Sustainable Technologies
EVALUATION PROGRAM

Supported by Toronto and Region Conservation Authority

Black Creek Pioneer Village Visitor Centre Variable Refrigerant Flow Air-source Heat Pump Retrofit

Final System Performance Report

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THE SUSTAINABLE TECHNOLOGIES EVALUATION PROGRAM

The Sustainable Technologies Evaluation Program (STEP) is a multi-agency initiative developed to support broader implementation of sustainable technologies and practices within a Canadian context. STEP works to achieve this overarching objective by:

- Carrying out research, monitoring and evaluation of clean water and low carbon technologies;
- Assessing technology implementation barriers and opportunities;
- Developing supporting tools, guidelines and policies;
- Delivering education and training programs;
- Advocating for effective sustainable technologies; and
- Collaborating with academic and industry partners through our Living Labs and other initiatives.

Technologies evaluated under STEP are not limited to physical devices or products; they may also include preventative measures, implementation protocols, alternative urban site designs, and other innovative practices that help create more sustainable and livable communities.

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NOTICE

EXECUTIVE SUMMARY

Background

TRCA's Black Creek Pioneer Village (BCPV) Visitor Centre is a 50,000 ft^2 industrial commercial institutional (ICI) building that is located at 1000 Murray Ross Pkwy in North York, ON. The main space heating system consisted of distributed water-to-air heat pumps connected to a central boiler and cooling tower and it was near end-of-life in 2017/18. A variable refrigerant flow (VRF), also known as variable refrigerant volume (VRV), air-source heat pump (ASHP) system was identified as an option with a better business case and lower carbon emissions than a like-for-like replacement of the existing system. TRCA undertook a retrofit of a VRF ASHP between 2018 and 2020. Concurrent with the VRF retrofit, the boiler used for perimeter heating and ventilation air was also upgraded to a higher efficiency model. The system received a third-party commissioning review in Spring/Summer 2020. Measurement & Verification (M & V) was performed to assess the utility savings of the full retrofit and detailed monitoring was performed on part of the ASHP system to characterize as-installed performance.

Measurement and Verification

M & V was adherent with the International Performance Measurement and Verification Protocol (IPMVP) Option C: Whole Facility. Savings was determined using pre- and post-retrofit utility bill data. The M&V estimated that there was no measurable change in the electricity consumption of the Visitor Centre due to the retrofit. It was estimated that **the natural gas consumption was reduced by 29,667 m^3 for one-year of operation, approximately 56% lower than pre-retrofit levels. This equated to 56 tons of carbon and $18,170 in utility savings for a full year** when current utility rates were applied. Savings were substantial but lower than modeled estimates from the feasibility study, which estimated an 83% reduction in gas consumption and a 30% reduction in electricity. The research team did not fully identify the root causes of the discrepancy but speculates it may be related to lower as-installed COPs from oversized equipment in some zones and higher return air temperatures.

Detailed Monitoring

The ASHP system is composed of many smaller subsystems – each consisting of one (or more) outdoor heat pump units, branch boxes, and indoor fan coils. Detailed monitoring was limited to one subsystem due to budgetary constraints. Post-retrofit detailed monitoring took place over nearly two years from 2020 to 2022. Sensors were deployed to monitor all parameters required to calculate the system efficiency, termed the coefficient of performance (COP). In the first year, the research team identified various system issues:

1. Simultaneous heating and cooling was provided by fan coils serving the *same* zone. Effectively, where a zone was served by two fan coils, it was common that one fan coil was heating while the other was cooling. This consumed electricity while having no net benefits to the zones.

2. One of the monitored zones experienced net *cooling* throughout the winter. It is believed to be a result of the zone receiving more tempered ventilation air than was required (possibly, in part, due to a balancing issue as well as the lack of a ventilation schedule).

3. The measured COP was lower than expected, specifically when the heat pump was operating with a low fraction of its rated load. The heat pump frequently operated in this regime because the heating load was small (from (2)).

4. The outdoor enclosure for the heat pumps was equipped with tempering from a natural gas unit heater. The unit heater turned on in relatively mild conditions, increasing the gas consumption.

Issue (3) was believed to be due to a combination of factors: oversized equipment, high return temperatures, and an airflow that was 20% below specifications on one fan coil. Year 2 offered more data for better characterization but the extent to which it was an issue in the full facility was not identified. Issue (4) was also not rectified for Year 2 because it was related to a condensate management problem within the outdoor enclosure that would need addressed first. Prior to Year 2, the following changes were made to the building controls:

- perimeter heating for the building was put on an outdoor reset schedule (it had been always on at full capacity during the heating season of Year 1);
- ventilation was set on a schedule to only operate during normal building occupancy hours; and
- the system was forced to only operate in either heating or cooling mode, and zones with multiple thermostats were reconfigured to be controlled by one.

These interventions, identified through the detailed monitoring, contributed to a drastic decrease in electricity consumption between the first and second year for the Visitor Centre. The impact on gas consumption between Year 1 and Year 2 was not fully known because the gas meter had seized and much of the Year 1 utility bill data was estimated by the utility rather than measured by the utility meter.

Key Lessons Learned for Future Retrofits

System Design

1. **Oversizing should be avoided.** It is common for conventional heating systems to be oversized by up to 40% because the first aim of a designer engineer is to ensure adequate heating and cooling. In conventional systems, there is neither a significant penalty to upfront costs nor to system performance by oversizing. However, this design approach can be problematic for heat pump systems. The cost premiums are greater for the extra capacity, and the impact on performance can be more significant. A key recommendation to future system owners is to invest more time into the heating load calculations for the building to ensure that the system sizing is well-matched to the heat loss, and that any oversizing is greatly diminished compared to standard practice for conventional systems. The monitoring data in this study showed that the COP was significantly reduced when the system was operating a low fraction of its rated heating capacity.

2. **Integration between the ASHP and natural gas systems needs to be carefully considered.** This is a key area that can cause the system to fall short of expectations. The Visitor Centre had three gas heating systems: hydronic perimeter heating, tempering for the ventilation air, and tempering for the outdoor enclosure. Data from Year 1 indicated that all three systems were providing more heating than was required. In the monitored subsystem, the ASHP was removing excess heat

provided by the tempered ventilation air (effectively fighting against it) and this created a notable system inefficiency.

3. **Outdoor enclosures for ASHPs should be avoided where possible.** The outdoor enclosures, while simple in concept, were one of the more challenging aspects of this retrofit. The control of the louvres and natural gas tempering introduced greater control complexity. The research team has since learned that designers are now moving away from outdoor enclosures in these systems.

Ensuring System Success

4. **M & V is crucial.** It is important to understand that despite the issues identified in Year 1, indoor set-points were being met, there were no equipment faults, and no obvious visual indications there were problems. Issues might therefore easily persist unnoticed. A basic check on system performance is an IPMVP-adherent analysis of energy savings based on utility bill or submetering data. If the savings deviates significantly from the expectations of the feasibility assessment, then it is an indicator that there may be issues meriting further investigation.

5. **Cloud-based monitoring and management may be helpful to ensure system success.** Site staff often have many competing responsibilities. System management can be made easier if it is partly done off-site with the help of cloud-based monitoring which is an add-on offered by some system manufacturers. This is especially relevant where a portfolio of buildings is being managed. There are powerful analytical tools available that can help to quickly identify and diagnose system problems that might otherwise be difficult to detect with visual inspections.

6. **Data-driven third-party system commissioning is important.** Controls in a retrofit like this are complex. Errors are likely to occur, and so are unexpected interactive effects with other building systems. As the complexity of the system increases, the commissioning should be increasingly data-driven and include detailed analysis of BAS trend data with a lens for full system optimization. This makes commissioning a retrofit with this level of complexity more challenging then with conventional systems. The development of commissioning guidelines for VRF ASHP retrofits would be helpful to both building owners and commissioning agents.

7. **A designated party should be assigned to ensure continued effective overall integration of the ASHP system with other existing building systems.** This may be internal staff or an external contractor. There should be an individual on the retrofit team tasked with ensuring effective integration for the *full* building. This a consequence of (2), where unexpected interactive effects between the ASHP and existing building systems are likely.

Conclusion and Future Work

This study demonstrated a VRF ASHP that reduced operating costs and significantly reduced natural gas consumption in a large institutional building. System implementation challenges have been documented and can be addressed in future retrofits based on key lessons learned from this study. The difference between the savings predicted in the feasibility study and those achieved in practice indicates that more work is required comparing modeled and actual performance. This will help to refine and build confidence in system modeling. Technical guidance from equipment manufacturers regarding best practices for system modeling and commissioning would be helpful to building owners as they procure feasibility studies and/or implement systems.

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1.0 INTRODUCTION

The Toronto and Region Conservation Authority's (TRCA) Black Creek Pioneer Village (BCPV) Visitor Centre is a 50,000 ft^2 industrial commercial institutional (ICI) building located at 1000 Murray Ross Parkway in North York. The building is used as the main hub of the site providing guest services, office space for staff, educational spaces, a commercial kitchen, event space, a gift shop, a gallery, washrooms, and a cafe. An external consultant identified the building as having significant energy conservation potential within the TRCA portfolio.

The building was previously served by a distributed water source heat pump system combined with a radiant perimeter heating system. The distributed water source heat pumps rejected heat to the primary hydronic circuit or extracted heat from the loop as needed to meet the temperature set-points of the zones they served. The temperature in the primary hydronic circuit was regulated by gas boilers and a cooling tower. This system allowed for heat recovery between the zones throughout the building.

As of 2017/2018, the existing system was due for replacement. TRCA hired an external consultant to evaluate different options. Based on building energy modeling, the consultant identified that a variable-refrigerant flow (VRF), also called variable refrigerant volume (VRV), air-source heat pump (ASHP) system would provide deeper carbon emissions savings and a better business case than like-for-like replacement of the existing system. Natural gas consumption with the VRF retrofit option was estimated to be approximately 70% lower than the like-for-like replacement and 83% lower than the current system (Figure 1-1). Electricity reductions were also expected – on the scale of a 30% reduction from the existing system.

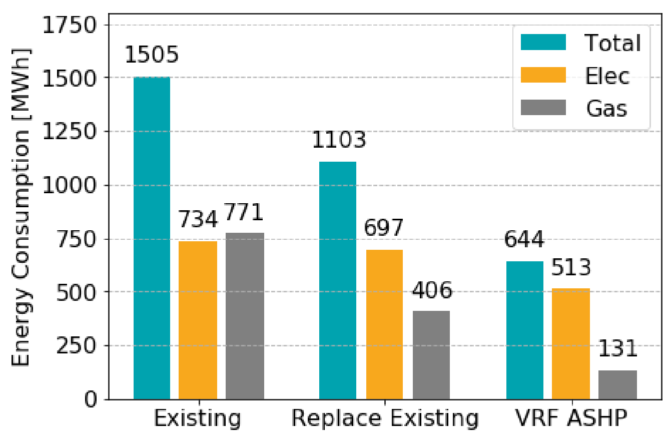


Figure 1-1. An external consultant analyzed the annual energy consumption of a like-for-like replacement of the existing system and that for a VRF ASHP retrofit. The VRF ASHP retrofit was estimated to consume much less gas and electricity than a like-for-like replacement.

Based on this information, TRCA went forward with a competitive bidding process for an ASHP retrofit of the BCPV Visitor Centre. The retrofit took place from 2018 to 2020, and third-party commissioning review was performed in Spring/Summer of 2020.

The Atmospheric Fund (TAF) provided funding to the Sustainable Technologies Evaluation Program (STEP) of the TRCA for measurement and verification (M&V) of the retrofit, detailed system monitoring, and analysis of barriers to more widespread deployment of the technology, among other deliverables. The broad aim of TAF's funding contribution was to help STEP provide data, analysis, and other activities that would support broader market transformation. This document outlines the analysis and results of the BCPV Visitor Centre ASHP retrofit as well as the issues that were confronted and the key lessons learned.

2.0 EQUIPMENT

The ASHP system was manufactured by Daikin and it was composed of many smaller subsystems. Each subsystem consisted of an outdoor unit,[1] indoor fan coils, branch box(es), and other components. As listed in Table 2-1, the retrofit comprised of 8 subsystems in total.

Table 2-1. Components of BCPV Visitor Centre VRF ASHP retrofit.

Unit	Model	Rated Heating Capacity [kBTU/hr]	Number of Indoor Fan Coils
OU-1	REYQ216TTJU	243	3
OU-2	REYQ192TTJU	229	5
OU-3	REYQ192TTJU	223	4
OU-4	REYQ120TTJU	146	2
OU-5	REYQ168TTJU	190	11
OU-6	REYQ216TTJU	243	3
OU-7	REYQ240TTJU	270	3
OU-8	REYQ144TTJU	162	6
	Totals:	**1,706**	**37**

Collectively, the subsystems connect to 37 indoor fan coils installed throughout the building. The outdoor units were installed in ventilated enclosures (Figure 2-1) heated by a natural gas unit heater (thermal efficiency of 82-83%) in cold conditions. This is one approach to provide back-up heating for the ASHPs. It was used as a cost-saving measure to avoid the additional cost of more advanced heat pumps that can continue to operate in extreme cold without the need of an enclosure.

Indoor fan coils can be locally controlled through Navigation Remote Controllers (NAVs)[2], as shown in Figure 2-2. NAVs connect to an Intelligent Touch Manager[3] (iTM) which is itself a “mini” building management system capable of central control for the ASHP system, as well as providing a user interface and other features. The iTM has the added option of being a BACNet gateway that interfaces the ASHP system with the building automation system (BAS) from Johnson Controls.

[1] Note that the outdoor “unit” may be composed of more than one heat pump.
[2] BRC1E73
[3] DCM601A71.

Figure 2-1. (Top left) Enclosure that houses outdoor units. (Top right) Inside view of enclosure showing heat pumps. (Bottom) Inside view showing dampers and gas unit heater.

Figure 2-2. (Left) Navigation Remote Controllers provide local controller for the indoor fan coils. (Right) The intelligent Touch Manager provides central control and a user interface for the system, as well as being the BACNet interface for the Johnson Controls building automation system.

Concurrently with the ASHP retrofit, the boiler used for perimeter heating and ventilation air was upgraded from an 80% efficient boiler to a 97% condensing boiler from Viessman (VITOCROSSAL 200) with a capacity of 851 kBTU/hr. The perimeter heating system control was also upgraded from manual on/off control pre-retrofit to an outdoor reset schedule post-retrofit. Note that some of the gas savings from the retrofit is attributable to these improvements.

3.0 RETROFIT TIMELINE

A timeline of key retrofit milestones is provided below.

- **November 2015**. "Black Creek Pioneer Village Energy Conservation Feasibility Study" released to TRCA identifying the BCPV Visitor Centre as having greatest conservation potential in the TRCA portfolio.
- **October 2016**. TRCA releases public RFP for "Services", to be provided in connection with "Development of a Concept Plan for the Renovation of the Existing Building Envelope, HVAC, Lighting and Automated Control Systems at Black Creek Pioneer Village."
- **Dec 2016**. RFP contract is awarded.
- **May 2017**. The Mechanical and Electrical Retrofit Business Case is presented to TRCA. This report employs energy modeling to present several mechanical retrofit use cases for the BCPV Visitor Centre.
- **June 2017**. Change order #001 - CONTRACT NO.: 10002310 was submitted for "Design, tender, permitting and construction management of a new heating, ventilation and air conditioning system (HVAC) at the Black Creek Pioneer Village Visitor Centre"
- **October 2017**. RFP documents for Electrical and Mechanical are prepared on TRCA's behalf.
- **October 2017.** RFP # 10006369 – submitted to public.
- **November 2017**. Five vendors submitted proposals.
- **November 2017.** Bid opening and review conducted internally by TRCA.
- **December 2017.** Contract awarded.
- **January 2018.** Kick-off meeting and announcement of subcontractors.
- **June 2018.** Testing and balancing report for ventilation system submitted.
- **November 2018**. Refrigerant connections complete. VRF Units start-up.
- **January 2020**. Substantial completion of outdoor enclosures.
- **March 2020**. Third-party system review (commissioning) provided.
- **July & August 2020.** Commissioning deficiencies were addressed.
- **October 2020.** Update close-out documents delivered.
- **January 2021.** Outdoor temperature reset control strategy was implemented on the boiler.
- **August 2021.** Multiple contractors attend on-site meeting to address thermostat issues. A decision is made to switch some thermostats to a manual mode (heating or cooling) to address the issues.
- **November 2021.** External HVAC contractor brought in to identify ongoing thermostat, and comfort issues.

- **May 2022.** External HVAC contractor follow-up. Performs analysis and testing of refrigerant systems. Minor issues discovered, further thermostat controls are changed to avoid simultaneous heating and cooling.

4.0 DETAILED MONITORING: SUBSYSTEM OVERVIEW

Detailed monitoring was performed on a single subsystem within this study since the distributed nature of the system meant that building-wide detailed monitoring would have been prohibitively challenging and costly. The aim of detailed monitoring was to help inform the M&V component of this project, but also to document as-installed system performance with special emphasis on the heat-recovery operation of the heat pump system. The monitored subsystem consisted of an REYQ216TTJU outdoor unit and three indoor fan coils – FXMQ72MVJU, FXMQ96MVJU, and FXMQ48PBVJU. Relevant data from the manufacturer is shown in Table 4-1 and Table 4-2.

Table 4-1. Selected manufacturer specifications for the REYQ216TTJU heat pump.

Parameter	Unit
Name	18-Ton VRV-IV Heat Recovery Unit – 230 V
Rated Heating Conditions	Indoor: 70 °F (21.1 °C) DB Ambient: 47 °F (8.3 °C) DB / 43 °F (6.1 °C) WB
Rated Heating Capacity	226 kBTU/hr (66.3 kW)
Rated Heating COP (Ducted)	3.7
Heating COP at 17 °F (-8.3 °C) (Ducted)	2.3
Capacity Control Range	5 to 100 %
Heating Input Power	22.20 kW
Rated Cooling Conditions	Indoor: 80 °F (21.1 °C) DB / 67 °F (19.4 °C) WB Ambient: 95 °F (35 °C) DB
Rated Cooling Capacity	200 kBTU/hr (58.7 kW)
Cooling Input Power	16.10 kW
EER (Ducted)	12.40

Relevant equations for capacity and efficiency calculations are provided in Appendix E. Assuming dry air near room temperature (return temperature of 21.1 °C, air density of 1.275 kg/m^3, and specific heat capacity of 1.006 kJ/(kg °C)), it is possible to calculate expected values for the maximum supply temperature and temperature rise across the fan coils. This is shown in Table 4-3.

Table 4-2. Selected manufacturer specifications for the fan coils in the monitored subsystem.

	Fan Coil 1.01	Fan Coil 1.02	Fan Coil 1.03
Model	FXMQ72MVJU	FXMQ96MVJU	FXMQ48PBVJU
Rated Cooling Capacity	72.0 kBTU/hr (21.1 kW)	96.0 kBTU/hr (28.2 kW)	48.0 kBTU/hr (14.1 kW)
Rated Sensible Cooling Capacity	52.7 kBTU/hr (15.5 kW)	68.4 kBTU/hr (20.1 kW)	35.0 kBTU/hr (10.3 kW)
Rated Heating Capacity	81.0 kBTU/hr (23.8 kW)	108 kBTU/hr (31.7 kW)	82.6 kBTU/hr (24.2 kW)
Fan speeds*	2	2	3
Low Speed Airflow	1,764 cfm	2,118 cfm	988 cfm
Medium Speed Airflow	-	-	1,165 cfm
High Speed Airflow	2,047 cfm	2,541 cfm	1,377 cfm

*In practice, there was an additional speed for each fan coil.

Table 4-3. Expected supply temperature and temperature rise for each indoor fan coil in the subsystem operating at rated conditions with maximum capacity.

Fan Coil	Maximum Expected Supply Temperature at Rated Conditions [°C]	Maximum Expected Temperature Rise at Rated Conditions [°C]
1.01	40.3	19.2
1.02	41.7	20.6
1.03	40.1	19.0

Fan coil 1.01 and 1.02 served the Student Assembly and Fan Coil 3 served the Offices. The Student Assembly is an area where visiting groups can gather for lunch, orientation, or other purposes. The Offices are used as office space for BCPV staff. The areas served by different subsystems are shown schematically in Figure 4-1. Note this is a simplified figure which only shows a single indoor fan coil for each subsystem, where there are typically multiple fan coils. The actual placement of the indoor fan coils is shown in Figure 4-2, which provides a more detailed mechanical drawing. Refrigerant connections for the monitored subsystem are shown in Figure 4-3. Interior images are shown in Figure 4-4 and 4-5.

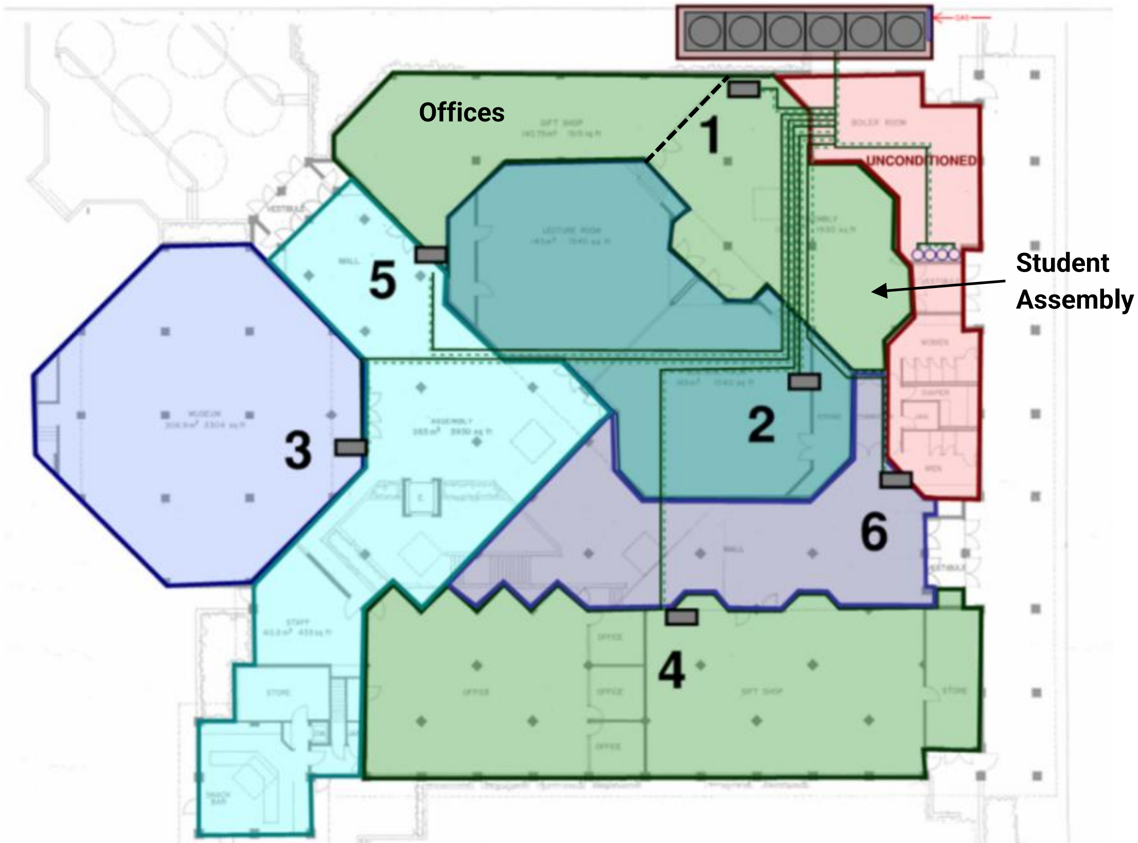


Figure 4-1. The subsystem (Zone 1) that received detailed monitoring provided heating and cooling to the Offices and the Student Assembly.

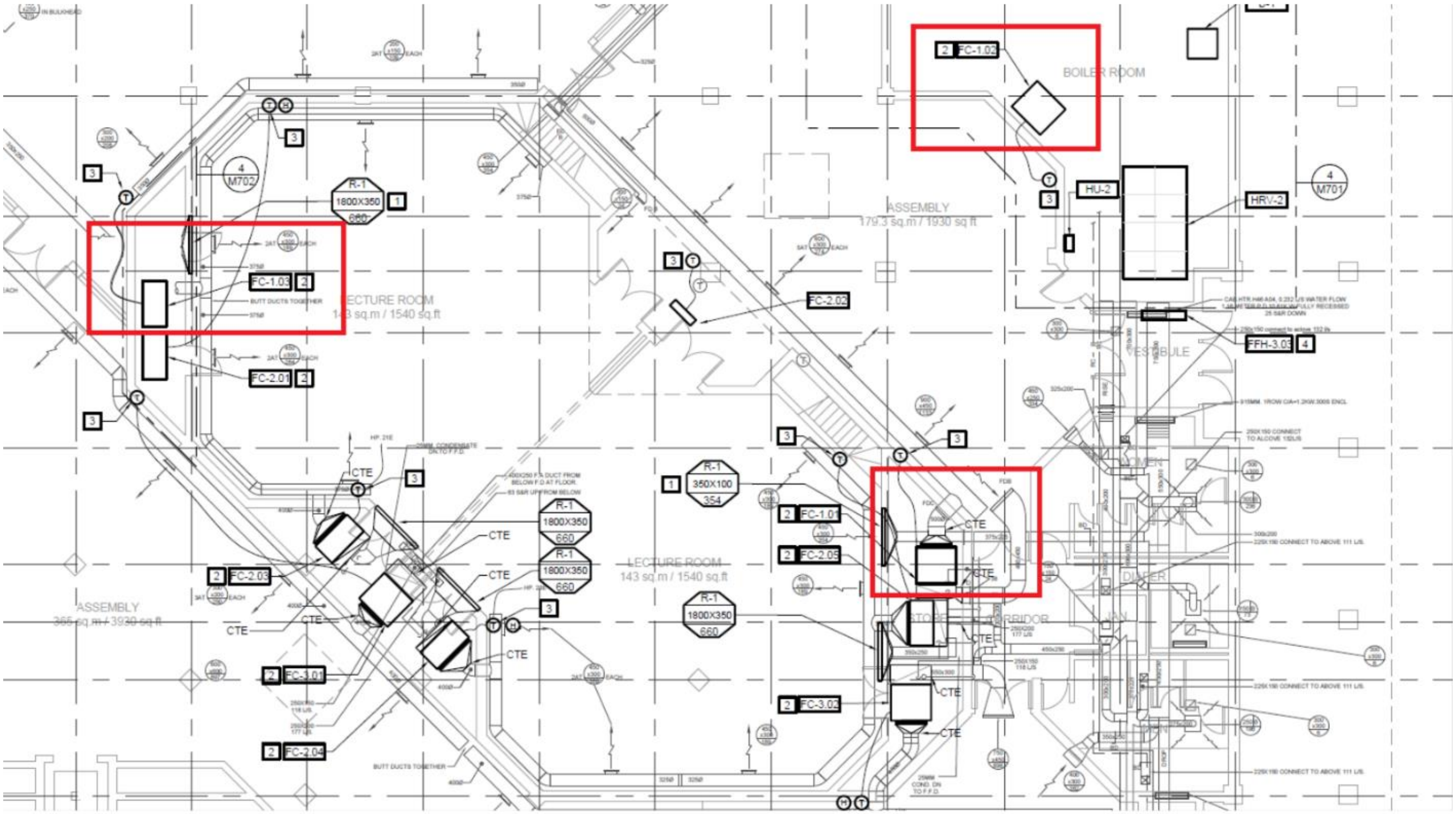


Figure 4-2. Fan Coils 1.01 and 1.02 serve the Student Assembly. Fan Coil 1.03 serves the Offices.

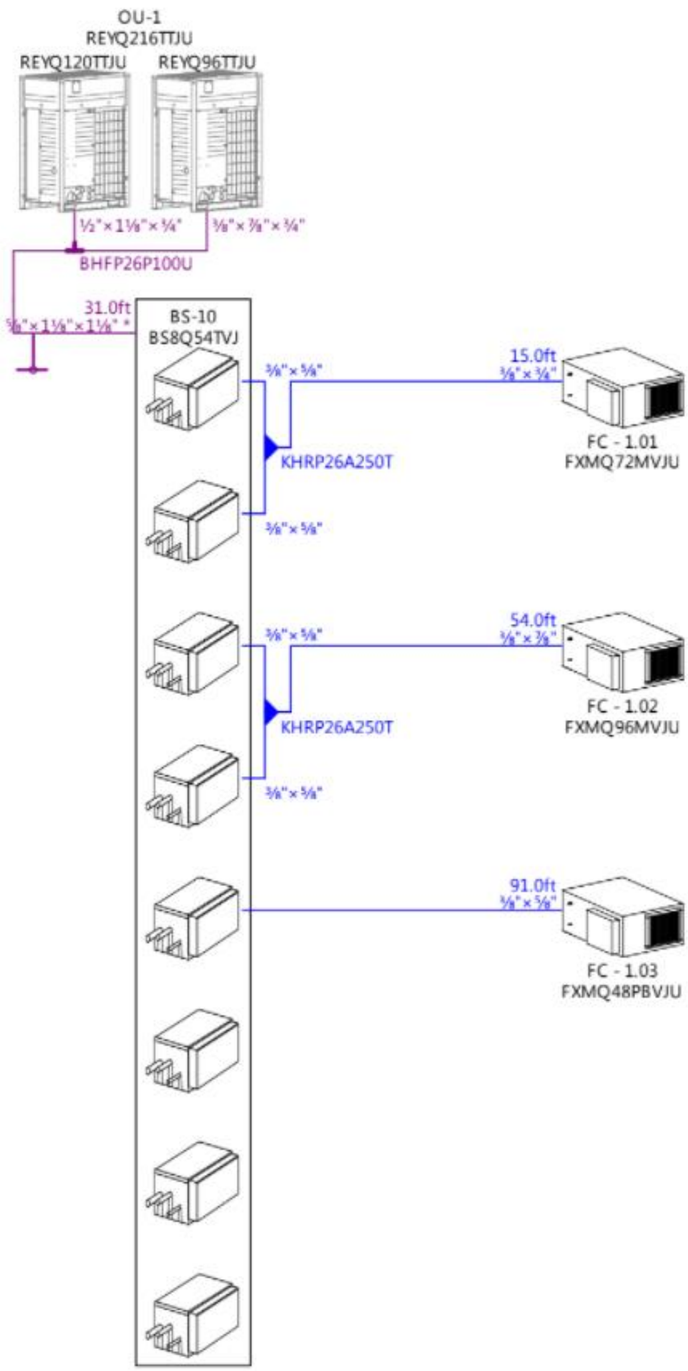


Figure 4-3. Refrigerant connections for the monitored subsystem (Zone 1). OU-1, represents the outdoor units serving this Zone. BS-10, represents the branch box capable of heat recovery. FC-1.01-1.03, represents the indoor fan coils.

Figure 4-4. View of Student Assembly from entry to Lecture Room and looking toward entry vestibule.

Figure 4-5. View of Student Assembly from entry to Lecture Room and looking toward the Offices.

5.0 DETAILED MONITORING: HARDWARE

The detailed subsystem monitoring considered the following monitoring points:

- electrical power consumption of the outdoor unit;
- supply and return air temperature for each indoor fan coil;
- supply and return relative humidity for each indoor fan coil;
- airflow for each indoor fan coil; and
- entering and leaving air temperatures for the outdoor unit.

In addition to these measurements, whole-building electricity and natural gas consumption was also submetered. The monitoring hardware is primarily from Monnit. Wireless sensors were connected to an online database via the internet connection of the building. The data was uploaded to a server provided by Monnit and then remotely downloaded for analysis using the server's API. Table 5-1 provides more detail on the instrumentation package that was used. Appendices B to D discuss the instrumentation in greater detail. Note that outdoor air temperature was obtained from Environment Canada data.

Table 5-1. Detailed monitoring instrumentation overview.

	Monitoring Point	Sensor/Transmitter	Expected Accuracy	Logging Interval	Logging Start
1	Electrical power of outdoor unit	Acuvim II from Accuenergy	0.2%	5 min	01/2021
2	Fan Coil 1.01 Return Air Temperature	Monnit ALTA Commercial AA Wireless Thermocouple Reader	± 2.2 ^{o}C	1 min after Jan 31st 2021; 10 min before Jan 31st 2021	01/2020
3	Fan Coil 1.01 Supply Air Temperature	Monnit ALTA Commercial AA Wireless Thermocouple Reader	± 2.2 ^{o}C	1 min after Jan 31st 2021; 10 min before Jan 31st 2021	01/2020
4	Fan Coil 1.02 Return Air Temperature	Monnit ALTA Commercial AA Wireless Thermocouple Reader	± 2.2 ^{o}C	1 min after Jan 31st 2021; 10 min before Jan 31st 2021	01/2020
5	Fan Coil 1.02 Supply Air Temperature	Monnit ALTA Commercial AA Wireless Thermocouple Reader	± 2.2 ^{o}C	1 min after Jan 31st 2021; 10 min before Jan 31st 2021	01/2020
6	Fan Coil 1.03 Return Air Temperature	Monnit ALTA Commercial AA Wireless Thermocouple Reader	± 2.2 ^{o}C	1 min after Jan 31st 2021; 10 min before Jan 31st 2021	01/2020
7	Fan Coil 1.03 Supply Air Temperature	Monnit ALTA Commercial AA Wireless Thermocouple Reader	± 2.2 ^{o}C	1 min after Jan 31st 2021; 10 min before Jan 31st 2021	01/2020
8	Fan Coil 1.01 Return Relative Humidity	Monnit ALTA Commercial AA Wireless Humidity Sensor	± 2.2%	1 min after Jan 31st 2021; 10 min before Jan 31st 2021	01/2020

9	Fan Coil 1.01 Supply Relative Humidity	Monnit ALTA Commercial AA Wireless Humidity Sensor	± 2.2 %	1 min after Jan 31^{st} 2021; 10 min before Jan 31^{st} 2021	01/2020
10	Fan Coil 1.02 Return Relative Humidity	Monnit ALTA Commercial AA Wireless Humidity Sensor	± 2.2 %	1 min after Jan 31^{st} 2021; 10 min before Jan 31^{st} 2021	01/2020
11	Fan Coil 1.02 Supply Relative Humidity	Monnit ALTA Commercial AA Wireless Humidity Sensor	± 2.2 %	1 min after Jan 31^{st} 2021; 10 min before Jan 31^{st} 2021	01/2020
12	Fan Coil 1.03 Return Relative Humidity	Monnit ALTA Commercial AA Wireless Humidity Sensor	± 2.2 %	1 min after Jan 31^{st} 2021; 10 min before Jan 31^{st} 2021	01/2020
13	Fan Coil 1.03 Supply Relative Humidity	Monnit ALTA Commercial AA Wireless Humidity Sensor	± 2.2 %	1 min after Jan 31^{st} 2021; 10 min before Jan 31^{st} 2021	01/2020
14	Fan Coil 1.01 Airflow	Dwyer Series AVU Air Velocity Transmitter connected to Monnit Alta Voltage Meter 0 -10 VDC	± 5% FS	1 min after Jan 31^{st} 2021; 10 min before Jan 31^{st} 2021	01/2020
15	Fan Coil 1.02 Airflow	Dwyer Series AVU Air Velocity Transmitter connected to Monnit Alta Voltage Meter 0 -10 VDC	± 5% FS	1 min after Jan 31^{st} 2021; 10 min before Jan 31^{st} 2021	01/2020
16	Fan Coil 1.02 Airflow	Dwyer Series AVU Air Velocity Transmitter connected to Monnit Alta Voltage Meter 0 -10 VDC	± 5% FS	1 min after Jan 31^{st} 2021; 10 min before Jan 31^{st} 2021	01/2020

6.0 DETAILED MONITORING: HEATING RESULTS YEAR 1

The monitoring data from November 15th, 2020, to April 15th, 2021, were analyzed to better understand heating season performance. Figure 6-1 shows time-series supply and return temperature data for each fan coil. Frequency histograms are shown in Figure 6-2.

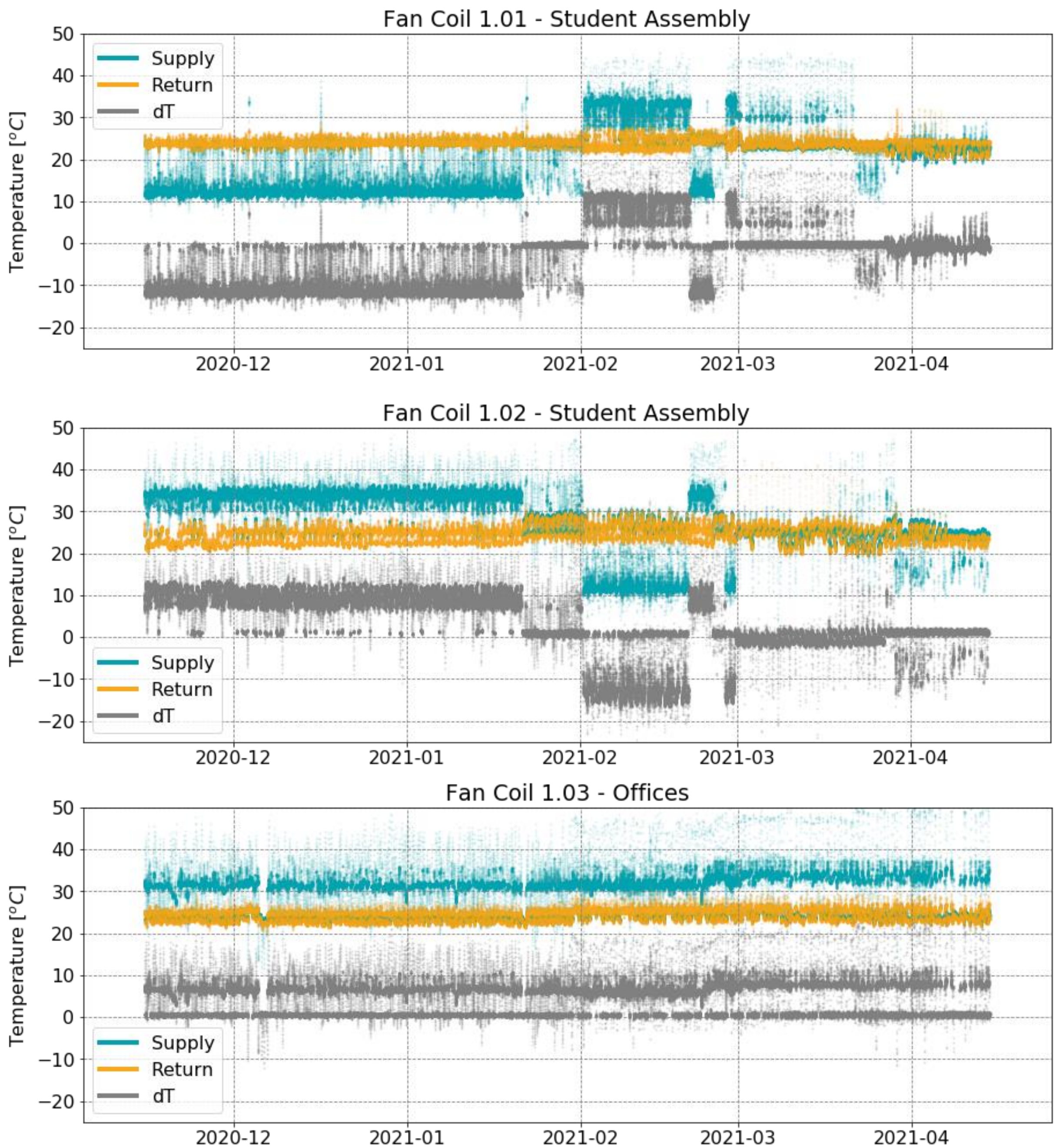


Figure 6-1. Supply and return temperatures (and their difference) are shown for each fan coil during Winter 2020/2021. It is evident that Fan Coil 1.01 provided heating to the Student Assembly while Fan Coil 1.02 simultaneously provided cooling (and vice versa)

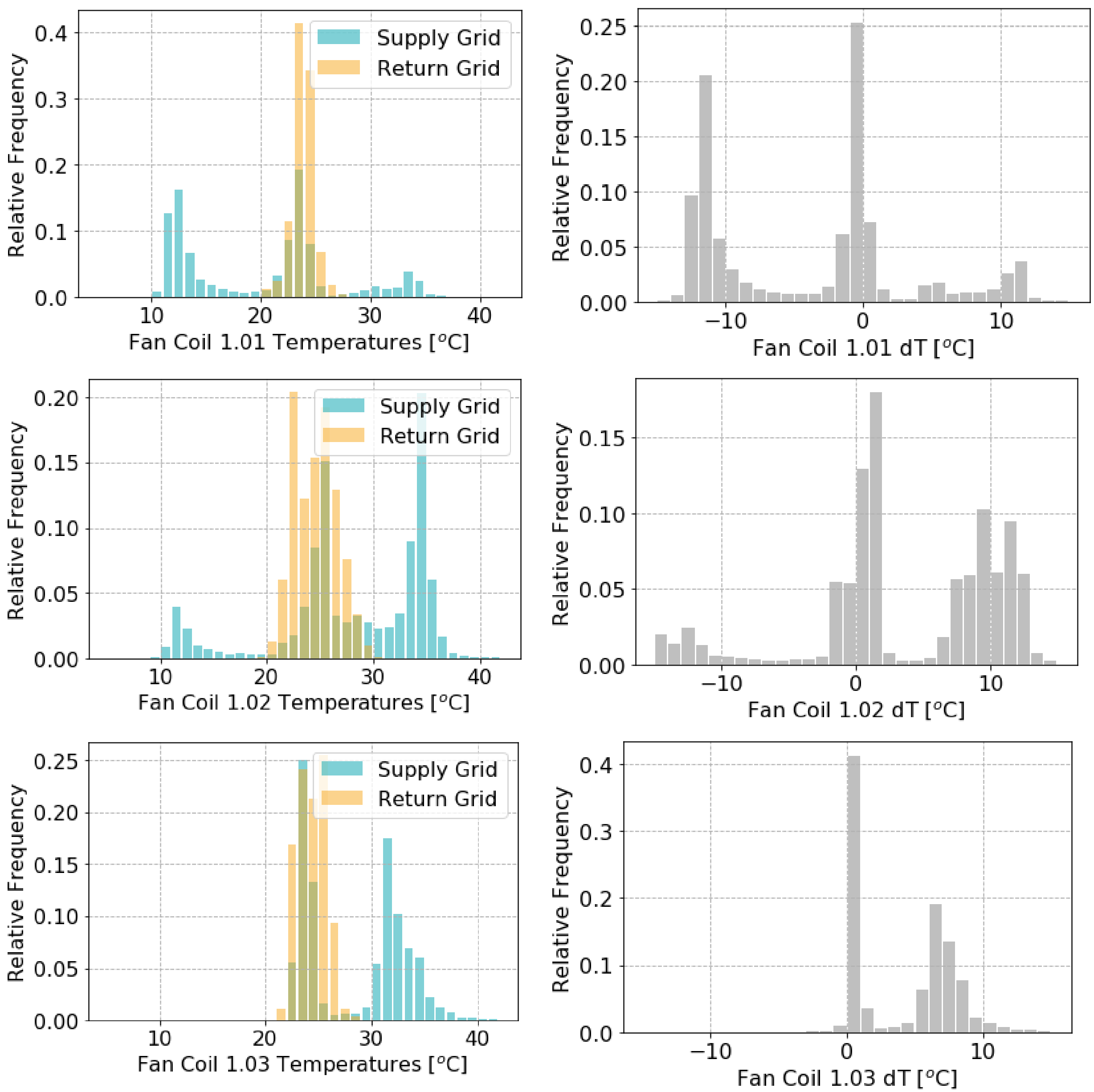


Figure 6-2. Relative frequency histograms are plotted for fan coil supply temperature, return temperature, and temperature difference (dT). Fan Coil 1.01 was primarily cooling the Student Assembly while Fan Coil 1.02 was primarily heating it. Fan coil 1.03 was only operating in heating mode.

Heat recovery operation can be beneficial for system performance when it is happening between *different zones*. Even in the wintertime, internal gains may be sufficiently high that cooling is required in one zone while heating is still required in other nearby zones. It would be the most efficient to transfer heat between the zones. However, in the monitored subsystem heat recovery was happening *within the same zone* – the Student Assembly – and this was not beneficial operation.

Recall from Table 4-1 that the rated values for heating capacity and efficiency assume a return temperature of 21.1 °C. Figure 6-2 shows that Fan Coil 1.01 and 1.03 return temperatures are typically measured between 23 and 25 °C. For Fan Coil 1.02, return temperatures are greater than 25 °C approximately half of the time, and even approach 30 °C in some cases. Zone temperatures measured at chest (thermostat) level are shown in Figure 6-3.

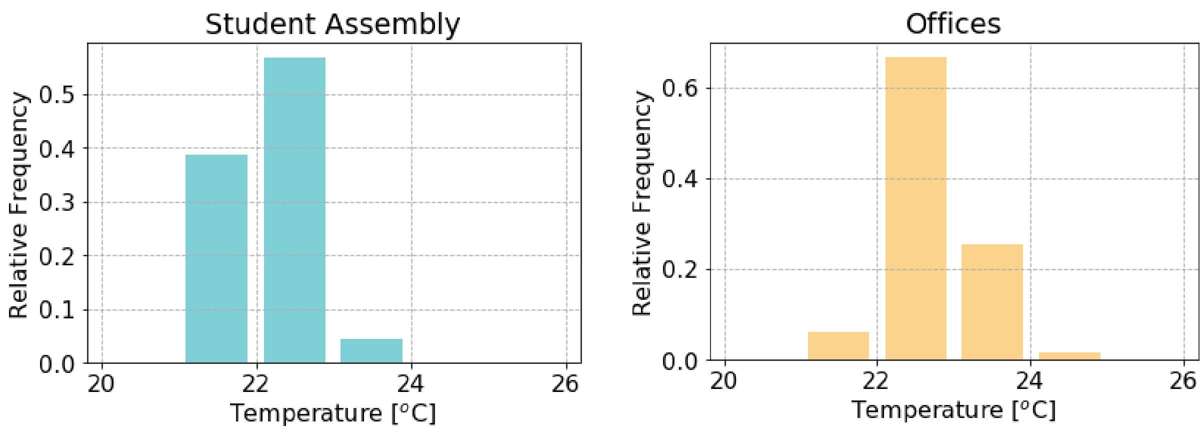


Figure 6-3. Zone temperatures during Winter 2020/2021 measured by monitoring system with sensors located at chest level in the zones near NAVs.

High return temperatures will negatively impact the efficiency of the heat pump system. The high return temperatures across all units are likely influenced by at least three factors. Firstly, Figure 6-3 shows that thermostats are set to higher than 21.1 °C. Next, there will be thermal stratification in the zones with warmer temperatures near the ceiling where the return air grills are located. Finally, for Fan Coil 1.02 specifically, there is a steam humidification system upstream of the fan coil and it is also located the closest to the mechanical room where the ERV is providing tempered ventilation air. There may be an air balancing issue where Fan Coil 1.02 is receiving more tempered air than needed.

Recall from Table 4-3, at maximum capacity, supply temperatures near 40 °C were expected as well as temperature rises near 20 °C across the fan coils. Supply temperatures only rarely surpass 35 °C. Temperature rises, when present, were typically in the vicinity of 10 °C or lower. The temperature rise across Fan Coil 1.03 was most typically between 6 and 8 °C.

Outdoor temperature data for this period is shown in Figure 6-4, both as a time-series and as a frequency histogram. Also shown is Toronto data from the Canadian Weather Year for Energy Calculations (CWEC) database which represents a typical meteorological year (TMY). Winter 2020/2021 had more hours below −5 °C and above 10 °C while the TMY data is more concentrated around 0 °C, but overall, Winter 2020/2021 was not a significant deviation from typical weather. Note that February 2021 had particularly cold weather and a greater heating capacity from the heat pump system might have been expected.

Airflows throughout the heating season are shown in Figure 6-5 and Figure 6-6. The airflow and temperature data were used to calculate the heating (and cooling) capacity according to the process described in Appendix E. Figure 6-7 plots the heating capacity of each fan coil as a time-series and as frequency histograms. It also plots the overall net heating capacity, and to the net heating capacity Student Assembly which is heated by both Fan Coil 1.01 and 1.02.

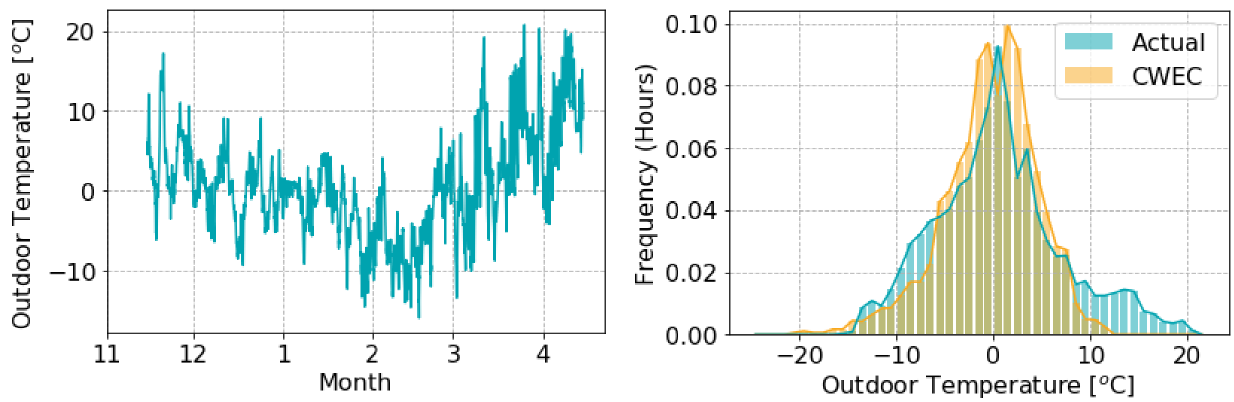


Figure 6-4. Outdoor temperature data during Winter 2020/2021 is shown both as a time-series and as a frequency histogram. Note that February had particularly cold weather.

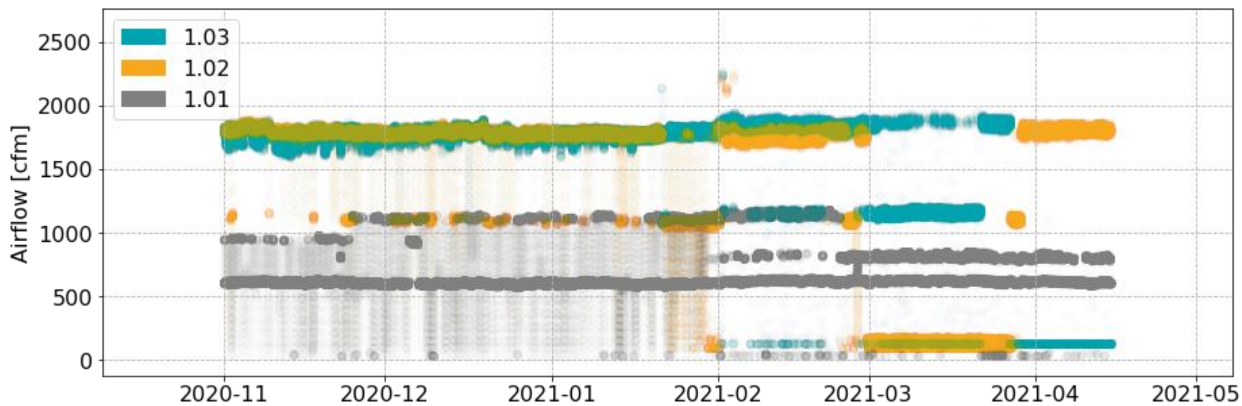


Figure 6-5. Time-series airflow data is plotted for each fan coil.

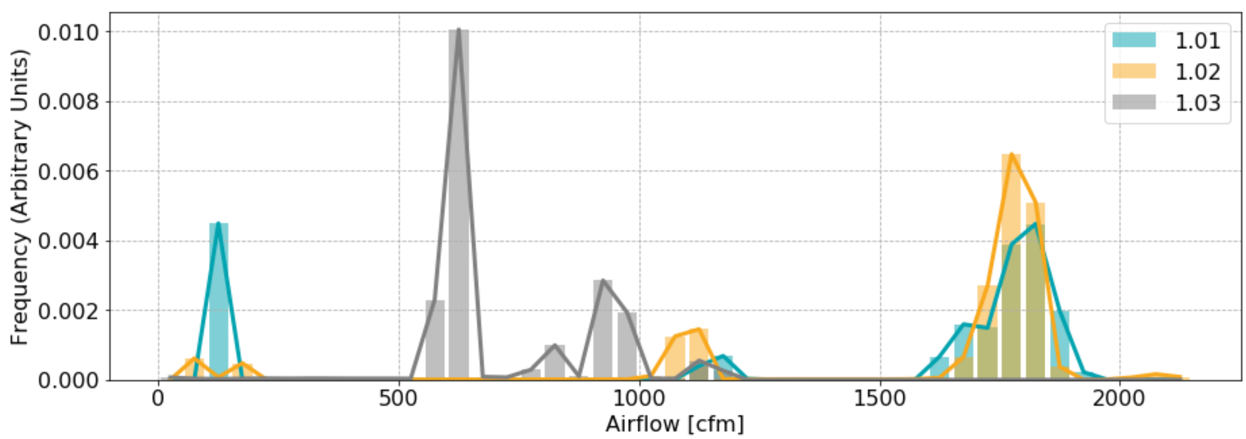


Figure 6-6. A frequency histogram shows the relative fraction of time that each indoor fan coil is providing different airflows during Winter 2020/2021.

From Table 4-1, the rated heating capacity (for an outdoor dry bulb of 8.3 °C) is 66.3 kW. The table also shows that at -8.3 °C the COP decreases by approximately a third, and so would the capacity. In contrast, the measured net total heating capacity across all fan coils shown in Figure 6-7 rarely exceeds 5 kW. Furthermore, during Winter 2020/2021 the heat pump system provided net *cooling* to the Student

Assembly. Table 4-2 shows the rated heat capacity for each fan coil, with Fan Coil 1.03 having a rated heating capacity of 24.2 kW. Figure 6-7 shows that it also very rarely exceeds 5 kW.

The 5-min electrical power consumption data is shown in Figure 6-8, also shown is daily-aggregated electrical energy consumption, and a frequency histogram of the 5-min kW electrical power consumption data. Note power consumption logging did not commence until mid-January 2021. Table 4-2 shows that the heating input electrical power at maximum capacity was 22.2 kW. Figure 6-8 shows that the heat pump is typically consuming 4 to 6 kW and rarely exceeds 10 kW. The heat pump is nearly always operating at a much lower level than its maximum capacity.

Due to the unnecessary simultaneous heating and cooling in the Student Assembly, a heating COP could not ultimately be calculated for much of the heating season. To better understand the heating efficiency of the system, Fan Coil 1.02 was turned off in early March to prevent the simultaneous heating and cooling. Near the end of March, Fan Coil 1.02 was turned back on and Fan Coil 1.01 was turned off. For all of March, the system was operating in heating mode exclusively. Daily-aggregated heating energy provided by each fan coil is shown in Figure 6-9, alongside the daily electricity consumption for the outdoor unit.

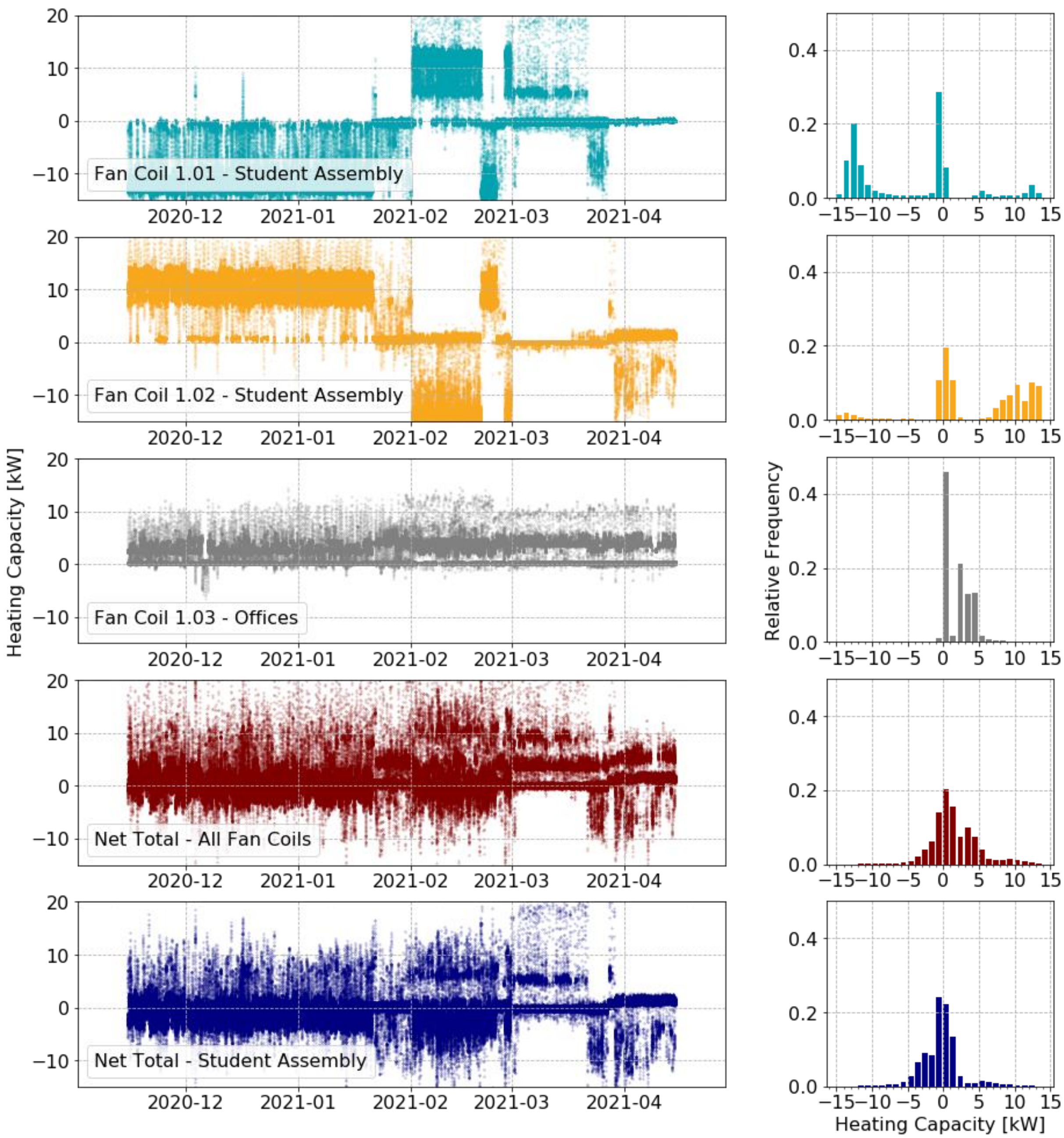


Figure 6-7. Instantaneous heat capacity is plotted for each fan coil. Also shown is the net total for all fan coils, and for the Student Assembly which is served by Fan Coil 1.01 and 1.02. Frequency histograms on the right-hand side indicate how often different instantaneous capacity values were observed during Winter 2020/2021. Note that negative capacity indicates cooling.

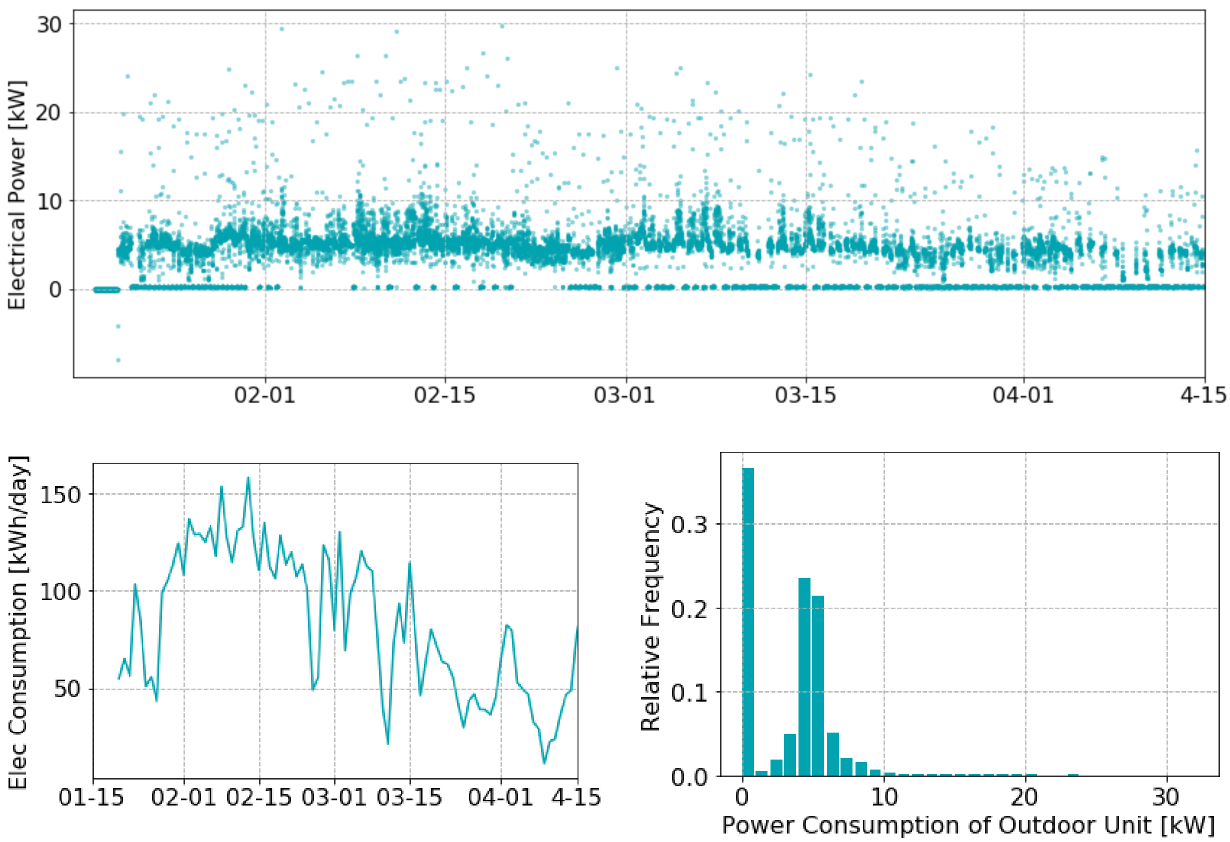


Figure 6-8. Electrical power consumption is expressed as a 5-min time-series, daily-aggregated kWh, and as a frequency histogram. Note that this does not include the electrical power consumption of the indoor fan coils. A maximum power consumption of 22.2 kW was expected if the system operated at maximum capacity.

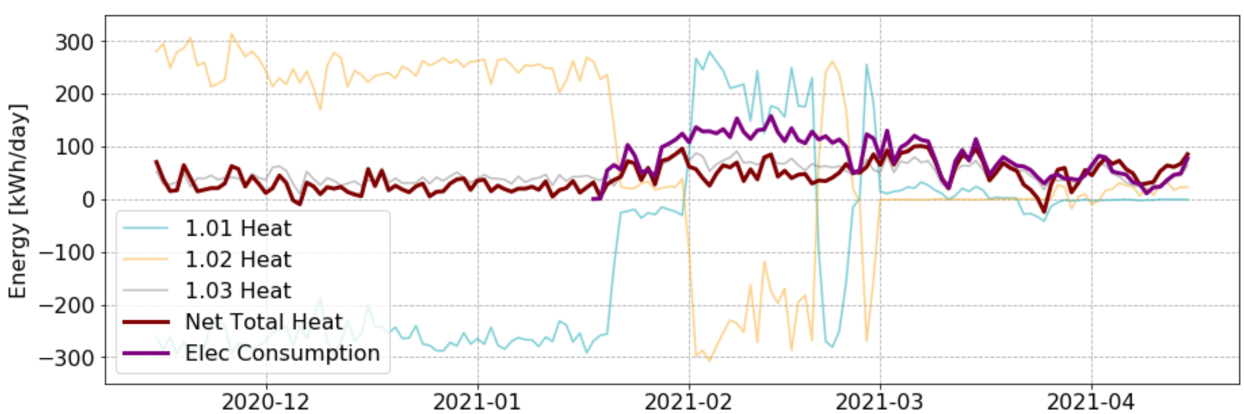


Figure 6-9. The daily-aggregated heating energy provided by the heat pump system, and electrical energy consumption, is shown for Winter 2020/2021. Note that until March, the system provided both heating and cooling – so the net heat is lower than the electrical consumption. However, starting in March, the system is only providing heating because Fan Coil 1.02 was forced off. During this time, the electricity consumption is very close in magnitude to the heating provided.

When the system is providing simultaneous heating and cooling to the Student Assembly the electricity consumption is notably greater than heat energy being provided. This may not be surprising, nor is it an indication that the system is performing poorly. However, in March, when the system is *only providing heating*, daily electricity consumption is still very close in magnitude to the heating being provided – indicating an average daily COP near 1. This is much lower than expected.

It is helpful to look at a higher resolution of the data to understand what is happening in the system. Figure 6-10 shows monitoring data from March 8th and 9th, 2021. Electrical power consumption and heating capacity provided are shown in the top plot. From 10 pm to midnight on March 8th, only Fan Coil 1.03 is on and providing approximately 5 kW of heating. It is also consuming approximately 5 kW of electricity, resulting in a COP near 1.

The temperature change across the outdoor unit is very low, on the scale of 0.5^{o}C. This suggests that there is not much heat being removed from the outdoor air and it corroborates the low COP. Note the inlet/exhaust temperatures are instructive but do not provide a complete picture because airflow is missing. At midnight, Fan Coil 1.01 turns on to provide heating. The total heat then exceeds the electrical consumption by approximately 50% and the COP is greater than 1.

An increase the heat removed from the outdoor air is also seen in the difference between inlet and exhaust temperatures of the outdoor unit. This provides a possible explanation for Figure 6-9. COP appears to be poor when the unit is operating at the bottom of its part-loading capabilities and improves as more capacity is required. However, Figure 6-8 showed that unit is nearly *always* operating at a high-degree of part-loading and the overall COP when evaluated daily is correspondingly low.

Lastly, Figure 6-11 shows a comparison of the outdoor temperature data against the inlet air temperature to the outdoor unit. Recall that the outdoor units are contained in an enclosure and the inlet air can be tempered by natural gas when the outdoor air is very cold. The temperatures agree closely in April when the outdoor air is warm. In the beginning of March, the natural gas heating in the outdoor enclosure heats the inlet air from -14^{o}C closer to – 5^{o}C as expected.

Later in March, outdoor air tempering is happening even when outdoor air temperatures are closer to 0^{o}C. Previous plots showed that the heat pump is not operating near its maximum capacity and therefore could be operated without the natural gas back-up heating in colder temperatures. Recall that the natural gas unit heaters in the enclosures are relatively low-efficiency (82 to 83%), and from a carbon perspective, should be avoided whenever possible.

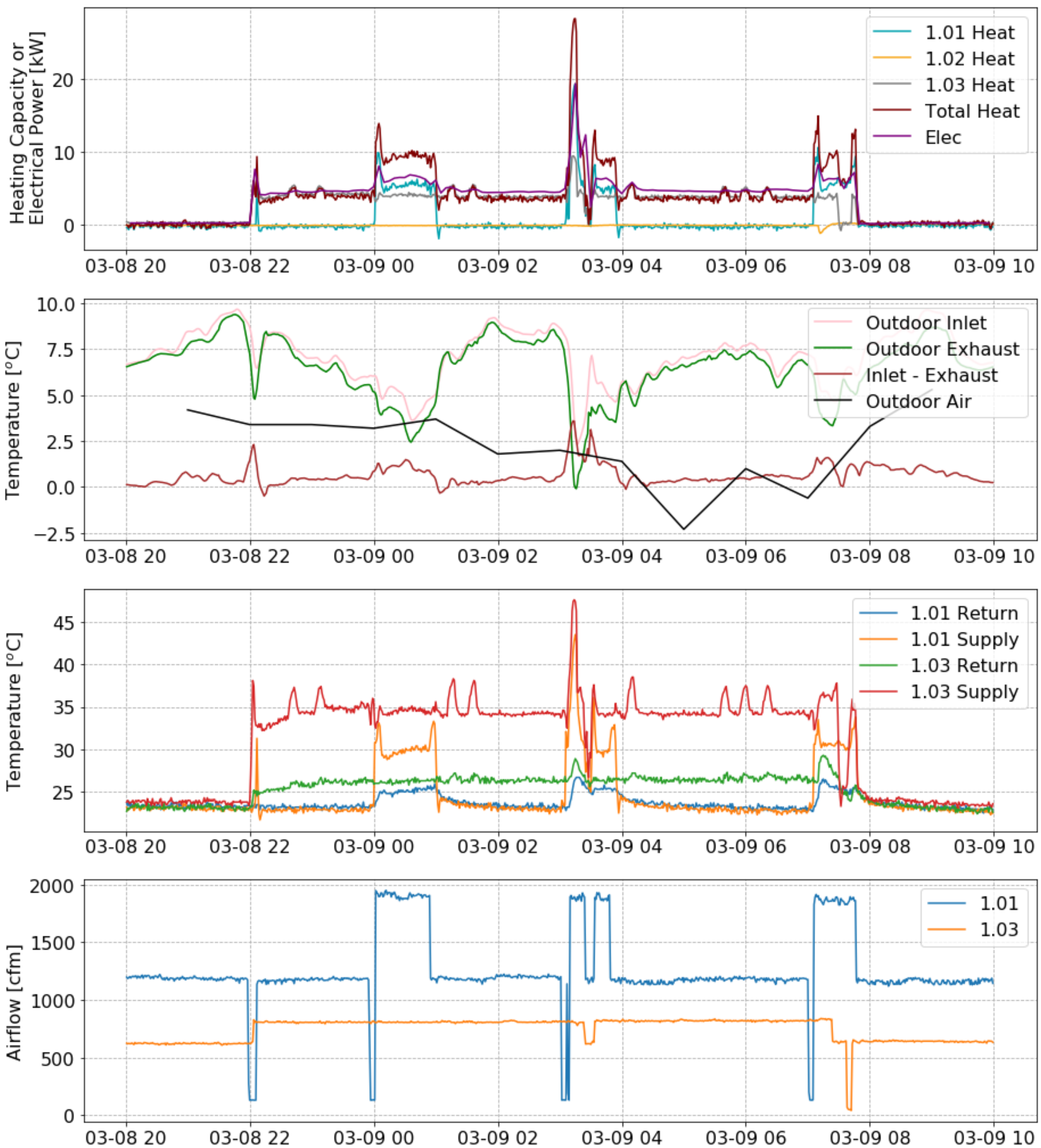


Figure 6-10. System performance data and parameters for March 8th to 9th, 2021.

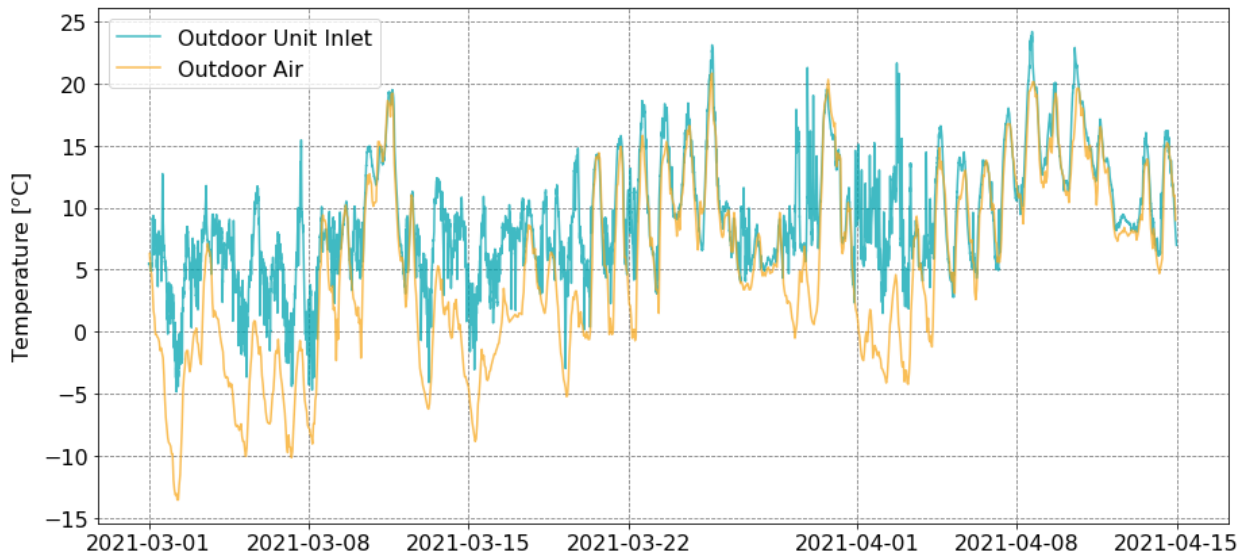


Figure 6-11. In cold conditions the “Outer Unit Inlet” air temperatures are greater than “Outdoor Air” temperature because the inlet air is tempered by natural gas heating.

7.0 DETAILED MONITORING: COOLING RESULTS YEAR 1

Monitoring data from June 15th, 2020, to September 15th, 2020, was evaluated to better understand the system operation in cooling mode. Time-series temperature data is shown in Figure 7-1. Frequency histograms of the temperature data are shown in Figure 7-2. Zone temperatures at chest level are shown in Figure 7-3.

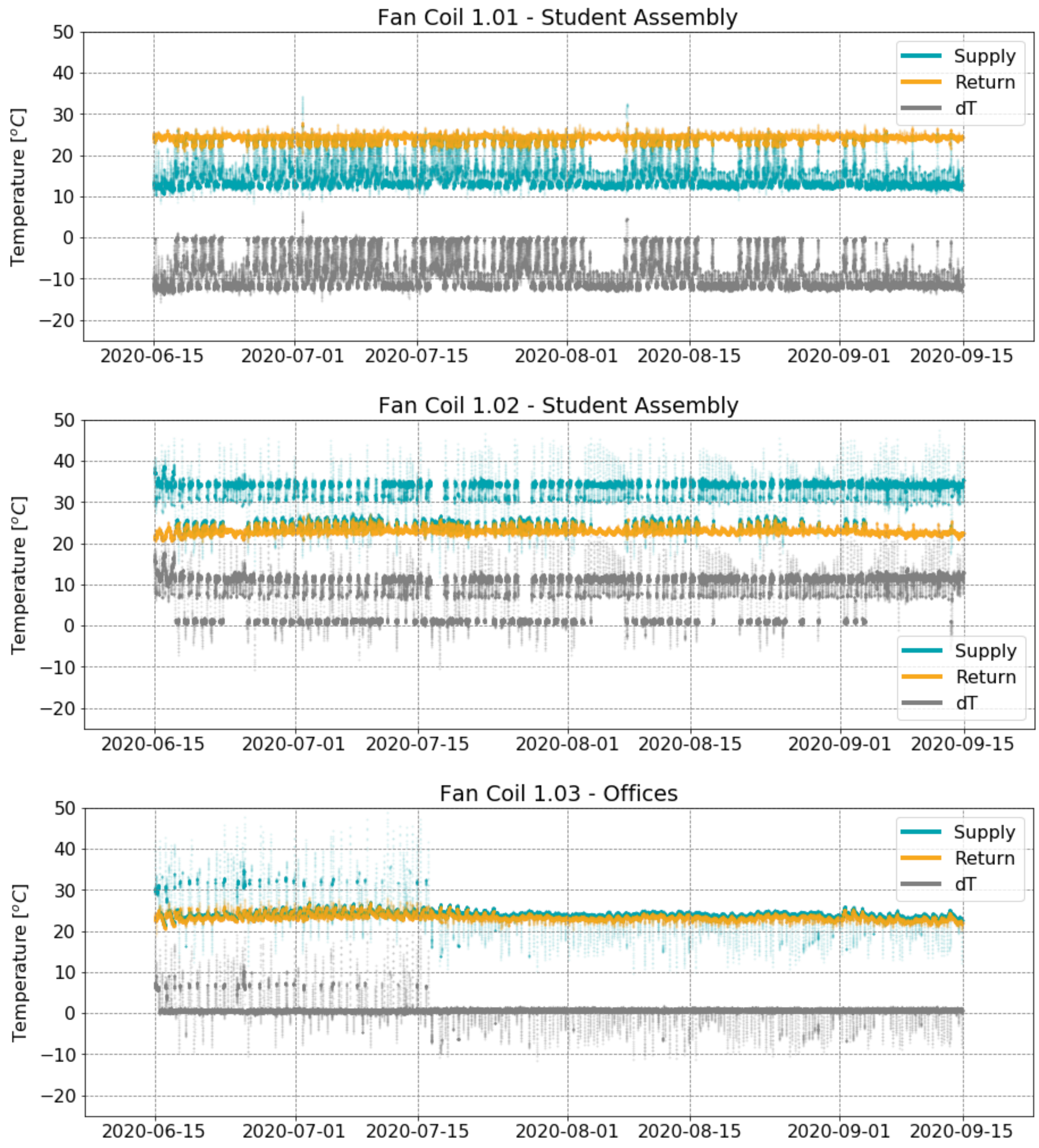


Figure 7-1. Supply and return temperature (and their difference) is shown for Fan Coils 1.01, 1.02, and 1.03.

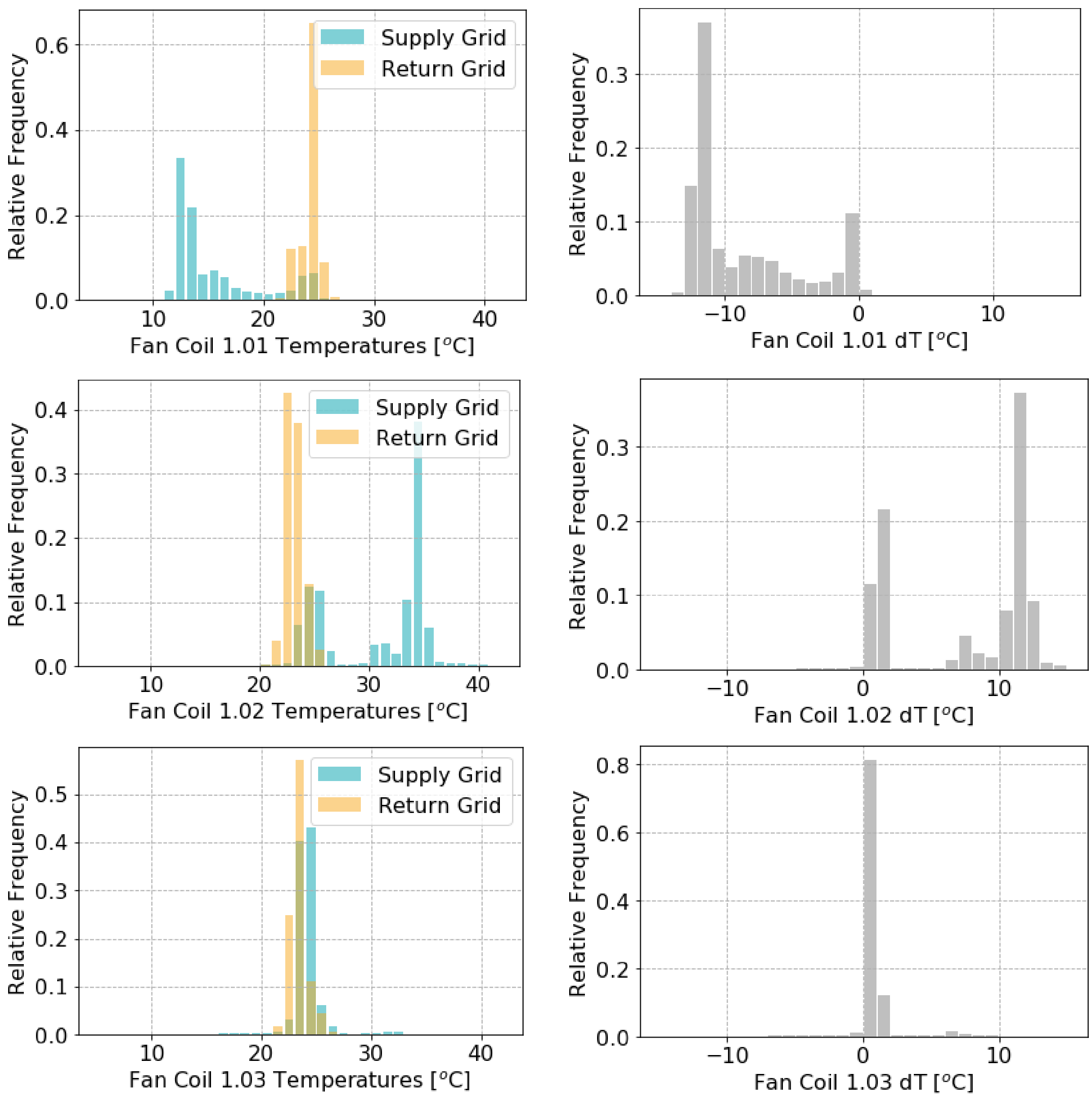


Figure 7-2. Frequency histograms are shown for the supply and return temperature (and temperature difference) for each fan coil in the monitored subsystem. It is evident that Fan Coil 1.01 is providing cooling, Fan Coil 1.02 is providing heating, and Fan Coil 1.03 is essentially off all summer. Recall that Fan Coil 1.01 and 1.02 are serving the same zone – the Student Assembly.

Figure 7-4 displays the outdoor air temperatures for this period alongside data from the CWEC database, showing a historically warm summer. The Summer 2020 airflow data is shown as a time series in Figure 7-5 and as a frequency histogram in Figure 7-6.

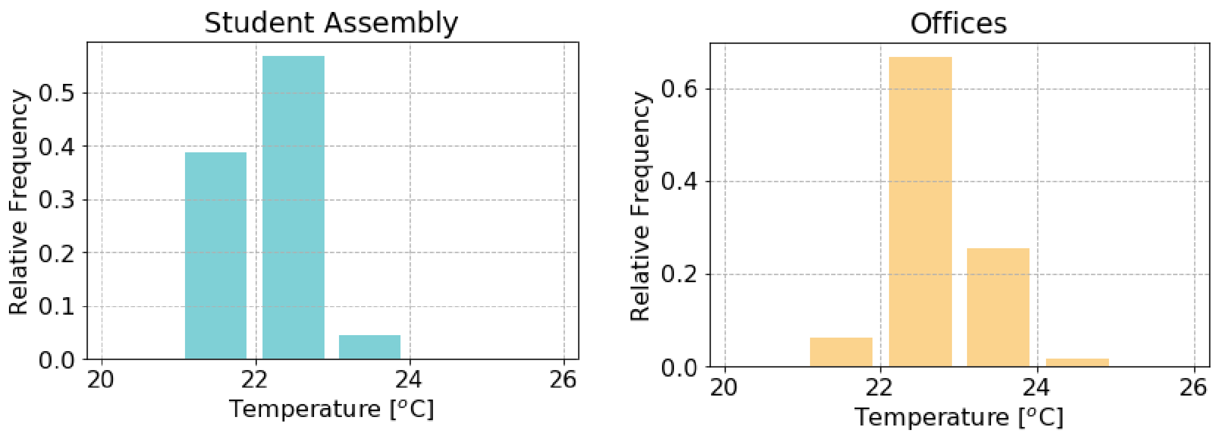


Figure 7-3. Zone temperatures at chest level during Summer 2020.

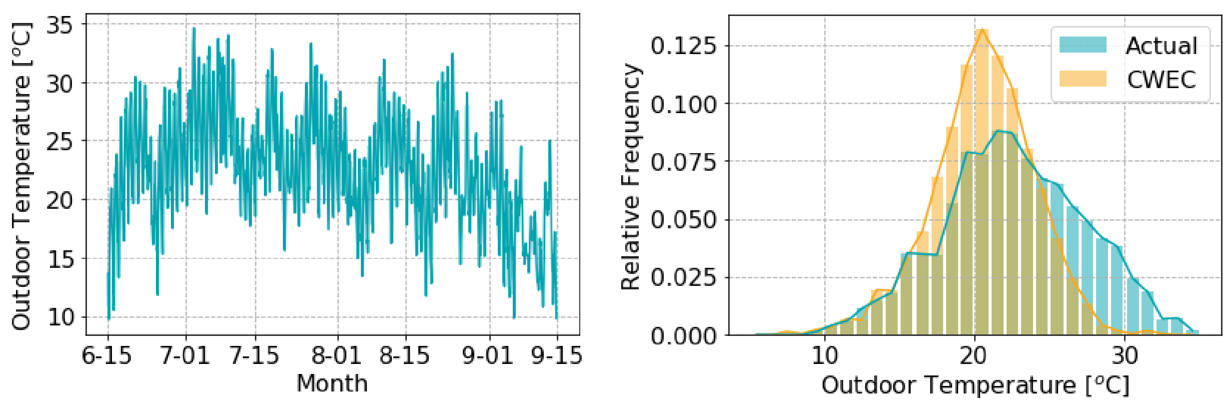


Figure 7-4. Outdoor air (dry bulb) temperatures are shown both as a time-series and as a frequency histogram. Also shown is a typical meteorological year from the CWEC database. Summer 2020 was much warmer than the TMY.

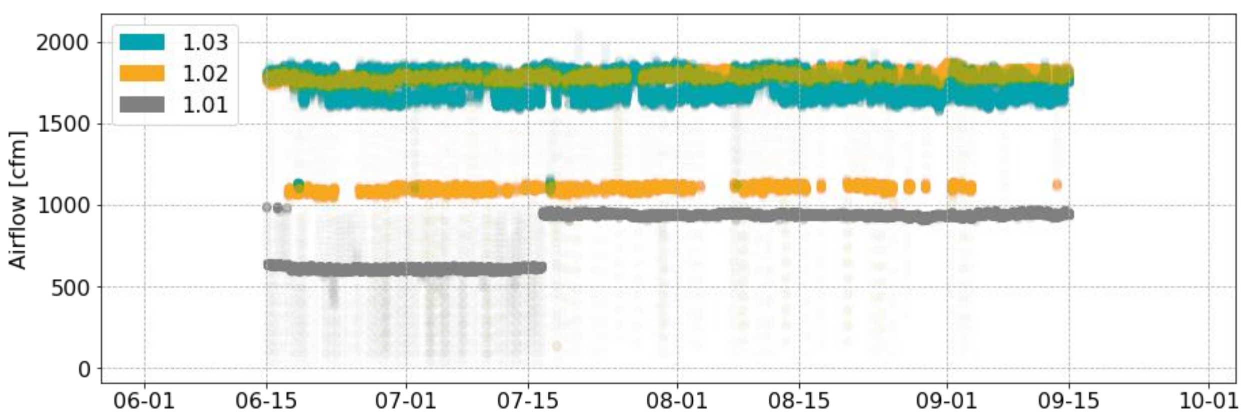


Figure 7-5. Time series airflow data from each fan coil is plotted for Summer 2020.

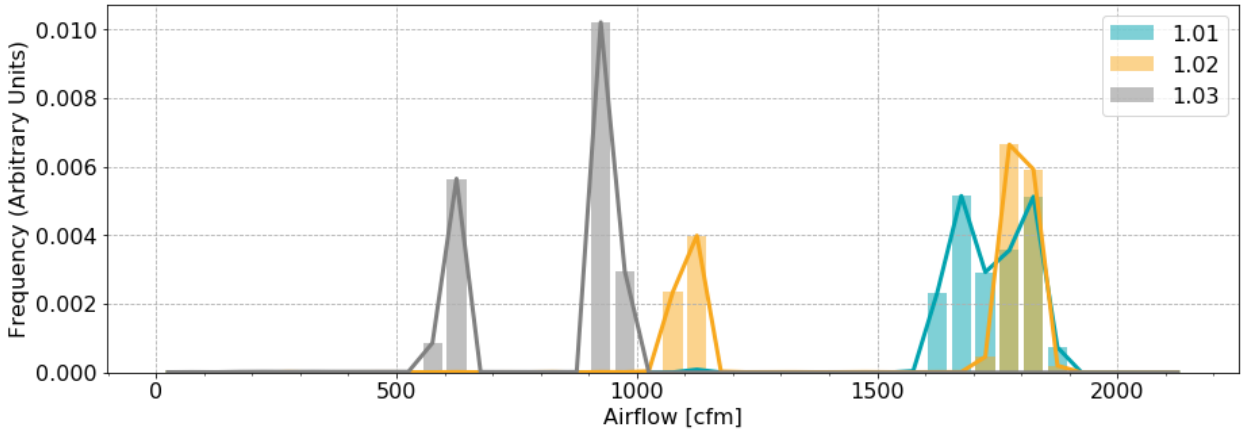


Figure 7-6. Airflow data for Summer 2020 is shown for each of the three fan coils in the monitored subsystem.

The system cooling (or heating) capacities were calculated using the process described in Appendix E and are shown in Figure 7-7. Note this figure includes only the sensible portion of cooling and *not* the latent capacity from dehumidification. Fan Coil 1.01 provides cooling only throughout the summer. Fan Coil 1.02 serves the same zone as Fan Coil 1.01 and provides heating only. Fan Coil 1.02 is providing heating approximately two thirds of the time during Summer 2020 while Fan Coil 1.01 is cooling.

There is essentially no cooling provided to the Offices and the net cooling to the Student Assembly is also near zero. Despite the lack of cooling being provided, zone temperatures were normal (Figure 7-3). The daily-aggregated heating and cooling provided by each fan coil is shown in Figure 7-8. It is apparent that Fan Coil 1.01 and 1.02 were directly working against each other on a daily basis. This caused unnecessary electricity consumption. Note that electricity monitoring was not set-up for summer operation in Year 1.

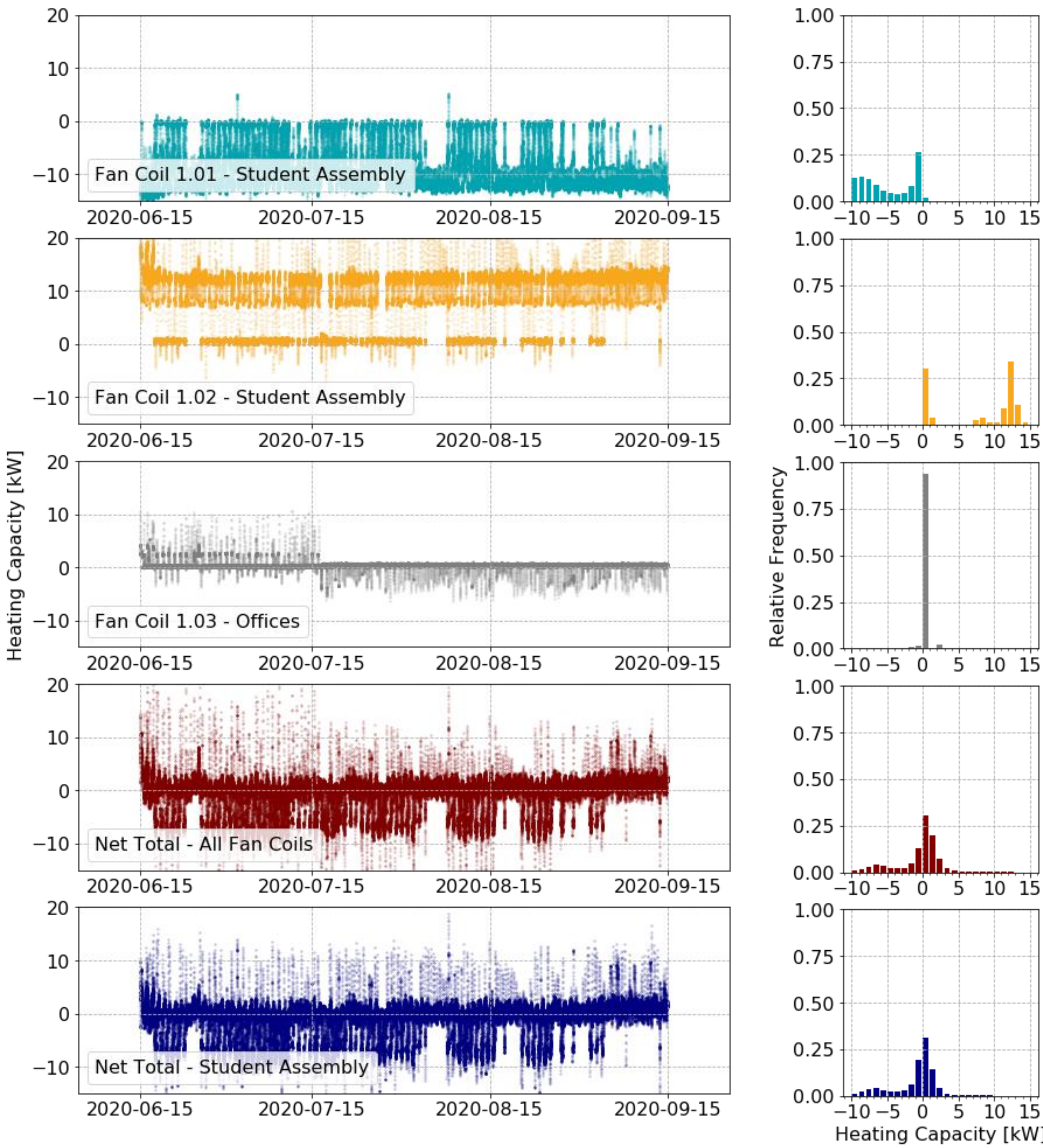


Figure 7-7. Instantaneous (minute-level) heating and cooling capacity are shown both as a time-series and as frequency histograms for Summer 2020. Fan Coil 1.03, serving the Offices, is essentially never on. Fan Coil 1.01 and 1.02 serve the Student Assembly but one is providing heating while the other is providing cooling throughout the summer. The net cooling provided to the Student Assembly over the summer is near zero.

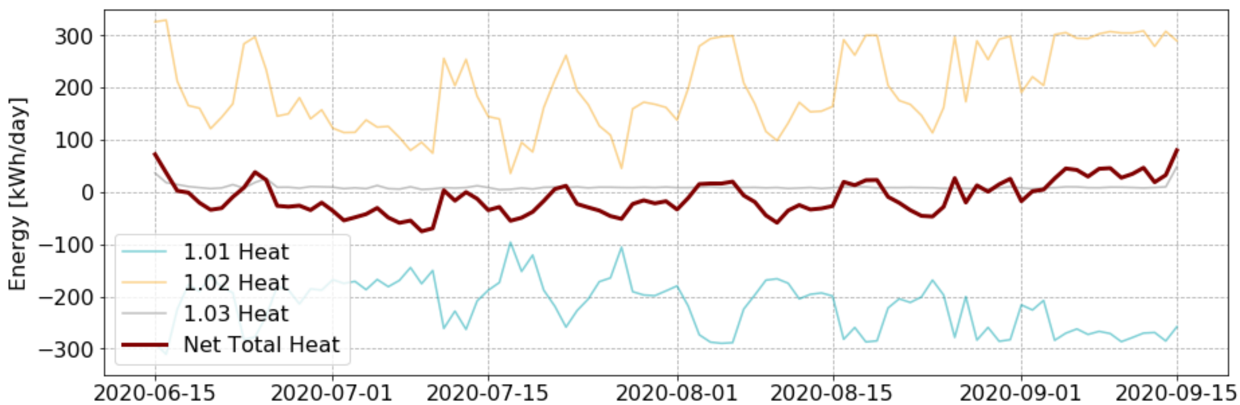


Figure 7-8. The daily-aggregated heating and cooling capacity for each fan coil is shown for Summer 2020. Fan Coil 1.03 serves the Offices and is essentially off for the summer. Fan Coil 1.01 and 1.02 serve the Student Assembly and are directly working against each other simultaneously heating and cooling the same zone.

8.0 DETAILED MONITORING: KEY SYSTEM ISSUES YEAR 1

The detailed monitoring data from Year 1 identified various system issues that needed to be addressed prior to Year 2. They are listed below.

8.1 Issue 1: Simultaneous heating and cooling in same zone

Fan coils were simultaneously heating and cooling the same zone. Effectively, where a zone was served by two fan coils, it was common that one fan coil was heating while the other was cooling. This was also happening in other zones beyond the monitored subsystem. The issue causes electricity to be unnecessarily consumed while having no net benefits to the building.

This problem was rectified building wide by: i) ensuring correct thermostat setpoints and spans; ii) setting the primary control of the building to heating or cooling to ensure that during any time period there could only be one mode of operation (heating or colling); and iii) disabling some thermostats in areas that contained more than one thermostat. Also note that ii) also meant the system could not be used for heat recovery when simultaneous heating and cooling was legitimately required between adjacent zones.

8.2 Issue 2: Natural gas systems overheating zones

The Student Assembly had net *cooling* throughout the winter. This could be attributed to three potential causes: i) this zone was directly adjacent to the mechanical room which may have contributed to internal heat gains; ii) it may have been receiving more tempered ventilation air than was required due to a balancing issue and lack of a ventilation schedule; and iii) perimeter heating was operating on full capacity at all times throughout the winter, providing more heat than required to all zones.

From analysis of the airflow and return temperature data the cause is likely ii). Fan Coil 1.02 is essentially always on and return air temperatures are very warm (much warmer than the temperatures measured at the thermostat in the zone). It was unlikely that this could have been reasonably foreseen by the system designer that specified the equipment.

A schedule was introduced for the ventilation air. This on/off schedule operate the ventilation during normal building occupancy. Demand control ventilation strategies were discussed and explored, but the complexity of the system and lack of zone dampers lead to this being an unviable option.

Note that while iii) was likely not the issue in the Student Assembly, it was an issue building-wide causing an over-reliance on natural gas heating. It was addressed by placing the boiler on an outdoor reset schedule such that it was not always operating at full capacity during the winter.

8.3 Issue 3: Lower than expected COPs at part-load

The measured COP appeared lower than expected, specifically when the heat pump was operating at a fraction of the rated capacity. The heat pump frequently operated in this regime because the heating load was much lower than the rated capacity of the equipment (from the previous issue). COP appeared to improve at greater levels of loading. The extent to which this was an issue building-wide was not known since COP monitoring is resource-intensive and was only completed in one zone.

Overall, it was difficult to fully diagnose the issue due to the lack of complete power monitoring data for cooling and heating. Year 2 offered an opportunity for more data collection. A compounding issue contributing to a lower-than-expected COPs was the high return temperatures, which is also related to Issue 2. The rated COP is provided for a return temperature of 21.1 ºC. The return temperature frequency histograms show they were much higher than that in the monitored subsystem.

8.4 Issue 4: Over-reliance on gas tempering in the outdoor unit enclosure

The outdoor enclosure for the heat pumps was equipped with tempering from a natural gas unit heater. The unit heater turned on in relatively mild conditions which increased the gas consumption of the facility. It is also worth noting that the gas unit heater does not have a high-efficiency and, from a carbon perspective, should be used only when needed.

The high set-point for gas tempering was ultimately related to a condensate management problem within the outdoor enclosure. There was no path for condensate from the defrost cycles to leave the building. When the building dropped to sub-freezing temperatures, the condensate froze on the floor. This occurred initially and ended up heaving the heat pumps in the outdoor enclosure. Heat trace was later installed under the equipment.

The floor in the remainder of the outdoor enclosure would still freeze over with ice, creating a safety hazard. The best solution was proper condensate management in the enclosure. A short-term fix was to keep the outdoor enclosure warmer with the gas unit heater. The latter is what was done at the Visitor Centre.

A broader issue is that, if the short-term fix appears to resolve the issue, there may be less motivation to implement the long-term fix. The consequence of the high tempering set-point is an over-reliance on inefficient carbon-intensive natural gas heating. System designers are typically no longer designing these systems with enclosed outdoor units, so this may be less of an issue with future systems.

Moving into Year 2 (Summer 2021 and Winter 2021/2022), the following changes were made:

- Perimeter heating was put on an outdoor reset schedule;
- Ventilation was scheduled to only operate during normal building occupancy hours;
- Fighting thermostats in the same zone were managed by switching the system to only operate in heating or cooling, effectively managing set-points, and by reconfiguring zones with multiple thermostats to be controlled by one.
- No changes were made to address the COP in the monitored subsystem or outdoor enclosure gas tempering setpoint.

9.0 DETAILED MONITORING: HEATING RESULTS YEAR 2

Monitoring data from November 15th, 2021, to April 15th, 2022 is shown in Figure 9-1. The simultaneous heating and cooling issue within the Student Assembly was addressed.

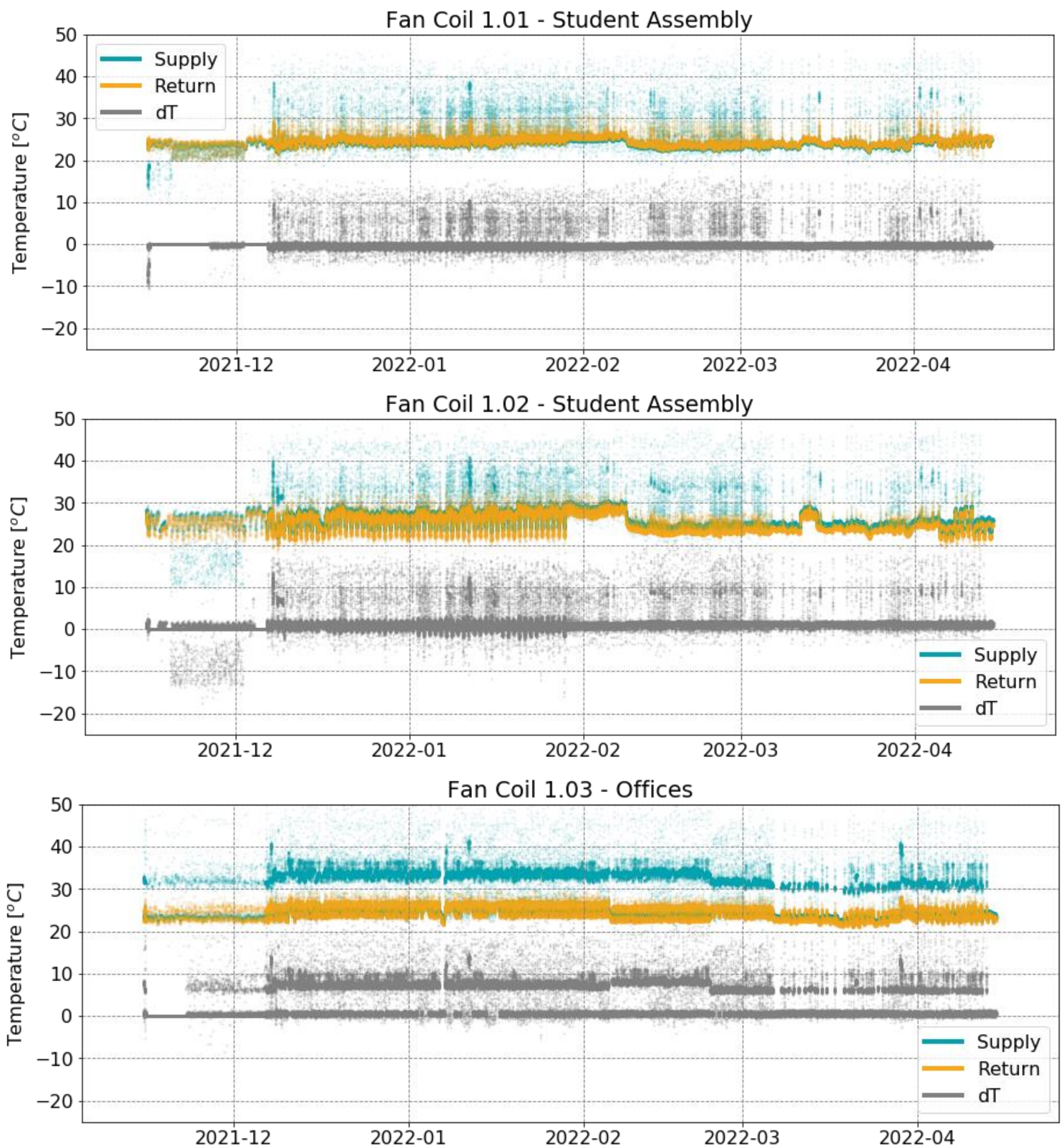


Figure 9-1. Supply and return temperatures (and their difference) is shown for each fan coil during Winter 2021/2022. The simultaneous heating and cooling issue within the Student Assembly was addressed.

Figure 9-2 shows histograms of fan coil supply and return temperatures as well as the temperature difference between supply and return. Fan Coil 1.01 and 1.02 are off nearly all the time. Fan Coil 1.03 is on less than half of the time, producing a small temperature rise of 5 °C to 10 °C across the fan coil.

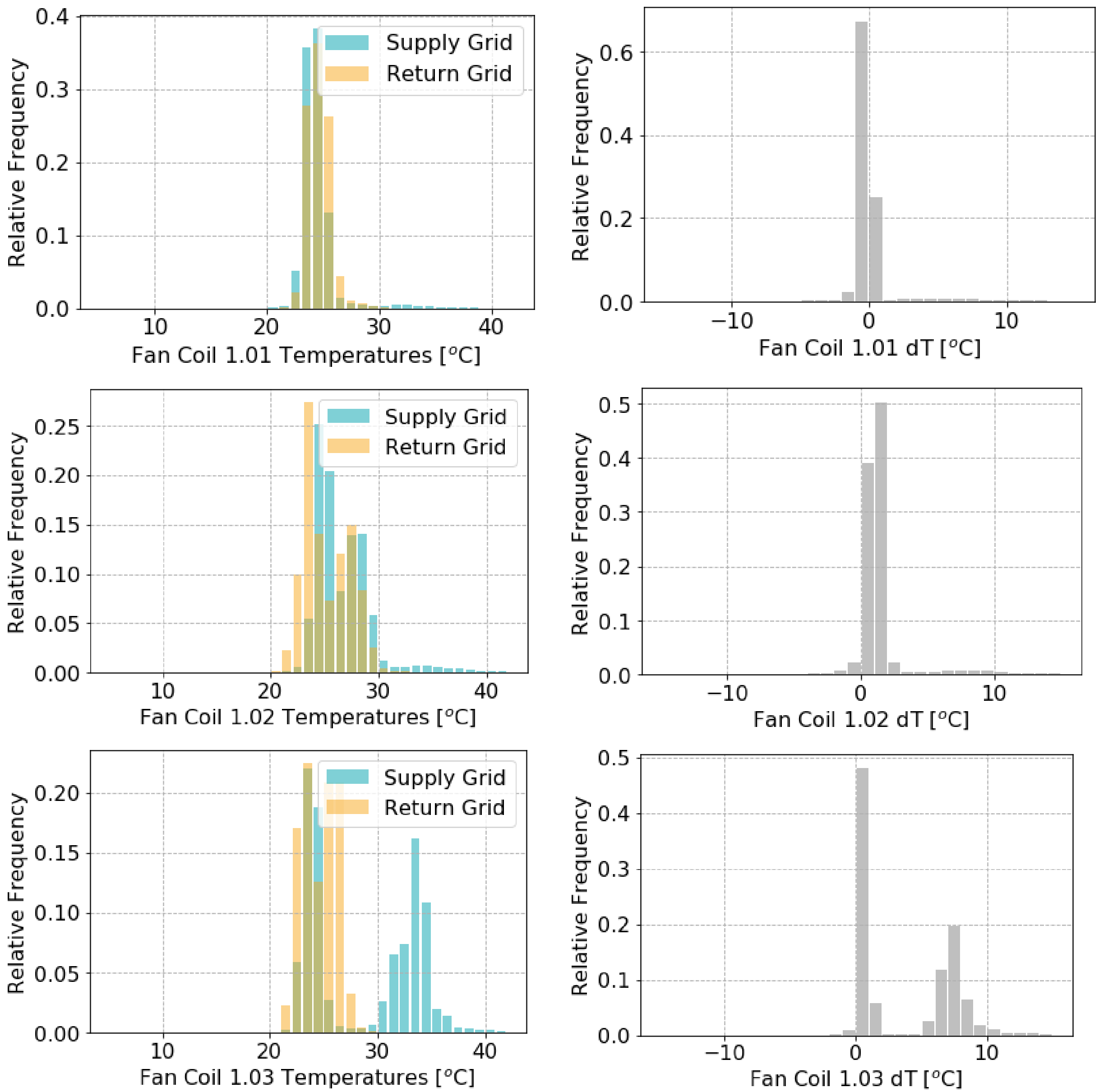


Figure 9-2. Relative frequency histograms are plotted for fan coil supply temperature, return temperature, and temperature difference (dT) for Winter 2021/2022. Fan Coil 1.01 and Fan Coil 1.02 appear to be off almost entirely for the heating season, while Fan Coil 1.03 is typically operating on a low setting.

Zone temperatures are typically between 22 °C and 24 °C, as shown in Figure 9-3. Outdoor temperature data is shown in Figure 9-4 and it is also contrasted with a typical meteorological year. Winter 2021/2022 had much more extreme cold than a typical year. A time-series plot of airflows is provided in Figure 9-5 and a frequency histogram of airflows is provided in Figure 9-6.

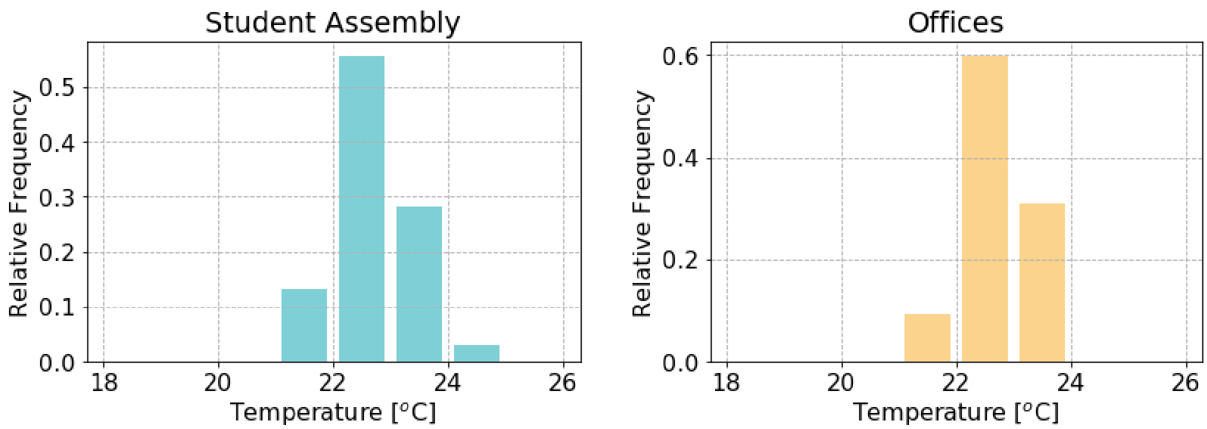


Figure 9-3. Zone temperatures during Winter 2021/2022 measured by monitoring system with sensors located at chest level in the zones near NAVs.

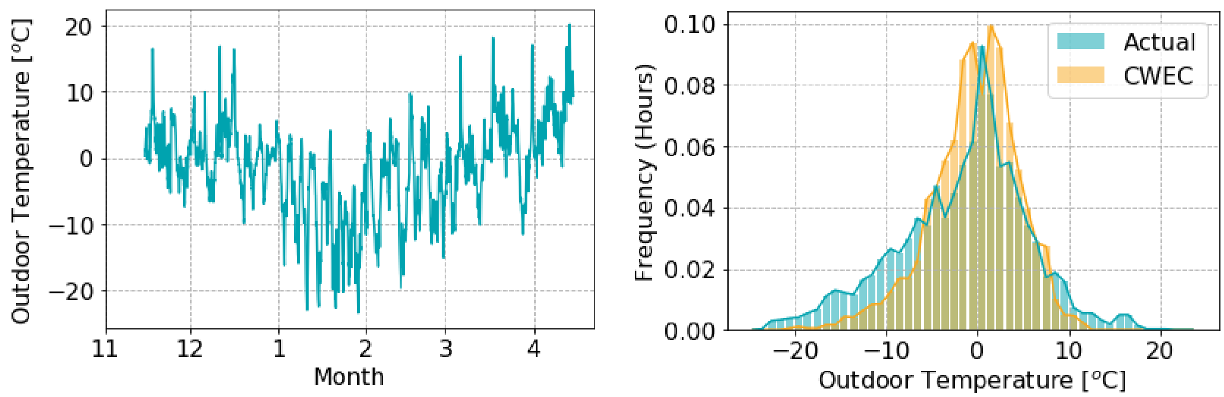


Figure 9-4. Outdoor temperature data during Winter 2021/2022 is shown both as a time-series and as a frequency histogram. Winter 2021/2022 had much more extreme cold than a typical winter.

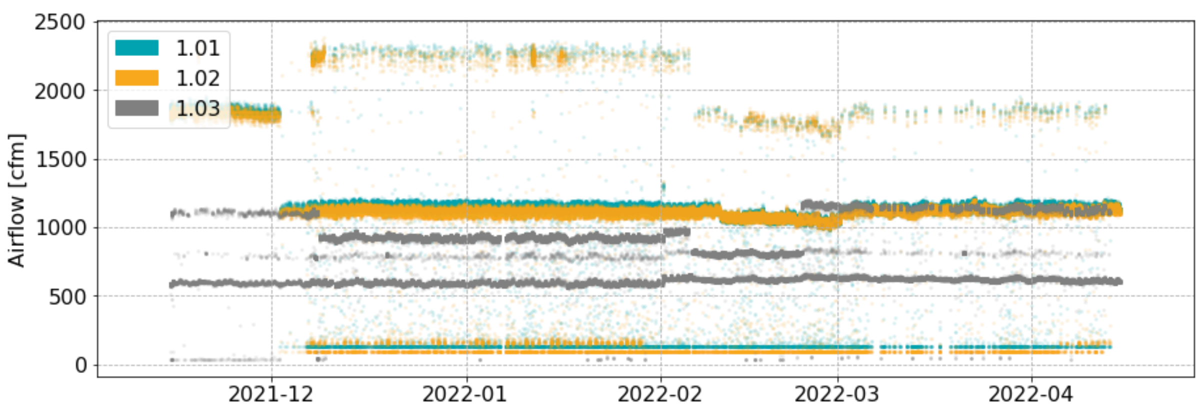


Figure 9-5. A time-series plot of the airflows shows the different fan speeds. While Fan 1.03 operates at four different speeds, Fan Coil 1.01 and 1.02 are nearly always at a single speed.

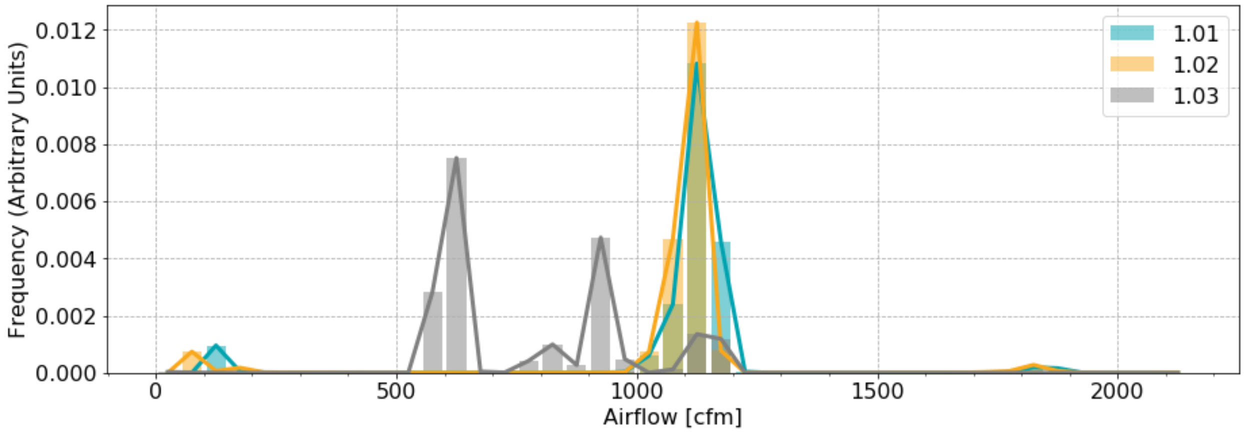


Figure 9-6. A frequency histogram showing the relative fraction of time that each indoor fan coil is providing different airflows during Winter 2020/2021. Fan coil 1.01 and 1.02 are always on the lowest speed. Fan coil 1.03 is most often on the first speed, almost never on the fourth (highest) speed, and sometimes on the intermediate speeds.

Heating Capacity for each fan coil, as well as the net heating for the Student Assembly and the overall system, is shown in Figure 9-7. Fan Coil 1.01 and 1.02 are almost never providing any heating capacity and Fan Coil 1.03 is providing a small amount of heating approximately half of the time.

In Figure 9-8, electrical power consumption is shown as a time series, as daily aggregated totals, and as a frequency histogram. The rated power for the unit at maximum capacity is 22 kW. The unit is off 50% of the time and for the remainder of the time it is consuming in the vicinity of 5 kW of electricity. Total daily heating output is contrasted against electricity consumption in Figure 9-9.

Figure 9-9 shows that, for the submetered system, the electricity energy consumption is typically greater than the heat being provided. It is also important to note that the heat being provided is very low and this trend is not necessarily indicative of the whole building. It is helpful to look at a full day of monitoring data where the daily aggregated heat delivery was less than the energy consumption. Figure 9-10 shows data from January 21st, 2022.

Figure 9-10 shows that Fan Coil 1.03 is primarily on, with Fan Coil 1.01 and 1.02 only turning on briefly. When only Fan Coil 1.03 is operating, the ASHP is operating at low fraction of its rated capacity, and the COP is low. The fact that no heat is being absorbed by the outdoor air is corroborated by the temperature difference across the outdoor coil. When the COP is low, the temperature difference across the outdoor coil is negligibly small and this indicates no heat is being removed from the outdoor air.

The temperature difference across the outdoor coil increases as COP increases. It becomes negative when everything is off since there is no longer any airflow across the coil; it just naturally floats to different values since the exhaust is ducted but the inlet is not. The research team did not determine why the COP would drop to as low as 0.5 instead of 1. Heat losses in the refrigerant lines and branch boxes may offer a partial explanation. Also note that the return temperatures are high, on the scale of 25 °C, and this degrades performance.

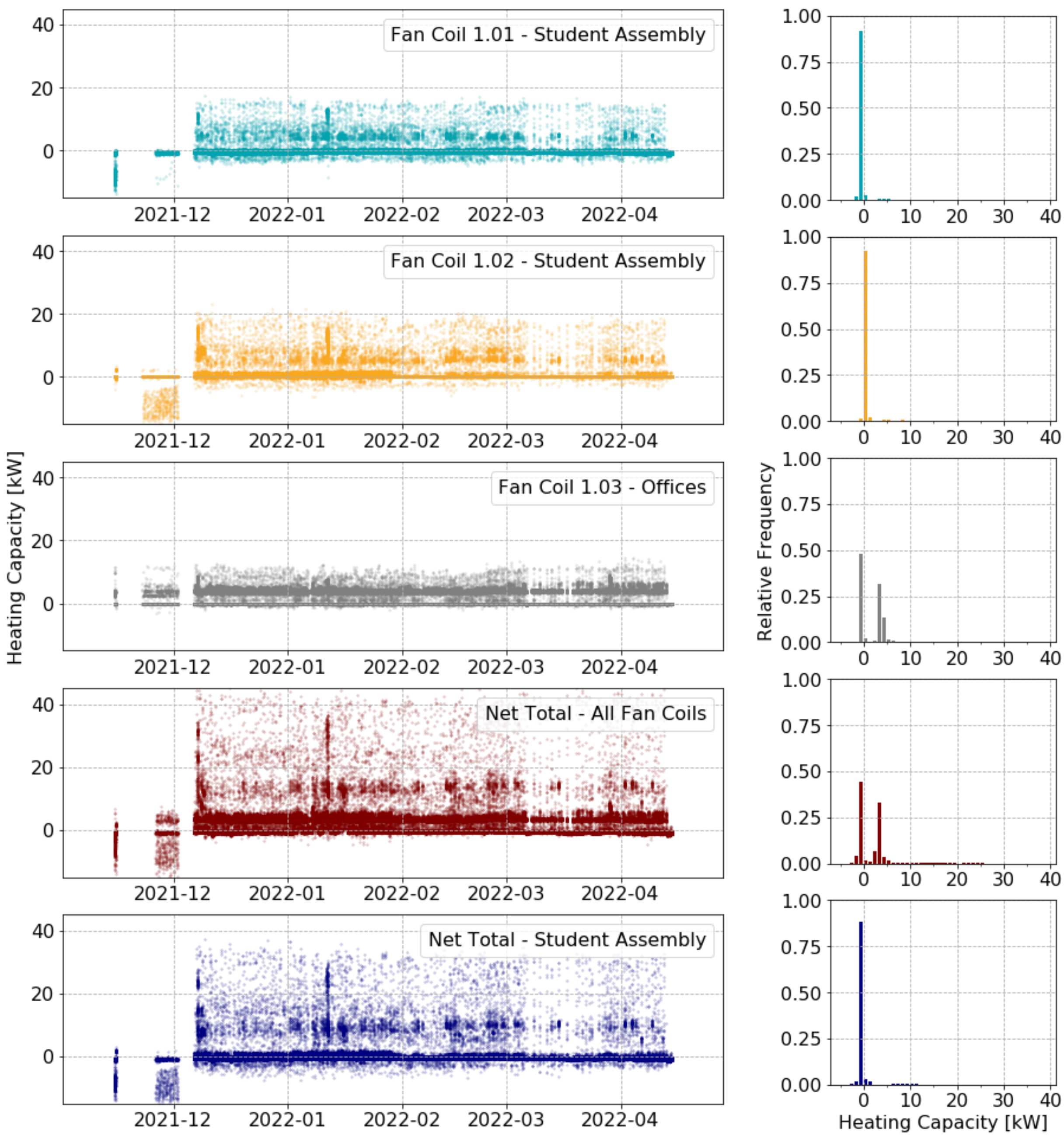


Figure 9-7. Instantaneous (minute-level) heating and cooling capacity are shown both as a time-series and as frequency histograms for Winter 2021/2022. Fan coils 1.01 and 1.02 are almost never providing any heating capacity, and Fan coil 1.03 was often on a low setting.

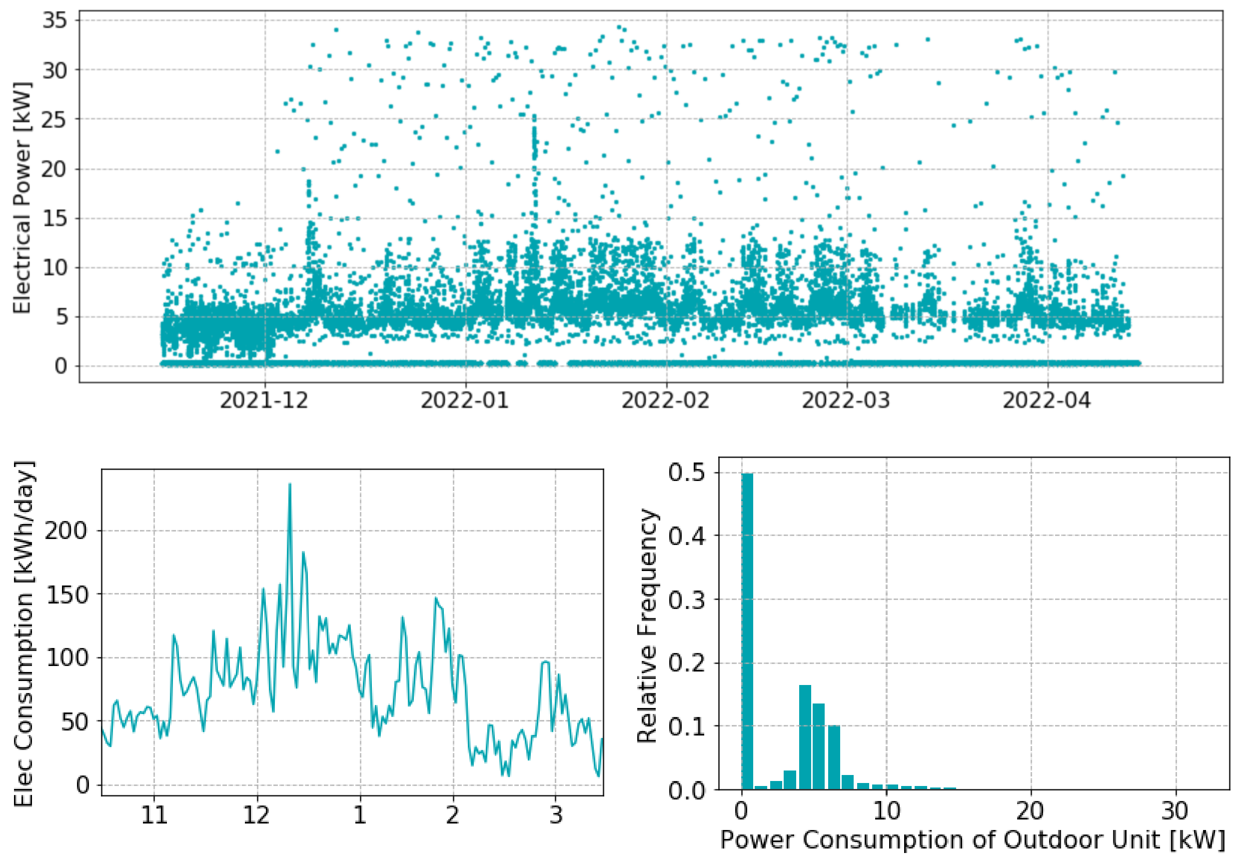


Figure 9-8. Electrical power consumption is expressed as a 5-min time-series, daily-aggregated kWh, and as a frequency histogram. Note that this does not include the electrical power consumption of the indoor fan coils. A maximum power consumption of 22.2 kW was expected if the system operated at maximum capacity.

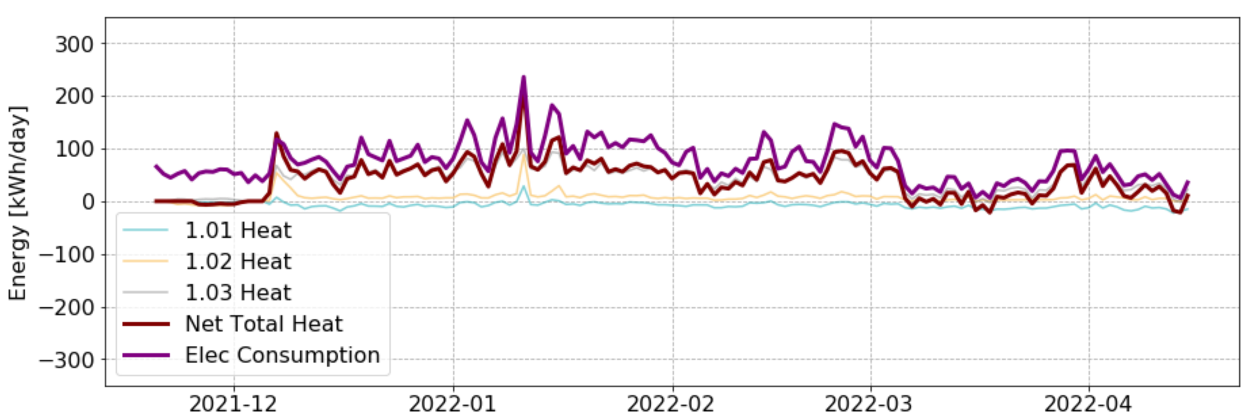


Figure 9-9. The daily-aggregated heating energy provided by the heat pump system, and electrical energy consumption, is shown for Winter 2021/2022. The electricity consumption is greater than the heat being delivered.

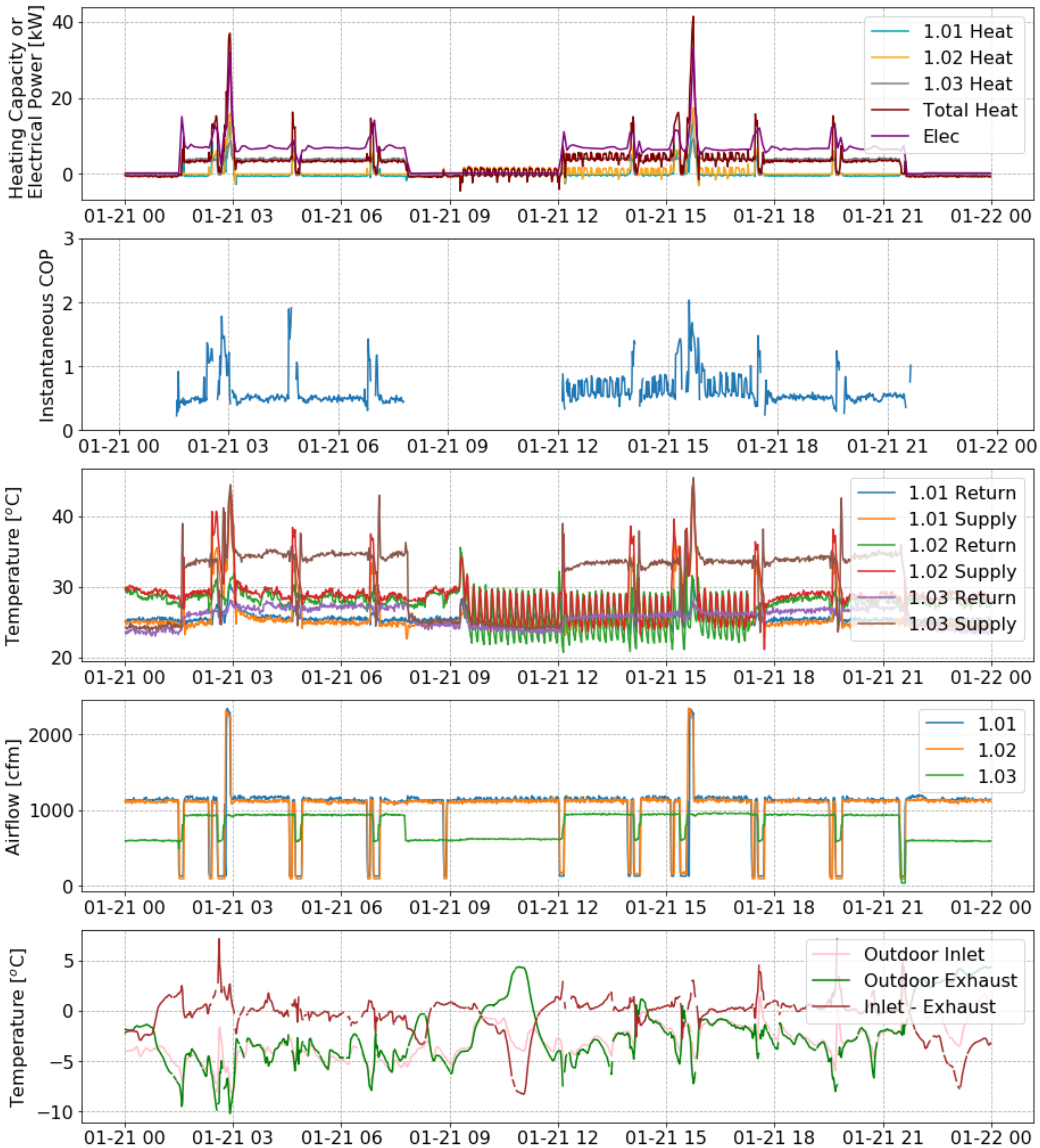


Figure 9-10. Key parameters are plotted for January 21st, 2022. It is primarily Fan Coil 1.03 that is on (it overlaps with "Total Heat"), and COP is low. COP increases when other fan coils turn on, but they only ever turn on briefly.

The total heating capacity is plotted against the electrical power draw (aggregated in 5-minute intervals) in Figure 9-11. The data is further aggregated according to the temperature of the air entering the outdoor unit. This plot only contains data when all fan coils were in heating mode (or off) so there is no heat recovery taking place. Recall that roughly half of the time the unit was consuming 5 to 8 kW and for the remainder it was primarily off. Around a power draw of 5 to 8 kW, the system heating output generally does not exceed the power input.

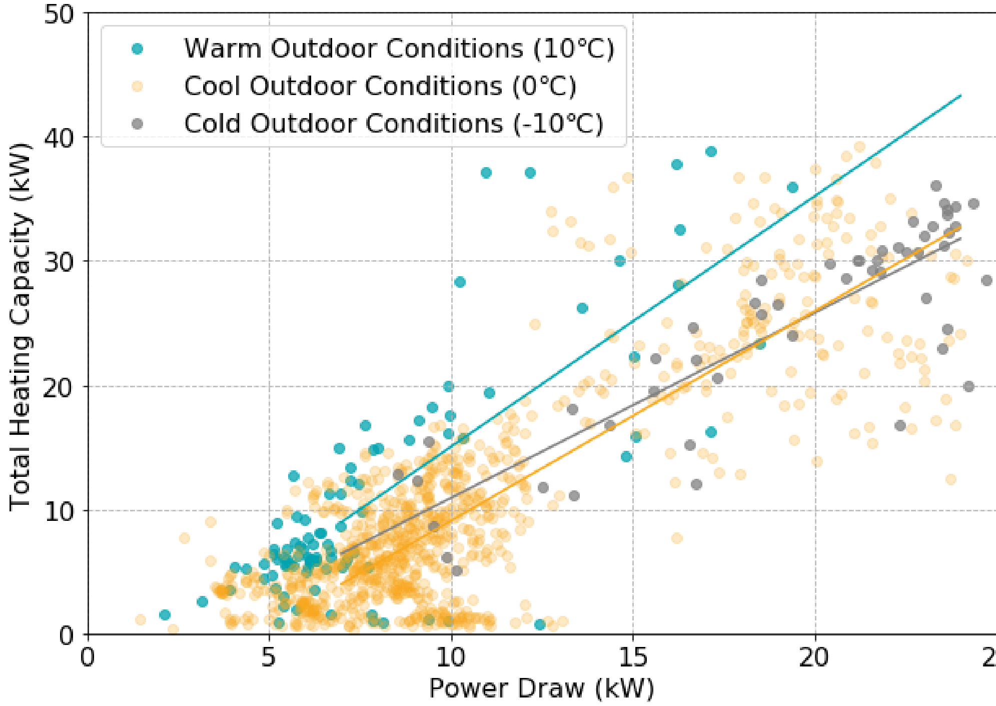


Figure 9-11. The total heating capacity is plotted against the electrical power draw of the outdoor unit in 5-minute intervals and aggregated according to the air temperature entering the outdoor unit. Above 10 kW, the system begins to produce a greater energy output than its electrical energy input. However, for the majority of the time the system power consumption is less than 10 kW.

Note that the system was forced to produce greater heating capacity and power draw on two different days (December 7th, 2021 and January 11th, 2022) where the research team opened the exterior doors to increase the heat load of the zone. This is how the data with higher power consumption was obtained. At higher power consumption the system does produce a greater heating output than its electrical power input, but typically by less than a factor of 2. Data from December 7th, 2021, when the system was forced to produce greater heating, is shown in Figure 9-12.

The system clearly produced greater COPs at greater fractions of the rated load, with COPs approaching 2.0. A possible reason the COPs did not increase beyond 2.0, was the high return temperatures, which were between 25 and 30 °C when the unit was on a higher capacity. The heat pump COP is rated assuming a return temperature of 21.1 °C. The outdoor temperature also reached -5 °C during this time. The same exercise was done on January 11th, 2022, and data is shown in Figure 9-13. COP approaches 1.5 when forced to produce high capacity. Outdoor temperatures dropped as low as -10 °C and the return temperatures were again, between 25 and 30 °C. Note that "Outdoor Air" is from a local weather station, and some discrepancy is expected.

There may also be an issue with the control of the dampers in the outdoor enclosure. Cold air from the heat pump exhaust can be recirculated. This may be happening when the unit turns on at higher capacity since the data shows that the inlet temperature drops rapidly.

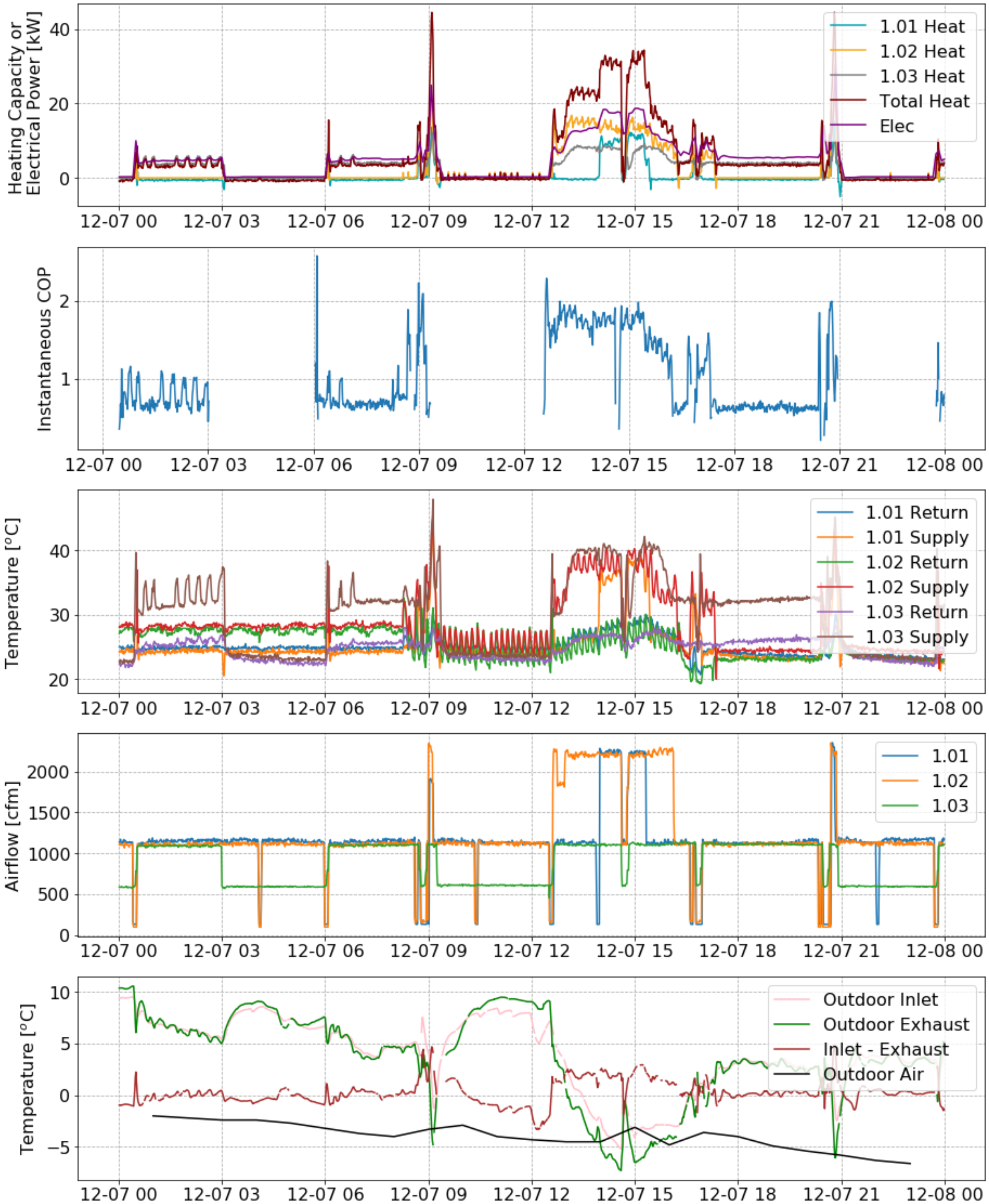


Figure 9-12. Key system parameters were plotted for December 7th, 2021, when the system was forced to provide greater capacity by opening the doors of the student assembly and letting cold air into the space.

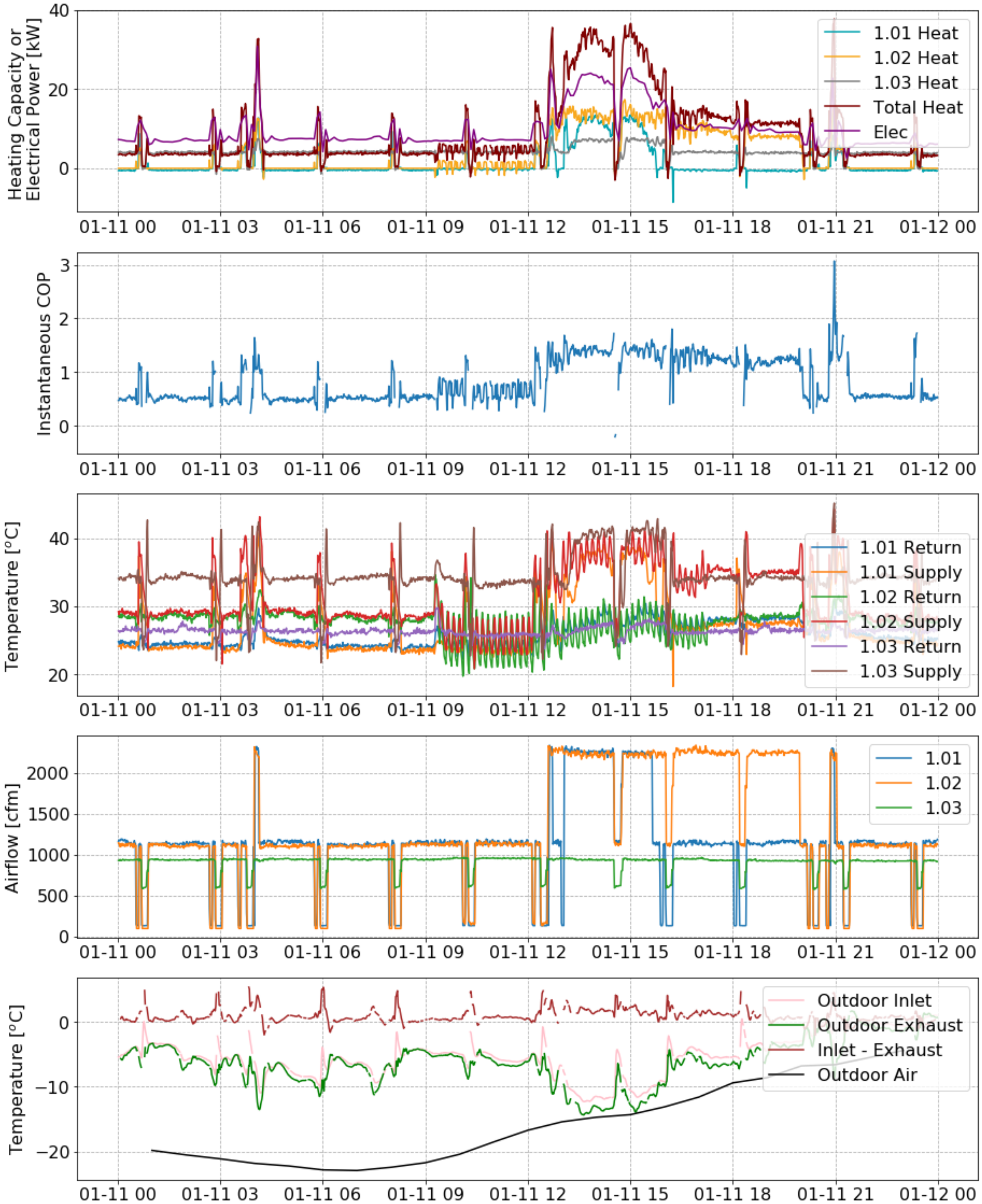


Figure 9-13. Key system parameters were plotted for January 11th, 2022, when the system was forced to provide greater capacity by opening the doors of the student assembly and letting cold air into the space.

The COP and power draw were aggregated into 5-minute intervals for Winter 2021/2022 and plotted in Figure 9-14. The system rarely exceeds a COP of 2.0. As previously mentioned, it is important to understand that return temperatures for the system were high (and the airflows for one fan coil were 20% below specifications – see Appendix D) and this partially explains lower than expected COPs. Furthermore, the data is not aggregated according to outdoor temperature, and this also has a significant impact on COP.

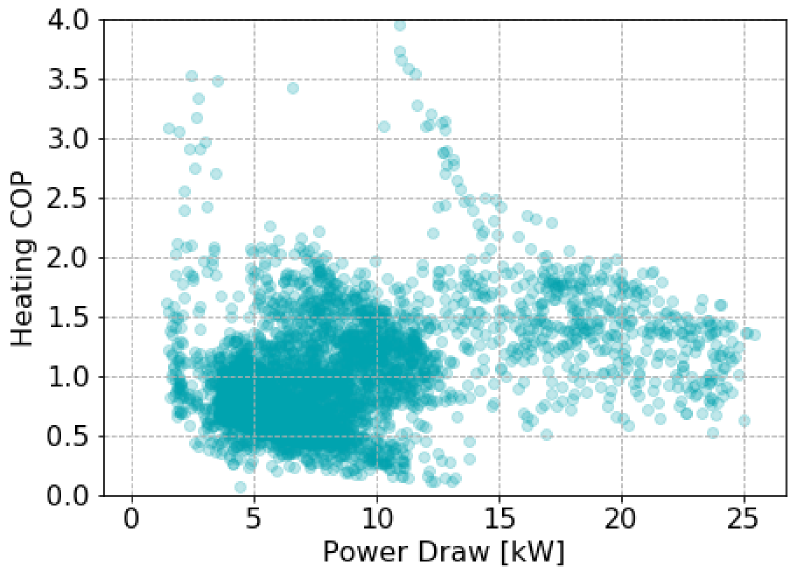


Figure 9-14. The 5-minute aggregate COP plotted against the electrical power draw for Winter 2020/2021.

Figure 9-15 shows a time-series plot of the outdoor temperature and the heat pump air inlet temperature. The difference is primarily due to natural gas tempering which appears to be on in temperatures as warm as 5 °C. The data is also shown as a frequency histogram in Figure 9-16.

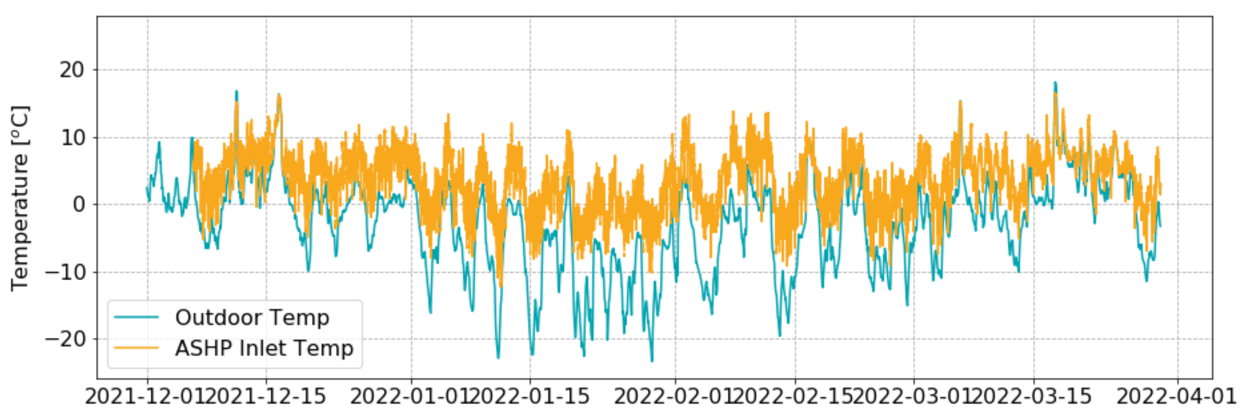


Figure 9-15. The outdoor air temperature is plotted as a time-series alongside the temperature of the air entering the outdoor unit. They diverge primarily due to the natural gas unit heater which appears to be on for temperatures as warm as 5 °C.

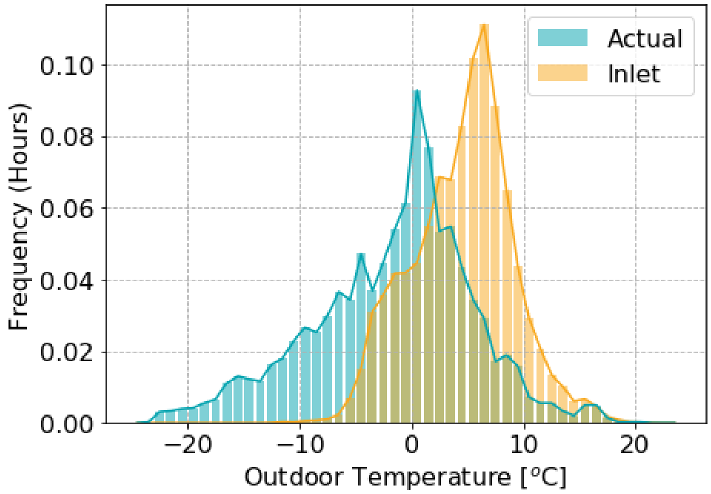


Figure 9-16. Frequency histograms of the outdoor temperature and outdoor unit inlet temperature show the impact of the natural gas tempering in the outdoor enclosure.

10.0 DETAILED MONITORING: COOLING RESULTS YEAR 2

Supply and return temperature are plotted for Summer 2021 in Figure 10-1 and Figure 10-2. The fan coils in the Student Assembly are no longer operating against each other.

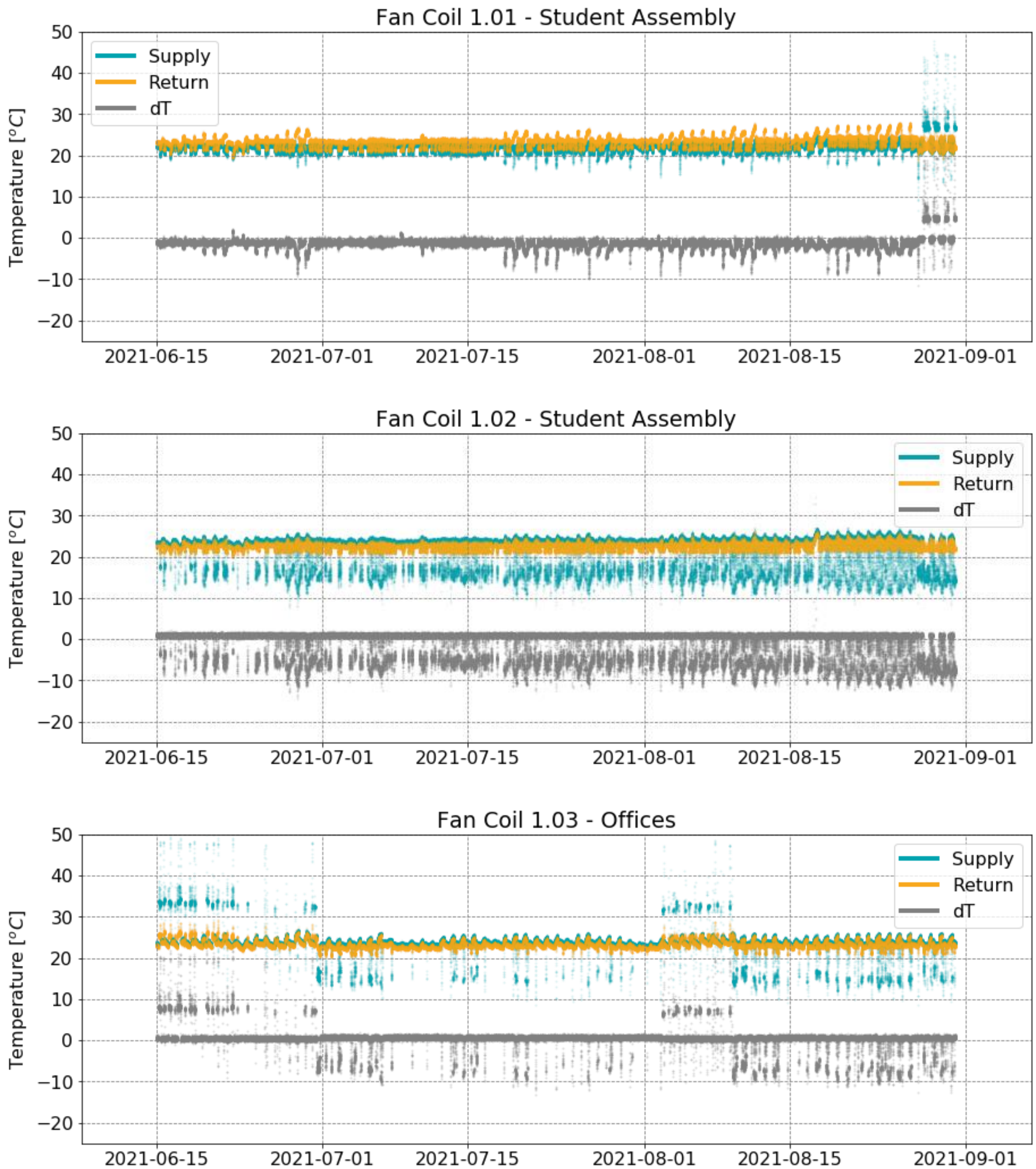


Figure 10-1. Supply and return air temperatures are plotted for Summer 2021. Fan Coil 1.01 is off nearly all the time. Fan Coil 1.03 is off most of the time. Fan Coil 1.02 is regularly providing cooling. Fan Coil 1.03 occasionally provides heating (potentially recovering heat for Fan Coil 1.02).

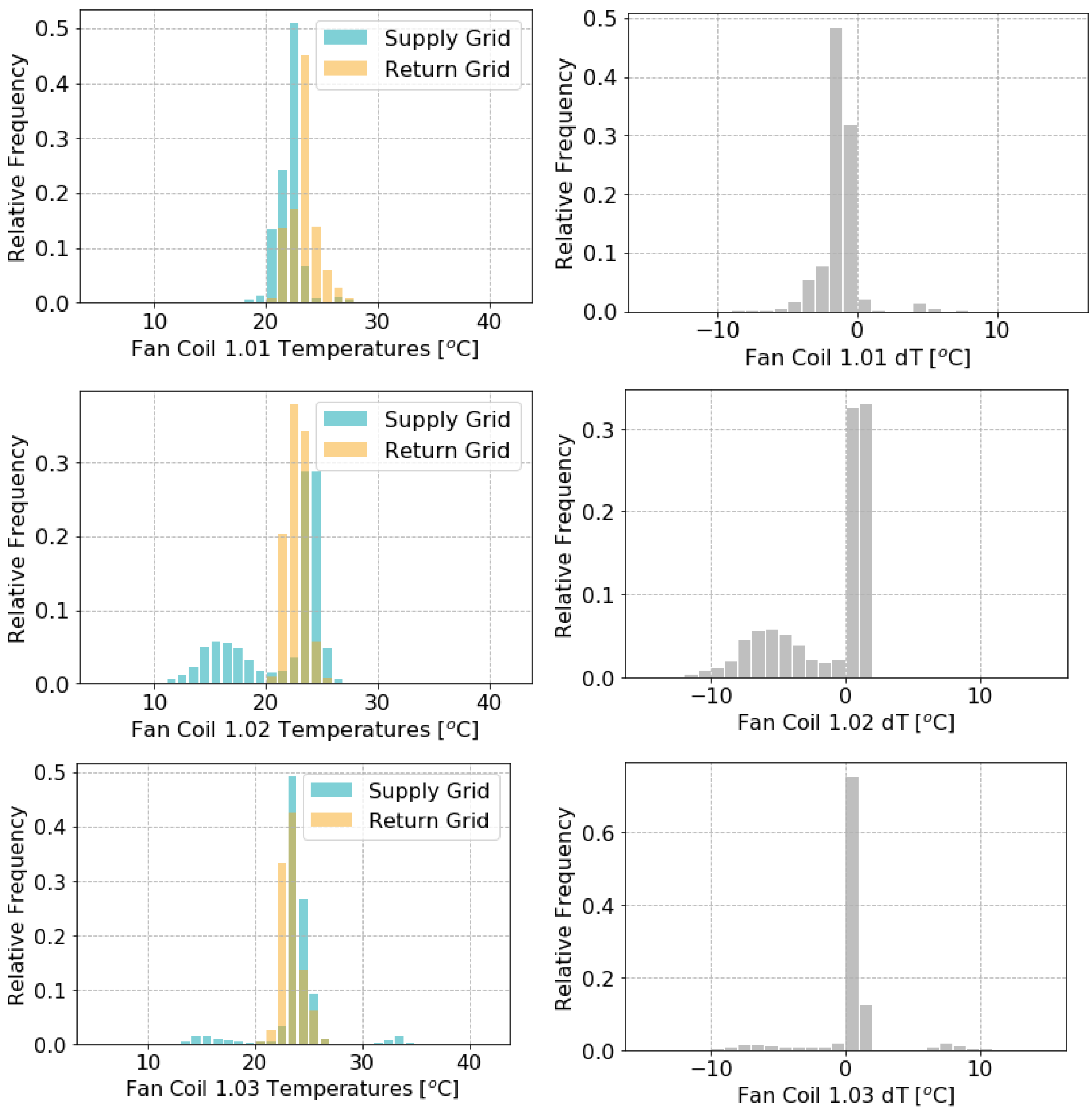


Figure 10-2. Relative frequency histograms are plotted for fan coil supply temperature, return temperature, and temperature difference (dT) for Summer 2021. Fan Coil 1.02 is the only fan coil providing substantial cooling.

Zone temperatures at thermostat level are typically between 22 and 24 °C, as shown in Figure 10-3. Figure 10-4 shows the outdoor temperatures both as a time-series and frequency histogram. Summer 2021 was recorded to be warmer than average. Figure 10-5 and 10-6 show the airflow data. Fan Coil 1.01 is typically off, Fan Coil 1.02 is on at a single speed for most of the summer, and Fan Coil 1.03 changes between two fan speeds. Capacity data in Figure 10-7 shows that only a small amount of cooling was provided.

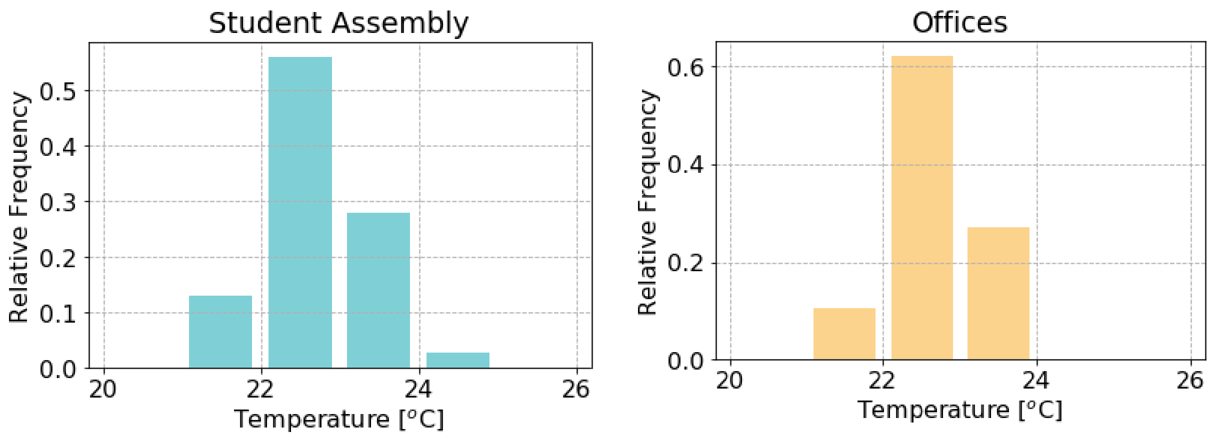


Figure 10-3. Zone temperatures during Summer 2021 measured by the monitoring system with sensors located at chest level in the zones near NAVs.

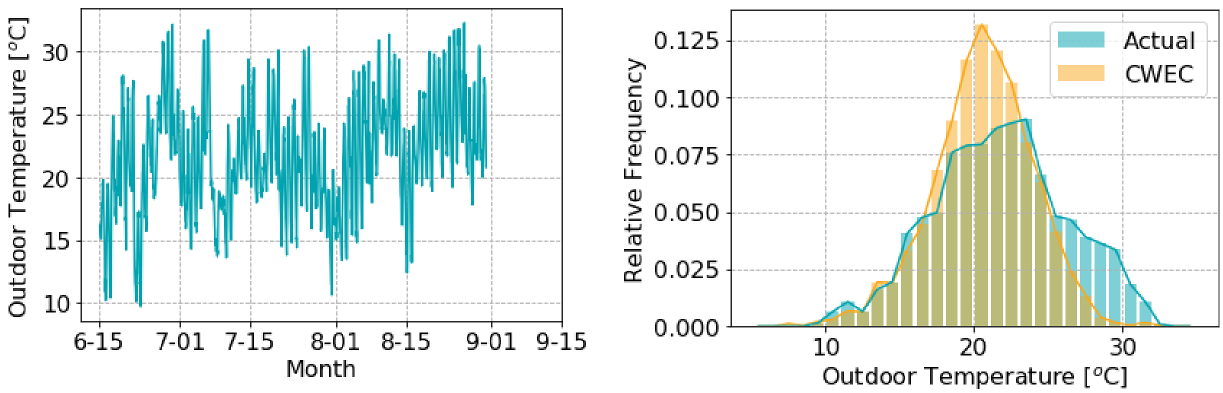


Figure 10-4. Outdoor temperature data during Summer is shown both as a time-series and as a frequency histogram. Summer 2021 was warmer than usual based on comparison against the CWEC data.

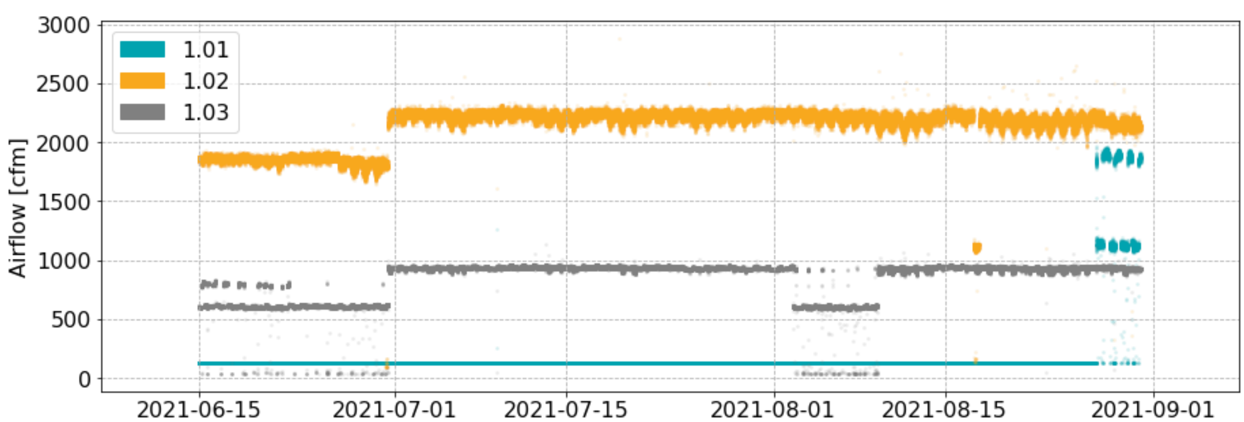


Figure 10-5. Airflow provided by each fan coil during Summer 2021. Fan coil 1.01 is off and Fan Coil 1.02 is nearly always at its highest speed.

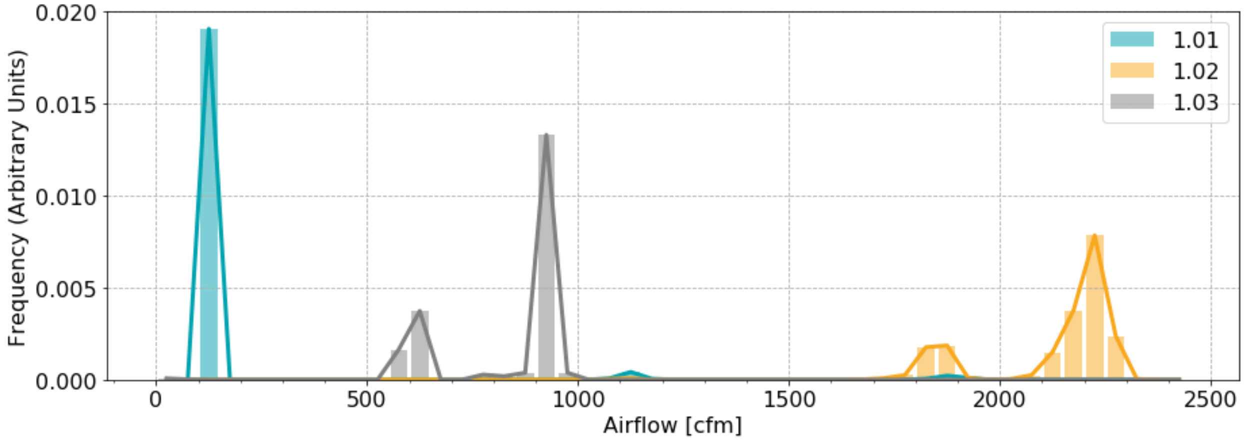


Figure 10-6. A frequency histogram showing the relative fraction of time that each indoor fan coil is providing different airflows during Summer 2021. Fan coil 1.01 is off and Fan Coil 1.02 is nearly always at its highest speed.

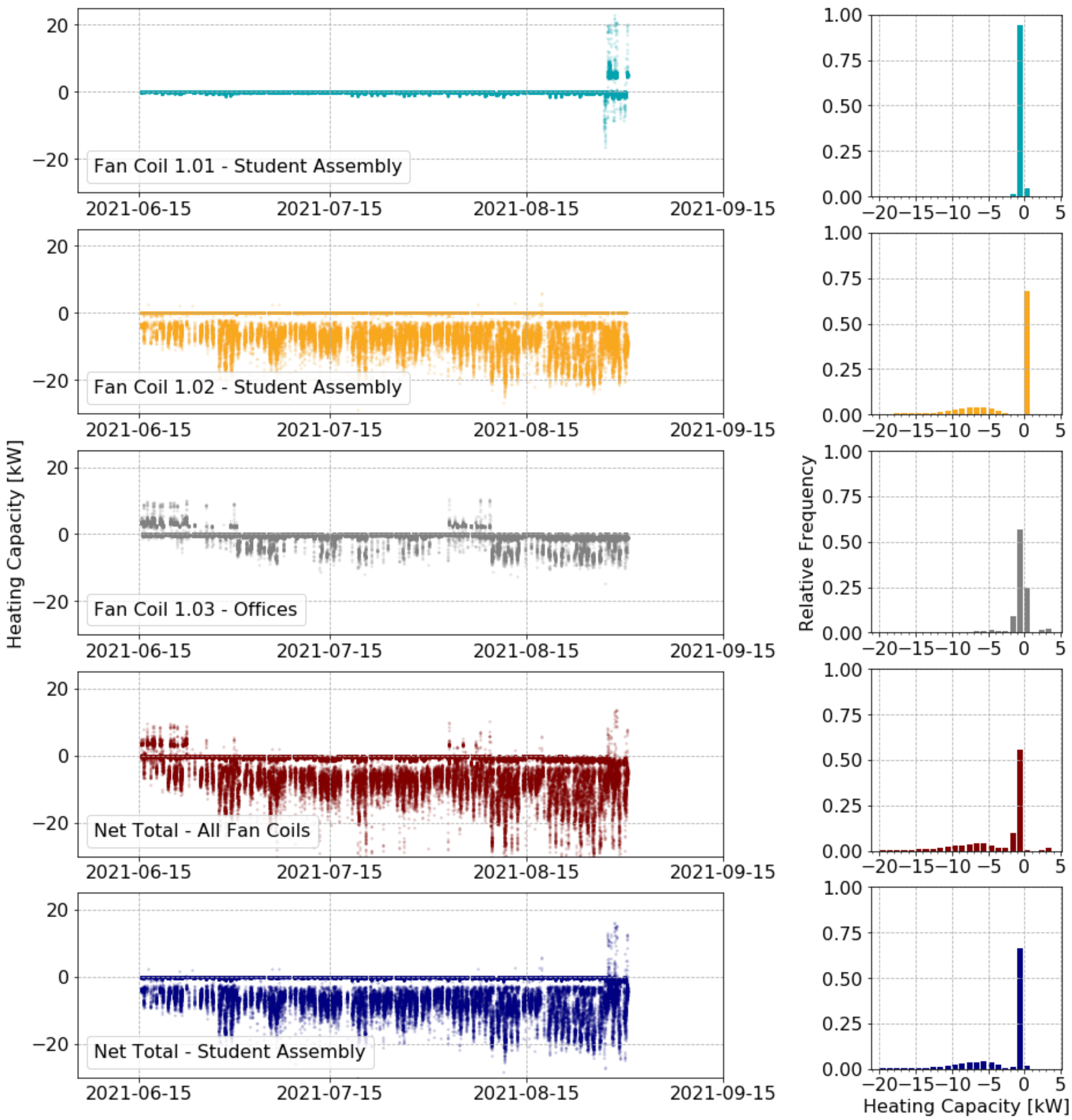


Figure 10-7. Cooling capacity (shown as negative heating capacity) is shown for Summer 2021.

Cooling capacity is plotted against electrical power draw in Figure 10-8. The heating energy removed greatly exceeds the electrical energy consumed, and the system is showing operation closer to expectations in cooling mode. Daily aggregated cooling capacity and electrical energy consumption is shown in Figure 10-9. Again, heat removed is much greater than electricity consumed.

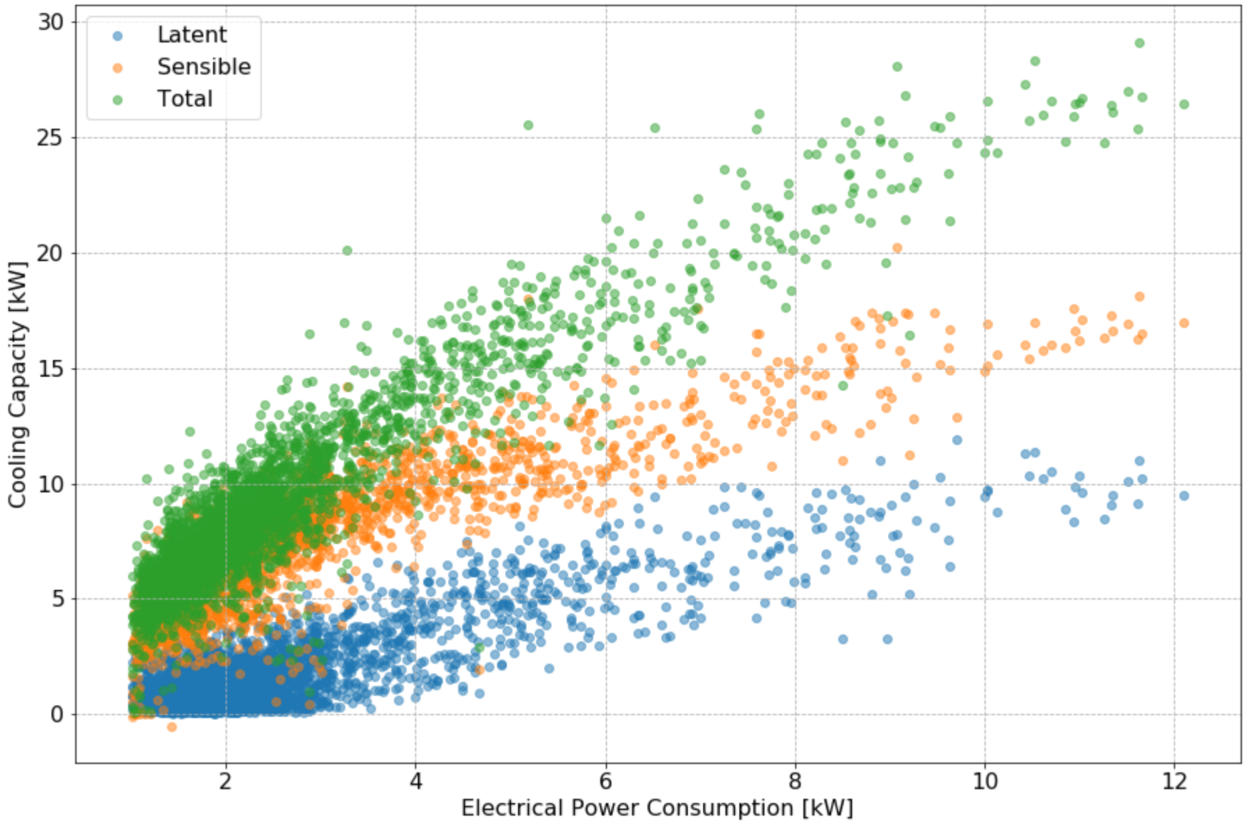


Figure 10-8. The daily-aggregated heating energy provided by the heat pump system is plotted against the electrical energy consumption for Summer 2021.

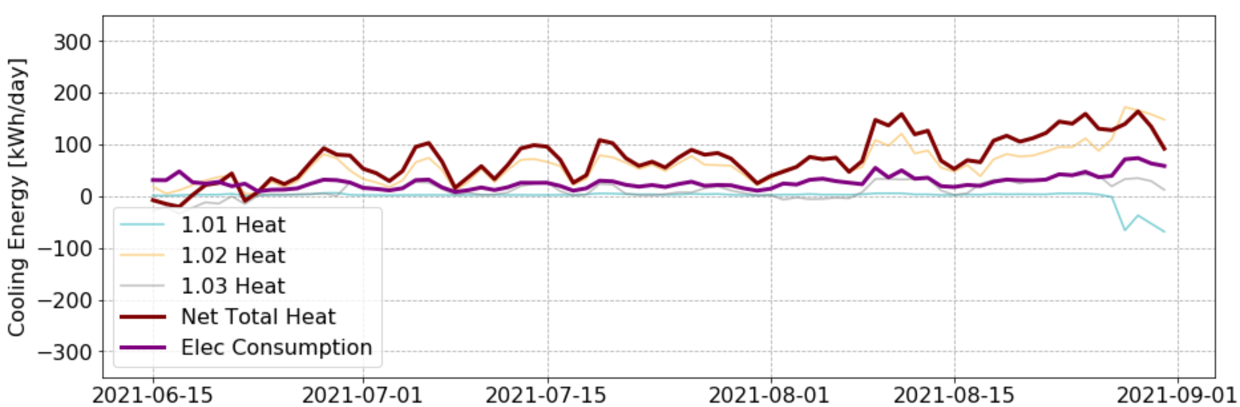


Figure 10-9. The daily-aggregated heating energy provided by the heat pump system and is plotted electrical energy consumption for Summer 2021. The cooling provided greatly exceeds the electricity consumption.

COP is plotted as a function of electrical power draw in Figure 10-10. The data is aggregated in 5-minute intervals and only includes data points where all fan coils were off or in cooling mode. Interestingly, the opposite trend from heating mode was observed. The cooling COP is much larger at low fractions of the maximum cooling capacity where the system typically operated, yielding a good COP when aggregated daily. The cooling COP was 2 to 3 at greater fraction of rated load. At first glance, it appears to be lower than expected but the system rarely operates in that range; it primarily operates a fraction of the rated capacity. The cooling efficiency will be lower for hotter ambient outdoor temperatures. It should also be noted that there is inherent error in determining latent heating capacity using relative humidity meters, so the error bars on the cooling capacity and efficiency calculations would be notable.

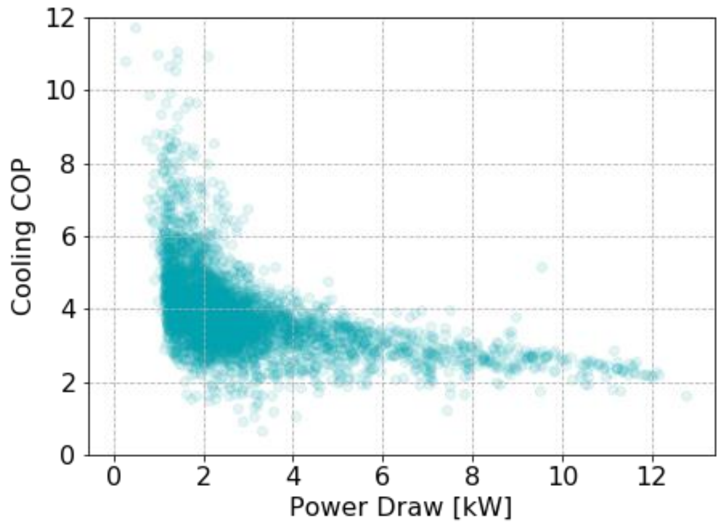


Figure 10-10. The daily-aggregated heating energy provided by the heat pump system and electrical energy consumption for Summer 2021.

Example system data for one day is shown in Figure 10-11. Negative capacity represents heat that was removed from the space (i.e. cooling capacity). By comparing the first two plots, when the power draw is lower, the cooling COP tends to be higher. Return temperatures tend to be 23 to 25 °C. In cooling mode, this is beneficial for COP and capacity since heat transfer will be greater given a greater temperature difference between the coil and return air.

Airflows are on at a constant rate. They are likely on when the fan coils are off to distribute the ventilation air. However, there may be a control issue in which the fan speed should reduce to a lower value when the unit is off (but not turn off entirely if ventilation is required). The heat rejected to the outdoor air is apparent from the temperature change across the outdoor coil, which increases whenever the system is on. On this particular day, the outdoor temperature approached the low-to-mid 30°C.

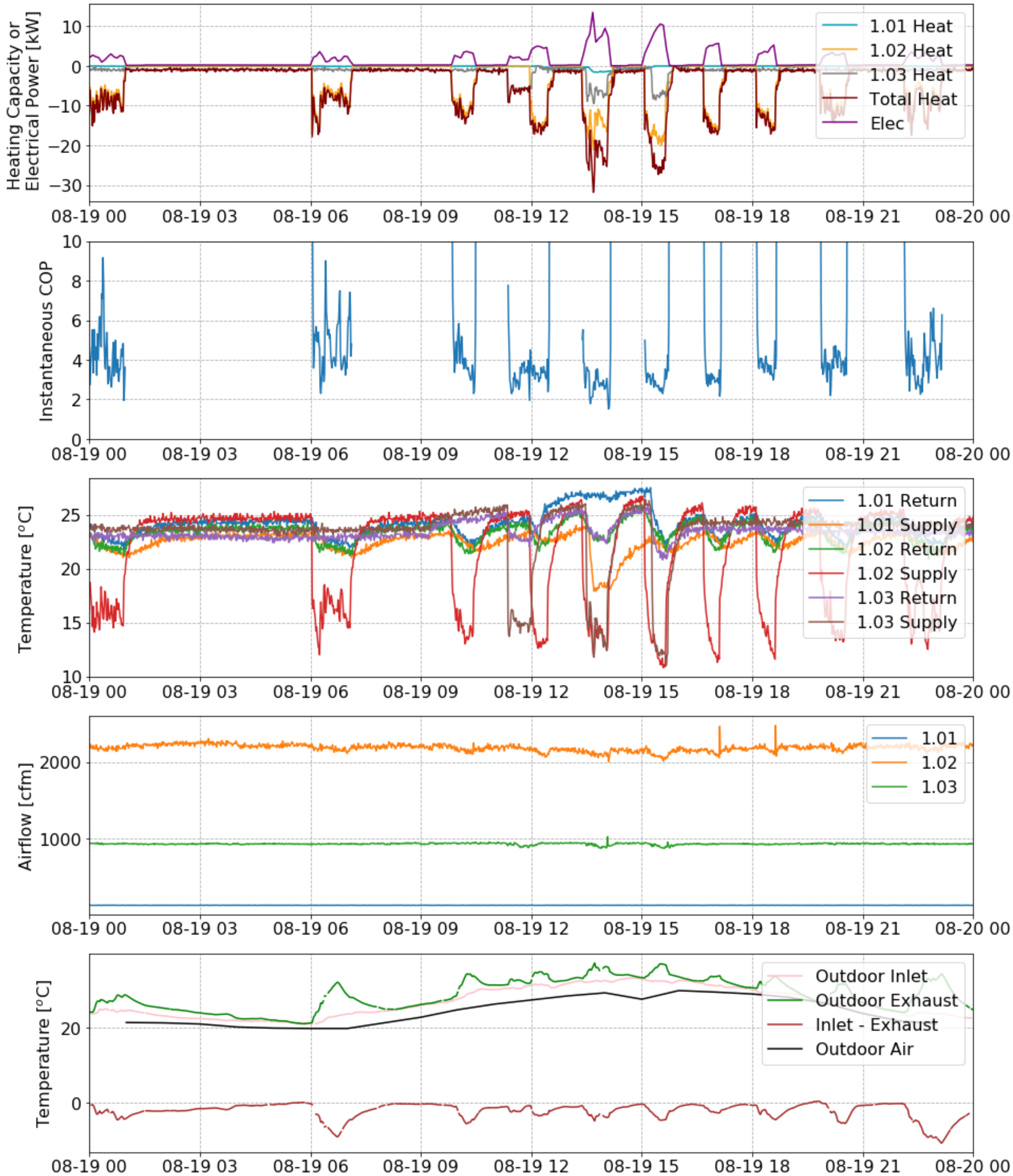


Figure 10-11. Example system operational data for August 19th, 2021.

11.0 DETAILED MONITORING: SUMMARY

Detailed monitoring for Year 2 shows that some of the major issues from Year 1 were resolved. Fan Coil 1.01 and 1.02 are no longer operating against each other, and the ASHP no longer appears to be operating against the heating provided through the tempered ventilation air (i.e. it is not providing net cooling throughout the winter).

Some issues remained. The ASHP in the monitored subsystem was still drastically oversized and only a fraction of its rated capacity was ever needed. The ventilation schedule that was implemented reduced the internal heat gains of the Student Assembly to the point where it no longer required cooling, but also hardly required active heating. The high set-point for tempered air in the outdoor enclosure was not adjusted due to the condensate management problem.

Year 2 also allowed for much greater characterization of the COP. The cooling mode data appeared closer to expectations. Interestingly, the performance trend appeared reversed to that observed in heating mode. The best performance was observed at lower fractions of maximum capacity, and as capacity increased it appeared to plateau between a cooling COP of 2 and 3. The plateau occurs partly because greater capacity is required for hotter ambient conditions, and efficiency decreases as outdoor temperature increases. This subsystem was nearly always operating at a low fraction of the load in cooling mode where better performance was observed.

In heating mode, the heat pump was also almost always operating a low fraction of its maximum capacity, and this resulted in relatively low COPs. The research team forced the system to operate at greater heating capacities for temporary periods by manually increasing the heating load of the zone (opening doors in the winter). This allowed the team to trace out performance closer to the rated power draw.

With close-to-maximum loading, the COP reached on the scale of 2.0 but this was still lower than expected. There are likely two important factors to take into consideration: (i) return temperatures tended to be much higher than those assumed in the equipment rating procedures and (ii) the airflow from Fan Coil 1.03 was 20% below its rated values. This may explain much of the deviation from expected performance. The return temperature merits special consideration in future modeling exercises. Ceilings are high and the return grills are located at ceiling level. There is likely significant thermal stratification within the rooms. A model which did not take this into account would over-predict the energy savings.

The system has no faults, and the system pressures and temperatures were reviewed by an HVAC service contractor in May 2022, also indicating no problems. Overall, due to the equipment sizing, this zone was not a good representative candidate for detailed performance monitoring but the research team could not have reasonably foreseen this in the study design.

12.0 MEASUREMENT & VERIFICATION

12.1 Introduction to M&V

A key issue confronting any M&V is that energy savings must be determined by comparing something that *did* happen (the energy consumption post-retrofit) with something that *would have* happened (energy consumption that would have occurred had there been no retrofit) – and only the former is directly measurable. It is not as simple as comparing a year of post-retrofit data against a year of pre-retrofit because utility consumption may have changed for reasons other than the retrofit; for example, due to the differences in weather, building usage, new or decommissioned equipment aside from the retrofit, and other factors.

The measurement & verification (M&V) plan was adherent with the International Performance Measurement & Verification Protocol (IPMVP) Option C. IPMVP Option C looks at the utility consumption of the entire facility, in this case, using electricity and natural gas utility bills. In IPMVP Option C, a pre-retrofit (baseline) linear regression model of the facility is created to predict the energy consumption of the facility using one or more independent variables prior to when an energy conservation measure (ECM) was implemented. In this M&V, the independent variable for the natural gas consumption, electricity consumption, and electricity demand was heating degree-days (HDDs). The baseline models were suitable without requiring any other independent variables (like occupancy or CDDs) to explain variations in utility consumption.

The baseline models are then used to predict the baseline utility consumption under post-retrofit (also called reporting period) conditions. This is an estimate of "what would have happened" post-retrofit had there been no ECM. The modeled baseline (also called the adjusted baseline) can then be compared against the actual utility consumption that occurred and it would be a fair "apples-to-apples" comparison considering the important factors that influence utility consumption. IPMVP guidelines also provide additional mathematical analysis that estimates the accuracy of the savings calculations. Data, models, and additional information are provided in Appendix A.

IPMVP Option C makes sense when ECMs are large enough to make a notable impact on the consumption reported in the utility bills (savings greater than 10%, for example). In many cases, Option C is the only M&V option because once the decision is made to undergo a retrofit, building owners may not want to wait extra time to collect better baseline data using submeters or other methods. The only data for baseline energy consumption would most commonly be the historical utility bills.

BCPV has many electricity and natural gas loads in outbuildings throughout the facility aside from the Visitor Centre, and the utility bills account for the whole facility. During the retrofit, electricity and natural gas submeters were installed at the Visitor Centre but there were no other submeters on the property, so the fraction of utilities consumed *pre-retrofit* by the Visitor Centre was not known precisely. It was estimated by facility staff that 35% of the natural gas load was the Visitor Centre, and 65% was the rest of the property. If the heat pump system achieved the savings estimated by the consultant that performed the feasibility assessment (83%), then the reduction on the utility bill would be 28%. This would be enough to evaluate energy savings using Option C, making it a viable method.

12.2 Non-Routine Adjustments

When using Option C, there may be energy-influencing factors that change after baseline data has been collected that are not accounted for in the baseline model. IPMVP terms the process of accounting for these factors as "non-routine adjustments." In the case of the BCPV facility, these factors may include:

1. conversions of outbuildings from electric resistance heat to high-efficiency gas equipment;
2. upgrades of outbuildings from low-efficiency to high-efficiency gas heating equipment;
3. changes in the temperature set-points of some outbuildings (due to COVID-19) to reduce energy consumption;
4. changes in the lighting schedule of the Visitor Centre and outbuildings because of low-occupancy (due to COVID-19);
5. no natural gas load from the kitchen facilities (due to COVID-19);
6. general lack of occupancy of the facility (due to COVID-19), reducing internal gains while also introducing other factors like significant changes in the frequency in which exterior doors are opened and closed.

The largest impacts on the energy consumption of the facility are from (1). The next largest is likely (2). Both factors were considered in the M&V. Factors (3) to (6) are related to COVID-19. For (3), set-points did not change significantly and this is estimated to be relatively small. Similarly, baseline data for the summer months showed that (5) is small. The analysis did not consider (4) or (6).

Table 12-1 describes the changes to the outbuildings at the facility after the baseline data was collected. To account for these changes, additional sensors were deployed in the outbuildings. Monnit ALTA temperature sensors were installed on the supply of the new gas equipment. The temperature sensors were used as an indicator to determine when equipment was on or off. This approach would not work with variable capacity equipment, but the new equipment was either single-stage or two-stage. Temperature monitoring was sufficient to determine when the equipment was on and whether it was high- or low-stage. This data was then used with gas input data from manufacturer specifications to estimate the gas consumption.

The installed sensor system began collecting monitoring data from the converted outbuildings in late November 2021. Sensor data collected from November to April 2022 along with manufacturer specifications were used to estimate gas consumption for each outbuilding for the heating season. Linear regression models were then derived for each outbuilding. They related outdoor daily mean temperature and the calculated daily gas consumption. The models were then used to estimate gas consumption for days in the heating season where no temperature monitoring data was available (including dates prior to November 2021). Further details are provided in Appendix A.

This process produced daily gas consumption estimates for each outbuilding from April 2021 to March 2022. This was subsequently converted to a monthly basis, coinciding with the utility billing periods. For natural gas, the estimated total monthly gas consumption for the converted outbuildings was *subtracted* from the actual monthly gas consumption on the utility bill. Similarly, the gas savings resulting from equipment efficiency upgrades in some outbuildings (Burwick Building, Church Building, and Second House) was *added* to the monthly actual gas consumption from the utility bill.

Table 12-1. Changes to equipment and facility after the baseline data was collected.

Date	Description of Change	Description of New Equipment
December 2018	Laskay's building converted from electrical heat to gas furnace	Make: ArcoAire Output: 80,000 BTU/hr Combustion Stages: 2 Stage Efficiency: 96%
January 2019	Bolton shop building converted from electric heat to gas furnace	Make: ArcoAire Output: 40,000 BTU/hr Combustion Stages: 1 stage Efficiency: 92%
September 2019	Flynn house converted from electric heat to gas furnace	Make: ArcoAire Output: 60,000 BTU/hr Combustion Stages: 1 Stage Efficiency: 92%
September 2019	Sam's workshop building converted from electric boiler to gas furnace	Make: ArcoAire Output: 40,000 BTU/hr Combustion Stages: 1 Stage Efficiency: 96%
October 2019	Burwick building upgraded old gas boiler to new high-efficiency gas boiler	Make: NTI Trinity Output: 108,000 BTU/hr Combustion Stages: N/A Efficiency: 95%
February 2020	Mckenzie house converted from electric heat to gas furnace	Make: ArcoAire Output: 60,000 BTU/hr Combustion Stages: 1 Stage Efficiency: 96%
April 2020	Second house upgraded from old gas furnace to a new high-efficiency gas furnace	Make: ArcoAire Output: 80,000 BTU/hr Combustion Stages: 2 Stages Efficiency: 96%
April 2020	Church building upgraded from old gas furnace to new high-efficiency gas furnace	Make: Maratherm Output: 100,000 BTU/hr Combustion Stages: 1 Stage Efficiency: 96%
March 2020	March COVID-19 shut down, gas in VC kitchen turned off, all lighting and heat turned down to minimum use (68-72 ^{o}F) in Visitor Centre and Village	-
April 2021	School house converted from electric heat to gas furnace	Make: ArcoAire Output: 100,000 BTU/hr Combustion stages: 2 Stages Efficiency: 96%

Recall that the aim of the non-routine adjustment was to allow for a fair comparison with a baseline, and since these changes happened after the baseline was determined, they needed to be considered for a fair comparison. In other words, by applying the non-routine adjustment to the actual post-retrofit utility bills, it was possible to estimate what the utility consumption *would have occurred* if the changes listed in Table 12-1 had not taken place.

The post-retrofit electricity consumption needed to be corrected as well. For buildings where an electric-to-gas conversion took place, the electricity saved from the conversion was equivalent to the energy content of the estimated gas consumption (with a small adjustment for the efficiency of the gas equipment). This electricity savings from outbuilding conversions was also determined on a monthly basis coinciding with the electricity billing periods – then it was *added* onto the actual electricity consumption. Again, it is an estimate of the electricity consumption that *would have* occurred had there been no conversions.

Total annual savings from the VRF ASHP retrofit for gas or electricity was then calculated separately as in Equation (1). The “Adjusted Baseline Consumption” is the utility consumption determined by applying the baseline linear regression model of utility consumption (determined from pre-retrofit utility data) to the post-retrofit HDDs. The “Actual Consumption Post-retrofit” is consumption data obtained directly from the post-retrofit utility bills. The “Non-Routine Adjustment” is a correction applied for the outbuilding conversions described above. The gas consumption of electric-to-gas conversions is *subtracted*, the gas consumption savings from the increases in efficiencies is *added*, and the electricity savings from the electric-to-gas conversions is *added*.

$$Gas\ or\ Electricity\ Savings = (Adjusted\ Baseline\ Consumption) - (Actual\ Consumption\ Post - retrofit) \pm (Non - routine\ Adjustment) \tag{1}$$

The non-routine adjustments allowed the M&V to focus on the impact of the Visitor Centre ASHP system retrofit alone, rather than the collective impact of the ASHP retrofit and the electric-to-gas conversions on the property. Further details are provided in Appendix A.

12.3 Utility Data Issues

Post-retrofit utility data was reviewed starting in November 2019 up until March 2022. However, the natural gas meter from Enbridge seized starting in July 2020 and was not replaced until February 2021. The gas consumption provided on the utility bills from this period were estimated from the utility and therefore not useful for the M&V. The M&V therefore focused only one full year encompassing March 2021 to February 2022.

12.4 Natural Gas Consumption

Baseline (pre-retrofit) natural gas consumption data for the facility was collected for 2015 to 2017 from the utility bills. Post-retrofit utility bill data was collected from March 2021 to February 2022. The data is provided in Appendix A. Figure 12-1 plots the pre-retrofit and post-retrofit natural gas consumption data as a time series.

Figure 12-2 normalizes the data for weather in terms of HDDs. HDDs were obtained from Environment Canada weather data for Toronto City Centre accessed via weatherstats.ca. A building balance point of 18°C was assumed. "Actual Pre-retrofit" and "Actual Post-retrofit" is the utility taken directly from the bills. "Post-retrofit with Non-Routine" includes the non-routine adjustment discussed in Section 12.2 Savings are more distinct when the non-routine adjustment is applied. Savings do not appear to be large but recall that the Visitor Centre is a smaller portion of the bill for the full facility so large savings for the Visitor Centre would still translate to small *relative* savings for the full facility.

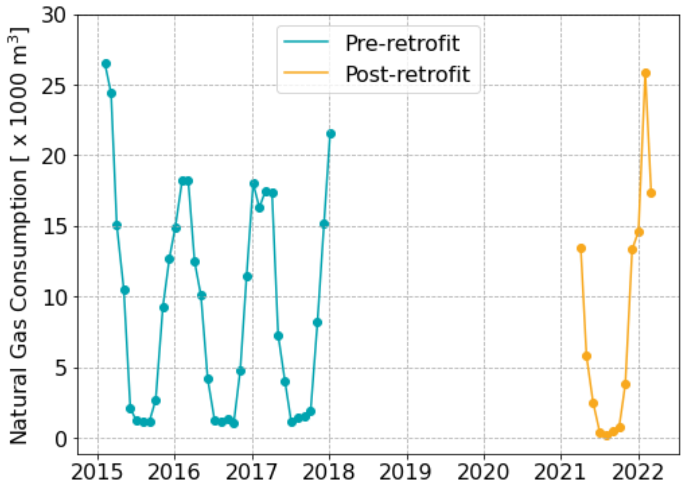


Figure 12-1. (Left) The natural gas consumption pre- and post-retrofit plotted as a time-series.

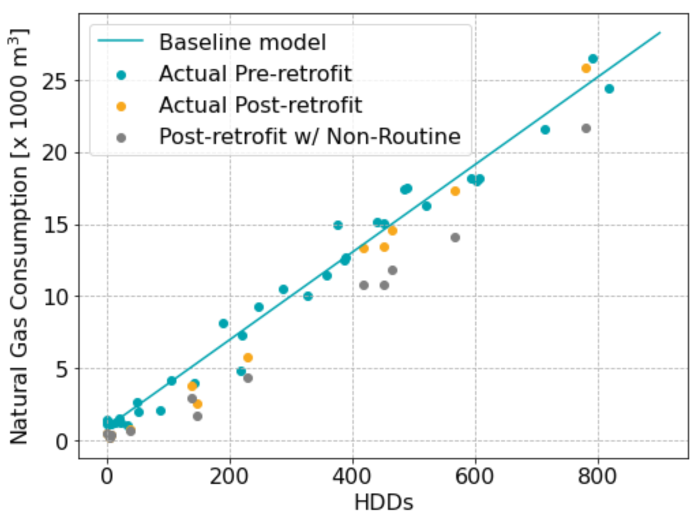


Figure 12-2. The natural gas consumption data pre-retrofit and post-retrofit show no significant change for the full facility (Visitor Centre *and* outbuildings). This is because the ASHP is reducing the natural gas consumption of the Visitor Centre while the electric-to-gas conversions of the outbuildings is simultaneously increasing it. Savings is clearer when the outbuilding conversions are accounted for within the non-routine adjustment.

Monthly gas data is plotted in Figure 12-3. The "Adjusted Baseline" is the result of the baseline model (from pre-retrofit utility data) applied to post-retrofit HDDs. "Actual Post-retrofit" is the actual gas consumed determined from the utility bill. "Actual Post-retrofit w/ Non-routine Adjustment" includes a correction for the equipment efficiency upgrades in the outbuildings and the electric-to-gas conversions. The utility data with the non-routine adjustment can be fairly compared against the “Adjusted Baseline” to determine the impacts of the Visitor Centre ASHP retrofit. “Savings” is the “Actual Post-retrofit w/ Non-routine Adjustment” subtracted from the “Adjusted Baseline”.

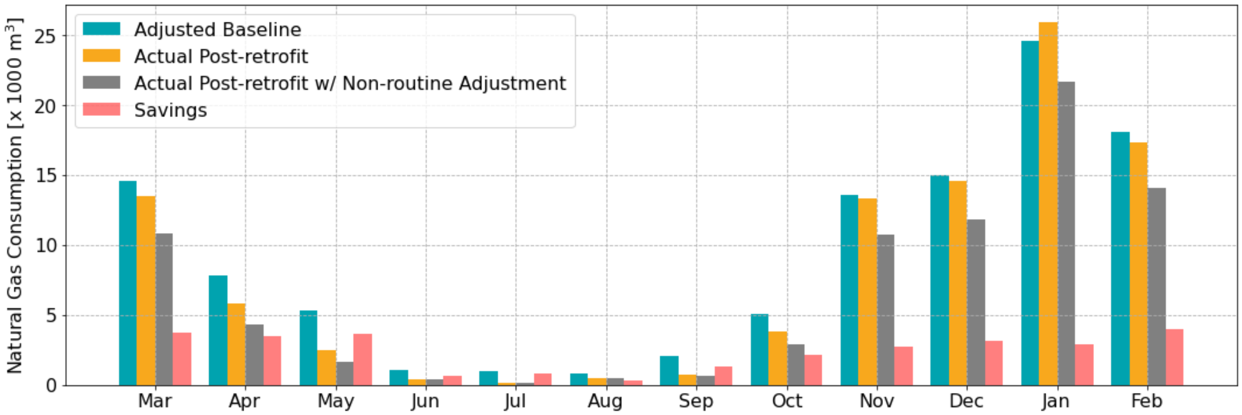


Figure 12-3. Natural gas savings resulting from the BCPV ASHP retrofit determined for one year of operation.

Concurrent with the ASHP retrofit, a natural gas submeter was installed at the facility to monitor the gas consumption of the Visitor Centre. The data was incomplete for March and April 2021. To fill in the data gaps, a linear regression model of the Visitor Centre gas consumption was created (Figure 12-4). The Visitor Centre gas consumption is plotted alongside the full facility gas consumption in Figure 12-5. The Visitor Centre was 23% of the full facility gas consumption for this period. Table 12-2 provides natural gas consumption totals for the one-year period.

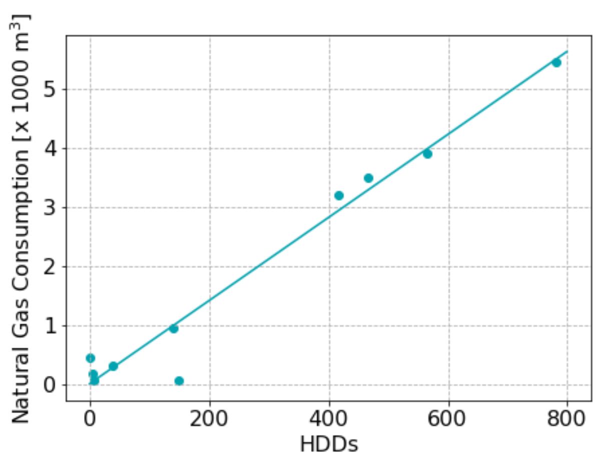


Figure 12-4. Natural gas savings resulting from the BCPV ASHP retrofit determined for one year of operation.

The estimated gas savings is 29,667 m^3 and is fully attributed to the ASHP retrofit of the Visitor Centre. The actual Visitor Centre gas consumption was 23,052 m^3.[4] This means relative gas savings for the Visitor Centre was estimated to be 56%.

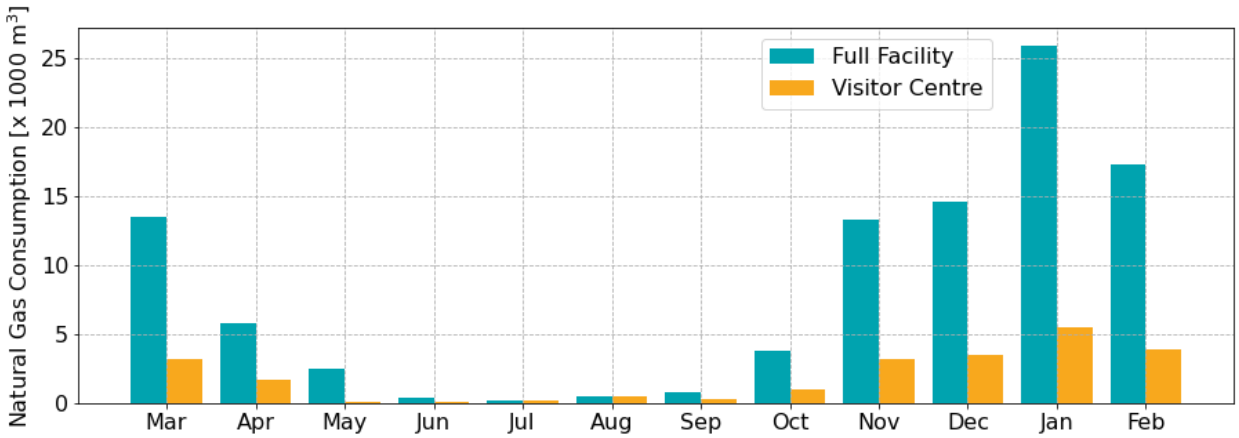


Figure 12-5. A natural gas submeter determined that the Visitor Centre uses 23% of the gas consumption for the full facility from March 2021 to February 2022.

Table 12-2. Natural gas consumption or savings totals for March 2021 to February 2022.

	Parameter	Natural Gas [m^3]
1	Full Facility Adjusted Baseline Gas Consumption	109,355
2	Full Facility Actual Post-retrofit Gas Consumption	98,681
3	Full Facility Post-retrofit Gas Consumption with Non-routine Adjustment	79,688
4	(Adjusted Baseline) – (Actual Post-retrofit)	10,674
5	Savings: (Adjusted Baseline) – (Post-retrofit with Non-routine Adjustment)	29,667
6	Visitor Centre Gas Consumption	23,052

12.5 Electricity Consumption

Pre- and post-retrofit electrical consumption and demand data was gathered from the utility bills and is provided in Appendix A. The time-series pre-and post-retrofit consumption data is shown in Figure 12-6. Pre- and post-retrofit data normalized to HDDs is shown in Figure 12-7 alongside the baseline electricity consumption model. Also shown is utility consumption after the non-routine adjustment was applied.

[4] The savings can be added on to the actual consumption to determine what the Visitor Centre gas consumption would have been without the retrofit and this can then be used to determine relative savings, as in: 29,667/(29,667 + 23,052) = 0.56

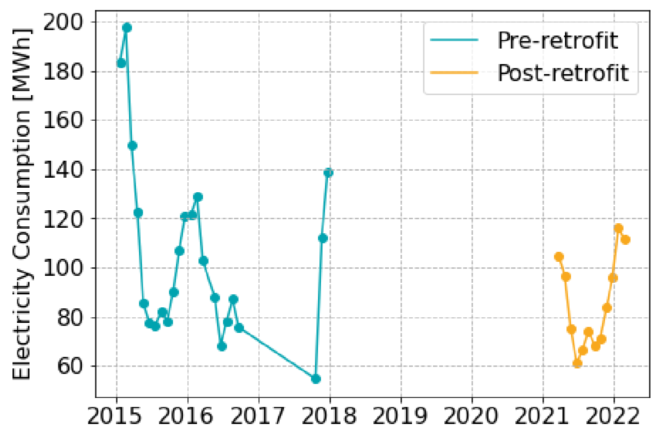


Figure 12-6. Electricity consumption is plotted as a time series pre- and post-retrofit.

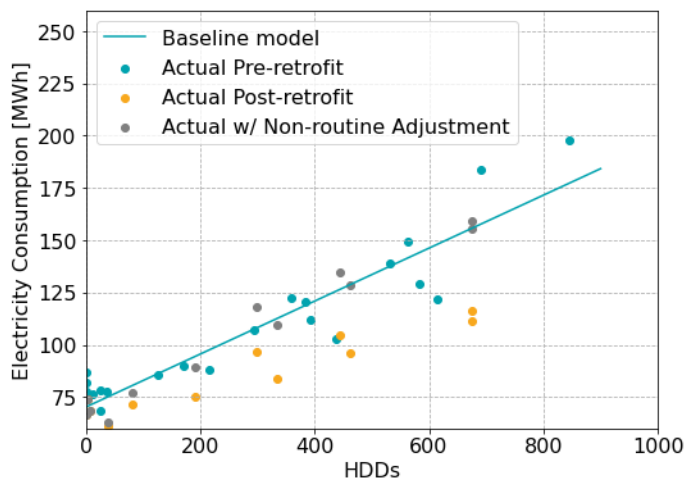


Figure 12-7. Weather normalized pre- and post-retrofit electricity consumption data is compared against post-retrofit data. Once the outbuilding conversions were considered via the non-routine adjustment, it is clear there was no measurable change in electricity consumption of the full facility (or the Visitor Centre).

A reduction in electricity consumption is apparent from simple inspection of the bills. However, the electricity reduction of the facility is entirely accounted for by the electricity-to-gas conversions of the outbuildings. This is clear because the “Actual Pre-retrofit” data aligns with the “Post-retrofit with Non-routine Adjustment” data. In other words, there was no discernable change in the electricity consumption of the full facility (and therefore the Visitor Centre as well) pre- and post-retrofit once the electric-to-gas conversions of the outbuildings were accounted for.

Monthly electricity consumption data is shown in Figure 12-8. The plot shows the “Adjusted Baseline,” the “Actual Post-retrofit” data, and the “Post-retrofit w/ Non-routine Adjustment” data. It is again evident that, once the electricity-to-gas conversions of the outbuildings is considered, the actual change in electricity consumption of the full facility (or the Visitor Centre) between the pre- and post-retrofit periods is negligibly small. Note the submeter data shows that the Visitor Centre constitutes approximately half of the electricity bill for the full facility. It follows that changes to the Visitor Centre electricity consumption would have a noticeable impact on the electricity consumption of the full facility.

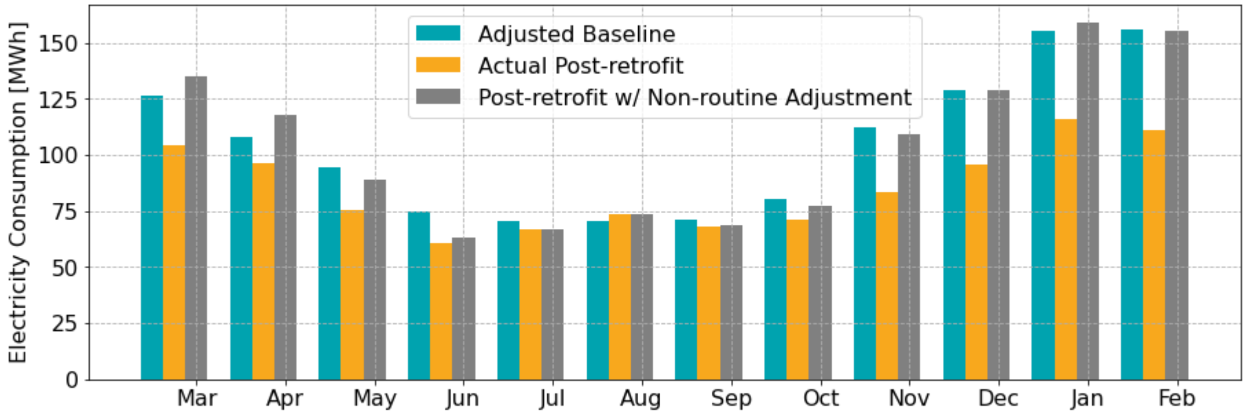


Figure 12-8. The “Adjusted Baseline” is the baseline model from Figure 12-7 applied to post-retrofit HDDs. The “Actual Post-retrofit” data is the consumption data taken directly from the bill. The “Post-retrofit w/ Non-routine Adjustment” data has been corrected for the electricity-to-gas conversions of the outbuildings – essentially the electricity savings from the conversions has been estimated and added back on to the “Actual” data.

Note there was a notable decrease in electrical consumption between 2020/2021 and 2021/2022 (for March to the following February in each year). This is believed to be due to two key system issues that were addressed: (i) the perimeter heating placed on an outdoor reset schedule rather than constant operation and (ii) the prevention of simultaneous heating and cooling occurring within the same zone. Both issues were seen in detailed monitoring from Year 1. In the first year, the ASHP was cooling the Student Assembly on average over the winter (due to overheating from the gas systems) and two fan coils serving Student Assembly where in different modes, one in heating and one in cooling. Both were broader issues within the building that were addressed early in the second year.

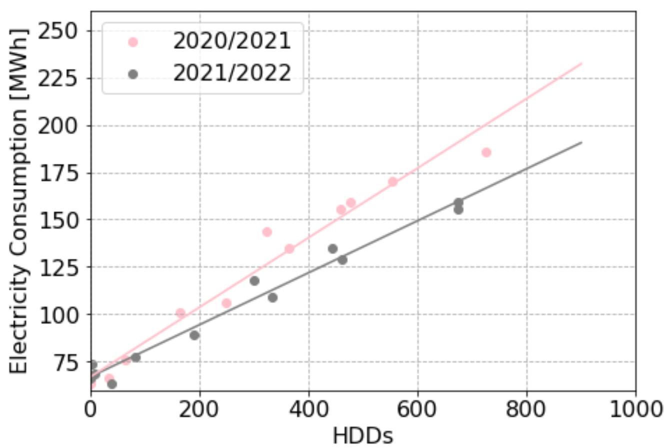


Figure 12-9. The electricity consumption notably reduced between 2020/2021 and 2021/2022. It is believed to be because the perimeter heating system was placed on outdoor reset control and the problem of simultaneous heating and cooling within the same zones was also addressed. The former issue resulted in overheating which forced the ASHP into cooling in some zones. The latter resulted in unnecessary operation of the ASHP. Both issues were addressed between the two years.

Electricity consumption and savings totals are provided in Table 12-3. Looking only at the weather normalized data (and not the non-routine adjustment), electricity of the facility reduced by an estimated 225 MWh (or 18%). However, once the non-routine adjustments for the electricity-to-gas conversions of the outbuildings was considered, electricity savings were negligibly small. In other words, the electricity consumption of the full facility was reduced but it was estimated to be entirely attributed to the outbuildings, with the ASHP system keeping electricity consumption of the Visitor Centre near the pre-retrofit levels.

Table 12-3. Electricity consumption and savings totals for March 2021 to February 2022.

	Parameter	Electricity [MWh]
1	Full Facility Adjusted Baseline Electricity Consumption	1,250
2	Full Facility Actual Post-retrofit Electricity Consumption	1,024
3	Full Facility Post-retrofit Electricity Consumption with Non-routine Adjustment	1,247
4	(Adjusted Baseline) – (Actual Post-retrofit)	225
5	Savings: (Adjusted Baseline) – (Post-retrofit with Non-routine Adjustment)	3

12.6 Electricity Demand

Electricity demand was evaluated similarly to electricity and natural gas consumption. Actual pre- and post-retrofit demand data is shown as a time-series in Figure 12-10. Data has been weather-normalized according to HDDs in Figure 12-11. Demand decreased post-retrofit, however, after applying the non-routine adjustment, it is clear that the decrease during heating months is largely explained by the outbuilding conversions. In June and July, the decrease in demand can be attributed to the ASHP retrofit. Monthly data is shown in Figure 12-12.

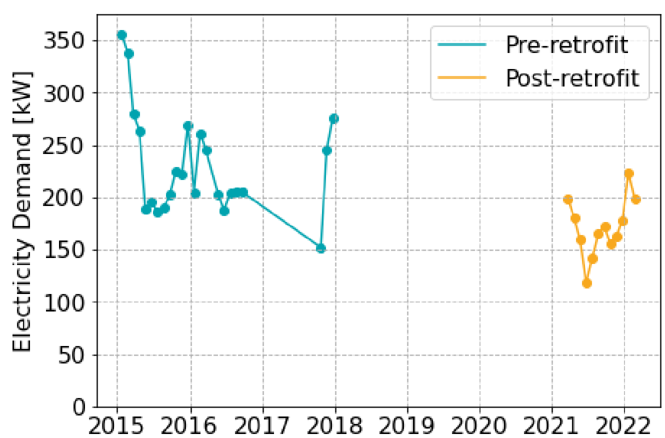


Figure 12-10. The actual pre- and post-retrofit electricity demand is plotted as a time-series.

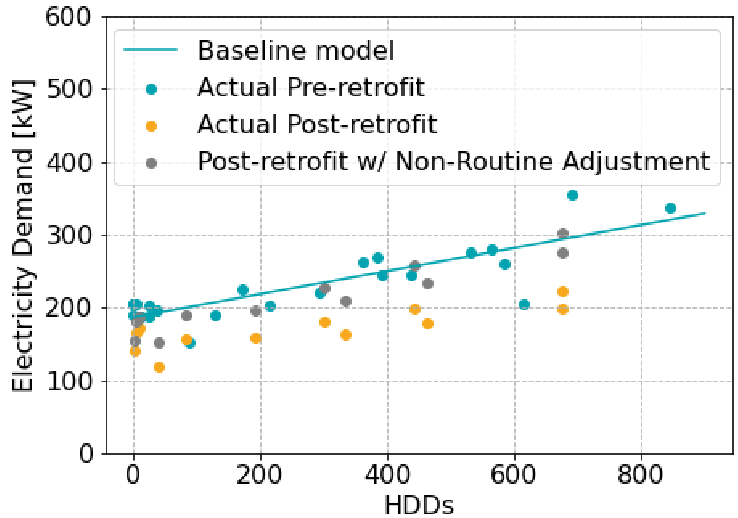


Figure 12-11. The actual pre- and post-retrofit demand from the utility bills was normalized according to HDDs and the pre-retrofit data was used to define a baseline electricity demand model. Demand decreased at the facility, but based on the non-routine adjustment, the decrease in demand for the heating months can almost fully be attributed to the conversions of the outbuildings.

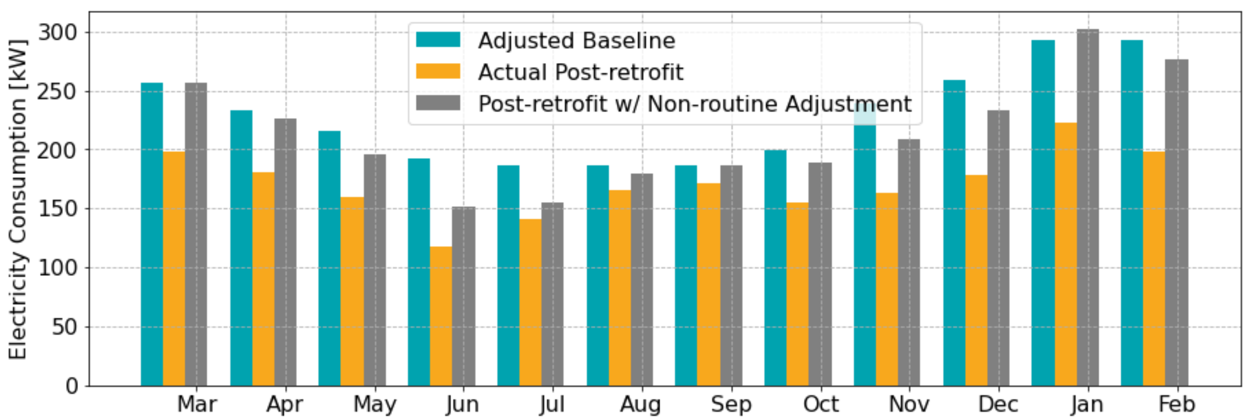


Figure 12-12. Demand data plotted on a monthly basis. The “Adjusted Baseline” is the model from Figure 12-11 applied to post-retrofit HDDs. “Actual Post-retrofit” is the demand data taken directly from the utility bills post-retrofit. “Post-retrofit w/ Non-routine Adjustment” has been corrected for the demand reductions from the outbuilding conversions.

Table 12-4. Electricity demand impacts from March 2022 to February 2021.

	(1) Adjusted Baseline Demand [kW]	(2) Actual Post-retrofit Demand [kW]	(3) Actual Post-retrofit Demand with Non-Routine Adjustment [kW]	(4) Actual Demand Reduction (1) – (2) [kW]	(5) Demand Reduction with Non-routine Adjustment [kW]
March	256	198	256	58	-1
April	233	181	227	52	6
May	216	159	196	57	20
June	192	118	152	74	40
July	186	141	154	45	32
August	187	165	180	22	7
September	187	172	187	15	0
October	199	155	188	44	11
November	239	163	209	76	30
December	259	178	233	81	26
January	293	223	302	70	-9
February	293	198	276	95	17

12.7 Annual Cost and Carbon Savings

The utility savings is expressed in terms of annual cost reductions in Table 12-5. Utility rates were derived from actual bills during the second year.

Table 12-5. Annual electricity/gas estimated energy savings and cost reductions.

Parameter	Energy	Energy Cost	Cost Reduction
Estimated Electricity Savings	3 MWh	91.5 $/MWh	$275
Estimate Demand Savings (Sum of Monthly Demand Savings)	179 kW	5.67 $/kW	$1,014
Estimated Natural Gas Savings	29,667 m^3	0.374 $/$m^3$	$11,095
		Total:	$12,384

Note that gas rates have risen substantially between Winter 2021/2022 and Winter 2022/2023, with the new gas rate at 0.569 $/m^3 as of October 2022. Applying the new gas rate yields a total annual gas savings of $16,881, and a total annual savings of $18,170. Assuming an emission factor of 1.89 CO_2e per m^3 of natural gas, the annual carbon savings is estimated at 56 T CO_2e.

12.8 M&V Summary

IPMVP Option C was used to estimate the utility savings associated with the ASHP retrofit. Without historical submeter data, the facility's utility bill data was used for this analysis. The utility bills encompassed the Visitor Centre and several outbuildings at the facility. A non-routine adjustment was required to account for numerous electric-to-gas conversions of outbuildings on the property that took place after the baseline model was developed.

Submeter data was used to determine the post-retrofit utility consumption of the Visitor Centre, and then the relative utility savings for the Visitor Centre alone. **The analysis estimated that the Visitor Centre electricity consumption did not have a measurable change pre- and post-retrofit, and the natural gas consumption was reduced by an estimated 29,667 m^3 (or 56% of the pre-retrofit Visitor Centre gas consumption)**. Additional details of the M&V are provided in Appendix A.

13.0 KEY IMPLEMENTATION CHALLENGES

13.1 Design and Construction

Two outdoor enclosures were initially designed to house the outdoor units. The main purpose of the enclosures was to supplement the ASHPs with natural gas heat in more extreme cold. This was intended as a cost-saving measure that would allow the system to use lower-cost "conventional" heat pumps instead of more advanced versions that would function better in extreme cold.

However, the outdoor enclosures introduced numerous cost increases, construction challenges, and bottlenecks for the installation. These included:

- challenges in building permits;
- complexity due to permitting and scheduling;
- siting of the enclosures;
- timelines of designing and constructing custom dampers;
- operation and integration of dampers, BAS points, heaters, etc.; and
- condensate management within the enclosures.

Some issues were site-specific, but others were related to the unique design considerations and requirements of the enclosures. It is still a relatively new concept to contractors, designers, and architects in the GTHA. One site-specific issue was the initial location of one of the enclosures. It was to be located near a delivery bay, but an existing food service contract required a minimum turning radius for large trucks that the initial planned location would interfere with. The selection and redesign of a new location introduced significant delays.

The initial design for footings of the enclosures exceeded what was required and they were redesigned, also introducing delays. The new location for the enclosure was too close to the existing building for fire code regulations and it needed to be relocated further back. The enclosures initially had a brick façade but it violated fire code and needed to be taken down to be replaced with a metal façade. There were also delays from the damper manufacturer to make custom dampers for the enclosures.

It follows that there was no single large issue with the enclosures but rather, several small issues that collectively amounted to significant delays and problems for the project management team. The "newness" of the enclosures used in these systems is therefore a barrier to successful implementation because it introduces a learning curve for vendors, consultants, and contractors.

Overall, the enclosures added significant unexpected expense and several months in delays for system completion. Furthermore, once completed, the natural gas unit heater was providing a greater amount of heating than anticipated due to the condensate management problem discussed in Section 8.0 . ASHPs that do not require outdoor enclosures are available and are strongly encouraged in subsequent retrofits. Discussions with system designers also indicated that the design community is moving away from the use of these enclosures.

13.2 Integration Between ASHP and Gas Heating Systems

One of the largest challenges in this retrofit was optimizing the integration of the ASHP system with the other subsystems that rely on natural gas. In this building, there were multiple pathways whereby the building could over-rely on natural gas:

- the perimeter heating system was initially set to full-on capacity for the heating season which offset the heating required from the ASHP system;
- the ventilation air was tempered by natural gas and, due to a suspected balancing issue, the tempered air alone was sufficient to heat the Student Assembly without even requiring the ASHP – pushing the ASHP to remove excess heat from the zone; and
- outdoor air was tempered by natural gas in the outdoor enclosure, and due to a condensate management problem, the set point for the enclosure was higher than was otherwise needed.

These issues are made more challenging by the fact that the systems may continue to maintain indoor temperatures at required set-points and overall show no faults or other obvious visual indications that there is an optimization issue. The research team helped to identify and address some of these optimization issues and a drastic decrease in building energy consumption was seen between Year 1 and Year 2.

13.3 Commissioning

Third-party commissioning is a way of safeguarding system performance by having an engineering consultant evaluate whether the installing contractor(s) provided what was agreed upon. It is clear to the research team that commissioning of a system like the BCPV VRF ASHP is different than commissioning of a more conventional HVAC system. There is a high degree of system complexity both in the design and operation of the system, but also in the controls coordination with other HVAC systems in the building. There are multiple points of failure or suboptimal system operation that might be difficult to detect without a detailed analysis.

A system may turn on to provide heating and cooling to meet thermostat set-points of the building – but it may not do so in an optimal way that achieves deep reductions expected from the feasibility assessment. It follows that the commissioning of a VRF ASHP retrofit in a building like the BCPV Visitor Centre retrofit should include a highly detailed review of BAS trend data with an emphasis on the optimization of system control for the *full building*.

The knowledge on how to effectively commission a VRF ASHP in the context of a retrofit may not be widespread. Project managers may not know the requirements for commissioning agents and the agents themselves may approach commissioning as they would in a conventional HVAC system. This could leave undiagnosed system issues and suboptimal performance. Guidance is needed for building owners and commissioning agents regarding the commissioning VRF ASHPs with a lens for full building optimization.

13.4 Operation

On-site staff have many competing responsibilities aside from building operations and the new VRF ASHP system was perceived as more challenging for the building operator to manage compared to a more conventional HVAC system. Some of the system controls are not easily accessible and this limits the building operators view of what is happening in the system. While a BAS may seem to simplify some of the system control – in practice, it may not be the case from the building operator's perspective.

Cloud-based monitoring and management of VRF ASHPs is now available as an add-on from some manufacturers. This would allow for some system management to be conducted off-site and could reduce the burden on site-staff. It is especially relevant for management of a portfolio of buildings. There are powerful analytical tools available that can help to quickly identify and diagnose system problems that might otherwise be difficult to detect with visual inspections.

14.0 BARRIERS TO BROADSCALE DEPLOYMENT

In addition to the implementation challenges mentioned previously, there are various barriers towards greater deployment of VRF ASHPs in the GTHA. These include:

- **Awareness of the Technology.** In the GTHA, and Ontario, the large majority of buildings are heated with natural gas. Many building owners are not aware of alternatives like VRF ASHPs and would continue to see conventional natural gas equipment as the only viable options.
- **Awareness of Carbon Pricing.** Carbon pricing has not yet increased to the extent where it is a reality that many building owners are considering in large capital expenditures. High-efficiency electric options like ASHPs can be a lower-cost option in terms of utility rates now or in the near future, but the perception remains that conventional natural gas equipment is the cheaper option.
- **Perception of Technological/Operational Risk.** For many building owners, VRF ASHPs are new and are therefore perceived as risky. It is important to have data-driven case studies of actual retrofits to hold as examples for prospective system owners.
- **Industry Capacity.** Similar studies have noted that there is a shortage of industry professional for the various stages of the heat pump implementation process.[5]
- **Modeling.** Modeling of VRF ASHPs utilizing heat recovery requires special expertise that is not available at every engineering consultancy and not every software package is capable of modeling VRF systems well. More studies are required to compare actual vs. modeled performance to build accuracy and confidence in modeling.
- **Feasibility Assessments.** Building owners interested in VRF systems will typically procure a feasibility assessment, but they must specify the modeling requirements in their procurement process. To the research team's knowledge, there is no best practice guide for VRF ASHP system to which building owners can refer when procuring feasibility assessments. Building owners may then not get high quality feasibility assessments which may under or over-predict systems savings, and overall diminish the confidence in the technology.

[5] TAF. Lessons from a Heat Pump Retrofit at CityHousing Hamilton. 2022. https://taf.ca/wp-content/uploads/2022/05/TAF_CityHousingHamilton-Retrofit-Case-Study_2022-1.pdf

15.0 LESSONS LEARNED

System Design:

1. **Oversizing should be avoided.** It is common for conventional heating systems to be oversized by up to 40% because the first aim of a designer engineer is to ensure adequate heating and cooling. In conventional systems, there is neither a significant penalty to upfront costs nor to system performance by oversizing. However, this design approach can be problematic for heat pump systems. The cost premiums are greater for the extra capacity, and the impact on performance can be more significant. A key recommendation to future system owners is to invest more time into the heating load calculations for the building to ensure that the system sizing is well-matched to the heat loss, and that any oversizing is greatly diminished compared to standard practice for conventional systems. The monitoring data in this study showed that the COP was significantly reduced when the system was operating a low fraction of its rated heating capacity.

2. **Integration between the ASHP and natural gas systems needs to be carefully considered.** This is a key area that can cause the system to fall short of expectations. The Visitor Centre had three gas heating systems: hydronic perimeter heating, tempering for the ventilation air, and tempering for the outdoor enclosure. Data from Year 1 indicated that all three systems were providing more heating than was required. In the monitored subsystem, the ASHP was removing excess heat provided by the tempered ventilation air (effectively fighting against it) and this created a notable system inefficiency.

3. **Outdoor enclosures for ASHPs should be avoided where possible.** The outdoor enclosures, while simple in concept, were one of the more challenging aspects of this retrofit. The control of the louvres and natural gas tempering introduced greater control complexity. The research team has since learned that designers are now moving away from outdoor enclosures in these systems.

Ensuring System Success:

1. **M & V is crucial.** It is important to understand that despite the issues identified in Year 1, indoor set-points were being met, there were no equipment faults, and no obvious visual indications there were problems. Issues might therefore easily persist unnoticed. A basic check on system performance is an IPMVP-adherent analysis of energy savings based on utility bill or submetering data. If the savings deviates significantly from the expectations of the feasibility assessment, then it is an indicator that there may be issues meriting further investigation.

2. **Cloud-based monitoring and management may be helpful to ensure system success.** Site staff often have many competing responsibilities. System management can be made easier if it is partly done off-site with the help of cloud-based monitoring which is an add-on offered by some system manufacturers. This is especially relevant where a portfolio of buildings is being managed. There are powerful analytical tools available that can help to quickly identify and diagnose system problems that might otherwise be difficult to detect with visual inspections.

3. **Data-driven third-party system commissioning is important.** Controls in a retrofit like this are complex. Errors are likely to occur, and so are unexpected interactive effects with other building systems. As the complexity of the system increases, the commissioning should be increasingly

data-driven and include detailed analysis of BAS trend data with a lens for full system optimization. This makes commissioning a retrofit with this level of complexity more challenging then with conventional systems. The development of commissioning guidelines for VRF ASHP retrofits would be helpful to both building owners and commissioning agents.

4. **A designated party should be assigned to ensure continued effective overall integration of the ASHP system with other existing building systems.** This may be internal staff or an external contractor. There should be an individual on the retrofit team tasked with ensuring effective integration for the full building. This a consequence of (2), where unexpected interactive effects between the ASHP and existing building systems are likely.

16.0 CONCLUSION AND FUTURE WORK

This study demonstrated a VRF ASHP retrofit that reduced operating costs and significantly reduced natural gas consumption in a large institutional building. System implementation challenges have been documented and can be addressed in future retrofits based on key lessons learned from this study. Despite notable gas reductions, there was a discrepancy between the savings predicted in the feasibility study and those achieved in practice. More work is required comparing modeled and actual performance to refine and build confidence in system modeling. This may start with technical guidance from equipment manufacturers regarding best practices for system modeling that building owners can use to inform their procurement for feasibility studies. Similar guidance on commissioning best practices is needed as well.

17.0 APPENDIX A: UTILITY DATA AND BASELINE REGRESSION MODELS

17.1 Data Tables

Table 17-1. Baseline natural gas consumption data gathered from utility bills.

From	To	HDDs	m^3
1/5/2015	2/5/2015	810	26,484
2/6/2015	3/6/2015	828	24,399
3/7/2015	4/2/2015	468	15,037
4/3/2015	5/5/2015	309	10,535
5/6/2015	6/5/2015	100	2,105
6/6/2015	7/3/2015	29	1,212
7/4/2015	8/5/2015	10	1,193
8/6/2015	9/3/2015	14	1,144
9/4/2015	10/3/2015	58	2,685
10/4/2015	11/6/2015	250	9,269
11/7/2015	12/7/2015	388	12,733
12/8/2015	1/4/2016	381	14,937
1/5/2016	2/5/2016	623	18,198
2/6/2016	3/5/2016	602	18,198
3/6/2016	4/4/2016	403	12,459
4/5/2016	5/4/2016	349	10,076
5/5/2016	6/5/2016	113	4,161
6/6/2016	7/8/2016	34	1,198
7/9/2016	8/5/2016	3	1,144
8/6/2016	9/6/2016	7	1,312
9/7/2016	10/6/2016	43	1,020
10/7/2016	11/6/2016	218	4,807
11/7/2016	12/6/2016	367	11,487
12/7/2016	1/6/2017	608	18,014
1/7/2017	2/4/2017	546	16,294
2/5/2017	3/5/2017	502	17,481
3/6/2017	4/5/2017	504	17,408
4/6/2017	5/4/2017	239	7,292
5/5/2017	6/5/2017	163	4,023
6/6/2017	7/5/2017	25	1,125
7/6/2017	8/5/2017	4	1,465
8/6/2017	9/4/2017	31	1,524
9/5/2017	10/5/2017	64	1,957
10/6/2017	11/6/2017	193	8,178
11/7/2017	12/7/2017	444	15,175
12/8/2017	1/5/2018	719	21,609

Table 17-2. Baseline electricity consumption data gathered from utility bills.

From	To	HDDs	kWh	kW
12/24/2014	1/23/2015	691	183,524	356
1/24/2015	2/23/2015	857	197,573	337
2/24/2015	3/23/2015	585	149,673	280
11/24/2017	12/23/2017	534	139,022	276
11/24/2015	12/23/2015	389	120,758	269
3/24/2015	4/23/2015	388	122,267	263
1/24/2016	2/23/2016	601	128,974	261
10/24/2017	11/23/2017	390	112,196	246
2/24/2016	3/23/2016	450	102,606	245
9/24/2015	10/23/2015	169	89,945	225
10/24/2015	11/23/2015	293	107,018	221
7/24/2016	8/23/2016	1	87,031	206
8/24/2016	9/23/2016	15	75,640	205
12/24/2015	1/23/2016	607	121,581	205
6/24/2016	7/23/2016	9	77,914	204
4/24/2016	5/23/2016	224	87,965	203
8/24/2015	9/23/2015	35	78,226	202
5/24/2015	6/23/2015	47	77,597	196
7/24/2015	8/23/2015	8	82,109	190
4/24/2015	5/23/2015	149	85,369	189
5/24/2016	6/23/2016	33	68,390	188
6/24/2015	7/23/2015	23	76,307	186
9/24/2017	10/23/2017	97	54,571	152

Table 17-3. Post-retrofit natural gas consumption gathered from utility bills.

From	To	HDDs	m^3	Actual/Estimated
3/3/2021	4/4/2021	452	13,484	Actual
4/5/2021	5/1/2021	230	5,819	Actual
5/2/2021	6/1/2021	148	2,530	Actual
6/2/2021	7/3/2021	7	386	Actual
7/4/2021	8/1/2021	5	193	Actual
8/2/2021	9/1/2021	0	479	Actual
9/2/2021	10/3/2021	39	793	Actual
10/4/2021	11/1/2021	139	3,821	Actual
11/2/2021	12/3/2021	417	13,325	Actual
12/4/2021	1/2/2022	465	14,623	Actual
1/3/2022	2/3/2022	780	25,903	Actual
2/4/2022	3/3/2022	566	17,325	Actual

Table 17-4. Post-retrofit electricity consumption data gathered from utility bills.

From	To	HDDs	kWh	kW
2/24/2021	3/23/2021	444	104,567	198
3/24/2021	4/23/2021	299	96,601	181
4/24/2021	5/23/2021	191	75, 220	159
5/24/2021	6/23/2021	38	60,964	118
6/24/2021	7/23/2021	1	66,591	141
7/24/2021	8/23/2021	3	73,883	165
8/24/2021	9/23/2021	8	68,399	172
9/24/2021	10/23/2021	82	71,303	155
10/24/2021	11/23/2021	334	83,553	163
11/24/2021	12/23/2021	463	95,755	178
12/24/2021	1/23/2022	674	116,176	223
1/24/2022	2/23/2022	675	111,328	198

Table 17-5. Summary of electricity consumption adjusted baseline and non-routine adjustment.

From	To	Adjusted Baseline [kWh]	Non-routine adjustment [kWh]
2/24/2021	3/23/2021	126,428	30,655
3/24/2021	4/23/2021	108,151	21,879
4/24/2021	5/23/2021	94,436	14,196
5/24/2021	6/23/2021	75,083	2,261
6/24/2021	7/23/2021	70,428	15
7/24/2021	8/23/2021	70,636	39
8/24/2021	9/23/2021	71,266	127
9/24/2021	10/23/2021	80,662	6,078
10/24/2021	11/23/2021	112,488	26,162
11/24/2021	12/23/2021	128,827	33,384
12/24/2021	1/23/2022	155,662	43,381
1/24/2022	2/23/2022	155,741	44,336

Table 17-6. Breakdown of non-routine adjustment for electricity consumption.

From	To	Bolton[[kWh]	Flynn [kWh]	Manse [kWh]	McKenzie [kWh]	Sams [kWh]	School [kWh]	Laskay [kWh]
2/24/2021	3/23/2021	1,996	3,690	4,141	2,996	2,762	6,663	8,408
3/24/2021	4/23/2021	1,344	1,979	2,745	2,162	1,993	4,441	7,214
4/24/2021	5/23/2021	843	1,052	1,706	1,411	1,300	2,774	5,110
5/24/2021	6/23/2021	122	138	246	210	193	557	794
6/24/2021	7/23/2021	0	0	0	0	0	15	0

7/24/2021	8/23/2021	0	0	0	0	0	39	0
8/24/2021	9/23/2021	0	0	0	0	0	127	0
9/24/2021	10/23/2021	328	294	654	582	536	1,347	2,338
10/24/2021	11/23/2021	1,628	2,430	3,329	2,609	2,405	5,129	8,632
11/24/2021	12/23/2021	2,140	3,938	4,953	3,284	2,956	7,112	9,000
12/24/2021	1/23/2022	2,925	6,244	5,708	4,161	3,841	10,310	10,191
1/24/2022	2/23/2022	2,936	6,273	6,142	4,439	3,993	10,349	10,203

Table 17-7. Summary of adjusted baseline and non-routine adjustment for electricity demand.

From	To	Adjusted Baseline [kW]	Non-routine Adjustment [kW]
2/24/2021	3/23/2021	256	59
3/24/2021	4/23/2021	233	46
4/24/2021	5/23/2021	216	38
5/24/2021	6/23/2021	192	34
6/24/2021	7/23/2021	186	14
7/24/2021	8/23/2021	187	15
8/24/2021	9/23/2021	187	16
9/24/2021	10/23/2021	199	34
10/24/2021	11/23/2021	239	46
11/24/2021	12/23/2021	259	55
12/24/2021	1/23/2022	293	80
1/24/2022	2/23/2022	293	78

Table 17-8. Breakdown of non-routine adjustment for demand.

From	To	Manse [kW]	Laskays [kW]	Bolton [kW]	Flynn [kW]	Sams [kW]	McKenzie [kW]	School [kW]
2/24/2021	3/23/2021	8	13	4	9	6	6	14
3/24/2021	4/23/2021	6	12	3	6	5	4	10
4/24/2021	5/23/2021	5	11	2	4	4	4	8
5/24/2021	6/23/2021	4	11	2	3	3	3	7
6/24/2021	7/23/2021	1	9	1	0	3	1	1
7/24/2021	8/23/2021	1	9	1	0	1	2	1
8/24/2021	9/23/2021	1	9	1	0	1	2	1
9/24/2021	10/23/2021	4	11	2	3	2	3	7
10/24/2021	11/23/2021	6	12	3	6	3	5	10
11/24/2021	12/23/2021	8	13	4	8	4	5	13
12/24/2021	1/23/2022	12	15	6	13	5	8	20
1/24/2022	2/23/2022	11	15	5	13	7	7	20

Table 17-9. Summary of natural gas adjusted baseline and non-routine adjustment.

From	To	Adjusted Baseline [m³]	Non-routine Adjustment [m³]
3/3/2021	4/4/2021	14,633	-2,699
4/5/2021	5/1/2021	7,878	-1,494
5/2/2021	6/1/2021	5,386	-851
6/2/2021	7/3/2021	1,085	6
7/4/2021	8/1/2021	1,006	9
8/2/2021	9/1/2021	867	6
9/2/2021	10/3/2021	2,057	-122
10/4/2021	11/1/2021	5,091	-936
11/2/2021	12/3/2021	13,576	-2,577
12/4/2021	1/2/2022	15,034	-2,825
1/3/2022	2/3/2022	24,638	-4,253
2/4/2022	3/3/2022	18,104	-3,255

Table 17-10. Breakdown of data used for non-routine adjustment of natural gas consumption.

From	To	Manse [m³]	Laskays [m³]	Bolton [m³]	Flynn [m³]	Sams [m³]	Burwick [m³]	McKenzie [m³]	Church [m³]	School [m³]	Second [m³]
3/3/2021	4/4/2021	-406	-891	-196	-342	-277	108	-301	121	-649	134
4/5/2021	5/1/2021	-212	-597	-104	-141	-158	66	-171	70	-337	90
5/2/2021	6/1/2021	-117	-357	-58	-70	-90	50	-97	40	-205	54
6/2/2021	7/3/2021	0	0	0	0	0	14	0	0	-8	0
7/4/2021	8/1/2021	0	0	0	0	0	14	0	0	-5	0
8/2/2021	9/1/2021	0	0	0	0	0	6	0	0	0	0
9/2/2021	10/3/2021	-11	-48	-5	-2	-10	4	-11	4	-51	7
10/4/2021	11/1/2021	-128	-405	-63	-72	-100	38	-108	44	-203	61
11/2/2021	12/3/2021	-385	-878	-187	-315	-266	102	-289	116	-607	132
12/4/2021	1/2/2022	-450	-860	-200	-383	-280	90	-308	123	-688	129
1/3/2022	2/3/2022	-659	-1039	-323	-705	-419	159	-461	179	-1137	151
2/4/2022	3/3/2022	-504	-865	-243	-507	-332	138	-373	138	-842	135

Table 17-11. Summary of baseline utility consumption models, where x is HDDs.

Parameter	Model	R^2	SE	CV(RMSE)	MBE
Natural gas consumption (m^3)	$y = 30.46 \cdot x + 867.2$	0.98	1,159	0.12	338.7
Electricity consumption (kWh)	$y = 126.7 \cdot x + 70{,}240$	0.86	13,970	0.13	1071
Electricity demand (kW)	$y = 0.1581 \cdot x + 186.7$	0.71	27.38	0.12	1.274

Table 17-12. Uncertainty of baseline models for 1-year (12-month) total consumption.

Parameter	Estimated Total	√12 x SE	t-statistic (90% Confidence)	Absolute Precision	Relative Precision
Natural gas consumption (m^3)	109,355	4,015	1.69	+/- 6,800	+/- 6.2%
Electricity consumption (MWh)	1,250	48.39	1.69	+/- 82	+/- 6.5 %
Sum of monthly electricity demand (kW)	2,740	94.85	1.69	+/- 160	+/- 5.7 %

Table 17-13. Uncertainty of savings estimates models for 1-year (12-month).

Parameter	Estimated Savings	Absolute Precision*	Relative Precision
Natural gas consumption (m^3)	29,667	+/- 6,800	+/- 23%
Electricity consumption (MWh)	3	+/- 82	N/A**
Sum of monthly electricity demand (kW)	175	+/- 160	+/- 91 %

* Only includes uncertainty from baseline model. Does not include uncertainty from non-routine adjustment. Uncertainty of non-routine adjustment was not calculated. It was primarily based on actual measurements of when the new gas equipment was on. Regression modelling was only used to fill in data gaps.

**Savings is less than absolute precision.

17.2 Overview of Non-routine Adjustment

New gas heating equipment was either single-staged or two-staged. Air temperature sensors from Monnit were installed on the supply of the new gas equipment. The temperature data was used to determine when the equipment was on, and whether it was on the high- or low-stage (if needed). The installed sensor system began collecting data on a minute-scale around November 18, 2021. The Laskay's building sensor failed to operate during this time and a replacement was installed on April 1, 2022. Using the equipment specifications, the estimates of when the equipment was on was used to estimate the gas consumption.

The data was filtered to eliminate days with significant data loss (days with less than 95% data). A linear regression model was then created for each outbuilding based on the relationship between daily mean outdoor temperature and daily gas consumption estimated from the sensor data as shown in Figure 18-1. This model was used to estimate daily gas consumption for days in the heating season where data was not recorded by the sensors (Figure 18-2). Note that in some cases it was a gas equipment *upgrade* rather than electric-to-gas conversion and this was accounted for in the non-routine adjustment by assuming the efficiency was upgraded by 15%. The total non-routine adjustment for each outbuilding was then derived for each outbuilding (Table 18-10). Historical daily average temperature was obtained from Environment and Climate Change Canada weather data for Toronto City Centre accessed via weatherstats.ca.

The reduction in electricity consumption and demand were calculated for the converted outbuildings from the estimated gas consumption values (Tables 18-5 to 18-8). To calculate the reductions, gas consumption was estimated for each electricity billing cycle and used to determine the equivalent electricity in kWh to deliver the same amount of heat. Electricity demand reductions were calculated by identifying the day with the lowest average temperature in each billing cycle, estimating the corresponding gas consumption for that day using the predetermined model parameters in Table 18-15, and converting it to electricity in average kW to determine the maximum demand expected for that period.

Table 18-15. Summary of natural gas consumption models for each outbuilding, where x is daily average outdoor temperature.

Outbuilding	Model	R^2
Bolton Shop	$y = -0.36x + 7.25$	0.96
Burwick building	$y = -1.06x + 25.53$	0.73
Church building	$y = -1.09x + 28.56$	-5.42
Flynn house	$y = -1.01x + 13.87$	0.87
Manse house	$y = -0.77x + 15.09$	0.78
McKenzie house	$y = -0.43x + 10.75$	0.81
Sam's workshop	$y = -0.40x + 9.92$	0.95
School house	$y = -1.38x + 24.45$	0.88
Second house	$y = -0.46x + 29.14$	0.42

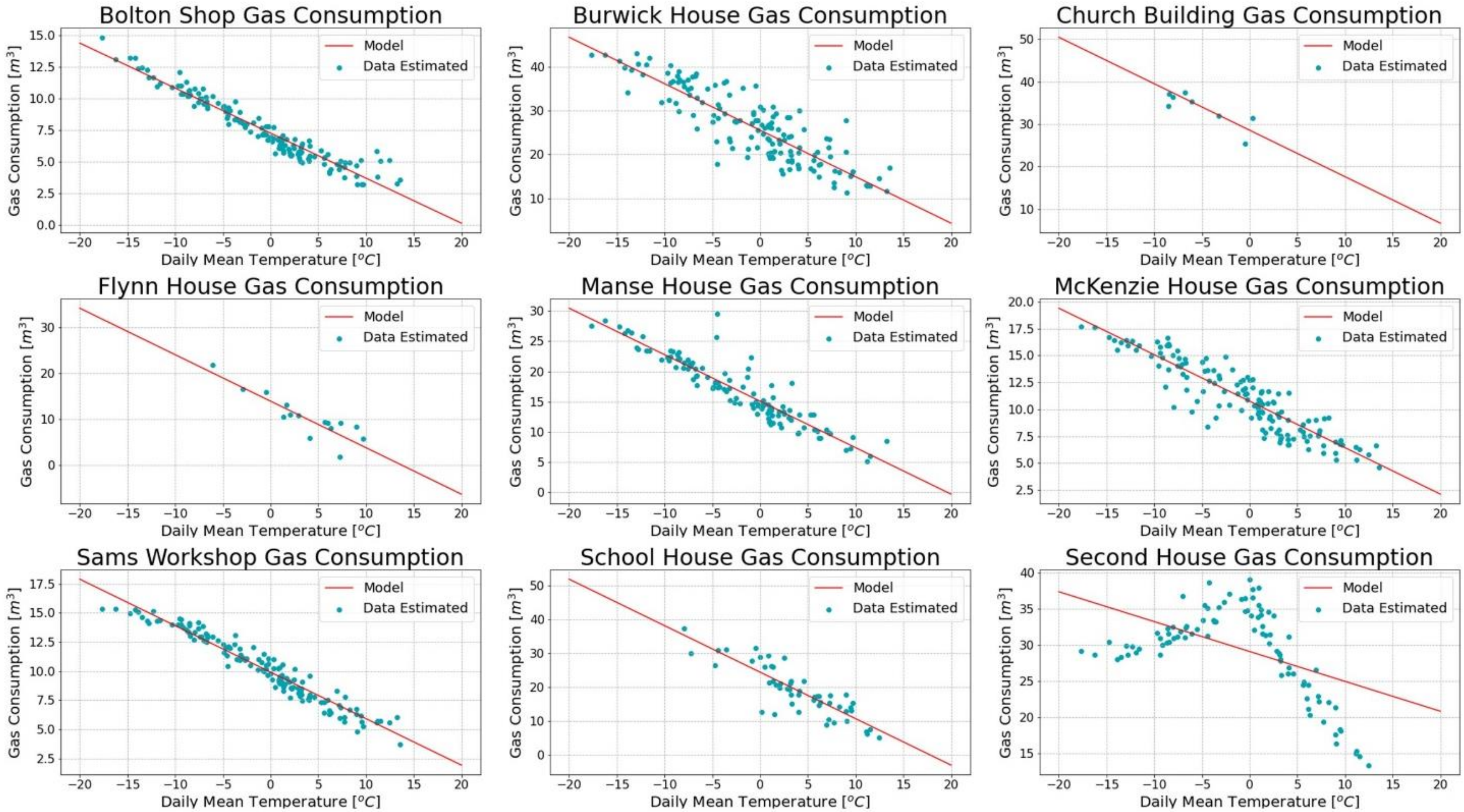


Figure 18-1. Gas consumption estimated from sensor data (blue) and linear regression model (red) plots for each outbuilding.

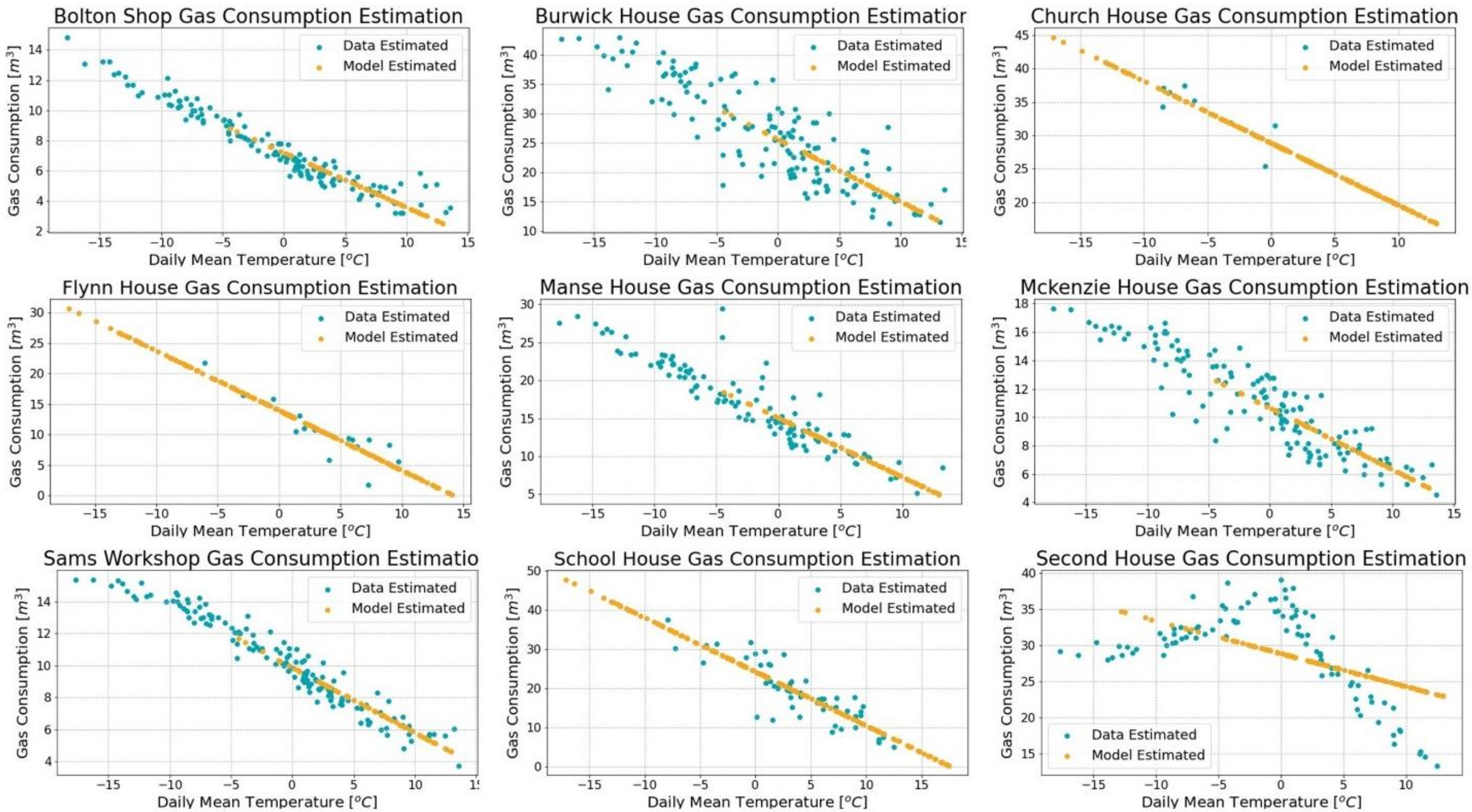
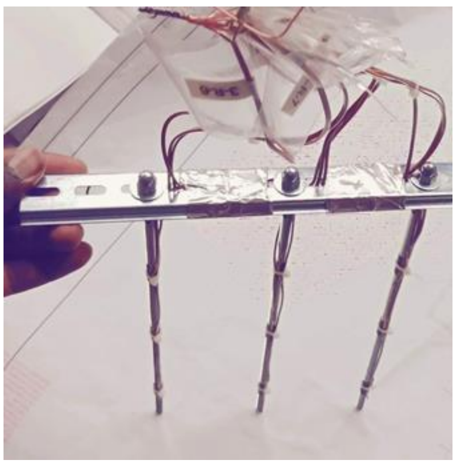


Figure 18-2. Estimated Natural Gas consumption for each outbuilding for the heating season (September 1, 2021 to May 5, 2022). Gas consumption estimated from sensor data (blue) and gas consumption estimated from models (orange).

18.0 APPENDIX B: SUPPLY AND RETURN TEMPERATURE SENSORS SET-UP AND VERIFICATION

The supply and return temperatures for each indoor fan coil were monitored using a grid of Type-K thermocouples attached to a Monnit Alta thermocouple reader. The grid was connected such that it automatically determined the average temperature across the duct cross-section. It is therefore better than a single-point temperature measurement. Each grid consisted of 9 thermocouples in a 3 by 3 arrangement. An example schematic from Fan Coil 1.03 is shown in Figure 18-1. Note that the ducting had transitions between square duct and round duct. Airflow measurements will be shown in round duct for Fan Coil 1.03, but the temperature measurements occurred in the square duct portion of the ducting.



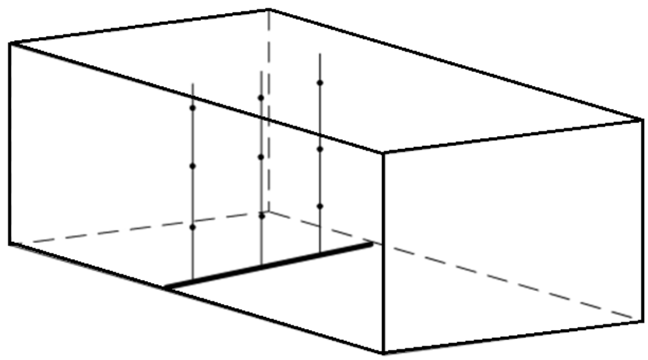


Figure 18-1. (Left) Thermocouple grids were created using threaded rod attached to DIN rail. (Right) Schematic of grid installed in duct.

In addition to the supply and return temperature sensor grids on the indoor fan coils, supply and return humidity (RH) sensors were installed as well, and these also had a built-in temperature sensor. It follows that there were redundant temperature sensors on the supply and return of each fan coil. To build confidence in the readings from the thermocouple grids, the secondary temperature measurements from the RH sensors were compared against the readings of the grids to confirm that the grids were configured properly. Exact agreement was not expected. Rather, this exercise was intended to identify if there were any large differences that might indicate an issue the supply and return thermocouple grids. This is shown in Figure 18-2 for Winter 2020/2021. The readings agree well.

Also shown in Figure 18-2 is a frequency histogram of the temperature difference across each fan coil during Winter 2020/2021. Both Fan Coil 1.01 and 1.02 are often cooling, as discussed in the body of the report. Note the peaks near 0 °C, this shows that supply and return grids are well-matched. Fan Coil 1.01 and 1.03 grids are matched to within less than ±1 °C. Fan Coil 1.02 grids are matched to less than ±2 °C. Overall, Figure 18-2 demonstrates that the temperature data is accurate and self-consistent. Redundant temperature sensors agree well with thermocouple grids, and the supply and return grids are well-matched.

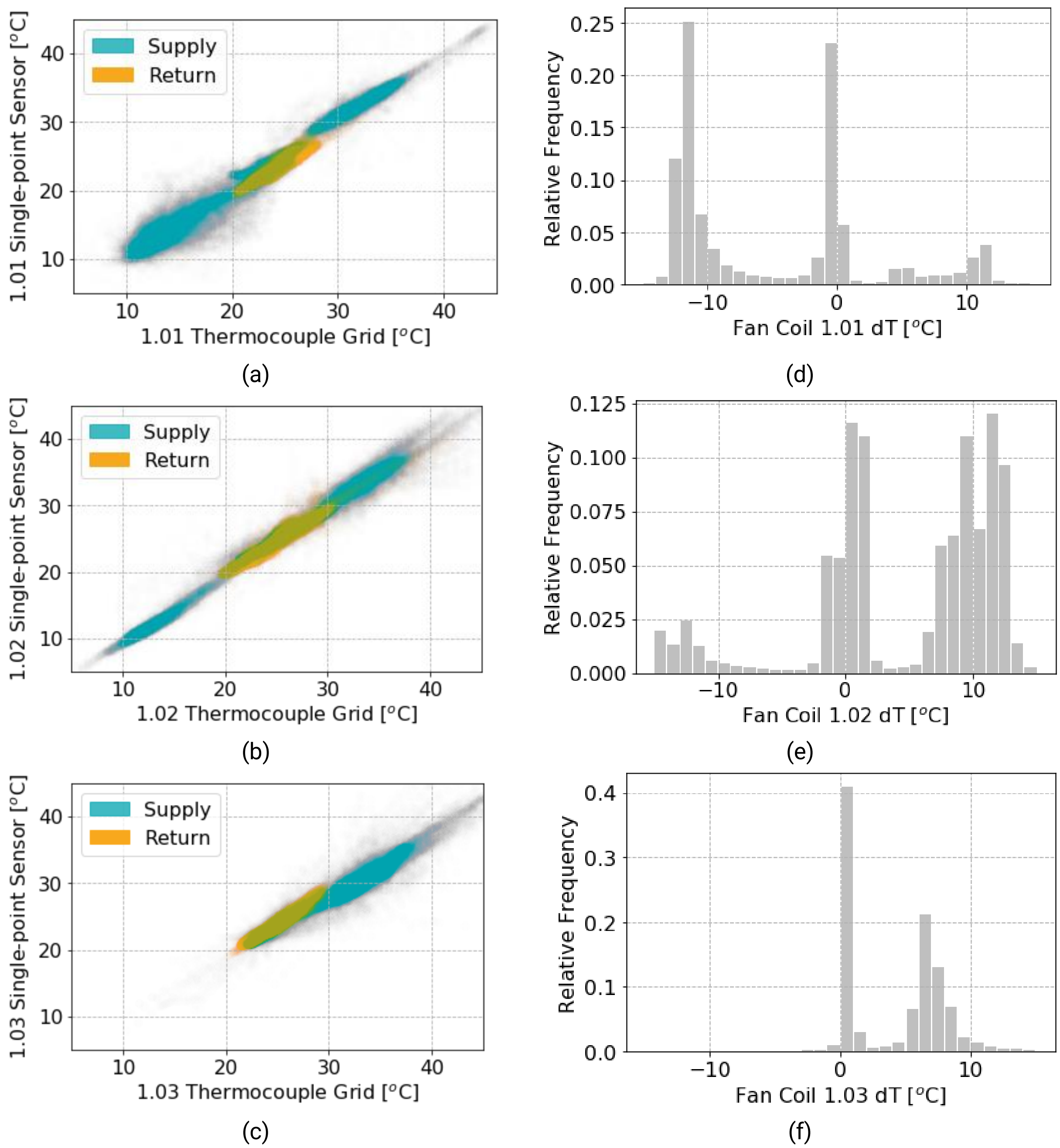


Figure 18-2. (a) to (c) For Fan Coil 1.01 to 1.03, the temperatures record by the single-point sensors are plotted against the thermocouple grids for Winter 2020/2021. The agreement is good and indicates that the thermocouple grids are properly configured. (d) to (f) Frequency histograms are shown for the temperature rise across each indoor fan coil for Winter 2020/2021. It is clear that each plot has a peak near 0 °C. This occurs when the unit is off. It indicates that the supply and return grids are well-matched since there is no measured temperature rise across the coil when the unit is off. Fan Coil 1.01 and 1.03 grids are matched to within less than ±1 °C. Fan Coil 1.02 grids are matched to less than ±2 °C.

19.0 APPENDIX C: ELECTRICAL POWER CONSUMPTION MONITORING SET-UP AND VERIFICATION

The electricity consumption of the heat pump was monitored using an Acuvim II from AccuEnergy. It is a revenue grade power meter (ANSI C12.20 (0.2 Class) and IEC 62053-22 (0.2S Class)). Using the AcuCloud platform, the data was also uploaded to the cloud where it was remotely downloaded for analysis. Note that only the heat pump outdoor unit was monitored for power consumption.

To ensure that the Acuvim II was configured properly for power monitoring it was compared against a Fluke 1730 Energy logger that was set-up to log energy consumption in parallel. The Fluke meter logged energy data for the heat pump outdoor unit between 2:28 pm and 4:53 pm on Feb 19, 2021. The starting totalized energy consumption read by the Acuvim II 3330.4 kWh and the ending value was 3342.3 kWh, yielding an energy consumption of 11.9 kWh for this interval. The Fluke meter recorded an energy consumption of 11.89 kWh during this time period. This is shown in Figure 19-1.

Figure 19-1. A Fluke 1730 Energy Logger was used to verify the electricity consumption measurements of the Acuvim II meter.

There was missing data from AcuCloud server for this time period. Data was uploaded at 5 min intervals so those measured in the field and uploaded to the cloud could not match exactly. The closest matching interval from the cloud data was from 2:25 pm to 4:45 pm, and during this time the totalized energy consumption recorded by the meter to the cloud was 11.4 kWh. The total duration of the field logging was 145 min, while the cloud interval was 140 min. The field logging interval was 3.6% longer than the cloud data interval. Similarly, the total energy consumption in the field was 4.4% greater. It follows that the difference in the values determined in the field and from the cloud data is explained by the slightly longer duration of the field logging. Overall, this exercise confirmed that the Acuvim II was properly configured and reading correct values.

20.0 APPENDIX D: AIRFLOW SENSORS SET-UP AND VERIFICATION

Airflow was the most challenging parameter to measure because there can be large variations in the airflow across the cross-section of the ducts. Integrating flow stations are often used for airflow measurements. These devices determine airflow via differences in the static and velocity pressure of the air at different points within the duct. However, this approach was prohibitively challenging and costly to implement at the BCPV Visitor Centre.

Instead, the approach to airflow monitoring in this project was based on single-point airflow measurements calibrated to traverses of the duct which determined total airflow. The single-point airflow sensor left in place throughout the monitoring in each duct was a Dwyer Series AVU Air Velocity Transmitter. It had a 0 to 10V output that was connected to Monnit Alta Voltage Meter 0 -10 VDC. It has an accuracy ± 5% full-scale but this is less relevant for this project because the sensor was calibrated in-situ using duct traverses.

For each of the three fan coils in the subsystem, and for each fan speed, a duct traverse was performed. Within each traverse, multiple airflow velocity measurements were taken across the duct cross-section. Air velocity was measured using a Fluke 922 Airflow Meter.

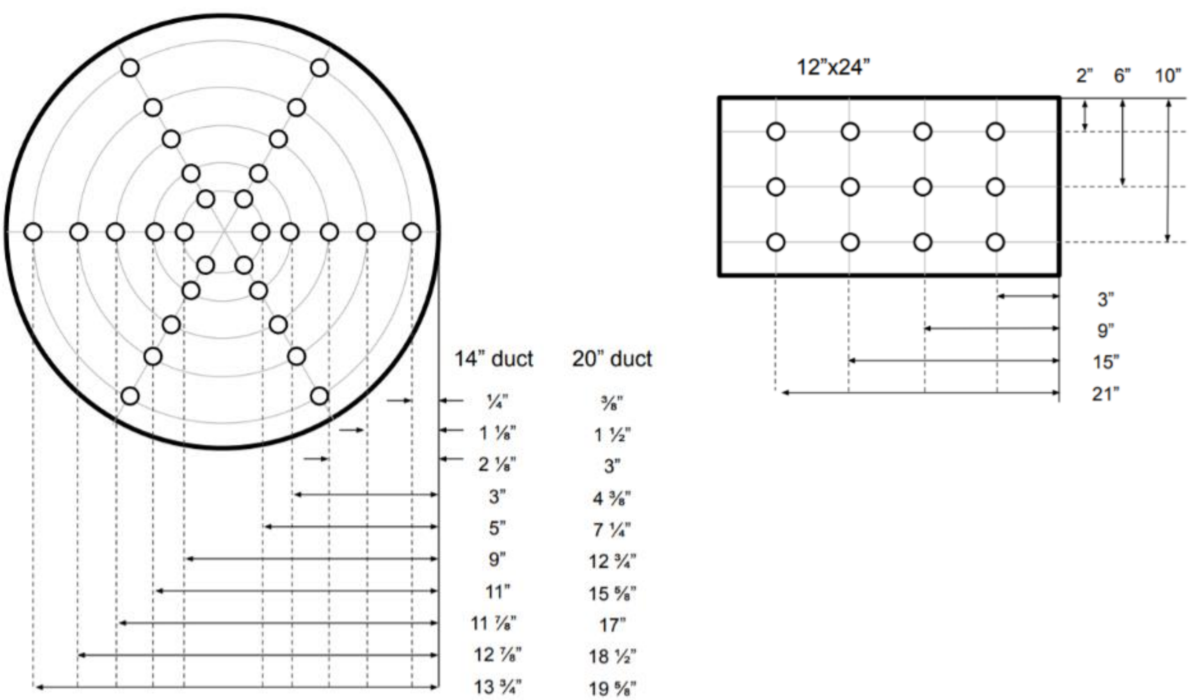


Figure 20-1. Schematic of the air velocity measurements taken during the duct traverses. Note that airflow measurements for Fan Coil 1.01 and 1.03 were in round ducts. Measurements for Fan Coil 1.02 were in a rectangular duct. This figure is shown for illustrative purposes. The specific points used in the traverse were slightly different than what is shown here. Dimensions are also shown for duct outside dimensions. Both 1.01 and 1.02 had insulation on the duct interior. The inside diameter was used for cfm calculations. Traverse data is provided at the end of this section.

The duct traverse data is provided at the end of this section. Python programming was used to interpolate and visualize the duct traverse measurements, as well as determine the average velocity of airflow through the duct. The velocity was then multiplied by the duct area to determine the airflow volume. Visualizations for the duct traverses are provided in Figure 20-2 to Figure 20-4.

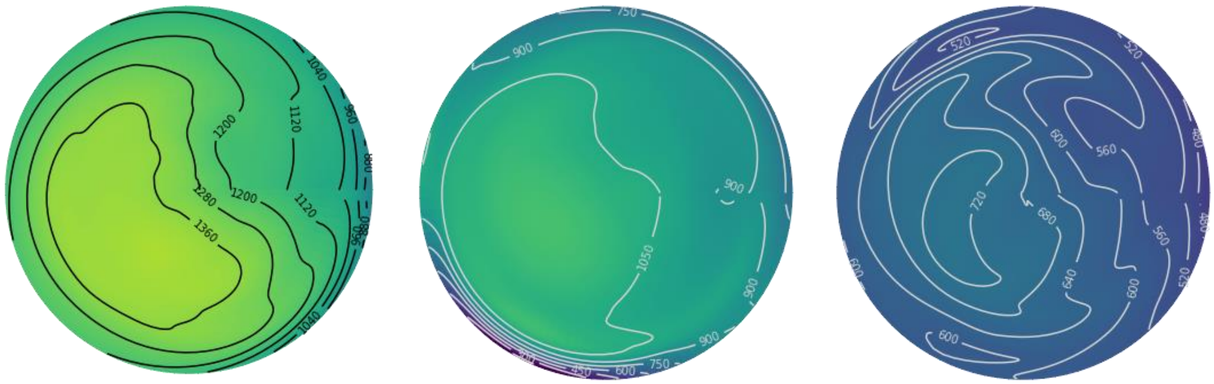


Figure 20-2. An interpolated air velocity map for the duct connect to Fan Coil 1.01 was produced using the duct traverse measurements for the duct. The values on the contour lines are the air velocity in fpm. The specifications indicated two speeds, but a third speed was also observed during the monitoring.

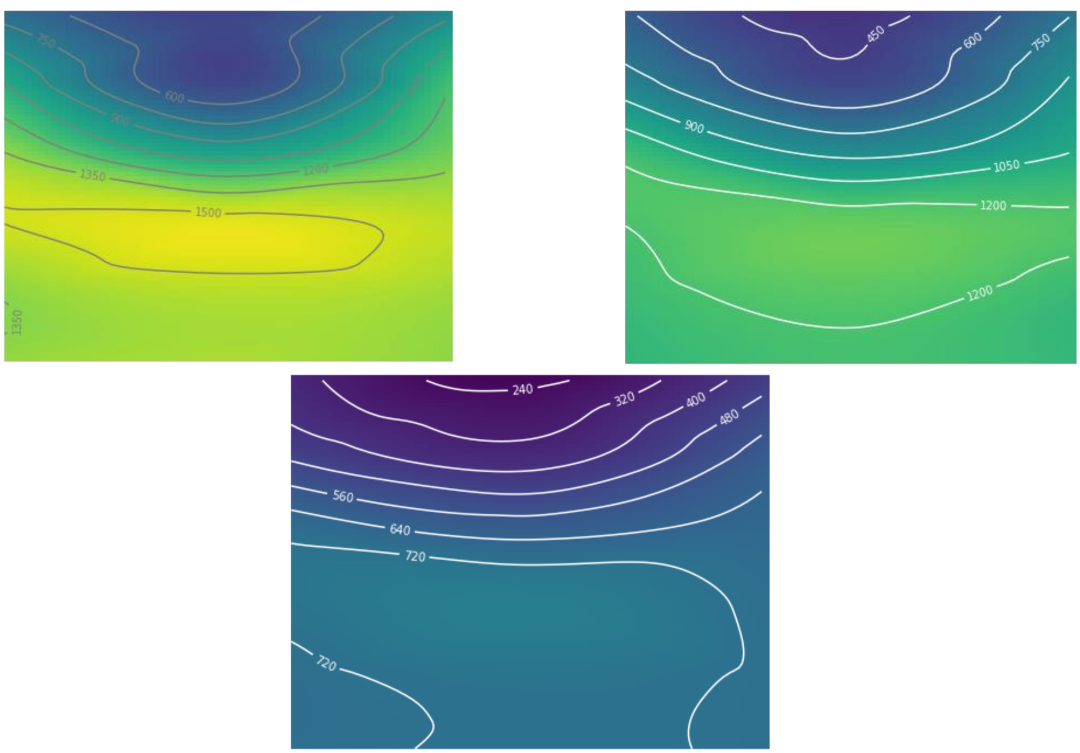


Figure 20-3. An interpolated air velocity map for the duct connect to Fan Coil 1.02 was produced using the duct traverse measurements for the duct. The values on the contour lines are the air velocity in fpm. The specifications indicated two speeds, but a third speed was also observed during the monitoring.

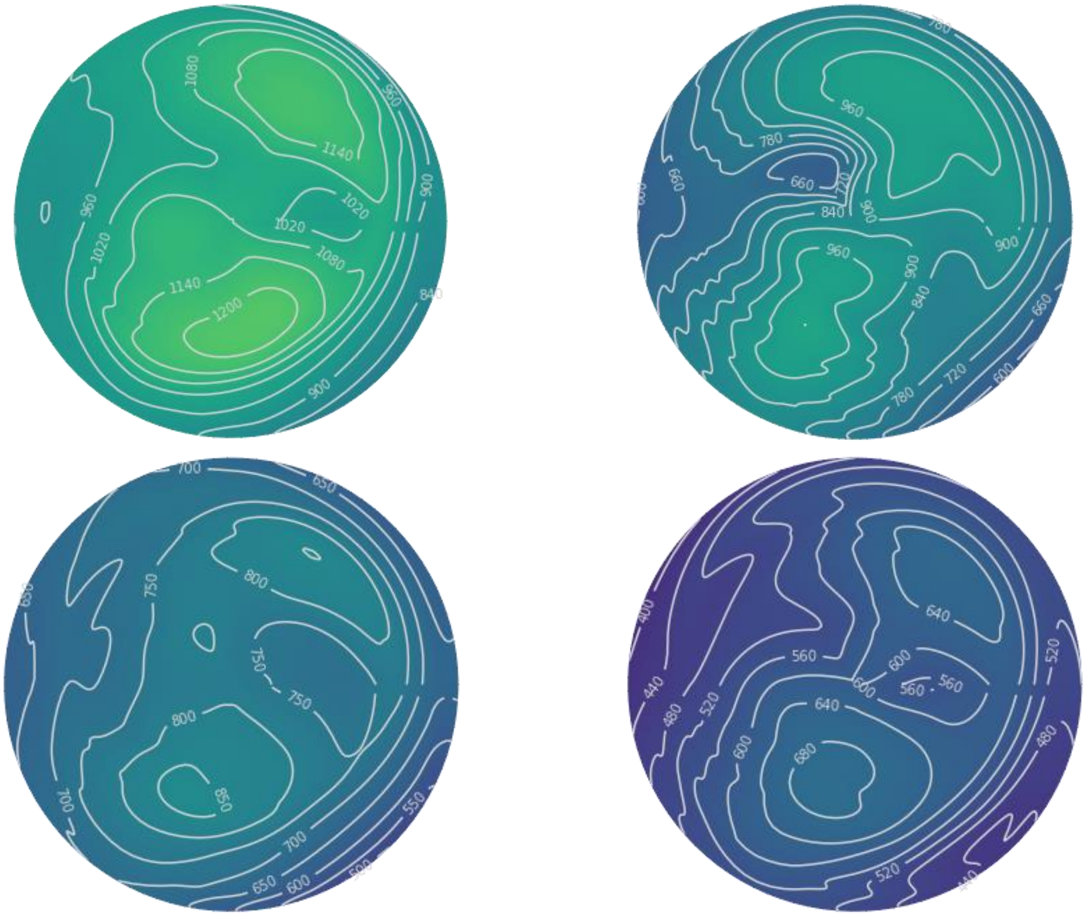


Figure 20-4. An interpolated air velocity map for the duct connect to Fan Coil 1.03 was produced using the duct traverse measurements for the duct. The values on the contour lines are the air velocity in fpm. The specifications indicated three speeds, but a fourth speed was also observed during the monitoring.

Table 20-1 provides a comparison of the duct traverse results against the expectation from the manufacturer specifications. Exact agreement was not expected because the actual static pressure created by the ductwork will be different than the assumptions used in the manufacturer ratings. Calibration curves were then created to relate the measured airflow to the voltage reading from the sensor. This is shown in Figure 20-5. The data is highly linear with no scatter. This indicates good measurement repeatability. Agreement between the measured airflow and the values provided in the manufacturer specifications was between -22% and +13%. Overall, the agreement shows that the measured airflow data is reasonable, although the airflow is low for Fan Coil 1.03.

Table 20-1. Comparison of airflow measurements against expected values from manufacturer specifications.

	Parameter	Fan Coil 1.01	Fan Coil 1.02	Fan Coil 1.03
	Model	FXMQ72MVJU	FXMQ96MVJU	FXMQ48PBVJU
	Duct size Interior	18.5’ diameter (circular)	12’ x 22’ (rectangular)	14’ diameter (circular)
	Duct area (ft^2)	1.87	1.83	1.07
Speed 1	Average velocity [fpm]	620	621	566
	Average voltage [V]	4.66	2.59	3.94
	Airflow measured [cfm]	1,157	1,138	605
	Airflow specs [cfm]	-	-	-
	Measured cfm vs. specs	-	-	-
Speed 2	Average velocity [fpm]	991	1,012	739
	Average voltage [V]	8.03	4.22	5.22
	Airflow measured [cfm]	1,850	1,857	790
	Airflow specs [cfm]	1,764	2,118	988
	Measured cfm vs. specs	**+5%**	**-12%**	**-20%**
Speed 3	Average velocity [fpm]	1244	1,215	849
	Average voltage [V]	9.84	5.18	6.25
	Airflow measured [cfm]	2322	2,228	908
	Airflow specs [cfm]	2,047	2,541	1,165
	Measured cfm vs. specs	**+13%**	**-12%**	**-22%**
Speed 4	Average velocity [fpm]	-	-	1040
	Average voltage [V]	-	-	7.23
	Airflow measured [cfm]	-	-	1,111
	Airflow specs [cfm]	-	-	1,377
	Measured cfm vs. specs [%]	-	-	**-19%**

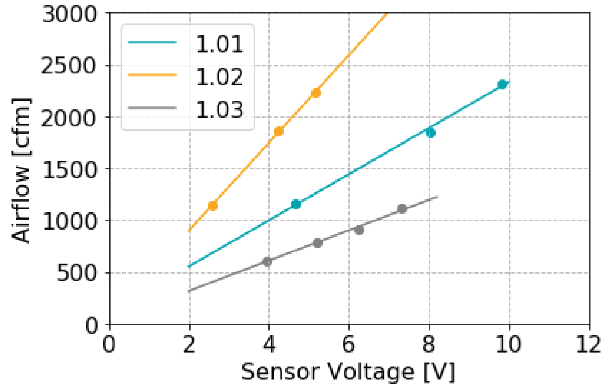


Figure 20-5. Data from the duct traverses and corresponding voltage values from the airflow sensors were used to create calibration curves for the airflow sensor voltage data.

Table 20-2. Fan Coil 1.01 duct traverse data where r is the distance from the centre of the duct and phi is the angular position in radians.

r [inches]	phi [radians]	Speed 1 [fpm]	Speed 2 [fpm]	Speed 3 [fpm]
8.8985	3.141593	600	1000	1180
7.8255	3.141593	640	1050	1280
6.4195	3.141593	680	1100	1380
5.2355	3.141593	710	1150	1380
2.5715	3.141593	730	1170	1380
2.5715	0	620	1050	1180
5.2355	0	580	900	1100
6.4195	0	530	900	1050
7.8255	0	510	850	1000
8.8985	0	440	690	800
8.8985	4.18879	600	100	1200
7.8255	4.18879	600	1050	1280
6.4195	4.18879	660	1100	1380
5.2355	4.18879	715	1120	1400
2.5715	4.18879	700	1170	1410
2.5715	1.047198	620	1020	1200
5.2355	1.047198	560	980	1200
6.4195	1.047198	600	950	1160
7.8255	1.047198	550	910	1130
8.8985	1.047198	520	710	1050
8.8985	2.094395	600	800	1120
7.8255	2.094395	520	900	1200
6.4195	2.094395	680	1050	1300
5.2355	2.094395	660	1090	1350
2.5715	2.094395	710	1080	1350

2.5715	5.235988	690	1080	1350
5.2355	5.235988	630	1000	1350
6.4195	5.235988	640	1000	1300
7.8255	5.235988	600	990	1250
8.8985	5.235988	580	820	1020

Table 20-3. Fan Coil 1.02 duct traverse data where x- and y-position are the horizontal and vertical positions of the measurement, and Area Fraction is the relative proportion of the duct cross-sectional area corresponding to the measurement point.

x-position [inches]	y-position [inches]	Area Fraction	Speed 1 [fpm]	Speed 2 [fpm]	Speed 3 [fpm]
2.83	3	0.088	700	1120	1380
6.50	3	0.069	700	1155	1360
10.17	3	0.069	740	1200	1500
13.83	3	0.069	690	1170	1360
17.50	3	0.069	460	830	1060
21.17	3	0.051	350	620	700
2.83	7	0.071	730	1185	1420
6.50	7	0.056	740	1235	1450
10.17	7	0.056	790	1270	1540
13.83	7	0.056	600	880	1000
17.50	7	0.056	350	500	500
21.17	7	0.040	220	400	490
2.83	11	0.053	710	1135	1360
6.50	11	0.042	710	1140	1400
10.17	11	0.042	700	1220	1460
13.83	11	0.042	670	1060	1300
17.50	11	0.042	600	930	1190
21.17	11	0.030	450	755	880

Table 20-4. Fan Coil 1.03 duct traverse data where r is the distance from the centre of the duct and phi is the angular position in radians.

r [inches]	phi [radians]	Speed 1 [fpm]	Speed 2 [fpm]	Speed 3 [fpm]	Speed 4 [fpm]
6.75	3.141593	410	620	600	900
5.875	3.141593	440	650	650	900
4.875	3.141593	480	700	680	940
4	3.141593	520	700	700	1000
2	3.141593	610	770	860	1120
2	0	550	730	880	1000
4	0	600	730	900	1000

4.875	0	600	770	930	1050
5.875	0	510	680	800	900
6.75	0	520	670	740	900
6.75	4.18879	550	700	880	940
5.875	4.18879	600	750	920	1000
4.875	4.18879	640	800	980	1150
4	4.18879	670	850	1000	1180
2	4.18879	680	830	990	1130
2	1.047198	610	750	950	1050
4	1.047198	660	840	990	1190
4.875	1.047198	660	840	1000	1180
5.875	1.047198	580	720	900	1120
6.75	1.047198	520	620	750	900
6.75	2.094395	420	700	770	980
5.875	2.094395	520	740	860	1010
4.875	2.094395	500	720	900	1020
4	2.094395	530	760	900	1030
2	2.094395	520	800	615	1020
2	5.235988	660	810	940	1180
4	5.235988	640	780	820	1180
4.875	5.235988	570	700	800	1010
5.875	5.235988	450	620	700	900
6.75	5.235988	440	500	600	800
2	6.283185	550	730	880	1000
4	6.283185	600	730	900	1000
4.875	6.283185	600	770	930	1050
5.875	6.283185	510	680	800	900
6.75	6.283185	520	670	740	900
0	0	605	782	873	1083
0	1.047198	605	782	873	1083
0	2.094395	605	782	873	1083
0	3.141593	605	782	873	1083
0	4.18879	605	782	873	1083
0	5.235988	605	782	873	1083
0	6.283185	605	782	873	1083

21.0 APPENDIX E: CAPACITY AND EFFICIENCY EQUATIONS

The equation for sensible heating or cooling capacity is shown in Equation (2), where ρ_a is the density of air, C_a is the specific heat capacity, F_a is the volumetric airflow, T_s is the supply air temperature, and T_R is the return air temperature.

$$Sensible\ Capacity = \rho_a \cdot C_a \cdot F_a \cdot (T_s - T_R) \qquad (2)$$

The air density shown in Equation (2) is not a constant. It depends on the air temperature, pressure, and moisture content. An accurate assessment of capacity and efficiency should take these factors into account. Figure 21-1 shows the density of air as a function of air temperature and atmospheric pressure.[6]

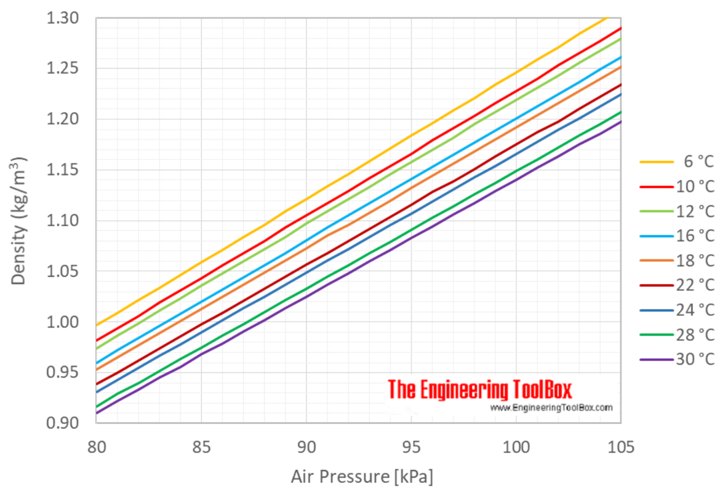


Figure 21-1. The density of air has a strong dependence on the atmospheric pressure and temperature.

The atmospheric pressure during Winter 2021/2022 is shown in Figure 21-2.[7] Variations in atmospheric pressure, and mean return air temperature, are relatively small but still significant enough to cause variations in the air density on the scale of a few percent.

[6] The Engineering Toolbox. Air - Density at varying pressure and constant temperatures. Accessed online Aug 2020: www.engineeringtoolbox.com/air-temperature-pressure-density-d_771.html

[7] Data taken from weatherstats.com for Toronto, which accesses Environment Canada data.

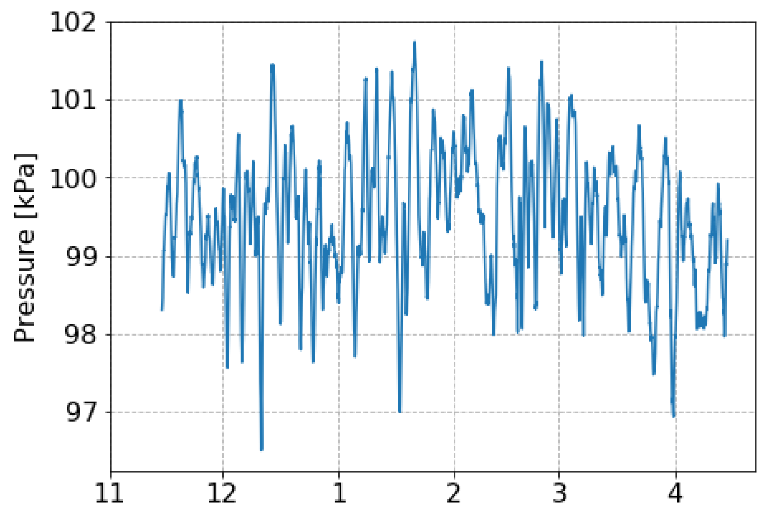


Figure 21-2. Atmospheric pressure during Winter 2021/2022.

At room temperatures, the moisture content is not expected to greatly impact the density (<1%) (Figure 21-3[8]) but it would be notable compared to the stated accuracy of the sensor (±2%). It follows that the airflow calculation did not assume standard air density but instead, took all these factors into account.

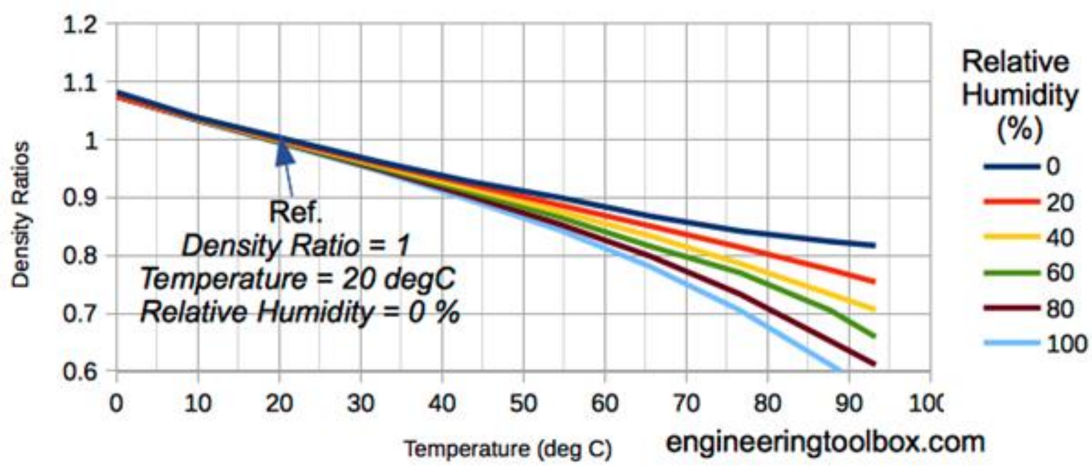


Figure 21-3. Impact of air moisture content on air density.

Air density as a function of temperature, humidity ratio, and pressure is given in Equation (3),[9] where p_a is the atmospheric pressure in units Pa, R_a is the individual gas constant for air (286.9 J kg^{-1} K^{-1}), T_a is the air temperature in units K, ω is the humidity ratio, and R_ω is the individual gas constant for water vapour (461.5 J kg^{-1} K^{-1}).

[8] The Engineering Toolbox. Density of Moist Humid Air. Accessed online August 2020: www.engineeringtoolbox.com/density-air-d_680.html

[9] Ibid.

$$\rho_{air} = \left(\frac{p_a}{R_a T_a}\right) \cdot \frac{(1+\omega)}{\left(1+\omega \cdot \left(\frac{R_w}{R_a}\right)\right)} \tag{3}$$

The humidity ratio can be determined from relative humidity data. The equation for relative humidity is shown in Equation (4),[10] where p_w is the water vapour partial pressure, p_{ws} is the saturation pressure for water vapour at the given dry bulb temperature, and $Corr$ is a correction factor that is dependent on the atmospheric pressure. The saturation pressure is the "holding capacity" that air has for water, beyond which the water will condense out of the air.

$$RH = \frac{p_w}{p_{ws}} \cdot Corr \tag{4}$$

Data for p_{ws} was plotted[11] and fit with a third-order polynomial to yield Equation (5) which is plotted in Figure 21-4.

$$p_{ws} = 7.097 \cdot 10^{-5} \cdot {T_a}^3 - 1.606 \cdot 10^{-5} \cdot {T_a}^2 + 5.385 \cdot 10^{-2} \cdot T_a + 7.193 \cdot 10^{-1} \tag{5}$$

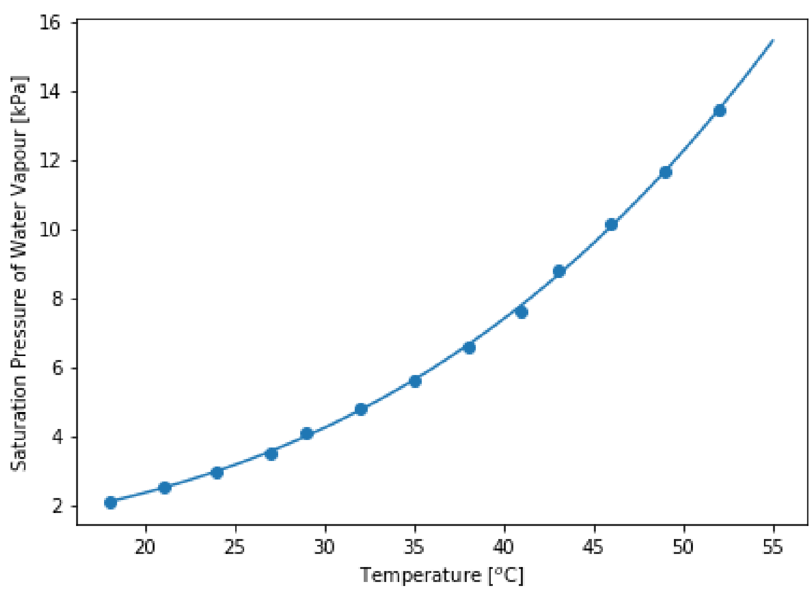


Figure 21-4. Water vapour saturation pressure as a function of temperature.

Data for $Corr$ was plotted[12] as well and fit with a linear equation. This is Equation (6), plotted alongside the data in Figure 21-5.

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[10] The Engineering Toolbox. Relative Humidity in Air. Accessed online August 2020: www.engineeringtoolbox.com/relative-humidity-air-d_687.html
[11] The Engineering Toolbox. Relative Humidity in Air. Accessed online August 2020: www.engineeringtoolbox.com/relative-humidity-air-d_687.html
[12] Ibid.

$$Corr = 0.01 \cdot p_a - 0.013 \tag{6}$$

With these relationships defined, it is possible to determine p_w from the data for RH, T_a, and p_a (Equation (7)). Finally, the humidity ratio can be determined according to Equation (8), and the air density can be calculated.

$$p_w = \frac{RH \cdot p_{ws}}{Corr} \tag{7}$$

$$\omega = 0.6220 \cdot \frac{p_w}{p_a - p_w} \tag{8}$$

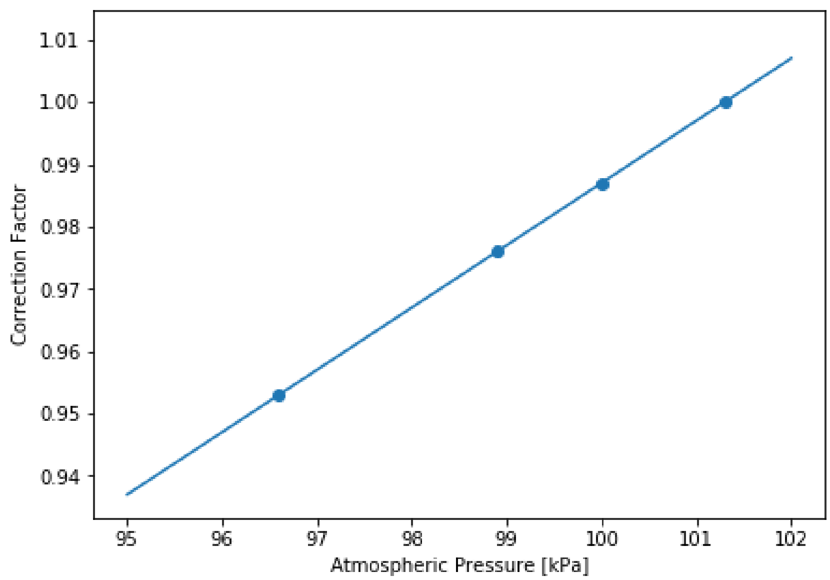

Figure 21-5. The relative humidity correction factor is a function of the atmospheric pressure.

It is possible to check these equations at standard air density, 0.075 lb/ft^3 for air at 68°F (20°C), 50% relative humidity, and 29.92″ Hg atmospheric pressure (101.3 kPa). A sample calculation is shown in Example 22-1, calculating the air density under these conditions.

Example 22-1. Air density calculation

Calculate the density of standard air.

- Calculate the saturation pressure for water vapour using Equation (5).

$$p_{ws}(20) = 7.097 \cdot 10^{-5} \cdot 20^3 - 1.606 \cdot 10^{-5} \cdot 20^2 + 5.385 \cdot 10^{-2} \cdot 20 + 7.193 \cdot 10^{-1}$$

$$p_{ws}(20) = 2.358\ kPa$$

- Calculate the correction factor for relative humidity using Equation (6).

$$C(101.3) = 0.01 \cdot 101.3 - 0.013$$
$$C(101.3) = 1$$

- Calculate the partial pressure of water vapour using Equation (7).

$$p_w = \frac{0.50 \cdot 2.358}{1}$$

$$p_w = 1.179\ kPa$$

- Calculate the humidity ratio using Equation (8).

$$\omega = 0.6220 \cdot \frac{1.179}{101.3 - 1.179}$$

$$\omega = 0.007325$$

Calculate the air density using Equation (3). Note that in this equation temperature is expressed in units of Kelvin and pressure in units of kPa.

$$\rho_{air} = \left(\frac{101.3}{286.9\ \cdot (273.15 + 20)}\right) \cdot \frac{(1 + 0.007325)}{\left(1 + 0.007325 \cdot \left(\frac{461.5}{286.9}\right)\right)}$$

$$\rho_{air} = 1.199\ \text{kg/m}^3$$

Units of kg/m^3 can be converted to lb/ft^3 using a conversion factor of 0.06243 to yield a result of 0.0747 lb/ft^3 – this is the standard air density.

Air is most dense at high pressure, low temperature, and low humidity. Assuming 18 ^{o}C, 102 kPa, and 20% RH, air density is 0.001219 kg/m^3 – a 1% increase over the standard density. Air is least dense at low pressure, high temperature, and high humidity. Assuming 24 ^{o}C, 97 kPa, and 80% RH yields 0.1127 - a 6% kg/m^3 decrease from the standard density. This shows that air density can vary considerably within the operating parameters of this study.

In practice within this study, air density could vary on the scale of several percent and impact the capacity and efficiency calculation by the same amount. Density was calculated for each data monitoring interval. The distribution of air density values used in the capacity and efficiency calculation for Winter 2021/2022 are shown in Figure 21-6.

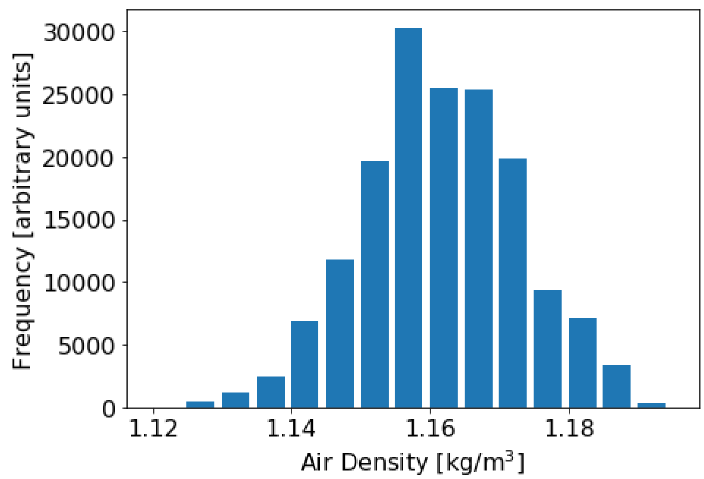

Figure 21-6. Distribution of air-density values used in efficiency and capacity measurements for Winter 2021/2022.

Specific heat capacity is impacted by the humidity ratio as well. The specific heat capacity of air is shown in Equation (9), where $C_{dry\ air}$ and C_{water} and are the specific heat capacities of dry air and water respectively. The value of $C_{dry\ air}$ is 1.006 kJ/(kg °C) and the value of C_{water} is 4.186 kJ/(kg °C). The distribution of specific heat capacities used in capacity and efficiency calculation for Winter 2021/2022 is shown in Figure 21-7. The specific heat capacity of air may vary by a small amount (on the scale of 1%) as a result of the changing moisture content. To calculate instantaneous sensible heating and cooling capacity, the density from Equation (3) and the specific heat capacity from Equation (9) were used in Equation (2).

$$C_a = C_{dry\ air} + \omega \cdot C_{p,water} \tag{9}$$

The latent cooling was estimated from the measured change in relative humidity across each fan coil, with the relative humidity converted to the humidity ratio using Equation (8). Note that this approach is not precise because of the accuracy of the relative humidity metres, but it was the best approach available to the team given the project budget. Equation (10) shows the latent capacity calculation, where L is the heat of condensation for water (2,260 kJ/kg). Airflow and air density are included as in Equation (2).

$$Latent\ Capacity = (\omega_S - \omega_R) \cdot F_a \cdot \rho_a \cdot L \tag{10}$$

The heating capacity only has the sensible component, while the cooling capacity is the sum of the latent *and* sensible capacities. To determine the total heating or cooling capacities, the individual sensible and latent capacities is summed across all indoor fan coils.

$$Heating\ Capacity = Sensible\ Capacity \tag{11}$$

$$\begin{aligned}Total\ Heating\ Capacity \\ &= (Heating\ Capacity\ Fan\ Coil\ 1.01) + (Heating\ Capacity\ Fan\ Coil\ 1.02) \\ &+ (Heating\ Capacity\ Fan\ Coil\ 1.03)\end{aligned} \tag{12}$$

$$Cooling\ Capacity = Sensible\ Capacity + Latent\ Capacity \tag{13}$$

$$\begin{aligned}Total\ Cooling\ Capacity \\ &= (Cooling\ Capacity\ Fan\ Coil\ 1.01) + (Cooling\ Capacity\ Fan\ Coil\ 1.02) \\ &+ (Cooling\ Capacity\ Fan\ Coil\ 1.03)\end{aligned} \tag{14}$$

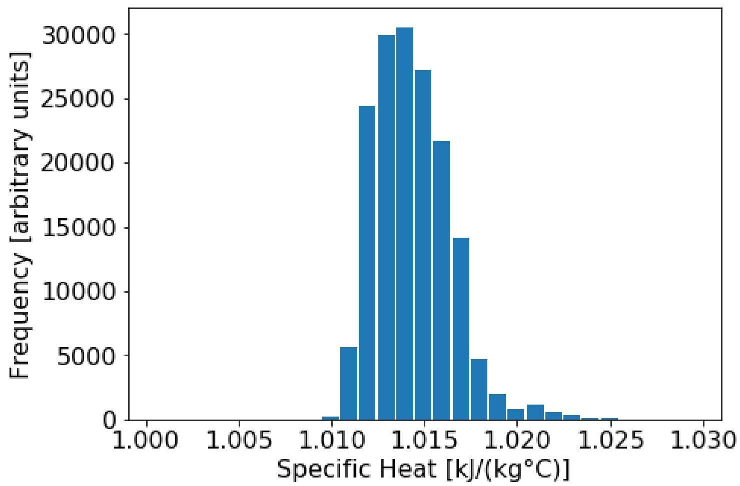

Figure 21-7. Distribution of specific heat capacity values used in efficiency and capacity measurements for Winter 2021/2022.

Lastly, the instantaneous efficiency (i.e. the coefficient of performance or COP) is the ratio of the total heating or total cooling capacity over the instantaneous electrical power draw of the heat pump. This did not include the power consumption of the fan coils in this study. Aggregated total daily COP is calculated as the total heating energy delivered (or removed) for each day divided by the electrical energy consumption of the heat pump for the day. Note this approach to COP calculation only applies when *all fan coils are operating in heating or all are operating in cooling.*

$$Instantaneous\ Heating\ COP = \frac{Total\ Heating\ Capacity}{Electrical\ Power\ Draw\ of\ Heat\ Pump} \tag{15}$$

$$Instantaneous\ Cooling\ COP = \frac{Total\ Cooling\ Capacity}{Electrical\ Power\ Draw\ of\ Heat\ Pump} \tag{16}$$

22.0 APPENDIX F: SET-UP OF INLET AND OUTLET TEMPERATURE SENSORS FOR THE OUTDOOR UNIT

The outdoor units were installed inside an enclosure and the enclosure could be heated with supplemental gas heat to keep the heat pumps operating in extreme cold conditions. It follows that the outdoor temperature is not necessarily the same as the inlet temperature to the outdoor units because the inlet temperature might be tempered by gas heat.

Outdoor temperature data was obtained from Environment Canada. Inlet temperatures to the outdoor unit were not initially monitored. This was due to budgetary considerations. However, after an initial review of the data which suggested a low heat pump efficiency, it was decided that the inlet/outlet temperatures would provide helpful context. Particularly, it would help to assess the if heat was being removed from the outdoor air.

In February 2021, the indoor temperature/humidity installed throughout each zone of the building were repurposed as inlet/outlet air temperature sensors for the outdoor unit. The outdoor unit is composed of two heat pumps, and air could enter the heat pumps via the front, side, or rear grills. For each heat pump, two temperature sensors were placed on each grill, and one in the exhaust as shown in Figure 22-1. Data losses for one of the exhaust sensors was significant for Winter 2021/2022, and the exhaust temperature for this that year was based on only one of the heat pumps.

Figure 22-1. Temperature sensors were placed on the front, side, and rear grills, as well as the exhaust, of the two heat pumps forming the outdoor unit for the monitored subsystem.