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Cold climate field evaluation of high performance air to air heat pump systems developed through the cold climate heat pump challenge program

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Abstract

CanmetENERGY-Ottawa (CE-O) evaluated the field performance of three (3) innovative cold climate air source heat pumps developed through the US Department of Energy / Natural Resources Canada's Cold Climate Heat Pump (CCHP) Challenge program. The program challenged participating manufacturers to develop, test and bring to market air to air heat pumps that would meet specific performance criteria. Several manufacturers participating in the program partnered with Natural Resources Canada's CanmetENERGY research laboratory to undertake field performance evaluation of their candidate systems. Some of the participating manufacturers have recently announced their intention to bring their systems to the market for purchase.

This paper presents coefficient of performance (COP) by outdoor temperature for three CCHP challenge systems evaluated, including two systems evaluated at the Canadian Centre for Housing Technology. A climate-specific bin model is then used to compute seasonal heating performance of the field collected test results. Results are compared to two (2) other cold climate air source heat pumps previously monitored. By evaluating performance according to operating mode, the study also analyzes low-temperature performance, part-load cycling behavior, and the effects of defrost cycles, standby modes, and crankcase/drain pan heaters on overall efficiency.

Results indicated that the three CCHP Challenge systems evaluated (units 1-3) outperformed the commercially available CCHPs, having greater than 20% higher seasonal heating COPs. When considering performance across all cycles / operating modes, units 1 and 3 were the higher performing CCHP Challenge systems in the coldest outdoor temperatures, achieving COPs of 1.8 at -26°C, while unit 2 was the highest performing CCHP Challenge system in the most moderate outdoor temperatures, achieving a COP of 4.0 at 6°C.

Keywords: heat pump; cold climate heat pump; cold climate heat pump challenge; heat pump performance; heat pump coefficient of performance

1. Introduction

The Canadian federal government's most recent climate change plan "2030 Emissions Reduction Plan: Canada's Next Steps for Clean Air and a Strong Economy'" commits to 40-45% reductions of greenhouse gas (GHG) emissions from a 2005 baseline by 2030 [1].

An evaluation of sources of GHG emissions across Canada shows that 22% of GHG emissions are from Buildings [2]. Further, space heating is the second largest contributor to end-use GHG emissions in Canada, with the majority of these emissions being produced by natural gas-fired heating systems (used to meet over 60% of space heating needs in homes across Canada) [3].

As an approach to reducing GHG emissions from space heating in the residential sector, many Canadian jurisdictions are interested in scaling up the market adoption of electric heat pumps. Studies have shown that new "cold climate" air source heat pumps can result in substantial GHG emissions reductions as compared to conventional oil, electric and natural gas heating systems in jurisdictions across Canada [4].

In 2021, Natural Resources Canada partnered with the US Department of Energy to undertake a program that would challenge manufacturers to develop heat pumps designed to achieve a number of specific criteria

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as set out in the program, with a particular focus on high performance in cold outdoor temperatures [5-6]. The program was called the Cold Climate Heat Pump Challenge (CCHP Challenge).

This paper seeks to evaluate the field measured performance of three different cold climate air source heat pumps developed through the CCHP Challenge program and contrast these with commercially available cold climate air source heat pumps (cc-ASHPs) that had been field monitored previously.

2. Testing Methodology

Centrally ducted cold climate heat pumps provided by three different manufacturers developed through the CCHP Challenge program were installed and tested for performance evaluation at field sites in Canada. Performance for these three pre-commercial prototype systems was compared to data collected from field testing of two commercially available high performance cold climate heat pump systems including one system installed at the Canadian Centre for Housing Technology (CCHT) and another system installed at an occupied home. The systems were installed by an installer certified by the equipment manufacturer and instrumentation was completed by CanmetENERGY research engineers. Data was collected and evaluated at a 1-minute or 5-minute time-step over a period long enough to enable seasonal performance evaluation (monitoring periods ranged from several weeks to over 1 year), with the focus of analysis being on cold climate heating season performance.

All systems were centrally ducted cold climate heat pumps sized with “an emphasis on heating”, equivalent to strategy 4c in the NRCan ASHP Sizing and Selection Toolkit [7]. This strategy requires the heat pump capacity to meet or exceed the building heating load at an outdoor temperature of -8.3 °C. All systems were fully electric and equipped with electric resistance auxiliary heating elements, sufficiently sized to cover the entire space heating load of the home, with the CCHP Challenge systems using multi-stage controlled auxiliary heating elements.

The three CCHP Challenge systems used low global warming potential refrigerants (R-32 or R-454b) while the two commercially available CCHP systems utilized R-410a refrigerant.

Table 1 below summarises the systems used and sites they were installed in. The site details are further described in Section 2.1.

Table 1: Summary of System and Site Details

System Designation	Capacity (kW at 8.3°C)	Refrigerant	Site Design Heat Loss (kW)	Site Design Heating Temperature (°C)	Site Measured Average Thermostat Temperature (°C)
CCHP Challenge Unit 1	10.6	R-454b	12.1	-25	21.4
CCHP Challenge Unit 2	10.6	R-32	12.1	-25	21.4
CCHP Challenge Unit 3	10.6	R-32	unknown	-23	19.8
Multi-stage CCHP	7.03	R-410a	6.96	-32	21.6
Variable Capacity CCHP	10.6	R-410a	12.1	-25	21.4

2.1. Site Descriptions

2.1.1. Canadian Centre for Housing Technology

The Canadian Centre for Housing Technology (CCHT) is a Twin Test House Facility located in Ottawa, Canada, used for evaluating whole-house performance of residential technologies. As shown in Figure 1, it consists of two adjacent houses that are identical in orientation, size and construction. Built to the R-2000 standard of the time, the research houses are primarily used for side-by-side testing of innovative housing technologies. Each is two storeys with 223 m2 of floor area, not counting the full basements. Each house has a design heating load of 12.1 kW at -25 °C and design cooling load of 7.2 kW at +30 °C [8]. The houses experience extremely high solar gains, which reduce heating loads during shoulder seasons and sunny cold days. Refer to [9] for further details related to solar gains and controls of solar gains at the CCHT.

Each house features a standard set of major appliances typically found in North American homes. The simulated occupancy system, based on home automation technology, simulates human activity (including water and energy use) by operating resistance heaters, major appliances, lights, water valves, fans, and other sources. The simulated occupancy schedule is intended to represent typical activities that would take place in a home with a family of two adults and one child. The heat given off by humans is simulated by low temperature heaters at various locations in the houses.

Figure 1: Canadian Centre for Housing Technology
Field Monitoring Site in Ottawa, Ontario

In 2022, two cc-ASHP systems provided by manufacturers participating in the CCHP Challenge program were installed in the houses. Both systems included multi-stage electric resistance auxiliary heaters and had equivalent rated capacities, but one used R-454b refrigerant while the other used R-32 refrigerant. Both had rated heating capacities of 10.6 kW at the rated heating condition of 8.3 °C outdoor temperature. Both heat pumps were installed with 10 kW multi-stage electric resistance auxiliary.

The two systems (each from a different manufacturer) were controlled by centrally located thermostats set to 21 °C constant setpoint temperature. All other settings were as determined by the manufacturer's approved installation contractor. Ventilation air was not delivered by these systems during testing (thus enabling the system to turn off the fan when not delivering heating).

The CCHP Challenge systems were compared in performance to a variable capacity cold climate heat pump that had been previously tested in the same homes in 2012. The system had a rated heating capacity of 10.6 kW at the rated heating condition of 8.3 °C outdoor temperature with 10 kW electric auxiliary. Unlike the CCHP Challenge systems, the variable capacity cold climate heat pump system was set to run on high-speed continuous fan to deliver ventilation air (through a heat recovery ventilator).

2.1.2. Occupied Home in Repentigny, Quebec (QC)

The second field site was in Repentigny, QC, a small town close to Montreal, QC as shown in Figure 2. The home was occupied by two adults working from home and one young adult who was away during the day. The home was approximately 219 m^2 of heated floorspace (2 storeys with finished basement) and was built in

Figure 2: Field Monitoring Site in Repentigny, Quebec

the last 15 years. The installed heat pump had a rated heating capacity of 10.6 kW at the rated heating condition of 8.3 °C source temperature with 15 kW three-stage electric resistance auxiliary. The system was replacing a pre-existing 8.8kW heat pump coupled with electric baseboard backup. The thermostat for the system was generally set to 21°C but varied depending on the user's preference. The system was set to run on high-speed continuous fan to deliver ventilation air through a heat recovery ventilator.

2.1.3. Occupied Home in Winkler, Manitoba (MB)

The third field site was in Winkler, MB, a small town close to Winnipeg, MB as shown in Figure 3. The home was a prefabricated modular home with approximately 139 m^2 of heated floorspace built in 2021 with a design heat loss of 6.96 kW at -32 °C. The heat pump installed was a 3-stage multi-capacity cold climate heat pump with a rated heating capacity of 7 kW at the rated heating condition of 8.3 °C outdoor temperature. The unit included three-stage integrated electric resistance auxiliary. The thermostat for the system was generally set to 21°C but varied depending on the user's preference. The system was set to run on low-speed continuous fan to deliver ventilation air through a heat recovery ventilator.

Figure 3: Field Monitoring Site in Winkler, Manitoba

2.2. Instrumentation and Monitoring

The heat pumps tested were instrumented to enable the determination of system thermal energy output, electrical energy input and coefficient of performance (COP) as expressed in equations 1, 2 and 3 respectively below.

$$E_{out} = mc_p(T_{Supply} - T_{Return}) \quad (1)$$

$$E_{in} = (E_{Outdoor\ unit} + E_{Indoor\ unit} + E_{Auxiliary}) \quad (2)$$

$$COP_{bin,j} = \frac{\sum E_{out}}{\sum E_{in}} \quad (3)$$

E_{out}, the thermal energy output (J) was given by m, the mass flow of air through the indoor unit (kg) multiplied by c_p, the specific heat of the air (J/kg*K) multiplied by the difference between T_{Supply}, supply air temperature (K) and T_{Return}, the return air temperature (K).

E_{in}, the electrical energy input (J) was given by the sum of the outdoor unit energy consumption $E_{\mathrm{Outdoor\ unit}}$ (J), the indoor unit energy consumption $E_{\mathrm{Indoor\ unit}}$ (J) and the auxiliary unit energy consumption $E_{\mathrm{Auxiliary}}$ (J).

$\mathrm{COP}_{\mathrm{bin,j}}$ for each temperature bin, j was given by the sum of E_{out} for that temperature bin divided by the sum of E_{in} for that temperature bin.

Temperature bins, j were in 2°C increments for all temperature bins for which there was sufficient time steps of measured data with the outdoor unit cycling on to allow for an evaluation across multiple cycles for each temperature bin[1].

For the heat pumps installed at the CCHT, airflow was measured using a calibrated airflow station. For the heat pump installed at the occupied home in Repentigny, QC, airflow was measured by a measurement grid with ongoing measurements obtained via correlation between airflow and indoor unit power consumption[2]. For the heat pump installed in Winkler, MB, airflow was measured by four high accuracy velocity sensors installed in a traverse pattern across a straight run of return duct.

Temperature and relative humidity were measured using high accuracy probes located in the equipment supply and return air ducts. Energy consumption was measured using high accuracy pulsed meters with current transducers chosen with a range suited to the expected current draw of the equipment being measured. All modes of operation were integrated in the measured results, including defrost cycles, on/off cycling, standby and auxiliary modes.

3. Results

3.1. COP by Outdoor Temperature

Coefficient of performance (COP) results for the five systems evaluated are summarized in Figure 4. The superior performance of the three CCHP systems (Units 1-3) as compared to the two commercially available CCHPs (Units 4-5) is evident across the operating range, including at cold outdoor temperatures.

A drop in COP can be observed for all units at more moderate temperatures, particularly for units 3, 4 and 5. This is because these systems were operating the blower fan continuously to deliver ventilation air to the conditioned space. As the operating mode of the system becomes increasingly dominated by ventilation mode, the negative impact on overall system performance becomes the dominating factor. When the system distributes ventilation air, it is consuming fan energy to move the air at near zero or, at times, negative delivered capacity, thus leading to low or even negative COP. This effect is most pronounced in Unit 5, in which the house experiences such high solar gains that there are significant periods of time in which the return air to the system is *warmer* than the supply air from it (leading to negative COPs).

Unit 4 experienced the coldest outdoor temperatures throughout the monitoring period because it was installed in the coldest location of the five systems evaluated.

[1] Through observation of measured data, the approximate value of 400 was determined to be sufficient to allow for the inclusion of multiple heat pump cycles in each temperature bin such that the bin would take into account multiple on cycles of the outdoor unit. A lesser number risks inaccurate representation by including partial cycles.

[2] The TrueFlow measurement grid imposes a pressure drop on the return airflow and thus cannot be left in place during ongoing monitoring (this is why power is correlated with airflow for ongoing measurement).

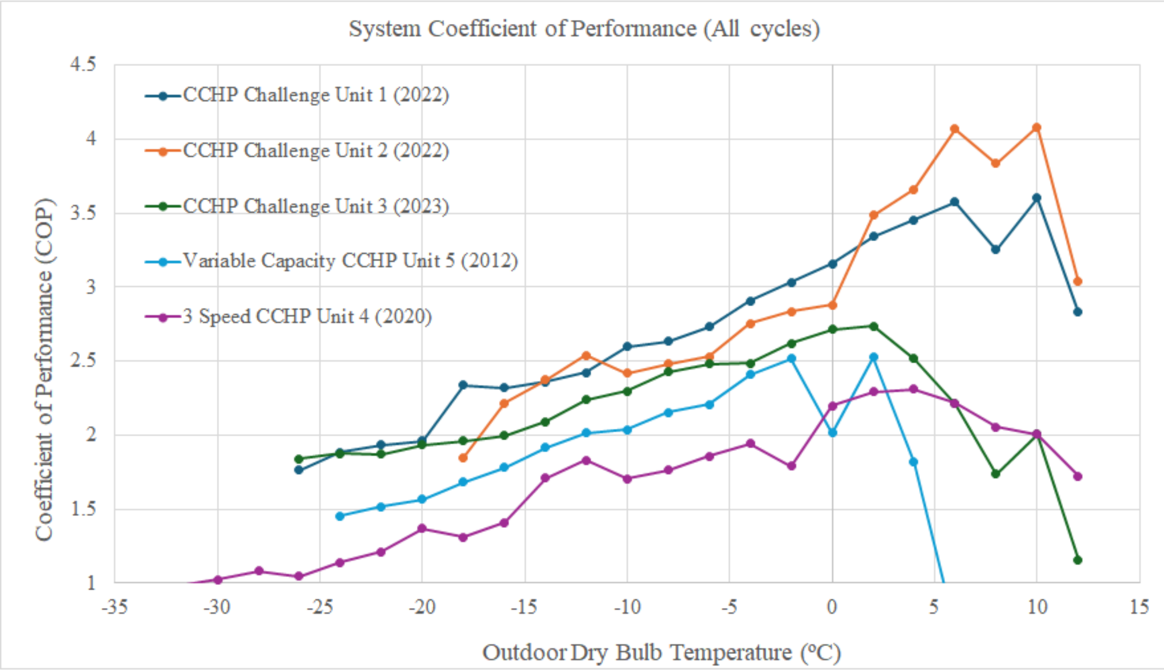

Figure 4: Coefficient of Performance for Five Cold Climate Heat Pump Systems Evaluated

In order to allow for a comparative analysis of seasonal performance, the systems were also evaluated on the basis of on-cycle performance which includes defrost cycles but excludes the ventilation distribution operating mode as well as standby and off cycle modes. Note that, for units that operated auxiliary while operating the outdoor unit and/or during defrost, this energy consumption is also accounted for in on-cycle performance. The comparative analysis of measured on-cycle COP results are depicted in Figure 5. In this comparison, the three CCHP Challenge systems consistently outperform the two commercially available CCHPs.

Two outlier trends observable in Figure 5 are the upwards spike in performance of unit 5, as compared to unit 4 at more moderate outdoor temperatures. This is likely due to the improved modulation of the fully variable compressor associated with unit 5 as opposed to the 3-stage compressor behaviour associated with unit 4. The second outlier trend observable in Figure 5 is the upwards spike in performance of unit 2 as compared to unit 1 at more moderate outdoor temperatures. Recall that both of these systems were installed in side-by-side identical homes, experiencing identical weather, loads, occupancy, solar gains, etc. The two systems are from different manufacturers utilizing different controls, etc. and they also differ in the refrigerant they used where unit 1 was charged with R-454b while unit 2 was charged with R-32. This sudden improvement in performance of unit 2 is due to the sudden decrease in incidence of defrost cycles and associated auxiliary energy, as is evident in Figure 7.

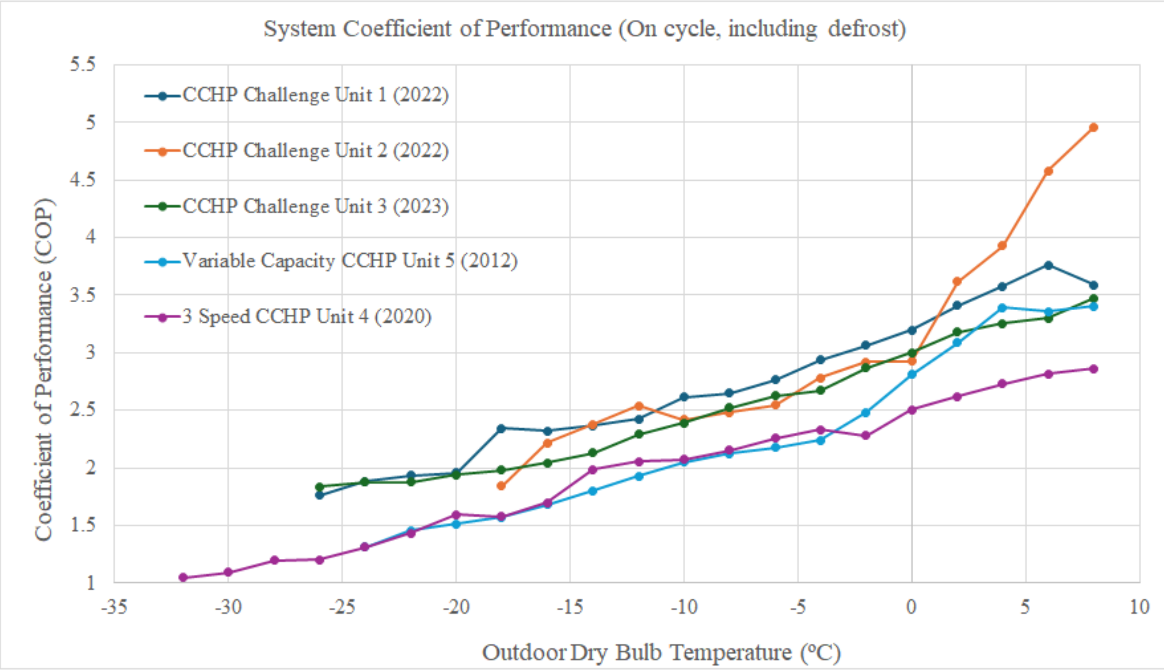

Figure 5: Coefficient of Performance for Five Cold Climate Heat Pump Systems Evaluated (on cycle including defrost)

3.2. Operating Modes by Outdoor Temperature

A methodology was developed to disaggregate the performance of the units by operating mode. For each timestep, measured unit energy consumption was allocated to individual operating modes based on observed system behaviour. A system of “flags” was developed to identify which operating mode the equipment was in. These flags were assigned as follows:

- On Cycle Flag (outdoor unit input power > 0 AND indoor unit input power > 0 AND indoor unit airflow > 0)
- Off Cycle Flag (all other modes = 0 AND airflow = 0)
- Standby Flag (on cycle flag = 0 for the past 30 minutes AND on cycle flag = 0 for next 30 minutes AND auxiliary = 0)
- Defrost Flag (outdoor unit input power > 0 AND indoor unit input power < during on cycle (for units that turn off airflow during defrost) AND standby flag = 0, may include auxiliary for timesteps in which auxiliary operates to support defrost)
- Auxiliary Flag (auxiliary input power > 0, defrost deducted for timesteps in which auxiliary operates to support defrost)
- Crankcase / Drain pan heater Flag (outdoor unit input power > during off cycle or during standby AND < during on cycle AND indoor unit input power < during on cycle AND off cycle flag = 0)

The correct allocation of the operating mode flags was confirmed by observation of input power data, airflow and temperature data. This level of analysis allows for the determination of the mode in which the unit consumes the majority of its energy. Ideally, energy use should be dominated by the on-cycle mode, as this is the mode in which useful heat is produced. Operating modes for units 1, 2, 3, 4, and 5 are shown in Figure 6, Figure 7, Figure 8, Figure 9, and Figure 10 respectively. For units 3, 4, and 5, the standby and/or off-cycle fractions include energy used to provide ventilation air distribution and recirculation. For all units, standby and off-cycle fractions include crankcase and drain pan heater energy input, which is examined in further detail in Figure 11. For systems in which auxiliary operated during defrost cycles (units 2, 4 and 5), this energy was accounted for in the defrost fraction.

Across the five units tested, the operating modes show a common seasonal pattern of predominantly on-cycle mode in the sub-zero ambient temperature bins, gradually transitioning to cycling off and then standby as the ambient temperature rises. The units differ markedly in how much auxiliary energy they require, with units 2 and 4 requiring substantially greater auxiliary energy use than the other units tested. Units 2 and

5 use significantly more energy during defrost cycles due to the fact that these units operate the auxiliary heater elements in conjunction with the defrost cycle. Unit 4 also employs auxiliary heating during defrost; however, defrost events for this unit were less frequent and less severe, resulting in a lower overall defrost energy fraction. Nevertheless, unit 4 showed a higher fraction of cycling-off operating mode due to the 3 stage (non-variable) operation of the unit and heavy reliance on auxiliary heating. Unit 3 did not operate the electric auxiliary at any time during the field monitoring. Possible explanations for this include: the fact that the home was kept cooler (19.8°C on average) than the indoor design heating temperature used for design heat loss calculations (22°C) and the homeowner implemented an overnight thermostat setback of 3-5°C, changing periodically over the measurement period. The homeowner also had the heat pump operating in efficiency mode, which allowed the heat pump to recover from overnight setback when the thermostat was set higher in the morning without switching to auxiliary, with recovery times reaching >2hrs in some cases.

Overall, the figures highlight three operating modes that affect overall seasonal performance: (i) the use of auxiliary energy and the temperature at which auxiliary becomes more dominant, (ii) the intensity of defrost cycles, and (iii) the switch to cycling off and standby modes that become the majority share in milder weather.

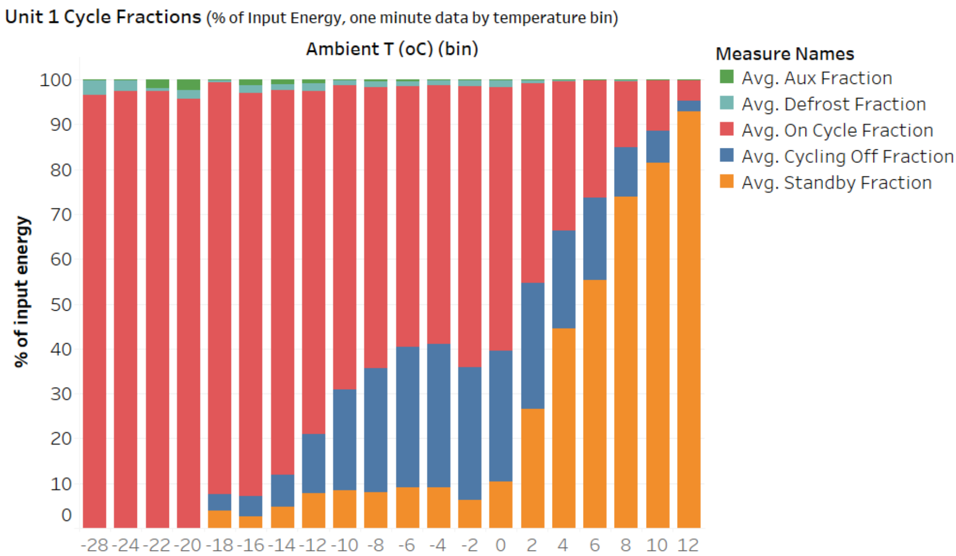

Figure 6: Unit 1 Operation Modes

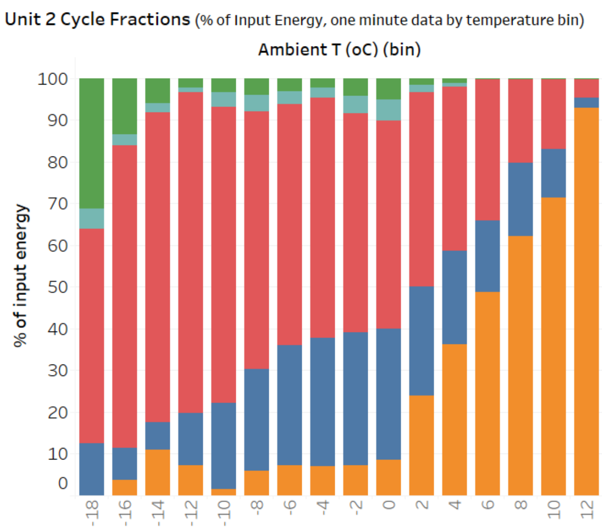

Figure 7: Unit 2 Operation Modes

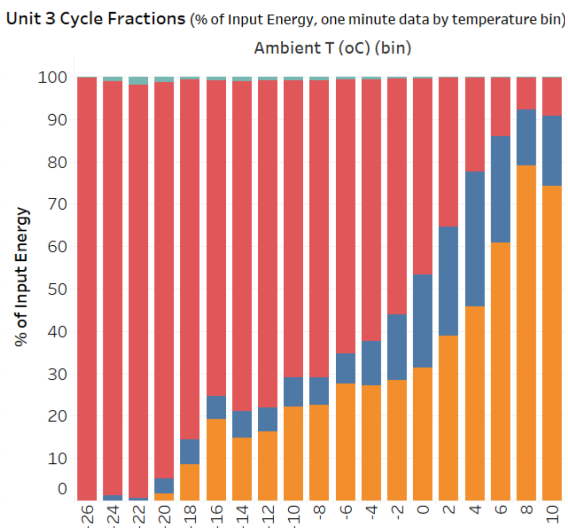

Figure 8: Unit 3 Operation Modes

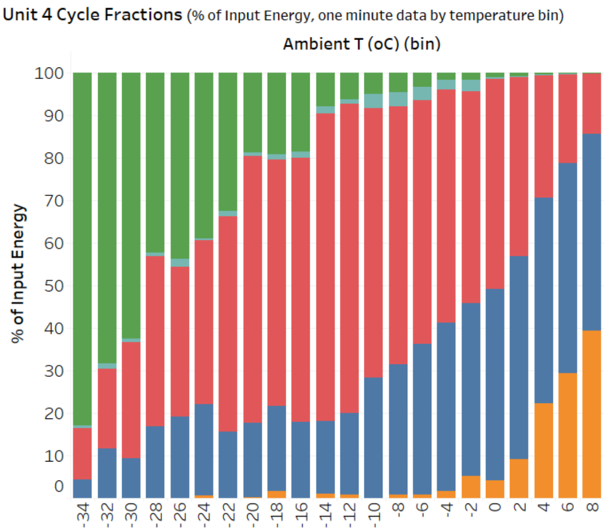

Figure 9: Unit 4 Operation Modes

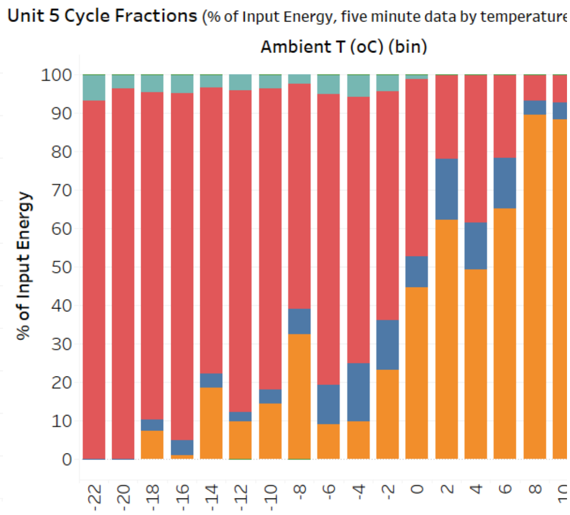

Figure 10: Unit 5 Operation Modes

Figure 11 depicts the energy consumption of the drain pan / crankcase heater as a fraction of total energy consumption for the five units monitored. There were significant variations in energy use and controls for the drain pan / crankcase heater for each of the units with unit 2 showing significantly higher consumption of the drain pan / crankcase heater across the operating range, particularly at warmer ambient temperatures. This was due to the drain pan / crankcase heater running for continuous periods of time in the absence of an effective control strategy to disable the heater when not required, such as at outdoor air temperatures above freezing. Despite the inefficient drain pan / crankcase heater control, unit 2 was shown in Figure 4 and Figure 5 to have the highest performance among the units tested in ambient temperatures above 0°C.

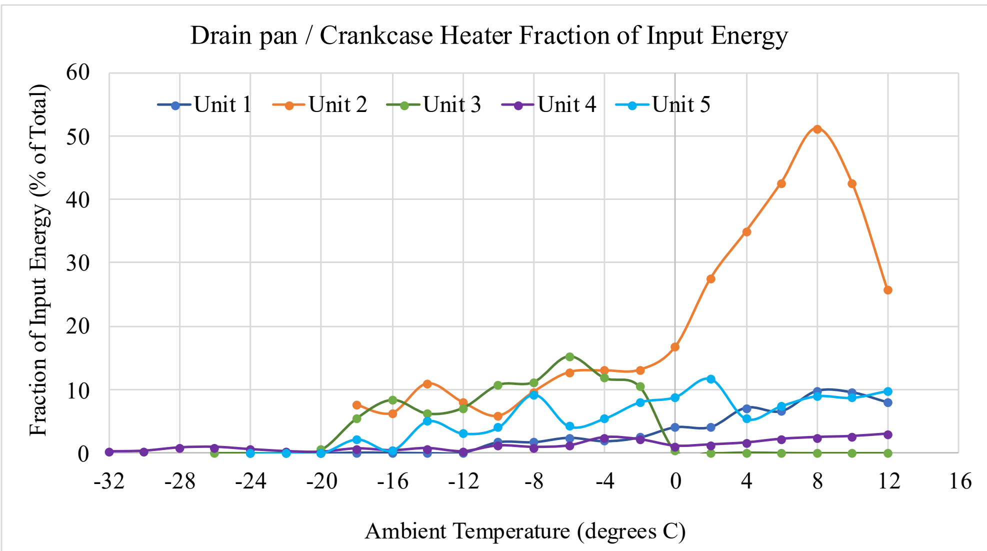

Figure 11: Drain pan / Crankcase Energy Fraction of Total Input Energy

3.3. Seasonal Heating Performance

The seasonal coefficient of performance in heating ($SCOP_H$) results for the systems evaluated based on performance during on cycles (which includes defrost cycles) are shown in Table 2. The values are calculated by first developing linear regressions of the measured on-cycle COP data shown in Figure 5 in order to determine performance across the entire range of heating bin fractions for the given climate location. Heating bin fractions for the climate locations where the heat pumps were installed were gathered from Canadian Weather Year for Energy Calculation database [10] by applying daily heating degree day and "previous 7-day" heating degree day "stage gates" as described in Annex G of the Canadian Standards Association CSA SPE-07:23 "Load-based and climate-specific testing and rating procedures for heat pumps and air conditioners" [11]. The field-measured COP data (with linear extrapolated data where needed) was then multiplied by the corresponding heating bin fraction and then summed to arrive at a seasonal COP.

To enable an evaluation of the performance benefits of the CCHP Challenge systems tested as compared to previous systems tested, CCHP Challenges systems 1 and 2 were compared to the variable capacity CCHP. Recall that all three of these systems were tested at the same test facility (the CCHT), under identical loads with seasonal performance evaluated using the same weather files. The average $SCOP_H$ results for the two CCHP Challenge systems tested at the CCHT was 3.35, while the $SCOP_H$ for the variable capacity system was 2.8, yielding a performance improvement for the CCHP Challenge systems of about 20% over the variable capacity CCHP under identical loads.

Table 2: On-Cycle Seasonal Coefficient of Performance for the CCHPs Evaluated

Field Measured Metric	CCHP Challenge Unit 1	CCHP Challenge Unit 2	CCHP Challenge Unit 3	Multi-stage CCHP	Variable Capacity CCHP
On-cycle Seasonal Coefficient of Performance in heating ($SCOP_H$)	3.2	3.5	3.0	2.3	2.8

4. Discussion and Conclusion

The units evaluated exhibited a range of operating modes that differed widely between units and installation peculiarities, which affected overall performance. Several operating behaviours were identified that negatively affected performance and therefore represent opportunities for improvement:

- Unit 2 operated auxiliary heating during defrost cycles and frequently during normal on-cycle operation, leading to excessive auxiliary heating usage in cold outdoor temperature bins;
- Unit 3 operated with the blower fan set to continuous high speed, a configuration selected by the homeowner for pet hair management with monthly filter replacement, which resulted in excessive fan energy consumption;
- Unit 4, installed in a net zero home, cycled frequently, particularly at moderate outdoor temperatures. It also operated with the blower fan set to continuous low speed operation so as to supply ventilation air throughout the home and consumed a great deal of electric auxiliary heating; and
- Unit 5 operated a single stage auxiliary heating element during defrost cycles and used a 3-speed indoor blower fan that did not fully modulate in sync with the outdoor unit in contrast to units 1 through 3, which employed fully modulating indoor blower fan motors.

In conclusion, the three CCHP Challenge systems evaluated (units 1-3) outperformed the commercially available CCHPs (units 4,5), having greater than 20% higher seasonal heating COPs. When considering performance across all cycles / operating modes, units 1 and 3 were the higher performing CCHP Challenge systems in the coldest outdoor temperatures, achieving COPs of 1.8 at -26°C, while unit 2 was the highest performing CCHP Challenge system in the most moderate outdoor temperatures, achieving a COP of 4.0 at 6°C.

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