Performance Testing of Variable Capacity Heat Pumps in the Pacific Northwest

Robert Davis and Ben Larson, Ecotope, Inc.

ABSTRACT

In recent years, heat pump manufacturers have expanded their line of variable capacity heat pumps (VCHPs) to integrate with *ducted* house heating systems (versus ductless heat pumps, which have received greater attention). These heat pumps are an evolution of fixed output or two stage heat pumps, and vary their output--both compressor and air handler speed--in response to building heating and cooling load. Currently available equipment carries heating seasonal performance factor (HSPF) ratings of 10-13, which are well above the minimum federal standard (HSPF 8.2 as of January 1, 2015, while this research was conducted in 2013).

The research described in this paper incorporates both detailed house and duct measurements and eight weeks of heating season monitoring and evaluates the capacity and efficiency of the combined heat pump and duct system. Quantification of duct losses is especially challenging in this situation, and duct losses are larger for variable speed systems because indoor unit airflow varies with load. Research was carried out in a mild continental climate, which resulted in a robust range of outdoor temperatures and variable heating loads. Also, parallel lab testing work on similar technology, as well as deployment of simulation tools, enabled an extension of short-term results to estimates of whole heating-season performance.

An additional finding from the project reinforced the need to pay close attention to system (capacity) sizing. This is especially true if this heat pump is used in concert with controls that limit compressor operation or fail to limit auxiliary (electric resistance) heat operation.

Introduction

Bonneville Power Administration (BPA) has offered, through its member utilities, a heat pump program for over 10 years. This program, initially 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 motor-driven compressors that can be used in ductless or ducted settings. Nominal HSPFs for these systems often exceed 10. 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 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), and Francisco et al. (2004). 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.

Research Methods and Site Characteristics

Summary

This project relied on field measurement and simulation, as well as incorporation of parallel laboratory testing done by others. Primary goals of the field testing were to take sufficient, detailed measurements over a range of environmental conditions (a varied heating load) in order to map performance. This map could be compared with laboratory data, which had recently been collected via work conducted by the Electric Power Research Institute (EPRI). Further, duct performance measurements would be made so complete system – heat pump and duct – efficiency could be simulated. 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.

Field Site Selection

Ecotope selected the central Oregon area to recruit field sites because it offered a populous cluster of the desired systems and it is located in International Energy Conservation Code (IECC) Climate Zone 5, which would offer a desirable temperature profile over the monitoring period. Table 1 summarizes salient characteristics of the sites. All houses but one were site-built homes. All but site 91004 had some ducts located in unheated buffer spaces such as crawlspaces or attics. (Table 3 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. All heat pumps used electric resistance elements for auxiliary heat. In a small survey of sites, it was fortunate to get a wide range of characteristics to study.

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	2	crawl & 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

*all sites single story except 91004 (2 story)

Metering Design

The metering plan included collection of heat pump operating data (electricity and temperature measurements) that extended over an approximately 8 week period. Data were collected on 5 minute logging intervals. Temperatures of supply and return air were collected and system airflow was determined through a combination of one-time tests and logging of air handler power.

All longer-term data were collected with loggers that included cellular data connectivity. The channels that were measured at each of the six sites were:

[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 to 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.

- 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 true RMS 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 (a single point for each)
- Supply duct buffer space temperature (to allow finer tuning of the SEEM duct model)
- Main living space temperature (via an independent temperature logger)

Data streams were checked regularly for anomalous values. Raw data were transferred periodically from the loggers into comma separated variable files and then fed into the R and Stata software packages for analysis.

House and Duct 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. Blower door tests were conducted so that seasonal and peak estimates of air leakage could be made using SEEM's infiltration module.

Table 2. Field Site Characteristics

Site ID	Heated Floor Area (ft^2)	Shell UA (BTU/hr-°F)	Total House Heat Loss Rate (BTU/hr°F)	Blower door leakage at 50 Pa (SCFM)	Average Heating Season Infiltration† (ACH)
91001	2694	692	896	4605	0.34
91002	1394	227	283	1326	0.22
91004	2663	864	1126	3849	0.46
91005	2515	421	580	2501	0.30
91009	1860	378	502	2178	0.33
91010	2660	509	632	1580	0.18

†Average infiltration over the heating season as calculated by SEEM using the tested leakage. Includes all sources: natural infiltration, duct leakage, and mechanical ventilation.

Duct characteristics (Table 3) are integral to calculating distribution efficiency. 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). 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 3. Duct Characteristics – Location, Surface Area, and Insulation

Site ID	Supply Ducts			Return Ducts		
	Location	Surface Area	Insulation	Location	Surface Area	Insulation
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 4 shows crucial information on the duct tightness and system airflow. 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 created 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. Once complete, static pressures in both return and supply side can be imputed at other flows and supply and return leakage fractions can be determined.

Table 3. 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)
91001	231	151	943, 1061	18	-84.5
91002	276	275	688, 774	16	-21
91004	n/a	n/a	889, 1000	21.5	-48
91005	273	131	924, 1040	26	-72.5
91009	329	208	1340, 1395	50	-139
91010	148	131	1271, 1430	116	-158

* 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) means the density is about 89% of the density of standard air. The air handler flow values show both the local CFM and the standard CFM (SCFM).
‡ "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).

House Heat Loss Rate and Heat Pump Size (Nominal Output Capacity)

The heat loss rate at design conditions (Table 2), 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. The contribution from duct losses must also be included since the duct effect will increase the heating load on the equipment. This calculation is critical to energy use since the balance point temperature will have direct bearing on the overall coefficient of performance (COP) of the system. In all these houses, any heating load not met by the heat pump will be supplied by the auxiliary electric resistance heat. The auxiliary heat source is typically sized so that it can meet all of the heating needs of the house if the heat pump compressor is non-functional.

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 1 shows the graphical representation of the balance point; note this graphic applies to a site where the sizing was relatively aggressive and therefore the heat pump should be able to keep the house comfortable under almost all outdoor conditions.

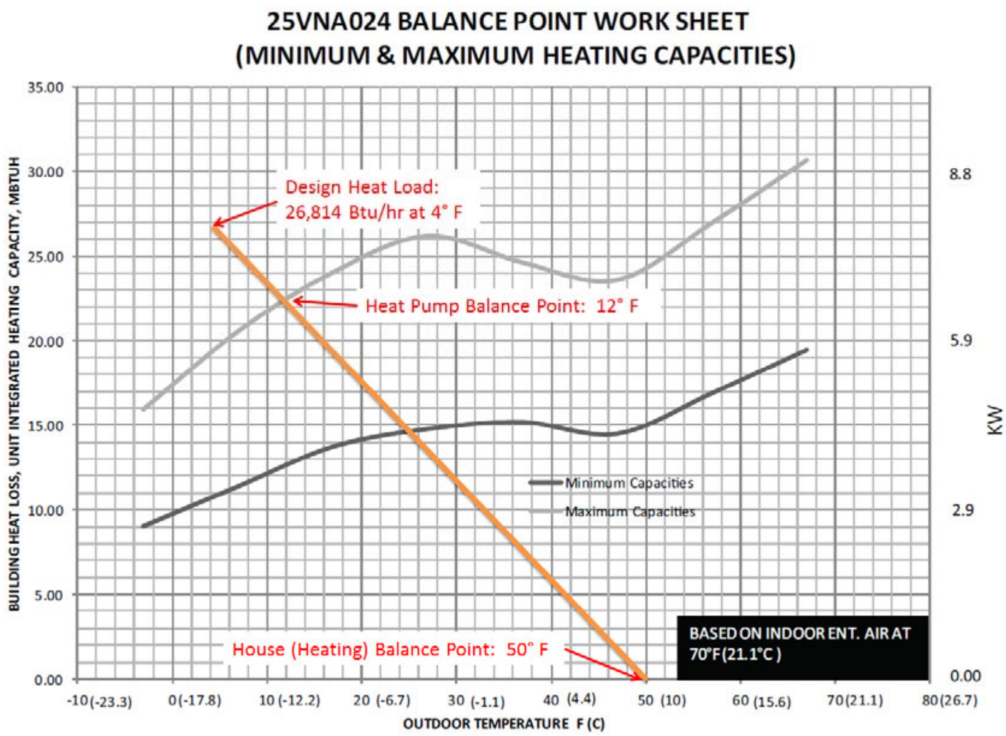

Figure 1. 2 Ton Heat VCHP Heat Pump Balance Point Chart. *Source*: Carrier 2012.

Table 5 shows the heat pump balance point temperatures for all sites. In some cases, such as one shown above, 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. Site 91004 is one of the two highest users of resistance heat in the study. This heat pump is simply sized too small to heat the house as efficiently as it might at this site.

Table 4. Heat Pump Balance Point Inputs and Result (Bend, OR design temp = 4° F)

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	66,391	17	4

†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

Results

Metering Period Outdoor Temperature

During the field monitoring period, the outdoor temperatures in the Bend, Oregon region ranged from 15° F to 70° F. Figure 2 combines operating mode data from all six sites as a function of outdoor temperature. As expected, the auxiliary heat runtime is heavily skewed to the lowest temperatures where it is needed to make up extra heating capacity, especially at sites where the heat pump is undersized. Compressor on-time is relatively even across the total hours; this type of compressor is designed to run at even very modest part loads.

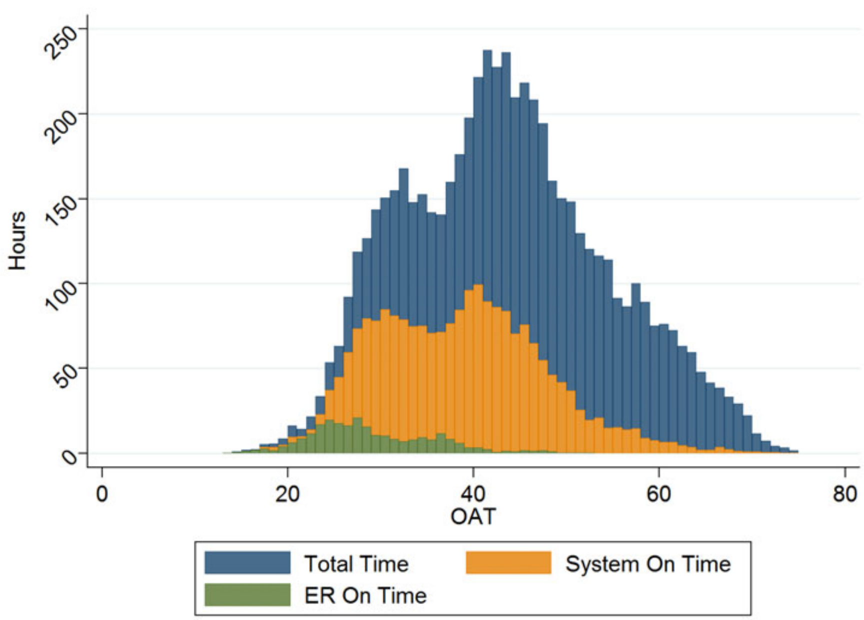

Figure 2. Heating System Runtime vs Outdoor Temperature (‘System’ includes both VCHP and ER while ‘ER’ equals indoor unit auxiliary electric resistance heat)

Metered Heating Energy Use

For five of the eight week monitoring period, all of the data loggers were installed and operating simultaneously at every site. All the sites were within 30 miles of one another, meaning they all experienced similar weather. The 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 6 lists 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 'Uutdoor 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 air circulation-only. (That is, 'Total Input' applies to heating periods only.

Table 6. 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)
91001	38.4	590	45	100	716
91002	39.3	221	30	56	277
91004	40.9	698	58	465	1185
91005	39.4	310	32	547	861
91009	38.2	591	35	58	662
91010	37.3	525	112	103	706

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. Table 7 shows that for all but two sites, the 'ER heat output 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 COP of 1).Part of the explanation for the excessive ER usage comes from the lack of outdoor temperature-based electric resistance heat lockout controls. Site 91001 does not have an element lockout control, either, but its resistance heat use is low. The metered indoor temperature at site 91001 revealed the occupants do not use a temperature setback. In contrast, sites 91004 and 91005 used a nighttime setback. Consequently, when coming off the nighttime setback, those two sites use resistance heat to make up the extra load.

Table 7 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 decreased the overall COP (especially at site 91005).

Table 5. Metered & Calculated Outputs – Common Five Week Period

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

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 – how it changes its compressor and air handler speed. 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 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; we divided the outdoor temperature into 5°F wide bins and assigned minimum and maximum values within those bins. Figure 3 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 outdoor temperature decreases.

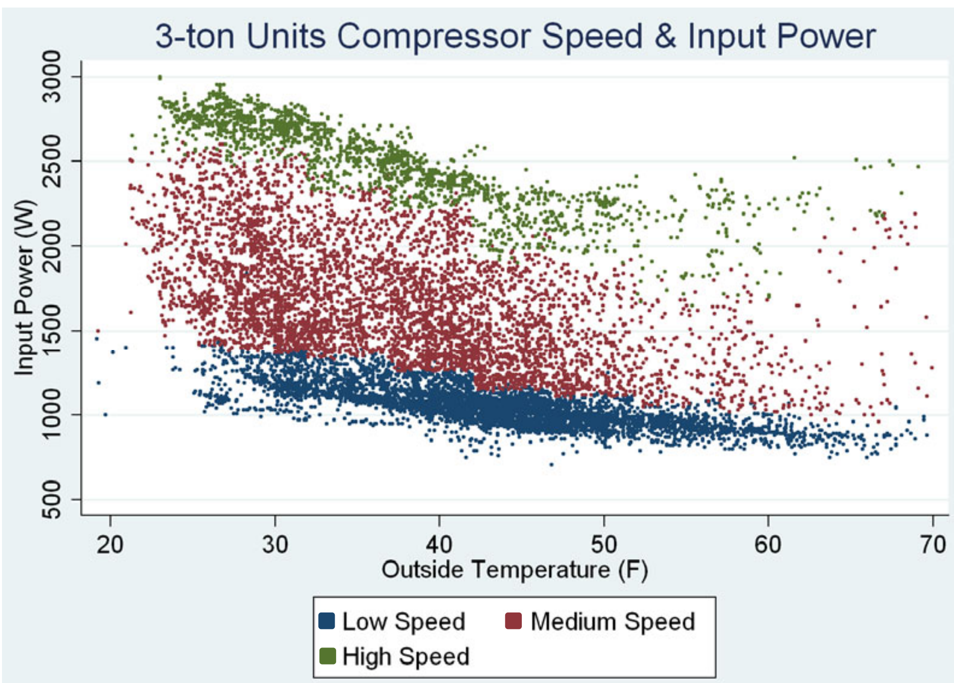

Figure 3. 3-ton Equipment Input Power vs. Outside Temperature

A further critical research question this study sought to answer was how the variable speed compressor and 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 mini-split 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 (Ecotope 2011).

To categorize the fan speed, we referenced the equipment specification sheet which lists minimum and maximum flows for each equipment size for either EFFICIENCY or COMFORT mode settings (Carrier 2012). 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 4 shows the airflow and input power relationship versus outside temperature for 3-ton systems. The figure is similar to Figure 3 except the color coding now represents the indoor air handler flow: low is blue, medium is red, and high is green. The graph is somewhat deceptive at the low input power settings because the red dots overprint many of the blue dots. Still, the pattern is clear. The high fan speeds correlate with the high input powers. The low fan speeds correlate with the low input powers. The medium fan speeds run with both medium and low input powers.

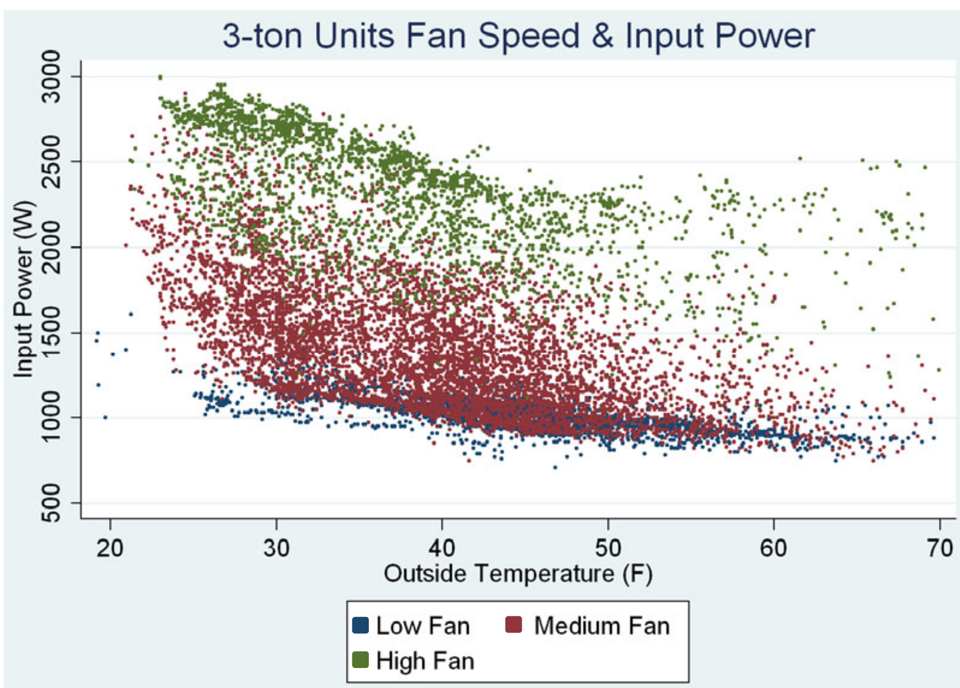

Figure 4. 3-ton Equipment Airflow and Input Power vs. Outside Temperature

A detailed comparison of the operating modes showed that 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 speeds. For low compressor speed, both low and medium flows are prevalent.

Distribution Efficiency

The duct system serves the necessary function of transporting heat from the indoor unit 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 identical between all the systems.

The measurement plan collected enough information to provide direct, reliable, and accurate estimates of duct losses. There are two main components to understanding the duct losses: air leakage out of and in to 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 air. The heat loss due to conduction is driven by the difference between the duct and buffer space air temperatures.

The field technicians performed a complete duct audit on site to characterize the two components of duct losses. Duct R-values were summarized in Table 3 and results of the duct tightness tests were given previously in Table 4. Using these test results, and additional measurements of operating static pressure, a leakage fraction for both the supply and return sides at the operating conditions of the duct system was calculated using the flow exponents and coefficients associated with the measurements at each site.

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. As air handler flow decreases so does the absolute leakage.

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. 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). Figure 5 shows the relationship that lower system flows lead to larger percentage losses.

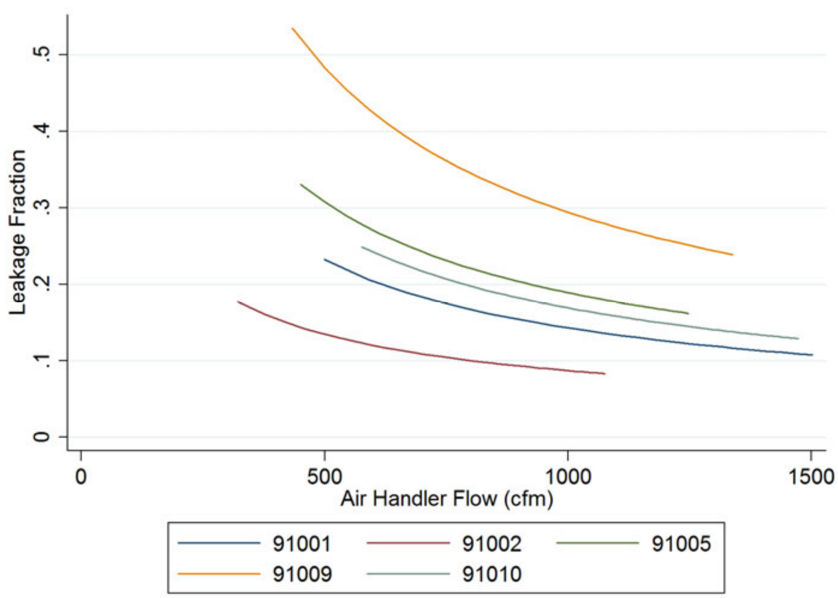

Figure 5. Leakage Fraction vs. System Airflow – All Sites

To paint the full picture of the impacts of duct leakage and conduction, we employ ASHRAE Standard 152 (ASHRAE 2004) and calculate the distribution efficiency at each time interval of our metered data. The distribution system efficiency is calculated using the following equation:

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

The α values are the leakage fractions for the supply side and return side. These values are the complements of the previously calculated leakage fractions (e.g.α_{supply} = 1 – Supply Leakage Fraction or 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 factors for the supply side and return side.

Table 8 illustrates the significance of the distribution system by showing the results of the duct efficiency calculations. The values shown are the averages over the common five week metering period. Houses with ducts inside have distribution efficiency (DE) of 1 and all the others incur losses. For example at site 91001, only 65% of all heat present at the air handler is delivered to heat the interior of the house. In addition to the duct air leakage, with long and expansive duct runs at many sites, the conduction losses contribute significantly to the low distribution efficiencies.

Table 6. Measured and Calculated Average Distribution System Efficiency

Site ID	System Airflow (CFM)	Supply Buffer Zone Temp (°F)	Supply Leakage Fraction	Return Leakage Fraction	Distribution Efficiency
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

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, 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 6 shows the steady-state system efficiencies for the lab and field measurements of a 2-ton system side-by-side (Hunt 2013). The figure shows agreement between the two measurements. For example, for 17°F temperatures in the lab, the COP ranged around 2. Likewise, examining the 20°F bins in the field shows 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 gives us confidence in our field measurements. Further important findings from the lab showed that the fan and compressor speeds generally tracked one another through the operating modes.

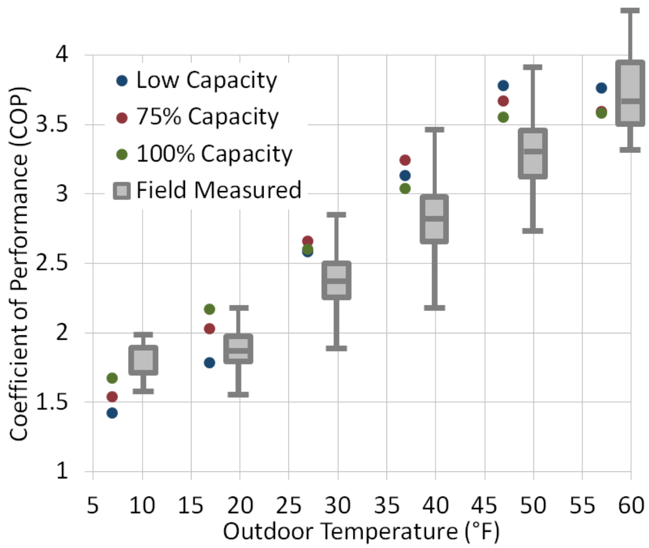

Figure 6. Lab and Field COPs Compared – 2-ton System .
Lab Data Source: Hunt 2013. *Field Data Source*: This study.

In considering the lab and field results, we concluded that performance curves of the heat pump itself can largely be determined 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 information. As we were able to observe performance variations of the 2, 3, and 4- ton systems in the field, to develop a fully accurate performance map in a lab setting, more than one system size should be tested.

What was not clear from the lab testing 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 can provide the needed insight in to operation. Alternatively, field measurements might be avoided here if the manufacturer provided detailed control strategy documentation.

Additional information obtainable only in the field showed just how the system interacts with the auxiliary resistance heat. This depends on outdoor temperature based resistance heat lockout 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.

Modeled Annual Heat Pump Efficiency (Distribution Efficiency Included)

To create an annual estimate of heat pump efficiency and duct losses, Ecotope used the SEEM energy use simulation program. Based on the data collected in this project, Ecotope updated the simulation to include a VCHP performance curve which 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 table shows the modeled efficiency (COP) for the equipment (the 'box' alone), and the combined equipment and distribution efficiency. The simulation was run with an electric furnace, a single speed heat pump with HSPF 7.9 (close to the current federal minimum HSPF of 8.2), and the new variable speed heat pump model. The left half of Table 9 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. The right half of Table 9 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 that, on average, the variable capacity heat pump is 25-30% more efficient than its single-speed counterpart.

Table 9. Modeled Equipment Efficiency (COP)

	Equipment Efficiency			Overall Efficiency		
Site	Electric Furnace, COP=1	Single Speed Heat Pump, HSPF=7.9	VCHP	Electric Furnace, COP=1	Single Speed Heat Pump, HSPF=7.9	VCHP
91001	1.00	1.99	2.46	0.72	1.33	1.61
91002	1.00	2.35	2.84	0.78	1.63	2.04
91004	1.00	2.12	2.59	1.00	2.12	2.59
91005	1.00	2.07	2.53	0.69	1.23	1.49
91009	1.00	2.48	3.05	0.64	1.28	1.58
91010	1.00	2.20	2.76	0.73	1.45	1.77

Conclusions

The residential variable capacity heat pump field study evaluated the energy performance of six VCHP heat pumps installed in houses in central Oregon (IECC Zone 5). 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 a model of VCHP behavior.

BPA funded a useful lab study of the same VCHPs observed in the field (Hunt 2013). The lab and field studies reinforce each other's findings in regards to the compressor and air handler operating speed combinations. The lab measurements provided detailed and accurate measurements of equipment input and output which agreed with the field work

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. 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 heat pumps. 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.

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