Measured Performance of a Low Temperature Air Source Heat Pump

R.K. Johnson
Consortium for Advanced Residential Buildings

September 2013

Building America SM
U.S. Department of Energy

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

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

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

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

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

Measured Performance of a Low Temperature Air Source Heat Pump

Prepared for:

The National Renewable Energy Laboratory

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

Office of Energy Efficiency and Renewable Energy

15013 Denver West Parkway

Golden, CO 80401

NREL Contract No. DE-AC36-08GO28308

Prepared by:

R.K. Johnson

Johnson Research LLC

1240 W. Calle Antigua

Pueblo West, CO 81007

of the

Consortium for Advanced Residential Buildings (CARB)

NREL Technical Monitor: Cheryn Metzger

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

September 2013

[This page left blank]

Contents

List of Figures

Unless otherwise noted, all figures were created by CARB.

List of Tables

Unless otherwise noted, all tables were created by CARB.

Definitions

AHU	Air handling unit
ASHRAE	American Society of Heating, Refrigerating and Air-conditioning Engineers
CT	Current transformer
GSHP	Ground source heat pump
HVAC	Heating, ventilation and air conditioning
Btu	British thermal unit
CFM	Cubic feet per minute
COP	Coefficient of performance
LTHP	Low temperature heat pump
ODU	Outdoor unit
RTD	Resistive temperature device
SCOP	Seasonal coefficient of performance

Executive Summary

A 4-ton low temperature heat pump (LTHP) manufactured by Hallowell International was installed in a residence near New Haven, Connecticut in April 2009 and monitored over two winters of operation. After some significant service issues were addressed, the heat pump operated as designed.

The LTHP was claimed to operate down to –30°F, and to supply essentially all needed heat to a home via its compressors with very minimal use of a supplemental heat source. The manufacturer claimed a steady-state coefficient of performance (COP) (efficiency) > 2.0 at –10°F. Standard air source heat pumps more rapidly lose heating capacity as the outdoor temperature drops below about +30°F and rely increasingly heavily on supplemental heat (usually an electric resistance heater). Minimizing the use of electric resistance heat not only reduces the homeowner's heating bill; it also reduces the heat pump's contribution to the local electric utility's winter peak load.

The primary objective of this study was to demonstrate the heating operation of an LTHP, as built by Hallowell International in Bangor, Maine. The tested model was named the Acadia 048 by the manufacturer; the 048 refers to the nominal capacity, 48,000 Btu/h.

The manufacturer has gone out of business, due apparently to economic issues. Its website is no longer available. As this report will demonstrate, however, its technology lived up to its promise, though not without some reliability and service issues along the way. This report should be considered a review of the dual compressor "boosted heat pump" technology, as implemented by Hallowell.

Extensive metering equipment was installed in September 2009 to monitor the performance of the heat pump over the 2009–2010 and 2010–2011 winters. Data were collected at 1-min intervals for nearly 20 months, through April 2011. Data were collected for system and subsystem power, various temperatures, airflow, and system status. The data enabled calculation of heat production (Btu output) and COP by minute, day, and full heating season.

System Performance

The LTHP system operates with four increasing levels of capacity (heat output) as the outdoor temperature drops. The system was shown to select capacity correctly, supplying the appropriate amount of heat to the house across the full range of outdoor temperatures.

The system's seasonal COP (SCOP) over two entire winters was calculated, based on measured data, to be 3.29 over the first winter and 2.68 over the second winter. A second seasonal efficiency calculation by a different method yielded a SCOP of 2.78 for the first winter and 2.83 for the second winter. This second seasonal efficiency calculation was determined by comparing measured heat pump energy use to the in situ energy use with resistance heat alone. This method calculates the ratio of the slopes of the daily energy use load lines.

The data showed the following heating energy, cost, efficiency and temperature environment during the two winters of testing. The local electricity rate was $0.1687/kWh.

Table 1. Summary of Heating Energy, Cost, Degree-Days, and SCOP

	Heating Season (kWh)	Heating Cost ($)	Degree-Days (65°F)	SCOP (2 Methods)
Winter 1 (2009–2010)	5,306 kWh	$895	5,326	3.22/2.78
Winter 2 (2010–2011)	7,776 kWh	$1,312	5,593	2.68/2.83

Economics

A rough estimate of the cost to heat the same home, if oil heat had been used, indicated that the owners would have spent about $1,600 (at $2.70/gal) under the same first winter weather conditions, versus $1,312 for the heat pump. Fuel oil is the most common heat source in this geographic area; this home had previously used an oil boiler. Table 2 compares estimated costs of heating the home with various fuels and heating systems at then current local fuel costs.

Table 2. Estimate of Heating Energy Cost for Various Heating Systems and Fuels in a Home That Requires 60 Million Btu per Winter

	Fuel Cost[a]	Fuel Cost per Million Btu Delivered	Annual Heating Cost to Deliver 60 Million Btu
Acadia Heat Pump, SCOP = 2.78[b]	$0.1687/kWh	$17.78	$1,067
Central, Ducted Electric Heat, 100% Efficiency	$0.1687/kWh	$49.43	$2,966
Baseboard Electric Heat, No Duct Losses	$0.1687/kWh	$42.01	$2,521
Premium 2-Speed Heat Pump,[c] SCOP = 1.78	$0.1687/kWh	$27.77	$1,666
Natural Gas, 70% AFUE[d]	$1.41/100 ft^3	$19.54	$1,172 + $135 = $1,307
Natural Gas, 90% AFUE[d]	$1.41/100 ft^3	$15.20	$912 + $135 = $1,047
#2 Fuel Oil, 70% AFUE[d]	$2.70/gal	$27.85	$1,671 + $135 = $1,806
Ground Source Heat Pump, SCOP = 4.0	$0.1687/kWh	$12.36	$741

[a] Electricity cost = $0.1687/kWh from www.cl-p.com/rates/averagebill.aspx, Natural gas cost = $1.41/ccf from Yankee Gas data. Fuel oil cost = $2.70/gal from the Energy Information Administration, past winter average for Connecticut.

[b] LTHP operating costs have been adjusted to eliminate the negative impact of operating the heat pump intentionally on strictly electric resistance heat.

[c] The "premium" heat pump was located in central New Hampshire and tested for National Rural Electric Cooperative Association-Cooperative Research Network.

[d] Estimated air handler fan operating cost of $135 (average of this system, both winters) has been added to the annual operating cost of fossil-fired systems. This system used an electronically commutated fan motor; a constant speed fan could use more energy.

1 Project Objectives

Homeowners in more densely populated areas generally have the option to heat with natural gas, but in less densely populated areas the heating fuel choices generally are electricity, propane, or #2 fuel oil. Oil prices have been volatile over the past few years, reaching more than $4/gal at times. Propane normally is significantly more expensive than natural gas, since it has to be manufactured and delivered to the site.

Homeowners looking for alternative heating sources who don't have access to natural gas often look to electric heating alternatives. Electric baseboards are commonly installed, particularly in lower cost homes, but are not economically efficient.

Heat pump technology, whether ground source or air source, is commonly seen as the alternative. The ground source heat pump (GSHP) has a significant first cost associated with installation of the ground loop. Air source heat pumps are sometimes used in an effort to avoid the high operating cost of electric resistance heating at a more moderate installed cost than GSHPs. Air source heat pumps have never reached the popularity in northern climates that they have achieved in the South. This is because they have been considered expensive to operate during the colder months; they rely heavily on supplemental electric resistance heating during cold weather.

This evaluation investigates the performance of an emerging technology, namely a dual compressor "boosted air-source heat pump" that claims to minimize the use of supplemental electric resistance heating, even at low outdoor temperatures. Specifically, the primary objective of this research is to demonstrate the heating operation of a low temperature heat pump (LTHP) as manufactured by Hallowell International and referred to as the Acadia model.

The demonstration includes these specific goals:

1. Measure the LTHP's net efficiency over two heating seasons in the local Connecticut climate under real-world operating conditions. This efficiency will be referred to as its seasonal coefficient of performance (SCOP). This measurement assists future users in the same climate in estimating their own operating cost with a similar heat pump. The SCOP can be roughly converted to the more commonly referenced heating season performance factor by dividing the SCOP by 0.293.
2. Determine if the heat pump meets its manufacturer's specifications in terms of control design (i.e., is the heat pump control selecting the proper operating stage for specific ambient conditions), heat output, and operational efficiency across the full range of winter ambient temperatures.
3. Provide feedback to the heat pump installer and the manufacturer in case any anomalous operation is seen and service is needed.

Air conditioning operation was also monitored though it is not the subject of this report.

2 Test Site Description

The test site is a two-story colonial style house in south-central Connecticut. The house has an attached garage and a full, unheated basement. The house was built in 1962. There are 2,010 ft^2 of heated floor area. Figure 1 shows the house. There is a pond at the bottom end of the backyard.

Figure 1. The test site. The heat pump is located at the back of the house.

The Acadia heat pump was installed in April 2009. The outdoor unit is located at the back of the house; the air handling unit (AHU) is in the attic.

The Acadia replaced an oil-fired boiler heating system. The boiler supplied hot water to baseboards on the second floor. Heat was supplied to the first floor from a hydro air heating coil and ducts. That system thus had two separate heating zones.

The heat pump was likewise installed in a two-zone configuration, with one AHU supplying both first floor and second floor zones using zone dampers in the supply trunk.

The previous high-velocity heating equipment was located in the unfinished basement. The new AHU was placed in the attic. The system was moved to allow for more space for maintenance and was more conducive for installing the new standard size ductwork to all rooms. To minimize the penalty of locating the heating, ventilation, and air-conditioning (HVAC) system in the attic, polyurethane spray foam was applied to the underside of the roof deck to convert this home to an unvented ("hot roof") attic. This retrofit provided significant air sealing to the building envelope and brought the HVAC within the conditioned space.

3 Basics of the Low Temperature Heat Pump

3.1 Low Temperature Heat Pump Components

The tested Acadia heat pump is typical of residential air source heat pumps (as opposed to GSHPs) in that it consists of two sections: an indoor unit (the AHU) and an outdoor unit (ODU—containing the two compressors). This is commonly referred to as a split system. It was manufactured by Hallowell International in Bangor, Maine. As a heat pump, the system supplies air conditioning as well as heating. Cooling operation (sensible cooling only) was monitored but is not the focus of this report.

The Acadia was manufactured in three nominal capacities: 24, 36, and 48 kBtu/h. The tested unit was the largest, 48 kBtu.

Figure 2 shows the ODU. This is the part of the heat pump that extracts heat energy from the outdoor air. It contains a wraparound coil, two compressors, a fan, a proprietary heat exchanger, and the system's controls. The legs are intended to keep the ODU above snow and to allow space for defrost condensate to drain away. The legs can be as much as 2 ft longer in snowier climates.

Figure 2 Acadia heat pump ODU

Figure 3 Acadia heat pump indoor unit

Figure 3 shows the AHU. The AHU receives hot refrigerant gas from the ODU and delivers its heat via ducts into the living spaces in the house. It contains a blower (fan), a heating coil or A-coil (named for its shape) through which hot refrigerant gas flows, and electric heater elements typically of 10–24 kW. This system uses two electric resistance coils of 5 kW and 7.5 kW for a total of 12.5 kW.

The AHU is located in the home's attic. It is suspended from roof rafters to sound-isolate it from the ceiling below. In Figure 3, the return duct enters the right end of the AHU and the supply exits the left end of the unit through several round, flexible ducts. Ducts are wrapped in foil-faced fiberglass insulation. The supply duct branches to two trunks; each has a zone damper. One zone serves the first floor and the other serves the second floor. Each zone is controlled by a thermostat.

A data recorder enclosure sits atop the AHU in the white box. The horizontal white tube carries condensate from air conditioning dehumidification to the outdoors.

3.2 How It Works—Stages and Modes of Operation

A typical gas furnace has one capacity: it is either on or off at its rated value, often 100,000 Btu/h. (There are also available two-stage and variable-capacity furnaces that can more closely match a home's varying heat requirements.) The basic gas furnace produces the same amount of heat whenever it runs. A building, in contrast, needs a little bit of heat at moderately cool outdoor temperatures and needs progressively more heat as the weather gets colder. The Acadia heat pump tries to just produce somewhat more heat than the building needs at any given outdoor temperature.

Figure 4 illustrates the LTHP's heat output according to the manufacturer's published laboratory test data. The figure is complex and requires some explanation.

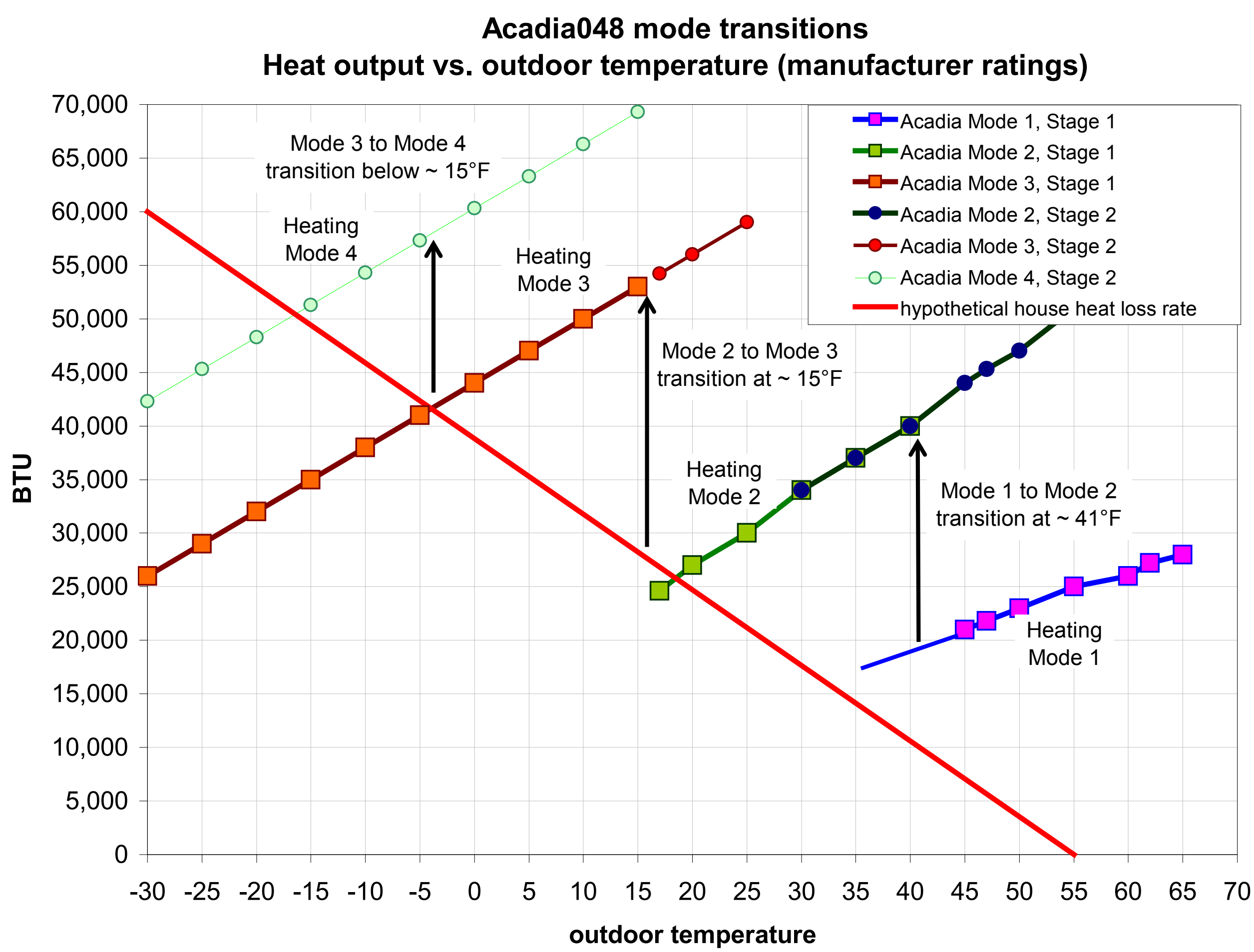

Figure 4. Factory data showing heat output in each operating mode, with transitions between modes

As with any heat pump, capacity (heat output) decreases as the "source" (outdoor air) temperature decreases. The LTHP, in contrast, increases its heat output in stages as the outdoor temperature drops. The square data points in blue/pink, green, and orange show the heat pump's basic Btu output capacities across its full range of operating temperatures. They illustrate how this heat pump "steps" to progressively higher capacity as the weather gets colder.

This is unlike a single-stage heat pump that would produce its "rated" output at one temperature, generally 47°F, with its output always less at lower temperatures. An ordinary, single-stage heat pump might have an output line similar to the Mode 2 line in this figure but would continue to lose capacity to about 0°F, where its output would be negligible and its compressor would be shut off. This ordinary heat pump would typically use increasing amounts of supplemental heat, starting around 30°F (commonly referred to as the balance point).

The Acadia heat pump works as follows. The heat pump uses two compressors, an internal heat exchanger, and a bank of electric heaters to deliver four stages of heat output that are called "Modes" by the manufacturer. The smaller, "primary" compressor has two cylinders and can use one or both cylinders for low (Mode 1) and high (Mode 2) outputs. The second compressor is called the "booster" compressor and is larger than the primary compressor. The booster compressor can operate in addition to the primary compressor, increasing the LTHP's total heat output (Mode 3) even more.

To summarize:

- Mode 1 uses one operating cylinder in the primary compressor.
- Mode 2 uses both cylinders of the primary compressor.
- Mode 3 uses both the primary compressor and the booster compressor. Mode 3 also incorporates a proprietary "economizer" heat exchanger arrangement that feeds back refrigerant into the system, further boosting its capacity.
- Mode 4 adds 1 or 2 stages of electric heat to the Mode 3 compressors' output.

A red line has been drawn on Figure 4 to represent the amount of heat a hypothetical building might need at any temperature between +55°F and –30°F.

- In mild weather the heat pump runs in Mode 1. Figure 4 indicates that the heat pump will run in Mode 1 until the outdoor temperature drops to about 40°–45°F, when the house will need more heat than can be provided by Mode 1.
- At that temperature the heat pump will switch to Mode 2.
- Around 15°–20°F, as the house needs yet more heat, the system jumps to Mode 3.
- If Mode 3 is not satisfying the home's need for heat as the outdoor temperature drops further below 15°F, the system might add 5–12 kW of electric heat to the output of the compressors.

Further, at any outdoor temperature below 41°F, there is a choice of a basic mode and the next higher mode. The building's thermostat always starts the heat pump in the lowest mode that is appropriate for the outdoor temperature. If the space temperature at the thermostat continues to drop, the thermostat will close its “Stage 2” contact and call for a higher mode.

3.3 Defrosting

All air source heat pumps, including the LTHP, accumulate frost on their outdoor coils under certain conditions. The outdoor coil is always colder than the outdoor air and when the coil temperature is below the ambient dew point, moisture in the air condenses on the coil. When the coil temperature is below freezing, this condensate turns to ice on the coil. The accumulated ice reduces heat transfer from the air to the refrigerant in the coil. If not removed the ice would eventually block airflow through the coil.

To maintain efficiency, the heat pump must periodically get rid of that accumulated ice by “defrosting.” The LTHP measures its own performance and recognizes when ice has accumulated. It sheds the ice by temporarily turning into an air conditioner. That is, it removes heat from the indoors and pushes that heat into the outdoor coil. This melts the ice off the coil.

In order for air coming from the registers not to feel cold during a defrost cycle, the LTHP simultaneously turns on a supplemental electric heating element (typically 5 kW) in the AHU.

The defrosting process unavoidably decreases any heat pump’s efficiency, so the defrost cycle and its use of electric heat are kept as brief as possible. Figure 10 in Section 6 will show an example of daily data that includes several defrost cycles.

There were issues with inadequate or improper defrosting that will be discussed in Section 6.

4 Test Methodology

Detailed data were collected on the operation of the heat pump at 1-min intervals through the entire 19 months (two full winter periods) of testing.

Measurements were made and recorded using a Campbell Scientific CR1000 data recorder. Data were transmitted daily via cell modem to the monitoring contractor's office for analysis.

Measurements each minute included:

- Outdoor temperature
- Indoor temperature
- Supply and return air temperatures
- Airflow through the AHU (conditioned air delivered to the house)
- Power consumed by the outdoor unit (compressors, fan, controls), by the AHU fan, and by the electric heater in the AHU
- System status (defrosting; air conditioning).

Data were retrieved on a dedicated data retrieval computer. Data were backed up on an external drive, on a second computer, and on an external data storage site. The data logger, though downloaded daily, has storage capacity for more than a week's data. Details on the logger and each measurement sensor are found in the Appendix.

All measurement points were selected for their contributions to overall system evaluation. The raw data were plotted daily to show that the heat pump was responding properly to calls for heat and selecting appropriate operating modes, as shown in Section 6. The most significant measurements included heat output (Btu output) and efficiency (COP). These numbers helped determine if the heat pump was meeting the manufacturer's ratings.

Btu heat output is calculated from temperature and flow measurements as follows:

$$Q_{OUT} = \dot{m}\, C_p\, \Delta T$$

where:

Q_{OUT} = rate of heat delivery out of the AHU, Btu/h
$\dot{m}$ = mass flow of air through AHU, lb/h = CFM *x* 1.08
C_p = specific heat of air, corrected for altitude, Btu/lb-°F
$\triangle T$ = temperature rise across the AHU (supply temperature minus return temperature), °F

COP is also calculated for each operating minute as follows:

$$COP = \frac{Q_{OUT}}{Q_{IN}} = \frac{CFM * \Delta T * 1.08}{(kWh_{ODU} + kWh_{AHU}) * 3{,}413}$$

where :

COP = efficiency
Q_{OUT} = heat delivered from the AHU, Btu/h
Q_{IN} = energy delivered into the system, Btu
CFM = air flow through the AHU,
$\triangle T$ = temperature rise across the AHU (supply minus return temperature)
1.08 = factor accounting for air density and specific heat, and minutes/hour (adjusted for elevation)
kWh_{ODU} = power drawn by the compressor(s), outdoor fan, and controls
kWh_{AHU} = power drawn by the AHU fan and the supplemental heat, if used
3,413 = converts kWh to Btu (1 kWh = 3,413 Btu).

These calculations are done each minute that the equipment is operating. Sample plots of Btu output and COP per minute versus outdoor temperature will be shown in Section 6.

5 Results

5.1 Full Heating Season Performance Summary

It will be evident in the following examples that data collection at 1-min intervals can reveal operating characteristics that 15-min data or bill analysis would never show.

5.1.1 Winter Weather Environment

One is normally interested in how a heat pump performs in worst-case weather conditions. ASHRAE (2009) lists lowest expected temperatures at several locations in Connecticut, shown in Table 3. The test site is actually closer to New Haven, Connecticut, though the Handbook does not have a New Haven listing. The test site is about 10 miles north of the Long Island Sound. The Table also shows the site's measured degree-days over the tested heating seasons and the site's lowest measured outdoor temperature during each winter. In general, one can consider the test site to have had similar weather to the nearby weather stations.

Table 3. Test Site Conditions and ASHRAE Design Conditions for Various Connecticut Sites

	Heating Degree Days Base 65°F	99.6%[a]	99%[b]
Bridgeport Sikorsky Memorial	5,290 (ASHRAE)	10.7°F	15.4°F
Hartford Bradley International	5,992 (ASHRAE)	3.1°F	8.0°F
Oxford	6,408 (ASHRAE)	3.1°F	8.7°F
New Haven	5,432 (2006–2009 average) (Degreedays.net 2013)	7°F (97.5% design temperature)	
Test Site, Year 1	5,326 (measured, 9/18/09[c] to 4/30/10)	4.3°F, lowest measured site temperature, January 10 and 30, 2010	
Test Site, Year 2	5,593 (measured, 10/9/10[c] to 4/30/11)	–2.8°F, lowest measured site temperature, January 24, 2011	

[a] Temperature exceeded for 0.4% of the hours (35 h) in an average year.
[b] Temperature exceeded for 1% of the hours (87.6 h) in an average year.
[c] First day of heating.

Figure 5 shows the daily maximum, minimum, and average temperatures over the entire test period. There were two days in the first winter, January 10 and 30, 2010, when the outdoor temperature dropped to 4.3°F. The lowest daily average temperature was 13°F on January 30, 2010.

The second winter experienced several days that were colder than during the first winter, with a low of –2.8°F on January 24, 2011. The average temperature over that day was 6.9°F.

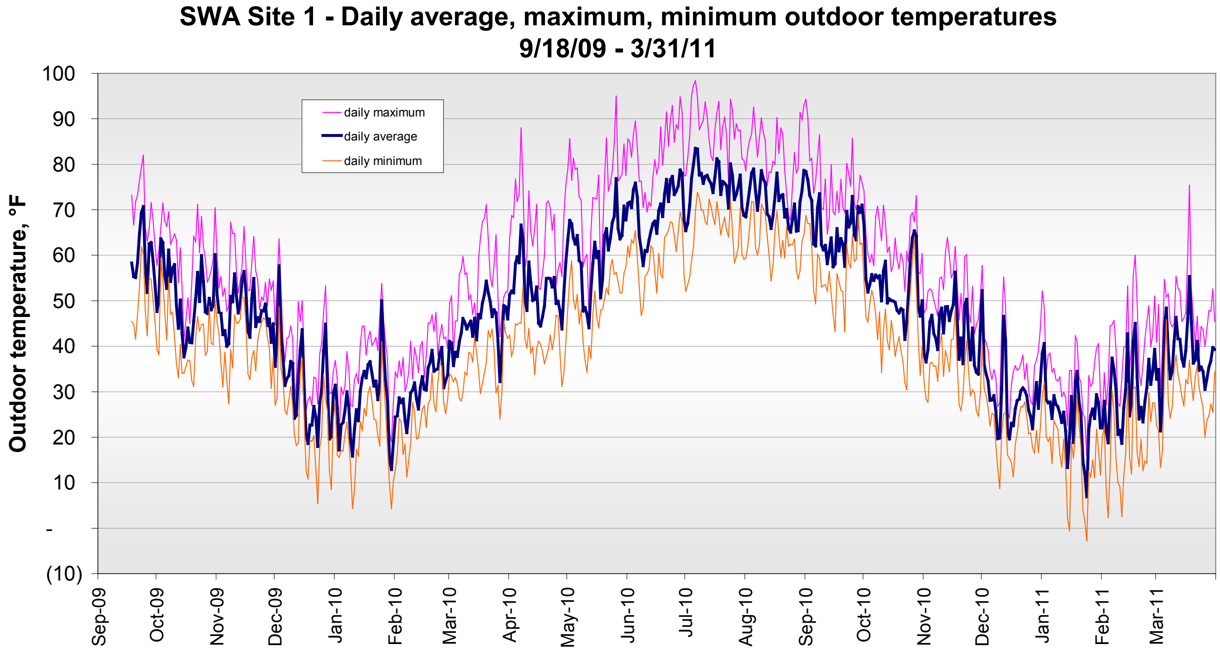

Figure 5. Daily outdoor temperatures at the test site: average, maximum, minimum

5.1.2 Heat Pump Energy Use

Figure 6 illustrates the amount of electric energy used by the heat pump during each day of the test.

- The blue portion of each bar (at the bottom) shows energy used by the AHU fan. The fan normally consumes less than 10% of the system's total use.
- The dark red bars show energy used by the outdoor unit—the compressors, fan, and master control.
- The orange bars show energy used by the supplemental electric heater in the AHU, both during defrosting and during Mode 4 operation. Under normal winter conditions when there is regular defrosting, the electric heat represents typically about 1%–3% of the system's total use.

It is important to note that on six occasions shown in Figure 6 the electric heater was run intentionally as part of the test procedure. The electric heat establishes a baseline heat loss rate of the house. The data are also used in a heat balance calculation to verify the calibration of the air flow measurement. This electric heating did not indicate any problem with the heat pump.

On most days with normal operation, the electric heat was used only for defrosting the outdoor coil. Mode 4, which adds electric heat to the compressors' output, was used for a total of 52 min during the entire first winter, all on January 10 and 11 and February 7, 2010. All Mode 4 operation was during periods when the outdoor temperature was below 12°F.

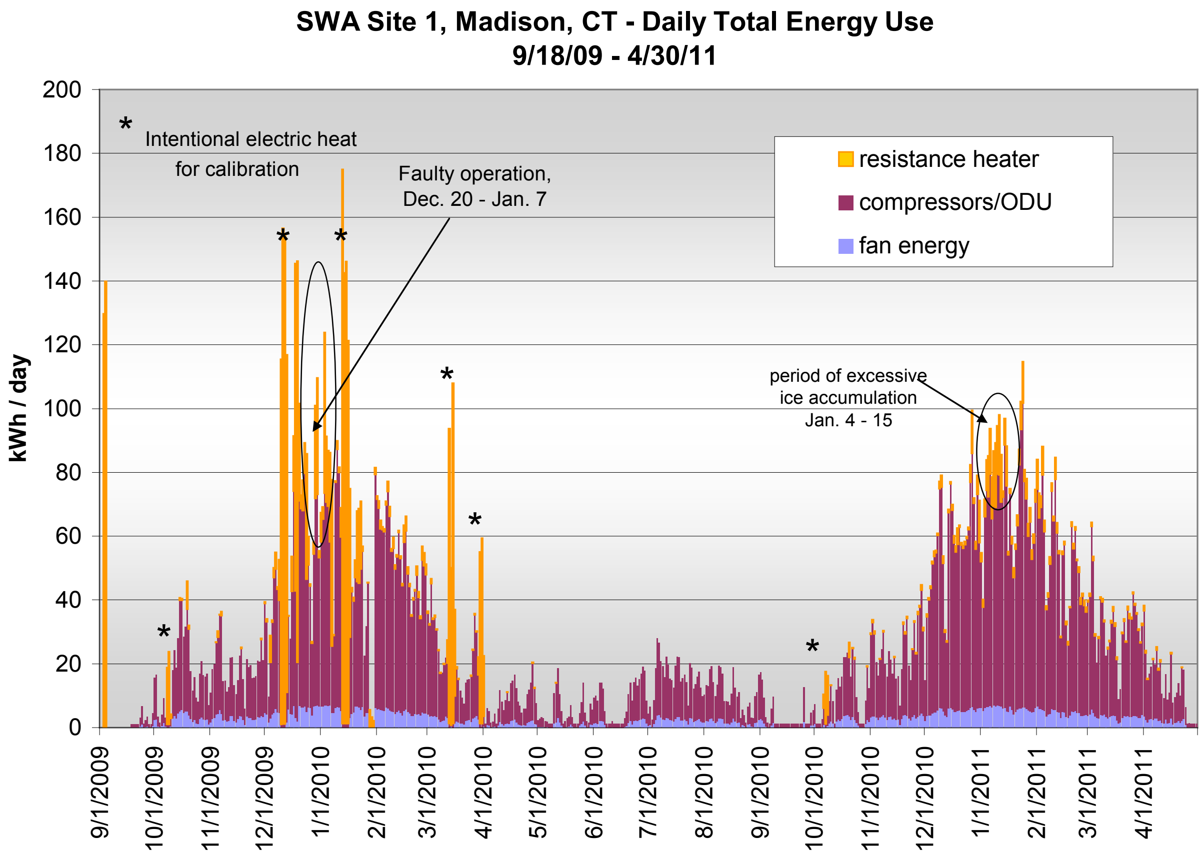

Figure 6. Daily electric energy used by each component of the heat pump: AHU fan, ODU (compressors), and supplemental electric heat. On certain days the electric heat was used intentionally instead of the heat pump for calibration purposes.

The period of faulty operation from December 20, 2009 to January 7, 2010 and January 4–15, 2011 will be discussed in Section 5.4 and Section 6.

Over the entire first winter heating period, September 26 through May 20, the heat pump used 5,306 kWh, excluding the excess energy use caused by intentional electric heating. At the then-current electricity rate of $0.1687/kWh, the owner spent $895 to heat the home over the September to May period. Had the erratic operation of December 20 to January 7 not occurred, the heating cost would have been $38 less.

During the second winter the system used 8,151 kWh over the heating period from October 9, 2010 to April 30, 2011, at a cost of $1,375. There were 90 days in the second winter with an average temperature below 35°F, but just 70 days below 35°F in the first winter.

Under the coldest weather conditions of the test, the system normally did not exceed 80–90 kWh/day, which equates to no more than $13.50–$15.00/day. Heating on the coldest day of the test cost about $20.

Air conditioning was not the focus of the field test but was monitored. The owner spent $183 for air conditioning from May through September 2010. The air conditioning operated as designed.

5.1.3 Seasonal Efficiency

Heat pumps are promoted and sold on the basis of their efficiency. It is important to know this system's seasonal heating efficiency so that it can be compared to other heat pumps in a similar climate. (Comparing seasonal energy efficiency ratios for various equipment is very approximate because the rating is calculated for a milder climate.)

As introduced in the previous discussion of test methodology, efficiency is calculated from two factors: heat produced (delivered) by the heat pump and electric energy consumed by the heat pump. Using the minute-by-minute measurements made on this system, one can calculate efficiency on a seasonal basis, on a daily basis, and minute by minute.

Seasonal efficiency, or SCOP, is the ratio of these two season total values (energy out/energy in). It is calculated from the formula

$$efficiency = \frac{Btuh\ delivered\ /\ 3{,}413}{input\ power}$$

Where:

Btuh delivered = the sum over the heating season of the Btu delivered during each minute of heat pump operation

3,413 = a unitary conversion factor: 3,413 Btuh = 1 kW

Input power = the sum over the heating season of the kWh used by the entire heat pump system during each minute of operation.[1]

During the first heating season the heat pump delivered a total of 58,375,000 Btu to the home. The heat pump consumed 5,306 kWh. This yields a SCOP of 3.22.[2] During the second heating season the system delivered a total of 74,673,000 Btu to the home and consumed 8,151 kWh. This yields a COP of 2.68. There was no intentional use of electric heat during the second season. The first heating season ended on May 20, 2010. The second heating season ended on April 26, 2011.

Bonneville Power Administration also sponsored research on this technology (Eckman et al. 2009) and found considerably lower efficiency values (COPs < 2.0). The large difference between these systems is that the Consortium for Advanced Residential Building's evaluation optimized the system performance by correcting all installation issues, while the Bonneville

[1] In this test actual power used by the AHU fan and ODU fan was used. Fan power is not counted when the fan is run without the compressor(s) for air circulation and filtration. In Standards testing the manufacturer may use standardized values for fan power, which are often lower than found in the field.

[2] As noted, the heat pump was overridden at times, shutting it off and heating with the electric resistance elements in the AHU. We have used its operation during normal periods to predict the energy that it would have used had the electric heat not been forced on. This estimated heat pump energy use is included in the calculation instead of the electric heat's energy use. The December 20 to January 7 period of faulty operation is included in the net SCOP without adjustment.

Power Administration study had several known installation issues (high external static pressure and broken outdoor temperature sensor). The impact of these installation issues on the overall performance from the Bonneville Power Administration study are unknown.

The monitoring contractor is currently testing a second method of estimating SCOP that could be implemented with simpler instrumentation. The method requires only the measurement of daily kWh used by the system and daily average outdoor temperature. It also requires that the electric heater in the AHU be used periodically, as was done during the first winter.

This method is called the "ratio of slopes" estimate of SCOP and is shown in Figure 7.

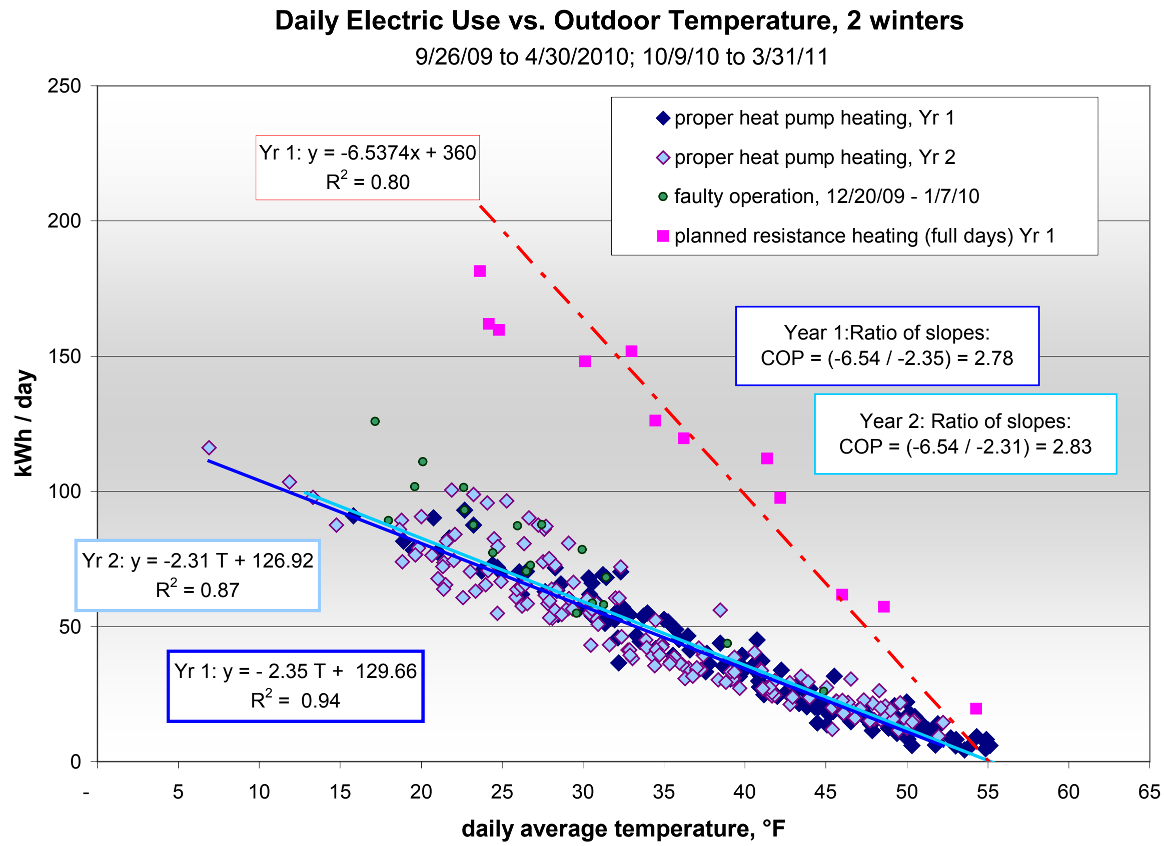

Figure 7. Daily total electric energy used by the heat pump is compared to daily total electric energy used by an electric resistance heater across the heating season. A linear regression is calculated for each heating system.

The method plots daily total kWh against daily average outdoor temperature for normal heat pump operation, shown in dark blue for the first winter and lighter blue for the second winter; and again for electric heat operation during the first winter, shown in red. Note that the electric heat is delivered through the same ducts as heat pump heat, so the method compares the two heat sources equally.

The ratio of slopes calculation requires that each heat source provide the same total Btu of heat to the house over a given day assuming the same outdoor temperature. This was tested and found to be true. (This might not be true if the electric heat had been from a room-by-room distributed source; e.g., baseboards, while the heat pump heat was ducted.)

The accuracy of the method is dependent on having little scatter in the data, as measured by the R^2 value of each linear regression. In this case the heat pump data are very consistent, with R^2 = 0.94 and 0.87, while the electric heat data are slightly noisier, with R^2 = 0.81. The method also assumes that the data points are best described by a linear regression. Thirteen days with mixed electric resistance and heat pump heating were excluded, as were the 17 days of faulty operation (shown in green).

This calculation estimates that the SCOP of the heat pump is 2.78 and 2.83 for the two winters. To illustrate the impact of "noise" (low R^2) in this calculation, eliminating the one or two highest or lowest electric heat data points in the first winter (those furthest from the regression line) caused the resultant SCOP to vary from 2.58 to 2.94.

5.2 Daily Efficiency

All commercially available heat pumps, including the LTHP, are tested by independent rating laboratories.[3] Figure 8 compares the laboratory rating of steady-state efficiency for this heat pump against its net daily efficiency for each day of the two winters.

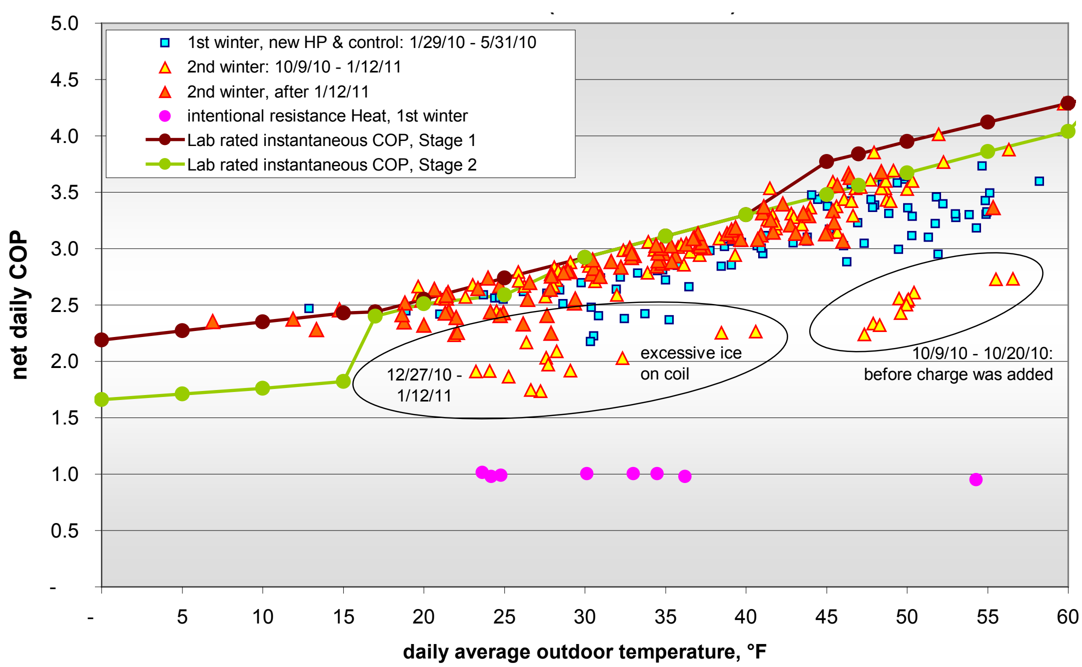

Figure 8. Net daily COP as measured on site is compared to the heat pump's laboratory-rated efficiency. The laboratory data are established in an environmental chamber, with the heat pump fully warmed up and operating under steady-state conditions.

[3] Tests were done on the LTHP by ETL Semko following Air-Conditioning and Refrigeration Institute Standard 210/240-2006. The standard test procedure had to be modified to include more than the normal two stages of operation and to include measurements below the normal low limit of 17°F.

Net daily efficiency is calculated by dividing total delivered energy per day by total kWh used per day (in like units).

Figure 8 shows the daily measured efficiency as discrete data points and laboratory-rated efficiency as points connected by lines. There are several groups of data points on the chart.

- The light blue squares represent days following the replacement of the heat pump and its control (discussed below) in the first winter.
- The yellow/orange triangles show the first part of the second winter. There were two issues evident in these data.
    - First, there was a loss of refrigerant in October 2010 that lowered the system's efficiency. The system was recharged and its efficiency returned to normal.
    - Second, there was a period of cold weather when the outdoor coil accumulated ice that was not fully removed during defrosting.

These issues will be discussed further in Section 7.

The chart shows that the system's daily efficiency was very close to its instantaneous efficiency, with the performance appearing to be better during the second winter. In other words, the system suffered very little from on/off cycling or from defrosting aside from the one ice-accumulation issue. The system may have cycled more frequently than normal because of its two zones.

The chart includes the days when electric heat was used intentionally shown as pink circles. Those days had an efficiency of one, as expected with electric resistance heat.

As Figure 8 illustrates, the system seemed to be operating slightly more consistently and efficiently over the second winter, without efficiency-robbing faults. That may be due to the charge being slightly better optimized. The one exception was the period of ice accumulation from December 27, 2010 to January 12, 2011, which will be discussed in the next section.

5.3 Noteworthy Events

This Acadia heat pump generally operated well with a few notable exceptions.

5.3.1 "Weak Mode 2"

One very strange operation, which we are calling a "weak Mode 2" for lack of a better term, was identified through the monitoring during the first winter.

During October and November 2009 the heat pump very occasionally operated oddly. It would draw power at a level that was between Modes 1 and 2. It drew 1.8–2.2 kW, while Mode 1 normally would draw 1.6 kW and Mode 2 would draw 3.3 kW. At the same time the supply air temperature would be only half as warm as it would have been in Mode 1. With the return temperature at 66°F, the supply air would be only 71°–74°F. In regular Mode 1 under similar conditions, the supply air would be 82°–85°F.

This happened only two or three times, briefly, in October and in November. The problem manifested more frequently in December, happening one to three times per day on about half the days. It appeared to happen immediately after a defrost cycle and would clear with the next defrost cycle. This issue will be discussed in more detail in Section 6.2.1.

The data did not indicate whether Mode 1 was drawing excessive power or Mode 2 was drawing low power. Airflow measurement indicated that the system "thought" it was in Mode 2. No cause for this "excess power/low output" problem was identified at the time, though the manufacturer later identified a motor contactor as being intermittently faulty at other installations. The contactor occasionally would not fully close both contacts, causing one of the two windings in the compressor motor to not be energized. The faulty contactor was apparently responsible for compressor failures at other installations.

5.3.2 Excessive Electric Heat

A second problem started to occur on December 20, 2009. The system began erratically to use electric heat as a supplement to Mode 2, instead of transitioning to Mode 3. The "weak Mode 2/low supply temperature" problem also continued to occur sporadically.

On December 30 when the outdoor temperature was below 15°F, the system would mostly use Mode 2 instead of the more appropriate Mode 3. It heavily supplemented Mode 2 with 4.7 or 12 kW of electric heat. It should have been using Mode 3, with minimal if any electric heat supplement. The excessive use of electric heat can be seen in the Figure 6 daily bar chart.

5.3.3 Outdoor Unit Replaced

The ODU was completely replaced on January 7, 2010. It is believed that the replacement was done as a precaution because the specific cause of the above problems could not be identified at the time, and because the Hallowell service technician had brought a replacement unit in case it was needed.

Both of the above-described problems continued to occur: low output with weak Mode 2 power draw resulting in excessive use of electric heat with Mode 2. The first winter data in Figure 8 show the period after that replacement only.

5.3.4 Control Replaced

On January 28, 2010 the older style Athena brand master control board in the new ODU was replaced with the manufacturer's updated control from a different supplier, Invensys.[4] Three internal temperature sensors were replaced along with the control.

Since the replacement on January 28, 2010 the system has operated as designed. The previous Figure 6 showed that there was very little electric heat used since the control replacement.

It has been speculated that the former control was somehow losing its ability to accurately measure outdoor temperature. This could have caused the system not to transition to Mode 3 as outdoor temperature dropped. The thermostat would then generate a Stage 2 call for additional heat as the indoor temperature dropped. The heat pump would try to satisfy the thermostat by adding electric heat.

[4] The Invensys control board can be identified by its two-digit light-emitting diode numerical display, used for diagnosing faults. The older control board did not have a digital display.

5.3.5 Refrigerant Loss

Late in the air-conditioning season the homeowner noted that the delivered air was not as cool as expected. Data verified that the system had lost cooling capacity. The service mechanic traced the problem to a refrigerant leak in the AHU coil. He repaired the leak and added refrigerant to replace the loss. The system then air conditioned properly, and has also been heating properly since then. It is noteworthy that the AHU was supplied by a major manufacturer directly to installers; it was not built by Hallowell.

5.3.6 Ice Accumulation

During the coldest part of the second winter a problem arose with ice accumulation on the ODU's cabinet. The ice is shown in Figure 9. As discussed in Section 3.3 on defrosting, it is normal for a thin layer of frost or ice to accumulate on the fins of the outdoor coil. A defrost cycle is normally sufficient to melt this ice away and return the heat pump to full efficiency.

Figure 9. Ice accumulation on the outdoor coil. The ice was not removed during defrost cycles.

The defrost cycles during this period may have been too short. The meltwater that ran off the coil during defrosting would apparently hit the cold steel frame of the ODU and refreeze. Over many defrost cycles the ice would build up to a point where it inhibited airflow through a good fraction of the coil. This lowered the system's efficiency and the temperature of the delivered supply air. Repeated defrosting could not clear the ice because, during defrosting, a layer of ice immediately against the coil would melt and run off, leaving a gap between the coil and the remaining ice.

The manufacturer recognized this as a problem. Their owner's manual specified that the ice should be removed by pouring hot water over the coil. This homeowner addressed the problem by connecting a garden hose to a basement faucet and flushing the ice away with tap water.

This ice accumulation has been known to cause catastrophic damage at other installations. An installation in central Canada had such a severe accumulation of ice that the ice crushed and cracked a copper tube in the outdoor coil, causing the refrigerant to be lost (Lamb 2013). The heat pump industry has not had to address this problem because other air source heat pumps normally do not operate in such cold conditions.

6 Operating Characteristics in Detail

6.1 Examples of Cold Weather Operation

Figure 10 shows the operation of the heat pump on the coldest day of the first winter. The figure demonstrates its use of several modes of operation. The figure shows minute-by-minute operation of the heat pump's compressors and fans, along with indoor, outdoor, and duct temperatures.

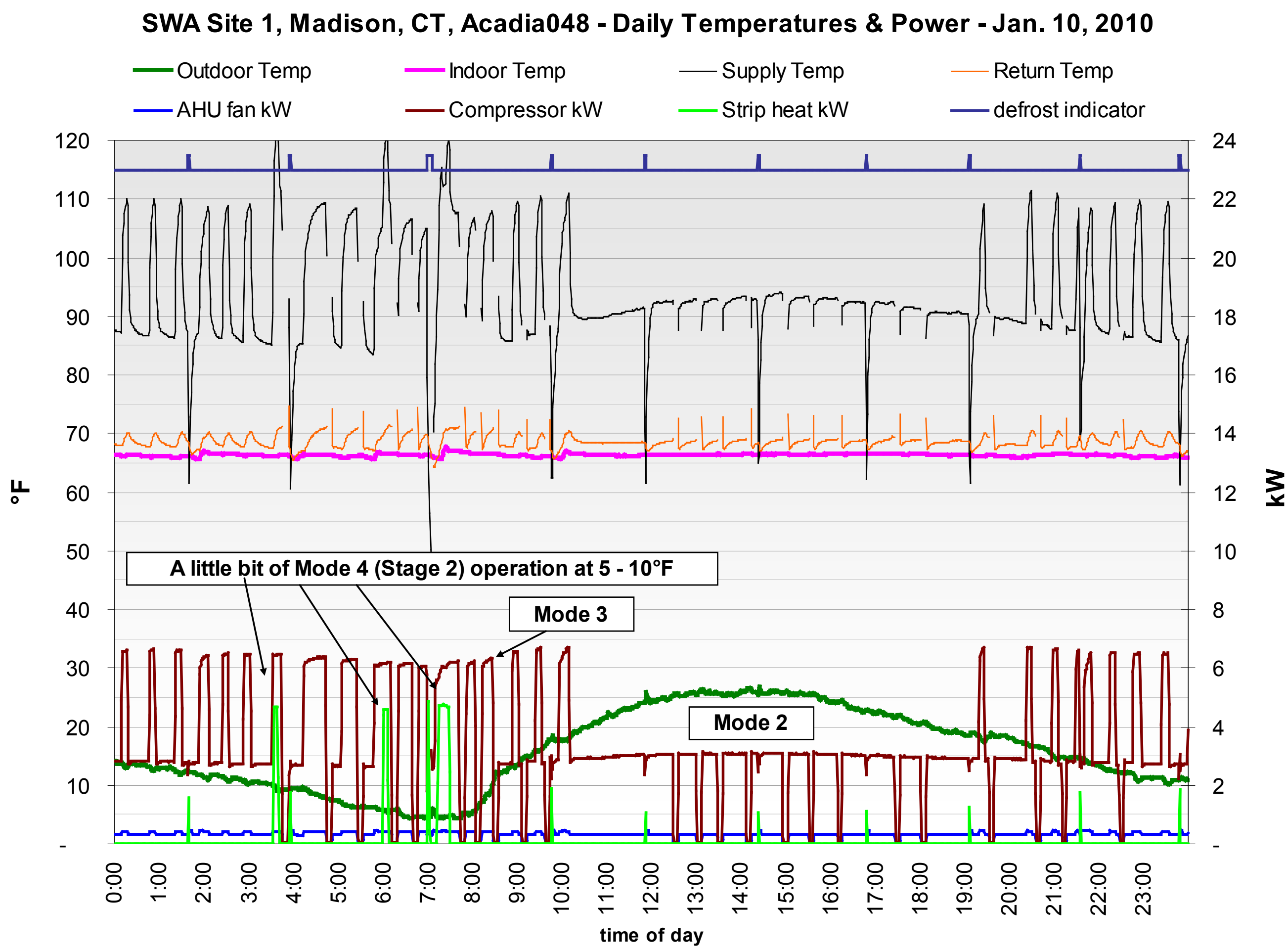

Figure 10. Minute-by-minute data for January 10, 2010. The chart shows indoor and outdoor temperatures; supply and return air temperatures; power drawn by the ODU, AHU fan, and electric heat; and defrost cycles.

The figure is complex. It shows the following:

- Outdoor temperature is shown in dark green; indoor temperature at the second floor thermostat is in bright red.
    - The indoor temperature was very steady at 66°–67°F.
    - The outdoor temperature reached a low of 4.4°F and a high of 26.8°F. It averaged 15.8°F over the entire day. It remained below 10°F for 5 h.

- Supply temperature is in black; return temperature is in orange.
    - The supply temperature varied with outdoor temperature and operating mode.
    - The return temperature generally remained at 66°–70°F. The return air is a mix of first and second floor air.
- Power drawn by the ODU (compressors, fan, and controls) is shown in brown.
    - Operating modes are identified by their power draw. Mode 2 generally draws 2.7–3.3 kW. Mode 2 was used from 10:00 a.m. to 7:00 p.m.
    - Mode 3 draws 6.0–6.6 kW. The exact power is determined by the outdoor temperature.
    - Mode 3 alternated with Mode 2 in the morning and evening as determined by the room thermostats' Stage 1 and Stage 2 "calls."
- The dark blue horizontal line near the top of the chart is an indicator showing when the system is defrosting. Each "blip" represents one defrost cycle. There were 10 defrost cycles over a total of 27 min on that day.
- The blue line near the bottom shows AHU power. On this day the AHU fan used less than 8 kWh of the 91 kWh total, or 9% of the total power.
    - Two fan power levels can be seen. In Mode 2 the fan draws 330 W, supplying air at around 1,180 CFM.
    - In Modes 3 and 4 the fan power increases to 420–450 W, supplying air at about 1,430 CFM.
    - Normalized fan power was typically about 3.6, 5.3, or 6.1 CFM/W, depending on mode selection.
- The lime-green line shows the power drawn by the electric resistance heater in the AHU.
    - There were three cycles where the heater drew about 4.7 kW. These were to supplement Mode 3, which then became Mode 4.
    - During defrosting the system operates a 4.7-kW heating element. In most cases the power measured about 0.5–1 kW over a 1-min measurement interval, indicating that the heater operated for a fraction of a minute during each measurement cycle.

The "bottom line" is that this figure shows that at the lowest temperature of the season, 4°F, the system barely used any supplemental heat. Over the entire day the system used 91 kWh, but only 2.7 kWh of that total (3% of the day's total power) was used by the resistance heater, including energy for defrosting.

An electric utility is normally concerned with a customer's peak demand during the coldest weather. Demand is typically measured over 30-min intervals. During the coldest hour on January 10 (7:00 a.m. to 8:00 a.m.) this house consumed 55,500 Btu of heat. The outdoor temperature was just under 5°F. That amount of heat could have been supplied by an electric heater with an average demand of 16.3 kW. The heat pump in comparison had an average demand of 6.7 kW from 7:00 a.m. to 7:30 a.m. and 4.7 kW from 7:30 a.m. to 8:00 a.m. on the day shown.

The system's COP over the entire day, with an average outdoor temperature of 15.8°F, was 2.4.

Figure 11 shows the coldest day of the second winter, and of the entire test period.

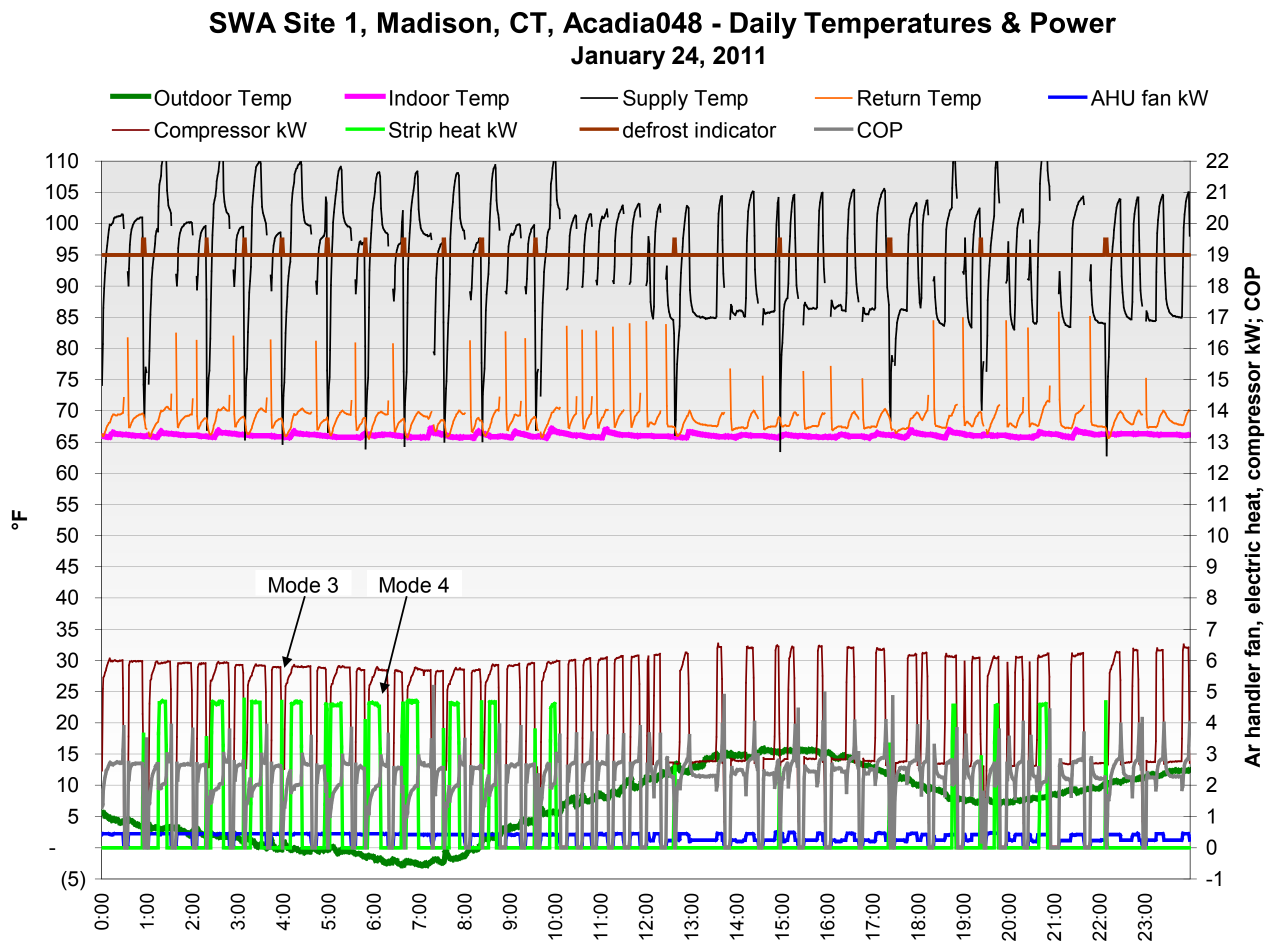

Figure 11. Minute-by-minute operating data for January 24, 2011, the coldest day of the field test. The chart shows indoor and outdoor temperatures; supply and return air temperatures; power drawn by the ODU, AHU fan, and electric heat; defrost cycles; COP; electric heat; and defrost cycles.

- This chart shows the electric heat supplementing the heat pump intermittently when the outdoor temperature drops below 5°F.
- The gray line on this chart shows minute-by-minute COP. When the outdoor temperature is below 5°F, Mode 3 achieves a COP > 2.5, while adding supplemental electric heat to Mode 3 drops the COP to about 2.0.
- The electric heat also boosts the supply air temperature by 10°F.

- Once the outdoor temperature rises above 12°F, Mode 2 is able to alternate with Mode 3 to satisfy the home's heating requirements without supplemental electric heat.

6.2 Verifying System Performance

One purpose of detailed, minute-by-minute measurement of all significant parameters is to reveal any operational anomalies that might occur. Data collection at 1-min intervals reveals details of operation that less-frequent measurements (i.e., 5-min interval or 15-min interval recordings) would hide. It is safe to say that monthly billing-meter data would reveal nothing about the unique characteristics of the system's performance.

6.2.1 Identifying Anomalous Operation

Odd compressor operation ("weak Mode 2") was noted in Section 5.4. The following discussion shows how the problem was identified.

Figure 12 shows a 6-h period on December 1, 2009. Charts like this one are made and examined for each day of the field test. This chart shows information that would never have been discovered from bill reviews, or from coarser interval data.

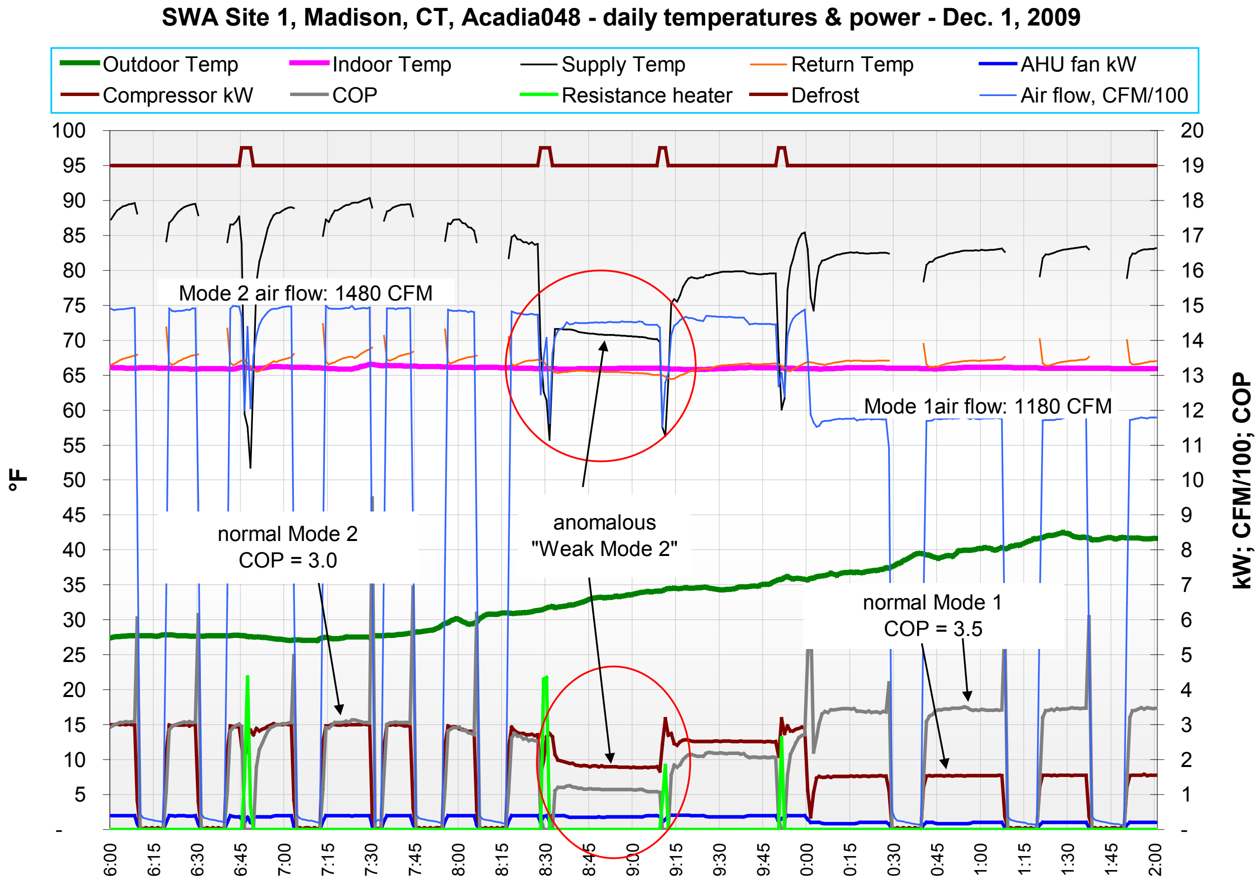

Figure 12. Minute-by-minute data for part of December 1, 2009. The chart shows the anomalous Mode 2 operation of the primary compressor.

The chart shows the following:

- The chart starts normally with Mode 2 operating as expected in the 25°–30°F outdoor temperature range. The primary compressor is using both cylinders and drawing about 3 kW. Airflow is normal for Mode 2 at a little less than 1,500 CFM. COP meets the manufacturer's specification at about 3.0 in this temperature range.
- After 10:00 a.m., Mode 1 operates as expected at the warmer outdoor temperature. The primary compressor, using one cylinder, is drawing about 1.5 kW. Airflow is again normal for Mode 1, around 1,180 CFM. COP again meets the manufacturer's specification at about 3.5 in this temperature range.
- The compressor power measurement is strangely low between 8:30 a.m. and 9:15 a.m.
    - It is drawing 1.8 kW. The airflow and fan power measurements indicate that it is trying to use Mode 2.
    - The supply air temperature is only 70° –72°F, significantly less than the 85°–90°F of the previous Mode 2 cycles.
    - COP is also significantly lower at only 1.1–1.2.
- The compressor seems to partially recover between 9:15 a.m. and 9:50 a.m. Its power is higher but not up to the normal Mode 2 level until after the defrost cycle at 9:50 a.m.
- This weak Mode 2 operation seems always to initiate immediately after a defrost cycle. It generally continues until there is another defrost cycle.

These data were shared with the manufacturer. The manufacturer tentatively diagnosed the problem as due to a sticking reversing valve.[5] The valve was replaced on December 16 but the problem continued the next day. That indicated that the reversing valve was apparently not sticking.

A second diagnosis indicated that there might be moisture in the refrigerant. Moisture could cause ice to form in an expansion valve or distributor (a device that divides refrigerant among several parallel coils in the outdoor unit). This seems unlikely because moisture would have been removed during the evacuation process following the valve replacement and before recharging with refrigerant, or the problem would have been more consistent. This was apparently not the cause of the problem.

Another possible diagnosis, in hindsight, might attribute the problem to a faulty compressor-motor contactor that was not sending proper voltage to both motor windings. This contactor issue had not yet been identified by the manufacturer, but was later found at other installations.

As noted, the problem had not recurred since the ODU and control were replaced. A specific cause was never identified by the manufacturer.

[5] The reversing valve changes the direction of flow of refrigerant. It sets the system into either heating or air-conditioning operation. It activates during defrost cycles.

6.2.2 Verifying British Thermal Unit Output

The Acadia's thermal output was measured for the manufacturer by an independent testing laboratory according to established Air Conditioning, Heating, and Refrigeration Institute standards. This type of testing is done in environmental chambers with very closely controlled and standardized temperature and humidity conditions. Equipment is operated until it stabilizes at fully warmed-up, steady-state condition before measurements are made. The information was published on the manufacturer's website.

To verify proper heat pump operation, the heat output of the heat pump was measured in the field during each minute of its operation. That measurement was then compared to the manufacturer's specification. Figure 13 shows an example of this measurement for every operating minute of February 2010.

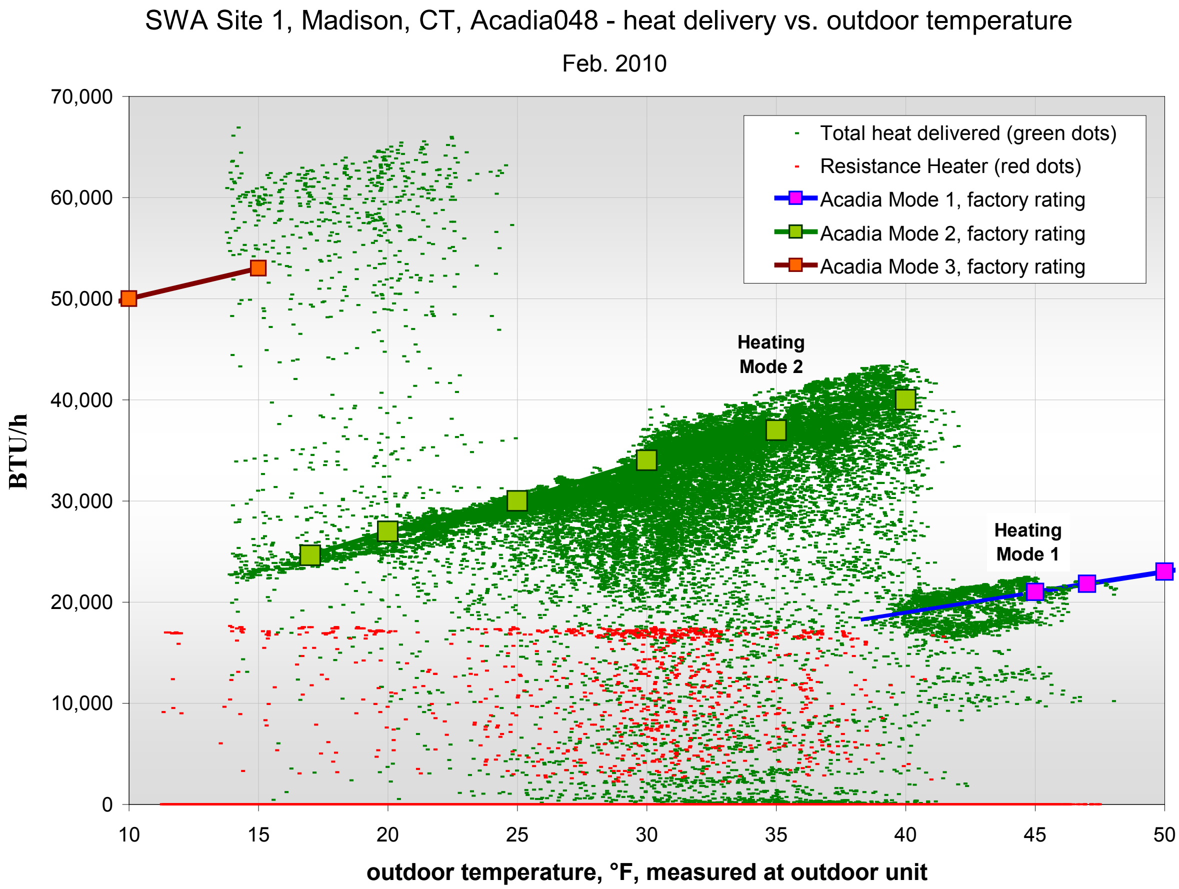

Figure 13. Minute-by-minute data for February 2010 showing Btu output rate during each operating minute

The large, square data points of Figure 13 show the manufacturer's laboratory-rated heat output of the Acadia048 model used at this site.

- The green dots show measurements taken each minute. Red dots show the operation of the electric heat during defrost cycles.

- The chart demonstrates that most the operation in February was in Mode 2, with some operation in Mode 1 and just a bit in Mode 3.
- The transition from Mode 1 to Mode 2 occurred at 40°–41°F, exactly as the manufacturer has programmed into the control.
- Mode 2 is allowed by the control to operate as low as 15°F, while Mode 3 can operate, when needed, as high as 25°F. This operation is exactly as programmed as well.
- The greatest density of Mode 1 and 2 operating points clusters around the rating lines. There are also a lot of operating minutes where the output is less than rated, particularly in Mode 2. This is because the system must warm up at the start of each operating cycle.

This system, since its repair in January 2010, has been heating as designed.

6.2.3 Verifying Efficiency

Efficiency can be verified in the same way that the Acadia's heat output was verified. Figure 14 shows COP as rated by the manufacturer and as calculated for each minute in February 2010 from Btu output and kW input to the system.

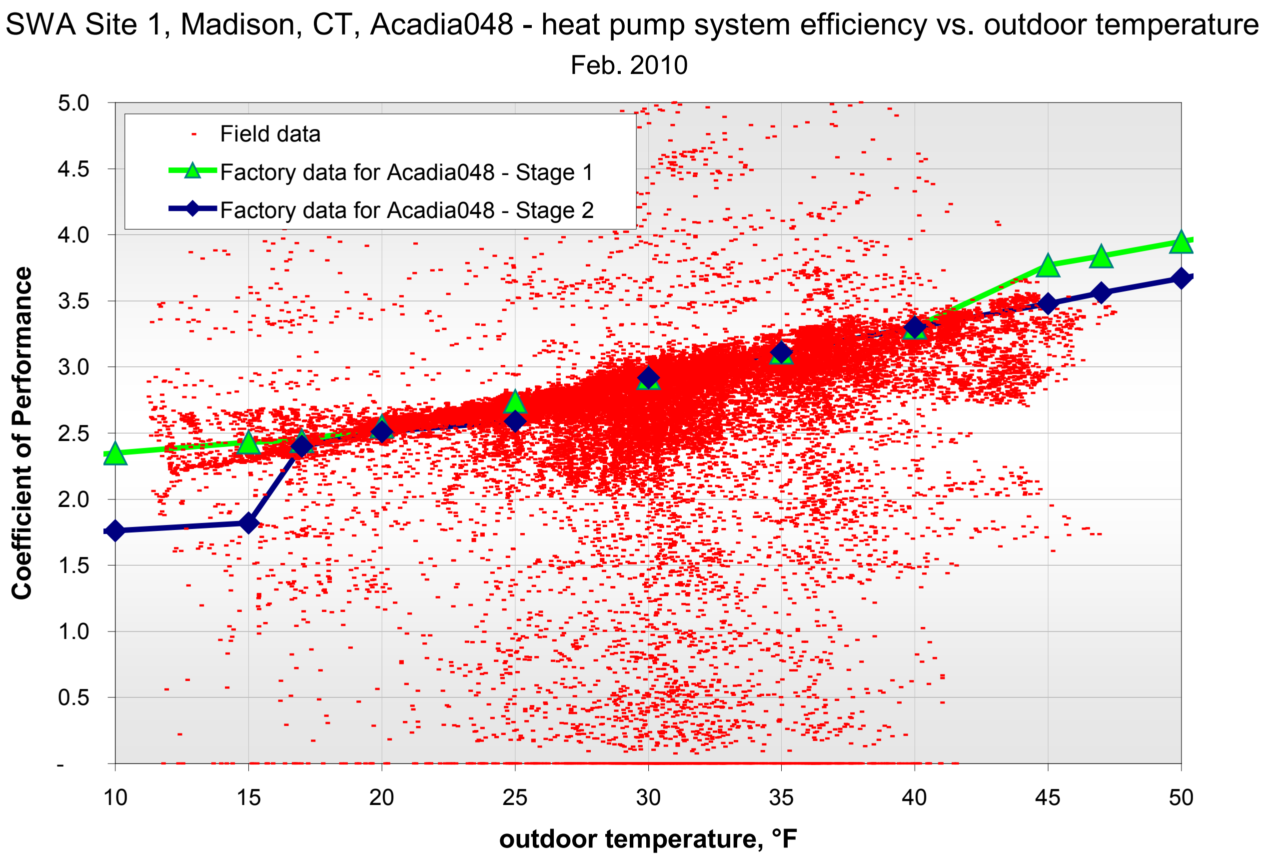

Figure 14. Minute-by-minute data for February 2010 showing COP during each operating minute

- Figure 14 demonstrates that the system's COP was fully meeting the manufacturer's laboratory-rated values across the full range of February's outdoor temperatures, 12°–47°F.
- The modest number of data points that fall well below the rating line shows that the system warmed up relatively quickly in each operating cycle. The loose scatter of data points above the rating line is due to the compressor having shut off, while the AHU fan "scavenges" heat that remains in the coil.
- As the rating line and data points indicate, there was no significant difference in COP between Mode 1 and Mode 2.

Again, these data indicate that the LTHP at this site was operating as it was expected to, without problems.

7 Discussion

7.1 Best Applications for a Low Temperature Heat Pump

7.1.1 Cold Climates

The best applications for the LTHP are, naturally, in cold climates. Other LTHPs from the same manufacturer have been field tested by the monitoring contractor in colder climates than Connecticut—in northern New Hampshire; northeastern Minnesota; western Colorado; Juneau, Alaska; and the plains of central Canada. The heat pumps were generally able to operate at the lowest temperatures seen (below –20°F), often with little to no supplemental electric heat being used.

Still, there is potential benefit in milder temperatures (though the cost/benefit will vary significantly). Several years ago the monitoring contractor tested an Acadia near Orlando, Florida. The outdoor temperature never dropped below 30°F and the heat pump never used Mode 3. The test was considered a relative success, however. The heat pump never used supplemental electric heat, while more typical air source heat pumps in that climate would generally be heavily supplementing with resistance heat, starting in the mid-30s. The concern in that climate is the winter peak load placed on utility companies by heat pumps. The manufacturer had at one time claimed to be developing a "light" version of the LTHP with only one compressor, specifically for use in milder climates.

7.1.2 Site and Heat Pump Selection Criteria

The most critical factor in the successful application of the LTHP is the equipment size selection, relative to the candidate house's design-temperature heating load. Other field testing has demonstrated the problem of a too-small heat pump in a house with a too-high heat loss (Johnson 2010). (Existing ducts in that house were too small to accommodate a properly sized heat pump.) In that installation the heat pump frequently called on electric heat, so heating costs were higher than the homeowner had anticipated.

An installer must be particularly careful in calculating a candidate site's "design heat loss." This is the rate at which a house loses heat at the lowest normally expected outdoor temperature at that site. The design heat loss is then compared to the various LTHP models' outputs at that design temperature. One must ignore the heat pump's rating value; i.e., 48,000 Btu for this monitored system, and select a heat pump whose actual Btu output is sufficient (or nearly so) at the design temperature.

This house has a measured heat loss of about 947,000 Btu/day at an outdoor temperature of 5°F, which is equivalent to a heat loss rate of 39,500 Btu/h. Note that this is an average value, not a worst-case value. In fact, the calculation in Section 6.1 from January 10, 2010 found that the home's heat loss at 5°F was about 55,500 Btu/h. The model Acadia048 used at this site has a factory rated output of 47,000 Btu/h at 5°F when operating in Mode 3 and without supplemental electric heat. This indicates that the Acadia048 was closely sized for the test site. It would be expected to use a moderate amount of supplemental electric heat at that temperature and below. The January 10, 2010 data in Figure 10 confirmed this, showing that the system used only a little supplemental heat at 4°F.

7.1.3 Low Temperature Heat Pump Operating Cost Compared to Other Heating Systems

It is instructive to compare the operating cost of this Acadia LTHP to other heating systems.

The tested house consumed about 58 million Btu of heat over the entire first winter. We shall assume a similar "standard" house that would use 60 million Btu and compare the approximate cost to heat that house with the LTHP and with other fuels and systems. Table 4 compares the results.

Table 4. Cost of Heating With Various Fuels and Systems

	Fuel Cost[a]	Fuel Cost per Million Btu Delivered	Annual Heating Cost to Deliver 60 Million Btu
Acadia Heat Pump, SCOP = 2.78[b]	\$0.1687/kWh	\$17.78	\$1,067
Central, Ducted Electric Heat, 100% Efficiency	\$0.1687/kWh	\$49.43	\$2,966
Baseboard Electric Heat, No Duct Losses	\$0.1687/kWh	\$42.01	\$2,521
Premium 2-Speed Heat Pump,[c] SCOP = 1.78	\$0.1687/kWh	\$27.77	\$1,666
Natural Gas, 70% AFUE[d]	\$1.41/100 ft^3	\$19.54	\$1,172 + \$135 = \$1,307
Natural Gas, 90% AFUE[d]	\$1.41/100 ft^3	\$15.20	\$912 + \$135 = \$1,047
#2 Fuel Oil, 70% AFUE[d]	\$2.70/gal	\$27.85	\$1,671 + \$135 = \$1,806
GSHP, SCOP = 4.0	\$0.1687/kWh	\$12.36	\$741

[a] Electricity cost = \$0.1687/kWh from www.cl-p.com/rates/averagebill.aspx; natural gas cost = \$1.41/ccf from Yankee Gas data. Fuel oil cost = \$2.70/gal from the Energy Information Administration, past winter average for Connecticut. All fuel costs are as of June 2010.
[b] LTHP operating costs have been adjusted to eliminate the negative impact of operating the heat pump intentionally on strictly electric resistance heat.
[c] The "premium" heat pump was located in central New Hampshire and tested for National Rural Electric Cooperative Association-Cooperative Research Network. See footnote 8, next page.
[d] Estimated AHU fan operating cost of \$135 (average of this system, both winters) has been added to the annual operating cost of fossil-fired systems. This system used an electronically commutated fan motor; a constant speed fan could use more energy.

At local utility and fuel rates in effect at the time of the test, this Acadia heat pump, at a SCOP of 2.78, cost less to operate than any other heating system except a GSHP with SCOP = 4 or a natural gas furnace at 90% AFUE.

The chart includes one "typical" air source heat pump that was previously monitored in New Hampshire (Johnson 2009a). This unit was chosen because it is a two-speed system near the high end of its manufacturer's line of heat pumps. The Table 4 data demonstrate, roughly, that the Acadia would cost about \$600/yr less to operate than that "ordinary" heat pump.

Please note that this exercise is hypothetical. Simplifying assumptions have been made about various other heating systems' seasonal efficiencies, though the Acadia efficiency value was determined in this test. The GSHP system efficiency is simply an optimistic value often claimed in advertising. One geothermal heat pump that was field tested by this monitoring contractor yielded a SCOP of 3.47 (Johnson 2009b). That system appeared to have used excessive circulator pump power, and might have otherwise rated slightly higher.

7.2 Gaps, Barriers, and Issues in Low Temperature Heat Pump Implementation

7.2.1 Manufacturer Issues

Perhaps the most significant factor impacting future use of LTHPs is the fact that, as this was being written, the Hallowell International Company has gone out of business. Hallowell had hoped to sell the company to a larger manufacturer, but nothing had been finalized. At last report (April 2011) its bank had control of its assets and was intending to sell the company assets in whole or pieces.

It is the monitoring contractor's impression that warranty repairs were costly to Hallowell, and that warranty work could not be financially supported from the revenue of new equipment sales.

The monitoring contractor had for a long time urged Hallowell to explore the possibility of variable-speed drives for its compressors. That concept was, in fact, part of the inventor's original prototype LTHPs. If a new manufacturer were to acquire rights to the patents and produce a similar heat pump, it would no doubt need to explore this option. Variable-speed compressor and fan operation may be necessary in order to compete with the high-tech capabilities of the newer ductless, mini-split heat pumps, some of which claim to operate at outdoor temperatures as low as 5°F.

7.2.2 Site Issues

7.2.2.1 Equipment Capacity and Design Load

A constraint on the implementation of LTHPs had been their available sizes. The Acadia heat pump was available in only three capacities, nominally 24, 36, and 48 kBtu. Other sites tested by the author would have benefited from larger or smaller capacities.

Heat pumps must be matched closely to the heat loss of the home. If the heat pump is too large it will mostly operate with very short cycles and will seldom reach its full, steady-state efficiency. If the heat pump is too small, it will work efficiently in mild weather, but will call on generous amounts of supplemental (usually electric) heat as the weather gets cold.

An installer must understand the unique characteristics of the LTHP before selecting the proper unit for a given home. He or she must do a careful heat loss calculation to determine the Btu requirement of the home at the design temperature of that location. If the home's heat requirement is greater than the capacity of the largest available LTHP, the installer should not recommend it for the home. As an alternative the installer might be able to use two LTHPs in a zoned configuration.

7.2.2.2 Retrofit Duct Issues

Retrofit applications can sometimes be problematic. Ducts are often buried in walls and under insulation in attics. Their rate of air leakage to unheated areas is unknown and can't be accurately incorporated in a load calculation, making selection of the proper heat pump capacity difficult.

Further, existing ducts typically have been sized for a fossil-fueled heating system. Those ducts were sized for a lower airflow rate at a higher temperature. In many cases the ducts are too small for a heat pump that may require a higher CFM flow rate. Forcing a high flow through small ducts will result in noisy air delivery and will increase the required fan power. In most homes the ducts were laid out and sized using traditional rules of thumb. If the duct size resulted in a room that was too warm, the flow was simply throttled down with a damper in the duct.

A good heat pump installation requires ducts that are more accurately sized for the heat requirement of each room in the house. One doesn't want to wastefully create excess heat and temperature, just so one can throttle flow in several rooms to achieve comfort.

Air register location is also more critical for heat pumps than for fossil systems. When air is delivered at 140°F, for example, the occupants can remain comfortable in spite of direct drafts. In contrast, the heat pump's typically lower air delivery temperature is often in the 80°–90°F range. One must be more careful in locating registers to ensure that the air does not blow directly at occupants. It should mix with room air away from normal seating areas.

7.2.3 Owners' Comments

The owners were satisfied with the comfort provided by the heat pump but have significant concerns about its reliability. The issue with loss of refrigerant arose near the end of the 2010 air-conditioning season and has caused the owners further concern.

The following comments were provided by the owners at the end of the first heating season:

> "Our Acadia system has provided a vast improvement in amount of thermal cooling comfort. Previously our home did not have central air conditioning and therefore has been extremely more comfortable in the summer months. However, the thermal comfort during the heating season was just slightly better that our previous system. The interior temperatures were more consistent than with our old oil fired system."

> "Over the past year and a half there have been many issues with our Acadia system. My wife and I are disappointed with how the system has been operating. There have been issues with a faulty control panel and most recently a loss of heating/cooling capacity due to a refrigerant leak. The only way to actually prove that there were operating issues is through the ongoing efficiency study by CARB. I can only imagine what other people might be experiencing when trying to explain odd system operation to their installer without daily monitoring and data feedback. Considering that this system has had major issues and is being monitored by a private consultant who will publicly be reporting results, the customer service at Hallowell has been fair at best. Hallowell has been slow to respond to their systems problems and always defer to HVAC installer. Only when all else failed did Hallowell send a service technician and replacement outdoor unit to try and solve the problems. Today, my wife and I are reluctant to recommend the Acadia system to another homeowner."

> "If the Acadia does run like it can and should, it is a very efficient system which does not have the invasive installation similar to a geothermal system."

The owners added this comment at the end of the second heating season:

> "In light of Hallowell closing their doors, I wish we had been able to get our hands on a replacement circuit board. The circuit board is the only part that cannot be easily obtained through our installer. I'm also concerned that 5 year warranty through Hallowell also no longer is in effect. I guess that's what happens when people take a chance on new technology. As long as it continues to operate the way it has since this past November, I'll be happy."

7.2.4 Low Temperature Heat Pump Service Issues

The most significant service issue in this test and other recent tests of the Acadia heat pump has been the system's electronic control. This test site and others have demonstrated that the previous control unit and its related temperature sensors were not sufficiently robust. Control failure typically lead to erratic operation and excessive supplemental electric heat usage. The new control installed in January 2010 (and its three related temperature sensors) made by Invensys had demonstrated complete reliability through the remainder of the test.

This field test has found no other design or service issues with this Acadia unit aside from the ice accumulation issue discussed in Section 5.3. The field test demonstrated complete reliability at this site in these major areas:

- No compressor failures
- No issues with refrigerant circulation or compressor lubrication
- No wiring failures
- No AHU mechanical or electrical failures (either the fan or the fan speed control, or the electric heating elements)
- No thermal expansion valve issues.

The AHU did develop a refrigerant leak during the air-conditioning season. That leak was repaired. As mentioned, the AHU is manufactured by another, major company and delivered directly to the installer. It was not built by Hallowell.

7.2.5 Barriers to Implementation of the Low Temperature Heat Pump

The two most significant barriers to further implementation of an LTHP had been cost (both development and installation) and infrastructure.

As noted previously, it is the author's opinion that the manufacturer did not have the financial capability to both support warranty claims and to continue refinement of the LTHP.

Installation cost has been considered relatively high by some installers and customers. They report that their costs are much higher than for other heat pumps. Some installers had felt that the LTHP's cost was similar to that of GSHP systems. It had been hoped that, as manufacturing volume picked up, costs would have benefited from higher volume and lower warranty overhead.

Infrastructure had also been a barrier to wider implementation. Hallowell management had wisely refused to sell equipment through untrained supply houses and installers. They were aware that an untrained installer could make mistakes that would reflect poorly on their company and the product in general. This had limited its geographic distribution.

8 Conclusions

This two-year evaluation provided valuable data on the true in-field performance of an LTHP. In addition, this information, though only a single site, demonstrates the potential of lower cost LTHP technology. Similar to the progression of heat pump water heaters over the years, it will likely take a large manufacturer to develop this technology into a reliable technology solution. One interesting new offering by Carrier is its Infinity Series Heat Pump with Greenspeed Intelligence. This takes the inverter driven technology that is common in mini-split heat pump systems and provides it in a more conventional central forced-air space conditioning system. Field evaluation is needed to verify the performance of this system, but it shows that even though the technology is different than the Acadia system evaluated in this report, the ability to extract more heat at lower outdoor temperatures is being pursued by the HVAC industry. Here is a summary of the initial research goals for this evaluation:

- Measure the LTHP's net efficiency over two heating seasons in the local Connecticut climate under real-world operating conditions. This efficiency will be referred to as its SCOP. This measurement assists future users in the same climate in estimating their own operating cost with a similar heat pump.
    - The system's SCOP over two entire winters was calculated, based on measured data, to be 3.29 over the first winter and 2.68 over the second winter. A second seasonal efficiency calculation by a different method yielded a SCOP of 2.78 for the first winter and 2.83 for the second winter.
- Determine if the heat pump meets its manufacturer's specifications in terms of control design (i.e., is the heat pump control selecting the proper operating stage for specific ambient conditions), heat output, and operational efficiency across the full range of winter ambient temperatures.
    - This system, since its repair in January, 2010, has been heating as designed. The transition from Mode 1 to Mode 2 occurred at 40°–41°F, exactly as the manufacturer has programmed into the control. Mode 2 is allowed by the control to operate as low as 15°F, while Mode 3 can operate, when needed, as high as 25°F. This operation is exactly as programmed as well.
    - The system's COP was fully meeting the manufacturer's laboratory-rated values across the full range of February's outdoor temperatures, 12°–47°F.
- Provide feedback to the heat pump installer and the manufacturer in case any anomalous operation is seen and service is needed.
    - Feedback was provided to the heat pump installer and the manufacturer on control board issues and ice accumulation on the ODU. Unfortunately the manufacturer of the evaluated system has gone out of business.

References

ASHRAE. (2009). *ASHRAE Handbook – Fundamentals 2009.* Atlanta, GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers, pp. 14–19.

Campbell Scientific. (2013). CR1000 Specifications and Technical Data. Logan, UT: Campbell Scientific, Inc. Accessed August 16, 2013. www.campbellsci.com/cr1000-specifications.

Degreedays.net. (2013). Bizee Software Limited. Accessed August 16, 2013. www.degreedays.net.

Ebtron. (2011). *Installation, Operation and Maintenance Technical Manual HF1/SF1.* Loris, SC: Ebron, Inc. Accessed August 16, 2013. www.ebtron.com/Web_Pdfs/SPSync/Hybrid%20Series/HYBRID%20SERIES%20TECHNICAL%20MANUALS/HF1,%20SF1%20-%20Fan%20Inlet%20Sensor/TM_HF1_SF1.pdf.

Ebtron. (2009). *Quick Installation Guide STA104 Analog Output Transmitter*. Loris, SC: Ebron, Inc. Accessed August 16, 2013. www.ebtron.com/Web_Pdfs/SPSync//Silver%20Series/SILVER%20104%20SERIES%20-%20DEPLETE%20EXISTING%20STOCK%20ONLY%20-%20REPLACE%20WITH%20HYBRID%20SERIES/SILVER%20104%20SERIES%20TECHNICAL%20MANUALS/STx104%20Transmitter/QUICK%20INSTALL%20GUIDES/IG_STA104.pdf.

Eckman, T.: Eklund, K.; Bedney, K. (2009). *Analysis of Hallowell Cold Climate Heat Pump Data, McCall, ID and Portland, OR*. Seattle, WA: Ecotope, Inc. Accessed August 23, 2012. www.bpa.gov/energy/n/emerging_technology/pdf/Hallowell_Cold_Climate_Heat_Pump_Data_Dec2009.pdf.

Johnson, R. (2009a). *Evaluating the All-Climate Heat Pump - Results from Field Tests of an Air-Source Heat Pump Designed to Reduce Demand, Improve Efficiency*, Project 06-16, NRECA Cooperative Research Network.

Johnson, R. (2009b). *Metered Performance of a Geothermal Heat Pump In Olathe, Colorado Monitored for the Delta-Montrose Electric Association*, Dec. 23, 2009.

Johnson, R. (2010). *Field Testing of Three Low Temperature Heat Pumps in Northeastern Minnesota, Using the Hallowell Acadia Low Temperature Heat Pump – Installation, Operation and Performance Results*. Prepared for Minnesota Power Co. by Johnson Research LLC.

Lamb, C. (2013). Anola, Manitoba: Frosty's Heating & Cooling, Personal communication.

Magnelab. (2013). Split-Core AC Current Sensor SCT-0750. Longmont, CO: Magnelab, Inc. Accessed August 16, 2013. www.magnelab.com/products/current-voltage-sensors/split-core-ac-current-sensor-sct-0750.

Measurement Specialties. (2013). Temperature Sensor - Epoxy Series IV BetaCURVE Bare. Hampton, VA: Measurement Specialities. Accessed August 16, 2013. www.meas-spec.com/product/t_product.aspx?id=6293.

Appendix: Monitoring and Measurement Details

Data at the test site were recorded at 1-min intervals for the following parameters:

- **Supply temperature** is measured at four points, with two sensors in each branch (zone) of the supply duct. The thermistor sensors (Measurement Specialities 2013) are negative temperature coefficient RTDs (resistive temperature devices). Sensors are about 2 mm diameter for fast response to changing air temperature. Measurements are typically made about 1 meter past the outlet of the AHU so as to avoid the sensors "seeing" radiant heat from the nichrome-wire strip heater elements.
- **Return temperature** is measured at four points and averaged. Measurement points are located near the entrance to the AHU in four quadrants of the duct.
- **Indoor temperature** is measured directly at the second floor thermostat using the same low-mass RTD.
- **Outdoor temperature** is measured within a weatherhead (radiation shield). The weatherhead is installed in the vicinity of the ODU but far enough away from the ODU that it does not "see" the air exhausting from the ODU fan. At this site it was hung under an eave at the second story level. A second outdoor temperature sensor is placed against the back side of the outdoor unit adjacent to the unit's own control temperature sensor.
- **Air flow** is measured at the inlet volutes of the AHU fan with two sensors on each side of the fan scroll. Ebtron SF1 Fan Inlet Sensors (Ebtron 2011) are used. The four sensors' signals are averaged by an Ebtron STA-104F Transmitter (Ebtron 2009), and sent to the logger as a 0–5 volt direct current signal representing 0–1000 ft/min. The airflow is treated as a relative value proportional to CFM. It is calibrated using a heat balance equation when the electric heat is operated such that the electric heat has a COP of 1.0.
- **Electric power** is measured individually for the ODU (total power), the AHU (total power), and the AHU fan. Measurements are made using individual WattNode WNB-3D-240P power transducers. All power measurements are 240 volt alternating current single-phase. The Watt transducers each measure current using two Magnelab current transformers (CTs). The WattNodes output their measurements to the data logger as pulses proportional to Watt-hours.
- The **current** signals into each WattNode comes from Magnelab (2013) SCT-0750 split-core CTs. CT sizes are 5 to 100 Amps, as appropriate. The CTs' output is 0–333 millivolts alternating current proportional to the CTs' rated full-scale current input. The size is selected to be close to the maximum amps of the measured load. Accuracy is ± 1.0% at 10%–130% of rated current. When the minimum load is expected to be less than 10% of the maximum load, a second watt transducer and CT set is used on the smaller part of the load (i.e., on the AHU fan, at 0.1–0.5 kW, versus the total AHU, which includes 12.5 kW of heating elements).
- **Defrost function.** The defrosting action of the ODU is sensed using a simple direct current-output CT. The transducer is placed on a wire going to the ODU's reversing valve solenoid. The solenoid valve is energized when the system is air conditioning or defrosting.
- **Relative humidity** is not monitored.

A total of 20 data channels are recorded in a Campbell Scientific CR-1000 data logger (Campbell Scientific 2013). The logger calculates temperature from each RTD's resistance measurements. The logger is powered from line voltage through a rechargeable battery so it can continue monitoring, uninterrupted, through power failures. The logger package includes a cellular modem to transmit data to the monitoring contractor's office when called each day.

buildingamerica.gov

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

DOE/GO-102013-3788 ▪ September 2013

Printed with a renewable-source ink on paper containing at least 50% wastepaper, including 10% post-consumer waste.