement systems were not simply the same system, with variable capacity added, but rather had entirely different components, including larger condenser and evaporator coils. Because of all of these factors, the degree to which energy and demand savings may be attributed to any particular change cannot be directly assessed from this data. For simplicity the systems will be referred to as the single-speed heat pumps (SSHP) and variable speed heat pumps (VSHP), though this is not mean to imply that all differences are attributable to single speed or variable speed systems. However, the data will provide a detailed assessment of the possibilities from a high-efficiency heat pump retrofit.

The VSHP systems operate with two modes: Efficiency Mode and Comfort Mode. While the full difference between the modes is not disclosed by the manufacturer, some generalities are known: Efficiency Mode provides a higher airflow rate of warmer (cooling mode) or cooler (heating) supply air. The proposed benefit is better efficiency by lessening the pressure difference that the compressor must maintain, at the expense of a smaller increase in fan power. In cooling mode, one implication of this is reduced latent cooling (humidity removal).

The outdoor units are shown in Figure 2-2. The units have a larger form factor than is typical for residential heat pumps.

Figure 2-2
VSHP Outdoor Units

The indoor controller is shown in Figure 2-3. The controller can connect to the home's wireless internet to gather information such as the weather, and to provide diagnostic information to the manufacturer.

Figure 2-3
VSHP Controller and One Indoor Temperature Sensor

The heat pump indoor units are shown in Figure 2-4 and Figure 2-5. Figure 2-4 shows the indoor unit for HP1, which serves the downstairs zone. HP1's indoor unit is in the garage. Some of the wiring for instrumentation can be seen in this figure.

Figure 2-4
HP1 Indoor Unit (Downstairs)

Figure 2-5
HP2 Indoor Unit (Upstairs)

Demand Response Testing

For demand response testing, a 24-Volt signal was supplied to the outdoor unit and controlled via the building's dedicated LabView controller. In order to enable a DR "event," the 24V signal was interrupted. This interruption, in combination with enabling DR on the devices' thermostat controllers, signals the heat pumps to enter utility curtailment mode. The heat pumps have two modes of demand response: "Min Cooling" mode, in which the device will not exceed its' minimum operating capacity, and "Off" mode, in which the system turns off. In order to understand both operating modes, all the while capturing valuable baseline (no DR event) data, a schedule was developed. The schedule mixed the different operating modes, and DR events of varying lengths and starting times. After some slight modifications over the course of the cooling season, the schedule shown in Table 2-1 was used. The project team agreed upon an equal balance of Comfort Mode and Efficiency Mode. The group also decided to use more "DR: Min" events than "DR: Off" events, since it was felt that the "DR: Off" response would be more predictable with limited data, while DR: Min is more complex. The schedule included baseline days with no events throughout, to attempt to capture all variation in weather with baseline operation.

Table 2-1
Schedule of Demand Response Tests

Week of…	Mode	DR Type	Monday	Tuesday	Wednesday	Thursday	Friday	Saturday	Sunday
5/6/2013	Comfort	None	-	-	-	-	-	-	-
5/13/2013	Efficiency	Min	-	-	-	-	5-7PM	1-9PM	-
5/20/2013	Comfort	Off	-	-	5-6PM	4-5PM	5-7PM	5-8PM	-
5/27/2013	Comfort	Min	-	-	3-8PM	5-8PM	5-7PM	1-9PM	-
6/3/2013	Efficiency	Min	-	-	3-8PM	5-8PM	5-7PM	1-9PM	-
6/10/2013	Comfort	Min	-	-	3-8PM	5-8PM	5-7PM	1-9PM	-
6/17/2013	Efficiency	Off	-	-	5-6PM	4-5PM	5-7PM	5-8PM	-
6/24/2013	Efficiency	None	-	-	-	-	-	-	-
7/1/2013	Comfort	Min	-	-	3-8PM	5-8PM	5-7PM	1-9PM	-
7/8/2013	Efficiency	Min	-	-	3-8PM	5-8PM	5-7PM	1-9PM	-
7/15/2013	Comfort	Off	-	-	5-6PM	4-5PM	5-7PM	5-8PM	-
7/22/2013	Comfort	None	-	-	-	-	-	-	-
7/29/2013	Efficiency	Min	-	-	5-6PM	4-5PM	5-7PM	5-8PM	-
8/5/2013	Comfort	Min	-	-	5-6PM	4-5PM	5-7PM	5-8PM	-
8/12/2013	Efficiency	Off	-	-	5-6PM	4-5PM	5-7PM	5-8PM	-
8/19/2013	Efficiency	None	-	-	-	-	-	-	-
8/26/2013	Comfort	Min	-	-	3-8PM	5-8PM	5-7PM	1-9PM	-
9/2/2013	Efficiency	Min	-	-	3-8PM	5-8PM	5-7PM	1-9PM	-
9/9/2013	Comfort	Off	-	-	5-6PM	4-5PM	5-7PM	5-8PM	-
9/16/2013	Comfort	None	-	-	-	-	-	-	-
9/23/2013	Efficiency	Min	-	-	3-8PM	5-8PM	5-7PM	1-9PM	-
9/30/2013	Comfort	Min	-	-	3-8PM	5-8PM	5-7PM	1-9PM	-
10/7/2013	Efficiency	Off	-	-	5-6PM	4-5PM	5-7PM	5-8PM	-

3
RESULTS: EFFICIENCY COMPARISON

Results

The monthly energy consumption for the baseline outdoor units, corresponding indoor units (excluding backup heat), and associated backup heat are shown for July, 2010 through October, 2012, along with a box-and-whisker plot of outdoor temperature, in Figure 3-1. The box-and-whisker plot shows the average, upper and lower quartile, and full range of temperatures. Note that here and throughout the report, Heat Pump 1 or HP1 will refer to the downstairs level heat pump and Heat Pump 2 or HP2 refers to the upstairs.

From Figure 3-1 some general trends can be observed. For both heat pumps, the seasonal variation of space conditioning equipment is clear, with low energy consumption during the shoulder months such as April and October, and relatively high energy consumption in the summer and winter. In the winter and in particular the winter of 2010-11, the energy consumption is particularly high and is driven in large part by backup heat. The winter of 2010-11 was particularly cold in Knoxville; for December through February, there were 2,398 65°F-basis heating degree days (HDD) in 2010-11, compared with 1,816 HDD for the same period in 2011-12, indicating a considerably larger heating load during the former [2]. Also, the downstairs heat pump can be seen to use more energy than the upstairs heat pump during the heating season while the upstairs unit used more during the cooling season. This is not uncommon, as heat tends to rise in homes.

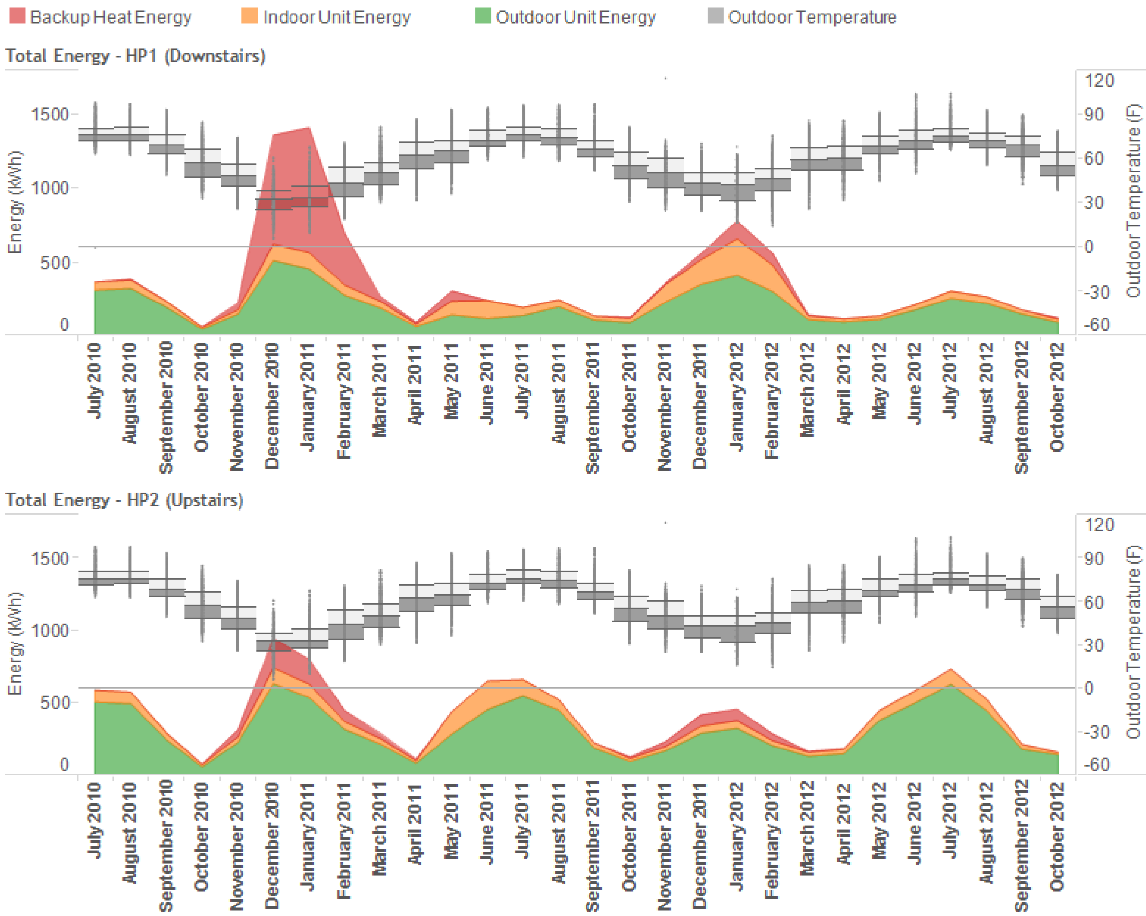


Figure 3-1
Monthly Energy Consumption and Outdoor Temperature, Baseline Heat Pumps

Figure 3-2 shows the same visualization of energy and outdoor temperature, for November, 2012 through October, 2013 for the variable speed heat pump systems. The energy consumption for each season can be seen to be generally lower, and the usage of electric backup heat in the winter was considerably lower. The number of heating degree days for December, 2012 through February, 2013 was 1,913, slightly higher than the winter of 2011-12 but lower than 2010-11. A significantly colder winter in 2013-14 will help provide data for the VSHPs with such weather; a brief Addendum at the end of this report shows preliminary results.

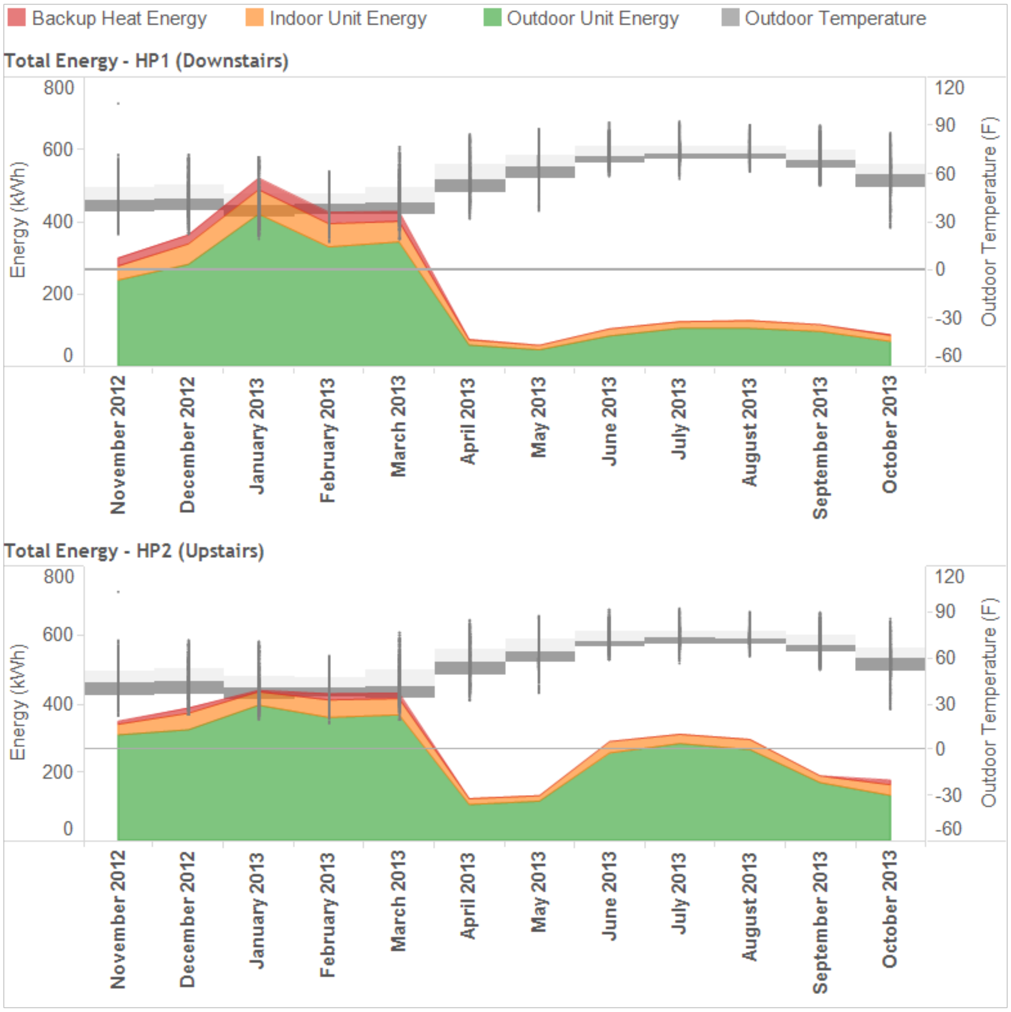


Figure 3-2
Monthly Energy Consumption and Outdoor Temperature, Variable Speed Heat Pumps

The total energy of all heat pumps, and the heating and cooling degree days for the entire test period are shown in Table 3-1. From this, general comparisons of energy consumption for periods of like weather can be made. The baseline system is shown with a blue background, the VSHP with green. As an example of this comparison, March of 2011, 2012 and 2013 all had very different weather and heating needs; the energy consumption is correspondingly different and cannot be fairly compared. A more apt comparison might be, for example, March 2012 (baseline system) to October, 2013 (VSHP), where the HDD and CDD are much more similar.

For the full year that the VSHP was monitored, November 2012 through October 2013, the energy consumption was 6,248 kWh, with 1,359 CDD and 3,635 HDD. With the baseline system, for November 2011 through October 2012, the consumption was 8,004 kWh, with 1,617 CDD and 2930 HDD. For the baseline period of November 2010 through October 2011, the consumption was 10,693 with 1,667 CDD and 3,832 HDD. So the VSHP used 22% less energy than the 2011-2012 baseline case, where the baseline saw a larger cooling load and smaller heating load. The VSHP used 42% less energy than the 2010-2011 case, but the CDD and HDD were both higher for the baseline case.

Table 3-1
Energy Consumption, CDD and HDD for the Entire Test Period; SSHP Periods in Blue, VSHP in Green

	2010			2011			2012			2013		
Month	**Energy (kWh)**	**CDD**	**HDD**	**Energy (kWh)**	**CDD**	**HDD**	**Energy (kWh)**	**CDD**	**HDD**	**Energy (kWh)**	**CDD**	**HDD**
January	-	-	-	2202	0	916	1225	0	691	956	1	719
February	-	-	-	1131	0	580	834	0	565	852	0	649
March	-	-	-	545	7	404	296	34	178	856	0	654
April	-	-	-	193	61	147	290	40	158	196	29	225
May	-	-	-	733	167	89	574	211	11	188	104	70
June	-	-	-	880	362	0	786	323	6	391	300	0
July	942	476	0	842	477	0	1024	491	0	432	333	0
August	947	473	0	754	439	0	778	336	0	419	348	0
September	516	221	5	346	142	16	379	173	21	302	197	4
October	129	7	167	242	12	223	269	9	244	263	47	178
November	527	0	450	582	0	424	646	0	548	-	-	-
December	2298	0	1007	968	0	632	747	0	588	-	-	-

In addition to the table above, the energy consumption in kWh per degree-day (both heating and cooling) is shown in Figure 3-3. The energy consumption was significantly reduced relative to the weather after the VSHP was installed. The degree-day comparison is inherently incomplete, and the efficiency improvements will be examined in much greater detail in the following sections, by examining the performance of each system across the range of operating conditions. However, this graphic provides a fast visual affirmation that there was a reduction in energy relative to weather variations.

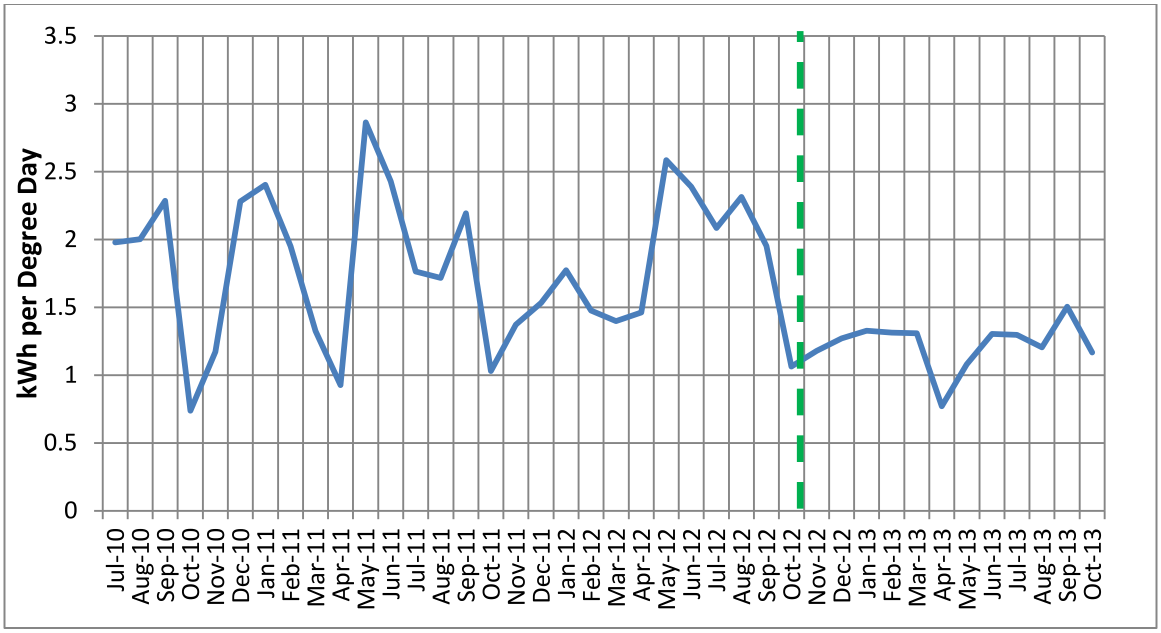


Figure 3-3
Average Heating and Cooling Energy per Degree-Day for Entire Test Period (Retrofit After Dotted Line)

The above data shows the energy consumption as it varied seasonally from a high-level perspective. To further understand the differences in operations of the heat pump systems examined, the following graphs examine the differences in operation of each system under different operating conditions. Figure 3-4 shows the average power for hourly average outdoor temperature bins in which there was sufficient data. For example, the average total power during all hours during which the average outdoor temperature was between 5°F and 10°F is found in the first column, and totals approximately 5,800 Watts for HP1. The data for the totals of HP1 and HP2 power, as well as the sum of both heat pumps, can also be found in Table 3-2, for temperature bins in which both systems operated. Observation of the data shows a considerable reduction in average power at the extremes, particularly during cold weather. The new VSHP used more heat pump power (outdoor unit power), but far less backup resistance heating when examining the lower floor heat pump which provided most of the space heating. For cooling, a similar trend is seen, with considerably lower average power during higher temperature bins. The reader may notice that there was more data in extreme temperature bins in both heating and cooling modes; the winter of 2013-2014 has produced more cold weather and results in a brief addendum at the end of this report will summarize the findings; the results will be evaluated in more detail in the future reports on this research effort.

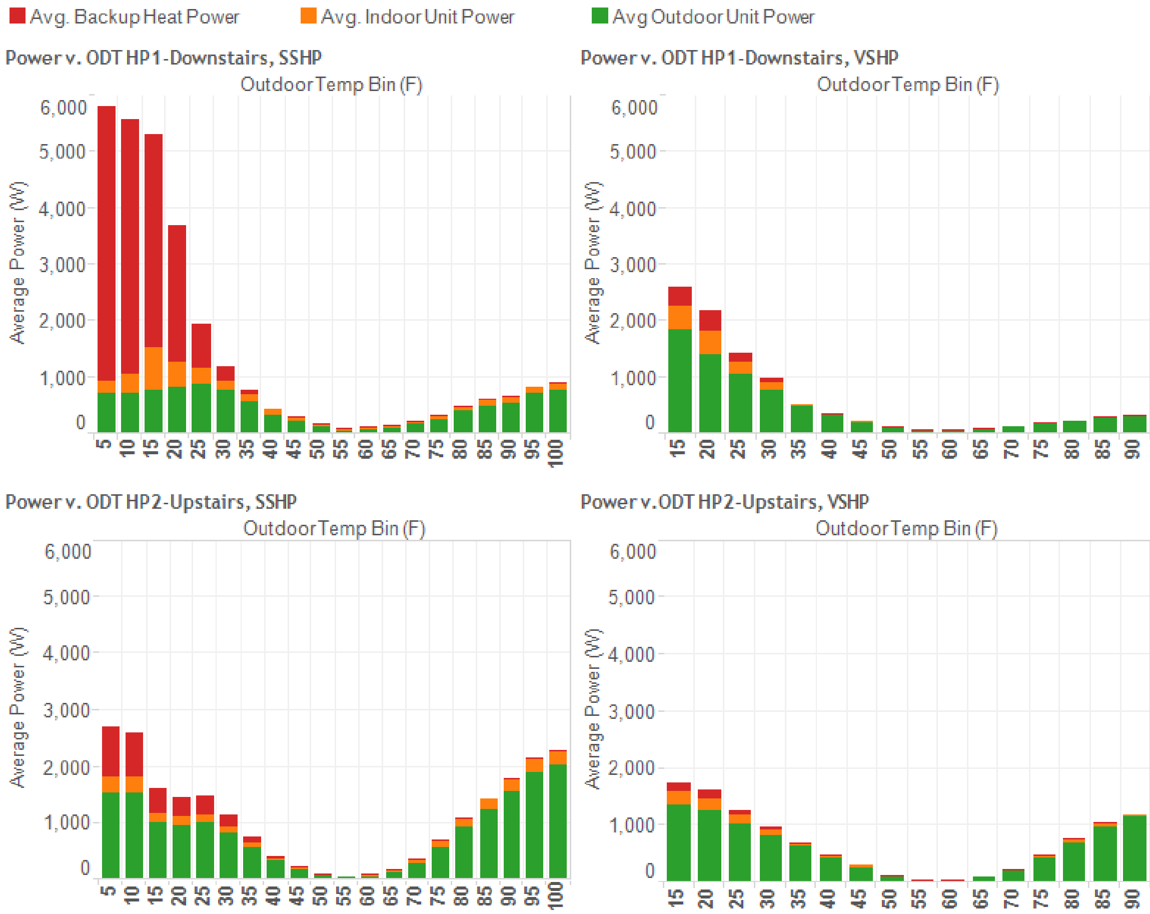


Figure 3-4
Power of SSHP and VSHP Systems vs. Temperature Bin

Examining Table 3-2, HP1-New used 52% less power during the 15°F temperature bin hours than HP1-Old; HP2-New used 9% more than HP2-Old. The total power difference of the two systems combined was 38%, a reduction of 2.6 kW on average. Similarly, in the highest temperature bin when temperatures were between 90°F and 95°F, the reduction of total power was again 38%, though the magnitude of consumption (and therefore savings) was lower: a total reduction of 0.9 kW. The largest percentage reduction was in the 60°F temperature bin; however the usage was very low during this period, so the savings are of less interest.

Table 3-2
Average Power of SSHP and VSHP Systems, and Reduction from SSHP to VSHP for Range of Temperature Bins

	Single-speed HPs			Variable Speed HP			Reduction (%)		
Outdoor Temp Bin (F)	HP1 (1.5-ton) Avg. Power (kW)	HP2 (2.5-ton) Avg. Power (kW)	HP1 + HP2 Avg. Power (kW)	HP1 (2-ton) Avg. Power (kW)	HP2 (3-ton) Avg. Power (kW)	HP1 + HP2 Avg. Power (kW)	HP1 Avg. Power (W)	HP2 Avg. Power (W)	HP1 + HP2 Avg. Power (W)
15	5.3	1.6	6.9	2.6	1.7	4.3	52%	-9%	38%
20	3.7	1.5	5.2	2.2	1.6	3.8	41%	-10%	27%
25	2.0	1.5	3.4	1.4	1.3	2.7	27%	14%	22%
30	1.2	1.1	2.3	1.0	1.0	1.9	17%	17%	17%
35	0.8	0.8	1.5	0.5	0.7	1.2	33%	11%	22%
40	0.4	0.4	0.9	0.4	0.5	0.8	19%	-13%	3%
45	0.3	0.2	0.5	0.2	0.3	0.5	25%	-33%	-1%
50	0.2	0.1	0.3	0.1	0.1	0.2	34%	-24%	11%
55	0.1	0.1	0.2	0.1	0.0	0.1	39%	20%	32%
60	0.1	0.1	0.2	0.1	0.0	0.1	49%	59%	54%
65	0.1	0.2	0.3	0.1	0.1	0.2	39%	49%	45%
70	0.2	0.4	0.6	0.1	0.2	0.3	40%	41%	41%
75	0.3	0.7	1.0	0.2	0.5	0.7	39%	31%	34%
80	0.5	1.1	1.6	0.2	0.7	1.0	51%	31%	38%
85	0.6	1.4	2.0	0.3	1.0	1.3	51%	28%	35%
90	0.7	1.8	2.4	0.3	1.2	1.5	51%	34%	38%

VSHP Performance Analysis - Cooling

The following section examines the difference between comfort and efficiency mode for the VSHP system, beginning with cooling mode operation. Figure 3-5 shows the average cooling capacity, Sensible Heat Ratio (SHR, the ratio of sensible cooling to total cooling), power and supply air temperature for HP1 in both Comfort Mode and Efficiency Mode. The values shown are for when the heat pump is active. The supply air temperature and sensible heat ratio were higher for Efficiency Mode. The general trend for HP1 was for capacity to decrease with increasing outdoor temperature. Since HP1 generally cycles at a relatively low duty cycle, between minimum capacity and off, this is expected.

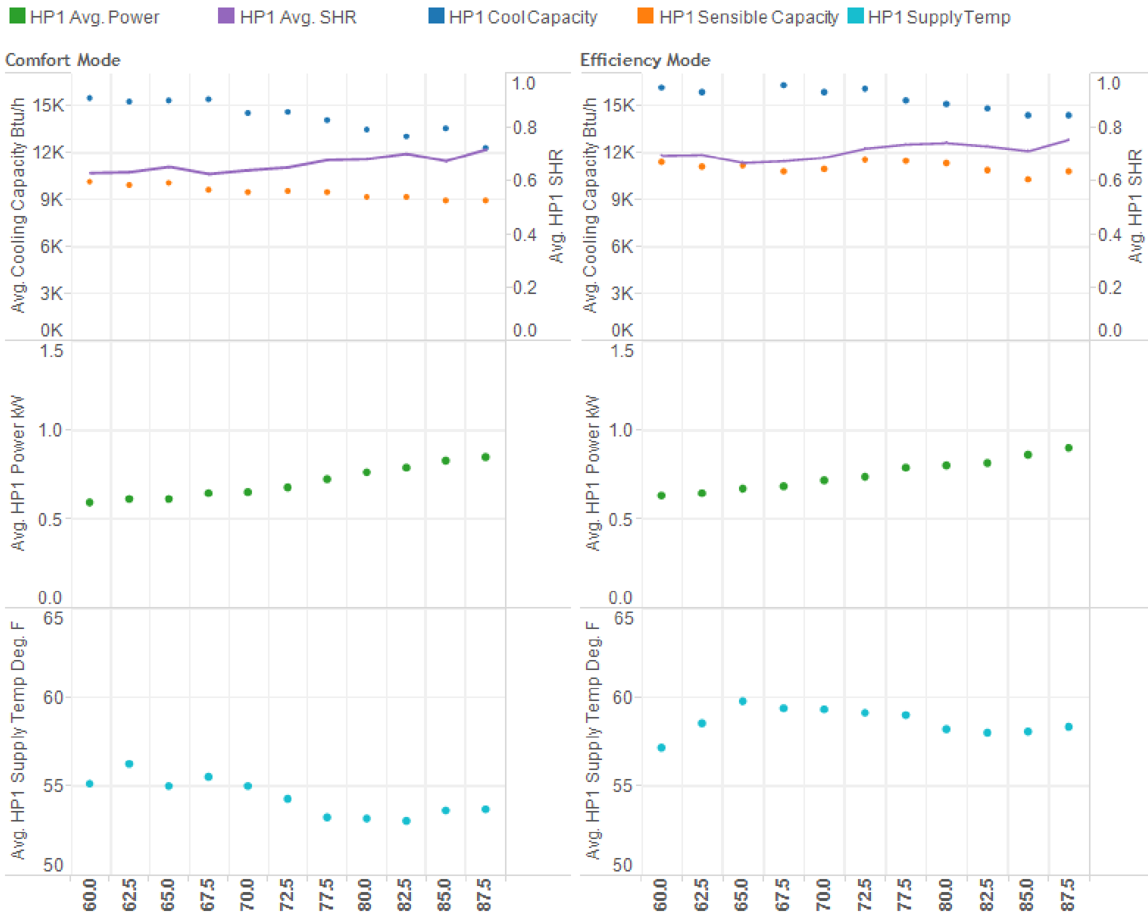


Figure 3-5
HP1 Average Capacity, SHR, Power and Supply Air Temperature (When Running) vs. Outdoor Temperature Bin

Figure 3-6 shows the average cooling capacity, SHR, power and supply air temperature for HP2 for both modes, again when the heat pump is active. The supply air temperature and sensible heat ratio were both higher in Efficiency Mode. The capacity of HP2 decreased with outdoor temperature up to approximately 70°F, indicating minimum-capacity cycling. Above 70°F, the capacity trended upwards, indicating that in these conditions the heat pump was modulating to increased capacity.

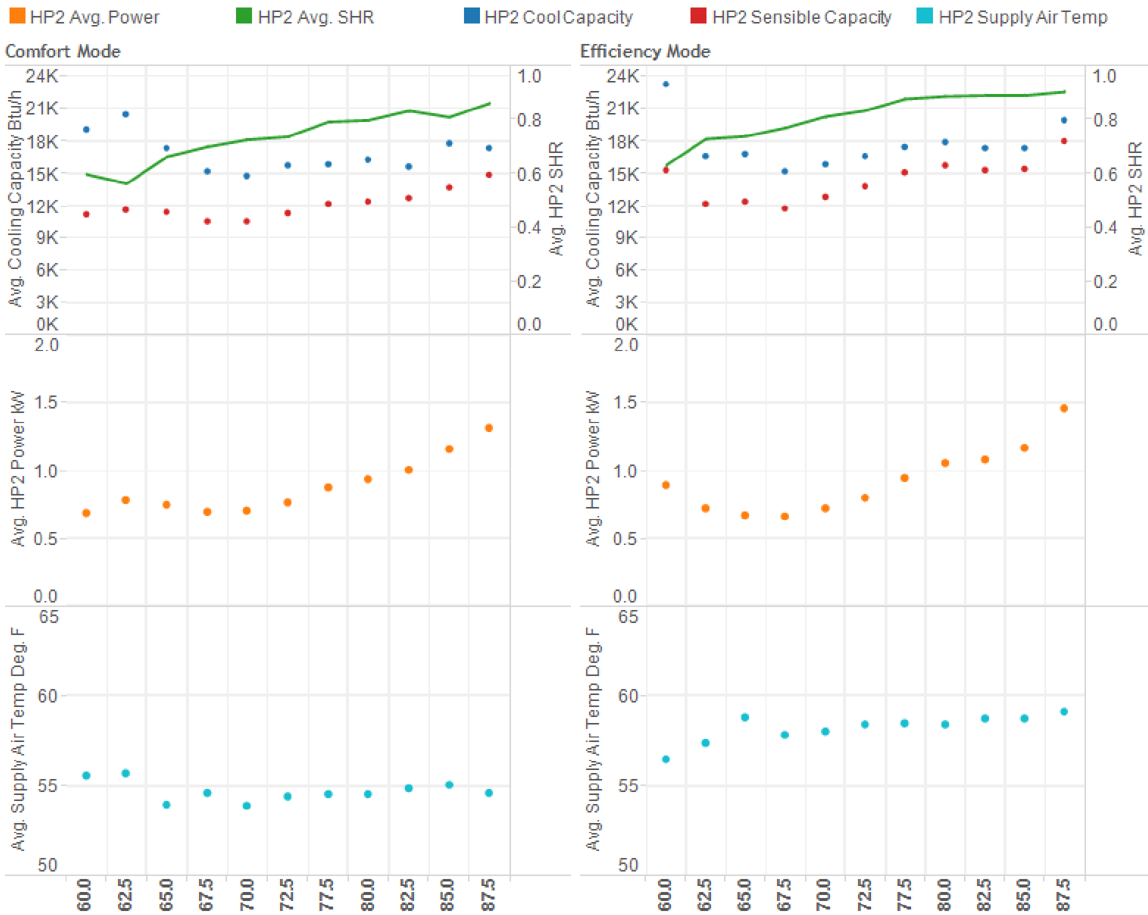


Figure 3-6
HP2 Average Capacity, SHR, Power and Supply Air Temperature (When Running) vs. Outdoor Temperature Bin

The Energy Efficiency Ratio (EER) is the ratio of capacity, in Btu/h, to power, in Watts. It is the standard metric for air conditioning single-point efficiency. The EER for both systems, for both modes is shown in Table 3-3. The EER for both systems was greater than 20 in low-load conditions, and decreased to 13-16 in higher-load conditions, which agrees with laboratory test results for similar conditions.

Table 3-3
EER of Heat Pumps in Cooling Mode Based on Average Capacity and Power, for Each Outdoor Temperature Bin

OD. Temp. Bin	HP1, Comfort	HP1, Efficiency	HP2, Comfort	HP2, Efficiency
60	26.2	25.6	28.0	26.2
62.5	24.9	24.3	26.2	23.0
65	25.0	26.6	23.4	25.1
67.5	24.0	23.8	22.0	23.2
70	22.2	22.2	21.0	21.9
72.5	21.7	21.9	20.6	20.9
77.5	19.5	19.3	18.1	18.5
80	17.6	18.8	17.4	16.9
82.5	16.6	18.2	15.5	16.1
85	16.2	16.6	15.3	14.9
87.5	14.6	15.9	13.2	13.7

Examining air flow rate for each mode shows a significantly higher airflow in Efficiency Mode, as expected. This can be seen in Figure 3-7, which shows the average airflow rate when the heat pump is running. It should be noted that the underlying data for this graph is in one-minute intervals, and in some instances the air flow was increasing or decreasing as the system started or stopped operation, causing skewed readings, which would be particularly prevalent during starting and stopping. These values should be considered rough values and only used to show the generally higher air flow rate in Efficiency Mode.

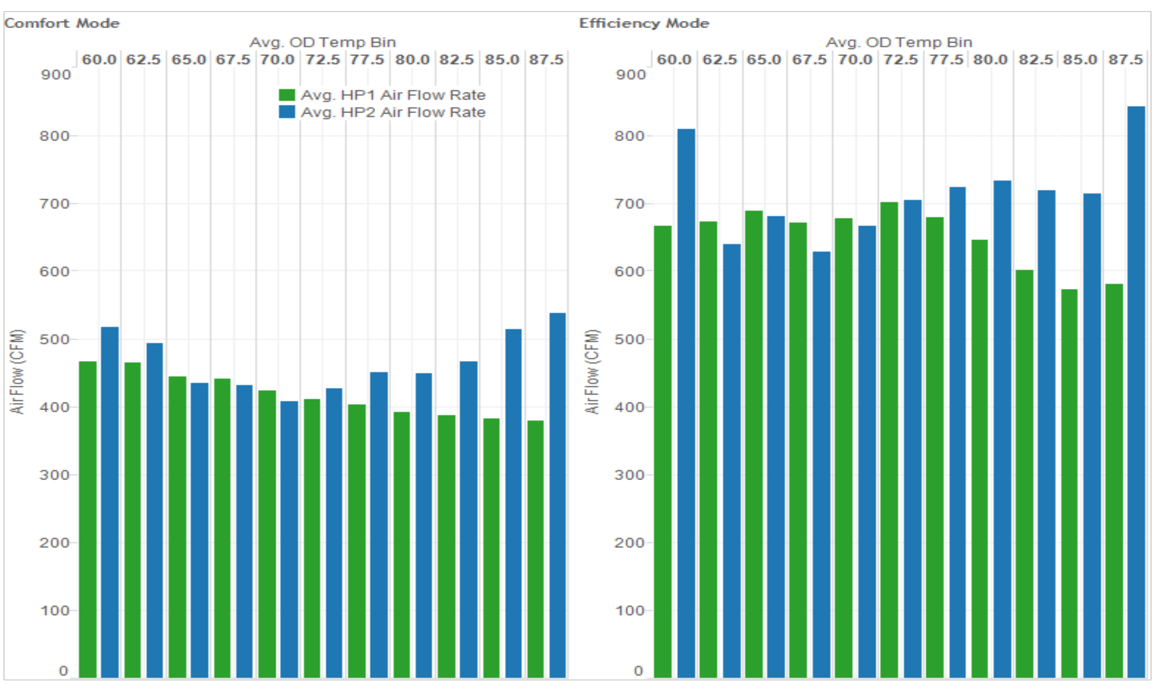


Figure 3-7
Average Airflow Rate for HP1 and HP2 in Comfort and Efficiency Modes

The above analysis was based on when the heat pumps were running; the following shows overall performance. Figure 3-8 shows the average power of both heat pumps plotted against temperature bin for each mode; also shown are the dry bulb temperature and wet bulb temperature. Comparing the power of each heat pump across modes, the total consumption was very similar in both modes, with a maximum power of 1.53 kW in Efficiency Mode, compared with 1.62 kW in Comfort Mode. Across the higher temperature bins, the power was generally approximately 5% lower in Efficiency Mode. Looking at each heat pump, HP2 actually uses more energy in Efficiency Mode than Comfort Mode, while HP1 uses more in Comfort Mode than Efficiency Mode. The reason for this subtle difference is unclear and likely related to the interaction between the two conditioned zones. Examining the indoor temperatures, both systems maintained temperature similarly, with a slight increase as measured in the downstairs zone and slight decrease in temperature upstairs, as outdoor temperature increased. In Efficiency Mode, where the upstairs heat pump ran more, the temperature decreased slightly more upstairs. The differences in temperature compared with set point are not indicative that the heat pump did not satisfy the thermostat set point; in all temperature bins the heat pumps generally ran at less than 100% duty cycle. Since the temperature displayed in this figure is an average of multiple zone temperatures, it simply reflects that the temperature across the whole zone was higher, even though the temperature near the thermostat may have been at set-point.

Examining wet-bulb temperature is where the difference between Comfort and Efficiency Modes becomes clear. In Comfort mode, the wet bulb temperature during warmer outdoor conditions was generally approximately 2°F lower for each zone. This indicates lower humidity, meaning that in Comfort Mode the heat pump was providing a larger amount of latent cooling.

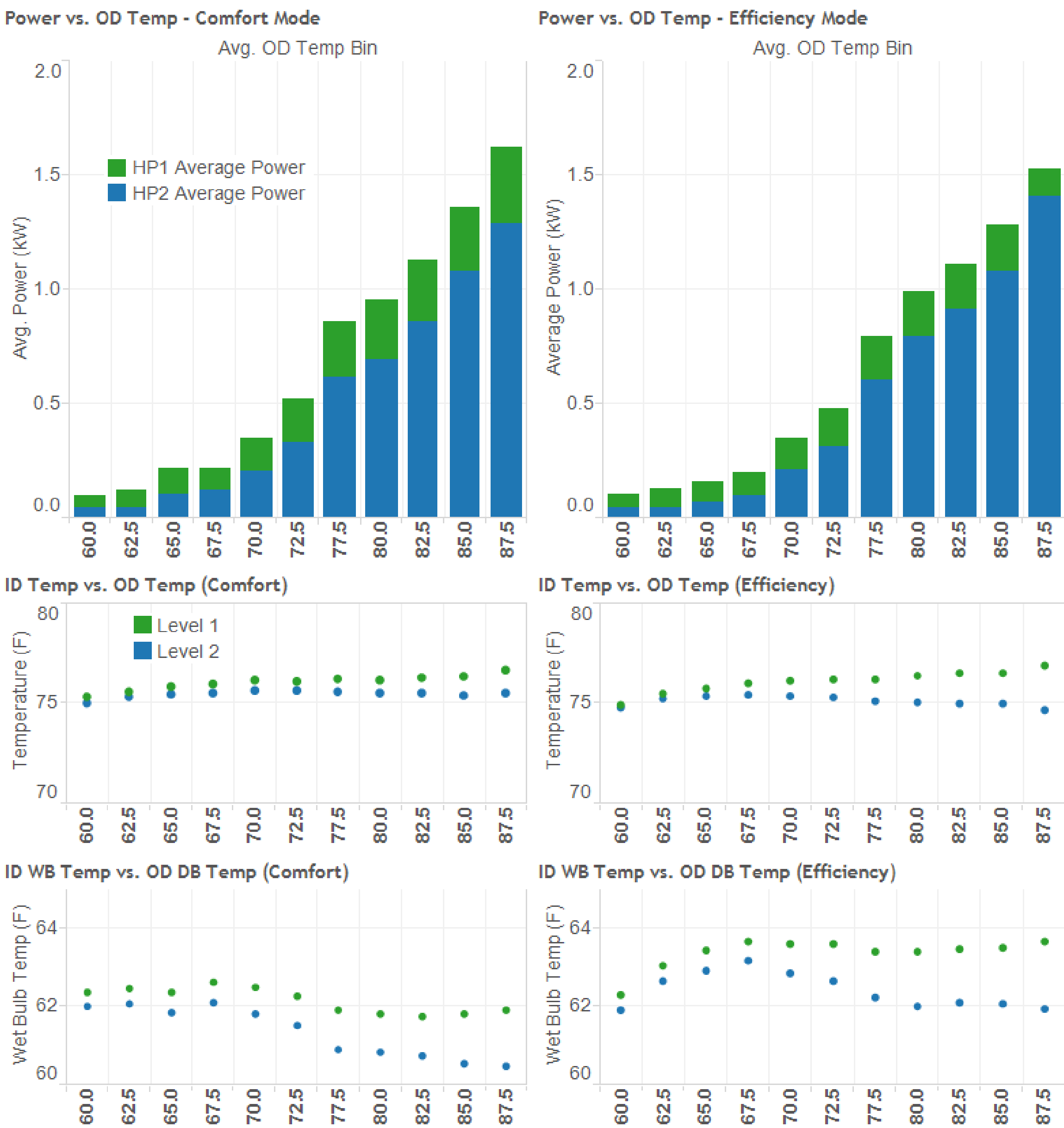


Figure 3-8
Average of Power of HP1 and HP2 for Comfort Mode and Efficiency Mode, and Indoor Dry-Bulb and Wet-Bulb Temperature, vs. Outdoor Temperature Bins

The load shape for each mode and the average outdoor temperature and relative humidity profile for the data analyzed for each mode are shown in Figure 3-9. This shows both the slight difference in load shape between HP1 and HP2 across modes, where HP1 consumption is slightly higher and HP2 consumption slightly lower during the afternoon peak in Comfort Mode than Efficiency Mode. The figure also shows that the weather for the days compared here was on average very similar, with a slightly higher average high temperature for the Efficiency Mode days. The very similar outdoor temperature and relative humidity suggest that the indoor wet-bulb differences noted above are a reflection of performance differences, and not weather differences between test days. Therefore it can be asserted that a humidity difference would exist in homes using one mode or the other, and some occupants may be less comfortable for the same set-point with higher humidity in Efficiency Mode. Though it cannot be directly verified from this study, this suggests that homeowners might naturally select a lower set-point (dry bulb) in Efficiency Mode than in Comfort Mode, since people tend to feel warmer with higher humidity.

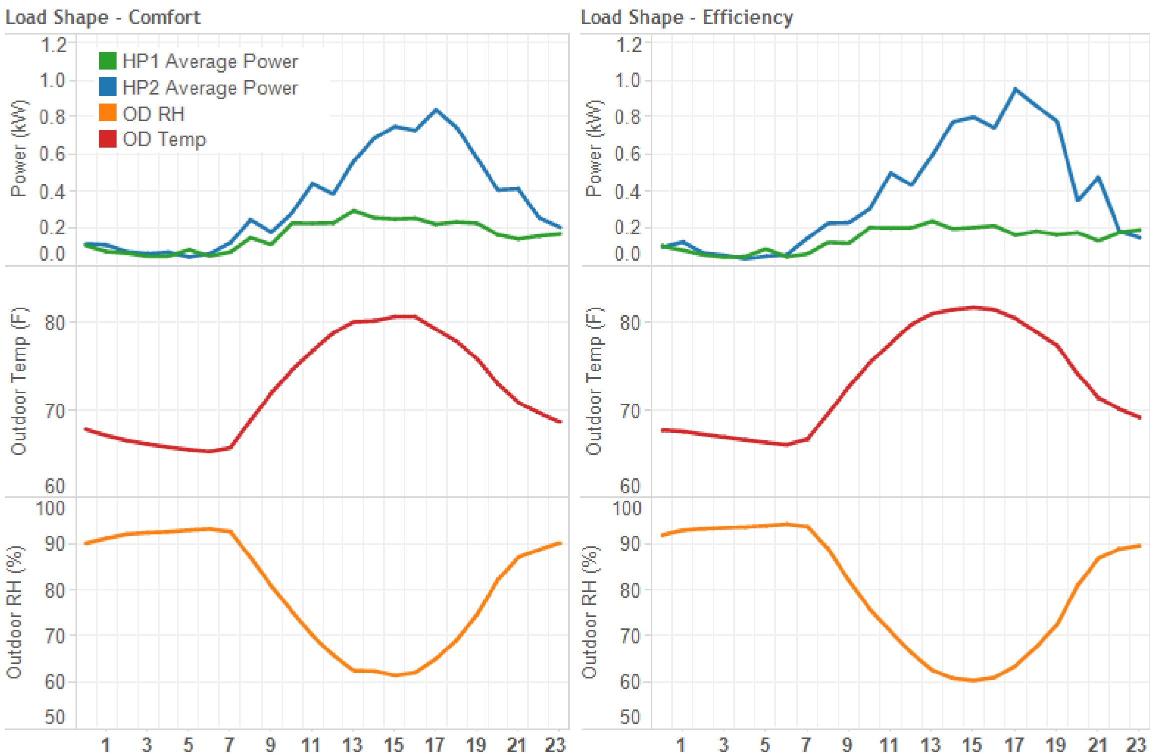


Figure 3-9
Hourly Load Shape and Outdoor Temperature and Humidity Profiles for Each Mode

The breakdown of indoor unit and outdoor unit power is also of interest in each mode; Figure 3-10 shows the power of each heat pump, separated to indoor and outdoor units for the same temperature bins in Comfort and Efficiency Modes. As can be seen, for similar conditions the indoor unit power is typically slightly higher in Efficiency Mode than in Comfort Mode, with the outdoor unit power lower. At the highest outdoor temperature bin, the HP1 power in Efficiency Mode can be seen to drop considerably, and the HP2 total power increased; the underlying data for this bin represents several different days, with periods from June, July and August represented. It does not appear that one anomalous day skewed the data, but rather it may be that HP2 operating at a higher speed caused HP1 to cycle somewhat less. The magnitude of power consumption by HP1 was much lower for all temperature bins above 70°F, again indicating that HP1 cycles at low duty cycle while HP2 absorbs much of the cooling load.

Figure 3-10
HP1 and HP2 Average Power vs. Outdoor Temperature Bin

VSHP Performance Analysis – Heating

The following graphs show the performance of the VSHP in heating mode. Figure 3-11 shows the average heating capacity, backup heat power, total power including backup heat, and supply air temperature for HP1 for periods during which the unit is operating. It should be noted that while whole-season analysis such as in Figure 3-4 shows outdoor temperatures reaching the 15°F to 20°F range, that temperature range was not reached in both modes; therefore, for comparative analysis only the bins which had adequate data in both modes are shown here. Again, the Winter of 2013-2014 has had colder temperatures, and some limited analysis can be seen in the addendum. HP1, which is the downstairs unit, had increasing capacity and average supply temperature as outdoor temperature decreased. The supply air temperature appears to increase in particular with use of backup heat. In Comfort Mode, the temperature was generally around 95°F except for the coldest periods where more strip heat was used. For efficiency mode the supply temperature ranged from 83-90°F, again increasing with backup heat usage. For both modes, backup heat began to be used starting with the 35°F temperature bin and increasing with reduced temperatures.

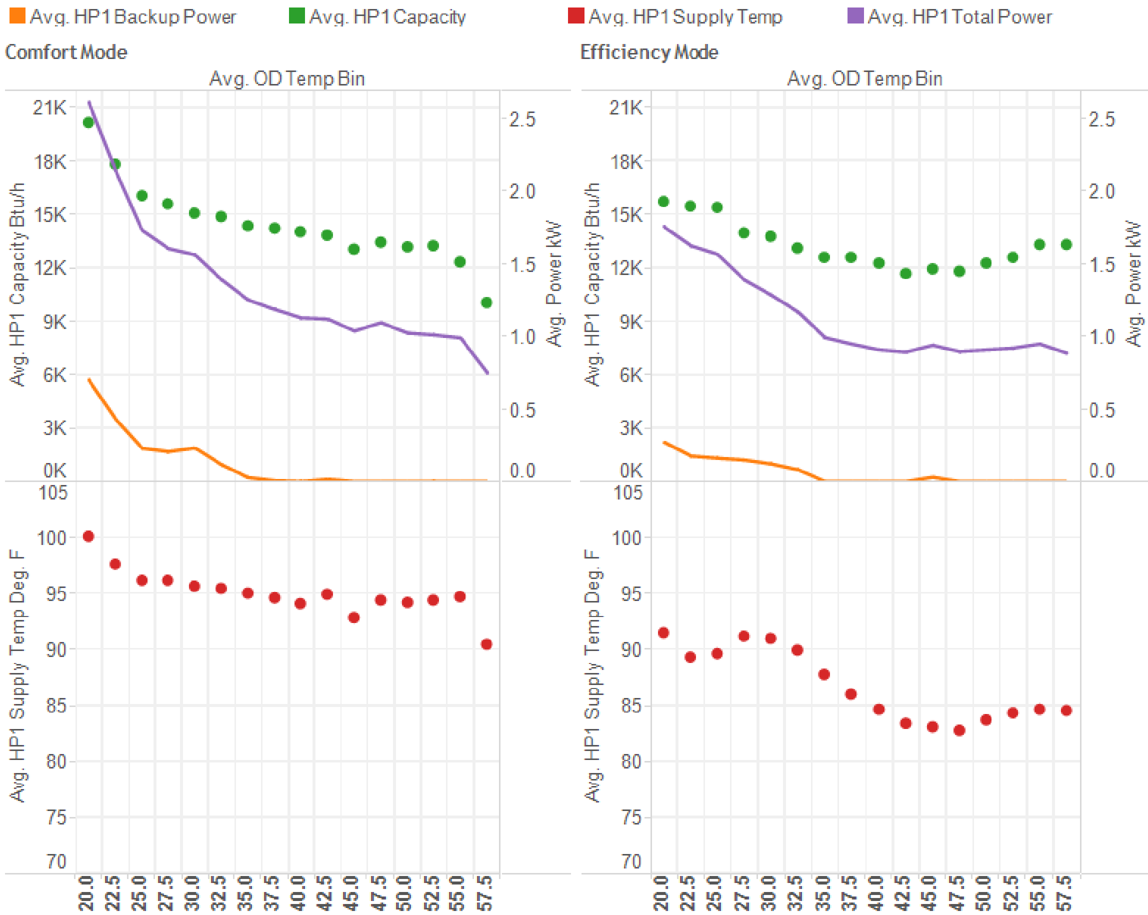


Figure 3-11
HP1 Average Capacity, Total Power, Backup Heat Power and Supply Temperature (While Running)

Figure 3-12 shows the same data for HP2. The heat pumps again used more power at lower outdoor temperatures, and had some backup heat usage starting at 35°F outdoor temperature. For this unit in comfort mode the temperature was consistently close to 90°F and ranged from 82-88°F in Efficiency Mode.

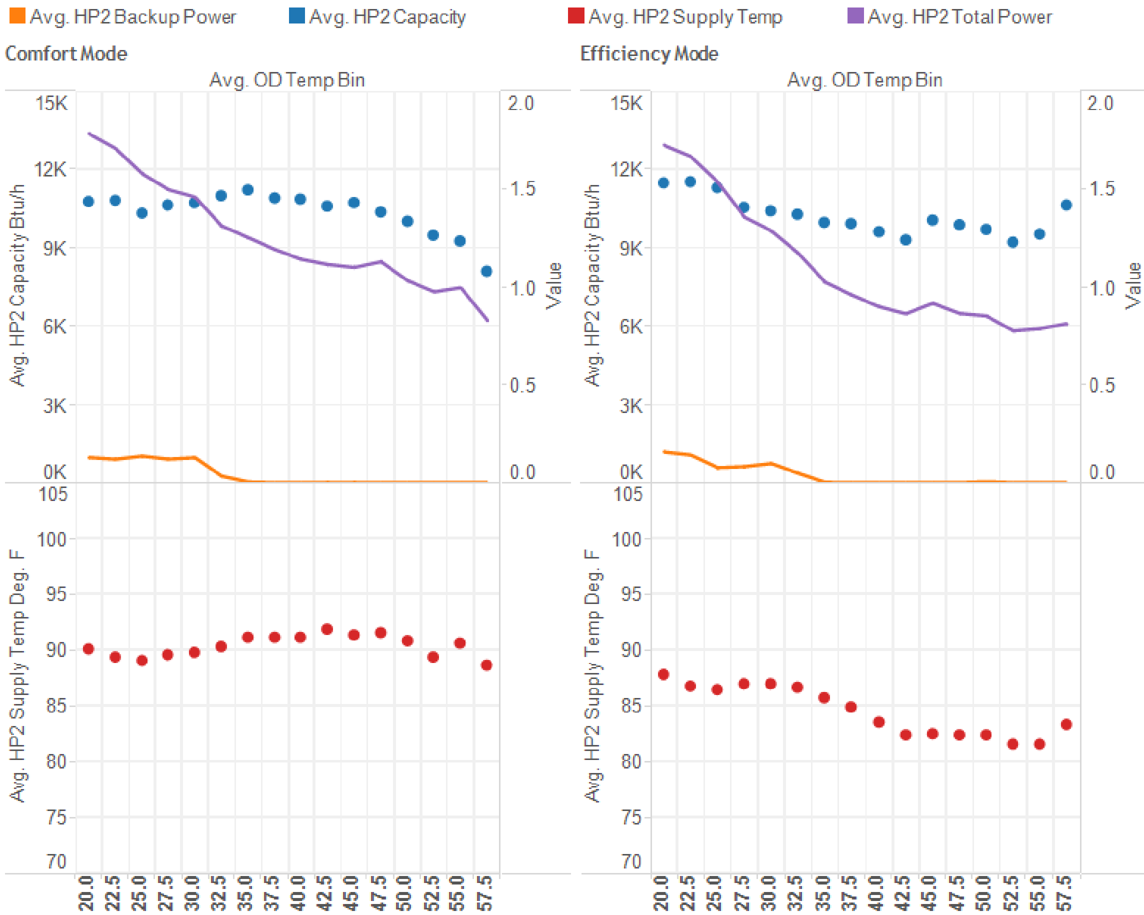


Figure 3-12
HP2 Average Capacity, Total Power, Backup Heat Power and Supply Temperature (While Running)

In heating mode it is conventional to use the metric Coefficient of Performance (COP) instead of EER. COP and EER are essentially the same, except the COP is reported with like units for power and capacity (e.g., Watts per Watt). EER values are 3.41 times the COP value. The COP for each temperature bin, calculated using the average values for capacity and power in each bin, is shown in Table 3-4. The average COPs for HP1 were considerably higher than for HP2, and the COPs in Efficiency Mode were higher than Comfort Mode for both heat pumps. The COP for HP1 ranged from 2.26 in the coldest bin, in Comfort Mode, to 4.42 for the mildest weather in Efficiency Mode.

Table 3-4
COP of Heat Pumps in Heating Mode Based on Average Capacity and Power, for Each Outdoor Temperature Bin

OD. Temp Bin	HP1, Comfort	HP1, Efficiency	HP2, Comfort	HP2, Efficiency
20	2.26	2.61	1.77	1.95
22.5	2.42	2.78	1.85	2.03
25	2.71	2.87	1.91	2.16
27.5	2.83	2.94	2.07	2.27
30	2.83	3.15	2.15	2.36
32.5	3.14	3.28	2.45	2.57
35	3.36	3.72	2.62	2.86
37.5	3.49	3.88	2.68	3.02
40	3.62	3.95	2.78	3.12
42.5	3.60	3.84	2.79	3.17
45	3.67	3.72	2.85	3.20
47.5	3.60	3.84	2.68	3.36
50	3.78	3.95	2.84	3.34
52.5	3.83	4.01	2.86	3.45
55	3.64	4.10	2.70	3.53
57.5	3.92	4.42	2.85	3.84

The airflow rate of each system is shown in Figure 3-13 and shows, like in cooling mode, that the Efficiency Mode airflow rate is higher, as expected.

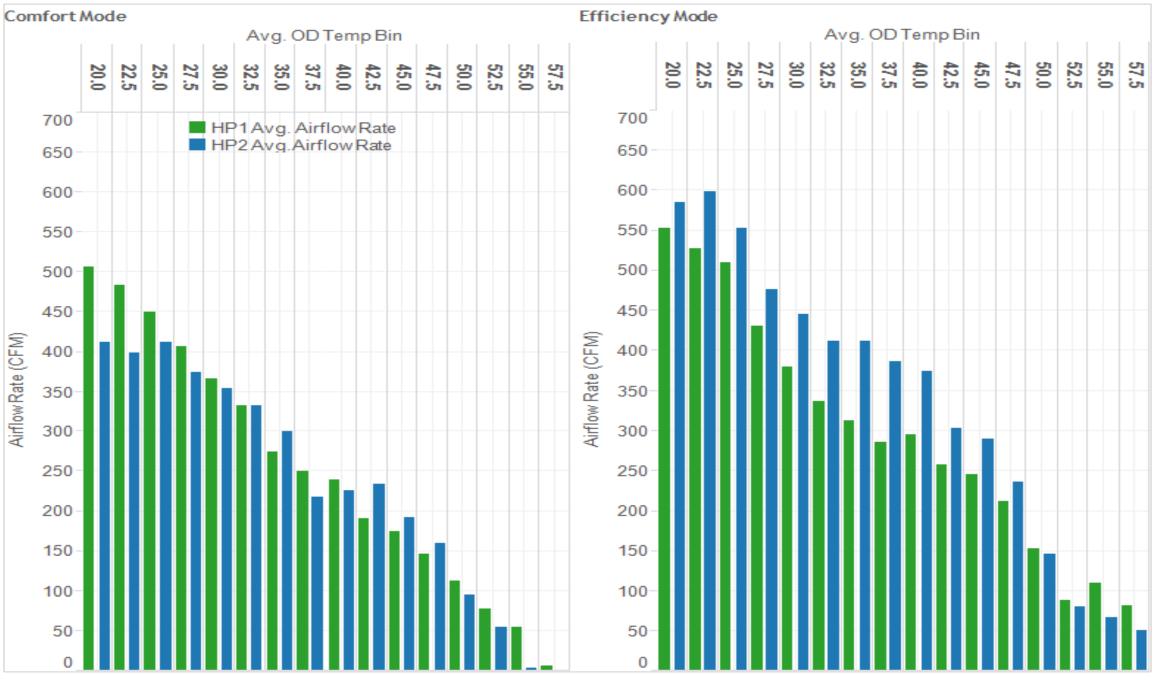


Figure 3-13
Average Airflow Rate vs. Outdoor Temperature Bin in Heating Mode

The above analysis showed performance for when each system was running; the following analysis looks at overall performance. Figure 3-14 shows the average power of each system plotted against outdoor temperature. On the left, Comfort Mode is shown and Efficiency Mode is shown on the right. The indoor temperature of each level is also shown. Efficiency Mode generally has slightly lower total power, with more power used in HP2 in Efficiency than Comfort, and more power used by HP1 in Comfort than Efficiency Mode. As will be shown below, the primary cause of this difference was higher electric resistance backup heat for HP1 in Comfort Mode. The indoor temperature for both zones and both modes never decreased significantly below the set-point of 71°F.

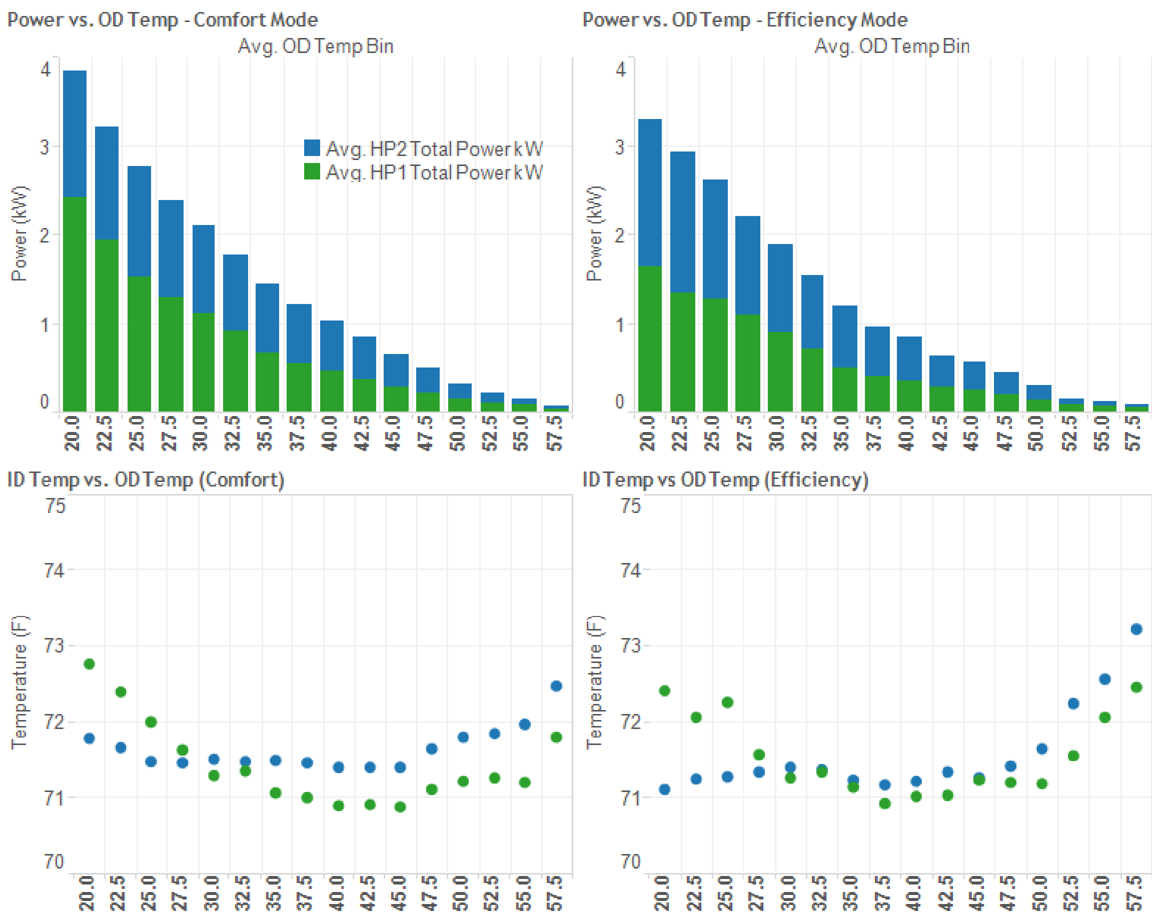


Figure 3-14
HP1 and HP2 Power, and Level 1 and Level 2 Indoor Temperature vs Outdoor Temperature for Comfort and Efficiency Mode

Each system operated in four different modes: the setting could be in Comfort or Efficiency Mode, and the defrost cycle could be performed with or without resistance heat. The following two graphs examine the entire test period. Figure 3-15 shows the indoor unit, outdoor unit, and backup heat average power for HP1 plotted against outdoor temperature bin for each of the four possible modes. Counter to what might be expected, Comfort Mode with electric resistance heat disabled during defrost resulted in the most electric resistance heat usage at low temperatures; this phenomenon was caused by a control strategy issue, and will be discussed below.

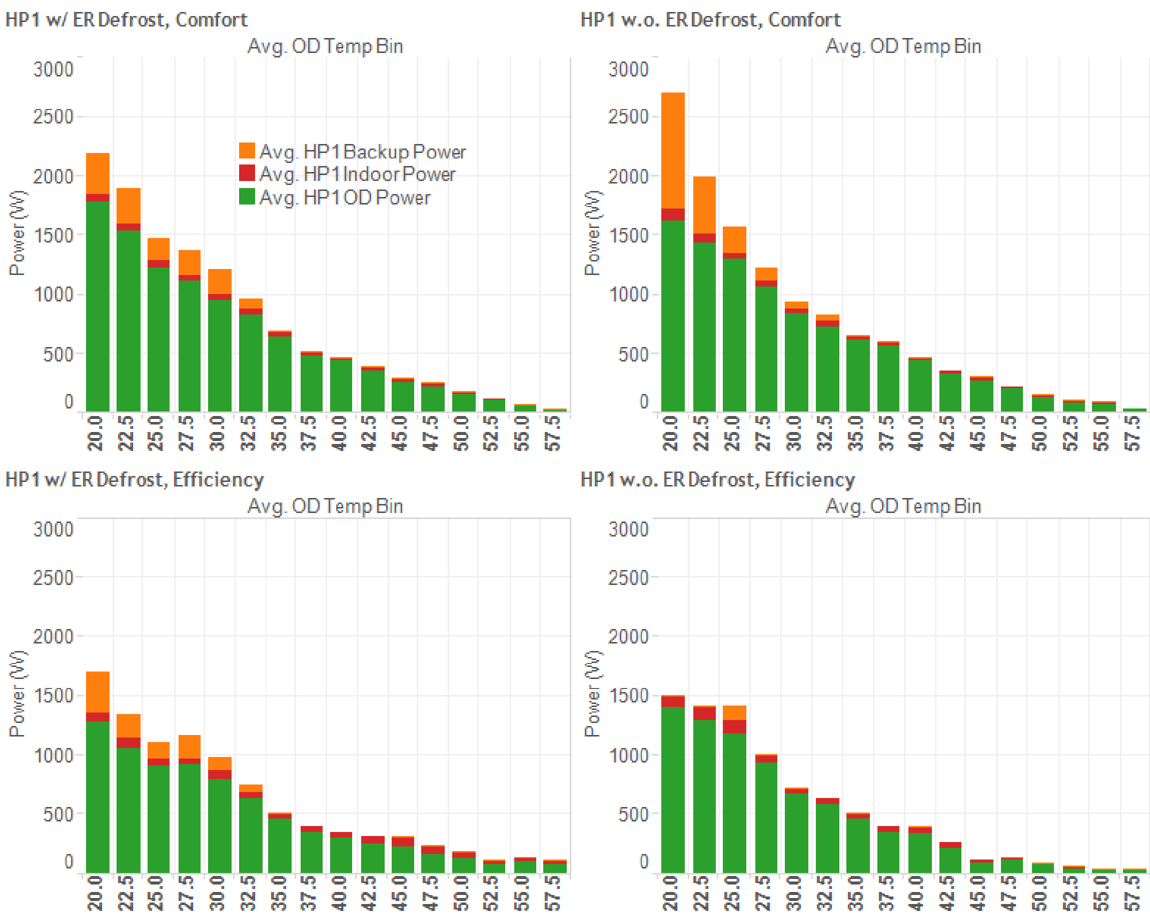


Figure 3-15
HP1 Average Power vs. Outdoor Temperature Bin in Heating Modes

Figure 3-16 shows the same graphs for HP2, the upstairs heat pump. HP2 used considerably less power in Comfort Mode at low temperatures than HP1; in Efficiency Mode the power was similar or slightly higher for HP2 than HP1. Comparing the two figures more closely, the main difference in total power between HP1 and HP2 in Comfort Mode is the electric resistance heat usage. HP1 can be expected to use more resistance heat, since HP2 receives some heating benefit from heat rising from the downstairs zone. HP2 itself uses slightly more power in Efficiency Mode than Comfort Mode for similar cold conditions.

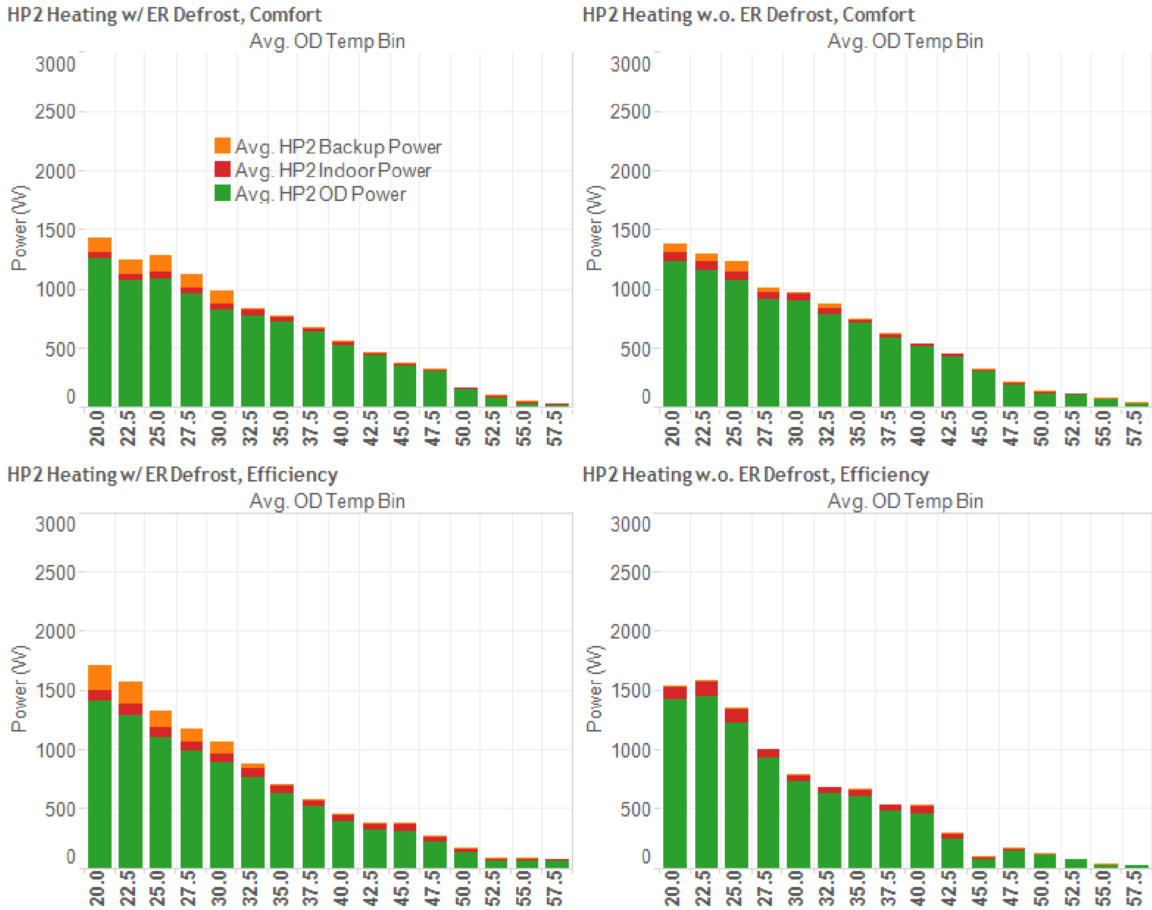


Figure 3-16
HP2 Average Power vs. Outdoor Temperature Bin in Heating Modes

As was briefly mentioned above, the defrost cycle could be performed with or without electric resistance heat and was tested in both configurations. In "with resistance heat" operation, the defrost cycle would be accompanied by backup heat to provide continuous, hot supply air. In "without resistance heat" operation, the system defrosts and briefly supplies cold air to the space, without backup heat to re-heat the air. However, the design of the system controls led to an inadvertent side effect of this feature, illustrated in Figure 3-17. In each graph, heat pump total power is shown in green, outdoor temperature in blue, and the supply air temperature, only when the unit is operating, is shown by red dots. The two graphs on the left side of the figure show January 18, 2013 in the morning. The defrost cycle can be identified by a dip in power while the heat pump is running, which is accompanied by a decrease in supply air temperature; typically the heat pump then returns to heating mode at a higher power briefly before settling to the same power level. However, four times for HP1 and once for HP2, the defrost cycle was followed immediately by the resistance heat operating, identified by the very high power draw and sharp increase in supply temperature. This is counter to the intent of defrost without resistance heat setting; the system calls for emergency backup heat in response to a short discharge of cold air, when it could recover without backup heat. Therefore, for these tests, on January 19th the electric resistance backup heat was locked out for outdoor temperatures above 10°F, with the lockout maintained for tests where defrost without resistance was to occur. The two graphs on the right show HP1 and HP2 for the morning of January 19th, where resistance heat was locked out. In that case, defrost occurs and cold air is briefly supplied before the heat pump resumes normal operation, at higher heat pump power for a brief period to recover.

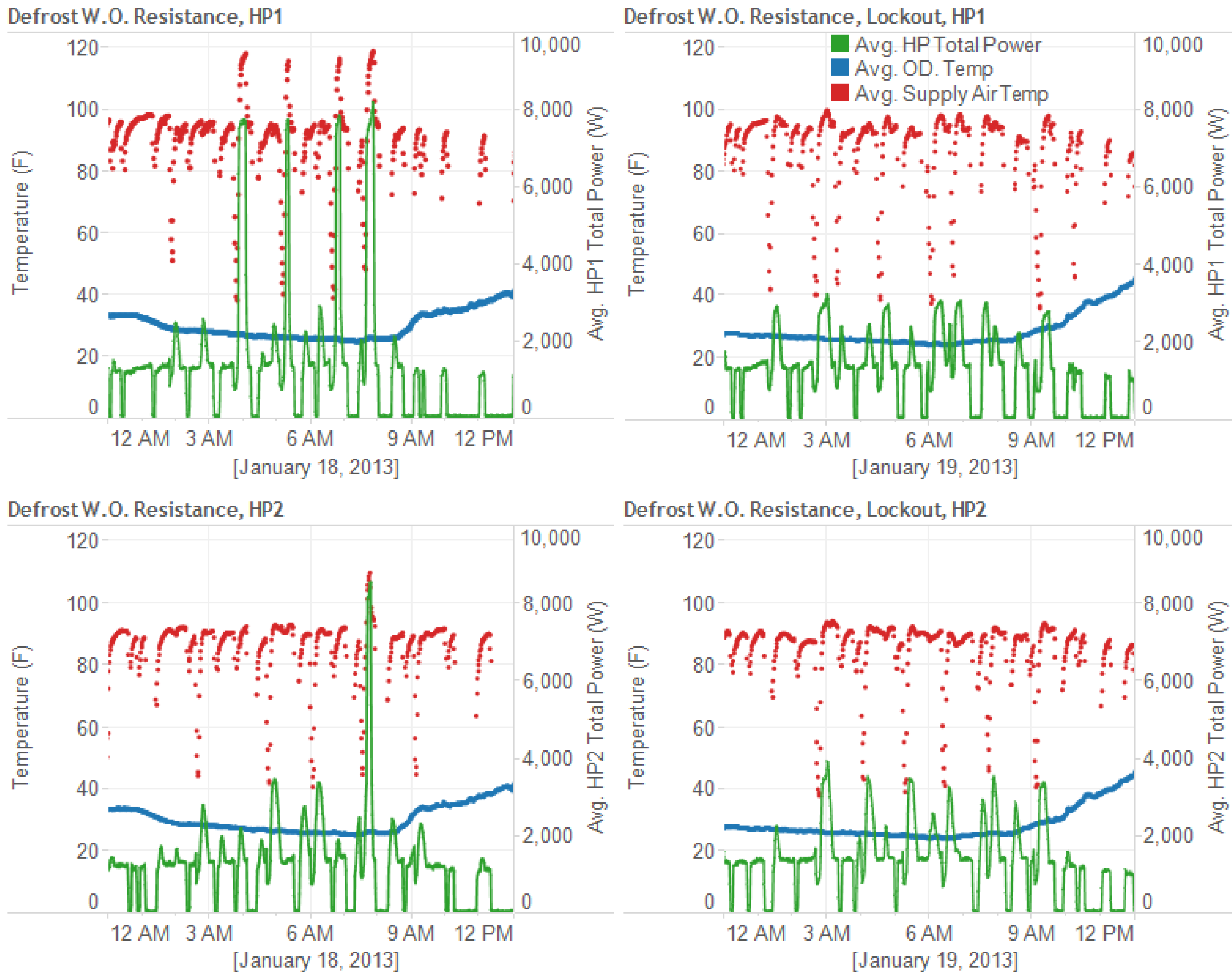


Figure 3-17
Power, Outdoor Temperature and Supply air Temperature showing Defrost Rebound with and without Resistance Heat Lockout

The defrost recovery behavior observed as the default for this system would negate efforts to reduce strip heat during peak periods; therefore, any effort to minimize strip heat through such controls should consider a reasonable resistance heat lockout, and the control strategy of future heat pumps should include the period immediately after defrost when disabling strip heat for defrost.

4

RESULTS: DEMAND RESPONSE

Throughout the cooling season, the heat pumps were tested in demand response simulations, and the following graphs illustrate the results. The nomenclature, "DR: Min" will be used for an event where the device is signaled not to exceed minimum-speed cooling, and "DR: Off" will be used for events where the device is configured to turn off. This chapter first discusses two example periods with similar weather to show how the system works in DR mode, and then quantifies the DR results across all tested conditions.

Figure 4-1 shows a day with a "DR: Min" event, on June 12 2013. The event lasted from 3:00 PM to 8:00 PM. The data shown is one-minute intervals. The two heat pumps are shown in the top graph. The bottom graph shows the outdoor temperature, and the dry-bulb and web-bulb average temperatures for both levels of the house. Both heat pumps run very little in the overnight hours, before outdoor temperatures increased quickly in the morning. The reader may notice that short cycles have a spike of higher power, followed by a "shoulder" of lower power for sustained operation, which is characteristic of the VSHP's start-up, and then power reduction to low-capacity output. HP1 ran in low-duty cycles at minimum capacity for much of the day leading to the event. HP2 ran at greater-than-minimum capacity for some period in the morning, and was at a higher capacity from approximately 1:00 PM until the event's beginning at 3:00 PM. At 3:00 PM, HP2 switched to minimum capacity and remained at minimum capacity for the duration of the event; the power can be seen to fluctuate some with outdoor temperature, as is characteristic of heat pumps. HP1, the downstairs heat pump, was already running at minimum capacity, low duty cycle, and continues to do so. It appears to cycle more frequently during the event period. After the event ends, HP2 resumes operation at higher-than-minimum cooling capacity. Since HP2 tends to remove heat from the downstairs zone as well, HP1 does not cycle during this recovery time and HP2 provides enough cooling to satisfy both zones. The indoor temperatures remained relatively steady for the entire period, though the Level 2 temperature can be seen to gradually increase by approximately 2 degrees during the minimum cooling period, before recovering after the event. The difference in power consumption for HP2 immediately before and immediately after the event, and during the event was approximately 0.5 kW; HP1 operated at minimum capacity, below 100% duty cycle for the entire day and a higher duty cycle during the event.

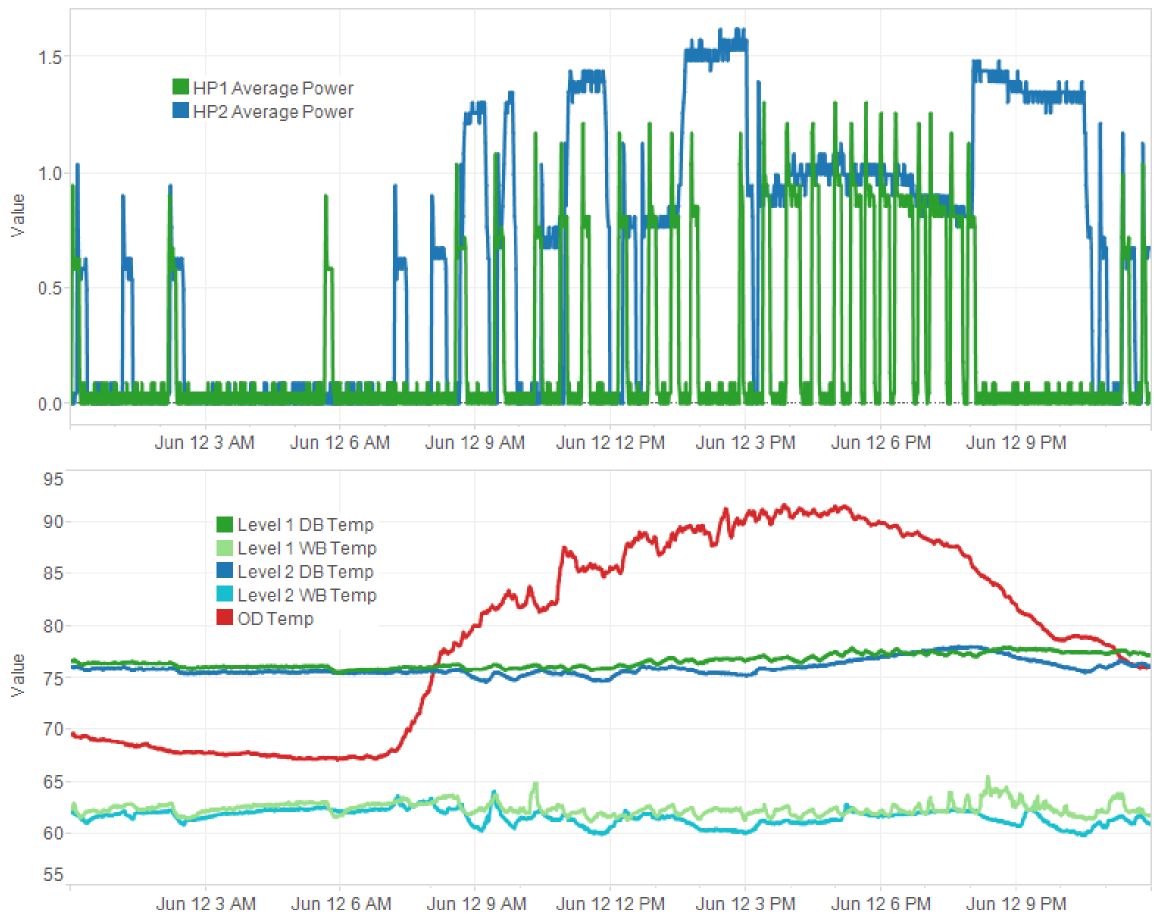


Figure 4-1
Heat Pump Power and Temperatures during a DR-Min Test Day

Figure 4-2 shows an example of a "DR: Off" test, on July 20, during which the DR signal is between 5:00 PM and 8:00 PM. This date was selected for having a similar temperature profile to that shown in Figure 4-1. In the early morning hours, both heat pumps can be seen to cycle on and off for short durations, with HP2 running for slightly longer duty cycles, and HP1 running very short duty cycles. As outdoor temperature increases, the frequency of cycles increases, and then in the mid-morning, HP2 in particular can be observed to modulate power up, above minimum capacity, before cycling off. In the afternoon, when the outdoor temperature is around 90°F, the upstairs heat pump operates at higher capacity for a sustained period while the downstairs system still cycles at low duty cycle. At the initiation of the event, HP1 was already off and HP2 was on at approximately 1.6 kW; both systems go to near-zero power. Indoor temperatures, which had remained fairly constant during the hours leading to the event, began to increase immediately for both zones. Both indoor temperatures reached a high of 81.5°F. At the end of the event, both heat pumps resume operation at a higher-than-minimum operating speed; HP1 reduces power to and then cycles off, resuming a low-usage cycling pattern at approximately 9:00 PM, while HP2 continues cooling until nearly midnight. The reduction for HP2 was from approximately 1.6 kW to zero, followed by a recovery at approximately 1.3 kW. HP1 reduced from a min-cooling, low duty cycle to off, and then recovered at approximately 1 kW for one hour.

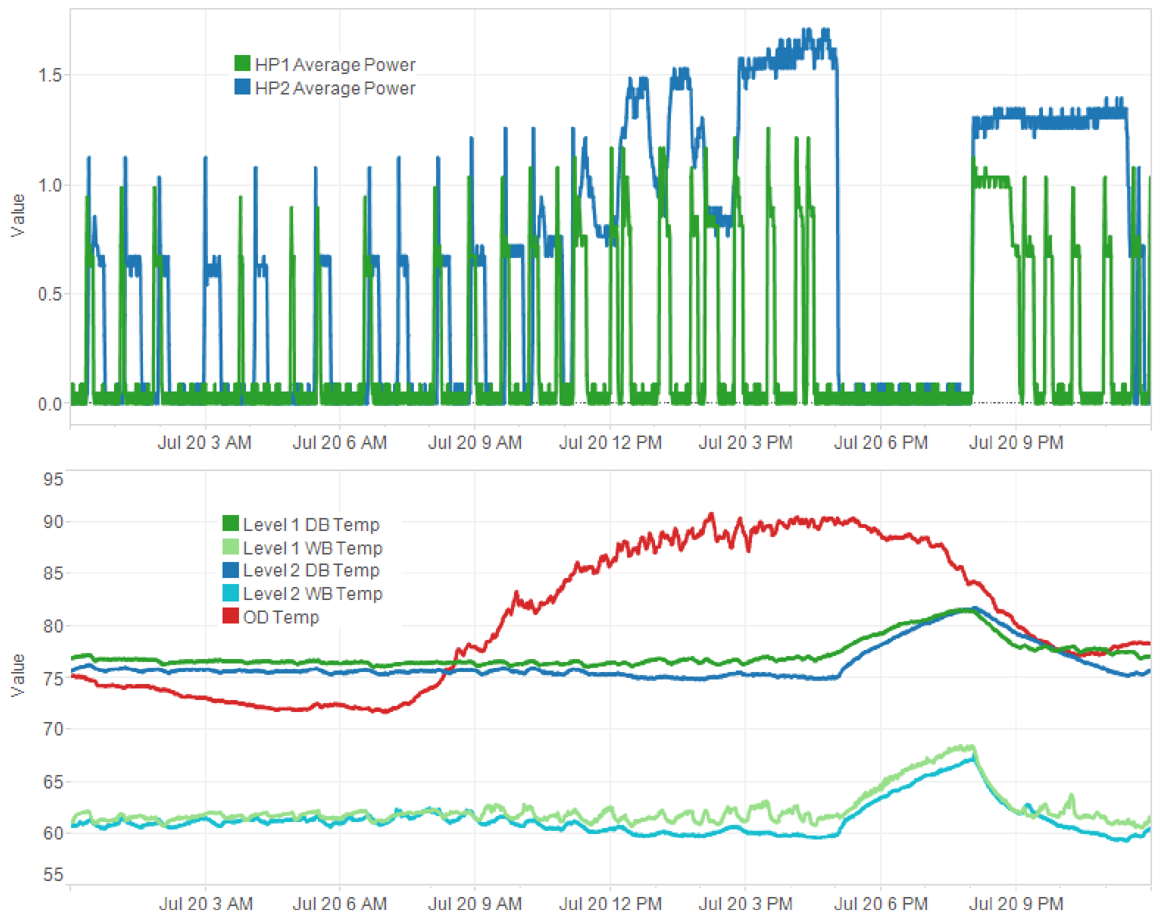


Figure 4-2
Heat Pump Power and Temperatures during a DR: Off Test Day

The above section showed examples of demand response events for two days with similar weather, but does not quantify the demand reduction provided by the heat pumps during the events. Figure 4-3 shows three graphs, each showing the average power of HP1 and HP2 plotted against the outdoor temperature bin. Each graph includes both Comfort and Efficiency Mode periods. The left-most graph shows all periods where there was no DR event. The center graph shows power only during DR: Min events. The right graph shows power during DR: Off events. The results during DR: Off events are clear: the heat pumps both turn off and draw almost no power, with the power reduction increasing with outdoor temperature. What little power is reflected in the graph may represent the device's standby power, and also some brief instances where the DR signal was on, and the device had not yet responded (for example the first few seconds after the signal was sent). Comparing this with the non-DR periods, the reduction during cooling periods was as high as 1.6 kW.

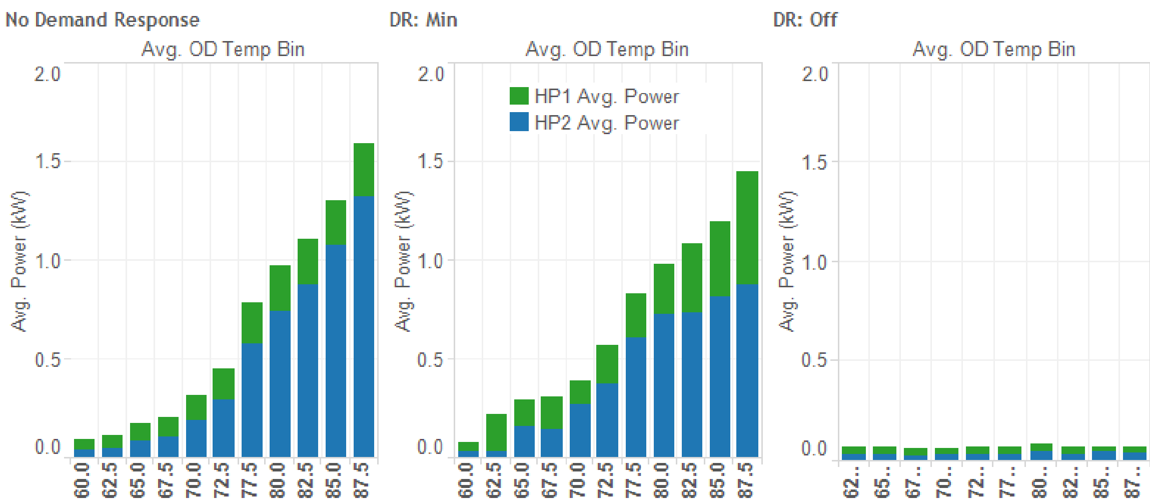


Figure 4-3
Heat Pump Power vs. Temperature for Periods of No DR Event, DR-Min Events, and DR-Off Events

The differences between the DR: Min power and the power with no DR event are less clear. With a DR: Min event the power at the highest temperatures was slightly lower, by 0.1 kW in the 87.5-90°F temperature range, a reduction of about 6%. The small reduction is explained by looking at the difference in distribution of power between HP1 and HP2. During non-DR hours, HP2 uses a large amount of power and HP1 uses very little, cycling at lower power. During a DR: Min event, HP2 does indeed reduce power considerably, for example going from 1.3 kW to 0.9 kW in the 87.5-90°F temperature range. HP1, which is already operating below full duty cycle in minimum cooling output, accommodates for the decrease of HP2 cooling by increasing its duty cycle, going from 0.3 kW to 0.6 kW.

The above observation may have very important consequences for understanding demand response with variable capacity systems. In this case, the equipment was over-sized for cooling such that one system almost always operated below minimum continuous capacity, and the interaction of the two cooling zones normally shifted much of the cooling burden to HP2, the upstairs system. Since HP1 was typically below minimum continuous capacity, it had the available capacity to operate more while still satisfying the “minimum cooling capacity” requirement. Because of this, HP2 indeed reduced its output during events, but HP1 simply compensated by operating more frequently. As a result, the DR: Off power reduction was 16 times greater than the DR: Min reduction for the hottest temperature bin.

For the systems tested here, a possible alternative approach may be to have one system in DR: Off and one in DR: Min. This approach would ensure reduced power and cooling output, while still providing some capacity to keep the temperature reasonably low. Further study in the 2014 continuation of this effort should examine this and other potential strategies. Additional research on the interactive effects between zones could be beneficial in developing future control strategies which would allow both zones to have direct cooling, but reduce capacity and power in both zones.

In addition to understanding demand reductions during DR events, it is also of interest to know how frequently the indoor temperature deviated away from set point during events. Figure 4-4 shows the hourly average indoor dry-bulb and wet-bulb temperatures plotted against the outdoor temperature, for all days with a demand response event. All hours of each day are shown, to show the normal range of temperatures during non-DR hours. The "DR: Minimum Cooling" days are shown with blue dots, the "DR: System Off" days with orange. Points from the days examined above are identified on each graph. From indoor dry-bulb temperature, it can be seen that the DR Min days do not have much deviation from the typical non-DR temperature range, with a small deviation in the upper range of outdoor temperatures for the upstairs system, and virtually no deviation for the downstairs system. This aligns with expectation, because as explained above the upstairs heat pump reduces operation but the downstairs system actually runs more. Likewise, wet-bulb temperature is well-controlled. For the DR: Off days, some major departures can be observed for both dry-bulb and wet-bulb temperature. This too aligns with expectation: with the system off, sufficient cooling cannot be provided. For any utility program considering these two options, the trade-off between occupant comfort and total power reduction is important to consider. The power reductions realized with DR: Off are greater in magnitude by far. However, occupants would likely have been uncomfortable during some of the DR: Off events, where temperatures were at times 5-6°F warmer than normal. How quickly the house deviates from set-point will depend upon the outdoor conditions and the specific construction and loading of the house. A generally safe approach may be to only consider DR: Off options for short-duration (e.g., one hour) events.

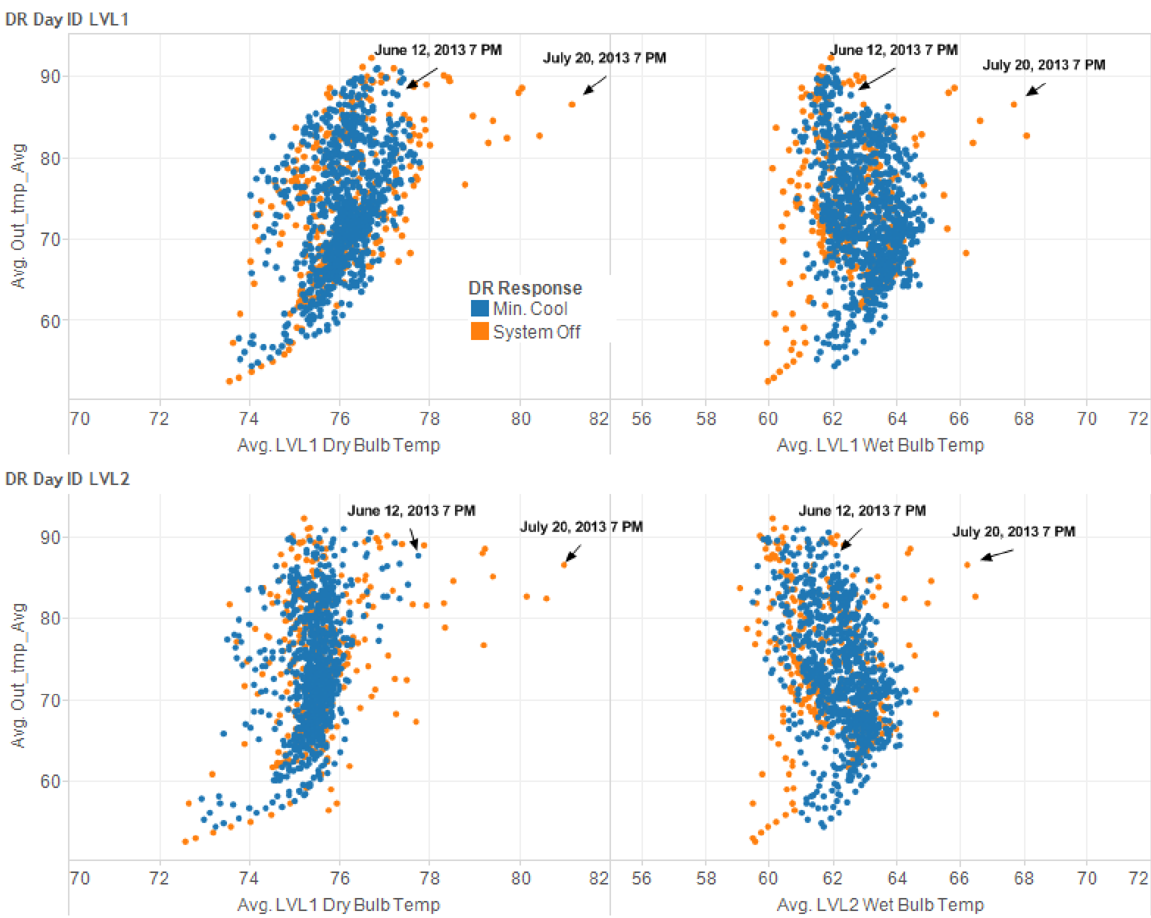


Figure 4-4
Indoor Dry-Bulb vs Outdoor Temperature for DR: Min and DR: Off Events

5
CONCLUSIONS

Through several years of simulated-occupancy field testing, two high-efficiency variable speed heat pumps and two standard-efficiency single-speed heat pumps were tested and compared at the Campbell Creek Research Homes in Knoxville, TN. The single-speed systems were installed for several years as a baseline, and the variable capacity systems are still under test at the time of this report. Though the heating and cooling seasons varied year-to-year, this report attempts to clarify the savings realized with the VSHP by comparing performance during like conditions. The report also examines the performance of the VSHP in its various operating modes and configurations including Demand Response scenarios.

The VSHP proved significantly more efficient than the SSHP, using 17-38% less power during active heating periods, and 34-38% less power during cooling conditions. The VSHP used around 50% less energy in the most moderate temperature ranges, though the magnitude of these reductions was small, on the order of 100-200 Watts. The VSHP used 22% less energy than the SSHP during one year (2010) of SSHP operating, and 42% less than the other year (2011) of SSHP operation. Though the weather was different each year and the summer and winter the VSHP has so far been tested in were comparatively mild, comparing shorter periods with similar heating and cooling degree days shows a marked improvement in both heating and cooling mode.

The performance of the VSHP was analyzed in its various modes. In both heating and cooling mode, the system was run in Efficiency and Comfort Mode. In Comfort Mode, the fan speed is lower and the supply air is hotter (in heating) or colder (in cooling), with increased compressor power; the inverse is true of Efficiency Mode. In addition to these two modes, in heating mode, the system was tested allowing or disallowing electric resistance heat to run during defrost. In cooling mode, the heat pump was found not to provide significant savings in efficiency mode, though the indoor wet bulb temperature (and humidity) increased in efficiency mode. The higher indoor humidity may reduce comfort for some people. In heating mode, the same was true, with small benefits in efficiency mode. In heating mode, with resistance heat disabled during defrost the heat pump exhibited an unexpected behavior: after briefly supplying cool air while defrosting, the electric resistance heat would often engage immediately following the defrost. This behavior could be corrected for testing by also enabling a “lock out” on the resistance heaters, but in the future heat pump manufacturers should consider extending the lockout of “defrost without resistance heat” to include the period shortly after defrost, to allow the heat pump to recover.

The VSHP system offers cooling-mode demand response capabilities of either limiting capacity to minimum speed, or turning off. These capabilities were tested in a series of scheduled, simulated demand response events. A rotation of Efficiency Mode and Comfort Mode, and Demand Response: Minimum Cooling and Demand Response: Off modes was enacted throughout the cooling season, with baseline days throughout. The results of this testing showed a significant reduction in demand with the DR: Off events, with a corresponding increase in indoor dry-bulb and wet-bulb temperature as would be expected. In DR: Min events, the interactions between the building’s two thermal zones shifted the cooling burden to be more evenly distributed between the two heat pumps, causing the average power reduction to be minimal. This finding suggests that extra considerations may be required in multi-system installations if a reduced-capacity DR approach is to be used; one possible approach would be to partner a "minimum cooling" system with a "cooling off" system, to ensure that some cooling is provided and a reduction is achieved. This should be an area of focus for the 2014 summer test schedule.

This research effort continues in 2014, with new research opportunities. CC House #2, which has a single VSHP serving both zones, will be included in data analysis. The CC House #2 data will include data from the period examined in this report, during which time CC House #2 was operated under the same DR schedule as discussed here. This portion of the study will provide interesting data for comparing a two-unit approach with a single-unit, dual zone approach. It is anticipated that the single-system approach will provide improved demand response operation. In addition, another VSHP made by a different manufacturer has been installed in CC House #3 and is being included in the study moving forward.

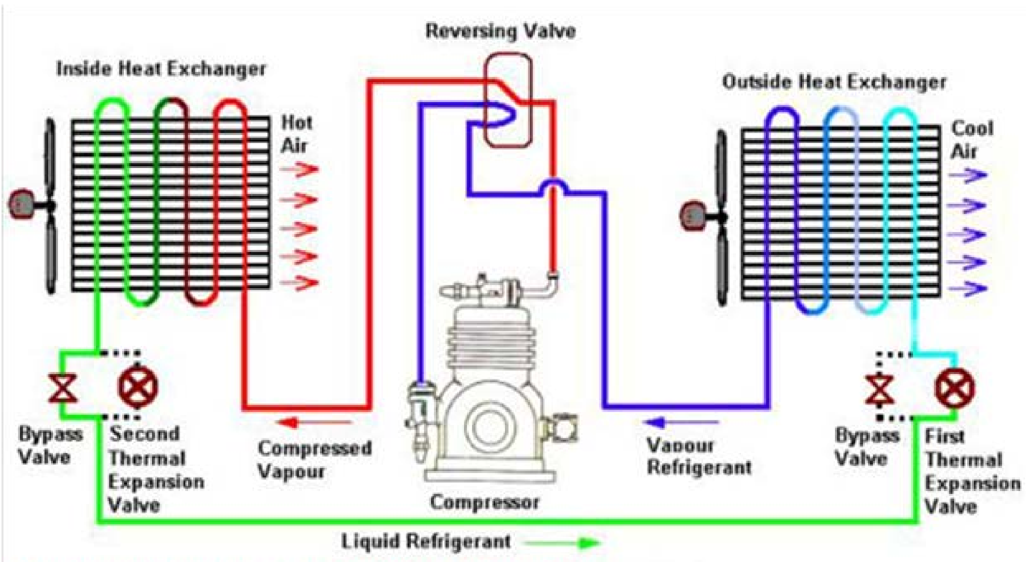
6
REFERENCES

1. Tennessee Valley Authority's Campbell Creek Research Homes Project: FY 2012 Annual Performance Report Test Results October 1, 2011—September 30, 2012; Anthony C. Gehl, Jeffery D. Munk, Philip R. Boudreaux, Roderick K. Jackson, Gannate Khowailed, SRA International. Oak Ridge National Laboratory, Oak Ridge, TN
2. "Historical Weather." *Weather History & Data Archive*. N.p., n.d. Web. 07 Jan. 2014. http://www.wunderground.com/history

A

HEAT PUMP BACKGROUND

Heat pump systems offer an alternative to the common HVAC configuration of single split unitary air conditioners with gas heating. In simple terms, heat pumps are traditional air conditioning units with the added capability of running in reverse. That is, where the traditional air conditioning unit has an indoor evaporator to remove heat from the space and an outdoor condenser to reject heat to the ambient, heat pumps can also reverse this configuration. This is accomplished using a system of reversing valves. A simplified schematic of a heat pump in heating mode is shown in Figure A-1.



http://www.nccc.gov.sg/building/heat_pump.shtm

Figure A-1
Heat Pump Schematic in Heating Mode

What gives the heat pump an advantage over other forms of electrical heat is that it uses the thermodynamic properties of a refrigerant to move more energy than is required to operate the system. By using a compressor to manipulate the pressure of the refrigerant, the heat pump causes the refrigerant to evaporate – absorbing heat in the process – at the conditions outdoors, and condense – rejecting heat – indoors. Since it is the refrigerant and not the compressor moving heat, it is possible to move a greater amount of heat than could be provided by directly converting the electrical input alone.

Heat pumps have advantages as well as disadvantages when compared with gas or oil heat. One advantage of heat pumps is that they only require electrical input. In cases in which infrastructure for gas or oil is not already in place, this can be a major consideration. Another advantage is having heating and cooling centralized to one device. This can simplify controls and ducting. Heat pumps can also offer highly efficient heating with coefficients of performance (COP) exceeding 4.0 in mild conditions.

Traditional single-speed unitary heat pumps are normally supplemented with a second stage of either electric resistance or gas heat that is called when the indoor set point cannot be maintained by the heat pump alone. This balance point generally occurs in the range of ~30˚F, but is dependent upon other factors including the indoor set point and the equipment sizing. Second-stage heat is also called during defrost cycles which occur every 30-90 minutes in typical timed-defrost control schemes. Frosting of the heat pump evaporator (outdoor coil) occurs when outdoor temperature is roughly between 42˚F and 20˚F. Frost creates an insulating layer on the heat exchanger and drives the heating capacity and coefficient of performance down, necessitating defrosting. At lower temperatures the moisture content of air is too low for significant frosting, but most traditional single-speed heat pumps go through a defrost cycle anyway, based only on the upper limit trigger. The entire frosting and defrosting process is a large energy penalty on the otherwise efficient operation of a direct expansion heat pump cycle.

In heating mode, a heat pump is removing heat from the cold outdoor space, and rejecting it into the warm indoor space. The indoor space can be 70°F warmer, or more, than the outdoor. The declining potential of heat pumps at lower outdoor temperatures can be demonstrated by looking at the thermodynamic ideal representation, the Carnot Cycle. The Carnot heat pump COP given by:

$$COP = \frac{Q_H}{W} = \frac{Q_H}{Q_H - Q_C}$$

This can be represented in the ideal case by:

$$\frac{T_H}{T_H - T_C}$$

where T_H is the temperature of the hot reservoir, in this case the indoor space, and T_C is the temperature of the cold reservoir; both temperatures are in Rankine units (or Kelvin, for SI). This is represented in Figure A-2. There is a thermodynamic limit to efficiency which becomes more apparent as higher temperature differences between outdoor and indoor air are encountered. However, heat pumps remain fundamentally able to provide adequate heat in the case of a large temperature difference (low outdoor temperature) with proper design considerations. In cases of a very high temperature difference, multi-stage compression or cascade heat pump arrangements can be used, but the single-stage equipment, if sufficient to meet the load, is most cost effective.

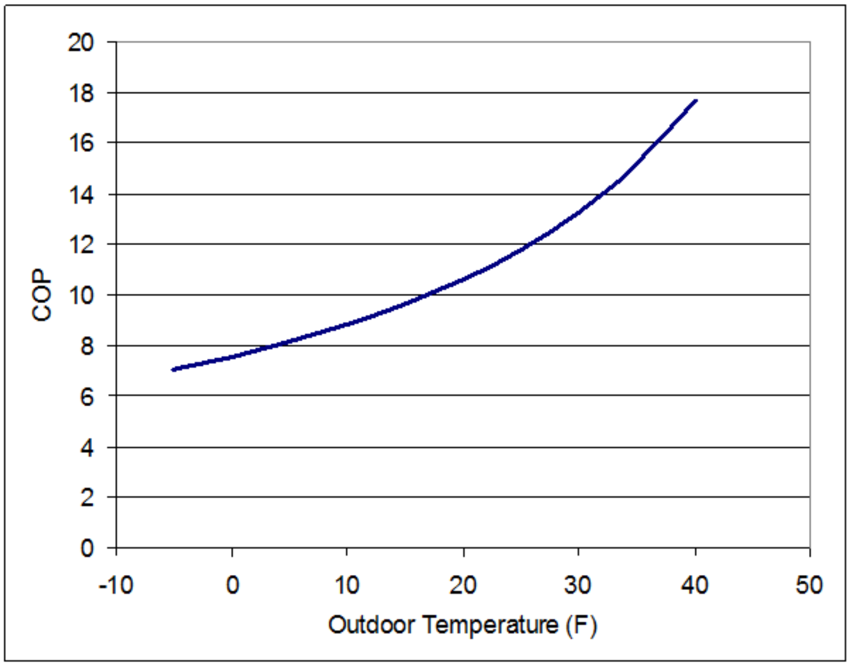


Figure A-2
Maximum COP of a Perfect Heat Pump

Fixed capacity (single-speed) heat pump systems have the unfortunate characteristic that capacity decreases in the direction of increasing load; this is true of air conditioners and heat pumps. Cooling capacity of an air conditioner decreases as the outdoor temperature rises (as the demand for cooling increases). Likewise the heating capacity of a heat pump decreases as outdoor temperature drops. This behavior of heat pumps creates a compounding problem; the colder the ambient temperature, the more heating is demanded in a building; at the same time, less heating is available from the heat pump.

The most common recourse for heat pump users in cold climates is to have electric resistance heating backup for particularly cold days. While this is certainly effective, resistance heating can provide at a maximum one unit of heating for each unit of electricity. In contrast a heat pump can provide two, three or four units of heating per unit of electricity.

Variable speed equipment offers advantages for increasing overall energy performance of heat pump equipment by having a wider range for providing full capacity, and by having the ability to control operating to minimize frost formation. Some manufacturers claim that variable speed heat pumps can be deployed without a second stage back-up in ambient temperatures down to -5˚F and below without need for electric resistance back-up.

Since real systems are hindered by factors such as mechanical friction and thermodynamic losses, the COP of real systems is always lower than the ideal discussed above. Coefficients of Performance in the range of 3-4 are common for the common design condition of 47°F outdoor temperature.

Another metric frequently used to rate the performance of heat pumps in heating mode is the Heating Seasonal Performance Factor (HSPF). The HSPF is the total heating provided (in BTU) divided by the electricity consumed (in Watt-hours) for a normal heating season. The HSPF is determined using tests defined in AHRI 210/240 [1]. The HSPF – like the COP - is useful as a comparative metric for heat pumps, but does not necessarily portray how the heat pump will operate in a given climate throughout all conditions of a heating season.

Source for Appendix A: *Air Source Heat Pumps: Laboratory Testing of the Heating Capacity of Heat Pumps at Low Outdoor Temperature Conditions.* EPRI, Palo Alto, CA: 2010. 1020130.

B
HERS CERTIFICATES

Home Energy Rating Certificate

1350 Hillman Road
Knoxville, TN 37932

5 Stars
Confirmed
HERS Index: 77

Registry ID	397043587
Rating Number	
Certified Energy Rater	Wes Soward
Rating Date	4/30/13
Rating Ordered For	Tennessee Valley Authority

General Information

Conditioned Area	2468 sq. ft.	House Type	Single-family detached
Conditioned Volume	21414 cubic ft.	Foundation	Slab
Bedrooms	3		

Mechanical Systems Features

Air-source heat pump:	Electric, Htg: 10.5 HSPF. Clg: 19.2 SEER.
Air-source heat pump:	Electric, Htg: 10.5 HSPF. Clg: 19.1 SEER.
Water Heating:	Heat pump, Electric, 2.40 EF, 50.0 Gal.
Duct Leakage to Outside	172.00 CFM.
Ventilation System	Exhaust Only: 38 cfm, 110.0 watts.
Programmable Thermostat	Heat=Yes; Cool=Yes

Building Shell Features

Ceiling Flat	R-38.0	Slab	R-5.0 Edge, R-0.0 Under
Sealed Attic	N/A	Exposed Floor	R-19.0
Vaulted Ceiling	N/A	Window Type	Double .50 .56*****
Above Grade Walls	R-13.0	Infiltration Rate	Htg: 2250 Clg: 2250 CFM50
Foundation Walls	N/A	Method	Blower door test

Lights and Appliance Features

Percent Interior Lighting	15.00	Range/Oven Fuel	Electric
Percent Garage Lighting	0.00	Clothes Dryer Fuel	Electric
Refrigerator (kWh/yr)	580.00	Clothes Dryer EF	3.01
Dishwasher Energy Factor	0.69	Ceiling Fan (cfm/Watt)	70.40

Estimated Annual Energy Cost

Use	MMBtu	Cost	Percent
Heating	20.9	$513	31%
Cooling	5.4	$133	8%
Hot Water	4.5	$110	7%
Lights/Appliances	30.8	$757	45%
Photovoltaics	-0.0	$-0	-0%
Service Charges		$162	10%
Total	61.5	$1675	100%

Criteria

This home meets or exceeds the minimum criteria for the following:

TITLE
Company
Address
City, State, Zip
Phone #
Fax #

The Home Energy Rating Standard Disclosure for this home is available from the rating provider.
REM/Rate - Residential Energy Analysis and Rating Software v14.2
This information does not constitute any warranty of energy cost or savings.

Air Leakage

Property	Organization	HERS
Tennessee Valley Authority	Conservation Services Group	Confirmed
1350 Hillman Road	865-202-6413	4/30/13
Knoxville, TN 37932	Wes Soward	Rater ID:0198007

Weather:Knoxville, TN
CC1
R2Builder - WS.blg

Builder
Rhodes Communities

Whole House Infiltration

	Blower Door Test	
	Heating	Cooling
Natural ACH	0.33	0.23
ACH @ 50 Pascals	6.30	6.30
CFM @ 25 Pascals	1434	1434
CFM @ 50 Pascals	2250	2250
Eff. Leakage Area (sq.in)	123.5	123.5
Specific Leakage Area	0.00035	0.00035
ELA/100 sf shell (sq.in)	2.17	2.17

Duct Leakage

Leakage to Outside Units	Main ducts	Floor 2
CFM @ 25 Pascals	43	129
CFM25 / CFMfan	0.0508	0.1159
CFM25 / CFA	0.0348	0.1045
CFM per Std 152	N/A	N/A
CFM per Std 152 / CFA	N/A	N/A
CFM @ 50 Pascals	67	202
Eff. Leakage Area (sq.in)	3.70	11.11
Thermal Efficiency	N/A	N/A
Total Duct Leakage Units	**CFM25/CFA**	**CFM25/CFA**
Total Duct Leakage	0.0535	0.1167

Ventilation

Mechanical	Exhaust Only
Sensible Recovery Eff. (%)	0.0
Total Recovery Eff. (%)	0.0
Rate (cfm)	38
Hours/Day	24.0
Fan Watts	110.0
Cooling Ventilation	Natural Ventilation

ASHRAE 62.2 - 2010 Ventilation Requirements

For this home to comply with ASHRAE Standard 62.2 - 2010 Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings, a minimum of 55 cfm of mechanical ventilation must be provided continuously, 24 hours per day. Alternatively, an intermittently operating mechanical ventilation system may be used if the ventilation rate is adjusted accordingly. For example, a 109 cfm mechanical ventilation system would need to operate 12 hours per day, as long as the system operates to provide required average ventilation once each hour.

Energy Cost and Features

Property
Tennessee Valley Authority
1350 Hillman Road
Knoxville, TN 37932

Organization
Conservation Services Group
865-202-6413
Wes Soward

HERS
Confirmed
4/30/13
Rater ID:0198007

Weather:Knoxville, TN
CC1
R2Builder - WS.blg

Builder
Rhodes Communities

Annual Energy Costs	$/yr
Heating	519
Cooling	139
Water Heating	110
Lights & Appliances	757
Photovoltaics	-0
Service Charges	162
Total	1687
Average Monthly($/Month)	141

Energy Features

Ceiling w/Attic	R-38 Blown, Attic***** U=0.026
Sealed Attic	None
Vaulted Ceiling	None
Above Grade Wall	R-13 G3***** U=0.097
Foundation Walls (Cond)	None
Foundation Walls (Uncond)	None
Doors	Steel-urth w/brk U=0.187
Windows	Double .50 .56***** U=0.500
Frame Floors	R-19 U=0.059
Slab Floors	R-5 Perimeter***** U=0.129
Infiltration	Htg: 2250 Clg: 2250 CFM50
Infiltration Measure	Blower door test
Mechanical Ventilation	Exhaust Only: 38 cfm, 110.0 watts.
Interior Mass	None
Mechanical Equipment 1	ASHP: Htg: 69.7 kBtuh, 10.5 HSPF. Clg: 33.4 kBtuh, 19.2 SEER.
Mechanical Equipment 2	ASHP: Htg: 60.6 kBtuh, 10.5 HSPF. Clg: 25.4 kBtuh, 19.1 SEER.
Mechanical Equipment 3	Water Heating: Heat pump, Elec, 2.40 EF.
Programmable Thermostat	Heat=Yes; Cool=Yes
Ducts	R-6.0 Conditioned space
Duct Leakage to Outside	43.00 CFM @ 25 Pascals
Total Duct Leakage	66.00 CFM @ 25 Pascals
Lights/Appliances	Defaults
Active Solar	None
Photovoltaics	0.00
Sunspace	No

Note: Where feature level varies in home, the dominate value is shown.

REM/Rate - Residential Energy Analysis and Rating Software v14.2
This information does not constitute any warranty of energy cost or savings.

Action Report

Property
Tennessee Valley Authority
1350 Hillman Road
Knoxville, TN 37932

Weather:Knoxville, TN
CC1
R2Builder - WS.blg

Organization
Conservation Services Group
865-202-6413
Wes Soward

Builder
Rhodes Communities

HERS
4/30/13
Rater ID:0198007
Rating Type: Confirmed

The following table identifies and ranks energy use and cost by building component. A maximum of six components are shown. Current mechanical equipment is assumed for this analysis. To determine the impact of varying the equipment efficiency, change the equipment specified in the building file and perform the energy calculations again.

ANNUAL ENERGY PROFILE

Energy End-Use	Component	Consumption(MMBtu/yr)	Cost($/yr)
HEATING	Above Grade Walls	8.4	207
	Slab Floors	4.6	112
	Infiltration	3.9	95
	Ducts	3.2	79
	Windows/Skylights	2.5	62
	Mechanical Ventilation	1.7	42
	Other	-3.2	-77
	Total	21.1	519
COOLING	Internal Gains	3.4	84
	Windows/Skylights	2.0	50
	Ducts	1.2	30
	Above Grade Walls	0.3	8
	Ceilings/Roofs	0.2	4
	Infiltration	0.1	3
	Other	-1.6	-41
	Total	5.7	139
WATER HEATING	Water Heater	4.5	110
LIGHTS & APPLIANCES	Lights & Appliances	30.8	757

Performance Factors

Property	Organization	HERS
Tennessee Valley Authority	Conservation Services Group	Confirmed
1350 Hillman Road	865-202-6413	4/30/13
Knoxville, TN 37932	Wes Soward	Rater ID:0198007

Weather:Knoxville, TN
CC1
R2Builder - WS.blg

Builder
Rhodes Communities

Normalized Loads	Btu/sf shell area/DD
Heating	2.38
Cooling	8.09

Normalized Consumption	kBtu/sf floor area/yr
Heating	8.6
Cooling	2.3
Water Heating	1.8
Lighting	3.7
Appliances	8.8
Photovoltaics	-0.0
Total w/o PV	**25.1**
Total	**25.1**

Normalized Consumption	Btu/sf floor area/DD
Heating	2.3
Cooling	3.9

Normalized Costs	$/sf floor area/yr
Heating	0.210
Cooling	0.056
Water Heating	0.044
Lighting	0.091
Appliances	0.216
Photovoltaics	-0.000
Total	**0.618**

Normalized Design Loads	Btuh/sf shell area/DD
Heating	0.0026
Cooling	0.0099

Normalization Factors	
Floor Area	2468
Shell Area	5695
Heating Degree Days (B65)	3666
Cooling Degree Days (B74)	592

REM/Rate - Residential Energy Analysis and Rating Software v14.2
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Performance Report

Property
Tennessee Valley Authority
1350 Hillman Road
Knoxville, TN 37932

Weather:Knoxville, TN
CC1
R2Builder - WS.blg

Organization
Conservation Services Group
865-202-6413
Wes Soward

Builder
Rhodes Communities

HERS
Confirmed
4/30/13
Rater ID:0198007

Annual Load	MMBtu/yr
Heating	49.7
Cooling	27.3
Water Heating	11.6

Annual Consumption	MMBtu/yr
Heating	21.1
Cooling	5.7
Water Heating	4.5
Lights & Appliances	30.8
Photovoltaics	-0.0
Total	**62.0**

Annual Energy Cost	$/yr
Heating	519
Cooling	139
Water Heating	110
Lights & Appliances	757
Photovoltaics	-0
Service Charges	162
Total	**1687**

Design Loads	kBtu/hr
Space Heating	53.9
Space Cooling	33.5

Utility Rates	
Electricity	LCUB**

REM/Rate - Residential Energy Analysis and Rating Software v14.2
This information does not constitute any warranty of energy cost or savings.

C

ADDENDUM: BRIEF OVERVIEW OF JANUARY AND FEBRUARY, 2014 HEATING DATA

This section provides a brief overview of preliminary findings from January and February, 2014. During January, Knoxville had 1,064 Heating Degree Days. Included during that time was a day in which TVA had record energy consumption early in the month, though there was even colder weather towards the end of January. In February, there were 665 HDD. In January, HP1 used 1,135 kWh, and HP2 used 855 kWh. In February, HP1 used 440 kWh and HP2 used 430 kWh. The totals were 1,990 kWh or 1.9 kWh/degree day in January, and 870 kWh or 1.3 kWh/degree day in February. For comparison, the next-nearest HDD month was December, 2010, and the SSHPs consumed 2,298 kWh (15% more energy). February, 2013 with 649 HDD was similar in HDD to February 2014, and the energy consumption was also similar: 852 kWh for the VSHPs in 2013, compared with 870 kWh in 2014 (a 2% difference).

The average power for each outdoor temperature bin with sufficient data is shown in Table C-1. It was decided for this period to filter periods during which less than ten hours of heat pump operation was recorded for aggregate numbers, though there were brief periods where the temperature reached below 0°F. For the winter 2013-2014 data, the system was in Efficiency Mode starting January 5, 2014. The data shown for winter 2012-13 is an aggregate of all operating modes.

Table C-1
Average Power vs. Outdoor Temperature Bin

	Winter 2012-13			January & February, 2014		
Outdoor Temp Bin (F)	**HP1 (2-ton) Avg. Power (kW)**	**HP2 (3-ton) Avg. Power (kW)**	**HP1 + HP2 Avg. Power (kW)**	**HP1 (2-ton) Avg. Power (kW)**	**HP2 (3-ton) Avg. Power (kW)**	**HP1 + HP2 Avg. Power (kW)**
0	Insufficient Data	Insufficient Data	Insufficient Data	6.3	2.8	9.1
5	Insufficient Data	Insufficient Data	Insufficient Data	4.7	2.4	7.1
10	Insufficient Data	Insufficient Data	Insufficient Data	3.2	1.9	5.1
15	2.6	1.7	4.3	2.3	1.6	3.9
20	2.2	1.6	3.8	1.7	1.6	3.2
25	1.4	1.3	2.7	1.3	1.2	2.5
30	1.0	1.0	1.9	0.9	0.9	1.9
35	0.5	0.7	1.2	0.5	0.7	1.2
40	0.4	0.5	0.8	0.4	0.4	0.8
45	0.2	0.3	0.5	0.3	0.3	0.6
50	0.1	0.1	0.2	0.2	0.1	0.3
55	0.1	0.0	0.1	0.1	0.1	0.2

The data shown in the table is also visualized in Figure C-1.

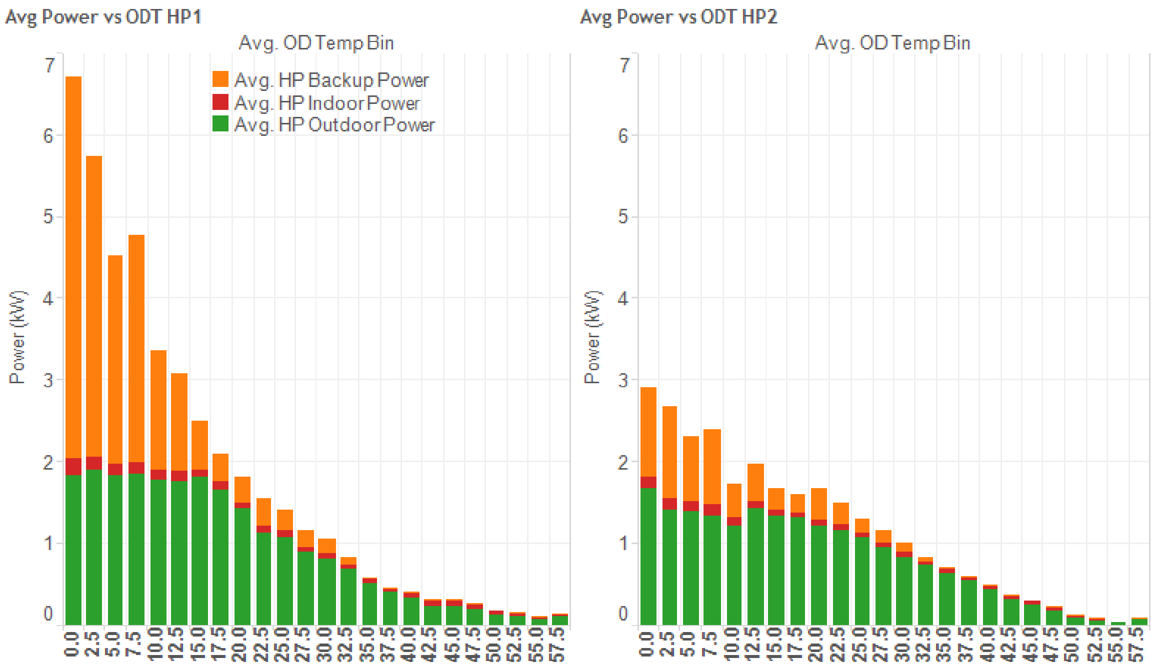

Figure C-1
Average Power of Heat Pump Outdoor Unit, Indoor Unit and Backup Heat vs. Temperature Bin for January and February, 2014

A period of unusually cold weather occurred in late January of 2014; the power and temperature for that week are shown in Figure C-2. An overnight low of -6.4°F was recorded on January 29th, and January 30th had an overnight low of 0.4°F. The downstairs system ran for several hours with extensive resistance heat both mornings, sustaining average hourly power draws of over 6.5 kW for approximately six hours each of those two mornings. The upstairs unit, as is typical for this house, used less power but still had sustained periods of high resistance heat usage during the morning.

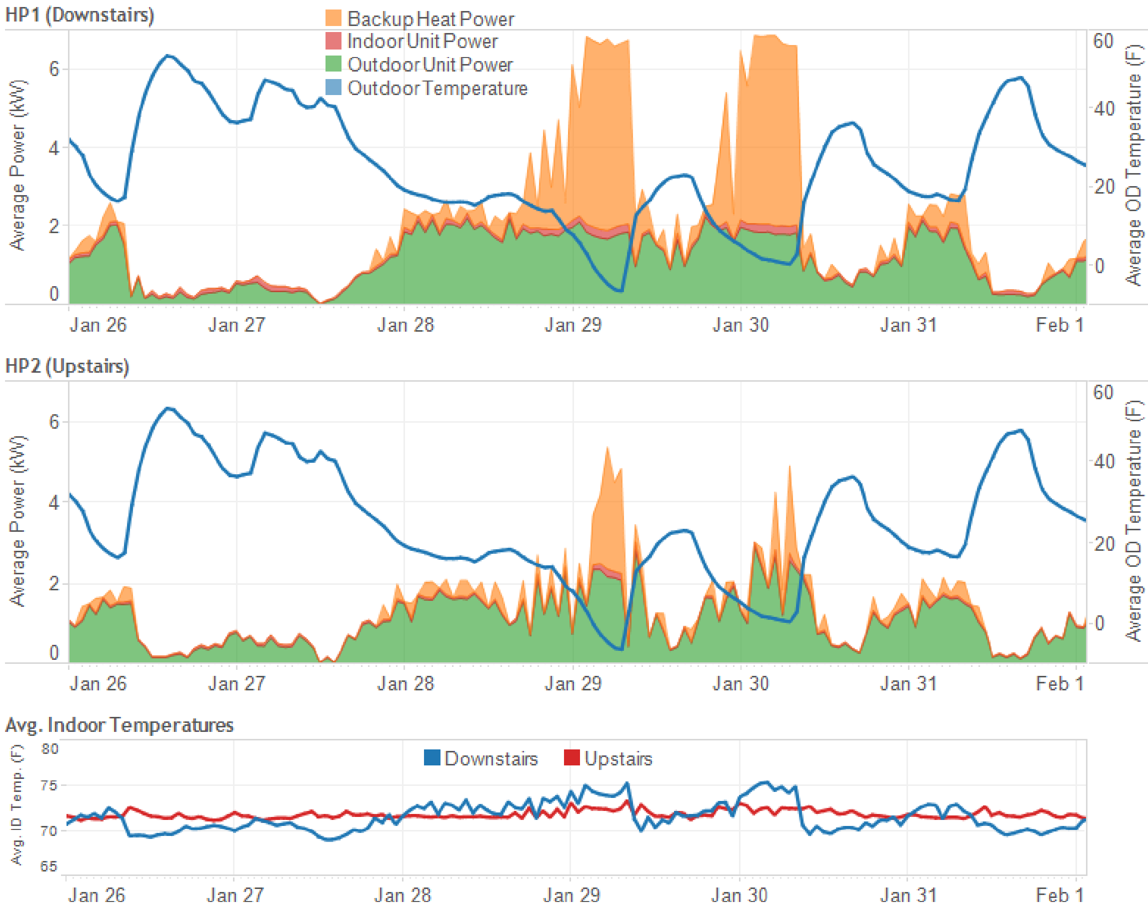

Figure C-2
One Week of Cold-Weather Heat Pump Operation

The period of peak heating overnight January 28-29 is shown for HP1 to illustrate how the system ran during the most extreme conditions. Figure C-3 shows the operation of the heat pump on morning of January 29, 2014 in one-minute time-steps; this figure shows that the electric resistance heat ran nearly continuously from approximately 1:30 AM, when the temperature outdoors approached 5°F, to 9:00 AM when outdoor temperature were rapidly increasing. The heat pump cycled periodically for defrosting throughout the night, and the backup heat ran if it was not already running during those times.

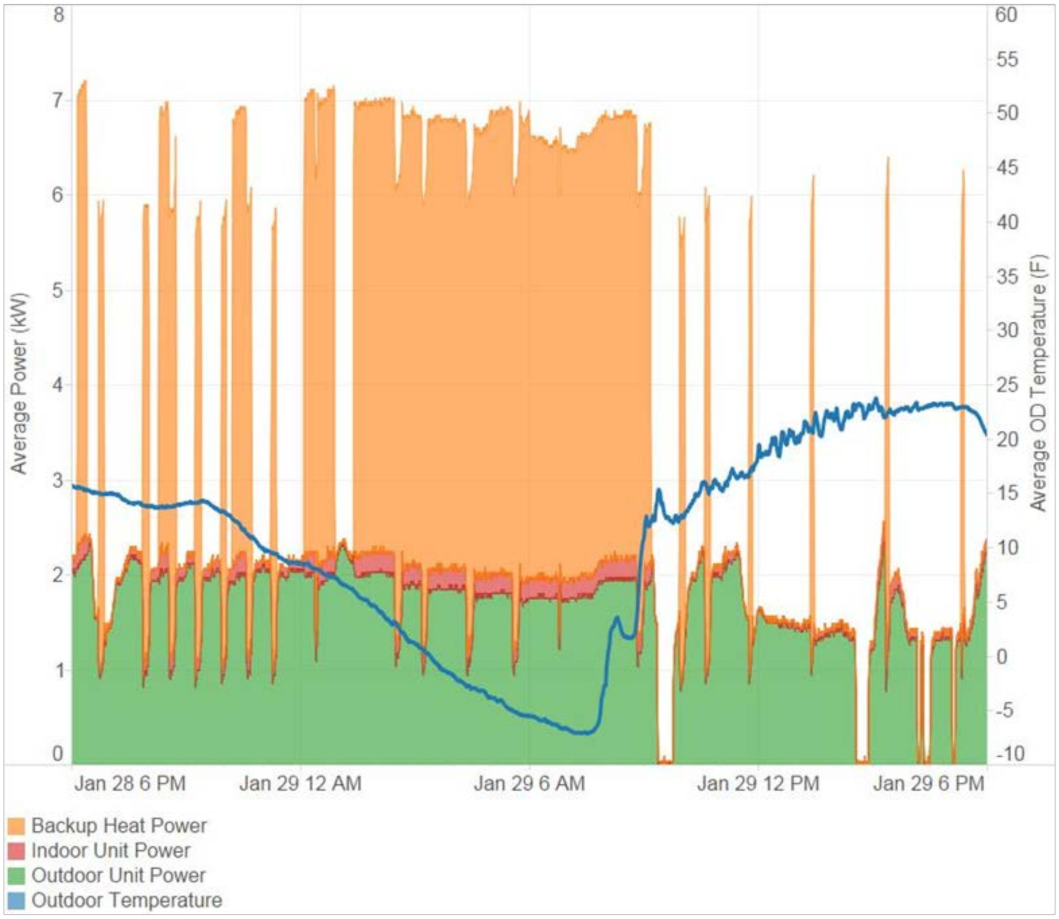

Figure C-3
Heat Pump and Backup Heat Operation during Coldest January Morning

The results for winter, 2013-14 will be fully analyzed and compared with past data in the final version of this report.

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