Performance Assessment of a Variable Capacity Air Source Heat Pump and a Single-Capacity Horizontal Loop Coupled Ground Source Heat Pump System

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Abstract

This study evaluates the performance of air source heat pump (ASHP) and ground source heat pump (GSHP) systems individually and as part of their associated heating and cooling distribution systems in two side-by-side semi-attached houses. Over the monitoring period, the efficiency of both heat pumps exceeded manufacturer and EnerGuide ratings, with seasonal Coefficients of Performance (COPs) above 5 in the cooling season and above 3 in the heating season. When the energy inputs associated with the heating and cooling distribution systems were accounted for, performance decreased by between 9 and 53%. Modelled optimization scenarios showed that considerable increases in system efficiencies could be achieved by configuring fans and pumps to operate only when the heat pump compressor is on, and by upgrading the GSHP from a single stage to a two stage system to reduce compressor cycling.

Performance of the ASHP was more adversely affected by declining winter temperatures than the GSHP. However, the ASHP continued to maintain indoor thermal comfort at temperatures as low as minus 24°C without supplementary heat. Model simulations for five major Canadian cities showed that both technologies can perform well in the Canadian climate, but that residential GSHP systems are better suited to climates where winter temperatures fall below minus 24°C when the ASHP would not be able to operate.

A simple cost analysis relative to electric resistance heating and central cooling revealed that although up-front equipment and installation costs are higher for both systems, the ASHP is more affordable with a simple payback of approximately 10 years. Even though the GSHP is slightly more efficient, simple payback is over two times longer due to higher initial capital costs. However, since the GSHP has a longer expected service life than the ASHP, the financial case for these systems would be more accurately assessed through a full life-cycle cost analysis. This simple cost analysis also omits the substantial benefits these systems offer in reducing greenhouse gas emissions which are directed related to the overall energy use.

Key words: Variable capacity air source heat pump, Ground source heat pump, coefficient of performance (COP), TRNSYS modeling, Performance evaluation and comparison, Archetype Sustainable House

1 INTRODUCTION

Between 1998 and 2004, the housing sector accounted for 17% of secondary energy use[1] in Canada and 16% of the country's greenhouse gas emissions (Natural Resources Canada, 2006). Space and water heating are the dominant residential end uses of energy, typically representing 58% and 22% of total household consumption, respectively (Cuddihy et al., 2005). Roughly 26% of the household contribution to total GHG emissions is from residential fuel use and the production of electricity for use in the home (Statistics Canada, 2008).

Heat pumps are among the most energy efficient technologies for heating and cooling buildings and providing hot water. Heat pumps function by moving heat from one place to another. A ground source heat pump uses the ground as the source and sink for heat, while air source heat pumps take heat from the outdoor air and transfer it indoors. Both can be used with a forced air or hydronics system. Since air temperatures fluctuate much more than ground temperatures, air source heat pumps often require a back-up source of heat during very cold weather to maintain indoor temperatures at desired levels.

This study assesses and compares the performance of a horizontal loop coupled 13.3 kW high efficiency ground source heat pump and a high efficiency cold climate variable capacity 10.5 kW air source heat pump. The heat pumps are installed in each of two side-by-side semi-detached LEED™ platinum houses at the Living City Campus at Kortright in Vaughan, Ontario (Fung et al., 2009; Dembo et al., 2010; Barua et al., 2010).

2 THE ARCHETYPE SUSTAINABLE HOUSES

The two side-by-side semi-attached houses, hereafter referred to as House A and House B, are 3-storey, south-facing houses with similar floor areas, internal volumes, and levels of insulation (R-30 above grade, R-20 below). Structural insulated panels were used for the roof of both houses. House B has roughly 20 % more window coverage than House A, and has triple glazed windows with higher thermal resistance than the double glazed windows in House A. The overall design heating loads of House A and House B are 7.91 kW and 7.94 kW when outdoor and indoor temperatures are -22°C and 22°C, respectively (Zhang et al., 2011). Brief description of the pertinent equipment in both houses is presented below. However, detailed differences in the mechanical systems are presented elsewhere in Barua (2010) and Zhang et al. (2011). The Archetype Sustainable House has been awarded LEED™ Platinum, EnergyStar and GreenHouse certifications (Dembo et al., 2010). Even though these are two side-by-side semi-detached houses with identical footprint, they are designed to show case different construction material and HVAC equipment. For example,

[1] Secondary energy is energy used by final consumers for residential, agricultural, commercial, industrial, and transportation purposes. It does not include intermediate uses of energy for transforming one energy form to another (e.g., natural gas to electricity) or transporting energy to market (e.g., fuel for gas pipeline compressors).

House B uses spray foam insulation in the exterior wall cavity versus rock wool insulation batt used in House A, resulting in slightly more air tight for House B.

Figure 1: Archetype Sustainable House

3 ENERGY AUDIT

Air leaks through the exterior envelope increase energy use. A blower door test was conducted to determine how much air was leaking into the houses. This entailed mounting a large fan to the door frame and drawing air out of the house to calculate the rate of air leakage and assess where the leaks were most prominent. The tests were conducted with outside temperature of –7.7 °C and inside temperature of 20.0 °C (Fung et al., 2009; Dembo et al., 2010).

Results of the blower test, presented in Table 1, showed that the exterior building envelopes were better sealed than most homes, particularly House B, which registered only 1.1 air changes per hour (ACH) during the blower test (more air changes mean leakier houses). By comparison, the Energy Star label requires a maximum of 2.5 ACH, and the new 2012 Ontario building code allows 3.1 ACH. Model simulation results using HOT2000 indicate that the energy efficiency of both houses exceeds the R-2000[2] energy efficiency standard, which is much higher than the energy efficiency ratings required under Canadian building codes (Dembo et al., 2010).

Table 1: Summary of Airtightness Results for House A and B (Fung et al., 2009; Dembo et al., 2010)

	Units	House A	House B
Net floor area	m^2	345	350 (excluding guest suite)
Internal volume	m^3	986	1036 (excluding guest suite)
Volumetric flow rate	CFM	699	665
ACH @ 50 Pa	ACH	1.204	1.091

ACH: Air Changes per Hour; CFM: Cubic Feet per Minute.

[2] R-2000 is a voluntary standard administered by Natural Resources Canada (NRCan) and is delivered through a network of service organizations and professionals across Canada. (http://www.nrcan.gc.ca/energy/efficiency/housing/new-homes/5051)

4 MECHANICAL SYSTEMS

Table 2 shows the mechanical systems in the two Houses. These mechanical systems were selected from 19 mechanical system alternatives based on a decision support matrix that used pre-defined criteria such as energy consumption, life cycle costs and greenhouse gas emissions to evaluate alternatives (Rad et al, 2007; Fung et al., 2010B).

Figure 2: Ground Source (top) and Air Source (bottom) Heat Pumps

Based on this analysis, the system selected for House A is a 10.5 kW (3 ton) high efficiency cold climate variable capacity air-to-air source heat pump with a direct-expansion-coil air handling unit (AHU) and a multi-speed fan to supply warm and cold air for space heating and cooling. The system is coupled with a natural gas mini-boiler which supplies hot water to the heating coil of the AHU when the ASHP alone is not able to supply sufficient heat when extreme low outdoor air temperature is encountered. The backup heating system is part of the overall commercial ASHP package intended to be used in clod climate like Canada. However, the natural gas mini-boiler was not operating during the study period.

The mechanical systems in House B consist of a 13.3 kW high efficiency ground source heat pump connected to two parallel 152.3 m (500 ft) horizontal loops in the yard. A propylene glycol mixture is used as the heat transfer fluid. In the cooling season, the GSHP supplies chilled water to the multi-zone AHU. A radiant in-floor heating system is used for space heating on each floor during the winter. A buffer tank is used between the GSHP and the infloor system/AHU to minimize equipment cycling. A Stirling Engine micro-cogeneration unit was installed in House B as a heating alternative to the GSHP but was not functioning over the duration of the testing period. Detailed heat pump systems and their configurations are presented elsewhere (Safa et al., 2011a; Safa et al., 2011b; Safa, 2012)

In houses with tight envelopes, ventilation systems are needed to maintain adequate indoor air quality. To avoid wasting energy, a heat recovery ventilator (HRV) is used in House A to pre-heat or pre-cool incoming fresh air by extracting the heat or coolness from indoor air being exhausted from the house. An energy recovery ventilator (ERV) installed in House B operates in a similar fashion but also provides moisture control. House A and B also include a 0.91 m long grey water heat exchanger for grey water heat recovery. Since the Archetype Sustainable Houses are research and demonstration facility, there is no real family living in them. During the controlled testing period, simulated activities of water draw and appliance/lighting usage were used (Tanha, 2012; Safa, 2012).

Table 2: Mechanical Features of the Archetype Sustainable House (Zhang et al., 2011)

Features	House A	House B	Guest Suite
Solar collector	Flat plate collectors	Evacuated tube collectors	From House-B
PV system	No	4.08 kWp	No
Wind turbine	No	1.8 kWp	No
Heating and cooling	Variable capacity air source heat pump packaged with AHU	Ground source heat pump (GSHP) with horizontal loops, desuperheater and buffer tank	From House B
	Wall mounted mini gas boiler*	Stirling engine micro-cogeneration unit with buffer tank (heating alternative to the GSHP)*	From House B
DHW system	Flat plate collector with one tank system	Evacuated tube collector with preheat tank and TOU based electric backup, GSHP desuperheater on the auxiliary tank, and auxiliary heat from the cogen.	None
Ventilation system	Heat recovery ventilator (HRV)	Energy recovery ventilator (ERV)	HRV with an air heater
Auxiliary water heating	Mini gas boiler	Time-of-Use (TOU) electric	From House B
Infloor heating	Basement only*	All three floors & basement	No
Drain water heat recovery	Yes	Yes	No
Air heater	No	No	Yes

***The infloor basement heating and wall mount natural gas mini boiler in House A and the Stirling engine micro-cogeneration unit in House B were not operating over the duration of this study.**

5 METHODOLOGY

The Archetype Sustainable House was designed and constructed as a laboratory for green building technology testing and research with over 300 calibrated sensors installed to monitor the performance of the electrical/mechanical systems and energy fluxes into and out of the house. A flexible and expandable commercial data acquisition system is adopted to process data received from the various sensors. A user-friendly GUI based commercial software has been programmed to provide real-time monitoring and data processing. Measurements are collected at 5 second intervals and recorded in an SQL database (Zhang et al., 2010; Zhang et al., 2011).

Data for this study were collected under controlled and non-controlled conditions year round except during system maintenance. During the controlled testing period, the two houses were closed off to any visitor/staff with simulated appliance/lighting load and hot water draw. The summer and winter controlled test periods extended from August 23 through September 15, 2010 and from December 24, 2010 to January 12, 2011, respectively. During this period, all uses of the house were controlled and recorded. Hot water was dumped daily in accordance with the IEA Schedule Task 26 model for a typical Canadian family of four people (Tahna et al., 2011; Tahna, 2012 Tahna et al., 2012). Data collected during controlled conditions were used to provide 'clean data' for benchmarking and TRNSYS model calibration. The calibrated model (TRNSYS) was subsequently used to simulate energy performance over the entire heating (October 1 to May 21) and cooling seasons (May 22 to September 30) based on climate normals derived from a 30-year historical record of solar irradiance and temperature.

Under non-controlled conditions, occupant activities were not restricted, resulting in energy gains and losses due to meetings, tours and other events in the houses. Since these activities were not recorded, and the effects on energy use were not quantified, the data collected during this period could not be used to benchmark energy use in the house. Nevertheless, data collected during this period provided general performance information that could be used to evaluate overall performance of the equipment, and verification of the model used for results extrapolation.

6 RESULTS

Both systems performed well over the one year study period, exceeding rated performance across all measured indicators. Table 3 compares actual performance during the heating and cooling seasons to the corresponding EnerGuide and equipment manufacturer ratings. Overall seasonal Coefficients of Performance (COP) for both heat pump systems exceeded 3.0 during the heating and 5 during the cooling, indicating that the systems provided over 3 kWh of heating output and 5 kWh of cooling output for each kWh of electricity consumed. The GSHP performed particularly well during the cooling season when performance well exceeded both the manufacturer and EnerGuide ratings for the technology. During the heating season, the measured seasonal COP for the GSHP was only slightly higher than the manufacturer and EnerGuide ratings. Although the ASHP had marginally lower seasonal COP and energy efficiency ratios compared to the GSHP, it exceeded the EnerGuide rating during the heating season by a much greater margin. Performance of the two technologies relative to manufacturer ratings was also significantly higher during both seasons (Table 3).

Table 3: ASHP and GSHP Performance Relative to Ratings

	Manufacturer	EnerGuide	Test	Season
Air Source Heat Pump				
Seasonal Energy Efficiency Ratio	**16**	**>=14**	**18**	**Cooling**
Seasonal Coefficient of Performance	**2.75**	**2.05**	**3.23**	**Heating**
Ground Source Heat Pump				
Seasonal Energy Efficiency Ratio	**12.9**	**>=14.1**	**19.7**	**Cooling**
Seasonal Coefficient of Performance	**3.00**	**>=3.30**	**3.44**	**Heating**

Electricity consumption by the heating and cooling distribution systems in the houses strongly influenced overall power consumption and performance of the HVAC systems. When analyzed as a standalone technology, without considering power consumption by the hydronics (circulation pumps for the buffer tank and infloor heating loops for House B) and forced air distribution systems (air handling unit fan for heating and cooling in House A and for cooling in House B) specific to each house, the heat pumps consumed roughly the same amount of electricity, while delivering similar heating and cooling outputs (Tables 4 and 5). However, when the entire system as installed is considered, including all related pumps and fans, the GSHP underperforms the ASHP by 29% during the cooling season, and outperforms it by 32% during the heating season. These differences are largely due to the electricity consumed by the associated heat pump distribution systems and how they were set up to operate. The air handling unit (AHU) in House A has a variable speed fan that ran continuously during the heating season, but was set up to operate only when the ASHP compressor was on during the cooling season. By contrast, the GSHP circulates water through the radiant floor during the heating season with a pump that consumes only 10% of the energy of the AHU fan in House A. During the cooling season, cool air is provided through an AHU system and a pump that circulates water from the buffer tank to the AHU. The House B distribution system consumed a similar amount of electricity during the heating and cooling season (519 kWh due to in-floor heating loop circulation pump and 482 kWh due to AHU fan, respectively). By comparison, the AHU fan integrated with the ASHP in House A consumed roughly 13 times more electricity during the heating season (2,753 kWh), when it was set operating continuously, than in the cooling season (219 kWh), when it operated only when required. It should be noted that one of the objectives of the current study is to evaluate the performance of the two heat pump systems as installed by the contractors/vendors and find potential issues and solutions for improvement. The continuous running of AHU fan in House A is setup because of the integration of the HRV ventilation with the heating/cooling distribution system. In House B, no fan power is used in the AHU to distribute the ventilation air from the ERV resulting in poorly distributed fresh outdoor air within the house.

Table 4: ASHP and GSHP System Performance During the Heating Season

Performance Metric	ASHP		GSHP	
	Stand-alone	System as installed	Standalone	System as installed
Seasonal power consumption	5442 kWh	8195 kWh	5460 kWh	5979 kWh
Seasonal heating output	17579 kWh		18764 kWh	
Seasonal Coefficient of Performance (COP)	3.23	2.14	3.44	3.14

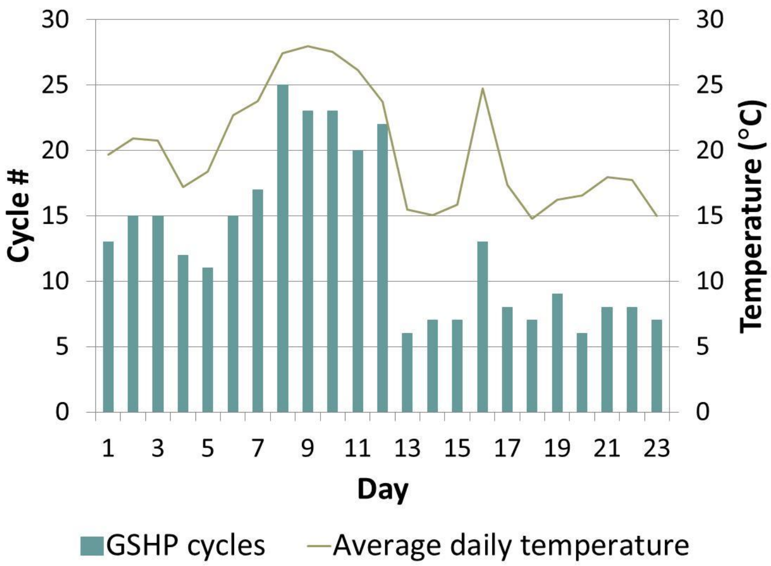
Table 5: ASHP and GSHP System Performance During the Cooling Season

Performance Metric	ASHP		GSHP	
	Stand-alone	System as installed	Standalone	System as installed
Seasonal power consumption	434 kWh	653 kWh	425 kWh	907 kWh
Seasonal cooling output	2289 kWh		2459 kWh	
Seasonal Energy Efficiency Ratio (SEER)	18	12	20	9
Seasonal Coefficient	5.37	3.50	5.78	2.71

of Performance (COP)

Modelled optimization scenarios showed substantial energy savings could be achieved by configuring the systems to operate more efficiently. When the AHU for the ASHP was modelled to operate during the heating season only when the compressor was on, rather than running continuously, electricity consumption fell by almost 37% and the COP increased from 2.25 to 3.54. Similarly, if the circulation pumps from the GSHP to the buffer tank and from the buffer tank to the AHU were optimized by operating them only when the compressor was on, resulting in a 28% reduction in electricity use and a dramatic increase in the as-installed COP from 2.64 to 3.68.[3] This highlights the importance of understanding the various components that make up the system, including HEPA filters and heat recovery ventilators, and ensuring these are optimally configured to maintain high levels of efficiency and energy performance.

The variable capacity ASHP compressor cycled on and off much less frequently than the single stage GSHP compressor resulting in improved overall efficiency**.** Lower on-off cycling of heat pumps improves performance by making the operation of heat pumps more efficient. The ASHP system achieved lower cycling through the variable capacity feature which allowed the compressor to operate primarily on part load, with higher efficiency than full-load, drawing less than half the electricity than would otherwise have occurred. During the cooling season test period the compressor operated between 3 and 11 hours per day and turned on and off only once. The single stage GSHP system operated during the cooling season test period between 1 and 6.5 hours and cycled on and off up to 25 times a day (Figure 3). This is an indication of an oversized system capable of operating only at a constant output. Equipment reliability and thermal comfort are adversely affected by frequent compressor cycling. In fact, the compressor of the GSHP failed twice during a three years of operation.



[3] Note that the optimization scenarios were simulated based on an extrapolation method that differed from the more detailed modeled results presented earlier in this section. Thus the COPs are similar but not identical.

Figure 3: The Number of Daily On-off Cycles for the GSHP in Relation to Outdoor Temperatures

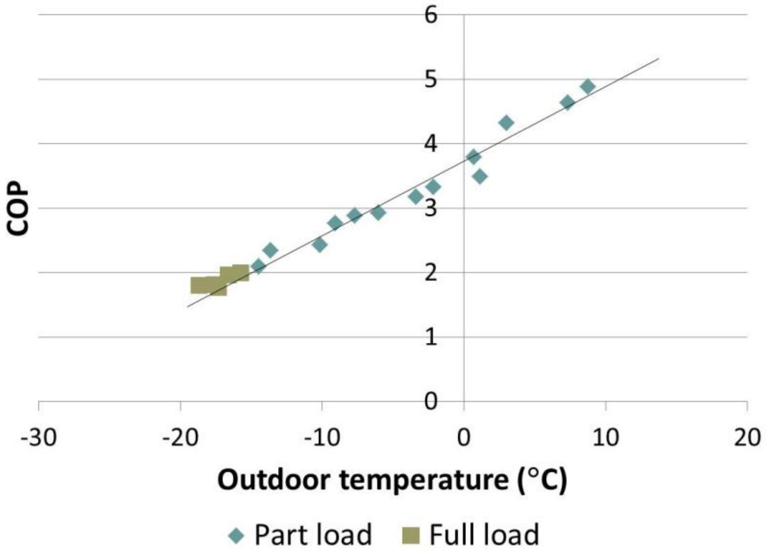
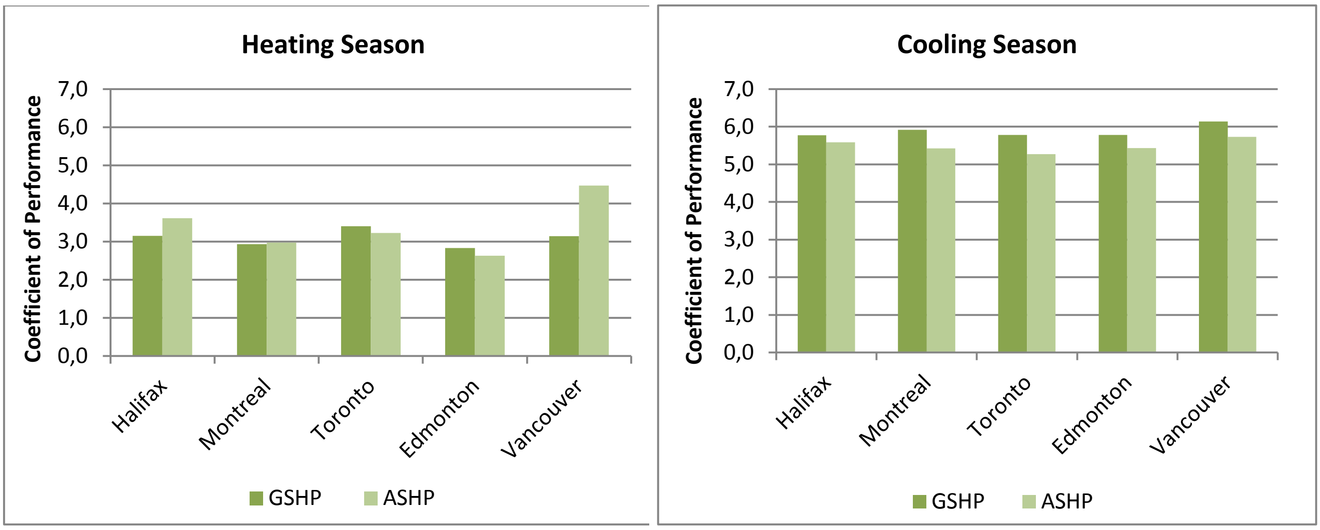


Figure 4: Coefficient of Performance (COP) for the Variable Capacity Air Source Heat Pump at Varying Outdoor Temperatures During the Winter Test Period (Dec. 1 to Feb. 9, 2011)

Efficiency of the ASHP declined more than the GSHP as temperatures declined in the winter, but even at temperatures as low as minus 24°C, the ASHP continued to maintain indoor thermal comfort at desired levels without supplementary heat. The efficiency of GSHP systems is more constant than ASHP systems during the heating season because the temperature of the ground from which heat is drawn is more constant than that of air. Figure 4 shows a decline in the COP of the ASHP from 4.9 to 1.6 as outdoor temperatures fell from 9 to -19 ˚C during the winter. Below -24 ˚C, the heat pump is less efficient than a conventional electric heating system, and a supplementary heat source would be required. At temperatures above roughly minus 15˚C, the variable capacity ASHP compressor operates on part load, drawing less than half the electricity (2 – 3 kW) required under full load conditions (6 kW). Only when outdoor temperatures dropped below -15˚C did the ASHP compressor operate at full speed (Figure 4). By comparison, the GSHP system maintained a more constant COP of about 3.0 regardless of outside air temperature conditions.

Model simulations for five Canadian representative cities (Halifax, Montreal, Toronto, Edmonton, and Vancouver) revealed the two technologies to function well under varying climates with comparable levels of performance except for Vancouver. For Vancouver, the estimated ASHP performance was higher than that of the GSHP. This was mainly due to the milder winter outdoor temperature that benefitted the ASHP system. The simulations were based on historical weather and ground temperature data from the selected cities. Figure 5 shows temperatures, degree days and coefficients of performance for each of the cities during the heating and cooling seasons. The ASHP displayed a wider range of COPs across the various cities during the winter because ASHP performance is more strongly influenced by differences in air temperature. This is evident from the lower COPs in Montreal and Edmonton, where a supplementary heat source would be required at temperatures below -24 °C. In Vancouver, the warmer and more even year round temperatures resulted in higher ASHP performance. During the cooling season, the GSHP system slightly outperformed the ASHP in all cities, with COPs ranging narrowly from 5.8 to 6.1.



Location	Heating Degree Days (°C-days)	Cooling Degree Days (°C-days)	Maximum Temperature (°C)	Minimum Temperature (°C)
Halifax	4297	710	28.1	-19.8
Montreal	4460	1120	32.2	-24.7
Toronto	4122	1114	33.9	-22.2
Edmonton	5514	812	29.4	-30.6
Vancouver	3034	785	26.3	-5.7

Figure 5: Heating and Cooling Degree Days, Temperatures and Modeled Coefficients of Performance for ASHP and GSHP Systems in Selected Canadian Cities

A simple cost analysis showed the ASHP to be more affordable than the GSHP based on performance and initial capital costs. In this analysis, the costs of both heat pump systems were compared to a conventional electric heater during the heating season, and a central air conditioning system during the cooling season. The electricity consumed by the forced air and hydronic distribution systems was ignored because this was assumed to be similar on both the conventional and more sustainable systems. At a Toronto retail electricity rate of 11.3 cents per kWh (including transmission, distribution, debt retirement, and regulation), the annual cost of energy for the ASHP was $664 while the conventional system (with electric resistance heating and centre air-conditioning system) energy cost was $2,074. At an initial equipment cost of $14,500 the simple payback would be 10.3 years. This does not include the cost of a supplementary heating system, which would be required when temperatures fall below -24 °C which is rare in Halifax, Toronto, and Vancouver.

The annual cost of energy for the GSHP was $725 while the conventional system energy cost was $2205. Accounting for an equipment and installation cost of $34,500 the simple payback was 23.3 years. Thus, although the GSHP is slightly more efficient, the higher GSHP equipment and installation costs make the ASHP a less expensive option. If life cycle costs and benefits were considered, this price gap would narrow because the ground loop is a one-time cost and the GSHP compressor is expected to have fewer mechanical and thermal stresses with a longer expected service life (20 to 25 years). If the two systems were compared to a natural gas furnace and air conditioner, the paybacks would have been considerably longer because current natural gas costs are less than one fifth the cost of electricity per unit of thermal energy provided.

Energy savings from the use of these more efficient heat pump systems translated into significant reductions in greenhouse gas emissions relative to conventional alternatives. Annual electricity savings relative to a conventional electric furnace and air conditioner were converted to the equivalent carbon dioxide based on electricity generation sources in Ontario to arrive at emission reductions of 2,330 and 2,449 kg eCO_2 for the ASHP and GSHP, respectively. If instead, the heat pump displaced natural gas during the heating season, the annual emissions reductions would rise to 3329 and 3549 kg eCO_2. As the electrical grid in Ontario continues to decarbonize, future emissions reductions associated with heat pump systems will also continue to rise.

7 CONCLUSIONS

In this study, two heat pumps were evaluated both as stand-alone technologies and as part of the overall systems installed in the houses using a common experimental data set. As stand-alone technologies, the systems performed exceptionally well, showing heating and cooling efficiencies above both EnerGuide and manufacturer rated performance. Both heat pump systems had COPs above 3 during the heating season and above 5 during the cooling season. Adding the energy inputs associated with the heating and cooling distribution systems installed in each house lowered overall performance by between 9 and 53%. Optimization scenarios showed that these systems could be set-up to function between 28 and 37% more efficiently by operating the fans and pumps only when the heat pump compressor is on. Upgrading the GSHP to a variable capacity system would further enhance performance by reducing cycling and increasing operating times on the more efficient part load setting.

The capital cost of the heat pump systems relative to conventional heating and cooling systems continues to be a barrier to wider adoption of these technologies. The ASHP system was shown to be a cost effective alternative to conventional electric furnace and air conditioner but the more expensive GSHP was less affordable with a simple pay back of over 20 years. A full life cycle cost assessment of the optimized technologies would be required to provide a more accurate assessment of the affordability of the two systems relative to one another and to conventional alternatives.

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