Next Gen Rooftop Units

PERFORMANCE REPORT: ROOM & BOARD HEAT PUMP ROOFTOP UNIT

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EXECUTIVE SUMMARY

Overview

Founded in 1980 and 100% employee-owned, Room & Board is a Minneapolis-based furniture company with a long-standing commitment to sustainability. The company is a **Department of Energy (DOE) Better Climate Challenge partner** and has set ambitious goals to reduce Scope 1 and Scope 2 greenhouse gas (GHG) emissions by 50% from 2019 levels. In 2024, they installed a solar array that now generates 120% of their corporate headquarters' electricity, according to their 2024 impact report.

As part of their ongoing decarbonization efforts, Room & Board replaced an aging 5-ton gas furnace and AC RTU that served the on-site gym with a 5-ton all-electric heat pump RTU with a resistance heater for auxiliary backup. The building is a single-story facility with 235,000 square feet of combined office and warehouse space. It is primarily heated and cooled by multiple gas and AC RTUs which are replaced on a regular 15-year schedule.

During the winter of 2024–2025, Room & Board partnered with Next Gen RTUs to evaluate the performance and efficiency of a heat pump rooftop unit (RTU) at the company's headquarters in Golden Valley, Minnesota.

To better understand the system's performance and potential energy savings, Room & Board partnered with Next Gen RTUs, a statewide initiative under Efficient Technology Accelerator. Next Gen RTUs strives to advance the performance of rooftop units to meet the growing demand for energy efficient and sustainable building solutions. The program strives to deliver energy efficiency and emission reduction goals where the market needs support, driving long-term sustainability in commercial buildings.

Through this partnership, Next Gen RTUs and Room & Board aimed to:

1. Demonstrate that heat pump RTUs can perform in Minnesota's cold climate without compromising comfort.
2. Showcase energy savings and operational benefits in real examples.

This report summarizes the findings from the monitored unit, highlighting energy and unit performance throughout the period studied.

Approach

To monitor the RTU, electrical power input was measured using one meter, while air temperatures and system actuator currents were tracked with a separate data logger, known as The Energy Conservatory (TEC) True Flow Grids. This approach allowed CEE to assess efficiency and energy consumption and is consistent with other studies performed by CEE. Detailed monitoring and analysis approaches are described in the following sections. CEE will collect qualitative and quantitative data and produce public case studies to disseminate benefits, occupant experiences, and energy savings results to market actors and building owners.

Results

This project demonstrates the potential of using all-electric heat pump RTUs in cold climate commercial buildings. The system consistently delivered heating through colder months and maintained occupant comfort. Performance was more efficient than a standard RTU with a gas furnace.

This study highlights the importance of maintaining sufficient airflow in heat pump RTU systems, as well as the importance of designing setpoints, ventilation, and occupancy control schedules to minimize electric auxiliary heat use as a backup source.

While the heat pump was able to provide sufficient heating even in cold weather, its efficiency was limited by poor airflow and excessive use of auxiliary heating. Early in the project, airflow issues were identified and improvements made, but it remained unusually low until the return grille was fully cleaned.

Once airflow was corrected, the monitored heat pump with electric backup used less total energy than all three modeled alternatives: a dual fuel heat pump, electric resistance heat, and a conventional gas furnace. While energy use decreased, energy costs and emissions increased. The all-electric heat pump RTU saved an estimated 40% of energy use, compared to a standard gas furnace RTU. Due to high electricity demand and the cost difference between electricity and natural gas, utility costs did not decrease at the same rate as energy savings. The system produced approximately 24% higher emissions than dual fuel alternatives during February and March due to grid emissions and a dependence on auxiliary heat.

Excessive use of auxiliary heat is likely due to setbacks installed to reduce heating when the building is unoccupied. Future studies plan to remove the setbacks to see if energy can be saved by maintaining a consistent temperature. Without the rapid increase of heat that is needed to meet the occupied heating needs in the morning, auxiliary heat may be significantly reduced.

Lessons Learned:

- **Commercial heat pumps are viable in cold climates:** With proper planning and controls, commercial heat pumps can reliably heat spaces even in cold conditions.
- **Airflow matters:** Low airflow significantly reduced system performance. Maintenance teams should ensure the return filters and upturned vents are properly sized and cleaned.
- **Auxiliary heat can decrease efficiency:** Controls that limit electric resistance use, such as reduced setbacks, are essential to both cost and energy savings. High demand charges may reduce the cost benefits of electric-only systems depending on rates.

METHODOLOGY

The following describes the methods used to monitor and verify unit performance and energy use. Power measurements were used to determine unit energy use. Data loggers were used to determine unit performance, which allowed us to determine energy use by fan, heat pump, and electric resistance heat. Both the power meter and data loggers continuously send data to the Next Gen RTU team via a cellular modem. Weather data was used to understand how the RTU operates at varying temperatures, as heat pump performance decreases with lower temperatures. Airflow was measured on site then correlated to power. Airflow helped determine maintenance barriers to unit performance.

Power Measurements

A power meter (eGauge 4105) was installed at the RTU to measure the power on each of the three phases feeding the RTU. The eGauge devices were connected to the internet via a cellular modem and continuously uploaded data to the eGauge server. Data at one-second intervals was retrieved daily from the server. The system's total electrical power input was calculated as the sum of the power measured on each phase.

RTU Operating Parameters

An independent data logger (Campbell Scientific CR3000) was installed to measure temperatures and actuator currents. Temperatures were measured with field-installed thermocouples and thermocouple arrays, while heat pump RTU actuator outputs were measured with current transducers (CTs).

Table 1. Datalogger fields and definitions

Field Name	Definition	Units
datetime	Timestamp	-
rec_nbr	Unique ID	-
sat	Supply Air Temperature (2 sensors)	°F
rat	Return Air Temperature	°F
mat	Mixed Air Temperature (6 sensors)	°F
cot	Coil Outlet Temperature (2 sensors)	°F
supfan	Supply Fan Current	mV
auxheat	Auxiliary Heat Status (2 sensors)	mV
rvalve	Reversing Valve Status	mV
cmp	Compressor Status	mV

Data at one-second intervals was transmitted daily from the logger to the storage server. Data was retrieved from the storage server on a weekly basis for processing and analysis.

Weather Data

Weather data from the NOAA NCEI local climatological data table for Minneapolis-St. Paul International Airport was downloaded on a weekly basis. Observations of outdoor temperature and relative humidity were interpolated to a one-second resolution before combining them with measurements from the data-logging equipment.

Airflow Measurements

Airflow was characterized through a one-time measurement and correlation approach. The initial measurements were made by inserting TEC Digital TrueFlow grids into the RTU filter slot and measuring the total airflow and supply fan current at various fan speeds. A correlation curve consisting of pairs of airflow and supply fan current measurements was used to calculate airflow based on the supply fan current, which was collected on an ongoing basis. These initial airflow measurements were much lower than expected, but the second possible measurement location of the RTU return grille was inaccessible on the day of the initial testing. A subsequent visit to measure airflow at the return on January 29, 2025, revealed that flow was severely restricted due to dust accumulation on the upward-facing return grille. Airflow was measured at the return grille before and after removing the dust. The measured airflow changed enough after cleaning that the performance analysis was divided into periods of operation before and after cleaning in some cases.

ANALYSIS

After monitoring the 2024–2025 heating season, the Next Gen RTU team analyzed the data. The data was cleaned of out-of-range sensor values before operation modes were determined. Performance metrics were established by determining operating modes. Energy, cost, and emissions savings were also calculated.

Data Preparation

The process for analyzing the data consisted of first combining the RTU data, power meter data, and weather data into a single table for each RTU. Then, out-of-range sensor values were removed and replaced with interpolated values. Values reported by sensors measuring the same physical value were averaged.

Operating Mode Determination

The first step in developing labels for different operating modes from the independent datalogger was to compare actuator current measurements with thresholds inferred from the operating data. Values were compared to the thresholds to determine whether each actuator was on (1) or off (0) at each observation. The system's operating mode was inferred from the combination of on and off statuses. In the case of defrost, the actuator statuses are similar to those of heat pump cooling, so the ambient temperature was compared to a threshold of 45°F to classify operation as defrost or cooling as an additional check.

Table 2. Operating mode mapping

Mode	Supply Fan	Compressor	Reversing Valve	Aux Heat	Ambient Temperature (°F)
Off	0	-	-	-	-
Fan Only	1	0	-	-	-
Heat Pump Heat	1	1	0	0	-
Aux. Heat	1	0	0	1	
HP + Aux. Heat	1	1	0	1	
Defrost	1	1	1	1	< 45
Heat Pump Cool	1	1	1	0	≥ 45

Performance Metrics

RTU performance metrics were calculated for each observation. Tracked metrics included supply volumetric airflow, temperature difference across the indoor coil, and temperature difference across the auxiliary heat section. These data features were combined with air properties to calculate heat pump sensible capacity, auxiliary heating capacity, and net sensible heating capacity. Electric power input from the separate electric power meter was the last piece of information required to calculate the system COP. The temperature difference across the heat pump indoor coil was the difference between the coil outlet temperature (COT) and the average mixed air temperature (MAT). The temperature difference across the auxiliary heat section was the difference between the supply air temperature (SAT) and the COT. Note that this formulation includes heat generated by the supply fan in the auxiliary heating calculation. Heating capacity was the product of the volumetric airflow, physical constants, unit conversion factors, and the appropriate temperature difference.

Analysis Interval

The analysis focused on performance in heating season. Heating season was defined as operation beginning the first day of monitoring (November 14, 2024) until the last day before the first instance of heat pump cooling (April 22, 2025). The return grille was cleaned on January 29, 2025. Performance before cleaning was subject to severely restricted airflow, so some metrics are calculated separately for the intervals before and after cleaning.

Energy Savings Calculations

The heating season site energy savings were calculated as the difference between the actual energy input and the energy input that would be required by a hypothetical RTU delivering the same net heating capacity with a gas furnace or electric resistance only. The savings were normalized by the hypothetical energy input to estimate percent savings.

Cost Savings Calculations

Cost estimates for the baseline and alternative equipment types were based on volumetric pricing for electricity and gas, as well as demand charges for electricity. The electric demand for each alternative was estimated as the average demand during a 15-minute period assumed to be the period of peak demand for the facility. The facility's 15-minute peak period was assumed to be the first 15 minutes of the morning warm-up period on the coldest day of each month.

Emissions Savings Calculations

The heating season source emissions savings were calculated as the difference between the estimated source emissions and the emissions from a hypothetical dual fuel RTU delivering the same net heating capacity using a heat pump and gas furnace. The savings were normalized by hypothetical emissions to estimate percent savings. Emissions associated with electricity use were estimated based on electrical energy input multiplied by the total output emissions factor for the eGRID subregion MROW of 0.420 kg CO_2e/kWh. Emissions associated with gas use were estimated based on the estimated gas use multiplied by an emissions factor of 53.11 kg CO_2e/mmBtu or 0.181 kg CO2e/kWh (U.S. EPA 2025). The hypothetical gas input was estimated based on the measured supplemental heating capacity divided by a typical furnace efficiency of 80%.

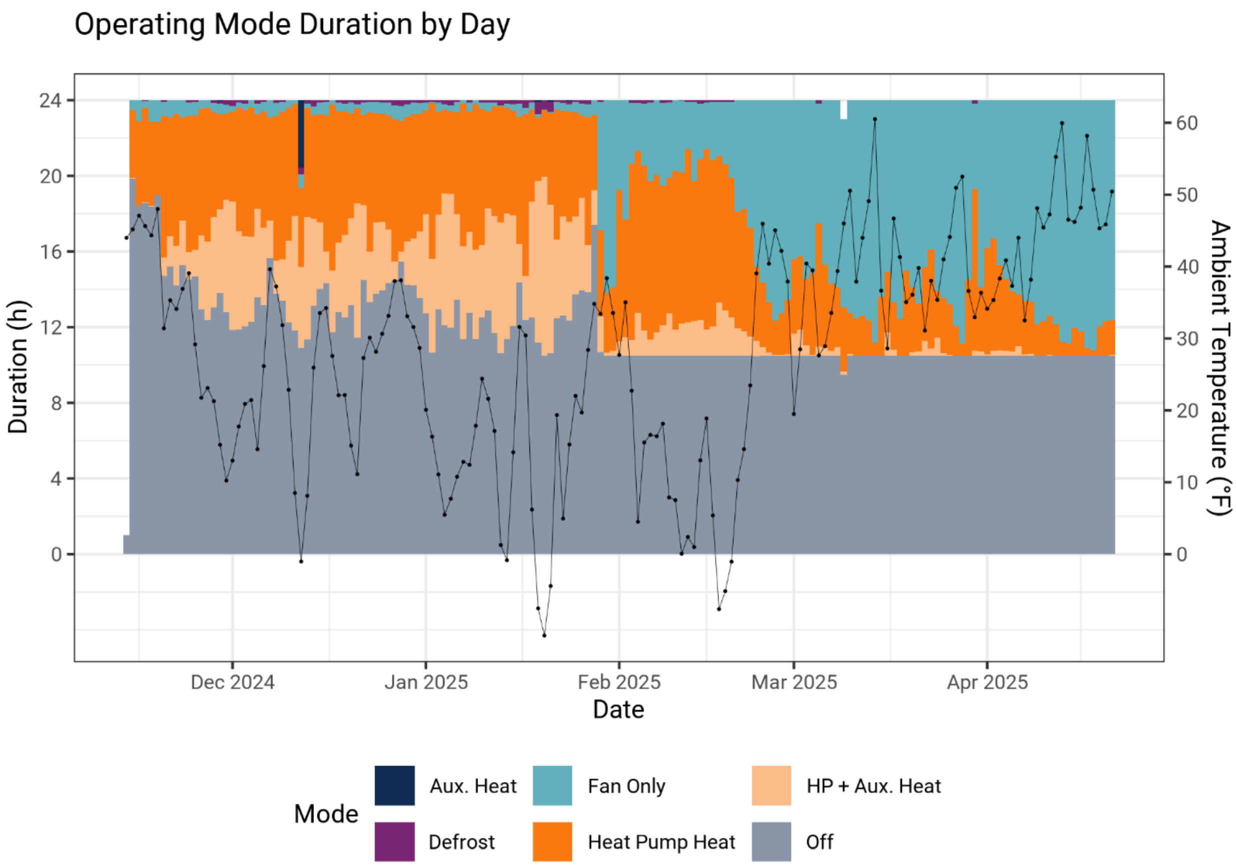
RESULTS

Plots from monitoring data describing operating modes, power input, delivered heating energy, and efficiency were created to characterize system performance. To determine energy and emissions savings relative to alternative equipment types, actual energy inputs were compared to the hypothetical energy inputs required to deliver the same heating capacity through alternative equipment.

Operating Modes

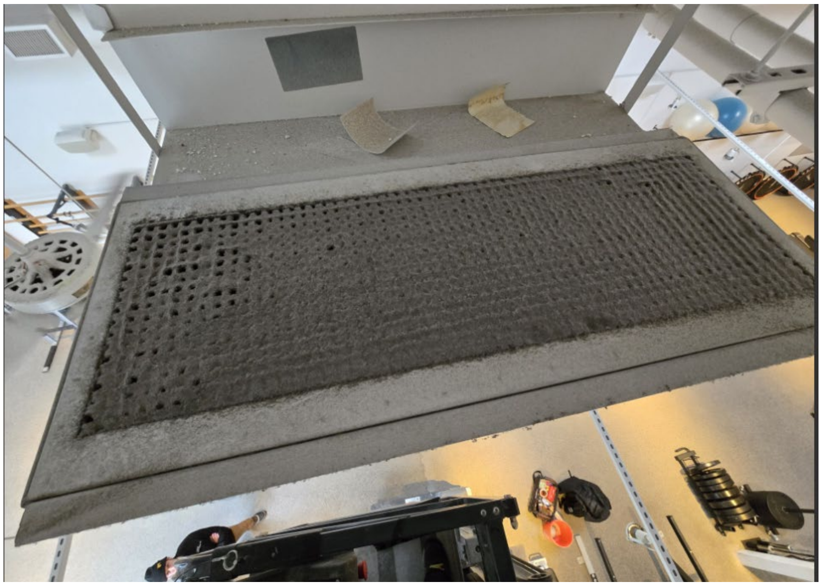
Operating modes were first described by a stacked bar chart with a bar corresponding to each date in the sample, as shown in Figure 1. The total height of the bar corresponded to the total duration of the operation observed (typically 24 hours). The color of the bar indicates the operating mode. The daily average outdoor air temperature trend overlaid on the plot shows the effect of weather on operating mode, with higher shares of heating modes when the temperature is farther away from the zone balance point temperature.

Figure 1. Operating Mode Duration by Day



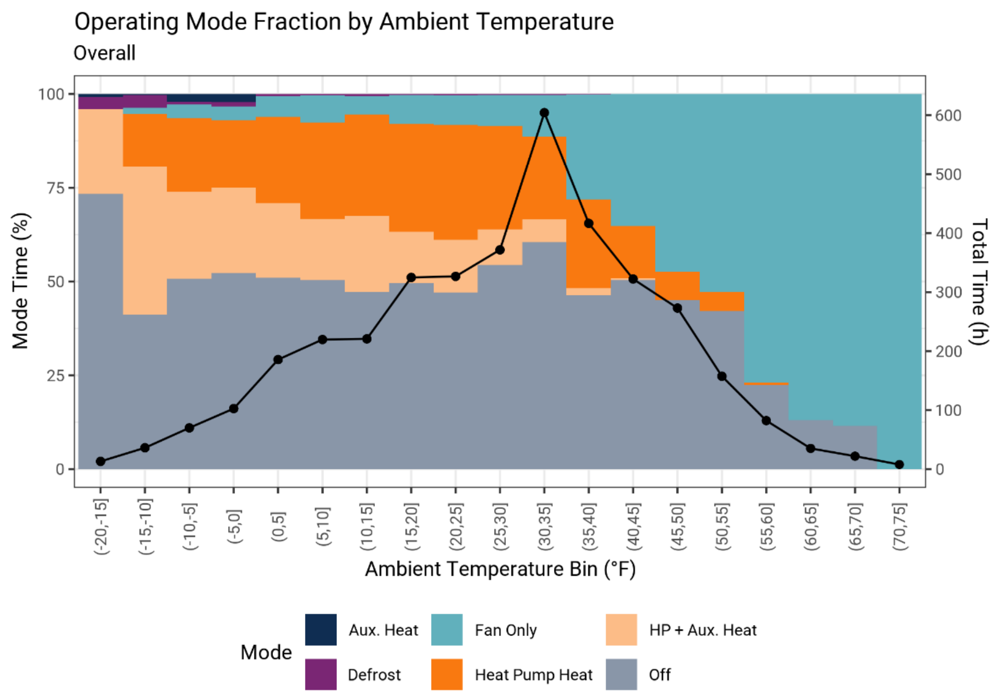
A distinct change to the mix of operating modes is evident following the cleaning of the return grille on January 29, 2025. This grille, shown in Figure 2, is turned up towards the ceiling as an aesthetic design choice, and collects dust on the surface. It was not cleaned at the time of install of the new RTU. Prior to cleaning, auxiliary heat use (with or without the heat pump) and defrosts were more common and fan only operation was rare. This mix of modes implies that the heat pump was struggling to meet setpoint due to impaired air distribution. Another feature of the pre-cleaning data is the irregular off time per day, which could indicate a change in thermostat settings (from "Auto" to "On") or fault conditions such as high supply air temperature that required the system to shut down. After cleaning, the system is off for a consistent ten hours per day, auxiliary heating use is significantly reduced, and the amount of time in heating modes seems to vary more consistently with outdoor temperatures. All these features point to corrected air distribution issues.

Figure 2. RTU Return Grille Before Cleaning



Operating modes were also described by a stacked bar chart showing the fraction of operating mode time by ambient temperature bin, as shown in Figure 3. The line overlaid on these plots indicates the total time spent in each ambient temperature bin during the study. These plots indicate the outdoor temperatures where the RTU transitions between modes in this application.

Figure 3. Overall Operating Mode Fraction by Ambient Temperature



The overall plot shows generally expected trends, with large percentages of fan-only operation at warmer temperatures giving way to more heat pump operation at mild temperatures, then increasing shares of auxiliary heating at colder temperatures. The larger share of defrost at the coldest temperatures is somewhat unexpected, as previous research has shown defrost to be more common in the 25°F–35°F range where the amount of moisture in the outdoor air is higher. This may be because this equipment used different control algorithms for initiating defrost than equipment monitored previously.

While the overall operating mode fraction plot is useful for general performance characterization, the impact of removing airflow obstructions is highlighted by comparing the operating mode fractions by temperature before and after cleaning. Before cleaning, fan only operation accounted for less than 6% of the operating time in even the warmest bins, as shown in Figure 4. After cleaning, fan only operation made up more than 40% of the operation down to the (35°F, 40°F] bin, indicating that the system was regularly satisfying the setpoint at conditions it was previously could not, as shown in Figure 5. Auxiliary heat use was significantly higher before cleaning as well, with the heat pump requiring auxiliary heat most of the time the heat pump was active up to the (10°F, 15°F] bin. After cleaning, the heat pump operated without auxiliary heat most of the time it was active down to the (-10°F, -5°F] bin.

Figure 4. Operating Mode Fraction by Ambient Temperature before Cleaning

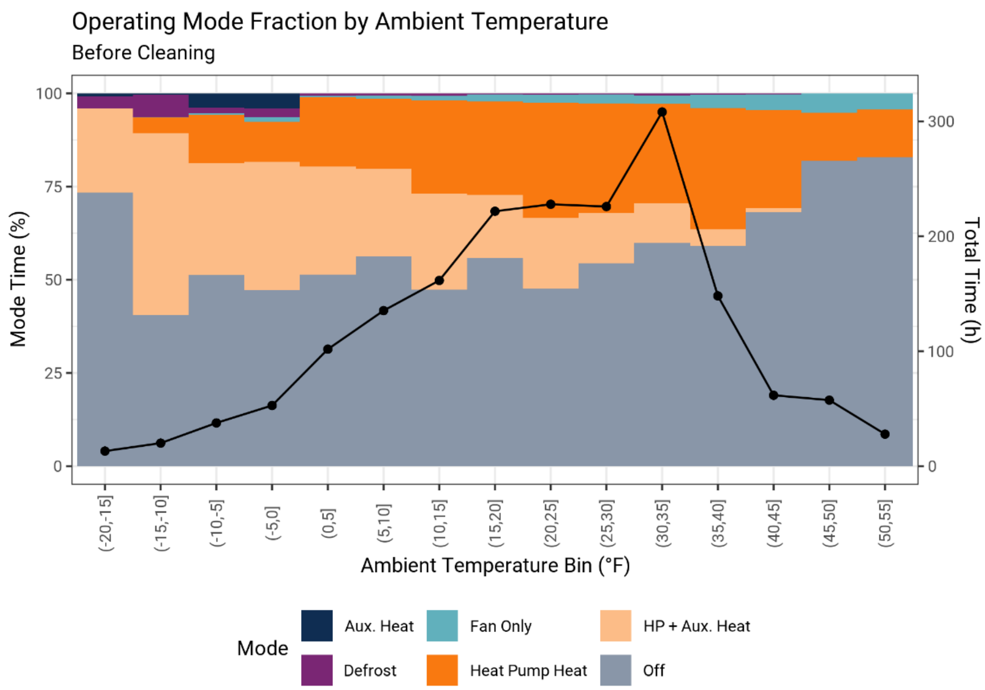
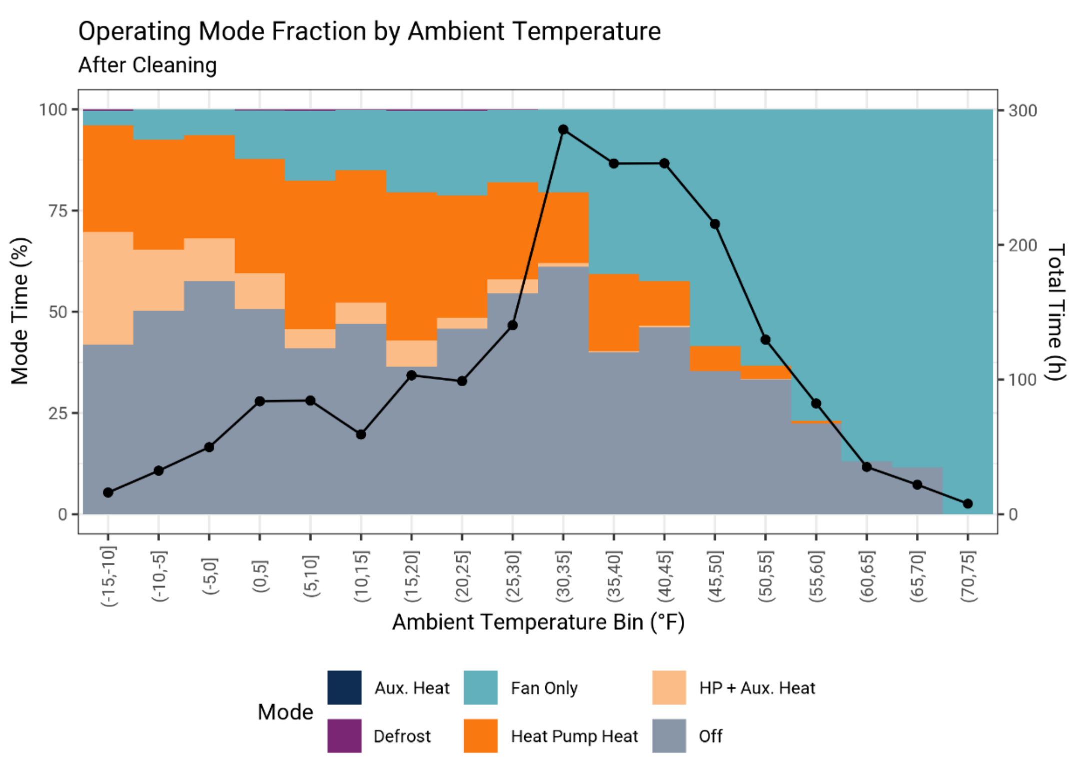
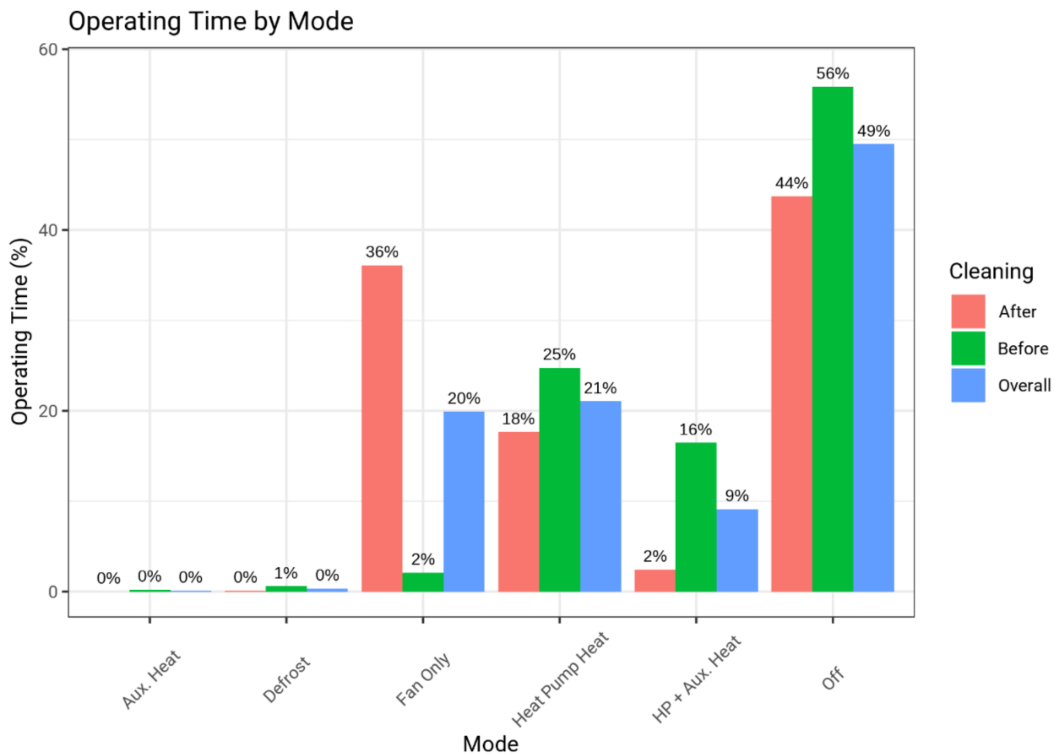


Figure 5. Operating Mode Fraction by Ambient Temperature after Cleaning



Overall mode fractions for the monitoring period and before and after cleaning are shown in Figure 6, highlighting the stark increase in fan-only operation and reduction in HP + Aux. Heat operation after cleaning.

Figure 6. Operating Time by Mode, Overall and Before/After Cleaning



Energy Output

The delivered energy plot in Figure 7 shows another time series view with a bar corresponding to each date. The height of the bar corresponds to the sensible heating energy delivered, while the color represents the operating mode. This plot also includes a line plot overlay that shows the daily average temperature on the right y-axis. Higher outdoor air temperatures are coincident with less delivered energy and vice versa. Here again, the impact of removing airflow obstructions on January 29, 2025, is evident. After cleaning, the delivered capacity per day was reduced because, with better distribution, the zone could reach setpoint and the system could transition to fan-only operation. The amount and share of capacity provided by the auxiliary heating section was reduced, and the heat pump became the primary heating source as intended.

Figure 7. Delivered Energy: Sensible Heat Transfer vs. Time by Mode

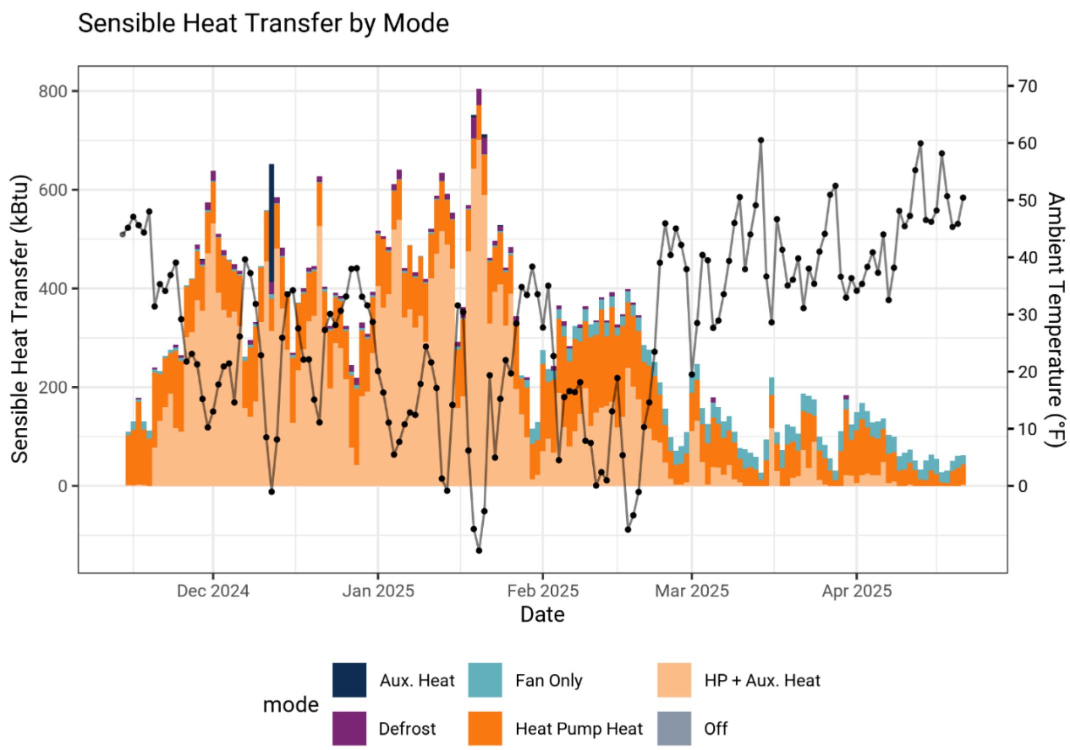
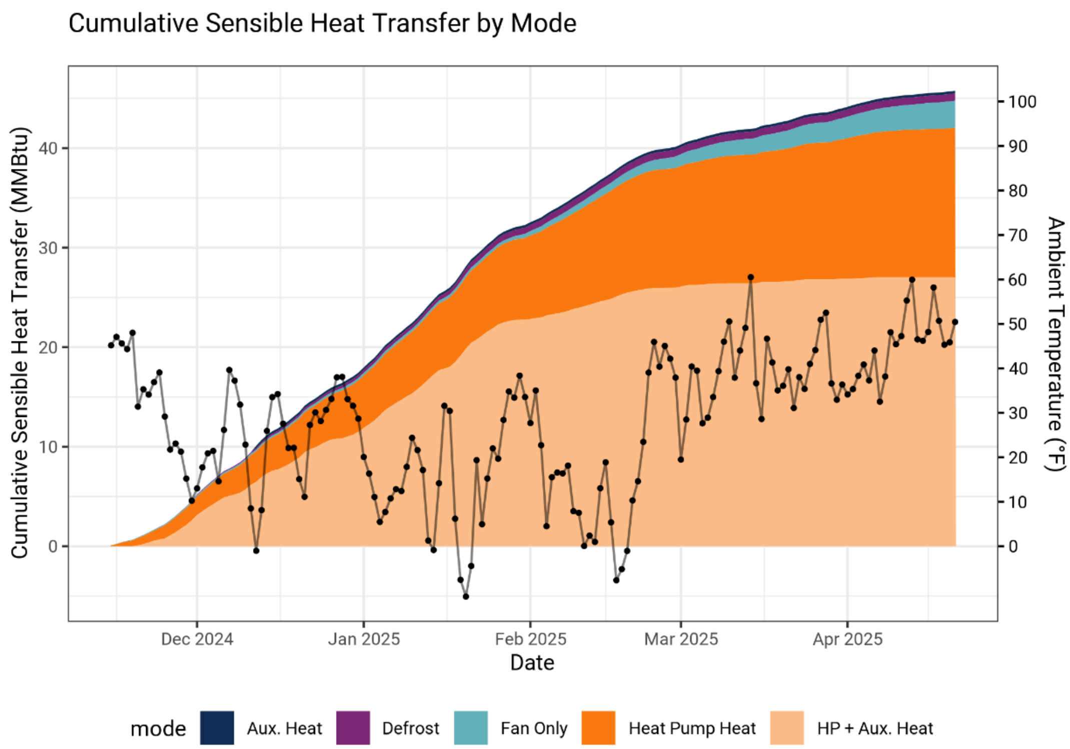


Figure 8 shows the cumulative view of the same information. It is easier to see how HP + Aux. Heat was responsible for most of the delivered energy up to the date of the cleaning, but the share of the heat pump heating capacity grew following the cleaning.

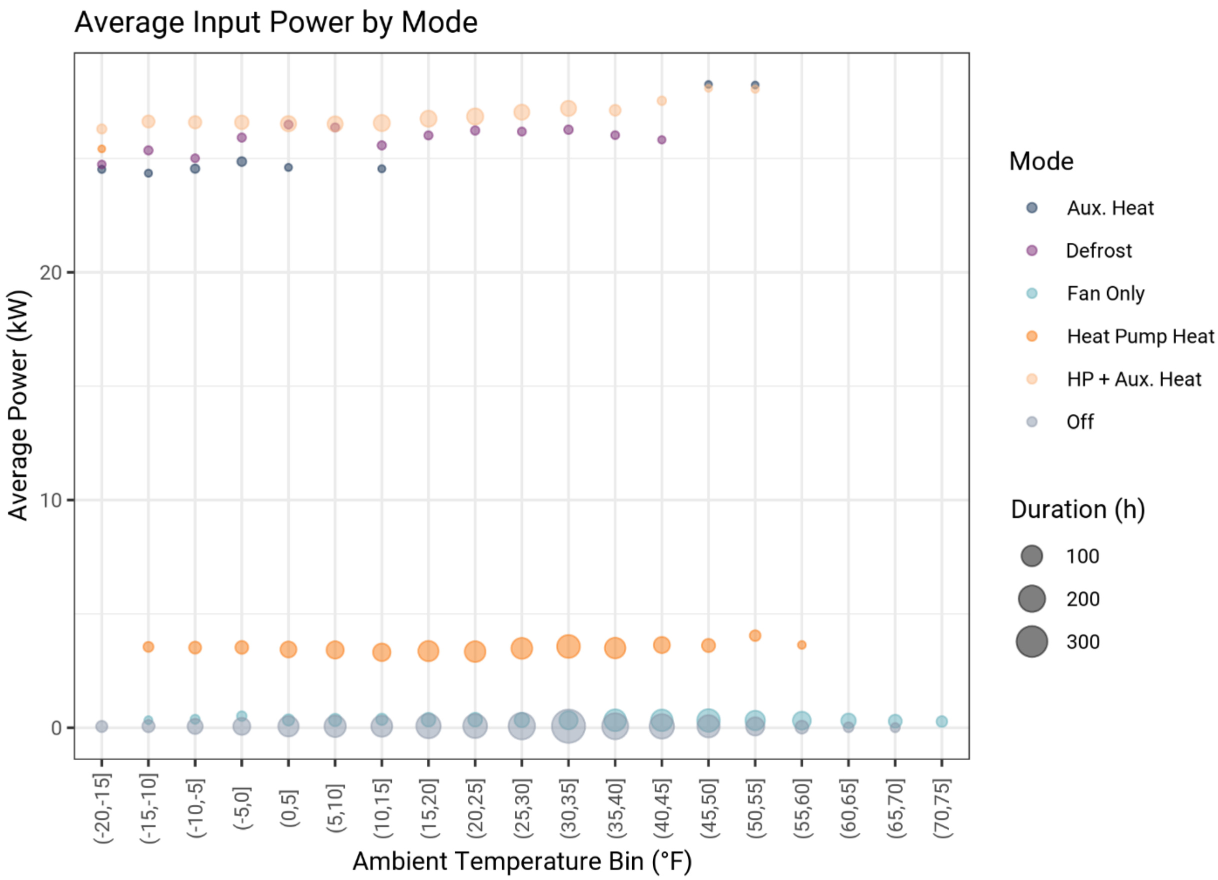
Figure 8. Cumulative Delivered Energy vs. Time by Mode



Power Input

Electric power input was characterized by plotting the average power by mode and ambient temperature bin in Figure 9. The size of the point maps to the total amount of operation in the corresponding conditions. As expected, the average power of the off mode is near zero, and the average power in fan-only mode corresponds to the power draw of the supply fan of about 300 W. Heat pump heating mode, which includes the supply fan, averaged around 3.3 kW. The system averaged over 25 kW average power consumption in auxiliary heat mode, which corresponds to the power rating of the installed resistance heating element. Since the system was configured to operate the heat pump and auxiliary heating simultaneously, it was surprising that the selected auxiliary heating element was large enough to provide the entire heating capacity of the system. This selection helps the system continue to function if there are failures related to the heat pump, but it could also mask issues with the heat pump system and allow the system to operate at a low efficiency indefinitely, resulting in unexpectedly high energy and power consumption and cost.

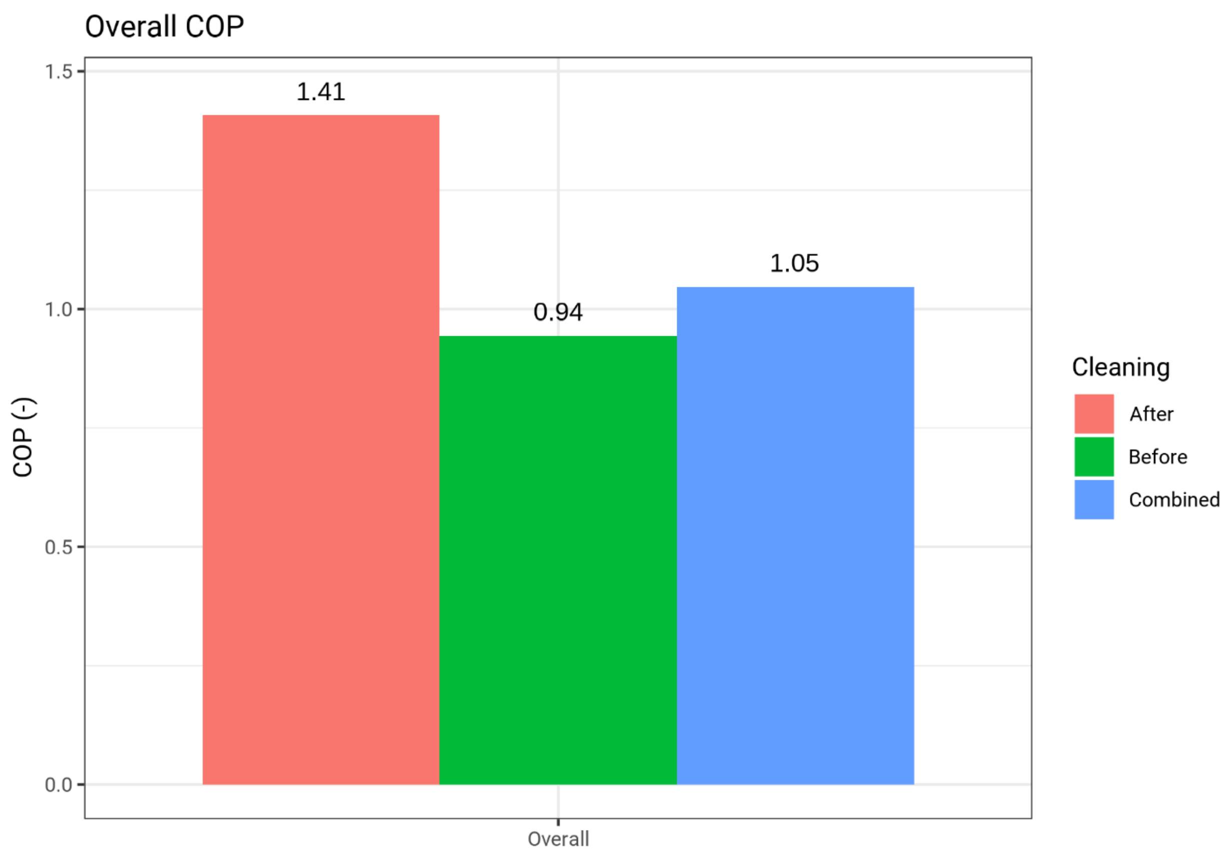
Figure 9. Average input power by mode



Efficiency

The overall system COP was calculated by dividing the total energy output by the total energy input for the monitored period. As shown in Figure 10, this calculation results in a COP of 1.05 for the entire period, 0.94 before cleaning, and 1.41 after cleaning. When comparing these numbers to the steady state COP of an electric resistance element, which is 1.0, it is important to remember that in practice, a forced-air heating system containing electric resistance heating will have a COP of less than 1.0 due to mode changes introducing transient behavior, heat loss to the surroundings, and the parasitic electrical load of the controls.

Figure 10. Overall system COP during the monitored period



As shown in Figure 11, the system COP by operating mode shows that heat pump mode resulted in a system COP greater than 1.5 even before cleaning. The jump in overall system COP after cleaning is more a result of heat pump mode's increased mode share than improvements within heat pump mode. Defrost mode had the lowest COP as expected because the power input to run the auxiliary heat and the compressor simultaneously was very high, but the heat pump system pulled heat from the indoor air to defrost the outdoor coil, reducing net capacity. There was no auxiliary heat use without the heat pump active after cleaning, but the system COP for auxiliary heating only was around 0.8 due to transient effects, heat loss to the surroundings, and parasitic electrical loads from controls.

Figure 11. Overall system COP by operating mode

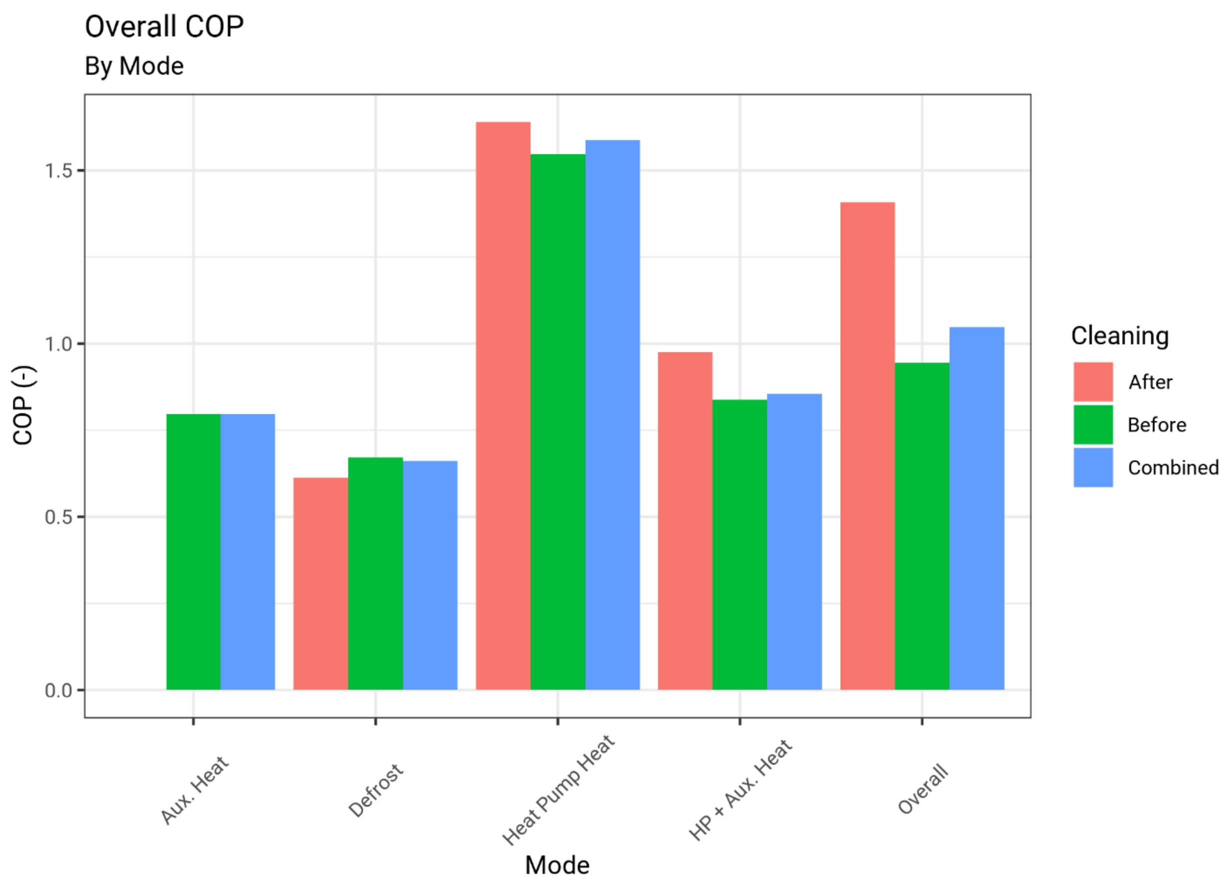
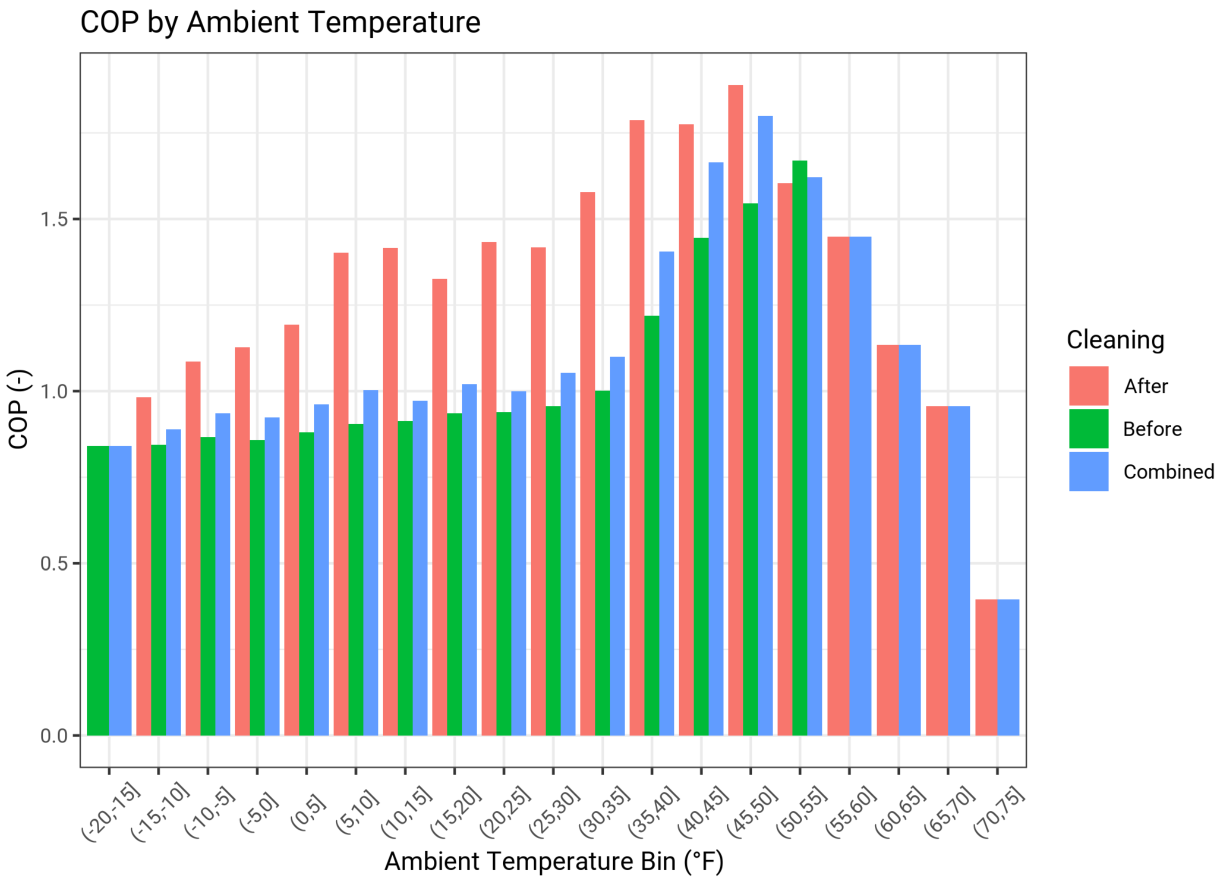


Figure 12 shows the overall COP (across all modes) by outdoor temperature bin throughout the heating season. Again, the increase in COP after cleaning is primarily due to increased use of the heat pump without the auxiliary heat. Lower COPs at warmer temperatures were due to shorter heating cycles and larger shares of fan-only operation.

Figure 12. Overall system COP by Ambient Temperature

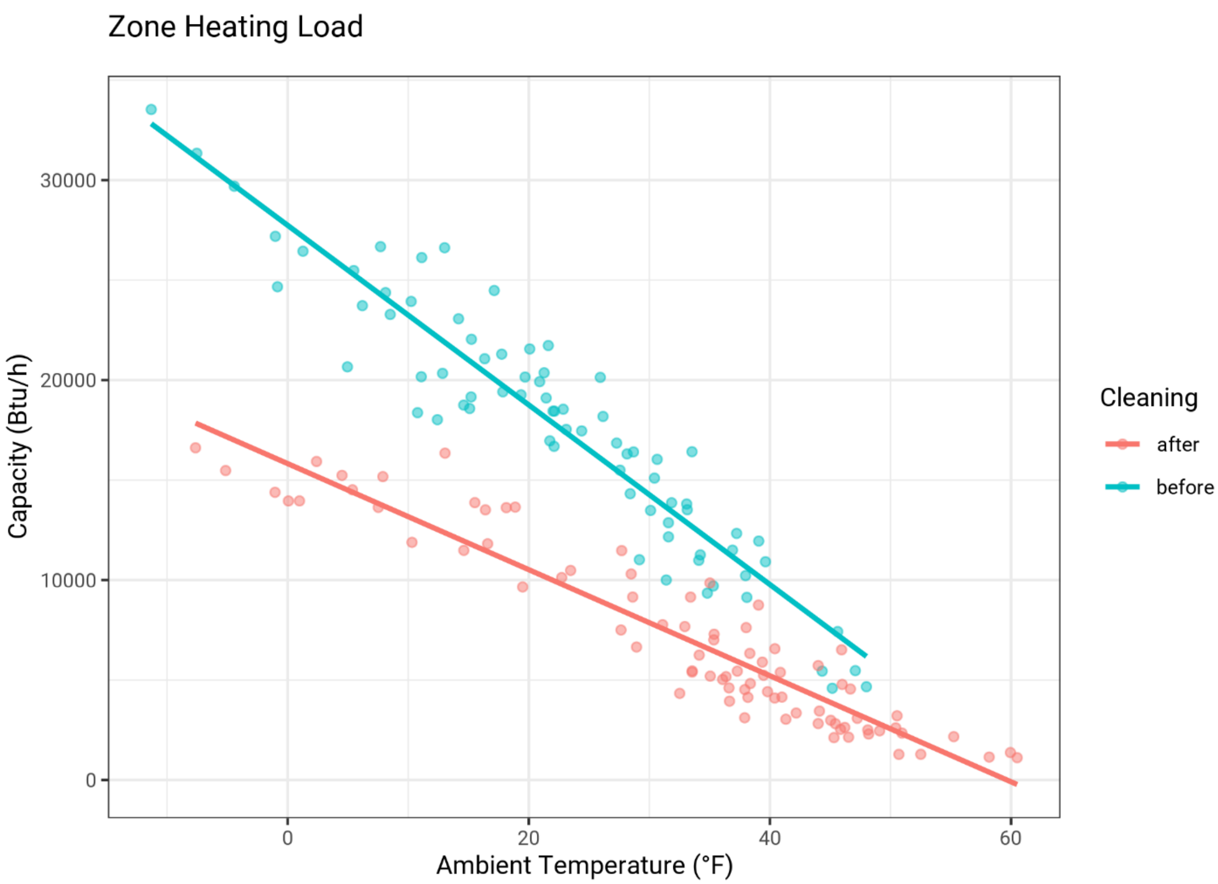


Heating Loads

Calculating the heating energy delivered by the heat pumps facilitates the empirical calculation of the zone heating load if the equipment meets the space setpoint temperature. In this case, the operating mode mix of the system prior to cleaning indicates that setpoint was rarely achieved, so the required capacity data from before cleaning is misleading. The system was transferring more heat to the indoor air at a given outdoor temperature than after cleaning, but that heat was not effectively distributed to the zone, therefore much of it was wasted.

In Figure 13, the daily average heating capacity delivered is shown with respect to the daily average temperature in a scatterplot with a different series for before and after cleaning. This scatterplot shows an inverse correlation during heating season, as the required heating capacity decreases with increasing temperature. Focusing only on the after-heating performance, the empirical zone design heating load was 18,740 Btu/h at a design temperature of -11°F. The measured airflow of around 1,150 CFM is lower than the lowest airflow for which this heat pump has published capacity data, but for reference, the rated heating capacity at -10°F ambient temperature, 70°F return air temperature, and 1,500 CFM was 18,300 Btu/h. This number represents the integrated capacity, or the long-term capacity including necessary defrosts.

Figure 13. Zone Heating Load by Temperature Before and After Cleaning



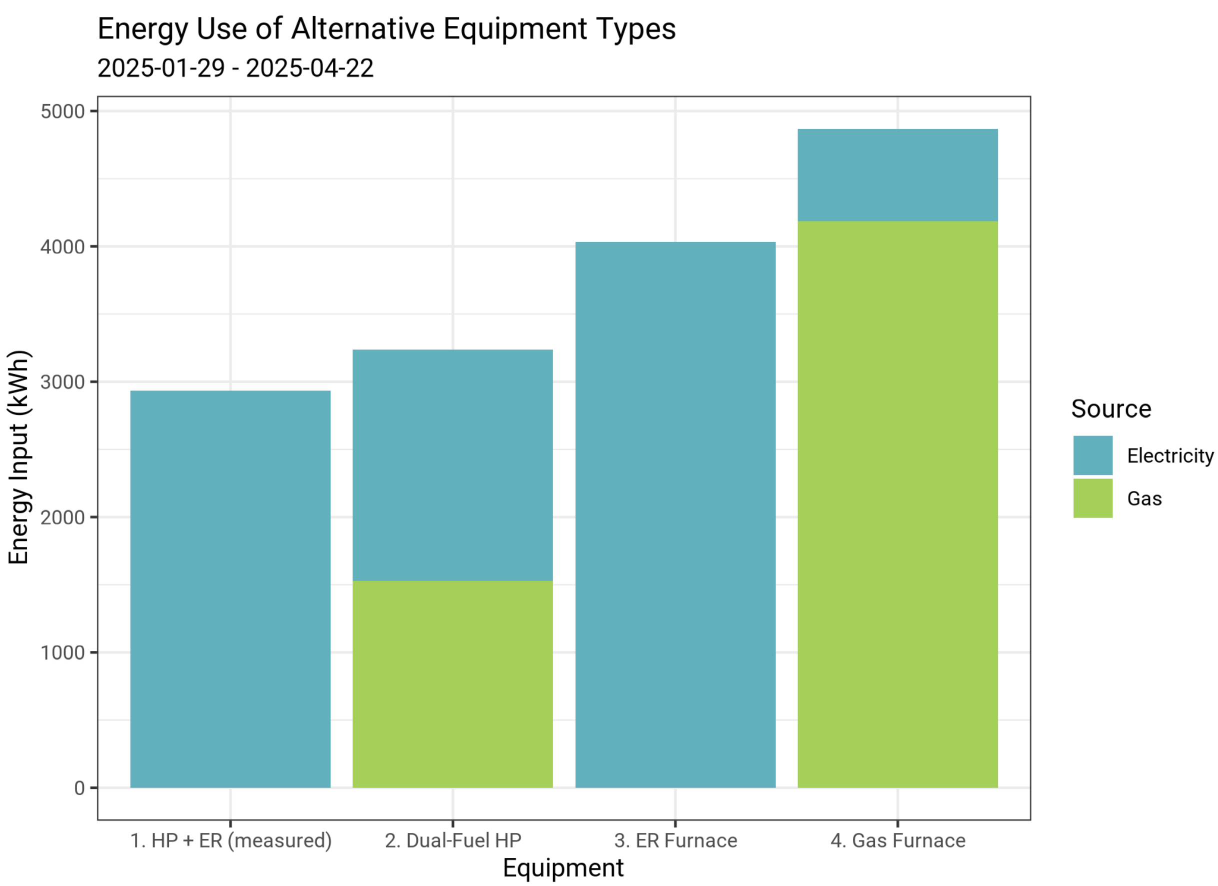
Energy Savings

The measured energy consumption of the system after the date the airflow obstructions were removed until the end of the heating season was 2,933 kWh of electricity. By disaggregating the energy consumption of the heat pump system, the supply fan, and the auxiliary electric resistance (ER) heating element, hypothetical energy consumption estimates for alternative equipment types delivering the same heating energy to the space were developed as shown in Table 3. As shown in Figure 13, each alternative used more energy in absolute terms relative to the baseline. In order of increasing energy consumption, the alternative systems considered were a dual fuel heat pump (heat pump with auxiliary gas heat), electric resistance furnace, and conventional gas furnace RTU.

Table 3. Energy Consumption Estimates by Equipment Type and Source

Equipment	Total Electricity (kWh)	Total Gas (kWh)	Total Energy (kWh)
HP + ER	2933	0	2933
Dual-Fuel HP	1709	1530	3239
ER Furnace	4031	0	4031
Gas Furnace	683	4186	4868

Figure 14. Energy Consumption Estimates by Equipment Type and Source



Cost Savings

While the installed system used less energy than the alternatives, it is not necessarily the lowest cost option due to the discrepancy in cost per unit of energy between electricity and gas. In addition, the impact of electric demand charges can significantly increase the cost of electric resistance as an auxiliary heat source. Table 4 presents the energy and demand estimates for the baseline and the dual fuel alternative for each full month of the heating season after cleaning. The savings values can be multiplied by relevant rates to determine the approximate costs of each alternative.

Table 4. Electricity and Gas Energy and Demand for Installed System and Dual Fuel Heat Pump

	HP + ER			Dual-Fuel HP			Savings		
Month	Electric Energy (kWh)	Electric Demand (kW)	Gas Energy (therms)	Electric Energy (kWh)	Electric Demand (kW)	Gas Energy (therms)	Electric Energy (kWh)	Electric Demand (kW)	Gas Energy (therms)
February	1759	26	0	865	3.0	38.1	894	23	-38.1
March	747	16	0	515	3.6	9.9	232	12.4	-9.9

Emissions Savings

Emissions savings estimates are calculated using CO2e grid emissions factors for the MROW eGRID subregion and fuel emissions factors from the U.S. EPA emissions factors hub combined with the energy consumption estimates from Table 3. Table 5 presents the emissions savings calculations for the installed HP + ER system and the hypothetical emissions from a dual fuel system. Due to the relatively high grid emissions factor for MROW and relatively low observed system COPs, the emissions of the installed system were about 23.8% higher than those that a dual fuel system would have produced over the same period.

Table 5. Emissions Calculations vs. Dual Fuel Heat Pump

ID	Parameter	Calculation	Value
A	Electric Emissions Factor (kg CO_2e /kWh)	[conversion]	0.420
B	Gas Emissions Factor (kg CO_2e /kWh)	[conversion]	0.181
C	HP + ER Electric Input (kWh)	[measured]	2933
D	HP + ER Gas Input (kWh)	[estimated]	0
E	HP + ER Emissions (kg CO_2e)	A * C + B * D	1233
F	Dual Fuel Electricity Input (kWh)	[measured]	1709
G	Dual Fuel Gas Input (kWh)	[estimated]	1530
H	Dual Fuel Emissions (kg CO_2e)	A * F + B * G	996
I	Emissions Savings (%)	100 * (H - E)/H	-23.8

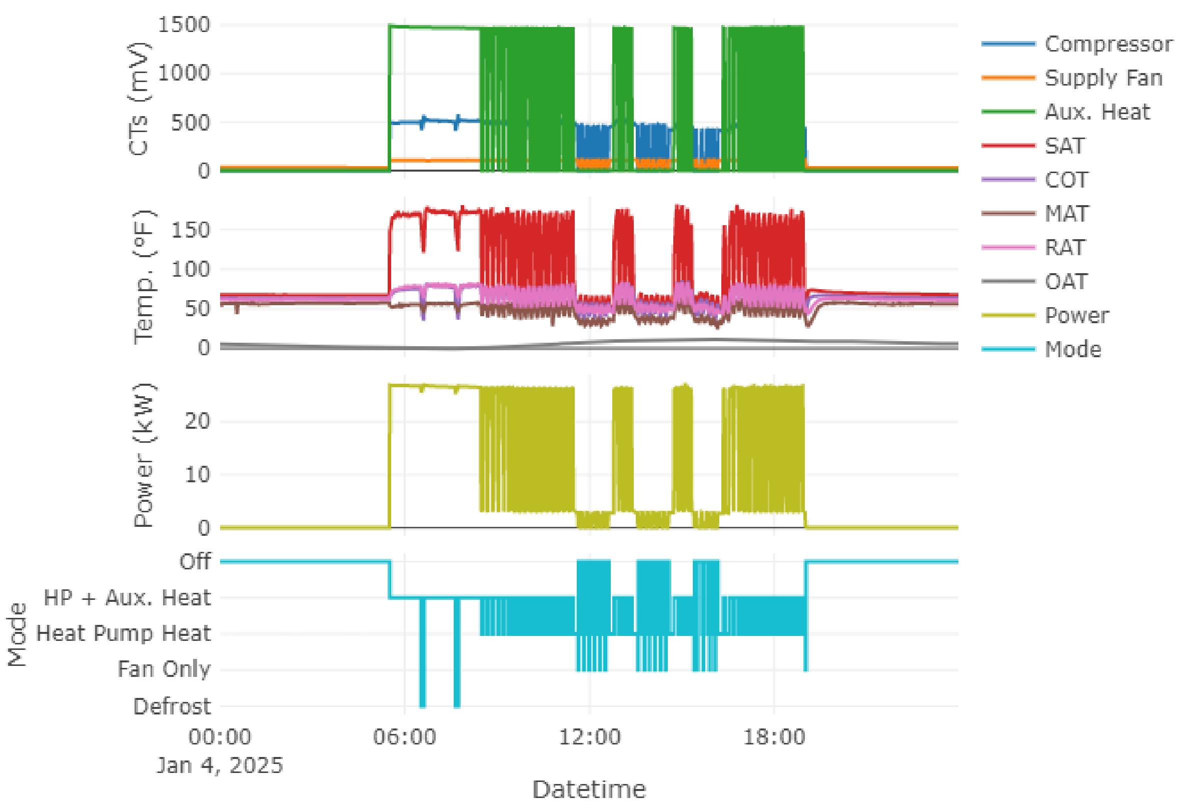
TIME SERIES EXAMPLES

The following plots show the underlying data to provide additional context and support for the results.

Typical Day before Cleaning

Before cleaning, the morning warm-up period after 5:30 a.m. was characterized by a long period of HP + Aux. Heat operation. The system did not enter fan-only mode until almost 12:00 p.m., then bounced between several modes including Fan Only, Heat Pump Heat, HP + Aux. Heat, and Off, as shown in Figure 15. Supply temperatures were well over 150°F when auxiliary heat was active.

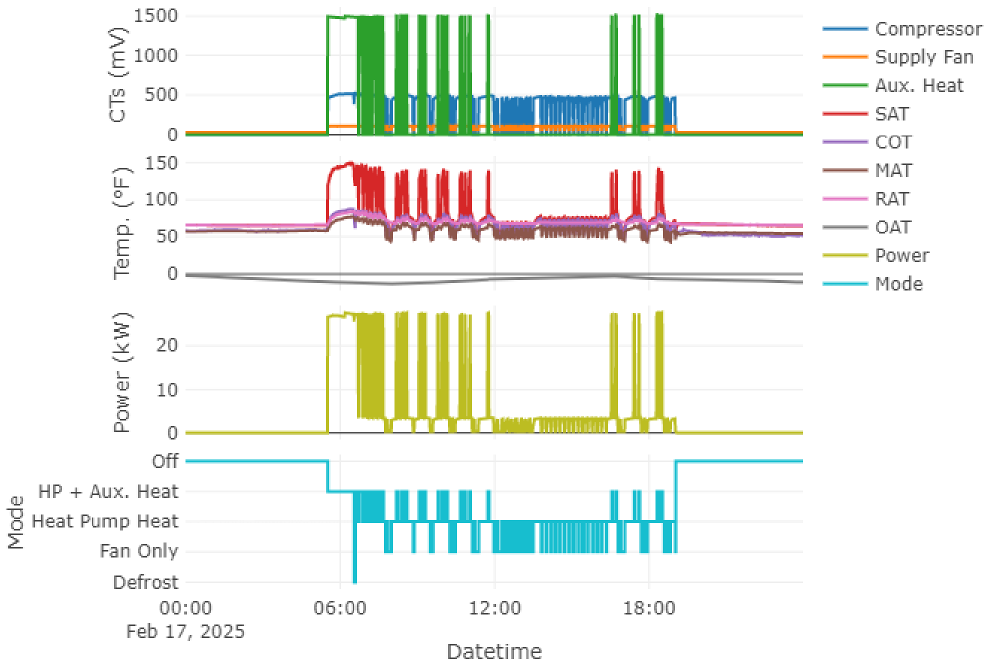
Figure 15. Timeseries Performance Before Cleaning



Typical Day after Cleaning

On a day after the grille cleaning with similar very cold ambient temperatures, use of auxiliary heat was drastically reduced. The system was able to alternate between fan-only and heat pump heating modes, maintaining more stable supply air temperature for several hours from around 12:00 p.m. until around 4:00 p.m. Peak supply air temperatures with auxiliary heating were reduced to less than 150°F, and the system operated continuously without turning off during the day, as shown in Figure 16. It is possible that this change in fan behavior is due to a change in thermostat settings from “Auto” to “On”. The following sections examine different periods of operation from this day.

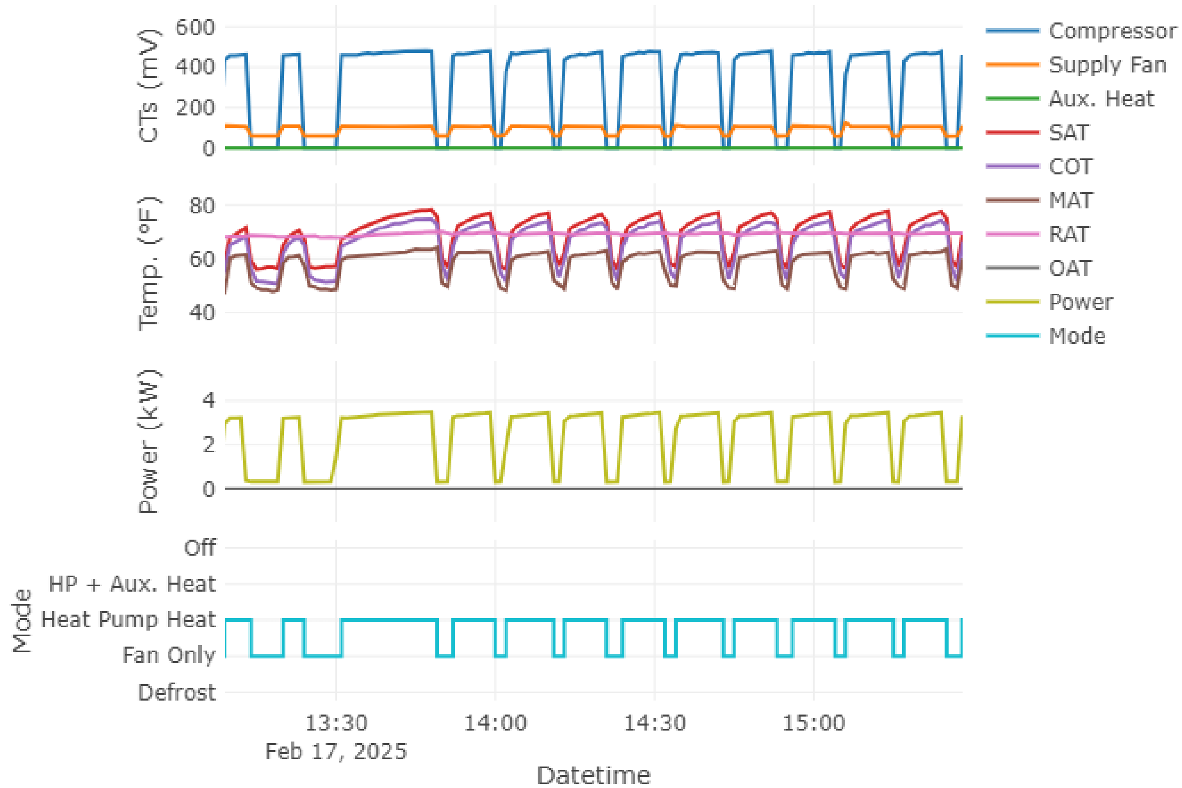
Figure 16. Timeseries Performance After Cleaning



Heat Pump Heating

Focusing on the heat pump cycles in the afternoon, a regular pattern of roughly five compressor cycles per hour is evident in Figure 17. The heat pump duty cycle is quite high, and the supply air temperature is low, around 80°F in this example, which is to be expected given the subzero ambient temperatures. Still, the system can maintain the return air temperature around 70°F without back-up heat in these conditions once the space has recovered from its nighttime setback.

Figure 17. Timeseries Heat Pump Cycles



Heat Pump + Aux. Heat

The morning warm-up period is characterized by simultaneous heat pump and auxiliary heating. Especially since this unit uses electric auxiliary heating, significant cost savings are likely possible by reducing or eliminating the overnight setback so that the unit controls call for auxiliary heat less often and thus require less peak power. Figures 18 and 19 show how the morning warm-up period consumes almost half the energy required for the entire day of operation at very cold temperatures. For improved energy performance, the heat pump should operate to maintain the space at a consistent temperature at all times, with outdoor air dampers opening in response to ventilation demand.

Figure 18. Timeseries Cumulative Energy Consumption

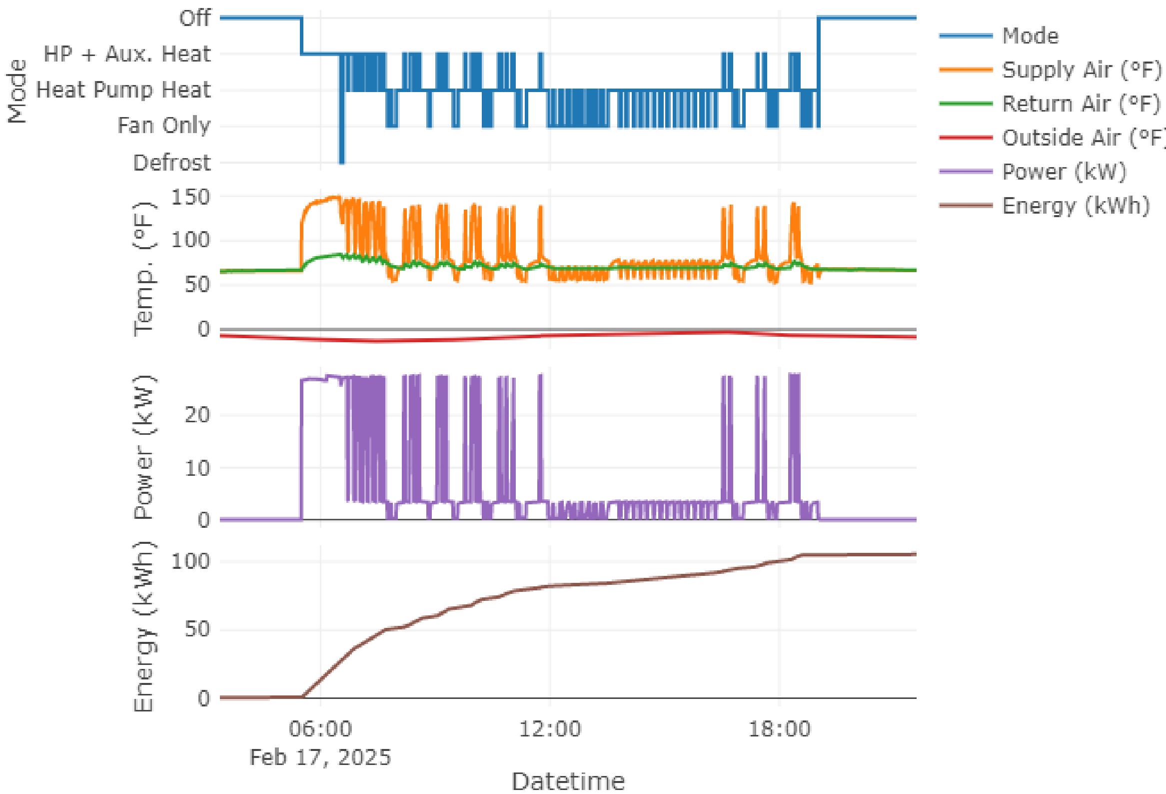
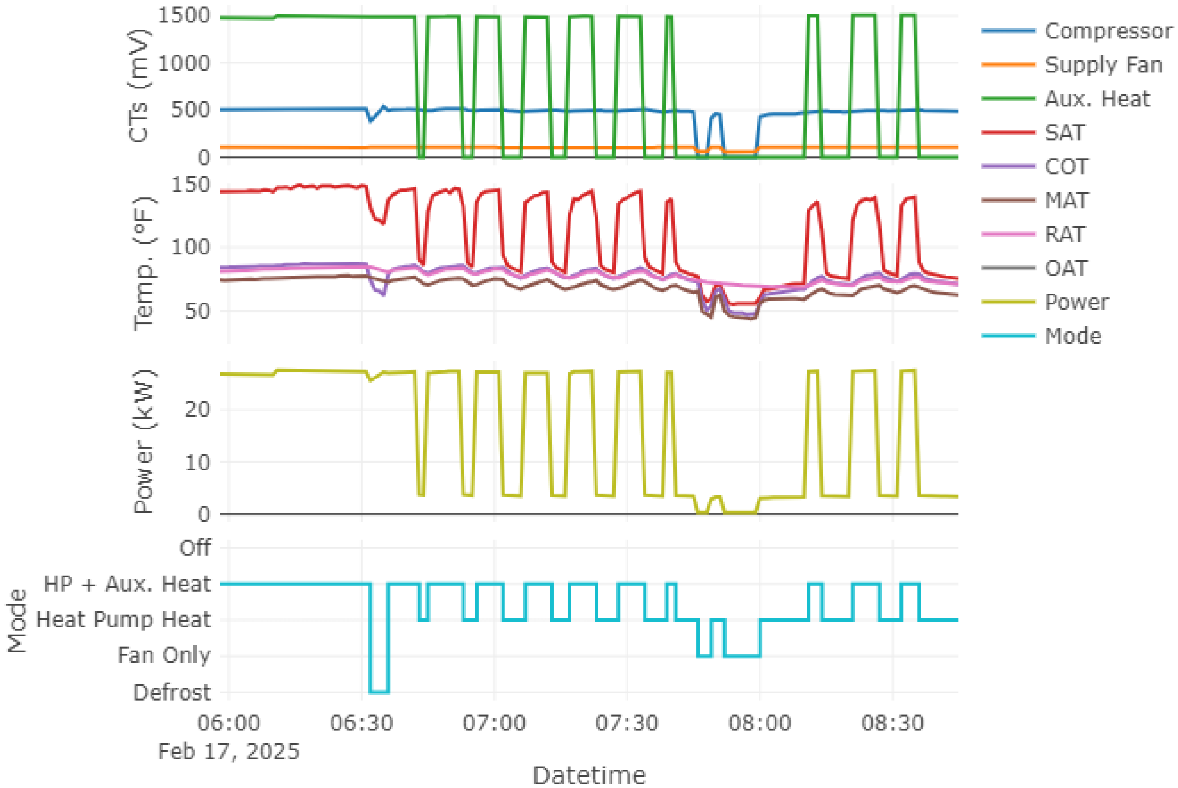


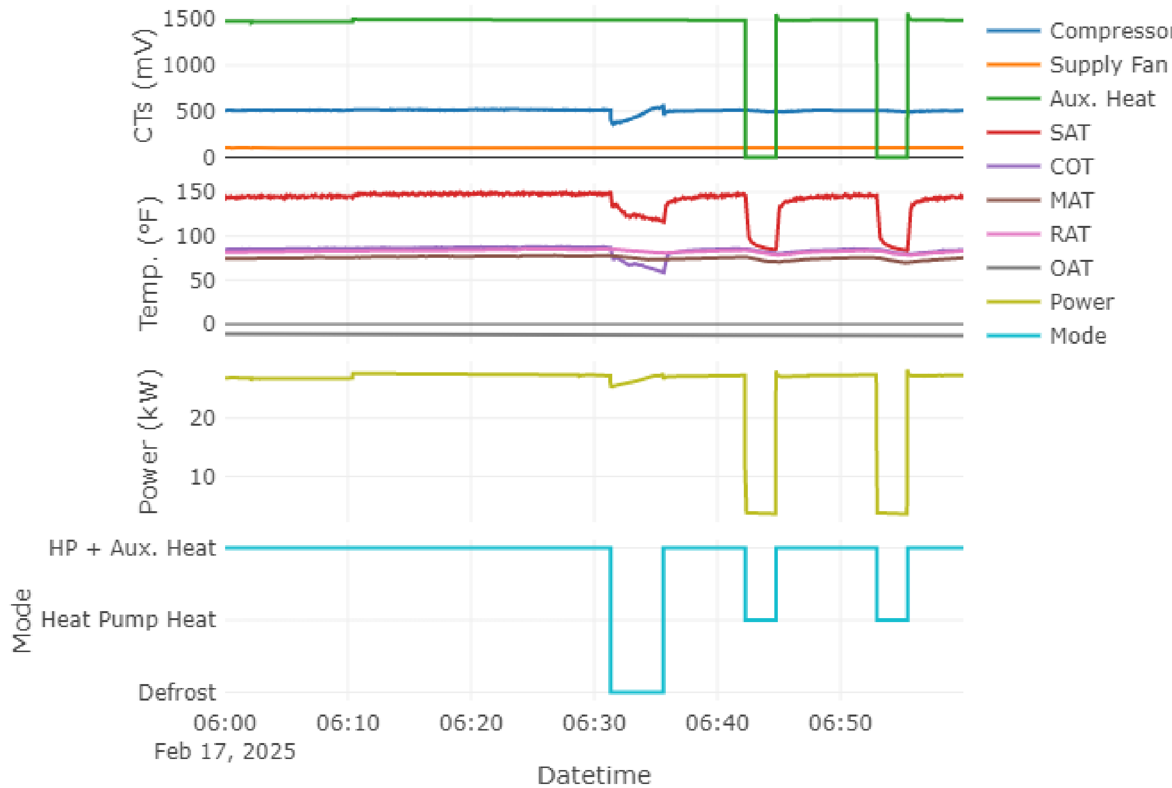
Figure 19. Timeseries Auxiliary Heat + Heat Pump



Defrost

The auxiliary heating system on this unit is large relative to the heat pump, so the effect of defrost on supply temperature is negligible. When in defrost, the heat pump system operates in cooling mode to reject heat to the outdoor coil and remove any accumulated frost, and the auxiliary heat compensates for the cooling effect of the heat pump. Supply temperatures remain above 100°F throughout defrost. Defrosts were relatively rare for this unit, usually only occurring fewer than three times per day, as shown in Figure 20.

Figure 20. Timeseries Defrost



CONCLUSION

This project demonstrates the potential and challenges of using all-electric heat pump RTUs in cold climate commercial buildings. The system consistently delivered heating through colder months and maintained occupant comfort, but performance was sensitive to airflow and control strategies. Key factors like decreasing or eliminating setbacks and improving return airflow would significantly reduce reliance on auxiliary heating and improve efficiency. While the heat pump RTU used less energy than other modeled alternatives, cost savings were limited by high demand charges and emissions were higher due to auxiliary heat use and grid mix. This report explores the importance of maintenance, especially when building electrification becomes critical to meeting carbon reduction goals.

This all-electric heat pump RTU provided comfort to occupants throughout the year, without complaints or down time. The unit had 40% energy savings compared to a standard efficiency gas furnace RTU and used no natural gas. With these results in mind, heat pump RTUs should be considered a primary heating source for commercial buildings in Minnesota. Continued field research, training, and utility support will be essential to the success of next gen RTUs. **To find more information on how to install and maintain heat pump RTUs on commercial buildings in Minnesota, visit nextgenrtus.org.**

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