FINAL REPORT

Analysis of
Heat Pump Installation Practices and Performance

Prepared for the

Heat Pump Working Group

ECOTOPE
CONSULTING
RESEARCH
DESIGN

Stellar Processes

David Baylon
Shelly Strand
Bob Davis
David Robison
Erin Kruse

December, 2005

TABLE OF CONTENTS

Executive Summary ES-1
1. Introduction 1
2. Billing Analysis and Field Sample Recruiting 2
  2.1. Program Description 2
  2.2. Sample Frame 3
  2.3. Utility Consumption Record Recruiting 4
  2.4. Field Sample Design 4
  2.5. C&RD / ConAug Sample Recruiting 6
  2.6. EWEB Sample Recruiting 8
3. Billing Analysis 8
  3.1. C&RD/ConAug Analysis 11
  3.2. Comparison (Control) Group 20
  3.3. Realization Rate 21
  3.4. EWEB CheckMe!® Billing Analysis 26
  3.5. EWEB Control Comparison Group 31
4. Market Actor Interviews 32
  4.1. HVAC Installers 32
  4.2. Trends in Heat Pump Manufacturing 43
  4.3. Interview Findings 46
5. Field Audits 47
  5.1. Field Protocol 47
  5.2. Field Characteristics 48
  5.3. Duct System Characteristics 51
  5.4. Heat Pump Characteristics 54
  5.5. Field Survey Findings 55
  5.6. Heat Pump HSPF 61
  5.7. System Sizing 62
  5.8. Homeowner Interviews 65
  5.9. Billing Analysis for the Field Sample 67
  5.10. Field Results 71
6. Laboratory Testing of Heat Pump Performance 73
  6.1. Adjustment Factors 73
  6.2. Results: Capacity and COP 74
  6.3. Results: Heating Seasonal Performance Factor 78
  6.4. Findings 82
7. Conclusions 73
  7.1. Program Savings 83
  7.2. Heat Pump Commissioning 84
  7.3. Heat Pump Controls 87
  7.4. Installation and Design Practice 90
  7.5. Overall Conclusions 91
8. Bibliography and Suggested Reading 92
Technical Appendix A A-1
Technical Appendix B B-1
Technical Appendix C C-1
Technical Appendix D D-1
Technical Appendix E E-1

Table of Tables

Table 1. C&RD & ConAug Heat Pump Installations by Area 3
Table 2. Utility Recruitment 4
Table 3. Billing Analysis Sample Frame (C&RD / ConAug) 5
Table 4. EWEB Sample Frame 6
Table 5. Field Sample Attrition 8
Table 6. Summary of Savings Estimates by Region 12
Table 7. Study Group Climate 15
Table 8. NAC Savings by Equipment Type 16
Table 9. NAC Savings by Equipment Type 17
Table 10. Square Footage by Equipment Type for C&RD Single Family 18
Table 11. Savings by Vintage, Single Family CR&D Cases 19
Table 12. Untreated Group, NAC kWh Consumption by Year 20
Table 13: Single Family Realization Rate by Pre-Retrofit System Type 21
Table 14: Manufactured Home Realization Rate by Pre-Retrofit System Type 22
Table 15: Realization Rate for Entire Sample by Pre-Retrofit System Type 22
Table 16. Analysis Results for Participant Groups 29
Table 17. EWEB Comparison Group Annual Consumption 31
Table 18. Sample UA and Air Tightness Characteristics 49
Table 19. Housing Type and Area 49
Table 20. Normalized Heat Loss Rate by Vintage 50
Table 21. Normalized Air Leakage Rate by Vintage 51
Table 22. Duct UA and Leakage Rates 52
Table 23. Duct Delivery Efficiency by Vintage 53
Table 24. HSPF of Heat Pumps by Region 61
Table 25. Heat Pump Sizing by Region 65
Table 26: Energy Use Estimates by End Use (N=122) 69
Table 27: Normalized Energy Use Estimates by Region and End Use 69
Table 28. Normalized Space Conditioning, Sample vs. RTF 71
Table 29. Purdue Adjustment Factors using Refrigerant-Side Capacity Measurements. 73
Table 30. C_d: From Test Data, ARI Default and Back-Calculated 79
Table 31. Defrost Multipliers: From Lab Data and ARI Default 81
Table 32. Comparison of HSPF results using Carrier YKC Catalog Data with Various C_d and Defrost Multipliers 82
Table 33. Summary of NAC Consumption and Savings by Region, Treated Cases A-1
Table 34. Summary of NAC Consumption by System Type, Treated Cases A-3
Table 35. Disaggregation of Savings by End use and Climate Zone A-5
Table 36. Test of Difference Between Regions A-6
Table 37. Test of Difference Between System Types A-6
Table 38. Difference Test, C&RD and non-C&RD Groups A-7
Table 39. Summary of NAC Consumption and Savings, ... A-8
Table 40. Test of Differences, Untreated Group, All Cases A-9
Table 41. Test of Differences, Untreated Group, Kitsap Cases A-9
Table 42. Test of Differences, Untreated Group, Yakima Cases A-10
Table 43. Test of Differences, Untreated Group, Oregon Redmond Cases A-10
Table 44. Analysis Results for Participant Groups B-3
Table 45. Purdue Adjustment Factors using Refrigerant-Side Capacity MeasurementsD-1
Table 46. Results of Duct Blaster® Tests D-4

Table 47. Nozzle and Duct Blaster Flows, and Duct Pressures with Respect to Ambient. Fan configuration identifies which fans were operating (DB = Duct Blaster®; Int. = Internal fan; Ext. = External fan). D-5
Table 48. Multipliers using Velocity Weighting D-6

Table of Figures

Figure 1. Disposition of Data Requests 6
Figure 2. NAC Regression Example 9
Figure 3. Heating-Only Temperature Regression Example 10
Figure 4. Heat/Cool Temperature Regression Example 11
Figure 5. NAC Savings by Region 12
Figure 6. Operations Profile, All Sites 13
Figure 7. Operations Profile, Cooling Zone 3 14
Figure 8. Enduse Consumption by Climate Zone 14
Figure 9. Enduse Savings by Climate Zone 15
Figure 10. Savings by Equipment Type 17
Figure 11. Building Size by System Type (N=470) 18
Figure 12. Savings by Vintage 19
Figure 13. Untreated Group NAC by Year with 90% C. L. 20
Figure 14. Pre- vs. Post-Installation Change in NAC 23
Figure 15. Square Footage by House Type and Pre-Existing System Type 24
Figure 16. *EZ*Sim® Bill Fit Example 28
Figure 17. Enduses Within Example Case 28
Figure 18. Example Post-Retrofit Match of Model to Consumption Records 28
Figure 19. Savings for Participant Groups 29
Figure 20. Pre/Post Operations Profile for Participants 30
Figure 21. Distribution Profile of Space Heat Consumption 31
Figure 22. Comparison Group Consumption 32
Figure 23. Heat Pump Sizing 34
Figure 24. Estimated Heating Seasonal Performance Factor (HSPF) 35
Figure 25. Factory-Installed TXVs (on Outdoor Units) 36
Figure 26. TXV Cost 37
Figure 27. TXV Acceptance 37
Figure 28. Outdoor Thermostat Penetration 38
Figure 29. Low Ambient Cut Out 39
Figure 30. First Stage Heating 40
Figure 31. R-410a Refrigerant 41
Figure 32. Customer Priorities 42
Figure 33. Influences on Consumer Buying Habits 43
Figure 34. House Size by Vintage 48
Figure 35. Heat Loss Rate (by Square Footage and Vintage) 50
Figure 36. Air Leakage Rate by Vintage 51
Figure 37. Distribution of Duct Efficiencies (by Vintage) 53
Figure 38. System Airflow Distribution 57
Figure 39. External Static Pressure 58
Figure 40. Refrigerant Charge Distribution 59
Figure 41. Field Sample Refrigerant Charge Evaluation 59
Figure 42. Indoor TXV 60

Figure 43. Outdoor TXV ... 60
Figure 44. HSPF by Region ... 62
Figure 45. Capacity Ratio (heat loss rate to nominal HP tons) ... 64
Figure 46. Capacity Ratio (heat loss rate to nominal HP tons) with Duct Losses ... 64
Figure 47. Distribution of Wood Heat ... 66
Figure 48. Distribution of Consumption by Load Type ... 68
Figure 49. Heating Load by Region ... 69
Figure 50. Cooling Load by Region ... 70
Figure 51. Capacity (fixed metering) versus Charge at three outdoor temperatures ... 74
Figure 52. Capacity (TXV) versus Charge at three outdoor temperatures ... 75
Figure 53. COP (fixed metering) versus Charge at three outdoor temperatures ... 76
Figure 54. COP (TXV) versus Charge at three outdoor temperatures ... 77
Figure 55. COP (fixed metering) vs Airflow Rate at three outdoor temperatures ... 77
Figure 56. COP (TXV) versus Airflow Rate at three outdoor temperatures ... 78
Figure 57. C_d versus Airflow Rate (row 1) and Charge (row 2) with a TXV (col. 1) and fixed metering (col. 2) ... 79
Figure 58. Capacity Defrost Multiplier versus Airflow Rate (row 1) and Charge (row 2) with a TXV (col. 1) and fixed metering (col. 2). ... 80
Figure 59. HSPF versus Airflow Rate (row 1) and Charge (row 2) with a TXV (col. 1) and fixed metering (col. 2). ... 81

Executive Summary

In 2004, a consortium of entities active in the conservation of energy in the Pacific Northwest, including the Bonneville Power Administration (BPA), the Northwest Energy Efficiency Alliance (Alliance), the Energy Trust of Oregon (Trust), the Northwest Power and Conservation Council (NPCC), Idaho Power and other regional utilities funded an in-depth study of heat pump performance in the Pacific Northwest. Heat pumps have enjoyed a significant increase in popularity in recent years, both with the public and with utility program designers. This study used a variety of analytical methods to assess the overall performance of heat pumps in Northwest climates and to identify the factors that have the most impact on the efficiency achieved. The study design was intended to address a number of project goals:

1. Assess the energy use and savings from heat pumps installed under the Conservation and Renewables Discount (C&RD) and Conservation Augmentation (ConAug) programs, and under the CheckMe!® program operated by the Eugene Water and Electric Board (EWEB).

2. Assess base case installation practices.

3. Assess heat pump performance under laboratory conditions to determine the performance impacts of variations from manufacturer-recommended refrigerant charge and air flow on system capacity and efficiency.

4. Assess the general approach of installers to control, sizing and performance issues; and of manufacturers to new technologies, etc.

To accomplish these goals, Ecotope conducted the following research steps:

1. ***Billing Analysis***: A large-scale billing analysis was conducted in targeted geographic areas across the region. These areas were chosen to represent a range of regional climate zones and building characteristics. A control group was selected to match each of the participant regions and also subjected to a billing analysis so that weather impacts could be removed from the savings calculations. A smaller analysis of about 400 customers of the Eugene Water and Electric Board (EWEB) was conducted to determine if savings could be attributed to use of a refrigerant charge and airflow field procedure used in the EWEB service territory and in other areas of the Pacific Northwest.

2. ***Field Review***: An extensive field review was conducted to examine the heat pump as installed, its set up and control strategy, and the characteristics of the ducts and house. This review included complete Duct Blaster® and blower door tests, as well as a check of the refrigerant charge. A separate billing analysis was conducted for this subset of the sample, which provided more detailed information for factors such as duct efficiency, Heating Seasonal Performance Factor (HSPF) and system sizing.

3. ***Laboratory Testing***: Ecotope developed a matrix of testing requirements to determine the impact of charge and airflow on overall heat pump efficiency under a variety of loads, and contracted with Purdue University to conduct these tests under strict laboratory conditions. These tests were conducted in heating mode and were meant to augment the data previously collected on cooling mode performance, primarily for the California climate.

4. ***Distributor & Installer Interviews***: Ecotope interviewed installers, distributors, manufacturer's representatives and other stakeholders to elicit data on equipment selection, sizing strategies, installation techniques and other issues that impact heat pump performance.

This effort has resulted in a much deeper understanding of the performance of heat pumps as installed under the C&RD and ConAug programs to date, as well as important information about what factors have the greatest impact on that performance.

Overall Findings and Conclusions

The various avenues of inquiry converged on some clear results regarding heat pump performance.

- From the billing analysis, it is clear that heat pumps are performing at or near what might be considered the expected level, at least for the C&RD/ConAug program participants. Savings averaged approximately 4,149 kWh/yr., representing about 15% of total electricity use. The overall realization rate was about 70% of savings anticipated when original Regional Technical Forum (RTF) estimates were prepared. The RTF had revised its original savings estimates downward as more data became available. When compared to the newer estimates, the realized savings were about 85% of expected.

- The EWEB billing analysis indicate average savings of about 360 kWh/yr. compared with a control group that did not receive the CheckMe!® service. Savings could not be attributed to the performance of the CheckMe!® procedure itself but since about 85% of the savings seen in the EWEB billing analysis came from the top 15% of heating energy consumers, it appears the correction of severe problems in a limited number of cases was more important than adjusting refrigerant charge and/or airflow.

- The Purdue laboratory data, the EWEB billing data, and the field review all indicate that running the system with non-optimal refrigerant charge does not have a significant impact on heat pump performance in heating mode. The system had to be run at 20% or more undercharge before any reduction in efficiency was noted. In fact, the data points to a slightly undercharged system as the optimal condition in heating mode. This was an unexpected finding, since previous research focused exclusively on the cooling mode did find significant efficiency impacts from over- or under-charged compressors.

- The field study of base case installations found a number of important findings. First, only about 10% of systems were found with undercharged compressors. Only a few systems out of the approximately 140 evaluated were seriously undercharged, and these systems were overdue for service. Low airflow across the indoor coil was noted in about 25% of cases. Airflow is an important determinant of field performance and remains a central part of ongoing field verification efforts in the Northwest. Control of auxiliary electric resistance heat in non-C&RD heat pumps is not carefully done and is assumed to be handled by adaptive recovery thermostats even though it can be easily circumvented by the system operator.

- There is a fairly high level of education about efficiency issues amongst regional installers, according to both our field audit results and contractor interviews. Installers generally understand the trade-offs inherent with heat pumps (more comfort compromises efficiency) and usually come down on the side of more comfort. This should be of concern to regional policymakers and utilities that expect rated efficiency from new heat pumps.

- Heat pump systems tend to be sized to about 70% of the required heating load according to the field research and interviews. Contractor interviews indicate that this is due primarily to first-cost considerations. Larger systems (more "tons") mean most or all of heating season requirements can be met by the refrigerant cycle rather than by auxiliary heat, but it is cheaper at the initial point of installation to install a smaller compressor and a larger resistance element combination. There is ongoing debate in the region on the best way to size a heat pump. The expected increasing use of multiple-capacity compressors will complicate this issue but may result in more heating energy coming from the refrigeration cycle and less from auxiliary heat, which will enhance the effectiveness of the heat pump in delivering conservation.

- Despite the continued positive development of regional installation standards, there is still a need for field verification of system performance. This field verification cannot be limited to evaluation of charge and airflow, as it has been in the past, but must be extended to system controls. Field verification cannot be limited to the installer's report but must include an additional layer of quality assurance. This is both to ensure the performance of the system and to maintain currency with the more advanced systems that will be installed in 2006 and later.

1. Introduction

In late 2004, the Ecotope team undertook the Heat Pump Maintenance Project. This project was funded by a consortium of entities active in the conservation of energy in the Pacific Northwest, including the Bonneville Power Administration (BPA), the Northwest Energy Efficiency Alliance (Alliance), the Energy Trust of Oregon (Trust), the Northwest Power and Conservation Council (NPCC), Idaho Power and other regional utilities. The study design was intended to address a number of project goals:

1. Assess energy use and savings from heat pumps installed under the Conservation and Renewables Discount (C&RD) and Conservation Augmentation (ConAug) programs and under the CheckMe!® program operated by the Eugene Water and Electric Board (EWEB) during the period from January, 2002 through December, 2004.

2. Assess base case installation practices.

3. Assess heat pump performance under laboratory conditions to determine the performance impacts of variations from manufacturer-recommended refrigerant charge and air flow on system capacity and efficiency.

4. Assess the general approach of installers to control, sizing and performance issues; and of manufacturers to new technologies, etc.

To accomplish these goals, Ecotope conducted the following research steps:

1. ***Billing Analysis***: A large-scale billing analysis was conducted in targeted geographic areas across the region. These areas were chosen to represent a range of regional climate zones and building characteristics to capture a thorough examination of the C&RD and ConAug savings estimates. A control group was selected to match each of the participant regions and also subjected to a billing analysis so that weather impacts could be removed from the savings calculations.

2. ***Field Review***: An extensive field review was conducted to examine the heat pump as installed, its set up and control strategy, and the characteristics of the ducts and house. This was a two-part protocol in which the bulk of the review was conducted by an energy researcher, including complete Duct Blaster® and blower door tests. A second visit to the home was typically scheduled for a certified HVAC contractor with the credentials needed to check the refrigerant charge. A separate billing analysis was conducted for this subset of the sample, which provided more detailed information for factors such as duct efficiency, Heating Seasonal Performance Factor (HSPF) and system sizing.

3. ***Laboratory Testing***: Ecotope developed a matrix of testing requirements to determine the impact of charge and airflow on overall heat pump efficiency under a variety of loads, and contracted with Purdue University to conduct these tests under strict laboratory conditions. These tests were conducted in heating mode and were

meant to augment the data previously collected on cooling mode performance, primarily for the California climate.

4. ***Distributor & Installer Interviews***: Ecotope interviewed installers, distributors, manufacturer's representatives and other stakeholders to elicit data on equipment selection, sizing strategies, installation techniques and other issues that impact heat pump performance.

This effort has resulted in a much deeper understanding of the performance of heat pumps as installed under the C&RD and ConAug programs to date, as well as important information about what factors have the greatest impact on that performance.

2. Billing Analysis and Field Sample Recruiting

2.1. Program Description

The utility heat pump installation programs we reviewed were developed as part of the BPA and Regional Technical Forum (RTF) efforts to provide some regional direction to utility conservation programs. The programs are collectively known as Conservation and Renewable Discount/Conservation Augmentation (C&RD/ ConAug). While they differ in administrative details, these two programs both began in late 2001 and use similar methodologies for claiming benefits under the BPA conservation programs. For this project, heat pump installations that took place from January 2002 through December 2004 were studied.

The RTF is a consortium representing all the region's utilities, as well as the State energy programs and various public groups. The RTF developed a series of specifications that described the heat pump selection and installation, and calculated an agreed-upon savings estimate for each installation type. This entire process was conducted over several years, and savings estimates were revised as more technical information became available or as utilities found the need to present alternative measures and program standards.

The result of this process was a series of "deemed" electric energy savings estimates that are used by the utilities to assign monetary value to their conservation programs (in this case, heat pump installations). This value is then used to calculate rate discounts for the utility's power purchases from the BPA. It is these energy savings estimates that are reviewed here. The effect of these reviews is to estimate the fraction of achieved savings observed and to propose more refinements to the calculation procedures to be used in the development of future deemed savings estimates.

A particular variation in this program was implemented in the EWEB service territory. In this utility, the benefit of a detailed maintenance review of existing heat pumps was tested as a potential source for cost effective energy savings. This program was based on a successful model used in California (and elsewhere) to review air conditioning units and check their refrigerant charge and system air flow. This approach was marketed and administered by the Proctor Engineering Group under the trade name CheckMe!®. In this approach, previously installed heat pumps were visited by technicians and evaluated for refrigerant charge and air flow. The program assumed that this review would also find and fix other problems. Savings would accrue from correcting variations between manufacturer's specifications and the observed conditions at the site.

This technique was also included in the installation standards for the new heat pumps installed under the C&RD/ConAug programs. However, this particular program run at EWEB afforded an opportunity to observe the impact of the CheckMe!® installation standards separately.

2.2. Sample Frame

The goal of the sampling design and methodology was to identify and target areas with sufficient climate and stock diversity to serve as an appropriate representation of the region as a whole. Several factors were considered when selecting the sample areas including climate, building practices, heat pump saturation, and C&RD/ConAug activity. EWEB's CheckMe! ® program was processed separately. The total C&RD/ConAug heat pump installation population served as the sample frame from which the representative regions were selected. The sample frame is shown in Table 1. The shaded rows indicate those areas selected for this study.

Table 1. C&RD & ConAug Heat Pump Installations by Area

Location	**FY01**	**FY02**	**FY03**	**ConAug**
Tri Cities	101	218	216	94
Spokane	3	125	126	661
Clallam	3	330	126	0
Columbia/Portland	0	98	169	198
Puget Sound	2	1	2	7
Central Oregon	17	196	317	69
Columbia Gorge	0	115	99	0
Willamette	0	3	6	502
Northern Tier	0	55	25	55
Coastal	11	376	107	47
EWEB				286
Grand Totals:	**137**	**1,654**	**1,193**	**1,919**
Selected Area Totals:	**132**	**1,218**	**935**	**694**

Note: The shaded rows indicate areas selected for this study.

2.3. Utility Consumption Record Recruiting

The original sample design targeted five sample areas (in addition to EWEB). Once these areas were identified, each utility in each sub-region was contacted by BPA, informed of the project goals, and urged to participate in this study effort. Following that initial introduction from BPA, Ecotope contacted each of the utilities and requested participation and billing data. Most of the regional utilities were easily recruited and provided timely, high quality data. Only in the Central region were we unable to successfully recruit any of the local utilities. Therefore, this region was dropped from the billing analysis, along with the control group from Bend (although the Bend field sample was retained). This is extremely unfortunate because the central Oregon area is a good representative of the Zone 2 heating climate. Table 2 shows the results of the utility recruitment effort.

Table 2. Utility Recruitment

Location	Contacted	Recruited	Control Area
Tri-Cities	4	4	Yakima / Walla Walla
Clallam County	2	2	Kitsap Peninsula
Portland/Columbia*	5	3	Portland
Central Oregon	2	0	Bend
Coastal Oregon / Washington*	3	2	Kitsap Peninsula
Eugene (EWEB)	1	1	Eugene (EWEB)
Totals:	**17**	**12**	

* One non-participating utility in this area was willing to participate, but could not for technical reasons.

2.4. Field Sample Design

For the C&RD/ConAug participant groups, the list of participating customers provided by the utilities served as the sample frame. A comparable random sample of homes with heat pumps located in a nearby non-participating utility service area was generated for each of the participant groups to serve as a control group. This control sample was drawn at random from a service territory with similar climate and other factors (see Table 3).

For the C&RD/ConAug participant groups, the sample frame came from the list of savings coupons provided to the BPA by the utility to claim savings credits. Only about 50 of the homes were from a ConAug program, which was designed to meet C&RD specifications. Therefore, for the remainder of this discussion, the ConAug cases will not be referred to separately.

The original sample design included a control group for the Portland region drawn from both Portland and southwestern Washington. While we did draw samples for these areas, we were much less successful in recruiting homeowners to participate in our study in Portland than in any other area, primarily due to the lack of heat pumps in the chosen utility service territory. Therefore, the Portland sample proved quite small. In southwestern Washington, the sample was comprised of customers that participated in a local utility (Clark County PUD) heat pump program. That utility did not participate in the C&RD nor use the specifications and requirements of that program. Nevertheless, the utility's program was very similar and the billing analysis revealed substantial savings. This area was included in our analysis and is reported with the other heat pump program results as "Non-C&RD". The field review in the Clark County PUD area suggested that the heat pump specification was less stringent than C&RD on the equipment HSPF but more inclusive of installation practices (although the Clark County PUD heat pumps did have much higher HSPF ratings than the standard practice control groups). These two factors may roughly cancel each other in the overall performance of these heat pumps.

Table 3. Billing Analysis Sample Frame (C&RD / ConAug)

	Expected	**Received**	**Complete**	**Analyzed**
Treated Group				
TriCities	584	388	347	318
NW, Kitsap	500	387	308	302
Central OR	272	0	0	0
Coast	502	194	194	134
Portld Area	364	366	288	268
C&RD	2,002	1,130	932	836
not C&RD	220	205	205	186
Total	2,222	1,335	1,137	1,022
Untreated (Control) Group				
TriCities	250	167	167	154
NW, Kitsap	249	251	225	81
Central OR	209	124	124	96
Ptld Area	39	37	12	11
Total	747	579	528	342

For the EWEB CheckMe!® analysis, the participant sample was chosen from EWEB customers that participated in the CheckMe!® program and received an airflow or refrigerant charge adjustment (or both). The original sample design called for the control group to consist of CheckMe!® customers that had not required any adjustments. However, when the initial analysis showed similar energy savings for both the sample and the control group, we recruited a second control group from other EWEB heat pump customers that had not participated in the CheckMe!® program during the period of interest. These results are shown in Table 4.

Table 4. EWEB Sample Frame

	Raw Data Received	Complete Data	Analysis Set
Participants	598	334	322
Non-Participants	372	131	80

2.5. C&RD / ConAug Sample Recruiting

There was significant attrition in both the billing analysis and field samples. For the billing analysis, this was primarily due to the quantity and quality of the data we were able to collect from the utilities. The attrition rates for the field sample varied considerably by area. For the most part, attrition was primarily due to homeowner disinterest. Schedule conflicts and existing maintenance agreements also contributed to the field sample attrition rate.

In this type of analysis, cleaning the data received from the utilities is an important part of the process. During this cleaning, cases are removed from the study group due to turnover in the customers and other factors. Such attrition was larger than expected for this project. By an unfortunate coincidence, the study period coincided with the time during which many utilities changed their accounting system. As a result, only partial data were available for many cases. Finally, the analysis identified a few cases that could not be used due to partial vacancy or otherwise incomplete data. The sample attrition is summarized in Table 3 above.

In general, the response to data requests was not as complete as desired, as shown for the treated cases in Figure 1. Consumption data was available for analysis for only about half the expected cases. A major shortfall was due to the fact that utilities supplied fewer bills than requested. As a result, many cases lacked sufficient usable data due to transition gaps in records. Cases that involved extended periods of vacancy were also classified as incomplete. As Figure 1 clearly shows, very few cases were dropped due to the inability to obtain a suitable regression fit – the number of cases is so small, the line is virtually invisible.

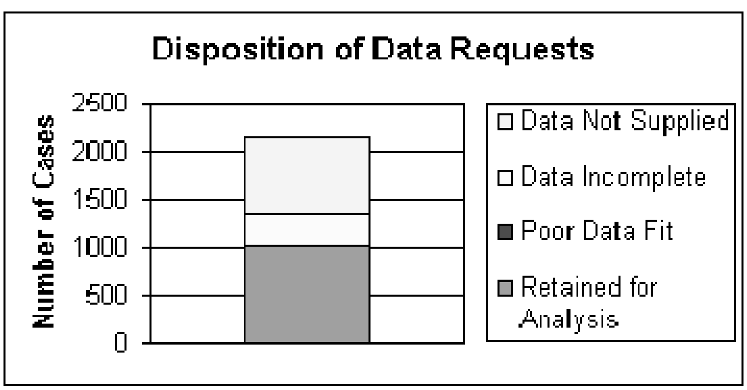

Figure 1. Disposition of Data Requests

The sampling strategy for the control group was designed to develop a representative sample of homes with heat pumps in each target locality. This was accomplished in different ways for each area, depending upon the available resources.

For the Kitsap County/Puget Sound control group, a random sample of accounts (with phone numbers) was drawn by the utility. This provided the basis for a phone recruiting survey that screened for homes with heat pumps to serve as a sample frame.

There was no comprehensive utility list in existence for the Bend area. We purchased a commercial list of about 2000 names and phone numbers of owners of homes with heat pumps. A similar commercial list was available for the Yakima/Walla Walla, Washington area. A phone survey team used these lists to recruit control groups from both areas. (The Bend control group was used only for the field survey). The protocol is included in Appendix C.

In Clark County, the control group sample was drawn from a utility list of homes participating in their (non-C&RD) heat pump installation or upgrade programs. In some cases, this list may have included homes receiving other weatherization services during the period prior to the year 2004. Although an effort was made to identify and screen out these cases, in this one group the final sample may not have provided an unbiased survey of heat pumps in Clark County. The total number of heat pumps in the set provided by the utility was about 700; this represented the customers that Clark County PUD knew to have heat pumps.

As a secondary part of the review, individuals from among the primary recruitment samples were asked if they'd like to participate in a field survey. Respondents were offered a $50 incentive for allowing the additional fieldwork to be conducted in their homes. Approximately 40 homes in each area were recruited, for a total of about 160 field audits.

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 better 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. Table 5 shows the final distribution of sites receiving all aspects of the field review. In future studies, the heat pump review should probably be delayed until both appointments can be made in quick succession. This is especially true given the laboratory results on the impact of charge discussed in Section 6.

For the most part, we believe that the field surveys are unbiased representations of the heat pumps in all localities. However, it would be a stretch to assume that this is representative of heat pumps throughout the region if taken as a whole. We assert that these are at least very good approximations of heat pump installation practices in the areas these units are located.

Table 5. Field Sample Attrition

Area	House & Duct Audit	Heat Pump Review
Central Oregon	40	27
Clark County	40	36
Kitsap	40	33
Yakima / Walla Walla	40	30
Total	160	126

2.6. EWEB Sample Recruiting

The EWEB sample was cleaned in the same manner as used in the C&RD analysis. The sample attrition is summarized in Table 4 above. Participants were dropped if any one of several factors was noted:

- customer turnover
- otherwise incomplete billing records
- when the participant record included a comment of "withdrawn"

3. Billing Analysis

For this project, the Ecotope team conducted two billing analyses:

- ***C&RD/ConAug Billing Analysis***: The primary analysis was intended to quantify the impact of heat pump installations on total energy use and annual savings. Bills were collected from 11 utilities in four geographic areas that participated in the C&RD or ConAug programs, and from matched utilities in the same areas that did not participate in either program. We conducted an engineering-based pre- vs. post-consumption comparison using a standard weather-normalized regression model in combination with an engineering model.

- ***EWEB Billing Analysis***: Separately, we analyzed a sample provided by EWEB to assess the impact of their program promoting the review of operating heat pumps and check of various operating parameters (especially refrigerant charge and airflow). This analysis used a similar methodology to the C&RD review. The goal was to quantify the annual energy savings available from an O&M program.

For this study, we used a standard pre- vs. post-installation cross-sectional consumption (billing) analysis. The purpose of the billing analysis is to compare energy consumption due to the treatment, while controlling for other extraneous variables. As a first approximation, we attempted to control for differences that may be due to weather or to gaps in the consumption records. To do so, we developed an estimate of normalized annual consumption (NAC) using a temperature-based regression technique. The NAC is defined as the energy consumption expected for a year of typical weather. In this manner, consumption is corrected for weather extremes. The same procedure can also fill in estimated consumption for months where data may be lacking. The weather-normalized annual consumption (NAC) before the treatment establishes a baseline, which can then be compared to weather-normalized consumption after the treatment. The difference in consumption determines gross savings. The formula used is shown in Equation 1.

$$\text{Gross savings} = NAC_{pre} - NAC_{post}$$

Equation 1

Gross savings are determined for the comparison group in the same way. The participant savings are corrected for any consumption change apparent in the comparison group. The result is net savings attributable to the program. This "difference of differences" approach is traditionally used in demand side management (DSM) evaluation methodologies to determine net savings due only to the treatment.

The temperature vs. consumption pattern in the billing data is reduced to three components: baseload, variable slope and balance temperature (Figure 2). The baseload represents energy used for lighting and appliances that is roughly constant each month. The space heating component is defined as consumption related to cold temperatures, and the balance temperature is the temperature below which heating occurs.

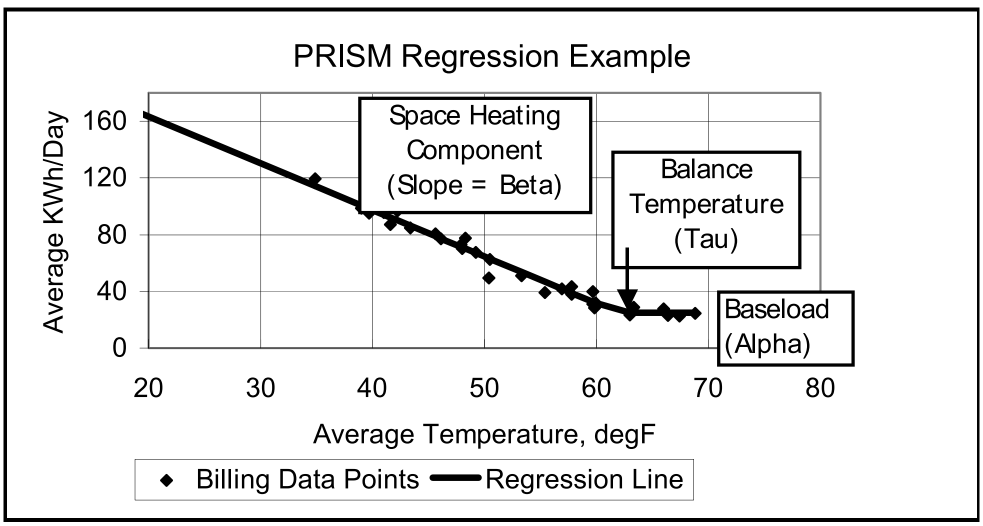

Figure 2. NAC Regression Example

We quickly found that the regression technique proved unreliable for our data. There were several factors that interfered with the ability of PRISM® to find a reliable regression model:

- We often had only partial data for the year
- There was occasional use of wood heat within the sample
- Many homes had air conditioning.

Instead, we developed our own temperature regression model. Figure 3 shows an example. In this model, we operated the regression over a range of balance temperatures and chose the one that provides the best fit to the data. Optimizing the balance temperature for both years together eliminated one regression variable and resulted in a more robust regression model when observations were sparse. We also reviewed each case for outlier observations that may be due to vacancy or wood heat use.

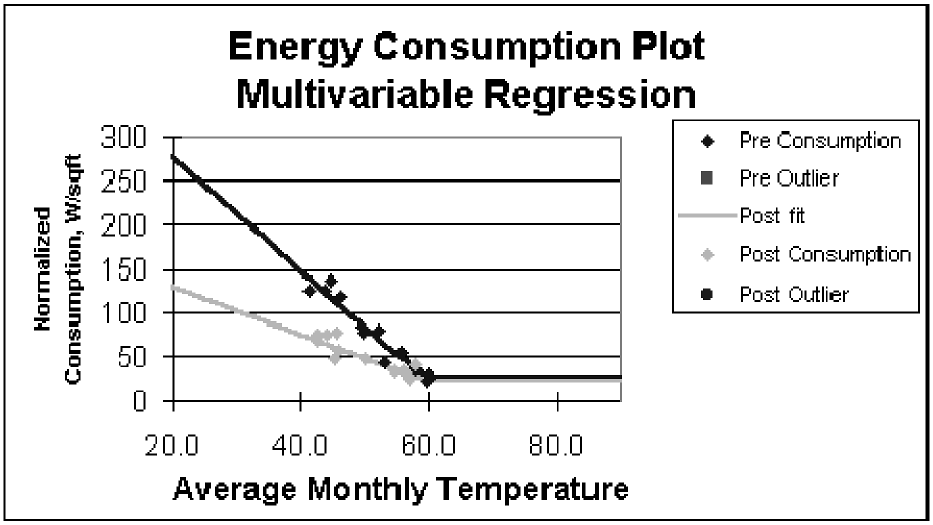

Figure 3. Heating-Only Temperature Regression Example

Finally, we developed a similar regression model that includes cooling as well as heating (Figure 4). This second model was necessary for cases in a cooling climate zone.

With the temperature regression method, we were able to analyze most of the sites with sufficient consumption records. It must be mentioned that the disaggregation of consumption into baseload, heating and cooling is only an approximation of enduses. Some enduses, such as lighting and water heating, have seasonal changes that will be included in the regression components. For that reason, it is best to look at the whole-building NAC as the most accurate indicator of consumption. The derived estimates of heating and cooling enduses are less accurate than the whole-building consumption.

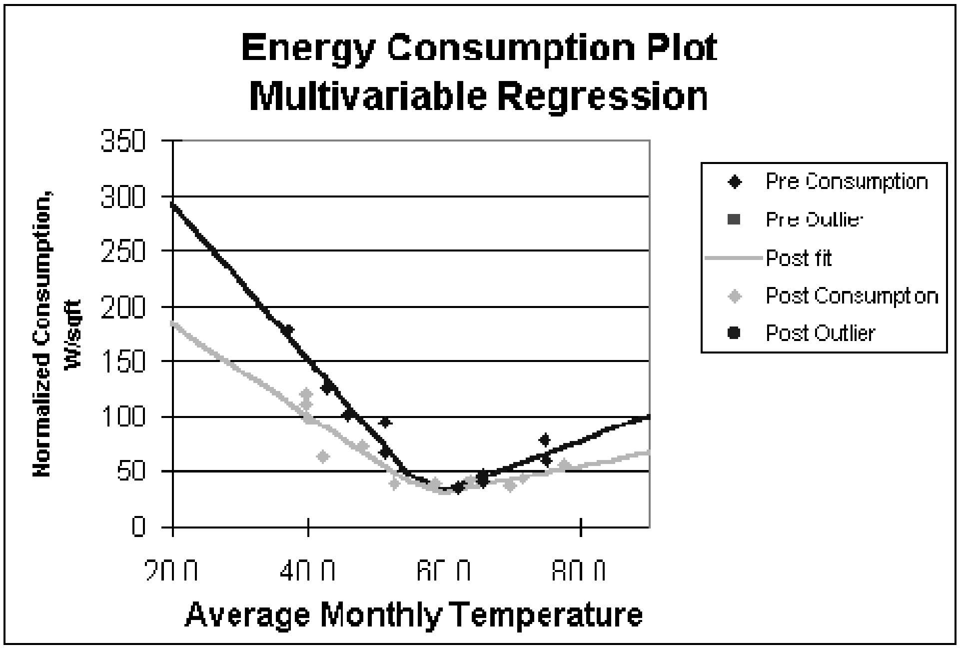

Figure 4. Heat/Cool Temperature Regression Example

3.1. C&RD/ConAug Analysis

Results of the analysis are shown below by region and by equipment type. It should be noted that, while estimates of the space heating (SH) must be recognized as only an approximation, that enduse accounts for most of the savings. NAC savings averaged 4,263 kWh per year with a confidence interval (C.I.) of +/- 292 kWh per year. Table 33 in Appendix A provides more detailed information about the NAC consumption and the savings results.

3.1.1. NAC Savings by Region

Results are shown by region in Table 6 and Figure 5. The differences between regions were confirmed as statistically significant using an analysis of variance test (ANOVA), which is detailed in Appendix A. However, those differences are due to a higher mean in the Northwest Washington area. The other regions are not significantly different. In general, there is little difference in savings between the different regions or between the C&RD or non-C&RD groups.

Table 6. Summary of Savings Estimates by Region

Region	n	Mean NAC Saved	SD	90% CL	Mean PreNAC	SD	Mean Saved %
TriCities	318	3,795	5,336	492	24,414	8,328	13.7
NW, Kitsap	299	5,111	6,198	587	24,226	10,363	17.4
Coast	137	3,986	6,516	927	24,415	15,888	12.2
Ptld Area	82	4,380	4,223	768	24,356	8,395	16.8
C&RD	836	4,354	5,767	328	24,343	10,601	15.1
not C&RD	186	3,851	5,185	626	25,379	8,936	12.3
Total	1,022	4,263	5,666	292	24,532	10,321	14.5

Table 6 and Figure 5 include savings for a non-C&RD group. This group is from a utility program that provided similar heat pump replacements but did so outside the C&RD program. This group is of interest to determine if the C&RD specifications provide any higher level of savings. Given the relatively small sample size of the non-C&RD group, any difference is not statistically significant. This hypothesis is confirmed by an ANOVA test detailed in Appendix A.

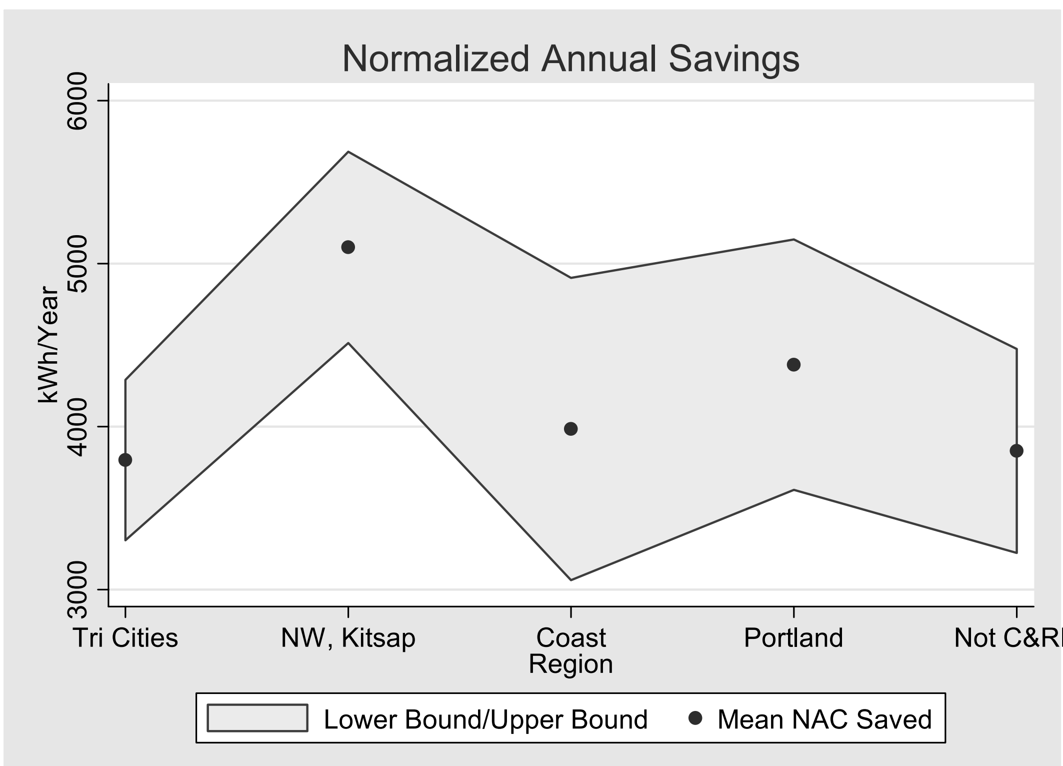

Figure 5. NAC Savings by Region

There is an important caveat to the assessment of overall savings from this program. The billing analysis summarized in Table 6 includes substantial variation in population characteristics depending on the individual utility program. Some of the variance observed here is due to differences in the population of homes treated. The principle variations are due to the conditions of the heating systems before the heat pump measure was installed (the base case). In all cases, the base case is used as the basis for calculating the deemed savings. Since the overall NAC savings shown here cut across all house types, house vintages and base case equipment types, these savings must be subdivided to be used to assess any particular utility or region.

3.1.2. Savings by Climate Zone

For this study, we were not able to acquire observations across the desired range of climate zones since we were unable to include Central Oregon. The cases studied are primarily considered to be in Heating Zone 1, Cooling Zone 1 and Heating Zone 1, Cooling Zone 2. The only exception was the Tri-Cities area, which is in Heating Zone 1 and Cooling Zone 3. To summarize the billing analysis, two climate zones were used. This was the result of the commonality between Cooling Zone 1 and Cooling Zone 2 observed in evaluating the savings, especially the observed cooling pattern. The Cooling Zone 3, however, offered a distinct pattern. These climates have been designated Zone 1 and Zone 3 respectively.

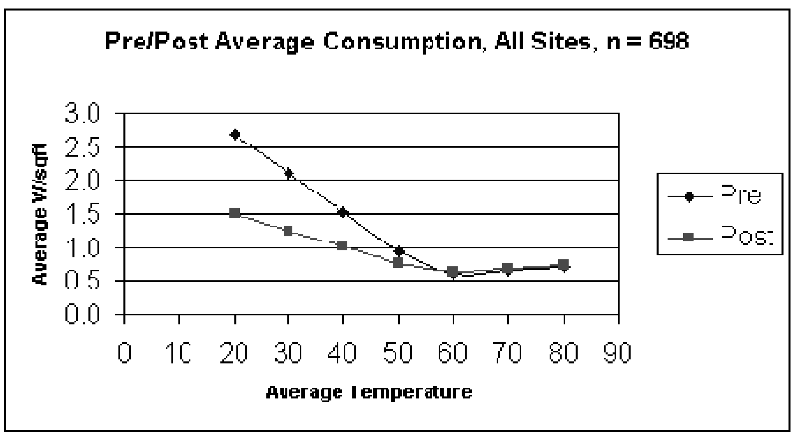

Figure 6. Operations Profile, All Sites

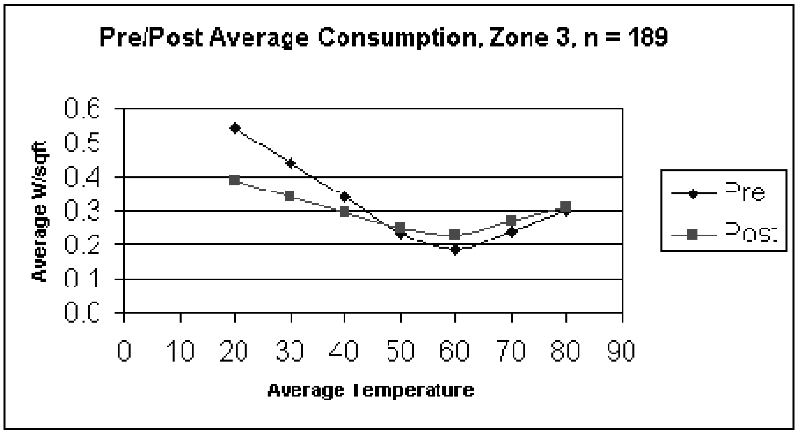

Figure 7. Operations Profile, Cooling Zone 3

The impact due to climate is presented in the following figures. The operations profile plots show the average rate of energy consumption per square foot, although square footage information was not available for all cases. Figure 6 shows the overall profile, while Figure 7 shows the profile for sites in the Tri-Cities area. In this figure, the effect of cooling consumption on overall operations is clearly apparent. Note that post-retrofit cooling extends over a broader range of temperatures. This may be an indication of "partial takeback" – which would mean that households are using air conditioning more.

There are differences in the enduses within the normalized annual consumption. Cooling Zone 1 exhibits very little air conditioning consumption. Table 35 in Appendix A details the enduse breakdown for consumption and savings by climate zone. The same information is shown below in Figure 8 and Figure 9.

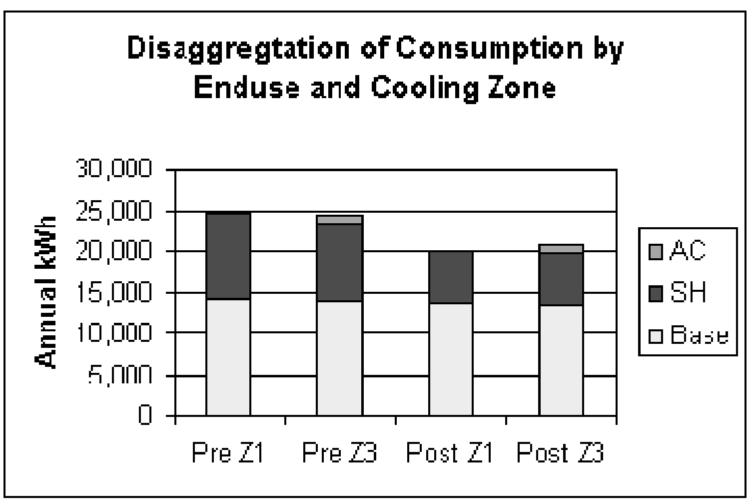

Figure 8. Enduse Consumption by Climate Zone

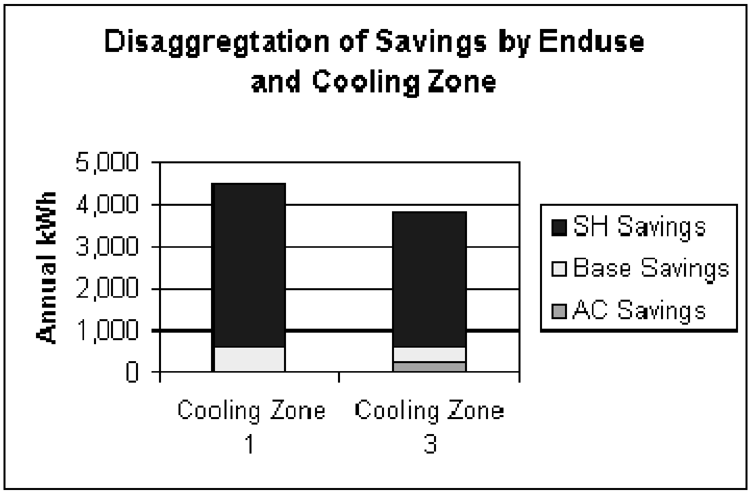

Figure 9. Enduse Savings by Climate Zone

In Cooling Zones 1 and 2, there is a small but significant "takeback" of savings due to a few sites that added air conditioning. In Zone 3, the net savings for cooling includes a positive increase in efficiency as well as the more complicated change due to increased operation, as shown in Figure 7.

The study group in Zones 1 and 2 included a slightly heavy weighting of cases in the colder Olympic Peninsula climate. However, examination of the savings-weighted average climate data is close to that assumed by C&RD as typical for the region. Climate data for the study group are shown in Table 7.

Table 7. Study Group Climate

Weather City	Heating Degree Days	Cooling Degree Days
Richland	4,828	883
Port Angeles	5,671	28
Hoquiam	5,164	31
Astoria	5,116	18
Portland	4,520	346
Not C&RD (Clark County)	4,520	346
Weighted Average		
C&RD Group	5,203	343
Total Study	5,091	344
Cooling Zone 1	5510	29
Cooling Zone 2	4520	346
Cooling Zone 3	4,828	883
C&RD Cooling Zone 1	5,008	320
C&RD Cooling Zone3	5,008	990

The summary of the savings estimates by climate zone show a distinct pattern as shown in Table 8. The savings are influenced by the cooling zone but heating degree days make more difference than the cooling load or cooling “take back”. This data set suggests that the cooler zones (especially the coastal zones) had about 20% more space heat savings. Virtually all the difference between these cases is accounted for by this extra savings.

Table 8. NAC Savings by Equipment Type

System Type	Mean Saved NAC	SD	n	90% CL
Cooling Zone 1	4,756	6,318	437	495
Cooling Zone 2	4,014	5,010	267	502
Cooling Zone 3	3,794	5,308	319	487
All Sites	4,263	5,703	1,022	294

3.1.3. Savings by System Type

Table 9 shows savings by the pre-retrofit system type. The system types include forced air furnace (FAF) with and without central air conditioning (CAC) as well as baseboard resistance (zonal) and replacement heat pumps.

Savings for the replacement heat pumps are surprisingly large. However, this is due to the defined base case rather than amazing efficiency. In the C&RD savings estimates, this measure was defined as the incremental upgrade from a low-efficiency replacement unit to a higher efficiency model and was expected to produce modest savings. In fact, we see savings comparable to replacing an electric furnace. One possibility for these high savings is that the home has a non-functioning heat pump system that is acting as a resistance furnace prior to replacement.

Savings for zonal systems are lower. However, these homes are about 500 square feet smaller, on average, than the homes with heat pump systems. These homes also do not have ducts or associated duct losses. Savings for systems with air conditioning are also lower; which may be due to some “take-back” in the form of increased air conditioning consumption, as suggested by Figure 7. Table 34 in Appendix A details results by system type. Appendix A is limited to the summary of cases that corresponded to the exact utility claim. Table 9 expands that definition to combine categories as long as the base equipment was as claimed. The principle difference comes from the fact that 21 cases had not claimed “PTCS” in their original installation. In Table 9 they were included where appropriate. In the equipment summary, only C&RD claims were accepted. For the total, in “All Sites” the non-C&RD cases were included as well as C&RD cases that had not reported base equipment type. A detailed summary of sub-groups in this sample is located in Appendix E.

Table 9. NAC Savings by Equipment Type

System Type	Mean Saved NAC	SD	n	90% CL
Heat Pump	4,352	5,861	99	970
FAF w/CAC	4,498	4,636	236	497
FAF w/oCAC	5,018	6,096	255	628
Zonal	2,627	5,581	130	806
All Sites	4,324	5,703	1022	294

These results show a pattern of savings for the forced air furnaces that is similar to the expected pattern in the deemed savings for the C&RD program. When the system being replaced by the new heat pump is an older heat pump, however, the observed savings is comparable to the cases where the equipment replaced is an electric resistance furnace. This suggests that the heat pump that is replaced has ceased to function and has run for some time as an electric furnace. When the heat pump replaces "Zonal" heating, a duct system must be added as well as a central air conditioning system. Even though the heat pump should be much more efficient, the addition of the duct losses and the added air conditioning load (especially in cooling zone 3) reduces the savings for this combination both in the deemed savings calculator and in the savings calculated from the billing analysis.

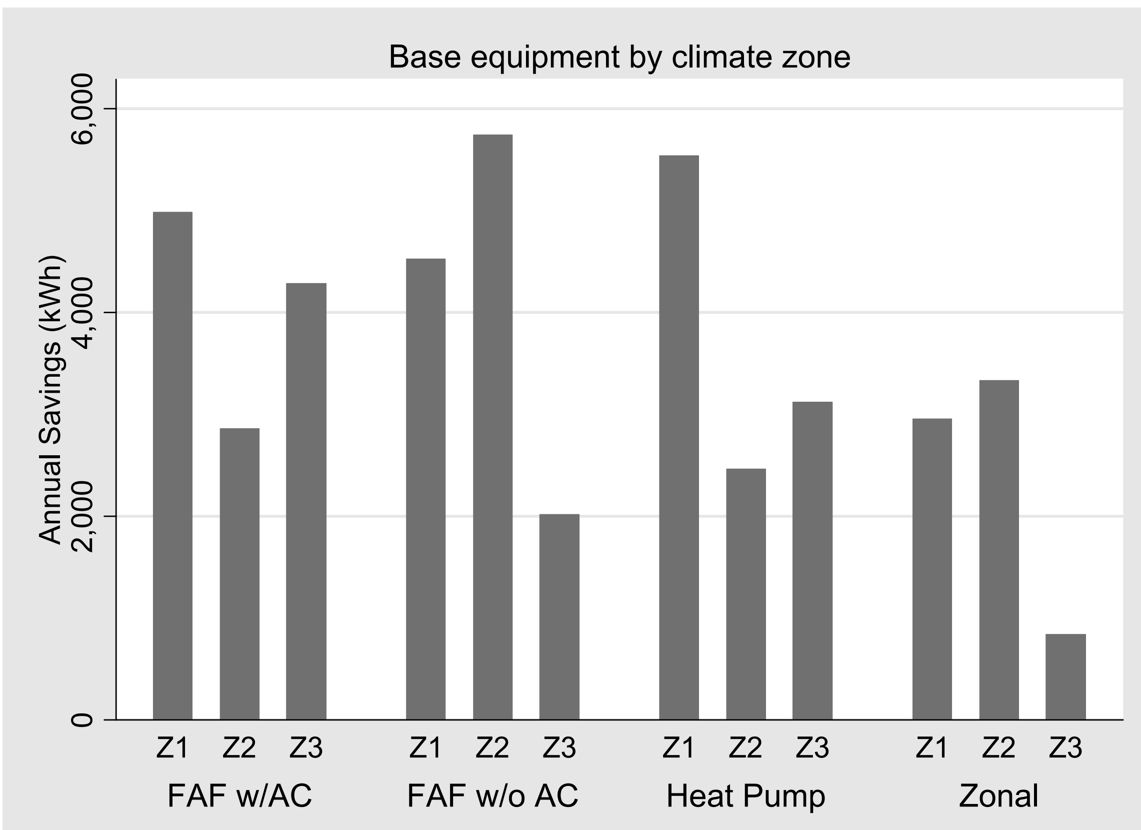

Figure 10. Savings by Equipment Type

Figure 10 recasts the savings estimates by base equipment to show the differences in savings by climate zone. While there are strong theoretical reasons to infer differences between the various climates for the individual equipment changes, the pattern here suggests that the sample is unpredictable. Indeed, when this combination is subjected to an ANOVA test, the result is a model that explains only about 3% of the savings variance in this population.

House size differs between the system types, but the differences are not large, as shown in Table 10 and Figure 11. These results apply to single family homes. Other types of homes (manufactured homes) provided sample sizes that were too small for meaningful comparisons to be made.

Table 10. Square Footage by Equipment Type for C&RD Single Family

System Type	Mean Sqft	SD	n	90% CL
FAF w/CAC	1,785	628	116	96
FAF w/oCAC	2,227	720	167	92
Heat Pump	2,403	924	72	179
Zonal	1,848	598	115	92
All Cases	2,052	707	470	54

These areas are also influenced by the type of house retrofit. About 18% of the C&RD claims were for heat pumps in manufactured homes. While there is little evidence that these systems are appreciably different, the homes themselves are smaller across the sample.

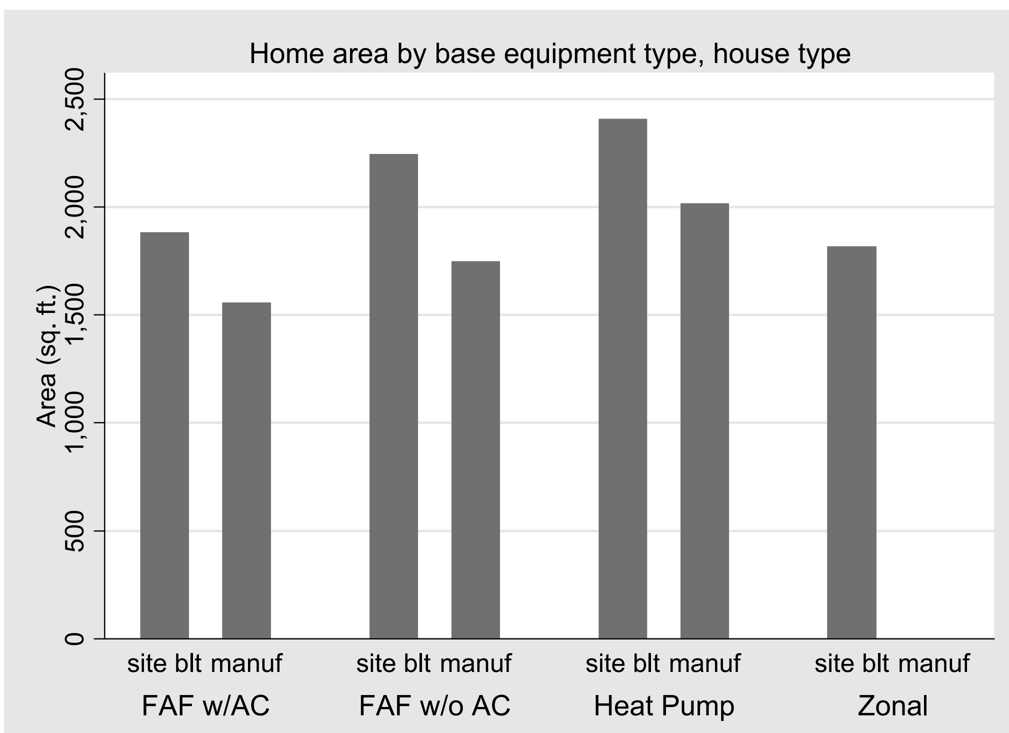

Figure 11. Building Size by System Type (N=470)

Figure 11 shows the difference between home sizes when area is expressed as a function of base system size. There were no cases in this sample where zonal electric heat occurred in a manufactured home. Overall, the manufactured homes were about 15 percent smaller than the site built homes. This variation persisted across most other segmentations of the sample.

There is also little difference in savings due to the age of the home. Savings by vintage bin are shown in Table 11 and Figure 12. These vintages correspond to the savings categories used in the C&RD calculation spreadsheet.

Table 11. Savings by Vintage, Single Family CR&D Cases

Vintage	Age Range	Mean NAC Saved	SD	n	90% CL
1	Pre 1981	4,597	5,937	394	491
2	1980-1994	4,079	5,744	160	744
3	Post 1994	3,939	6,702	54	1,495
All Cases		4,401	5,955	608	396

The savings by house age in the single family homes in this sample show a pattern of increasing savings with the age of the home. This is not surprising, although the size of this increase is small and the confidence intervals overlap. As shown in Figure 12, the savings follow a similar pattern when the vintage is compared to climate zone.

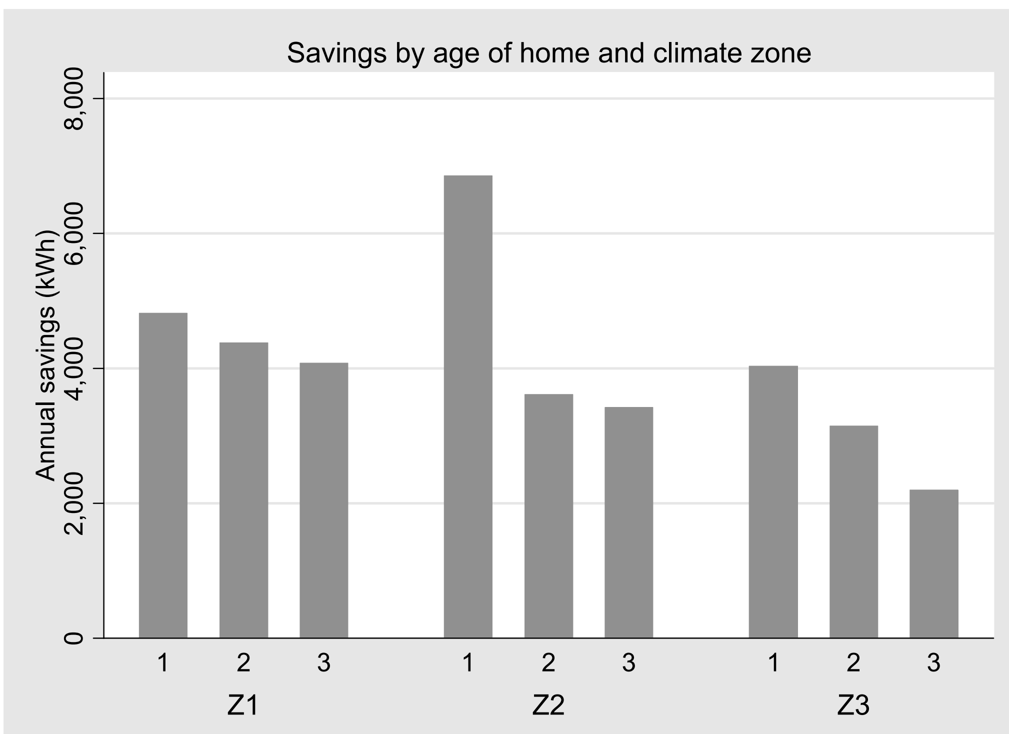

Figure 12. Savings by Vintage

3.2. Comparison (Control) Group

The process of identifying a control group presented some difficulties. First, the participants received treatment over a range of installation dates. It is not possible to identify a control group with pre/post billing data for exactly the same time period. Instead, we looked at the annual consumption over a three-year period for a group of untreated customers. Second, it was difficult to find a group of untreated customers. We utilized consumption records from investor owned utilities (IOUs) as the untreated group. Since the treated participants are all from public utilities, the groups are not perfectly matched.

The untreated group was verified using a phone survey to assure that they did have a heat pump and installation occurred before (not during) the study period. The consumption for this group is shown in Table 12 and Figure 13.

Table 12. Untreated Group, NAC kWh Consumption by Year

Variable	2001	2002	2003
NAC	20,160	20,467	19,932
Standard Deviation	8,712	8,096	8,036
N	342	342	342
90% C. L.	775	721	715
Annual Change	-	-307	535
Average Trend	-	-	114

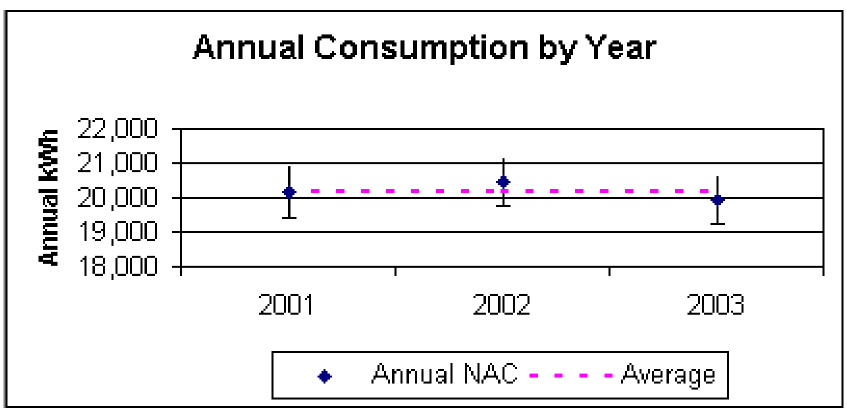

Figure 13. Untreated Group NAC by Year with 90% C. L.

There is no significant difference in consumption from year to year for the untreated group, although a slight downward trend in 2003 was noted. However, this is partly due to an artifact in the consumption records. Data obtained from the Energy Trust of Oregon included some consistent errors for specific months in 2003 that we were not able to correct. Review of these results by regional subgroup also finds no significant change in annual consumption.

The minor year-to-year changes that were seen fell within the error band, leading to a "null" result. The same null result was confirmed with an ANOVA test (see

Appendix A for details). However, we were requested to take any underlying trend in consumption, however small, into account. For that reason, we have decreased gross savings by 114 kWh as the "best estimate" correction for net savings. Thus, net savings are estimated as 4,263 minus 114 or 4,149 kWh per year. Inclusion of the trend adjustment term with its own uncertainty has an effect on the precision of the net savings estimate. We applied a pooled standard deviation to estimate confidence limits for the net estimate, as shown in Table 12. Since the size of the study groups is relative large, there was only a slight increase in the confidence limits.

3.3. Realization Rate

One goal of this study was to compare the savings verified through the billing analysis with those predicted under the C&RD program. Approximately 80% of the C&RD records (about 700 cases) included sufficient detail to identify the initial conditions before the installation of the heat pump. Savings from this group were subdivided into the main categories used in the C&RD spreadsheet. This resulted in 72 categories, including three climate zones, two house types (single family and manufactured homes), three vintages (pre 1980, 1981 to 1992, and post 1993) and four initial equipment types (forced air furnace (FAF) with and without central air conditioning (CAC), zonal electric, or an older heat pump).

Table 13: Single Family Realization Rate by Pre-Retrofit System Type

Equipment Type	N	Savings (NAC)	C&RD 2003	C&RD 2005	Realization Rate 2003	Realization Rate 2005
FAF w/AC	198	4705	7856	6055	59.8%	77.7%
FAF w/o AC	172	5159	7319	6337	70.5%	81.4%
Zonal	124	2614	3121	2199	83.8%	118.9%
Heat Pump	83	4648	982	1900	473.3%	244.6%
Totals	577	4383	5689	4713	77.0%	93.0%
Totals w/o HP	494	4338	6480	5186	66.9%	83.6%
Totals w/ HP as FAF	577	4383	6719	5477	65.2%	80.0%

Unfortunately, the variance on these estimates precluded using this complete set of divisions. Table 13 and Table 14 show comparisons for single family homes and manufactured homes, respectively. These are based on four system types, three vintages, and two house types as defined by the C&RD calculator. The predicted savings are derived from this calculator for each climate zone. The expected C&RD savings were then assigned to the individual cases based on their characteristics. In all cases, we used the C&RD value as generated from the prototype analysis used to calculate the deemed savings value. The realization rate calculation was based on the pre-existing system type and other characteristics of the home. This was done so that the amount of data in each cell would be statistically significant (usually at a more relaxed statistical criterion). When the sample is further subdivided, very few of the aggregate savings estimates are statistically useful.

Table 14: Manufactured Home Realization Rate by Pre-Retrofit System Type

Equipment Type	N	Savings (NAC)	C&RD 2003	C&RD 2005	Realization Rate 2003	Realization Rate 2005
FAF w/AC	34	3691	5202	5179	70.9%	71.2%
FAF w/o AC	78	4875	4227	4301	87.0%	113.3%
Heat Pump	10	3859	928	2186	415.8%	176.5%
Totals	122	4405	4229	4372	96.0%	99.2%
Totals w/o HP	112	4453	4524	4568	98.4%	97.4%
Total w/ HP as FAF	122	4405	4611	4646	95.5%	94.8%

The realization rates are calculated for the C&RD savings estimates that were used in the 2002-2003 program year (the year from which this sample is drawn). After 2003, the C&RD spreadsheet was revised extensively to account for better information on heat pump and duct performance in regional climates. This resulted in a reduction in savings estimates, particularly for conditions in which the PCTS installation guidelines for commissioning were not employed. Table 15 shows both the 2003 and 2005 realization rates for the entire sample. The 2005 program year estimates are based on these calculation revisions. These realization rates assume that the values from the calculator were used to claim savings. We do not have the actual savings claim in each case so the claimed values were inferred from the calculator for these tables.

Table 15: Realization Rate for Entire Sample by Pre-Retrofit System Type

Equipment Type	N	Savings (NAC)	C&RD 2003	C&RD 2005	Realization Rate 2003	Realization Rate 2005
FAF w/AC	232	4556	7467	5927	61.0%	76.9%
FAF w/o AC	250	5042	6354	5702	79.3%	88.4%
Zonal	124	2614	3121	2199	83.8%	118.9%
Heat Pump	93	4563	997	1931	457.7%	236.3%
Totals	699	4387	5435	4654	80.7%	94.2%
Totals w/o HP	606	4360	6119	5071	71.2%	86.0%
Total w/ HP as FAF	699	4387	6351	5332	69.1%	82.3%

As can be seen, there are significant differences between the predicted savings and the results of the billing analysis. This is most significant in the “Heat Pump Upgrade” category. The heat pump calculation in the C&RD spreadsheet assumes an upgrade from a working and properly installed heat pump that meets the current federal and state performance standards to a higher efficiency unit. It seems apparent that the base case in this sample is a heat pump that has ceased to function, and thus the savings are calculated from an installation that is operating as an electric furnace. While we have no direct indication of the nature of this base case heat pump, it is apparent that savings estimated here would need to be reduced substantially to accurately assess the realization rate for this measure. To compensate for these anomalies, we have summarized the total realization rate in three combinations:

1. A realization rate was calculated using the heat pump deemed savings from the C&RD spreadsheet. This has the effect of increasing the program-wide realization rate.

2. A program realization rate was calculated without considering the heat pump cases. This ignores the savings from the heat pumps on the assumption that the true nature of the base case systems could not be determined from the available information.

3. The heat pump was assigned to the Forced Air Furnace with Central Air Conditioning (FAF w/CAC) category. This has the effect of decreasing the realization rate, since the estimated savings for this measure are higher than the savings estimated for the heat pump cases.

This analysis suggests that the realization rate for the heat pump measures was about 70% when compared to the original (2003) deemed savings values and about 84% when compared to the revised 2005 deemed savings estimates.

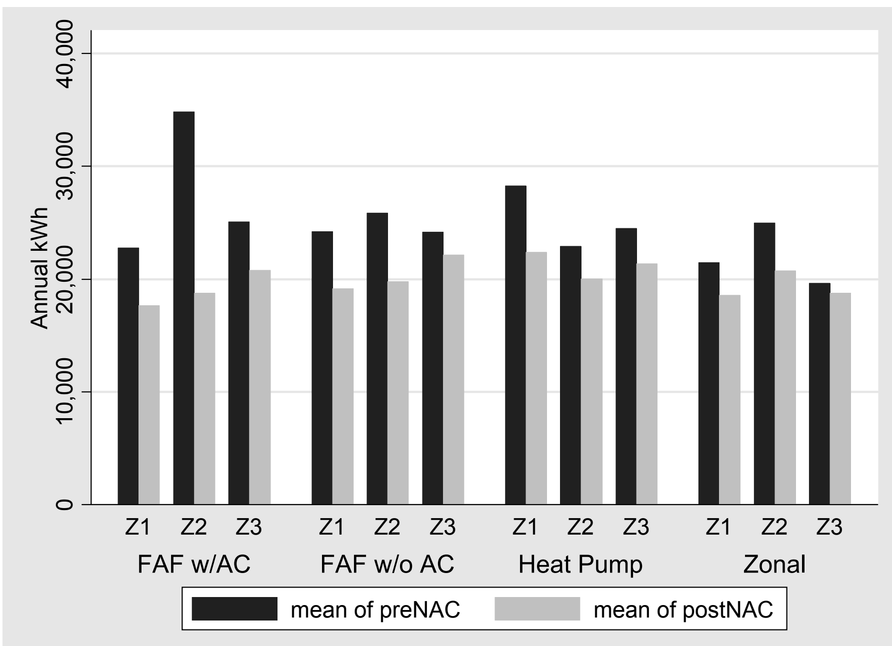

Figure 14. Pre- vs. Post-Installation Change in NAC

Appendix E includes realization calculations sub-divided by climate zone and vintage in addition to the rates discussed above. It is important to remember that the overall program realization rate will remain the same. However, given the variation in cell size and distribution, the detailed realization rates may not be statistically significant.

Figure 14 illustrates the variation between homes in different climate zones and with different base case heating systems. While the difference in consumption between the homes before the heat pump installation (preNAC) and the homes after the installation (postNAC) is apparent, it is also apparent that there are substantial differences between the data sets represented. When this data is subjected to an ANOVA test, the impact suggests that only a weak model could be inferred from the house type, house age, base case equipment and climate zone. Overall, the savings and consumption are not well explained by the categories of climate zone, house type, base equipment, and age of home. Much of the apparent variance here is the result of small sample sizes in many of these cells.

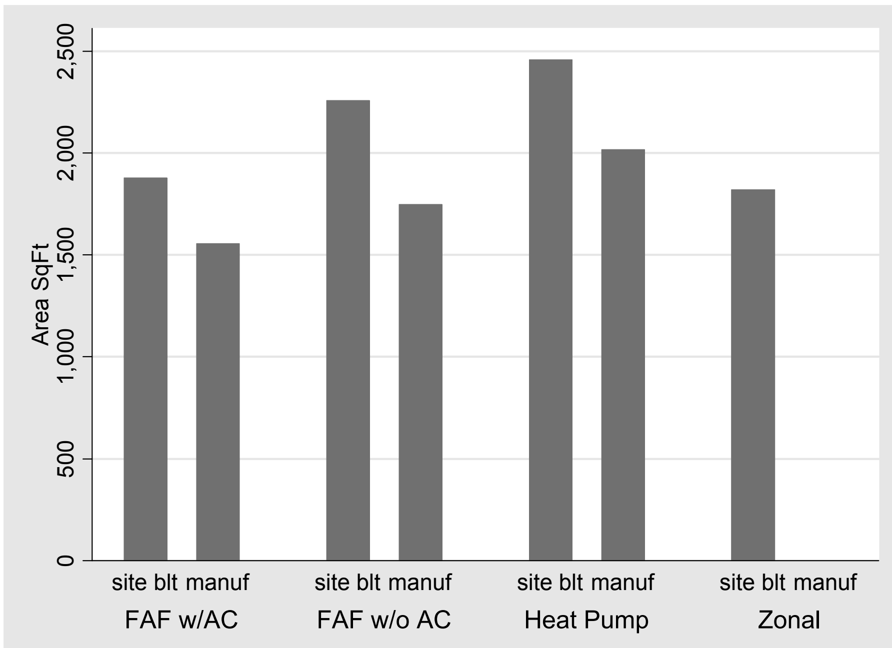

Figure 15. Square Footage by House Type and Pre-Existing System Type

Figure 15 explains part of the consistency in savings and use estimates. The size of the homes in this sample is fairly consistent across housing types and climate zones. There is a significant difference between the house area for manufactured homes and site built homes. This pattern can be seen in Figure 15. While it is important to note the differences between the housing types, it should be pointed out that the correlation between house size (of either house type) and the observed savings or the observed consumption is weak in this sample. Indeed, it is difficult to escape the conclusion that the determinants of savings for these heat pump installations are not particularly correlated to the major variables collected here after reviewing the data both for savings and for energy consumption. We suspect that this would be very different if some colder climates had been included. It does appear that there is much more consistency between the heat pump installations than there is between the homes that received these heat pumps.

It should be noted that there are several sources of bias in these calculations. In comparing billing analysis results to the C&RD calculator, the savings used assumed that the installation did not use the Performance-Tested Comfort System (PTCS) duct specification. For the most part, there is no information in the files indicating if this set of specifications was used. Therefore, in those cases where PTCS was actually employed, the estimated savings would be higher than the values used in Table 13 through Table 15. There is some indication this bias may affect about 25% of the cases.

A second source of bias could come from the weighting used in calculating the results from the C&RD calculator. The distribution of vintage assumes a building size associated with each vintage. In this sample, the average heated floor area in all the vintages is nearly identical and somewhat smaller than the size of the larger prototype used in the C&RD calculator for the "post 1993" vintage. This has the effect of comparing savings calculated for smaller prototypes in the 70% of the sample that is older to savings observed in houses that were actually somewhat larger. While the effect is partly offset by the newer homes, it could result in a small upward bias in the realization rate.

Finally, the method of estimating the NAC savings is based on the ability of the regression analysis to estimate the base consumption from the minimum bill. In most cases, this value was estimated from the base year. In cases where the base has no cooling (either because the climate is mild as it is in Northwest Washington, or because there is no cooling installed in the base case system, as in an electric furnace without central air conditioning) the estimate of the base case should be unbiased. However, in cases where there is a fully operating central air conditioner in the base case, the estimate of the minimum bill could include some cooling load.

This could create a bias in the base load estimate. If this bias is present, it would impact the base load itself and influence each month in the billing sample. This would have the effect of underestimating heating and cooling use in the base case and in the year with the new heat pump measure (since they both use the same base load calculation). The effect of this bias could be small, but it would always have the effect of reducing the estimated impact of changes in the heating and cooling load. Generally, this problem would be confined to climates with relatively large cooling loads (such as the Tri-Cities area). In those cases, the realization rate could be underestimated. This effect would partly offset the other effects identified above.

3.3.1. C&RD / ConAug Billing Analysis Findings

- The study population was smaller than expected due to our inability to obtain all the requested data. However, the study group of 1,022 cases is large enough for a relatively precise estimate of mean savings. The breakdown of estimates into subsets of the study population suffers from missing information and small sample sizes for sub-categories.

- Our best estimate of net savings is 4,149 kWh per year and is highly significant. The best estimate for the C&RD participants only is 4,240 kWh per year. This savings estimate is a function of the specific mix of system types, climate zones and housing types in the sample; it may not reflect future program results.

- Savings are approximately 70% of the predicted amount in the savings claim. There is ambiguity because the deemed savings estimates were not delivered as part of the billing records. The savings were compared to the deemed savings from the C&RD spreadsheet.

- Savings compared to the predicted C&RD deemed savings calculator revisions in 2003 had a 17% lower realization rate than the revised 2005 deemed savings. This reflects the impact of revisions in the calculator designed to more carefully account for heat pump performance in the Pacific Northwest climates.

- There is little difference in overall savings that can be attributed to the climate zones here. Indeed there is little of the actual variance in savings estimates that can be attributed to the category variables in this dataset. The ANOVA analysis suggests that these variables do not explain more than 3% of the variance in savings estimates. However, Cooling Zone 3 exhibits more cooling consumption and savings, as would be expected.

- There are significant differences according to the type of equipment that was replaced. The heat pump category appears to generally refer to the replacement of a non-functioning heat pump and should probably be considered equivalent to replacing an electric furnace. When this adjustment was made, the savings estimates were more consistent with the C&RD deemed savings calculator.

- As with any billing analysis, there is a substantial uncertainty introduced by the variability of the bills themselves. This variability, coupled with the small sample size, severely limits the capability to draw significant implications across the entire range of system types and climates.

- The inability to include utilities from more severe heating climates in eastern Oregon and northeastern Washington compromised our ability to observe differences in heating climates. This analysis cannot reliably be extended to C&RD installations in those climate zones.

3.4. EWEB CheckMe!® Billing Analysis

The purpose of the separate billing analysis of the EWEB program was to examine participants that received an adjustment of system airflow and/or refrigerant charge to determine if the adjustment alone produces energy savings. We examined two participant groups – those receiving an adjustment and those who were tested but found not to need any adjustment. The test-only group was expected to serve as a comparison control group. However, since we found equivalent savings in both groups, we added a second control group to verify our results. The second control group consisted of untreated customers.

This billing review focused on actual performance in the field. In addition, a laboratory review was conducted on the impact of refrigerant charge and airflow. The results of this testing are discussed in Section 6.0.

There is some ambiguity regarding the classification of participants. Contractors were not always careful to document which customers received an adjustment and which were “test-only”. EWEB has done its best to verify the classification and it is believed to be reasonably accurate.

We used the same methodology to weather-normalize annual consumption for this analysis as used for the C&RD analysis. As noted in Section 3.1, the weather normalized annual consumption (NAC) before the treatment establishes a baseline, which can then be compared to weather normalized consumption after the treatment. The difference in consumption determines gross savings. The formula used is the same as shown in Equation 1.

Gross savings are determined for the comparison group in the same way. The participant savings are corrected for any consumption change apparent in the comparison group. The result is net savings attributable to the program. This difference-of-differences approach is traditionally used in DSM evaluations to identify net savings due only to the treatment.

As with the C&RD analysis, we initially attempted to conduct the normalization using PRISM®. However, the same difficulties were inherent in the EWEB sample. Furthermore, we were concerned about having a model that would explicitly recognize both heating and cooling consumption. For that reason, we modeled the cases using the *EZ*Sim® tool. This tool applies a simplified engineering model that can be calibrated to monthly consumption records. Unlike a regression model, this model is based only on the building physics.

Other than the building area, we did not have detailed physical parameters for these cases. We applied typical residential parameters and then empirically adjusted operational parameters (i.e., setpoint) until the model fit the consumption records. This procedure provides an accurate estimate of the weather-normalized whole-building consumption, or NAC. However, estimates of the specific enduses within the NAC are less precise. In this case, lighting and appliance loads (baseloads) are set to match each specific case but are assumed to be the same average value throughout the year. Thus, “noise” in the consumption is assigned to heating and cooling. In fact, there will be behavioral changes in the baseload as well. Without detailed sub-metering, there is no way to quantify monthly variation in baseload enduses. The assumption of constant baseloads is consistent with assumptions inherent in regression analysis. Figure 16 shows an example application of the analysis tool to fit the model to the actual bills. In this case, the model fits to the bills with an R^2 of 95% and bias of 0. This high level of agreement is typical.

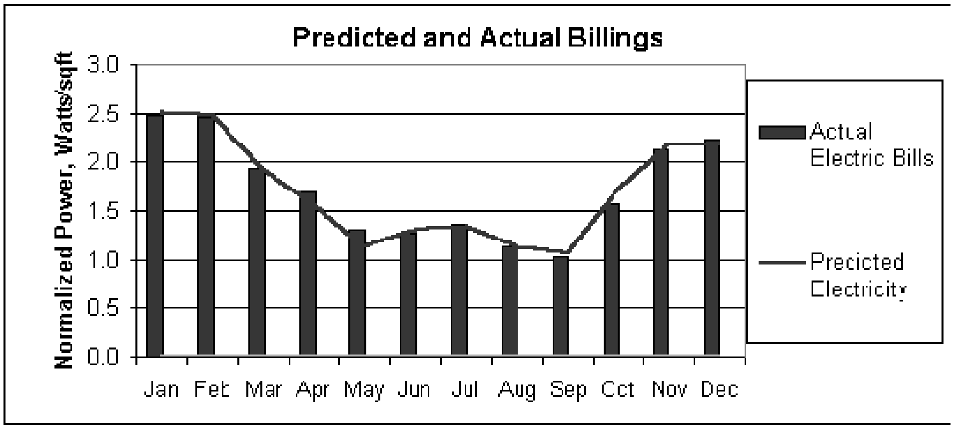

Figure 16. *EZ*Sim® Bill Fit Example

Figure 17 shows the enduses implicit in the example model, as computed for "normal' weather conditions. Figure 18 compares the modeled performance post-retrofit with the baseline model. In this case, the model fits the bills with an R^2 of 97% and bias of 0.

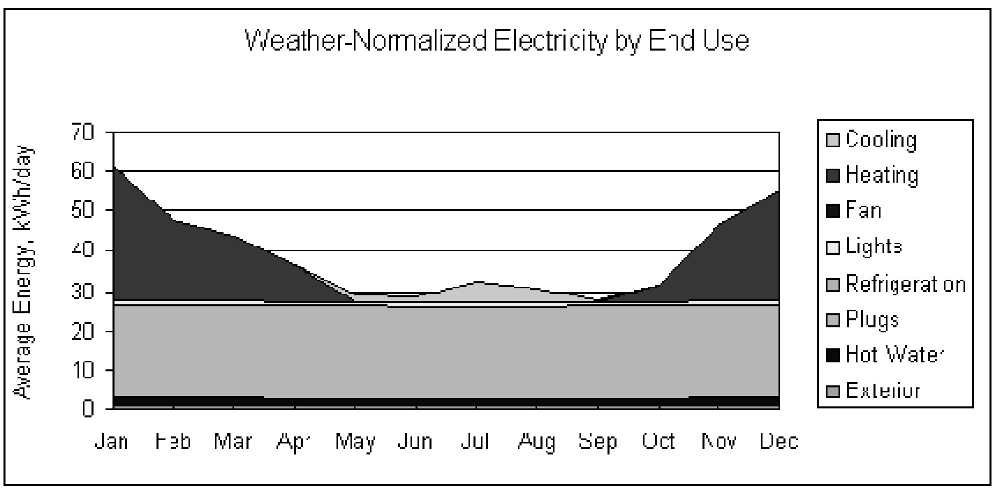

Figure 17. Enduses Within Example Case

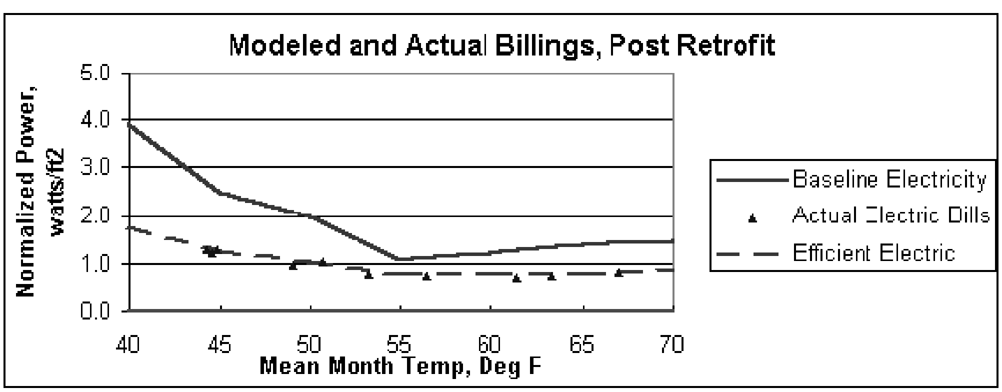

Figure 18. Example Post-Retrofit Match of Model to Consumption Records

As discussed, participants were divided into two primary groups – those that received an airflow and/or refrigerant adjustment and those that were only tested without any adjustment. There were also two smaller groups that received other repairs at the same time. These smaller groups were also analyzed but they are too small for the sample size to be meaningful. They are included in the "All Participants" group. Analysis results for all participant groups are reported in Table 16.

Table 16. Analysis Results for Participant Groups

Service type	NAC Savings	Std.Dev.	N	90% CL
Airflow and Refrigerant Adjustment	310*	3,410	100	677
Test Only	508	2,585	183	379
Adjust and Repair	553*	2,757	20	1,223
Repair Only	265*	2,582	19	1,175
All Participants	446	2,875	322	263

*Not statistically significant

Although savings are small, the "All Participant" group has a sufficient sample size to demonstrate that the savings are statistically significant. The savings appear to be associated with space heating rather than cooling, showing that a positive impact occurs for heat pumps. Although the "Refrigerant Adjustment" group shows savings of similar magnitude, the variance is much higher for this group. As a result, the savings lack significance, as shown in Figure 19.

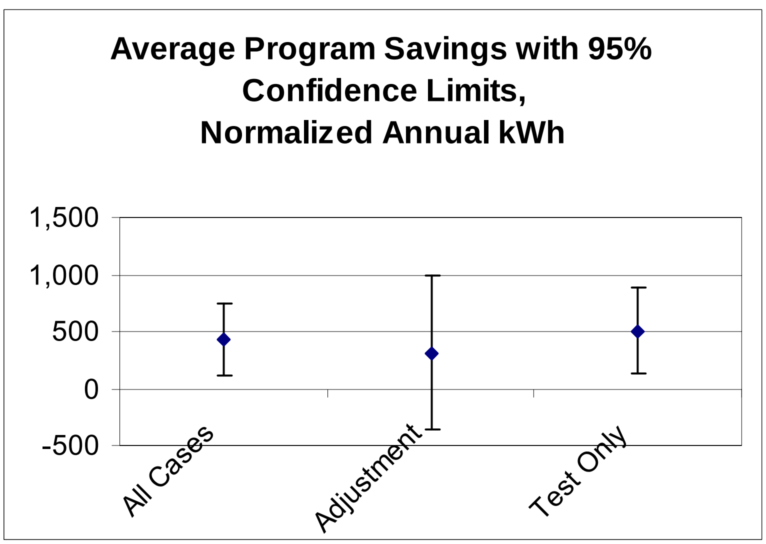

Figure 19. Savings for Participant Groups

The fact that the average savings are close suggests that, while there is a treatment impact associated with the technician's site visit, there is no additional impact due to the airflow or refrigerant adjustment. This hypothesis was confirmed by the fact that a difference-of-means test shows no significant difference in savings between the two major groups (see Appendix B for details). While the classification of "test-only" participants may include some errors due to faulty documentation, it is important to note that no such classification errors are expected for the group receiving an adjustment. Thus, we believe the aggregated group of all participants provides the best estimate of program savings.

Figure 20 shows an operations profile that plots energy consumption against the average monthly temperature. Note the small difference between pre- and post-retrofit. The pre- and post-operations profile lines are almost identical. The savings are small but statistically significant at the 95% level, as indicated in Figure 19 and Table 17. Figure 20 also demonstrates that the analysis model does identify some cooling energy consumption associated with higher temperatures.

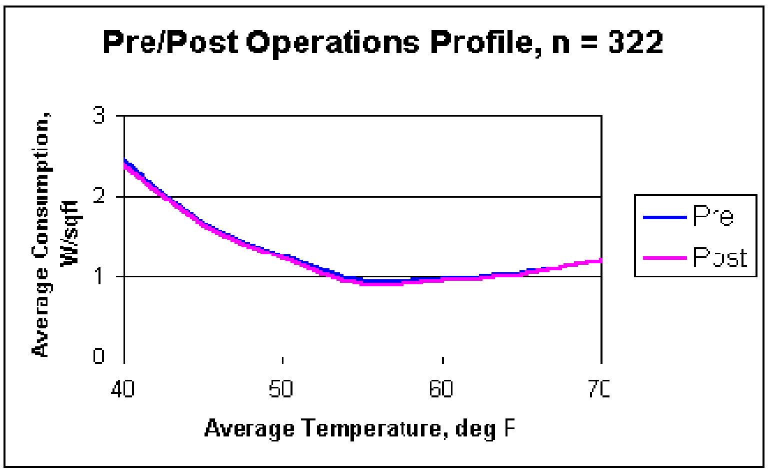

Figure 20. Pre/Post Operations Profile for Participants

The fact that there is no additional savings due to airflow and/or refrigerant adjustment is consistent with the results of laboratory testing discussed in Section 6. Apparently, the explanation for the fact that "test-only" participants demonstrate small but significant savings is related to the fact that, in order to conduct the testing, the contractor is required to restore the unit to operational order. This may include repair of electrical components or the compressor, coil cleaning, or other service necessary for the unit to operate. Evidently, these actions improve the performance of at least some of the units.

This is supported by the cumulative distribution profiles seen in Figure 21, which show that the savings occur in a few high consumption participants in the upper tail. This suggests that the program benefits occur as a few badly-performing systems are restored to proper operation as a result of the site visit. The cases in Figure 21 were not noted to have received any CheckMe!® repairs; they were only tested.

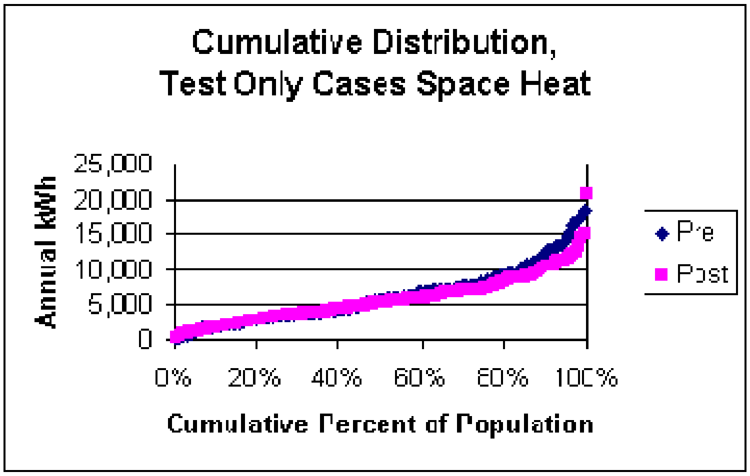

Figure 21. Distribution Profile of Space Heat Consumption

3.5. EWEB Control Comparison Group

The finding that the two participant groups show similar savings was a surprise. The original expectation was that the "test-only" group would provide a control group. However, since we found equivalent savings in both groups, we added another group of untreated customers to serve as the comparison group. EWEB staff provided a list of customers that had participated in various conservation programs other than CheckMe!®. We screened out any that participated in another program during the study period as well as those with occupant changes or otherwise incomplete data. Of the remaining customers, time permitted analysis of 80 customers.

It should be noted that this comparison group used about 25% less total electricity than the participant group. While not ideal, this was considered acceptable since the point of this part of the analysis was to see if there were underlying changes in space conditioning loads in the EWEB service territory in the period used for this study.

The participants received treatment over a range of installation dates. Thus, it is not possible to identify a control group with pre/post consumption for exactly the same time period. Instead, we looked at the annual consumption over a three-year period for a group of untreated customers. As shown in Table 17 and Figure 22, the year-to-year differences are not statistically significant.

Table 17. EWEB Comparison Group Annual Consumption

	2001	2002	2003
Mean NAC	13,084	13,629	12,913
SD	5,330	5,231	5,415
n	80	80	80
90% C.L.	981	963	996
Annual Change		-545	716
Average Change			86

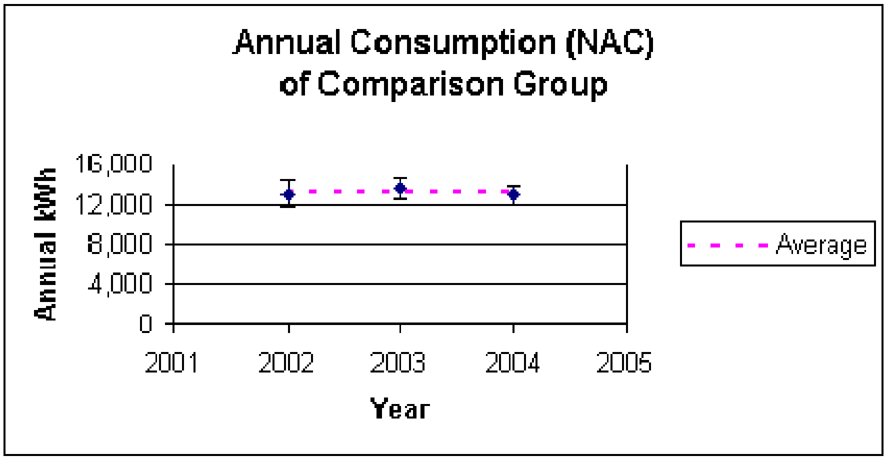

Figure 22. Comparison Group Consumption

The same result was confirmed with an ANOVA test (see Appendix B for details). However, we are dealing with a very small amount of savings and any underlying trend in consumption must be taken into account. For that reason, we decreased gross savings by 86 kWh as the best estimate correction for net savings. Thus, net savings are estimated as 446 minus 86, or 360 kWh per year.

3.5.1. EWEB Analysis Findings

- EWEB's CheckMe!® program provides an effective average savings of 360 annual kWh. These savings are small but statistically significant.

- Savings due to system airflow and/or refrigerant adjustment (normal CheckMe!® repairs) cannot be disaggregated from savings due to testing the unit (and perhaps performing other repairs to get the heat pump running).

4. Market Actor Interviews

Ecotope conducted a series of interviews with installers, manufacturer's representatives, and other stakeholders to elicit data on equipment selection, sizing strategies, installation techniques and other issues that impact heat pump performance. The goal was to understand the common practices used by these professionals and the reasons behind their decision-making, particularly regarding heat pump sizing, system set up and control strategies.

4.1. HVAC Installers

All of the heat pump programs in the Pacific Northwest depend on the cooperation of HVAC tradesmen. Interacting with this group is challenging, given the differences in motivations and day-to-day pressures for the contractors versus energy efficiency experts. However, in one sense, the goal of the installer is relatively simple: sell product to a compliant consumer. Entities such as utility personnel and policymakers must remember this basic motivation when designing and refining installation protocols and related incentives.

Well before this project got underway, information about installers' experiences, challenges, and expectations had been gathered anecdotally. This information was very useful in shaping the technical specifications of various regional heat pump installation programs. An updated survey of installers was proposed along with the fieldwork and billing analysis so that ongoing efforts to ensure high quality heat pump installations could be integrated into current practice by the trade as much as possible.

A total of 32 independent HVAC installation companies throughout the region were contacted. Full interviews were completed for 29 shops. (In the other 3 cases, the interview could not be completed for various reasons.) Generally, the person interviewed was the shop owner or general manager. In larger shops, the service manager was interviewed. In many cases, the interviewee consulted with others in the office to complete the interview. Questions covered sales information, equipment questions, and tax credits/incentives, as well as general impressions of where the industry is headed.

Geographically, interviews were conducted region-wide, not just in the areas included in the billing analysis and field work. The bulk of the interviews were carried out in the Portland and Upper Willamette Valley areas, the Olympic Peninsula, Tri-Cities, Spokane, and central/southern Oregon. Additional sites were in the mid-Columbia, Boise, and Seattle areas.

Shop size is typically measured in number of trucks. In this sample, shops ranged in size from 2 to 30 trucks (which included both service and installation functions), with a median shop size of 6 trucks. Shops were asked what percentage of their business (by gross revenue) was new construction. The median response was 50%, but the distribution was skewed downward (meaning more shops were involved in the retrofit/replacement market than in new construction). This indicates that the majority of shops interviewed are generally not competing in the aggressive, lower-profit margin arena of new construction). Most shops were familiar with the various credits/incentives available for heat pumps and ducts, but not all had submitted jobs for credits or incentives.

Installers were asked about the biggest obstacles preventing them from installing equipment in what they consider the best (most efficient) style. The two most common responses were competing with low bidders (who may over-promise) and finding and keeping good installation crews. These are common laments in the industry.

A few additional descriptive data are of interest. Installers were asked what percentage of their service technicians are NATE-certified. NATE (North American Technician Excellence) is now the standard industry means of certifying technician competence, with tests on several types of equipment and trade topics. The mean level of NATE-certification in this group is 50%, but there were a number of shops with no NATE certification and a few with 100% of their technicians certified. The businesses represented install all major brands of equipment (Carrier, Trane, Lennox) and some also offer other product lines such as Amana (Goodman), Rheem, and Coleman.

4.1.1. Installation Practices: TXVs

One of the major items of interest in this study had to do with the availability of thermostatic expansion valves (TXVs) on the outdoor unit. The use of this technology was thought to be a technologically effective way to ensure unit capacity even if system charge was not optimal (and therefore could be used as a stand-in for a full charge check during installation). The availability of the factory-installed outdoor TXV is determined largely by manufacturer choice. Field-installed TXVs are also an option but are not recommended by one major manufacturer (Carrier). This issue is explained in more detail below.

Regional technical specifications provide guidance on how heat pump systems should be sized; installers are directed to perform heat loss and heat gain calculations as part of selecting a unit. Almost all interviewees indicated they used standardized methods to size their systems. A combination of pencil-and-paper and computer sizing methods were reported. Interviewees were also asked what load tended to dominate in their service territory, and what they do when heating and cooling loads differed by more than a ton (12,000 Btu/hr) of capacity. Figure 23 shows heating loads were given the slight edge in determining unit size unless the calculations differ. In that case, cooling tended to dominate.

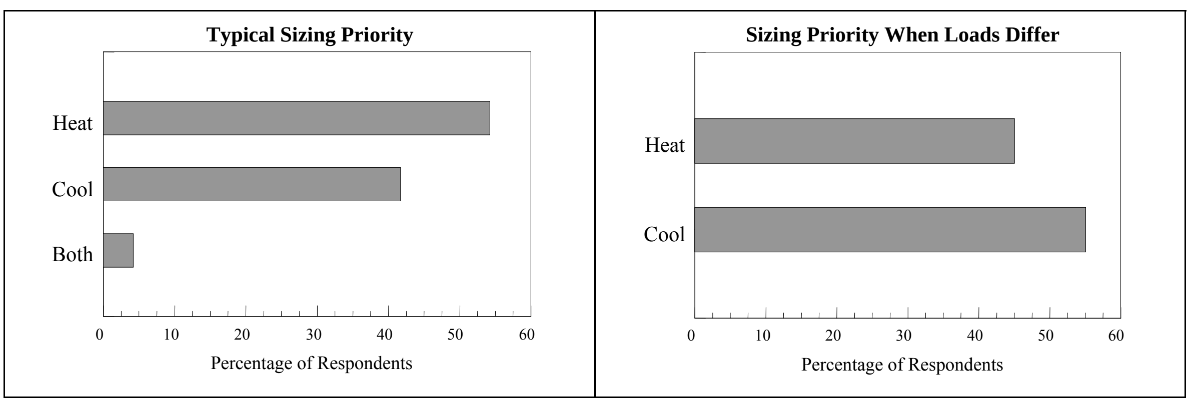

Figure 23. Heat Pump Sizing

Interestingly, this is not what we found when we used the field review results to calculate heat pump capacity. Based on the individual housing characteristics, it appears that much more conservative sizing is employed than these interviews suggest. Using a balance point sizing method, our estimates indicate that the average system size was 30% smaller than would be expected, even without including a safety margin or duct losses. When combined with questions regarding first cost versus operating costs (see Figure 32), it appears that HVAC contractors are installing smaller systems (by up to a ton of capacity) in order to reduce first costs at the expense of increased operating costs due to more extensive use of electric back up heat.

Installers were also asked about duct system design and features. Most reported using more than “rules of thumb” for sizing duct systems; a majority said they preferred their systems to deliver 400 CFM/ton, which is the manufacturers’ most commonly specified system flow rate. Interestingly, the actual average flow rate measured in the study for base case homes was about 325 CFM/ton. Surprisingly, 85% of respondents said they “sometimes” or “always” installed air volume dampers in their duct systems so that airflow could be adjusted to meet the specific heating and cooling needs of rooms. This is relatively high percentage, but it may reflect an increased emphasis on comfort amongst this group of installers.

Respondents were also asked about the efficiency characteristics of their heat pumps. Specifically, they were asked to estimate the average HSPF of their most efficient line (which should be the unit most likely to receive incentives or credits). Figure 24 shows the results. The graphic shows that the clear majority of installers do not have a large number of lines with the highest efficiency heat pumps (i.e., HSPF significantly above 8.0). This should change as the federal minimum efficiency standard increases to SEER 13 in January, 2006, but the overall impact will not be known for some time since the new minimum SEER will provide only an indirect indication of how HSPF will be affected.

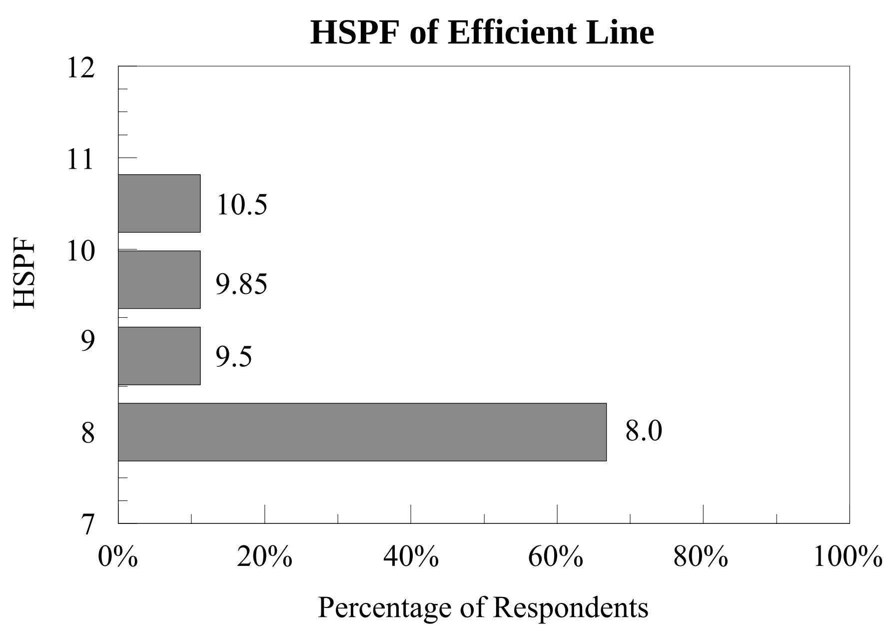

Figure 24. Estimated Heating Seasonal Performance Factor (HSPF)

There has been considerable interest in the impact of using a TXV on the outdoor coil during heating mode. Research has suggested that the TXV, when installed on the indoor unit, maintains capacity and efficiency better during cooling operation than a fixed metering device. Installers were asked to estimate the percentage of their heat pump product lines that offer the outdoor unit TXV as a standard item; they were also asked the cost of field-installing a TXV on the outdoor unit. Results are shown in Figure 25 and Figure 26.

About half of the units installed have an outdoor TXV; it is a common feature of certain product lines (Trane, Lennox) and very uncommon on others (most notably, Carrier). Since these questions were asked, the Purdue bench testing has shed considerable doubt on whether use of an outdoor unit TXV is as important as originally expected. The cost of a field-installed TXV ranges from $100 to $400, with an average of about $310. Installers were asked about their general acceptance of TXV technology. About 80% gave a positive report (see Figure 27).

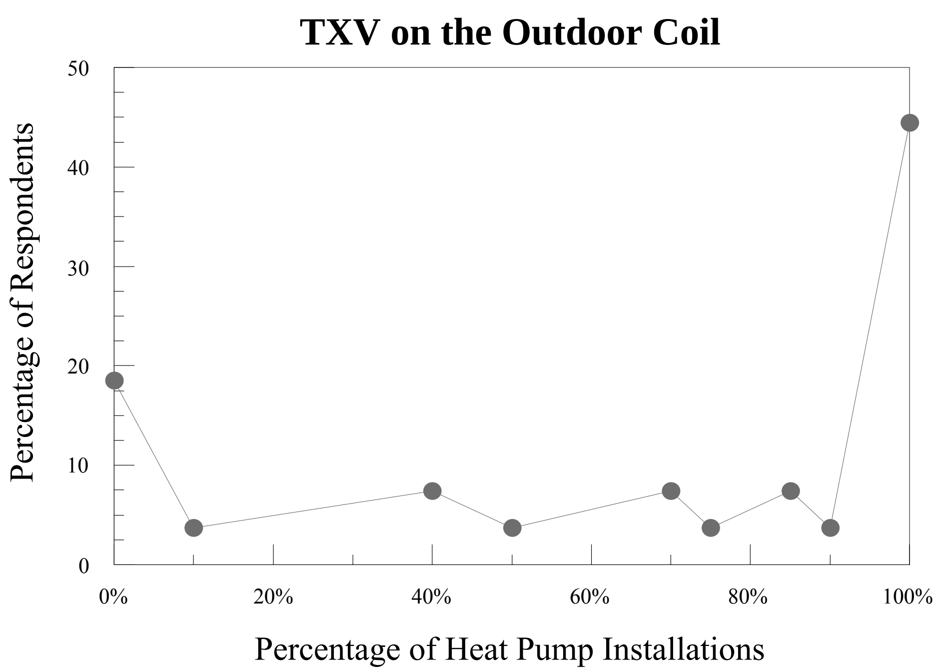

Figure 25. Factory-Installed TXVs (on Outdoor Units)

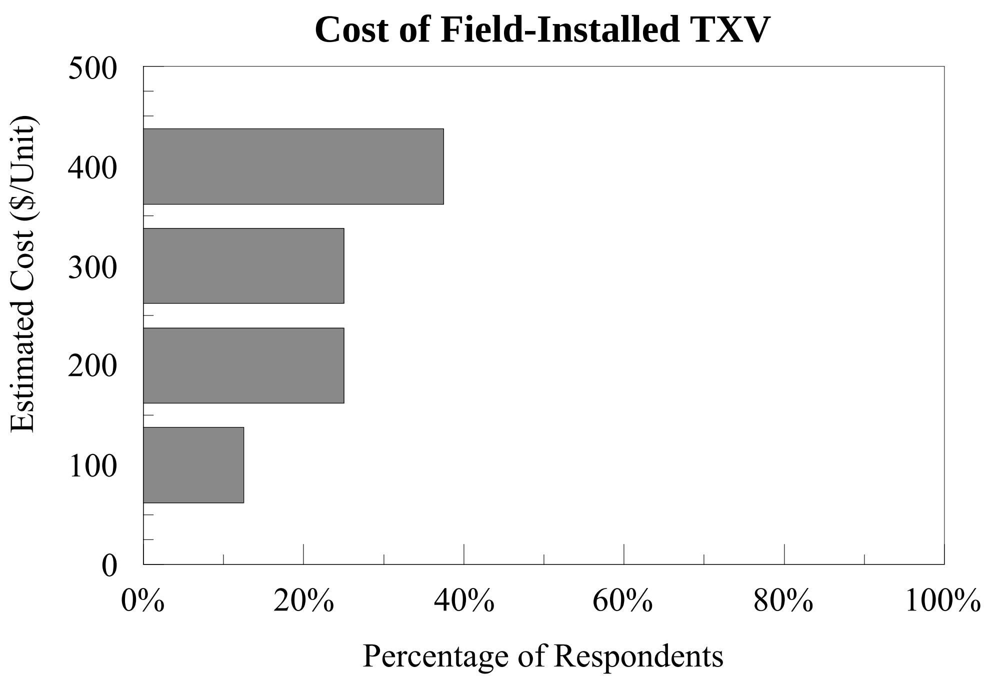

Figure 26. TXV Cost

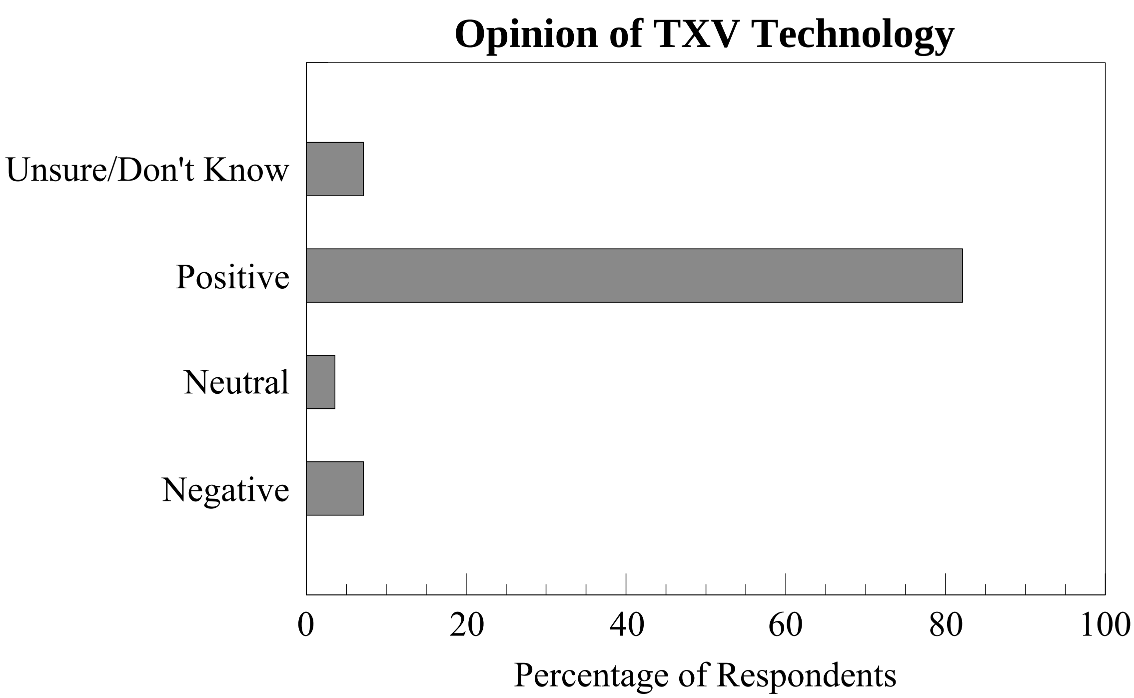

Figure 27. TXV Acceptance

4.1.2. Installation Practices: Back-Up Heat Control

Of central importance to heating energy usage is the interplay of compressor-produced heat and auxiliary (electric resistance) heat. Installers were asked a number of questions about these issues.

1. They were asked if they enable the low ambient cut out, which turns off the compressor below a certain ambient temperature.

2. They were also asked if they installed an outdoor thermostat, which disables auxiliary heat unless it is below the outdoor thermostat setpoint. If they did install an outdoor thermostat, they were asked the setpoint temperature.

3. They were asked if they ever installed auxiliary heat so that at least one resistance element always came on when there was any call for heat at the thermostat.

Only about 35% of respondents said they installed an outdoor thermostat. This corresponded (coincidentally) with the percentage of systems in the field study that had an outdoor thermostat. The highest concentration was in the Clark County PUD service territory, which began requiring outdoor thermostats in 2001 for all systems financed through utility loans; this was the database we used to identify field study sites. The average setting was 40°*F*, which should assure delivery temperatures would not dip below about 90°*F* if the system were sized properly and ducts were handled well.

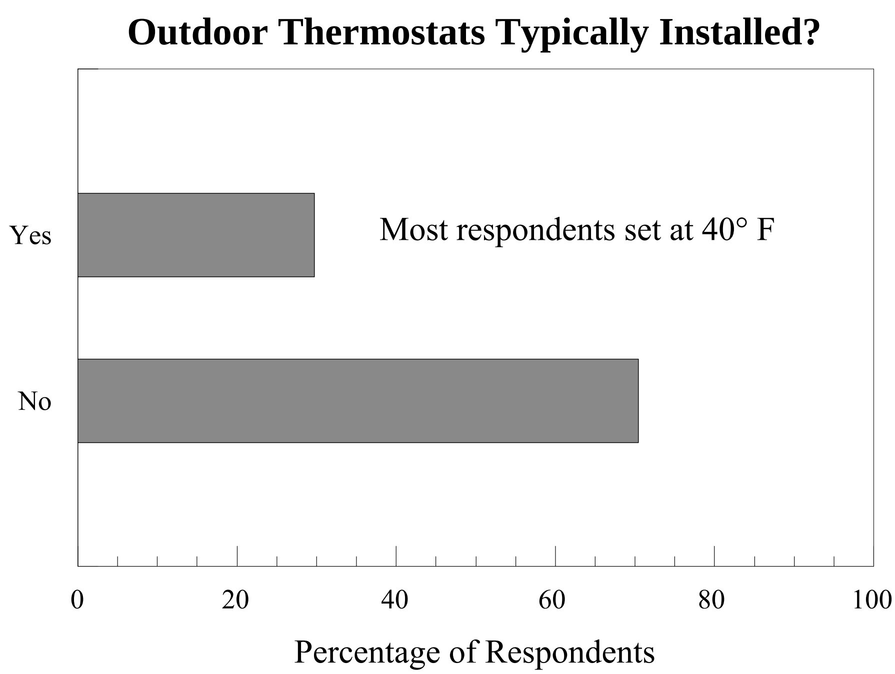

Figure 28. Outdoor Thermostat Penetration

A relatively high percentage of respondents said they enabled the low ambient cutout feature at least sometimes, as shown in Figure 29. The field study did not find a case that had the low ambient cutout enabled. However, the respondents that reported doing this were in areas not included in the field study. Low ambient cut out seems to be used most commonly in the Spokane area, which may be a product of utility-sponsored programs over the last 15 years to convert gas furnace customers to heat pumps. Because the temperature of the air delivered from the ducts is much cooler in a heat pump system than in a furnace, converting this direction could be expected to lead to more homeowner complaints at lower outdoor temperatures. The other respondents who reported using the low ambient cut-out feature were in the TriCities and mid-Columbia areas.

A related installation feature which has been of some concern is the practice of wiring a resistance element so that it operates whenever the thermostat calls for heat (rather than being activated by a Stage 2 heat call, which is the usual way). Previous field work conducted in various parts of the region has suggested this practice is sometimes used. In this study, seven respondents stated they did this at least occasionally. All of these respondents were from eastern Washington or Idaho, so the practice is apparently a comfort measure. In the field study, no sites were found that used this practice.

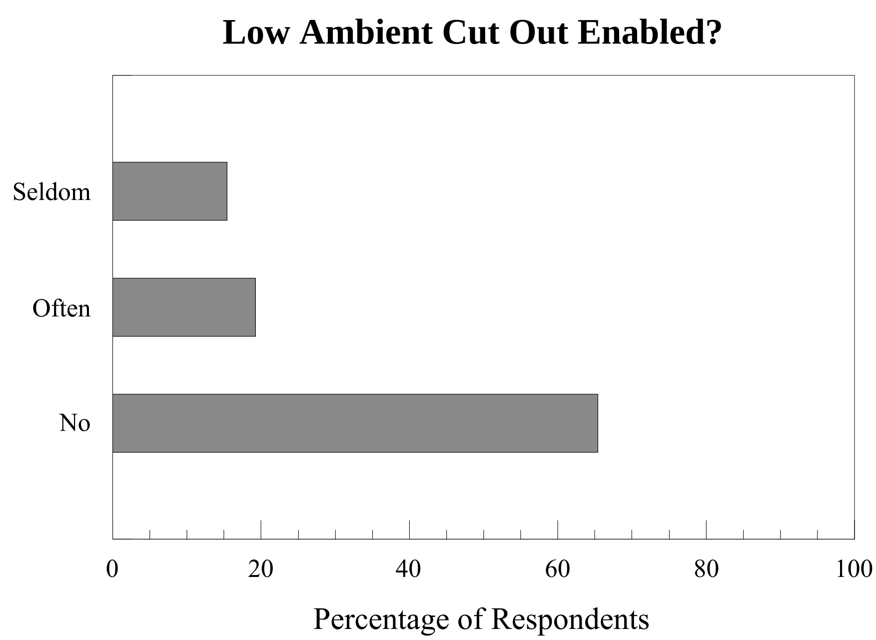

Figure 29. Low Ambient Cut Out

The majority of the installers interviewed indicated that they did not install wiring for auxiliary heat that would automatically turn on the electrical element when a call for heat was received. However, about 5% of the sample indicated that they always did so (see Figure 30). The perception amongst that group is that customers are more comfortable with the warmer air than would be produced by the compressor alone.

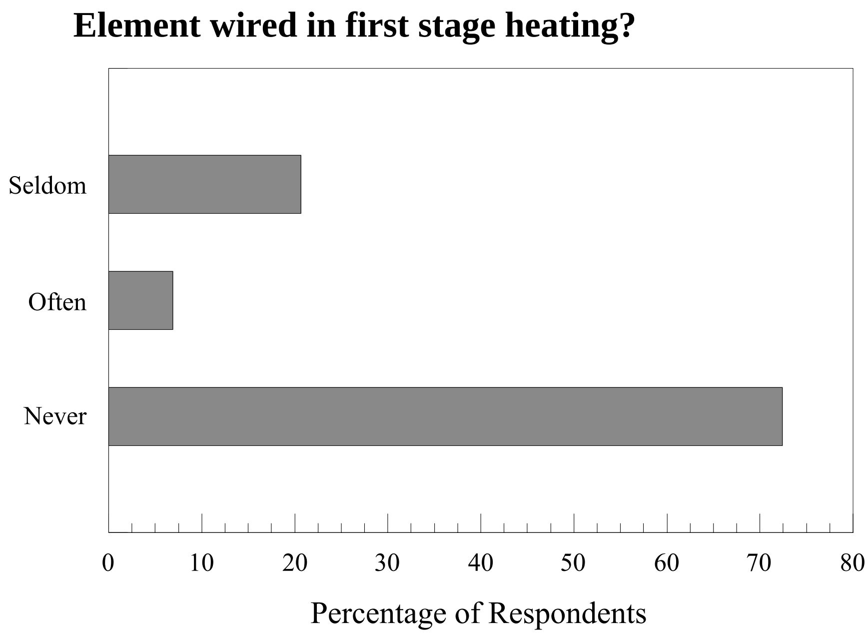

Figure 30. First Stage Heating

To summarize, there is substantial potential to increase existing heat pump efficiency by adding outdoor thermostats to systems. Savings have been estimated at 5 to 20 percent of annual heating energy, depending on climate and control strategy. This measure should not be applied without a review of the house heat loss rate. In some cases, depending on the system tonnage and site heat loss rate, it might not be advisable to set the outdoor thermostat aggressively (to, say, 35°*F*). In general, though, this measure should receive more attention in both existing and new heat pump systems.

Installers were also asked about their customers' opinion of the indoor thermostat. About 70% of respondents said their customers complain that the indoor thermostat is too hard to program. The technology has started to change, however, and the introduction of touch-screen heat pump thermostats is viewed by several installers as a positive evolutionary step. More time is needed to assess the overall effectiveness of these newer thermostats, but the consumer reaction has so far been mostly positive.

A topic of great interest in this survey and the overall study is R410A refrigerant, which is gradually being introduced as part of the eventual complete phase-out of R22. Opinions on R410A vary greatly from shop to shop (see Figure 31). Those shops not doing many R410A installations were usually less positive and expressed doubts about whether R410A was even necessary. A few shops even said they thought the phase-in was some sort of conspiracy that had nothing to do with protecting the ozone layer or improving heat transfer efficiency and homeowner comfort.

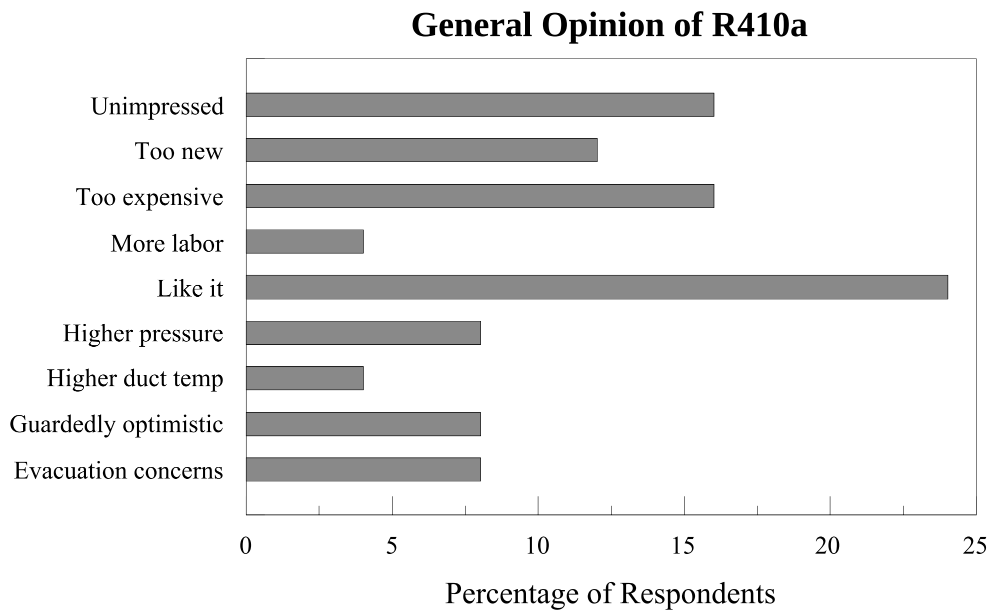

Figure 31. R-410a Refrigerant

It is true that more care must be used with evacuating lines and charging with R410A since it uses POE oil, which is much more hydrophilic than the mineral oil used with R-22. The early installation guidelines for R410A were not as specific and direct as they later became.

More attention should be paid to R410A system performance and installation quality to ensure these systems are performing as advertised. Specifically, it would be beneficial to engage installers and manufacturer's representatives so that installation standards and protocols keep step with new R410A data.

4.1.3. Incentives and Tax Credits

Installers were asked several questions about the impact of state or regional tax credits and incentives on their business. Familiarity with credits and incentives was mixed, but about 60% of those interviewed considered themselves familiar with PTCS specifications (which cover both heat pumps and ducts). Almost all respondents (90%) were at least "somewhat" familiar with Energy Star® heat pump efficiency standards. About 80% of respondents (both in Oregon and overall) said incentives or tax credits had improved their bottom line. More than 50% of respondents considered explaining incentive and tax credit procedures to their customers "not difficult," but 35% considered it "too complicated." Only about 20% of Oregon installers said the Energy Trust's $200 incentive for heat pumps with HSPF of 8.1 or better had improved their ability to sell heat pumps not meeting the ODOE minimum requirement of HSPF 8.5.

Installers were also asked about their use of third-party refrigerant charge checking programs (such as CheckMe! ®, ACRx, and eenalysis), which are required to gain PTCS certification and Oregon tax credits. Of those installers using these programs, about half said they thought the improved record keeping was a plus, as was the ability to use the third-party verification to upsell to higher efficiency units. On the down side, about 60% of contractors who used one of these programs said it was "hard to sell" overall, meaning in many cases the cost of using the service added too much to the price tag of a service visit (if not part of a new installation).

4.1.4. Customer Concerns

The final questions on the survey asked the installers for their assessment of major trends affecting consumer behavior and a "crystal ball" prediction of future influences on buying habits. Figure 32 and Figure 33 summarize the results. Installers report the most influential factor in buying a heat pump today is its initial cost. More than 40% of the sample indicated they expect consumer interest in heat pumps to improve as fuel costs continue to rise. About 25% of the sample thought better technologies would fuel increased consumer interest.

The cost of energy is listed as the next most influential factor. This makes good sense in the context of a higher minimum standard for heat pumps (which will equalize cost of an "entry" model for all consumers); those consumers opting for a more efficient heat pump should have lower overall energy usage and operating expenses.

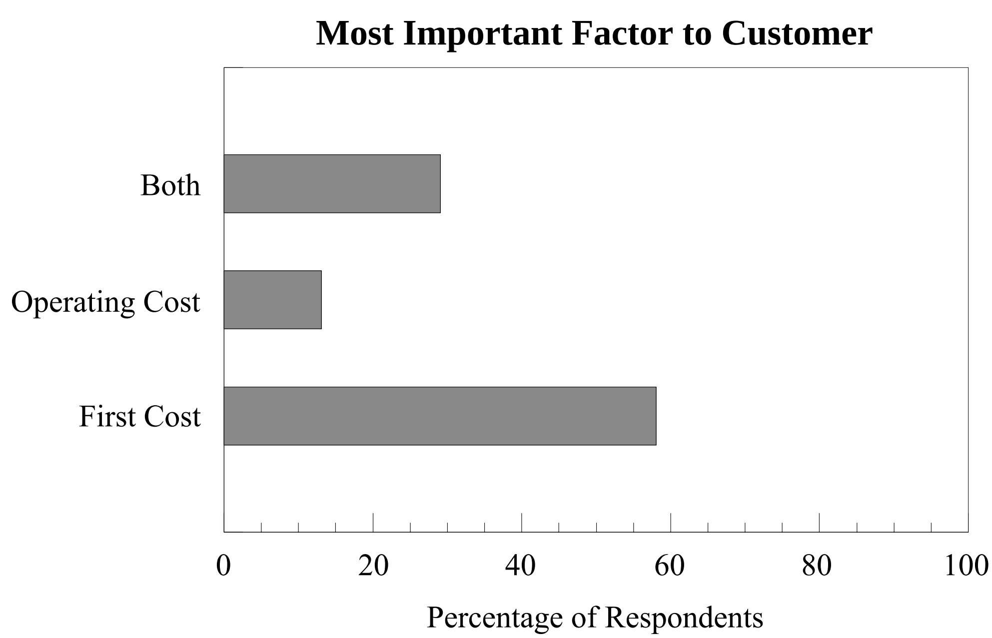

Figure 32. Customer Priorities

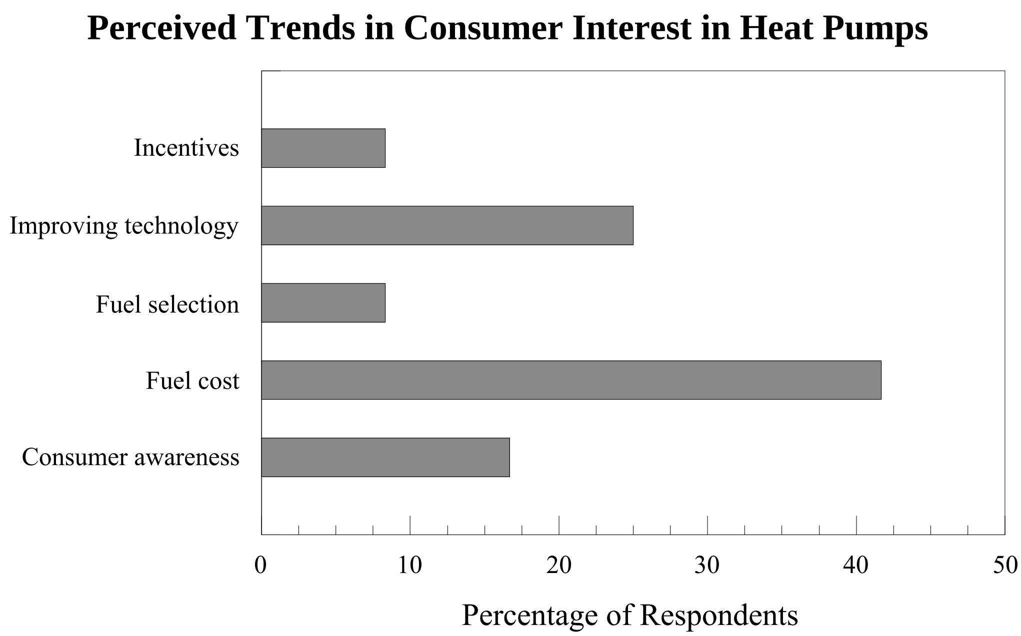

Figure 33. Influences on Consumer Buying Habits

> Installers also report there is still confusion about how heat pumps work and the pros and cons of a heat pump versus other air conditioning systems. In some cases, customer inquiries about the technology have resulted in the purchase of a heat pump even though the customer was initially very skeptical. This trend may continue, at least in the near term, given the relative costs of electricity and natural gas.

4.2. Trends in Heat Pump Manufacturing

In an effort to gather information about current hardware, industry trends (and rumors) and issues facing suppliers, manufacturer's representatives were interviewed. One representative from each of the major brands (Carrier, Trane, Lennox) and two reps from smaller-market brands were interviewed. These reps, for the most part, have daily contact with installation companies and have considerable field experience investigating complaints. Because of the small sample size for this set of interviews, the summary results are not as useful as are the general impressions gleaned from the installer interviews.

Information was also gathered at several national and regional HVAC conferences and workshops, and by reviewing popular trade journals. Many of the same themes of concern to regional reps were stressed. This is certainly a reflection of a particular view being passed downward; however, it is becoming increasingly apparent that emerging international factors are more of an influence on the industry than in years past.

A number of “warm-up” questions were asked to determine the degree of contact of the manufacturer’s reps with the installers, product line features, and regional efficiency programs. All but one regional representative interviewed said they commonly assist installers in performing heat loss/gain calculations and sizing heat pumps and ducts. Respondents were also asked about programs and technologies intended to promote efficiency. All representatives had at least some familiarity with state and utility incentives/credits (although they didn’t know, for the most part, the specific incentives or credit levels) and with services such as CheckMe®!. In general, the manufacturers believed these incentives and services were beneficial and noted installer interest had been increasingly expressed. The Oregon tax credits had the most positive effect on sales (versus the Energy Trust incentive, which is smaller but which applies to units with lower HSPF ratings).

Most reps said they would recommend their installer upsell to a system with a TXV on the outdoor coil if it was a standard feature or common option. Manufacturer reps were supportive of outdoor thermostat installation but expressed concern about the possibility of “cold blow”. Even though they typically said they thought installers were doing an adequate job of sizing systems, apparently they are still nervous about potential customer comfort complaints.

Another area of concern was the “new” refrigerant, R410A. Manufacturer reps accept it as the future (even though it has been available for about eight years), but they report there is still significant resistance to its use amongst their installers, mostly because of its higher operating pressures and first cost. This is more of an issue for the installers who deal in lower-efficiency lines (almost all of which use R-22 refrigerant). Reps report many questions from their installers on R410A and expect it will take some time for it to be widely accepted.

The 2006 change in minimum SEER is a topic of great discussion and concern, both amongst regional representatives and factory representatives higher up the chain. All distributors are concerned about the physical size of units that meet SEER 13 and up. That is, both the indoor and outdoor units are bigger than previous units (since coil surface area is a determinant of efficiency, along with compressor design, indoor/outdoor fan type, and other factors) and therefore require more materials to build and more space both in the warehouse and at the installation site.

Raw materials (copper, steel, and aluminum, most notably) are becoming more expensive as world market demand increases; this means, quite simply, that heat pumps will become more expensive. Bigger units built with more expensive materials translate into higher prices to consumers, and this is a substantial concern to many industry personnel.

The irony of the efficiency bar being raised to SEER 13 is that the energy efficiency marketing edge is dulled, since the minimum is now so much higher than it was only a few years ago. Only a fraction of consumers will spend even more to get units that have a higher efficiency than SEER 13; however, manufacturers have recognized there is a growing concern among consumers that energy prices will continue to increase.

At least one major equipment manufacturer has been asking installers how they plan to deal with the SEER 13 transition. Emerson, the parent company of Copeland Compressor, carried out the survey to gauge contractor knowledge of transition details, including phase out of SEER 10 and the amount of contractor preparedness in marketing new equipment. It appears there is still quite a bit of confusion on when the final firm order for SEER 10 units can be placed; only about 25% of the contractors surveyed in April had been advised of this date by their distributor. It is interesting to note that 12% of contractors plan to continue aggressively selling SEER 10 units after the official transition date of January 23, 2006. (It's not clear how this would work out, but certainly the enforcement environment is different throughout the U.S.) Emerson also found that about a third of contractors had started working on SEER 13 marketing plans.

One manufacturer rep reported his company had announced release of a new heat pump product line specifically designed for the Pacific Northwest (with an HSPF expected to meet ODOE minimum requirements but with a more attractive price point than before). So far, we have not found any more mention of this product but it is under active investigation.

A bright spot in the new heat pump (and furnace) product lines involves controls. Manufacturers are actively promoting the next generation of thermostats. This means, among other things, that the homeowner has more scheduling flexibility. Also, the control is said to provide both better comfort and lower operating cost (through a newer adaptive recovery algorithm and "smart" utilization of the indoor fan) but there is little evidence to support this claim. Such evidence is less a concern to the manufacturer than consumer perception of what the newer control could do; the marketing approach is similar to that used to sell a new line of automobile. The main problem with this thermostat is that it can be more complicated to set up than previous thermostats; indeed, it is so new that installers may themselves not be completely fluent in its installation and programming.

Other new thermostats have become less complicated, still provide the needed level of heat pump control, and cost less than the last generation of adaptive recovery thermostats. Since a common homeowner complaint has been the complexity of thermostats, design engineers have apparently gotten the message that having the ability to quickly reconfigure a thermostat on-site with a simpler algorithm is very important. That is, even though the thermostat has many advanced capabilities, these may not be appropriate for many homeowners, who prefer a system that does the same thing every day and has little or no "mysterious" operation (such as coming on hours before the homeowner wakes up, a common feature of adaptive recovery). Whereas the last generation of controls could be configured more simply, they were not as transparent to program as many new controls.

In general, the news from the manufacturer/distributor side is that, while there is concern with the SEER change, the selling environment in the Northwest is favorable, especially given credits/incentives and the overall level of consumer interest in higher efficiency units.

4.3. Interview Findings

The most important findings from talking with installers and manufacturer representatives are as follows:

- Most installers say they use some sort of engineering approach to size heat pumps; however, the overall average size of systems suggests that first-cost economics also influence the decision on what size to recommend to their customer.

- Control of backup heat is a critical part of heat pump performance. The interviews suggest that the use of low ambient cutout and inclusion of resistance heat in first stage heating are less common across most of the Northwest than had been previously believed. Based on this finding, the assumptions used in C&RD savings calculations should be adjusted.

- Differences among standard equipment features largely determine which units can be installed with factory outdoor TXVs. The cost of adding an outdoor TXV is not insignificant (about $300 on average). It is unclear what benefits accrue from an outdoor TXV in light of the lab tests reported in Section 6 of this report.

- There is considerable resistance to installing heat pumps that use R410A. However, installers that specialize in higher efficiency units have more experience with R410A and report fewer concerns.

- Indoor thermostats create problems for some homeowners. More control options have not necessarily made homeowners feel more in control. Thermostats are changing and becoming simpler to control, and early reports from both installers and homeowners are encouraging.

- The federal requirement for equipment to meet a minimum SEER of 13 has created turmoil in the Northwest just as elsewhere; however, the perception (hope?) is that the demand for heat pumps will remain relatively strong in 2006 and beyond. All major manufacturers already have product lines that will meet these requirements.

- Installers report that first cost is still the most important influence on consumer behavior but expect that increased energy costs (for both electricity and natural gas) will have a greater impact in coming years.

5. Field Audits

One goal of this study was to establish characteristics of homes with existing heat pumps installed without utility intervention. This control group was drawn from homes in areas where no C&RD utility programs operated. Each area was targeted to match the weather characteristics of a participant area. The control group sample was drawn using the same strategy as the participant sample to correspond to the C&RD review. The control sample was drawn from the service territories of PacifiCorp in Bend/Redmond and Yakima/Walla Walla; Clark County PUD in the Portland/Vancouver area; and Puget Sound Energy in Kitsap County.

5.1. Field Protocol

The field protocol was divided into two separate reviews:

- House/duct audit
- Heat pump audit

The first review was designed to characterize the house, its heat loss rate and the overall characteristics of both occupants and particular building components. The second review was specifically focused on the heat pump. Each review required a separate visit by a designated specialist. The protocol is shown in Appendix C. This protocol was designed to parallel numerous field evaluations conducted throughout the region. It focused on the overall building size and characteristics, and included blower door and Duct Blaster® tests of the HVAC system and building shell.

The protocol was designed to require about four hours to complete by a trained auditor, augmented by a two hour assessment by an HVAC specialist with the appropriate certification and experience for handling refrigeration and HVAC systems. This provided a more thorough assessment of the heat pump's current operating conditions, state of maintenance, and the status of airflow charge and individual controls than previous studies have attempted.

The timing of the second visit was controlled largely by weather. The assessment of cooling charge must be conducted in relatively mild conditions (when the heat pump can be put into the cooling mode); therefore, homes that received an initial audit during the winter were not scheduled for the second phase of the protocol until spring. Due to this limitation, about 20% of the homes receiving an initial audit could not be scheduled for the second phase of the protocol. In the future, all testing should be postponed until both tests can be done within a short time frame to reduce the amount of attrition experienced in this effort.

5.2. Field Characteristics

The only criterion qualifying homes for inclusion in the field review was the presence of a heat pump in the home. We did not consider, for example, house type, equipment installation strategy, age of system, or heat pump make and model. The sample was designed so that both house and heat pump characteristics are reasonably representative of the regional practices within which they are located.

The primary housing characteristics that were reviewed included:

- House size
- Vintage
- Heat loss rate
- Information about the duct construction and location

These direct observations were supplemented by a blower door test, a Duct Blaster® test, system airflow test, and a test of the system operating static pressures. The blower door test was performed to assess the air sealing of the house and estimate the contribution of infiltration to its heat loss. The Duct Blaster® test was used to determine the overall air leakage of the supply and return ducts in order to assess their impact on the efficiency of the system. Airflow and static pressure measurements were taken in order to evaluate heat pump capacity and to normalize duct leakage readings.

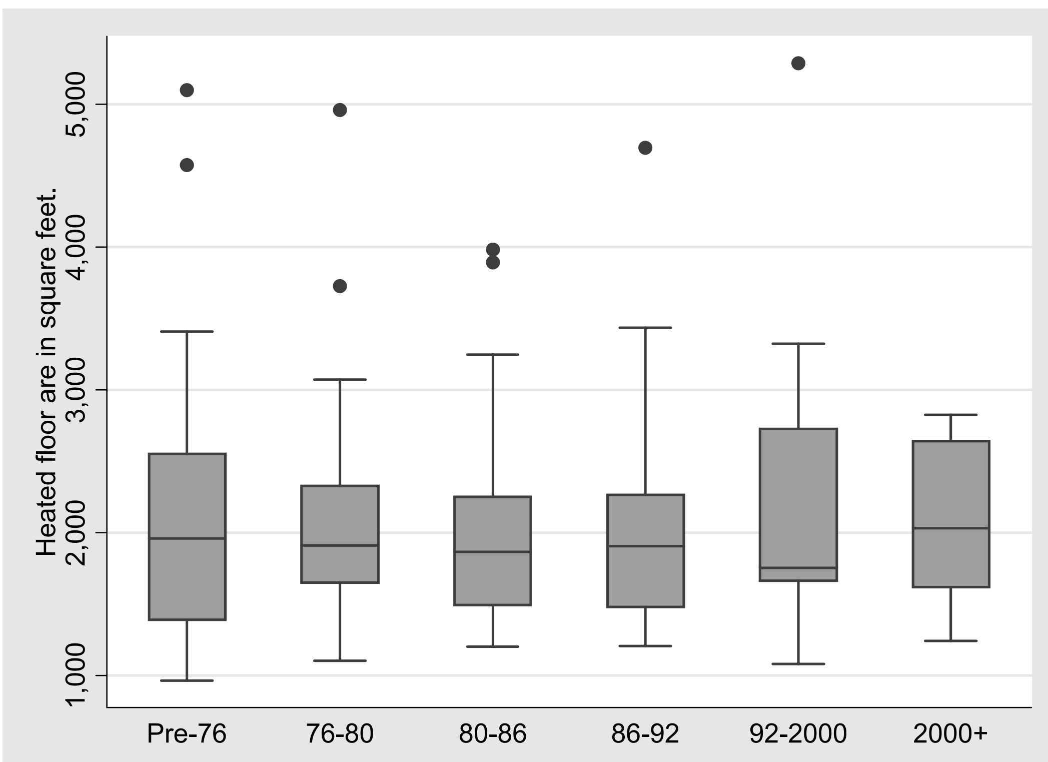

Figure 34. House Size by Vintage

Approximately five to ten percent of the field sample had data anomalies or missing data from the field review that made the calculation of the duct efficiencies, infiltration test results and/or overall building heat loss rates problematic. Since these calculations involve several separate pieces of information, there is a cascading effect that results in an overall attrition rate of 15 percent in the most complex of these calculations (duct efficiency).

Figure 34 shows the distribution of building area by vintage of house. Home size averaged approximately 2000 square feet. Size did not correlate well to the vintage of the home.

Table 18 summarizes house size, UA, and infiltration rates for each of the four localities. As can be seen, the values differ very little among the localities. This suggests that, at least for major characteristics, the sample is reasonably homogeneous in spite of the relatively diverse geographic areas included.

Table 18. Sample UA and Air Tightness Characteristics

Region	N	House UA/sf	Air Tightness		
			N	ACH50	ACH_{NAT}
Bend	41	.199	40	9.06	.45
Clark	40	.261	31	9.52	.48
Kitsap	40	.260	39	7.50	.37
Yakima	40	.218	38	7.52	.38
All	160	.235	148	8.34	.42

Table 19 shows the distribution of manufactured homes versus site built in the sample. 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 distribution of homes with basements in the Yakima and Bend areas seem to be somewhat lower than seen in previous new construction reviews. The overall heat-loss rates are characteristic of homes of similar vintages seen in other regional assessments.

Table 19. Housing Type and Area

Region	N	House Area (ft^2)	House Type	
			Manuf. (%)	Basement (%)
Bend	41	2,033	14.6	7.3
Clark	40	1,873	10.0	17.5
Kitsap	40	2,301	2.5	15.0
Yakima	40	2,155	10.0	12.5
All	161	2,090	9.3	13.0

Figure 35 and Table 20 show the distribution of heat-loss rate normalized by square footage across the various vintages. 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 energy codes, particularly after 1985.

Table 20. Normalized Heat Loss Rate by Vintage

Vintage	Heat Loss Rate (UA/sf)		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

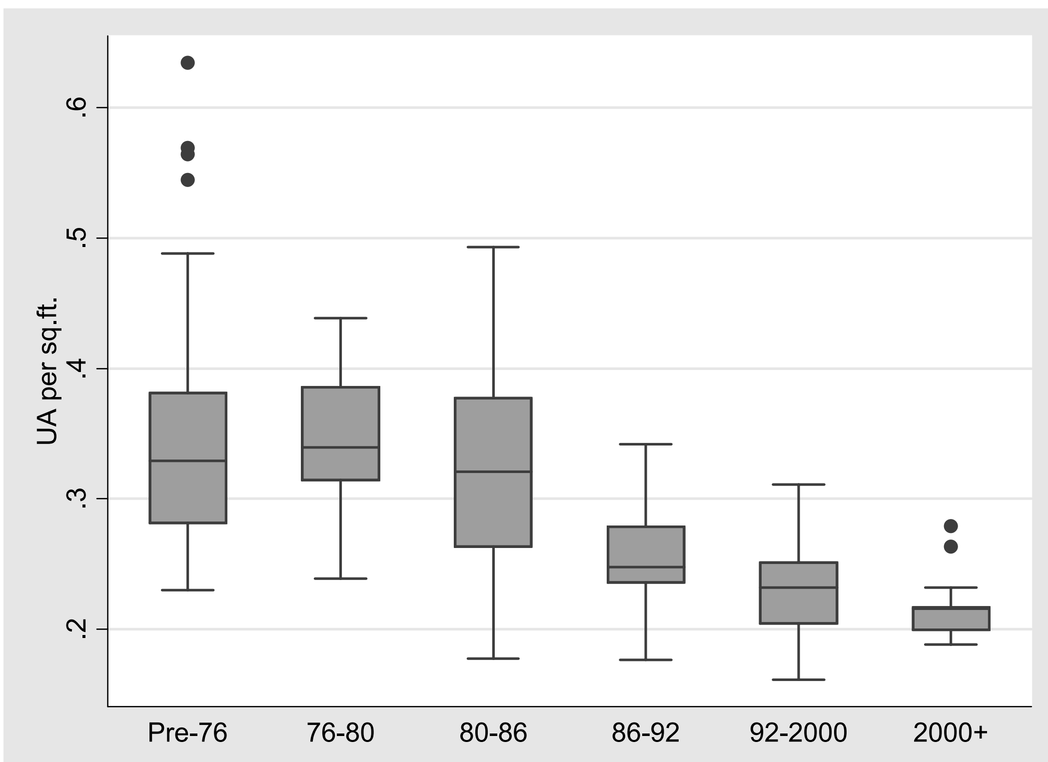

Figure 35. Heat Loss Rate (by Square Footage and Vintage)

Overall, even with the incidences of weatherization and home remodel upgrades, 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. This result is shown in Figure 36 and Table 21.

Table 21. Normalized Air Leakage Rate by Vintage

Vintage	ACH		N
	Mean	Std.Dev.	
Pre-76	.475	.151	41
76-80	.517	.181	22
80-86	.461	.213	18
86-92	.400	.113	22
92-2000	.304	.108	30
2000+	.292	.097	13
Total	.417	.168	146

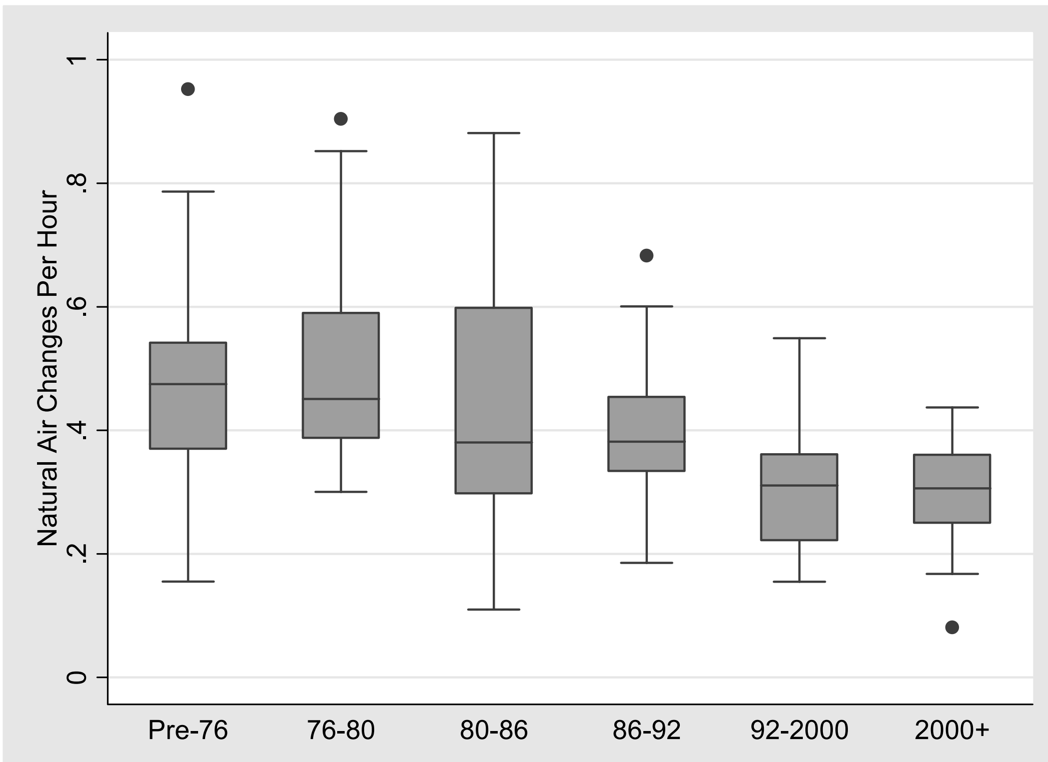

Figure 36. Air Leakage Rate by Vintage

5.3. Duct System Characteristics

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 building. In all current codes, duct insulation is required in such applications and, nominally, duct sealing is also required.

Duct systems in the field review did not follow the same pattern as the heat loss rate of the building nor the infiltration rates. 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. All this information was collected and evaluated, although complications in interpreting the results of Duct Blaster® tests resulted in some minor attrition of the sample.

Table 22 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). 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. Because this model requires an estimate of all four parameters of duct efficiency the attrition on the sample was almost 15%.

Table 22. Duct UA and Leakage Rates

		Leakage (% of airflow)		% Interior Ducts		Distribution Efficiency	
	N	**Supply**	**Return**	**Supply**	**Return**	**Entire Sample**	**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

The model (SEEM) was developed by Larry Palmiter of Ecotope and implements the ASHRAE Standard 152 duct calculation protocol. 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.

Figure 37 and Table 23 show the distribution of duct efficiency for the population by vintage. For this graphic, homes with ducts inside the conditioned shell (an additional 15% of the sample) were excluded. Table 23 includes the same data but 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 are considered to have no significant heat loss or air leakage, since any leakage occurs within the heated space.

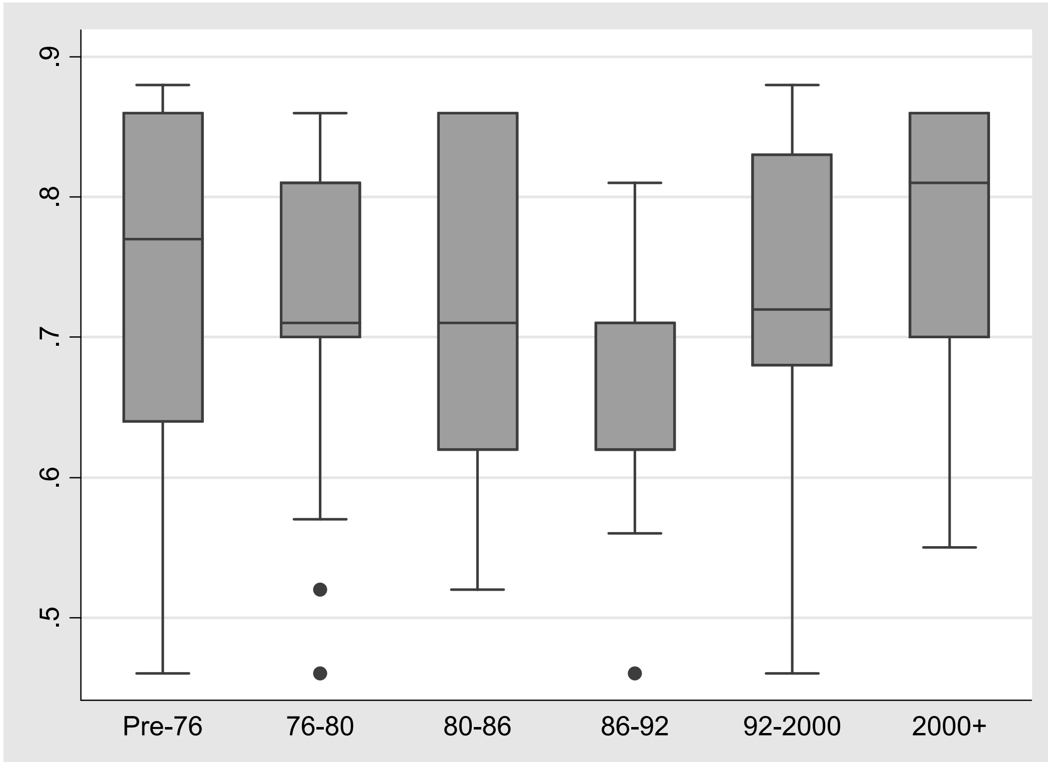

Figure 37. Distribution of Duct Efficiencies (by Vintage)

Table 23. 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 37 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.

Many of these homes had some duct sealing upgrade as part of the installation of the heat pump, even when the heat pump was installed more than fifteen years ago. In Kitsap County, PSE ran a popular heat pump program in the late 1980s in which some duct improvements and upgrades were mandated. Clark County PUD has recommended or required duct sealing as part of its heat pump financing program.

As shown in Table 22, the distribution in Clark and Kitsap Counties suggests somewhat higher efficiency in these two localities when compared to the other areas where such duct sealing practice was not included.

5.4. Heat Pump Characteristics

Heat pump data were collected in a comprehensive service call that typically took about two hours per site. The protocol is provided in Appendix C. A total of 126 sites were visited. The most important data were those that could be used to modify the assumed efficiency of systems in the field (most importantly, controls, system airflow, and system refrigerant charge).

About half of the field visits were carried out by Bob Davis of Ecotope; the remainder were performed by HVAC service techs using Ecotope's protocol. Mr. Davis was in frequent contact with the technicians as paperwork arrived so that he could ask clarifying questions and provide input into the testing process. In discussing the project and data collection with the technicians, Mr. Davis' intent was to get the data he needed but to also get as much of the technician's professional opinion on the condition of the equipment and critical performance factors as possible. There are many variations on the general heat pump installation theme and technicians are a tremendous resource in understanding these systems.

In general, this process was effective, but there was still considerable attrition between the house/duct audit and the heat pump audit (as shown in Table 5). This was primarily due to the time delay between the house/duct and heat pump audits. When the project began, Ecotope believed it would be most desirable to perform the heat pump audit during the cooling season to get a more definitive refrigerant charge evaluation. This may not have been as critical as originally thought, given findings from the Purdue work that showed only drastic charge levels would be expected to affect capacity and/or efficiency. Nonetheless, about 25% of sites only received a house/duct audit. For those not scheduled for the heat pump audit, the most common reason (60%) was that the homeowner and service company could not work out a visit time. The next most common reason (30%) was that the homeowner had lost interest in the project or had a different company service the system (despite having to pay for the visit) because of the time lag.

The protocol was developed to inspect or measure the following major items of interest:

- Outdoor (cut-out) thermostat setting and operation. Depending on setting and operation, this device, along with the low ambient control, has the most potential to reduce unnecessary electric resistance heat usage.

- Characteristics of the heat pump, including performance and capacity ratings. For the most part, this review was based on manufacturer's information either recorded on-site or derived from catalogue data after the site visit.

- Presence of compressor low ambient control, which shuts off the compressor below a specified temperature.

- Airflow across the indoor coil, which was usually measured by the house/duct auditor but sometimes also was measured by the HVAC technician when he did not have easy access to the measured flow data. This occasionally occurred when the data were taken in a different visit and couldn't always be conveyed to the HVAC tech in time for his use.

- Refrigerant charge level (found from heating and/or cooling season measurements). This evaluation, along with an assessment of airflow across the indoor coil, is a major component of current commissioning programs such as CheckMe!®.

- Indoor thermostat type (programmable or not), thermostat schedule, and presence or absence of control wires needed to enable an outdoor lockout thermostat.

- Type of refrigerant metering device on indoor and outdoor coil. Depending on whether a TXV is in place, the implications of non-optimal refrigerant charge and system airflow differ. The pre-study assumptions about the impact of the type of metering device on system capacity ("tons") and coefficient of performance (COP) were altered by the laboratory findings. However, knowing the type of metering device is important.

Other items were also inspected: indoor and outdoor coils, type of defrost system (and whether it is operational), presence of safety devices such as crankcase heater and suction line accumulator, the condition of the reversing valve (from a temperature split test), and the operating amperage of the compressor, indoor fan, and outdoor fan. These are items which are often (but not always) inspected during a heat pump service visit; some of this information is often useful in assessing the key measurements such as airflow and refrigerant charge.

5.5. Field Survey Findings

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:

- None of the field sites had compressor low ambient cut out enabled. Based on information gathered some time ago, Ecotope believed this practice was used by some installers to enhance comfort. However, the only new information collected that supports this practice (installer interviews) is from sites in areas where field work was not done (TriCities and Spokane). This finding will have an effect on the characterization of the energy usage of the base case heat pump in RTF calculations. The current C&RD calculator assumes low ambient cut out is used in 10 percent of new installations. Such an assumption should be reduced considerably if not removed altogether.

- About 35% of sites visited had an operating outdoor thermostat (ODT). About 75% of the Clark County PUD sites have ODTs; they also had a program which required an ODT during the time many of these sites were selected. This is in line with pre-survey expectations. The average outdoor thermostat setting found in the field was 40°*F*. The C&RD calculator currently assumes a 30°*F* setting or compressor staging that effectively eliminates resistance operation above 40°*F*.

- About two-thirds of sites without ODTs had the extra wires needed to install one without requiring new wiring runs, providing the potential for added savings without prohibitive cost for this measure. For PTCS, the regional assumption is that the ODT could be retrofit as part of any re-commissioning. This review suggests that as many as 15% of the systems don't have an ODT and would incur substantial installation expenses to install one.

- About 80% of indoor thermostats are programmable units (with approximately 75% of these being 7-day units, according to the model number). 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 probably 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 was 65°*F*. The average setback (when it was used) was 6.7°*F*55% of systems had a setback greater than 5°*F* from the nominal heating setpoint.

  The current assumptions for heat pump performance assume a setback of 6°*F* with no adaptive recovery. The algorithm assumes the elements are turned on after one cycle if the set point is not reached. This reflects morning warm-up operation, with electric resistance supplementing the compressor in the warm-up hour(s). Adaptive recovery is not modeled and it remains unclear what algorithm is used by the thermostat designers. Only detailed monitoring is likely to resolve the modeling assumptions and performance questions, given the current state of understanding regarding this technology.

- In general, indoor thermostats behaved as expected. On a normal Stage 1 heating call (thermostat turned up by 1°*F*), 15% of thermostats activated a resistance element; the median time the element stayed on was 5 minutes. Depending on the internal programming and the response of the thermostat, it is possible for an element to be activated during Stage 1. However, the duration of the activation is what will determine the added electricity used during a heating season. It is hard to know the overall effect of intermittent backup heat usage during Stage 1 heating without detailed monitoring. However, this usage will be captured in the billing analysis, although it cannot be disaggregated from the heating provided by the compressor. For the most part, current modeling assumptions do not assume elements operate in the first stage unless there is a specific control override. Approximately 60% of the systems modeled use elements in the first stage. In about 20% of those cases, the elements are on much more than we observed here. In the remaining cases they are on less. Re-specifying these modeling assumptions is probably necessary, given these findings.

- System airflow (across the indoor coil) was measured with a TruFlow® air handler meter. The median value was 326 CFM/ton, based on size of the outdoor unit. This size 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 38. The lower quartile of flows starts at 292 CFM/ton. Therefore, even though the measures of central tendency do not show critically low flows, a significant number of systems are moving less than 300 CFM/ton.

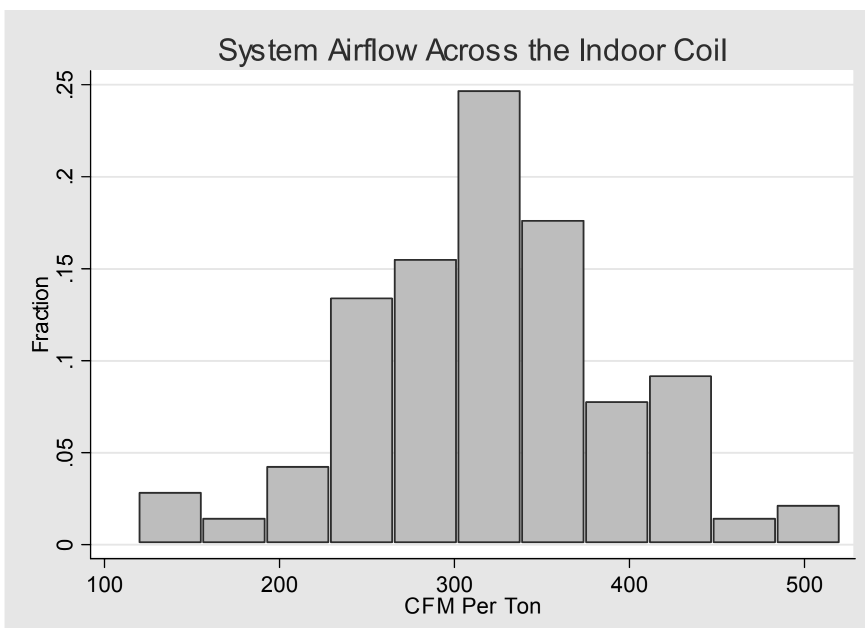

Figure 38. System Airflow Distribution

The current specification and 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.

- Many of the newer heat pumps systems use a variable speed indoor fan which is designed to deliver consistent flow over a wide range of entering and leaving air conditions and pressures. At least 30 cases 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 indicates the ducts might be restricting airflow. 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 39.

    - For the lowest quartile of air handler flows (CFM/ton <= 292) the external static readings were not notable; 90% of these cases have an external static pressure of <=100 Pa.

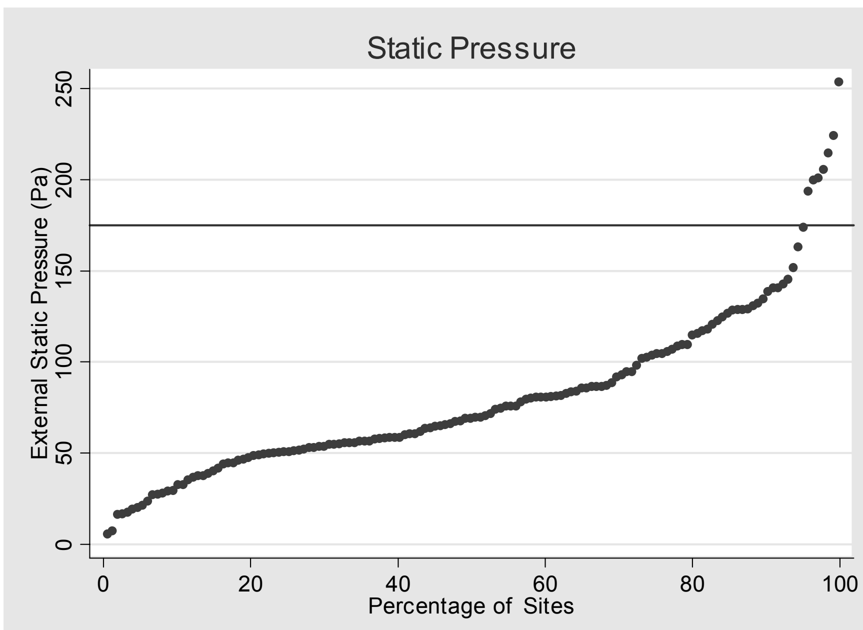

Figure 39. External Static Pressure

- 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 technicians 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). The distribution of these results is shown in Figure 40.

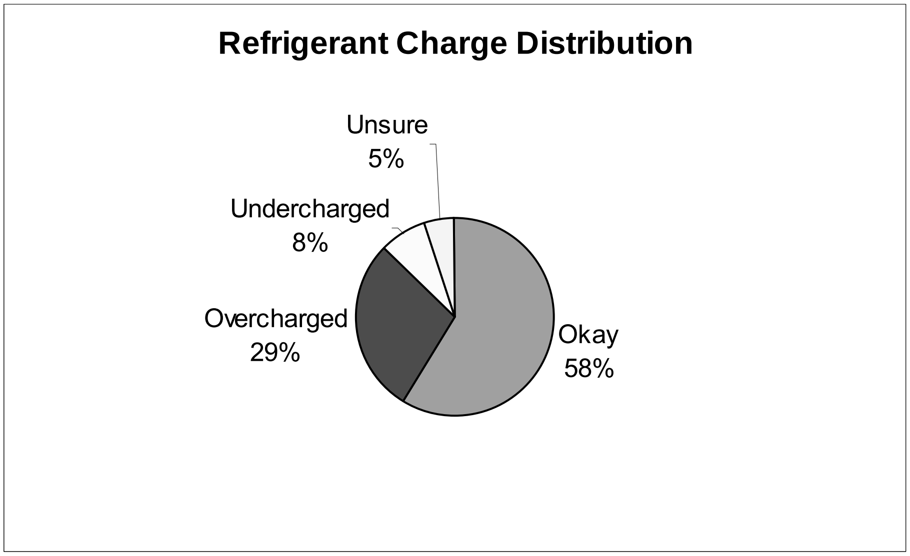

Figure 40. Refrigerant Charge Distribution

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.

Where possible, operating pressures were also compared with expected pressures based on ambient conditions. This combination of evaluation techniques is less exact than removing charge and weighing it and then comparing the result to the expected amount (including an allowance for the refrigerant line set), but the field procedure required evaluation of a number of other components and therefore a quicker method was used. Results are shown in Figure 41.

Figure 41. Field Sample Refrigerant Charge Evaluation The survey results for the refrigerant metering devices are shown in Figure 42 and Figure 43. Overall, 65% of systems were equipped with an indoor TXV, and 51% were equipped with an outdoor TXV. The outdoor TXV influences heating performance (since the indoor TXV is bypassed during heating operation).

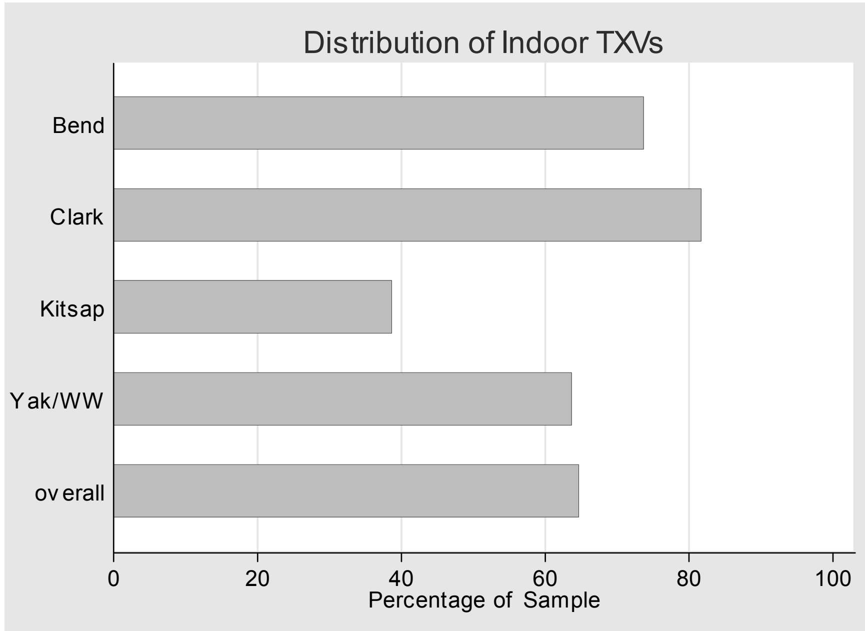

Figure 42. Indoor TXV

- Kitsap and Clark County have the highest percentage of units with outdoor TXVs. This is to be expected in Clark, since the average HSPF for its program heat pumps is higher than the sample average. The reason for Kitsap's outdoor TXV percentage is mostly related to the predominant manufacturers in that region; they include an outdoor TXV in more of their heat pump product lines regardless of nominal efficiency.

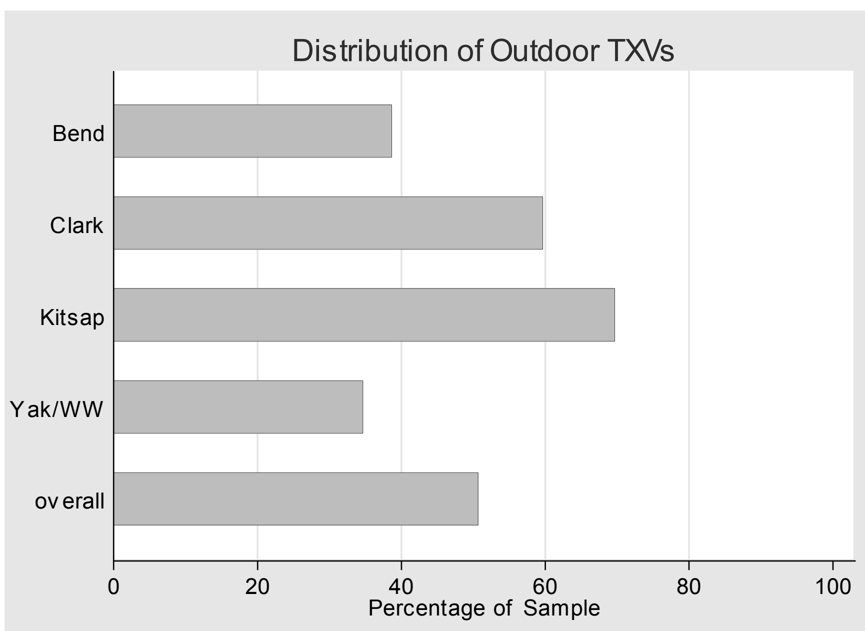

Figure 43. Outdoor TXV

5.6. Heat Pump HSPF

The performance of a heat pump is summarized by a wide variety of capacity and energy use at different temperature bins. Each heat pump manufacturer is required to rate their products using several standardized tests at various temperatures. 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. This index is the Heating System 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. Since this is the minimum standard, the mean HPPF 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 24. This is probably the result of the Clark County PUD heat pump program, which has encouraged higher performance heat pumps in this sample.

Table 24. 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

Figure 44 shows the relative distribution of HSPF across the sample. Except for Clark County the distribution reflects a pattern that would be expected in the absence of any utility intervention. In these areas, the distribution appears reasonably consistent. The last entry in Table 24 shows the average distribution without the Clark County sample.

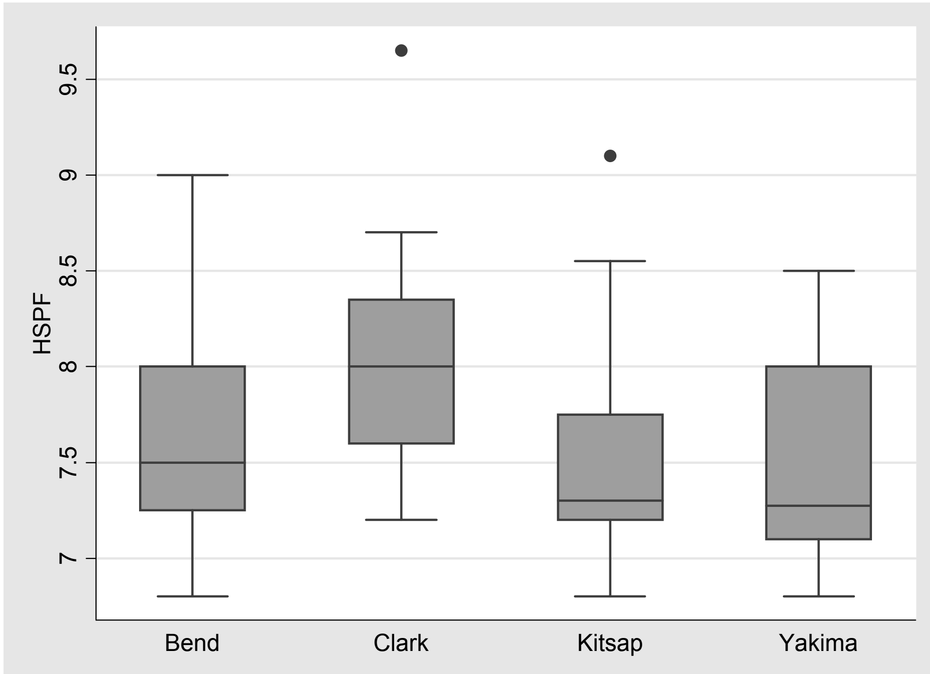

Figure 44. HSPF by Region

5.7. System Sizing

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. The audit was not sufficiently detailed regarding window orientation and shading to allow a cooling sizing estimate.

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.

To assess the heat pump, the nominal capacity of the equipment was used. This 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 heat load of the house at this temperature is less than half the load at design conditions. This is problematic because the typical heat pump loses 25 to 50 percent of its capacity at 17°*F* outside temperature (much closer to the design temperature). 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 25 to 50 percent larger than the heat load calculated from the design temperature.

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 sized compressor. Only homes with substantial solar heat gain (through south and west facing glass) would routinely require compressors sized above the nominal size from the heating calculation. In reviewing the cooling sizing, it does appear 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. It is difficult to explain the equipment sizes we observed as a response to the overall engineering sizing.

Given a heating load sizing criterion, the heat pump sizing observed in this sample is surprisingly small. Table 25 shows the results of the sizing summary across the four study regions. As can be seen in Figure 45, the sizing methods used for this sample appear to be very close to the nominal capacity sizing with little or no compensation for any drop in heat pump capacity or duct losses. Indeed, when duct losses are taken into account, the sizing of the equipment is less than the calculated heating load, even though the nominal equipment tons were used in this calculation. The impact of duct losses can be seen in Figure 46.

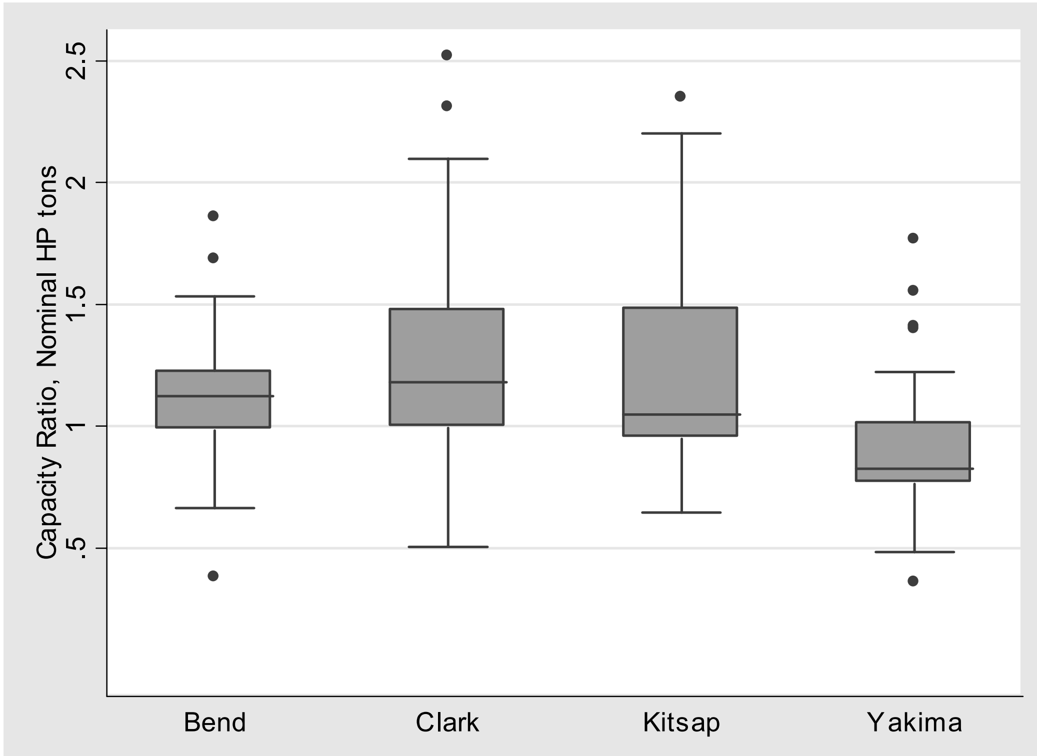

Figure 45. Capacity Ratio (heat loss rate to nominal HP tons)

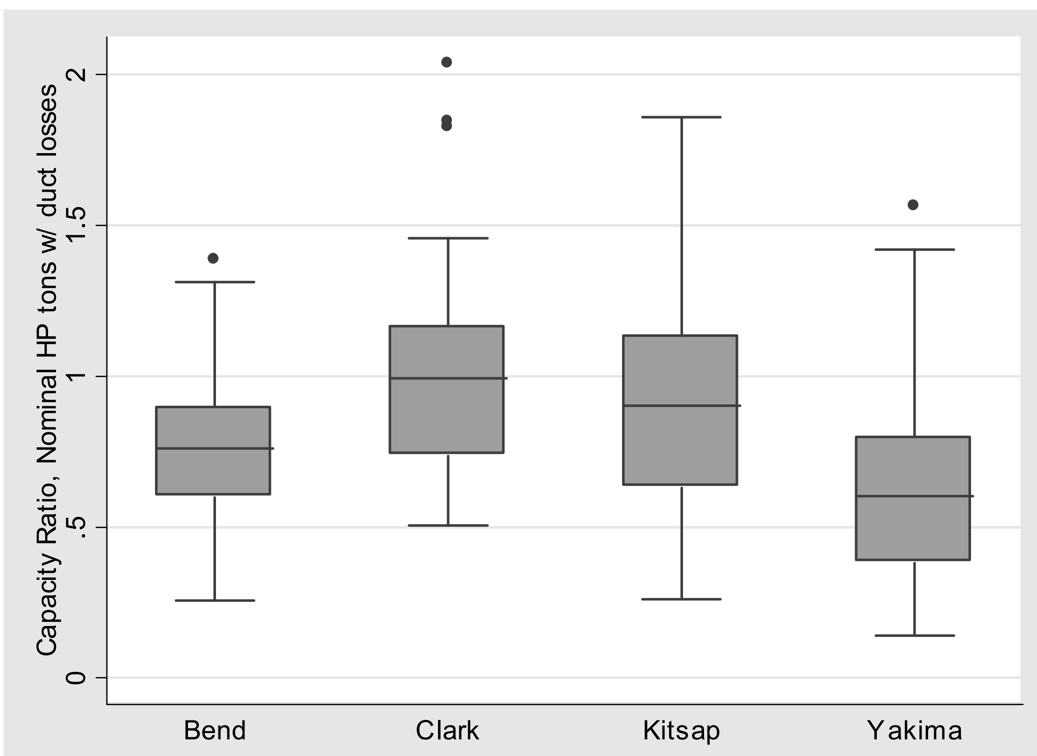

Figure 46. Capacity Ratio (heat loss rate to nominal HP tons) with Duct Losses

Other researchers have expressed the concern that this equipment tends to be oversized. At least in this study, it appears to have been undersized considerably. The impact of this is to reduce the first cost of the equipment itself and thus reduce the cost of the installation (at the expense of additional use of back up heat and associated heating energy expense).

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. The impact of 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 well be that, in the context of utility rates, the installer and consumer are actually making a more optimal economic decision. In other words, the increased cost in energy use may not be enough to justify the increased expense of a correctly sized heat pump, even though the smaller system would limit the energy savings available from the heat pump.

Table 25. Heat Pump Sizing by Region

Area	Capacity Ratio (heat load/compressor size)	
	With Ducts	Without Ducts
Bend	.77	1.13
Clark County	1.03	1.29
Kitsap	.91	1.20
Yakima	.63	.91
Total	.83	1.13

5.8. Homeowner Interviews

During the initial audit, homeowners were asked about their satisfaction with their heat pump. They were also encouraged to supply other feedback regarding their heating and cooling system. For the most part, participants had relatively few comments. Approximately 40% of the sample had specific comments on the function of their systems. These have been divided into two major categories:

- Complaints about the distribution system.
- Complaints about service.

The most common complaint came from about 12% of the homeowners and referred to one or more areas of the home being "too hot or too cold" relative to their expectations; i.e., inadequate heating or cooling had been observed. About 11% of the sample complained about difficulty getting service scheduled or with repeat service calls. About 10% of the sample complained about more general comfort considerations. These were primarily focused on the inability of the heat pump to blow adequately warm air (at least for part of the cycle). This is a well-documented feature of heat pumps. When the results of the interviews were compared to observed duct characteristics, there was no relationship between duct efficiency and perceived occupant comfort.

About 6% of those interviewed complained about high bills. The homeowners said that their bills were "just higher than they should be" and they attributed this to a poorly functioning heat pump. On this variable, even though the sample size is quite small, it does appear that an unusually significant difference in duct efficiency could be observed. In one case, the duct efficiency tested 20% lower than the average for the remainder of the sample.

Another 4% of the sample complained about the behavior of the thermostat; these were usually regarding programming difficulties. Certainly the most significant problem related to the thermostat was the complaint that the thermostat caused the heat pump to power on at some times when it wasn't desired or to fail to go on at necessary times. In these cases, the technician often provided significant testing of the thermostat along with resetting or reprogramming when necessary.

A surprisingly large number of homes surveyed had a supplementary heating system in the form of a wood, propane, or natural gas fireplace. However, under only a few isolated circumstances did wood heating represent a substantial fraction of the heat supplied to the building and impact the frequency of the heat pump operation. In two of the cases, the total amount of heat generated by the woodstove appeared to be as much or more than half of the heating that might be expected for a home of that size.

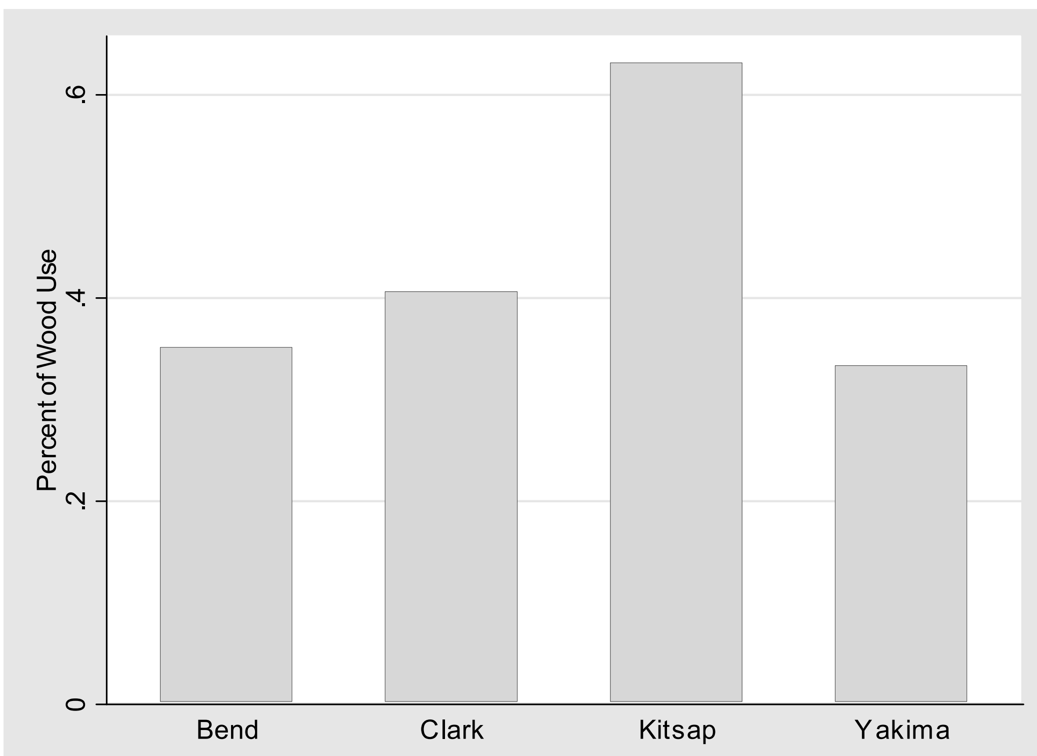

Figure 47. Distribution of Wood Heat

Figure 47 shows the distribution of wood-heat use. It shows that in most of the localities studied, between 30%-40% of homes used wood heat. Only in Kitsap County did the number exceed 60%, indicating that a significant fraction of heating energy used in this area is provided by wood.

5.9. Billing Analysis for the Field Sample

While the field sample was relatively small, we did conduct a detailed billing analysis for this group. 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 is essentially a comparison using a median low bill methodology, which uses an inspection of the lowest bill 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. While the analysis is fairly arbitrary, it does allow for adjustments to the base load estimates that can account for the difficulties in evaluating any heat pump system for both heating and cooling.

In all of these cases, the minimum bill is a “shoulder month” and included a substantial amount of heating and cooling. Thus, the base load is significantly biased by the actual behavior of the HVAC equipment. In order to accommodate this, the analysis was modified to derive an estimate of the base load from occupancy characteristics (the number of occupants, water heat fuel, etc.) that were noted in the occupant survey. This was then modified to accommodate the observed bills. The base load is a derivation of this value over an annual period.

The heating and cooling estimate is calculated as the difference between this base load and the observed bills in any particular month. It is also necessary to separate cooling and heating. In order to do this, the entire billing record was summarized for each climate for all houses and the “minimum bills” months. These were assumed to represent the swing months, and the differences between the base load and the values observed in these months were taken to be half heating and half cooling. The overall result was to have a cooling season that was more-or-less consistent across the entire climate zone observed and a heating estimate that was similarly consistent over that same period.

Note that these estimates are approximate at best. We believe that, while they may include substantial errors, they are essentially unbiased since we have made the effort to accommodate the base load and attempted to prevent this estimate from being contaminated by the HVAC loads that might occur in the swing months. The result of this analysis is that heating, cooling and base load estimates were calculated for each home. This information was subsequently summarized and compared with the characteristics data, as shown in Figure 48.

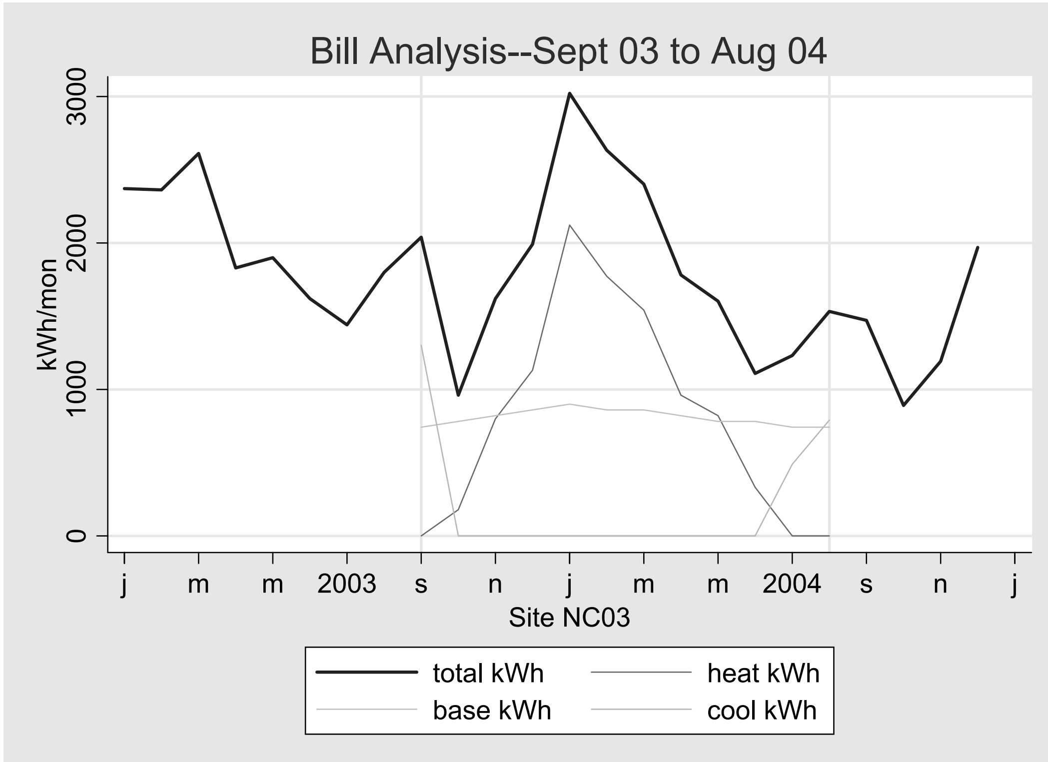

Figure 48. Distribution of Consumption by Load Type
(Note: Bills spanning 23 months were collected; the analysis period was 9/03 - 8/04.)

Figure 48 illustrates the distribution of bills and the disaggregation of the bills into the three major categories for each site: heating, cooling, and base loads. This disaggregation was conducted on each site but limited to the year from September 2003 through August 2004. This year was selected because virtually all the sites had this year present. In the case illustrated, 23 months were actually present but only the 12 months from September to the following August were used.

This process allocates any given bill to each of the three enduse categories depending on time of year and the assessment of base load characteristics. Therefore, the sum of the enduse kWh is equal to the total consumption. Given the highly variable and erratic nature of the bills, the individual months for the year are separately aggregated into the three major categories. The annual estimates become the basis for evaluating home performance against the relevant variables that might determine heating and cooling consumption; i.e., duct efficiency, heat pump efficiency, and overall building UA.

Although the billing analysis inherently contains a certain amount of error from the allocation necessary in the median low bill analysis, we believe this sample is unbiased in that the cooling load itself is not either artificially deflating the heating load nor artificially inflating the base load estimate.

Table 26: Energy Use Estimates by End Use (N=122)

Load	kWh per Year	Standard Deviation
Heating	8,286	5,183
Cooling	1,669	1,451
Base	11,474	3,598
Total	21,136	6,825

Table 26 summarizes the heating, cooling and base load actual observed kilowatt hours for the test year used in this analysis. Table 27 summarizes the same data by region after normalizing for building size.

Table 27: Normalized Energy Use Estimates by Region and End Use

Area	N	kWh per Ft^2 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

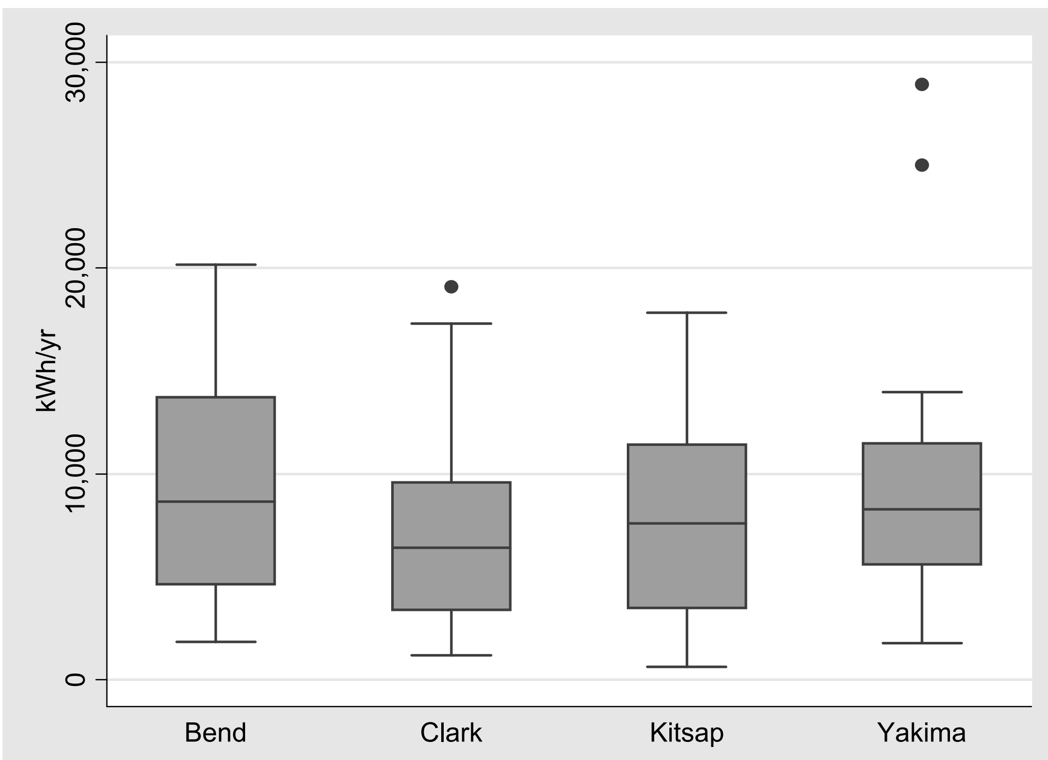

Figure 49. Heating Load by Region

As can be seen, there are reasonably consistent values assigned to heating and cooling once the sample is normalized by building size. Figure 49 illustrates this same result for heating only, with the expected pattern that the two coldest climates (Bend and Yakima) show more heating energy used than Clark and Kitsap Counties. It should be noted, however, that the Kitsap sample had a large amount of wood or other supplemental heat.

Similarly, with the cooling load shown in Figure 50, less than half of the energy is used for this end use in Kitsap County than for the rest of the sample, while the cooling load is noticeably higher in Yakima.

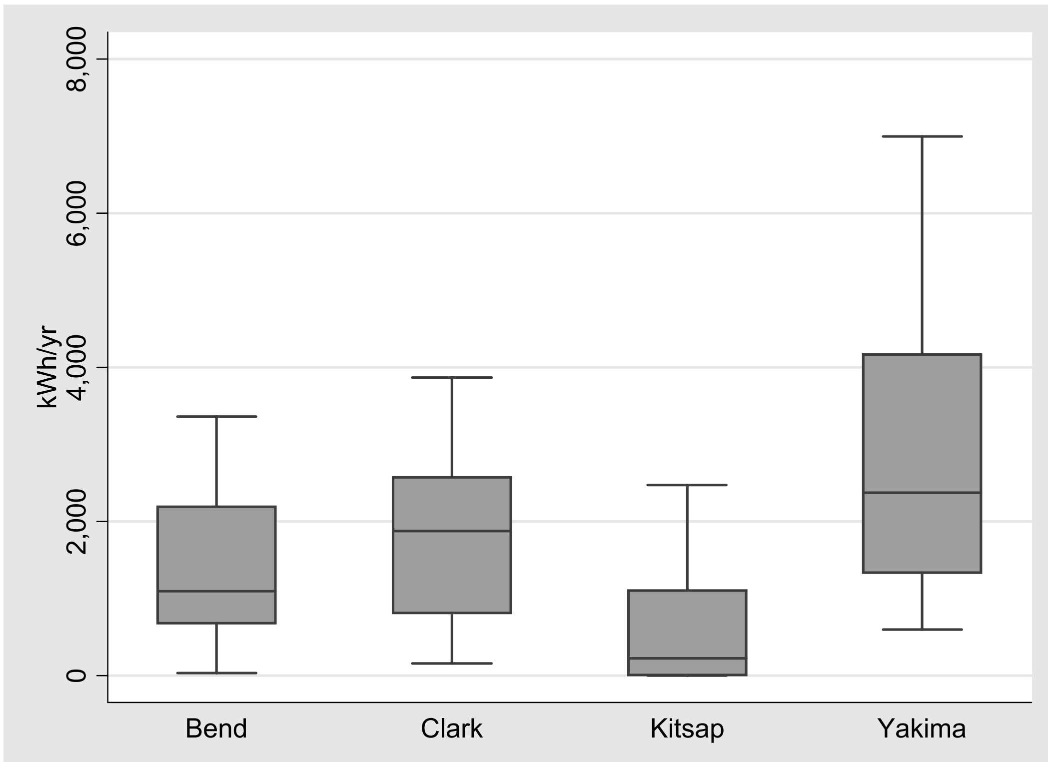

Figure 50. Cooling Load by Region

This analysis suggests an overall heating load estimate of about 4 kWh/ft^2/yr. When this is reviewed in the context of the expected values, it is clear that the heating levels used are about 50% of what would be expected if the homes were heated with an electric forced air furnace. While an accurate assessment of this comparison is impossible, when the runs were done to estimate duct efficiency, a similar estimate was made for heating. This suggested that the expected value for space heat is approximately 8.5 kWh/ft^2/yr in this sample.

The observed values in this sample include a substantial amount of wood heat in about 25 percent of the sample. Some amount of supplemental gas or propane heat was also noted. Nevertheless, the amount of heating observed in this sample is approximately 4 kWh/ft^2/yr. This suggests an effective COP over the entire sample of about 2. While this value is probably lower than would be expected given the observed heat pump HSPF, it does suggest that heat pump performance is providing a substantial benefit to the homes in this study in spite of difficulties in commissioning and airflow.

A further comparison was made between these space conditioning (combined heating and cooling) results for the sample and the space conditioning assumptions used in the RTF spreadsheet to generate the deemed savings used by the C&RD. To do this, comparison the vintage bins in this analysis was collapsed into the three bins shown in Table 28. The space conditioning load was then summarized and compared to the estimates developed in the RTF analysis. Table 28 shows the result when these assumed loads are normalized to be comparable to the loads in this sample.

The RTF spreadsheet divides the region into six climate zones. For this comparison we used Heating Zone 1, Cooling Zone 2 to represent the western climates (Kitsap and Clark) and Heating Zone 2, Cooling Zone 3 to represent the eastern climates (Bend and Yakima). While sample sizes are small, there is reasonable consistency between the estimates in the RTF spreadsheet for existing homes with heat pumps and the space conditioning load observed in this sample for the 2003 base lines (especially in the western climates). Agreement with 2005 is better in the eastern climate zones. None of the 1980 vintages seem to be consistent with the baseline assumptions. This may be the result of supplemental heat in this subset of the sample (about a third of this group mentioned more than one ton of wood or pellet use per heating season).

Table 28. Normalized Space Conditioning, Sample vs. RTF

		Climate					
		Western			Eastern		
Vintage	N	Sample	RTF 2005	RTF 2003	Sample	RTF 2005	RTF 2003
		(kWh/sf)	(kWh/sf)	(kWh/sf)	(kWh/sf)	(kWh/sf)	(kWh/sf)
Pre 1980	62	5.13	6.19	5.28	6.71	6.68	6.06
1980-1992	28	3.76	7.28	4.74	5.03	7.87	5.15
Post 1992	26	3.15	4.96	3.48	5.80	5.47	3.80

5.10. Field Results

The field review was meant to provide data which would cast light on the base case conditions for heat pumps installed in the region. Many of the modeling and calculation assumptions used in the RTF calculator and in other regional heat pump programs are addressed, in part, in this sample:

- House heat loss rates were used to calibrate the prototypes. The findings here suggest a consistent improvement in the heat loss rate over the last three decades. Between homes built before 1980 and homes built after 2000, the normalized heat loss rate dropped 41 percent. This includes numerous weatherization measures in the homes that were retrofit after the home was constructed. This reduction in heat loss reduced space conditioning energy use by about 25% across the same time period.

- Unlike the results of other studies, there was no particular change in home size over the vintages in this sample. In fact, there was remarkable consistency among the various subgroups in this sample, except for the manufactured homes, which were 23% smaller than the site built homes.

- Duct leakage and efficiency were measured on virtually every home in the field sample. The duct leakage measurements resulted in a supply leakage fraction of about 13% and a return leakage fraction of about 12%. This translates into an overall estimated duct efficiency of about 70%. The distribution of this result across home vintage was remarkable. The most inefficient duct systems were from homes built from the mid-1980s to the mid-1990s. Surprisingly, the newest homes and the oldest homes in the sample had the most efficient duct systems.

- A variety of components of the heat pump installation were reviewed.

    - The most significant finding was that the air flow provided by the indoor unit averaged about 320 CFM/ton of compressor capacity. Over a third of the sample had airflows below 300 CFM/ton.

    - A third of installations in the sample had outdoor thermostats (ODT) that would control the operation of the electric back-up elements. The majority of these were in the Clark County PUD service territory, where the utility mandated them as part of their program. Two-thirds of the remaining homes had wiring capable of receiving an ODT retrofit.

    - The heat pump sizing observed in this study was somewhat smaller than expected. On average, heat pump compressors in this study were sized to about 83% of the heating load when ducts are taken into account.

    - The average HSPF of the heat pumps in this study was 7.7. However, the Clark County PUD sample was generated from homes that had participated in some utility programs. When this group is removed, the average HSPF falls to 7.5.

- Cooling loads make up about 17% of the space conditioning energy in the systems.

- Space conditioning energy use observed in these homes was reasonably consistent with the base case assumptions used in the RTF spreadsheets.

- Occupants had relatively few complaints, but about 22% of homeowners noted problems associated with comfort or the distribution of heat in the home. Only about a quarter of this group mentioned “cold blow.” About 12% of homeowners had complaints about the HVAC service they had received.

6. Laboratory Testing of Heat Pump Performance

Testing of a 3-ton Carrier YKC heat pump was performed in a laboratory setting at Purdue University. The goals of this study were to determine the effects of refrigerant charge and airflow variations on the heat pump performance in heating mode. In addition, the impact of a thermostatic expansion valve (TXV) versus a fixed metering device (FEO) on the outdoor unit (evaporator in heating mode) was explored in the context of variations in both charge and air flow. The results were also intended to provide measures of cycling degradation (C_d) and the defrost degradation factor on a typical heat pump over the range of testing. These are essential to the calculation of the HSPF rating used to establish the relative performance of heat pumps. In all configurations, only the heating mode of the heat pump was tested. These tests parallel the Air-Conditioning and Refrigeration Institute (ARI) Standard 210/240 and allow calculation of overall performance indicators used to rate equipment under this standard. The combined effects of all the variables are demonstrated through HSPF ratings calculated using the test data applied to various climate zones relevant to the Pacific Northwest region.

6.1. Adjustment Factors

Problems in the test setup led to corrections of the raw air-side capacity and airflow measurements based on the more reliable refrigerant-side measurements. The resulting measurement error embodied in the final results, after the corrections to the raw data, is approximately 1-3%, depending on the flow conditions. Lower airflow rates result in uncertainties at the upper end of this range. The adjustment factors applied to all subsequent capacity and airflow data, and are shown in Table 29.

Table 29. Purdue Adjustment Factors using Refrigerant-Side Capacity Measurements

	800 cfm_{nom}	**1100 cfm_{nom}**	**1300 cfm_{nom}**	**1500 cfm_{nom}**	**1700 cfm_{nom}**
FEO	1.0403	1.0909	1.0570	1.0088	1.0068
TXV	0.9986	1.0763	1.0504	0.9957	0.9872

Although this method of adjustment was determined to be the best available, Appendix D details the adjustment analysis and discusses the shortfalls of this adjustment methodology. Appendix D also describes several unsuccessful attempts to find an even more robust adjustment. Despite the fact that the post-test analysis did not result in a better adjustment method, much was learned about preventing these problems and otherwise improving the data in future laboratory testing. We believe that we could eliminate most of the capacity discrepancy with minor alterations to the test configuration and methodology.

6.2. Results: Capacity and COP

In general, the results show that, for this heat pump, the impacts of refrigerant charge are minimal except at greatly reduced levels (70% of manufacturer's specification). The impacts of airflow are also limited to cases with very low air handler flow (less than 300 CFM/ton). Use of a TXV improves overall performance for this heat pump, as seen in the HSPF plots in the following section, but has minimal impacts on the effects of low charge and low airflow under most conditions.

Figure 51 shows the capacity versus charge for a heat pump with a fixed metering device, with points labeled by nominal airflow rate; each line represents a common airflow rate. As expected, the capacity increases with increased outdoor temperature. The trend with respect to charge levels is fairly flat, with the largest effects seen for low charge levels at high temperatures (47º*F*) and very little effect seen as the charge increases beyond the specified charge level (100%). Within each temperature group, the effect of airflow rate can be seen. The capacity is most affected at the lowest flow rate, but it is important to note that measurements at this flow rate were the most difficult to measure and have the most uncertainty. Figure 52 is the same as Figure 51 for a unit with a TXV instead of a fixed orifice. A comparison of these two plots reveals relatively little impact at 17°*F* and 35°*F*, with the capacity at low charge levels improving slightly with a TXV.

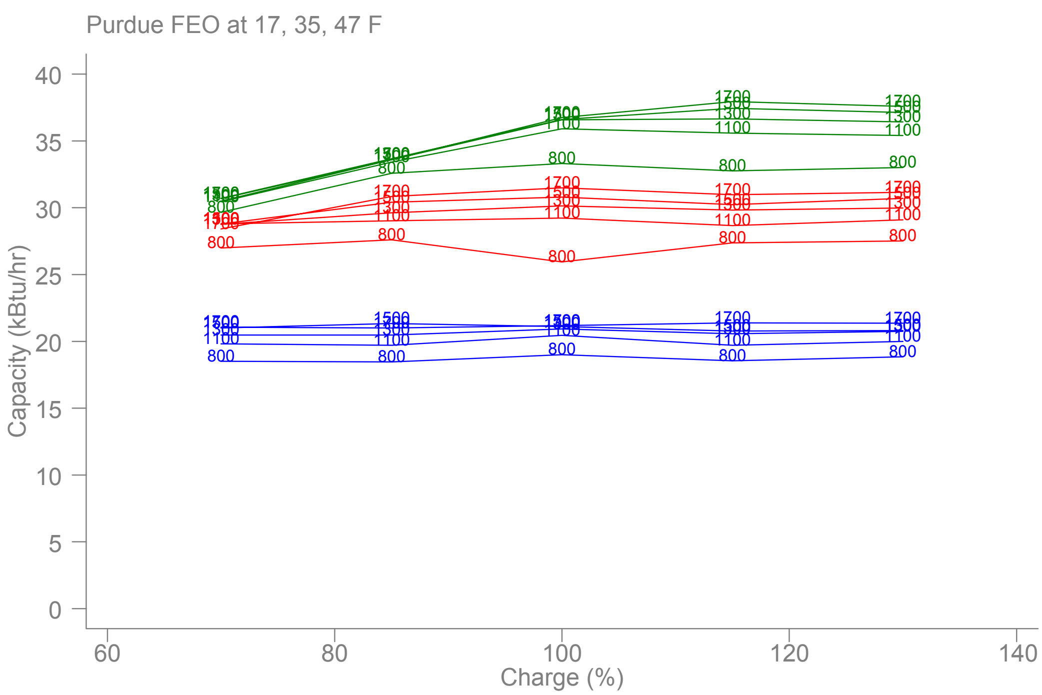

Figure 51. Capacity (fixed metering) versus Charge at three outdoor temperatures
17*F*(blue/lower group), 35*F*(red/central group), 47*F*(green/upper group). Points labeled by nominal airflow rate (cfm).

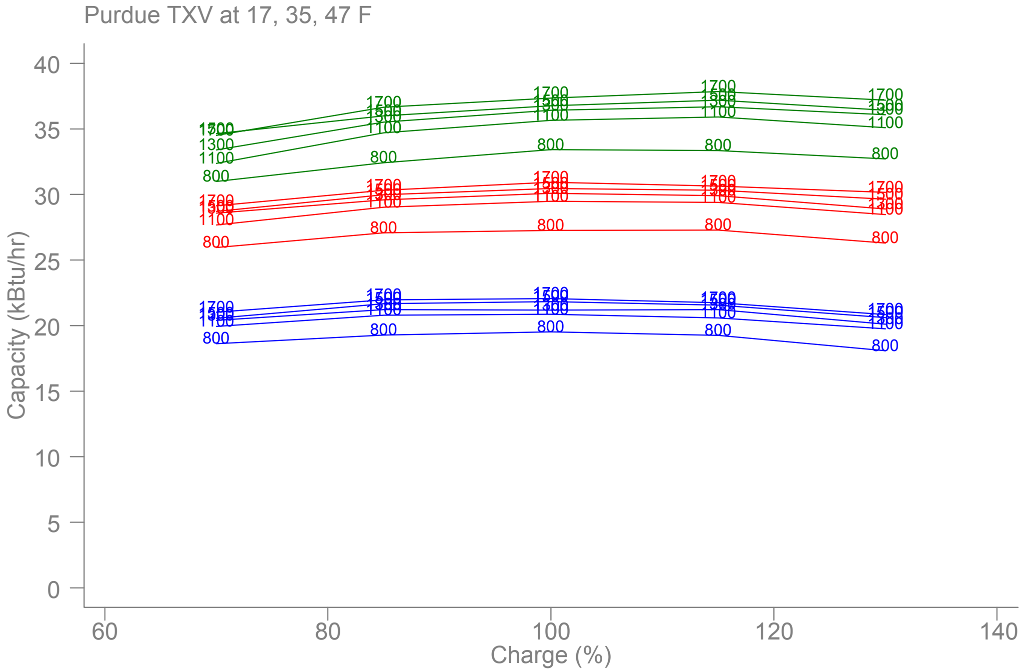

Figure 52. Capacity (TXV) versus Charge at three outdoor temperatures
17F(blue/lower group), 35F(red/central group), 47F(green/upper group). Points labeled by nominal airflow rate (cfm).

The effect on heating capacity is not necessarily the most relevant benchmark of the implications of changes in these variables. The common practice of over-sizing heat pumps means that, particularly in mild climates, the heat pumps are rarely operating near their capacity limits in the heating mode, especially in the higher temperature bins. In the sample of homes examined in this study, (Section 5.0), the equipment tended to be under- rather than over-sized. Therefore, the capacity limits of the heat pump are rarely reached until temperatures fall below 40°*F*. In this case, the heat pump capacity would impact performance when outdoor temperatures fell below 35°*F*, which would have a marginal impact on overall performance.

In the climates in this study, there are few hours in these lower temperature bins so the capacity reduction would be of little concern. In colder climates, a similar sizing strategy would result in a significant reduction in capacity when even greater variations in charge and airflow are present. As can be seen in Figure 52, there is little change in capacity within the range of these tests. However, as the charge level falls below 70% of manufacturer's specifications, a more serious impact can be observed. As discussed above, the greatest effects were often seen at high temperatures, where the heating capacity limits of the heat pump are even less problematic. The capacity trends at lower, more critical temperatures, level out with respect to charge and become more tightly grouped with respect to airflow.

The effect of variations in charge, airflow rate and metering device on the heat pump efficiency can be seen in the coefficient of performance (COP), as illustrated in Figure 53 and Figure 54. The trends are similar to those seen in the capacity plots

(Figure 51 and Figure 52) with minimal effects from refrigerant charge level, but with the effect of low flow even more pronounced than any of the capacity effects.

Figure 55 and Figure 56 give a different perspective by plotting the COP versus the airflow rate with the points labeled by charge level. Here it is clear that the effect of low airflow rate on the COP increases with outdoor temperature. Again, the difference between the metering devices is difficult to discern in all four of the COP plots.

It is important to note that all of the testing was performed on one heat pump and it is unclear how greatly the results might vary with a different heat pump model. In particular, it is expected that a heat pump without an accumulator would perform very differently, especially with respect to variations in charge level. Additional testing is needed to identify the range of performance results possible across the range of heat pumps available. It should also be noted that this heat pump used R22 refrigerant. These effects could be different for an alternative refrigerant, such as R410A.

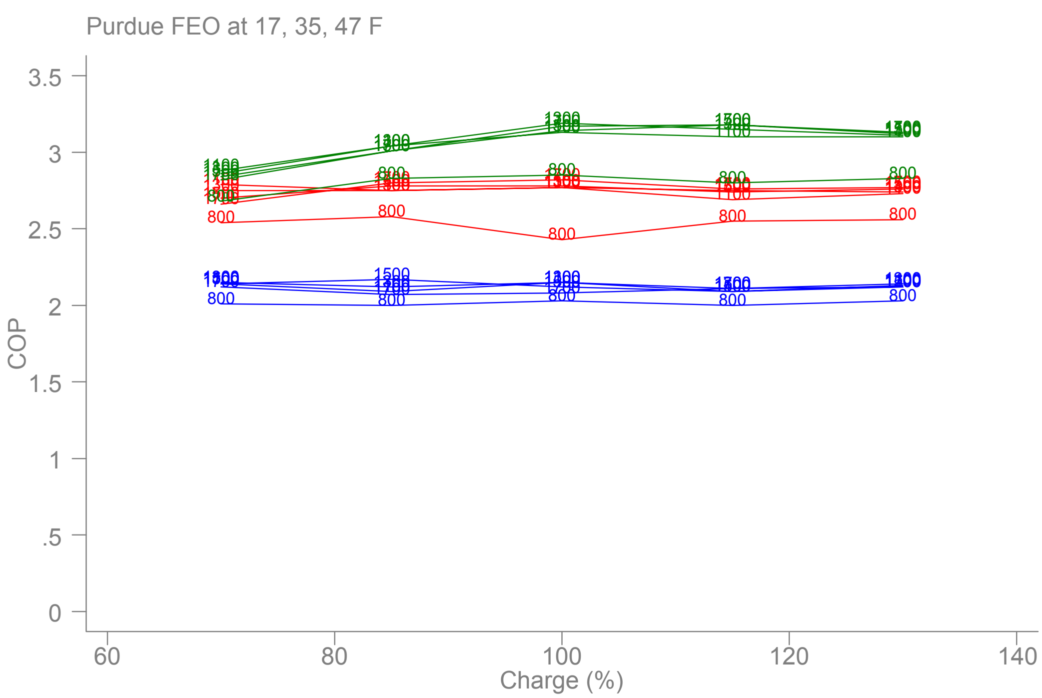

Figure 53. COP (fixed metering) versus Charge at three outdoor temperatures
17F(blue/lower group), 35F(red/central group), 47F(green/upper group). Points labeled by nominal airflow rate (cfm).

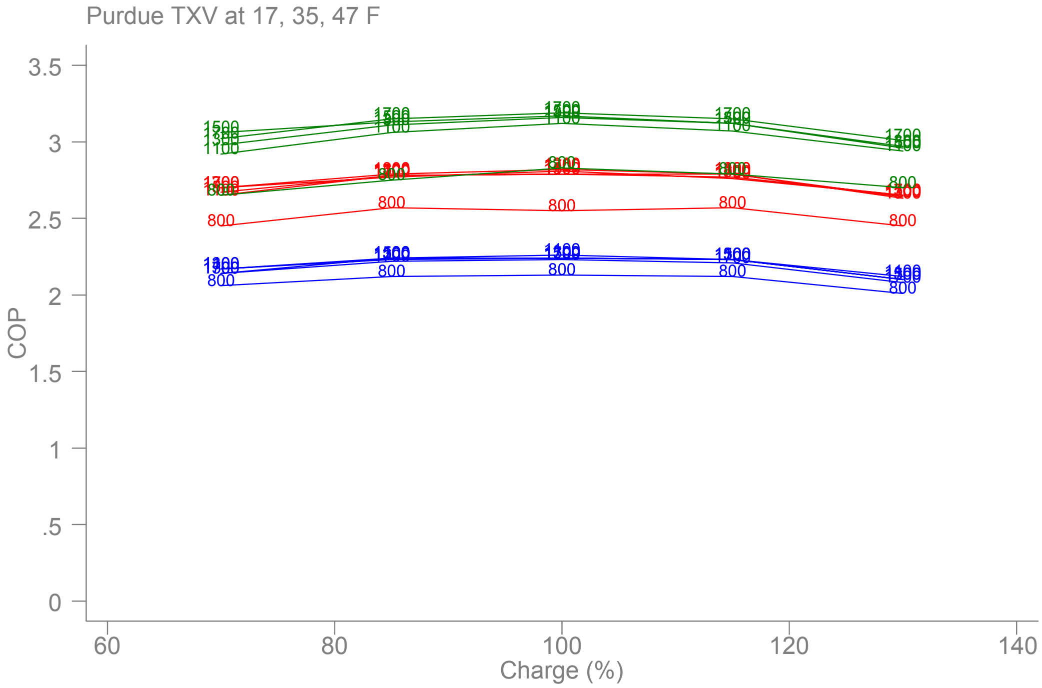

Figure 54. COP (TXV) versus Charge at three outdoor temperatures
17F(blue), 35F(red) 47F(green). Points labeled by nominal airflow rate (cfm).

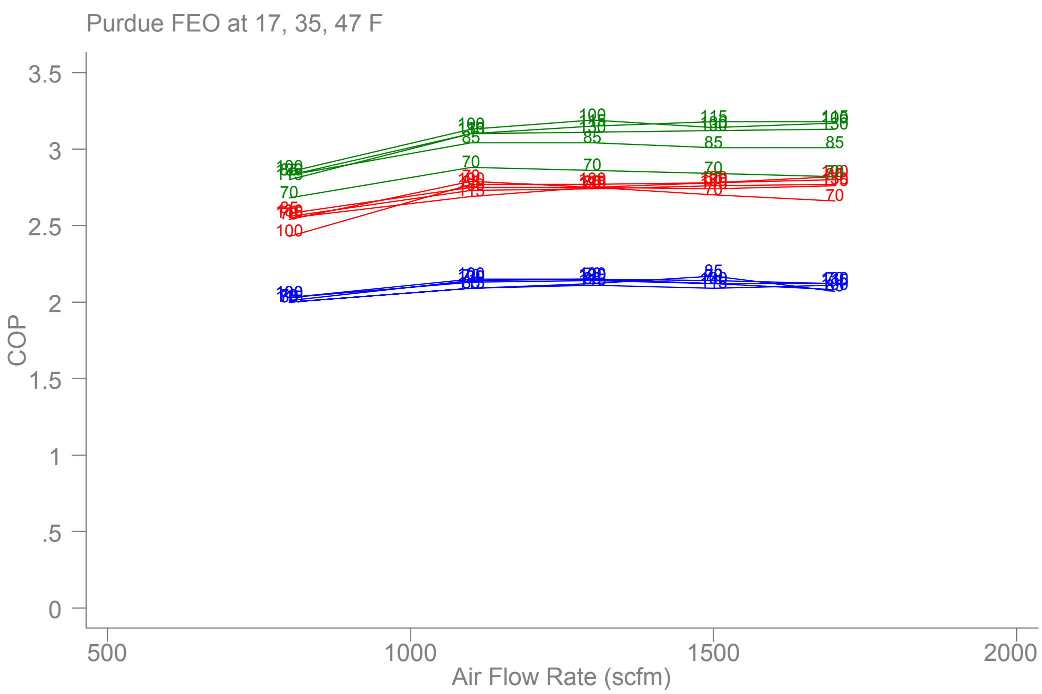

Figure 55. COP (fixed metering) vs Airflow Rate at three outdoor temperatures
17F(blue/lower group), 35F(red/central group) 47F(green/upper group). Points labeled by charge.

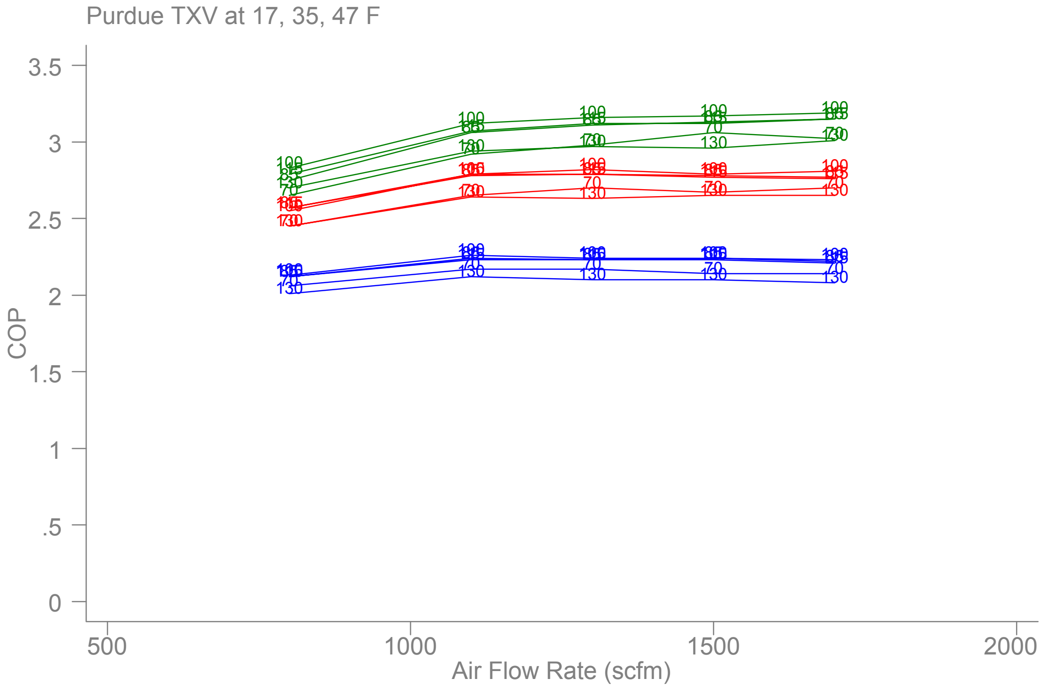

Figure 56. COP (TXV) versus Airflow Rate at three outdoor temperatures
17F(blue/lower group), 35F(red/central group), 47F(green/upper group). Points labeled by charge.

6.3. Results: Heating Seasonal Performance Factor

The test data generated by Purdue University was also used to calculate the heating seasonal performance factor (HSPF) for multiple climate zones. This number represents an efficiency measure designed to reflect a specific climate.

6.3.1. Calculation Method

Standard 210/240, published by ARI, defines the test methods used to measure the HSPF of a heat pump, as well as the required equations. The variables required for the calculation are flow rates, capacities and electrical power consumption from steady-state tests at 17°*F* and 47°*F*, a cyclic test at 47°*F*, and a defrost test at 35°*F*. For the tested heat pump, all of the steady-state values are available from the Carrier catalog and also from the Purdue testing. However, the values for the defrost penalty and the cycling penalty (C_d), derived from the defrost and cycling tests, are not available from the manufacturer.

Heat pump manufacturers are not required to publish results of the defrost or cyclic tests. ARI Standard 210/240 includes equations for estimating the defrost capacity and power consumption using steady-state values as well as a default C_d which may be used by manufacturers for the heating mode. It is possible to calculate these variables for comparison with the default values and a previously calculated C_d.

In the past, instead of using the default value (which we suspected to be outdated), Ecotope used a C_d of 0.14 for Carrier's YKC heat pump based on back-calculations using the catalog cooling data. The back-calculation is only possible for cooling because Carrier does not publish the values necessary to do the same calculation on the heating side. Previously, this value was used for both heating and cooling. The Purdue testing allows us to resolve the issue of the heating C_d value for the Carrier YKC heat pump. The results suggest that the C_d is considerable higher (larger cycling penalty) than previous modeling assumptions, but similar to the default value for the fixed orifice, and considerably lower than the default for TXV. Table 30 summarizes the results of the various methods of estimating the C_d.

Table 30. C_d: From Test Data, ARI Default and Back-Calculated

	Over All Variables	Fixed Metering	With TXV
Mean C_d (Lab) Data	0.188	0.225	0.152
C_d at Rating Point (Lab) 1300 cfm 100% charge	-	0.233	0.130
ARI Default C_d	0.25	0.25	0.25
Back-Calc'd Cooling C_d	-	0.14	-

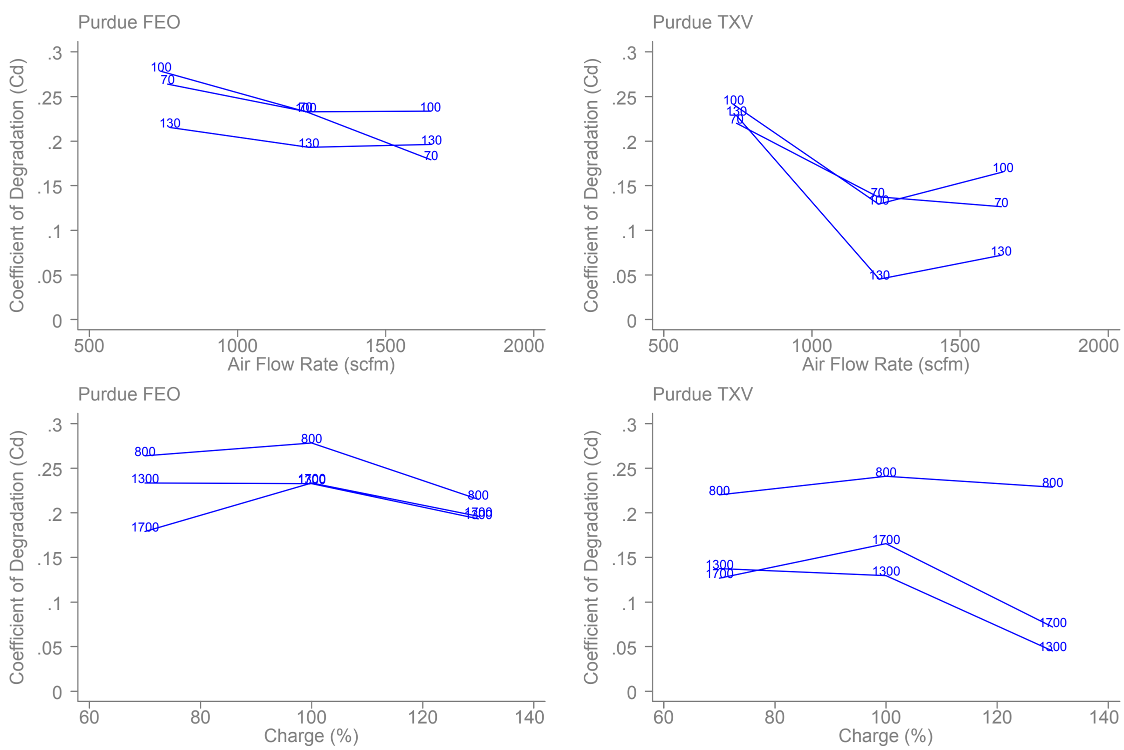

Figure 57. C_d versus Airflow Rate (row 1) and Charge (row 2) with a TXV (col. 1) and fixed metering (col. 2).
Note: Points labeled by charge (%) or airflow rate (cfm).

Figure 57 illustrates the effect of airflow rate and charge on the fixed metering heat pump and on the TXV heat pump. Here the effects of the metering device are more evident. With a TXV, the C_d shows more sensitivity to changes in the flow rate and charge levels. The TXV heat pump C_d is also considerably lower than that of the fixed orifice heat pump except at the lowest airflow rates. Much of the performance improvement in heat pumps using the TXV expansion device can be traced to this change in C_d.

Figure 58 illustrates the stability of the capacity defrost multipliers across charge, flow rate and metering device. Variation in charge or airflow has little effect on this factor.

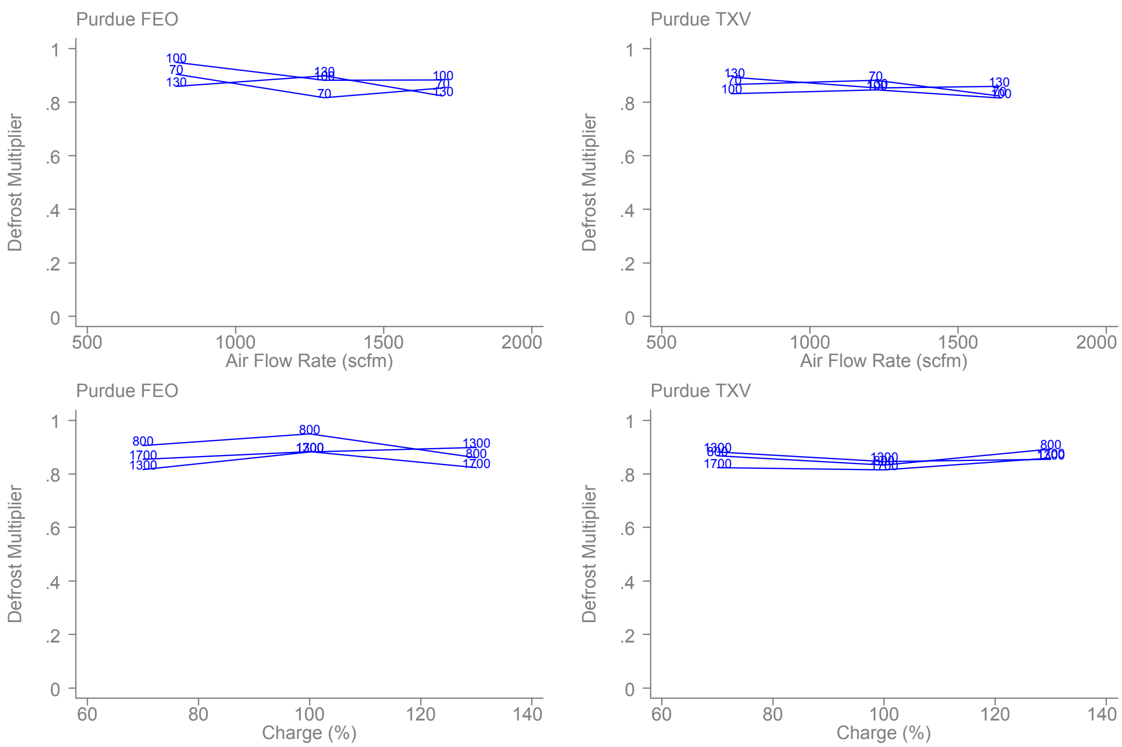

Figure 58. Capacity Defrost Multiplier versus Airflow Rate (row 1) and Charge (row 2) with a TXV (col. 1) and fixed metering (col. 2).
Note: Points labeled by charge or airflow rate (cfm).

The defrost degradation factors, or defrost multipliers, also necessary for the calculation of the HSPF, are largely stable at values just below the modeled and manufacturer's assumptions. Table 31 shows the ARI default defrost multipliers, the mean values and the values at the rating point (1300 cfm, 100% charge) calculated from the laboratory data.

Table 31. Defrost Multipliers: From Lab Data and ARI Default

	Over All Variables	Fixed Metering	With TXV
Mean Capacity Defrost Mult. (Lab)	0.864	0.875	0.852
Rated Capacity Defrost Mult. (Lab) 1300 cfm 100%charge	-	0.882	0.846
ARI Capacity Defrost Mult.	0.9	0.9	0.9
Mean Power Defrost Mult. (Lab)	0.968	0.970	0.967
Rated Power Defrost Mult. (Lab) 1300 cfm 100%charge	-	0.974	0.966
ARI Power Defrost Mult.	0.985	0.985	0.985

6.3.2. HSPF Comparison

The HSPF values of ARI standard climate zones 4, 5 and 6 are shown in Figure 59, representing the published rating zone and two northwest zones respectively. In most literature, the values published are for climate zone 4, which roughly corresponds to Medford, OR or Boise, ID. Zone 5 approximates the colder northwest climates of Spokane and Missoula. Zone 6 corresponds to the marine climates west of the Cascade Mountains (Seattle and Portland).

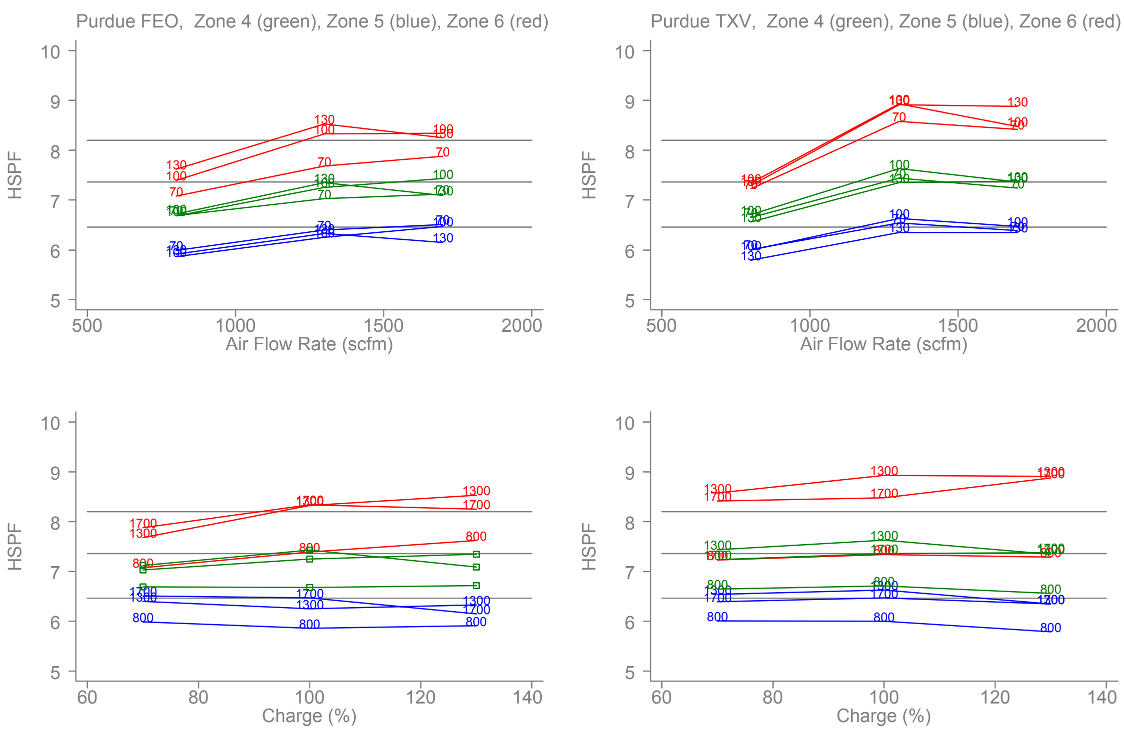

Figure 59. HSPF versus Airflow Rate (row 1) and Charge (row 2) with a TXV (col. 1) and fixed metering (col. 2).
Note: Points labeled by charge or airflow rate (cfm). Zone 4: green/central group, Zone 5: blue/lower group, Zone 6: red/upper group

The reference lines at 6.46 (zone 5), 7.36 (zone 4) and 8.20 (zone 6) represent the HSPF calculated for the YKC heat pump using catalog data combined with the default defrost multipliers of 0.9 for capacity and 0.985 for power. The C_d used (0.233) was calculated from the laboratory test data for the case most closely matching the catalog rating point: 1300 cfm nominal airflow, 100% charge and fixed metering.

The HSPF values generated using the steady state catalog data and the various options available for the C_d and the defrost multipliers are shown in Table 32. When varying the C_d or defrost multipliers in the HSPF calculations, the other variable was held at the value based on the laboratory test data. All of the variables are for the fixed metering heat pump.

Table 32. Comparison of HSPF results using Carrier YKC Catalog Data with Various C_d and Defrost Multipliers

	C_d	Defrost Cap. Mult.	Defrost Power Mult.	HSPF: Vary C_d			HSPF: Vary Defrost		
Zone	-	-	-	4	5	6	4	5	6
ARI Default	0.25	0.9	0.985	7.29	6.41	8.05	7.36	6.46	8.20
Lab. Data	0.23	0.882	0.974	7.34	6.45	8.17	7.34	6.45	8.17
Back Calc'd	0.14	-	-	7.63	6.66	8.81	-	-	-

Comparing rows within each climate zone shows that the HSPF is not sensitive to defrost multipliers within the degree of variability seen here. However, HSPF is sensitive to the degree of variability in the C_d.

6.4. Findings

The laboratory tests should not be looked at without the context provided by the rest of the study. The major implications are reviewed in Section 7.0. Nevertheless, there are general implications regarding the results of the lab tests on this heat pump:

1. The impacts of charge levels within the range tested here are minimal and probably do not materially affect the performance or capacity of the heat pump in heating mode.

2. The performance of the heat pump with variations in airflow is similarly minimal, except in the case where airflow is allowed to fall below about 300 CFM/ton. This may be a more significant result since about 30% of the heat pumps reviewed in the field had airflow set to comparable levels.

3. Cycling tests suggested that the modeling assumption for C_d used in this region was set too low for the FEO device and too high for the TXV device.

4. Use of a TXV has very little impact on charge and air flow, but a large effect on C_d. This results in better HSPF ratings, particularly in the milder climates where cycling in warmer temperature bins is a significant fraction of the heat pump operation.

5. Defrost tests revealed almost no impacts on overall performance from variation in defrost conditions due to charge and air flow.

7. Conclusions

The over-arching goal of this project was to utilize a variety of independent methodologies to answer several basic research questions about heat pumps and their performance in the heating climates in the Pacific Northwest. These units are a popular means of acquiring energy conservation in various utility programs throughout the region. Due to the nature of this technology, there are several complex and interconnected questions related not only to the performance of the equipment, but also to the specifications and installation practices that might enhance or compromise the savings that could be anticipated from such programs.

To probe these issues, this project really was composed of four interrelated but largely independent research efforts. Each effort was designed to address some portion of the uncertainty in the regional programs. The results are intended to inform the assumptions behind, and the process of delivering, these programs. Another goal is to evaluate potential savings from utility heat pump programs.

7.1. Program Savings

Over the study period, utilities in the region installed approximately 4,000 individual heat pumps in localities throughout Idaho, Montana, Washington and Oregon under the C&RD and ConAug programs. These heat pump installations were conducted using C&RD specifications and deemed savings estimates. The savings estimates were derived from the spreadsheets, calculators and other support materials integral to the C&RD process.

The energy savings associated with heat pumps installed under C&RD or ConAug are essentially deemed. That is, the utility is asked to estimate the vintage of the house, the type of heating system used prior to the heat pump installation, and the type of installation practice used. From this, an average savings is derived based on simulation research conducted previously for a variety of prototype houses. Therefore, the details about the particular house or individual heat pump are superceded by the deemed savings calculator, which is meant to describe an overall average condition. Thus, it is expected that substantial variation in savings from those estimates would occur in individual cases, but that, on the whole, this calculator delivers an accurate average savings over the range of heat pump installations and the range of service territories.

In addition to the C&RD heat pump equipment assumptions, there were numerous assumptions related to control strategy (indoor and outdoor thermostat, etc.) and duct distribution efficiency used to generate the original savings estimates. One of the most important of these was the estimated thermal load of the building. We need to compare this value as well since it is just as important as the controls, etc. This study was designed to allow utilities and the region to answer two important questions regarding these calculated savings:

1. Did they accurately forecast the savings observed (RTF 2003)?

2. Did revisions to these calculations made with more detailed technical analyses after these heat pumps were installed (RTF 2005) provide any additional certainty about the deemed savings calculations?

The billing analysis was conducted on about 725 C&RD heat pump installations and about 300 non-C&RD installations done under the Clark County PUD heat pump program. While this represented only a fraction of the participants in the C&RD program, the billing analysis suggested that not only were the results statistically significant, but the imputed size of the savings estimated in this way were consistent with the deemed savings derived from the RTF spreadsheet.

The overall realization rate calculated from the billing analysis was about 70% when only the C&RD cases were included. This realization rate did not include cases where a heat pump was in the base case. In those cases, the RTF spreadsheet includes the assumption that the heat pump is functioning properly and meets current federal equipment efficiency standards. It was quite apparent that the base case heat pump was probably not functioning for at least part of the time. This resulted in an inflated estimate of the realization rate. When the same results were compared to the savings calculations introduced to the program in 2005, the realization rate increased to more than 80 percent.

Because of the final sample size, a more detailed review of the program components by house type, house age, and system type, could not be reliably parsed. Thus, individual realization rates were quite variable in these categories and savings estimates were often not statistically significant.

7.2. Heat Pump Commissioning

Implicit in the C&RD guidelines and savings calculators is the assumption that the installation practice should include a commissioning step that ensures the airflow across the heat pump indoor coil and refrigerant charge levels are at or near the manufacturer's specifications. It was originally asserted that a 10% savings could be expected when commissioning was applied to individual existing heat pumps. This was based on work done by Proctor Engineering in California and lab studies used in estimating the impact of these commissioning issues on central air conditioners. It was further asserted that the inclusion of this commissioning step in the set-up of new heat pumps would ensure the system behaved more like manufacturer's predictions. The implication was that the heat pump would under-perform these predictions in the absence of such commissioning.

The assertion of commissioning-related energy savings sparked a heated debate. The main questions focused on the incidence of problems (and their severity) and the application of cooling season results to the heating-dominated climates of the Pacific Northwest.

7.2.1. Billing analysis

To address these regional questions, two independent efforts were mounted within this study. The first was a review of a commissioning program run by EWEB on existing heat pumps at about the same time the heat pump systems for the C&RD billing analysis were being installed. Because the commissioning effect was thought to be relatively small (approximately 10% of the total heating load), an alternative billing analysis methodology was employed that used a simulation applied to each home to attempt to establish changes in consumption on the heating and cooling components of the total electricity use.

These approaches successfully highlighted relatively small savings that would have been difficult to identify using conventional billing analysis. Overall, the CheckMe!® protocol used by the EWEB program (written by Proctor Engineering) showed savings of about 60% of the total predicted by the C&RD spreadsheet calculators. Unlike the C&RD/ConAug billing analysis results, however, the distribution of the savings for the EWEB program was very narrow. In fact, no significant savings were found in 85% of the cases. In those 15% of cases where significant savings occurred, it may be that the benefit came from identifying units that exhibited severe maintenance or set up problems through conducting the commissioning process. Averaged through the entire sample, these results were fairly encouraging. However, when the results of units that received charge and airflow adjustments were reviewed individually, these factors did not result in any statistically significant savings by themselves.

7.2.2. Bench Testing (Purdue University)

Simultaneous with this billing analysis effort, laboratory tests were conducted by Purdue University with the specific goal of reviewing set-up and commissioning on heat pump performance in heating mode. Purdue was contracted to review a specific heat pump, which was selected to represent a base case for the Pacific Northwest region. This review was intended to establish the impacts of refrigerant charge and airflow on heat pump capacity ("tons") and the coefficient of performance ("COP"). While the Purdue research was aimed at more than this single issue, the initial results of these tests over a wide range of temperatures, charges and airflows suggested that issues of charge or incorrect charge do not have any significant impact on heat pump performance in the heating mode.

The impact of airflow only showed significant impacts when it was allowed to fall below about 75% of the manufacturer's recommended airflow rate (usually 400 CFM/ton). This suggests that commissioning the charge and airflow would probably not result in any significant savings under ordinary operating conditions in the heating mode.

7.2.3. Field Review

While the lab and billing analyses were being conducted, a field study was also undertaken to review approximately 160 heat pump installations throughout the region that were installed without C&RD intervention. In order to determine the impacts of commissioning and airflow, the field review process took direct measurements of airflow and charge for about 125 of these sites. The overall results suggest that the penalties associated with incorrect charge, even from heat pumps that have been in the field for a long time, are rare. Only about 10% of the cases were found to be undercharged.

About 30% of cases were evaluated as overcharged, but only about 1/3 of these cases (or about 10% of the overall sample) were designated as very overcharged. In most cases of overcharge, the system was equipped with a suction line accumulator, which is a fail safe device to store extra charge. Bench testing at Purdue confirmed very little effect on system capacity and efficiency from overcharge. Where severe undercharge was noted, it had caused comfort and related performance complaints and therefore was easily identified and repaired.

It is quite apparent that current practice is to sometimes lower system airflow much more than was anticipated by the manufacturer. This does have the effect of increasing supply air delivery temperature; however, in about 25% of the cases reviewed, the airflow is sufficiently low (less than 75% of manufacturer's recommendations) that it would suggest the need for correction.

Reviewing large numbers of heat pumps as part of a commissioning program does seem to have the beneficial effect of identifying those units in serious need of maintenance or those nearing the end of their functional life. The savings resulting from this aspect of the commissioning program were significant and may actually be sufficient to justify the cost of the program.

One additional outcome of this review is the suggestion that, in the design of commissioning and installation practices, the review of sizing, charge and airflow may provide appreciable benefits beyond the obvious one of ensuring that the airflow across the heat pump coil is within 80 to 85 percent of the manufacturer's specified range. This is a relatively straight-forward installation practice. Although it might require some specialized equipment, the time and documentation required are not as significant as that required by the CheckMe!® or ACRX protocols. Furthermore, especially in the case of the ACRX (where airflow is inferred rather than directly measured), these tests suggest that attention to airflow and charge in the commissioning of new installations is unlikely to produce any substantial savings.

7.3. Heat Pump Controls

Throughout the C&RD specifications, there are efforts to describe particular strategies for sizing the heat pump and for controlling the heat pump once installed. This study attempted to establish the nature of the 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.

As with other aspects of this study, more than one effort was undertaken to identify the procedures currently in use:

1. The technical question was addressed by the field review of 160 heat pumps installed outside of the C&RD specifications (although, in some cases, other non-C&RD utility programs impacted the heat pumps reviewed). Because of this feature of the sample, there is some bias toward better control systems.

2. For some issues, the input was gathered from the installer interviews. In addition, the responses from the large number of installers were used to confirm or inform the findings from the field sample.

This effort was focused on describing the particular installation practices and control strategies actually used. The field review of the heat pumps focused on the nature of the controls managing the electric backup elements of the equipment and on the nature of the thermostat setback and recovery algorithms.

Outdoor "lockout" thermostats were installed in about 40% of the cases reviewed. However, most of those cases were in the Clark County PUD service territory. That utility was operating a heat pump program that influenced virtually all of the heat pumps reviewed. While that program did not use C&RD protocols, it did require many of the same commissioning and installation practices, including the use of an outdoor thermostat to limit the amount of second- and/or first-stage electric element operation in relatively warmer outdoor temperatures. More importantly, however, the review suggested that a retrofit of outdoor thermostats would be fairly easy and that the control wires needed to enable an outdoor thermostat were already in place in about 65% of the heat pumps reviewed that did not already have an outdoor thermostat.

The indoor thermostat was another matter. Occupants complained about the thermostat in about 15% of the cases. Almost always, the complaints involved the complexity of the thermostat and difficulty setting it properly. Setbacks were routinely used in spite of the fact that this often defeated the adaptive recovery cycle. The heat pump operation "in the middle of the night" to begin warming up the house was considered improper operation by most occupants; thus, adaptive recovery became somewhat less practical. Presumably, in these cases, the use of an outdoor thermostat to defeat the elements as part of morning warm-up would be problematic. In these cases, occupants are usually employing "emergency" electric backup heat in the warm-up cycle even during relatively warm outdoor temperatures.

In the calculation of base case practices in the C&RD spreadsheet, potential control packages are weighted. Depending on the climate, these packages result in a 30% to 40% reduction in the nominal COP of the base case heat pump. While many of the control issues remain somewhat uncertain, the major control decisions in current practice are well represented in either the interview results or the field results. These control decisions include:

1. First stage heating: The use of electric elements in the first stage of the heating call was noted in about a third of the field cases. This effect would be reduced in cases where the outdoor thermostat was present and set to override the element operation above the temperature setting. In about 10 percent of the cases where first stage elements were used, such an override was possible. In only two cases was the setting such that a significant change in the element operation would be expected. In the current modeling assumptions, the first stage element is on during the entire duty cycle. This was rarely seen in the field sample. In the current RTF spreadsheet, 10 percent of the cases are thought to use first stage heating. This is probably conservative and a substantial adjustment would be appropriate to account for the reduced operation of the elements in the observed control logic.

2. Low ambient cutout (compressor): This installation strategy was thought to be common, especially in colder Northwest climates. No cases of this control strategy were observed in the field review, which included Bend (a colder climate). From the interviews, it appears that this strategy is used in the Spokane market but no other installers thought it was used in their locality. About 25 percent of the interview respondents said they used this strategy. Given the results of the field survey, we would assert that this could not affect more the 10% of the new heat pump installations in the region. In the current modeling assumptions, this was thought to occur 25% of the time. An adjustment in this assumption is therefore appropriate.

3. Comfort assist: This strategy was based on the idea that electric elements are activated in the first stage at low temperatures (less than 32°*F*). While there are many installations where this is possible, only about 3% of the field sample employed this strategy and almost no installers had used that approach. The

nature of this control logic may more closely mimic the operation of heat pumps controlled to allow elements to operate during a first stage heating call if an ODT prevents the elements from operating above 40°*F*. The current modeling assumption is that this operation occurs 10 percent of the time. It may be appropriate to increase this to capture the impact of the observed control strategies.

4. Thermostat setback: The speed and responsiveness of the electric elements changes the overall electric usage of the heat pump over the heating season. This is particularly true as the unit is turned on after a night setback. There was some capability to do "adaptive recovery" but most homeowners found this option inappropriate and employed either the "emergency heat" option (to decrease warm-up time) or allowed the second stage to bring in the auxiliary elements. This suggests that the calculator (which currently does not take adaptive recovery into account) probably represents the current operating practice. Current modeling assumptions assume a 6°*F* setback during heating operations. The elements are forced on only if the heat pump is not capable of meeting the load in the first cycle. This occurs when temperatures fall below 45°*F*. In an adaptive recovery scheme, the elements would have been off at a temperature this high. Our review of current thermostat practice closely mirrors current modeling assumptions.

7.3.1. Heat Pump Sizing

A review of heat pump sizing issues associated with heat pump installation practices and specifications suggest that balance point sizing should still be the recommended method. This part of the installation practice and specification 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. 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.

This study addressed heat pump sizing issues in two ways:

1. The first was to attempt to assess the heating and capacity requirements of the individual homes in the field study. To do this, heat loss rates were estimated using the building audit, in addition to the detailed description of infiltration and duct efficiency for each site. Using these measures, a fairly precise estimate of capacity could be derived. Sizing for this capacity was then calculated at some pre-selected balance point temperature. For this purpose, a somewhat simplified method was used to try to determine, on average, what sizing practices were actually being employed. For the most part, the field

sizing results were actually lower than the nominal sizing estimate that could be derived from the characteristics of the building. They did not appear to be consistent with either the ASHRAE sizing methods or with the RTF-specified sizing methodology in either the older specifications or in any of the current revisions.

In most cases, it is apparent that the heat pump is undersized relative to the heating load. It is clear that there is some effort on the part of the mechanical contractor to size the unit so as to minimize the incremental first cost associated with heat pump capacity, at least in the areas studied, rather than minimize energy consumption or minimize electric element operation. Presumably, installers assume that electric elements make up for any capacity shortfall when outdoor temperatures are quite cold. Leaving out the performance implications of these sizing criteria, this does in fact reduce the initial cost of the heat pump. Installers were asked about this. They argued, for the most part, that they were installing these units in accordance with estimated heating and cooling loads, but that a large part of the decision was to minimize first cost for the potential customer.

2. To further understand the impact of this undersizing, the Purdue University laboratory was asked to measure actual cycling degradation in the heat pump tests. This produced a surprising result: The C_d was about 20% lower than the standard assumptions in cases where a fixed orifice was employed as part of the refrigerator and expansion device. It was, however, about 40% higher than the assumptions used in modeling the base case heat pump in the RTF spreadsheets. In cases where a TXV was employed, however, the impact of cycling degradation was reduced dramatically (more than 65%). In fact, most of the apparent improvement in performance between the fixed orifice and TXV units came not from improvements in charge and airflow behavior but rather from an improvement in cycling losses. Overall, this combination may suggest that the modeling assumption could characterize average installations. For new high performance heat pumps, however, the RTF modeling assumption is probably too conservative.

A specification that limits the <u>minimum</u> size of the heat pump in relatively cold conditions would tend to maximize savings. (That is, this specification would suggest larger heat pumps be installed). However, the first cost of the heat pump tonnage must be considered a significant factor in the cost-effectiveness of the installation. 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).

7.4. Installation and Design Practice

Direct interviews with suppliers and installers were conducted to provide better insights into the nature of heat pump installation and design practices, and to assess the potential changes such practices might undergo as new federal standards are put into place in 2006. Overall, the installers were somewhat apprehensive about the required changes, but this apprehension mostly centered on the cost of the new equipment rather than on the improved efficiency or technical complexity of these high efficiency units.

By and large, the change of prevailing refrigerant from R-22 to R-410A was embraced by most contractors who had experience installing R-410A systems. In general, the installers interviewed seemed ready to accept improvements in the heat pumps mandated by new standards or by utility and state incentive programs.

For the most part, installers seemed to utilize a reasonable approach to the selection and installation of heat pumps. This approach was intended to maximize the performance of the heat pump and to improve the chances of making a heat pump sale by selecting options that result in a more cost-effective (less expensive) initial installation.

Our results suggest that installation guidelines from the utilities within and outside of the C&RD program would interact well with this strategy. For the most part, contractors are familiar with most aspects of the installation requirements and are reasonably familiar with the reasons for the specifications. By adding additional specifications and backing those up with incentives, it is likely that contractors would respond well to the added requirements. The contractors were contacted throughout the region rather than in the four areas reviewed for the rest of this study, and relatively few of them were experienced with the utility programs. However, they did believe that the utility programs would improve both the market for, and the quality of, heat pump installations.

7.5. Overall Conclusions

Heat pumps, for the most part, are behaving at or near the expected efficiency level, at least for C&RD/ConAug program participants. The performance of this equipment shows a consistent reduction in space conditioning energy even in areas where substantial amounts of wood heat also offset the space heating load. It should also be pointed out that most of the heat pumps reviewed in the field in this study did not include many of the advanced compressor and controls now available or under development. Therefore, the nature of these conclusions is clearly dominated by existing equipment and technologies using single-stage compressors. Future programs will have to address these newer technologies as the industry moves to higher efficiency equipment and alternative refrigerants in response to federal mandates. If anything, these moves would make the installation and equipment specifications used in the utility program even more desirable and more likely to deliver high quality and efficient equipment.

Installation practices intended to circumvent many of the potential problems in heat pump performance (low airflow, first stage resistance heating) are often employed by installers. This is due not so much to actual occupant complaints but to the fear of occupant complaints. It is apparent that, particularly in relatively recent installations, occupants are reasonably pleased with heat pump technology and believe it can deliver both energy savings and a high level of satisfaction.

This work suggests the need for substantial improvements in understanding these newer technologies (and refrigerants) and for improvements in the understanding of thermostat settings and behaviors that can defeat the efficiency of high performance heat pumps. Our overall impression, however, is that this technology has been embraced by many markets and could be effective in the majority of the markets in the Northwest, including those areas where this technology has not traditionally been installed (e.g. Boise).

Additional research should focus on these more problematic areas; notably, the interaction between compressor heating and auxiliary heating, so that both occupant comfort and savings can be delivered. The new generation of efficient equipment could invalidate many of these conclusions, as the design and control of heat pumps is modified to meet new federal standards. We strongly urge that the detailed testing of this type of equipment be conducted in a laboratory environment or as installed in utility programs.

It is notable that this review of heat pumps throughout the region shows that there are often difficulties in both performance and operation that stem from the complexity of this equipment. Detailed specification of installation practices as well as detailed quality control is necessary to ensure system performance. Certainly, better information and assessment would be desirable, both from the perspective of convincing installers to push high efficiency equipment and from the perspective of designing these installation practices and quality control procedures to be effective at delivering this product to the market.

8. Bibliography and Suggested Reading

Air-Conditioning and Refrigeration Institute (ARI). 2003. *ARI Standard 210/240-2003: Unitary Air-Conditioning and Air-Source Heat Pump Equipment*. Arlington, VA.: Air Conditioning and Refrigeration Institute.

Air Conditioning, Heating, and Refrigeration News. July 10, 2005.

Baylon, D., S. Borrelli, and M. Kennedy. "Baseline Characteristics of the Residential Sector in Idaho, Montana, Oregon and Washington. For the Northwest Energy Efficiency Alliance. Portland, OR. 2001.

Baxter, V. and J. Moyers. November, 1984. "Air-Source Heat Pump: Field Measurement of Cycling, Frosting, and Defrost Losses, 1981-83", Report ORNL/CON-150, Oak Ridge National Laboratory, Braun, J., and H. Li. May, 2005. "Heat Pump Testing for Optimal Heating Mode Performance: Progress Reports and Final Report". Prepared for Ecotope, Inc. by Purdue University, Lafayette, IN.

Bullock, Charles, D. Schuster, and R. Lambert. July, 16, 2004. "Heat Pump Heating Performance Comparison Between Short-tube Orifice and TXV". Prepared by Carrier Residential Engineering in response to questions from Bob Davis, Ecotope.

Domingorena, A., and S. Ball. January, 1980. "Performance Evaluation of a Selected Three-Ton Air-to-Air Heat Pump in the Heating Mode", Report ORNL/CON-34, Oak Ridge National Laboratory.

Farzad, M. and D. L. O' Neal. January, 1993. Influence of the Expansion Device on Air-Conditioner System Performance Under a Range of Charging Conditions. ASHRAE Paper 3622, ASHRAE Transactions.

Fels, M. *The Princeton Scorekeeping Method: An Introduction*. 1984. Princeton University, Center for Energy and Engineering, Princeton, NJ, PU/CEES 163.

Fels, M. *Special Issue Devoted to Measuring Energy Savings: "The Scorekeeping Approach"*, Energy and Building, 9(1-2), Feb/Mar 1986.

Francisco, P., D. Baylon, B. Davis, and L. Palmiter. Winter, 2004. "Heat Pump System Performance in Northern Climates". ASHRAE Transactions. Atlanta, GA.

Francisco, P. and L. Palmiter. May 2003. Revised: Improvements to ASHRAE Standard 152P. Prepared for the US Dept. of Energy under Subcontract No. 324269-AU1. Prepare by Ecotope, Inc. Seattle, WA.

Kao, James, W. Mulroy, and D. Didion. 1987. "Field Performance of Three Residential Heat Pumps in the Heating Mode", Report NBSIR 87-3528, U.S. Department of Commerce, National Bureau of Standards.

Levins, W., K. Rice and V. Baxter. 1996. *Modeled and Measured Effects of Compressor Downsizing in an Existing Air Conditioner/Heat Pump in Cooling Mode*. Prepared for ASHRAE by Oak Ridge National Laboratory, Oak Ridge, TN.

Miller, R. and H. Jaster. November, 1985. "Performance of Air-Source Heat Pumps, " Report EPRI EM-4226, Electric Power Research Institute.

Palmiter, L. and P. Francisco. July 1997. *Development of a Practical Method for Estimating the Thermal Efficiency of Residential Forced-Air Distribution Systems.* Prepared for the Electric Power Research Institute. Prepared by Ecotope, Inc. Seattle, WA.

Proctor, J., B. Davids, F. Jablonski, and G. Peterson. 1990. *Pacific Gas and Electric Heat Pump Efficiency and Super Weatherization Pilot Project.* Prepared by Building Resources Management Corporation for PG&E.

Proctor, J., M. Blasnik, and T. Downey. 1995. *Southern California Edison Coachella Valley Duct and HVAC Retrofit Efficiency Improvement Pilot Project.* Prepared by Proctor Engineering Group, San Rafael, CA.

Proctor, J. November 4, 2004. "Energy Savings from Verified Refrigerant Charge and Airflow on Heat Pumps in the Heating Mode". Presented to the Northwest Power and Conservation Council Regional Technical Forum. Prepared by Proctor Engineering Group, San Rafael, CA.

Technical Appendix A

C&RD/ConAug Statistical Results and Tables

Table 33. Summary of NAC Consumption and Savings by Region, Treated Cases

Participant Group	Consumption Variable, kWh/yr	Mean	Standard Deviation	n	90% CL	t-test	Significance P(T<=t) two-tail
Cooling Zone 1 Treated Cases	NAC, Pre	24,583	11,110	704	689		
	NAC, Post	20,109	9,480	704	588		
	NAC, Saved	4,474	5,861	704	364	20.25	0.00
	Baseload, Pre	14,250	7,265	704	451		
	Baseload, Post	13,616	6,676	704	414		
	Baseload, Saved	635	3,616	704	224	4.66	0.00
	Space Heat, Pre	10,314	5,980	704	371		
	Space Heat, Post	6,455	4,469	704	277		
	Space Heat Saved	3,859	4,700	704	292	21.79	0.00
	Cooling, Pre	19	178	704	11		
	Cooling, Post	40	224	704	14		
	Cooling Saved	-21	171	704	11	-3.35	0.00
Cooling Zone 1 Non C&RD Treated Cases	NAC, Pre	25,379	8,936	186	1,078		
	NAC, Post	21,528	7,247	186	875		
	NAC, Saved	3,851	5,185	186	626	10.13	0.00
	Baseload, Pre	15,063	6,226	186	751		
	Baseload, Post	14,981	5,361	186	647		
	Baseload, Saved	82	3,579	186	432	0.31	0.38
	Space Heat, Pre	10,278	5,016	186	605		
	Space Heat, Post	6,475	3,648	186	440		
	Space Heat Saved	3,803	3,953	186	477	13.12	0.00
	Cooling, Pre	39	277	186	33		
	Cooling, Post	76	300	186	36		
	Cooling Saved	-37	193	186	23	-2.64	0.00
Cooling Zone 3 Treated Cases	NAC, Pre	24,419	8,329	318	769		
	NAC, Post	20,624	6,726	318	621		
	NAC, Saved	3,795	5,316	318	491	12.73	0.00
	Baseload, Pre	13,884	7,054	318	651		
	Baseload, Post	13,521	5,810	318	536		
	Baseload, Saved	363	4,823	318	445	1.34	0.09
	Space Heat, Pre	9,418	4,646	318	429		
	Space Heat, Post	6,220	3,140	318	290		
	Space Heat Saved	3,197	3,692	318	341	15.44	0.00
	Cooling, Pre	1,117	1,174	318	108		
	Cooling, Post	883	795	318	73		
	Cooling Saved	234	824	318	76	5.07	0.00

Participant Group	Consumption Variable, kWh/yr	Mean	Standard Deviation	n	90% CL	t-test	Significance P(T<=t) two-tail
All C&RD Cases Treated Cases	NAC, Pre	24,343	10,601	836	603		
	NAC, Post	19,989	8,991	836	512		
	NAC, Saved	4,354	5,811	836	331	21.66	0.00
	Baseload, Pre	13,930	7,385	836	420		
	Baseload, Post	13,276	6,591	836	375		
	Baseload, Saved	654	4,119	836	234	4.59	0.00
	Space Heat, Pre	9,981	5,738	836	327		
	Space Heat, Post	6,361	4,198	836	239		
	Space Heat Saved	3,620	4,519	836	257	23.16	0.00
	Cooling, Pre	432	906	836	52		
	Cooling, Post	353	659	836	38		
	Cooling Saved	80	537	836	31	4.26	0.00
All Treated Cases	NAC, Pre	24,532	10,321	1,022	531		
	NAC, Post	20,269	8,717	1,022	449		
	NAC, Saved	4,263	5,703	1,022	294	23.89	0.00
	Baseload, Pre	14,136	7,198	1,022	371		
	Baseload, Post	13,586	6,417	1,022	330		
	Baseload, Saved	550	4,030	1,022	208	4.36	0.00
	Space Heat, Pre	10,035	5,613	1,022	289		
	Space Heat, Post	6,382	4,102	1,022	211		
	Space Heat Saved	3,653	4,420	1,022	228	26.42	0.00
	Cooling, Pre	360	842	1,022	43		
	Cooling, Post	302	619	1,022	32		
	Cooling Saved	58	495	1,022	25	3.75	0.00

Note: t-tests for the savings are those based on a difference of means test for the two means. The significance represents the likelihood that the observed difference could have occurred due to random variability. A low significance confirms that the observed change is significant and not due to random events.

Table 34. Summary of NAC Consumption by System Type, Treated Cases

System Type	Consumption Variable, kWh/yr	Mean	Standard Deviation	n	90% CL	t-test	Significance P(T<=t) two-tail
FAF w/CAC	NAC, Pre	24,672	7,595	236	814		
	NAC, Post	20,174	6,866	236	736		
	NAC, Saved	4,498	4,636	236	497	14.91	0.00
	Baseload, Pre	13,916	6,253	236	670		
	Baseload, Post	13,337	5,667	236	607		
	Baseload, Saved	578	3,964	236	425	2.24	0.01
	Space Heat, Pre	9,855	4,980	236	534		
	Space Heat, Post	6,124	3,418	236	366		
	Space Heat Saved	3,731	4,114	236	441	13.93	0.00
	Cooling, Pre	901	1,129	236	121		
	Cooling, Post	713	782	236	84		
	Cooling Saved	188	746	236	80	3.88	0.00
FAF w/oCAC	NAC, Pre	23,855	13,973	234	1,504		
	NAC, Post	19,077	12,559	234	1,351		
	NAC, Saved	4,778	6,245	234	672	13.05	0.00
	Baseload, Pre	13,245	8,518	234	917		
	Baseload, Post	12,415	8,194	234	882		
	Baseload, Saved	830	3,838	234	413	3.33	0.00
	Space Heat, Pre	10,545	7,076	234	761		
	Space Heat, Post	6,608	5,604	234	603		
	Space Heat Saved	3,937	4,609	234	496	14.49	0.00
	Cooling, Pre	65	440	234	47		
	Cooling, Post	57	323	234	35		
	Cooling Saved	8	231	234	25	-0.55	0.29
Heat Pump	NAC, Pre	25,993	10,629	99	1,758		
	NAC, Post	21,641	8,539	99	1,413		
	NAC, Saved	4,352	5,861	99	970	7.39	0.00
	Baseload, Pre	15,333	7,741	99	1,281		
	Baseload, Post	14,705	6,655	99	1,101		
	Baseload, Saved	628	3,983	99	659	1.57	0.06
	Space Heat, Pre	10,278	6,218	99	1,029		
	Space Heat, Post	6,633	4,031	99	667		
	Space Heat Saved	3,645	5,164	99	854	7.02	0.00
	Cooling, Pre	382	953	99	158		
	Cooling, Post	303	682	99	113		
	Cooling Saved	79	524	99	87	1.49	0.07

System Type	Consumption Variable, kWh/yr	Mean	Standard Deviation	n	90% CL	t-test	Significance P(T<=t) two-tail
Zonal	NAC, Pre	21,623	6,353	95	1,073		
	NAC, Post	18,771	5,356	95	905		
	NAC, Saved	2,852	5,258	95	888	5.29	0.00
	Baseload, Pre	13,008	4,632	95	782		
	Baseload, Post	12,532	4,572	95	772		
	Baseload, Saved	475	4,053	95	684	1.14	0.13
	Space Heat, Pre	8,456	4,001	95	676		
	Space Heat, Post	6,058	3,085	95	521		
	Space Heat Saved	2,398	4,009	95	677	5.83	0.00
	Cooling, Pre	159	344	95	58		
	Cooling, Post	180	387	95	65		
	Cooling Saved	-17	297	95	50	-0.68	0.25
All C&RD Cases	NAC, Pre	24,532	10,321	1,022	531		
	NAC, Post	20,269	8,717	1,022	449		
	NAC, Saved	4,263	5,703	1,022	294	23.89	0.00
	Baseload, Pre	14,136	7,198	1,022	371		
	Baseload, Post	13,586	6,417	1,022	330		
	Baseload, Saved	550	4,030	1,022	208	4.36	0.00
	Space Heat, Pre	10,035	5,613	1,022	289		
	Space Heat, Post	6,382	4,102	1,022	211		
	Space Heat Saved	3,653	4,420	1,022	228	26.42	0.00
	Cooling, Pre	360	842	1,022	43		
	Cooling, Post	302	619	1,022	32		
	Cooling Saved	58	495	1,022	25	3.75	0.00

Note: t-tests for the savings are those based on a difference of means test for the two means. The significance represents the likelihood that the observed difference could have occurred due to random variability. A low significance confirms that the observed change is significant and not due to random events.

Table 35. Disaggregation of Savings by End use and Climate Zone

Climate	Variable	NAC1	NAC2	Saved NAC	Base1	Base2	Saved Base	SH1	SH2	Saved SH	AC1	AC2	Saved AC
Zone1	Mean	24,583	20,109	4,474	14,250	13,616	635	10,314	6,455	3,859	19	40	-21
	S. D.	11,110	9,480	5,861	7,265	6,676	3,616	5,980	4,469	4,700	178	224	171
	n	704	704	704	704	704	704	704	704	704	704	704	704
	90% C. L.	689	588	364	451	414	224	371	277	292	11	14	11
Zone3	Mean	24,419	20,624	3,795	13,884	13,521	363*	9,418	6,220	3,197	1,117	883	234
	S. D.	8,329	6,726	5,316	7,054	5,810	4,823	4,646	3,140	3,692	1,174	795	824
	n	318	318	318	318	318	318	318	318	318	318	318	318
	90% C. L.	769	621	491	651	536	445	429	290	341	108	73	76

* The value for baseload savings in Zone 3 is not statistically different from zero at the 90% level. All other values, including negative cooling savings in Zone 1, are statistically significant. The fact that there are some savings allocated to baseload suggests that the disaggregation into enduses is not perfect. The baseload includes some heating and/or cooling. For this reason, NAC savings are a better estimate of the effective overall savings.

Table 36. Test of Difference Between Regions

Anova: Single Factor

SUMMARY

Groups	Count	Sum	Average	Variance
Coast	134	533,960	3,985	42,280,716
NW, Kitsap	302	1,540,286	5,100	38,679,381
Portland Area	82	359,186	4,380	20,836,215
Tricities	318	1,206,686	3,795	28,262,644
Non C&RD	186	716,278	3,851	26,886,286

ANOVA

Source of Variation	SS	df	MS	F	P-value	F crit
Between Groups	324,561,970	4	81,140,493	2.51	0.04	2.38
Within Groups	32,886,783,266	1,017	32,337,053			
Total	33,211,345,236	1,021				

Conclusion: Differences between regions are significant.

Table 37. Test of Difference Between System Types

Anova: Single Factor

SUMMARY

Groups	Count	Sum	Average	Variance
FAF w/CAC	236	1061423	4497	2148791
FAF w/oCAC	255	1279799	5018	37744363
Heat Pump	99	430809	4351	34347762
Zonal	130	341542	2627	30994697

ANOVA

Source of Variation	SS	df	MS	F	P-value	F crit
Between Groups	504556586	3	168185528	5.473394534	0.001009	2.617
Within Groups	22001125249	716	30727828			
Total	22505681836	719				

Conclusion: Differences between system types are significant.

Table 38. Difference Test, C&RD and non-C&RD Groups

Anova: Single Factor

SUMMARY

Groups	Count	Sum	Average	Variance
Non-C&RD	186	716,278	3,851	26,886,286
C&RD	836	3,640,118	4,354	33,771,076

ANOVA

Source of Variation	SS	df	MS	F	P-value	F crit
Between Groups	38,533,731	1	38,533,731	1.18	0.28	3.85
Within Groups	33,172,811,505	1,020	32,522,364			
Total	33,211,345,236	1,021				

Conclusion: No significant difference between groups.

Table 39. Summary of NAC Consumption and Savings, Untreated Comparison (Control) Group

Participant Group	Variable	2001	2002	2003
Kitsap	NAC	20,175	21,126	19,983
	Standard Deviation	7,884	8,137	7,325
	n	81	81	81
	90% C. L.	1,442	1,488	1,340
	Annual Change	-	-951	1,143
	Average Trend	-	-	96
Tricities	NAC	20,923	21,724	21,167
	Standard Deviation	7,620	7,339	7,571
	n	154	154	154
	90% C. L.	1,011	973	1,004
	Annual Change	-	-801	556
	Average Trend	-	-	-122
Oregon	NAC	19,050	18,160	18,116
	Standard Deviation	10,552	8,669	8,885
	n	107	107	107
	90% C. L.	1,679	1,380	1,414
	Annual Change	-	890	44
	Average Trend	-	-	467
All Untreated Cases	NAC	20,160	20,467	19,932
	Standard Deviation	8,712	8,096	8,036
	n	342	342	342
	90% C. L.	775	721	715
	Annual Change	-	-307	535
	Average Trend	-	-	114

Table 40. Test of Differences, Untreated Group, All Cases

Anova: Single Factor

Groups	Count	Sum	Average	Variance
2001	342	6894600	20160	75901438
2002	342	6999743	20467	65547279
2003	342	6816828	19932	64570727

ANOVA

Source of Variation	SS	df	MS	F	P-value	F crit
Between Groups	49280083	2	24640042	0.359	0.699	3.005
Within Groups	70252630314	1023	68673148			
Total	70301910398	1025				

Table 41. Test of Differences, Untreated Group, Kitsap Cases

Anova: Single Factor

Groups	Count	Sum	Average	Variance
2001	81	1634167	20175	62161604
2002	81	1711196	21126	66215470
2003	81	1618642	19983	53649708

ANOVA

Source of Variation	SS	df	MS	F	P-value	F crit
Between Groups	60661551	2	30330776	0.500	0.607	3.033
Within Groups	14562142621	240	60675594			
Total	14622804172	242				

Table 42. Test of Differences, Untreated Group, Yakima Cases

Anova: Single Factor

Groups	Count	Sum	Average	Variance
2001	154	3222128	20923	58070920
2002	154	3345419	21724	53854886
2003	154	3259783	21167	57320307

ANOVA

Source of Variation	SS	df	MS	F	P-value	F crit
Between Groups	51844560	2	25922280	0.459	0.632	3.015
Within Groups	25894655215	459	56415371			
Total	25946499776	461				

Table 43. Test of Differences, Untreated Group, Oregon Redmond Cases

Anova: Single Factor

Groups	Count	Sum	Average	Variance
2001	96	1878908	19572	116905083
2002	96	1796472	18713	77398133
2003	96	1797414	18723	82230052

ANOVA

Source of Variation	SS	df	MS	F	P-value	F crit
Between Groups	46660315	2	23330157	0.253	0.777	3.027
Within Groups	26270660403	285	92177756			
Total	26317320718	287				

Technical Appendix B

EWEB Statistical Results and Tables

Difference of Means Test Between Adjustment and Test-Only Groups
z-Test: Two Sample for Means

	NAC Savings 1	NAC Savings 2
Mean	310	508
Known Variance	11,629,774	6,684,319
Observations	100	183
Hypothesized Mean Difference	0	
z	-0.505	
P(Z<=z) one-tail	0.307	
z Critical one-tail	1.645	
P(Z<=z) two-tail	0.614	
z Critical two-tail	1.960	

Difference of Means Test Between Adjustment and Test-Only Groups

z-Test: Two Sample for Means

	SH Savings 1	SH Savings 2
Mean	121	609
Known Variance	7,891,530	6,411,976
Observations	100	183
Hypothesized Mean Difference	0	
z	-1.445	
P(Z<=z) one-tail	0.074	
z Critical one-tail	1.645	
P(Z<=z) two-tail	0.149	
z Critical two-tail	1.960	

Difference of Means Test Between Adjustment and Test-Only Groups
z-Test: Two Sample for Means

	Cooling Savings 1	Cooling Savings 2
Mean	189	-101
Known Variance	2,029,032	880,792
Observations	100	183
Hypothesized Mean Difference	0	
z	1.832	
P(Z<=z) one-tail	0.033	
z Critical one-tail	1.645	
P(Z<=z) two-tail	0.067	
z Critical two-tail	1.960	

Conclusion: no significant difference between the two groups

Anova: Single Factor

Groups	Count	Sum	Average	Variance
Bin 1	424	2,002,003	4,722	33,081,893
Bin 2	197	798,073	4,051	30,218,302
Bin 3	82	339,510	4,140	35,304,790

ANOVA

Source of Variation	SS	df	MS	F	P-value	F crit
Between Groups	70,325,057	2	35,162,528	1.08	0.34	3.01
Within Groups	22,776,115,938	700	32,537,308			
Total	22,846,440,995	702				

Conclusion: No significant difference by vintage bin; higher savings of older homes are not statistically different given the variance of the study population.

Anova Test of Comparison Group Annual Consumption by Year

Groups	Count	Sum	Average	Variance
2002	80	1,062,561	13,282	28,402,462
2003	80	1,133,899	14,174	27,262,148
2004	80	1,085,187	13,565	29,318,072

ANOVA

Source of Variation	SS	df	MS	F	P-value	F crit
Between Groups	33,223,946	2	16,611,973	0.59	0.56	3.03
Within Groups	6,713,631,917	237	28,327,561			
Total	6,746,855,863	239				

Conclusion: Year-to-year differences are not statistically significant.

Table 44. Analysis Results for Participant Groups

Participant Group	Variable	Mean	Standard Deviation	n	90% CL	t-test	Significance P (T<=t) two-tail
Airflow / Refrigerant Adjustment	NAC, Pre (kWh/yr)	17,927	16,450	100	3,264		
	NAC, Post (kWh/yr)	17,613	17,394	100	3,451		
	NAC Difference (kWh/yr)	310	3,410	100	677	0.92	0.36
	Space Heat, Pre (kWh/yr)	6,980	5,119	100	1,016		
	Space Heat, Post (kWh/yr)	6,858	4,846	100	961		
	Space Heat Saved (kWh/yr)	121	2,809	100	557	0.43	0.67
	Cooling, Pre (kWh/day)	2,559	2,669	100	529		
	Cooling, Post (kWh/day)	368	33	100	7		
	Cooling Saved (kWh/day)	189	1,424	100	283	0.28	0.78
Test Only	NAC, Pre (kWh/yr)	17,577	10,555	183	1,548		
	NAC, Post (kWh/yr)	17,069	9,626	183	1,412		
	NAC Difference (kWh/yr)	508	2,585	183	379	2.66	0.01
	Space Heat, Pre (kWh/yr)	6,422	4,194	183	615		
	Space Heat, Post (kWh/yr)	5,813	3,325	183	488		
	Space Heat Saved (kWh/yr)	609	2,532	183	371	3.25	0.00
	Cooling, Pre (kWh/day)	1,847	957	183	140		
	Cooling, Post (kWh/day)	1,948	1,194	183	175		
	Cooling Saved (kWh/day)	-101	939	183	138	0.28	0.78
Adjustment Plus Repair	NAC, Pre (kWh/yr)	16,368	6,967	20	3,091		
	NAC, Post (kWh/yr)	15,816	6,246	20	2,771		
	NAC Difference (kWh/yr)	553	2,757	20	1,223	0.90	0.38
	Space Heat, Pre (kWh/yr)	5,965	3,890	20	1,726		
	Space Heat, Post (kWh/yr)	5,448	2,375	20	1,054		
	Space Heat Saved (kWh/yr)	516	2,823	20	1,252	0.82	0.42
	Cooling, Pre (kWh/day)	1,455	786	20	349		
	Cooling, Post (kWh/day)	321	6	20	3		
	Cooling Saved (kWh/day)	36	828	20	367	6.43	0.00
Repair Only	NAC, Pre (kWh/yr)	18,530	7,153	19	3,256		
	NAC, Post (kWh/yr)	18,265	7,832	19	3,565		
	NAC Difference (kWh/yr)	265	2,582	19	1,175	0.45	0.66
	Space Heat, Pre (kWh/yr)	6,500	3,169	19	1,443		
	Space Heat, Post (kWh/yr)	6,269	3,407	19	1,551		
	Space Heat Saved (kWh/yr)	231	2,667	19	1,214	0.38	0.71
	Cooling, Pre (kWh/day)	1,752	825	19	376		
	Cooling, Post (kWh/day)	1,718	917	19	417		
	Cooling Saved (kWh/day)	34	542	19	247	0.27	0.79
All Participants	NAC, Pre (kWh/yr)	17,622	12,386	319	1,376		
	NAC, Post (kWh/yr)	17,176	12,351	319	1,372		
	NAC Difference (kWh/yr)	446	2,875	319	319	2.77	0.01
	Space Heat, Pre (kWh/yr)	6,570	4,442	319	493		
	Space Heat, Post (kWh/yr)	6,141	3,856	319	428		
	Space Heat Saved (kWh/yr)	429	2,654	319	295	2.89	0.00
	Cooling, Pre (kWh/day)	2,027	1,713	319	190		
	Cooling, Post (kWh/day)	2,011	2,274	319	253		
	Cooling Saved (kWh/day)	17	1,078	319	120	0.28	0.78

Technical Appendix C

House / Ducts Survey Protocol

Heat Pump Survey Protocol

Recruiting Telephone Interview Protocol

Field Protocol / Control Group Questionnaire

Name:	**Date:**
Address:	**Technician(s):**
Phone:	**Organization:**
Utility:	

Homeowner Acknowledgment:
I acknowledge that I have given permission for Ecotope, Inc. or its representative to test my heat pump system and house as part of the Northwest Energy Efficiency Alliance Heat Pump Testing Program. Ecotope and its subcontractors are covered by $1 million professional liability insurance. Ecotope will repair or cause to be repaired any damage caused as the result of the testing.

_______________________________________Homeowner signature   Date

By signing below, I allow Ecotope, Inc. to request and use utility billing information to evaluate the energy performance of heat pumps. The information will be kept strictly confidential and only used for pooled summaries of results.

_______________________________________Homeowner signature   Date

Electric utility account #(if available):__________________

Account holder name (if different from above): ________________________________

Order of tests

1. Homeowner questions
2. House characterization (age, number of stories, UA)
3. Duct characterization: location/insulation
4. As found system static pressures (plenums and registers)
5. Blower door & Duct Blaster tests
6. TrueFlow test (if not performed by AC tech)

House type:	**Rambler** **2 story** Split level attached Garage Other (specify):	**Year house built**	
		Indicate major remodel details/dates (especially if weatherization occurred):	
		Location of heat pump indoor unit: In house Garage Crawl Attic Other:	

Homeowner interview:

How many people live here full-time? Adults (age 12 or over):_____Children (under 12):______

Thermostat make/model (if known):_______________________________
Type:_____programmable ____non-programmable

At what temperature do you usually set your thermostat (for heating):_____
Do you set back your thermostat? ___yes ___no. If yes, to what temperature ____?

During heating season, first thing in the morning, do you turn your thermostat up if you feel the house isn't heating up fast enough? ___yes ___no.

Do you know if there is a control on your system to limit the use of the backup heat? ___ yes ___no

Does your house experience brownouts or other power problems? Y N
How many times/year?_________

How much wood do you burn in a typical winter?_________
What is your water heat fuel
____

Does the house have a LPG fireplace ____ or stove/oven _______ or dryer _____?
About how many gallons of LPG do you use per year? _______

Other auxiliary electric loads: well pump____
extra refrig/freeze ________ shop equipment ____
Spa/hot tub ______
Other______________________________ _________

Do you have a whole house ventilation system? ___yes ___no
If yes, what type: ____spot fan on timer ___other whole house fan ___AAHX other________________

Do you have any problems to report with your heating system?

House Audit and U-value Tables

We need to know enough about the house to estimate its heating load. ***You therefore need to calculate a house UA.*** *The purpose of this is to compare the load with the heat pump size. Areas can be reported to the nearest 50 ft². Accuracy is more important in poorly insulated houses. For windows, the big break is between single and double glazed units; within double-glazed units with metal frames, older units have smaller air spaces and non-thermally improved frames.* ***Calculate house volume; you do not need to calculate infiltration UA.***

Windows

Single glazing	1.1
Double glazing metal	0.75
Double glaze metal improved	0.65
Double with wood/vinyl frame	0.55
Dbl wood/vinyl low e	0.40

Above Grade Walls

Uninsulated	0.25
R-11	0.09
R-19	0.065

Doors

Hollow wood*	0.50
Panel wood*	0.40
Solid wood*	0.35
Insulated metal	0.20

*subtract 0.15 from U-value if storm door installed. If more than half glass, use appropriate glass U-value.

Below Grade Walls (fully below grade; assumes uninsulated slab)

Uninsulated	0.2
R-11	0.06
R-19	0.04

Floor Over Crawlspace

Uninsulated	0.12
R-11	0.055
R-19	0.04
R-30	0.03

Slab Floors (use lineal feet, not ft²)

Uninsulated on grade	0.75
Uninsulated below grade	0.50
Insulated on grade	0.55

Attics/vaults

Uninsulated	0.3
R-11	0.06
R-19	0.05
R-30	0.04
R-38	0.03

Worksheet page for house audit:

Record house UA (no infiltration) here: __________________Btu/ft^2 ºF

Record house volume here: _______ft^3

Ducts

We need enough information to estimate the system efficiency of the ducts. This means getting the length and diameter and insulation level of the ducts in unconditioned spaces such as garage, attic and crawlspace. If the ducts run between-floors, also note this. ***Ducts fully inside the conditioned space do not need to be measured.*** *Measure diameters to nearest inch and lengths (overall) to nearest 3'. Estimate as needed to save time by pacing off runs inside the house, using stud spacing as an estimating device, etc. If insulation is damaged or missing, note as needed. The duct audit should take no more than 30 minutes.* ***Describe both supply and return sides of system.***

It is not necessary to make a sketch but use of grid paper is helpful.

Supply ducts (list all unique dimensions/insulation levels)

Duct type (metal/flex)	Duct Zone Location (garage, attic, crawl, other)	Dimension (LxW or inside diameter if round)	Length (feet)	Area (ft^2) (convert dimension to ft first)	Insulation (best guess on R-value)*	UA to Duct Zone

*R-value/inch is about 3 for fiberglass; derate if damaged or missing

Return ducts (list all unique dimensions/insulation levels)

Duct type (metal/flex)	Duct Zone Location (garage, attic, crawl, other)	Dimension (LxW or inside diameter if round)	Length (feet)	Area (ft^2) (convert dimension to ft first)	Insulation (best guess on R-value)*	UA to Duct Zone

*R-value/inch is about 3 for fiberglass; derate if damaged or missing

If any ducts in crawl, check blank if crawl is vented (more than 4 open vents):______
If any ducts in attic, check blank if attic vented (soffit and ridge or gable vents):______

Notes on duct system (use back of sheet as needed):

2-Point Blower Door Test (ducts unsealed)

Depressurize to near 50 and 25 Pa with respect to outside. **Note the house pressure WRT outside doesn't have to be exactly 50 or 25 Pa; the actual values will be corrected to 50 Pa during analysis.**

Make and model of blower door used

____

Blower Door (BD) Depressurization Test Procedure:

1. *Close all windows and doors to the outside. Open all interior doors and supply registers.*
2. *Close all dampers and doors on wood stoves and fireplaces. Seal fireplace or woodstove as necessary to prevent ash disaster.*
3. ***Make sure furnace and water heater can not come on during test. Put water heater and/or gas fireplace on "pilot" setting. Make sure all exhaust fans and clothes dryer are off. Make sure any other combustion appliances will not be backdrafted by the blower door**.*
4. ***Make sure doors to interior furnace cabinets are closed**. Also make sure crawlspace hatch is on, even if it is an outside access. Check attic hatch position. Put garage door in normal position.*
5. *Set fan to de̲pressurize house. Run pressure tap out through door shroud.*
6. *Depressurize house to –50 Pa or thereabouts. Record house pressure, BD flow pressure, and BD ring (below). If you cannot reach –50 Pa, get as close as possible and record information.*
7. *Now take the house down to –25 Pa WRT outside and record information.*

Blower Door Tests	House P near 50 Pa (P_{50})	BD fan pressure	BD Ring	BD flow near 50 Pa (Q_{50})	House P near 25 Pa (P_{25})	BD fan pressure	Ring	BD flow near 25 Pa (Q_{25})
Test 1								
Test 2								

8. *To check test, calculate the flow exponent, n. Use the following formula, $n = \ln(Q_{50}/Q_{25})/\ln(P_{50}/P_{25})$. Note Q_{50} and Q_{25} are the flows through the blower door at the testing pressures (which are denoted P_{50} and P_{25}. Depending on the test, you may not get the house to exactly –50 or –25 Pa WRT outside. Use the exact ΔP you measure when checking the flow exponent. For example, if the house gets to –48 Pa for the high ΔP, use this as the P_{50} in the equation. If the flow exponent is not between 0.50 and 0.75, repeat the test.*

Note testing conditions (if windy, inaccessible room(s), garage door open or closed, etc):

Duct Pressurization Tests (Total and Exterior Duct Leakage)

Set-up procedure for duct pressurization tests:

1. *Set blower door to pressurize house. May have to flip fan (from normal BD test position) if house leaky.*
2. *Set up for total test first; attach to best point in system.*
3. *If testing supply side, attach duct tester fan to furnace cabinet with cardboard and tape or directly to blower mount. (If blower removed, be careful with wires and record how to re-connect them.)*
4. *Tape all registers. Use appropriate tape (Long Mask) for friable surfaces.*
5. *Set up pressure tubes so that pressure gauge can read duct pressure WRT outside, duct tester fan pressure, and house pressure (for exterior duct leakage test).*
6. *Measure duct pressure in plenum or register. If you select a register, make sure it is not disconnected from the rest of the duct system. Specify on protocol sheet where duct pressure is measured. Use Pitot tube or static pressure tap for this measurement.*
7. *Make sure crawlspace access door is on (even if access is from outside house).*

Performing total duct leakage test

1. *For "both sides" test, pressurize supply and return side to about 50 Pascals WRT outside with smallest flow ring possible. Fan pressure should be at least 20 Pa to ensure accuracy.*
2. *If split test (testing supply or return alone), once ducts are pressurized to near 50 Pa, check pressure in other side of system WRT outside. (Check return if testing supply; supply if testing return.) This pressure should be zero or close to zero. If not, check system split.*
3. *Measure the duct system pressure WRT outside. Record in table (next page).*
4. *Measure duct tester fan pressure. Look up flow in table, use gauge* ***(make sure the pressure gauge you are using is paired with the right duct tester)*** *or use flow equation.*
5. *Repeat steps 1-3 with ducts at about 25 Pa WRT outside.*
6. *Check flow exponent. (Formula on next page.) Repeat tests as needed.*
7. ***If you cannot reach 50 Pa or 25 Pa, test to the highest pressure you can reach and enter this in the 50 Pa column. Use a test pressure of half this pressure for the low pressure test.***
8. *Note any unusual testing conditions (wind, etc.):*

Total Duct Leakage Data (note duct pressure WRT outside does not have to be exactly 50 or 25 Pa)

	Both sides		Supply or Return (circle one)	
	50 Pa	25 Pa	50 Pa	25 Pa
Duct P	____	____	____	____
Ring	____	____	____	____
Fan P	____	____	____	____
Flow	____	____	____	____

Note position of pressure tap(s) in supply and return system:
To check each test, calculate flow exponent as for the blower door test (previous page).
The flow exponent, n, = $ln(Q_{50}/Q_{25})/ln(P_{50}/P_{25})$. *If flow exponent not between 0.50 and 0.75, repeat test.*

Performing exterior duct leakage test:

1. *Exterior house doors and garage doors should be closed for exterior duct leakage test.*
2. *Pressurize the house to about 50 Pascals WRT outside.*
3. *Pressurize tested part of duct system to about 50 Pascals with smallest flow ring possible.*
4. *Measure pressure of ducts WRT house. Make sure blower door flow does not impinge on pressure tap measuring house pressure.*
5. *Adjust duct tester speed controller so that duct pressure WRT house is zero or very close.*
6. *Re-check pressure of ducts WRT outside.*
7. *Measure duct tester fan pressure. Look up flow in table, use gauge* ***(make sure gauge is paired with the right duct tester)*** *or use flow equation. Record duct pressure WRT out, DB fan pressure, DB fan ring.*
8. ***If you cannot reach 50 Pa or 25 Pa, test to the highest pressure you can reach and enter this in the 50 Pa column. Use a test pressure of half this pressure for the low pressure test.***
9. *Repeat steps 2-7 with house and ducts at about 25 Pa WRT outside.*
10. *Check flow exponent (as above).*
11. *Note any unusual testing conditions (wind, etc.):*

Duct Leakage to Outside Data (note duct pressure WRT outside may not be exactly 50 or 25 Pa)

	Both sides		Supply or Return (circle one)	
	50 Pa	25 Pa	50 Pa	25 Pa
Duct P	____	____	____	____
Ring	____	____	____	____
Fan P	____	____	____	____
Flow	____	____	____	____

<u>System Airflow (TrueFlow or Duct Blaster)</u>

Set-up: *Turn on air handler (by using fan-only switch or by turning on heat/AC). It is best to call for the flow that will be used during most of the year (probably heating) so that the leakage fraction will be applied to the predominant use. Drill access hole as needed and point hooked end of Pitot tube into airflow.* ***Do not drill into the duct at any point where you are concerned with hitting something.***

Measure pressure in supply plenum. Record pressure below as Normal System Operating Pressure (NSOP). Also measure pressure in return plenum and record:__________

Place appropriate plate and spacers into filter slot. Turn on air handler and record register static with TrueFlow in place (TFSOP) and pressure drop across plate.

Plate used (14 or 20) _______________

Normal System Operating Pressure (NSOP) _____Pa
True Flow System Operating Pressure (TFSOP) _____Pa
Correction Factor* √(NSOP/TFSOP) _________

Plate pressure drop ________Pa
Raw Flow (CFM)________
Corrected Flow ________CFM

*if using DG-700, unnecessary to record CF (but still a good idea)

<u>Air Handler Flow Measurement Using Duct Blaster (if TrueFlow cannot be used)</u>

Record normal system operating pressure (NSOP) as described in flow plate test. Install split between supply and return so that all air flowing through Duct Blaster will go into supply side. Install Duct Blaster on furnace. Turn on air handler. Turn Duct Blaster on and slowly increase flow until the supply plenum pressure is the same as NSOP. Check to make sure the pressure in the return system is 0 or very close to 0 (to confirm system split is good). Record Duct Blaster flow pressure, ring#, and CFM.

NSOP ______Pa
Ring # ______
Flow pressure ______**Pa**
Air Handler flow ______**CFM**

Supply/Return Register Static Pressure Measurements

Put all registers in normal condition (as normally operated). Measure pressures in as many boots as possible; use long Pitot tube and point hooked end into flow. Turn on air handler (heating speed). These measurements will be used to adjust Duct Blaster results to operating conditions.

Register Pressures

Reg #	Toe kick? Normally partially or fully closed? Other notes:	Static P (Pa)	Reg #	Toe kick? Normally partially or fully closed? Other notes:	Static P (Pa)
S1			**S16**		
S2			**S17**		
S3			**S18**		
S4			**S19**		
S5			**S20**		
S6			**S21**		
S7			**S22**		
S8			**S23**		
S9			**S24**		
S10			**S25**		
S11					
S12			**R1**		
S13			**R2**		
S14			**R3**		
S15			**R4**		

Exit Protocol

____ *Remove fireplace seal.*
____ *Turn breakers on where applicable and confirm furnace operation.*
____ *Turn any gas appliances back **ON** and confirm operation.*
______ *Inspect home, garage, crawlspace, attic for any equipment, garbage, etc.*

Northwest Energy Efficiency Alliance Heat Pump Marketing Research Control Group Questionnaire

Good morning (afternoon). My name is ___________________ and I am calling you as part of a research project being conducted by several regional utilities and the Northwest Energy Efficiency Alliance. The goal of the project is to determine whether heat pump efficiency could be improved if stricter attention were paid to the installation procedures and control settings. Would you be willing to answer a few questions about your home's heating system?

1. Do you have a heat pump? Yes No
   *If **No**, end survey. If **Yes**, continue to Question 2.*

2. Do you have a gas furnace? Yes No
   *If **Yes**, end survey. If **No**, continue to Question 3.*

3. How often do you use wood to heat your home? Never Rarely Often Usually
   *If **Usually**, end survey. Otherwise, continue to Question 4.*

4. Where are the majority of the warm air ducts located? *(If homeowner is unsure, prompt "Under the floor? In the attic? In the walls?").*
   ____

5. What is your home's approximate heated floor area (in square feet)?
   ____

6. Approximately how old is your home?
   ____

7. How many occupants live in the home?
   ____

8. Has your home ever been insulated through a utility program or privately? Yes No

9. May we use your monthly billing history in our analysis? We will be comparing the energy used by homes in your area against homes in similar climates that participated in a utility program. We will not report anything about particular houses; we will only report the statistical results for all the homes in your area as a group. Yes No
   *If **Yes**: Thank you very much. If required by your utility, we will send you a release form to sign so that the utility will provide your billing history.*

Take down address and any additional phone numbers at bottom of page.

10. Would you be interested in participating in the field review portion of our research? An experienced, licensed HVAC technician would inspect your heat pump to see if it is operating at the correct charge and settings. After recording the findings, the technician would provide a tune up to optimize the heat pump's operation, although no major repair work would be done. As a thank you for your time, we are offering $25, and the assurance that your heat pump is operating at its most efficient. Yes No
    *If **Yes**, take down address and any additional phone numbers.*

    Address: ____

    City, State, Zip: ____

    Work / Cell / Fax phone: ________________________________

Technical Appendix D
Laboratory Data Adjustments

Purdue's original test results showed differences between the measured air-side and refrigerant-side capacities as large as 13%. The sensitivity of the HSPF calculation to measurement errors of even a few percent in the capacity requires careful measurement. Investigation of Purdue's test facilities, setup and methodology revealed several possible sources of measurement error, including air leakage, as Purdue asserted. Purdue reported corrected airflow rates, capacities and COP values in the final report based on the refrigerant-side capacity measurements as described in this section. In addition to investigating air leakage as a cause for the discrepancy, and the validity of the refrigerant-side capacity as a method of adjustment, we also looked at other possible methods of adjustment.

Air Side Measurements

The adjustment factors used as the final results are Purdue's corrections to the air-side capacity using the refrigerant-side measurements, where available. Refrigerant-side measurements are known to be less error-prone than on the air side and the results of the correction are consistent across variables. The adjustment factors used in the final reported capacity and airflow data are shown in Table 45.

Table 45. Purdue Adjustment Factors using Refrigerant-Side Capacity Measurements

	800 cfmnom	1100 cfmnom	1300 cfmnom	1500 cfmnom	1700 cfmnom
FEO	1.0403	1.0909	1.0570	1.0088	1.0068
TXV	0.9986	1.0763	1.0504	0.9957	0.9872

Two difficulties with this method of adjustment are:

1. Correcting the air-side measurements where the refrigerant-side measurements could not be taken

2. Determining whether the airflow rate should also be adjusted

To adjust the cases with missing refrigerant-side capacities, Purdue averaged the air-side to refrigerant-side deviations at each flow rate and applied those averages as the multipliers. Note that there are only four refrigerant-side capacities available for the 800 cfm nominal cases without a TXV. These four values, which are all at 47°*F*, are used to generate the multiplier used on eleven other measurements at 800 cfm.

When Purdue suggested the refrigerant and air-side capacity measurement discrepancy was due to air leakage we contracted Paul Francisco of The Building Research Council at the University of Illinois at Urbana-Champaign to travel to Purdue and perform leakage tests on the test facilities. Mr. Francisco reviewed the Purdue facility and reported on the results of various adjustments to the data and help correct for errors in the testing results.

Purdue adjusts the airflow rate by the same multipliers as used to adjust the capacity on the basis that the discrepancy is caused strictly by air leakage. One independent test of the Purdue adjustments to the airflow rate lends confidence that they are appropriate. Using pressure measurements taken by Mr. Francisco, a relationship can be determined between the pressure across the filter (P_{filter}) at each nominal flow rate and the flow rates (Q_1, Q_2). The flows used to determine this power law relationship were those at which Mr. Francisco measured the pressures, but adjusted by the same multipliers used to correct the air-side capacity measurements (Table 45). The robust regression using the multipliers intended for use with the FEO data produces

$$Q_1 = 195.7 \cdot P_{filter}{}^{0.522}$$

Equation 2

and using the multipliers in Table 45, intended for use with the TXV data, produces

$$Q_2 = 191.2 \cdot P_{filter}{}^{0.524}$$

Equation 3

Equation 2 and Equation 3 both have physically reasonable coefficients and exponents, providing one added measure of confidence in the adjustment multipliers suggested by Purdue. Further, a plot of the natural log of the adjusted flows against the natural log of the pressure across the filter produces a straight line. Although this process doesn't verify that the value of the multipliers in absolute terms is correct, it does indicate that the relative adjustments at each flow rate are reasonable.

One concern with this method of adjustment is that it results in an increase of flow at 800 cfm. The pressure tests show the ducts to be depressurized in this case though, implying that the adjustment should be in the direction of reduced flow. In other words, the correction should compensate for leakage into the duct after the coil but before the airflow is measured. If the airflow were decreased, as we believe to be more appropriate, the effect of the capacity and COP dropping off at low flows seen in almost all graphs would be diminished. Unfortunately, no method of analysis could produce a more consistent and robust correction to replace the one presented by Purdue.

Air- and Refrigerant-side Capacity Measurement Adjustments

The difference between the refrigerant- and air-side capacity measurements are as much as 13% different (at 47*F*, 130% charge, 1100 cfm). Purdue suggests that the difference is entirely attributable to leakage in the test setup. We further noted that the placement of the temperature grid directly after the internal fan outlet also may have contributed to this discrepancy. We investigated the test setup and methodology, and a number of different methods of data analysis to test the validity of the adjustment factors.

Airflow Measurement

Test measurements were controlled using airflow rates measured in cubic feet per minute (cfm) instead of standard cfm (scfm). This standardization is used to account for differences in atmospheric pressure and air density. After applying the conversion, the resultant airflow rates often differed greatly from the nominal test rates specified. However, note that after the adjustments are applied to the airflow rates, the values are often once again very close to the nominal value specified.

Leakage Testing

Leakage tests on the Purdue test facilities were not specified in conjunction with or otherwise performed prior to the testing for this project. Mr. Francisco was asked to collect sufficient details that would allow us to determine the veracity of Purdue’s adjustments.

The overall system duct leakage tests were performed with a Duct Blaster® attached via a flexible duct to the return-side inlet in the air handler cabinet. Most tests were performed with the Duct Blaster® pressurizing the duct system. The tests were as follows:

- Test 1 was done with no sealing of penetrations or bypasses in the return-side cabinet.
- Test 2 was done with bypasses at the inlet taped (e.g. where temperature sensors for the coil entered the unit) and annuluses around the refrigerant lines also taped.
- Test 3 was done with the flow nozzles also sealed off, so as to measure leakage only between the return-side inlet and the nozzles. This test was also in pressurization mode.
- Test 4 was identical in setup to the third test (leakage only to nozzles), except that it was done in depressurization mode.

Tests 1, 2, and 3 were done in pressurization mode while test 4 was done in depressurization mode. The results of these tests are in Table 46, and the power laws are written out in Equation 4 through Equation 7 below.

Pressures are reported in units of Pascals (Pa) and flows are reported in cubic feet per minute (cfm). Pressures were measured in the location used by Purdue to measure pressures.

Table 46. Results of Duct Blaster® Tests

	Test 1	Test 2	Test 3	Test 4
Flow at 25 Pa	136 cfm	112 cfm	46 cfm	45 cfm
Flow at 50 Pa	216 cfm	176 cfm	71 cfm	69 cfm
Power Law Coefficient	15.9	13.7	6.1	6.2
Power Law Exponent	.667	.652	.626	.617

The resultant power law leakage equations are

$$Q1=15.9\Delta P^{.667} \qquad \text{Equation 4}$$

$$Q2=13.7\Delta P^{.652} \qquad \text{Equation 5}$$

$$Q3=6.1\Delta P^{.626} \qquad \text{Equation 6}$$

$$Q4=6.2\Delta P^{.617} \qquad \text{Equation 7}$$

where Qn is the leakage flow (cfm)
ΔP is the pressure between the ducts and their surroundings (Pa)

Note Equation 7, the depressurization test, is virtually identical to the results in pressurization mode (Equation 4, Equation 5, and Equation 6). The final two tests indicate that, for the same magnitude of pressure, the leakage is identical regardless of whether the duct is pressurized or depressurized.

Pressure Measurements

Mr. Francisco also measured pressures at different points in the ducts and air handler cabinet. These tests were not duct leakage tests, but were conducted at normal operation, with the Duct Blaster® serving as an additional measurement device at the indoor end of the system. These results are shown in Table 47.

Table 47. Nozzle and Duct Blaster Flows, and Duct Pressures with Respect to Ambient. Fan configuration identifies which fans were operating (DB = Duct Blaster®; Int. = Internal fan; Ext. = External fan).

Test ID	Fan Configuration	Duct Pressure (Pa)	Flow Nozzle flow (cfm)	Duct Blaster flow (cfm)
1	DB only	26	505	700
2	Ext. only	-278	800	680
3	Int. only	48	715	740
4	Ext. and Int.	-142	1125	1000
5	Ext. and Int.	-196	1200	1060
6	Ext. and Int.	-270	1300	1150
7	Ext. and Int.	-443	1515	1300
8	Ext. only	-361	900	770
9	Ext. only	-505	1080	900
10	Ext. and DB	-481	1260	1100

These results show that, as expected, the flow nozzle reads a higher value when the duct is depressurized, since air is entering the duct through holes downstream of the Duct Blaster®. The Duct Blaster® reads a higher value when the duct is pressurized.

Temperature Grid

The temperatures entering and exiting the air handler are reported as an average of eight thermocouples arranged on each of two grids. Further downstream, the temperature through the flow nozzles is also recorded. The outlet temperature grid is separated from the nozzle by some length of ductwork which travels both within and outside of the indoor environmental chamber. However, most of the reported nozzle temperatures were greater than those recorded at the outlet temperature grid, despite the lack of heat source and the conduction losses in the length of duct. Photographs of the test setup show the outlet temperature grid located immediately downstream of the air handler fan outlet. The uneven velocity distribution at this point in the system can cause inaccurate temperature measurements if the average temperature calculations are not velocity-weighted.

Using the individual temperature measurements from the temperature grid on the outlet side of the air handler, the velocity-weighted temperature averages were created. These were calculated by averaging the individual temperatures weighted with a set of nine-point velocity measurements performed at each of the nominal flow rates. This new average greatly reduced, but did not eliminate, the occurrence of apparent temperature increase in travel to the nozzle.

A capacity correction multiplier can then be created from the ratio of the velocity-weighted temperature rise to the original temperature rise across the equipment. Looking at the multipliers in Table 48, it is clear that the only significant adjustment would occur at the lowest flow; all others are negligible. This low flow adjustment agrees with the 4% adjustment suggested by Purdue (Table 30) for the FEO cases at the lowest flow. However, Purdue's low flow multiplier for the TXV case (0.9986) is very different, as are both adjustments at the nominal 1100 and 1300 cfm flow rates.

Table 48. Multipliers using Velocity Weighting

	800 cfmnom	1100 cfmnom	1300 cfmnom	1500 cfmnom	1700 cfmnom
FEO	1.040	1.004	1.003	1.006	1.005
TXV	1.038	1.005	1.005	1.006	1.004

The results of the velocity weighting corrections to the measured air temperature produced results that corroborated Purdue's adjustment factors to a high degree in certain cases, but not in others (as seen in Table 45 and Table 48). Most of the adjustment factors which did differ from those based on the refrigerant-side measurements were very small.

Technical Appendix E

Realization Rate Calculations
by Vintage and Climate Zone, House Type & Equipment Type

This appendix summarizes the billing results (NAC savings) by climate zone, house type and base equipment type regardless of the statistical significance or N of the analysis. In many cases the results cannot be considered significant. In those cases the results are superimposed with a gray background. The realization rates are calculated against both the deemed savings for the heat pump programs in the 2003-2003 spreadsheets and the 2005 spreadsheets (shown in the table for each combination of vintage, house type, and equipment type). In general all the units installed in this sample applied under the 2002-2003 deemed savings calculations.

The climate zone designations reflect the sample used in this analysis: Climate Zone 1 in this analysis refers to the coastal climates and the northern Olympic Peninsula of Washington (Heating Zone 1, Cooling Zone 1); Climate Zone 2 refers to the Portland area and the Oregon Willamette Valley (Heating Zone 1, Cooling Zone 2); Climate Zone 3 refers to the Tri-Cities area of Eastern Washington (Heating Zone 1, Cooling Zone 3).

The equipment divisions refer to the base conditions. Used only the "non PTCS" entries to calculate the realization rates. In general, there were very few cases in this sample that used the PTCS options so they were combined to develop the savings estimates.

Climate Zone **1**
Site Built
Equip. Base

	Vintage			
	pre1980	1981-1992	1993+	Total
FAF w/AC	6432	5728	1215	5220
Std. Dev	6485	5086	8921	7086
N	19	2	6	27
yr 2003	7642	9011	6733	7541
yr 2005	5442	8986	5752	5773
Realization 2003	84.2%	63.6%	18.0%	69.2%
Realization 2005	118.2%	63.7%	21.1%	90.4%
FAF w/o AC	5869	4376	4880	5044
Std. Dev	8242	6647	5755	7225
N	57	66	18	141
yr 2003	6928	8283	5889	7430
yr 2005	5114	8283	5239	6613
Realization 2003	84.7%	52.8%	82.9%	67.9%
Realization 2005	114.8%	52.8%	93.1%	76.3%
Heat Pump	7846	4857	6539	5986
Std. Dev	7299	5021	10791	7162
N	11	23	10	44
yr 2003	888	1032	812	946
yr 2005	1542	2129	1483	1835
Realization 2003	883.5%	470.6%	805.3%	632.8%
Realization 2005	508.8%	228.1%	440.9%	326.2%
Zonal	3041	2971	1830	2868
Std. Dev	6154	6124	3507	5828
N	63	15	12	90
yr 2003	2960	4430	3647	3297
yr 2005	2245	3565	2101	2446
Realization 2003	102.7%	67.1%	50.2%	87.0%
Realization 2005	135.5%	83.3%	87.1%	117.3%
Total	4897	4307	3967	4548
Std. Dev	7260	6184	7189	6877
N	150	106	46	302
yr 2003	4909	6178	4311	5263
yr 2005	3689	6293	3671	4600
Realization 2003	99.8%	69.7%	92.0%	86.4%
Realization 2005	132.8%	68.4%	108.1%	98.9%

Climate Zone **2**
Site Built

Equip.	**Vintage**			
	pre1980	1981-1992	1993+	Total
FAF w/AC	23812	-	-	23812
Std. Dev	0	-	-	0
N	1	-	-	1
yr 2003	7678	-	-	7678
yr 2005	5519	-	-	5519
Realization 2003	310.1%	-	-	310.1%
Realization 2005	431.5%	-	-	431.5%
FAF w/o AC	8988	5323	-	7831
Std. Dev	4281	2577	-	4138
N	13	6	-	19
yr 2003	6730	8026	-	7139
yr 2005	4767	7678	-	5686
Realization 2003	133.6%	66.3%	-	109.7%
Realization 2005	188.6%	69.3%	-	137.7%
Heat Pump	3476	2859	2806	3161
Std. Dev	1820	3753	3055	2640
N	8	6	2	16
yr 2003	924	1078	850	973
yr 2005	1619	2263	1587	1857
Realization 2003	376.2%	265.2%	330.1%	325.0%
Realization 2005	214.7%	126.3%	176.8%	170.3%
Zonal	4912	-2135	4649	4247
Std. Dev	4219	0	0	4328
N	9	1	1	11
yr 2003	2762	4143	3439	2949
yr 2005	1898	2961	1632	1970
Realization 2003	177.8%	-51.5%	135.2%	144.0%
Realization 2005	258.8%	-72.1%	284.9%	215.6%
Total	6861	3612	3420	5742
Std. Dev	5346	3625	2408	4980
N	31	13	3	47
yr 2003	4110	4521	1713	4071
yr 2005	3146	4816	1602	3509
Realization 2003	166.9%	79.9%	199.6%	141.1%
Realization 2005	218.1%	75.0%	213.5%	163.6%

Climate Zone **3**
Site Built

Equip.	**Vintage**			
	pre1980	1981-1992	1993+	Total
FAF w/AC	4516	4569	3613	4511.424
Std. Dev	4010	3509	459	3922.068
N	148	20	2	170
yr 2003	7755	9141	6849	7907.4
yr 2005	5664	9365	6047	6103.918
Realization 2003	58.2%	50.0%	52.7%	57.1%
Realization 2005	79.7%	48.8%	59.7%	73.9%
FAF w/o AC	2296	-	-	2296
Std. Dev	6641	-	-	6641
N	12	-	-	12
yr 2003	6307	-	-	6307
yr 2005	4134	-	-	4134
Realization 2003	36.4%	-	-	36.4%
Realization 2005	55.5%	-	-	55.5%
Heat Pump	5468	154	-641	3123
Std. Dev	5506	3174	0	5263
N	13	9	1	23
yr 2003	1001	1162	929	1061
yr 2005	1764	2507	1778	2055
Realization 2003	546.2%	13.3%	-69.0%	294.4%
Realization 2005	310.0%	6.1%	-36.1%	151.9%
Zonal	564	2701	-	842
Std. Dev	5478	3012	-	5224
N	20	3	-	23
yr 2003	2338	3713	-	2517
yr 2005	1265	1899	-	1348
Realization 2003	24.1%	72.7%	-	33.5%
Realization 2005	44.5%	142.2%	-	62.5%
Total	4032	3152	2195	3885
Std. Dev	4633	3828	2477	4510
N	193	32	3	228
yr 2003	6649	6388	4876	6589
yr 2005	4850	6736	4624	5112
Realization 2003	60.6%	49.3%	45.0%	59.0%
Realization 2005	83.1%	46.8%	47.5%	76.0%

Climate Zone **1**
Manufactured Homes

Equip.	**Vintage**			
	pre1992	1992+	SGC	Total
FAF w/AC	5737	11744	1181	4971
Std. Dev	2916	3186	3645	4323
N	12	2	6	20
yr 2003	5827	4599	3276	4939
yr 2005	5735	4641	3066	4825
Realization 2003	98.5%	255.3%	36.1%	100.6%
Realization 2005	100.0%	253.0%	38.5%	103.0%
FAF w/o AC	5449	4129	4742	5153
Std. Dev	4239	4111	2068	3943
N	45	9	10	64
yr 2003	5066	3765	2344	4458
yr 2005	4979	4145	2717	4508
Realization 2003	107.6%	109.7%	202.3%	115.6%
Realization 2005	109.4%	99.6%	174.5%	114.3%
Heat Pump	5238	-	-	5238
Std. Dev	5268	-	-	5268
N	6	-	-	6
yr 2003	947	-	-	947
yr 2005	947	-	-	947
Realization 2003	553.1%	-	-	553.1%
Realization 2005	553.1%	-	-	553.1%
Total	5484	5514	3407	5118
Std. Dev	4065	4901	3188	4070
N	63	11	16	90
yr 2003	4819	3917	2694	4331
yr 2005	4862	4235	2848	4428
Realization 2003	113.8%	140.8%	126.5%	118.2%
Realization 2005	112.8%	130.2%	119.6%	115.6%

Climate Zone **2**
Manufactured Homes

Equip.	**Vintage**			
	pre1992	1992+	SGC	Total
FAF w/AC	8387	-	-	8387
Std. Dev	0	-	-	0
N	1	-	-	1
yr 2003	5865	-	-	5865
yr 2005	5885	-	-	5885
Realization 2003	143.0%	-	-	143.0%
Realization 2005	142.5%	-	-	142.5%
FAF w/o AC	2987	4587	1934	3444
Std. Dev	0	3508	3824	3571
N	1	7	5	13
yr 2003	4873	3495	2138	3079
yr 2005	4533	3769	2424	3310
Realization 2003	61.3%	131.3%	90.5%	111.8%
Realization 2005	65.9%	121.7%	79.8%	104.0%
Heat Pump	2433	-	-129	1793
Std. Dev	1650	-	0	1859
N	3	-	1	4
yr 2003	985	-	646	900
yr 2005	985	-	646	900
Realization 2003	247.0%	-	-20.0%	199.1%
Realization 2005	247.0%	-	-20.0%	199.1%
Total	3735	4587	1591	3352
Std. Dev	2860	3508	3522	3418
N	5	7	6	18
yr 2003	2739	3495	1889	2750
yr 2005	3519	3769	2224	3185
Realization 2003	136.4%	131.3%	84.2%	121.9%
Realization 2005	106.1%	121.7%	71.5%	105.2%

Climate Zone **3**

Manufactured Homes

Equip.	**Vintage**			
	pre1992	1992+	SGC	Total
FAF w/AC	1861	-1382	-	1362
Std. Dev	4226	928	-	4055
N	11	2	-	13
yr 2003	5949	3406	-	5558
yr 2005	6087	3376	-	5670
Realization 2003	31.3%	-40.6%	-	24.5%
Realization 2005	30.6%	-40.9%	-	24.0%
FAF w/o AC	-1306	-	-	-1306
Std. Dev	0	-	-	0
N	1	-	-	1
yr 2003	4453	-	-	4453
yr 2005	3940	-	-	3940
Realization 2003	-29.3%	-	-	-29.3%
Realization 2005	-33.1%	-	-	-33.1%
Heat Pump	-	-	-	-
Std. Dev	-	-	-	-
N	-	-	-	-
yr 2003	-	-	-	-
yr 2005	-	-	-	-
Realization 2003	-	-	-	-
Realization 2005	-	-	-	-
Total	1597	-1382	-	1171
Std. Dev	4132	928	-	3960
N	12	2	-	14
yr 2003	5824	3406	-	5479
yr 2005	5908	3376	-	5546
Realization 2003	27.4%	-40.6%	-	21.4%
Realization 2005	27.0%	-40.9%	-	21.1%

q:\te\rtf\evaluations\heat pumps\heatpumpfinalreport7.0.doc