Combined Space and Water CO_2 Heat Pump System Performance Research

Final Report

Bonneville Power Administration
Technology Innovation Project 326

Organization

Washington State University –
WSU Energy Program in Olympia, WA

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Contents

Tables

Figures

Abbreviations

AC	alternating current
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BPA	Bonneville Power Administration
Btu	British thermal unit
CO_2	carbon dioxide
COP	coefficient of performance
DOE	U.S. Department of Energy
DHW	domestic hot water
DR	demand response
EF	energy factor
ER	electric resistance
FEF	Field Energy Factor
GPD	gallon per day
GPM	gallons per minute
GWP	Global Warming Potential
HFC	hydrofluorocarbons
HPWH	heat pump water heater
HSPF	Heating Seasonal Performance Factor
kWh	kilowatt hour
NEEA	Northwest Energy Efficiency Alliance
NSH	NEEA's Next Step Home Program
OAT	outside air temperatures
PNNL	Pacific Northwest National Laboratory
PSI	pound per square inch
SEEM	Simple Energy and Enthalpy Model
TIP	Technology Innovation Program
UL	Underwriters Laboratory
WSU	Washington State University
XPB	Heat Exchange Pump Block

A Technology Innovation Project Report

The research described in this report was funded by Bonneville Power Administration (BPA) to:

- Assess the potential for emerging technologies, and
- Provide for development of those technologies to increase the efficiency of electricity use and provide other benefits, such as capacity reduction and demand response services.

BPA is undertaking a multi-year effort to identify, assess, and develop emerging technologies with significant potential for contributing to the goals of efficiency, capacity reduction, demand response, and climate change remediation.

Neither Washington State University (WSU) nor BPA endorse specific products or manufacturers. Any mention of a particular product or manufacturer should not be construed as an implied endorsement. The information, statements, representations, graphs, and data presented in these reports are provided as a public service. For more reports and background on BPA's efforts to "fill the pipeline" with emerging, energy-efficient technologies, visit Energy Efficiency's Emerging Technology (E3T) website at http://www.bpa.gov/energy/n/emerging_technology/.

Ken Eklund is the Building Science and Standards Team Lead for the WSU Energy Program. His background includes research organization and management spanning 40 years in the energy-efficiency field. His work at WSU includes facilitating and coordinating staff involved in building science research, and developing and implementing research projects like the current one, which leverages the experience and capabilities of WSU staff and of skilled subcontractors – all blended into a collaborative team.

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Abstract

The CO_2 refrigerant, split-system heat pump water heater was evaluated both in lab tests and in 10 new, high-efficiency homes representing the three heating zones in the Pacific Northwest. This technology served as the heat source in a combined space and water heating system concept design. This report includes both the lab and the field test results, as well as the data and experience collected after the homes were built. This is a promising technology; if optimized it can be part of a comprehensive, low climate impact, space and water heating solution for the Pacific Northwest.

Executive Summary

This report was prepared for Technology Innovation Project (TIP) 326 conducted by Washington State University (WSU) and funded by Bonneville Power Administration (BPA). The research created a prototype combined space and water heating system using a CO_2 refrigerant split system heat pump water heater (HPWH) manufactured by Sanden International (Sanden) and other commercially available components. The system was tested in both lab and field studies. This report describes the research and analysis, delivers results, and offers conclusions and recommendations.

The field study included 10 sites located in all three regional heating climate zones. These new low-load homes were primarily recruited through the Northwest Energy Efficiency Alliance's (NEEA) Next Step Home (NSH) program, and most were built during the project.

The field tests were a voyage of discovery. The cold weather sites revealed defrost and capacity issues. In response, the manufacturer redesigned the split system heat pump and offered a unit with over twice the capacity for cold climates and higher-load homes. After the first site, back-up auxiliary tanks were abandoned in favor of on-demand electric heaters—this decision is now being revisited due to issues with the demand heaters. System control issues plagued system operation until the standard programming designed for condensing gas boilers was customized for hydronic heat pumps. Near the end of the project, cross flow of hot water to the tank inlet from the tempering valves was discovered and suspected of contributing to poor system efficiency at most sites.

The lab test protocol was developed in the context of the first site system design, and included questions arising from that design process. The most important questions were the way in which water should be returned from the heating system to the storage tank and the impact of high- and low-temperature return water on system performance.

The preliminary lab test results (provided in Appendix A) showed that low-temperature water should be returned to the bottom of the tank. Three systems were re-plumbed to implement this change and two systems in progress were plumbed that way. The other five systems retained the original plumbing.

In addition, the lab test recommended that the best heat distribution systems are low temperature, and that storage tanks for combined split systems should be increased to 120 gallons with multiple ports for return water of different temperatures.

The research found major issues in installation that must be solved for basic system functionality, Including:

- Heat pump capacity—not including auxiliary heat—must match design load,
- X-Block programming must be modified to optimize hydronic heat pump performance,
- Tank stratification must be maintained, and
- Cross flow at the tempering valve must be eliminated to allow proper heat pump function.

The system created and tested in this project works for homes with design loads within the heat pump capacity. It is not a solution for cold climates or higher-load homes in other climates. A higher-capacity CO_2 heat pump is needed for these situations.

Introduction

This is the final report on the Washington State University (WSU) Energy Program research into the performance of CO_2 refrigerant heat pumps used for combined space and water heating in high-efficiency new homes. The research was funded by the Bonneville Power Administration (BPA) through its Technology Innovation Program (TIP). The equipment tested in this study was manufactured by Sanden International in Australia.

This research is based on previous research into CO_2 refrigerant heat pump technology conducted by WSU as TIP 292, which demonstrated the ability of the system to provide hot water to a large family during extremely cold weather while operating only 25% of the time. This capacity was corroborated during the demand response (DR) testing under TIP 302 at Pacific Northwest National Laboratory (PNNL) Lab Homes Test Center. The tests used an unusually high daily draw of 130 gallons, and the split system was able to meet the demand while turned off for up to 12 hours.

This research consisted of a field study and lab test focused specifically on combined space and water heating in homes with design load temperatures within the capacity of the heat pump where possible. At the coldest sites, the load was within the combined capacity of the heat pump and auxiliary heat. The lab test took place at Cascade Engineering Services and was conducted by Ecotope, Inc.

Because of the long timeline needed to build the homes, site recruitment began as soon as the project started on October 1, 2014 and continued until June 2015. Multiple leads were pursued to obtain the six sites proposed for the field study, resulting in a total of ten sites. Of these, seven were located in the coastal climate zone; one in the cold inland zone; and two in the very cold, mountainous zone. The sites were selected primarily from builders participating in NEEA's Next Step Home (NSH) program. WSU coordinated the development of engineering and monitoring plans by the project team, which included WSU, CLEAResult, NEEA, and Ecotope.

The lab test was delayed until the project design choices were clear. It was then used to address the research questions defined in the project proposal, and to obtain data on alternative design choices and their performance implications. The test results impacted the system installation at the two last homes constructed, and at the three homes that were retrofitted to implement the findings on heating loop return water configuration. Five homes retained the original configuration for heating loop return.

The project succeeded in:

- Developing a workable design for combined systems,
- Identifying system operation, distribution and heat pump equipment issues,
- Collecting sufficient data to obtain basic operating characteristics of the projects,
- Identifying key variables, and
- Defining development opportunities and design recommendations.

The region desperately needs a natural refrigerant solution for space and water heating. Every heat pump or heat pump water heater (HPWH) installed with climate-destroying refrigerants is a step in the wrong direction (described further in Appendix B). A solid set of natural refrigerant solutions must be developed and implemented.

Basis for Combined Space and Water Heating Experiment

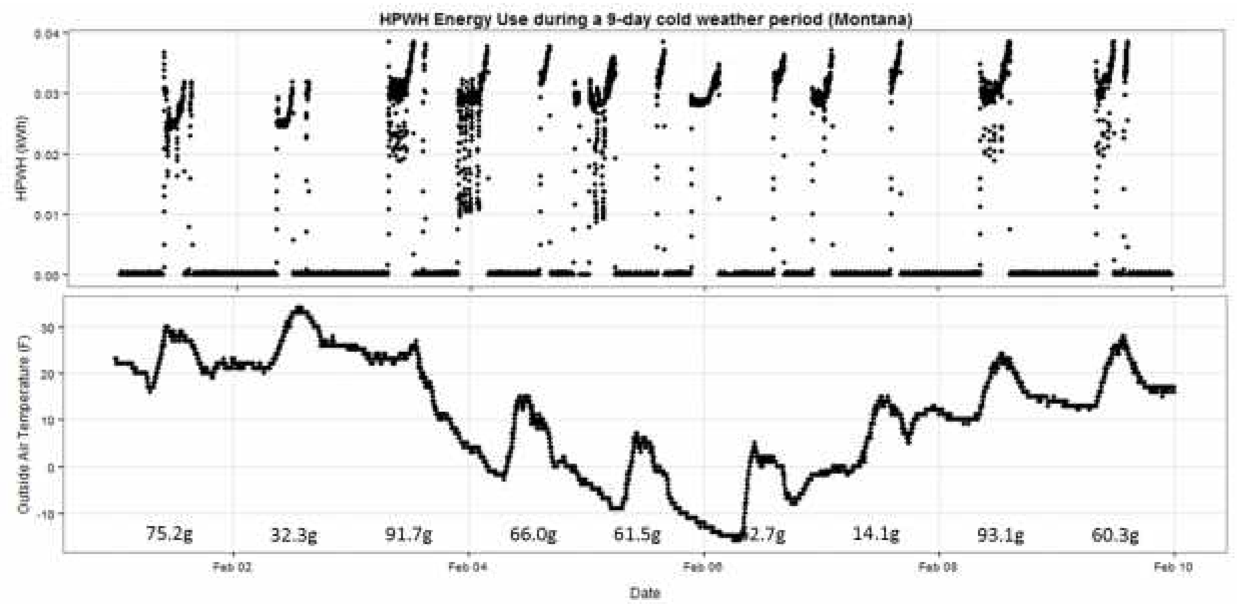
This research is based on the performance of the CO_2 refrigerant heat pump documented as noted in:

- Lab and field tests done under TIP 292: *Laboratory Assessment of Sanden GAU Heat Pump Water Heater* (Larson, September 2013).
- Performance of HPWHs: *Advanced Heat Pump Water Heater Final Report* (Eklund and Banks, December 2015).
- Controlled field tests performed by PNNL and lab tests conducted by Ecotope for TIP 302 to study the demand response (DR) performance of these systems while they functioned as HPWHs: *Demand-Response Performance of Sanden HPWH* (Sullivan, July 2015).

In these studies, specific findings indicated the technology had the capacity to provide heat to end uses while also meeting a substantial hot water load.

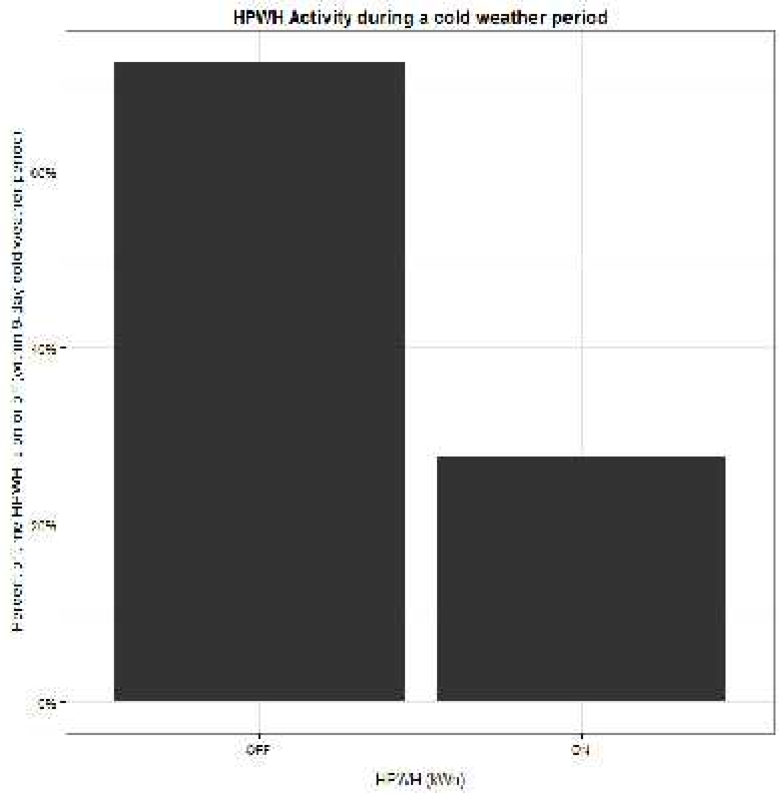
In the original research under TIP 292, the specific finding was a very cold nine-day period at the Montana site when the outside air temperature (OAT) remained almost entirely below freezing and went as low as -16.5°F. The hot water load consisted of a family of four, including two teenagers, who take an average of 22 showers per week. **Figure 1** shows this period. The top graph is the energy use by the system and when it was operating; the bottom graph shows the OAT (including total daily tempered water use along the x-axis) during this period, showing that compressor use was fairly regular.

Figure 1. Hot Water Load During 9-Day Below-freezing OATs – Montana Site



It would seem that a system drawing all of the heat for this end use from this cold air source would be operating frequently. **Figure 2** answers this question by showing the percentage of time the system was on versus the time it was off. The tall bar represents the 75% of the nine-day period when the heat pump was not operating, suggesting that the heat pump could be providing heat for another use. This was the first solid evidence that the system could serve as a combined space and water heating system because it has this heating capacity even during very cold weather.

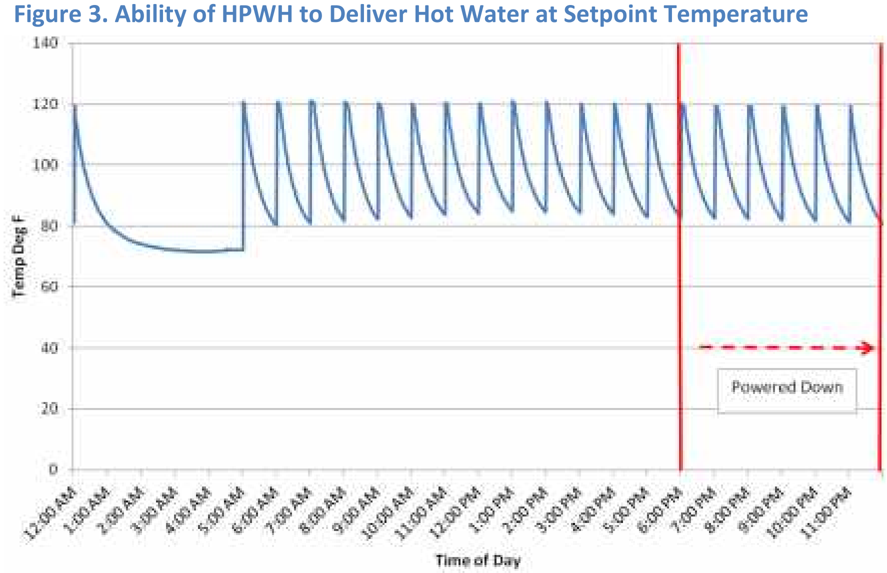
Figure 2. Percentage of Time HPWH On/Off During Cold Weather Period



The second specific finding was from the DR research at PNNL for TIP 302. This set of experiments was based on a daily draw of 130 gallons of hot water in order to test the system under very high-use conditions. Given that the average hot water use in the Pacific Northwest is 42 gallons per day (GPD), the test condition could be considered extreme.

The Oversupply Mitigation test was designed to test the capacity of systems to store energy when there is a surplus of generation. To create a storage bank for nighttime generation, the split system water heater was turned off for up to 12 hours while still supplying 130 GPD. **Figure 3** shows its ability to deliver water at the set temperature without missing a draw. This ability corroborates the field results at the Montana site.

Figure 3. Ability of HPWH to Deliver Hot Water at Setpoint Temperature



Field Study

Ten split system CO_2 refrigerant HPWHs were installed in nine highly insulated new homes and one highly-insulated deep retrofit home across the region beginning in fall 2014. These homes are located in Bellingham, WA; Coeur d'Alene, ID; McCall, ID; Milwaukie, OR; Olympia, WA; Seattle, WA; and Tacoma, WA. The field study was designed to test the performance of the technology in all three of the Pacific Northwest's heating climate zones. The host organizations were Avista, Energy Trust of Oregon, Puget Sound Energy, and Tacoma Public Utilities.

Description

Site Selection

Ten sites were recruited and analyzed through NEEA's NSH program, which is managed by CLEAResult. For each home, CLEAResult staff determined the heat loss rate and annual load using SEEM™, a simulation tool developed by Ecotope, Inc. They also calculated the design heat load using SpecPro©, by Bruce Manclark. The final determination of whether the home would be part of the combined space and water heating experiment was made by Ken Eklund of WSU.

Five of the homes are in the NSH program and built according to its specifications. The four other new homes were built to Passive House™ standards. The one retrofit home in the program, located in Olympia, is a prototype for retrofit application of the combined space and water heating concept. The goal of the project was to sample three primary heating climate zones corresponding to International Energy Conservation Code (IECC) Zones 4C, 5, and 6.

Code Issues and Solutions

The CO_2 HPWH used in these experiments was not yet UL listed. Electrical and building permits were obtained for each of the ten installations. The situation was complicated by the fact that the HPWH was providing space heat as well as hot water. The addition of the second use made obtaining permits in most jurisdictions more difficult than installing the systems simply as water heaters, as was done in TIP 292 and TIP 302. As in those earlier projects, the building official was required to exercise discretion under Section 104 of the International Residential Code, which allows use of alternate materials and systems.

Ken Eklund worked with building officials. The initial permit in Bellingham took eight months to obtain. It required engineered drawings of the system, which proved instrumental in obtaining that permit and all the ones that followed. The engineering was done by Jonathan Heller, PE, at Ecotope. At the Idaho sites, the building official was local and the electrical official was a state inspector. Obtaining these permits required working with both jurisdictions, but once the Coeur d'Alene site was permitted, the McCall permits proved easy because the state officials were already educated and on board.

Sanden International, the manufacturer of the HPWH, has since obtained UL listing for the split system installed in these projects. It was a long and expensive process, and much of the knowledge and experience from these TIP projects was incorporated into the product that is UL listed and sold in North America.

System Design and Installation

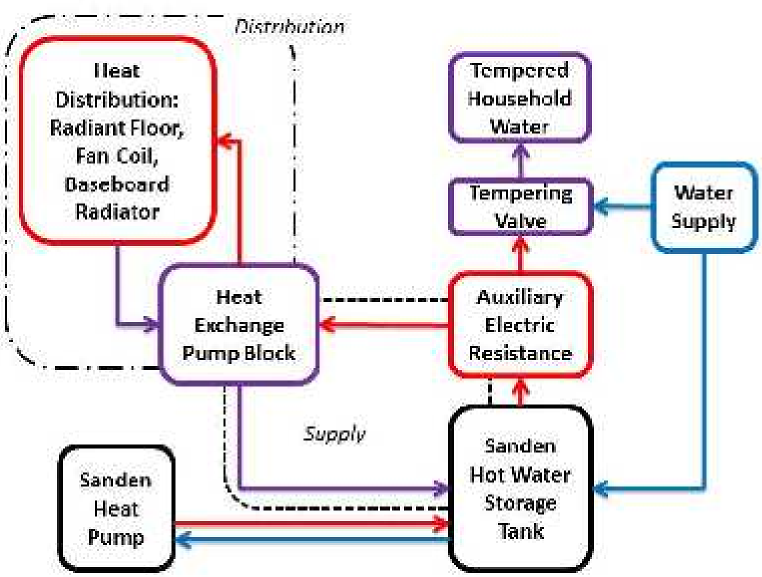
The main source of space and water heating was a Sanden GAUS-315EQTD, CO_2 refrigerant, split system HPWH equipped with an 83-gallon storage tank. An inverter-driven, variable-speed compressor, gas cooler (heat exchange from CO_2 transcritical gas refrigerant to water), evaporator, and water pump are located in the outdoor unit. Plumbing lines transport water between the tank and the outdoor unit.

The combined space and water heating system adds a heating loop to the HPWH. This heating loop consists of two parts:

- The supply side moves heated water from the tank to a heat exchanger, and
- The distribution loop delivers heat to the living space.

A device called the Taco X-Block contains the pumps, controls, and heat exchanger for the space heating loop in one integrated package. The system design also includes an instant electric heater to provide auxiliary heat. It is located between the tank and the heat exchanger. **Figure 4** shows a basic schematic of the combined system.

Figure 4. Schematic of the Combined Space and Water Heating System



A significant amount of system design – more aptly termed evolution – took place in the context of the Bellingham installation. The technical design committee consisted of Ken Eklund, WSU; Jonathan Heller Colin Grist and Ben Larson, Ecotope; Mark Jerome, CLEAResult; Charlie Stephens, NEEA; and John Miles, Sanden. Weekly calls to exchange ideas and make design decisions took place with frequent email traffic in between for several months in the fall of 2014.

The original system design proposed for the Bellingham site called for replacing the Sanden tank with a tank that had an integral heat exchanger. However, the Sanden tank is equipped with a precisely located sensor that alerts the system controller when to operate the outdoor unit and the system is carefully engineered to maintain stratification necessary for system operation. To ensure proper operation and maintain the heat pump system warranty, the original design was tabled.

A great deal of discussion took place concerning the best way to return water from the space heating distribution system to the Sanden tank. Sanden's concern was that returning warm water to the bottom of the tank would:

- Interfere with defrost function in cold weather, because warmer water causes the system to misread the temperature and turn off the defrost (this issue was solved in the UL listed system); and
- Reduce efficiency in operation, which depends on maintaining a temperature gradient in the tank to deliver cool water to the outside heat exchanger.

It was decided to return heating loop water to the top of the tank. This caused warm water to mix with hot, and resulted in some cool showers at the Bellingham site. A device called a diversion fitting was developed and built by WSU to direct the incoming warm water down toward the center of the tank so it could find its proper stratification level (**Figure 5**). A copy of this device is installed at five sites.

Figure 5. Diversion Fitting

The backup tank was equipped with heating elements to provide additional capacity if the HPWH could not provide sufficient hot water for space heating. After the Bellingham installation, the design team decided it would be simpler and better to use an electric resistance (ER) demand heater for backup. This was done in all subsequent installations except the Olympia and Portland sites, which have no backup heating due to owner preference. Further auxiliary heat issues surfaced later in the project and are discussed in a special section.

The Bellingham site was retrofitted in early October 2015 to move the heating loop return from the top of the Sanden tank to the bottom, and to replace the auxiliary tank with a demand heater. Relocation of the heating loop return was based on the combined space and water heating lab test conducted by Ecotope in August 2015, which showed clearly that returning 70°F to 80°F water to the bottom of the storage tank is more efficient than returning it to the top of the tank or introducing it through a diversion fitting. Five of the original ten sites have this configuration. This lab test is described later in this report. Diversion fittings were left at three sites with high temperature distribution systems and at two sites where the owners were not enthusiastic about the changes.

Auxiliary heat was originally designed only to serve the space heating loop. After reports of cool showers the hot water was also plumbed to take advantage of the backup heat.

Challenges in Monitoring

NEEA provided all of the monitoring equipment and supported the installation, calibration, and monitoring of that equipment by WSU. The monitoring used for all ten sites is the same as that used for the detailed monitoring done by NEEA in its first generation of NSH, including four of the NSH homes in this study. The equipment was designed primarily for use by homeowners to monitor energy use, and has been expanded through its use in the NSH program to provide a wide array of monitoring services. The monitoring equipment requires Internet access in order to operate, so the home must be occupied and have Internet installed and accessible before monitoring can be installed and commissioned. The equipment does not record data if it is not connected to the web, resulting in substantial loss of data.

The biggest monitoring challenges were flow meter accuracy and data gaps caused by the monitoring system. Calibration of flow meters on site using a micro-weir or an ultrasonic flow meter is recommended to test flow measurement and provide correction factors if needed. Loss of data by the monitoring system and by failure of Internet connections on which they depended was not expected, and this loss affected some sites more than others. Temperature sensors incorporated into the flow meters were also subject to failure; in some cases, plumbing and electrical system reconfiguration was not accompanied by monitoring adjustments. Notwithstanding these issues, usable data was available for over half the sites. The data analyzed in this report is carefully selected and filtered to provide accurate information representing all types of heat distribution systems in the study.

Data collection at the cold climate sites in McCall and Coeur d'Alene was limited by the cessation of proper function of the outdoor unit defrost cycle after below-freezing weather set in. Ultimately the sites were decommissioned. This is discussed in more detail in the results section of this report.

Field Study Details

Site Summaries

The test sites are typical of the regional heating zones they represent, as shown in **Table 1**. Most Heating Zone 1 sites are warmer than the median value for that zone but represent the most populated areas in the region. Bellingham and Olympia are colder than the median. Coeur d'Alene is a solid representative of Heating Zone 2 and McCall is colder than the Heating Zone 3 median. Characteristics of the test sites are summarized in **Table 2**.

Table 1. Heating Zones of Ten Test Sites

Heating Zone	Number of Sites	Median HDD_{65} [1]	Site Location	Site HDD_{65}
Heating Zone 1	1	5,182	Milwaukie, OR	4,461
Heating Zone 1	3	5,182	Seattle, WA	4,867
Heating Zone 1	1	5,182	Tacoma, WA	4,696
Heating Zone 1	1	5,182	Bellingham, WA	5,622
Heating Zone 1	1	5,182	Olympia, WA	5,655
Heating Zone 2	1	6,824	Coeur d'Alene, ID	6,239
Heating Zone 3	2	8,363	McCall, ID	8,851

Table 2. Test Site Characteristics

Site #*	1	2	3	4	5	6	7	8	9	10
HDD	5,622	6,239	8,851	8,851	5,655	4,461	4,867	4,867	4,867	4,696
Design T	19	-1	-16	-16	23	24	27	27	27	24
Conditional floor area	2,057	1,062	2,812	1,533	1,152	2,000	2,218	1,936	3,764	1,538
Heating load Btu/hour	13,098	11,760	28,864	21,061	12,430	6,226	10,285	8,516	10,853	11,007
Dist. system**	RF	RF	RF	RF	RP	RP	RF	RF	RF	RFF+FC
Tstat heating set point	67	73	67	63	70	60	71	73	68	73
DHW T°F	120	120	122	120	120	130	120	130	120	120
# Occupants	4	2	2	0	2	0	3	2	4	7

* 1=Bellingham, 2=Coeur d'Alene, 3=McCall, 4=McCall, 5=Olympia, 6=Portland, 7=Seattle, 8=Seattle, 9=Seattle, and 10=Tacoma

** RF = radiant floor, RP = radiant panel, and RFF+FC = radiant first floor and fan coils on second floor

[1] Source: Northwest Power and Conservation Council, 6th Power Plan Assumptions

Monitoring Setup

The main monitoring collection device was a SiteSage Energy Monitor with internet connection so data can be downloaded and settings on the logger can be controlled remotely. Temperature and flow information and electrical use data were collected. All data was taken at 1-minute intervals. A schematic of this monitoring system is provided in **Figure 6**. The following monitoring equipment was used:

- Emonitor + Gateway
- INDAC sensor controller
- (2) Temperature + % relative humidity (RH) 1-wire sensors (indoor and outdoor)
- (3) Temperature wells with 1-wire temperature sensors
- (2) Grundfos flow sensors + temp model VFS 2-40
- (1) Grundfos flow sensor +temp model VFS 1-20

In some instances, on-site HOBO link® monitoring was also required to capture all data streams. These data are downloaded manually at several-month increments.

The measurements recorded by the monitoring system are listed below. Please note the code names that match the identification of each channel on the schematic:

Water flow, time, and volume (FM = flow meter)

- Through hot water tank measured at the cold water inlet (FM-2)
- Through space heating supply loop measured on return to tank (FM-1)
- Through space heat distribution loop measured on return to heat exchanger (FM-3)

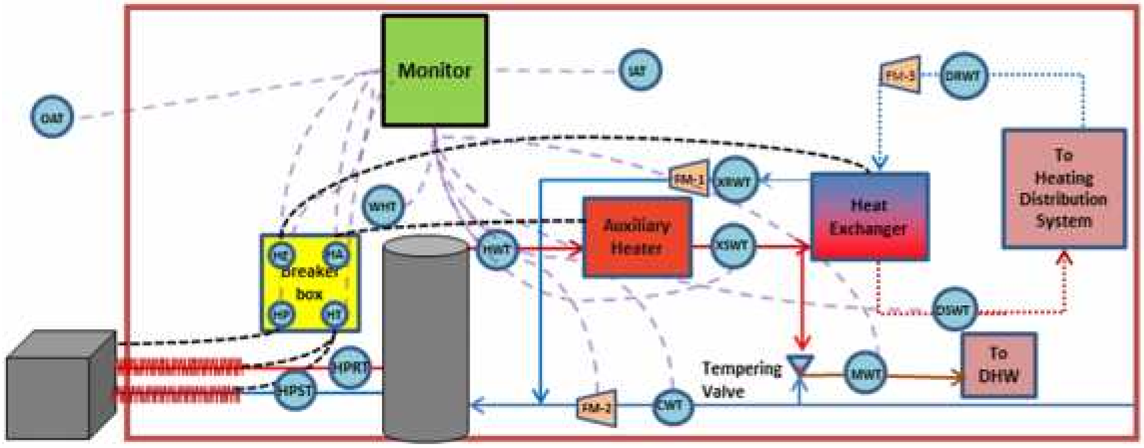
Temperatures

- Cold water supply (CWT)
- Hot water to auxiliary heater (HWT)
- Tempered water to house (MWT)
- Outside air temperature (OAT)
- Inside air temperature near the hot water tank (WHT)
- Inside air temperature in conditioned space (IAT)
- Hot water to heat exchanger and tempering valve (XSWT)
- Return water from heat exchanger to hot water tank (XRWT)
- Hot water to heating distribution system (DSWT)
- Return water from heating distribution to heat exchanger (DRWT)
- Temperature of water supplied from the tank to the heat pump (HPST)
- Temperature of water returned from the heat pump to the tank (HPRT)

Power measurements

- Time and amperage of outdoor compressor unit (compressor, fan, and pump) (HP)
- Time and amperage of outdoor pipe freeze protection (heat tape) electricity use (HT)
- Time and amperage of backup heating loop electricity use (at all but two sites) (HA)
- Time and amperage of heat exchange supply and distribution pumps and controllers (HX)

Figure 6. Field Monitoring Setup



Field Study Data Analyses

The period covered by this analysis is from the time monitoring began at the Bellingham site on December 30, 2014 through June 30, 2016.

The analysis examined the performance of the system for both space and water heating, and a number of its operating parameters, including: the temperature of the system cold water supply, heated water, and tempered water; and the calculated volume of water used to temper the hot water before use. The total volume of water used and daily use averages were also calculated for domestic hot water (DHW). In addition, the characteristics of the space heating loop were examined for temperatures, operating parameters, and energy used under representative conditions.

Domestic Hot Water

Calculating DHW use requires the following elements:

- Average temperatures by flow event or by day for cold water supply, hot water, and tempered water for the DHW supply.
- Thermal energy required to heat cold supply water for each flow event.
- Volume of water added to temper hot water for each flow event.
- Volume of total water for each flow event.

To calculate accurate temperatures for cold supply water, hot water, and tempered water for DHW, at least 3 minutes of consecutive flow was required. Temperatures were then calculated by dropping the initial reading and averaging over the remaining readings for a given flow event (or draw). Daily averages were used as the representative temperatures for short-duration draws that were less than 3 consecutive minutes. When only short draws occurred during a given day, the daily average water temperatures from adjacent days were used.

Only water volume flowing into and out of the HPWH tank was metered via data loggers, so additional water added to temper the hot water was calculated for each flow event by using the known water flow (gallons) and the difference between the average daily tempered water flow and the average daily cold or hot water temperatures, respectively. Total tempered water flow for each flow event was the sum of the cold water flow and the added water.

Average water temperatures were used to calculate the thermal energy needed to heat the cold water for each draw. The energy is calculated via the familiar calorimetric equation shown below where p is the density and C_p is the heat capacity of water.

Equation 1: Energy = Volume x p x C_p x (Temperature 1 - Temperature 2)

In the specific case of DHW use, the energy in Btu is defined as Qdhw, Temperature 1 is the tank outlet (HWT), and Temperature 2 is the tank inlet (CWT) temperature.

Space Heat

The relevant energy values for the space heating system were calculated using Equation 1 but with values substituted as shown in **Table 3**.

Table 3. Measured Flow and Average Temperature Values Used to Calculate System Loads

Calculated Variable	Flow Volume	Temperature 1	Temperature 2
Qaux	Supply return after heat exchange (FM-1)	Auxiliary heat outlet (XSWT)	Hot water from tank (HWT)
Qsystem	Supply return after heat exchange (FM-1)	Hot water from tank (HWT)	Supply return after heat exchange (XRWT)
Qdistribution	Distribution return before heat exchange (FM-3)	Distribution after heat exchange (DSWT)	Distribution return before heat exchange (DRWT)

Overall System Efficiencies

Water heating is rated with Energy Factors; space heating is rated by Coefficient of Performance (COP) or Heating Season Performance Factor (HSPF). The combined system performance has been designated as a Field Energy Factor (FEF). This accounts for all system inefficiencies such as tank loss, pipe loss, pump energy, controls, defrost, and freeze protection. FEF efficiencies are calculated as:

Equation 2: FEF = (Qdhw + Qsystem) / Qinput

where Qinput is the sum of energy inputs to the HPWH (HP), auxiliary heat (HA), heat exchanger block (HX), and heat tape (HT).

When data was unavailable for the supply side of the heat exchanger, an FEF was calculated using data from the distribution side of the system:

Equation 3: FEFdis = (Qdhw + Qdistribution) / Qinput.

Space and Water Heating Efficiencies

Given that heat is simultaneously provided by one heat source through a single tank for both space and water heating, it is impossible to calculate a definitive efficiency for each end use. This is particularly true for a heat pump because its efficiency varies with OAT, supply water temperature, and load. Thus, a period of water heating only during the summer cannot be used to determine its portion of the load in winter. The lab test was designed to quantify the individual efficiencies for space and water heating as well as combined function efficiencies.

Field Study Results

The project began October 1, 2014, with the goal to conduct a field study on six new homes. Recruitment was successful and a total of nine new homes plus a major thermal remodel comprised the final cohort. These homes were completed over a period of a year and monitored as they were finished. A great deal was learned about system design and performance, which resulted in changes to the system plumbing at some sites during the monitoring period.

Auxiliary heat strategy: The first site had an ER tank for auxiliary heat. Monitoring showed that most of the minimal auxiliary energy at this site was used to keep the tank warm. A demand electric water heater then became the standard design. Eight sites adopted this system and two sites have no auxiliary heat.

Heating supply water return location: Potable water is taken from the bottom of the 84-gallon tank to the outdoor unit, where it is heated and then delivered to the top of the tank. Hot water is taken from the top of the tank for both DHW and space heat. At the first site, the return water from the radiant floor, averaging 83°F, was initially returned to the top of the tank. On cold days the home occupants had cool showers due to mixing of this cool return with the hot water.

An additional concern in determining return water location was the warning by the heat pump manufacturer that both efficiency and defrost function depended on cold water supply to the heat pump, making it vital to maintain tank stratification. The heating system return water was cooler than the 149°F water at the top of the tank, but hotter than the normal cold water supply. The ideal location for the return was thought to be in the central portion of the tank, but no port was available. A fitting to divert heating supply return water to the center of the tank was installed at Site 1, where it cured the cold showers. This strategy was adopted at the next six sites.

Subsequently, a lab test was conducted to compare the impact on tank temperature stratification of three different return strategies: top of the tank, top of the tank with diversion fitting, and bottom of the tank. The best location among these choices for maintaining tank stratification with lower temperature distribution systems (radiant floors) was found to be at the bottom of the tank; second best was the diversion fitting; and third was the top of the tank. For high temperature systems (radiators) the bottom entry and the diversion fitting perform almost the same. The ultimate recommendation was that a tank designed for combined systems should have multiple ports to allow installers to match the return to the proper temperature level in the tank. The two sites constructed after this finding had return water from the heating system plumbed to the bottom of the tank, and the plumbing was revised at three existing sites – all with radiant floors – to implement this design change.

It should be noted that the need for cold water supply to optimize performance of the heat pump is incompatible with strategies to preheat the supply water. Site 1 had such effective pre-heating strategies, that its supply water was often hotter than the return water from the heating system. This is part of the reason for its reduced system performance.

Auxiliary heat for DHW: The original system design provided auxiliary heat only to the space heating system. As sites in colder locations came online, home occupants experienced cool showers when space heat was operating. Five of the sites were re-plumbed to connect the DHW to the electric demand auxiliary heat source. Site 9 had auxiliary heat only for DHW while Sites 5 and 6 had no backup heat.

Monitoring combined systems: Several challenges in monitoring the systems limited the data set available for analysis:

- This is the first time scientific monitoring has been done with the monitoring system used, and many days of data were lost due to data collection issues. The system was used, because it was part of an effort by NEEA, which provided the equipment and its installation, to develop low cost monitoring options.
- The system plumbing revisions resulted in loss of data and changed operation. Moving the heating return to the bottom of the tank caused the temperature sensor, which was integrated into the flow meter, to end up on the upstream side of the return entry point, resulting in loss of the incoming water temperature at three sites.
- Some temperature sensors and flow meters malfunctioned, preventing calculation of key variables, and it was difficult to obtain replacement parts.
- The monitoring required Internet service to collect and store data, and the provider cut service at Site 1 in November, 2015. It took four months to fully restore service.

The resulting analysis was conducted on sites that had complete data sets for the periods analyzed, and data were screened to ensure that periods with missing data were not used. Sites were excluded from the analysis because of failures in the systems. The sites used in the analysis are 1, 4, 5, 6, 7, and 10 which represent all distribution system types and two climate locations.

Daily DHW Use by Site

Average daily hot water use in the Pacific Northwest is approximately 15 gallons per person per day (totaling 45 GPD for a family of three), as illustrated in **Figure 7**. Several of the sites have water use lower than average. Site 10 used substantially more hot water than other sites. Sites 4 and 6 were unoccupied during monitoring, so any domestic hot water use was related to construction cleanup.

Daily Average Outside Air Temperature by Site

Sites 1, 5, 6, 7, and 10 are in the Maritime Northwest; Site 4 is in McCall, ID, a cold location, and one of the last systems to come online where it operated only a short time. The longest-term location is Site 1 in Bellingham. Its data flow was interrupted when the Internet provider cut service. These findings are shown in **Figure 8**.

Daily Heat Pump Energy Use (kWh) by Site

Figure 9 shows the energy use by site. Site 4 in McCall, ID, the coldest location, shows the highest daily energy use in the 40 kWh per day range. Site 6 in Portland, OR, shows much lower energy use during the same period, with a high of 20 kWh per day. These sites were both unoccupied during the monitoring period and, therefore, all heat pump energy use is for space heating. Regardless of the OAT, the systems were able to operate and produce heat. At all the sites in the coastal climates, the systems – including auxiliary heat – were able to provide space and water heating. At the very cold location for Sites 3 and 4, a larger-capacity heat pump would be an asset.

Figure 7. Daily Tempered Water Flow

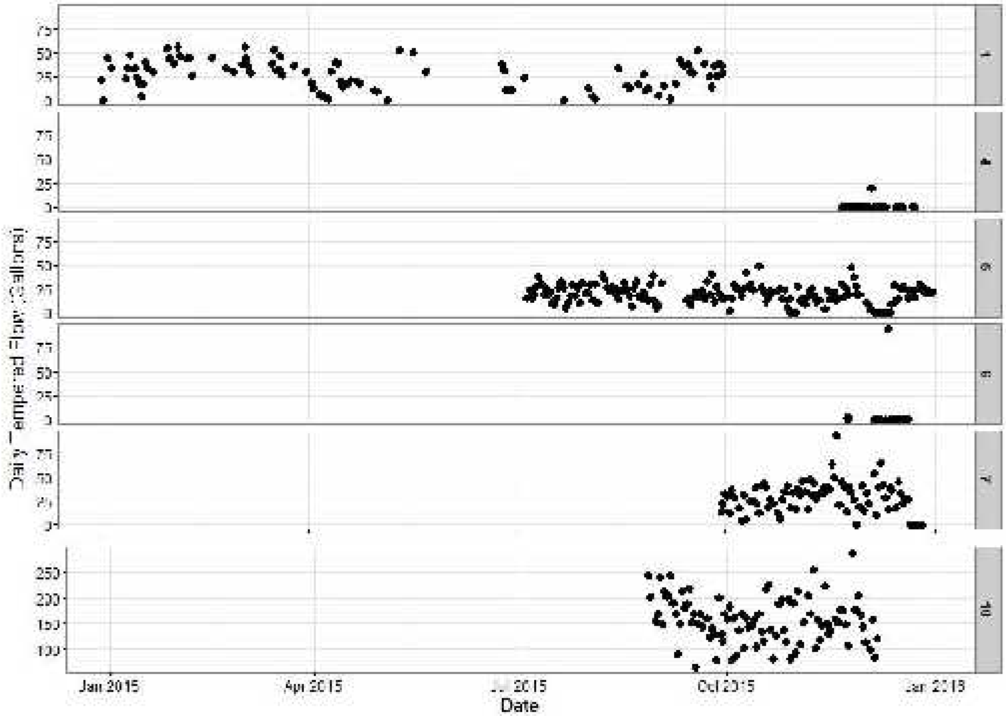


Figure 8. Daily Average OAT

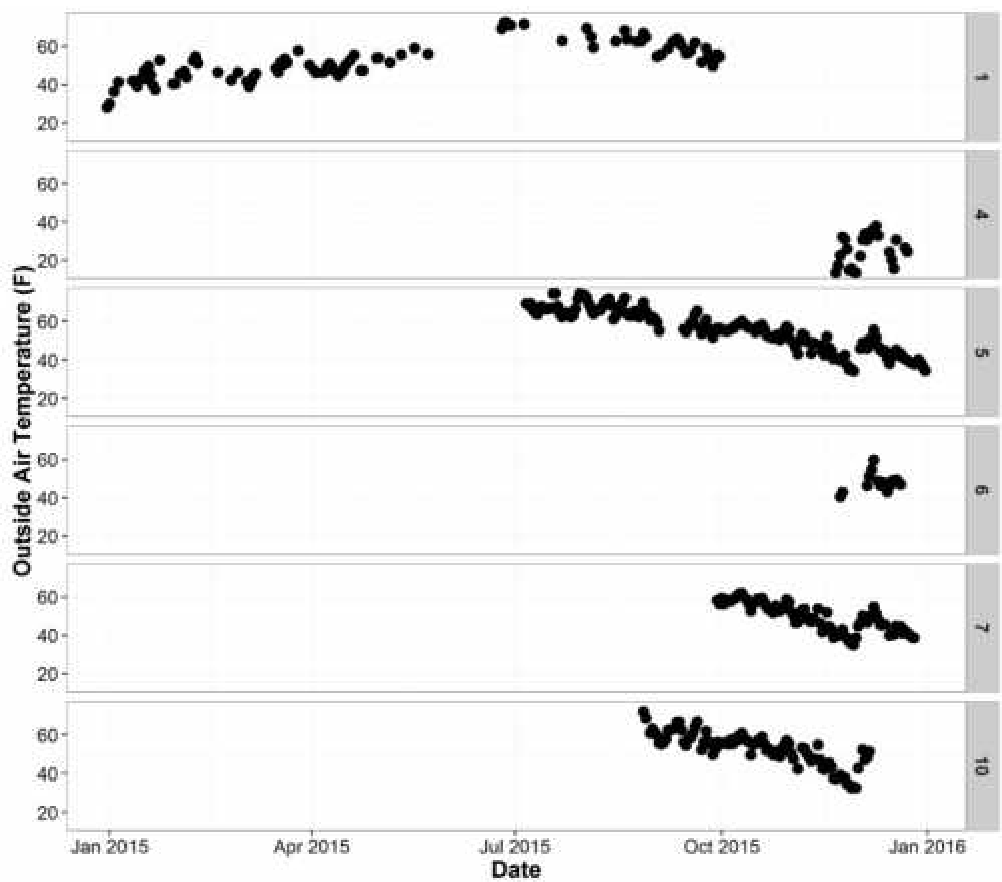


Figure 9. Daily Heat Pump Energy Use by Site

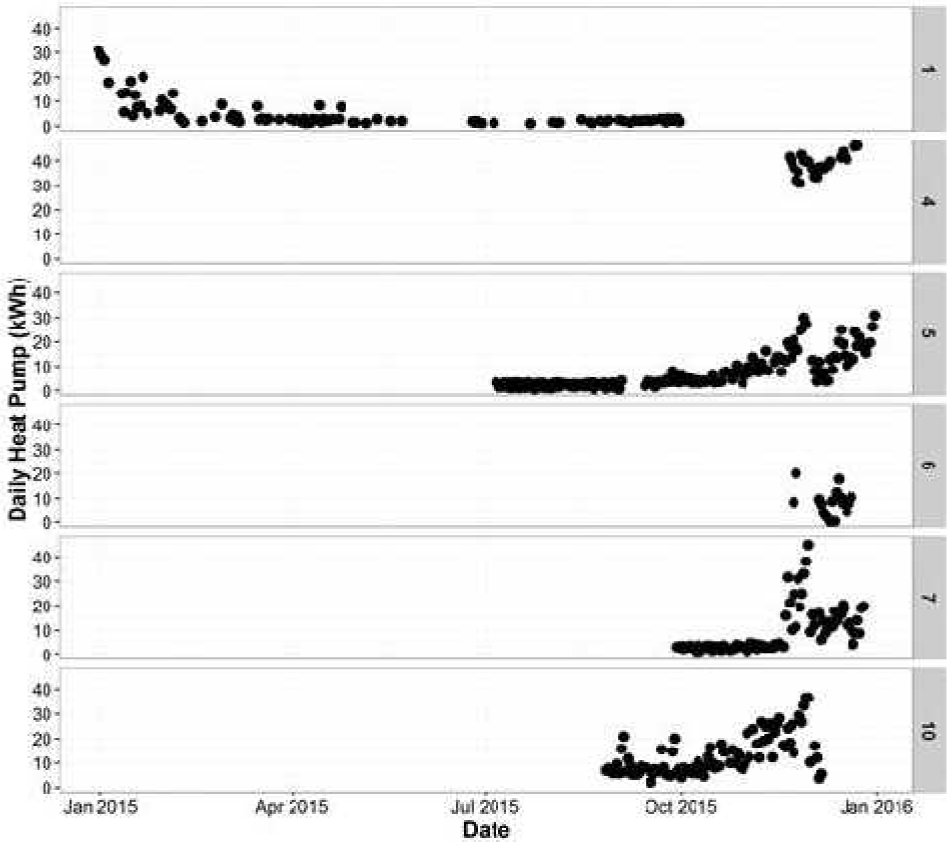
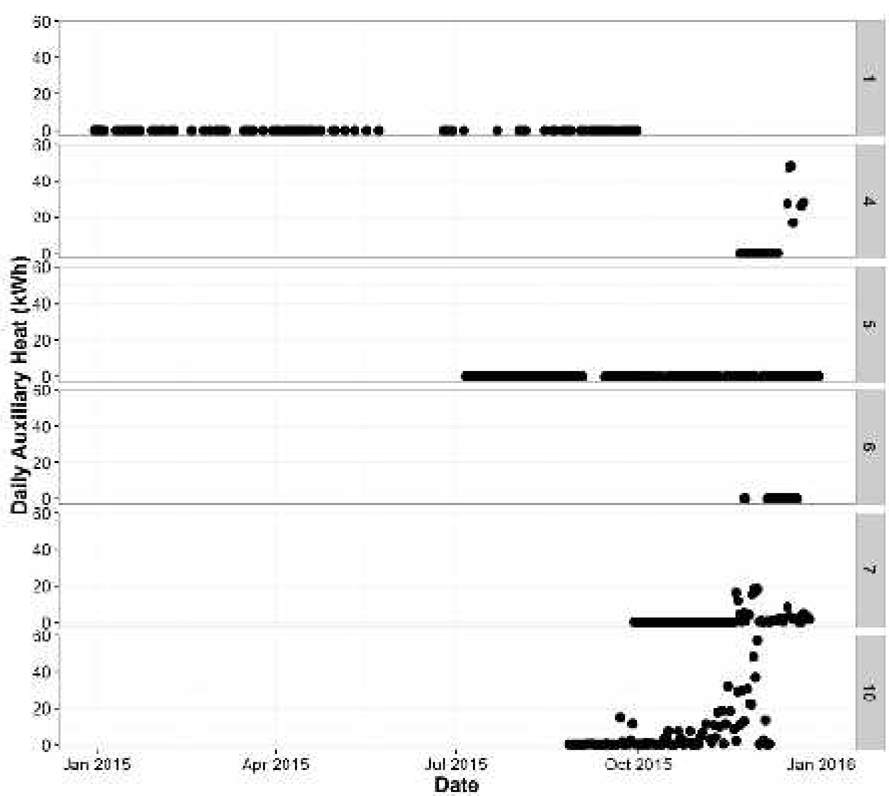


Figure 10. Daily Auxiliary Heat Energy by Site



Daily Auxiliary Heat Energy Use (kWh) by Site

The highest auxiliary use was at Site 4, the coldest site, and Site 10, which had the highest occupancy and a high-temperature fan coil system on the second floor. Sites 5 and 6 did not have auxiliary heat. These findings are shown in **Figure 10**.

Daily Field Energy Factor by Outside Air Temperature

Figure 11 shows the daily FEF for the analyzed sites arranged by OAT. Daily data for the heating season (October 1 to March 15) and non-heating season were averaged to examine seasonal differences for distinct system types (see Table 4: "H" for heating and "NH" for non- heating). Select sites, most of which have more than 30 days of sampled data from a given season, are presented. The combined space and water heating efficiencies vary according to temperature and other variables, such as DHW use. The most interesting comparison is between Site 5 and Sites 1 and 7. Site 5 has hydronic radiators for distribution, and Sites 1 and 7 have radiant floors.

Figure 11. Daily Field Energy Factor (including freeze protection)

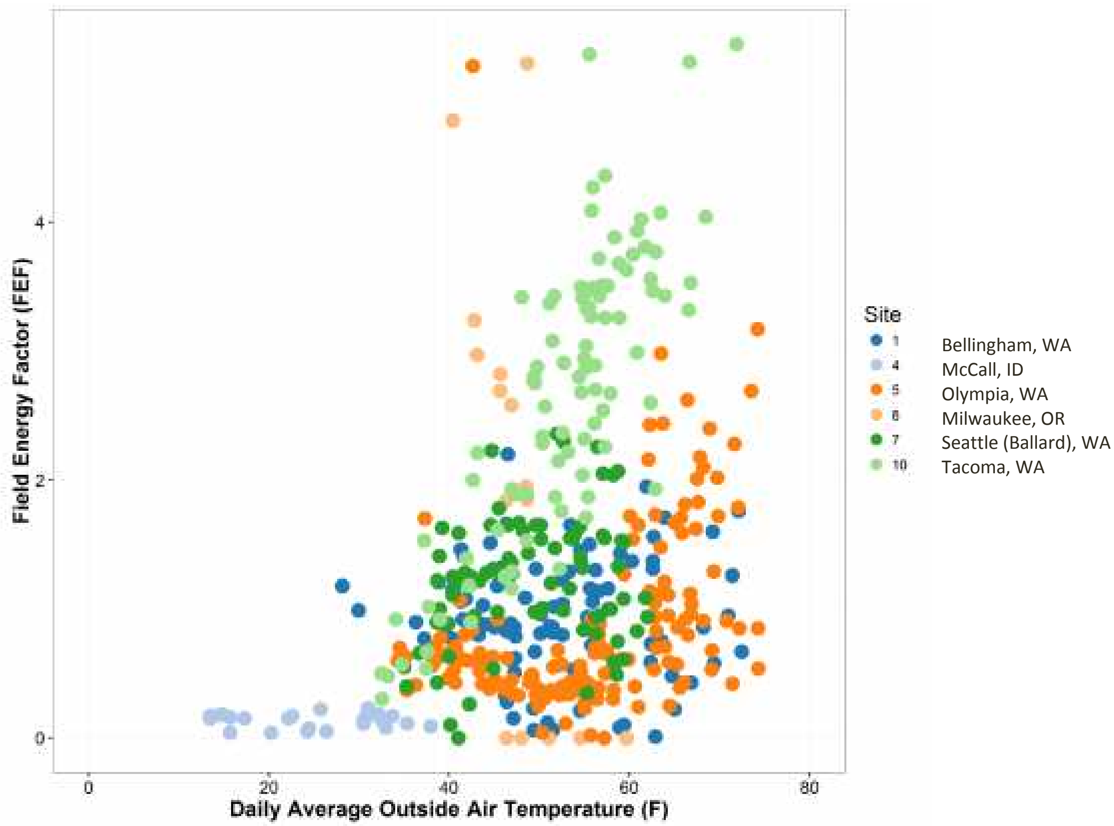


Table 4 shows seasonal averages for key factors that impact performance on sites with both space heat and hot water use. The systems with the high return loop temperatures (XRWT) have radiators (Site 5) or fan coils (Site 10).

Table 4. Average Daily Values for Heating (H) and Non-heating (NH) Seasons for Select Sites

Site	OAT (F)		CWT (F)		XRWT (F)	DHW (GPD)		FEF		Days Sampled	
	H	NH	H	NH	H	H	NH	H	NH	H	NH
1	43.1	56.3	76.34	77.51	82.93	34.28	23.46	1.04	0.88	31	65
5	47.47	64.11	60.21	69.93	111.4	17.87	21.02	0.58	1.18	83	75
7	48.94	57.25	60.37	67.06	89.7	28.57	18.3	1.24	0.76	80	2
10	49.12	59.64	58.12	58.99	101.43	151.91	167.31	2.28	3.35	60	33

The system with the lowest average heating season performance in Table 4 is Site 5, which has the highest return water temperature. Its FEF doubled during the non-heating season, due in large part to the reduction in supply water temperature going to the outdoor unit.

The non-heating season average FEFs for Sites 1 and 7 are lower than those for the heating season. This appears to be related to the drop in daily water use at these sites. (Although there are only two non-heating season days in this sample for Site 7, it is considered instructive on this point.) A contributing reason is that tank and pipe losses continue while there is less useful energy delivered to allocate it to. At sites 5 and 10, daily hot water use and FEF increased during the non-heating season.

The large daily water use at Site 10 coincides with the only outstanding performance in this sample; this performance was despite the fact that its system operated at a higher return loop temperature than sites with only radiant floors. Hot water use brings cold water into the storage tank, which results in colder water going to the heat exchange with the refrigerant in the outdoor unit resulting in higher heat transfer. The cold water also reduces tank loss.

Energy Use

The amount of energy used at the sites is another way to look at the data. **Table 5** contains information on energy inputs into the systems for which data in both heating and non-heating seasons is available. In addition, two systems have been added that have data in either the heating or non-heating season.

Table 5. Energy Inputs by Function

Site	Site Location	Season	HP kWh	Sys kWh	Aux kWh	# Days	Full Season	Total Annual	Std. HP & HPWH kWh
1	Bellingham	Heat	305	24	1	31	1,769		
1	Bellingham	Nonheat	160	8	1	65	516	2,285	3,110
5	Olympia	Heat	977	224	0	83	2,403		
5	Olympia	Nonheat	184	35	0	75	481	2,884	1,905
6	Milwaukie	Heat	133	13	0	16	1,519		1,442
7	Seattle. Ballard	Heat	756	23	154	80	1,334		
7	Seattle. Ballard	Nonheat	6	0.1	0	2	647	2,581	2,749
9	Seattle. Madrona	Nonheat	156	0	10	65	512		669
10	Tacoma	Heat	930	62	526	60	4,200		
10	Tacoma	Nonheat	277	22	40	33	2,042	6,242	8,192

Table 5 shows seasonal energy use at six sites: the electricity used by the heat pump, the distribution system and the auxiliary heater, the number of days of clean data, the energy use extrapolated to a whole season, and the sum of the heating and non heating seasons to a total annual estimate. The column on the far right is the modeled annual energy use of a standard air source heat pump and HPWH for comparison purposes.

The energy usage contains interesting facts that are not apparent in the FEF numbers. The energy used by the system (Sys kWh) is the electricity used by the Taco X-Block, which contains the heat distribution pumps, heat exchanger, and controls, plus any other system controls and operating devices such as zone control valves. On most of the systems, the system energy use is a small part of the total energy used.

Site 5, where the heating season system energy use is 19% of the total, is the exception. Where only water heating is occurring, no system energy is used because it pertains to space heat. If system energy is seen in the non-heat season, it indicates that heating took place during that time.

Auxiliary energy use for space heating is not possible at three sites. Sites 5, 6, and 9 do not have backup space heat, although Site 9 has an auxiliary heater on the DHW line. The largest auxiliary heat use is at Site 10, which has a radiant floor downstairs and four fan coils upstairs in the bedrooms. It has seven water users and hot water use that is almost four times the average regional volume. Note that Site 10 uses auxiliary heat in both the heating and non-heating seasons. The system, auxiliary, and heat pump energy uses are all included in the FEF calculation, and Site 10 still has the highest performance of all the sites with 2.28 heating FEF and 3.35 non-heating season performance.

The heat pump energy use is highest at Sites 5 and 10. Heating is the main cause at Site 5, which has the highest system energy use and return loop temperature due to radiant heat distribution. Hot water use is probably the main factor at Site 10, with seven users all using more than the average use per person.

The energy use totals for each season were reduced to daily values that were extrapolated to seasonal results shown in the column labeled “Full Season” and summed to annual totals where data for both seasons were available and shown in the column labeled “Total Annual”. These are conservative estimates because both data sets generally represent the coldest part of the season.

This energy data invites comparison to the annual use of more conventional heat pumps and the column labeled “Std. HP & HPWH kWh” in Table 5 shows this. Using SEEM, Ecotope’s Simple Energy and Enthalpy Model, Version 97, to simulate the Site 1 house in Bellingham using an HSPF 9 air source heat pump and a unitary HPWH, the modeled space heat input is 1,869 kWh with TMY3 data adjusted for the mild winter of 2015-16. This provides a direct comparison for Site 1 and was adjusted using degree days and conditioned space area to estimate standard heat pump comparison loads at the other sites. The hot water comparison was adjusted for each site using the average energy per gallon for the unitary HPWH and the actual hot water consumption (*ibid*). At Site 10, which used an average of 160 gallons of hot water a day, four times the regionally monitored average, the unitary HPWH would use approximately 7,000 kWh per year to heat this water at its measured efficiency. This is impressive considering that an electric resistance water heater would use 12,800 kWh.

The advantage of combined systems can be seen in the data provided in Table 5. Even with performance that is lower than expected, most of the combined systems compare favorably with systems using discrete heating and hot water systems. This may be due to several factors: the generally lower energy use by hydronic distribution systems; the low auxiliary heat use by a heat pump with no integrated backup system; and the fact that the end uses gain thermal advantage from using a common tank, piping, and heat source.

The main implication is that the technology is promising, but these systems require significant development to increase average overall performance. This report now moves to issues that impacted the performance and a lab test conducted by Ecotope that shows the level of performance that can be expected with this type of technology in different climates if they are optimized.

Issues Impacting System Performance

A number of issues impacted performance of the combined systems in the study. Discovering issues is expected when researching a brand new system made up of components repurposed from the original uses. The goal is to learn enough to determine what needs to be done to enhance the system so that it runs efficiently and can be easily installed.

The issues began to appear as soon as the first system was installed (this discussion is presented on pages 5 and 6). More issues surfaced as more systems were installed. This section brings together all of the performance issues and the action or resolution taken. The main issues that were examined are:

- Defrost failure caused systems in cold climates to undergo repeated manual defrost to continue system operation. Systems in McCall and Coeur d'Alene, ID, were removed at the homeowner's request.
- Power-out freezing caused one system in McCall to be disabled and shut down during a 10-hour power outage at 20°F.
- Systems worked best where design load was within heat pump limits.
- Standard programming for combined heat exchange, control, and pump (Taco X-Block) did not operate the system properly.
- Tank destratification occurred, especially in cold climates and with high-temperature heating systems, which reduced efficiency.
- Cross flow through tempering valves resulted in reduced operating efficiency.
- The auxiliary demand heaters developed water leaks at several sites.
- Cold Water preheating is incompatible with optimum hydronic heat pump performance.

Defrost

Sanden warned that the defrost system on the outdoor unit would be disabled by water above 100°F supplied to the outdoor unit. Efforts were made to keep the return water as cool as possible, especially at the cold temperature sites. Radiant concrete floors generally returned water at 90°F or lower. For reasons that are not altogether clear, the defrost logic on the heat pump was tricked into not defrosting the unit as it would if only water heating was taking place at the sites in McCall and Coeur d'Alene. The causes may have been cross flow or tank destratification delivering high-temperature water to the heat exchanger in the outdoor unit.

Sanden has squarely faced this defrost issue and redesigned the heat pump operation logic to allow defrost operation regardless of the temperature of the water entering the outdoor unit. The UL listed unit will soon contain this change and other design improvements.

Power-Out Freeze Protection

A system to protect the outdoor water lines between the tank and outdoor unit, the inner piping, water pump, and heat exchanger in the outdoor unit from freezing during a power outage in cold weather was recommended in the final report for TIP 292. It was to be installed at the cold weather sites in this project, but before it could be done, one of the sites in McCall was subjected to 10 hours of 20°F temperature without electricity. The water pump in the outdoor unit cracked due to ice expansion, requiring system shutdown. The other system in McCall was not impacted.

A solenoid automatic drain system has been developed by Sanden for installation in cold climates. It has been tested as part of a marketability study funded by BPA for WSU to conduct on the UL listed system. The solenoid drains the system, does not cause air locks, and uses negligible electricity. A full report on this and other tests will be issued in early 2017 by Ecotope and WSU.

Capacity

System capacity – the output in Btu per hour – is relatively constant, with the main determinant being OAT. That output is 13,000 Btu per hour below freezing and 15,000 Btu per hour at higher temperatures. An example is provided by two unoccupied homes with the combined systems that were heated during the same period in winter 2015:

- Site 4 is located in McCall, ID, which has a design temperature of -16°F and a design load of 21,061 Btu per hour.
- Site 6 is located in Milwaukie, OR, with a design temperature of 24°F and a design load of 6,226 Btu per hour.

Figure 12 shows the daily heat pump electricity usage during the period from November 21 through December 21, 2015. The data is not normalized to weather, but provides a comparison of the capacity demands during the same window of time when both sites were in space heating-only mode.

Figure 12. Daily Heat Pump Electricity Usage

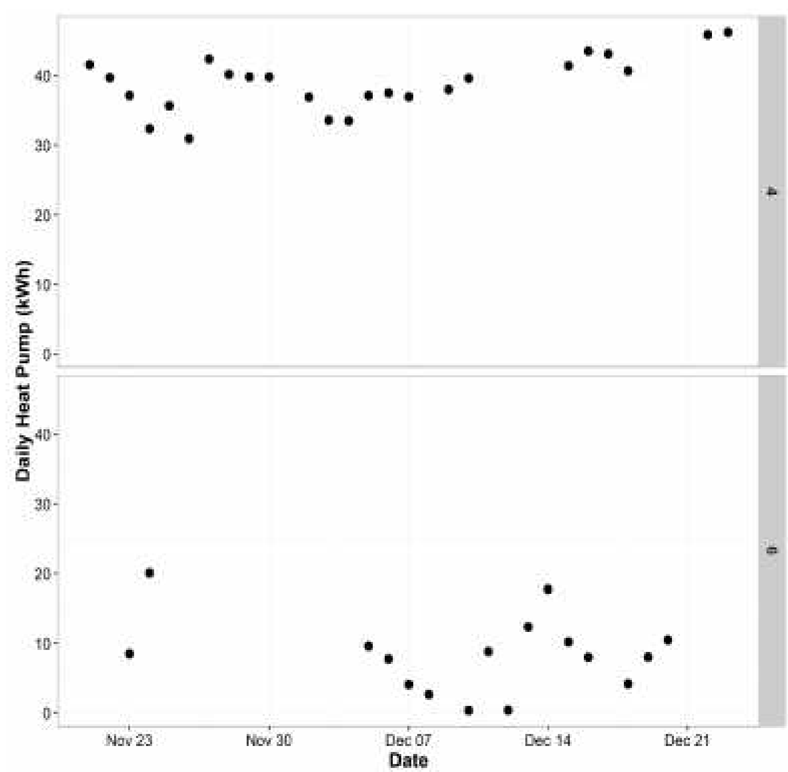
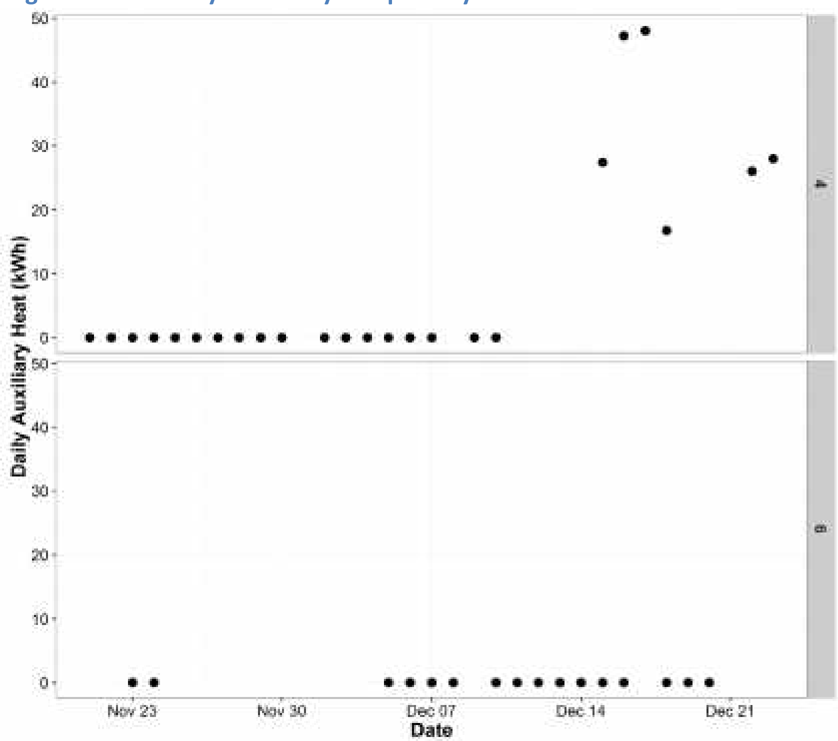


Figure 13 shows the auxiliary electricity use per day at each of the two sites. Note that the heat pump handled the load at Site 4 until mid-December, when backup energy use soared to almost 50 kWh per day. The heat pump and backup heat operated simultaneously as the temperature plunged from a mean temperature of 28°F and a low of 24°F on December 14, to a mean of 8°F with a low of -8°F on December 17. The heat pump began having defrost issues as the tank destratified and higher temperature water was sent to the outdoor unit. It had to be turned off and manually defrosted which took it out of commission and put the entire load on the auxiliary heat.

Figure 13. Auxiliary Electricity Use per Day



The performance of each system is compared in **Table 6**. Note that the average OAT at Site 4 was close to 25°F during the period, while at Site 6 it was almost 48°F. The supply water temperature was also much colder at Site 4. Interestingly, the heating system at Site 4 returned much colder water to the tank than the Site 6 system, showing the performance difference between the radiant floor and radiant panels. Lower temperatures indicate better performance, but when the overall performance is compared, the Site 6 system shows an FEF over 2 while the Site 4 system was 0.13, clearly indicating the system was not capable of performing adequately under the circumstances to which it was subjected at Site 4. Note that the FEF includes the auxiliary heat.

Table 6. Performance of Each System

Site	OAT (°F)	CWT (°F)	XRWT (°F)	FEF	Days sampled
4	24.89	50.3	79.85	0.13	24
6	47.83	62.71	105.99	2.05	16

At Site 6, the 15,000 Btu capacity heat pump was more than twice the design load, while the Site 4 load exceeded its 13,000 Btu capacity by about 8,000 Btu. With backup heat, the Site 4 system capacity was 36,884 Btu per hour (almost twice the design load), but this did not solve the capacity issue because the heat pump operated at all times if not turned off manually, regardless of its ability to operate effectively. The best solution for very cold climates appears to be a heat pump that better matches the design load and is not at the mercy of tank destratification.

The heat pump at Site 4 was destroyed by freezing during a power outage in late December 2015. It was replaced by a 28 kW Seisco instant electric heater, which is providing all space and water heat until a larger CO_2 hydronic heat pump is available.

Taco X-Block Programming

The X-Block by Taco is an integrated heat exchanger, pump system, and controller that exchanges heat from the source fluid to the working fluid that serves the heat load. It performs that function at all of the research sites. Many plumbers and heating contractors automatically program the X-Block to operate as it would with a gas boiler. This destroys the efficiency of a hydronic heat pump by increasing flow rates and inducing tank destratification. During fall 2015, five systems came online, and the X-Blocks at all of these sites required reprogramming.

The recommended setup is to use Outdoor Reset, which requires an outdoor air temperature sensor. This allows the system to vary the heating delivery temperature to match outdoor conditions. The system should also be set up to enable rather than control the heat source. Heat source protection is designed for condensing boilers, and should be turned off.

Programming the X-Block requires moving through a series of screens in sequence. **Table 7** shows the recommended X-Block programming for a hydronic heat pump at three locations representing the coldest (McCall), cold (Spokane), and moderate (Olympia) climates, according to source at Taco.

Table 7. Recommended X-Block Programming for Hydronic Heat Pump at Three Locations

				For McCall	For Spokane	For Olympia
VIEW	MIX	Targ		"- - -	"- - -	"- - -
VIEW	MIX	SUPP	DEM	77	77	77
VIEW	MIX	Targ	DEM	83	83	83
VIEW	BOIL	MIN	DEM	OFF	OFF	OFF
VIEW	OUT		DEM	37	37	37
ADJUST	OUT	DSGN	DEM	0	10	20
ADJUST	MIX	DSGN	DEM	100	100	100
ADJUST	MIX	MAX	DEM	115	115	115
ADJUST	MIX	MIN	DEM	OFF	OFF	OFF
ADJUST	BOIL	MIN	DEM	OFF	OFF	OFF
ADJUST			WWSD	70	70	70
	DEM					
ADJUST	DEM					

If programming fails to provide improved function, the sensor should be checked for accuracy. A sensor reading high can shut down the supply pump when heat is needed by falsely showing the mix design temperature is met.

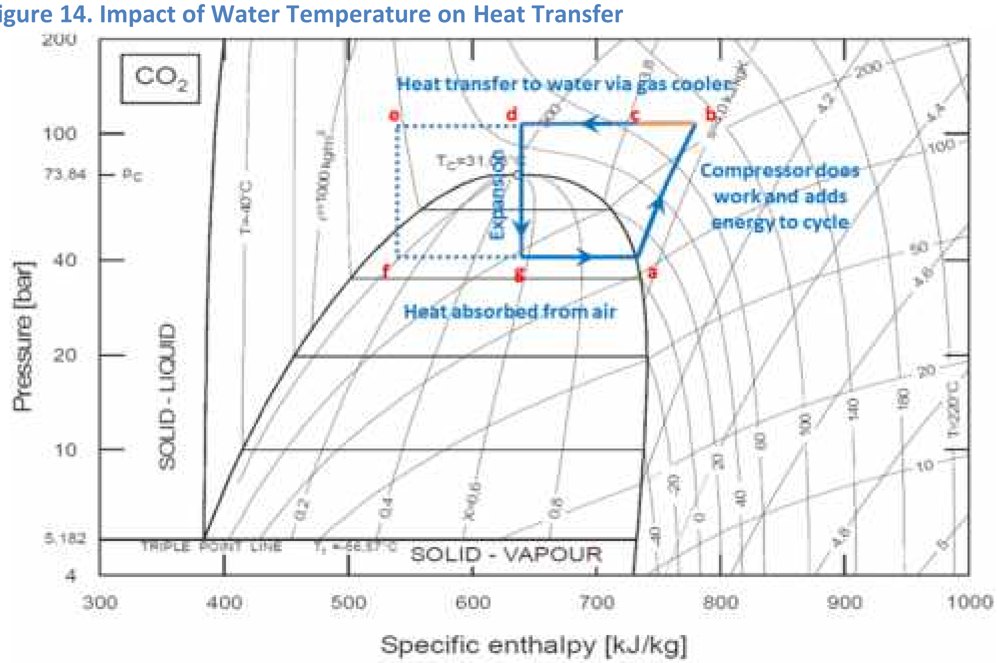
Destratification

Proper function of the split system depends on tank stratification, where cold water resides at the bottom of the tank and hot water is placed at the top. This allows the transcritical CO_2 refrigeration cycle to perform as designed, with colder water going to the heat exchanger in the outdoor unit.

The CO_2 refrigerant in the transcritical zone does not condense at constant temperature as in typical refrigerant cycles that are below the critical point.. Instead, the CO_2 cools as it transfers heat to water in the heat exchanger called the gas cooler. After it leaves the gas cooler at about the temperature of the incoming water, it drops down into the evaporator and goes through the air-to-vapor exchange at a lower constant pressure and temperature. The compressor then lifts the CO_2 back to the high temperature and pressure transcritical zone, where it transfers the absorbed heat to the colder water.

In normal operation, the split system heats water in a single pass to 149°F. There is, however, a catch. **Figure 14,** taken from the lab assessment of the combined system by Ecotope (Larson, et al., July 2015), shows the truncation that occurs when the water coming into the system is too warm. The efficiency of the transcritical cycle is reduced because the invested compression energy remains the same but the heat that can be transferred to the water in the gas cooler or absorbed from the air in the evaporator is reduced.

Figure 14. Impact of Water Temperature on Heat Transfer

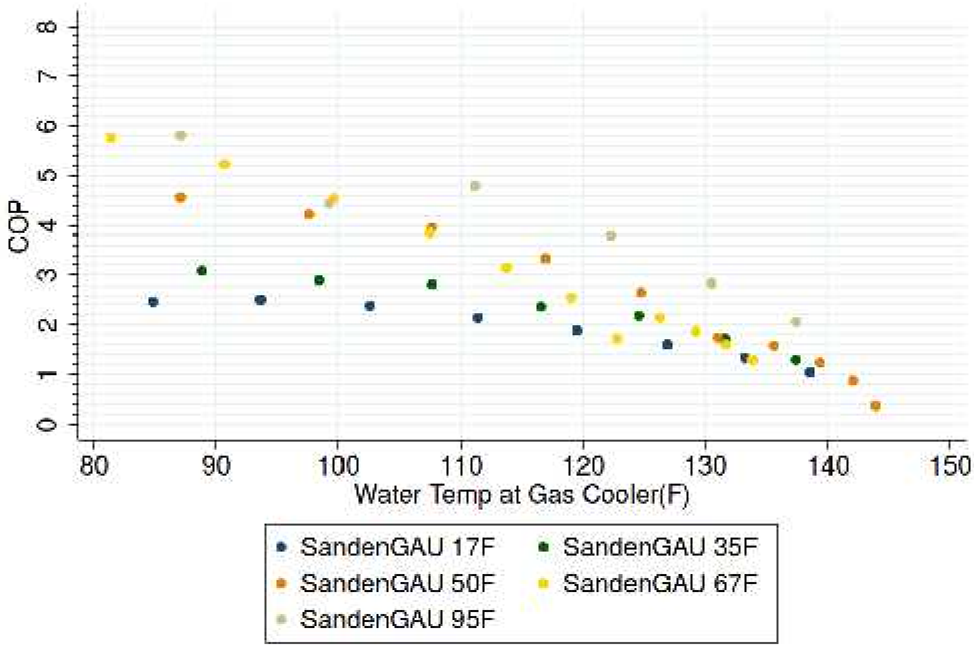


Tank destratification means that temperature difference between the top and bottom of the tank decreases. This can happen if heating demand is high and sustained. In this case the X-Block will circulate enough water to exceed the tank capacity, thus causing the tank to completely mix. Also, if heating demand exceeds the rate at which water is heated in the heat pump the auxiliary heat will turn on and increase the temperature of the heating return water to the tank.

The temperature ranges that optimize performance depend on system operation. In general, low-temperature distribution systems, such as radiant floors that return water below 90°F, have higher performance than higher-temperature systems such as radiant panels that return water above 100°F. Optimum performance depends on return temperatures no higher than 80°F, which implies a radiant slab for heat delivery and a moderate (68°F to 70°F) thermostat setting.

Figure 15 is a graph taken from the Demand Response Lab Test Report by Ecotope (Larson, September 2015). It shows the impact of water temperature on the COP. The colored dots are the OAT (as simulated in the lab). The X axis shows the incoming water temperature to the gas cooler heat exchanger and the Y axis shows the COP.

Figure 15. Impact of Water Temperature on COP



COP decreases as water temperature increases. While the decrease caused by higher temperature is most dramatic at higher OAT, the most critical decreases are at colder temperatures, where destratification is most likely to occur. For example, at 35°F OAT, the COP with 80°F water going to the gas cooler is 3.1, but if the temperature of the water being heated increases to 110°F, the COP drops to 2.4.

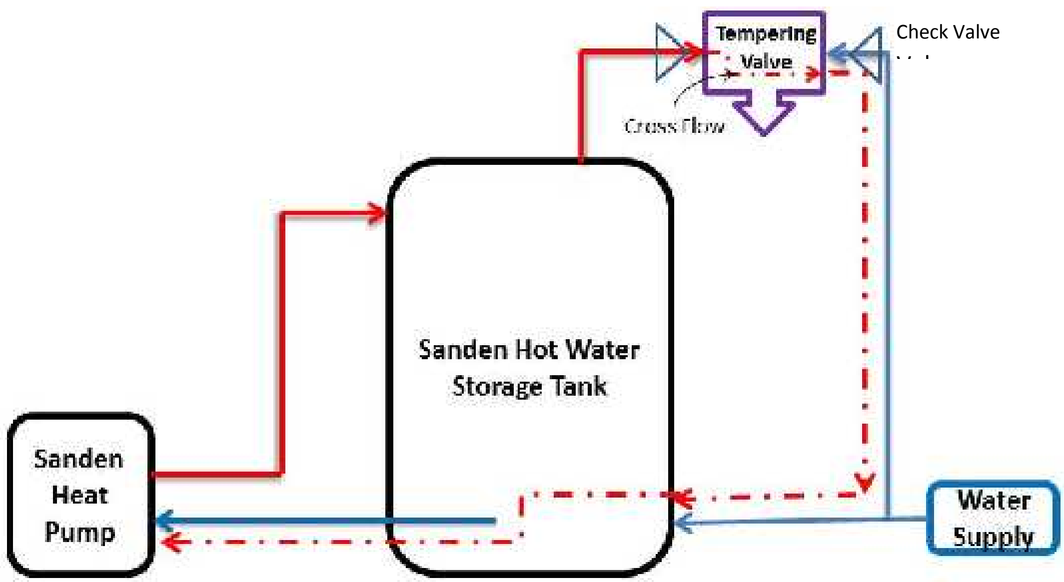
Destratification can be reduced. Site 10 has the second highest average return temperature listed in Table 5, but has the best performance in both heating and non-heating seasons. The factor that distinguishes this site from other sites with high return temperatures is very high hot water use averaging 152 GPD during the heating season and 167 GPD during the non-heating season. The result is a flood of cold water into the tank, which increases stratification and causes optimum performance at the heat exchanger.

Cross Flow

A specialized form of destratification occurs when hot water crosses the tempering valve and mixes with the cold water supply. This can happen when the bottom of the tank is depressurized as the result of the pump in the outdoor unit pulling hot water from the tank to the gas cooler. This is illustrated **Figure 16.**

Late in the project period (June 2016), WSU noticed that this was occurring at Site 9. The lines going to and from the outdoor unit have now been equipped with temperature sensors at all sites, and review of the temperature data shows that the water going to the outdoor unit from the bottom of the tank reaches temperatures as high as 121°F midway or earlier in the cycle every time the heat pump operates. The pathway for this hot water movement is through the tempering valve and down the cold water line. As demonstrated in the section on tank destratification and **Figure 15**, this has a serious negative impact on system performance.

Figure 16. System Schematic with Cross Flow Dynamic Superimposed



A temperature rise at the end of the heat pump cycle is normal. The cycle ends when heated water reaches the lowest tank level, and that heated water is sent to the outdoor unit at the very end of the heat pump operation. **Figure 17** shows the progressive temperature increases at succeeding tank levels as the heating cycle progresses, as measured during a lab test conducted by Ecotope.

Figure 17. Progressive Temperature Increases at Succeeding Tank Levels as Heating Cycle Progresses

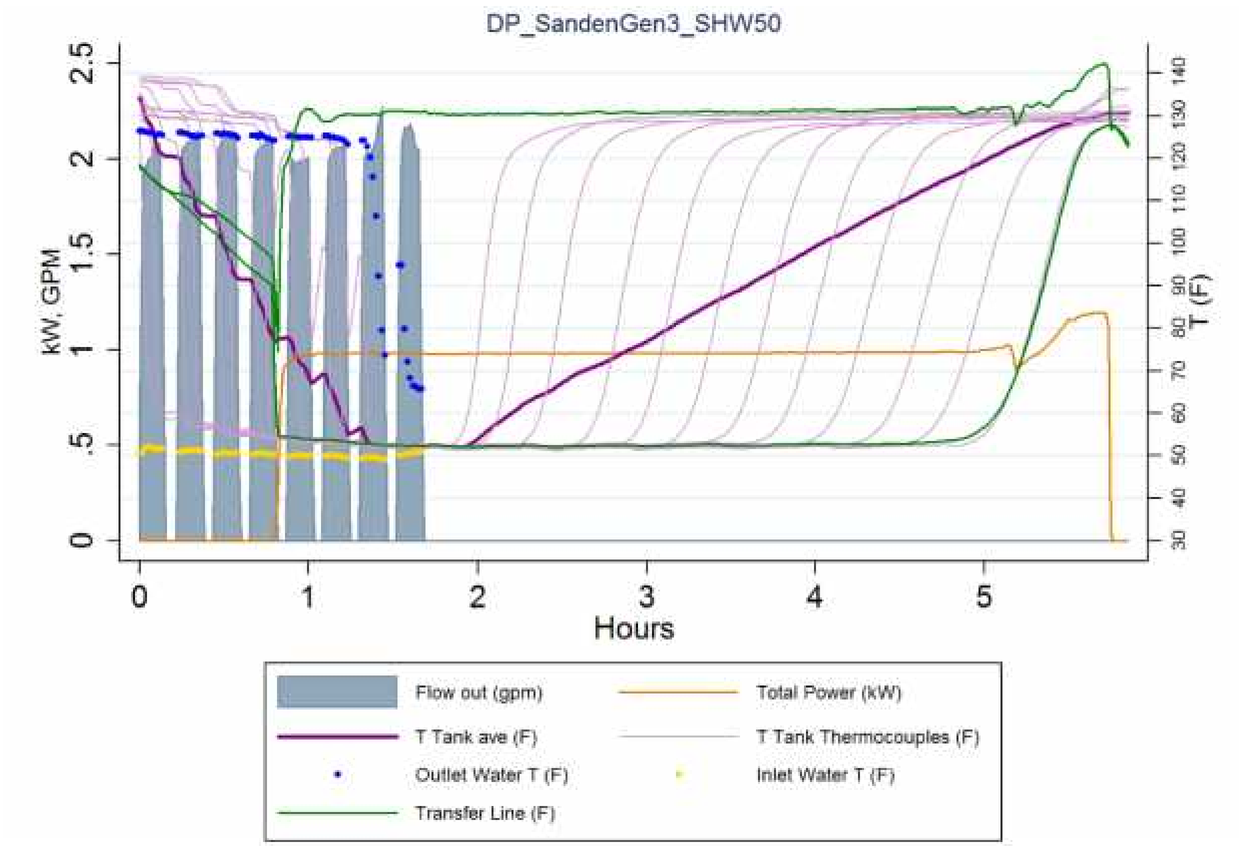
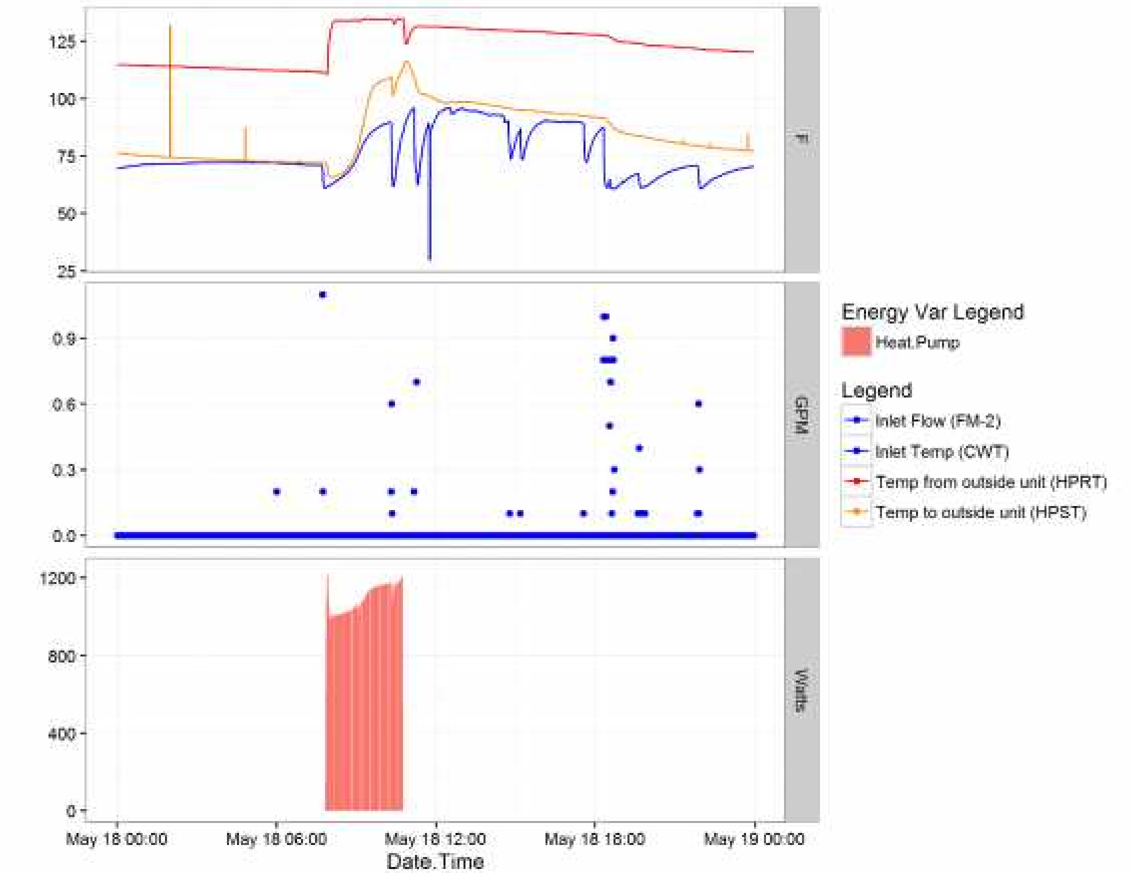


Figure 17 plots the transfer line temperatures to and from the Sanden outdoor unit in green. The bottom green line is the temperature going out; its temperature starts rising just before hour 5. An hour later, it has risen to setpoint. The lavender line shows the temperature at progressively lower heights in the tank and tracks the progression in tank temperature as the cycle progresses.

Cross flow is not normal operation. It begins earlier in the heat pump cycle and short circuits the standard progression of heating from top to bottom of the tank by bringing heated water directly to the bottom of the tank and displacing cold water that would otherwise go to the outdoor unit.

Figure 18 shows the system temperatures and operation on one day in May 2016 at Site 9. The three water temperatures in the top graph are from the heat pump to the tank (red), from the tank to the heat pump (orange), and inlet water (blue). Inlet flow is in the second level of the plot, and the heat pump wattage is at the bottom. Note the initial decrease followed by an increase in the water temperature going to the outdoor unit, as well as the temperature in the inlet water pipe when the heat pump turns on. The initial temperature is that of the cold water drawn by the pump in the outdoor unit from the bottom of the tank. Because cross flow at the tempering valve occurs, hot water is pulled through the cold water side of the tempering valve to mix with the cold water supply to the bottom of the tank, where it is pumped to the heat pump for exchange at the gas cooler. This temperature rise starts at the beginning of the heat pump cycle and peaks before the heat pump stops working. This is not normal operation.

Figure 18. System Temperatures and Operation at Site 9 on One Day in May 2016



The heat pump uses the same amount of energy to bring the CO_2 to high temperature in all cases. The most efficient heat transfer occurs at larger differences in temperature between the water and gas in the gas cooler. With warmer water entering the gas cooler, the electricity used by the compressor produces less heat transfer, which reduces overall efficiency. The denominator (the energy in) is larger and the numerator (the energy out) is smaller (**Equation 2**: FEF = (Qdhw + Qsystem) / Qinput).

At Site 9, the water temperature going to the heat pump (HPST) is significantly higher than 100°F for half the operation. This reduces the COP, and probably reduces the overall annual system efficiency (FEF) at least a full point. At sites 1, 5, 7 and 10 sufficient data was collected during the project to determine performance as shown in **Table 8** (taken from Table 4). Available data from Site 9 has been added.

There is a big discrepancy between the efficiencies shown in Table 8 and the simulated efficiencies stated in **Table 9** (also discussed in the next section on the lab test). The efficiencies stated in Table 8 are derived from detailed lab tests and highly accurate simulations using lab and field data for the climate zone where these houses are located. At an FEF of 2.9 for water heating only (NH) and 2.6 for space and water heating (H), they are much higher than those noted in Table 9. The heating season efficiency in the simulations is predicated on houses being heated with radiant slab systems using low supply and return water temperatures – the optimum design for this equipment – but the non-heating season efficiencies do not have space heat return water so should be comparable for any type of heat distribution.

These efficiencies are much higher than those noted in Table 8. While low domestic hot water use may contribute to the measured lower efficiencies, it is likely that the inefficiency caused by feeding hot water to the gas cooler every time the heat pump operates also impacts the performance. Solving this problem will likely bring these systems closer to the forecast efficiency.

Tempering valves are designed to prevent a burst of scalding water from going out the hot water tap. Manufacturers of tempering valves claim to make models containing integrated check valves. WSU found an integral check valve stuck in the open position on the cold water inlet at Site 9. This finding demonstrates that valves should be field tested to ensure they actually block the cross flow – many do not or cease to do so after a period of time. The recommended approach is to install independent spring-loaded check valves on every tempering valve feed line.

Table 8. Performance Impacted by Cross Flow

Site	XRWT (°F)	HPST	DHW (GPD)		FEF		Days Sampled	
	H	Peak T on Av. Day	H	NH	H	NH	H	NH
1	82.93	110	34.28	23.46	1.04	0.88	31	65
5	111.4	121	17.87	21.02	0.58	1.18	83	75
7	89.7	121	28.57	18.3	1.24	0.76	80	2
9		116		28		1.7		65
10	101.43	124	151.91	167.31	2.28	3.35	60	33

Table 9. Performance Predictions by Climate

Climate	Annual Efficiency		
	Water Heating	Space Heating	Combined
Boise	2.9	2.3	2.5
Kalispell	2.6	2.1	2.2
Portland	3.0	2.6	2.7
Seattle	2.9	2.6	2.7
Spokane	2.8	2.2	2.4
Heating Zone 1	2.9	2.5	2.6
Heating Zone 2	2.8	2.2	2.4
Heating Zone 3	2.6	2.1	2.2

The space heating values in Table 9 are directly comparable only to low-temperature distribution systems such as radiant floors. Compare the Seattle space heating values to the heating season (H) FEF values in Table 8 for Sites 1 and 7. All of the non-heating season (NH) FEF values in Table 8 may be compared to Seattle's water heating annual efficiency. The qualifier there is that values in Table 9 were calculated using average daily hot water consumption of 46 gallons. Four sites are substantially below that value, which reduces water heating efficiency. The higher efficiency values provided for Site 10 in Table 8 are certainly related to the high hot water use at the site.

Auxiliary Heaters

The original auxiliary heater was a 40-gallon tank installed in Bellingham. It was replaced with an electric demand heater to avoid tank loss, and to conform to the revised design guideline applied to all other sites that elected to install backup heating.

After installation, the demand heaters at three sites developed water leaks, causing extensive site damage in one case. At Site 10, a leaking heater was replaced, but the replacement also leaked. The technical design committee met and decided to assess all remaining sites, determine those that needed auxiliary heat, replace the demand heaters at those sites with 40-gallon electric water heaters, and remove the demand heaters at the remaining sites. **Table 10** summarizes the auxiliary electricity use during the heating season at all the sites with backup heaters.

Table 10. Auxiliary Heat Analysis

Site #	Location	kWh	# Days	kWh/Day
1	Bellingham	1.1	31	0.04
7	Seattle, Ballard	154.4	80	1.93
8	Seattle, Leschi	421.4	167	2.52
10	Tacoma	526	60	8.77

Figure 19. Tank Installed at Site 10

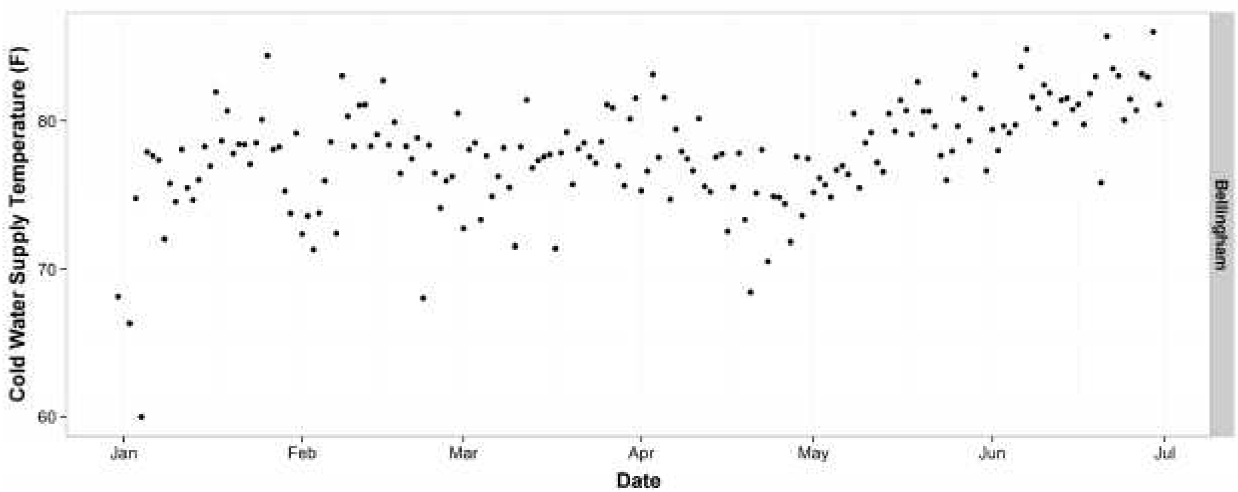
Table 10 does not include Sites 5, 6, and 9, which do not have backup space heat. This table was used to determine if auxiliary heat was needed at those sites that had demand heaters. The comparison revealed that Site 1 does not need auxiliary heat while Sites 7 and 8 probably do, and Site 10 clearly does. Tanks, such as the one installed at Site 10 (**Figure 19**), are now the recommended form of auxiliary heat and also provide additional thermal storage.

The question remains as to the best location for auxiliary heat. Should it serve both space and water heating or be focused solely on assuring hot showers? The thermal advantages of using it only to boost DHW temperature is that it does not increase the temperature of the return water going to the heat pump tank. One of the sites had a backup demand heater on the mixed hot water line, because the owners would prefer to put on a sweater than to suffer a cool shower.

Preheating Cold Water

The plot in **Figure 20** (Eklund and Banks, 2015) shows the average cold water supply temperatures at Site 1. As pointed out in the section on destratification, the temperature of the incoming water impacts the system efficiency due to the thermodynamic properties of CO_2 at the pressures used in the system. The water supply at the Bellingham site is rainwater that is stored in above-ground cisterns and then held in a pressurization tank in conditioned space and then preheated by running it through a solarium. The resulting higher temperatures are above the recommended range.

Figure 20. Average Cold Water Supply Temperature in Bellingham, WA (mean = 77.8°F)



In general, strategies that preheat cold water going to a water heater of this type reduce performance. While heat recovery technologies are beneficial for use with standard electric resistance water heaters, they are counterproductive when used in conjunction with this technology, whether it is used only for water heating or for combined space and hot water.

Combined Space and Water Heating Lab Test

As part of TIP 326, Ecotope was subcontracted to perform lab tests on the combined space and water heating system based on the Sanden split system HPWH. The lab tests were designed to test the ability of the combined system as designed to provide space and water heating. It also examined the alternatives available for returning the heating loop water in terms of impact on tank stratification. This synopsis provides a detailed overview of the test report, including procedure and findings. The full report is provided as a link in Appendix A.

Introduction

The report begins with an explanation of why CO_2 is used as a refrigerant (see also Appendix B). It reviews the history of refrigerants and ozone depletion, and the contribution of hydrofluorocarbons (HFCs) to climate change because of their high Global Warming Potential (GWP). It notes that they are being phased out and concludes that "carbon dioxide is an unusually environmentally friendly refrigerant," noting that it is not toxic, flammable, or corrosive; is inexpensive and readily available; and has no impact on the ozone layer. The report discusses the transcritical vapor compression cycle.

The research plan addressed the following research questions:

- What is the best way to test a combined hydronic system with two different loads?
- What is the COP of each load?
- What impact do the two loads have on each other and on system performance?
- How does the system perform over a wide range of OATs?
- How do the individual functions perform compared to dedicated systems previously tested?

The following questions were added to address issues noted in the first site installation:

- Where in the storage tank should the water returning from the heat exchanger be delivered?
- What happens to the system under high temperature and low temperature space heating applications?
- What is the heat exchanger heat transfer effectiveness?
- How much energy do the circulating pumps use and what are their flow rates?

The report then discusses the equipment tested and previous test results for the Sanden system in its dedicated function as a water heater.

Test Methods

Ecotope discusses the development of the test method and why the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Standard 206 for combined system evaluation and rating was not used. The reasons were that Standard 206 would not answer the practical questions posed by the project, including system design and operation, and determining appropriate applications and technical recommendations for performance improvement.

The test plan focused on the following areas:

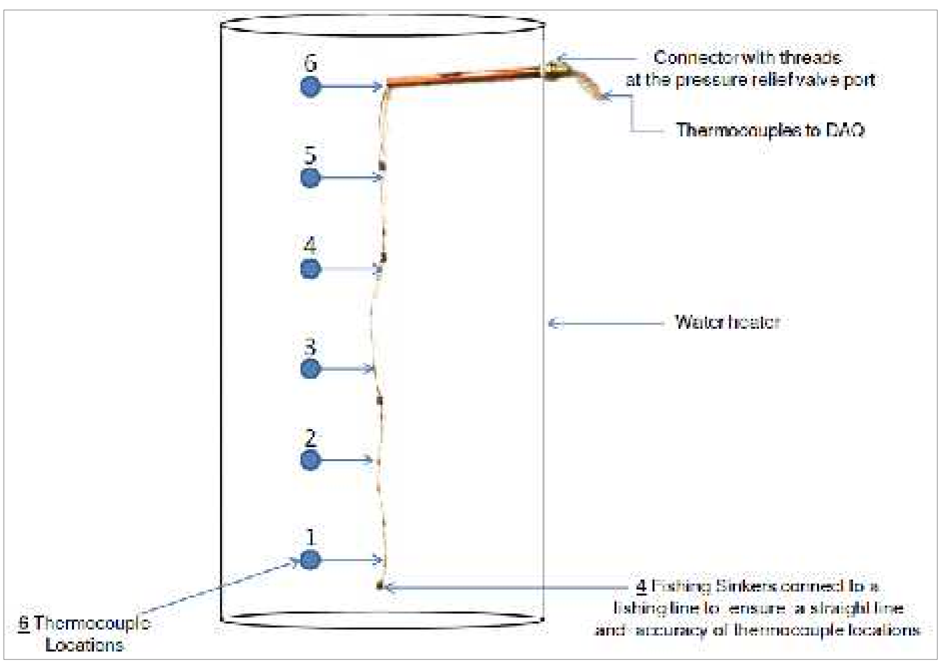
- Measuring space heating only performance,
- Measuring combined space and water heating performance, and
- Tempering valve tests.

The space heating tests measured system efficiency under low-temperature (80°F) heating return emulating radiant floors and high-temperature (110°F) representing radiant panels at three different locations:

- Top of the tank, at the pressure and temperature relief port;
- Top of the tank, at the pressure and temperature relief port with a diffuser; or
- Bottom of the tank, sharing the port traditionally used for cold city inlet water.

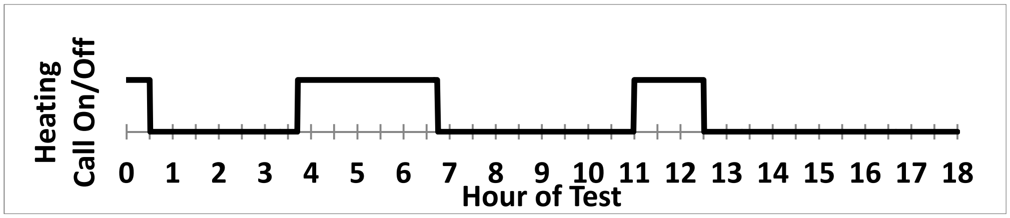
Thermocouples were placed vertically in the tank at intervals to observe the impact on tank stratification of the different return locations under different return water temperatures, as shown in **Figure 21**. This illustration is taken from the first Sanden split system lab test report demonstrating the value of experience (Larsen, 2013). The tank used for the combined system tests had 13 thermocouples instead of the six shown here.

Figure 21. Illustration of Thermocouples Placed in Tank to Examine Tank Stratification



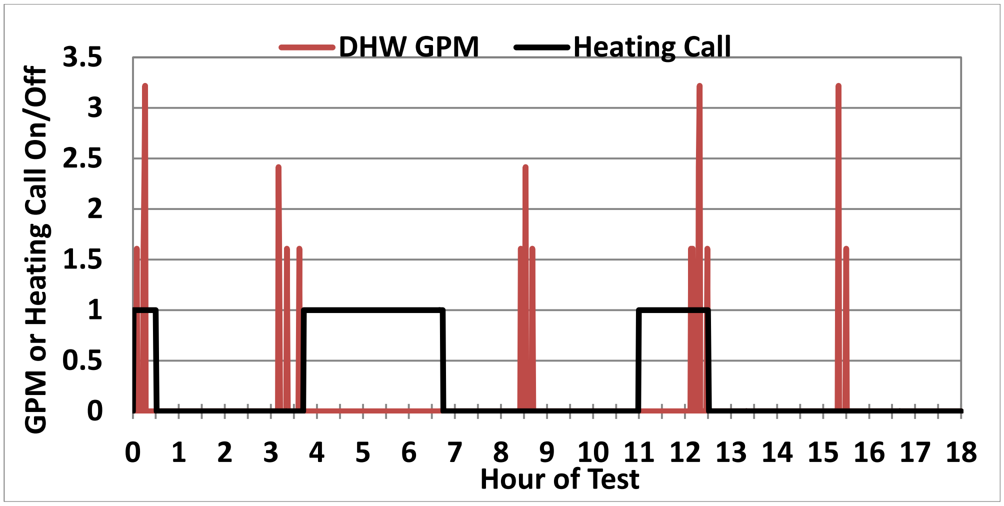
The space heating demand profile consisted of three heating calls over an 18-hour period, which is the amount of time the lab had to reset the equipment during the day and run the tests at night. **Figure 22** shows the space heat demand profile. The total heating time was five hours. The heating load of 93.5 kBtu was constant for both the low- and high-temperature return water tests.

Figure 22. Space Heat Demand Profile



The combined space and water heating tests used a standard hot water draw pattern for a family of three totaling 46 GPD. The water heating load totaled 28.6 kBtu; space heat totaled 93.5 kBtu, for a total of 122.1 kBtu. **Figure 23** shows the combined space and water heating test demand profile, with red showing the hot water calls in gallons per minute (GPM).

Figure 23. Combined Space and Water Heating Test Demand Profile



The outdoor unit was installed inside a thermally controlled chamber, and the tank was kept next to the chamber in a space kept between 60°F and 70°F. A total of 24 independent measurements were taken.

The overall lab test findings indicate:

1. In general, the low-temperature tests showed greater stratification than the high-temperature return water tests.
2. In combination mode, the high-temperature flow rate of 3.75 GPM quickly destratified the tank regardless of return location.
3. Where the return temperature of 80°F was returned to the bottom of the tank in either space heating only or combination mode, the stratification was greatest, followed by the diffuser and then top of the tank return.
4. The result of lower stratification is higher temperatures to the gas cooler, which results in lower system efficiency.

Based on the lab results and previous tests, Ecotope determined a system performance function that was used to estimate annual system efficiency by applying it to different climate temperature profiles. **Table 9** (on page 28) shows these estimates for annual water heating, space heating, and combined performance efficiency. **Table 11** summarizes the results of the lab testing. It provides guidance for all future system design.

Table 11. Performance Predictions by Climate

Test	XPB Load Supply T (°F)	XPB Sanden Supply T (°F)	XPB Sanden Return T (°F)	XPB Sanden Flow (GPM)	XPB Load Flow (GPM)	HP Waterline In (°F)	HP COP	System COP
Low Temp, Combo, Bottom Inlet	80.2	146.6	76.5	0.6	3.7	73.7	2.61	2.30
Low Temp, Combo, Diffuser	80.2	127.6	78.0	0.7	3.7	88.2	2.29	2.01
Low Temp, Space Heat, Bottom Inlet	81.4	143.7	77.9	0.6	3.8	80.3	2.49	2.21
Low Temp, Space Heat, Diffuser	81.1	133.8	78.4	0.7	3.8	92.4	2.30	2.11
Low Temp, Space Heat, No Diffuser	80.4	111.0	80.2	1.5	3.8	91.3	2.10	2.03
High Temp, Space Heat, Bottom Inlet	109.1	119.4	110.2	3.6	3.6	108.8	1.90	1.68
High Temp, Space Heat, Diffuser	108.7	118.7	109.7	3.5	3.6	109.9	1.90	1.66
High Temp, Space Heat, No Diffuser	109.3	119.5	110.6	3.5	3.6	110.6	1.67	1.56

Based on the detailed lab findings, Ecotope provided the following recommendations for optimizing system design:

- The overriding system design principle is to provide the coldest water possible to the heat pump.
  - Very cold water provides exceptional performance, but 80°F is acceptable.
  - Radiant floors are best for this system because they operate at lower temperatures.
- Extract as much heat as possible from the water circulating from the tank through the heat exchange block.
  - This lowers the temperature of the return water.
  - It also reduces the flow rate, which provides more time before the tank destratifies.
- The ideal storage tank would have return ports at multiple levels for different temperature returns: bottom for radiant floors, first third to one half for radiant panels, and top for cassettes and hydronic coils returning shower-temperature water.
- Increase the tank size to 120 gallons.

For detailed findings with graphs and tables, please see the entire lab test report, available through a link in Appendix A.

Conclusion

The experiment was to combine a set of component parts designed for other purposes into a new system that would handle space and water heating in energy-efficient new homes located in all three of the region's climate areas. The experiment succeeded in finding the capacity limits of the system application, the installation issues that must be solved, the features that could optimize performance, and the efficiency limits of this approach to combined space and water heating.

The region needs a working natural refrigerant solution for loads that are currently served by heat pumps using high global warming factor refrigerants. During the final work on this report, amendments were adopted to the Montreal Protocol on October 15, 2016 that will begin the phase out of hydrofluorocarbon (HFC) refrigerants beginning in 2018. With ambitious plans being discussed in Vancouver B.C. and Seattle to convert natural gas furnaces to heat pumps, carbon dioxide systems have great potential.

The system resulting from the experiments conducted in TIP 326 can be optimized for low design load homes using low-temperature heat distribution, such as high mass radiant floors, and achieve overall combined annual efficiency, including all energy costs such as tank loss, distribution energy, and freeze protection, approaching an FEF of 3 in Portland and Seattle. An optimal system would include: perfect installation (no cross flow); a 120-gallon tank with multiple return options; and a household that sets the thermostat no higher than the mid-70s and uses an average of 15 gallons per person of hot water per day. Performance would be enhanced by the new X-Block with electronically commutated motors.

A future design improvement would be to eliminate the possibility of destratified operation during cold weather. This would lock out the tank when heating flow increases to levels that will cause destratification, and switch completely to backup heat to provide all heat during these times without returning hot water to the tank. These controls need to be developed to avoid periods of heat pump efficiency that are lower than the efficiency of electric resistance heaters.

For cold climates and higher load homes in milder climates, a higher capacity CO_2 heat pump is needed. Sanden has an 11 kW output system designed specifically for space heating. It has not been tested in the U.S., but BPA and NEEA are working with WSU to test that system. If lab testing demonstrates that it has potential to solve the capacity issue identified in the TIP 326 research, it may be the technology needed to offer a comprehensive natural refrigerant solution for combined space and water heating.

The optimizations identified in this research have not been field tested. The impacts of cross flow and other design solutions need to be quantified through continued field testing. NEEA has supported updating all the monitoring systems and implementing cross flow solutions. WSU has used some of its remaining TIP 326 resources to test various cross flow solutions. It is essential that this work be done and the results be disseminated. It would be very beneficial if the data collected over the coming year could be analyzed and reported.

Recommendations

1. The cross flow issue may impact all of the seven systems that remain in operation. It must be thoroughly investigated and a practical solution found because it appears to a have a significant impact on system performance. To date, three different Honeywell tempering valve models with integral check valves, including the model that Sanden is shipping with every split system sold, have failed to consistently stop cross flow.
2. The recommendations by Ecotope from its lab test should be field tested. Foremost among these is the large 120-gallon tank with return ports at different levels to optimize performance to the heating system type.
3. Power-out freeze protection requirements need to be carefully considered. This adds cost to the system, and it needs to be determined if it should be required at all locations regardless of climate. Freeze time was investigated by Ecotope in the lab test for TIP 302 (Larson, September 2015, p. 25).
4. The TIP 326 field test with the seven surviving sites should be continued for another nine months. The data gathered and analyzed in this report was impacted by monitoring equipment issues, re-design and re-plumbing issues, and cross flow issues. None of these issues were related to the heat pump, details of its performance are really unknown.
    a. The metering equipment is now updated and operational at all sites. The monitoring licenses have been extended by NEEA and the homeowner agreements all extend to June 30, 2017.
    b. It will provide an opportunity to determine an effective cross flow solution and install it at all sites in October and early November of 2016. This will allow data from most of the ensuing heating season, spring, and summer to be collected without interference from this performance debilitating issue.
    c. Monitoring, data analysis, and an updated report could be obtained at modest cost, considering that the major investment is already made. It was overly optimistic to attempt to build new homes, design and install a new prototype unit, and monitor it during this process and expect that representative data would be obtained.
5. The two remaining cold weather sites in McCall, ID, need to undergo testing of the large-capacity unit design specifically for space heating. The monitoring systems are installed and functioning. NEEA is committed to installing the large-capacity units. Time and funding are needed to collect and analyze the data and report the results. Once again, the major part of the investment is already made or committed.

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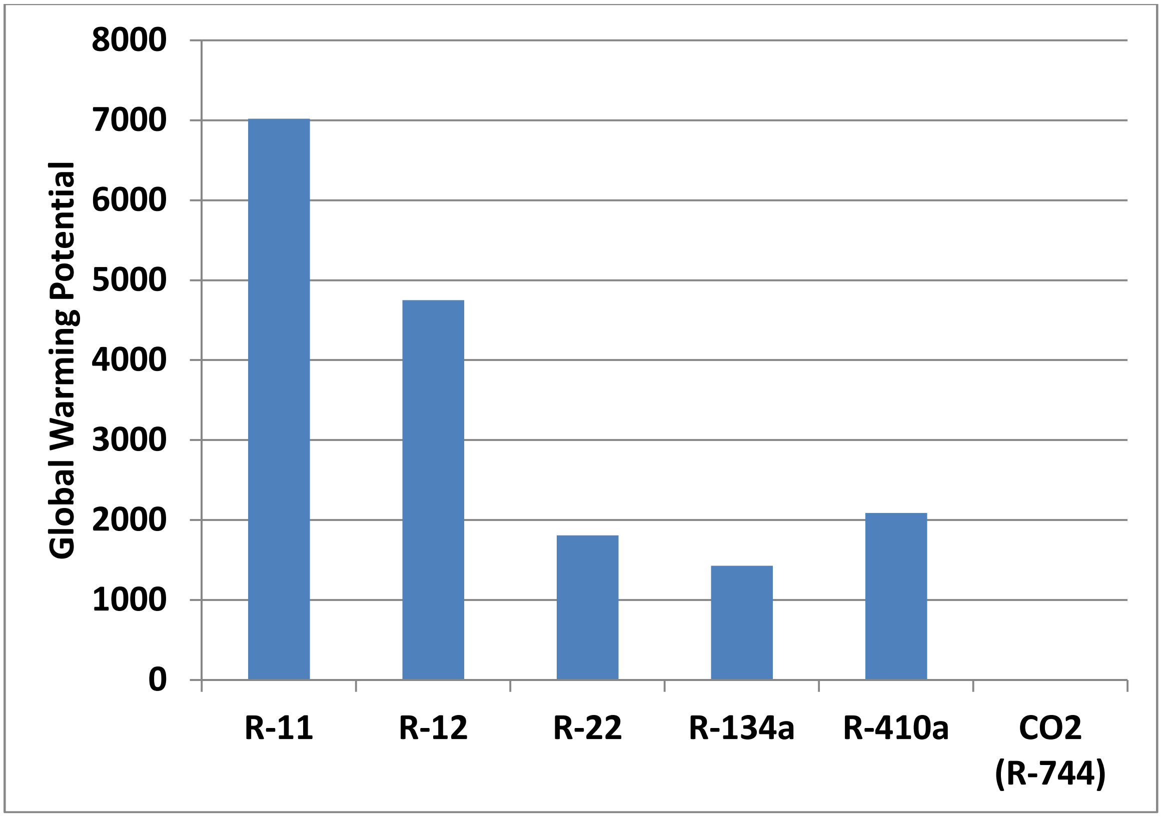
Appendix A: Laboratory Assessment

The *Laboratory Assessment of Combination Space and Water Heating Applications of a CO2 Heat Pump Water Heater*, July 2016, is provided as a link: Lab Test Report for Combined Space and Water Heating Systems

Appendix B: Climate-Destroying Refrigerants

The Kyoto Protocol addressed the contributions to climate change of hydrofluorocarbons (HFCs) such as R-134a and R-410a. The European Union, Australia, and Japan have pioneered domestic reductions in HFC use. For example, HFCs are now banned for use in automotive air conditioners in the EU, and by 2030 the EU is targeting to produce only one-third of 2014 HFC emissions. In addition, in 2016 both the EU and North American countries submitted proposals to the Montreal Protocol to phase down the use of HFCs. The reason for these actions is shown in Figure B-1 based on EPA information.

Figure B-1. Global Warming Potential of Refrigerants

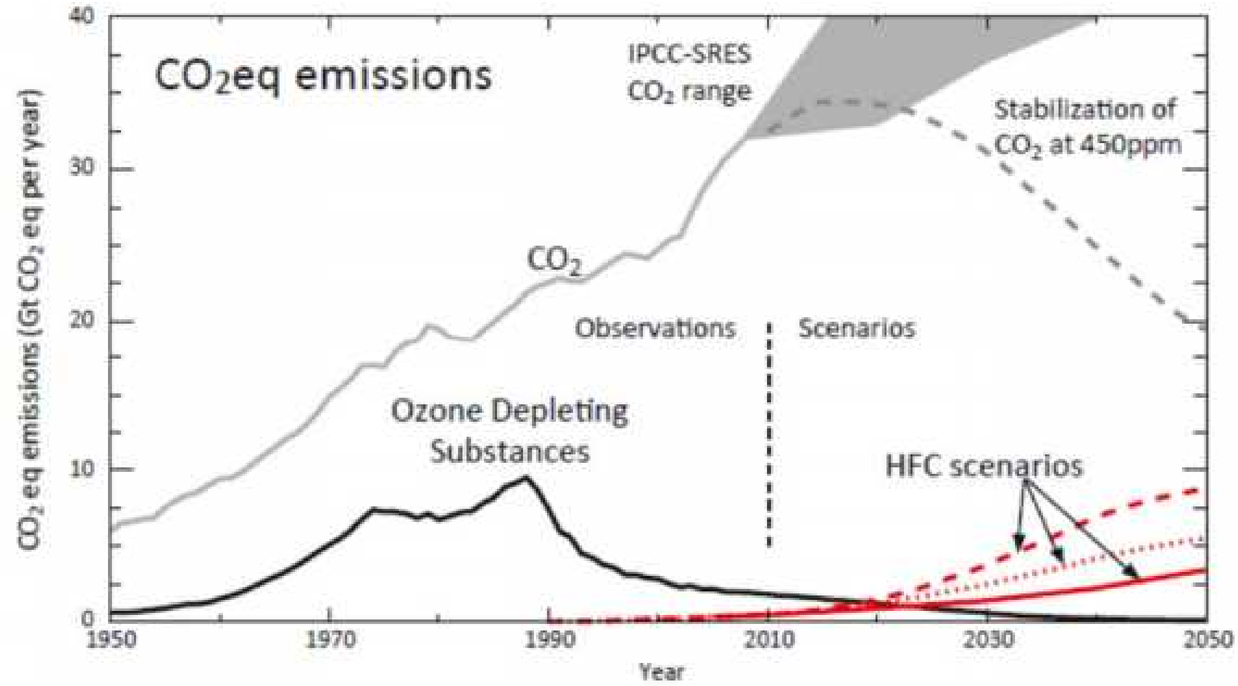


A pound of R-410a is equivalent to more than a ton of CO_2 in climate impact. And heat pumps normally leak up to 25% of their refrigerant per year, which usually result in service calls.

The problem is growing. Currently, the anthropogenic forcing from refrigerants is 1% to 2% of the total. With the growth in use of heat pumps and reductions in other sources of climate change, it is possible that if HFCs are not curbed, they will increase to 9% to 19% of the total by 2050 (Velders, et al., 2009, http://www.pnas.org/content/early/2009/06/19/0902817106.abstract).

The following graphic illustrates how the levels increase over time as the level of CO_2 emissions is brought under control while refrigerant use for air conditioning increases and technology spreads to developing nations. Figure B-2 was taken from Merging the Transition to Next Generation HVAC Refrigerant Technology with Effective Climate Policy.

Figure B-2. CO_2 Levels Affected by HFC Scenarios



As Ecotope stated in its lab test report: "Basically, carbon dioxide is an unusually environmentally friendly refrigerant. It is not toxic, flammable or corrosive, and it has no impact on the ozone layer. It is inexpensive and readily available (Austin and Sumathy, 2011). By definition, carbon dioxide has a GWP of one, as compared to 1,430 for R-134a or 2,100 for R-410a (U.S. EPA, 2015). There are essentially no fears of CO_2 being regulated out of existence for heat pump applications. If CO_2 can be utilized to provide comparable efficiency to an HFC-based heat pump, then it would be an ideal refrigerant to use."

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