It Sounded Good on Paper: Field Performance of Residential Heat Pumps in the Pacific Northwest

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ABSTRACT

Air-source heat pumps have been touted as energy conservation measures in the Pacific Northwest for at least two decades, and many regional utilities have subsidized their installation. However, no comprehensive study of heat pump installations has been done and several regional actors, including the Bonneville Power Administration, electric utilities, and State Energy Offices, have long been interested in the assumptions that underpin energy savings estimates, especially assumptions regarding system sizing ("tons") and control of auxiliary (electric resistance) heat.

A total of 160 regional heat pump installations were evaluated in this project as part of a larger study that included a billing analysis and heat pump bench testing. The field evaluation utilized a detailed protocol and included a review of system controls, evaluation of duct system performance, evaluation of system airflow and refrigerant charge, and estimation of building shell heating load and its relationship to heat pump capacity.

Results from this work were used to revise estimates of regional heat pump energy usage and to inform ongoing efforts to improve field installation protocols that are required as prerequisites to receive utility incentives.

Introduction

This paper summarizes the field evaluation of heat pumps installed in four areas of the Pacific Northwest. A total of 160 homes were visited in this study, as part of a larger study that included billing analysis of about 1,000 homes and heat pump bench testing at an ARI-certified laboratory (Baylon, et al, 2005). Homes visited were heated and cooled by "base case" heat pump systems; that is, systems NOT installed as part of utility incentive programs. This was done to establish the baseline performance of systems against which energy consumption of newer systems could be estimated. The focus of the work in this study was on the heating performance, as heating energy usage is typically ten times that of cooling usage in most Pacific Northwest climates.

The field protocol required two separate visits. The first visit evaluated building shell heat loss (including air leakage) and the duct system. The second visit was conducted by a qualified refrigeration technician and included a review of nominal system capacity ("tons"), system controls, system airflow, and refrigerant charge.

Results from this study were designed to inform both the energy savings calculations used in utility sponsored programs (Bonneville Power Administration's Conservation and Renewables Discount, C&RD) and the specifications used to qualify heat pump installations for incentives under this program. This work was adapted to develop installation and quality assurance testing guidelines for use in improving the effectiveness of this program.

Field Sample Selection

Field sites were selected in four parts of the Pacific Northwest to represent baseline heat pump practice in each sub-region. Given the sample size, we assert that these are at least very good approximations of heat pump installation practices in those areas.

The four regions selected were Kitsap County (west Puget Sound), Bend (central Oregon, the coldest site), Yakima/Walla Walla (south/central Washington), and Clark County (just north of Portland). Individual sites were selected randomly from utility-provided lists or from commercially available lists of heat pump customers. Homeowners were asked a short set of questions. Those expressing interest were asked if they would be interested in a visit and were told they would receive a $50 incentive if they participated. Approximately 40 homes in each area were recruited, for a starting possible total of 161 field audits.

In Clark County, the control group sample was drawn from a utility list of homes participating in their (non-C&RD) heat pump program. These systems were not installed to C&RD requirements. However, Clark County PUD has, for some time, operated a heat pump program which includes some design and installation requirements. As is noted below, these requirements resulted in some marked differences in nominal system efficiency.

The final number of field sites which received a full review (house/duct audit and heat pump service check) was 126. Because the heat pump review was best carried out during warmer times of the year (to enable a more accurate assessment of refrigerant charge), most of the heat pump visits were done in spring/early summer 2005. This time lag meant some homeowners either were not interested in having their heat pump looked at, had hired someone to look at it, or could not agree on a time for the review.

Field Protocol Elements

The primary housing characteristics that were reviewed included house size, age, heat loss rate (as determined by assessing component areas and insulation levels/window types while on site), duct location and insulation, and presence/operation of a mechanical ventilation system. These direct observations were supplemented by a blower door test, a Duct Blaster® test (duct leakage), system airflow test, and a test of the system operating static pressures. Airflow and static pressure measurements were taken in order to evaluate heat pump capacity and to normalize duct leakage readings (by expressing them as supply and return leakage fractions).

The heat pump protocol was developed to inspect or measure the following major items of interest: nameplate information (which relates to "box spec" efficiency), settings and operability of system controls (indoor and outdoor (cut-out) thermostat, low ambient compressor cut-out control) airflow across the indoor coil (measured with a calibrated flow plate), and refrigerant charge.

Control strategies were seen as critical in estimating system performance. Any amount of auxiliary heat usage cuts into the nominal seasonal efficiency of the heat pump. Some auxiliary heat usage is assumed in the seasonal rating, but the amount can be greatly increased by installer preference (low-ambient cut-out set above 20°F, outdoor thermostat not installed, or first stage element operation) and homeowner decisions (thermostat fiddling). The field study enabled an informed review of assumptions that had been used in original estimation of C&RD incentives. These assumptions were based on limited field study and anecdotal data; in fact, the paucity of these data was one of the major spurs to carrying out this study.

Field Audit Results

In the following discussion, house physical characteristics are first reported in order to set the stage for a discussion of duct system distribution efficiency, heat pump field findings, and heating energy usage.

Shell Heat Loss and Air Tightness

Table 1 shows the distribution of house size and type/foundation. The significance of foundation type is that it affects duct distribution efficiency. For the most part, these values are consistent with other survey data collected in the past for these regions. The exception is overall house size, which appears to be somewhat larger than observed in previous studies, while somewhat smaller than the more recent new construction reviews done for the region (Baylon, et al., 2001). The proportion of homes with basements in the Yakima and Bend areas seem to be somewhat lower than seen in previous new construction reviews.

Table 1. House Type and Area

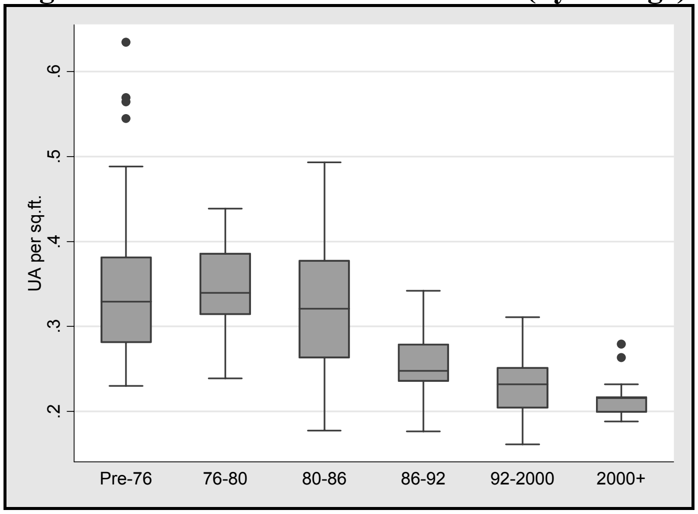
	N	House Area	House Type	
			Manuf.	Basement
		ft^2		
Bend	41	2033	14.6	7.3
Clark	40	1873	10.0	17.5
Kitsap	40	2301	2.5	15.0
Yakima	40	2155	10.0	12.5
All	161	2090	9.3	13.0

Table 2 shows the distribution of heat-loss rate (UA) normalized (by heated floor area) across the various vintages. Figure 1 includes the same data summarized in box plots that illustrate the medians and quartiles for each region. These UA data include contributions from air leakage, which was determined by using a two-point blower door test. Heat loss rate rather than heat gain was the focus in this study given the dominant heating loads throughout the Pacific Northwest. There is a clear link between age of the home and a higher UA. This can and should be interpreted as due to the impact of regional energy codes, particularly after 1985.

Table 2. Normalized Heat Loss Rate (by vintage)

	Heat Loss Rate (UA/ft^2)		n
	Mean	Std. Dev.	
Pre-76	.287	.087	47
76-80	.260	.041	25
80-86	.247	.053	20
86-92	.194	.034	22
92-2000	.182	.030	31
2000+	.170	.027	13
Total	.234	.073	158

Figure 1. Normalized Heat Loss Rate (by vintage)



Overall, even with the incidence of weatherization and home remodel upgrades, the graph shows homes of more recent vintages have a 30% lower heat-loss rate per square foot than homes built prior to 1985. A similar 30% lower rate for air leakage in the building shell was observed in homes built in 1986 or later.

Duct System Results

In order to evaluate the overall duct system efficiency, both leakage and conduction were taken into account. Of primary concern was the location of ducts in unheated areas (buffer spaces) where heat loss would negatively impact the distribution efficiency of the duct system. In all current codes, duct insulation is required in such applications and, nominally, duct sealing is also required by state energy codes.

Field tests were conducted using the Duct Blaster® and blower door to simultaneously pressurize the ducts and the home. The result was a leakage estimate that netted out leakage to the inside of the home. Field auditors were required to test the flow exponent derived from the two-point duct leakage test to determine the validity of the test. In some cases, despite repeating the test, the flow exponent indicated an invalid test. These cases were excluded from the analysis and result in a diminished sub-sample size.

Duct systems in the field study did not follow the same pattern as the heat loss rate of the building nor the infiltration rates. Table 3 summarizes the duct UA and the supply and return leakage fraction for each of the samples in the region. These components determine the duct delivery efficiency (i.e., the fraction of heat produced at the air handler that is delivered to the space). The duct efficiency is a function of the supply/leakage fraction (the amount of air leaking from the supply systems out to the unheated buffer spaces), the return/leakage fraction (the amount of air leaking into the return system from unheated and unconditioned areas), and the conduction losses from both supply and return ducts in these areas. To develop an estimate of the duct efficiency, these parameters were combined in a duct simulation model which accounted for heat pump operation as well as duct location.

The model (SEEM) was developed by Larry Palmiter of Ecotope and implements the ASHRAE Standard 152 duct calculation protocol (ASHRAE, 2003). This model also takes into account the impact of the heat pump operation on the duct efficiency. The duct efficiency is shown for the sample as a whole, and for the subset of the sample with ducts in unconditioned buffer spaces. The characteristics observed in the individual sites were then summarized as an aggregate for each home. The duct efficiencies take into account that a heat pump is used to supply both heating and cooling. Considering duct efficiency alone only reflects the amount of heat loss associated with the duct systems once the heat pump is operating. Off-cycle losses are not included in this calculation.

Table 3. Duct UA and Leakage Rates

	n	Leakage (% of airflow)		% Interior Ducts		Distribution Efficiency (%)	
		Supply	Return	Supply	Return	Entire Sample	Sites with no interior ducts
Bend	28	14.5	10.9	2.4	24.4	64.0	62.7
Clark	34	9.7	11.3	15.0	30.0	82.2	78.4
Kitsap	40	14.1	14.9	22.5	30.0	74.7	70.2
Yakima	33	12.6	13.1	12.8	36.8	70.2	66.1
Total	135	12.8	12.7	13.1	30.2	73.3	69.4

Figure 2 and Table 4 show the distribution of duct efficiency for the population by vintage. Homes with ducts inside the conditioned shell were excluded. In the table, the same data is summarized as means rather than the medians shown in the graphic. Note that, for the SEEM model (and for practical purposes), ducts that are located within the heated space have no significant heat loss or air leakage, since any leakage occurs within the heated space.

Figure 2. Distribution of Duct Efficiency (by vintage)

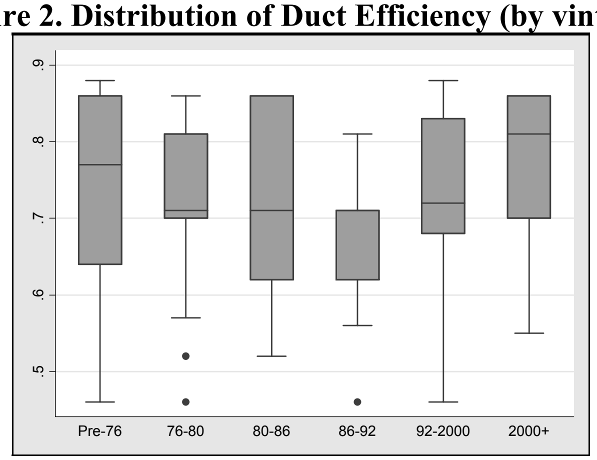


Table 4. Duct Delivery Efficiency (by vintage)

Vintage	Duct Efficiency Ratio		n
	Mean	Std. Dev.	
Pre-76	.701	.170	26
76-80	.686	.156	20
80-86	.675	.170	15
86-92	.599	.125	18
92-2000	.721	.153	24
2000+	.790	.189	10
Total	.691	.163	113

Figure 2 suggests that the distribution efficiency in older (pre-1980) homes was better than in homes built between about 1980 and about 1992. At about that time, the importance of ducts began to be more clearly recognized both in energy codes and utility practices. Thus, while some improvements have been made in duct efficiency over the past decade, the improvements have scarcely done more than return duct systems to the level of sealing and installation practice common prior to 1980.

Heat Pump Field Results

The most important findings of the heat pump field survey are listed below, in order of importance, in terms of impact on heating energy usage. Because of attrition between the characteristics survey and the field test of the heat pump equipment, the number of cases summarized was reduced from 161 to 127.

Controls

None of the field sites had compressor low-ambient cut-out enabled. Based on information gathered from installer interviews in 2001, it was believed that this practice was used by some installers to enhance comfort. However, in this work the field test included no instances of this practice, and added interviews with installers in other regions of the Northwest suggests that compressor cut-out is used rarely except in areas of eastern Washington, especially Spokane. C&RD calculations had assumed low-ambient cut-out is used in 10% of new installations. The effect of this assumption will be reduced in ongoing calculations.

About 35% of sites visited had an operating outdoor thermostat (ODT), which eliminates operation of auxiliary heat above a preset temperature (typically 35-40°F). About 75% of the Clark County PUD sites have ODTs; they also had a utility program which required an ODT during the time many of these sites were selected. The average outdoor thermostat setting found in the field was 40°F. About 2/3 of sites without ODTs had the extra wires needed to install an ODT without requiring new wiring runs, which means there is a potential for added savings without prohibitive cost for this measure.

About 80% of indoor thermostats are programmable units (with approximately 75% of these being 7-day units). About 80% of the 7-day model thermostats have an adaptive recovery mode, which is often marketed by installers as performing the same function as an outdoor thermostat. This is accurate in some cases, but if the homeowner performs manual adjustments of their heating setpoint (especially during morning warm-up) adaptive recovery is typically overridden. About 35% of homeowners said they often "turned up" the thermostat in cases where they didn't think the house was warming up as quickly as desired.

The median heating setpoint was 70°F and median setback temperature was 65°F. The average setback (when it was used) was 6.7°F, and 55% of systems had a setback greater than 5°F from the nominal heating setpoint.

In general, indoor thermostats behaved as expected. However, on a normal Stage 1 heating call (thermostat turned up by 1°F), 15% of thermostats activated a resistance element. Median element operation time was 5 minutes. It is hard to know the overall effect of intermittent backup heat usage during Stage 1 heating without detailed monitoring, but these short-term measurements suggest relatively limited operation.

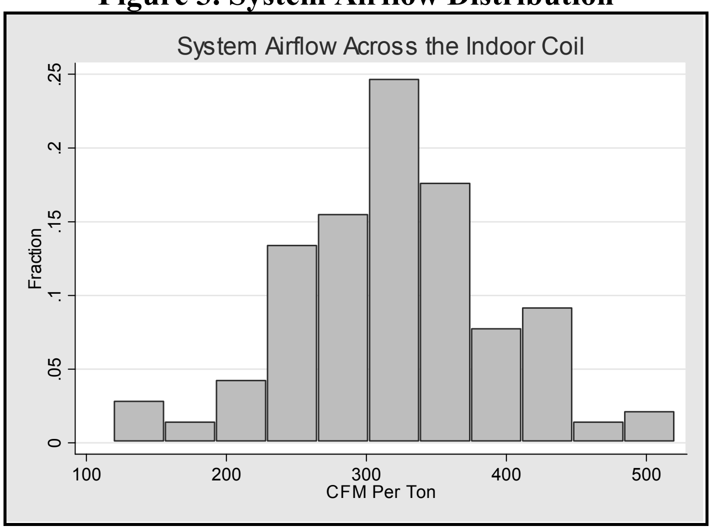
For the most part, current C&RD modeling assumptions do not assume elements operate in the first stage unless there is a specific installer choice. However, in sum, these assumptions have included more use of auxiliary heat than observed in this field study. The overall effect of revisions in assumptions will be to reduce the effect of auxiliary heat on annual heating energy usage.

Indoor Unit Airflow

System airflow (across the indoor coil) was measured with a TruFlow® air handler meter. The median value was 326 CFM/ton, based on the heating capacity of the outdoor unit. This heating capacity is listed in tons, with one ton equal to 12,000 BTU/hr of capacity at a standard rating point of 47°F. The mean value was 327 CFM/ton. Sites were screened out if values fell below 175 CFM/ton or above 500 CFM/ton. The remaining 135 sites with valid combined flow and outdoor unit size are summarized in Figure 3. The lower quartile of flows starts at 292 CFM/ton. A significant number (20%) of systems are moving less than 300 CFM/ton.

The current C&RD modeling assumptions assume that heat pump equipment meets the manufacturers' specifications. This is generally between 380 CFM/ton and 420 CFM/ton. It appears from this data that some adjustment in this assumption would be appropriate.

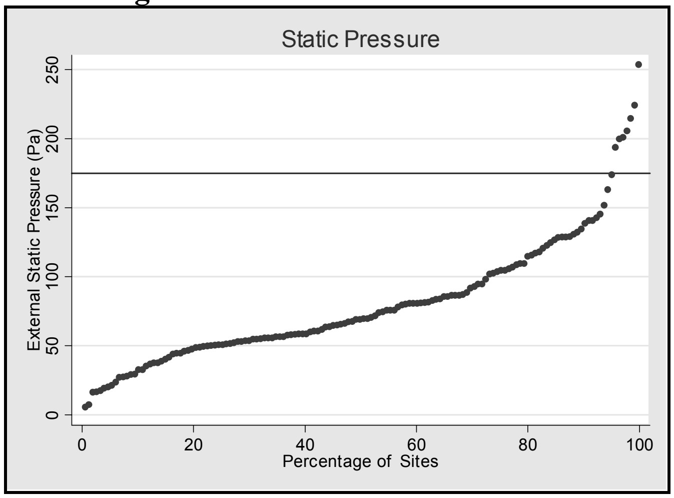
Figure 3. System Airflow Distribution



Many of the newer heat pump systems use a variable speed indoor fan and an electronically commutated motor (ECM), which is designed to deliver consistent airflow over a wide range of entering and leaving static pressures. At least 30 cases (about 24%) were identified with these motors in this study; the median flow for these cases is 337 CFM/ton.

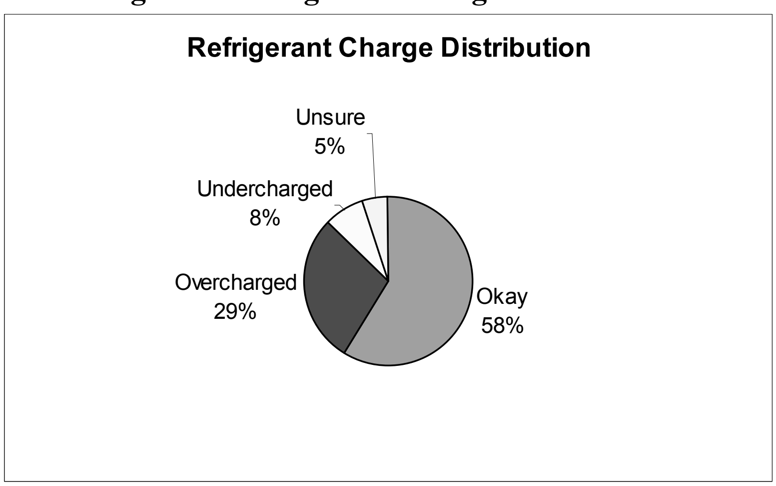
Normal operating supply and return static pressures were measured as part of system airflow and duct leakage fraction evaluation. The sum of the absolute value of these numbers is a reasonable estimate of the system external static pressure. Generally, external static pressure of more than about 175 Pa is an indicator of reduced system airflow (PSC fans) or increased energy usage (ECM fans). In this study, the distribution of external static pressure was such that only a few cases came in above 175 Pa, as shown in Figure 4.

Figure 4. External Static Pressure



Refrigerant charge was evaluated using a combination of tests (depending on season). In some cases, both heating and cooling performance tests could be run. Service techs were encouraged to use manufacturer's information, look-up tables provided by Ecotope, and their own experience in assessing whether the charge was correct. Ecotope reviewed each site to see what the technician had decided and to provide a final ruling. In the heating season, the evaluation started with a comparison of the measured temperature split with the expected split based on the air flow rate (which was also measured at each site). In cooling season, system superheat or subcooling was compared with expected values (when available). The distribution of results is shown in Figure 5.

Figure 5. Refrigerant Charge Distribution



Only about 8% of systems were undercharged; they were so designated if the data suggested a major deviation from expected performance values. About 30% of systems were overcharged. About 60% of cases had the correct charge, and a small fraction of systems could not be evaluated because of confusing or partially missing data. Four systems could not be evaluated when the technician arrived. In two cases, there had been a serious refrigerant leak. In two other cases, the compressor had a serious mechanical problem and would not turn on or would not run long enough to allow an evaluation.

System Sizing and Nominal Heat Pump Efficiency

Throughout the C&RD calculation process, there are efforts to describe particular strategies for sizing the heat pump and for controlling the heat pump once installed. The field study (and related work) attempted to establish control packages used in non-C&RD installations (so that operating assumptions could be revised, if necessary), and to gain insights that could result in optimizing performance of C&RD control strategies.

A review of heat pump sizing suggests this part of the installation practice is somewhat controversial, since oversizing can lead to cycling losses and undersizing can lead to excessive use of backup heat. This controversy is fueled by the fact that the actual nature of cycling losses is derived from equipment and computer simulations that were conducted on heat pumps that have not been made for more than 25 years (ARI, 2003). This is because manufacturers do not publish this data for heat pumps operating in heating mode (although it is typically published for the cooling mode). Thus, the assumptions made in the 1970s and 1980s remain the best available estimates of cycling losses.

A specification that limits the minimum size of the heat pump (i.e. larger installed heat pumps) in relatively cold conditions would tend to maximize savings. However, the first cost of the heat pump tonnage must be considered a significant factor in the cost-effectiveness of the installation.

The field audits were sufficiently detailed to establish the heat loss rate of the home. Using this estimate, it was possible to assess the sizing of the heat pump relative to the heating load. This comparison is very useful in estimating expected heating energy usage, since a system that is well-matched to the load will be expected to provide most or all heating energy with the vapor compression cycle (rather than auxiliary electric resistance heat).

In reviewing the heating load, some added assumptions were necessary. The most significant of these is the design temperature difference. This factor is multiplied by the house UA to generate the minimum equipment size needed to meet the heating requirements of the house. There are a wide variety of temperature differences that would be applicable across the sample. For this evaluation, we used 50°F for the western locations (Clark and Kitsap counties) and 70°F for sites in the Bend and Yakima regions. This calculation estimates the overall heat loss rate. The efficiency of the duct system modifies this calculation. The duct efficiency number generated from the audit is not typically available to the installer who is selecting the heat pump, but it has a real impact on the sizing estimate.

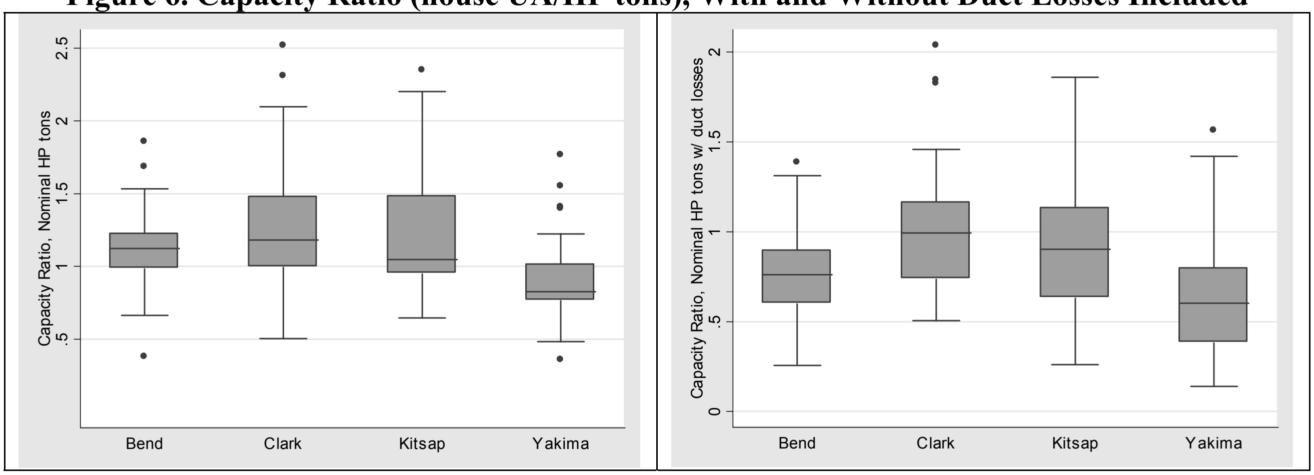
The rated heat pump capacity is published as part of the standard rating for all heat pumps and represents the output of the heat pump at 47°F outside temperature. This value is typically available even if more detailed manufacturer's specifications are not. Furthermore, the nominal size is often used by installers as a reliable method of determining the minimum size of the equipment to be installed. The problem with this value is that the heating load of the house at this temperature is less than half the load at design conditions. The strategy for sizing then usually involves increasing the nominal size of the heat pump to take this into account. Moreover, while the installer seldom has any detailed information on the duct efficiency, there is a general understanding that the size of the heat pump should be increased somewhat to take duct losses into account. Considering these factors, it would be reasonable to expect that the heat pumps installed in this sample would have a nominal size about 50% larger than the heat load calculated from the design temperature.

Given a heating load sizing criterion, the heat pump sizing observed in this sample is surprisingly small. When duct losses are taken into account, the sizing of the equipment is less than the calculated heating load. The comparison is shown in Figure 6.

About 60% of the installers interviewed suggested that the cooling sizing was the principle factor in their equipment selection. In the Pacific Northwest cooling climates, this would explain a smaller compressor. The cooling sizing data (including window characteristics and orientation), do indicate that such an approach would result in a slightly smaller compressor in the Clark and Kitsap regions, but a larger compressor in the Yakima and Bend regions.

Given the observed equipment sizes in the study, it would appear that a 30% increase in size would be required to meet the design heating load at about 30°F outside temperature. In many cases, additional capacity would be required to overcome the substantial duct losses. Such a change in sizing criterion would probably increase the cost of the heat pump installation by about $1,000 in the current market. It may be that current practice, at least in the four regions reviewed for this study, delivers a more cost-effective system overall for the homeowner (or at least reduces the first cost of the installation to an acceptable level). In the context of low utility rates, the installer and consumer may be making a sensible economic decision

Figure 6. Capacity Ratio (house UA/HP tons), With and Without Duct Losses Included



Heat Pump Performance Rating (HSPF)

Each heat pump manufacturer is required to rate their product with several standardized tests at various outdoor temperature bins. These tests are used to develop a factor which combines the results as a single rating point that allows equipment to be compared based on a single performance index. For the heating season (which is of primary interest in the Pacific Northwest), this index is the Heating Seasonal Performance Factor (HSPF). There are actually six separate climates for which the HSPF is calculated; however, the standard rating manual only uses one of these climates (zone 4) for the values published in rating manuals. Climate Zone 4 is closely related to the Bend climate. The other climates in this study are milder and should have better performance than the standard rating.

For most of the last 20 years, there has been a Federal Standard in place that regulated the minimum HSPF for heating equipment that could be produced and sold throughout the country.

This value was set at 6.8 BTU/Wh. In effect, this regulates the heat pumps to have an average seasonal COP of 2.0. The mean HSPF for our sample was somewhat higher. The HSPF ratings of the Clark County PUD installations were noticeably higher than the other regions, as shown in Table 5. This is probably the result of the Clark County PUD heat pump program, which encouraged installation of higher performance heat pumps.

Table 5. HSPF of Heat Pumps by Region

Region	n	Rated HSPF		Confidence Int. (95%)
		Mean	SD	
Bend	32	7.64	.546	.190
Clark	37	8.01	.509	.164
Kitsap	32	7.52	.564	.196
Yakima	22	7.44	.527	.220
Total	123	7.69	.576	.102
Total w/o Clark	86	7.55	.548	.116

Billing Analysis for the Field Sample

While the field sample was relatively small, we did conduct a detailed billing analysis for this group as a check on expected space conditioning usage. This analysis was confined to a single year's activities for which a complete billing record was available for the majority of the homes.

The analysis uses a median low bill methodology, which takes the middle of the three lowest bills to determine a base load and then assigns heating and cooling by month based on the size of this low bill. In this process, the total energy use for the year is recovered from the sum of base, heating, and cooling loads. The analysis allows for adjustments to the base load estimates that can account for the difficulties in evaluating any heat pump system for both heating and cooling. The results of this analysis are shown in Table 6.

The annual estimates become the basis for evaluating home performance against the relevant variables that might determine heating and cooling consumption; e.g., duct efficiency, heat pump efficiency, and overall building UA. The expected usage based on these inputs was found for each case in the study by running a separate SEEM model.

Table 6. Energy Use Estimates by Region and End Use

Area	n	kWh per Ft2 per Year			
		Heating	Cooling	Base	Total
Bend	29	4.92	0.78	6.07	11.77
Clark County	38	3.96	1.11	6.64	11.71
Kitsap	27	3.17	0.23	5.49	8.89
Yakima	28	4.65	1.37	5.89	11.91
Total	122	4.15	0.87	6.10	11.12

This analysis suggests an overall space conditioning energy usage of about 5 kWh/ft^2/yr. When this is reviewed in the context of the expected values based on local weather, building UA, and duct efficiency, the heating levels observed in the billing analysis are about 50% of what would be expected if the homes were heated with an electric forced air furnace. This implies a COP over the entire sample of about 2. While this value is less than would be expected if only the “box spec” of the heat pump were used, it does show the compression cycle is providing a substantial benefit to the homes in this study. However, the combination of system undersizing, duct losses, and (in some cases) low airflow has increased usage over what might be predicted using just the nominal HSPF (from Table 5). Furthermore, outside of Clark County, use of outdoor thermostats is relatively rare. This would be expected to increase heating usage on a seasonal basis.

Conclusions

Primary conclusions from this study of 161 air-source heat pump sites in the Pacific Northwest have been used to revise estimates of heat pump energy usage and have reinforced the importance of directly specifying the following activities required to receive incentives for new heat pump installations.

- Nominal capacity of the system shall be found using a design temperature (generally 20°F to 25°F depending on climate zone) and shall include a reasonable allowance for duct losses.
- Auxiliary heat activity shall be eliminated during mild conditions (35°F to 40°F) through the use of an outdoor thermostat to ensure seasonal performance. Unless this type of control is specified by an organized program, it is not likely to be installed.
- System airflow is still low in a significant minority of systems and needs to be measured and adjusted as part of the installation process.
- Refrigerant charge problems, at least in this sample, are limited and are more on the overcharge side, which has little effect on heating season performance.

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