Demonstrating Cold Climate Packaged Terminal Heat Pumps in New Construction

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Demonstrating Cold Climate Packaged Terminal Heat Pumps in New Construction

Final Report

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Abstract

This demonstration project investigated the performance of twelve Ice Air RSXC cold climate packaged terminal heat pumps (ccPTHP) installed in four apartments in a multifamily new construction building on Coney Island, New York. This project sought to inform the market about the potential for ccPTHPs to meet New York State greenhouse gas emission and electrification goals. After the installation of the ccPTHPs, measurement and verification systems were installed to monitor thermal output, energy consumption, and operational status of the units, as well as the conditions of the spaces being served.

When the outdoor air temperature (OAT) was below 40°F, the ccPTHPs maintained comfortable room temperatures and overall comfortable supply air temperatures. The seasonal heating COP of the units ranged from 1.2 to 2.3 for the individual units. The smaller RSXC09 units, with a rated heating capacity of 10,200 Btu/h, had an average seasonal heating COP of 1.7. The larger RSXC13 unit, with a rated heating capacity of 12,000 Btu/h, had an average seasonal heating COP of 2.3. The researchers observed increased cycling of the ccPTHP units installed in the bedrooms of the apartments potentially impacting the comfort of supply air temperatures and reducing their efficiency. Differences in occupancy and resident behavior along with oversizing of the RSXC09 units relative to the bedroom heating loads possibly explain the differences between models and variations between different apartments' units. When the OAT was below 40°F, the ccPTHPs operated in defrost mode for ~6% of the total operating time. The energy consumed by the units during defrost operation had a slight impact on COP and minimal impact on occupant comfort.

During the cooling season, the cooling output of the units indicated that the units were either adequately sized or potentially oversized. Daily trends of the room temperatures compared to the OAT confirm that the ccPTHPs effectively meet cooling demands.

Keywords

heat pump, cold climate packaged terminal heat pump, field monitoring, measurement and verification, PTHP, ccPTHP, efficiency, COP, demonstration

Acknowledgements

Steven Winter Associates, Inc. (SWA), would like to express appreciation to the team at L+M Development Partners, LLC, for overseeing the installation of the cold climate packaged terminal heat pumps in this research project. Additionally, the team at Ice Air, LLC, spent significant time and effort to provide background information and assistance during the installation, monitoring, and analysis of the performance data. On the Ice Air team, SWA specifically acknowledges Director of Engineering Bill Liu, Senior Sales Engineer Tamer Bou Matar, Jose Liang, and Nadim Afiouni. SWA would like to acknowledge the contributions of Robin Neri and Jason Block in the development of this project and for their continued insight throughout the analysis and reporting phase. The team greatly appreciates Robert Carver and Anne Ruckman of NYSERDA for their patience and invaluable insights.

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Acronyms and Abbreviations

°F	degrees Fahrenheit
Btu	British thermal units
Btu/hr.	British thermal units per hour
ccPTHP	cold climate packaged terminal heat pump
CFM	cubic feet per minute
COP	coefficient of performance
DB	dry bulb
dT	difference in temperature
ERH	electrical resistance heating
HPD	NYC Housing Preservation and Development
HPO	heat pump output
HVAC	heating, ventilation, and air conditioning
kΩ	kiloohm
kWh	kilowatt hours
L+M	L+M Development Partners
M&V Plan	measurement and verification plan
NYS	New York State
OAT	outdoor air temperature
NEEP	Northeast Energy Efficiency Partnership
PTAC	packaged terminal air conditioners
PTHP	packaged terminal heat pump
RH	relative humidity
SF	square foot
SWA	Steven Winter Associates, Inc.
TEC	The Energy Conservatory
TSI	TSI Incorporated Alnor balometer
WB	wet bulb

Executive Summary

An efficient cold climate packaged terminal heat pump (ccPTHP) that operates at low ambient winter temperatures common throughout New York State could reduce local greenhouse gas emissions, improve air quality, and reduce costs for building owners. Many buildings currently use packaged terminal air conditioners (PTACs), so developers and contractors are familiar with their form factors. Units designed as ccPTHPs with the same form factor can easily replace PTACs in the new construction market, offering a widespread solution for electrifying multifamily heating and cooling.

This report presents the results of an experimental installation of ccPTHPs in a new construction project in New York City. The findings illustrate how ccPTHPs can help meet greenhouse gas emission and electrification goals set by New York State, New York City, and local utilities. This demonstration aims to inform the market—including developers, contractors, and building owners—about the effectiveness and efficiency of ccPTHPs for heating and cooling new apartments in New York State.

Steven Winter Associates, Inc. (SWA), analyzed the electrical energy consumption and thermal energy output of 12 ccPTHPs manufactured by Ice Air, LLC, installed in a new multifamily building in Brooklyn, NY. The installed units consisted of eight RSXC09 PTHPs with a heating capacity of 10,200 thousand British thermal units per hour (Btu/hr) and four RSXC13 PTHPs with a heating capacity of 12,000 Btu/hr. Researchers monitored and analyzed these units from January 1, 2022, through March 6, 2023. The RSXC product line has undergone testing and has been rated under standard lab conditions, which serve as a comparison to the real-world conditions studied here.

The study team comprised researchers from SWA and Ice Air. Ice Air manufactures the ccPTHP units analyzed in this study and is a major producer of PTACs and other heating, ventilation, and air conditioning (HVAC) equipment. L+M Development Partners (L+M) provided the host site for the demonstration and managed the construction and installation process. Sentient Buildings installed the monitoring equipment and hosted the data.

ES.1 Summary of Findings

ES.1.1 Comfort

When the outdoor air temperature (OAT) was below 40 degrees Fahrenheit (°F), the ccPTHP units maintained comfortable room and supply air temperatures without requiring backup heat.[1] Even with OATs between 15°F and 20°F,[2] room temperatures averaged approximately 75°F. Supply air temperatures dropped below 80°F for approximately 43% of the operating time; however, room temperatures rarely fell below 70°F, even during periods with low supply air temperatures. While variations occurred from apartment to apartment and room to room, the overall stability and comfortable room temperatures suggest that these ccPTHP units can supply sufficient thermal energy to adequately heat the spaces, even at low outdoor temperatures.

The heating and cooling capacities of the ccPTHP units installed in the bedrooms exceeded the expected loads based on modeling, despite the installed units being the lowest capacity RSXC model available. This oversizing, combined with a limited turndown ratio, resulted in these units more frequently cycling on and off. Frequent cycling can reduce the overall efficiency of a heat pump and increase the likelihood of the supply temperature falling below comfortable ranges.

Uncertainties exist regarding the unexpectedly low supply air temperatures, but open outdoor air dampers are a possible explanation. The southern exposures of the studied apartments might contribute to warmer room temperatures at cold outdoor air temperatures, despite the low supply air temperatures. Interviews with building staff confirmed no comfort complaints from the residents in the apartments with PTHPs installed.

ES.1.2 Efficiency

The seasonal heating coefficient of performance (COP)[3] resulted from selecting all points where the unit operated in heating mode below 50°F OAT, excluding defrost mode. The seasonal heating COPs[4] of the individual units ranged from 1.2 to 2.3. Grouping the units by model revealed a seasonal heating COP of 1.7 for the RSXC09 and 2.3 for the RSXC13.

The published COPs for the RSXC09 and RSXC13 at 10°F OAT are 2.2 and 2.14, respectively. In this study, the temperature-binned COPs in the 15°F–20°F OAT range (the lowest observed temperature range) were 1.5 for the RSXC09 and 2.1 for the RSXC13.

During the cooling season, the cooling output of the units indicated either adequate sizing or oversizing for the space where they were installed. Room temperature data and daily trends demonstrated that the units effectively met the cooling demands of the tenants.

ES.1.3 Defrost

Instances of defrost mode were minimal throughout data collection, with minimal impact on occupant comfort. When the OAT fell below 40°F, the ccPTHP units operated in defrost mode for about 6% of the total operating time. Including defrost operation in efficiency calculations had a minor effect—the temperature-binned COP decreased by approximately 0.1. The seasonal COP for the RSXC09 dropped to 1.4, while the RSXC13 dropped to 2.0 when accounting for the energy consumed during defrost operation in the COP calculations.

1 Background

Based on a 2018 analysis Taitem Engineering prepared for the New York State Energy Research and Development Authority (NYSERDA),[5] approximately 250,000 packaged terminal air conditioners (PTACs) currently operate in New York City. An estimated 100,000 of these existing PTACs and packaged terminal heat pumps (PTHP) could easily undergo replacement with high-efficiency cold climate packaged terminal heat pumps (ccPTHP) units. Taitem found that these retrofits could proceed without requiring fuel switching or wall renovations.

The current PTAC models are generally inexpensive and inefficient, relying on electrical resistance heating (ERH) or fossil fuel plants and distribution. Construction costs for installing PTACs are increased due to the need for a hot water or steam coil and natural gas piping. In contrast, an efficient PTHP designed for colder New York State climates can significantly reduce greenhouse gas emissions, installation complexity, and operating costs.

Taitem's report identified the low efficiency of traditional PTHP units as a significant barrier to their adoption in New York State and New York City. Traditional units often require substantial ERH to compensate for poor performance at low outdoor air temperatures (OAT). To serve as a reliable replacement for PTACs, PTHPs must demonstrate higher capacity and efficiency in such conditions.

Steven Winter Associates, Inc. (SWA), analyzed the electrical energy consumed and thermal energy delivered by 12 ccPTHPs installed in a new multifamily building in Brooklyn, NY. Manufactured by Ice Air, LLC, these ccPTHPs represent a viable solution for electrifying space heating in new construction. This demonstration seeks to inform the market about the effectiveness and efficiency of these heat pumps in a real-world, new construction setting.

The 12 ccPTHPs were installed in four two-bedroom apartments. Each apartment contained three heat pumps: one RSXC09 unit in each bedroom and one RSXC13 unit in the living room. The apartments were vertically stacked on the second through fifth floors of a larger building. The project included eight RSXC09 and four RSXC13 heat pumps. Figure 1 illustrates the building layout, the relative positions of the apartments, and the locations of the ccPTHPs in each room.

Figure 1. Building Layout and Location of Test Apartments

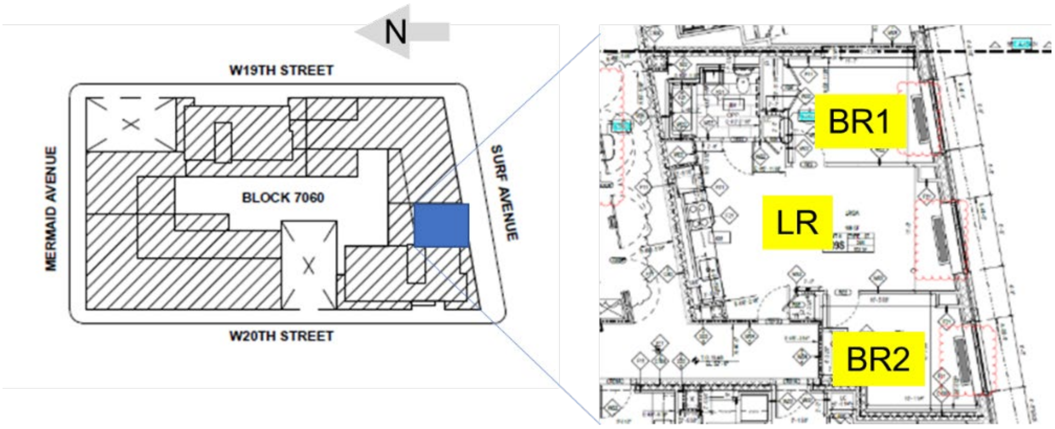


Table 1 shows the cooling and heating capacities for the specific brand and model of cold climate packaged terminal heat pumps (ccPTHP) installed and measured during this analysis. The values were obtained from the Ice Air documentation.

Table 1. Ice Air RSXC Packaged Terminal Heat Pump Rated Capacity and Efficiency

Series Model Number	RSXC09	RSXC13
Cooling Capacity (Btu/hr) at 95°F	9,200	12,500
Heating Capacity (Btu/hr) at 47°F	10,200	12,000
COP at 47°F	3.6	3.1
Heating Capacity at 10°F	6,600	7,700
COP at 10°F	2.2	2.14
Heating Capacity at 5°F	6,100	6,900
COP at 5°F	1.98	1.91
Heating Capacity at -5°F	5,500	6,400
COP at -5°F	1.74	1.62

The RSXC series allows supplemental heating from either a central building system or an ERH option. Ice Air recommends using the ERH option in markets that experience ambient temperatures below -5°F. When the ERH is used, the heat pump operation shuts down, and ERH activates automatically below -5°F (+/-3°F).

In addition to the rated capacities provided in the Ice Air documentation, maximum and minimum capacities provided in Northeast Energy Efficiency Partnerships' (NEEP) Cold Climate Air Source Heat Pump Lists[6, 7] are available through the links in the reference section. Table 2 displays the heating and cooling capacities for the RSXC09 and RSXC13 based on the outdoor dry bulb temperatures. Figure 2 compares the NEEP heating capacities with the heating capacity from Ice Air.

Table 2. Ice Air RSXC Unit Capacities from Northeast Energy Efficiency Partnerships Data

Capacities shown in Btus/hr.

Mode	OAT (°F)	Max. Cap. (NEEP)	Rated Cap. (Ice Air)	Min. Cap. (NEEP)
		Ice Air RSXC09		
Heating	-5	6,300	5,500	5,500
Heating	5	7,300	6,100	5,200
Heating	17	8,200	—	6,500
Heating	47	12,600	10,200	9,200
Cooling	82	11,800	—-	11,800
Cooling	95	11,800	9,200 (7,200 sensible)	9,200
		Ice Air RSXC13		
Heating	-5	7,350	6,400	6,400
Heating	5	8,200	6,900	5,600
Heating	17	9,500	—	7,700
Heating	47	14,200	12,000	11,000
Cooling	82	14,900	—	14,900
Cooling	95	14,000	12,500 (8,800 sensible)	12,500

Figure 2. Rated Heating Capacity Comparison

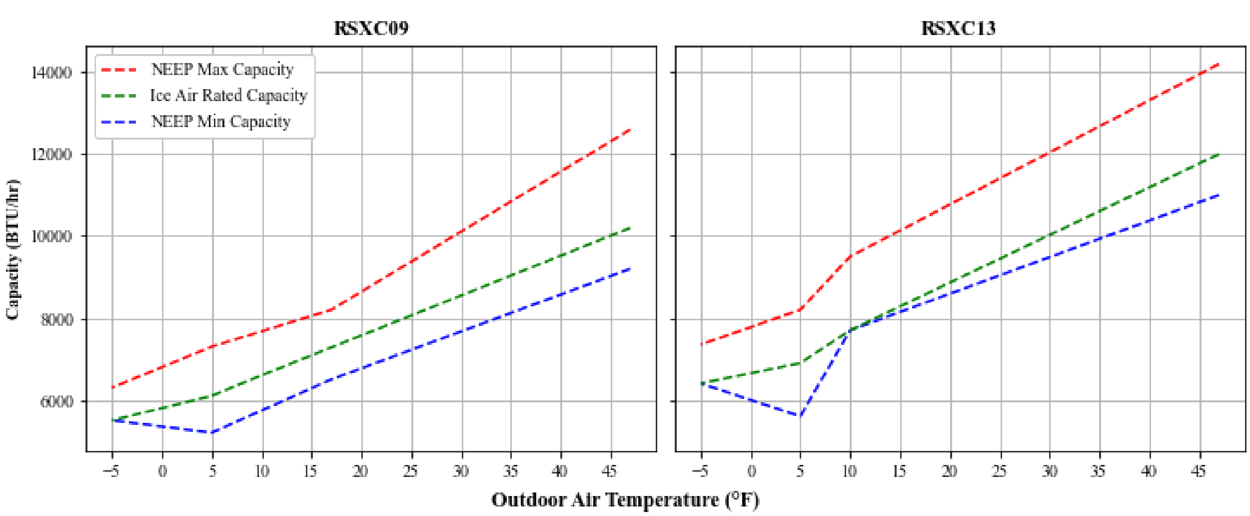


Figure 3. Rated Cooling Capacity Comparison

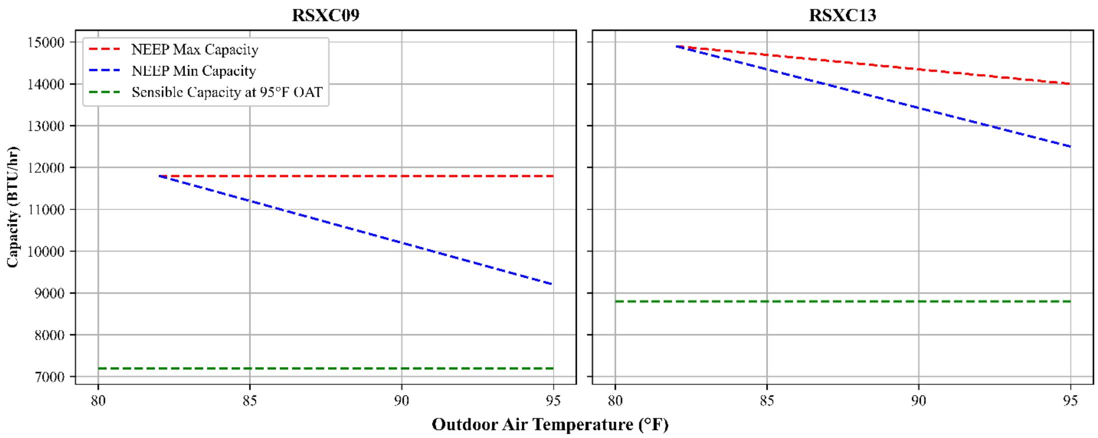


Figure 3 displays the NEEP maximum and minimum capacities for cooling. Because no relative humidity data was recorded during the monitoring period, the Ice Air sensible capacity is a point of comparison for evaluating heat pump output in cooling mode. Ice Air measured the rated sensible performance while operating in cooling mode at indoor conditions of 80°F dry bulb (DB) and 67°F wet bulb (WB), with outdoor conditions of 95°F DB and 75°F WB ambient. The units can function in cooling mode down to an ambient air temperature of 38°F and are tested to operate in heating mode at ambient temperatures as low as -20°F.

Table 3 outlines the reference tags and locations of each PTHP. These reference tags facilitate the analysis of the units' performance across different apartments and models. Each bedroom contains the smaller RSXC09 model, while the living rooms feature the larger capacity RSXC13 model.

Table 3. Packaged Terminal Heat Pump Reference Tags

Reference Tag	Apartment Reference	Apartment	Room	PTHP Model
1_1	1	209S	BR1	RSXC09
1_2	1	209S	LR	RSXC13
1_3	1	209S	BR2	RSXC09
2_1	2	309S	BR1	RSXC09
2_2	2	309S	LR	RSXC13
2_3	2	309S	BR2	RSXC09
3_1	3	409S	BR1	RSXC09
3_2	3	409S	LR	RSXC13
3_3	3	409S	BR2	RSXC09
4_1	4	509S	BR1	RSXC09
4_2	4	509S	LR	RSXC13
4_3	4	509S	BR2	RSXC09

2 Methodology

SWA analyzed data collected from January 2022 through March 2023 for the 12 ccPTHP units installed in a multifamily building in Brooklyn, NY. Each ccPTHP unit was outfitted with several sensors by the research team to measure electrical energy consumption and calculate the thermal energy delivered. The sensors were also used to infer the operation mode of the heat pumps.

2.1 Research Questions

SWA conducted this study to address eight research questions. The fundamental questions and the methods used to answer them are outlined in the following text.

1. **Does the RSXC heating system provide enough heat without backup?**

   Method: Evaluate the frequency of room temperature drops when the heat pump operates in heating mode. If room temperature drops for more than 30 minutes with the heat pump on, then the unit is supplying insufficient heat.

   Limitations: This method cannot directly calculate the heating load of each room. Open windows may significantly increase the heating load, potentially misleading assessments of the heat pump's performance.

2. **Can the RSXC maintain comfortable supply air temperatures?**

   Method: Record the supply air temperature while the heat pump operates in heating mode. Summarize instances when supply air temperature falls below 80°F at various OATs. Additionally, interview building staff to identify tenant complaints or other issues not captured by the monitoring system.

   Limitations: Comfort levels are subjective; 80°F might not suffice for all occupants under varying conditions (clothing, activity level, airflow direction, etc.). Interviews may capture only significant issues, and the limited number of monitored apartments reduces opportunity for feedback in a building with many residents.

3. **What is the seasonal efficiency of the systems for heating? What is the seasonal efficiency of the systems for cooling?**

   Method: Sum the heat pump output and input during mode when OATs drop below 50°F to determine a seasonal coefficient of performance (COP). Use temperature bins to calculate a temperature-binned COP and fit a curve to predict efficiency at low OATs. Calculate the average heat pump output (HPO) for heating with rated capacity at different OATs to determine the heating seasonal efficiency. Calculate the average HPO for cooling with rated capacity at different OATs to determine the cooling seasonal efficiency.

   Limitations: Sample size depends on occupant usage patterns. Some heat pumps may operate infrequently due to low set points or being turned off, limiting the data available for aggregation.

4. **How much condensate/ice melt is produced?**

   Method: Calculate the frequency of detected ice meltwater in the drain line and analyze trends against outdoor temperature and humidity, normalized to hours of heat pump operation. Compare results with theoretical calculations from earlier psychrometric analysis.

   Limitations: This method provides frequency data but does not directly measure condensate volume. No viable method existed for directly measuring condensate volume in these units.

5. **Does the defrost mechanism operate effectively without sacrificing comfort or efficiency?**

   Method: Calculate seasonal efficiency both with and without including defrost cycle energy consumption to assess the energy impact of defrost. Ensure that neither room temperature nor supply air temperature (when the fan is on) drops by more than 5°F during a defrost cycle.

   Limitations: Measurements do not account for open windows, so drops in room temperature may result from occupant behavior rather than faulty heat pumps.

6. **Are there other design, installation, commissioning, or operations issues?**

   Method: Gather feedback from the manufacturing and construction teams regarding product installation and system balance, including electrical work and condensate drains.

   Limitations: This process is anecdotal and may overlook some details.

7. **What is the estimated operating cost of a ccPTHP in new construction?**

   Methods: Sum the seasonal energy usage of the heat pumps per apartment and multiply it by the typical electricity rates paid, regardless of whether the heat pumps connect to a common area meter or the individual resident's meter.

   Limitations: This estimate only applies to a building with this specific heating load and usage pattern. Variations in room set points and differences in heating loads across apartments will affect the estimate's precision.

8. **What is the estimated construction cost of the ccPTHP in new construction?**

   Method: Review as-built installation costs with the construction team.

   Limitations: This analysis represents only one building developed by a single developer with a small number of installations, which may not reflect larger-scale installations.

2.2 Hardware

SWA used the monitoring hardware detailed in Table 4 to record data points. An Invonics Receiver/Repeater network transmitted data to a Sentient Neuro Edge Gateway panel for remote access. SWA collected and analyzed samples in 15-minute increments, employing internally developed tools for data review.

Table 4. Directly Measured Physical Parameters

Measurement	Quantity	Sensor Type	Sensor Accuracy
Supply Air Temperature	1 per ccPTHP	Inovonics EN1723 with external 10k thermistor	+/- 1.2 °F
Return Air Temperature	1 per ccPTHP	Inovonics EN1723 with external 10k thermistor	+/- 1.2 °F
Supply-Return Air Pressure Differential	1 per ccPTHP	Inovonics EN1702 with Honeywell P7650A1026	+/- 1% full scale
			(+/-.01 inH2O)
Room Air Temperature	1 per ccPTHP	Inovonics EN1723 internal sensor	+/- 0.8 °F
Outdoor Coil Temperature	1 per ccPTHP	Inovonics EN1723 with external 10k thermistor	+/- 1.2 °F
Condensate Drain Temperature	1 per ccPTHP	Inovonics EN1723 with external 10k thermistor	+/- 1.2 °F
ccPTHP Amperage	1 per ccPTHP	Inovonics EN1702 with Honeywell Clamp CT CTP-A-50-RMS	+/- 0.1 Amps
Outdoor Air Temperature / RH / Weather Conditions	1 for the site	Sentient Virtual Weather Station	+/- 2°F
			+/- 2% RH

The original measurement and verification (M&V) plan indicated a sampling frequency of 5-minute increments; however, limitations in data transmission for the individual sensors and cloud storage capacity necessitated a change to 15-minute intervals. This change reduced the resolution of recorded cycles. Despite this, the team believes that the length of the monitoring period and substantial quantity intervals provides a reasonable representation of the units' performance across various temperatures. Approximately 41,000 intervals were captured for each unit, with compressors operating for about 11% of those intervals across all monitored PTHPs.

2.3 Monitoring Equipment Installation

Sentient installed monitoring equipment according to the M&V plan and connected it to the Skyspark and Neuro platforms for cloud-based diagnostics and analysis.

Figure 4. Sensor Transmitter Boxes

Sensor transmitter boxes were installed in each ccPTHP to collect readings from the sensors and transmit the data to the internet gateway.

Figure 5. Internet Gateway Panel and Cell Network Transmitter

The internet gateway and cell network transmitter, located in a common area hallway closet, received data from the sensor transmitter boxes and relayed it to Skyspark.

Figure 6. Temperature and Air Pressure Sensors

Temperature and air pressure sensors installed at the supply (left) and return (center) of the indoor coil, with a condensate drain temperature sensor (right) extending to the outdoor section of the PTHP.

Figure 7. Differential Pressure Sensor and Current Sensor

Differential pressure measured by an octagonal sensor across the indoor coil. The red current sensor, located inside the protected electrical section of the PTHPs, monitored unit amperage draw.

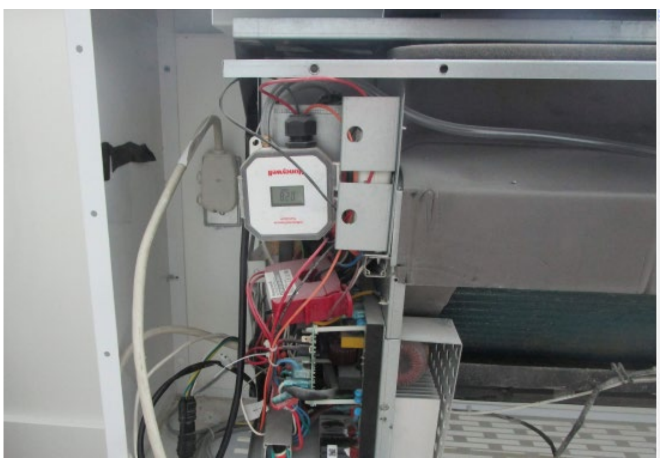
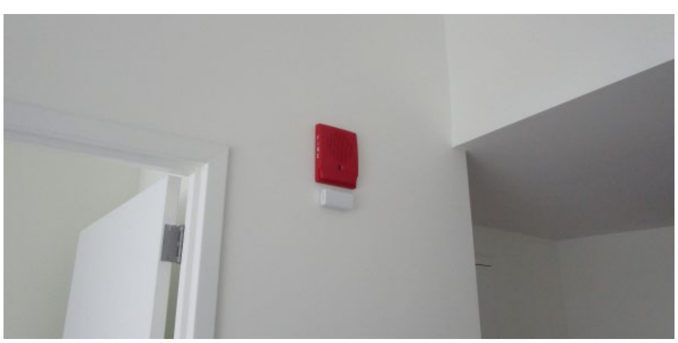


Figure 8. Room Temperature Sensor

Small rectangular room temperature sensor installed below the fire alarm in the living room, positioned approximately 6 feet above the floor and centrally located within each room.



After installation, SWA identified two persistent sensor issues which persisted throughout data collection. First, sensors for PTHP 1_3's were missing after installation. SWA, Sentient, and L+M could not ascertain the cause, but the construction team still working in the building prior to occupancy may have inadvertently removed them. Consequently, no data was collected from that PTHP because Sentient did not replace them. The Ice Air and SWA team verified the functionality of that heat pump during site visits, but without monitoring equipment, data was not collected. Second, the return air temperature sensor installed in PTHP 4_3 experienced intermittent communication issues. SWA attempted to reboot and reconnect the sensor, but it continued to lose signal intermittently. During those periods, SWA substituted room temperature for the missing return air temperature values to maintain the integrity of the dataset, noting consistency between room temperature and return air temperatures in other PTHP units.

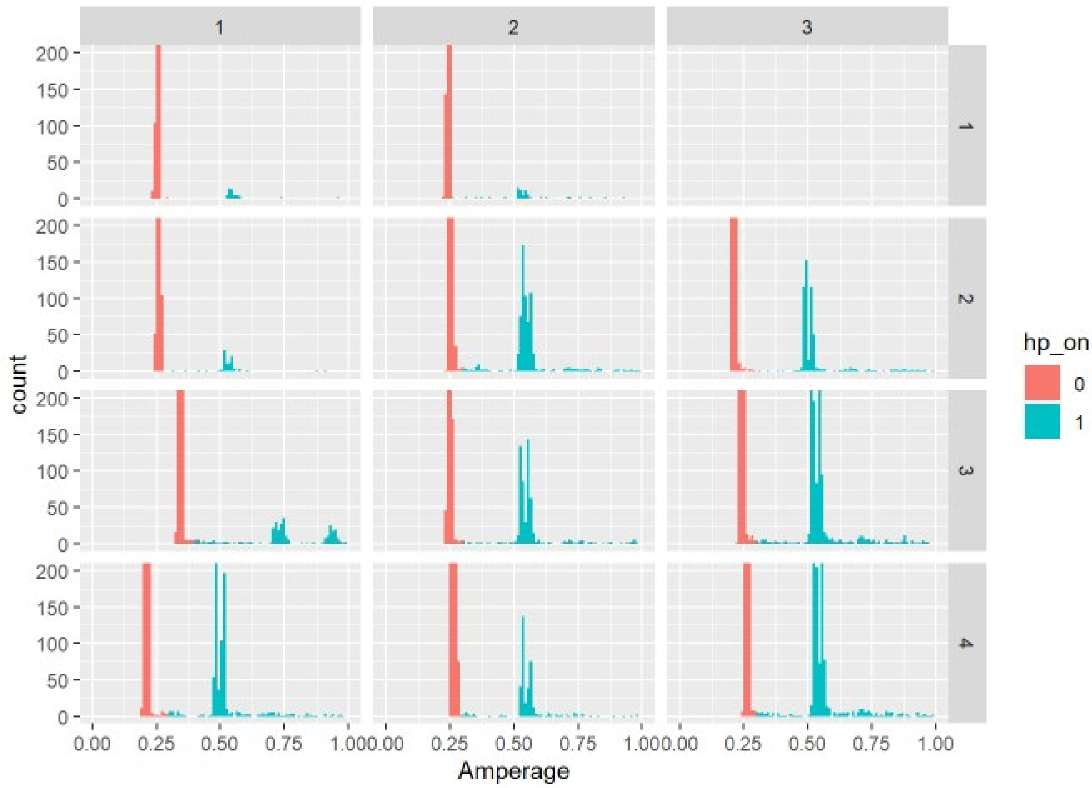
2.4 Determining Operating Mode and Calculated Values

2.4.1 Determining Heat Pump Operating Mode

SWA inferred the operating mode and associated current draw thresholds of the ccPTHP units from the collected data. Ice Air ccPTHP units operate in various modes, including fan-only mode, low and high fan speeds with compressor on, and defrost mode.

To distinguish between operating modes, SWA analyzed the current draw distribution from each unit. The threshold for the unit turning on was determined based on these observations, as shown in the histograms in Figure 9. This threshold was approximately 0.3 amps. Note that units are arranged vertically by floor and horizontally by room, so the top left histogram represents unit 1_1, and the bottom right histogram represents unit 4_3.

Figure 9. Histograms of Current Draw for Each Unit With On/Off Threshold



SWA determined the remaining operating modes based on the analysis of current draw distributions for each unit and manufacturer data.

1. Compressor off, fan on
    - Current draw greater than 0.3 amps, but less than 1.25 amps (the rated maximum current draw of the fan)
2. Compressor on (amperage greater than 1.25 amps)
   Low fan speed: Compressor on, amperage below ~5 amps
    - Heat mode, high fan speed: Compressor on, amperage above ~5 amps
    - Defrost mode (all criteria must be met):
        - Heat pump compressor on for at least 15 minutes (two samples)
        - Supply air temperature dropped at least 5°F between the first and second readings
        - Outdoor coil temperature increased between the first and second readings
        - Outdoor coil temperature in the first reading was less than 40°F

When the compressor was operating, the mode was labeled as cooling mode if the supply air temperature was 2°F lower than the return air temperature. Conversely, if the supply air temperature was 2°F higher than the return air temperature, the mode was labeled as heating.

2.4.2 Determining Derived Values

Using directly measured data points, SWA derived several values. Table 5 summarizes these parameters and the equations/logic used to calculate them.

Table 5. Derived Values and Calculation Methods

Parameter	Description	Calculation	Units
dT	Temperature difference across the indoor coil of the ccPTHP	$Supply\ Air\ Temp - Return\ Air\ Temp$	°F
Heat pump mode	Heating or cooling mode	dT higher than 2°F: Heating dT lower than -2°F: Cooling	Logical
CFM	Cubic feet per minute of airflow across the indoor coil of the ccPTHP	Lookup value based on fan speed. Airflow rates were determined as described in section 2.4.4	CFM
Heat pump output[8]	Thermal energy delivered by the heat pump to the room in a 15-minute period	$dT \cdot C_{p_{air}} \cdot \rho_{air} \cdot CFM \cdot \frac{60\ minutes}{hour} \cdot 0.25\ hour$	Btu
Heat pump input	Electrical energy consumed by the heat pump	$Amperage \cdot \frac{208\ V}{1000} \cdot Runtime(0.25\ hours) \cdot PF$[9]	kWh
COP[10]	Units of energy delivered to the room for each unit of input electricity	$\frac{Heat\ Pump\ Output}{Heat\ Pump\ Input}$	N/A
Condensate produced	Whether heating mode condensate is produced during a defrost cycle	Label as "Condensate" if unit in "Defrost" mode and condensate drain temperature drops by more than 5°F in 15 minutes	Logical

2.4.3 Calculating Coefficient of Performance

The 15-minute data collection intervals complicated the COP calculation. Some data points showed a very high change in temperature (dT) across the coil but low energy consumption at the time of measurement, resulting in a very high COP for that individual measurement. This condition likely occurred when a measurement captured the end of a cycle, at which point the compressor had turned off, the coil remained hot (or cold), and the fan continued to run. Similarly, high energy consumption associated with a low dT across the coil likely indicated a measurement taken at the beginning of a cycle.

Additionally, because the PTHPs feature variable speed compressors, the energy input and output are not the same during each time interval. Calculating the COP at each data point and averaging those values would overrepresent low-output measurements and underrepresent high-output measurements.

To address these factors, SWA summed each unit's total energy consumption and total thermal energy output by 5°F OAT bins. These total input and output energy values were then used to calculate a temperature-binned COP for each unit within each OAT range. Following is a simplified example equation for the COP:

$$Temperature - Binned\ COP_{35-40°F} = \frac{\sum_{35°F}^{40°F} HPO}{\sum_{35°F}^{40°F} HPI}$$

To ensure that the heat pump output value was not artificially inflated, SWA included only data points where the unit operated with the heat pump on. Additionally, points where the heat pump was off but the fan was on were included if the previous time increment/measurement indicated the heat pump had been on. This inclusion accounted for the heat delivered by the fan blowing over the hot coil at the end of the cycle, even after the compressors stopped running.

SWA applied a similar method to determine the seasonal COP. Instead of using temperature bins to divide the COP, SWA included data for all winter months when the unit operated in heating mode at or below 50°F OAT.

2.4.4 Determining Supply Airflow

The team measured airflow on the installed ccPTHPs using a TSI Alnor balometer calibrated with a powered flow hood manufactured by The Energy Conservatory (TEC). Table 6 displays the airflow readings from the heat pumps. After taking measurements in each operating mode, the team connected the TSI balometer to the TEC powered flow hood to calibrate the balometer readings. The TEC equipment used a powered fan to measure and confirm the flow rate. Results indicate that the TSI equipment recorded approximately 19% lower airflow than the TEC powered flow hood in the expected airflow range for the PTHPs, which is <400 cubic feet per minute (CFM). Table 7 shows the differences between the TEC and TSI airflows used for calibrations.

The calibration setup between the TEC and TSI introduced potential sources of error. The TEC equipment created significant negative pressure on the TSI equipment within the airflow range of the PTHPs. Additionally, small leakage points in the connection between the TEC fan and TSI balometer could have affected the accuracy of the measurements. Therefore, the TEC equipment likely reported a higher airflow than the TSI equipment. To address this discrepancy, the team decided to adjust airflows up to half the difference in readings, resulting in a 9% increase from the TSI readings. Table 8 shows the adjusted airflow values for each heat pump at each fan speed. These values, used in calculations, incorporate the adjustments made after comparing the equipment. Figure 10 summarizes the calibration process for the airflow measurements from the PTHPs.

Table 6. Supply Airflow Measurements

TSI Alnor Measurements	Low Fan Speed (CFM)			High Fan Speed (CFM)		
	Minimum	Average	Max	Minimum	Average	Max
Small (RSXC09) n = 8	200	223	240	300	311	350
Medium (RSXC13) n = 4	210	227	240	310	313	320

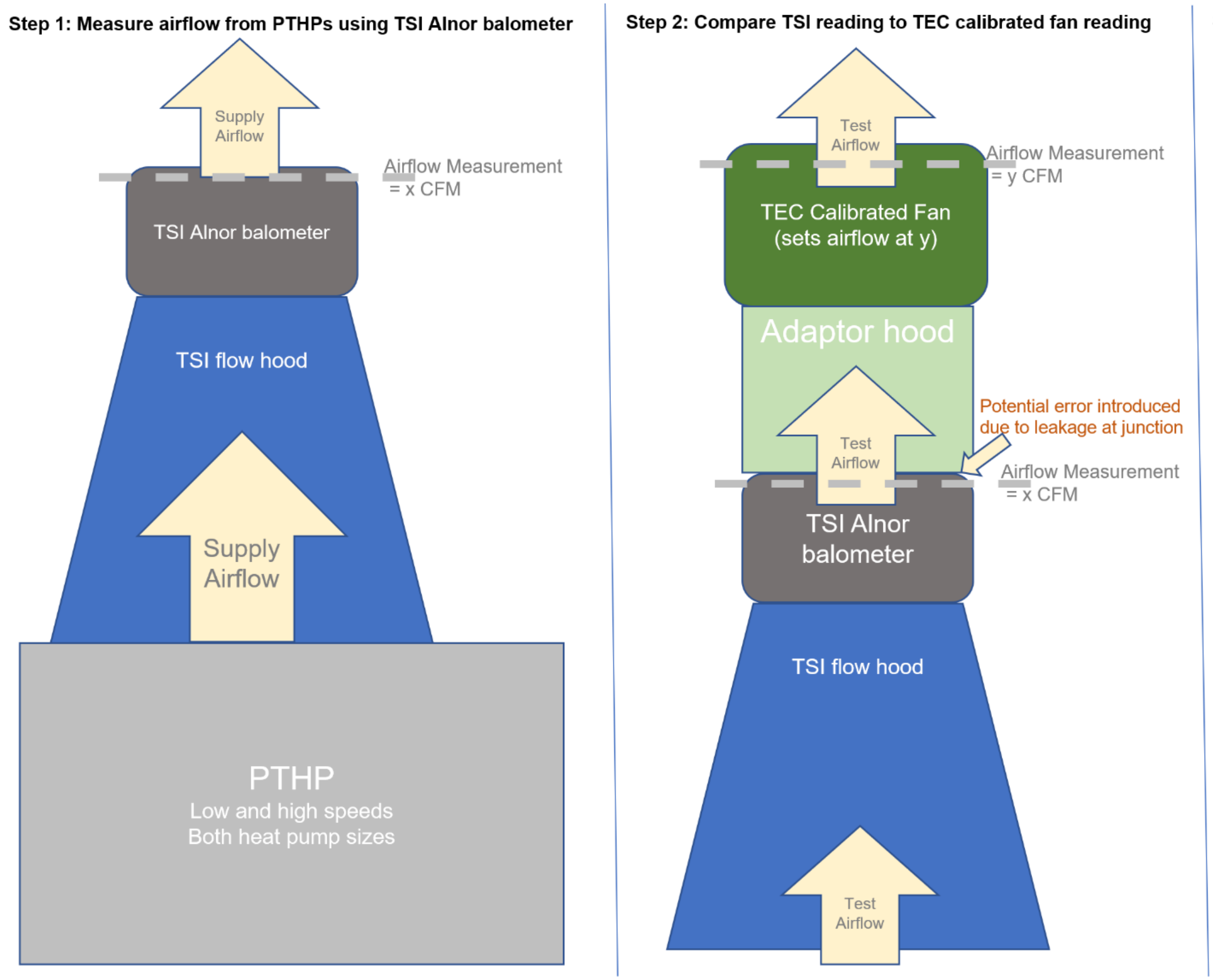
Table 7. Comparison of The Energy Conservatory and TSI Airflows

TEC Fan Reading (CFM)	TSI Alnor (CFM)	Difference (TEC-TSI)/TSI
245	206	19%
403	340	19%
502	435	15%

Table 8. Airflow Values Used in Calculations

	Low Fan Speed (CFM)			High Fan Speed (CFM)		
	Minimum	Average	Max	Minimum	Average	Max
Small (RSXC09) n = 8	219	243	262	328	339	382
Medium (RSXC13) n = 4	229	248	262	339	342	350

Figure 10. Summary of Airflow Measurement and Calibration Process



Step 3: Adjust readings for use in calculations

TSI measured error: (y-x)/x = 18.5%

Estimate of introduced error in test setup: 9%-10%

Result: adjust all TSI readings from PTHP upward by 9%

2.4.5 Determining Uncertainty

The team calculated the uncertainty based on the accuracy of the sensors and their typical measurement ranges. The sensors used for temperatures included Invonics EN1723 sensors. The team completed all temperature measurements with a 10 kiloohm (kΩ) thermistor, except for room temperature, which used an internal sensor. Table 9 shows the parameters for the sensors.

Table 9. Temperature Sensor Accuracy

EN1723 with External 10kΩ Thermistor[11]	Value	Units	EN1723 with Internal Sensor[5]	Value	Units
Accuracy	0.5	% full scale	Accuracy	0.5	% full scale
Min reading	-22	°F	Min reading	-13	°F
Max reading	212	°F	Max reading	140	°F
Full scale range	251	°F	Full scale range	151	°F
Accuracy	1.255	°F +/-	Accuracy	0.8	°F +/-

The team measured the amperage of the PTHPs using an Invonics EN1702 with Honeywell Clamp CTs. Table 10 displays the accuracy of this sensor apparatus.

Table 10. Amperage Sensor Accuracy

Amperage	Value	Units
Accuracy	1	% full scale
Min reading	0	Amps
Max reading	10	Amps
Full scale range	10	Amps
Accuracy	0.1	A +/-
Anticipated range of current	0 to 8	Amps

The team determined airflow values following the process outlined in section 2.4.4, categorizing readings based on data from the Invonics EN1702 with Honeywell P7650A1026, which measures differential pressure. Table 11 shows the accuracy of this process.

Table 11. Airflow Measurement Accuracy

Airflow[12]	Value	Units
Typical measurement (low speed)	240	CFM
Min measurement	216	CFM
Max measurement	259	CFM
Range of potential error	43	CFM
Accuracy	9%	+/- of avg

The temperature data was rounded to the nearest whole number during analysis to account for the limited accuracy. Sentient Buildings completed the installation of the sensors and facilitated data collection. The team calculated potential error in heating output using the formula:

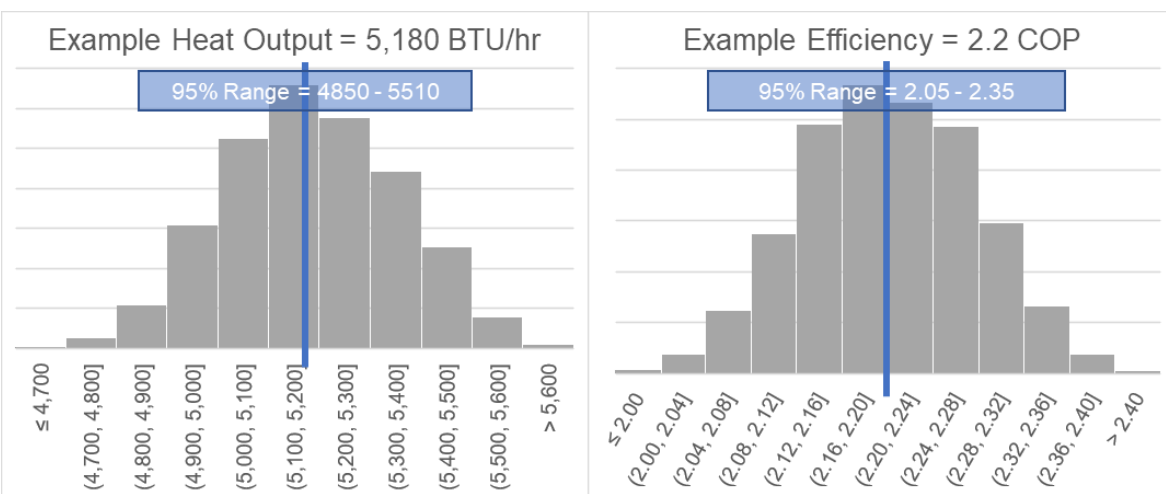
$$\frac{\boldsymbol{BTU}}{\boldsymbol{hr}} = \mathbf{1.08} * \boldsymbol{CFM} * \boldsymbol{dT}$$

The team treated CFM uncertainty as uniformly distributed and dT uncertainty as normally distributed. The team calculated the error range of the heating output using random sampling from within each variable's error distribution. Monte Carlo simulation (n = 3000) results show the expected distribution in Figure 11.

After incorporating the potential error from the current draw measurements, which determine heat pump input, the team determined the uncertainty of the COP, as represented in Figure 11. The expected heat pump output error is 6% and the expected efficiency error is 7%.

Figure 11. Monte Carlo Simulation of Heating Pump Output and Coefficient of Performance

Results based on 3,000 iterations



3 Results and Discussion

The primary goal of the ccPTHP effort is developing a technology solution to displace and ultimately replace fuel-fired space heat.

During construction, the Coney Island multifamily building installed 12 Ice Air ccPTHP units. The team collected and analyzed data from 11 of these units from January 1, 2022, through March 15, 2023. This study focused on cold climate performance, emphasizing the collection and analysis of winter data. Due to warmer-than-expected outdoor air temperatures, the dataset included a low percentage of low OAT data, with only ~8% of data points below freezing. Therefore, the researchers used temperature bins of 5°F to allow for calculations within temperature ranges that had low quantities of data. Generally, summaries in this section concentrate on the outdoor temperature bins below 40°F, the temperature below which non-cold-climate PTHPs tend to rely on electric resistance heating elements instead of the heat pump to satisfy heating requirements.

The Ice Air ccPTHP units were installed during the building's initial construction. The developer installed all units with hot water coils to provide heating from the central boiler if needed, addressing concerns about backup heat. The coils' isolation valves were closed at the time of installation; however, the valves for apartments 1 and 2 were opened during construction, allowing hot water to flow in heating mode which skewed supply air temperature measurements. As a result, the heat pumps in these apartments were excluded from heating analyses because their heating was supplemented by the central system.

3.1 Heating Mode and Space Temperature

Research question: Does the RSXC heating system provide enough heat without backup?

For an initial analysis, the team filtered the data to focus on instances when the heat pump operated in heating mode. Table 12 presents the maximum, minimum, and average room temperatures within each temperature bin. This data shows that all units maintained warm, comfortable space temperatures even at the coldest observed temperatures. Occasionally, space temperatures dipped into the mid- to high 60s°F, but these occurrences were infrequent.

Table 12. Room Temperature at Cold Outdoor Temperatures Less Than 40°F

Unit	OAT (°F)	Room Temperature (°F)			Unit	OAT(°F)	Room Temperature (°F)		
		Max	Mean	Min			Max	Mean	Min
3_1	[15, 20)	79	76	70	4_1	[15, 20)	81	77	74
	[20, 25)	79	77	71		[20, 25)	83	76	71
	[25, 30)	79	76	72		[25, 30)	80	75	74
	[30, 35)	80	76	68		[30, 35)	82	75	65
	[35, 40)	80	76	72		[35, 40)	87	75	67
3_2	[15, 20)	78	76	71	4_2	[15, 20)	82	76	74
	[20, 25)	78	76	72		[20, 25)	81	75	73
	[25, 30)	77	76	74		[25, 30)	79	74	71
	[30, 35)	80	76	68		[30, 35)	80	75	65
	[35, 40)	84	76	74		[35, 40)	82	75	71
3_3	[15, 20)	77	74	69	4_3	[15, 20)	81	76	74
	[20, 25)	76	74	71		[20, 25)	82	75	73
	[25, 30)	76	74	72		[25, 30)	83	76	73
	[30, 35)	76	74	66		[30, 35)	84	76	65
	[35, 40)	80	74	71		[35, 40)	85	75	72

Table A-2 shows the average room temperatures, regardless of the PTHP units' operational status. When the units operated, average room temperatures increased significantly compared to the overall average, indicating that the PTHPs effectively increased space temperature at low OAT. Monitored apartments did not contain backup heat sources. All instances where setpoints were satisfied indicate that the installed PTHP units maintained comfortable space temperatures.

To further assess whether the units provided adequate heat, the team reviewed consecutive periods of heating operation for changes in room temperature. If a unit's output was greater than or equal to the heat loss of the space (i.e., the heating load), the room temperature should either have increased or remained steady from one period to the next, assuming the setpoint remained constant. If the heat pump operated in heating mode for two consecutive periods but room temperature decreased from one measurement to the next, assumedly the ccPTHP did not provide adequate heat. Exceptions would occur only if the unit had operated in defrost mode between measurements.

Table 13 shows the room temperatures and changes in room temperature for consecutive periods of operation within each OAT bin. Consecutive periods of usage represent approximately 25% of the total heating mode data points; however, about 38% of those data points showed decreases in room temperature from the previous measurement. This means that room temperature decreased from the previous period less than 10% of the time when the units were operated in heating mode below 40°F.

Since the units rarely ran for consecutive 15-minute periods, and when they did the room temperature increased more often than it decreased, the team concludes that the ccPTHPs provided sufficient heat to the space without requiring backup.

Table 13. Heating Mode Room Temperature Change at Cold Outdoor Temperatures in Consecutive Data Points

Unit	Temperature		Room Temperature Change (°F)		Room Temperature (°F)		
	(°F)	Count	Max	Min	Max	Mean	Min
3_1	[15, 20)	5	1	0	75	73	70
	[20, 25)	12	1	-1	78	76	73
	[25, 30)	13	1	-1	79	76	74
	[30, 35)	14	2	-1	79	78	75
	[35, 40)	13	3	-2	79	77	72
3_2	[15, 20)	8	1	0	77	75	71
	[20, 25)	8	2	-2	77	75	72
	[25, 30)	5	0	0	77	76	76
	[30, 35)	8	8	-4	80	76	68
	[35, 40)	3	2	1	83	82	81
3_3	[15, 20)	35	3	-2	77	75	69
	[20, 25)	35	1	0	76	74	71
	[25, 30)	20	0	-1	75	73	72
	[30, 35)	51	1	-7	76	73	66
	[35, 40)	70	3	-3	80	74	72
4_1	[15, 20)	52	2	-4	81	78	74
	[20, 25)	77	2	-3	83	76	71
	[25, 30)	42	2	-3	78	76	74
	[30, 35)	92	3	-12	81	75	65
	[35, 40)	139	4	-12	87	75	67
4_2	[15, 20)	82	1	-6	82	77	74
	[20, 25)	104	1	-4	77	76	73
	[25, 30)	40	2	-3	79	75	71
	[30, 35)	91	2	-7	79	75	71
	[35, 40)	85	6	-8	80	75	71
4_3	[15, 20)	34	3	-1	80	77	74
	[20, 25)	69	2	-4	82	75	73
	[25, 30)	76	2	-1	82	76	73
	[30, 35)	143	4	-7	83	76	65
	[35, 40)	145	3	-10	84	77	72

The fourth-floor units experienced more frequent and larger drops in space temperature compared to the third-floor units. By isolating the data points where the temperature decreased between consecutive measurements while the heat pumps were on (see Table 14), the researchers found that average temperature decreases remained below 1°F. Larger temperature drops often occurred when rooms were already warm, possibly due to occupant behavior, such as opening a window or door, rather than inadequate heating capacity.

Table 14. Heating Mode Room Temperature, Decreases Only

Unit	Temperature (°F)	Count	Average Room Temperature Change (°F)	Average Room Temperature (°F)
	[15, 20)	11	-1	75
	[20, 25)	30	0	75
4_1	[25, 30)	17	-1	76
	[30, 35)	40	-1	75
	[35, 40)	58	-1	74
	[15, 20)	24	-1	77
	[20, 25)	22	-1	75
4_2	[25, 30)	17	-1	73
	[30, 35)	49	-1	74
	[35, 40)	43	-1	75
	[15, 20)	15	0	77
	[20, 25)	29	-1	76
4_3	[25, 30)	36	0	76
	[30, 35)	61	-1	76
	[35, 40)	57	-1	76

3.2 Maintaining a Comfortable Supply Temperature

Research question: Can the RSXC maintain comfortable supply air temperatures?

Even when room temperatures remained comfortable, low supply temperatures during heating mode could have affected resident comfort. Low supply air temperatures can make residents feel cold, despite an overall comfortable room temperature. Table 15 compiles the average supply temperature for the units. The data is limited to instances when the unit operated with the compressor on and in heating mode, excluding defrost mode operation. The table also shows the percentage of total points where supply temperatures fell below 80°F.

Table 15. Supply Air Temperature Average and Percentage Below 80°F Threshold

OAT (°F)	Avg (°F)	% Below Threshold	Avg (°F)	% Below Threshold	Avg (°F)	% Below Threshold
	3_1		3_2		3_3	
[15, 20)	80	53%	74	93%	79	76%
[20, 25)	81	39%	75	94%	76	82%
[25, 30)	82	28%	75	98%	75	91%
[30, 35)	83	30%	79	79%	76	82%
[35, 40)	85	15%	88	60%	77	72%
Overall	83	30%	78	87%	76	81%
	4_1		4_2		4_3	
[15, 20)	91	14%	95	8%	87	15%
[20, 25)	88	33%	91	19%	84	31%
[25, 30)	91	35%	89	18%	85	19%
[30, 35)	87	43%	89	21%	85	30%
[35, 40)	88	40%	89	18%	86	32%
Overall	88	37%	90	18%	86	28%

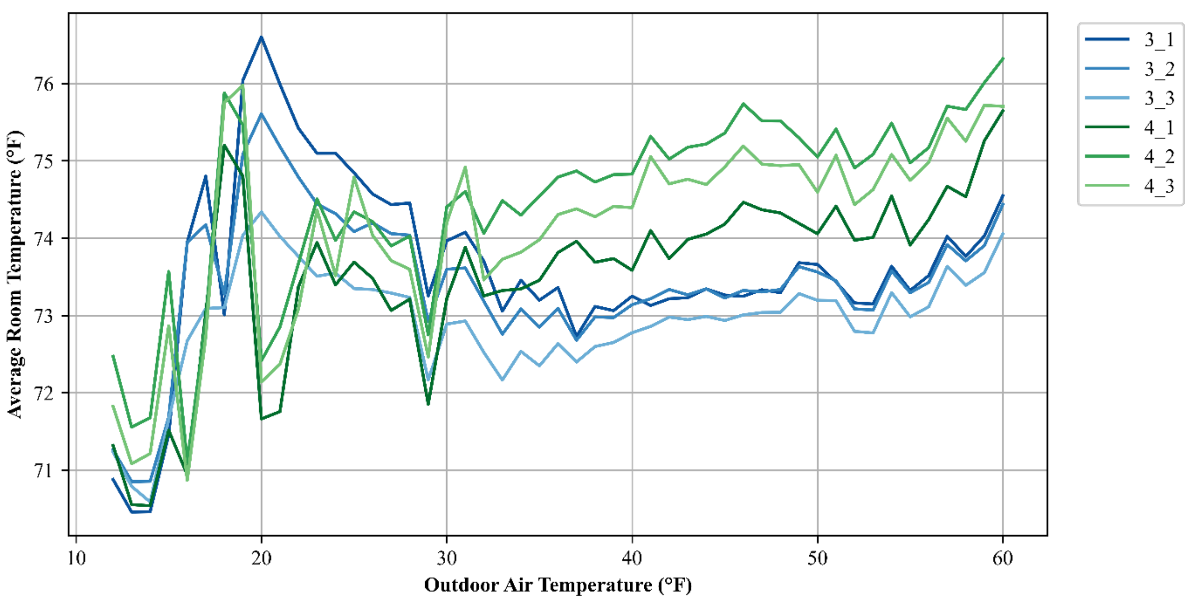
For this study, we considered 80°F the minimum supply air temperature acceptable for occupant comfort. The units in apartment 3 frequently supplied air below this 80°F threshold and provided lower overall supply temperatures compared to the units in apartment 4. The disparity likely results from how occupants used the PTHPs. Trends on the coldest day of the year indicate that the PTHPs in apartment 3 turned on infrequently and only at specific times, whereas the PTHPs in apartment 4 operated more continuously, allowing for extended periods of use. Figures A-4 and A-5 in Appendix A show full performance trends for the coldest day in the data collection window. To further assess the impact of supply temperatures below 80°F, the researchers isolated points below the threshold. The team calculated and grouped average and minimum room temperatures when the supply air temperature was below 80°F within OAT bins. Table 16 details these findings.

Table 16. Minimum and Average Room Temperatures at Low Supply Temperatures

Unit	3_1		3_2		3_3		4_1		4_2		4_3	
OAT (°F)	Min	Mean	Min	Mean	Min	Mean	Min	Mean	Min	Mean	Min	Mean
[15, 20)	70	75	71	76	72	75	74	74	74	75	74	75
[20, 25)	71	75	72	76	71	74	74	75	73	74	73	75
[25, 30)	73	75	74	76	72	74	74	75	72	74	73	74
[30, 35)	73	75	74	76	72	74	74	75	71	74	73	75
[35, 40)	72	74	74	76	71	74	73	74	71	74	73	75
[40, 45)	72	75	75	76	71	73	70	75	72	74	73	75
[45, 50)	69	74	77	77	72	73	74	75	74	75	73	76
[50, 55)	72	74	73	74	72	73	74	74	n/a	n/a	73	77
[55, 60)	72	73	72	73	72	73	74	75	73	74	73	77

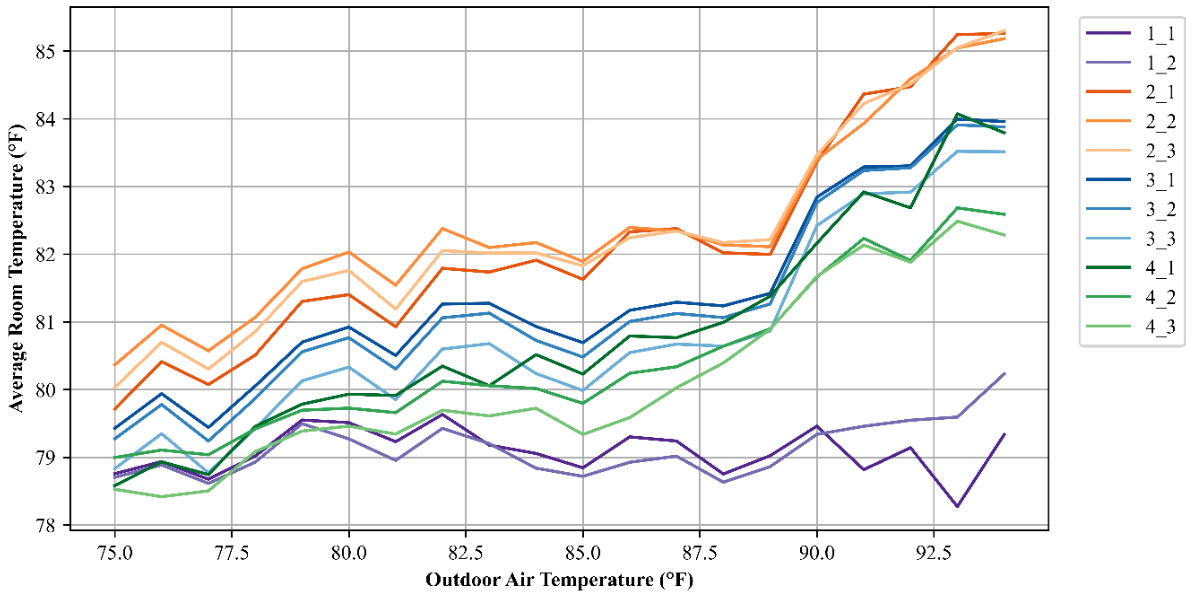
Minimum room temperatures rarely dropped below 70°F, with average room temperatures hovering around 75°F even within low OAT bins. These space temperatures indicate that, despite the low supply air temperatures, these rooms maintained comfortable temperatures. Figure 12 displays the overall room temperatures of the units in heating mode. This data portrayed is unfiltered and includes periods where the units did not operate. On average, room temperatures in apartment 4 were higher than those in apartment 3, indicating a difference in occupant preference and usage.

Figure 12. Average Room Temperatures in Heating Season



Although the analysis primarily focused on the heating season, the team also analyzed room temperature during the cooling season. Figure 13 presents the average room temperature of each unit based on the OAT during this season. The chart includes data for all units and is not filtered based on the operational mode.

Figure 13. Average Room Temperatures in Cooling Season



3.3 Temperature Profile on the Coldest Day

Evaluating heat pump performance at temperature extremes is key to understanding their effectiveness in cold climates. To evaluate the performance of the units at extreme temperatures, the researchers plotted temperature trends for each unit on the day with the coldest average OAT. Figure 14 displays supply temperature, return temperature, coil temperature, and room temperature for unit 4_1 with shaded regions indicating when the unit operated.

Throughout the day, room temperature did not fall below 70°F and remained close to 80°F while the unit operated. The unit did not run for most of the day; therefore, the bedroom temperature served by unit 4_1 could have received support from the living room unit or benefited from solar gains.

Figure 14. Coldest Day Performance of Unit 4_1

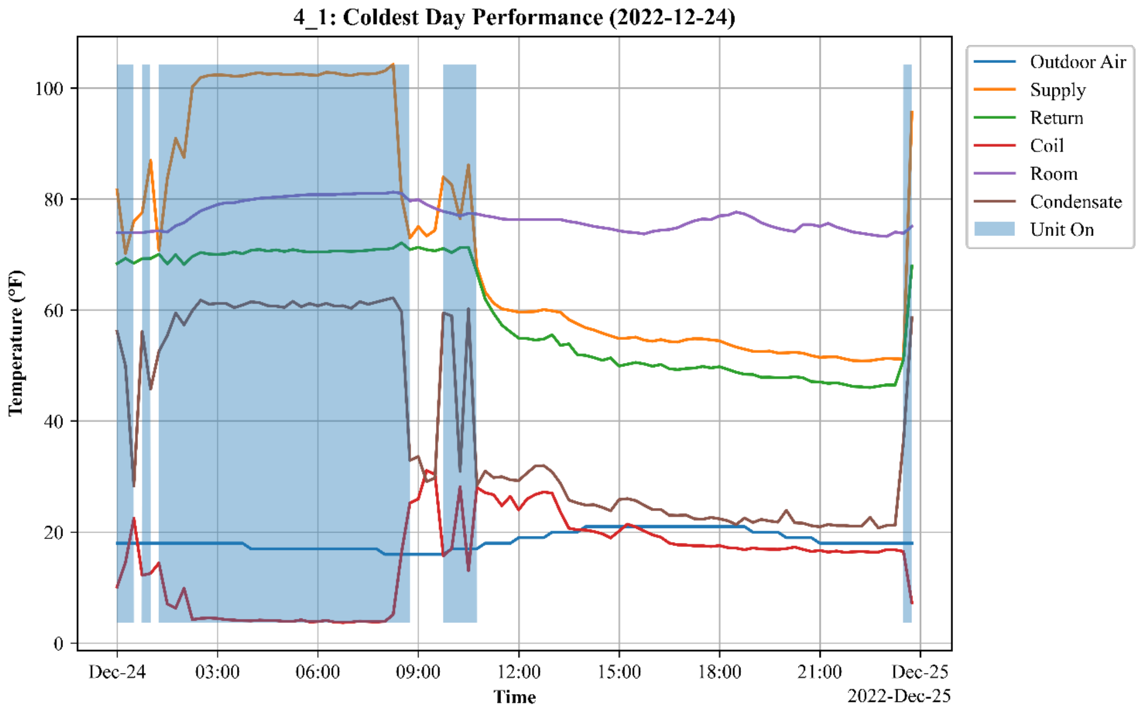
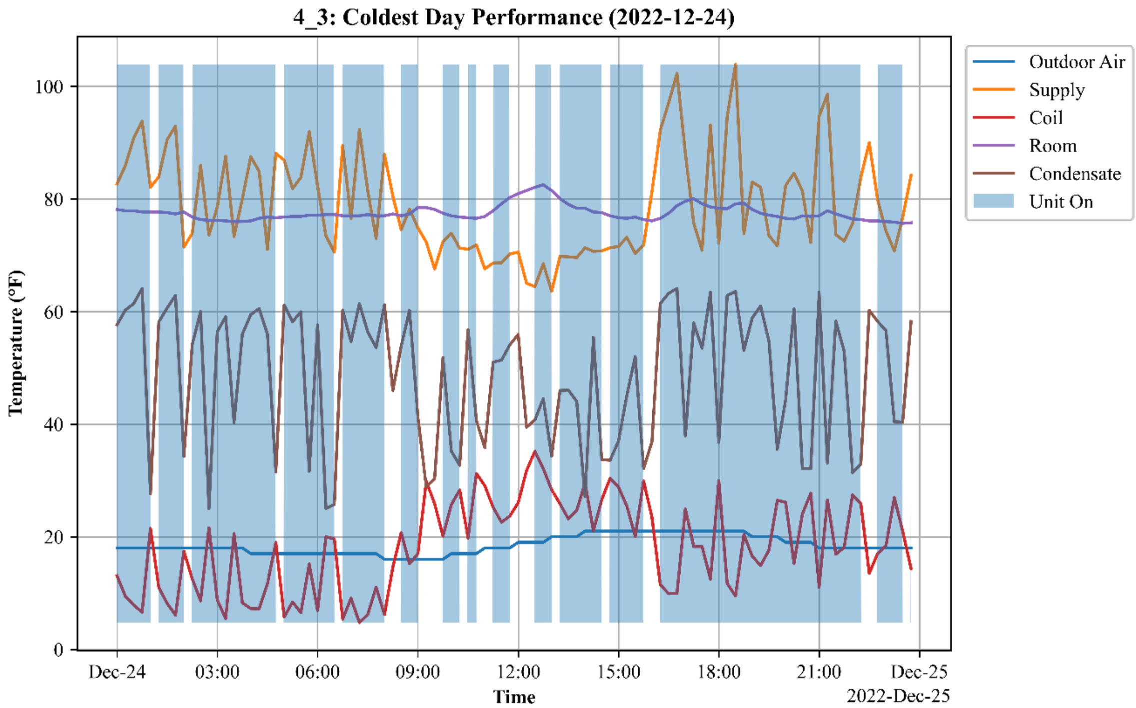


Figure 15 shows the performance of unit 4_3, which served the other bedroom. Unit 4_3 operated for most of the day, maintaining high room and supply temperatures. Fluctuations in the coil temperature indicate that the unit frequently switched on and off throughout the day, even during periods where amperage measurements suggested continuous operation. This discrepancy may result from the 15-minute data intervals exceeding the unit's typical run time for one cycle. Additional figures showing the temperature profiles for all units in apartments 3 and 4 units are available in Appendix A.

Figure 15. Coldest Day Performance of Unit 4_3



3.4 Temperature Profile on the Warmest Day

The temperature trends on the day with the highest average OAT were plotted. The units from apartment 1, shown in Figures 16 and 17, represent typical temperature profiles of the PTHP units on warm days. The units had the highest usage among all PTHP units on the warmest day of data collection. Usage varied throughout the day, with room temperature responding quickly when the units switched to cooling mode. The living room unit, 1_2, has a slower rate of temperature change due to the larger space being cooled.

Figure 16. Warmest Day Performance of Unit 1_1

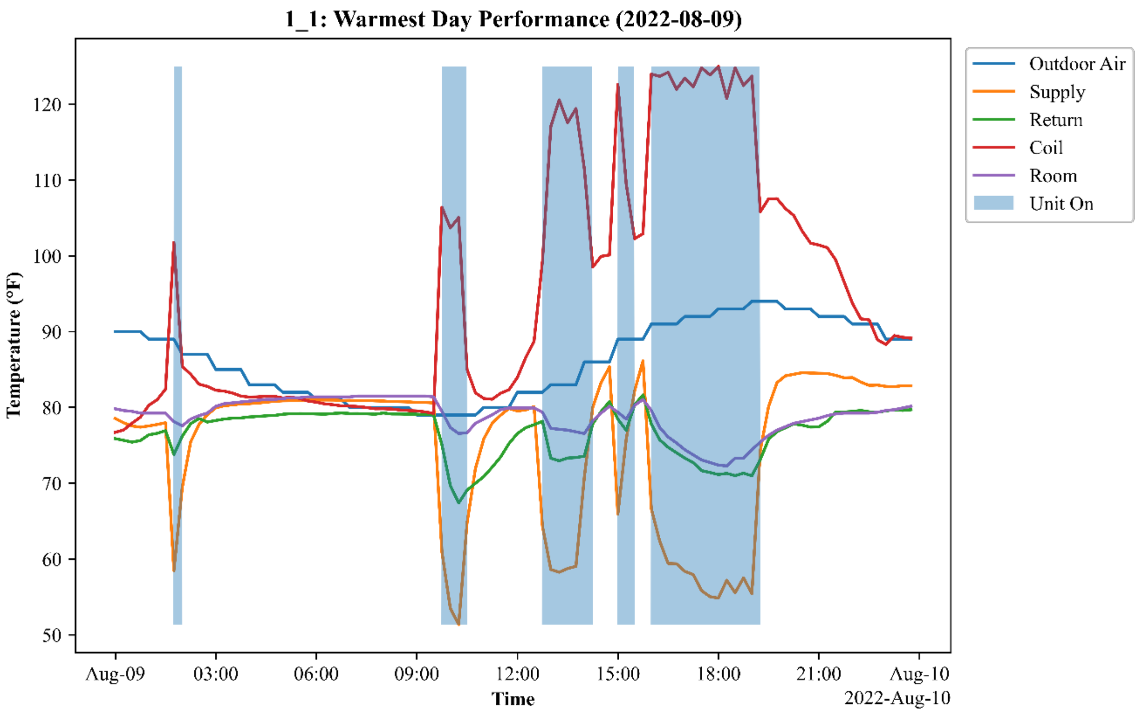
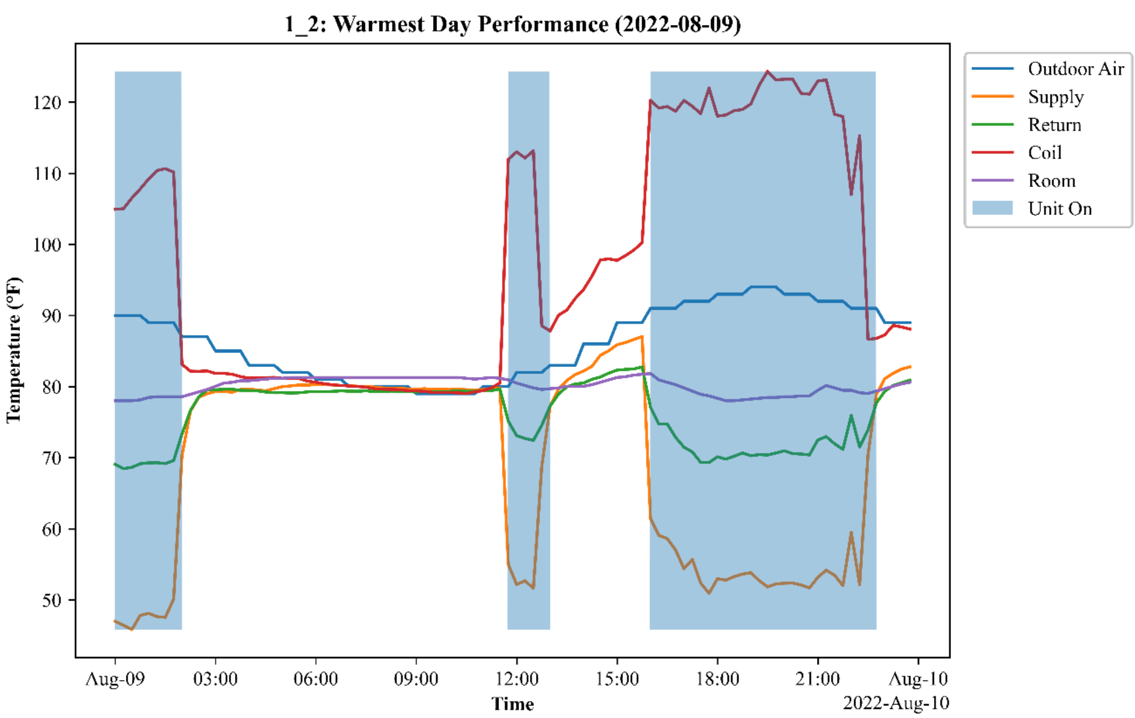


Figure 17. Warmest Day Performance of Unit 1_2



3.5 Heating Efficiency at Different Outside Air Temperatures

Research question: What is the seasonal efficiency of the systems for heating?

To determine how efficiently the units operated, the team calculated the COP across different OAT bins, both excluding and including points where the units were in defrost mode. Table 17 shows the COP values across different OAT ranges, excluding defrost points. Section 3.9 further explores the effect of defrost mode on efficiency.

Table 17. Coefficient of Performance at Different Outdoor Air Temperatures by Unit

OAT (°F)	3_1	3_2	3_3	4_1	4_2	4_3
[15, 20)	1.4	1.6	1.1	2.1	2.2	1.3
[20, 25)	1.6	1.9	1.1	1.9	2.1	1.1
[25, 30)	1.8	1.9	0.9	2.3	2.2	1.2
[30, 35)	1.8	2.2	1.1	1.9	2.1	1.3
[35, 40)	2.0	2.7	1.1	2.1	2.3	1.5
[40, 45)	1.4	1.9	1.2	1.7	2.5	1.5
[45, 50)	1.4	2.2	1.2	1.3	2.4	1.2
Seasonal	1.7	2.1	1.2	2.2	2.3	1.5

The COP differed between the different capacity units, with the RSXC09s having a lower average COP than the RSXC13s. The team conducted a comparison between the models of PTHPs using total energy input and output for each model type to calculate both models' temperature-binned COP in each temperature range. Table 18 presents this comparison. The discrepancy between the COPs of the two models possibly stems from differences in the loads each unit served relative to their design capacities. Section 3.6 discusses the relationship between sizing and efficiency further.

Table 18. Coefficient of Performance at Different Outdoor Air Temperatures by Packaged Terminal Heat Pump Model

OAT (°F)	RSXC09	RSXC13
[15, 20)	1.5	2.1
[20, 25)	1.4	2.0
[25, 30)	1.4	2.1
[30, 35)	1.4	2.1
[35, 40)	1.6	2.3
[40, 45)	1.5	2.5
[45, 50)	1.2	2.3
Seasonal	1.7	2.3

3.6 Heat Pump Output at Different Outdoor Air Temperatures

The heat pumps' outputs at various OATs determined whether the installed PTHPs reached or exceeded their rated capacities. For a unit with a rated output capacity closely matching the space loads, the measured heat pump output (HPO) should increase toward the rated capacity as the OAT decreases. If a unit's minimum output capacity significantly exceeds space loads, the measured output would likely remain relatively constant since it does not need to increase output to maintain space temperatures.

3.6.1 Heat Pump Output in Heating

The analysis of heating mode output focused on apartments 3 and 4. The average heating mode thermal output for each unit in Figure 18 calculated in 5°F bins. The living room units, 3_2 and 4_2, have higher capacity than the bedroom units, and the difference in their operation is notable in the chart. Unit 4_2 had an increasing output at colder OATs, whereas the output of unit 3_2 at warmer OATs.

Apartment 3 was largely unoccupied during the coldest weather, explaining the sudden shift in usage between temperature bins. Figure 19 breaks down the heat pump output of each unit, showing all instances of measured heat pump output recorded while the unit operated in heating mode. The NEEP-listed maximum capacity and Ice Air–rated capacity for each unit appear on the charts for comparison.

Figure 18. Average Heat Pump Output of Each Unit: Heating

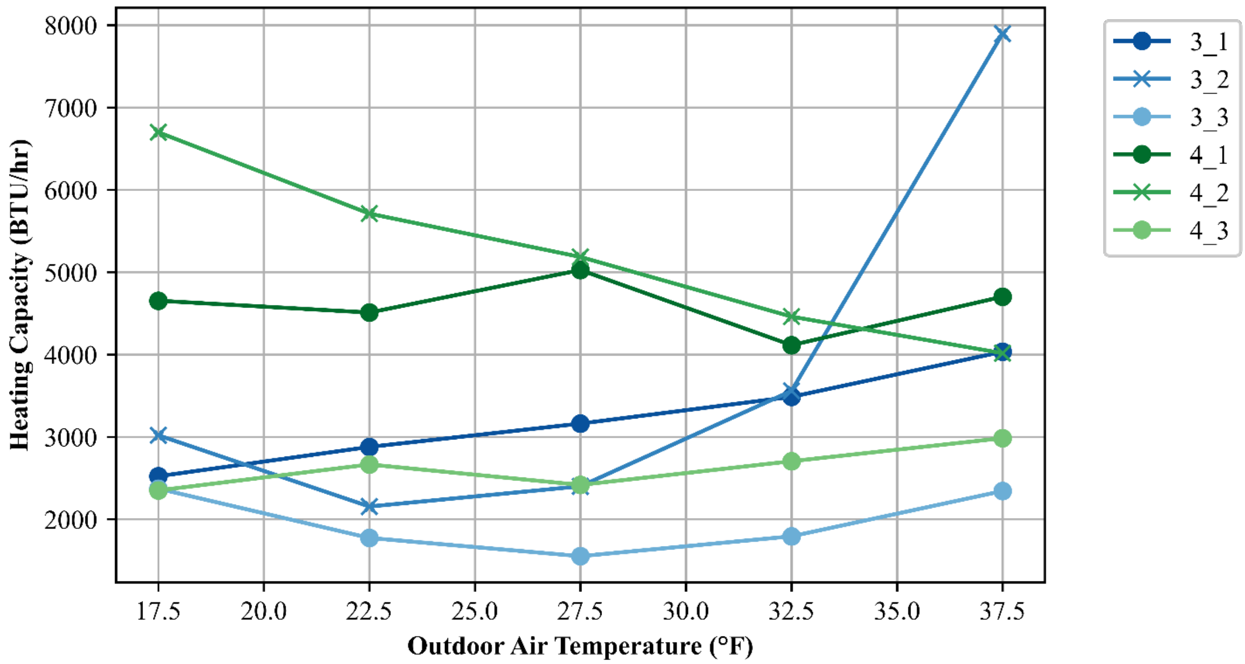
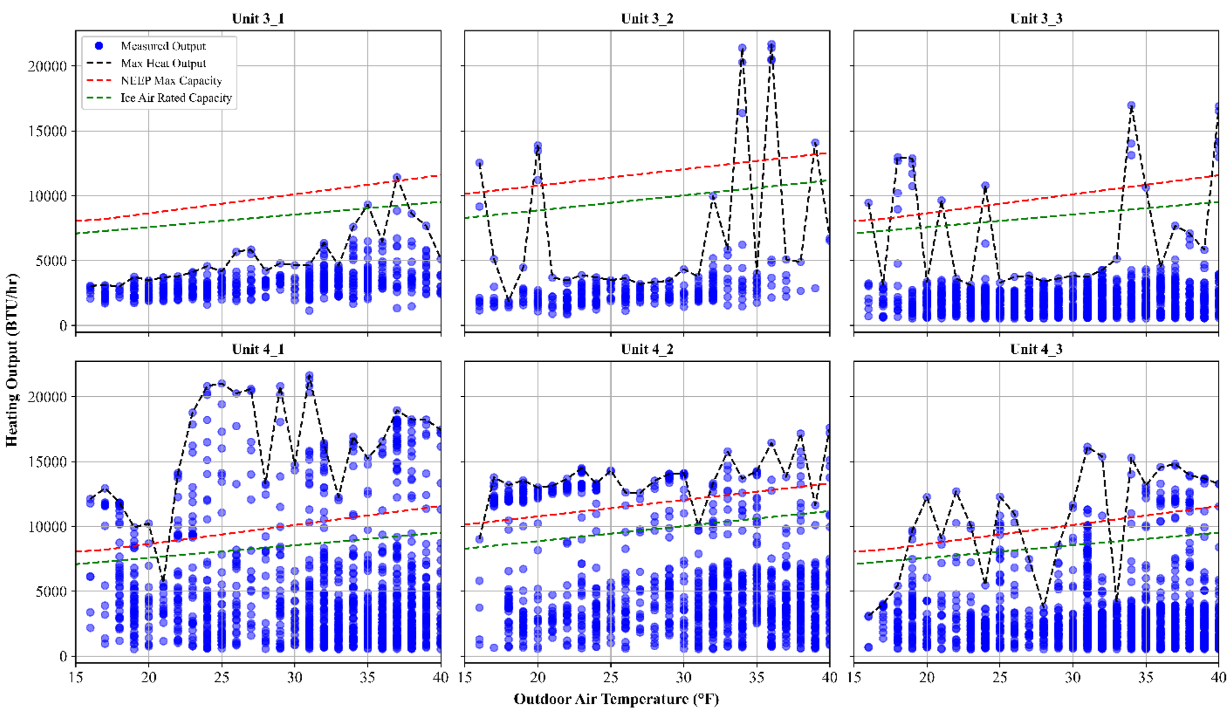


Figure 19. Heat Pump Output per Unit Comparison: Heating



Some individual output measurements significantly exceeded the rated capacity of the units, and the general trends for each apartment differ. Apartment 3 appears as if the installed unit capacities may be too large because the heat pump output remains well below the rated capacity at all OATs. Apartment 4 had higher heat pump output, but most measurements still fell well below the rated capacity, even at low OATs. While setpoint temperatures were not recorded, room temperatures in apartment 4 generally remained higher throughout the heating season, as shown in Figure 12, which could explain the higher HPO. Although many measured output points were above the NEEP maximum heating capacity, most were within expected ranges. Table 19 shows the percentage of output data points exceeding the listed NEEP maximum heating capacity.

Table 19. Percentage of Heat Pump Output Points above Maximum NEEP Rating per Unit

Unit	Points Above Max (%)
3_1	0.2%
3_2	3.9%
3_3	1.5%
4_1	15.1%
4_2	14.9%
4_3	4.1%

Figure 18 shows that unit 4_2 experienced increasing average output as OAT decreased, indicating that the unit's output capacity aligns better with the actual space loads, leading to less cycling and more efficient operation. Indeed, Table 17 shows that unit 4_2 had the highest COPs across the six units.

3.7 Cooling Efficiency at Different Outside Air Temperatures

Research question: What is the seasonal efficiency of the systems for cooling?

Because relative humidity was not monitored, the team could not calculate cooling COP, as total cooling output consists of both latent cooling and sensible cooling. Therefore, the analysis of cooling effectiveness focused on the sensible cooling capacity for the 11 units monitored.

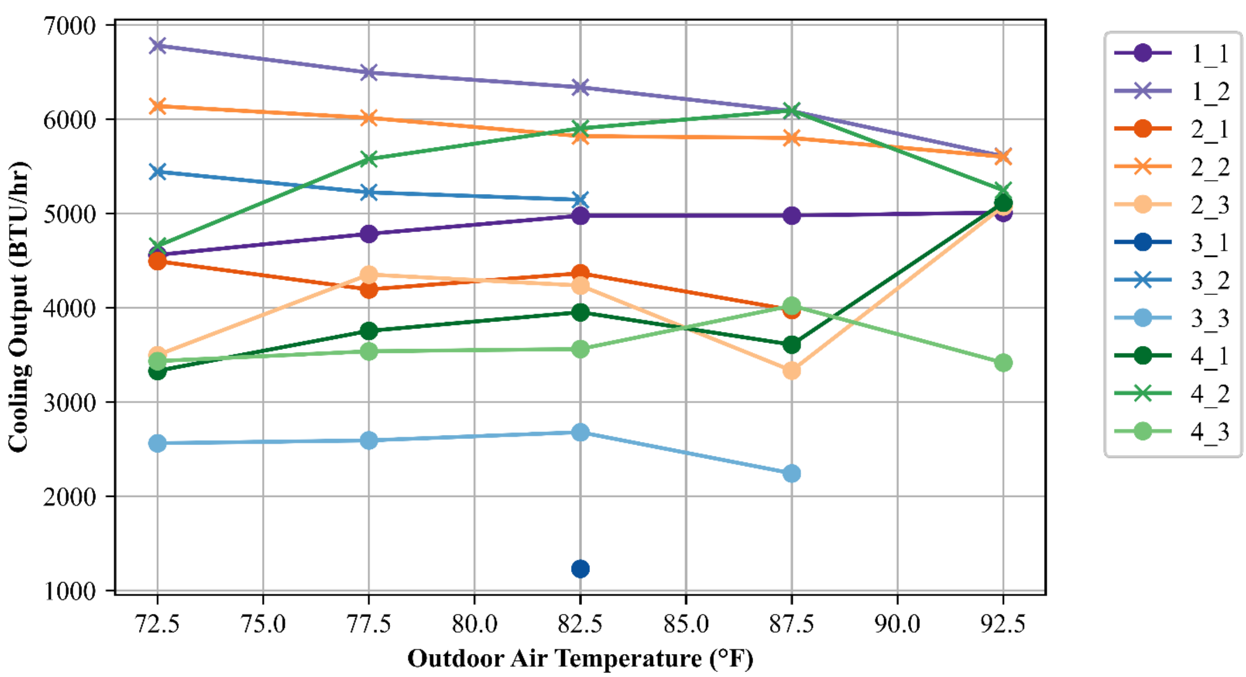
Table 20 displays the usage hours for each unit. Significant cooling began only above 65°F OATs. The team limited calculations for cooling mode to this range.

Table 20. Cooling Mode Usage (Hours) at Different Outside Air Temperatures

OAT (°F)	1_1	1_2	2_1	2_2	2_3	3_1	3_2	3_3	4_1	4_2	4_3
[60, 65)	0.50	0.00	0.25	1.25	2.00	0.00	3.00	26.00	0.50	5.50	9.50
[65, 70)	2.00	5.50	4.25	0.75	4.25	0.25	6.50	81.50	5.50	10.00	36.25
[70, 75)	9.00	29.25	30.00	1.50	2.50	0.00	1.25	123.25	34.75	33.25	135.25
[75, 80)	31.00	130.25	11.75	13.75	15.75	0.25	2.75	105.00	53.50	102.25	126.50
[80, 85)	25.75	161.25	13.00	22.00	15.25	0.00	2.25	54.75	38.00	91.50	72.50
[85, 90)	17.25	116.75	3.00	17.75	7.00	0.25	0.25	30.25	9.25	52.75	50.00
[90, 95)	13.75	63.50	0.00	12.50	2.75	0.00	0.00	0.00	3.25	11.25	7.50
Total	99.25	506.50	62.25	69.50	49.50	0.75	16.00	420.75	144.75	306.50	437.50

PTHP 3_1 operated in cooling mode very rarely across any temperature range, likely due to the apartment being unoccupied and the unit powered off for most of the cooling season (the resident moved in at the end of August 2022). Figure 20 presents the cooling output for each unit at different OATs. The relatively flat lines indicate that the units may be oversized for cooling as the output did not increase at higher OATs.

Figure 20. Average Cooling Output per Unit



3.7.1 Heat Pump Output in Cooling

Figures 21 and 22 display the comparison of the cooling output for each unit against the NEEP minimum and maximum cooling output.

Figure 21. Cooling Output by Unit (Apartments 1 and 2)

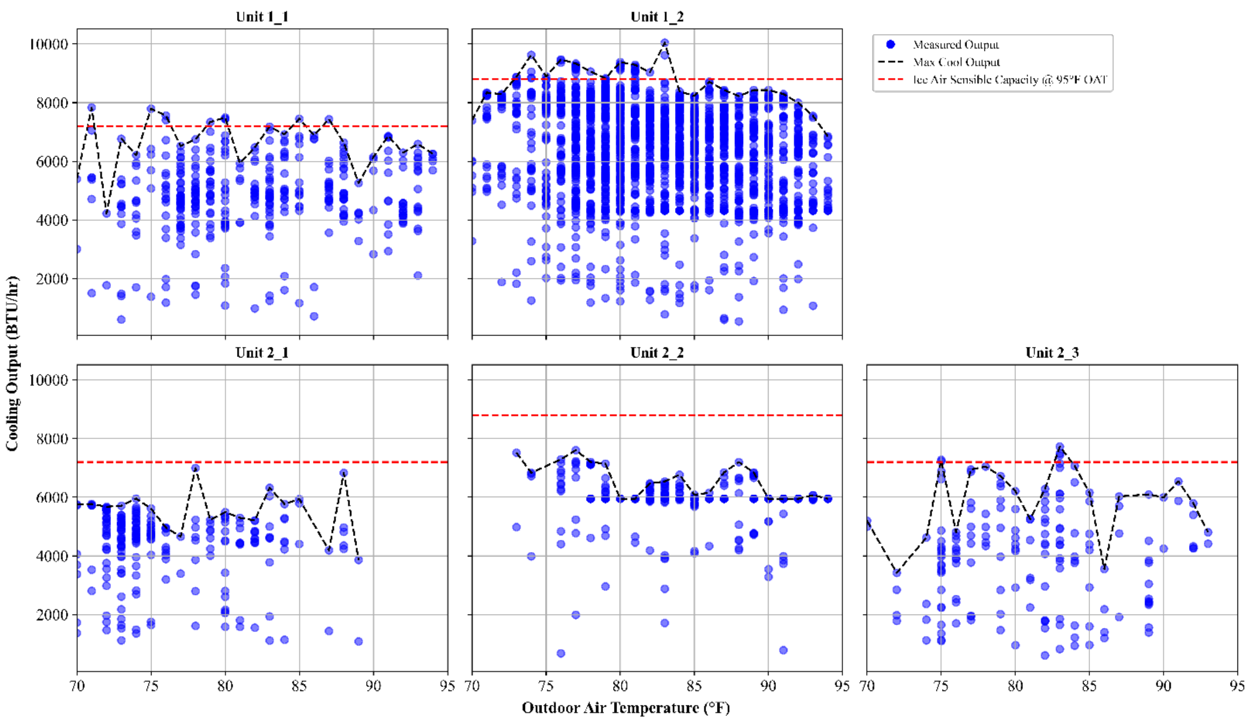
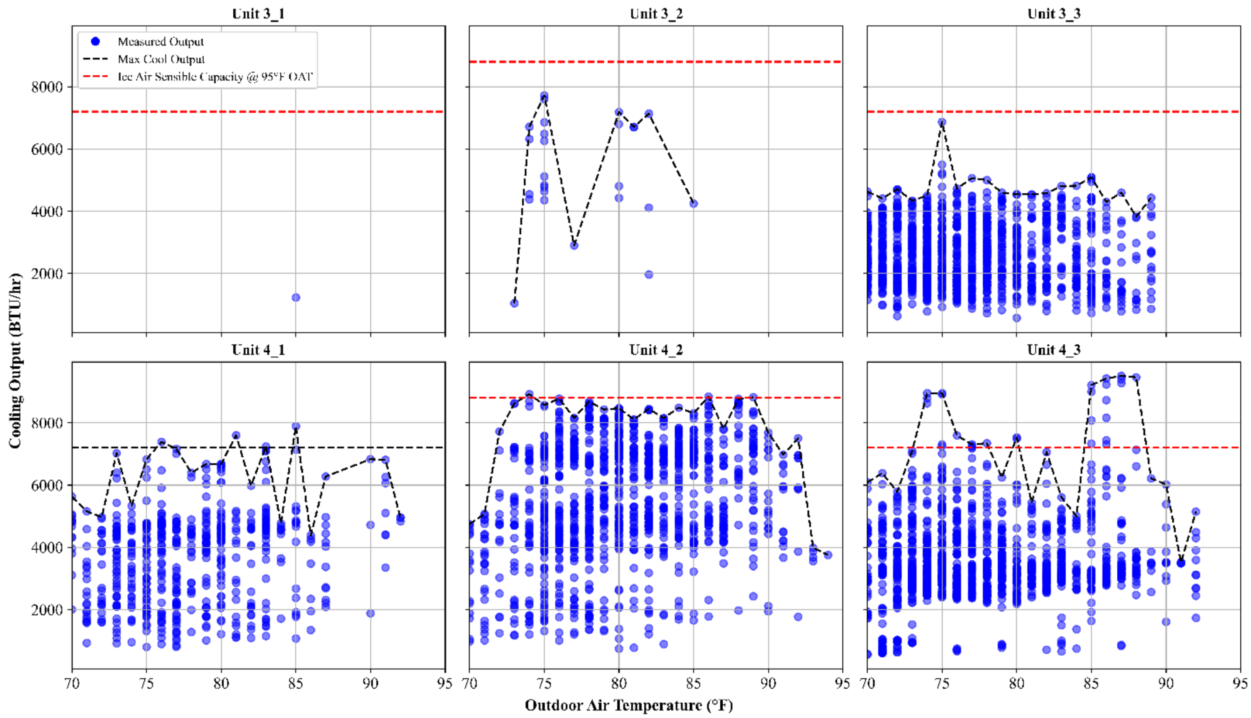


Figure 22. Cooling Output by Unit (Apartments 3 and 4)



3.8 Space Load Versus RSXC Capacities

SWA modeled the design heating and cooling loads for different zones during the building's design phase. All analyzed apartments shared the same layout and aligned vertically within the building. Table 21 presents the design loads for the entire apartment and the individual spaces. Selecting heat pumps for a specific typically depends on the cooling capacity of the heat pump and the cooling load of that space. The living room, kitchen, and bathroom received service from a larger capacity RSXC13 unit, which matched the modeled cooling and heating loads. The RSXC09 units, the smallest PTHP in the RSXC line, served the bedrooms. The cooling capacity of the bedroom units exceeded the cooling load by two times, while the heating capacity surpassed the heating load by three times. Oversized equipment increases the number of cycles per day while decreasing the length of each cycle, reducing the time a unit operates in a steady state.

Table 21. Apartment Heating and Cooling Loads

Space	SF	Sensible Cooling Load (Btu/hr)	Heating Load (Btu/hr)	Sensible Cooling Capacity (Btu/hr)	Heating Capacity (Btu/hr)
APT	652	15,412	16,025	23,200	32,400
BR1	130	3,073	3,195	7,200	10,200
LR/Kitchen/BR	382	9,030	9,389	8,800	12,000
BR2	140	3,309	3,441	7,200	10,200

3.9 Defrost Mode Effects on Efficiency and Comfort

Research question: Does the defrost mechanism work effectively without sacrificing comfort or efficiency?

3.9.1 Efficiency

Older or non–cold-climate PTHP models generally disable their compressors at low OATs, relying on electric resistance elements for heating below a certain threshold. In contrast, Ice Air cold climate PTHP units are designed to use their compressors down to -5°F OAT. Because heat is being pulled from the ambient air via the outdoor coil, the coil becomes cold and condenses moisture from the air. This moisture can freeze on the coils, prompting the unit to enter defrost mode, which consumes energy without providing heat output to the space.

The team determined the impact of the defrost mode on energy efficiency by calculating the COP of the PTHP units at different OATs, both including and excluding the energy consumed during the defrost cycles. These COPs for each unit, shown in Table 22, are marginally lower than the COPs found without including defrost.

Table 22. Coefficient of Performance With Defrost at Outdoor Air Temperatures by Unit

OAT (°F)	3_1	3_2	3_3	4_1	4_2	4_3
[15, 20)	1.4	1.6	1.0	1.9	2.1	1.0
[20, 25)	1.6	1.8	1.0	1.5	1.9	1.0
[25, 30)	1.8	1.8	0.8	2.0	2.0	1.2
[30, 35)	1.7	2.1	1.0	1.8	1.9	1.2
[35, 40)	1.9	2.7	1.1	1.9	2.0	1.4
[40, 45)	1.4	1.9	1.2	1.7	2.4	1.4
[45, 50)	1.4	2.2	1.2	1.3	2.4	1.2
Seasonal	1.7	2.0	1.2	1.9	2.1	1.4

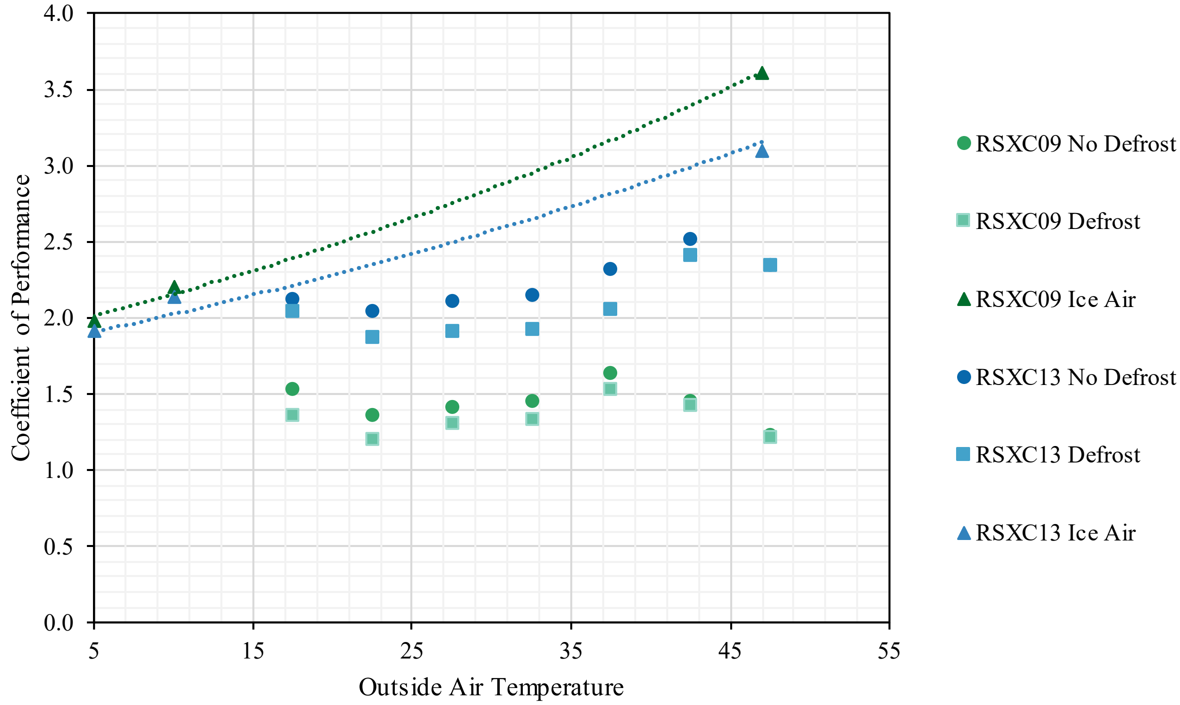
Table 23 compares COP values with defrost energy included versus excluded, revealing a decrease of between 0.1 and 0.3. The table also indicates how often the units operated in defrost mode as a percentage of the total hours in heating mode.

Table 23. Change in Coefficient of Performance With Defrost

OAT (°F)	RSXC09				RSXC13			
	Hours	COP	% Defrost Hours	Δ COP w/ Defrost	Hours	COP	% Defrost Hours	Δ COP w/ Defrost
[15, 20)	141	1.4	12%	-0.2	43	2.0	6%	-0.1
[20, 25)	282	1.2	13%	-0.2	84	1.9	12%	-0.2
[25, 30)	248	1.3	8%	-0.1	58	1.9	10%	-0.2
[30, 35)	456	1.3	9%	-0.1	99	1.9	11%	-0.2
[35, 40)	496	1.5	7%	-0.1	100	2.1	11%	-0.3
[40, 45)	262	1.4	2%	0.0	43	2.4	5%	-0.1
[45, 50)	105	1.2	0%	0.0	7	2.3	0%	0.0

Figure 23 presents the COPs of the RSXC models at different OATs, both including and excluding defrost, compared to the rated COP values based on Ice Air documentation.

Figure 23. Coefficient of Performance at Different Outside Air Temperatures with Defrost



3.9.2 Comfort

To determine the impact of defrost operation on comfort, the team limited the dataset to moments when the units operated in defrost mode. Average room temperature served as the primary indicator of occupant comfort. Table 24 shows the average room temperature in the spaces served by each unit while operating in defrost mode at different OATs.

Table 24. Average Defrost Operation Room Temperature

OAT (°F)	Room Temperature (°F)					
	3_1	3_2	3_3	4_1	4_2	4_3
[15, 20)	-	-	75	76	76	76
[20, 25)	-	77	75	75	75	75
[25, 30)	77	76	74	75	74	78
[30, 35)	76	75	74	76	75	77
[35, 40)	78	-	74	75	76	76
[40, 45)	-	-	78	76	77	76

Room temperatures were well within the comfortable range for all units during defrost operation. Defrost mode typically occurs between 20°F and 40°F OAT, when both the refrigerant coil temperature is low enough and the ambient air contains enough moisture to form frost on the outdoor coils.[13]

3.10 Condensate and Ice Melt

Research question: How much condensate/ice melt is produced?

Initially, the team intended to monitor how frequent ice melt water appeared in the drain lines; however, this detection proved infeasible, and water sensors were not installed. Sentient did not have a sensor capable of accurately detecting flow in the drain line, particularly because of the unreliability of detecting a relatively small volume of water in a non-full pipe.

Without the water sensor, the team could not determine whether the condensate temperature sensor measured actual condensate temperature or simply ambient air conditions. Nevertheless, during defrost cycles, the condensate temperature sensor recorded an average temperature of about 52°F, indicating low risk of condensate freezing in the drain lines.

3.11 Construction and Operation

Research question: Are there other design, installation, commissioning, or operations issues when installed?

L+M noted the following issues:

- Finding a pathway for the condensate line due to horizontal collection occurring below a beam on the second floor.
- Installing a relay switch to change between heating and cooling meters presented clearance and design challenges.
- Setting up monitoring devices proved difficult because of limited space within the PTHPs and poor signal quality.

Building operations staff reported no resident complaints or maintenance issues across the four apartments with ccPTHPs installed.

The relay switch L+M referenced allows the PTHPs to draw power from the apartment meter while operating in cooling mode and to switch to drawing power from an owner-paid meter while operating in heating mode. This installation avoids regulatory complications regarding charging residents for heating and Housing Preservation and Development utility allowances, particularly for a small subset of apartments in the building. This setup allows ownership to cover heating costs for residents without relying on a traditional central boiler plant.

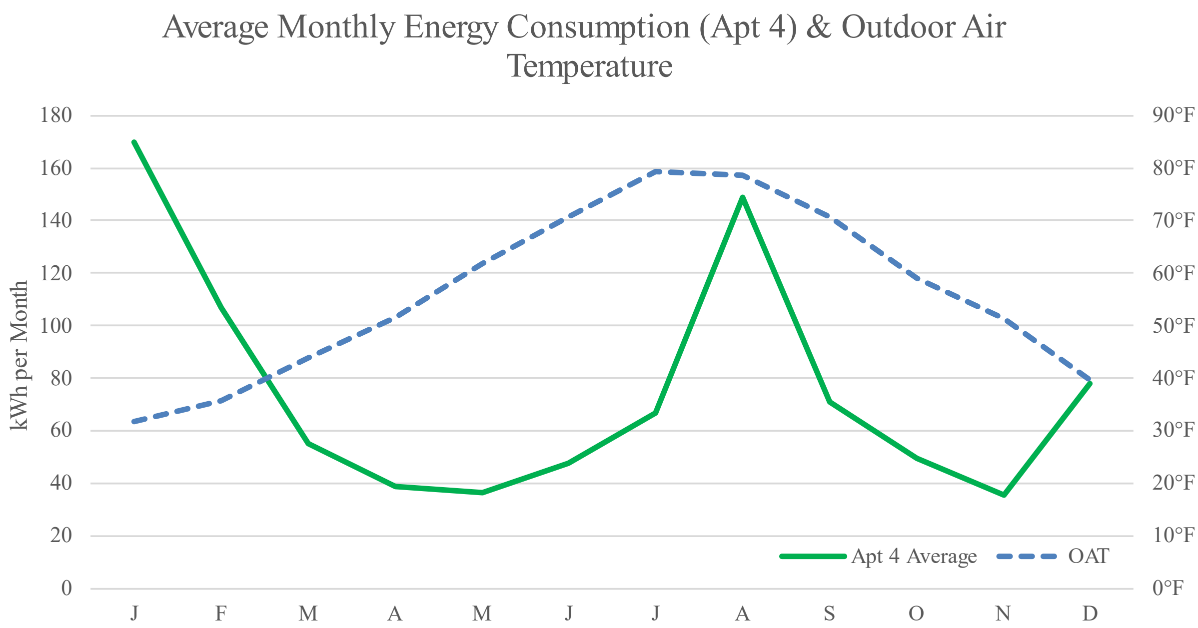
Research question: What is the estimated operating cost of a ccPTHP in new construction?

The average annual kilowatt hours (kWh) input across all monitored ccPTHPs was approximately 800 kWh. At an estimated utility rate of $0.20/kWh, the average unit would incur an annual cost of $160 to operate. Apartment 4, occupied for the largest portion of the monitoring period, had its ccPTHPs consume an average of 900 kWh in 2022. At an estimated utility rate of $0.20/kWh, these units would cost $180 each to operate annually, totaling $540 for the two-bedroom apartment.

To compare against a base case hot water PTAC receiving hot water from a central boiler plant, the team assumed an overall heating system efficiency of 80% and a natural gas utility rate of $1.20/therm.[14] Each unit would cost approximately $44 in gas to operate annually, totaling $132 for the two-bedroom apartment. The annual cooling costs for electricity would be approximately $75 per unit, or $225 for the entire apartment. In this scenario, total utility costs would be approximately $119 per unit, or $357 per apartment.

In the base case, heating costs would fall on the building owner, while residents would bear cooling costs. The building owner also covered the heating costs of the monitored ccPTHPs, which had relays enabling the units to draw electricity for heating from the base building electrical meter and for cooling from the apartment meter. A different property could implement a different metering configuration where residents are responsible for both heating and cooling electrical consumption.

Figure 24. Apartment 4 Average Monthly Energy Consumption and Outdoor Air Temperature



Research question: What is the estimated construction cost of the ccPTHP in new construction?

L+M reported an approximate premium of $41,000 for the 12 ccPTHPs included in the study, averaging to $3,417 per unit or $10,250 per apartment. The mechanical and plumbing scope for the base case PTAC was $29,000 per apartment, indicating an approximate 35% premium to install the ccPTHPs. This premium could likely decrease if the ccPTHPs were integrated into the design from the beginning, thereby reducing design and construction costs associated with condensate lines and allowing for bulk purchases of equipment.

4 Opportunities for Future Research

4.1 Increased Sample Size

This study initially included only 12 ccPTHP units. Missing monitoring equipment and operational issues, such as heating with central hot water, narrowed the final sample size for heating to six ccPTHP units, which consisted of two RSXC13 units and four RSXC09 units. A study with a larger sample size could generate clearer results and reduce the impact of outlying data.

4.2 Increased Data Collection Frequency

Data collection for this study occurred at 15-minute intervals. Reducing the time to 1- to 5-minute intervals could provide additional insight into unit cycling time. Ice Air confirmed that the unit typically runs for 10–40 minutes consecutively under normal conditions. A 15-minute reporting frequency may have excluded data points toward the lower end of that range. The data confirmed relatively few consecutive periods when the compressors were on. More frequent data collection could capture additional data points within a single run cycle.

4.3 Testing at Lower Outdoor Air Temperatures

Winter conditions at this site in New York City during data collection were uncharacteristically mild, with very few days recording OATs below 20°F. As shown in Table 1, the Ice Air PTHP units installed are rated to perform down to -5°F OAT. Testing these PTHPs in climates that reach such extreme OATs would offer further insight into the efficiency and effectiveness of these units at low ambient temperatures. Given the site's coastal location, milder weather was expected compared to other inland areas, including parts of New York City farther from the moderating effects of the ocean.

5 References

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Appendix A. Load Sizing Calculation Models

A.1 Modeled Apartments Photos and Floor Plans

Figure A-1. Basic Apartment Floor Plan

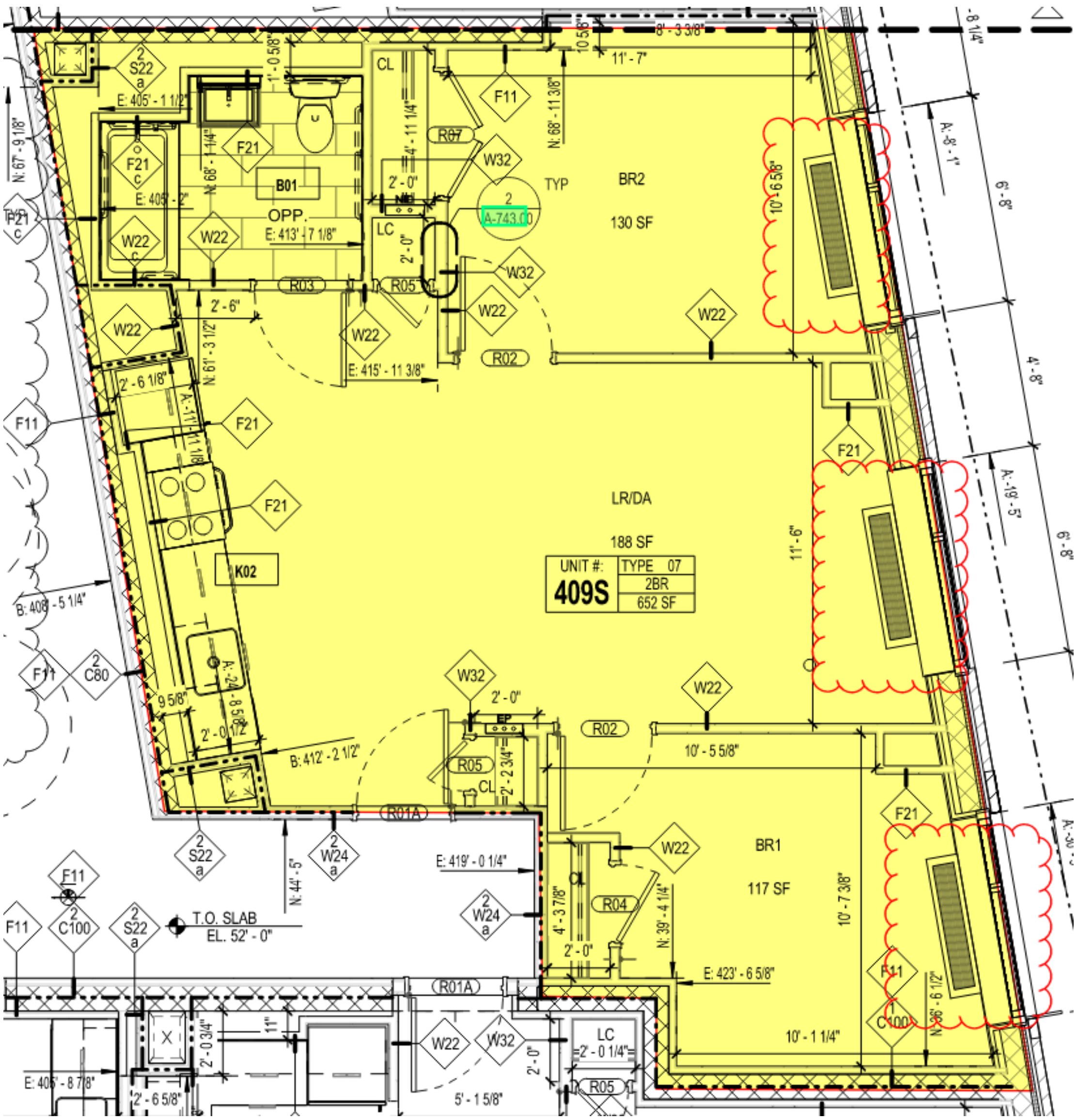


Figure A-2. Exterior View of Monitored Apartments

Figure A-3. Installed Cold Climate Packaged Terminal Heat Pump Images

A.2 Model Outputs and Reports from Software

Table A-1. Instances of Low Supply Temperature at Different Outdoor Air Temperatures

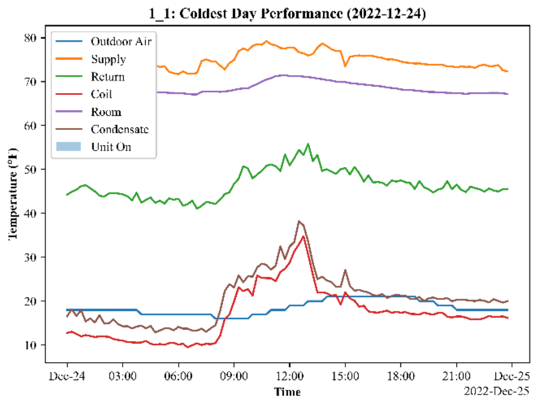
OAT (°F)	3_1	3_2	3_3	4_1	4_2	4_3
	Count	Count	Count	Count	Count	Count
[15, 20)	21	25	57	14	9	13
[20, 25)	36	64	157	54	32	48
[25, 30)	20	42	181	41	20	34
[30, 35)	36	46	219	117	46	118
[35, 40)	16	15	169	147	44	148
[40, 45)	15	3	113	61	21	114
[45, 50)	24	1	108	12	8	46
[50, 55)	42	5	129	24		32
[55, 60)	63	15	48	8	3	38

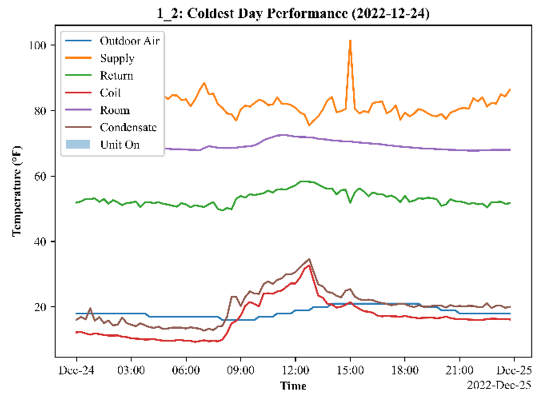
Table A-2. Average Room Temperatures

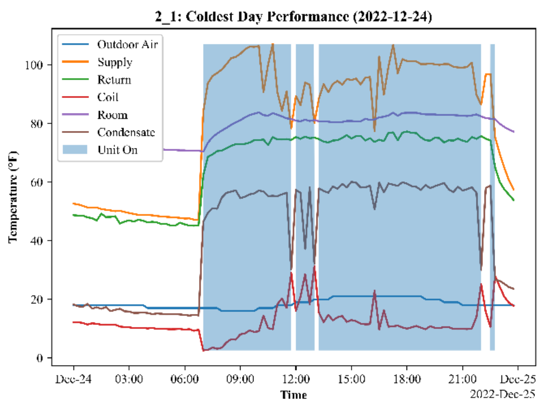
Unit	OAT (°F)	Room Temperature (°F)			Unit	OAT (°F)	Room Temperature (°F)		
		Max	Mean	Min			Max	Mean	Min
	[10, 15)	71	71	70		[10, 15)	72	71	70
	[15, 20)	80	75	69		[15, 20)	81	74	65
3_1	[20, 25)	80	76	70	**4_1**	[20, 25)	83	73	66
	[25, 30)	79	74	64		[25, 30)	80	73	62
	[30, 35)	81	74	65		[30, 35)	82	73	63
	[35, 40)	80	73	65		[35, 40)	87	74	63
	[10, 15)	72	71	70		[10, 15)	74	72	71
	[15, 20)	78	74	70		[15, 20)	82	75	66
3_2	[20, 25)	78	75	69	**4_2**	[20, 25)	81	74	67
	[25, 30)	77	74	64		[25, 30)	81	74	63
	[30, 35)	80	73	65		[30, 35)	80	74	64
	[35, 40)	84	73	66		[35, 40)	83	75	64
	[10, 15)	72	71	70		[10, 15)	73	71	70
	[15, 20)	77	73	69		[15, 20)	83	75	65
	[20, 25)	77	74	67		[20, 25)	86	73	66
3_3	[25, 30)	76	73	63	**4_3**	[25, 30)	86	74	62
	[30, 35)	77	73	65		[30, 35)	85	74	63
	[35, 40)	80	73	65		[35, 40)	85	74	63

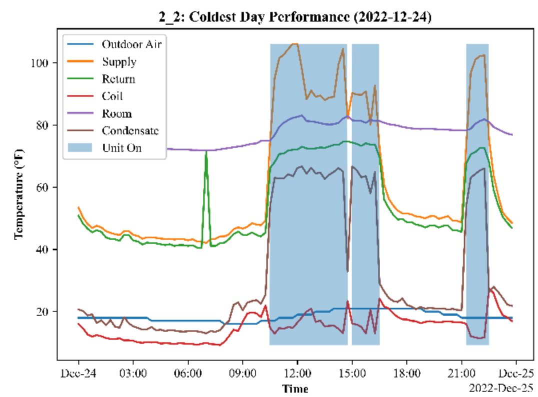
A.3 Daily Performance Trends

Figure A-4. Coldest Day Performance Unit Trends









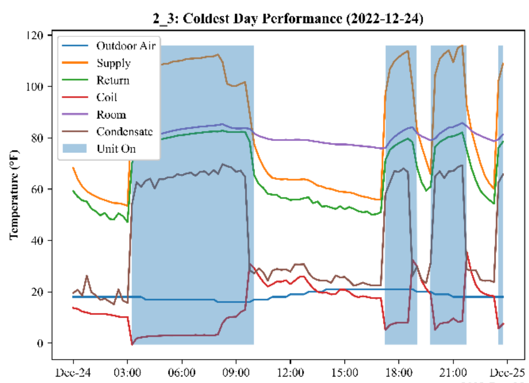
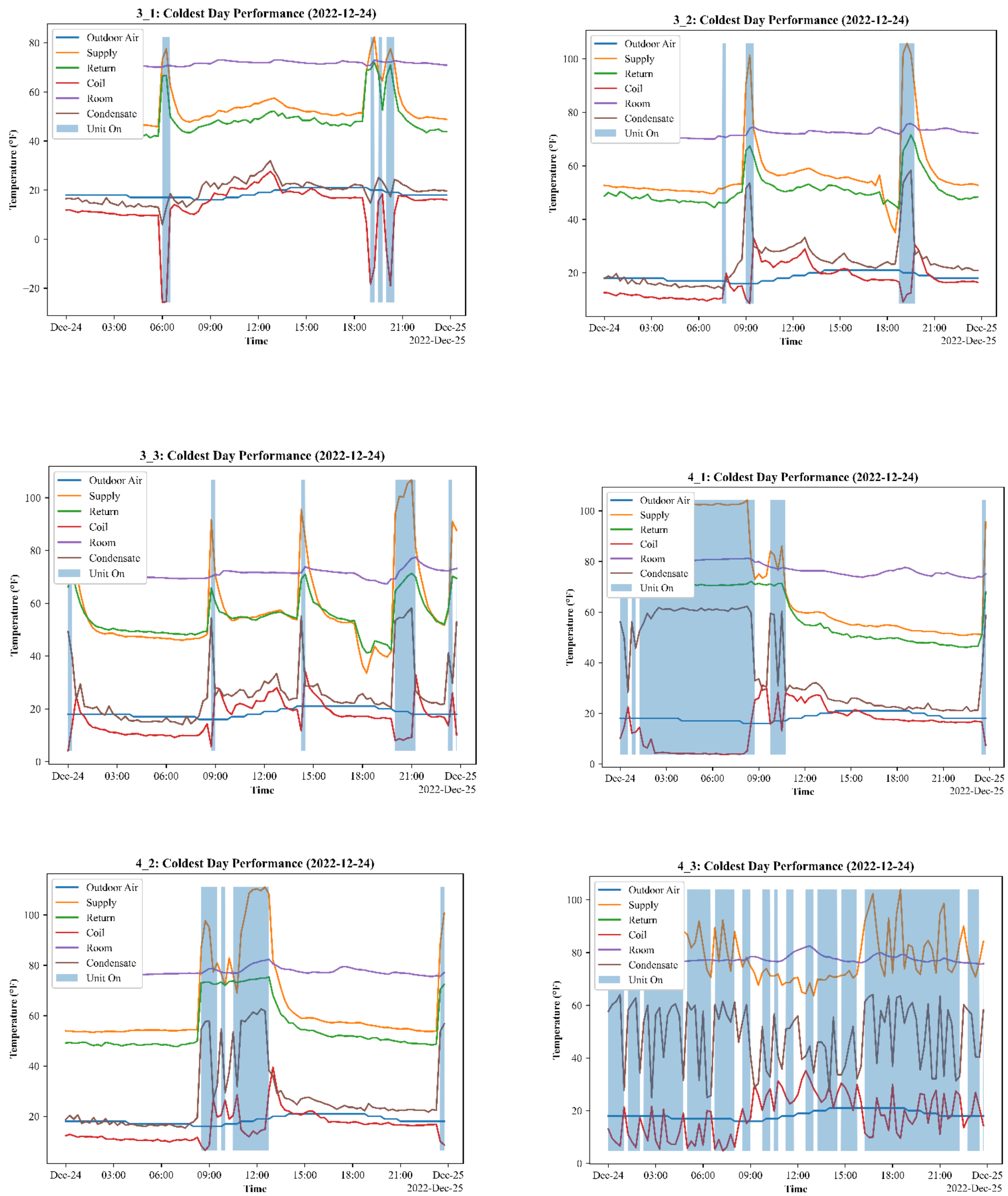


Figure A-5. Coldest Day Performance Unit Trends



Appendix B. Ice Air Packaged Terminal Heat Pump RSXC Product Sheet

PTHP

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World Class Comfort®

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Packaged Terminal Air Conditioner for Replacement

RSXC Series

Ice Air's breakthrough cold climate technology is a paradigm shift. It allows Packaged Terminal Heat Pumps (PTHPs) to efficiently provide space heating down to -5°F and below. And our advanced Variable Refrigerant Flow (VRF) technology ensures that your unit is pinpointing the exact amount of heating or cooling required for the desired room conditions. Efficient, sustainable, heat pumps designed for cold climates are finally a reality.

Defining Cold Climate

- Heating performance laboratory tested and certified to -5°F
- The theoretical lower limit for heating operation is -25°F ambient
- Provides cooling operation down to 38°F

What You Would Expect

- Industry leading efficiency
- Sustainable R-410a Refrigerant
- Fits within a standard size wall sleeve (42" W x 16")

AHRI CERTIFIED®
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Globally Recognized. Industry Respected.

Series Model #	RSXC09		RSXC13		RSXC18
Cooling Capacity (Btu/hr)[1]	9,200		12,500		16,300
Sensible Capacity (Btu/hr)[1]	7,200		8,800		11,200
Cooling Capacity Range (Btu/hr)	6,300 - 11,800		6,500 - 14,900		7,300 - 18,000
EER[1]	12.1		11.1		10.0
Heating Capacity (Btu/hr)[2]	10,200		12,000		17,300
Heating Capacity Range (Btu/hr)	5,200 - 12,600		5,600 - 14,200		9,500 - 18,700
COP[2]	3.6		3.1		3.0
HSPF[2]	9.6		9.5		9.0
Voltage	115	208	115	208	208
Electric Heater (kW)[3]	1.5	3.0 | 3.5	1.5	3.0 | 3.5 | 4.3	3.0 | 3.5 | 4.3
Electric Heater (A)[3]	13	14.4 | 16.8	13	14.4 | 16.8 | 20.7	14.4 | 16.8 | 20.7
Current in Cooling Operation (Amps)	6.6	3.7	9.8	5.4	7.8
Power in Cooling Operation (Watts)	760		1,126		1,630
Current in Heating Operation (Amps)	7.2	4.0	9.9	5.5	8.1
Power in Heating Operation (Watts)	830		1,134		1,690
MCA (without Electric Heat)	11.7	5.9	12.6	8.5	10.4
MOCP (without Electric Heat)	20	15	20	15	15
MCA (with Electric Heat)	17.0	18.4 | 21.5	17.0	18.4 | 21.5 | 26.4	18.4 | 21.5 | 26.4
MOCP (with Electric Heat)	20	20 | 25	20	20 | 25 | 30	20 | 25 | 30
Evaporator Motor Nominal HP	1/25	1/25	1/25	1/25	1/25
Airflow (CFM)	380		400		480
Outside Air (CFM)	60		60		60
Weights (lbs.)	127		134		151
	Low Ambient Performance				
Heating Capacity @ 10°F	6,600		7,700		11,600
COP @ 10°F	2.2		2.14		2.02
Heating Capacity @ 5°F	6,100		6,900		10,600
COP @ 5°F	1.98		1.91		1.93
Heating Capacity @ -5°F	5,500		6,400		8,100
COP @ -5°F	1.74		1.62		1.6

SPECIFICATION NOTES:

1. Rated performances in cooling mode @80°F/67°F DB/WB Indoors and 95°F/75°F DB/WB Ambient
2. Rated performances in heating mode @ 70°F/60°F DB/WB Indoors and 47°F/43°F DB/WB Ambient
3. If the electric heat option is selected, the heat pump operation is disabled and electric heat enabled below -5°F (+/- 3 °F).
4. Units without electric heat will operate below -5°F with derated performance. Performance below -5°F has not been certified.
5. Electric heat is recommended in markets that may experience ambient temperatures below -5°F.

Due to Ice Air's ongoing product development programs, the information in this document is subject to change without notice.

ICE6050.RSXC.1 10/22

Endnotes

1 According to Taitem Engineering, the most common outdoor temperature at which PTHPs switch to electric resistance heat is at 40°F (NYSERDA 2018, p. 16).

2 OATs did not drop below 15°F throughout the data collection period.

3 Temperature-binned and seasonal COP are defined in section 2.4.3.

4 Inadvertently, building staff or the installing contractor did not isolate the first and second floor units from the central hot water heating plant and were therefore excluded from heating calculations. All units were included in cooling season calculations.

5 NYSERDA 2018, p. ES-1.

6 NEEP 2023a, p. 2.

7 NEEP 2023b, p. 2.

8 At typical conditions close to sea level:

ρ_{air} is the density of air: $0.075\ lb/ft^3$

$C_{p_{air}}$ is the specific heat of air: $\frac{0.24\frac{BTU}{lb}}{°F}$

9 We used a power factor of 0.99 in our calculations based on testing conducted by Ice Air (B. Liu, personal communication, March 3, 2023).

10 Equation used in basic calculations. Method used for overall analysis is explained in section 2.4.3.

11 Errors in temperature and current sensors were anticipated to be normally distributed around zero error, as these would be calibration errors in the sensors out of the box.

12 Error in airflow is assumed to be uniformly distributed between the minimum and maximum measurements of each heat pump size.

13 NYSERDA 2018, p. 20.

14 The monitored heat output of the PTHPs in apartment 4 was summed and converted to heating plant input assuming 80% system efficiency and 100,000 btu/therm.

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