PERFORMANCE MONITORING OF LARGE CENTRAL AIR-TO-WATER HEAT PUMPS IN A TORONTO MULTI-UNIT RESIDENTIAL BUILDING

Results from French Quarters Shared Facilities

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About STEP

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

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Executive Summary

Introduction

In 2023, French Quarters Shared Facilities (FQSF)—two 11-storey condominium buildings in downtown Toronto—replaced their aging chiller with an air-to-water heat pump system designed to provide both cooling and low-carbon heating. The retrofit installed two heat pumps on secondary glycol loops connected to the building's hydronic system, with a natural gas boiler retained for backup heating. Analysis of detailed monitoring data over one summer and two winters evaluated performance, energy use, carbon savings, and operating cost. Key lessons were identified.

Findings Summary

- **Carbon and Energy Savings**: The heat pumps fully met the cooling demand and supplied approximately 70% of the space heating load, reducing natural gas and carbon emissions by 21,000 m^3/year and 30 t CO_2e/year, respectively. The main constraint on greater gas displacement was the heat pumps' supply temperature limitations during colder weather.
- **Efficiency:** Seasonal heating COPs ranged from 2.1 to 2.2, and cooling COPs were 2.9 for both heat pumps, with notable additional reductions once the circulator pump energy was included.
- **Operating Costs**:
  - *Heating season costs* rose in part due to a control strategy that maximized heat pump usage, even during peak time-of-use periods when electricity was significantly more costly.
  - *Cooling season costs* increased due to the added electricity consumption of the glycol circulator pumps and efficiency penalties from the secondary loop configuration.
  - There are identifiable pathways to achieving cost-neutral or even cost-saving performance in both modes through optimization.
- **Refrigerant Management**: A full refrigerant loss from one unit resulted in 125 t CO_2e released. This was equivalent to the cumulative carbon savings from 4 years of operation for the full system. While this could be an issue in any refrigerant system, including chillers, it underscores the need for robust preventative maintenance, safeguards to limit refrigerant loss, and contractor training. In the long term, low-GWP refrigerants will be essential.
- **Market and Risk Considerations**: Current natural gas prices are low but historically volatile. Because HVAC systems may remain in place for 20 years or more, systems that include a high-efficiency electric heating option can mitigate financial risk. In times of high gas prices, the business case for electric heat pumps strengthens significantly.

Recommendations Summary

- **Start with Comprehensive Feasibility Planning:** Determine the energy use breakdown of the facility to set realistic expectations. Optimize outdoor reset curves to promote performance and utilization. Assess heat pump potential by examining capacity and supply temperature constraints. Evaluate financial performance across various utility rate scenarios, emphasizing time-of-use control opportunities for cost optimization.
- **Engineer for System Efficiency:** Carefully size all ancillary components, accounting for glycol effects. Component sizing can restrict performance. In this study, pump, buffer tank, and heat exchanger sizing introduced performance issues.
- **Integrate with Broader Retrofit Strategy:** Combine the heat pump retrofit with other energy upgrades to improve overall cost-effectiveness.
- **Optimize Control Strategies:** Design controls to avoid short cycle times, reduce circulator runtimes, and align with owner goals (i.e. cost/carbon).
- **Prioritize Preventative Maintenance**: Follow manufacturer protocols. Require signed maintenance checklists. Use manufacturer-trained contractors. Monitoring can help catch faults early.
- **Support Product Innovation:** Manufacturers should pursue design improvements (e.g., split refrigerant systems, leak-limiting safeguards) and continue transitioning to low-GWP refrigerants. Training should address known issues.

This study highlights the potential of air-to-water heat pumps to contribute to the decarbonization of multi-unit residential buildings. Success depends on setting realistic expectations, careful system sizing and design, strategic controls, rigorous maintenance, and ongoing optimization. Lessons from the FQSF retrofit provide valuable guidance to inform future projects and accelerate broader adoption.

1 Introduction

Air-to-water heat pumps are a promising retrofit technology for advancing the decarbonization of large buildings multi-unit residential buildings (MURBs). The French Quarters Shared Facilities (FQSF) consists of two condominium buildings located in Toronto, at 120 Lombard Avenue and 115 Richmond Street East. These 11-storey buildings, constructed in 2003, contain a total of 151 condo units. The need to replace a chiller that had reached end-of-life prompted the condominium board to install air-to-water heat pumps instead of a like-for-like replacement.

These heat pumps provide both heating and cooling. The system was designed to meet all cooling needs of the building and to offset a significant portion of the space heating provided by a natural gas boiler. The decision was motivated by both environmental and economic considerations: a desire to reduce carbon emissions and to mitigate long-term operating costs by diversifying the facility's heating energy sources.

The system was completed and commissioned in 2023. A comprehensive data monitoring system was installed to assess performance, quantify energy and cost savings, and capture lessons learned for future adopters. The monitoring system was set up to collect data for two full years, covering Summer 2023, Winter 2023/24, Summer 2024, and Winter 2024/25. However, Summer 2023 was omitted from the analysis because the system was still being commissioned. This report summarizes the key findings and lessons learned from the monitoring.

2 System Description

The FQSF condominiums are served by a two-pipe central hydronic system that provides both heating and cooling. Seasonal switchover occurs in May and October to reconfigure the system between heating and cooling modes. The building's primary hydronic loop is connected to forced-air hydronic fan coil units.

Prior to the installation of heat pumps, the primary hydronic loop was heated by a condensing boiler[1] and cooled by an air-cooled chiller.[2] As part of the retrofit, the chiller was replaced by heat pumps, while the boiler was retained to provide backup heating capacity.

Figure 1. (Top) FQSF consists of two condo buildings sharing a common space heating system. (Bottom) ASHP installed on rooftop.

The 2-stage heat pumps are installed in parallel and integrate into the building as shown in Figure 2. HP1 is a Climaveneta (Mitsubishi Electric) NX-N-G02-U-0812P. HP2 is a NX-N-G02-U-0662P. Rated heating capacity outputs are 248 kW (846 kBtu/h) and 192 kW (655 kBtu/h), respectively. Rated cooling capacities are 226 kW (771 kBtu/h; 64 Tons) and 180 kW (614 kBtu/h; 51 Tons), respectively.[3]

Each heat pump serves a hydronic circuit filled with 40% propylene glycol. Heat is transferred from the glycol loop to a water-based hydronic circuit via a heat exchanger. As illustrated in the schematic, this configuration requires a pump on both sides of each heat exchanger. The original intent was to locate the heat pump connection upstream of the boiler to allow the heat pumps to preheat return water. However, the system was initially installed with the boiler positioned upstream and this was corrected in Fall 2024.

3 Controls

Heating and cooling in the building are coordinated by a building automation system (BAS). The BAS determines which heat pump(s) are permitted to operate and provides them with a target setpoint glycol

[1] Camus DynaFlame DFNH-2002-MGI; 2,000,000/1,900,000 Btu/h max. heating input/output; 400,000 Btu/h min. output.
[2] York YCAL0114EC; 111 Tons (Nominal) cooling capacity; IPLV: 13.7 EER; 9.8 EER (2.88 COP) at 95 °F (35°C).

[3] Rated in accordance with AHRI Standard 550/590. Rating conditions are discussed in Section 6.1 and 9.4.

temperature. Each heat pump then uses its internal control logic to regulate the glycol loop temperature. This includes decisions such as operating in high- or low-stage, selecting which compressor to run in low-stage, initiating defrost cycles, and turning on or off.

In heating mode, a heat pump shuts off when its supply temperature exceeds the setpoint by a user-defined margin and will turn back on once the temperature falls below the setpoint by a specified amount. A similar control strategy was applied during the cooling season.

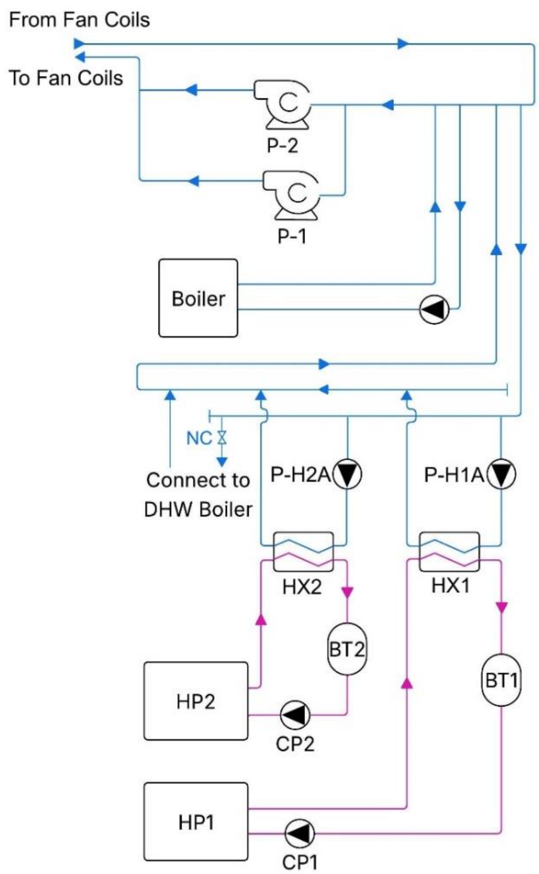

Figure 2. The heat pumps were connected in parallel. In heating mode, each heat pump heated a separate glycol loop containing a buffer tank (BT1 & BT2). Heat was then transferred to the primary loop via heat exchangers (HX1 & HX2). This set-up required two glycol pumps (CP1 & CP2) and two "injector" pumps (P-H1A & P-H2A).

During Winter 2023/24, the heat pumps were not configured to operate concurrently in heating mode. This continued for most of Winter 2024/25 due to operational issues with HP1 (discussed in Section 13). However, toward the end of Winter 2024/25, the issues were resolved, and concurrent operation was enabled.

In Summer 2024, the system was configured to allow concurrent operation of both heat pumps, which was necessary to meet the building's cooling demand. During the cooling season, the heat pumps were generally unable to operate in high-stage. Poorer heat transfer across the heat exchangers meant that the heat pumps would quickly cool the glycol loop and trigger protection mode due to low fluid temperatures. To avoid this, both heat pumps were operated in parallel at low-stage in periods of high cooling demand. This is discussed further in Section 9.1.

4 Monitoring Description

A wireless data acquisition system from Monnit was used for monitoring. Data was collected at one-minute intervals. Monnit temperature sensors were deployed to measure temperature, while Monnit pulse counters recorded outputs from third-party sensors used to monitor energy consumption and flowrate. Sensors were verified using custom setups at STEP's Archetype Sustainable House (ASH) Lab prior to deployment. A detailed description of the monitoring system is provided in the Report Addendum, while a high-level summary of the monitored points is outlined below:

Boiler & Primary Loop

- Water flow rate
- Water supply and return temperatures

HP1 & HP2

- Glycol flow rate
- Glycol supply and return temperatures
- Heat pump electricity consumption
- Glycol circulator pump electricity consumption

Other

- Outdoor temperature and humidity
- Transformer primary electricity consumption (the heat pumps required a transformer for the necessary voltage)

The monitoring system was sufficient to determine energy inputs and outputs, enabling the calculation of heating efficiency, capacity, and total heating output for the heat pumps, boiler, and overall system. Redundancy was built into the monitoring approach—heating energy was measured both at the equipment level and at the primary loop—providing a means of cross-validation to improve confidence in the results.

5 Analysis Overview

Heating capacities of the heat pumps and boiler were calculated at one-minute intervals using measured temperatures and flowrates. These were adjusted using the total heating output measured at the primary loop to ensure the sum of individual outputs of each component matched the system-level output.

Data was then aggregated hourly, daily, monthly, seasonally, and for the monitoring period. The coefficient of performance (COP) was calculated as total thermal energy output divided by total electrical energy input, both including and excluding the glycol circulator and injector pumps. Time-series plots, scatterplots, and histograms were used to evaluate performance and operation patterns.

Heating season gas savings were determined using International Performance Measurement and Verification Protocol (IPMVP) Option B (All Parameter Measurement). Boiler efficiency and delivered heat were derived from monitoring, and baseline natural gas consumption was calculated as the natural gas required to meet the measured heating load at the measured efficiency. Savings were the difference between baseline and actual natural gas use.

Electricity increases during heating season were assessed using IPMVP Option A (Key Parameter Measurement), based on metered consumption from the heat pumps and circulator pumps. IPMVP Option A was also used to evaluate cooling-mode electricity savings. Submetering of the previous chiller was used to establish a baseline electricity use relative to cooling degree days (CDDs), which was then compared to the actual heat pump energy consumption.

Modelling was used to estimate cost savings under alternative control strategies, including one that optimized heat pump use to achieve lowest cost of operation. Time periods were removed from the analysis due to data loss on key sensors. Overall, 91% of the 2023/24 heating season from October 20th to April 30th was included, and 84% of the 2024/25 heating season was included.

The main body of this report summarizes key findings. Recommendations are provided alongside findings to position the results in a way that is actionable for prospective system owners, decision-makers, designers, and others. Detailed IPMVP analysis results and additional information are provided in the Report Addendum.

6 Findings: Heating COP & Utilization

6.1 Heating efficiency & outdoor temperature

Figure 3 shows the hourly-aggregated COP (excluding circulator pump energy) plotted against mean daily outdoor temperature. As expected, heating efficiency decreased as outdoor temperatures dropped. The rated COP of 3.0—based on water as the heat transfer fluid, 8.3°C outdoor temperature, 43.3°C return water temperature, and flow rates of 155 GPM (HP1) and 124 GPM (HP2)—is also shown. However, actual conditions differed: glycol was used instead of water, return temperatures varied due to the outdoor reset curve, and flowrates were lower (HP1: 121–130 GPM; HP2: 100–111 GPM). Hourly COP values also reflect cycling and defrost losses, and increasing return temperatures as the outdoor temperature decreases. Under similar outdoor temperatures to the rated conditions, the actual COP ranged from 2.5 to 2.6 for both heat pumps.

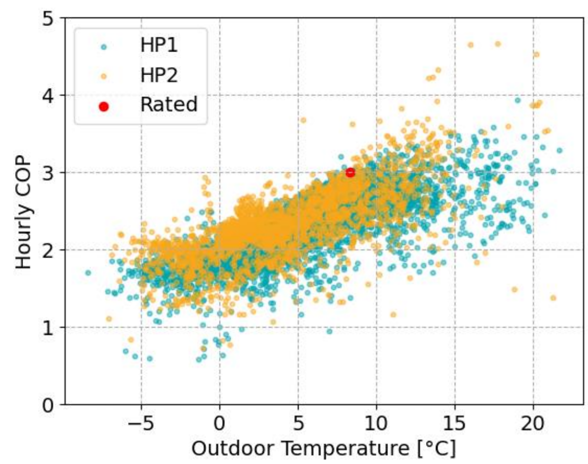

Figure 3. Daily aggregated COP is plotted for each heat pump. It includes losses from real-world factors like cycling and defrost.

Recommendations

1. ***Consider real-world COP in feasibility assessments:*** *Feasibility assessments should account for the difference between the rated COP and the real-world COP achievable in practice. The rated COP, which is determined under ideal conditions, may not reflect the performance in an actual installation due to a variety of factors.*

6.2 Heating COP & cycling

The amount of time the heat pump remains on in response to a heating call—referred to in this report as the "cycle time"— correlated with COP, shown in Figure 4. The equipment needs to "warm up" and time is required for the electronic expansion valve to find its optimal position. This relationship was *most apparent when the mean outdoor temperature was below 5°C. These days are plotted in Figure 4*. HP2 was more affected by short cycle times, although the reason for this discrepancy was not identified.

Figure 5 highlights outdoor temperature as a major influence on cycle time. The plot uses a logarithmic scale where 10^0 represents 1 minute, 10^1 represents 10 minutes, and 10^2 represents 100 minutes. In warmer conditions, cycle times for both heat pumps are typically short. Time-series data in Figure 6 shows an example of short cycle times. These shorter cycles are not ideal for equipment longevity. As temperatures drop, cycle times tended to increase.

The primary issue behind the short cycle times was identified as the buffer tank sizing. The mechanical room is space constrained, and the tanks are small, at 115 gallons each. Technically, this is near the manufacturer-specified minimum required circuit volume of 150 gallons and 120 gallons for HP1 and HP2, respectively. However, the flowrate through the heat pumps is 100 to 130 GPM. The data showed that within ~1-minute, heated glycol from a heat pump's supply arrived at the return. The low volume of the glycol circuit caused the glycol temperatures to increase rapidly, triggering the heat pump to shut off.

The project team is exploring potential control solutions to address this issue. Future projects should avoid it through larger buffer tank sizing. Section 12.5 discusses how hydronic circuit design could be improved for greater buffer tank capacity.

Recommendations

2. ***Consider strategies to foster longer cycles:*** *Technical practitioners involved in the design and controls of air-to-water heat pump systems should consider strategies to foster longer cycle times as this boosts overall system efficiency and promotes equipment longevity.*
3. ***Consider buffer tank sizing****: Larger buffer tanks promote longer cycle times. In space-contained environments, phase-change materials may help increase thermal storage capabilities.*
4. ***Implement minimum run time logic****: Equipment manufacturers should consider enforcing longer cycle times through a "desired minimum on time" control setting. This would widen the gap between on and off around the setpoint but it would allow the temperature to go higher in heating and lower in cooling. Efficiency also depends on temperature and there is a balance to be had between longer cycle times and optimal temperatures. The heat pumps in this study did have a minimum runtime parameter that will generate longer runtimes provided the unit is within safe operational limits.*

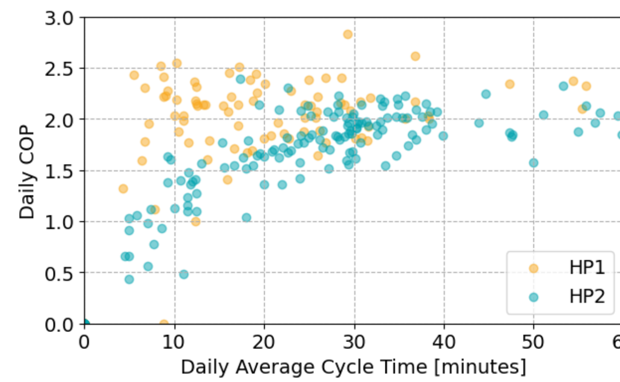

Figure 4. Short cycle times negatively impact COP.

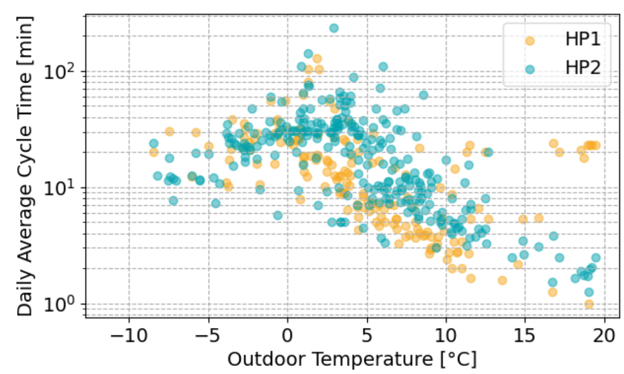

Figure 5. Short cycle times occurred predominantly in warm outdoor temperatures.

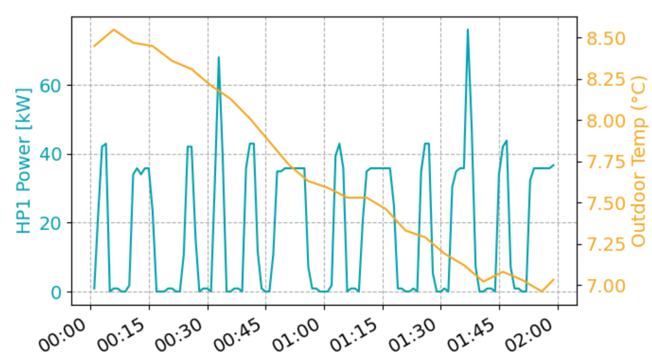

Figure 6. Time-series data over 2 hours showing short cycle times. Average cycle time for this period is 5 minutes.

6.3 Overall heating season COP

Figure 7 plots average annual heating energy inputs and outputs of the heat pumps. Results have been partly corrected for an issue with a fused fan contactor on HP1, discussed separately in Section 13.1. When considering only ASHP energy consumption, the overall COPs were 2.2 and 2.1 for HP1 and HP2. When including the electrical power consumption of the circulators, the COPs reduced to 1.7 and 1.8.

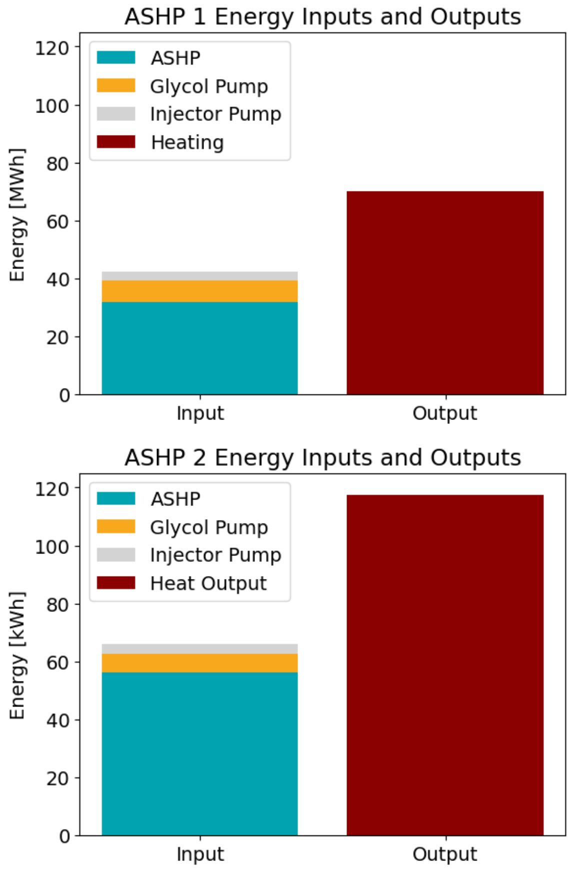

Figure 7. Average annual energy inputs and outputs are plotted for HP1 (top) and HP2 (bottom) considering Winter 2023/24 and 2024/25.

Recommendations

5. ***Consider all electrical loads**: Feasibility assessments should consider the electricity consumption of all components required to run heat pumps. The energy required by circulators is not insignificant for overall efficiency.*

6. ***Consider other topologies**: Manufacturers should consider developing system topologies (like air-to-water systems with split refrigerant circuits) that negate the need for extra circulators and eliminate efficiency losses related to the secondary loop.*

6.4 Heating season ASHP utilization

Figure 8 shows that the ASHPs provided 69% of the heating for the primary loop, with the boiler providing the remaining 31%. HP2 provided notably more heating than HP1 because HP1 was periodically out of commission in Winter 2024/25 due to issues discussed in Section 13. Figure 9 shows annual ASHP utilization with respect to the outdoor temperature. The ASHPs provided most of the heating down to an outdoor temperature -2°C. ASHP utilization improved near the end of Winter 2024/25, as discussed in Section 6.5.

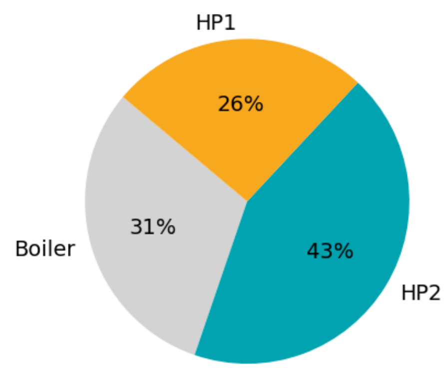

Figure 8. Breakdown of heating energy provided to the primary loop considering both winters.

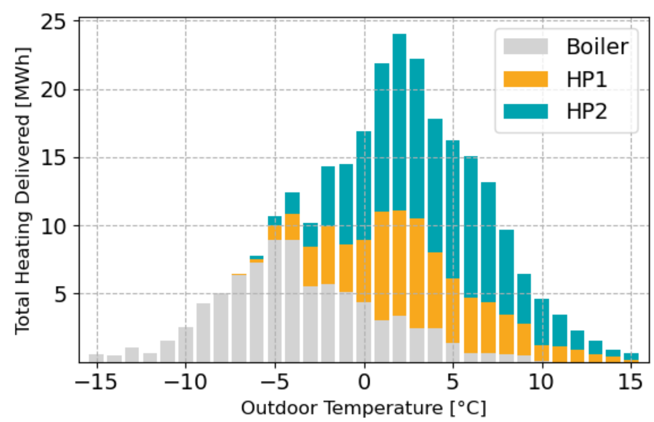

Figure 9. Utilization of ASHPs with respect to outdoor temperature.

Recommendations

7. ***Expect that an air-to-water heat pump can provide a substantial level of heating**: MURB owners may expect that well-sized air-to-water heat pumps may supply most of the space heating in a cold climate—in this case, more than two-thirds of the space heating needs.*

6.5 Achievable utilization

Figure 9 presented data from the full monitoring period across both winters, during which various control parameters were adjusted and system issues were identified and resolved. In contrast, Figure 10 focuses on a two-month period at the end of Winter 2024/25 including March and April 2025, which is more representative of a properly functioning and optimized system moving forward.

Figure 10 shows that the heat pumps were capable of meeting nearly all heating demand down to an outdoor temperature of -3°C. This was 87% of the heating load for this time period. At -4°C and -5°C, they provided a smaller share of the total heating. This was often due to periods when the heat pump could not meet the supply temperature setpoints for the primary loop but had not undershot the setpoint for long enough to activate the boiler. Below -5°C, the boiler was required to meet heating demand.

Note that the heat pumps can provide up to a 50°C supply temperature down to a -5°C outdoor temperature. However, this supply temperature is not translated into actual supply to the building due to the use of glycol and external heat exchangers.

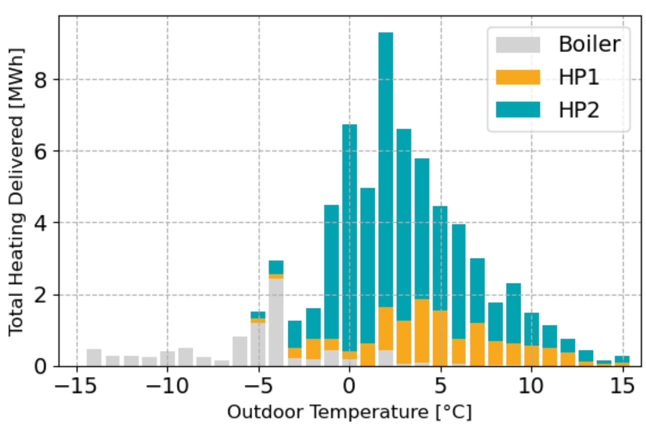

Figure 10. With more optimized control and fully functioning heat pumps, the heat pumps were capable of meeting nearly all of the building's heating load down to an outdoor temperature of -3°C.

7 Findings: Heating Cost & Carbon

7.1 Demand increase

Monthly electricity demand was assessed using post-retrofit utility bills and pre-retrofit 15-minute interval data. Monitoring data from submetered systems was not used, as demand charges are based on the peak draw of the *entire facility*. Demand was defined as the highest average facility-wide power draw over a 15-minute period each month. Pre-retrofit, this was determined based on interval data. These monthly peak demands were then averaged across the heating season (October to April).

Post-retrofit demand was taken directly from utility bills. Before the retrofit, the facility's average monthly demand during the heating season was 162 kW. After the retrofit, this increased by 44 kW to 206 kW. The facility's highest monthly demand pre-retrofit occurred in August at 244 kW. Post-retrofit during the heating season, the highest monthly demand was 225 kW, occurring in January 2024. It follows that the highest heating season monthly demand post-retrofit did not exceed the facility's historical summertime maximum demand.

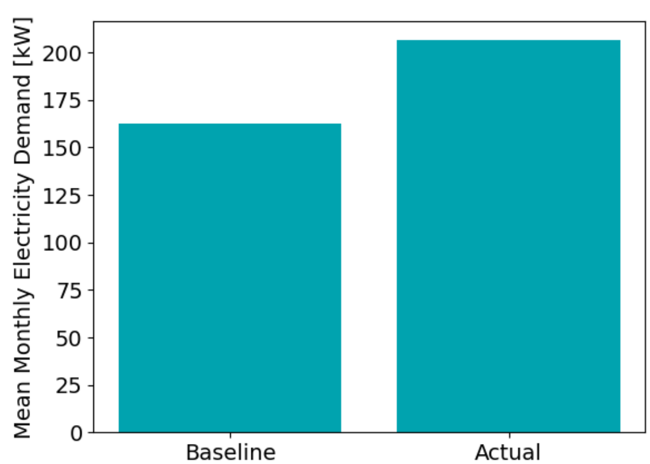

Figure 11. The mean monthly electricity demand during the heating season increased after the retrofit.

7.2 Electricity increase

The annual electricity increase due to the heat pumps in the heating season is shown in Figure 12 broken down by source. This figure has been corrected to remove the unnecessary outdoor unit fan consumption discussed in Section 13.1. The correction removed 8 MWh from the heat pump energy consumption. The heat pumps themselves consumed 87 MWh, which was 81% of the total electricity increase. The glycol circulators consumed 14 MWh (13%) and the injector pumps consumed 6 MWh (6%).

The injector pumps were running continuously and were not directly monitored. The injector pump electricity consumption is based on the nameplate power consumption. Glycol circulators were partially interlocked with the heat pumps. The BAS would select either heat pump as the lead to meet the load, and the corresponding glycol circulator ran more-or-less continuously while the glycol circulator for the other heat pump was turned off. Glycol circulator pump optimization is discussed in Section 8.6.

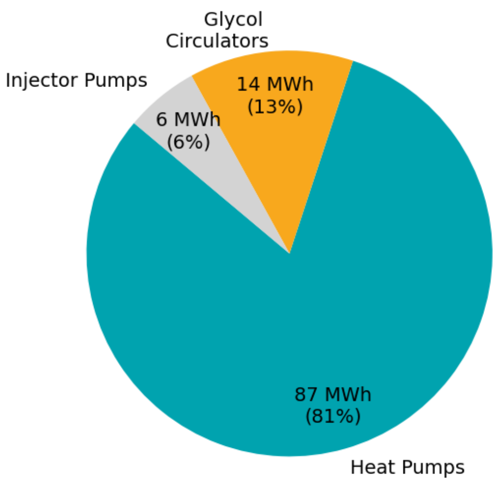

Figure 12. The electricity use increased in the building as a result of the heat pumps, glycol pumps, and injector pumps.

7.3 Electricity cost increase

The electricity cost increase has two components: the cost for increased electricity demand and the cost for increased electricity consumption. The electricity increase of the heat pumps was weighted in each TOU similarly for the full building consumption – approximately two thirds in an off-peak TOU, with the remainder split between the peak and mid-peak TOU brackets.

Utility bill analysis was used to estimate that the weighted electricity cost for this distribution was 13.3 ¢/kWh during the monitoring period. This includes the per kWh cost weighted according to TOU, distribution charges, regulatory charges, HST, and the Ontario Energy Rebate. The demand rate was 6.76 $/kW per month and this includes transmission connection and network charges.

In total, the cost of the increased electricity consumption was $15,900, with 90% of the increased cost coming from the per-kWh charges and 10% coming from the increased demand charges.

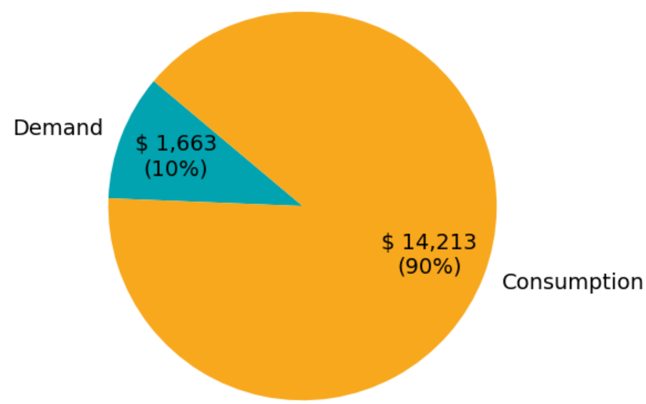

Figure 13. The electricity cost increases were composed of demand charges and per-kWh consumption charges.

Recommendations

8. ***Consider electricity demand costs:*** *When assessing the potential operating cost changes from a heat pump retrofit, it is important to consider the cost impacts of increased demand charges.*

7.4 Gas reduction

Figure 14 shows that, using IPMVP Option B, the gas consumption of the boiler was reduced by 20,900 m^3 annually and natural gas costs were reduced by $9,200. This is substantial, but because the space heating boiler was only a portion of the total gas load of the building, it resulted in a relative gas savings of 19% for the full building. This is shown in Figure 15.

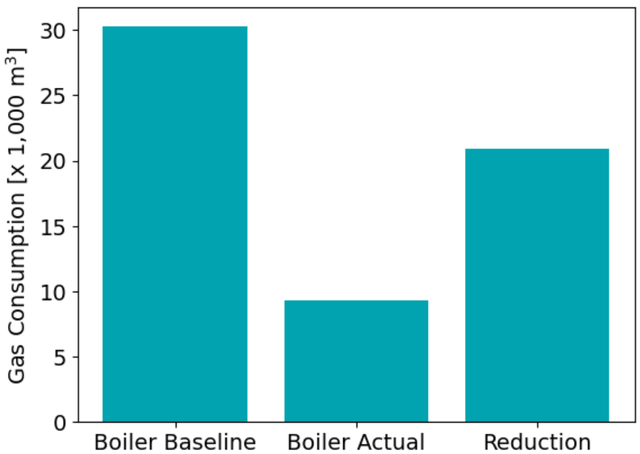

Figure 14. The heat pumps reduced the consumption of the space heating boiler.

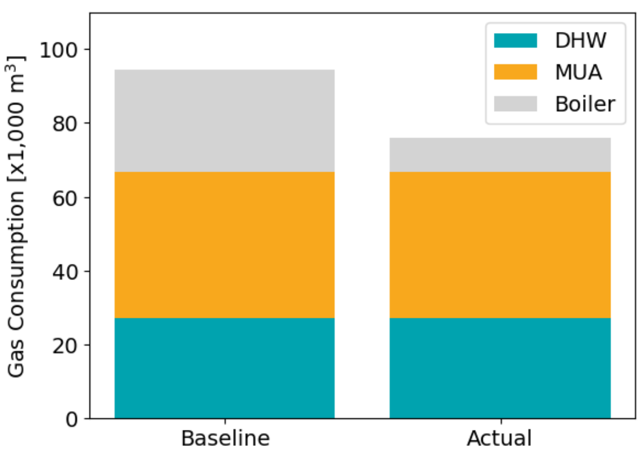

Figure 15. The relative gas reduction for the full building was lower because the space heating boiler is only one component of the total building gas consumption.

Recommendations

9. ***Understand the gas-use breakdown:*** *Future system owners should seek to estimate how gas consumption breaks down in their facility. This will inform other decarbonization and savings opportunities and set reasonable expectations for gas bill reductions.*

7.5 Net operating cost change

The mean gas rate was determined from the post-retrofit billing to be 44 ¢/m^3. Applying the electricity rates to the electricity increase, and the gas rate to the gas savings, it was estimated that there was an average annual cost increase of $6,700 for the two heating seasons that were evaluated. For the sake of comparison, the full utility cost for the building including electricity ($132,958) and gas ($53,887) was $186,845 for 2021 (pre-retrofit). The relative cost increase due to the heat pump retrofit for the full bill is then on the scale of a few percent.

The key factors impacting the relative cost savings or increase are the heat pump COP, the electricity rate, and the gas rate. For a gas rate of 44 ¢/m^3 and an average electricity rate of 13.3 ¢/kWh, the heat pump would need a COP near 2.7 to break-even on costs with a boiler that was providing heat at an efficiency of 86%. Effectively, for the utility rates that occurred during the heating season, the heat pump COP was not high enough to drive a heating season cost savings with the control strategy that was used. A cost-optimized control algorithm would have limited heat pump operation to off-peak TOU brackets where electricity is lower-cost and in milder conditions where efficiency is higher. This is explored in Section 8.1.

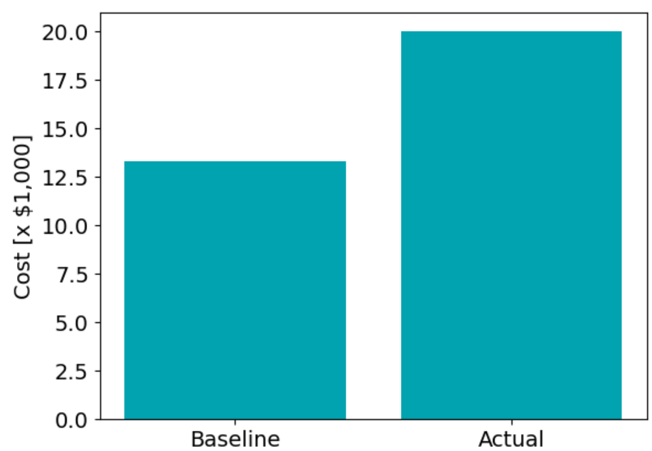

Figure 16. Post-retrofit, there was a net cost increase to provide space heating to the building.

Recommendations

10. ***Recognize the cost impacts of control strategies:*** *Future system owners should understand that optimizing for ASHP utilization—rather than cost—can increase operating costs.*

7.6 Carbon emissions

A marginal emission factor (EF) of 84 g CO_2e per kWh was assumed, based on TAF's projections for 2024.[4] The EF of natural gas consumption was assumed to be 1,927 g CO_2e per m^3. The carbon reduction from natural gas savings was 40 t CO_2e and the carbon increase from the new electricity load was 9 t CO_2e, yielding a net carbon reduction of 31 t CO_2e per year, considering only heating mode operation.

[4] TAF. Ontario Electricity Emissions Factors and Guidelines, 2024 Edition. https://taf.ca/publications/ontario-electricity-emissions-factors-2024/

8 Findings: Heating Season Control Optimization

8.1 Operating costs

The heat pump control strategy prioritized the heat pumps as much as possible to reduce carbon emissions, with the boiler used only when the heat pump could not meet the load. This approach was not intended to optimize utility costs. Alternate control strategies were assessed with modelling. The modelling used empirical data from the 2023/2024 heating season to define the building's hydronic heating load. Electricity rates were fixed at 10.8 ¢/kWh (off-peak), 14.1 ¢/kWh (mid-peak), and 20.0 ¢/kWh (on-peak), reflecting current marginal electricity costs with all electricity bill line items. Various natural gas rates were considered.

System COP, including heat pumps and circulators, was modeled based on empirical data. The model evaluated the cost of meeting heating demand using either the heat pump or the boiler and applied different control strategies. Four control strategies were modeled:

1. boiler-only operation;
2. maximize heat pump use;
3. operate heat pump only during off-peak periods;
4. operate heat pump only when it's lower cost.

Each scenario was run across a range of gas rates. Outputs included the fraction of heating provided by the heat pump (Figure 17) and relative cost savings compared to boiler-only (Figure 18). Results showed there is a trade-off between the amount of heating load met by the heat pump and the overall utility costs.

Operating the heat pump as much as possible produces the greatest cost increases and does not achieve savings until gas rates exceed ~65 cents/m³. Operating the heat pumps only in an off-peak TOU bracket significantly mitigates utility cost increases while still meeting much of the heat load using the heat pumps. Cost-optimized control always matched or beat boiler-only costs but it prevented heat pump use until gas rates increased above ~45 cents/m³.

Gas rates have fluctuated significantly over the past several years. During planning for the heat pump installation, they peaked at greater than 60 cents/m³. They've since lowered significantly with the removal of the Federal Carbon Charge. Overall, it is important to understand that cost savings are a product of system efficiency, utility rates, and control strategy. Rates vary over time, but the control strategy is user-selectable.

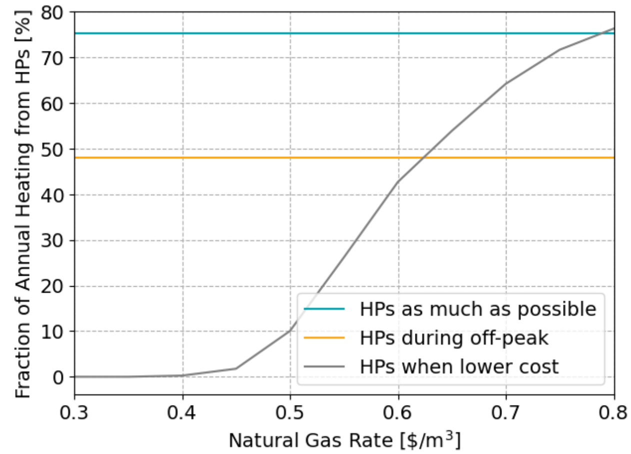

Figure 17. The modelled fraction of annual heating from the heat pump is shown for different control approaches.

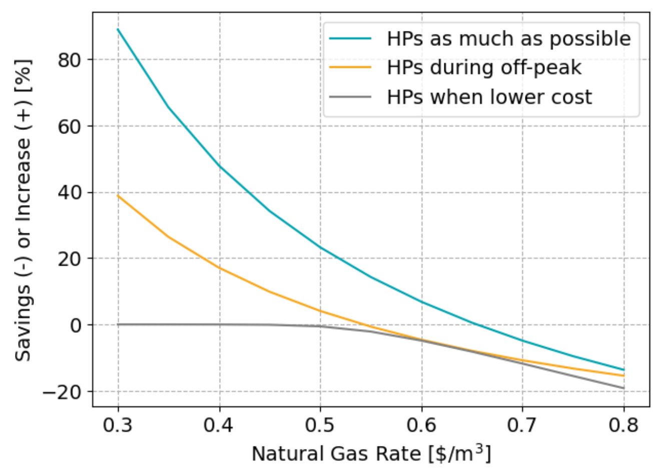

Figure 18. Modelled cost increases (+ values) or savings (- values) are shown for different control approaches.

Recommendations

11. ***Select a control strategy aligned with project goals:*** *Prospective system owners should recognize that control strategies may prioritize carbon reductions or operating cost savings and that these may conflict. Controls that allow different strategies should be part of the initial installation.*

12. ***Support time-of-use (TOU) electricity pricing:*** *Policy-makers should consider maintaining or expanding TOU electricity pricing. These rate structures can enable cost savings sooner or otherwise provide the opportunity to significantly mitigate increases.*

8.2 Outdoor reset curve

One of the key factors limiting air-to-water heat pump utilization is the maximum supply temperature they can deliver. During Winter 2023/24, the outdoor reset curve governing the primary loop was reduced, and it was further lowered early in Winter 2024/25 (see Figure 19). These adjustments were made iteratively, with each change monitored for impacts on tenant comfort. Initial and final reset curve setpoints are below.

Table 1. Original and final reset curve setpoints

Outdoor Temp [°C]	Original Reset Curve [°C]	Final Reset Curve [°C]
16	34	-
12	-	32
-10	65	55
-20	75	70

The revised reset curve enabled the heat pumps to manage the heating load under colder conditions, as shown in Figure 20. The figure plots the hourly averaged return temperature of the primary loop (indicating the outdoor reset curve setpoints) alongside minute-level supply temperatures delivered by HP2 in second-stage operation. The data show that HP2 can supply glycol temperatures up to approximately 51°C, though this capacity declines when outdoor temperatures fall below −4°C.

At outdoor temperatures of −5 to −6°C, the boiler is used to meet the primary loop supply setpoint. In this range, the heat pump's *supply* temperature capability is equal to the primary loop *return* temperature. The heat pumps would not be able to drive heat into the primary loop with those return temperatures and therefore they could not meet the supply setpoint. In reality, the boiler needs to activate in warmer outdoor temperatures than −5 to −6°C because there is temperature loss across the heat exchangers.

If the reset curve had not been optimized, this crossover point would have occurred above 0°C—significantly reducing the heat pump's contribution to space heating.

Recommendations

13. ***Optimize the outdoor reset curve:*** *System owners/operators should aim to optimize the outdoor reset curve of the primary hydronic loop. This supports both efficiency and heat pump utilization. Ideally, this optimization should be completed prior to heat pump sizing, as the reset curve may be a key limiting factor for heat pump performance and should be reflected early in feasibility assessments.*

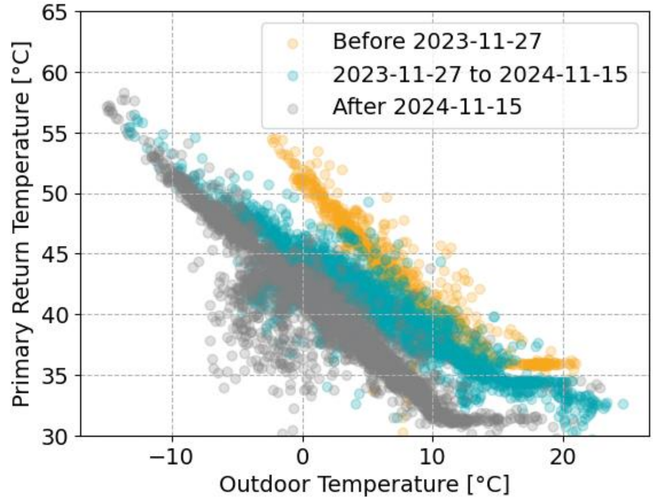

Figure 19. The trends in the primary loop return temperature show that changes were made to the outdoor reset curve of the primary loop early in Winter 2023/24, and again in Winter 2024/25. Lower primary loop return temperatures foster improved heat pump utilization and efficiency.

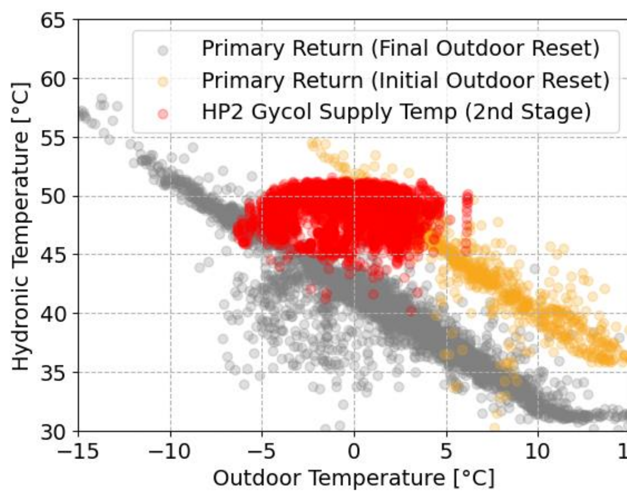

Figure 20. The primary loop return temperature with the initial and final outdoor reset setpoints is plotted alongside the HP2 supply temperature in high stage. When the HP2 supply temperature drops below the primary loop return temperature, it is a hard limit below which the heat pump can no longer provide useful heating. The outdoor reset curve optimization allowed the heat pump to heat the building in temperatures that were 5°C cooler than the original setpoints.

8.3 Primary loop flowrate

The pumps on the primary loop were approaching end-of-life and replaced with more energy efficient models between the two winters. Previously the primary loop was operated at a single flowrate of approximately 250 GPM. The new pumps had variable frequency drives and were configured to operate at different flowrates depending on the heating needs of the building (Figure 21), while also considering the flowrates required by the heat pumps on the water-side of the heat exchangers.

The new pumps consumed substantially less energy (Figure 22). The previous pumps would have consumed 23 MWh/year during heating season, while the new pumps would consume 12 MWh (Figure 23). This is a savings of 11 MWh ($1,600). If the new pumps were left at a constant flowrate, the consumption would have been 15 MWh. It follows that the energy savings are largely due to the improved efficiency of the new pumps, but the variable control also contributed to the energy savings.

Recommendations

14. ***Consider electricity savings opportunities**: To mitigate the electricity increase from a new heat pump installation, it is beneficial to consider other energy savings opportunities within a facility. Converting older pumps to higher-efficiency pumps with VFDs can generate notable energy reductions.*

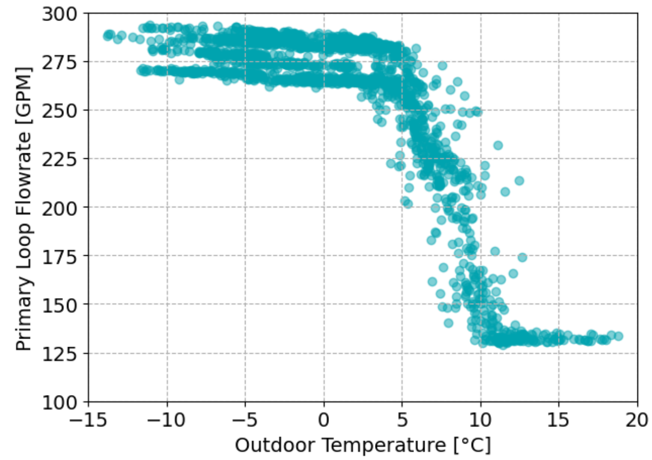

Figure 21. The new primary loop pumps were configured to vary their flowrate with the heating needs of the building.

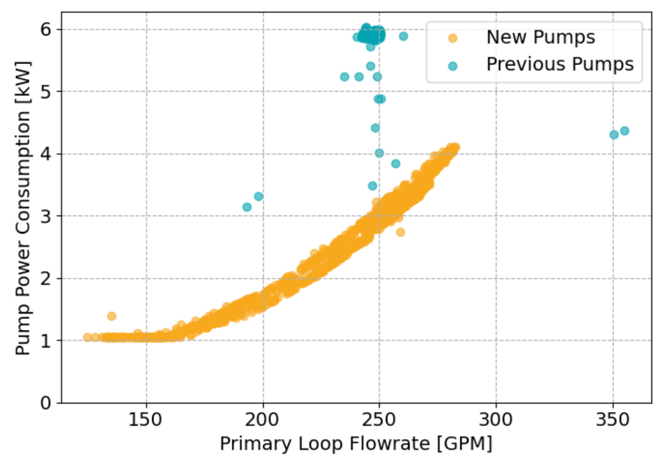

Figure 22. The new primary loop pumps consumed much less energy.

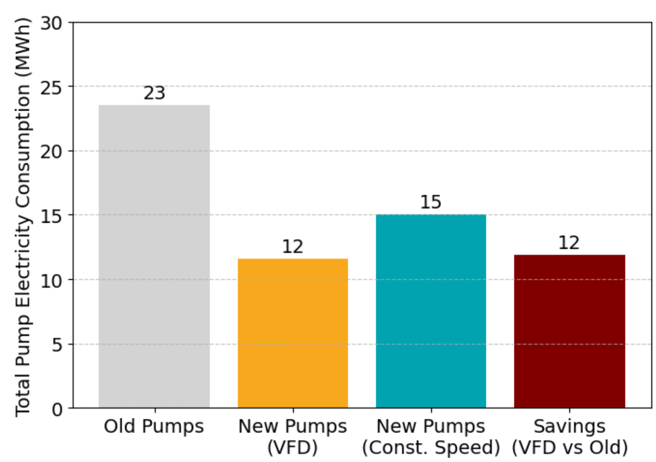

Figure 23. The new primary loop pumps generated notable electricity savings. This was mostly from an increase in efficiency, but also due to variable control.

8.4 Controlling heat pumps in parallel

Control strategies for cold-weather operation were evaluated to maximize heat pump use at low outdoor temperatures. One strategy tested whether both heat pumps operating in low stage could outperform a single unit in high stage. Performance could be increased because the heat exchanger surface area would increase and the temperature differential between the primary loop and glycol loops should be reduced — potentially improving primary loop supply water temperatures.

Figure 24 shows primary loop supply temperature versus outdoor temperature for several modes:

- **Only Boiler** (target supply temperature reference)
- **Both HPs (Low Stage)**
- **Both HPs (One in High Stage)**
- **Only HP1 (High Stage)**
- **Only HP2 (High Stage)**

By -2 to -3 °C, all heat pump scenarios begin to fall below target supply temperatures; by -5 °C, the shortfall is significant. Running both heat pumps in low stage produced primary loop supply temperatures marginally warmer (<1 °C) than HP1 alone in high stage. However, running both units with one in high stage did not result in the warmest temperatures—possibly because this mode was infrequently sustained for long periods.

Figure 25 shows that during parallel operation, one unit often cycled rapidly (average 7-minute cycles in the figure), reducing COP as noted in Section 6.2. This mode also required both glycol circulators, further lowering system efficiency.

In this installation, temperature—not capacity—is the limiting factor (discussed in Section 8.5). One heat pump in high stage could meet the load until supply temperature became a constraint. One heat pump has longer run times and reduced circulator pump energy – both of which would drive a better COP. In systems where capacity is the constraint, parallel operation may offer greater benefit.

It is possible that a single heat pump operating in high-stage may enter defrost more often, briefly lowering glycol temperatures enough to trigger boiler backup. The project team is still exploring this consideration further and identifying potential control refinements to address this behavior.

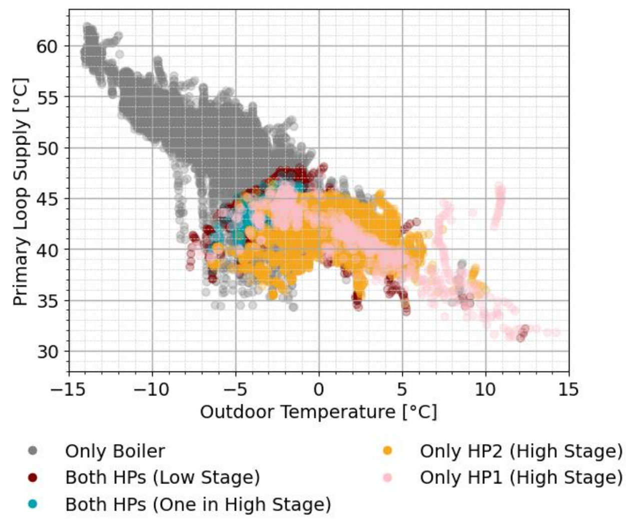

Figure 24. Starting at -2°C to -3°C, different heat pump operational modes began to fall short of the target supply temperature for the primary loop.

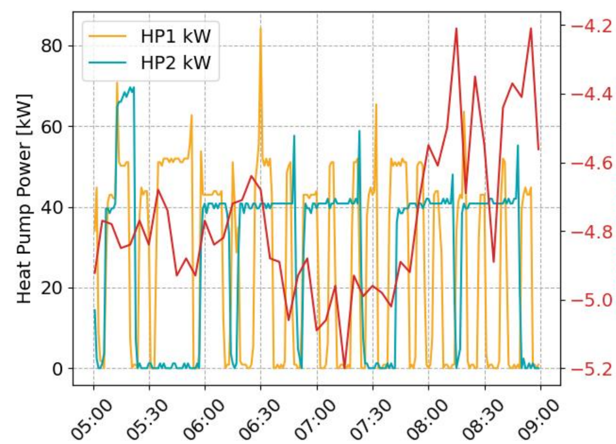

Figure 25. When both heat pumps operated in parallel shorter cycles (here on HP1) were observed.

Recommendations

15. ***Favor high-stage operation of a single unit:*** *In this installation, prioritizing running one heat pump (preferably the larger one) in high stage is likely preferable to operating both units in a low-stage. This approach delivered similar capacity and primary supply temperatures, while avoiding short cycle times and additional pump energy use.*
16. ***Monitor and minimize short cycle times:*** *Short cycle times, particularly when both heat pumps are on in cold temperatures, can significantly degrade system performance. Control logic should ensure reasonable cycle durations.*
17. ***Match control strategy to system limitations:*** *Installations limited by heat pump capacity may benefit more from operating heat pumps in parallel.*

8.5 Preheating for boiler

One of the design objectives of placing the heat pumps upstream of the boiler was to allow the ASHPs to preheat water for the boiler. This approach might extend heat pump operation into colder temperatures where they lack sufficient capacity alone.

Figure 26 shows the total heating load on the primary loop versus outdoor temperature, using four-hour averaged data. The heating load was calculated as the sum of output from the heat pumps and boiler. Also shown is the hourly maximum capacity of HP2 in both high- and low-stage operation. The figure indicates that HP2 has enough capacity to meet the heating load down to approximately -4°C. Additional capacity from HP1 may support operation at lower temperatures.

However, as shown in Figure 24Figure 26, the primary limitation is not capacity, but supply temperature. When the system switched to the boiler, it was because the heat pumps could not supply the needed temperatures to meet the primary loop setpoint.

Greater potential for preheating may occur in other buildings. Notably, this building had a relatively low temperature differential between the supply and return of the primary loop, often between 1 and 3°C. If the temperature differential were greater, and the return was cooler, there is greater potential for preheating. This differential can be adjusted with optimization of the primary loop flow – but multiple factors must be considered.

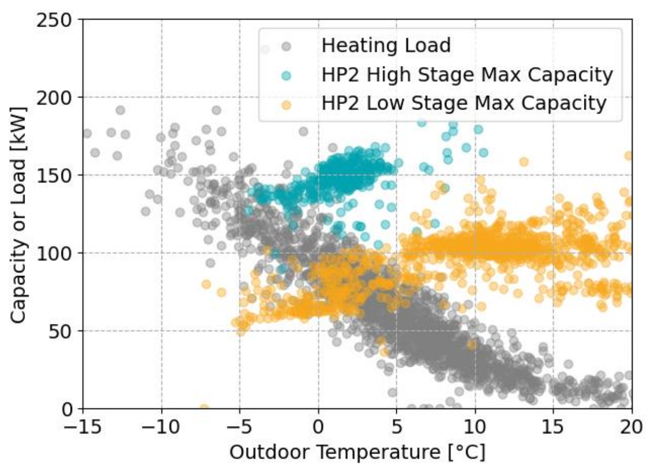

Figure 26. HP2 capacity is plotted alongside the building load. This illustrates that the total capacity of the heat pumps is not a constraint, since one heat pump alone has enough capacity down to approximately -4°C. Since the constraint is supply temperature, preheating does not likely have significant benefits for the system.

Recommendations

18. ***Assess whether heat pump constraints are capacity- or temperature-driven:*** *Preheating strategies may only offer benefits if capacity—not supply temperature—is the limiting factor. In this case, supply temperature was the main constraint. Designers should evaluate from the outset whether system limitations are likely to be temperature- or capacity-driven to determine whether preheating has practical value.*

8.6 Interlocking pumps

At the start of Winter 2023/24, the glycol and injector pumps operated continuously. Early in the season, glycol pump control was partially optimized. Heating during any period was typically only from one of the heat pumps and only the circulator for the active heat pump remained on. However, glycol circulators were not interlocked with the heat pumps. They continued to run as the heat pump cycled on and off.

There is a benefit to allowing the glycol pump to continue running after the heat pump shuts off—so residual heat in the loop can be transferred— but data showed that this heat dissipated within minutes. The data analysis flagged points when this residual heat had fully dissipated, allowing the injector and glycol pump energy use to be isolated to periods when they were actively transferring heat to the primary loop. This filtered data is shown in Figure 27.

With fully interlocked control—where circulator pumps operate only when their associated glycol loop is actively transferring heat—circulator energy use could be reduced by approximately 66%. This would amount to a savings of 14 MWh and $1,800 annually, considering only heating season operation.

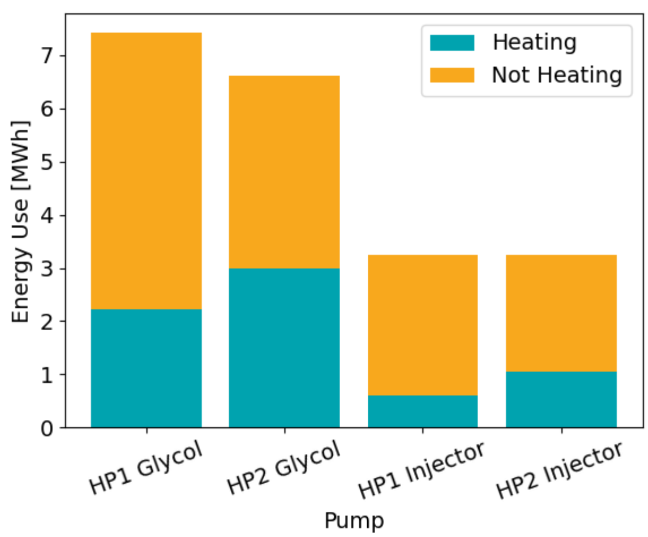

Figure 27. The circulator pump energy consumption was broken into periods when the glycol loops were providing active heating to the water loop ("Heating"); and when they were not ("Not Heating").

Recommendations

19. Interlock circulator pumps: *Insofar as possible, pumps should be turned off when not needed. This may offer a notable energy and cost reduction.*

8.7 Make-up air optimization

Based on the measured full building and boiler gas consumption, the natural gas loads were disaggregated. As shown in Figure 28, the MUA was the largest pre-retrofit load. In early 2024, the MUA schedule and setpoints were reduced—still within accepted standards—resulting in a ~25% reduction in MUA gas use (Figure 29). There were further changes in the 2024/25 heating season which brought the MUA consumption down and then up again. These adjustments aimed, in part, to shift heating load onto the heat pumps.

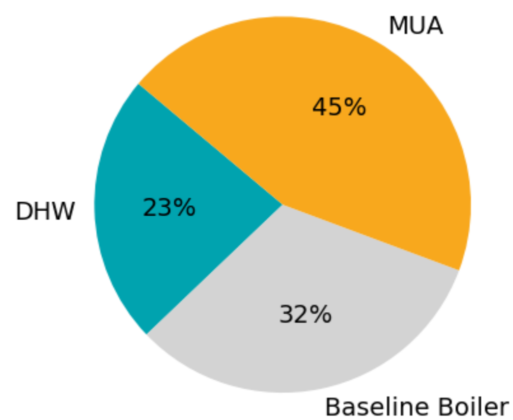

Figure 28. Estimated pre-retrofit gas use breakdown.

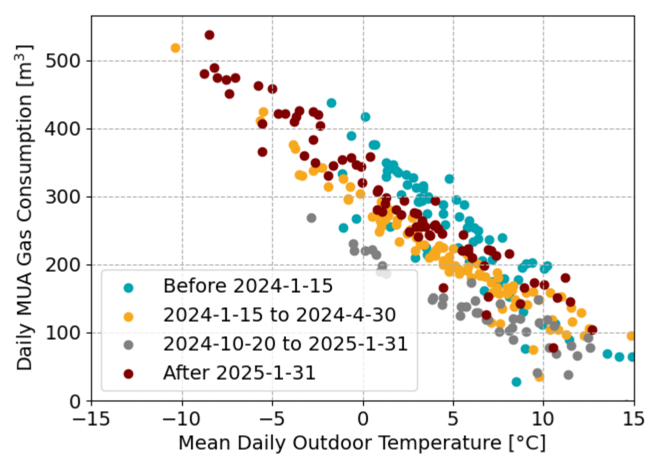

Figure 29. MUA setpoints were re-configured throughout the monitoring period as evident through the different MUA gas consumption trends.

Recommendations

20. Optimize other natural gas loads*: A heat pump retrofit provides a good opportunity to assess and optimize other natural gas loads in a building.*

8.8 Holistic cost savings

Section 7.5 showed that, at observed utility rates, the heat pump retrofit led to a cost increase. However, a primary loop circulator upgrade yielded a savings (Section 8.3) and circulator pump optimization could reduce costs further (Section 8.6). Section 8.1 showed that limiting heat pump operation to off-peak TOU periods could still achieve substantial heat pump utilization while notably reducing utility cost increases.

By considering these improvements as a bundled strategy—heat pump retrofit, circulator upgrade, and optimized control—the cost impact curves from Section 8.1 were recalculated and are presented in Figure 30. It shows that (1) with off-peak-only heat pump control, holistic savings are achievable across nearly all feasible gas prices; and (2) with maximum heat pump use, cost increases are substantially reduced, and net savings are realized at lower gas rate. An optimized and holistic design and control approach, considering additional upgrades, is therefore key to ensuring cost savings overall.

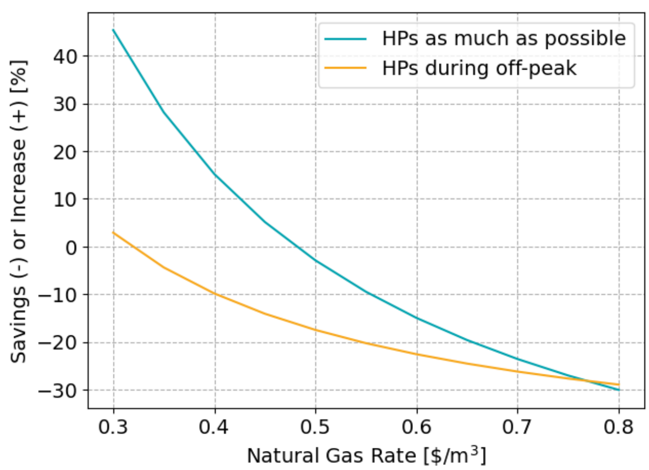

Figure 30. By considering additional cost optimization measures for the circulator pump control, and the already achieved savings from the primary loop pumps, it is possible to consider potential savings with different control approaches.

Recommendations

21. ***Bundle heat pump with other efficiency upgrades and optimize controls****: To maintain or reduce operating costs when adding a heat pump, consider additional energy efficiency upgrades, optimize the system, and choose a control strategy that is more cost-optimized.*

9 Findings: Cooling Capacity & COP

9.1 Initial cooling capacity issues

In May/June 2024, the system in its original configuration was unable to meet the primary loop temperature setpoints. Two main issues were identified. First, the original outdoor reset curve for cooling was too warm to enable effective cooling in mild weather. Figure 31 shows June data when only HP2 was operating. In mild conditions, the primary loop supply temperature was 13°C, which led to occupant complaints of insufficient cooling. This setpoint was later lowered to 11°C in July.

Second, the control strategy prioritized using a single heat pump if possible. A heat pump would start in first stage and activate second stage if needed, based on its internal controls. However, one heat pump alone couldn't meet the cooling load under warm or hot conditions. As shown in Figure 31, while HP2 drove the glycol loop to progressively colder temperatures, the primary loop supply remained relatively steady once outdoor temperatures exceeded ~20°C.

A one-hour snapshot of HP2 operation in June (when only HP2 was on) illustrates this further (Figure 32 and Figure 33). Cooling capacity is shown as negative values. At 15:14, HP2 switched from first to second stage, causing glycol temperatures to drop. After ~25 minutes, the glycol supply reached 2°C and hit the equipment protection limits. At that point, there was a 7°C differential between the HP2 and primary loop supply temperatures.

Much of the cooling did not transfer to the primary loop. Instead, the glycol loop continued to get colder until it hit its operational limit, preventing continued second-stage operation. A likely contributing factor is that the heat exchangers could not transfer sufficient cooling capacity.

Recommendations

22. ***Start with a warmer outdoor reset curve in cooling mode****: The curve can then be adjusted based on observed performance and occupant feedback.*
23. ***Ensure heat exchangers are adequately sized*** to transfer the full cooling capacity—particularly during second-stage operation—when glycol is used as the heat transfer medium

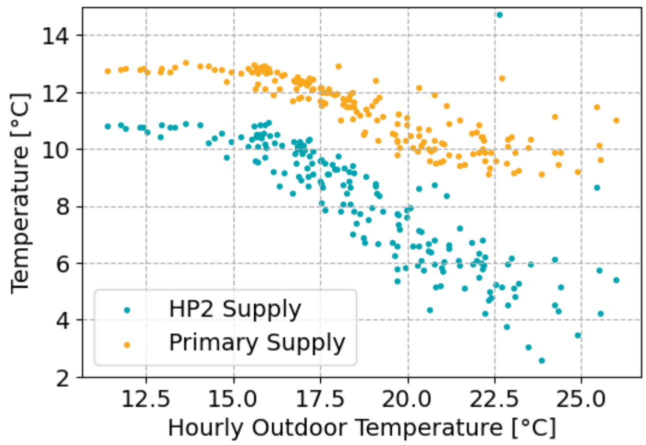

Figure 31. This data is from May/June 2024 when only HP2 was operating. In mild conditions, the primary loop supply setpoint was too warm to provide adequate cooling. In warm and hot outdoor temperatures, the primary loop supply was near a constant 10°C (indicating insufficient cooling) despite HP2 producing colder and colder glycol temperatures.

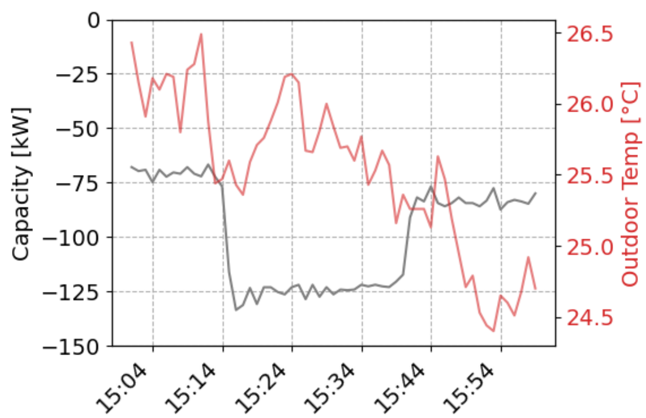

Figure 32. This is one hour of operational data when only HP2 was operating. At 15:14 it turned on in second stage.

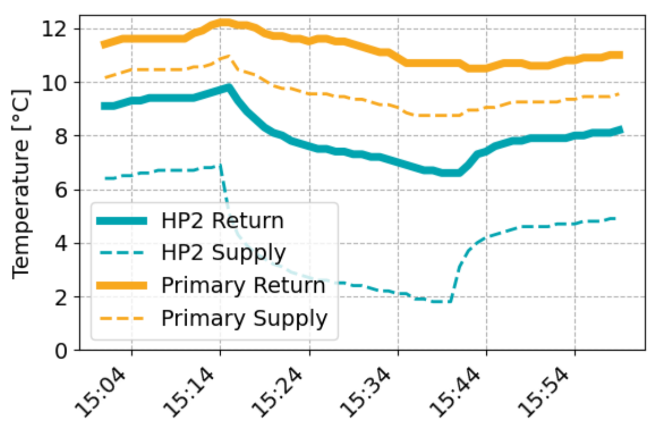

Figure 33. Primary loop and HP2 supply and return temperatures are shown for 1 hour of operation. The HP2 supply temperature drops quickly when in second stage but doesn't drive enough cooling into the primary loop before it reaches its low temperature operational limit.

9.2 Heat exchanger sizing

The heat exchangers selected for HP1 and HP2 are the Alfa Laval CB110-150H and CB110-124H, respectively. These were sized in cooling mode to deliver 780 kBtu/h and 660 kBtu/h, compared to the rated cooling capacities of the heat pumps, which are 771 kBtu/h for HP1 and 614 kBtu/h for HP2.

The heat exchanger schedule specified:

- Cold-side entering water temperature (EWT): 5°C
- Building-side EWT: 15.5°C
- Leaving water temperature difference: 5.6°C
- Glycol flow rates: 178 GPM (HP1) and 151 GPM (HP2)

However, after replacing the glycol pumps due to initial underperformance, the actual flow rates were 128 GPM for HP1 and 115 GPM for HP2.

This discrepancy between scheduled and actual flowrates may have contributed to insufficient heat transfer, which in turn prevented second-stage operation in cooling mode. However, the exact cause of the heat exchanger sizing issue remains undetermined.

Additionally, when only one heat pump operates, approximately half of the glycol flow bypasses the heat exchanger without being cooled. This forces the active heat pump to operate at lower inlet temperatures to meet the primary loop cooling setpoint. Lower glycol temperatures reduce system efficiency and push the heat pump closer to its operational limits.

Section 12.5 describes a change to the hydronic circuit design that would have helped to alleviate the heat transfer issues with second stage cooling.

9.3 Cooling capacity solution

The heat transfer problem was addressed by allowing both heat pumps to operate in tandem in first stage, rather than relying on one unit to progress to second stage. This would deliver a cooling capacity roughly equivalent to one heat pump in second stage, but it effectively doubles the heat exchange area, eliminating the heat exchanger as a system bottleneck. It also allows no (or minimal) flow to bypass without being cooled.

This improvement is illustrated in Figure 34, which shows data with both heat pumps operating in tandem. The temperature difference between the average glycol supply temperature and the primary loop supply temperature remained consistently around 2°C. The trade-off with this approach is increased pump energy consumption: it requires twice the glycol circulator energy. As a result, while effective in meeting the load, this solution reduces overall energy efficiency.

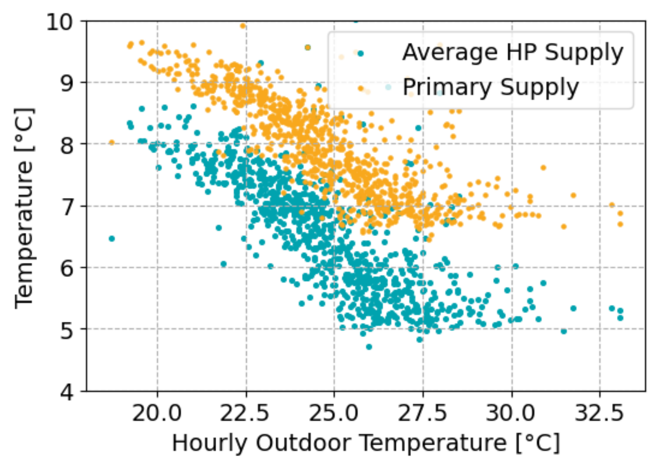

Figure 34. The heat pumps met cooling setpoint when allowed to operate in tandem in first stage.

Recommendations

24. ***Operating two heat pumps in parallel in first stage can be used to alleviate heat transfer issues,*** *but future retrofits should seek to optimize heat exchangers.*

9.4 Outdoor temperature & COP

Figure 35 plots the hourly heat pump cooling COPs as a function of the outdoor temperature. HP1 is split into two trends: one trend for when it was operated independently, and another when it was operating in tandem with HP2 (typically with both in a low stage). HP1 is split into two trends because it achieved a lower efficiency when it operated in tandem with HP2. The reason for the reduced performance is shorter cycle times as illustrated in Section 9.5.

Overall, as expected, the cooling COP decreases with increasing outdoor temperature. Also shown is the rated cooling COP. The rated cooling COP is determined for a specific set of conditions (water as the heat transfer fluid, outdoor temperature of 35°C, 6.7°C supply temperature, flow rate of 155 GPM for HP1, and 124 GPM for HP2). In this installation, glycol was used as the heat transfer fluid, the supply and outdoor temperatures varied, and the flowrates were lower (128 GPM for HP1 and 115 GPM for HP2).

Lower flowrates can cause short cycles due to the elevated temperature differentials, and this can degrade performance. Unlike heating mode, for outdoor conditions similar to the rated conditions, the real-world cooling COP was trending *notably below the rated cooling COP*.

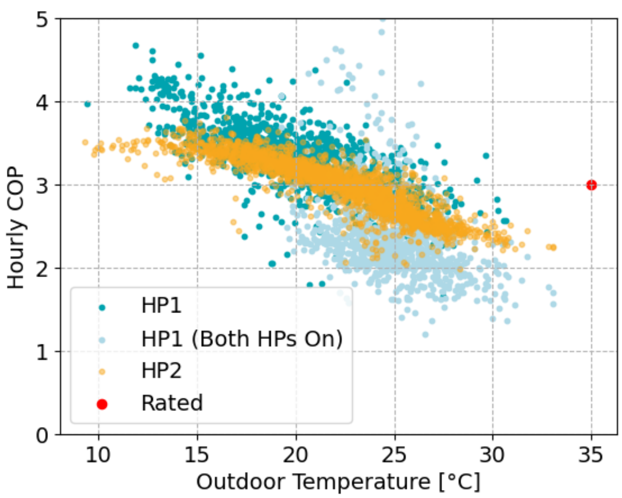

Figure 35. The cooling COP is plotted against outdoor temperature for each heat pump. Data for HP1 when it was operating in tandem with HP2 is separated because it followed a different trend with a lower COP.

Recommendations

25. Practitioners should expect that the real-world cooling efficiency may differ from the ratings *insofar as the real-world conditions differ from the rating conditions.*

9.5 Cycling & COP

Figure 35 separated the COP data for HP1 into periods when it operated independently and when it ran in tandem with HP2. It shows that HP1 achieved a lower COP when operating in tandem. The primary reason is shorter cycle times.

Figure 36 plots COP as a function of cycle time, using only data with an outdoor temperature ±1°C of 25°C to isolate the effect of cycle time from the effect of outdoor temperature. The figure otherwise includes all operational data from the cooling season where the heat pump delivered a minimum level of cooling. The analysis shows that short cycle times—particularly those under 10 minutes—resulted in a reduced COP. HP1 had more short cycle times than HP2, which had no cycle times below 10 minutes for this subset of the data. At longer cycle durations, both heat pumps followed similar performance trends.

Figure 37 includes data for all outdoor temperatures but groups data by whether the heat pumps operated independently or in tandem (labelled as “Both On”). It is a log plot that looks at cycle time against outdoor temperature. Both units experienced shorter cycle times when running in parallel, but HP1 was driven to operate with shorter cycle times than HP2, frequently below the 10-minute threshold. Both heat pumps had short cycle times in mild outdoor temperatures.

As discussed in Section 9.2, operating both heat pumps in tandem in first stage was a technical solution to meet the cooling load. It was already noted that this strategy increased circulator pump energy and therefore lowered system efficiency. Figure 36 and Figure 37 show that parallel operation also degraded COP further by inducing shorter cycle times on HP1.

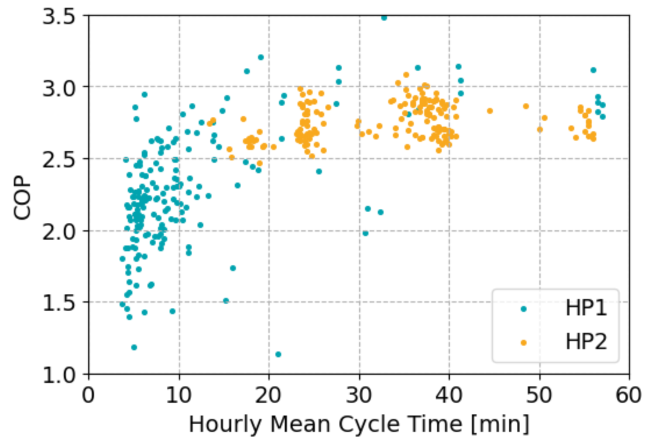

Figure 36. COP is plotted against cycle time for both heat pumps considering only data with an outdoor temperature of 25°C ± 1°C to remove the impact of outdoor temperature on COP. Shorter cycle times introduce COP degradations.

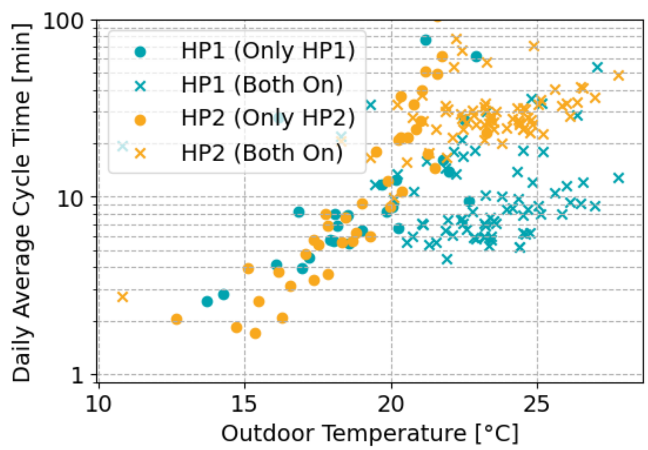

Figure 37. In warmer conditions, HP1 generally operated with shorter cycle times than HP2. When operating in tandem with HP2, HP1 was more frequently pushed into the sub-10-minute range, where the most significant COP degradation was observed.

Recommendations

26. ***Mitigate short cycling during parallel cooling operation:*** *As in heating mode, operating two heat pumps in parallel can lead to short cycle times and reduced performance. This should be addressed through improved control strategies or system design adjustments.*

9.6 Glycol temperature & COP

Figure 35 showed that for comparable outdoor conditions, the actual COP that was achieved in practice was lower than the rated COP. Section 9.4 discussed that other real-world conditions were different than the rated conditions, including the flow rates that were achieved and the heat transfer fluid that was used. Also important were the glycol loop temperatures that were achieved in practice.

Figure 38 shows HP2's supply temperature as a function of outdoor temperature. When outdoor temperatures approached or exceeded 30°C, the mean hourly supply temperature from HP2 was ~4.5°C. This was lower than the value that was used in ratings, which was 6.7°C. This is related to the heat transfer issues discussed in 9.1 and 9.2.

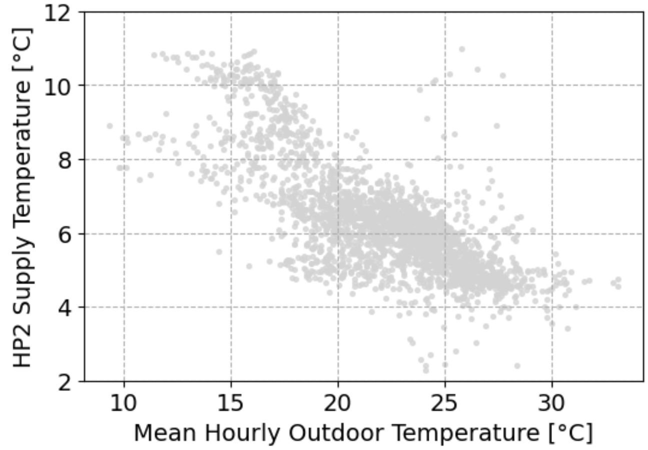

Figure 38. In very warm conditions, the supply glycol temperature from HP2 reached ~4.5°C. This caused a notable performance degradation compared to the rated conditions which used 6.7°C.

Figure 39 illustrates the effect of supply temperature on COP, with data grouped into different fixed outdoor temperatures. At an outdoor temperature of ~20°C, colder glycol temperatures resulted in a small COP drop; at ~25°C, the drop was more pronounced. A reduction from 6.7°C to 4.5°C corresponded to a ~0.5 COP loss. Data were not available to quantify the impact at 30–35°C, but the 20°C and 25°C trends suggest COP degradation due to return temperature worsens with higher outdoor temperatures.

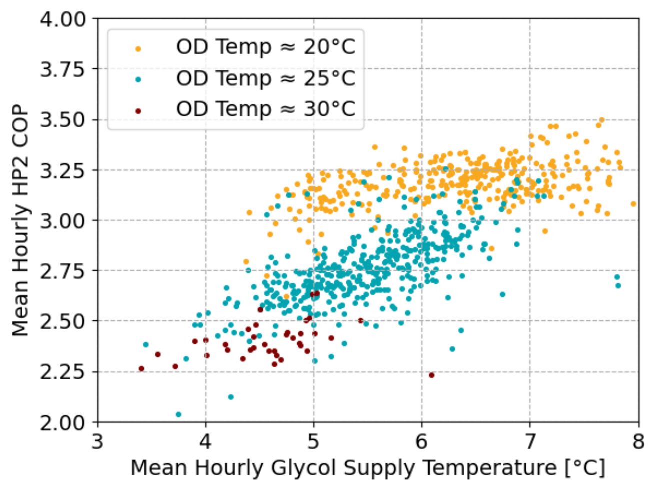

Figure 39. COP was plotted against glycol temperatures for HP2 for different fixed outdoor temperatures. Colder glycol temperatures introduce greater COP degradation as the outdoor temperature increases.

Notably HP1 did not show a correlation that was as clear and typically behaved with glycol temperatures that were not as cold. However, HP1 was affected by the fan contactor issue (see Section 13.1), which may have reduced airflow and ultimately pushed it into operational conditions that were not typical behaviour.

Overall, it is reasonable to conclude that much of the lower-than-rated COP can be attributed to lower supply temperatures in the field relative to those used in rating conditions. Additional performance degradations are likely due to other parameters, like the glycol heat transfer fluid or flow rate, deviating from rated conditions. The impact of glycol is believed to have had a more pronounced impact in cooling mode than in heating mode, because when glycol is warm it is more "water-like" but when cold it becomes more viscous.

Recommendations

27. ***In cooling mode, glycol temperatures should be kept as warm as is feasible to achieve the best efficiency:*** *However, this must be balanced with providing sufficiently low temperatures to provide adequate cooling.*

9.7 Overall cooling COP

Figures in previous sections plotted the cooling mode COP considering *only the heat pump energy consumption*, and not the energy consumption of the glycol and injection circulator pumps. Figure 40 and Figure 41 plot the total energy inputs and cooling delivered from each heat pump, separating the energy consumption from the heat pumps, the glycol circulators, and the injection circulators.

The average cooling mode COP from HP1 and HP2, considering only the heat pump energy consumption, was 2.9 for both heat pumps. When the circulator pump electricity consumption was included, it reduced to 2.2 and 2.3 for HP1 and HP2, respectively. The circulator energy consumption therefore reduced COP by 0.6 – 0.7. The plot for HP1 was corrected to remove the impact of the fan contactor issue (Section 13.1) which created a phantom load of several kW when the heat pump was in standby mode.

The heat pumps have close to comparable COPs. HP1 had cycling issues when it was operating in tandem with HP2, which reduced its COP. However, HP2 was used to drive lower glycol supply temperatures which also caused a reduction in COP. It may be that these factors evened out to an extent. HP1 also appeared to have a better COP in milder temperatures, although the reason was not identified, and this would have also brought up the overall average COP.

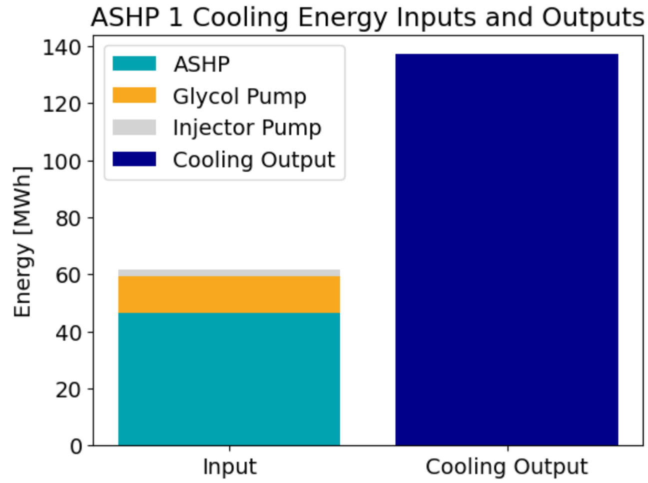

Figure 40. Energy inputs and outputs for HP1.

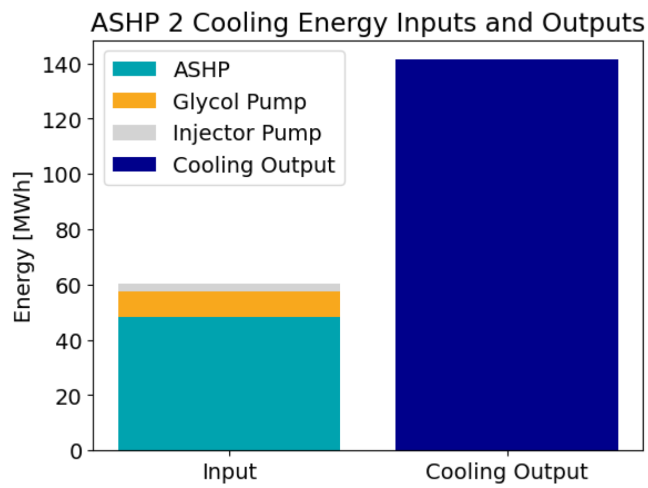

Figure 41. Energy inputs and outputs for HP2.

10 Findings: Cooling Season Optimization

10.1 Optimization considerations already addressed

Two key optimization considerations were partly discussed in Section 9:

- **Outdoor Reset Curve Adjustment**
  The initial cooling reset curve was too warm to maintain comfort in mild weather and was later lowered. This adaptive approach—starting with a conservative reset curve and adjusting based on occupant feedback—helped balance system efficiency and comfort. The final outdoor reset curve setpoints for the primary loop supply water temperature were 12°C at an outdoor temperature of 18°C and 8°C at 30°C. Two outdoor temperature sensors were used, one on the north of the building and one on the south, and the higher of the two temperatures was used to determine the outdoor temperature for the reset curve. The south-facing sensor was added to address complaints of insufficient cooling during periods of high solar gain. The cooling system only turns off if temperatures drop below 10°C. The setpoints of the previous chiller were not recorded, but it is suspected that it did not have an outdoor reset curve.

- **Parallel Operation of Heat Pumps**
  A single heat pump in second stage could not meet the cooling load. To address this, both heat pumps were operated in parallel at first stage. While this resolved the capacity issue, it resulted in increased circulator pump energy and shorter cycle times for HP1—both of which reduced COP. It follows that both design and controls should be optimized in future installations to avoid these issues.

10.2 Circulator pump optimization

Similarly to the heating season, the circulator pump energy consumption was sorted into those times when the glycol loops were actively cooling the primary loop and when they were not. This is plotted in Figure 42. In total, the circulators consumed 27 MWh during the cooling season with 15 MWh consumed when the heat pumps were actively providing cooling. It follows that a reduction of 12 MWh (~$1,600) is possible if the circulators turned off when they were not actually needed.

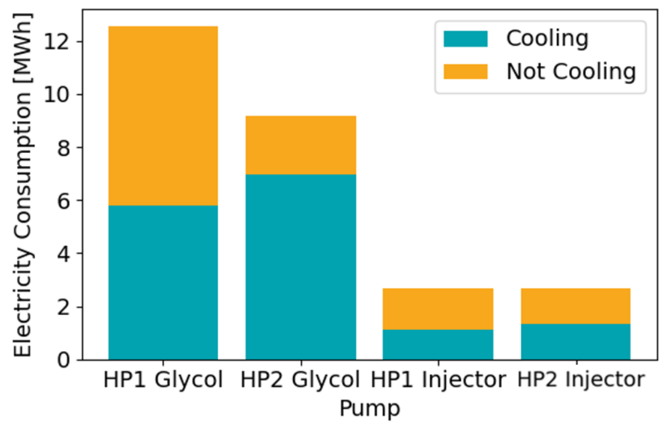

Figure 42. Circulator pump energy consumption was sorted according to whether the glycol loops were providing active cooling to the primary loop. In total, 12 MWh were consumed when the circulators were not actually needed (of 27 MWh).

Recommendations

28. ***Optimize glycol circulator pump operation*** *in cooling mode such that they are only on when the corresponding glycol loop is actively providing cooling. This can boost COP and reduce operating costs.*

11 Findings: Cooling Cost & Carbon

11.1 Electricity changes

Electricity consumption during the 2024 cooling season was assessed relative to the building's previous chiller system using IPMVP Option A: Key Parameter Measurement, with full details provided in the Report Addendum. The previous chiller was submetered, allowing for the development of a baseline linear regression model of cooling electricity use versus cooling degree days (CDDs). This model was applied to post-retrofit CDDs to estimate what electricity use would have been had the chiller remained in place.

Figure 43 compares energy use against CDDs pre- and post-retrofit. Post-retrofit values are shown for heat pumps alone and for the combination of heat pumps and circulator pumps. All figures show data that has been corrected for fused contactor (Section 13.1) phantom load, removing ~5 MWh from the total heat pump energy. While the full system used more electricity than the chiller, the heat pumps alone consumed less. Figure 44 shows monthly results; and Figure 45, seasonal. The adjusted baseline was 109 MWh, compared to 122 MWh in Summer 2024 (split into 95 MWh for heat pumps and 27 MWh for circulators). This is an overall increase of 12%, though energy use was lower (by 13%) when considering only the heat pumps. The increase is due to several factors:

- **Circulator pumps**, which were not needed with the chiller and reduced heat pump COP by ~0.6.
- **Heat exchangers losses** requiring lower heat pump supply temperatures and reducing COP.
- **Short cycle times of HP1** when operating in tandem with HP2.
- **Glycol** was needed for freeze protection of the heat pumps, but it is a poorer heat transfer fluid.

Recommendations

29. ***The identified factors (circulator pumps, cycle times, fan contactor) should be addressed to reduce utility cost increases:*** *However, the secondary loop required by this system type introduces performance degradation that is unavoidable and cost neutrality in the cooling season may be more realistic than savings.*

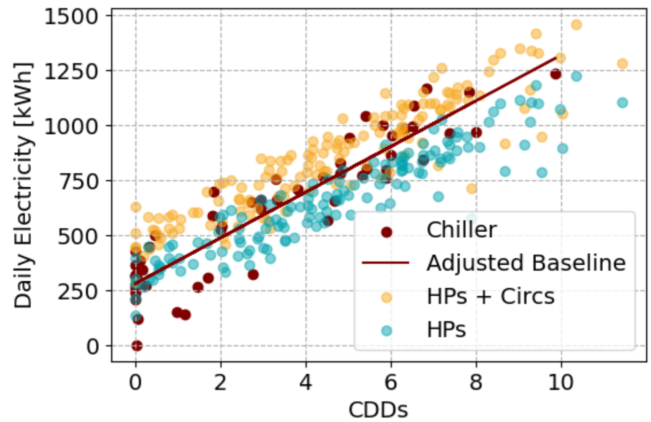

Figure 43. Pre-retrofit chiller consumption data was plotted against CDDs and this was compared against post-retrofit heat pump and circulator consumption.

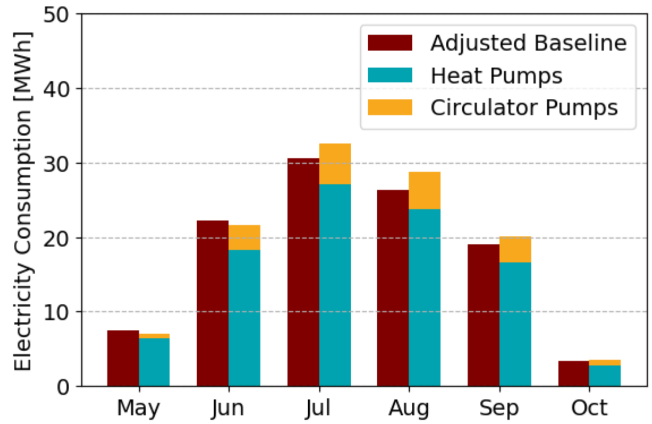

Figure 44. Electricity consumption was plotted for the adjusted baseline on a monthly basis, and this was compared against the post retrofit-energy consumption.

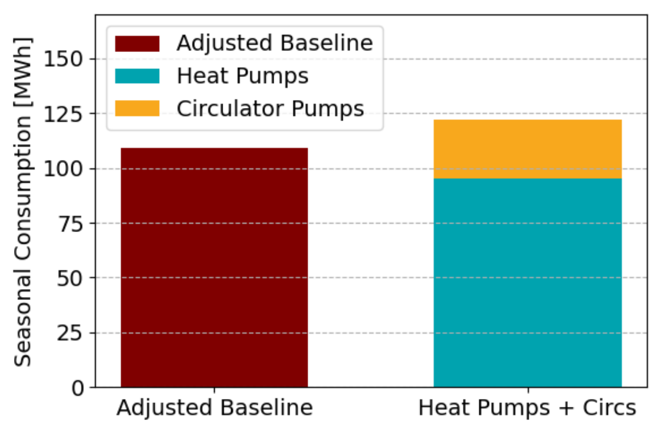

Figure 45. Electricity consumption was plotted for the adjusted baseline for the full season, and this was compared against the post retrofit-energy consumption.

11.2 Cost changes

Assuming an electricity rate of 13.3¢/kWh, the increase in electricity costs over the summer was $1,700. As noted in Section 10.2, improved circulator pump optimization could potentially reduce this by $1,600, and then the system would be net neutral on operating costs in cooling mode. Addressing other factors contributing to COP degradation may bring total utility costs lower than the previous chiller.

Changes in demand charges were evaluated for heating due to the heat pump system representing a new electrical load. However, in cooling mode, this was not assessed, as the heat pump electricity use replaced that of the chiller and was not considered an additional load that would notably impact demand.

30. ***Set realistic expectations for cooling season operating costs:*** *Owners should expect that a well-optimized air-to-water heat pump system may achieve comparable, but likely not substantially lower, cooling season utility costs relative to the chiller it replaces. This study suggests substantial savings are unlikely due to inherent efficiency penalties for this system topology—such as glycol use instead of water, added circulator pump energy, and the lower supply temperatures required with a secondary loop, all of which degrade efficiency.*

11.3 Carbon changes

The additional electricity consumption results in a carbon emissions increase of ~1 t CO_2e for the cooling season. Yielding a total net annual carbon reduction, including both the heating and cooling season utility consumption changes, of 30 t CO_2e.

12 Findings: Planning & Design

12.1 Impacts of the glycol loop

This secondary glycol loop was needed for this heat pump system, but it influenced system performance in several important ways, summarized below:

- **Reduced efficiency**, particularly in cooling mode, due to the poorer thermal properties of glycol.
- **Increased design complexity**: The secondary glycol loops required multiple additional components that needed to be sized but, in many cases, the size of components (glycol pumps, buffer tanks, heat exchangers) introduced performance issues. These were not all addressed in the study period.
- **Higher energy use** from the additional required circulator pumps.
- **Cooling mode limitations**, as heat could not be transferred quickly enough to prevent the glycol loop from dropping below the ASHP's operational limits, effectively preventing second-stage cooling.
- **Temperature loss across the heat exchangers**, impacting utilization in heating mode and efficiency in cooling mode.
- **Contribution to short cycle times**, due to the differences in heat transfer rates into and out of the glycol loop.

Recommendations:

31. ***Reassess glycol concentration****: A 40% mix may be overly conservative in systems where glycol is warm when the system is operating during cold weather. A lower concentration, such as 33%, may be sufficient to provide burst protection down to -34°C, while improving flow, heat transfer, and overall efficiency. Designers should reference ASHRAE's Cold-Climate Buildings Design Guide burst protection limits (Table 7.3) when selecting glycol concentrations.*
32. ***Explore alternative fluids****: Consider other heat transfer fluids that may have lower viscosity and/or better thermal properties to reduce efficiency penalties.*
33. ***Avoid secondary loops when possible****: Manufacturers should explore system topologies that eliminate the glycol loop altogether—for example, using refrigerant-to-water heat exchangers to directly heat the primary loop.*
34. ***Improve design practices****: Engineers should account for the significant impact glycol has on pump sizing, heat exchanger performance, and overall system efficiency. Lessons from this installation highlight potential pitfalls.*

12.2 Location where HPs connected

While the schematic showed the heat pumps connected upstream of the boiler—to allow for potential preheating—they were installed downstream. The monitoring team did not determine the root cause, though the error may have been preventable with a clear and detailed piping layout. In this installation, preheating did not offer notable benefits. However, in other systems where preheating could improve performance, upstream placement of heat pumps relative to the boiler is critical.

Recommendations:

35. ***Provide piping layout drawings:*** *Layout drawings are essential for calculating loop head and specifying circulator pumps. Without them, there may be more fittings than anticipated, increasing head and reducing flow.*
36. ***Follow piping layout:*** *Ensure heat pumps are installed according to the intended piping layout. Where preheating is beneficial, installing heat pumps upstream of the boiler is essential and should be clearly communicated in design documentation.*

12.3 Air separator with automatic vent

The monitoring team and manufacturer observed that the system used manual air vents rather than an air separator with an automatic vent. The best practice is to use automatic air separation. They more effectively and consistently remove air from the system.

Recommendations:

37. ***Use air separators with an automatic vent:*** *Install air separators with an automatic vent instead of manual vents to improve air removal and ensure long-term system performance.*

12.4 Pump sizing & replacement

The desired flowrates were not achieved by the glycol pumps as originally designed. They were replaced in late 2024 with new pumps. The monitoring team noted that the system had a large number of 90º bends which may have contributed to a greater than expected head requirement.

Recommendations:

38. ***Carefully size pumps:*** *Mechanical engineers should consider the impact of cold glycol on head, and also properly assess the pressure loss of system components based on a piping layout.*

12.5 Hydronic circuit design

The initial hydronic design incorporated a single large buffer tank and heat exchanger connected to both heat pumps. Due to supply chain issues, the design was changed to that shown in Figure 2, with two separate glycol circuits each with their own buffer tank and heat exchanger. This design change exacerbated some of the performance issues that were observed. Cycling behaviour would have been improved with the original design because of the larger buffer tank. Similarly, the heat transfer issues in second stage during cooling mode would have been alleviated due to the larger heat exchanger. Essentially, during part load conditions with a single heat pump operating, the original design would have provided approximately twice the buffer tank size and heat exchanger size.

Recommendations:

39. ***Do not isolate heat pump hydronic circuits from each other:*** *Mechanical engineers should consider that part load performance is improved when multiple heat pumps connect to a single buffer tank and heat exchanger, rather than designing them as isolated hydronic circuits.*

13 Findings: Operation & Maintenance

13.1 Fan contactor fault on HP1

A fan contactor failure on HP1 caused the outdoor fan to run continuously, drawing approximately 6 kW regardless of whether the heat pump was operating. The issue began in May 2024 and went unnoticed until January 2025, when performance monitoring identified a phantom load. Further investigation in February 2025 confirmed that the fan contactor had fused closed, resulting in continuous power to the fan circuit. The issue was resolved in March 2025. Over this period, the fan consumed approximately 4.3 MWh per month—most of it during periods when HP1 was not providing active heating. It also may have caused COP degradation in cooling mode.

The manufacturer provides an inspection schedule and checklist that includes annual checks of fan contact wear. These procedures are intended to support early detection, but this issue persisted undiagnosed for nine months until it was flagged through analysis of the monitoring data. Although maintenance was performed by a factory-trained contractor, it is unclear whether these specific checks were completed during either of the seasonal heating/cooling changeovers. A signed and dated checklist was neither requested by the building owner nor provided by the contractor.

Recommendations

40. ***Implement basic performance monitoring:*** *Where feasible, building operators should consider basic performance monitoring of key system parameters such as power draw, to help flag emerging issues and verify ongoing system health. These systems may be configured with automatic e-mail alerts to flag when parameters fall outside of expectations.*
41. ***Operation and maintenance should be provided according to the manufacturer schedule:*** *Building owners should require signed maintenance checklists from service providers to support accountability, confirm completion of key inspections, and aid in future troubleshooting.*
42. ***The manufacturer should assess the fan contactor fault further:*** *If needed, they should take appropriate corrective action regarding the design of the heat pump and selection of the components.*
43. ***Site staff should regularly inspect the equipment:*** *Some issues do not require specialized diagnostic equipment to identify, but can be flagged based on sight, sound, or smell. In the case of the fan contactor, the fan was audibly and visually running during periods when it should not have been, and this observation could have flagged the issue sooner.*
44. ***Consider having trained building operations professionals on-site more regularly:*** *Historically, large buildings with a certain class boiler system were required to have a certified Stationary Engineer on-site to ensure safe and efficient operation. Modern mechanical systems generally do not carry this regulatory requirement, and responsibilities are typically distributed across site staff, equipment manufacturers, maintenance contractors, BAS providers, and other professionals. While this arrangement offers cost efficiency, it may lack the regular on-site practical expertise once provided by Stationary Engineers. Building Operator training programs available in the GTA offer a modern equivalent, though they are not mandated. For larger or more complex buildings, owners may benefit from evaluating whether dedicated, trained operational staff that are on-site more regularly could improve system reliability and long-term performance.*

13.2 Refrigerant leak from HP1

In December 2024, HP1 experienced a total loss of refrigerant due to a cracked joint on one of its liquid-line filter driers. The failure was traced to vibration during the defrost cycles. This vibration is believed to have contributed to the filter-drier mounting brackets loosening over time which further exacerbated the vibrations.

The cracked joint allowed the full refrigerant charge to be released into the atmosphere. Repairs included replacing the filter-driers and brackets, recharging the system, and modifying the defrost control sequence to minimize vibration. HP1 contained approximately 60kg or R410a refrigerant, which has a global warming potential of 2,088. It follows that the full loss of the refrigerant charge produced approximately 125 Tons CO_2e. It would take approximately 4 years of operation (at 30 Tons CO_2e per year) to break even, at which point the system will overall be producing net carbon reductions.

The manufacturer preventative maintenance checklist includes a line item for inspection of refrigerant filter-driers and brackets annually. Similar to 9.1, it is unclear whether this specific check was completed during either of the seasonal heating/cooling changeovers.

Recommendations

45. ***Assess design of failed brackets****: The manufacturer should consider revisiting the design of mounting brackets for the filter-driers to achieve greater resistance to loosening from vibrations.*
46. ***Highlight known failure points in contractor training****: Items related to loose mounting brackets and vibration, and potential consequences, should be highlighted in contractor training materials.*
47. ***Review maintenance checklist language****: The checklist language should be evaluated and potentially clarified to help ensure that the checks detect early signs of fatigue or failure, as intended.*
48. ***Consider engineering safeguards****: For example, design features that limit refrigerant loss in the event of a crack — should be explored.*
49. ***Prioritize low-GWP refrigerants****: Future systems should prioritize equipment that uses low-GWP refrigerants as they become available. This will mitigate the negative consequences of a full refrigerant loss.*
50. ***Comprehensive performance monitoring may identify early signs of problems****: If a catastrophic leak was preceded by a small leak there may be measurable signatures in performance monitoring data, like a decline in efficiency, that could be used to alert staff of an issue.*
51. ***Preventative maintenance might include vibration diagnostics****: Since the root of the refrigerant loss was excessive vibration, it is feasible that out of specification levels of vibration could be measured with appropriate diagnostics during preventative maintenance.*

14 Findings Summary

14.1 Heating

- The heat pump system was able to meet the heating needs of the building down to an outdoor temperature of -3°C, which translated to 70% of the total heating load.
- The heat pumps had a rated COP of 3.0 for defined rating conditions. However, in practice, the conditions were different and the total *seasonal* heating COPs were 2.1 to 2.2. When the additional circulator pump energy was included, it reduced to between 1.7 and 1.8.
- The heat pumps reduced gas consumption by 21,000 m^3 per year. Monthly demand increased by 44 kW on average during the heating season, and total electricity consumption increased by 107 MWh. Approximately 20% of the increase was due to the glycol and injection loop circulators.
- Using the utility rates from monitoring period, and noting that the control strategy sought to optimize heat pump utilization rather than operating costs, the net utility cost change was an increase of $6,700 per heating season. This was a 50% increase compared to boiler-only operation for the hydronic heating system but overall increased the total utility cost of the building on the scale of a few percentage points.
- The primary loop circulators were also upgraded late in the monitoring period and this would have produced savings of $1,600/year for the full period. An additional $1,800/year reduction may be achievable with improved circulator pump optimization.
- The above sources of savings were considered alongside modelled control approaches that sought better cost-optimization by operating the heat pump preferentially in an off-peak TOU bracket. The modelling found that bundling the primary loop pump upgrade, circulator optimization, and cost-optimized “off-peak only” control together, would result in net cost savings while still managing 50% of the heating load with the heat pumps.
- Optimizing the outdoor reset curve setpoints for the primary loop supply temperature was crucial for achieving significant heat pump utilization. By reducing the curve to the lowest values, while still ensuring occupant comfort, the heat pumps were able to manage the heating in temperatures that were approximately 5°C colder than they otherwise would have been able to with the original unoptimized outdoor reset curve.
- It was initially expected that the heat pumps could preheat water for the boiler to extend cold-weather operation. However, the system was limited primarily by the supply temperatures of heat pumps—rather than their capacity. As a result, boiler preheating offered little benefit in this case, though it may be valuable in other systems.
- The hydronic heating system gas consumption was reduced by 70% and the facility’s overall gas use dropped by 20%. This full building gas reduction was less than expected and was a consequence of the fact that space heating only accounted for about one-third of total gas use—make-up air was the dominant load. During monitoring, make-up air setpoints and schedules were adjusted with the goal of shifting more heating onto the hydronic system.
- Different cold-weather control strategies were tested. Operating a single heat pump in high-stage is believed to be more effective than running two in parallel, which caused short cycling, increased pump energy use, and reduced COP.
- Cycle time was a key factor impacting efficiency. Short cycle times primarily occurred in mild outdoor conditions during the heating season, and when the heat pumps operated in tandem. The root cause was identified to be the buffer tank sizing. Short cycle times were not fully addressed within the study but different solutions are possible.

14.2 Cooling

- The original system configuration was initially unable to meet cooling setpoints. In mild weather the outdoor reset curve was set too warm, and it was later adjusted based on occupant comfort complaints.
- In warm and hot weather, one heat pump alone was not able to handle the cooling load of the building because of insufficient heat transfer between the glycol and primary loops. The glycol loop would turn on in second stage and cool down until it reached its operational limits and then shut off, preventing continuous second stage operation.

- Operating two heat pumps in tandem at first stage, rather than one in second stage, solved the cooling capacity issue by effectively doubling heat exchanger area. It also reduced the flow which bypassed cooling. However, this increased circulator pump energy consumption and reduced overall efficiency.
- The measured COP was notably below the rated values when outdoor conditions were close to the rated conditions. However, several other parameters deviated from rated conditions: the achieved flow rates were lower; glycol was used instead of water; supply temperatures were colder; and cycle times influenced the achieved COPs.
- Short cycle times—especially below 10 minutes—were linked to lower COP; HP1 experienced more short cycle times than HP2, particularly when running in tandem with HP2. This degraded the overall HP1 efficiency.
- On HP2, the glycol supply temperatures in hot conditions were generally colder than rating conditions. This significantly contributed to COP degradation, causing a reduction that is believed to be greater than 0.5.
- The average seasonal cooling COPs, considering only heat pump energy, were approximately 2.9 for both HPs. HP1 was corrected for the fan contactor phantom load discussed in Section 13.1. Including circulator pump energy reduced these to 2.2 and 2.3 respectively, a reduction of 0.6 – 0.7.
- Circulator pumps consumed considerable energy even when not actively cooling, indicating inefficient operation during standby periods. A savings on the scale of ~$1,600 is possible with better circulator pump optimization, and this would bring the total system operational cost in the cooling season near net neutral with the previous chiller.
- The total cooling season electricity consumption increased by 12% compared to the previous chiller system, but heat pumps alone consumed 13% less energy than the chiller. It follows that the increase can be attributed to the energy used by the extra circulator pumps that were not needed with previous chiller.
- Beyond the circulator pump optimization there are additional optimization opportunities that could drive the system into a small level of cost savings. This would include addressing the cycle times on HP1 and the colder glycol supply temperatures on HP2. However, substantial savings are unlikely due to inefficiencies that are built into this heat pump topology - such as glycol use instead of water, added circulator pump energy, and the lower supply temperatures required with a secondary loop.

14.3 Design and O&M

- Two significant operation and maintenance issues occurred and were rectified. Firstly, a fused fan contactor caused the outdoor fan of HP1 to run continuously for nine months, resulting in significant unnecessary energy consumption. It may have also generally reduced COP due to there being less airflow on the outdoor unit. Secondly, a cracked refrigerant filter-drier joint on HP1 led to a complete loss of 60 kg of R-410A, equivalent to 125 T CO_2e, which was equivalent to the savings produced from 4 years of operation.
- Design and installation issues included the following: heat pumps were initially connected to the system downstream of the boiler rather upstream as planned, use of manual air venting was used instead of automatic air separators, and the glycol pumps were initially undersized and replaced. It is also suspected that the heat exchangers are undersized for cooling mode operation. The small buffer tank sizing resulted in short cycle times.
- While necessary for this equipment, the use of a secondary glycol loop introduced multiple performance and design challenges. It reduced heating and cooling efficiency and heating season utilization, contributed to short cycle times, resulted in higher energy use, and added system complexity with additional pumps, heat exchangers, and tanks that needed to be sized, and sometimes re-sized.

15 Recommendations Summary

15.1 Feasibility assessment & planning

At this stage, practitioners should evaluate the potential for heat pumps to meet the heating and cooling loads. They should account for auxiliary energy use, such as circulator pumps, and estimate real-world seasonal COPs in both modes.

Optimizing the outdoor reset curve should be completed during feasibility, as it may directly affect how much of the heating and cooling seasons the heat pumps can cover. Designers should determine whether the system is likely to be constrained by capacity or supply temperature — this will influence whether boiler preheating is beneficial in heating mode.

A breakdown of the facility's gas use (e.g., space heating, DHW, make-up air) is essential for setting reasonable expectations and identifying other decarbonization opportunities.

Feasibility assessments should include an assessment of cost-optimization opportunities based on time-of-use control, if applicable. They should evaluate system financial metrics under a range of utility rate scenarios, noting that even if rates are less favourable today, this may change over the system's lifetime, as recent history has shown.

15.2 System design & engineering

Air-to-water heat pump systems that rely on a secondary glycol loop are inherently more complex, and this complexity must be actively managed. Accurate sizing of pumps, heat exchangers, and buffer tanks is critical — designers should account for the effects of glycol on component performance.

In cooling mode, undersized heat exchangers may prevent full second-stage operation of a single unit, leading to lower system performance. Instead, systems may be forced to run two units in parallel at low-stage, which increases circulator pump energy use. Automatic air separators should be included to support system reliability.

As per the original design, connecting multiple heat pumps to a single buffer tank and heat exchanger would improve the cycling and heat transfer issues that were observed.

15.3 Holistic retrofit integration

A heat pump retrofit should not be treated in isolation — it should be seen as part of a broader facility upgrade. Other opportunities to reduce energy use, such as converting older constant-speed pumps to high-efficiency variable-speed pumps, should be considered at the same time.

Identifying and reducing other natural gas loads (like MUA systems) can also help maximize benefits. Bundling multiple upgrades together can mitigate increases in electricity consumption and improve the overall business case. Planning for performance monitoring and optimization from the start can also support commissioning, troubleshooting, and long-term efficiency.

15.4 Control strategy optimization

Once installed, the control system determines how the heat pumps interact with building loads and backup systems. Operators should pay attention to short cycle times and strive to avoid them. They should also seek to avoid operating heat pumps in tandem when it is possible to meet the load with one. Tandem operation of two heat pumps — while sometimes necessary to meet the load — can induce shorter cycles and degrade COP. This was observed in both heating and cooling mode. Controls should seek to mitigate this.

Ideally, cycling could be addressed through a minimum run-time parameter, but it can also be supported through appropriately-sized buffer tanks and thresholds that govern on/off logic. Control logic should reflect the building owner's goals: for example, if reducing operating costs is a priority, controls may favor off-peak operation under TOU pricing. If emissions reductions are the goal, a higher degree of heat pump utilization may be justified even when it is not cost-optimal.

15.5 Operation & maintenance

Routine maintenance should follow manufacturer recommendations, and owners should require signed checklists from service providers to support accountability. Manufacturer-trained maintenance contractors are preferred. The experience from this project suggests the language of maintenance checklists might be improved to better detect known potential issues related to component fatigue or early signs of failure.

In addition to temperature monitoring done by the BAS, additional monitoring points — for example, electrical power consumption — are helpful to flag operational issues early.

15.6 Manufacturer-specific opportunities

Split refrigerant circuit systems, which can deliver heat directly to the primary loop without a glycol intermediary, represent a promising direction for reducing system complexity and improving efficiency. Component-level failures — including a fused fan contactor and a cracked refrigerant filter-drier bracket — point to the importance of continual improvements in equipment design. Manufacturers should also explore safeguards that limit refrigerant loss in the event of a leak or failure. Where refrigerant loss does occur, using low-GWP alternatives will help mitigate climate impacts. Known failure points should be highlighted in contractor training materials to support better installation and servicing practices in the field.

16 Conclusion

The FQSF retrofit successfully demonstrated the technical feasibility of replacing a chiller with an air-to-water heat pump system that offsets most of the space heating gas consumption in a large multi-unit residential building. The system handled approximately 70% of the building's heating needs and fully met the cooling load, reducing natural gas use by 21,000 m^3 per year—contributing to an annual carbon emissions reduction of 30 t CO_2e. The main constraint preventing greater gas reduction was the supply temperature limits of the heat pumps.

This performance came with important caveats. The secondary glycol loops required by this heat pump topology greatly increased system complexity and reduced efficiency. In both heating and cooling modes, auxiliary energy use from the glycol circulator pumps was a key factor impacting overall performance and cost. Achieving acceptable seasonal performance requires careful engineering to minimize penalties related to the secondary loops and to optimize system setpoints like outdoor reset curves.

The real-world seasonal COPs were lower than rated values due to differences between actual operating conditions and test conditions. Seasonal heating COPs ranged from 2.1 to 2.2, while the seasonal cooling COP from both heat pumps was 2.9, with further reductions once circulator pump energy was included.

Utility costs increased in both heating and cooling seasons, though for different reasons. In the heating season, the increase was largely due to the chosen control approach, which aimed to maximize heat pump utilization rather than minimize cost. As a result, the heat pumps were operated during peak electricity price periods, when the cost per unit of delivered heat was significantly higher than gas. A more strategic approach—favoring off-peak time-of-use (TOU) operation—could improve cost performance while still achieving substantial gas displacement.

In the cooling season, the cost increase stemmed from the added electrical loads of the glycol circulator pumps and efficiency losses from the secondary loop, which were not required with the original chiller. That said, identifiable potential pathways to cost-neutral operation—or even savings—exist in both heating and cooling modes through further optimization.

Refrigerant loss was a serious issue during the monitoring period. One heat pump lost its full charge, releasing 125 T CO_2e into the atmosphere. This was equal to the savings from four years of operation. This risk is not unique to air-to-water heat pumps, but applies to any refrigerant-based system, including chillers. Preventative maintenance—following manufacturer recommendations and using detailed checklists—is essential to detect and limit leaks. Design safeguards to prevent full-charge loss are also recommended, as is contractor training to identify known faults. Looking ahead, low-global warming potential refrigerants will be important to reduce the climate footprint of these systems.

Currently, air-to-water heat pump systems with secondary glycol loops, as installed at FQSF, represent a practical solution for supporting the decarbonization of large residential buildings. While complex and not without trade-offs, they remain a technology that can be deployed today. Broader implementation, along with ongoing optimization, will be critical to meeting building decarbonization goals. Looking ahead, emerging systems with split refrigerant circuits—which deliver heat directly to the primary loop without a secondary loop as an intermediary—offer potential to simplify design, reduce auxiliary energy use, and improve efficiency.

While natural gas rates are currently low, they are volatile and have recently reached historic highs before falling. In periods of high gas prices, the business case for electrically-driven heat pumps improves. Because building owners may live with today's equipment selection for 20 years or more, it is important to recognize that energy prices will likely fluctuate dramatically over that time. Systems that incorporate a high-efficiency electric heating option offer flexibility and reduce exposure to price swings. When paired with thoughtful controls—and supported by TOU rate structures—this flexibility can deliver low operating costs, manage financial risk, and reduce emissions.

Ultimately, this study underscores the potential of air-to-water heat pumps as a decarbonization tool for multi-unit buildings, while emphasizing the importance of realistic expectations, careful design, proactive commissioning, ongoing maintenance, and continuous optimization. The lessons from the FQSF retrofit provide valuable guidance for future deployments and broader sector adoption.