

Retrofitting Forced Air Combi Systems

A Cold Climate Field Assessment

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Definitions

AFUE	Annual fuel utilization efficiency
AHU	Air handling unit
BC	Best case (installation)
BEopt™	Building Energy Optimization software
DHW	Domestic hot water
EF	Energy factor
ND	Natural draft
NEAT	National Energy Audit Tool
NOAA	National Oceanic Atmospheric Administration
NQBC	Not quite best case (installation)
OAT	Outdoor air temperature
RTD	Resistance temperature detector
SIR	Savings-to-investment ratio
SWH	Storage water heater
TWH	Tankless water heater
WH	Water heater

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Executive Summary

This project analyzed combined condensing water heaters or boilers and hydronic air coils to provide high efficiency domestic hot water (DHW) and forced air space heating. Called “combi” systems, they provided similar space and water heating performance less expensively than installing two condensing appliances. The system’s installed costs were cheaper than installing a condensing furnace and either a condensing tankless or condensing storage water heater. However, combi costs must mature and be reduced before they are competitive with a condensing furnace and power vented water heater (energy factor of 0.60).

Better insulation and tighter envelopes are reducing space heating loads for new and existing homes. For many homes, decreased space heating loads make it possible for both space and domestic water heating loads to be provided with a single heating plant. These systems can also eliminate safety issues associated with natural draft appliances through the use of one common sealed combustion vent.

The combined space and water heating approach was not a new idea. Past systems have used non-condensing heating plants, which limited their usefulness in climates with high heating loads. Previous laboratory work (Schoenbauer et al. 2012a) showed that proper installation was necessary to achieve condensing with high efficiency appliances. Careful consideration was paid to proper system sizing and minimizing the water temperature returning from the air handling unit to facilitate condensing operation.

Experiment. Twenty sites were selected for a detailed monitoring project to characterize how combi systems perform when installed in real homes. The homes were monitored for 1–2 months to evaluate the performance of the existing systems. A combi system was installed in each home after the completion of the existing system monitoring. At each of the 20 sites a detailed monitoring system was installed, collecting data on energy usage, household load, and system efficiency. This report describes the monitoring protocol, installed combi systems, energy savings and installed performance of combi system. Combi system monitoring was conducted for a full year in order to obtain information on both combined space/DHW operation and DHW-only operation.

Analysis. Monitored field data were used to characterize combination systems. Collected data were analyzed to determine:

- The annual energy consumption of both the existing furnace and water heater and the combi system
- The installed efficiency of the combi systems
- The delivery capabilities for space heating and DHW.

The output measurement data were also analyzed to assess the delivery capabilities of these systems. Capabilities were assessed both in terms of occupant comfort and sizing capacity. Both delivered air and water temperatures were analyzed to assess occupant comfort.

Results/Conclusions. Fifteen sites have been monitored and analyzed for both the existing and combi systems (instrumentation and occupancy issues prevent complete analysis at five sites). Combi systems, on average, saved 19% of natural gas usage for space and water heating. The measured annual combined efficiency was 81%–92% with an average 87%.

System installation was very important to achieve these savings. Careful equipment pairing and operation parameter selection guidelines were required to achieve high efficiencies and good savings. Deviation from these installation parameters would have reduced performance considerably.

1 Problem Statement

1.1 Introduction

Better insulation and tighter envelopes are reducing space heating loads for new and existing homes. For many homes, decreased space heating loads make it possible for both space and domestic water heating loads to be provided with a single heating plant. For these systems, called dual integrated appliances or combination (combi) systems, the use of a direct vent burner can also eliminate safety issues associated with natural draft (ND) appliances, especially in tighter homes (Bohac and Cheple 2002; Bohac 2002).

The project used combi systems consisting of water heater (WH) or boiler heating plants teamed with forced air distribution space heating systems. In each house, a hydronic air handling unit (AHU) that included an AHU, water coil, and water pump to circulate water between the heating plant and coil replaced the existing forced air furnace. Either a WH or a boiler with a separate circuit for domestic hot water (DHW) replaced the existing ND SWH. Various options for DHW priority, DHW tempering, and heating plant temperature set point control were considered.

This project characterized the installed performance of combi systems that used several types of condensing heating plants. Measurement of installed performance of combi systems was the primary objective of this project; a pre/post analysis was also used to estimate energy savings. The monitoring included key system parameters such as water return temperature in order to better understand variations from expected performance and identify improved system designs. The project also tracked the installation costs and evaluated potential cost reductions with improved contractor familiarity with the systems.

1.2 Background

Historically, mechanical contractors have most often custom engineered and pieced together combi systems in the field. They focused on assembling functional systems and often paid little attention to efficiency and optimization. As high efficiency condensing WHs and boilers gain a larger share of the residential market, there is greater potential to use these systems to improve the efficiency of providing both space heating and DHW loads.

Field research and demonstration projects are needed to address several outstanding questions about combi systems. For example, What is the actual installed energy savings of properly configured combi system with a condensing heating plant? What are the installation costs and paybacks of these systems? Can contractor familiarity and experience with combi systems reduce installation prices?

The concept of a single heating plant to supply both space and water heating has been in use for many years. Bohac et al. (1995) installed and monitored combis in small commercial and multifamily buildings in 1989. These systems used an ND SWH as the heating plant. The 1.5 years of monitored operation demonstrated that these systems could be reliably installed and perform without failure while saving energy. The combi systems in this project had annual efficiency ratings of at least 78% and replaced ND WHs with energy factor (EF) rating of about 50% and furnaces with annual fuel utilization efficiencies (AFUEs) around 60%. The study found an average energy savings of 24%.

Twenty years ago combi systems using non-condensing WHs could provide energy savings when they were used to replace older furnaces and WHs. Space heating equipment with 90%+ furnaces have become more common in the past 20 years (Comstock 2013). Laboratory testing (Thomas et al. 2011) demonstrated that when used to replace mechanical equipment in modern homes, combi systems must utilize condensing heating plants to achieve similar or improved energy performance.

Combi systems using high efficiency heating plants are relatively new on the market. Several laboratory test and field installations identified potential problems. Laboratory tests by Brookhaven National Laboratory (Butcher and Arena 2011) showed that the manufacturer-specified plumbing configuration with a primary and secondary loop made it difficult to achieve the high efficiency potential of condensing combi boilers. A field installation by the New York State Energy Research and Development Authority (Rudd 2012) examined durability issues for systems using tankless water heaters (TWHs). The study assessed problems with hard water, scaling, and short cycling. The combi system installed in that project had buildup on the inlet water filter on a TWH. This buildup eventually prevented the TWH from activating. The study installed an industrial strainer on the inlet water line, which prevented WH failure and reduced the maintenance interval to an annual filtering.

In 2009, the Consortium for Advanced Residential Buildings, analyzed the performance of six condensing boilers in existing homes (Steven Winter Associates 2011). The analysis discovered that the high efficiency boilers were operating well below their rated efficiencies. In space heating mode the frequency of condensing (as a fraction of total runtime) for the boilers was 14%–69%. In domestic water heating mode the frequency of condensing operation was 18%–65%. Measuring return water temperatures in the system and relating them to a criterion for condensing determined in the laboratory allowed for analysis of the condensing of each system. Three major factors contributed to the lack of condensing at these six sites: set point temperature higher than 180°F for all sites, high secondary or distribution system flow rates produced low temperature drop across the radiation system, and primary/secondary plumbing loop configuration increased return water temperatures.

This large sample field study was designed to help determine how these systems work in the real world, as well as assess the actual installed efficiency and performance of combi systems. These field tests used commercially available, high efficiency products laboratory tested and configured to achieve condensing operation (Schoenbauer et al. 2012a) to determine the actual energy savings of well-designed combi systems.

1.3 Relevance to Building America's Goals

Combi systems have the potential to significantly reduce home energy use. A properly installed combi system can provide both space and water heating with 90+% efficiency compared to a minimum efficiency 78% AFUE furnace and a 0.59¹ EF WH. Additionally, replacing an ND SWH with a direct vent heating plant allows the home to be more airtight without causing combustion safety issues and eliminates combustion makeup air. These two measures can further improve the energy performance of a home.

¹ For gas-fired water heater with a rated storage volume of 40 gal.

Combi systems are feasible for typical houses in most climate regions. In colder climates the application may not be possible in larger homes with poorly insulated and leaky envelopes. Some currently available equipment can meet space heating design loads up to 60,000 Btu/h while still operating in the condensing mode. Higher capacity hydronic coils would allow current heating plans to meet even higher space heating loads.

The implementation portion of this project installed more than 200 combi systems in Minnesota homes. It was expected that by the completion of this project contractors would be ready to install the systems across the state. In addition, this project is developing installation guidelines and specifications to increase the success of installations in all climates.

1.4 Cost Effectiveness

The installation cost of a high efficiency combi system may be lower than the cost of a similar efficiency separate furnace and WH. In a retrofit application, a homeowner can expect to pay approximately \$4,250 for a high efficiency (90%–98% AFUE) furnace and \$5,300 for a high efficiency (0.80–0.95 EF) storage water heater (SWH).² As the number of high efficiency WH installations increased over the past few years, the installed cost dropped dramatically. In the Minneapolis area, contractors with limited experience with combi systems currently bid a high efficiency system for \$8,200, on average with a typical range from \$6,500 to \$10,000.

A preliminary EnergyPlus analysis of a high efficiency boiler used for space and domestic water heating estimated 12% natural gas and 7% source energy savings compared to an 80% AFUE furnace and 0.55 EF WH. However, EnergyPlus is not easily adapted and may not properly model high efficiency combi system performance. For example, the heating plant steady-state efficiency varies with return water or inlet water temperature and the form of that relationship varies for different plants. In addition, the boiler used for the EnergyPlus model was found to have high off-cycle losses and was not included in the field installations. It is expected that the recently available test laboratory measurements for individual combi components will provide performance data necessary for improved EnergyPlus models. The laboratory results should also help verify the accuracy of the predicted EnergyPlus combined space and domestic water heating efficiency. Project results will be processed to generate data that National Renewable Energy Laboratory staff can use in the development of accurate EnergyPlus combi system models, with the ultimate goal of incorporating combi system models into Building Energy Optimization (BEopt™) software.

1.5 Tradeoffs and Other Benefits

There are several secondary benefits to the combi system. The system replaces a separate furnace and WH with a single boiler or WH. This reduced the number of gas lines and exhaust vents and can reduce the equipment footprint. A combi system using an SWH will be the same size as a forced air furnace and traditional WH. A system using a TWH or hybrid WH can be wall mounted and reduce the footprint. These small types of WHs can allow for a more versatile mechanical room. In new construction, this may allow the mechanical equipment to be placed closer to the end uses, reducing delivery losses and hot water wait times.

² Cost estimates come from the standard bid amounts from the project contractors.

A single high efficiency burner also has combustion safety and venting benefits. The high efficiency combi heating plants have forced mechanical draft venting systems. These eliminate combustion safety depressurization issues that pose a significant problem as homes become tighter. Direct vent burners may provide an additional energy benefit by eliminating combustion makeup air openings and may allow an existing chimney common vent to be sealed.

For the implementation portion of this project the selected combi system components include a warranty and their durability is expected to equal or exceed that of the alternatives. In addition, the single heating plant for combi systems is expected to require less annual maintenance than a separate furnace and WH.

As part of the laboratory portion of the project, properly installed combi systems were demonstrated to code officials and contractors expected to bid on the installations. The demonstrations allowed code officials to identify potential code concerns, such as water stagnation in potable rated AHUs, and recommend acceptable solutions. This improved code-official acceptance of this newer technology and reduced issues requiring attention during the installation process.

2 Experiment

2.1 Research Questions

The field research was designed to address the following research questions:

- What are the minimum performance criteria, installation specifications, and quality control methods that must be included in the install to ensure proper performance and expected efficiency? What trouble areas of the installation can be addressed through these criteria and quality control methods?
- What is the installed performance of combi systems and what is the savings potential of this technology in real homes? How do monitored performance and savings results compare to modeled results?
- Are there heat delivery advantages and/or disadvantages of different combi systems? What are system response times? How consistent are delivered temperatures? Etc.
- What were the initial installed costs for combi systems and does increased contractor training and installation experience reduce these costs?
- How can these systems benefit from improved combi products (boilers, WHs, hydronic AHUs, controls, etc)?

2.2 Technical Approach

This project examined energy savings achieved by using a single high efficiency appliance to provide both space and water heating. The Sustainable Resource Center in Minneapolis, Minnesota received a Sustainable Energy Resources for Consumers grant to install approximately 400 combined space and water heating systems in homes participating in the State of Minnesota Low-Income Weatherization Assistance Program. The limited contractor awareness and agency experience with the systems led to extensive laboratory tests to generate detailed design and installation specifications. The grant required that each home have participated in the low income weatherization program. Older mechanical systems are frequently replaced in the weatherization process and this project did not want to remove a new condensing furnace. That meant that only homes that had gone through weatherization and did not have the furnace replaced could qualify for this installation. The added time required for system design and the increased difficulty of finding homes reduced the total number of installations to just over 200. Building America funds were used to install more extensive field monitoring to measure and analyze the system's performance for a subset of the homes. The original goal was to monitor 20 of the installations, but one site dropped from the study after the pre-monitoring when the owner did not proceed with the combi installation. In addition, a pre/post analysis of daily average space and domestic water heating energy use was included for 18 homes. Pre-monitoring was not performed at the final site because there was not enough heating season remaining to accurately characterize the existing system.

Laboratory tests were conducted on a variety of complete systems and individual components before combi systems were installed in the field. The results from the laboratory work were reported in the Building America report titled *Retrofitting Integrated Space and Water Heating Systems: Laboratory Tests* (Schoenbauer et al. 2012b). The laboratory analysis was used to develop a set of specific parameters for several categories of home space and DHW design loads

so that each system would achieve condensing operation during the space heating mode, provide acceptable supply air temperatures, and produce space heat sufficient to meet or exceed design conditions.

At 19 of the 200 sites, data on the natural gas and electricity use of the existing furnace and WH were collected for at least 1 month prior to the installation of the combi system. Combi installations were conducted between January and March 2012, allowing 2011/2012 heating season data to be collected on both systems. Models of daily average energy use with local National Oceanic Atmospheric Administration (NOAA) weather station outside temperature and seasonal variations in inlet water temperature were used to characterize the daily energy use of the existing systems. These models were used to compute the annual gas use of the existing systems. A similar analysis was conducted for the installed combi systems to estimate the combi system energy savings.

Instruments were added to each combi system to measure the space and DHW energy output at 1-second intervals. The energy output and input data were used to compute short- and long-term system efficiencies and disaggregate the space and water heating loads. Measured efficiencies were compared to rated efficiencies. The field data collected will be made available to National Renewable Energy Laboratory staff to assist them in developing more representative EnergyPlus models, with the ultimate goal of including a high efficiency combi system option in BEopt.

Water flow rates and water supply and return temperatures were recorded for both the space and DHW loads along with AHU return/supply air temperature and air temperature at the thermostat. The measurements were made at 1-second intervals. Portions of the short interval data were used to help identify system installation issues, quality control measures, and design changes to improve the performance of installed systems.

Quality control procedures established for previous projects were used to calibrate instruments before field deployment, verify operation at the time of installation, and confirm operation on an ongoing basis after installation. Site data were collected and reviewed 1 week after the initial installation and the collection process continued at 2- to 4-week intervals through the end of the monitoring period.

2.3 Monitoring Equipment

The energy performance and operating characteristics of the existing and combi systems at the 19 test sites were monitored using a Campbell Scientific model CR-3000 data logger. The natural gas and electricity use of the existing furnace and WH were computed from burner runtime measurements. WH burner runtimes were monitored using either gas valve pressure switches or thermocouple sensors near the burner; both strategies were tested and verified under simultaneous operation at multiple sites. Furnace burner on times were monitored using current switches on the burner signal. Air temperature data near the thermostat, furnace return air temperature, and supply air temperature were also monitored. The outdoor air temperature (OAT) data were obtained from a local NOAA weather station. Figure 1 shows the instrumentation configuration for monitoring the existing equipment. All existing SWHs and air furnaces had single-stage, constant input burners. This type of equipment enabled the use of natural gas valve runtime monitoring and measured gas input rate to determine the appliance gas use. Electricity use was computed from runtime monitoring and one-time measurements of

circulation fan and off-cycle power. Surface mount thermocouples were used to track WH inlet and outlet temperatures, but were not used for the energy analysis.

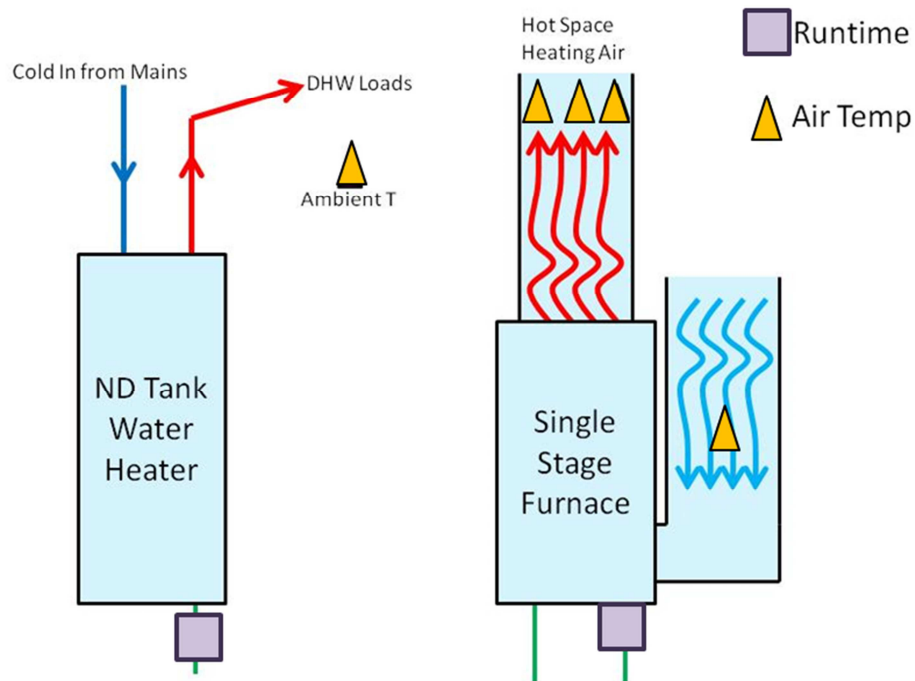


Figure 1. Monitoring instrumentation for existing space and DHW systems

At the time of the combi system installation a comprehensive and accurate monitoring system utilizing high precision instruments (see Table 1) was installed to monitor space and DHW heating energy input and output along with several key operating characteristics of the systems. Water temperature and flow rate measurements were used to calculate the hot water energy output by the heating plant for the DHW and space heating loads. The two energy outputs were summed to obtain the total output. The gas use was measured with a diaphragm meter and the electricity use by a Watt transducer. The instrument configurations for storage tank WHs, TWHs, and boilers are shown in Figure 2, Figure 3, and Figure 4 respectively. The purple triangles indicate the location of the immersion RTDs (resistance temperature detectors) used for energy output calculations. In addition, AHU supply air temperatures and conditioned space temperatures were used to help identify customer comfort issues and diagnose complaints.

Table 1. Instrumentation Specifications

Measurement	Sensor Type	Resolution ^a	Precision ^a	Range
Water Volume Flow Rate	Nutating disk flow meter	198.4 pulses/gal	2% of reading	0.5–25 gpm ^b
Natural Gas Volume	Diaphragm meter with pulse output	40 pulses/ft ³	0.3% of reading	0–250 cfm
Water Temperatures	Matched pair of immersion RTDs	0.002°F @ 140°F	1/10 DIN: 0.03°F @ 32°F	–148° to 752°F
Electric Energy	Watt transducer	0.02 Watts	0.2% of reading	0–1000 Watts
Air Temperature	Thermocouple array	0.003°F @ 140°F	Greater of 1.8°F or 0.75% of reading	–454° to 725°F

^a Resolution is used to measure the granulation of instrument measurement and precision is used to indicate the ability of the instrument to measure the actual value.

^b The meter measures flow rates < 0.5 gpm, but the precision decreases for flow rates outside the specified range.

For the DHW load calculation the water flow meter and inlet temperature immersion RTD were placed before the tee to the mixing valve and the RTD for the supply temperature was placed at the mixing valve outlet. This provided an accurate measurement of the energy output, as well as the volume and temperature of hot water provided to the house. An immersion thermocouple was placed near the heating plant DHW outlet (before the mixing valve) to measure the transient performance of the heating plant (see yellow triangles in Figure 2 through Figure 4).

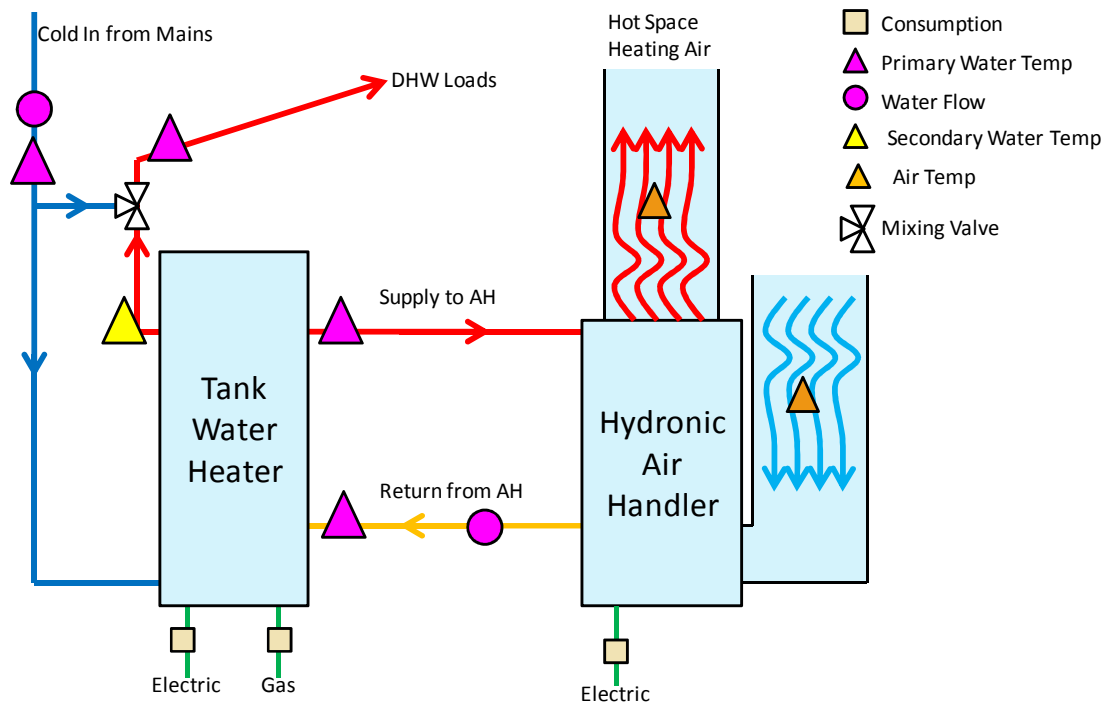


Figure 2. Field monitoring instrumentation for an SWH system

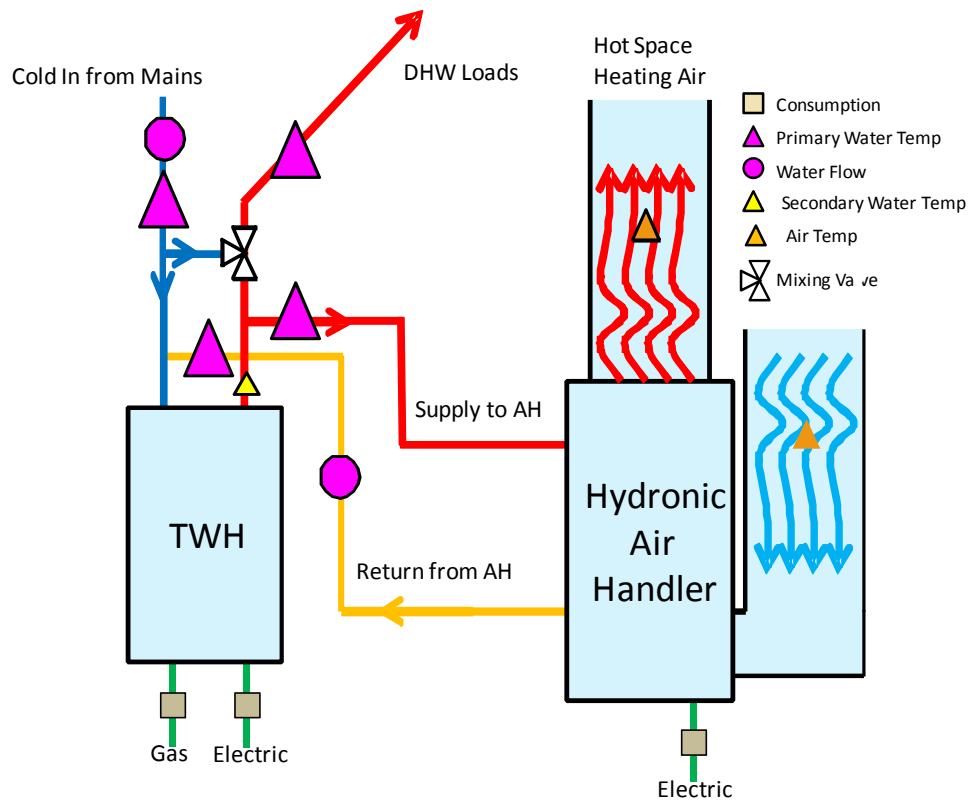


Figure 3. Field instrumentation for a TWH combi system

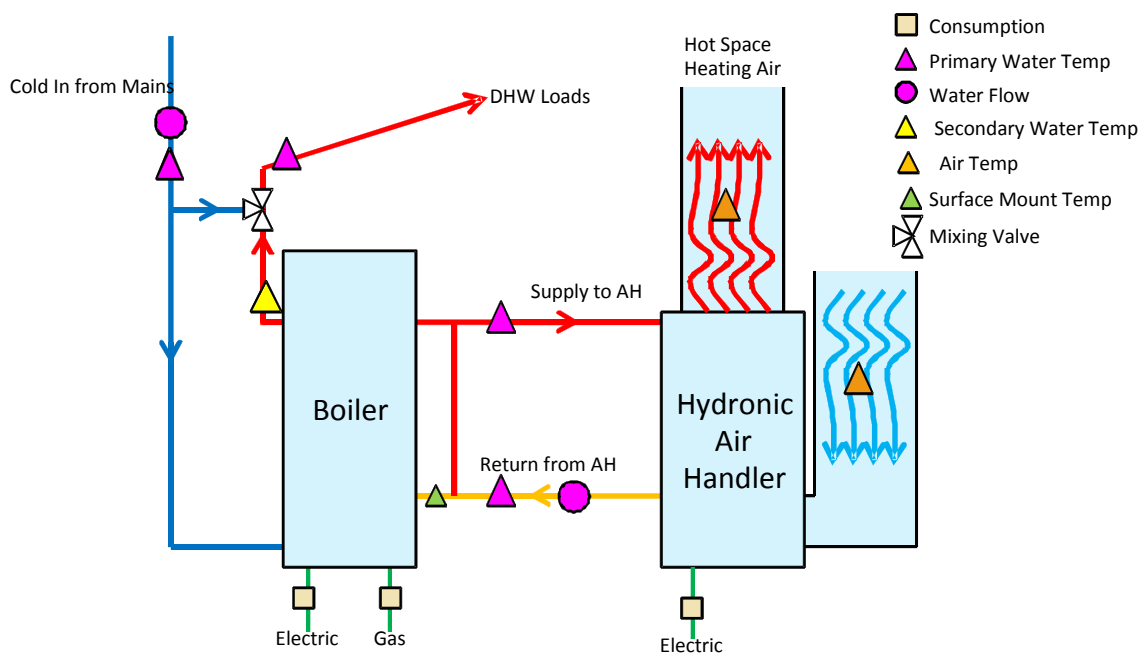


Figure 4. Field instrumentation for boiler-based combi systems

2.4 Systems Monitored

The existing space and water heating equipment was measured in 18 homes. Each home had an ND SWH and a forced air furnace. Sixteen of the sites had ND furnaces with AFUE ratings around 80% and the other two sites had condensing furnaces with efficiencies listed around 90%. Table 2 lists the characteristics of the existing equipment at each site.

Table 2. Characteristics of the Existing WH and Furnace at Each Site

Site Number	Existing WH					Existing Furnace			
	Install Year	Input (Btu/h)	Storage (gal)	EF	Vent Type	Install Year	Input (Btu/h)	AFUE	Vent Type
1026	2009	40,000	40	0.59	ND	2001	88,000	80%	ND
1027	2006	38,000	40	0.59*	ND	1999	90,000	80%	ND
1028	1999	40,000	40	0.56	ND	1979	–	80%	ND
1030	2004	36,000	40	0.59	ND	2004	66,000	80%	ND
1031	2001	40,000	40	0.56	ND	2007	60,000	93%	Direct Vent
1037	2009	38,000	50	0.58	ND	–	75,000	80%	ND
1045	–	40,000	40	0.59	ND	1999	80,000	80%	ND
1047	1982	32,000	40	0.56*	ND	2003	75,000	80%	ND
1049	2004	40,000	40	0.59	ND	2004	88,000	80%	ND
1052	2008	38,000	50	0.58	ND	–	80,000	90%	Direct Vent
1054	2006	34,000	40	0.59	ND	2003	88,000	80%	ND
1055	1991	32,000	40	0.56	ND	1991	75,000	78%	ND
1056	2006	40,000	40	0.59	ND	1997	75,000	80%*	ND
1061	2004	40,000	40	0.59	ND	2004	88,000	80%	ND
1062	2005	40,000	40	0.59	ND	1998	60,000	80%*	ND
1063	2001	40,000	40	0.59	ND	1989	95,000	80%*	ND
1065	2002	40,000	40	0.59*	ND	1989	71,000	80%	ND
1070	2003	40,000	40	0.59*	ND	2007	75,000	80%	ND
1078	N/A					N/A			

* Efficiency ratings could not be determined and were estimated based on vintage and other model characteristics

Table 3 shows the manufacturer and model information for the combi system heating plants and hydronic AHUs used for the field installations. The project tested a standard model of both the American Polaris (PG10 34-130) and HTP Phoenix (ph100-55) in two homes, with higher capacity models (PG10 50-150 and ph160-50) for homes with larger loads. The third storage model, the AO Smith Vertex, was installed in only two homes. This WH does not have a larger model suitable for higher load homes. Two combination boilers (Navien Combi Boiler and Rinnai 37AHB075) and two TWHs (Rinnai 98Lsi and Grand Hall Eternal) were installed in three homes.

Table 3. Field-Installed Combi System Equipment

Equipment Type	Manufacturer	Model	Rating		
			AFUE	Thermal Efficiency	EF
SWH	AO Smith	Vertex	–	0.96	–
SWH	American	Polaris	–	0.95	–
SWH	HTP	Phoenix	–	0.95	–
TWH	Rinnai	98LSI	–	–	0.95
Hybrid WH	Grand Hall	Eternal	–	–	–
Boiler	Navien	Combi Boiler	91%	–	–
Boiler	Rinnai	Q175C	96%	–	–
Hydronic AHU	Rinnai	37AHB75	–	–	–
Hydronic AHU	Ennerzone	7500%	–	–	–

Sites were selected from homes participating in the State of Minnesota low income weatherization program. These sites had received building envelope air sealing and insulation upgrades, but no mechanical system updates. Homes with an estimated design space heating load less than 50,000 Btu/h and one or two showers were chosen for the study. The design space heating loads were determined from National Energy Audit Tool (NEAT) modeling used to select cost-effective energy improvements. Table 4 shows the combi heating plant and hydronic AHU specified for each site along with the site load information. Homes with lower heating loads (20,000 to 37,000 Btu/h) and a single shower were considered to be a “best case” (BC) installation. Homes with a larger heating load (37,000 to 50,000 Btu/h) and a single shower were considered to be “not quite best case” (NQBC) installations. Homes with two showers were considered “stress case” (STRESS) installations. The load category specification indicates the expected level of difficulty for the heating plant to satisfy the home’s combined space heating and DHW loads.

Table 4. Combi Equipment and Loads by Site

Site Number	Combi Appliance		Calc Load (Btu/h)	Number of Showers	Number of People	Load Category
	Heating Plant	AHU				
1026	Vertex	Ennerzone	46,248	1	3	NQBC
1027	Polaris	Ennerzone	38,774	2	4	STRESS
1028	Polaris	Rinnai	35,716	2	1	BC
1030	Rinnai TWH	Ennerzone	32,064	2	5	STRESS
1031	Polaris	Ennerzone	38,260	2	1	NQBC
1037	Navien	Ennerzone	28,534	2	3	STRESS
1045	Rinnai TWH	Ennerzone	31,461	1	3	BC
1047	Navien	Ennerzone	42,636	1	2	NQBC
1049	Eternal	Ennerzone	36,277	2	5	STRESS
1052	Rinnai Boiler	Ennerzone	33,474	2	1	BC
1054	Eternal	Ennerzone	40,101	2	1	NQBC
1055	Navien	Rinnai	30,729	1	2	BC
1056	Phoenix	Ennerzone	26,112	1	1	BC
1061	Phoenix	Ennerzone	37,368	1	2	NQBC
1062	Rinnai TWH	Ennerzone	41,966	2	1	NQBC
1063	Rinnai Boiler	Ennerzone	32,640	2	2	STRESS
1065	Vertex	Ennerzone	35,136	1	1	BC
1070	Eternal	Ennerzone	29,111	1	1	BC
1078	Phoenix	Ennerzone	40,320	2	5	STRESS

Large load installations required some larger model heating plants.
These cases are noted as stress models.

2.5 System Sizing and Proper Installation

Previous laboratory test results were used to develop guidelines for component selection, sizing requirements, and system installation. These results were presented in the 2012 Building America report titled *Retrofitting Combined Space and Water Heating Systems: Laboratory Tests* (Schoenbauer et al. 2012b). The first step in the combi design process is to measure or determine the key household characteristics: space and water heating loads, occupancy, distribution systems, and hot water end use.

The space heating loads are estimated from the building envelope characteristics and used to calculate the space heating design load. The home's gas utility billing history and corresponding outdoor temperatures can help verify the space heating load estimates. The water heating load can be estimated with a survey of end uses and number of residents. For sizing purposes, the large hot water draws are more important than the total DHW load. The shower events are typically the largest loads and have the greatest impact on occupant satisfaction with DHW delivery (Schoenbauer et al. 2012a). The actual shower hot water flow rates provide more reliable sizing estimates, but person-to-person variances of flow, cold water temperature into the home, and shower temperature selection necessitate the use of typical values. Low flow shower heads were recommended to replace shower heads with flow rates greater than 2.0 gpm for the field portion of this project to avoid excessive shower flow rates that could cause occupant dissatisfaction.

Codes handle sizing for WHs and space heating systems differently and manufacturers often use their own proprietary software. Neither codes nor manufacturers adequately address combined appliance sizing. Contractors have traditionally sized systems using their own rules of thumb or guidance from local manufacturer representatives. For this project the heating plants were sized for each home using the expected shower load and estimated space heating load. The natural gas burner input rate, storage capacity, and heating plant controls (DHW priority) were considered when sizing the system. Table 5 lists sizing guidelines for the three types of combi systems studied in this project. The sizing guidelines were developed by assuming an output capacity for each system based on the unit specifications. A conservative thermal efficiency of 85% was used to determine the energy output capacity for the burner size. The output capacity of the hot water storage volume was considered to be the energy available from a 30°F temperature drop in the storage volume over 20 minutes. These assumptions were similar to residential usage, around 2 gpm and the 20-minute time interval was similar to shower lengths. Inlet water temperatures were assumed to be 60°F and the mixed shower temperature was assumed to be 105°F. Additionally, the energy rates that resulted from these calculations were compared to rates measured in the laboratory during hot water events prior to the burner firing. The total output capacity of the unit was the combined output from storage and the burner. This output was compared to the combined DHW and space heating load, unless the heating plant used DHW priority. In that case the output capacity was compared to each load independently. Each row corresponds to a heating plant that was considered for use in the field monitoring portion of the project. The first four columns of the table (type of heating plant, burner maximum input, water storage capacity, and DHW priority) characterized the combi system. The next six columns display the system output capacity as a percentage of load for the six load profiles. These columns indicate whether the system would be properly sized for the specified loads. For example, a boiler with 199 kBtu/h input, zero storage, and DHW priority would have more than 2 times the capacity needed in a 40 kBtu/h, one shower home. A heating plant would be undersized if the number listed was less than 100%, meaning the required output was greater than the available capacity. The loads in Table 5 were selected to represent the range of the housing stock considered. When sizing was done for a specific home, the estimated loads of that home were used.

Table 5. Combination System Sizing Chart

	Max Input (kBtu/h)	Storage (gal)	DHW Priority?	1 Shw	2 Shw	1 Shw	2 Shw	1 Shw	2 Shw
				SP: 40,000 Btu/h (%)		SP: 50,000 Btu/h (%)		SP: 60,000 Btu/h (%)	
Boiler-Based Combination Systems	75	0	Yes	92	53	92	53	92	53
	100	0	Yes	122	71	122	71	122	71
	150	0	Yes	183	107	183	107	183	107
	199	0	Yes	243	142	243	142	243	142
	75	6	Yes	97	57	97	57	97	57
	100	6	Yes	128	74	128	74	128	74
	150	6	Yes	189	110	189	110	189	110
	199	6	Yes	248	145	248	145	248	145
	75	12	Yes	102	60	102	60	102	60
	100	12	Yes	133	78	133	78	133	78
	150	12	Yes	194	113	194	113	194	113
	199	12	Yes	254	148	254	148	254	148
TWH-Based Combination Systems	150	2	No	117	85	108	76	99	72
	199	2	No	155	113	142	101	131	95
	150	0	No	116	84	107	75	98	71
	199	0	No	154	112	141	100	130	94
	150	2	Yes	185	108	185	108	185	108
	199	2	Yes	245	143	245	143	245	143
	150	0	Yes	183	107	183	107	183	107
	199	0	Yes	243	142	243	142	243	142
SWH-Based Combination Systems	100	50	No	106	77	98	69	90	65
	150	50	No	145	105	133	94	123	89
	199	50	No	183	133	168	119	155	112
	100	80	No	124	90	113	80	105	76
	150	80	No	163	118	149	105	137	99
	199	80	No	201	145	184	130	170	123
	100	50	Yes	106	77	98	69	90	65
	150	50	Yes	145	105	133	94	123	89
	199	50	Yes	183	133	168	119	155	112
	100	80	Yes	124	90	113	80	105	76
	150	80	Yes	163	118	149	105	137	99
	199	80	Yes	201	145	184	130	170	123

Note: The SP value refers to the space heating load and 2 shw refers to 2 simultaneous showers. The percentage in each column specifies the ratio of system capacity to household load. Values less than 100% indicate the system would be undersized and greater than 100% means the system would be oversized.

An appropriate AHU can be selected based on the required space heating output. The detailed selection process was described in Section 2.3 of the laboratory test report (Schoenbauer et al. 2012a). The capacity range of the properly installed AHU must be greater than the home space heating demand plus a safety factor. In order to achieve condensing heating plant operation, the AHU coil must be large enough to transfer the required heat output at an acceptable air temperature (115°F was selected for this study) and a low return water temperature. In order to meet acceptable operating parameters (especially 115°F air temperature) combi system heating capacities often exceeded the required capacity to meet the home load. Figure 5 shows the

relationship between return water temperature and system efficiency as determined by laboratory testing. There was a 2% reduction in efficiency of the Polaris system from reducing return water temperature from 100°F to 90°F. Return water temperatures below 100°F resulted in loads on the system near the maximum input rating of the WHs. These conditions resulted in small reductions in output, reducing the overall efficiency. A maximum return water temperature of 105°F was selected to provide a good tradeoff between higher efficiency and reasonable coil size and cost. These conditions will help maximize the combi system energy efficiency. Larger capacity hydronic coils were not available at the time of this study that would meet the installation criteria for high load homes.

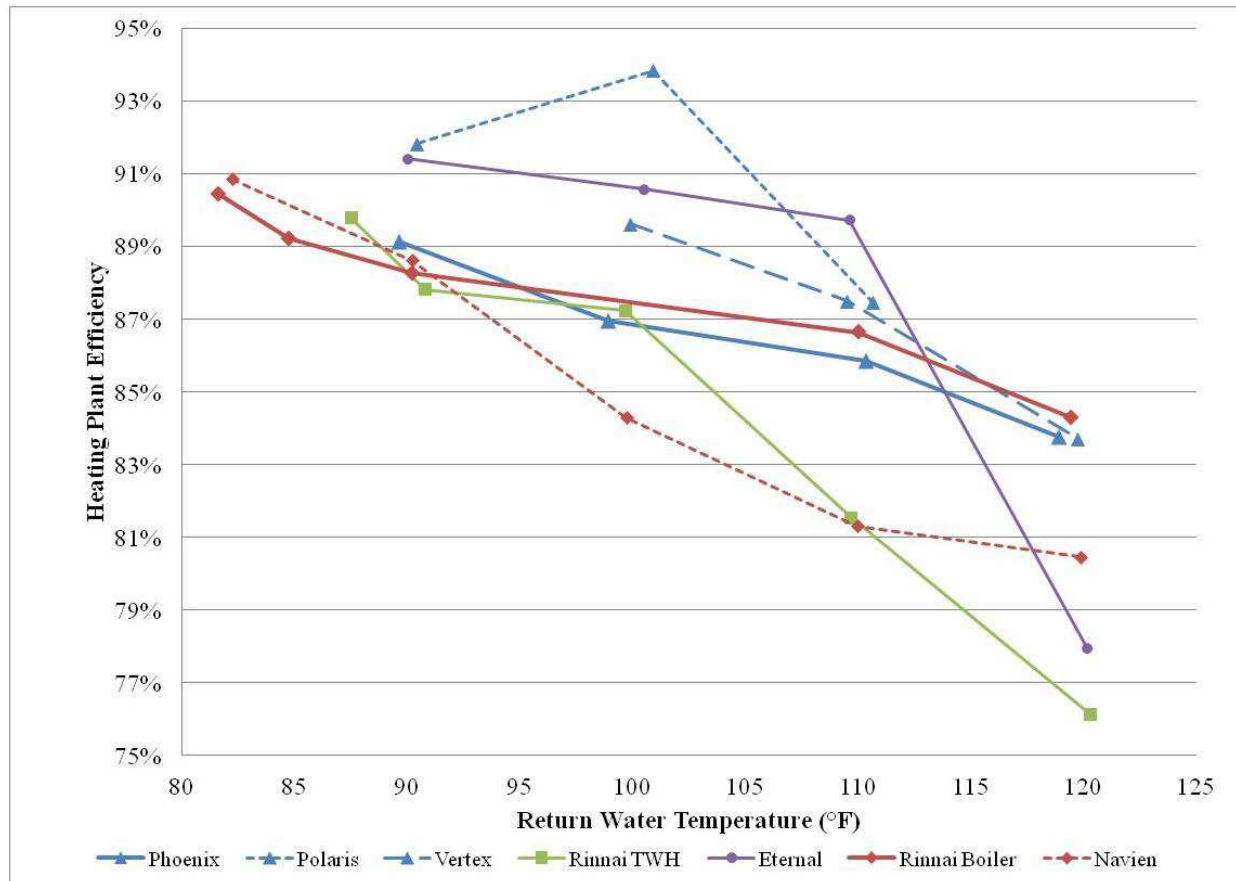


Figure 5. Laboratory testing of heating plant efficiency over a range of return water temperatures

A DHW mixing or tempering valve was included on every combi system to provide acceptable hot water temperatures at the fixtures. A mixing valve blends the heating plant DHW supply water with cold inlet water to produce the hot water supplied to the fixtures. Space heating supply water temperatures are often required to be higher than 120°F in order to meet the demand at an acceptable supply air temperature. For this study the systems used a set point or operating temperature of 140°F. The mixing valve set point was initially set to 120°F at each site to decrease the risk of scalding. The mixing valve temperature control capabilities also helped to reduce some of the transient changes in the temperature of the hot water delivered to the fixtures.

3 Analysis

Two types of analysis were performed. The first was an energy savings calculation computed from the difference between the annual energy use of the existing furnaces and WHs and that of the combi systems. The second computed the combi system water heating, space heating, and combined annual efficiency.

3.1 System Efficiency

When the combi systems were installed, additional instruments were added to calculate the space and DHW heating energy outputs and document system operating characteristics. The space heating energy output rate was computed from the hydronic coil's water flow rate and temperature difference across the coil (see equation 2). The DHW energy output was calculated from the DHW flow rate and temperature difference between the water entering the tee to the mixing valve and leaving the mixing valve. The relationship between the system energy input and output was used to determine the system efficiency over a wide range of loads.

$$q_{in} = Q_g \cdot HF \quad (1)$$

$$q_{out} = C_p \cdot \rho \cdot Q_w \cdot 60 \cdot \Delta T \quad (2)$$

$$\eta = q_{out}/q_{in} \quad (3)$$

where:

q_{in} = natural gas energy input, Btu/h

q_{out} = energy output, Btu/h

C_p = specific heat of water (varies by temperature), Btu/(lb*°F)

ρ = water density at the flow meter (varies by temperature), lb/gal

Q_w = water flow rate, gal/min

ΔT = water outlet – inlet temperature difference, °F

Q_g = burner natural gas flow rate, ft³/h

HF = natural gas heat factor, Btu/ft³

η = thermal efficiency

The approach described above generated a linear relationship between daily average energy input and output of the combi system. This approach has been used previously for WHs (Schoenbauer et al. 2011) and commercial boilers (Butcher 2011). Because the combi system efficiency changed with the space/DHW load ratio and there was not simple linear relationship between the two loads, the combi system energy use analysis required a more complicated modeling scheme than that for the existing systems. Energy use and system efficiencies could not be estimated by year or season, they were determined for each day and then summed to get annual results. A Minneapolis/St. Paul Typical Meteorological Year 3 dataset was used to generate the daily average outdoor temperatures and the site specific inlet water temperature models were used to determine the daily inlet water temperature. The same set of site specific linear relationships between the DHW inlet water temperature and load used for the existing systems (see Figure 11) were used to compute the combi system daily average DHW load. A linear regression of the combi system daily average space heating output and outside temperature (see Figure 6) was

used to determine the daily space heating loads. The daily total load was simply the sum of the DHW and space heating loads.

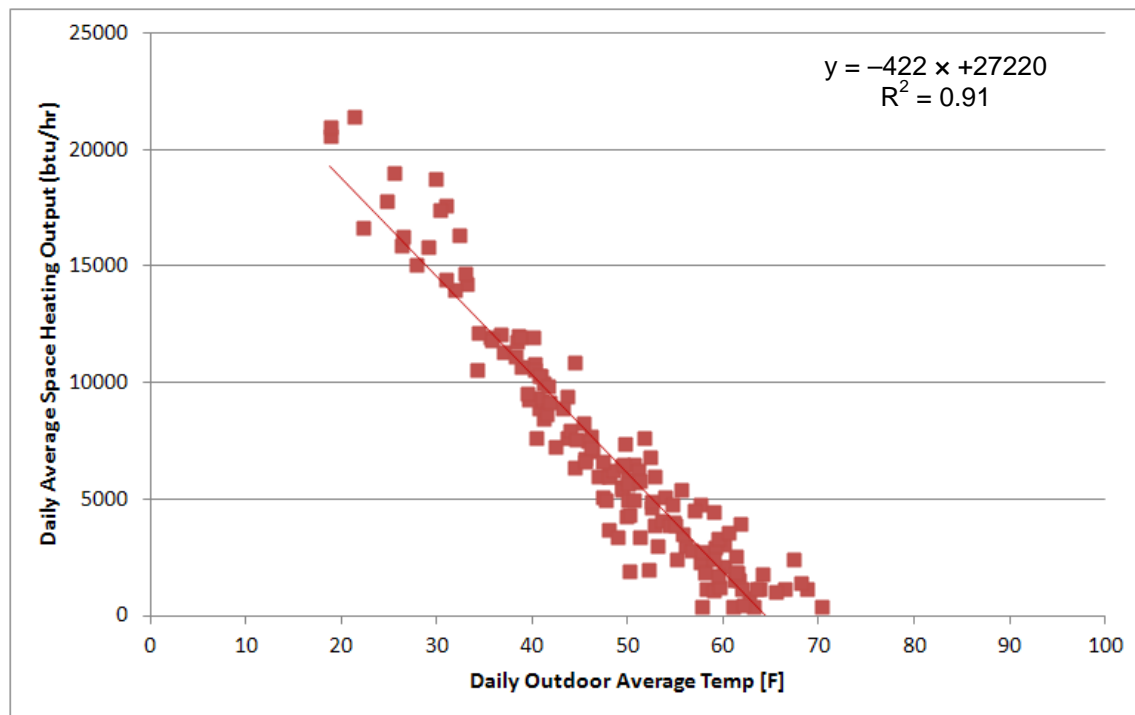


Figure 6. Daily space heating load by OAT for Site 1061

A model of space heating output versus the temperature difference of the conditioned space and the outside air was also considered. This model was less reliable than the output versus outdoor air model described above.

Monitoring days were split into two categories: those with space and water heating and those with DHW only. The data from each category were used to develop two separate linear energy input/output relationships. Figure 7 shows the natural gas input versus the combined space and water heating energy output (or load) for days when there was space heating. Figure 8 shows the same relationship for days with water heating only. Figures for these relationships for all sites are available in Appendix A and Appendix B.

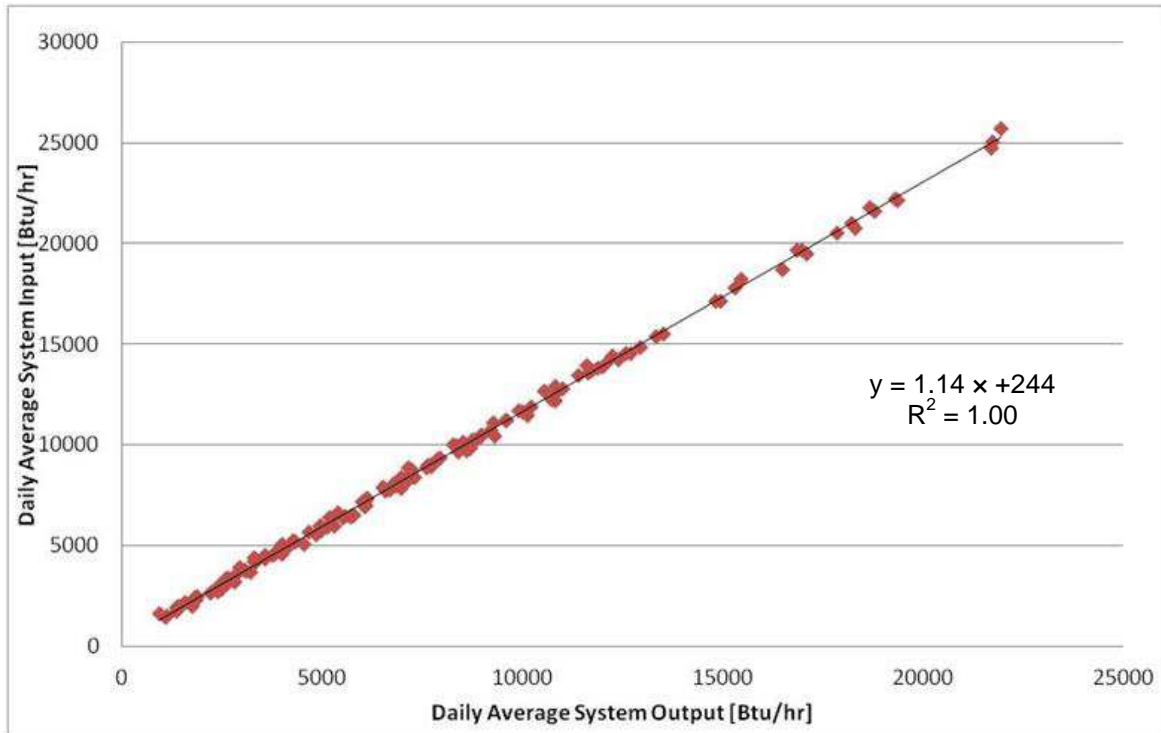


Figure 7. Input versus output relationship for the combi system on days with space and water heating at Site 1061

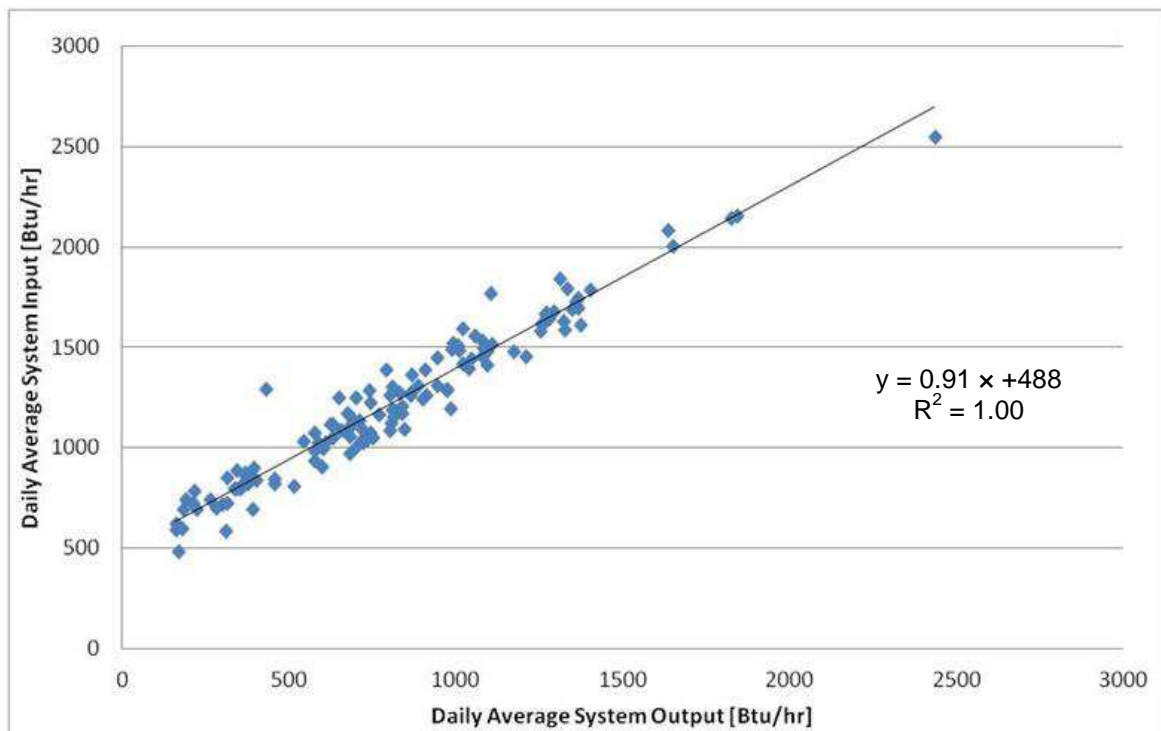


Figure 8. Input versus output relationship for the combi system on days with DHW only at Site 1061

3.2 Annual Energy Savings

The estimated annual energy uses of the existing systems were computed from models of daily use with outside temperature applied to historical outside temperature data. The existing system energy use was computed using furnace and WH burner runtimes with measured burner gas flow rates and furnace electricity use. A linear model of space heating gas use with outside temperature was generated from a linear regression of the furnace daily average gas use and outside temperature (see Figure 9). This was used to determine the house balance point temperature or outside temperature above which no space heating is required. The balance point temperature was applied to a Typical Meteorological Year dataset for the local NOAA station (Minneapolis-St. Paul International Airport) to determine the heating season average outside temperature and duration. The average outside temperature was used with the regression model to compute the average energy use and that was multiplied by the heating season duration to determine the annual energy use. A linear model of the energy use with the difference between inside and outside temperature was also generated. The heating season average inside temperature was applied to that model to determine the balance point temperature and then the method described above was used to determine the annual space heating energy use. The model using OAT alone proved to be more reliable and was used at all sites. Figure 9 shows the comparison of the R-squared values for both models. Most sites showed a very small difference in the models, likely due to very little day to day change in the indoor air temperature. Two sites had significantly worse models using the temperature difference; therefore the OAT model was selected for the analysis.

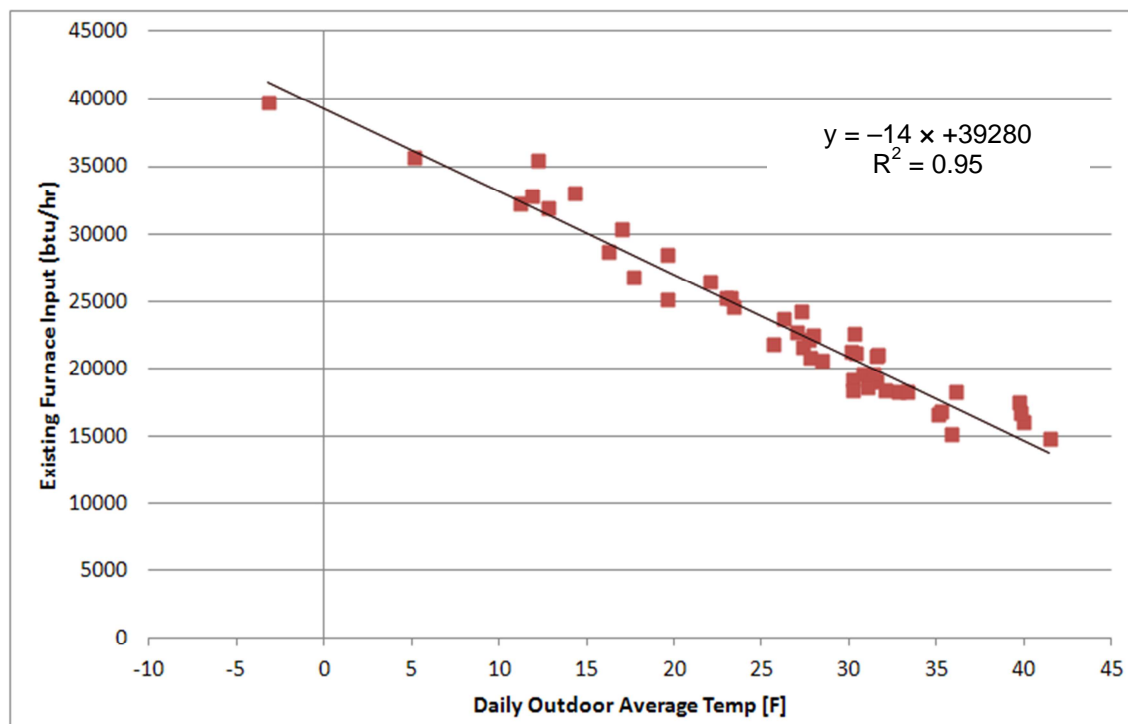


Figure 9. Existing furnace energy consumption versus OAT for Site 1061

Table 6. Comparison of the Space Heating Output Models

Site #	OAT Model (R-Squared)	Indoor-OAT Model (R-Squared)	% Difference
1026	0.85	0.87	1
1027	0.83	0.84	2
1028	0.95	0.95	1
1030	0.42	0.31	-35
1031	0.87	0.91	5
1037	0.66	0.68	3
1045	0.88	0.88	1
1047	0.85	0.90	6
1049	0.84	0.84	0
1052	0.84	0.84	0
1054	0.64	0.68	7
1055	0.80	0.86	8
1056	0.87	0.86	-2
1061	0.91	0.90	-2
1062	0.86	0.86	0
1063	0.70	0.72	3
1065	0.98	0.97	0
1070	0.72	0.58	-24
1078	0.65	0.63	-3

A three-step process was used to compute the annual DHW daily or annual energy use based on inlet water temperatures, DHW demand models, and appliance efficiency models. First, a model was developed for inlet water and OAT by day of the year to establish the seasonal variation and annual average inlet water temperature. The inlet temperature has been found to have a sinusoidal relationship with day of the year (Burch and Christensen 2007). The inlet temperature model parameters are functions of the average OAT, time lag from the OAT, and difference in the maximum and minimum average monthly temperatures. Figure 10 shows the measured and modeled weekly inlet water temperatures at site 1052. For each site the existing and combi system inlet water temperature measurements were used to develop the model for daily average inlet water temperature.

The second step was to develop a linear relationship between the inlet water temperature and DHW demand or load. Due to the high variance in the daily water usage and the limited change of the inlet water temperature over the monitoring period for the existing systems, it was not possible to generate separate models for the existing and combi systems. However, Schoenbauer et al. (2011) demonstrated that hot water energy demand does not differ significantly for a TWH and tank type WH in the same house. The data from both the existing and the combi system was combined to establish a single inlet water temperature to DHW load linear relationship for each site. Figure 11 shows the inlet water temperature and hot water energy output relationship for site 1062.

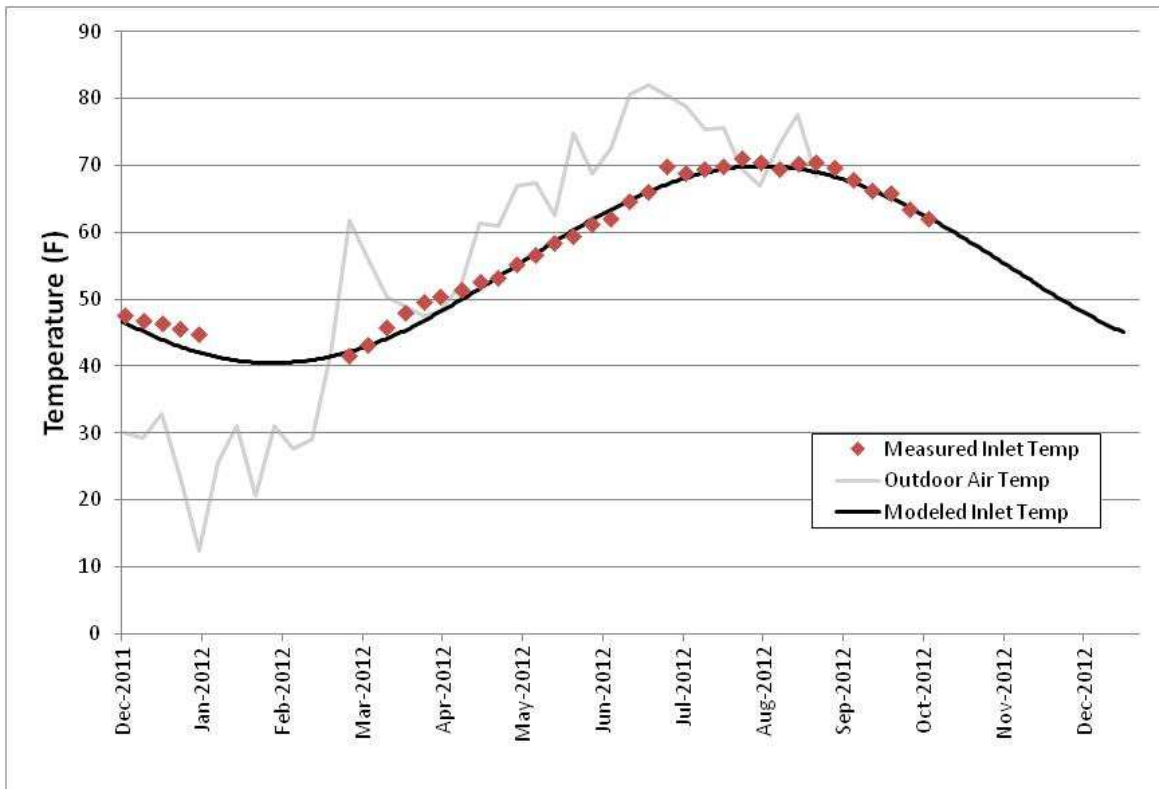


Figure 10. Inlet water temperature measured and modeled data at site 1052

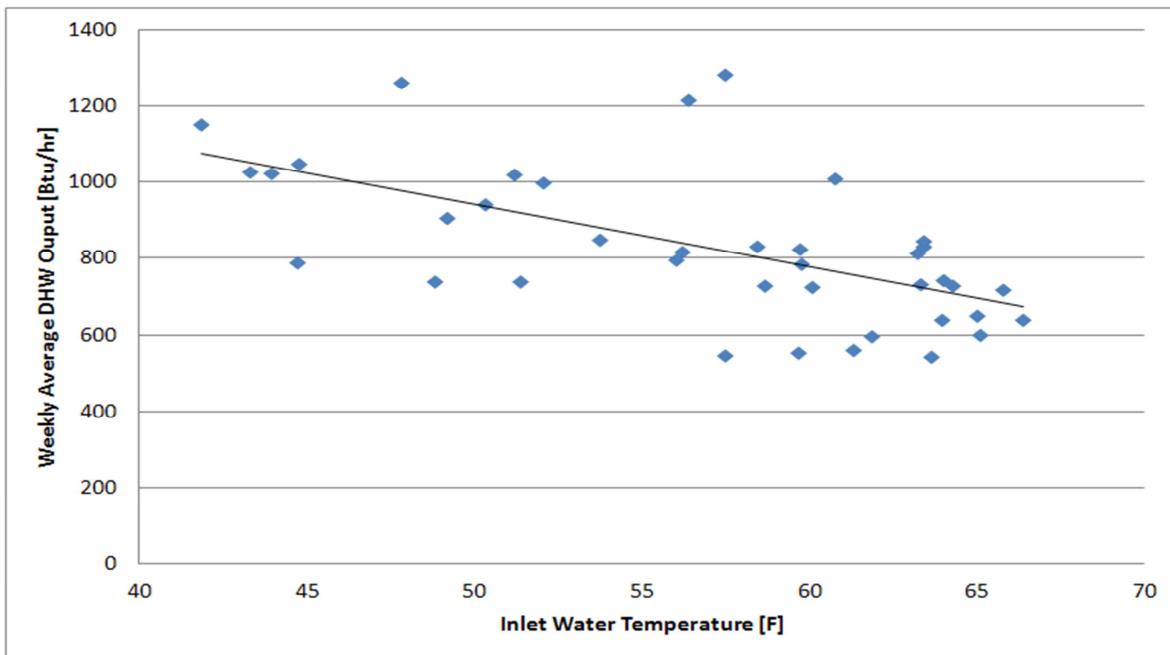


Figure 11. Average domestic water heating load for Site 1061

The third and final step was to use a linear model of appliance energy use with DHW load to compute daily or annual energy use. The energy load or output of the existing WHs was not

monitored. The input versus output relationship was based on results for eight identical 0.60 EF, 40-gal, ND SWHs from a previous study (Schoenbauer et al. 2011). The SWH models were very well defined, each model had R-squared values higher than 0.94. The eight models were also consistent; at a daily load of 30,000 Btu/h (about 50 gal at a 70°F temperature rise) the models varied by only 8%. This approach was expected to slightly underestimate the actual energy use of the existing WHs that had confirmed or estimated EFs from 0.56 to 0.59. The energy input versus output relationship for the combi system DHW operation was based on linear regressions of measured data from days when the systems were not used for space heating.

The measured natural gas use of the existing WH during the pre period was compared to the modeled value over the same time period to evaluate the accuracy and bias of the modeling method. The three step process described previously was performed using the OATs for the monitoring period, inlet water temperature and DHW load models for each site, and the energy input versus output model for the 0.60 EF WH. Figure 12 shows a comparison of the measured and modeled gas use. The average difference between the measured and modeled use was 0.6%. The median percent difference was 0% with an interquartile range of –0.04% to 0.1%.

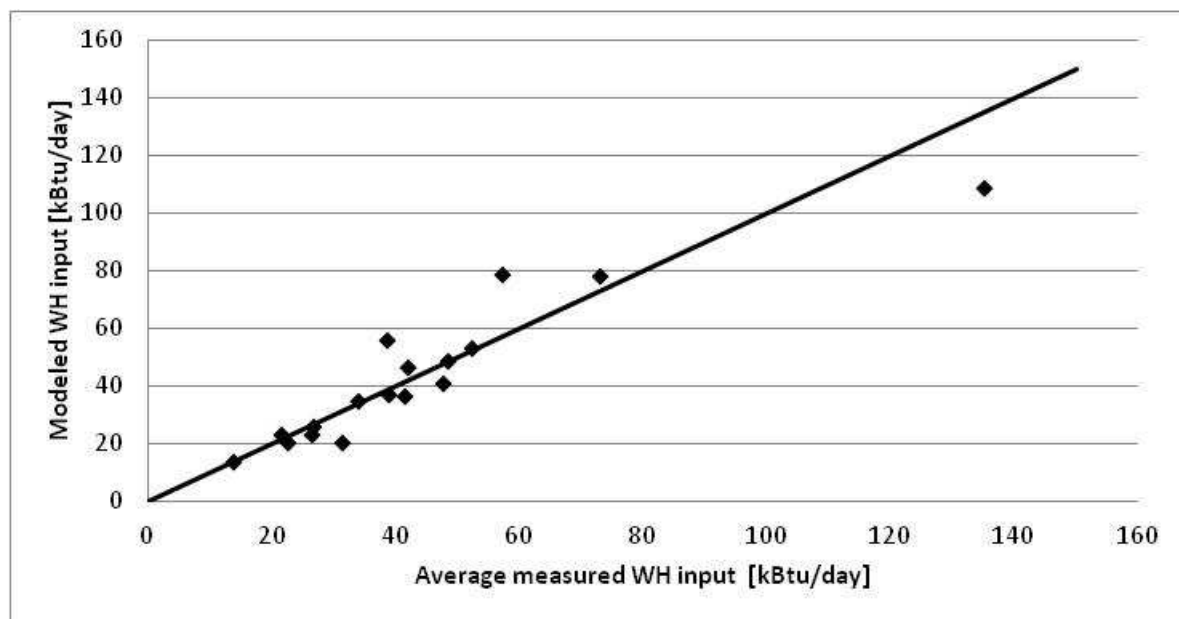


Figure 12. Comparing existing WH energy use model to measured data

The combi system gas consumption was computed from the measured gas flow rate and heating value (see equation 1), and a Watt transducer measured electricity use. The combi system annual energy use was computed using a similar analysis approach as that used for the existing systems. However, since the heating plant input versus output relationship was different for the DHW only and combined DHW/space heating operation, it was necessary to use a daily calculation of energy use. The details of the calculation procedure are described in the following section. The uncertainties of the existing and combi system annual energy uses were determined using a standard procedure for propagating the instrumentation and regression uncertainties.

3.3 Other Analysis

The AHU supply air and conditioned space temperatures helped identify occupant comfort issues and complaints. An analysis of the frequency distribution of water temperature returning from the AHU and the supply air temperature and transient temperatures when the AHU was first turned on helped determine whether the systems were properly sized and correctly installed with an acceptable control strategy.

Detailed cost data were collected from each installation in the implementation project of 200+ systems. These data were used for a cost benefit analysis, a simple payback calculation, and evaluation of the change in cost as contractors became more familiar with the installations.

The space heating design load for each house was estimated from the linear models of space heating output with outside temperature. Those values were compared to the hydronic AHU steady-state output to determine the degree of coil under or over sizing. In addition, the analysis compared the computed design loads to values from the NEAT software and contractor calculations to assess the reliability and bias of the NEAT and contractor estimates. The space heating models included linear regressions of existing furnace gas use with outside temperature and inside/outside temperature difference. The computed use for the outdoor design temperature was multiplied by the furnace AFUE to estimate the space heating design load. Linear regressions of the hydronic AHU energy use with outside temperature provided two separate calculations of the space heating design load.

4 Results and Discussion

Combi systems were characterized using detailed field monitoring data. The energy efficiency, savings potential, installed cost, delivered load, and occupant comfort were all assessed under typical installed conditions with real occupants. Each category was analyzed to determine the potential of combi systems to provide improved efficiency, lower energy cost, and improved combustion safety.

4.1 Energy Efficiency

The primary goal of this project was to assess whether it was possible to provide high installed efficiency with a single heating plant. The summer efficiency was used to characterize the combi system DHW-only performance and the winter efficiency used to characterize performance when space heat is the dominant load. The annual efficiency showed the combined efficiency of all space and water heating in a year.

Efficiency was calculated by summing the daily output energy for space and water heating over an entire year and dividing by the year's input energy. Figure 13 and Table 7 show the calculated efficiencies at each site.

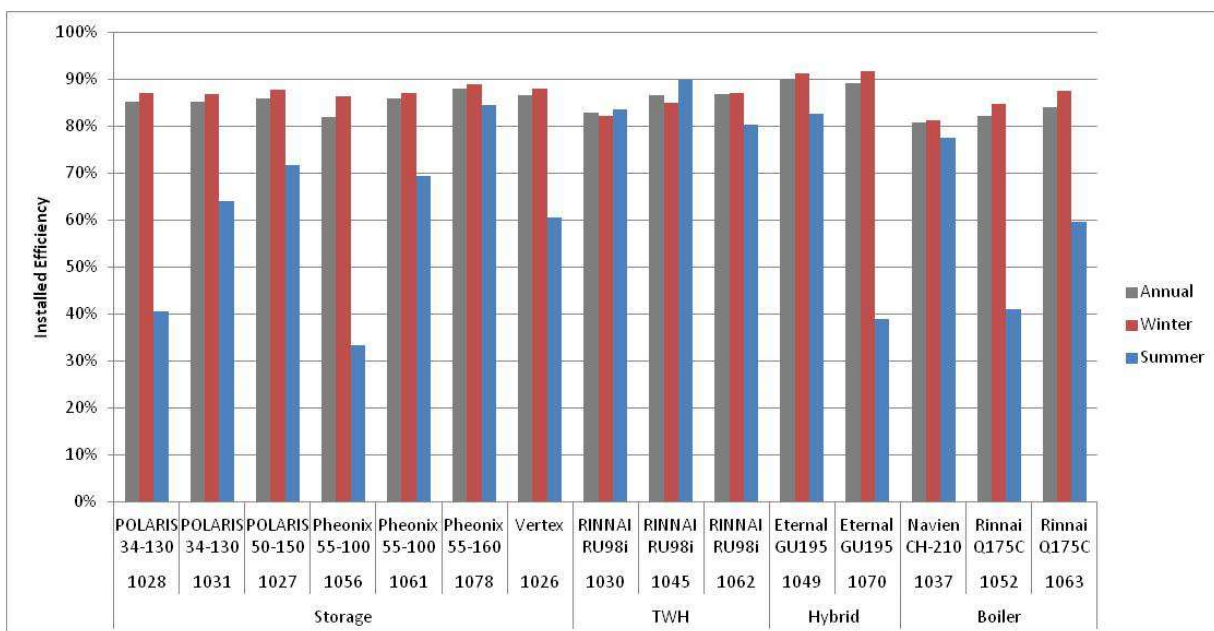


Figure 13. Combi system installed efficiencies

Note: Analysis of this figure continues throughout Section 4.

Table 7. Installed Efficiency of the Combi Systems at Each Site

Site Number	Heating Plant	Combination Installed Efficiency		
		Annual (%)	Winter (Space Heat) (%)	Summer (DHW) (%)
1028	Polaris	85	87	41
1031	Polaris	85	87	64
1027	Polaris	86	88	72
1056	Phoenix	82	86	33
1061	Phoenix	86	87	69
1078	Phoenix	88	89	84
1026	Vertex	87	88	60
1030	Rinnai TWH	83	82	84
1045	Rinnai TWH	87	85	90
1062	Rinnai TWH	87	87	80
1049	Eternal	90	91	83
1070	Eternal	89	92	39
1037	Navien	81	81	77
1052	Rinnai Boiler	82	85	41
1063	Rinnai Boiler	84	87	60

The combination system energy load, use, and efficiency were calculated for each day in a typical year as described in Section 3. Field data were used to define the relationships between environmental conditions, the household loads and the energy consumption of each system. These relationships characterized the household and the occupants as well as system performance.

Regressions of the system inputs versus output showed the natural gas consumption necessary to produce a given amount of DHW and space heat (Table 8). Days with both space heat and DHW loads (winter) were analyzed separately from days with only DHW (summer). Water use at one site (1056) was low on most days, and the low usage prevented the system from being fully characterized, resulting in a low R-squared value. However, the remaining system performance regressions had very strong fits (R-squared values from 0.89 to 1.00).

Table 8. Input Versus Output Models for the Combi Space Heating Systems

Site Number	Heating Plant Type	DWH Only Days			Space Heating and DHW Days		
		Slope	Intercept (Btu/h)	R-Square	Slope	Intercept (Btu/h)	R-Square
1028	Storage	1.03	414	0.90	1.12	439	1.00
1031		0.99	542	0.95	1.13	392	1.00
1027		1.04	437	0.97	1.12	443	1.00
1056		0.57	764	0.34	1.11	617	1.00
1061		0.91	488	0.94	1.14	244	1.00
1078		1.08	253	0.89	1.11	284	0.98
1026		1.24	433	0.97	1.12	499	1.00
1065	TWH	–	–	–	1.19	314	0.99
1030		1.09	218	1.00	1.22	31	1.00
1045		1.07	72	1.00	1.19	–196	1.00
1062	Hybrid	1.09	153	1.00	1.14	106	1.00
1049		1.02	286	1.00	1.08	220	1.00
1054		1.06	323	0.97	1.08	367	1.00
1070	Boiler	1.03	320	0.94	1.06	348	1.00
1037		1.17	165	1.00	1.22	137	1.00
1047		–	–	–	1.15	227	1.00
1055		1.33	28	0.99	1.23	308	0.96
1052		1.28	445	0.96	1.15	463	1.00
1063		1.24	391	0.98	1.10	535	1.00

The relationships between the loads and environmental conditions were more strongly dependent upon occupant behavior and had weaker fits (see Table 9). Daily variance in occupant behavior affects the load. For example, the data showed variance in daily thermostat operation and homeowners indicated short term variable occupancy, which changed the daily DHW load. Additionally, the relationship between DHW use and environment (inlet water temperature) was not statistically significant at sites with high variability in DHW use, low DHW loads, and/or small seasonal variance in inlet water temperature. In these cases, when the load does not vary with inlet water temperature, we used the average DHW.

Table 9. Load Curves for Space and Water Heating

Site Number	Heating Plant Type	DHW Output Versus Inlet Water Temperature					Space Heating Output Versus OAT			
		Slope	Intercept (Btu/h)	R-Squared	P-Value	Signif?	Slope	Intercept (Btu/h)	R-Squared	Balance Point (°F)
1028	SWH	-3.52	325	0.09	1.000	No	-311	21,017	0.95	68
1031		-2.94	946	0.02	0.048	No	-287	19,253	0.87	67
1027		-21.53	2,300	0.17	0.010	Yes	-344	21,081	0.83	61
1056		-7.39	667	0.61	0.000	Yes	-263	16,425	0.87	62
1061		-16.42	1,762	0.36	0.000	Yes	-422	27,220	0.91	65
1078		-40.62	4,406	0.28	0.001	Yes	-272	16,803	0.65	62
1026		2.64	547	0.00	0.820	No	-497	31,767	0.85	64
1065	TWH	0.00	0	0.00	0.130	No	-369	24,221	0.98	66
1030		-30.85	3,994	0.45	0.000	Yes	-245	16,052	0.42	65
1045		-36.09	3,664	0.23	0.001	Yes	-297	21,351	0.88	72
1062	Hybrid	-13.14	1,521	0.03	0.274	No	-315	18,822	0.86	60
1049		-48.30	4,163	0.50	0.000	Yes	-310	17,978	0.84	58
1054		-3.80	452	0.04	0.207	No	-230	13,654	0.64	59
1070	Boiler	0.74	61	0.02	0.384	No	-239	15,479	0.72	65
1037		-29.09	2,934	0.33	0.000	Yes	-183	12,812	0.66	70
1047		-24.02	1,643	0.00	0.898	No	-439	27,709	0.85	63
1055		-10.10	875	0.34	0.000	Yes	-221	14,527	0.80	66
1052		-3.61	520	0.03	0.289	No	-309	20,536	0.84	67
1063		-8.32	1,260	0.06	0.172	No	-229	15,288	0.70	67

We do not report efficiencies at four sites. These sites had data issues preventing full analysis of the combi systems. The occupants of sites 1047 and 1065 relocated before monitoring was completed. An additional space heating source was discovered at Site 1054 immediately prior to monitoring. The variability in secondary heat source use made the load difficult to characterize. Site 1055 had several instrumentation and installation issues, such as instrumentation installed in the wrong location and a heating plant wiring issues that caused atypical operation. These issues prevented data collection over an extended period.

Homes with statistically significant relationships between DHW load and inlet water temperatures had R squared values between 0.17 and 0.63. Occupant behavior had smaller impacts on the space heating load versus OAT relationships (R-squared from 0.42 to 0.98).

Summer Efficiency. The summer efficiency was the measured efficiency from July when space heating loads were nonexistent and there were only DHW loads. Summer or DHW-only efficiencies had a large variance from site to site (33% to 90%). The summer efficiency was impacted by the heating plant type, household DHW load, and average cycle length.

Heating Plant Type. Standby losses were largely determined by the heating plant type. Systems with storage capacity kept water hot at all times. Poor insulation and/or large storage volumes increased the standby losses, which were previously tested in a laboratory (see Figure 14) (Schoenbauer et al. Upcoming). The higher volume SWHs had the highest standby losses. The hybrid WHs and boilers with storage (Rinnai) also had significant standby losses. The TWHs and low mass boilers (Navien) had very small standby losses. The range of efficiency for the tankless units was due to hot water usage profile. The site with the highest efficiency (90% at Site 1045) had a higher percentage of water usage from large draws, 91% of hot water volume was consumed in draws of greater than 3 gal at Site 1045 compared to an average of 80% at the other sites. The impact of standby loss was higher in the lowest load homes.

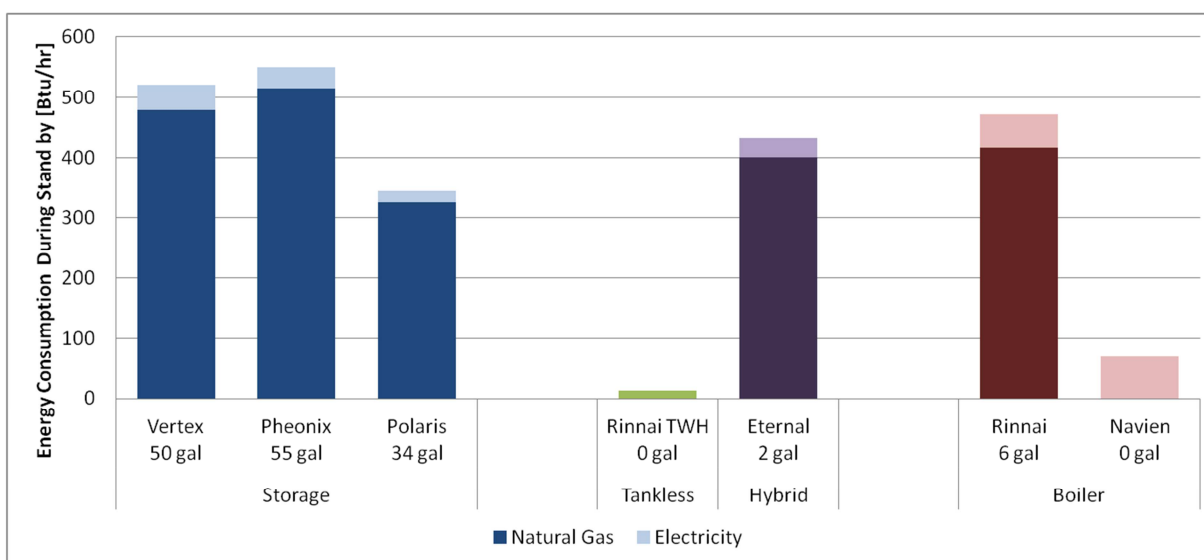


Figure 14. Standby energy use for combi systems

DHW Load. Standby or idle time increased with lower DHW loads. The lower efficiency sites (1028, 1056, 1070, and 1052) were those that had a combination of: (1) heating plants with

greater standby loss and (2) low DHW use (Table 10). Homes with less water heating usage reduce the summer efficiency for systems with hot water storage (see Figure 15). The efficiency of a condensing SWH from field data collected during the WH only mode is included for comparison purposes.

Table 10. DHW Loads at Each Site

Site Number	Heating Plant	DHW Load	
		Energy (Btu/h)	Volume (gpd)
1028	Polaris	131	12.7
1031	Polaris	790	42.3
1027	Polaris—Stress	1,041	54.6
1056	Phoenix	197	10.8
1061	Phoenix	830	33.3
1078	Phoenix—Stress	2,115	89.0
1026	Vertex	715	33.9
1065	Vertex	337	19.5
1030	Rinnai TWH	1,966	106.8
1045	Rinnai TWH	1,726	81.1
1062	Rinnai TWH	832	41.5
1049	Eternal	1421	67.7
1054	Eternal	251	13.5
1070	Eternal	101	5.7
1037	Navien	1,227	72.7
1047	Navien	768	23.7
1055	Navien	264	16.5
1052	Rinnai Boiler	301	13.0
1063	Rinnai Boiler	789	34.0

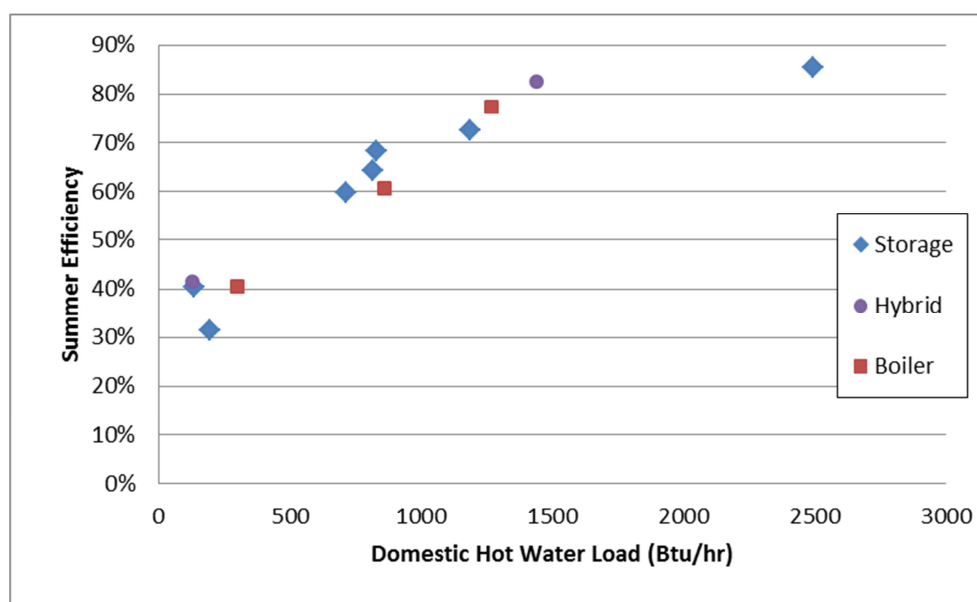


Figure 15. Combi system summer efficiency versus DHW load

Burner Cycle Length. Cycle length had a large impact on daily efficiencies for the low mass systems (TWH and low mass boiler), with shorter cycle length decreasing the daily efficiency. Low mass systems require the heat exchanger and burner to heat up for each event; therefore, for short events a significant portion of the event energy was used to heat up the unit. Low mass systems have no hot storage capacity; therefore, most of the energy at the beginning of a cycle is used to heat up the heat exchanger. If there are no additional draws soon after this short cycle, the heat dissipates to the surrounding area and has no useful effect. Because of this, days with short cycle lengths often have lower efficiencies than days with longer events where the startup effects were minimized (Figure 16).

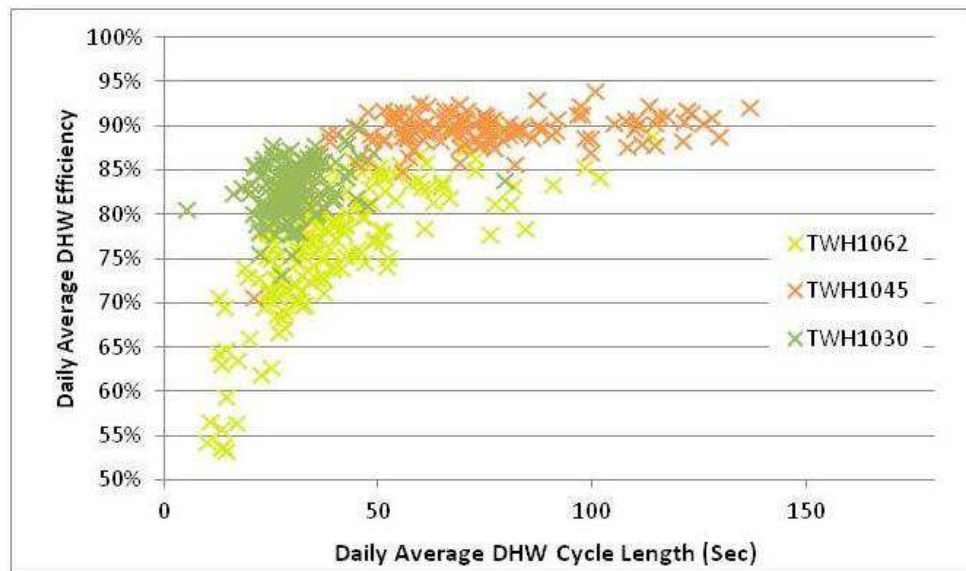


Figure 16. Impact of daily average event length on DHW efficiency

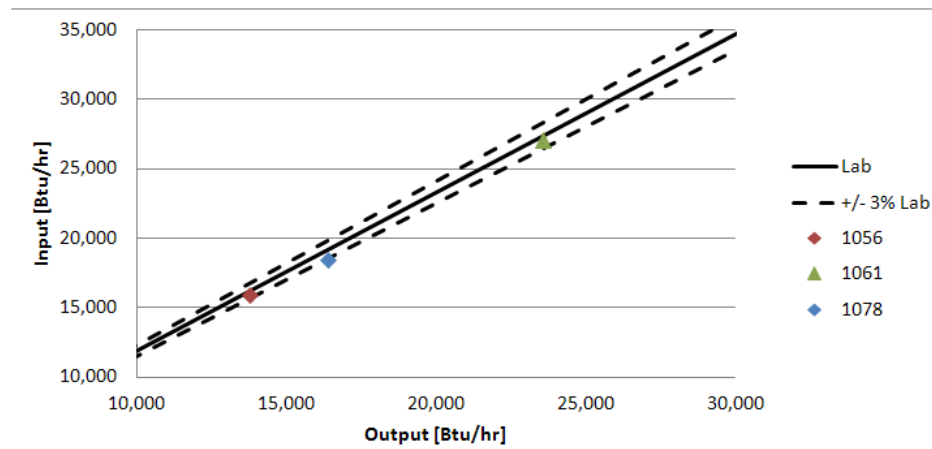
Winter Efficiency. The winter efficiencies were calculated from January data. During this time space heating loads were much greater than DHW loads; making winter efficiency mostly dependent on space heating efficiencies. Winter efficiencies (87% on average) were higher than summer efficiencies (65% on average) at most sites. Winter efficiencies had smaller site-to-site variances (82% to 90%). Operational parameters, such as return water temperature and flow rate, have a high impact on space heating efficiency. Combi systems were installed carefully so that these parameters allowed for the best possible performance and were similar at each site. The variance in winter efficiency was mostly a result of differences in the heating load.

Heating Plant Type. System type had the largest impact on winter efficiencies. The hybrid WHs had the highest winter efficiencies in the field (92% on average). SWHs had slightly lower installed efficiencies (87%). The TWH had lower space heating efficiencies (85%). Laboratory testing, at the field installation condition of 105°F return water temperatures, showed the same rankings (Figure 5).

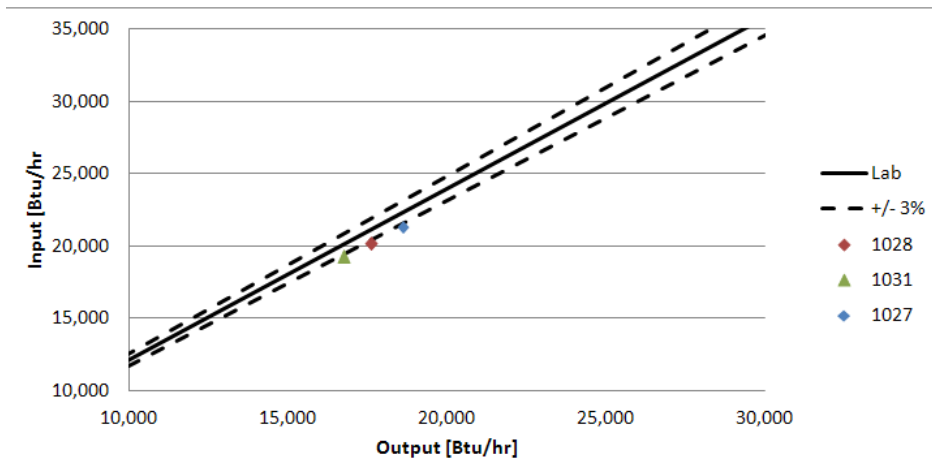
Boiler manufacturers required a primary secondary loop plumbing configuration. The primary loop flow rates were controlled by an internal boiler pump. This configuration made proper installation and set up at two sites with boilers (1037 and 1052) difficult, resulting in an

increased temperature of the return water and reduced the efficiencies (83% on average) (for more information see Butcher, 2010). At site 1063, optimization was more successful. The heating load and house characteristics allowed the system to achieve a lower return water and better efficiency, 87% during peak heating season.

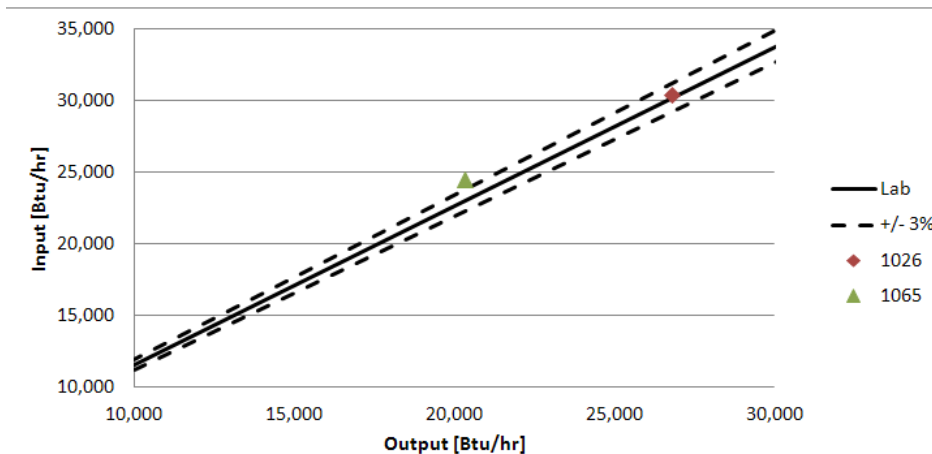
Field and Lab Data Comparison. We compared winter efficiencies measured in the field to the system efficiencies measured in the laboratory for each system type (Figure 17 through Figure 19). Due to the constraints of real homes and loads, the steady-state laboratory conditions were not the same as the field installs. Despite the small differences in laboratory and field conditions the data were consistent. Laboratory data from previously completed testing (Schoenbauer et al. Upcoming) were used to verify field data. Laboratory tests of steady-state operation at 4 gpm and 110°F return water temperature and idle tests were used to determine the laboratory input versus output relationship (Figure 17 through Figure 19). Standby results were used for the zero output intercept and the steady-state test's measured input and output were used with the intercept to define the linear relationship. For comparison, we plotted field data from January when the ratio of space heating to DHW was highest. Field data from 12 of the 19 field sites fall within the $\pm 3\%$ efficiency bands, which was the combined uncertainty for the laboratory and field measurements, shown on each plot. And 16 of the 19 sites were within 5%, while the remaining three sites had less than a 7% difference. The average difference in the field and laboratory measurements was 0.1%. In general, the data were consistent between the field and the laboratory, meaning that two laboratory tests, steady-state and standby, were enough to reasonably predict the installed performance.



a) Phoenix

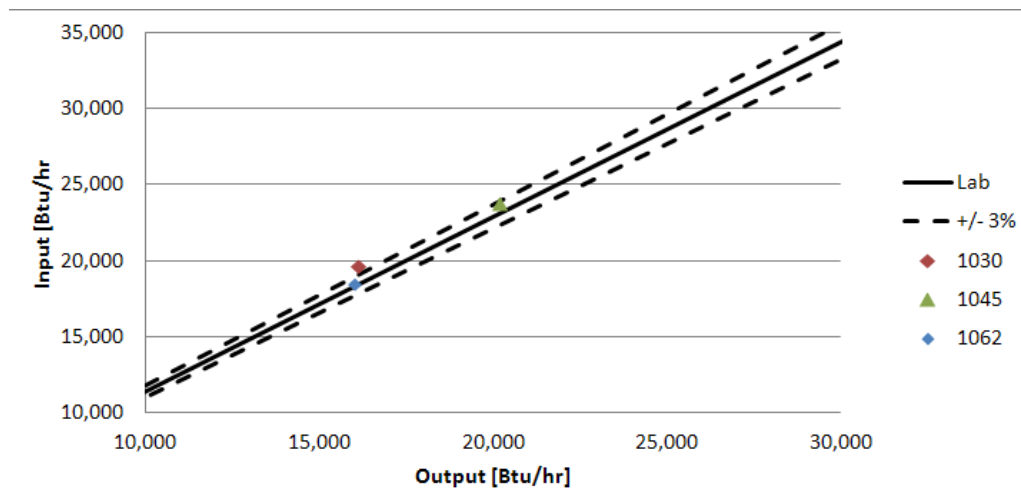


b) Polaris

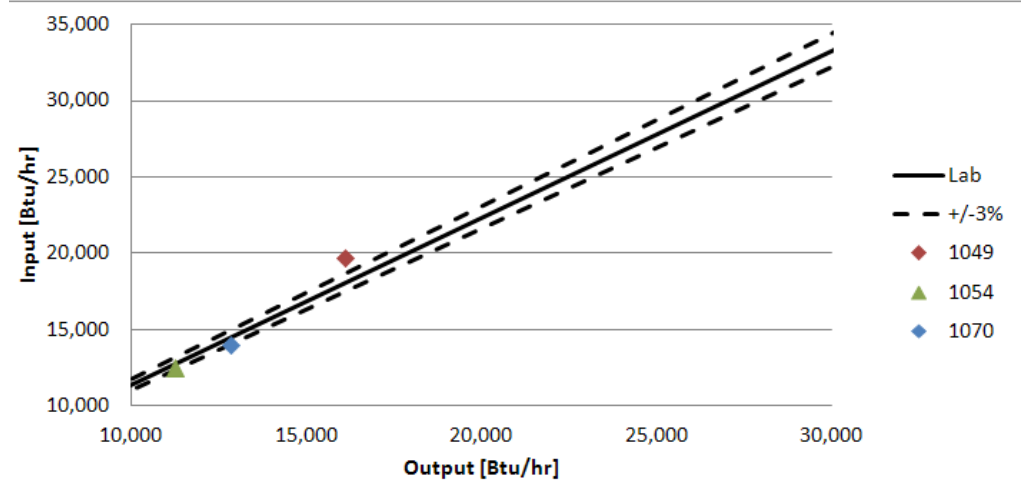


c) Vertex

Figure 17. Comparison of field and laboratory efficiencies for storage-based combi systems



a) Rinnai TWH



b) Eternal

Figure 18. Comparison of field and laboratory efficiencies for TWH and hybrid-based combi systems

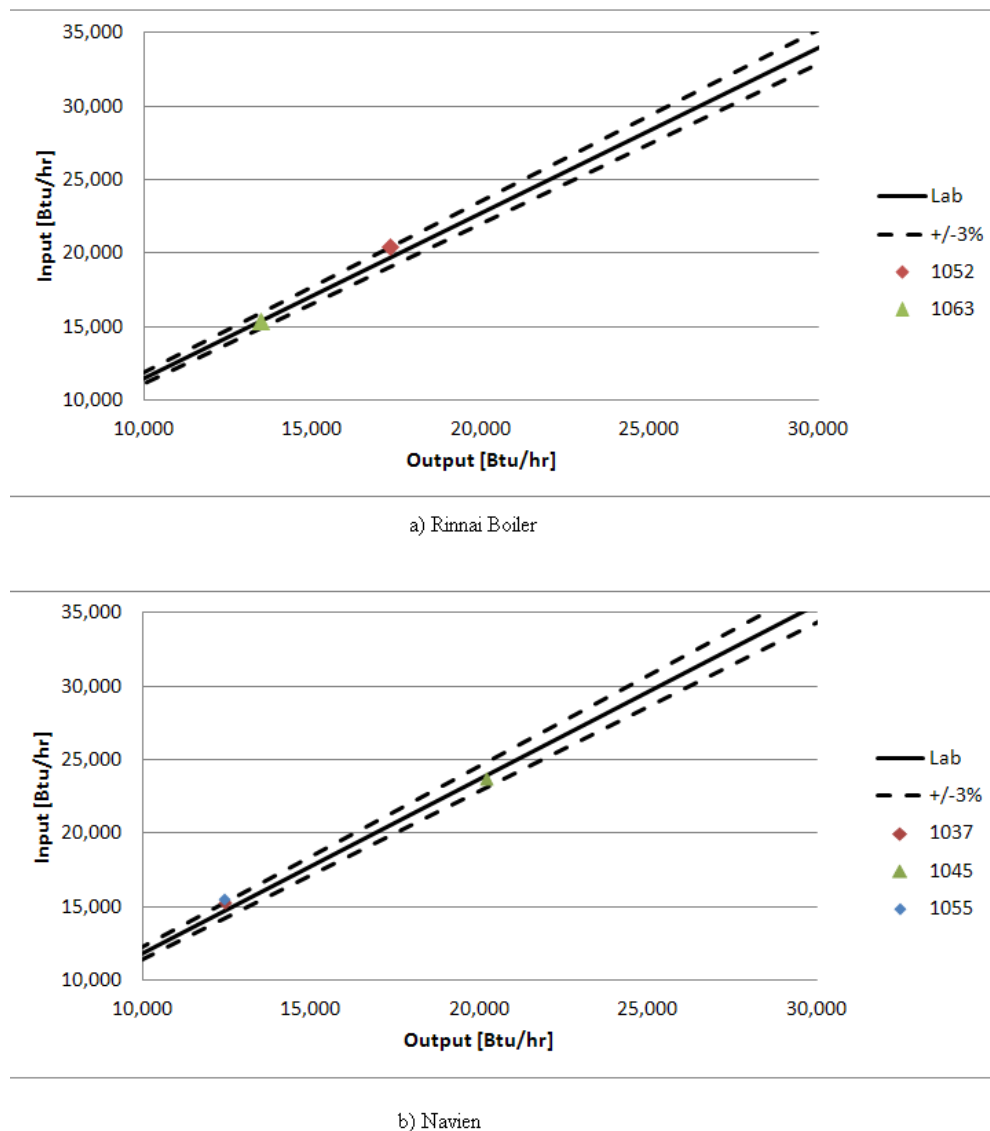


Figure 19. Comparison of field and laboratory efficiencies for boiler-based combi systems

Annual efficiencies were calculated by summing the total daily energy consumption and energy delivered by each system over a typical year. On average the annual efficiencies of the combi systems were 87%. Annual efficiencies ranged from 81% to 90%. DHW (or summer) efficiencies were lower than space heating (or winter) efficiencies for all systems other than the TWH combi systems. Combi system efficiencies were impacted by the size of the load. Systems with smaller loads were typically less efficient than those with larger loads. In general, the combi systems had high efficiency and performed well.

4.2 Energy Savings

We compared existing system and combi systems and found significant gas savings (Figure 20). Energy savings were computed as described in Section 3.2. Savings depended on:

- The existing equipment
- The quality of installation and setup of the combi system

- The space heating and water heating loads of the home.

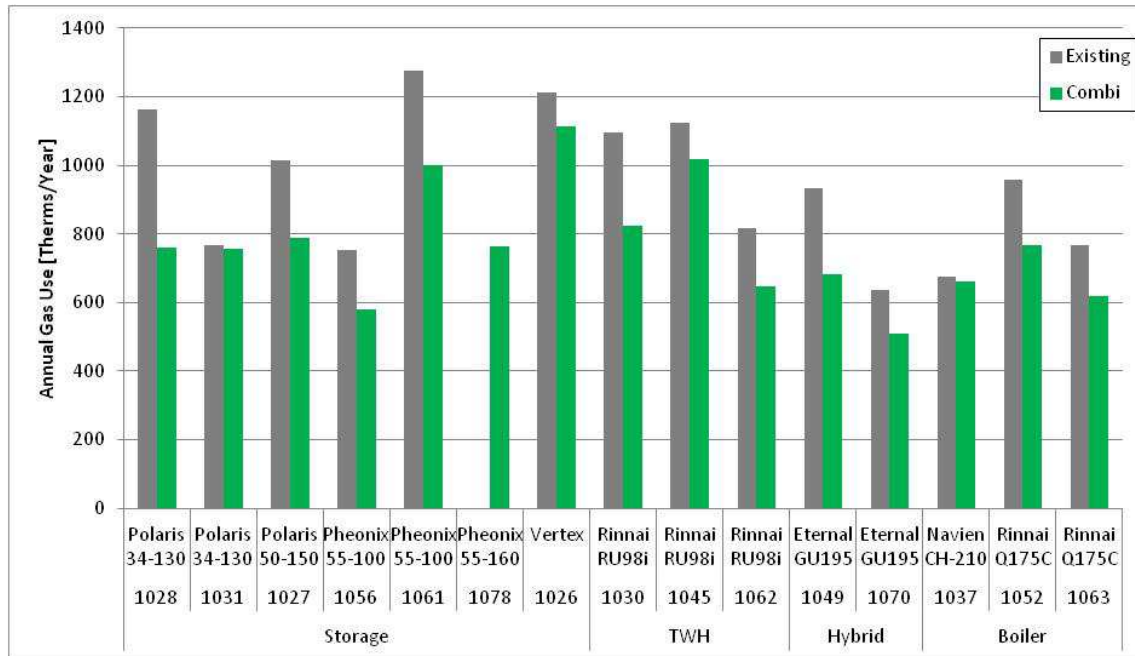


Figure 20. Annual natural gas consumption for space and water heating pre- and post-combination system installation

Figure 21 shows the percentage of natural gas that the combi systems saved. On average, the systems saved 18% of the natural gas previously required to provide space and water heating to these homes.

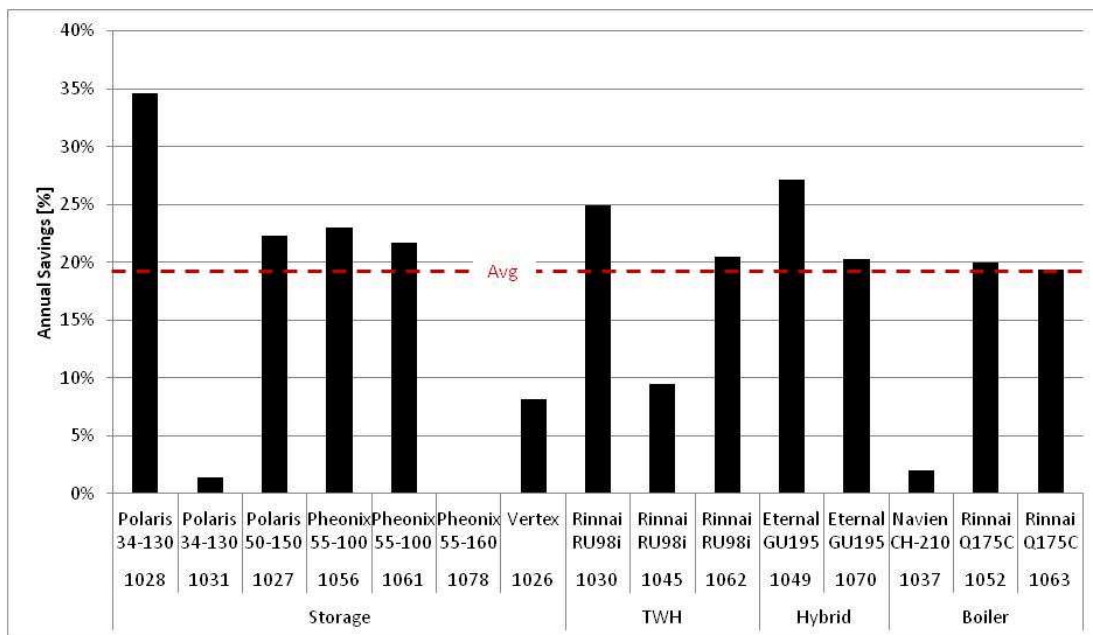


Figure 21. Natural gas savings at each site

Existing Equipment. The performance of the existing furnace had a large impact on the savings potential at two sites. The highest (35%) and lowest (1%) savings came from sites 1028 and 1031. At Site 1028, the existing furnace was the oldest of all the homes, likely dating to the early 1980s. The unit's AFUE was not available, but its age suggests a rating of 70%–78%. The large savings were likely due to a low AFUE (70%) or the system was rated higher but had degraded over time. The existing furnace at site 1031 was a condensing furnace. The furnace was eligible for replacement in this project because it was installed in 2004 and was nearing the end of the expected lifetime. Despite its age, the furnace had an AFUE rating of 93%. The SWH combi system selected to replace the system had a 95% thermal efficiency rating and an installed winter efficiency of 87%. The high efficiency of the existing system reduced the potential for improvement and limited the savings to 1% at this site.

Quality of Installation. We found that it was challenging to balance multiple installation requirements (delivered air temperatures, loads, system configurations, and the conditions necessary to achieve high efficiency). At Site 1037, these difficulties resulted in low savings (only 2% over the existing system). The difficulties were accentuated by the additional installation requirements of boiler systems. Mixing between the loops resulted in water temperatures around 120°F entering the boiler during space heating, reducing performance.

Load. Homes with larger loads saved relatively more energy per year (see Figure 22). The percentage saved remained fairly constant; therefore, in general larger loads had greater savings. The homes available as test sites for this project had all undergone weatherization, greatly reducing their heating loads. There was concern at the outset of this project about combi system capacity, so the selection criteria favored homes with smaller total (DHW and space heating) loads. The results showed that initial capacity concerns were unwarranted and combi systems could have met much higher loads. Additionally, the actual space heating loads in the monitored homes were smaller than initially estimated. AHU maximum output was much greater than the measured space heating design loads (see Section 4.4). Additionally, analysis of the delivered water and air temperatures showed that adequate DHW and air temperatures were supplied to meet demands (see Section 4.5). The project's focus on low-load homes, limited the absolute annual savings, and impacted the cost effectiveness of these systems.

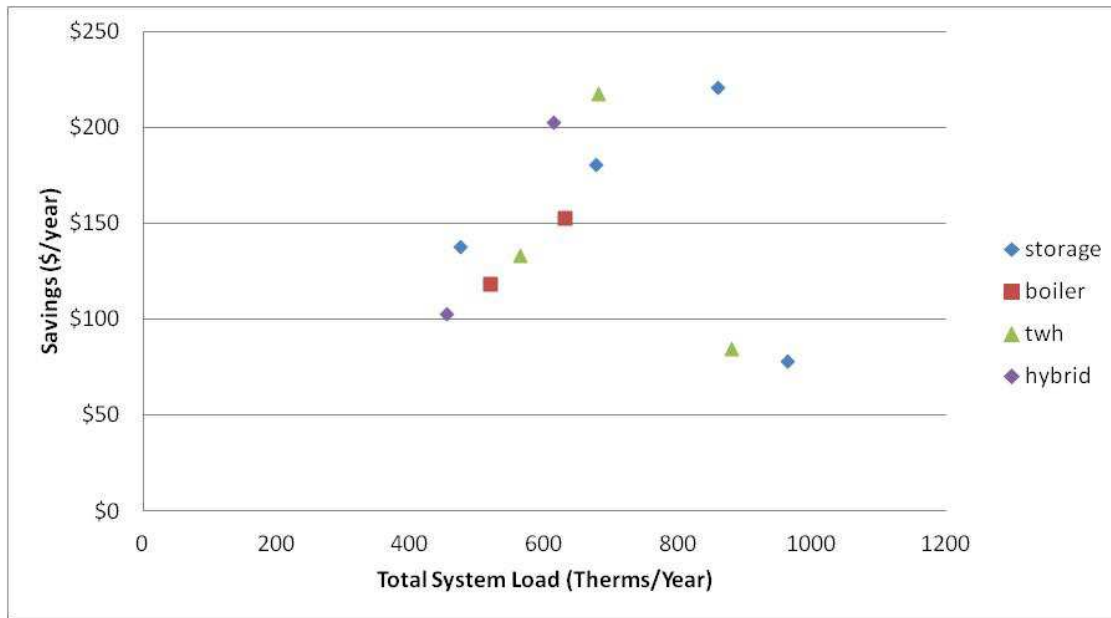


Figure 22. Natural gas savings per site by annual load

Homes selected for this project had undergone low income weatherization prior to any monitoring. These homes were on average 1,540 ft² of conditioned space and had fully insulated 2 × 4 walls with R-50 insulation targets for open attics. This resulted in homes with low space heating loads.

4.3 Cost Analysis

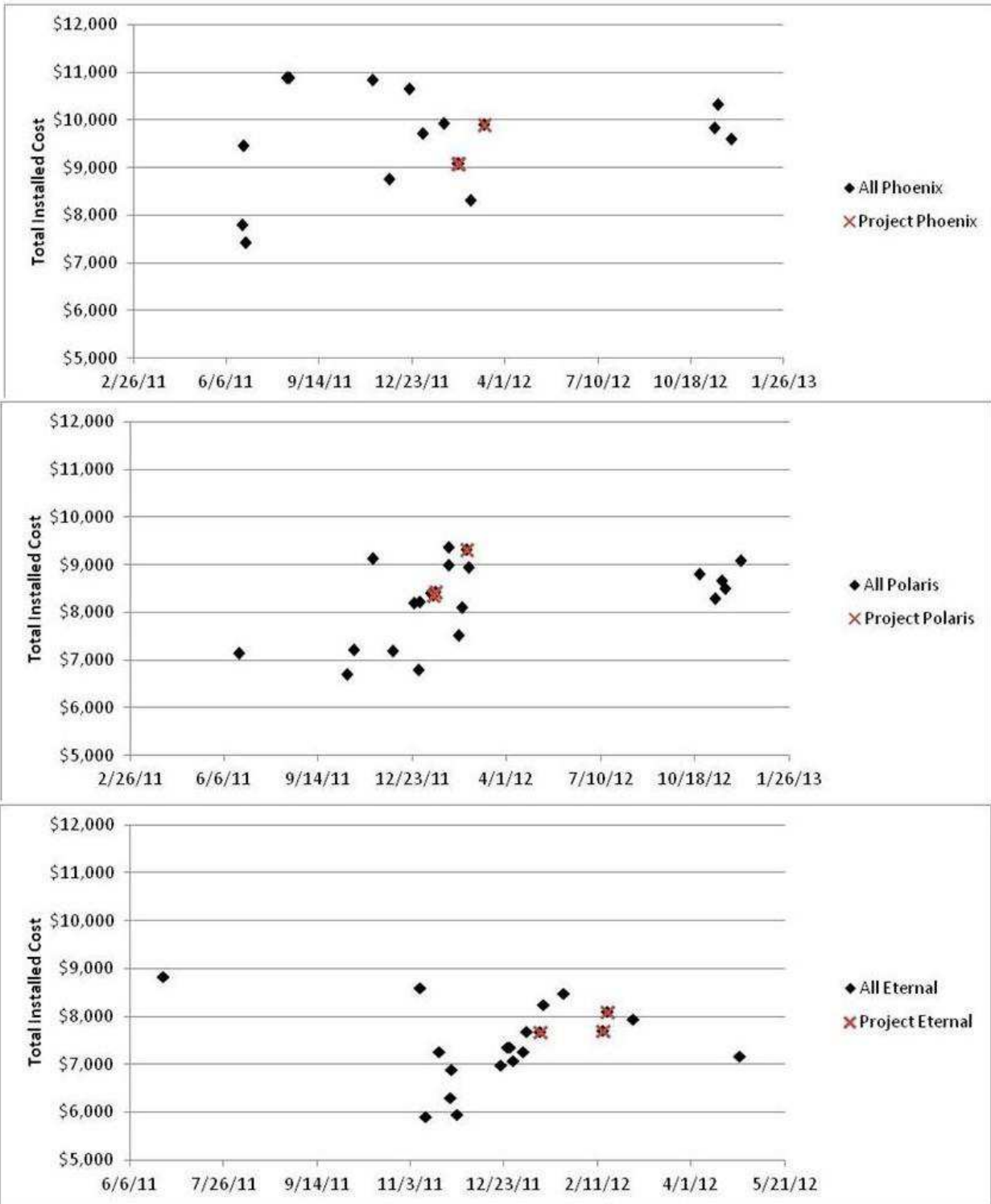
An economic analysis determined how combi systems can increase energy efficiency and the effectiveness of energy efficiency programs. We analyzed installed cost, the savings to investment ratio (SIR), and simple paybacks.

Installation Costs. Detailed costs were recorded for 70 installations using a variety of heating plants and AHUs. Several factors influence the combi systems' installed costs:

- The installations were part of a large project.
- Jobs were sent out to bid separately with multiple contractors bidding each install.
- The number of contractors changed over the course of the project.
- The pressure on contractors for low bids changed over time.
- Contractors installed the same system several time and became more familiar with the equipment.
- Several of the factors that influenced the installation costs for these systems were specific to this project.

Figure 23 shows how the costs of systems changed over the course of the project, initial bids tended to be lower, and bids near the end of the project were typically higher in the price range. Figure 23 shows the installed costs for the three heating plants with the most installations. Other systems did not have sufficient installations to see cost trends. The figure shows the bids for two

contractors over time. In general, initial low bids were likely due to contractors' desire to be awarded jobs. The contractors knew of the potential for a large number of jobs allowing them to reduce their typical profit per job. Over time, only two contractors continued to bid on jobs. The contractors became more familiar with the costs of the installs and what bids were winning jobs. Additionally, at the onset of the combi installation project reducing costs was of high concern. However, the goal for the total number of installations was reduced toward the end of the project, which increased the weatherization agencies' allowable cost per job. Contractors were not informed of this directly, but pressure from the weatherization agency to keep costs down decreased and higher priced bids started to be accepted.



Note: Figures show installed costs for similar heating plants installed as part of the 200 home implementation project ("All") and the detailed monitoring project ("Project").

Figure 23. Cost changes over time for the most installed systems (Phoenix, Polaris, and Eternal)

Table 11 shows the average costs by system type. There were differences in how the installers broke down prices into different categories. It was common for several jobs with similar equipment to have similar total costs, but there could be big differences in the breakdown between heating plant cost, labor costs, and miscellaneous materials costs. The total costs were known to be accurate and were used for the cost analysis. The tankless and hybrid systems were generally less expensive to install and the storage tanks were the most expensive. In general, total installed cost scaled with the equipment costs.

Table 11. Average Installation Costs From Implementation Project

System	n	Heating Plant (\$)	AHU (\$)	Misc (\$)	Labor (\$)	Cost (\$)		
						Average	Min	Max
Storage—Vertex	3	3,223	2,345	1,280	1,943	8,792	8,120	9,505
Storage—Polaris	21	4,175	2,186	476	1,369	8,207	6,495	9,375
Storage—Phoenix	15	4,397	2,081	650	2,479	9,607	7,135	11,000
TWH—Rinnai	4	2,521	2,421	1,025	1,616	7,584	6,945	8,445
Hybrid—Eternal	22	2,815	2,323	590	1,562	7,289	5,696	8,751
Boiler—Navien	5	4,129	2,375	495	1,124	8,123	6,650	8,840

Installed costs of combi systems were compared to other systems types (see Figure 23). Several different costs were considered for this comparison. Low income weatherization agencies in Minnesota have installed a large number of power vented WHs and condensing furnaces. The large volume of installations has resulted in a price break for these agencies (see Table 12). Local contractors were contacted for their typical pricing structures for open market installations. These costs were used for the analysis in this project. The least expensive space and water heating option would be an ND furnace and an ND SWH. This inexpensive option is still common in some northern climate states, and was useful for replacement comparisons. Typical costs for this base system are around \$3,075 (\$1,875 for the furnace and \$1,200 for the WH). A common system upgrade consisted of a condensing gas furnace (with an average AFUE of 94%) and power vent WH (EF of 0.60). This was the system typically installed by Minnesota low income weatherization programs. On the open market this system costs \$6,350 to install (\$4,250 for a condensing furnace and \$2,100 for the WH), which includes local utility rebates. Local weatherization paid, on average, \$3,500 for a condensing furnace and \$1,800 for the WH. A homeowner seeking high efficiency space and water heating could install a condensing furnace and condensing SWH (thermal efficiencies of 95%) or condensing TWH (EF of 0.90 to 95%). Open market installation costs for a condensing SWH were found to be \$5,300 and previous research (Schoenbauer et al. 2011) showed condensing TWHs with installed costs of \$4,000. Condensing combi systems were the lowest cost method of providing both condensing water heating and space heating (Table 13). Costs of installing a condensing furnace and power vent WH are lower, but only provide water heating with a 0.60 EF rating. Contractors also had little experience with combi systems and their costs will likely change over time if the numbers of installations were increased. Combi system equipment can currently have three different ratings (EF, AFUE, and thermal efficiency). Confusion surrounding multiple ratings has prevented local utilities from rebating combi systems, which ultimately make them a less desirable option. Condensing furnaces, power vent WHs, TWHs, and condensing SWHs are more mature technologies with better defined costs.

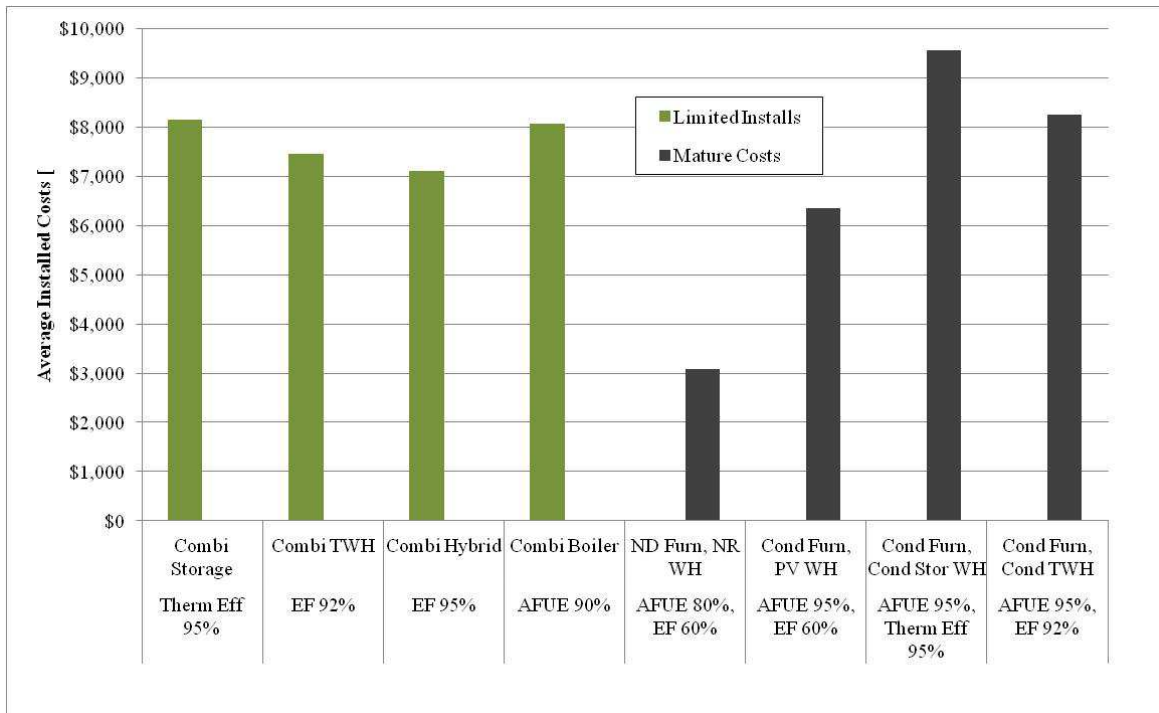


Figure 24. Installation costs for residential HVAC systems

Table 12. Installation Costs for Furnaces and WHs

	Program or High Volume (\$)	Open Market (\$)	BEoptE+ v1.01 (\$)
Condensing Furnace	3,500	4,250	3,318
Minimum Efficiency Furnace	–	1,875	1,408
Photovoltaic WH	1,800	2,100	970
Condensing SWH	5,100*	5,350	–
Condensing TWH	–	–	1,800

* Price was based on the installed cost of a photovoltaic WH and the increased cost of a condensing SWH.

Table 13. SIR for Combi Systems

Site Number	Heating Plant	AHU	Cost (\$)	Space and WH Savings (\$/yr)	Lifetime (years)	Wx Program SIR	Install Cost Needed for SIR = 1 (\$)
1028	Polaris	Rinnai 75	8,410	321	15	0.6	4,820
1031	Polaris	Enerzone 70	8,840	9 ^a	15	0.2	1,365
1027	Polaris—Stress	Enerzone 70	9,760	181	15	0.3	2,712
1056	Phoenix	Enerzone 70	9,535	138	15	0.2	2,076
1061	Phoenix	Enerzone 70	9,535	221	15	0.3	3,315
1026	Vertex	Enerzone 100	9,000	79	15	0.1	1,184
1030	Rinnai TWH	Enerzone 70	7,605	218	15	0.4	3,266
1045	Rinnai TWH	Enerzone 70	6,760	85	15	0.2	1,280
1062	Rinnai TWH	Enerzone 70	7,850	134	15	0.3	2,006
1049	Eternal	Enerzone 70	8,500	203	15	0.4	3,047
1070	Eternal	Enerzone 70	8,155	103	15	0.2	1,549
1037	Navien	Enerzone 70	9,200	11	15	0.0	163
1052	Rinnai Boiler	Enerzone 70	12,500	153	15	0.2	2,297
1063	Rinnai Boiler	Enerzone 70	12,507	119	15	0.1	1,785
Condensing Furnace (AFUE = 95%)^b			3,500	105	20	0.6	2,098

^a Existing system was a condensing furnace and an ND WH.

^b Looking at the space heating SIR of a condensing furnace under average projected load for comparison.

Simple Paybacks. Payback calculations were used to determine the number of years the combi systems would take to pay off the incremental cost of the update. These calculations assumed that equipment needed to be replaced (which was not the case for the SIR). For these simple paybacks the savings are compared to the incremental cost for the system, because a new system was necessary. The equipment could be replaced with a baseline system non-condensing furnace and ND SWH (\$3,075 install cost), a condensing furnace and power vent SWH (\$6,350 install cost), a condensing furnace and condensing WH (\$8,250 install cost with a TWH or \$9,550 install cost with a SWH), or a condensing WH-based combi system (\$7,300 to \$8,000 average install cost). Installed performance was not available for all options. The energy consumption of combi systems, non-condensing furnaces, and condensing SWHs were determined from the field data collected for this project. The energy consumption of condensing TWHs, power vent WHs, and ND SWHs were characterized in a previous field monitoring project (Schoenbauer et al. 2011). Condensing furnace performance was assumed to match the performance of the space heating combi systems, an assumption that was supported by the data collected for a condensing furnace in this project (Site 1031). Assuming the space heating performance of a combi system matched the installed performance of a condensing furnace; the remaining benefit would be from water heating savings. The final column of Table 14 addressed this scenario with comparison to condensing furnaces. The paybacks versus a condensing furnace system are long because the only savings come from DHW, where the load was small. The installation costs of combination systems were lower than the cost of installing a condensing furnace and a condensing WH in most cases making the payback immediate.

Future research is underway to improve to installation costs and performance of the systems. Section 6 discussed the planned continuation of combi system work. Research will include control optimization to improve system efficiency and/or increase space heating capacity. For

example, properly configured outdoor resets could increase space heating capacity using the same hydronic coil with no change or improved efficiency. Improving capacity may allow for smaller systems to be used for more installations, reducing the equipment costs.

Table 14. Simple Payback for Combi Systems

Combination System	Load	Average Install Cost (\$)	Simple Payback	
			AFUE = 80%, EF = 60%	AFUE = 95%, EF = 60%
Storage WH	Low	8200	47	185
	High	8800	23	136
TWH	Low	7600	44	28
	High	7600	21	20
Hybrid WH	Low	7300	30	23
	High	7300	13	11

Notes:

AFUE 80%, EF 0.6: Refers to an ND furnace and WH.

AFUE 95%, EF 0.6: Refers to a condensing furnace and power vent WH.

Listed install costs are total costs, but the incremental cost is used to calculate the payback.

Boiler systems had a limited number of installations over a range of costs and were excluded from this analysis.

4.4 Calculated Space Heating Loads

Careful consideration was taken to size the combi systems for each home. System parameters were set to minimize return water temperature while meeting or exceeding the required design OAT load and 115°F supply air temperature. We used two methods to determine the design load for space heating prior to each installation. Contractors were asked to perform a Manual J calculation and weatherization staff used house data to model each home in the NEAT software.

The contractor's design condition space heating load calculations were compared to the load for the home using the measured data and temperature to determine the actual load at design conditions (Figure 25 and

Table 15). For each site, both contractors calculated a higher design load than the measured design load. For two of 19 sites Contractor A's computed load was within 20% of the measured load (10%–20% is a typical oversize factor used by the industry). Contractor B had three sites within 20%. In contrast, the NEAT models overpredicted the design heating load at 17 of the 19 sites and were within a 20% oversize factor at two other sites (Figure 26).

Table 16 shows the percent difference of space heating load calculations and the measured design load of each home. The results indicate that the design heating loads were typically significantly overestimated. The average Contractor A heating load estimate was more than 100% oversized. Personnel for contractor B were a little closer, about 80% oversized on average. The NEAT modeling software was the closest to actual loads, overpredicting by 51% on average. The NEAT software required several hours of data collection and entry to determine the estimated load, while the contractor's personnel spent significantly less time conducting their Manual J calculations; therefore, it was expected that the NEAT software would be more accurate. Finally, the agencies do not use a comparison of modeled and utility bill annual gas use to modify NEAT model parameters. Applying that process could further improve NEAT design load estimates.

Table 15. Space Heating Load Comparisons

Site Number	Contractor A (Btu/h)	Contractor B (Btu/h)	NEAT (Btu/h)	Measured Data (Btu/h)	Uncertainty (Btu/h)
1062	46,103	40,476	41,966	16,104	1,688
1056	52,368	36,440	26,112	17,797	1,705
1065	51,893	44,820	35,136	18,504	3,245
1078	40,400	58,753	40,320	19,415	2,165
1061	54,359	41,916	37,368	19,785	1,748
1049	—	46,154	36,277	20,466	5,438
1030	41,106	35,004	32,064	21,167	1,145
1037	40,723	36,948	28,534	21,699	2,657
1055	43,502	42,466	30,729	23,549	1,555
1027	53,058	41,736	38,774	24,422	1,002
1054	55,909	53,828	40,101	24,498	1,961
1070	55,140	41,097	29,111	26,092	1,695
1026	58,014	45,524	46,248	26,621	1,002
1052	61,305	45,819	33,474	26,690	1,042
1028	46,984	42,343	35,716	27,280	2,061
1047	55,917	52,961	42,636	30,855	4,990
1031	38,541	38,900	38,260	34,815	1,450
1063	56,420	36,800	32,640	35,609	3,391
1045	42,955	47,693	31,461	40,709	2,891

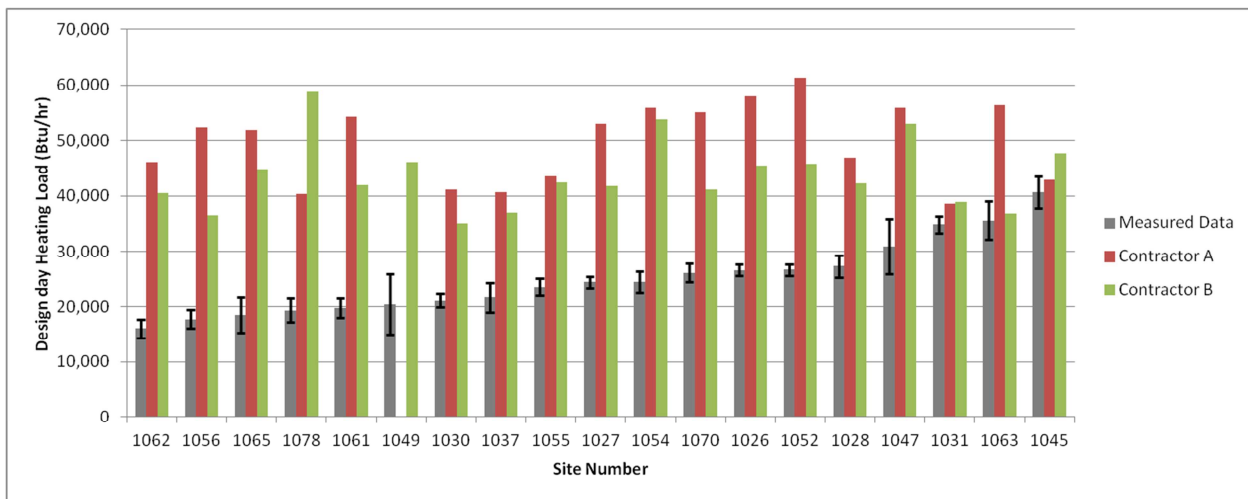


Figure 25. Comparison of the measured space heating load and the contractors estimates

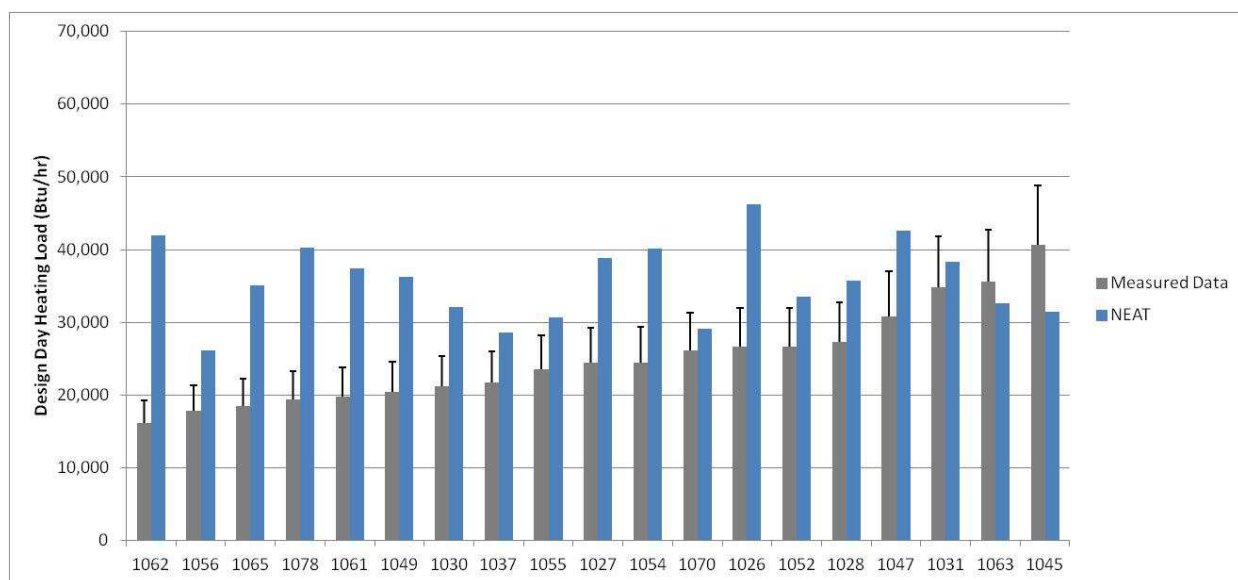


Figure 26. Comparison of the measured space heating load and the NEAT software

Table 16. Overprediction of Space Heating Loads

Site Number	Percentage Difference From Measured		
	Contractor A (%)	Contractor B (%)	NEAT (%)
1062	186	151	161
1056	194	105	47
1065	180	142	90
1078	108	203	108
1061	175	112	89
1049	—	126	77
1030	94	65	51
1037	88	70	31
1055	85	80	30
1027	117	71	59
1054	128	120	64
1070	111	58	12
1026	118	71	74
1052	130	72	25
1028	72	55	31
1047	81	72	38
1031	11	12	10
1063	58	3	-8
1045	6	17	-23
Mean	108	84	51
Median	110	72	47
1 st Quartile	82	61	28
3 rd Quartile	126	108	69

4.5 Occupant Comfort

4.5.1 *Delivered Air Temperature*

A common concern with space heating systems has been occupants feeling “cold blow,” discomfort from moving air that, while warmer than the air in the room, feels cold when blowing directly on a person. For new systems cold blow can be avoided by delivering higher air temperatures and avoiding air blowing directly on occupants. Because ductwork and grilles were not retrofitted in this project, we could not change the location or type of supply registers to minimize the occurrence of air blowing directly on occupants. Instead, we focused on achieving higher supply air temperatures by setting a steady-state delivered air temperature guideline of 115°F. Each of the four system types had slightly different delay times to achieve steady-state temperatures performance.

An alternative solution to the “cold blow” discomfort is a control strategy of delaying the blower initiation after the call for heat to allow the temperature to ramp up before airflow begins. This control strategy is common in forced air applications, but was not available for most hydronic air handling units.

SWHs. Almost half of the SWH-based combi systems achieved a median supply air temperature of 115°F. The average supply air temperatures of combi systems using storage-type WHs was 109°–120°F (Figure 27 and Table 17). The median supply air temperature was 112°F or greater for all of the systems and 115°F or greater for three of the seven systems. For storage-based systems it was more difficult to set the supply air temperature due to the variability in the stored water temperature. The SWHs had different sized dead bands for temperature control, resulting in different ranges in the minimum/maximum stored water temperatures and delivered air temperatures. On a call for heat from the thermostat, hot water immediately circulated through the hydronic coil. The AHU fan turned on at the same time as the circulation pump and the delivered air temperature raised as the coil came up to temperature (Figure 28). The delivered air temperature reached 110°F around 45 seconds on average.

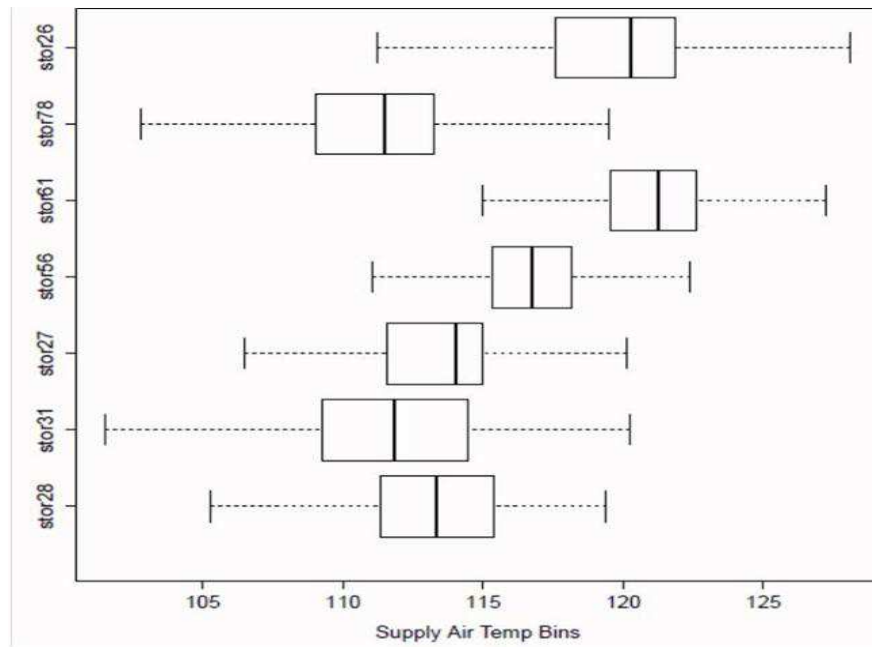


Figure 27. Supply air temperature distributions for SWH-based combi systems

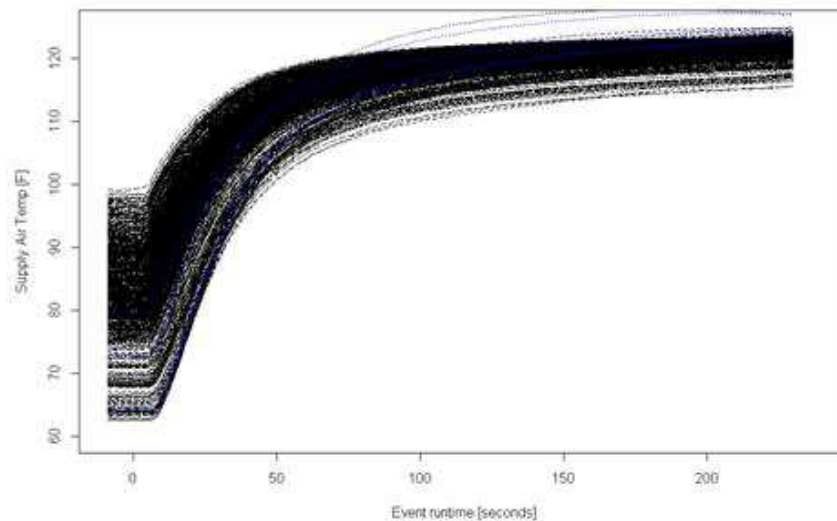


Figure 28. Delivered air temperature of each space heating event at Site 1061 with a SWH-based combi system

TWHs. Combi systems using TWHs had average delivered air temperatures of 113°–118°F and medians of 115°–119°F (Figure 29). TWHs control the steady-state outlet temperature to a small range of allowable temperatures, making the outlet water temperature more constant. The fairly constant water temperature made it possible to more accurately set the supply air temperature to the desired value. As a result, all three systems had a median supply air temperature at or slightly above the desired value of 115°F. Since the TWHs had no storage capacity and heated water as necessary when space heat was called for, cold water circulated through the hydronic coil until

the WH came up to temperature. This control strategy resulted in the TWH-based combi system taking a relatively long time of about 100 seconds to reach 110°F (Figure 30).

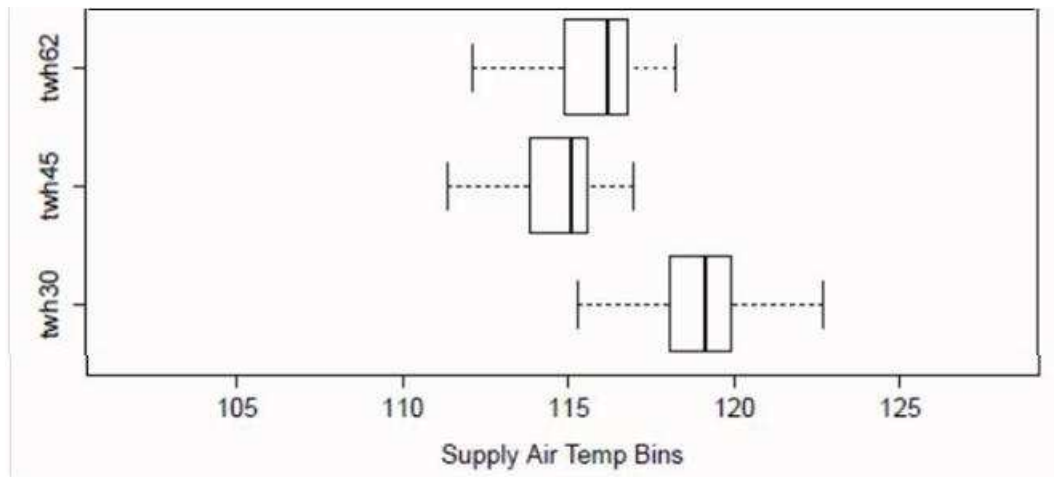


Figure 29. Supply air temperature distributions for TWH-based combi systems

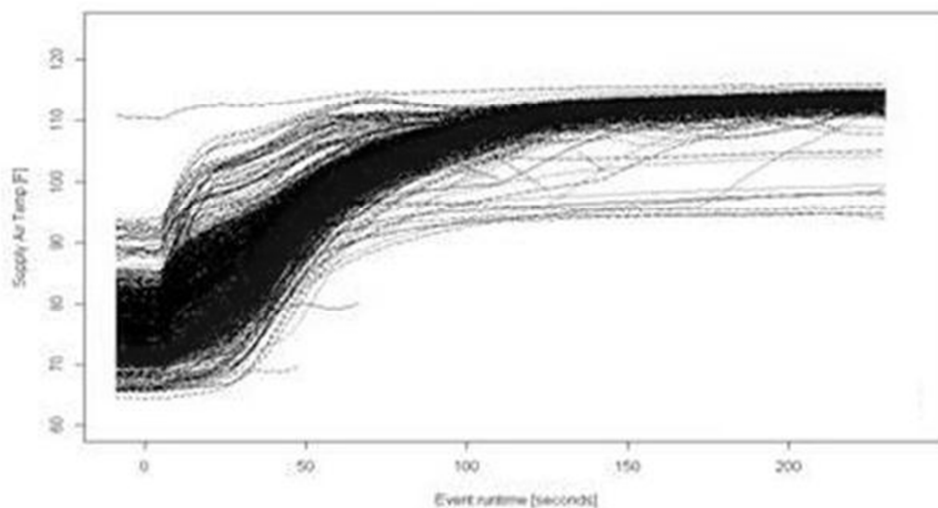


Figure 30. Delivered air temperature of each space heating event at Site 1045 with a TWH-based combi system

Hybrid WHs. Combi systems that used hybrid WHs had average delivered air temperatures of 115°–117°F with median air temperatures of 116°F and 119°F (see Figure 31). Like tankless units, the hybrid WHs had a tight range of delivered water temperatures, allowing steady-state air temperatures to be set up easily. These units had 2 gal of storage, and this quantity of hot water helped to bridge the gap between a call for space heating and the burner reaching steady-state. On average hybrids delivered 110°F air 50 seconds after a call for heat.

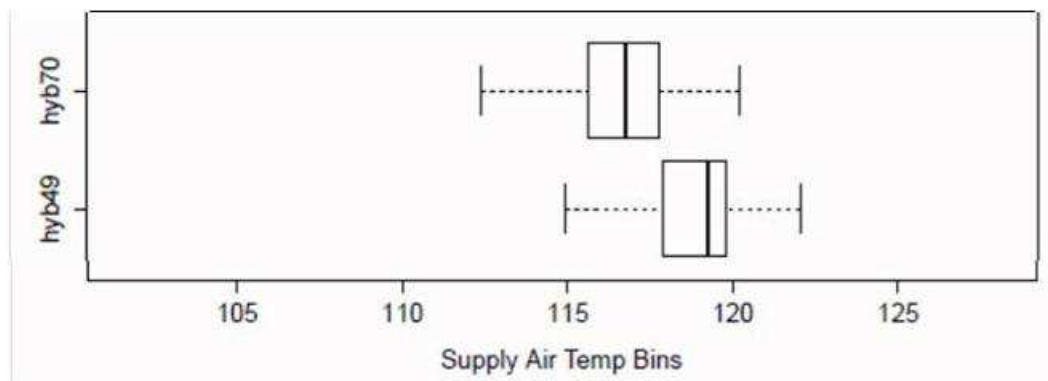


Figure 31. Supply air temperature distributions for hybrid WH-based combi systems

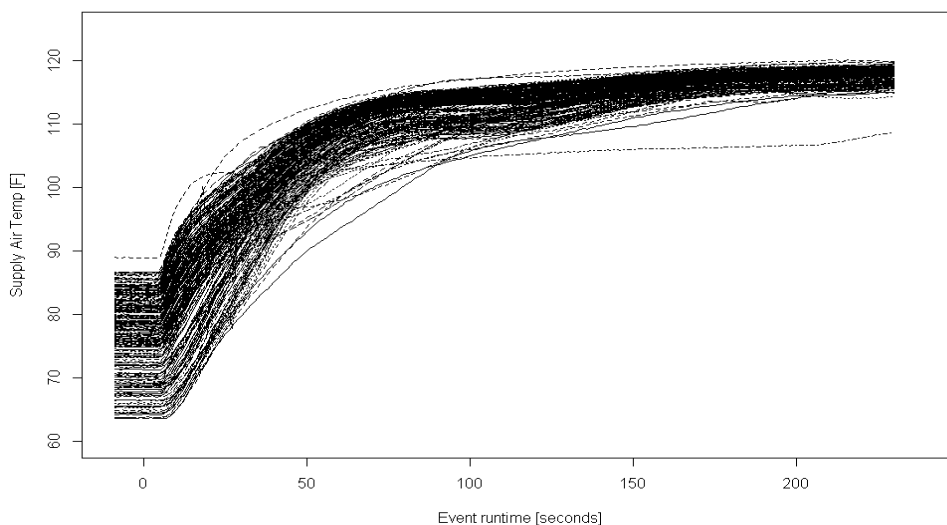


Figure 32. Delivered air temperature of each space heating event at Site 1049 with a hybrid WH-based combi system

Boilers. Most boiler-based combi systems, both low mass systems and those with storage capacity, had delivered air performance similar to that of other combi units, 116°–120°F average temperature (Figure 33). However, one boiler (a Rinnai at Site 1063) used an OAT reset control that reduced the delivered water temperature as the OAT increased. This resulted in the delivered air temperature being reduced in the shoulder seasons, which resulted in a larger inter-quartile range than for other systems. Low mass boilers have no storage capacity, and these performed similar to a TWH. These systems took around 110 seconds to reach 110°F (Figure 34). A boiler with storage capacity (Site 1063) performed similar to storage-based system. Air temperatures reached 110°F around 40 seconds into the event (Figure 35).

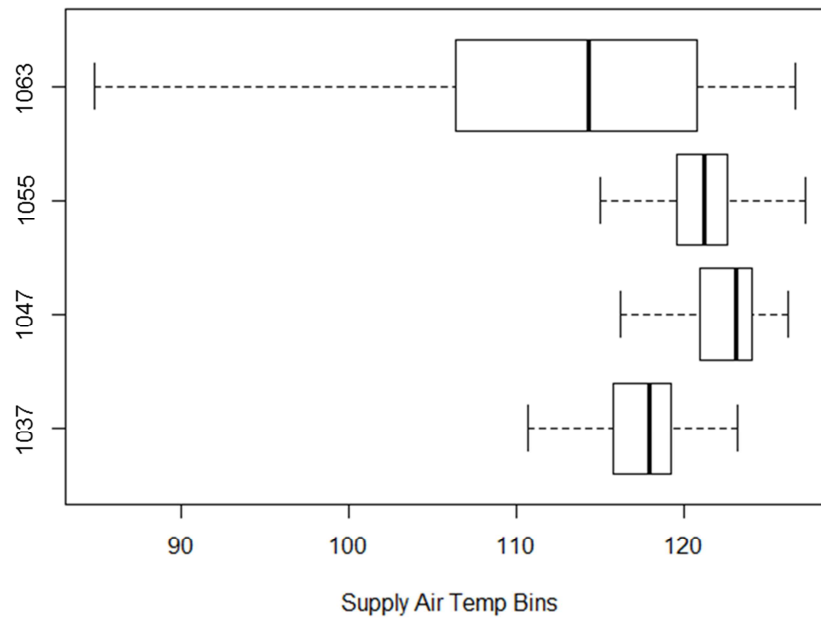


Figure 33. Supply air temperature distributions for boiler-based combi systems

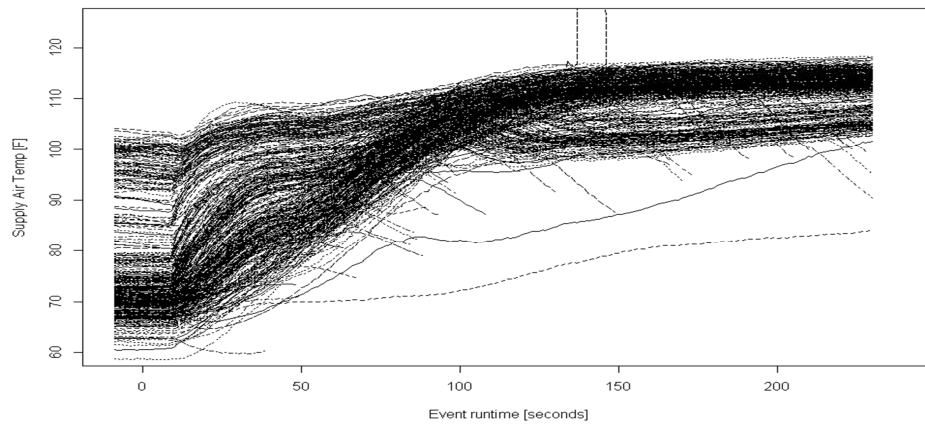


Figure 34. Delivered air temperature of each space heating event at Site 1037 with a low mass boiler-based combi system

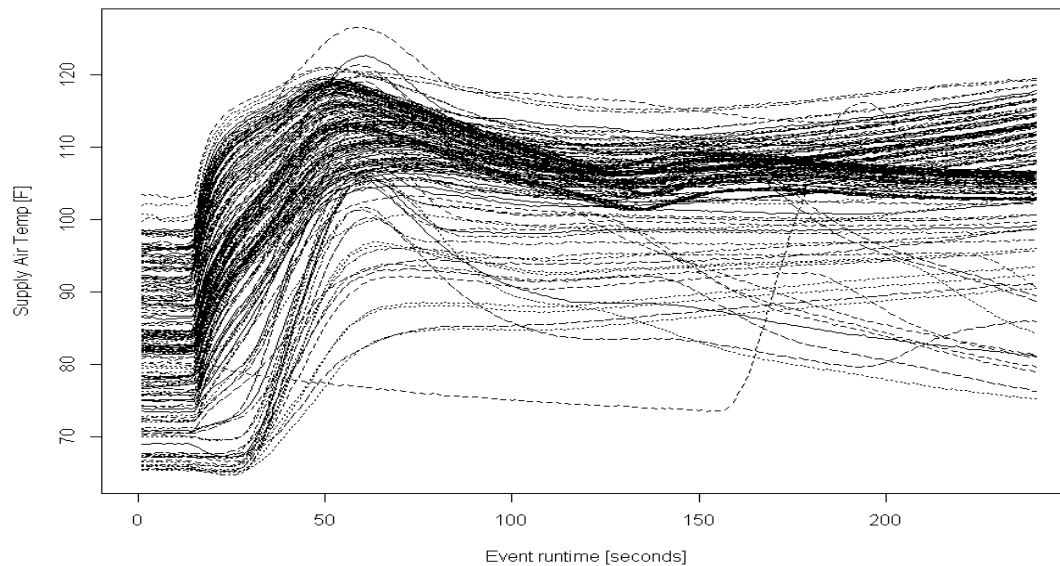


Figure 35. Delivered air temperature of each space heating event at Site 1063 with a boiler-based combi system

Table 17. Supply Air Temperature Distribution Characteristics

Site Number	Combination Appliance Heating Plant	Supply Air Temperature			
		1 st Quartile (°F)	Median (°F)	Mean (°F)	3 rd Quartile (°F)
1028	Polaris	111	113	113	115
1031	Polaris	109	112	111	114
1027	Polaris	112	114	111	115
1056	Phoenix	115	117	116	118
1061	Phoenix	120	121	120	123
1078	Phoenix	109	112	109	113
1026	Vertex	118	120	119	122
1030	Rinnai TWH	118	119	118	120
1045	Rinnai TWH	114	115	113	116
1062	Rinnai TWH	115	116	114	117
1054	Eternal	118	119	117	120
1070	Eternal	116	117	115	118
1037	Navien	116	118	116	119
1047	Navien	121	123	121	124
1055	Navien	120	120	120	123
1052	Rinnai Boiler	106	114	111	121

Comparisons Between Combi Systems and Other Space Heating Systems. Delivered air temperatures for combi systems were compared to those for other system types. The existing furnaces that were ND non-condensing furnaces had a preset time delay between burner fire and the fan turning on. Figure 36 shows that these furnaces (time equals 0 when the fan turns on) compared to the combination systems. The existing furnaces reached 110°F about 15–20 seconds

after the fan turned on. The existing non-condensing ND furnaces also had higher steady-state delivered air temperatures. The furnaces monitored in this study had air temperatures of 125°–150°F. Other projects have shown that condensing gas furnaces typically deliver air temperature of 115°–130°F. Heat pump systems have air temperatures of 77°–115°F (Natural Resources Canada 2009). These systems have had significant numbers of installations without occupant comfort complaints.

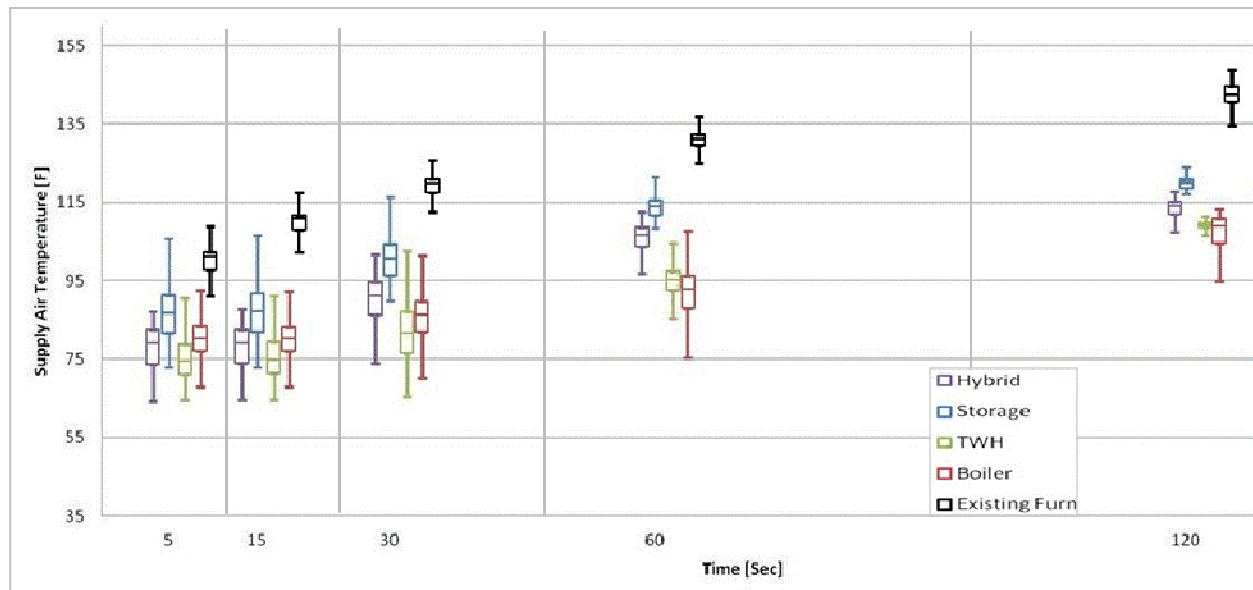


Figure 36. Time series discharge air temperatures for combi systems

The 115°F supply air design temperature was considered to achieve good occupant comfort without modifications to ductwork or delivery grilles. Combi systems have air temperatures similar to other high efficiency space heating systems (e.g., condensing furnaces and heat pumps). The combi system with outdoor reset (Site 1063) and large numbers of heat pump systems have provided sufficient capacity and comfort at temperatures below 115°F, with longer runtimes at lower temperatures and flow rates while avoiding blowing directly on people. Delivered air temperatures of combi systems could potentially be lowered. The air temperature would be lowered by reducing the heating plant water temperature thus lowering return water temperature and improving efficiency.

4.5.2 Hot Water Delivery Capabilities

There were differences in the capabilities of each combi system to provide hot water. Three areas were analyzed to identify the origins of these differences. The first and most significant was the ramp up time, the time between hot water being turned on and hot water leaving the system. The second was the range of the steady-state water temperatures. The third was the percentage of time that water delivered was usable (temperature greater than 105°F). The delivered water temperature of each system was measured approximately 6 in. downstream of the mixing valve. Each system was initially set to deliver 120°F water from the mixing valve, but occupants were allowed to adjust the system to meet their needs. This allowance for adjustment meant that not all systems were set for the same steady-state water temperature.

SWHs. Figure 37 shows the water temperatures exiting the mixing valve for a combi system with a condensing SWH. Within about 5 seconds of the draw starting, the water leaving the mixing valve reached its steady-state temperature—in this case about 125°F. The control mechanism for SWHs allowed the stored water to cool before firing to reheat. This resulted in a range (about 10°–15°F) of steady-state delivered water temperatures. Excluding site 1028, for SWHs 91% of the time when there was a hot water draw the delivered temperature was greater than 105°F, the temperature water is typically assumed to become usable (Figure 38). The Phoenix WH (site 1078, 1056, and 1061) had greater variance in delivered water temperature than the other two heaters. This was likely due to a difference in (1) the reduction of temperature variance allowed in the tank before reheat; (2) temperature measurement technique that allows greater variance in temperature; and (3) stratification in the tank or a combination of these effects. The homeowner at site 1028 preferred much cooler water and had lowered the water delivery temperature. At this site only 31% of the time was the temperature greater than 105°F. The average delivered water temperatures for the SWHs varied from 105°F to 135°F.

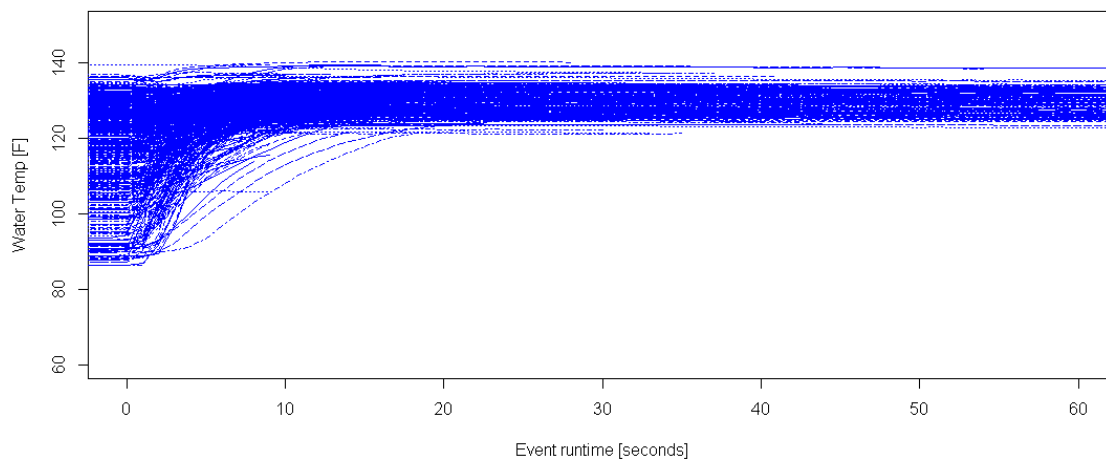


Figure 37. Time series plots of DHW temperatures after mixing valve of the SWH at Site 1078

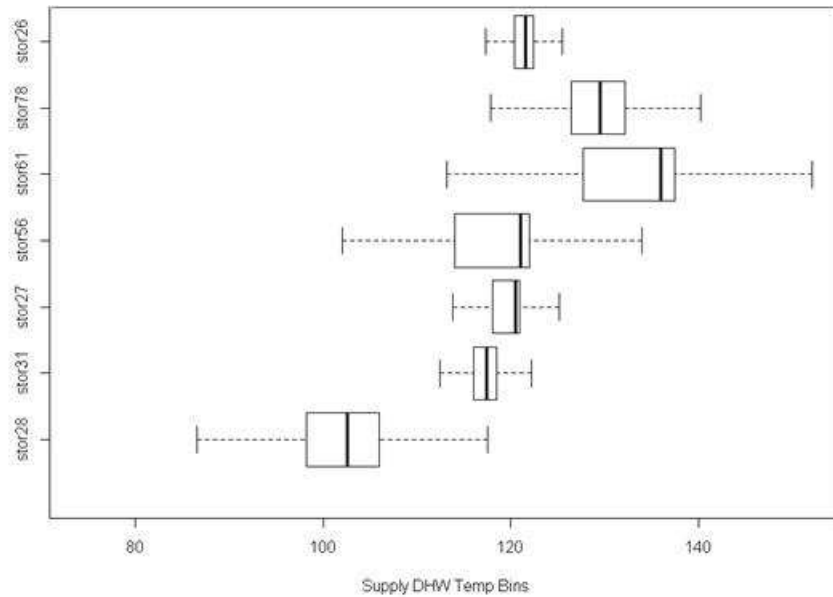


Figure 38. Delivered water temperatures for the storage-based combi systems

TWHs. TWHs initiate heating when a hot water draw is sensed (Figure 39). Not surprisingly, these systems had slower ramp-up times, typically taking 20–30 seconds longer to reach the desired temperature than the systems using SWHs. The delay time increased at lower flow rates and the figure shows some draws that took longer than 60 seconds to reach steady-state temperature. In addition, the unit would not produce hot water when flow rates fell below the minimum fire rate of the TWH (~0.5 gpm). Variance in a site’s delivered water temperature increased with low water flow rates, long idle periods between events, and large seasonality in inlet water. At site 1030, low water flow rates caused increased delay time, making WH ramp-up time a larger percentage of the total runtime and increased the temperature variance. However, once the system reached steady-state, the continuous burner modulation allowed the WH to provide a very consistent temperature over a range of flow rates. At steady-state temperature, variance was typically less than 2°F (Figure 40). The occupants using the TWH systems kept the settings close to 120°F (Figure 40). Outlet water temperatures were greater than 105°F 84% of the time hot water was being used, 8% lower than percentage of time for the SWHs.

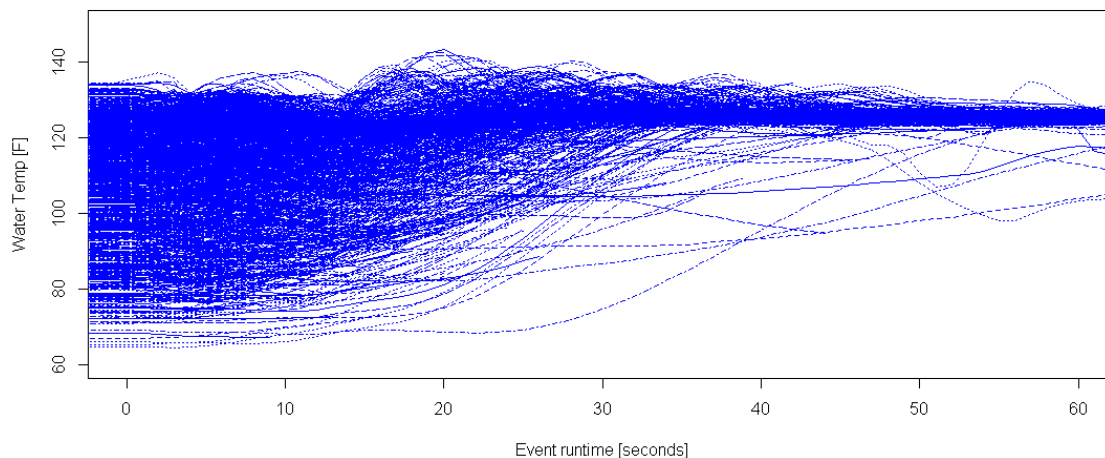


Figure 39. Time series plots of DHW temperatures after mixing valve of the TWH at Site 1030

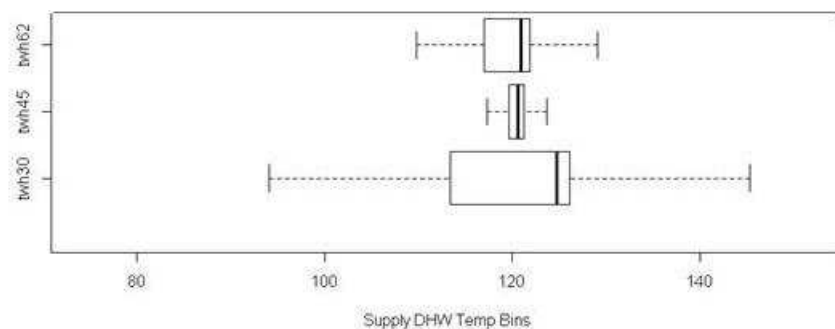


Figure 40. Delivered water temperatures for the TWH-based combi systems

Hybrid WHs. We found a clear benefit for added storage volume of the combi system using a hybrid WH (Figure 41). The 2-gal storage volume provided usable hot water while the heater came up to temperature. The water temperature leaving the mixing valve reached the steady-state temperature about 5–10 seconds after the start of a water draw. This was comparable to a system using a SWH and an improvement over systems using a TWH. The hybrid system also provided hot water for low flow draws below 0.5 gpm, but delivered water temperatures were reduced as the draw continued until the stored water temperature dropped enough to require the tank to reheat (Figure 42). There were several events where the heater generated hot water 10–15 seconds into the event and then cooled before heating up again. The modulating burner allowed the hybrid unit to provide very consistent steady-state water temperatures (within 2°F). Temperatures were greater than 105°F on average 84% of the active time (Figure 42). None of these sites changed water delivery set point from 120°F design.

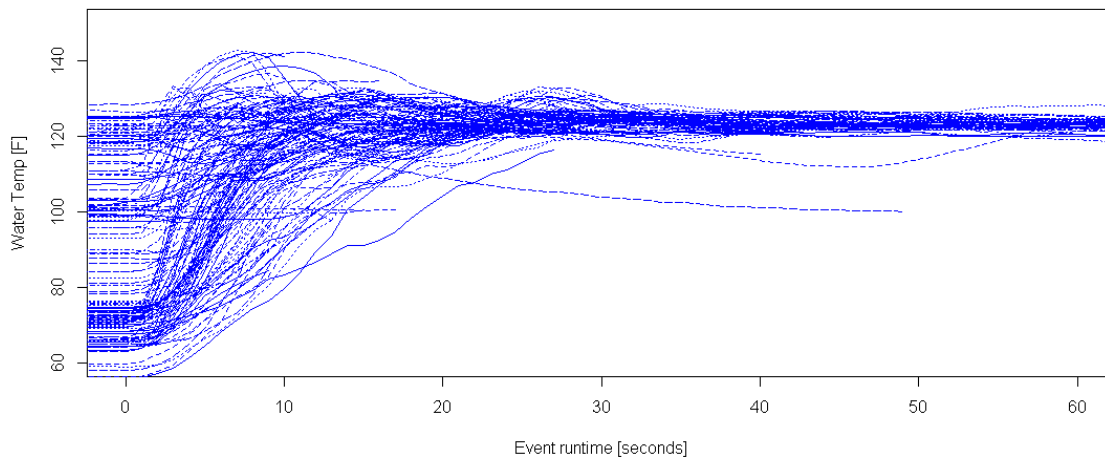


Figure 41. Time series plots of DHW temperatures after mixing valve of the hybrid WH at Site 1070

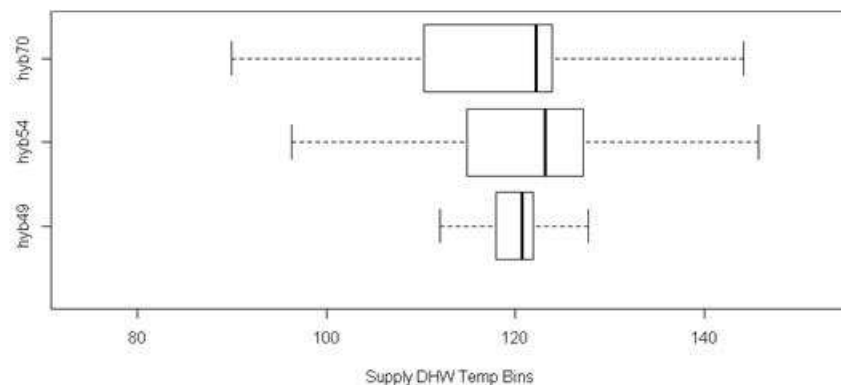


Figure 42. Delivered water temperatures for the hybrid WH-based combi systems

Boilers. The mechanism for setting the delivered water temperature was different for the boiler combi systems than for those based on WHs. Whereas each of the WHs was set at 140°F to provide enough heat for space heating and mixed with cold water for hot water use, each boiler had an internal mixing valve that was set to the desired DHW temperature.

Combi systems based on low mass boilers (Sites 1037, 1047, and 1055) did not have any storage capacity and created hot water as needed. The systems used a digital set point control and internal mixing valve. Testing of the unit prior to installation left some concerns about the unit's ability to control the DHW temperature; therefore, the same mixing valves were installed for the other combi systems were also installed on the boiler combi systems. The water delivery performance of a low mass combi boiler was similar to that of a TWH (Figure 43). These boilers had the slowest ramp up time of any DHW system tested taking 20–40 seconds to reach temperature. The low mass boilers had steady-state water temperatures that fluctuated 5°–10°F often up to 2 minutes into events.

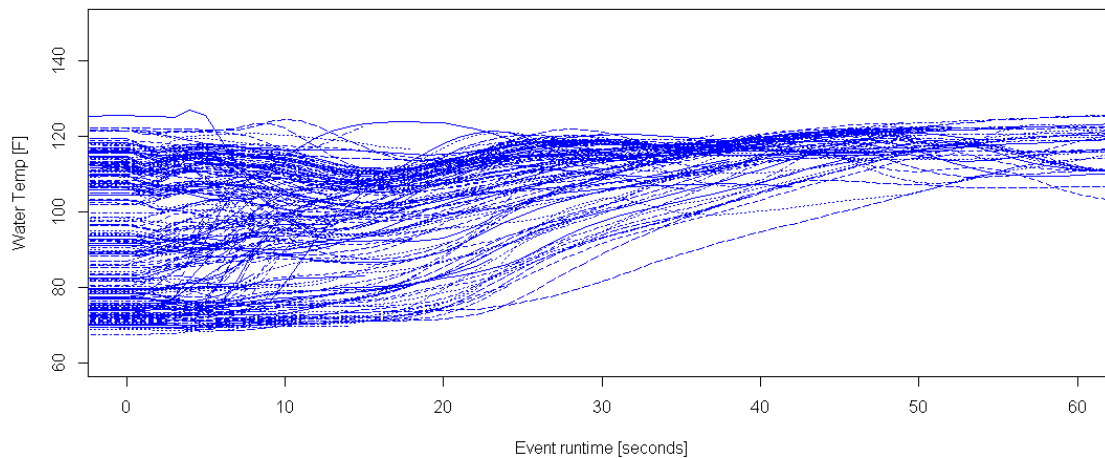


Figure 43. Time series plots of DHW temperatures after mixing valve of the low mass boiler at Site 1055

The boilers installed at sites 1052 and 1063 had internal storage tanks with an approximately 12-gal capacity. The storage temperature of these tanks was fixed by the manufacturer, and an internal mixing valve was used to reduce the DHW temperature. Both sites had consistent water delivery (interquartile ranges less than 5°F). Average temperatures were 128°F at each site, which was greater than 120°F design, meaning that occupants increased the water temperature set point. Twelve gal of storage capacity yielded DHW performance similar to an SWH with 5- to 10-second ramp-up time (Figure 44). The combi boilers with internal storage had a range of steady-state temperatures of 10°F.

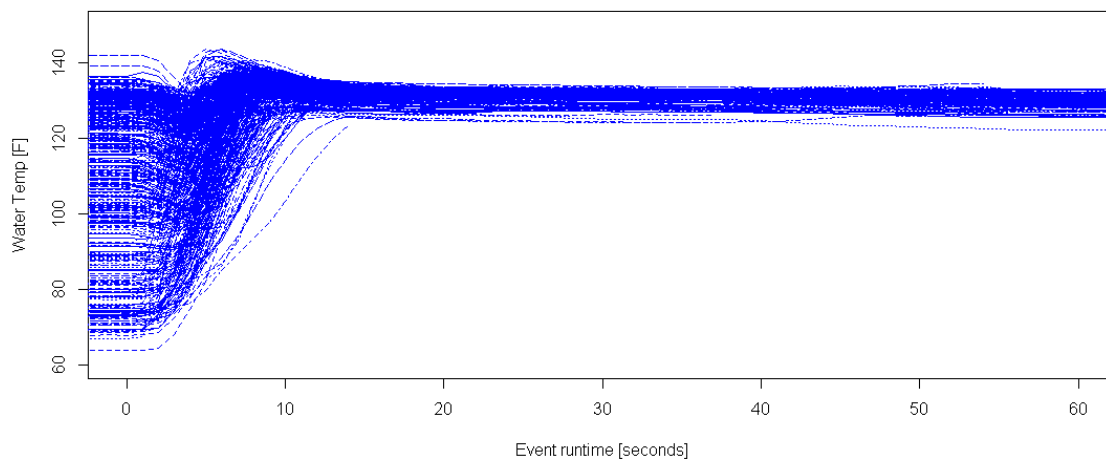


Figure 44. Time series plots of DHW temperatures after mixing valve of the boiler at Site 1063

The average delivered water temperatures show a significant difference between the low mass boilers and boilers with internal storage. The low mass boilers had an average delivered water temperature of 109°F (Figure 45) with a steady-state temperature of 120°F. The boilers with

storage capacity had an average delivered water temperature of 128°F, with a steady-state temperature of 130°F (both occupants increased the set point temperature). The quick ramp-up time with storage capacity made the average temperature much closer to the steady-state temperature.

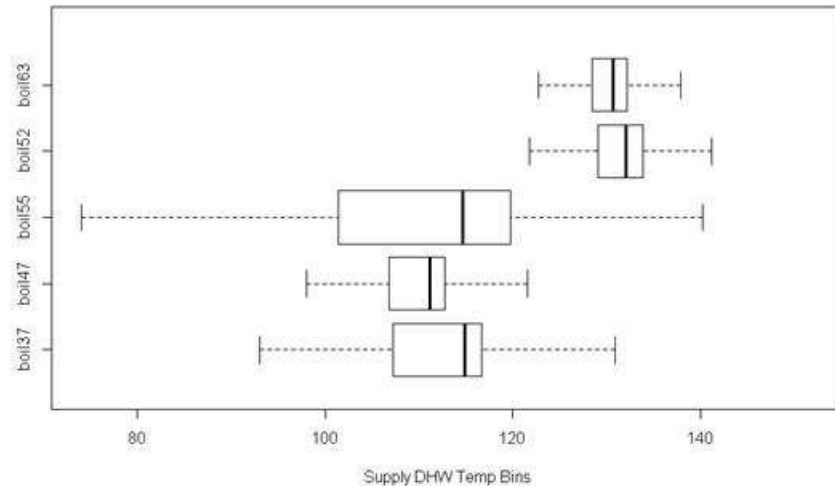


Figure 45. Delivered water temperatures for the boiler-based combi systems measured 6 inches after the mixing valve

For comparison, Figure 46 shows the delivery capability of a standard natural-draft SWH, a system that was monitored in a previous study (Schoenbauer, Hewett, and Bohac 2011). The ramp-up time was about 2 seconds. This installation did not require a mixing valve, reducing the ramp-up temperatures by a couple of seconds. The ramp-up time for this system was very similar to that of the storage-based combi system. This system has a wide range (about 30°F) of steady-state water temperatures. This range of temperatures was much greater than any of the combi systems.

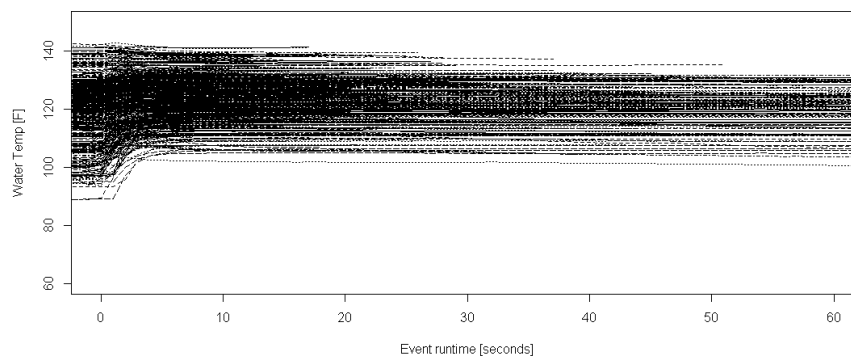


Figure 46. Time series plots for DHW temperatures at outlet of WH for an ND storage tank WH

5 Conclusions

The field research conducted for this project has demonstrated that forced air natural gas combination systems can be used to provide high efficiency space and water heating for residential retrofits in a cold climate. These systems had acceptable occupant comfort and DHW capabilities. Combi systems provided on average a 19% natural gas savings over ND forced air furnaces and ND SWHs (AFUE ~80% and EF ~58%). Combination systems provided annual space heating at 85%–92% efficiency and improved domestic water heating efficiency at every site. Summer DHW efficiencies ranged from 35%–90% depending on water usage and system type. Several homes in the study had very low usage resulting in storage tanks operating almost totally in standby mode.

Despite good savings and occupant comfort there were two hurdles to widespread implementation discovered. The technology has not matured enough to be cost competitive with condensing forced air furnaces that have almost complete penetration in cold climates. The second hurdle was the dependence on a proper installation to ensure good performance. The systems installed for this project were supported by a large laboratory research project. Knowledge from that project was used to inform the installation and setup the operating parameters of these systems. Without careful attention to detail in these areas, systems would not be able to perform to the levels measured in this report.

6 Future Work

Future work will build on the results from this research showing that combi systems utilizing condensing WHs and boilers can provide both space and water heating with efficiencies of 90%. However, combi systems, using condensing equipment, are still relatively new to the market and both end users and contractors are not familiar with their installation and performance. Future work will concentrate on system control options to improve ease of installation and use, reduce on-site design and engineering, improve system efficiency, and increase system capacity.

Monitoring results have highlighted areas for potential improvement. A preliminary list of control measures to evaluate in the laboratory includes temperature set point control (outdoor reset, turn down after heating season), space heating modulation (both water flow and airflow) and DHW priority.

The set point temperature reset varied by outdoor air control method has been widely used for both commercial and residential boilers. This control method has potential benefits for combi systems as well. Past projects have required WHs be kept constant at 140°F to meet design space heating loads. Reducing the set point temperature to 130°F or below in the shoulder heating season and summer would increase the thermal efficiency of all the combi appliances and it would reduce the standby losses of SWHs. Initial laboratory testing showed an increase in efficiency for one condensing SWH from 86% to 93% with a 10°F reduction in the setback temperatures. These effects combine to create estimates 50 therms/year savings for a very simple outdoor reset control. These types of controls can increase performance, save energy, and improve the sizing and design process. Detailed laboratory and field testing is needed to determine the best control options and the impacts on the occupants. Laboratory testing will be used to determine the balance point between efficiency tradeoffs and reductions in runtime that can lead to short cycling.

Controlling air and water flow modulation would also improve combi systems. Current equipment requires air and water flow, which can only be manually modulated and typically remains fixed through the product's lifetime. Full modulation or multiple step modulation would allow the combi to operate at different optimized output rates depending on demand. There are several potential benefits from this approach:

- Lower output rates will increase run times, reducing the impacts of short-cycling, and improving comfort.
- Preliminary laboratory testing indicates that lower output rates may yield lower return water temperatures improving efficiency.
- Multiple firing rates can result in downsizing equipment. Large air coils are required to achieve high efficiency at peak loads. However, if a combi system with a smaller coil were optimized for less than peak load with flow modulation, it could be ramped up to peak load at a slightly lower efficiency. Because most hours of operation for a space heating system occur well below the design load, the system would operate at max efficiency for more hours with a small reduction in efficiency for the few hours at peak design loads.

- This process would also reduce some of the site specific optimization and engineering. Modulation control would allow the combi system to change the output based on the homes need, thus reducing the dependence on optimizing for the specific conditions of the home.

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