FINAL

Development of a Best Practices Guide for Integrated Hydronic and Ductless, Air-source Heat Pump Systems

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Abstract

Modern low-ambient, mini-split heat pumps offer the potential to provide both heating and cooling without the cost associated with duct retrofit. It is common for these heat pumps to be installed with the existing heating system. The focus of this work has been hybrid arrangements of this type involving hydronic heating as the existing base system. In such a hybrid arrangement, mini splits offer the potential to achieve high efficiency during milder parts of the heating season, when traditional fuel-fired boilers may have low efficiency due to light load and cycling. During the colder part of the heating season the performance of a heat pump decreases, but the efficiency of an oil-fired heating system peaks. Hybrid systems are currently being installed with a wide variety of integration and control strategies that do not achieve their efficiency potential. In this project, field studies have been conducted to better understand how these systems are currently controlled. These field studies involved six sites in New York State at which mini-split heat pumps were installed to meet part of, or in one case, all of the heating load of the home. Sensors were added and logged to understand when each of the two systems was meeting the heat demand and some basic operating characteristics of the heat pump.These studies did not include measurement of heat pump Coefficient of Performance (COP) or energy use. User motivation for heat pump installation and the basic approach to control the hybrid system were captured.

A separate effort was undertaken to allow analysis of component performance, sizing, and control strategies on annual energy use in model homes in upstate, mid-state, and downstate New York. Performance curves for the performance of a range of heat pumps were used based on available published field and laboratory data. Performance curves for boilers were developed based on prior published laboratory studies. Different strategies for the control of these hybrid systems were explored including switchover at a specified outdoor temperature and switchover at a specified date. Annual heating costs were determined based on New York State Energy Research and Development Authority (NYSERDA) surveys of energy pricing. Colder temperatures, which are associated with lower heat pump performance, and longer system run times for fuel fired boilers, suggest that a boiler system (hydronic) should be used more; the underlying costs of electricity and fuel will obviously be critical factors. A best-practices guide for the use of these heat pumps in a hybrid combination with oil-fired boilers is in the appendix.

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Introduction

Interest in the use of mini-split heat pumps has been increasing in the northeast. These systems can provide a low-cost option for heating parts of a home. Mini splits also provide summer cooling for homes. When retrofit to a home with an existing high efficiency hydronic heating system, the two systems are typically operated as a hybrid combination. There are, however, technical options and significant challenges associated with achieving the efficiency potential of such an integrated system. In a hybrid arrangement, this technology offers the potential to achieve high efficiency during moderate heating periods, when traditional boilers may have low efficiency due to light load and cycling. During the colder part of the heating season the performance of a heat pump falls off, but the efficiency of an oil-fired heating system is at its peak. Reportedly, systems are currently installed with a wide variety of integration and control strategies, which prevents achieving the efficiency potential. One control option is to operate only the heat pump during milder seasons and only the boiler during the colder part of the winter. With this approach, potential limited benefits of the heat pump during intermittent warm periods in the winter are not achieved. An alternative approach has the heat pump as the base system, but when the demand exceeds the capacity of the heat pump, the oil-fired boiler covers the excess heating load. Usually this is accomplished by having the heat pump thermostats set at a higher temperature than that of the hydronic heating system. This approach may result in uneven heating in the home and if the hydronic pipes are in an outside wall, they could potentially freeze. Both approaches overlook heating of domestic hot water. In most cases,an oil-fired boiler will provide hot water by heating a tank in an indirect hot water setup or using a tankless coil, while some may use a standalone water heater. When the boiler provides domestic hot water, it must be kept on during the summer only for that purpose, leading to someinefficient operation with lower performing boilers. Newly emergent heat pump water heaters are an option for this as well, but it cannot be integrated into the existing mini-split heat pump system. Production and usage of domestic hot water systems in this study, however, was not explored extensively, thus, was limited.

To better understand these issues, NORA reviewed available published studies on laboratory/certification tests of mini-split heat pumps, and field tests conducted under this project. The field tests focused on how hybrid systems are installed and controlled/operated and did not evaluate direct energy use. The measurements included documentation of the equipment and setup, indoor temperature throughout the homes, boiler temperature, outdoor air temperature and heat pump delivered air temperature. This data when combined with both boiler and heat pump basic performance data from other sources provides insight into achieved energy efficiency and opportunities for improvements. The sites for the field studies were selected by NORA and approved by New York State Energy Research and Development Authority (NYSERDA). All of the sites were in NewYork State; however, the wide range of temperature profiles in the State make this data applicable tomuch of the northeast.

Site Descriptions

The sites were chosen with the following criteria. The most important of those criteria is for each site to contain a hydronic system powered by an oil boiler, and a mini-split system that overlaps with the hydronic heating system. The study included three sites from Long Island, one in the Hudson Valley, and two in Upstate New York. The selection of sites on a regional basis provided data on the performance of hydronic and heat pump systems working in conjunction in different climates. The selected sites would invariably have unique control methods used by homeowners for the thermostats of both systems. From each site, the zones chosen for the test would also provide many varying conditions which, in one way or another, determine the load placed on the heating systems.

Detailed descriptions of each site and the chosen zone, along with each of their heating systems, are provided below.

Site 1

This site is a 2,000-square feet (ft^2) colonial-style home located in Mount Sinai, NY on the north shore of Eastern Long Island. It has three heating zones in total, one of which is an indirect domestic water heater with a 32-gallon tank. A Dynatherm FP18 boiler with a firing rate of 0.7 gallons per hour (gph) is used for the hydronic heating system. The mini-split system was a Comfort Star 12,000 British thermal units (Btu) indoor unit located in the dining room, with an accompanying external condensing unit in the south side of the house. Specifications are provided in Table 1.

Table 1: Site 1, Unit 1 Specifications

Manufacturer	Comfort Star
Outdoor Unit Model Number	CPG012CA(O)
Indoor Unit Model Number	CPG012CA(I)
Nominal Heating Capacity	12,000 Btu/hr
Nominal Cooling Capacity	12,000 Btu/hr
HSPF	9
Cooling EER	10.5
Cooling SEER	17.2

The mini split is rated by the manufacturer for use to outdoor temperatures as low as 5°F. During mild weather (nominally 32°F and higher) conditions, the homeowner sets the thermostat on the heat pump to 68°F and on the hydronic system on the selected zone to 60°F. On colder days when temperatures are below freezing and they feel the heat pump system cannot keep up, the hydronic system’s thermostat setting is manually increased to 70°F, making it the primary source of heating. These changes were made manually, on a day-by-day basis.

A front view of the site and the heat pump unit are shown below.

Figure 1: Front view of site 1

Figure 2: Heat pump for site 1

Site 2

This site is a Cape Cod style house located in Bethpage, NY. It consists of two heating zones alongwith an indirect water tank for domestic hot water, resulting in 3 zones that the boiler services. The boiler in this site is a Weil-McLain Gold three-section cast iron section boiler with a Riello burner. Both indoor and outdoor units for the mini-split system are Fujitsu. The indoor has a capacity of 27,000 British thermal units per hour (Btu/hr) and the outdoor has a capacity of 36,400 British thermal units (Btu/hr). More detailed specifications are in Table 2.

Table 2: Site 2, Unit 1 Specifications

Manufacturer	Fujitsu
Outdoor Unit Model Number	AOU36RLXFZ
Indoor Unit Model Number	ASU24RLF
Nominal Heating Capacity	27,000 Btu/hr
Nominal Cooling Capacity	27,000 Btu/hr
HSPF	9.4
Cooling EER	8.8
Cooling SEER	16

The first floor is the zone chosen for this study, which includes a kitchen, living room, dining room and a bathroom. The heat pump unit analyzed at this site is in the living room. The hydronic system thermostat automatically sets the target temperature to 66°F every day at 5am. On some days, the heat pump thermostat is manually set to 70°F, making it the primary source of heating during the day. Once the zone has reached a comfortable temperature, the homeowner may shut it off. The hydronic thermostat is automatically set back to 62°F and heat pump is manually turned off at night.

A front view of the site along with both heat pump and boiler systems are shown below.

Figure 3: Front view of site 2

Figure 4: Boiler system for site 2 with hydronic piping attached

Figure 5:Heat pump for site 2 with temperature sensor attached

Site 3

This site is in Bethpage, NY. The house is a split-level house with two heating zones for the Burnham V-84 boiler installed. This boiler has a tankless coil setup for domestic hot water. The indoor unit for the mini-split system is a Klimaire KWM09-H2 and the outdoor unit is a Midea M30C-HRDN1-M. The specifications are detailed in Table 3.

Table 3: Site 3, Unit 2 Specifications

Manufacturer	Klimaire (indoor) & Midea (outdoor)
Outdoor Unit Model Number	M30C-27HRDN1-M
Indoor Unit Model Number	KWM09-H2
Nominal Heating Capacity	10,000 Btu/hr
Nominal Cooling Capacity	25,400 Btu/hr
HSPF	12.8
Cooling EER	12.5
Cooling SEER	24.6

The indoor unit is installed in one of the two bedrooms on the second floor. The other bedroom also has an indoor unit, but that was not analyzed. The boiler thermostat is kept at a constant 72°F, with no nighttime setback. The homeowner claims that the heat pump thermostat is set to 80°Fand is used to boost the temperature in the room whenever the occupant feels the heat from the hydronic system is insufficient. The switchover control strategy between the heat pump and hydronic systems for this site, then, is fully manual.

The heat pump and boiler systems are shown below.

Figure 6: Heat pump indoor unit for site 3 with temperature sensor attached

Figure 7: Boiler system for site 3

Site 4

This site is a split-level house with three heating zones with a domestic hot water coil for the B. Smith Series 8 boiler. The house is located in Mahopac, NY. The relevant zone in this study includes the living area and "den" zone which is also heated by a ductless heat pump system with matched Lennox indoor and outdoor units. The specifications of the mini-split system are shown below.

Table 4: Site 4, Unit 1 Specifications

Manufacturer	Lennox
Outdoor Unit Model Number	MH7-HO-18P1A
Indoor Unit Model Number	MH7-HI-18P1A
Nominal Heating Capacity	25,000 Btu/hr
Nominal Cooling Capacity	18,000 Btu/hr
HSPF	10.2
Cooling EER	12.0
Cooling SEER	18.0

The hydronic heating system thermostat for the zone was set to 67°F for the duration of this test, with no nighttime setback implemented. The heat pump was operated manually when the zone was occupied, typically afternoon and evenings. The homeowner prefers the zone to be "toasty for things like movie night," so the heat pump was set to 78°F or higher whenever it was being operated.

A front view of the site is shown below. Also shown is the heat pump in operating mode and the boiler system with hydronic piping.

Figure 8: Front view of site 4

Figure 9: Close-up view of heat pump for site 4 during operation with temperature sensor attached

Figure 10: Boiler system for site 4

Site 5

This site is a split-level house with one main heating zone. The house is in Feura Bush, NY. The boiler in this site is a Weil-McLain 366, which was not utilized for the duration of this test. Domestic hot water was provided by an electric water heater. The whole house was heated using multiple ductless mini-split systems, one of which was analyzed in this study. The indoor unit for this heat pump is in a small office space and the heat pump's thermostat for this space is set to 65–68°F, as reported by the homeowners. On some days, they turn off the heat pump, which acts as a nighttime setback setting. Occasionally, the heat pump is shut off during the night or when the homeowners are not in the house for more than a day. Both indoor and outdoor units are manufactured by Mitsubishi. The following table shows more detailed descriptions of the systems.

Table 5: Site 5, Unit 1 Specifications

Manufacturer	Mitsubishi
Outdoor Unit Model Number	MXZ-3C30NAHZ2
Indoor Unit Model Number	MSZ-GL09NA
Nominal Heating Capacity	28,600 Btu/hr
Nominal Cooling Capacity	28,400 Btu/hr
HSPF	11
Cooling EER	12.5
Cooling SEER	18

A front view of the site showing the location of the zone tested and the outdoor unit is below. Following that figure is a picture of the boiler system.

Figure 11: Front view of site 6 with outdoor unit

Figure 12: Boiler system for site 5

Site 6

This site is a ranch-style house located in Branchport, NY with 2 heating zones in the hydronic heating system and an indirect water heater, resulting in 3 zones in total. The first zone covers the sole bedroom and the attached bathroom while the second covers the rest of the house, which includes a living room and dining room. While both zones also have a mini-split system, the second zone was chosen for this test. The boiler is a three-series H.B. Smith Series 8. Both indoor and outdoor units for the mini-split system are manufactured by Mitsubishi. More details are provided in Table 6 below.

Table 6: Site 6, Unit 2 Specifications

Manufacturer	Mitsubishi
Outdoor Unit Model Number	MUZ-FH18NA
Indoor Unit Model Number	MSZ-FH18NA
Nominal Heating Capacity	20,200 Btu/hr
Nominal Cooling Capacity	17,200 Btu/hr
HSPF	12
Cooling EER	12.5
Cooling SEER	21

The homeowners stated that they did not use the heat pump system in the winter after the outdoor temperature decreased to 25–35°F. They may wait to begin the use of this manual control method until mid- to late-December, or even January. In late winter/spring after the homeowners revert to use the heat pump. Outside of this cold season the heat pump and hydronic systems are used with both thermostats set for 73°F. While the hydronic system remained on at all times, the heat pump was manually turned on only when one of the two occupants of the home was in zone used in the study. Once turned on, the heat pump would sometimes be left running overnight while on other days it would be turned off before nightfall.

A side view of the sight and where the zone for this test is located is shown below. Also shown is the boiler system along with hydronic piping to all zones.

Figure 13: Side view of site 6 showing location of zone chosen for this test

Figure 14: Boiler system for site 6

Experimental

At each of the test sites the number, location, manufacturer, and model numbers for the heat pumps were documented. Also documented were the type, manufacturer, model number, and zone structure for the hydronic heating system that was in place prior to the installation of the heat pump and that operates in conjunction with the heat pump. The NORA team had discussions with the homeowners to understand how they use the heat pump and what approach they take to transitioning from the heat pump to the hydronic heating system when needed. Also, information was collected on temperature setback practice in the space.

Temperature sensors were installed at each site and Table 1 provides a list of measurement points as planned.

Table 7: Field Test Measurement Points

Point	Sensor Location
1	Heat pump delivered air temperature
2	Ambient zone temperature
3	Hydronic supply to zone with the heat pump
4	Hydronic return from zone with the heat pump
5	Boiler flue gas temperature
6	Outside heat exchanger inter-fin temperature

Measurement Point 1 provides information that shows the time at which the heat pump was operating. Point 2 provides general information about the ability of the heat pump to meet the space heating load and, possibly the state of temperature setback in the space. Points 3 and 4 were planned to determine times at which the hydronic heating system was delivering heat to the space. Point 5 shows when the boiler is firing. Point 6 was installed with an interest in determining when the system is in defrost mode.

Details of the installation at each house were somewhat different, and these differences are discussed in the results section of this report. For most of the sites, Points 1 and 2 were measured using ModelUX100-001 loggers from Onset Computer Corporation. They are tiny, battery-powered loggersand a 1-minute recording interval was used.

Points 3–6 were measured using model UX100-014M loggers from Onset Computer Corporation with a 10 second measurement interval. For sites where winter temperatures commonly drop below 0°F (i.e., sites 5 and 6), weatherproof loggers from Omega, model OM- CP-ETR101A, were used for measurement point 6 only. For all the sites it was necessary to visitto download the data.

In addition to these measurement points, ambient outside weather data was obtained from the National Climate Data Center. This data is based on local airport measurements. Sites with the Omega temperature loggers for point 6 did not require local airport temperatures because the loggers measured outdoor ambient temperatures.

Results

Site 1

The outside ambient temperatures from January 30, 2019 to March 23, 2019 were analyzed to determine the appropriate period for a detailed analysis, shown below in Figure 15. The first of three periods was the coldest, which lasted from January 31 to February 2 of 2019. In this period, the lowest recorded temperature was approximately 3°F. A second time period that consisted of average winter temperatures for the region lasted from February 5 to 8 of 2019, where temperatures were at or higher than 32°F and reaching upwards of 40°F at times. The third period was March 14 to 17 of 2019. This period oversaw higher than average temperatures, even reaching over 50°F. The periods and the temperatures during them are circled in the figure below.

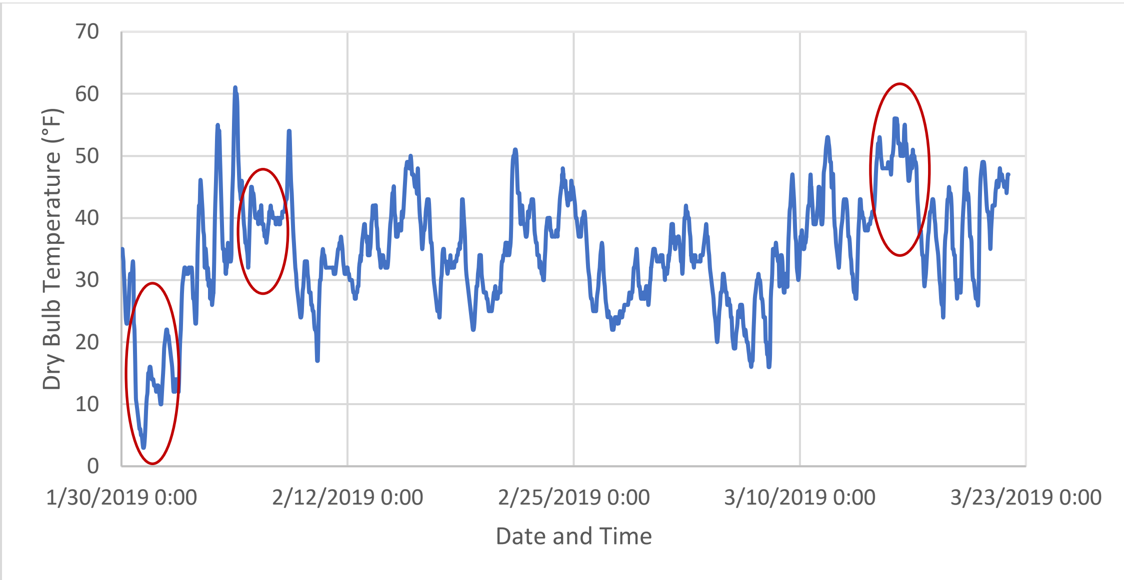

Figure 15: Outside ambient temperatures over test period for site 1

Site 1 — Period 1 Temperature Plots

Figures 16–19, below, provide plots of ambient zone (room air), hydronic supply water to the zone, heat pump delivered air, and outdoor unit fin temperatures for this selected time period.

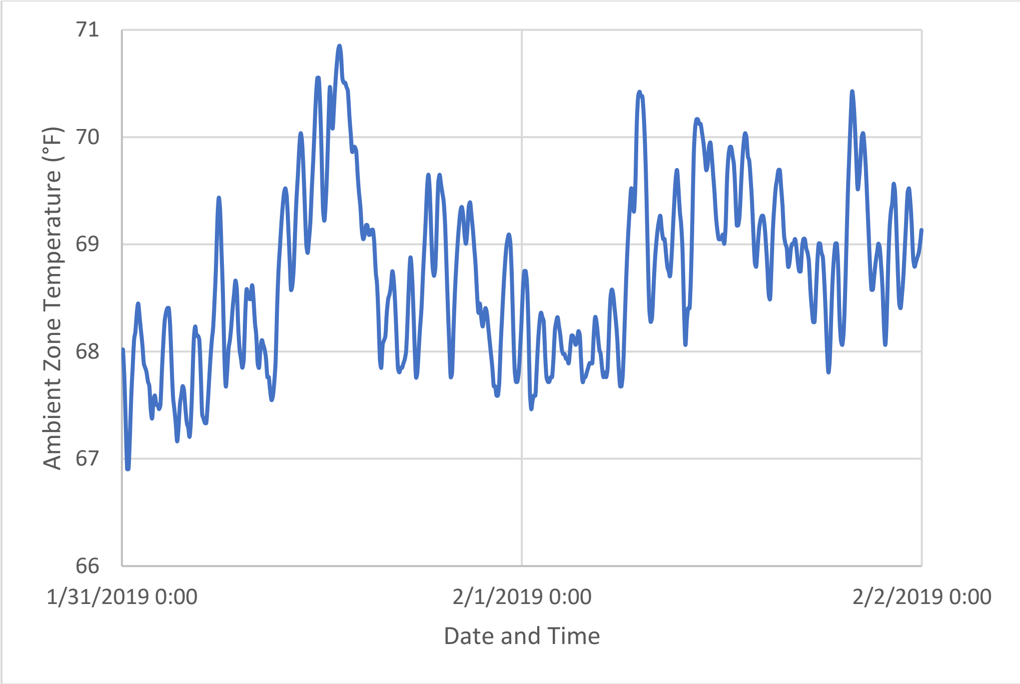

Figure 16: Ambient zone temperature for period 1 in site 1

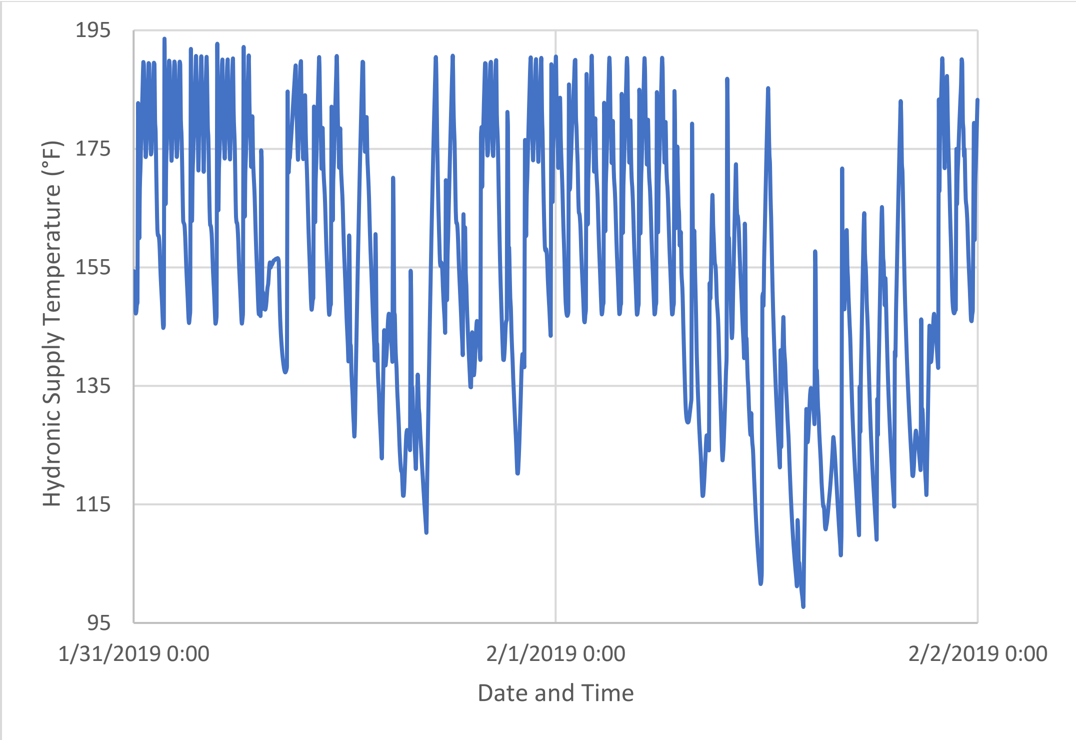

Figure 17: Hydronic supply temperature for period 1 in site 1

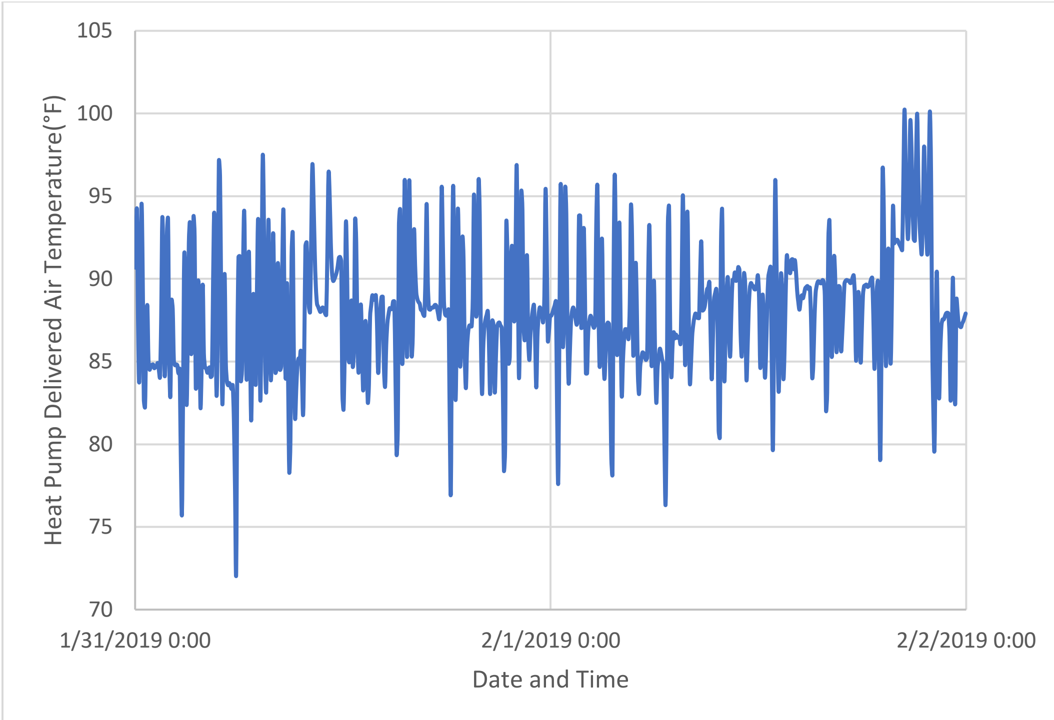

Figure 18: Heat pump delivered air temperature for period 1 in site 1

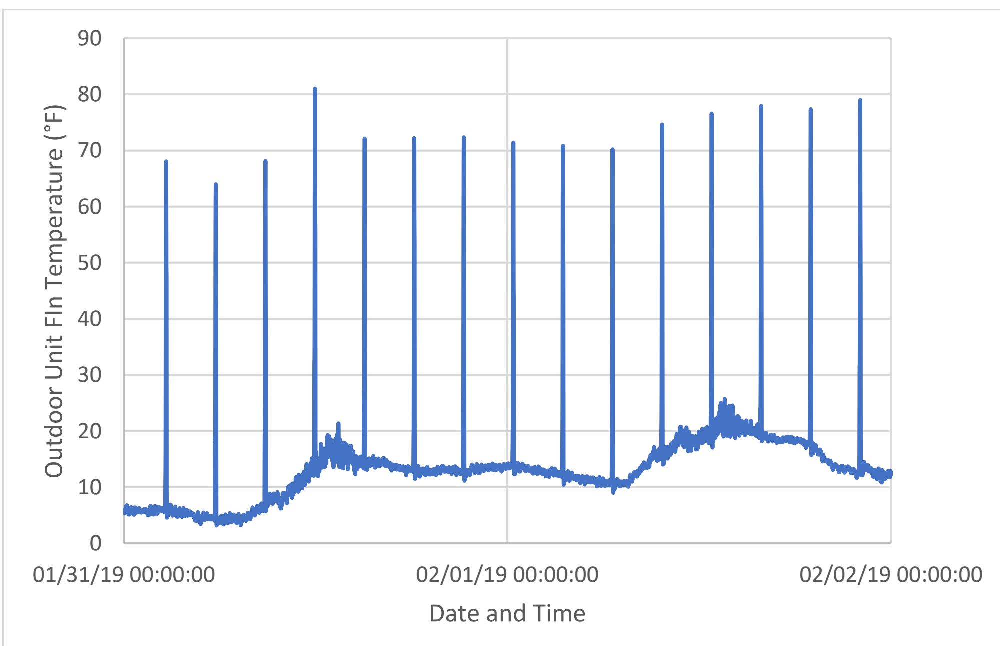

Figure 19: Outdoor unit fin temperature for period 1 in site 1

Site 1 — Period 1 Discussion

During this period, the outdoor temperature ranged from 3°F to 22°F, making it the coldest period examined for site 1. The ambient zone (room) temperatures in Figure 16 show that the heating systems in the house were able to keep up with the demand despite the cold weather in period 1. This is evident from the range of temperatures close to or greater than the daytime setpoint of 68°F.

Both the hydronic and heat pump systems simultaneously provided heat to the zone during this period. In Figure 17, any spike of temperature over 150°F suggests a heat call from the thermostat, showing that there was a hydronic demand at most times in this period. The heat pump, on the other hand, provided heat almost the entire period. This is observed in Figure 18 from always delivered air temperatures at or above 85°F, barring a few instances. The operation of both heating systems at once indicates that in this period, the homeowner had manually changed the settings to allow the hydronic system to provide heat. An important addition to this data is the condenser temperature. Usually, the condenser operates below outside ambient temperatures. However, it reverses the heat pump cycle when ice or frost builds up on its fins, increasing its temperature. This is known as a defrost cycle and can be seen in Figure 19 where condenser temperatures very often reach well over 50°F. Manufacturers use different control strategies for defrost. Some are simply time-based when the outdoor temperature is likely to lead to frosting of the condenser coils. Others include measures of more parameters and evaluation of heat pump performance. In this case, there are regular defrost cycles every 2.5–3 hours, which last about 4 minutes each. The presence of these cycles corresponds with the heat pump delivered air temperature dropping closer to room temperature, indicating the air supply had shut off for a prolonged period.

Site 1 — Period 2 Temperature Plots

Figures 20–24 show ambient zone, hydronic supply, heat pump delivered and outdoor unit fin temperatures for period 2 along with sub-plots showing temperature behavior over shorter periods of time for the heat pump delivered air temperature.

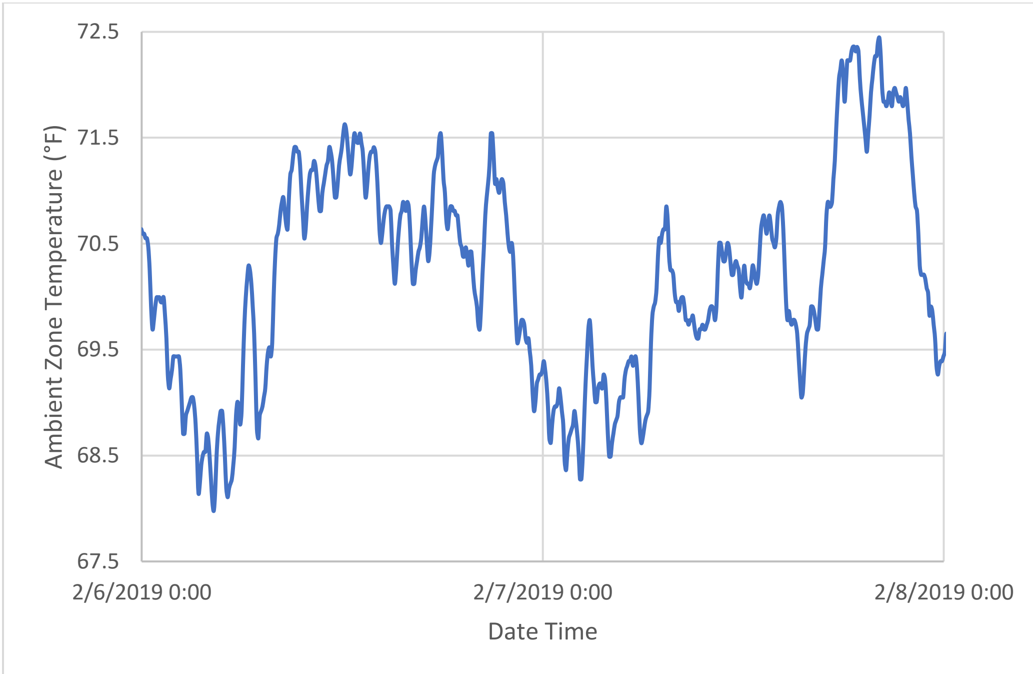

Figure 20: Ambient zone temperature for period 2 in site 1

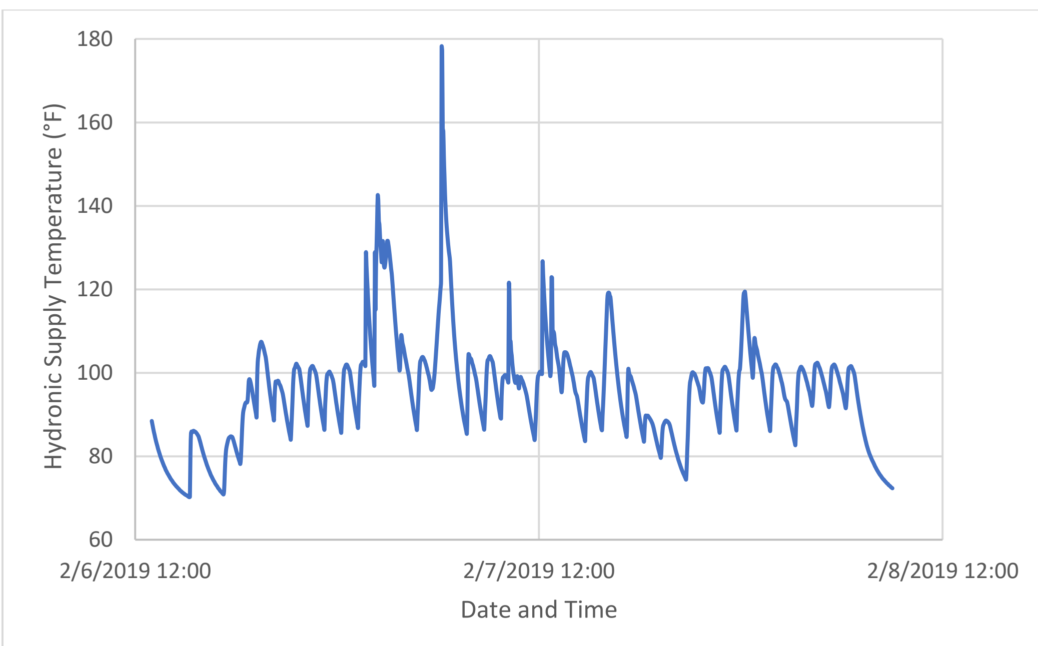

Figure 21: Hydronic supply temperature for period 2 in site 1

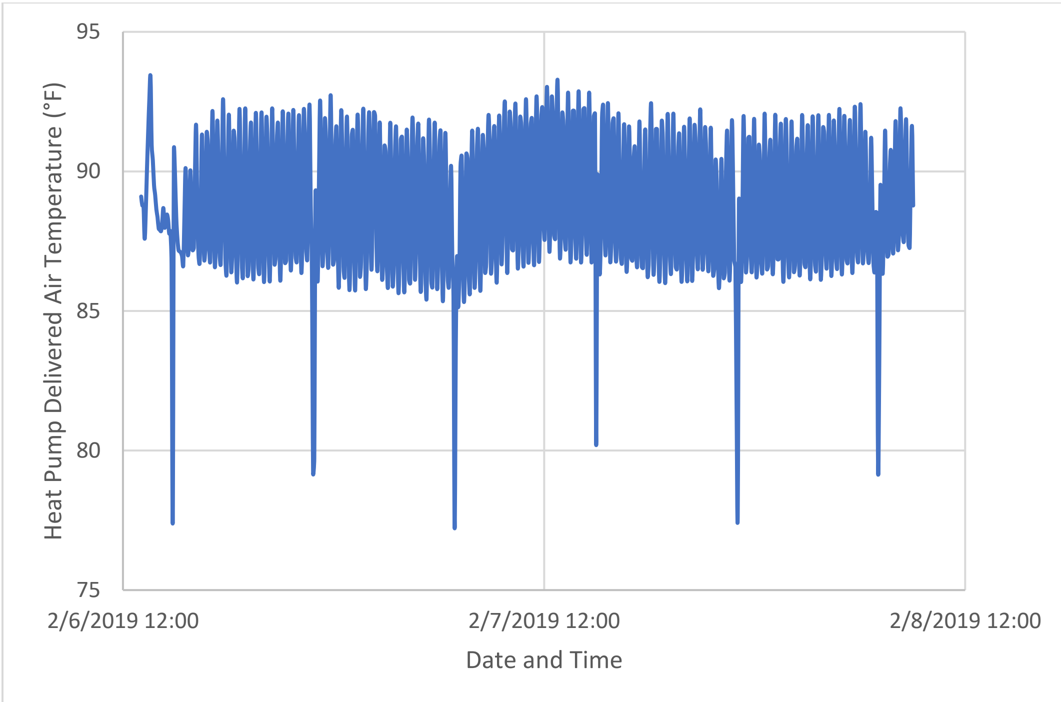

Figure 22: Heat pump delivered air temperature for period 2 in site 1

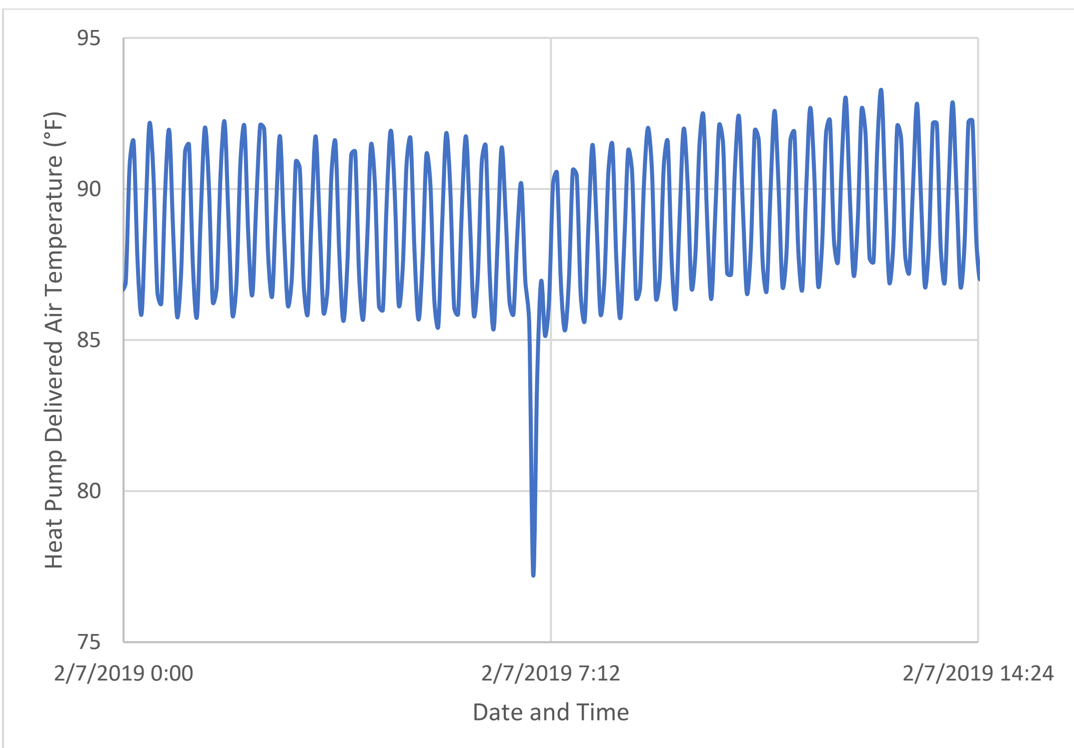

Figure 23: Sub-plot of heat pump delivered air temperatures for period 2 in site 1

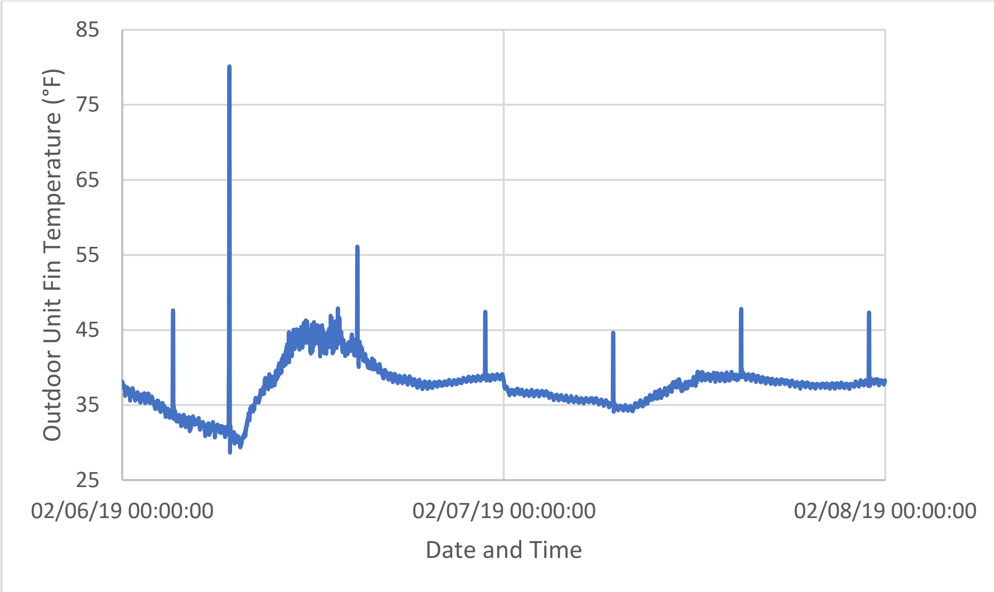

Figure 24: Condenser temperature for period 2 in site 1

Site 1 — Period 2 Discussion

This is a milder period, which represents average winter weather for the region. temperatures were close to 32°F at the beginning of the period but rose to, and generally remained very close to 40°F.

The hydronic system in this period provided heat on two separate occasions. This is shown by the two spikes over 140°F in supply temperature in Figure 21. The heat pump, on the other hand, was providing heat nearly the whole time, as shown by average delivered air temperatures approaching 90°F as seen in Figure 22. Following each spike in temperature of delivered air, there is a drop. This pattern repeats continuously, indicating heavy cycling of the heat pump. The heat pump also delivered, on average, warmer air in this period compared to the previous period. The heat pump was the primary source of heating for this period. The outdoor unit did not defrost very often. But even when it did defrost, peak fin temperatures only remained within the 45°F to 55°F range, with one spike to 80°F, all of which lasted ~4 minutes. The defrost cycles during this period did not appear to correct a deficiency in operation or impact delivered air temperature. This is the result of higher outdoor temperatures and lower relative humidity.

Site 1 — Period 3 Temperature Plots

Figures 25–30 show ambient zone, hydronic supply, heat pump delivered and condenser temperatures for period 3 along with sub-plots showing temperature behavior over shorter periods of time when necessary.

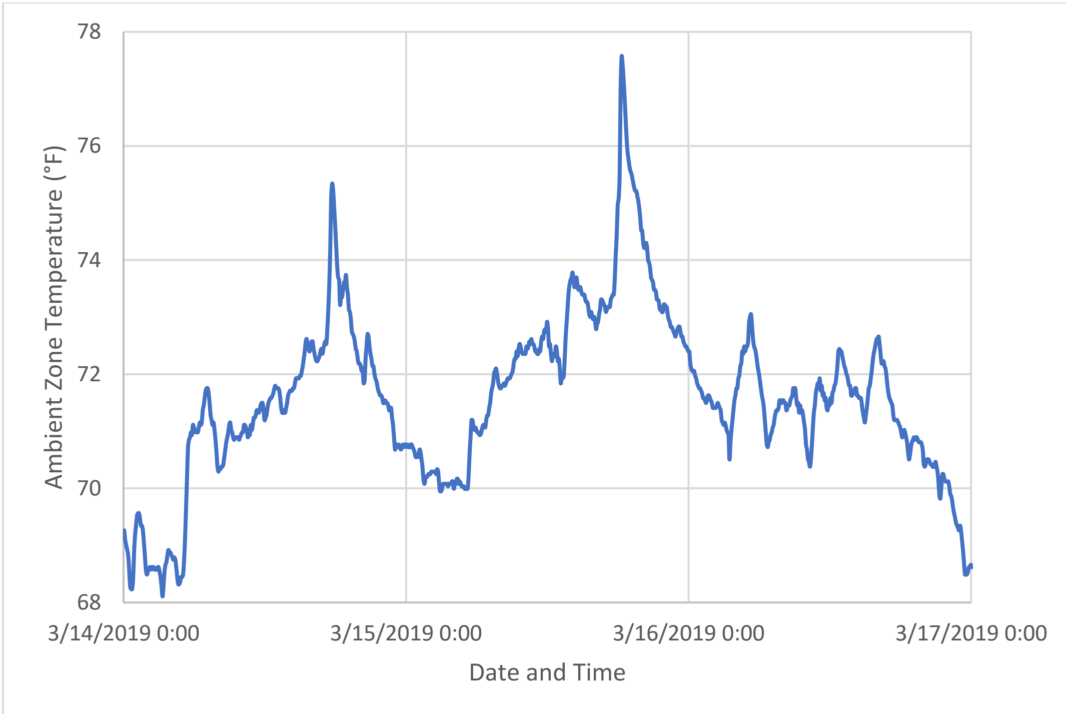

Figure 25: Ambient zone temperature for period 3 in site 1

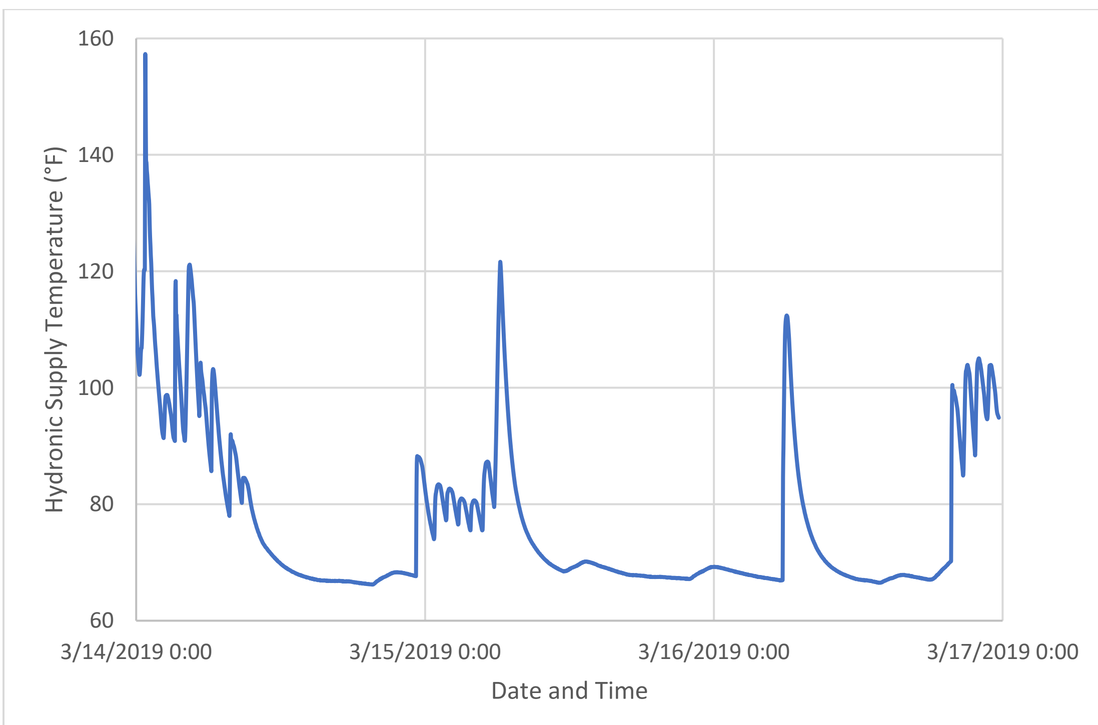

Figure 26: Hydronic supply temperature for period 3 in site 1

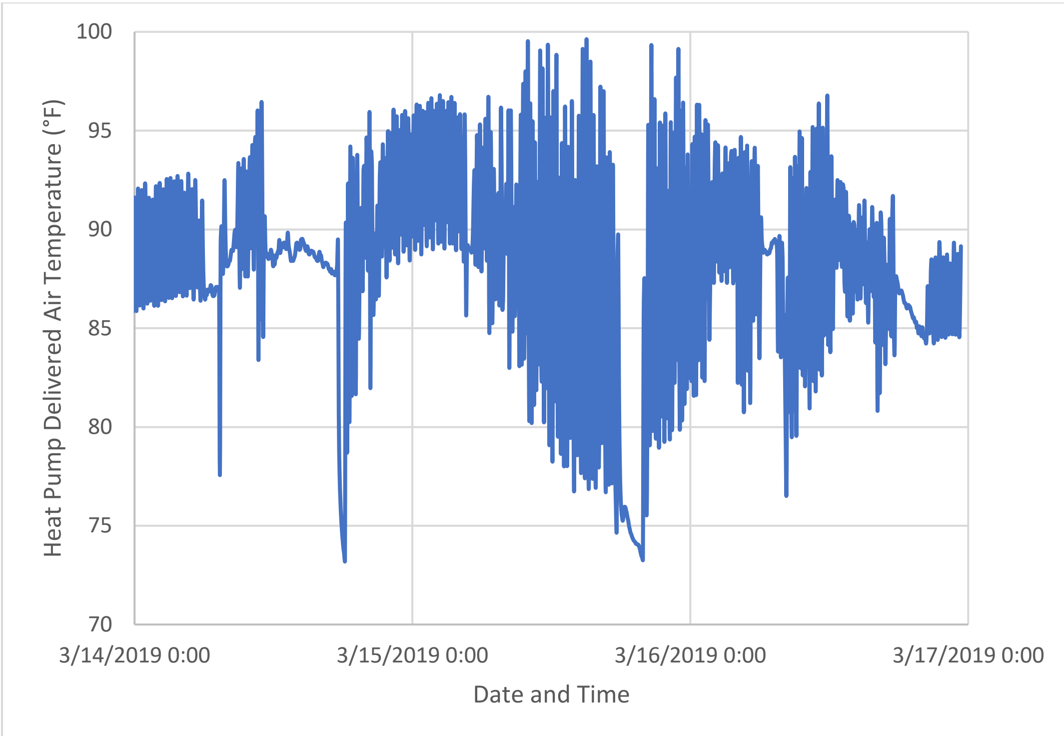

Figure 27: Heat pump delivered air temperature for period 3 in site 1

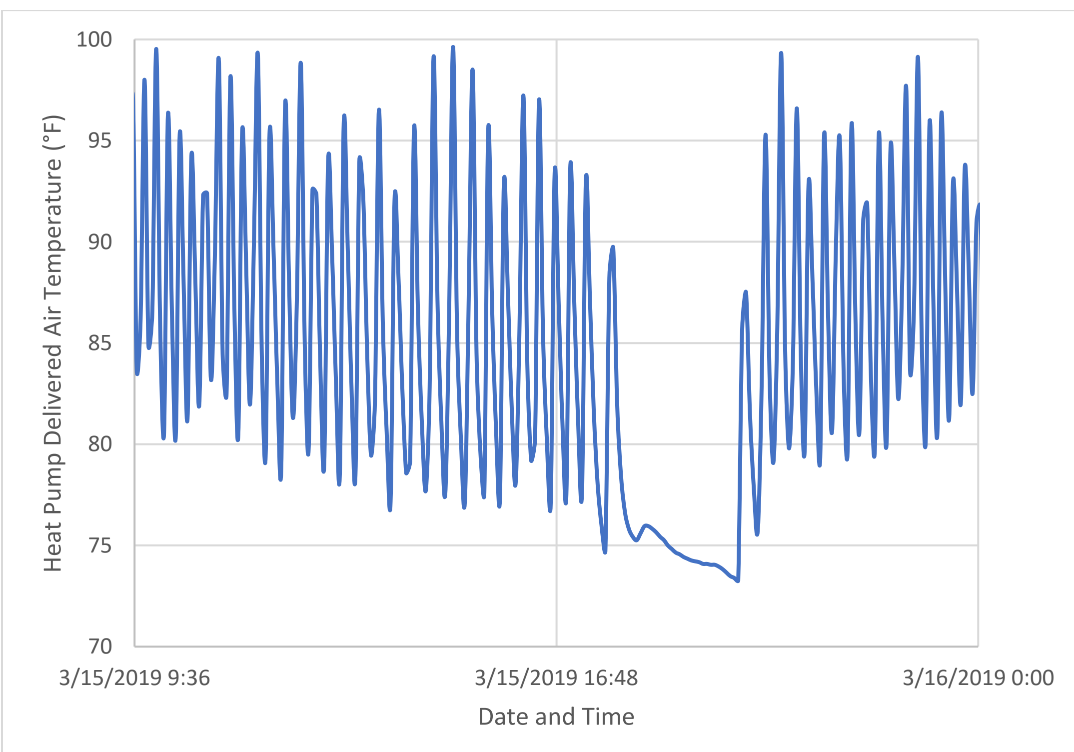

Figure 28: Sub-plot of heat pump delivered air temperatures for period 3 in site 1

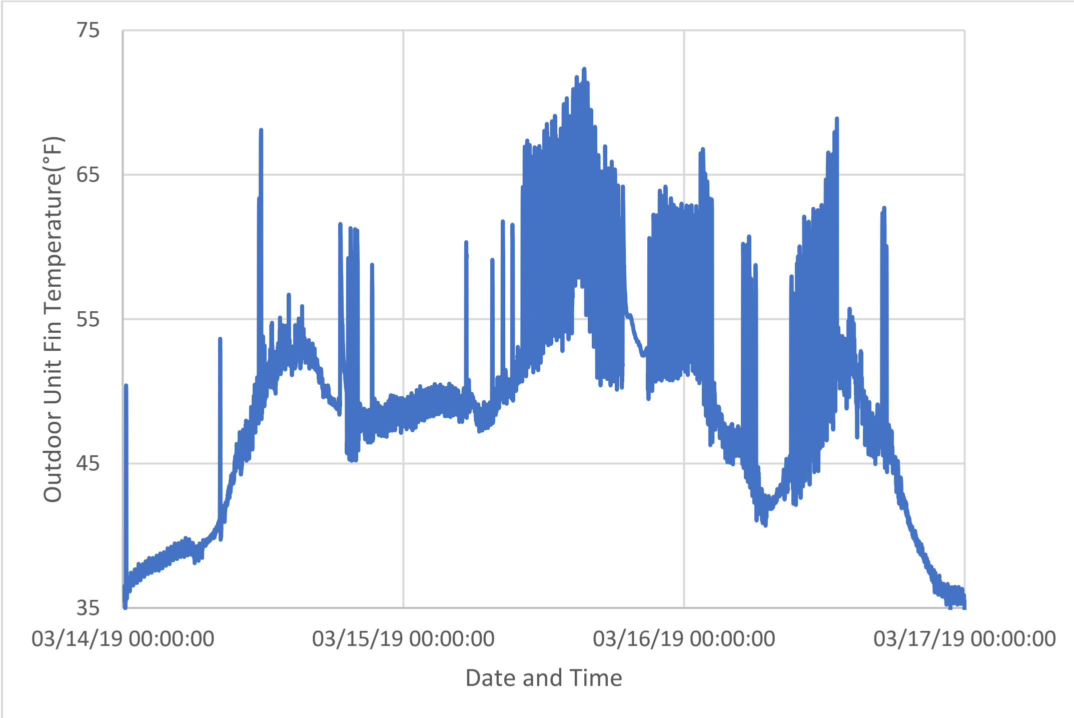

Figure 29: Condenser temperature for period 3 in site 1

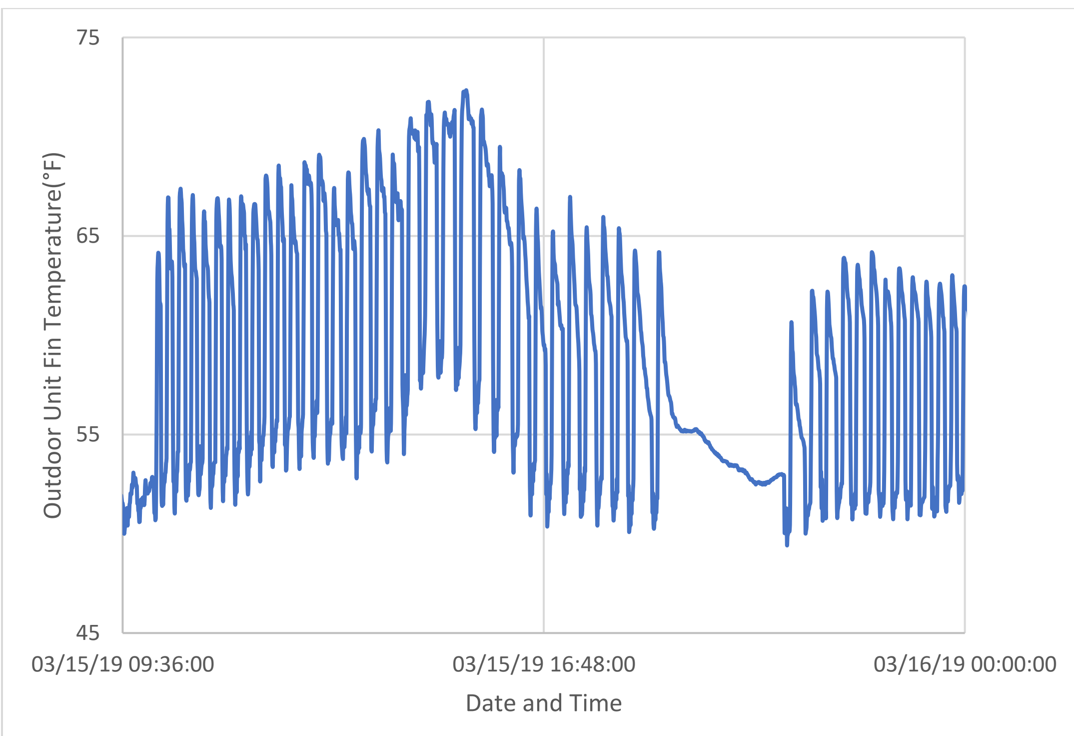

Figure 30: Sub-plot of condenser temperature for period 3 in site 1

Site 1 — Period 3 Discussion

This is the warmest period chosen for this site, with average outdoor temperatures of ~50°F. The zone temperature is also warmer than those of other periods, indicating that the thermostat settings may have been set higher in this period.

There were two cases during period 3 when the zone temperature rose to well above 70°F. Both were at about the same time in the evening and during these times, the delivered hydronic temperature was low indicating only the heat pump provided heat to the zone. It seems likely that the occupant raised the heat pump thermostat setting, leading to this temperature spike. After the spike, on both days, the heat pump stopped operating, indicating the occupant probably changed the thermostat setting back to the original level.

The hydronic system did not provide heat in period 3 except on three occasions, as seen from the steep spikes in Figure 26. The heat pump delivered air temperatures as shown in Figure 27 indicate that it provided heat during the entire period. Despite this continuous and high load, the heat pump experienced heavy cycling as shown in the close-up view in Figure 28.

Also notable were the very frequent defrost cycles, as shown in Figures 29 and 30, even though the ambient outside temperatures were well above freezing. This is likely due to the high (80–90%) outdoor humidity. There was no precipitation during this time period.

Site 2

As noted above, this is a Cape Cod style home located in Bethpage, NY. The data selected for analysis for this site range from January 31 to April 1 of 2018. Figure 31 shows the ambient temperatures for that period. A relatively cold period was chosen for analysis from February

2 to 4 where temperatures reached a low of 12°F. A comparatively warmer period from 20th to 22nd February was chosen as the second period where the highest temperature was 63°F. A third, higher temperature time period was March 7 to March 9, and temperatures ranged from 30°F to 40°F. The three chosen periods are circled in Figure 31.

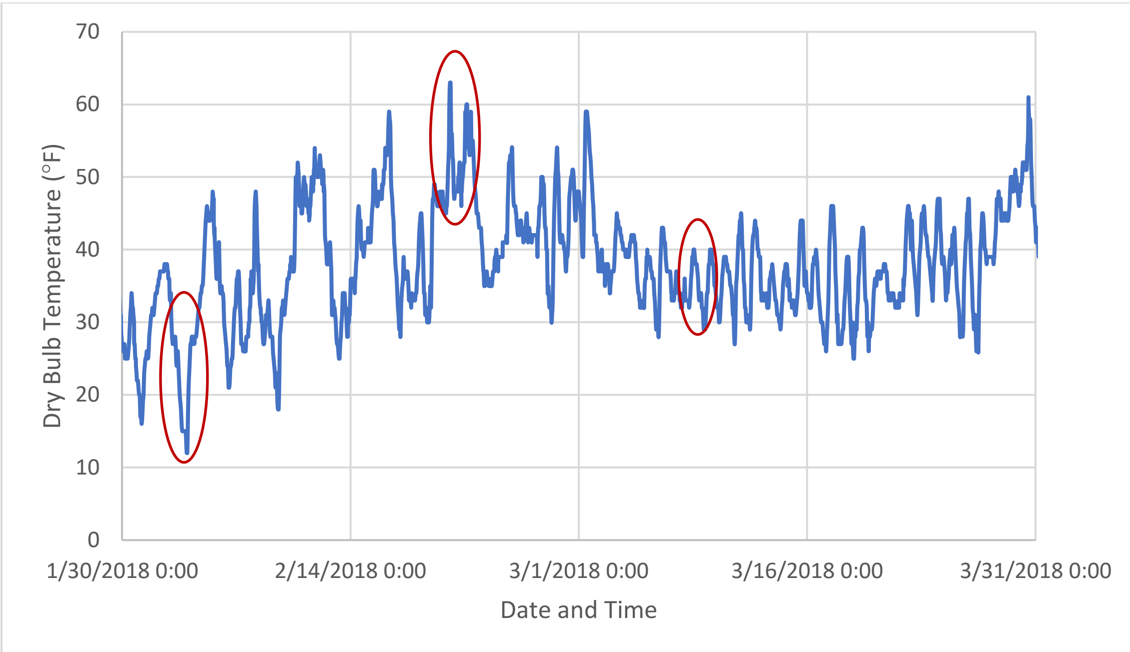

Figure 31: Outside ambient temperatures over test period for site 2

Site 2 — Period 1 Temperature Plots

Plots of heat pump delivered, ambient zone, hydronic supply and outdoor unit fin temperatures were made for the three selected periods. *Figures 32–35* below show those plots.

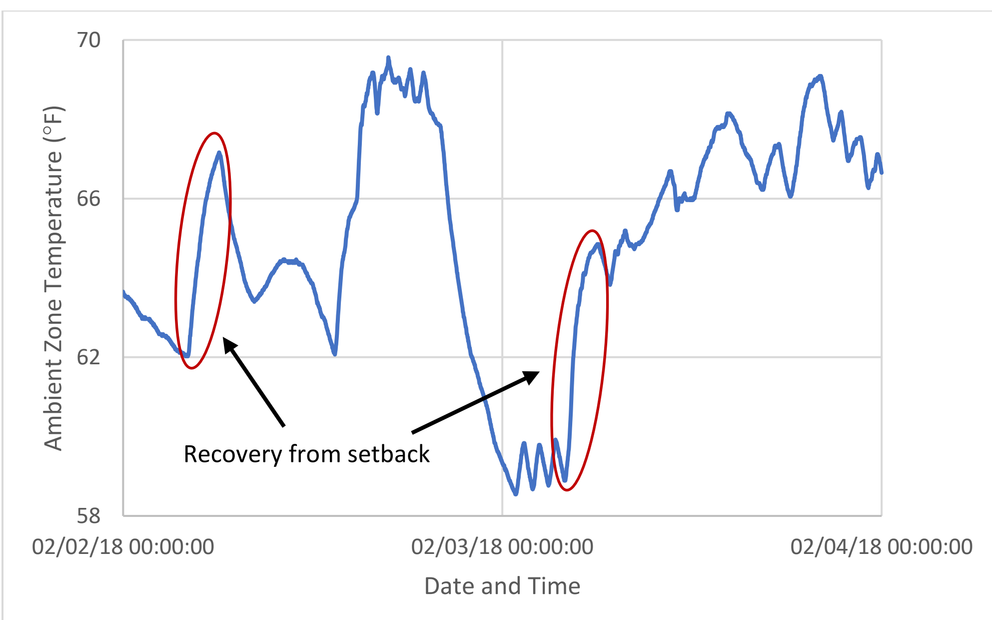

Figure 32: Ambient zone temperature for period 1 in site 2

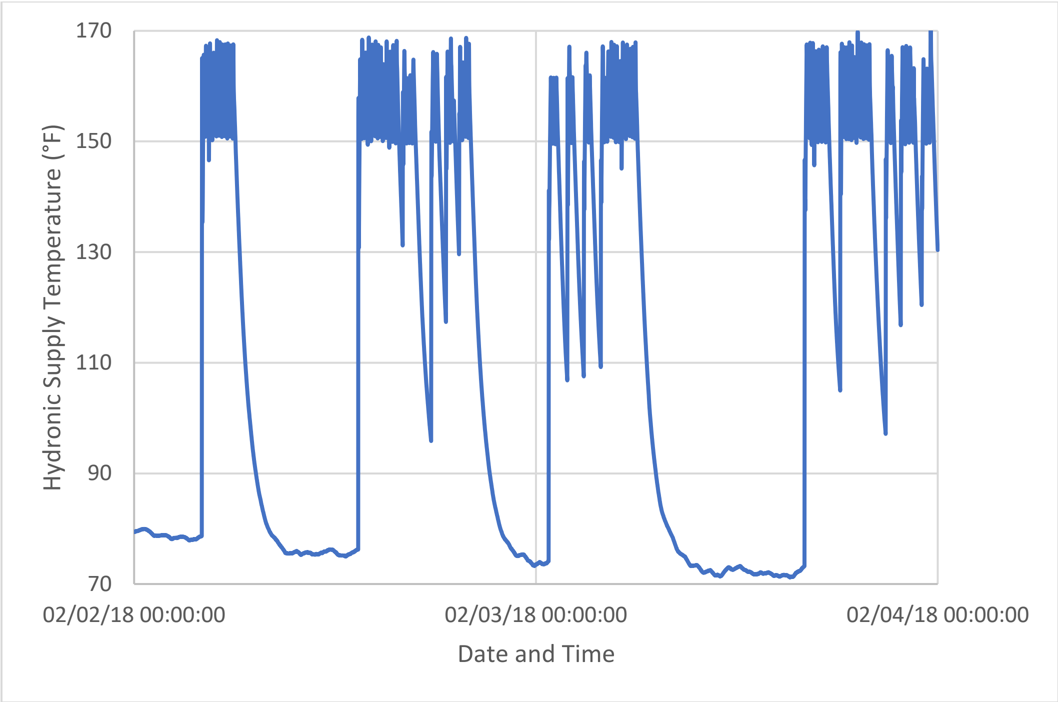

Figure 33: Hydronic supply temperature for period 1 in site 2

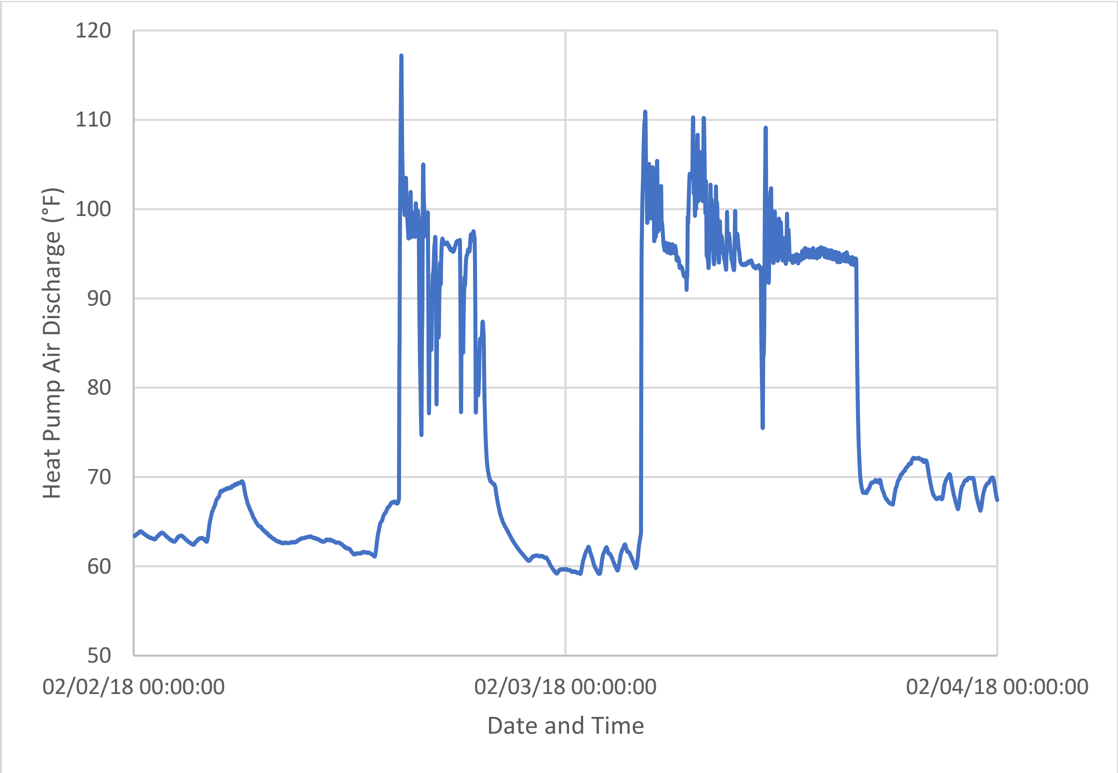

Figure 34: Heat pump delivered air temperature for period 1 in site 2

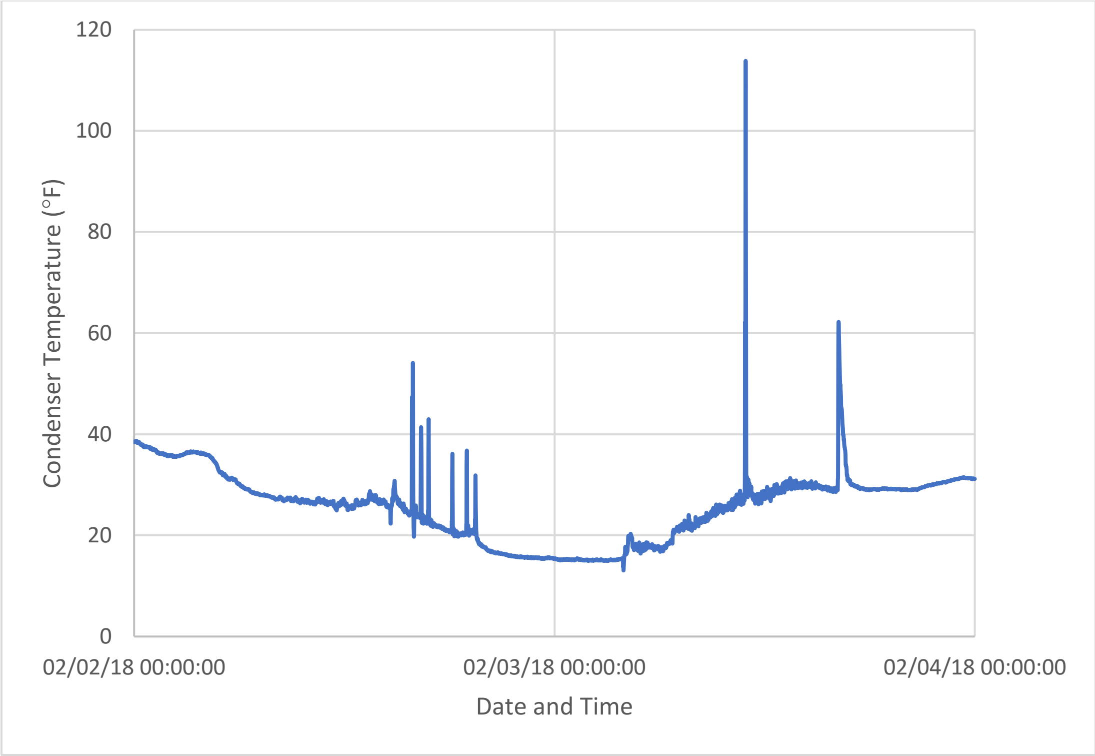

Figure 35: Outdoor unit fin temperature for period 1 in site 2

Site 2 — Period 1 Discussion

This is the coldest period of the three that were analyzed, where outside temperatures ranged between ~10°F and ~30°F. The ambient zone temperatures for this period are shown in *Figure 32*. The circled areas in that plot are the two instances of recovery from nighttime setback, each of which takes ~2 hours. Each of those recoveries were facilitated by the hydronic system, as indicated by the high hydronic supply temperatures shown in *Figure 33*. Elevated heat pump delivered air temperatures during those periods, seen in *Figure 34*, indicate that the heat pump assisted in the second of the two recoveries. At this site the heat pump is turned on and off manually by the homeowner. In this case the cold outdoor temperatures likely prompted the homeowner to enable operation of the heat pump. *Figure* 31 indicates that the ambient outside temperature for the second night in this period was as low as 12°F. *Figure 35* shows that the outdoor unit fin performed defrost cycles, which only occur 8 times during the entire two-day period, each lasting 2 to 3 minutes. This illustrates the intermittent and limited use of the heat pump by the homeowner during this time.

Site 2 — Period 2 Temperature Plots

Figures 36–39 show ambient zone, hydronic supply, heat pump delivered and outdoor unit fin temperatures for period 2 along with sub-plots showing temperature behavior over shorter periods of time when necessary.

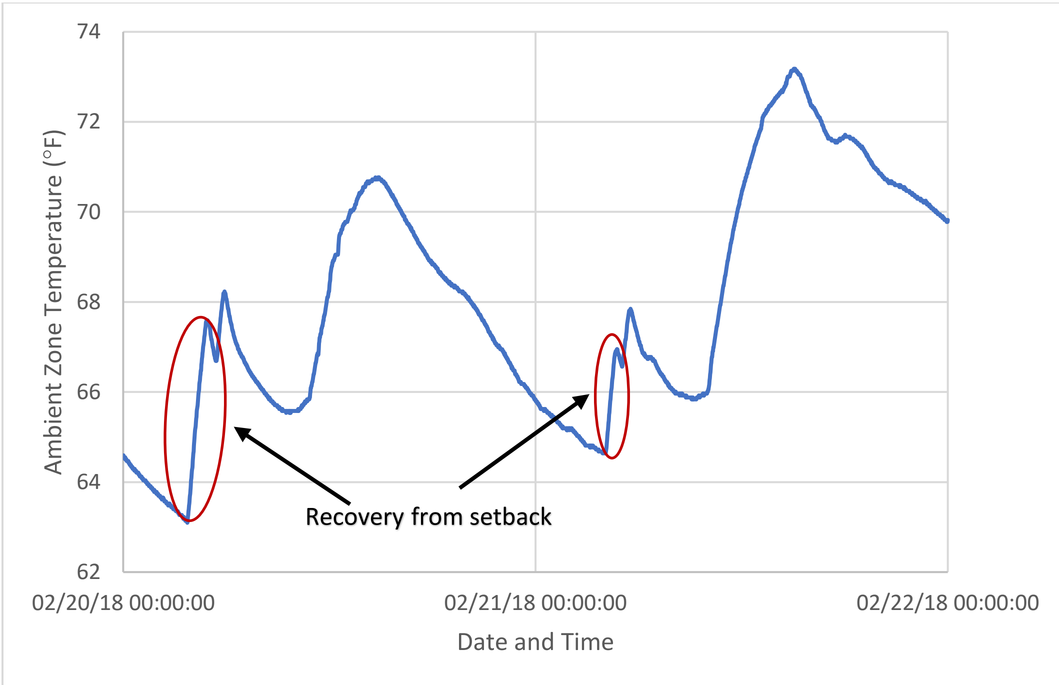

Figure 36: Ambient zone temperature for period 2 in site 2

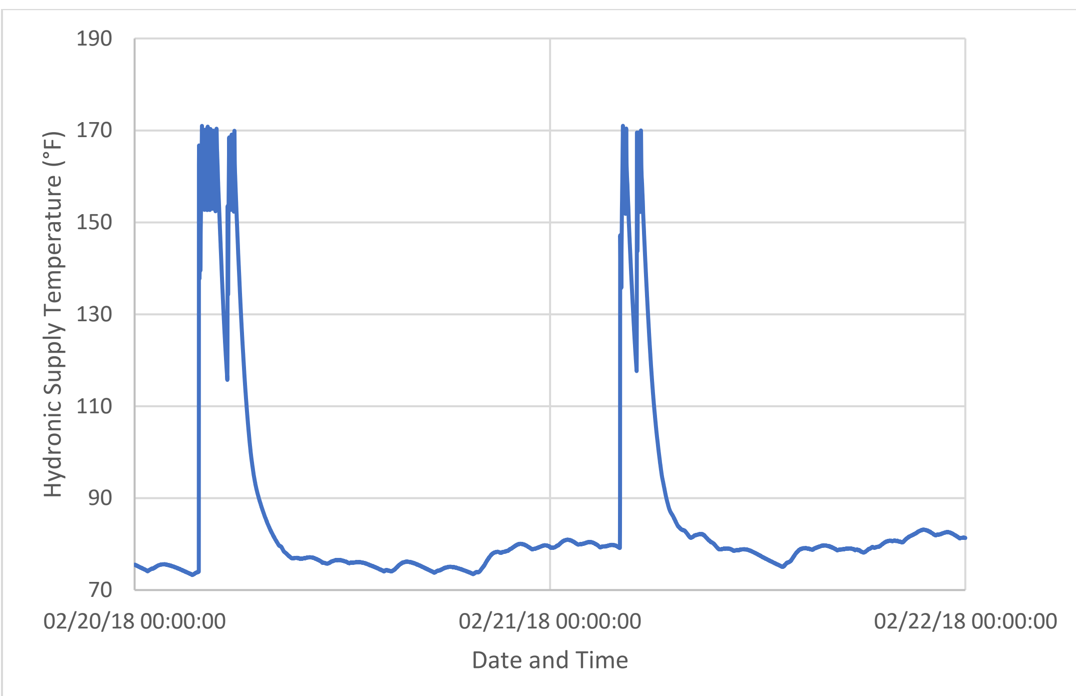

Figure 37: Hydronic supply temperature for period 2 in site 2

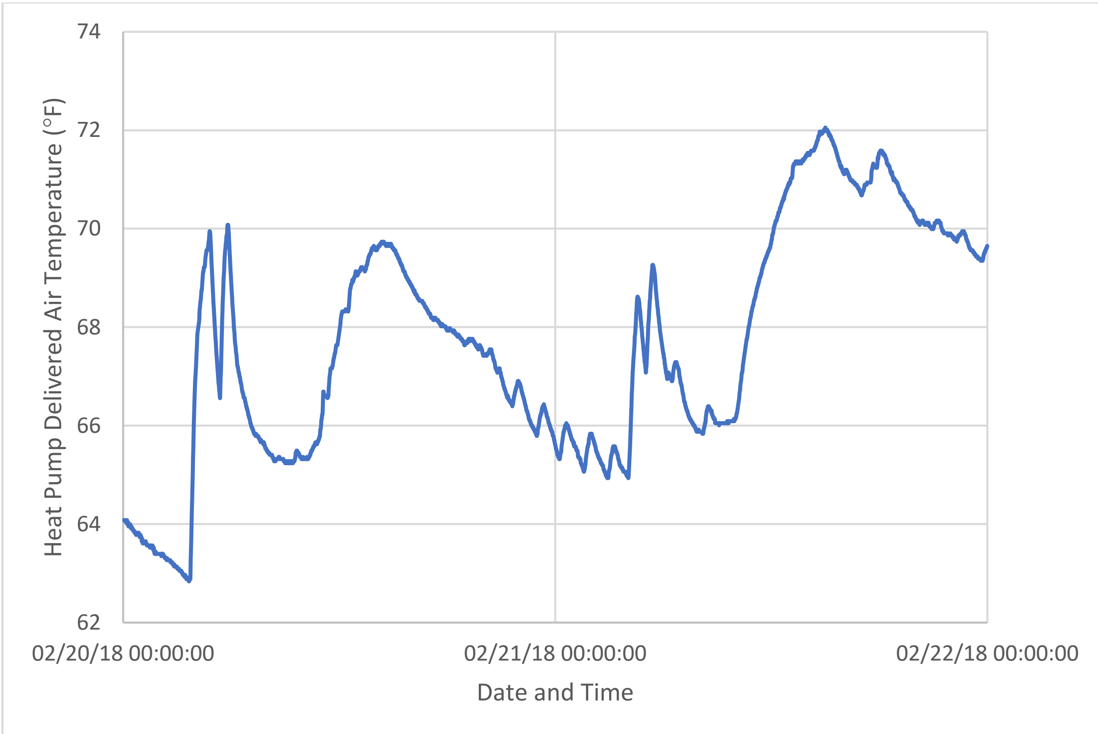

Figure 38: Heat pump delivered air temperature for period 2 in site 2

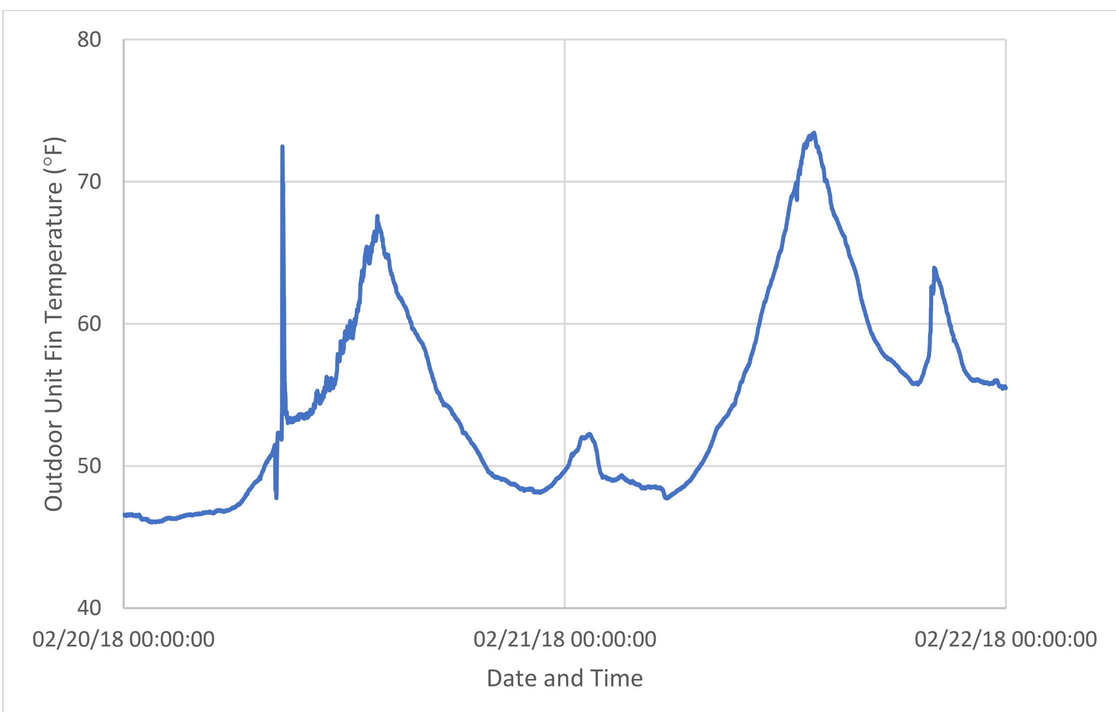

Figure 39: Outdoor unit fin temperature for period 2 in site 2

Site 2 — Period 2 Discussion

Outside ambient temperatures in this period ranged between 45°F and 62°F, which makes it the warmest of the three periods analyzed. The heat pump delivered air temperatures shown in *Figure 38* track very closely with the ambient zone temperatures in *Figure 36*, which indicates that the heat pump did not provide any heat during this period. The boiler, however, provided heat on two occasions, as seen from the temperature spikes in *Figure 37*. Both of those instances coincide with a recovery period from nighttime setback, as point out in *Figure 36*. The outdoor unit fin temperature in *Figure 39* indicates there was one defrost cycle despite the heat pump not operating during the period. This occurred due to recovery from a previous cycle before the period began.

Site 2 — Period 3 Temperature Plots

Figures 40–43 show ambient zone, hydronic supply, heat pump delivered and outdoor unit fin temperatures for period 3 along with sub-plots showing temperature behavior over shorter periods of time when necessary.

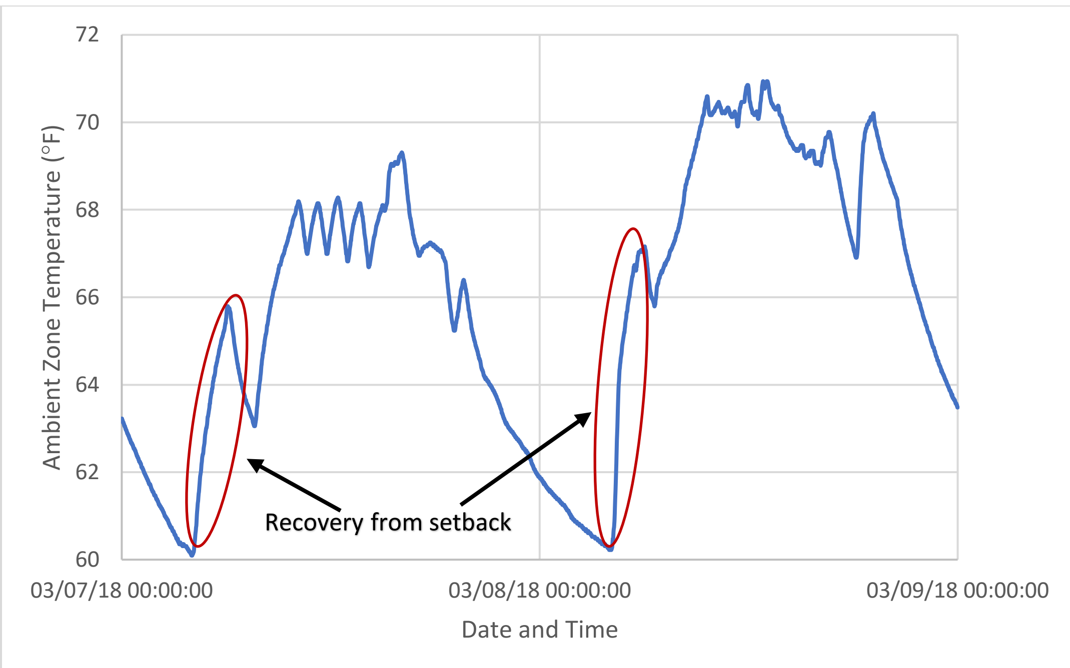

Figure 40: Ambient zone temperature for period 3 in site 2

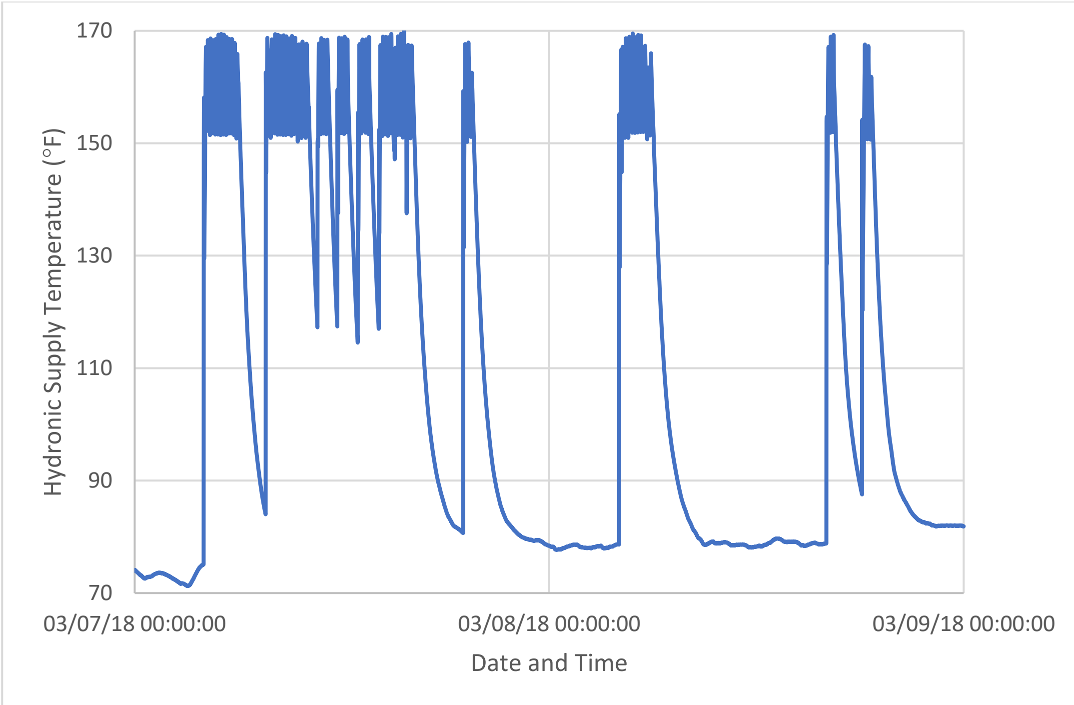

Figure 41: Hydronic supply temperature for period 3 in site 2

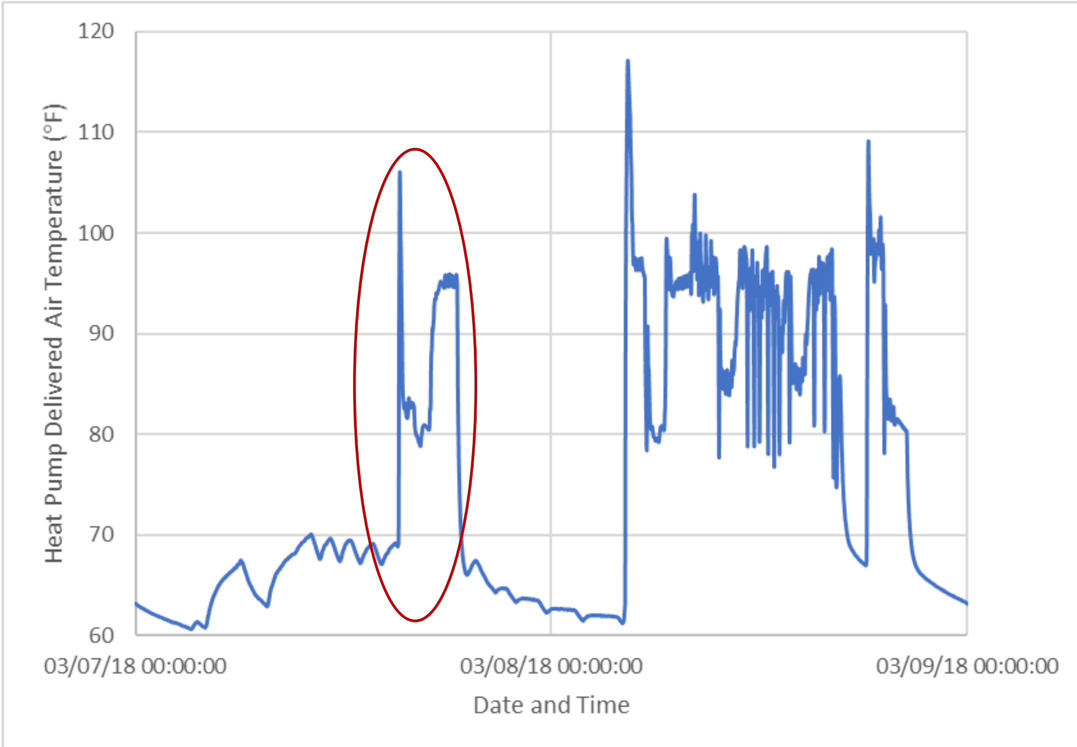

Figure 42: Heat pump delivered air temperature for period 3 in site 2

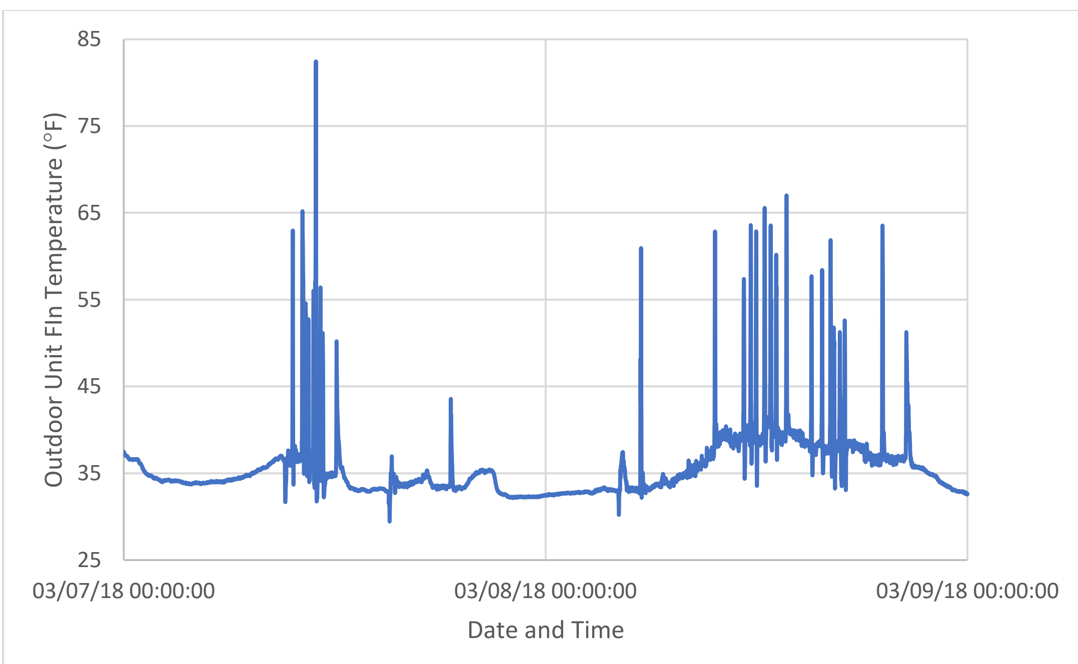

Figure 43: Outdoor unit fin temperature for period 3 in site 2

Site 2 — Period 3 Discussion

Outside ambient temperatures in this period range between ~30°F and ~40°F, which typical for winter in the Long Island region. Both heat pump and hydronic systems provided heat, as shown by the temperature spikes in Figures 41 and 42.

Recovery from nighttime setback occurs on both days in the period, as shown in Figure 40. The first recovery is facilitated by the hydronic system only, since only the hydronic system provides heat at the corresponding time. The second is performed by both systems, since there are temperature spikes showing operation during that period. The heat pump is manually turned on in this period to heat the space during other times of the day, when recovery is not taking place. The thermostat for the heat pump at those times is set to a temperature greater than that of the hydronic system. This is evident in Figure 40 where ambient zone temperatures are greater when the heat pump is operating (either on its own or with the hydronic system) compared to the zone temperature when the hydronic system is the sole source of heating. The outdoor unit fin temperatures in Figure 43 show the outdoor unit going through defrost cycles very often during a 16-hour period where the heat pump operated continuously apart from a short period in between. These defrost cycles were as close as 20 minutes apart in some cases.

Part of Figure 42 has been circled and shaded to illustrate an interesting operating condition observed in some of the sites. During this time period, the heat pump operates but, based on the outdoor unit fin temperatures as shown in Figure 43, it does not defrost. As also shown on these same plots, however, on the following day, the heat pump operated but defrosted often. Based on regional airport weather data for this site during this time period on March 7, the outdoor temperature averaged 32.5°F and ambient humidity was 100%. During the time period on March 8, the outdoor ambient temperature averaged 39°F and ambient humidity was 63%.

Site 3

The outside ambient temperatures from March 3, 2019 to March 23, 2019 are shown in Figure 44. The first of the three periods chosen for analysis is March 6 to March 8 of 2019, when the lowest temperature was 16°F, representative of colder than average winter weather for the region. The next is the mid-range analysis from March 12, 2019 to March 14, 2019, when temperatures ranged from 27°F to 43°F. The third period is March 15 to 17 of 2019, when temperatures reached 56°F, which is warmer than average for the winters in this region. The three chosen periods are shown in Figure 44.

Data collection was performed for this site over the winters of 2017/2018 and 2018/2019. However, various issues with instrumentation meant that only data in the period shown below could be analyzed.

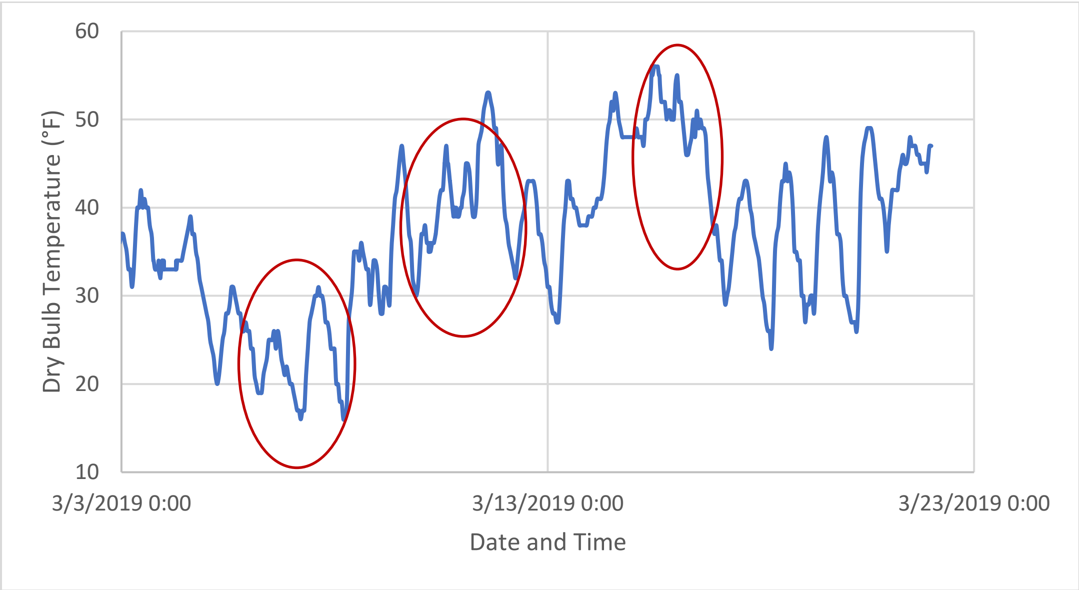

Figure 44: Outside ambient temperatures over test period for site 3

Site 3 — Period 1 Temperature Plots

Plots of heat pump delivered air, ambient zone, and hydronic return and temperatures were made for the three periods. Analysis of the data showed that the heat pump was not operated in any of those periods. Only the plots for period 1 are reported to demonstrate that the heat pump was not used in any of the three chosen periods. This is shown by the lack of temperature spikes in Figure 46 where the heat pump delivered air temperatures closely track the ambient temperatures in Figure 45. This is also true for the other two periods. After analyzing the data, more information was gathered from the homeowners to better understand why this happened. The homeowner stated that the boiler thermostat was incrementally increased throughout the day because the occupant, who is an elderly gentleman, felt cold. Thus, the hydronic system heated the room to the point where the heat pump thermostat was satisfied. It must be noted that the hydronic supply temperature data was not recorded for all the relevant time periods because of a logger malfunction. As an alternative, the hydronic return temperature was used to identify operation of the hydronic system. Figures 45–48 are for period 1.

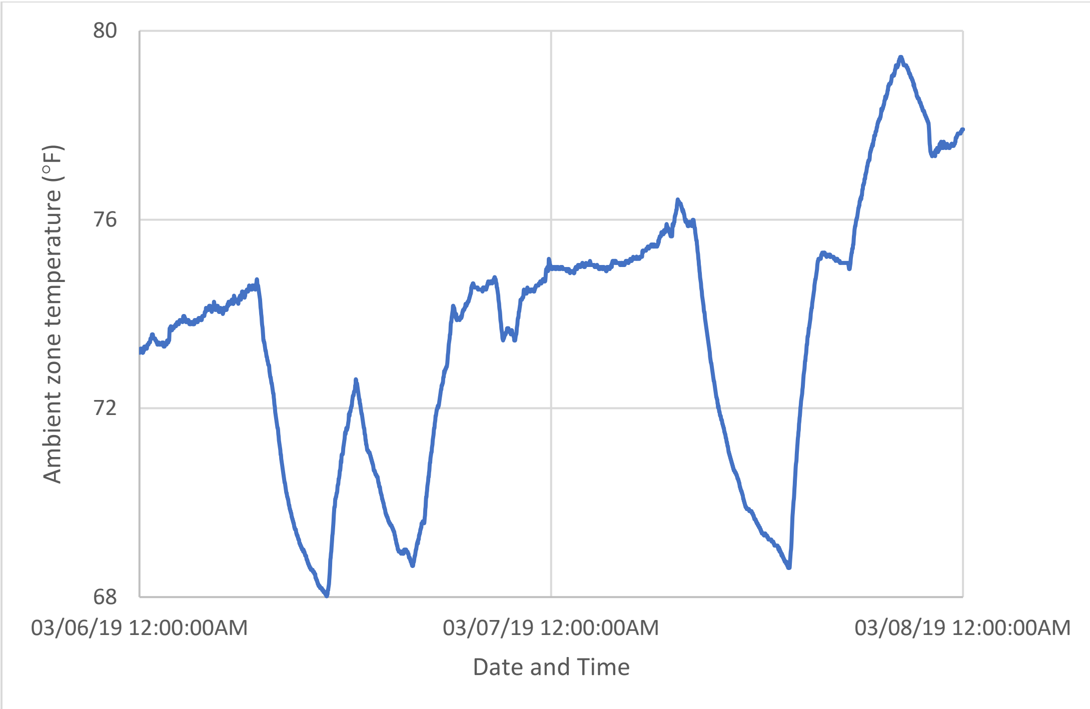

Figure 45: Ambient zone temperature for period 1 in site 3

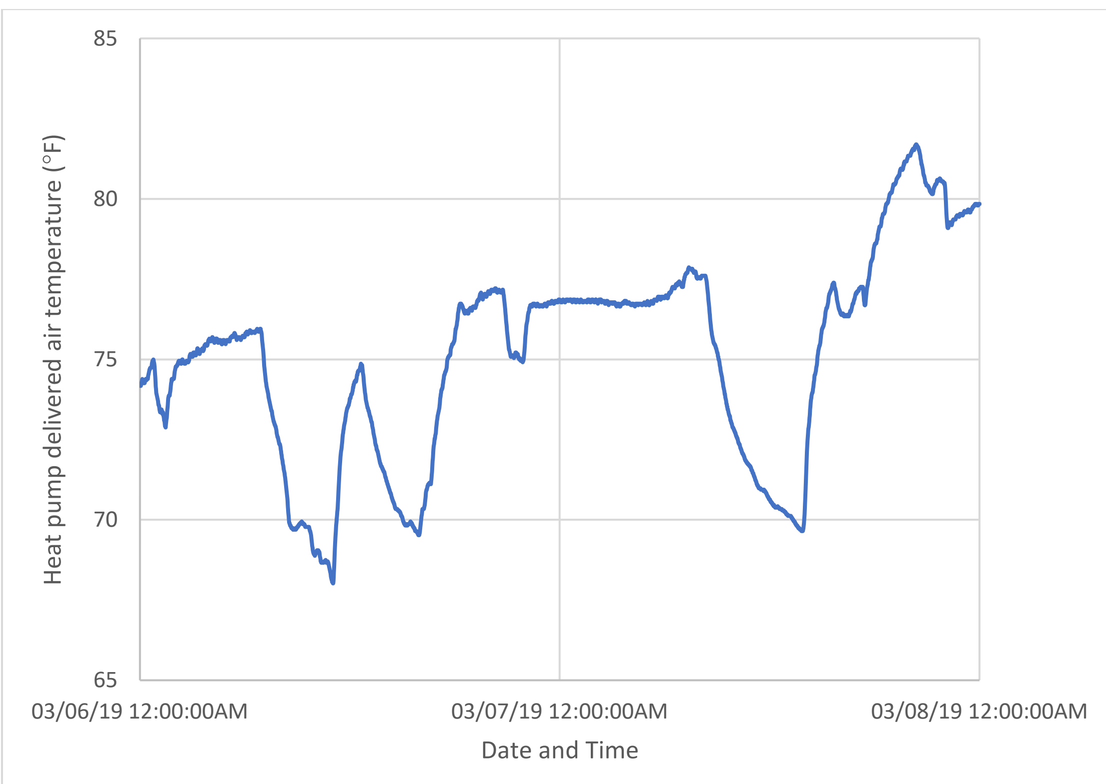

Figure 46: Heat pump delivered air temperature for period 1 in site 3

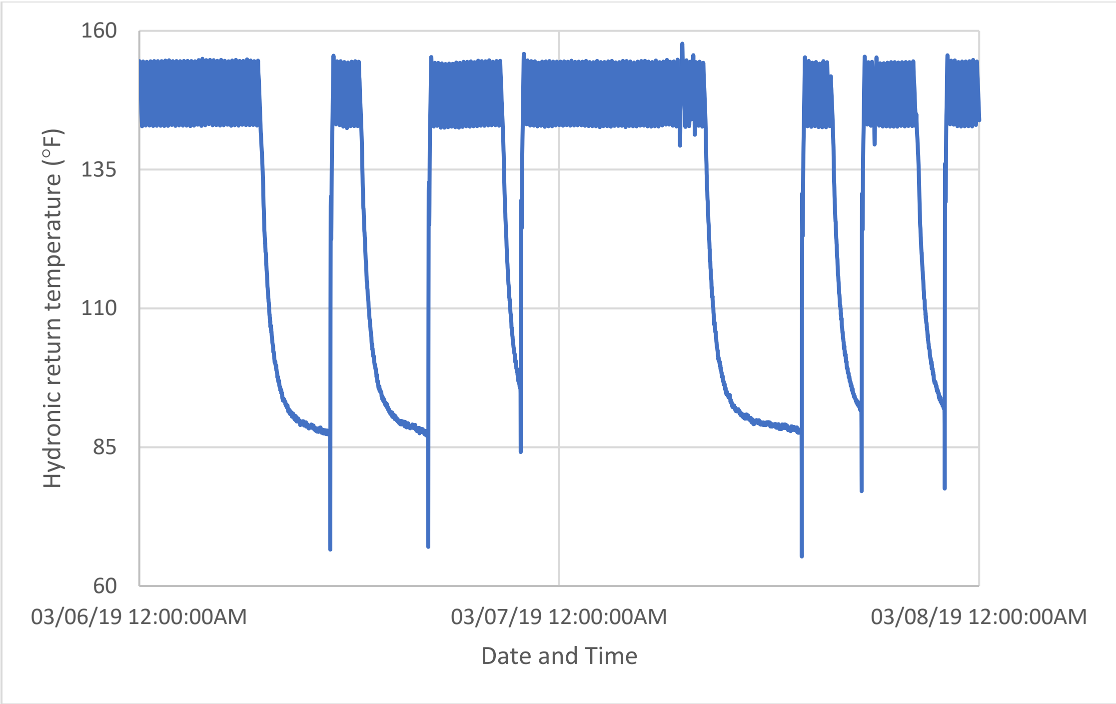

Figure 47: Hydronic return temperature for period 1 in site 3

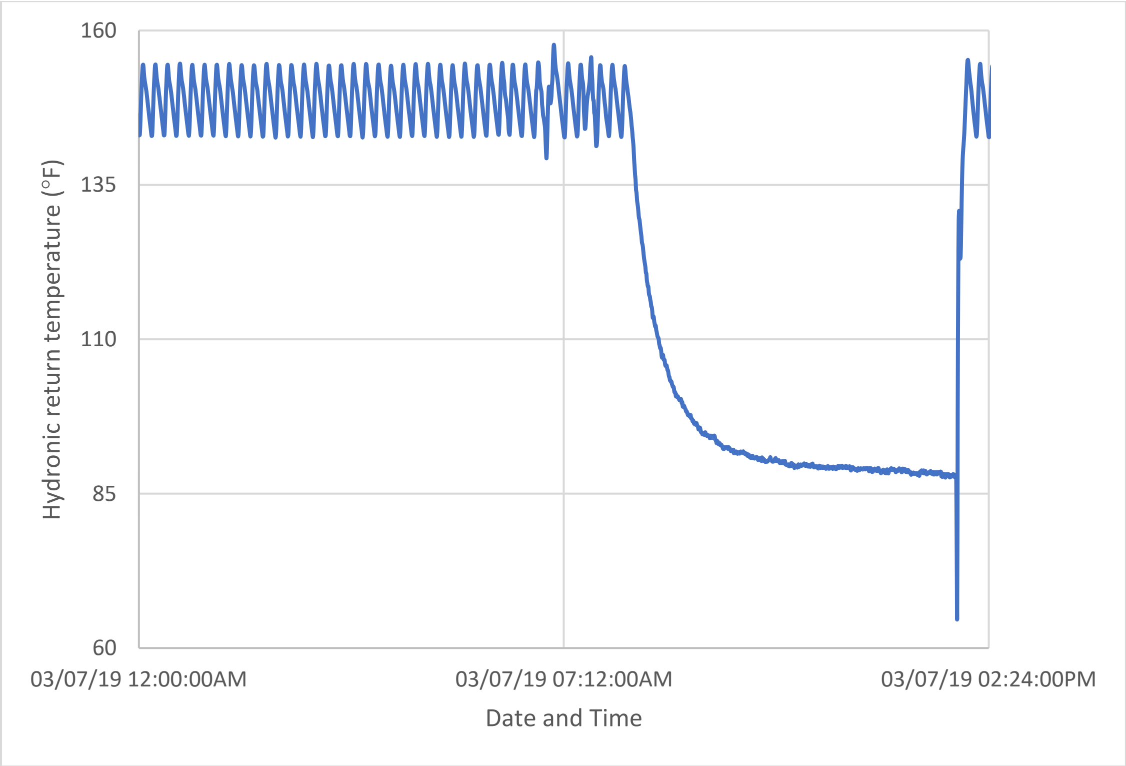

Figure 48: Sub-plot of hydronic return temperature for period 1 in site 3

Site 3 — Period 1 Discussion

The occupants did not utilize the heat pump in this, or any other period chosen for the analysis of this site, as mentioned before. The ambient zone temperatures in Figure 45 show temperature rises that occur when the hydronic system provides heat. Operation of the hydronic system are indicated by the spikes in hydronic return temperature in Figure 47. The "target" temperature based on the peak of each of those temperature rises is different every time the hydronic system provides heat. This means that the homeowners changed the thermostat settings manually, sometimes raising it as high as 78°F and as low as 68°F.

Site 3 — Period 4 Temperature Plots

Despite the heat pump not operating during the three selected periods for this site, there is one instance in the winter of 2018 when it did. The plots for the dataset for the period when the heat pump operated are shown below. This period includes data up to 12 hours before the heat pump cycles. The date range is from 2/23/2018 to 2/24/2018. Figures 49–52 below show the ambient zone, hydronic supply, and heat pump delivered air temperatures.

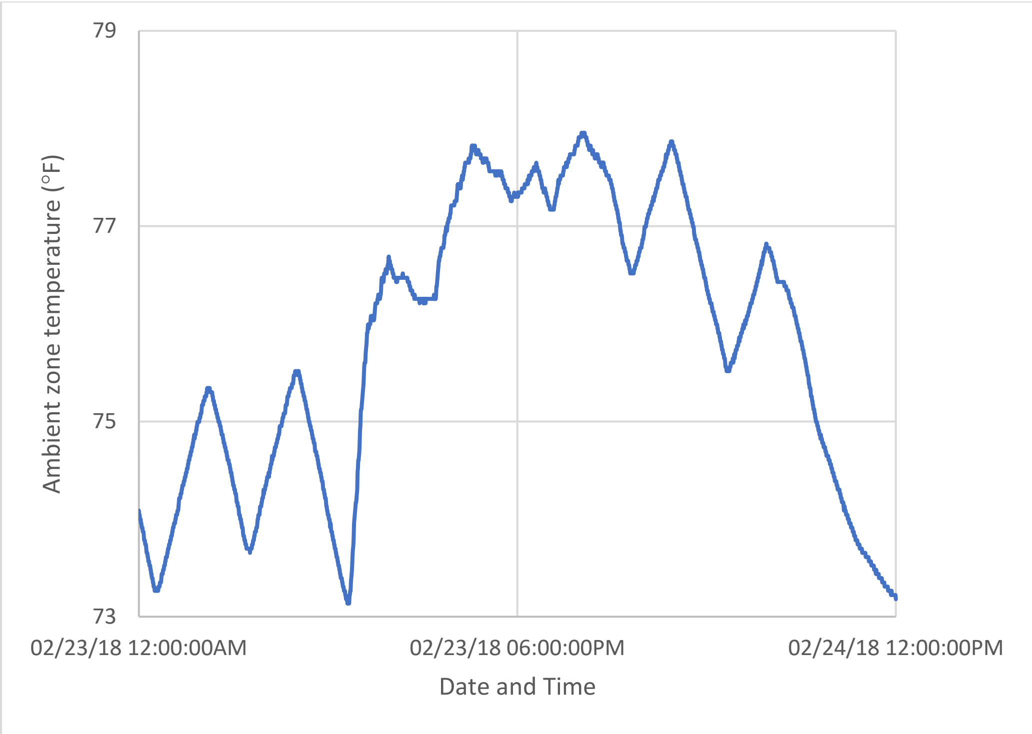

Figure 49: Ambient zone temperature for period 4 in site 3

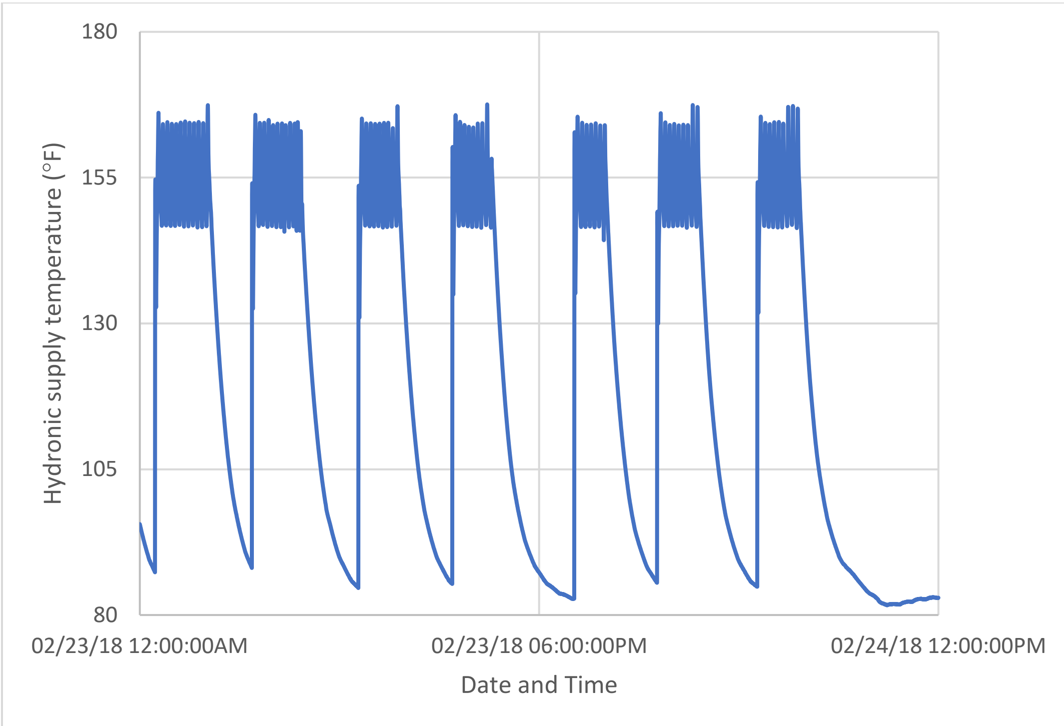

Figure 50: Hydronic supply temperature for period 4 in site 3

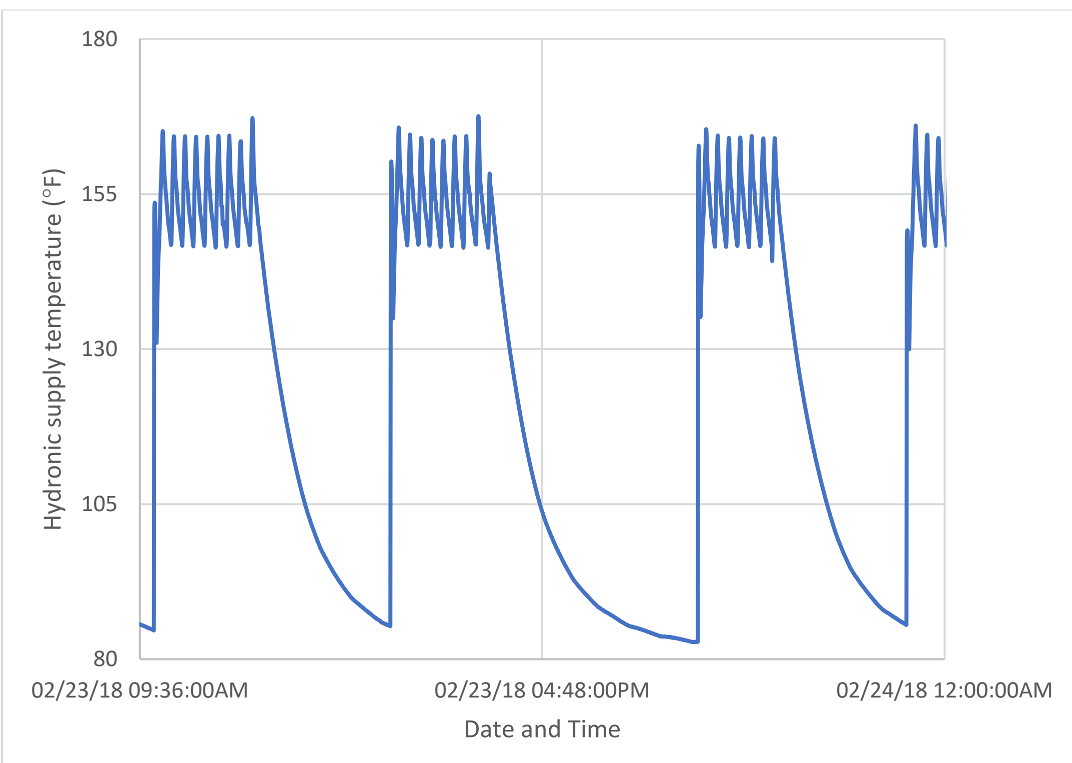

Figure 51: Sub-plot of hydronic supply temperature for period 4 in site 3

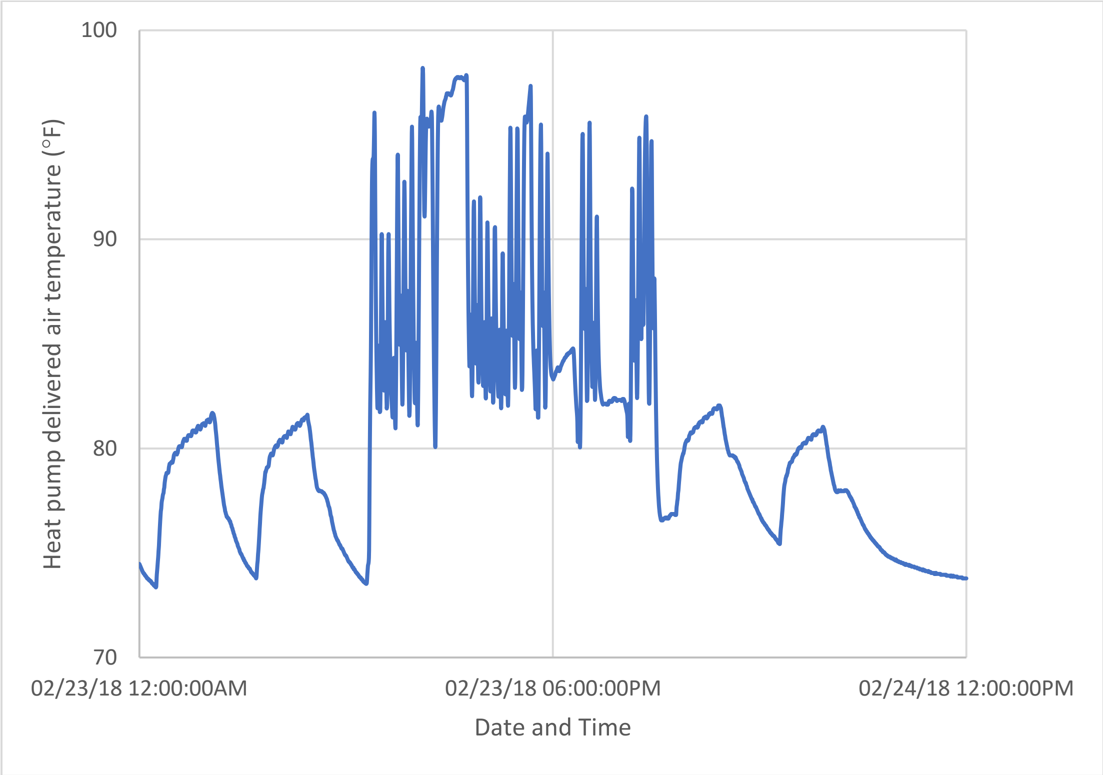

Figure 52: Heat pump delivered air temperature for period 4 in site 3

Site 3 — Period 4 Discussion

The outside ambient temperature in this period ranged between 45°F and 61°F. In this period of 36 hours, the heat pump provides heat for 12 hours, shown by spikes in heat pump delivered air temperature between 90°F and 98°F in Figure 52. The hydronic system provides heat during the whole period with 1.5–2 hours between a cluster of cycles, as seen in Figure 50. A direct effect of these off periods are the drops in room temperatures in Figure 49. This effect is also observed even when the heat pump is providing heat. Outdoor unit fin temperatures were not available for this period because of a logger failure. However, drops in delivered air temperature indicate that the outdoor unit defrosted often during the short period it provided heat.

Site 4

This site is in Mahopac, NY, about 50 miles north of the first 3 sites and so has consistently lower outdoor temperatures. The analyzed data is from December 2017 to February 2018. The first selected period of lower-than-average winter temperatures (with a low of -13°F) ranges from January 6 to 8 of 2018. The second period is from January 11 to 13, 2018, where temperatures reached 61°F – much warmer than average winter conditions. The third period is from the 6 to 8 of February 2018, with average winter temperatures ranging between 18°F and 35°F.

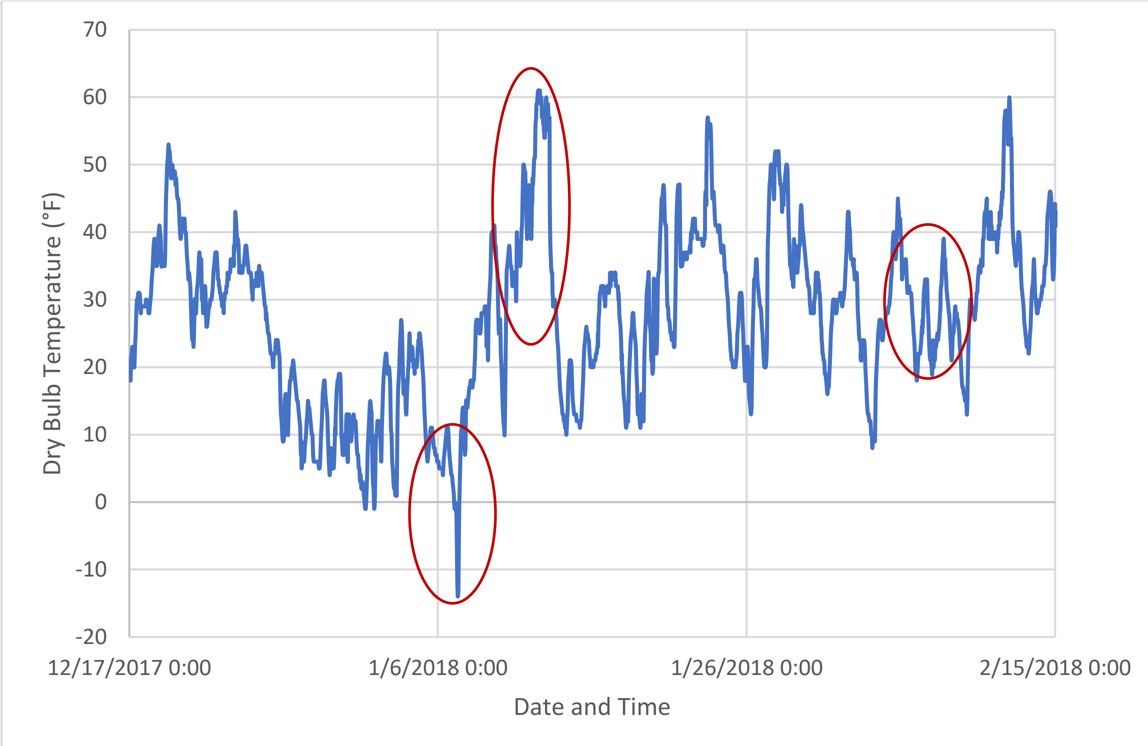

Figure 53: Outside ambient temperatures for site 4

As an aside, the following must be noted about the outdoor unit temperature data for this site. The outdoor unit fin temperatures do not show the high spikes (at least upwards of 50°F). There are, however, small spikes in temperature that align with a drop in heat pump delivered air temperature, which resemble defrost cycles. This is also observed in the rest of the data for the outdoor unit temperature, which means that the temperature probe was not sufficiently inside the fins and thus was not able to capture the full extent of the temperature rise during the defrost cycles. Therefore, the shorter temperature spikes, during which the heat pump does not provide heat will be counted as defrost cycles for the remainder of the discussion of this site.

Site 4 — Period 1 Temperature Plots

Plots of heat pump delivered, ambient zone, hydronic supply and outdoor unit fin temperatures were made for the three selected periods. *Figures 54–58* show those plots.

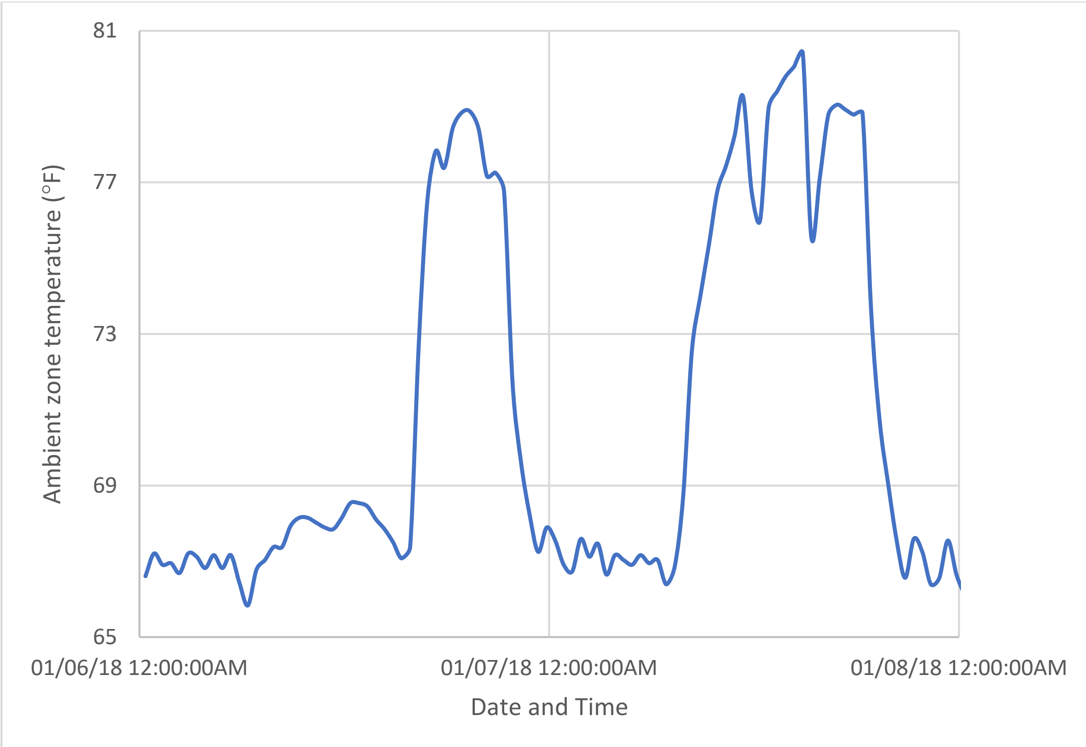

Figure 54: Ambient zone temperature for period 1

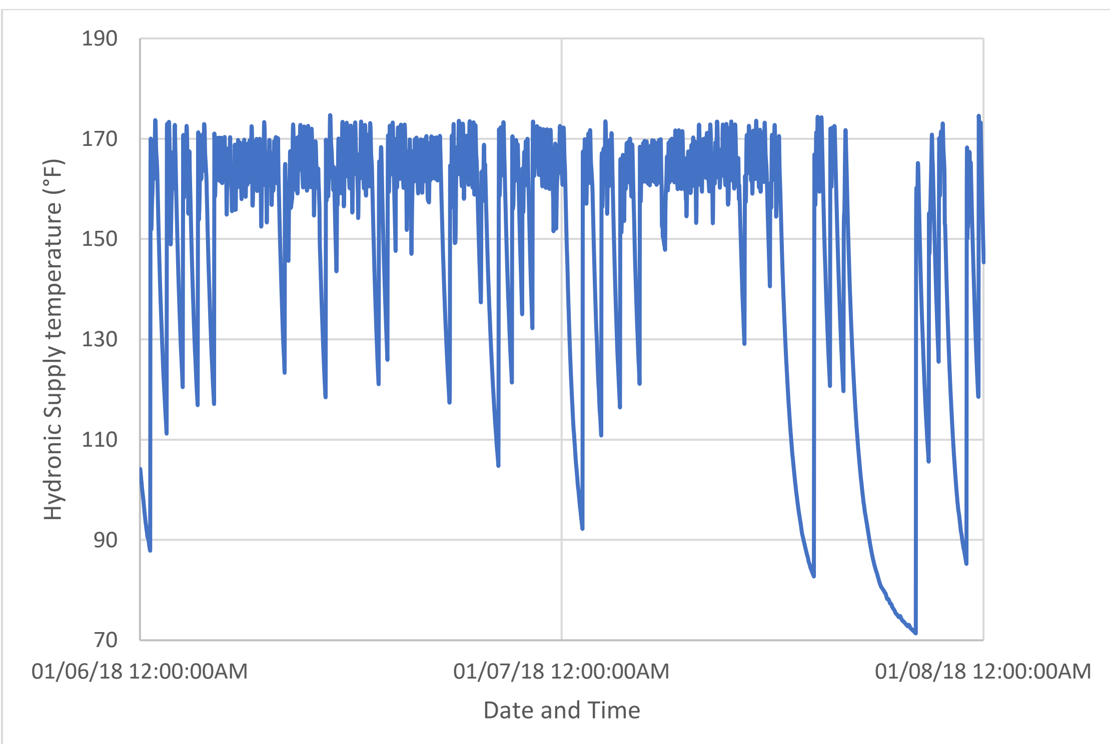

Figure 55: Hydronic supply temperature for period 1 in site 4

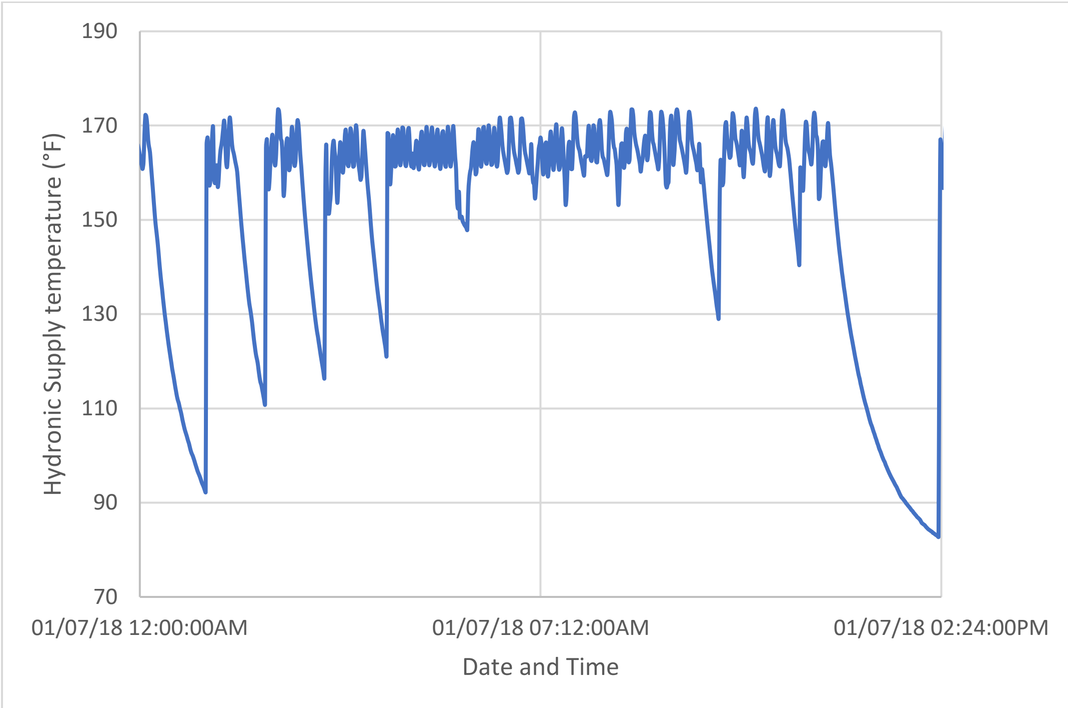

Figure 56: Sub-plot of hydronic supply temperature for period 1 in site 4

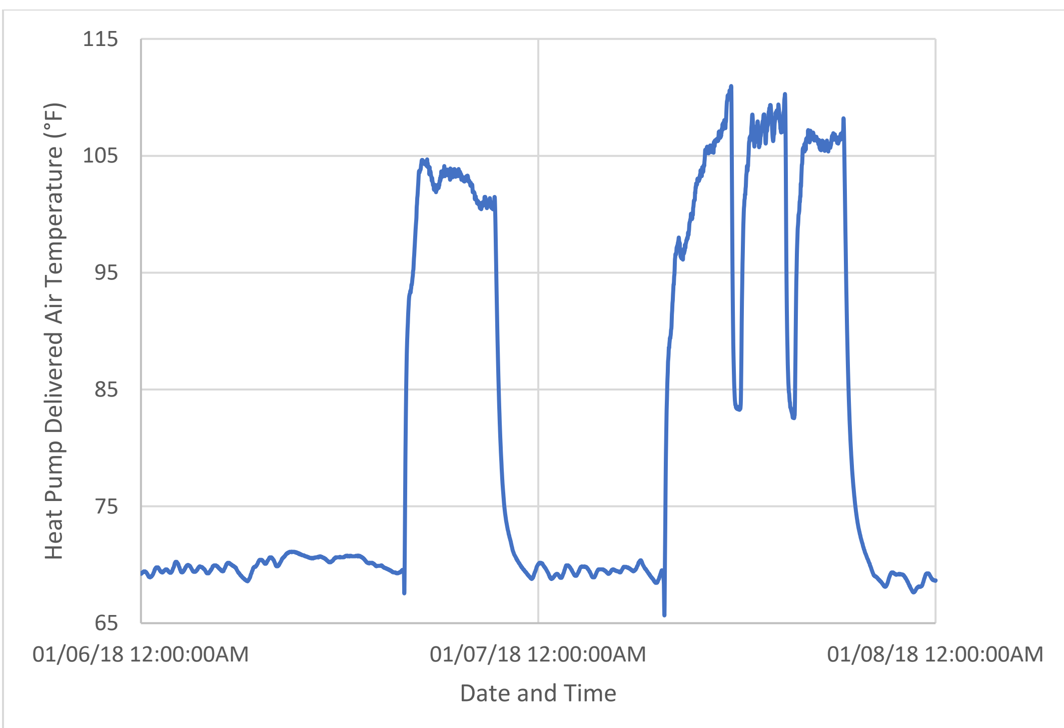

Figure 57: Heat pump delivered air temperature for period 1 in site 4

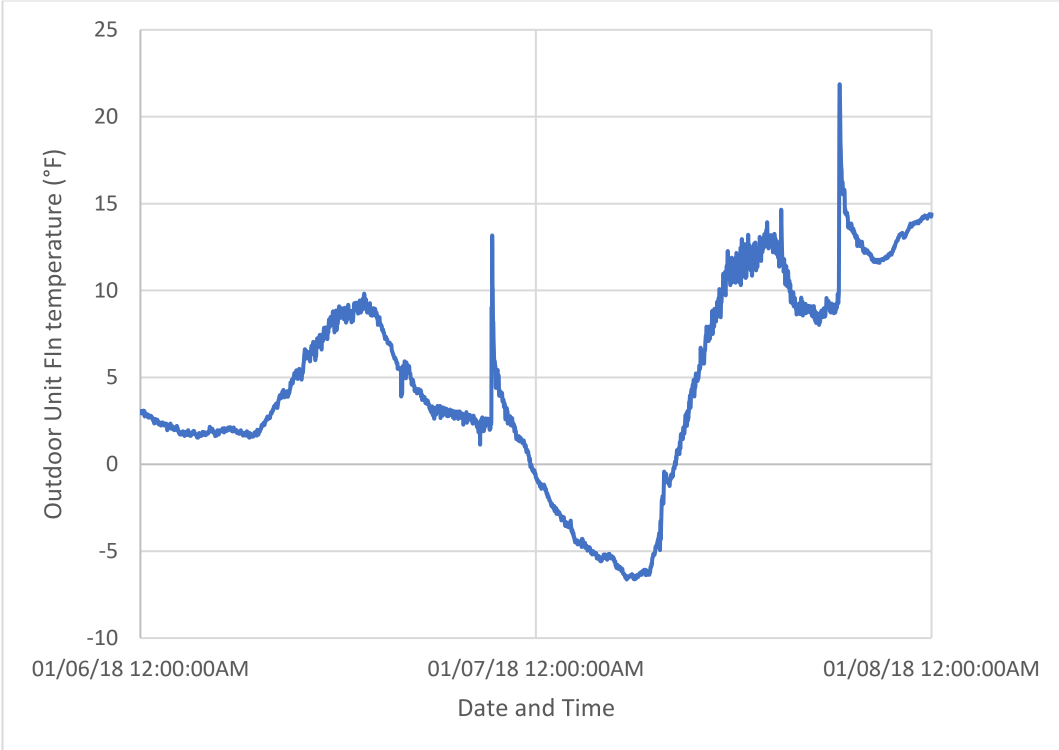

Figure 58: Outdoor unit fin temperature for period 1 in site 4

Site 4 — Period 1 Discussion

This period was the coldest of the three periods analyzed for this site. A plot of the ambient zone temperature (Figure 54) shows that the zone stayed at an approximate average temperature of 67°F. There are, however, two separate occasions (one on each day of the two-day period) where there is an incremental rise to ~80°F until late evening. These rises coincide with the times that the heat pump was delivering warm air (between 95°F and 110°F, as shown in Figure 57. During those same periods, the hydronic supply temperatures in Figure 55 show that the hydronic system was also providing heat. For the remaining parts of the period, the hydronic system was the only one meeting the heating demands. This shows that the heat pump was used manually as a supplementary source of heat, and also agrees with the homeowner's described pattern of usage. The outdoor unit fin temperatures in Figure 58 only indicate one defrost cycle on each of the days in this period, which is much lower than expected given prolonged hours of heat pump operation in cold weather.

Site 4 — Period 2 Temperature Plots

Figures 59–62 show ambient zone, hydronic supply, heat pump delivered and outdoor unit fin temperatures for period 2.

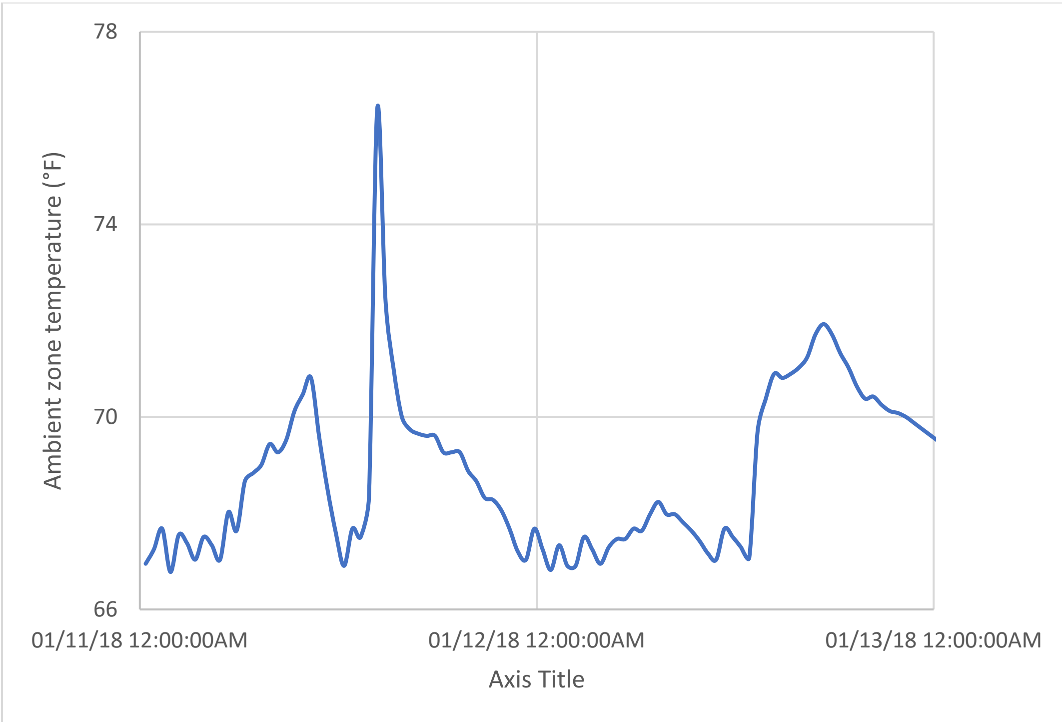

Figure 59: Ambient zone temperature for period 2 in site 4

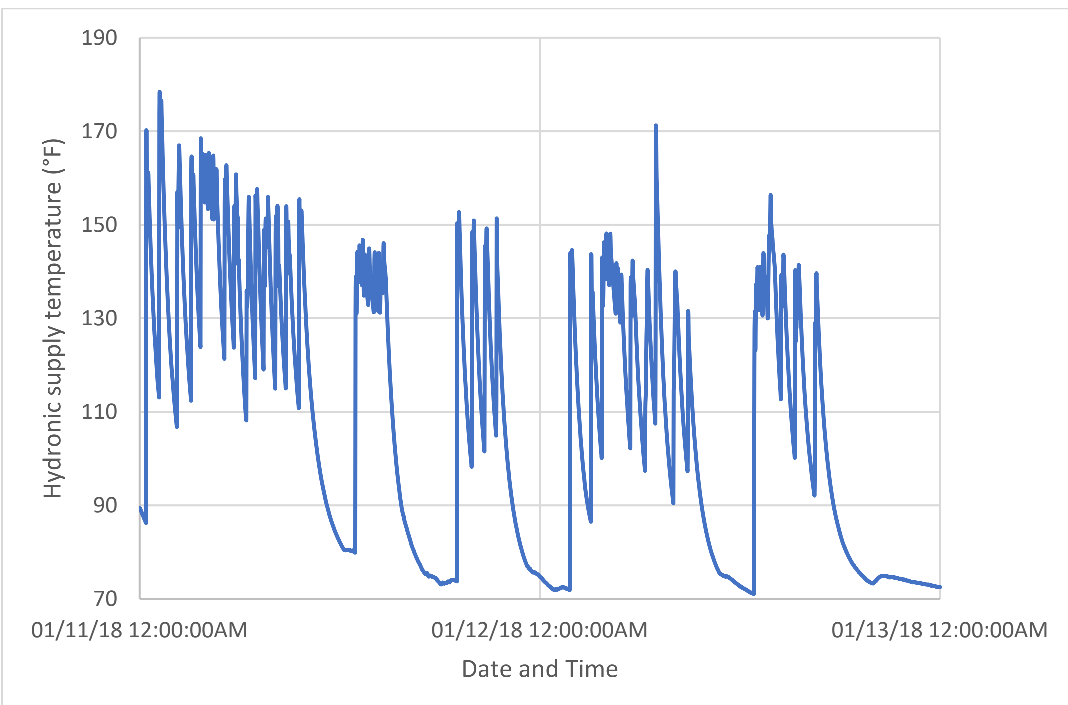

Figure 60: Hydronic supply temperature for period 2 in site 4

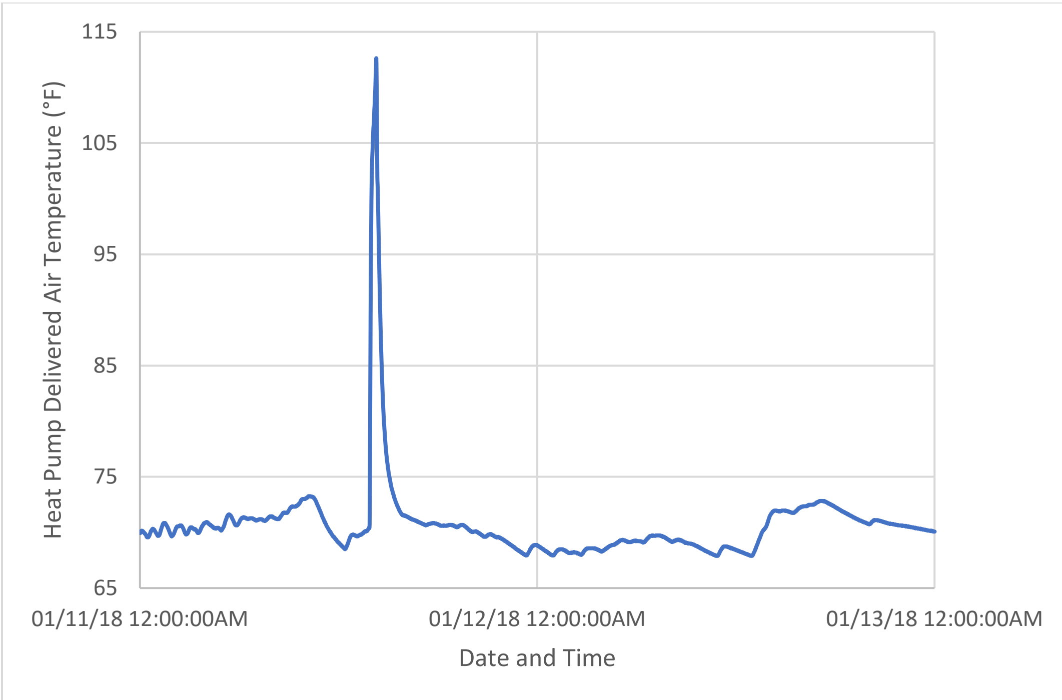

Figure 61: Heat pump delivered air temperature for period 2 in site 4

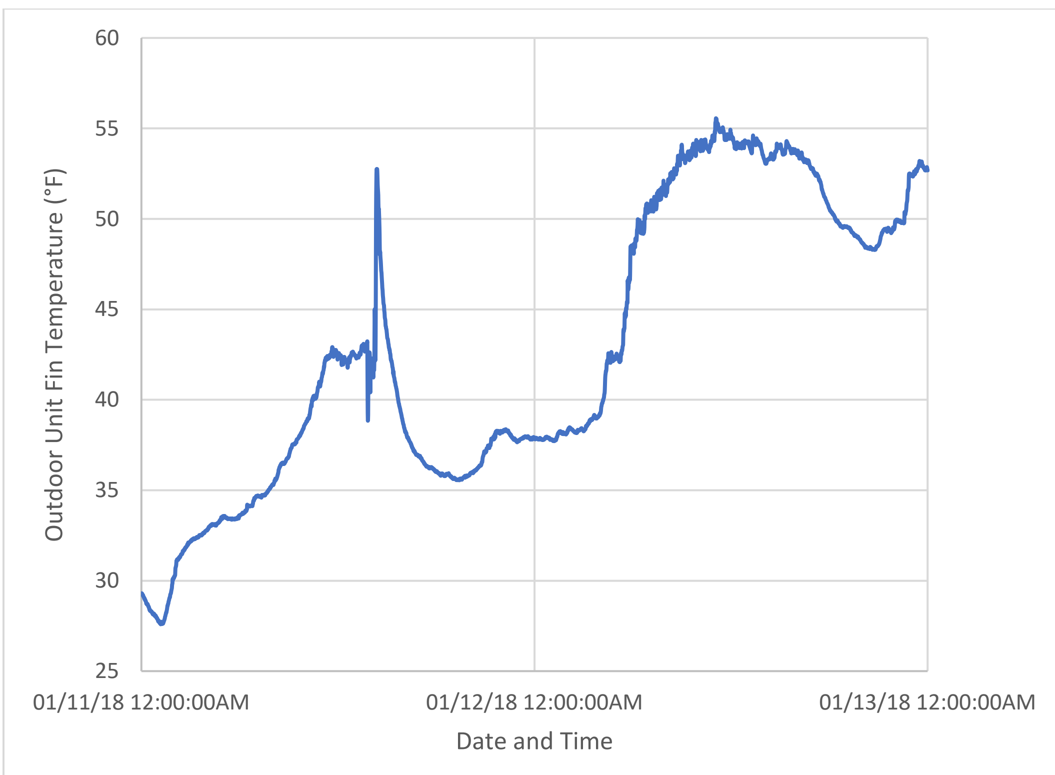

Figure 62: Outdoor unit fin temperature for period 2 in site 4

Site 4 — Period 2 Discussion

This period was warmer than average. The ambient zone temperatures in Figure 59 show that the zone generally maintained an average temperature of 68°F except a few instances of elevated temperatures. One of those instances is facilitated by heat provided by the heat pump, as shown by the sole temperature spike in Figure 61. Figure 62 shows that subsequently, the outdoor unit performs a defrost cycle. Apart from that short period where the heat pump operates, the hydronic system, as indicated by temperature spikes in Figure 60, provides heat the rest of the period. The frequency of operation of the hydronic heating is much lower in this period 2 as compared to that of period 1 because outside temperatures in the latter are much greater.

Site 4 — Period 3 Temperature Plots

Figures 63–66 show ambient zone, hydronic supply, heat pump delivered and outdoor unit fin temperatures for period 3.

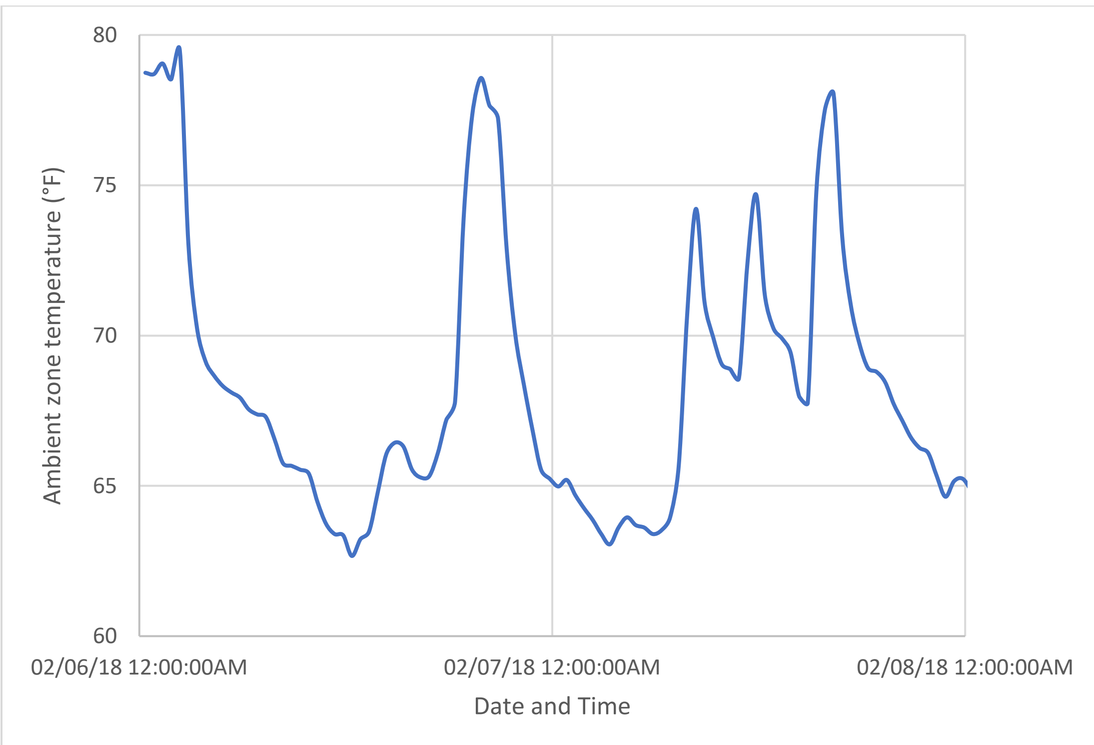

Figure 63: Ambient zone temperature for period 3 in site 4

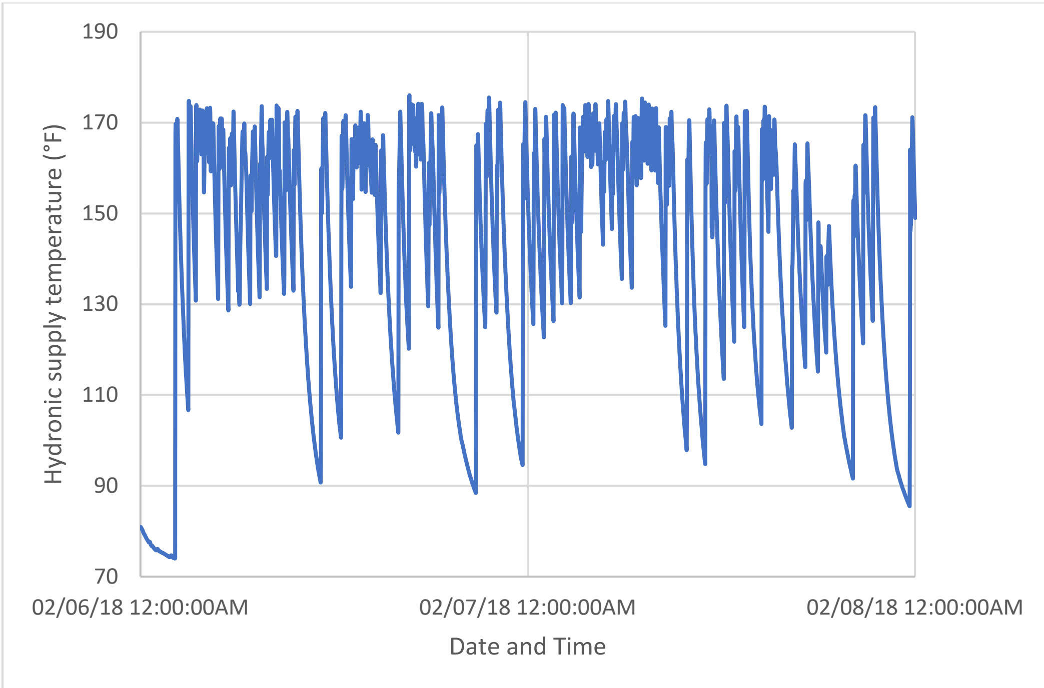

Figure 64: Hydronic supply temperature for period 3 in site 4

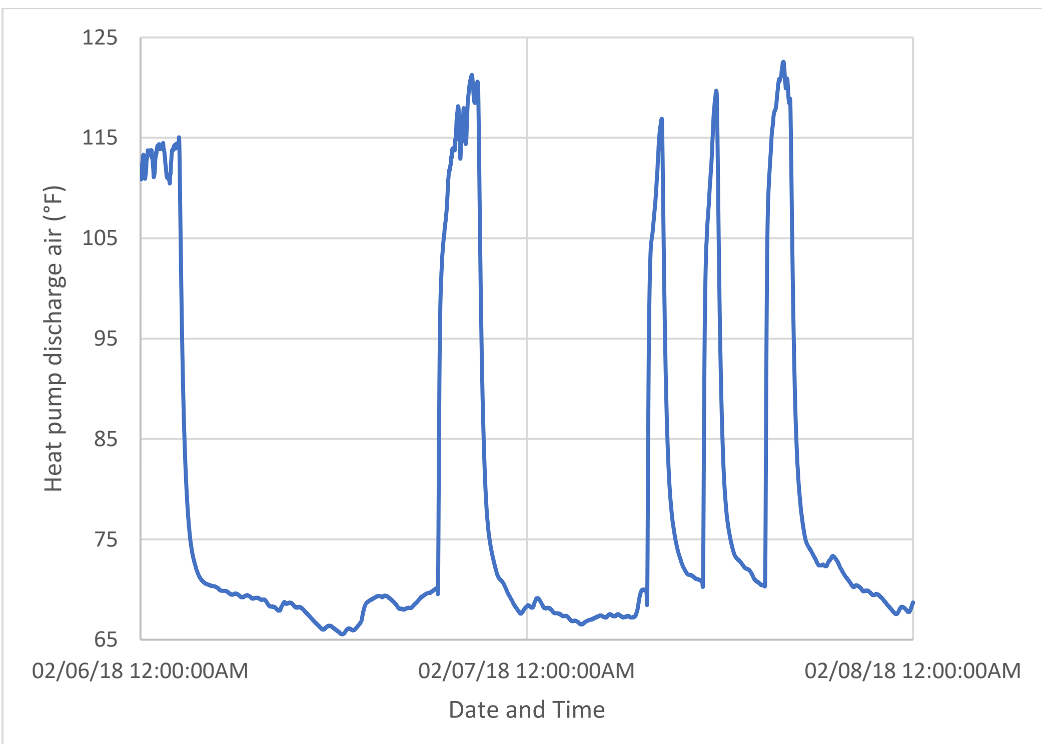

Figure 65: Heat pump delivered air temperature for period 3 in site 4

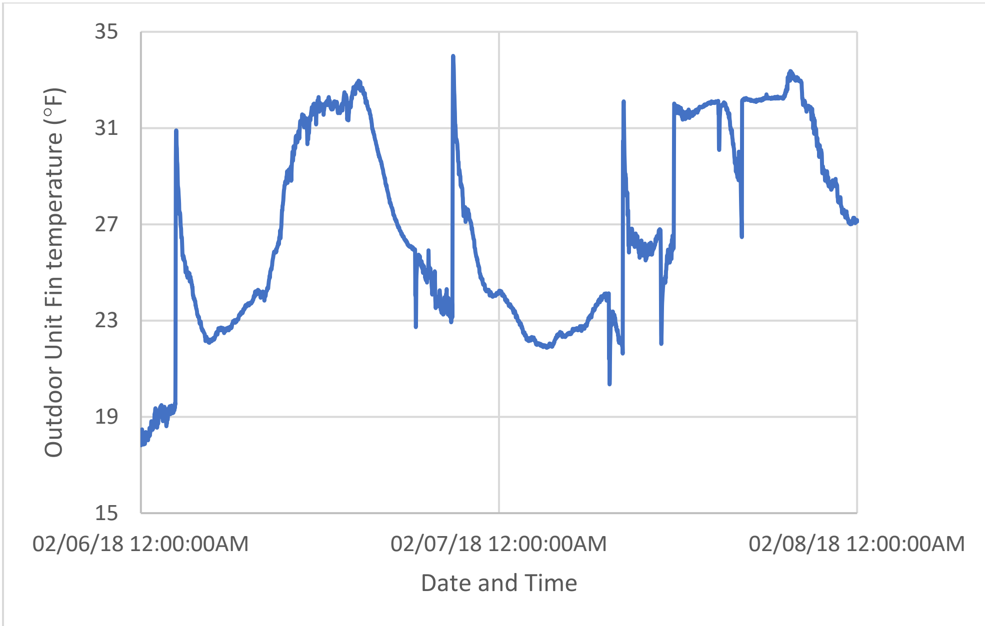

Figure 66: Outdoor unit fin temperature for period 3 in site 4

Site 4 — Period 3 Discussion

The ambient zone temperatures for this period, shown in *Figure 63*, fell below the baseline setting for the hydronic thermostat of 67°F on two occasions. During those periods, the hydronic system was providing heat, as seen in *Figure 64*. This suggests that the hydronic thermostat setting was manually set lower than 67°F during this period. The heat pump provided heat during the evening on both days in the period, and earlier in the day on the second day. Each of the periods of operation of the heat pump corresponds to a subsequent, steep rise in the ambient zone temperature. Once again, this is consistent with the homeowner's described pattern of usage. The outdoor unit fin temperatures in *Figure 66* indicate three defrost cycles occurred in this period.

Site 5

This site in Feura Bush, NY presents a unique dataset in this study since the homeowners did not utilize the hydronic heating system for their heating needs for any period. Therefore, the hydronic supply temperature was not included in the following datasets. *Figure 67*, which shows data over a ~20-day period, was used to select three periods for analysis. The first of those periods was from January 31st to February 3rd of 2019, where temperatures were as low as -7°F. There is a temperature rise included at the end of this period, as shown in the area marked period 1, to analyze recovery by the heat pump system after a period of colder than average weather. The second period is a warmer than average one with temperatures over 50°F, which lasted from February 4th to February 6th of 2019. The third period is from February 9th to February 13th of 2019, which consists of average winter temperatures for the area ranging mostly between 20-30°F.

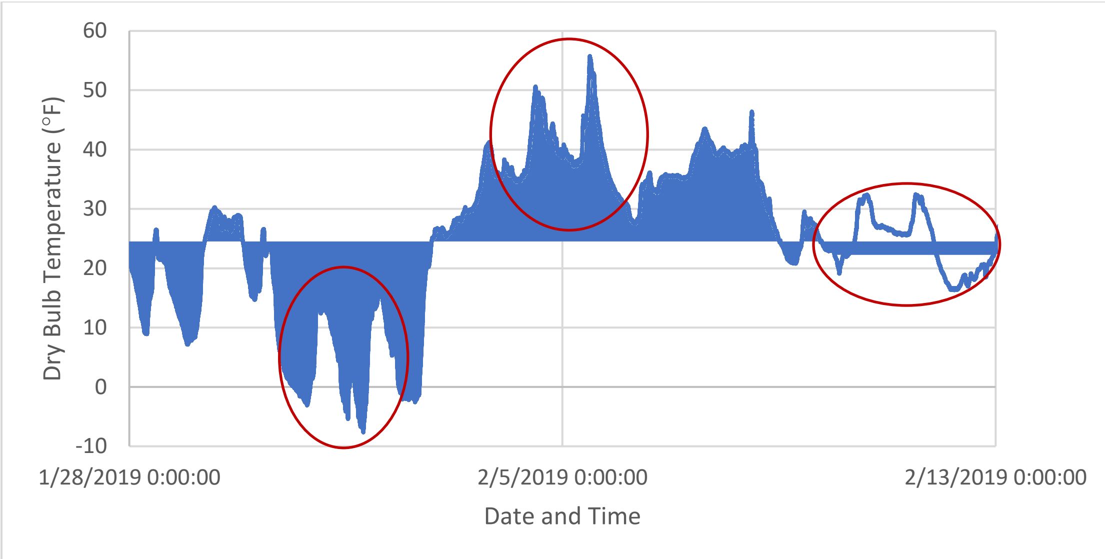

Figure 67: Outside ambient temperature for site 5

Site 5 – Period 1 Temperature Plots

Figures 68-71 below show the plots of ambient zone, heat pump delivered air and outdoor unit fin temperatures for period 1 along with sub-plots showing temperature behavior over shorter periods of time when necessary.

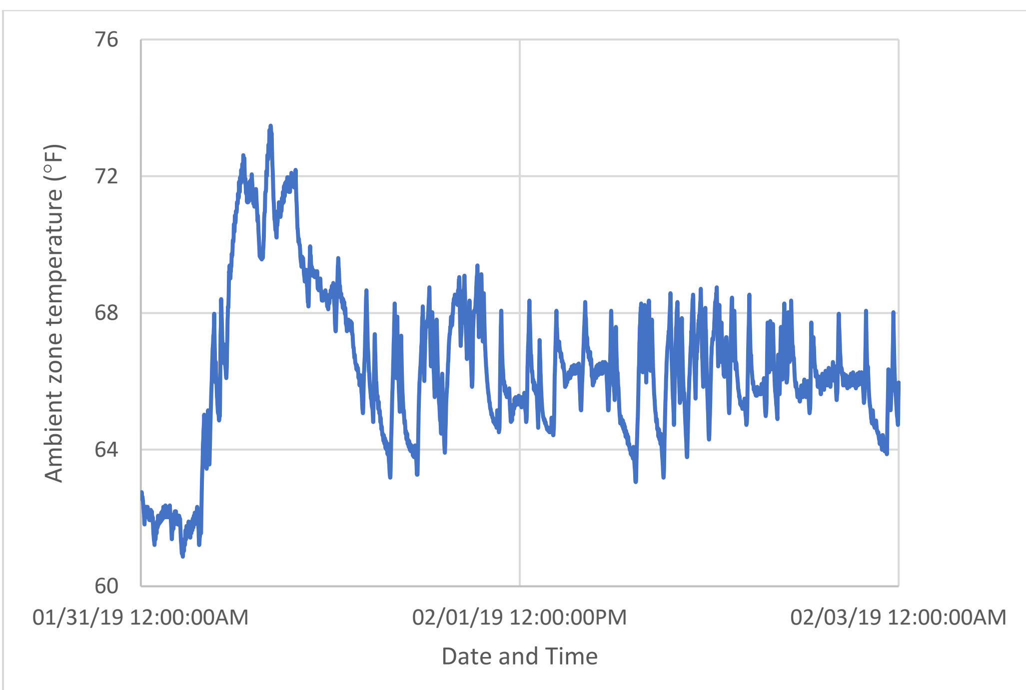

Figure 68: Ambient zone temperature for period 1 in site 5

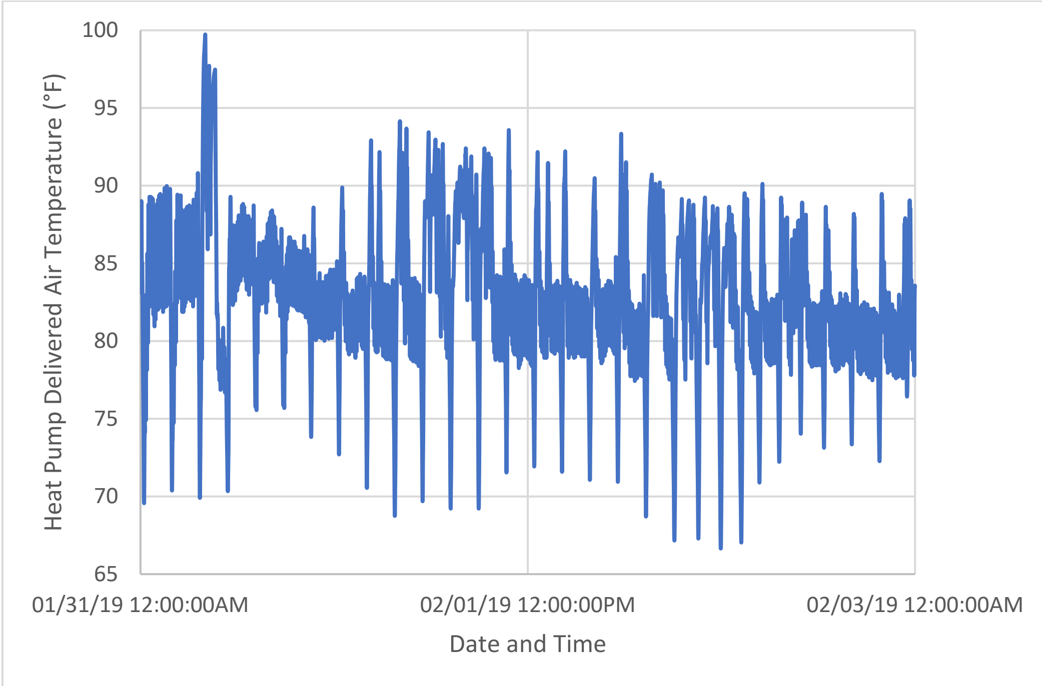

Figure 69: Heat pump delivered air temperature for period 1 in site 5

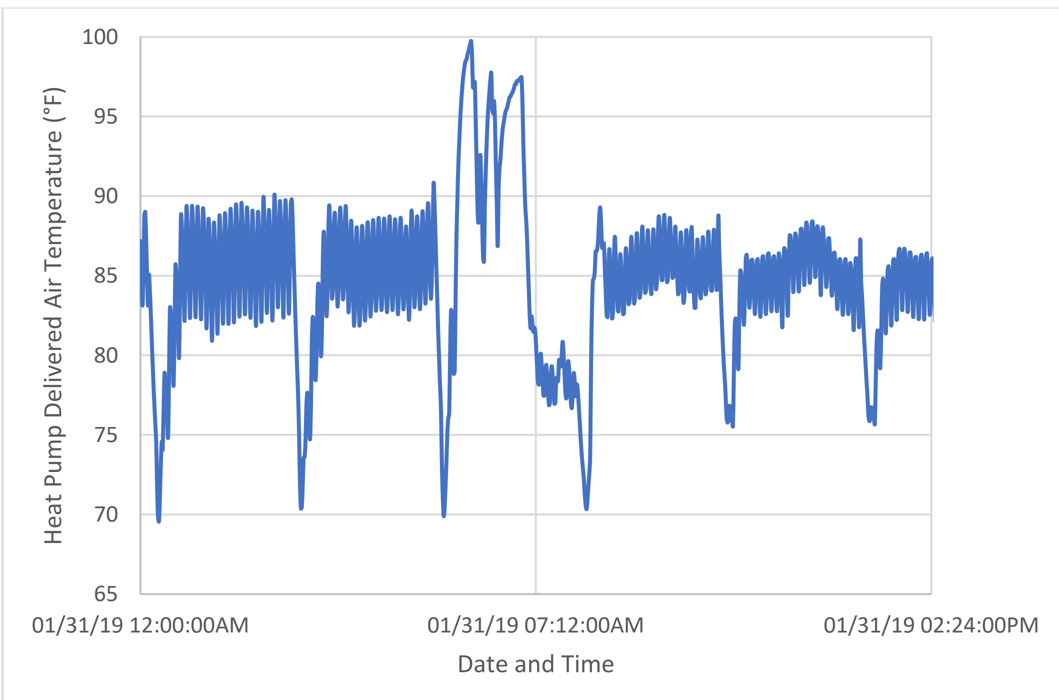

Figure 70: Sub-plot of heat pump delivered air temperature for period 1 in site 5

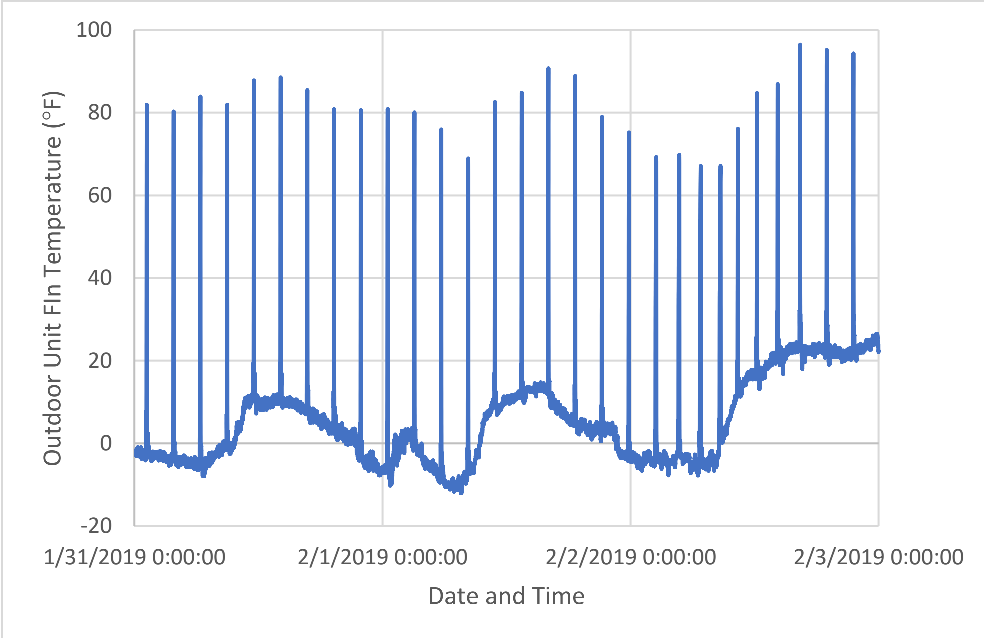

Figure 71: Outdoor unit fin temperature for period 1 in site 5

Site 5 – Period 1 Discussion

The ambient zone temperatures for period 1 in *Figure 68* indicate that the heating system was able to keep up with the heating demands. Temperatures in the zone maintain an approximate average of 66°F for most of the period. There is a rise in ambient zone temperature to ~73°F, which remains in that range for a few hours. This occurred at the same time that the heat pump delivered air temperature, shown in *Figure 69* (and more closely in *Figure 70*), provided air between 95°F and 100°F. This temperature range shows the air delivered at that time was warmer than that of the rest of the period, suggesting that the occupants had manually turned up the air setting. *Figure 69* also shows localized rises in heat pump delivered air temperature, generally up to 90°F. This occurs after the heat pump has turned back from a defrost cycle of the outdoor unit, possibly in the attempt to recover from the time the zone did not receive heat. Given the zone is a relatively small office room, the sudden rise in heat pump delivered air temperature causes the zone temperature to rise by 3-4°F rapidly. The outdoor unit executes defrost cycles during this period at regular intervals of ~2.5 hours during which its fin temperatures rise higher than 80°F, each lasting between 2 and 3 minutes.

Site 5 – Period 2 Temperature Plots

Figures 72-76 below show the plots of ambient zone, heat pump delivered air and outdoor unit fin temperatures for period 2 along with sub-plots showing temperature behavior over shorter periods of time when necessary.

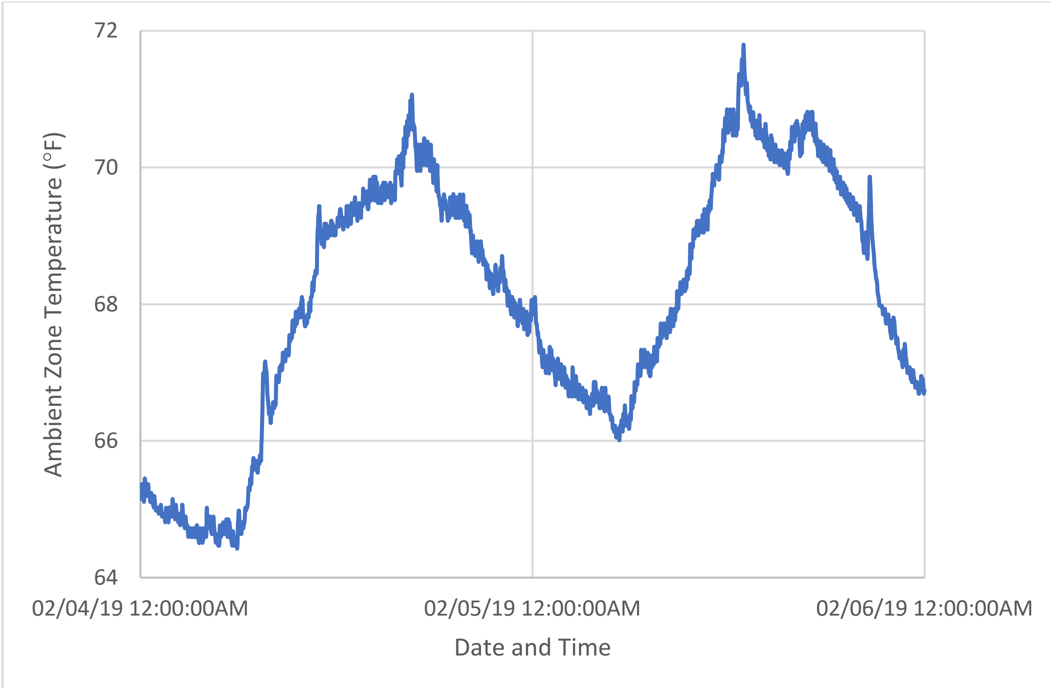

Figure 72: Ambient zone temperature for period 2 in site 5

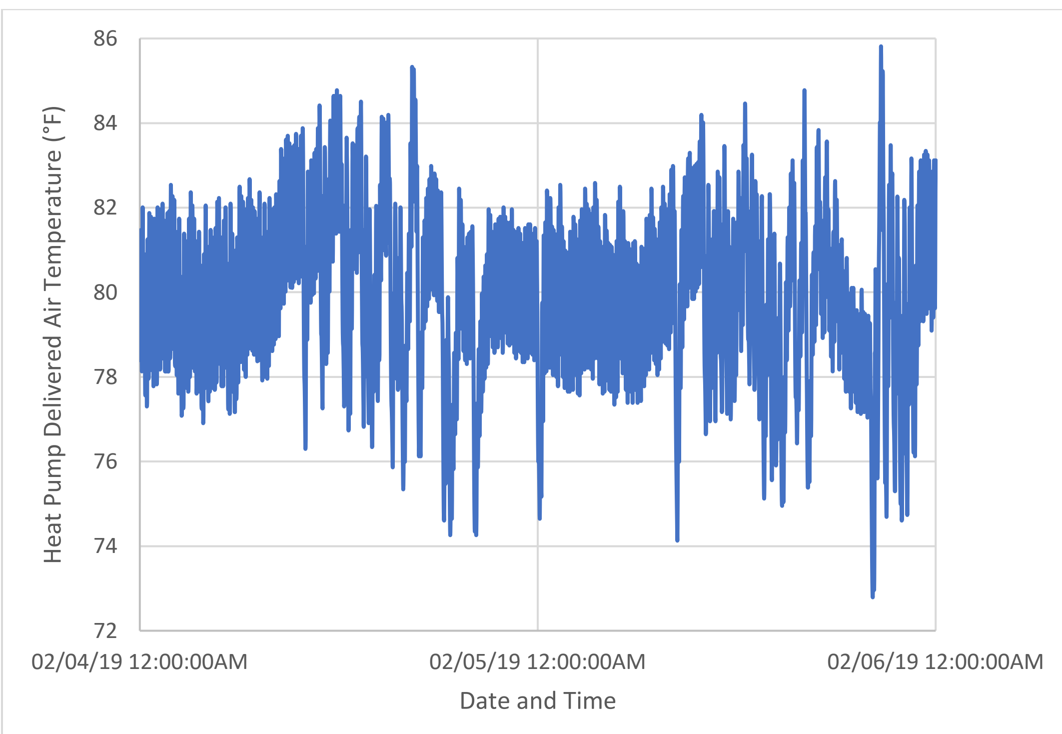

Figure 73: Heat pump delivered air temperature for period 2 in site 5

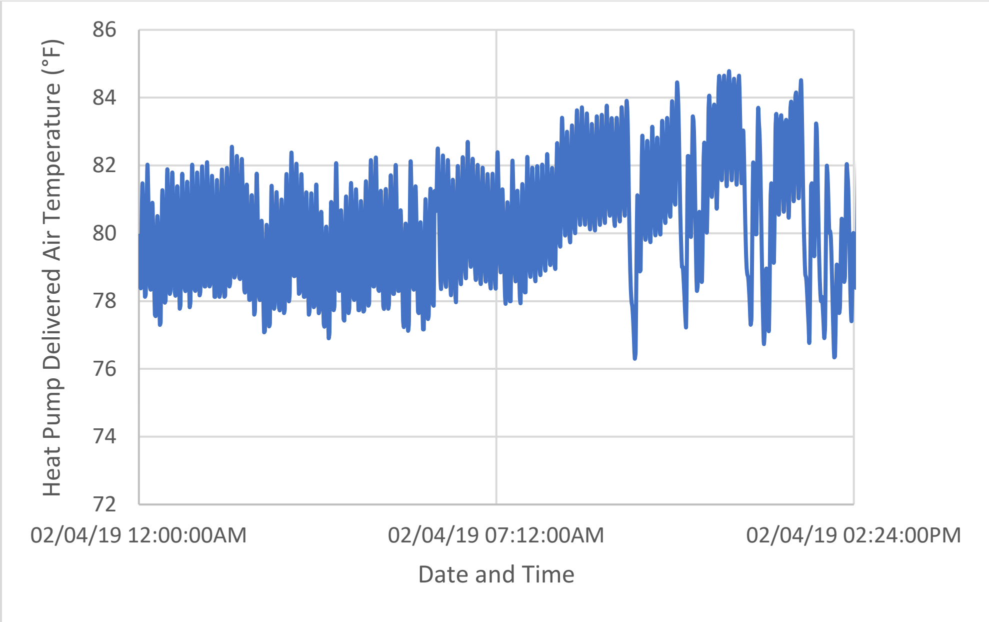

Figure 74: Sub-plot of heat pump delivered air temperature for period 2 in site 5

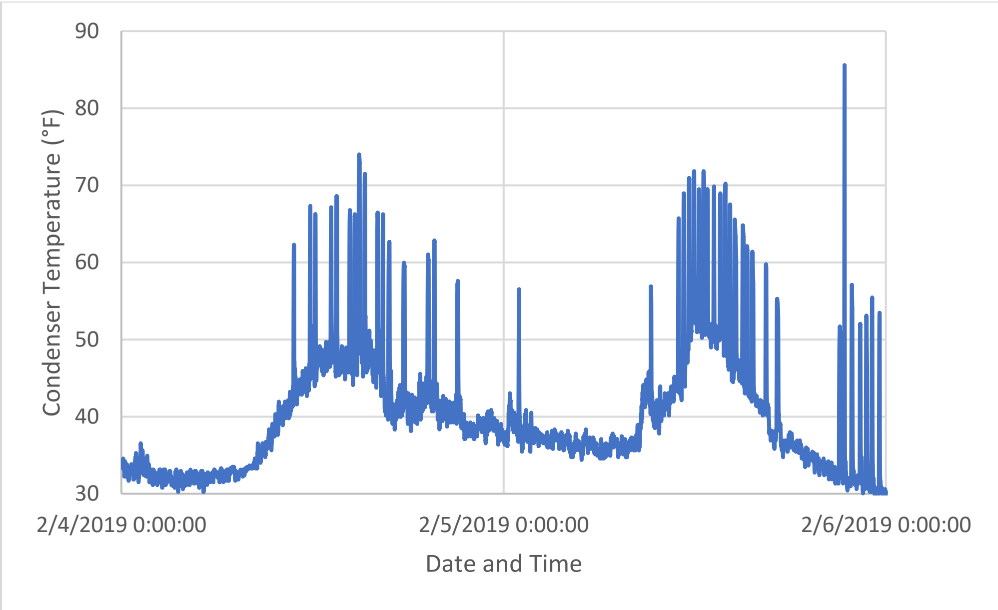

Figure 75: Outdoor unit fin temperature for period 2 in site 5

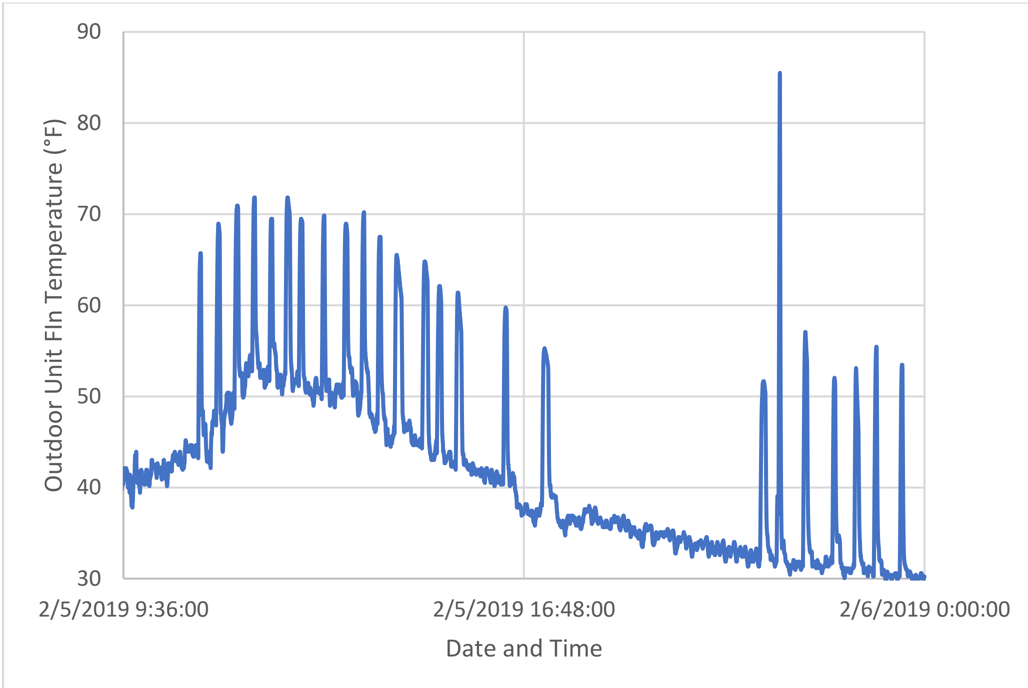

Figure 76: Sub-plot of outdoor unit fin temperature for period 2 in site 5

Site 5 – Period 2 Discussion

Ambient zone temperatures for period 2, shown in *Figure 72*, had two daily, steady temperature rises. The temperature rises slightly above the general setpoint that the thermostat setting of 65-68°F that the homeowner claimed to have used. This suggests that the setting was manually changed to a higher temperature. The heat pump cycles much more frequently compared to the colder first period. The temperatures of the delivered air are lower in this period. This is because the outside ambient temperatures were, on average, ~30°F warmer in this period than those in the first period. The outdoor unit performs defrost cycles that have lower peaks than those seen in period 1. These defrost cycles occur very frequently, with some cycles being separated by as low as 20 minutes, lasting ~4 minutes each. The occurrence of these cycles was unexpected given outside temperatures exceeding 40°F.

Site 5 – Period 3 Temperature Plots

Figures 77-81 below show the plots of ambient zone, heat pump delivered air and outdoor unit fin temperatures for period 3 along with sub-plots showing temperature behavior over shorter periods of time when necessary.

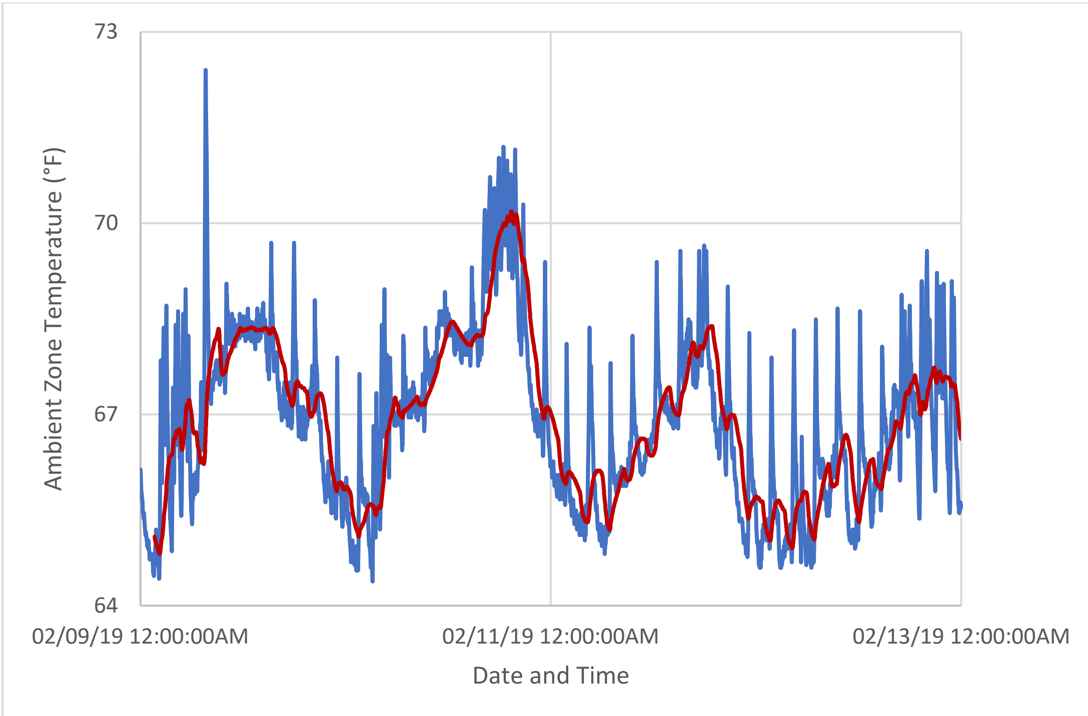

Figure 77: Ambient zone temperature (blue) and moving average of ambient zone temperature (red) for period 3 in site 5

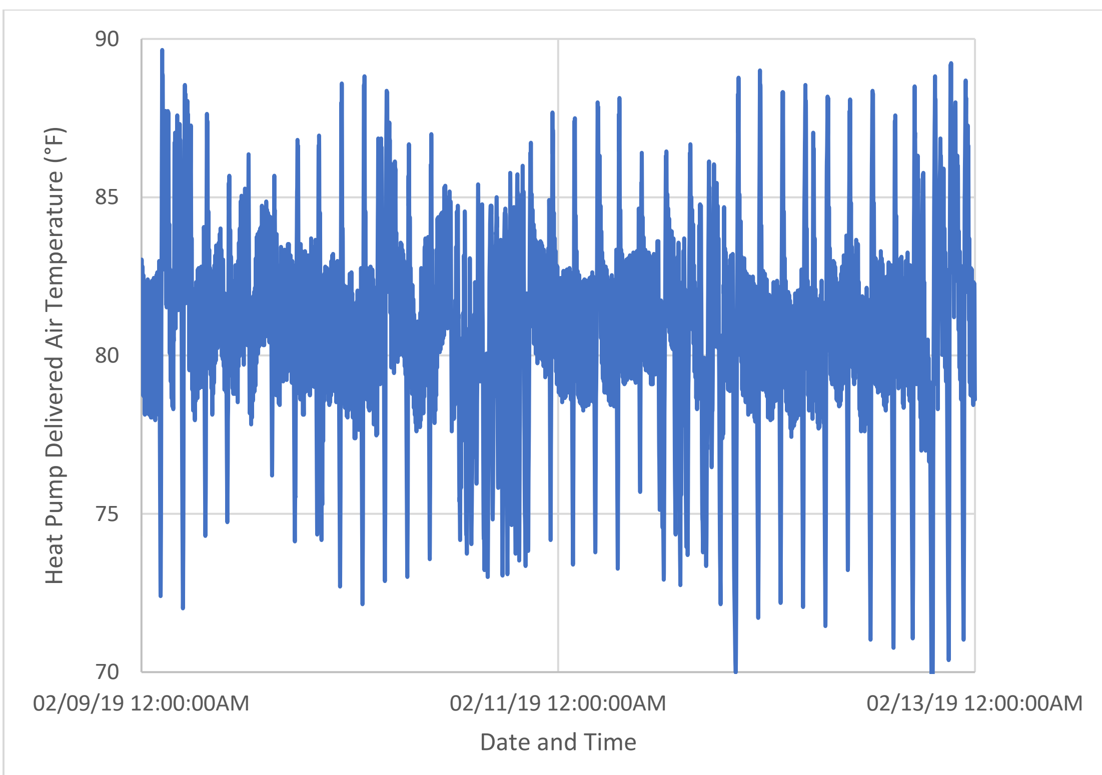

Figure 78: Heat pump delivered air temperature for period 3 in site 5

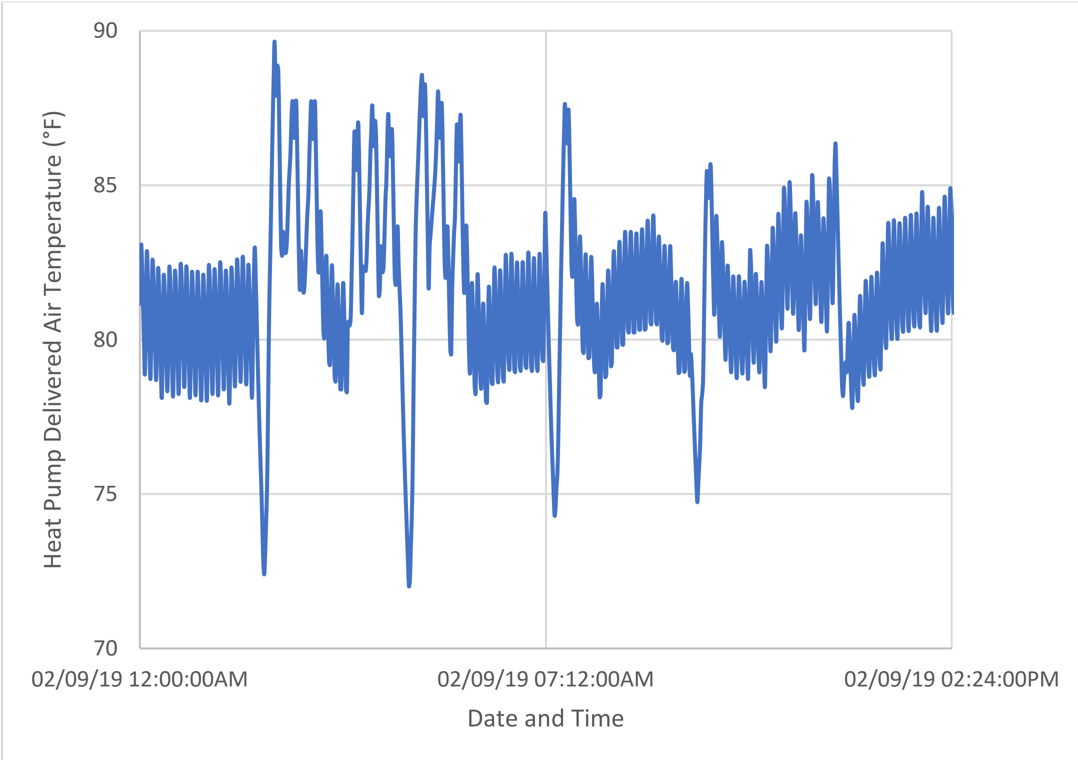

Figure 79: Sub-plot of heat pump delivered air temperature for period 3 in site 5

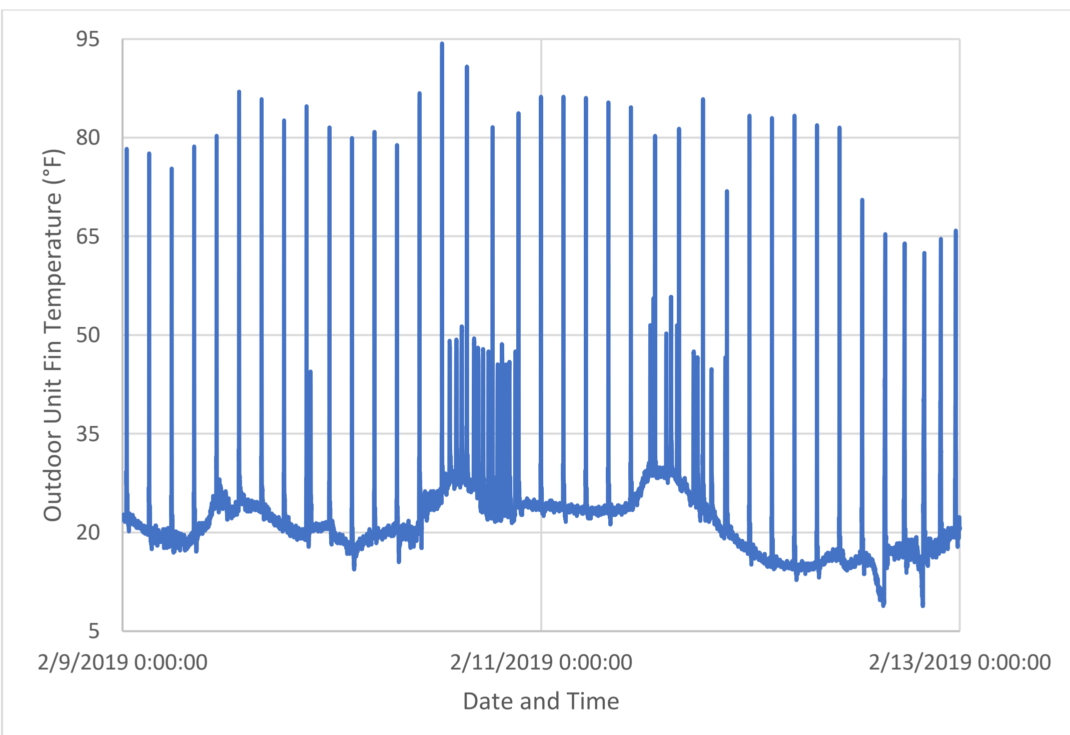

Figure 80: Outdoor unit fin temperature for period 3 in site 5

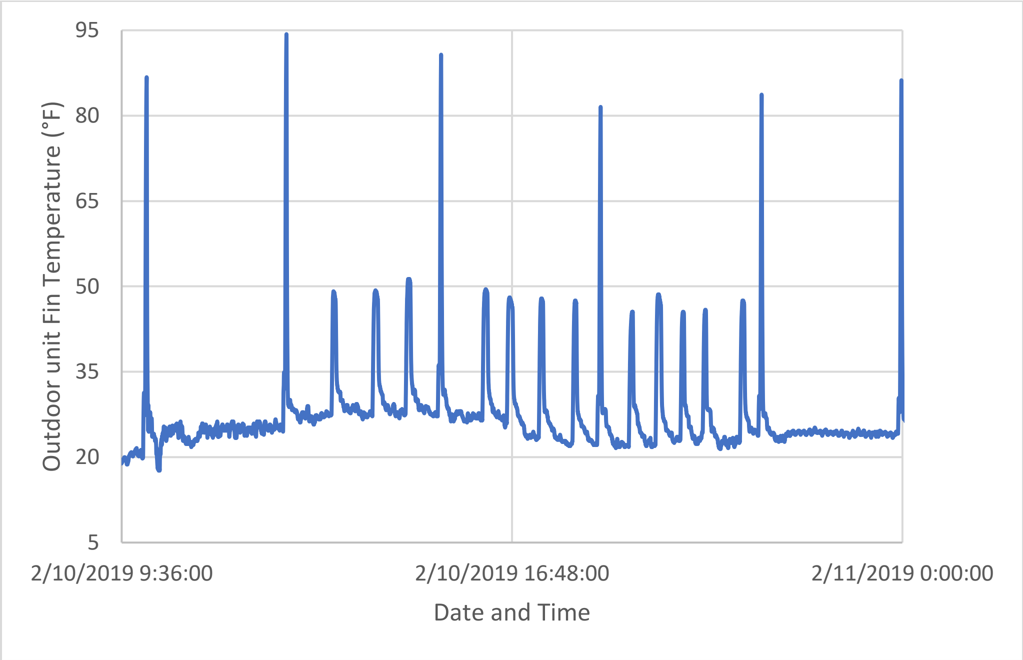

Figure 81: Sub-plot of Outdoor unit fin temperature for period 3 in site 5

Site 5 – Period 3 Discussion

A plot of the ambient zone temperature in *Figure 77* shows a gradual rise in temperature on each of the four days included in this period, as demonstrated by trend of the moving average in the same plot. This trend was also observed in period 2. The peak heat pump delivered air temperatures, shown in *Figures 78* and *79*, are greater than those of period 2, but not as high as those of period 1. This is expected because the outside ambient temperatures and, therefore, heating loads for this period are generally between those of periods 1 and 2 (given that the temperature settings are not different). Defrost cycles in the outdoor unit appear to be mostly time dependent, as shown by the periodic high spikes (upwards of 65°F to 90°F) in fin temperature in *Figure 80*. These spikes occur once every 2.5-3 hours. A few defrost cycles with lower peaks (~50°F), however, occur between the high peak ones, albeit much more frequently (as low as 25 minutes between each cycle). These defrost cycles lasted ~4 minutes each.

Site 6

Three periods were chosen for analysis for this site. *Figure 82* below shows the ambient outside temperature over a portion of the test period and isolates the chosen periods. The first is a cold period from November 21 to November 24, a three-day timeframe in 2018 where the lowest temperature was 4°F. The second period is an average winter period from December 6 to December 8 of 2018, with temperature ranges from 12 to 35°F. The final period chosen is a warmer period from December 27 to December 30 of 2018. In this period, temperatures rise to 50°F and then fall to ~27°F.

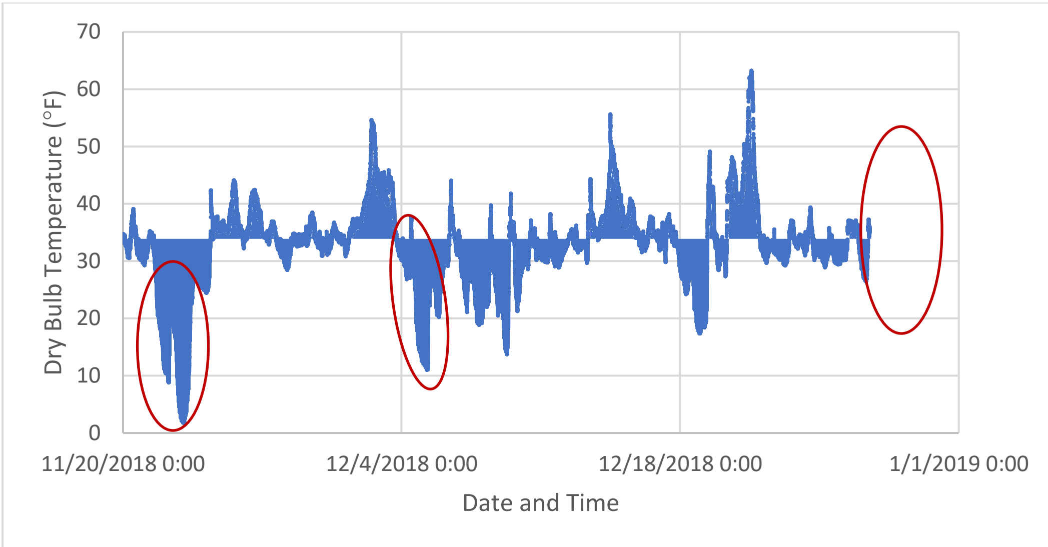

Figure 82: Outside ambient temperature for site 6

Site 6 — Period 1 Temperature Plots

Figures 83-88 show plots of ambient zone, hydronic supply, heat pump delivered air and outdoor unit fin temperatures for period 1 along with sub-plots showing temperature behavior over shorter periods of time when necessary.

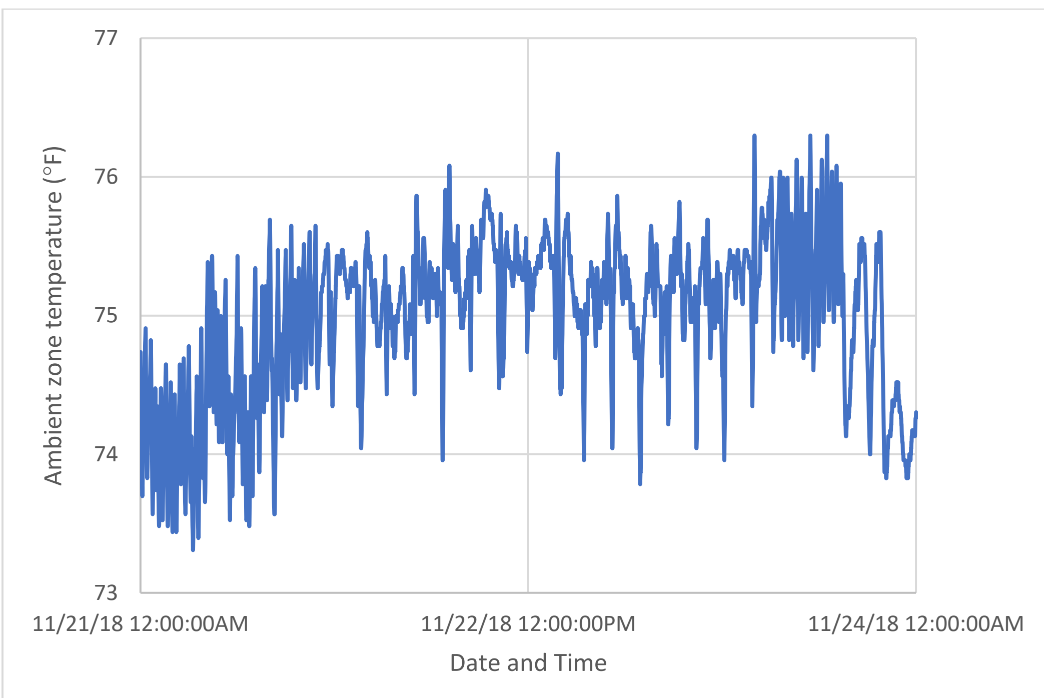

Figure 83: Ambient zone temperature for period 1 in site 6

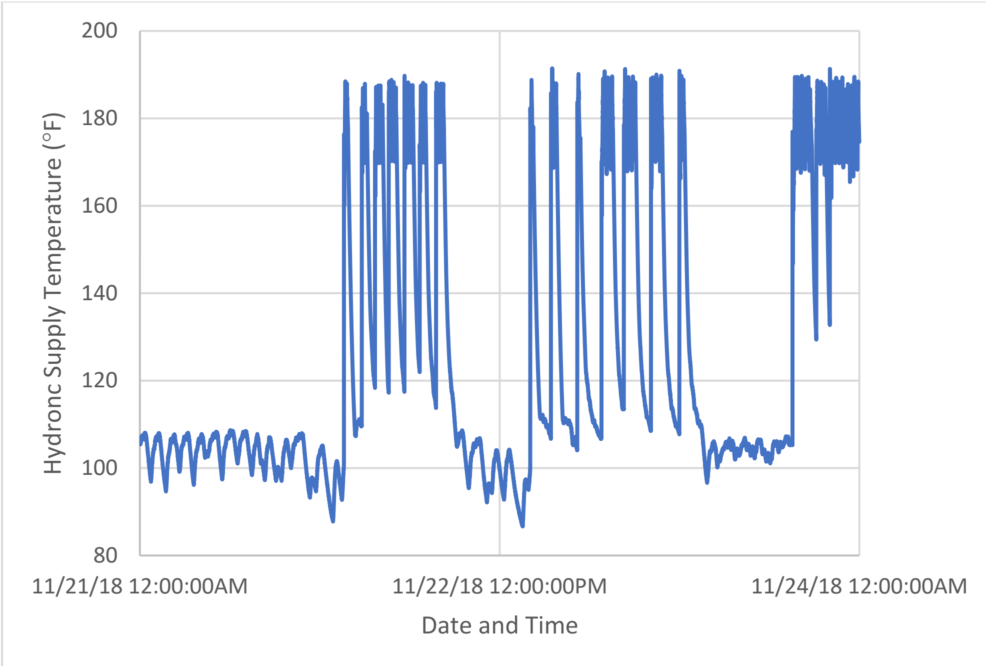

Figure 84: Hydronic zone temperature for period 1 in site 6

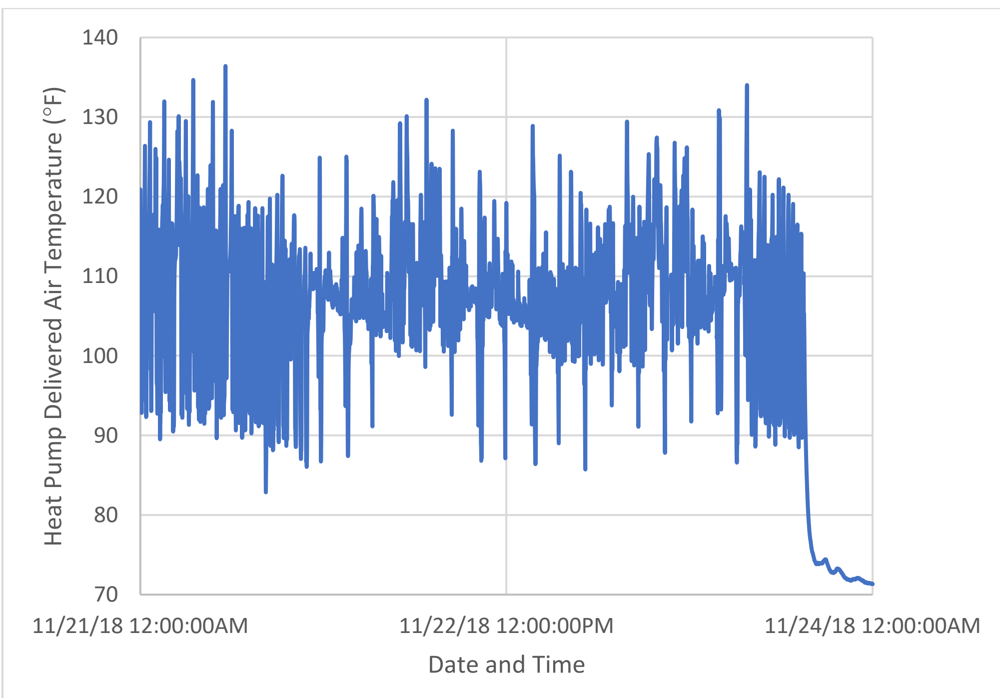

Figure 85: Heat pump delivered air temperature for period 1 in site 6

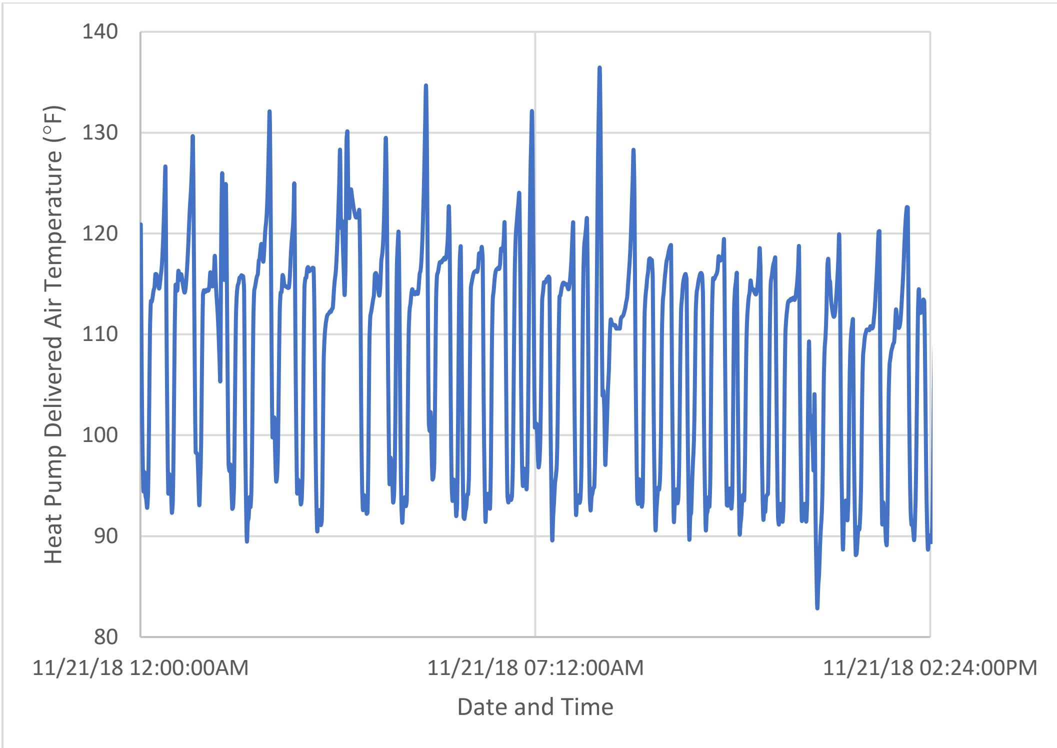

Figure 86: Sub-plot of heat pump delivered air temperature for period 1 in site 6

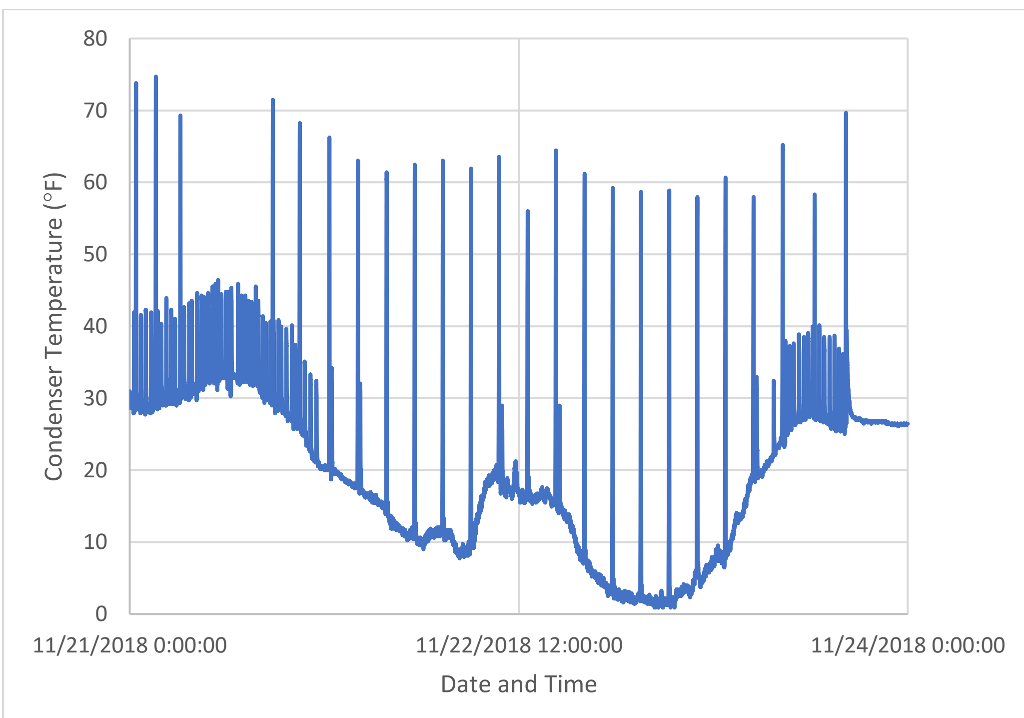

Figure 87: Outdoor unit fin temperature for period 1 in site 6

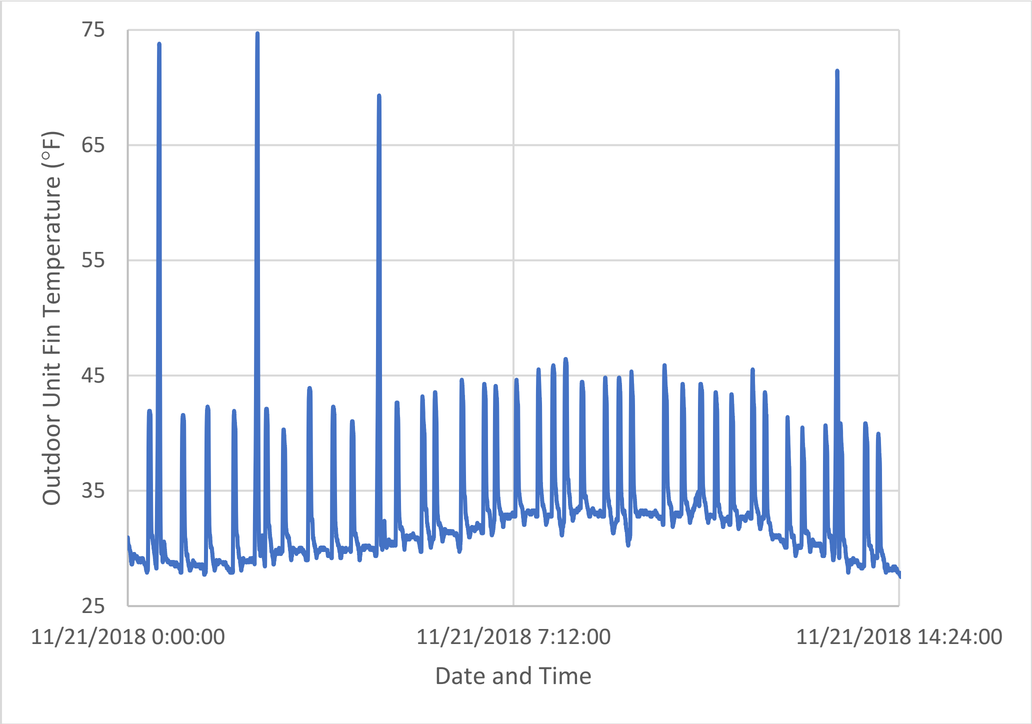

Figure 88: Sub-plot of outdoor unit fin temperature for period 1 in site 6

Site 6 — Period 1 Discussion

Ambient zone temperatures shown in *Figure 83* for the first period indicate that the temperature only varied ±1.5°F from an approximate average of 75°F, which generally higher than the nominal 73°F setting that the homeowner reported using. The reason for the higher recorded temperatures is because the temperature sensors in this site were placed close to the ceiling, where temperatures are higher compared to those closer to the floor. *Figure 85* shows that the heat pump operated to provide heat for most of this period. The hydronic system also operated, albeit much less often, as seen in *Figure 84*. The frequency of operation and peak air temperatures of the heat pump were reduced when the hydronic system was operating. The outdoor unit cycled into defrost mode in regular periods of 2.5 to 3 hours with outdoor unit heat exchanger fin temperature peaks ranging from ~60°F to ~75°F throughout this period, which lasted ~5 minutes each. There were two distinct periods where more frequent defrost cycles, 15 to 30 minutes apart, with lower peak temperatures occurred and each lasted ~3 minutes. These defrost cycles coincide directly with the periods the hydronic system was not providing heat.

Site 6 — Period 2 Temperature Plots

Figures 89–92 show plots of ambient zone, hydronic supply heat pump delivered air and outdoor unit fin temperatures for period 2 along with sub-plots showing temperature behavior over shorter periods of time when necessary.

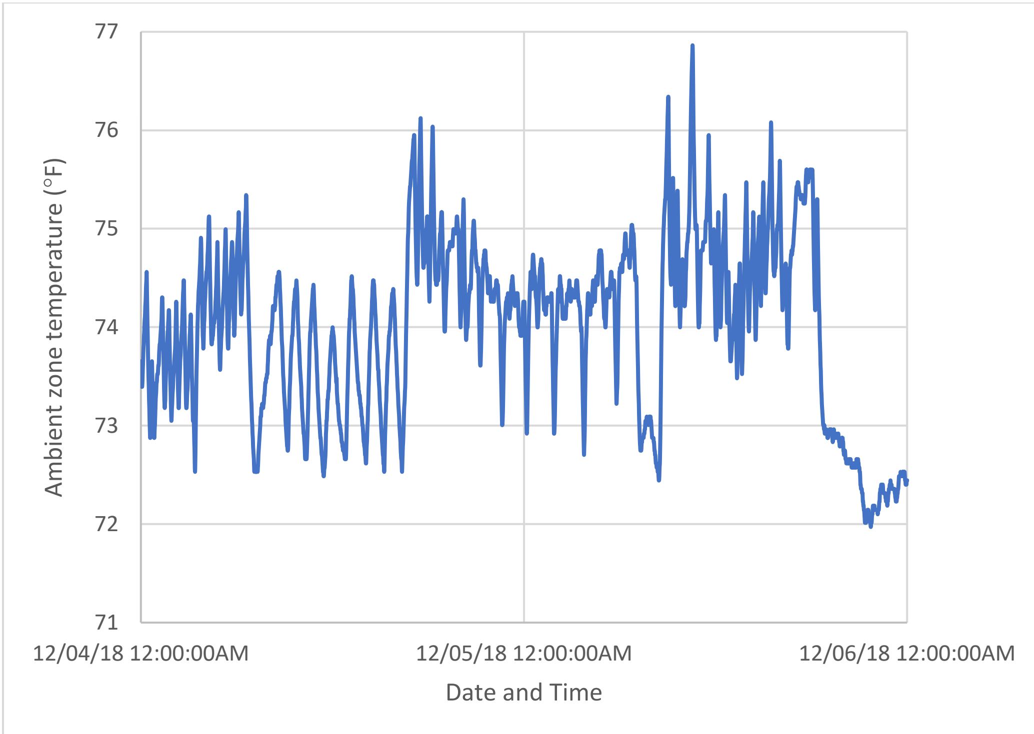

Figure 89: Ambient zone temperature for period 2 in site 6

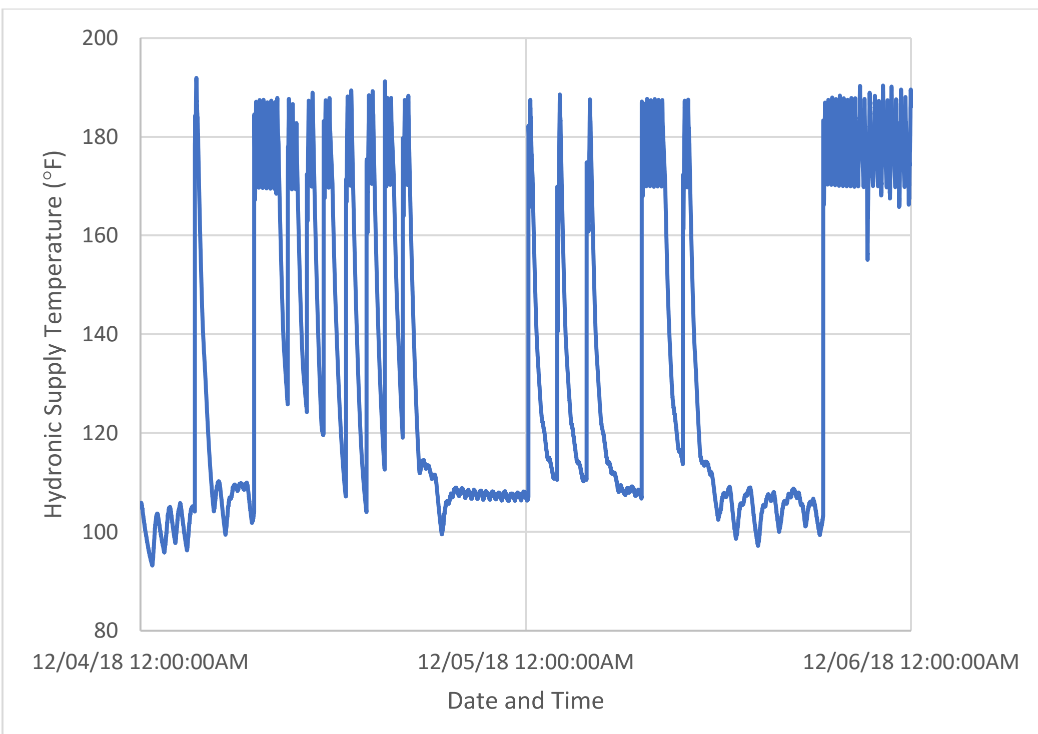

Figure 90: Hydronic supply temperature for period 2 in site 6

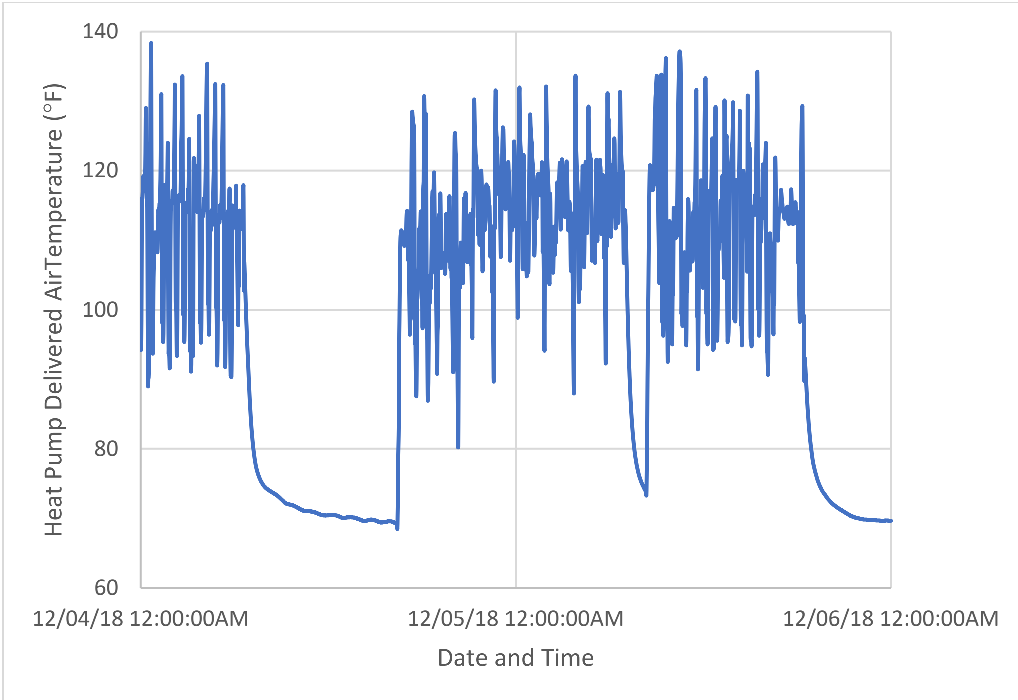

Figure 91: Heat pump delivered temperature for period 2 in site 6

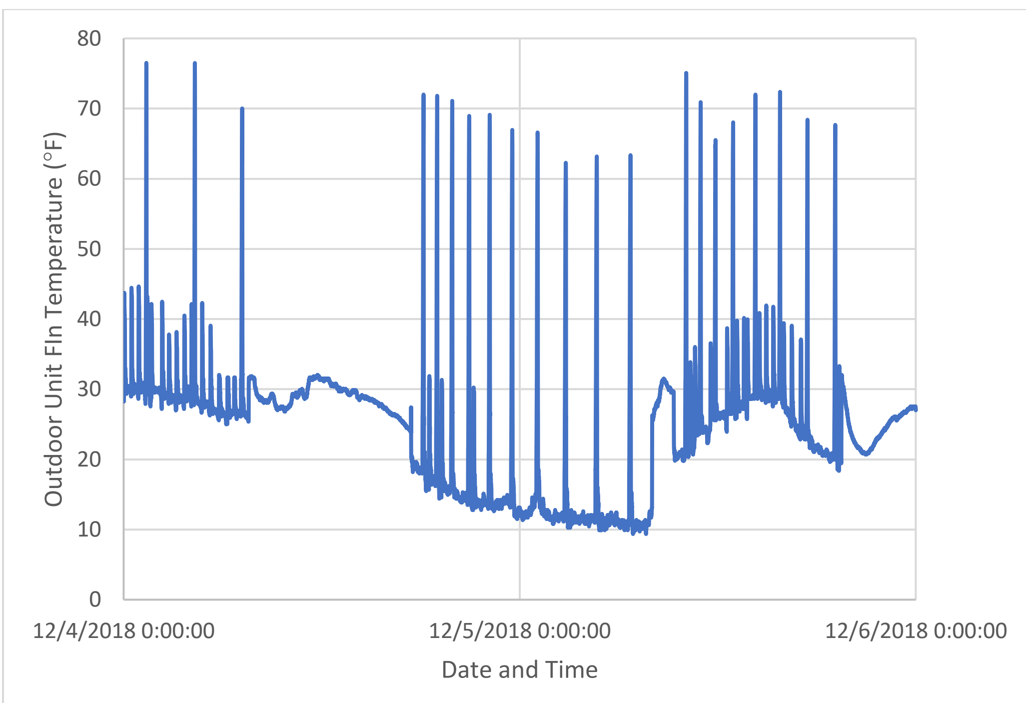

Figure 92: Outdoor unit fin temperature for period 2 in site 6

Site 6 — Period 2 Discussion

The ambient zone temperatures shown in *Figure 89* are similar to those of period 1, albeit slightly lower in some instances. This suggests continuity of thermostat settings from the previous period. A comparison of *Figures 90* and *91* show that the heat pump provided heat more often than the hydronic system did. There are two distinct periods where the hydronic system provided heat while the heat pump was not running. When the heat pump does provide heat, the outdoor unit defrosts periodically. The periodic defrost cycles generally occur every 2–3 hours (~4 minutes each) with high (greater than 60°F) peak temperatures, whereas low (~40°F) peak defrost cycles occur in between every 15–30 minutes (3 minutes each). The length of each defrost cycles was the same from the previous period.

Site 6 — Period 3 Temperature Plots

Figures 93–98 below show plots of ambient zone, hydronic supply, heat pump delivered air and outdoor unit fin temperatures for period 3 along with sub-plots showing temperature behavior over shorter periods of time when necessary.

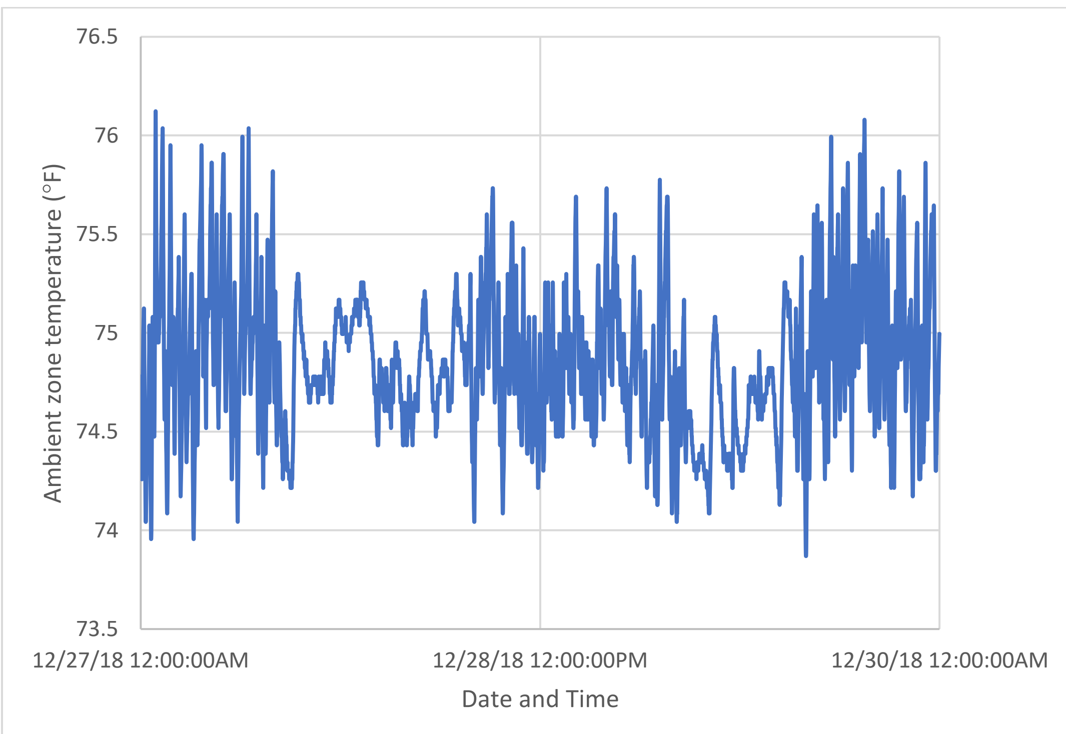

Figure 93: Ambient zone temperature for period 3 in site 6

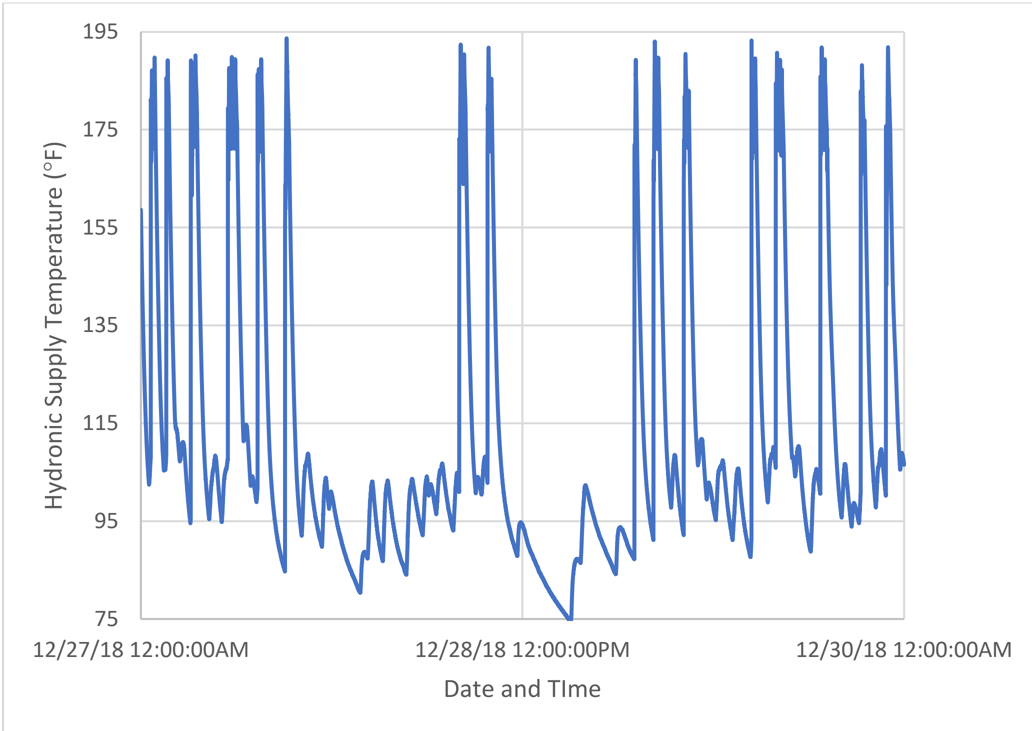

Figure 94: Hydronic supply temperature for period 3 in site 6

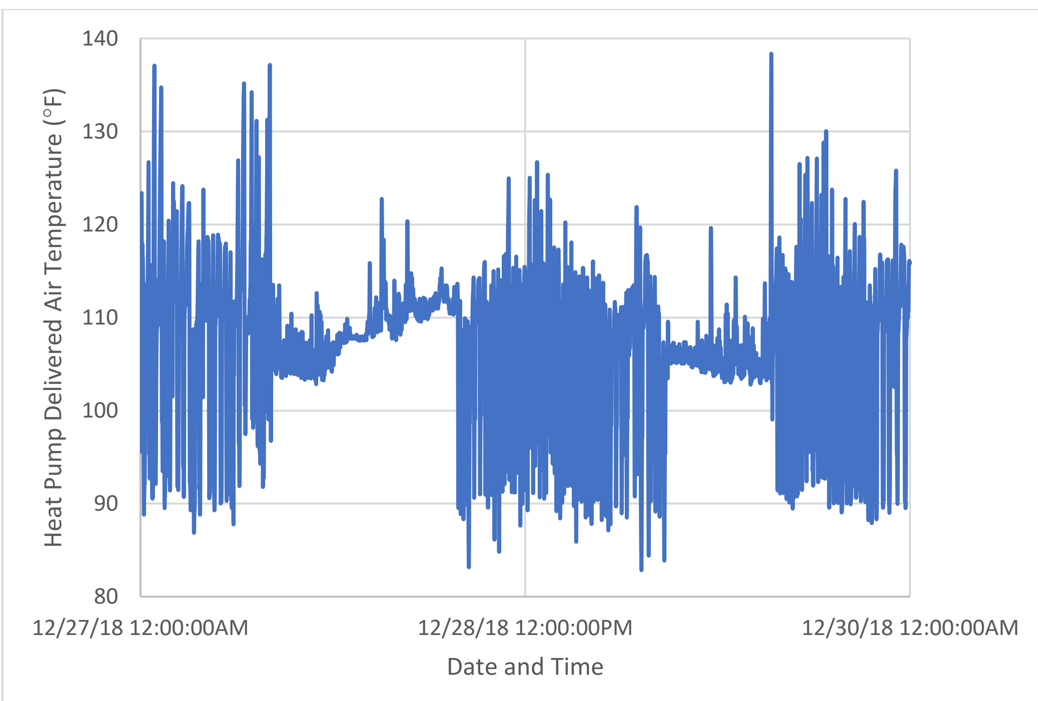

Figure 95: Heat pump delivered temperature for period 3 in site 6

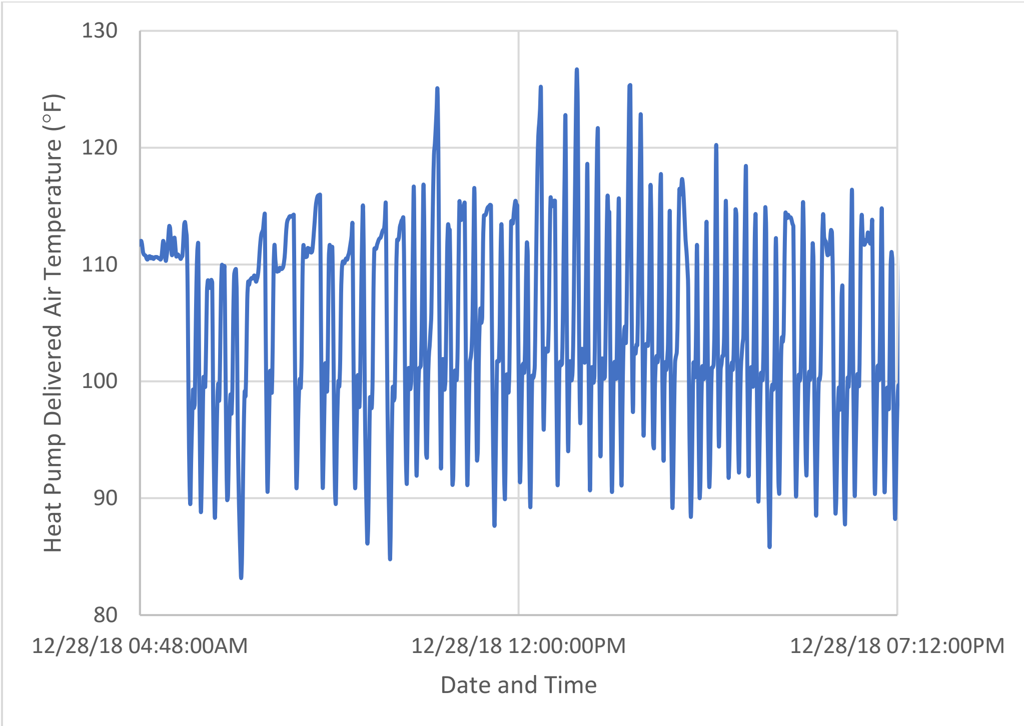

Figure 96: Sub-plot of heat pump delivered air temperature for period 3 in site 6

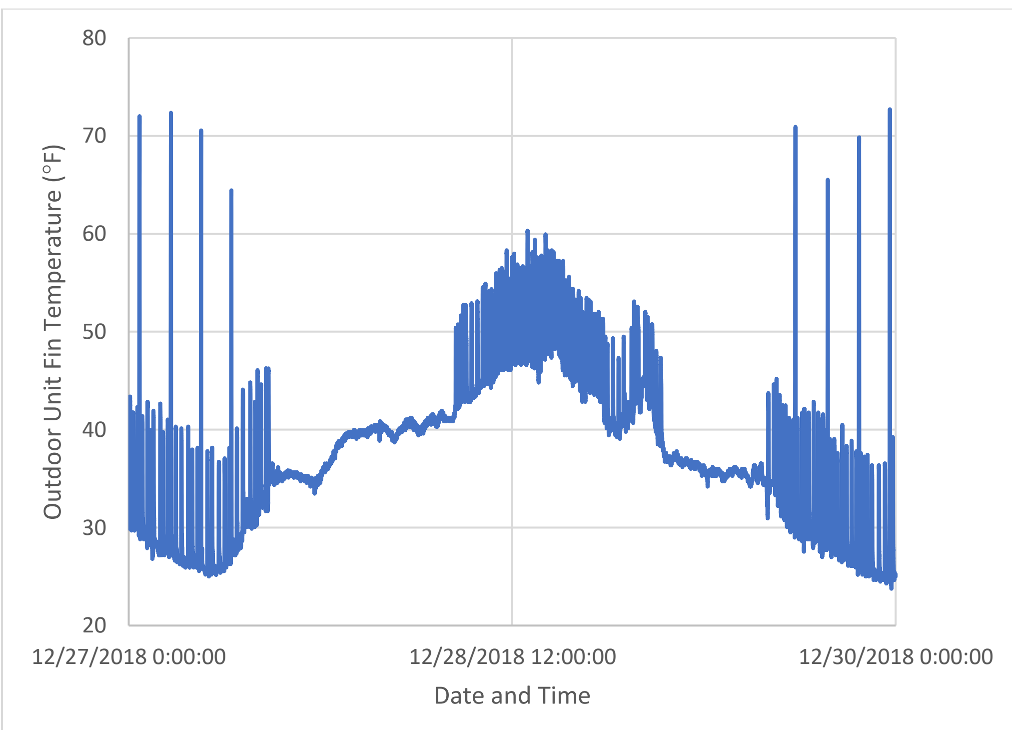

Figure 97: Outdoor unit fin temperature for period 3 in site 6

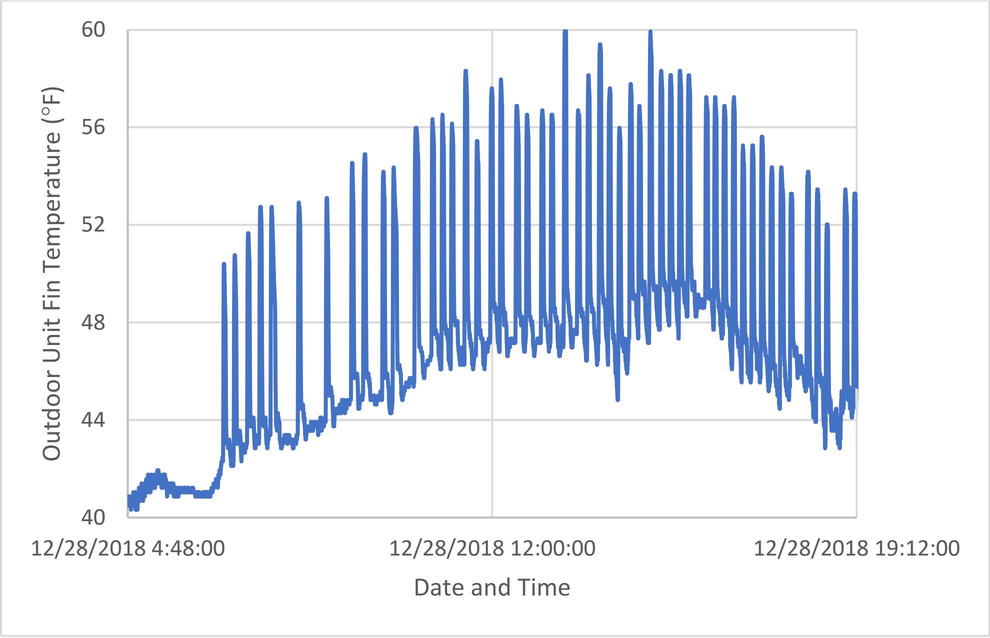

Figure 98: Sub-plot of outdoor unit fin temperature for period 3 in site 6

Site 6 — Period 3 Discussion

The ambient zone temperatures for this period shown in *Figure 93* vary between ~74°F and ~76°F throughout the period. This is consistent with the other two periods and with the thermostat settings that the homeowner described. The hydronic system does not provide heat very often in this period, as seen in *Figure 94*. The heat pump, however, delivers warm air for the entirety of the period, as seen in *Figure 95*. There are two prolonged periods when there it provided warm air without cycling at all. Those two periods correspond with times when the outdoor unit did not defrost at all, as seen in *Figure 97*. For the remainder of the period, however, defrost cycled with low (~40°F to ~55°F) peaks occurred every 15 to 30 minutes. Higher peak temperature defrost cycles also took place towards the beginning and end of the period. All defrost cycles last between 2 and 3 minutes.

Discussion

Analysis of the data from the sites and homeowner reported use patterns shows that control "strategies" vary greatly. In fact, none of the sites utilized the same control strategies. For example, occupants of site 1 use the heat pump as their primary form of heating at most times, while those of site 4 only use the heat pump for comfort when they occupied the zone. These variations, however, not only exist from site to site, but can also be observed in day-to-day operation for each site. This is seen in site 3, where the homeowners incrementally increased the thermostat setting for the hydronic system from the baseline temperature of 72°F on most days for comfort reasons.

Coefficient of Performance (COP), capacity, and defrost patterns of air source heat pumps vary with outdoor temperature and this is clear from the test data. Sites 1, 5, and 6 all used the heat pump consistently on colder days (outdoor temperatures less than 22–27°F). In comparison, sites

2 and 4 used the heat pump for short periods during the day under similar cold conditions. As a result, a significant increase in heat pump operation time is observed for sites 1, 5, and 6 in cold weather compared to those of other sites (and warmer periods). Moreover, peak heat pump delivered air temperatures are generally higher during colder periods at those sites where there was high heat pump use. This is particularly apparent for site 5, and less so for sites 1 and 6. It must be noted that site 1, in particular, showed this increase in delivered air temperature as utilization of the hydronic system decreased. As this phenomenon becomes more prevalent due to increased load (decrease in outside ambient temperatures), both Coefficient of Performance (COP) and heating capacity are reduced [1]. During these periods, the outdoor units for each site change to clearly operating defrost cycles at regular intervals. The length of the cycles varied between 4 and 5 minutes depending on the site. This behavior is consistent with listed control methods at temperatures below 32°F in the respective manufacturers' guides, and with those of various other manufacturers whose products were not a part of this study. The pattern of defrost cycles were very similar in a less cold, yet below 32°F, periods. Added between this pattern were more frequent, lower peaking defrost cycles which lasted 2 to 3 minutes depending on the site. (Note: these observations on Coefficient of Performance (COP) and capacity are based on typical heat pump performance and published performance curves for the tested equipment and not on Coefficient of Performance (COP) and capacity measurements).

A performance comparison for colder periods can then be made between sites leaning towards the use of heat pump for daily heating (sites 1, 5, and 6) and those that mostly utilized the heat pump only for certain parts of the day (sites 2 and 4). Especially interesting of the latter two is site 4, which had the coldest outdoor temperatures (period 1) of any site in this test, where reliance on the heat pump for a few hours each day resulted in a very consistent warm air supply with little to no defrost cycles performed by the outdoor unit. The same is also observed for the outdoor unit for site 2. The hydronic systems for both sites were able to keep up with the regular heating demands of the zone. Even though a decrease in overnight indoor zone temperature for site 2 during the coldest period is observed, it is not due to a failure of the hydronic system to provide heat but rather consistent with the homeowners' implementation of nighttime setback (also observed in many other instances). This is the only site that utilized an automatic nighttime setback. Others, such as site 5, manually shut off their heating system or lowered the temperature setting occasionally. The hydronic system for site 3, which was the sole heating source for the whole test period except one short period, was able to keep up with the highly fluctuating heating demands during the coldest period. For all sites examined, it cannot be concluded that the heat pump could not keep up with heating demands on cold days. This is because the homeowners operated the system in a way that they were comfortable either through the use of the hydronic system and/or setback.

The performance of the heat pump systems changed over the course of the year, obviously because of the change in ambient outside conditions. Beyond outdoor temperature, part of this performance change was also due to a general shift of increased usage of the heat pump compared to the hydronic system. Sites 1 and 6 displayed such behavior and this can easily be determined from less frequent supply of heat from the hydronic system in the warmer periods. In fact, site 1 was almost solely dependent on the heat pump for heat. This increase in load caused the heat pump to deliver high temperature air, as was the case when loads increased due to low outdoor ambient temperatures.

Periods with warmer outside ambient temperatures allowed the heat pumps for sites 1,5, and 6 to operate for longer periods of time without the outdoor unit shifting to defrost mode. However, when the units changed to a mode where defrost was needed, they did so very frequently. The worst-case scenario in this circumstance (site 1 during period 3) showed low peak defrost cycles every 10-15 minutes lasting ~3minutes each even when the ambient outdoor temperature was between 40 and 50°F. The outdoor unit for site 5 also displayed similar behavior in similar outdoor weather in period 2, albeit at a much lower frequency. Since the hydronic heating was not a factor for site 5, delivered air temperatures for the heat pump during the warmer period decreased because of a lower load placed on the heat pump compared to the colder period(s). Site 6 showed the least heat pump performance change in this site cluster between the colder and warmer periods. The outdoor unit, however, repeated the more frequent, lower-peak defrost patterns observed in sites 1 and 5.

Warmer periods for site 2 did not prevent the frequent defrosting patterns for its outdoor unit despite operation of the heat pump for short periods of time. During period 3, when outside ambient temperatures were mostly above ~32°F and below ~40°F, the outdoor unit for this site operated many frequent defrost cycles during the times the heat pump was operating. The frequency of the defrost cycles was, however, lower than those of sites 1, 5 and 6 in the same weather conditions. The outdoor unit for site 4, however, did not perform any defrost cycles other than a few isolated ones throughout the warmer periods. It must be noted that the heat pump for this site generally operated for 2-3 hours at a time, compared to longer ranges of time for the other sites.

The pattern of frequent defrost cycles as seen in sites 1, 5, and 6 due to heat pump operation for prolonged periods of time decreases efficiencies because the system operates and consumes energy in reverse mode to rid the outdoor unit of frost while providing no heat. Despite being able to keep up with heating demands, the heat pump systems spent a significant amount of time defrosting. This only seems to occur for temperatures slightly higher than 32°F up to the low 40°F range, and, in the worst-case scenario, up to 50°F. This appears to be a control method utilized by the outdoor unit in those weather conditions when chances of frost buildup would be high and accompanied with near-freezing temperatures, increasing the chances of frost forming on the fins. There was, however, no specific pattern observed for such behavior in any of the sites.

The outdoor unit avoided defrost cycles for a lengthy period of time despite continuous operation of the heat pump is generally on days warmer than ~32°F, seen especially in site 1 during period 3, and site 5 during period 2 and site 6 during period 3. For sites that utilized the hydronic system primarily during the warmer periods (sites 2 and 4), it is seen that the thermostat for the respective zones was satisfied at most times. That is because boiler design is performed for the worst-case scenario, which causes a non-modulating system to be highly oversized on warmer days. Therefore, utilizing the heat pump as the primary source of heating on such days would prompt less circulation of hydronic water and, hence, less cycling of the boiler.

Further examination of outdoor unit temperature data was performed in order to better understand defrost patterns as a function of temperature. The relevant data for site 5 was chosen for this portion of the analysis because it was the only site where the hydronic system does not provide heat. This means that patterns of defrost cycles performed by the outdoor unit in this site were not skewed by varying heat pump load due to hydronic system operation. For this analysis, outdoor unit fin temperatures were analyzed to identify frequency of defrost cycles per hour for various outdoor temperatures during the test period. For select outdoor temperatures varying between ~-3°F and ~57°F, periods with most frequent defrost cycles were chosen. The frequency of defrost cycles in each of the periods was then attributed as the "Maximum Defrost Cycles Per Hour" for the respective temperature. This metric is useful because it identifies the worst case scenario in terms of defrost cycles. If, instead, an average value was presented, it would not be a fair representation because that value would be skewed due to absence of defrost cycles when the heat pump was either under low to zero load. The maximum cycle rates are shown in the figure below as a function of outdoor temperature.

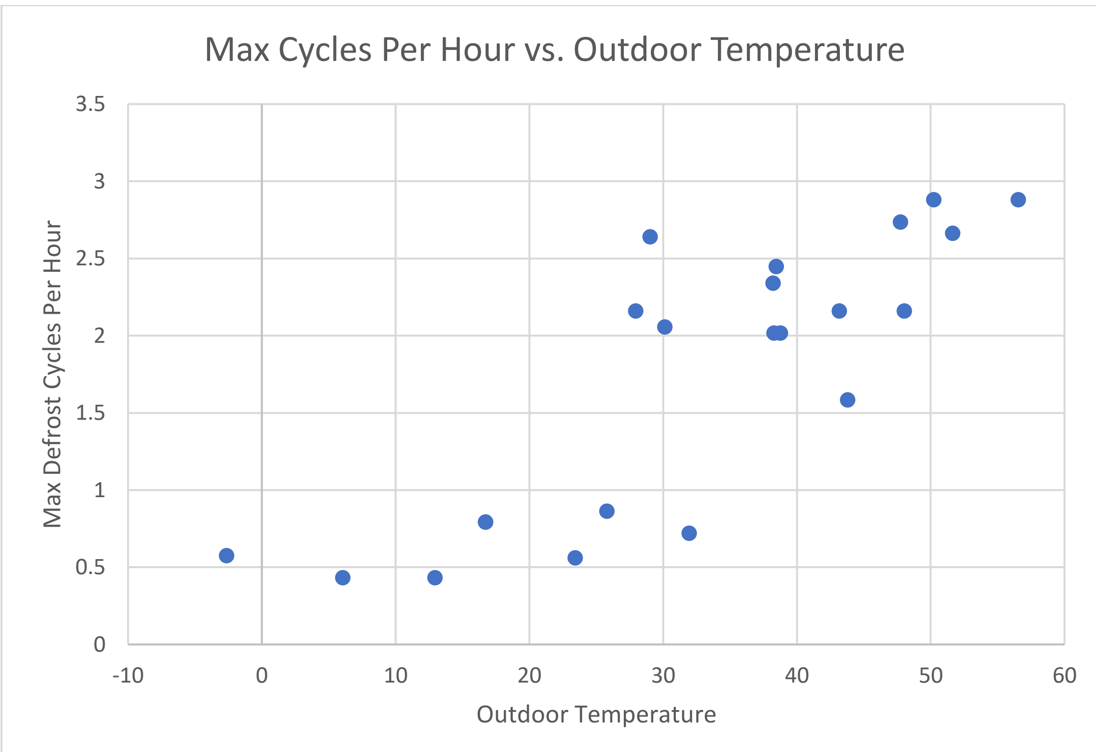

Figure 99: Maximum defrost cycle rates as a function of outside ambient temperatures

While the correlation above does not suggest a pattern that would provide a reliable curve fit, it does corroborate the point made previously that frequency of defrost cycles peaks above ~32°F. Most of those cycles, as discussed previously, have lower peak temperatures compared to defrost cycles that occur below ~32°F.

Based on the system performances observed and deduced, certain methods of operation were identified that offer more adequate heating while possibly providing more efficient operation. When it is colder than ~32°F, it is more efficient to operate the heat pump occasionally and, if possible, avoid prolonged run time. This method can greatly lower the number of defrost cycles that take place (both less frequent, high peak cycles and more frequent, low peak ones), which lowers energy losses. Lower usage of the heat pump will require the hydronic system to provide more heat. This will ensure the supply of more heat produced by the boiler to a zone rather than to losses to its surroundings. During milder weather (i.e., warmer than 32°F and up to 40 or 45°F), the heat pump can be allowed to operate more often, especially because actual Coefficient of Performance (COP) at those temperatures is higher than those in colder weather. However, to avoid the low peak defrost cycles that were observed, operation of the heat pump at each time should be limited to 2–3 hours for intermittent heating. It must be noted that this period of time may possibly be much longer if the load for heating is decreased. As outside ambient temperatures climb higher than ~40°F, more of the heating requirement can be shifted to the heat pump system since this is the temperature range where it will operate most efficiently and avoid any defrost cycles from occurring. Further energy savings can be achieved if the boiler is then shut off when there is no dependence on hydronic heating unless it provides domestic hot water.

Annual Performance Analysis

This study conducted an annual performance analysis of houses heated by both heat pumps and oil-fired hydronic heating systems for various climate zones in New York State. This analysis was based on a 2,500-ft2 typical ranch-style house whose energy and temperature parameters were generated using Energy-10 software. This theoretical house was assumed to have standard code, including 2 x 4 stud wall construction with basement and roof. The roof has an R30 insulation level. The daily temperature setpoint was 70°F with nighttime setback of 63°F. Using this information and local climate data from three New York State regions, the software could determine heating load, indoor temperature and outdoor temperatures on average for each hour of the year. These parameters were obtained for the upstate, mid-state and downstate regions. Each region was represented by the following locations/cities, respectively—Long Island, Albany, and Ottawa. The last of the three regions was used in place of Upstate New York since it closely resembled the climate in Upstate New York.

The generated data could then be used as a baseline for the simulation. The chosen software for the simulation was MATLAB. A code was written to specify boiler and heat pump performance parameters and assign either or both of the systems to satisfy the heating load at any given time. The assignment of heating loads to the heating systems were governed by certain control methods that provided conditions for operating each system. The following sections provide an explanation of the various systems and control methods used.

Operating Parameters and Assumptions

Several heat pump and oil-fired boiler systems were chosen for the performance analysis of the hybrid heating systems. Various control methods were then formulated to simulate heating cost for any combination of boiler and heat pump. Each method had unique conditions that dictated load distribution between the two systems. Either the boiler or heat pump system would provide all the heat for a given hour depending on which conditions are met. Boiler sizing was performed such that it would be greater than that of the maximum hourly load. This is highly representative of what occurs in the field. Oversizing of the boiler means that it could cover 100% of the load whenever it is asked. Four heat pump units were assigned in every instance, each intended to cover 25% of the load of the entire home. Since the capacity of these units decreases as it gets colder, the boiler was allowed to idle and standby as back up. This resulted in an additional cost of operating the boiler and must be considered when viewing the results. For the type of hydronic systems being considered here, it is common for these to provide domestic hot water as well as space heating. It is common for these boilers to remain hot even when there is no heating load.

Heating load during generally warmer months was limited for all modes by avoiding use of any form of heating between May 15 and October 1 even if the outdoor temperature was below a point where heat might be required. During this period, it was assumed that a short period of colder outdoor weather would not motivate the occupants to activate the heating system. The remainder of the year can be considered as part of the heating season and will be referred as such from this point on. Operating principles and descriptions of each control method is provided below.

Control Method 1 allowed only the boiler to provide heat during the entire heating season. This was chosen as a baseline for comparison against the other methods, all of which include some combination of the boiler and heat pump systems. The simulation code, when prompted to perform calculations with this method, assigned the load for every hour of the heating to the boiler. Total boiler energy output and efficiency can be calculated from the load using the boiler load equation presented in the following section. The output of the boiler is then utilized by the program to calculate the fuel input and cost of heating per hour and, ultimately, annual cost of heating.

In the analysis, a fixed percentage of the total heat load of the building is assigned to the zone which is heated by the heat pump under evaluation. This load could also be split into any number of heat pumps if desired. As mentioned previously, the entire heating load was split into four units, each carrying 25% of the load. This is equivalent to an assumption that four identical ductless heat pumps are used to meet the full load of the house. Any time the conditions for heat pump operation were met, the assigned load was divided by four. This value was then compared to the heating capacity of the heat pump, which was calculated based on the outdoor temperature. Loads that were less than the capacity were assigned to the heat pumps. In the case that the load was larger than the capacity, the portion that the heat pumps could not cover was satisfied by the boiler. The boiler would also satisfy the entire load when the operating conditions did not call for heat pump operation. The cost of heating using each system could then be calculated based on the prices.

The heat pumps acted as the primary source of heating for the entire heating season for control method 2 with the boiler acting as backup as discussed above. In the case of control method 3, the heat pump would be the heating source any time the outdoor temperature was at or above a chosen crossover point, which was set as 25°F for this analysis. Control method 4 made the decision of primary heating based on the time of the year. Between March 1 and November 1, the heat pump would attempt to provide all the heat, while the remaining time it would be the boiler. For all these methods, the code calculated, using the load and Coefficient of Performance

(COP) for each hour, a value to total energy consumed by the heat pump system. Based on the energy consumed and cost of electricity, the total cost per hour was also calculated.

A study done by the National Renewable Energy Laboratory [2] evaluated performance for test heat pump units for various operating conditions. Part of the study included calculating Coefficient of Performance (COP) degradation at different temperatures due to defrost cycles, which were utilized to estimate reduction in Coefficient of Performance (COP) for the simulation of heat pump performance. The following are the temperature ranges and corresponding approximations of percentage degradation due to defrost cycles in Coefficient of Performance (COP):

Table 8: Percent degradation in Coefficient of Performance (COP) for various temperature ranges

Temperature	***% Degradation***
T ≤ 5°F	10
5°F ≤ T ≤ 17°F	5
17°F ≤ T ≤ 35°F	3
T ≥ 35°F	0

An equation for Coefficient of Performance (COP) losses due to cycling was also utilized from a model created to analyze partial load behavior in heat pump systems [3]. The following is the equation used to obtain a part-load factor, or *PLF*:

$$PLF = 1 - C_d(1 - PLR),$$

where C_d is a degradation coefficient due to cycling, usually assumed to be 0.25 and PLR is the part-load ratio calculated by finding the ratio between the load and 30% of the maximum capacity (minimum modulation point), given that the latter is less than the former. Once the *PLF* is found, the degraded COP or COP_{deg} can be found using following equation:

$$COP_{deg} = COP \cdot PLF$$

Cost of fuel and electricity were chosen from the most recent average for New York State on the NYSERDA website add link. Price of no. 2 Oil was chosen as $2.39 per gallon, which was the state average for August 2020. Price of electricity used was $0.191 per kilowatt-hours (kWh), which was the state average for June 2020. It must be noted that wholesale prices of heating oil are generally lower (~20%) than the value used.

Boiler Modeling

For this analysis, a linear input/output model for boiler efficiency was used and this is based on prior studies of residential systems and a boiler model being used in ASHRAE Standard 155P, currently under development [4] [5] [6]. For a specific boiler, the inputs for this model include:

1. Idle Loss or ILP (%) — Rate of energy loss (energy input needed) from the boiler under zero load expressed as a percentage of the full load, steady state burner heat input rate.
2. Maximum Output or $Output_{max}$ (Btu/hr) — this is the rated maximum full load steady-state heat delivery rate from the boiler

3. η_{ss} (%) — this is the full load, steady state delivered efficiency of the boiler. This includes flue gas sensible and latent heat losses and boiler jacket losses.

The linear input/output model is expressed as:

$$Input = a \cdot Output + b$$

Where:

- Output (Btu/hr) — Output rate at a partial load point, average over time including on and off periods.
- Input (Btu/hr) — Input rate at a partial load point, average over time including on and off periods.
- a,b — constants

The constants a and b can be determined from the three inputs above using the equations below:

$$Input_{max} = Output_{max} \cdot \frac{100}{\eta_{ss}}$$

$$Input_{zero\ load} = Input_{max} \cdot \frac{ILP}{100}$$

$$\mathrm{a} = \frac{Input_{max} - Input_{zero\ load}}{Output_{max}}$$

$$b = Input_{zero\ load}$$

For this analysis, the full load, steady state boiler efficiency was assumed to be 86% and 90% in different case studies, which represent a typical modern boiler and a high efficiency common boiler, respectively. Values of the $Output_{max}$ parameter are based on typical sizes of boilers sold for this market. Based on prior studies *ILP* ranges from 0.2% for a well-insulated boiler with modern controls to ~ 4% for an older boiler which remains hot at all times and is poorly insulated. The older boiler would be representative of a unit which should soon be replaced. The following table provides the operating parameters for each boiler used in this analysis:

Table 9: Performance Parameters for the Various Boiler Models Used in the Simulation

Boiler	**Max Output (Btu/hr)**	**Idle Loss (%)**	**η_{ss} (%)**
1	120000	1.0	86
2	84260	0.2	90
3	84260	0.2	86

Heat Pump Modeling

Several of the heat pumps for the simulation were chosen from the NEEP Cold Climate Air-Source Heat Pump database (https://ashp.neep.org/#!/). Data such as capacity and Coefficient of Performance (COP) at various temperatures were obtained from this database for some units. Reports on field studies of the performance of heat pumps were also used as a source of data for this analysis. Using the combination of the two sources, eight distinct sets of model heat pumps for the analysis were generated. Upon further analysis of the parameters of these models, it was concluded only five out of eight of the generated "units" would be used in the actual analysis. The following sections provide details of each heat pump model and how the units were chosen.

Heat Pump 1

The heat pump is manufactured by Daikin. The indoor unit model is FDMQ24RVJU and the outdoor unit model is RX24RMVJU. The maximum capacity at temperature points of 5°F, 17°F, and 47°F were obtained from the NEEP database. The Coefficient of Performance (COP) at the maximum capacity for the respective temperature was used for the first two points and Coefficient of Performance (COP) at minimum capacity was used for the third point. Capacity and Coefficient of Performance (COP) values at various outdoor temperatures between the low and high temperatures were extracted using interpolation. Extrapolation was used to obtain values outside that range. This unit was chosen as an example heat pump with average capacity and high Coefficient of Performance (COP) values compared to the rest of the units. All data points are shown below:

Table 10: Performance Parameters for Heat Pump 1

HP Temp Points (°F)	*HP Capacity Points (BTU/hr)*	*HP COP Points*
5	13760	2.24
17	17250	2.69
47	27600	4.82

Heat Pump 2

This unit is manufactured by CanAir. The indoor unit model is C19SEH18H21 and the outdoor unit model is C19SCH18H21. The maximum capacity at temperature points of 5°F, 17°F, and 47°F were obtained from the NEEP database. The Coefficient of Performance (COP) at the maximum capacity for the respective temperature was used for the first two points and COP at minimum capacity was used for the third point. Capacity and Coefficient of Performance (COP) values for various temperature points were calculated using the same method as that for heat pump 1. This unit represents a low capacity and low Coefficient of Performance (COP) heat pump. All data points are shown below:

Table 11: Performance Parameters for Heat Pump 2

HP Temp Points (°F)	*HP Capacity Points (BTU/hr)*	*HP COP Points*
5	10198	1.97
17	11041	2.2
47	18278	3.22

Heat Pump 3

This is the same unit as heat pump 1. The same capacity values were used, but the Coefficient of Performance (COP) values were different. A linear regression was performed on performance data from a field study done by Williamson and Aldrich [7], which provided a relationship between Coefficient of Performance (COP) and outdoor temperatures. The data drawn from this study was from a combination of two sites that provided the most numerous data points that were obtained in the outdoor temperature ranges used in the simulation. This linear relation was utilized to calculate Coefficient of Performance (COP) above the low temperature point of 5°F. A Coefficient of Performance (COP) value of 1 was assigned for temperatures below that point. Capacity values at various outdoor temperatures between the low and high temperatures were extracted using interpolation. Extrapolation was used to calculate values outside that range. The results of the extrapolation were values that more closely represented performance in the field. The following are the data points used for the model heat pump in this analysis:

Table 12: Performance Parameters for Heat Pump 3

HP Temp Points (°F)	***HP Capacity Points (BTU/hr)***	***HP COP Points***
5	13760	1.09
17	17250	1.43
47	27600	2.3

Heat Pump 4

This unit is the same as heat pump 2, as are the values for capacity at each temperature set point. The Coefficient of Performance (COP) values used for this unit were also obtained in the same method as those of heat pump 3. The following are the data points:

Table 13: Performance Parameters for Heat Pump 4

HP Temp Points (°F)	***HP Capacity Points (BTU/hr)***	***HP COP Points***
5	10198	1.09
17	11041	1.43
47	18278	2.3

Heat Pump 5

This unit is manufactured by Fujistu. The indoor unit model is ASU12RLS3Y and the outdoor unit model is AOU12RLS3H. The maximum capacity at temperature points of 5°F, 17°F, and 47°F were extracted. The Coefficient of Performance (COP) at the maximum capacity for the respective temperature was used for the first two points and COP at minimum capacity was used for the third point. Capacity and Coefficient of Performance (COP) values for various temperature points were calculated using the same method as that for heat pump 1. The following are the data points:

Table 14: Performance Parameters for Heat Pump 5

HP Temp Points (°F)	HP Capacity Points (BTU/hr)	HP COP Points
5	16500	2.15
17	17500	2.32
47	22110	4.54

Heat Pump 6

This unit is manufactured by Mitsubishi. The indoor unit model is MSZ-FE12NA and the outdoor unit model is MUZ-FE12NAH. The maximum capacity at temperature points of 5°F, 17°F, and 47°F were extracted. The Coefficient of Performance (COP) at the maximum capacity for the respective temperature was used for the first two points and COP at minimum capacity was used for the third point. Capacity and Coefficient of Performance (COP) values for various temperature points were calculated using the same method as that for heat pump 1. The following are the data points:

Table 15: Performance Parameters for Heat Pump 6

HP Temp Points (°F)	HP Capacity Points (BTU/hr)	HP COP Points
5	12500	2.3
17	15100	2.34
47	21000	5.86

Heat Pumps 7 and 8

These two units contain the same capacity values that of heat hump 1. The Coefficient of Performance (COP) values, however, were derived from field data obtained and reported in a study done on over 100 homes across Massachusetts and Rhode Island [8], which included cold-climate and non-cold-climate heat pumps. The report contains plots of average Coefficient of Performance (COP) values of all homes in a group at various outdoor temperatures.

A best fit line was obtained from the cold-climate heat pump Coefficient of Performance (COP) data in the report. These values were used for heat pump 7 Coefficient of Performance (COP) points. The same action was performed for the non-cold-climate air-source heat pumps and used as Coefficient of Performance (COP) points for heat pump 8. The following are the data points for both heat pumps:

Table 16: Performance Parameters for Heat Pumps 7 and 8

HP Temp Points (°F)	HP Capacity Points (BTU/hr)	HP7 COP Points	HP8 COP Points
5	13760	2.1	1.6
17	17250	2.47	1.86
47	27600	3.4	2.5

A search of field studies was conducted to obtain performance parameters in the field that confirm the manufacturer provided values from the NEEP website. Only a handful of studies provided performance curves for a varying range of sites and products. The curves chosen in this study to reflect field data were determined to have a few criteria. The first of those is a dataset that empirically demonstrated repeatability within the study period. Another criterion is the use of a relatively large sample size. For example, reported Coefficient of Performance (COP) values from a study containing only two units tested in lab conditions was not utilized. The chosen studies also demonstrated data that was not skewed or interrupted due to factors such as equipment malfunction, extreme events and others.

After comparing the data from the field studies and the manufacturer values obtained from NEEP, it was concluded that latter of the two was impractical and most likely could not be found in the field. Therefore, heat pumps 5 and 6 were not part of the final analysis. Heat pump 1 was, however, used for the analysis to reflect discrepancies between theoretical and field data. Heat pump 2 was also kept as part of the analysis. This was due to two reasons: firstly, to show output based on a heat pump with relatively lower capacities and secondly, Coefficient of Performance (COP) values most closely resembled field data. Reduced Coefficient of Performance (COP) values were calculated for heat pumps 1 and 2 since they do not account for losses due to cycling and defrost cycles. The Coefficient of Performance (COP) degradation equations due to cycling and defrost derived in a previous section were applied to find the reduced Coefficient of Performance (COP) values.

Based on the above heat pump data and Coefficient of Performance (COP) degradation percentages, a plot of the actual Coefficient of Performance (COP) as a function of outdoor temperature was made for all five heat pumps analyzed in the study. The plot is shown below.

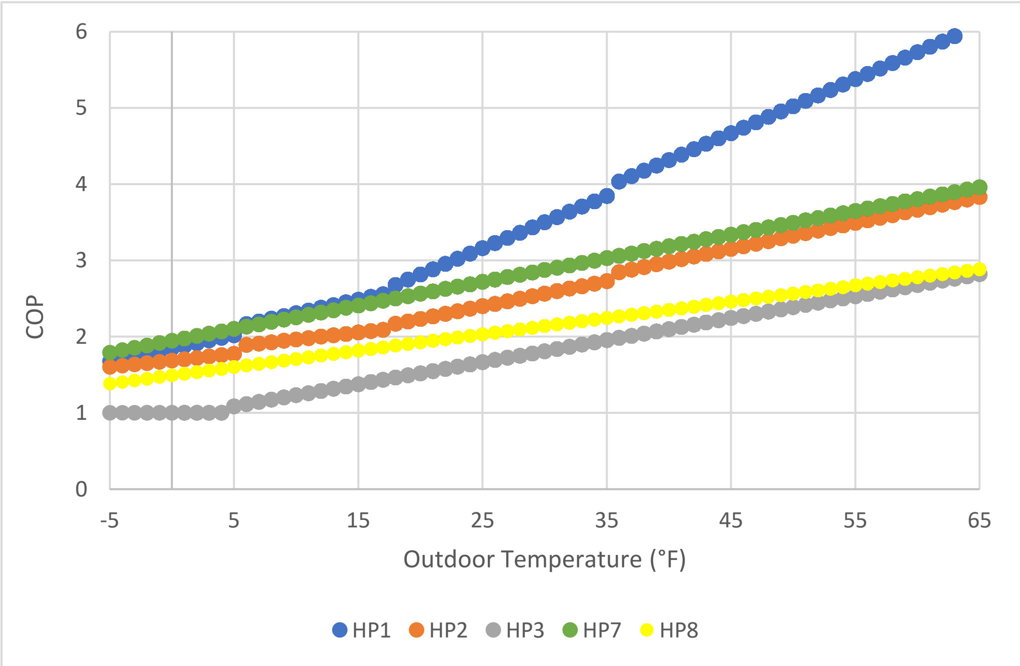

Figure 100: COP as a function of outdoor temperature after accounting for degradation due to defrost cycles

Data and Analysis

Mid-State Region Data

The flexibility of the code allowed for data output for any combination of inputs of region, boiler, heat pump and control method. Give the multitude of data obtained, it is best to present them region by region. The first region examined is mid-state (Albany). This region was deemed as the focal point of the annual performance analysis since it is roughly representative of "average" climate for New York State. As a precursor to heating cost for a combination of boiler and heat pump system, analysis of standalone boiler systems was performed, the output of which is shown below.

Table 17: Annual Heating Cost for Various Boiler Models in Mid-State Region

Boiler Model	***Annual Heating Cost ($)***
Boiler 1	1953
Boiler 2	1774
Boiler 3	1857

Annual performance data was simulated for all heat pumps that are part of the analysis in combination with boiler 1 (lowest efficiency boiler) for control methods 2, 3, and 4. The following is the annual performance data for heat pumps 1 and 2, each working with boiler 1 using control methods 2–4. The boiler and heat pump (or HP) load expressed in percentage represents the total portion of the load that is covered annually by each system. Boiler backup time is a value obtained by finding the number of hours the total load is greater than the total capacity of the heat pumps. This is the instance when the boiler covers the load that cannot be satisfied by the heat pumps.

Table 18: Annual Performance Data for Heat Pumps 1 and 2 Combined with Boiler 1 in Mid-State Region

Heat Pump/Boiler Unit	***Control Method***	***Boiler Load (%)***	***HP Load (%)***	***Backup Time (hours)***	***Boiler Cost ($)***	***HP Cost ($)***	***Total Cost ($)***
Boiler 1	1	100	0	N/A	1953	0	1953
HP 1	2	1.04	98.96	110	149	1704	1853
	3	43.12	56.88	0	950	803	1753
	4	71.64	28.36	0	1492	379	1871
HP 2	2	6.39	93.61	455	251	2038	2289
	3	44.06	55.94	125	968	1064	2032
	4	72.55	27.45	86	1510	486	1996

Heat pumps 1 and 2 were grouped together in the above table since they are the only units with data that is not representative of the field. Heat pumps 3, 7, and 8 contain data derived from the field and will be grouped together. The annual performance data for those units combined with boiler 1 for each control method is shown below.

Table 19: Annual Performance Data for Heat Pumps 3, 7, and 8 Combined with Boiler 1 in Mid-State Region

Heat Pump/Boiler Unit	***Control Method***	***Boiler Cost ($)***	***HP Cost ($)***	***Total Cost ($)***
Boiler 1	1	1953	0	1953
3	2	149	3171	3320
	3	950	1583	2533
	4	1492	743	2235
7	2	149	1906	2055
	3	950	961	1911
	4	1492	448	1940
8	2	149	2548	2697
	3	950	1295	2245
	4	1492	602	2094

The boiler and heat pump load percentages are not shown in the above table because for each control method, the respective values are the same as those of heat pump 1. This occurs because the annual load percentages depend solely on heat pump capacities, which are the same for heat pumps 2, 3, 7, and 8.

Further analysis was performed on heat pump 7 by combining with boilers 2 and 3 as well. This was only done for the mid-state region. The following is the output data.

Table 20: Annual Cost Data for Heat Pump 7 in Combination with Boilers 2 and 3 in the Mid-State Region

Boiler Unit	***Control Method***	***Boiler Cost ($)***	***Heat Pump Cost ($)***	***Total Cost ($)***
2	1	1774	0	1774
	2	36	1906	1942
	3	808	961	1769
	4	1330	448	1778
3	1	1857	0	1857
	2	38	1906	1944
	3	845	961	1806
	4	1392	448	1840

Upstate Region Data

The next set of data shown is from Ottawa, which represents the Upstate New York region. Temperatures are generally lower in this region than the mid-state region, which means heating loads are also greater. The following table compared annual cost data for each of the boiler models in the upstate region.

Table 21: Annual Heating Cost for Various Boiler Models in Upstate Region

Boiler Model	Annual Heating Cost ($)
Boiler 1	2535
Boiler 2	2335
Boiler 3	2443

Once again, heat pumps 1 and 2 were grouped together, and the annual performance data for those units in conjunction with boiler 1 is shown below, followed by the other heat pumps with the same boiler.

Table 22: Annual Performance Data for Heat Pumps 1 and 2 Combined with Boiler 1 in Upstate Region

Heat Pump/Boiler Unit	Control Method	Boiler Load (%)	HP Load (%)	Backup Time (hours)	Boiler Cost ($)	HP Cost ($)	Total Cost ($)
Boiler 1	1	100	0	N/A	2535	0	2535
HP 1	2	5.63	94.37	430	270	2469	2739
	3	64.03	35.97	0	1729	641	2370
	4	72.5	27.5	39	1940	523	2463
HP 2	2	12.77	87.23	889	448	2742	3190
	3	65.04	34.96	81	1754	840	2594
	4	73.98	26.02	119	1977	638	2615

Table 23: Annual Performance Data for Heat Pumps 3, 7, and 8 Combined with Boiler 1 in Upstate Region

Heat Pump Unit	Control Method	Boiler Cost ($)	HP Cost ($)	Total Cost ($)
3	2	270	4620	4890
	3	1729	1267	2996
	4	1940	1010	2950
7	2	270	2621	2891
	3	1729	771	2500
	4	1940	601	2541
8	2	270	3477	3747
	3	1729	1039	2768
	4	1940	806	2746

Once again, annual performance data was obtained for heat pump 7 alongside boilers 2 and 3 for each control method. The data is shown below.

Table 24: Annual Cost Data for Heat Pump 7 in Combination with Boilers 2 and 3 in the Upstate Region

Boiler Unit	***Control Method***	***Boiler Cost ($)***	***Heat Pump Cost ($)***	***Total Cost ($)***
2	2	36	1906	1943
	3	808	961	1769
	4	1330	448	1778
3	2	38	1906	1944
	3	845	961	1806
	4	1392	448	1840

Downstate Region Data

Long Island is a region with much milder climate compared to the rest of New York State. Lower heating loads and higher temperatures result in overall lower heating costs annually. The data for each boiler system in standalone mode is shown below.

Table 25: Annual Heating Cost for Various Boiler Models in Downstate Region

Boiler Model	***Annual Heating Cost ($)***
Boiler 1	1492
Boiler 2	1330
Boiler 3	1392

The following table shows annual performance data for heat pumps 1 and 2 for the Long Island region.

Table 26: Annual Performance Data for Heat Pumps 1 and 2 Combined with Boiler 1 in Downstate Region

Heat Pump/Boiler Unit	***Control Method***	***Boiler Load (%)***	***HP Load (%)***	***Backup Time (hours)***	***Boiler Cost ($)***	***HP Cost ($)***	***Total Cost ($)***
Boiler 1	1	100	0	N/A	1492	0	1492
HP 1	2	0.01	99.99	4	129	1144	1273
	3	19.48	80.52	0	404	855	1259
	4	72.31	27.69	0	1148	263	1411
HP 2	2	1.91	98.09	196	156	1485	1641
	3	19.66	80.34	63	406	1159	1565
	4	72.51	27.49	23	1150	356	1507

Next, the data for heat pumps 3, 7, and 8 working in conjunction with boiler 1 are shown.

Table 27: Annual Performance Data for Heat Pumps 3, 7 and 8 Combined with Boiler 1 in Downstate Region

Heat Pump Unit	Control Method	Boiler Cost ($)	HP Cost ($)	Total Cost ($)
3	2	129	2226	235
	3	404	1692	2096
	4	1148	526	1674
7	2	129	1343	1472
	3	404	1032	1436
	4	1148	320	1468
8	2	129	1805	1934
	3	404	1392	1796
	4	1148	432	1580

Annual performance data for heat pump 7 was also generated with boilers 2 and 3 for each control method. The following is the cost output for said system combinations.

Table 28: Annual Cost Data for Heat Pump 7 in Combination with Boilers 2 and 3 in the Downstate Region

Boiler Unit	Control Method	Boiler Cost ($)	Heat Pump Cost ($)	Total Cost ($)
2	2	18	1343	1361
	3	282	1032	1314
	4	998	320	1318
3	2	18	1343	1361
	3	295	1032	1327
	4	1045	320	1365

Analysis

The annual performances and costs of the various boiler and heat pump systems, in standalone and hybrid operation, have provided a variety of data that provides many avenues of analysis. First off, there is an obvious difference in annual operating cost in standalone operation for the three boilers utilized. Decreased idle losses in a boiler provides considerable savings, as shown by lower costs in operating boilers 2 and 3 compared to boiler 1 in any given region. As is expected, a decrease in loss of energy to surroundings lowers heating costs.

The above can also be observed when the boiler is run in conjunction with a heat pump. The data generated for heat pump 7 working with each of the three boilers for all control methods, shown in *Figures 20*, *24* and *28*, serves to show that better performing systems have lower operating costs. Control method 2 is especially effective in this aspect since this instance requires the boiler to operate solely to provide back up to the heat pump. For all cases shown in *Tables 20, 24 and 28,* hybrid heat pump operation with boilers 2 and 3 using control method 2 yield much lower boiler operation costs than doing the same with boiler 1.

Idle losses can be lowered using a number of methods. The first, and most expensive, is to purchase a tightly designed, well-insulated boiler. A cost-effective method compared to that is to incorporate piping and jacket insulation into the boiler system. Another cheaper, very effective method is to switch to a boiler control that lowers burner usage depending on outdoor temperature and/or ongoing heat demand. Such controls facilitate longer, steady-state operation more often and avoid needless cycling of the boiler.

Before starting any discussion on the modes with heat pump operation, it is important to recall the conditions for each control method. The following table outlines those. It is important to note that whenever the heat pump was providing heat, the boiler was assigned to idle as a backup in case the heat pump was not able to satisfy the demand. If the boiler were to not operate as a backup, the cost of heating would be lower because boiler idling would be eliminated. This would, however, come at the risk of losing comfort when the heat pump cannot meet the heating demand. In the case that the homeowners use the boiler for domestic hot water, the boiler would need to be on.

Table 29: Conditions for the Various Control Methods of Hybrid Operation

Control Method	***Condition***
2	HP only
3	HP only above 25°F
4	HP between 3/1 and 11/1

Heat pump operating costs under mode 2 vary greatly from one unit to another. The annual cost of heating, regardless of location, was greater for heat pumps with lower Coefficient of Performance (COP) values on average. As expected, heat pump 1 performed the best, followed by heat pumps 7, 2, 8 and 3. The differences from best performing to worst performing were more pronounced for nominally colder regions. This is a direct result of operating at lower Coefficient of Performance (COP) values while covering greater loads.

Switching from heat pump operation with boiler backup at all times to distributing load based on certain conditions will usually provide savings. The exception to this was heat pump 1 in the downstate and mid-state regions. This is because of higher Coefficient of Performance (COP) values compared to the other model heat pumps driving down cost of operation even on very cold days. However, it is imperative to note, once again, that the Coefficient of Performance (COP) values for heat pump 1 are not verified by field data but are rather manufacturer reported "optimal" data.

The actual savings between control methods 2 and 3 from one heat pump to another vary based on their relative Coefficient of Performance (COP) values. A heat pump with Coefficient of Performance (COP) values generally greater than compared to another heat pump yields greater savings when switching from method 2 to 3. For example, in the mid-state area, heat pump 8 saw upwards of $400 savings compared to only about $100 for heat pump 1 when switching from method 2 to 3. The variation in savings increased drastically for the upstate region, while it decreased for the downstate region. The cause of this can be attributed to the climate of the regions. As seen in the following table, the upstate region spends more hours below 25°F compared to the mid- and downstate regions. Therefore, limiting operation of the heat pump, especially on a heat pump with relatively lower Coefficient of Performance (COP) values, below the set temperature avoids operation at the least efficient points.

Table 30: Hours in the Year with Average Temperatures Below 25°F for Each Region

Region	***Hours Below 25°F***
Upstate	2102
Mid-state	1191
Downstate	414

The choice of 25°F as the crossover temperature was made after performing sensitivity testing with different temperatures. This was done for each of the three regions using a combination of heat pump 7 and boiler 1. The following tables show that data:

Table 31: Annual Cost Data at Various Crossover Temperatures Using Mode 3 for Heat Pump 7 and Boiler 1 in Mid-state Area

Crossover Temperature (°F)	***Boiler Cost ($)***	***Heat Pump Cost ($)***	***Total Cost ($)***
5	213	1814	2027
15	431	1539	1970
25	950	961	1911
35	1548	364	1911
45	1892	51	1943

Table 32: Annual Cost Data at Various Crossover Temperatures Using Mode 3 for Heat Pump 7 and Boiler 1 in Upstate Area

Crossover Temperature (°F)	***Boiler Cost ($)***	***Heat Pump Cost ($)***	***Total Cost ($)***
5	664	2030	2694
15	1117	1454	2571
25	1729	771	2499
35	2185	315	2500
45	2487	41	2528

Table 33: Annual Cost Data at Various Crossover Temperatures Using Mode 3 for Heat Pump 7 and Boiler 1 in Downstate Area

Crossover Temperature (°F)	***Boiler Cost ($)***	***Heat Pump Cost ($)***	***Total Cost ($)***
5	132	1,339	1,471
15	173	1,288	1,461
25	404	1,032	1,436
35	1,006	434	1,441
45	1,406	70	1,476

Switching from control method 2 to 4 resulted in savings as well. This is because the coldest temperatures are avoided by entirely switching away from heat pump operation in the coldest months. These savings are mostly achieved in the upstate region given average winter temperatures are much lower there. For the other regions, these savings are only limited to heat pumps 2, 3 and 8, while the other two show little to no savings.

The above analysis shows output differences due to changing control methods to be very similar for heat pumps 1 and 7, even though the former is not representative of the field and the latter is. It must be noted that this occurs because their Coefficient of Performance (COP) behaviors at the colder temperatures are almost identical, as seen in *Figure 99*. Despite that similarity, overall comparisons show that heat pump 7 performance cannot reach the cost savings that heat pump 1 can provide. This discrepancy further highlights the difference in performance between heat pump parameters provided by manufacturers (heat pump 1) and those that are seen in the field (heat pump 7 and others).

Additional Data and Analysis

While the metric used to identify which system and operational mode combinations would work best was in terms of cost, a short discussion on energy consumption is also useful in this context. To that end, annual energy consumption data that led to the calculation of annual heating costs were compiled for a few systems in each of the three regions of interest.

Firstly, annual energy usage was obtained solely for the best and worst performing boiler units (boilers 2 and 1, respectively). Also obtained was the energy usage for the best performing heat pump based on field data (HP7) in hybrid mode with boilers 1 and 2 for modes 2-4. For each mode and system combination described, energy consumption in mmBtu for the boiler and heat pump were calculated separately. Furthermore, source energy consumption for the heat pump was calculated using various average values of electrical power plant and transmission efficiencies. Another set of data for total energy consumption was calculated by adding the boiler and source annual energy consumption values. The following is the annual energy consumption data for the upstate, mid-state and downstate regions:

Table 34: Annual Energy Usage Data for Mid-State Region

System and Mode			***Annual Energy Usage (mmBtu)***			
Boiler	***Heat Pump***	***Mode***	***Boiler***	***HP Site***	***HP Source***	***Boiler and HP Source***
1	N/A	1	114	0	0	114
2	N/A	1	104	0	0	104
1	7	2	9	34	93	102
1	7	3	56	17	47	103
1	7	4	87	8	22	109
2	7	2	2	34	93	95
2	7	3	47	17	47	94
2	7	4	78	8	22	100

Table 35: Annual Energy Usage Data for Upstate Region

System and Mode			***Annual Energy Usage (mmBtu)***			
Boiler	***Heat Pump***	***Mode***	***Boiler***	***HP Site***	***HP Source***	***Boiler and HP Source***
1	N/A	1	149	0	0	149
2	N/A	1	137	0	0	137
1	7	2	16	47	157	172
1	7	3	101	14	46	147
1	7	4	114	11	36	150
2	7	2	9	47	157	166
2	7	3	91	14	46	137
2	7	4	103	11	36	139

Table 36: Annual Energy Usage Data for Downstate Region

System and Mode			***Annual Energy Usage (mmBtu)***			
Boiler	***Heat Pump***	***Mode***	***Boiler***	***HP Site***	***HP Source***	***Boiler and HP Source***
1	N/A	1	87	0	0	87
2	N/A	1	78	0	0	78
1	7	2	8	24	80	88
1	7	3	24	18	62	85
1	7	4	67	6	19	86
2	7	2	1	24	80	81
2	7	3	17	18	62	78
2	7	4	58	6	19	78

The electrical source efficiencies for each region used to calculate source energy are shown in the table below:

Table 37: Source Efficiencies for the Three Regions Examined in This Study [9]

Region	***Efficiency (%)***
Mid-state	36.5
Upstate	42.8
Downstate	29.9

The lowest annual energy consumption is achieved in all regions by the more efficient boiler in combination with the heat pump with a temperature-based switchover method. This is because in such an operating method, both hydronic and heat pump systems operate closer to their optimal efficiencies. In the upstate region, the lowest energy usage is also achieved by the most efficient boiler alone. This occurs because greater loads (due to colder temperatures) allow for more efficient operation. Source energy usage using the heat pump alone goes up because of lower temperatures in the upstate region despite having the greatest electrical source energy efficiencies. Lower source efficiencies in the downstate region result in more source and, hence, overall energy consumed for heat pump with boiler backup mode compared to hybrid operation with both temperature and date-based switchover methods.

A short analysis of domestic hot water usage was also performed below to provide better insight on how it can impact recommended use of the oil-fired system during periods when the boiler is not providing heat. For this model, the average daily consumption of domestic hot water was assumed to be 55 gallons at a temperature rise (ΔT) of 70°F. Based on these numbers, the average daily energy required for domestic hot water is 32126 Btu. The prices of heating oil and electricity used in this analysis were the same as those utilized in the calculation of the annual heating cost analysis. Given these parameters, the daily cost of heating for various systems can be calculated. The following table shows the systems, efficiencies, and daily cost for each:

Table 38: Various Domestic Hot Water Systems and Their Efficiencies Used for the Analysis

DHW System	***Efficiency (%)***	***Daily Cost ($)***
Oil-fired indirect	74.9	0.73
Oil-fired tankless coil	50	1.09
Direct oil-fired water heater	66	0.83
Heat pump water heater (HPWH)	200	0.90
Electric resistance water heater	95	1.89

Given the above information, an existing indirect domestic hot water (DHW) heating system should still remain in operation even if the boiler is not providing space heating. If a mid- to high performance tankless coil DWH system is used, it may be replaced with a heat pump water heater (HPWH) to provide savings on a daily basis. However, given a cost of $2400 for purchase and installation, the payback period for such a replacement would be ~35 years. HPWH also do not provide savings over a direct oil-fired water heater. Moreover, a HPWH draws heat from its surroundings, thereby increasing the heating load of that space, which is not accounted for in the hot water cost shown. Therefore, replacing the oil-fired system with a HPWH is not recommended. An existing electric resistance hot water tank should be replaced by any of the more efficient systems shown above.

Conclusion

This study included two primary parts: the field tests and the analysis. Conclusions for each of these are provided separately here.

Field Tests

Field tests were completed as planned on six sites across New York State that had hybrid heating arrangements containing both ductless mini-split heat pumps and hydronic heating systems. The purpose of these tests was to understand how they are operated together in common practice and provide suggestions that would help achieve more efficient and reliable operation.

At five of the six sites, a single heat pump was installed to provide heating and cooling for just a part of the home. The primary motivation was to provide direct comfort control over this space. In one case the heat pump was considered a supplemental heater for a den where the family spent time in the evening. In another case the heat pump provided independent temperature control for a home office space. At this site, three ductless heat pumps were installed and the building owner sought to totally displace the use of the oil-fired heating system.

For the sites which had just one heat pump, heating could be provided in the affected spaces by either the heat pump or the hydronic heating system. The practice for the control of these systems varied and often involved regular manual actions. When the heat pump was activated for use in heating, the setpoint on the heat pump thermostat was typically set for a higher level than the hydronic thermostat so it would serve as the primary heat source. For example, in the morning, the homeowner would turn the heat pump on if the outside temperature was not too cold. With this type of manual control there were often cases where the heat pump was simply not turned on at all. In one case a homeowner reported the heat pump ran regularly but a review of the data indicated it was only used for one day during the entire heating season.

From a review of the data, it can be concluded that prolonged operation in weather close to and slightly warmer than 32°F substantially increased frequency of defrost cycles, which led to further efficiency reductions. On the other hand, sites that utilized the heat pumps occasionally during days with colder than and slightly warmer than 32°F showed sporadic outdoor unit defrost cycles, which should limit energy losses and allowed the heat pump to provide warm air without needing to shut down too often. Therefore, continuously operating the heat pump with cycles under 2.5 hours showed potential to greatly improve performance by avoiding inefficient defrost cycles.

Analysis

As a general (and perhaps obvious) conclusion, for a combination of the best performing heat pump and the worst performing boiler, the lowest annual operating cost would be associated with a control strategy that emphasizes the heat pump as meeting most of the annual load. The opposite is true with the best performing boiler and a middle- or poor-performing heat pump.

In the mid-state (Albany) region, operation with the worst performing boiler alone has an annual cost of $1,953. Operation with the best performing, "ideal" heat pump (#1) with this boiler only providing backup heat as needed, leads to an annual operating cost $100 lower. Changing from the worst to the best boiler option (#2), operating without a heat pump, leads to an annual cost of $1,774, or $179 lower. For the heat pump with the best field-verified performance parameters (#7), operation with the lowest performing boiler (#1) was $103 more expensive than operating the boiler on its own. Given these results, anyone considering/needing an upgrade would benefit more from switching to a highly efficient boiler.

In the far upstate region, operation with the best performing boiler has a lower annual cost than operation with the best heat pump with any control strategy. Also, in this region, operation with the worst heat pump considered (3) with boiler backup as needed has an annual operating cost of $2,300 higher than operation with the worst boiler alone.

In the downstate (Long Island) region, operation with the best (not based on field data) heat pump (HP1) with boiler backup, leads to an annual cost savings of $219 compared to annual operation with the worst performing boiler. The best performing heat pump based on field data provides $20 annual savings compared to the worst performing boiler. The best performing boiler has an annual operating cost $57 higher than that of the best heat pump with boiler backup in this region.

In general, the best control strategy evaluated for use of the heat pump and hydronic heating system together is one which makes a decision on which to operate based on outdoor temperature. There are exceptions to this in some cases.

It should be noted that these conclusions are based on very specific current assumptions about heating oil and residential electricity prices. As the relative value of these parameters change, the conclusions would need to be reevaluated. It can also be generally concluded that, to be an attractive alternative, the heat pumps installed should only be those with the highest level of performance. Of the six sites included in this field study, only one heat pump can be considered in this category. The other five are more similar to heat pump units 7 and 2 and so only "fair" in performance.

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Attachment I – Best Practices Guide

Best Practices Guide

Hybrid Heating – Mini-Split and Oil-Fired Hydronic Heating Systems

Prepared by the National Oilheat Research Alliance

Description of Technology

Hydronic heating is a generally popular form of heating in New York State. The focus area in this study was on hydronic systems operated with an oil-fired boiler. These boilers use heating oil or, depending on the region, a mixture of heating oil and biodiesel, as fuel to heat a mass of water. This mass is then circulated when there is a heat requirement in a certain area. Heat exchangers (baseboard radiators) situated in that zone then dissipate the heat into the area.

Oil-fired boilers are also capable of providing domestic hot water, either as a tankless coil system or an indirect zone system. The former acts as an "instant" heater by transferring heat from the boiler water to domestic water using a heat exchanger coil. The latter system transfers heat into a water tank, which acts as a separate heating zone. For either method, the boiler needs to remain hot throughout the year.

Air-source heat pumps (ASHPs), on the other hand, move heat from the outside circulating a refrigerant through a thermodynamic cycle that enables heat transfer. The heat is then transferred from the refrigerant to air. This warmed air is released into the heating zone to satisfy the demand. The ASHPs used in this study are those of the ductless mini-split variety. When compared to their ducted counterparts, ductless mini-split ASHPs are useful for heating singular, small spaces or zones. They consist of an outdoor unit and an indoor heat pump unit.

Because of the way they provide heat, hydronic and heat pump systems achieve optimal performances at different conditions. For example, oil-fired boilers will operate at their greatest efficiencies when steady state is reached. This becomes more likely as the load increases, which can be either due to very cold weather or recovery from nighttime setback. Conversely, idling time increases as loads decrease, which reduces operating efficiencies. ASHPs, on the other hand, rely heavily on outdoor temperature for their performance parameters. Maximum capacities and efficiencies (measured as COP) decrease as it gets cooler outside, which means that operation in cold weather may entail sacrificing comfort and/or efficiency.

Given the circumstances under which each system provides best performance, there is potential to utilize a hybrid system, for a house that has both available, to achieve lower-cost heating.

Summary of Work to Support Best Practices

The analysis for this guide was performed based on a field study that analyzed operation of hybrid heat pump and hydronic systems and an annual cost analysis based on various performance factors.

The field components involved setting up loggers in six sites across New York State to measure temperature at various points of interest in a hybrid system. These measurements determined frequency and condition of operation as a function of indoor and outdoor conditions. For example, hydronic supply water temperatures above a certain temperature indicated that the hydronic system was providing heat to a zone. If this temperature sustained for a prolonged period or cycled on and off very frequently while the indoor temperature was low in the early hours of the morning, it could be concluded that recovery from nighttime setback is occurring. Heat pump delivered air temperatures substantially above the room temperature indicated that the heat pump is providing heat. Frequency of defrost cycles were also determined using outdoor unit temperatures, which could then be presented as a function of outdoor temperatures.

Data was obtained for all six sites for at least one full heating season. Based on the data, operation of the combined systems was evaluated under various indoor and outdoor conditions.

The annual performance analysis was performed for three regions in New York State using modeling data that provided hourly demands and outdoor temperatures. Then, based on performance data for a collection of model heat pumps and boilers, annual cost was evaluated for different chosen hybrid control methods. The first control strategy involved heating using the boiler only (method 1). Another control strategy was to provide heat with the heat pump only (method 2). The next two involved switching between boiler and heat pump as primary heating sources. The switch was made either based on a crossover outdoor ambient temperature (method 3) or a date (method 4). For most cases with control method 3, the crossover temperature was chosen to be 25°F. There was, however, some sensitivity testing performed by varying that temperature to 5°F, 15°F, 35°F and 45°F for only one combination of heat pump and boiler, and one location. Output data for control method 4 was obtained by stipulating the heat pump provide heat solely (with the boiler on standby) between March 1 and November 1, while the boiler was the sole heating source the rest of the year. It is important to note that at any instance the heat pump is providing heat, the boiler was assigned to cover any loads that could not be satisfied because of a lack of capacity.

Recommendations

Based on the work above, a list of best practices was produced for the operation of hybrid mini-split and oil-fired hydronic systems to minimizes cost and maximizes efficiencies. The following is that list of best practices:

1. For either heat pump or boiler system, the greatest cost reductions will occur by switching to the most efficient available appliances. This factor provides more annual heating cost reduction than any control method that switches heating loads between the hydronic and heat pumps systems. Low performance heat pumps under any control strategy will most likely yield annual costs higher than using the hydronic system on its own in combination with a low performance boiler.

2. Using a control strategy to switch between heat pump and hydronic system operation based on either outdoor temperature or date provides lower annual operating cost than a control system where the heat pump alone provides heat with the hydronic system covering unsatisfied loads. The lower the performance of the heat pump, the greater this difference.

3. The savings obtained from using a crossover temperature point to switch between heat pump and hydronic heating peak at 25°F for any region. Using crossover temperatures less than 25°F that generally yields more heating costs than those greater than 25°F. The differences, however, are not substantial.

4. In general, especially for good- and mid- performing heat pumps, the difference between a control strategy based on outdoor temperature and one based on date is generally not substantial. Thus, a date-based control strategy can be utilized to obtain similar savings while avoiding the task of switching back and forth many times within the heating season. In the case of poor performing heat pumps, the savings of using this strategy are greater than the one based on outdoor temperature.

5. When used in combination with the hydronic system, the heat pump should not be used as the main source of heating near 32°F or slightly warmer since frequency of defrost cycles peaks around those temperatures. Light operation of the heat pump at those temperatures will greatly decrease the number and frequency of defrost cycles, thus minimizing a loss of comfort due to them. This is also highly advisable for temperatures well below 32°F so operation of heat pump at low Coefficient of Performances (COPs) can be avoided.

6. Many oil-fired hydronic heating systems provide domestic hot water in addition to space heat and, so, will remain in a warm or hot state even when they have no heating load. However, when using a date-based switchover control strategy, a boiler with an indirect water heater or tankless coil system should remain on for domestic hot water heating.

7. If, during the coldest part of the year, an existing hydronic heating system is not used, there is a concern that the hydronic system piping could freeze. In this case the use of antifreeze in the circulating piping is recommended.