B O N N E V I L L E P O W E R A D M I N I S T R A T I O N

Residential Variable Capacity Heat Pump Field Study

Final Report

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A Report of BPA Energy Efficiency's Emerging Technologies Initiative

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An Emerging Technologies for Energy Efficiency Report

The study described in the following report was funded by the Bonneville Power Administration (BPA) to provide an assessment of the state of technology development and the potential for emerging technologies to increase the efficiency of electricity use. BPA is undertaking a multi-year effort to identify, assess, and develop emerging technologies with significant potential for contributing to efficient use of electric power resources in the Northwest.

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Abstract

The residential variable capacity heat pump (VCHP) study evaluated the field performance of variable capacity heat pumps to provide reliable estimates of energy use and savings. The field study measured the system performance at six houses over eight weeks. On top of the field study findings, the project developed the analytical tools necessary to evaluate VCHP energy use on an annual basis. A comparison of the field data to lab data show good agreement.

Overall, the field project showed that the VCHP systems are efficient and perform at higher levels than single speed systems. The modeled performance based on the field data suggests a 25-30% improvement. In contrast, the measurements showed that duct losses are larger for variable speed systems because they operate at lower airflows. The losses, however, are only ~5% greater. Taken as a whole, the VCHP and duct system provide improvements over a federal minimum single-speed system.

A further central finding is, just like single speed air-source heat pumps, the equipment size, auxiliary resistance heat controls, and thermostat setback behavior still matter. Although the VCHP is able to boost its heating output at lower temperatures, the equipment still needs to be properly sized for the house heating load.

Glossary of Acronyms and Abbreviations

BPA	Bonneville Power Administration
Btu	British thermal unit
Btu/ft^2F	British thermal units per square foot per degree Fahrenheit
CFM	Cubic Feet per Minute
COP	coefficient of performance
DE	distribution efficiency
EPRI	Electric Power Research Institute
ER	electric resistance
ft	feet
ft^2	square feet
HP	heat pump
HSPF	heating seasonal performance factor
HVAC	heating, ventilation, and air conditioning
kWh	kilowatt hours
kWh/DD	kilowatt hours per degree-day
kWh/yr	kilowatt hours per year
NEEA	Northwest Energy Efficiency Alliance
NWPCC	Northwest Power and Conservation Council
NWS	National Weather Service
OAT	Outdoor Air Temperature
OD	Outdoor
ODT	Outdoor thermostat
Pa	Pascal
PTCS	Performance Tested Comfort Systems
RAT	Return Air Temperature
RETC	Residential Energy Tax Credit (Oregon)
RFO	Request For Offers
RLF	Return leakage fraction
SAT	Supply Air Temperature
SCFM	Standard Cubic Feet per Minute (at standard temperature and pressure)
SEEM	Simplified Energy and Enthalpy Model
SLF	Supply leakage fraction
UA	The sum of the thermal transfer coefficient (U) times the area (A) of the components of the building. Can also include convective losses from infiltration (Btu/ft^2F).
VCHP	variable capacity heat pump

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Executive Summary

In recent years, heat pump manufacturers have expanded their line of variable capacity heat pumps (VCHPs) to integrate with ducted house heating systems. Currently available equipment carries heating seasonal performance factor (HSPF) ratings of 10-13 which are well above the minimum federal standard of HSPF 7.7 and the current baselines in the Pacific Northwest. Bonneville Power Administration (BPA) has offered, through its member utilities, a heat pump program for nearly ten years to provide incentives for air-source heat pumps that use less energy than the base case systems. This project seeks to evaluate the field performance of variable capacity heat pumps to provide reliable estimates of energy use and savings.

Variable capacity heat pumps, sometimes called variable speed heat pumps, consist of direct current compressor and air handler motors which change their speed and output in response to a building heating load. These are an evolution of single speed systems which only operate at a fixed output capacity. Both systems are connected to ducts to deliver the conditioned air to the house. Duct losses are known to affect overall system delivery efficiency and, in a further complication, vary with system air flow.

The VCHP field study measured the performance of six systems at six different houses in the Bend, OR area over an eight week period. The field work consisted of a day of intensive, on-site, one-time measurements, and then an eight week long data logging period recording VCHP behavior over a range of outside temperatures as they operated in real houses. In addition to the field study findings, the project developed the analytical tools necessary to evaluate VCHP energy use on an annual basis.

BPA has also funded a lab testing project to evaluate VCHP performance (Hunt 2013). A comparison of the field data to the lab data show good agreement. The biggest differences between the two arose from the controller mode settings. The lab measured performance in the manufacturer's "efficiency" mode while all the field sites were measured in "comfort" mode (the mode found at each site). The agreement between the two datasets gives us confidence in both measurements.

Overall, the field project showed that the VCHP systems are efficient and perform at higher levels than single speed systems. The modeled performance based on the field data suggests a 25-30% improvement. In contrast, the measurements showed that duct losses are larger for variable speed systems because they operate at lower airflows. The losses, however, are only ~5% greater. Taken as a whole, the VCHP and duct system provide improvements over a federal minimum single-speed system.

A further central finding is that, just like single speed air-source heat pumps, the equipment size (heating output), auxiliary resistance heat controls, and thermostat setback behavior still matter. Although the VCHP is able to boost its heating output at lower temperatures, the equipment still needs to be properly sized for the house heating load. An undersized VCHP will still resort to auxiliary resistance heat just like a fixed speed system. The issue is further compounded when the occupants use a thermostat setback. In two of the six cases, the morning recovery period was a clear driver of resistance heating use. A better controller could anticipate the recovery sooner to avoid the low efficiency resistance heat. Complicating the issue further is the control (or lack thereof) of the resistance heat based on outside temperature. Best practices dictate locking out resistance heat above 35°F (Regional Technical Forum 2007). This was not done at several of the field sites which then showed more resistance heating use. These issues are important in that if resistance heat is not carefully controlled, it will erode the efficiency gains made by the variable speed system.

In addition to the field measurements, the project lays the groundwork for our future understanding of variable speed systems. The investigation of duct losses is crucial to understanding overall heating performance. The framework developed in this project can be leveraged for all future work. Next, the project developed a performance curve and engineering model which was added to the Simple Energy Enthalpy Model (SEEM). The model will allow accurate predictions of VCHP heating energy use for simulated buildings.

Further work is needed to fully assess the cost effectiveness and energy savings potential of the VCHP. Both the lab and field findings demonstrate these VCHPs perform better than the baseline systems. The next step is to simulate energy use over a full year in a variety of climates and house types in the Pacific Northwest. The metered sites can be used to further calibrate the simulation as needed. In general, the VCHP systems offer increased capacity at lower outdoor temperatures which can provide large energy savings. An important alternative to consider to a VCHP installation is simply upsizing the planned single speed system. Where a 3-ton system may be installed, it would be useful to evaluate the savings and cost effectiveness of a 4-ton system. A

simulation and cost analysis can examine the full suite of base case and efficient case scenarios and provide estimates of energy savings and relative cost-effectiveness.

1 Introduction

Bonneville Power Administration (BPA) has offered, through its member utilities, a heat pump program for nearly 10 years. This program, called Performance Tested Comfort Systems (PTCS), offers incentives for installation of high efficiency air-source heat pumps. “High efficiency” has typically meant a heat pump with HSPF at least 0.5 points above the federal minimum; installation requirements have included control settings, indoor coil airflow testing, and recording of refrigerant operating pressures.

In recent years, various manufacturers have developed direct current compressors that can be used in ductless or ducted settings. Nominal HSPFs for some of the ducted systems have been listed at 10 and well above. BPA would like to assess the performance of these systems to understand how much energy they may use. Duct losses also affect overall system delivery efficiency. BPA has offered a provisional incentive for this type of product pending further field and laboratory testing. The project described in this report was designed to add to existing laboratory and field results to allow a more informed evaluation of the product.

Ecotope, Inc. has performed or overseen studies of heat pump and duct performance in the Pacific Northwest since the early 1990s. Examples of this work are Olson et al. (1993), Palmiter and Francisco (1997), Francisco and Palmiter (2004) and Reichmuth et al. (2006). Much of this research fed development of the Simplified Energy and Enthalpy Model (SEEM – Ecotope 2013), which is now used throughout the Northwest to model residential building performance. Ecotope responded to the Request for Offer in August 2012, was selected to carry out the work, and then engaged in subsequent discussion with BPA about the research. The intent was to agree upon a Scope of Work and perform field testing during the 2012-2013 winter.

2 Methods

The primary goals of the field testing were to take sufficient detailed measurements over a range of environmental conditions that would dictate the heating load so a performance map could be developed. This map could be compared with laboratory data, which had recently been collected via work funded by the Electric Power Research Institute (EPRI) and, further, since measurements of duct performance would also be taken, the basic map could be revised based on a duct model that would be developed for each site. The latter step is critical since the heat pump, regardless of its inherent efficiency, is connected to a distribution system that will incur various conductive and air leakage losses.

The field data collection would be divided into two categories: one-time measurements such as a house and duct thermal audit, blower door test, duct leakage test, and air handler profile (power versus airflow and static pressure); and long-term measurements of electricity usage and operating temperatures. Ecotope’s past experience with combining these measurement categories convinced them a similar approach would work on this type of equipment.

2.1 Project Development and Planning

BPA’s request for offers (RFO) was released in late July, 2012. The original RFO included language about VCHPs as a PTCS measure and suggested that work to be completed under the RFO would inform a revised field-testing procedure for both installers and QA agents. This aspect of the work was eventually dropped from the contract and work focused only on field performance and integration of field results with EPRI’s lab and site testing, the latter done at Campbell Creek, TN.

Ecotope engaged in discussions with BPA and an Advisory Committee made up of interested parties such as electric utility and HVAC industry personnel during this time in efforts to refine the field testing process. Various suggestions were advanced and an agreement was finally reached in early 2013 to test 8 field sites.

2.2 Field Work

2.2.1 Site Selection

Because of the warm winter conditions in early 2013, Ecotope suggested to BPA that efforts be focused on Heating Climate Zone 2, which historically has between 6,001 and 7,499 Heating Degree Days (base 65°F), and which is found in central and eastern Oregon and Washington and much of Idaho**.** The most populous cluster of sites was in central Oregon, so that is where recruiting was centered. A recruiting script was developed and about 20 sites in central Oregon were called, with prospects found from BPA's list of sites that included both PTCS and Oregon Residential Energy Tax Credit (RETC) installations; this list included sites served by both public and private electric utilities. Ten sites agreed to the testing but eventually only 6 were able to be instrumented after closer investigation of the actual equipment at the site. The remaining four sites were found to have an incorrect or ambiguous entry in the PTCS database; three of these sites were filtered during recruiting and one, where the recruiter could not fully verify the system type, was visited by the field team but was found to have the wrong heat pump.

Table 1 summarizes salient characteristics of the test sites. All but one were site built homes. All but site 91004 had some of their ducts located in unheated buffer spaces such as crawlspaces or attics. (Table 2 details duct locations.) Site 91004 was included in the field test so that a "perfect ducts"[1] case could be evaluated. The sites comprised a range of nominal heat pump sizes from 2-4 tons and offered a range of duct leakage. In such a small survey of sites, it was fortunate to get a wide range of characteristics to study.

Also, heat pumps were set up by the original installer to run in COMFORT mode. The operating mode is mentioned because it differs from the mode that was investigated by EPRI during laboratory testing. A conversation with the heat pump manufacturer found that COMFORT mode is the default mode programmed into the main system controller so this is likely the mode that most of these heat pumps operate in during normal conditions.

Table 1. Field Site Characteristics

Site ID	Location	Occupants	House Type	Heat Pump Size (tons)	Duct Location (supply/return)
91001	Powell Butte, OR	2	Site built	3	crawl / attic
91002	Powell Butte, OR	2	Manufactured Home	2	crawl and belly/ inside
91004	Bend, OR	2	Site built	3	inside
91005	Bend, OR	2	Site built	3	crawl / attic
91009	Redmond, OR	2	Site built	4	crawl / attic
91010	Redmond, OR	2	Site built	4	crawl / attic

2.2.2 Metering Design

The metering plan included both one-time measurements and collection of heat pump operating data (electricity and temperature measurements) that extended over an approximately 8 week period. As outlined above, there was considerable discussion about the nature of the data logging interval and the channels to be collected. Initially, there were to be two types of sites, "basic" and "detailed", but because of project timing, all sites became "detailed"; that is, data were collected on 5 minute logging intervals (versus hourly). Temperatures of supply and return air were collected and airflow was determined through a combination of one-time tests and logging of air handler power/energy.

All longer-term data were collected with an Onset Computing Corp. U30 data logger equipped with integral 3G connectivity to allow near-real-time remote access to data. The data logger and most of the associated wires and hardware were installed adjacent to the main electrical panel (which was in the garage at most sites). Ecotope confirmed at each site that there was adequate 3G coverage before proceeding with the installation.

[1] "Perfect ducts" implies there are no duct losses. For ducts installed inside the conditioned envelope of the house, any air leakage or conductive losses are transferred to the conditioned space of the house and not outside or to a buffer space. Since this is useful heating or cooling supplied to the house (although not always in the desired location), we say there are no duct losses or that the ducts are "perfect" distributors of the heating or cooling.

The channels that were measured at each of the six sites were:

- Air handler power/energy (along with one-time measurement of system airflows and supply/return static pressures, the air handler power could be used to construct a fan curve for each site and also to construct duct leakage flow equations)
- Outdoor unit power/energy (compressor and outdoor fan combined in this channel)
- Indoor unit auxiliary heat (electric resistance) power/energy
- Outdoor temperature
- Temperature in supply and return plenum. (This was a single-point measurement and was checked against a Fluke 52 thermocouple thermometer.)
- Supply duct buffer space temperature (to allow finer tuning of the SEEM duct model; see below)
- Main living space temperature (via an independent temperature logger)

Because of the relative brevity of the data logging period, Ecotope did not use semi-automated error-checking for these data streams. Instead, the data were looked at periodically (typically every other day) via Onset's Hobolink web interface to make sure values looked to be within range. Raw data were transferred from Hobolink into comma separated variable files and then fed into the R and Stata software packages for analysis.

2.3 Measurement Uncertainty Analysis

When it was decided that all sites would receive detailed metering, it became of utmost important to ensure the measurements of air handler flow, coil entering (RAT – return air temperature) and leaving (SAT – supply air temperature) temperature and system power were accurate. These measurements are used directly to calculate thermal output (system capacity) over a desired monitoring interval (in this case, every 5 minutes). The ratio of system capacity (converted from Btu/hr to kW) to average 5 -minute input power (to run the compressor, outdoor fan, and air handler) is the COP.

Systematic error checking was used for all measurements. The power measurements were checked with a true-power multimeter for comparison with the pulse counts employed by the data logging system. If an issue was found, wiring and CT placement were checked and corrected. Air handler flow was measured at at least three points along with air handler true power and supply and return plenum static pressure. This process allowed an overall curve of air handler flow to be created (Section 3.1.4). Air handler measurements were also compared with the manufacturer's information to cross-check that they were within close agreement.

Ecotope's past experience in testing duct efficiency with the coheating method (for example, Olson et al. (1993)) suggested the most problematic measurements would be that of temperature, especially supply temperature. A primary issue here, other than the type of/accuracy of sensor, is the position of the sensor. In the past, most of the problems measuring supply temperature have occurred when working with electric resistance furnaces (or heat pumps operating in electric furnace mode). This is due to both radiant gains from an element (which can bias the supply temperature high) or to the presence of "cold spots" (where the supply air is not completely mixed). Ecotope's experience in measuring supply temperature when the indoor coil is providing most of the heat (via the compression cycle) has mostly been positive, with very little variation across the coil and therefore relatively uniform temperature distribution in the supply plenum.

In this study, during the data logger installation, temperatures measured by the Onset sensors were checked with a recently calibrated Fluke 52 digital thermocouple thermometer and Omega steel-jacketed Type K thermocouples (which Ecotope customarily uses in all of its field work). Because both supply and return temperatures were measured at a single point, the cross-check was taken to determine if there was significant variation across the plenum cross-section. Given the amount of air moved, we generally found very little variation in the return plenum temperature. The supply plenum temperature was also typically very consistent across the plenum cross-section.

We performed an additional cross-check of supply air temperature. We looked at the amount of temperature change corresponding to adding a known amount of electrical resistance heat to the airstream. We used this only as a rough check of supply sensor accuracy given our experience with electric furnaces. Mostly we were concerned that our measurements could be biased low and therefore we would be underestimating thermal output and dragging down COP.

Figure 1 shows a typical plot of supply air temperature (SAT) as a function of electric resistance heat (ER) added over the given time interval (in this case, 30 minutes, so that we minimize transient effects). The points are labeled with the volumetric air handler flow to allow better interpretation. The values plotted here are pooled for all of the 3 ton capacity sites. What we are trying to see is whether the SAT is obviously biased low. This would appear as SAT values which are significantly depressed versus the expected rise.

The general calorimetric equation for heat added to an airstream is:

$Q = \Delta T$ * CFM * (conversion factors for density of air and altitude, about 0.96 overall at these sites)

CFM is the volumetric flow, as shown in Figure 1

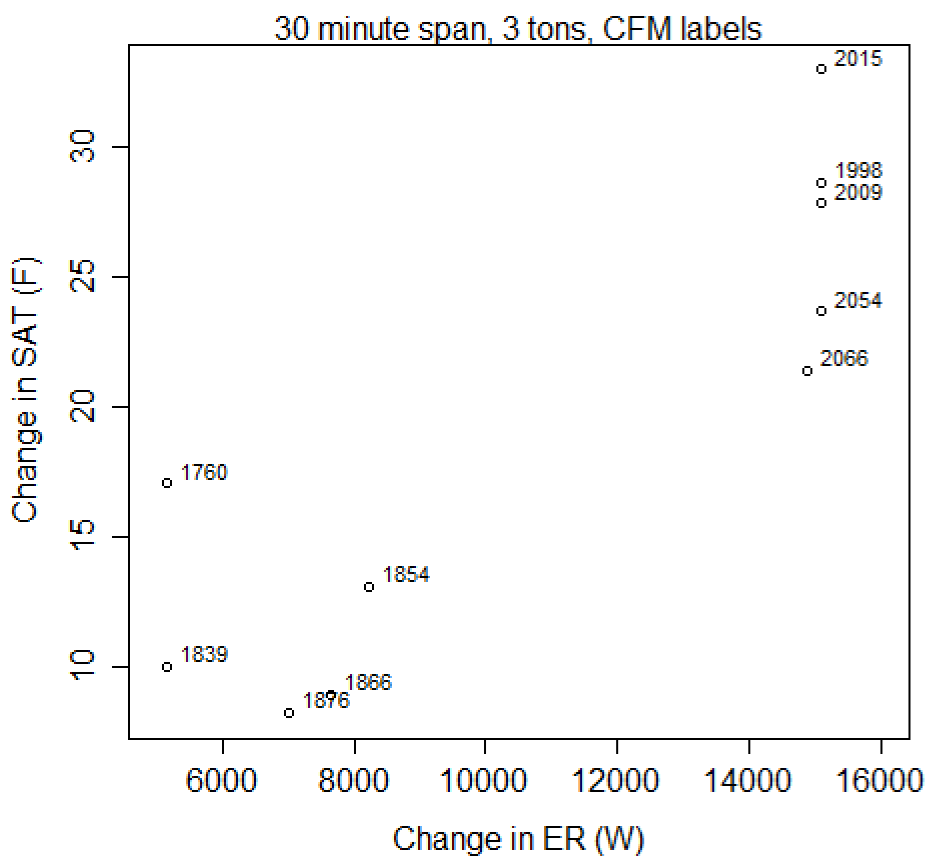
So $\Delta T = Q/(0.96 \text{ (CFM)})$

If we have 2000 CFM of air moving through the system and add 15,000 Watts of electric resistance heat, we would expect to see an increase of roughly:

(15,000 W * 3.414 Btu/hr-W)/0.96*2000 = 26.6 °F

The plotted values range from a bit under 25° F up to a bit over 30° F and we therefore concluded we did not see a systematic bias that would cause concern.

Figure 1. Comparison of Supply Air Temperature Measurements and ER



The points further left in the graph indicate as well that earlier in the 30 minute period, we had SAT increases that scaled as expected when fewer Watts were activated (initial sequencing of the system).

3 Findings

3.1 House, Heat Pump, and Duct Characteristics

The heat pump interacts directly with the house and duct system so understanding them is important to interpreting any metered findings. The characteristics of the house, heat pump, and ducts are summarized in this section. The field technicians collected the characteristics onsite as part of the intensive, one day site visit.

3.1.1 House Characteristics

Table 2 displays summary information on the house size, insulation, air tightness, and overall heat loss rates. Much as there are a range of house types, there are a range of house sizes and heat loss rates.

Table 2. House Characteristics

Site ID	Heated Floor Area (ft^2)	Shell UA (BTU/hr°F)	Peak Infiltration Heat Loss Rate* (BTU/hr°F)	Total Heat Loss Rate (BTU/hr°F)	Blower door leakage at 50 Pa (SCFM)	Average Heating Season Infiltration† (ACH)	# Floors
91001	2694	692	204	896	4605	0.34	1
91002	1394	227	56	283	1326	0.22	1
91004	2663	864	262	1126	3849	0.46	2
91005	2515	421	159	580	2501	0.3	1
91009	1860	378	124	502	2178	0.33	1
91010	2660	509	123	632	1580	0.18	1

*Infiltration under peak design conditions as calculated by SEEM. Note that average heating season infiltration is less than the peak.
†Average infiltration over the heating season as calculated by SEEM using the tested leakage. Includes all sources: natural infiltration, duct leakage, and mechanical ventilation.

3.1.2 Heat Pump Characteristics

Table 3 shows the characteristics of the heat pumps at each site. The table lists the outdoor model number as well as the indoor unit it was paired with for installation. For each combination, there is a different HSPF rating which is also shown. Generally, the larger indoor coils and air handlers are paired with the larger outdoor units. Due to the physics of heat transfer, larger coils, both indoors and out, lead to more efficient operation.

Some of the sites went through the PTCS programs for heat pump configuration. Interestingly, the sites that got PTCS heat pump incentives were found to not have their outdoor lockout controls set up. A key element of the PTCS program is to lock out use of electric resistance heat when the outdoor temperature is above 35°F, thereby using the compression cycle to provide all the heat. This finding likely represents a failure in the PTCS quality assurance program.

Although discussed previously, it is worth mentioning again that the controllers for all the field sites were set to operate in COMFORT mode. An email exchange with the manufacturer confirmed that is the default mode when shipped from the factory. The manufacturer also confirmed that one of the main differences between COMFORT and EFFICIENCY mode is the airflow. In practice, EFFICIENCY mode operates with higher system airflows and therefore delivers slightly lower temperature supply air to the house. COMFORT mode means maximum delivered airflow is usually in the 315-350 CFM/ton range compared with 400 CFM/ton when the system is running at full capacity EFFICIENCY mode.

Table 3. Heat Pump Characteristics

Site ID	Heat Pump Size (tons)	Manufacturer	Outdoor Unit Model #	Indoor Unit Model #	HSPF	ODT setting* (°F)	PTCS?†
91001	3	Bryant	280ANV036-A	FE4ANB006	13	not set	Yes
91002	2	Carrier	25VNA024A	FE4ANF003	10.5	35	No
91004	3	Bryant	280ANV036-A	FE4ANF005	12.4	not set	Yes
91005	3	Bryant	280ANV036-A	FE4ANB006	13	not set	Yes
91009	4	Carrier	25VNA048A	FE4ANB006	12.5	35	No
91010	4	Carrier	25VNA048A	FE4ANB006	12.5	35	No

* "ODT" means "outdoor thermostat". Where a number is shown, this is the lockout temperature for the electric resistance elements; the elements should not operate above this temperature unless the system is undergoing a defrost cycle or the homeowner switches into emergency heat mode at the thermostat.

†"Yes" means a PTCS incentive was paid for HP set-up.

3.1.3 Duct Characteristics

The duct characteristics are important to calculating the distribution efficiency. When located outside the conditioned envelope, ducts can lose heat through conduction to buffer spaces. The surface area and insulation values shown in Table 4 are the determining components in the heat conduction losses. The table shows that most of the houses have considerable duct surface area. For comparison, a more typical house in the Northwest (one used as a prototype in modeling for the Power Plan and the Regional Technical Forum) may have 250-300ft^2 of supply duct area and 80-100ft^2 of return duct area. As will be shown later, the large duct area leads to significant efficiency losses for these houses.

The average insulation values are only slightly lower than one would expect for a well-insulated system. Importantly, the insulation values are not the nominal values but the actual R-values (Palmiter and Kruse 2006). Further, they are averaged over all the duct types. Typically, most of the ducts were of the flex-duct type with very little sheet metal duct present. The insulation value for site 91002 is high since the supply ducts are buried in the belly blanket insulation of the manufactured house. Additionally, for that site, the return air flows through the interior of the house and not through ducts so there are no return duct characteristics. For site 91004, with all interior ducts, the insulation R-value and surface area are not applicable.

Table 4. Duct Characteristics – Location, Surface Area, and Insulation

Site ID	Supply Ducts			Return Ducts		
	Location	Surface Area (ft^2)	Insulation (average R-value)	Location	Surface Area (ft^2)	Insulation (average R-value)
91001	Crawl	524	4.0	Attic	71	7.8
91002	Belly/Crawl	255	22.0	n/a	n/a	n/a
91004	Interior	n/a	n/a	Interior	n/a	n/a
91005	Crawl	628	7.1	Attic	81	7.8
91009	Crawl	509	5.5	Attic	204	7.8
91010	Crawl	825	6.9	Attic	253	7.8

Table 5 shows crucial information on the duct tightness characteristics. The duct leakage, air handler, and static pressure measurements are critical in this analysis because they allow (combined with a measurement of duct system size and R-value) a full model of duct distribution efficiency to be developed for use in our field measurements and in SEEM. Multiple air handler flows (and accompanying power) were measured so that an actual air handler curve could be constructed and compared with the manufacturer's values. Once this curve is constructed, static pressures in both return and supply side can be imputed at other flows and supply and return leakage fractions can be determined as well. Using known relationships for airflow, we can calculate the leakage at any flow and pressure once we have measured it at the standard testing pressures of 50 Pa and 25 Pa.

Table 5. Duct Characteristics – Airflow and Leakage

Site ID	Both sides duct leak to out at 50 Pa* (SCFM)	Supply duct leak to out at 50 Pa* (SCFM)	Reference‡ Air Handler Flow (CFM, SCFM)*	Reference‡ supply static pressure (Pa)	Reference‡ return static pressure (Pa)	PTCS?†
91001	231***	151***	943, 1061	18	-84.5	Yes
91002	276	275	688, 774	16	-21	No
91004	n/a	n/a	889, 1000	21.5	-48	No
91005	273	131	924, 1040	26	-72.5	Yes
91009	329	208	1340, 1395	50	-139	No
91010	148	131	1271, 1430	116	-158	No

* Leakage and air handler flow results corrected to standard air (68°F and 1 atmosphere). The elevation of houses in the Bend area (about 3,000 ft above sea level) results in lower air density. More specifically, the density is about 88.9% of the density of standard air. The air handler flow values show both the local CFM and the standard SCFM.
† "Yes" means a PTCS incentive was paid for duct sealing
‡ "Reference" airflow corresponds to the supply and return static pressure measurements shown in the table. Typically this airflow represents the highest flow that could be attained using the User Interface (thermostat); this measurement was taken to make sure the air handler was not working against an extreme external static pressure (above 200 Pa) at its highest flow. No adverse static pressure conditions were found. All of these systems were set up by the installer in COMFORT mode (so maximum flows typically average 325-350 CFM/nominal ton of capacity).
***These measurements taken at 45 Pa; not yet corrected to 50 Pa.

3.1.4 System Air Handler Flow

Figure 2 shows the measurement process and result of constructing an air handler curve specific to an individual site. At site 91009, which has a 4-ton outdoor unit, the field team measured the air handler flow with a flow plate and the corresponding power input at four different points. From the fan affinity laws, we know that the airflow is proportional to the cubed root of the input power. Therefore, we fit a power curve to the measured data points. Ideally, the exponent in the curve fit is equal to 1/3. Field measurements are rarely made under ideal conditions so we expect the exponent to deviate somewhat from 1/3. The coefficient of the curve fit represents the external static pressure of the duct system. As the static pressure increases, more input power is needed to move the same amount of air so the coefficient will decrease. This is key because although we expect all the sites to have the same exponent, we expect the coefficients to differ. Different coefficients represent variations in installations and duct layouts. In essence, a larger coefficient means there is less external static pressure inside any given duct system (less resistance to airflow). Figure 3 shows the curve fits for all of our sites. The measured points are not plotted but the points fall along the fitted lines nicely because of stable behavior of the 1/3 power law.

Figure 2. Airflow & Fan Power: Site 91009

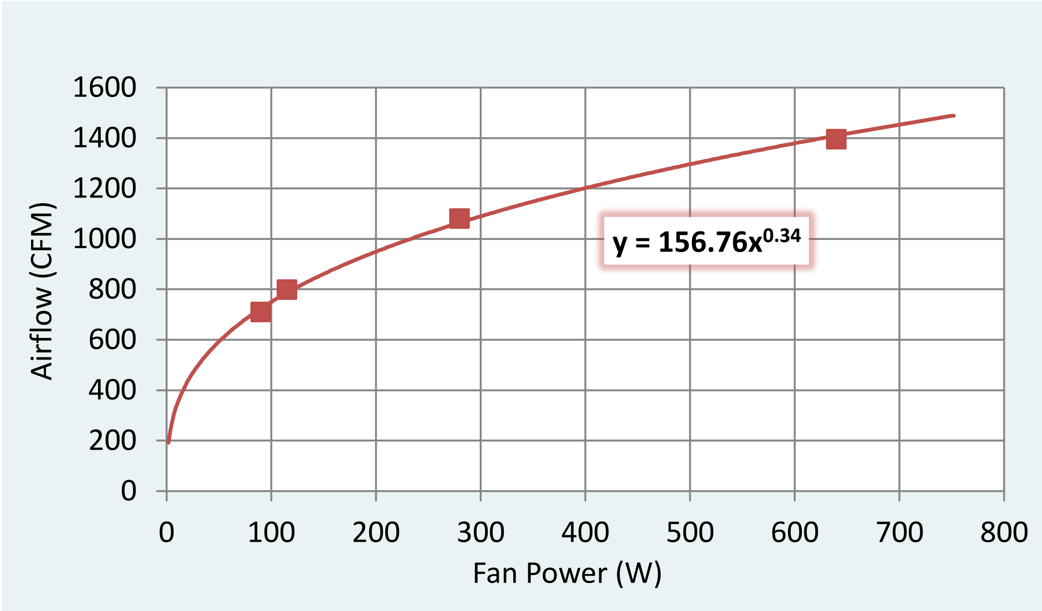
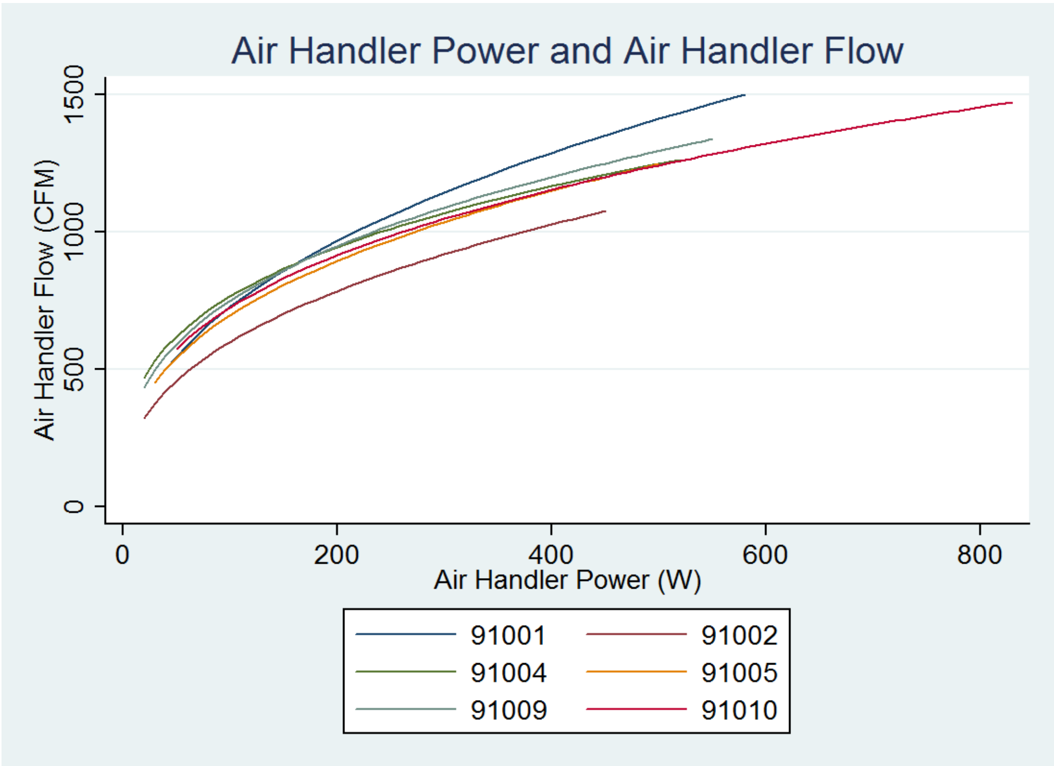


Figure 3. Airflow & Fan Power: All Sites

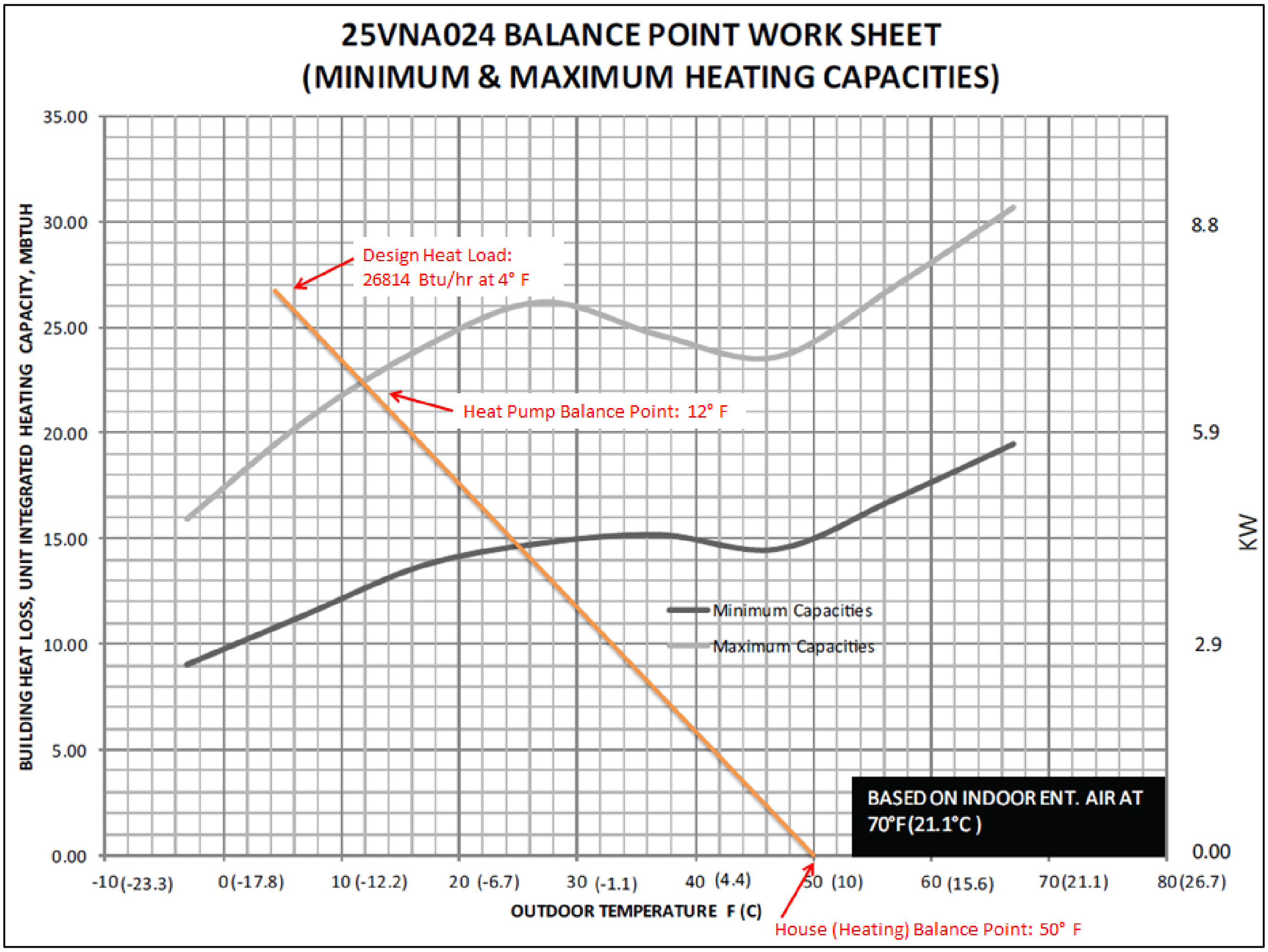


3.2 House Heat Loss Rate and Heat Pump Size

The heat loss rate at design conditions, including the contribution from all sources of air infiltration/exfiltration (as determined by SEEM's infiltration model), was calculated so that a heat pump balance point temperature could be determined. The *heat pump balance point temperature* is the lowest outdoor temperature at which the capacity of the heat pump can be expected to meet the heating load. Also, the contribution from duct losses must be included since the duct effect will increase the peak heating load (except in the case of Site 91004, where all ducts are included in the building thermal envelope).

Figure 4 shows the graphical representation of the balance point for Site 91002. One end of the orange line is anchored at upper left, at the outdoor design temperature: 4° F for Bend, OR based on ASHRAE (1989). The other end is anchored at the *building balance point temperature,* which is calculated in SEEM and which is the outdoor temperature above which no heat must be added to the building to keep it at or above an indoor temperature of 70°F. Better insulated buildings can meet this comfort set point with internal and solar gains at outdoor temperatures in the mid to high 50°s F while more poorly insulated buildings typically need heat at outdoor temperatures below 60-65°F.

Figure 4. Example Heat Pump Balance Point Calculation



Determining balance points for the variable speed products is more intriguing than usual since the manufacturer lists both a "maximum" and "minimum" output curve. However, one would expect that in colder winter conditions, such as those that occur near design temperature, the system would be operating on the maximum output curve. In another sense, it might not matter what capacity the system was operating under as long as the compression cycle could meet the heating load without requiring auxiliary heat.

Figure 4 shows the balance point graphs for two of the 3 ton sites: 91001 and 91004. Site 91001 has the highest design heating load and warmest balance point in this group of homes; the heat pump here is obviously undersized (literally "off the chart"). If a 4 ton unit had been selected for this home, the heat pump balance point would have been 30° F. Site 91004 has all-internal ducts but the heat pump is still undersized, with a calculated heat pump balance point of 36° F. Using a 4 ton heat pump at this site would have resulted in a heat pump balance point of 22° F.

Figure 5. Undersized 3 Ton Heat Pumps

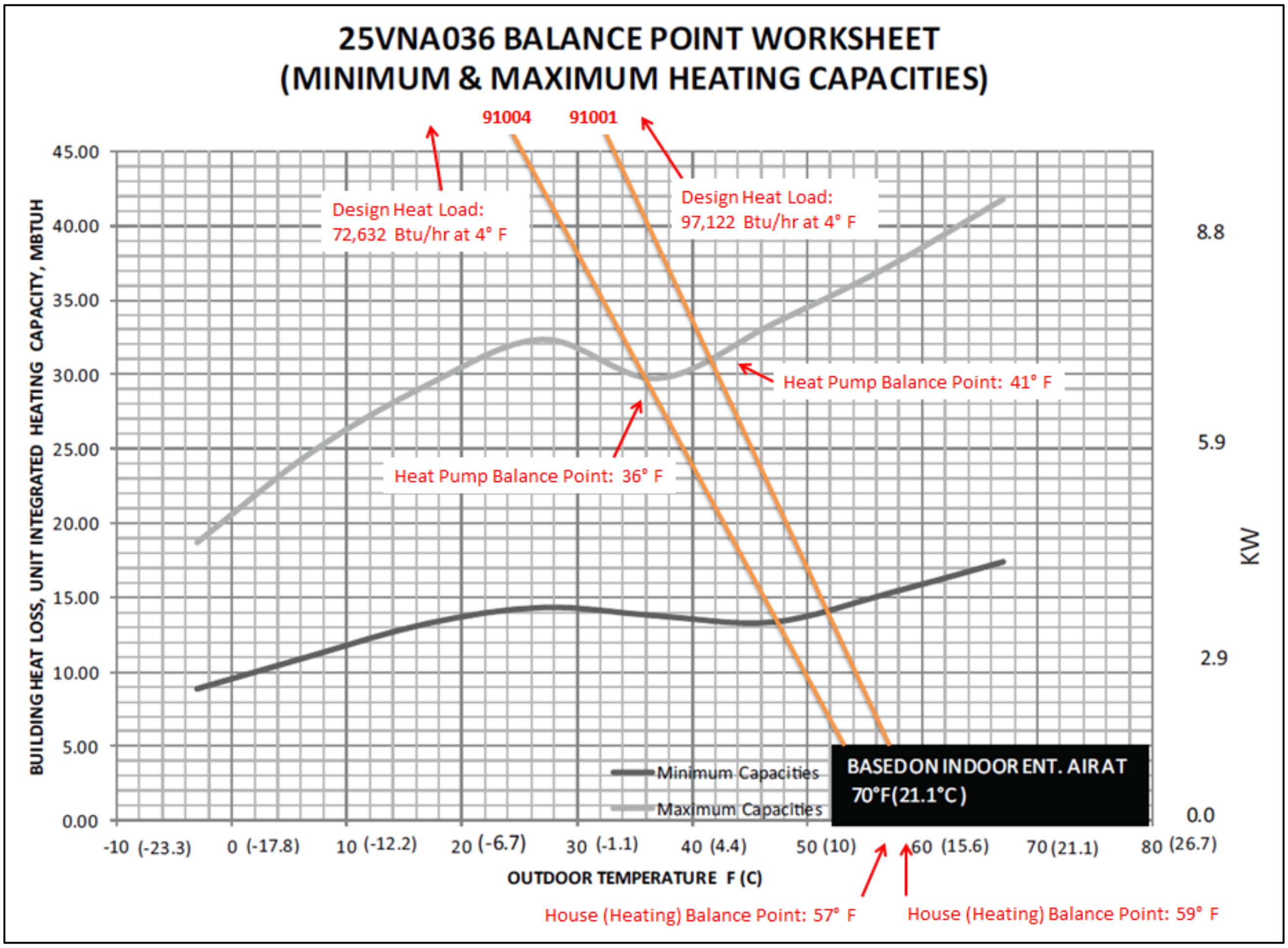


Table 6 shows the heat pump balance point temperatures for all sites. In some cases, the balance point is much colder than the regionally-recommended value of 30°F (Regional Technical Forum 2007). However, it is apparent that even with this product, calculating a balance point is still an important exercise if one wants to maximize compressor-only heating. The heat pump sizing has real implications for energy use. As is shown in detail later, site 91004 is one of the two highest users of resistance heat in the study. This is despite the VCHP's ability to increase output at lower ambient temperatures. Overall, the heat pump is simply sized too small to heat the house as efficiently as it might at this site.

Table 6. Heat Pump Balance Point Inputs and Result*

Site ID	Peak total UA† (BTU/hr °F)	DHL base‡ (BTU/hr)	DHL w/DE†† (Btu/hr)	HP Balance Point (°F)	Heat Pump Size (tons)
91001	896	63,638	97,122	41	3
91002	283	19,260	26,814	12	2
91004	1126	72,632	72,632	36	3
91005	580	40,996	69,719	26	3
91009	502	37,452	72,367	20	4
91010	632	42593	66391	17	4

* All Design Heating Load (DHL) calculations based on 4° F design temperature.
†Shell plus infiltration heat loss at heating design temperature
‡Design Heating Load without duct losses at the design temperature
††Design Heating Load with duct losses included

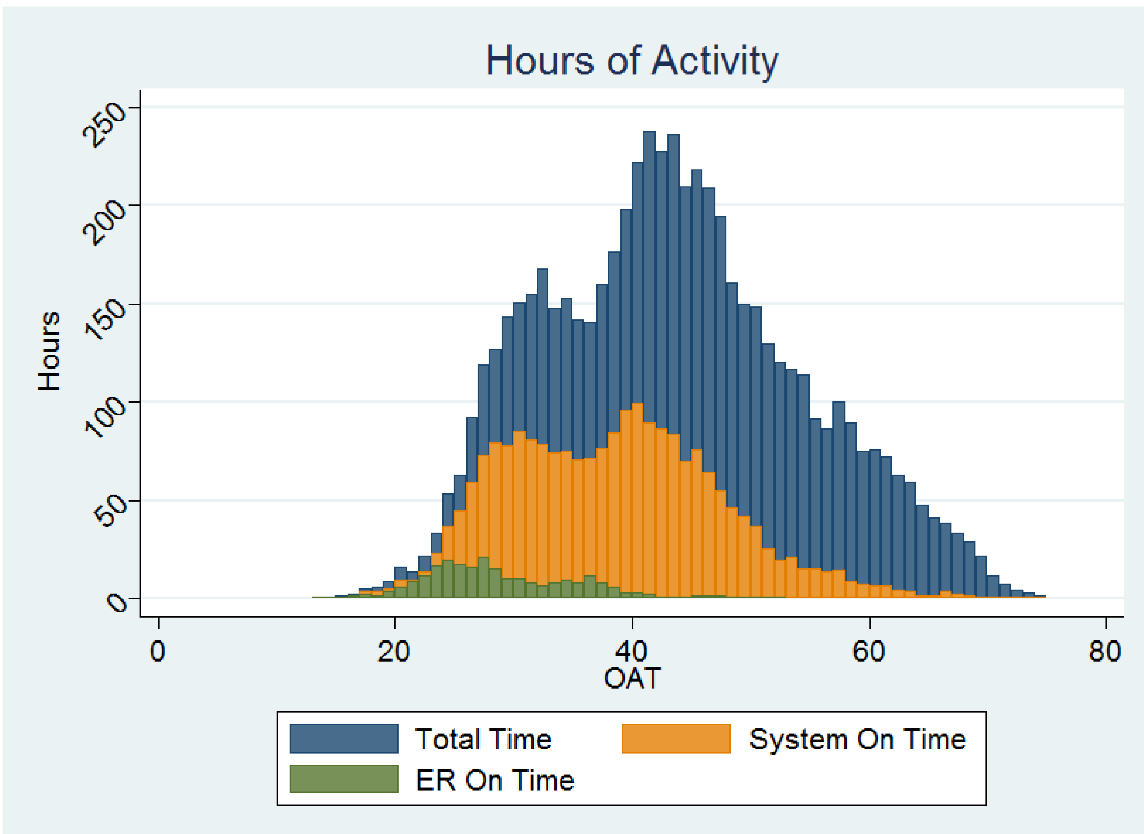
3.3 Metered Findings

This section discusses the findings from the continuous data logging systems installed at each site. The data loggers recorded approximately 6-8 weeks of heat pump operation at the sites.

3.3.1 Outdoor Temperature

During the field monitoring period, the outdoor temperatures in the Bend, Oregon region ranged from 15° F to 70° F. Figure 6 shows the total number of hours in a given temperature range. The figure displays all the data combined from all six sites. The graph also shows the outside temperature profile when the heat pump system was running and when just the ER heat was running. As expected, the ER heat runtime is heavily skewed to the lowest temperatures where it is needed to make up extra heating capacity. Interestingly, the distribution of compressor on-time is relatively even across the total hours. This is likely due to the compressor being able to run at lower outputs when there is less heating demand at the warmer temperatures.

Figure 6. Outside Temperature during Project



3.3.2 Metered Heating Energy Use

The entire field monitoring period lasted approximately eight weeks. For five of those weeks, all of the data loggers were installed and operating simultaneously at every site. All the sites were within 30 miles of one another meaning that they all experience similar weather. The weather similarity is convenient because it allows us to more directly compare the energy use across sites without manipulating the data using heating regression models. Table 7 shows that the measured average outdoor temperatures at each site differ by no more than 3.6°F. Since we expect some differences in temperature due to measurement uncertainties and thermistor placement, this constitutes good agreement and shows the weather across the sites is nearly identical.

Table 7 shows the measured energy use for each of the input power channels plus a combined value. We metered the outdoor unit, air handler, and ER auxiliary heat separately. The outdoor unit channel consists of the compressor and outdoor fan. The air handler value consists of the time when the air handler was running in heating mode and also in air circulation mode. The ER value shows the auxiliary heating energy used. The sum of those three channels exceeds the total input listed because the input total was restricted to exclude the periods of fan air circulation. Total input applies to heating periods only. Due to different house UA values, we expect the total input energy to differ; however, two sites show heavy use of the backup resistance heat which dramatically increases heating energy use at those sites.

Table 7. Metered Temperature and Energy – Common Five Week Period

Site ID	Average Outdoor Temperature (F)	Outdoor Unit (kWh)	Air Handler (kWh)	ER Auxiliary (kWh)	Total Input (kWh)	Metered Time Span (hrs)
91001	38.4	590	45	100	716	834
91002	39.3	221	30	56	277	834
91004	40.9	698	58	465	1185	834
91005	39.4	310	32	547	861	834
91009	38.2	591	35	58	662	834
91010	37.3	525	112	103	706	834

To examine the energy use in more detail, we turn to Table 8 which shows the fraction of total heating output provided by the ER auxiliary system. For all but two sites, we see the fraction is less than 10%. For sites 91004 and 91005, the sites with largest amount of total ER energy use, we see the fraction is large. For site 91005, nearly half the heat is provided by the ER elements with a COP of 1. Part of the explanation for the excessive ER usage comes from the lack of outdoor thermostat lockout controls at sites 91004 and 91005. Site 91001 does not have an element lockout control either but its resistance use is low. The metered indoor temperature at site 91001 shows the occupants do not use a temperature setback. They keep the thermostat setting the same throughout the day. In contrast, sites 91004 and 91005 use a nighttime setback. Consequently, when coming off the nighttime setback, those two sites use resistance heat to make up the extra load.

Table 8 also shows the metered average coefficient of performance at each site. This value is the total heat pump and ER heat output divided by total input. The inclusion of the ER operation will decrease the overall COP (seen especially in site 91005). The COP calculations are discussed in detail in section 4.1.

Table 8. Metered & Calculated Outputs – Common Five Week Period

Site ID	Total Output (kWh)	COP (ave)	ER heat output fraction	Metered Time Span (hrs)
91001	1979	3.21	0.05	834
91002	627	2.56	0.09	834
91004	2475	3.09	0.19	834
91005	1172	2.23	0.47	834
91009	1485	2.41	0.04	834
91010	1457	2.39	0.07	834

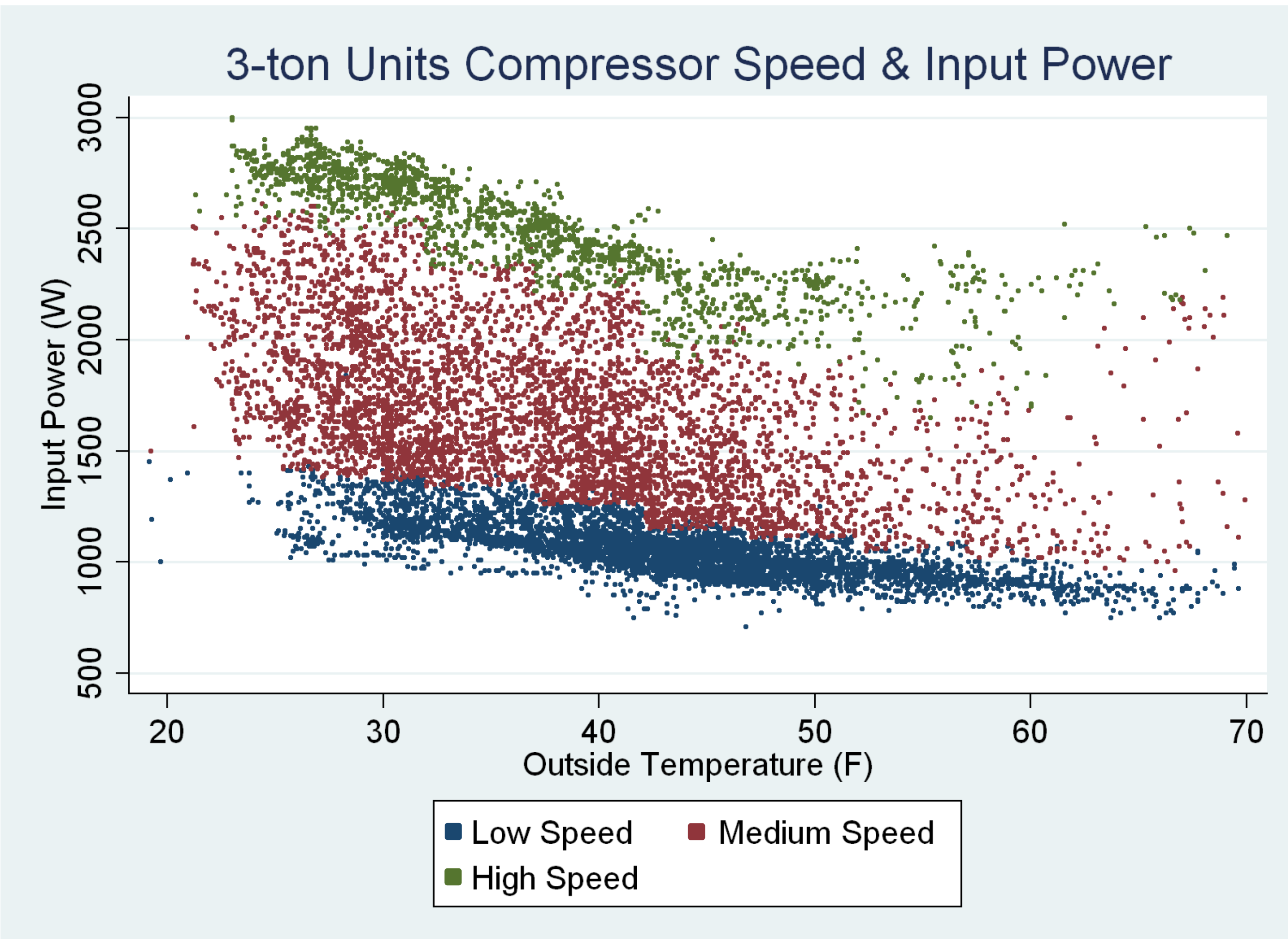
3.3.3 Fan, Compressor, and Auxiliary Heat Runtime by Mode

A key research question was to understand how the VCHP runs in response to changing heating loads. The VCHP changes its compressor speed and indoor airflow to match a heating load. We refer to these speeds and outputs as different operating modes. For this project, we did not measure the compressor speed directly but instead used the compressor power as a proxy for speed. For visualization and analytical purposes, we categorized the speed into low, medium, and high bins. The categories are defined as follows:

- Low compressor speed: input power < 1.25 x observed minimum input
- Medium compressor speed: anything in between low and high
- High compressor speed: input power > 0.87 x observed maximum input

The observed maximum and minimum inputs were inferred from the field measurements themselves. Those values change with outdoor temperature. Ecotope divided the outdoor temperature into 5°F wide bins and assigned minimum and maximum values within those bins. Figure 7 shows the input power versus outdoor temperature for all the data collected on all three of the 3-ton systems. The figure shows the distribution of compressor speeds: low is in blue, medium is in red, and high is in green. As expected, there is a clustering of high and low speeds and the high speeds are not prevalent at the warmer temperatures. There is also a trend of increasing input power as the temperature decreases.

Figure 7. 3-ton Equipment Input Power vs. Outside Temperature



A further critical research question this study sought to answer was how the variable speed compressor and variable speed air handler operate together. The field measurements showed that both varied their speeds continuously between the maximum and minimum ranges. By monitoring the equipment over several weeks, we were able to observe how the compressor and fan speeds match up. We wanted to know if the compressor and fan tracked each other (low/low, medium/medium, and high/high) or if they also worked in high/low, low/high, or other combinations. The combinations have important implications for performance and are fundamental to understanding if a simulation of the equipment is going to be accurate. Previous research on ductless heat pumps, which also have variable speed compressors and fans, shows that the systems are most efficient with high air flow and low compressor speed and least efficient with low air flow and high compressor speed.

To categorize the fan speed, we referenced the equipment specification sheet (Carrier 2012). The specification sheet lists minimum and maximum flows for each equipment size for either EFFICIENCY or COMFORT mode settings. The controllers at all the sites used the COMFORT mode setting so we referenced those for the fan speeds. The categories are assigned as follows:

- Low fan speed: airflow < 1.25 x listed minimum airflow
- Medium fan speed: anything in between low and high
- High fan speed: airflow > 0.87 x listed maximum airflow

Figure 8 shows the airflow and input power relationship versus outside temperature for the 3-ton systems. The figure is similar to Figure 7 except the color coding now represents the indoor air handler airflow: low fan is blue, medium fan is red, and high fan is green. The graph is a bit misleading at the low input power settings because the red dots actually overprint many of the blue dots. Still, the pattern is clear. The high fan speeds always go with the high input powers. The low fan speeds always go with the low input powers. The medium fan speeds run with both medium and low input powers.

Figure 8. 3-ton Equipment Airflow and Input Power vs. Outside Temperature

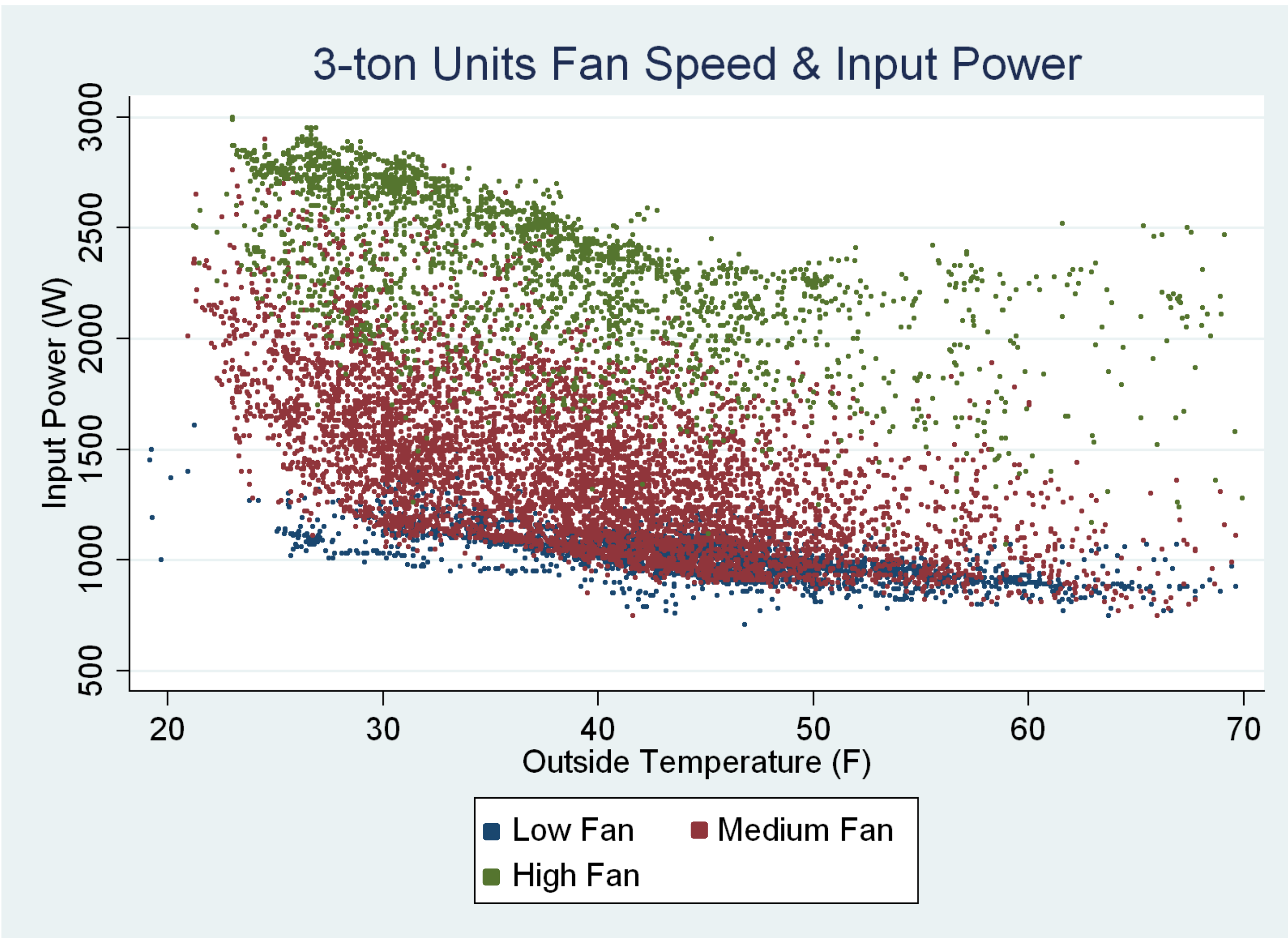


Table 9 through Table 11 quantify the matching of fan and compressor modes for all system sizes. Much as in Figure 8, the tables show that the fan and compressor speeds generally track one another. In other words, for a given compressor speed, there is a preferred fan speed. This is particularly true for high compressor outputs which almost never pair with low and medium fan. For low compressor speed, both low and medium flows are prevalent. Table 10 shows, as expected, that when the ER heat is used, the high fan speed dominates. Overall, Table 11 shows the significant finding that very little crossover in high/low or low/high speeds occurs.

Table 9. Fraction of Runtime by Mode when No ER Backup Used

Compressor Speed	Fan Speed			
	Low	Medium	High	All
Low	0.26	0.14	0.09	0.48
Medium	0.10	0.20	0.09	0.39
High	0.01	0.01	0.12	0.13
All	0.37	0.34	0.29	1.00

Table 10. Fraction of Runtime by Mode when ER Backup Heat On

Compressor Speed	Fan Speed			
	Low	Medium	High	All
Low	0.08	0.05	0.07	0.19
Medium	0.04	0.03	0.10	0.16
High	0.00	0.03	0.60	0.64
All	0.13	0.11	0.77	1.00

Table 11.Fraction of Runtime by Mode for Both Compressor and ER Operation

Compressor Speed	Fan Speed			
	Low	Medium	High	All
Low	0.24	0.12	0.08	0.45
Medium	0.09	0.18	0.09	0.36
High	0.01	0.01	0.18	0.19
All	0.34	0.31	0.35	1.00

4 Analysis

As with any study of heat pumps, the major question to answer is how efficiently the system performs. The coefficient of performance of the heat pump (“the box”) itself depends on the outside temperature, return air temperature, supply air temperature, compressor speed, and airflow. The box is also connected to a duct system to deliver heat to the house. The duct system can have a distribution efficiency of less than 100% due to air leakage and heat conduction losses. The final efficiency of the system is then a product of the box efficiency and the distribution efficiency. Ecotope measured the heat pump efficiency directly and modeled the distribution efficiency using standard techniques (ASHRAE Standard 152).

4.1 Output Capacity and Heat Pump Coefficient of Performance

In calculating the coefficient of performance we must know the input power and the heating output capacity of the heat pump.

Ideally, input power is calculated using the sum of the compressor power and the air handler power, both in watts. Our devices recorded pulses for electric resistance, air handler, and heat pump outdoor unit energy use. For each sensor, there is a known conversion of pulse counts to watt-hours.

We calculate input power for each observation as the sum of compressor power and air handler power. We must add an exception to the input power calculation as the homes recorded have electric resistance systems to complement the heat pumps. In order to have a complete picture of the power draw of the system as a whole we must consider times when the resistance heat is active.

Our devices were wired such that the device recording electric resistance power also recorded the air handler power. Even though the ER channel also includes the air handler circuit, the current transducer used for the large ER power draws does not accurately measure the smaller currents used by the air handler. Accordingly, when no ER elements are in use, we want to use just the sum of the air handler channel and heat pump outdoor channel to determine input power. This provides the most accurate measure of input energy.

When the resistance heat is running we take a different approach. To isolate resistance power we must simply subtract air handler power. We use a threshold of 600W (well above the maximum air handler power and well below the minimum resistance power draw) to determine if the electric resistance system is active. When we determine that resistance heat is being supplied we use that power in place of the air handler power to calculate the input power.

Now that we have calculated the input power of the whole system we must calculate the heat output capacity. Heat output capacity is calculated using the equation $Q = mc_p\Delta T$, where

Q = heat output (Btu/hr)

m = mass flow rate of air (lb/hr)

ΔT= supply air temperature – return air temperature (°F)

c_p= heat capacity of air (Btu/lb-°F)

Field measurements were taken at each site to model a relationship (using the fan affinity laws) between air handler wattage draw and CFM produced. Heat output capacity (watts) is then the change between supply air temperature and return air temperature multiplied by air handler volume flow multiplied by local air density divided by 3.412 (to convert from Btu to watts). The coefficient of performance (COP) is then calculated as the ratio of heat output capacity to system input power.

Figure 9 shows the results of the output capacity and COP calculations for a 4-ton system. All of the calculations are conducted without ER heat running. Compared to a single-speed heat pump, the figure shows a noticeable difference in output capacity. The observed capacity increases as the outdoor temperature decreases which is the exact opposite of a single speed system. The graph explains exactly why this happens – the input power also increases as temperature decreases. At the lower temperatures, the compressor is running faster to increase output. The boxplots in the figure show the median and range of operation for each temperature bin. The bar in the middle of the box is the median value. The box represents the inner quartile (50% of all data). The whiskers on the end mark the outer quartile of operation. Due to the variable speed operation, there is always a broad range of data in each temperature bin.

Figure 9. Heating Capacity, Input, and COP for Site 91010

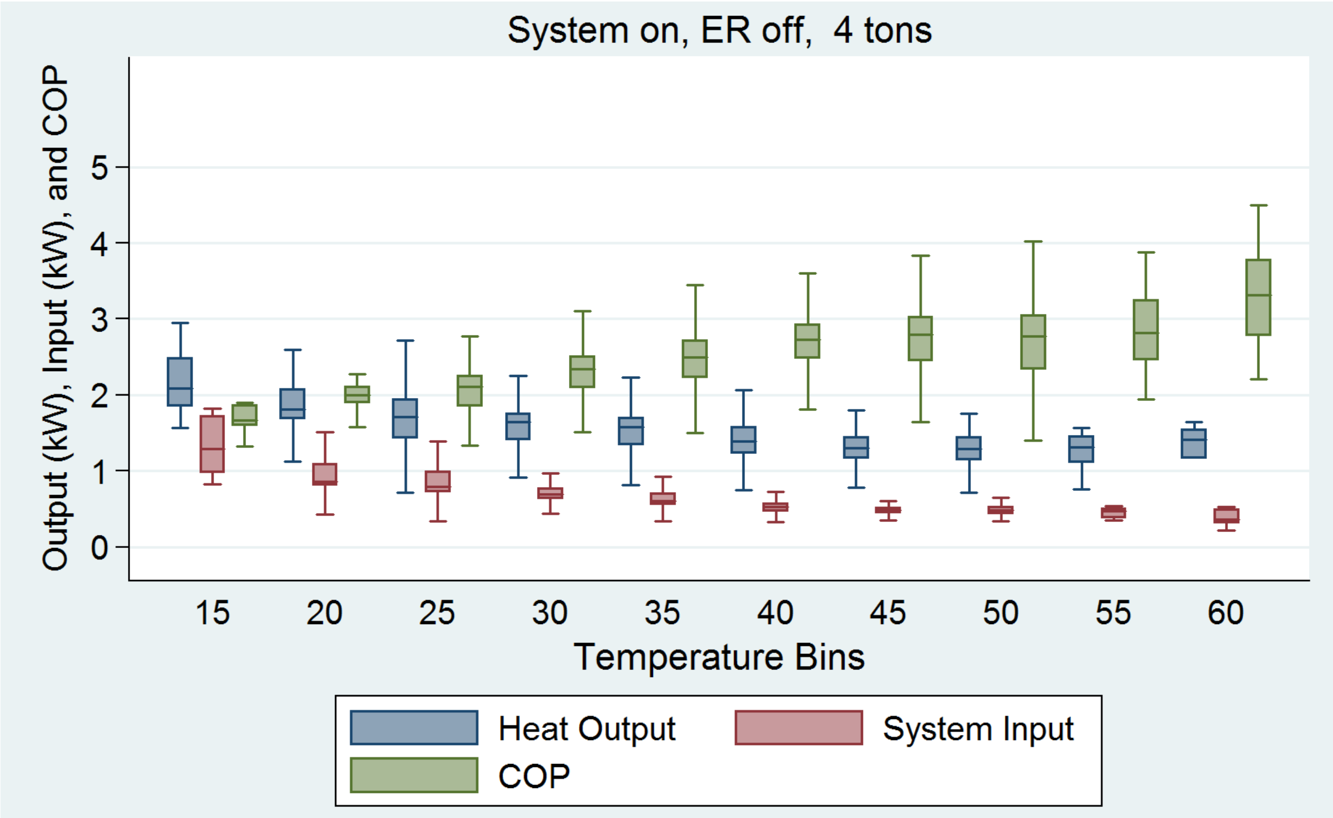
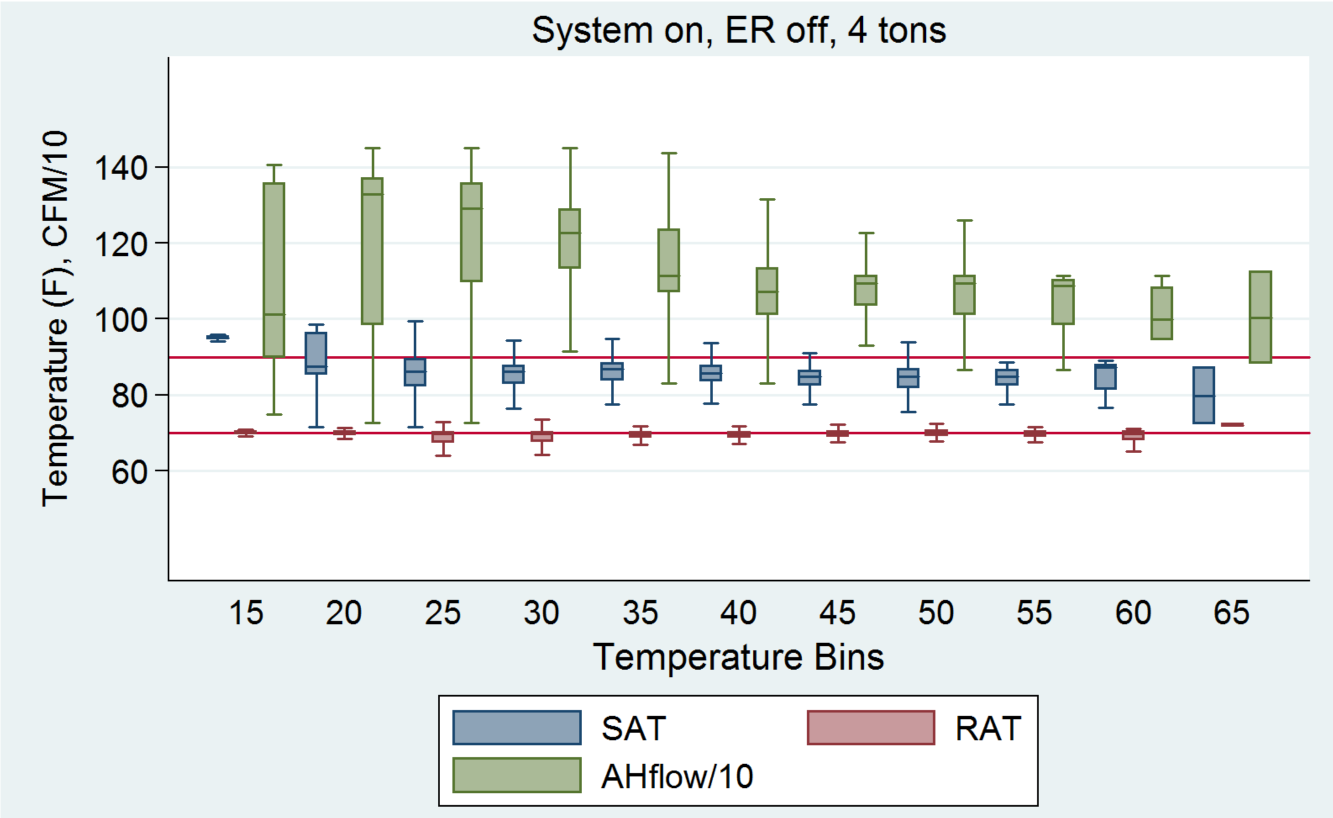


Figure 10 shows the measurements of airflow and air temperature used to calculate the output capacity. All temperatures are in Fahrenheit. To plot on the same graph, the air flow (AHflow) is scaled by 10. In other words, a flow of 120 on the graph corresponds to 1,200 CFM. The red horizontal lines are at 70° F and 90° F corresponding to typical return and supply air temperatures encountered in the study.

The output, input, and COP plots for all sites are shown in Appendix A. Supplemental Graphs The general shape and magnitude of the COP boxes in those graphs changes somewhat between sites. The differences can largely be explained by the differences in HSPF rating for each specific box at a given site. Site 91002, with the 2-ton unit, has the lowest HSPF rating at 10.5. This is likely due to the smaller indoor and outdoor heat exchangers and lower airflows characteristic of this coil combination. In contrast, Site 91001 has a rated HSPF of 13 and shows accordingly higher performance than 91002 (the difference is about 15%). Examining the 4 ton sites, 91009 and 91010 (12.5 HSPF), shows similar behavior between them but also lower COPs than the 13 HSPF systems. On closer examination, site 91009 shows better COPs than 91010. One contributing factor there is the air handler and duct system. Site 91010 has higher duct static pressures, which means the air handler always draws more power to deliver the same air flow (thereby lowering the COP). The one outstanding site is 91005, a 3-ton system with rated HSPF 13 that shows lower COPs than its 3-ton counterparts. The duct system has slightly higher static pressures but not dramatically so. It's unclear what exactly is reducing the steady-state COP at that site. Overall, with the exception of site 91005, the observed performance differences of the systems can largely be explained through the different nominal HSPF ratings and by comparing the air handler powers.

Figure 10. Supply Air Temperature, Return Air Temperature, and Airflow at Site 91010



4.2 Duct Losses

4.2.1 Duct Leakage

The duct system serves the critical function of transporting heat from the furnace or indoor heat exchanger to the rest of the house. For a given house, any equipment type creating the heat - furnace, single-speed heat pump, or variable-speed heat pump - will be connected to the same duct system. The duct losses, however, are not incurred identically between all the systems. It is important to understand if the duct losses occur more or less readily in variable-speed systems than in single-speed systems.

The measurement plan for this project collected enough information to provide direct, reliable, and accurate estimates of duct losses. Duct losses can be directly measured in “coheating” tests (Olson, et al., 1993). This project took a less intensive approach and leverages the method of testing set out in ASHRAE Standard 152. There are two main components to understanding the duct losses: air leakage out of and into the ducts and heat conduction through the ducts. For ducts installed in crawl spaces or attics, the air and heat transfers between the duct and those buffer spaces. The supply-side duct leaks send hot air into the crawlspace (some of which leaves the house through crawl vents and some of which is regained back into the house), while the return-side duct leaks draw in colder buffer space air instead of house inside air. The heat loss due to conduction is driven by the temperature difference between the duct and buffer space air temperatures.

The field technicians performed an extremely detailed duct audit on site to characterize the two components of duct losses. The first step was conducting a duct tightness test to quantify air leakage to outside the conditioned space. The results of the duct tightness tests were given previously in Table 2. Using the test results, a leakage fraction for both the supply and return sides at the operating conditions of the duct system can be calculated. The duct tightness test includes two test points; the duct openings to the house are taped off and the ducts pressurized to levels near 50 Pa and 25 Pa. The tests give us the following measurements:

$Q_{high\ pressure}$	(flow into the duct system at/near 50 Pa; this is the high pressure test point leakage)
ΔP_{high}	(actual high test pressure, typically 50 Pa or very close to it)
$Q_{lowpressure}$	(flow into the duct system at/near 25 Pa; this is the low pressure text point leakage)
ΔP_{low}	(actual low test pressure, typically 25 Pa or very close to it)

Duct air leakage, like house air leakage, follows a function described by the form $Q=C\Delta P^n$, where

Q	is the actual leakage in the duct system
ΔP	is the pressure acting on the leaks
n	is an empirically-determined flow exponent, and
C	is a constant which relates to the type of test and configuration of leaks in the system.

During the duct tightness test, two flows are measured and two test pressures are also measured; this allows the flow exponent and constant to be determined for each unique duct system that was tested.

We therefore solve for the coefficient and exponent using our field reference measurements.

$n = \ln(Q_{high\ pressure}/Q_{low\ pressure})/\ln(\Delta P_{high}\ /\Delta P_{low})$

$C = (Q_{high\ pressure}/\Delta P_{high})^n$ or $(Q_{low\ pressure}/\Delta P_{low})^n$

At any given pressure in the duct system, we can then calculate a flow (Q) as

$Q_{supply\ or\ return\ loss}=C\Delta P^n$

4.2.2 Leakage Fraction Calculations

Now that we have the coefficient and exponent as constants we must calculate ΔP for each observation. Field measurements are taken for later transformation reference: CFM reference, supply reference (or return depending on the site). The ASHRAE 152 method for calculating the ΔP is to simply take half the plenum pressure, but in this research, we have field measurements from which we can more directly calculate the pressure differential between inside and outside the ducts. We calculate plenum pressure at each air handler flow separately and then divide it by two to put into the leakage fraction calculation. At each air handler flow, the plenum pressure is calculated as:

$$P_{plenum} = \sqrt{\frac{Q_{system}}{CFM_{reference}}} \times P_{reference,supply}$$

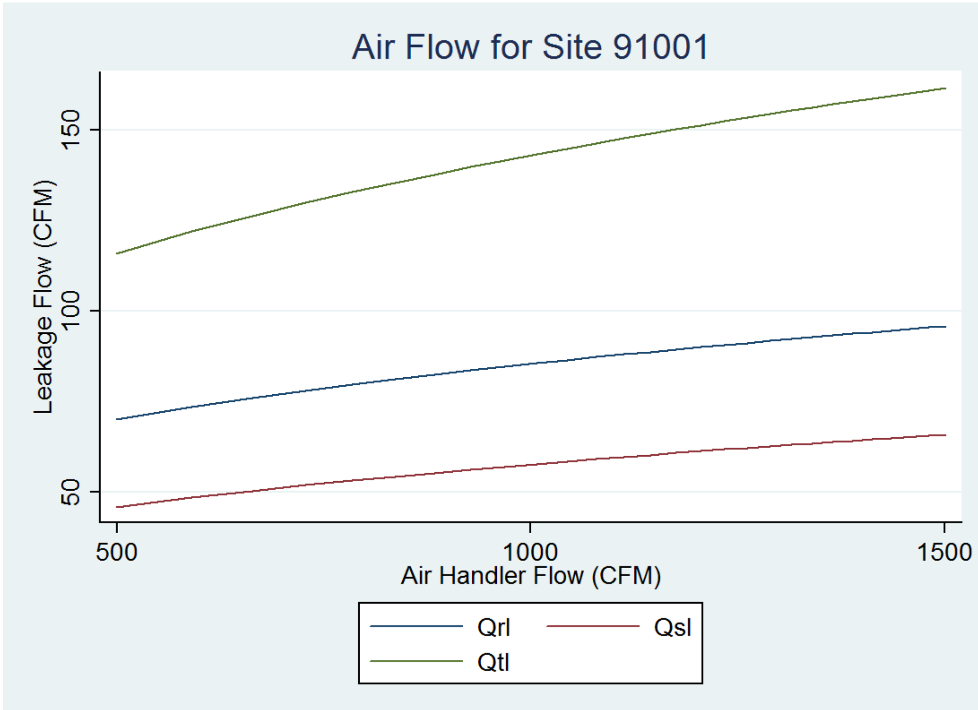
where Q_{system} is our air handler flow measurement in the current time interval and $CFM_{reference}$ is the air handler CFM that we measure at the same time we measure reference supply and return static pressures. $P_{reference,supply}$ can be interchanged with $P_{reference,return}$ depending on which side of the plenum the reference pressure was measured. The final supply and return leakage fractions are:

$$SLF = \frac{Q_{supply\ loss}}{Q_{system}}, \qquad RLF = \frac{Q_{return\ loss}}{Q_{system}}$$

We now have the supply volume flow, return volume flow, system flow, supply leakage fraction, return leakage fraction, and total leakage fraction (sum of SLF and RLF) for each observation.

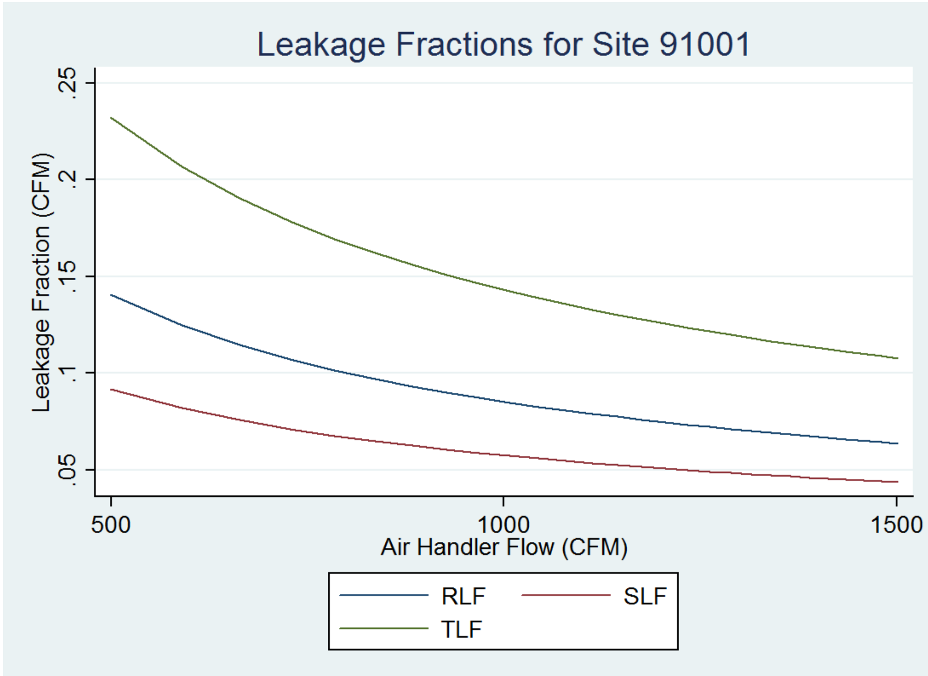
For a single-speed system, the operating conditions consist of a single air flow and static pressure regime within the ducts. For variable speed systems, the operating conditions constantly change. The static pressure depends on the airflow and, with it, the leakage fraction. Figure 11 shows the relationship between the air handler flow and leakage fraction. The air handler flow is plotted as “Air Handler Flow (CFM)” and the duct leakage is plotted as “Leakage Flow (CFM)”. The total (Qtl), supply (Qsl), and return (Qrl) flow are all plotted. For a single-speed, 3-ton system, we might expect an air handler flow of 1050 CFM (350 CFM/ton). The corresponding supply leakage is 55 CFM in Figure 11. In contrast, the variable speed systems change airflow and the duct leakage correspondingly changes. The figure shows the intuitive result that leakage decreases when airflow decreases.

Figure 11. Duct Leakage vs. System Airflow at Site 91001



For the duct delivery efficiency, the quantity of interest is not just the leakage in absolute terms but also the *relative* fraction of total air handler flow that is leaked. The duct leakage fraction represents the relative amount of air either not delivered to the house (supply side) or not returned to the air handler (return side). Due to the physics of air flow, as illustrated in the aforementioned equations, the absolute duct leakage does not decrease as quickly as the air handler flow decreases. Figure 12 shows the relationship that lower system flows lead to larger percentage losses. As in Figure 11, the figure also plots the system air flow but the vertical axis now shows the leakage fraction (RLF – return leakage fraction, SLF – supply leakage fraction, and TLF – total leakage fraction). If the air handler flow is fixed, as in our example of the single-speed system, the SLF would be fixed at ~7%. For a variable speed system, Figure 12 shows that as the flow decreases, the leakage fraction increases.

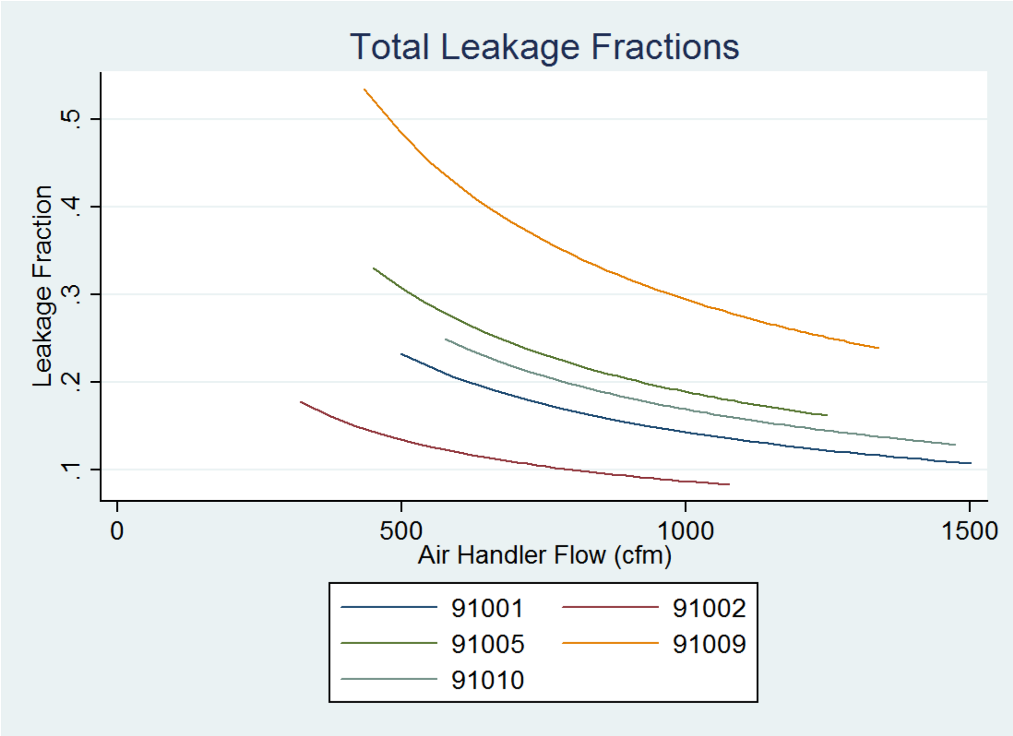
Figure 12. Leakage Fraction[1] vs. System Airflow at Site 91001



[1]RLF is return leakage fraction, SLF is supply leakage fraction, and TLF is total leakage fraction

Figure 13 provides a comprehensive view of the *total* leakage fractions at all the sites as a function of air handler flow. Note that site 91004 has all its ducts inside the conditioned space. Based on observations from the field, we found that the controls favor low and medium fan speed operation. The figures show that this lower airflow regime actually *increases* the duct losses. Therefore, compared to single-speed heat pumps, we expect their variable-speed counterparts to incur larger duct losses.

Figure 13. Leakage Fraction vs. System Airflow – All Sites



4.2.3 Calculated Total Duct Losses

To develop the full picture of the impacts of duct leakage and conduction, we employ ASHRAE Standard 152 and calculate the distribution efficiency at each time interval of our metered data.

The distribution system efficiency is calculated using the following equation:

$$DE = \alpha_{supply}\beta_{supply} - \alpha_{supply}\beta_{supply}(1 - \beta_{return}\alpha_{return})\frac{\Delta t_{return}}{\Delta t_{system}} - \alpha_{supply}(1\text{-}\beta_{supply})\frac{\Delta t_{supply}}{\Delta t_{system}}$$

The α values are the leakage factors for the supply side and return side. These values are the complements of the previously calculated leakage fractions (e.g.α_{supply} = 1 – SLF). α is the fraction of air that is delivered to the house or return to the air handler.

The β values are the conduction fractions for the supply side and return side. The model for conduction fraction is as follows:

$$\beta_{side} = \exp[\frac{-A_{side}}{60Q_{system}\rho_{input}C_pR_{side}}]$$

Where A_{side} is the duct surface area for the supply or return side and R_{side} is the thermal resistance for the corresponding supply or return side. Both values are recorded manually on site after a complete inspection of the duct system. Q_{system} is the air handler flow (cfm) of the entire system, ρ_{input} is the air density of the region, and C_p is the heat capacity of air. Q_{system} is metered with our data logging system and changes as the heat pump controller changes fan speeds.

Δt_{system} is the difference between the supply air temperature and return air temperature which is continuously metered with the data loggers.

Δt_{supply} is the difference between the indoor temperature and the supply buffer space ambient temperature. The supply buffer space (generally the crawl space) temperature was continuously logged in the field.

Δt_{return} is the difference between the indoor temperature and the return buffer space ambient temperature. In the study, most of the returns were in the attic. Due to study design constraints, we did not log the return buffer space temperature. Instead, we refer to a table of default seasonal values in ASHRAE Standard 152. The table shows that for the Redmond/Bend OR area, the seasonal temperature for attics at our sites is 44°F.

The results of the duct efficiency calculations are shown in Table 12. The values shown are the average values over the common five week metering period. The table illustrates the significance of the distribution system. Houses with ducts inside have a DE of 1 and all the others incur losses. For example at site 91001, only 65% of all the heat that is present at the air handler is delivered to heat the interior of the house. In addition to the duct air leakage, the conduction losses contribute significantly to the low distribution efficiencies. The duct runs are long and expansive at many sites.

Table 12. Measured and Calculated Average Distribution System Efficiency (DE)

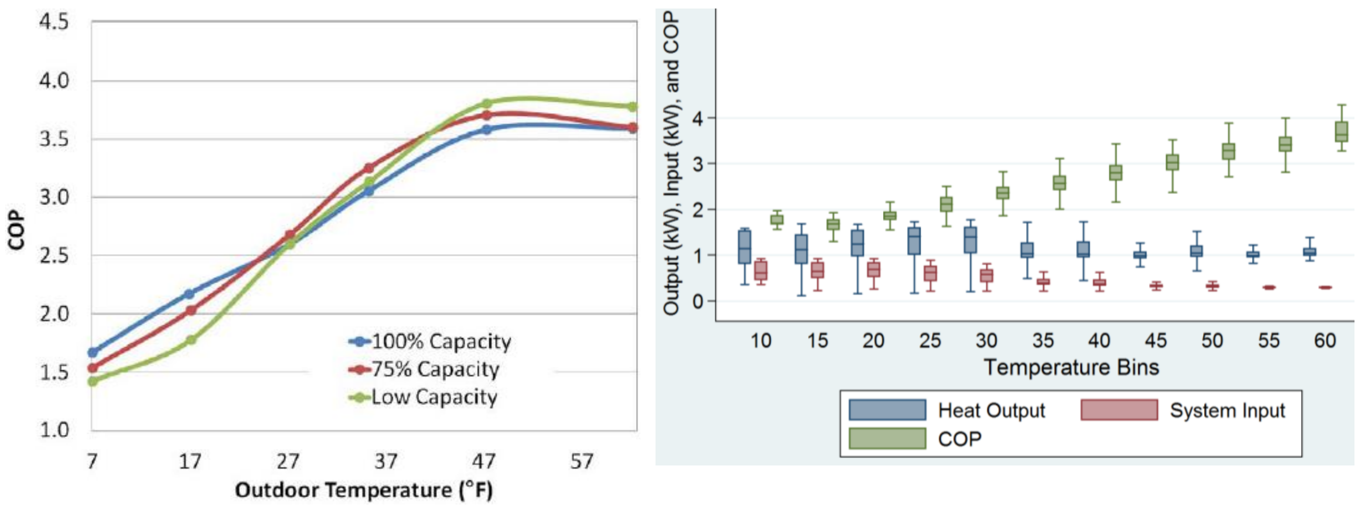
Site ID	System Airflow (CFM)	Avg Supply Buffer Zone Temp (°F)	SLF	RLF	DE
91001	675	56.8	0.08	0.11	0.65
91002	550	55.2	0.13	0	0.82
91004	722	n/a	0	0	1.0
91005	766	55.9	0.07	0.17	0.61
91009	748	65.3	0.18	0.19	0.49
91010	1159	57.7	0.13	0.02	0.68

4.3 Comparison to Lab Testing Results

As part of BPA's research effort on variable capacity heat pumps, EPRI recently completed a lab study of a 2-ton VCHP (Hunt 2013). The lab measurements are made in a highly controlled setting which allows a detailed and accurate set of measurements. They are extremely useful when compared to the field measurements. Inherently, field measurements are made under less than ideal conditions so it is important to see if they produce results similar to the lab measurements. When they agree, our confidence in both measurements increases. The one significant difference between the two systems was that the lab unit was set in EFFICIENCY mode in contrast to the COMFORT mode we observed at all field sites. Based on the known differences between modes, we expect the field data to show lower airflows, higher supply air temperatures, and lower COPs.

Figure 14 shows the steady-state system efficiencies for the lab and field measurements of a 2-ton system side-by-side. The graph on the left of the figure is from the EPRI lab testing report (Hunt 2013). The graph on the right shows the field measurements. The figures generally show good agreement between the two measurements. For example, for 17°F temperatures in the lab, the COP ranged around 2. Likewise, examining the 15°F and 20°F bins in the field also show a COP around 2. Clearly, however, the COPs in the field are slightly lower. We expect that based on the use of COMFORT mode as compared to EFFICIENCY mode. Overall, the agreement with the lab measurements, a "gold standard", gives us confidence in our field measurements.

Figure 14. Lab and Field COPs Compared – 2-ton System



Further important findings from the lab showed that the fan and compressor speeds generally track one another through the operating modes (Hunt 2013). That finding matches ours as discussed in section 3.3.3.

In considering the lab and field results, we conclude that performance curves of the heat pump itself can largely be developed from the lab data alone. If the lab conducted tests in both EFFICIENCY and COMFORT mode, it would be possible to build a performance map for the equipment. This would obviate the need for *in situ* field measurements for that parameter. In this particular project, the performance does change somewhat with different sized equipment. The lab tested a 2-ton system whereas the field work was able to observe 2, 3, and 4-ton systems. To develop an accurate performance map in a lab setting, testing more than one system size should be considered.

What was not clear from the lab testing in a controlled environment was how the controller would set operating modes in response to outdoor temperature, heating load, and recovery from setbacks. It is often not possible to test enough conditions in the lab to observe all those modes. The project showed that short-to-medium term (several weeks) field monitoring study can provide the needed insight in to operation. Alternatively, field measurements could be avoided here if the manufacturer would divulge the control strategy information.

Additional information obtainable only in the field showed just how the system interacts with the auxiliary resistance heat. This depends on ODT settings, heat pump size relative to heating load, and use of setbacks. Those, in turn, are determined by the HVAC installer and occupant and cannot be readily explored in the lab setting.

4.4 Modeled Heat Pump Efficiency and Duct Losses

To create an annual estimate of heat pump efficiency and duct losses, Ecotope used the SEEM energy use simulation program. Ecotope updated the simulation to include a variable capacity heat pump performance curve. The performance curve is based on the measured field data and consists of separate input power and output capacity equations. Those equations are functional fits of outdoor temperature, air handler flow, and return air temperature to the field data. The simulation implements a simple "load following" algorithm to have the heat pump change its output to track the house heating load. The simulation checks to see if the minimum load is below the minimum allowed by the equipment. If so, it cycles under part-load. If the maximum heating load exceeds what can be provided by the equipment, the simulation adds in auxiliary resistance heat. For the duct losses, SEEM implements the ASHRAE Standard 152 method so the losses vary depending on equipment operation and airflow.

The following tables show the modeled efficiency for the equipment (the box alone), the distribution efficiency, and the combined equipment and duct efficiency. For comparison, the simulation was run with an electric furnace, a single speed heat pump with HSPF 7.9, and the new variable speed heat pump model. Table 13 shows the annual efficiency of the box alone for each of the six houses. As expected, because of the higher performance values from the box specification, the table shows that the annual average COP of the variable speed heat pump exceeds that of the single speed version. Table 14 compares the duct losses and shows they are greatest for the variable speed system. Of note, the electric furnace, which is the least efficient in terms of the box, has the best distribution efficiency. This is due to furnaces operating at higher airflows and delivery temperatures. Table 15 shows the final delivery effectiveness or overall heat pump and duct system efficiency. It is the product of the seasonal heat pump COP and the distribution efficiency. The simulations suggest the variable capacity heat pump is 25-30% more efficient than its single-speed counterpart.

Table 13. Modeled Equipment Efficiency

Site	Electric Furnace, COP=1	Single Speed Heat Pump, HSPF=7.9	Variable Speed Heat Pump
91001	1.00	1.99	2.46
91002	1.00	2.35	2.84
91004	1.00	2.12	2.59
91005	1.00	2.07	2.53
91009	1.00	2.48	3.05
91010	1.00	2.20	2.76

Table 14. Modeled Distribution System Efficiency

Site	Electric Furnace, COP=1	Single Speed Heat Pump, HSPF=7.9	Variable Speed Heat Pump
91001	0.72	0.67	0.66
91002	0.78	0.70	0.72
91004	1.00	1.00	1.00
91005	0.69	0.60	0.59
91009	0.64	0.52	0.52
91010	0.73	0.66	0.64

Table 15. Modeled Overall Efficiency

Site	Electric Furnace, COP=1	Single Speed Heat Pump, HSPF=7.9	Variable Speed Heat Pump
91001	0.72	1.33	1.61
91002	0.78	1.63	2.04
91004	1.00	2.12	2.59
91005	0.69	1.23	1.49
91009	0.64	1.28	1.58
91010	0.73	1.45	1.77

5 Conclusions

The residential variable capacity heat pump field study evaluated the energy performance of six VCHP heat pumps installed in houses in the Bend, OR area. Through a combination of intensive, one-day site investigations, and an eight week data logging period, the project characterized the house, heat pump, and duct system. The project also set forth the analytical groundwork for evaluating variable speed ducted systems and developed an engineering model of VCHP behavior.

BPA funded a useful lab study of the same VCHPs observed in the field (Hunt 2013). By comparing our field measurements to those of the lab, and finding agreement, we have confidence in our field findings. Further, the lab and field studies work to reinforce each other in regards to the compressor and air handler operating speed combinations. In examining what we learned from both the lab and field, we can better design future studies. The lab measurements provided detailed and accurate measurements of equipment input and output (which the field also matched). This somewhat obviates the need for field measurements of capacity. The caveat is that the lab measured in EFFICIENCY mode whereas all the field sites were set up by the installer in COMFORT mode. Further, the lab only tested one size of the heat pumps whereas the field measurements addressed three system sizes.

What was not clear from the lab testing in a controlled environment was how the controller would set operating modes in response to outdoor temperature, heating load, and recovery from setbacks. It is often not possible to test enough conditions in the lab to observe all those modes. The project showed that a short-to-medium term (several weeks) field monitoring study can provide the needed insight into operation. Alternatively, field measurements could be avoided here if the manufacturer would divulge control strategy information. Additional information, obtainable only in the field, showed just how the system interacts with the auxiliary resistance heat. This depends on outdoor (lockout) thermostat settings, heat pump size relative to heating load, and use of setbacks. Those, in turn, are determined by the HVAC installer and occupant and can't be readily explored in the lab setting.

Overall, the field project showed that the VCHP systems by themselves are efficient and perform at higher levels than single speed systems. The modeled performance, based on the field data, suggests a 25-30% improvement. However, the measurements showed that duct losses are larger for variable speed systems because they typically operate at lower airflows. The losses, however, are only ~5% greater. Taken as a whole, the VCHP and duct system provide improvements over a federal minimum single-speed system.

A further finding is that, just like single speed air-source heat pumps, the equipment size (heating output), auxiliary resistance heat controls, and thermostat setback behavior still matter. Although the VCHP is able to boost its heating output at lower temperatures, the equipment still needs to be properly sized for the house heating load. An undersized VCHP will still resort to auxiliary resistance heat just like a fixed speed system. The issue is further compounded when the occupants use a thermostat setback. In two of the six cases, the morning recovery period was a clear driver for resistance heating use. A better controller could anticipate the recovery sooner to avoid the low efficiency resistance heat. Complicating the issue further is the control (or lack thereof) of the resistance heat based on outside temperature. Best practices dictate locking out resistance heat above 35°F (Regional Technical Forum 2007). This was not done at several of the field sites which then showed more resistance heating use. These issues are important in that if resistance heat is not carefully controlled, it will erode the efficiency gains made by the variable speed system.

In addition to the field measurements, the project lays the groundwork for our future understanding of variable speed systems. The investigation of duct losses is crucial to understanding overall heating performance. The project developed a performance curve and engineering model which was added to the Simple Energy Enthalpy Model (SEEM). The model will allow accurate predictions of VCHP heating energy use for simulated buildings.

Further work is needed to fully assess the cost effectiveness and energy savings potential of the VCHP. Both the lab and field findings demonstrate these VCHPs perform better than the baseline systems. The next step is to simulate energy use over a full year in a variety of climates and house types in the Pacific Northwest. The metered sites can be used to further calibrate the simulation as needed. In general, the VCHP systems offer increased capacity at lower outdoor temperatures which can provide large energy savings. An important alternative to consider to a VCHP installation is simply upsizing the planned single speed system. Where a 3-ton system may be installed, it would be useful to evaluate the savings and cost effectiveness of a 4-ton system. A simulation and cost analysis can examine the full suite of base case and efficient case scenarios and provide estimates of energy savings and relative cost-effectiveness.

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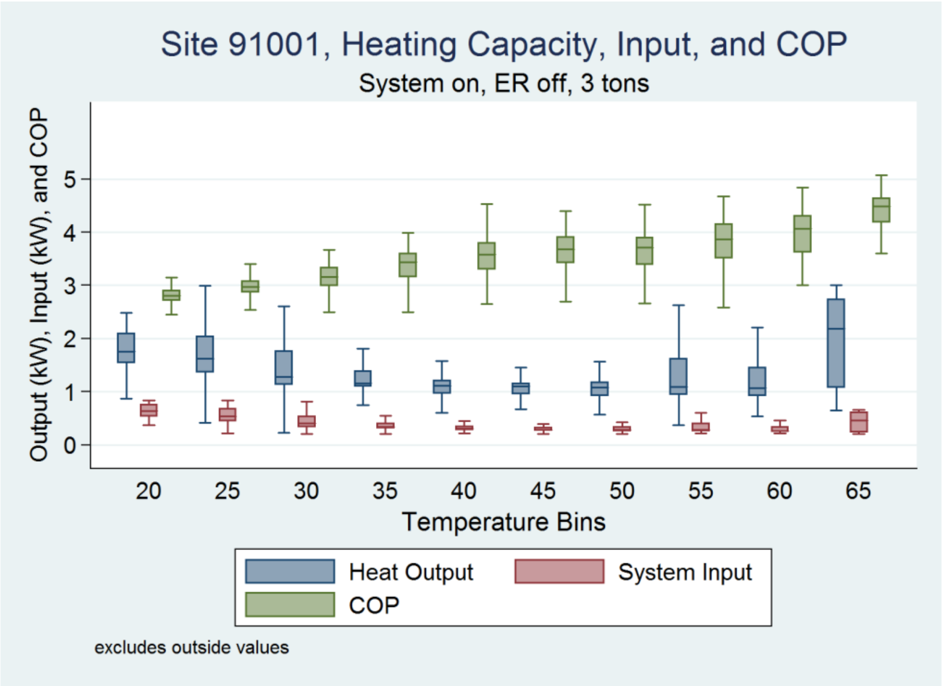
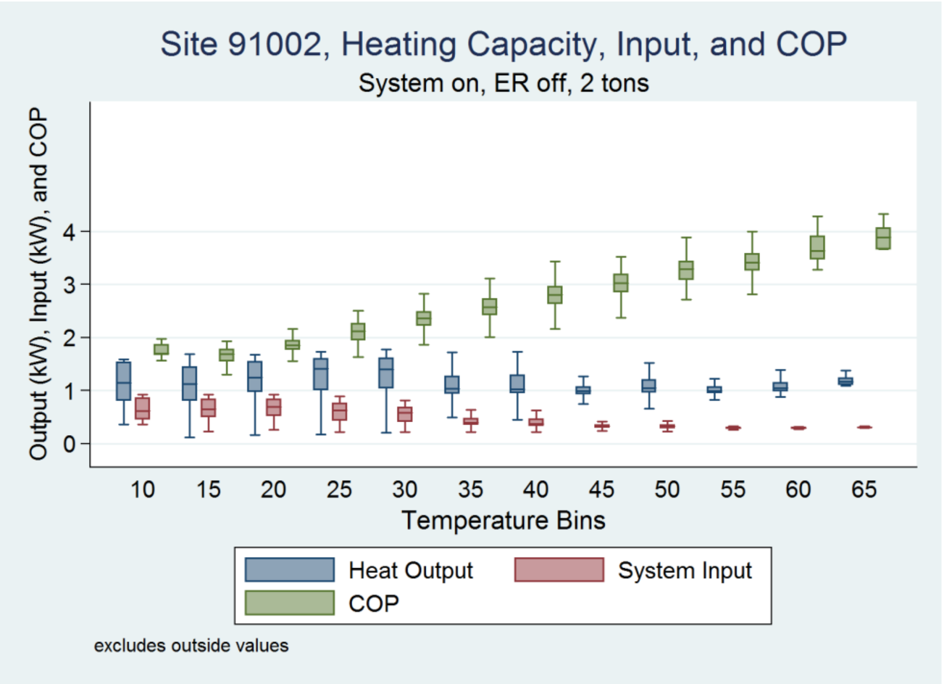
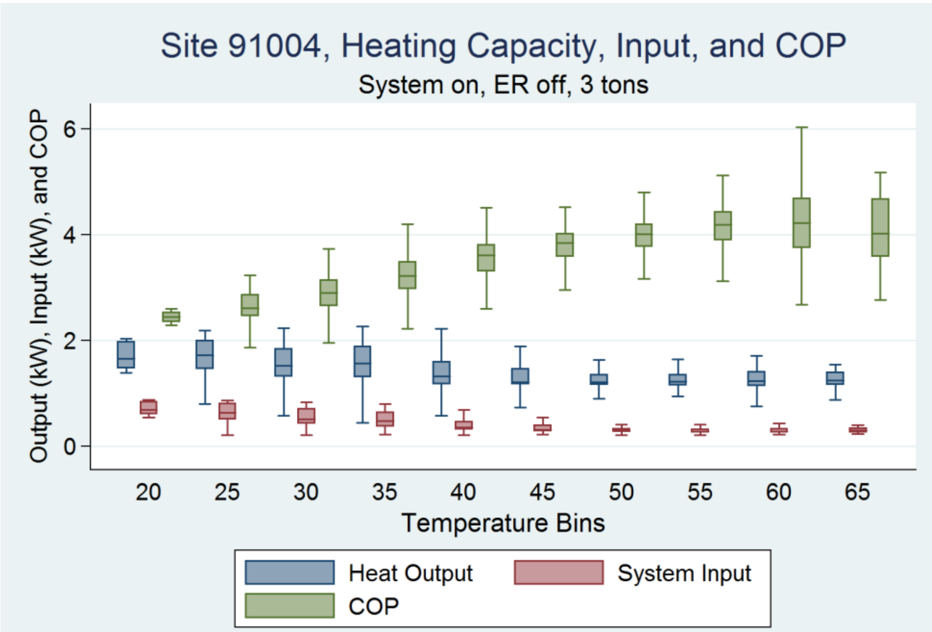
Appendix A. Supplemental Graphs

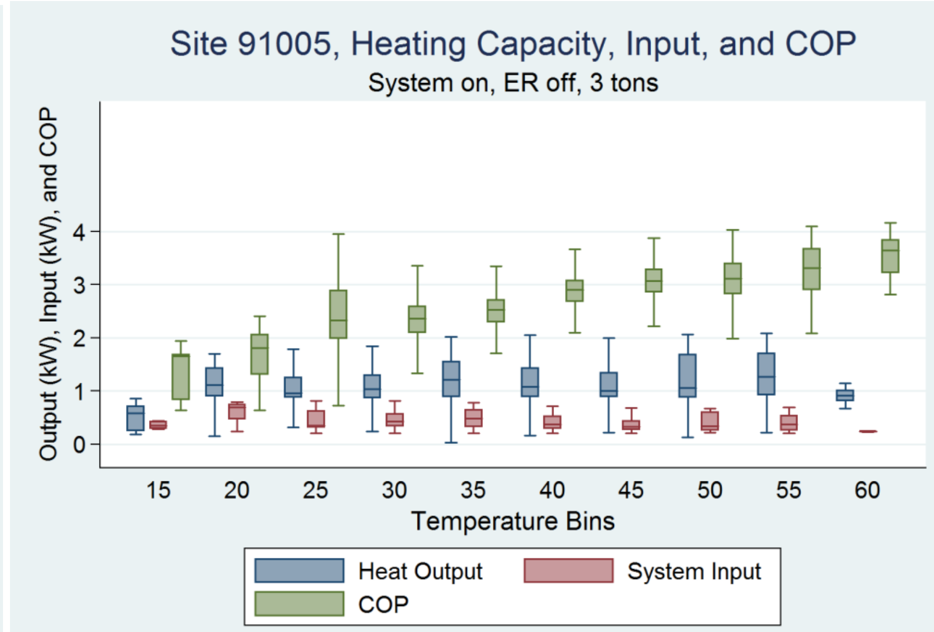
Heating Capacity, Input Power, and COP by Temperature for Each Site.



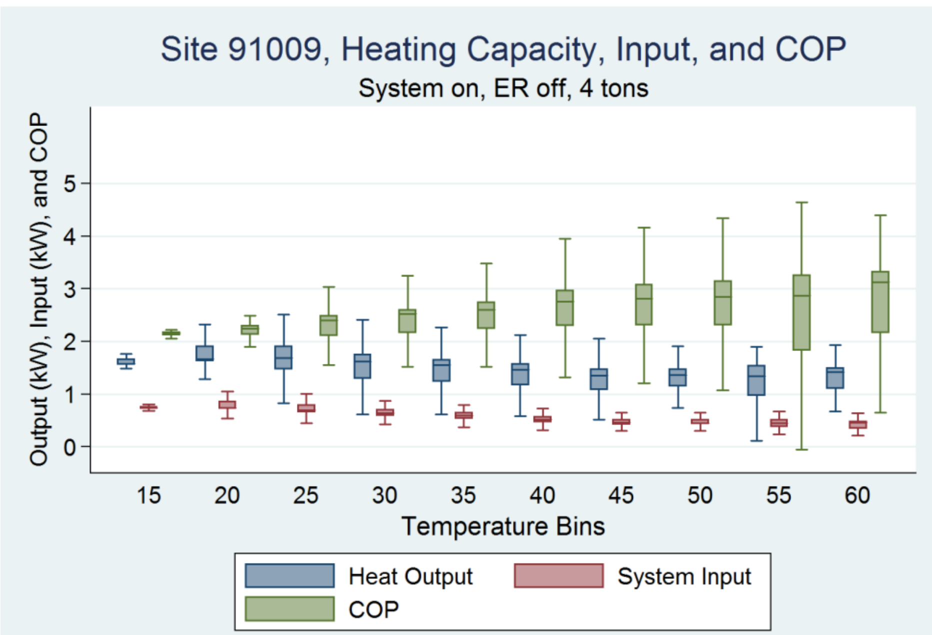




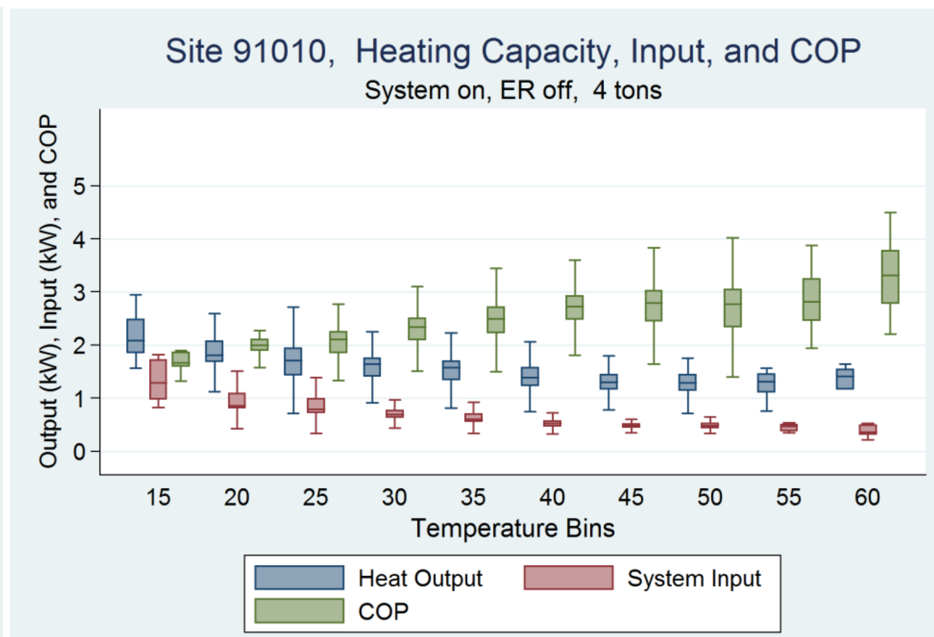
Site 91004, Heating Capacity, Input, and COP
System on, ER off, 3 tons
Output (kW), Input (kW), and COP
0
2
4
6
20
25
30
35
40
45
50
55
60
65
Temperature Bins
Heat Output
System Input
COP



Site 91005, Heating Capacity, Input, and COP
System on, ER off, 3 tons
Output (kW), Input (kW), and COP
0
1
2
3
4
15
20
25
30
35
40
45
50
55
60
Temperature Bins
Heat Output
System Input
COP

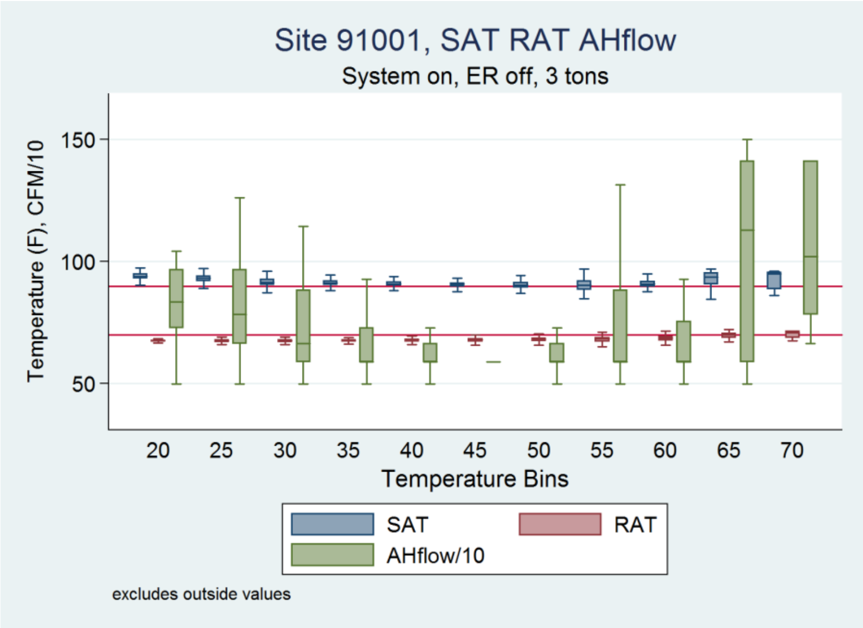


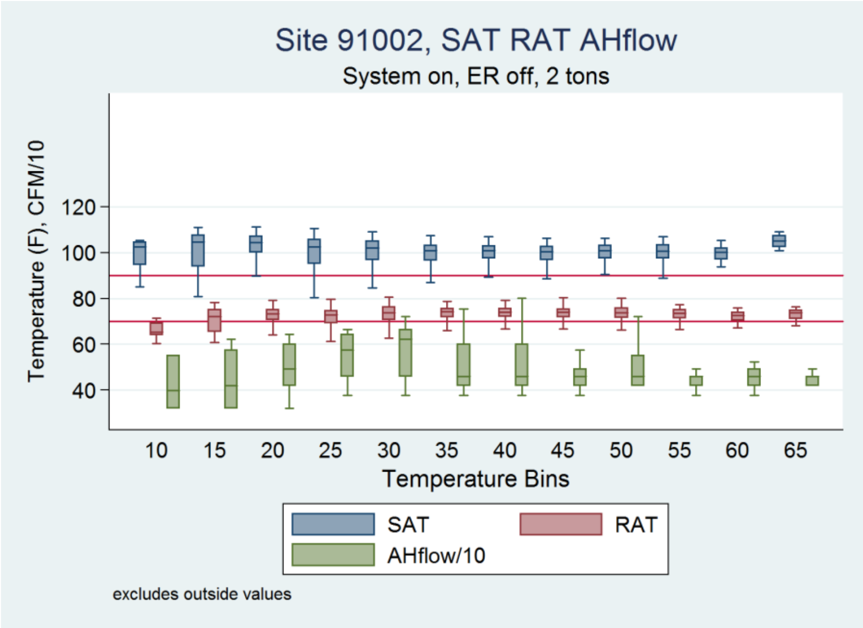
Site 91009, Heating Capacity, Input, and COP
System on, ER off, 4 tons
Output (kW), Input (kW), and COP
0
1
2
3
4
5
15
20
25
30
35
40
45
50
55
60
Temperature Bins
Heat Output
System Input
COP

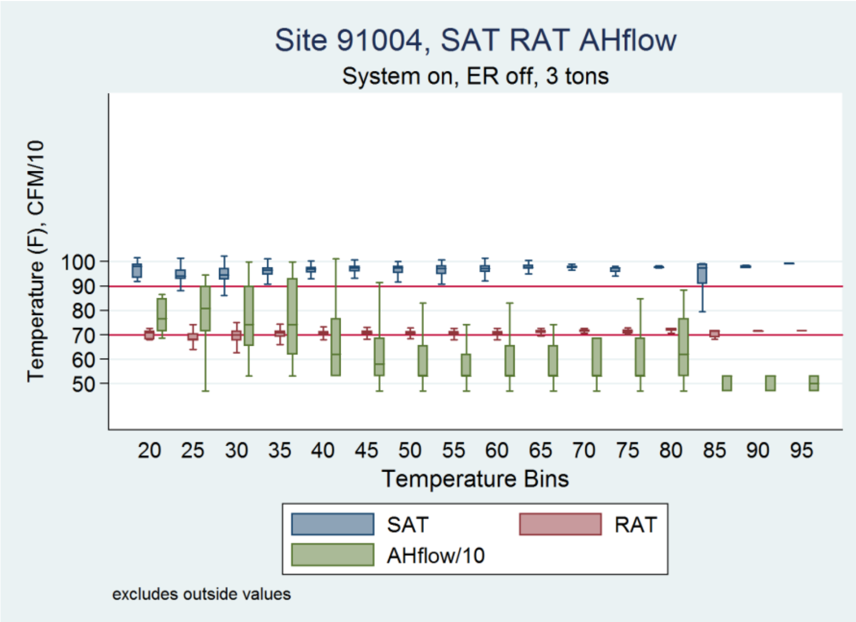


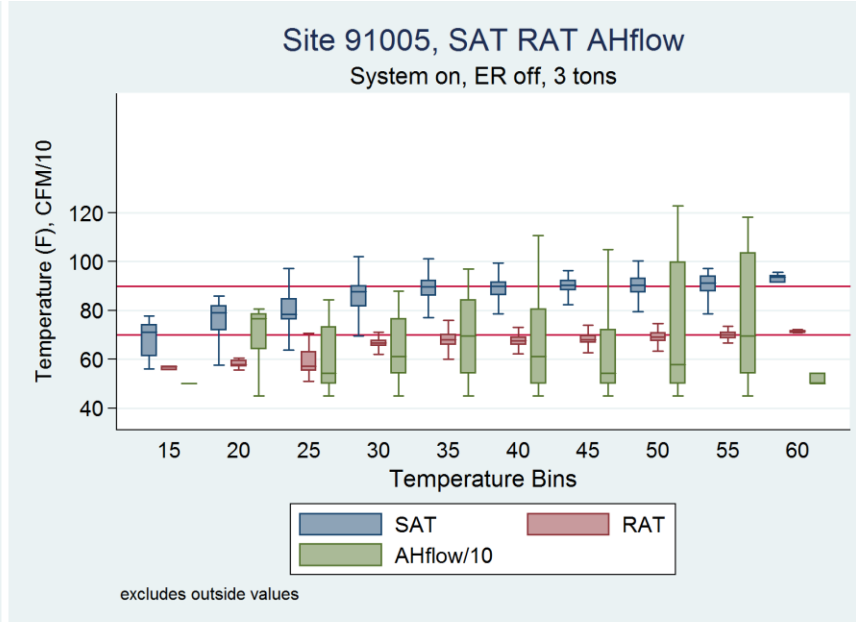
Site 91010, Heating Capacity, Input, and COP
System on, ER off, 4 tons
Output (kW), Input (kW), and COP
0
1
2
3
4
5
15
20
25
30
35
40
45
50
55
60
Temperature Bins
Heat Output
System Input
COP

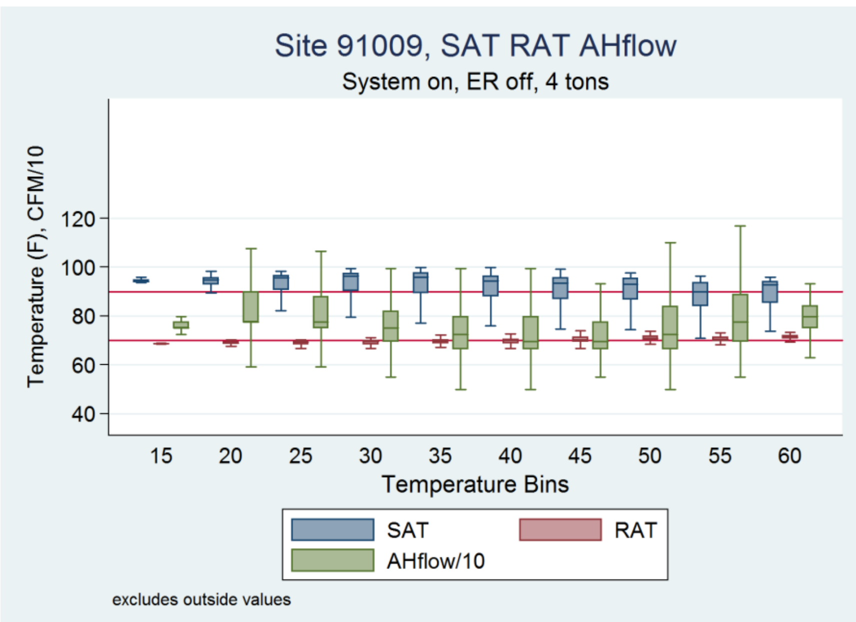
Supply Air Remperature, Return Air Temperature, and Airflow by Temperature for Each Site

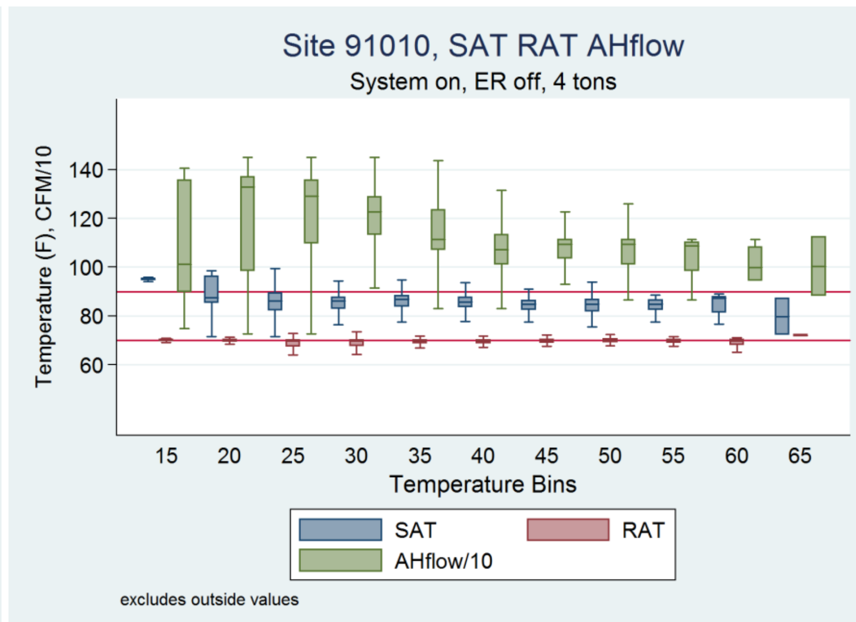




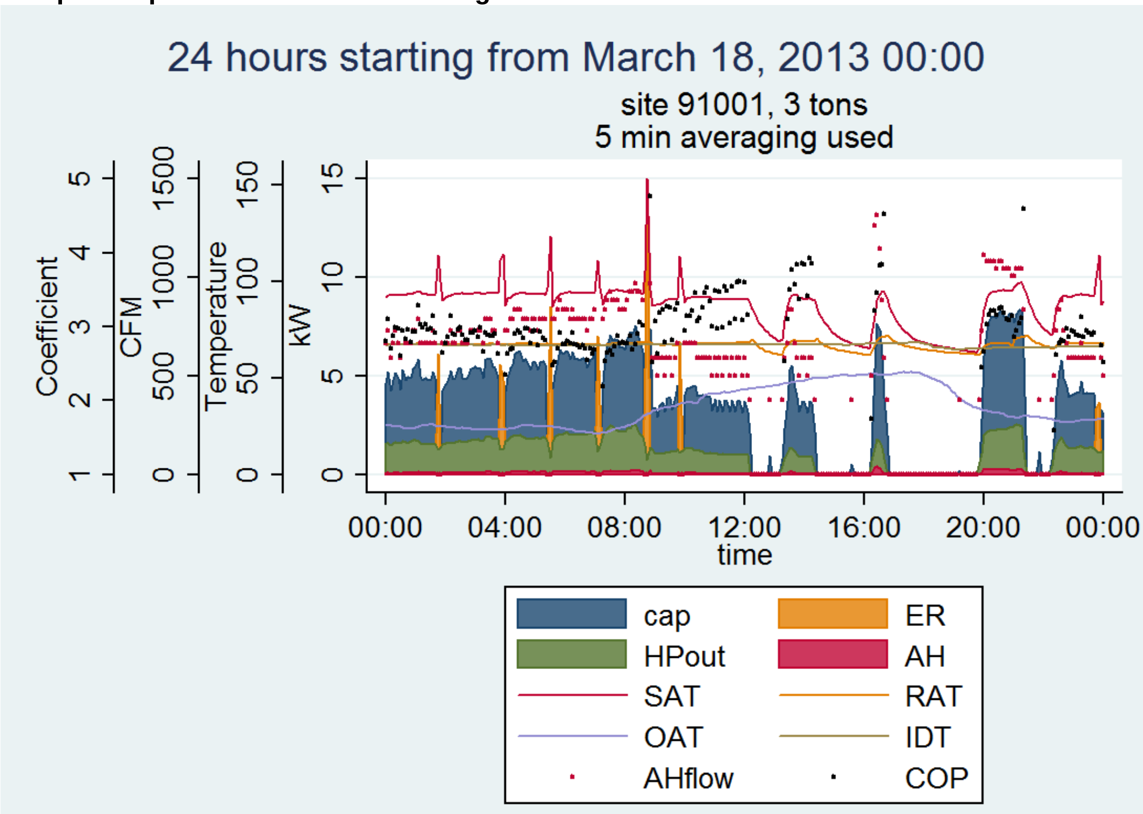

Site 91004, SAT RAT AHflow
System on, ER off, 3 tons
Temperature (F), CFM/10
100
90
80
70
60
50
20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95
Temperature Bins
SAT
RAT
AHflow/10
excludes outside values

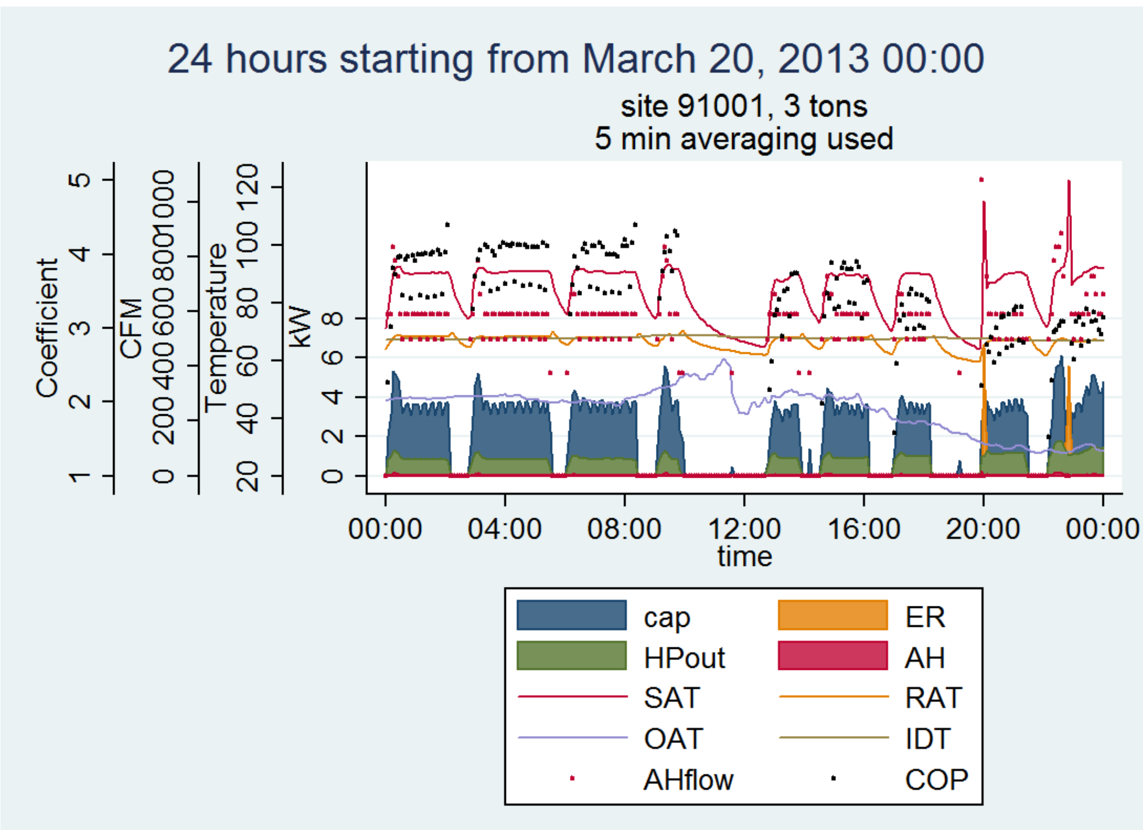

Site 91005, SAT RAT AHflow
System on, ER off, 3 tons
Temperature (F), CFM/10
120
100
80
60
40
15 20 25 30 35 40 45 50 55 60
Temperature Bins
SAT
RAT
AHflow/10
excludes outside values

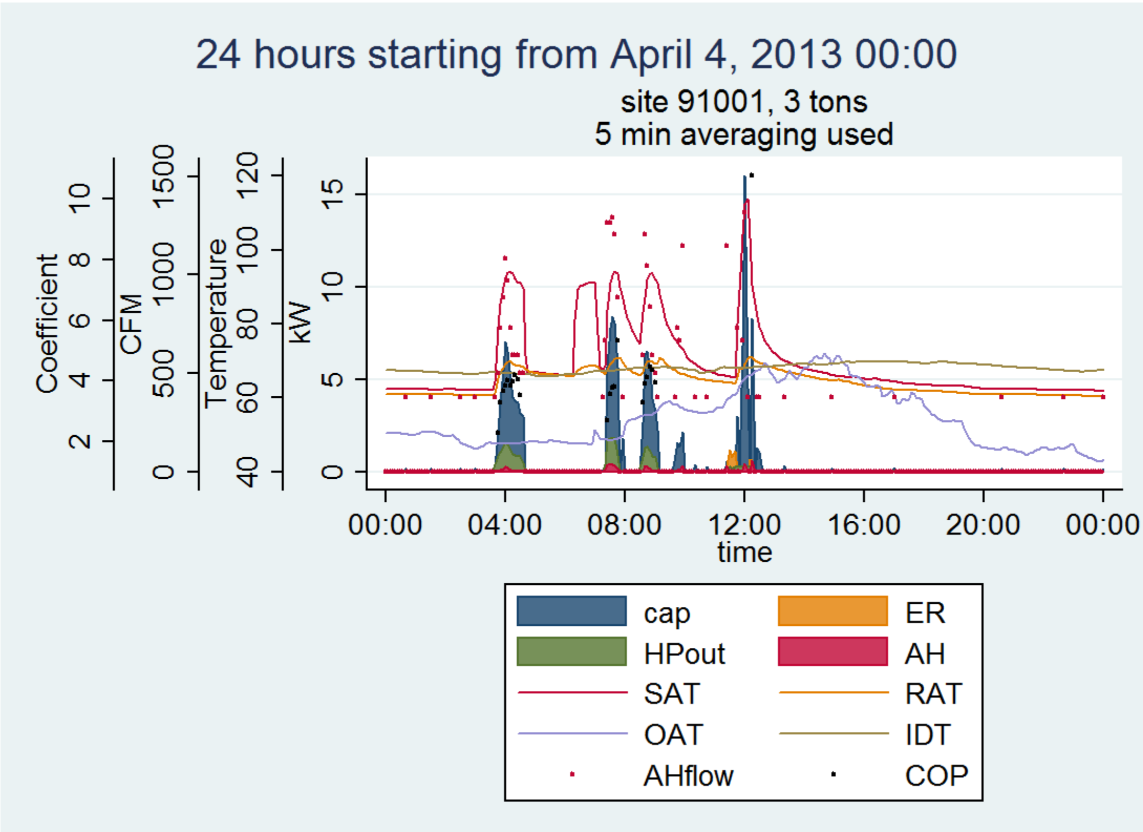

Site 91009, SAT RAT AHflow
System on, ER off, 4 tons
Temperature (F), CFM/10
120
100
80
60
40
15 20 25 30 35 40 45 50 55 60
Temperature Bins
SAT
RAT
AHflow/10
excludes outside values


Site 91010, SAT RAT AHflow
System on, ER off, 4 tons
Temperature (F), CFM/10
140
120
100
80
60
15 20 25 30 35 40 45 50 55 60 65
Temperature Bins
SAT
RAT
AHflow/10
excludes outside values

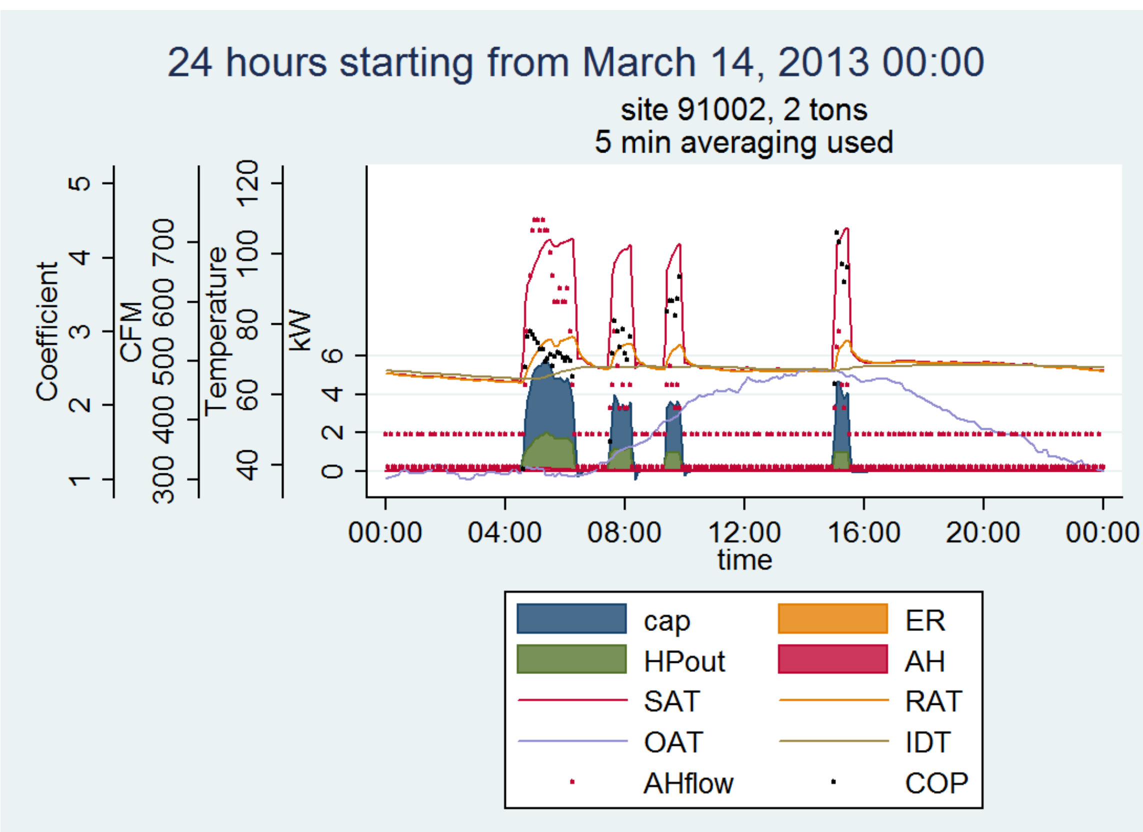
Sample Graphs of Selected Monitoring Periods at Various Sites







24 hours starting from April 4, 2013 00:00
site 91001, 3 tons
5 min averaging used
Coefficient
2 4 6 8 10
CFM
0 500 1000 1500
Temperature
40 60 80 100 120
kW
0 5 10 15
00:00 04:00 08:00 12:00 16:00 20:00 00:00
time
cap
ER
HPout
AH
SAT
RAT
OAT
IDT
AHflow
COP



24 hours starting from March 14, 2013 00:00
site 91002, 2 tons
5 min averaging used
Coefficient
1 2 3 4 5
CFM
300 400 500 600 700
Temperature
40 60 80 100 120
kW
0 2 4 6
00:00 04:00 08:00 12:00 16:00 20:00 00:00
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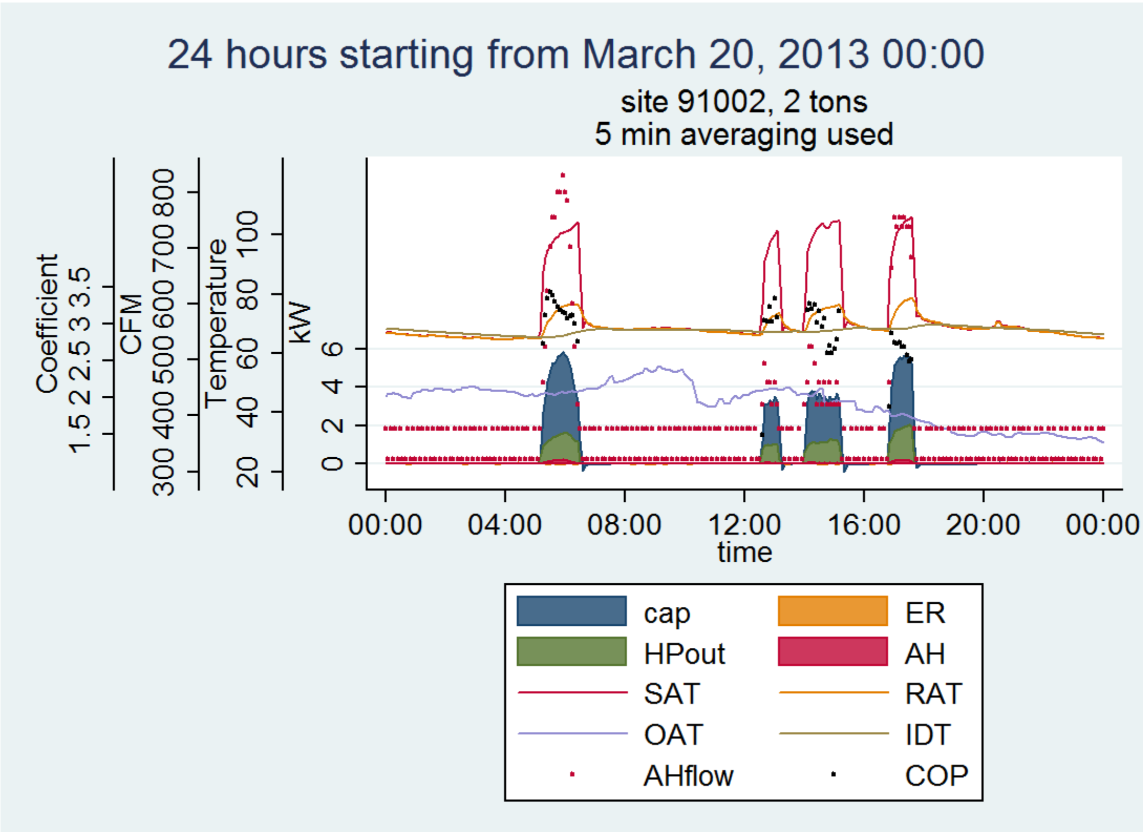
OAT
IDT
AHflow
COP


24 hours starting from March 20, 2013 00:00
site 91002, 2 tons
5 min averaging used
Coefficient
1.5 2 2.5 3 3.5
CFM
300 400 500 600 700 800
Temperature
20 40 60 80 100
kW
0 2 4 6
00:00 04:00 08:00 12:00 16:00 20:00 00:00
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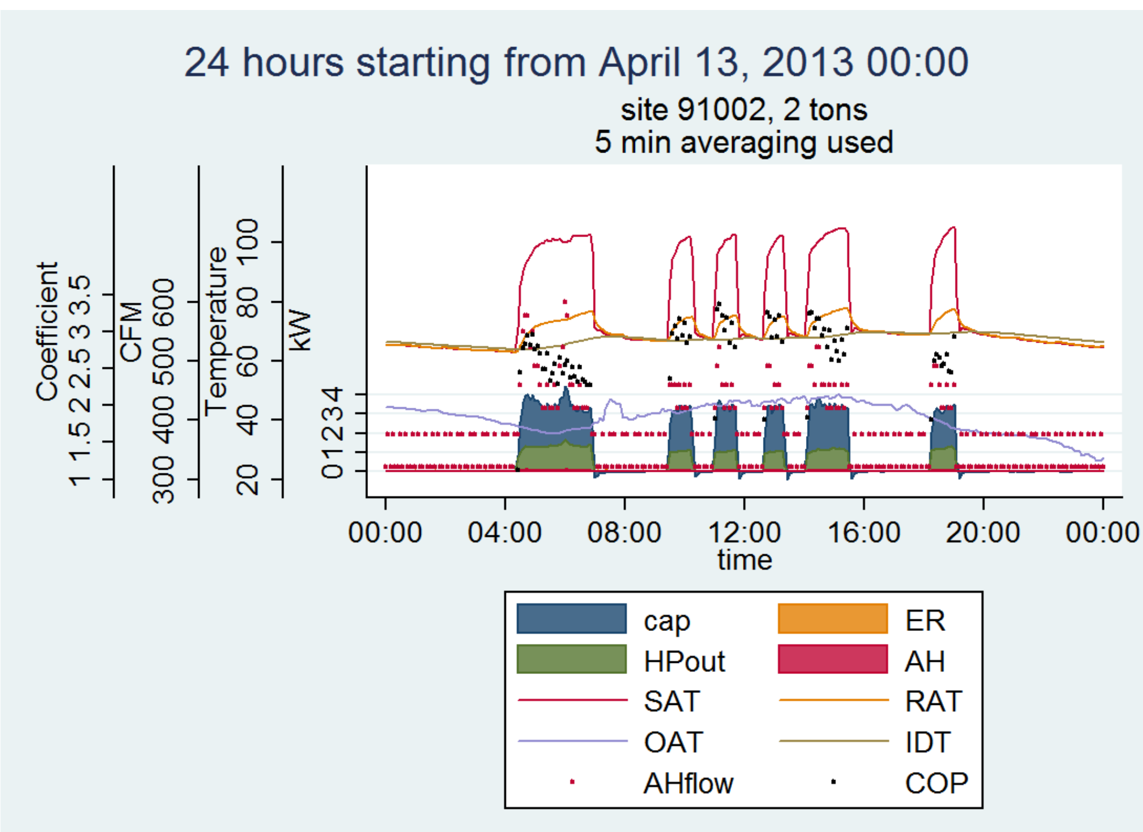
OAT
IDT
AHflow
COP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site 91002, 2 tons
5 min averaging used
Coefficient
1 1.5 2 2.5 3 3.5
CFM
300 400 500 600
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kW
01234
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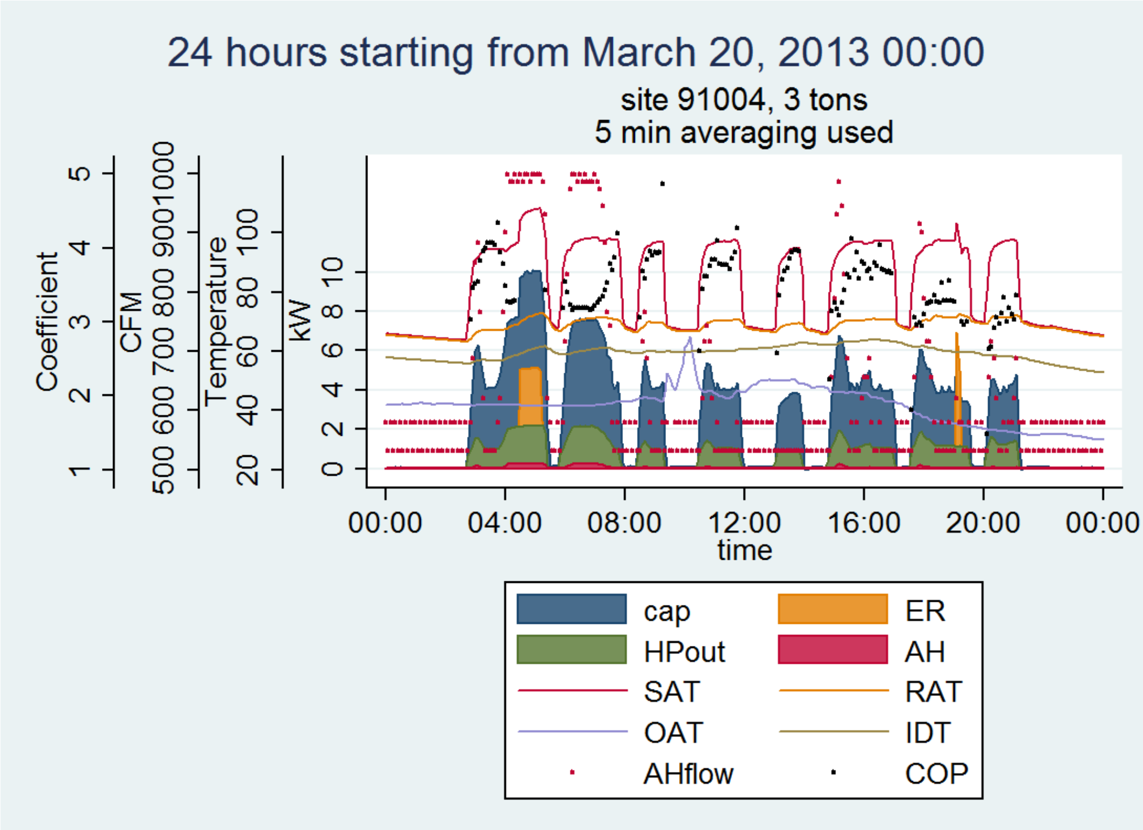
OAT
IDT
AHflow
COP


24 hours starting from March 20, 2013 00:00
site 91004, 3 tons
5 min averaging used
Coefficient
1 2 3 4 5
CFM
500 600 700 800 900 1000
Temperature
20 40 60 80 100
kW
0 2 4 6 8 10
00:00 04:00 08:00 12:00 16:00 20:00 00:00
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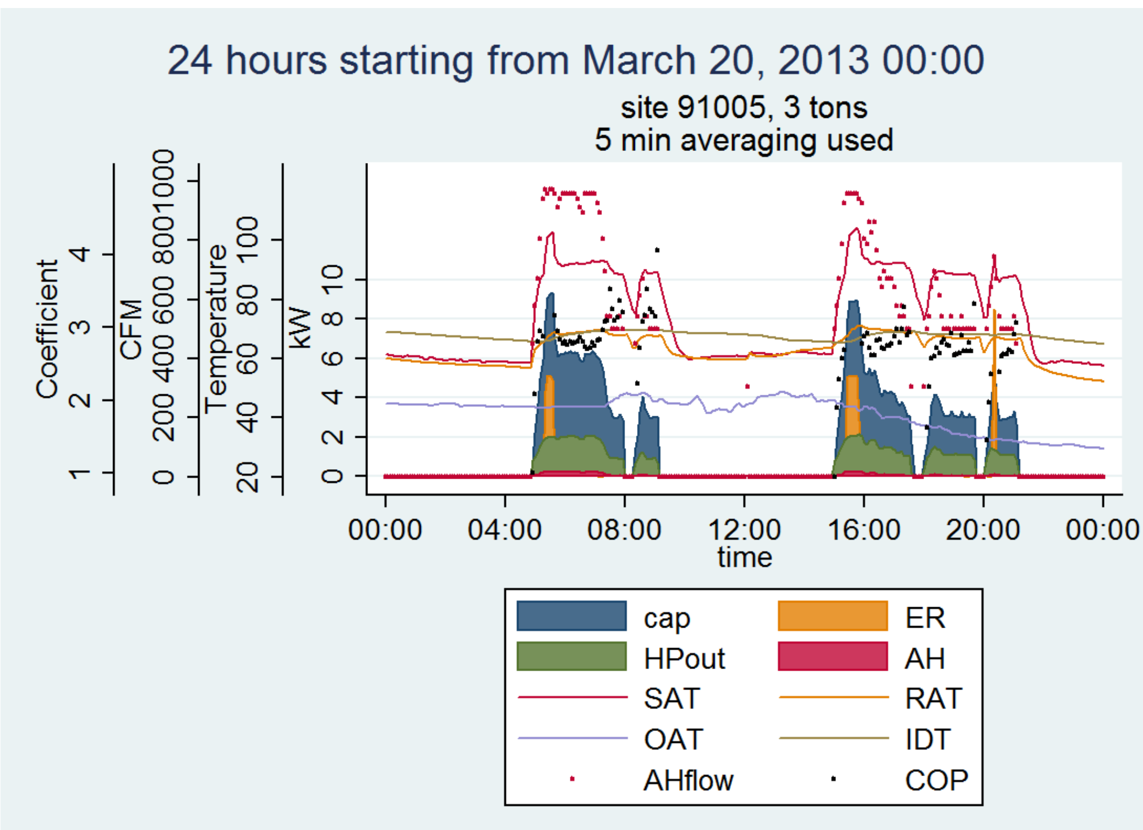
OAT
IDT
AHflow
COP


24 hours starting from March 20, 2013 00:00
site 91005, 3 tons
5 min averaging used
Coefficient
1 2 3 4
CFM
0 200 400 600 800 1000
Temperature
20 40 60 80 100
kW
0 2 4 6 8 10
00:00 04:00 08:00 12:00 16:00 20:00 00:00
time
cap
ER
HPout
AH
SAT
RAT
OAT
IDT
AHflow
COP