BONNEVILLE POWER ADMINISTRATION

Laboratory Assessment of Combination Space and Water Heating Applications of a CO_2 Heat Pump Water Heater

05 July 2016

Prepared for
Ken Eklund, Project Principal Investigator
Washington State University Energy Program
Under Contract to
Bonneville Power Administration
Janice Peterson, Project Manager

Prepared by
Ben Larson, Michael Logsdon, and Nick Kvaltine
Ecotope, Inc.

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A Technology Innovation Project Report

The study described in the following report was funded by the Washington State University Energy Program under contract to Bonneville Power Administration (BPA) to provide an assessment of the state of technology development and the potential for emerging technologies to increase the efficiency of electricity use. BPA is undertaking a multi-year effort to identify, assess, and develop emerging technologies with significant potential for contributing to efficient use of electric power resources in the Northwest.

Michael Logsdon is a statistician and research scientist at Ecotope. He has a meandering background in engineering, biotech, and mathematics. Mr. Logsdon generates web dashboards for datasets, proposes novel methodologies for analyzing measured energy use, and generally helps people understand their data.

Nicholas Kvaltine is an analyst at Ecotope. He has a background in particle physics, experimental design, quantitative analysis, and numerical modeling. He is a skilled experimentalist working on Ecotope's lab and field energy measurement projects. He is also the lead developer on Ecotope's heat pump water heater simulation. Dr. Kvaltine's doctoral work involved the collection, quality control, and manipulation of large amounts of particle scattering data to refine the understanding of the spin structure of the deuteron.

Ben Larson is a senior analyst and research scientist at Ecotope. He has a multifaceted background in physics, alternative energy, climate change, and energy efficiency. Among his work at Ecotope is a key role maintaining and developing the SEEM energy simulation program, which is used to model energy efficiency improvements in the residential sector. Mr. Larson also manages projects at Ecotope including Pacific Northwest regional efforts to investigate heat pump water heater performance.

Acknowledgements

Ecotope acknowledges Kumar Banerjee, Robert Sant, and Tu Bui of Cascade Engineering Inc, in Redmond, WA for the timely and excellent work in conducting the lab testing. We also thank John Miles, Maho Ito, and Charles Yao of Sanden International for support and technical assistance. Additionally, Ecotope thanks Ken Eklund at WSU for being the conceptual and driving force behind the project.

Abstract

Under the *Combined Space and Water CO_2 Heat Pump System Performance* project, the Washington State University Energy Program (WSU) contracted with Ecotope, Inc. and Cascade Engineering Services Inc. to assess the ability of the of the Sanden International GAU 315EQTA CO_2 heat pump water heater to provide space and water heating. The assessment consisted of a series of lab tests designed to replicate space heating and combined space and water heating needs in real houses. The test series was conducted once for low-temperature heating systems, to simulate heat distribution like a radiant slab and once for high-temperature heating systems like fan coils.

The results showed that performance is highly dependent on the system configuration and design requirements. Moreover the findings reflected the theoretical predictions of examining the transcritical CO_2 cycle: Higher temperature space heating systems have reduced efficiencies compared to low temperature space heating systems. Further results shows combining the hot water needs with space heating bring no deleterious interactions between the two load types. If anything, layering water heating on top of space heating improves the space heating only efficiency.

Glossary of Acronyms and Abbreviations

A	amps
ASHRAE	American Society of Heating, Refrigeration, and Air Conditioning Engineers
BPA	Bonneville Power Administration
Btu	British thermal unit
C	Celsius
CFC	chlorofluorocarbon
CO_2	carbon dioxide
COP	coefficient of performance
DAQ	data acquisition system
DHW	Domestic hot water
DOE	Department of Energy
DR	Demand Response
EF	Energy Factor
EPA	Environmental Protection Agency
EU	European Union
F	Fahrenheit
GAU	Sanden model 80 gallon heat pump water heater
GES	Sanden model 40 gallon heat pump water heater
GPM	gallons per minute
GPD	gallons per day
GWP	Global Warming Potential
HCFC	hydrochlorofluorocarbon
HFC	hydrofluorocarbon
HFO	hydrofluoro olefins
HPWH	Heat Pump Water Heater
Hz	hertz
kJ	kilojoule
kPa	kilopascals
kW	kilowatt
kWh	kilowatt hours
NEEA	Northwest Energy Efficiency Alliance
ODP	Ozone depletion potential
PNW	Pacific Northwest
PSI	pounds per square inch
R-12	Refrigerant 12
R-22	Refrigerant 22
R-134a	Refrigerant 134a
R-410a	Refrigerant 410a
RH	relative humidity
V	volts
WSU	Washington State University
XPB	X Pump Block

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Executive Summary

Under the *Combined Space and Water CO_2 Heat Pump System Performance* project funded by Bonneville Power Administration (BPA), the Washington State University Energy Program (WSU) contracted with Ecotope, Inc. and Cascade Engineering Services Inc. to assess the ability of the of the Sanden International GAU 315EQTA heat pump water heater to provide space and water heating. The project built on several previous assessments of the equipment when used for traditional water heating applications and as a utility demand response device (Larson 2013, Larson 2015). Further, it was conducted in tandem with a field study monitoring the combined system, as installed, in several houses across the Pacific Northwest.

Testing Plan

Previous work has quantified the HPWH performance for supplying water heating only so the test plan focused on new, space heating related tests (Larson 2013). To explore the feasibility of using a heat pump to provide both space and water heating needs, the test plan focused work on measuring space heating only performance, and then measuring combined space and water heating performance. Informed by the engineering design questions from the field study, the test objectives were to measure the system efficiency under low and high temperature configurations and observe the tank temperature stratification profile with different return water plumbing configurations. We examined three configurations:

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

The test series was conducted once for low-temperature heating systems, to simulate heat distribution like a radiant slab and once for high-temperature heating systems like fan coils. Additionally, we layered a hot water draw profile on top of two of the low-temperature heating tests for a total of eight unique test patterns.

Findings

Overall measurement results confirmed that the transcritical CO_2 cycle operates at the highest efficiencies when applying a large temperature lift to cold inlet water. Such a scenario is nearly guaranteed for water heating only applications but must be engineered for space heating systems. In general, the low temperature tests displayed greater stratification (and more pooled, cooler water in the bottom of the tank), and hence greater efficiency, than the high temperature tests. Further, the bottom inlet configuration maintained more tank stratification, followed by the top of the tank with the diffuser, followed by the top of the tank with no diffuser. The high temperature space heating scenarios consistently returned warmer water to the storage tank and mixed the tank more quickly. The warm water reduced the coefficient of performance for the heat pump and the tank mixing reduces hot water availability. There was so much water turnover in the high-temperature tests that no plumbing scenario was able to maintain stratification.

Table ES1 summarizes the measured efficiencies at 35°F for a number of different modes and heating scenarios. The “Water Heating Only” measures come from previous work (Larson 2013). “HP COP” can be thought of as the operating efficiency of the heat pump, while “System COP” calculates the efficiency including tank standby losses. Significantly, the results show that using the Sanden heat pump in a low-temperature space heating system, with or without combined water heating, yields efficiencies similar to water heating alone. Further, they show that the high-temperature space heating scenarios lag in performance by at least 25%.

Table ES1. Measured Efficiency at 35°F Ambient

Test	HP COP	System COP
Water Heating Only	2.75	2.2
Space Heating Only (Low T. Bottom Inlet)	2.5	2.2
Space Heating Only (High T. Any scenario)	1.9	1.7
Combination (Low T. Bottom Inlet)	2.6	2.3

The current lab testing explored the space heating performance at only 35°F while previous work mapped water heating performance over a range from 17-95°F. Given that the *low temperature* space and water heating performance matched the earlier work well at 35°F, it is reasonable to assume that performance will match at other ambient conditions. Consequently, we leveraged the existing performance map to predict combined space and water heating performance over a broad temperature range.

Table ES2 shows the results of the temperature bin-weighted calculation exercise for five climates. The Water Heating column shows system efficiency providing water heating only, Space Heating shows space heating only, and Combined shows the heat pump providing both services. Since space heating always happens at cold temperatures, its efficiency is necessarily less than water heating on an annual basis. Water heating sees a significant efficiency boost from warmer, summertime outdoor temperature. The combined column is the time- and temperature bin-weighted result. Since most of the system energy output goes to space heating, the combined efficiency closely resembles the space heating only efficiency. All cases account for standby losses. High temperature space heating applications are not modeled here but the lab test data show they will have a much lower annual performance.

Table ES2. Annual Performance Predictions by Climate (Assuming Low Temperature Space Heating)

Climate	Annual Efficiency		
	Water Heating	Low Temperature 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

Heat pump water heating works best when maintaining tank stratification, such that hot water is available at the top of the tank while cold water provides maximal heat pump efficiency at the bottom of the tank. For combined space and water heating with the Sanden CO_2 HPWH, this arrangement is most achievable with a radiant slab and heating loop return water plumbed to the bottom of the hot water storage tank. In this case, the Sanden HPWH should provide enough heat to meet hot water and space heating demand for a well-insulated, 3-occupant home at an annual COP of 2.2-2.7 depending on climate.

Optimized System Design

The main concern is that the space heating load can quickly de-stratify the tank, especially with heating systems other than radiant slabs. CO_2 transcritical cycle heat pumps work best when applying a large temperature lift to cold inlet water. These types of heat pumps do not work as well for adding small amounts of heat to already warm water, which is the scenario under de-stratification of the storage tank. The domestic hot water demand actually helps significantly by introducing cold water to the bottom of the tank. There do not appear to be any deleterious interactions between the two load types. If anything the domestic hot water helps re-stratify the tank after lengthy space heating events.

After examining the test results, a clear set of concepts to optimize the design and system performance emerged. The most important is to always provide the coldest possible water for the heat pump to heat. This can be achieved with low-temperature heating distribution systems like radiant slabs. In practice this is further done by designing the space heating portion of the combination system to extract as much heat as possible from the water circulating out of, and back to, the tank.

Several modifications to the Sanden system itself could improve performance; however, the most important influence on performance remains the design of the heating system to which the equipment is applied. The first equipment modification is adding inlet water ports at various heights in the tank. Ideally, they are 1/3 to 1/2 the height from the bottom. These ports would be for space heating return water. Based on the expected, average return water temperature, the system designer can select between the ports for best the placement. If only one is made, the 1/3 height is likely ideal. Next, enlarging the tank size to 120 gallons would make incrementally more hot water available so that under heavy space and water heat usage times, there is more available for both needs. Further, the bigger volume makes it less likely to mix the entire volume under a single heating call. Related, the Sanden heat pump controls were designed around water heating needs. Space heating is somewhat different and overall hot water availability could be improved if the heat pump turned on sooner. For an integrated system, the heat pump could monitor the heating use in the house. When there is a call for heating, the heat pump could wait a few minutes and then engage, instead of waiting for the temperature to drop at the control thermocouple which is approximately 1/3 of the tank height from the top. This control won't likely reduce heat pump efficiency but would place more hot water at the top of the tank. These changes, while theoretically increasing the efficiency, could also increase the standby losses by storing more hot water. The optimum tank volume is not clear from this testing although it seems likely that a larger tank and revised controls could increase the overall system efficiency.

Even with further optimizations, it should be emphasized that primary determinant of performance remains the type of space heating application. Higher temperature space heating systems, with their corresponding high temperature return water will still have reduced efficiencies relative to their low temperature counterparts. Further, even low temperature systems will perform worse than simple water heating only systems because incoming mains water temperature is always colder on an annual basis than any return water from a space heating system. The extent to which efficiency degrades with warmer return water temperature is in the nature of the transcritical CO_2 cycle.

1 Introduction

Using a single device to provide both the space and water heating needs of a dwelling is a concept with historic roots. Wood burning stoves, used to heat a house, were often used in conjunction with a kettle on top to heat water.[1] More recent wood stoves offer a "side-arm" addition for more integrated water heating.[2] Modern applications of combined space and water heating have most commonly been implemented with natural gas-fired devices. The Center for Energy and Environment has conducted extensive research on how to optimize gas-fired combination systems performance (Schoenbauer 2012, 2014a, 2014b). This report explores the feasibility of using a heat pump, with electricity as the energy source, to provide space and water heating demands.

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The field study began in late 2014 and provided preliminary data on the system operation in early 2015. The data suggested several avenues of investigation which Ecotope used to draft the lab test protocol. Having the field test site up and running was useful in targeting the lab work. Previous work has characterized water heating performance so this work focused on space heating applications and the combination of both. Under the direction of Ecotope, Cascade Engineering Services carried out all the lab testing. To do so, they constructed a simulated space and water heating load in the lab with the same components used to install the systems in the field. A narrative and table describing all tests performed for this report is included in Appendix A: Lab Test Protocol.

1.1 Background

1.1.1 Why CO_2? Environmental and Regulatory History of Common Refrigerants

To understand the use of CO_2 as a refrigerant it helps to review the history of environmental implications and regulations for common refrigerants.

Previously common refrigerants, such as R-12 (commonly known as Freon), a chlorofluorocarbon (CFC), contributed to ozone depletion. Due to concerns over the ozone layer integrity, the 1987 Montreal Protocol called for global elimination of CFCs and closely related HCFCs, which led to the phase-out of some common refrigerants (Montreal Protocol 1989).

In their place came now-familiar hydrofluorocarbons (HFCs) such as R-134a and R-410a. These compounds have zero Ozone Depletion Potential (ODP), a measure of harmfulness to the ozone layer, but outsized Global Warming Potential (GWP): potency as a greenhouse gas compared to carbon dioxide. The 1997 Kyoto Protocol called for limitations on greenhouse gas emissions, including the contributions of HFCs (Kyoto Protocol 1998). While not explicitly banning HFCs (like the Montreal Protocol did for CFCs and HCFCs), these common refrigerants were put effectively "on-notice" for their large GWP under the Kyoto Protocol.

The longevity of common usage for R-134a or R-410a – two common refrigerants used in heat pump water heating applications – is, at this time, unknown, and will likely depend on future regulatory and political action on curbing greenhouse gas emissions and the feasibility of low-GWP refrigerants. The European Union (EU), Australia, and Japan have thus far led with recent domestic targets on HFC phase-down. For example, new automobiles in the EU may not use HFC air conditioners, and 2030 EU HFC emissions are targeted at one third of 2014 levels (European Commission 2015). In addition, earlier this year both the EU and the North American

[1] For an example, see http://www.morsona.com/morsoe-7110
[2] See http://kitchenqueenstoves.com/

countries submitted proposals to the Montreal Protocol to include an HFC phase-down (Proposed Amendments 2015). Due to the overwhelming success of the Montreal Protocol at phasing out CFCs, and the environmental concerns of their replacements, some international actors intend to simplify regulation of HFCs by including them under the Montreal Protocol rather than broader climate policy.

Three primary replacements for HFCs emerge: hydrocarbons, hydrofluoro olefins (HFOs), and natural refrigerants such as water, ammonia, and carbon dioxide. Due to different thermodynamic and safety properties, none of these apply to all scenarios. HFOs work most effectively as a drop-in replacement to HFCs, but present flammability concerns and a slightly higher GWP than natural refrigerants. Hydrocarbons present even greater flammability concerns. Ammonia works well thermodynamically, but toxicity limits the breadth of its application. Carbon dioxide presents high-pressure issues and must utilize an alternate refrigeration cycle, but holds GWP of one and zero ODP.

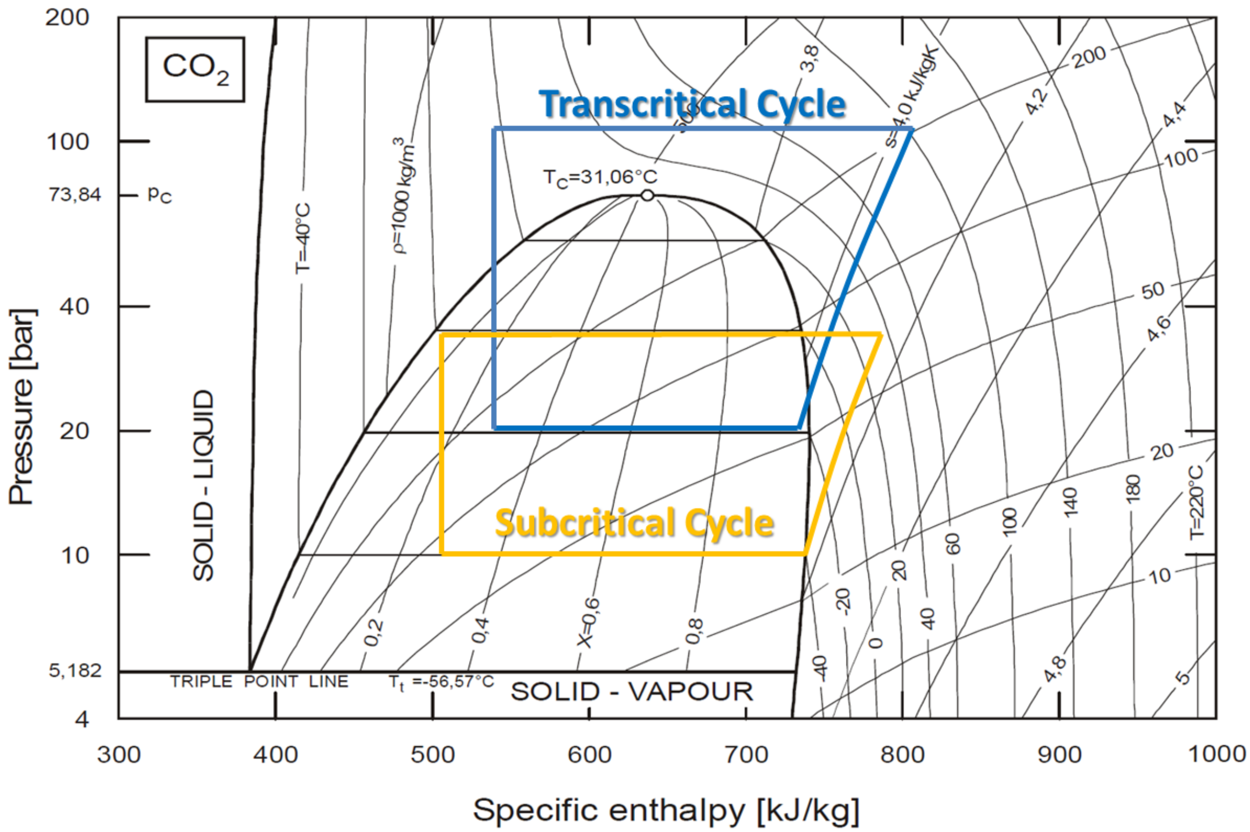
Basically, carbon dioxide is an unusually environmentally friendly refrigerant. It is “not toxic, flammable or corrosive, and it has no impact on the ozone layer. It is inexpensive and readily available.” (Austin and Sumathy 2011). By definition carbon dioxide has a GWP of one, as compared to 1430 for R-134a or 2100 for R-410a (US 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.

1.1.2 The Transcritical Vapor-Compression Cycle

Given that CO_2 holds desirable environmental properties, how does it work as a refrigerant? First off, the “critical point” of CO_2 occurs at a much lower temperature than many common refrigerants. This is the temperature and pressure above which distinct liquid and gas phases do not exist. For a traditional vapor-compression refrigeration cycle – in which heat is extracted from a low-temperature reservoir by evaporating the refrigerant, and ejected to a high-temperature reservoir by condensing the refrigerant – the secondary fluid (in this case the water to be heated) cannot be heated warmer than the critical temperature. CO_2 becomes a supercritical fluid at 7380 kPa and 31°C (1070 psi and 88°F). A traditional vapor-compression cycle with CO_2 refrigerant could not heat water suitable for domestic hot water or space heating applications.

Instead, heat pump water heating with CO_2 refrigerant uses a so-called “transcritical” cycle, where, rather than condensing the refrigerant to eject energy to the incoming domestic water, a “gas cooler” transfers heat through sensible cooling of supercritical CO_2. This cycle is so named for operating in both sub and supercritical zones. Transcritical and subcritical cycles are represented on the pressure-enthalpy diagram of Figure 1, annotated from a figure in the excellent paper by Cavallini (2004). Conventional wisdom suggests that energy is most effectively harnessed in the phase change – hence the evaporator and condenser on familiar residential heat pump applications – so why should we even consider this transcritical cycle?

Figure 1. CO_2 cycle diagram, annotated from Cavallini



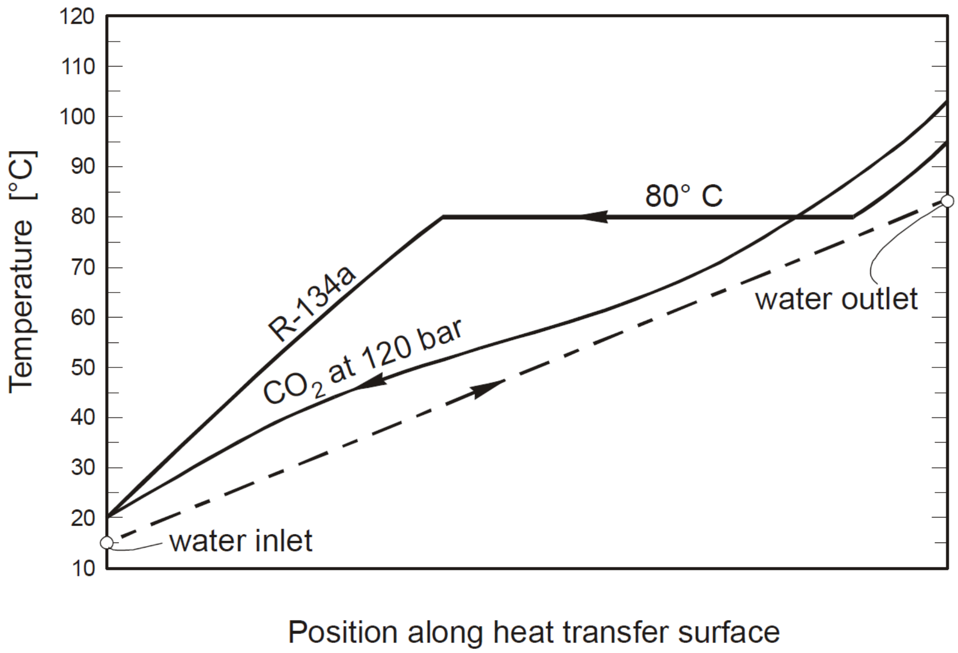
Cavallini explains relative cycle effectiveness is all in the system design:

> *Because of its low critical temperature (around 31 °C), CO_2 does not compare favourably against traditional refrigerants, as far as energy efficiency is concerned, when simple theoretical cycle analyses are carried out. But this situation can be mitigated, and in some cases completely reversed, by proper design of the system aimed at fully exploiting the unique characteristics of CO_2 and/or the exclusive features of transcritical cycles.*

First off, freed from the constraint of staying below the critical point to condense refrigerant, a transcritical cycle can make extremely hot water. For example, R-410a systems struggle to produce hot water much above 135 °F (Larson and Logsdon 2011). The useful hot water temperature with a transcritical heat pump is limited only by feasible refrigerant pressure and acceptable system COP. Essentially, a transcritical CO_2 heat pump can deliver water as hot as desired for residential space and water heat applications, given the engineering/manufacturing constraints on high pressure.

In addition, the nature of the single phase sensible cooling – rather than refrigerant condensation – lends a transcritical CO_2 cycle heat pump well to inducing a large temperature lift. Graphically, the advantage of transcritical CO_2 cycles for inducing large temperature lifts have been visualized in Figure 2, also from Cavallini. The temperature profile of the sensibly-cooled CO_2 along the heat exchanger more closely matches the temperature profile of domestic water, rather than the condensing temperature of the R-134a system. In practice, the refrigerant-to-water heat exchanger is designed in a counter-flow fashion. That is, in Figure 2, the high temperature CO_2 flows from right to left and cools off as it exchanges heat with the water flowing from left to right. At the far end of the heat exchanger, hot CO_2 enters to heat the warmest water. This temperature can exceed the phase change temperature of R-134a which means the water can be made hotter. Conversely, at the other end of the heat exchanger, the coldest water comes in to thermal contact with the coldest CO_2 dropping the refrigerant temperature as far as possible.

Figure 2. Temperature profile along gas cooler or condenser, from Cavallini



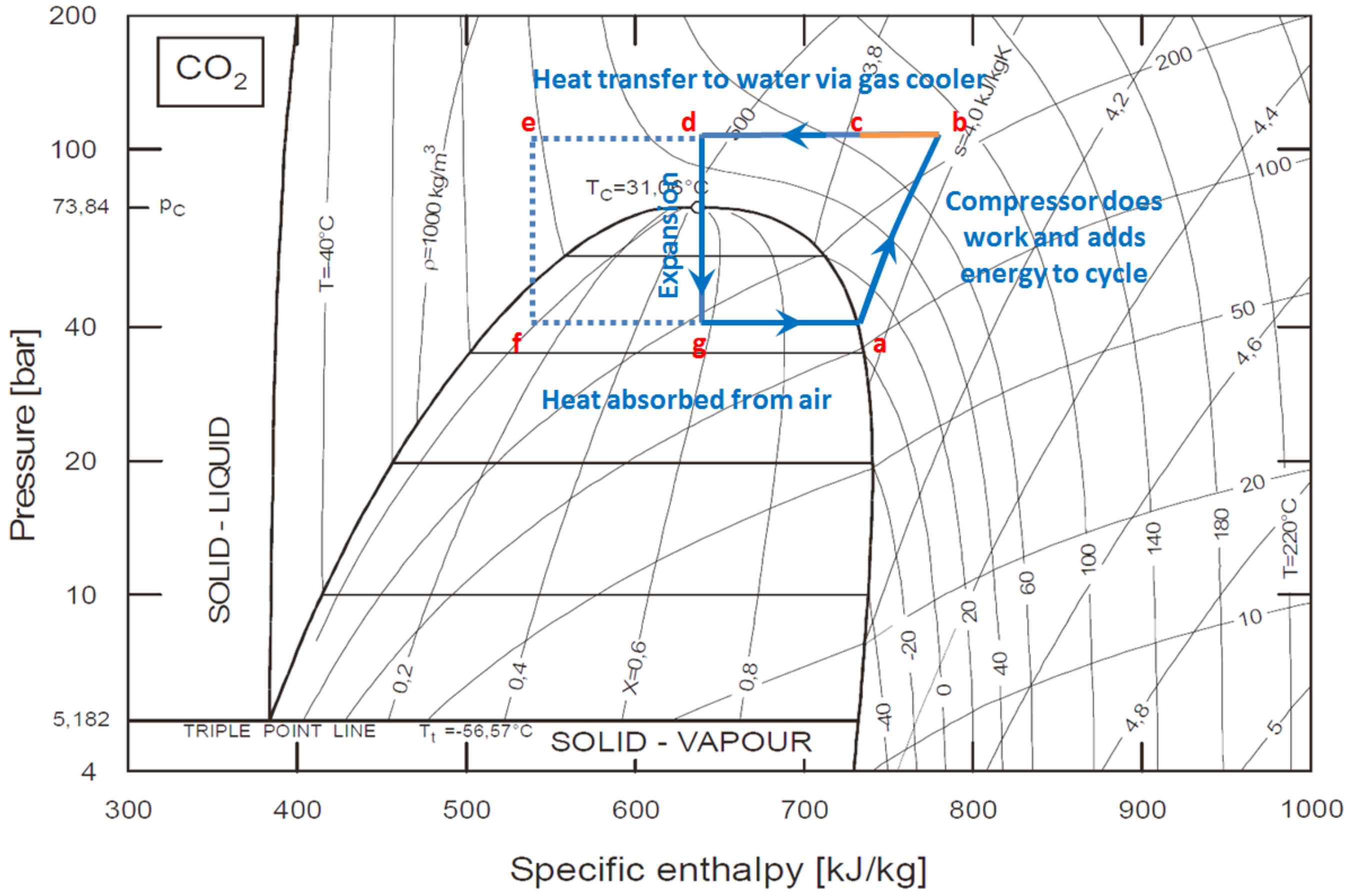
Heat pump water heating is an ideal application for large temperature lifts, as maintaining tank stratification is paramount to an efficient HPWH. In a stratified tank, hot water is available on top, and cold water is heated most efficiently on bottom. Drawing cold water from the bottom of the tank, heating it to setpoint in a single pass, and injecting at the top of the tank maximizes the utility of a HPWH by maximizing tank stratification, and also plays to the strengths of the transcritical CO_2 cycle. These features suggest promise for transcritical CO_2 heat pump water heating and possible applications to space heating as well.

An exergy analysis by Cavallini, comparing a traditional, R-22 vapor compression cycle to a transcritical CO_2 cycle operated under similar conditions, suggests the greatest thermodynamic losses of the CO_2 cycle occur during the isenthalpic expansion process. Physically, this step corresponds to gas expansion occurring at the expansion valve between the gas cooler and the air-to-refrigerant heat exchanger. Further, these throttling losses can be minimized with cooler CO_2 temperature exiting the gas cooler, which would correspond to a greater temperature lift of the secondary fluid (the water to heat).

Overall, to increase the efficiency of the CO_2 cycle, the incoming water temperature needs to be as cold as possible in order to drop the CO_2 temperature as far as possible as it leaves the gas cooler. Conversely, as the incoming water temperature is incrementally warmer, the overall cycle efficiency will decrease. Consequently, this tells us that the system designer’s priority is to configure the entire space and water heating system to provide the coldest possible water. Moreover, it demonstrates that higher temperature applications will inevitably have lower COPs and may not be good candidates for this equipment.

Expanding on this point, Figure 3 demonstrates an idealized transcritical vapor compression cycle with truncation, representing the scenario in which warm source water only allows the gas cooler to sensibly cool refrigerant to around 45°C, as opposed to 20°C in the fully depicted cycle (dashed lines). The COP of each can be expressed as the ratio of the specific enthalpy changes. In the case with cooler source water, this change manifests as the length of segment “b-e,” and in the case with warmer source water as the shorter segment “b-d.” In both cases the compressor contributes segment length “b-c.” Qualitatively this implies a dramatic difference in the efficiency between the cooler source water and the warmer source water, possibly on the order of two times higher. Maximizing sensible cooling in the gas cooler is hence paramount to efficient operation of the cycle, meaning that optimal applications for this type of equipment require cool water for heating. Higher temperature heating applications will, by definition, achieve a lower COP, with the reduction in efficiency likely severe.

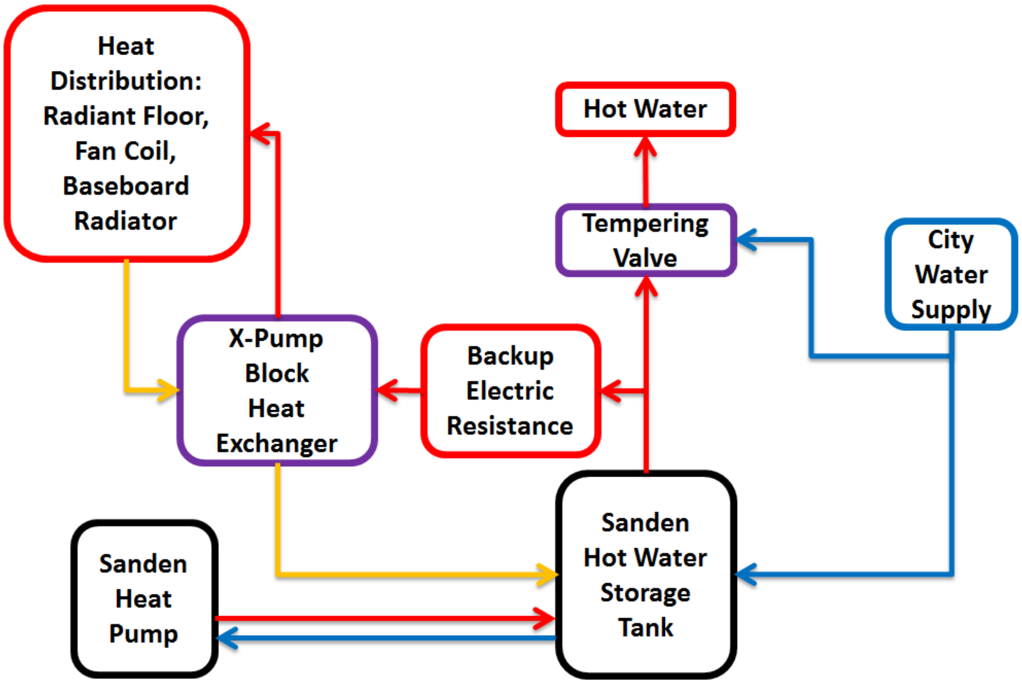
Figure 3. Idealized Transcritical Vapor Compression Cycle with Truncation



1.2 System Design and Operation

The combination space and water heating system requires components of both individual systems to be considered and designed at once. Figure 4 presents the basic components to the system. The arrows on the diagram indicate the water flows with colors loosely corresponding to temperatures (red is hot and blue is cold). The heat pump water heater provides the heat source. It is split in two parts: the heat pump unit, located outdoors, and the hot water storage tank, placed indoors. The heat pump draws cold water from the bottom of the tank, heats it, and circulates it back to the top of the tank. New water is added to the system from the city (or well) water supply. Hot water leaves the tank for one of two uses. For DHW use, it passes through a tempering valve to reduce the temperature to the occupant's set point and then flows in to the house's distribution system. For space heating, it flows through a backup heater, through a heat exchanger, and returns at a colder temperature to the tank. Much like an air-to-air space heating heat pump system, the backup heat is electric resistance. The backup augments the heating capacity if the heat pump can't meet the demand. The heat exchanger has two circulation pumps: one on the Sanden tank side (variable speed) and the other on the load side (fixed speed). Conceptually, the heating load can be any hydronic heat distribution system including radiant floors, forced-air fan coils, or baseboard radiators. The space heat loop is a closed system. The heat exchanger circulates water to the distribution system where it gives up heat to the house and returns to the exchanger colder.

Figure 4. Basic Schematic for Combination Space and Water Heating



1.3 Research Plan Review and Overall Approach

The project set out to address the following research questions in the lab test:

- 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 outside air temperatures?
- How do the individual functions perform compared to dedicated systems previously tested?

Both the overall system design exercise and the first field site deployed and reporting data provided valuable insight to what the lab testing protocol should be. Under a separate contract, Ecotope's mechanical engineering team was contracted to design the space and water heating systems for the field sites. The design process raised clear questions about the best circulation pump configuration, piping layout, and temperature settings to use given the need to meet space and water heating demands and do so as efficiently as possible. Early data from the first field site further confirmed lines of investigation. Both of these sources added more specific research questions to address in the lab including:

- 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?

1.4 Equipment Overview

There are two major pieces of equipment necessary for the system operation. The first is the Sanden HPWH itself, the main focus of the project, and the second is the Taco X-Pump Block. Both bear additional mention which is necessary to understanding system operation and interpreting results.

1.4.1 Sanden Heat Pump

The heat plant for the project is the Sanden GAU-315EQTA CO_2 heat pump. Ecotope previously evaluated the unit, investigating its function under standard water heating conditions (Larson 2013a and Larson 2013b). The reader is encouraged to reference those two previous studies for in-depth explanations of the equipment. The water heater uses a transcritical CO_2 refrigeration cycle to extract heat from the ambient air and transfer it to the water. The product is designed so that the air-to-refrigerant and refrigerant-to-water heat exchange both take place in a single unit. A variable speed pump circulates water from the tank, past the “gas cooler” (heat exchanger) and returns the water to the top of the tank. The variable speed compressor and pump work to heat incoming cold water at any temperature to 149°F in a single pass through the heat exchanger.

The GAU, currently built and sold in Australia, stores over 80 gallons of hot water and has a heat exchange unit fully separated from the hot water tank. Typically, the outdoor unit will be placed on the house exterior with water lines running between it and the water tank, placed inside. The electrical connections accept standard power input– 240V, 15A, at 60Hz. Table 1 presents salient, basic equipment characteristics.

The water heater is directed at markets outside the United States which results in different design decisions than typical of equipment sold within the US. For example, the tank does not have electric resistance elements and has a fixed temperature set point at 149°F. Importantly, the equipment was not designed specifically for space heat applications so this project was, in part, assessing its feasibility for such a use. We evaluated the unit as-is, however, any equipment destined for the United States and for the combined space and water heating market would likely have a slightly different configuration of tank size, controls, and set point possibilities.

Table 1. Basic Equipment Characteristics

Component	GAU-315 EQTA
Tank Volume (Gallons)	84.6
Resistance Elements	None
Heat Pump* (W)	900 – 2,400
Standby (W)	< 1
Tank Heat Loss Rate (Btu/hrF)	4.7
Refrigerant	R-744 (CO_2)

*Includes compressor, circulation pump, and fan for both products. Range depends on water and ambient temperature.

Previous work summarized the HPWH performance in DHW-only mode, shown in Table 2 (Larson 2013). The tests conducted in the earlier work were based on the Department of Energy 24 hour test procedure draw profile (US DOE 1998). The standard ambient air conditions for that test are 67.5°F. These tests expanded on that temperature range. The Energy Factor (EF) follows the standard definition. The COP, Output Capacity and Input Power are all averages over the entire test. Output capacity was calculated using the tank temperature thermocouple readings: for each time step, capacity was computed as the product of mass of water in the tank, heat capacity of water, and incremental temperature difference. COP was then calculated as the ratio of output capacity defined in this way to measured input power. The difference between EF and COP is due to tank standby losses in the 24 hour period. The COP reports only the efficiency of the heat pump when it is running.

Table 2. DHW Only Tests: Efficiency, Output, & Input vs. Outside Temperature

Outside Air Temperature (F)	Energy Factor (EF)	COP	Output Capacity (kW)	Input Power (kW)
17	1.74	2.1	4.0	1.9
35	2.21	2.75	3.6	1.3
50	3.11	3.7	4.0	1.1
67	3.35	4.2	4.1	0.97
95	4.3	5.0	4.6	0.93

For this project, the particular temperatures of interest to heating are those of 17°F, 35°F, and 50°F. The table shows the equipment maintains a 4kW (13.6 kBtu/hr) output capacity over almost any temperature range. If run continuously, the system can supply 48kWh of heating over the course of a day which can be enough to provide the space and water heating needs for a low load house.

1.4.2 Heat Exchanger X-Pump Block

The heat exchanger between the heat source side and the hydronic loop side is the other principal component to the system. This design used the Taco X-Pump Block (XPB).[3] The XPB combines a heat exchanger and two pumps into one device. The heat exchanger is of the double-wall counter flow type. The hydronic load side pump is fixed speed and constant flow. The heat source side pump is variable speed and flow. The XPB contains sensors to measure temperatures on the incoming and outgoing water lines to balance the flows and achieve the desired heat transfer. There are numerous configuration options but the lab used the conceptually simple one of setting the hydronic load side to a target temperature. Given that this is a fixed temperature and flow, the XPB varies the flow rate on the heat source side to achieve the necessary heat transfer.

[3] https://www.taco-hvac.com/products/systems/radiant_systems/x-pump_block_xpb/index.html?action=file_list&category=345&type=2

2 Methods

In order to determine the viability and performance of the Sanden HPWH for combined space and water heating applications, Ecotope, in conjunction with the project advisory committee, drafted a lab test plan.

2.1 Test plan

The field study began in late 2014 and provided preliminary data on the system operation in early 2015. The data suggested several avenues of investigation which Ecotope used to draft the lab test protocol. Having the field test site up and running was useful in targeting the lab work. Ecotope considered using ASHRAE Standard 206 as the method of test but opted for a more customized and focused test plan (ASHRAE 2013). Plainly, the procedure to conduct Standard 206 was too lengthy and costly for the budgeted project. More practically, the field test sites raised numerous design questions which we wished to answer in the highly instrumented lab environment including space heating loop return water location. Answers to these questions would not be provided conducting the Standard 206 procedure. The scope of Standard 206 is for rating equipment while our questions had more to do with the balance of system design and determining appropriate applications.

Previous work has characterized water heating performance so this work focused on space heating applications and the combination of both. The test plan centered work in to three areas: measuring space heating only performance, measuring combined space and water heating performance, and mixing valve tests.

2.1.1 Space Heat Tests

The test objectives were to measure the system efficiency under low and high temperature configurations and observe the tank temperature stratification profile with different return water plumbing configurations. We examined three configurations:

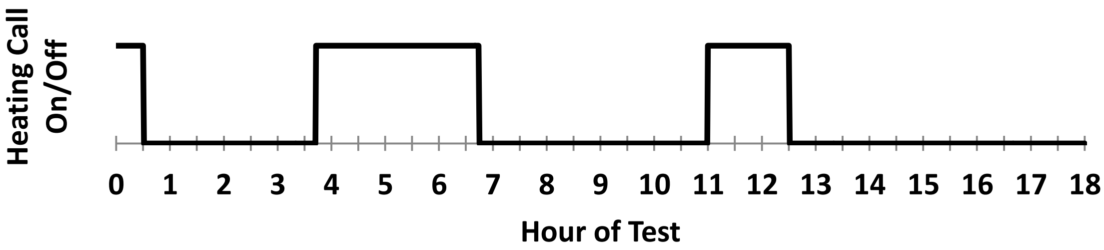
- Top of the tank, at the pressure and temperature relief port
- Top of the tank, at the pressure and temperature relief port with a custom diffuser
- Bottom of the tank, sharing the port traditionally used for cold city inlet water

Each plumbing configuration changes the water mixing and stratification within the tank. Thermocouples, immersed in the tank show how it changes. The different mixing regimes lead to differing amounts of hot water availability and operating efficiencies. The tests are intended to find the optimal design. Note that return water from the space heating loop must be reinserted into the tank at either the top or bottom: there is no intermediate port through which to direct spent water from the XPB.

Figure 5 shows the space heating demand profile used for all tests. It consists of three heating calls over the course of eighteen hours. The first heating call of thirty minutes starts the test. The second, lasting three hours, occurs between 3 hours 45 minutes and 6 hours 45 minutes. The third, and final, heating call begins at hour eleven and lasts 1.5 hours. The total heating time is five hours. The tests are run on an 18 hour cycle to allow the lab personnel to reset the equipment during normal working hours and run the tests overnight.

All heating tests were configured such that the return water on the hydronic load side entered the XPB 10F below its target supply value. That temperature rise, multiplied by the flow rate, yields the heating load. The hydronic side pump flows at ~3.75 GPM. For a 10F temperature difference, that amounts to 18.6 kBtu/hr (5.5kW). This heating load is higher than the output capacity of the heat pump but, because of the energy stored in the tank, the system can meet the load on a short-term basis. Over a full day, the five hours of heating demand amount to only 93.5 kBtu (27.4kWh) of heat. The tank, full of completely hot water (at 149°F), has ~49kBtu of stored heat available above 80°F and ~28kBtu above 110°F.

Figure 5. Space Heating Demand Profile

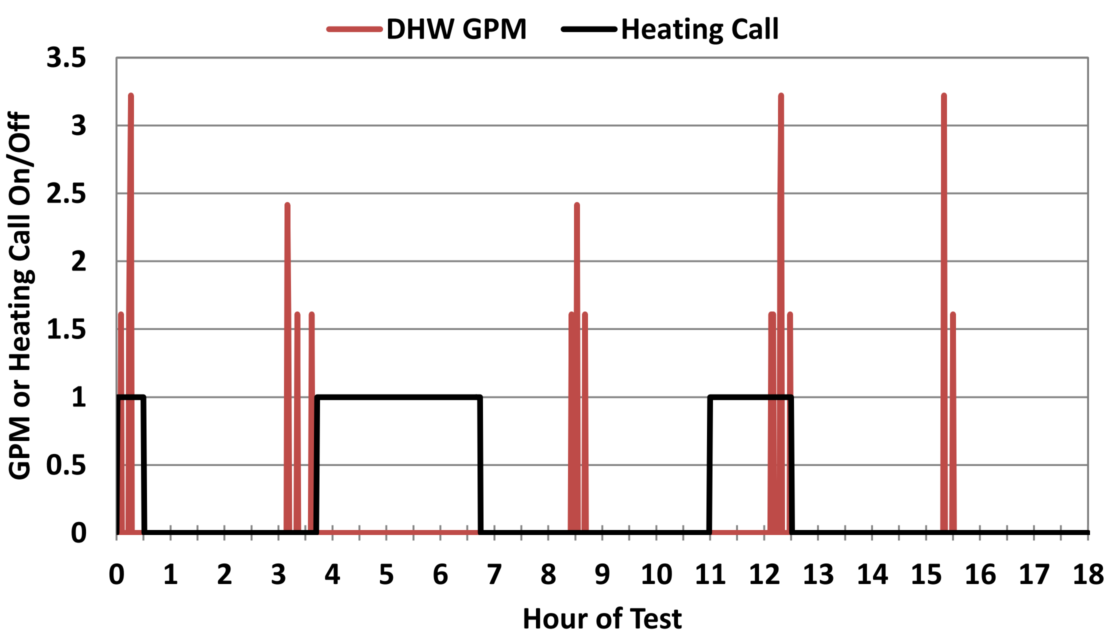


Hydronic loops can distribute heat to a house in many ways, of which three typical ones are: radiant slabs, forced-air fan coils, and radiators (like radiant baseboards or panels). Different distribution methods require different water supply temperatures. To explore the effect on the HPWH system, we opted to test a low temperature application, supplying water at 80°F to the hydronic loop and a higher temperature application at 110°F. In both scenarios, the tests were managed so that the return water was ~10F below the target set point. This enabled us to draw a constant, known, load. In all application scenarios (high and low temperature), the heating load was the same. The temperature difference and flow rates of the heat load side were constant. The only thing that varied was the heating load required temperature (80°F or 110°F).

2.1.2 Combination Tests

The combination tests build on the space heating only tests. They impose a hot water draw profile on top of the heating load. Figure 6 shows both loads on the same graph. The hot water draw profile is 46 gallons per day derived from a 3-person household as observed in PNW field studies (Ecotope 2015). The average occupancy house, of 2.7 people, uses 42 gallons, supplied from a tank with a 128°F setpoint, so this is only slightly more. These hot water draws directly remove hot water from the system placing a greater load on the heat pump. That water heating load (assuming a 75F temperature rise) is 28.6 kBtu. The combined energy load on the system is then 122.1 kBtu (35.8 kWh) in a given test.

Figure 6. Combined Space and Water Heating Demand Profile



Two combination tests were conducted - both for low temperature heating - one with the return water plumbed to the bottom inlet and the other plumbed to the pressure/temperature relief port using a custom diffuser. The diffuser consists of a copper tube with downward facing perforations meant to direct lukewarm water returning from the XPB to the lower reaches of the tank. The approximately 10” long tube is inserted at the return water location at the top of the tank (pressure/temperature relief port). Figure 7 shows a prototype and finished custom diffuser. Note the combined area of the outlet holes is much greater than that of a ¾” pipe which acts to reduce the water speed as it reenters the tank.

Figure 7. Custom Return Water Inlet Diffuser

2.2 Lab Testing Setup

Ecotope collaborated with Cascade Engineering and WSU to devise methods and protocols suitable for carrying out the testing plan. The general approach and methodological overview for this test are provided here.

The outdoor unit was placed in a thermal chamber where the ambient air conditions are tightly regulated (Figure 8). The hot water tank itself is placed next to the chamber in the large lab space (Figure 9). That lab space is kept thermally controlled only by a space heating thermostat. The temperature varied from 60°F to 70°F. The small changes in temperature will lead to slight changes in the heat loss through the tank but the impacts on the overall system efficiency measurements are minimal.

Figure 8. Sanden GAU Outdoor Unit Installed Inside Thermal Chamber

Figure 9. Hot Water Tank Instrumented and Installed Adjacent to Thermal Chamber

2.2.1 System Layout

Figure 8 presents the detailed system layout and measurement points. Ecotope drafted similar plans for use in house installations. This one is specific for the lab testing. The figure shows the plumbing configuration using the custom diffuser. Not shown are the configurations without the diffuser (the return water piping is the same) and the return to the bottom inlet of the tank. For the latter, the water return from the XPB is routed, with a “Tee” fitting, to the bottom inlet between the tank and the temperature and flow sensors.

Additional changes made to the implementation and not shown in the figure is a closed loop on the heating load side. Original plans called for an open loop on the heating load side (drawing on tap water and then dumping it down the drain) but Cascade Engineering devised a reliable way of using a closed loop to provide a constant load (heat sink). They built a rudimentary cooling tower which sprayed the heated water in to a 300 gallon tank of water. The water cooled off and the tank of water remained at reliable temperatures throughout the testing. The cooling off of the water was enough to emulate the design building heating load we wished to observe.

The following is a list of the system components and measurement points to provide a key to Figure 10.

System Components:

- Heat Pump Outdoor Unit, Model GAU-A45HPA, (HP-1)
- Storage Tank, 80 gallon, Model GAU-315EQTA, (ST-1)
- Expansion Tank (EXP-1)
- Pump, Taco XPB-DW-1, (XPB-1)
- Instantaneous Electric Heater (EH-2)

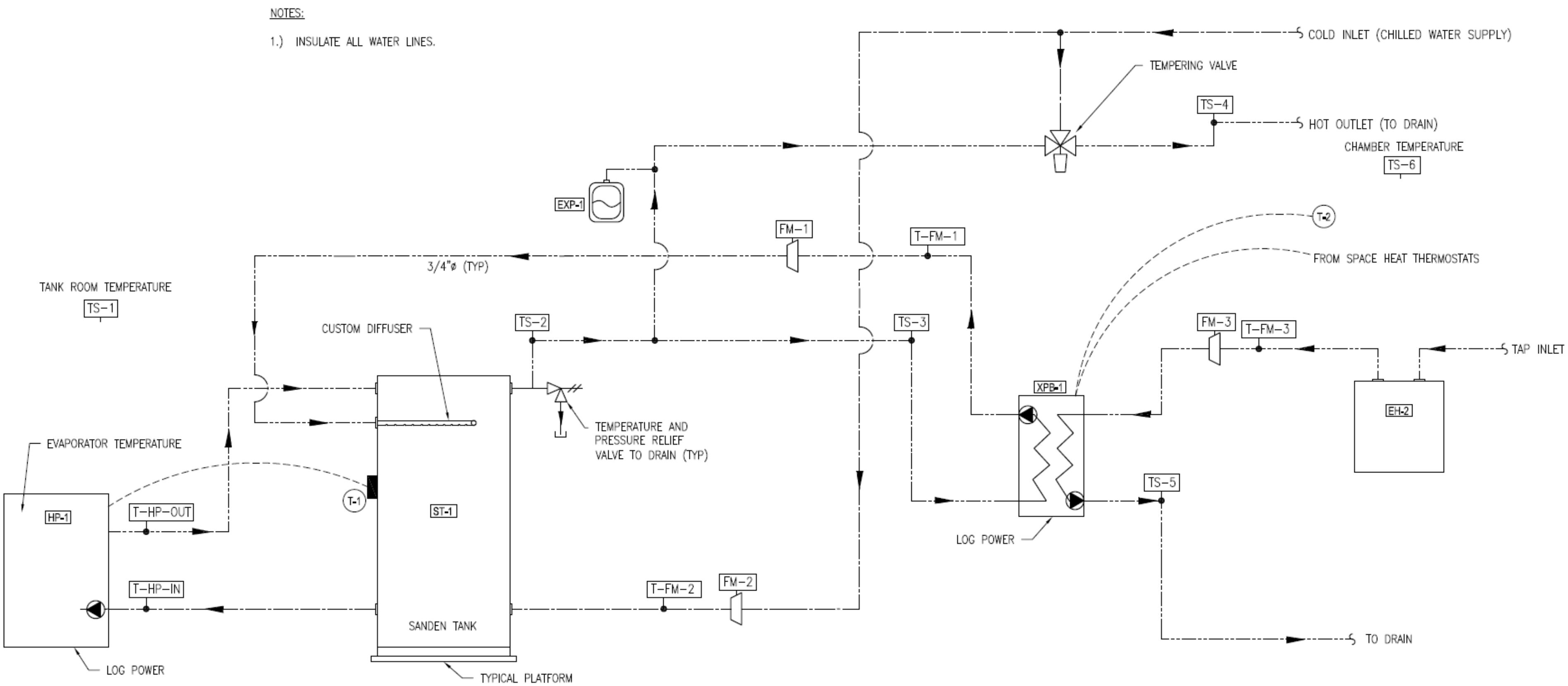
Measurement Points:

Power Measurements

- Heat Pump, complete outdoor unit, (HP-1)
- Taco X-Pump-Block, (XPB-1)

- Heat Pump Measurements (Outdoor Unit)
  - Evaporator surface temperatures:
    - Refrigerant Inlet
    - Mid-Point
    - Refrigerant Outlet
  - Exhaust air temperature, (T-EXH)
  - Cold Inlet Water T, (T-HP_{IN})
  - Hot Outlet Water T, (T-HP_{OUT})
  - Chamber Temperature and RH (TS-6)
- Tank Measurements
  - Lab air temperature (TS-1)
  - Tank thermocouple temperature tree (13 sensors)
    - Spaced so each represents an equal-volume segment
  - Cold Inlet Water (T-FM-2)
  - Hot Outlet Water (TS-2)
  - Cold Water Inlet Flow (FM-2)
- Measurements for/near Taco X-Pump-Block, (XPB-1)
  - Heating Supply Water T, (TS-3)
  - Heating Return Water T, (T-FM-1)
  - Heating Water Flow (FM-1)
  - Load Supply Water T, (TS-5)
  - Load Return Water T, (T-FM-3)
  - Load Water Flow (FM-3)
- Measurements near Mixing Valve
  - Mixing Valve Outlet Water T, (TS-4)

Figure 10. Detailed System Schematic



2.2.2 Measurement Equipment and Specifications

In accordance with Figure 10, Cascade Engineering installed an instrumentation package to measure the required points. All instrumentation was calibrated and has accuracies well within those specified by the Department of Energy for water heater and heat pump testing. Cascade Engineering measured inlet and outlet water temperatures with thermocouples immersed in the supply and outlet lines. Three thermocouples mounted to the surface of the evaporator coil at the refrigerant inlet, outlet, and midpoint monitored the coil temperature to indicate the potential for frosting conditions. Power for the equipment was monitored for the entire unit including the compressor, fan, and pump all at once.

Tank mixing and stratification was of utmost interest in the lab tests, so Cascade Engineering devised a method to insert thirteen temperature sensors at different heights, corresponding to equal water volume segments, within the tank. Most electric water heaters have an anode rod port at the top of the tank which offers convenient access for inserting a straight thermocouple tree near the central axis. Because the Sanden tanks are all stainless steel and there are no resistance heating elements, there is no need for an anode rod. Without the convenient anode port, Cascade Engineering used the pressure and temperature relief valve port for access. With the unique “fishing rod” approach, the thermocouples hang freely in the tank so, to keep them positioned, the lab attached weights to the bottom of the thermocouple wire. Further, the lab fabricated two different rods – one integrating the custom diffuser, with the thermocouple wire affixed on top, and the other with a standard piece of pipe. In all instances, extra care was taken to minimize disruption to the normal water flow regime.

For the DHW flow events, Cascade Engineering conditioned and stored tempered water in a large tank to be supplied to the water heater at the desired inlet temperature. A pump and a series of flow control valves in the inlet and outlet water piping control the water flow rate. A flow meter measures and reports the actual water flow. The data acquisition (DAQ) system collects all the measurements at five-second intervals and logs them to a file. In a post processing step, Ecotope merged the temperature log of the thermal chamber with the DAQ log file to create a complete dataset for analysis.

3 Findings

To begin, all laboratory testing data have been uploaded to an interactive dashboard at the following web address: http://ecotope.shinyapps.io/Sanden_Combi_Lab. The custom analysis tool – written with the statistical software R and RStudio add-ons – allows one to view a selection of measured and/or derived variables for one or more tests. The test results can be viewed graphically or numerically. The findings section of this report contains a selection of curated views from the web dashboard.

Secondly, because tank stratification proves so crucial to the success of a HPWH, we introduce a "Stratification Index" (Appendix B: Stratification Index Definition) to quantify the extent to which the water tank remains stratified. This index spans a continuous scale from zero to one, where zero indicates no stratification at all, i.e. complete mixing, and one indicates perfect stratification, i.e. an imaginary barrier with all hot water above and all cold water below. The graphics in this section, at times, compare the stratification indices between tests, with the premise that more stratification is better (hot water available for delivery at the top of the tank and cold water available at the bottom to be efficiently heated).

In addition we should note a number of inconsistencies in test execution that add difficulties to the analysis. These include several instances of the hot water tank beginning a test not fully heated, and a mistake in the timing of a space heating interval. An unforeseen complication also arose in the low temperature space heating test in which the tank unexpectedly did not recover after the final space heating interval. These nuances do not detract from the usefulness of the study, but do create some additional hurdles to jump when examining the findings.

3.1 Overall Findings

Table 3 presents summary values for all tests. In general, the low temperature tests displayed greater stratification, and hence greater efficiency, than the high temperature tests. Further, in general, the bottom inlet return location displayed greater stratification, and hence greater efficiency, than either the diffuser or no diffuser case at the top of the tank. This is demonstrated in the "HP Waterline In" (the temperature of water sent to the Sanden gas cooler for heating), "XPB Sanden Supply Temp" (the temperature of water sent from the Sanden to the XPB), the HP COP, and the System COP. The HP COP is the efficiency of the heat pump, when it runs excluding tank standby losses. The System COP includes the standby losses in the calculation.

In the high temperature tests – meant to represent a fan-coil or radiator heating system – the supply water for heating was roughly 110°F. These tests met an equivalent heating load but with the need for higher supply water temperature. The return water from the XPB to the tank was nearly the same. The flow rate required from the tank to supply 110°F water to the heating system was around 3.75 GPM, which mixed the hot water tank quickly during heating events. This mixing resulted in lower hot water availability, and also in warm water sent to the heat pump as demonstrated with the nearly 110°F temperature "HP Waterline In." There was so much water turnover in the tank that the water temperature being returned to the tank from the XPB was essentially identical to that being sent out to the heat pump for heating. This caused the lower COPs of 1.6 to 1.7. The tank mixing in this scenario was extreme enough that the location of the return water from the XPB played only a minor role. Under no scenario did the tank maintain stratification, as seen in the similar temperatures supplied to the XPB and the heat pump.

The low temperature tests – both space-heating-only and combination – showed more differences between the three XPB return water scenarios. The bottom inlet configuration maintained more tank stratification, followed by the top of the tank with the diffuser, followed by the top of the tank with no diffuser. This can be seen in greater supply temperatures from the Sanden to the XPB, lower waterline inlet temperatures to the heat pump, and higher COP. Not listed in the figure is the hot water temperature delivered in the combination tests. For the bottom inlet case it was 149.5 °F, while for the diffuser case it was 138.2 °F. Returning water to the bottom inlet increased overall hot water availability.

Table 3. Test Summary Statistics

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

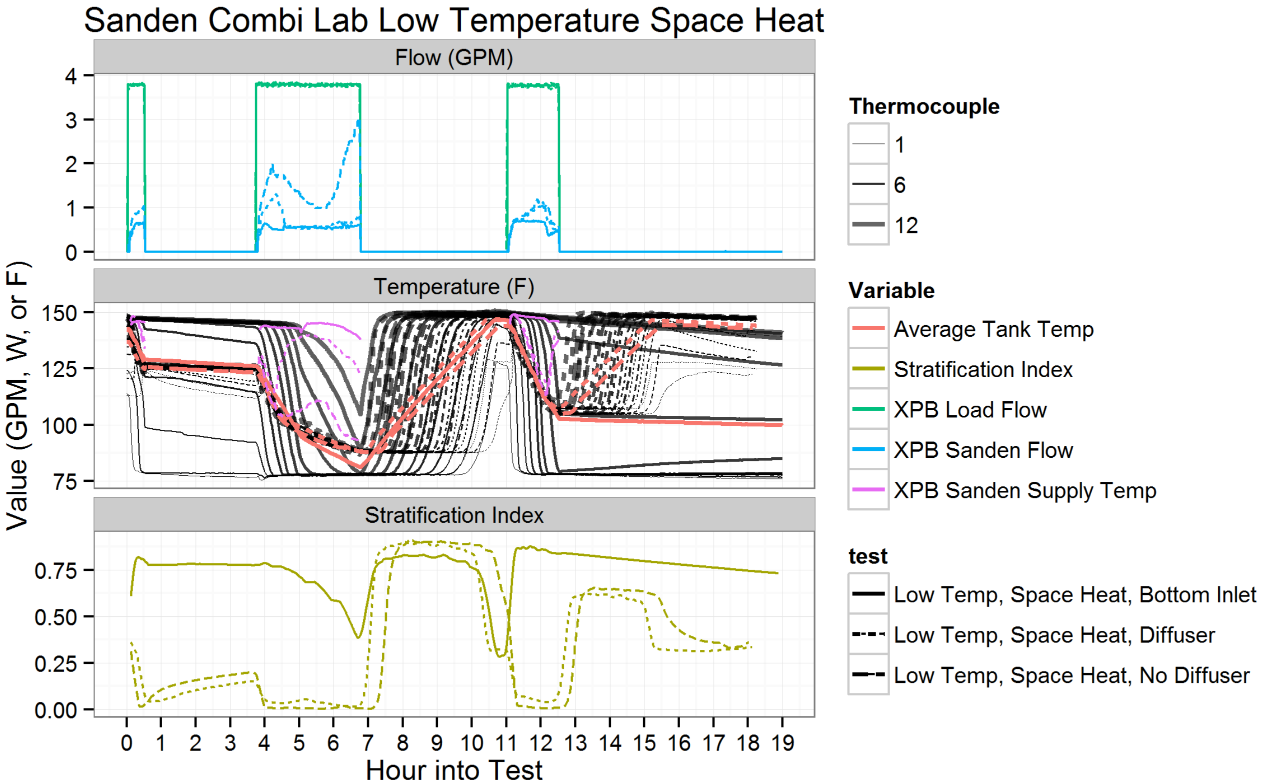
3.2 Space Heating Tests

3.2.1 Low Temperature Space Heating Tests

Figure 11 shows flow, tank temperatures, and stratification for the three low temperature space heat tests. These tests were meant to represent a heating-only scenario for a radiant floor. The three intervals of space heating are visualized in the XPB Load and Sanden Flow variables – whenever there is a call for heating, there is flow past these sensors. A minor difference in initial tank temperature occurred during the bottom inlet test, but this discrepancy did not change the interpretation of the results.

The load side flow from the XPB for the radiant slab was constant at roughly 3.75 GPM, and the Sanden side flow modulated depending on Sanden supply temperature to the XPB. Immediately striking was that the bottom inlet case maintained decent stratification during the space heating events, whereas both cases with XPB return plumbed to the top of the tank de-stratified almost immediately upon triggering of the space heating load. The magenta line for XPB Sanden supply temp shows one consequence of the immediate de-stratification, which was a lower temperature of water delivered from the Sanden to the XPB and hence the higher flow rate. The middle box of Figure 11, expanded into its own figure, can be found in Appendix C: Full Page Graphics. Unexpectedly, the bottom inlet scenario remained so stratified that the tank did not experience a recovery after the final space heating event. Consequently, the tests should only be compared through hour 11.

Figure 11. Low Temperature Space Heating

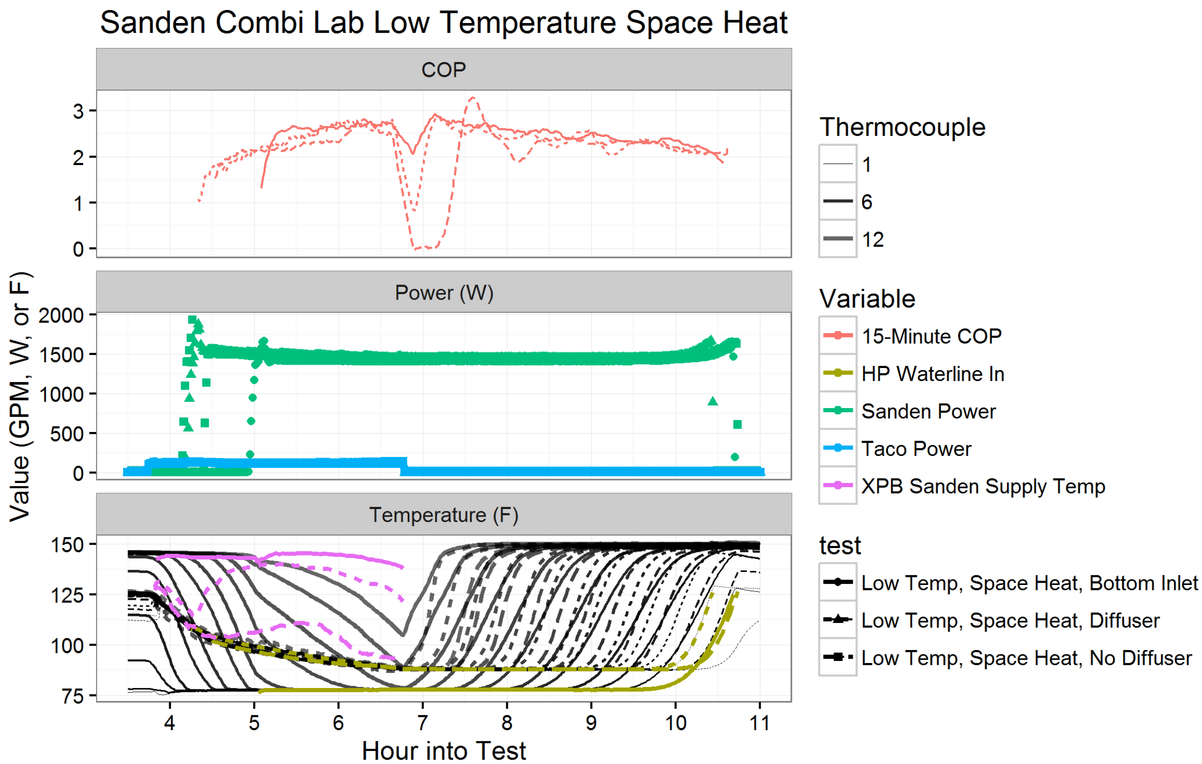


Zooming in closer on the second space heating event and subsequent recovery gives the following graphic, Figure 12. This figure introduces a temporally smoothed "instantaneous" COP based on the average tank temperature, the amount of heat transferred at the XPB, and the input power of the Sanden. This is only defined during operation of the Sanden unit. The intervals in which the COP drops – just after termination of the space heating load – is an artifact of the discrete thermocouples and the time lag between heating water and when that heat is observable at the thermocouple locations.

In this figure we see a bit more clearly the benefit of the diffuser. Even though the tank had completely de-stratified during the space heating event, while running, the Sanden unit was still able to provide hot water to the XPB. This is because the hot water injected at the top of the tank by the Sanden was available to the XPB, rather than immediately mixing with lukewarm return water from the XPB. The dashed line, representing the "no diffuser" case, shows that, as the tank mixed during the space heating event, the apparatus struggled to maintain a supply of hot water to the XPB.

The flip side of the stratification – maintaining cold water at the bottom of the tank – shows that the "bottom inlet" case was able to send colder water to the Sanden unit and thus operate at a higher efficiency.

Figure 12. Low Temperature Space Heating – First Eleven Hours



Numerically, Table 4 describes the low temperature space heating tests from hours zero to 11 (since the bottom inlet scenario did not recover after the final space heating interval). In all three arrangements the Sanden generated 150°F water, although the consequences of tank mixing were evident with higher "HP Waterline In" temperatures in the "Diffuser" and "No Diffuser" cases where XPB return water was plumbed to the top of the tank. In addition, the supply water to the XPB was highest in the bottom inlet configuration at 143°F, followed by the "Diffuser" at 134°F, and finally "No Diffuser" at 111°F. This was also evident in the differing flow rates from the hot water tank to the XPB. The ultimate consequence for efficiency was a range of COP from 2.0 to 2.2, with the "Bottom Inlet" scenario the most efficient and the "No Diffuser" scenario the least efficient, with the "Diffuser" in the middle. This follows mainly from the differences in tank mixing and stratification.

Table 4. Low Temperature Space Heat Test Summary Statistics

Summary Statistic	Low Temp, Space Heat, Bottom Inlet	Low Temp, Space Heat, Diffuser	Low Temp, Space Heat, No Diffuser
HP Waterline In (F)	80.3	92.4	91.3
HP Waterline Out (F)	151.0	150.5	150.5
Taco Load Flow (GPM)	3.8	3.8	3.8
Taco Sanden Flow (GPM)	0.6	0.7	1.5
Taco Load Return Temp (F)	71.8	71.3	70.0
Taco Load Supply Temp (F)	81.4	81.1	80.4
Taco Sanden Return Temp (F)	77.9	78.4	80.2
Taco Sanden Supply Temp (F)	143.7	133.8	111.0
Stratification Index	0.72	0.33	0.33
Sanden Energy (kWh)	8.18	9.19	9.37
Taco Energy (kWh)	0.41	0.40	0.44
Total Energy (kWh)	8.58	9.59	9.81
Space Heat Output (kBtu)	61.63	62.47	67.10
Total Output (kBtu)*	64.67	69.15	67.94
Equipment COP	2.49	2.30	2.10
System COP	2.21	2.11	2.03

*Adjusted for differences in starting and ending average tank temperature

3.2.2 High Temperature Space Heat Tests

The high temperature space heating tests were meant to represent a radiator or fan-coil based heating system. In these tests the XPB supplied roughly 110°F water to the heating system. In actual application, such systems may well need to be supplied with even hotter water. There was a mistake in the test execution for the diffuser case in which the final space heating interval ran roughly two hours later than scheduled. In addition, the tank temperature was not identical at the start of each test. These problems were not deemed important enough to re-run the tests and mathematical adjustments were made wherever necessary to achieve comparable data.

In all cases, the tank de-stratified almost immediately due to the high flow rate required from the hot water tank to the XPB (Figure 13). At a flow rate of 4 GPM, even a completely hot tank will turn-over completely in 20 minutes. The location in the tank of the XPB return and presence of the diffuser made little difference in supply temperature from the hot water tank and inlet temperature to the heat pump. Due to the rapid mixing and warm inlet water to the heat pump, these tests showed much lower efficiency than the low temperature tests. As discussed in the introduction, CO_2 transcritical heat pumps work most effectively when inducing a large temperature lift, which is not the case with a high temperature space heating application.

Figure 13. High Temperature Space Heating

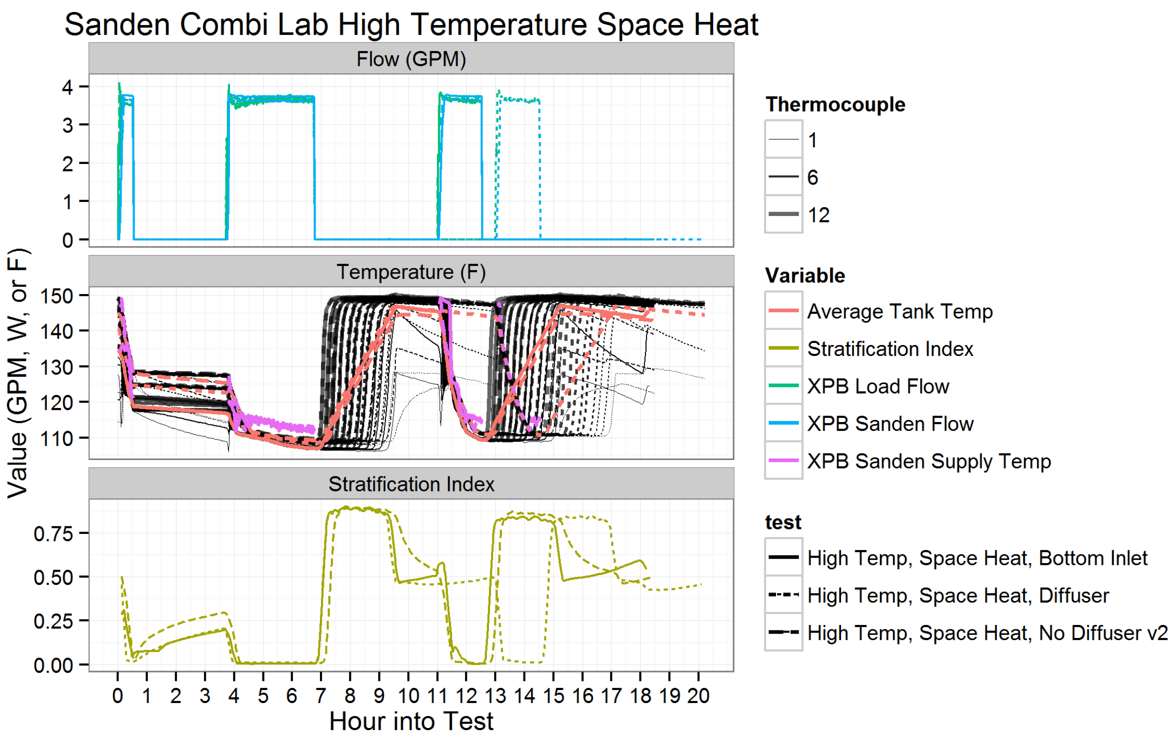


Table 5 tabulates summary statistics for the three return water scenarios in high temperature space heating. The story is largely the same between the three. Returning 3.6 GPM of 110°F water to the hot water storage tank causes rapid mixing in all cases, which leads to uncomfortably high heat pump waterline temperatures – on average sending 110°F water to the heat pump for heating. This results in COPs of 1.6 to 1.7 pulling from 35°F outdoor air, contrasted to COPs up to 2.2 with low temperature space heating.

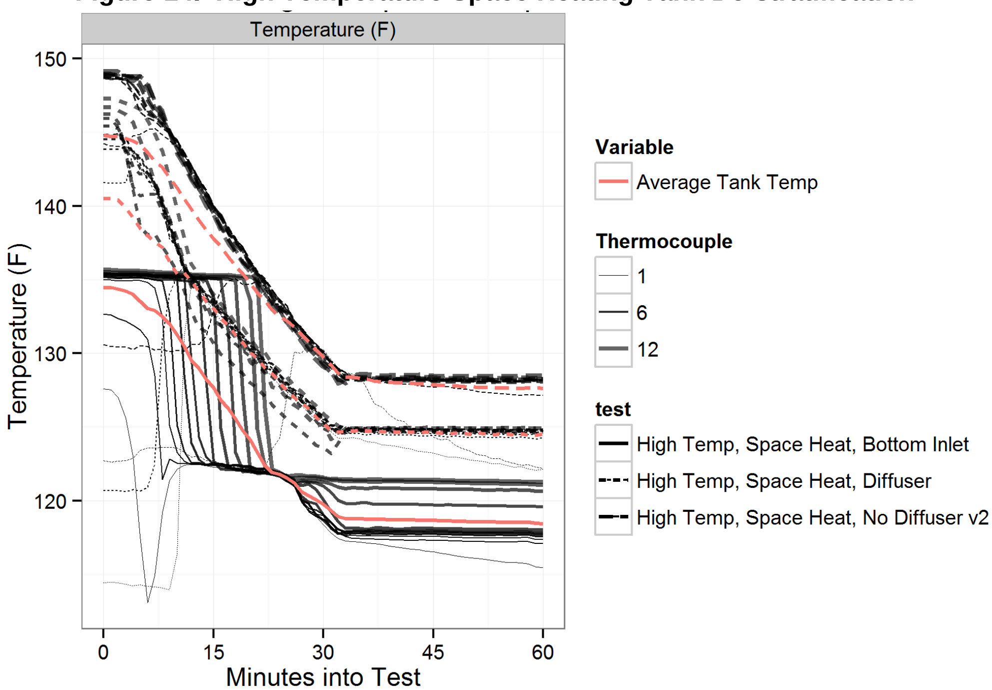
Table 5. High Temperature Space Heat Test Summary Statistics

Summary Statistic	High Temp, Space Heat, Cold Inlet	High Temp, Space Heat, Diffuser	High Temp, Space Heat, No Diffuser v2
HP Waterline In (F)	108.8	109.9	110.6
HP Waterline Out (F)	150.5	150.4	150.6
Taco Load Flow (GPM)	3.6	3.6	3.6
Taco Sanden Flow (GPM)	3.6	3.5	3.5
Taco Load Return Temp (F)	100.2	100.3	100.9
Taco Load Supply Temp (F)	109.1	108.7	109.3
Taco Sanden Return Temp (F)	110.2	109.7	110.6
Taco Sanden Supply Temp (F)	119.4	118.7	119.5
Stratification Index	0.40	0.38	0.42
Sanden Energy (kWh)	14.22	12.80	13.01
Taco Energy (kWh)	0.90	0.91	0.89
Total Energy (kWh)	15.12	13.71	13.89
Space Heat Output (kBtu)	79.10	74.73	75.12
Total Output (kBtu)*	86.70	77.66	73.77
Equipment COP	1.90	1.90	1.67
System COP	1.68	1.66	1.56

*Adjusted for differences in starting and ending average tank temperature

Whereas the observed efficiency was slightly different between the three return water scenarios, this is in part an artifact of inconsistencies in testing procedures. Figure 14 shows the tank temperature distributions for the first hour of the test. The “Bottom Inlet” scenario started the test with an average tank temperature around 135°F, the “Diffuser” at 140°F, and the “No Diffuser” near 145°F. The differences in observed efficiency are partially a product of these initial differences in average tank temperature. Since the tank fully mixed in all scenarios, the bottom inlet case saw the highest efficiency partly because it started with the coolest tank, although it was a subtle difference. Regardless of the difference in COP tabulated above, these three scenarios should be treated as mostly equivalent with respect to efficiency. Although note that the bottom inlet did maintain complete hot water availability at the top of the tank for twenty or so minutes.

Figure 14. High Temperature Space Heating Tank De-stratification

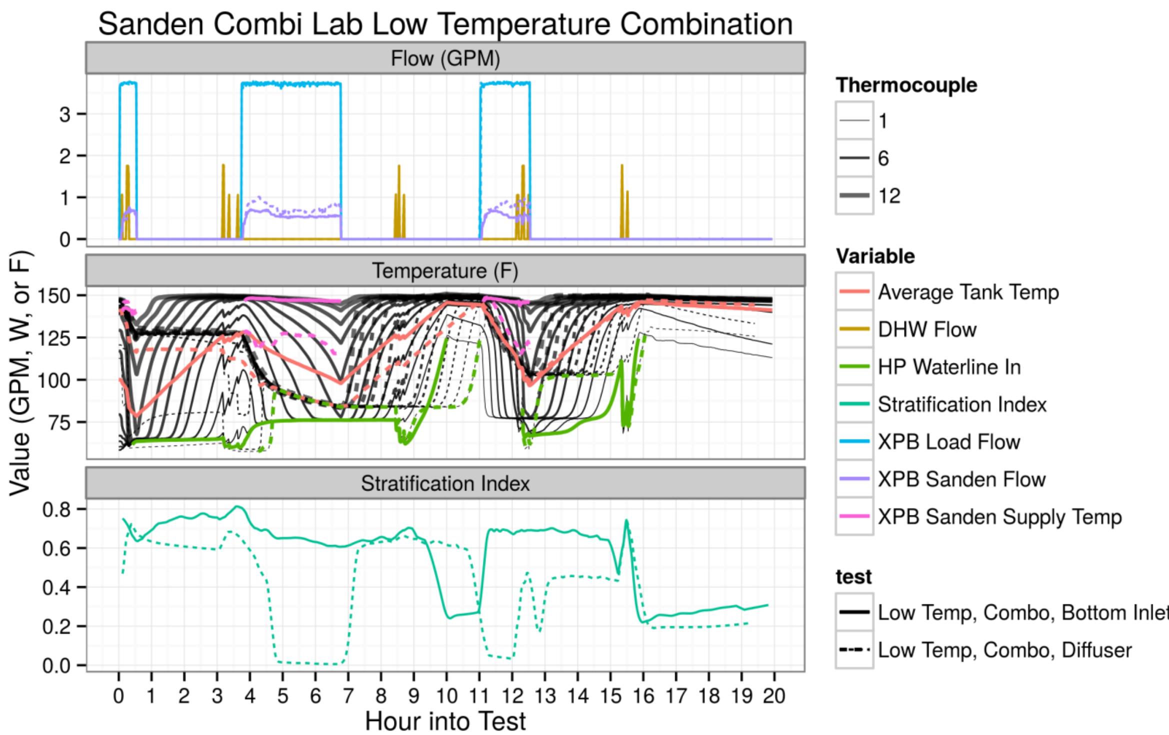


3.3 Combination Space and Water Heating Tests

The results for combined space and water heating were similar to the space-heating-only low temperature tests. Returning XPB water to the bottom of the tank led to better stratification and better performance than returning to the top of the tank. Although there was no test of the “No Diffuser” scenario for the combined space and water heat, it seems likely that the diffuser was again an improvement when returning to the top of the tank, but not as effective as returning to the bottom of the tank.

Most importantly, there did not appear to be any deleterious interactions between the two load types. If anything, the water heating helped by bringing cold water to the bottom of the tank. The Sanden system works best when heating cold city water, rather than lukewarm or warm water returned from the space heating loop.

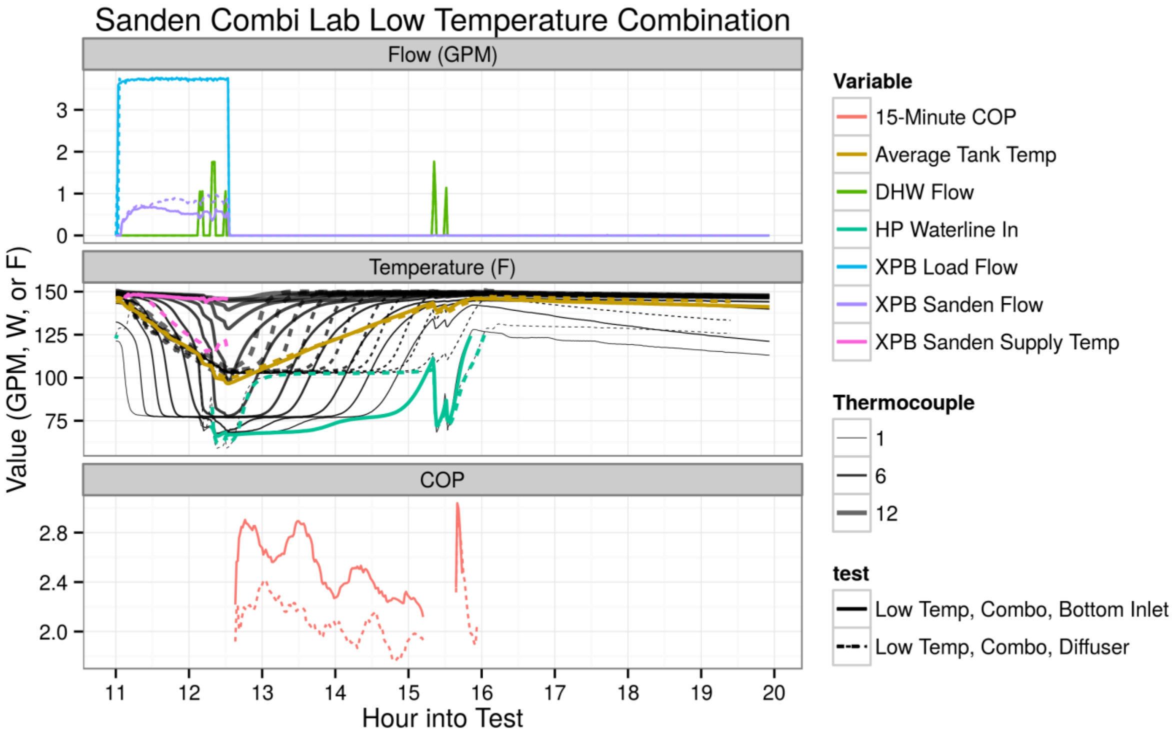
Figure 15. Combination Space+Water Heating Tests



Unfortunately, the “Bottom Inlet” test did not start with the tank fully heated which means that a fair comparison between tests could only be made between hours 11 and 20. Figure 16 shows the tests for that period.

The difference in COP was fairly stark, and driven largely by the difference in HP waterline temperature. Notice that with better stratification the bottom inlet scenario sends ~75°F water to the heat pump, whereas the diffuser scenario sends ~100°F water. In addition, the bottom inlet scenario sends constant 150°F water to the XPB. The outlet water temperature sent to the XPB in the diffuser case flagged as the tank mixed, with a corresponding increase in XPB Sanden-side flow. Once again, we saw that an improvement in stratification caused an improvement in efficiency and hot water availability. It should be noted that the fluctuations in COP are an artifact of using a discrete number of thermocouples to measure temperature: the output capacity is derived from the change in average tank temperature, and as the column of hot water shifts downward during heat pump heating, the finite spacing of the thermocouples causes a perceived oscillation in output capacity that is really just a limitation of calculating average tank temperature from the fixed locations. A vertically continuous measurement of tank temperature would eliminate the perceived oscillations.

Figure 16. Combination Tests – Hours 11-20



The following summary statistics of Table 6 are from hour 11 to finish of the test – the only comparable time period due to the inconsistency in starting tank temperature. The bottom inlet saw an average of 74°F water in the heat pump waterline, an improvement from the 80°F waterline temperature of the space-heating-only test. This led to a slight increase in COP, from 2.2 to 2.3. Otherwise the introduction of a water heating load to the space heating profile caused very little change in operational profile.

Table 6. Low Temperature Combination Test Summary Statistics

Summary Statistic	Low Temp, Combo, Cold Inlet	Low Temp, Combo, Diffuser
HP Waterline In (F)	73.7	88.2
HP Waterline Out (F)	150.8	150.7
DHW Flow (GPM)	38.4	38.4
Taco Load Flow (GPM)	3.7	3.7
Taco Sanden Flow (GPM)	0.6	0.7
Taco Load Return Temp (F)	69.5	70.5
Taco Load Supply Temp (F)	80.2	80.2
Taco Sanden Return Temp (F)	76.5	78.0
Taco Sanden Supply Temp (F)	146.6	127.6
Stratification Index	0.57	0.4
Sanden Energy (kWh)	18.62	15.68
Taco Energy (kWh)	0.56	0.58
Total Energy (kWh)	19.18	16.26
Space Heat Output (kBtu)	95.96	87.23
Domestic Water Output (kBtu)	25.4	22.22
Total Output (kBtu)	150.37	111.7
Equipment COP	2.61	2.29
System COP	2.3	2.01

*Adjusted for differences in starting and ending average tank temperature

3.4 Evaporator Temperatures

To obtain a non-invasive view in to the operating state of the refrigeration cycle we installed surface-mount thermocouples at three locations along the evaporator heat exchanger (inlet, midpoint, and outlet). The difference between the three XPB return locations amounted to 2-3 degrees (F) in the evaporator. Table 7 shows mean evaporator temperatures for the low temperature space heating tests during the primary recovery interval between hours 6 and 10. In general the "no diffuser" arrangement produced the coldest evaporator temperatures, the "diffuser" arrangement produced the warmest evaporator temperatures, with the bottom inlet case in between. The average temperature differences are small and there is likely no meaningful difference in operation between the scenarios.

Table 7. Evaporator Temperatures

Test	Evaporator In T (F)	Evaporator Middle T (F)	Evaporator Out T (F)
Low Temp, Space Heat, Bottom Inlet	22.6	24.0	25.3
Low Temp, Space Heat, Diffuser	24.4	25.9	27.2
Low Temp, Space Heat, No Diffuser	21.3	23.1	24.7

3.5 Taco X-Pump Block Heat Transfer

Here we discuss the observed heat transfer characteristics of the space heating loop Taco X-Pump Block (XPB). It appears as though better heat transfer effectiveness was realized at the XPB with a low flow rate of extremely hot water from the Sanden. Table 8 shows the heat transfer effectiveness[4], defined as the ratio of temperature reduction on the Sanden side to the total possible temperature reduction (cooling the Sanden side flow to the temperature of the space heating return). The highest effectiveness occurred in both low temperature space heating, bottom inlet return configurations. As we saw above, those scenarios best preserved tank stratification and were able to send a low flow rate of very hot water. Within the low temperature space heating regimes, as we saw previous, the diffuser and no diffuser top of the tank return exhibited progressively more tank mixing, which caused higher flow rates of cooler outlet water sent to the XPB. This evidently resulted in lower effectiveness at the XPB. All high temperature space heating tests showed much lower effectiveness at the XPB.

Table 8. XPB Heat Transfer Effectiveness

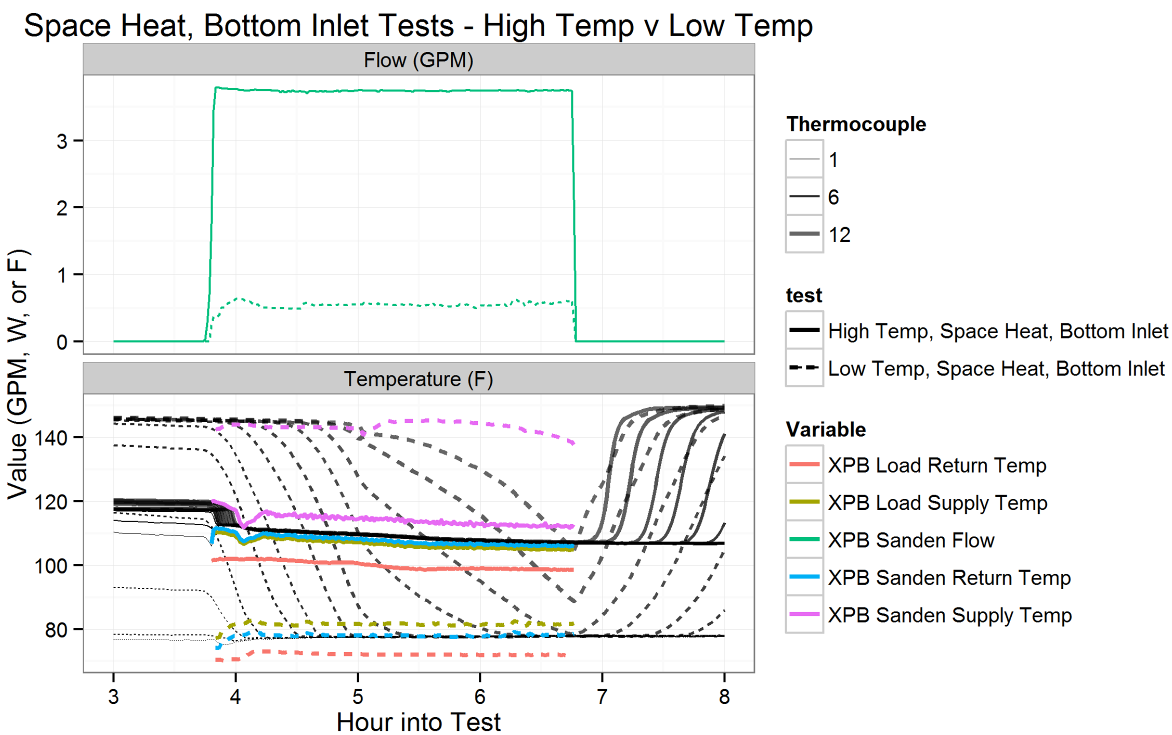
Test	XPB Return Location		
	Bottom Inlet	Diffuser	No Diffuser
Low Temp Space Heating	0.91	0.88	0.76
High Temp Space Heating	0.46	0.48	0.47
Low Temp Space Heating + DHW	0.91	0.87	

Figure 17 illustrates how a higher heat transfer effectiveness leads to more desirable system operation. Both high and low temperature scenarios are considered. As we have seen before, the superior stratification of the bottom inlet, low temperature space heating case results in 140-150°F water supplied from the hot water tank to the XPB at all times (the dashed magenta line). This resulted in a low flow rate sent to the XPB and an extremely large temperature drop on the Sanden side, all the way to 80°F. In contrast, the mixed tank in the high temperature space heating test sent ~120°F water to the XPB and cooled it around 10 degrees (F), half of the potential 20 degrees (F). Once again we see superior performance corresponding to greater tank stratification. Overall, a

[4] Heat transfer effectiveness is not a measure of how much heat was transferred (and lost) across XPB but rather of how well the heat exchanger does at reducing the temperature of the Sanden side flow while transferring the energy described in that temperature reduction to the heating load side. It is the ratio of actual heat transfer to maximum possible heat transfer. For example, take hot water entering the exchanger at 150°F and leaving at 90°F, and cold water entering at 70°F and leaving at 80°F. The heat exchanger could have cooled the hot water down to 70°F but only did so to 90°F. Therefore the effectiveness is (150-90)/(150-70) or 0.75.

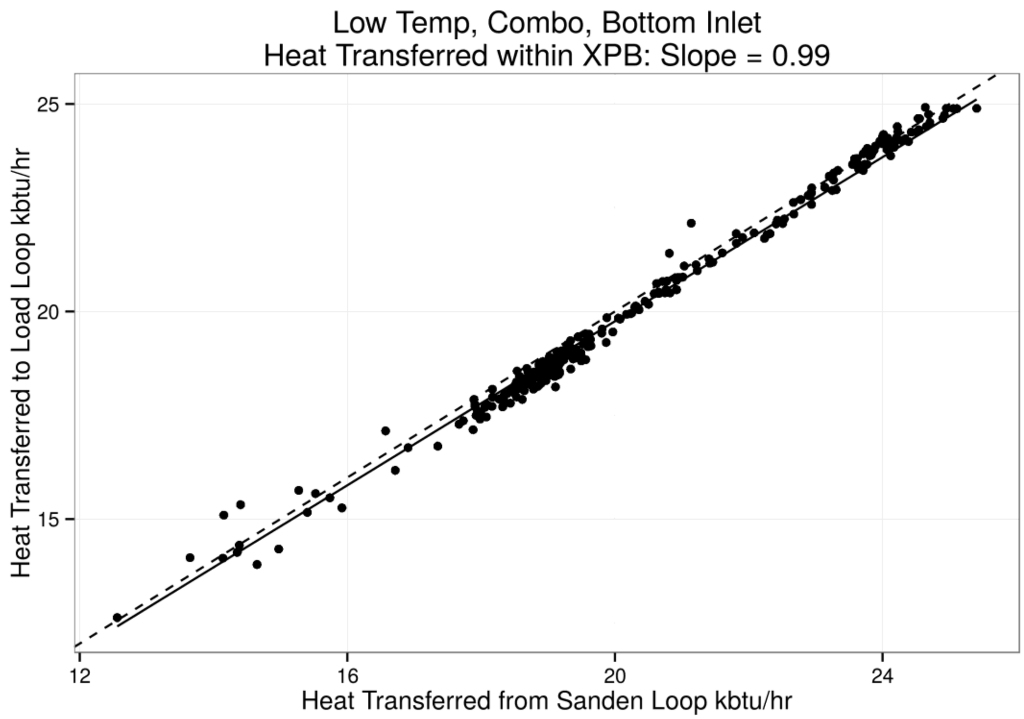
higher heat transfer effectiveness is more desirable for optimum system operation. It would allow the Sanden to heat the coolest water possible at all times.

Figure 17. Comparative Heat Transfer Effectiveness Outcomes for Space Heat Bottom Inlet Tests



Another consideration (and good sanity check on measurements) is to examine heat loss within the XPB itself. With flow rates and temperatures of the water lines on both sides of the XPB, we compared the calculated heat transfer from the Sanden side and load side. This is demonstrated in Figure 18 for the low temperature, bottom inlet, combination space and water heating test. On average in that test, the heat transferred to the load side of the XPB was 99% of the heat lost from the Sanden side of the XPB. Other tests similarly showed negligible heat loss from the XPB itself.

Figure 18. Heat Transfer within the XPB



3.6 Discussion

3.6.1 Annual Efficiency Estimates

With combination equipment, there are three possible usage scenarios: water heating only, space heating only, and combined space & water heating. This project directly observed the latter two and draws on previous work for the water heating only information (Larson 2013). Table 9 summarizes the most salient test measurement results for efficiency comparisons. Again, "HP COP" can be thought of as the operating efficiency of the heat pump, while "System COP" calculates the efficiency including tank standby losses. With low temperature space heating, a bottom inlet return location from the heating loop yielded the best results, so that configuration is displayed. With high temperature space heating, the heating loop return location made little difference (hence, any scenario). Overall, the table shows that low temperature space heating, either with or without water heating, can provide a similar system COP to water heating alone. The introduction of warm return water from space heating reduced the HP COP slightly, but the increase in overall heat demand lowered the relative impact of standby losses, leading to comparable system COP.

Table 9. Lab Measured Efficiency at 35°F Ambient Air

Test	HP COP	System COP
Water Heating Only	2.75	2.2
Space Heating Only (Low T. Bottom Inlet)	2.5	2.2
Space Heating Only (High T. Any scenario)	1.9	1.7
Combination (Low T. Bottom Inlet)	2.6	2.3

The current lab testing explored the space heating performance at only 35°F while previous work mapped water heating performance over a range from 17-95°F. Given that the low temperature space and water heating performance matched well at 35°F, it is reasonable to assume that performance will match at other ambient conditions. Consequently, we can leverage the existing performance map to predict combined space and water heating performance over a broad temperature range. The water heating system efficiency was shown to be a linear function of outdoor temperature: 0.033*temperature + 1.196, with temperature in Fahrenheit (Larson 2013). Note that at 35°F, the linear prediction would give a COP of 2.35 whereas Table 9 reports the specific measured efficiency. In the previous lab test work, the data used to derive the fit contained COPs below the predicted line at 35°F, above it at 50°F, and more or less on the line at 17°F, 67°F, and 95°F.

Given a system performance function, we can estimate annual system efficiency by applying it to different climate temperature profiles. The procedure is a standard temperature-bin weighted calculation using TMY3 data to provide the number of hours for each temperature bin. Consider a well-insulated house with a heat loss rate of 300 Btu/hr/F, a balance point of 55°F, and radiant floor with the optimal plumbing configuration. Note that the Sanden heat pump has a near constant output of 4kW, which is relatively small for space heating, so it makes the most sense to install it in a well-insulated building. Under scenarios where the outdoor temperature drops below the point at which the Sanden can fully meet the load, we add in auxiliary resistance heat. For simplicity in this example, assume constant water heating load in each temperature bin of 46 gallons per day – an average three person use pattern.

Table 10 presents the results of the annual efficiency analysis. The Water Heating column shows system efficiency providing water heating only, Space Heating shows space heating only, and Combined shows the heat pump providing both services. Since space heating always happens at cold temperatures, its efficiency is necessarily less than water heating on an annual basis. Water heating sees a significant efficiency boost from warmer, summertime outdoor temperature. The combined result is the time- and temperature-bin-weighted result. Since most of the system energy output goes to space heating, the combined efficiency closely resembles the space heating only efficiency. All cases account for standby losses.

Table 10. 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

This analysis is likely a best case estimate for the system. Crucially, it assumes the heating will be via a low temperature radiant floor and requires meticulous heating system design. Further, it requires careful consideration of where the heating loop return water is injected in to the storage tank. If a higher temperature heating system is used, the performance is likely to be 25% worse, or more, as indicated by Table 9.

3.6.2 Optimizing System Design

The research clearly shows the dependence of heat pump efficiency on the temperature of the water to be heated. The heat pump efficiency also depends on outside air temperatures but the water temperature was shown to be just as important. Given the nature of the transcritical CO_2 cycle, the overriding system design principle needs to be to provide the coldest water possible to the heat pump. For traditional water heating applications, this temperature is governed by the ground water and seasonal temperature swings ranging from 40-80°F (Ecotope 2015). 40°F water gives exceptional COPs while 80°F is still acceptable. The high temperature heating tests showed that performance suffers when the water to be heated is at 100°F or warmer. Therefore, the natural fit for space heating is the low temperature delivery needed for radiant floors. While other heat delivery systems like fan coils and radiators likely still provide adequate heat, their efficiency will be much lower.

In designing the space heating portion of the combination system, it is essential to extract as much heat as possible from the water circulating out of, and back to, the tank. Returning the water at 70-80°F, or colder, is ideal. Of course, the load-side return temperature determines how cold this water can be. Obtaining a high temperature drop in the heat-source loop is additionally beneficial in that it decreases the flow rate. A lower flow rate means more time until the tank completely mixes. This will leave the tank stratified longer increasing the availability of hot water for domestic use.

The ideal storage tank for a combination system would have a variable-height dip tube for the heating return water. The dip tube would match the returning water temperature to a height in the tank with the exact same temperature. Doing so would reduce de-stratification, keeping the coldest water at the bottom and the hottest water at the top. Such a device is not realistic but a practical alternative is to have additional return water ports at multiple heights in the tank. The ports should be at approximately 1/3 and 1/2 the height from the bottom.[5] Then, based on the expected, average return water temperature, the system designer can select the placement. If only one is made, the 1/3 height is likely ideal.

Adding an additional return port makes the following scenario possible. In the heating season, fresh inlet water is generally 50°F or colder in the NW and would enter the tank at the bottom. The water being used for space heating would return to the tank at 70°F at the 1/3 height and start to fill up the tank from there. Then, the heat pump unit would turn on maintaining hot water at the top of the tank. Essentially, there are three layers to the tank all serving different, yet related purposes.

Additional improvements could likely be obtained by using a larger storage tank of 120 gallons. The larger tank makes more hot water available so that under heavy space and water heat usage times, there is more available for both needs. Further, the bigger volume makes it less likely to mix the entire volume under a single heating call. Related, the Sanden heat pump controls were designed around water heating needs. Space heating is somewhat different and overall hot water availability could be improved if the heat pump turned on sooner. For an integrated system, the heat pump could monitor the heating use in the house. When there is a call for heating, the heat pump could wait a few minutes and then engage, instead of waiting for half the tank to cool down. This control won't likely reduce heat pump efficiency but would place more hot water at the top of the tank.

These changes, though, while theoretically increasing the efficiency, could also increase the standby losses as well from storing more hot water. It is not clear from this testing the optimized tank volume, although it seems likely that a larger tank and optimized controls could increase the overall system efficiency.

Even with further optimizations, it should be pointed out that high temperature space heating applications, with their corresponding high temperature return water will still have reduced efficiencies relative to their low temperature counterparts. The extent to which efficiency degrades with warmer return water temperature is a feature of the transcritical CO_2 cycle.

[5] Water storage tanks used for solar-thermal heating have a similar port configuration. See http://www.rheem.com/product/solar-water-heaters-universal-solaraide for an example.

4 Conclusions

Due to the necessity of heating primarily cold water with the Sanden heat pump, the combined space and water-heating configuration tested seems most feasible for radiant floor heating systems. Higher-temperature heating systems introduce rapid mixing and de-stratification of the hot water storage tank, which hurts efficiency and hot water availability.

The main concern is that the space heating load can quickly de-stratify the tank, especially with heating systems other than radiant slab. CO_2 transcritical cycle heat pumps work best when applying a large temperature lift to cold inlet water. These types of heat pumps do not work as well for adding small amounts of heat to already warm water, which is the scenario under de-stratification of the storage tank. The domestic hot water demand actually helps significantly by introducing cold water to the bottom of the tank. There do not appear to be any deleterious interactions between the two load types.

For low temperature heating of a radiant floor, return water from the XPB should be piped to the bottom of the tank to preserve stratification. For other heating systems, the optimal return location is not clear, nor is it clear whether the Sanden heat pump combined system is optimal. Adding multiple inlet ports at various heights to accommodate different return water temperatures could provide system-specific optimization of performance, by returning heating loop water to the appropriate height in the tank to best reduce mixing.

With a bottom inlet space heating loop return and radiant floor heating, one can expect an overall COP of 2.2 to 2.3 for combined space and water heating at 35 °F outdoor temperature, given the loads specified in this testing procedure. Combining these estimates with measurements from the water heating only project (Larson 2013) suggests annual space and water heating system efficiencies between 2.2 and 2.7 for a variety of Northwest climates.

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Appendix A: Lab Test Protocol

Configuration

Install heat pump outdoor unit in walk-in thermal chamber.
Install all other components outside thermal chamber in lab space.
Insulate all water lines, both hot and cold.

Components

- Heat Pump Outdoor Unit, Model GAU-A45HPA, (HP-1)
- Storage Tank, 80 gallon, Model GAU-315EQTA, (ST-1)
- Expansion Tank (EXP-1)
- Pump, Taco XPB-DW-1, (XPB-1)
- Instantaneous Electric Heater (EH-2)

Measurement Points

Power Measurements
- Heat Pump, complete outdoor unit, (HP-1)
- Taco X-Pump-Block, (XPB-1)

Heat Pump Measurements (Outdoor Unit)
- Evaporator surface temperatures:
  - Refrigerant Inlet
  - Mid-Point
  - Refrigerant Outlet
- Exhaust air temperature, (T-EXH)
- Cold Inlet Water T, (T-HP_{IN})
- Hot Outlet Water T, (T-HP_{OUT})
- Chamber Temperature and RH (TS-6)

Tank Measurements
- Lab air temperature (TS-1)
  - Position sensor as per typical DOE location relative to tank
- Tank thermocouple temperature tree (13 sensors)
  - Tank stratification is of supreme interest so add more sensors than standard 6
  - Space thermocouples so each represents an equal-volume segment
- Cold Inlet Water (T-FM-2)
- Hot Outlet Water (TS-2)
- Cold Water Inlet Flow (FM-2)

Measurements for/near Taco X-Pump-Block, (XPB-1)
- Heating Supply Water T, (TS-3)
- Heating Return Water T, (T-FM-1)
- Heating Water Flow (FM-1)
- Load Supply Water T, (TS-5)
- Load Return Water T, (T-FM-3)
- Load Water Flow (FM-3)

Measurements near Mixing Valve
- Mixing Valve Outlet Water T, (TS-4)

General Overview

We plan on 3 different styles of measurement: space heating only, combined space and water heating, and mixing valve measurements. Each test style will require a different set of measurements. The combined space and water heating is the most comprehensive; space heating-only tests are conducted with a subset of those components; and the mixing valve measurements are a different, simple configuration.

From the early field site, we have observed we see problems of tank mixing which is potentially leading to lower efficiency and the under-delivery of DHW. This has motivated us to devise a lab setup to directly observe tank mixing and test several plumbing scenarios. We will use the lab tests to optimize system design.

In order to measure the whole system, Ecotope is focusing on running a series of tests at one temperature rather than repeating tests at different temperatures. Sufficient data is available from previous lab and field testing to simulate performance at different temperatures if detailed measurements are taken in the proposed test series.

Space Heating Only Tests

Objectives:

- Measure system efficiency under low and high loads
- Observe tank temperature stratification profile with different return water plumbing configurations
  - Plumbed to PTR port with the diffuser
  - Plumbed to PTR port without diffuser
  - Plumbed to cold inlet at bottom (without diffuser)

The concept for a single test run is to subject the system to a simulated heating load over 18 hours. Over this period, we will monitor all energy use and temperatures. At the end of the period, we will be able to calculate a COP of the test run. The heating load will be simulated and the outdoor unit will run in response to it. The tests will be long enough that we will get to observe all combinations of operating possibilities including heat pump on/off, and XPB on/off.

To simulate the heating load, we will send a call for heating to the XPB. We will have an instantaneous water heater set to a preset temperature to simulate the heating "load." There will be two different heating load profiles, low and high, at 70°F or 100 °F with a 10 or 20 °F temperature rise requirement. The 70 °F "return" temperature and 10 °F temperature rise will approximate a radiant slab heating system while the 100 °F "return" and 20 °F rise will approximate a fan coil heating system. Both systems are used in the field. The length of run time will remain the same for each load condition, but the amount of heat provided to the load differs by requiring a greater temperature rise. The larger temperature rise will require the XPB to flow more hot water from the Sanden tank giving us an opportunity to observe mixing differences.

For all loads, we will set the outdoor unit ambient conditions to 35°F. The 35°F ambient temperature is not so much representative of a winter outdoor temperature as it is useful. The limited testing resources require us to focus on a single dry-bulb condition. That could be 47°F or 50°F but the houses these equipment are being installed in have low heating loads so we don't expect much heating need at 47°F. That leads us to 35°F as a temperature to use which might make the lab test more "realistic". Further, from previous projects, we have existing performance data at 35 °F that we can leverage.

Next, we consider the effect of hot water return plumbing on operation. We will run both low and high loads with each of the return plumbing options. With the thermocouple tree installed inside the tank, we expect to see different stratification and mixing patterns for each of the plumbing patterns and load conditions.

Test Setup

Install all equipment and instrumentation as per the schematic. No hot water draws are to happen during these tests so that equipment does not necessarily need to be operational.

Combined Space & Water Heating Tests

Objectives

- Measure system efficiency under combined hot water with low and high heating loads

These tests build off the space heating only tests. On top of the low and high heating load profiles we will impose a hot water draw profile. The profile will be based on the, field-observed 3-person household draw schedule. These draws will directly remove hot water from the system placing a greater load on the heat pump. They will also change the tank temperature profile. We will measure all energy and temperature points throughout these tests.

The intention is to use only one DHW draw profile – the 3 person schedule. We will run the same DHW draw pattern while we switch up the space heating load between low and high. We expect this to lead to different amounts of tank mixing which we are keen on observing.

Test Setup

These tests require all equipment and instrumentation to be installed including the mixing valve. The hot water return plumbing location will be determined after the space heating only tests have been conducted and we determine which configurations should be examined further. An additional pair of tests could be run if it is desired to test more than one plumbing return option.

Space Heating Tests

Test Name	Ambient Air Conditions					Inlet Water		Load Return		Hot Water Return Plumbing	Load Profile	XPB Settings	Test Time	Notes
	Dry-Bulb		Wet-Bulb											
	F	C	F	C	RH	F	C	F	C					
Low Heat	35	2	33	1	~80%	na	na	~80	~27	With Diffuser	Low Heating	**10F** dT ~3 GPM	1 day	Use stock tank water directly as the "Load Return". Set Taco control target temperature to 10F above incoming stock tank temperature. Send "Load Supply" water to stock tank. See "space_heat_load" profile.
Low Heat no diffuser	35	2	33	1	~80%	na	na	~80	~27	Without Diffuser	Low Heating	**10F** dT ~3 GPM	1 day	
Low Heat return to cold inlet	35	2	33	1	~80%	na	na	~80	~27	Plumbed to Cold Inlet (no diffuser)	Low Heating	**10F** dT ~3 GPM	1 day	
High Heat	35	2	33	1	~80%	na	na	100	38	With Diffuser	High Heating	**10F** dT ~3 GPM	1 day	Use an instantaneous electric heater to increase stock tank water temperature to "Load Return" temperature. Send the "Load Supply" water to stock tank.
High Heat no diffuser	35	2	33	1	~80%	na	na	100	38	Without Diffuser	High Heating	**10F** dT ~3 GPM	1 day	
High Heat return to cold inlet	35	2	33	1	~80%	na	na	100	38	Plumbed to Cold Inlet (no diffuser)	High Heating	**10F** dT ~3 GPM	1 day	

Combined Space and Water Heating Tests

Test Name	Ambient Air Conditions					Inlet Water		Load Return		Hot Water Return Plumbing	Load Profile	XPB Settings	Test Time	Notes
	Dry-Bulb		Wet-Bulb											
	F	C	F	C	RH	F	C	F	C					
Combi Low	35	2	33	1	~80%	50	10	~80	~27	With Diffuser	Combi Low	**10F** dT ~3 GPM	1 day	See combined space heating and DHW demand profile
Combi Low	35	2	33	1	~80%	50	10	~80	~27	Plumbed to Cold Inlet (no diffuser)	Combi Low	**10F** dT ~3 GPM	1 day	See combined space heating and DHW demand profile

Appendix B: Stratification Index Definition

The stratification index was calculated according to the following two-step procedure:

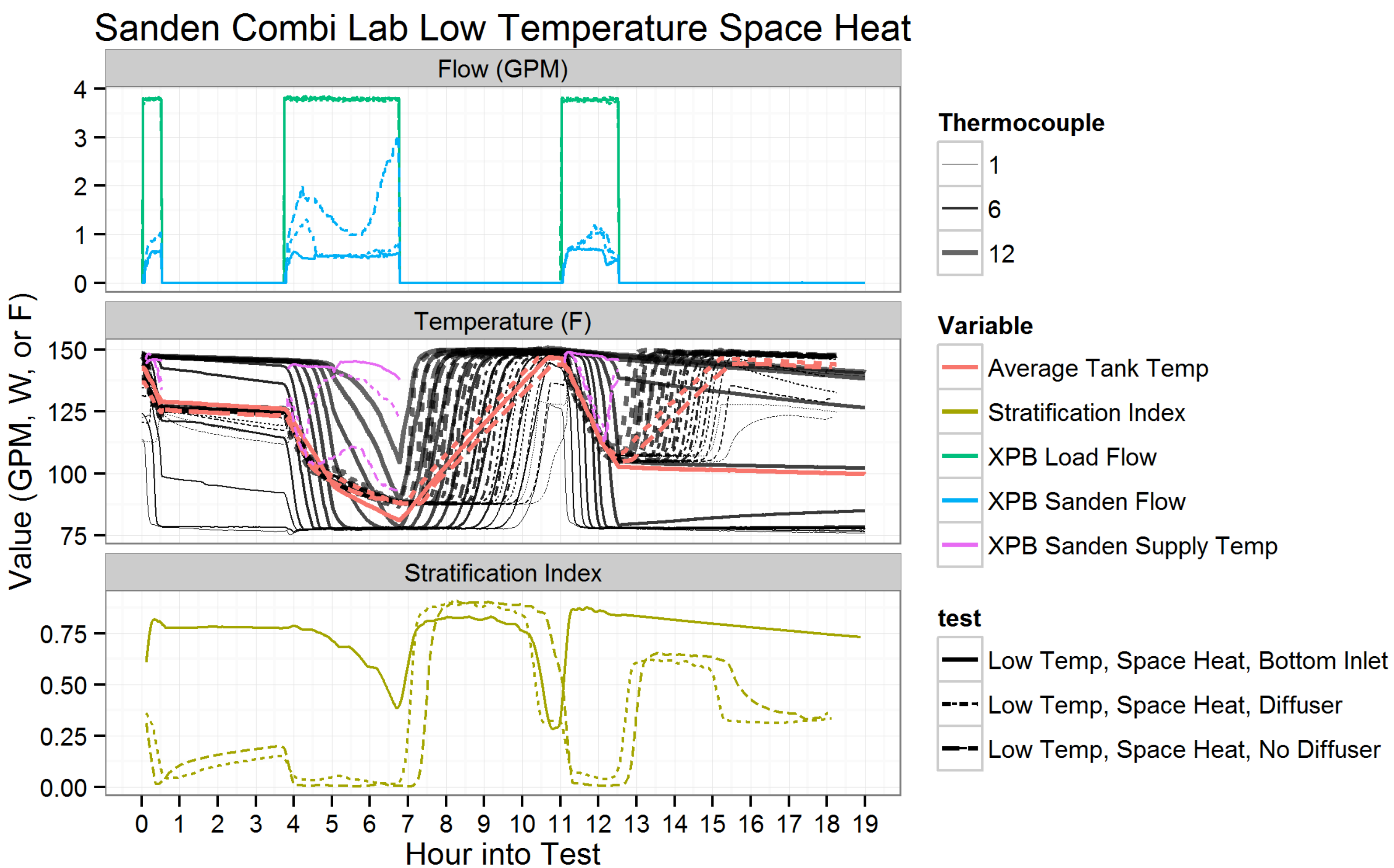
1) Identify the partition of the 12 thermocouple temperatures into 2 groups, minimizing the within-group variation.
2) Calculate the mean pairwise distance between thermocouples in different groups, normalized by the distance between the minimum and maximum temperatures observed in the test.

To understand how this works, suppose that the top 9 thermocouples read 150°F, while the bottom 3 thermocouples read an inlet temperature of 50°F. The partition from step one would be to separate into the 9 top thermocouples and the bottom 3 thermocouples. Because every pairwise distance between the temperatures is equal to the span from inlet to setpoint temperatures, then the stratification index would be identically one.
Now suppose that all thermocouples read essentially the same temperature. The partition from step one will be fairly arbitrary, and all distances between pairs of temperature readings will be essentially zero, leading to a stratification index of zero. This scale then varies continuously between zero and one for all actual and intermediate levels of stratification.

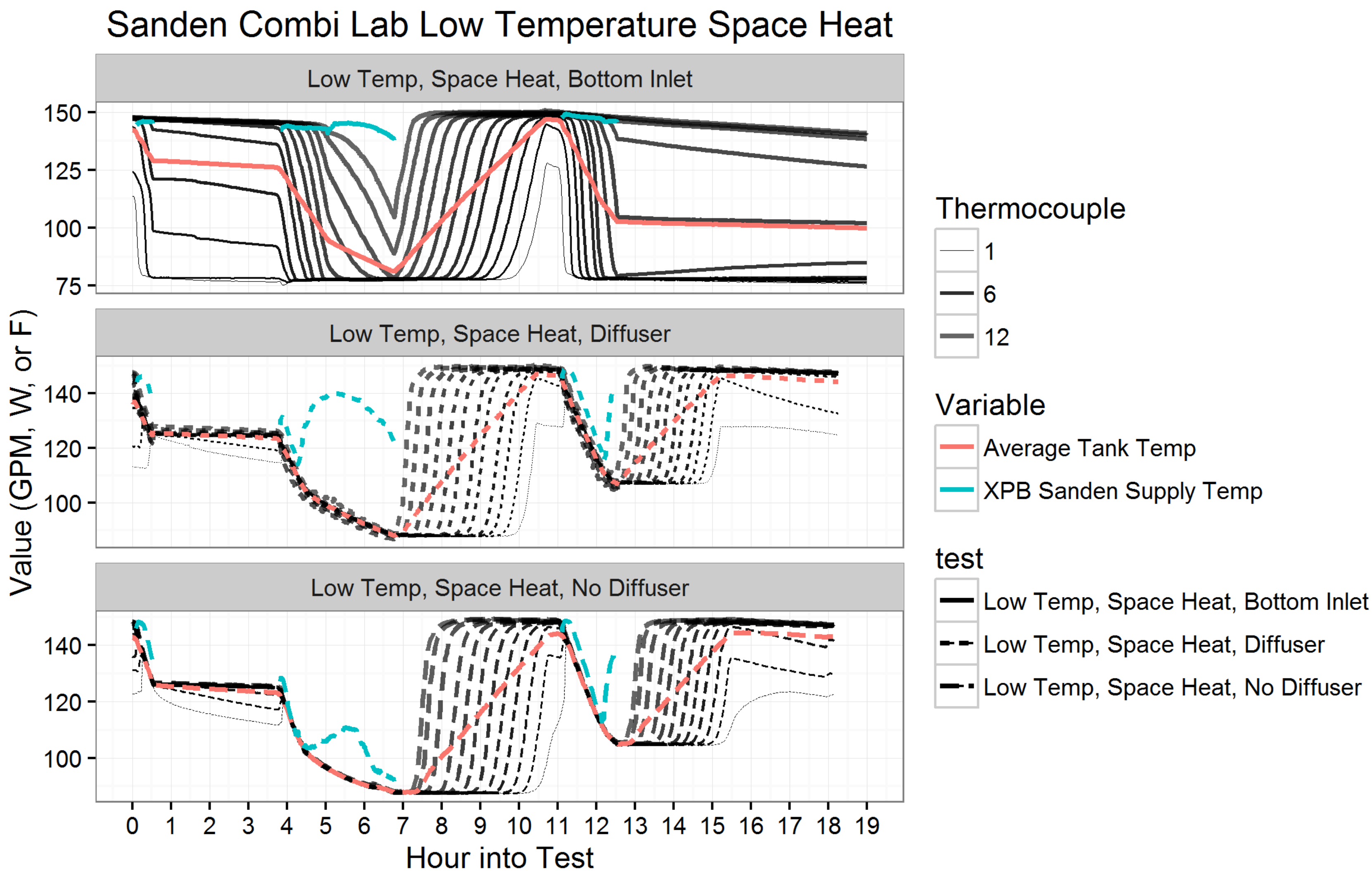
Appendix C: Full Page Graphics

Figures in this section are larger, higher resolution than those in the main body of the report. They replicate, or increase the detail of the main body report figures. For reference, the figure that they replicate is listed above each.

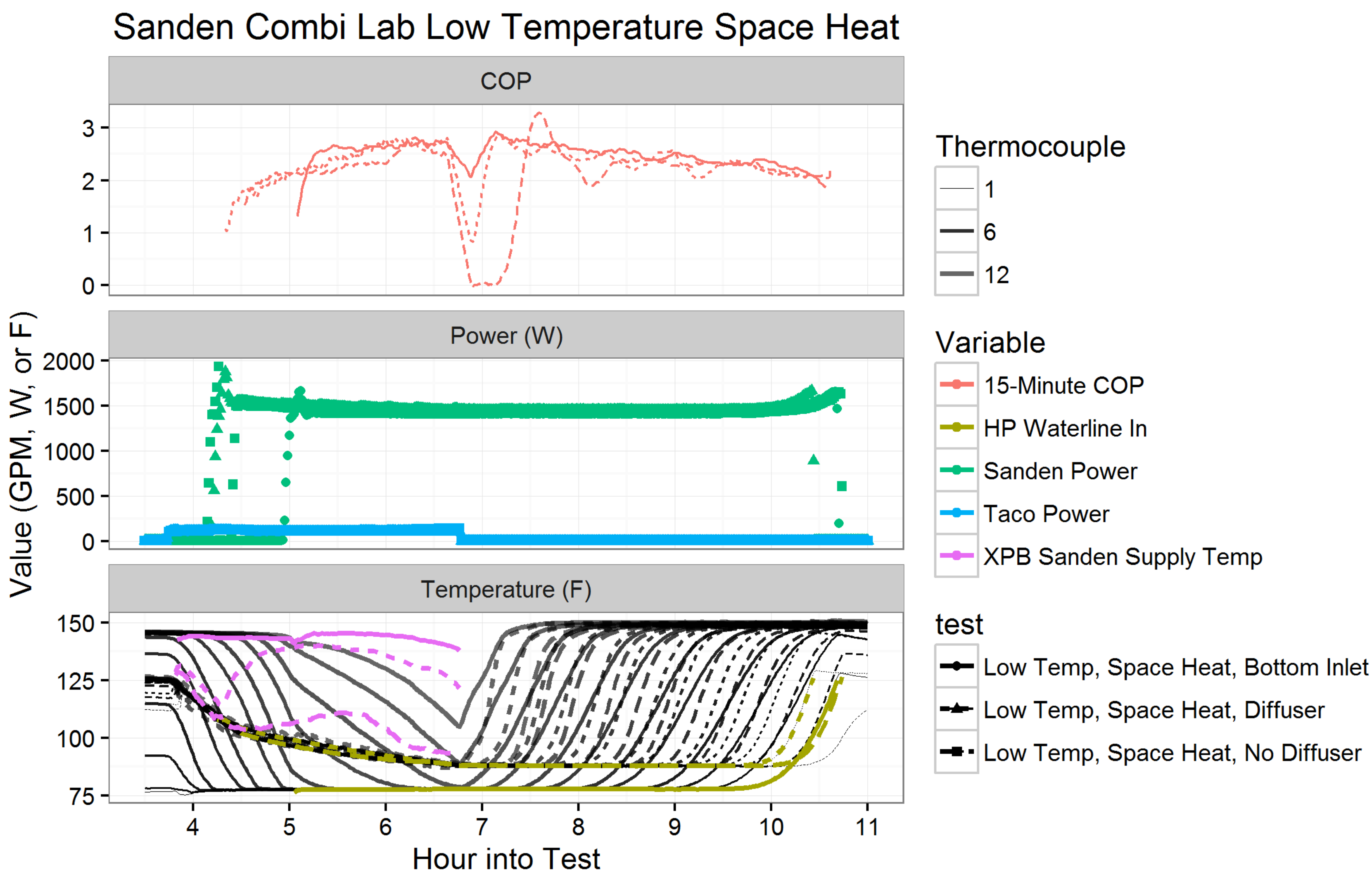
Pairs with Figure 11



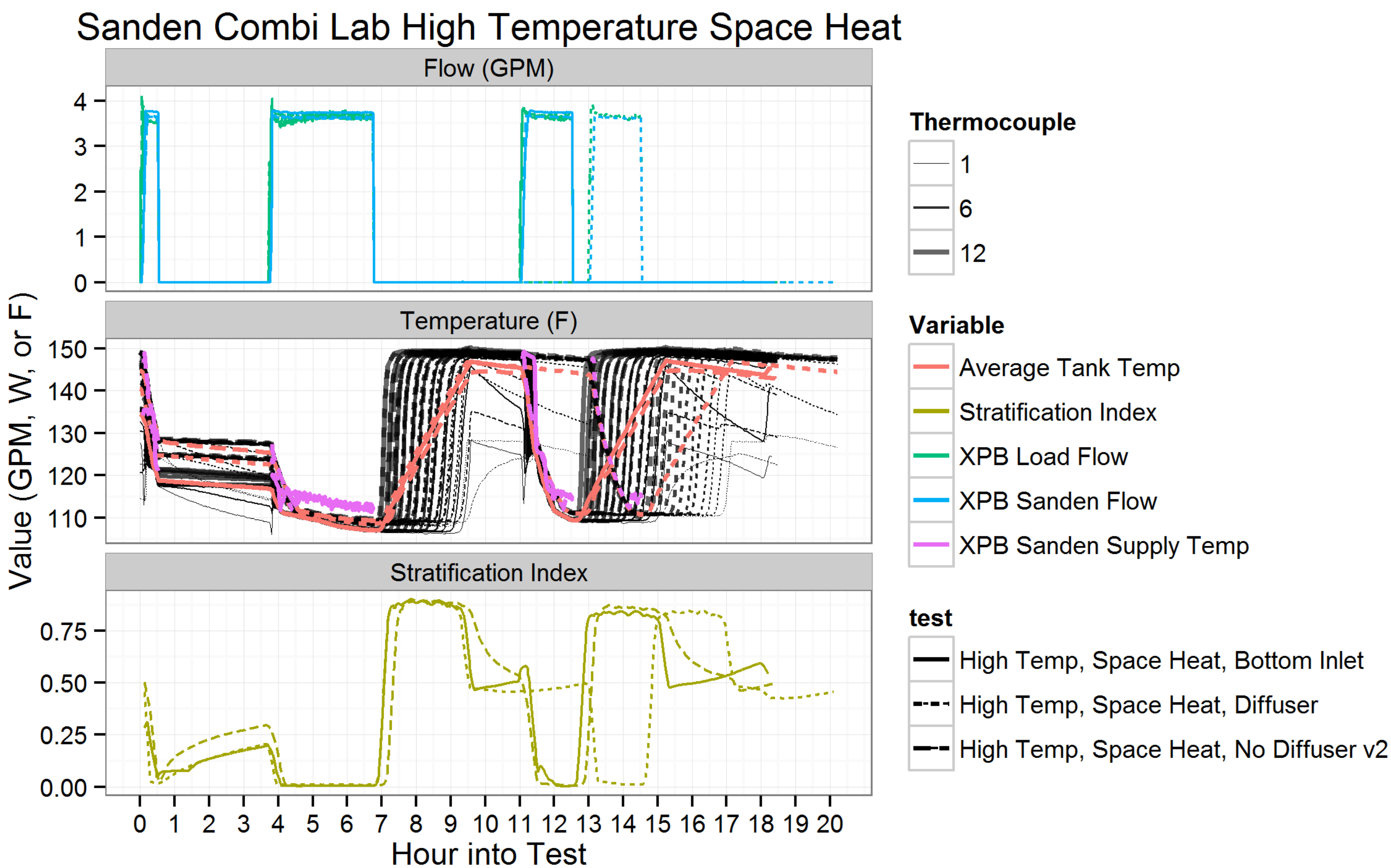
Pairs with Figure 11



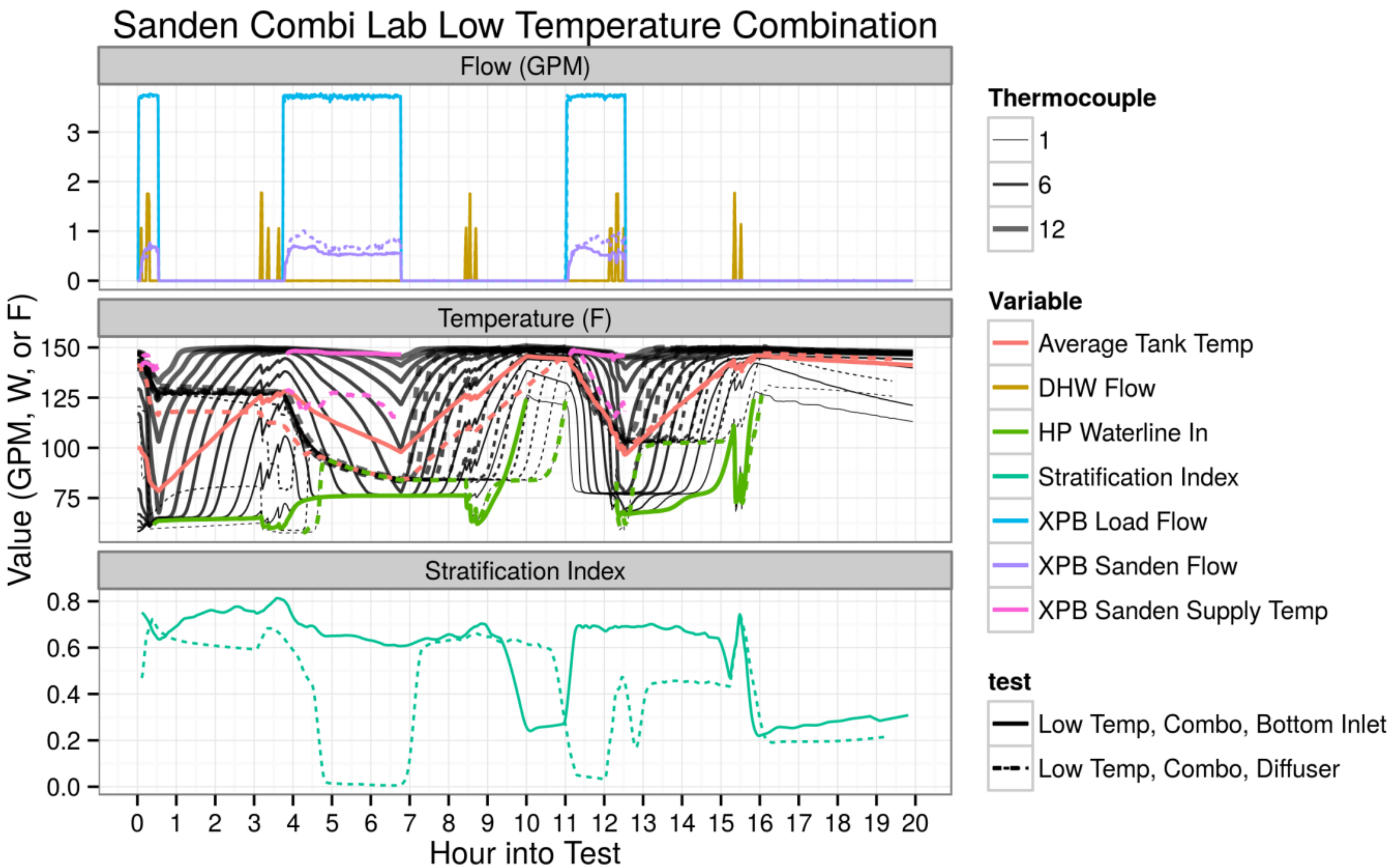
Pairs with Figure 12



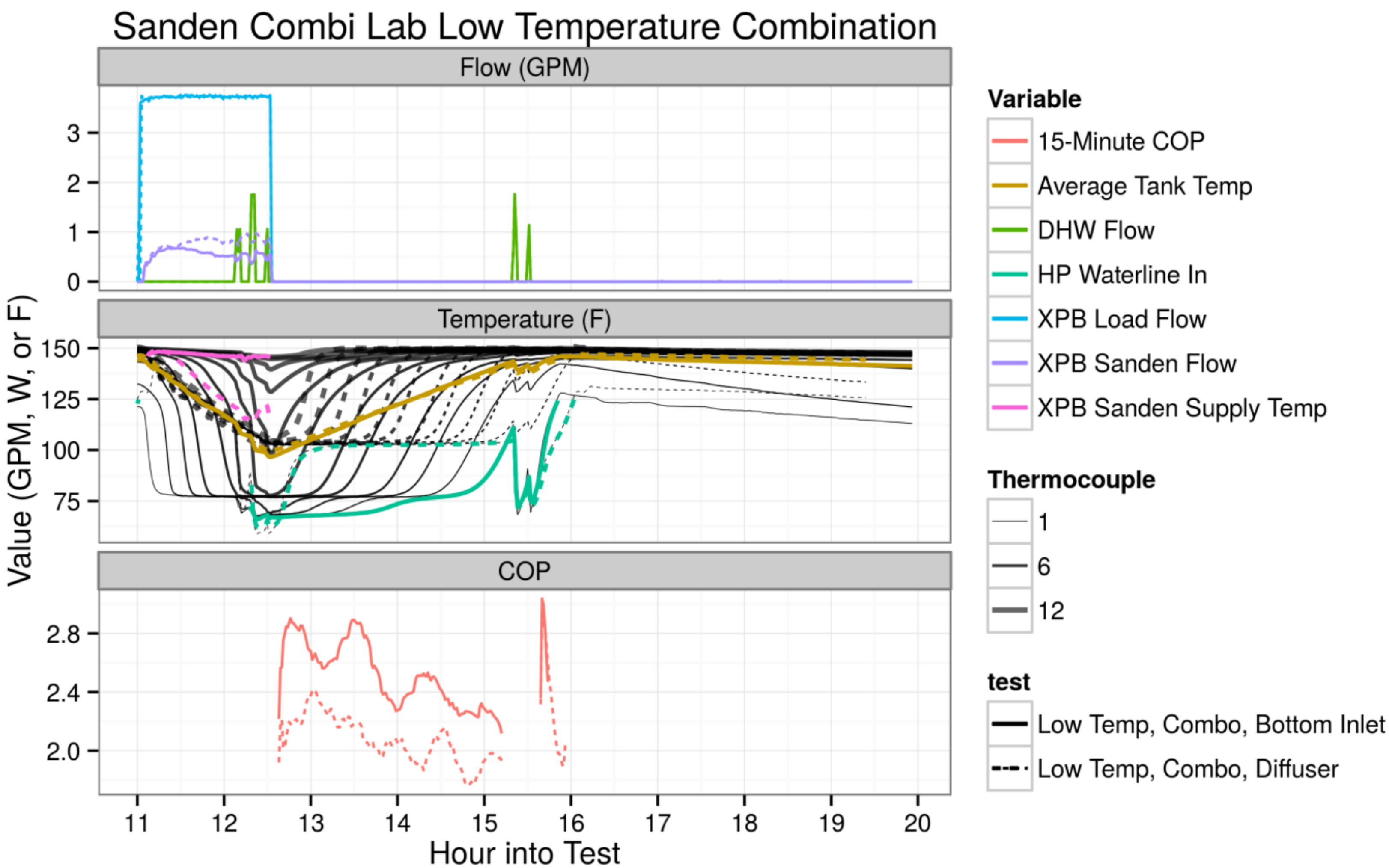
Pairs with Figure 13



Pairs with Figure 15



Pairs with Figure 16



Pairs with Figure 17