# BC Cold Climate Heat Pump Field Study | Project 21090.000

Final Report


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# Executive Summary

Heat pump performance ratings are based on test procedures that generally test the equipment in a laboratory under steady-state conditions. These operating conditions rarely occur in the real world and because there is limited research on how this equipment functions in service, it is difficult to understand their true efficiency at partial loads and throughout the year. New test procedures aim to provide designers with better metrics that are both load-based & climate-specific.

This study provides a better indication of the real-world efficiency and the resulting energy reduction and potential cost savings of air-source heat pump retrofits in existing homes in cold and moderate regions of British Columbia. The intended outcomes are to develop a clearer understanding of the performance of cold climate air-to-air heat pumps in Canadian climates; to evaluate design and installation considerations that may affect this performance; and, where possible, to identify design and installation best practices that positively affect the performance of heat pump systems. The information gathered in this study will inform the development of heat pump testing and rating procedures.

In April 2019, twenty-six ductless mini split and central heat pump systems were monitored at twenty-four distinct single-family residential sites on Vancouver Island and in the interior of British Columbia in Climate Zones 4 and 5. In-situ measurements of key heat pump system parameters and corresponding outdoor environment conditions were collected at five-minute intervals for each site over a one-year monitoring period. The following are key takeaways:

→ The average seasonal Coefficient of Performance (COP)[1] for cooling was estimated to be 5.0, 4.1 and 4.5 for ductless mini split, central single stage and central variable speed systems, respectively. The results show that participants are using active cooling across a wide range of outdoor temperatures. Generally, the measured heat pumps appear to be performing with an average COP greater than 1 in cooling season for all outdoor conditions, even during extreme heat above 38°C.

→ It was found that many participants are using heat pumps to cool the interior when outdoor temperatures are below typical interior temperatures. The efficiency of some units fluctuated significantly (as a result of short-cycling) when operated at outdoor temperatures below the average interior temperature. Although this phenomenon was not exhibited for all units, educating homeowners on the strategies and benefits of passive cooling strategies (i.e., natural ventilation) could reduce hours of heat pump operation during mild outdoor temperatures.

→ The average seasonal COP for heating was estimated to be 2.4, 2.6 and 3.3 for ductless mini split, central single stage and central variable speed systems, respectively. In heating season, the average seasonal COP of central units was relatively higher than ductless mini split units. However, most of the central systems were also found to reduce their

[1]COP is an efficiency ratio that represents the units of energy output per unit of energy consumed by the system.

operation or stop heating between 3°C and -5°C; relying on a backup heating system to supplement the heating load at lower outdoor temperatures.

→ The average COP for heating was estimated for the entire heating season and correlated with outdoor temperature. Results show that the overall average COP for all heat pumps is greater than 1, even down to -14°C. However, two units were found to have a COP less than 1 at outdoor temperatures around 0°C and below. For the poorest performing unit, it was found that heating capacity of the system dropped below the energy demand around 0°C and continued to drop as outdoor temperature got colder. Based on conversations with the homeowner, there is reason to believe that leaked refrigerant may be responsible for the poor performance during the winter season.

→ Energy savings and cost evaluations were performed. Of the 18 participants with available utility data, 12 (67%) experienced annual energy savings after the heat pumps were installed. Cases that used electricity as a primary heating source were separated and average savings were found to be 5650 kWh and $810 for the year-long monitoring period. Cases that used non-electric primary heating fuel sources varied significantly, where homes either saw a decrease or an increase in their electricity consumption after the heat pump installation. In all, the average savings for these homes was still found to be 1520 kWh and $231 for the year-long monitoring period. The use of non-utility based fuel sources before and/or after heat pump installation means that a full accounting of energy consumption and costs was not possible for most sites, and while overall energy and GHG savings were demonstrated, it is difficult to draw broader conclusions about energy and GHG savings potential from the field study.

→ Lessons learned based on some site monitoring intricacies discovered during this study were outlined to improve future in-situ heating pump instrumentation techniques, including suggested return and supply air temperature sensor placement.

Some key factors affecting heat pump performance were also explored, including volumetric flow rate, equipment sizing and short-cycling, defrost control, backup heating, and some installation considerations, with the following findings:

→ During the initial site visit, volumetric flow rate of all units was measured at every fan speed and compared to manufacturer rated flow rates. Overall, the average measured volumetric flow rates were 64% of (or 36% lower than) the flow rates listed in manufacturer data sheets. The lower measured results compared to listed flow rates are consistent with previous studies. The central systems were found to exhibit average flow rates closer to manufacturer ratings (72%) compared to mini split systems (62%), despite central system flow rates being measured at supply louvers after the air has travelled through existing (likely leaky) ductwork within the homes.

→ Lower than rated volumetric flow rates for ductless mini split units are potentially due to lab testing methodologies that typically do not include back pressure caused by the presence of the supply louvers. It was also found that many of the mini split indoor heads were installed with limited ceiling clearance, which could be restricting the flow of air at the return airstream intake. Research into the manufacturer specified clearances shows a listed minimum clearance range between 3.9” (100mm) and 1.2” (30mm). While most of the units were installed within the specified acceptable minimum clearances, low ceiling clearances could still be negatively affecting the volumetric flow rate of the indoor units. Future research should include controlled testing of units with various clearance ranges for a greater understanding of its overall impact on volumetric flow rate.

- Samples of heating cycles were examined and evidence of short-cycling (i.e., cycles less than 5-8 minutes) during typical heating periods was found in 33% of units, suggesting some units may be oversized. Oversized units can cause a space to quickly reach its set point and shut off before the unit can reach an optimal efficiency, which can negatively affect the overall efficiency of the unit. During the initial site visit, participants were asked if they had access to any documentation related to heat pump sizing. With the exception of one case, no participant had seen or received any documentation to confirm that their units had been formally sized for their home.
- An analysis of energy consumption from electric resistance backup heating coils in central systems was performed. Results show that the average backup heat consumption accounted for 22% of total space conditioning, and as much as 63% for one unit. Differentiation of electric resistance consumption for defrost versus backup heat was not possible for central systems, though some inferences were made suggesting that more than half of this coil use was allocated to defrost cycle rather than supplementary heating at cold temperatures. Because ductless mini split units typically are not equipped with backup heat, the defrost cycles are more easily measurable by isolating periods of cooling during the winter season. A defrost cycle analysis for ductless mini split systems was therefore conducted and results showed that defrost cycle energy accounted for less than 1% of total mini split heat pump consumption during winter.
- Conditions and variables that made definitive conclusions challenging include small sample size, variations in home size and construction, different primary and backup heating sources, occupant behaviour, and instrumentation limitations.

This study has shown that there is potential for widespread adoption of heat pumps in British Columbia, particularly for retrofit applications in homes that rely primarily on electricity as their primary heating source. Since the analysis was limited to homes in Climate Zone 4 and 5, further studies investigating the in-situ performance of heat pumps in colder climates is recommended prior to widespread adoption in these regions.

# Contents



# Appendices

# 1 Introduction

Air-source heat pumps are increasingly being used as an efficient means of interior space conditioning. Rather than converting heat from fuel, heat pumps use a compressor with refrigerant to draw heat from the outdoors during the heating season and reject heat to the outdoors during the cooling season. Heat pumps can achieve high efficiencies as the amount of energy required to power the compressor and fans should be less than the total amount of heat that is being moved. These types of systems have been known to deliver 1.5 to 3 times more heat energy than the electrical energy they consume, with the variation tied primarily to outdoor temperatures.

Heat pump performance ratings are based on test procedures that generally test the equipment in a laboratory under steady-state conditions. These operating conditions rarely occur in the real world and because there is limited research on how this equipment functions in service, it is difficult to understand their true efficiency at partial loads and throughout the year. This study provides a better indication of the real-world efficiency and the resulting energy reduction and potential cost savings of air-source heat pump retrofits in existing homes in cold and moderate regions of British Columbia.

The main outcomes of this study are to develop a clearer understanding of the performance of cold climate air-to-air heat pumps in Canadian climates; to evaluate design and installation considerations that may affect this performance, and to identify design and installation best practices that positively affect the performance of heat pump systems. The information gathered in this study will inform the development of heat pump testing and rating procedures.

Twenty-six ductless mini split and central heat pump systems were monitored at twenty-four distinct single-family residential sites on Vancouver Island and in the interior of British Columbia. In-situ measurements of key heat pump system parameters and corresponding outdoor environment conditions were collected at five-minute intervals for each site over a one-year monitoring period. Pertinent installation information and observations were also recorded at the time of monitoring instrumentation.

This report summarizes the collected data and analysis for the complete monitoring period of this study from April 2019 to April 2020.

## 1.1 Background

This section briefly describes the metrics used in this study to describe heat pump efficiency; common factors including heat pump short-cycling and defrost control and their effect on performance, as well as other system installation considerations.

### *Coefficient of Performance (COP)*

The coefficient of performance (COP) is a unitless value often used to measure a heat pump's efficiency. The COP is a ratio of heat energy delivered to or removed from an indoor environment compared to the amount of energy supplied to the system. For example, a typical electric resistance baseboard heater should have a COP of 1 since all the energy supplied to the system is converted to heat energy. Because heat pumps move energy through mechanical advantage rather than directly from electricity, they can achieve a COP greater than 1.

$$COP = \frac{heat\ delivered\ to\ or\ removed\ from\ indoor\ environment\ (W)}{power\ consumed\ by\ the\ heat\ pump\ (W)}$$

### *Seasonal Energy Efficiency Ratio (SEER)*

The Seasonal Energy Efficiency Ratio (SEER) is a measurement of the efficiency of a cooling system for an entire season. It is calculated by summing the total amount of cooling removed by the heat pump system (in BTU) during the cooling season and dividing that by the total electric energy consumption (in watt-hours) during the same period. SEER values are typically determined in a laboratory setting by conducting tests at varying indoor and outdoor conditions, including a measure of performance under cyclic operation. The equivalent SEER can also be solved by dividing the seasonal cooling COP by 0.293.[2]

$$SEER = \frac{total\ cooling\ energy\ removed\ over\ a\ season\ (BTU)}{input\ electrical\ energy\ during\ the\ cooling\ season\ (Wh)}$$

### *Heating Season Performance Factor (HSPF)*

The Heating Seasonal Performance Factor (HSPF) is a commonly used metric for rating heat pump heating efficiency. HSPF is calculated by dividing the total heat energy delivered to the space by the heat pump system (including backup heat) during the heating season (in BTU) by the total input electrical energy of the unit (in watt-hours) during the same period. HSPF is also determined in a laboratory setting by conducting tests at very limited conditions, including a measure of cyclic performance. Like SEER, the equivalent HSPF can also be solved by dividing the seasonal heating COP by 0.293.

$$HSPF = \frac{total\ heating\ energy\ delivered\ over\ a\ season\ (BTU)}{input\ electrical\ energy\ during\ the\ heating\ season\ (Wh)}$$

It should be noted that SEER and HSPF are derived at a full output at specific ambient temperatures. Therefore, they do not utilize the on-board algorithms that are an essential component of variable-speed systems during in-field operation.[3] For this reason, SEER & HSPF are typically viewed as poor indicators of field performance. New test procedures aim to provide designers with better metrics (e.g., seasonal COP) that are both load-based & climate-specific.

### *Short-Cycling*

When a heat pump is short-cycling, the system repeatedly shuts on and off. Because the system requires more energy when it begins a cycle, the efficiency benefits associated with running the system for long periods of time are not achieved. For this reason, short-cycling of heat pumps has a significant negative impact on energy efficiency. One study found that degradation in the efficiency of heat pumps is greatest when units ran the compressor for less than six minutes and suggest that air source heat pump units should run for at least

[2] ASHRAE (2016). HVAC Systems and Equipment (SI Edition)

[3] CSA EXP07:19. *Load-Based and Climate-Specific Testing and Rating Procedures for Heat Pumps and Air Conditioners*. CSA Group 2019. Toronto, Canada.

eight minutes.[4] Although there are other factors that can cause a heat pump to short-cycle, the most common is due to improper sizing of the unit. Oversizing a heat pump will cause the room temperature to reach its set point more quickly, signaling the unit to turn off. Oversizing a unit in a cold climate can also promote ice build-up on the compressor coil and potentially cause damage, as short runtimes may not allow for the heat pump to activate it's defrost control mode.

### *Defrost Control*

When the outdoor unit coil temperature is below the dew point, condensation can occur on the coil as moist air passes over it. The condensation on the coil can turn to ice when the temperature is below freezing. Ice build-up can reduce the efficiency of the heat transfer from the coil to the outdoor environment. Heat pump units are commonly equipped to remove or defrost any ice build-up on the outdoor unit coil by temporarily reversing the refrigerant cycle and forcing the unit into cooling mode, which delivers heat to the outdoor coil. Though all system types can be equipped with backup heating (e.g., electric resistance heat coil) during these cycles, they are more commonly found in central systems. Unless the heat pump is equipped with a backup electric or propane heating element, the unit will temporarily provide cooling to the indoor space while in defrost mode. Research has found that the average heat pump energy demand from defrosting is relatively small, around 1kW.[5]

There are two types of defrost control: time-temperature and demand-defrost. Time-temperature defrost is controlled by either a timer or a temperature sensor located at the outside unit coil. Once triggered, the defrost control will typically run automatically for 30, 60 or 90 minutes depending on system design and climate. Demand-defrost control, on the other hand, can detect frost accumulation by monitoring airflow, refrigerant pressure, air and coil temperature. Generally, the demand-defrost control is more efficient because it only runs when and as long as needed to defrost the coil.[6]

### *Backup Heating*

Both ductless mini split and central heat pump systems can be equipped with a backup heating coil (i.e., strip heaters), though they are more commonly found in central systems. Backup heating coils can be programmed to operate at low temperatures, when a heat pump may begin to struggle to extract heat from the outdoor environment to meet the heating demand of the home. For example, ductwork that was designed to deliver relatively hotter air from a fuel burning appliance, may be too small to allow the relatively cooler air from a heat pump to be delivered because of the additional volume of air required to provide the same amount of heating throughout the home. This is an important detail given that all of the central heat pumps in this study were connected to existing ductwork.

Since the efficiencies of electric resistance (100%) and natural gas combustion (80-95%) are relatively lower than the efficiency of a heat pump compressor, the contribution of backup

[4] Green, R. *The Effects of Cycling on Heat Pump Performance (2012).* Prepared for Department of Energy and Climate Change (DECC). Available at https://assets.publishing.service.gov.uk/government/uploads/system/uploads/attachment_data/file/65695/7389-effects-cycling-heat-pump-performance.pdf

[5] Mei, V.C., Domitrovic, R.E., & Chen, F.C. 2002. *The Development of a Frost-Less Heat Pump.* Oak Ridge National Laboratory. Residential Buildings: Technologies, Design, Performances Analysis, and Building Industry Trends – 1.189.

[6] https://www.nrcan.gc.ca/sites/oee.nrcan.gc.ca/files/pdf/publications/infosource/pub/home/heating-heat-pump/booklet.pdf

heating can affect the overall performance of a home's heating system as a whole.[7] In other words, the overall performance of the heating system will be negatively affected as more backup heating is relied upon. Therefore, backup heating should be operated as sparingly as possible to maximize the efficiency of heat pump systems. Note that, for the purposes of this study, backup heating does not refer other supplementary forms of heat external to the heat pump furnace system, such as baseboards or fireplaces.

### *Design & Installation Considerations*

Equipment design and installation can significantly affect the performance of heat pumps. Following best practice guidelines can improve the potential heating, cooling and indoor air quality benefits of heat pumps. Publicly available documents such as the Home Performance Stakeholder Council's *Heat Pump Best Practice Guide for Existing Homes* are available for designers, installers and homeowners.[8] Tips and other considerations are provided, including relevant standards such as CSA F280-12 (heat pump sizing) and CSA C273.5-11 (installation), to ensure that heat pumps are designed and installed to operate to their full potential.

[7] Palmiter, L. and P. W. Francisco. 1997. *Development of a Practical Method for Estimating the Thermal Efficiency of Residential Forced-Air Distribution Systems*. Electric Power Research Institute report TR-107744, Palo Alto, CA.

[8] Home Performance Stakeholder Council. *Heat Pump Best Practices Installation Guide for Existing Homes*. ICF Canada. 2019.

# 2 Methodology

The methodology for this study was designed in accordance with the International Performance Measurement and Verification Protocol (IPMVP) core concepts (Volume I EVO 10000 – 1:2016).[9] The study incorporates a hybrid approach of *Option A: Retrofit Isolation: Key Parameter Measurement* for heat pump efficiency (COP) and *Option C: Whole Facility* for savings analysis before and after heat pumps were installed. The main objectives of this study, based on in-situ field performance and whole-building energy savings, are as follows:

→ Compare field performance results with reported performance data,

→ Estimate heat pump COP throughout the range of outdoor temperatures experienced in winter, summer and shoulder season, and

→ Evaluate actual energy and cost savings resulting from the heat pumps by comparing pre- and post-installation utility bills, normalized for weather.

## 2.1 Site & System Selection

Twenty-four sites were selected within Vancouver Island (Victoria) and the Interior of British Columbia (Salmon Arm, Kelowna, Princeton, Summerland, and Penticton). *Figure 2.1* is a map with approximate locations of monitored sites in bold. Note that all of homes were located in either Climate Zone 4 or 5. A variety of system types were monitored including central systems with single or variable speed compressors, and variable speed ductless mini split systems with a single or multiple indoor head(s). Table 1 lists all the indoor and outdoor heat pump types and model information. To qualify as a cold-climate heat pump, a minimum seasonal performance of HSPF greater or equal to 10 was required for single-head ductless mini split systems; 9.5 for multi-head, and 8.5 for central single speed compressor systems, respectively.

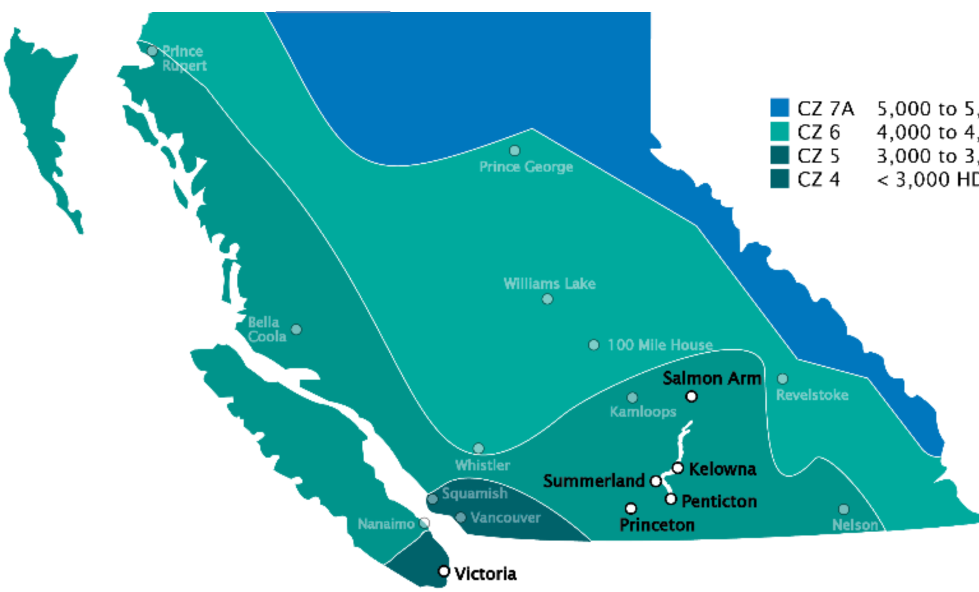


*Figure 2.1 – schematic map of southern British Columbia with site locations, climate zones and heating degree days (based on NECB 2017).*

9 https://evo-world.org/en/products-services-mainmenu-en/protocols/ipmvp

TABLE 1 - INDOOR AND OUTDOOR HEAT PUMP MODEL INFORMATION

| ID | System Type | Manufacturer | # of Outdoor Units | # of Indoor Units | Outdoor Unit A Model No. | Indoor Unit 1 Model No. | Indoor Unit 2 Model No. | Outdoor Unit B Model No. | Indoor Unit 3 Model No. | HSPF | SEER |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **KEL01A** | Ductless | Mitsubishi | 1 | 1 | MUZ-FH12NAH | MSZ-FH12NA | | | | 11.5 | 26.1 |
| **KEL01B** | Ductless | Mitsubishi | 1 | 1 | MUZ-FE09NAH | MSZ-FE09NA | | | | 10 | 26 |
| **KEL02** | Central Ducted (Single Stage Compressor) | Payne | 1 | 1 | PH15NB04800G | PF4MNB049 | | | | 9 | 15 |
| **KEL03** | Ductless | Daikin | 1 | 2 | 3MXS24RMVJU | CTXG18QVJUW | CTXGO9QVJUW | | | 12.5 | 18 |
| **KEL04** | Central Ducted (Single Stage Compressor) | York | 1 | | YZF04813CA | (unknown) | | | | 9 | 15 |
| **KEL05** | Central Ducted (Variable Speed Compressor) | Carrier | 1 | 1 | 25VNA024A300 | FE4ANF002 | | | | 10.5 | 19.1 |
| **KEL06** | Ductless | Daikin | 2 | 3 | 2MXS18NMVJU | FTXS12LVJU | FTXS09LVJU | RXS12LVJU | FTXS12LVJU | 13/10.7 | 23/18.9 |
| **KEL07** | Ductless | Fujitsu | 1 | 1 | AOU12RLS3H | ASU12RLS3Y | | | | 13.9 | 29.3 |
| **PRI01** | Ductless | Fujitsu | 2 | 2 | AOU12RLS3H | ASU12RLS3Y | | AOU9RLS3H | ASU9RLS3Y | 13.9/14.1 | 29.3/33 |
| **SAL01** | Ductless | Fujitsu | 1 | 1 | AOU18RLXFWH | ASU18RLF | | | | 10.4 | 20 |
| **SUM01** | Central Ducted (Single Stage Compressor) | Payne | 1 | 1 | PH14NB030-A | CNPVU3017ALA | | | | 8.2 | 14 |
| **PEN01** | Central Ducted (Variable Speed Compressor) | Mitsubishi | 1 | 1 | PUZ-HA36NHA5 | PVA-A36AA4 | | | | 11 | 17 |
| **VIC01** | Central Ducted (Variable Speed Compressor) + Ductless | Mitsubishi | 1 | 2 | MXZ-3C30NA2 | MSZ-GL06NA | SVZ-KP18NA | | | 10.3 | 17.6 |
| **VIC02** | Ductless | Fujitsu | 1 | 2 | AOU24RLXFZ | AGU15RLF | AGU9RLF | | | 9.5 | 18 |
| **VIC03** | Ductless | Fujitsu | 1 | 2 | AOU24RLXFZ | ASU15RLF1 | ASU12RLF1 | | | 9.5 | 18 |
| **VIC04** | Central Ducted (Single Stage Compressor) | York | 1 | 1 | YZF03013CA | AHV36C3XH21CC | | | | 9 | 15 |
| **VIC05** | Ductless | Mitsubishi | 1 | 1 | MUZ-FH18NAH2 | MSZ-FH18NA2 | | | | 11 | 21 |
| **VIC06** | Ductless | Daikin | 1 | 2 | 2MXS18NMVJU | FTXS09LVJU | FTXS09LVJU | | | 11.6 | 21.7 |
| **VIC07** | Ductless | Fujitsu | 1 | 1 | AOU12RLS3 | ASU12RLS3Y | | | | 14 | 29.3 |
| **VIC08** | Central Ducted (Single Stage Compressor) | York | 1 | 1 | YZF03013CA | AHV36C3XH21CC | | | | 9 | 15 |
| **VIC09** | Ductless | Fujitsu | 1 | 1 | AOU12RLFW1 | ASU12RLF1 | | | | 11 | 22 |
| **VIC10** | Ductless | Fujitsu | 1 | 1 | AOU12RLS3 | ASU12RLS3Y | | | | 14 | 29.3 |
| **VIC11** | Central Ducted (Single Stage Compressor) | Lennox | 1 | 1 | XP14-024-230-09 | CBX32MV-024/030-230-6-08 | | | | 8.7 | 15.2 |
| **VIC12** | Ductless | Daikin | 1 | 1 | 3MXS24RMVJU | FTXS18LVJU | | | | 11.7 | 19.2 |

## 2.2 Field Data Collection

This section describes the variables that were measured in the field to estimate the in-situ performance of the selected heat pumps.

### *System & Fan Consumption*

Power and energy meters (wattnodes) with current transformers (CTs) were installed by Structure Monitoring Technology Ltd. (SMT) with the assistance of local electricians to measure both the total heat pump system consumption (250A CTs) and indoor unit fan power (20A CTs). For mini split systems, wiring for the wattnodes and CTs was routed from the heat pump to a waterproof box mounted next to the outdoor unit (*Figure 2.2*). For central systems, the boxes were typically mounted indoors next to the circuit breaker panel. Note that for central systems, the current transformers measuring furnace fan energy also registered backup heating energy from the backup heating coil (if electric resistance), which was located on the same circuit. This detail is relevant to the defrost cycles and backup heating methodology in Section 2.3. Finally, Linux-based cellular modems (Data Gateways) were installed in each of the participants' homes to facilitate collection and cloud-based transfer of the data.

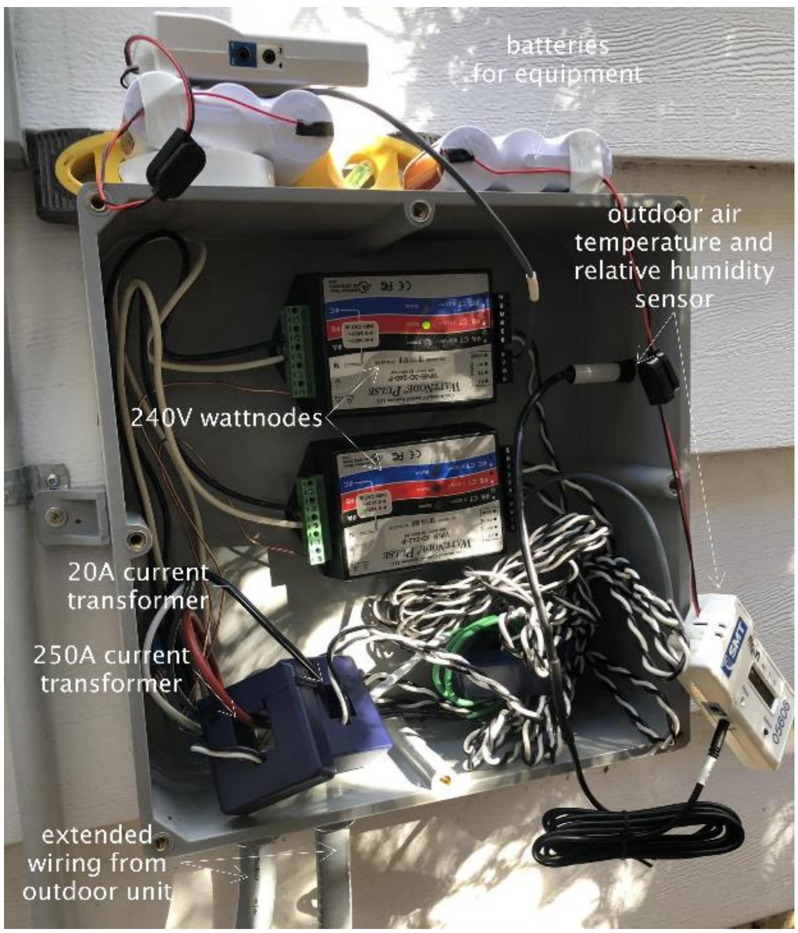


*Figure 2.2 –mounted box for housing electricity consumption monitoring equipment. All equipment (shown here on top of box during installation), including batteries, was placed inside the box with a waterproof lid for weather protection.*

### *Airflow Measurements*

To estimate the volumetric flow rate of the air passing through the indoor units throughout the monitoring period, the volumetric flow rate was measured at each fan setting during the initial site visit.[10] Indoor unit fans were sub-metered and used as a proxy for fan speed, to which measured flow rates were assigned depending on the fan speed setting. The method used to determine the ductless mini split volumetric flow rate at the indoor unit supply was as follows (see *Figure 2.3*):

- Secure a sealed airtight box around supply louver
- Connect a flexible duct from the box to a variable speed fan
- Measure with a dual channel manometer:
  - Pressure difference inside box in relation to the ambient indoor environment
  - Pressure difference across the fan
- Adjust fan speed until pressure difference is null between airtight box and indoor ambient environment (i.e., until CFM of variable speed fan matches heat pump fan CFM)
- Record CFM across the variable speed fan

For central ducted systems, the volumetric flow rate was determined by measuring the cumulative flow rate of all supply outlets with a powered flow hood, for each fan setting as applicable (see *Figure 2.4*).

[10] Williamson, James and Robb Aldrich. *Field Performance of Inverter-Drive Heat Pumps in Cold Climates*. Prepared for U.S. Department of Energy. August 2015.

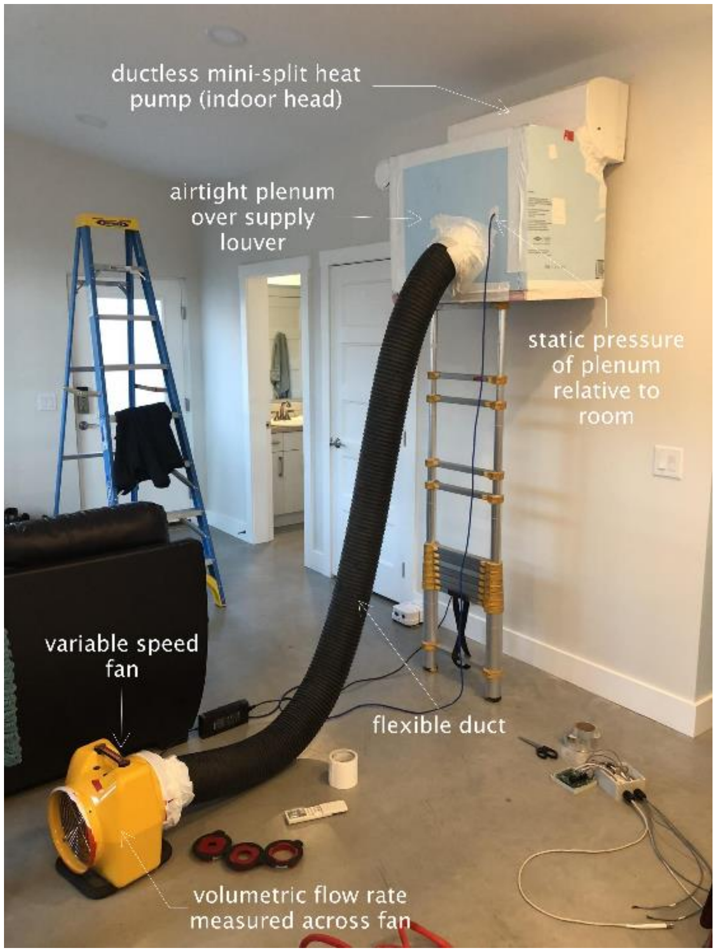


*Figure 2.3 – volumetric flow rate test configuration for ductless mini split units.*

*Figure 2.4 – powered flow hood measuring volumetric airflow at each supply vent for central unit (at every fan setting for variable systems).*

## *Air Temperature & Relative Humidity*

Sensors for air temperature (MF52 thermistor) and relative humidity (HTM2500) were installed by SMT in the airstream of the return and supply louvers of each ductless mini split heat pump indoor unit (*Figure 2.5* and *Figure 2.6*). For central systems, sensors were located within the ductwork (see *Figure 2.7*). These parameters were monitored to determine the energy provided or removed by the heat pump. Data was wirelessly transmitted to the Data Gateway and retrieved via a cloud-based transfer. Outdoor ambient air temperature and relative humidity were also monitored on-site to correlate the efficiency of the units with corresponding outdoor conditions.

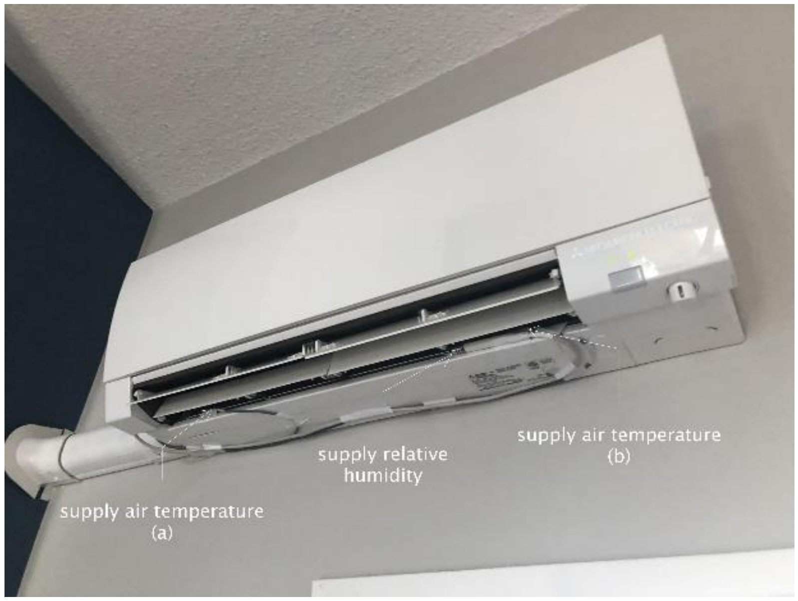


*Figure 2.5 – indoor unit with temperature and relative humidity sensors at supply louver (bottom of unit).*

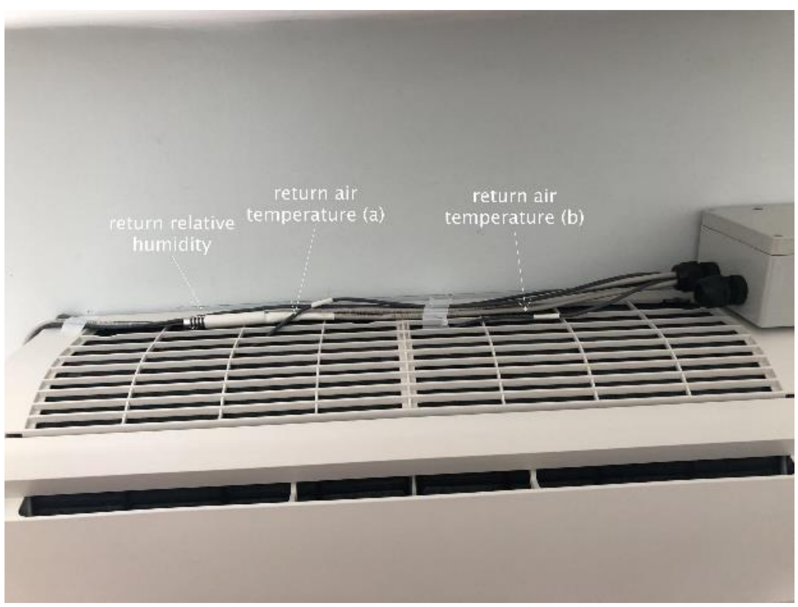


*Figure 2.6 - indoor unit with temperature and relative humidity sensors at return louver (top of unit).*

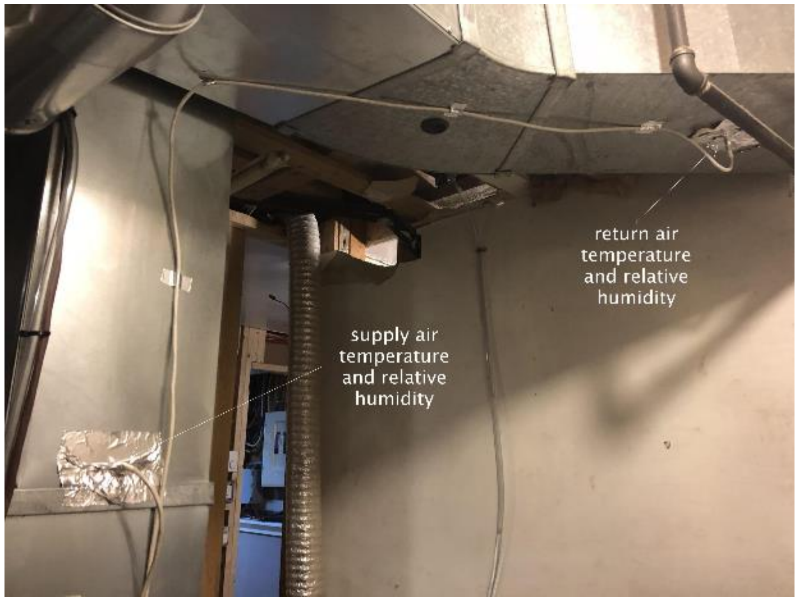


*Figure 2.7 – air temperature and relative humidity sensor locations for central system unit.*

## 2.3 Field Data Analysis

Data collected in the field were used to estimate heating and cooling capacity of heat pumps and resulting performance metrics. This section describes the methodologies and assumptions used to conduct the analysis.

### *Heating & Cooling Cycles*

For the analysis, a heating or cooling cycle was defined as a period when both the fan and the compressor were in operation. This section explains the rationale for this definition and adjustment made to ensure that heating and cooling cycles were precisely captured.

In some cases, heat pumps appeared to be conditioning the air with no apparent consumption in power. This phenomenon was found to occur if there was even a slight difference between return and supply air temperature, for example, due to the buoyancy of air. Occasionally, the opposite was also found where the indoor unit was consuming power but hardly providing any heating or cooling to the space (less than 1˚C). This instance was found to occur when the unit was set to the fan-only mode, which is available for the purpose of satisfying ventilation needs of the occupant without activating the refrigerant cycle. Technically, energy output could be calculated during these instances, though the results would not accurately represent the actual heating or cooling performance of the system since the compressor is not operating. Therefore, to account for these moments, heating and cooling parameters were defined which excluded data that did not satisfying a minimum change in temperature between the measured return and supply air temperatures. These temperature differences were typically set at 1˚C for ductless mini split units and 5˚C for central systems.

It was also found that the placement of the return temperature sensors can have a significant impact on the measured data. *Figure 2.8* is an example of supply and return temperature measurements for a ductless mini split unit. The plot shows that after each heating cycle, a sharp spike in return temperature occurs as residual heat from the coil rises due to the buoyancy of the warmer air.

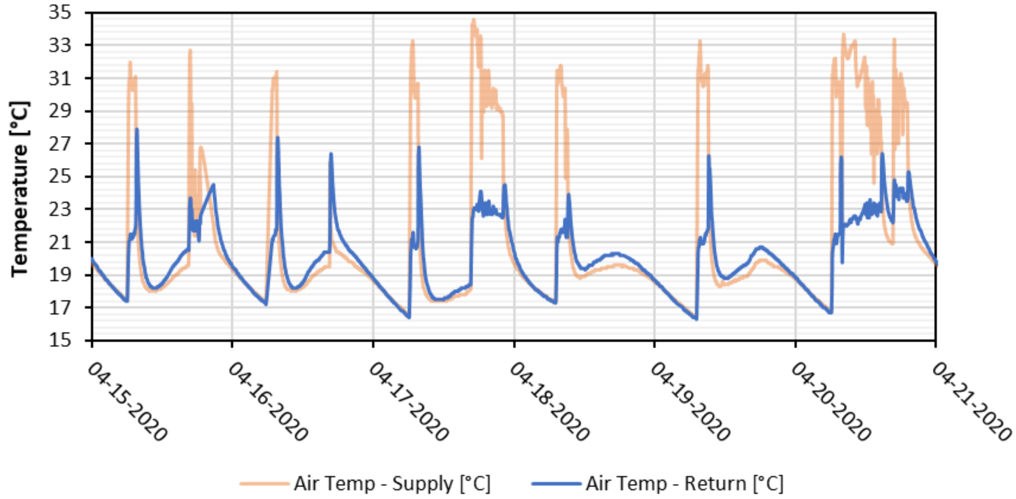


*Figure 2.8 – example of return temperature rise at the end of each heating cycle suggesting residual heat rising from coil once the compressor and fan are turned off (VIC05 – ductless mini split).*

A single heating cycle is plotted in *Figure 2.9* which demonstrates how supply temperature rapidly rises over a five-minute period as the compressor and fan are turned on. The return temperature, indicative of ambient indoor air, begins to rise and becomes stable throughout the cycle. Once the compressor and fan turn off, residual heat radiating from the coil begins to rise since the fan is no longer directing it through the indoor head. Because the compressor and fan are not operating outside of the gold box annotated in the plot, the data outside that range is not included in the COP calculation. In addition, the residual heat causes a short period where the return heat is higher than the supply and therefore the equations used would erroneously assume that cooling is occurring after every heating cycle. Note that the extra heat provided to the space, in theory, could be measured, though was not included in this study as not all return sensors captured this residual heating effect.

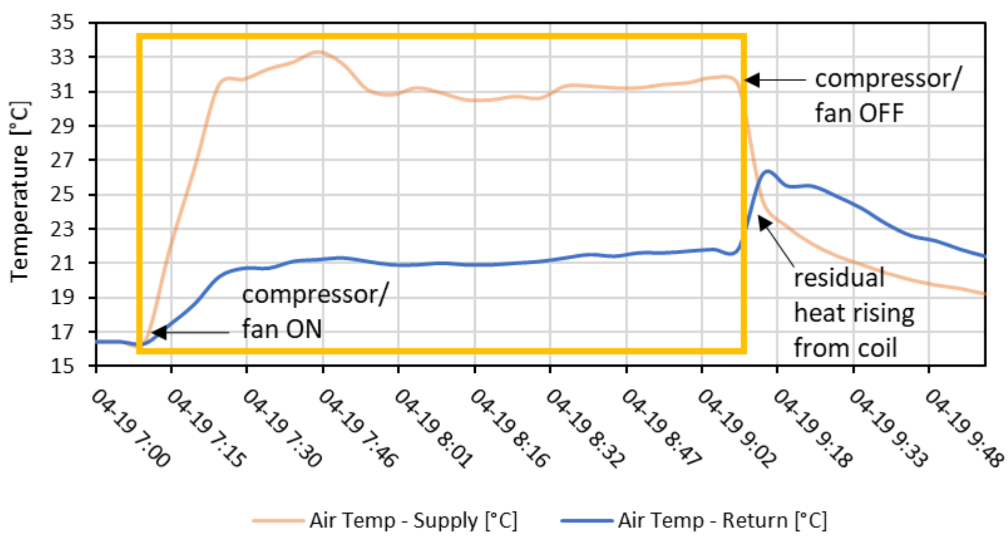


*Figure 2.9 – example of a typical heating cycle. Note how the return temperature rises as the supply temperature falls, suggesting residual heat from the coil is rising after the compressor and fan have been turned off. The gold box is the measurement period for calculating the efficiency of the cycle i.e., excludes residual heat when the compressor and fan are off during this time (VIC05 – ductless mini split).*

It is assumed in the *Figure 2.9* example that the compressor and fan have been turned off since the heat is beginning to rise to the return sensor. However, if the fan were to turn on, a false cooling COP could potentially be calculated during any period where the return temperature is greater than supply (see plotted temperatures after the gold box in *Figure 2.9*). To further ensure that the measurement periods used to calculate efficiencies did not include anything outside of the *gold box* in the figure above, the fan-only mode was excluded by ignoring any measurements at or below a typical fan-only system consumption. Therefore, typical fan consumption was determined for each unit and only cases when system energy demand exceeded fan consumption (i.e., compressor on) were included. For future studies, it is recommended that return temperature sensors be placed at a distance far enough from the return louvers as to not capture residual heat from the rising coil. Also, an additional temperature sensor at the indoor thermostat is also recommended for a better understanding of interior set points.

## *Heating & Cooling Capacity*

The heating and cooling capacity, or energy output from the return to supply airstream is calculated using their respective psychrometric properties, which were derived from the measured dry-bulb temperatures and relative humidity. The temperature difference between return and supply establishes the amount of sensible heat being supplied or removed at the indoor unit as air moves across the supply coil. Then, as discussed in Section 2.1, the sub-metered fan energy is used as a proxy to estimate the volume of air passing through the indoor unit. The added or removed energy is thus a function of both the difference in temperature between return and supply air and the volume of air passing through the system.

Because heat pump systems rely on convective heat transfer of forced air, the mass will typically enter the unit (return air louver) as a mixture of air and water vapour and match the mass of the exiting mixture (supply air louver). In a cooling process, however, water vapour can condense out of the supplied air when its dry-bulb temperature reaches its dew-point temperature. This dehumidification process during cooling is shown below in *Figure 2.10*.

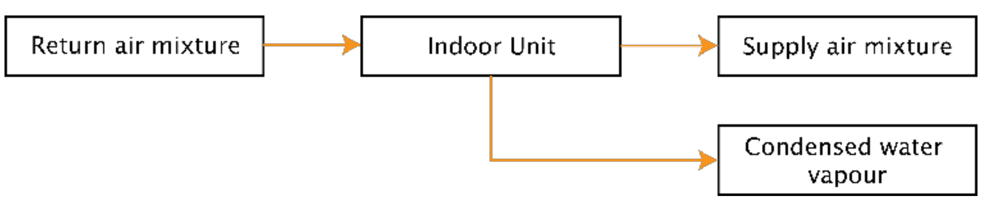


*Figure 2.10 – Mass flow through indoor unit under cooling and dehumidification.*

Latent energy generated by the phase change process of the return air from gas to liquid should, in theory, increase the effectiveness of the cooling process since a greater amount of energy is being removed via a condensate drain before it is exhausted as supply air. Typically, the latent heat energy contributed by the removed water vapour is considered negligible and ignored.[11] In this study, however, latent energy was included in heat pump

[11] CADMUS (2016). *Ductless Mini-Split Heat Pump Impact Evaluation*. Available online: http://ma-eeac.org/wordpress/wp-content/uploads/Ductless-Mini-Split-Heat-Pump-Impact-Evaluation.pdf

performance when the difference in measured conditions between return and supply air suggest that condensation has occurred.[12]

For example, when the return air is cooled by the coil to a temperature below the air's dew point, relatively less water vapour should be present in supply air. This process of cooling and dehumidification must satisfy a conservation of air mass and an energy balance between return and supply states. The mass and energy balance equation for this case is described below:

$$\Delta E = \dot{V}_2\rho_2[(h_1 - h_2) - (W_1 - W_2)h_{w2}]$$

Where subscript 1 refers to the return air, subscript 2 refers to the supply air, $V$ is the volumetric flow rate (m³/s), $\rho$ is the density of the moist air mixture (kg/m³), $h$ is the specific enthalpy of moist air (kJ/kg), $W$ is the humidity ratio (kg/kg), and $h_w$ is the specific enthalpy of condensed water (kJ/kg).

When measured conditions between return and supply air suggest that no condensation has occurred, the mass of the air entering the indoor unit (the return air) matches with the mass of the air exiting the indoor unit (the supply air), representing a sensible heating or cooling process. Note that a negative value will result when heat energy is removed, indicating a cooling process. The energy balance equation for this case is described below:

$$\Delta E = V_2\rho_2(h_2 - h_1)$$

Where subscript 1 refers to the return air, subscript 2 refers to the supply air, $V$ is the volumetric flow rate (m³/s), $\rho$ is the density of the moist air mixture (kg/m³), and $h$ is the specific enthalpy of moist air (kJ/kg).

To calculate the performance metrics in scenarios with multiple indoor units, the useful heat provided or removed by all indoor units was added together and divided by the input electrical energy of its respective outdoor unit.

## *Psychrometrics & Equipment Accuracy*

As described in the previous *Heating & Cooling Capacity* section, a mass and energy balance must be conserved through the conditioning process of the indoor unit. Therefore, in theory, it is possible to calculate an expected relative humidity of the exhaust air based on the measured relative humidity of the intake air. However, during an initial comparison between the measured and expected relative humidity, results suggested in some cases that the mass and energy balance were not conserved (i.e., expected did not match measured). This phenomenon is attributed to the accuracy of the instruments and affects the calculated performance of the studied heat pumps.

For the purposes of this study, it is important that the humidity ratio remains constant between supply and exhaust air, particularly in heating mode. For example, an error in relative humidity measurement that would falsely suggest that moisture has been removed during the heating process could result in a significantly lower COP, as this would imply that some moisture-related energy was removed.

To identify the inherent error of the measurements made in this study, the following methods were used to preserve the mass and energy balance:

[12] ASHRAE (2017). Fundamentals (SI Edition)

- → Calculate the measured partial vapour pressure of the return and supply states of air using the measured dry-bulb temperature and relative humidity
- → Calculate the expected partial vapour pressure of the supply state of air by equating it to the measured specific vapour pressure of the return state.
- → Calculate the expected relative humidity of the supply state of air knowing the expected partial vapour pressure.
- → Check the agreement between the expected and measured relative humidity of the supply state of air.

The methodology used to evaluate agreement between the expected and measured values of relative humidity was to compare the range of uncertainty via the combination of errors in quadrature, also known as the square root of the sum of squares.[13] Listed accuracy for the instruments allowed the computation of uncertainty ranges for each measurement and their calculated derivatives. The agreement between the expected and measured values was evaluated based on the propagated error of both calculated values and is related to the accuracy of the instruments used. This process is illustrated in *Figure 2.11*. Although this may exclude some data points, this technique reduced the variability of the calculated heat pump performance from the overall sample of collected data.

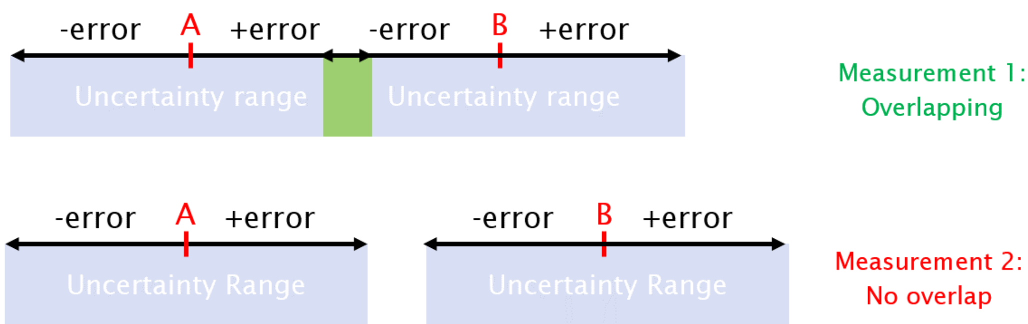


*Figure 2.11 – examples of propagated error between two calculated values. In this example, Measurement 1 would be accepted as its error band (B) overlaps with expected result's error band (A). Measurement 2 error band (B) does not overlap with expected error band (A) and therefore would be rejected.*

## *Defrost Cycles & Backup Heating*

As discussed in Section 1.1, the intent of a defrost cycle is to heat the outdoor coil to thaw and prevent the coil from excessive ice build-up. The defrosting strategy is to temporarily reverse the refrigerant path from heating mode to cooling mode, which provides heat to the outdoor coil in order to melt any ice accumulation. Because ductless units are not typically equipped with backup heating, sporadic periods of cooling during the winter were clearly identifiable for some units (see *Figure 2.12* and *Figure 2.13*). The energy consumption associated with the defrost cycle was therefore estimated by isolating these sharp decreases in supply temperature (i.e., active cooling) during the heating season. [14]

[13] Wolfram (2019) Experimental Data Analyst Documentation. Available online: https://reference.wolfram.com/applications/eda/ExperimentalErrorsAndErrorAnalysis.html

[14] Johnson, R.K. *Measured Performance of a Low Temperature Air Source Heat Pump*. Prepared for U.S. Department of Energy. September 2013.

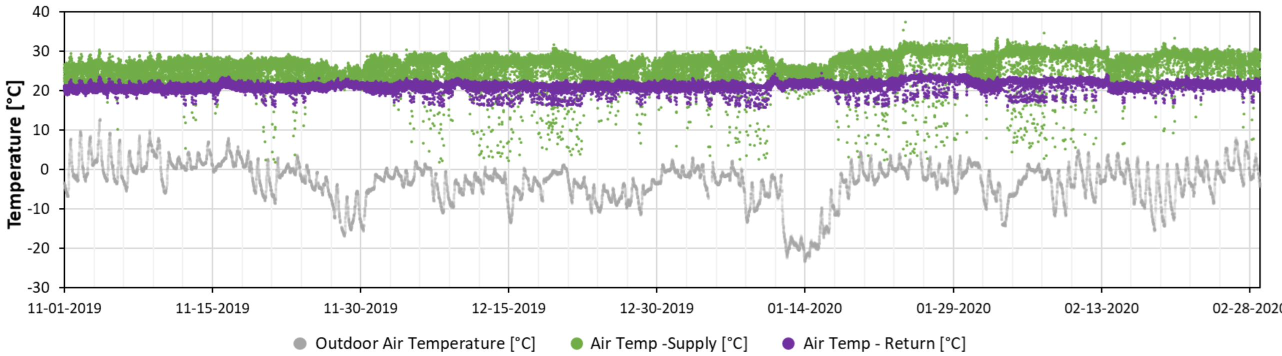


*Figure 2.12 – example of sporadic periods of cooling (green dots well below the purple return air temps) during winter season (Nov 1 – Mar 1) as a result of defrost cycle for a ductless mini split system (PRI01ii – ductless mini split).*

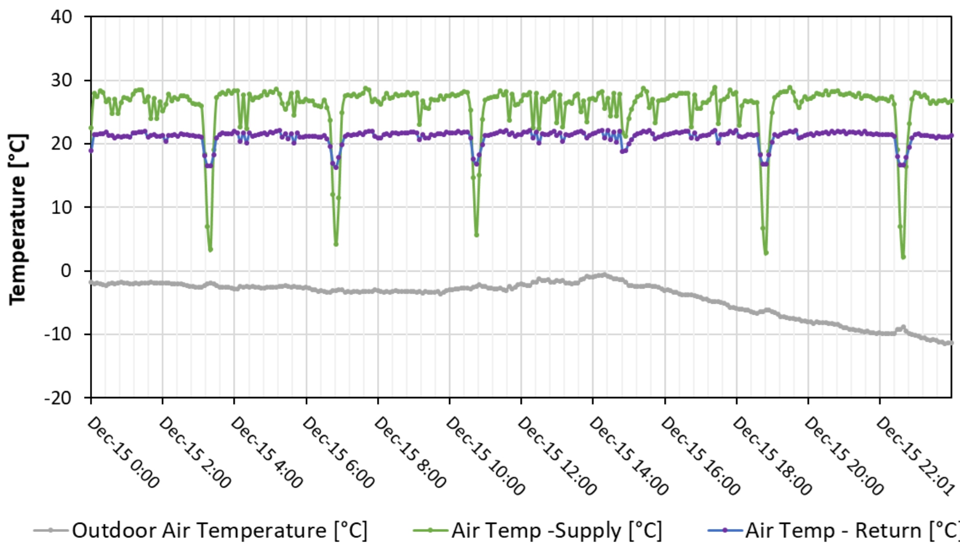


*Figure 2.13 – example of several clear defrost cycles during a 24-hour period during winter season for a ductless mini split system (PRI01ii – ductless mini split).*

Central heat pump systems are often equipped with backup heating for both backup heat and to provide continuous heating while the heat pump is in a defrost cycle. For central systems, the backup heating energy (if delivered by an electric resistance heating coil within the ducted system) was metered by the current transformer on the same circuit measuring the fan (i.e., the circuit of the furnace). For example, *Figure 2.14* is an example of a central heat pump system operating for a sample week in heating season. The blue band is the typical heat pump system consumption (i.e., compressor and electronics) and the orange bands are sub-metered fan and backup heat consumption. Note that the fan is consistently consuming roughly 15 Watt-hours (Wh) over each five-minute interval, though when the backup heat is activated, the consumption typically exceeds the heat pump consumption. It was therefore possible to sum the total backup heat consumption for a given period by adding all of the sub-metered energy above the fan consumption (in this case approximately 15Wh). Calculating backup heating (within the ducted system) for central heat pump systems with variable speed fans uses the same methodology as each fan speed has its own distinct energy consumption band.

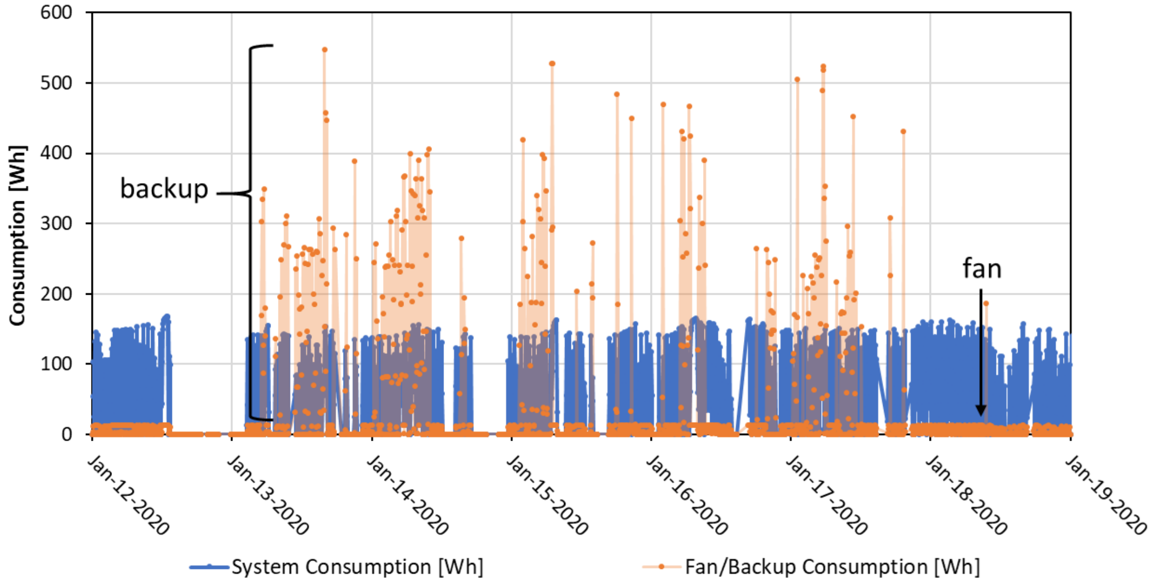


*Figure 2.14 – example of central heat pump system consumption (blue) and fan/backup consumption (orange) for a sample week in heating season. Note that the fan is consistently consuming approximately 15Wh, whereas the backup heating typically exceeds the heat pump system consumption (VIC08 – central single stage).*

## 2.4 Energy Savings Analysis

Electric and gas utility bill data (where applicable) was provided by BC Hydro and FortisBC for each of the participant buildings. These data contained monthly whole-building electric and gas energy consumption values for at least one year before heat pump installation as well as for the one-year post-installation monitoring period.

Baseline energy consumption during the measurement period was estimated by adapting the pre-heat pump installation consumption data to the weather conditions of the measurement period. This was done by establishing the quantifiable trend between energy consumption and heating degree day values (HDD) during the pre-heat pump installation data period. The resulting polynomial fit of the curve was then used to predict energy consumption at any HDD within the regression. Using the monthly HDD values during the measurement period, a reporting period baseline energy consumption was obtained for most participant buildings. The majority of buildings used a variety of fuel sources to heat their homes, both before and after the heat pump installation. For buildings that also utilized non-utility provided energy for heating (e.g., propane or wood fireplaces) a full picture of whole home energy consumption was not available.

The estimated baseline electricity and natural gas consumption was then compared to consumption after the heat pump was installed to estimate potential energy and cost savings. The difference between the estimated baseline and measured utility data post-heat pump installation reflects the whole home energy impacts of the heat pump. This analysis assumes that building characteristics other than the installation of the heat pump and removal of the previous space conditioning equipment (e.g. building enclosure performance, occupancy) do not significantly change, unless noted as a static factor adjustment.

The estimated cost and greenhouse gas (GHG) emission savings from the whole building energy analysis were calculated for each participating home.

Current utility rates and emission factors were used for the analysis, as follows:

- BC Hydro[15]
  - 0.0935 $/kWh (for first 1,350 kWh in an average two-month billing period)
  - 0.1403 $/kWh (over 1,350 kWh)
- FortisBC[16]
  - Electricity
    - 0.10799 $/kWh (for first 1,600 kWh in an average two-month billing period)
    - 0.14320 $/kWh (over 1,600 kWh)
  - Natural Gas
    - 9.150 $/GJ (0.0329 $/kWh) total delivered commodity cost
      - 4.596 $/GJ (0.0165 $/kWh) delivery charge
      - 1.019 $/GJ (0.0037 $/kWh) storage charge
      - 1.549 $/GJ (0.0056 $/kWh) cost of gas
      - 1.986 $/GJ (0.0072 $/kWh) carbon tax
- City of Vancouver Energy Modelling Guidelines[17]
  - Electricity
    - 0.011kg $CO_2$e/kWh emission factor
  - Natural Gas
    - 0.185kg $CO_2$e/kWh emission factor

[15] https://app.bchydro.com/accounts-billing/rates-energy-use/electricity-rates/residential-rates.html (accessed May, 2020).

[16] https://www.fortisbc.com/about-us/regulatory-affairs/our-electricity-utility/electric-bcuc-submissions/electricity-rates (accessed May, 2020).

[17] https://vancouver.ca/files/cov/guidelines-energy-modelling.pdf (accessed May, 2020).

# 3 Results & Discussion

## 3.1 General Information

The intent of the study was to understand how heat pumps perform in-situ under a variety of different outdoor environmental conditions. *Figure 3.1* is a plot of the average outdoor temperature of each region based on measurements taken at each site. The total cooling degree days (CDD) and heating degree days (HDD) based on site measurements were also calculated and results are presented in TABLE 2.

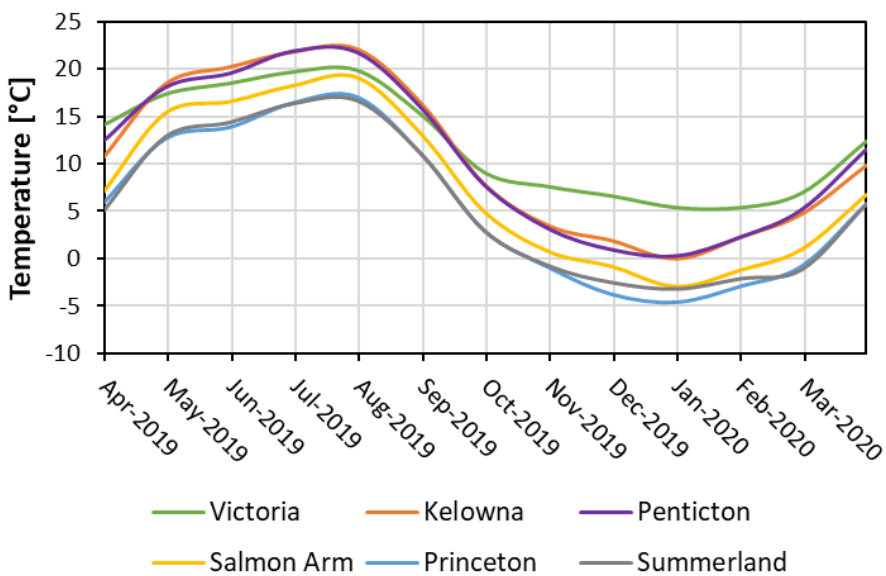


*Figure 3.1 – average temperature of each participant location for the monitoring period based on site measurements.*

| TABLE 2 – MEASURED COOLING & HEATING DEGREE DAYS FOR MONITORING PERIOD | | | |
|---|---|---|---|
| **Location** | **CDD Measured** | **HDD Measured** | **HDD BCBC 2018** |
| Victoria | 204 | 2516 | 2650 |
| Penticton | 359 | 3028 | 3350 |
| Kelowna | 394 | 3247 | 3400 |
| Summerland | 127 | 4024 | 3350 |
| Princeton | 146 | 4129 | 4250 |
| Salmon Arm | 125 | 4262 | 3650 |

Building and participant information was collected during an initial equipment instrumentation site visit. All buildings that participated in this study were defined as single-family detached homes of various ages and geometries.

The majority of the buildings were pre-1990 with various types of heating systems prior to the installation of a heat pump system. It was noted that all homes are equipped with backup heating systems such as electric resistance baseboards and gas or wood fireplaces. The average floor area of the buildings was measured at 145m² (1560ft²). *Figure 3.2* represents a general distribution of participant and building information.

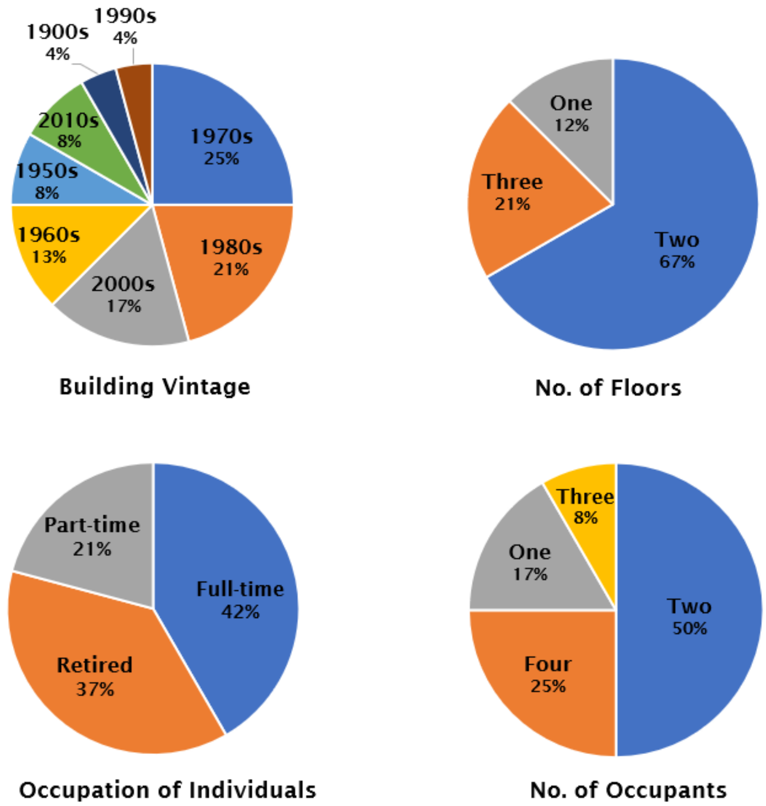


*Figure 3.2 – general distribution of participant and building information.*

Compared to controlled laboratory measurements, field monitoring studies occasionally experience conditions that could compromise the validity of measured data. Due to the nature of field monitoring, some results were found to be suspicious and may have affect the estimated performance of some heat pump units. Therefore, erroneous data largely associated with instrument installation ultimately led to some sites being excluded from the overall results. TABLE 3 is a list of excluded heat pumps from the analysis results, which also includes one case of voluntary removal and a heat pump that was rarely operated.[18] These cases do, however, provide insight into the intricacies of in-situ heat pump monitoring and some of these lessons learned are provided in Section 3.5.

| TABLE 3 – SITES EXCLUDED FROM OVERALL DATA ANALYSIS | |
|---|---|
| **ID** | **Description** |
| KEL02 | Participant requested removal from study |
| KEL05 | Relative humidity propagated error method resulted in fewer than 50 hours for monitoring period – suggests supply RH sensor not properly reading RH in airstream |
| KEL07 | Supply and return temps vary significantly – resulting in low heat pump capacity (almost zero) – suggests supply sensors not placed in airstream |
| PRI01i | Very few data points after applying inclusion/exclusion criteria for analysis – suggests that supply temperature sensor did not accurately capture the supply airstream. |
| SAL01 | Supply and return temps roughly same for monitoring period – suggests supply sensors not placed in airstream |
| VIC07 | Supply and return temps vary significantly – resulting in low heat pump capacity (almost zero) – suggests supply sensors not placed in airstream |
| VIC09 | Heat pump operated less than 25 hours in both heat/cool |

[18] VIC09 unit was installed primarily as backup cooling for their 3rd floor bedroom. It was expressed by the participant during initial site visit that this unit would likely not be used unless under extreme conditions.

## 3.2 Heat Pump Performance

### 3.2.1 Volumetric Flow Rate

As described in the Section 2.2, the volumetric flow rate of each pump was measured at all fan settings and compared with manufacturer data sheets. In some cases, the measured volumetric flow rate (expressed in cubic feet per minute, or CFM) of ductless mini split units was similar to what the manufacturer's technical literature stated (see *Figure 3.3*). The majority of the CFM measurements, however, were lower than published manufacturer data. *Figure 3.4* is a more typical example of how the measured results compared to manufacturer data. Similar plots for all units can be found in Appendix A.

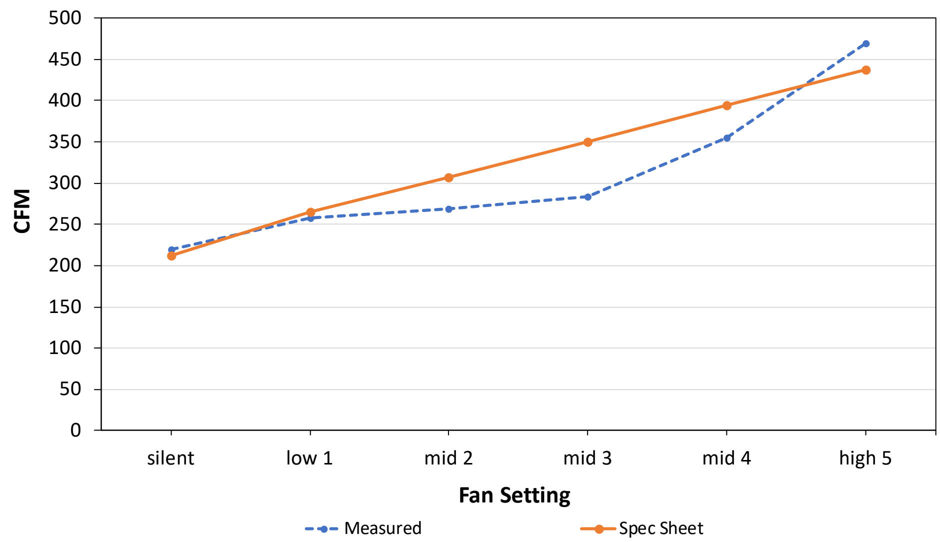


*Figure 3.3 – measured volumetric flow rate of a variable speed mini split heat pump with generally strong agreement (93%) with manufacturer data (KEL03 – ductless mini split). Silent is a setting on most mini split heat pumps which means very low CFM and little heating or cooling*

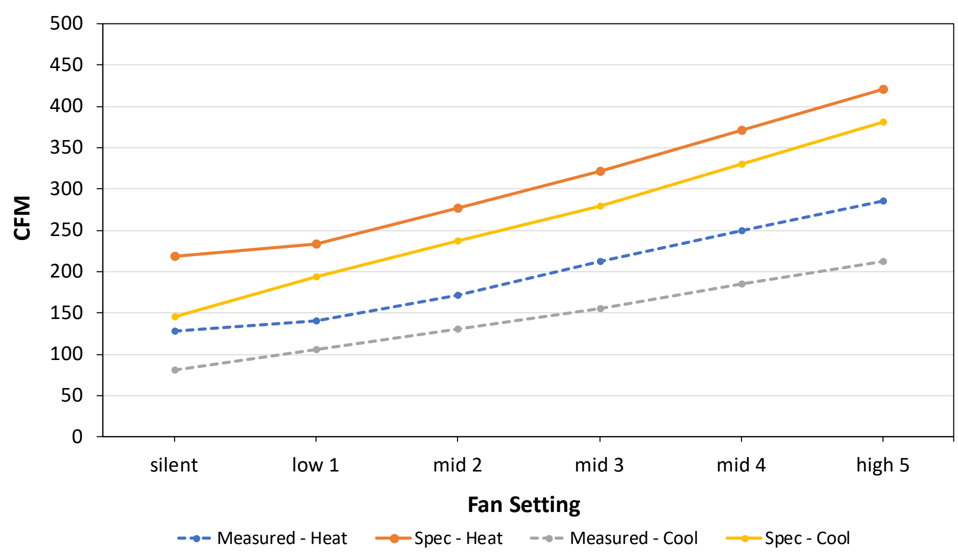


*Figure 3.4 – measured volumetric flow rate of a variable speed mini split heat pump with generally poor agreement (65% heating and 55% cooling) with manufacturer data (VIC02 – ductless mini split). Silent is a setting on most mini split heat pumps which means very low CFM and little heating or cooling*

Lower than rated volumetric flow rates for ductless mini split units are potentially due to lab testing methodologies that typically do not include back pressure caused by the presence of the supply louvers. It was also found that many of the mini split indoor heads were installed with limited ceiling clearance, which could be restricting the flow of air at the return airstream intake. Research into the manufacturer specified clearances shows a listed minimum clearance range between 3.9" (100mm) and 1.2" (30mm).[19] Based on the specified minimum clearances, some of the units were in fact within the acceptable listed range; however, it appears that low ceiling clearances could be negatively affecting the volumetric flow rate of the indoor units. *Figure 3.5* shows some photos of mini split indoor head locations with a variety of different ceiling clearances and installation locations relative to walls and ceilings.

*Figure 3.5 – various ductless mini split indoor heads with limited ceiling clearance.*

In addition to the potential flow rate issues associated with limited ceiling clearance, indoor heads mounted too closely to the ceiling can run the risk of prematurely recirculating supply air. *Figure 3.6* is a plot of a sample heating period where warm air supply air appears to be prematurely recirculating into the return air intake, potentially prompting what is interpreted as system short-cycling since each data point represents a five-minute interval. Future research investigating how ceiling clearance affects heat pump performance is encouraged.

[19]Mitsubishi and Fujitsu listed greater clearances (2.4" – 3.4") whereas Daikin listed either 1.2" or no listing.

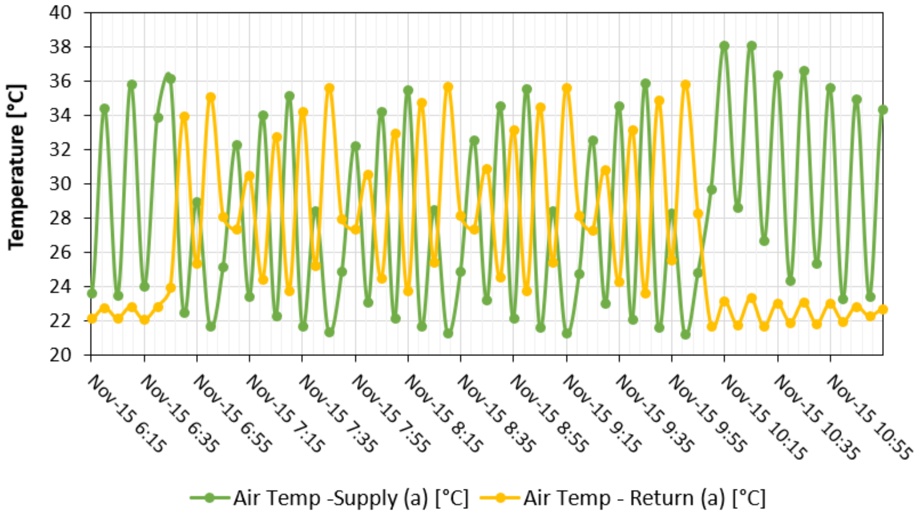


*Figure 3.6 – sample heating period where warm supply air appears to be prematurely recirculating into the exhaust air intake as a result of the indoor head being installed with limited ceiling clearance.*

The measured volumetric flow rates of the central ducted systems were found to be closer to their rated CFM, for most cases. One unit, however, was found to be significant less (48%) than the rated CFM, though it appears that for this case that the comparably lower CFM is due to a seemingly optimistic flow rating, given that this level of CFM was not achieved by any central system in this study. The measured central single stage systems were found to deliver an average volumetric flow rate of 670 CFM and variable speed systems in the range between 485 and 670 CFM, with a maximum of roughly 830 CFM for both types. Ductless mini splits were found to deliver between 129 and 290 CFM, with a maximum of 600 CFM in one case. Figure 3.7 shows this significantly lower than rated CFM example (left bar chart) compared to a more typical measurement scenario (right bar chart). It is important to also note that the rated CFM is typically measured across the unit itself whereas the CFM of central systems was measured after the air had travelled through existing (likely leaky) ductwork, and as such, some air had likely escaped the ductwork prior to reaching the exhaust vents.

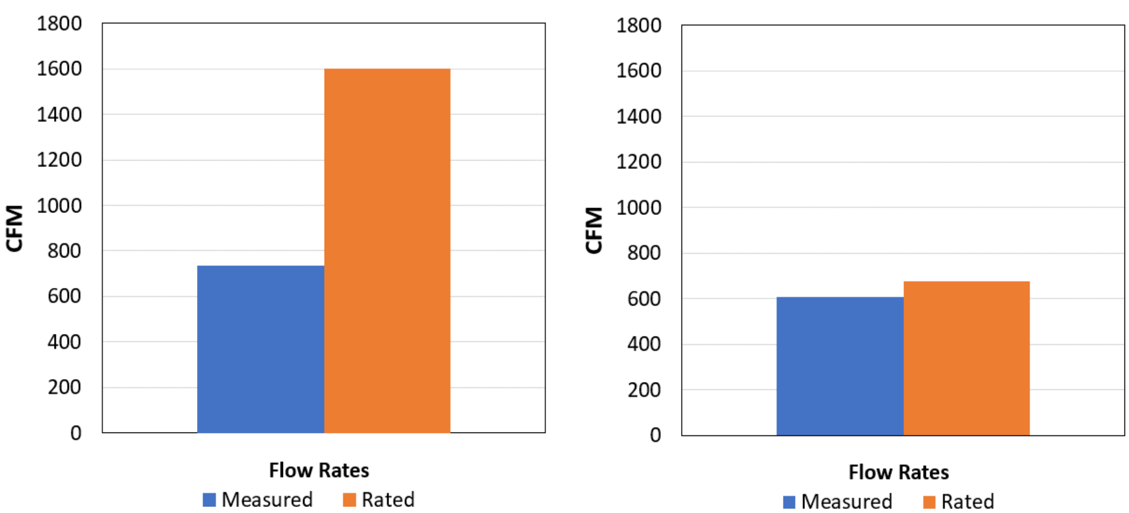


*Figure 3.7 – measured vs. rated volumetric flow rate for two central single stage systems: a case with poor agreement (KEL04, left) and another with generally strong agreement (KEL05, right).*

TABLE 4 lists the percent difference between measured volumetric flow rates and manufacturer data. Note that some data sheets express different flow rates depending on when the unit is in heating or in cooling mode. Per the table, the average measured volumetric flow rates were 64% of (or 36% lower than) the flow rates listed in manufacturer data sheets. The lower measured results compared to listed flow rates are consistent with previous studies. The DOE, for example, reported that measured flow rates were typically 50% to 80% of rates listed by manufacturers.[20] Results from the table below also show that the measured flow rates of central systems are generally closer to manufacturer data than those of ductless mini split systems.

TABLE 4 – DIFFERENCE BETWEEN MEASURED VOLUMETRIC FLOW RATES AND MANUFACTURER DATA SHEETS.

| **Volumetric Flow** **% measured vs. data sheets** | **Overall** | **Heating Mode** | **Cooling Mode** | **Central Systems** | **Mini Split Systems** |
|---|---|---|---|---|---|
| Max | **100%** | 79% | 93% | 100% | 93% |
| Average | **64%** | 56% | 58% | 72% | 61% |
| Min | **22%** | 20% | 24% | 46% | 22% |

Generally, greater flow rates are required to force air through a system of ductwork. Because fan energy is relatively small compared to the compressor energy of a heat pump, central systems in principle are predisposed to operate at higher efficiencies, given that a high volumetric flow rate is important to achieve a high COP. In other words, a relatively small increase in fan power translates to a larger increase in flow and hence a larger volume of heated or cooled air is distributed to the space. In theory, ductless mini systems could also be equipped with higher powered fans, although this would have an impact on the acoustical performance of the indoor head and comfort within the zone. Because mini split systems are often located directly in common rooms, a balance between fan flow rate, air velocity, and acoustical comfort must be met. A central system is typically located in place of the existing furnace, which is usually remote from the areas of the home it is conditioning and can thus accommodate more noise and higher air volume.

### 3.2.2 Cooling Season Performance

The cooling season analysis is based on all active cooling data measured during the monitoring period, rather than a specified time period, though cooling mainly occurred during the summer and shoulder seasons.

Based on the monitored data, it was found that five participants rarely used active cooling during the shoulder and cooling season (fewer than 25 hours, the majority of which were in Victoria), and were therefore not included in the cooling season analysis. The overall range of cooling COP values, which includes data from all participants using active cooling across their respective range of outdoor temperatures is shown in *Figure 3.8*. The minimum and maximum COP lines represent the heat pump with the lowest efficiency and the heat pump with the highest efficiency, respectively, while the mid COP line is the overall average of all the measured units.

[20] Williamson, James and Robb Aldrich. *Field Performance of Inverter-Drive Heat Pumps in Cold Climates*. Prepared for U.S. Department of Energy. August 2015.

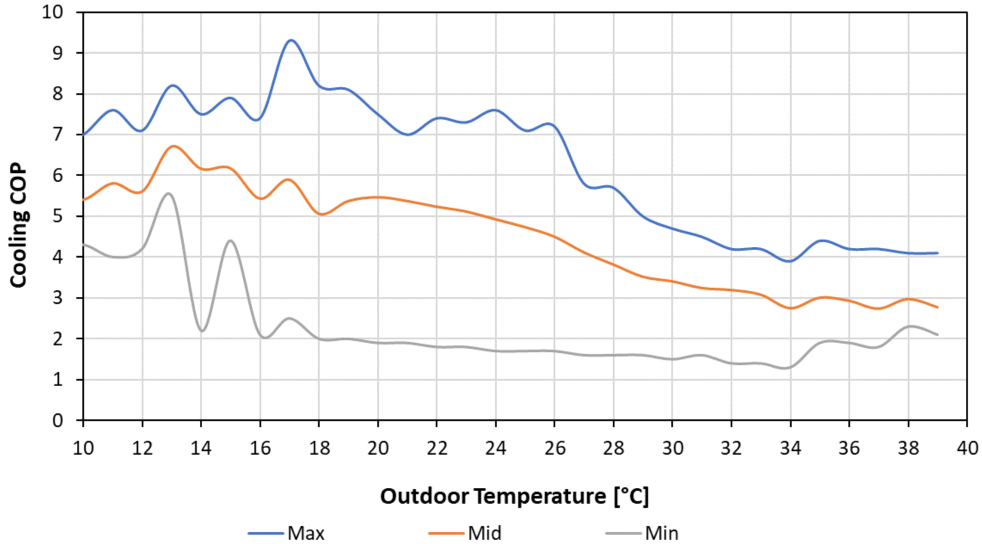


*Figure 3.8 – average cooling COP range of all heat pumps vs. outdoor temperature.*

The results show that that participants are using active cooling across a wide range of outdoor temperatures, and that average cooling COP values as high as 9.3 were achieved, though during moderate cooling conditions. Note that measured outdoor temperatures of 40˚C are likely a result of direct sunlight hitting the monitoring equipment box. However, the equipment boxes are typically mounted next to the outdoor unit of the heat pump, and therefore the outdoor unit is likely experiencing similar temperatures. This is relevant when considering that the outdoor coil must effectively transfer the heat collected from the interior to the outdoor environment. In theory, higher ambient temperatures around the outdoor unit could reduce heat transfer from the coil to the outdoors relative to a cooler ambient outdoor environment.

It is evident from the figure above that participants are using heat pumps to cool the interior when outdoor temperatures are below typical interior temperatures (~21˚C). Passive cooling measures (i.e., opening windows to promote natural ventilation) could be an effective way of conditioning the interior when the outdoor temperature is below the interior temperature, though the data suggests that people typically are not passively cooling. Note that when the outdoor temperature is below the typical interior temperature, the indoor environment could be significantly warmer due to high solar heat gains, for example. This also seems to occur largely during shoulder season when heat pumps are set to cool during warm days. As the outdoor temperature drops during the evening and at night, the interior temperature may remain relatively warm due to factors like stored daytime heat, and therefore the system remains in the cooling mode even though outdoor temperatures are below interior temperatures. Also, some heat pumps require a manual switch between heating and cooling, which may explain why some homes continue to cool throughout the night when outdoor temperatures are typically at their lowest. For example, only if ductless units are set to automatic mode will the system switch between heating and cooling. Alternatively, central systems must be manually changed over.

The overall cooling COP was evaluated according to system type: ductless mini split systems (*Figure 3.9*), central systems with variable speed compressors (*Figure 3.10*) and central systems with single stage compressors (*Figure 3.11*). Of the two central variable systems, one of the units experienced significant data loss in the cooling season. Generally, the average cooling efficiency of all systems types appears to be similar. Also, the ductless mini split units appear to be operating at a wider range of temperatures. In all, the average COP ranges, including the relatively poorer operating systems ("min" lines) are still operating above a COP of 1 across the measured range of outdoor temperatures.

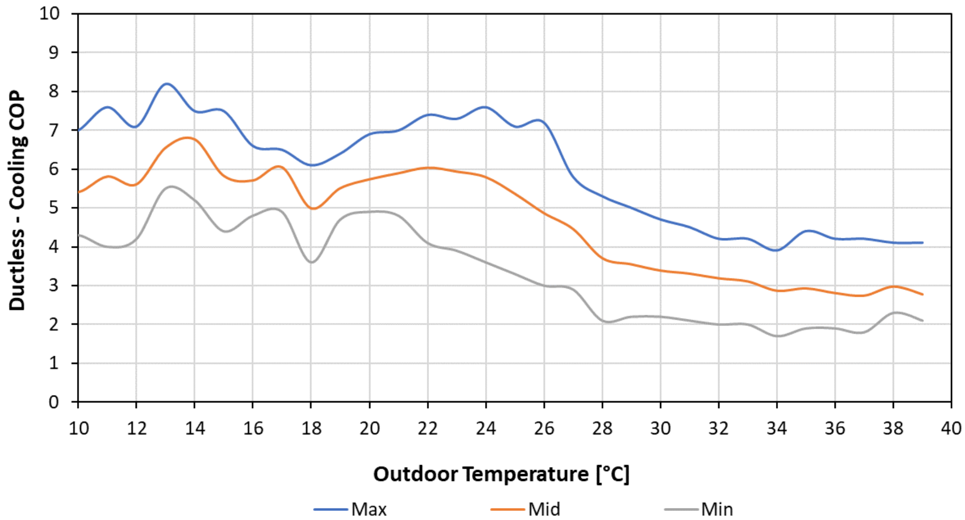


*Figure 3.9 – average cooling COP range vs. outdoor temperature for all ductless mini split systems.*

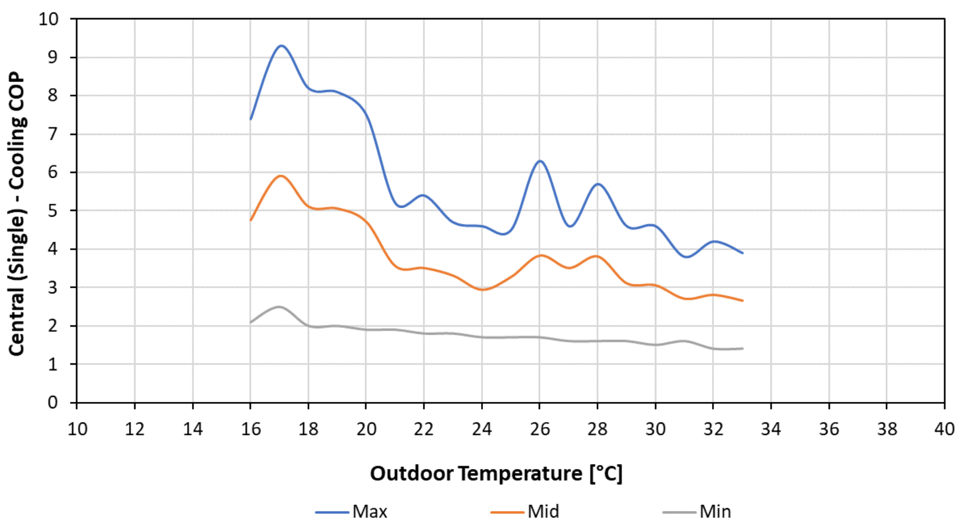


*Figure 3.10 – average cooling COP range vs. outdoor temperature for all central systems with single speed compressors.*

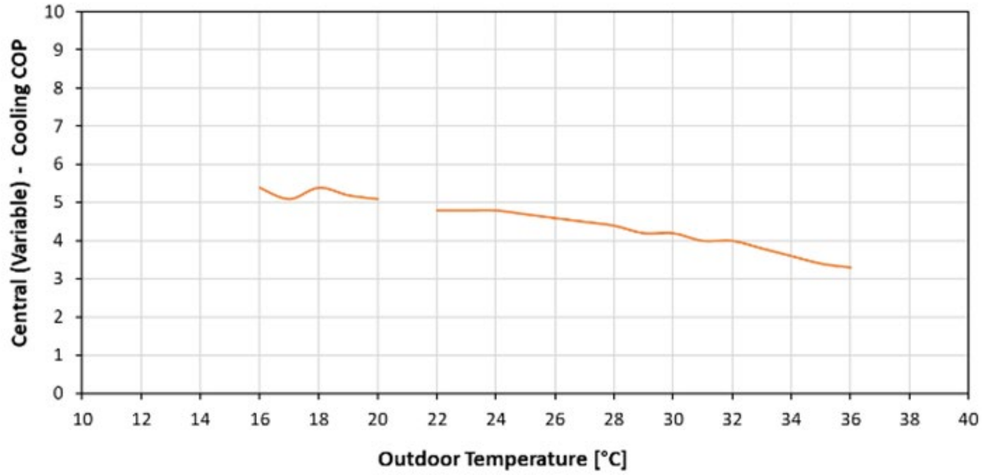


*Figure 3.11 – average cooling COP vs. outdoor temperature for the central system with variable speed compressor.*

To gain some insight into how the heat pump systems are operating on a daily basis, *Figure 3.12* is a sample of temperature measurements and corresponding cooling COP values for a single site, plotted over a warm four-day period in June 2019. *Figure 3.13* is a magnified plot to exhibit what is occurring over the course of two days instead of four. Based on the plots, it appears that this particular heat pump is maintaining a steady indoor temperature (i.e., return temperature) of roughly 21˚C, despite daily outdoor temperatures reaching above 30˚C. A diurnal COP trend, which is correlating well with outdoor temperature is also apparent; however, it appears that the COP consistently drops when outdoor temperature is below the average interior temperature.

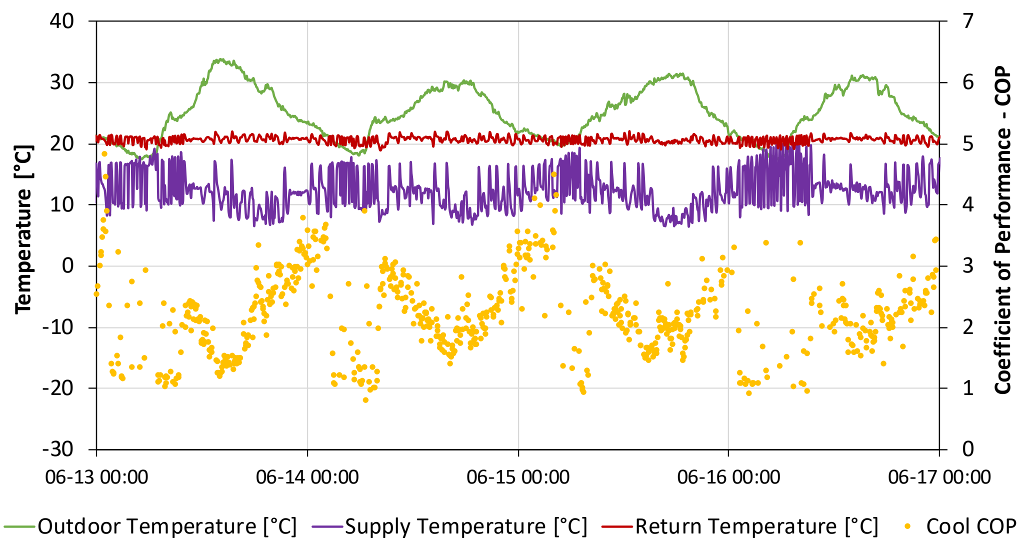


*Figure 3.12 – sample daily cooling COP and outdoor, return and supply temperatures for four days in summer 2019 (KEL01b – ductless mini split).*

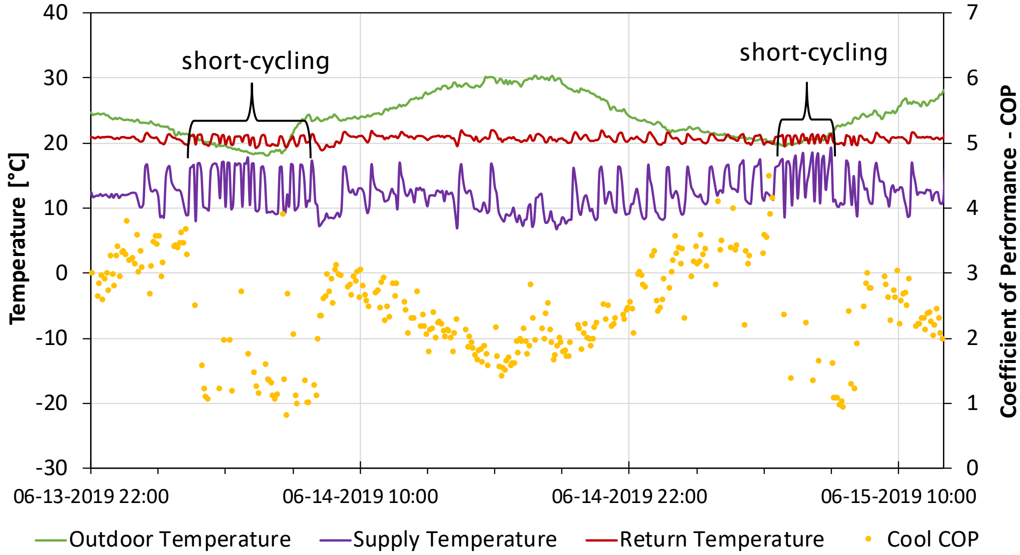


*Figure 3.13 – sample daily cooling COP and outdoor, return and supply temperatures for two days in summer 2019 (KEL01b – ductless system).*

The magnified graph shows that when the outdoor temperature is below the interior temperature, the heat pump cooling cycle begins to operate for shorter periods of time compared to when the outdoor temperature is above interior ambient conditions, suggesting some short-cycling is occurring. As described in Section 1.1, shorter operating cycles can reduce the efficiency of the heat pump, which is evident from the data results presented above.

It was found that this phenomenon is only occurring with ductless mini split units, though not for all units. Analysis showed that 4 of 7 (57%) of the mini split units that exhibited significant active cooling were experiencing fluctuations and reductions in overall COP when units were operated below the average indoor temperature. For example, *Figure 3.14* is a plot of the average COP across the monitored outdoor temperature range. Note that this figure is for the same participant case shown in *Figure 3.12* and *Figure 3.13* above. The plot shows clearly that there is a significant fluctuation in COP at temperatures below the average interior temperature. *Figure 3.15* though is an example of a unit where the COP does not appear to be affected by outdoor temperatures below the average interior temperature. Therefore, future research focused on this phenomenon as an isolated variable is encouraged.

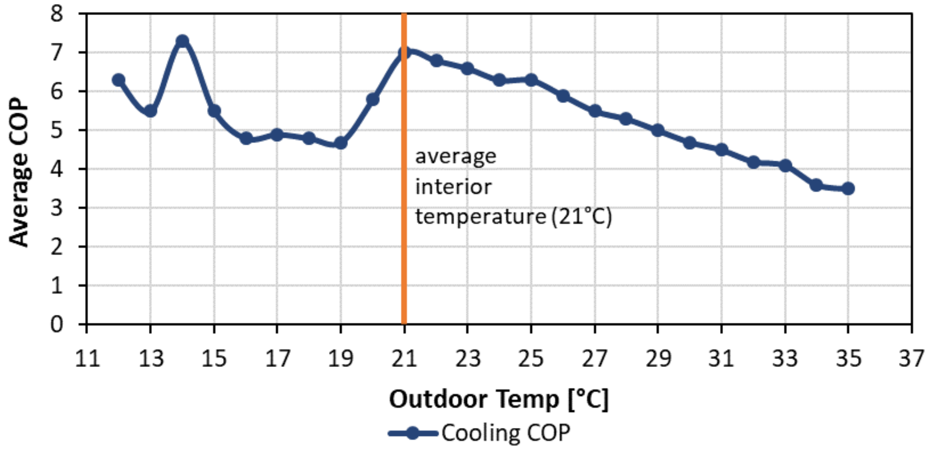


*Figure 3.14 – example of significant variability in COP at temperatures below the average interior temperature (KEL01b – ductless mini split).*

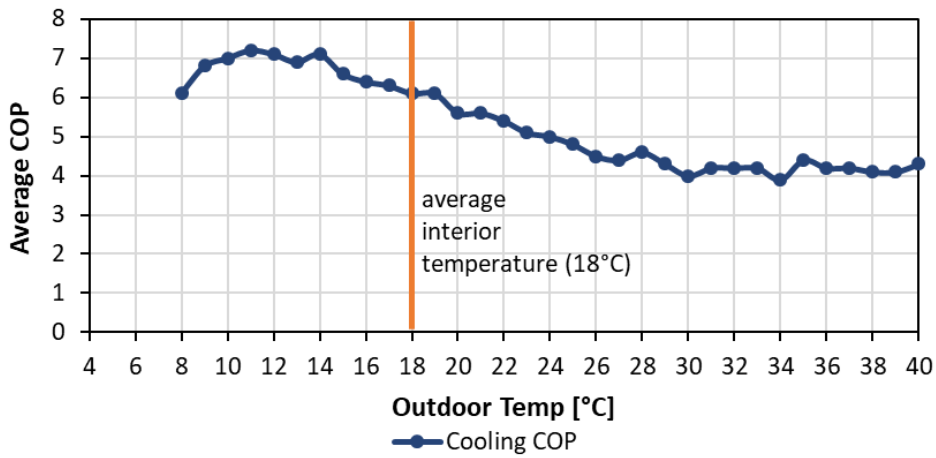


*Figure 3.15 – example of no variability in COP at temperatures below the average interior temperature (VIC02 – ductless mini split).*

## *Seasonal Cooling Efficiency*

The seasonal efficiency metrics of units are often considered an important factor when comparing different heat pump manufacturers and units. However, it is difficult to compare the estimated in-situ seasonal performance against manufacturer ratings, or to compare in-situ seasonal performance of one heat pump to another, without considering other variables that affect seasonal performance. Some of the many variables found in this study include refrigerant line/duct length, indoor head ceiling clearance, building vintage, occupant behaviour, and average outdoor temperature during heat pump operation.

TABLE 5 is a summary of the estimated cooling season efficiencies of each unit expressed as seasonal COP and SEER. The seasonal cooling COP was determined by averaging all COP values throughout the monitored cooling and shoulder seasons. This was performed for all participants who actively cooled their homes with a heat pump for 25 hours or more throughout the monitoring period. The estimated seasonal efficiency (SEER) was determined based on seasonal cooling COP using the multiplication factor described in Section 1.1. Note that the rated SEER is generally higher than the estimated SEER. Also noted in the section, the rated SEER is measured at steady-state conditions and therefore does not account for temperature and load-based performance.

TABLE 5 – COOLING SEASON SUMMARY

| System | ID | Estimated HP Operating Hours | Avg. Outdoor Temp. During HP Operation [˚C] | Estimated Seasonal COP± | Estimated SEER | Rated SEER |
|---|---|---|---|---|---|---|
| **Ductless** | KEL01A (s) | 244 | 27˚C | 3.9 | 13.3 | 26.1 |
| | KEL01B (s) | 1391 | 23.4˚C | 5.8 | 19.8 | 26 |
| | KEL03 (m) | 337 | 21.9˚C | 5.3 | 18.1 | 18 |
| | KEL06i (m) | 1369 | 21.8˚C | 4.5 | 15.2 | 23 |
| | KEL06ii (s) | <25 | - | - | - | 18.9 |
| | PRI01ii (s) | * | * | * | * | 13.9 |
| | VIC02 (m) | 1144 | 19.8˚C | 5.9 | 20.2 | 18 |
| | VIC03 (m) | <25 | - | - | - | 18 |
| | VIC05 (s) | <25 | - | - | - | 21 |
| | VIC06 (m) | <25 | - | - | - | 21.7 |
| | VIC10 (s) | 1172 | 13.3˚C | 2.5 | 8.5 | 29.3 |
| | VIC12 (s) | 56 | 19.2˚C | 6.7 | 22.9 | 19.2 |
| **Single Head (s) -Averages** | | **484** | **20.7˚C** | **4.7** | **16.1** | **22.1** |
| **Multi Head (m) - Averages** | | **578** | **21.2˚C** | **5.2** | **17.8** | **19.7** |
| **Ductless Overall - Averages** | | **531** | **20.9˚C** | **5.0** | **17.0** | **20.9** |
| **Central (Single Stage)** | KEL04 | 247 | 23.6˚C | 1.8 | 6 | 15 |
| | SUM01 | 31 | 17.6˚C | 7.1 | 24.1 | 14 |
| | VIC04 | <25 | - | - | - | 15 |
| | VIC08 | 55 | 29.7˚C | 3.9 | 13.3 | 15 |
| | VIC11 | 27 | 24.8˚C | 3.4 | 11.5 | 15.2 |
| | **Averages** | **72** | **23.9˚C** | **4.1** | **13.7** | **14.8** |
| **Central (Variable)** | PEN01 | 876 | 25.2˚C | 4.5 | 15.5 | 17 |
| | VIC01† | * | * | * | * | 17.6 |
| | **Averages** | **876** | **25.2˚C** | **4.5** | **15.5** | **17.3** |

*† Hybrid system with combined with central variable speed and ductless mini split*
*± Accuracy between ±0.01 and ±0.43*
** Significant data loss in cooling season*
*- fewer than 25 hours of operation*

In all, the average seasonal Coefficient of Performance for cooling was estimated to be 5.0, 4.1 and 4.5 for ductless mini split, central single stage and central variable speed systems, respectively. The HVAC equipment performance requirements outlined in Table 9.36.3.10 of the 2018 BC Building Code state that approved mini split and central systems must have minimum rated SEER and approximately 15. Therefore, the average estimated SEER for all system types suggests that the units are performing at efficiencies that exceed code minimum requirements, with the exception of two units.

### 3.2.3 Heating Season Performance

The heating season analysis is based on all active heating data measured during the monitoring period, rather than a specified time period, though most of the heating did occur during the winter and shoulder season.

The overall range of heating COP values, which includes data from all participants using active heating across their respective range of outdoor temperatures is shown in *Figure 3.16*. Plotted results show that active heating is being used across a wide range of outdoor temperatures. The overall average COP (mid line) is shown to be greater than 1 even down to -14˚C, however, the average COP of relatively poorer performing units (min line) shows that some units are performing at a COP below 1 starting at around 2˚C to 0˚C.

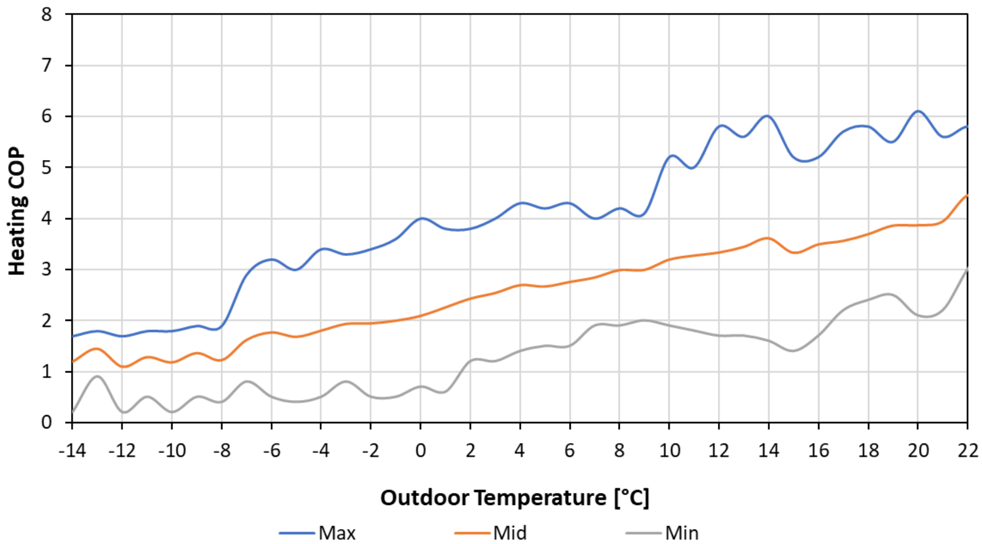


*Figure 3.16 – average heating COP range of all heat pumps vs. outdoor temperature.*

Further analysis shows that the poorest performing unit was a ductless mini-split system. For a better understanding why the unit may not be performing as expected, *Figure 3.17* is a plot of the average heat pump consumption and capacity in heating and cooling for the unit.

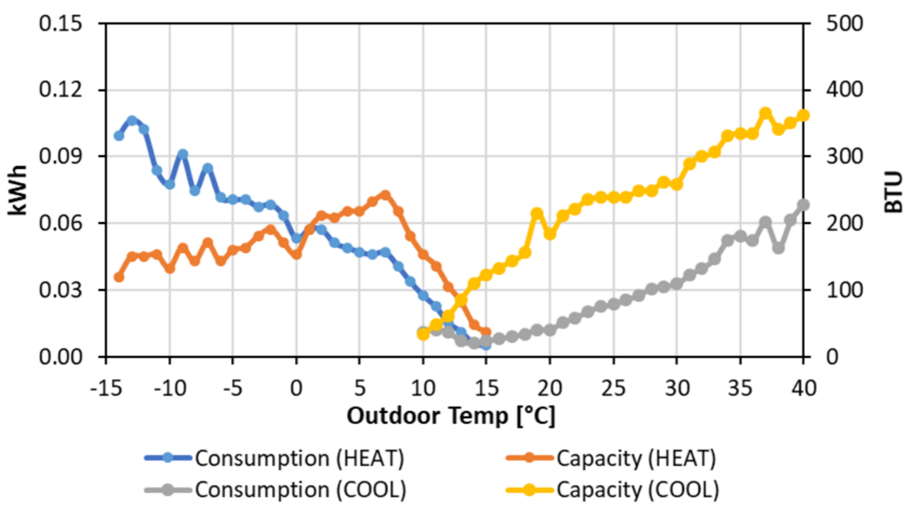


*Figure 3.17 – average heat pump consumption and capacity in heating and cooling (KEL06i).*

The poorest performing heat pump appears to be delivering adequate coooling capacity at all outdoor temperatures and heating capacity during milder outdoor conditions, which suggest that the results are based on system performance rather than equipment error. However, during relatively cold outdoor conditions, the heating capacity of the system drops while the energy demand increases rapidly, which results in a COP below 1. Note that this type of plot has been generated for all measured heat pumps in this analysis – see Appendix A.

Interestingly, the homeowners of the poorest performing unit had expressed that the heat pump refrigerant pressure was tested and it was found that over half of the refrigerant had leaked from their unit and needed to be topped up. This information was reported shortly after the initial site visit. Therefore, between the refrigerant top up and the heating season, the added refrigerant could have leaked again resulting in lower than rated effiency, since the unit appears to have functioned at a reasonable effiency throughout the cooling season. Given that the full length of refrigerant lines are sometimes difficult to access, anecdotal evidence shows that refrigerant leaks are often mitigated by simply replacing lost refrigerant as opposed to sourcing and remediating the leakage path (e.g., holes or loose connections in the line) to ensure leaks do not occur in the future.

The overall heating COP of units was also evaluated according to system type: ductless mini split systems (*Figure 3.18*), central systems with variable speed compressors (*Figure 3.19*) and central systems with single stage compressors (*Figure 3.20*).

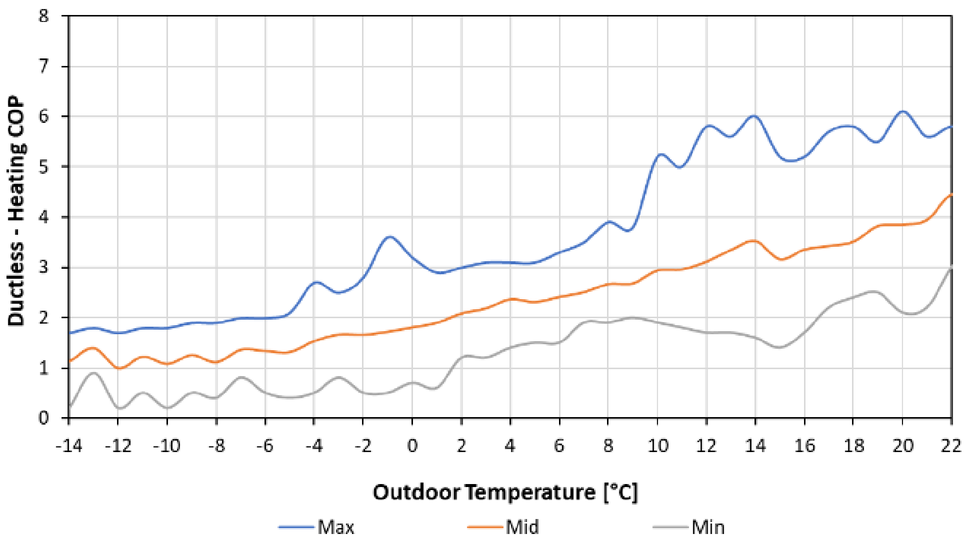


*Figure 3.18 – average heating COP range vs. outdoor temperature for ductless mini split systems.*

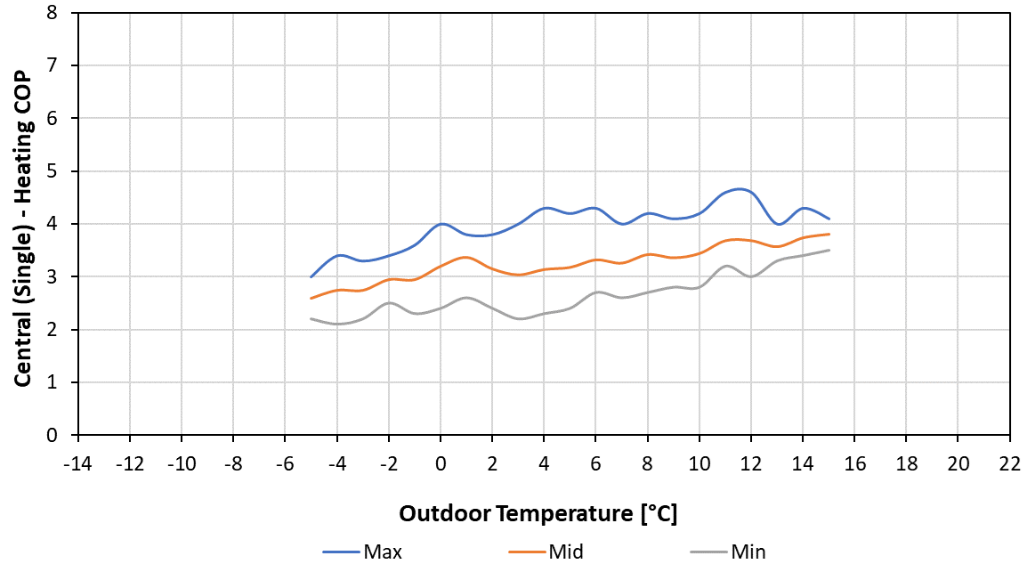


*Figure 3.19 – average heating COP range vs. outdoor temperature for central systems with single speed compressors.*

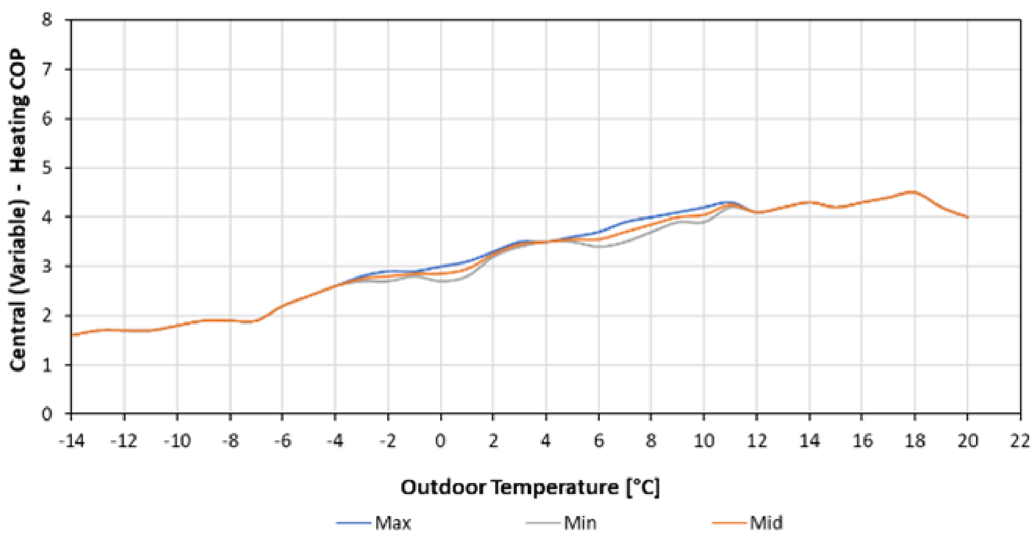


*Figure 3.20 – average heating COP range vs. outdoor temperature for central systems with variable speed compressors.*

Results show that the ductless mini split units are being operated across a large outdoor temperature range. Most of the central systems, with the exception of the one central variable unit, reduce their operation or stop heating around -5˚C and -8˚C and switch over to their backup heating system within the furnace to supplement the heating load at low outdoor temperatures. Interestingly, the central variable speed system without electric resistance backup heat does appear to be operating relatively well under extreme cold conditions (e.g., COP of 1.6 at -14˚C).[21] A summary of the backup heating analysis is described later in this section.

When comparing the average heating and cooling COP ranges, the cooling efficiencies tend to be higher (*Figure 3.21*). This is largely due to the greater temperature differences between

[21] PEN01 home is equipped with a wood burning stove for backup heat; therefore, the heat pump may not be fulfilling the full heat demand at extreme low outdoor temperatures.

outdoor air and the temperature of supplied air during winter compared to the temperature difference between the indoor air and the temperature of air supplied during the summer.

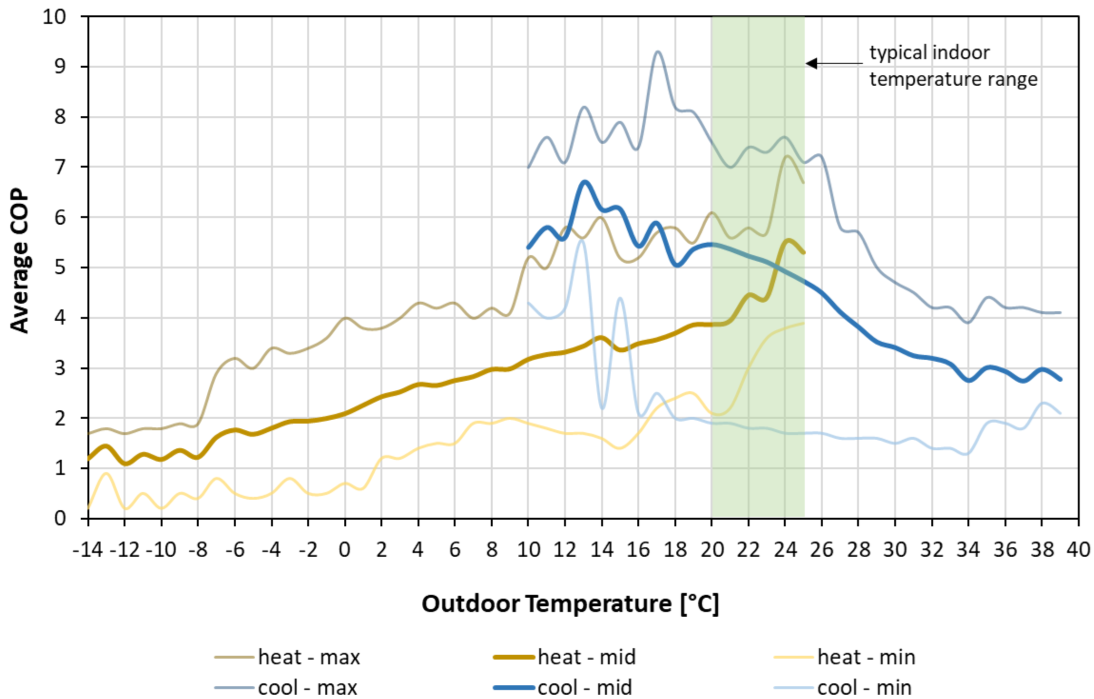


*Figure 3.21 – overall average heating and cooling COP range vs. outdoor temperature and typical indoor temperature range.*

*Figure 3.22* illustrates how the difference between the outdoor air temperature (grey, top graph) and the supply air temperature (green, top) in winter is greater compared to the return air (i.e., indoor air - purple, top) and the supply air temperature in summer (green, top). This generally results in less energy consumption required during the summer compared to winter (blue, middle) to provide roughly the same amount of energy provided or removed (red, middle). Because the energy required is less in summer compared to winter, the efficiency is generally higher (light blue, bottom) compared to winter (orange, bottom). Note that these three types of plots have been generated for all measured heat pumps in this analysis – see Appendix A.

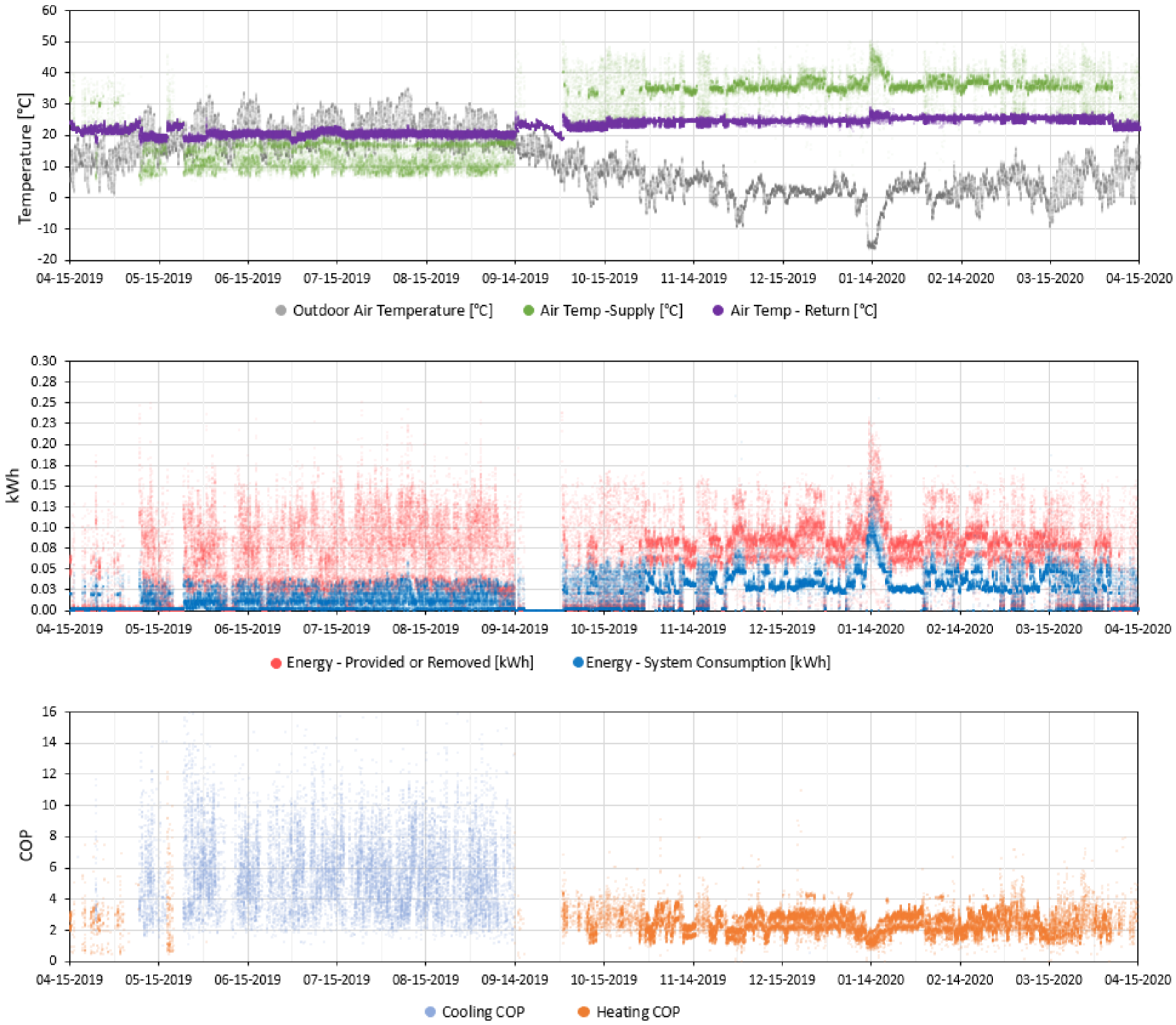


*Figure 3.22 – various measured and estimated parameters for a ductless mini split unit (KEL01b – ductless mini split).*

## *Seasonal Heating Efficiency*

As noted in the previous section, the seasonal efficiency metrics of units are often considered an important factor when comparing different heat pump manufacturers and units. However, the conditions under which in-situ performance is estimated can vary. Therefore, it is not recommended to compare the estimated in-situ seasonal performance against manufacturer ratings, or to compare in-situ seasonal performance of one heat pump to another.

*Figure 3.23* is a distribution of the total number of calculated COP data points used to obtain the seasonal COP of 2.1 for a specific heat pump. Note that this unit was operated at an average outdoor temperature of 6.7˚C. *Figure 3.24* is a distribution of the total calculated COP data points used to obtain a seasonal COP of 1.7 for another heat pump. The average outdoor temperature that the second heat pump was operated at was 1.2˚C. In all, in-situ seasonal efficiency metrics were found to be affected by the average outdoor temperature at which they were operated at. An analysis investigating the correlation between seasonal COP and average outdoor temperatures was performed; however, a weak correlation ($R^2$=0.22) was found. This is likely due to the impact of competing variables and resulting difficulty in isolating outdoor operation temperature as a single variable.

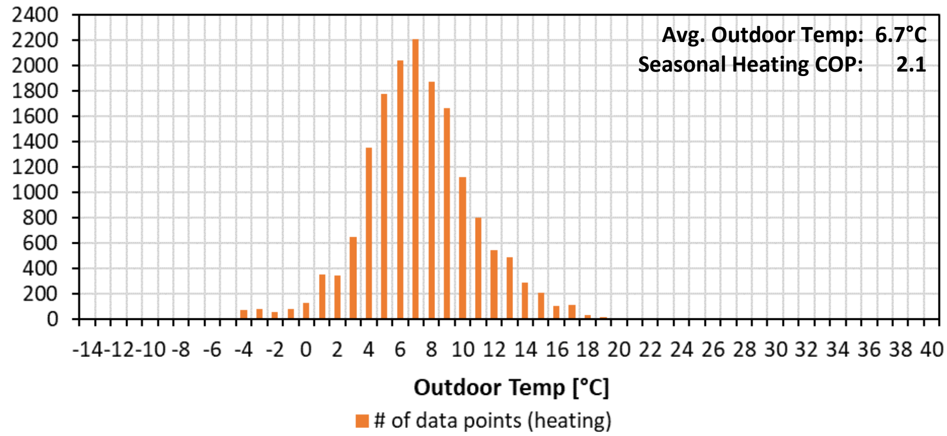


*Figure 3.23 – distribution of total calculated COP data points used to calculate the seasonal COP (VIC05 – ductless mini split). The average outdoor temperature listed in the top corner represents the average temperature at which the units were operated at.*

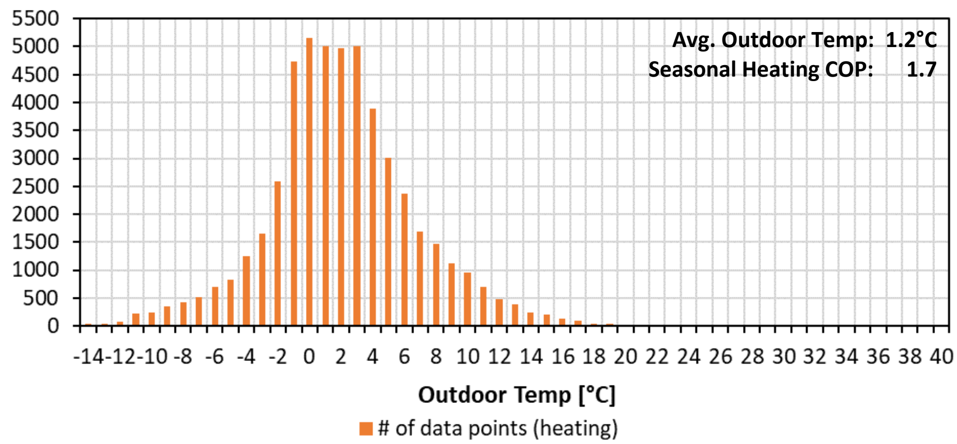


*Figure 3.24 – distribution of total calculated COP data points used to calculate the seasonal COP (KEL06ii – ductless mini split). The average outdoor temperature listed in the top corner represents the average temperature at which the units were operated at.*

TABLE 6 is a summary of the estimated heating season efficiencies of each unit expressed as seasonal COP and HSPF. The seasonal heating COP was determined by averaging all COP values throughout the monitored heating and shoulder season. The seasonal efficiency (HSPF) was determined based on seasonal heating COP using the multiplication factor described in Section 1.1. Note that, compared to central systems, the rated HSPF of ductless mini split systems is generally higher than the estimated HSPF. For central systems, the values in brackets represent the estimated seasonal COP and HSPF including backup heat. Results show that, in some cases, the backup heat can significantly reduce the seasonal heating efficiency of central units. The backup heat analysis for central systems is presented later in this section. In all, the average seasonal heating COP (accounting for backup heat) is estimated to be 2.4, 2.6 and 3.3 for ductless mini split, central single stage and central variable speed systems, respectively.

| TABLE 6 – HEATING SEASON SUMMARY | | | | | | |
|---|---|---|---|---|---|---|
| **System** | **ID** | **Estimated HP Operating Hours** | **Avg. Outdoor Temp. During HP Operation [˚C]** | **Estimated Seasonal COP±** | **Estimated HSPF** | **Rated HSPF** |
| Ductless | KEL01A (s) | 4854 | 4.3˚C | 2.9 | 9.8 | 11.5 |
| | KEL01B (s) | 2929 | 2.6˚C | 2.6 | 8.8 | 10 |
| | KEL03 (m) | 4723 | 4.3˚C | 2.2 | 7.4 | 12.5 |
| | KEL06i (m) | 3454 | 2.8˚C | 2.7 | 9.3 | 13 |
| | KEL06ii (s) | 4220 | 1.2˚C | 1.7 | 5.8 | 10.7 |
| | PRI01ii (s) | 4545 | 0.7˚C | 1.6 | 5.2 | 13.9 |
| | VIC02 (m) | 2233 | 7.1˚C | 3.0 | 10.1 | 9.5 |
| | VIC03 (m) | 4142 | 6.7˚C | 2.5 | 8.5 | 9.5 |
| | VIC05 (s) | 1370 | 6.7˚C | 2.1 | 7.3 | 11 |
| | VIC06 (m) | 4098 | 9.6˚C | 2.4 | 8.1 | 11.6 |
| | VIC10 (s) | 3458 | 6.1˚C | 1.9 | 6.6 | 14 |
| | VIC12 (s) | 3446 | 6.2˚C | 3.4 | 11.8 | 11.7 |
| | **Single Head (s) -Averages** | **3546** | **4.0˚C** | **2.3** | **7.9** | **11.8** |
| | **Multi Head (m) - Averages** | **3730** | **6.1˚C** | **2.6** | **8.7** | **11.2** |
| | **Ductless Overall - Averages** | **3638** | **5.0˚C** | **2.4** | **8.3** | **11.5** |
| Central (Single Stage) | KEL04 | 1329 | 1.4˚C | 2.6 (n/a#) | 9.0 (n/a#) | 9 |
| | SUM01 | 428 | 0.6˚C | 3.8 (1.4*) | 13.1 (4.8*) | 8.2 |
| | VIC04 | 236 | 8.8˚C | 2.9 (2.9*) | 9.8 (9.8*) | 9 |
| | VIC08 | 1149 | 7.4˚C | 4.0 (3.4*) | 13.7 (11.6*) | 9 |
| | VIC11 | 997 | 7.6˚C | 2.9 (2.6*) | 10.0 (8.9*) | 8.7 |
| | **Averages** | **828** | **5.6˚C** | **3.2 (2.6*)** | **11.1 (8.8*)** | **8.8** |
| Central (Variable) | PEN01 | 1938 | 1.1˚C | 3.2 (3.2*) | 10.9 (10.9*) | 11 |
| | VIC01† | 2062 | 7.8˚C | 3.7 (3.3*) | 12.7 (11.3*) | 10.3 |
| | **Averages** | **2000** | **4.5˚C** | **3.5 (3.3*)** | **11.6 (11.1*)** | **10.9** |

*† Hybrid system with combined with central variable speed and ductless mini split*
*± Accuracy between ±0.01 and ±0.07, with exception of VIC06 at ±0.7*
** including electric resistance backup heating*
*# propane backup heating (not measured)*

The HVAC equipment performance requirements outlined in Table 9.36.3.10 of the 2018 BC Building Code state that approved mini split and central systems must have minimum rated HSPF performance of approximately seven. Therefore, based on the average estimated HSPF for each system type, the results suggest that the units are performing in general conformance with code minimum requirements. In all, the overall heating COP range across the measured outdoor temperatures appears to be somewhat consistent with performance values reported by the Government of Canada (COP around 3.3 at 10˚C) and U.S. Department of Energy (COP between 1.5 and 3).[22,23]

[22] https://www.nrcan.gc.ca/energy/publications/efficiency/heating-heat-pump/6831
[23] https://www.energy.gov/energysaver/heat-pump-systems/air-source-heat-pumps

### *Short-Cycling*

Appropriate sizing of heat pumps is often a concern with respect to system efficiency. Since the total heat demand of each home is not known, a reasonable indication as to whether a system is right-sized is to measure the length of its operating cycles. As noted in Section 1.1, the heat pump's efficiency can degrade if units are operating in cycles shorter than six minutes. *Figure 3.25* is an example of a heat pump that is, on average, only operating for five minute cycles during a typical winter period, whereas *Figure 3.26* is the same unit though operating for much longer periods (hours) during an extreme winter period. The owner of this heat pump was able to confirm that this unit was deliberately over-sized.

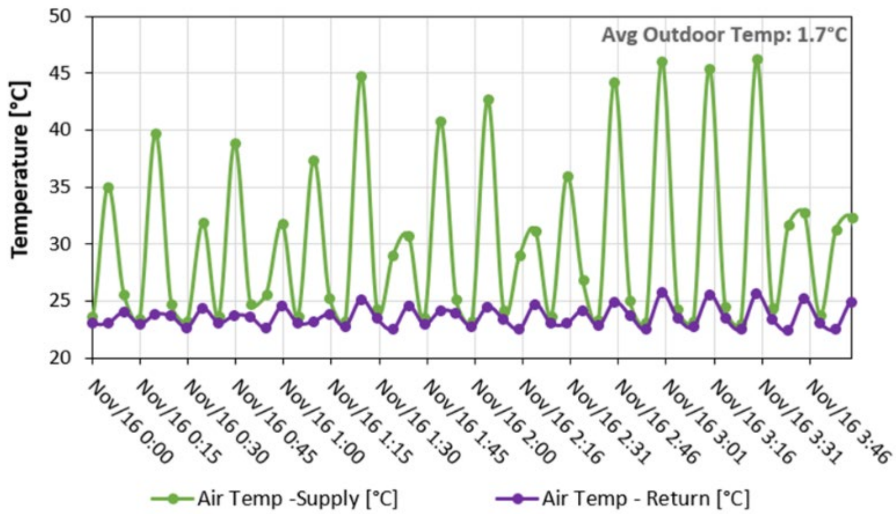


*Figure 3.25 – example of heat pump short-cycling during typical winter period (average outdoor temperature 1.7°C). Note that each dot represents a five-minute interval (KEL01b – ductless mini split).*

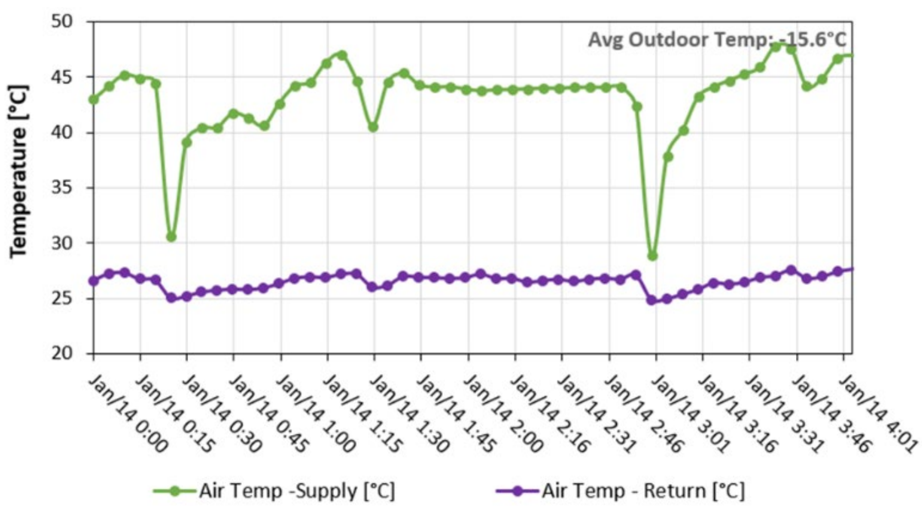


*Figure 3.26 – example of heat pump operating cycles greater than ten minutes during extreme winter period (average outdoor temperature -15.6°C). Note that each dot represents a five-minute interval (KEL01b – ductless mini split).*

*Figure 3.27* is an example of a heat pump that appears to be adequately sized, where the average heating cycles are greater than ten minutes during a typical winter period. *Figure 3.28* is the same unit operating during an extreme winter period and continues in cycles greater than ten minutes. It is important to note, however, that during the extreme winter period, the deemed oversized unit does appear to be maintaining the return temperature (i.e., interior ambient temperature) to a higher temperature. Four-hour samples of heat pump operating cycles during typical and extreme winter periods for all measured heat pumps were produced and can be found in Appendix A. Based on the plots that exhibit active heating, 33% (6/18) of ductless mini split units appear to be operating for ten minutes or less per cycle during a typical winter period. The central heat pump systems, however, appear to be operating for cycles longer than ten minutes for both typical and extreme winter periods.

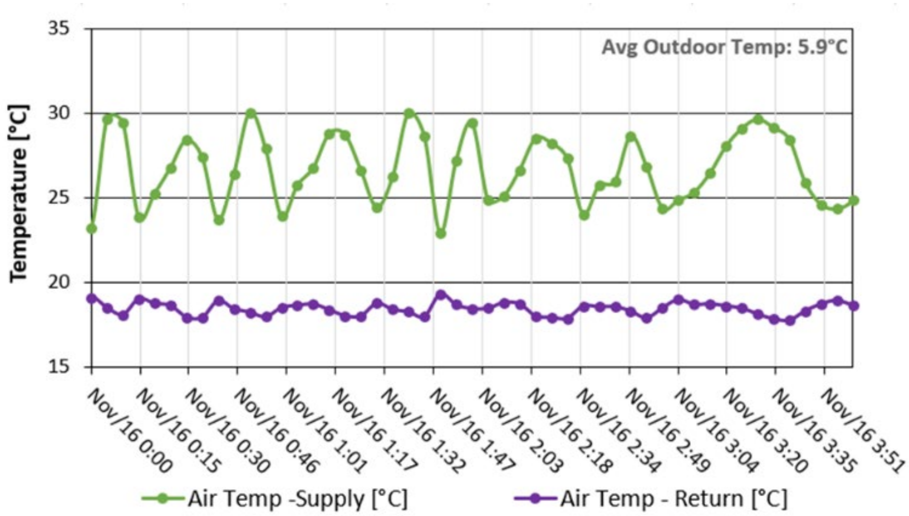


*Figure 3.27 – example of heat pump operating cycles greater than ten minutes during typical winter period (average outdoor temperature 5.9˚C). Note that each dot represents a five-minute interval (KEL01a – ductless mini split).*

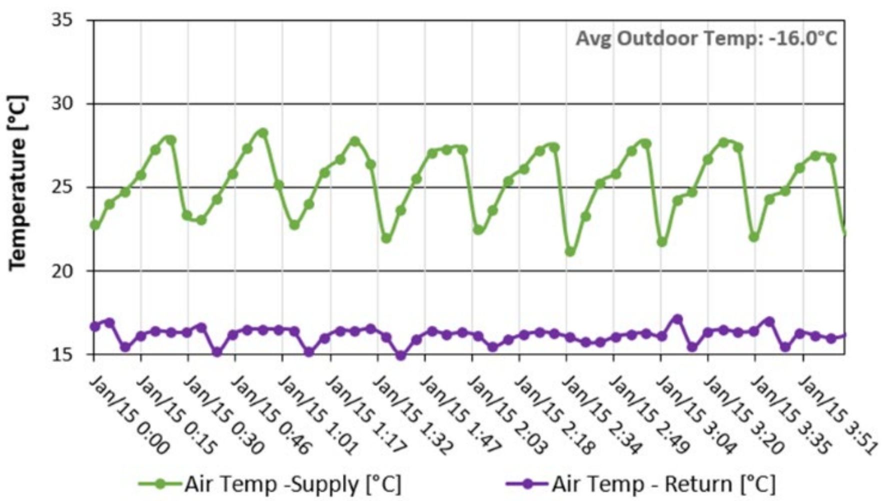


*Figure 3.28 – example of heat pump operating cycles greater than ten minutes during extreme winter period (average outdoor temperature -16˚C). Note that each dot represents a five-minute interval (KEL01a – ductless mini split).*

### *Defrost Cycling*

As described in Section 2.3, the defrost cycles for ductless mini split systems were determined by isolating periods of sporadic cooling during the winter. The section also describes that, for central systems, this method of isolating defrost cycles is not possible as the furnace backup coils are typically designed to provide heat while the heat pump is temporarily operating in defrost (i.e., cooling mode). Therefore, periods of cooling were not noted during heating season for central systems. However, some inferences were made with respect to central system defrost cycling in the following section.

*Figure 3.29* is an example of a single defrost cycle for a ductless mini split system which, in this case, is occurring for roughly ten minutes, given that each data point represents a five-minute interval. Defrost intervals between five and ten minutes were found to be typical for mini split units. Note that the outdoor temperature is slightly below freezing (around 2˚C) which is expected as defrost cycles typically occur when outdoor temperatures fluctuate between 5˚C and -5˚C.

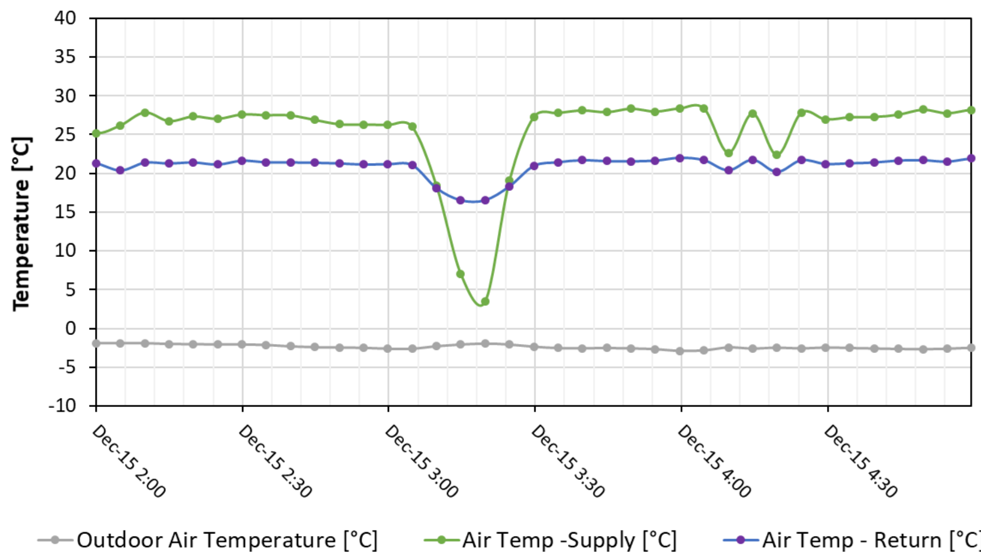


*Figure 3.29 – example of a single defrost cycle during heating season. Each dot represents a five-minute interval (PRI01ii – ductless mini split).*

TABLE 7 is a summary of the total defrost energy of each unit that exhibited sporadic cooling during the winter months between November 1st, 2019 and March 1st, 2020. The table shows that, compared to the total heat pump consumption for this winter period, the average defrost energy was found to be less than 1%. This fraction of time spent in defrost is somewhat lower than anticipated; however, the total amount of time that each unit spent in cooling during the analysis period was found to be relatively small compared to heating. The data also suggests that units are equipped with demand-defrost control (i.e., defrost operates only when required as opposed to operating on a timer), give that cycles are typically shorter than 30 minutes. In addition, not all of the ductless mini split heat pumps showed signs of cooling during the winter, which suggests that some indoor heads may be equipped with electric resistance backup heating coils. Note that backup heat consumption for ductless mini splits would be included in the total system consumption, whereas central system backup is separately metered. Any backup heat consumption for ductless mini splits would be included as part of the system's total efficiency. For central systems, the backup energy consumption is measured separate from the system consumption; though for this study, both central heat pump efficiencies have been presented (with and without backup) for illustrative purposes.

| TABLE 7 –DEFROST CYCLE SUMMARY FOR DUCTLESS SYSTEMS (NOV 1, 2019 - MAR 1, 2020) | | | | |
|---|---|---|---|---|
| **ID** | **System Consumption [kWh]** | **Defrost Cycle Consumption** | | **Avg. Temp During Defrost Cycle [˚C]** |
| | | **[kWh]** | **[%]*** | |
| KEL01a | 1278.7 | 1.6 | 0.1% | 0.9˚C |
| KEL01b | 1211.5 | 1.8 | 0.1% | 0.9˚C |
| KEL06ii | 1735.6 | 19.1 | 1.1% | -1.2˚C |
| PRI01ii | 1184.9 | 17.9 | 1.5% | -3.7˚C |
| VIC03 | 1711.8 | 5.9 | 0.3% | -0.6˚C |
| VIC10 | 897.9 | 1.4 | 0.2% | -2.0˚C |
| VIC12 | 1742.6 | 2.2 | 0.1% | 1.3˚C |

**percentage of total system consumption*

### *Backup Heating*

The methodology for measuring electric resistance backup heat for central heat pump systems is described in Section 2.3. In summary, the back-up heat consumption was sub-metered simultaneously on the same circuit as the heat pump fan consumption. Therefore, to isolate the backup heat from fan consumption, the typical fan consumption for each central unit is subtracted from the total sub-metered electricity.

TABLE 8 is a summary of the backup heat consumption for central systems compared to the heat pump system consumption and total home electricity consumption from November 1st, 2019 to March 1st, 2020. The percentages in the table are based on the total electricity consumption.

| TABLE 8 – BACKUP HEAT SUMMARY FOR CENTRAL SYSTEMS (NOV 1, 2019 - MAR 1, 2020) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| **ID** | **Total Electricity Consumption†** | **Heat Pump Consumption** | | **Backup Heat Consumption** | | **Total Space Heating Consumption** | | **Estimated Temp at which Backup Heat Begins** | **Estimated Electric Coil Size** |
| | **[kWh]** | **[kWh]** | **[%]*** | **[kWh]** | **[%]*** | **[kWh]** | **[%]*** | **[˚C]** | **[kW]** |
| KEL04 | 10620 | 2146 | 20% | - | - | - | - | - | - |
| SUM01 | 8503 | 1114 | 13% | 1895 | 22% | 3009 | 35% | 3˚C | 30 |
| PEN01 | 8411 | 3510 | 42% | 10 | 0.1% | 3519 | 42% | -5˚C | 5 |
| VIC01 | 5336 | 1445 | 27% | 43 | 1% | 1488 | 28% | -5˚C | 5 |
| VIC04 | 5544 | 1117 | 20% | 11 | 0.2% | 1128 | 20% | 0˚C | 5 |
| VIC08 | 4476 | 1344 | 30% | 230 | 5% | 1574 | 35% | 5˚C | 15 |
| VIC11 | 3854 | 902 | 23% | 122 | 3% | 1023 | 27% | 0˚C | 10 |
| **Average** | **6678** | **1654** | **25%** | **385** | **5%** | **1957** | **31%** | | |

*† Includes non-space conditioning electricity (i.e., lighting, appliances, etc.)*
** percentage of total electricity consumption*
*- Propane backup (not measured)*

Results from the table above show that the average heat pump consumption represented 25% of the total electricity consumption for the analysis period. Interestingly, when factoring the additional energy consumed for backup heating, the average space heating consumption increases to 31% of total electricity consumption. The backup heating from the SUM01 system in particular represented 22% of the total electricity consumption, compared to 13%

consumed by heat pump. In addition, the average backup heat consumption accounted for 20% of total space conditioning for the units above, and as much as 63% for SUM01. TABLE 9 and TABLE 10 demonstrate how backup heat can significantly affect the efficiency of the complete heating system and is perhaps not always considered when analyzing central heat pump consumption. For example, TABLE 10 shows that the seasonal COP was reduced by 0.6 on average when including the energy consumption from backup heat and as much as 2.4 in one case.

TABLE 9 – TOTAL ESTIMATED HEAT PUMP CONSUMPTION AND CAPACITY WITH PERCENT BACKUP VS. TOTAL SPACE CONDITIONING

| ID | Total Heat Pump Consumption [kWh] | Estimated Total Heat Pump Capacity [kWh] | Backup Heat Consumption [kWh] | Total Space Heating Consumption [kWh] | % backup vs. total space heating |
|---|---|---|---|---|---|
| SUM01 | 1114 | 4233 | 1895 | 3009 | 63% |
| PEN01 | 3510 | 11232 | 10 | 3520 | 0.3% |
| VIC01 | 1445 | 5347 | 43 | 275 | 1.8% |
| VIC04 | 1117 | 3239 | 11 | 1128 | 1.0% |
| VIC08 | 1344 | 5376 | 230 | 1574 | 15% |
| VIC11 | 902 | 2616 | 122 | 1024 | 12% |
| **Avg** | **1572** | **5340** | **385** | **1955** | **20%** |

TABLE 10 – HSPF AND SEASONAL HEATING COP REDUCTION FOR CENTRAL SYSTEMS AS A RESULT OF BACKUP

| ID | Estimated Average HSPF (Seasonal COP) | | Estimated Average HSPF incl. Backup (Seasonal COP) | | HSPF Reduction from Backup (Seasonal COP) | |
|---|---|---|---|---|---|---|
| SUM01 | 13.1 | (3.8) | 4.8 | (1.4) | 8.2 | (2.4) |
| PEN01 | 10.9 | (3.2) | 10.9 | (3.2) | 0 | (0) |
| VIC01 | 12.7 | (3.7) | 11.3 | (3.3) | 1.3 | (0.4) |
| VIC04 | 9.8 | (2.9) | 9.8 | (2.9) | 0 | (0) |
| VIC08 | 13.7 | (4.0) | 11.6 | (3.4) | 2.1 | (0.6) |
| VIC11 | 10.0 | (2.9) | 8.9 | (2.6) | 1.0 | (0.3) |
| **Avg** | **11.6** | **(3.4)** | **9.6** | **(2.8)** | **2.0** | **(0.6)** |

Although it was not possible to definitively distinguish whether the backup heat consumption was allocated to defrost or supplemental heat, it is understood that defrost cycles typically occur between 5°C and -5°C. Therefore, the total backup heat consumption for central systems was separated into two bins: backup heat consumption above -5°C and below -5°C (TABLE 11).

| TABLE 11 – BACKUP HEAT DISTRIBUTION FOR CENTRAL SYSTEMS (NOV 1, 2019 - MAR 1, 2020) | | | | | | |
|---|---|---|---|---|---|---|
| **ID** | **Backup Heat Consumption** | **Avg. Outdoor Temp during Backup** | **Backup Above -5˚C** | | **Backup Below -5˚C** | |
| | **[kWh]** | **[˚C]** | **[kWh]** | **[%]*** | **[kWh]** | **[%]*** |
| SUM01 | 1895 | -1.7˚C | 1080 | 57% | 815 | 43% |
| PEN01 | 10 | -2.9˚C | 7 | 72% | 3 | 28% |
| VIC01 | 43 | 1.3˚C | 43 | 100% | 0 | 0% |
| VIC04 | 11 | 3.2˚C | 11 | 100% | 0 | 0% |
| VIC08 | 230 | 1.9˚C | 230 | 100% | 0 | 0% |
| VIC11 | 122 | 2.3˚C | 122 | 100% | 0 | 0% |

**percentage of backup heat consumption*

Results from the table show that in the colder regions, a larger percent of the backup heat is being consumed at temperatures below -5˚C, whereas the more temperate climate exhibits all of its backup heat consumption above -5˚C. Assuming backup heat consumption above -5˚C is indicative of defrost energy, an inference can be made that significantly more energy is allocated to defrost for central systems compared to ductless mini splits.

## 3.3 Energy Savings Evaluation

The energy consumption results shown here are a comparison of whole home electricity and natural gas consumption before versus after the installation of the heat pumps.

### 3.3.1 Electricity Consumption Analysis

Electrical utility data for 18 participants was provided by FortisBC or BC Hydro for at least one year prior to heat pump installation and was compared with post-installation utility data. Sufficient pre-heat pump installation electricity utility data was not available for four participants, so they were excluded from this analysis. The data presented in TABLE 12 shows the savings in estimated annual electricity consumption and the savings in estimated annual electricity cost for the reporting period of April 2019 to end of March 2020. Positive values indicate energy savings, while negative values indicate an energy increase relative to the estimated pre-installation baseline. Note that supplementary heating sources using fuel types other than electricity or natural gas were used in many homes both before and after the installation of heat pumps that were not captured by utility bills; for example, wood fireplaces or oil furnaces. In addition, the whole-home electric utility data includes base loads (lighting, plugs, etc.). Base loads are not expected to change significantly from pre to post-heat pump installation; however, there may be differences in occupant behaviour from year to year (e.g., appliances and their operation, home renovations, changes in occupancy or habits, etc.) which could have some impact on the results.

| TABLE 12 - ESTIMATED SAVINGS IN ELECTRICITY CONSUMPTION AND ASSOCIATED COST IMPACTS FROM INSTALLING HEAT PUMPS | | | | | |
|---|---|---|---|---|---|
| **ID** | **Savings in Electricity Consumption [kWh/yr]** | **Area Normalized Savings in Electricity [kWh/m²/yr]** | **Savings in Electricity Costs [$/yr]*** | **Area Normalized Savings in Electricity Costs [$/m²/yr]** | **Pre-heat pump installation fuel source(s) (primary in bold)** |
| VIC 01 | -2600 | -27 | $(360) | $(4) | **oil furnace** + electric radiant |
| VIC 02 | 2800 | 14 | $390 | $2 | **BB** + wood fireplace + electric radiant |
| VIC 03 | 5500 | 34 | $770 | $5 | **electric baseboards (BB)** |
| VIC 04 | -400 | -3 | $(60) | $(0) | **oil furnace** |
| VIC 05 | -2000 | -12 | $(280) | $(2) | **BB** + wood stove |
| VIC 06 | -2700 | -33 | $(380) | $(5) | **oil furnace** + BB + wood fireplace |
| VIC 07 | 2200 | 22 | $310 | $3 | **BB** + wood fireplace |
| VIC 08 | 2500 | 18 | $350 | $3 | **oil furnace** + BB + wood fireplace |
| VIC 09 | 100 | 2 | $10 | $0 | **gas fireplace** + BB + electric radiant |
| VIC 10 | 4500 | 37 | $630 | $5 | **BB** + wood fireplace |
| VIC 12 | -800 | -20 | $(110) | $(3) | **gas fireplace** + BB |
| KEL 02 | 2100 | 8 | $300 | $1 | **oil furnace** w/ electric backup |
| KEL 04 | 7300 | 2 | $1,100 | $4 | **propane fireplace** + wood fireplace +BB |
| KEL 06 | 18000 | 100 | $2,600 | $15 | **electric baseboards** |
| KEL 07 | 5300 | 40 | $760 | $6 | **BB** + wood fireplace |
| PRI 01 | 10000 | 50 | $1,500 | $7 | **wood-fired boiler** for radiant floor w/ electric backup + BB |
| PEN 01 | 8900 | 38 | $1,300 | $5 | **electric furnace** + woodstove |
| SUM 01 | -300 | -2 | $(40) | $(0) | **gas fireplace** + electric furnace + heat pump |

**Rate of $0.14 per kWh was used to calculate cost savings for the BC Hydro participants.[24] Rate of $0.14 per kWh used to calculate cost savings for the FortisBC participants.[25]*

Of the 18 participants with available utility data, 12 (67%) experienced annual energy savings after the heat pumps were installed. KEL06 experienced the greatest energy savings post-installation with savings of 18,000 kWh and $2,600 for the year-long reporting period, while VIC06 experienced the greatest increase in energy consumption post-installation of 2,700 kWh and $380 over the one-year period.

A main reason that some participants would consume more electricity is that they were previously heating their home using a different fuel source. In addition, there may also be added load from actively cooling with their heat pump, given that most customers reported that they did not previously have air conditioning. In all, the average savings for cases that used electricity as a primary heating source were found to be 5650 kWh and $810 for the year-long monitoring period.

TABLE 13 is the condensed list of participants who explicitly stated that they used electricity as their primary heating source prior to the heat pump.

TABLE 13 - ESTIMATED SAVINGS IN ELECTRICITY CONSUMPTION AND ASSOCIATED COST IMPACTS FROM INSTALLING HEAT PUMPS. SORTED TO INCLUDE ONLY CASES THAT USED ELECTRICITY AS PRIMARY HEATING SOURCE PRE-HEAT PUMP INSTALLATION.

| ID | Savings in Electricity Consumption [kWh/yr] | Area Normalized Change in Electricity [kWh/$m^2$/yr] | Change in Electricity Costs [$/yr]* | Area Normalized Change in Electricity Costs [$/$m^2$/yr]* | Pre-heat pump installation fuel source(s) (primary in bold) |
|---|---|---|---|---|---|
| VIC02 | 2800 | 14 | $390 | $2 | **BB** + wood fireplace + electric radiant |
| VIC03 | 5500 | 34 | $770 | $5 | **electric baseboards (BB)** |
| VIC05 | -2000 | -12 | $(280) | $(2) | **BB** + wood stove |
| VIC07 | 2200 | 22 | $310 | $3 | **BB** + wood fireplace |
| VIC10 | 4500 | 37 | $630 | $5 | **BB** + wood fireplace |
| KEL06 | 18000 | 100 | $2,600 | $15 | **electric baseboards** |
| KEL07 | 5300 | 40 | $760 | $6 | **BB** + wood fireplace |
| PEN01 | 8900 | 38 | $1,300 | $5 | **electric furnace** + woodstove |
| **Average** | **5650** | **34** | **$810** | **$5** | |

**Rate of $0.14 per kWh was used to calculate cost savings for the BC Hydro participants.[26] Rate of $0.14 per kWh used to calculate cost savings for the FortisBC participants.[27]*

TABLE 13 shows that the majority of homes that used electricity as primary heating source prior to heat pump installation had baseboards; the exception being PEN01 which replaced its electric furnace with a central variable speed system. Of these homes, only one case

[24] https://app.bchydro.com/accounts-billing/rates-energy-use/electricity-rates/residential-rates.html (assessed May 2020)

[25] https://www.fortisbc.com/about-us/regulatory-affairs/our-electricity-utility/electric-bcuc-submissions/electricity-rates (assessed May 2020)

[26] https://app.bchydro.com/accounts-billing/rates-energy-use/electricity-rates/residential-rates.html (assessed May 2020)

[27] https://www.fortisbc.com/about-us/regulatory-affairs/our-electricity-utility/electric-bcuc-submissions/electricity-rates (assessed May 2020)

showed an increase in electricity consumption (VIC05). Given the mixed fuel uses, the cause for increased electricity consumption is unclear; however, the increase could be related to a greater reliance on the heat pump over wood. In all, the average amount of savings for cases that used electricity as a primary heating prior to their heat pump installation is 5650 kWh and $810 for the year-long monitoring period.

TABLE 14 lists the remaining participants who either explicitly stated or it was inferred that they used non-electric fuel as their primary heating source prior to the heat pump installation. The table shows that the electricity consumption post-heat pump installation for these cases varies significantly. For example, PRI01 exhibited savings of 10,000 kWh and $1,500 for the year-long monitoring period, whereas VIC06 had an increased utility cost of 2,700 kWh and $380. In all, the average savings for cases that used a non-electric primary heating source were found to be 1520 kWh and $231 for the year-long monitoring period.

TABLE 14 - ESTIMATED SAVINGS IN ELECTRICITY CONSUMPTION AND ASSOCIATED COST IMPACTS FROM INSTALLING HEAT PUMPS. SORTED TO INCLUDE ONLY CASES THAT USED A NON-ELECTRIC PRIMARY HEATING SOURCE PRE-HEAT PUMP INSTALLATION.

| ID | Savings in Electricity Consumption [kWh/yr] | Area Normalized Savings in Electricity [kWh/m²/yr] | Savings in Electricity Costs [$/yr]* | Area Normalized Savings in Electricity Costs [$/m²/yr]* | Pre-heat pump installation fuel source(s) (primary in bold) |
|---|---|---|---|---|---|
| VIC01 | -2600 | -27 | $(360) | $(4) | **oil furnace** + electric radiant |
| VIC04 | -400 | -3 | $(60) | < $1 | **oil furnace** |
| VIC06 | -2700 | -33 | $(380) | $(5) | **oil furnace** + BB + wood fireplace |
| VIC08 | 2500 | 18 | $350 | $3 | **oil furnace** + BB + wood fireplace |
| VIC09 | 100 | 2 | $10 | < $1 | **gas fireplace** + BB + electric radiant |
| VIC12 | -800 | -20 | $(110) | $(3) | **gas fireplace** + BB |
| KEL02 | 2100 | 8 | $300 | $1 | **oil furnace** w/ electric backup |
| KEL04 | 7300 | 2 | $1,100 | $4 | **propane fireplace** + wood fireplace +BB |
| PRI01 | 10000 | 50 | $1,500 | $7 | **wood-fired boiler** for radiant floor w/ electric backup + BB |
| SUM01 | -300 | -2 | $(40) | < $(1) | **gas fireplace** + electric furnace + heat pump |
| **Average** | **1520** | **-1** | **$231** | **< $1** | |

**Rate of $0.14 per kWh was used to calculate cost savings for the BC Hydro participants.[28] Rate of $0.14 per kWh used to calculate cost savings for the FortisBC participants.[29]*

Understandably, fuel switching from a non-electric heating source to electric will likely increase the electricity consumption of the home. However, what is not captured in this section (due to insufficient data) is the reduction, or even elimination of natural gas in some cases. The following section shows two examples of homes with non-electric primary space heating where both electricity and natural gas data was obtained, though a whole home energy consumption analysis for most homes is incomplete.

[28] https://app.bchydro.com/accounts-billing/rates-energy-use/electricity-rates/residential-rates.html (assessed May 2020)

[29] https://www.fortisbc.com/about-us/regulatory-affairs/our-electricity-utility/electric-bcuc-submissions/electricity-rates (assessed May 2020)

### 3.3.2 Whole Home Energy Consumption Analysis

For the three participants whose heating and cooling energy consumption before and after heat pump installation was fully captured via utility electricity and/or natural gas, a whole home energy consumption analysis was conducted to gauge the impact of heat pump installation on whole home energy use. The figures below show the energy usage for the April 2019 to March 2020 reporting period for the three participants.

The whole home energy impact of installing heat pumps was assessed on an annual basis by summing the monthly energy consumption for the three participants with complete utility data. Figure 3.30 shows both the measured annual energy consumption and the estimated weather-normalized baseline for the three participants. Both VIC03 and VIC09 show a decrease in overall energy consumption from the estimated baseline. The overall energy consumption for SUM01 is very similar to the baseline although there is a slight consumption increase.

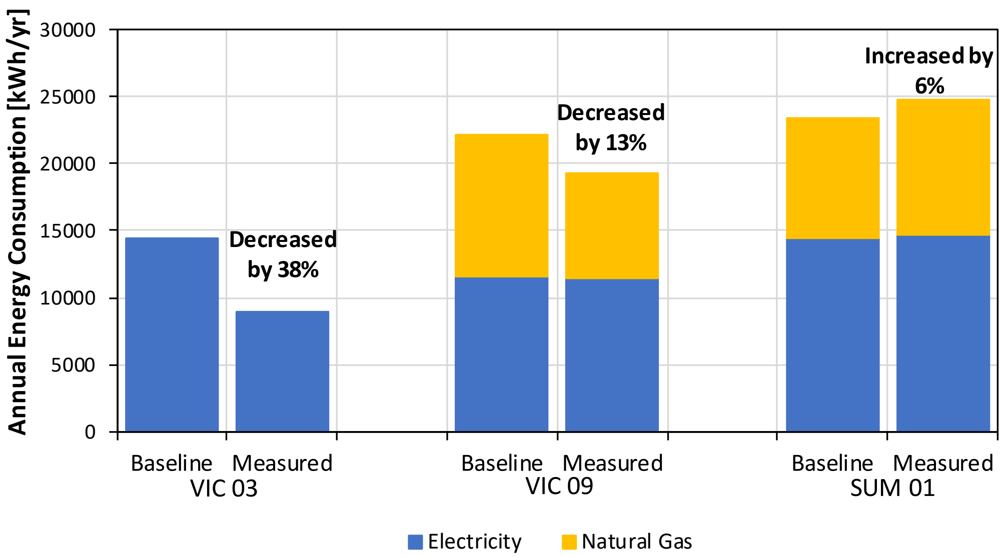


*Figure 3.30 Annual whole home energy consumption for the three participants. Electricity in shown in blue, natural gas in yellow. Both measured and estimated baseline energy consumption are shown.*

The whole home energy consumption and GHG savings are summarized for the three participants with complete home energy data in TABLE 15 for April 2019 to March 2020.

| TABLE 15 - SUMMARY OF THE ESTIMATED WHOLE HOME ENERGY CONSUMPTION & GHG IMPACTS DURING THE REPORTING PERIOD | | | | | | |
|---|---|---|---|---|---|---|
| **ID** | **Floor Area [m²]** | **Savings in Electricity Consumption [kWh/yr]** | **Savings in Natural Gas Consumption [ekWh/yr, GJ/yr]** | **Savings in Overall Utility Cost [$/yr]** | **GHG Savings [kg $CO_2$e/yr]*** | **Pre-heat pump installation fuel source(s) (primary in bold)** |
| VIC03 | 160 | 5500 | N/A | $770 | 60 | **electric baseboards (BB)** |
| VIC09 | 60 | 100 | 2900, 10 | $110 | 540 | **gas fireplace** + BB + electric radiant |
| SUM01 | 120 | -300 | -880, -3 | $(10) | -160 | **gas fireplace** + electric furnace + heat pump |

**Emissions factor of 0.011kg$CO_2$e/kwh was used for electricity and 0.185 kg$CO_2$e/kwh was used for natural gas.*[30]

As expected, VIC03 experienced the greatest electricity savings as a result of the installation of their heat pump, largely due to the higher efficiency of the heat pump compared to electric resistance baseboards. The GHG savings, however, were modest compared to VIC09 for example, due to the lower emissions factor of electricity compared to natural gas. Electricity consumption for both VIC09 and SUM01 did not change significantly, though VIC09 consumption increased slightly. Interestingly, the natural gas consumption appeared to vary for VIC09 and SUM01. Both homes were equipped with gas fireplaces both pre- and post-heat pump installation, rather than a replacement. It is likely that VIC09 relied less on their gas fireplace after the installation of the heat pump, whereas SUM01 showed relatively stable consumption.

### 3.3.3 Heating Load Analysis

The fraction of heat delivered by the heat pump compared to the whole home heating load was estimated for homes with all electric heating (KEL01a, KEL01b, and VIC03). Heating load refers to the amount of delivered heat to maintain a desired set point temperature in a home. The data that were used for this analysis include hourly whole home electricity consumption from smart meters and heat pump energy consumption data from the submetering. As such, hourly energy consumption is used as a proxy for average hourly heating load or heating demand (i.e., delivered heat).

The heating demand of the home at the code specified design temperature was estimated by averaging the hourly whole home electricity consumption during the coldest hours of the measurement period. Since the coldest hours typically occur at night, base loads were assumed to be negligible, which results in underestimating the percent contribution of delivered heat by the heat pump.[31] The delivered heat from the heat pump was estimated by taking the average hourly consumption of the heat pump and multiplying by the average COP at the design temperature. The average hourly whole home energy consumption was then converted to delivered heat by adding the delivered heat from the heat pump that is not captured by the whole home energy meter (i.e., capacity of heat pump is greater than metered consumption since the COP of the unit is greater than 1). The delivered heat from

[30] https://vancouver.ca/files/cov/guidelines-energy-modelling.pdf (assessed May 2020)

[31] The coldest temperature during the measurement period for Kelowna was -16 °C, which is one degree Celsius from the design temperature of -17 °C, therefore a regression analysis was used to extrapolate the consumption at the design temperature for KEL01.

the heat pump is compared to the whole home heating demand to estimate the percent of heat provided by the heat pump, shown in TABLE 16.

| TABLE 16 – SUMMARY OF ESTIMATED FRACTION OF HEATING ENERGY PROVIDED BY HEAT PUMPS AT DESIGN TEMPERATURE | | | |
|---|---|---|---|
| Participant ID | KEL01a | KEL01b | VIC03 |
| Design Temperature | -17˚C | -17˚C | -4˚C |
| Estimated Whole Home Design Heating Demand (kW) | 4.3 | 2.2 | 3.9 |
| Estimated Heat Pump Delivered Heat (kW) | 2.4 | 2.1* | 2.7 |
| Heat Pump % of Delivered Heat | 56% | 96% | 71% |

**HOT2000 modelled peak heat load of 2.13kW.*

Results show that the percent of delivered heat from the heat pumps varied between 56% and 96% for this small sample size of three participants. Additional communication with the KEL01 homeowner confirms that KEL01a heat pump was intended to provide heating for the living room and kitchen area of the home, whereas KEL01b was intended to provide nearly all the heating in the home (with the exception of the bathroom which is equipped with a baseboard heater). Therefore, the heat pumps for these homes appear to be providing an adequate percent of delivered heat based on their design intent. No conclusions could be drawn for VIC03 as further information regarding the design intent of the heat pump was not provided.

It is important to note that, with the exception of KEL01b, no formal heat pump sizing calculations were provided for any of the homes in this study. Most homeowners could not confirm the extent to which their heat pump had been sized. Some mentioned that an installer or salesperson either measured the total floor area, evaluated existing duct sizing, referenced and matched the size of previous heating equipment, asked questions over the phone, and/or simply arrived with a quote upon their initial site visit.

## 3.4 Additional Findings

This section lists additional findings and general observations from the initial site visit and equipment instrumentation.

### *Heat Pump Installation Observations*

Based on field measurements, the distance between indoor and outdoor units varied with 38% of units between 1 to 5m from each other, 54% between 6 to 10m and 8% between 11 to 15m, respectively. As noted in installation requirements of CSA standard C273.5-11, refrigerant runs are to be as straight and short as possible. Some home designs allowed for very simple refrigerant runs, while others required lines to bend in multiple locations and span longer distances, with greater potential for losses between the indoor and outdoor units. In addition, most refrigerant lines exposed to the outdoor environment were insulated, though six units (23%) that were noted as partially insulated.

### *Occupant Feedback*

During the initial site visit questionnaire session, some participants expressed that they were somewhat unsatisfied with their heat pumps, including the following comments:

- → Two participants noted that their system is good except their space occasionally gets too hot.
- → One participant noted that their living room and bedroom were not being heated sufficiently, and that an additional electric resistance coil had to be installed on both of their outdoor units because of excessive use of defrost. They mentioned, "the heat pump can't keep up", referring to two incidents of refrigerant leakage and the requirement for valve replacement and maintenance.
- → Another participant mentioned that, during the winter months prior to this study, that ice build up had formed on the fan of the outdoor unit.
- → Some felt that their units were undersized for the zone they were trying to condition.
- → One participant said that installers came back several times for issues with the outdoor unit, and after 13 months of the unit working improperly, they discovered that roughly 70% of the refrigerant had leaked. Once the valve was replaced and refrigerant line was refilled, they said performance had improved significantly.

Despite some heat pump issues expressed by participants during the questionnaire period, many homeowners were very satisfied with the results of their system, and made the following comments:

- → Some participants complimented the aesthetics, acoustics and overall performance of their heat pump.
- → One participant mentioned that their heat pump seems to be more efficient based on their utility bills.
- → Another noted that, while tracking their utility expenses, they found that they used 40-60% less electricity during the winter even though the winter season was abnormally cold.

## 3.5 In-Situ Instrumentation Techniques: Lessons Learned

Some instrument installation issues resulted in the exclusion of some measured heat pump systems from the study. This section is intended to share instrumentation lessons learned and thereby inform future heat pump monitoring research methodologies.

During the period of initial site visits, it was found that each ductless mini split indoor head had slightly different louvers for delivering supply air. Many of the louvers mechanically shut when they are off and open to an approximate 45˚ angle during operation. Some of the louvers also oscillate during operation or can manually be set to different angles. Therefore, it was difficult to adopt a universal supply sensor installation methodology. *Figure 3.31* shows two mini split indoor heads. The image on the right had a louver that allowed a large enough gap so the sensors' wiring (placed behind the louvers) would not damage the louver when fully closed. The image on the left, however, shows a unit that did not provide any gaps when the louver was fully closed. For this reason, it was decided to adhere the sensors to the rotating louvers.

*Figure 3.31 – two different strategies for mounting supply temperature sensors: adhered to oscillating louver (left – PRI01i) and adhered behind the louver (right – KEL01a).*

This method of adhering the sensors to the louvers, used on many sites, proved problematic as the rotating of the sensors may have also over time rotated the sensor head such that is was no longer in the direct stream of supply air. In some cases, we were also notified that the adhesion of the sensor had failed (likely due to the routine opening and closing of the louvers) and that it was hanging just below the louver. It is also important to note that the sensors had to be installed in a way that would allow homeowners to maintain the unit, such as cleaning the airstream filters, typically accessed on the front face of the unit.

*Figure 3.32* is an example of a sensor install that did not adequately measure the supply temperature during the monitoring period. Based on the metered system consumption, the unit consumed 3,431kWh during the study, which does not agree with the measured temperatures which suggest it was operating without providing or removing heat from the space (the maximum and minimum temperatures of the supply were relatively equal to return (i.e., ambient indoor conditions)). For comparison, *Figure 3.33* is an example of a system configuration that measured the expected variation in supply and return temperatures.

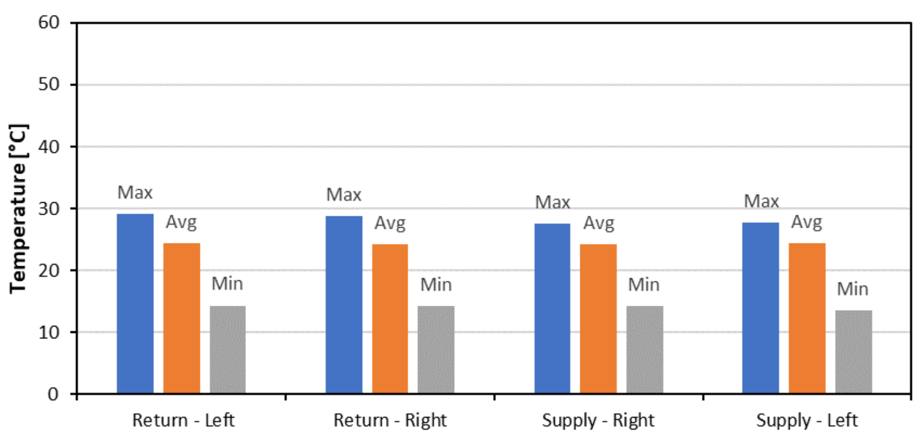


*Figure 3.32 – example of erroneous return and supply temperature measurements for complete monitoring period as a result of sensor placement (SAL01 - ductless mini split).*

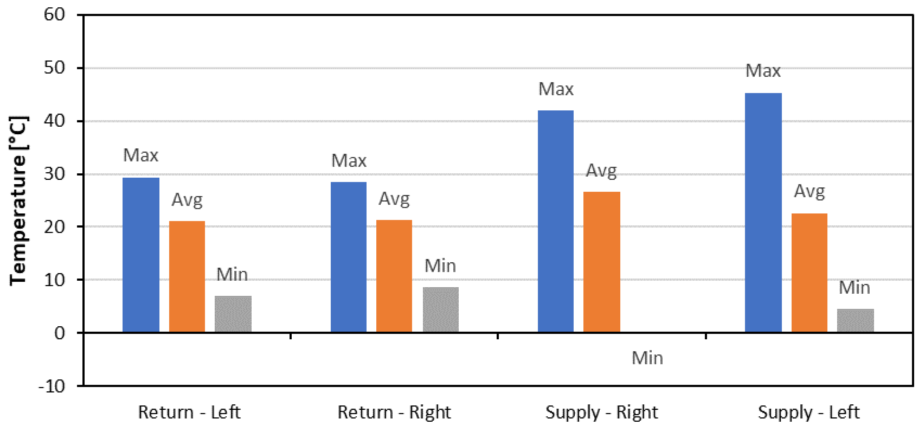


*Figure 3.33 – example of typical return and supply measurements for a complete monitoring period (KEL06ii - ductless mini split).*

Interestingly, a slight variation in left and right supply temperature measurements were also noted in some instances as a result of sensor placement. *Figure 3.34* is a right and left supply temperature plot for one unit throughout the monitoring period. The resulting differences are likely the result of sensor location as it is unlikely that the temperature difference across the supply air outlet would be so significant. Due to the large difference between the supply sensors in some cases, the original methodology to average the two sensors was found to be unsuitable. For cases with significantly different supply temperatures, an analysis was performed to attempt to use a sensor with relatively stronger correlation with system consumption (i.e., consistently measuring cooling and heating when compressor is running) over a less reliable sensor. If this was not possible, the heat pump unit was removed.

Given the variability that was found in the field when measuring the mini split units, it is recommended that, if possible, sensors be installed behind the louvers (i.e., within the mini split head). Sensors with thinner diameter wiring would facilitate this type of install.

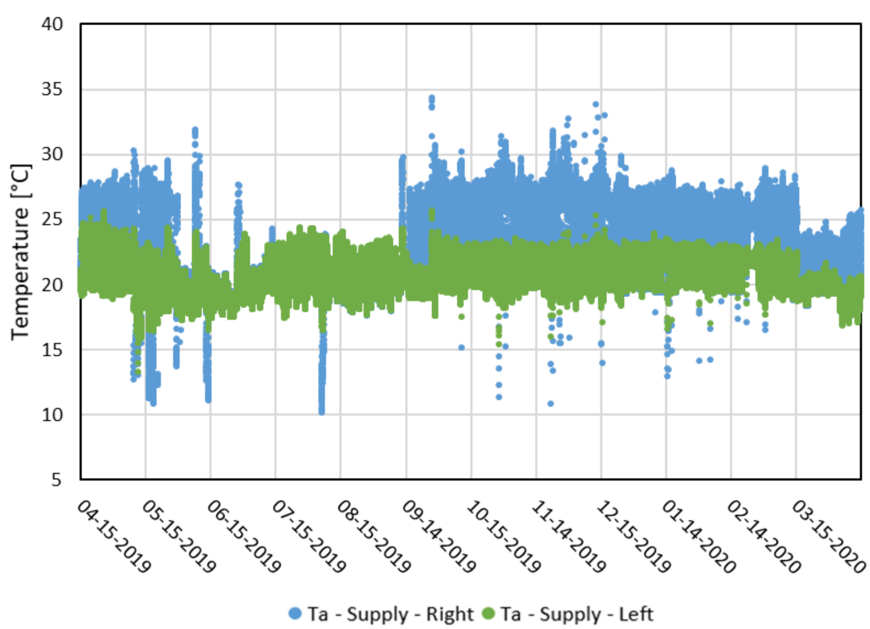


*Figure 3.34 – example of two supply temperature sensors reading significantly different results (VIC07 – ductless mini split)*

As noted in Section 2.3, residual heat from the coil was measured in some cases once the compressor and fan turned off, which led to false cooling periods to be registered after each heating cycle. Not all return temperature sensors captured residual heat and therefore we did not consider the additional heat that was distributed to the interior space while the fan and compressor were off. However, it would be possible to calculate this extra heat energy and add it to the capacity of the unit. Per cycle, the amount of extra heat may appear insignificant though when added up over a year it may amount to a slight increase in efficiency. For this reason, it is recommended that return temperature sensors be either placed far enough from the return louver so that no residual heat is captured, or install sensors close to the louver in order to intentionally capture the heat, depending on the accuracy of the unit capacity one is trying to achieve.

# 4 Key Findings

The intended outcomes of this study were to develop a clearer understanding of the performance of cold climate air-to-air heat pumps in Canadian climates; to evaluate design and installation considerations that may affect this performance, and to identify design and installation best practices that positively affect the performance of heat pump systems. Key findings are summarized below.

→ The average seasonal COP for cooling was estimated to be 5.0, 4.1 and 4.5 for ductless mini split, central single stage and central variable speed systems, respectively. Generally, the measured heat pumps appear to be performing with an average COP greater than 1 in cooling season for all outdoor conditions, even during extreme heat above 38˚C. Many participants are using heat pumps to cool the interior when outdoor temperatures are below typical interior temperatures. Cooling during periods when outdoor temperatures are below the average indoor temperature (~21˚C) causes significant fluctuation in COP for some units, likely as a result of system short-cycling, though not all exhibited this phenomenon. Training homeowners on the strategies and benefits of passive cooling strategies (i.e., natural ventilation) could reduce hours of heat pump operation during mild outdoor temperatures.

→ The average COP for heating was estimated for the entire heating season and correlated with outdoor temperature. Results show that the overall average COP for all heat pumps is greater than 1, even down to -14˚C. However, two units were found to have a COP less than 1 at outdoor temperatures around 0˚C and below. For the poorest performing unit, it was found that heating capacity of the system drops below the energy demand around 0˚C and continues to drop as outdoor temperature gets colder. Based on conversations with the homeowner, there is reason to believe that leaked refrigerant may be responsible for the low performance.

→ The average seasonal COP for heating was estimated to be 2.4, 2.6 and 3.3 for ductless mini split, central single stage and central variable speed systems, respectively. In heating season, the average seasonal COP of central units was higher than ductless mini split units. Most of the central systems were also found to reduce their operation or stop heating between 3˚C and -5˚C; relying on a backup heating system to supplement the heating load at lower outdoor temperatures.

→ Evidence of short-cycling (i.e., cycles less than 5-8 minutes) during typical heating periods was found in 33% of units, suggesting some units may be oversized. Oversizing units can cause the conditioned space to quickly reach its set point and shut off before the unit can reach an optimal efficiency, which can negatively affect the overall performance. During the initial site visit, participants were asked if they had access to any documentation related to heat pump sizing. With the exception of one case, participants had not received or seen any formal documentation to confirm that their units had been appropriately sized for their home.

→ An analysis of the backup heating demand for electric resistance backup heating coils in central systems was performed. Results show that the average backup heat consumption accounted for 22% of total space conditioning for the units above, and as much as 63% for one unit. Differentiation between electric resistance use for defrost or backup heat was not possible for central systems, though some inferences were made suggesting that more than half of backup heat was allocated to the defrost cycle rather than supplementary heating at extreme cold temperatures.

- → A defrost cycle analysis for ductless mini split systems was conducted. Results showed that defrost cycle energy accounted for less than 1% of total heat pump consumption during winter.
- → The backup heating and defrost analysis supports the inclusion of these impacts when testing or rating heat pump systems. Particularly for central systems, the exclusion of backup for supplementary heating and defrost energy consumption leads to an incomplete picture of overall system performance.
- → The volumetric flow rate of all indoor units was measured at each fan speed setting during the instrumentation of monitoring equipment. It was found that, on average, the measured volumetric flow rate of all measured indoor units was 36% lower than manufacturer listed rates. These results are consistent with previous studies. It was also found that some ductless mini split indoor heads were installed with limited ceiling clearance (less than 75mm), which may be restricting air flow to the return airstream.
- → Results show that current testing procedures produce rated volumetric flow rates that are often overestimating the rates typically found in in-situ environments. Updated rating procedures could provide standard guidelines to ensure units represent more typical as-installed conditions. For example, flow rates for central systems could account for flow restrictions that are likely to result from traveling through a duct system.
- → Impacts to energy and greenhouse gas emissions (GHG) and associated costs were evaluated. Of the 18 participants with available utility data, 12 (67%) experienced annual energy savings after the heat pumps were installed. Cases that used electricity as a primary heating source were separated and average savings were found to be 5650 kWh and $810 for the year-long monitoring period. Cases that used non-electric primary heating fuel sources varied significantly, where homes either saw and decrease or an increase in their electricity consumption and cost after the heat pump installation. In all, the average savings for these homes was still found to be 1520 kWh and $231 for the year-long monitoring period. The use of non-utility based fuel sources (e.g., propane, wood) before and/or after heat pump installation means that a full accounting of energy consumption and costs was not possible for most sites, and while overall energy and GHG savings were demonstrated, it is difficult to draw broader conclusions about energy and GHG savings potential from the field study.
- → Conditions and variables that made definitive conclusions challenging include small sample size, variations in home size and construction, different primary and backup heating sources, occupant behaviour, and instrumentation limitations.
- → Lessons learned based on some site monitoring intricacies discovered during this study were outlined to improve future in-situ heating pump instrumentation techniques, including suggested return and supply air temperature sensor placement.

# 5 Potential for Widespread Adoption in British Columbia

Based on the results of the study, the potential for widespread adoption of cold climate heat pump air-source heat pumps in British Columbia was evaluated.

Generally, the heat pumps in this study were found to have performed as expected within Climate Zones 4 and 5, though an estimate of heat pump performance in colder climate zones of British Columbia is not feasible based solely on the results of the current study. As noted in Section 3.2.3, a poor correlation was found between system performance and outdoor temperature; likely the result of competing variables found throughout the study. In addition, analysis suggests that that factors such as system type, installation quality, occupant behaviour, system operations, and more can affect the performance of units. The impacts of these factors are largely unknown for systems operating in Climate Zones 6, 7 and 8.

Despite the poor correlation between heat pump performance and average outdoor operating temperature exhibited in this study, it is widely understood that outdoor temperature does affect heat pump performance, particularly at extreme cold temperatures. Thus, a general assumption can be made that heat pump performance would be poorer, particularly for heating, in colder parts of British Columbia (e.g., colder average operating temperatures, greater reliance on backup heating, etc.). A similar study in colder parts of British Columbia would provide a more complete understanding of the potential for widespread adoption of heat pumps throughout the entire province.

When evaluating and quantifying the viability of air-source heat pumps as home retrofit options, both economic and environmental impacts should be considered. For example, per Section 3.3.1, homes that previously relied primarily on electricity for heating (e.g., electric resistance baseboards) exhibited fairly consistent electricity and cost savings. In contrast, a wide range of results were exhibited for homes that pervious relied on non-electricity for heating (e.g., oil and gas furnaces), and in some cases electricity and cost had increased.

From a homeowner's perspective, the economic benefits of retrofitting their home with a heat pump are generally greatest for homes that previously relied on electric resistance heating. However, the environmental benefits of retrofitting a home with a heat pump (though not well exhibited in this study) are generally greatest for homes that previously relied on non-electric sources to heat their home, given that the emissions factor ($kgCO_2e/kWh$) of natural gas, for example, is roughly 17 times higher than electricity in British Columbia.[32]

In all, this study suggests that there is potential for widespread adoption of heat pumps in British Columbia, although since the analysis was limited to homes in Climate Zone 4 and 5, further studies investigating the in-situ performance of heat pumps in colder climates is recommended prior to adoption in these regions.

[32] https://vancouver.ca/files/cov/guidelines-energy-modelling.pdf (accessed May, 2020).

# 6 Recommendations for Future Studies

Based on the findings from this study, the following are recommendations for further research:

→ It is recommended that a similar study be conducted in Climate Zone 6, 7 and/or 8 to provide a more complete understanding of heat pump performance and potential in these colder climates.

→ Specific to ductless mini split systems, future studies should explore the isolated impacts of the many variables noted in this study in a controlled laboratory setting:

  → How the duration of heating/cooling cycles affects the efficiency of conditioning cycles

  → How ceiling clearance of the indoor mini split head affects volumetric flow rate and premature re-circulation of conditioned air

  → How refrigerant line distances and bends affect heat pump capacity

  → How fully insulated vs. partially insulated refrigerant lines affect heat pump capacity at low temperatures

  → How unit sizing impacts system performance

→ Specific to central heat pump systems, future studies should focus on the development of the knowledge base around backup heating systems:

  → Could central heat pump systems deliver sufficient heat capacity at outdoor temperatures below their cut-off temperatures for backup heating?

  → What is the optimal cut-off temperature for backup heating in central heat pump systems?

We trust that the information and analysis presented above meets the intent of the final report. Please contact us if you have any questions or comments.

Yours truly,

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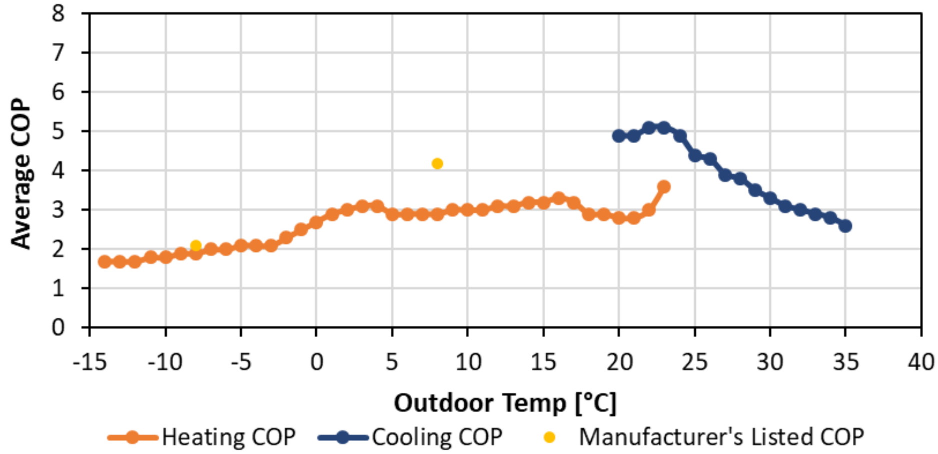
# Appendix A

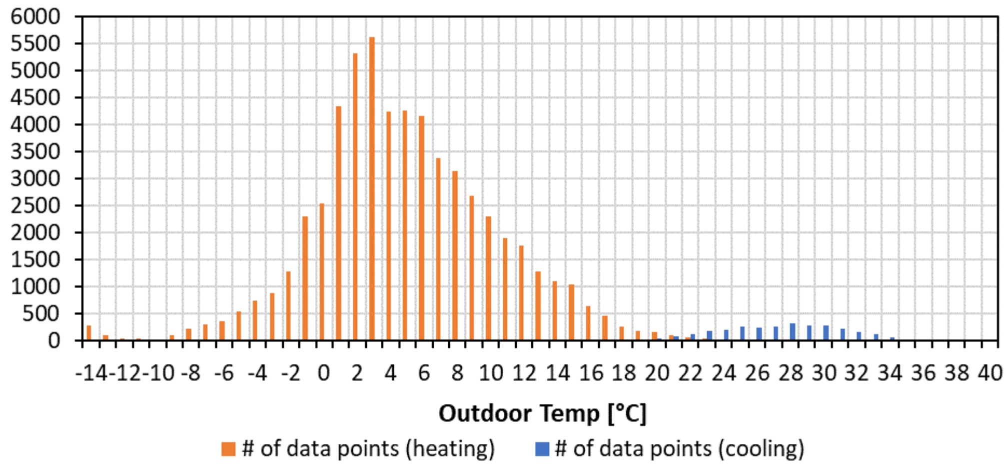
## Additional Plots

## Ductless Mini Split Systems

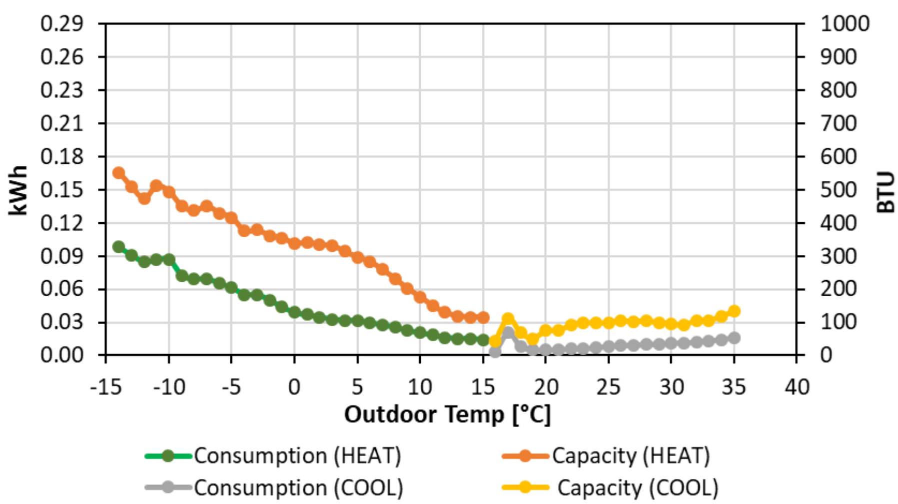
KEL01A – Ductless (Single Head) – Mitsubishi: MUZ-FH12NAH | MSZ-FH12NA



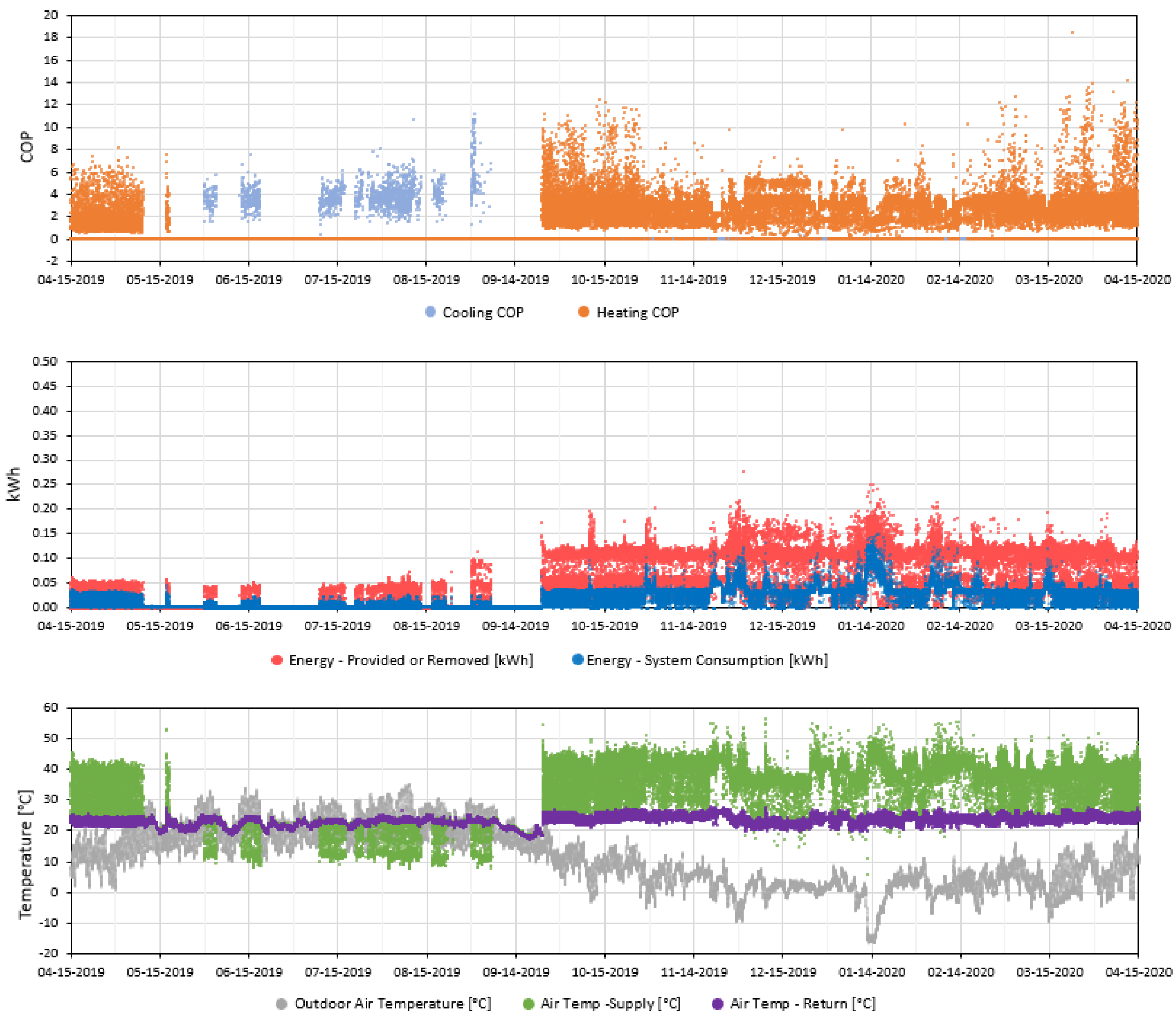

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



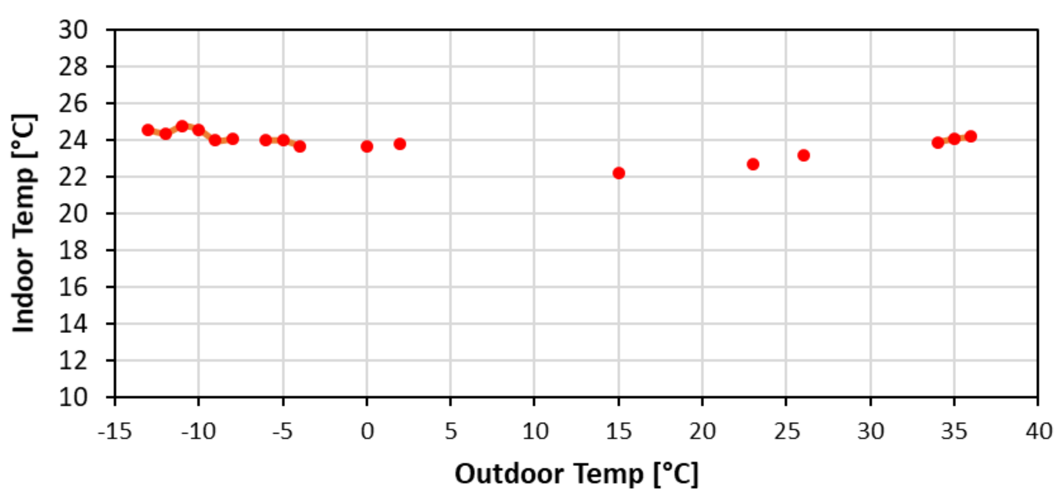

*Total number of heating and cooling data points throughout monitoring period.*



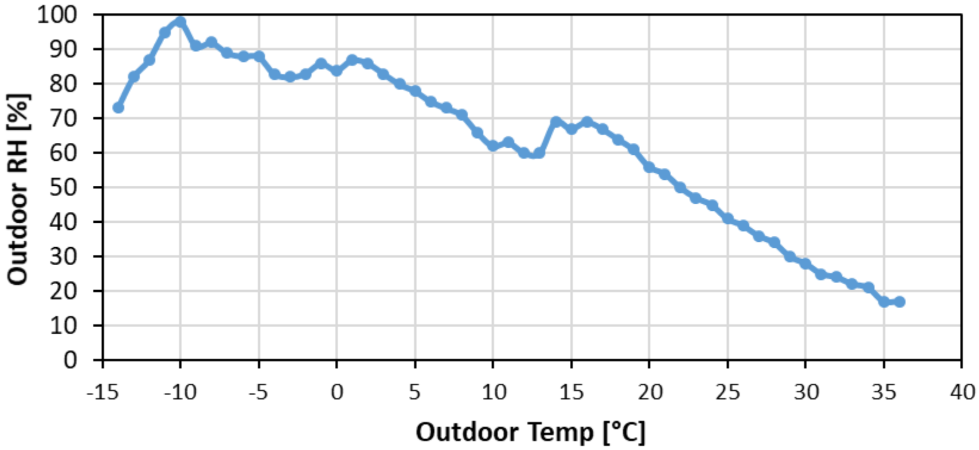

*Average system consumption and capacity for monitored outdoor temperature range.*



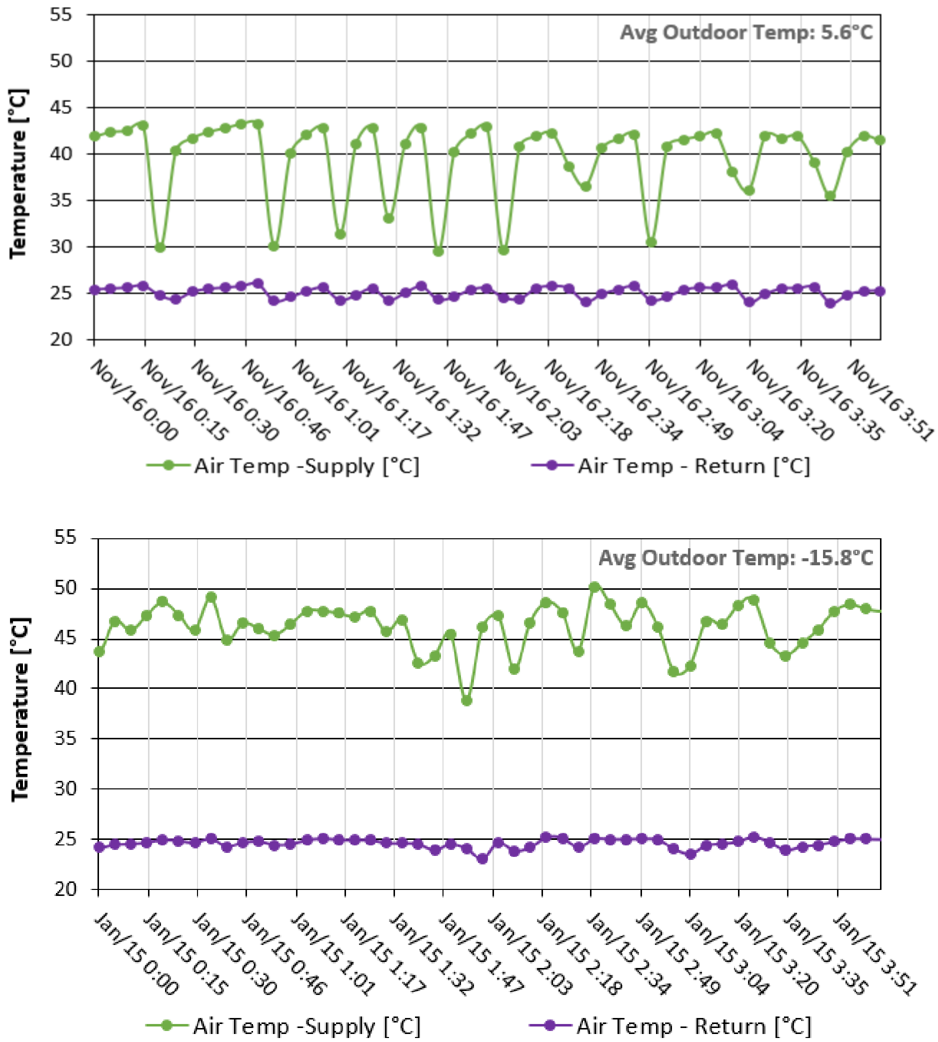

*Measured variables and corresponding COP for monitoring period.*



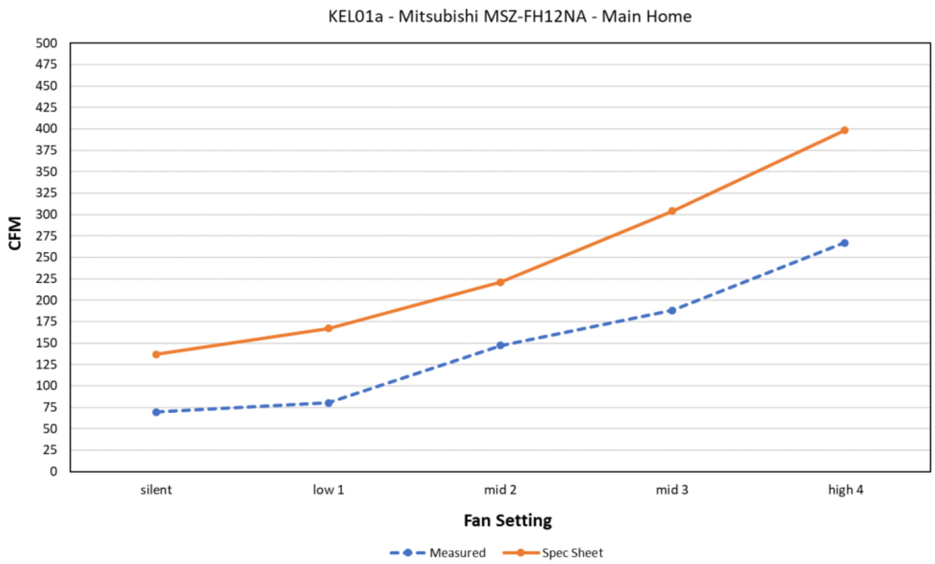

*Average indoor air temperature (return air) for monitored outdoor temperature range.*




*Average outdoor relative humidity for monitored outdoor temperature range.*

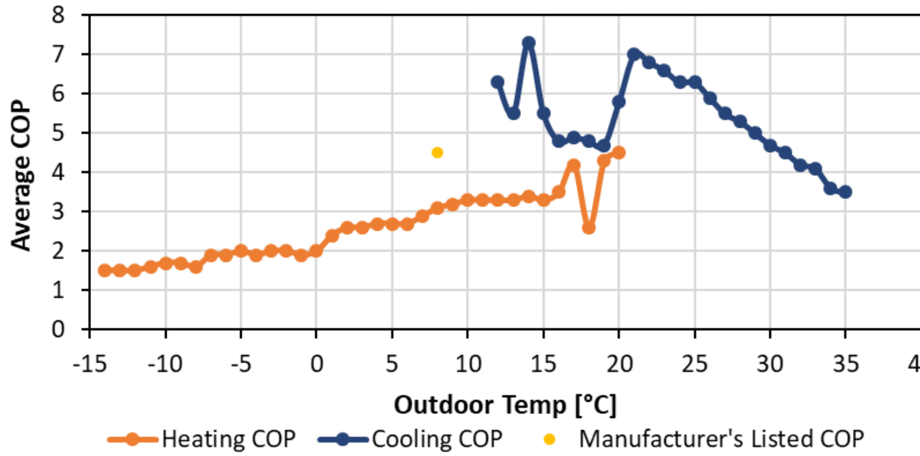



*Sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*

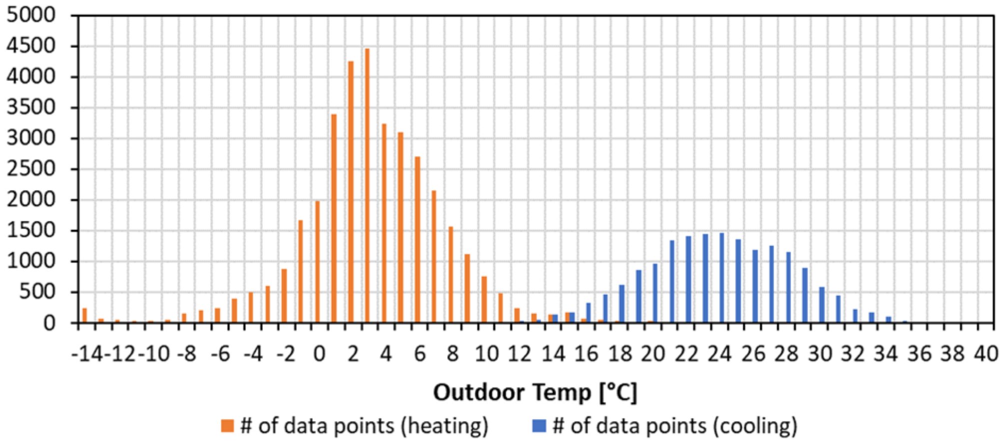



*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

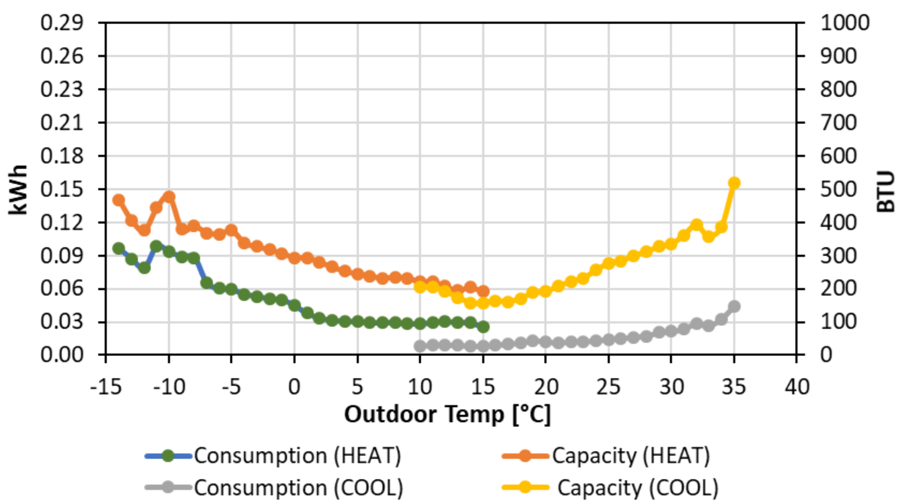
KEL01B – Ductless (Single Head) – Mitsubishi: MUZ-FE09NAH | MSZ-FE09NA



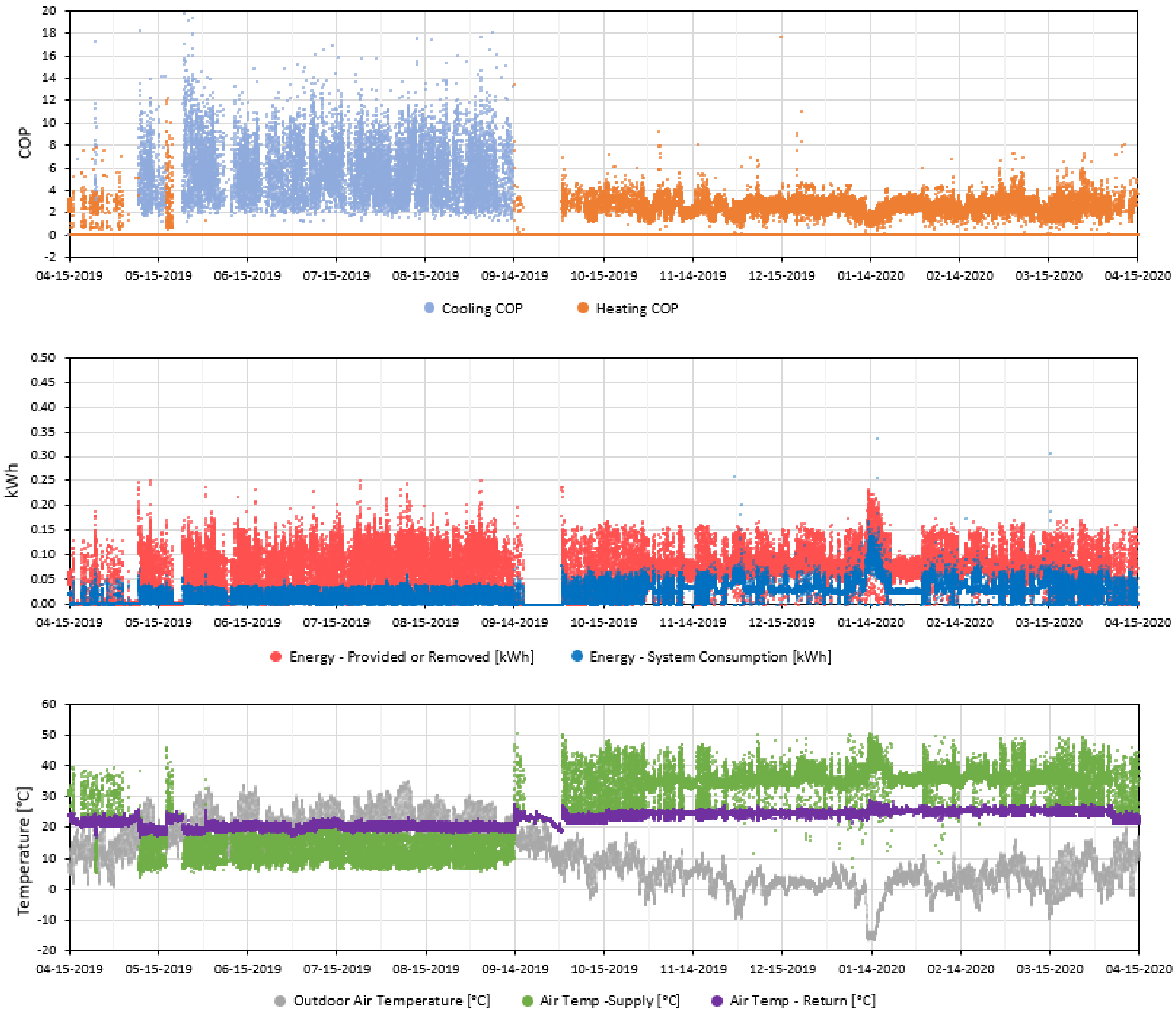

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



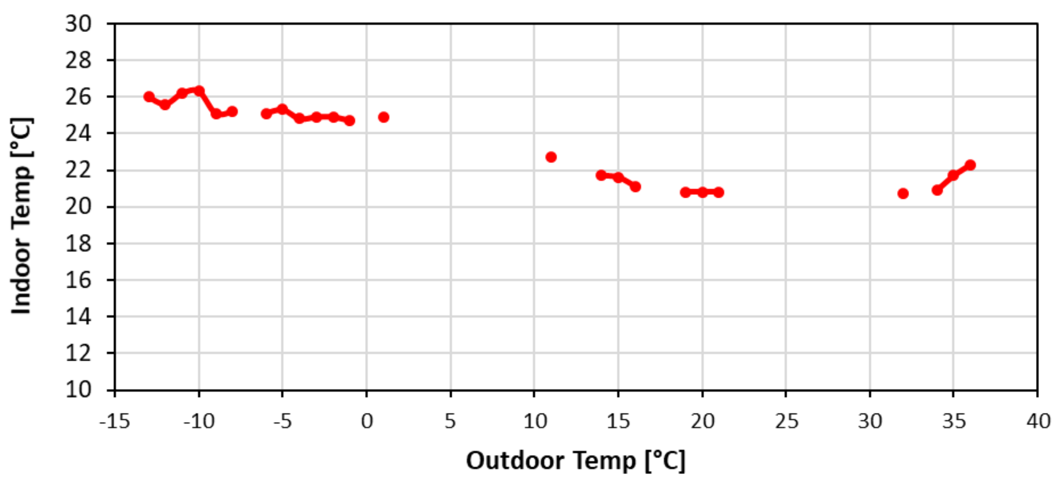

*Total number of heating and cooling data points throughout monitoring period.*



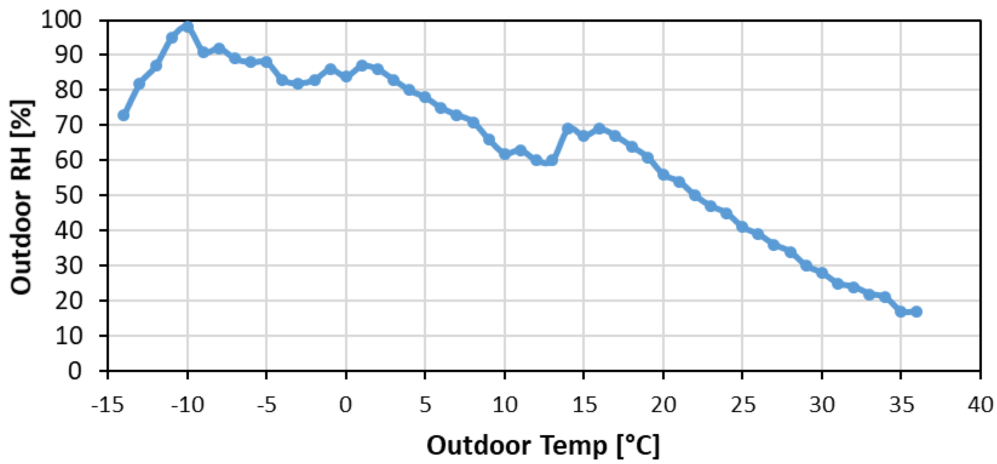

*Average system consumption and capacity for monitored outdoor temperature range.*



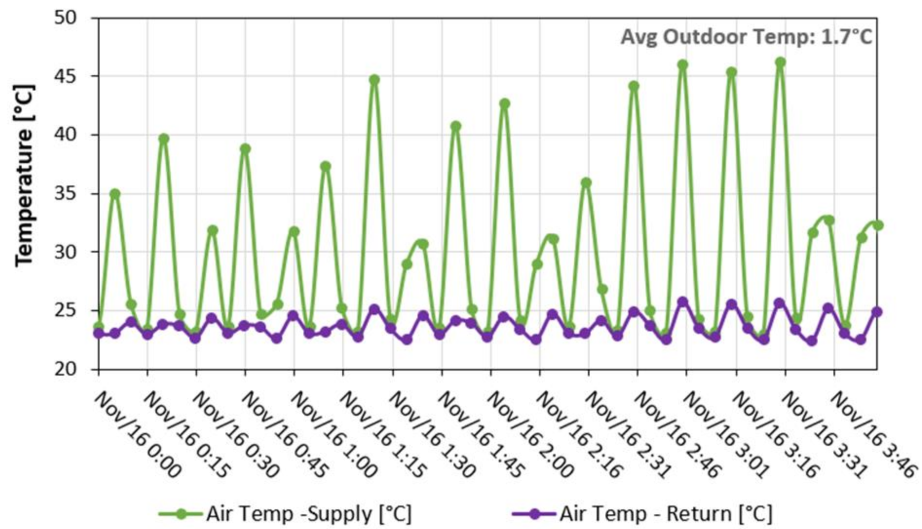

*Measured variables and corresponding COP for monitoring period.*

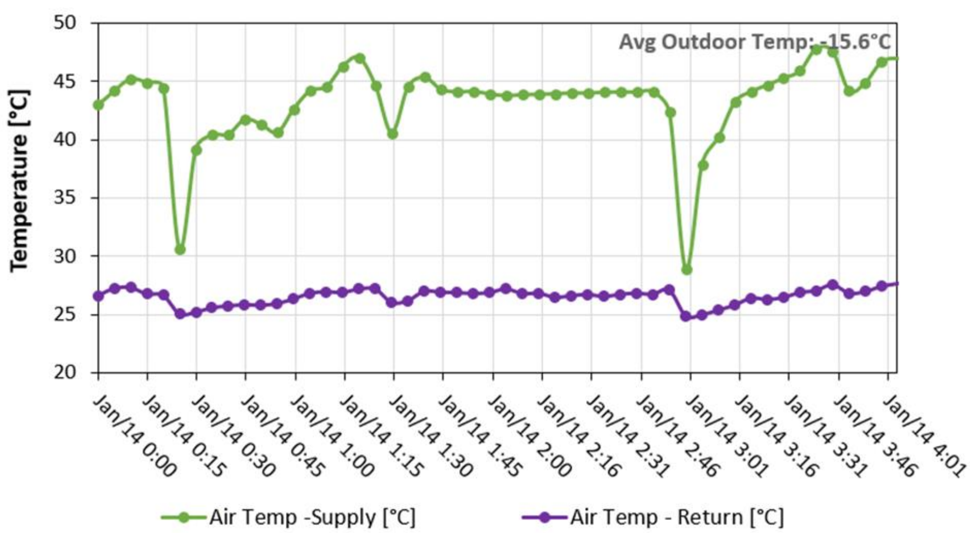



*Average indoor air temperature (return air) for monitored outdoor temperature range.*

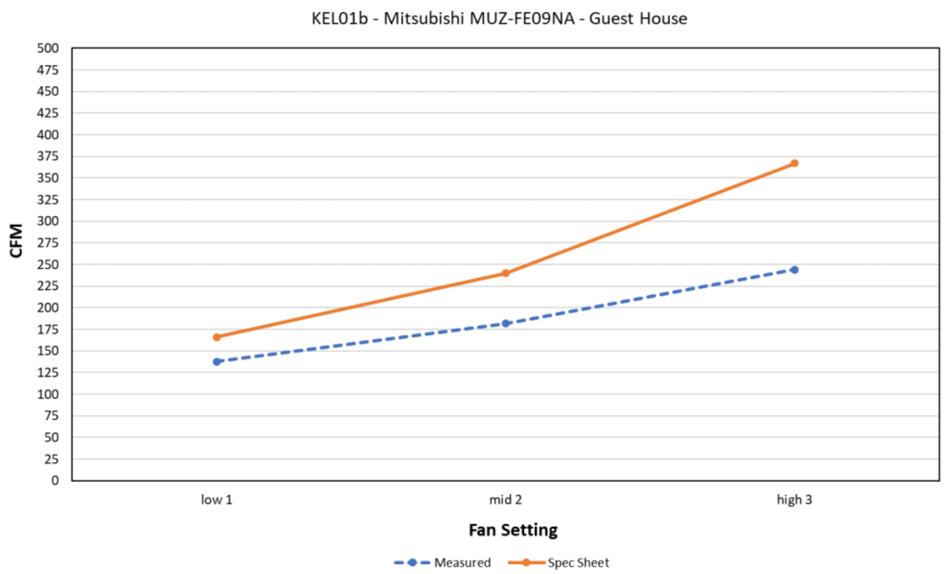



*Average outdoor relative humidity for monitored outdoor temperature range.*



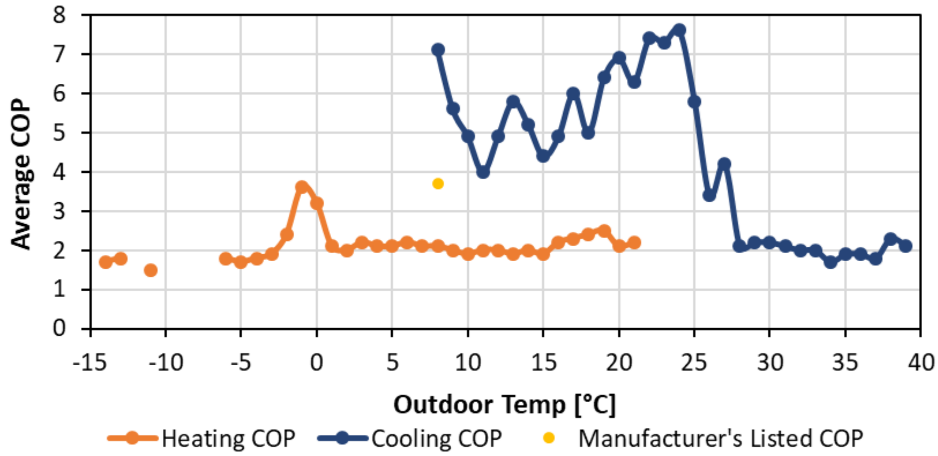




*Sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*

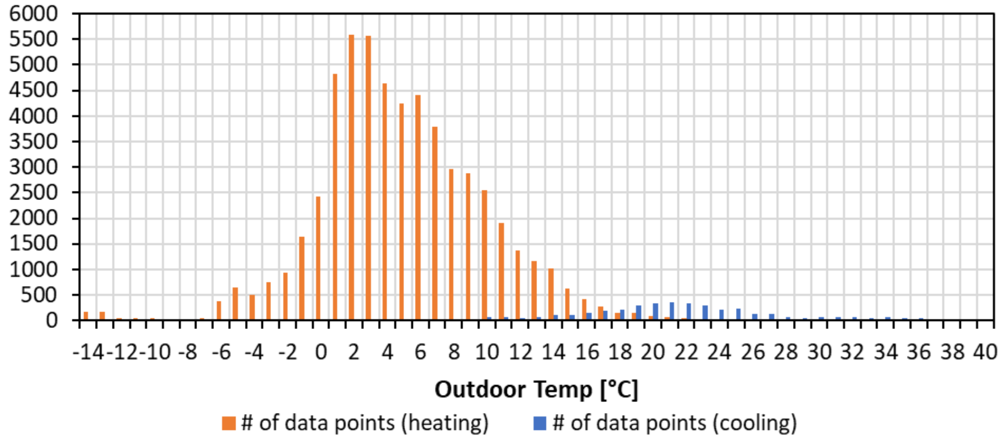



*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

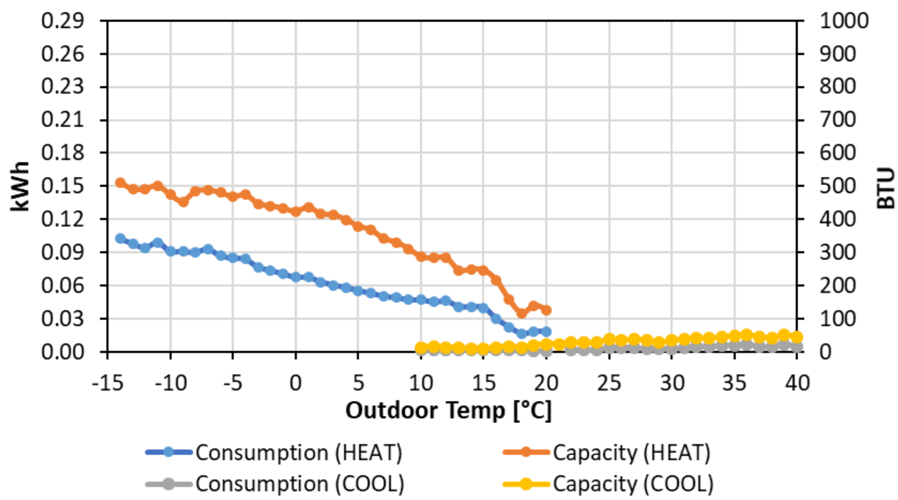
KEL03 - Ductless (Multi Head) - Daikin: 3MXS24RMVJU | CTXG18QVJUW | CTXGO9QVJUW



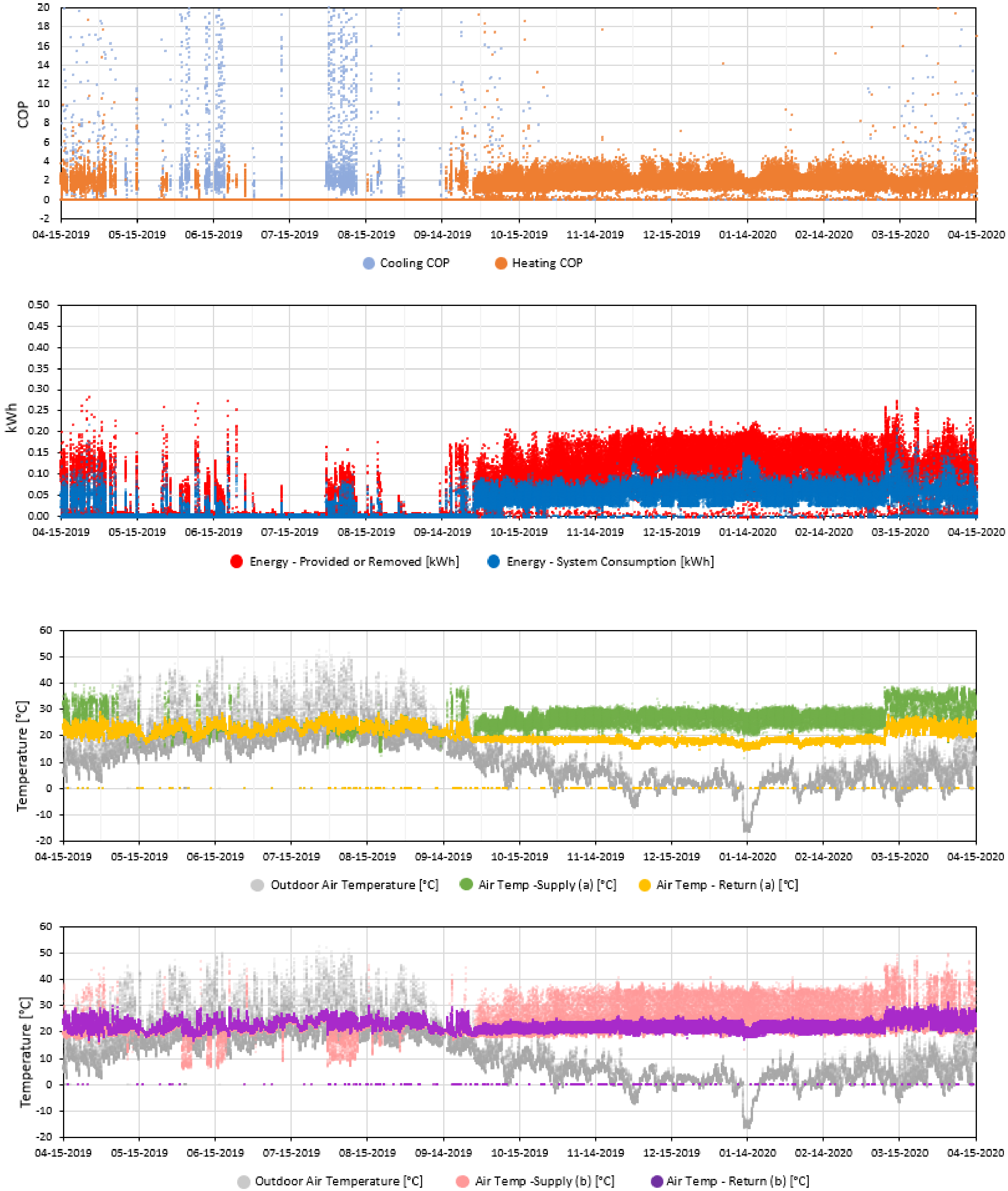

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



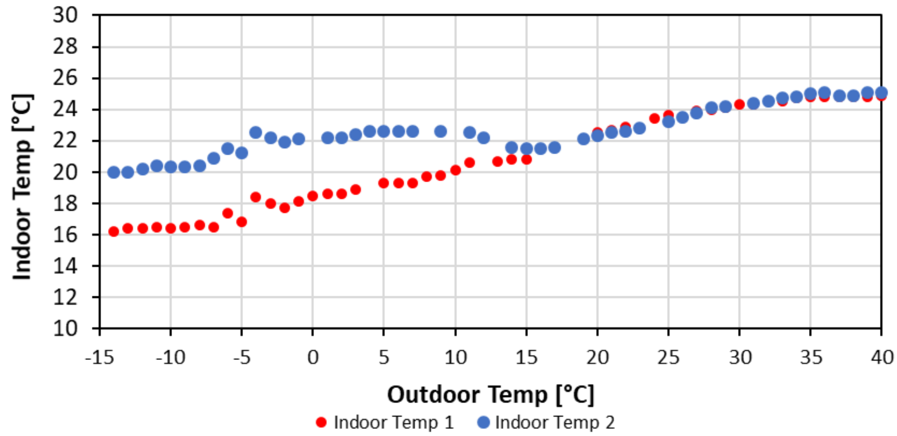

*Total number of heating and cooling data points throughout monitoring period.*



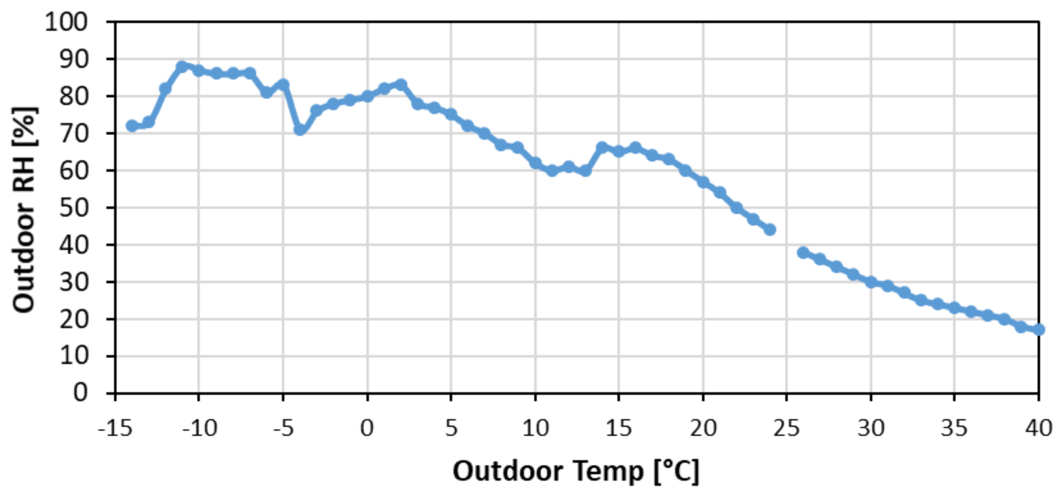

*Average system consumption and capacity for monitored outdoor temperature range.*



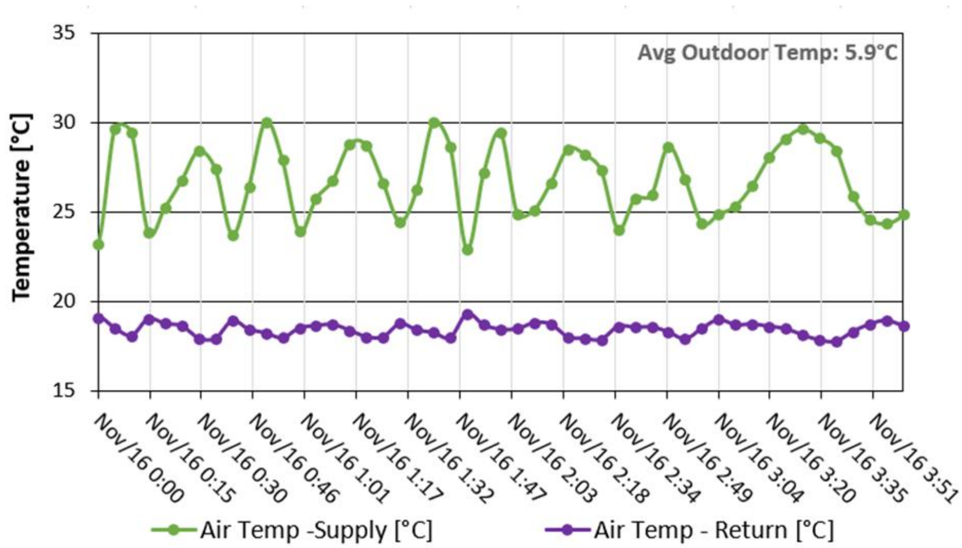

*Measured variables and corresponding COP for monitoring period.*

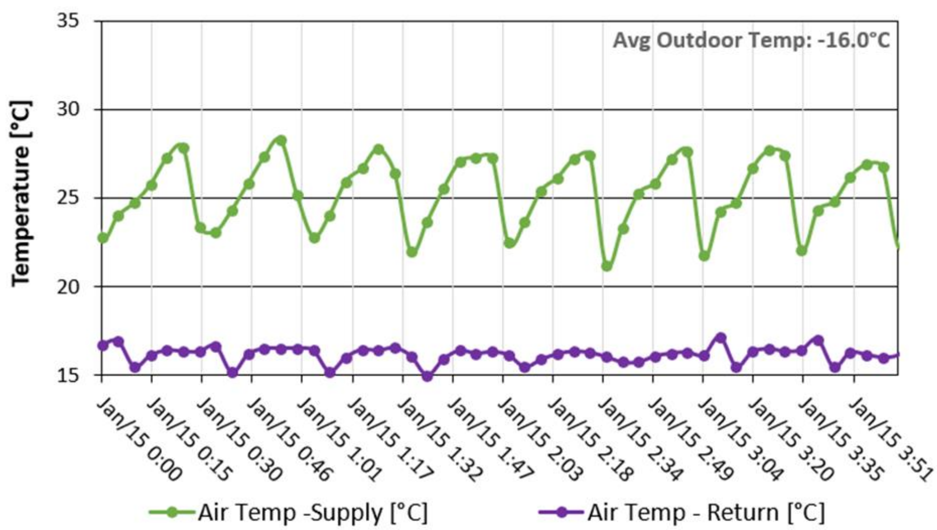



*Average indoor air temperature (return air) for monitored outdoor temperature range.*

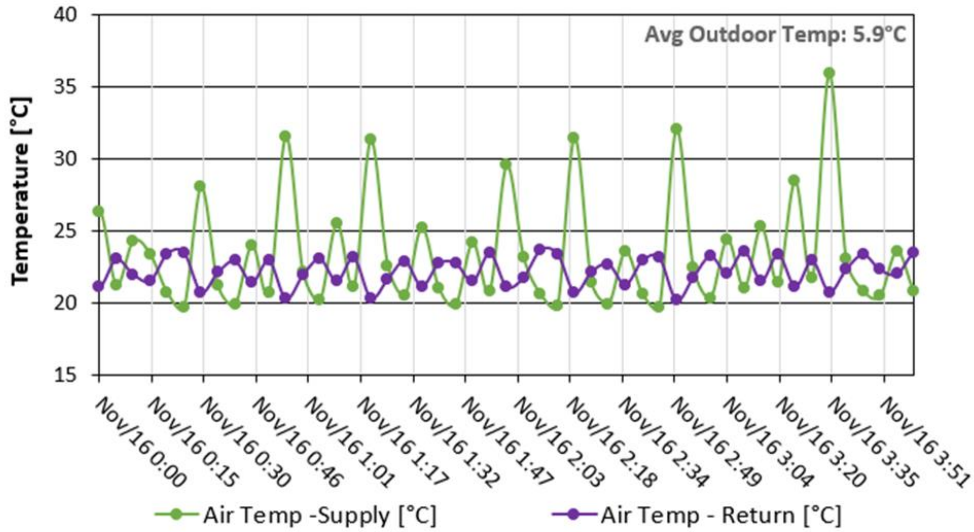


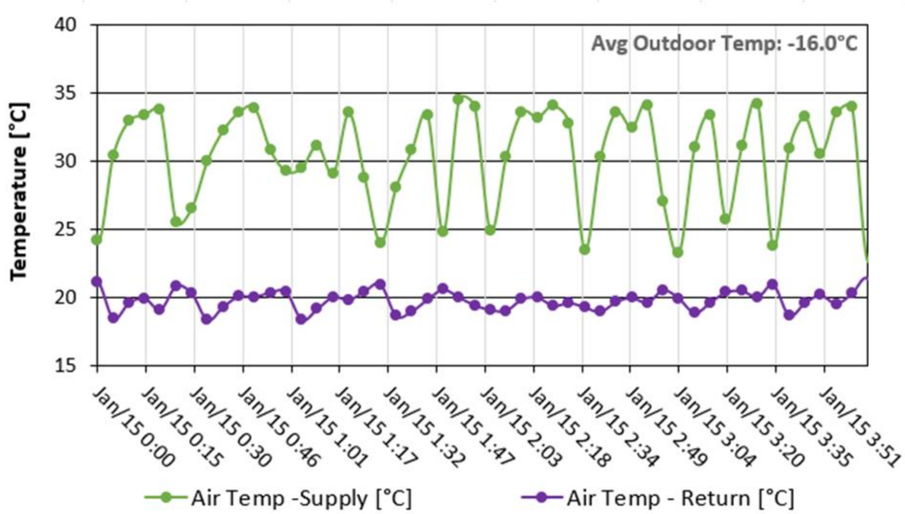

*Average outdoor relative humidity for monitored outdoor temperature range.*



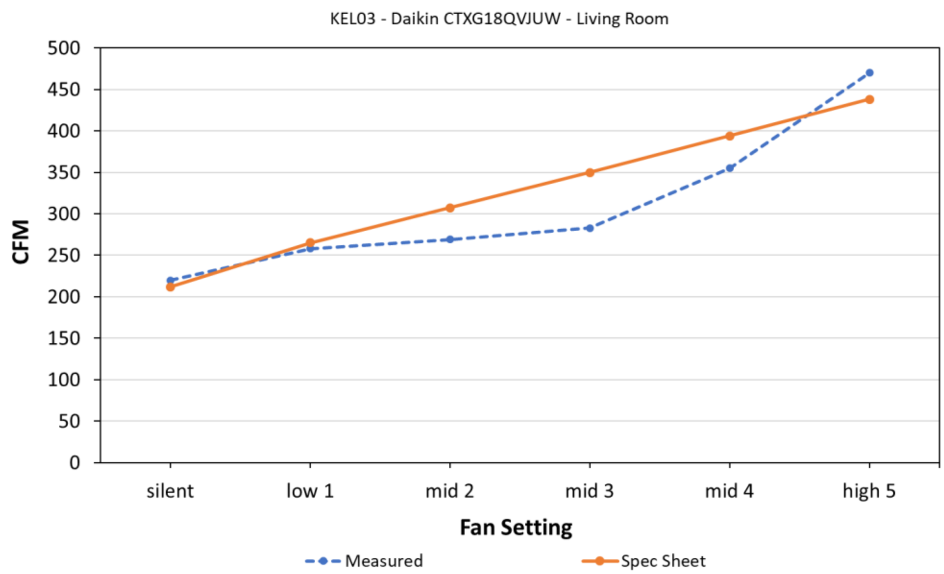



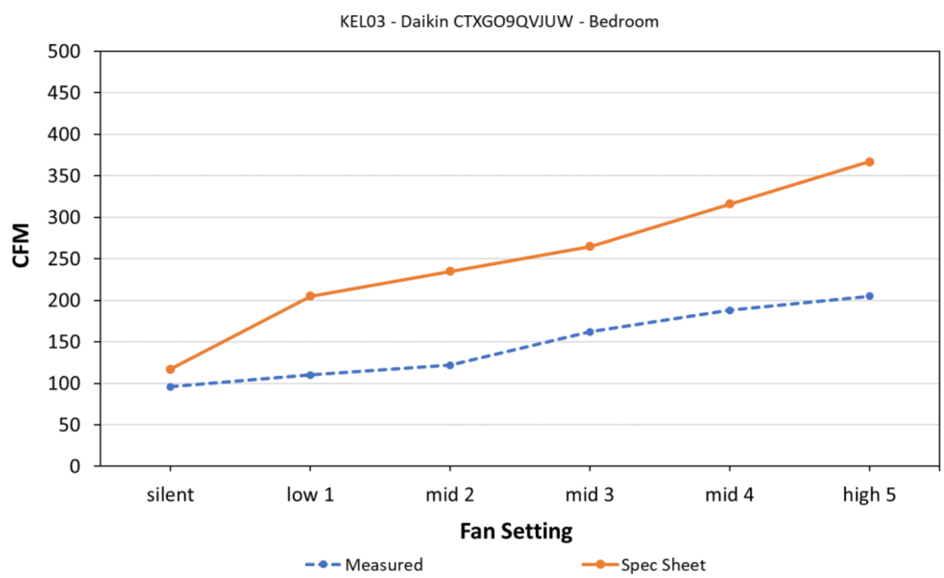

*Unit A: sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*



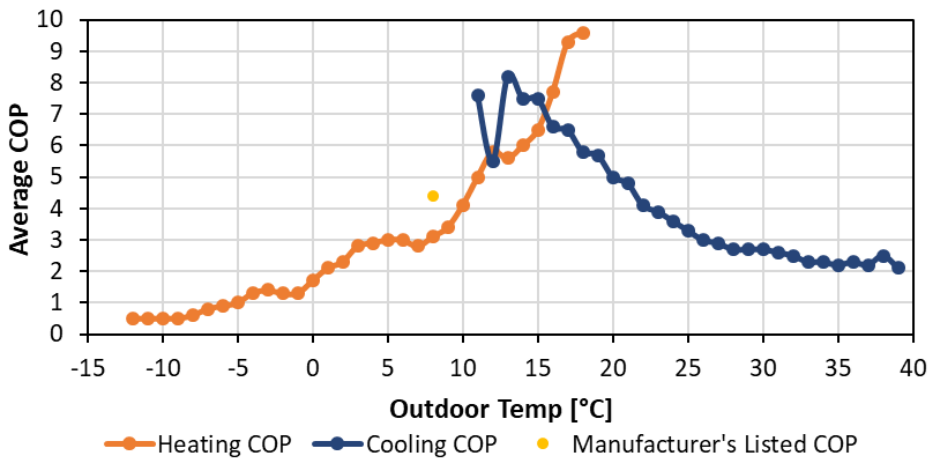

*Unit B: sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*



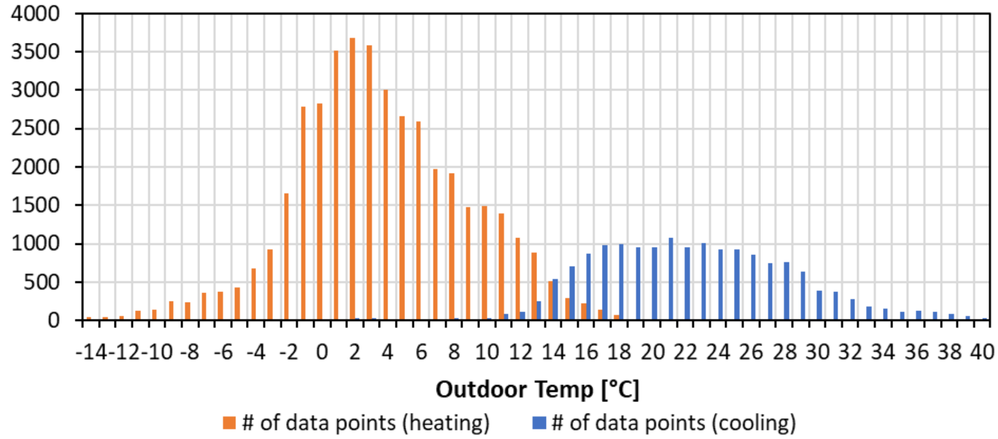




*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

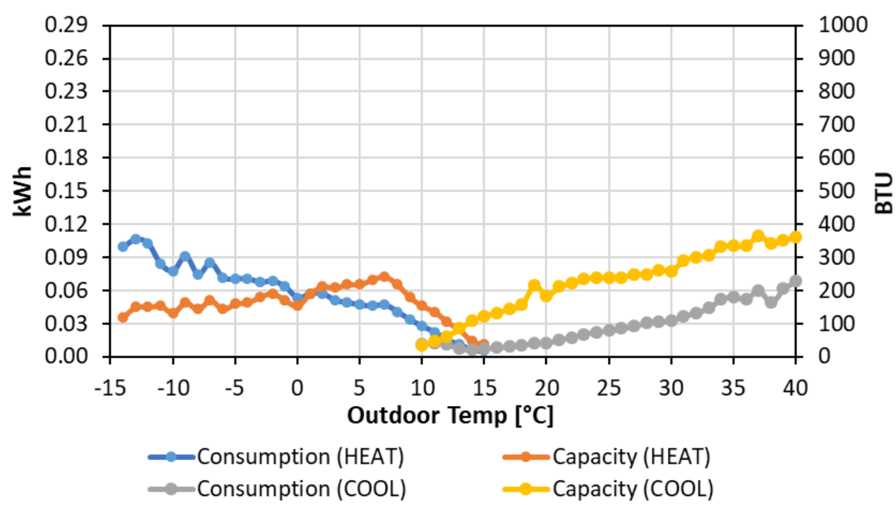
KEL06i – Ductless (Multi Head) – Daikin: 2MXS18NMVJU | FTXS12LVJU | FTXS09LVJU



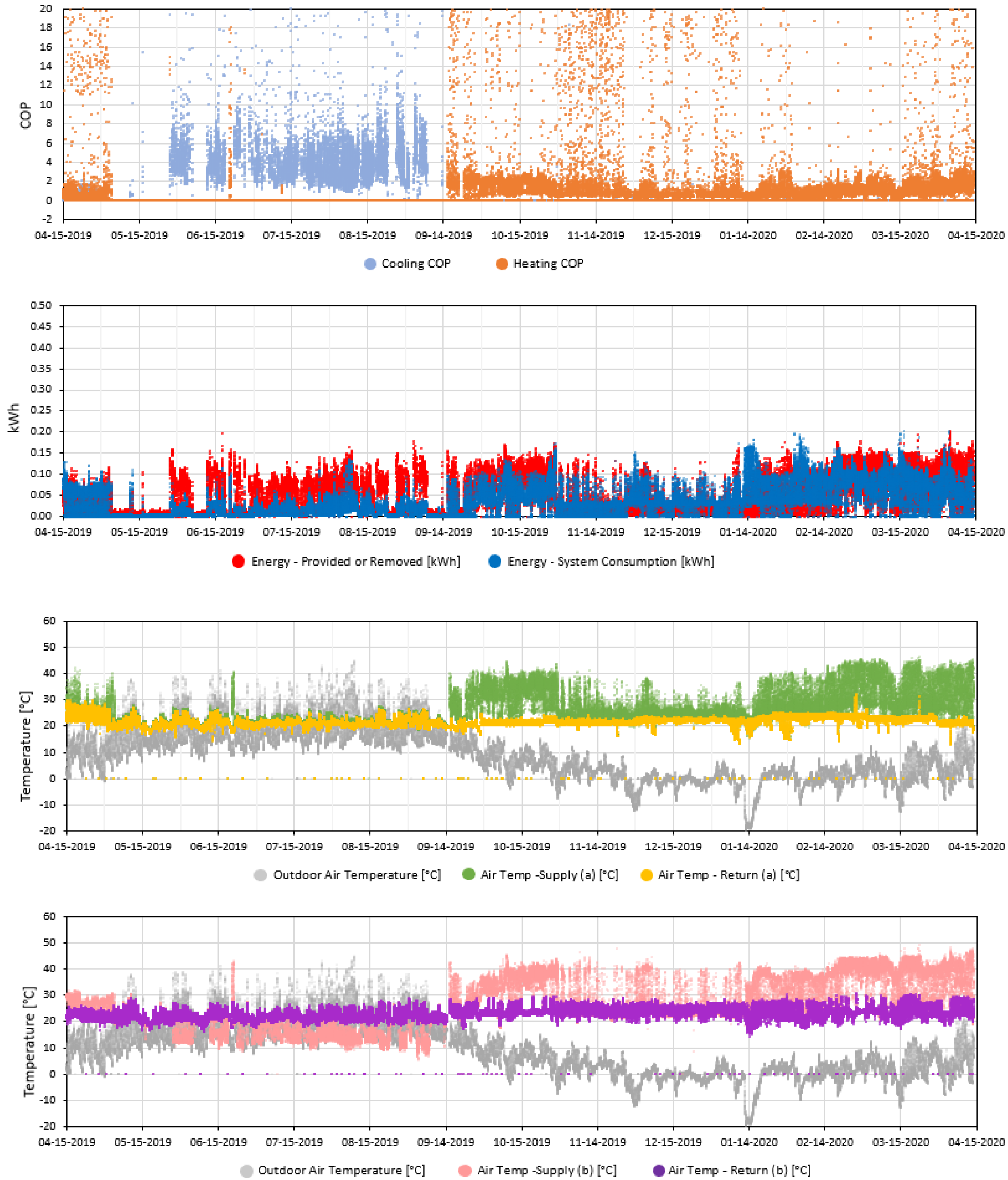

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



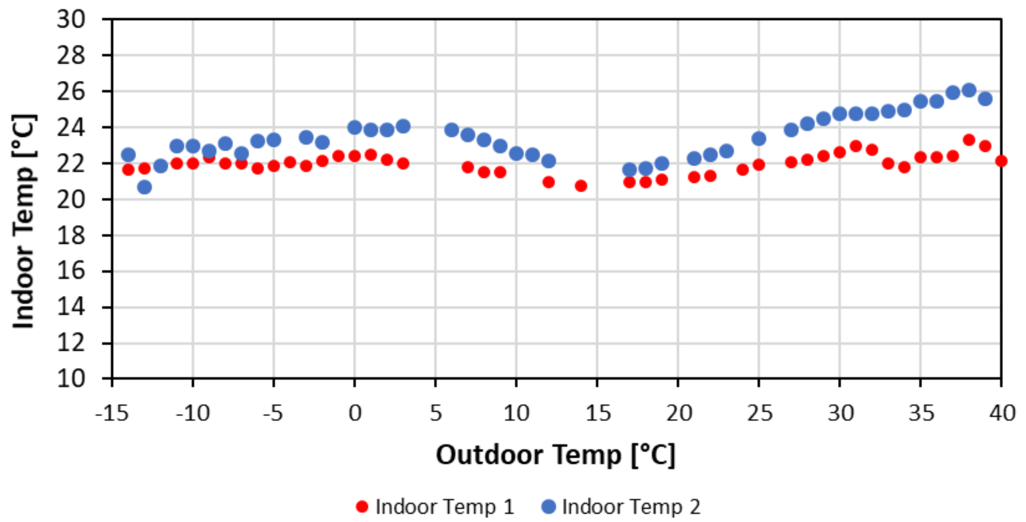

*Total number of heating and cooling data points throughout monitoring period.*



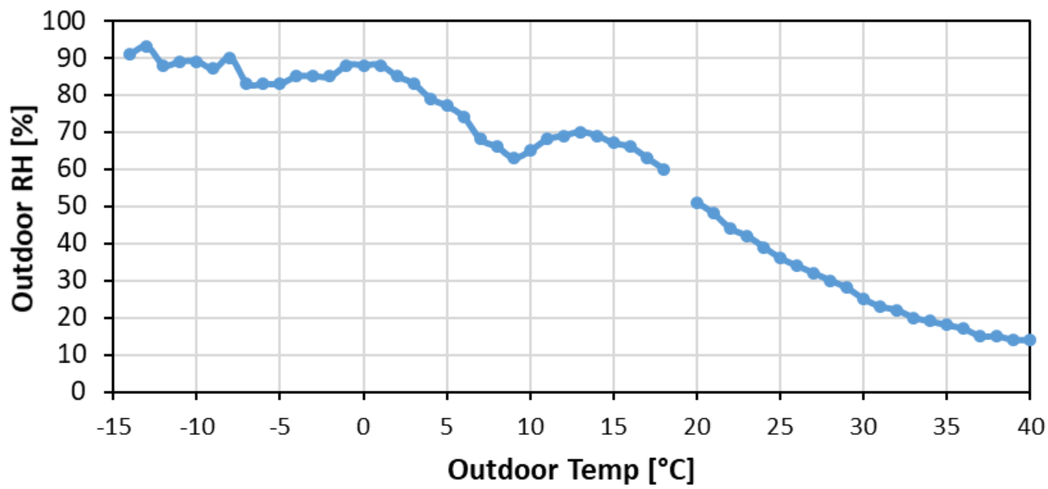

*Average system consumption and capacity for monitored outdoor temperature range.*



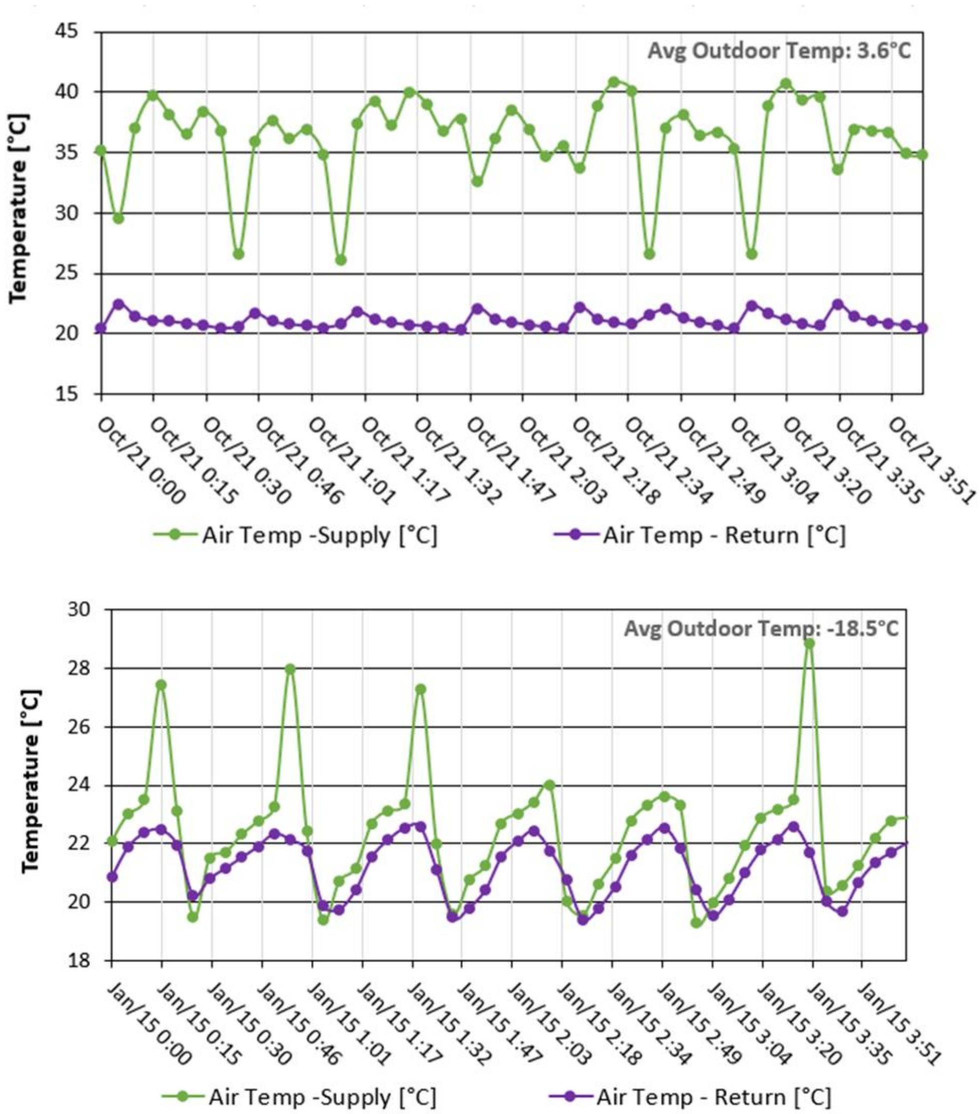

*Measured variables and corresponding COP for monitoring period.*



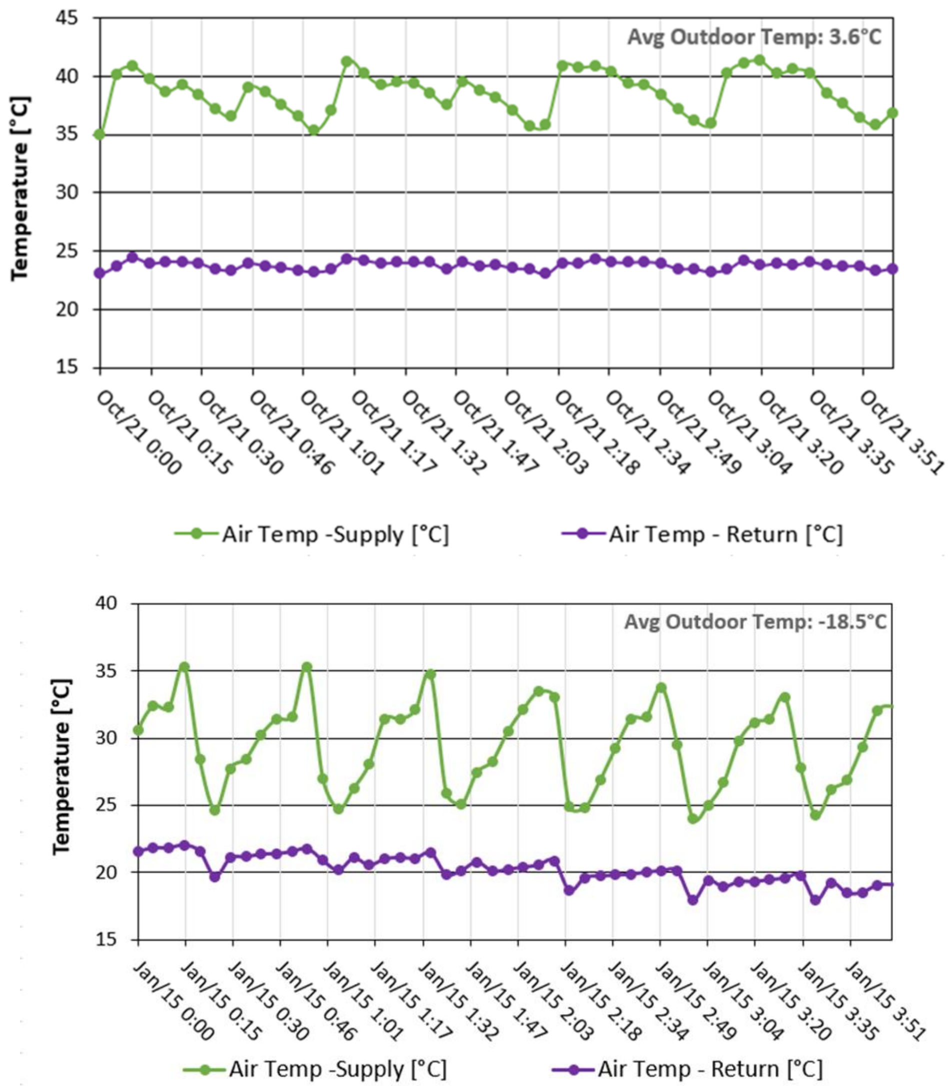

*Average indoor air temperature (return air) for monitored outdoor temperature range.*



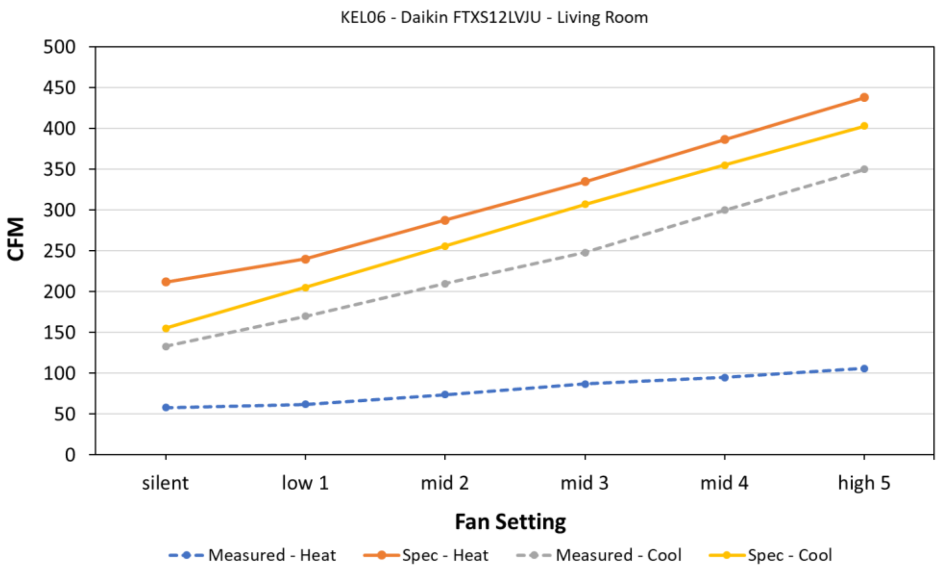

*Average outdoor relative humidity for monitored outdoor temperature range.*

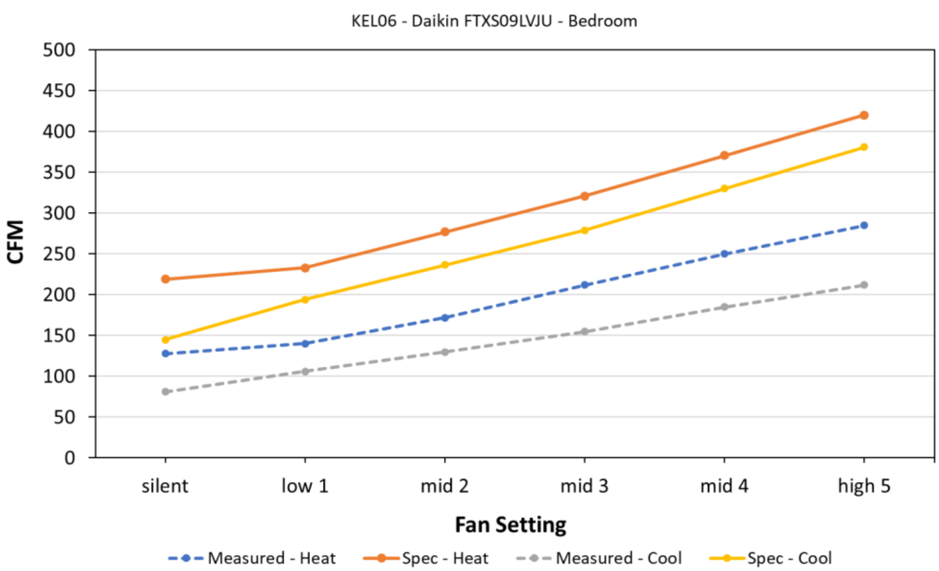



*Unit A: sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*



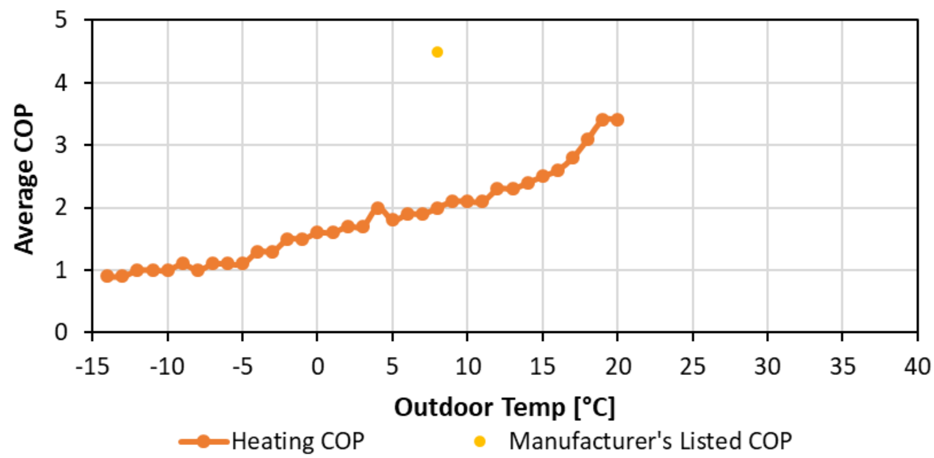

*Unit B: sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*



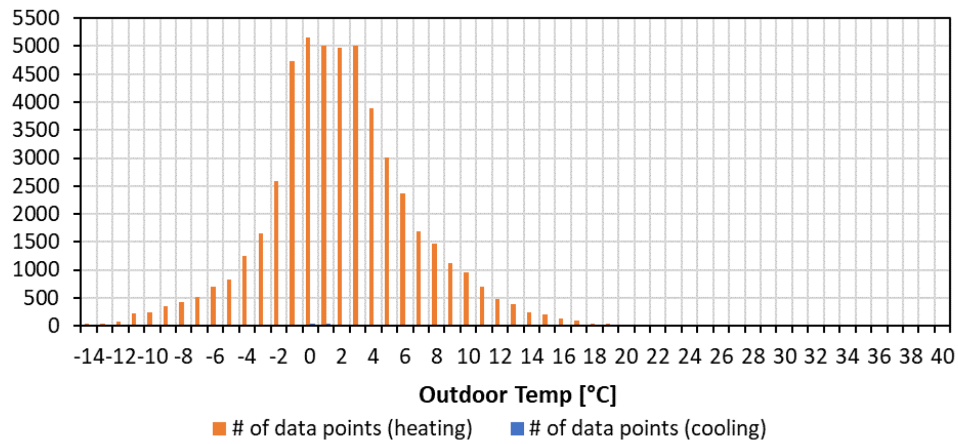




*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

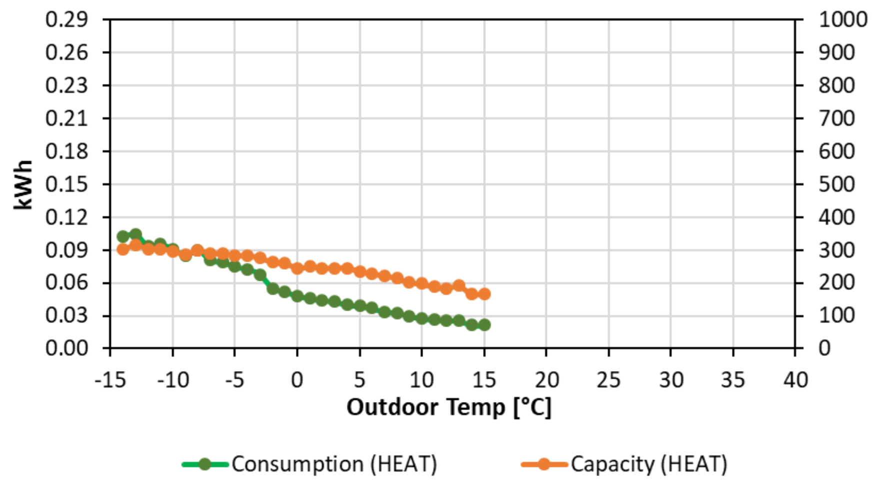
KEL06ii – Ductless (Single Head) – Daikin: RXS12LVJU | FTXS12LVJU



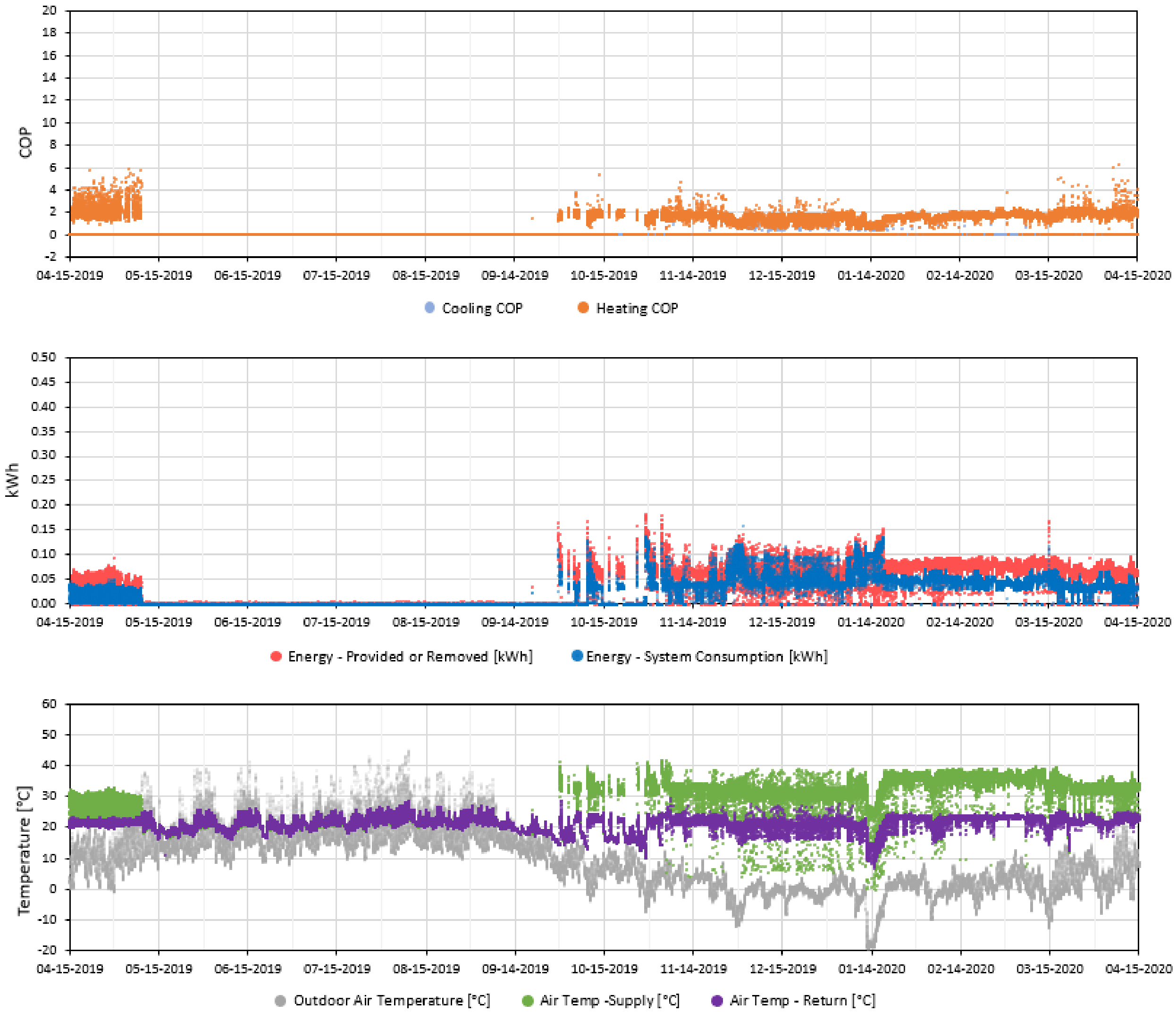

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



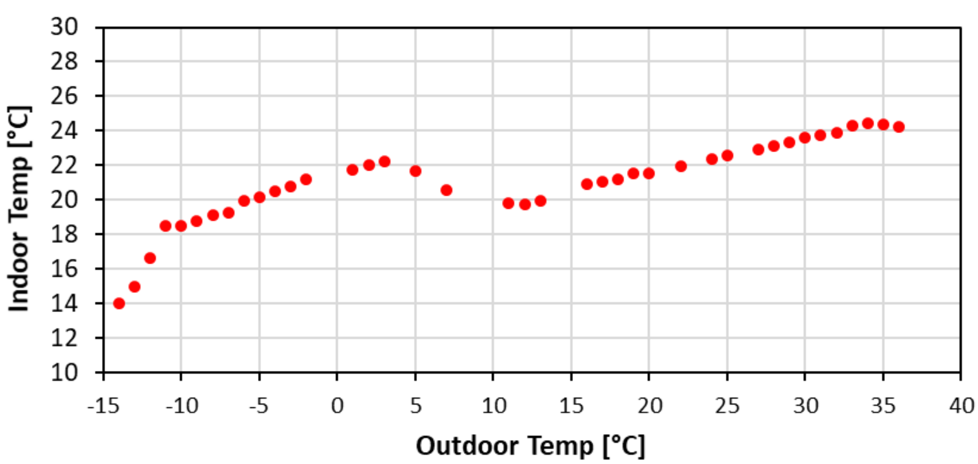

*Total number of heating and cooling data points throughout monitoring period.*



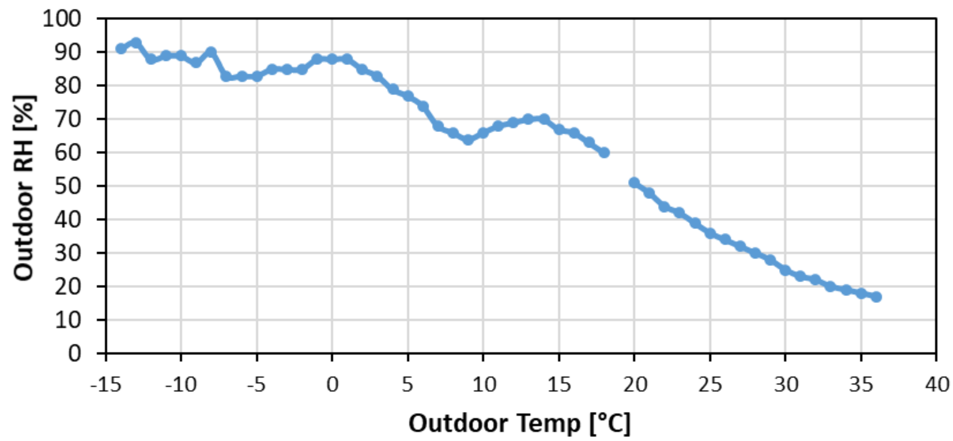

*Average system consumption and capacity for monitored outdoor temperature range.*



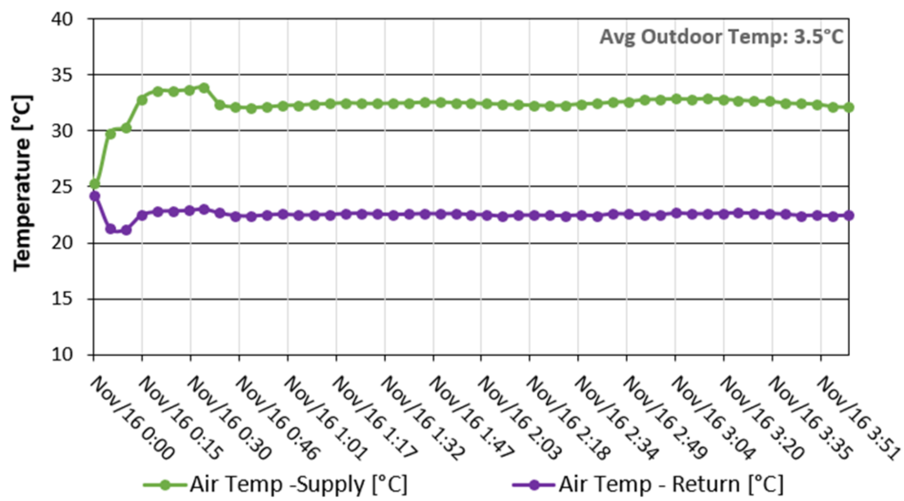

*Measured variables and corresponding COP for monitoring period.*

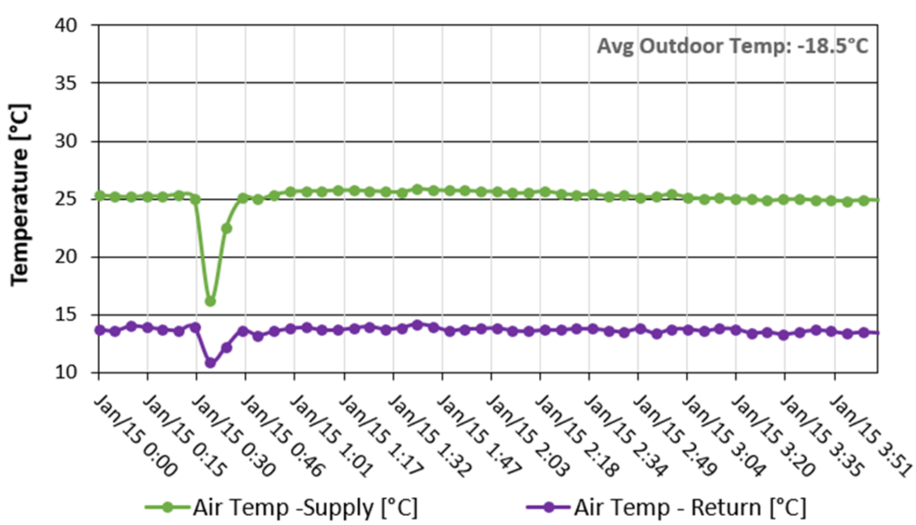



*Average indoor air temperature (return air) for monitored outdoor temperature range.*

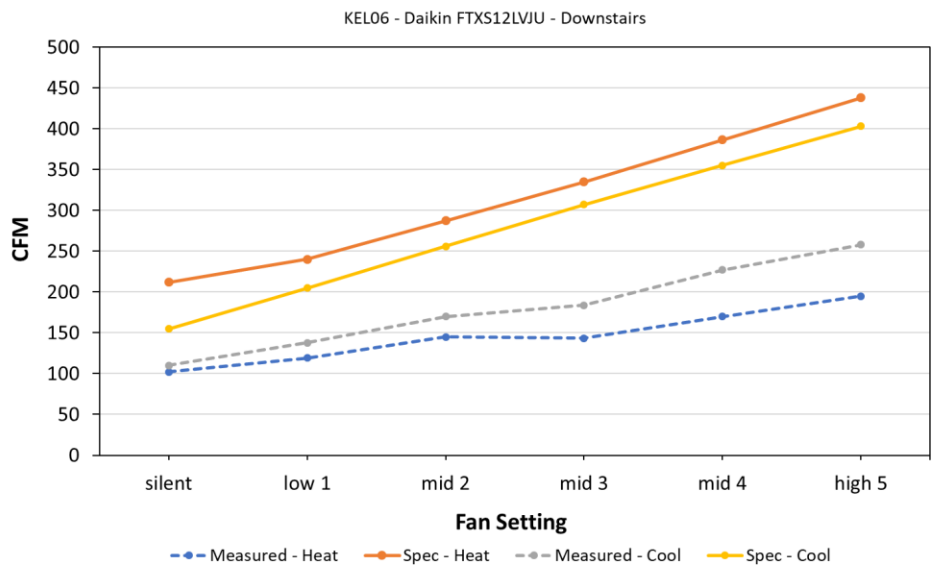



*Average outdoor relative humidity for monitored outdoor temperature range.*



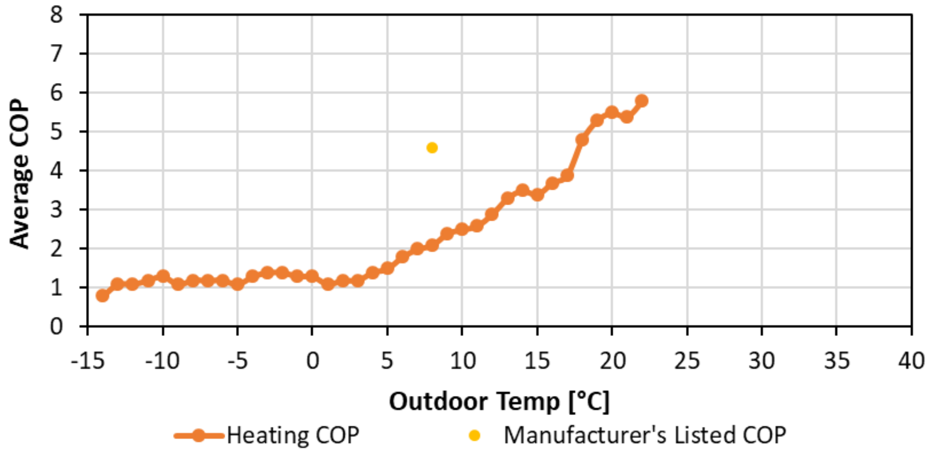




*Sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*

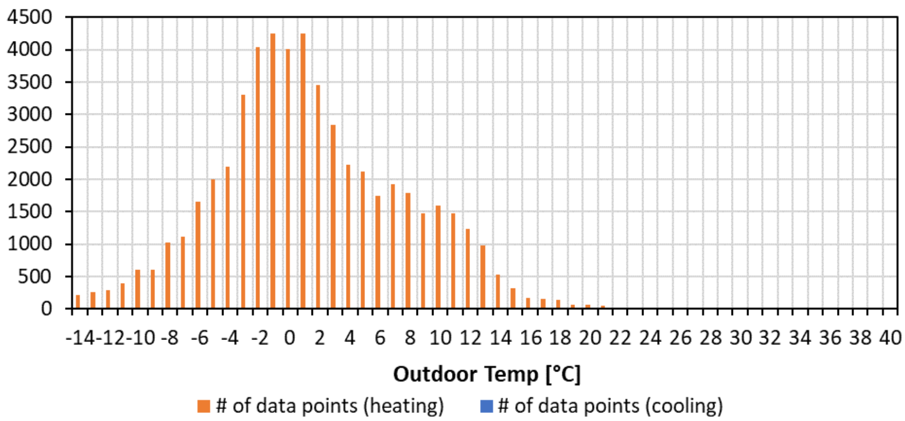



*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

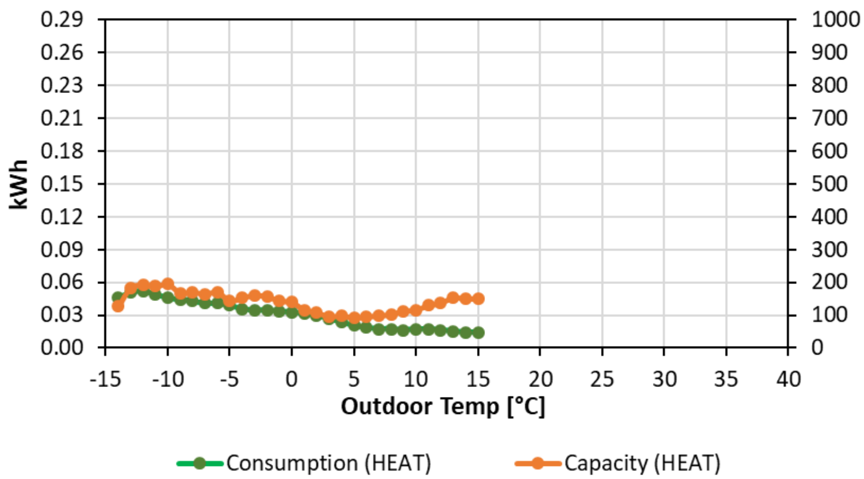
PRI01ii – Ductless (Single Head) – Fujitsu: AOU9RLS3H | ASU9RLS3Y



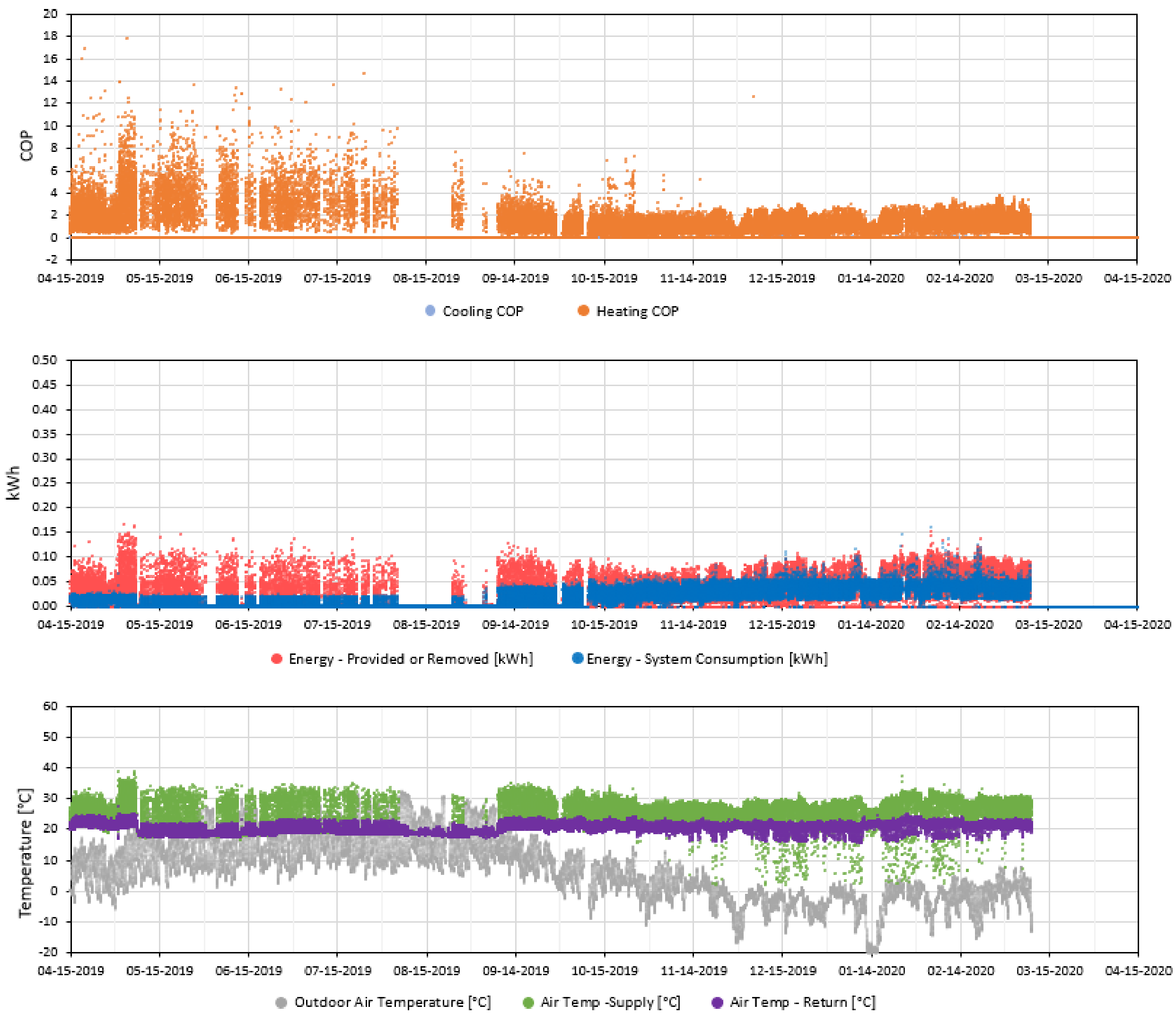

*Average estimated heating and cooling COP for monitored outdoor temperature range.*

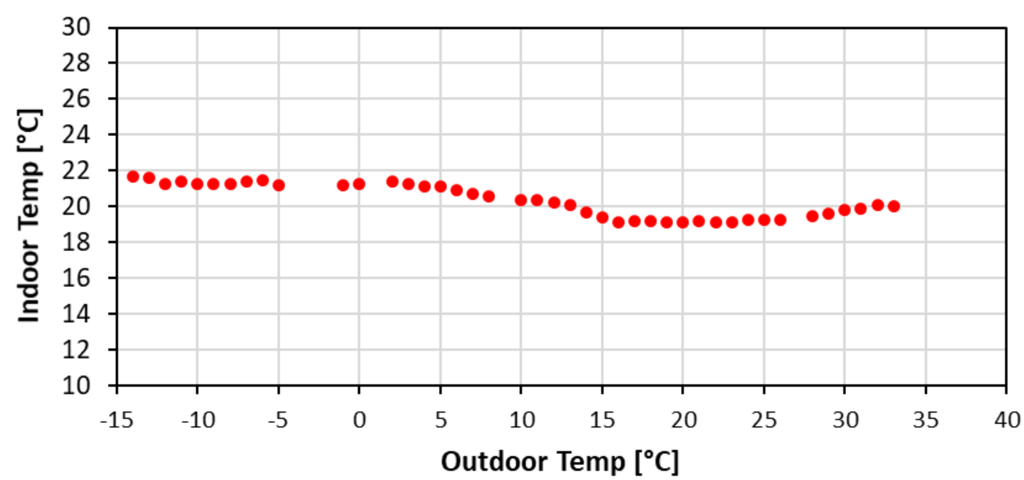



*Total number of heating and cooling data points throughout monitoring period.*

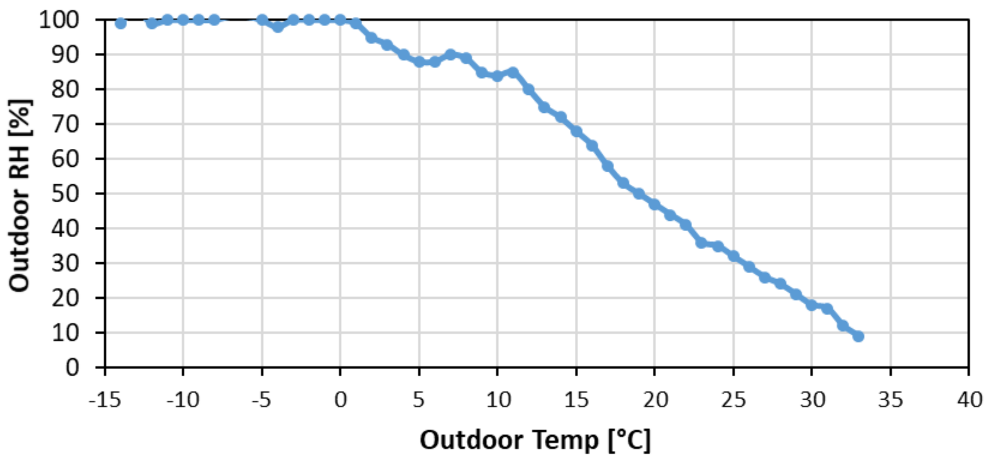



*Average system consumption and capacity for monitored outdoor temperature range.*

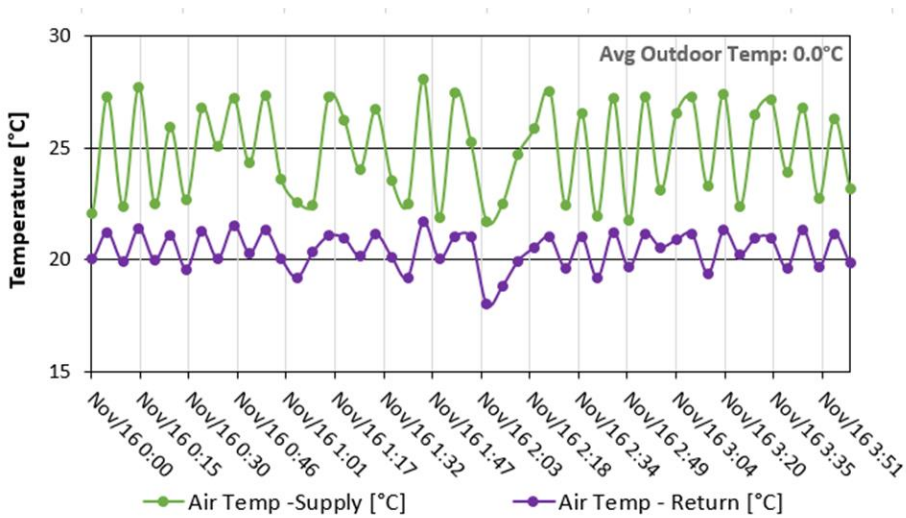


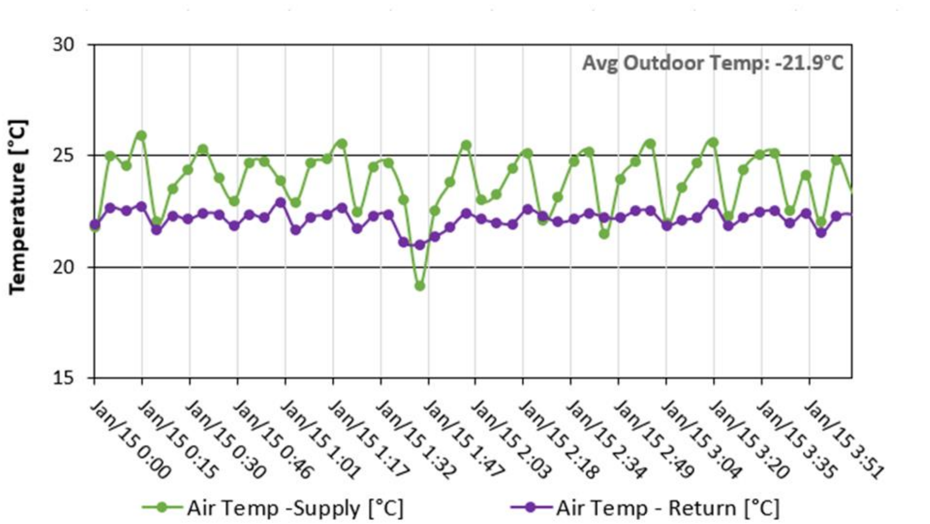




*Average indoor air temperature (return air) for monitored outdoor temperature range.*

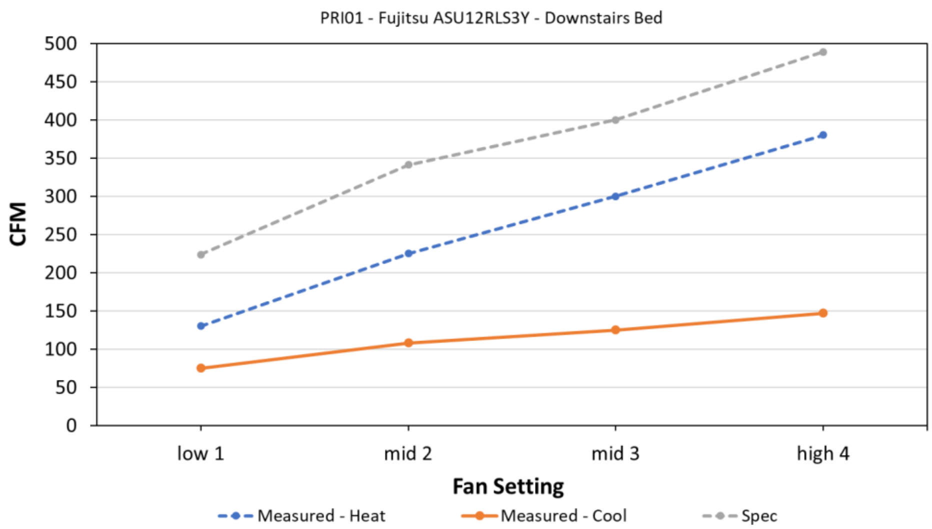



*Average outdoor relative humidity for monitored outdoor temperature range.*



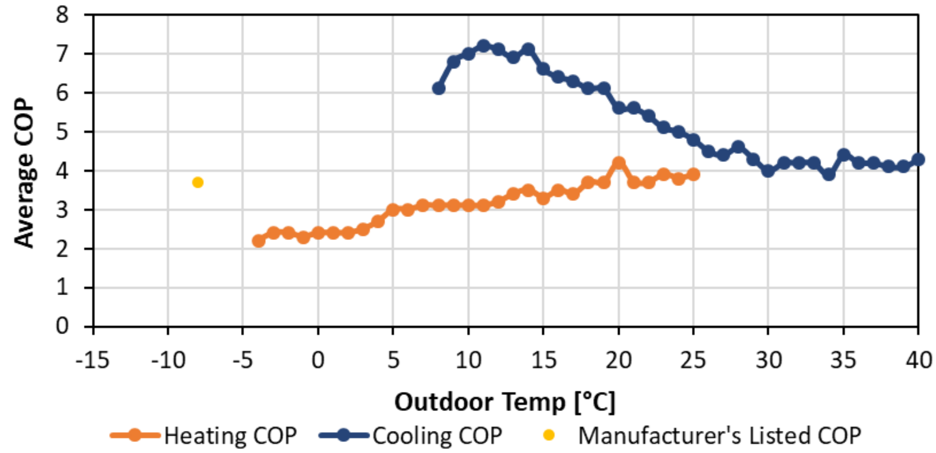




*Sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*

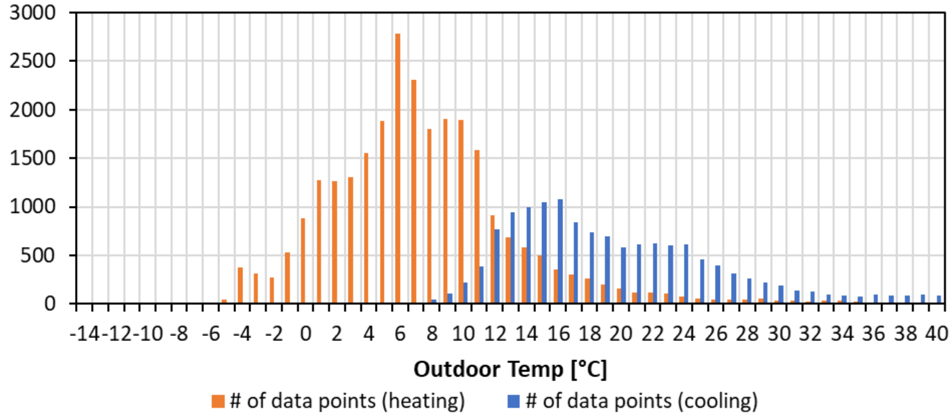



*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

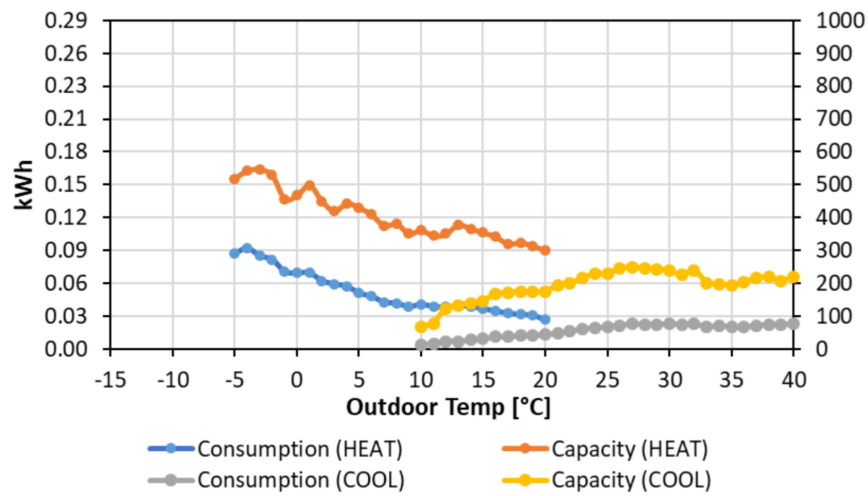
VIC02 – Ductless (Multi Head) – Fujitsu: AOU24RLXFZ | AGU15RLF | AGU9RLF



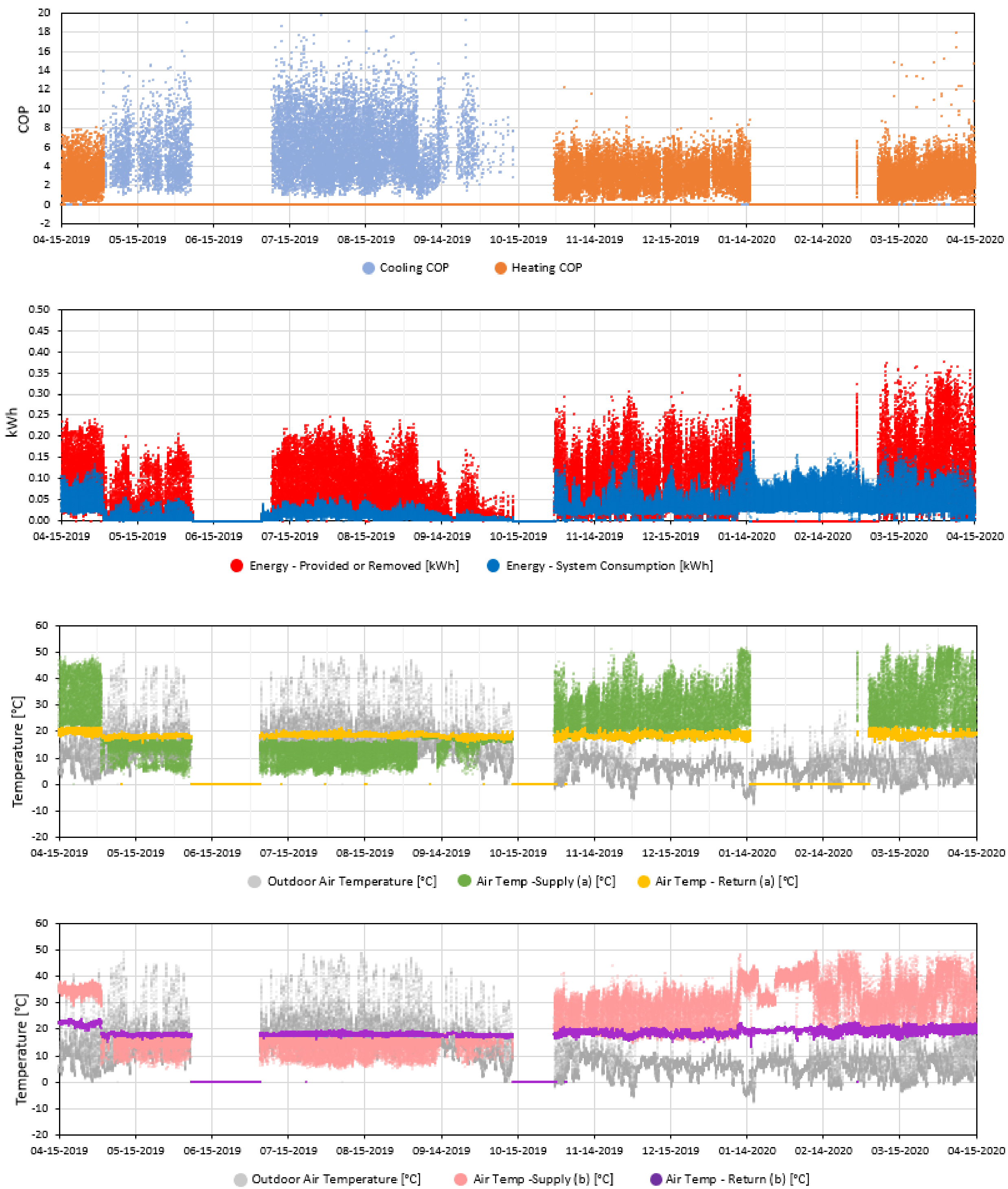

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



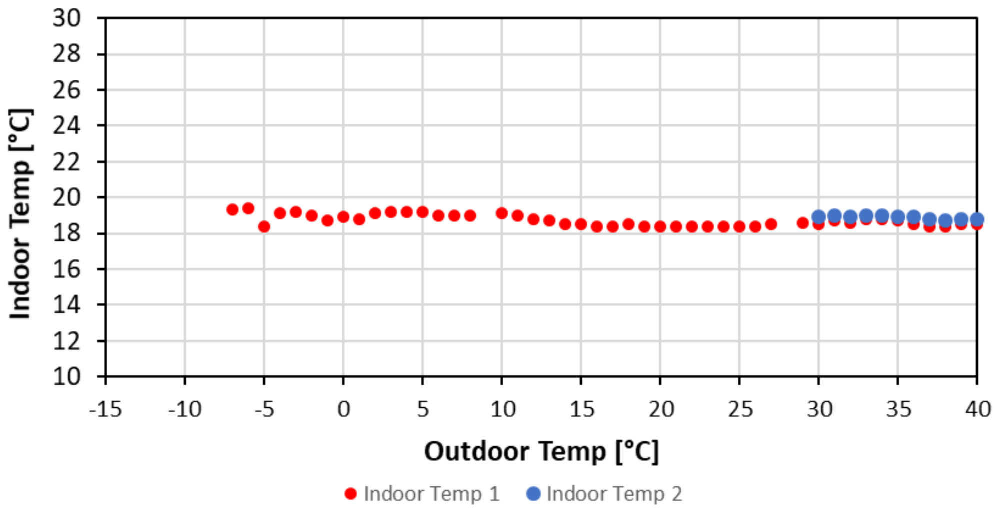

*Total number of heating and cooling data points throughout monitoring period.*



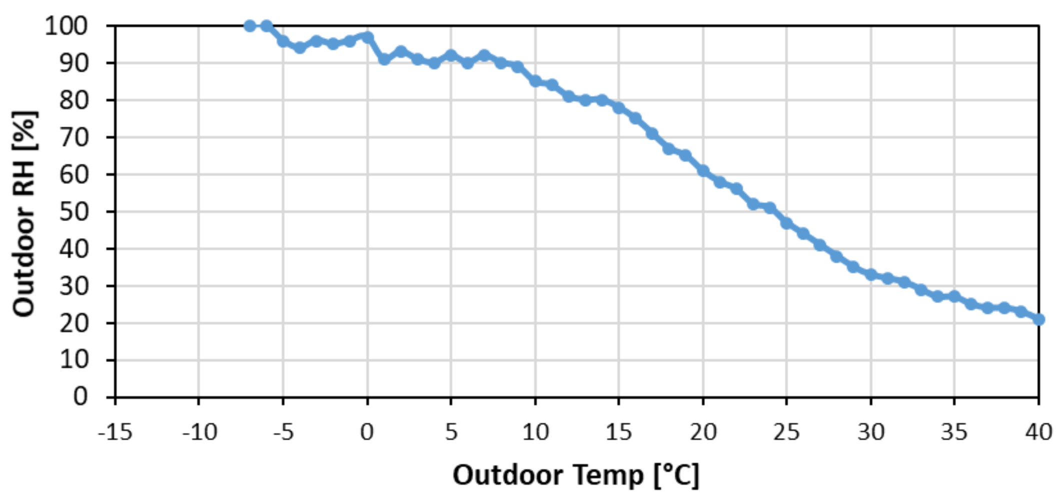

*Average system consumption and capacity for monitored outdoor temperature range.*



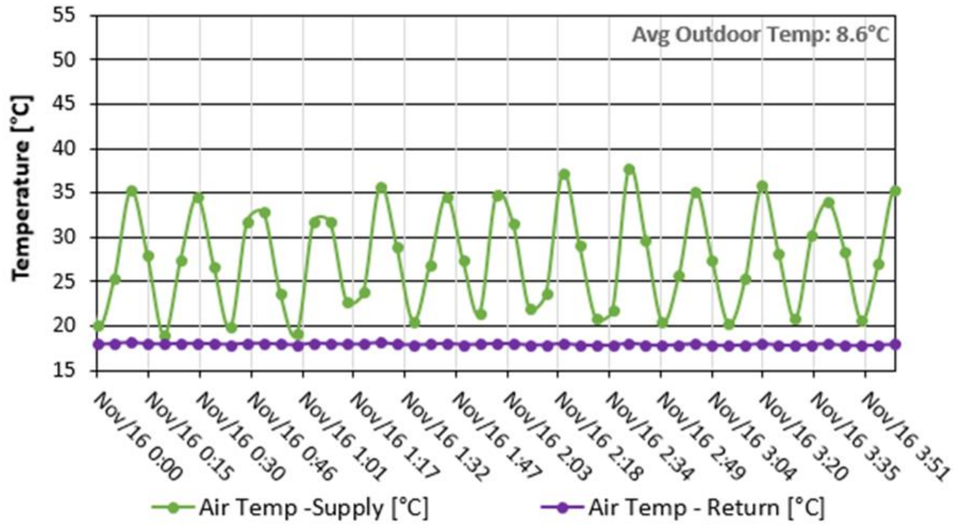

*Measured variables and corresponding COP for monitoring period.*

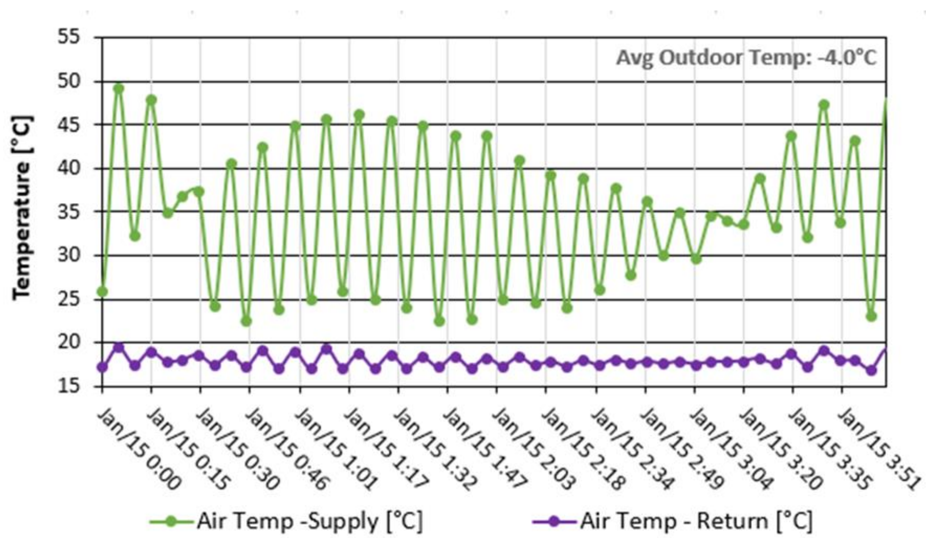



*Average indoor air temperature (return air) for monitored outdoor temperature range.*

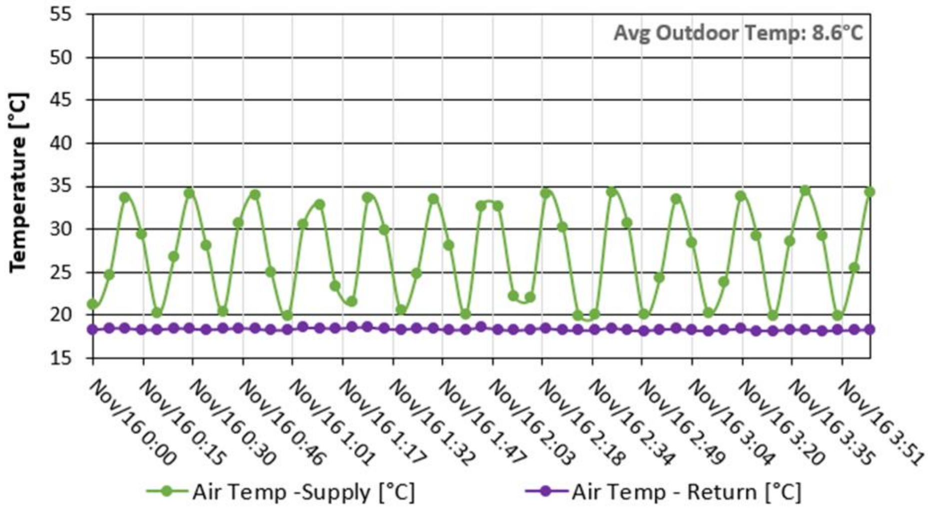


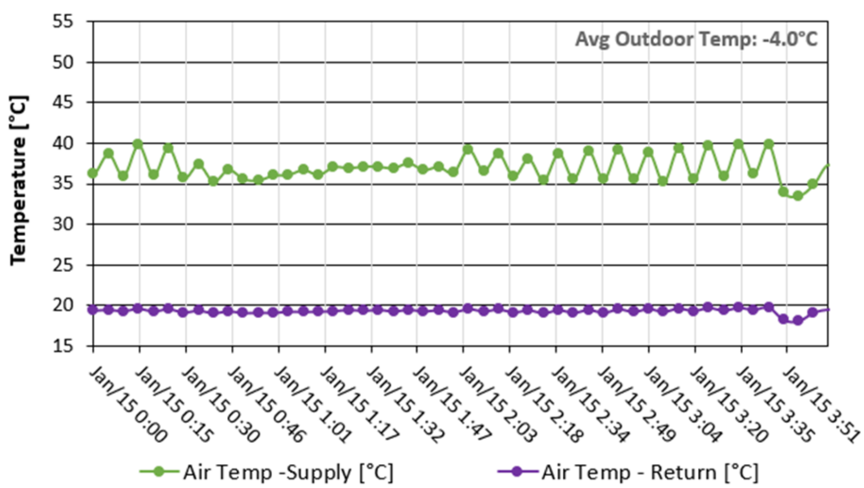

*Average outdoor relative humidity for monitored outdoor temperature range.*



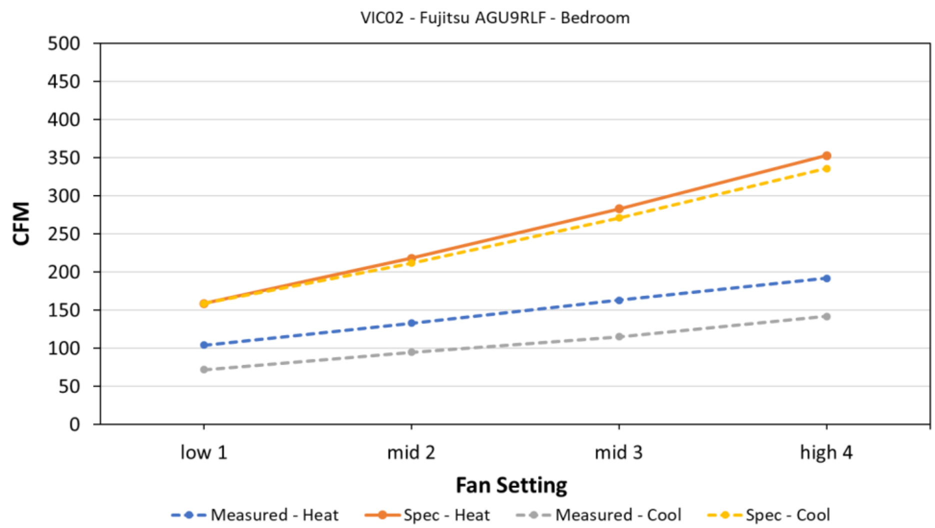



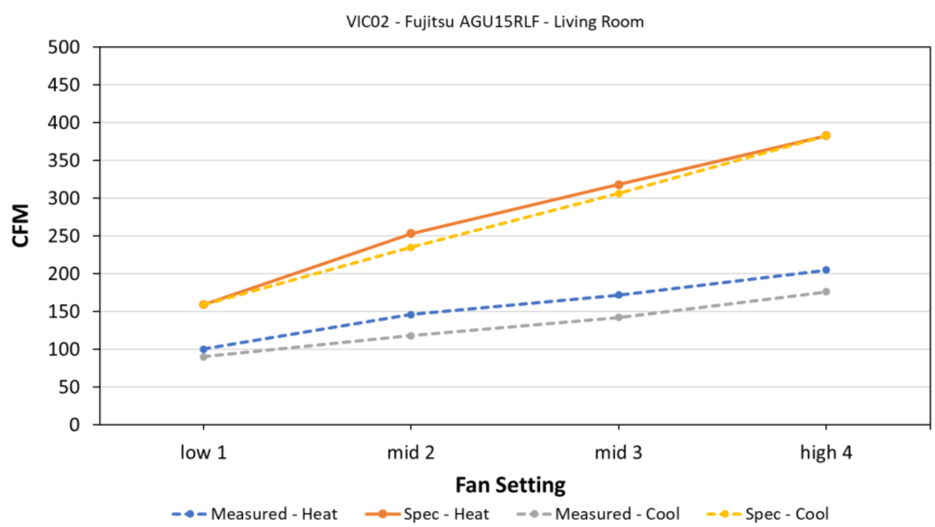

*Unit A: sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*



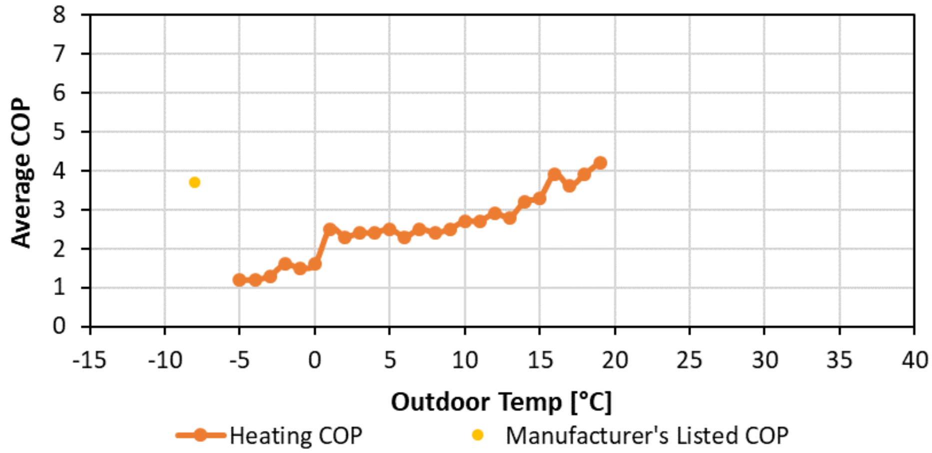

*Unit B: sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*



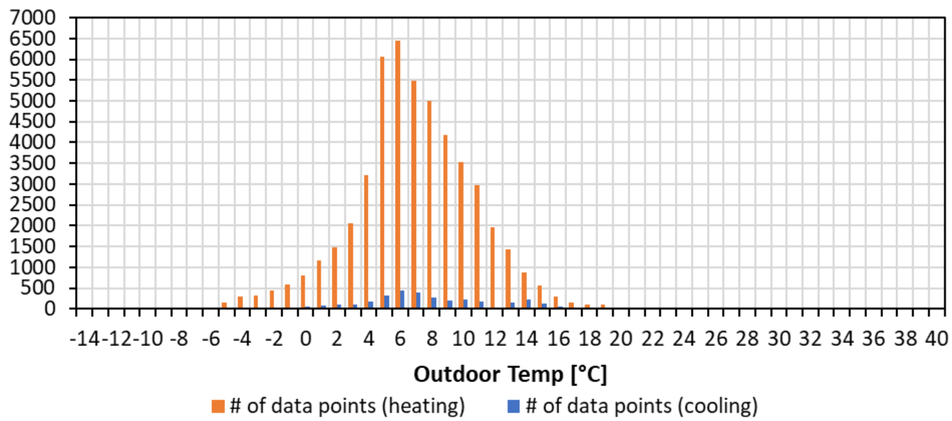




*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

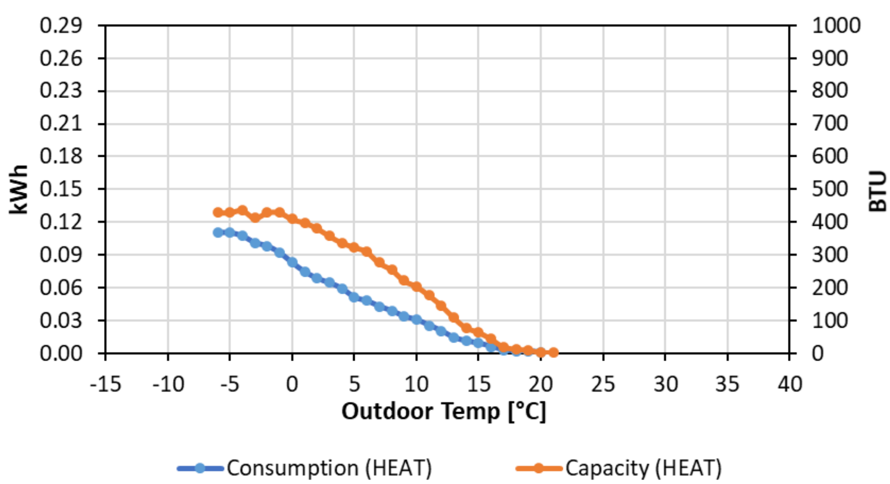
VIC03 – Ductless (Multi Head) – Fujitsu: AOU24RLXFZ | ASU15RLF1 | ASU12RLF1



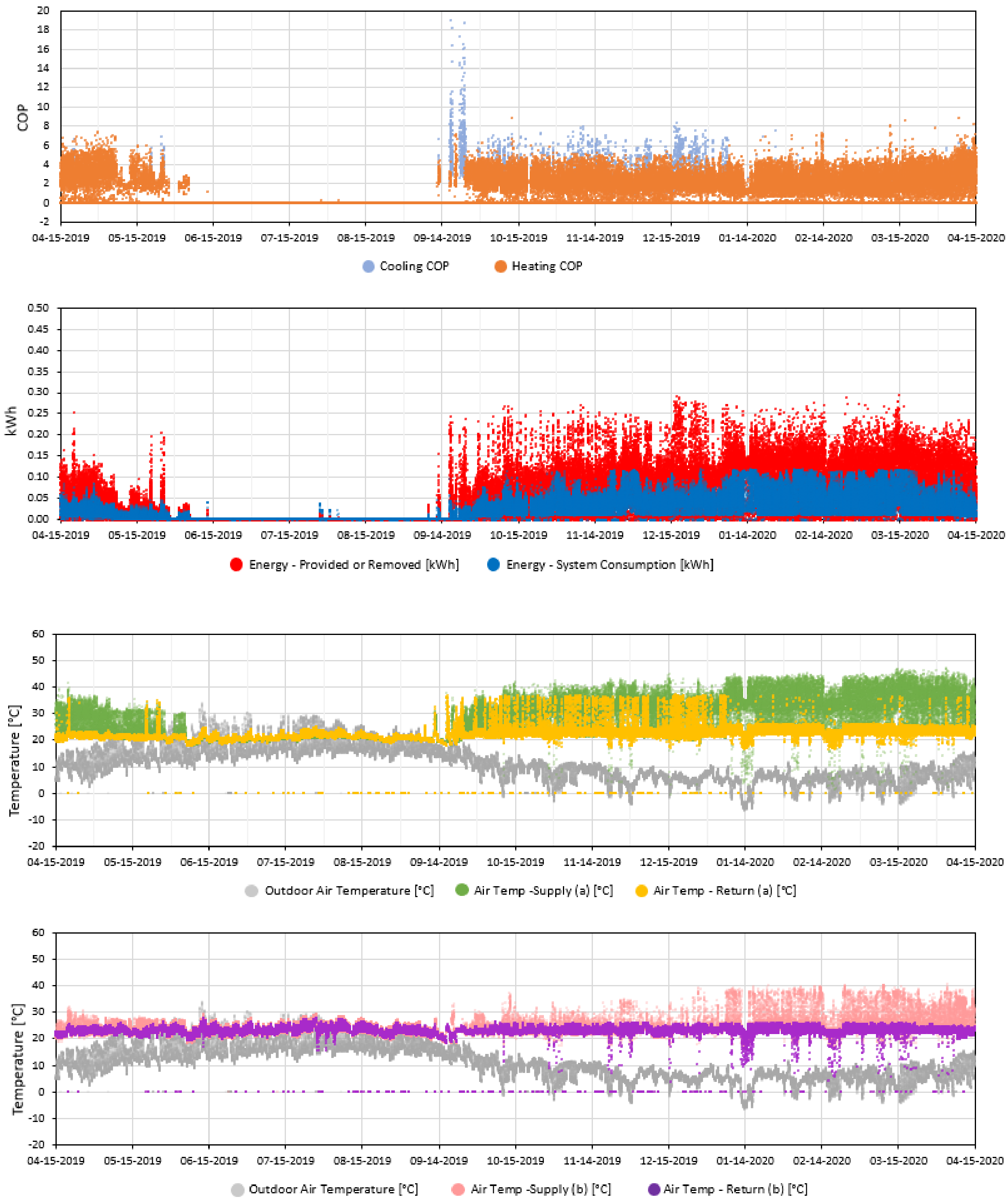

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



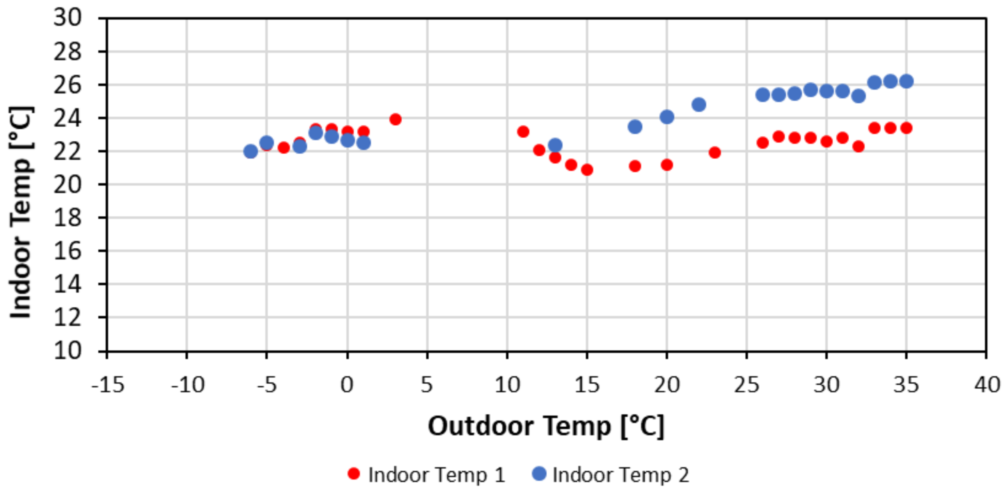

*Total number of heating and cooling data points throughout monitoring period.*



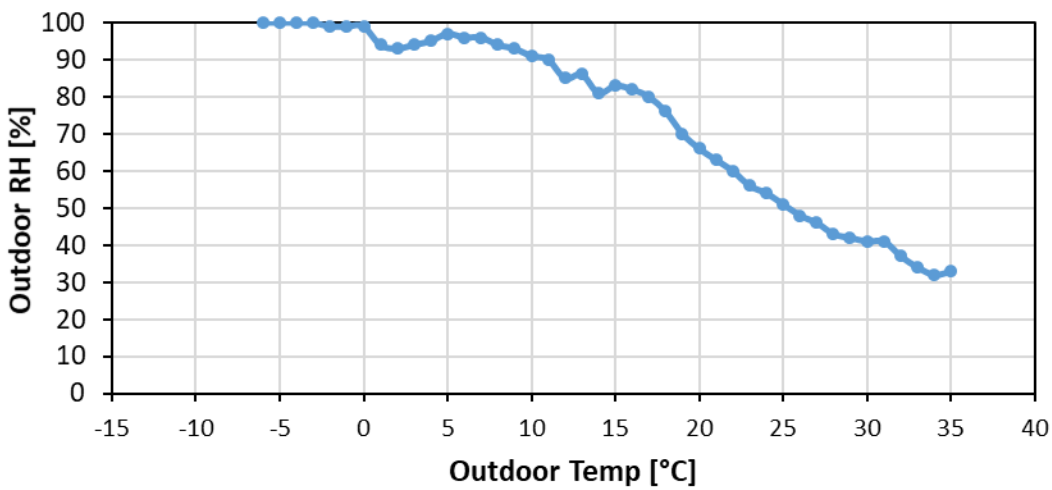

*Average system consumption and capacity for monitored outdoor temperature range.*



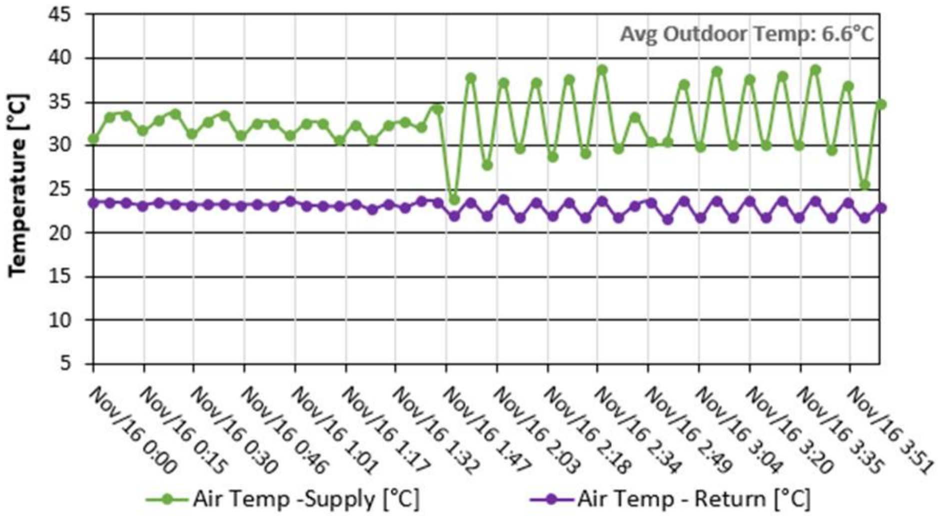

*Measured variables and corresponding COP for monitoring period.*

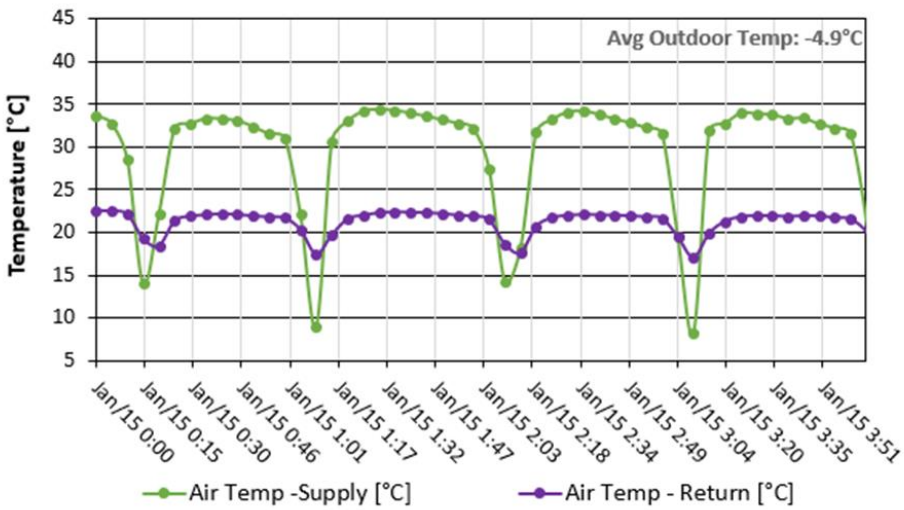



*Average indoor air temperature (return air) for monitored outdoor temperature range.*

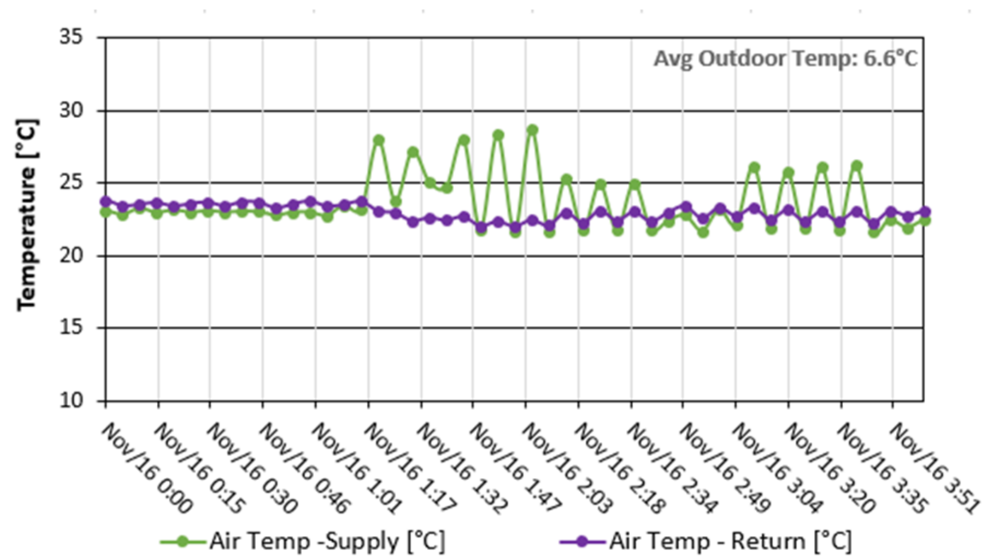


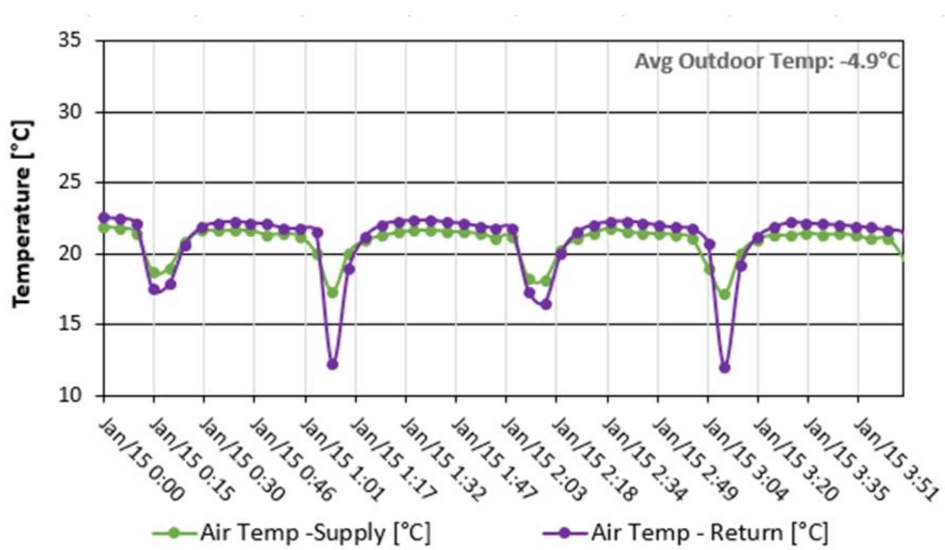

*Average outdoor relative humidity for monitored outdoor temperature range.*



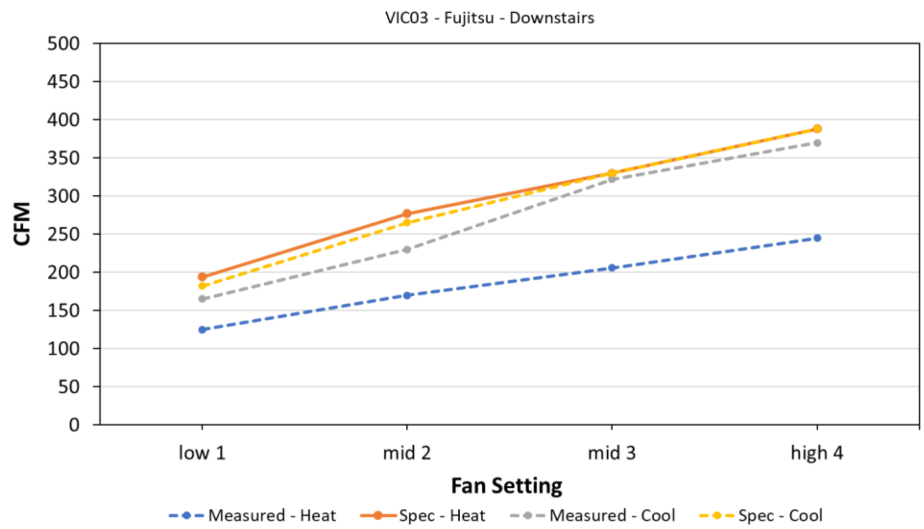



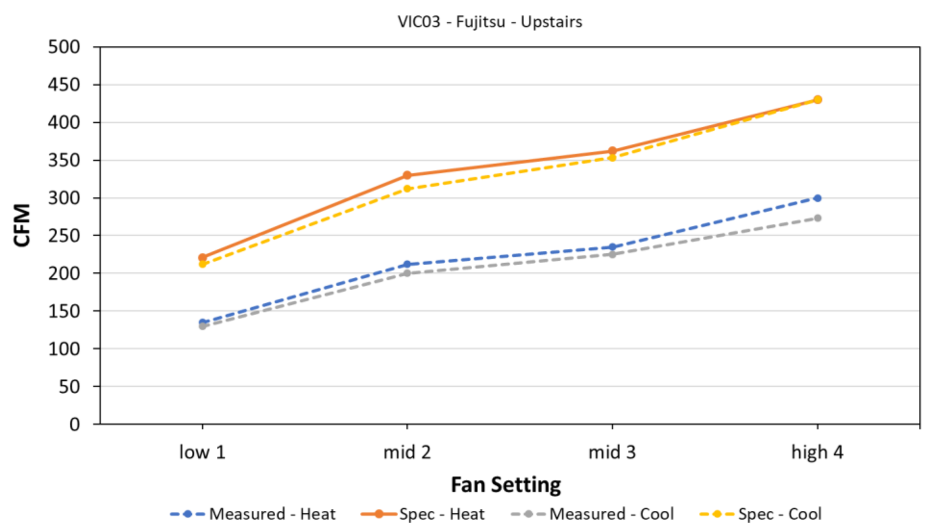

*Unit A: sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*



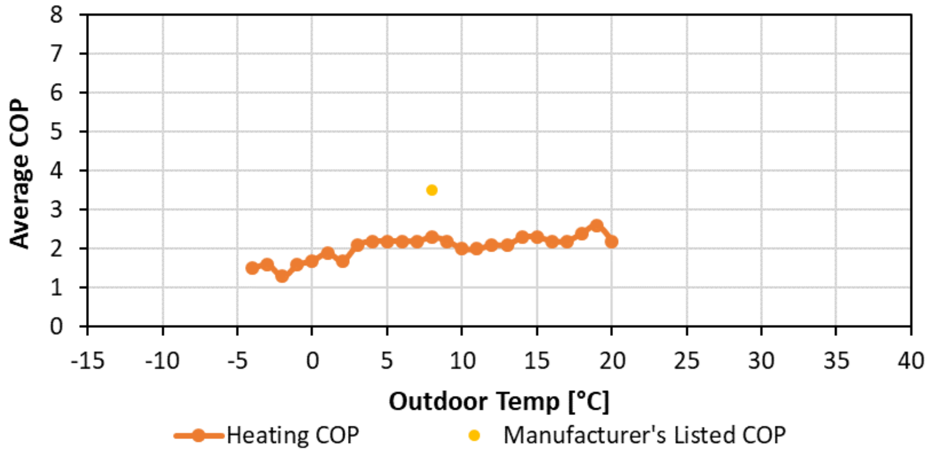

*Unit B: sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*



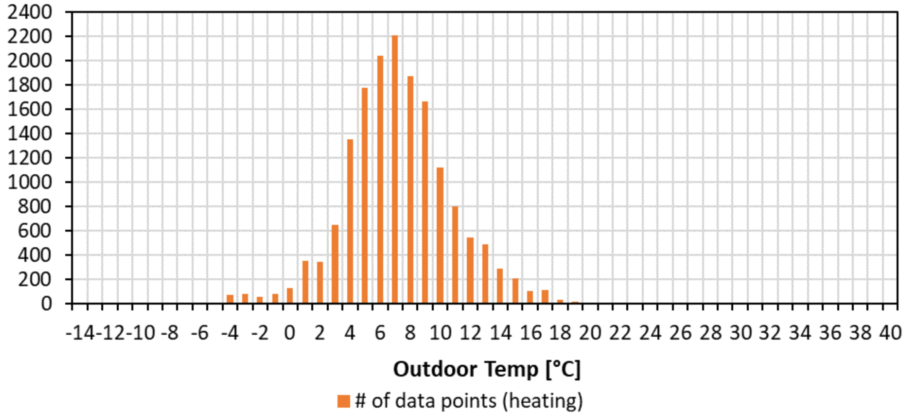




*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

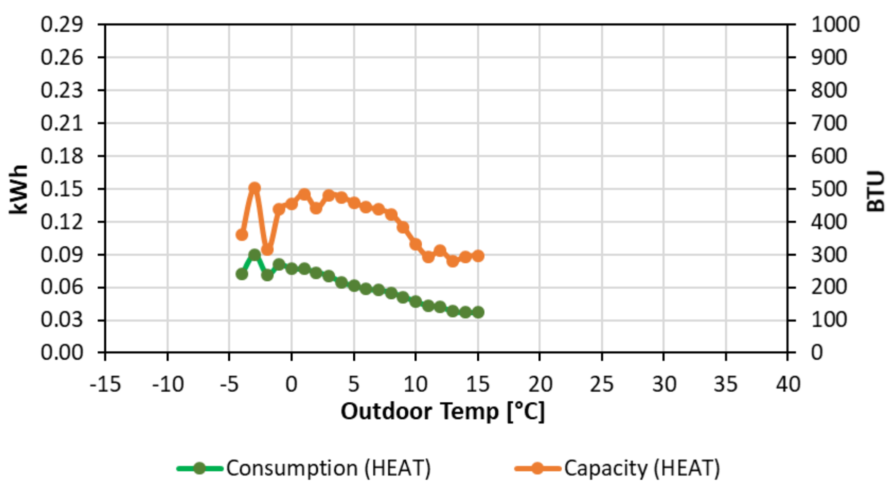
VIC05 – Ductless (Single Head) – Mitsubishi: MUZ-FH18NAH2 | MSZ-FH18NA2



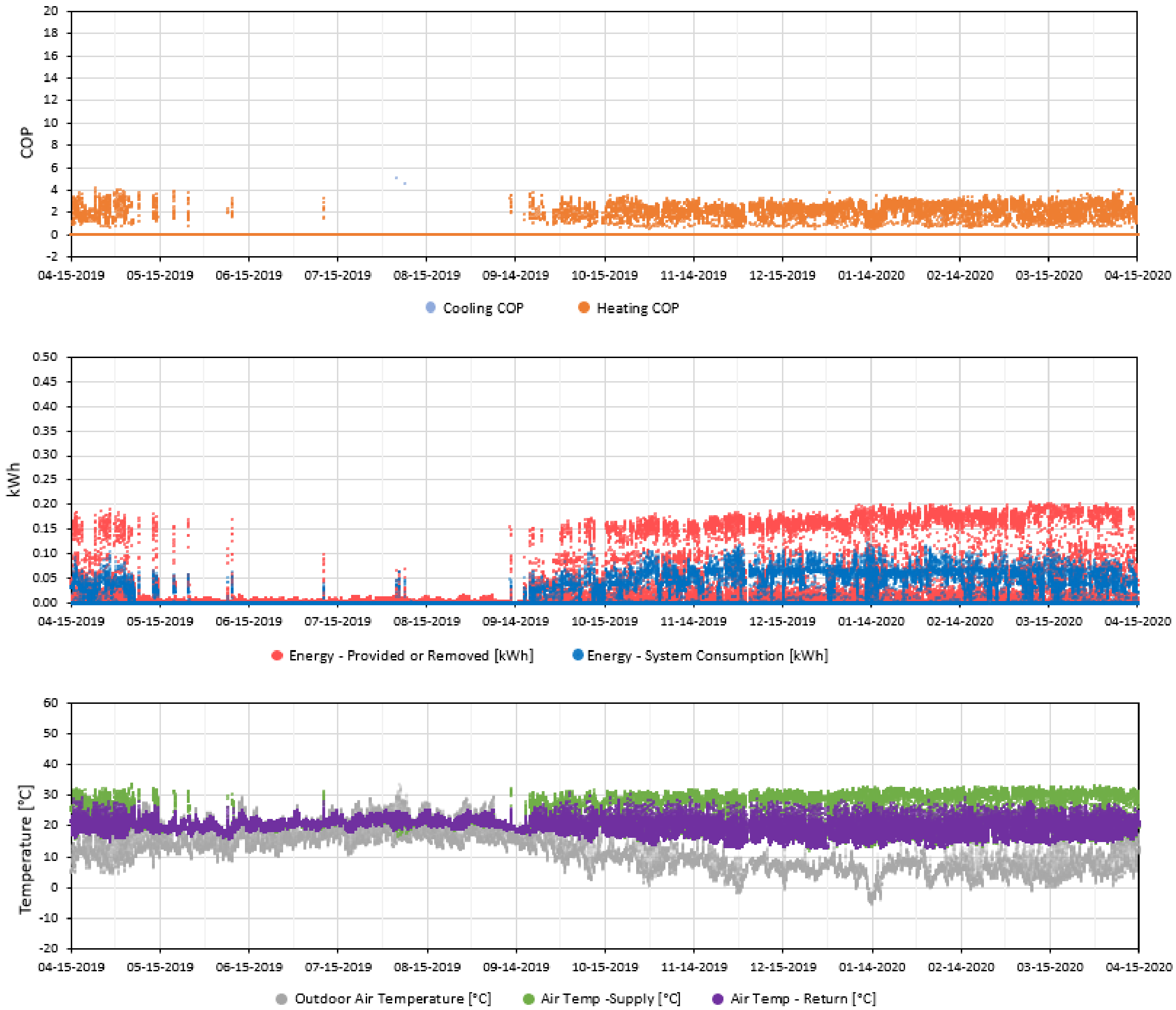

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



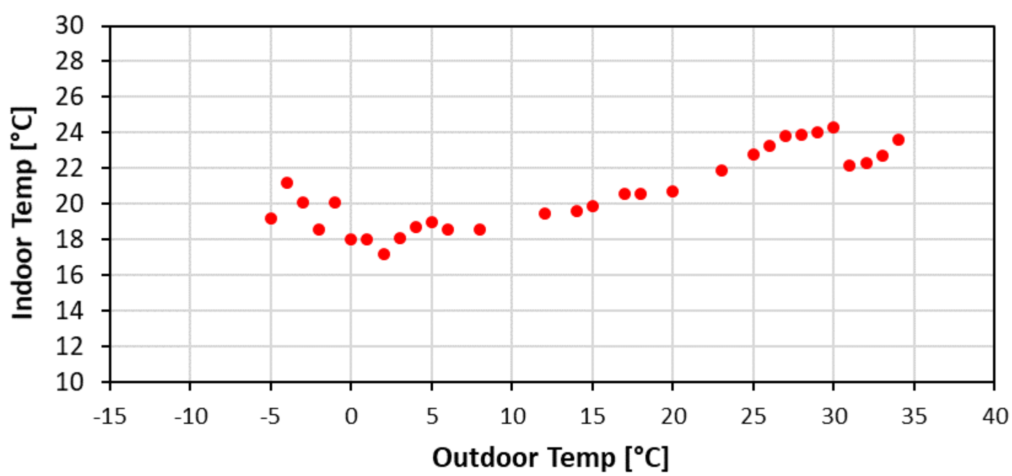

*Total number of heating and cooling data points throughout monitoring period.*



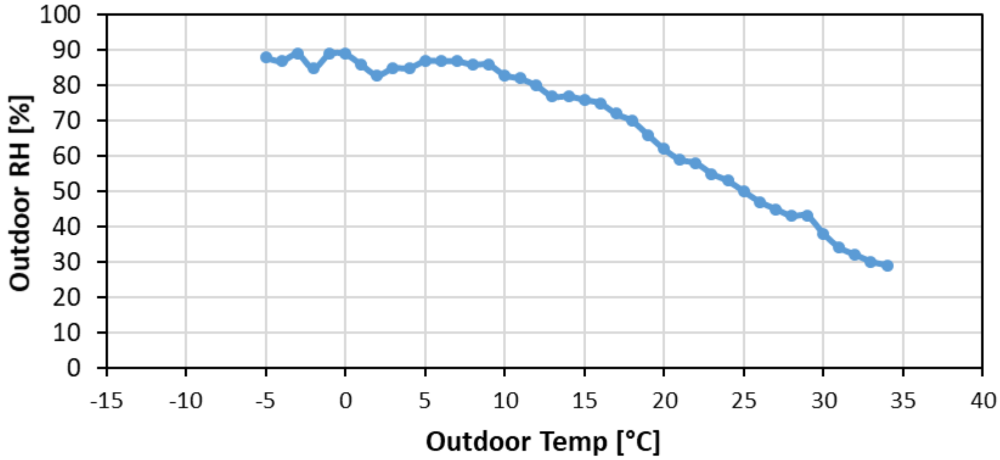

*Average system consumption and capacity for monitored outdoor temperature range.*



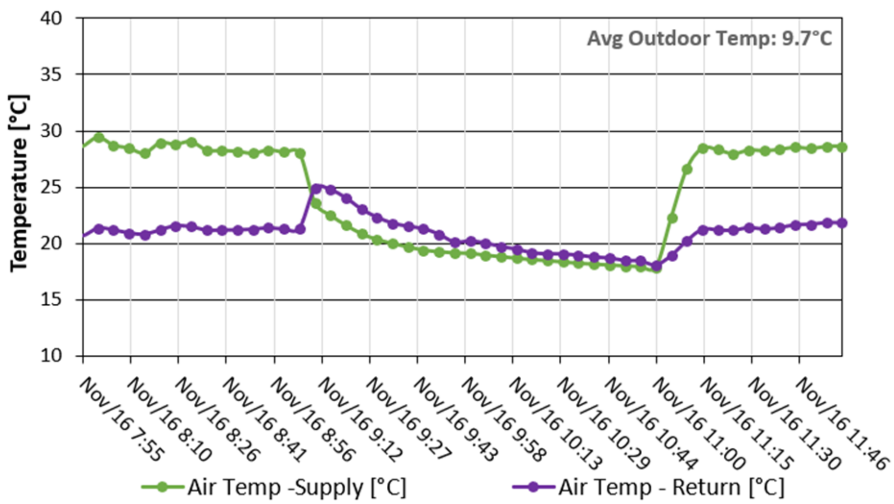

*Measured variables and corresponding COP for monitoring period.*

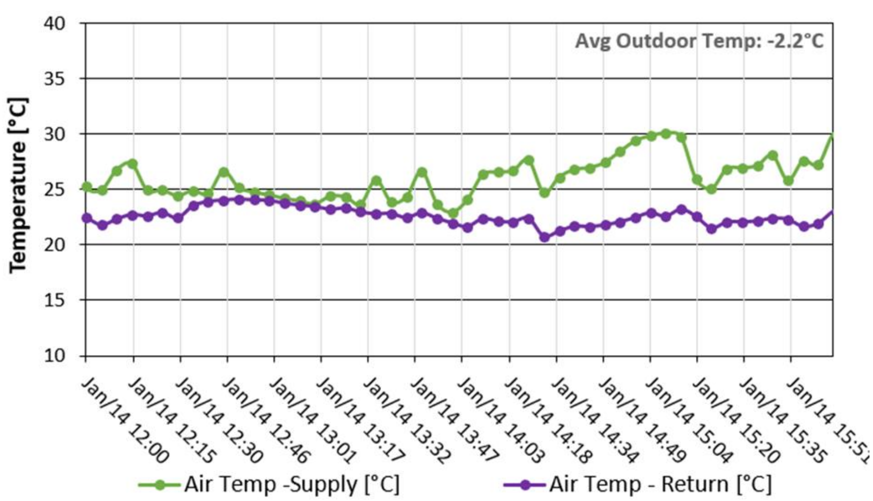



*Average indoor air temperature (return air) for monitored outdoor temperature range.*

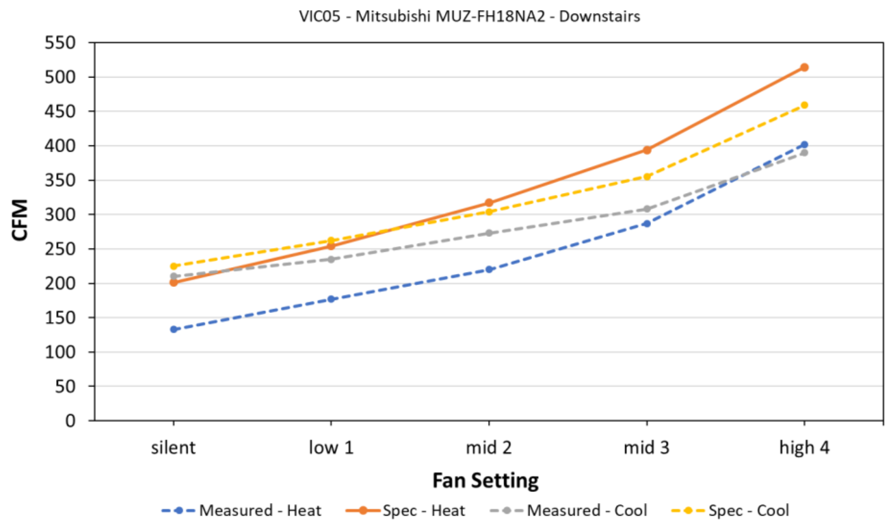



*Average outdoor relative humidity for monitored outdoor temperature range.*



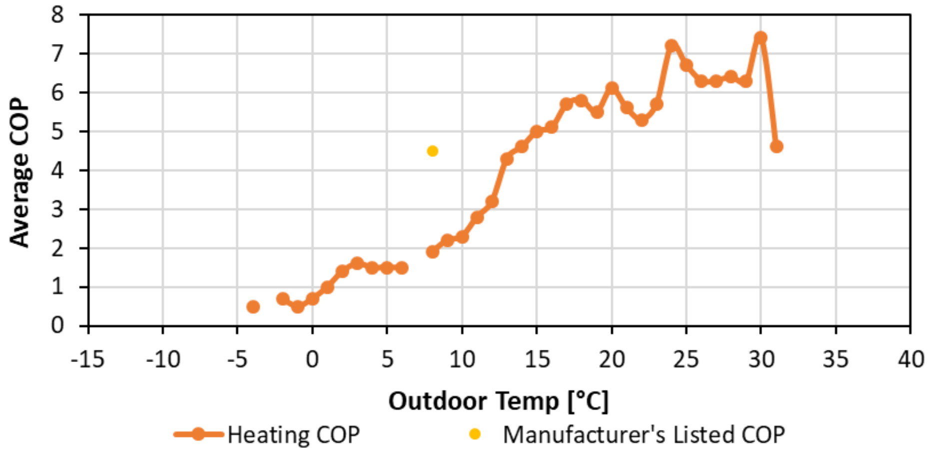




*Sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*

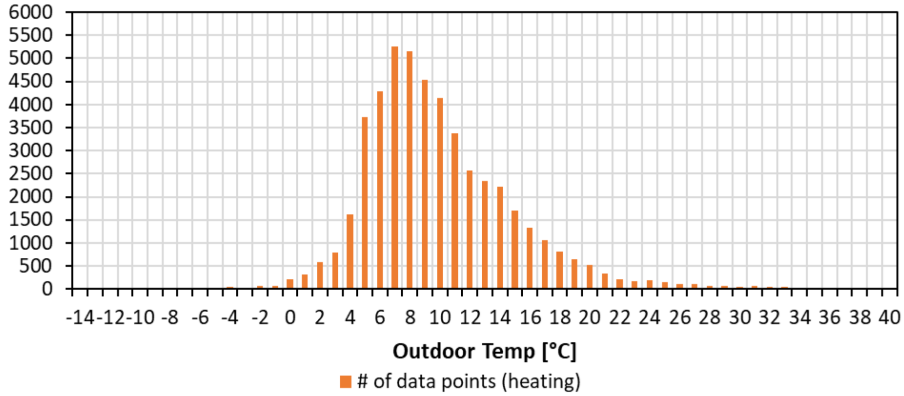



*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

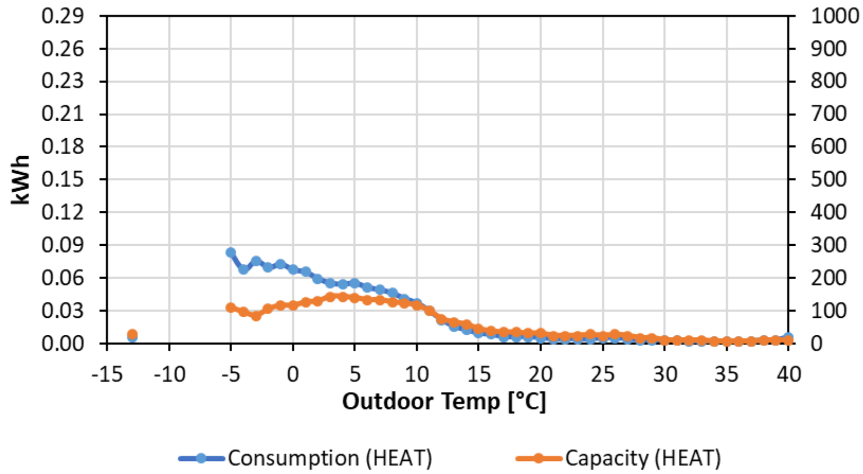
VIC06 – Ductless (Multi Head) – Daikin: 2MXS18NMVJU | FTXS09LVJU | FTXS09LVJU



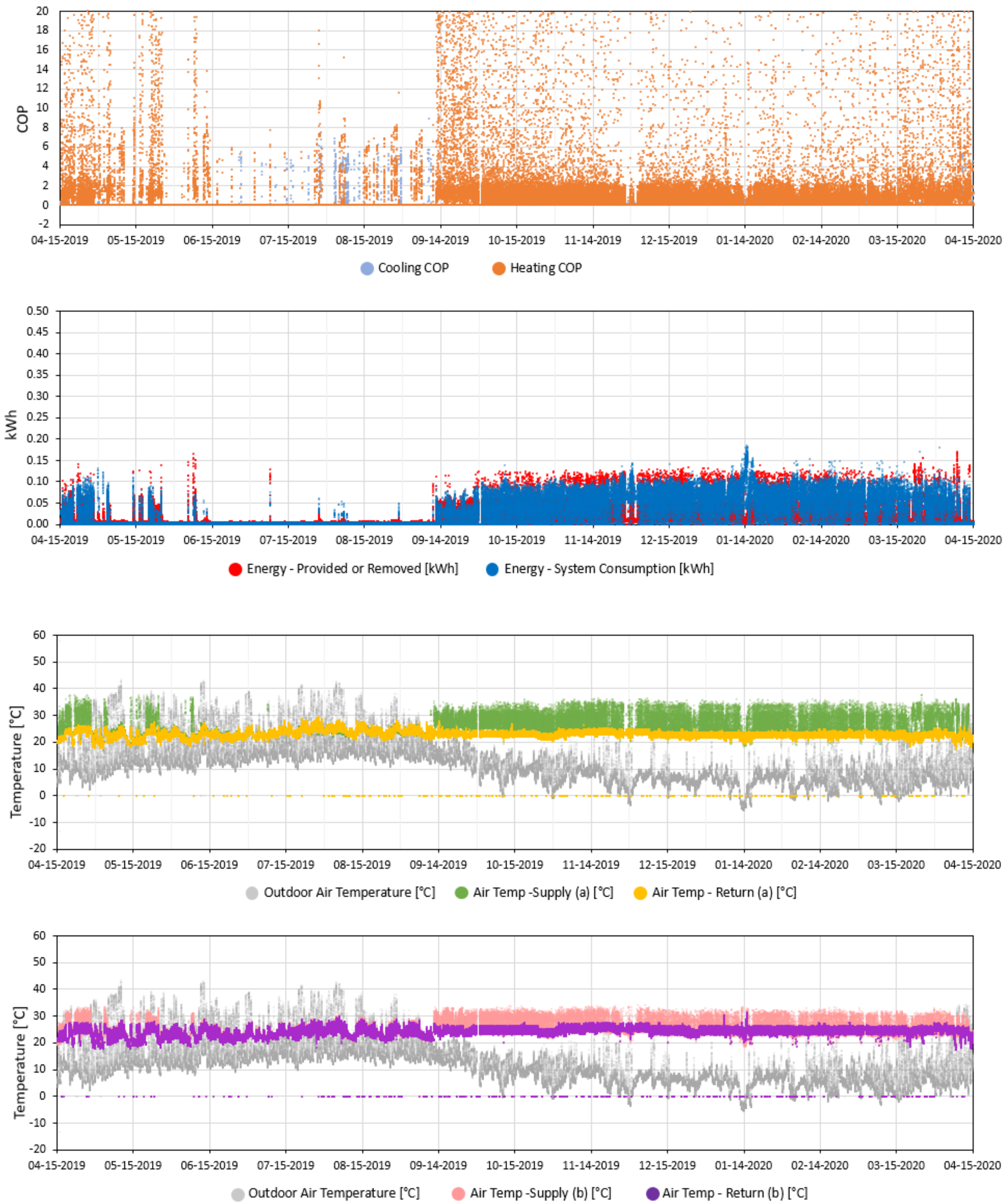

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



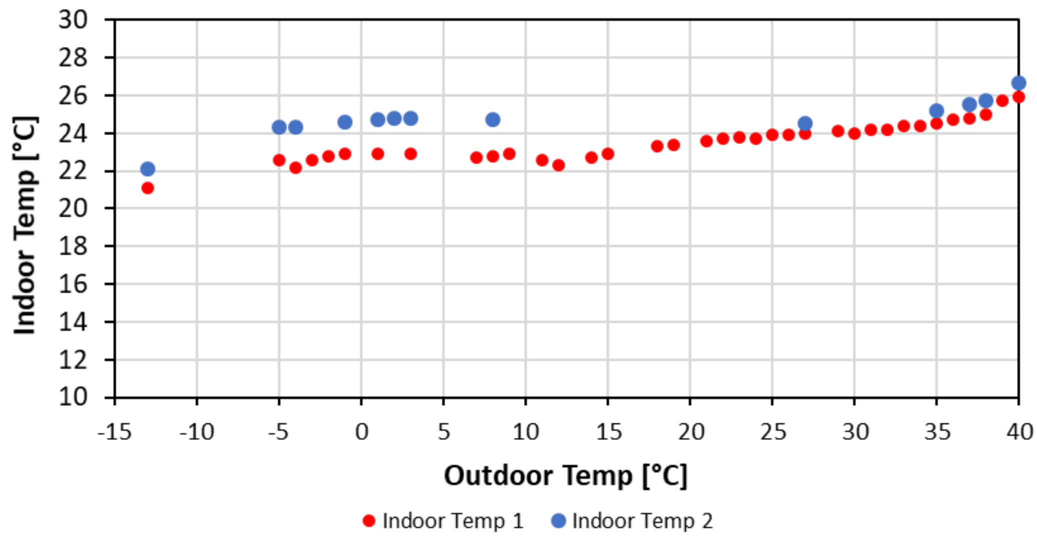

*Total number of heating and cooling data points throughout monitoring period.*



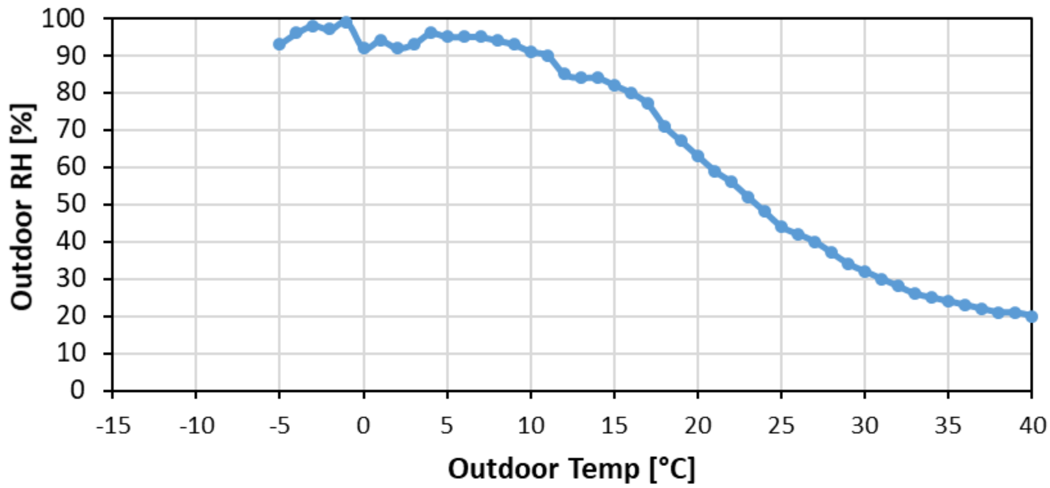

*Average system consumption and capacity for monitored outdoor temperature range.*



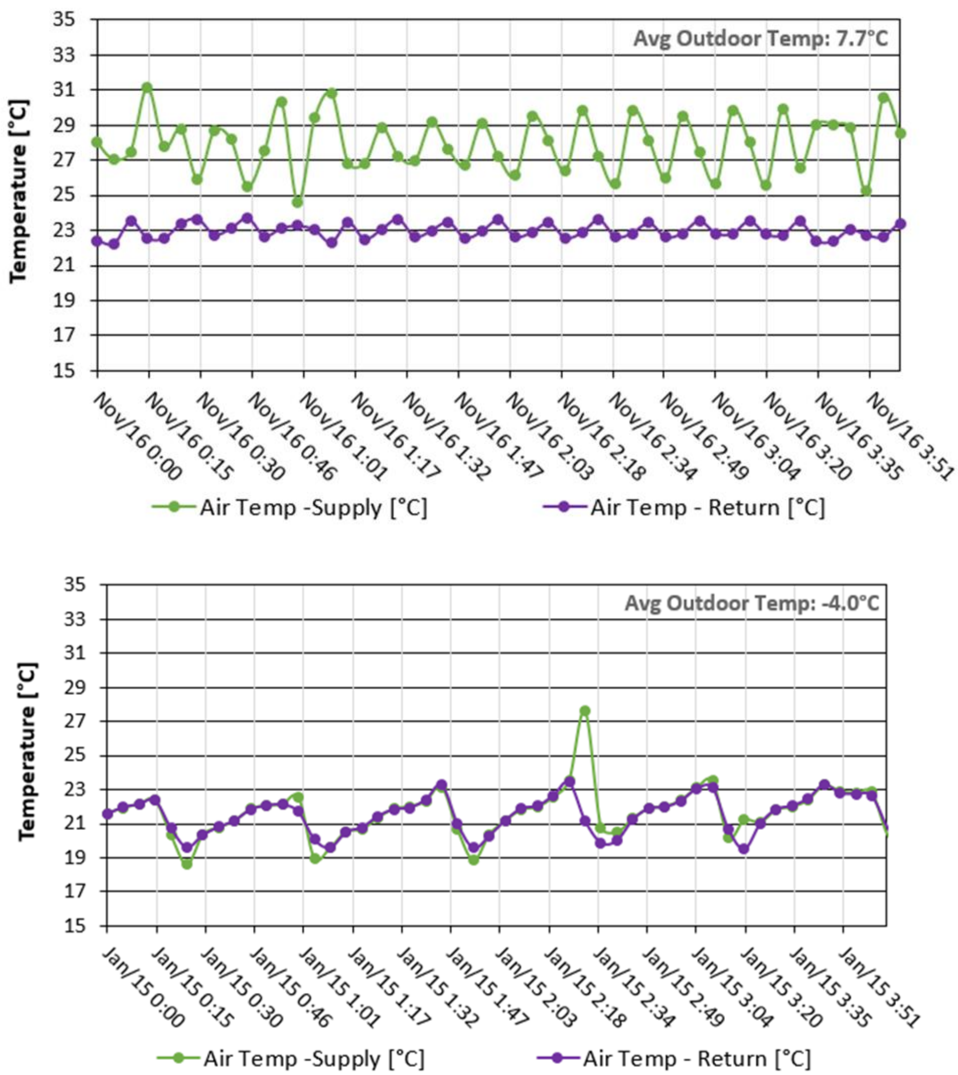

*Measured variables and corresponding COP for monitoring period.*



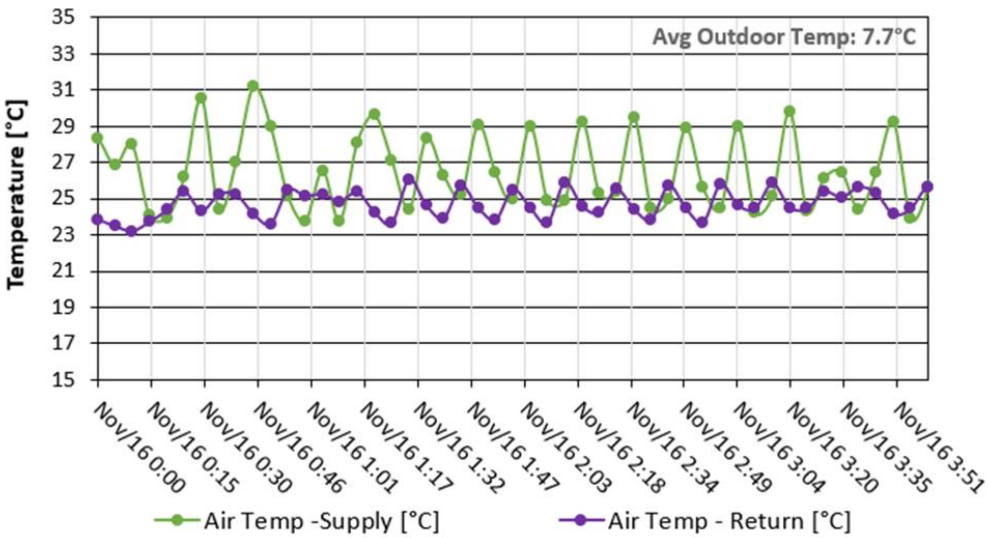

*Average indoor air temperature (return air) for monitored outdoor temperature range.*

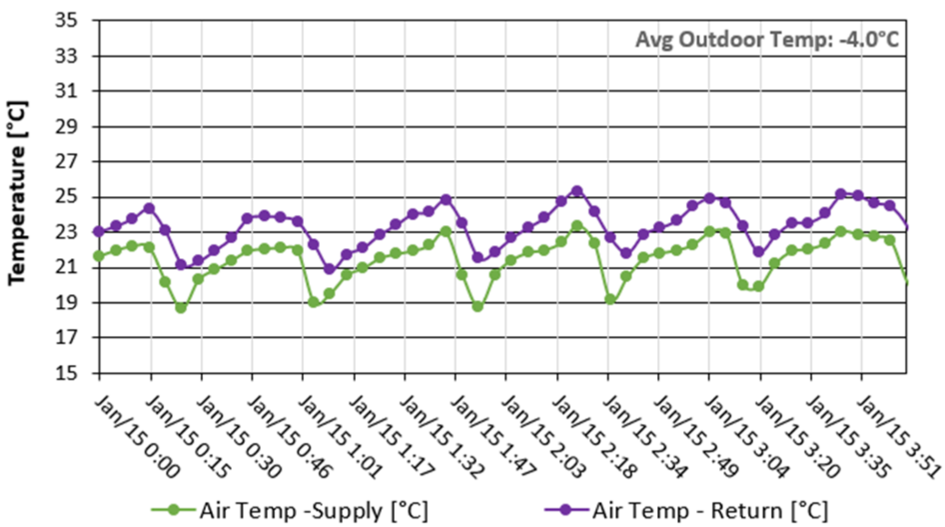



*Average outdoor relative humidity for monitored outdoor temperature range.*

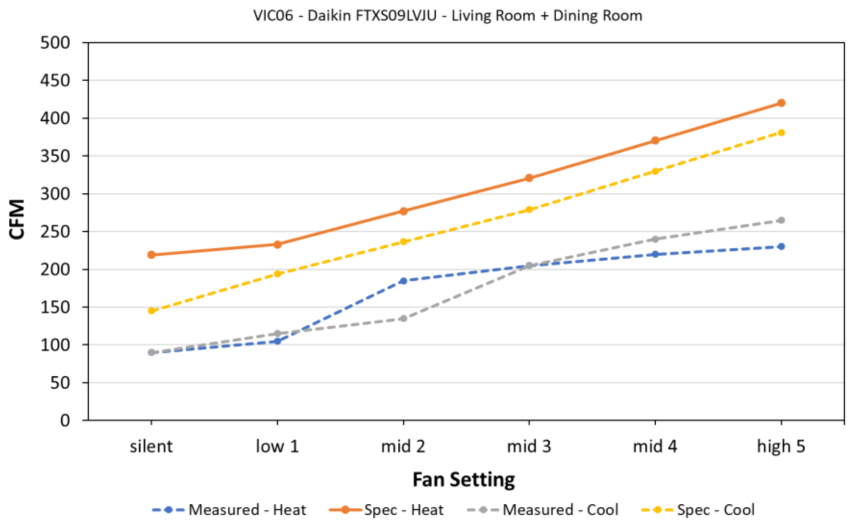



*Unit A: sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*



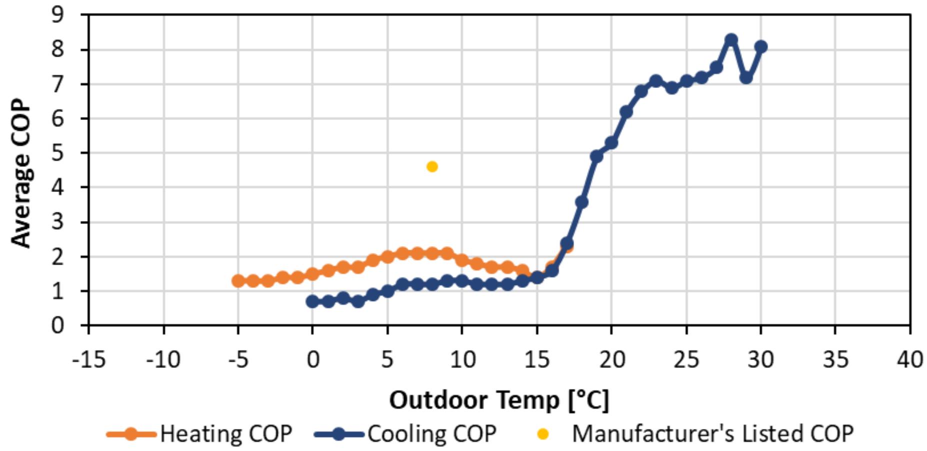




*Unit B: sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*

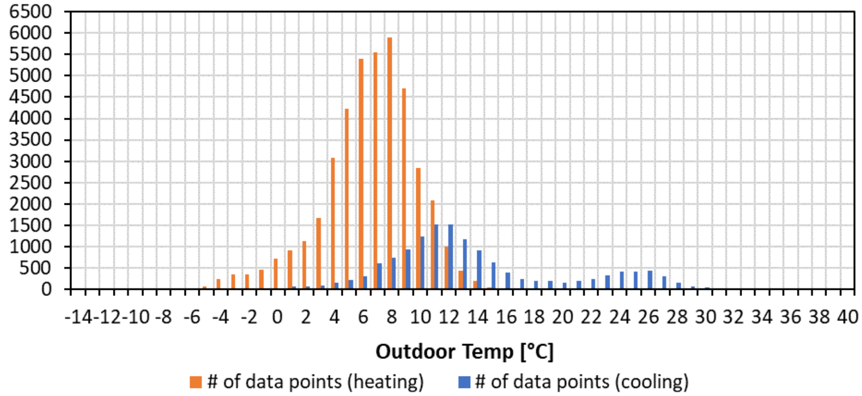



*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

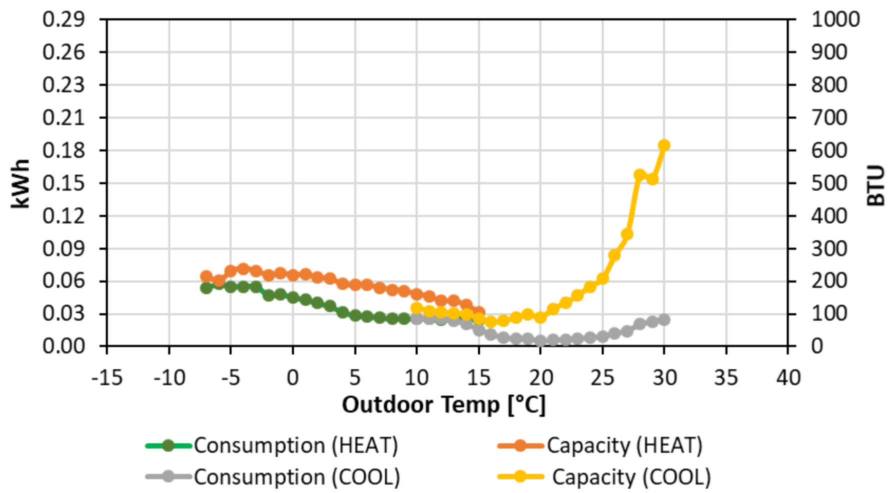
VIC10 – Ductless (Single Head) – Fujitsu: AOU12RLS3 | ASU12RLS3Y




*Average estimated heating and cooling COP for monitored outdoor temperature range.*

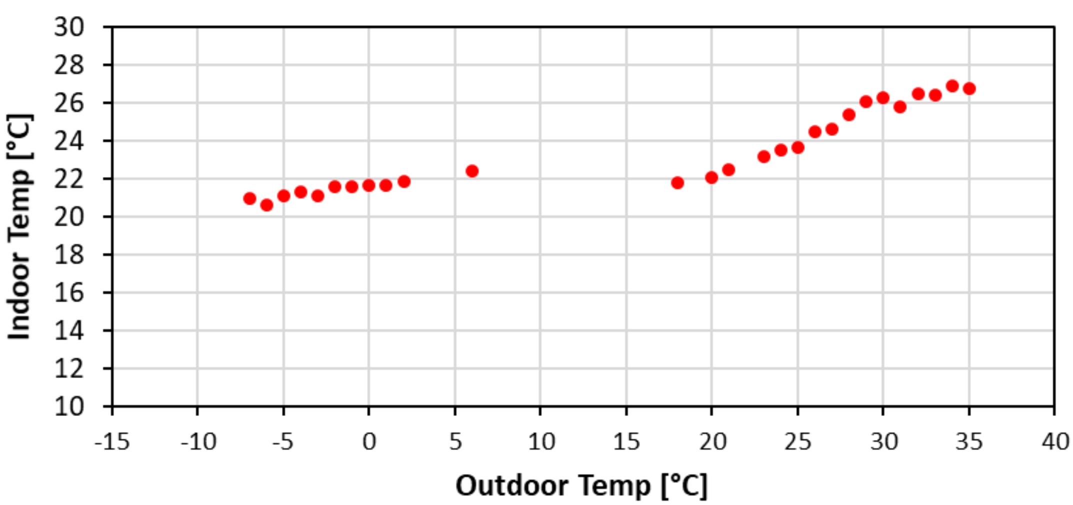



*Total number of heating and cooling data points throughout monitoring period.*

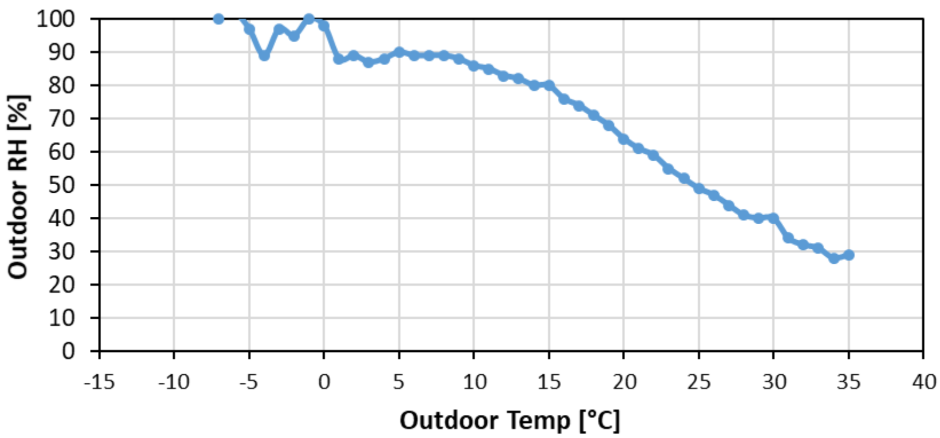



*Average system consumption and capacity for monitored outdoor temperature range.*

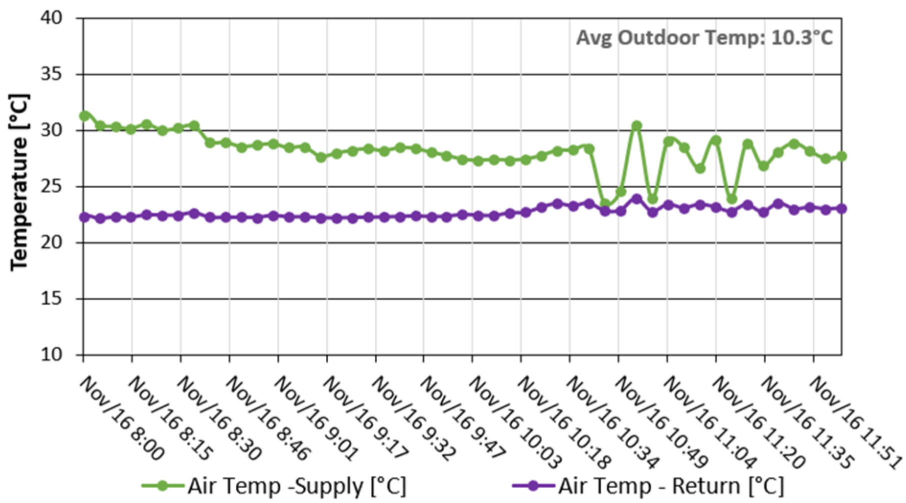

*Measured variables and corresponding COP for monitoring period.*

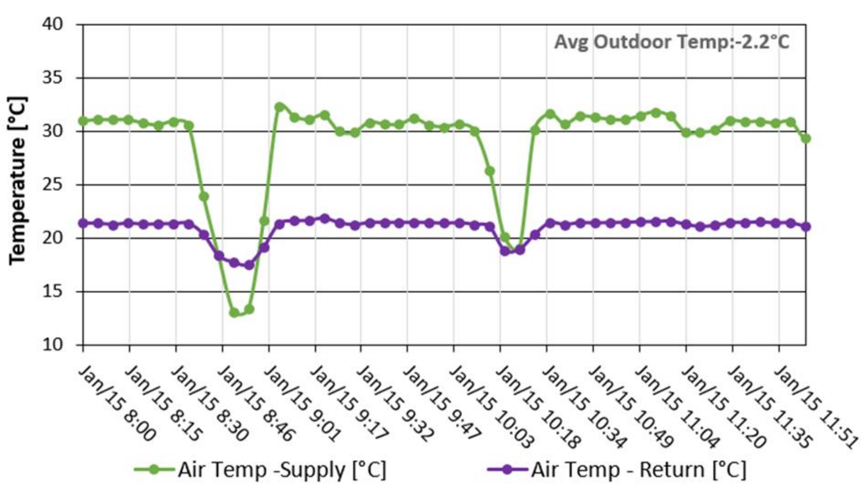



*Average indoor air temperature (return air) for monitored outdoor temperature range.*

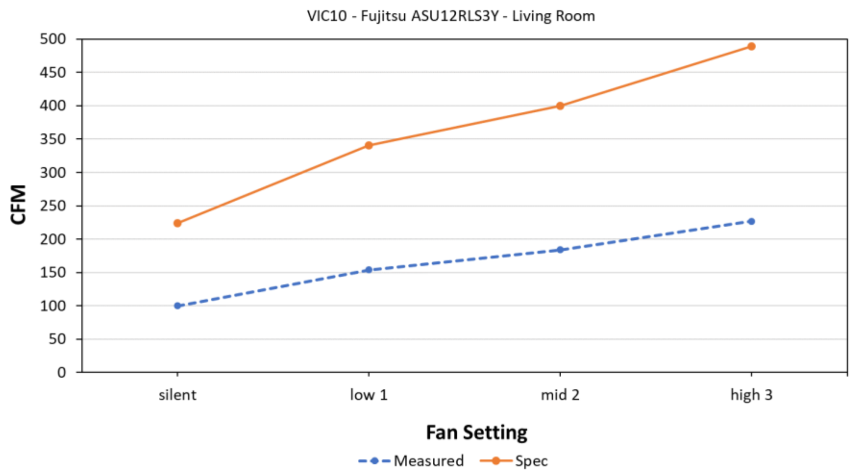



*Average outdoor relative humidity for monitored outdoor temperature range.*



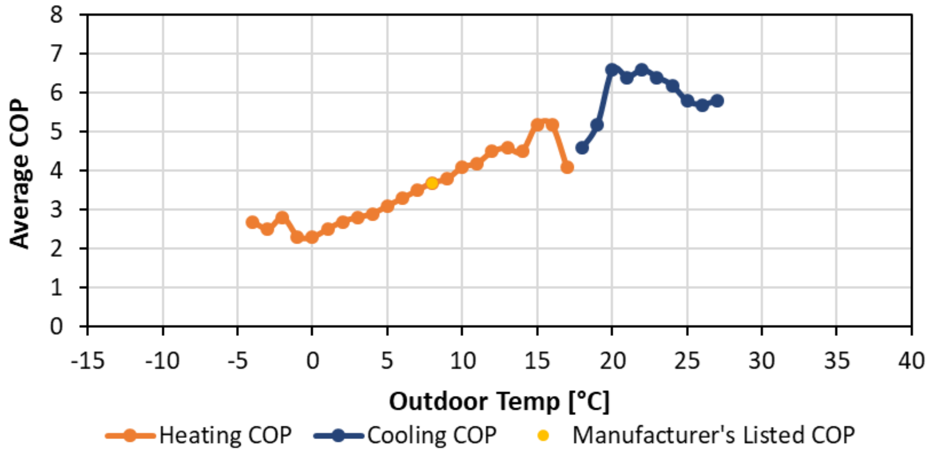




*Sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*

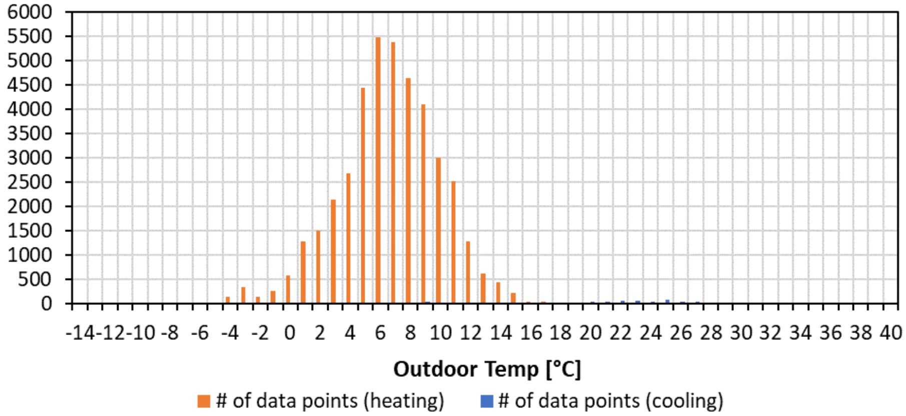



*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

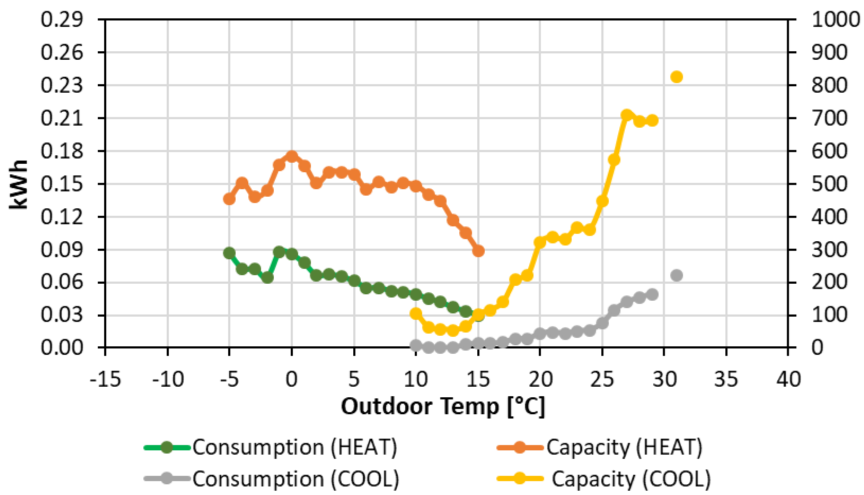
VIC12 – Ductless (Single Head) – Daikin: 3MXS24RMVJU | FTXS18LVJU



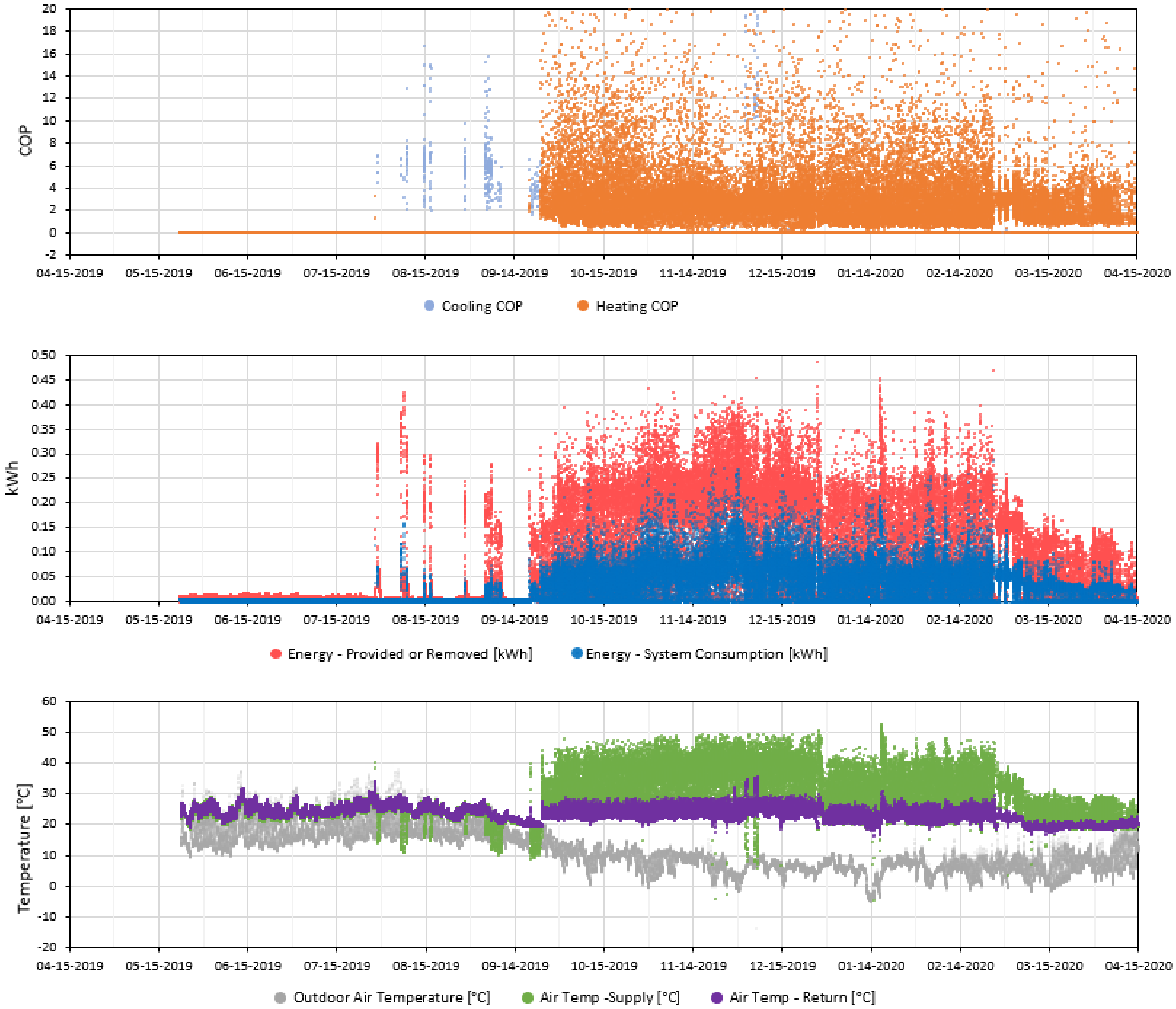

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



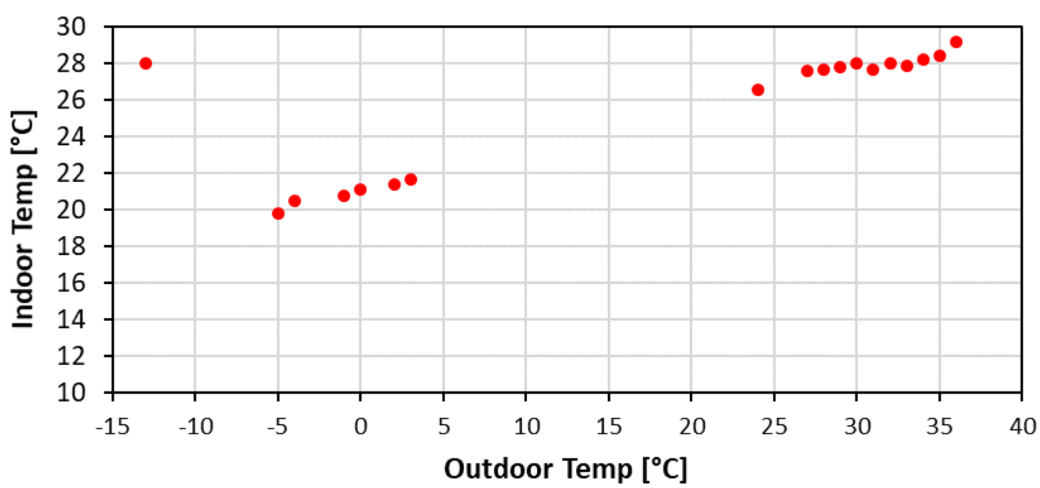

*Total number of heating and cooling data points throughout monitoring period.*



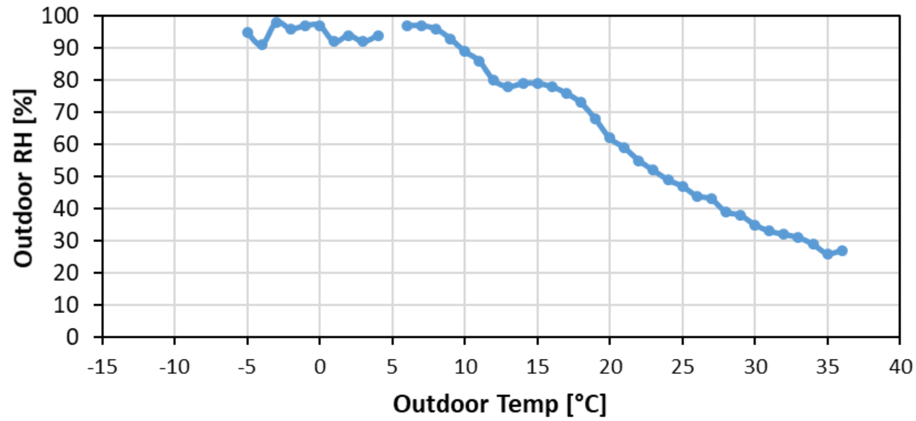

*Average system consumption and capacity for monitored outdoor temperature range.*



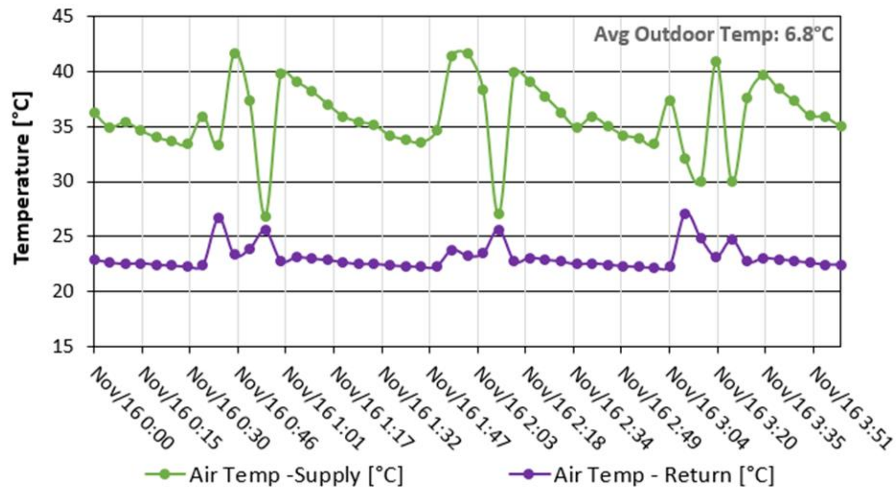

*Measured variables and corresponding COP for monitoring period.*

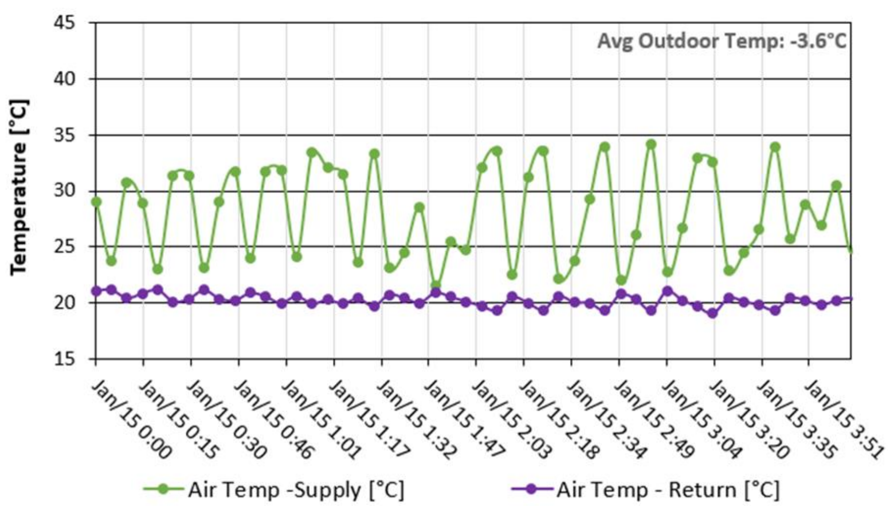



*Average indoor air temperature (return air) for monitored outdoor temperature range.*

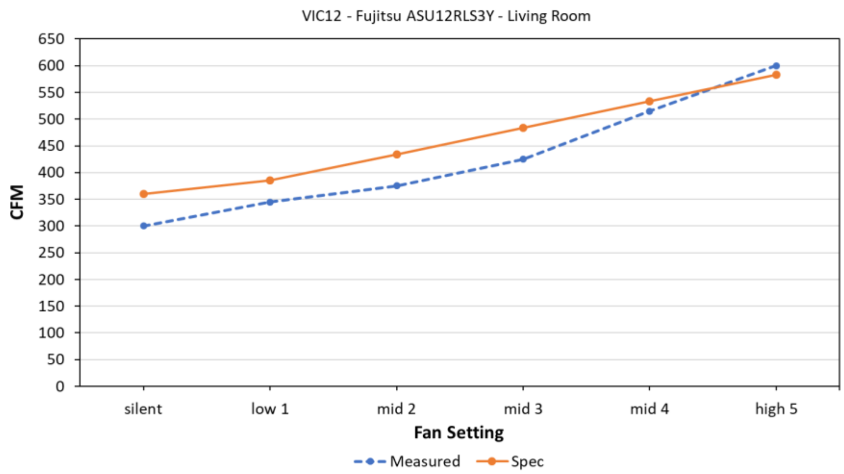



*Average outdoor relative humidity for monitored outdoor temperature range.*



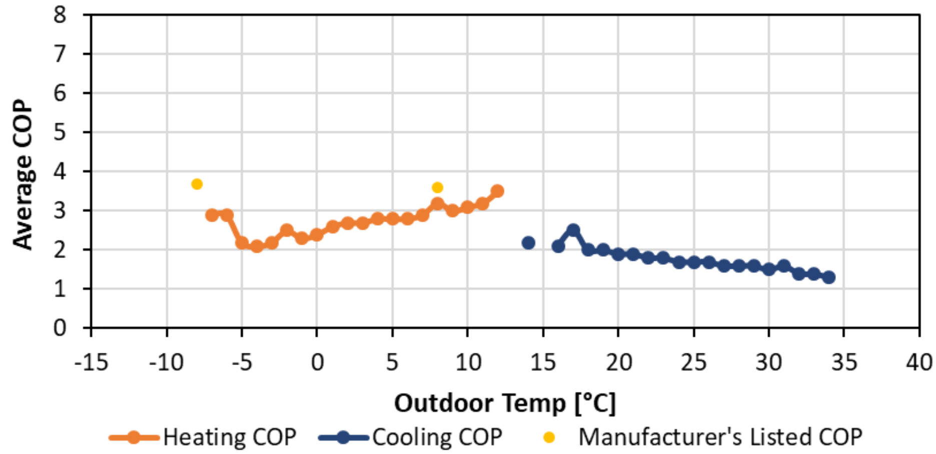

*Sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*



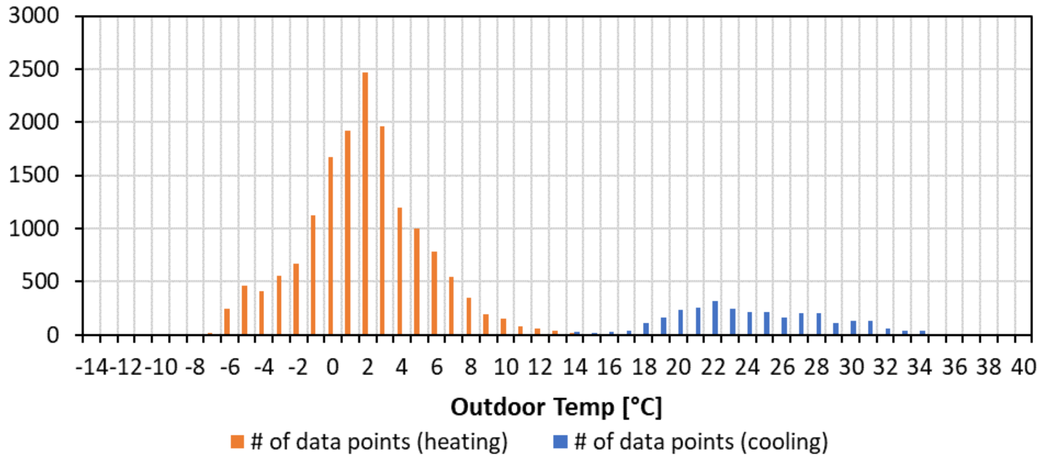

*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

## Central – Single Stage Systems

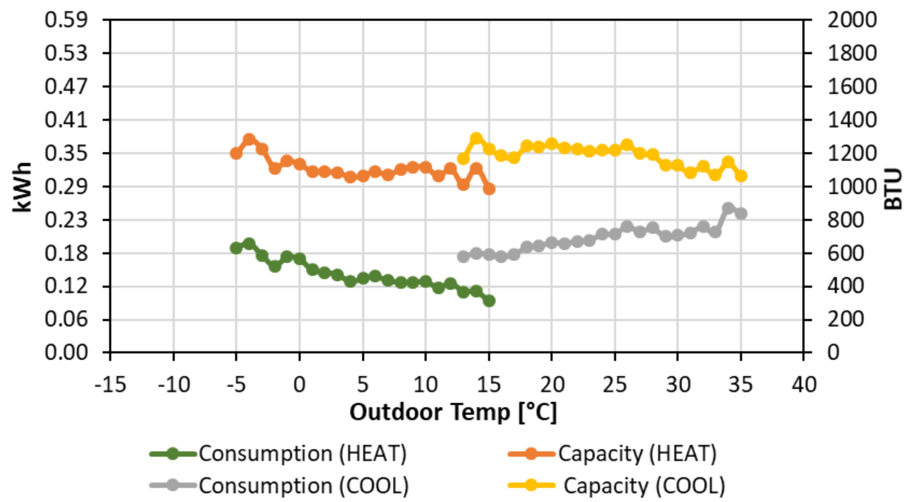
KEL04 – Central S – York: YZF04813CA



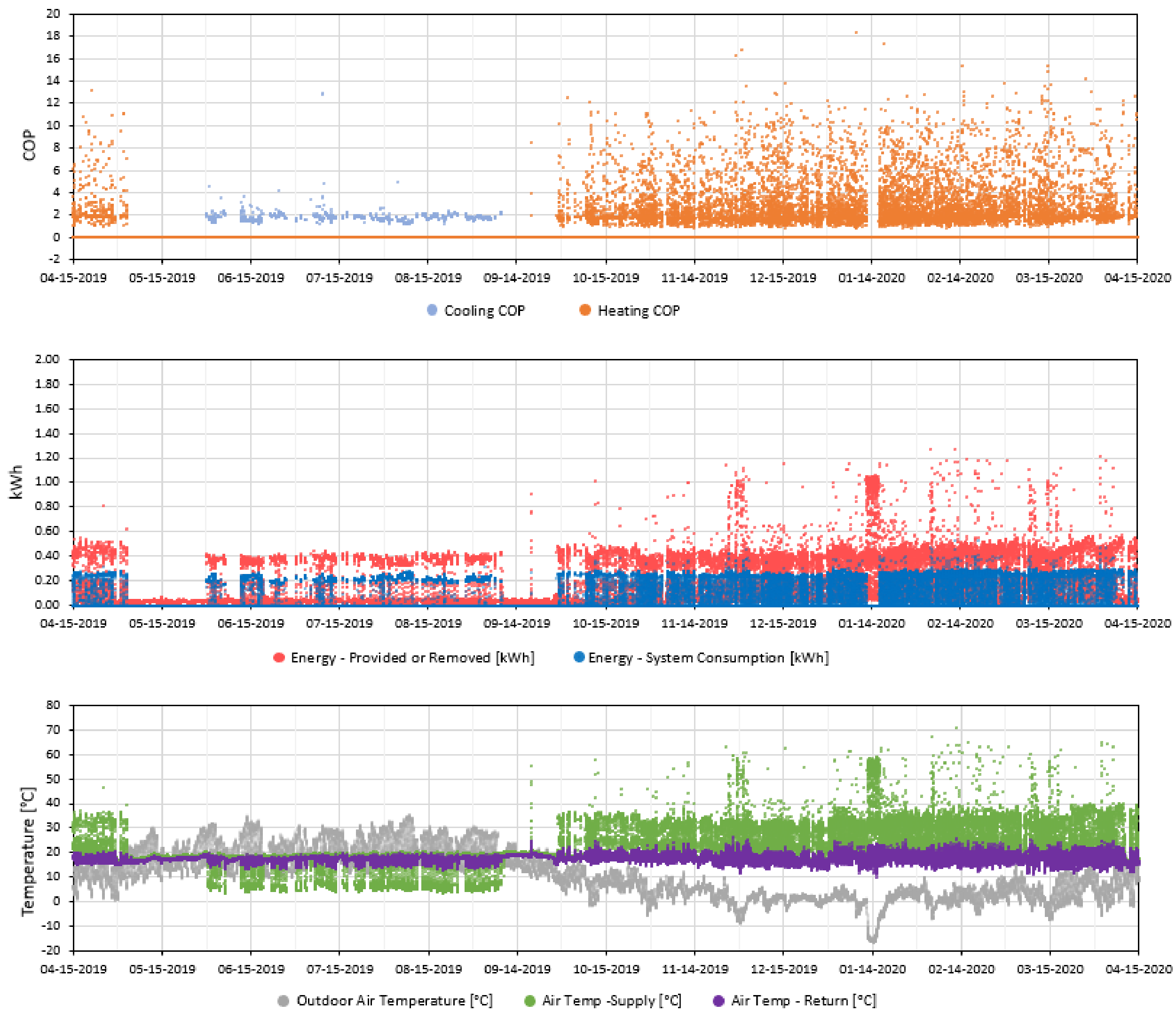

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



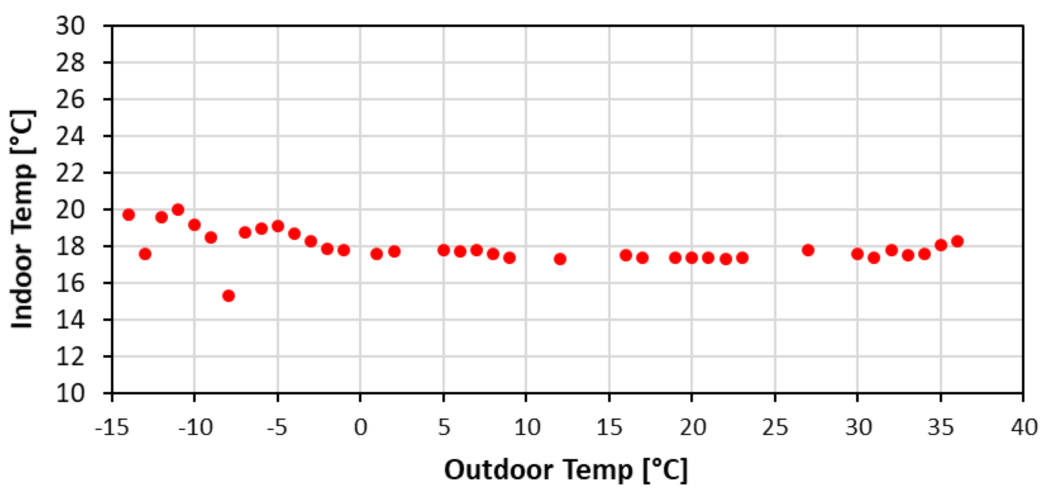

*Total number of heating and cooling data points throughout monitoring period.*



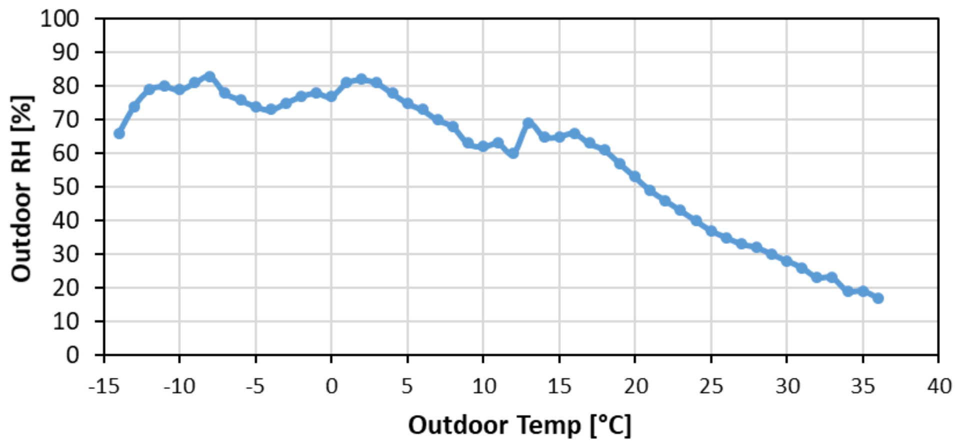

*Average system consumption and capacity for monitored outdoor temperature range.*



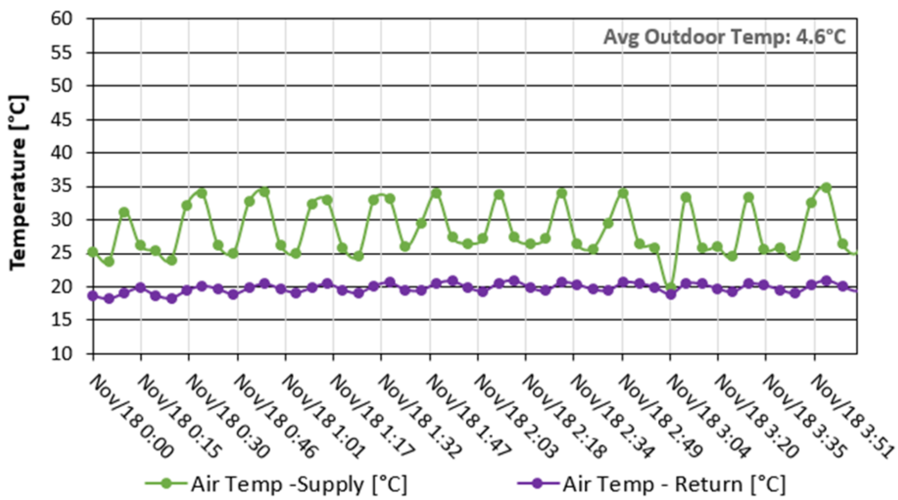

*Measured variables and corresponding COP for monitoring period.*

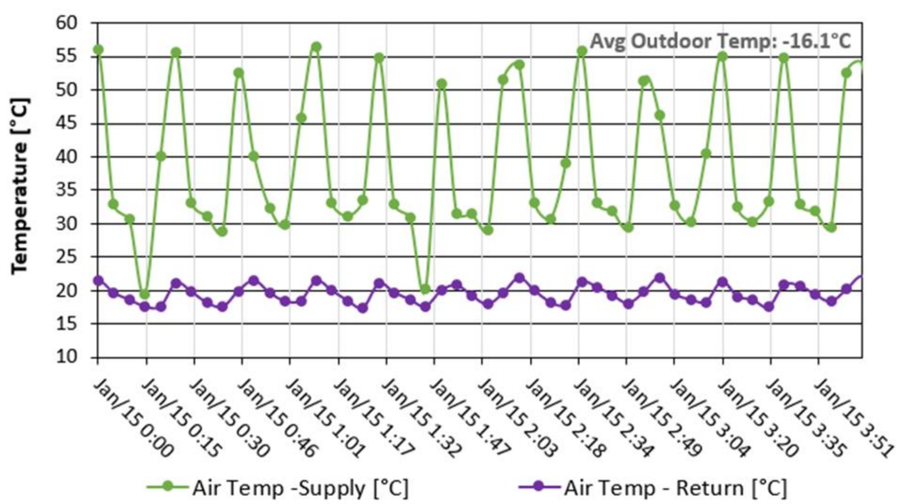



*Average indoor air temperature (return air) for monitored outdoor temperature range.*

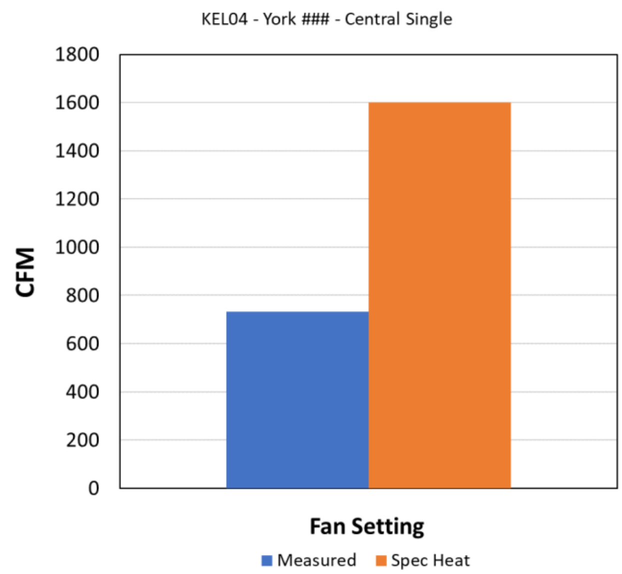



*Average outdoor relative humidity for monitored outdoor temperature range.*



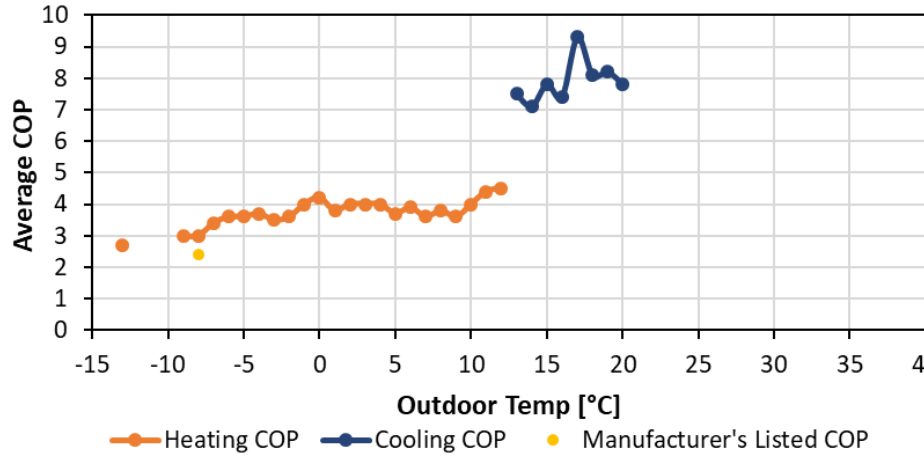




*Sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*

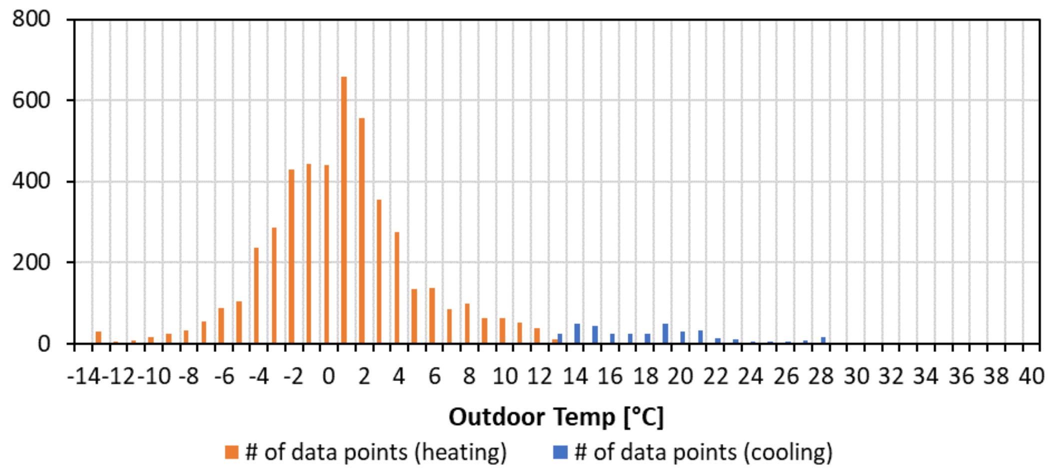



*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

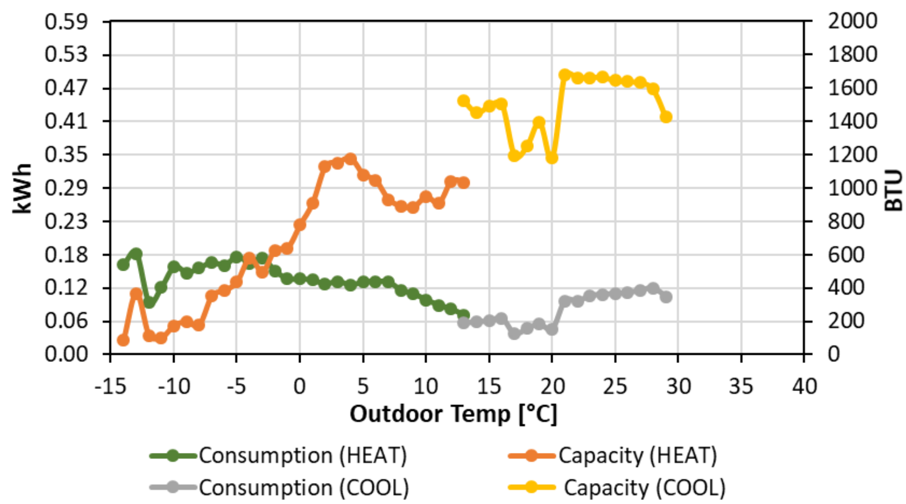
SUM01 – Central S – Fujitsu: PH14NB030-A | CNPVU3017ALA



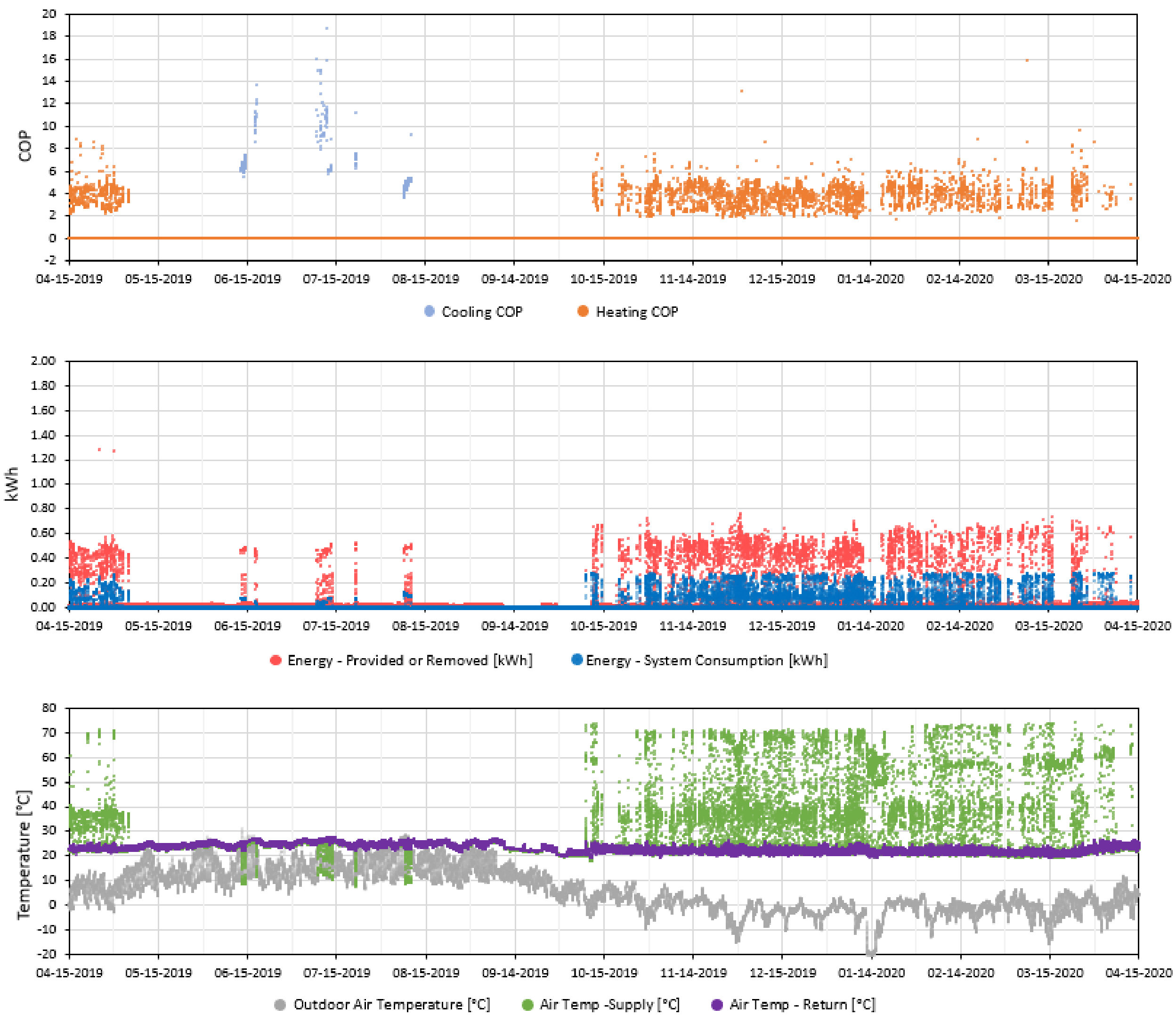

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



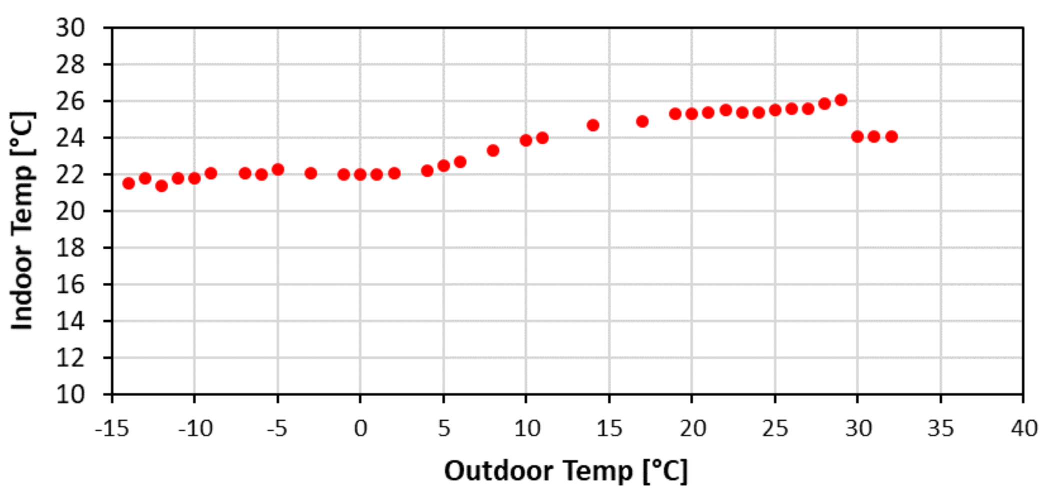

*Total number of heating and cooling data points throughout monitoring period.*



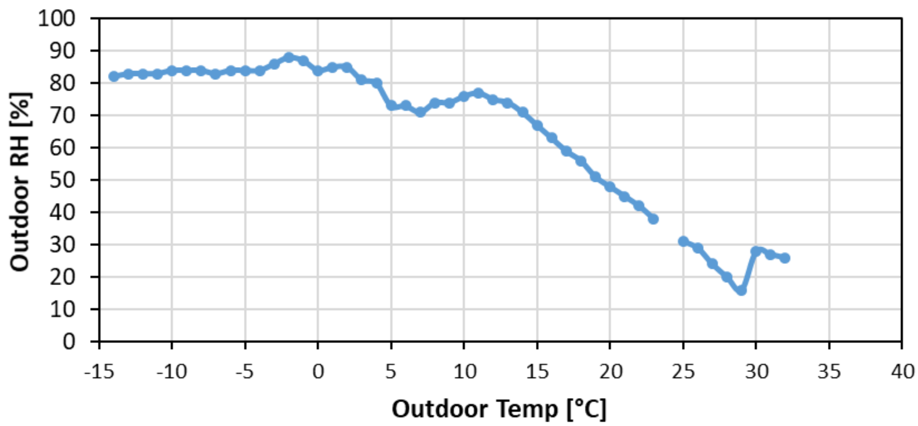

*Average system consumption and capacity for monitored outdoor temperature range.*



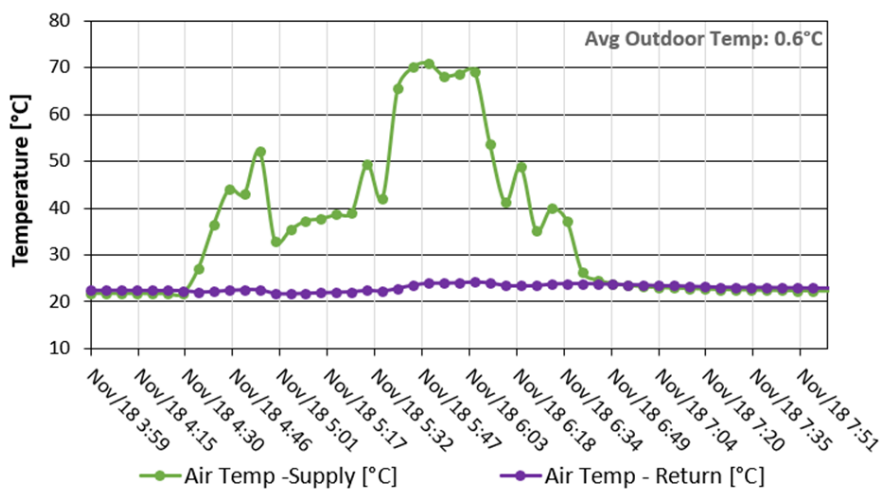

*Measured variables and corresponding COP for monitoring period.*

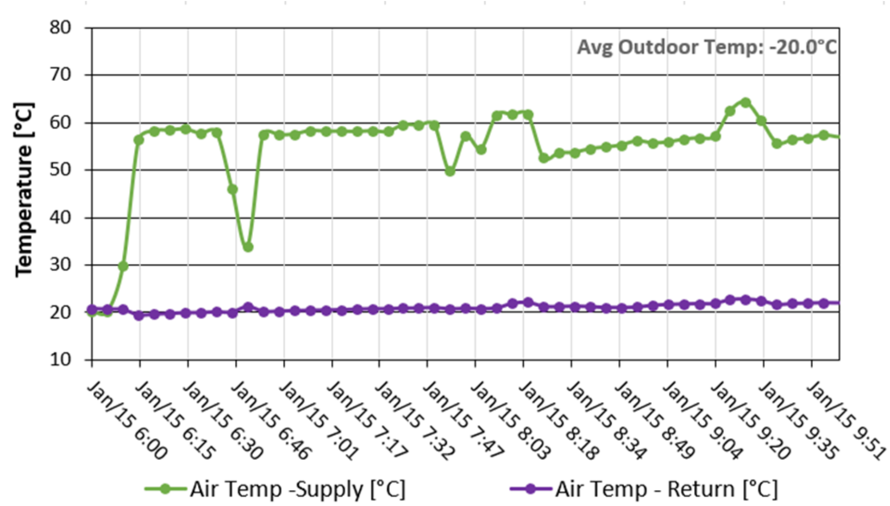



*Average indoor air temperature (return air) for monitored outdoor temperature range.*

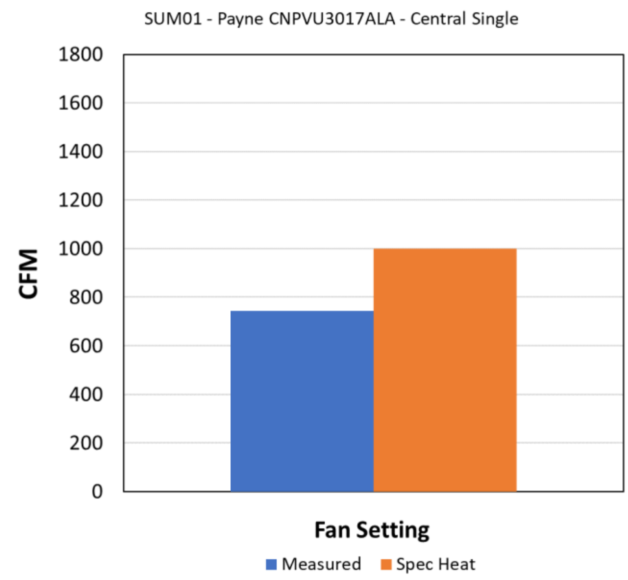



*Average outdoor relative humidity for monitored outdoor temperature range.*



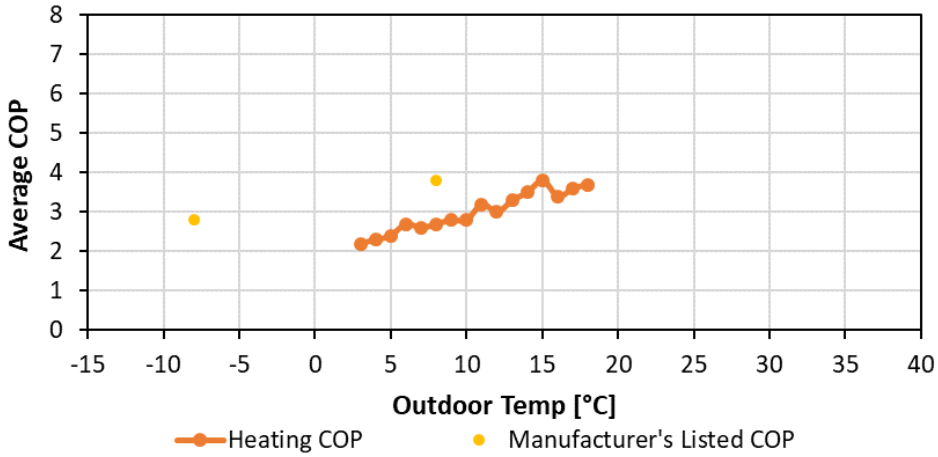




*Sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*

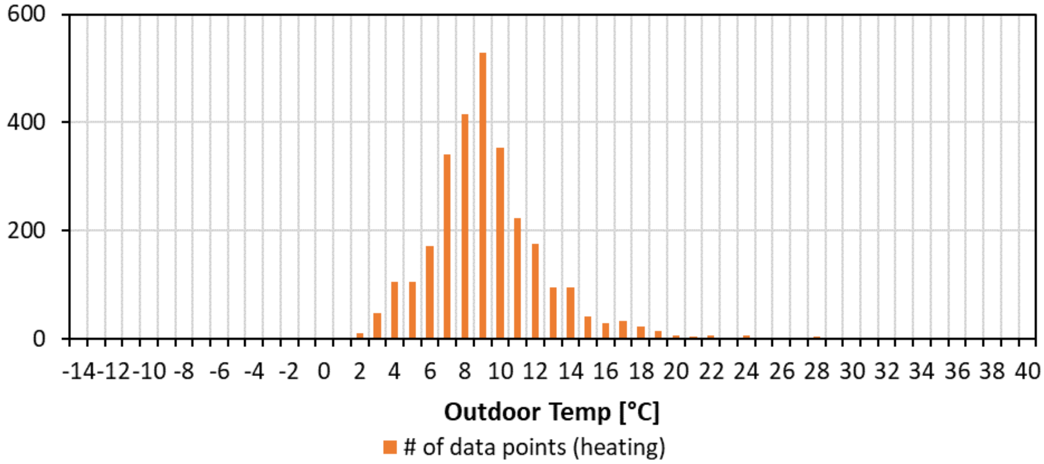



*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

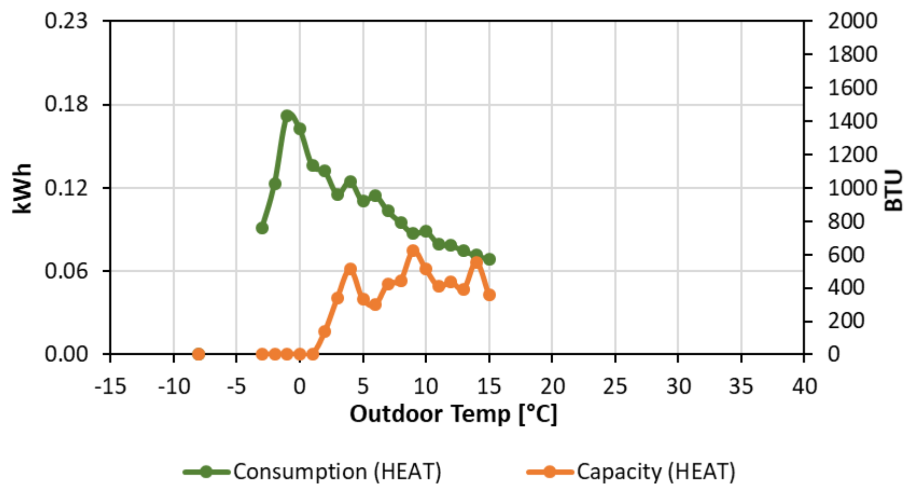
VIC04 – Central S – York: YZF03013CA | AHV36C3XH21CC



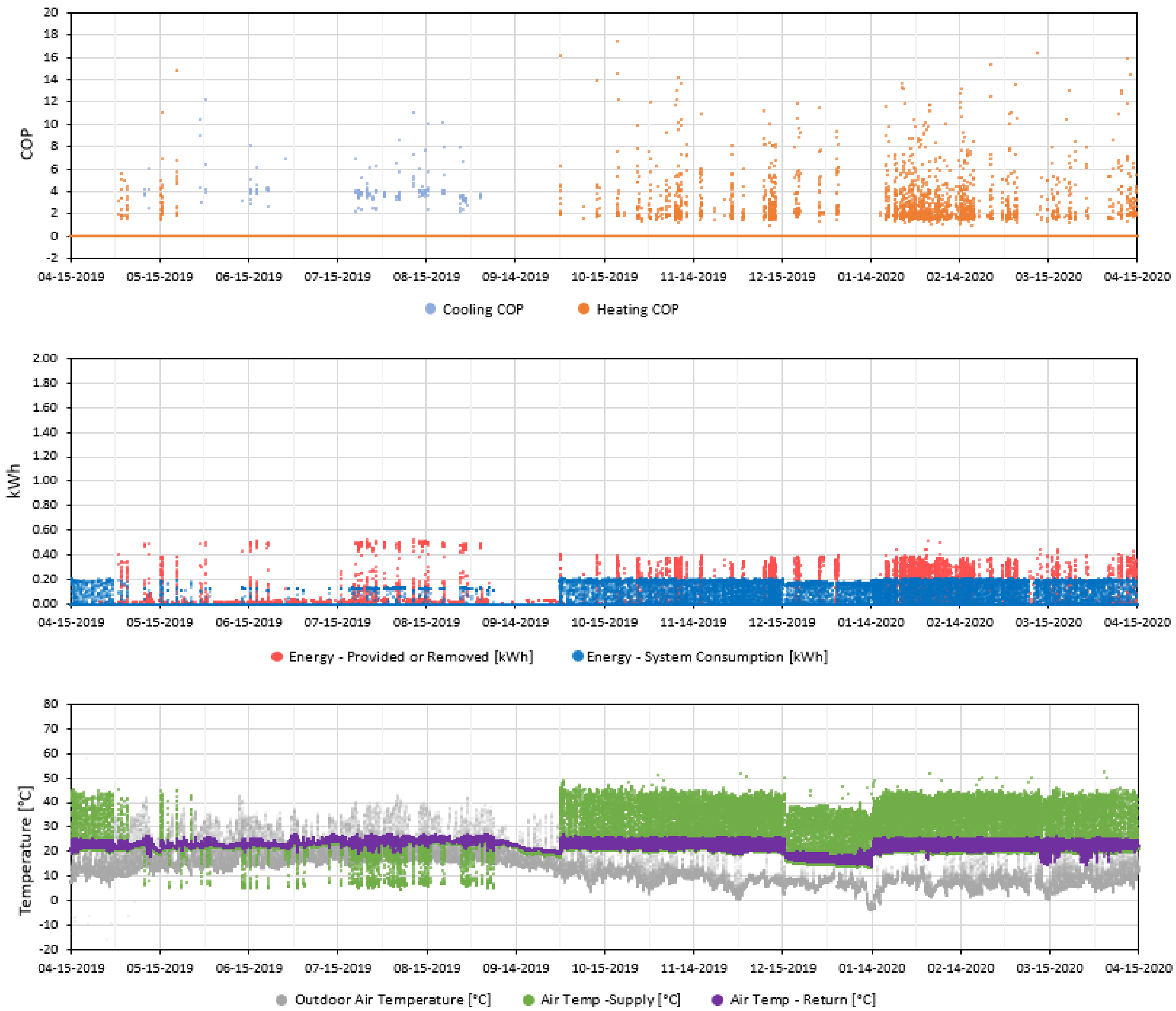

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



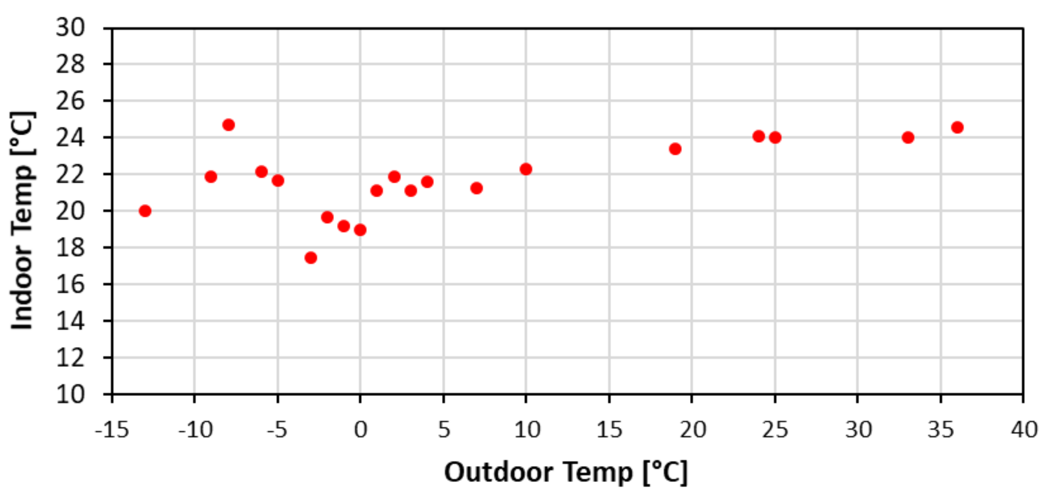

*Total number of heating and cooling data points throughout monitoring period.*



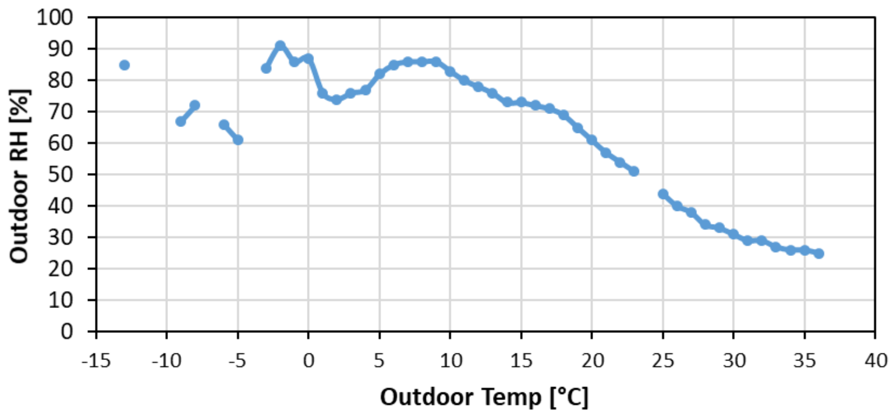

*Average system consumption and capacity for monitored outdoor temperature range.*



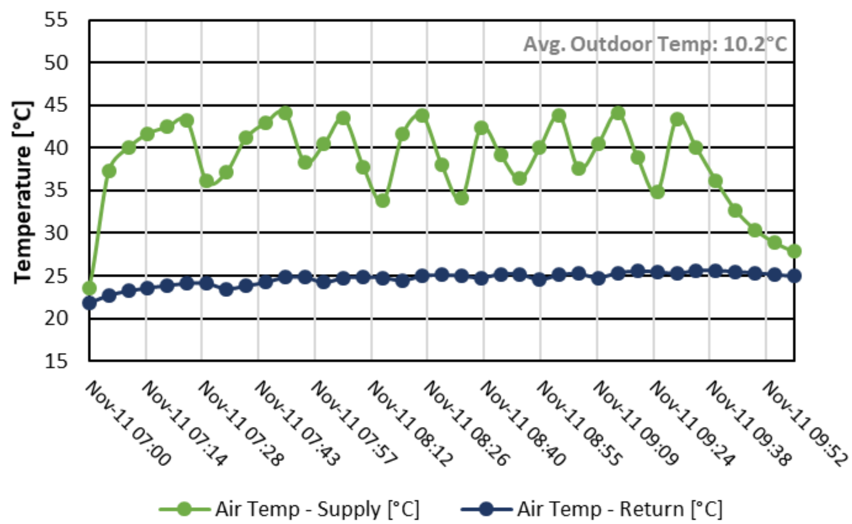

*Measured variables and corresponding COP for monitoring period.*

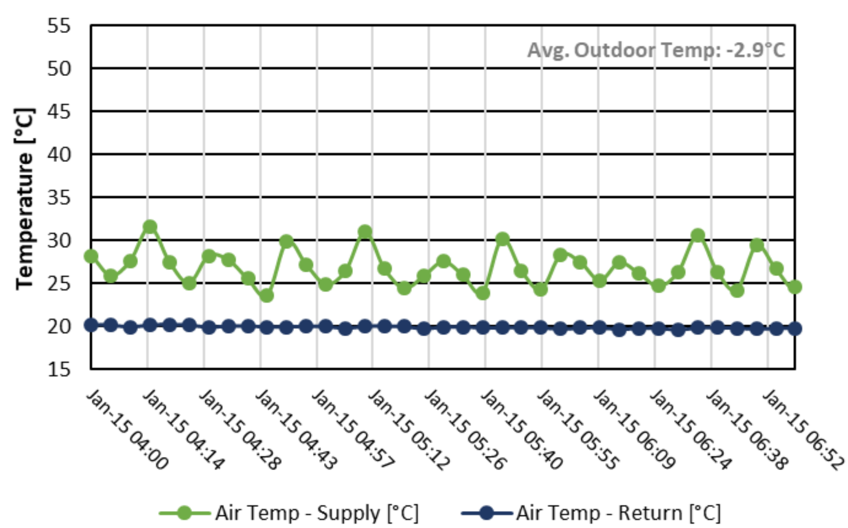



*Average indoor air temperature (return air) for monitored outdoor temperature range.*

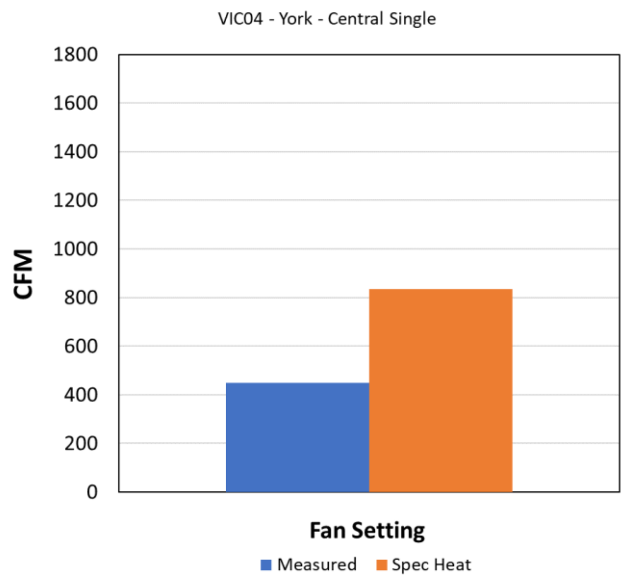



*Average outdoor relative humidity for monitored outdoor temperature range.*



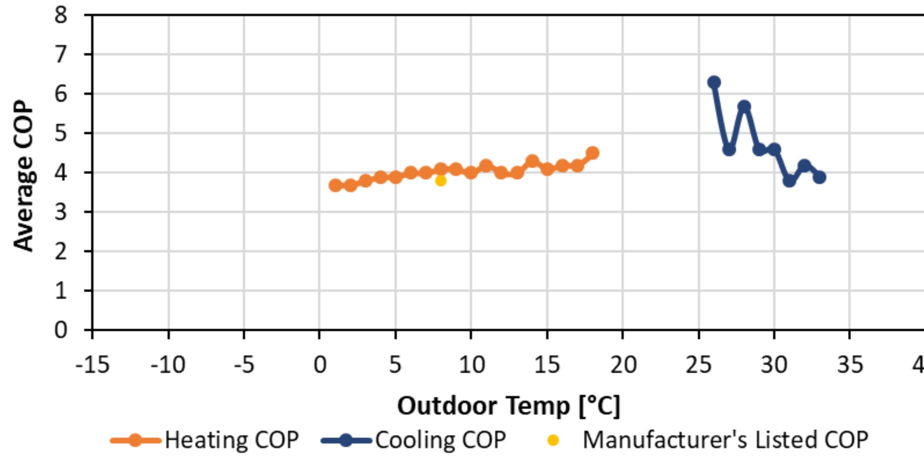




*Sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*

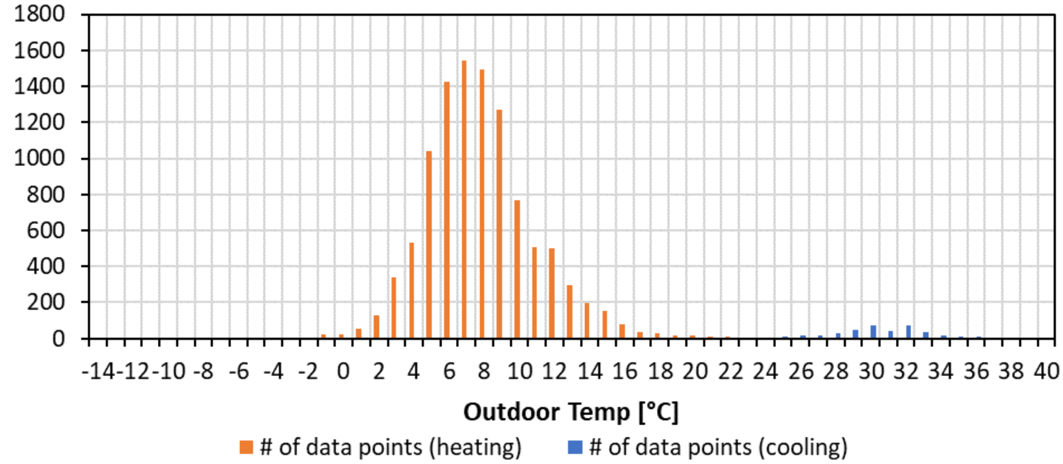



*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

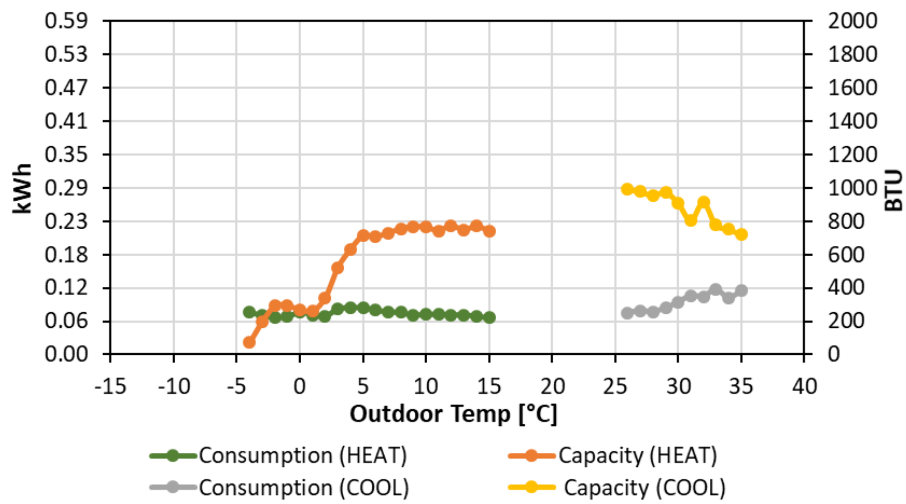
VIC08 – Central S – York: YZF03013CA | AHV36C3XH21CC



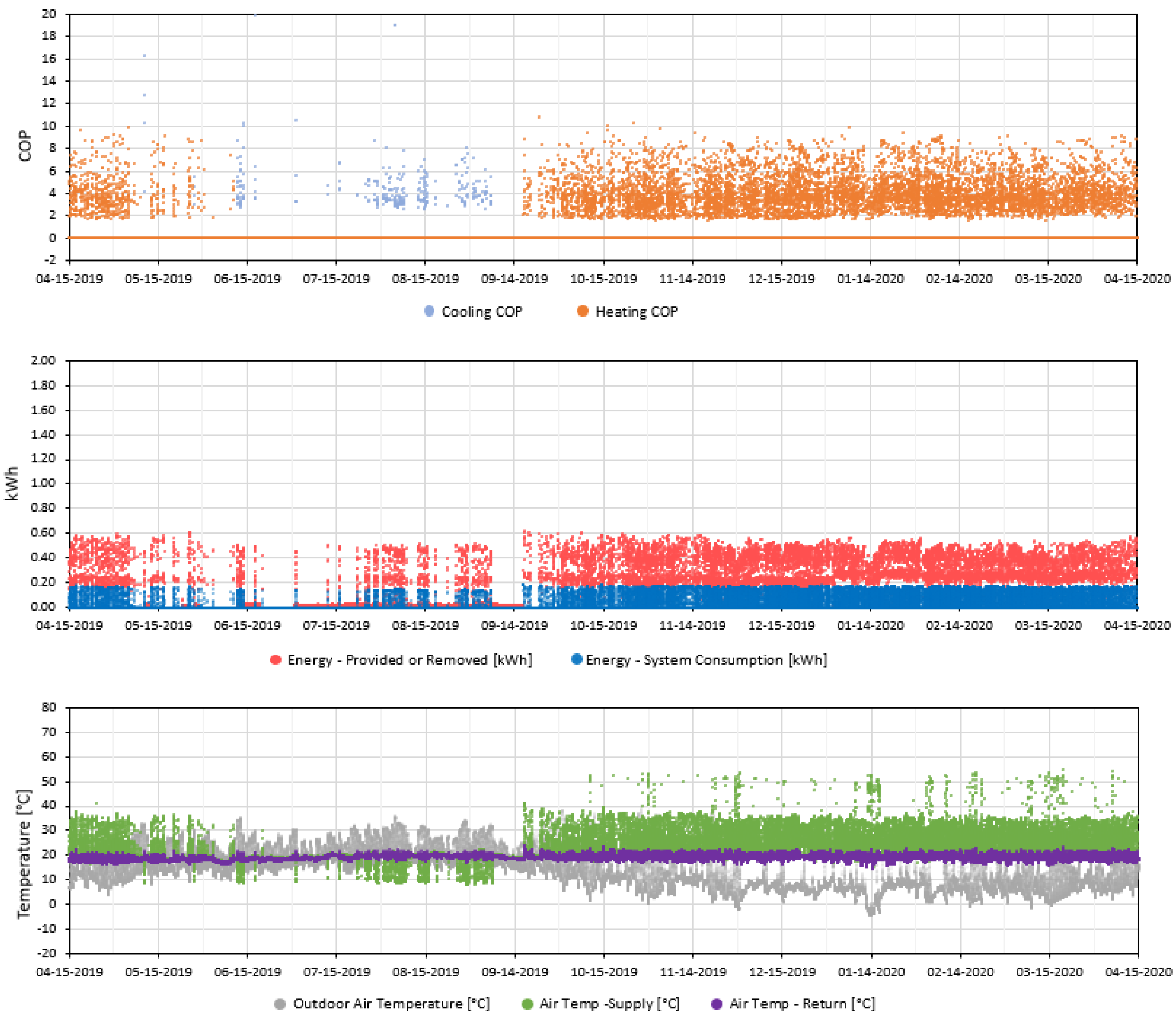

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



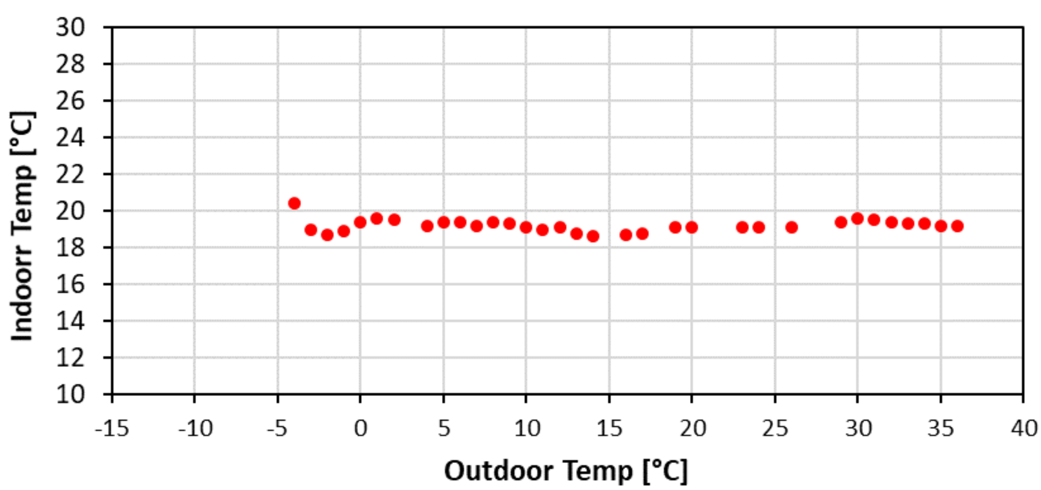

*Total number of heating and cooling data points throughout monitoring period.*



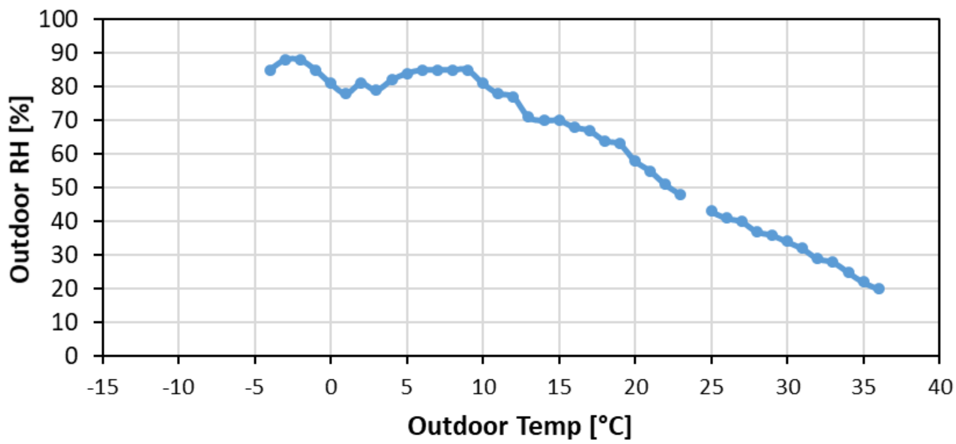

*Average system consumption and capacity for monitored outdoor temperature range.*



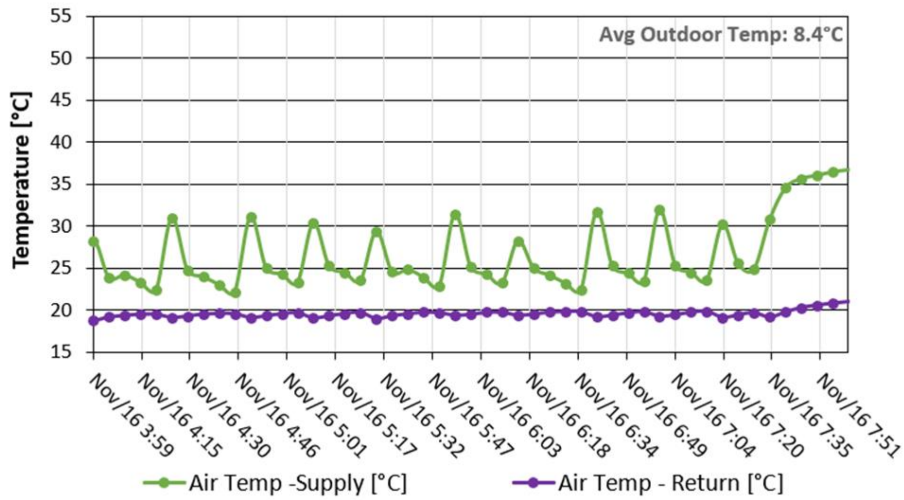

*Measured variables and corresponding COP for monitoring period.*

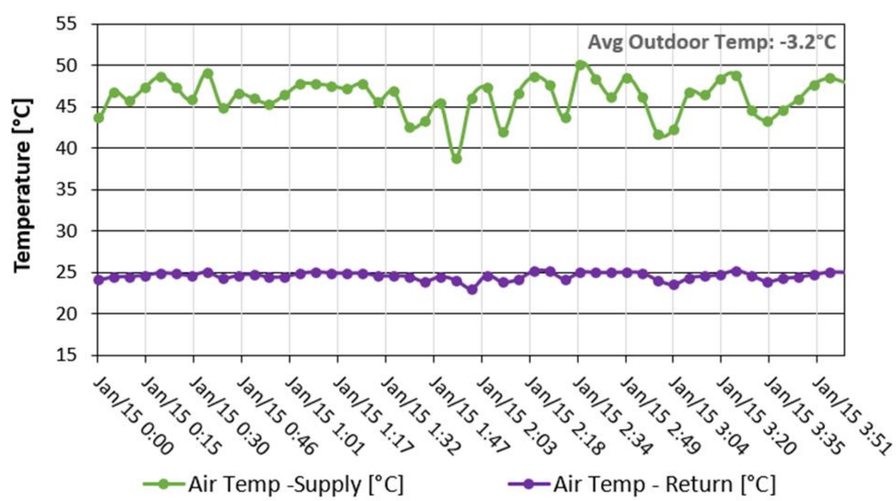



*Average indoor air temperature (return air) for monitored outdoor temperature range.*

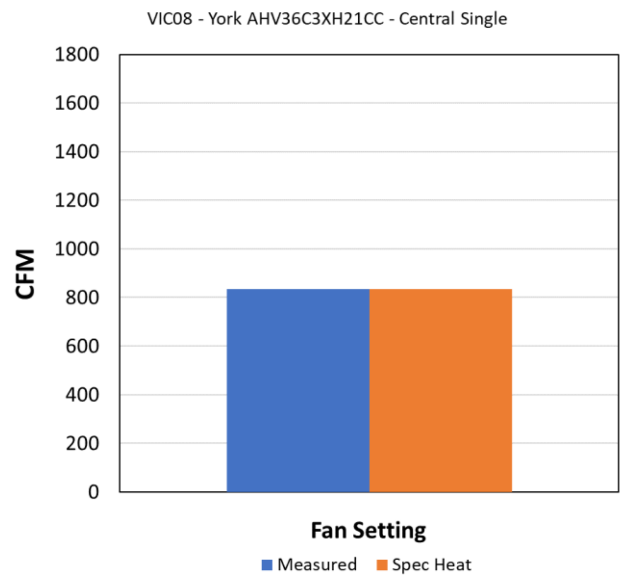



*Average outdoor relative humidity for monitored outdoor temperature range.*



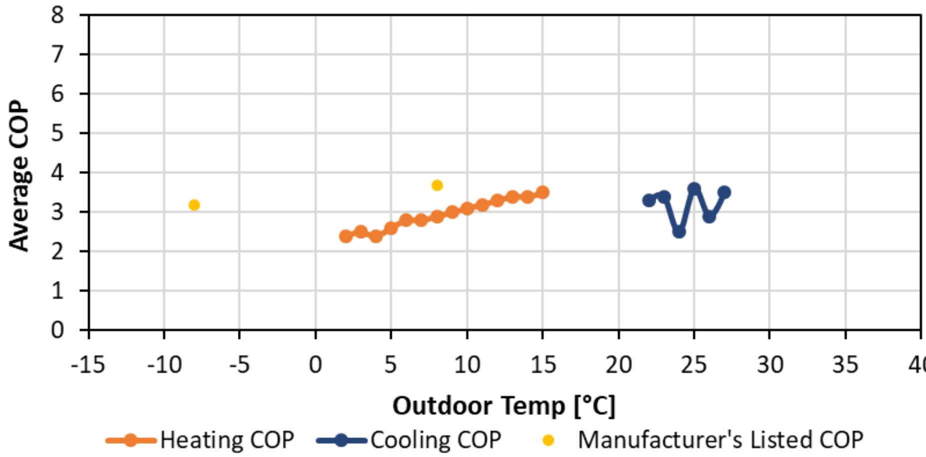




*Sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*

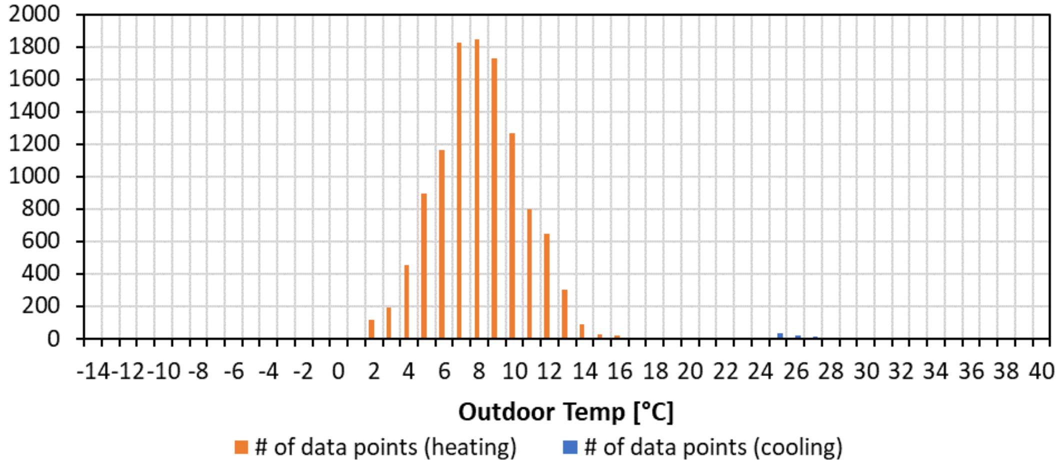



*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

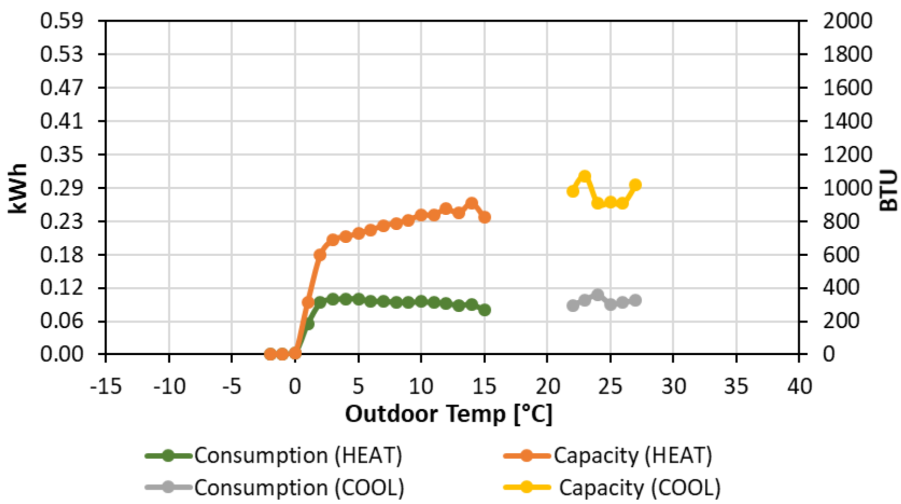
VIC11 – Central S – Lennox: XP14-024-230-09 | CBX32MV-024/030-230-6-08




*Average estimated heating and cooling COP for monitored outdoor temperature range.*

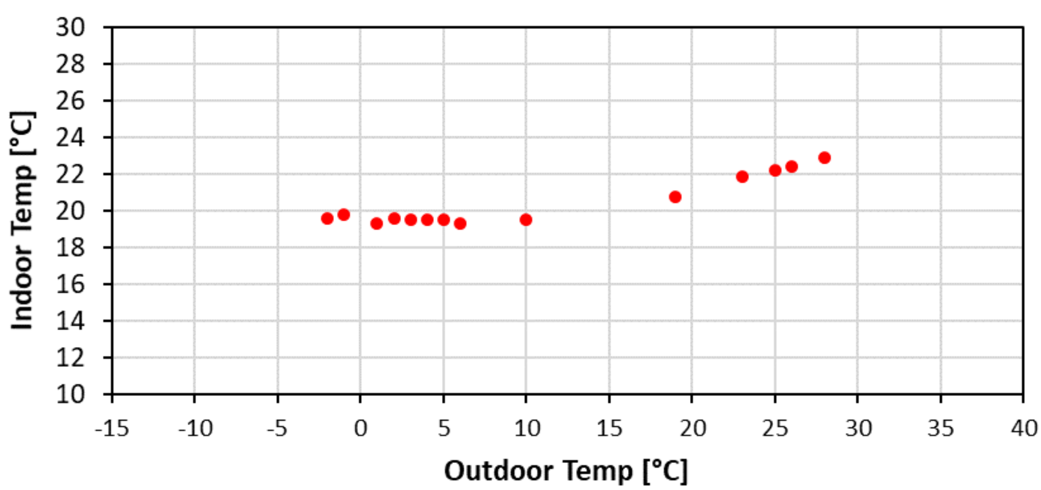



*Total number of heating and cooling data points throughout monitoring period.*

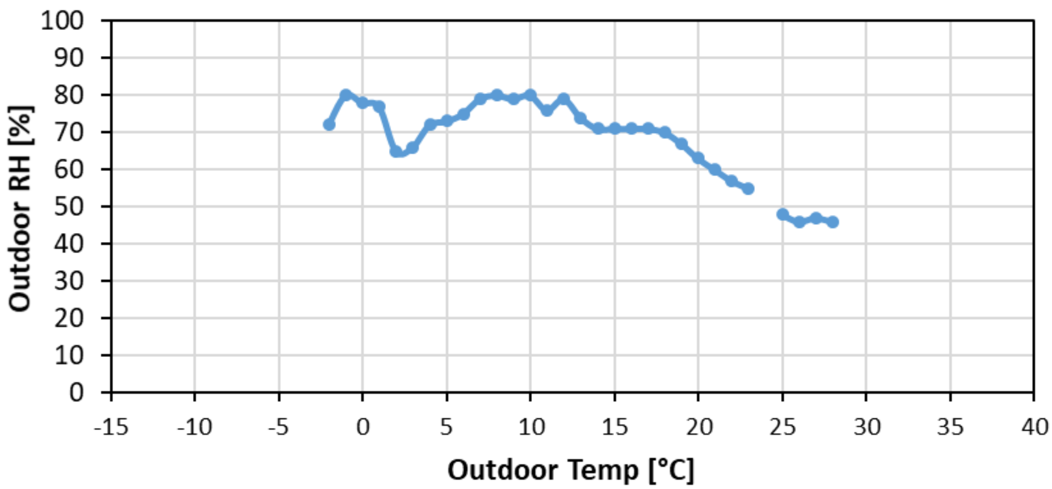



*Average system consumption and capacity for monitored outdoor temperature range.*

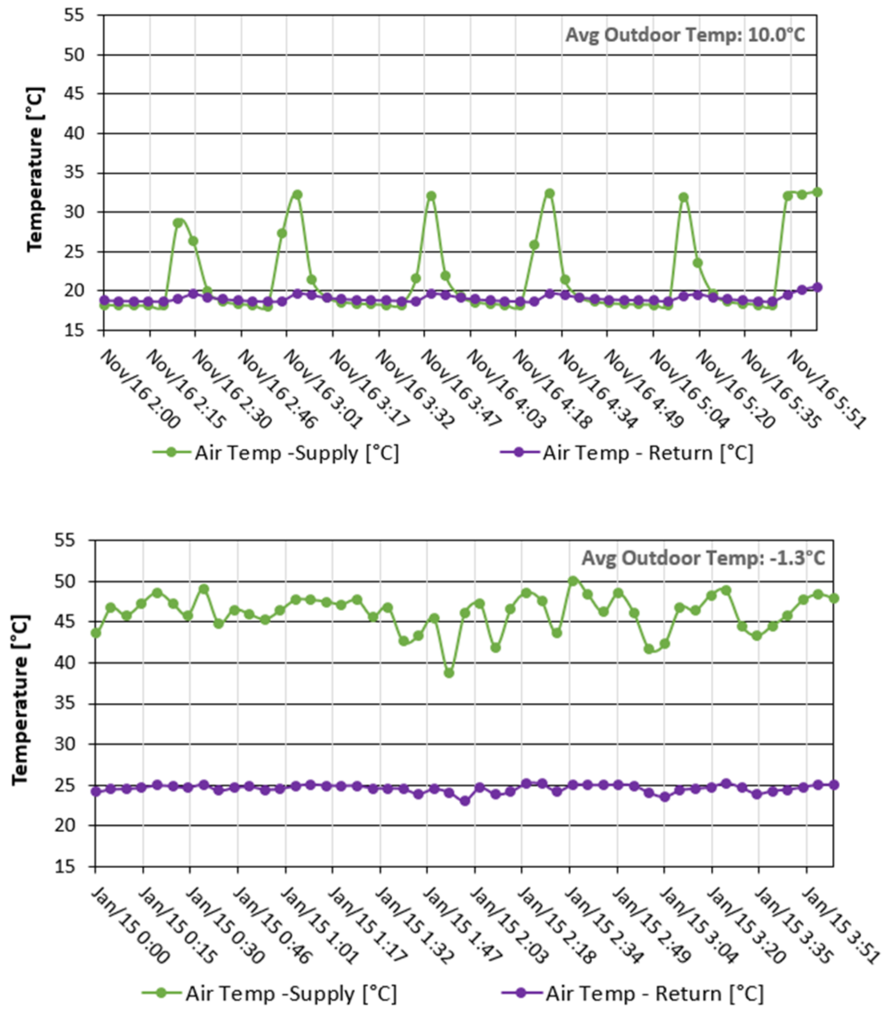

*Measured variables and corresponding COP for monitoring period.*



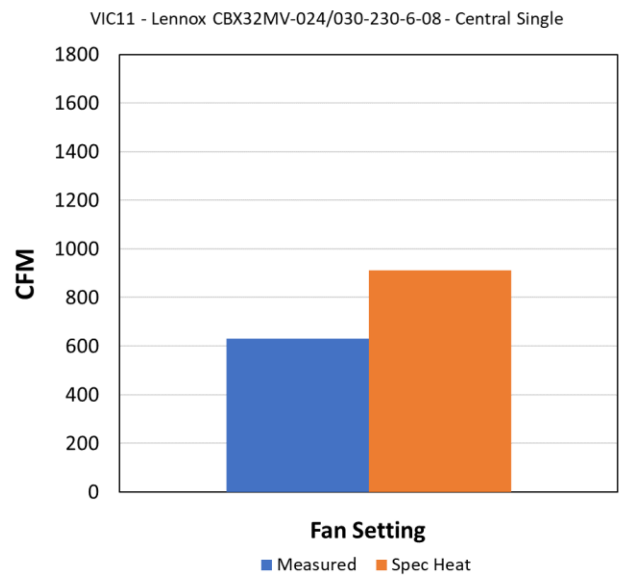

*Average indoor air temperature (return air) for monitored outdoor temperature range.*




*Average outdoor relative humidity for monitored outdoor temperature range.*



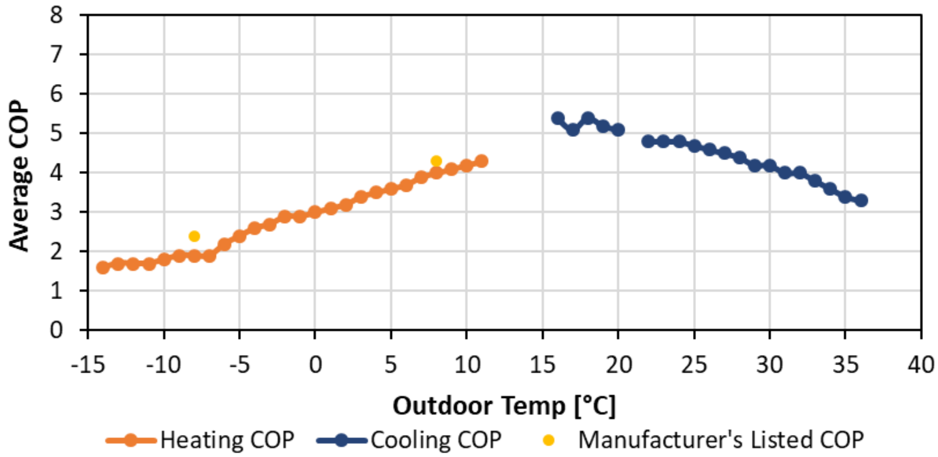

*Sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*



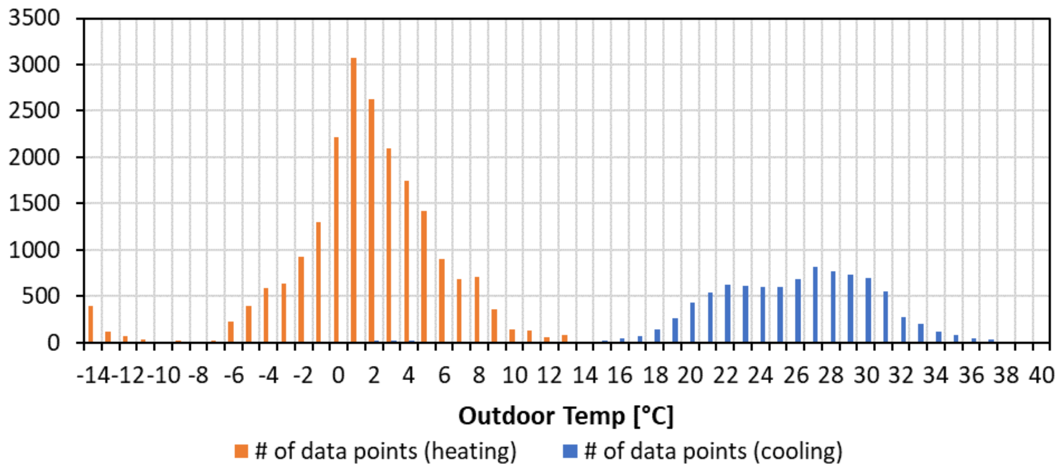

*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

## Central – Variable Speed Systems

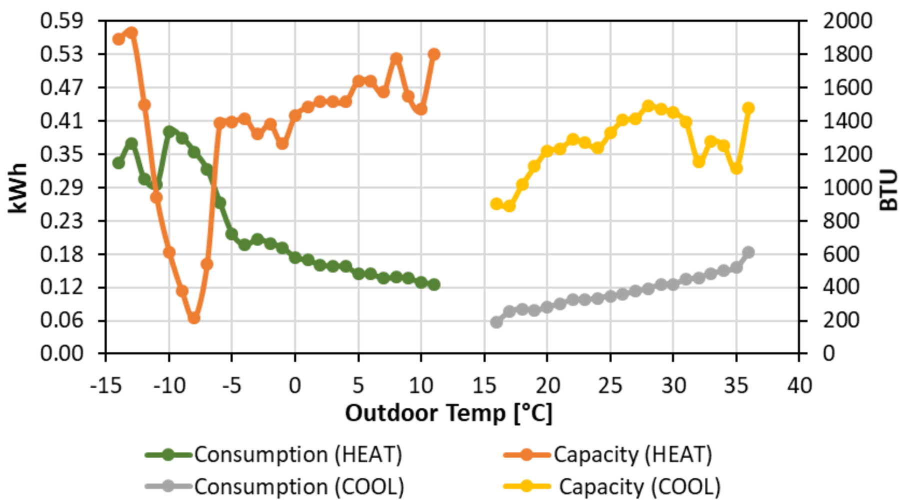
PEN01 – Central V – Mitsubishi: PUZ-HA36NHA5 | PVA-A36AA4



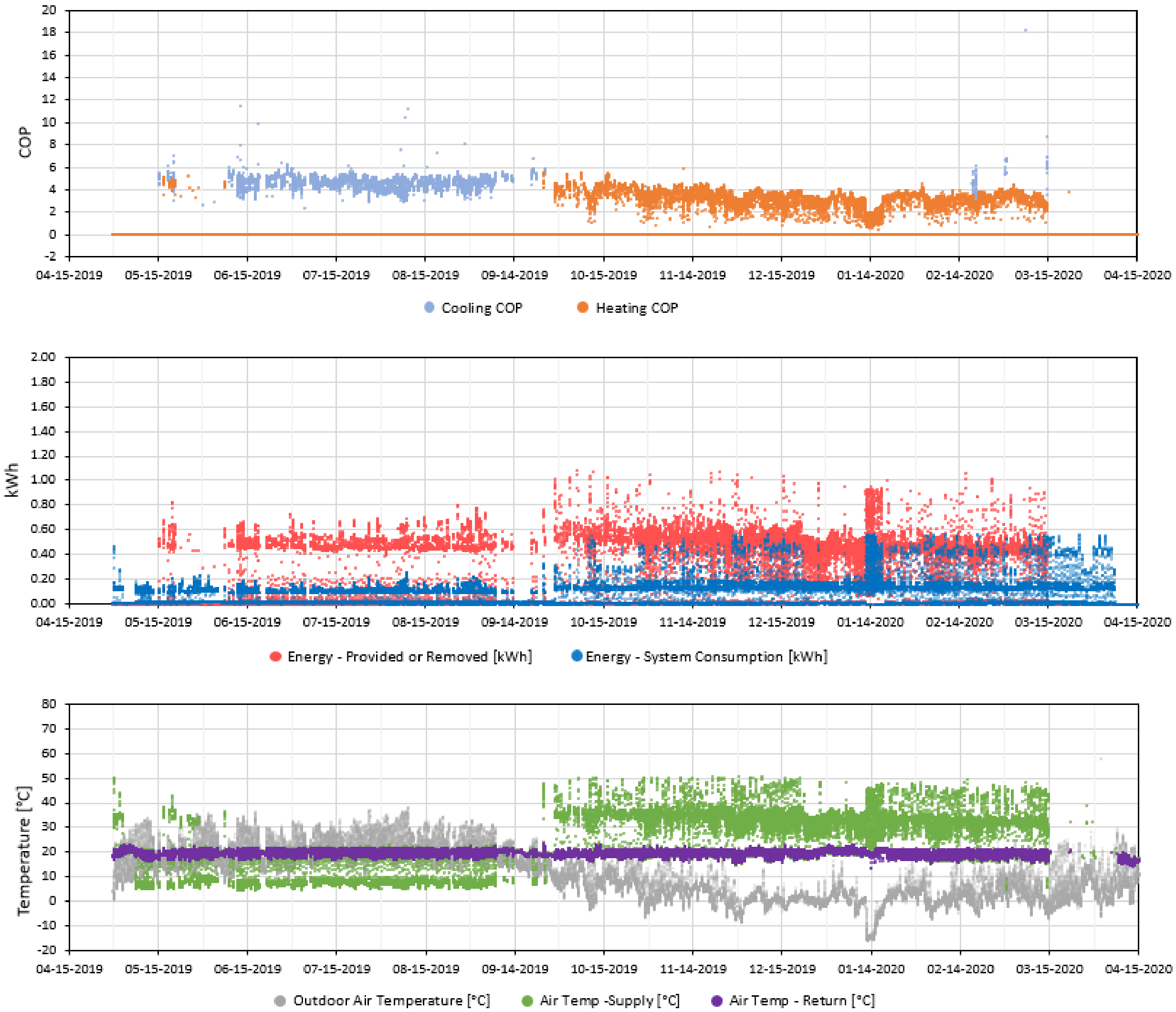

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



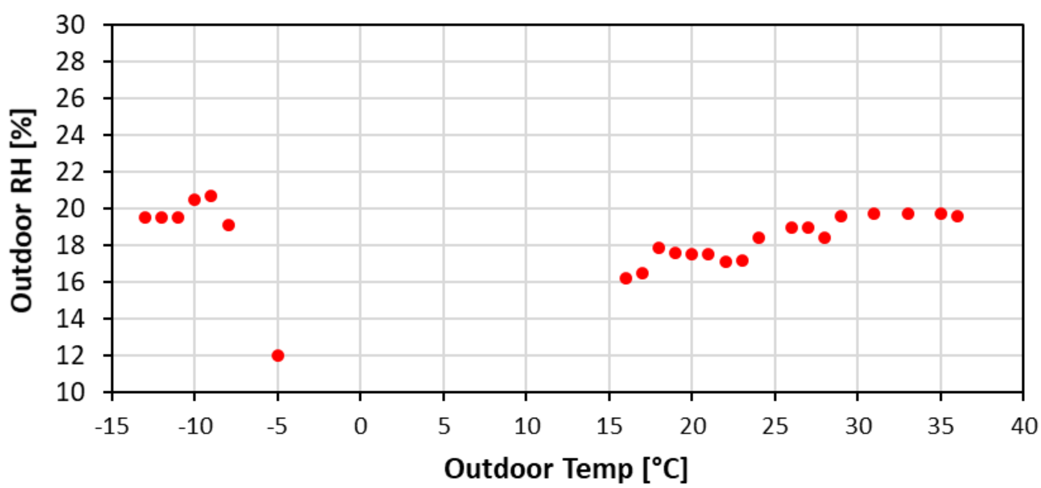

*Total number of heating and cooling data points throughout monitoring period.*



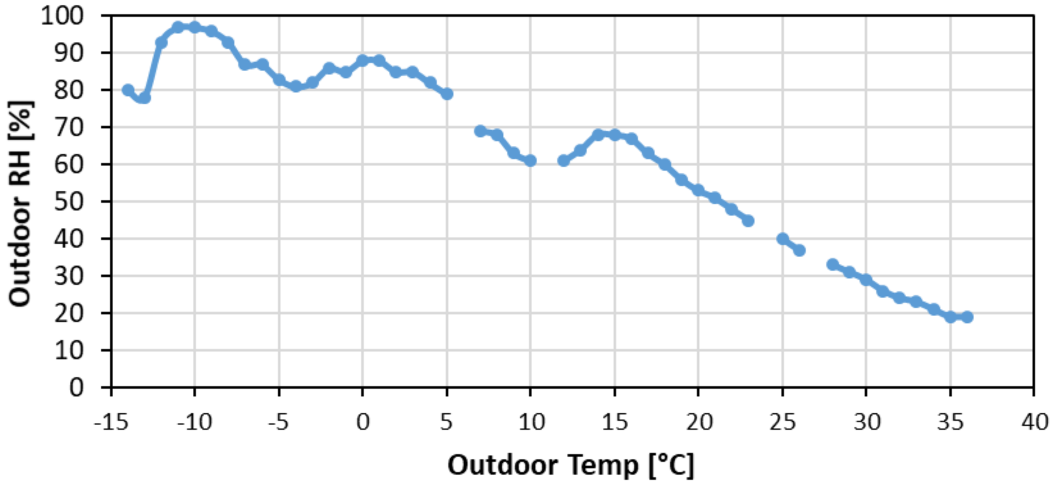

*Average system consumption and capacity for monitored outdoor temperature range.*



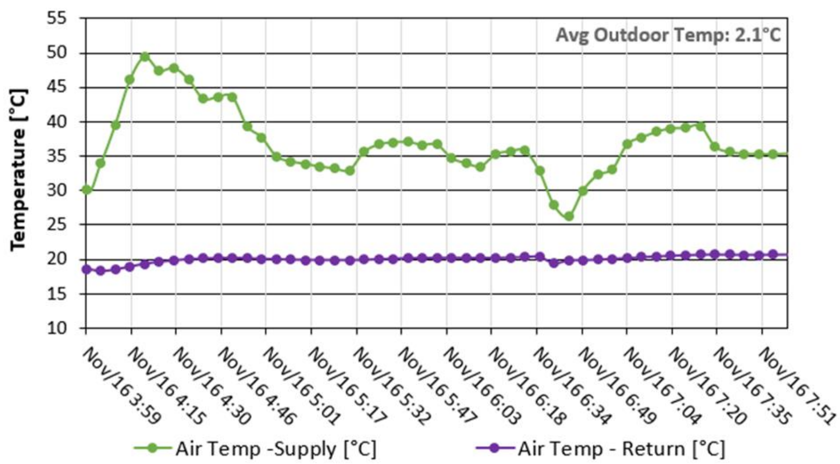

*Measured variables and corresponding COP for monitoring period.*

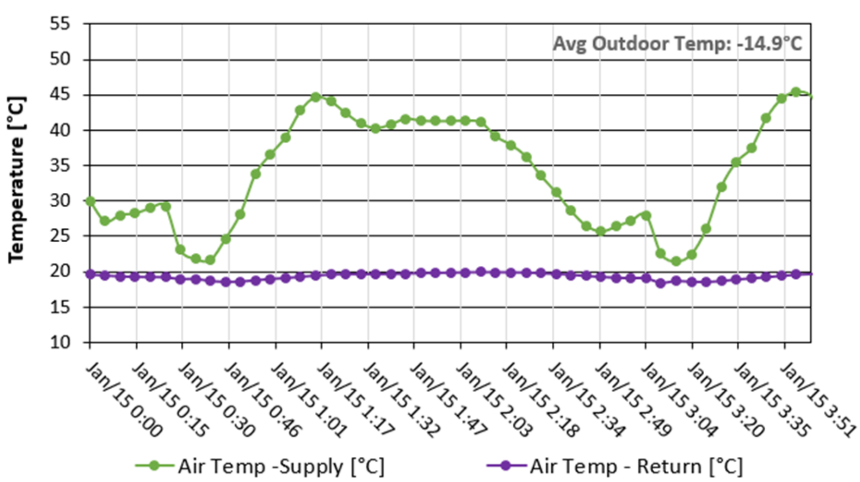



*Average indoor air temperature (return air) for monitored outdoor temperature range.*

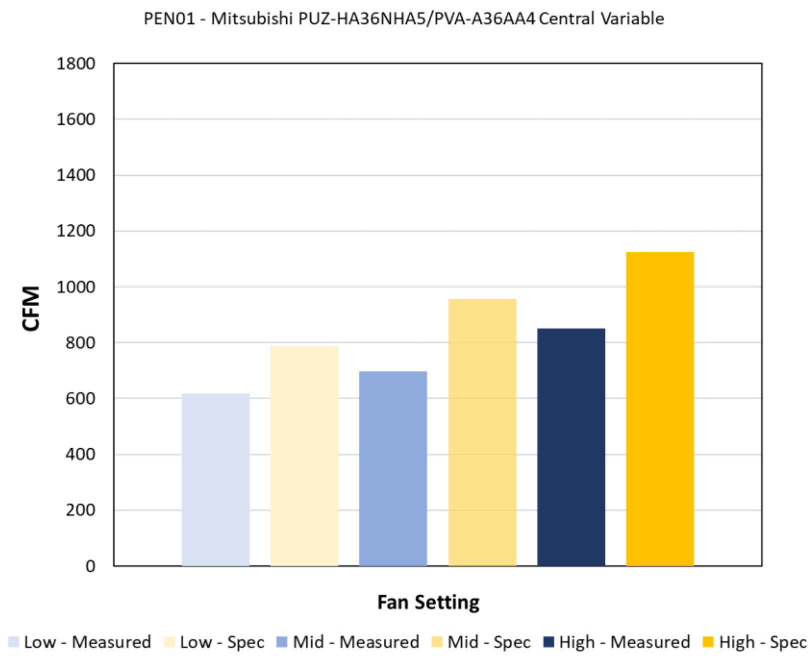



*Average outdoor relative humidity for monitored outdoor temperature range.*



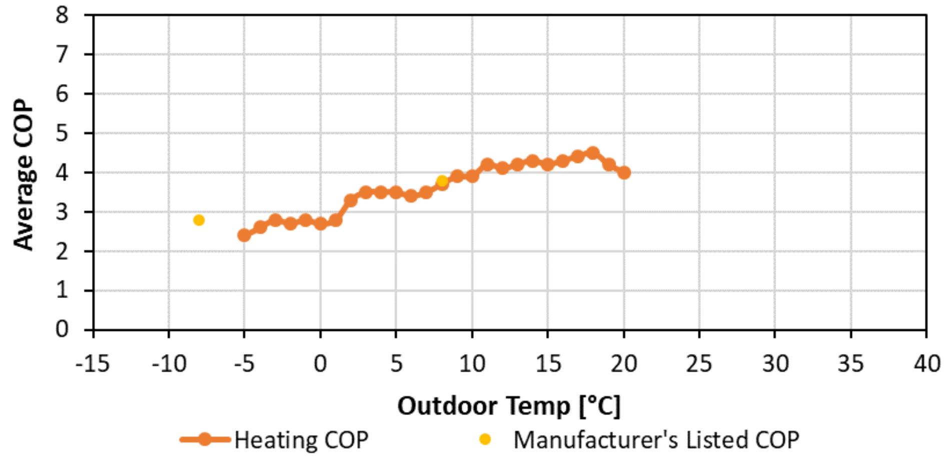




*Sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*

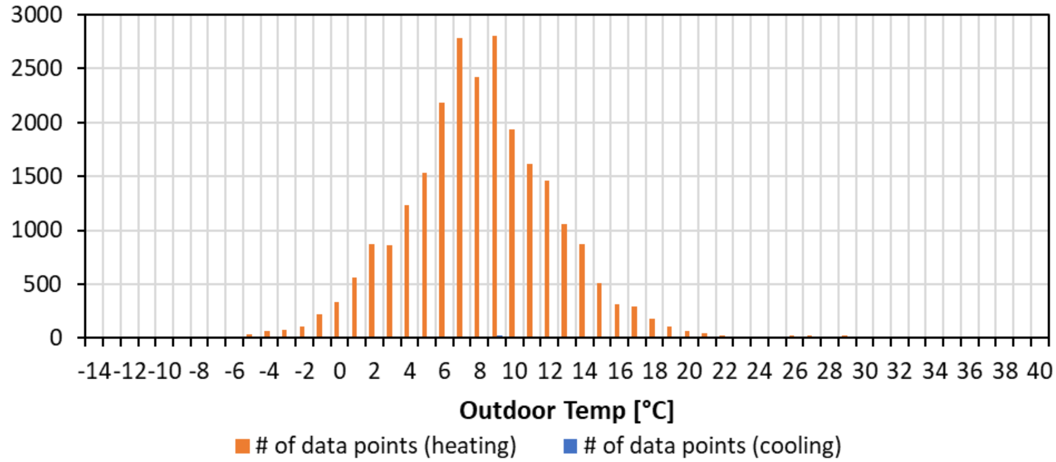



*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*

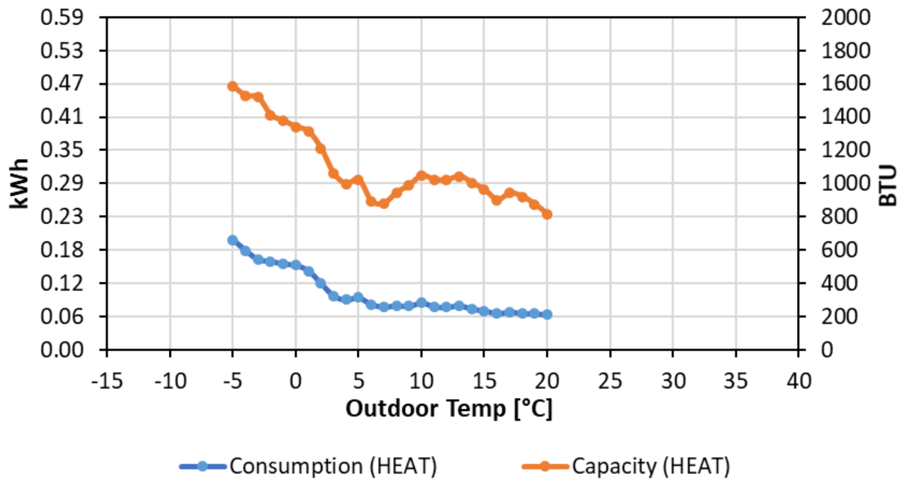
VIC01 – Central V & Ductless – Mitsubishi: MXZ-3C30NA2 | MSZ-GL06NA | SVZ-KP18NA



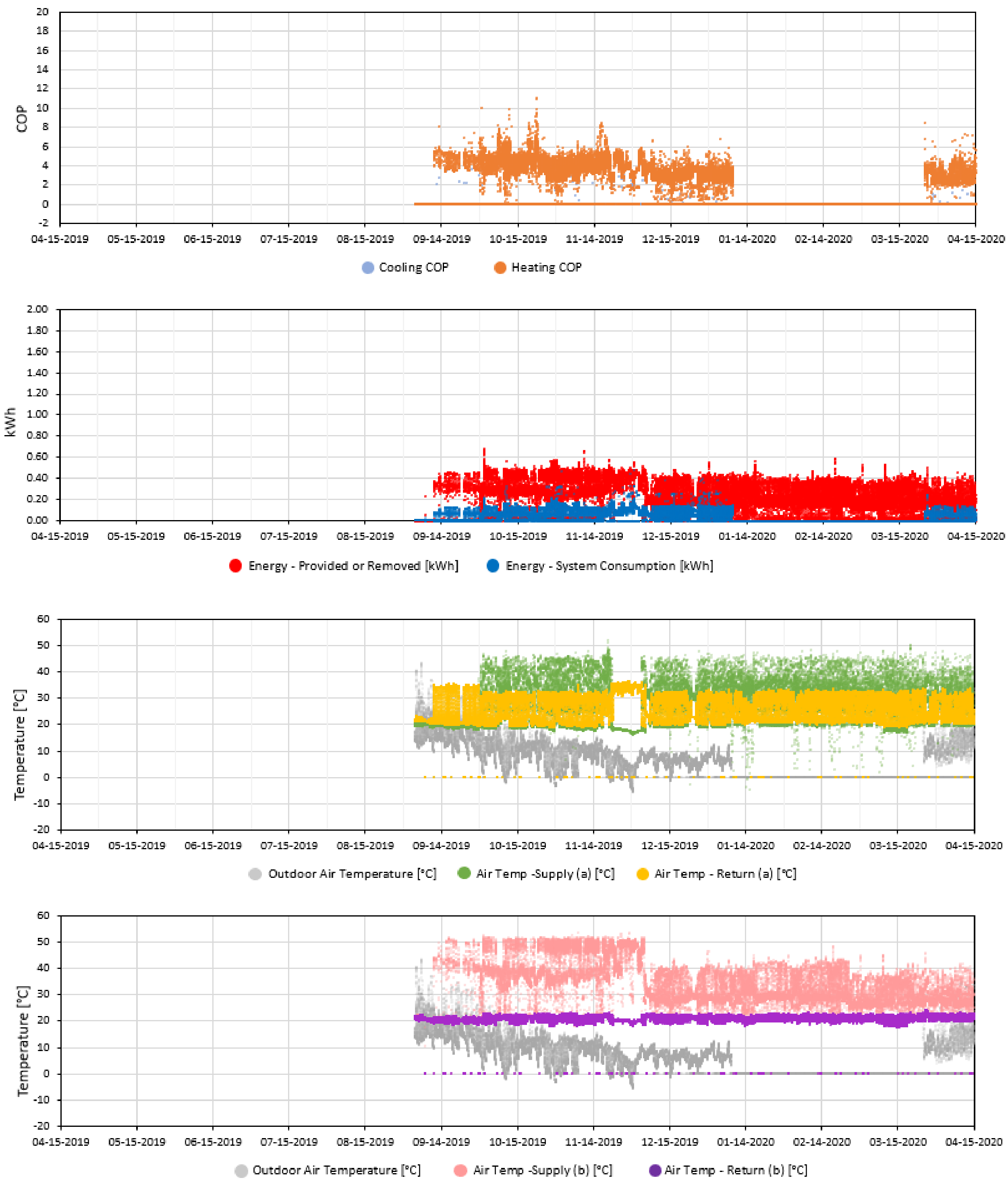

*Average estimated heating and cooling COP for monitored outdoor temperature range.*



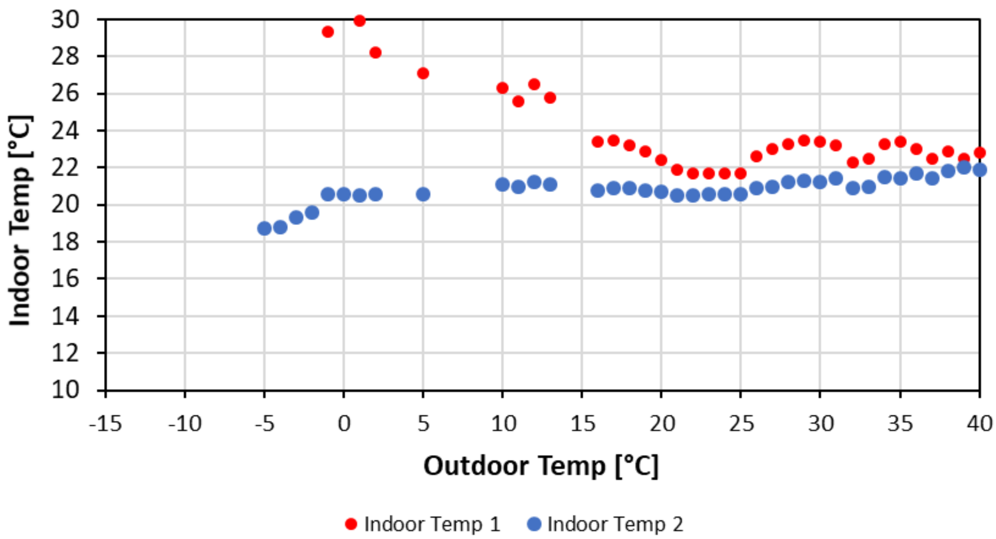

*Total number of heating and cooling data points throughout monitoring period.*



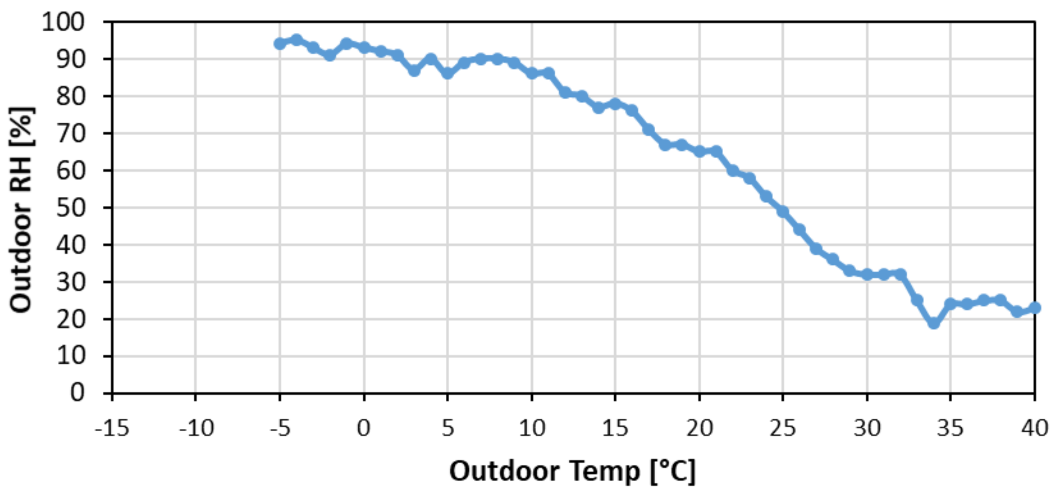

*Average system consumption and capacity for monitored outdoor temperature range.*



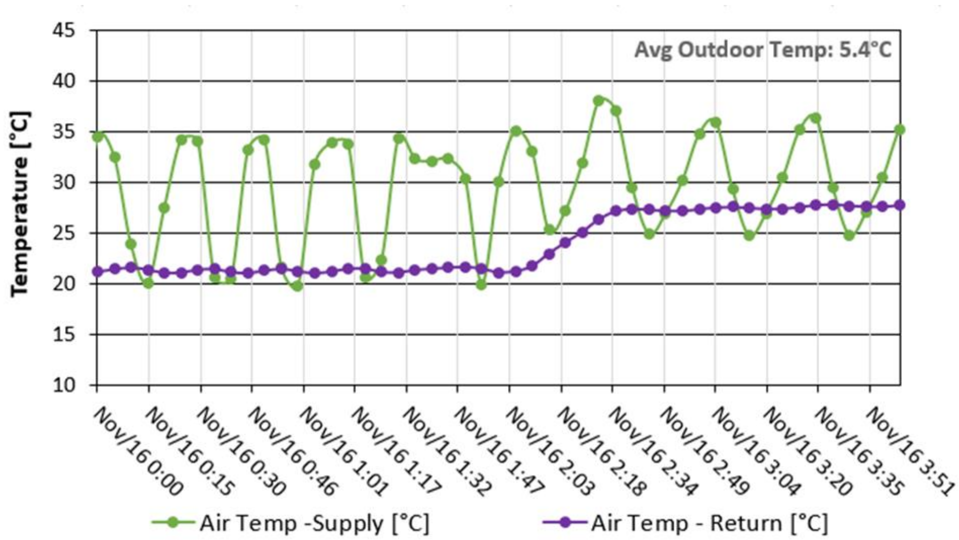

*Measured variables and corresponding COP for monitoring period. Note significant data loss from 04/15/19 to 09/03/19.*

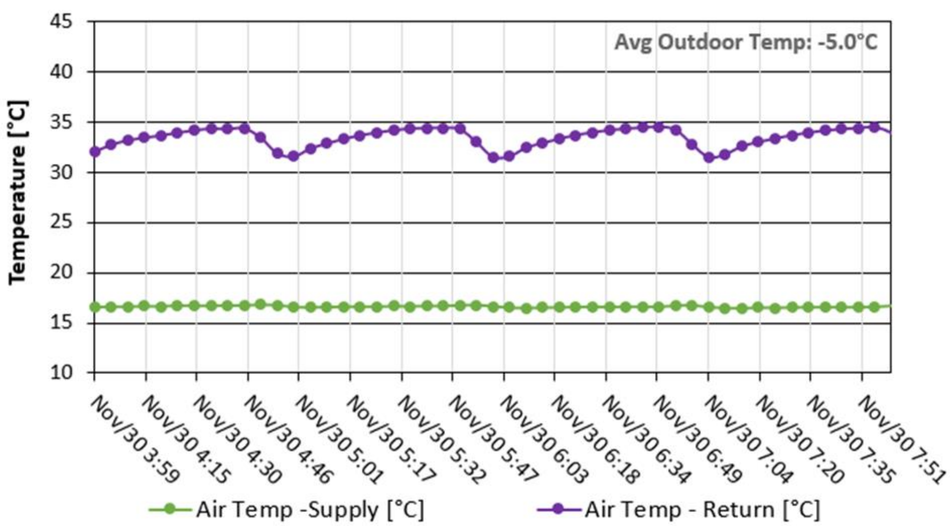



*Average indoor air temperature (return air) for monitored outdoor temperature range.*

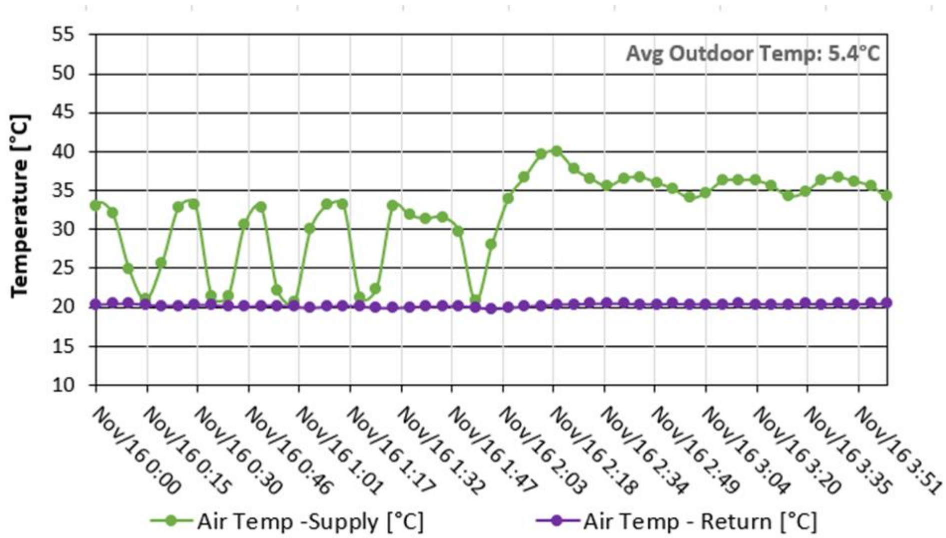


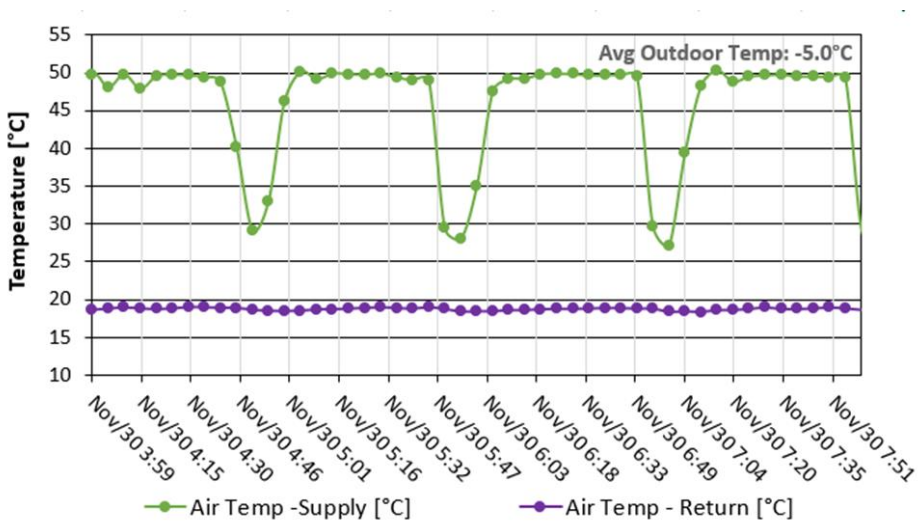

*Average outdoor relative humidity for monitored outdoor temperature range.*



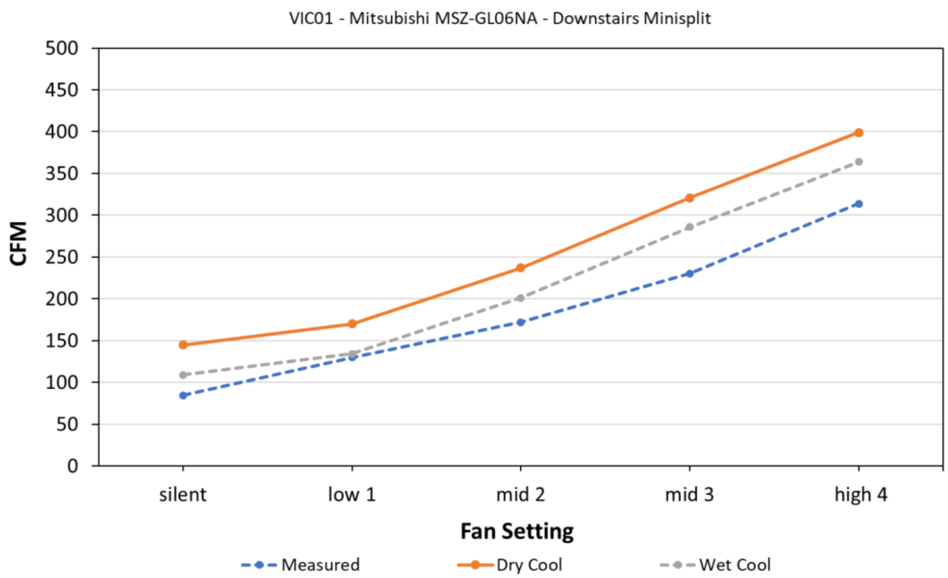



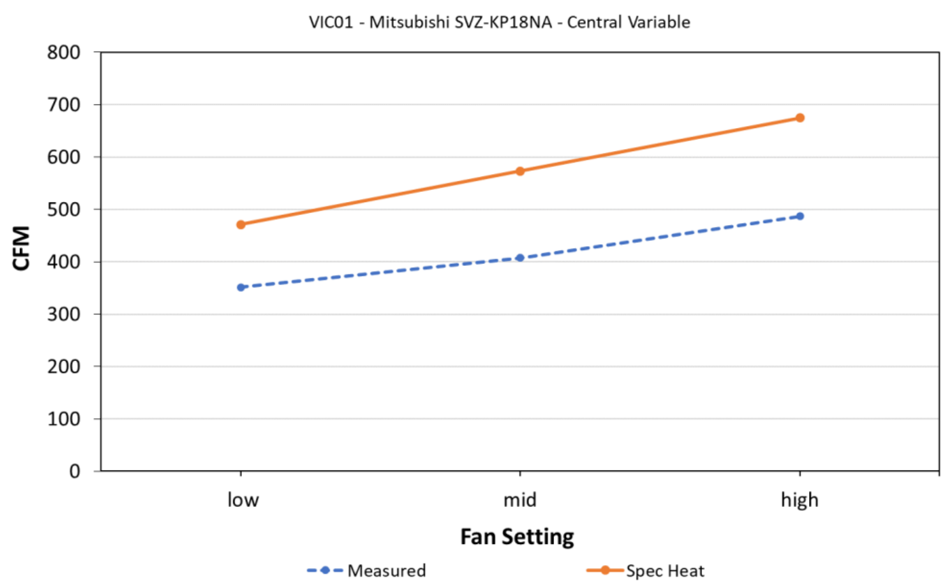

*Unit A: sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*




*Unit B: sample heating cycle plots for typical winter period (top) and extreme winter period (bottom). Every dot represents a 5-minute interval.*




*Measured vs. rated indoor unit volumetric flow rate (cubic feet per minute – CFM)*