New York State Energy Research and Development Authority

Condensing Boilers and Low-Temperature Baseboard Convection

Final Report

August 2011

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CONDENSING BOILERS AND LOW-TEMPERATURE BASEBOARD CONVECTION

Final Report

Prepared for the

NEW YORK STATE
ENERGY RESEARCH AND
DEVELOPMENT AUTHORITY

Albany, NY
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1 Executive Summary

1.1. Overview

High-efficiency, condensing boilers have been available in the U.S. since the 1990s and are now common in the residential market, but in many instances they are not achieving consistent condensing performance levels due to high return water temperatures. This is particularly true with hydronic baseboard heating systems and their traditional sizing and design methods. In high performance, low-load homes, a change in the baseboard sizing and overall system configuration is warranted.

In response to this need, SWA teamed with the New York State Energy Research and Development Authority (NYSERDA), Ithaca Neighborhood Housing Services (INHS) and Brookhaven National Laboratory (BNL) and devised a three phase research study to evaluate the performance of condensing boilers using baseboard convector delivery systems. Specific objectives included:

- evaluate and define the optimal operating parameters of the condensing-boiler/hot-water baseboard combination;
- evaluate and define the optimal operating parameters of the condensing boiler/indirect domestic hot water combination;
- test those findings in real-world residential settings;
- document those parameters in a technically accurate, installer friendly manner;
- quantify costs of properly sized and installed systems.

The first phase of the project involved monitoring boiler performance in six existing homes in Ithaca, NY. This research began in late January 2009 and was concluded in August of 2009. Information gained from the first phase was used to design and size systems for three new homes also located in Ithaca, NY. Bench top research conducted by BNL (Phase II) in conjunction with follow-up testing and monitoring of the new homes (Phase III) has been conducted to further define the best design parameters to ensure maximum boiler efficiencies.

1.2. Key Findings

All homes monitored in Phase I and Phase III of this project contained Munchkin condensing boilers from Heat Transfer Products, Inc. Each boiler was also responsible for supplying the domestic hot water via an indirect tank. The boiler analyzed by BNL in the Bench Top Testing was a Peerless Pinnacle, Model 80M. The Peerless Pinnacle and the Munchkin use the same heat exchanger. Following are key findings from each phase of this study.

Phase I: Existing Home Monitoring

1. The typical primary/secondary loop plumbing configuration contributes to higher than optimum return water temperatures to the boiler.
2. The flow rates in these systems are higher than recommended, contributing to higher than optimum return water temperatures.
3. The baseboard lengths being installed in these homes are consistent with the lengths needed for a low temperature, low flow system.
4. The maximum boiler output temperature is typically set to 180°F or higher. For homes with low loads, it is unlikely that the return water temperature would be below 130°F, especially with the high flow rates measured and the added hot water circulating in the primary loop.
5. Boiler supply temperatures to the domestic hot water tank were set at 180°F or higher for five of the six homes.

Phase II: Benchtop Research

1. Any control technique that reduces the return water temperature, including lowering the boiler setpoint and/or reducing the loop flow rate, will significantly improve the achieved efficiency.
2. Condensing boilers can be operated with flow rates significantly lower, and temperature rises significantly higher than the manufacturer's recommendations.

Phase III: New Home Design & Monitoring

1. Recovery from Setback:
    a. Recovery time from setback is extremely slow and often not achieved.
    b. Recovery time is primarily affected by baseboard length, outdoor reset controls and the boiler's differential settings.
2. Flow rates:
    a. Flow rates were higher than optimal.
    b. Contractors do not have standard, simple methods for measuring/setting flow rates.
    c. Different boiler manufacturers have different recommendations for minimum flows through the heat exchanger.
3. Primary Loop:
    a. Elimination of the primary loop did not appear to adversely affect system operation, though
    b. Removing the primary loop leads to control issues such as domestic hot water (DHW) priority and may require wiring diagram changes.

4. Boiler Controls:

    a. Each boiler manufacturer has proprietary controls – low pressure cutoffs, high temperature cutoffs, differential settings to reduce cycling, etc.—making it difficult to determine the most efficient setup at install.
    b. Manuals provide little to no recommendations for ensuring condensing.
    c. The low limit on the boiler curve can lead to “no heat” situations even though the zone is calling for heat.
    d. Outdoor reset and low limit on the boiler curve may result in water temperatures too low to heat up the space.

5. First floor zones ran approximately 22–35% more often than 2nd floor zones.
6. There was a significant amount of time (25% to 66%) when the zones were calling for heat, but the boiler was not firing; indicating over-sizing, insufficient length of baseboard, less than optimal boiler settings, or some combination of the three.
7. The smallest boilers available are too large for these homes; short cycling is common.

The market barriers concerning condensing boilers paired with baseboard convectors apply more to the system configuration and settings than to the actual equipment. The boiler output temperatures are consistently set too high to promote condensing, as is the flow rate through the zones. The primary loop (recommended by some boiler manufacturers) also appears to reduce the frequency of condensing.

Although there is a lot of information on condensing boilers, baseboard convectors, and indirect DHW components, there is little information explaining the best combination of settings when these technologies are combined.

2 Introduction

2.1. Relevance

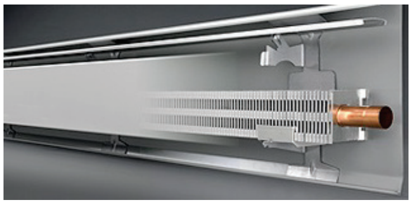
In the Northern U.S., most residential space heating is accomplished with either natural gas or fuel oil, and less frequently, propane. Of the available fuels and heating plant/heating distribution combinations, few are able to achieve very high overall efficiencies. Where natural gas as a fuel source is available, options for very efficient space heating (over 90% combustion efficiency) include condensing gas furnaces, and to a lesser degree, condensing boilers in combination with radiant floors or hydro-coils in low temperature operations. Where natural gas is not available, options are more limited and include propane fired condensing furnaces and propane condensing boilers. Until very recently, very high efficiency oil equipment (in the form of condensing boilers) did not exist. Moving down a notch to mid-level, or lower, efficiencies yields a multitude of additional space-heating options, but at the cost of increased fuel usage and higher energy expenses. Direct vent gas and oil boilers and furnaces have proven reliable, but generally achieve efficiencies of approximately 85%. Atmospheric boilers and furnaces are generally rated at approximately 80% efficiency.

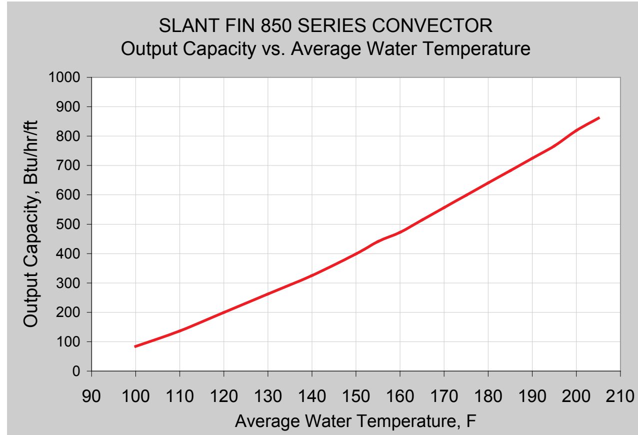
Aside from the efficiencies of the heating plants themselves is the relative effectiveness of the manner in which the heat is distributed throughout the residence. Ducted systems (furnaces and hydro-systems with air-handlers) are inherently inefficient. Ten percent total duct leakage is considered a “tight” system. Depending on where the ducts are located relative to unconditioned space, conductive losses can be substantial. Integrating ducted systems into conditioned space can be challenging, resulting in many attic and crawlspace space installations. Duct leaks to unconditioned space can cause interior pressure imbalances and another layer of problems.

Space heating through hydronic distribution can provide a more efficient option. As compared to a single-family air-handler blower motor that uses between 200 and 800 watts during operation, a circulator pump capable of serving an entire residence may use less than 100 watts. Furthermore, hydronic distribution systems are small and easily integrated into conditioned space reducing or eliminating thermal distribution losses.

To achieve an overall highly efficient hydronic system, these efficient distribution systems must be combined with a highly efficient heating source: a condensing boiler. Matched to a properly sized hot-water baseboard convection distribution system, a low-temperature condensing boiler can achieve optimal efficiencies and flexibility in meeting even the smallest of loads.

Figure 1. SLANTFIN Hot Water Baseboard Convectors and Performance Data





Market barriers to this technology are minimal. Even though additional length of convective baseboard is theoretically required due to lower BTU/hr output under low temperature operations, in practice hot-water baseboard systems are nearly always oversized, with substantial excess capacity. This practice of redundant capacity means that the dimensional requirements for a typical hot-water baseboard system would need to be increased only minimally, if at all, to meet design conditions under low-temperature operation.

CARB believes that evaluation and documentation of “real-world” installations – especially in colder climates – has provided a more accurate understanding of the current technology’s performance. It is critically important to know how systems are actually performing in order to gauge what advancements need to be made to achieve long-term Building America goals. It’s also critical for modeling tools to accurately predict the performance of advanced systems installed in Building America homes. These field evaluations of condensing boilers have identified specific areas of opportunities for improvements in the technology and will allow more accurate modeling of these systems.

2.2. Evaluate and Define Optimal Design and Operating Parameters

The research project draws upon current Brookhaven Nation Lab (BNL) condensing boiler research (Butcher 2006) and Steven Winter Associates’ experience with affordable, high-performance housing and advanced systems integration. Critical to the success of this endeavor is a clear understanding of the engineering principles involved and the current state of technology development. Condensing boilers are capable of recovering the latent heat of condensation from the combustion process that would otherwise be wasted. As is illustrated in Figure 2, realizing the higher combustion efficiency possible, due to the energy of condensation, requires return-side water temperatures below 130^{o}F for gas equipment. With oil equipment, return side water temperatures must be below 118^{o}F.

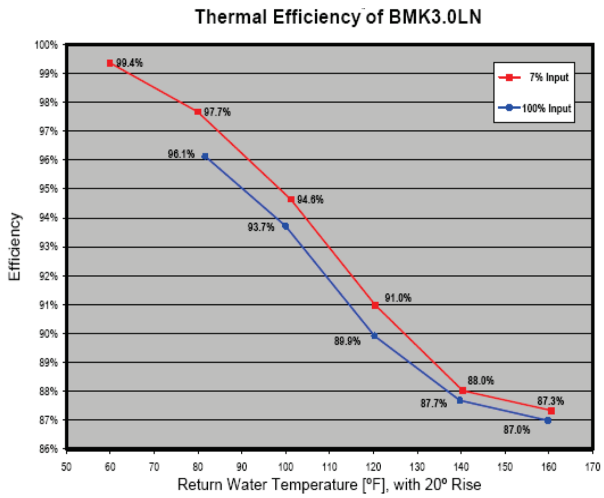


Figure 2. AERCO Boiler Performance Data

Therefore, in order to meet design loads in a typical dwelling, a combination of factors must be resolved: overall building loads must be low enough to allow the application of low-temperature hydronic heat delivery, and the delivery mechanism (baseboard convectors in this study) must be optimally specified and designed to deliver sufficient heat within the space constraints of a room.

The individual technologies needed to resolve these factors are established. What is required for market adoption is proof-of-concept, bench-top research to establish the optimal operating parameters of the low-temperature convection/condensing boiler system in combination with indirect DHW along with in-field testing, monitoring, and optimization of the concept under actual operating conditions. Documenting and disseminating the study results will allow the technology to be adopted on a large scale, in particular in affordable housing where energy and first costs are especially sensitive.

This project will provide the building industries with the information and tools needed to design, engineer, install, and operate very high-efficiency, condensing-boiler based, low-temperature space heating using low-cost, off-the-shelf distribution components. At present, the market-ready technologies able to deliver low-temperature hydronic-heat distribution are radiant-floor systems and some specialized cast-aluminum radiation systems, both of which are very expensive and therefore not suited for affordable applications. A typical radiant-floor distribution system will cost in the neighborhood of $15 per square foot of living space, or about $30,000 for a 2000 ft^2 residence. A cast-aluminum radiation system for the same residence would cost approximately $20,000. By comparison, the envisioned low-temperature convector technology would cost less than $2,000.

Based on the above cost analysis, SWA concluded that these systems – baseboard convectors with condensing boilers – are relevant in providing cost-effective, high-efficiency solutions to the residential construction market. Therefore, the following research was conducted to evaluate and define the optimal operating parameters of condensing boilers when combined with a hot water baseboard delivery system and an indirect domestic hot water system.

3 Phase I. Evaluate System Performance in Existing, Single-Family Applications

There were several objectives for Phase I of this research project. The highest priority was to analyze the frequency of condensing and determine the factors most likely to affect the overall performance of these systems. SWA wanted to identify typical operating conditions in existing installations and to analyze how the boilers were functioning: did they cycle too frequently; what were the typical flow rates; etc. Based on the results of this phase, recommendations for the design and operating conditions of several new systems would be made. The first phase was also used to evaluate the monitoring plan and make revisions and refinements where necessary for Phase III.

Working with Ithaca Neighborhood Housing Services, a non-profit developer and operator of for-sale and rental affordable housing in central New York, the research team evaluated several low-temperature, space heating and distribution systems in existing homes.

Selection of the first six test sites was based on the following:

1. geographical location (prefer cold climate: greater than 5500 ^{o}F HDD);
2. system configuration (type of fuel, type of delivery system);
3. overall efficiency of the home (documented levels of insulation, above average efficiency);
4. homeowner approval of monitoring for a minimum of six months.

Table 1 summarizes the operating conditions of the boilers in the first phase of this project and the estimated frequency of condensing during the monitoring period when there was a call for space heat. Frequency of condensing was calculated by dividing the number of boiler return water temperature readings below 130°F by the total number of measurements made when the boiler was firing in response to a call for heat. It should be noted that the data collection interval (15 minutes) was too large to make very accurate predictions, but one conclusion can definitely be drawn: the boilers in these homes were far from condensing 100% of the time. In fact, these numbers are most likely an over-estimate of the frequency considering that the average temperatures for the 15 minute interval were recorded. A smaller collection interval and better resolution on the gas meter were used in the last phase of this project to isolate the conditions when the boiler was firing, and therefore allow for a more accurate prediction of the frequency of condensing.

There were a few other key findings during the initial phase of this project. First, it became clear that the presence of the primary loop was most likely raising the temperature of the water returning from the zones, and subsequently the water returning to the boiler, thereby decreasing the frequency of condensing.

Second, the flow rates through the primary loop ranged from over 3-gpm to over 5-gpm. This, in combination with the higher temperatures of the primary loop, was keeping the ΔT of the boiler very small, under 10°F in most cases, even at design conditions.

Table 1. Summary of Space Heating Operating Conditions from Existing Home Monitoring

							Boiler Curve Settings [°F]			
House	Baseboard Length ft	Boiler Capacity kBtuh	# of Zones #	Flow Rate[1] gpm	Frequency of Condensing	Outdoor Reset	$T_{s,max}$	$T_{out,min}$	$T_{out,max}$	$T_{s,min}$
#1	52	unknown	1	3.1	69%	Y	180	0	72	95
#2	38.5	50	2	5.3	59%	Y	185	5	68	95
#3	61	80	3	4.8	60%	Y	180	5	68	95
#4	32	80	1	3.3	20%	N[2]	200	5	68	95
#5	41	50	2	5.2	14%	Y[3]	185	5	68	145
#6	54	80	2	4.3	16%	N	201	5	68	95

[1]Flow rate recorded through primary loop.

[2]The outdoor reset, although installed, is not registering in the controller.

[3]The minimum boiler supply temperature was set to 145 °F because the toe kick heater in the kitchen would not activate below that.

Third, the baseboard lengths being installed in these homes are consistent with the lengths needed for a low temperature, low flow system. In Phase III of this project,SWA's recommended linear feet of baseboard for the new homes did not raise any objections from the builder since these lengths were in line with their current practice. Since over-sizing is a common practice, it is anticipated proper sizing of baseboards will not be a hardship to any builder.

Also, as can be seen in Table 1, the maximum boiler supply temperature ($T_{s,max}$) was set to 180°F or higher at a corresponding outdoor temperature ($T_{out,min}$) of 5°F for all the homes monitored. For homes with such low loads, it is unlikely that the return water temperature would be below 130°F for most of the heating season if the supply temperature is 180°F, especially with the high flow rates and added heat from the primary loop.

Another common problem is that the outdoor reset control is sometimes not installed because it is an additional expense. The purpose of the outdoor reset is to lower the boiler supply temperature as the outdoor temperature rises. A lower boiler supply temperature results in a lower return temperature and a higher frequency of condensing. Unfortunately, a condensing boiler without an outdoor reset control will condense no more frequently than a standard boiler. Houses 4, 5 and 6 in Table 1 either didn't have an outdoor reset or it wasn't operating properly. It is very apparent that without a functioning outdoor reset control, condensing is unlikely to occur except for at cold start ups. The result is efficiency in the mid to upper 80s as opposed to the lower to mid 90s.

Finally, after analyzing the data on the indirect hot water tanks, it was clear that further analysis was needed to determine the optimal parameters to ensure condensing when indirect water heaters are paired with condensing boilers. Table 2 summarizes the DHW system settings and characteristics along with the frequency of condensing when there was a call from the DHW tank sensor.

Table 2. Summary of DHW Operating Conditions from Existing Home Monitoring

House	$T_{s,DHW}$* °F	Tank Setpoint °F	Tank Size gallons	Flow gpm	# of Occupants	% of Time Condensing	Data Interval minutes
#1	180	125	30	3.3	2	60%	15
#2	180	115	30	5.6	2	65%	5
#3	180	125	40	4.2	4	44%	5
#4	145	115	30	6.0	4	36%	5
#5	180	?	30	5.4	3	18%	5
#6	185	115	30	2.7	1	35%	15

* Boiler supply temperature setpoint for DHW.

During a follow-up visit In May of 2009, the data collection interval was decreased to five minutes in four of the six homes. Although the heating season was almost over at that point, this allowed a more detailed analysis of the domestic hot water systems. Frequency of condensing when the boiler was supplying heat to the domestic hot water ranged from 18 to 65% for these six homes. In Phase III of this study, an even smaller time interval was used for data collection. The results of that phase are discussed later in this report, but based on that data, these estimates appear reasonable.

Flow rate, boiler supply temperature, boiler capacity, heat exchanger size and storage volume are parameters that must be optimized to ensure condensing during domestic hot water calls.

4 Phase II. Bench Top Research

Conducted and reported by: Thomas Butcher, BNL

The Buildings Group at Brookhaven National Lab (BNL) is involved with emerging technologies for building heating and cooling equipment, distributed generation concepts, and research in support of reducing energy losses in building thermal distribution systems. BNL partnered with SWA in this research and performed the following “bench top” analysis:

1. Thermal efficiency testing of one, natural-gas fired, Peerless Pinnacle condensing boiler
2. Evaluation of low-flow applications using a condensing boiler in combination with baseboard convectors.
3. A comparison of the output performance of existing baseboard convectors
4. A comparison of available indirect-fired, storage water heaters

A detailed performance map for representative boilers was developed using the existing BNL boiler performance test system. This data was used in the design study to do a trade-off evaluation of options. Thomas Butcher at BNL led this effort.

4.1. Efficiency Tests on Peerless Pinnacle Condensing Boiler

In support of the field study project on the performance of condensing boilers with baseboard radiators, BNL conducted thermal efficiency testing of one condensing boiler of the type included in the field study. The boiler tested was a natural-gas fired, Peerless Pinnacle, Model 80M with a maximum rated input of 80,000 Btu/hr. These studies included three parts: 1) steady state efficiency, 2) efficiency under a cyclic load pattern, and 3) idle energy input requirements.

The arrangement for the tests is illustrated in Figure 3.

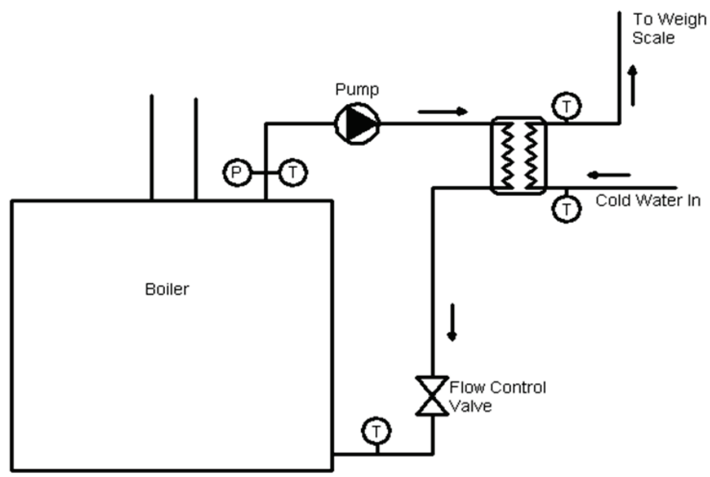


Figure 3. Arrangement for efficiency tests

As shown, boiler water is circulated through a plate type heat exchanger and direct cooling on the secondary side of the heat exchanger is used to reject generated heat. After the heat exchanger the cooling water is directed to a weigh scale which is linked to the lab's data acquisition and control system (DAQ). Heat output from the boiler is taken as the product of cooling water mass flow and temperature rise.

Gas energy input is measured using a positive displacement, dry test meter with a pulse pickup which provides an output of 1000 pulses per cubic foot of gas (roughly 1 pulse per Btu). The pulse output is also logged on the lab's DAQ system.

Steady State Efficiency Tests

The steady state tests were done at three different setpoints for the boiler water temperature control: 165°F, 145°F, and 125°F. At each water temperature efficiency was measured over a range of cooling water flow rates (and input rates).

Results of tests at 165, 145, and 125°F are presented in Table 3, Table 4, and Table 5 respectively.

Table 3. Results of Steady State Tests at 165°F

Test No.	Output Btu/hr	Input Btu/hr	Efficiency %	T_{supply} °F	T_{return} °F	Boiler Flow gpm	$T_{flue\ gas}$ °F	Flue gas O_2 %	Heat Exchanger Flow gpm
1	27,418	31,837	86.1	167	150	3.3	164	5.8	0.68
2	34,267	39,348	87.1	167	146	3.3	163	5.3	0.89
3	38,564	43,995	87.7	167	143	3.3	162	5.3	1.07
4	44,916	51,270	87.6	167	139	3.2	160	5.1	1.35
5	57,898	65,686	88.1	167	131	3.2	157	4.9	2.34
6	67,481	76,165	88.6	167	125	3.2	156	5.1	3.67

Table 4. Results of Steady State Tests at 145°F

Test No.	Output Btu/hr	Input Btu/hr	Efficiency %	T_{supply} °F	T_{return} °F	Boiler Flow gpm	$T_{flue\ gas}$ °F	Flue gas O_2 %	Heat Exchanger Flow gpm
1	41,950	45,731	91.7	144	118	3.3	136	5.2	1.9
2	47,821	51,867	92.2	144	115	3.3	136	5	2.6
3	49,340	53,503	92.2	144	111	3.3	136	5	2.8
4	53,575	57,719	92.8	144	114	3.3	135	5	3.7

Table 5. Results of Steady State Tests at 125°F

Test No.	Output Btu/hr	Input Btu/hr	Efficiency %	T_{supply} °F	T_{return} °F	Boiler Flow gpm	$T_{flue\ gas}$ °F	Flue gas O_2 %	Heat Exchanger Flow gpm
1	32,372	34,011	95.2	125	105	3.2	122	5.7	1.9
2	37,053	38,948	95.2	125	102	3.2	122	5.5	2.6
3	38,288	40,531	94.4	125	101	3.2	122	5.4	2.9
4	41,992	44,392	94.5	125	99	3.2	121	5.3	3.8

The tests for which results are described in the above tables were all conducted with the maximum boiler loop flow rate provided by the circulator and system used, roughly 3.3 gpm. One of the interests in this project is in operation of the hydronic loop at lower flow rates to increase the supply return differential temperature and effect condensing even at high supply temperatures. An additional set of efficiency data was taken with the boiler loop flow reduced by throttling the Flow Control Valve shown in Figure 3. Results of this test are presented in Table 6.

Table 6. Results of Steady State Tests at 165°F, Reduced Boiler Loop Flow

Test No.	Output Btu/hr	Input Btu/hr	Efficiency %	T_{supply} °F	T_{return} °F	Boiler Flow gpm	$T_{flue\ gas}$ °F	Flue gas O_2 %	Heat Exchanger Flow gpm
1	40,167	44,618	90	162	128	2.3	145	-	1.4
2	45,185	49,932	90.5	163	124	2.3	142	-	1.9

Taken together these results show, as expected considerably higher efficiency at the lower boiler temperature. These results are also presented graphically in Figure 4.

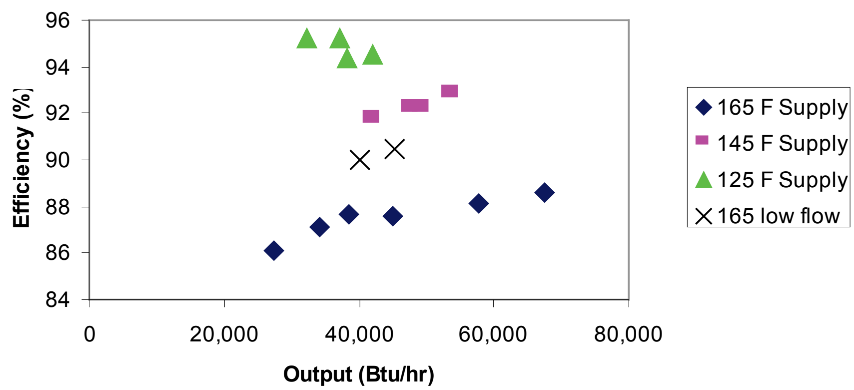


Figure 4. Summary of steady state efficiency test results

In condensing boilers, the degree of condensation and the efficiency is particularly affected by the return water temperature. Figure 5 again illustrates the steady state efficiency for all of the data against the return water temperature.

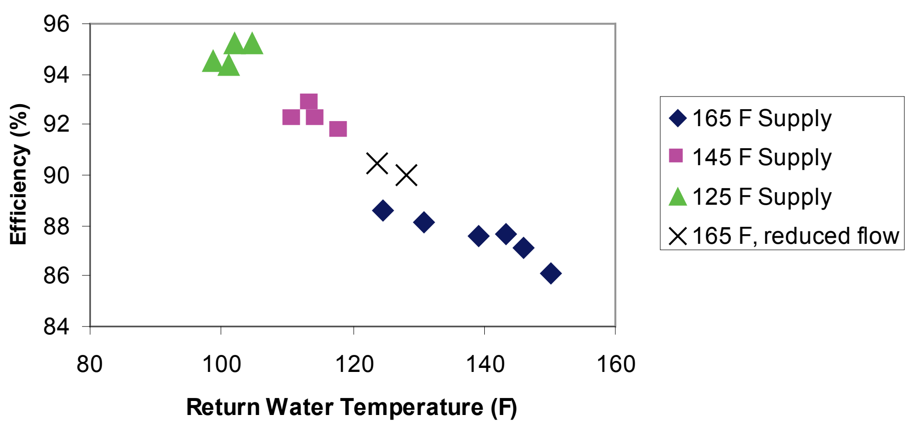


Figure 5. Steady state efficiency vs. return water temperature

Efficiency Under a Cyclic Load

Tests under cyclic load conditions were done to evaluate the degradation in efficiency of the condensing boiler under low load, cyclic conditions more typical of those observed during a heating season. For these tests cyclic loads were imposed on the boiler by opening and closing the solenoid valve controlling cooling water flow to the plate heat exchanger. The boiler circulating pump ran continuously during these tests. Testing was done at each of the three temperature setpoint conditions used in the steady state tests: 165, 145, and 125°F. At each of these temperatures, three different load patterns were applied:

Pattern 1: 5-minute draw, 10-minute idle

Pattern 2: 5-minute draw, 25-minute idle

Pattern 3: 15-minute draw, 105-minute idle

All tests were done with a fairly low cooling water flow rate, about 1.6 gpm. The load patterns were applied to the boiler by the lab DAQ system under full automatic operation. For patterns 1 and 2, eight repeat cycles were used and for Pattern 3, five repeat cycles were used.

For all setpoint temperatures the boiler supply and return water temperatures varied considerably over the draw period. For the 125 setpoint temperature the burner cycled during the draw period as the boiler quickly reached the setpoint temperature. For example, during each of the the 15 minute draws the burner cycled five times.

For these cycling draw patterns it is expected that the performance would be affected by both the draw pattern and the water temperature, leading to a less-clear performance image.

Figure 6, below, shows the results of all cyclic tests in the form of efficiency vs. output with setpoint temperature as a parameter. This generally shows improved efficiency at lower output but lower output also corresponds to lower boiler water temperatures and so the relationship is not very clear. In general efficiency seems to be dominated by temperature. The efficiency levels during these cyclic tests are clearly lower than during the steady state tests, as would be expected. At the lower water temperature the difference between the steady state and cyclic test efficiencies is greater than at higher water temperature. For example, comparing Figure 4 and Figure 6, at 125°F the cyclic efficiency is roughly six points lower than the steady state efficiency. Nevertheless, at 165°F, the cyclic efficiency is roughly three points lower than the steady state efficiency.

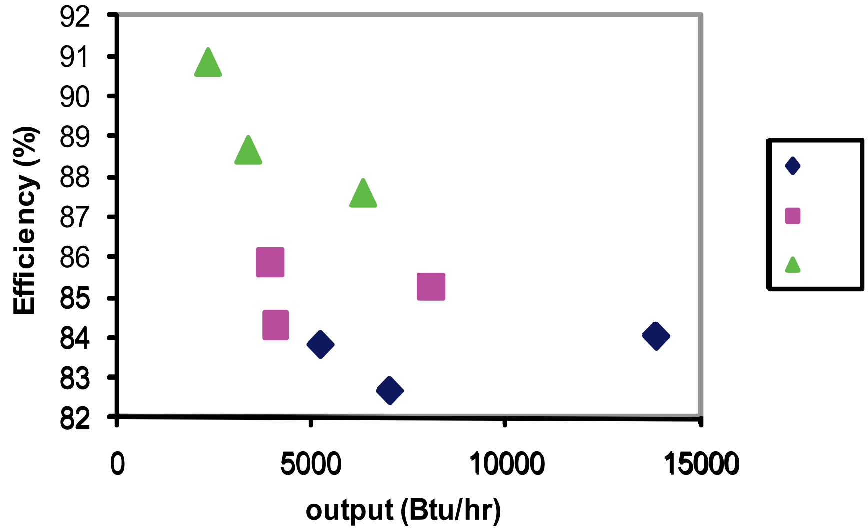


Figure 6. Cyclic tests – efficiency vs. output

The results in Figure 6 can also be used to get a very rough estimate of the energy savings potential of an outdoor reset control (or control that functions in a similar way). Reducing water temperature from 165 to 125 in cyclic operation leads to an increase in efficiency from roughly 83% to 88-91%.

Figure 7 is an effort to correlate efficiency vs. average return temperature during the draw period. While clearly the highest efficiencies result with the lowest return water temperatures as in steady state, the relationship is not a perfect one.

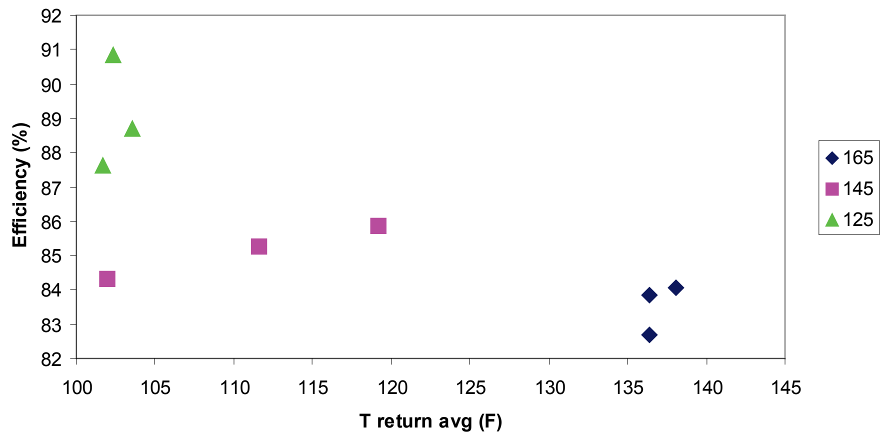


Figure 7. Cyclic tests – efficiency vs. return water temperature

In Figure 8, an input/output plot is presented for all of the cyclic tests and this shows a fairly uniform relationship.

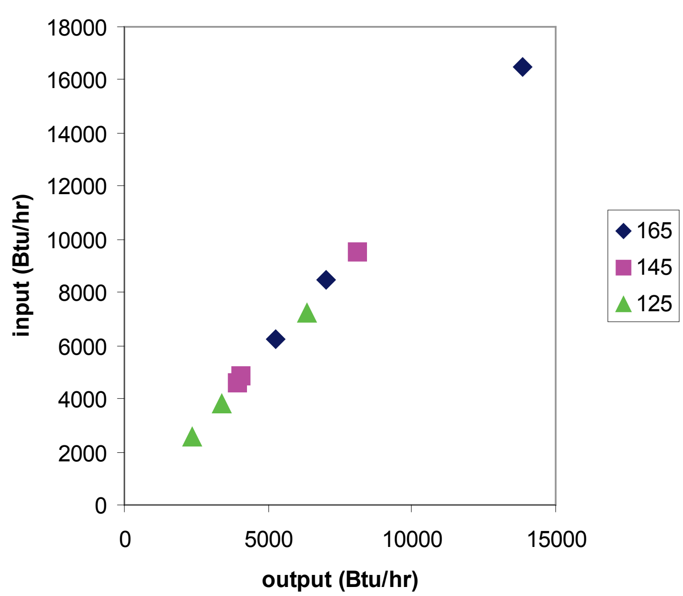


Figure 8. Cyclic tests – input/output relation

For the plot in Figure 8 both input and output are averaged over the idle plus draw periods.

Adding a single steady state point to the input/output relationship in Figure 8 with 90% efficiency at 42,000 Btu/hr output and using a linear regression for all points leads to the results presented in Figure 9.

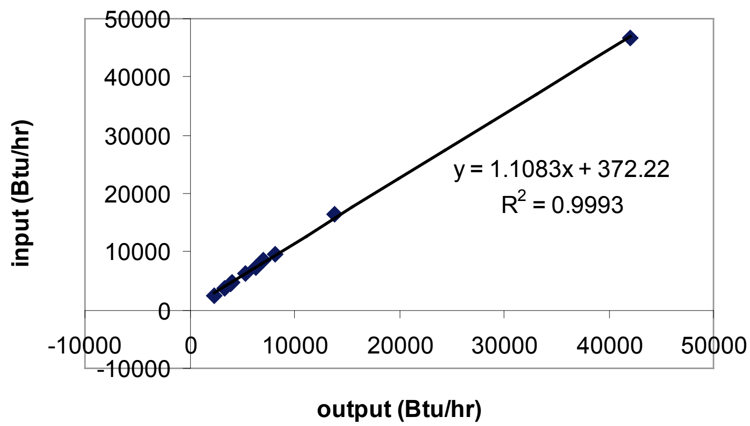


Figure 9. Input output relation including all cyclic data and a steady state point

Idle Loss Tests

Idle loss tests were done by closing the cooling water flow, communicating a call for heat to the boiler control, running the circulator continuously, and recording gas consumption over extended time periods (12 hours). In this mode the burner fires every few hours to make up for heat lost from the boiler jacket. Tests were done at two setpoint temperatures – 165°F and 145°F. The measured heat input rate averaged over the entire period for these two conditions was found to be 433 Btu/hr and 232 Btu/hr respectively. Since this is a cold start boiler this parameter is not useful for estimating energy consumption during extended idle periods. It is a useful parameter, however, for evaluating the off cycle losses and so the magnitude of the performance degradation in cyclic operation. These values are very low – on the order of 0.5 % (at 165) of the boiler's steady state input. The default value for boiler jacket loss in the ASHRAE 103 Standard is 1%. Typically boilers range from 1% to as high as 5% with older, poorly insulated units. These values can be compared with the intercept for the linear regression shown in Figure 9.

Conclusions

Overall, these tests show this to be a very efficient boiler with very low idle losses and very good performance even under low, part-load conditions. These results also show that the potential of this boiler is really only achieved with low return water temperatures and that this parameter is the most significant, particularly for steady state efficiency. Any control technique that reduces the return water temperature, including lowering the boiler set-point and/or reducing the loop flow rate, will significantly improve the achieved efficiency.

4.2. Evaluation of Low-Flow Applications

The water condensation rate and efficiency of a condensing boiler is directly related to the return water temperature. If a condensing boiler is being used in a heating system with baseboard radiators, which require higher temperature water to meet the building heat load, a low return water temperature can be achieved by reducing the boiler water flow rate.

The boilers being used in this project are low mass, modulating, gas-fired products with helical coiled stainless steel heat exchangers. The manufacturer specifies a minimum flow rate (model dependent) which is acceptable for use, and this minimum flow rate is inconsistent with the goal of low flow rates and high efficiencies in baseboard applications. The purpose of the lab tests described in this topical report was to evaluate performance problems at low flows with these boilers.

The specific boiler included in these tests was the Peerless Pinnacle, Model 80M with a maximum rated input of 80,000 Btu/hr. The specified minimum cooling water flow rate is 4.0 gpm. At the maximum firing rate this flow yields a boiler supply/return temperature difference of 36°F. At lower firing rates, where the boiler will operate for most of the time in the field, the temperature difference will be much lower, reducing the opportunity for condensing.

In discussions with the manufacturer, the key concern raised about lower water flow rates is boiling in the heat exchanger, leading to local metal temperatures that can exceed safe limits and pressure surges.

For laboratory testing two different arrangements were used. The first, illustrated in Figure 10, involves use of a closed boiler loop and compact plate heat exchanger for cooling. The boiler loop includes an expansion tank (not shown) and the boiler pressure is controlled by the fill level on the primary side. In addition to a minimum flow rate the manufacturer specifies a minimum boiler pressure of 15 psi. For these tests, this was bypassed and studies were done with pressures ranging from 0 to 25 psi. To enable evaluation of the occurrence of significant boiling, a rapid response pressure transducer was installed as shown in Figure 10 (Omega model PX302-030 GV). During selected periods pressure data was logged at 1-second intervals. Also as shown in Figure 10, system temperatures were measured using thermocouple probes and recorded at 5-second intervals.

The second test arrangement used is illustrated in Figure 11. Here the plate heat exchanger has been removed, and the boiler is directly cooled with cold "city" water. This arrangement was planned in discussions with the manufacturer to provide higher burner firing rates for a given flow rate and outlet temperature and increase the potential for observance of the boiling situation.

Table 7 reports on a set of tests that were done with the boiler in the arrangement shown in Figure 10. For all of these tests the boiler pressure was approximately four psi, and there was no sign of pressure pulses. Figure 12 illustrates the stability of the boiler pressure readings during one minute of the test done at the lowest flow rate.

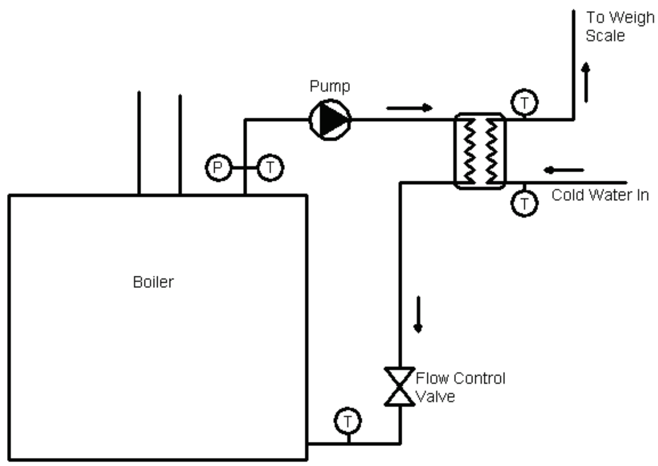


Figure 10. Test arrangement using a plate heat exchanger and primary/secondary arrangement

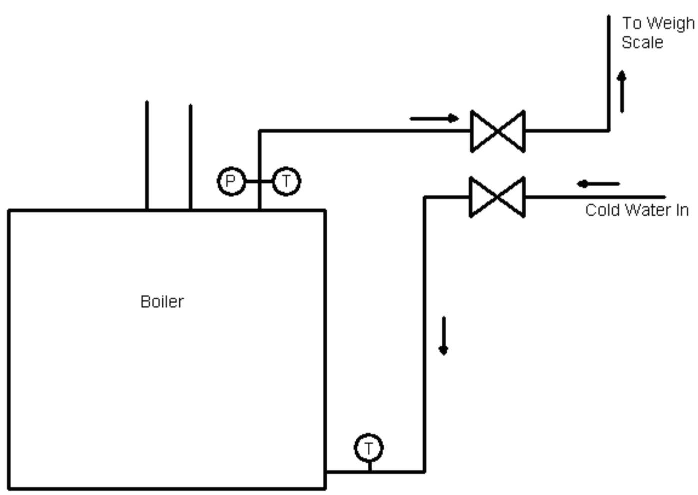


Figure 11. Test arrangement with direct cooling

Table 7. Results of Tests with Primary/Secondary Arrangement with Low Boiler Flow Rate

Test	Boiler Loop Flow	Cooling Water Flow	Return Water Temp	Supply Water Temp
	gpm	gpm	°F	°F
1	1.91	2.59	120	164
2	1.35	2.6	111	167
3	0.92	2.7	98	169
4	0.75	3.1	84	171
5	0.61	3.1	79	175

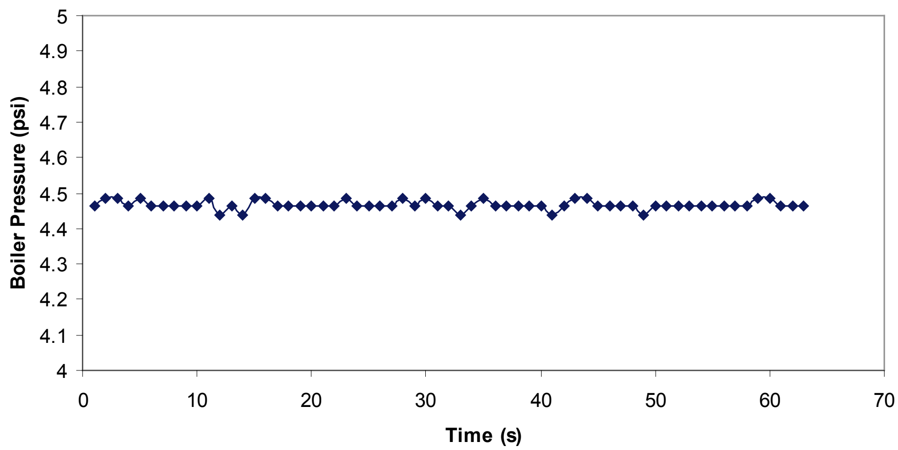


Figure 12. Illustration of the stability of the boiler pressure during test at 0.61 gpm flow rate

During the tests with arrangement 2 (Figure 11) boiler pressure was adjusted by balancing the inlet and outlet flow control valves. Some of the testing was done at higher pressure, 20 psi, and here again no observation of boiling in the heat exchanger was observed, even at fairly low flows. Increasing the pressure, of course, increases the boiling point (saturation temperature), and this is illustrated over the range of interest in this work in Figure 13.

When operating with the direct cooling arrangement we attempted to have as high a temperature rise as possible through the boiler. This led to the burner shutting off periodically due to over temperature, but allowed the basic evaluation to be completed.

Figure 14 shows the results of one test conducted with the boiler pressure in the 21 psi range and cooling water flow in the 0.7 gpm range. There was some upward trend in both pressure and temperature during the test period but no indication of the strong pressure rise that would be expected with boiling.

Figure 15 shows the results of a test conducted with the direct cooling arrangement but a lower pressure and slightly higher flow rate. Here a clear indication of boiling within the heat exchanger was observed. This pattern was very repeatable – as the boiler approached the limit temperature pressure increased sharply, the temperature, then rose above the setpoint, and the burner shut down. Both pressure and temperature decreased after this point.

It must be noted that for the data in Figure 15, the temperature and pressure data were collected with two different devices and there may have been a poor clock correlation between these. As a result it may have happened, for example, that the pressure peaked before the temperature did.

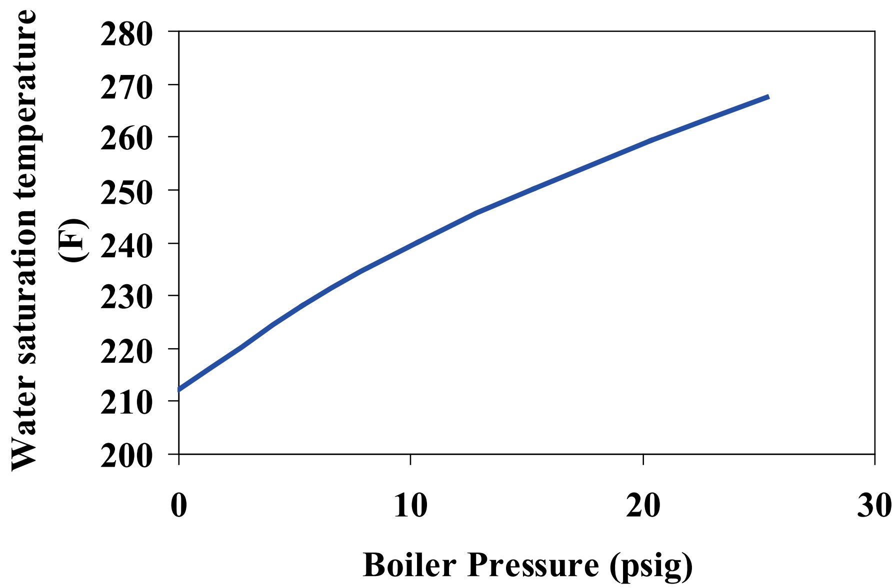


Figure 13. Relationship between boiler pressure and saturation temperature over range of interest for this work.

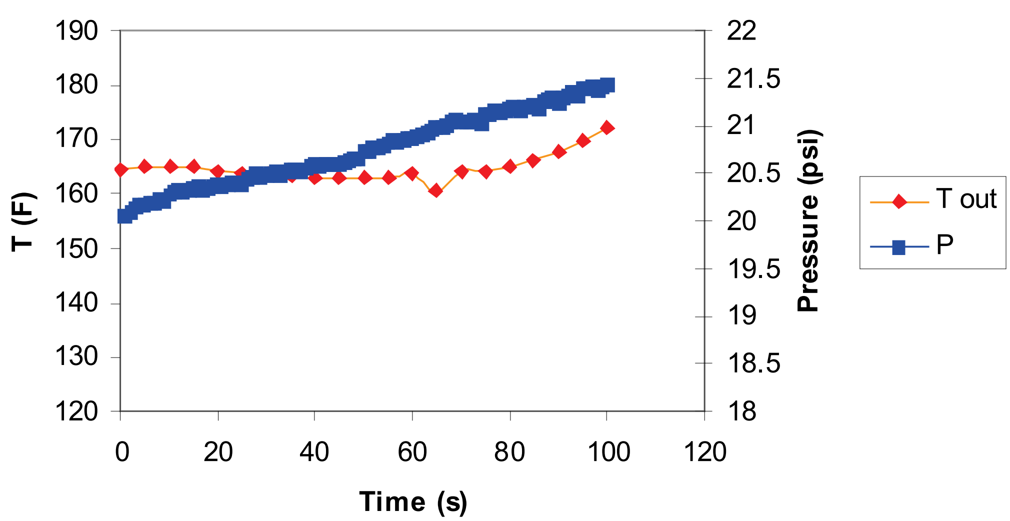


Figure 14. Direct cooling mode, higher boiler pressure, illustration of the temperature and pressure trends. 0.73 gpm flow

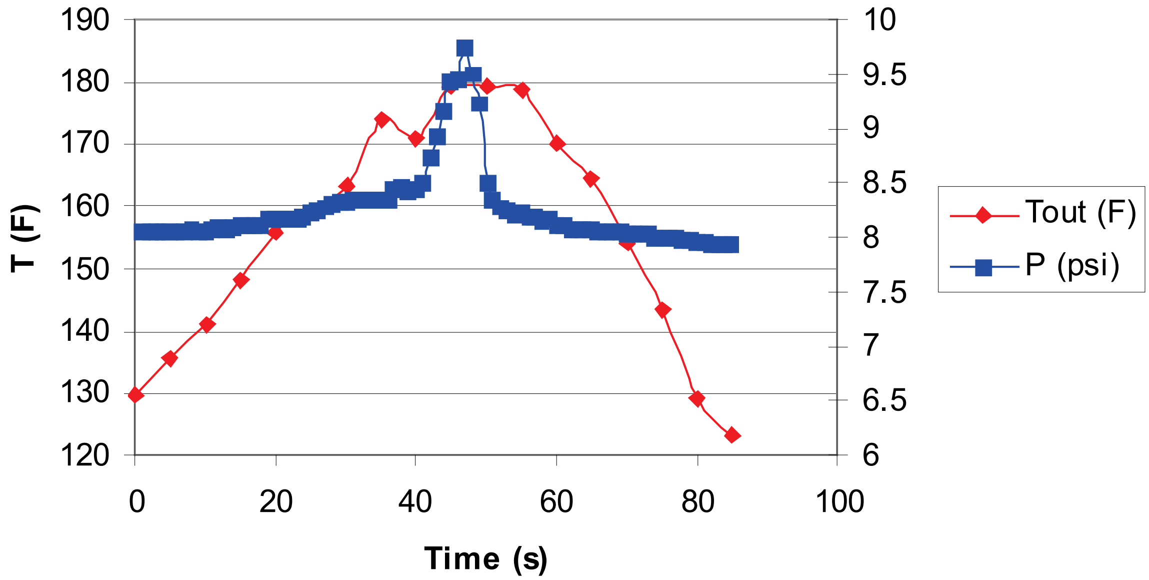


Figure 15. Direct cooling mode. Lower boiler pressure. 0.9 gpm flow. Illustration of temperature and pressure trends showing a boiling occurrence.

For all tests with the direct cooling mode the entering water temperature was 70°F.

Based on these test results it is concluded that these boilers can be operated with flow rates significantly lower, and temperature rises significantly higher than the manufacturer's recommendations. The boiler pressure should be maintained at the high end of its allowable range.

5 Phase III. Design & Evaluate System Performance in New, Single-Family Applications

Phase III involved evaluating the plans and specifications for three new homes, making design recommendations based on information gained from Phase I, and monitoring the performance of these systems once installed.

Construction of all three homes was completed in December 2009/January 2010. During the week of January 11th through the 17th, 2010, SWA set up the monitoring equipment and performed initial testing on the condensing boilers. Temperature and flow sensors were installed by the HVAC contractors for the respective homes and were connected to the data logging equipment by SWA. Data collected through September 2010 has been analyzed for this report.

5.1. House Characteristics

Three separate homes in Ithaca, NY were monitored during this phase of the project. One, House #1, is a single family, detached dwelling. The other two units make up a duplex: House #2 and House #3. Each unit has its own heating and domestic hot water system, which is comprised of a condensing boiler, baseboard convectors, two separate heating zones, and an indirect domestic hot water tank.

Figure 16. House #1

Figure 17. House #2 & House #3

In general, all three homes have the same or similar characteristics and/or efficiency levels. The following table lists the basic specifications for each dwelling.

Table 8. House Characteristics for Monitored Homes in Phase III

Component	House #1	House #2	House #3
Size	1328 ft^2	1300 ft^2	1100 ft^2
Beds/Baths	3 beds/1.5 baths	3 beds/1 bath	2 beds/1 bath
Foundation	Unconditioned Walkout Basement	Unconditioned Basement	Unconditioned Basement
Foundation Insulation	R-10 Exterior Rigid	Same	Same
Wall Insulation	R-21 blown cellulose	Same	Same
Windows	Low-e, argon, wood (U-0.25/0.25 SHGC)	Same	Same
Ceiling	R-50 blown cellulose	Same	Same
Infiltration*	< 3 ACH@50	< 3 ACH@50	< 3 ACH@50
Mechanical Ventilation	Exhaust only, Panasonic WhisperGreen bath fan	Heat Recovery Ventilator	Heat Recovery Ventilator
Space Heating	95 AFUE Condensing Boiler w/ baseboard convectors & outdoor reset	Same	Same
Space Cooling	None	Same	Same
DHW	Indirect, 40 gallon tank	Indirect, 30 gallon tank	Indirect, 30 gallon tank
Appliances	EnergyStar Dishwasher & Refrigerator	Same	Same
Lights	100% Fluorescent	Same	Same
Design Heating Load	19,500 Btuh	16,000 Btuh	15,000 Btuh

*Based on previous experience with this builder

5.2. Design Recommendations

Space Heating

Phase I confirmed the suspicion that condensing boilers are often not installed and configured in a manner that promotes condensing. Although the GAMA ratings for the Munchkin Contender are as high as 95 AFUE, this was not the annual efficiency of the installed systems as determined by monitoring the return water temperature to the boiler. As discussed in Section 3, the reasons for this are essentially design and configuration problems. Boiler maximum supply temperature, boiler capacity, flow rate through the system, and baseboard length all play a part in the overall efficiency.

To determine the best combination of settings, SWA began by performing a bin temperature analysis to predict the frequency of condensing in the new homes to be built in Ithaca, NY for a variety of boiler supply temperatures and flow rates. These supply temperatures were then used to size the baseboard convectors.

The following chart shows the boiler curve for the Munchkin Contender if the maximum boiler supply ($T_{s,max}$) temperature is 160°F at an outside temperature ($T_{out,min}$)of 5°F and a minimum supply temperature ($T_{s,min}$) of 95°F at an outside temperature ($T_{out,max}$) of 68°F.

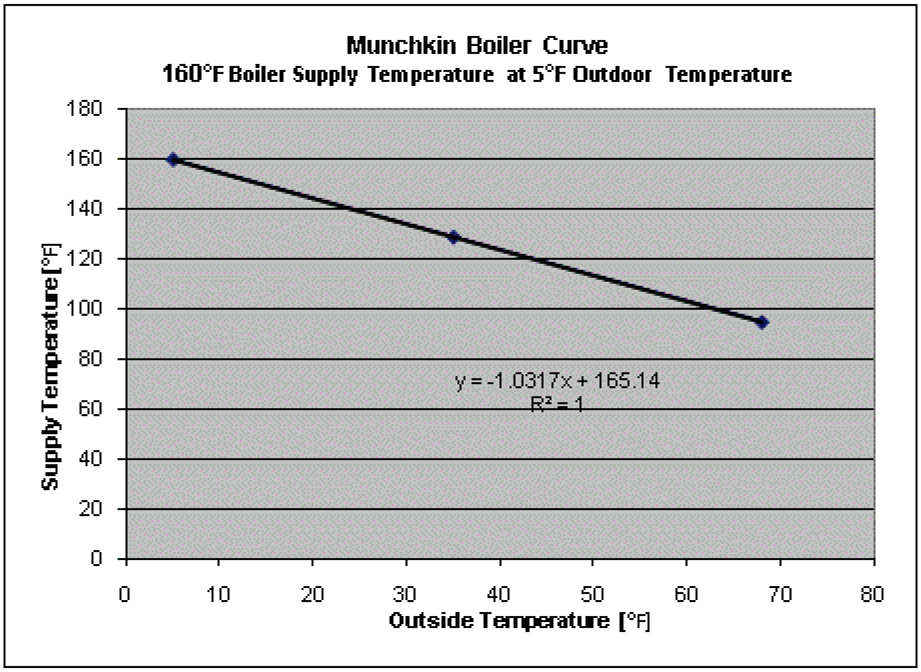


Figure 18. Boiler Curve @ 160°F Max Output Temperature

The return temperatures can be predicted at various outside temperatures by using 1) the equation for this curve, 2) the flow rate through the zones and 3) the UA of the home to be monitored. The following table shows the bin temperature profile for Ithaca, NY and the predicted boiler conditions for each bin.

Table 9. Bin Temperature Profile & Frequency of Condensing: 160°F Maximum Boiler Supply Temperature & 1 gpm Flow Rate

Bin Mean [F]	Bin Hrs	T_s [F]	Load @ Bin Temp [Btuh]	T_r [F]	ΔT [F]	% of Heating Season
62	738	101.2	1065.8	99.0	2.1	11.1%
57	715	106.3	1731.8	102.9	3.5	10.7%
52	691	111.5	2397.9	106.7	4.8	10.4%
47	643	116.7	3064.0	110.5	6.1	9.7%
42	686	121.8	3730.1	114.3	7.5	10.3%
37	776	127.0	4396.2	118.2	8.8	11.7%
32	767	132.1	5062.3	122.0	10.1	11.5%
27	507	137.3	5728.4	125.8	11.5	7.6%
22	381	142.4	6394.5	129.6	12.8	5.7%
17	297	147.6	7060.6	133.5	14.1	4.5%
12	209	152.8	7726.7	137.3	15.5	3.1%
7	119	157.9	8392.8	141.1	16.8	1.8%
2	68	160.0	9058.9	141.9	18.1	1.0%
-3	28	160.0	9725.0	140.5	19.5	0.4%
-8	20	160.0	10391.1	139.2	20.8	0.3%
-13	8	160.0	11057.2	137.9	22.1	0.1%
-18	2	160.0	11723.3	136.5	23.5	0.0%
	Percent annual heating load met in a condensing mode					89.7%

The boiler supply temperature (T_s) is calculated using the boiler curve and the mean temperature for each bin (T_{bin}).

$T_s = -1.0317 \times T_{bin} + 165.14$ (Eq. 5-1)

The load for each bin temperature was then calculated by multiplying the UA for the zone by the difference between the indoor (T_{in}) and outdoor temperatures (T_{bin}).

$Q_{Tbin} = UA \times T_{in} - T_{bin}$ (Eq. 5-2)

Finally, assuming the flow rate through the zones is one gallon per minute, the return temperature (T_r) was calculated as follows:

$T_r = T_s - [Q_{Tbin}/(\dot{m} \times cp \times m/h)]$ (Eq. 5-3)

where:

Q_{Tbin}: load [Btuh]

$\dot{m}$:	mass flow rate	[lb/m]
c_p:	specific heat	[Btu/lb °F]

For gas fired boilers, condensing occurs when the return temperature to the boiler is 130°F (Butcher, 2006) or less. The occurrences when the return temperatures are under 130°F have been highlighted in Table 9.

Domestic Hot Water (DHW)

Data collected in Phase I was not detailed enough to make specific recommendations for the optimal DHW design. SWA recommended that the builder install the same type of system as was installed in home #2 in Phase I of this study (see Table 2), which consisted of a 30 gallon indirect storage tank controlled by the Vision I controller. Of the six homes monitored during that phase, that home had one of the higher frequencies of condensing during DHW calls. Smaller data collection intervals were used and flow rates through the DHW heat exchanger were collected during this portion of the study.

5.3. Specifications of Recommended vs. Installed Equipment

Based on the findings from the research conducted in Phase I and the bin temperature analysis, SWA made the following design recommendations for the homes in Phase III:

- Maximum boiler supply temperature should be set to 160°F
- Flow rate through each zone should be 1-gpm
- Baseboard sizing was based on an average water temperature of 150°F
- 30 gallon, indirect storage tanks controlled by Vision I controller
- The primary loop should be removed.

As shown in Table 9, it was predicted that the boilers in these homes would condense about 90% of the heating season if the above design recommendations were implemented.

SWA recommended a maximum boiler supply temperature of 160°F for multiple reasons. First, this temperature resulted in baseboard lengths and specifications consistent with current practice. Second, if a lower temperature was selected resulting in 100% condensing frequency, it would be difficult to analyze the validity of these recommendations as non-condensing conditions may not have been encountered. In order to accurately predict when condensation stops, we need to determine the conditions under which no-condensing occurs. Knowing the limits of condensing performance is essential to creating general guidelines for effective installations.

The design heating loads were calculated using WrightSoft's Manual J8 software and the building specifications listed in Table 8. Based on those calculations, SWA developed the heating system specifications listed in Table 10. This table also shows what was actually installed. “As Recommended” means that SWA’s recommended specifications were implemented.

Table 10. Summary of Recommended and Installed Specifications for Phase III of Advanced Systems Condensing Boiler Research

Address	Recommended Spec’s			Installed Spec’s		
	Boiler	Baseboards	DHW	Boiler	Baseboards	DHW
House #1	50 kBtuh Munchkin Contender	26’ - 1st floor, 22’ – 2nd floor	30 gallon indirect tank, Vision I controlled	80 kBtuh Munchkin	31’ - 1st floor 23’ - 2nd floor	40 gal indirect tank, Vision I controlled but not compatible
House #2	50 kBtuh Munchkin Contender	21’ - 1st floor 18’- 2nd floor	30 gallon indirect tank, Vision I controlled	As Recommended	29’ - 1st floor + 12” toe kick heater in the kitchen* 32.5’ - 2nd floor	As Recommended
House #3	50 kBtuh Munchkin Contender	19’ - 1st floor 16’ – 2nd floor	30 gallon indirect tank, Vision I controlled	As Recommended	27.5’ - 1st floor + 12” toe kick heater in kitchen* 30.5’ - 2nd	30 gal indirect tank, aquastat controlled

*The toekick heaters provide approximately 3,100 Btuh at 140°F inlet on low speed. This is approximately equal to 3.5 times the calculated load for that space.

These specifications were provided to INHS before heating system installation began. Variations from CARB’s recommendations are due to one or more of the following:

- different heating contractors used for the single family home and the duplex
- timing of the projects – older specifications may have been used at the time the projects went out to bid
- contractor preference and availability of materials.

In House #1, the additional baseboard length on the first floor (over what was originally specified) was installed in the living room. The 5’ of baseboard recommended in the foyer was split between the powder room and the foyer, as was the baseboard recommended for the second floor bath, which was split between the bath and the landing area at the top of the stairs.

For both units of the duplex, all rooms appear to have approximately double the baseboard over what was originally recommended. Additional baseboard over what was recommended is not a problem in general. Lengths of baseboard above the recommended design will only allow more heat to transfer to the space before the water returns to the boiler, thereby lowering the return temperature below what it would have been with shorter lengths of baseboard. Lower return temperatures are definitely desirable. This may also help shorten recovery time.

Floor plans of each home along with room by room summaries of the recommended baseboard lengths for each dwelling are located in the appendix. Each summary was provided to the builder before construction on the heating systems began.

A significant change to the normal system design was the elimination of the primary loop that included both the primary loop pump and the closely spaced tees. In primary loop set-ups, an independent circulator loop is activated whenever any of the heating zones call for heat. The primary loop is a supply-to-return loop from which each of the individual heating zones are fed. This loop is thought to keep a consistent flow of water through the heat exchanger to protect it from boiling, and to stop one zone in the system from affecting the flows to other zones.

Based on the results from the first phase of this study, it is suspected that the return temperatures from the heating zones were being increased when they mixed with the warmer water circulating in the primary loop thus reducing the frequency of condensing. Increasing the return water temp to the boiler could also increase cycling, as it would take less time to raise the return water to the desired supply temperature.

CARB specifically requested that the primary loop be eliminated in these three homes to prevent this from happening. Because this is in contrast to the manufacturer's recommendations, the systems were closely monitored to ensure that they were providing adequate heat to the zones and the hot water tank, and that the boiler is not short cycling due to overheating or low pressure conditions across the heat exchanger.

Phase III Long Term Monitoring

The following system parameters were monitored at one minute intervals:

- Gas Use
- Flow Rates
- Water Temperature
    - Boiler supply and return water temperature
    - Supply water temperature to the zone
    - Return water temp for each zone
    - Domestic hot water supply and return temperatures
    - Domestic water temperature from the tanks to the loads.
- Air Temperatures
    - Outdoor air temperature
    - Indoor temperatures in each of the bedrooms and the main living area.
- Pump Power

Figure 19 shows the intended location for the boiler's flow and temperature sensors. Verification of sensor operation was conducted during the initial site visits. Data has been output every minute and downloaded regularly via a wireless connection.

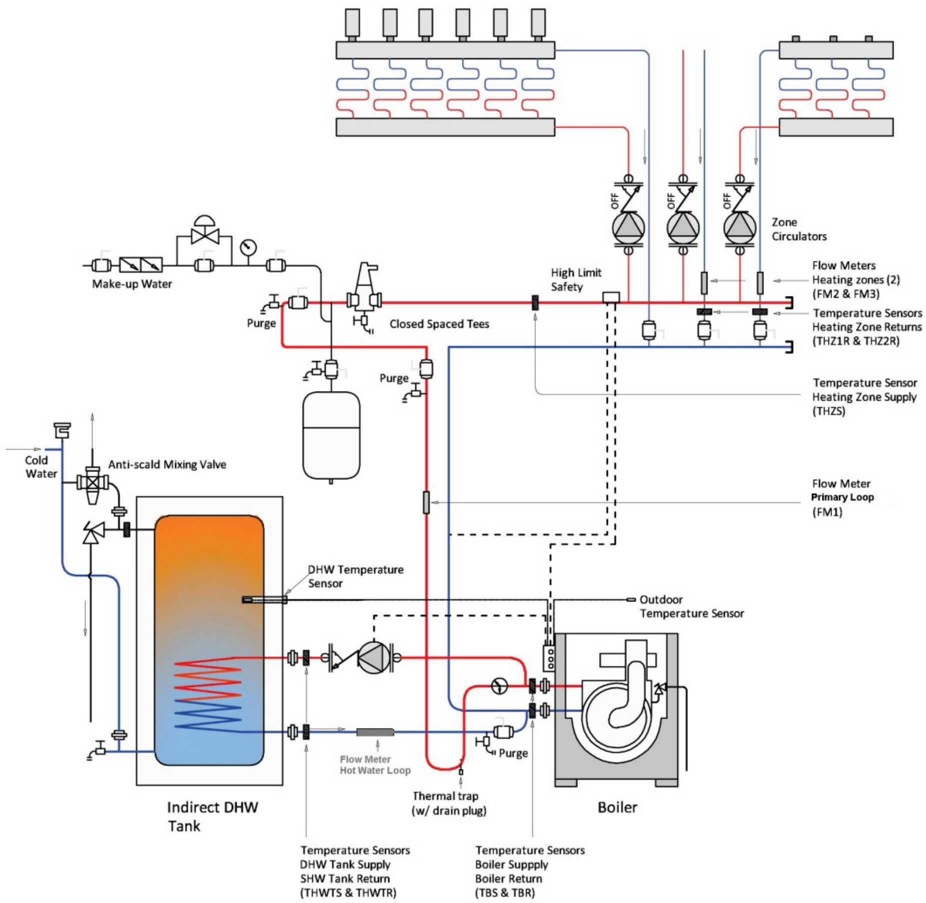


Figure 19. Flow and Temperature Sensor Location

5.5. Tests Performed During Initial Site Visit

During the initial site visit in January 2010, tests were performed on each system to ensure they were running properly and the recommended boiler setup did not cause any system performance problems. These initial tests were also conducted to ensure our sensors and data logging equipment were working. Finally, it was SWA's intention to use the results of these tests to determine the best combination of settings for each home in this study.

In a couple of instances problems with the zone controls were detected and corrected. Air needed to be bled out of one system and serious problems with the DHW system at House #1 were identified.

Also, discrepancies between SWA's temperature sensors and the boilers' sensors were noted and recorded. It is presumed that the reason for this is the installation of SWA's sensors. They were not installed in the specified locations in more than one instance, and the pipe fittings used to house the sensors held the tips of the sensors out of the main stream of the water.

There was a 6°F difference between our return sensor and the return water temperature being reported by the Vision I controller at House #1and House #3 and a 10°F difference at House #2. Values were adjusted accordingly, and all discussion of the frequency of condensing has been performed using the adjusted values.

Below is a general description of the tests performed. Not all tests were possible at each home.

Domestic Hot Water (DHW)

The first test performed on each DHW system involved a normal draw at the faucet. The hot water was turned on at one or more of the sinks until the hot water tank called for heat. Once the DHW pump started circulating water through the heat exchanger, the faucet was shut off. The time it took for the boiler to satisfy the tank was noted along with the boiler supply and return temperatures, which were recorded in order to evaluate the percent of the time the boiler was condensing. The pump flows were altered and this test was repeated to see if the flow had a significant effect on the recovery rate.

During the second test, the DHW tank was drained of all hot water until the temperature of the water leaving the tank was close to that of the water coming in from the main. The faucets were then shut off, and the time to come back to temperature and boiler supply and return temperatures were noted.

Finally, the shower was allowed to run for 15 minutes. The intent here was to see if the boiler could keep up with the demand at a given boiler supply temperature to provide enough hot water for a 15 minute shower. Table 11 summarizes the tests performed and the results of each.

Table 11. Results from Testing on DHW Systems During Initial Site Visits

	System Specifications	Sink Draw	Test Performed Cold Start	Shower Call	T_s °F	Frequency of Condensing for Each Test %
House #1	80 kBtuh Munchkin/Tank incompatible	30 minutes (4.5 gpm)*	n/a	n/a	180	n/a
House #2	50 kBtuh Contender/30 Gallon Indirect, Vision I Controlled	11 minutes (4.0 gpm) 13 minutes (5.0 gpm) 12 minutes (5.4 gpm)	62°F starting tank temp 31 minutes	108°F supply temp to shower after 15 minutes, 8 min. till boiler started	180	70/?**/70
House #3	50 kBtuh Contender/30 Gallon Indirect, Aquastat	13 minutes (4.2 gpm)	62.7°F starting tank temp 26 minutes	97°F supply temp to shower after 15 minutes, 4 min. till boiler started	160	70/80/70

*Tank was incompatible with system and was not functioning properly. No further tests were conduct on the DHW at House #1.

**Loss of data during this test.

Space Heat

Four separate tests were conducted on each heating system. Because recovery from setback in the first phase of this study was so slow in all the homes tested, this was a focus of the initial tests. Each heating zone was tested individually, as well as together, to see how quickly the space would come up to temperature. The boiler's outdoor reset was also bypassed to evaluate the affect on recovery time. To accomplish this, the lower limit of the boiler curve was set at the same temperature as the high limit, essentially eliminating the reset function.

All pumps used were Grundfos, UPS15 58PC, three speed pumps except for the DHW pump at House #1 which was a Taco 007-F51IFC. The pump flows were recorded and adjusted during each test. Ultimately, flows of 1 gpm were desired, but in most cases, flows were above 2 gpm on the lowest pump settings when the pumps were running individually. If the flow fell below 1gpm, the pump setting was increased. This only happened at one of the homes when both zones were calling for heat simultaneously.

The temperature of the zones was recorded before and after each test. Frequency of condensing was also evaluated. Table 12 provides a summary of the space heat tests and the results for each home. Each day of testing the outdoor temperature was approximately 40°F.

Table 12. Results from Testing on Space Heating Systems During Initial Site Visits

		Temperature Rise in Each Zone for Test Performed [°F/hour]					
	System Specifications	**Zone 1**	**Zone 2**	**Zone 1 & 2 Outdoor Reset**	**Zone 1 & 2 No Outdoor Reset**	**$T_{s,max}$ Reset/ No Reset °F**	**Frequency of Condensing for Each Test %**
House #1	80 kBtuh Munchkin	0.5	1.2	0.6 &1.3*	2.4 & 4.2	160/180	100/100/100/9
House #2	50 kBtuh Contender	1.1	1.3	2.1 & 2.4	5.3 & 5	160/180	100/100/100/19
House #3	50 kBtuh Contender	1.6	2.5	2.1 & 3.9	4.3 & 7.7	150/150	100/100/100/40

*It is anticipated that these numbers are low. During this test, the door to the basement was left open and not discovered until the beginning of the following test listed in the table.

The highlighted row in the above table illustrates the rate at which the zones came to temperature when both zones call for heat and the outdoor reset was working. At an outdoor temperature of 40°F the outdoor reset control kept the maximum boiler supply temperature to approximately 120°F when the zones called for heat. This is consistent with the predicted boiler supply temperatures in Table 9.

In addition to this low boiler setpoint, there is a 30°F differential programmed into the Vision I controller that prevents the burner from firing until the return temperature is 30°F lower than the boiler's setpoint. For example, if the setpoint is 120°F based on the outdoor reset, and the supply temperature exceeds 120°F by approximately 7°F, the burner will shut down and will not fire again until the return temperature has decreased to 90°F.

This combination of factors plays a very large part in why the zones take so long to come to temperature. If the heat is set back 5-degrees at night, it will take House #1 more than 4-hours to come back up to temperature and between 2- and 2.5-hours for House #2 and House #3 on a day when the temperature is 40°F outside. These numbers are consistent with the rates observed in the first phase of this study.

Solutions to this problem may include a bypass switch, or override features, built into the boiler if the temperature has not been satisfied within a set amount of time, or eliminating setback altogether. If the outdoor reset was bypassed, the response time would be cut approximately in half.

5.6. Final Settings

The following table lists the thermostat and boiler settings that were programmed during the site visit and are currently being evaluated. Because these systems seem to bring the heating zones to temperature so slowly, heating zone setback schedules were varied to determine the best options for performance and comfort.

Because the flow rates are higher than our initial recommendations, boiler setpoints were set slightly lower than our original design recommendations for these homes. When the outside temperature is 5°F, the maximum boiler supply temperature has been set to 150°F instead of 160°F. At 150°F maximum boiler supply temperature and 2.5 gpm flow, the predicted frequency of condensing and baseboard sizes remained relatively unchanged.

Table 13. Thermostat and Boiler Settings During Monitoring Period as Programmed by CARB

Settings		House #1	House #2	House #3
Thermostats (°F)				
on thru Fri	6:00 am	68	68	68
	8:00 am	65	62	65
	6:00 pm	68	68	68
	10:00 pm	65	62	65
Sat & Sun	8:00 am	70	70	70
	10:00 pm	62	62	65
Boiler Settings (°F)				
	Supply to Zones at 5°F outside	150	150	150
	Supply to zones at 68°F outside	95	95	95
	Differential to zones	30	30	30
	Supply to DHW	160	180	160
	DHW Tank Setpoint	95*	119	130
Pump Speeds (gpm)				
	Zone 1(alone/+Zone2)	3.1/2.4(low)	3.0/2.6(low)	3.2/2.3(low)
	Zone 2(alone/+Zone1)	3.0/2.4(low)	1.9/1.4(high)	2.5/2.1(low)
	DHW (alone)	4.5**	4.0 (low)	4.2(low)

*This is a temporary setting until tank is fixed/replaced. Final setting should supply 120°F water at the tap.

**DHW pump at House #1 was a Taco. Speed was not adjustable. All other pumps were Grundfos, 3-speed pumps. All pumps set to "low" except zone-2 at House #2.

The aquastat in House #3 was set to 130°F by SWA. At this setting, 120°F water was measured at the kitchen sink. The tank setpoint in House #2 should probably be increased as well. During testing, with the tank setpoint of 119°F, the water at the kitchen sink was approximately 108°F according to the HVAC contractor for that home. SWA did not change this setting.

6 Results and Discussion

6.1. Evaluation of System Performance

From mid January through the end of September of 2010, data was collected at one minute intervals from all sensors as described in Section 5.4. Figure 20, Figure 21 and Figure 22 display a representative weekend day near design conditions for each home. The boiler at House #2 is attempting to recover from an 8°F setback, and at #3 a 5°F recovery. The graph for House #1 was included for comparison, but the thermostat settings were changed by someone after SWA left the site, and were holding at 60°F in both zones.

These figures show that, once thermostat settings were reached, the boilers were able to maintain the temperatures in each home. The design heating loads, as listed in Table 8, range from 15,000 Btuh for the smallest home to 19,500 Btuh for House #1. The boilers installed in each unit of the duplex have a capacity of 50,000 Btuh (833.3 Btu/min), and an 80,000 Btuh (1333.3 Btu/min) unit was installed at House #1.

Although the lowest modulating rate is higher than the design load for each home, these boilers modulate to higher rates regularly. The algorithm in the boiler control increases the firing rate based on certain conditions such as 1) the difference between the boiler supply temperature and the boiler supply setpoint (based on the outdoor temperature) and 2) the difference between the supply and return temperatures. This is why, even though the lowest firing rate should maintain an interior temperature of 70°F under design conditions, the boilers modulate to higher output rates.

Unfortunately, the manufacturer was reluctant to share detailed information about the control algorithms with SWA, as the manufacturer felt it was proprietary. Still, as can be seen in the graphs, even under design conditions, these boilers rarely reach max firing rate. In fact, with few exceptions, they only reach the maximum firing rate when there is a call for heat from the DHW tank.

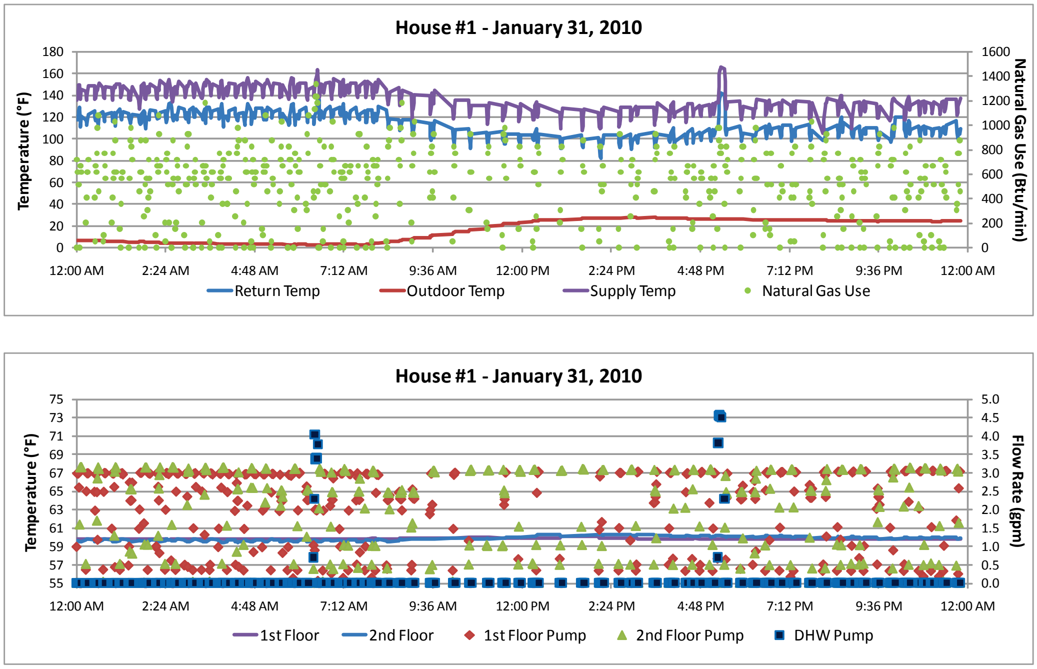


Figure 20. Boiler Operating Conditions for House #1 at Design Conditions: January 31, 2010.

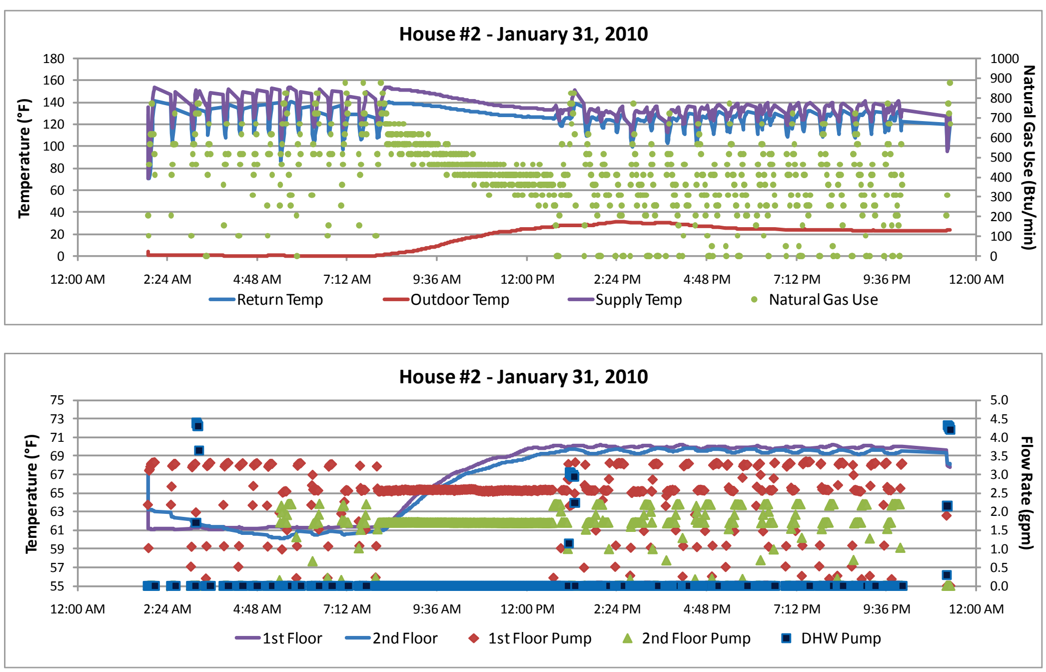


Figure 21. Boiler Operating Conditions for House #2 at Design Conditions: January 31, 2010.

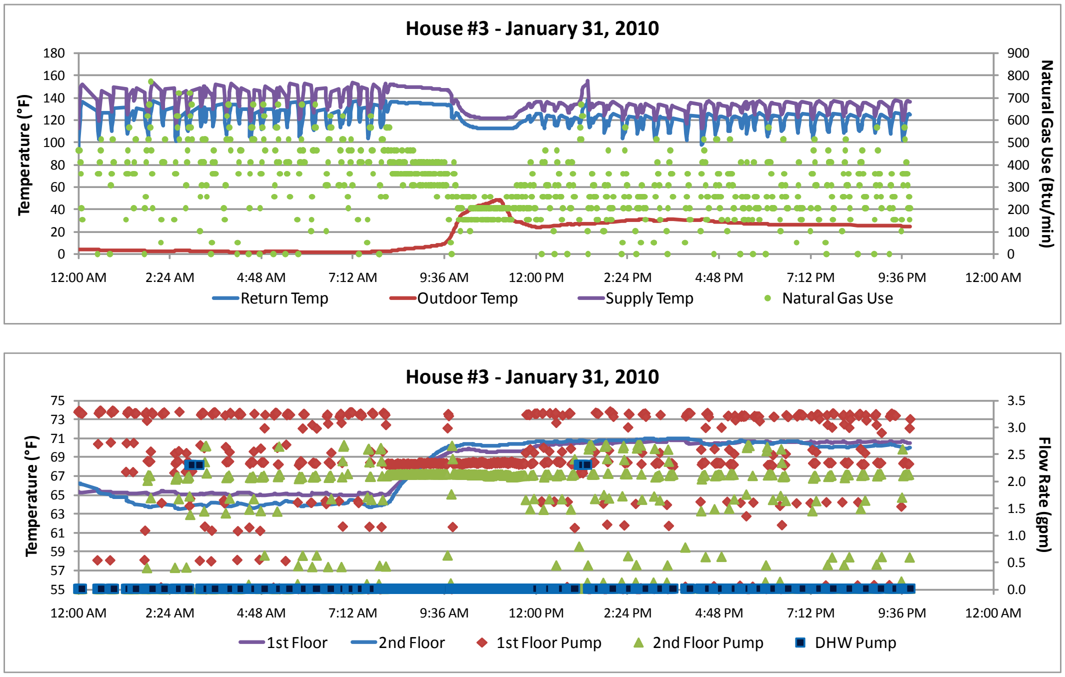


Figure 22. Boiler Operating Conditions for House #3 at Design Conditions: January 31, 2010.

Table 14 summarizes the frequency of condensing in each home for space heat, DHW calls only, and a combination of the two for the entire monitoring period. Based on the weather conditions recorded during the monitoring period, the frequency of condensing for space heat shown in Table 14 is similar to the predicted frequency based on the bin temperature analysis.

As noted earlier, these homes were unoccupied for most of that period. In July 2010, all three homes were sold and occupied. The frequency of condensing during DHW calls was analyzed for July through September 2010 for each home to see if occupant use affected system performance. In each case, the frequency of condensing during a call from the DHW tank was similar to the value listed in the table for the entire monitoring period. No reduction or improvement in efficiency was detected due to occupant behavior.

Table 14. Evaluation of System Performance for Each Home for the Period of January 2010 through September 2010.

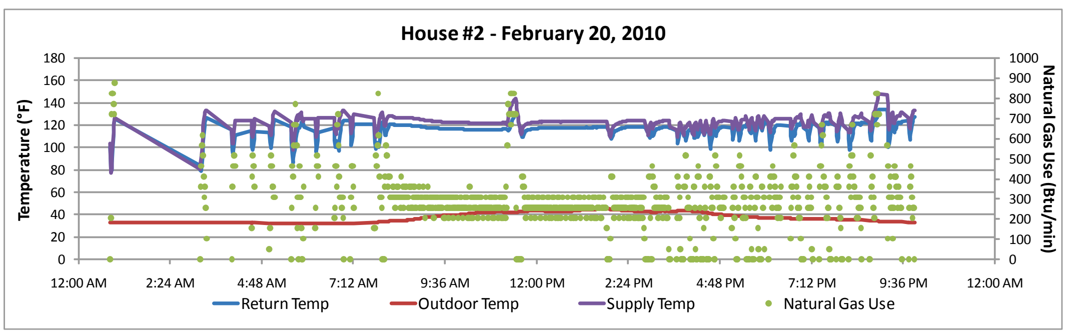
	Frequency of Condensing (%)			% of Time Boiler Fires During Call for Heat
	Space Heat Only	DHW Only	Overall	
House #1	97	64	97	34
House #2	96	56	93	68
House #3	96	64	94	75

Although the system at House #1 appears to be functioning in condensing mode most of the time, it should be noted that this system cycles on-an-off much more frequently than those installed at the duplex. During a call for heat, the boiler at House #1 fired only 34% of the time as compared to 68% and 75% for the units in the duplex. This could be indicative of severe oversizing, insufficient baseboard capacity and/or less than optimal boiler settings. Excessive cycling could possibly lead to reduced life of the unit.

One of the issues noted in Phase I was the lengthy recovery from setback. More than two hours to recover from a 6°F setback was not uncommon and in some cases the thermostat setpoint was not achieved. This issue was evaluated again in Phase III and the same problems identified. In each of the three homes, recovery time from setback is extremely slow and often not achieved.

As can be seen in Figure 21 and Figure 22, it took both units of the duplex four hours to come to temperature. For House #2, the setback was 8-degrees and for House #3 it was 5-degrees. It should be noted that Unit 2 would have come to temperature sooner if the outdoor temperature sensor hadn't spiked, thereby causing the boiler to lower the supply temperature. It appears from the data, that the outdoor reset sensor at House #3 is located where the sun hits it, causing an inaccurate reading of the outdoor temperature. Other data during that time period indicates that House #3 would recover from a 5-degree setback in about 2.5 hours at similar outdoor conditions.

The recovery time gets worse as the outdoor temperature increases as can be seen in Figure 23 and Figure 24. At outdoor temperatures in the upper 30s, House #2 took almost six hours to recover from an 8-degree setback, and House #3 recovers from a 5-degree setback in approximately four hours.



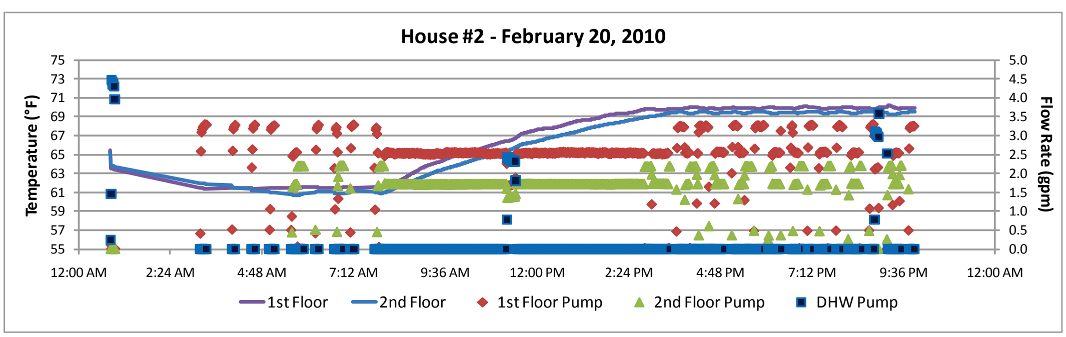


Figure 23. Boiler Operating Conditions for House #2: February 20, 2010

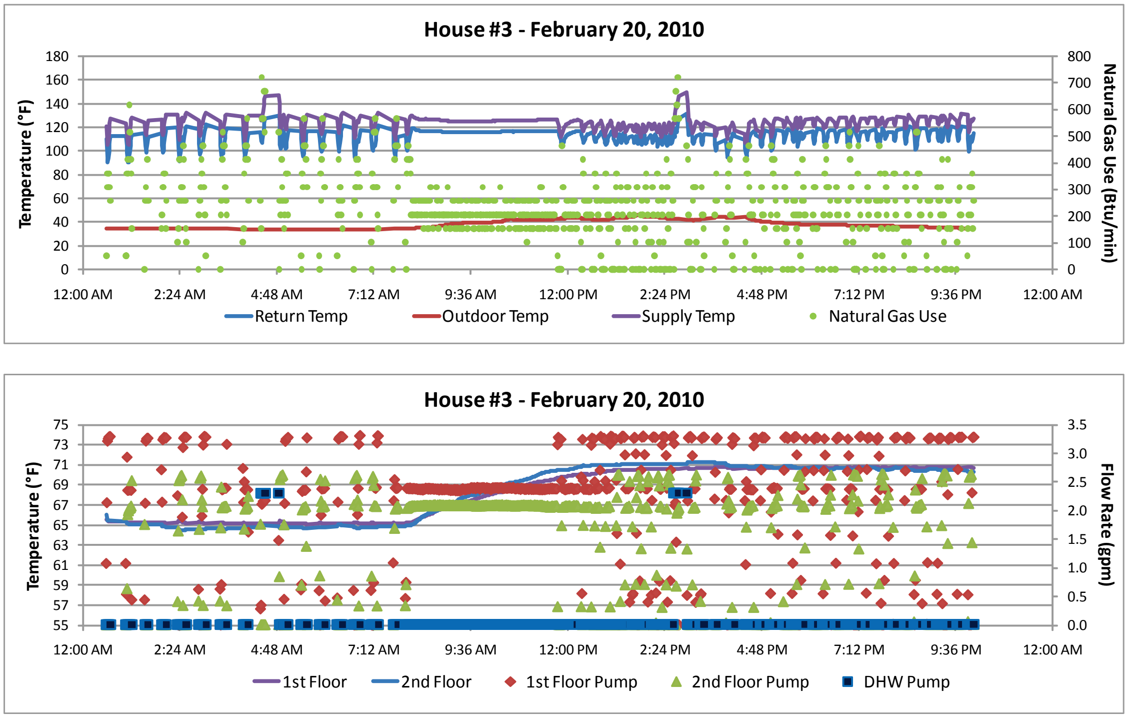


Figure 24. Boiler Operating Conditions for House #3: February 20, 2010

Recovery from setback was often not achieved in the duplex, and it was never achieved on the first floor at House #1 (see Figure 25 & Figure 26). As noted earlier, the baseboard length in the living room was increased by 50% from 12' to 18' but the kitchen baseboard was slightly shorter than recommended, and half the length intended for the foyer was installed in the powder room. A review of the temperature sensors located in the living room and the kitchen show average temperatures of 5°F+ in the living room during the heating season. While the living room reached the desired temperatures, the kitchen and foyer (where the thermostat was located) did not.

It is suspected that the two story configuration in these homes is contributing to the inability of the first floor system to come to temperature. These thermostats were placed in the stair wells at the top and bottom. The thermostat on the first floor of House #1 was also located directly next to the door to the unconditioned walkout basement. On several occasions, it was discovered that the door to the basement had been left open by someone working on or visiting the home. This would undoubtedly exacerbate the stack affect in the home, causing the warm air on the first floor to rise to the second.

Upon investigating this further, it was discovered that the first floor zones ran approximately 22–35% more often than second floor zones in all three homes. A different design method may be warranted, such as installing more capacity in the living spaces and first floor vs. the bedrooms and second floor spaces.

Another possible explanation for this is that the heat loss calculations underestimate the losses from the first floor to the basement resulting in a predicted design load lower than actual.

Besides the affects of the outdoor reset control, the boiler's differential control also affected recovery. The differential is an adjustable setting programmed into the Vision I controller that prevents the burner from firing until the return temperature meets this differential. The intention of the differential is to reduce cycling. For example, if the setpoint is 120°F based on the boiler curve settings, once the supply temperature exceeds 120°F plus an allowable overshoot (not necessary half the differential), the burner will not fire again until the return temperature has decreased to 30°F below the boiler's setpoint. The differential comes set from the factory at 30°F, but can be adjusted by the installer.

System cycling at House #1 was analyzed for the period from 4:00 a.m. to 8 a.m. on February 12, 2010. During this period, the burner fired continuously for an average of 10 minutes and then was off for 12 minutes on regular intervals. Supply temperatures ranged from 133 to 108°F, averaging 121 degrees. Given the outdoor temperature of 25°F and flow rates of 2.5 gpm recorded during that period, the boiler curve calls for a supply temperature of 133°F. Therefore, the average temperature supplied to the zone is too low. The warmer it gets outside, the more pronounced this becomes considering that this boiler is so oversized. As shown in Figure 27, when the boiler finally fires at the end of the day, the supply temperature setpoint is met in six minutes, and then the water circulates through the baseboard for 40 minutes before the boiler fires again. At 60°F outdoor temperature, the boiler is attempting to supply 100°F water to the baseboards. With the differential of 30°F, the average temperature being supplied to the zone is 89°F.

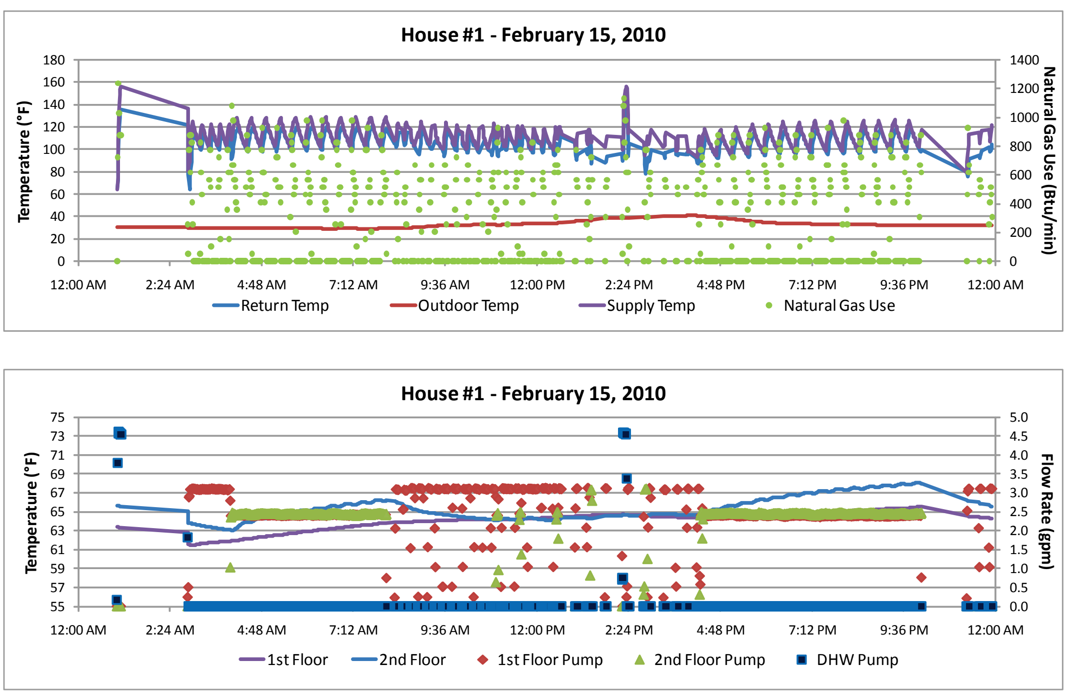
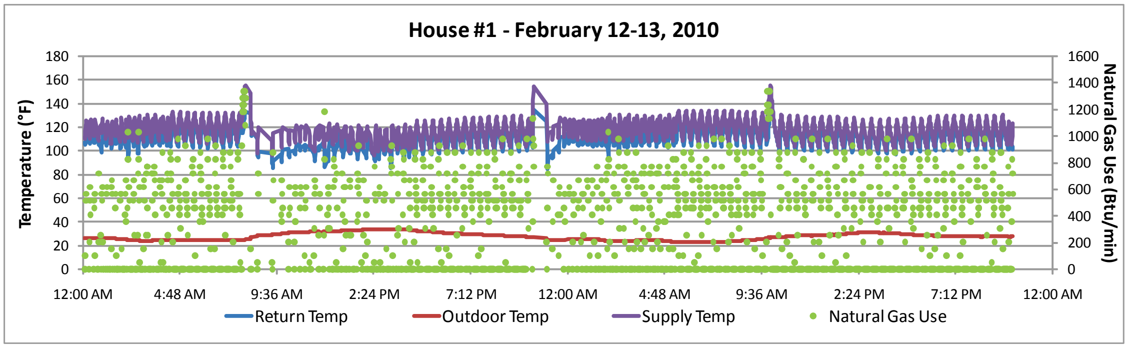


Figure 25. Boiler Operating Conditions for House #1: February 15, 2010



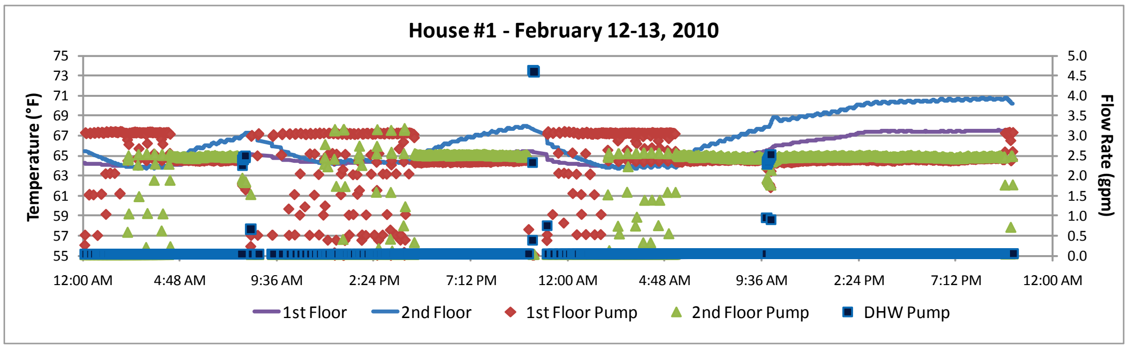


Figure 26. Boiler Operating Conditions for House #1: February 12 & 13, 2010

One option for decreasing recovery time is to increase the minimum boiler supply ($T_{s,min}$) temperature on the boiler curve. Unfortunately, increasing the minimum supply temperature on the boiler has a significant negative effect on efficiency (see Table 15).

Table 15. Predicted Frequency of Condensing for Various Boiler Curve Settings: Maximum Boiler Supply Temperatures vs. Minimum Boiler Supply Temperatures

	Frequency of Condensing at Different $T_{s,max}$ (1, 2 & 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%

This affect on the efficiency was seen in Phase I in the home that had difficultly with the toe kick heaters. The fan in the heaters would not start if the supply temperature was below 140°F. This left the occupants without heat in the kitchen for much of the heating season. To solve the problem, the minimum boiler supply temperature was set to 145°F so that the heaters would turn on. The frequency of condensing in that home was estimated at 14% based on the data collected. When looking at the boiler curve for a maximum supply of 180°F and a minimum supply of 145°F, no condensing is predicted. Of course, there will be some condensing at cold start up as with a non-condensing boiler, but this will be minimal.

Another problem detected occurs in the swing seasons when the nights are cool, but the daytime temperature rises above the maximum outdoor temperature ($T_{out,max}$) on the boiler curve. This temperature prevents the boiler from firing when the outdoor temperature passes a certain level. The upper limit comes from the factory set at 68°F. The problem occurs when the indoor temperature is lower than the thermostat setpoint and the zone calls for heat. The pumps will run, but the boiler will not fire. As can be seen in all three homes in Figure 27 through Figure 29 for the same day in April, the pumps ran for the entire setback period without the boiler firing.

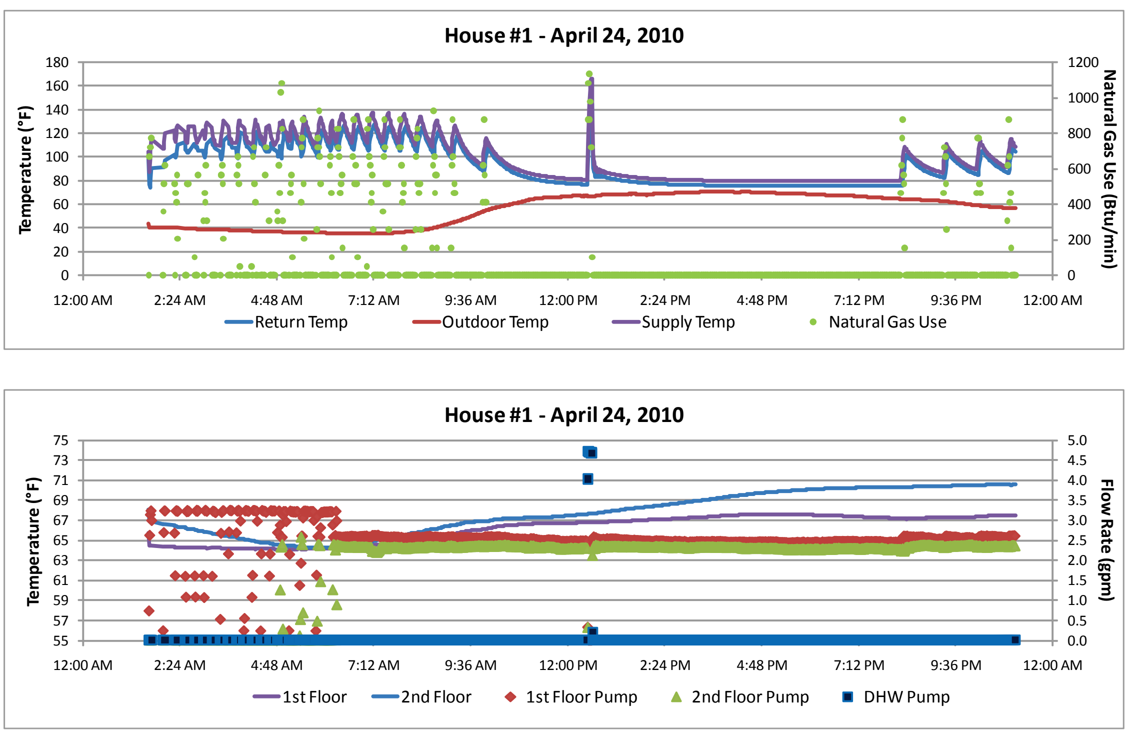


Figure 27. Boiler Operating Conditions for House #1: April 24, 2010

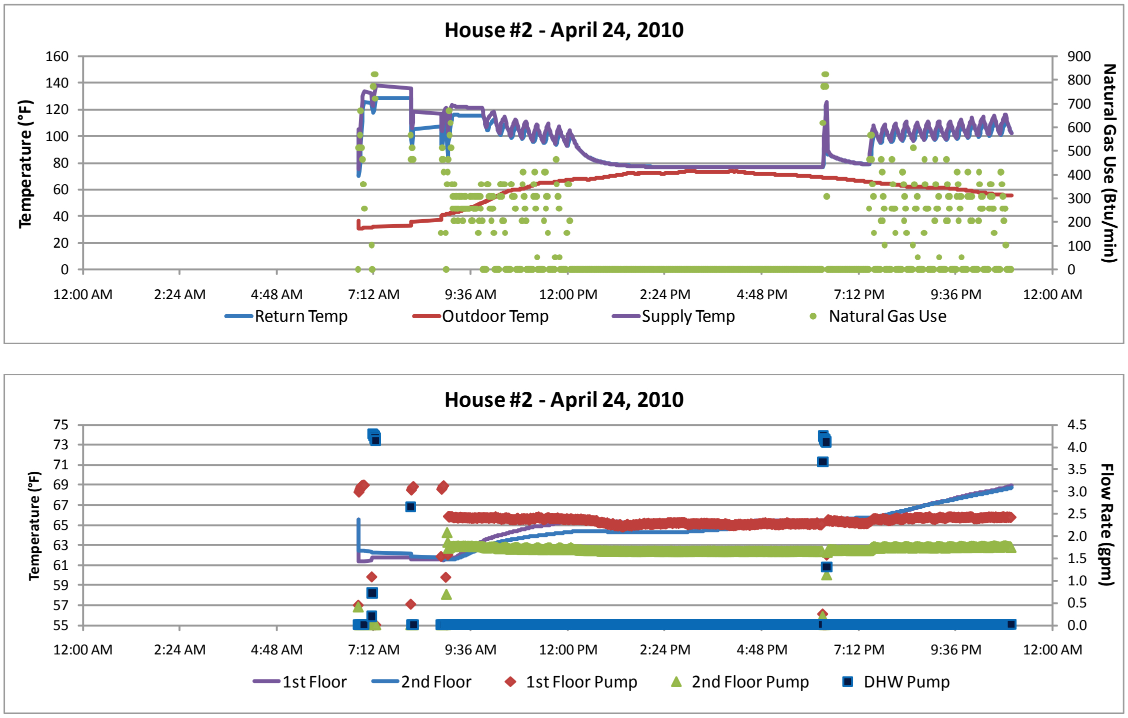


Figure 28. Boiler Operating Conditions for House #2: April 24, 2010

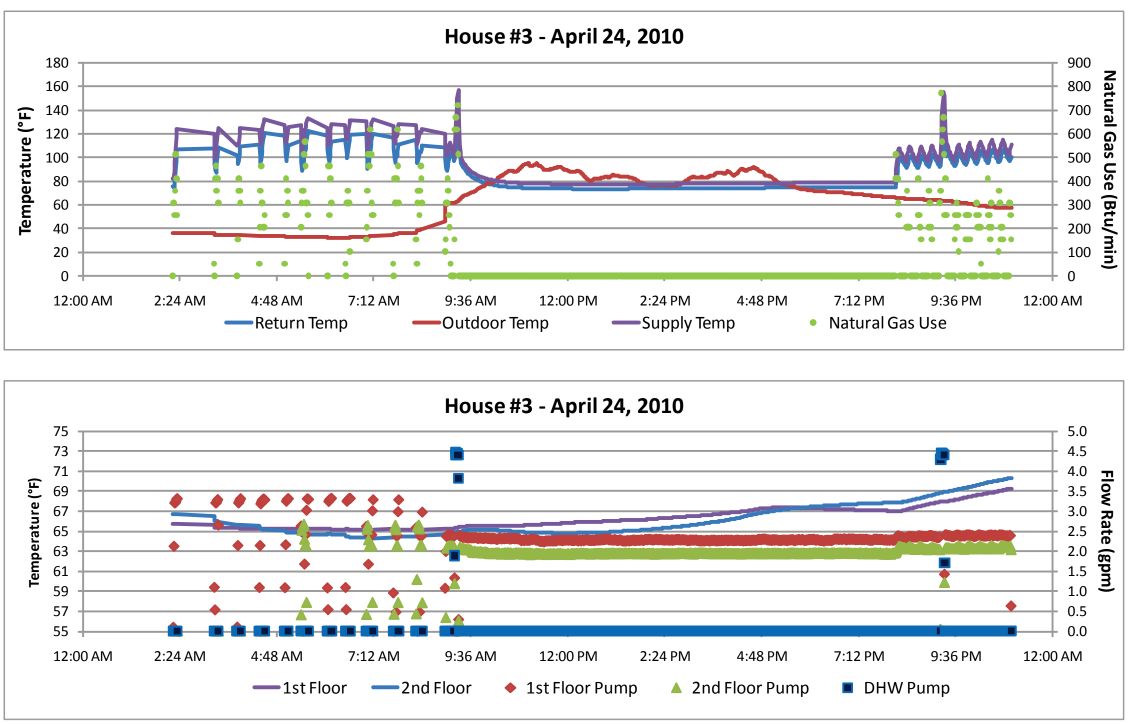


Figure 29. Boiler Operating Conditions for House #3: April 24, 2010

One solution is to raise the maximum outdoor temperature at which the boiler will fire. As can be seen in Table 16, this has a small affect on the frequency of condensing.

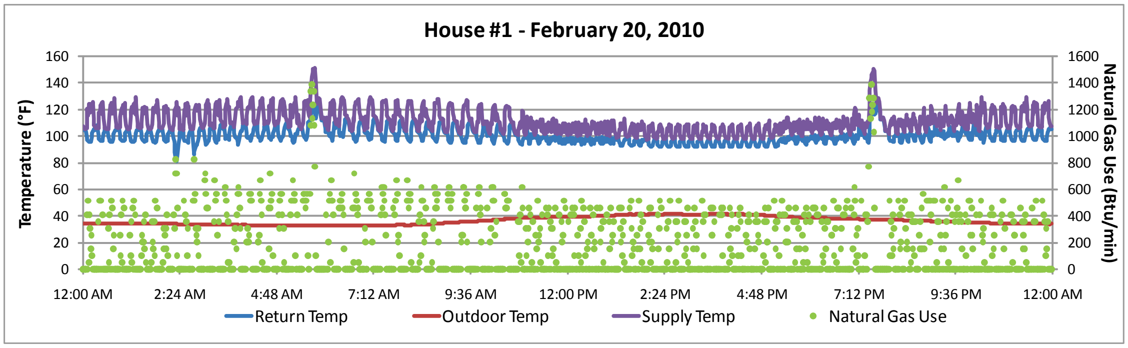
Table 16. Predicted Frequency of Condensing for Various Boiler Curve Settings: Maximum Boiler Supply Temperatures vs. Maximum Outdoor Temperatures

	Frequency of Condensing at Different $T_{s,max}$ (1, 2 & 3 gpm)											
$T_{out,max}$		150			160			170			180	
68	99%	91%	87%	90%	80%	77%	79%	68%	64%	66%	57%	53%
70	99%	90%	87%	89%	79%	76%	78%	66%	64%	64%	54%	52%
72	99%	90%	86%	88%	78%	74%	77%	64%	61%	63%	52%	51%

Another parameter affecting efficiency and boiler operation is flow rate. The recommended flow rates for these systems were 1gpm through each zone. Actual measured flow rates were higher than optimal and are displayed in Table 13. At the flow rates recommended and a maximum boiler setpoint of 150°F, it was predicted that the boilers would condense 99% of the time. At 2.5 gpm, the prediction is 89% frequency of condensing.

There are several barriers to achieving the specified, low-flow rates that would optimize these systems. First, contractors don't have standard, simple methods for measuring and/or setting flow rates. Second, until just recently, low flow, modulating residential pumps for which the flow can be adjusted by the installer have been unavailable. Also, different boiler manufacturers have different recommendations for minimum flows through the heat exchanger, under which they will not warranty the equipment, and therefore, installers are reluctant to design systems with very low flows. This is one of the main purposes of the primary loop: it maintains a constant flow through the heat exchanger regardless of how many zones are calling for heat. Thomas Butcher from Brookhaven National Laboratory evaluated these systems under very low flows as explained in Section 4.2 of this report.

CARB evaluated low flows for a few weeks at House #1 by closing off the ball valves until 1 gpm was flowing through each zone. Because there is no primary loop built into these systems, there were times when the flow rates through the boiler's heat exchanger were only 1 gpm. A summary of the operating conditions under the different flow rates is provided in Table 17. Data from two days in February with similar outdoor conditions was used.



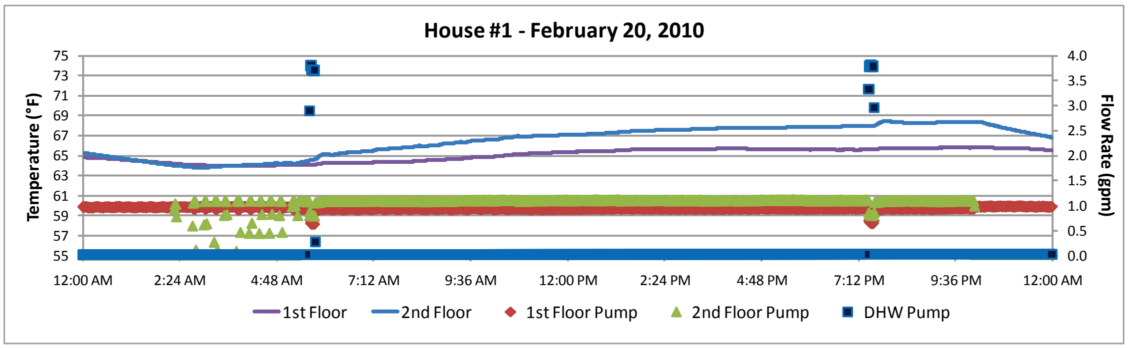
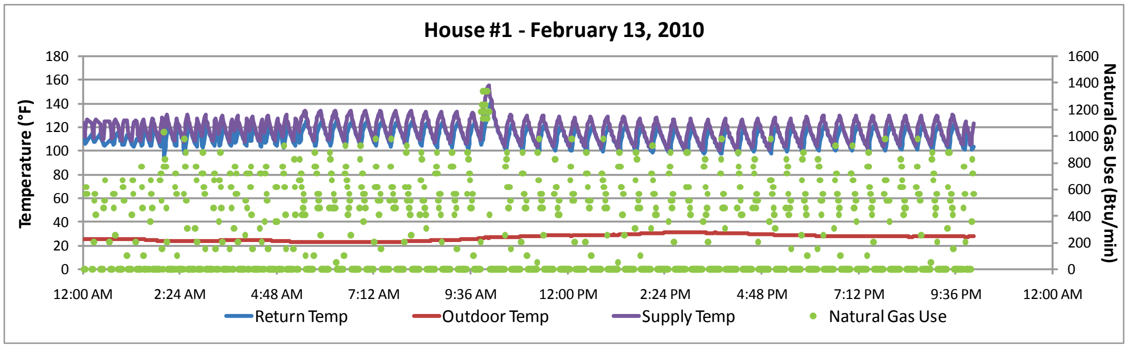


Figure 30. Boiler Operating Conditions for House #1 at 1 gpm Through Each Zone: February 20, 2010



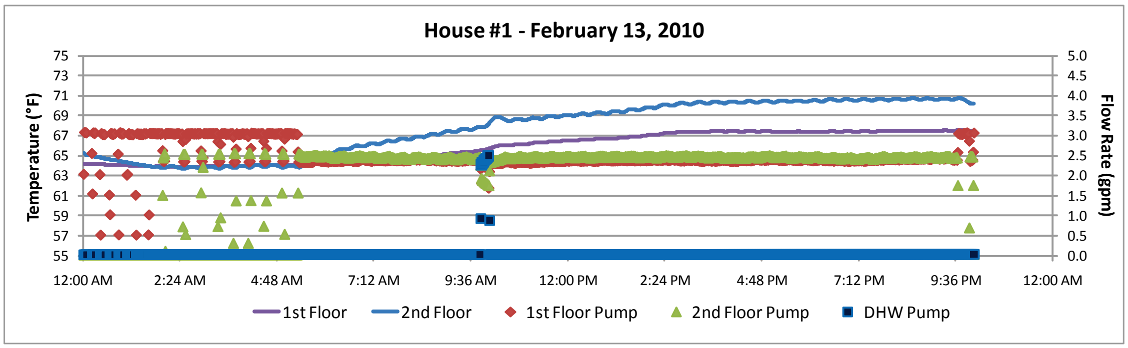


Figure 31. Boiler Operating Conditions for House #1 at 2.5 gpm Through Each Zone: February 13, 2010

Positive outcomes of slowing the flow include an increased temperature difference between the boiler supply and the return, lower maximum firing rates, and less boiler cycling. Unfortunately, the recovery time increased, and for this home, which is already having difficulty coming back to temperature, that is an undesirable effect. This could be offset by increasing the boiler supply temperature in relation to the increased ΔT of the supply and return.

Table 17. Comparison of Boiler Operation at House #1 with Two Different System Flow Rates

Pump Flows (gpm) Zone1/Zone2	Average ΔT	$T_{s,ave}$	$T_{out,ave}$	Max Firing Rate	Average Firing Rate	% Firing During Call for Heat	Recovery Time
	°F	°F	°F	kBtuh	kBtuh		°F/hour
2.5/2.4	13	126.2	29.6	65	32	31	0.5/0.9
0.9/1.1	19.3	125.5	33.5	49	21	47	0.35/0.35

Elimination of the primary loop did not appear to adversely affect system performance, but removing the primary loop leads to control issues such as domestic hot water (DHW) priority set up, and may require wiring diagram changes. Not one of the DHW systems was given priority over the space heating zones.

When brought to the attention of the contractor, he immediately understood the issue and how to fix the problem, but this is not something that is included in the install manuals.

6.2. Energy Savings/Cost Analysis

EnergyGauge v.2.8.02 was used to compare these systems to baseline heating system consisting of an 80 AFUE atmospheric boiler and a 0.53 EF atmospheric domestic hot water tank.

As can be seen in the following table, installing a condensing boiler with baseboard convectors and an indirect domestic hot water tank increases the source energy savings by 7% when not condensing and by 11% when condensing consistently.

Table 18. Source Energy Savings: Boiler Operating in Condensing Mode vs. Benchmark Values

Source-Energy Savings Improvement vs. 80 AFUE Boiler/053 EF DHW				
	Whole House	**Heating**	**DHW**	**Costs Savings[1]**
Non-Condensing 87%	7%	8%	28%	$166.37
Condensing 95%	11%	15%	33%	$251.74

[1]$0.20/kWh, $13.68/MMBtu

The above analysis is based on a 2400 ft^2 home located in Ithaca, NY built to meet the 50% source energy savings goal as outlined in "Maximizing Residential Energy Savings: Net Zero Energy Home Technology Pathways"[1]. The particular boiler analyzed was the Munchkin Contender, the boiler monitored during this study. The efficiency of the boiler in condensing mode is assumed to be 95% (as rated by GAMA), and when not condensing, 87%. The efficiency of the water heater is assumed to be 92% of the AFUE of the boiler for each case. This is the recommended method for calculating the efficiency of an indirect water heater according to the national HERS guidelines.

An initial assessment of the annual energy and cost savings between a Munchkin Contender operating in condensing mode 100% of the time vs. 0% of the time is shown below.

Table 19. Annual Savings Associated with Boiler Operating in Condensing Mode

	Space Heat (MMBtu)	**Domestic Hot Water (MMBtu)**	**Annual Cost**
Non-Condensing	56.2	19.1	$1,030.75
Condensing	51.5	17.6	$945.38
Savings:	4.7	1.5	$85

[1] Anderson, R and D. Roberts, National Renewable Energy Laboratory.

The Department of Energy's Building America Cost Neutrality Requirements (used here as a standard reference) are based on comparing the incremental source energy and cost for each measure compared to a baseline, in this case the Building America Benchmark. A measure or strategy is deemed cost-beneficial if the incremental annual cost of the improvement, when financed as part of a 30 year mortgage, is less than the annual reduction in utility bill costs relative to the BA Benchmark reference house.

Table 20. Condensing Boiler Cost Neutrality Analysis

vs. 80 AFUE Boiler/0.53 EF DHW	Estimated Incremental System Cost	Additional Mortgage Cost	Cash Flow	Additional Mortgage Cost (w/ Federal Incentive)	Cash Flow (w/ Federal Incentive)
Non-Condensing 87%	$1,925	($155)	$11	($34)	$132
Condensing 95%	$1,925	($155)	$97	($34)	$217

Based on an incremental cost of $1,925 (provided by the builder) to install a properly sized and configured condensing boiler heating system with baseboard convectors and an indirect domestic hot water tank as opposed to an atmospheric boiler and hot water heater, this advanced system meets this criteria.

As shown in Table 19, the annual savings associated with a sealed combustion boiler running in condensing mode 100% of the time, as opposed to 0% of the time, is approximately $85 for a 2400 ft^2 home in Ithaca, NY. No additional costs are anticipated to achieve this higher efficiency.

The cost trade offs between installing an atmospheric boiler and a sealed combustion, condensing boiler are mainly associated with the elimination of a flue chase and B-vent that must be installed with an atmospheric boiler. Condensing units use pvc or stainless steel pipe and vent through the wall; usually the rim if located in the basement. Some material and labor savings are realized, but the boilers themselves and the controls necessary to ensure proper operation are more expensive than a standard system. In addition, a condensate pump needs to be installed. INHS estimates that the cost to upgrade to a condensing boiler with an indirect DHW tank is $1,925 more than a conventional boiler and water heater.

If a condensing boiler is being installed, the costs to ensure condensing, however, are virtually nothing. Baseboard convectors are commonly oversized in standard practice so proper sizing will not result in additional costs to the builder. The recommendations to remove the primary loop, lower the boiler's maximum output temperature and lower the flow rates should not result in additional costs and could present a cost savings with the elimination of the additional pump and plumbing associated with the primary loop.

7 Conclusions/Remarks

The combination of a gas-fired condensing boiler with baseboard convectors is a low-cost, energy efficient solution for high-efficiency residential space heating in cold climates. The modulating capability of some gas boilers makes them an excellent option for low-load homes. Condensing boiler technology has been around for many years and has proven to be a durable, reliable, and code compliant method of heating. The sealed combustion, direct vent arrangement results in a reduction of exposure to combustion byproducts inside the home. The likelihood that combustion gases will enter the home is extremely minimal, practically eliminating exposure to carbon monoxide and other combustion byproducts. Reductions in carbon emissions are also a result of the higher combustion efficiency.

The optimum parameters for a condensing boiler combined with baseboard convectors include properly balancing the boiler supply temperature at design conditions with the proper flow rate through the system and baseboard output capacity. Higher capacity baseboards, lower flow rates through the system, and lower supply temperatures, result in lower return temperatures and thus a higher frequency of condensing.

Nevertheless, during Phase I of this study, it was discovered that:

- condensing boiler systems when combined with baseboard convectors are often installed no differently than a system with a conventional boiler
- the boiler supply temperatures are commonly left at the manufacturer's setting of 180°F or higher
- the primary loop configuration (recommended by some boiler manufacturers) also appears to reduce the frequency of condensing
- baseboards are sized using this high output temperature
- Circulators move the water at speeds at least 2-to-3 times higher than optimum.

Other major findings from this study include:

1. All systems in this study experienced extreme delays in recovery from setback. These ranged from 2.5 hours to over six, and in many cases recovery was never achieved.
2. Slowing the flow rates at House #1 increased the temperature difference between the boiler supply and the return, lowered the maximum firing rates and resulted in less boiler cycling. This also, however, increased the recovery time.
3. Outdoor reset and low limit on the boiler curve may result in water temperatures too low to heat up the space.
4. No-heat conditions arise in swing seasons when the nights are cool, but the daytime temperature rises above the maximum outdoor temperature ($T_{out,max}$) on the boiler curve, preventing the boiler from firing even though the zones are calling for heat.

5. First floor zones ran approximately 22 - 35% more often than second floor zones in all three homes. A different design method may be warranted, such as installing more capacity in the living spaces and first floor vs. the bedrooms and second floor spaces.
6. During a call for heat, the boiler at House #1 fired only 34% of the time as compared to 68% and 75% for the units of the duplex indicating severe oversizing, insufficient baseboard capacity and/or less than optimal boiler settings.
7. Based on research conducted by BNL, it can be concluded that:

   a. any control technique that reduces the return water temperature, including lowering the boiler setpoint and/or reducing the loop flow rate, will significantly improve the achieved efficiency, and
   b. condensing boilers can be operated with flow rates significantly lower, and temperature rises significantly higher than the manufacturer's recommendations for the boiler analyzed.

8. Components that require a minimum supply water temperature, such as toekick heaters, should be specified with the knowledge that boiler temperatures vary depending on the outside conditions, and supply temperatures may drop below that required by various components.
9. Each boiler manufacturer has proprietary controls – high and low pressure cutoffs, high temperature cutoffs, differential settings to reduce cycling, etc. - making it difficult to determine the most efficient setup at install.
10. Elimination of the primary loop did not appear to adversely affect system performance, but removing the primary loop leads to control issues such as domestic hot water (DHW) priority set up, and may require wiring diagram changes.

There are several barriers to achieving the specified, low-flow rates that would optimize these systems. First, contractors don't have standard, simple methods for calculating, measuring and/or setting flow rates. Second, until recently, low and variable flow residential-scale circulator pumps have not been readily available. Also, each boiler manufacturer has different recommendations for minimum flows through the heat exchanger under which they will not warranty the equipment, and therefore, installers are reluctant to design systems with very low flows.

Although all the systems in Phase I and Phase III of this study were having trouble meeting setback, feedback from industry partners suggested that elimination of a setback option would not be an acceptable way to deal with this problem. Most home owners prefer the option of setting back their thermostats. Using an indoor reset control to detect a lag in recovery was the preferred way to deal with this problem. The indoor reset control would bypass the outdoor reset control and provide warmer water to the baseboards if, after a set amount of time, the zone was still calling for heat. More analysis is needed to determine if these systems are more efficient if setback is completely eliminated vs. bypassing the outdoor reset to decrease recovery time.

The market barriers concerning condensing boilers paired with baseboard convectors apply more to the system configuration and settings than to the actual equipment. The boiler output temperatures are consistently set too high to promote condensing as is the flow rate through the zones. Although there is a lot of information available on condensing boilers, baseboard convectors, and indirect DHW components, there is little information explaining the best combination of components and settings when these technologies are combined.

Although several baseboard manufacturers have started to publish sizing charts that list the output of their product at temperatures consistent with those of a condensing boiler, and several sections of the Air Conditioning, Heating and Refrigeration Institute's *Residential Hydronic Heating Installation and Design* guide provide information on using outdoor reset controls and properly sizing baseboard convectors, little advice is given on how to design the whole system to ensure condensing.

8 References

AHRI; “Residential Hydronic Heating: Installation & Design Guide”; Air Conditioning, Heating and Refrigeration Institute, Arlington, VA, 2009.

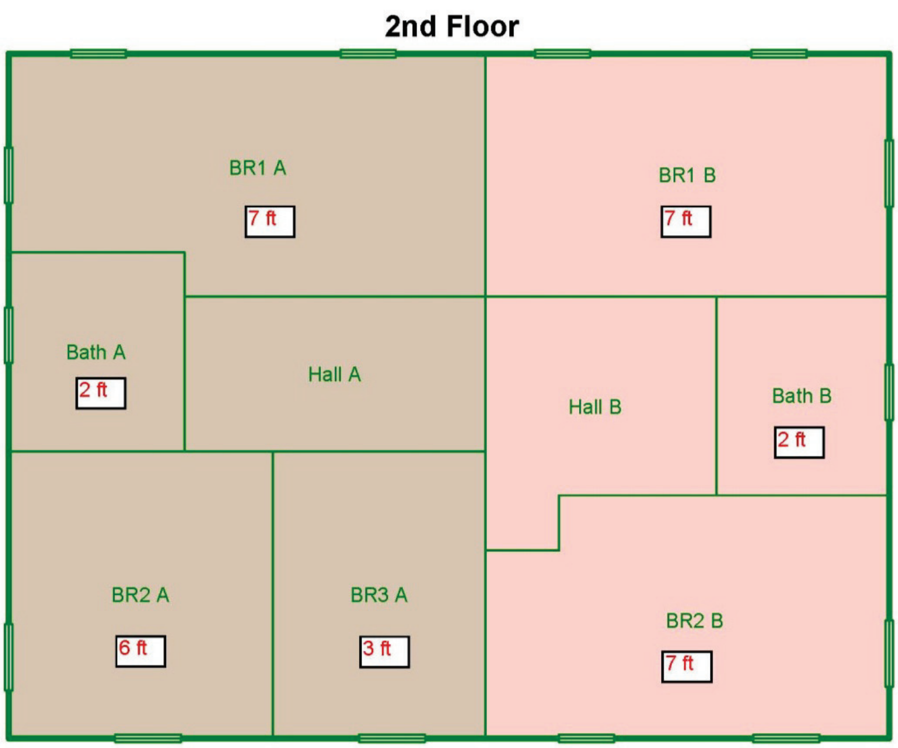
ASHRAE; “ASHRAE Handbook of Fundamentals”; American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, 2009.

Butcher, T. “Condensing boilers and baseboard hydronic distribution systems.”; *ASHRAE Transactions;* Vol. 112, part 1, 2006.

Kreider, J.F., Rabl, A.; “Heating and Cooling of Buildings”; McGraw-Hill, 1994.

9 Appendix

Recommended Baseboard Lengths for House #1



Recommended Baseboard Lengths for House #2 & House #3

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