U.S. DEPARTMENT OF ENERGY | Energy Efficiency & Renewable Energy

BUILDING TECHNOLOGIES OFFICE

Optimizing Hydronic System Performance in Residential Applications

L. Arena and O. Faakye
Consortium for Advanced Residential Buildings

October 2013

Building America
U.S. Department of Energy

Available electronically at http://www.osti.gov/bridge

Available for a processing fee to U.S. Department of Energy
and its contractors, in paper, from:
U.S. Department of Energy
Office of Scientific and Technical Information
P.O. Box 62
Oak Ridge, TN 37831-0062
phone: 865.576.8401
fax: 865.576.5728
email: mailto:reports@adonis.osti.gov

Available for sale to the public, in paper, from:
U.S. Department of Commerce
National Technical Information Service
5285 Port Royal Road
Springfield, VA 22161
phone: 800.553.6847
fax: 703.605.6900
email: orders@ntis.fedworld.gov
online ordering: http://www.ntis.gov/ordering.htm

Printed on paper containing at least 50% wastepaper, including 20% postconsumer waste

Optimizing Hydronic System Performance in Residential Applications

Prepared for:

The National Renewable Energy Laboratory

On behalf of the U.S. Department of Energy's Building America Program

Office of Energy Efficiency and Renewable Energy

15013 Denver West Parkway

Golden, CO 80401

NREL Contract No. DE-AC36-08GO28308

Prepared by:

L. Arena and O. Faakye

Steven Winter Associates, Inc.

of the

Consortium for Advanced Residential Buildings

61 Washington Street

Norwalk, CT 06854

NREL Technical Monitor: Cheryn Metzger

Prepared under Subcontract No. KNDJ-0-40342-03

October 2013

[This page left blank]

Contents

List of Figures

Unless otherwise noted, all figures were created by CARB.

List of Tables

Unless otherwise noted, all tables were created by CARB.

Definitions

AFUE	Annual Fuel Utilization Efficiency
BA	Building America
BEopt™	Building Energy Optimization software
CARB	Consortium for Advanced Residential Buildings
DHW	Domestic hot water
DOE	U.S. Department of Energy
ECM	Electronically commutated motor
EF	Energy factor
gpm	Gallons per minute
HDD	Heating degree day
SHGC	Solar heat gain coefficient
SWA	Steven Winter Associates, Inc.
W	Watt

Acknowledgments

Steven Winter Associates, Inc. (SWA) acknowledges the U.S. Department of Energy Building America program and its funding and support for the development of this technical report as well as research that informed it. SWA would like to thank John Siegenthaler for his assistance with the design of the systems in Houses #2 and #3 and for his invaluable consulting services throughout this project. SWA would also like to thank all the manufacturers and product suppliers who donated their time, expertise, and equipment: Dave Davis from Heat Transfer Products, Dave Scearce from Peerless, Mark Handzel with Bell & Gossett, Bob Reimund from Grundfos, and Parker Wheat with Emerson Swan. A special thanks to Scott Reynolds with Ithaca Neighborhood Housing Services for his continued support and the use of his homes for this research.

Executive Summary

Even though new homes constructed with hydronic heat comprise only 3% of the market, of the 115 million existing homes in the United States, almost 14 million (11%) are heated with steam or hot water systems, according to 2009 U.S. Census data. Therefore, improvements in hydronic system performance could result in significant energy savings in the United States.

When operating properly, the combination of a gas-fired condensing boiler with baseboard convectors is a viable option for high-efficiency residential space heating in cold climates. Based on previous research efforts, however, it is apparent that these types of systems are typically not designed and installed to achieve maximum efficiency. Furthermore, guidance on proper design and commissioning for heating contractors and energy consultants is hard to find and is not comprehensive.

Through modeling and monitoring, CARB sought to determine the optimal combination(s) of components—pumps, high efficiency heat sources, plumbing configurations and controls—that result in the highest overall efficiency for a hydronic system when baseboard convectors are used as the heat emitter. The impact of variable-speed pumps on energy use and system performance was also investigated along with the effects of various control strategies and the introduction of thermal mass.

Three different natural gas-fired systems were analyzed: (1) a modulating, condensing boiler with an on-demand domestic hot water (DHW) generation; (2) a high-mass condensing water heater with an external brazed plate heat exchanger for supplying space heating; and (3) a modulating, condensing boiler with a standard primary/secondary loop and an indirect DHW tank. The third system listed was also tested with a buffer tank between the heating zones and the boiler to analyze the effects of added mass on cycling and overall system efficiency.

According to the modeling, the condensing boilers installed in the three houses in Ithaca, New York were anticipated to result in source energy savings of 13%–14% compared to the Building America benchmark. Reduced annualized energy costs indicated that these systems would be cost-effective options. However, the modeling software could not adequately simulate the differences between the three systems. In particular, the effects of variable-speed pumps, boost controls, the presence of mass to reduce cycling, and alternative DHW options such as indirect tanks or on-demand DHW heat exchangers could not be modeled. These features were evaluated further using the measured data.

Lessons learned from this study and the resulting recommendations include the following:

- Even though short-term tests revealed combustion efficiencies for these boilers ranged from the upper 80%'s to upper 90%'s, long-term data show that standby losses to unconditioned spaces may be as large as 20%–30%. These losses are primarily due to standby losses from the water storage tanks, exposed piping and boiler jackets, and cycling losses. It is recommended that installers insulate all exposed supply and return piping and storage tanks located in unconditioned spaces.
- Using thermostat setback and boost controls for recovery resulted in reduced cycling and less energy consumption than systems operated in constant temperature mode. If setback

operation is desired, the systems should be designed to provide the necessary output under design conditions at supply temperatures lower than the boiler's maximum supply temperature. Then, when recovery is needed, the boiler can boost to higher temperatures to meet the added load. Both the boiler and the baseboard need to be able to provide this added output. For modulating, condensing boilers, design the system to meet the design heating load such that the return temperature does not exceed 130°F. If using a condensing water heater, design the system to meet the design heating load such that the tank temperature does not exceed 130°F.

- Inadequate baseboard capacity will result in extended run times, increased standby losses, and an overall decrease in system efficiency. Baseboard should be sized for low temperature operation under design conditions and be able to provide twice that capacity under boost operation to reduce recovery time from thermostat setback.
- High efficiency pumps prove to be cost-effective enhancements and have a simple payback of 4–5 years. Pumps that have displays showing flow rate and energy consumption are particularly useful during commissioning to optimize performance and ensure design conditions are being met.
- If oversized for the heating load, short cycling of the boiler can be reduced by controls that limit the boiler's space heating input, added mass in the system, and the use of thermostat setback.
- Controls that enable a post-purge to the DHW after a space heating call are encouraged to reduce standby losses.

These results and recommendations are intended to benefit heating, ventilation, and air conditioning contractors, installers, energy designers, manufacturers of hydronic equipment, and other researchers looking to optimize hydronic heating system performance in residential applications.

1 Introduction and Background

1.1 Introduction

Condensing boiler technology has been around for many years and has proven to be a durable, reliable method of heating. The modulating capability of some gas boilers makes them an excellent option for low-load homes. The sealed combustion, direct vent arrangement results in reduced risk of exposure to combustion byproducts inside the home and is often a money-saving feature in retrofit applications. Reductions in carbon emissions are also achieved due to the higher combustion efficiency.

The basic components of a modern hydronic space heating system are shown in Figure 1 and include a heat source, a delivery system consisting of piping and circulation pumps, heat emitters, and controls. The water in the system is heated by the heat source and circulated through the piping to the heat emitters, where it is delivered to the space.

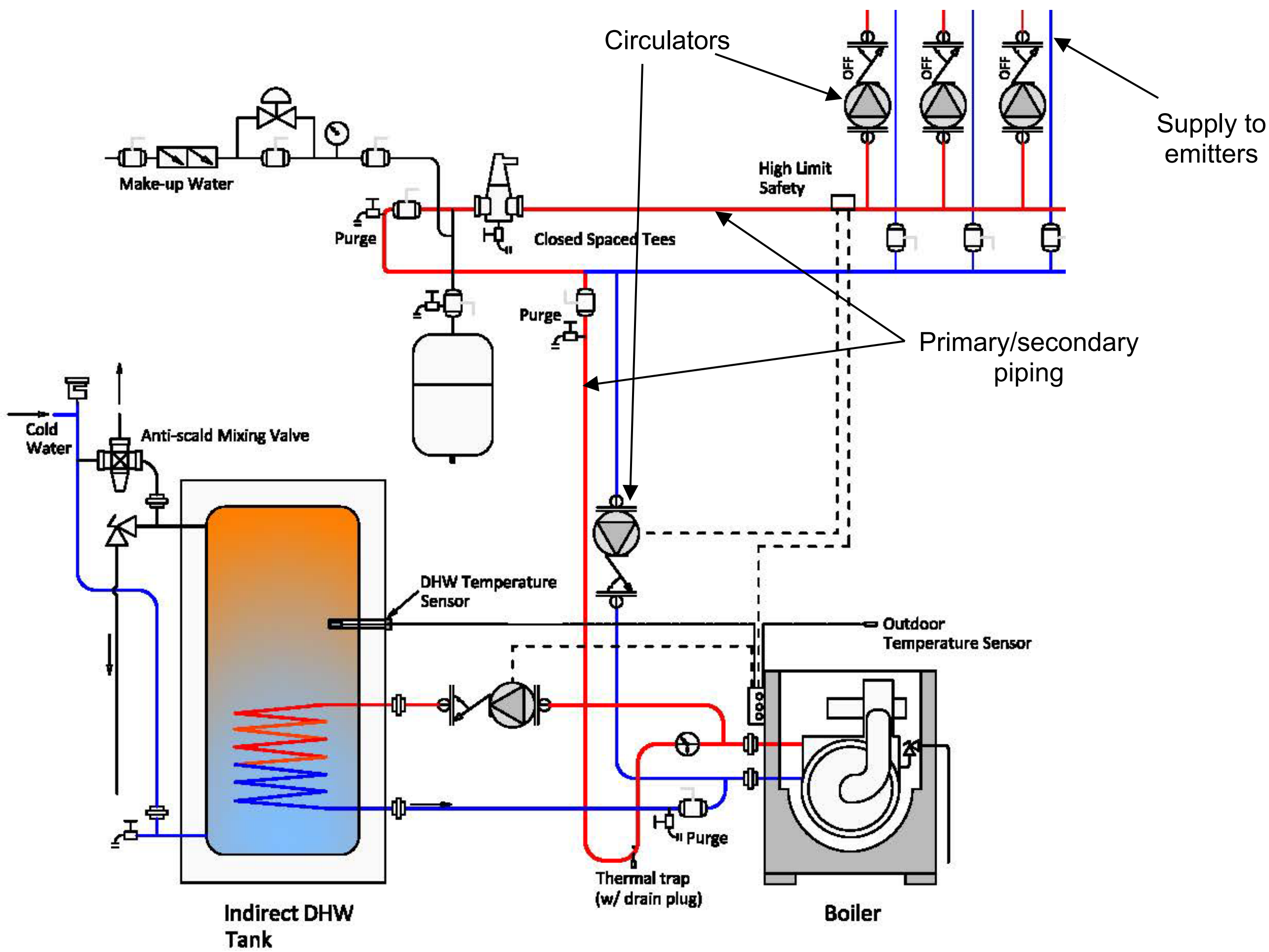


Figure 1. Hydronic heating system composed of a condensing boiler, outdoor reset control, circulation pumps, and indirect domestic hot water (DHW) tank

Table 1 provides a few common examples for each of these basic components. An individual system may employ one or more of each component. For example, the heat source could be a boiler coupled with a solar thermal system that supplies hot water to both baseboards and radiant floors in different parts of the home and may employ all the controls listed. In addition to these items, it is common that a boiler will supply the DHW.

Table 1. Examples of the Various Components in a Hydronic Heating System

Component	Examples
Heat Sources	Boiler, water heater, solar thermal system, heat pump
Piping Configurations	Primary/secondary loop, oversized headers, home run configuration, hydraulic separators
Circulation Pumps	Constant speed, constant pressure, proportional pressure
Heat Emitters	Baseboard convectors, radiant panels, radiant flooring, radiators
Controls	High temperature limits, outdoor reset curves, differential settings, thermostats

Overall system efficiency will be affected by the efficiency and interactions of each component chosen. While the boiler may come with an efficiency rating in the upper 90s, when pump energy, pipe and boiler jacket losses, and boiler settings are factored in, the overall system efficiency could be significantly lower.

Based on previous research efforts, it is apparent that hydronic systems are typically not designed and installed to achieve maximum efficiency, especially when paired with baseboard convectors: one of the most cost-effective emitters for hydronic systems (Arena 2010). Part of the reason for these low efficiencies is that little information is available on combined system efficiencies where all the components in the hydronic system are considered. Over the past several years, researchers have conducted several studies into the efficiency of hydronic heating. Such work includes laboratory testing on boiler efficiencies (Butcher 2009), control strategies (Butcher 2004) and system interactions (Butcher 2006, 2011), as well as laboratory and field testing on hydro air systems that provide both the space heating and DHW (Rudd et al. 2011; Butcher 2011; CEE; Schoenbauer et al. 2012). However, there are no known field studies that assess the combined effects of several high efficiency components as listed in Table 1. The goals of this research were to assess several combinations of these components and make recommendations for cost-effective, responsive, energy-efficient packages.

1.2 Background

In an attempt to optimize system performance and to create guidelines for the industry, Steven Winter Associates, Inc. (SWA)—the lead for the U.S. Department of Energy's (DOE) Building America (BA) team CARB (Consortium for Advanced Residential Buildings)—has been monitoring and evaluating condensing boiler systems since 2008. Over the past several years, SWA has teamed with BA, the New York State Energy Research and Development Authority, Ithaca Neighborhood Housing Services, and Brookhaven National Laboratory to evaluate the installed performance of condensing boilers using baseboard convectors for the heat emitter.

There were several critical findings from SWA's previous research (Arena 2010). First, in order to protect their equipment in the field, manufacturers of low-mass, condensing boilers typically recommend design strategies and components that ensure steady, high flow rates through the heat exchangers. These strategies ultimately result in a decrease in condensing frequency and overall efficiency. The requirement that a primary loop be designed into the system, along with the high flow rates of the pumps recommended in the installation manuals, result in return temperatures that are higher than optimal for best efficiency.

It was also found that there is a significant lack of information for contractors on how to configure these systems to optimize overall efficiency. For example, there is little advice on selecting the best settings for the boiler reset curve or how to measure and set flow rates in the system to ensure that the return temperatures are low enough to promote condensing. It was also observed that recovery from setback was extremely slow in all homes evaluated in the previous research effort and was often not achieved. Recovery was affected by the outdoor reset control, the differential setting on the boiler and over-sizing of the boiler itself. Again, little mention of this issue was found in boiler manuals or hydronic design guides.

Another significant finding from past research is that the factory settings for the boiler's reset curve are usually not adjusted by the installer when, in fact, this is one of the keys to achieving high efficiency. A reset curve adjusts the boiler's supply temperature based on the outdoor temperature to more closely match the boiler's output to the heat load. For example, Figure 2 shows a typical factory default boiler reset curve.

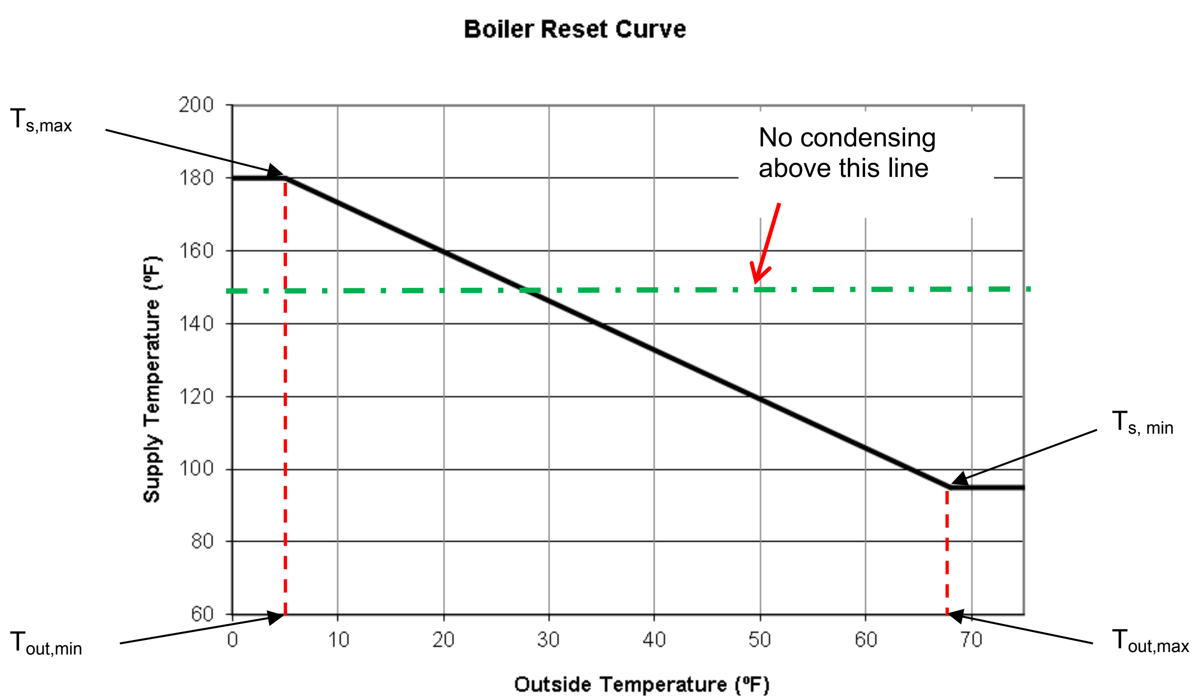


Figure 2. Outdoor reset curve @ 180°F maximum output temperature

Note that for gas fired boilers, most condensing occurs when the return temperature to the boiler is 130°F or lower (Butcher 2004). If the system was designed for a 20°F temperature difference between the boiler supply and return, and the factory settings were the same as those mentioned above, the return temperature could be above 130°F for a good portion of the heating season, depending on the climate. In fact, with a 20°F design difference, any supply temperature above 150°F would result in return temperatures above 130°F. And, as the loads decrease with increasing outdoor temperatures, the difference between the supply and return temperatures also decreases. So, if the system was designed with a 20°F delta at 180°F, at 150°F the difference may only be 10°F, resulting in reduced condensing performance.

Table 2 demonstrates the impact of flow rates and various settings for the reset curve on the frequency of condensing, which is the percent of the heating season the return temperature is predicted to be below 130°F. Data shown are valid for a Munchkin Contender and are based on bin temperature data for Ithaca, New York.

Table 2. Predicted Frequency of Condensing for a Boiler in Ithaca, New York

	Frequency of Condensing at Different $T_{s,max}$ at 1, 2, and 3 gpm											
$T_{s,min}$	150			160			170			180		
95	99%	91%	87%	90%	80%	77%	79%	68%	64%	66%	57%	53%
105	99%	87%	83%	86%	72%	67%	71%	58%	54%	56%	47%	44%
110	99%	84%	79%	82%	66%	60%	62%	50%	45%	48%	41%	39%
115	98%	80%	73%	72%	56%	50%	48%	42%	35%	40%	34%	32%
120	97%	70%	60%	66%	45%	40%	43%	34%	24%	32%	25%	23%

$T_{s,max}$: Maximum boiler supply temperature at the outdoor temperature used to size the boiler
$T_{s,min}$: Minimum boiler supply temperature that will be delivered at the maximum outdoor operating temperature

Findings indicate that the maximum boiler supply temperature ($T_{s,max}$) is typically not adjusted by the installer and is left at the factory setting of 180°F. Also, in the systems analyzed in CARB's previous studies, flow rates through the boiler were typically found to be at least 3 gpm or higher. At these settings, the frequency of condensing for the boilers in Ithaca (i.e. periods when return temperatures are below 130°F) would be approximately 53%. This means that the boiler would be operating in condensing mode only 53% of the time, and the rest of time it would operate as a standard boiler.

1.3 Research Questions

The findings outlined in the previous section were presented to industry leaders at a BA experts meeting in October of 2010. Several boiler, pump, and baseboard manufacturers attended along with leading hydronic designers. Several manufacturers proposed solutions that would likely decrease the costs for installation and increase the overall efficiency of these systems resulting in increased savings and cost effectiveness for homeowners in both new construction and retrofit applications. Based on feedback and input from these industry partners, the following research questions were proposed:

1. What combination(s) of components—pumps, high efficiency heat sources, plumbing configurations and controls—will result in the highest overall efficiency for a hydronic system where baseboard convectors are used as the heat emitter?

2. If variable-speed pumps that sense system pressure are used, can the primary loop be eliminated without endangering the heat exchanger? If so, what are the implications for energy use?
3. What is the tradeoff in efficiency associated with using a boost control to decrease recovery times following setback versus maintaining a constant temperature in the home? Which would result in lower overall energy use?
4. What is the value of thermal mass in high efficiency boiler systems?

To answer these questions, three different heating systems were monitored. Each system was set up to optimize specific heating system parameters: boiler supply and return temperatures, flow rate, and temperature control configurations. Modeling results have been compared to field data collected from October 2012 to February 2013.

2 Modeling

All three systems were modeled in BEopt™ (Building Energy Optimization), an hourly simulation tool developed by the National Renewable Energy Laboratory for DOE's BA program. The software was developed to evaluate residential building designs and identify cost-optimal efficiency packages at various levels of whole-house energy savings along the path to zero net energy. BEopt can be used to analyze both new construction and existing home retrofits through evaluation of single building designs, parametric sweeps, and cost-based optimizations.

All homes analyzed were compared to a standard reference home of the same size and built with the same construction methods. Simulation assumptions for the reference home are based on the BA House Simulation Protocols (Hendron and Engebrecht 2010).

The following sections explain the results of the simulations, discuss the predicted source energy savings and annualized energy costs of the test homes compared to the BA benchmark, and explain the limitations encountered with the software.

2.1 Source Energy Savings Compared to the Building America Benchmark

Although quite similar to each other, each test home was modeled in BEopt and compared to BA's benchmark. Specifications for all three test homes have been listed in Table 3. House #1 is a 1,083-ft^2 single-family detached house with a design heating load of approximately 16,000 Btu/h. Houses #2 and #3 are each one side of the same duplex. Design heating loads are 14,695 Btu/h and 15,426 Btu/h, respectively. Conditioned floor area is similar to that of House #1: 1,140 ft^2 for House #2 and 1,236 ft^2 for House #3.

Table 3. Test Home Specifications

Component	House #1	House #2	House #3	BA Benchmark
Type	Single-family detached	Duplex	Duplex	Single-family detached
Conditioned Area	1,083 ft^2	1,140 ft^2	1,236 ft^2	Same as the test home
Volume	8,664 ft^3	9,120 ft^3	9,888 ft^3	Same as the test home
Beds/Baths	2 beds/1.5 baths	2 beds/1.5 bath	3 beds/1.5 bath	Same as the test home
Foundation	Unconditioned basement	Same	Same	Same as the test home
Foundation Insulation	R-7.5 exterior rigid	Same	Same	R-15 exterior rigid
Wall Insulation	R-21 blown cellulose	Same	Same	R-13 batts 2 × 4 16 in. o.c. + R-5 foam
Windows	Low-e, argon, wood (U-0.25/ 0.23 solar heat gain coefficient (SHGC)	Same	Same	Low-e, vinyl (U-0.35/0.35 SHGC)
Ceiling	R-54 blown cellulose	Same	Same	R-49 blown cellulose
Infiltration*	< 3 air changes per hour @50 Pascals	Same	Same	7 air changes per hour @50 Pascals
Mechanical Ventilation	exhaust only, Panasonic WhisperGreen bath fan	Same	Same	Exhaust only
Space Cooling	None	None	None	SEER 13 (11.09 energy efficiency ratio)
Appliances	ENERGY STAR® dishwasher and refrigerator	Same	Same	ENERGY STAR dishwasher and refrigerator
Lights	100% fluorescent	Same	Same	100% fluorescent

*Based on previous test results from this builder.

While all three of these homes have similar efficiency levels, they differ from the BA benchmark in several ways. The benchmark building has a better insulated basement in comparison to these homes, but the benchmark home has a higher infiltration rate, less efficient space and DHW systems, less efficient lightening, lower levels of insulation in the ceiling, and less efficient windows. Table 4 summarizes the whole-house source energy savings compared to the benchmark home. These savings include differences in envelope efficiencies as well as mechanical system efficiencies.

Table 4. Source Energy Savings Compared to the BA Benchmark

Test House	Source Energy Savings (%/yr)
#1	20.4
#2	21.6
#3	21.0

2.2 Cost of Each Hydronic System

A cost comparison of these systems is provided in Table 5. Total costs were calculated from information provided by the heating contractor, supply houses, manufacturers, and from defaults in BEopt.

Table 5. Cost Comparison for Each System Analyzed

Component	House #1	House #2	House #3A[1]	House #3B	Benchmark
Heat Source	BaxiLuna condensing boiler 92.5 annual fuel utilization efficiency (AFUE[3]) $3,000	VersaHydro condensing water heater 96% thermal efficiency $4,700	Peerless Pinnacle PF50 condensing boiler 95.1 AFUE $2,800	Peerless Pinnacle PF50 condensing boiler 95.1 AFUE $2,800	Non-condensing boiler 80 AFUE $2,000
DHW[2]	On-demand DHW heat exchanger 0.83 energy factor (EF)	0.87 EF	Indirect tank 0.86 EF $1,240	Indirect tank 0.86 EF $1,240	Standard gas tank 0.62 EF $350
Buffer Tank	None	None	$625	None	None
Primary Loop (Plumbing and Labor)	$1,260	None	$1,260	$1,260	$1,260
Pumps	2 B&G 3-speed pumps $280	1 Grundfos Alpha VSP $150	4 B&G Vario VSP $600	4 B&G Vario VSP $600	4 small 3-speed pumps $336
Emitter	Under floor cross-linked polyethylene radiant system $1,625	42 ft HeatingEdge 2-pipe baseboard $1,680	44 ft HeatingEdge 2-pipe baseboard $1,760	44 ft HeatingEdge 2-pipe baseboard $1,760	44 ft high output, single pipe baseboard $352
Total	$6,165	$6,530	$8,285	$7,660	$4,448

[1]System3A uses a buffer tank in lieu of a standard primary loop. 3B uses a standard primary loop.
[2]Costs for DHW for House #1 & #2 were assumed to be 10% of total system costs.
[3]ANSI/ASHRAE Standard 103-2007 (2007).

This table shows that the system for House #1 is the least expensive: condensing boiler with an on-demand DHW heat exchanger and under floor heating. The system in House #2—the condensing water heater with high output baseboard—is approximately $370 more expensive than that of House #1. The systems in House #3 are $1,500 more for the setup with the primary/secondary loop and $2,100 more for the configuration using the buffer tank. Significant cost savings in House #1 and House #2 resulted from the elimination of extra components such as the indirect DHW tank and the buffer tank, the primary/secondary loop piping, and the elimination of several pumps.

2.3 Optimization

Using BEopt's optimization function, various hydronic space heating and DHW systems were evaluated and plotted along a cost-effective optimal curve. Optimization mode sequentially searches the available building options for the lowest cost designs at various levels of energy savings. Figure 3 shows how each system in this study (green dots) compares to the optimal curve.

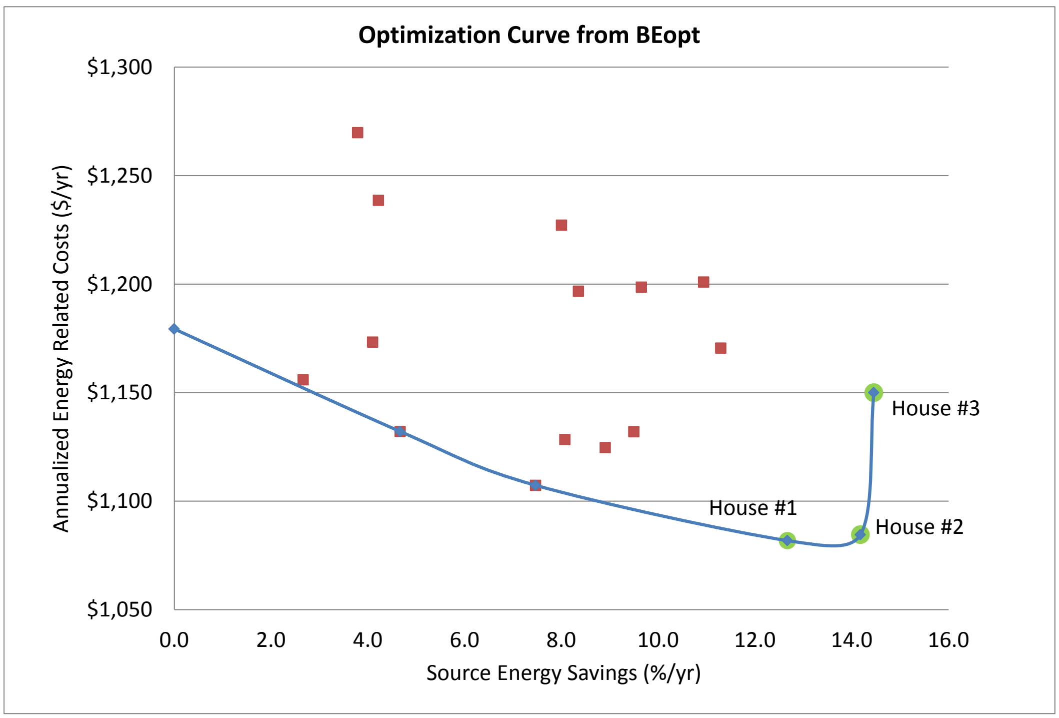


Figure 3. Optimization curve from BEopt for various combinations of hydronic heat and DHW systems

The cost effective optimal curve is the lowest boundary of all the points plotted on the graph with percent source energy savings along the x-axis and the annualized energy costs on the y-axis. This annual cost is made up of the energy costs each year plus the cost of the energy efficiency measures that have been incorporated into the mortgage payments. Annualized energy related costs were calculated over a 30-year analysis period. Cash flows consist of mortgage/loan payments, replacement costs, utility bill payments, mortgage interest rates, and residual values.

For this optimization, the BA benchmark was run for climate zone 6, and only the heating and DHW systems were altered. This presents a clearer comparison of the various mechanical systems than if the test homes were compared to the benchmark. The better envelope efficiency values in the test homes would skew the analysis of the mechanical systems.

All three systems in this study were included in the optimization (see Table 5 for efficiencies used) along with the following options for hydronic space heating and DHW production. The buffer tank could not specifically be modeled.

- 80% AFUE boiler
- 95% AFUE boiler
- 95% AFUE boiler without outdoor reset
- 0.62 EF water heater
- 0.82 EF instantaneous water heater
- 0.96 EF condensing instantaneous heater

With cost values from Table 6, operating costs from the model and cost analysis parameters in Table 7, the annualized energy-related cost was calculated for each condensing boiler with BEopt. Annual energy-related cost represents the equivalent annual cost of the complex cash flow in present dollars.

Table 6. Fuel Prices and Characteristics for Energy and Cost Analysis

Energy Source	Site-to-Source Ratio[1]	Cost[2]
Gas	1.092	$1.3644/therm
Electricity	3.365	$0.18/kWh

[1] Deru and Torcellini (2007)
[2] Prices obtained from (2012a, 2012b). Electricity and natural gas prices represent average national prices for 2011.

Table 7. Cost Analysis Parameters

Parameters	Value
Analysis Period	30 years
Inflation Rate	3%
Discount Rate	3%
Mortgage Period	30 years
Mortgage Interest Rate	7%
Marginal Interest Rate	28%
Fuel Escalation	0%

Table 8 summarizes the results from the optimization for the three study homes. All three systems result in a negative annualized energy cost, indicating that all are cost-effective options compared to the BA benchmark's heating and DHW systems. These values are based solely on differences in mechanical system efficiencies used in the optimization, and do not include building envelope efficiency differences.

Table 8. Summary of BEopt Modeling Results for the Three Condensing Boilers

	Condensing Boiler #1	Condensing Boiler #2	Condensing Boiler #3
Source Energy Saving (%/yr)	12.7%	14.2%	14.5%
Annualized Energy Cost Increase ($/yr)	–$97	–$94	–$29

2.4 Software Limitations

With respect to attached housing, BEopt has currently been upgraded to handle adiabatic wall surfaces, but this is limited to attached walls only; it does not handle common floors or ceilings. The duplex in our study actually has a slightly more complex configuration than can be entered in BEopt and has some adiabatic floor/ceiling surfaces (Figure 4). The second floor of one side of the duplex actually extends over the first floor of the other half.

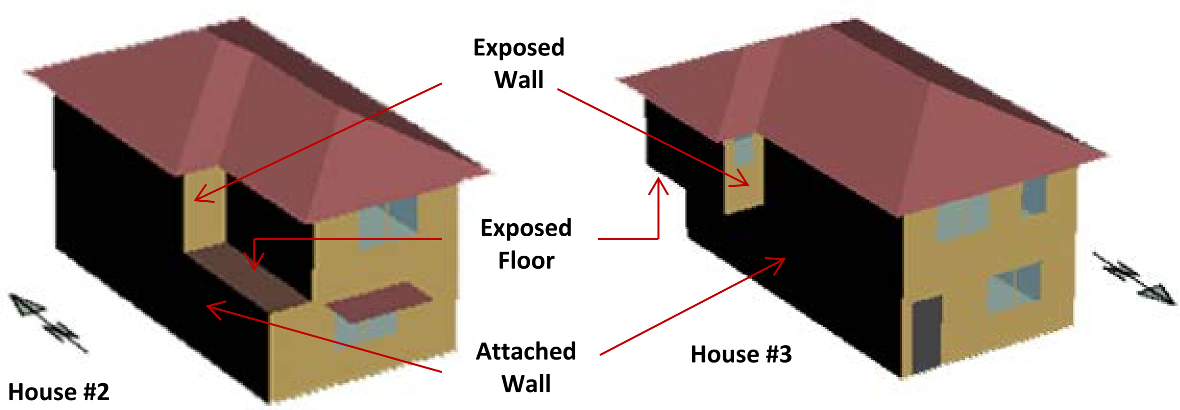


Figure 4. BEopt geometry screen for duplex: Houses #2 and #3

A workaround for this issue was to increase the thermal resistance of the exposed surfaces in the model to simulate the same heat loss that would actually occur through a smaller exposed area.

The hydronic heating options in BEopt have recently been updated, allowing more accurate input of the system characteristics. The basic system configuration consists of a boiler and baseboard convectors. The user can input information for outdoor reset controls including maximum and minimum supply temperatures at user-defined outdoor temperatures. Setback schedules for the thermostats can also be altered.

However, there are a several other components being analyzed in this research effort that cannot be modeled. These components include:

- High mass condensing systems
- Indirect DHW tanks
- Buffer tanks

- Variable-speed pumps
- Boost controls.

These components have been singled out because they offer the opportunity to increase efficiency, improve durability, and/or improve occupant comfort. Field data collected during this study was used to analyze these features and will hopefully aid in expanding this software's capabilities.

The final limitation discovered during this exercise is the heating system auto-sizing function in the software. BEopt automatically choses a boiler capacity based on design calculations. Unfortunately, in this circumstance, this results in selection of equipment that does not exist. The minimum boiler capacity available on the market is a 50 kBtu/h boiler. It can modulate down to approximately 15 kBtu/h. For these homes, BEopt chose a boiler with a maximum capacity of 13 kBtu/h. This results in lower predicted space heating input values than actual and prevents proper modeling of recovery from setback. When the boiler is sized that small and that closely to the load, the system does not have the capacity to bring the space back to temperature after a thermostat setback.

3 Experimental Methods

3.1 Description of Test Homes

Three different hydronic heating systems were monitored. In general, all three test homes have the same or similar characteristics and/or efficiency levels as reported in Table 3. Overall installed cost and annual efficiency were evaluated taking into account energy required for pumps, DHW generation and the effects of different control strategies.

The equipment specifications along with the method for supplying DHW and the minimum and maximum space heating output capacities are displayed in Table 9.

Table 9. Mechanical System Specifications

Component	House #1	House #2	House #3
Space Heating Equipment Make/Model #	Baxi/Luna HT 380 modulating condensing boiler	HTP/Versa Hydro PHE130-55 condensing water heater	Peerless/PF50 modulating condensing boiler
Space Heating Efficiency	92.5% AFUE	96% thermal efficiency (provided through external heat exchanger)	95.1% AFUE
Emitter (total output)	Radiant floor cross-linked polyethylene tubing under subfloor/no fins (19,494 Btu/h)	Heating Edge Baseboard/Smith's Environmental Products (19,080 Btu/h)[1]	Heating Edge Baseboard/Smith's Environmental Products (21,264 Btu/h)[1]
Pumps	(2) Bell & Gossett NRF-25 (1) Grundfos UPS-15/62	(1) Grundfos Alpha (ECM[2]) (1) Taco 008 (variable speed Delta-T)	(4) Bell & Gossett Vario (ECM)
DHW	On demand DHW heat exchanger in Baxi Luna	Directly from Versa Hydro 55-gal tank	Peerless Partner PP40 Indirect 28-gal tank
Space Heating Output (max/min)	112,601 Btu/h 35,452 Btu/h	100,000 Btu/h 10,000 Btu/h	46,000 Btu/h 14,720 Btu/h
Design Heat Load	16,047 Btu/h	14,695 Btu/h	15,426 Btu/h

[1] Output is based on performance tests witnessed by BSRIA @ 140°F and 1 gpm
[2] Electronically commutated motor

Following is a detailed discussion of the differences in each system's configuration and control strategy.

3.1.1 House #1: Combination Boiler With On-Demand Domestic Hot Water Heat Exchanger

The Baxi Luna 380T installed in House #1 is a modulating, condensing boiler with an integrated on-demand DHW heat exchanger that provides both central heating and DHW (Figure 5). The maximum heating capacity of the boiler is 112 kBtu/h and the minimum is 35.5 kBtu/h (more than twice that of the design load). The supply temperature to the zones is controlled with an outdoor reset sensor and has been capped at 150°F at an outdoor temperature of 0°F. The reset curve for this system is plotted in Figure 6.

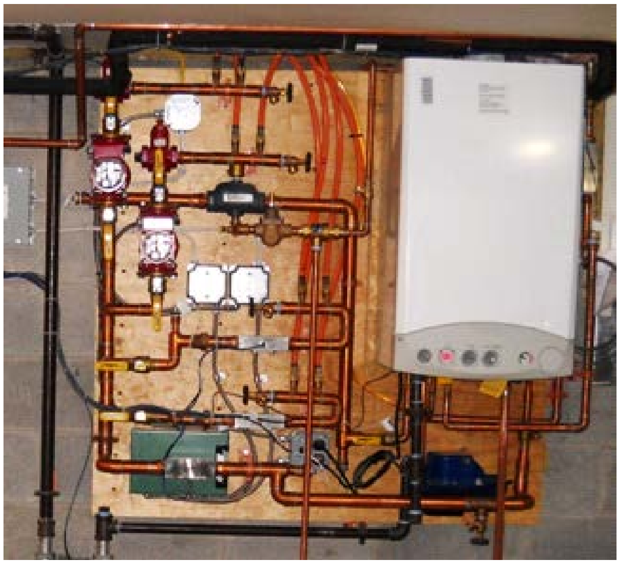


Figure 5. Picture of boiler setup for House #1

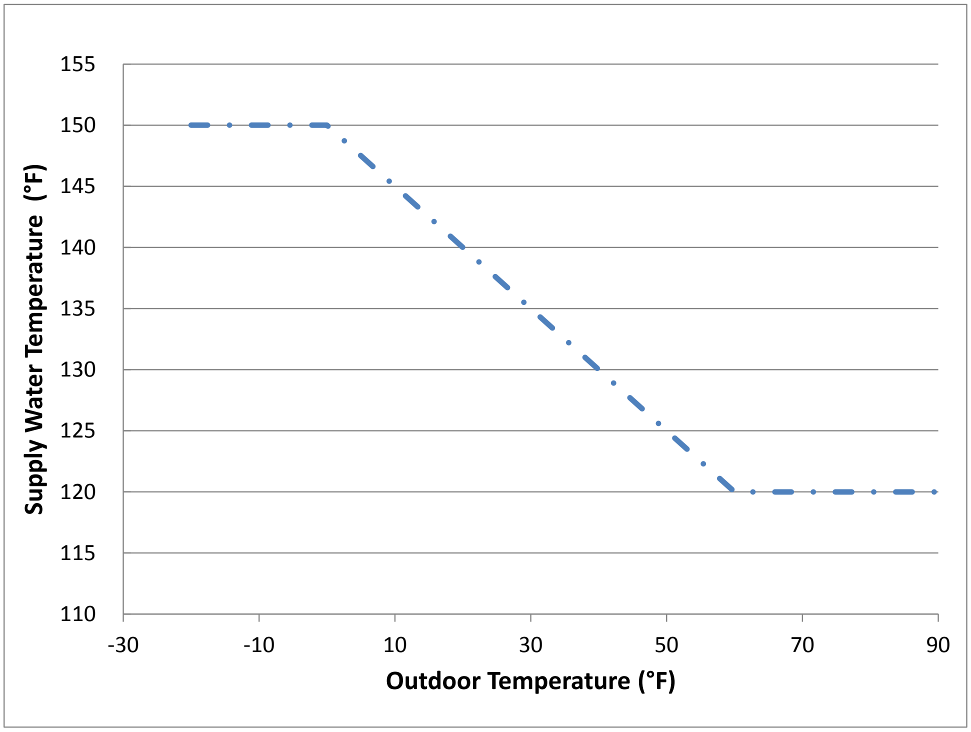


Figure 6. Outdoor reset curve for House #1

Figure 7 shows a schematic of the heating system setup, including the location of the sensors installed.

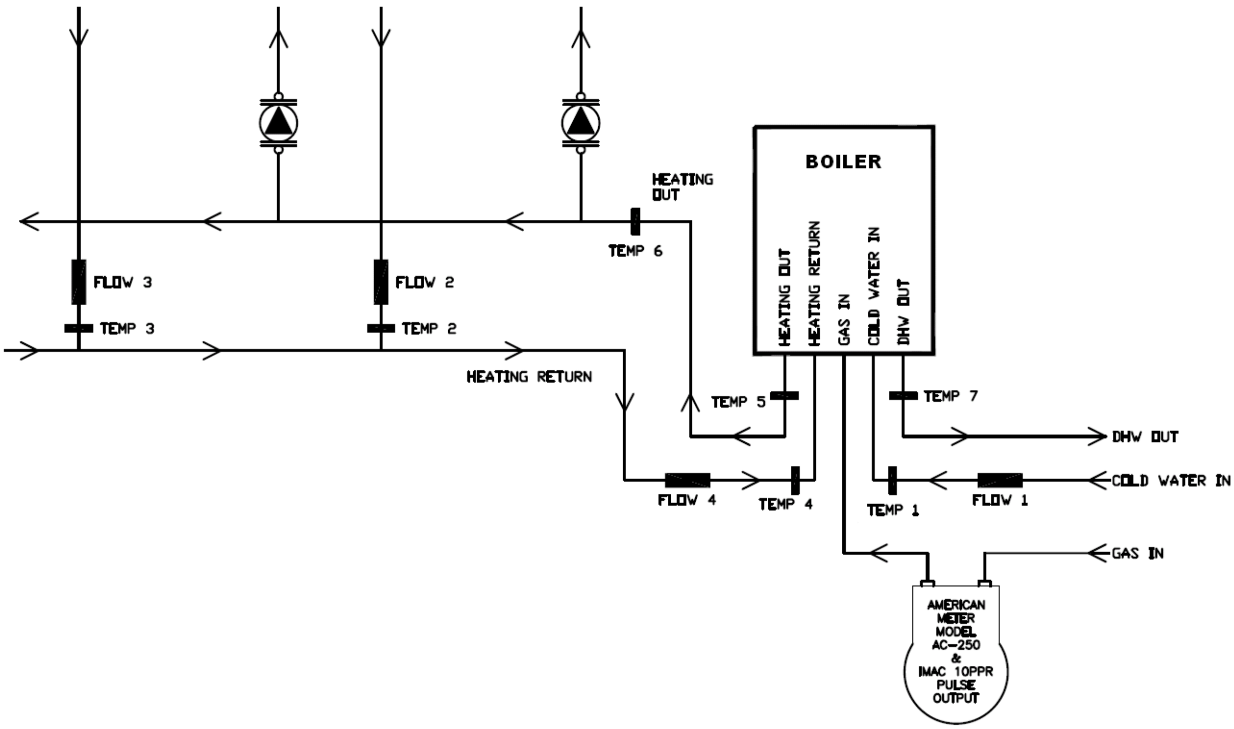


Figure 7. Schematic of boiler setup for House #1

Though the focus of this study is optimizing systems using baseboard convectors, this home has radiant floor heating with tubing installed between the floor joists under the subfloor on both the first and second floors. The heating, ventilation, and air conditioning contractor offered this as a free upgrade after construction was already underway, and the builder accepted. Because the delivery temperatures are similar to those of the homes with baseboard, it was anticipated that data from this house would still be of value.

3.1.2 House #2: High Mass Condensing Water Heater

The heating system in House #2 (VersaHydro from HTP) consists of a high mass, condensing water heater that provides both space heating and DHW (Figure 8). The heater is rated for a combustion efficiency of 98% and thermal efficiency of up to 96%. The maximum firing rate of the heating system is 130 kBtu/h and was capped at 80 kBtu/h to reduce cycling.

The material and design of the appliance's heat exchanger are intended to minimize buildup of lime and scale associated with hard water, a major concern expressed by manufacturers during the expert meeting. The 5-in. diameter combustion tube and 1⅛-in. secondary heat exchange spirals are intended to enhance heat transfer and eliminate the blockage risk of small passageways that exist in low-mass boilers.

Figure 8. High mass condensing water heater installed in House #2

The water in the 55-gal tank is supplied directly to the DHW end uses. When there is a call for heat by the thermostat, space heating is provided by cycling the water in the tank through a brazed plate heat exchanger that transfers heat to the heating loop (Figure 9). This heated water is then circulated to the conditioned space and distributed via baseboard convectors.

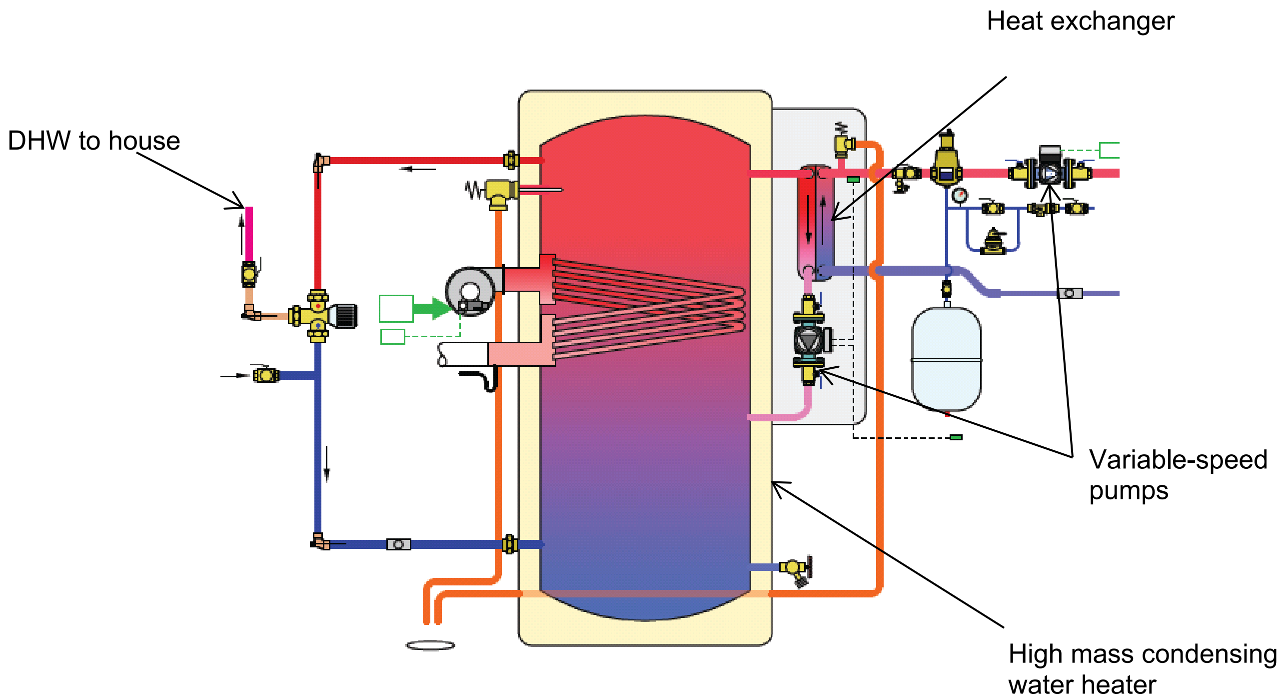


Figure 9. Heating system schematic for House #2

This appliance has a 5:1 turndown ratio (able to fire at 20% of its maximum input capacity) on the main combustion system and a 10:1 turndown ratio on the space heating module resulting in a minimum output capacity of 10,000 Btu/h. The additional reduction in space heating capacity is achieved through modulation of the pump feeding the brazed plate heat exchanger. Greater turndown should allow this appliance to modulate at lower firing rates, which conserves energy and improves component reliability. Modulation, along with thermal mass, helps to reduce the short-cycling effect when there are small space heating or hot water loads.

House #2's system is simpler to install because a primary/secondary loop is not required on the heating side, nor much near-boiler piping, boiler/indirect electrical connections, or isolation valves. This results in cost savings in materials and labor and decreased complexity.

The controls are set at the factory to maintain a minimum internal DHW temperature of 120°F. The system was designed so that the tank temperature will start to increase when the outdoor temperature falls to 25°F, as shown on the reset curve in Figure 10. At an outside temperature of 0°F, the water temperature would reach its maximum setting of 145°F.

The minimum water temperature supplied by the space heating module was initially set to 100°F. The variable-speed circulator within the boiler monitors the supply water temperature to the space heating distribution system, and attempts to maintain the target supply water temperature as dictated by the outdoor reset curve.

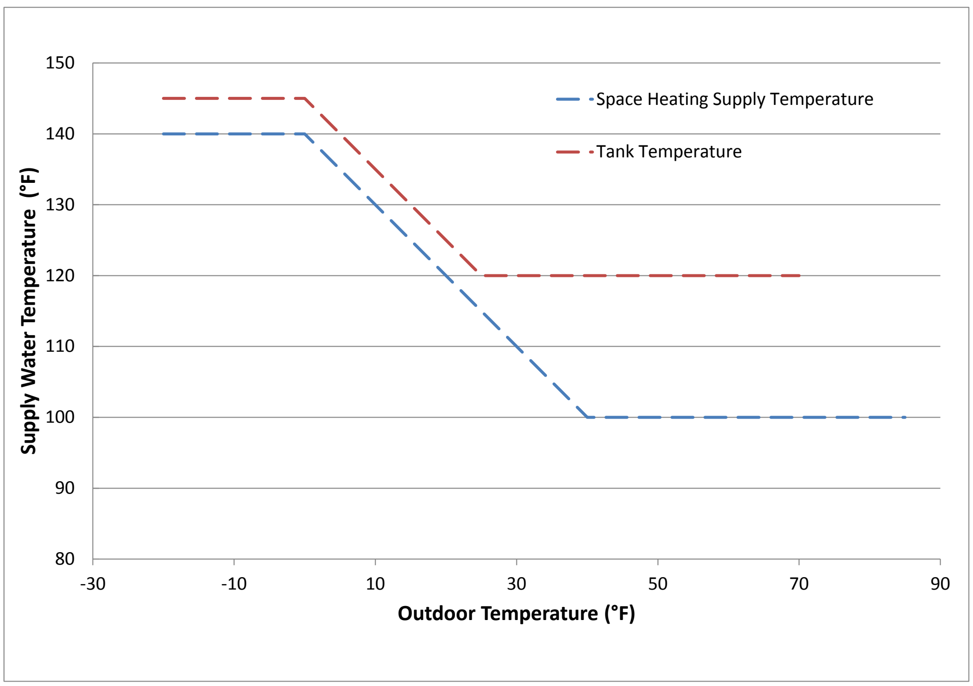


Figure 10. Outdoor reset curve for House #2

For health and safety reasons, the water in the tank cannot be kept any lower than 120°F.The effects of this on condensing performance were analyzed along with the standby losses associated with increasing the tank temperature to 145°F during design conditions.

The fact that the water in the tank will be kept higher than 120°F for part of the heating season results in the need for a tempering valve to prevent scalding. Although the builder of these test homes typically installs these valves on all his systems, this could be an added cost to someone who does not typically install them.

A second variable speed pump, a Grundfos Alpha, circulates the heated water to the zones (Figure 11). This pump has seven different settings: three constant-speed settings, three constant-pressure settings, and an automatic setting. The pumps can either run at a constant flow rate while the pressure in the system changes or run at constant pressure by regulating the flow rate through the system (GRUNDFOS 2010). During commissioning, this pump was set to the second constant-pressure setting resulting in a flow rate of just over 1 gpm to each zone.

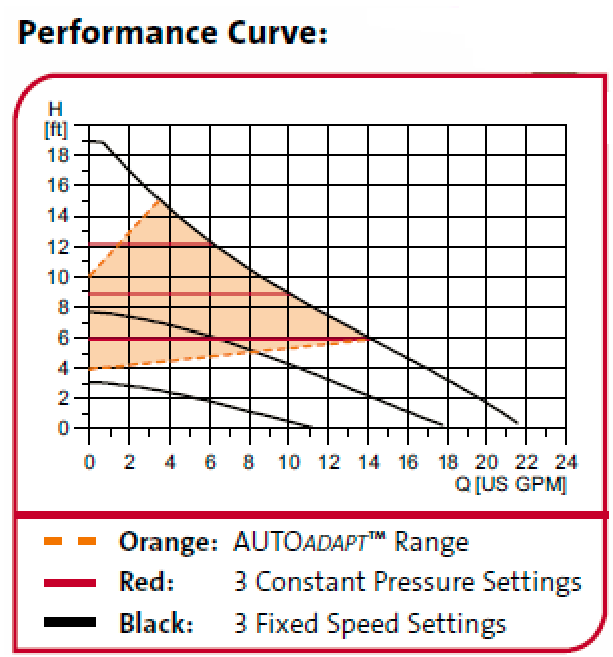


Figure 11. Pump curve for variable-speed pump in House #2

(courtesy of Grundfos)

During field testing, various settings were tested to determine the most efficient mode for the system installed. Both power consumption and flow rate were analyzed.

Heating Edge high output, double-pipe baseboard convectors from Smith's Environmental Products were installed and evaluated in this home. Comparisons to listed manufacturer data and those of other baseboard products on the market were conducted.

3.1.3 House #3: Low Mass Condensing Boiler With Indirect Domestic Hot Water (Buffer Tank Versus Primary/Secondary Loop)

The Peerless PureFire 50 (Peerless 2011) installed in House #3 is a modulating, condensing, low-mass boiler that also provides the DHW via an indirect tank. The boiler has an efficiency rating of 95.1% AFUE. This system was set up to compare two different approaches to providing hydraulic separation: a buffer tank (#3A) and a standard primary/secondary plumbing configuration (#3B). The image in Figure 12 shows the components and plumbing configuration installed in House #3.

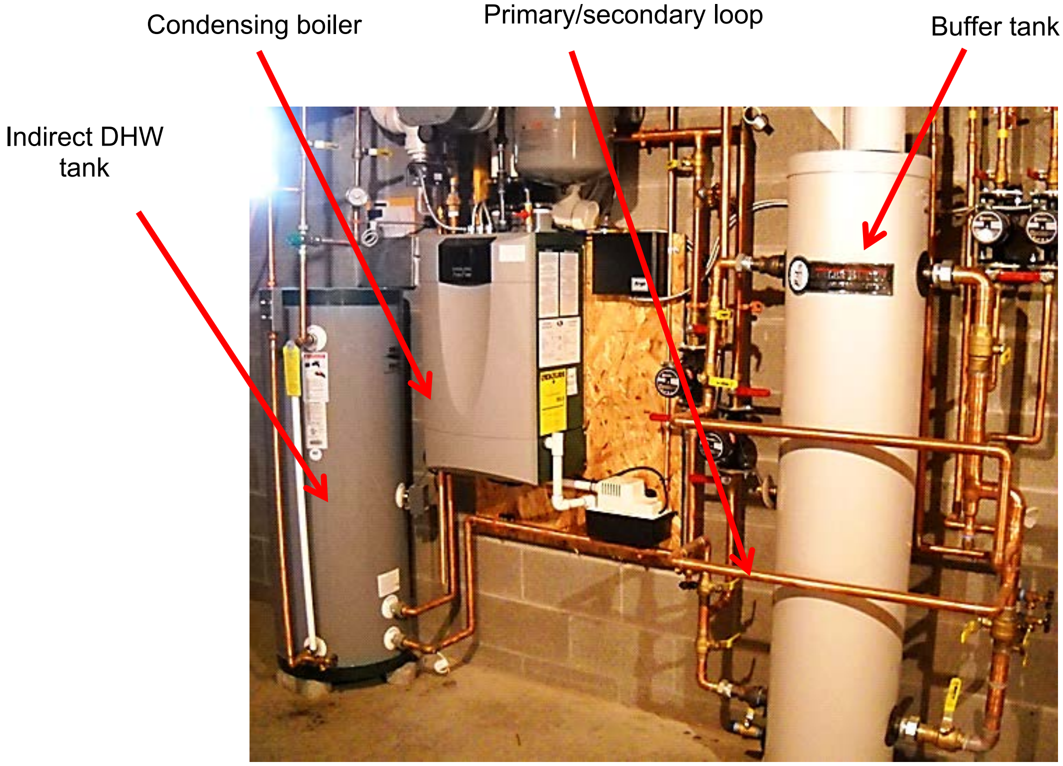


Figure 12. Low mass boiler installed in House #3

Figure 13 shows a detailed schematic of the heating system setup. Ball valves have been installed to switch the flow between the two plumbing configurations, and a toggle switch was used to direct the boiler's controller to either the temperature sensor in the buffer tank or one in the primary loop.

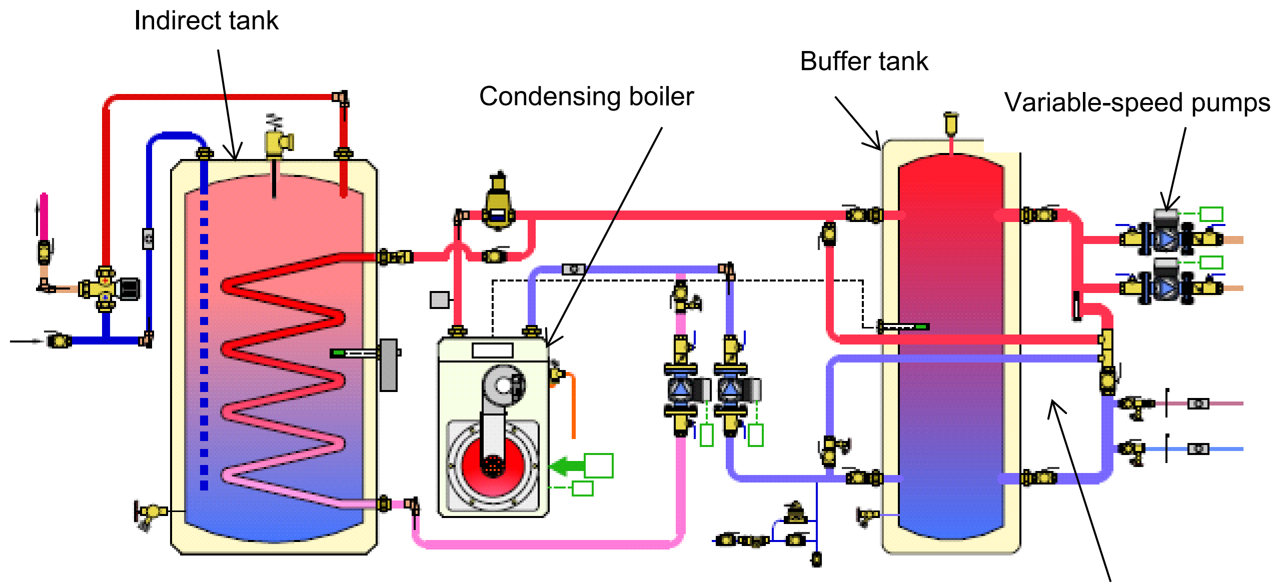


Figure 13. Heating system schematic for House #3

The outdoor reset curve for the boiler is displayed in Figure 14. The boiler fires under the operation of its own internal reset controller which monitors the temperature of the sensor in the well of the buffer tank when the buffer tank is “online” or the temperature of the sensor in the well of the distribution header when the buffer tank is “offline.” Like the system in House #2, the maximum supply temperature to the space was set to 140°F at the design outdoor temperature of 0°F. However, this curve allows the minimum boiler supply temperature to drop to 95°F when the outdoor temperature is 68°F, whereas the lower limit on the boiler supply for the high-mass boiler was 100°F (see Figure 10).

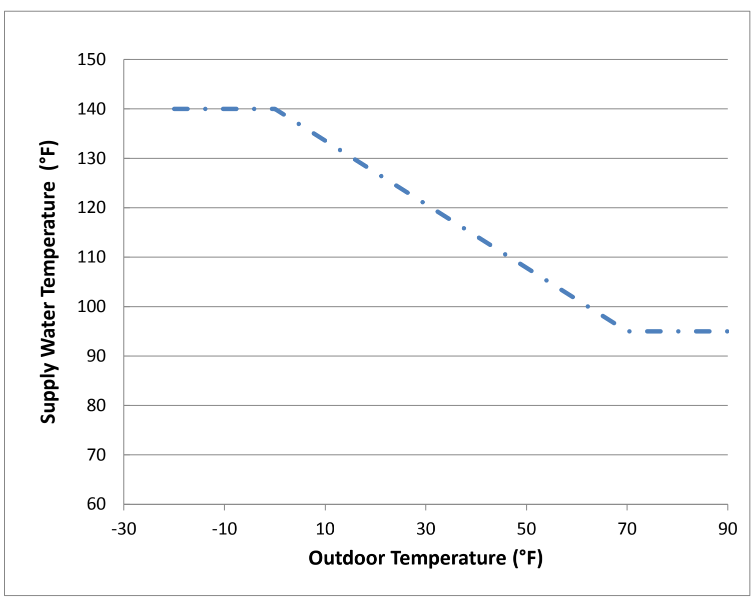


Figure 14. Outdoor reset curve for House #3

This boiler is equipped with controls to prevent short cycling. The installer can limit the maximum space heat input of the boiler up to 50%, if desired. This feature is beneficial in circumstances where the boiler is oversized for the space heating load either because the DHW demand is larger than the space heating load, or the space heating loads are simply smaller than the smallest available boiler on the market, which is the case here. Short cycling in previous studies was severe in similar size homes, because boilers typically go to "high fire" upon startup to ensure a properly lit burner and then modulate down. This causes a rapid rise in supply temperature resulting in the boiler shutting off quickly. Reducing the input to the space heating function is intended to reduce this short cycling. For this study the boiler's maximum space heating input was capped at 50%.

Another feature available with this boiler is a boost control that can be used to decrease the time to recover from thermostat setback. This boost control overrides the outdoor reset control as follows: if there is a heating demand from any combination of zones for *t* minutes, an offset of *d* degrees F is added to the target temperature of the boiler. For example, consider the case where the default values are used (t = 20 min and d = 18°F) and the outdoor reset algorithm calculates a target space heating supply temperature of 110°F. If the central heating call exceeds 20 min, the target supply temperature will be increased to 128°F. If the central heating thermostat continues to call for another 20 min, the temperature will jump again to 146°F, and so on. In systems with many zones, the boiler may boost to the highest temperature if calls from several zones overlap, which often occurs. Effects of this control strategy on the overall efficiency were evaluated and compared to eliminating setback completely and maintaining a constant indoor temperature 24 h/day.

DHW in this system is provided via a 37-gal indirect hot water tank that is heated by the boiler. When the tank thermostat calls for heat, the boiler will supply a higher temperature (160ºF).

A total of four Bell & Gossett Vario pumps were used in this system: one to circulate hot water to the DHW tank, one to circulate hot water through the buffer tank or primary loop, and two to circulate hot water to the space heating zones. All zone circulators have ECMs and have an input wattage range of 6–60 W. These pumps operate on a fixed-speed curve based on the setting of the dial on the side of the circulator. The operating zone for the models installed is displayed in Figure 15 in the zone titled "vario." During commissioning, the pumps were set up to provide 1.5 gpm through each heating zone loop and approximately 5 gpm through primary loop/buffer tank and the indirect DHW tank.

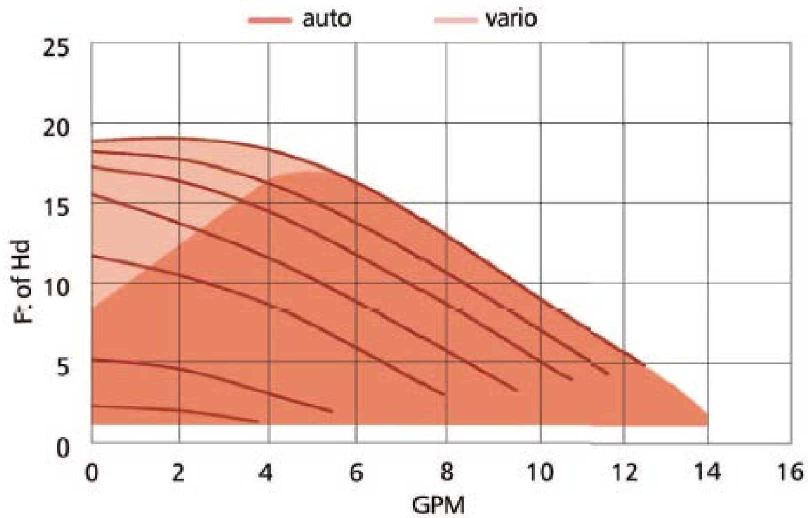


Figure 15. Pump curve for variable-speed pumps in House #3

(from Bell & Gossett install manual)

As in House #2, the Heating Edge baseboard was installed here.

3.2 Monitoring Setup

All three systems were set up to capture total energy input to the heating system (gas and electricity), total heat output to the conditioned space, output to each heating zone, and output for DHW generation. To calculate these values, gas input, water flow rates, system temperatures, electric energy consumption, and indoor and outdoor temperatures were recorded. All sensors were hard wired to a data logger located in the basement of each home. Data were recorded every minute and transmitted daily via wireless modem. Table 10 provides an overview of the parameters measured in each system and the location of the sensors.

Table 10. Parameters Measured in Each Home

Parameter	Sensor Location
Gas Input	Main gas line into the boiler/water heater
Pump Electrical Energy	At each pump
Boiler Electrical Energy	At main electrical line into the boiler
Boiler Supply Return Temperatures	Inlet and outlet temperatures on boiler/water heater main supply and return piping
Zone Supply/Return Temperatures	Right before the zones split form the primary loop and on each return line before the primary loop
Water Flow Rates	Each zone, the primarily loop and the cold water from the main into the water heater
DHW temperatures	On the hot water line right after the boiler/DHW tank and on the main cold water line into the system
Interior Temperatures	Each bedroom, kitchen, living room, dining room, and basement
Exterior Temperature	At the outdoor reset sensor location

A detailed description of each sensor installed for this study is located in Appendix A and includes its function, accuracy rating and model number. All temperature sensors are fast-response. Boiler, DHW, and heating zone water temperature measurements were made using tubular immersion thermistor sensors. These sensors include a 4.5-in. long × $^{5}/_{32}$-in. diameter probe and are plumbed directly into the water's flow. Exterior temperature measurements were made using hermetically sealed thermistor elements which allow for use in wet environments. Both the water and exterior sensors have an interchangeability rating of ± 0.1°C at 0°–70°C. Interior temperature measurements were made using exposed-element thermistor sensors, and have an interchangeability rating of ± 0.2°C at 0°–70°C.

The measurement circuit used for all thermistor temperature measurements was a direct current half-bridge circuit. The two resistors making up the half-bridge consisted of the 10KΩ thermistor sensor itself and a fixed-value 10KΩ, 0.1% precision resistor. An excitation voltage of 2.5Vdc was used.

To verify water temperature sensor accuracy prior to installation, all thermistors were connected to a Campbell Scientific CR10X data logger and submerged in a water bath. To verify interior and exterior temperature sensor accuracy prior to installation, all thermistors were connected to a Campbell Scientific CR10X data logger, then allowed to settle to room conditions. All sensors performed to manufacturer specifications.

Boiler gas consumption was measured using an American Meter Company residential diaphragm gas meter. The meter's standard pointer index was upgraded to an IMAC Systems Domestic Meter Pulser. This allows for 10 pulses per revolution, which translates to 1 pulse per 0.025 ft^3 of gas.

Electric energy consumption was measured using split-core, high-accuracy current transformers. Depending on the load, the current transformer rating was either 5A or 15A. Low current loads (less than 1A) were measured more accurately by winding the load's power line five times through a 5A current transformer's window, which converted the 5A current transformers to 1A transformers.

Boiler system, heating zone, and DHW flow measurements were made using low-flow, impeller flow meters. The accuracy rating for all meters is ± 1% of full scale down to 0.5 gpm. During monitoring it was clear that flow rates on the DHW side of the system often fell below the flow meter's accuracy range and were giving incorrect readings. Therefore, field measurements were made at the taps and compared to flow meter readings at the data logger. Calibration curves were generated for each of the DHW flow meters in the various test houses. The calibration curves for each system are shown in Appendix B.

3.3 Experiments Conducted

The intention of this research effort was to evaluate different high efficiency hydronic systems and their corresponding control and plumbing configurations to determine the most efficient combination of components; therefore, several changes were made during the monitoring period to evaluate the impacts on overall system efficiency. Several combinations of control settings, thermostat settings and plumbing configurations were evaluated in House #2 and House #3. Table 11 shows the testing schedule and the parameters that were altered during each period (highlighted in red). Each period lasted approximately 13 days except for period 1, which lasted 8 days.

Table 11. Settings for Each Two-Week Test Period

Test Period	Dates	Parameter	House #1	House #2	House #3
1	10/16–10/28/12	Thermostat	Constant	Constant	Constant
		Ave. Ind. Temp	71	67	68
		Plumbing Config.	n/a	n/a	Primary loop
		Boost Control	No	No	Yes
		Occupied	Yes	No	No
2	11/1–11/14/12	Thermostat	Constant	Setback	Setback
		Ave. Ind. Temp	70	63	66
		Plumbing Config.	n/a	n/a	Primary loop
		Boost Control	No	No	Yes
		Occupied	Yes	No	No
3	11/15–11/27/12	Thermostat	Constant	Constant	Setback
		Ave. Ind. Temp	68.5	66	66
		Plumbing Config.	n/a	n/a	Buffer Tank
		Boost Control	No	No	Yes
		Occupied	Yes	No	No
4	11/30–12/12/12	Thermostat	Constant	Constant*	Constant
		Ave. Ind. Temp	68	67	67
		Plumbing Config.	n/a	n/a	Buffer Tank
		Boost Control	No	No	No
		Occupied	Yes	No	No
5	12/17–12/29/12	Thermostat	Constant	Constant	Constant
		Ave. Ind. Temp	68.5	67	67
		Plumbing Config.	n/a	n/a	Buffer Tank
		Boost Control	No	No	No
		Occupied	Yes	No	No
6	1/9–1/21/13	Thermostat	Constant	Constant	Constant
		Ave. Ind. Temp	68.5	67.5	67
		Plumbing Config.	n/a	n/a	Primary loop
		Boost Control	No	No	Yes
		Occupied	Yes	Yes	No
7	1/25–2/6/13	Thermostat	Constant	Constant	Setback
		Ave. Ind. Temp	68	69	65.5
		Plumbing Config.	n/a	n/a	Primary loop
		Boost Control	No	No	Yes
		Occupied	Yes	Yes	No

*Outdoor reset curve was adjusted to provide warmer water to conditioned space.

Note that House #1 was not altered during the entire experiment because it was occupied when the heating season began. This system also lacked control and plumbing options available in the other two systems, which were of particular interest in this study—boost control, the ability to limit the boiler's firing rate, and the presence of mass. The settings in House #3 were altered the most because it was plumbed with two different piping configurations and had a boost function that allows it to bypass the outdoor reset control for faster response times.

4 Mathematical Methods

4.1 System Performance

To evaluate performance of the systems, monitored data were used to calculate the following:

- Total system input including boiler and pump power
- Output for DHW generation
- Total heating output to the conditioned space
- Output to each heating zone.

The calculation for all output values is displayed in Equation 1:

$$Q_{out} = \dot{V} \times 8.33 \times c_p \times (T_{out} - T_{in}) \tag{1}$$

where:

Q_{out}: heat flow out of the heating system (Btu/min)

$\dot{V}$: volumetric flow rate (gpm)

c_p: specific capacity of heat transfer medium water (1 Btu/lbm·ºF)

T_{in}: Temperature of water entering the system (ºF)

T_{out}: temperature leaving the system (ºF)

The constant 8.33 converts volumetric flow rate, gpm, to pound-mass per minute.

Output energy, Q_{out}, to each subsystem was calculated each minute and summed for each test period to obtain the total output energy. For heat transfer to the DHW, the main line water temperature was the inlet temperature (T_{in}) and the hot water out of the boiler/DHW tank going to the DHW end uses (before the mixing valve) was the outlet temperature (T_{out}). Flow rates through this subsystem were measured on the main water line into the boiler/tank.

Input energy, Q_{in}, was calculated for space heat and DHW and on a whole system basis. Gas used was added to the electric power consumed by the burner's combustion fan, the boiler controls, and the pumps.

Where possible the efficiency of the space heat and DHW was calculated individually. This was possible for House #1 and House #3 only. The hydronic systems in these two houses have DHW-priority. As such, if there is a space heating call and a DHW call simultaneously, the hydronic system diverts all input energy to DHW side. For House #2, however, the input energy could not be separated between space heat and DHW because both systems can operate simultaneously; it is impossible to account for losses and auxiliary boiler power associated with each separately.

When this separation could be performed, space heating input energy was calculated by summing the gas used and power consumption of auxiliary components when there was no flow to the DHW end uses. DHW input energy was calculated by summing all input energy only when there was flow to the DHW side.

4.2 Heating Degree Day Calculations

In order to compare system performance across each test period, input energy values were normalized against heating degree days (HDDs). HDDs were calculated by summing the differences between the daily average outdoor temperatures and the balance point temperature of 65°F (Equation 2). The balance point temperature is the temperature below which heating is required. HDDs were calculated for each test period, each of which lasted approximately 13 days.

$$HDD(T_{bal}) = \sum_{days}^{N} (T_{bal} - T_o)_+ \qquad 2$$

where,

HDD = heating degree days (days - ºF)
T_{bal} = balance point temperature (ºF)
T_o = average daily temperature (ºF)
N = number of days

5 Results

The systems in each of the three homes provide both the space heating and the DHW. Where possible and applicable, actual input and output values for each of the categories shown have been provided. Due to equipment used and plumbing configurations, some categories were not able to be separated out and have been given as aggregate values. The flowchart in Figure 16 provides an overview of the system end uses and the corresponding losses.

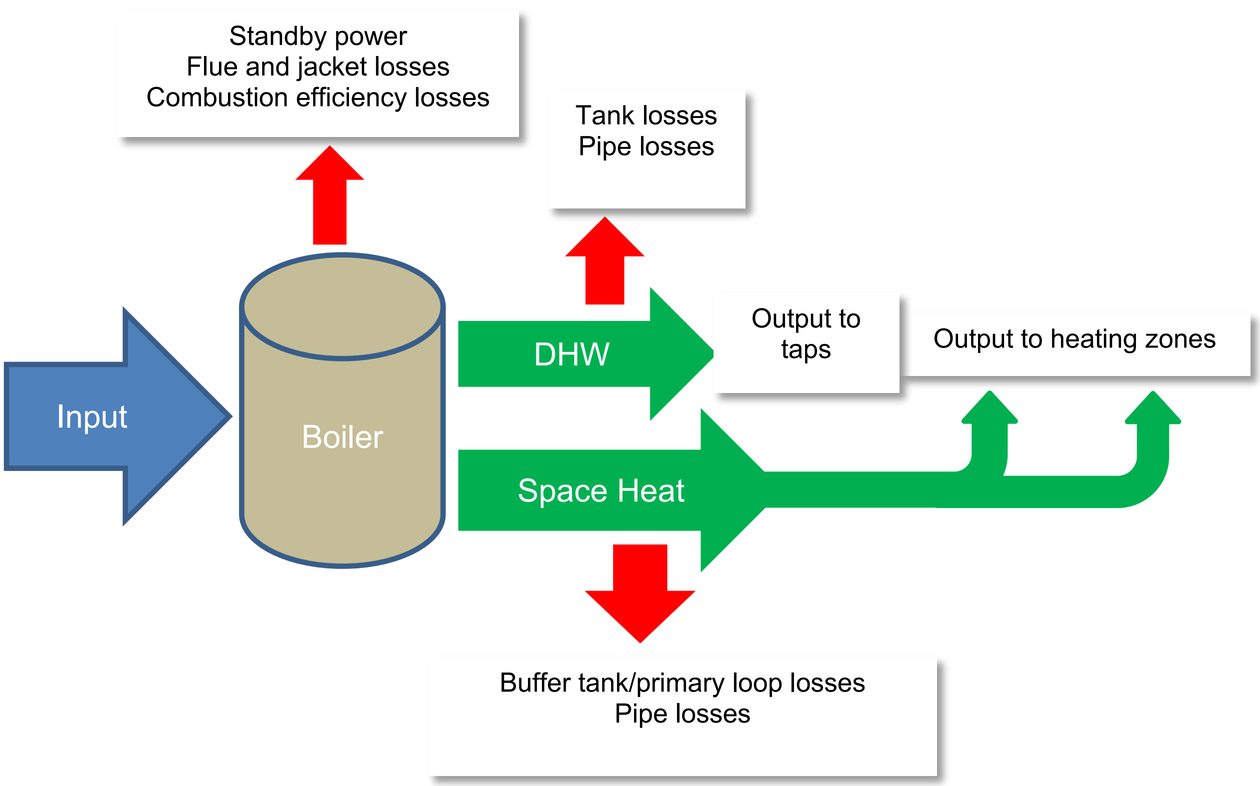


Figure 16. Flow chart of hydronic system energy balance

With a couple of exceptions, each system experienced the losses shown in Figure 16 to various extents. For example, House #1 did not experience tank losses because the DHW is provided by an on-demand DHW heat exchanger in the boiler, and House #2 had no primary/secondary plumbing. Results of the monitoring are presented in the following sections and have been presented for each system individually as well as compared side by side.

5.1 House #1

The system designed for House #1 includes a combination condensing boiler that provides central heating as well as DHW through an on-demand DHW heat exchanger. Heat is delivered to the conditioned space via radiant floor tubing under wood subfloors. Unfaced R-19 fiberglass batt insulation was installed under the tubing on each floor. The output capacity of the floor is estimated to be 18 Btu/h/ft^2 based on output tables published by Uponor (2011). Considering tubing was installed under the entire floor surface on both floors of the home, this would result in an output capacity of 19,800 Btu/h at a supply temperature of 140°F. This home was occupied during the entire four month test period. Thermostats settings were primarily kept at a constant temperature (as opposed to using a nighttime setback).

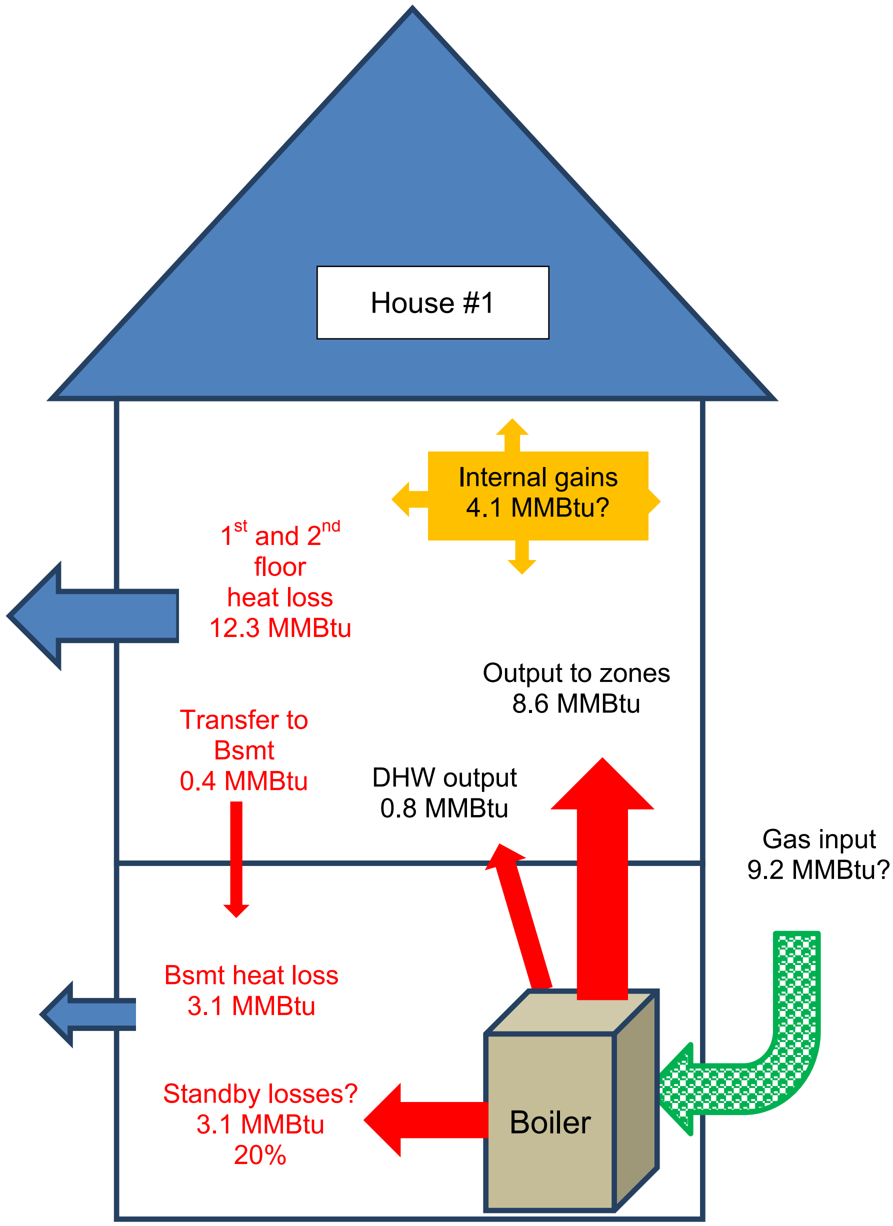
The total energy consumed during the monitoring period by the hydronic system is presented in Table 12. Energy output for space heating and DHW is broken out along with the electrical energy use of the pumps and boiler. Note that overall system efficiency was not calculated for this home. This value is the total output to space heating and DHW divided by the total natural gas input plus electricity consumption of the pumps and boiler. A problem with the gas meter in this home is suspected; therefore, the overall system efficiency was not calculated. Reasoning to support this suspicion follows.

Table 12. Space Heating and DHW Input and Output Summary for House #1

End Use	Measured Value	Result
Space Heat	Gas input (MMBtu)	8.2
	Space heat pumps (kWh)	146
	Boiler electricity (kWh)	49
	Total input (MMBtu)	8.6
	Total output to zones	8.6
DHW	Gas input (MMBtu)	1.1
	DHW pump (kWh)	1.4
	Boiler electricity (kWh)	5.5
	Total input (MMBtu)	1.1
	Total output (MMBtu)	0.7
Total System	Gas input (MMBtu)	9.2
	Entire system electricity (kWh)	208
	Total system input (MMBtu)	9.9
	Total system output (MMBtu)	9.4
	Overall system efficiency	n/a
	Number of on/off cycles	16,314
	Total cost of gas	$126
	Total cost of electricity	$38
	Total utility cost	$163

Assumed: $1.3644/therm, $0.18/kWh

Figure 17 is a graphical representation of the system performance for the entire 4-month monitoring period, and shows the interactions between conditioned and unconditioned spaces, internal gains, and hydronic system losses.



(Values in black are from measured data; values in red are estimates based on a combination of measured and calculated parameters).

Figure 17. Energy flows for House #1

There are a couple of important issues to point out regarding the values in Figure 17. First, the calculated space heating load of 15.8 MMBtu is much larger than the total measured gas output of 8.6 MMBtu. The calculated total space heating load is a combination of the heat loss from the first floor to the outside (12.3 MMBtu), from the conditioned space to the basement (0.4 MMBtu), and from the basement to the outside (3.1 MMBtu). The heat loads were calculated by multiplying the heat loss coefficient or building heat loss coefficient (Btu/h·°F) by the difference between the interior temperature and the outdoor temperature for each minute of the monitoring period and then totaling the values. The difference between the calculated load and measured output from the heating system is approximately 7.2 MMBtu or 81,000 Btu/day (46% of the load). If there is no problem with the monitoring equipment, this energy must come from solar and internal gains. Otherwise the interior set point temperatures could not have been maintained. An analysis of internal and solar gains was performed to determine if this deficit could reasonably be made up by those sources.

It is very evident when looking at the daily graphs like the one in Figure 18 that a significant portion of the heating load is being satisfied by gains from occupants and related activities. It is not uncommon for the first floor temperature in this home to rise during the day (even on the coldest days) while the heating system is off. In Figure 18, spikes in DHW supply temperatures indicate occupant activity. Considering that these spikes are generally higher in the kitchen than the other areas on that floor, it can be assumed that cooking or increased use of kitchen appliances is occurring. Heat gain to those spaces from DHW supply lines in the walls is also likely. The spikes in room temperature when no heat is being provided by the hydronic system support the assumption that internal gains are contributing to the heating needs of this home.

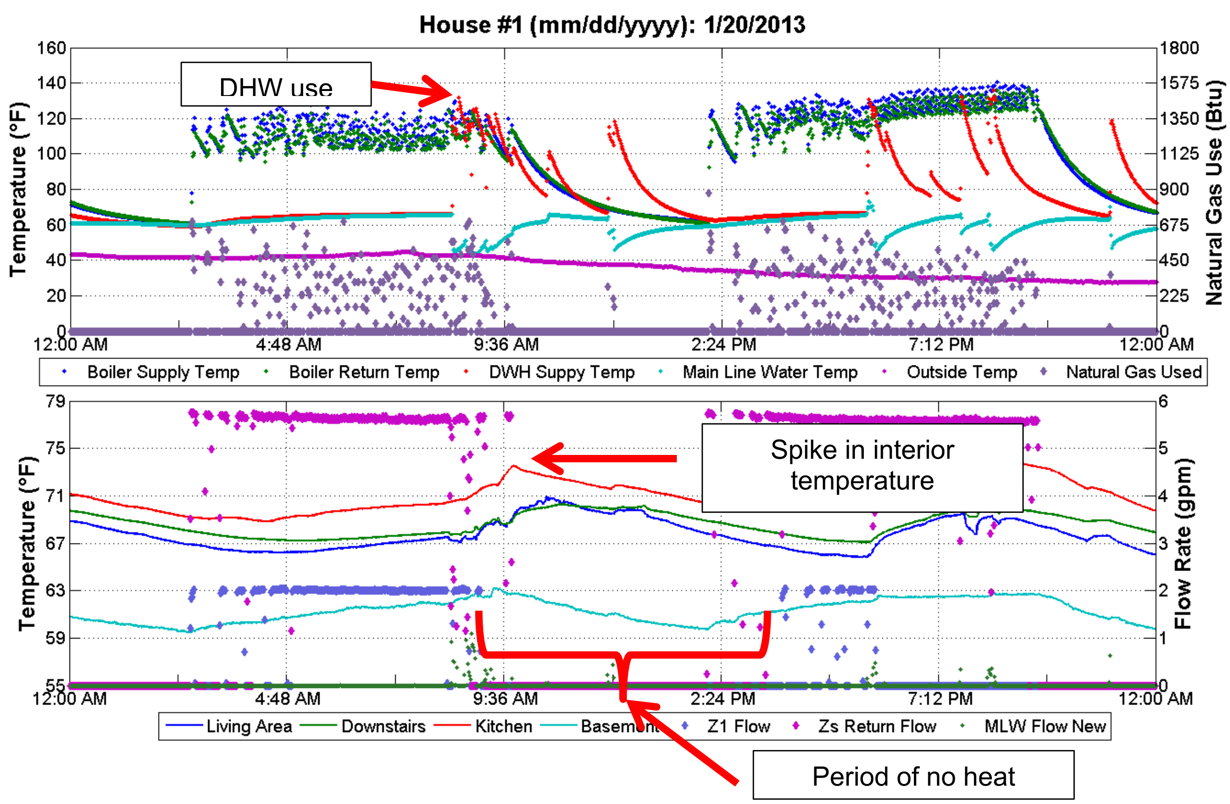


Figure 18. Effects of internal gains on interior temperatures

Actual electric bills to help determine internal gains were not available at this time. Instead, predictions from BEopt were used to estimate site energy consumption for electricity for lights, appliances and miscellaneous electrical loads (Table 13). If it is assumed that all the electricity consumed by lights and miscellaneous appliances contributes heat to the home and up to 50% of the large appliances (Hendron and Engebrecht 2010), these loads would help to make up a significant portion of the deficit between the heating load and the measured output.

Table 13. Predicted Internal Gains Using Results From BEopt

Source of Internal Gain	Annual Contribution (MMBtu)	Contribution per Day (Btu)
Miscellaneous Electric Loads	8.7	24,000
Large Appliances	3.4	9,500
Lights	2.7	7,000
Total	18.2	40,500

Solar gain will also make up some of the deficit between the load and the output. If an average value of 3.0 kWh/m^2/day is assumed for the solar radiation on the vertical face of the south wall, the solar contribution through the south windows alone would be approximately 14,300 Btu/day for 50 ft^2 of south glass assuming an SHGC of 0.3. This would account for about 20% of the deficit. Combined, these estimated internal and solar gains make up only 66% of the deficit between the calculated loads and the gas input.

A comparison of the total gas flow through the boiler's dedicated gas meter and the outside gas meter for the entire house shows a discrepancy of approximately 30% with the exterior meter reading higher. The only other gas appliance in the home is the kitchen range. This difference of 30% translates to approximately 11,000 cf of gas, or 11.4 MMBtu over 10 months (10 months of data were available for this home), or 38,000 Btu/day. This is about four times the predicted gas use for a cooking range from BEopt and twice as much as estimates published by DOE (2009).

In summary, a problem with the gas meter is suspected for the following reasons:

1. Discrepancies between calculated loads and measured gas input cannot be adequately accounted for based on estimates of solar and internal gains.

2. The difference in measured gas input between the whole-house gas meter and the dedicated meter to the boiler cannot be reasonably accounted for based on standard predictions of energy consumption for natural gas ranges.

Because of this, efficiencies for House #1 were not calculated, but other performance criteria such as runtime, cycling, and system losses will still be compared to House #2 and House #3.

5.2 House #2

Heating and DHW for this home were provided by a 55-gal condensing water heater paired with dual-pipe baseboard convectors. Boiler supply temperatures for space heating were controlled by an outdoor reset control. The home was initially set up to maintain a constant temperature of 67°F for 24 h/day. It was unoccupied for the first five test periods, but occupied for the last two. The total input to the space heating and DHW for each test period is shown in Figure 19. Electric energy consumption of the pumps and boiler is included. Note that the values have been normalized for HDDs in each period. HDDs for each of the seven monitoring periods are listed in Table 14.

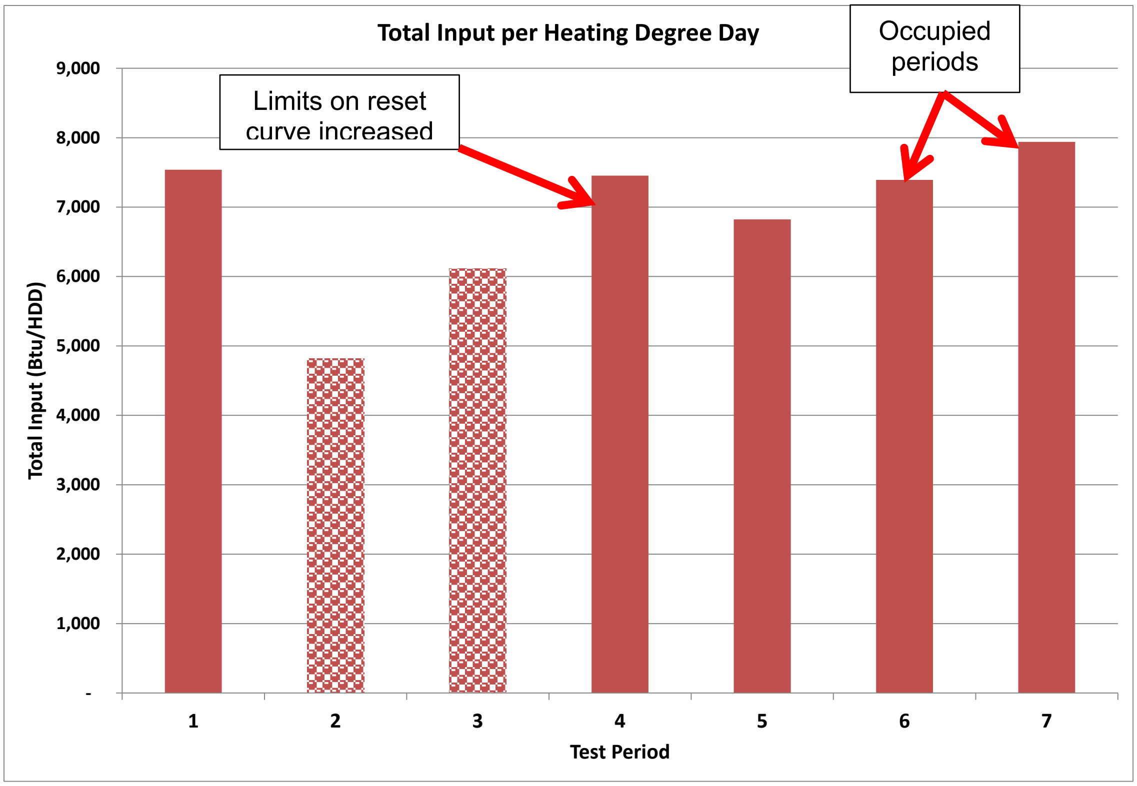


Figure 19. Total hydronic system input per HDD per test period for House #2; pump and boiler power is included

Table 14. HDDs for Each Test Period (Balance Point Temperature of 65°F)

Test Period	HDDs
1	94
2	297
3	355
4	305
5	415
6	395
7	465

Note that periods 2 and 3 are differentiated in Figure 19. During both of these periods the system was unable to maintain the desired indoor temperature of 67°F. Night setback was employed during period 2, but the system was unable to fully recover from the 7°F night setback over the 12-h daytime recovery period. When using an outdoor reset control, this is to be expected, especially if the baseboards are not oversized and there is no boost function to override the boiler's reset curve, as is the case here. Because the system could not bring the house to the desired temperature during period 2, setback is not considered a viable option for the current system configuration in House #2.

This is not to say that these components could not be used with a setback strategy. The condensing water heater used has sufficient capacity to provide the extra needed to recover from setback. If sufficient baseboard capacity was installed, recovery from setback could have been achieved. How fast recovery was achieved would depend on how much extra capacity was provided.

In period 3, the system was once again set up to run at a constant temperature of 67°F. Shortly after this change was made, it was discovered that the system was unable to maintain the desired set point once the outdoor temperature fell below 40°F. The pumps ran continuously with no interruption and interior temperatures began to drop.

The main reason the original design settings were not able to keep up with the load appears to be underperformance of the baseboard, primarily on the second floor. Each baseboard on the second floor was plumbed to promote parallel flow through both pipes in the emitters, as opposed to series flow where the supply water enters the top of the baseboard and exits through the bottom pipe. The parallel flow arrangement should have resulted in a higher output per linear foot (Table 15) than the baseboard plumbed in series on the first floor. This was not the case. Heating Edge is a two-tube baseboard. It is suggested by the manufacturer that the underperformance may be due to the fact that the upper tube did not see any or much flow at all and that this was most likely due to it being air bound and/or unbalanced flow. It is also likely that the ½-in. cross-linked polyethylene tubing and special fittings that were installed created a higher pressure drop that originally thought thus limiting available flow to each baseboard section.

Table 15. Comparison of Baseboard Output for Single Pipe Versus Double Pipe Models

	Average Water Temperature (°F)									
	90	100	110	120	130	140	150	160	170	180
	Btu/h Output/ft									
Multi-Pak 80 Single Pipe	–	–	200	270	340	410	490	570	650	730
Heating Edge Double Pipe (Series Flow)	101	165	226	289	356	426	498	572	647	725
Heating Edge Double Pipe (Parallel Flow)	130	205	290	385	460	546	637	718	813	911
Heating Edge Single Pipe (Bottom Flow)	75	127	169	208	260	311	362	408	470	524

While the temperature sensors are not accurate enough to precisely determine the output per linear foot, estimates indicate that only 50% of the rated output per foot is being achieved. In looking at the manufacturer's published ratings in Table 15 (AHRI 2009), it appears that the suppositions above are correct in that the baseboard is performing as if only one tube were heating This is supported by the fact that total circulator run time for the second floor zone is two to three times higher than that of the first floor circulator despite having very similar design loads. This increased runtime was very consistent for periods 1, 4, and 5, all unoccupied periods using constant temperature controls. Since occupancy, this has changed and appears to be due to changes in thermostat settings by the occupants.

Insulation levels were verified through visual inspection and with an infrared camera. No significant voids were discovered that would explain an increased load on the second floor as compared to the first floor. It is suspected that the reduced output is a function of uneven flow through the two pipes in the baseboard. Special fittings were fabricated for the ends of the baseboards to accommodate the temperature sensors. These fittings could be resulting in uneven or laminar flow and reduced output.

At the beginning of period 4 the lower and upper temperature limits on the outdoor reset curve were increased, which resulted in higher supply temperatures to the heating zones (Figure 20).

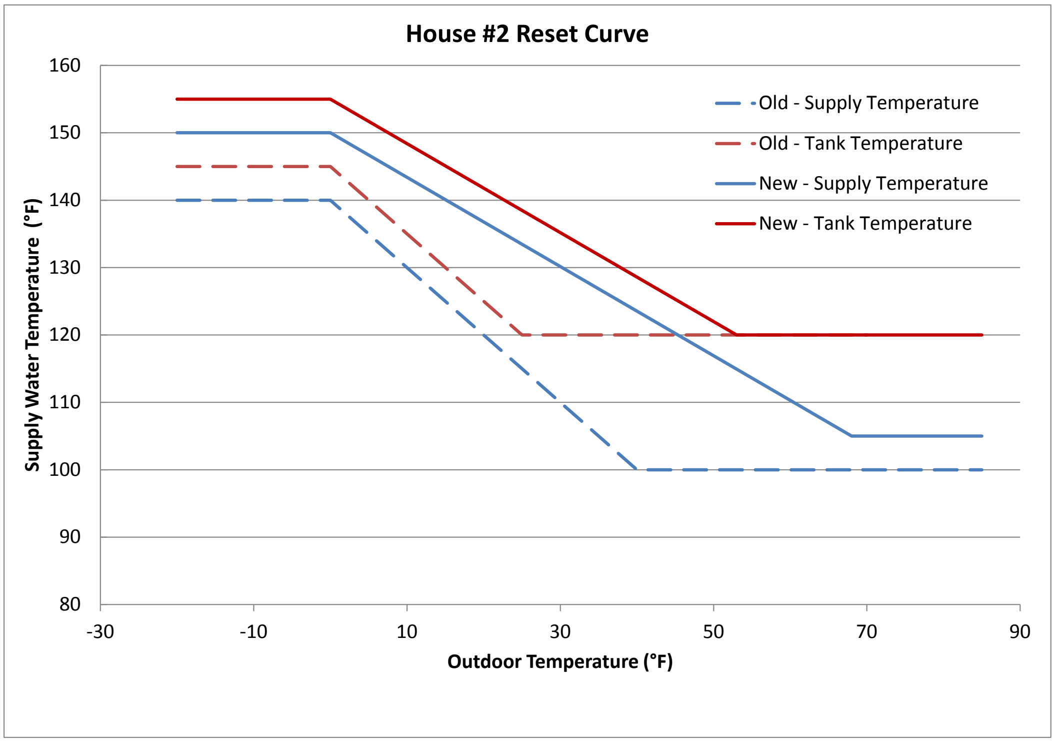


Figure 20. Revised outdoor reset curve for House #2

The new curve was determined by calculating how much heat was being delivered to the space to maintain a set point of 65°F (the interior temperature to which the house was falling overnight) at the corresponding outdoor temperature. The resulting heat loss was then compared to how much heat was needed to maintain the space at the design condition of 70°F (used when sizing the system). The percent difference between the actual output and the desired output was applied to the old reset curve to generate the new curve. A safety factor was added in as well to ensure there were no comfort problems once the home was occupied. There have been no problems maintaining the indoor temperature setting since this change was made.

Unfortunately, the new settings reduce the amount of time the boiler will condense over the course of the heating season and will increase standby losses through the tank's shell, because the entire tank will now have to be kept at a higher temperature than required by the previous curve. The prior curve did not require that the tank temperature be increased above 120°F (minimum for DHW) until the outdoor temperature fell below 25°F. With the new curve, once the outdoor temperature falls below 55°F, the tank temperature will start to increase above 120°F. Once the tank temperature exceeds 130°F, condensing will be severely limited. This was confirmed with the use of a Bacharach combustion analyzer during a period when the tank was being maintained at 144°F. The resulting combustion efficiency reading on the meter was 89%, not the rated 96% for condensing operation as recorded when the tank was being maintained at 120°F.

The input energy associated with space heating only is presented in Figure 21. Variations in input per HDD are due to:

1. Increases in the limits of the outdoor reset curve at the beginning of period 4
2. Increases in efficiency as HDDs increase (standby losses become a smaller percent of overall load)
3. Internal gains from occupants in periods 6 and 7
4. Increases in thermostat settings in period 7.

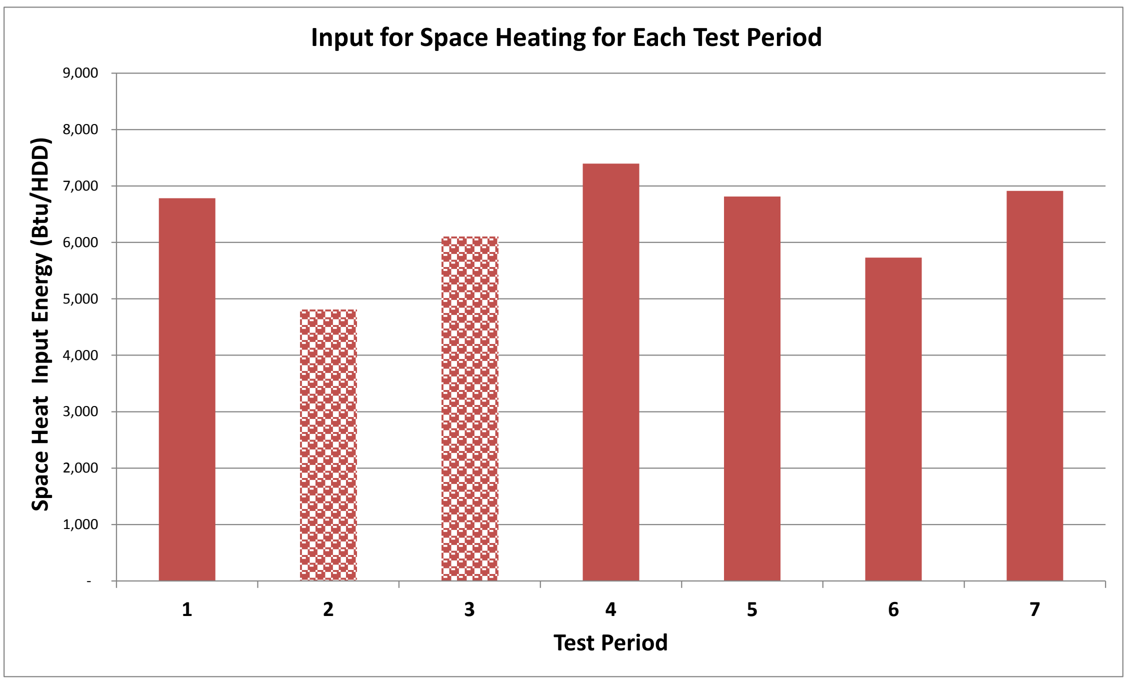
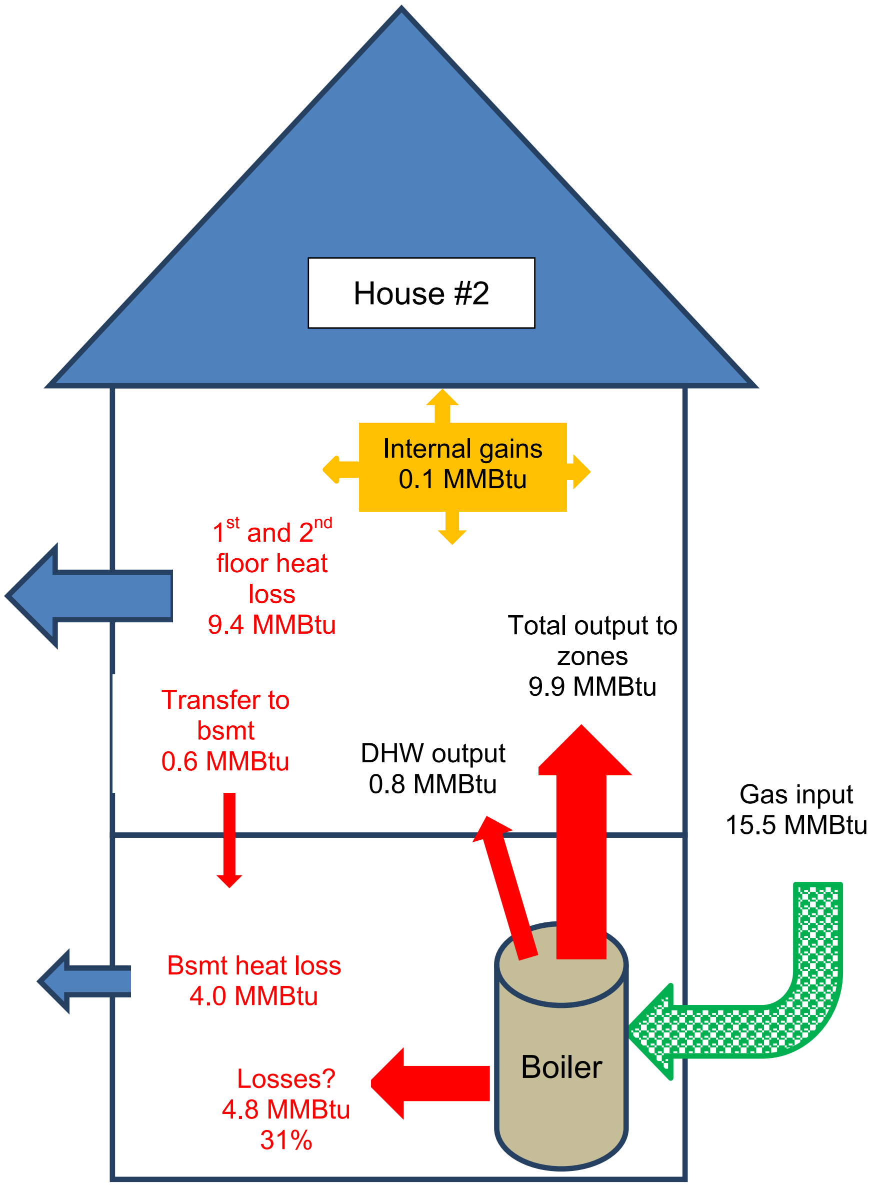


Figure 21. Space heating input for each test period for House #2; pump and boiler power is included

Total input and output energy for all 7 test periods is summarized in Table 16 and Figure 22. Note that DHW and space heat input values have not been separated out. Heat operation does not stop when there is a call for hot water at the taps, as is typical in the other two systems. Therefore, properly assigning losses to each end use is not possible.

Table 16. Space Heating and DHW Input and Output Summary for House #2

End Use	Measured Value	Result
Space Heat	Total output to zones (MMBtu)	9.9
DHW	Total output (MMBtu)	0.8
Total System	Gas input (MMBtu)	15.5
	Entire system electricity (kWh)	131.3
	Total system input (MMBtu)	16.0
	Total system output (MMBtu)	10.7
	Overall system efficiency	67%
	Number of on/off cycles	5,758
	Total cost of gas	$212
	Total cost of electricity	$24
	Total utility cost	$235



(Values in black are from measured data; values in red are estimates based on a combination of measured and calculated parameters).

Figure 22. Energy flows for House #2 for the entire test period

Pipe and standby losses to the space can be seen in Figure 23. This set of data is from a warm day with no heating activity and no call for DHW. The three sets of purple dots represent gas use. The gas is being used to maintain the tank's internal temperature of 120°F.

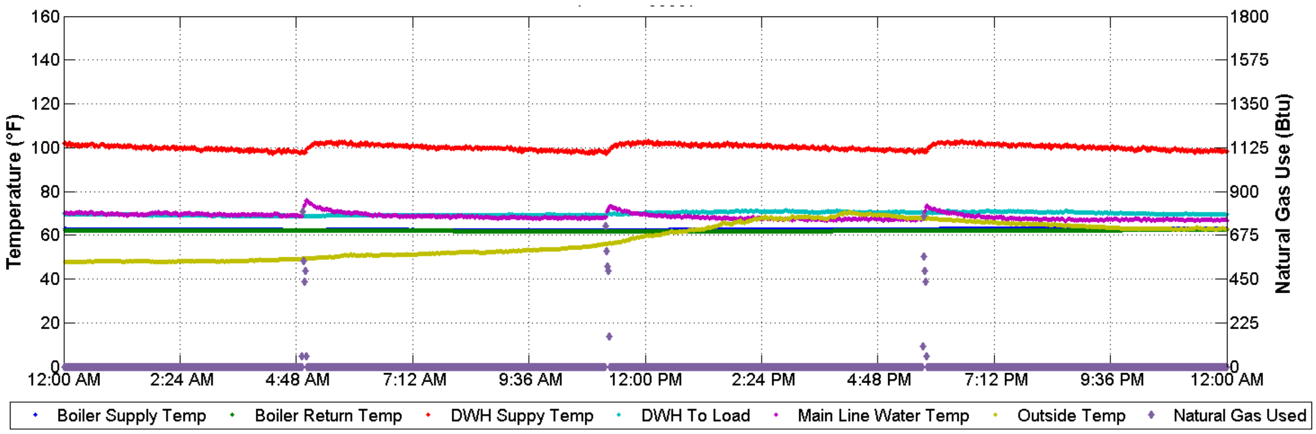


Figure 23. Example of gas consumption needed to maintain the condensing water heater tank temperature of 120°F

The gas input into the tank during these periods is approximately 2,500 Btu/call or 7,500 Btu/day: 100% of this input is standby loss. This value will increase as the basement temperature decreases and vise versa. As heating and DHW needs increase, this standby loss becomes a smaller percentage of the total load. During the summer months, it will be a much larger percentage of the overall load.

The basement in this home, as in the others, is not directly heated, yet has maintained a constant average temperature of 66°F over the entire test period. Losses from the hydronic system (pipes and tank losses), combined with heat transfer from the first floor to the basement are responsible for this level of conditioning. Heat loss from the first floor to the basement is estimated at 0.6 MMBtu over the entire monitoring period. The heat load of the basement was estimated by multiplying its heat loss coefficient by the difference between the basement room temperature and the outdoor temperature for each minute of the monitoring period. The resulting load is approximately 4.6 MMBtu.

5.3 House #3

House #3 was equipped with a modulating, condensing boiler and an indirect DHW tank. The same baseboard that was used in House #2 was installed in this home as well and was plumbed identically—parallel flow in each baseboard on the second floor and series flow on the first. In addition to an outdoor reset control, this boiler came with boost controls that would override the reset control if response time was slow. This feature was enabled during the test period and the factory defaults left in place. To help prevent short cycling, this boiler is also equipped with the ability to limit the input for space heating, thereby preventing it from going to high fire and shutting off prematurely. Space heating input was capped at 50% of the maximum.

Seven distinct tests were performed on this system. Details of the system setup and operating conditions were listed in Table 11. General results for House #3 are presented first followed by a comparison of the performance for each set of operating conditions.

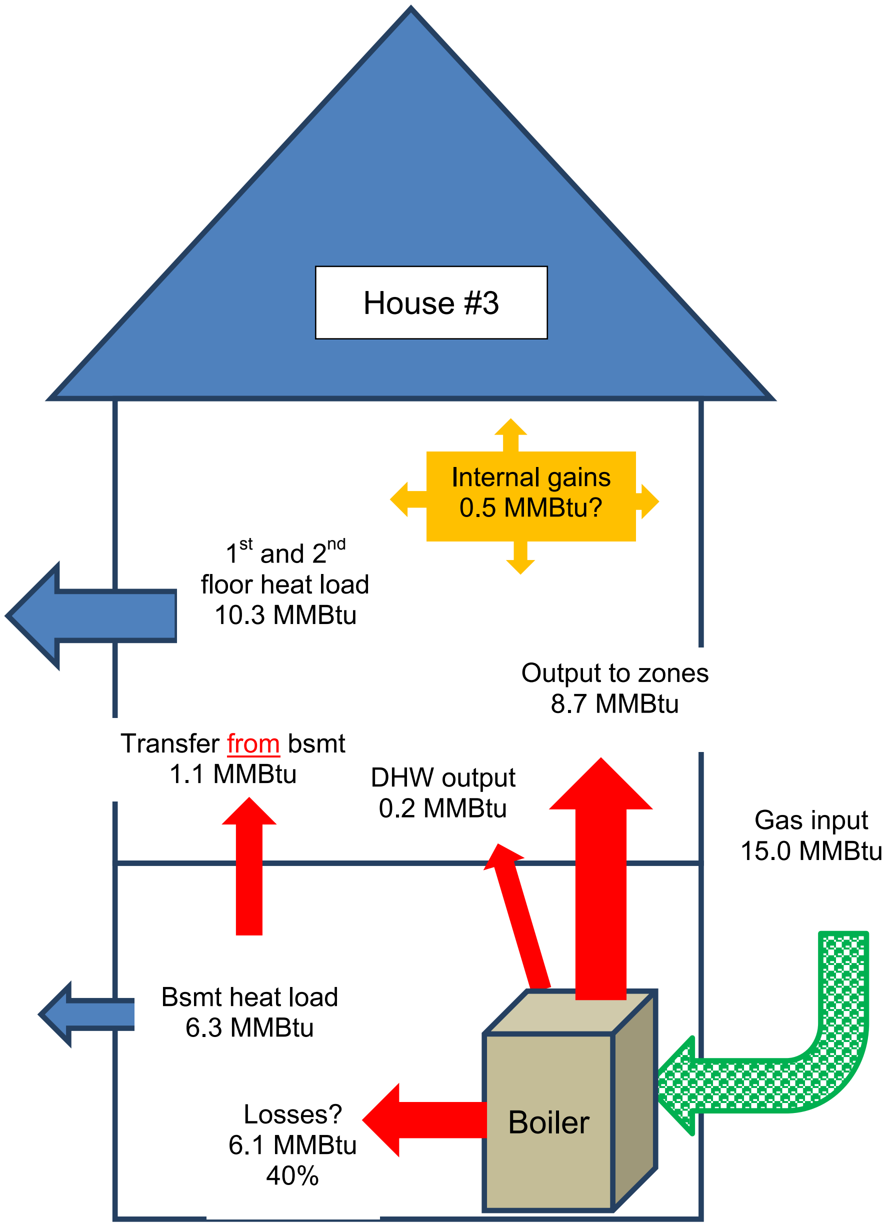
5.3.1 General Results for House #3

Results of the entire monitoring period are presented in Table 17 and are graphically presented in Figure 24. When evaluating overall efficiencies for House #3, the fact that the building is currently unoccupied must be taken into account. Because of this, any input to the DHW to maintain the tank temperature is considered standby loss.

Table 17. Space Heating and DHW Input and Output Summary for House #3

End Use	Measured Value	Result
Space Heat	Gas input (MMBtu)	13.4
	Space heat pumps (kWh)	61.8
	Boiler electricity (kWh)	34.4
	Total input (MMBtu)	13.8
	Total output to zones	8.7
DHW	Gas input (MMBtu)	1.6
	DHW pump (kWh)	1.5
	Boiler electricity (kWh)	2.2
	Total input (MMBtu)	16
	Total output (MMBtu)	0.2
Total System	Gas input (MMBtu)	15.0
	Entire system electricity (kWh)	109.3
	Total system input (MMBtu)	15.4
	Total system output (MMBtu)	10.0
	Overall system efficiency	65%
	Number of on/off cycles	3,950
	Total cost of gas	$205
	Total cost of electricity	$20
	Total utility cost	$224

One big difference between House #3 and the other two homes is the direction of heat transfer between the conditioned space and the basement (see Figure 24). In House #1 and House #2, heat flow was from the conditioned space to the basement. The opposite was found for House #3.



(Values in black are from measured data; values in red are estimates based on a combination of measured and calculated parameters).

Figure 24. Energy flows for House #3

The reason for this direction of heat transfer stems from the high level of system losses that come primarily from the indirect tank, exposed plumbing of the primary loop, and the buffer tank when in use. These losses can easily be detected in the data. The data in Figure 25 are from a day in October when there was no call for space heating. The activity seen in the top graph is due to the boiler heating the indirect DHW tank to maintain a temperature of 130°F. The input is strictly being used to replenish standby losses. There is no call for DHW during this period. Approximately 6,000 Btu are being consumed for each episode, or about 18,000 Btu/day.

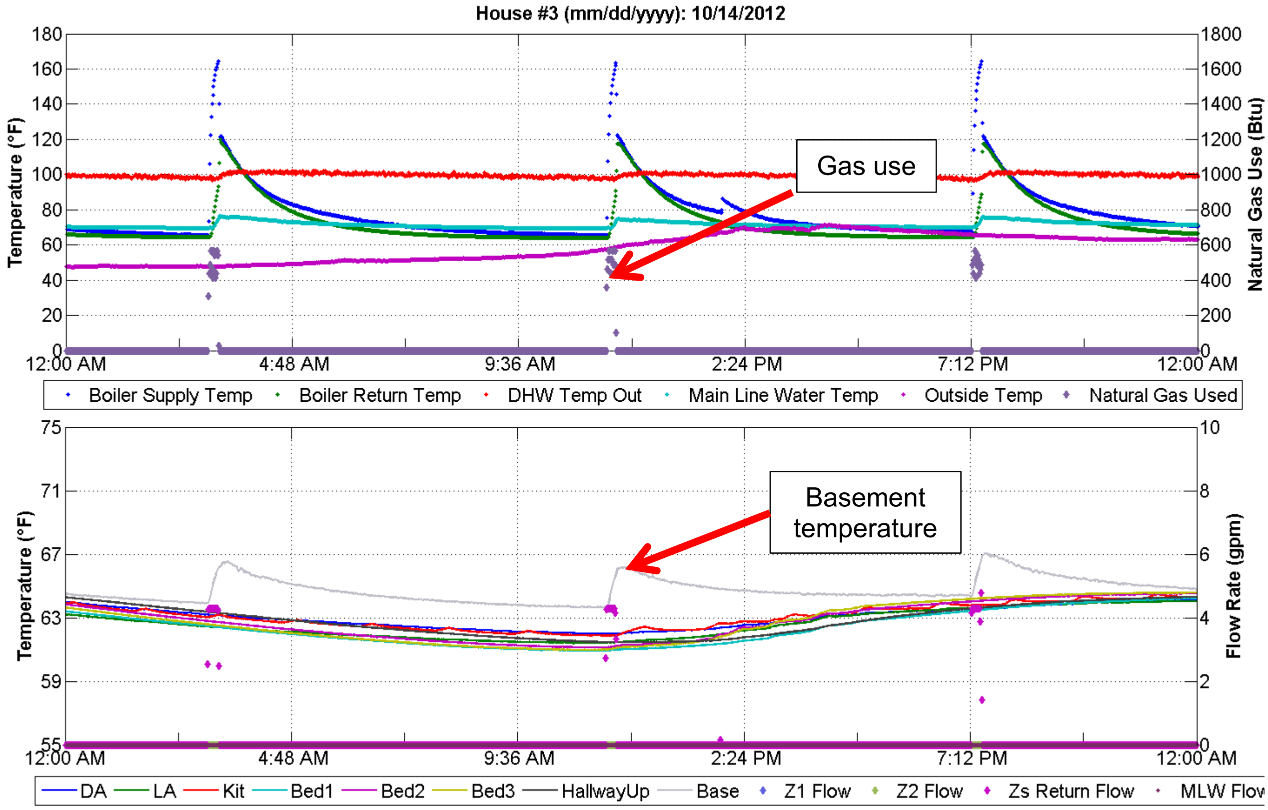


Figure 25. Example of gas consumption needed to maintain the indirect DHW tank temperature at 120°F

Note the corresponding increase in basement temperature in the lower graph (light gray line). Every time the tank is heated, the basement temperature increases about 2°–3°F. This activity increases as the basement temperature gets cooler, and vise versa. It should be noted that during periods of setback, the basement cools down with the rest of the home if the system is not running. While this saves on space heat, it actually results in increased standby losses from the tank and more frequent boiler operation to maintain the tank's internal temperature.

The effect that boiler operation has on the basement temperature can be seen in Figure 26. During periods of setback, the average basement temper was 67°–68°F with typical ranges from the low 60s to the low 70s, as compared to periods of constant temperature control where the temperatures typically remained at 69°–70°F the entire time. The hydronic system losses to the basement were so substantial that calculations predict a positive heat flow from the basement to the conditioned space. Reduced run time of the heating loop to zone one and the comparison of total output (8.7 MMBtu) to the calculated heat loss (10.3 MMBtu) for the entire monitoring period, support the assumption that the heat loss from the system to the basement was at least, in part, being transferred to the first floor. Other than solar gain, there are few internal gains available to offset the load.

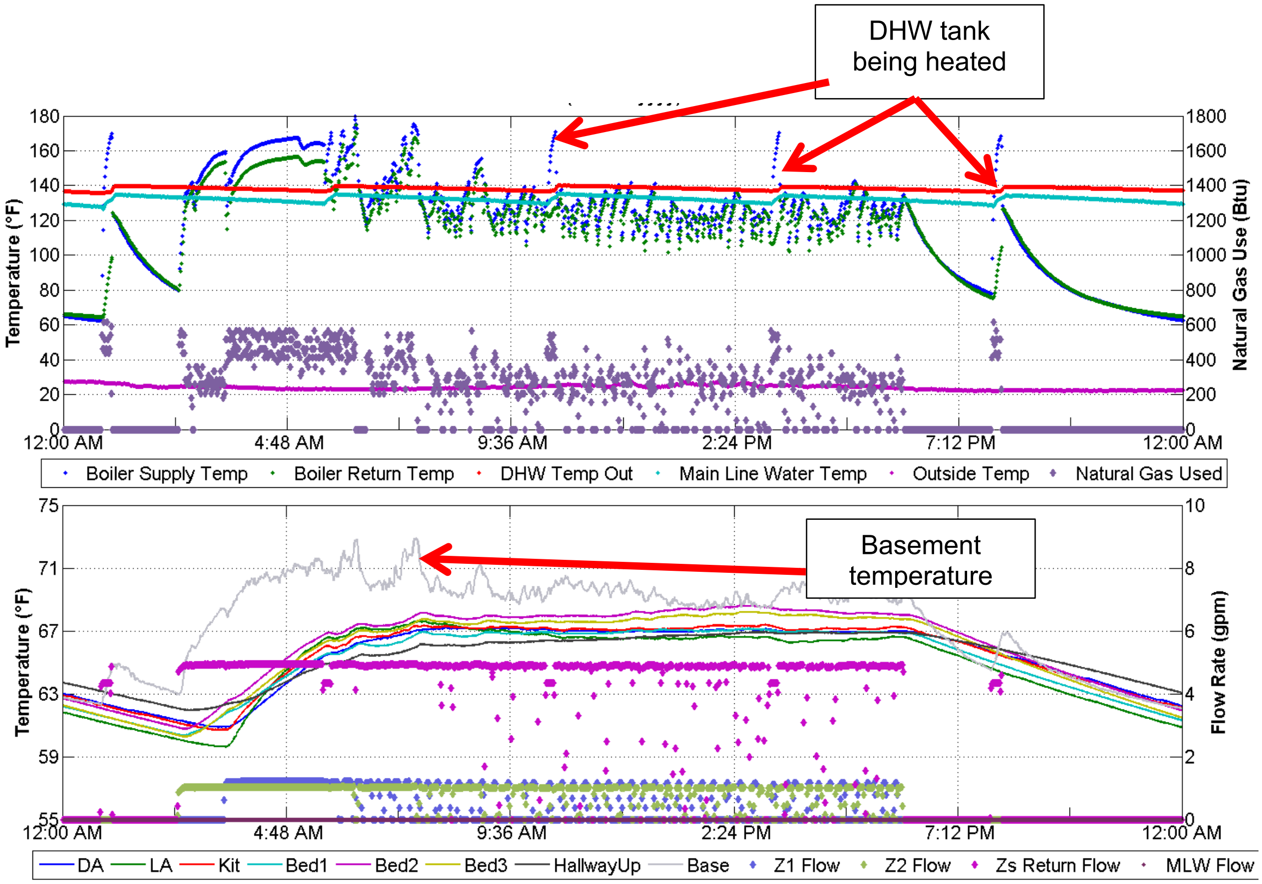


Figure 26. Test period #7 showing hydronic system behavior with thermostat setback and boost controls in House #3. Increases in basement temperature are notable during boiler operation.

5.3.2 Comparison of Different Operating Conditions

Tests were conducted to evaluate four major configurations:

1. Constant thermostat settings combined with a buffer tank for hydraulic separation
2. Nighttime thermostat setback with boost override of the outdoor reset sensor combined with the buffer tank
3. Constant thermostat settings with a primary/secondary loop
4. Nighttime thermostat setback with boost override combined with a primary/secondary loop.

Total input normalized for HDDs for each of the 7 test periods is displayed in Figure 27. A key for the x-axis legend is given on the same page. A definitive conclusion from these data is that when operated in setback mode, the system consistently used less energy than constant temperature operation, even though operation of the boost control regularly results in return water temperatures above 130°F. In fact, this occurs about 50% of the time the boiler is operating during periods of recovery from setback. During periods of operation at constant temperature, the return water temperature stays below 130°F more than 90% of the time, and yet, constant temperature operation still results in increased energy use. Maintaining the space at a constant temperature uses more energy than operating in setback mode even though the boiler is running at a slightly decreased efficiency.

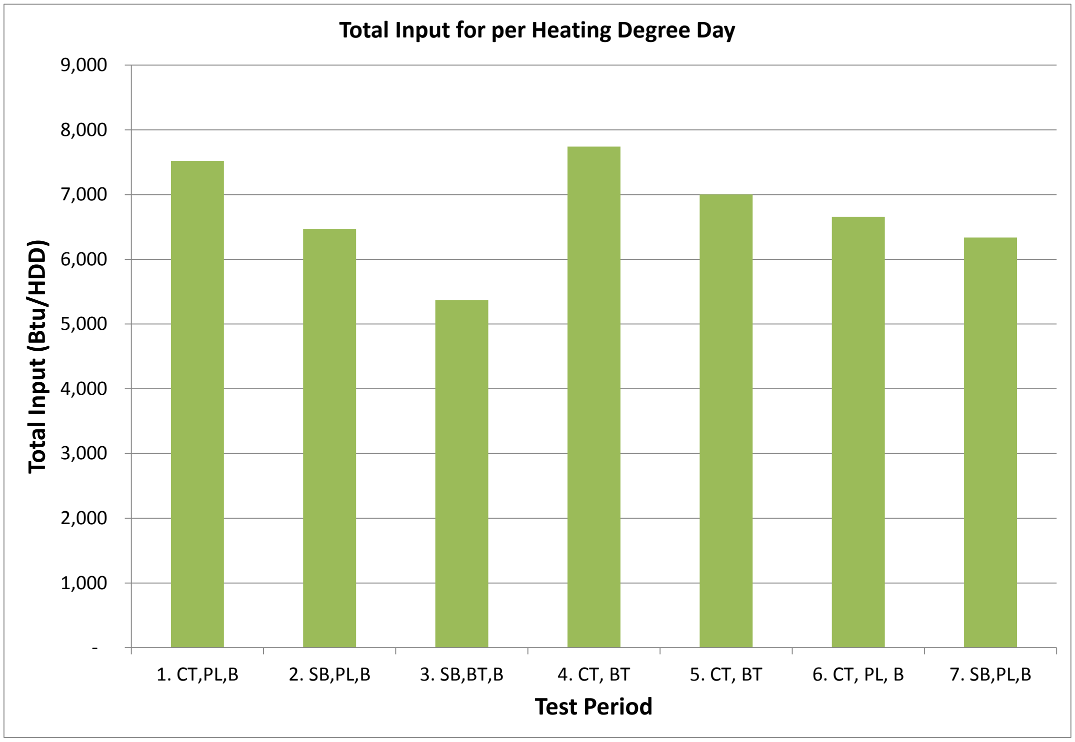


Figure 27. Total hydronic system input per HDD per test period for House #3; pump and boiler power is included

Figure 28 shows the space heating input per HDD for House #3. Beside the changes made between control strategies and plumbing configuration, variations in input per HDD from period to period are due to:

1. Decreases in the pump flow to both space heating zones at the beginning of period 4
2. Increases in efficiency as HDDs increase (standby losses become a smaller percent of overall load)
3. Increased activity inside the home in period 3 most likely due to continued construction efforts
4. High solar gain in period 3, as was evidenced in the room temperatures on the south side, especially on the second floor.

Even though these items affect system operation and efficiency, it is very clear that operation in setback results in lower energy use. Differences between primary loop operation and buffer tank operation are not that obvious.

Legend Key for Graphs:
1st Set of Letters: CT = Constant Temperature, SB= Setback
2nd Set of Letters: PL = Primary Loop, BT = Buffer Tank
3rd Letter: B = Boost

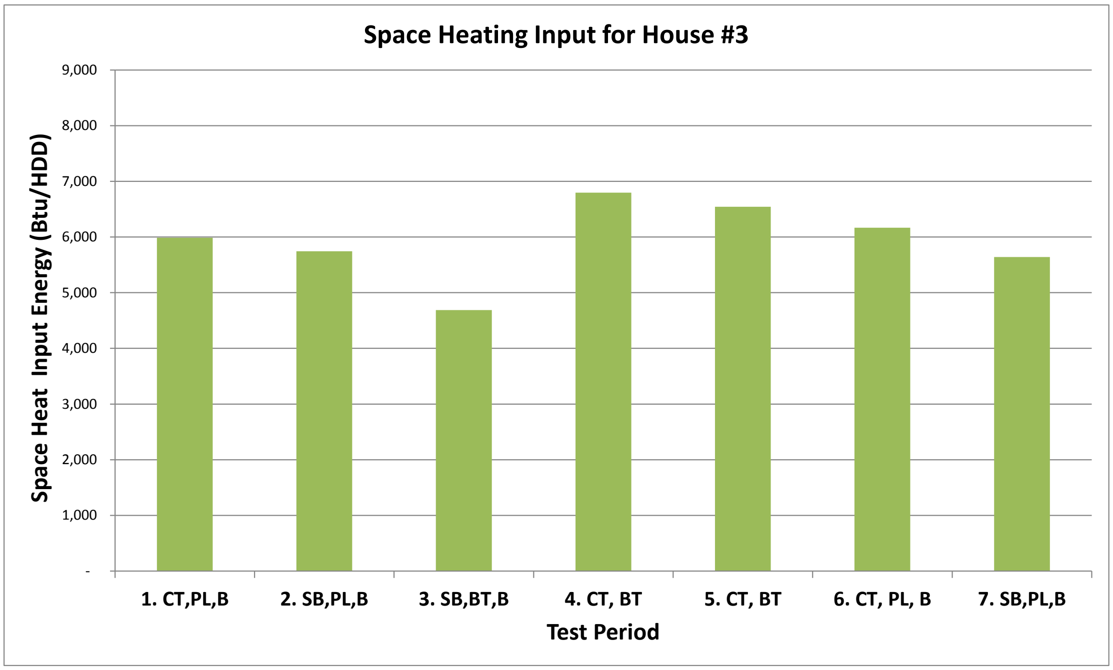


Figure 28. Space heating input for each test period for House #3; pump and boiler power is included

As stated above, the pump flow to both space heating zones was reduced from 1.5 gpm to approximately 1.0 gpm at the start of period 4. It was suspected, for the same reasons as discussed under results for house #2, that the baseboard on the second floor was not delivering the rated output listed in the manufacturer's charts. Since the baseboard ratings are given for 1.0 and 4.0 gpm, the flows were reduced to more accurately compare actual output to rated values. This reduction in flow rate resulted in an increase in overall space heating energy use despite the fact that the pump power was reduced from 40 to 18 W. A 30% reduction in flow rate results in a lower average baseboard temperature and, hence, a reduction in baseboard output capacity and a longer system run time.

As noted above, the four major variables altered during the monitoring period included primary loop versus buffer tank configuration and constant temperature control versus setback operation.

The data were grouped for each of these four major differences and are provided in Table 18. All values have been normalized by the total number of HDDs in each data group.

Table 18. Comparison of Performance for Each System Configuration in House #3; All Values Normalized per HDD

End Use	Variable	Control Settings			
		Primary Loop	Buffer Tank	Constant Temperature	Setback
Space Heat	Gas input (Btu)	5,729	5,846	6,274	5,251
	Space heat pumps (kWh)	59	75	80	52
	Boiler electricity (kWh)	14	16	18	11
	Total input (Btu)	5,857	6,002	6,441	5,365
	Output to zones (Btu)	3,801	3,702	3,999	3,492
Total System	Gas input (Btu)	6,408	6,502	6,935	5,929
	Total system electricity (kWh)	44	50	53	40
	Total input (Btu)	6,558	6,674	7,116	6,066
	Total output (Btu)	5,493	6,018	6,309	5,116
	Overall system efficiency	66.8 ± 12%	67.0 ± 11%	66.6 ± 12%	67.2 ± 11%
	Space heat efficiency	72.5 ± 11%	69.8 ± 11%	69.7 ± 11%	73.2 ± 11%
	Boiler heating efficiency	95 ± 5%	99 ± 4%	98 ± 5%	96 ± 5%
	Estimated heat transfer to Z1 from base (Btu)	442	489	490	436
	Space heat runtime (min)	29	36	41	22
	# on/off cycles	2.230	1.082	2.223	1.133
	Total cost	$0.095	$0.098	$0.104	$0.088

Operation in setback mode results in the lowest gas input, lowest overall cost and the highest space heating efficiency. Boiler cycling is also significantly reduced. The lower room temperatures at the beginning of the recovery period result in greater differences between the boiler's supply and return temperatures than are seen during constant temperature operation. As long as a minimum temperature differential is maintained, the boiler will continue to fire, reducing short cycling. In this study, operating the system with a setback (as opposed to constant temperature operation) resulted in longer periods of time when the supply and return temperature differentials remained larger than the boiler's minimum limit for firing. Therefore, the boiler short cycled less.

The difference in the results for primary loop versus buffer tank operation are much too small to make a definitive conclusion on which configuration is more efficient. It is clear, however, that buffer tank operation reduces system cycling by 50% or more over primary loop operation.

Three different efficiencies are given in Table 18 along with the corresponding error associated with the sensors used. The error ranges were calculated using the accuracies of the sensors shown in Appendix A, and show potential errors ranging from 4% to 12%. For more details on the uncertainty calculations refer to Taylor (1939).

Overall system efficiencies were calculated by adding the output to the space heating zones and the DHW and dividing by the total input—gas and electric. The values are comparable for all four operating modes shown and indicate that more than 30% of the total input to space heating and DHW is being lost. Even with if the most conservative values are calculated using the accuracy percentages, system losses still approach 20%.

Space heating efficiencies were calculated by adding the output to each space heating zone (loss to the basement was not included) and dividing that total by the total gas and electricity input associated with space heat operation only. DHW was not included. The space heating efficiencies are better than the overall system efficiencies because DHW input is currently all standby losses, due to the fact that there are no occupants and no DHW use.

Boiler efficiency was calculated by dividing the total boiler output by the total system input. Boiler output was calculated by multiplying the temperature difference of the supply and return water right near the boiler and multiplying that by the flow rate through the primary loop/buffer tank. This calculation results in boiler efficiencies ranging from 95% to 99% because it ignores all but combustion losses. These values are consistent with measurements taken with a combustion analyzer during system commissioning at the start of the monitoring period.

5.3.3 Comparison of System Performance for All Three Homes

When comparing all three systems, it is important to consider the following facts:

1. The design loads and orientation (and hence the solar gain) for each house are very similar.
2. House #1 was occupied for the entire test period providing internal gains and resulting in better DHW efficiencies.
3. House #2 was occupied during periods 6 and 7.
4. House #3 was unoccupied for the entire test period.

Note that House #1 is discussed in this section; however, it was eliminated from any comparison of system efficiencies because it is strongly suspected that the gas meter is reading 30%–50% below the actual gas use. Information from the bills and exterior meter readings support this conclusion, as do simple energy balance calculations, as discussed in Section 5. Further investigation into the accuracy of this meter is underway. Table 19 provides a snapshot of each of the three mechanical systems, their output capacities, and the design load for each home.

Table 19. Summary of System Designs

Component	House #1	House #2	House #3
Space Heating	92.5% AFUE combination condensing boiler with radiant flooring	93% combustion efficiency condensing water heater with baseboard convectors	95.1% AFUE condensing boiler (buffer tank versus primary/secondary loop) with baseboard convectors
DHW	On-demand DHW heat exchanger	55-gal tank	Indirect, 37-gal tank
Space Heating Output (max/min)	112,601 Btu/h 35,452 Btu/h	100,000 Btu/h 10,000 Btu/h	46,000 Btu/h 14,720 Btu/h
Design Heat Load	16,047 Btu/h	14,695 Btu/h	15,426 Btu/h

Note that the minimum output capacity of the boiler in House #1 is more than twice the design load. The effects of this can be seen in Table 20. Over the entire monitoring period, the number of on/off cycles in House #1 is more than three times that of House #2 and more than four times that of House #3. Standard startup procedure for the boiler in House #1 is to start up at maximum fire. When this happens, the supply set point temperature from the outdoor reset control is quickly satisfied before the boiler modulates down to a lower rate.

Table 20. Comparison of System Performance

	House #1	House #2	House #3
System Runtime (h)	1,087	1,403	1,234
Pump Electricity (kWh)	146 (60–65 W)	79 (20–22 W)	62 (16–18 W)
Boiler Electricity (kWh)	49	95	34
# of On/Off Cycles	16,314	5,758	3,950

[1]Watts in parentheses shown for zone pumps only, not primary loop or DHW.

Short cycling decreases efficiency in a couple of ways. First, fixed losses such as radiation losses from the jacket and pipes are amplified under part load conditions (D&R International 2012). For example, if jacket losses are 2% of the input at full load, at half load they would be 4%, and so on. Second, pre- and post-purge losses are increased. Purge occurs during fan operation at the beginning of the firing cycle as well as at the end as the fan removes any combustible gas mixture that may have accumulated. Consequently, heat is removed from the boiler as the air is purged. House #1 would see three to four times more losses than the other two systems due to this part load operation.

During the initial install, the maximum firing rate of the heating system in House #2 was lowered from 130 kBtu/h to a firing rate of approximately 80 kBtu/h. Based on system performance, the firing rate could have been lowered even further. This lower firing rate combined with the extra mass in the system reduces the cycling rate drastically compared to House #1.

The plumbing configuration in House #3 was alternated between a buffer tank and a conventional primary/secondary loop configuration. Cycling during buffer tank operation is 50% less than that during primary loop operation. Even in primary loop operation, cycling is significantly less than in House #1 because the maximum space heat firing rate was reduced to approximately 26 kBtu/h, preventing the boiler from firing at its maximum capacity of 50 kBtu/h. This prevents the boiler from overshooting the set point temperature and the resulting cycling that would come from that.

Table 20 also lists the total pump power consumption in each system. The pumps in House #2 and House #3 have ECMs and use significantly less energy than the pumps in House #1. Figure 29 shows the pump power consumption per test period for all three homes, and Figure 30 provides the pump run time during space heat operation. Even during periods when the system in House #1 ran less than the other two homes, the pump energy consumption is significantly higher. During periods 4 through 7, House #2 and House #3 either ran for a longer or a similar amount of time, but still used significantly less energy. Switching to similar pumps would save approximately \$53/year at \$0.18/kWh. At \$80 increased cost per pump for three pumps, the simple payback would be approximately 4.5 years.

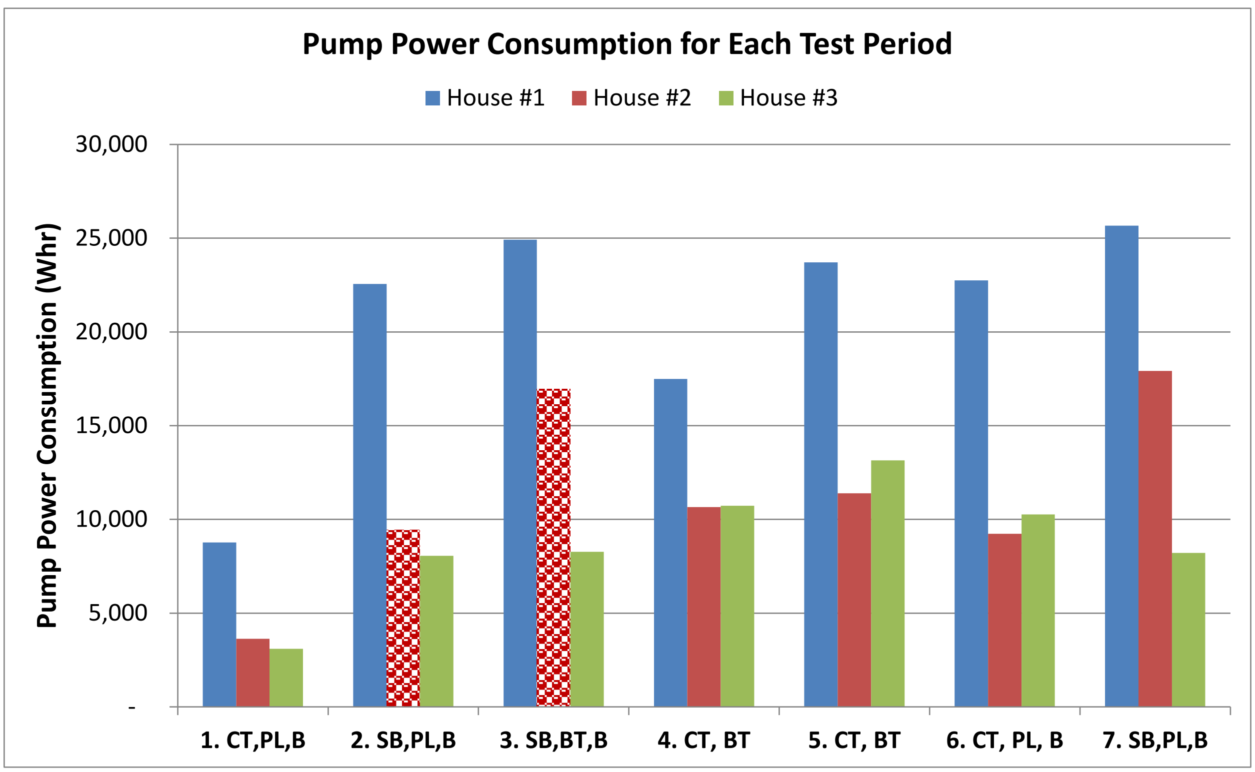


Figure 29. Total pump electricity consumption for space heating for each test period

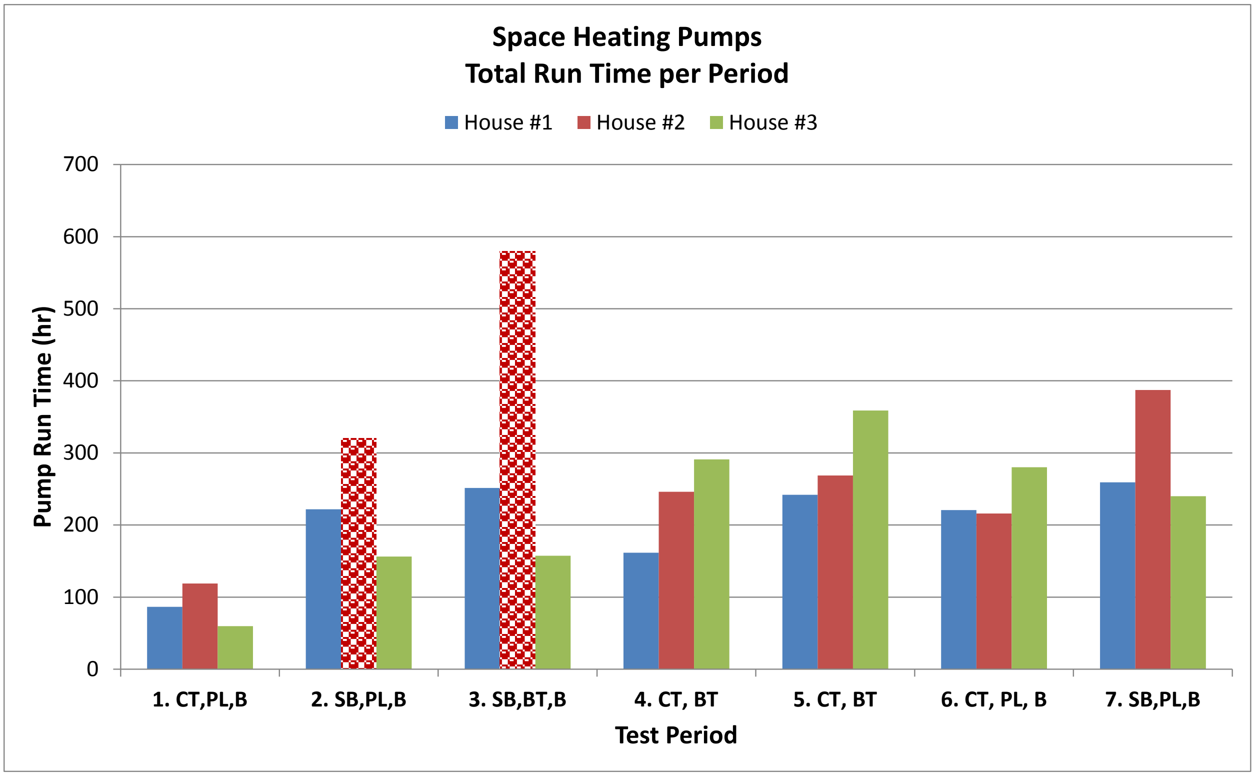


Figure 30 Total pump runtime for space heating for each test period

Figure 31 shows the total input energy normalized per HDD. Input includes all gas used for space heat and DHW generation as well as consumption of the pumps and auxiliary electricity of the boiler controls and combustion fan.

Differences in input for House #2 are related to an increase in the reset curve temperatures between period 3 and period 4, occupant water use in periods 6 and 7, and an increase in thermostat settings, particularly in period 7. Differences for House #3 are directly related to thermostat settings and use of the buffer tank versus the primary loop. Results consistently show that constant temperature control results in increased gas use per HDD as compared to setback.

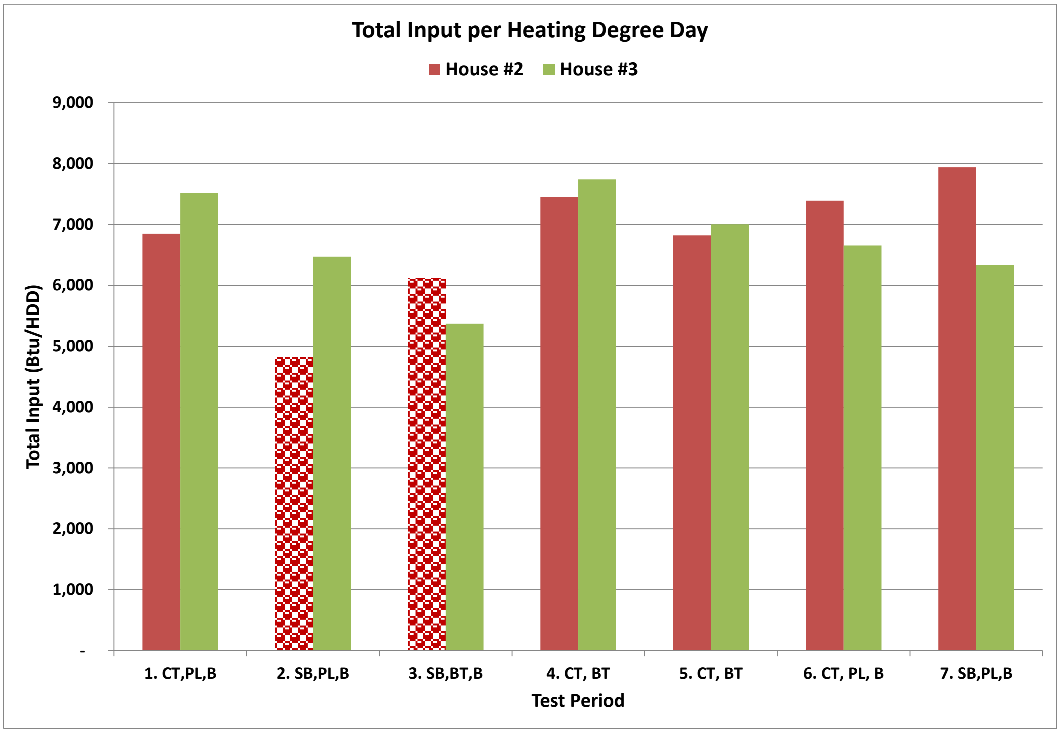


Figure 31. Total input energy for DHW and space heating per HDD for each period. Labels along the x-axis apply to House #3.

Figure 32 displays the input energy per HDD associated with space heating only. Control strategies employing setback perform the best in House #3, even when boost controls override the outdoor reset control.

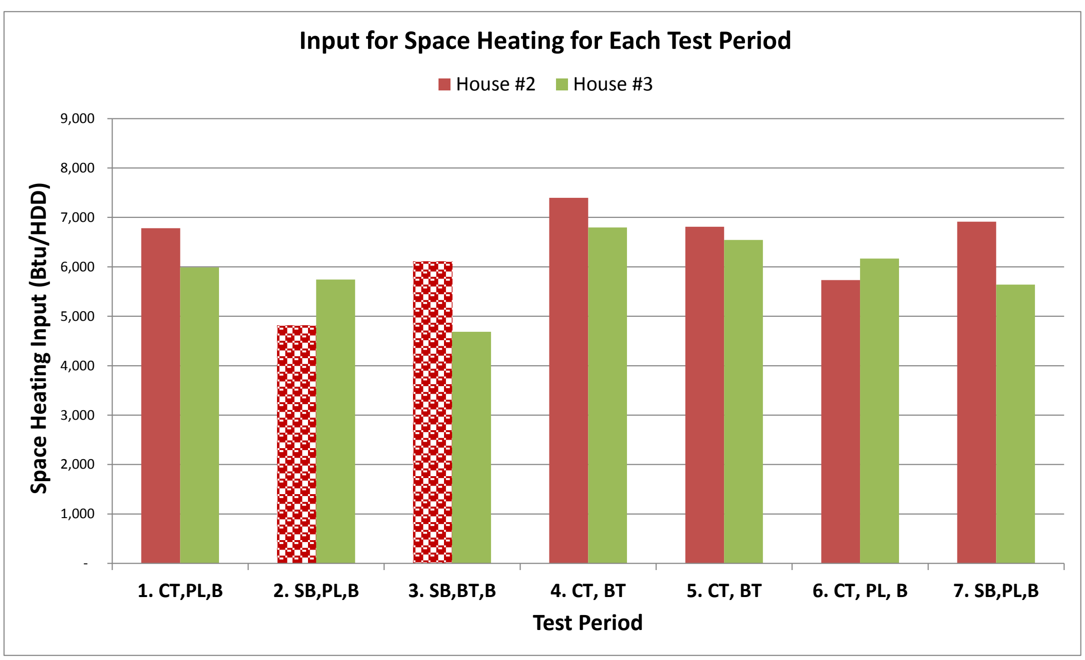


Figure 32. Input associated with space heating only; pump and boiler power included

Idle losses played a large part in the overall performance of the systems. Because House #2 and House #3 have storage tanks that need to be kept at a constant temperature, there will inherently be more standby losses with these two systems as compared to House #1, which heats the domestic water with an on-demand DHW heat exchanger. Therefore, if there is no call for DHW, the only standby losses will be from the length of pipe to the end use directly after a DHW call and losses from the boiler through the jacket and flue.

Although these losses could not be calculated for House #1, they can be seen in Figure 33. When there is a call for DHW, spikes in basement temperature are common. The light blue line in the lower graph of Figure 33 is the basement temperature. The red line in the upper graph is the DHW supply temperature. There is a definite correlation between DHW operation and basement temperature.

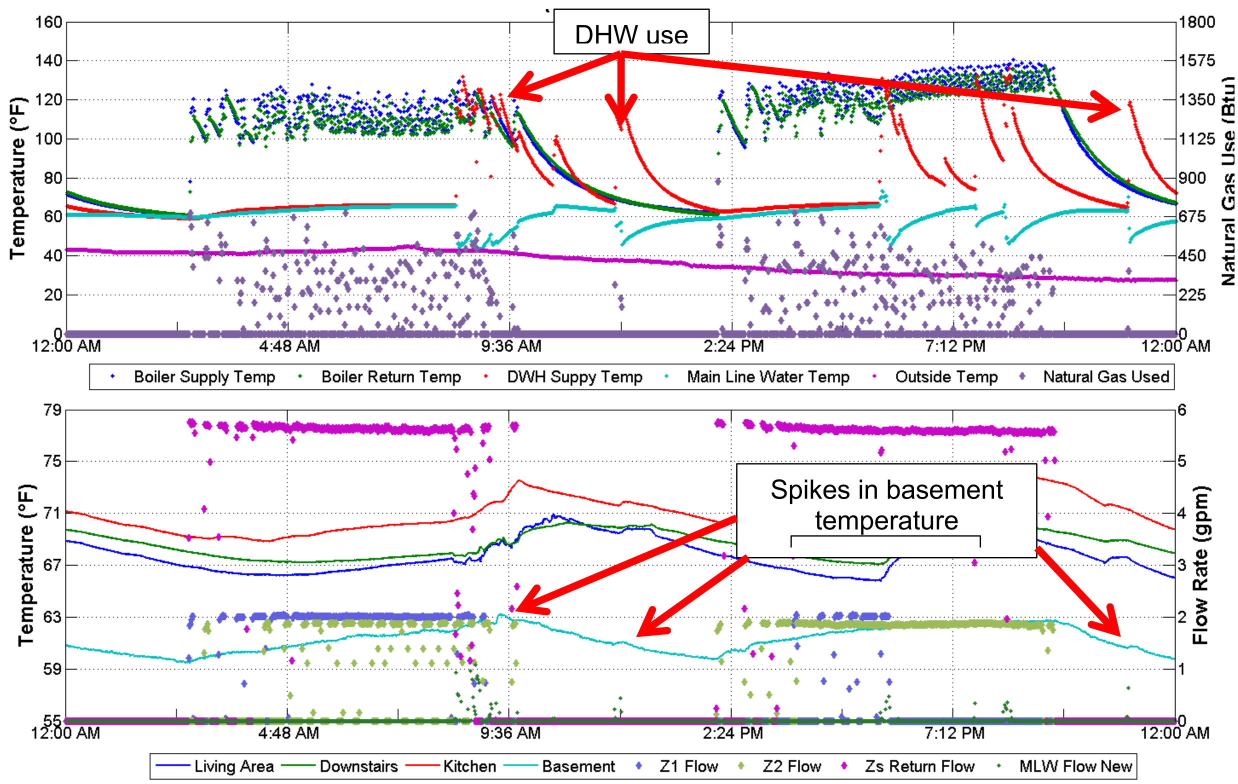


Figure 33. Temperature interactions in basement in House #1 during DHW calls

Table 21 summarizes the energy required to replenish standby losses from the storage tanks in House #2 and House #3. Periods with no hot water draws and no space heating activity were examined to isolate these values. As seen in Figure 23 and Figure 25, replenishment of standby losses occurs approximately three times per day for each house and increases or decreases corresponding to the basement temperature.

Table 21. Energy Used To Replenish DHW Standby Losses

	House #2	House #3
Energy Consumption per Replenishment Cycle[a] (Btu/cycle)	2500	6000
Annual Energy Consumption (MMBtu/yr)	2.7	6.57
Annual Cost[b]	$38	$92

[a]Measurements take for a tank temperature set point of 120°F in House #2 and 130°F in House #3.
[b]Assumed $1.34/therm.

For House #2, the losses listed in Table 21 should be seen as minimum values. At the time these values were recorded, the tank was being heated to 120°F (the minimum setting for DHW), but the tank temperature rose to a maximum of 155°F in this study. Standby losses increase the warmer the tank is kept. These losses can be reduced by insulating hot water lines into and out of the tank, insulating all exposed space heating piping in the basement, and increasing the baseboard capacity to allow the system to run at a lower temperature.

The DHW tank set point temperature in House #3 is 130°F. DHW standby losses in this home could be reduced by lowering the tank temperature to 120°F and insulating the hot and cold water lines into and out of the tank as well as all exposed piping running to and from the boiler. Given the number of times per day the heat in the tanks is replenished, additional tank insulation would benefit both houses.

Resting electric energy consumption for each of the three systems is minimal and costs less than $2.00 for the monitoring period (Table 22).

Table 22. Resting Power Consumption for All Three Homes

	House #1	House #2	House #3
Resting Power Consumption (W)	8.4	12.5	13.1
Costs Over Monitoring Period	$1.22	$1.62	$1.77

Figure 34 compares overall system efficiency calculated for House #2 and House #3 for each of the seven test periods. Overall system efficiency was calculated by adding the output to each zone to the output to the DHW end uses, and dividing that total by the total input into the boiler. Total input includes the gas input and the electrical power input to all pumps and the boiler. Losses to the basement were considered losses, and were not included in the system output.

House #3 is shown with and without an adjusted efficiency. When evaluating the results, it became apparent that the output calculated to the first floor zone was too small to meet the load. From further investigation and more in-depth heat transfer calculations, it became clear that the basement was warmer than the first floor and some of that heat was rising up through the floor to offset that load. On average, calculations predict that approximately 160,000 Btu per 2-week period are transferred through the basement ceiling to the first floor. The overall system efficiency increases by about 7.5% over the entire monitoring period when this heat transfer is taken into account.

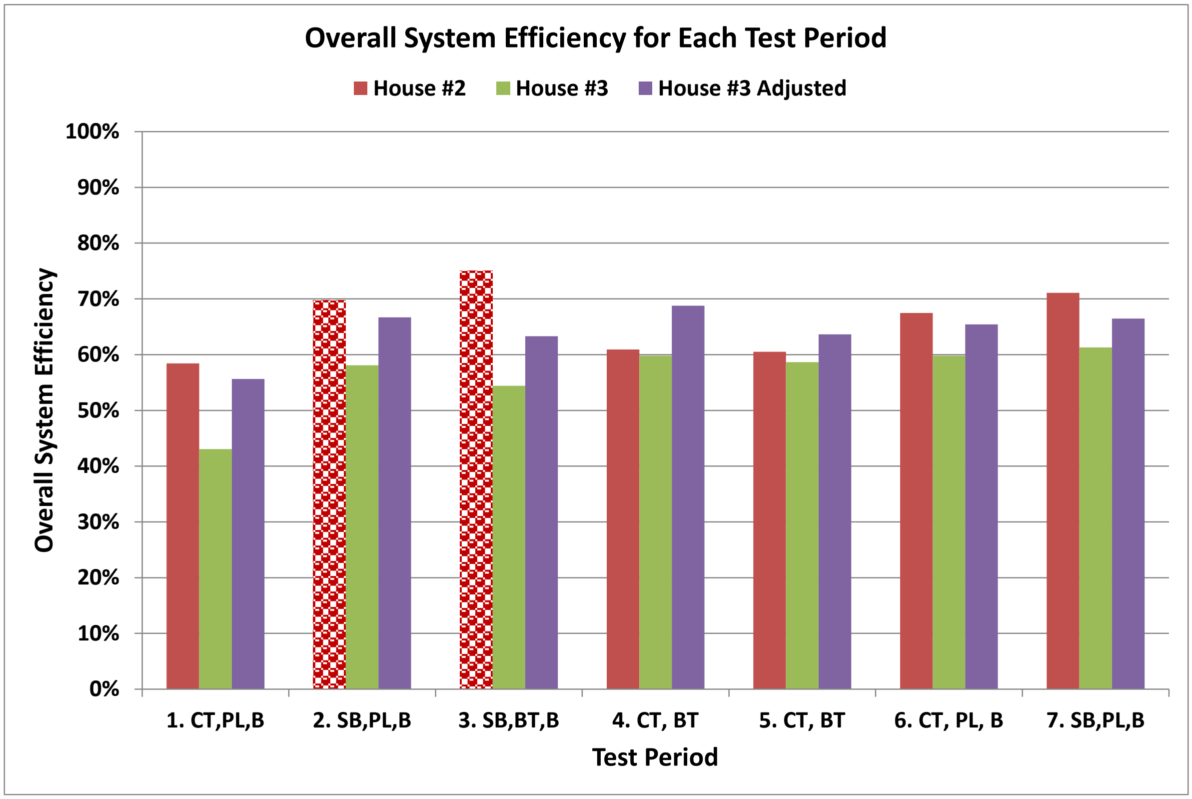


Figure 34. Comparison of overall system efficiency for space heat and DHW for each test period. Values include pump and boiler power and exclude heat lost to the basement.

In general, both systems performed similarly and overall efficiency increased as both the space heating and DHW loads increased. Under higher loads, standby losses became a much smaller portion of the overall load. Tested combustion efficiencies for the Versa Hydro ranged from 89% to the upper 90%'s depending on the tank temperature being maintained. Disregarding periods 2 and 3 for House #2, the overall system efficiencies range from the mid-50s to the low 70% for both homes, indicating losses of 30% and higher. The argument over whether or not losses to the basement should instead be considered useful heat output is sure to be a lengthy one. Including those losses in the calculations would result in much higher overall system efficiencies. But, how much loss to the basement is OK? What basement temperature is considered sufficient before it is excessive and wasteful? Although determining the answers to these questions is outside the scope of this research, one thing is very clear from this study: losses from exposed piping and storage tanks are significant and should be minimized.

A comparison of the utility costs per HDD for each test period is shown in Figure 35. These results show that setback operation for House #3 results in the least cost option. Operating costs for House #2 and House #3 during periods 1, 4, and 5 show nearly identical results. Costs increase for House #2 in periods 6 and 7 after the occupants moved in. This is due to increased thermostat settings and DHW use.

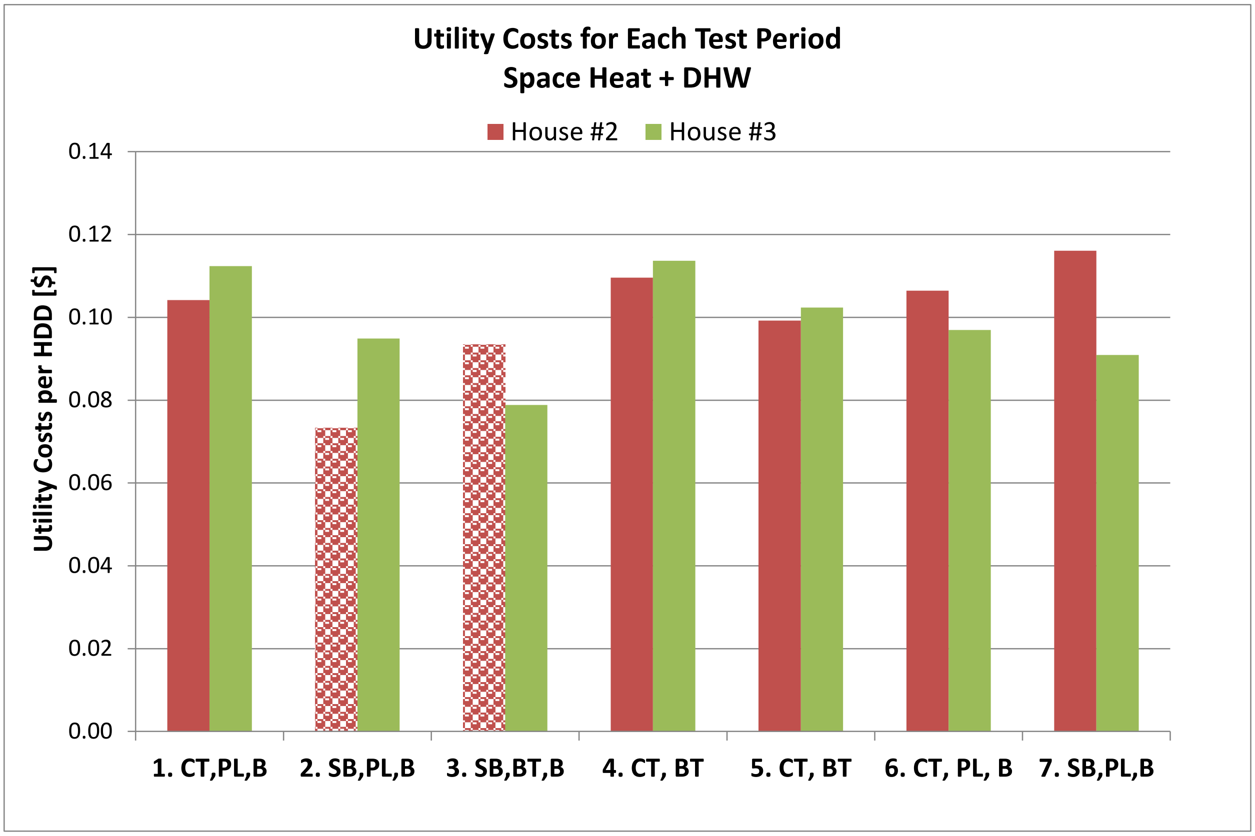


Figure 35. Utility costs per HDD for each test period

6 Discussion

6.1 Adjustments to Calculation of Space Heat Output and Accuracy

The basic equations used to calculate output to the space heating zones were provided in Section 4, but this basic equation doesn't account for the lag between the time the water is heated by the boiler and that same water is returned to the heat exchanger. If this lag is not accounted for, errors in calculated output can be significant. To determine the system lag, the length of piping was calculated from the drawings and multiplied by the velocity in the loop. The resulting number of minutes was then used to find the corresponding return temperature at time t+lag, which was then used in the output calculation. This calculation was performed for each space heat loop and the main supply and return to the boiler. While not 100% accurate, this calculation results in a more realistic map of system performance.

6.2 Decreasing System Losses Through Proper Design

The losses calculated in House #2 and #3 were quite significant. Simple solutions such as insulating exposed piping and storage tanks could easily be implemented to help reduce these losses. Other solutions should be implemented during the design phase and include reducing excess piping where possible and properly sizing the boiler and the baseboard to reduce cycling and system run time. Because the baseboard in these homes on the second floor did not perform as specified, short cycling of the boiler was increased along with system runtime. In addition, supply temperatures had to be increased to meet the loads and to recover from thermostat setback.

Increased baseboard capacity would have resulted in the following improvements in performance and efficiency:

- Shorter system runtime equating to less pump and boiler fan energy consumed and decreased standby losses.
- Boosting to such high temperatures would not have been necessary because the system would have recovered faster.
- Lower temperature operation, which in turn would have resulted in less heat loss through pipes and storage tanks used for space heating.
- Short cycling would have been decreased because the temperature difference between the boiler's supply and return temperatures would have been larger preventing the burner from turning off.
- Lower return temperatures to the boiler would have resulted in increased combustion efficiency.

Research conducted by Brookhaven National Laboratory has shown that oversizing the baseboard by 1.5–2 times the design load will result in condensing operation even if the boiler's supply temperature is 180°F (Butcher 2006). Based on this and past research, it is recommended to oversize the baseboard by 2 times at a design temperature of 150°F. Past research efforts showed recovery from setback took 2–4 h for an 8°F setback and double the baseboard. If this configuration is combined with a boost control that can override the outdoor reset sensor, recovery time will be drastically reduced. With the limited baseboard capacity, on the second floors in the homes in this study, boost control resulted in recovery times of approximately 3 h on the second floors and 2 h on the first floors for a 7°F setback. This time could be cut in half through proper design and the use of a boost control.

6.3 High Mass Versus Low Mass Operation

The systems in both House #2 and House #3 had added mass. A 55-gal condensing water heater was installed in House #2 and an additional 30-gal buffer tank was installed between the boiler and the space heating zones in House #3. Because the system in House #2 already provides hydraulic separation, it didn't require the standard primary/secondary loop plumbing typically found in low-mass systems. This means that one less pump and less piping were required as compared to a low-mass boiler with an indirect tank.

Due to the extra plumbing and labor involved, the system in House #3 resulted in added costs over the primary/secondary loop configuration. There was no reduction in the number of pumps and piping needed, and the buffer tank introduced an additional cost.

Boiler cycling, pump energy use, and system response time were evaluated along with overall system efficiency during each mode of operation (see Table 18). The differences in these values were not significant enough to conclude that one operated more efficiently or resulted in lower operating costs. The buffer tank definitely reduced cycling on the order of 50% compared to the primary/secondary loop configuration, but that is the only solid conclusion.

The boiler in House #3 was also equipped with additional controls intended to reduce cycling when used in primary/secondary operation. The first, limiting the boiler's space heating input was discussed in Section 3.1.3 and is intended to prevent the boiler from going to high fire on start up, limiting the speed at which the supply temperature will reach the set point. The second control reduces or increases the boiler's response time. The slower the response, the longer it will take for the supply temperature to reach the set point. Although buffer tank operation resulted in the least amount of cycling, both these controls were in place for buffer tank operation as well. Therefore, operation in that mode benefitted from these extra controls and not only the benefit of the mass. In hind sight, a comparison of buffer tank operation set up with the factory defaults should have been made to the primary/secondary loop with these controls optimized for this home.

The benefits of reduced cycling, however, are not well documented. There are claims that short cycling reduces boiler life and efficiency, but no conclusive evidence was found. There is also conflicting opinions about the effects. Two different DOE publications were found: one stating cycling had little effect on efficiency (Henderson et al. 1999), the other claiming a significant effect (D&R International 2012). There are no conclusive results from this research that support the claims that cycling reduces efficiency.

One conclusion that can be drawn is that increases in mass temperature result in increased standby losses. For House #2, an increased tank temperature also meant reduced condensing efficiency because the combustion chamber is located in tank. When the tank temperature was raised over 130°F, condensing was seriously reduced. Systems such as this should be designed to provide space heat at very low temperatures.

6.4 Comparison of Predicted Results Versus Measured Data

To investigate the accuracy of BEopt with respect to measured data, gas, pump power, and total operating cost were compared to that of the model for each hydronic system. BEopt uses Typical Meteorological Year 3 data, which were close in value to the measured outside temperatures. Measured gas input for the hydronic system in House #1 was not compared with model results because of the possibly faulty gas meter, as mentioned in Section 5.1. The hydronic system for House #3 was compared to the modeled results for two thermostat settings: setback and constant temperature.

Figure 36 shows the average measured and modeled gas input per HDD for House #3 in constant temperature and setback modes. The measured gas usage is higher than that predicted in the model for both thermostat settings. Possible reasons for this difference include:

1. BEopt does not consider standby and pipe losses. As such, the entire boiler output is assumed to be used for space heating and DHW. This is not the case in the measured results, as discussed in Section 5.
2. The home was unoccupied for the entire monitoring period and did not benefit from internal gains. The model assumes internal gains from occupants, lights, and appliances.

If the same internal gains that BEopt predicts are subtracted from the total measured input, the predicted and actual results are much closer. The "Actual Adj" bar in Figure 36 is the result of this calculation. Consistent with the measured data, BEopt predicts lower gas usage per HDD during setback operation.

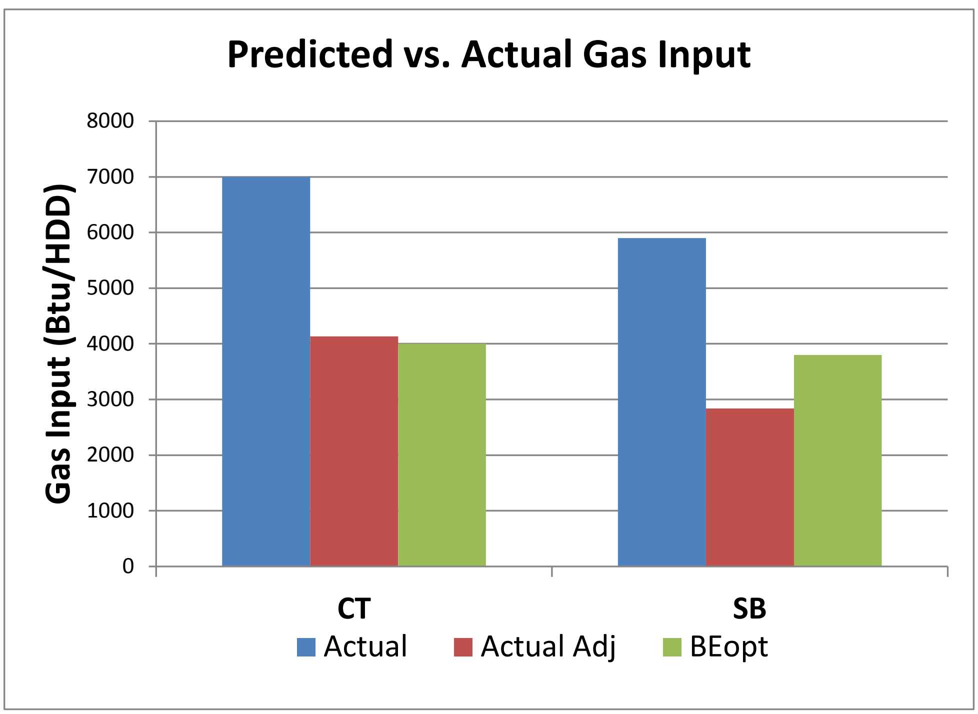


Figure 36. Measured versus predicted gas input per HDD for House #3

In terms of pump energy consumption, BEopt predicted lower pump energy consumption for all three houses compared to measured data (see Figure 37). This is primarily due to the fact that BEopt uses only one constant speed pump as opposed to four pumps in House #3, three in House #1, and two in House #2. The ability to enter the actual number of pumps installed and the pump specifications into the software would result in more accurate predictions. The model supports the measured data that setback operation results in the least pump energy consumption and lowest operating costs (Figure 38). Actual cost savings between constant temperature and setback operation were approximately 16% as opposed to predicted savings of 7%.

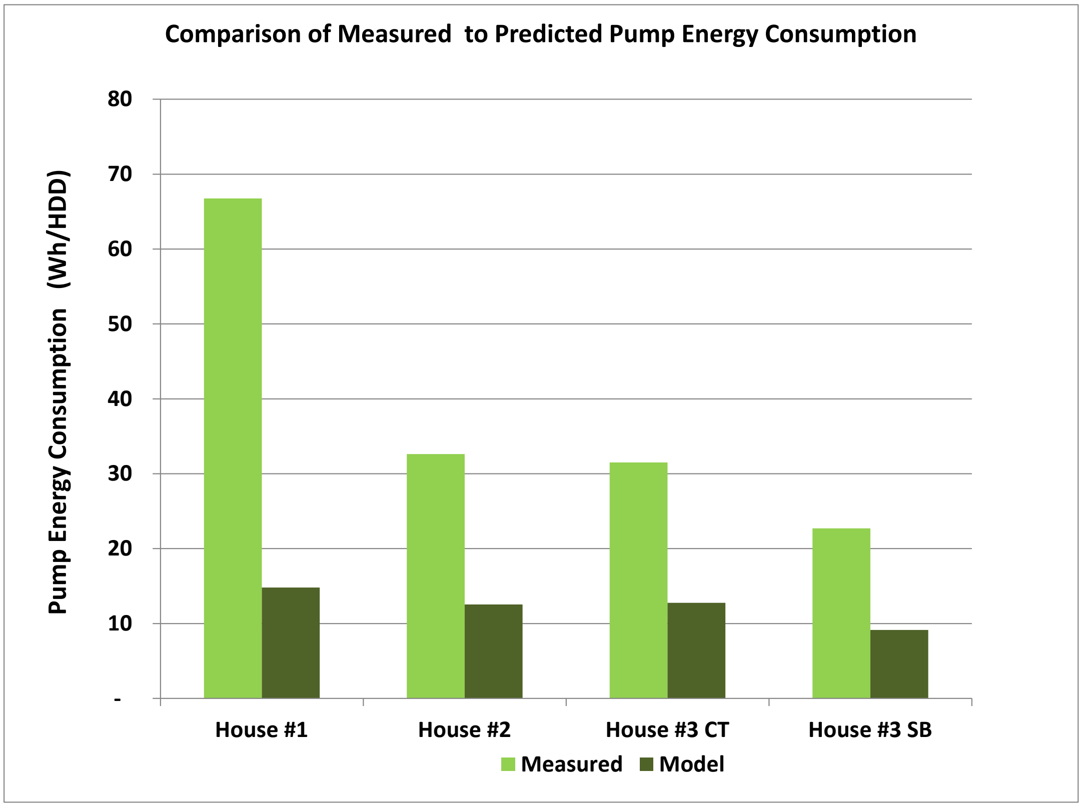


Figure 37. Measured versus predicted pump energy use

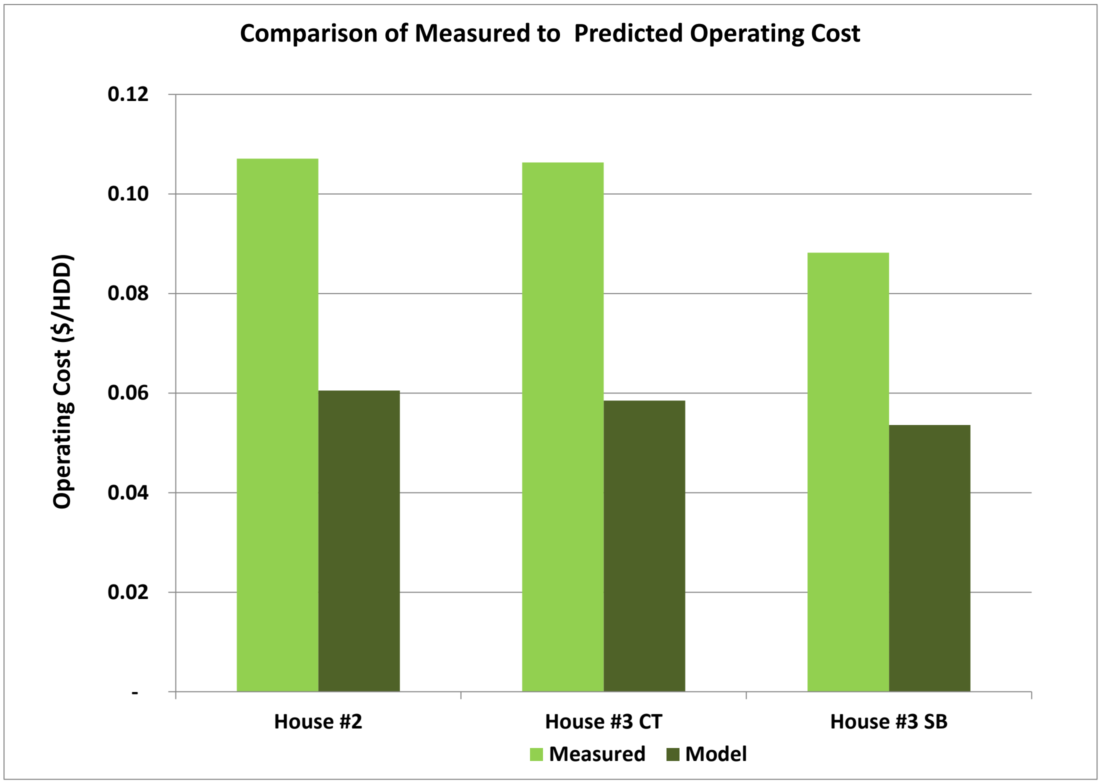


Figure 38. Measured versus predicted operating costs

7 Conclusions and Recommendations

Based on previous research efforts, it is apparent that hydronic systems are typically not designed and installed to achieve maximum efficiency, especially when paired with baseboard convectors: one of the most cost-effective emitters for hydronic systems. Part of the reason for this is that little information is available on combined system efficiencies where all the components in the hydronic system are considered. With help from builder partners, interested manufacturers, and hydronic designers, three different hydronic systems were designed, installed and monitored. Several control strategies, DHW production methods, plumbing configurations, and heating plants were evaluated. The goals were to assess several combinations of components and make recommendations for cost-effective, responsive, energy-efficient packages. Following are the research questions defined and the corresponding conclusions based on modeling and monitoring conducted over a 5-month period.

What combination(s) of components—pumps, high efficiency heat sources, plumbing configurations, and controls—will result in the highest overall efficiency for a hydronic system where baseboard convectors are used as the heat emitter?

Although measured boiler combustion efficiencies ranged from the high 80%'s to the high 90%'s, system interactions in both House #2 and House #3 resulted in similar overall system efficiencies—upper 60%'s to lower 70%'s. These low overall efficiencies are very conservative estimates because they do not include heat loss to the basement as useful heat. That energy was eliminated from the calculations because the basements in all three homes are unfinished, unconditioned spaces. Any heat to that space is unintentional. This point could easily be countered, but it is not the intention of this study to define useful heat. What was found was that system losses were primarily the result of three problems.

First, the baseboards installed on the second floors were not performing as anticipated for reasons discussed previously in this report. It is suspected that the overall system efficiencies would have been higher if the installed baseboard capacity on the second floors had been higher allowing for lower temperature operation. These systems were designed to meet the design heating load using 140°F supply water at 1.5 gpm. But because the output of the baseboard was lower than expected, supply temperatures had to be increased, thereby increasing gas input and standby losses. The biggest losses came from the storage tanks and the exposed piping in the basements. Second, House #2 and House #3 were unoccupied for most or all of the monitoring period. This means that any energy to maintain the DHW tank temperature has to be considered standby loss. When occupied, these losses will become a smaller percentage of the overall input resulting in higher overall system efficiencies. Lastly, the standby losses from the exposed piping and storage tanks in the basements were significant and exacerbated by the fact that the systems had to be operated at higher temperatures than expected to meet the heating demand and to recover from setback. These losses could be drastically reduced by insulation and lower temperature operation.

When operated in setback mode with a boost for recovery, the condensing boiler installed in House #3 resulted in the lowest operating costs and the highest overall efficiencies compared to the other configurations tested in that home.

It is anticipated that the boiler in House #1 with the on-demand DHW heat exchanger results in the most efficient DHW production due to the lack of standby losses from a storage tank, but the lack of data from the other two houses prohibits a definite conclusion. House #3 was unoccupied for the entire monitoring period and House #2 was occupied for only the last two test periods.

In general, the following combination of components would result in the highest overall efficiency when baseboard convectors are used:

- A condensing, modulating boiler properly sized for design loads and recovery from setback: for modulating, condensing boilers, design the system to meet the design heating load such that the return temperature does not exceed 130°F; if using a condensing water heater, design the system to meet the design heating load such that the tank temperature does not exceed 130°F.
- An on-demand DHW heat exchanger in the boiler to provide DHW, given the boiler is not maintained at a constant temperature.
- Baseboard sized for low temperature operation under design conditions and able to provide twice that capacity under boost operation to reduce recovery time from thermostat setback.
- Thermostat setback control with provisions for quick recovery such as boost control or proper oversizing of the boiler and baseboards.
- High efficiency pumps: pumps that have displays showing flow rate and energy consumption are particularly useful during commissioning to optimize performance and ensure design conditions are being met.
- Insulated piping and storage tanks, especially in unconditioned space.
- An outdoor reset control set up to provide supply temperatures that would result in condensing operation under design conditions.
- Controls that result in a purge to the DHW tank after a space heat call (if DHW is provided by an indirect storage tank).

From the BEopt model, all three systems resulted in a negative annualized energy cost indicating that all are cost-effective options compared to the BA benchmark's heating and DHW systems. House #1 showed the highest annualize cost savings followed by House #2 and House #3.

If variable-speed pumps that sense system pressure are used, can the primary loop be eliminated without endangering the heat exchanger? If so, what are the implications for energy use?

In previous phases of this research, it was determined that primary/secondary plumbing configurations can result in higher return temperatures to the boiler than are actually returning from the zones. The reason for this is that the flow rates and water temperatures in the primary loop are generally much higher than those returning from the space heating zones. When the water returning from the zones mixes with that in the primary loop, the result is an increase in temperature. This in turn can reduce the boiler's efficiency. Therefore, this research proposed to investigate using specialized pumps as a way of eliminating the need for the primary loop. This study does not provide any encouraging results, although they are not conclusive in ruling out the possibility. The systems designed here had much higher system pressures than anticipated. It is suspected that the specialized fittings for the temperature sensors in the second floor baseboard restricted the flow more than the normal fittings would have. It is also true that there was no way to guarantee balanced flow to the two tubes without balancing valves. This resulted in lower (possibly laminar) flow rates than anticipated. The ECM pumps installed were not able to provide more than 1.5 gpm to the second floor zones in House #2 or House #3. If the primary loops had been eliminated, the system pressure would have been even higher, resulting in even lower flows. This is not to say that this couldn't be done, but this study did not result in solid recommendations to the industry on how to do this safely.

What is the tradeoff in efficiency associated with using a boost control to decrease recovery times following setback versus maintaining a constant temperature in the home? Which would result in lower overall energy use?

One of the major findings from this study was that systems operated with thermostat setback and boost controls for recovery cycle less and use less energy than systems operated in constant temperature mode. It must be noted that in order for a hydronic system to recover from setback, the system must have the extra capacity needed to recover. The systems should be designed to provide the necessary output under design conditions at supply temperatures lower than the boiler's maximum supply temperature. For example, if the boiler's maximum supply temperature is 180°F, the system should be designed to meet the design load at 150°F. Then, when recovery is needed, the boiler can boost to higher temperatures to meet the added load. Both the boiler and the baseboard need to be able to provide this added output.

What is the value of thermal mass in high efficiency boiler systems?

It was clearly observed in this research that added mass resulted in less cycling of the boiler's burner. Buffer tank operation resulted in the lowest amount of cycling. It is suspected that House #2 would have seen a similar level of cycling if the boiler's maximum space heating input was reduced to a similar level to that in House #3. However, no significant differences were found in efficiencies or operating costs between primary/secondary loop and buffer tank operation. Considering that the system with the buffer tank was the most costly to install and there were no measured savings, installing a buffer tank with the system in House #3 does not appear to provide much added benefit. If the boiler were severely oversized like that in House #1, a buffer tank might provide a bigger benefit.

When using a condensing water heater for space heating, maintaining a tank temperature above 130°F will result in decreased combustion efficiency, and should be avoided by designing the system for low-temperature operation. In general, increases in mass temperatures, as may be needed for space heat operation, will result in increased standby losses. Added tank insulation should be considered, especially for storage tanks located in unconditioned spaces.

8 Future Research Needs

Although this research resulted in some very useful findings, a few questions remain. Post-purge through the indirect DHW tank has been found to reduce standby losses in laboratory testing (Butcher 2011), but was not tested in the field in this study. An evaluation of how much this feature reduces standby losses would be useful. Does post-purge result in the same amount of savings under constant temperature operation as it would if setback is used?

Also, though cycling was four times higher in House #1 compared to House #2, does the extra cycling equate to increased energy use? What are the effects on boiler durability? Would a buffer tank in House #1 have resulted in decreased energy use? There are conflicting reports on the effects of short cycling on boiler efficiency. Field measurements would help quantify the impacts.

Finally, one of the conclusions of this study and the past study conducted by CARB was that oversizing baseboard capacity will result in improved response time and reduced energy use. Unfortunately, the baseboard in this study did not perform as anticipated for reasons previously discussed. Although the effects of inadequate capacity can clearly be seen in the performance of the systems in House #2 and House #3, verification of the benefits would be useful. What amount of oversizing results in optimal performance? What is the most cost-effective oversizing factor?

References

AHRI; (2009). Certified Product Directory—Boilers, Baseboard Radiation, Finned Tube (Commercial Radiation), Indirect-Fired Water Heaters, Air-Conditioning, Heating, and Refrigeration Institute (AHRI).

ANSI/ASHRAE Standard 103-2007 (2007). Methods of Testing for Annual Fuel Utilization Efficiency of Residential Central Furnaces and Boilers.

Arena, L.; (2010). "In-Field Performance of Condensing Boilers in Cold Climate Region." National Energy Technology Laboratory, Morgantown, WV; US Department of Energy, Building Technologies Program.

Butcher, T. (2004). "Hydronic Baseboard Thermal Distribution System with Outdoor Reset Control to Enable the Use of a Condensing Boiler." Brookhaven National Laboratory; Uptown, NY; US Department of Energy, Office of Buildings Technology.

Butcher, T. (2011). "Performance of Combination Hydronic Systems". ASHRAE Journal, December 2011, p36-41.

Butcher, T.; (2006). "Condensing boilers and baseboard hydronic systems.", ASHRAE Transactions, V. 112, part 1, 2006.

Butcher, T; (2009). "Optimal Design and Operating Parameters of the Condensing Boiler/Hot Water- Baseboard Combination: Task Report." NYSERDA Agreement, No. 10927. Albany, NY; NYSERDA.

Deru, M; Torcellini, P. (2007). *Source Energy and Emission Factors for Energy Use in Buildings* NREL/TP-550-38617. Golden, CO: National Renewable Energy Laboratory.

D&R International, Ltd; (2011). "2010 Building Energy Data Book." Pacific Northwest National Laboratory; U.S. Department of Energy's Office of Energy Efficiency and Renewable Energy (EERE).

EIA (2009). "2009 Residential Energy Consumption Survey." Washington, DC: United States Energy Information Administration. Accessed October 31, 2012: http://www.eia.gov/consumption/residential/data/2009/.

___ (2012a). "Electricity Data Browser." Washington, DC: United States Energy Information Administration. Accessed February 18, 2013: http://www.eia.gov/beta/enerdat/.

___ (2012b). "U.S. Natural Gas Prices." Washington, DC: United States Energy Information Administration. Accessed February 18, 2013: http://www.eia.gov/dnav/ng/ng_pri_sum_dcu_nus_m.htm

GRUNDFOS; (2010). "GRUNDFOS ALPHA: Installation and operating instructions." Olathe, KS; GRUNDFOS Pumps Corporation.

Henderson, H., Huang, Y., Parker, P. (1999). "Residential Equipment Part Load Curves for Use in DOE-2". LBNL-42175. US Department of Energy, Office of Building Systems, Contract No. DE-AC03-76SF00098.

Hendron, R.; Engebrecht, C. (2010). "Building America House Simulation Protocols". Golden, CO: National Renewable Energy Laboratory.

HTP (2013). "Versa-Hydro: Installation, Start-Up, Maintenance, Parts." East Freetown, MA; Heat Transfer Products, Inc..

Kratz, A.; Harris, W.; (2007). A Study Of Radiant Baseboard Heating in The I=B=R Research Home. The Engineering Experiment Station University Of Illinois. Urbana-Champaign, IL.

Peerless; (2011). "PEERLESS PureFire Rev2: Installation, Operation, & Maintenance Manual." Bally, PA; PB Heat, LLC.

Polly, B.; Gestwick, M.; Bianchi, M.; Anderson, R.; Horowitz, S.; Christensen, C.; Judkoff, R. (2011). A Method for Determining Optimal Residential Energy Efficiency Retrofit Packages. DOE/GO-102011-3261. Golden, CO: National Renewable Energy Laboratory.

Rudd, A.; Ueno, K.; Bergey, D.; Osser, R. (2011). "Draft Report on the Expert Meeting for Recommendations for Applying Water heaters in Combination Space and DHW Heating Systems". National Energy Technology Laboratory, Morgantown, WV; US Department of Energy, Building Technologies Program.

Schoenbauer, B.; Bohac, D.; Huelman, P.; Olson, R.; Hewett, M. (2012). "Retrofitting Combined Space and Water heating Systems: Laboratory Tests". US Department of Energy, Building Technologies Program.

U.S. Census Bureau (2009). Report accessed March 2012. http://www.census.gov/const/C25Ann/sftotalheatsystem.pdf. US Department of Commerce.

Taylor, John R.; (1939). An introduction to error analysis. 2nd Edition. University Science Books, 55D Gate Five Road, Sausalito, CA 94965 USA.

DOE (2009). "Technical Support Document: Energy Efficiency Program for consumer Products and Commercial and Industrial Equipment: Residential Dishwashers, Dehumidifiers, and Cooking Products, and Commercial Clothes Washers". US Department of Energy, Building Technologies Program.

Zoeller, W.; (2011). "Condensing Boilers and Low-Temperature Baseboard Convection." NYSERDA Agreement, No. 10927. Albany, NY; NYSERDA.

Uponor; (2011). "Radiant Heating and Cooling Systems: Complete Design Assistance Manual". 7th Edition. 5925 148th Street West Apple Valley, MN 55124 USA www.uponor-usa.com.

"Retrofitting Integrated Space & Water Heating Systems." (2013). Center for Energy and Environment. Accessed August 12, 2013: http://www.mncee.org/Innovation-Exchange/Projects/Current/Retrofitting-Integrated-Space---Water-Heating-Syst/.

"Annualized Energy Related Costs." (2013). BEopt. National Renewable Energy Laboratory. Accessed August 12, 2013: http://beopt.nrel.gov/sites/beopt.nrel.gov/files/help/Annualized_Energy_Related_Costs.htm

"Occupational Safelty & Health Administration." (2013). United States Department of Labour. Accessed August 12, 2013: https://www.osha.gov/dts/osta/otm/legionnaires/faq.html.

"A Performance Calculator for Grid-Connected PV Systems." (2013). PV Walls. National Renewable Energy Laboratory. Accessed August 12, 2013: http://rredc.nrel.gov/solar/calculators/pvwatts/version1/.

Appendix A: Sensor Type, Location and Accuracy

Measurement	Equipment and Specifications
Temperature: Water to/from Boiler, Heating Zones, and DHW	Omega Engineering P/N ON-910-44031, fast response tubular thermistor sensor with 4.5 in. long × 5/32 in. diameter probe and ⅛ in. NPT threaded pipe fitting, Operating range = -80°–75°C, Interchangeability = ± 0.1°C, 10K-Ohm thermistor element, equipped with 12 in. of 28AWG PVC insulated wire and PVC jacketed cable
Temperature: Interior Spaces	Omega Engineering P/N TH-44006-40-T, fast response thermistor sensor with exposed element, Operating range = -80°–120°C, 3nterchangeability = ± 0.2°C, 10K-Ohm bead, equipped with 40 in. of 26AWG PFA insulated wire and PFA jacketed cable
Temperature: Exterior	Omega Engineering P/N HSTH-44031-40, Fast Response Hermetic Flexible Thermistor Sensor with Sealed Element, Operating Range = -80°–75°C, Interchangeability = ± 0.1°C, 10K-Ohm thermistor element, equipped with 40 in. of 26AWG PFA insulated wires and PFA jacketed cable
Gas Consumption: Boiler	American Meter Company residential diaphragm gas meter Model AC250-TC, temperature compensated, 1 ft^3 and ¼ ft^3 drives, 1 in. connection set, Equipped with IMAC Systems domestic meter pulser Model 400-10P-10 mounted on ¼-drive hand, Pulser output = 10 Pulses Per Revolution, Pulser resolution = 1 Pulse Per 0.025ft^3 of Gas, Output = dry contact, form-C switch closure
Water Flow: Boiler, Heating, DHW	Impeller type, current-sinking pulse output, Accuracy = ± 1% full scale Omega Engineering flow meter models: FPR-302 = ½ in. FNPT, 0.1–10 gpm, FPR-303 = ¾ in. FNPT, 0.2–20 gpm, FTB4810 = 1 in. FNPT, 0.5–25 gpm Seametrics flow meter models: SPT-100 = 1 in. FNPT, 0.5–40 gpm, SES-100 = 1 in. FNPT, 0.5–25 gpm
Electric Energy Consumption: Boiler System Total, Pumps—Zone, Return, Heat Exchanger	Continental Control Systems WattNode Watt/Watt-hour transducer Model WNB-3Y-208-P Option P3, Output = optically isolated solid state relay, Accuracy = ± 0.50% of reading, Continental Control Systems revenue-grade, split-core high-accuracy current transformers, Model ACT-075-XXX (5A, 15A), Accuracy = ± 0.5% from 1%–100% rated current

Appendix B: Flow Rate Calibration Results

Flow rates at various speeds were measured by recording the time required to fill a fixed volume container. Measurements were repeated several times at each setting and averaged. The measured flow rates were calculated using Equation 3.

$$\dot{V} = \frac{c}{t_{\sec}} \times 60 \quad (3)$$

where:

$\dot{V}$: flow rate, gpm
c : container size, gallons
t_{sec}: time to fill cup, seconds

The constant 60 converts flow rates from gallons per second to gallons per minute (gpm).

Flow meter readings are represented by ‘x’ and the cup flow test measurement, ‘y’, in the calibration equation. For House #1, if flow meter recorded was 0 gpm while the boiler was firing, the flow rate was adjusted to 0.55 gpm, since that is the minimum flow required for the boiler to fire. According to the calibration equation shown in 40, for House #2, all flow meter readings of zero would be adjusted to 0.514 gpm, which is not always the case when there is actually no flow through the flow meter. To avoid marking up all 0 gpm readings, the calibration curve was applied to 0 gpm flow meter readings when there was a 5ºF increase in DHW supply temperate within the same minute. A 5ºF increases was chosen based on DHW supply temperature pattern shown in Figure 41. Whenever there is a hot water draw, shown with the purple diamond markers, there is at least a 5ºF increase in the DHW supply temperature, shown in blue dots.

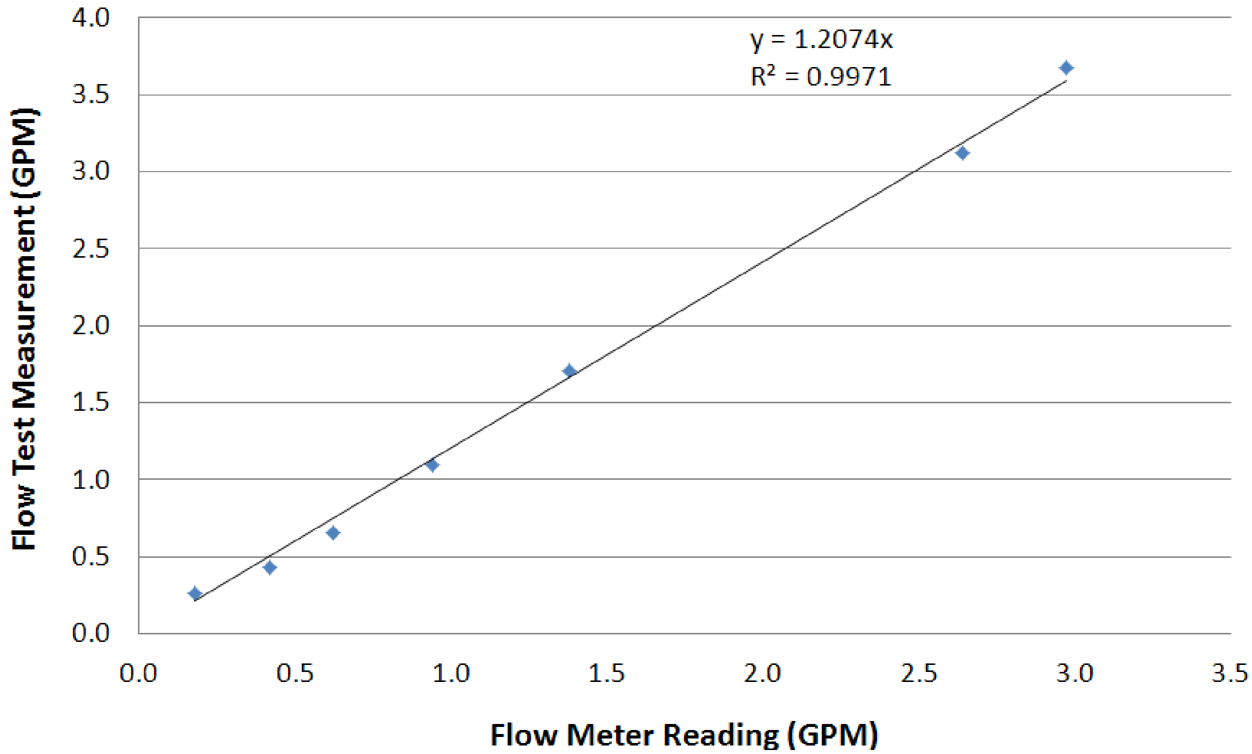


Figure 39. Calibration curve for DHW flow meter for House #1

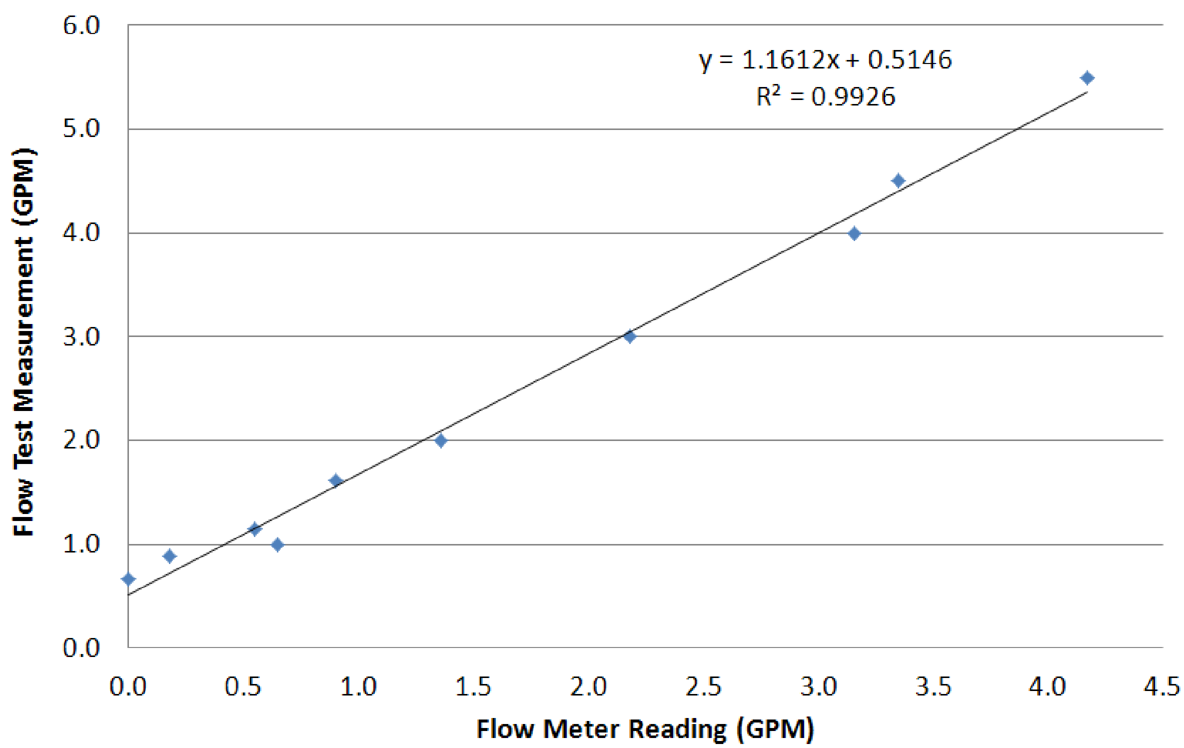


Figure 40. Calibration curve for DHW flow meter for House #2

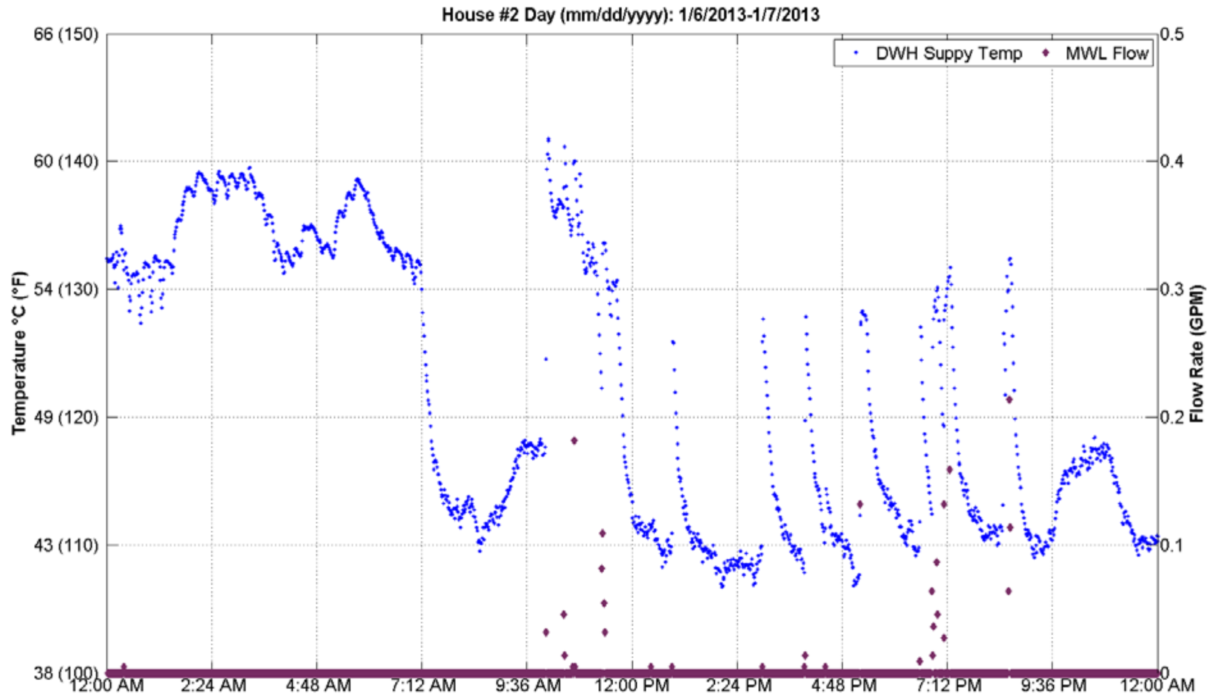


Figure 41. Typical DHW supply temperature pattern

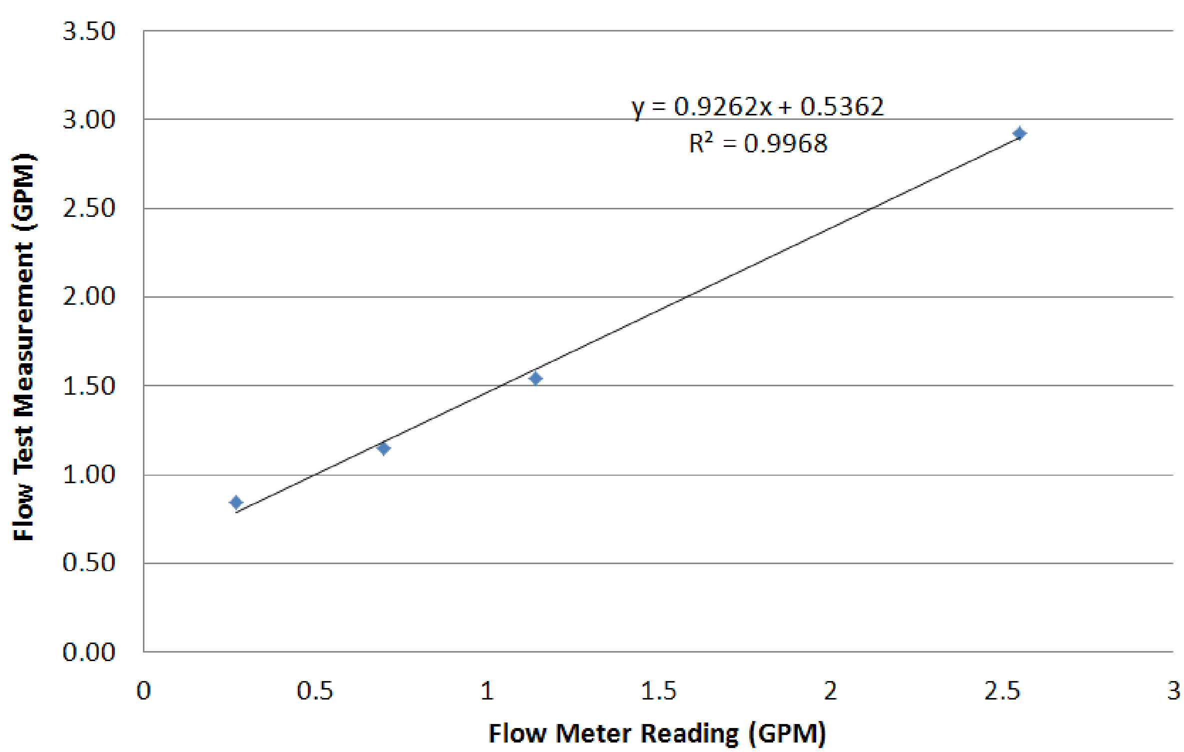


Figure 42. Calibration curve for DHW flow meter for House #3

buildingamerica.gov

U.S. DEPARTMENT OF ENERGY | Energy Efficiency & Renewable Energy

DOE/GO-102013-4108 ▪ October 2013