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EVALUATION OF A HIGH-PERFORMANCE MULTIFAMILY RESIDENTIAL BUILDING IN ASPEN, COLORADO*

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ABSTRACT

Multifamily housing is an essential component of urban sustainable design, because finite available urban space necessitates greater use of attached housing and shared community spaces. Improving the energy efficiency of this category of housing presents special challenges, because units share walls and the space heating and domestic hot water systems are frequently centralized, requiring recirculation and long pipe runs to reach all units. These challenges were investigated as part of the first phase of a high-performance multifamily housing community called Burlingame Ranch in Aspen, Colorado. The first phase of the project has been completed, and features such energy efficiency measures as insulated slab-on-grade foundations, condensing boilers, solar preheat for hot water, heat recovery ventilation, and energy-efficient lighting and appliances. The authors participated in a thorough evaluation of key building systems for one prototype building and provided insights into potential design improvements that can be implemented in the second phase of construction.

INTRODUCTION

The City of Aspen, in partnership with the U.S. Department of Energy's Building America (BA) program and Building Science Corporation (BSC), is constructing a series of affordable, attached multifamily homes in multiple phases at Burlingame Ranch in Aspen, Colorado. The project is targeting very high levels of durability, comfort, and energy efficiency. The source energy savings target for the homes in Phase I was 51% [1] compared to the BA Research Benchmark [2]. This target was based on preliminary modeling performed by BSC using a relatively simple model and recommended design features, some of which were not installed in the final building. Key features included airtight spray foam insulation, insulated slabs, unvented attics, low-emissivity windows, and heat recovery ventilators (HRVs). Baseboard heating was provided by a primary loop heated using two high-efficiency boilers located in a central mechanical room. The domestic hot water (DHW) system consisted of an auxiliary tank heated by either a solar pre-heat tank or the primary space heating loop when insufficient solar energy was available. Individual pumps served each boiler and the DHW auxiliary tank, circulating water parallel to the primary heating loop. Building F-1, a 4-unit building constructed as part of Phase I, underwent detailed short-term performance testing by the authors in late March 2007. Follow-up testing was conducted on June 8, 2007. Long-term monitoring began once the homebuyers moved into the building in September 2007.

A photo of Building F-1 is shown in Fig. 1. The floor plan is laid out in Fig. 2. A schematic of the integrated space heating and DHW system is presented in Fig. 3. The as-designed specifications for the building are summarized in Table 1.

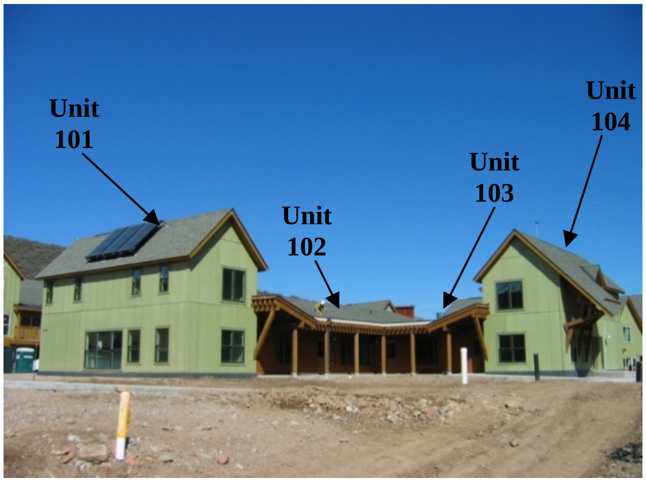


FIGURE 1. BURLINGAME BUILDING F-1, UNITS 101-104, VIEW FROM SOUTHEAST

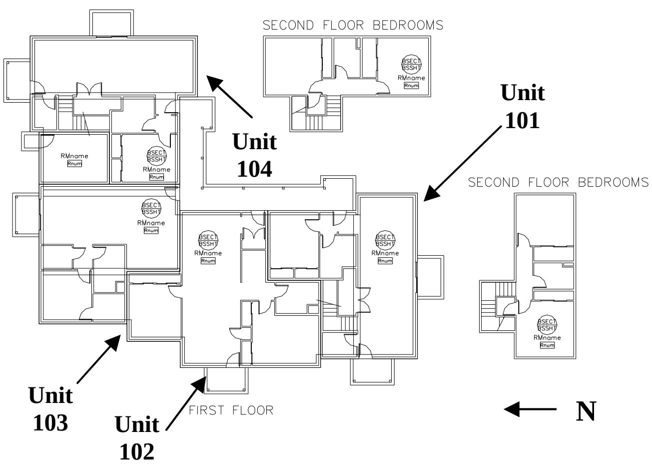


FIGURE 2. FLOOR PLAN FOR BUILDING F-1

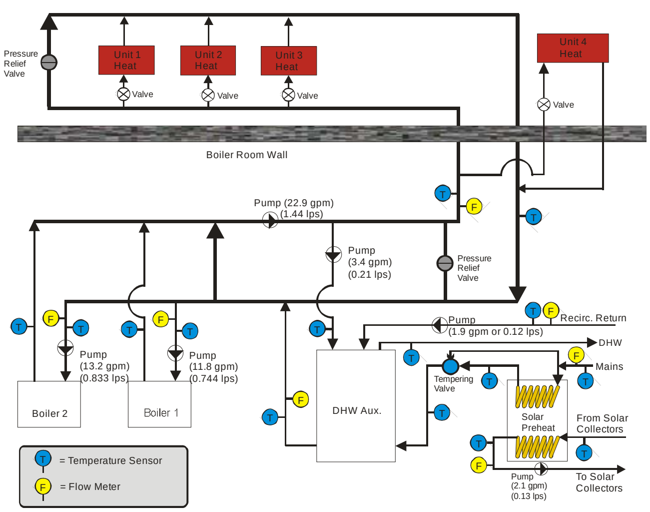


FIGURE 3. INTEGRATED SPACE HEATING AND DHW SYSTEM SCHEMATIC

TABLE 1. AS-DESIGNED SPECIFICATIONS FOR BUILDING F-1

Feature	Specification
Building	F1 (4 units)
Bedrooms	Units 101 and 104: 3-BR; Unit 102: 2-BR; Unit 103: 1-BR
Floor area	Units 101 and 104: 1,622 ft^2 (151 m^2); Unit 102: 1,149 ft^2 (107 m^2); Unit 103: 877 ft^2 (81 m^2)
Ceiling	Combination of cathedralized unvented attics and flat roofs, R-50 hr•ft^2•°F/Btu (R-8.8 m^2•°C/W) high-density foam at sloped roof, R-38 hr•ft^2•°F/Btu (R-6.7 m^2•°C/W) at flat roofs
Walls	2 in. × 6 in. (0.051 m × 0.152 m) with 3.5 in. (0.089 m) of high-density spray foam cavity insulation, R-24 hr•ft^2•°F/Btu (R-4.2 m^2•°C/W)
Slab	2-in. (0.051-m) XPS perimeter insulation extending 2 ft (0.61 m) below grade
Windows	Spectrally selective, low-emissivity glazing, U = 0.37 Btu/hr•ft^2•°F (2.1 W/m^2•°C), solar heat gain coefficient (SHGC) = 0.33
Infiltration	Target: Equivalent leakage area (EqLA) to the outside <2.5 $in.^2$/100 ft^2 (0.017 m^2/100 m^2) thermal envelope area
Space heating	Two 93% annual fuel utilization efficiency (AFUE) condensing boilers (central system) in mechanical room, baseboard heating, all plumbing in conditioned space or mechanical room
Cooling	None
DHW	120-gal (454-l) indirect tank off boiler in mechanical room, DHW recirculation system, 120-gal (454-l) solar preheat tank for DHW in mechanical room, 92-ft^2 (8.5-m^2) solar collector
Ventilation	HRV, 60% average heat transfer efficiency, ASHRAE 62.2 [3] rate for each unit when operating @ 33% duty cycle, supply to bedrooms and living room, exhaust from first floor bathroom
Ducts	Only for ventilation system, entirely in conditioned space, leakage target: <5% to outside
Lighting, appliances, and miscellaneous electric loads (MELs)	90% fluorescent hard-wired lighting; ENERGY STAR® dishwasher, refrigerator, clothes washer, and ceiling fans

The primary focus of this field test and analysis project was to evaluate the actual energy savings of advanced energy efficiency measures installed in a prototype multifamily building, and identify alternative approaches that could save additional energy in Phase II. These alternatives could then be evaluated by the builder, and perhaps by others pursuing similar multifamily projects, considering how they might affect purchase, installation, and maintenance costs. We did not specifically address the installed cost or the cost effectiveness of the energy efficiency measures discussed in this paper, primarily because first costs tend to be significantly higher for the first few houses that are built. Mature costs in a production context are more appropriate for examining cost effectiveness, and such projections are beyond the scope of this project.

SHORT-TERM TEST RESULTS

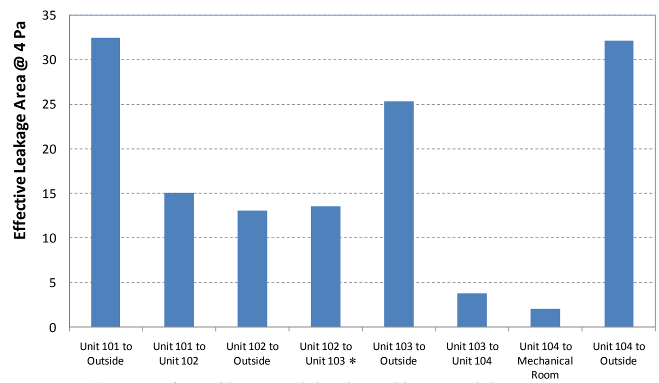
Blower Door and Flow Hood Testing

We conducted standard multipoint blower door tests for 3 of the 4 units on Sunday, April 1, 2007 to estimate the total effective leakage area (ELA) at 4 Pa (0.0006 psi). Unit 103 was not tested at that time because high wind conditions prevented accurate measurements. During a return visit on June 8, several tests were repeated because some measurements taken during the first test had questionable results. This was partly because of the high winds and partly because the builder had subsequently tightened some leaky areas in the attic between the units. Unit 103 was also tested during the second test period. Additional blower door tests were conducted with and without the adjacent units pressurized using a second blower door, to determine the fraction of leakage area that was in attached walls between adjacent units, as opposed to exterior walls. That fraction was then used to calculate the ELA to the outside and to any nonguarded adjacent units. The term *guarded* is used in this paper when an adjacent unit is pressurized to the same level as the test unit. Leakage from one unit to another usually does not affect energy use, so it is important to disaggregate the location of leakage area in each unit for the purpose of energy analysis. Even though inter-unit air leakage does not affect energy use, minimizing leakage between units of attached housing is important for inhibiting the spread of odors and other pollutants.

The effective leakage areas to the outside and to adjacent units based on the blower door tests are shown in Fig. 4. The net ELA to the outside and to each adjacent unit was calculated by apportioning the total leakage area based on the guarded test results. The guarded tests indicated that 32% of the leakage area for Unit 101 was in the attached wall with Unit 102. Similarly, 69% of the leakage area for Unit 102 was to adjacent units, along with 41% for Unit 103, but only 16% for Unit 104. Units 102 and 103 are interior units, and therefore have considerably more common wall area than do the two end units. Spray foam insulation was not used for the common walls, making them less resistant to air flow than the exterior walls. There were also interconnected attic spaces above Units 101, 102, and 103, contributing to the relatively large percentage of leakage area between these units.

Because there was significant communication between Units 101 and 103, most likely through leakage in the attic space, some of the calculations in Fig. 4 are suspect. The attic spaces and adjacent units were partly pressurized up to 25 Pa (0.0036 psi) when the test unit was pressurized to 50 Pa (0.0073 psi). These spaces were further pressurized when a guarded test was conducted, inhibiting some of the leakage from the test unit that would have been to the outside via the attic, or to unguarded adjacent units. As a result, we believe the leakage areas attributed to common walls are somewhat overstated, especially between Units 102 and 103. One indication of this error is that leakage area calculations for common walls were 20-30% different depending on which unit was pressurized relative to the other. Some directionality could be associated with the leakage between units, but given that the common walls are symmetrical, it seems unlikely that this could explain the difference in effective leakage area when flow direction was reversed. The more likely cause was the added complication of a buffer space in the attic, making the interpretation of guarded test results more difficult.

The normalized equivalent leakage area (EqLA) compared to the design target for each unit is shown in Fig. 5. All units met the target of less than 2.5 in.^{2}/100 ft^2 (0.017 m^2/100 m^2), which was based on EqLA at 10 Pa (0.0015 psi) and the envelope area adjacent to the outside.



* Average of 102 with 103 guarded, and 103 with 102 guarded

FIGURE 4. LEAKAGE AREA BETWEEN ADJACENT UNITS AND TO THE OUTSIDE BASED ON GUARDED BLOWER DOOR MEASUREMENTS (APRIL 1 AND JUNE 8, 2007)

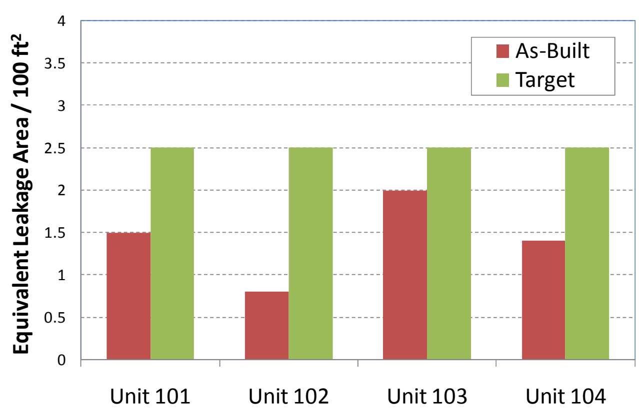


FIGURE 5. LEAKAGE AREA TO THE OUTSIDE COMPARED TO DESIGN TARGETS

Flow hood measurements were conducted on March 27 to determine flow rates for the HRV in each unit. Each HRV exhausts air from the first floor bathroom and supplies fresh air to each bedroom and the living room. Second floor bathroom exhaust fans operate independently from the HRV systems, and the ducts are physically separate. The HRV can be set for "min" or "max," depending on the intensity of the pollutant sources. It also has an "intermittent" setting that operates the

HRV at low speed (“min”) with a duty cycle of 10 minutes on and 20 minutes off. The builder used “intermittent” as the default setting.

The HRV in each unit met the minimum ventilation rate recommended by ASHRAE 62.2 when operating at high speed, after adjusting for high altitude. The flow rate at low speed in Unit 103 may be a little lower than ASHRAE 62.2, but the measurement accuracy is in doubt because the air flows barely registered on the flow hood readout. None of the HRVs would meet the ASHRAE 62.2 recommendations when operating at the expected duty cycle of 33%.

Tracer Gas Testing

We next performed a series of single-zone tracer gas tests in each unit to determine the envelope leakage under actual weather conditions, and to evaluate the net ventilation rate of the HRVs. A single-zone tracer gas system monitored the decrease in concentration of sulfur hexafluoride over time, while maintaining a uniformly mixed state throughout the test, allowing the calculation of overall air change rate for each unit in air changes per hour (ACH). Portable heaters were used to control the interior temperatures, and a combination of small portable fans, ceiling fans, and destratification fans (for 2-story units) were operated to help maintain uniform mixing throughout each unit. The results from 6 sampling points were averaged to calculate whole-house concentration at each time step.

The results of the tracer gas testing of Unit 101 conducted on March 30–31 are shown in Figure 6. The natural air change rate ranged from about 0.05 to 0.15 ACH, depending on wind speed and outside temperature. This was fairly consistent with the annual average air change rate based on the guarded blower door test, which was calculated to be 0.14 ACH to the outside. It was especially windy during the tracer gas tests, and the temperature was in the seasonable range of 20º–40ºF (–7º–4ºC). Tracer gas results generally offer a better representation of actual air exchange than estimated natural infiltration based on blower door test data for a given set of weather conditions. Unfortunately, we cannot use tracer gas results to separate leakage to the outside from leakage to adjacent units, but the driving pressure differences were likely much smaller between units compared to the outside.

An HRV bump test at low speed was conducted between 5:00 a.m. and 8:00 a.m. on March 31. A bump test is used to determine the change in whole house air change rate caused by changing the operating state of a building system such as an air handler or ventilation system when natural infiltration is steady. In this case, the bump test indicated that the net ventilation rate was about 0.18 ACH, or 39 cfm (18 l/s), based on a house volume of 12,976 ft^3 (367 m^3). This ventilation rate was lower than the 68 cfm (32 l/s) measured with a flow hood, but this was not surprising. Interactions can occur between infiltration and ventilation, even for nominally balanced systems, when there are local changes in pressurization or when supply or exhaust registers are proximate to large leaks in the building envelope.

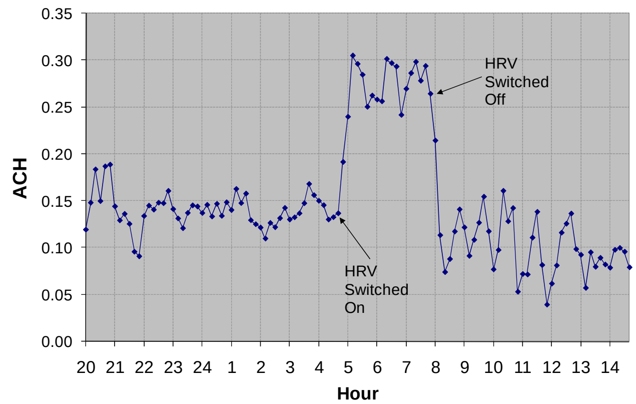


FIGURE 6. SINGLE-ZONE TRACER GAS TESTING OF UNIT 101 (MARCH 30–31, 2007)

A similar single-zone tracer gas test was conducted in Unit 102. The natural infiltration rate relative to the outside and the two adjacent units was about 0.05 ACH, which was consistent with the annual average of 0.06 ACH predicted based on blower door tests. Turning off the HRV appeared to reduce the air change rate by about 0.3 ACH, or 46 cfm (22 l/s) based on a volume of 9192 ft^3 (260 m^3). This was close to the 57 cfm (27 l/s) indicated by the flow hood measurements. However, the average wind speed dropped from 8 mph (3.6 m/s) to 2 mph (0.9 m/s) around the time the HRV was turned off, so the bump test probably suggested a larger change in net air exchange rate than would have occurred under steady weather conditions.

An additional bump test was conducted in Unit 102, where the bathroom exhaust fans in both adjacent units (101 and 103) were turned on and off to determine whether the pressure difference resulted in a higher air change rate in Unit 102. Gusty winds during the afternoon made it difficult to quantify the increase in ACH with any certainty, and a rapid drop-off in average wind speed around 7:00 p.m. again made it difficult to separate the wind effect from the effects of the exhaust fans in adjacent units. However, we were confident that the change in ACH was no more than 0.03 ACH, or 5 cfm (2 l/s), even if the entire effect was attributed to the exhaust fans.

A final single-zone tracer gas test was performed in Unit 103. The total air change rate to the outside and to adjacent units ranged between 0.1 and 0.2 ACH without the HRV operating. This was consistent with the estimated annual average rate of 0.16 ACH to the outside calculated based on the guarded blower door measurements. When the HRV was on, the air change rate increased to about 0.53 ACH, indicating that the net ventilation rate was about 62 cfm (29 l/s), based on a volume of 7016 ft^3 (199 m^3). This ventilation rate was substantially higher than what was measured with the flow hood, which was less than 38 cfm (18 l/s). Because there was no significant drop-off in wind speed correlated with turning off the HRV, it is difficult to explain why the net ventilation rate appears to be so much higher than the nominal rate, but local pressurization effects can often create complex interactions between ventilation and natural infiltration.

A multizone tracer gas sampling system was used in Unit 104 to evaluate natural infiltration and ventilation rate. The approach used to quantify the air exchange rate was similar to that used for the single-zone tests in the other units, except the samples were collected sequentially and analyzed individually instead of being mixed together before analysis. The average natural infiltration rate for Unit 104 during the test period was about 0.14 ACH, including air entering from the adjacent Unit 103. This result is consistent with the estimated annual average of 0.13 ACH based on the guarded blower door tests. The air change rate increased by an average of about 0.32 ACH during the HRV bump test, corresponding to about 69 cfm (33 l/s) based on a volume of 12,976 ft^3 (367 m^3). This result was consistent with the flow hood measurement of 70 cfm (33 l/s).

Space Heating and Hot Water System

We next turned our attention to the integrated, centralized space heating and DHW systems. We quickly noticed that the flow rates in both the heating loop and the DHW recirculation loop were quite high, resulting in very small temperature differences between supply and return, approximately 0.2–0.5°F (0.1–0.3ºC) for each loop. This small temperature change meant that we could not rely on calculations of thermal losses from the recirculation loops based on traditional flow-ΔT measurements. This was especially true during the short-term test periods, which occurred in the spring and summer months before the units were occupied and significant heating loads were present. However, the measurements of gas use were expected to be more accurate, so we ran a few experiments to isolate the two heating loads and measure the gas used by the boilers. First, the heating loop valve was closed, but the DHW recirculation loop was kept running. This resulted in a change in gas use from about 10.7 ft^3/hr (0.30 m^3/hr) to about 2.9 ft^3/hr (0.08 m^3/hr), implying that the space heating loop used about 7.8 ft^3/hr (0.22 m^3/hr) (input to the boilers). Next, the DHW recirculation loop pump was shut off. The resulting change in gas use was indistinguishable in the data, indicating that the heat losses from the DHW recirculation loop were much smaller than the other energy flows being met by the boilers.

On the morning of July 3, 203 gal (770 l) of water were drawn over a period of about 20 minutes to cool the solar preheat tank. At the end of the draw, the temperature of the water leaving the solar preheat tank had decreased to about 77ºF (25ºC), indicating that the tank was cool at the beginning of the day. The solar collector circulation pump started at about 7:30 a.m., cycling on and off until about 8:20 a.m., at which time the pump ran continuously until about 2:00 a.m. At the end of the day the pump cycled on and off, which is normal when the tank is hot and the sun is low. This experiment indicated that the solar water heating system was operating normally during the day. However, a few small flows during the night suggested that reverse thermosiphoning was occurring when the collector cooled off and the preheat tank was hot, resulting in losses equal to about 20% of the energy collected during the day. The effect was not large on this particular day, but it could be a significant problem under more extreme conditions. We recommended adding a check valve to the solar collector loop to make sure reverse flow could not occur. The builder agreed and added the check valve in September 2007.

Several additional problems were identified with the solar system before the units were occupied: (1) the outdoor temperature sensor was not connected properly to the solar controller; (2) there was no flow in the solar system, even with the pump running, because of an airlock somewhere in the system; and (3) there was a high cycling rate on the pump caused by shorting of the temperature sensor's wire nut connectors, which were immersed in water that had collected in the dip tube. The builder addressed all these issues before the building was occupied and instituted enhanced commissioning procedures for future buildings.

On March 30 and April 1, we evaluated the hot water distribution system by turning on the fixtures furthest from the recirculation loop in each unit and measuring the flow rate and the length of time before the hot water reached the typical mixed-water temperature of 105ºF (41ºC). The tests were performed in the early morning to make sure the water in the distribution system was at approximately the same temperature as the surroundings. Either the kitchen sink or the most remote bathroom sink was used to provide the first draw. The recirculation loop for the building operated continuously during the test, but individual units did not have dedicated recirculation loops.

The maximum wait time of 60 seconds occurred at the kitchen sink in Unit 101, which was the unit furthest from the mechanical room. In fact, the kitchen sink had the longest wait time in each of the four units, suggesting it probably had the longest pipe run, even if it was not physically the furthest from the recirculation loop. On average, the kitchen sink wait time was 50% shorter when there was an earlier draw at one of the bathroom sinks. Conversely, the bathroom sink wait time was 62% shorter when a kitchen draw occurred first. Clearly the order in which hot water events occur can have a significant impact on average wait time, and perhaps the amount of hot water wasted while waiting for hot water to arrive. Overall, the wait times are a concern, but appear to be no better or worse than they would be in a typical detached home with a dedicated hot water tank.

LONG-TERM MONITORING RESULTS

We installed long-term monitoring equipment and sensors to track the performance of building F-1 under occupied conditions over the course of a year. Data were collected at 1-minute intervals beginning on March 27, 2007, but only a limited number of conclusions could be drawn because the homebuyers had not yet moved in. By October 1, 2007, all units had been purchased and occupied; this marked the start of the long-term test period.

HRV Sensible Efficiency and Operation

The inlet and outlet air temperatures of the HRV in Unit 104 were monitored continuously starting in March 2007. This allowed its sensible heat recovery efficiency to be calculated in both hot and cold weather. The sensible efficiency, after adjusting for fan energy, is plotted in Figure 7 as a function of indoor-outdoor temperature difference. When the temperature difference was very small, the calculation of sensible efficiency was not meaningful because uncertainties in the measurements were larger than the heat recovered. However, the efficiency under these conditions is unimportant because the ventilation air would have a minimal impact on the space conditioning load. When the temperature difference was at least 9ºF (5ºC), the efficiency ranged between 65% and 90%, and appeared to approach 75% at very large temperature differences. The manufacturer's website asserts a sensible heat recovery efficiency of 69%–77% at an outdoor temperature of 32ºF (0ºC), corresponding to a temperature difference of about 38ºF (21ºC). The measured data show that the installed efficiency falls within the published range.

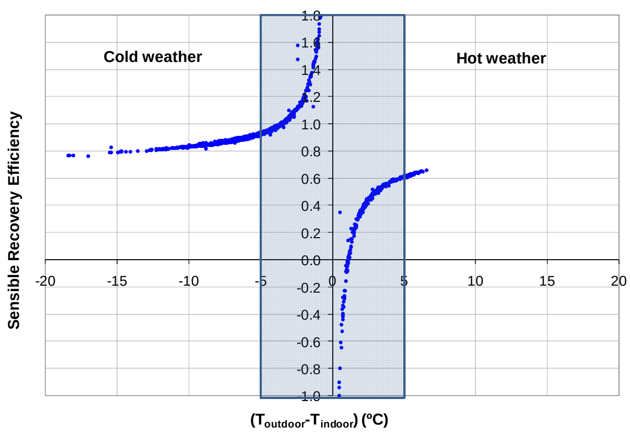


FIGURE 7. SENSIBLE RECOVERY EFFICIENCY OF THE HRV IN UNIT 104 UNDER A VARIETY OF WEATHER CONDITIONS

The status of the HRVs in all 4 units has been monitored continuously to determine how the occupants use them over the course of a year. We observed that the HRVs remained off following a power failure. Because the HRV controls were mounted near the HRVs, which were in the attics of some units, the occupants could not easily access them. In those units, the occupants (if they even realized the HRV was off) would have had to climb through a closet and into the attic to turn the HRV back on. We therefore had a concern that the HRVs would not be turned back on if a power failure occurred during the long-term test period and the controls were neither moved to more accessible locations, nor modified so the default state was the same as when the power went off. Our understanding is that the builder is looking into changing the default settings to "min" and providing the homeowners with remote controls for the HRVs.

Space Heating and Hot Water System

Recirculation loops for both the hot water and space heating systems were necessary because without them, the distance from the mechanical room to the furthest unit would have caused unacceptable delays in the delivery of hot water and space heating. These recirculation loops ran continuously, which may not have been necessary to meet the expectations of the occupants, especially during the summer when there was rarely a call for space heating. About 930 W (3170 Btu/hr) of electrical energy were used continuously for pump operation, not including the solar collector pumps. The primary space heating loop operated at 35 gpm (2.2 l/s) (whether in recirculation mode or space heating mode), as opposed to the design flow rate of 25 gpm (1.6 l/s), resulting in higher pumping energy than expected. The DHW recirculation system flow rate was about 7 gpm (0.4 l/s), which was twice as high as the specified flow rate of 3.5 gpm (0.2 l/s). At 7 gpm (0.4 l/s) the water was returning within 1º–2ºF (0.6º–1.1ºC) of its supply temperature. Electrical energy to power the recirculation pump could be reduced considerably by reducing the size of the pump (and thus the flow rate) so the returning temperature would be on the order of 10ºF (5.6ºC) below the supply. Because the power to run a pump generally rises with the cube of the flow rate, a reduction in the flow rate by a factor of 5 would result in a fivefold increase in temperature drop, and reduce the energy needed to run the recirculation pump by a factor of 125.

After our short-term test, the builder hired a professional balancer to throttle back the valves to reduce the flow rates for both recirculation loops. This did not result in any electricity savings, but it did help improve the efficiency of the boiler. The builder is considering variable-speed pumps for Phase II, which would allow the space heating recirculation loop to run at a much lower flow rate when there is no demand for heat, and a higher flow rate in cold weather to simultaneously meet the space heating needs of 4 units. A smaller single-speed pump should be adequate for the DHW recirculation loop. Our building simulation model was used to analyze other approaches to the design and control of the recirculation loops. The results are presented later in the paper.

We calculated an average boiler efficiency of 65% from June 1 through August 19, 2007, instead of the rated 93%. The boilers supplied water at 190ºF (88ºC) and it returned at 180ºF (82ºC), which was higher than the condensing temperature of the combustion products. Recapturing heat by condensing the combustion products gives a condensing boiler its high efficiency (see Figure 8). Also, both boilers often ran at the same time instead of being staged. Given the small heating loads we expect in the units, and the number of baseboard heaters, the heating loop could be maintained at less than the 160ºF supply temperature typically used for hydronic baseboard systems. In addition, a lower pump flow rate should be used, and only one boiler is needed. These measures would be less expensive on a first cost basis, and would contribute to lower return temperatures, resulting in higher boiler efficiencies. If two boilers are needed from a reliability perspective, a controller should be installed to ensure that the boilers are staged appropriately.

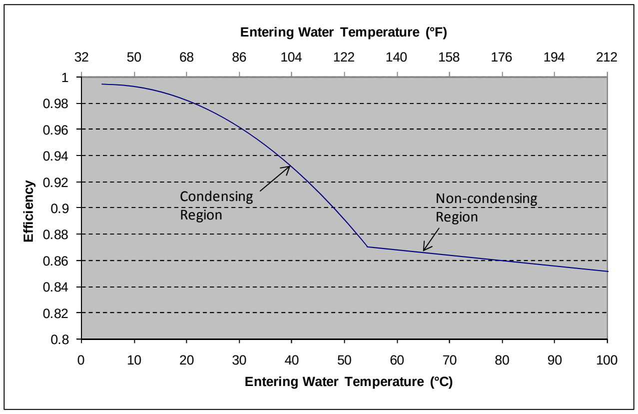


FIGURE 8. TYPICAL EFFICIENCY OF CONDENSING BOILERS AS A FUNCTION OF ENTERING WATER TEMPERATURE [4]

The efficiency of the boilers was revisited in January 2008, when the weather was cold and the homeowners were using a more typical amount of hot water. Even though there was very low boiler utilization (~15%) during this period of maximum load, the boiler efficiency averaged 86%. This efficiency was much closer to the rated AFUE of 93% than was the efficiency observed during the summer, but it still indicated that the hot water return temperature was below the condensation temperature of the combustion products.

The fraction of hot water energy met by solar energy from October 2007 until early March 2008 averaged 38%. A very significant fraction was met on most days, indicating that the solar preheat system was performing as expected given the size of the collectors and solar preheat tank.

ANNUAL SIMULATIONS

Modeling Approach

Building F-1 was modeled using TRNSYS, an advanced energy analysis software package with the flexibility to accurately model the complex space heating and hot water systems present in the building. The analysis was conducted in accordance with the 2007 BA Benchmark Definition [2]. The BA Benchmark is a house with the same basic size and shape as the prototype being analyzed, except the thermal envelope characteristics and equipment efficiencies are consistent with mid-1990s standard practice. Our approach to performing energy simulations for this project was to begin with the Benchmark and add energy efficiency packages one at a time in order to determine the incremental energy savings for each package. Once the as-built prototype was modeled, we examined some what-if scenarios to see if simple design modifications might be warranted for the next generation of houses at Burlingame as part of Phase II.

The inputs for the Building F-1 as-built model were based on a combination of design specifications and measured performance data when available, including the short-term test results and some of the long-term monitored results presented earlier in this report. A summary of the key differences between the Prototype model of Building F-1 and the corresponding Benchmark model is presented in Table 2. The table also identifies the grouping of efficiency measures into 6 packages, which were then analyzed incrementally.

Benchmark Analysis

The Benchmark building was modeled with 10 thermal zones, including the first and second floors of each unit, 3 attic zones, and the mechanical room. The floor area and conditioned volume were the same as the Prototype, with the exception of the attic, which was unconditioned in the Benchmark model.

The thermal characteristics of the Benchmark building envelope were consistent with mid-1990s construction as specified in the performance path of the 1995 Model Energy Code [5]. These included typical wall, ceiling, and slab insulation levels, air infiltration, and other envelope features. Window area was fixed at 18% to credit more efficient use of glazing. Other details can be found in the Benchmark Definition.

A 3-D finite difference model was used to model the slab for the Benchmark. The model was designed for rectangular slabs and uses symmetry to model only one-fourth of the slab. This building does not have a rectangular slab, but the added accuracy of using the detailed model was deemed more important than the exact geometry. The actual ratio of perimeter length and slab area was used to create a rectangular slab with the same ratio, which was then used in the model.

The exhaust ventilation rates for the Benchmark were based on the measured values from the actual building. Actual ventilation rates were used because we believe that energy savings for ventilation should be calculated based on how efficiently fresh air is provided to the occupants. Therefore, we do not credit under-ventilation relative to ASHRAE Standard 62.2 as an energy efficiency measure. The flow was assumed to be continuous and the power draw for the ventilation fan was set at the Benchmark value of 0.5 W/cfm (1 W/l/s) of ventilation air.

The heating system for the Benchmark included a single boiler, a pump, and hot water baseboard convectors inside the units. The thermal interactions between the piping and the mechanical room and attics were included in the model. The baseboard convectors are piped in three parallel loops inside each unit, and each unit is assumed to receive one-fourth of the total main circulation flow when that unit calls for heat.

TABLE 2. KEY DIFFERENCES BETWEEN THE PROTOTYPE AND BENCHMARK MODELS FOR BUILDING F-1

Pkg. No.	Abbr.	Feature	Benchmark	Prototype
1	Env	Total Window Area	706 ft^2 (66 m^2)	621 ft^2 (58 m^2)
1	Env	Attic Type	Vented	Unvented, insulated
1	Env	Attic Insulation U-Value	0.026 Btu/hr•ft^2•°F (0.15 W/m^2•°C) (attic floor)	0.026 Btu/hr•ft^2•°F (0.15 W/m^2•°C) (attic ceiling)
1	Env	Exterior Wall U-Value	0.076 Btu/hr•ft^2•°F (0.43 W/m^2•°C)	0.044 Btu/hr•ft^2•°F (0.25 W/m^2•°C)
1	Env	Slab Insulation R-Value	7 hr•ft^2•°F/Btu (1.2 m^2•°C/W)	10 hr•ft^2•°F/Btu (1.8 m^2•°C/W)
1	Env	ELA (to the outside)	441 in.^{2} (0.285 m^2)	105 in.^{2} (0.068 m^2)
2	IG	Hard-Wired Lighting	14% Fluorescent, 4476 kWh/yr (15.3 MBtu/yr)	90% Fluorescent, 1244 kWh/yr (4.2 MBtu/yr)
2	IG	Appliance Electric Loads	Standard dishwasher, refrigerator, clothes washer, 4031 kWh/yr (13.8 MBtu/yr)	ENERGY STAR® dishwasher, refrigerator, clothes washer, 1955 kWh/yr (6.7 MBtu/yr)
2	IG	Appliance Gas Loads	Standard clothes washer and dryer, 185 therms/yr (5420 kWh/yr)	Reduced dryer energy resulting from ENERGY STAR® clothes washer, 101 therms/yr (2960 kWh/yr)
2	IG	Appliance Hot Water	Standard dishwasher and clothes washer, 70 gal/d (265 l/day)	ENERGY STAR® dishwasher and clothes washer, 27 gal/d (102 l/day)
2	IG	MELs	Standard misc. electric loads, 10,732 kWh/yr (36.6 MBtu/yr)	ENERGY STAR® ceiling fans, otherwise standard misc. electric loads, 10,580 kWh/yr (36.1 MBtu/yr)
3	HRV	Ventilation	Continuous Exhaust	HRV
4	Boiler	Boiler Efficiency	75% AFUE	85% AFUE
4	Boiler	Boiler Capacity	190 kBtu/hr (56 kW)	130+260 kBtu/hr (38+76 kW) (2 staged boilers)
4	Boiler	Pump Power	800 W (2700 Btu/hr) continuous	800 W (2700 Btu/hr) continuous + 375 W (1280 Btu/hr) for 1st boiler + 242 W (830 Btu/hr) for 2nd boiler
5	DHW	Central Water Heater	100 gal (379 l), 0.45 EF	120 gal (454 l), heated by boilers
5	DHW	Pump for DHW Recirculation	None	150 W (510 Btu/hr)
6	Solar DHW	Solar DHW Preheat	None	92-ft^2 (8.5-m^2) collector, 120-gal (454 l) tank

The main circulation pump operates continuously; however, hot water flows to the baseboard convectors only when individual units call for space heating. The boiler controls were modeled as turning on when the return water temperature drops to 150ºF (66ºC) and turning off when the return water temperature reaches 160ºF (71ºC). This appeared to be the control strategy and deadband employed for the actual space heating system based on the monitored performance data, and we believe these control points are fairly typical for centralized heating systems in multifamily housing.

The building as-constructed does not include a cooling system. However, the Benchmark procedures only credit reductions in the cooling load or equipment efficiency as energy efficiency improvements, not energy savings resulting from the elimination of the cooling system. Therefore, the models for both the Benchmark and Prototype include 10 SEER air-conditioning units for each conditioned zone. The load analysis showed that only the two upper zones have any cooling load, and this load is less than 0.5 ton (1,760 W). Therefore the model includes 0.5 ton (1,760 W) air-conditioning units for each unit.

The Benchmark DHW system was modeled as a single centralized water heater located in the mechanical room, with a continuous recirculation loop in the unconditioned attics of Units 102 and 103. The methods outlined in the ASHRAE HVAC Applications Handbook [6] were used to determine the size of the tank and the capacity of the burner. The heat losses from the tank were added to the mechanical room, and losses from the piping were added to the attic space.

As-Built Prototype Analysis

Six incremental changes to the Benchmark model were necessary to model the Prototype:

1. The building envelope was modified based on the actual construction of the building (see Package 1 in Table 2). This influenced the amount of window area, the exterior wall construction, the slab insulation, and the building tightness. The insulated ceilings and ventilated attics in the Benchmark were changed to insulated roofs with sealed attics. The natural infiltration into the zones was calculated using the Sherman-Grimsrud method, and the leakage areas for each unit were based on the values of ELA to the outside measured during the guarded blower door tests. For the zones with multiple floors or attics, the leakage area was applied in proportion to the exterior wall area.
2. The energy use and internal gains for the installed lighting and equipment package were added to the model. The hourly schedules applied to these values were the same between the models, but the peak values of lighting, appliances, and MELs were reduced based on the installed equipment. The daily volume of hot water was also reduced for the ENERGY STAR clothes washer and dishwasher.
3. The HRVs were added to the model instead of exhaust ventilation. Measured values of flow rate, power, and heat transfer efficiency were used in the model.

4. The heating system was modified to reflect the installed system in Building F-1. The system included two boilers running in parallel, each with its own pump. The efficiency of the boilers was calculated based on the ASHRAE curve for the typical efficiency of condensing boilers [4]. This curve is shown in Figure 8. Other performance elements of the space heating system were based on measured data.
5. We added the installed tank-in-tank DHW system, which was heated indirectly by the condensing boilers via the primary space heating loop. The assumed control strategy was that water was drawn into the outer tank when the DHW temperature dropped below 115ºF (46ºC) and was stopped when the temperature reached 125ºF (52ºC).
6. The solar preheat system, which heated the mains water before it entered the DHW tank, was added to the model. The DHW makeup water came from the solar preheat tank unless that tank was too hot, in which case it was tempered with mains water to be at the hot water tank set point. The solar preheat tank had two heat exchangers, one for the collector loop and one for the DHW loop, and was filled with water as the heat transfer medium. The collector loop used a propylene glycol mixture to prevent freezing.

The predicted annual source energy for the Benchmark and Prototype buildings are summarized by end-use category in Table 3. Source energy is defined as the total energy associated with a particular fuel type, including generation, transmission, and distribution losses. For electricity, a multiplier of 3.16 was applied. For natural gas, the multiplier was 1.02.

TABLE 3. PREDICTED ANNUAL SOURCE ENERGY SAVINGS FOR THE AS-BUILT PROTOTYPE BUILDING COMPARED TO THE BA BENCHMARK

End Use	Benchmark, MBtu/yr (kWh/yr)	Prototype, MBtu/yr (kWh/yr)	Savings as Percent of End-Use Energy	Savings as Percent of Total Energy
Space Heating	635 (186,100)	351 (102,900)	45%	28%
Space Cooling	2.3 (700)	1.6 (500)	21%	0%
DHW	103 (30,200)	59 (17,400)	43%	4%
Lighting	91 (26,700)	36 (10,600)	61%	5%
Appliances + MELs	186 (54,500)	148 (43,400)	20%	4%
Total Energy Use	1017 (298,100)	596 (174,700)	41%	41%

The predicted net energy savings for Building F-1 is 41% compared to the Benchmark, based on the actual installed performance of all building systems as of December 2007. This is somewhat less than the expected energy savings of 51%. This difference is not surprising, partly because our analysis was conducted with TRNSYS, a more sophisticated and flexible simulation analysis tool that allows more accurate analysis of the complex space heating and DHW systems than the tool that was used for the original analysis. Some minor changes to the Benchmark were also made after the original analysis, which could have contributed to the difference. Also, the Benchmark does not currently provide detailed guidance for modeling multifamily buildings. This allows different interpretations of how to model centralized space heating and DHW systems, depending on the modeler. Most importantly, the final design configuration and the measured energy performance of key elements of the building, including air leakage, HRV efficiency, solar collector efficiency, boiler efficiency, pump power, and actual control settings, were used in our model. These data were not available at the time of the preliminary analysis.

Significant energy savings are predicted in all end-use categories except cooling, which is a very small component of total energy use, and therefore was not targeted to any great extent in the design of the building. The cooling energy savings was a consequence of reductions in the cooling load, primarily as a result of improved windows and reduced internal gains from lighting.

Table 4 summarizes the source energy savings attributable to the six packages of efficiency measures listed in Table 2. The envelope improvements ($1316/yr), the lighting and appliance measures ($737/yr), and the HRVs ($145/yr) appear to be the most important energy-saving measures. The space heating and DHW measures, including the solar preheat system, do not produce a large benefit in terms of whole-house source energy savings or energy cost savings. However, the sequencing and grouping of measures in the analysis can have a very large effect on incremental energy savings. If solar hot water were added to the Benchmark as the first package, the energy savings would be much larger. Our choice of sequencing is somewhat subjective, but is generally based on which packages save the most incremental energy from one step to the next.

TABLE 4. PREDICTED INCREMENTAL ANNUAL SOURCE ENERGY SAVINGS FOR EACH PACKAGE OF ENERGY EFFICIENCY MEASURES

	Site Energy		Source Energy		Energy Cost*	
Package**	Electric, kWh (MBtu)	Nat Gas, kWh (MBtu)	kWh (MBtu)	% Saved	$/yr	% Saved
Benchmark	31,306 (107)	195,200 (666)	298,000 (1,017)		$6461	
(1) Env	31,242 (107)	118,400 (404)	219,500 (749)	26%	$5144	20%
(2) IG	23,621 (81)	119,900 (409)	196,900 (672)	34%	$4407	32%
(3) HRV	24,676 (84)	105,200 (359)	185,300 (632)	38%	$4263	34%
(4) Boiler	25,067 (86)	99,000 (338)	180,200 (615)	40%	$4197	35%
(5) DHW	25,304 (86)	96,400 (329)	178,300 (608)	40%	$4175	35%
(6) Solar DHW (Proto)	25,418 (87)	92,500 (316)	174,700 (596)	41%	$4122	36%

* Calculated using approximate national average electricity cost = $0.10/kWh ($30/MBtu) and gas cost = $5/MBtu ($0.017/kWh).
** The packages are numbered in accordance with Table 2, and are cumulative. The last row includes all 6 packages and therefore represents the Prototype.

What-If Scenarios

A series of "what if" scenarios was run to examine the sensitivity of energy savings to various design characteristics, and to quantify potential energy savings for design changes that might be considered for Phase II.

A space heating approach with individual high-efficiency furnaces in each unit was modeled. The DHW was still provided with a centralized tank and distribution system. The furnace efficiencies were 92% AFUE. The furnaces, including the air handlers, were predicted to use a total of 214 MBtu (62,700 kWh). The equivalent central boiler system would be expected to use 265 MBtu (77,600 kWh) for the boiler and pumps combined. This suggests that a 19% reduction in energy consumption for space heating (6% whole house) could be achieved by using small, individual condensing furnaces.

We examined the potential energy savings if the primary pump operates only when there is a call for heating. Electricity consumption would decrease by about 9% and gas consumption would increase by about 1%, resulting in a net decrease in total source energy use of about 2%.

The model was then run with a single noncondensing boiler (capacity = 191 MBtu/hr [202,000 kJ/hr], boiler efficiency = 0.75, combustion efficiency = 0.82) instead of the installed pair of condensing boilers to determine whether the efficiency gains for the two condensing boilers was significant. The analysis suggested that the condensing boilers provide significant whole-house source energy savings (~5%), even though they do not operate at their optimal efficiency levels.

We analyzed DHW system options without the building model to compare different options without the complication of interactions between the space heating system and the DHW system. The matrix of DHW systems included a centralized DHW tank with a recirculation loop (as-built, except using aquastat control instead of continuous operation), individual DHW tanks in each unit, tankless DHW systems in each unit, and each of these three systems with a solar preheat system. When there was a DHW system in each unit, individual solar preheat systems and a single central solar preheat system were both analyzed. To simplify the model and shorten the run time, the tanks and distribution systems were assumed to be entirely within conditioned space at a constant 68ºF (20°C). The results of the analysis are summarized in Table 5.

The current centralized water heating system seems to be at least as efficient as individual tanks inside the units. The energy for the additional pumps and the recirculation loop needed for a centralized DHW system is small relative to the energy savings for more efficient condensing boilers compared to standard efficiency tank systems. The most efficient option is predicted to be individual high-efficiency tankless water heaters combined with a centralized solar preheat tank. We believe this is the current plan for Phase II at Burlingame. However, there are potential temperature stability issues when gas tankless water heaters are used in conjunction with solar preheating, because the minimum firing rate may cause the hot water supply temperature to exceed the set point when the temperature into the tankless heater is within about 20º–30ºF (11º–17ºC) of the set point, depending on the flow rate and the control logic. These issues must be addressed for the tankless system with solar preheat to meet homeowner expectations.

TABLE 5. SITE AND SOURCE ENERGY USE FOR ALTERNATIVE WATER HEATING DESIGNS

System	Natural Gas, MBtu	Electricity, kWh	Source Energy, MBtu (kWh)
Central	51	70	52 (15,200)
Central + Solar	42	119	44 (12,900)
Individual Tanks	51	0	52 (15,200)
Individual Tanks + Solar	45	53	46 (13,500)
Individual Tanks + Central Solar	43	48	44 (12,900)
Individual Tankless	35	0	36 (10,600)
Individual Tankless + Solar	30	53	31 (9100)
Individual Tankless + Central Solar	28	48	29 (8500)

When the air conditioners were removed from the model, only two zones exceeded the cooling set point for any length of time. As a result of thermal stratification, the upper floor of Unit 101 exceeded the set point of 76ºF (24ºC) for 751 hours (9% of the time over the course of a year), and the upper floor of Unit 104 exceeded the set point for 567 hours (6% of the time). The difference between units was most likely caused by a greater fraction of the window area on the south and west sides of Unit 101. The analysis assumed that windows were not opened to mitigate high temperatures in the models, so the number of hours with uncomfortably warm interior temperatures will likely be much smaller in the actual houses.

The HRVs were modeled with constant operation at low speed and with an intermittent strategy where the HRVs operated at low speed for 10 minutes and then were off for 20 minutes. The measured flow rates and powers were used. The main purpose of this analysis was to examine how much the heating energy would decrease if the HRVs were to operate at a 33% duty cycle instead of at 100%. The whole-house source energy savings would be significant (~3.5%) if intermittent operation were used, but these savings would have to be traded off against potential negative effects on indoor air quality as a result of ventilating at a lower rate than recommended by the ASHRAE residential ventilation standard [3]. The authors generally recommend compliance with the ASHRAE standard, despite the potential for significant energy savings if the duty cycle is reduced. Because the HRVs appear to meet the ASHRAE standard in each unit when operating at their lowest flow rate at 100% duty cycle, our recommendation to the homeowners was to use the "min" setting for their HRVs.

The model was next run with continuous exhaust fans sized to remove the same amount of air as the current HRVs. An exhaust fan with the same nominal ventilation rate as a balanced HRV system will result in a lower net ventilation rate because of interactive effects with natural infiltration. The model suggested that the decrease in fan energy would be overwhelmed by the increase in space heating energy if exhaust fans were used, resulting in a net source energy penalty of about 8%.

Model Validation

Once we had a year of measured data, the actual weather conditions, number of occupants, hot water use, thermostat settings, and electricity use for lighting, appliances, and MELs were used to confirm the accuracy of the model, including assumed operating conditions such as the use of windows, blinds, and spot ventilation. In Figure 9, the predicted natural gas energy for the two boilers is compared to the measured values for the period from March 2008 through February 2009. The measured gas usage was less than predicted for most months. We are currently investigating possible explanations for this difference, such as lower setpoints for the tanks and circulation loops, warmer mechanical room temperatures, smaller ground losses, or non-uniform temperatures inside the units. Details of the model itself are also being re-examined.

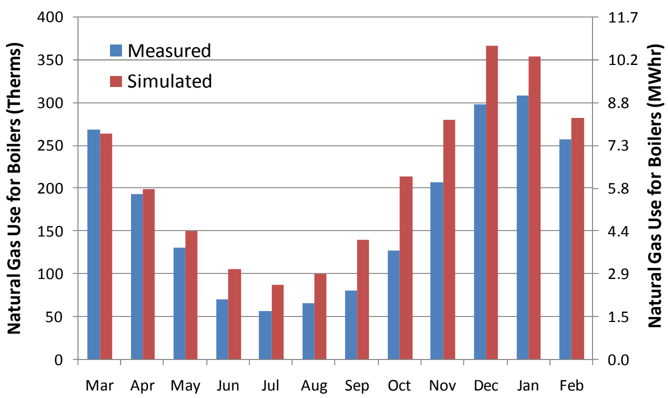


FIGURE 9. MEASURED NATURAL GAS USE FOR SPACE HEAT AND DHW COMPARED TO THE TRNSYS SIMULATION

The predicted whole-building electricity use for the same time period, including the mechanical room, is compared to the measured electricity use in Figure 10. Actual electricity use was slightly higher than predicted. The primary reason for the difference is that the energy used by the 6 pumps in the mechanical room was higher than expected. The higher electricity use in the mechanical room could also partly explain the lower gas energy use. We are currently investigating the cause of the difference in pump energy, and inputs to the model will be adjusted if necessary.

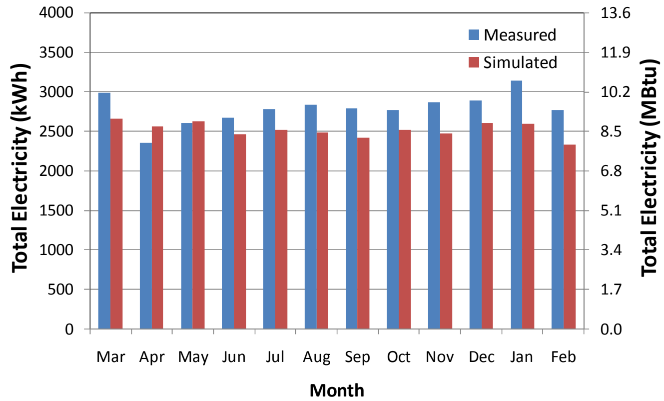


FIGURE 10. MEASURED WHOLE-HOUSE ELECTRICITY USE COMPARED TO THE TRNSYS SIMULATION

CONCLUSIONS

Overall, the performance of the building was very good, achieving 41% energy savings compared to the BA Benchmark, which is a reference house built in accordance with typical 1990s construction. This level of energy savings is especially difficult to achieve in multifamily housing, because space conditioning is a much smaller fraction of energy use than it is in single-family detached housing (primarily because of smaller thermal envelope areas), and end-uses other than space conditioning tend to be more difficult to reduce. For this project, it was necessary to target water heating, ventilation, lighting, appliances, and MELs in addition to space heating.

Most of the energy efficiency improvements performed as intended, and sometimes performed better. The HRV heat transfer efficiency and outside air distribution met design expectations. Outside air infiltration was significantly better than the target values. The solar hot water system appeared to work as expected, contributing about 38% of the energy required for DHW once the builder corrected a few initial problems.

However, a number of important lessons were learned from the evaluation of the test house, and the authors made several recommendations for improving the performance of designs for Phase II.

- Guarded blower door tests showed that approximately one-third of the measured leakage at 50 Pa (0.0073 psi) was to adjacent units for Units 101 and 103, and two-thirds of the leakage area for Unit 102 was to adjacent units. The leakage area to adjacent units for Unit 104 was only about one-sixth of the total. This indicates that some of the attached walls were relatively leaky. We would expect the air leakage fraction between units to be much less under normal conditions, because the pressure differences that drive airflow from one unit to another should be much smaller than those between the inside and outside. However, the builder should try to reduce leakage in the attached walls to minimize the exchange of pollutants between units.

- The continuous space heating recirculation loop lost energy at a rate of about 5300 Btu/hr (1552 W) during the summer with little or no space heating load on the units. The continuous DHW recirculation loop did not appear to lose a significant amount of thermal energy. The total pump energy associated with the two recirculation loops was about 3170 Btu/hr (930 W). Both pumps were significantly larger than necessary, and the space heating pump could be operated only during demands for heat, which would reduce the electricity drawn in the mechanical room.
- The estimated combined efficiency of the boilers appeared to be about 65% on average during the summer while the units were unoccupied. The efficiency increased to ~86% once there were significant DHW and space heating loads, but was still lower than the rated efficiency of 93%, because the return temperature remained well above the condensing temperature of the combustion products.
- Additional source energy savings (approximately 19% of space heating energy and 6% of whole-house energy) can be achieved by using high-efficiency condensing furnaces in each unit instead of a centralized space heating system, which is hampered by high pump energy, significant thermal losses from the continuous recirculation loop, and the inability to achieve the rated boiler efficiency because of high return temperatures.
- Based on annual simulations using TRNSYS, the solar water heating system is only expected to contribute about $53 of energy cost savings for the entire building. It is unlikely that this system will be cost-effective as it is currently designed. However, the energy savings could be approximately tripled if individual tankless water heaters are deployed in each unit, while continuing to use a centralized solar pre-heat tank.

As a result of the lessons learned during this project, a number of follow-up steps have been identified:

- Work closely with the City of Aspen and the builder selected for Phase II of the project to ensure that the energy features that worked well are continued, and the features that did not perform well are corrected or abandoned.
- Improve the commissioning procedures for the remainder of Phase I as well as Phase II.
- Continue to refine the TRNSYS model of Building F-1 based on measured data, and identify additional cost-effective improvements to the energy efficiency, comfort, and reliability of the design for Phase II.

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