Field Monitoring of High-Efficiency Residential Heat Pumps

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ABSTRACT

Four high-efficiency heat pumps and one older unit, serving residential loads in Oregon, were monitored for one year to characterize heating and cooling performance. The monitoring sites were climatically diverse; coastal, mild valley and high desert climates were represented. The monitoring was thorough enough to quantify the true electrical input power and output thermal power for all operating modes including defrost. The results showed all units performing below expectations for identifiable reasons. The most significant failures were associated with excess icing, improper use of resistance backup heat and faulty staging of a scroll compressor. This work led to corrections that improved the system operating efficiencies as monitoring proceeded. The work also led to recommendations that can be generally applied toward improved heat pump performance.

Introduction

The purpose of this research is to improve the understanding of heat pump performance in the Pacific Northwest. This monitoring project involves performance measurements of four high-efficiency (nominal HSPF >= 8.0) heat pumps at residential sites, with one older heat pump site as a reference. These sites were monitored for electric energy use and key heat pump operating temperatures for approximately one year, since August 2004, to provide a complete characterization of heating and cooling season performance.

The monitored systems were selected from a range of climate types and a range of compressor types (single- and multi-stage), air handler type (standard and variable speed), duct placement, and thermostatic and related controls. The fundamental characteristics of the monitored sites are summarized in Table 1.

Table 1. Monitored Site Characteristics

Site Location	Climate	Compressor	Fan/ducts
The Dalles, Oregon existing stock reference unit	Low elevation hot dry summer with cold winter	Single-stage piston, 15 yr old unit, HSPF<6	Standard fan basement location and under floor distribution
Sunriver, Oregon	4000 ft elev. mountain/ high desert cold winter mild summer	Single-stage scroll, new unit HSPF>8	Standard fan interior location with attic return and under floor distribution
Ashland, Oregon	2000 ft elev. dry mountain warm summer cool dry winter	Two-stage scroll, new unit HSPF>8	VSD fan in attic with attic distribution
Eugene, Oregon	Mild interior valley mild moist winter and summer	Two-stage piston	VSD fan basement location underfloor distribution
Manzanita, Oregon	Oregon Coastal cool moist summer and mild moist winter	Single-stage scroll, new unit HSPF>8	Standard fan interior location under floor distribution

One of the primary reasons for selecting the diverse range of test sites was to examine the system control responses to a full range of weather and de-icing conditions.

Before investing in an extensive monitoring effort, it was important to bring all the sites to proper operating condition.

Methodology

The methodology consisted of four very interrelated primary activities: site characterization, data collection, data analysis and follow-through.

Site Characterization

The sites' physical characteristics were documented well enough to estimate the heat loss rate (UA) for the site. Duct leakage and system airflow were measured, and an airflow vs. fan power function was developed. These data, along with the estimated UA and component details, were used in Ecotope's SEEM model (Palmiter & Francisco 1997) in order to estimate distribution system efficiency. The system refrigerant charge was checked against specifications, as were unit controls (defrost, outdoor thermostat).

Data Collection

Logged data consisted of the four key temperatures: outside air, indoor air at thermostat, return air and supply air. Accurate temperature measurement required careful sensor placement. The outdoor temperature sensor was in a shielded enclosure with a fan, and the supply sensor was at least five duct diameters downstream of the fan to allow for sufficient mixing. The indoor temperature was measured within one inch of the thermostat. Also logged was true power at the outdoor unit (compressor), indoor unit (fan/resistance) and whole house power. Control status at the reversing valve and second stage were also sensed and logged via relays. The data logger scanned all the sensors every six to ten seconds (depending on the complexity of the logging instructions) and accumulated averages for a specified interval. The interval was an hour for most logging, but it was at times set to two minutes to get a closer look at individual operating cycles. The data loggers were capable of identifying the heat pump operating mode on the fly and accumulating key data for each operating mode. For example, first-stage heating could be identified from compressor power and reversing valve status. Then duty cycle, return and supply air temperatures, and compressor and fan power, all for the identified first-stage heating mode, could be accumulated for that monitoring interval. This is referred to here as conditional logging. Including all the conditional data, the monitoring data consisted of hourly values for ~75-100 measurements. Dataloggers would initiate phone contact, via the existing phone line, to an 800 number at the data collection computer. To avoid conflict with the residents, each data logger had a unique daily call-out time prior to about 5:00 a.m. Overall, the project achieved a data recovery rate in excess of 90 percent.

Data Analysis

Data analysis is an often under-budgeted effort. The first few months of the project, into November 2004, required intensive data collection and analytical effort in order to find and exercise all the heating modes (including defrost), verify the methodology and prepare analysis templates. In general this analysis is intended to compliment other work done in this field, but in other regions (Maisello, Bouchette, Matthew, Parker & Sherwin 2004; Proctor, Hammarlund, Cast & Ward 1992; Francisco, Baylon, Davis & Palmiter 2004).

As the monitoring proceeded, the data were blocked into two-week units. Each of these two-week units was examined in detail in an analysis template unique to each site. The analysis templates showed the data from a variety of perspectives and produced summary results for that two-week interval. There were enough detailed perspectives to highlight most data irregularities and operational glitches. At all sites the detailed review would reveal unusual operating situations, such as unusual defrost. The logger would then be set to high resolution (two-minute interval) data, or further site inspection and consultation would be carried out. Active follow-up of the data analysis was an important team effort. The two-week summary reports are produced in Excel and formatted to allow ready aggregation seasonally and at the end of the project.

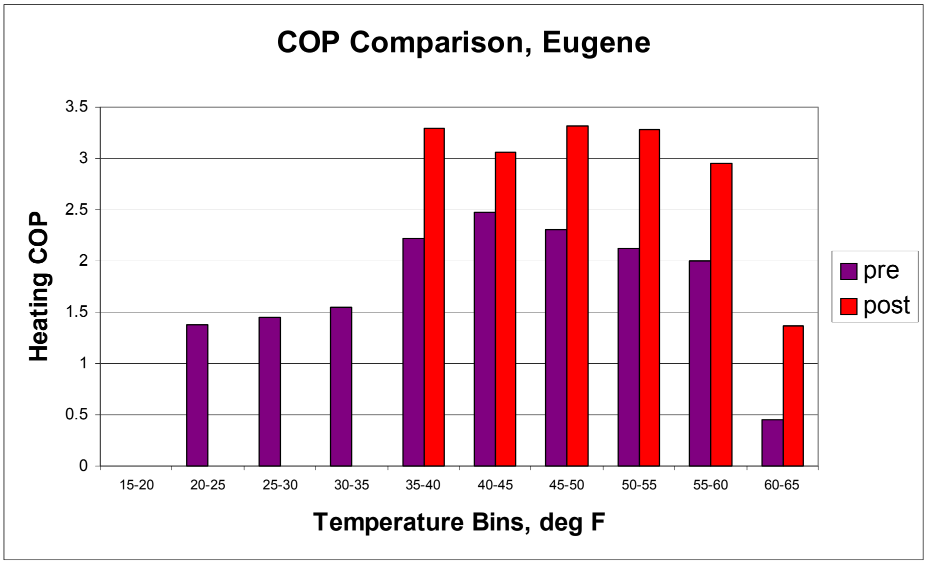
As a check on accuracy, the monitored performance of each unit (power, airflow, Btu output, outside temperature and return temperature) was compared to the manufacturer's heating performance tables for that particular combination of outdoor/indoor conditions (Trane; Carrier; York). The manufacturer's estimate is based on steady-state operation, requiring at least 30 minutes to reach such a state. It was usually difficult to find monitoring intervals with the same conditions as those in the manufacturer's tables, but several appropriately close instances were found. At two sites, Sunriver and Eugene, the monitored data compared closely with the manufacturer's estimate. At the Manzanita site, it was difficult to find a suitable interval with a long enough cycle of operation for comparison. At the Ashland site, both monitored power and output exceeded that of the manufacturer's table, however the COP was very close.

It is notable that this comparison to the rated steady-state performance confirmed the need for an altitude-corrected air density at Sunriver, which is at 4,000 feet. Each site has its own "personality," and understanding the typical operating behavior is key to verifying that the unit is operating properly and that the data collection scheme accurately characterizes that operation.

In this work it is quickly evident that heating and cooling system thermal efficiencies have a complex relationship to outdoor temperature, dependent on both thermal/mechanical properties and system controls. A single seasonal number or rating point could not usefully characterize the observed results, except for gross seasonal comparisons. Instead these thermal efficiency measurements are calculated and aggregated in 5°F dry bulb temperature bins. See Figure 1. This is also the format used by BPA researchers on the Nyle heat pump project, and it facilitates comparison of this work with the BPA work (Callahan 2006). A significant benefit of this aggregation of efficiency measurements is the ability to readily use it with existing (hourly or bin temperature) annual performance models.

This has proved to be a broad and versatile way to compare the performance of heat pump units operating in different climates and despite variations in other system parameters. And significantly, it has also proved to be a very fluent way to show the performance improvements from subtle control changes as shown below.

Figure 1. Heat Pump COP by Temperature Bins



In Figure 1, the control change was the replacement of an improperly sized refrigerant expansion orifice.

Summary Performance Results

Thermal Output

The observed thermal output of the heat pumps at the five monitored sites consists of both heating energy and cooling energy. The observed annual thermal output is summarized in Table 2.

Table 2. Annual Delivered Thermal Energy

	Ashland	The Dalles	Eugene	Manzanita	Sunriver
Btu Heat	15,507,468	39,093,512	52,009,285	16,645,263	30,065,828
Btu Cool	576,667	1,498,644	4,860,324	1,562,116	2,616,404
Cooling Fraction	3.6%	2.1%	8.5%	8.6%	8.0%
Unnecessary Use of Cooling	1.0%	0.0%	32.7%	97.3%	32.7%

Most evident in Table 2 is that observed cooling energy is rarely more than 8 percent of total thermal output, referred to as the cooling fraction. It is noteworthy that the highest percentage observed cooling fraction was at the coastal Manzanita site, the site with the lowest actual cooling load. This shows dramatically that the cooling fraction varies most with operator intention. The 2-4 percent cooling fractions observed at the Ashland and The Dalles sites correspond to a typical cooling set point of about 78°F. The unnecessary cooling noted in Table 2 is an estimate of the portion of the observed cooling that could have been achieved by admitting the cool outside air. Note the Manzanita site again: it had the highest observed cooling fraction, but most (97.3 percent) of that cooling could have been achieved instead by simple ventilation, i.e. opening a window. Even the coastal Oregon climate has low humidity during the occasional episodes of high dry bulb temperature.

Thermal Efficiency

The presentation of thermal efficiency relies on the temperature bin presentation shown in Figure 1. The noted COPs are the "unit COP," and are intended to be consistent with the regional COP or HSPF assumptions. This type of COP is measured at the unit by referencing the supply and return air temperatures at the unit. It does not include the downstream effects of system losses such as duct and air handler losses. The heating and cooling COP are based on the sensible temperature changes and do not include the latent heat changes in the air. For heating this is a satisfactory approximation, and heating is the primary energy use in the Northwest. The data loggers did include a special flow gage to record condensate during cooling, but this was not included in estimates of cooling COP.

Heating System Efficiency Summary

The heating system efficiency derived from an entire heating season is given in Figure 2. This figure gives the COP at each site with respect to outdoor temperature bins.

Figure 2. Heating COP by Temperature Bins

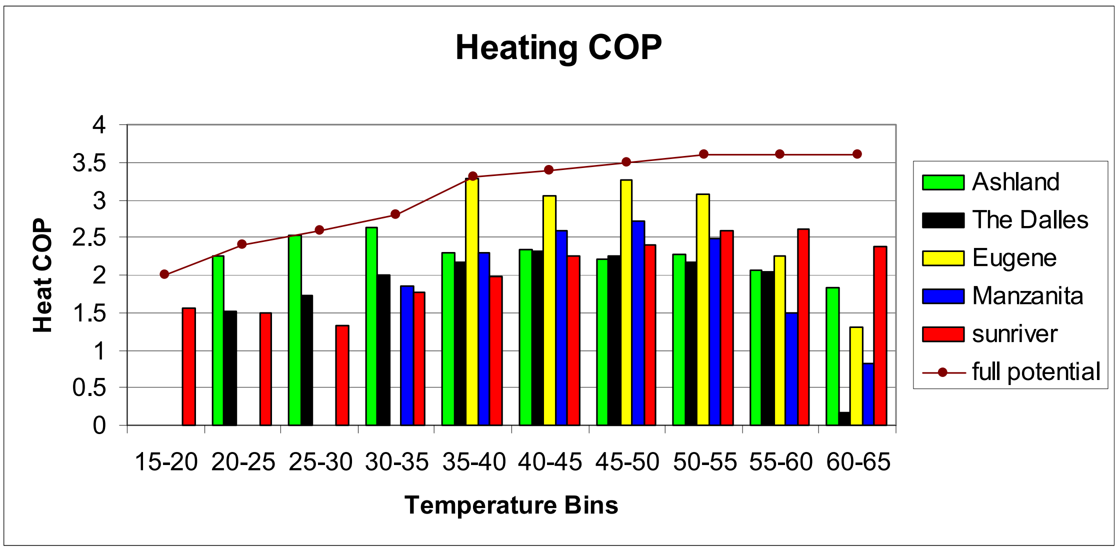


Figure 2 presents a comparison of the overall heating COP by temperature bins for all five sites. (This is full-year, output-weighted, average heating performance. The temperature-binned heating performance in any particular two-week review period often differs from this average annual performance.)

In theory, the heating COP should increase as the outside temperature increases. However, the COP for all sites decreases in the temperature bins above about 55°F. This is predominantly due to short-cycling. Not much heating is needed at these temperatures, and the units do not operate long enough to achieve steady-state efficiency. Also from theory, the COP should decrease as the outside temperature decreases. In Figure 2, this appears to be the case for all units but Ashland, where the highest COPs are in the cold 20-35 degree range. This is because at these cold temperatures the Ashland unit operated with its more efficient second stage. Had the Ashland first stage operated as efficiently as intended (or if the first stage had been bypassed), the green bars would have showed Ashland with COPs of about 3 (much like Eugene) in the 35-55 degree temperature bins.

Note the COPs of the site in The Dalles (black bars). This unit was so old that its nominal performance was not available from the manufacturer. This stalwart 15-year-old unit often outperforms the newer units at Sunriver, Ashland and Manzanita. The low COPs for the Sunriver unit are quite significant and probably result from excess icing as discussed later in this paper.

The full potential of the newer heat pump technology is probably best represented by Ashland in the lower temperature bins and by Eugene in the higher temperature bins – these demonstrate a significant improvement over the older technology. However, realizing the full potential of the newer technology over all the temperature bins requires proper attention to many installation details. ***None of the units did all things right at all times,*** as might be achieved by a hypothetical "full-potential" (but achievable) unit. Such a unit would perform as the best performing unit in each temperature bin in Figure 2.

Annual Cooling Performance

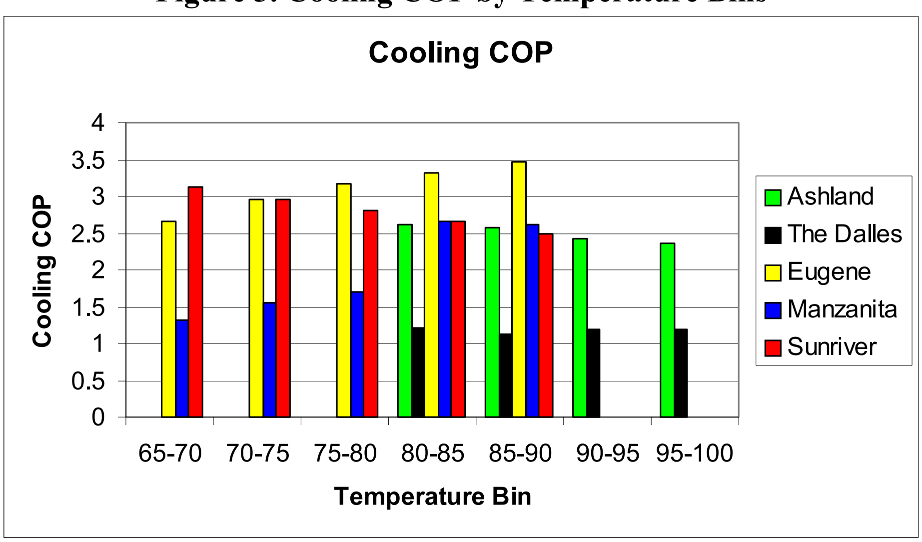
The overall annual cooling performance is given in Table 3 along with performance targets.

Table 3. Summary of Estimated Cooling Performance

Site	COP	Quasi EER	Target EER	Comments
Ashland	2.5	8.5	11.5	Inefficient first Stage
The Dalles	1.2	4.1	N/A	15 year old unit
Eugene	3.1	10.6	11.1	Efficient first stage
Manzanita	2.6	8.9	10.89	Short cycling
Sunriver	2.6	8.90	11.81	

The overall cooling performance, COP, in Table 3 and Figure 3 is based on a weighted average COP. That is, the COP for each temperature bin is weighted by the thermal output in the temperature bin for each site. Note that the quasi EER is simply the COP converted to BTU/watt; it is not the formal EER based on test measurements at 95°F. The target EER is calculated from the HSPF using Northwest specific planning correction factors.

Figure 3. Cooling COP by Temperature Bins



The Manzanita site shows anomalously low COPs in the 65-80 degree temperature range due to contradictory heating. The cooling setpoint at this site was only about 2 degrees higher than the heating set point, and it was common to find cooling one hour and heating the next.

It is probable that all four high-efficiency units would perform at the EER 11+ level if the thermostats dead bands were at least 5-10°F and if compressor cycles were of sufficient duration. Commonly observed compressor cooling cycles were of the order of 5 minutes duration, and in this short time the system does not achieve its steady state operation.

The Ashland site showed poorer performance than expected, COP 2.5 and EER 8.4. This is because it predominately used a less efficient first stage as discussed later in this paper. The same situation dragged down the heating COP.

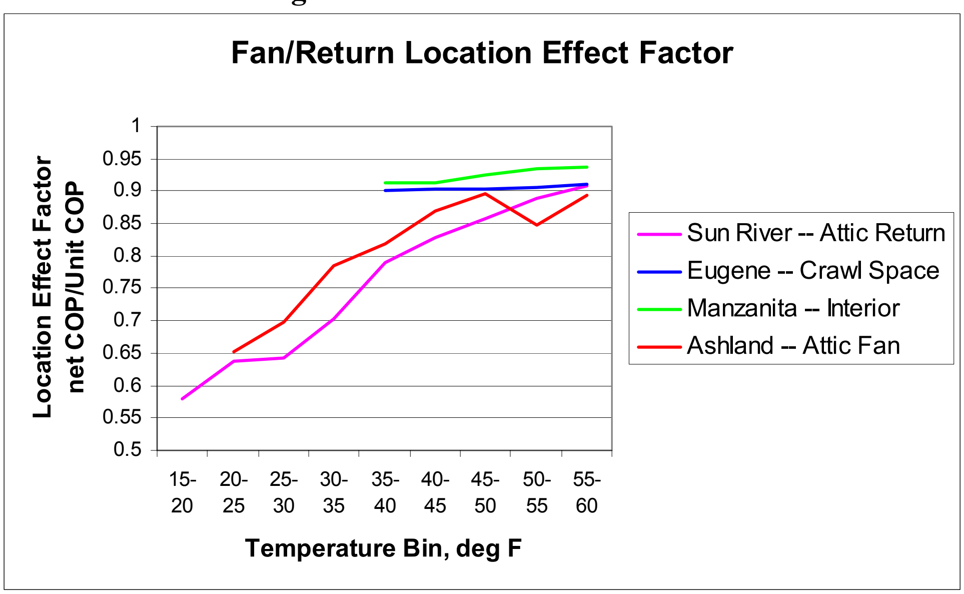
Specific Problem Areas

This monitoring highlighted several important themes relevant to high-efficiency heat pump applications in general:

System Layout and Design

Even if the mechanical components in a system work perfectly, there can still be inefficiencies due to location of the return duct and fan box. The location of these components is determined in the design phase and cannot be easily remedied later. At two sites, the return duct (Sunriver) or both the return duct and fan unit (Ashland) are located in the attic above the insulated envelope. These components are subject to temperature extremes that are about 30°F cooler during heating season and 30°F warmer during cooling season. This location creates a disadvantage for either heating or cooling because the compressor must first bring the return air duct and fan unit to the return air temperature that enters the ducts from inside the house. This takes several minutes of operation and, in the case of heating, reduces the supply air temperature during more than half a typical heating cycle. The effect is strongly temperature dependent, as shown in Figure 4.

Figure 4. Location Effect Factor



This temperature dependency is referred to here as a “location effect.” The net heat delivered to the space is related to the supply temperature–space temperature. Yet the unit efficiencies are calculated from the supply temperature-return temperature. In exposed systems where the return temperature has drifted several degrees from the space temperature, there will be a significant difference between unit efficiency, (unit COP) and net efficiency, (net COP). In any temperature bin, the location effect is defined as the ratio of net COP to unit COP. Figure 4 shows the temperature-dependent location effect functions for four sites.

Both sites with attic components show a location effect with very strong losses in the colder temperature bins – as much as 40 percent of performance can be lost. The attic location of return ductwork is clearly a disadvantage in colder locations such as Sunriver or Ashland. In warmer locations, such as the coast or Willamette Valley the ducts were interior or in the crawl space, and therefore showed high location effect factors and negligible degradation in efficiency.

Short-Cycling

Another system layout consideration pertains to short-cycling. The Manzanita site short-cycled such that a typical heating cycle was rarely more than 6 minutes long. That unit never reached its steady state efficiency. The remedies for short-cycling heat pump systems via thermostat adjustments are limited. A better remedy for short-cycling is proper system layout or design. (The nominal unit size at Manzanita is not out of line with the heat loss rate of the home but still short-cycling occurs.) The thermostat should not be located near a supply register where it will be influenced by the supply airflows. Ideally, the thermostat should be the last place to come up to temperature, not the first. In a small, well-insulated building with tight, well-insulated ducts it is often difficult to find a suitable thermostat location. It may be necessary to “starve” the supply registers nearest the thermostat to encourage longer cycles.

System Sizing and Multi-Stage Systems

Short-cycling is particularly important where a slightly oversized heat pump is employed. Over-sizing the heat pump, especially a two-stage one (Eugene), is a viable way to maintain high efficiency and eliminate resistance heating. But this strategy will not work if short-cycling reclaims the efficiency gains. The Eugene site employed a true two-stage compressor. The first stage had a reasonably high COP of 3+, and the second stage had a COP of about 2.2. This was an oversized unit (4 tons) and could meet the maximum encountered heat load with the second stage with a COP in excess of 2.2. A smaller unit with a resistance boost on second stage would have had a COP under similar circumstances of about 1.2-1.5.

The Ashland site was also labeled and controlled as two-stage. However, this unit is not a true two-stage compressor. This unit was a scroll compressor with an unloading mechanism. The second stage employed the full output of the compressor, and it operated very well with a COP of 3+. The first stage operated at almost the same compressor energy as the second stage, but air handler flow was reduced so drastically that overall mass heat flow (and therefore, COP) was diminished significantly. This type of pseudo-two stage unit is less suitable for an over-sizing strategy because it uses predominantly the first stage, which has lower efficiency than the second stage. The COP of this first stage was about 2 instead of 3. The two-stage thermostat, which was operating properly, invoked the first stage as the first priority. This situation was discussed with the manufacturer and the unit appeared to be operating properly. In all but deep winter, most compressor activity was at the low efficiency first stage. This sequence of events worked to produce an unusually poor annual average COP.

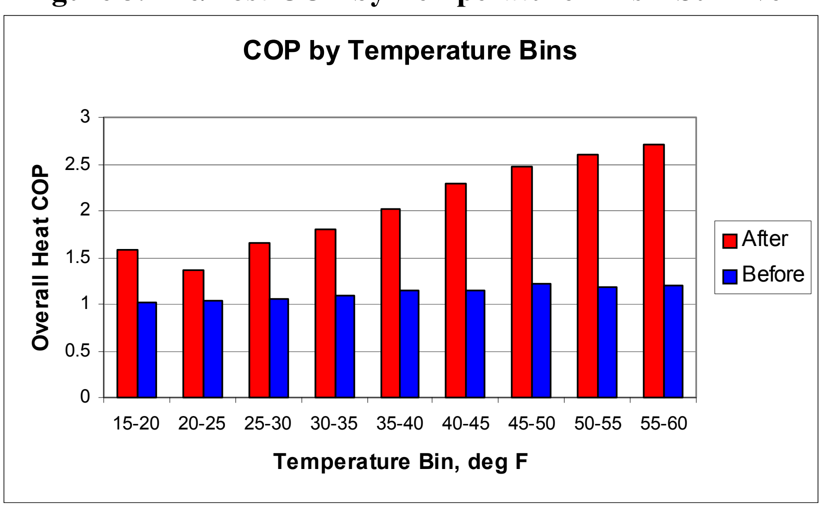
Excess Ice Formation

The initial winter data for Eugene showed an unusually high amount of defrost energy - often as much as 25 percent of input electric energy. Inspections revealed that there had been a factory recall on the outdoor unit refrigerant metering device. The metering device shipped with the unit was overly restrictive in heating mode, resulting in a starved evaporator under some conditions and therefore lower operating pressure, lower evaporating temperature and increased ice formation. Also, back-up elements were wired so that up to 15 kWh of element capacity could be used during defrost. This problem was corrected in March 2005 along with the other repairs. Figure 1 compares the binned COP performance before and after this repair. It is interesting to note that the significant fix at this site altered the defrost energy (and average COP) only -- it did not alter the operating COP in the heating modes. The Sunriver site also showed similar high fractions of defrost energy relative to total energy, 25+ percent. But this unit has a thermostatic expansion valve, TXV, not an orifice, and the cause of this high defrost energy has not been identified for certain.

Comfort Energy

Most resistance heating was for perceived "comfort" reasons. This comfort energy use plays no key role in the protection or performance of the unit; it is typically used only to add about 10°F to the supply air stream to prevent the appearance of drafts. But this use of "comfort energy" is embedded in the as-purchased control logic of most units. The most significant efficiency failure observed in this project was due to an unnecessary discharge air sensor installed at the Sunriver site. The sensor communicated with the indoor unit board, turning on backup (resistance) heat if the discharge air temperature did not reach 85°F within 4 minutes after the compressor began. The efficiency deterioration due to this sensor was large as shown in Figure 5. With the sensor removed, the supply air was often in the range of 80-85°F instead of above 90-95°F.

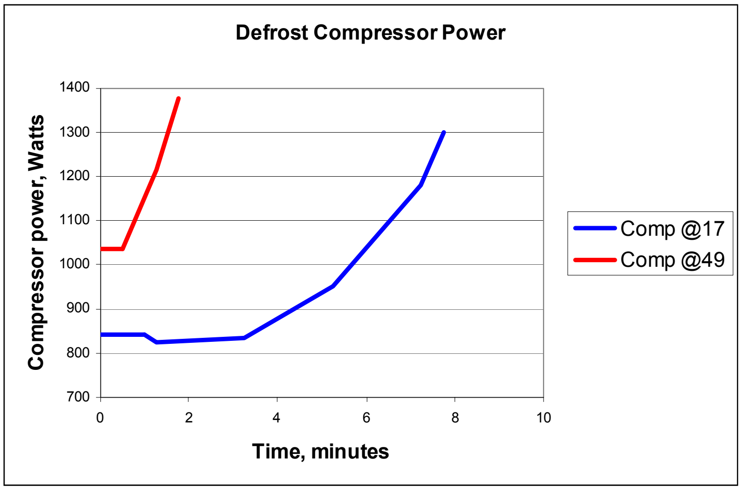
Figure 5. Pre/Post COP by Temperature Bins – Sunriver



Use of Resistance Heat During Defrost

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 be minimally impacted.

Figure 6. Defrost Compressor Power



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. Typically, the defrost process lasts for a few minutes and proceeds as in Figure 6. The compressor power associated with a frozen coil is less than 900 Watts. As the ice is melting (an isothermal process), the coil will remain at the same temperature and compressor power will remain below 900 Watts. As soon as the ice has fully melted, the coil will quickly heat up and compressor power will dramatically increase. Figure 6 shows the defrost compressor power for two conditions: (1) where there really is ice on the coil with an outdoor temperature of 17°F, and (2) where there is no ice on the coil and the outdoor temperature is 49°F. In the 17°F case, the compressor power remains at ~850 watts for about 6 minutes while the ice melts then the power increases to ~1,200 watts in the next few minutes. In the 49°F case, the compressor power rises to more than ~1,000 watts in the first few minutes. This would not occur if there were ice on the coil.

In both cases the cycle ends when the compressor power reaches more than 1,200 watts. The 49°F cycle is in essence a "probing cycle." It is checking for the presence of ice. At intermediate temperatures, there may be a "probing cycle" every half hour or so.

Both cycles end with use of resistance heat. In the case of the 49°F cycle, resistance heat is used for the last minute of the cycle, and it only serves to heat the ducts. This application of resistance heat is totally unnecessary. In the case of the 17°F cycle, about 6 minutes of 15 kW resistance heat is used. This is a subjective judgment issue: the air could probably be adequately tempered with 5 kW. It is important to note that resistance heat during defrost only reduces efficiency; the air tempering role can usually be done by the thermal mass of the heated ducts.

Occupant Control-Unnecessary Cooling

The focus of heat pump control is at the thermostat, and it is tempting to regard the thermostat as the sole control option. But for proper cooling in the Northwest, occupants need a broader understanding of their control options. The cooling operation at the Manzanita site was the most extensive, yet the outdoor air temperature was almost always below the cooling set point. A less extreme version of the cooling at Manzanita was observed at Sunriver and Eugene. At all three of these sites, mechanical cooling was observed during periods when cooling could easily have been accomplished by the available cool outside air. At these three sites, the occupants appear to have relied on the automaton; they did not realize that there was a role for them beyond the thermostat (namely, to open windows for manually controlled ventilation). Unnecessary cooling will be triggered whenever the house is treated as a sealed space and the outside air is colder than the cooling set point.

Tight Thermostat Deadband

At the Manzanita site, the cooling set point was 75°F, and the heating set point was about 72°F. Frequently heating would be observed immediately followed by cooling and vice versa. Such a tight control deadband forces the application of heating to compensate for over-cooling. These tight thermostat settings seriously degraded the heating and cooling efficiencies in the 55-70°F temperature bins at that site.

Summary of Improvements

This project was not confined to an observer role. As operational irregularities were found, they were corrected. In total, the analysis, the site inspections and re-inspections, and consultations with industry technical representatives led to the list of actual and recommended system improvements shown in Table 4.

Table 4. Actual and Recommended System Modifications

Site	Improvements	Additional Possible Improvements	Control improvements
The Dalles	Repaired ducts, deployed programmable electronic thermostat (with outdoor sensor which locks out resistance above 40° F)	Add duct insulation. Consider unit replacement (now 16 years old)	None
Sunriver	Disabled supply air sensor	Reduce defrost power, possible outdoor unit metering device	Fresh air cooling
Ashland	Added outdoor thermostat (in initial bid but mistakenly not installed)	Refine first stage heating, remove indoor unit from attic (not practical but would improve performance)	None
Eugene	Repaired ducts, repositioned outdoor sensor, replaced outdoor unit metering device (factory recall)	Reduce resistance heat during defrost	Fresh air cooling
Manzanita	None	Relocate thermostat and/or damper supply flow to reduce short-cycling	Fresh air cooling – adjust dead band

Recommendations

1. Avoid attic placement of return ducts or fan units in colder locations.
2. Carefully plan thermostat location in the design phase.

3. Avoid resistance heating by using a bona-fide two-stage unit, sized to meet load in the second stage.
4. At installation, check the fan airflow against the manufacturer's recommended airflow and configure the controls to provide the recommended airflows.
5. Avoid a thermostat deadband of less than 5°F. Avoid unnecessary cooling by increasing the cooling set point to about 78°F and use windows and doors for cooling when outside air is cool.
6. Do not allow use of a discharge air sensor on systems with only one stage of compressor heat. In warm locations it will never be needed. In cold locations, it can seriously degrade the efficiency of the unit. In cold locations it is best to recognize that supply air may be at temperatures as low as 80°F and to locate the supply discharge to minimize drafts in sedentary activity areas.
7. Disable all or at least halve the resistance heat associated with the defrost cycle. We doubt that backup heat is needed during the defrost cycle. Typically, cycles are short enough that the boost to supply air temperature is not significant relative to the amount of energy used.
8. Reduce the auxiliary heat sizing to 75 percent of design heating load from its current, and conservative, 150 percent. Connected resistance load is a regional demand "liability." The expectation that modern heat pumps have less energy use should be matched by the expectation that modern heat pumps have lowered demand.
9. Excess icing is very difficult to detect without monitoring. This type of problem can significantly depress heating performance. Defrost operation should be an explicit monitoring target, and there should be ongoing winter monitoring of a modest sample of heat pumps. If this type of problem is prevalent, then a means of detecting it at installation should be devised.
10. Be cautious about "two-stage scroll compressors". These were developed where precise control of compressor output is important (as in dehumidification), but current applications do not lead to better efficiency. If one is used, check the first stage running power and compare it to the second stage power. If the difference is less than 15 percent, then bypass the first stage.

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