Field Monitoring of High Efficiency Residential Heat Pumps

Bob Davis, Ecotope, Inc.
David Robison, Stellar Processes

ABSTRACT

Contemporary air-source heat pumps have great potential to reduce space-conditioning loads, especially when installed in houses built with low thermal loads. These heat pumps are sensitive to installation quality, and problems may not always be identified quickly and repaired. Electric utilities in t he Pacific Northwest have provided incentives to HVAC installers to ensure system installation quality, and regional energy planners have paid increasing attention to heat pump performance. This study looked at a small number of heat pumps in great detail, monitoring energy usage for a year. After initial evaluation and repair (as needed), most sites performed up to the label specifications (HSPF). Even at sub-performing sites (measured on basis of climate-adjusted HSPF), annual kWh usage for heating mostly met expectations (in part because of very efficient thermal shells).

Introduction

Pacific Northwest utility incentive programs have subsidized higher efficiency heat pumps and installation measurement protocols for several years. Previous detailed monitoring using similar methods (Reichmuth et al. 2005) found a number of problems and determined that some installation errors may not be caught and corrected by one-time startup protocols (which have become more common in utility incentive programs in recent years). If systems are not installed correctly, and if problems are not identified and repaired, heating loads will not be as expected and the value of the conservation resource will be degraded.

The monitoring described in this paper was funded as part of a State Technologies Assessment Collaborative (STAC) grant awarded by the U.S. Dept of Energy. The project included a diversity of local climates (from maritime to Rocky Mountains) and system types, including multistage compressors. Homes in this study were built to latest energy codes so thermal loads were relatively modest. Duct systems, in most cases, were sealed and insulated to meet modern efficiency standards. Monitoring collected true power, temperature, and control status data every few seconds; hourly averages were prepared by operational mode (heating/cooling, defrost, fan scavenging, etc.). System airflow was measured with an averaging velocity pressure grid for each mode of operation so that thermal output could be determined. Thus it was possible to determine overall system performance during a cycle, including the effects of duty cycling, defrost and auxiliary (electric resistance) heating.

The most interesting findings from this study were as follows:

- Some installers are unfamiliar with newer heat pump technology and can make serious mistakes in system installation. Depending on the inspection environment, these problems may not be identified for some time.
- After proper installation, units can perform at expected efficiency (climate-specific HSPF), although improvements can reduce unnecessary usage of electric resistance heating during defrost.
- Even in cases where climate-specific HSPF did not meet expectations, heating season usage (when normalized by house size), was not out of line with recent regional billing analysis of similarly-built homes.
- Mechanical cooling energy was only a very small part of annual HVAC energy at all sites (as expected). At two sites, there was no mechanical cooling.

Site Summaries

The test sites were located in diverse climate zones in the Pacific Northwest, including central Oregon (Bend), the greater Portland metro area (Deer Island), the Olympic Peninsula (Shelton), Boise, north central Washington (Moses Lake), and at higher elevation in far-eastern Idaho (Ashton). Monitoring started during the 2006 cooling season and continued through 2007. The Bend site was withdrawn in November, 2006 due to a death in the family. A replacement site in southern Oregon (Roseburg) was set up in early March, 2007.

The monitoring system used multiple temperature sensors, current transducers, a measurement of system airflow, and a condensate tipping gauge (to measure latent cooling energy usage). Data were stored on a laptop computer that served as the interface between sensors and the Internet for data retrieval.

The Pacific Northwest is a heating-dominated climate, ranging from about 4250 Heating Degree Days (base 65º F) at the Shelton site to about 8611 HDD at the Ashton site. Most sites had thermal shells that we sufficiently efficient that base 65º F heating degree days are perhaps not the most accurate estimators of annual heating energy usage (that is, base 50 or 55º F degree days would be more descriptive). The Boise, Moses Lake, and Roseburg sites had the most cooling demand (but it amounts to no more than 1500 kWh/year at these sites given the combination of relatively mild climate, efficient building shell, and minimal latent cooling load). The Shelton site, located about 100 feet from the Puget Sound, used no mechanical cooling during the monitoring period. The Deer Island site, located near Portland, OR, also used no mechanical cooling.

Table 1. Test Site Summary

Site	Heated sq.ft.	Flrs	House UA, Btu/ hr-°F	Whole House U_o[2],Btu/ Hr-°F-Ft^2	USDOE Zone[5]/ HDD_{65}	Heat Pump Size (Tons), Type[3]	Refrig	Duct Location Supply/ Return
Bend[1]	2 2750	2	Not	Calc'd	IV/7040	4, M	R-410a	Crawl/ attic
Boise, ID	2 2550	2	396	0.064	IV/5727	3, M	R-410a	Crawl/ interior
Ashton, ID	1 1960	1	235	0.053	V/8611	4, M	R-410a	All interior
Moses Lake, WA	1 1568	1	228	0.042	IV/5836	2.5, S	R-22	All interior
Deer Island, OR	1 1837	1	365	0.053	VI/4491	3.5, S	R-410a	Crawl/ attic
Shelton, WA	3 3100	2	576	0.066	VI/4250	3.5, S	R-410a	Crawl/ interior
Roseburg, OR	1 1750	1	402	0.061	VI/4020	3, M	R-410a	Crawl

[1]site withdrawn from study, November, 2006

[2]heat loss rate of home normalized by area of all components; does not include contribution from air leakage

[3]S=single stage compressor; M= multi-stage compressor

[4]heat pump heats 1,960 ft^2 main level; fully finished lower level of same square footage heated by separate electric furnace

[5]ARI Zone corresponds to DOE weather zone (I through VI). HDD_{65} is Heating Degree Days (base 65º F).

Because these systems are certified by the Air Conditioning and Refrigeration Institute (ARI) to be higher efficiency units (SEER of at least 14 and HSPF of at least 8.5 in all cases), their average nominal performance, as expressed as a Coefficient of Performance (COP), should be expected to be between 2 and 3. This means that at a given outdoor temperatures, two to three units of heat will be delivered for each unit of electrical energy used to run the compressor, outdoor unit fan, and air handler.

Most units use R410a refrigerant, employ a variable speed (ECM-based) air handler, and are equipped with thermostatic expansion valves (TXVs) on both coils. All test homes are built to modern energy –efficiency standard (in some cases, Energy Star or near-equivalent), and the U_o in Table 1 reflects this (as U_os of about 0.05 are equivalent to 2007 Energy Star specifications). Duct systems are insulated to at least R-8 when run in unheated buffer spaces and meet Energy Star leakage standards (exterior leakage of less than 6% of heated floor area at a test pressure of 25 Pa). Note two of the sites (Ashton and Moses Lake) have duct systems that are located completely within the house's conditioned space.

A number of repairs were needed at these sites. Major installation errors were noted early in the monitoring period and corrected. A malfunctioning TXV was identified and replaced at the Moses Lake site during system review. There is reason to believe that the same error was present at the Bend site, although there were likely other problems (which unfortunately were never fully identified because the site was withdrawn). The Ashton site had several problems. A non-heat pump thermostat was installed, the incorrect indoor unit was installed, and numerous control wiring problems needed to be resolved to facilitate proper system staging. These problems were eventually ironed out by mid-October, 2006 (but the system still performed poorly, due in part to its being oversized for the heating load.) The heating performance figures reported in the next section were determined after these major fixes were performed. The auxiliary heat elements at the Roseburg site were wired so that they operated in cooling mode; this problem was quickly fixed. At the Boise site, the defrost control resulted in abnormally high amounts of electric resistance heat usage (at least in terms of what would be expected based on ARI defaults (around 10% of heating-related usage). Revisions to the defrost system were made and improvement was measured. These problems are a focus of this paper and are discussed in depth below.

Heating Season Performance

The "bottom line" for these systems is how well they perform over the entire year and especially during the heating season. Table 2 summarizes the performance in terms of both COP and various ways of estimating the HSPF. The "Overall COP" noted in Table 2 is the empirically measured COP over the full cycle, including the energy used for auxiliary heat (electric resistance heat used to supplement compressor heat), part-load compressor operation, and defrost. The "Observed HSPF" is the same overall value expressed in HSPF units (kBtu/KWh). COP and "Observed HSPF" at each site are weighted by the actual duty hours. That is, the average COP and "Observed HSPF" use the as-operated energy in each temperature bin to calculate the overall seasonal rating. The "Quasi HSPF" uses the observed COPs by temperature bin but weights them according to USDOE-specified operating hours in each bin. Since these are not the same as the observed operating hours, this performance rating is different. The difference between "Observed" and "Quasi" ratings provides an indication of whether the USODE-specified operating hours match observed operations. The difference between the "Quasi" and "Target" ratings provides an indication of whether system performance matches the "label spec" rating using similar assumptions regarding operating hours. These performance ratings do not include the effects of duct losses (as these are not included in ARI ratings). These losses would include extra infiltration induced by unbalanced duct leakage; these effects are very limited at these sites because ducts are well sealed.

Table 2. Summary of Annual Heating Performance (HSPF)

Site	Overall COP	Observed HSPF	Quasi HSPF	Nominal HSPF*	Target HSPF**	Comments
Bend	1.4	4.8	5.5	9.0	9.0	Dropped out of program after 3 months of monitoring
Boise	2.2	7.4	8.1	8.6	8.6	Excessive defrost energy usage corrected late in monitoring period
Ashton	1.1	3.6	4.9	8.8	7.7	System oversized vs. heating load
Moses Lake	2.7	9.3	6.2	9.1	9.1	Defective TXV on indoor unit replaced soon after initial installation
Deer Is	2.9	10.0	10.1	9.6	10.9	
Shelton	2.4	8.2	6.2	8.9	10.2	
Roseburg	2.4	8.2	5.3	9.05	10.3	Same unit as Bend; much better performance. Very limited heating data given system installed in late March.

*DOE Zone IV HSPF ("label spec")

**Target HSPF is nominal system HSPF with DOE climate zone correction applied as needed.

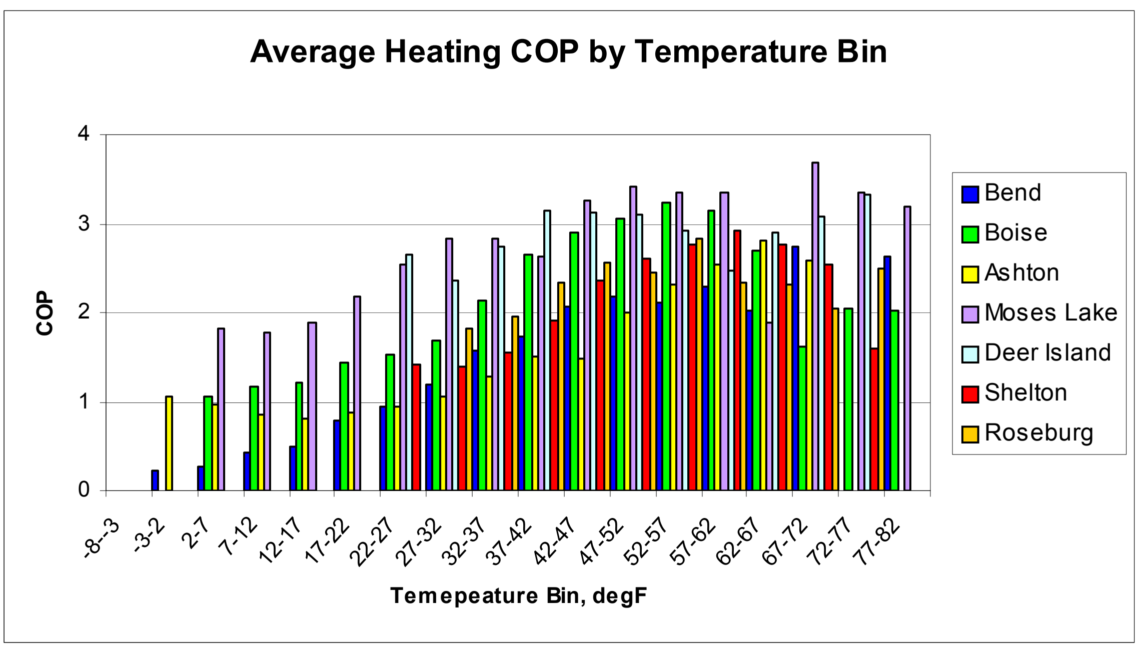
Note the target HSPFs are not the nominal HSPFs reported by Air Conditioning and Refrigeration Institute (ARI, 2003). In most cases, the nominal HSPF reported by the manufacturer and found in marketing literature is the HSPF determined for USDOE climate zone IV (Washington, DC area). If a specific weather site has significant different ambient temperatures than Zone IV, the nominal HSPF will have to be adjusted (since the heat pump output depends on outdoor temperature). For example, a system rating for Zone IV would overestimated heating seasonal performance in Ashton (Zone V). The Zone IV rating for Shelton would underestimate performance given Shelton is in the milder Zone V climate. This issue is discussed in detail in Francisco et al. (2004).

The COPs calculated from the monitored data reflect the performance of the system after the monitoring team checked out system controls, airflow and refrigerant charge. The check out procedure is a more in-depth version of that required by installers who participate in the April, 2007 Performance Tested Comfort Systems (PTCS) program (administered by Bonneville Power Administration (BPA) for new heat pumps that receive incentives from public utilities who are BPA customers).

These COPs also capture performance after major repairs (detailed below) were made; most repairs were made before much monitoring had occurred. The exception to this is the Bend site, where we were unable to have repairs made. There were obvious problems at this site, and its overall performance was not much better than that of an electric furnace (which should have a COP of 1 if there are no duct losses).

Another way to represent heating season performance is by showing the side-by-side COPs as a function of outdoor temperature. The difference is striking between sites that performed well (Moses Lake, Deer Island, Boise) and sites that performed poorly (Bend, Ashton, Shelton). Note the Roseburg site had very limited heating operation given when it was instrumented (late March, 2007).

Figure 1. Heating COP by Site



Heating energy usage was also summed for the year and normalized by house size. This was useful because even if COP/HSPF is not as expected, the true bottom line is how much energy was used to heat the home over the winter. The usage could be compared with simulations, but the intent here is to make a comparison to measured usage for similar buildings sited in the Northwest. Most of the homes in this study were constructed to aggressive energy codes and unfortunately there is a limited amount of submetered heating usage data for heat pump homes that could be used for comparison.

Table 3. Measured Heating Energy Usage

Site	Measured Annual Heating Energy (kWh)	Annual Heating EUI* (kWh/ft^2)	Whole House U_o ($Btu/ Hr\text{-}°F\text{-}ft^2$)	ARI Climate Zone
Bend	N/A	N/A	Not calculated	IV
Boise	6,402	2.8	0.064	IV
Ashton	4,232	2.2	0.053	V
Moses Lake	1,437	0.92	0.042	IV
Deer Is	4,535	2.5	0.053	VI
Shelton	4,086	1.3	0.066	VI
Roseburg	N/A	N/A	0.061	VI

*EUI = (measured annual heating energy/house size)

A different comparison is offered. In the early 1990s, a set of Pacific Northwest manufactured homes were constructed to a target whole house U_o of about 0.055. The U_o, as shown in Tables 1 and 3, is the area-normalized heat loss rate of the home. This metric does not include the contribution of infiltration or duct losses to annual heating energy usage. Note, though, that duct performance and infiltration are roughly comparable between the manufactured homes and the sites in this study.

The billing results from the evaluation of the manufactured homes, found in Baylon et al. (1995), are helpful in benchmarking performance. However, only about 20% of the manufactured homes had heat pumps. Most of the utility billing data (about 100 sites) are for electric forced-air furnaces. Therefore, usage needs to be adjusted by a multiplier equivalent to the assumed average COP. This average COP is set at 2.5 in the comparison here (implying an HSPF of 8.5).

Approximately 100 sets of utility bills for the manufactured homes can be used; these are split into two groups of about 50 bills, with one group sited in heating climates applicable to DOE Zone IV and the other group sited in heating climates applicable to DOE Zone VI. A small group (less than 20 bills) was evaluated for the coldest part of the Northwest (DOE Zone V) but results are not reported because of the limited data.

Average usage for the manufactured homes, before the COP correction, is 4.1 kWh/ft^2-yr in Zone VI and 5.3 kWh/ft^2-yr in Zone IV. Correcting for COP, the expected brackets for heat pump usage are about 1.7 – 2.1 kWh/ft^2-yr. (Standard deviation for Zone VI is about 35% of the mean and about 28% of the mean in Zone IV.)

The Boise site comes in at the high end of expected usage; the building's U_o is noticeably higher than 0.055 and there is also abnormally high defrost-related usage. Moses Lake uses less than 1 kWh/ft^2-yr for heating, not surprising given the very low U_o (but Moses Lake is also in Zone VI). Ashton's EUI does not suggest a problem (even though its HSPF is disappointing); the lower level of the Ashton house is heated by a separate electric forced-air furnace (using the same heating set point as the main level) and this may have more of an effect than predicted. Deer Island's usage is somewhat above what is expected using this approach, and Shelton's usage is below expected. Predicting individual building energy usage is often folly, but some comparison with regional billing data does help place annual heating usage in context.

Cooling Performance

The overall annual cooling performance is given in Table 5 along with performance targets. It should be noted that at all sites, the nominal available cooling capacity is well in excess of the cooling load (including solar effects). Note there were no hours of cooling operation at the Shelton or Deer Island sites. The Boise site used the most for cooling (1,424 kWh); this represented operation for only about 10% of all potential cooling hours (defined as times where the indoor temperature was more than 1° F above the implied cooling set point, which is determined by looking at the indoor temperature just before mechanical cooling is activated. This is done by the datalogging system.) The cooling duty cycle was even more modest at the other Idaho sites; the Roseburg site had a similar cooling fraction (9%).

Table 4. Summary of Cooling Performance

Site*	Average COP	EER*	ARI EER	Measured Annual Cooling kWh	Cooling Duty Cycle**	Comments
Bend	1.4	4.5	12.3	N/A	17%	Dropped out of program after 3 months of monitoring
Boise	3.4	10.3	11.7	1424	10%	Spends most time in compressor Stage 1
Ashton	2.8	8.3	12.0	871	5%	Almost always in part load operation
Moses Lake	3.4	10.0	12.5	412	6%	
Roseburg	3.5	11	12	521	9%	Same unit as Bend but correctly installed

Deer Island and Shelton sites did not require mechanical cooling during the monitoring period.

*EER taken as average cooling performance (KW/ ton) at 95 ºF.

**Percentage of time mechanical cooling operates (not including any defrost usage in winter).

The overall cooling performance, COP, is based on a weighted average COP. That is, the COP for each temperature bin is weighted by the duty hours in the temperature bin for each site. The EER in this table is the observed average performance in just the single 95 ºF temperature bin.

The ARI EER is used as a performance target instead of SEER. The SEER rating bears an opaque relationship to the weighted average COP. SEER is not well suited for Northwest locations, which uniformly have minimal latent load. The EER is a much more rational target. The target EERs noted in Table 5 are taken directly from the manufacturers' values. All the achieved EER values are somewhat short of the Target EER. In practical terms, this is not a big problem because there is so little cooling load relative to heating load.

Because of project timing, not every site was monitored for a complete cooling season. Boise, the warmest site, has a complete year of cooling data. Ashton is missing half of August and all of September. Based on July usage, we would expect to see at least another 300 kWh usage for cooling. Moses Lake is missing August data; based on July usage, we would expect to see another 200 kWh of cooling usage.

Key Performance Themes

1. Incorrect Specification/Installation of Equipment

The Ashton site had several major problems when the monitoring instrumentation was first installed in early September, 2006. The thermostat did not support a heat pump (let alone a multiple-stage heat pump) and the incorrect indoor unit had been installed. At initial installation of monitoring equipment, only one stage of cooling operated properly and the compressor would not operate in heating mode. We urged the homeowner to contact the installer and he did. However, the installer was not able to fix the problems and needed assistance from the distributor's technical staff. Approximately 20 labor hours were needed to fix the major problems. Finally, by mid-October, 2006, monitoring could begin. The results from this site were not impressive even after repairs, at least on an efficiency basis. More discussion on this point can be found below.

As mentioned above, the Moses Lake site had a malfunctioning TXV. The installer at this site was following the start-up protocol required by the local utility to obtain an incentive. An optional part of the procedure, a check of temperature split at indoor coil, had not been performed. When we were setting up the monitoring, we noticed an inadequate cooling temperature. Even thought he installer had performed due diligence using the required procedure (and had even gone beyond basic requirements and measured system subcooling in cooling mode), he would have left a serious problem at the site (which is one in the study with significant cooling usage. Note this problem would not have affected heating energy usage (since the indoor unit TXV is bypassed in heating operation.)

2. Excessive Parasitic Energy Usage during Defrost

The Boise site utilized a defrost control that seemed to trigger defrost more frequently than the measured coil temperature would indicate it was needed. This issue by itself was not serious; however, the system would also typically call for at least 15 kW of resistance heat during the defrost cycle. In the defrost cycle, the heat pump is briefly reversed into the cooling mode so that it will pump interior heat to melt ice that has collected on the outdoor coil. It has been common (since the late 1970s) to activate resistance heat to buffer air temperatures delivered to the conditioned space during defrost. Depending on how often defrost is triggered and how much resistance heat is activated, the overall parasitic energy usage can be significant. This problem was noted by Reichmuth et al. (2005) in their earlier detailed study of Pacific NW heat pumps.

Figure 2. Example of Defrost Operation Before Repair

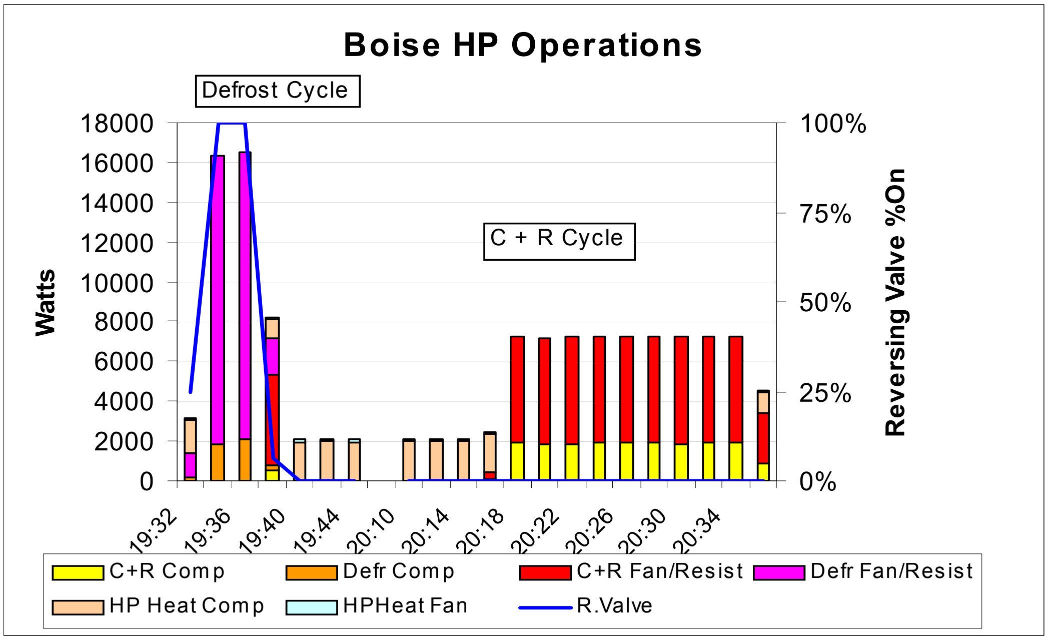
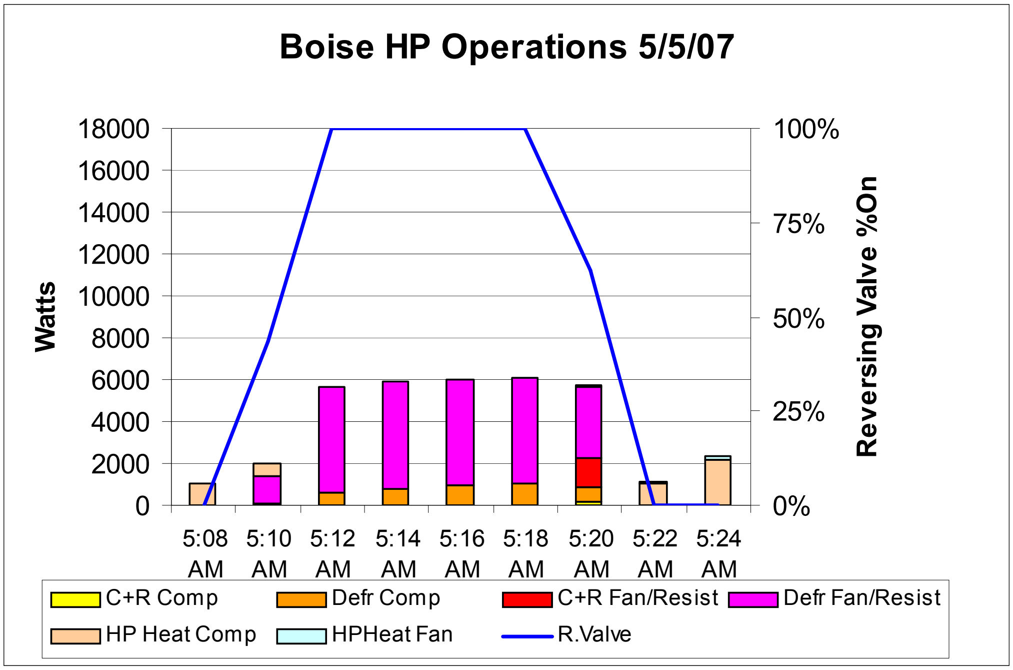


Figure 2 shows a typical Boise defrost cycle (November, 2006) in 2 minute intervals. The blue line indicates the reversing valve switching to cooling mode so that hot refrigerant vapor will be directed to the outdoor coil. Shortly after this occurs, the approximately 15 kW of resistance heat is energized (magenta bars). Resistance heat plays no functional role in the defrost process; all de-icing proceeds from the compressor activity. The resistance energy is used strictly for perceived comfort reasons. As long as the supply air is maintained at more than 55°F for the brief defrost cycle, comfort will not be compromised. Given that the ducts have been heated prior to the defrost cycle, the delivery temperature is well above 55º F without need for any resistance heat.

After the defrost cycle ends, the system returns to normal compressor-only heating mode (peach-colored bars). Also, this graphic shows some (relatively rare) use of auxiliary heat (right side of graph; red bar is approximately 5 kW of auxiliary heat). Ambient temperature was about 20º F during this monitoring, and the balance point of this system was measured at about 22º F (Figure 3). Therefore, some amount of auxiliary heat use is not surprising.

Figure 3. Boise Site Defrost Behavior After Repair



The installer was persuaded to remove two of the three resistance elements from the auxiliary heat circuit. The resulting change in defrost wattage is shown in Figure 3. In coarse terms, the parasitic energy usage during defrost was reduced by a factor of 3, since the resistance elements are by far the biggest load during defrost. At this site, the pre-repair defrost usage, as a percentage of overall heating energy, was about 18%. The average of the other sites was about 8%. The repair was done late in the year at the Boise site so there was very little heating season left. Using the factor of three reduction estimate, however, the Boise site defrost usage as a percentage of all heating energy should drop to below 10%. An amount around 10% is in line with what is described in U.S. Department of Energy (2005).

It is encouraging an improvement could be made at this site. In looking at the equipment in this study, a variety of defrost algorithms were noted. Some systems are easier to modify to reduce parasitic energy usage. The challenge going forward is to construct an inspection protocol which can be applied prescriptively across heat pump manufacturers. This effort is underway but will be difficult given this difference in equipment and also given the concern of installers that comfort will be adversely affected.

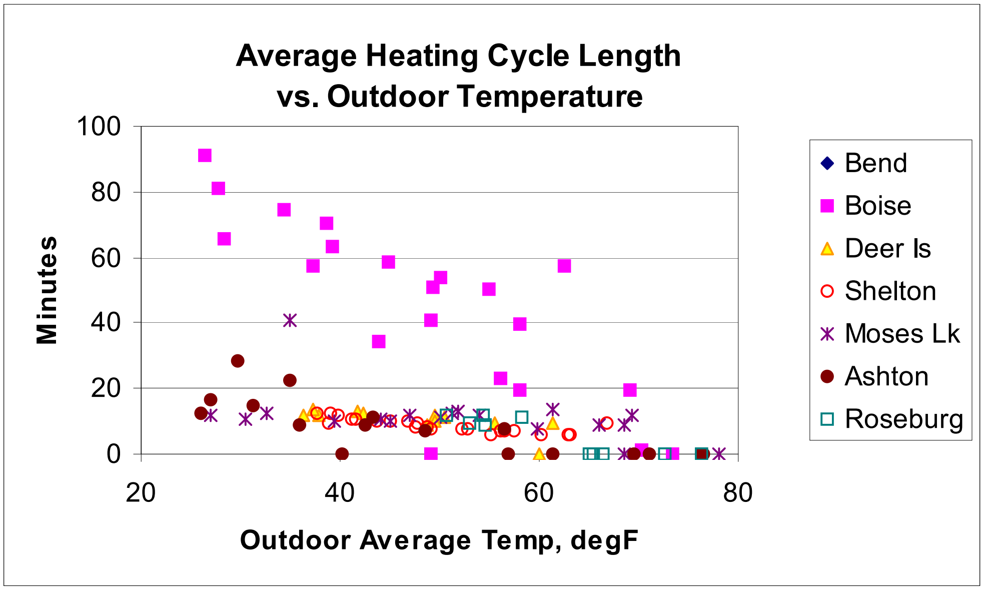
3. Unresolved Contributors to Apparent Under-performance

The Shelton and Ashton sites did not perform up to the manufacturer's HSPF. Despite extensive checkouts and follow up based on various data signals, we were not able to make changes that would get the system to perform at the expected COP/HPPF levels.

Ashton had a two-stage compressor, but there is no clear evidence that the system worked with staged compressor operation. Quite simply, the problem at Ashton was that the supply air was usually not warm enough relative to the electricity used to run the compressor and fans! This problem occurs at all outdoor temperatures and is especially confusing since the system is being run at only about 325 CFM/ton of indoor unit airflow (when volumetric airflow is corrected to mass flow). Running the system at lower should result in warmer supply air, but this is not what was observed at Ashton.

Part of the problem has to do with average heating cycle length. Ashton does have the shortest average cycling length (vs. outdoor temperature). This does have some influence on the heating COP and quasi-HSPF. Shelton is also near the bottom in this regard. However, Moses Lake and Deer Island are not far off the cycle length of the poorer performers. Systems with dual TXVs tend to get to steady-state performance within 7 or 8 minutes, whatever that steady-state performance is.

Figure 4. Heating Cycle Length



The Shelton site also had a lower than expected measured efficiency. This could be in part to an unresolved measurement problem. We know that the COPs were measured early on were artificially deflated due to the placement of the return air sensor (in relation to the bypass duct back into the return plenum to support the supply duct zoning). However, even when a second return air sensor was added so that the return air temperature could be more accurately described overall COP was still not impressive.

Apart from the binned COP performance or quasi-HSPF, we do also have the kWh totals and Energy Use index that can be compared to what might be production from a simulation or by looking at sets of bills for homes built to similar energy codes.

Relying only on COP or quasi HSPF is probably not sufficient to describe a system as underperforming. Still, after several months of monitoring and attempts to improve the efficiency at these sites, we cannot say the results were fully successful.

Conclusion

New, high-HSPF air-source heat pumps have great potential to reduce space-conditioning loads. Systems are sensitive to installation quality, and problems may not always be identified quickly and repaired. In this study, the monitoring crew was motivated to identify and repair problems. At most sites, once problems were repaired, the systems performed to expected levels.

Even at sub-performing sites (measured on basis of climate-adjusted HSPF), annual kWh usage for heating mostly met expectations (in part because of very efficient thermal shells).

Some problems were relatively easy to identify and repair (such as replacing a defective TXV or removing one or two auxiliary heat elements from the defrost cycle) but others required extensive intervention by service and distributor personnel (wrong thermostat type installed plus incorrect control wiring at indoor unit).

Defrost cycle parasitic energy use is of concern to Northwest energy policy makers and will likely receive some attention in upcoming reviews of utility programs that provide incentives to heat pump installers.

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