



12th IEA Heat Pump Conference 2017



Performance Testing of Cold Climate Air Source Heat Pumps

Martin Kegel^{a*}, Jeremy Sager^b, Martin Thomas^b, Daniel Giguere^a, Roberto Sunye^a

^a Natural Resources Canada, CanmetENERGY, 1615 Blvd Lionel Boulet, Varennes, J3X-1S6, Canada

^b Natural Resources Canada, CanmetENERGY, 1 Haanel Drive, Ottawa, K1A-1M1, Canada

Abstract

The recent advancement of variable capacity technologies has led to the introduction of cold climate air source heat pumps (CCASHP) capable of efficiently meeting space heating loads at low ambient temperatures. Widespread adoption in Canada is still hindered however due to a lack of detailed and reliable performance data to accurately assess the benefits of CCASHPs. This paper presents the results and observations of the detailed performance tests conducted on centrally ducted and ductless CCASHP systems with the goal of addressing barriers and identifying potential improvements.

Performance tests revealed that while the systems were able to efficiently heat at low ambient temperatures, the manufacturer performance curves were difficult to interpret and not entirely suitable to assess the benefit of these system types. The efficient modulation to meet varying space heating loads does result in improved seasonal efficiencies over single stage systems. Further energy savings can be achieved by improving or modifying the defrost control strategies. In ducted systems, avoiding the use of the auxiliary heating system during defrost can result in 40% energy savings per defrost cycle. In ductless systems, the system operates at full power immediately following the defrost cycle, which can be of concern for utility companies as increased adoption could result in more strain on the local grid.

Selection and/or peer-review under responsibility of the organizers of the 12th IEA Heat Pump Conference 2017.

"Keywords: Cold climate air source heat pump; performanc testing; ducted; ductless"

1. Introduction

The Canadian residential sector accounts for 17% of Canada's secondary energy end use and 14% of the country's greenhouse gas (GHG) emissions [1]. Space heating represents 63% of this energy end use and space cooling represents only 1% of the residential sectors energy end use although the amount of floor space cooled through air conditioners has tripled since 1990 [1] due to increased demand for summer thermal comfort. The use of heat pumps to efficiently meet space heating and space cooling loads can be an attractive solution for reducing residential energy end use, especially if introduced through the space cooling market.

Conventional air-source heat pump systems often have difficulty providing sufficient heating capacity at the low outdoor temperatures common during a Canadian winter. With the advancement of variable capacity technologies, cold climate air source heat pumps have been introduced in the market, which can efficiently meet space heating loads at these low temperatures. Previous studies [2, 3] have shown that cold climate air source heat pump systems can be a financially viable option for homeowners to reduce energy, utility costs and ultimately GHG emissions; however their exact performance characteristics and benefits are not widely known, which can hinder increased adoption. Furthermore, in regions where fuel for space heating is being offset through these heat pump systems, utility companies have an interest in knowing what kind of impact these systems can have on the local utility grid. Some studies have been performed on field trialing cold climate air

* Corresponding author: martin.kegel@canada.ca

source heat pumps; however their exact performance was often estimated from manufacturer performance curves [4] and thus a need was identified to better map the performance of these variable capacity systems.

The type of space heating system and predominant space heating fuel source in the Canadian residential sector varies depending on the region and age of the house. From the survey of household energy use [5], centrally ducted air based distribution systems are most widely used, representing 54% of the type of space heating systems, with electric baseboard heating representing the second highest proportion at 30%. Hydronic boilers, heating stoves and gas fireplaces and heat pumps make up the remaining 16%. It is worth noting that only 2.5% of the Canadian households have a heat pump system installed. Natural gas is the predominant fuel source for space heating and is used in 46% of the houses, with electricity representing the 2nd highest at 41%. Heating oil, wood, propane and dual fuel system make up the remaining 13%. Thus, with the high use of fossil fuel and electric resistance for space heating, the potential of heat pump systems achieving significant energy savings and GHG emission reductions in the residential sector are apparent. Fig. 1 summarizes the space heating system type and space heating fuel source percentages in Canada.

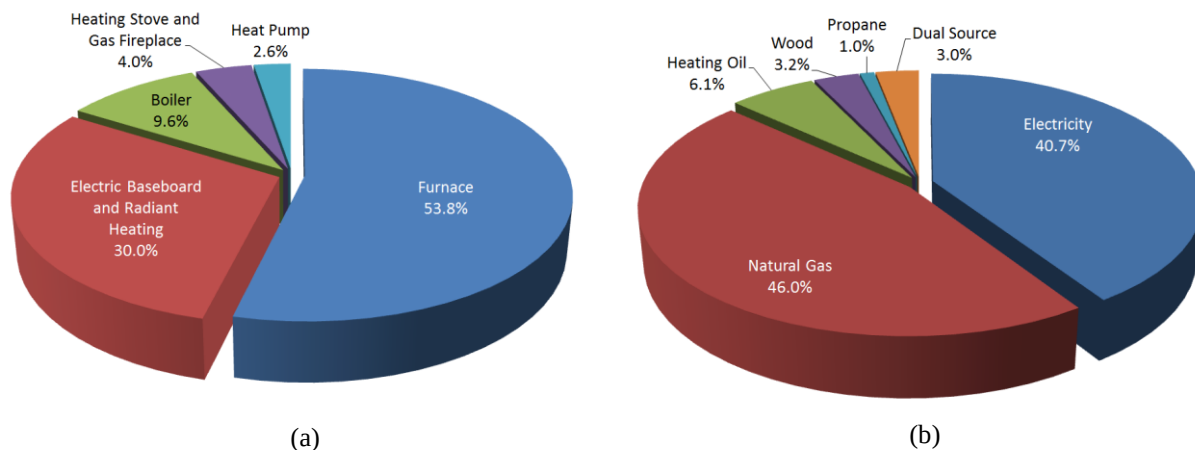


Fig. 1. Percentage of Canadian Households (a) Space Heating System Type (b) Space Heating Fuel Source

Regionally, electric space heating systems are more prominent in Eastern Canada and Quebec due to the lack of natural gas or higher natural gas costs. In Ontario and the Prairies, natural gas is the prominent fuel source due to the extremely low rates with British Columbia having an equal split between electricity and natural gas for space heating fuel.

Cold climate air to air heat pumps come in two forms in the North American market – ducted and non-ducted systems. Ducted systems have the indoor coil installed in the ductwork delivering space heating and cooling throughout the household. Non-ducted systems have indoor coils inside an ‘indoor unit’ mounted on the wall or ceiling to individually heat and cool a space. In some instances, up to eight indoor coils can be connected to one outdoor unit to efficiently heat/cool different zones within a household. Through the variable capacity technology of cold climate air to air heat pumps, space heating loads can be met at low ambient temperatures while still providing the capability to modulate to low speeds (and lower capacities) at warmer ambient temperatures. While these systems are advertised as being efficient, their actual performance is still largely unknown. Builders and design consultants across Canada have identified the need for better simulation tools and knowledge of these systems to increase market adoption.

To address this, the Natural Resources Canada/CanmetENERGY laboratories in Ottawa, Ontario and Varennes, Quebec have performed tests on ducted and non-ducted cold climate air to air heat pumps systems to acquire performance data and identify potential areas of improvement. Testing was performed in a simulated household at the Canadian Centre for Housing Technology [6] on a ducted cold climate air to air heat pump system. Testing of a non-ducted cold climate air to air heat pump was performed in an environmental test chamber at CanmetENERGY-Varennes (additional comparative analysis was completed at the Canadian Centre for Housing Technology [7]).

2. Ducted Cold Climate Air to Air Heat Pump Performance Testing

2.1. Test Facility Description

The Canadian Centre for Housing Technology (CCHT) is a facility located in Ottawa, Ontario, designed to evaluate the whole-house performance of residential technologies [8]. It includes two highly instrumented, identical, unoccupied houses (Fig. 2) comprised of two storeys and 223 m² (2,400 ft²) of total heated floor area not counting the full basements. Each house has a design heating load of 10.7 kW (36,600 Btu/h) and design cooling load of 4.3 kW (14,700 Btu/h or 1.25 tons) (calculated according to the Canadian Standards Association (CSA) F280-12). Occupancy is simulated by computer controlled operation of lights and appliances, use of hot water, and generation of heat to simulate the presence of occupants.



Fig. 2. Images of CCHT Twin House Test Facility

Measurements are captured by two data monitoring systems located in each of the two houses. One system measures comfort and performance data on a 5 minute basis and is used to determine comparative performance of the houses in all experiments. The second system is used to capture detailed performance data on a 5-minute and minute-by-minute basis, providing a backup source of data for the first system, while offering more detailed data used for system commissioning and research purposes.

Data collected for the experiment at both the five minute and one minute interval included:

- Indoor and outdoor unit electrical consumption.
- Indoor temperature and relative humidity (average for house) and room-by-room temperatures.
- Outdoor ambient temperature, relative humidity and global solar radiation.
- CCASHP air supply and return duct temperatures.
- CCASHP air supply flowrate.

2.2. Test Results

A centrally ducted cold climate air source heat pump with a rated heating and cooling capacity of 11.7 kW (40,000 BTU/hr) and 10.5 kW (36,000 BTU/hr) respectively, was selected and tested in 2012/2013 to acquire heating cooling and shoulder season performance data. Additional testing was completed in 2015/2016 with a next generation air handler and controls. The system indoor and outdoor units are shown in Fig. 3a and Fig. 3b respectively.

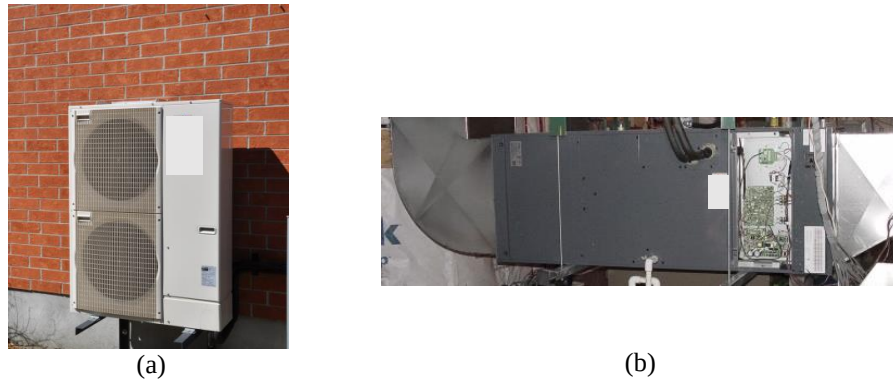


Fig. 3. Images of (a) Ducted CCASHP Outdoor Unit, (b) Ducted CCASHP Indoor Unit

The coefficient of performance (COP) during the heating and cooling season (2013) was calculated based on output power divided by input power. Output power (Heat Output or Cooling Output) was calculated based on the enthalpy difference between return air to the system and supply air from the system and the measured airflow. Output power also includes output heat provided by the auxiliary electric resistance heating element when it operated during defrost cycles or when the system was unable to meet the entire household heating load during cold temperatures. It should also be noted that the heat recovery ventilator (HRV) installed in the house had a fresh air supply that was connected to the return air system. Return air temperature therefore included the influence of fresh air supplied by the heat recovery ventilator. Input power includes electricity consumption of the auxiliary heating element occurring during defrost cycles as well as electricity consumed by the air handler during low speed continuous air circulation. These factors, as well as factors unique to the CCHT Test House may result in performance data that differs from standard performance results completed according to the Canadian Standards Association CSA-656 methodology. The measured performance of the ducted CCASHP at various temperature bins is shown in Fig. 4.

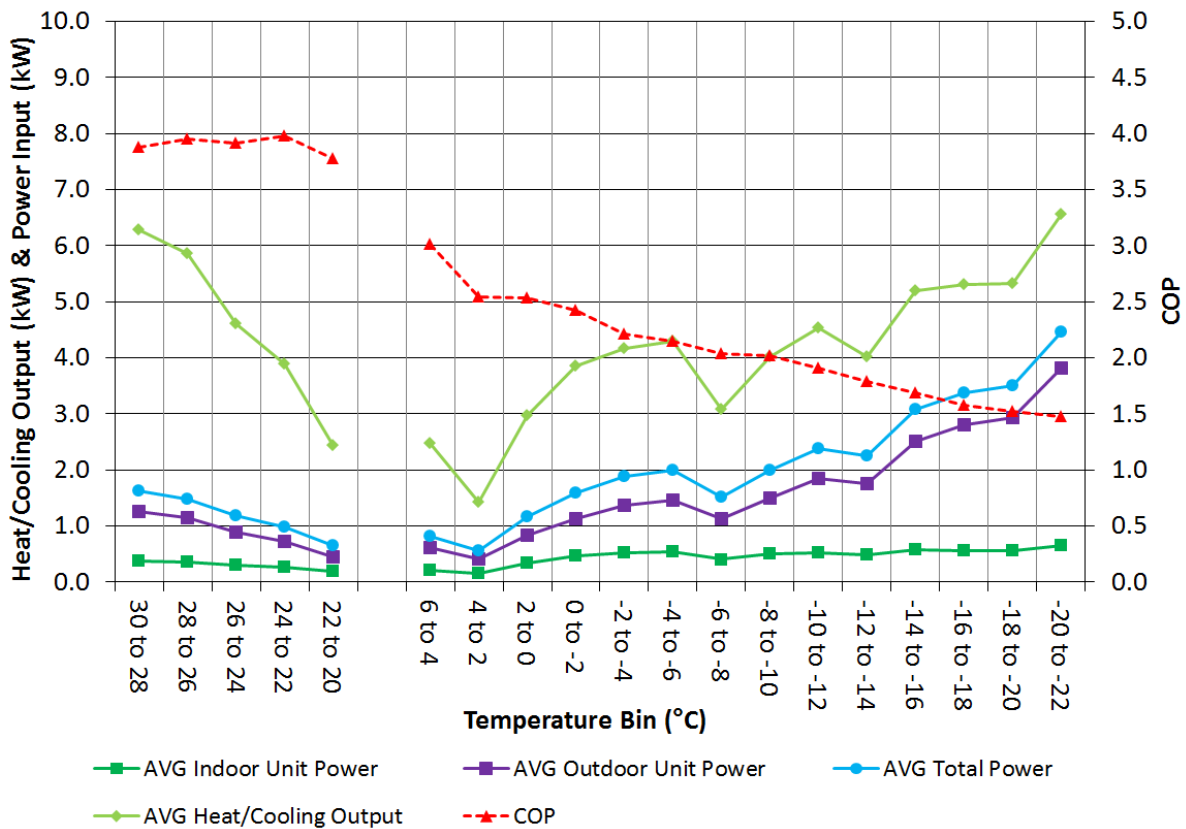


Fig. 4. Electrical Input, Output and Coefficient of Performance of a Cold Climate Air Source Heat Pump at the CCHT

From the testing performed in 2013, the heating and cooling season measured performance was:

- Heating Season: The COP values for the CCASHP ranged from an average of 1.5 at -21.1°C outdoor temperature to 3.0 at 4.9°C outdoor temperature.
- Cooling Season: The COP values in cooling ranged from an average of 3.7 at 20.5°C outdoor temperature to 4.1 at 23.5°C outdoor temperature.

The heating performance is as anticipated with a decrease in COP as ambient temperatures fall (associated with the higher temperature lift between the evaporating and condensing temperatures). In cooling mode, the COP remained fairly constant across the temperature bins. This is attributed to the likely oversizing of the heat pump for the actual household space cooling load. With a design cooling load of 4.3 kW and a rated system cooling capacity of 10.5 kW, the system operates at low part loads during cooler ambient temperature conditions. Although the system is designed to be variable capacity, inverter driven systems cycle on-off if there is insufficient load (less than 30% of rated capacity [9]). With the blower fan on continuously, the system results in having a lower COP cycling on-off than at higher ambient temperatures where the compressor operates more continuously; albeit at a slow speed effectively meeting the household cooling load more efficiently. This phenomenon is further highlighted during the shoulder season data (not shown) temperature ranges from 6°C to 20°C, where the calculated COP data drops off substantially and even becomes negative. This is attributed to the low space heating loads met by solar gains and the continuous blower operation to maintain heat recovery ventilation.

2.3. Defrost Strategies

It was noted during testing in 2013 that the air handler fan was set to run during defrost cycles and, as a result, the electric resistance backup heaters activated in order to temper the cooler air being supplied by the system as it defrosted (note that the electric resistance backup heaters in this setup were two stage). In Fig. 5 below, this setup is denoted “CCASHP Defrost with Auxiliary”. This approach resulted in substantial energy consumption during defrost cycles for the indoor unit (which includes the auxiliary heater), with total electrical consumption for the defrost cycle in this setup being 1.18 kWh. An alternative defrost strategy was evaluated in 2016 (under comparable conditions as the 2013 test) such that the air handler fan would not operate during defrost cycles, therefore also preventing operation of the electric resistance auxiliary heaters. This was done with onboard dip switch settings for the system tested. With such a strategy, the “heat” used for defrost was taken from the thermal mass and standing air within the air handler. As a result, one would expect perhaps higher outdoor unit energy consumption and/or longer defrost cycles, but perhaps less overall electrical consumption due to less use of the auxiliary heaters. This can indeed be seen in Fig. 5, in which the total electrical consumption for the defrost cycle in this setup was 0.65 kWh, a reduction of about 40% per defrost cycle compared to the previous strategy. For an entire heating season, with a range of 5 to 10 defrost cycles daily (depending on ambient temperature and humidity conditions), it can be seen that an improved defrost control strategy can have significant energy saving potential.

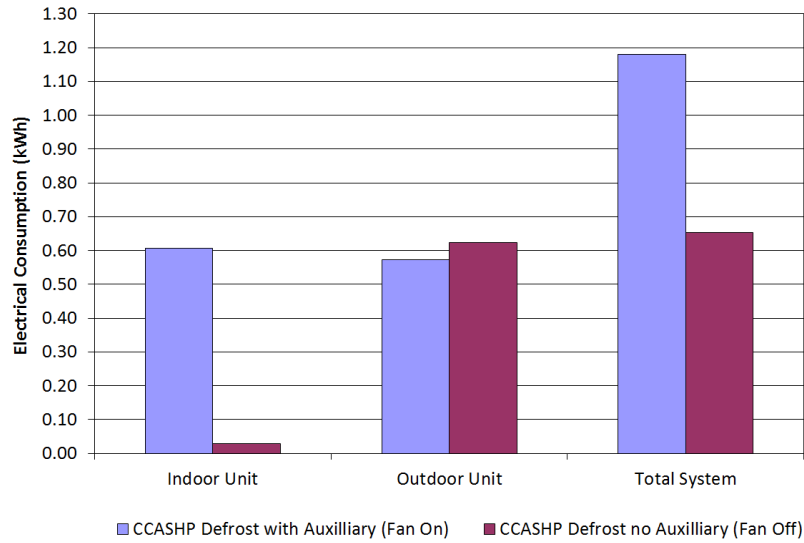


Fig. 5. Comparison of Measured Electrical Consumption Before, During and After a Defrost Cycle for Two Different System Setups

3. Non-Ducted Cold Climate Air to Air Heat Pump Performance Testing

3.1. Test Bench Description

An environmental controllable test chamber was constructed at the CanmetENERGY-Varennnes laboratory to provide the capability to induce a wide range of heating loads at different user-defined ambient temperatures. The test bench is comprised of two insulated 3.6 m wide x 4.9 m long x 3.6 m high sheds with variable speed exhaust fans and intake louvers to induce a wide range of potential operating conditions. The indoor unit is installed in one shed, while the outdoor unit is installed in the other (Fig. 6).



Fig. 6. Fully Instrumented (a) Outdoor Unit and (b) Indoor Unit

The outdoor ambient air is used to induce a load on both sheds in order to vary operating conditions. In the outdoor unit shed, the evaporator (in heating mode) continues to cool the shed, while ambient air is brought in to heat the space. In the event that test conditions warmer than current outdoor temperatures are desired, electric heaters have been installed to reheat the space. In the indoor unit shed, ambient air is brought in to induce a heating load (during a heating mode test). As no chiller was installed to induce a cooling load on the shed, performance testing could only be done when ambient temperatures were suitable. The amount of ambient air delivered to each shed was varied using variable speed exhaust fans and louvers. While relative humidity levels

could not be controlled, the test bench was deemed to provide suitable flexibility to obtain a wide range of performance data. Both the indoor and outdoor units of the air to air heat pump were fully instrumented with thermocouples, a refrigerant mass flow meter, pressure transducers and watt transducers. Using REFPROP, the enthalpies at the various points in the cycle were calculated using knowledge of the temperatures and pressures (Fig. 7).

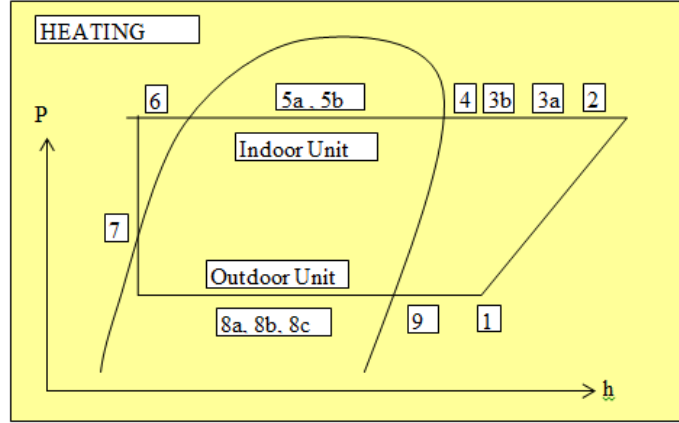


Fig. 7. Pressure Enthalpy Diagram of Heat Pump in Heating Mode

System performance was calculated using the enthalpies, refrigerant mass flow, and electrical power input obtained during testing. The heat output of the system was calculated using the entering and exiting enthalpy of the indoor unit (equation 1), while the renewable energy input was estimated by calculating the enthalpy change across the evaporator assuming isenthalpic expansion (equation 2).

$$Q_{Heat} = \dot{m} (h_4 - h_6) \quad (1)$$

$$Q_{Input} = \dot{m} (h_1 - h_6) \quad (2)$$

To ensure results were valid the energy balance of the system was also verified (equation 3).

$$\dot{m} (h_1 - h_6) + W_{comp,in} = \dot{m} (h_2 - h_4) + \dot{m} (h_4 - h_6) \quad (3)$$

The compressor power was not directly measured. However, it was estimated by subtracting the measured indoor unit power and the estimated outdoor fan power from the total measured system power consumption. Preliminary testing indicated that the compressor outlet temperature was incorrect, as isentropic efficiencies exceeded 100%. This was expected, as the temperature of the refrigerant exiting the compressor could not be directly measured (only the surface temperature was recorded at the compressor exit). Unfortunately, compressor performance curves are unavailable to estimate the isentropic efficiency of the system, and thus the performance of the system was calculated by estimating the isentropic efficiency to achieve an energy balance across the system. Also, the enthalpy at point 4 was estimated from the measured temperature at that point and the pressure exiting the compressor. In some instances, the resulted calculated enthalpy was in the sub-cooled region (point 6) (slow response of thermocouple and thermocouple accuracy) and thus point 4 was taken at the saturated vapor point at the respective pressure.

In order to compare the measured heat pump performance versus the manufacturer published data, the compressor operating speed must be known, as heat pump performance data is estimated using rated capacity correction curves. The compressor speed during testing could not be measured at the time, but was instead estimated through the refrigerant mass flow measurement. The heat pump was operated at two known operating speeds, and the subsequent refrigerant volumetric flow rate was correlated to that operating speed. The volumetric flow was then assumed to vary proportionally with the compressor speed at different operating conditions.

3.2. Test Results

A market available cold climate air source heat pump system with a rated heating and cooling capacity of 4.0 kW (13,600 BTU/hr) and 3.5 kW (12,000 BTU/hr), respectively, was used for all ductless testing. Several tests at different ambient temperature conditions were conducted over an 8 hour period in order to obtain a wide range of performance data. The heat pump was operated in such a way as to provide as much heat as possible (setpoint of 27°C). Sample performance test results are given in Fig. 8, showing the heat output, power input, COP, evaporating temperature and condensing temperature over an 8 hour period at an outdoor shed temperature of -25°C. The sudden decrease in the heat output indicates periods when the heat pump entered a defrost cycle – approximately every 2 ½ hours. The heat pump uses a reverse cycle strategy for defrost, pumping hot refrigerant gas through the outdoor coil.

Although the heat pump was operating to provide the maximum heat output, the system was found to only operate at maximum speed for short periods (20 minutes) after the defrost cycle. After these short periods, the heat pump would then ramp down and operate at approximately 70% of the maximum speed despite the desired temperature setpoint not being met (Fig. 9). This was observed at ambient test conditions of -15°C and below, and is attributed to heat pump controls which are designed to avoid possible compressor overheating, particularly during times when the heat pump is unable to meet the temperature setpoint within a reasonable amount of time.

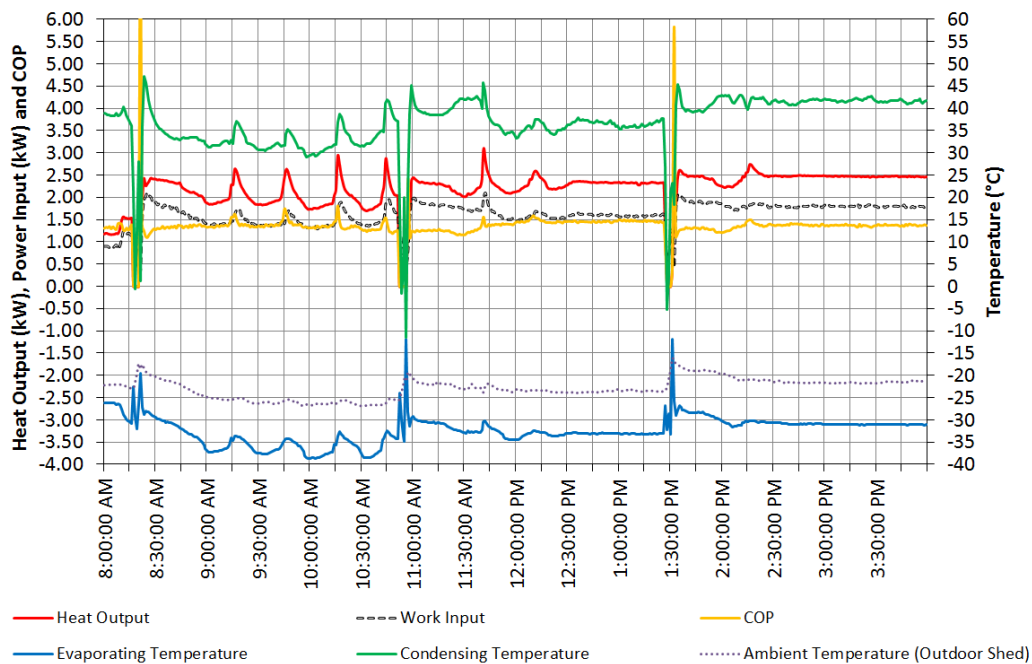


Fig. 8. Measured Heat Output, Power Input, COP, Evaporating Temperature and Condensing Temperature at -25°C Ambient

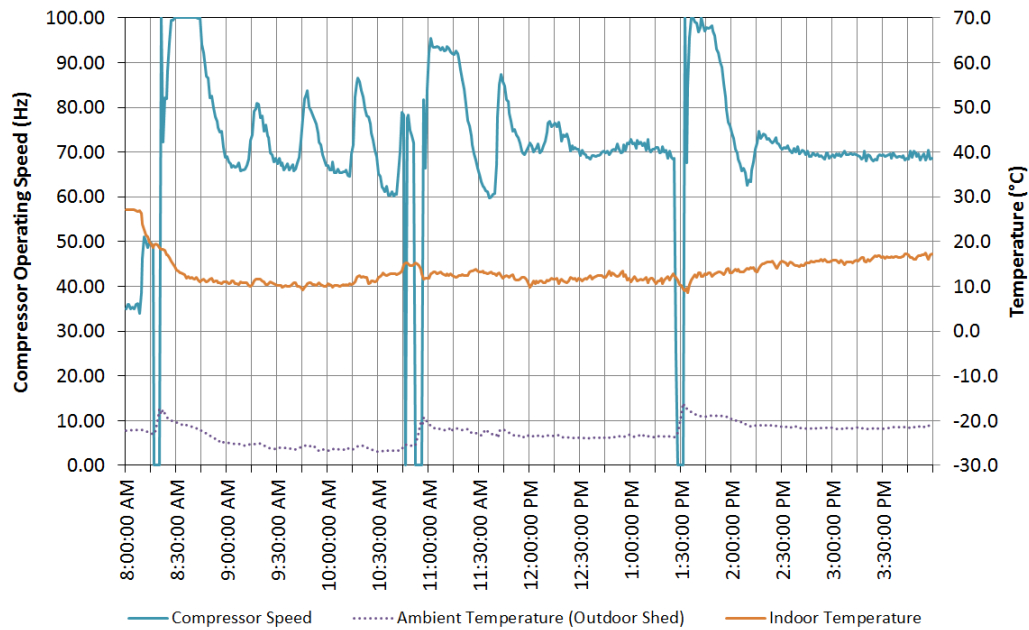


Fig. 9. Measured Compressor Frequency and Maintained Indoor Temperature at -25°C Ambient

Above -15°C, the heat pump was able to meet the indoor shed temperature but appeared to operate at an average frequency around 60 Hz regardless of the ambient test temperature. Fig. 10 summarizes the measured heat output, power input and average compressor operating frequency at different ambient temperature conditions. The integrated results include the heat pump going through at least one defrost cycle.

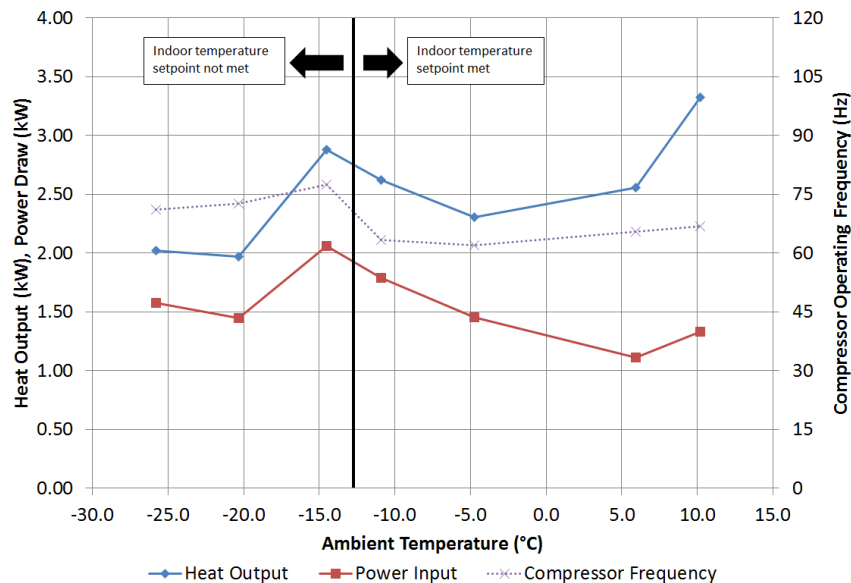


Fig. 10. Measured Heat Output, Power Draw and Compressor Operating Frequency at Different Ambient Temperatures

The somewhat constant compressor speed at warmer temperatures is attributed to the number of the defrost cycles occurring over the sampling period. Essentially, the closer the outdoor air temperature was to 0°C, the more frequent the defrost cycles became. Although the compressor operates at a slower speed to meet the heating load at warmer ambient temperatures, the frequent defrost cycles result in the system operating at maximum speed more frequently. The frequency of defrost cycles appeared to be timed depending on the ambient temperature. Defrost cycles would last between 1 ½ to 5 minutes depending on the ambient temperature, with longer defrost times occurring at colder ambient temperatures. Table 1 summarizes the observed defrost

cycle frequency and length at various ambient test temperatures. Ambient relative humidity was not measured during testing, which likely has an impact on the length between defrost cycles.

Table 1. Defrost cycle time at various ambient temperatures

Ambient Test Temperature	Defrost Cycle	Defrost Length
> 5°C	No Defrost	No Defrost
0°C to 5°C	2 ½ hours	1 ½ minutes
-5°C to 0°C	1 ¼ hours	3 minutes
-10°C to -5°C	2 ½ hours	3 ½ minutes
-15°C to 10°C	2 ¼ hours	1 ½ minutes
-20°C to -15°C	2 ½ hours	4 ½ minutes
-25°C to -20°C	2 ½ hours	4 ½ minutes

To see how the steady state results compared to those published by the manufacturer, the measured and published heat output is plotted for various ambient temperatures (Fig. 11). The manufacturer published data was estimated from performance curve correction factors and rated capacities at different entering outdoor coil temperatures (with a 21.1°C dry bulb entering indoor coil temperature).

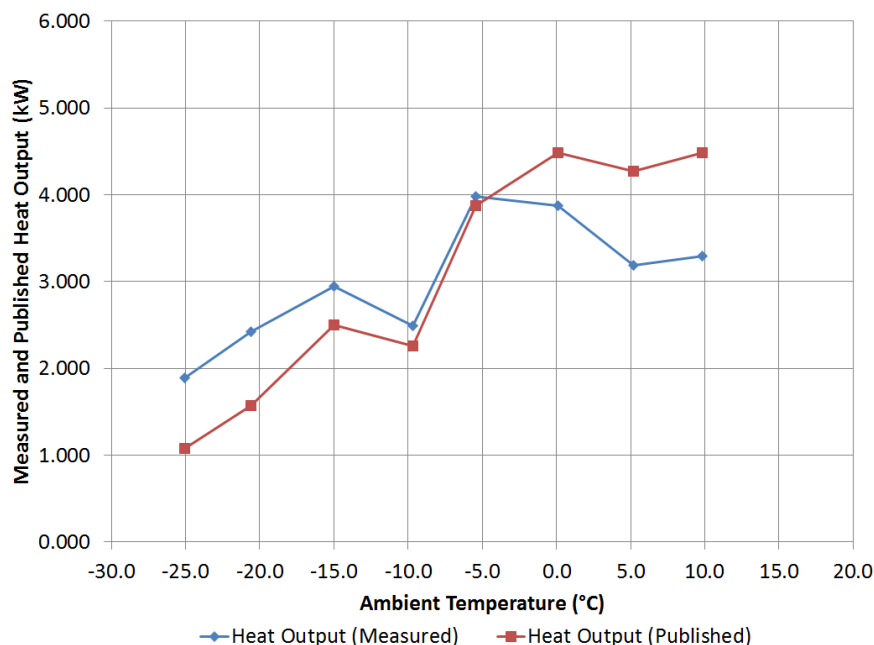


Fig. 11. Measured Heat Output versus Manufacturer Published

Results indicate a higher than anticipated heat output at ambient temperatures below -10°C, attributed to lower condenser inlet air temperatures than those published by the manufacturer. At ambient temperatures above -5°C, the measured heat output under-predicted the manufacturer published data, which can be attributed to the difficulty in deriving the manufacturer performance data from the published heat capacity correction curves. The power output was not compared as the estimated manufacturer published power input resulted in very low power draws at low ambient temperatures.

4. Conclusion and Future Work

The recent advancement of variable speed compressor technology has led to the introduction of cold climate air to air heat pumps in the Canadian market. Variable capacity systems are constructed to efficiently meet space heating loads at low ambient temperatures, while providing the ability to efficiently modulate compressor speeds to maintain comfort conditions at warmer ambient temperatures. With the abundance of natural gas and electric resistance space heating systems used in the Canadian residential sector, cold climate air source heat pumps can

significantly reduce building energy consumption and greenhouse gas emissions. However, their adoption is hindered by a lack of reliable performance data and a resulting inability to fully estimate their energy saving benefits. To address this, Natural Resources Canada/CanmetENERGY has tested ducted and non-ducted air source heat pump systems to acquire performance data and identify potential areas of improvement.

A ducted cold climate air to air heat pump was installed at the Canadian Centre for Housing Technology to assess system performance in the winter, summer and shoulder seasons. Performance testing showed that the heat pump was capable of efficiently meeting space heating loads down to -21°C with a COP of 1.5, while still demonstrating a strong COP in cooling mode. It was also found that the defrost settings on the heat pump can have a significant impact on “as installed” system performance. The standard system controls operate the air handler fan while defrosting, which in turn activate the electrical auxiliary heaters to alleviate any potential discomfort associated with blowing cooler air into the space. Alternatively, the system can be setup so as not to operate the air handler fan while defrosting, therefore eliminating the need for electric auxiliary heating. This latter strategy can result in significant energy savings versus the former strategy for equivalent defrost cycles, with potential reductions in electricity use of 40% per defrost cycle.

A ductless cold climate air source heat pump was tested in an environmental test chamber capable of inducing a wide range of heat pump operating conditions. Preliminary testing showed that the non-ducted system was capable of efficiently meeting space heating loads down to -25°C . However, the system would operate at a reduced capacity if the temperature setpoint could not be met in a sufficient period of time. It was also observed that regardless of the desired heating load, the system would operate at maximum capacity after a defrost cycle before ramping down, which can be of concern for local utility companies. The measured heat output of the heat pump also did not match the manufacturer derived performance, highlighting the need for more testing and improved published performance curves.

Future work is planned to perform more testing on the installed ductless heat pump in the environmental test chamber. Recent test bench improvements include submetering the compressor and outdoor fan power, in addition to monitoring the compressor speed and relative humidity inside both test huts. Test bench controls have also been improved to better induce realistic loads, allowing for improved performance mapping and assessment of heat pump control strategies. Using this performance test data, an improved simulation component model will also be developed to help consultants better predict the impact that these systems can have on building energy performance and greenhouse gas emissions.

References

- [1] Natural Resources Canada. *Energy Efficiency Trends in Canada 1990 – 2013*. Ottawa: Natural Resources Canada; 2016, ISSN 1926-8254.
- [2] Kegel M., Tamauskas J., Sunye R., Giguere D., 2014. “Heat Pumps in the Canadian Residential Sector,” 11th International Energy Agency Heat Pump Conference. Montreal, Canada, paper #O2.4.4.
- [3] Kegel, M., Tamasauskas, J., Sunye, R., 2014. “Integration and Evaluation of Innovative and Renewable Energy Technologies in a Canadian Mid-rise Apartment,” 9th System Simulation in Buildings Conference. Liege, Belgium, paper #038.
- [4] Le Lostec, B., Nouanegue, H. F., 2014. “On-site Performance of Air Source Heat Pumps”, 11th International Energy Agency Heat Pump Conference. Montreal, Canada, paper #O2.4.3.
- [5] Natural Resources Canada. *Survey of Household Energy Use 2011 Detailed Statistical Report*. Ottawa: Natural Resources Canada; 2014, ISBN 978-1-100-234468-7.
- [6] Sager, J., Armstrong, M., and Szadkowski, F., 2013. “Cooling and Heating Season Performance Assessment of a Cold Climate Air Source Heat Pump,” Natural Resources Canada, CanmetENERGY. Ottawa, Canada.
- [7] Sager, J., Thomas, M., Armstrong, M., Gusdorf, J., and Szadkowski, F., 2014. “Cooling and Heating Season Performance Assessment of a Mini-Split Cold Climate Air Source Heat Pump,” Natural Resources Canada, CanmetENERGY. Ottawa, Canada.
- [8] Swinton, M. C., Moussa, H., Marchand, R. G., 2001. “Commissioning Twin Houses for Assessing the Performance of Energy Conservation Technologies,” Performance of Exterior Envelopes of Whole Buildings VIII. Clearwater, Florida, USA.
- [9] Filliard, B., Guiavarch, A., Peuportier, B., 2009. “Performance Evaluation of an Air to Air Heat Pump Coupled with Temperature Air-Sources Integrated into a Dwelling,” International Building Performance Simulation Conference, Glasgow, Scotland, p. 2266 to 2273.