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# The Potential of Heat Pump Technologies for the Replacement of Rooftop Units in Small Commercial Buildings

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***Abstract:*** *This paper presents an energy and lifecycle cost analysis of the impact of replacing conventional rooftop units with electrically driven heat pumps for small office buildings in Canada. First, an energy model of a small office is created in EnergyPLUS for five Canadian regions. This model serves as the baseline for three heat pump integrations, including (i) conventional air-source, (ii) cold climate air-source, and (iii) ground-source systems. Each integration is modelled in TRNSYS, with special focus on implementing a methodology to assess both full and part load performance. Annual simulations demonstrate strong heating/cooling energy savings for heat pumps in all regions. However, low natural gas rates present a challenging economic context for heat pumps, with lifecycle costs consistently higher than conventional rooftop units. A sensitivity analysis is used to assess the impact of a carbon tax, natural gas escalation rates, and the potential of hybrid natural gas/heat pump systems.*

***Keywords:*** *Heat pump, TRNSYS, cold-climate, lifecycle analysis*

## INTRODUCTION

Commercial buildings in Canada account for nearly 10% of national secondary energy use, with 60% of this total directed towards space heating and cooling (NRCan, 2017). Within this sector, office buildings account for approximately 20% of total floor space and energy use, and thus represent a prime opportunity for strong energy use reductions (NRCan, 2012). A large portion of small commercial buildings use packaged rooftop units (RTU) to provide space heating and cooling. Information from the US Department of Energy (DOE) estimates that RTUs serve over 60% of commercial floor space in the US (Wang *et al.*, 2011), with a similar percentage likely in Canada. While RTUs offer a simple and low initial cost solution to building heating and cooling, they primarily rely on natural gas or electric resistance elements for space heating, with little opportunity to use renewable energy sources.

Heat pump systems offer an attractive alternative to traditional RTU systems, delivering space heating and cooling efficiencies of nearly 300%. Their integration is also complimentary to the increased electrification of building systems brought about by a higher fraction of onsite renewable electricity generation, and the push towards Net Zero Energy Buildings. However, the addition of heat pumps to a building is often a complex task, and involves examining both the indoor and outdoor environments, building envelope, and mechanical systems. From an economic perspective, heat pump integration or conversion can be more difficult to justify given that the price of natural gas can be up to six times less expensive per unit of energy then electricity in some Canadian regions.

Previous studies have examined the energy performance of RTU units, and the viability of heat pump systems in commercial buildings. Both Studer *et al.* (2012) and Wang and Katipamula (2013) presented the potential of retrofitting existing RTUs with variable speed fan controls and higher efficiency units. The authors noted the significant potential impacts of widespread improvements to RTU systems, and the need for region specific results to provide building owners with the supporting data required to make retrofit decisions. Both papers were primarily focussed on cooling dominated regions in the United States, and did not examine the impact of reducing heating energy use via heat pumps. Tamasauskas *et al.* (2014) and Kegel *et al.* (2014) examined the role of heat pumps integrated into Canadian buildings. However, these studies were limited to larger commercial buildings with centralized HVAC systems, and did not address RTU systems. If heat pump systems are to have a significant impact in the RTU market in Canada, a comprehensive set of region-specific energy and economic performance data is required to better support building owners and policy makers.


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*Table 1: Climate summary* (CCBFC 2010, CCBFC 2011)

| | Vancouver British Columbia | Toronto Ontario | Halifax Nova Scotia | Montreal Quebec | Whitehorse Yukon |
|---|---|---|---|---|---|
| NECB Climate Zone | Zone 4 | Zone 5 | Zone 6 | Zone 6 | Zone 7B |
| Heating Degree Days (18°C) | 2950 | 3520 | 4000 | 4200 | 6850 |
| Heat $T_{Design}$ (°C,DB) | -7 | -18 | -16 | -23 | -41 |
| Cool $T_{Design}$ (°C, DB/WB) | 28/20 | 31/23 | 26/20 | 30/23 | 25/15 |

The objective of this paper is to examine the energy and cost saving potential of replacing fuel-fired RTUs with various heat pump technologies for small office buildings in Canada. First, an energy model of the building is created in EnergyPLUS for five Canadian climate zones. This model is then used as a base for detailed TRNSYS simulations of several heat pump systems, including (i) conventional air-source, (ii) cold-climate air-source, and (iii) ground-source heat pumps. Results are examined from an energy and lifecycle cost perspective to identify the potential of each integration.

## DEVELOPMENT OF BASE CASE MODELS

A set of region-specific small office models has been developed using EnergyPLUS. These models form part of a larger project examining the impact of energy efficient technologies in commercial/institutional buildings. Five regions are selected in this study in order to present the results in a concise manner while demonstrating the impact of different climates and utility rate structures. Climate parameters are summarized in Table 1.

Building geometry is based on the US Department of Energy (DOE) archetypes. Envelope parameters, mechanical system performance, operating and occupancy schedules are derived from the National Energy Code of Canada (NECB) 2011 (CCBFC, 2011). Important building and system parameters are summarized in Table 2. Mechanical systems are modelled as a one-zone system, in accordance with the base case in the NECB 2011.

*Table 2: Parameters for small office*

| Parameter | Value/Description |
|---|---|
| Total Floors | 1 |
| Total Floor Area | 511 $m^2$ |
| Aspect Ratio | 1.50 |
| Glazing Percent | 21% |
| Heating System | RTU, Fuel fired, 80% AFUE |
| Cooling System | RTU, DX cooling, COP=3.9* |
| DHW | Fuel fired, EF=0.62 |

** COP @ 35°C outdoor, 26.7/19.4°C indoor*

## HEAT PUMP SYSTEM INTEGRATIONS

The developed base case models serve as a reference for three separate heat pump integrations:

I. Conventional air-source heat pump (ASHP)
II. Cold-climate air-source heat pump (CCHP)
III. Ground-source heat pump (GSHP)

### Conventional Air-Source Heat Pump

This integration examines the performance of a conventional air-source heat pump, which uses the ambient air as a thermal source/sink. The proposed heat pump system replaces the conventional RTU, with the heat pump cooling capacity selected using the peak cooling load of the building. As conventional air-source heat pumps lose a significant portion of heating capacity at colder outdoor temperatures, a fully sized auxiliary system is also integrated. This auxiliary system consists of electric baseboards, and is activated if the heat pump is unable to meet the heating setpoint within a specified time period.

Key parameters for the ASHP are derived from manufacturer data (York, 2017a) and are shown in Table 3. The ASHP modelled uses a single speed compressor. Performance as a function of outdoor temperature is modelled using curves from the NECB (CCBFC, 2011).

*Table 3: ASHP system parameters*

| | Unit Size (Cooling) | |
|---|---|---|
| | <7.5 ton | 7.5-10 Tons |
| Heating COP | 3.4* | 3.35* |
| Heating Auxiliary | Electric BB | Electric BB |
| Cooling COP | 4.0** | 4.1** |
| Cooling Stages | 1 | 2 |

** @8.3°C outdoor, 21.1°C indoor*
*** @ 35°C outdoor, 26.7/19.4°C indoor*


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## Cold Climate Air-Source Heat Pump

The cold-climate heat pump offers improved low temperature heating capacities and an extended operating range. Figure 1 compares the heating capacity and (full load) power draw for the ASHP and CCHP units used in this study, normalized per rated heating conditions: outdoor temperature 8.3°C, indoor temperature of 21.1°C.

The CCHP unit is able to maintain nearly 75% of its rated capacity at an ambient temperature of -25°C, while the ASHP exhibits a strong decline in available capacity at lower ambient temperatures. Interestingly, the improved low temperature heating capacity of the CCHP also brings about a significant increase in full load power draw at low temperatures. Selection of these units thus requires careful analysis and sizing to ensure that CCHP integrations result in an appropriate reduction in system energy use.

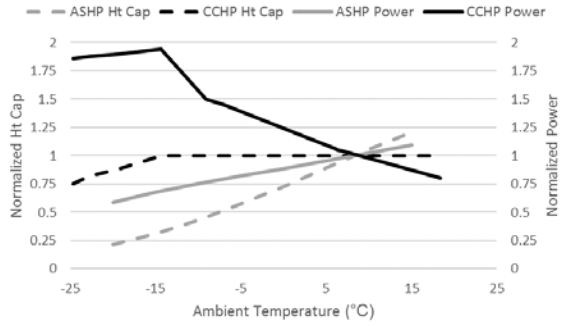


*Figure 1: Comparison of ASHP and CCHP performance*

Unlike the ASHP, no standard CCHP rooftop integrations exist in the North American market. The proposed system thus requires a custom design utilizing a field supplied DX coil along with a manufacturer supplied connection kit to interface with the compressor section of the CCHP unit. While technically feasible, this integration is also associated with added costs, as shown later in this paper.

The selected CCHP unit has a rated heating COP of 3.6 (indoor 21.1°C, outdoor 8.3°C), a rated cooling COP of 3.4 (indoor 26.7°C, outdoor 35°C) (Mitsubishi, 2017), and uses a variable speed compressor. Variation of performance with indoor and outdoor temperatures follows data from the manufacturer. Each unit is sized for the larger of either (i) the peak cooling load, or (ii) approximately 60% of the peak heating demand. Avoiding oversizing is a key focus of CCHP system design: Proper sizing is vital in order to ensure higher average equipment loads, and therefore, improved operating efficiencies. Due to the operating range and performance of the CCHP, electrical auxiliary system sizes (modelled as duct heaters) can be reduced for all regions with the exception of Whitehorse (due to its extreme cold climate).

## Ground-Source Heat Pump

Vertical ground-source systems make use of relatively stable deep earth temperatures as a thermal source/sink for the heat pump. Proper sizing of the borehole is critical in order to optimize the energy performance of these systems. In this study, borehole sizing was performed using the peak hourly, monthly, and annual ground loads, following the methodology of Philippe *et al.* (2010). Soil types were derived from Hanova *et al.* (2007), with soil properties obtained via ASHRAE (2017). A summary of borehole sizing in each region is provided in Table 4. Each borehole is 0.150 m in diameter, and uses HDPE piping in a single U tube configuration. Borehole spacing was set at 6.1m. For all GSHP cases, a 25% propylene glycol fluid was circulated in the boreholes.

*Table 4: Borehole sizing*

| | Boreholes | Depth (m) |
|---|---|---|
| Halifax | 4 | 107.9 |
| Montreal | 4 | 60.0 |
| Toronto | 4 | 82.6 |
| Vancouver | 2 | 83.8 |
| Whitehorse | 20 | 92.1 |

Each ground source heat pump has a rated heating COP of 3.6 and a rated cooling COP of 4.75 at ISO 13256 ground loop conditions (ClimateMaster, 2017), and employs a variable speed compressor. The selected heat pumps include low temperature heating capabilities, and are capable of operating down to an entering fluid temperature of -6.7°C in heating, and -1.1°C in cooling (limits dependant on operating flow rate). Heating performance, normalized based on rated capacity and power draw at a 0°C entering water temperature and 21°C entering air temperature, is provided below in Figure 2.

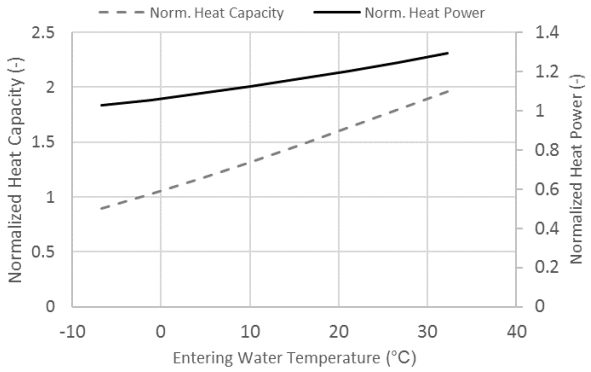


*Figure 2: GSHP performance in heating mode*


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## LIFECYCLE COST ANALYSIS

In addition to the energy performance of the system, lifecycle costs are critical to determining the potential market impact of different heating and cooling technologies. This section presents key information regarding the lifecycle cost analysis.

### Main Lifecycle Parameters

Each integration is assessed using a lifecycle of 15 years, which is typical of packaged RTU systems (ASHRAE, 2015). Lifecycle calculations are performed using an inflation rate of 1.5% and a discount rate of 4.0%, based on current financial markets in Canada.

### Capital and Maintenance Costs

Capital costs were assessed at the component level, and included material and installation for all required equipment, controls, and piping/ducting. These costs were primarily derived from available suppliers (Budget Heating & Cooling, 2017) and RS Means (2017). Both material and labour costs were adjusted for each region using location factors from RS Means. Total capital costs are provided in Table 5, while a summary of included costs is shown in Table 6.

Maintenance costs were derived from Bloomquist *et al.* (2004), and assumed equal for all systems given the nature of integration.

*Table 5: Capital cost summary*

| | Base | ASHP | CCHP | GSHP |
|---|---|---|---|---|
| Halifax | $30,912 | $38,318 | $52,809 | $67,207 |
| Montreal | $37,977 | $46,274 | $59,680 | $72,658 |
| Toronto | $42,276 | $49,758 | $64,453 | $88,648 |
| Vancouver | $31,999 | $37,654 | $54,353 | $51,320 |
| Whitehorse | $32,007 | $42,833 | $74,394 | $292,715 |

*Table 6: Components for lifecycle cost analysis*

| Category | Components |
|---|---|
| Heat/Cool | RTU/ASHP/GSHP<br>CCHP (Condenser/Coil/Connection Kit) |
| Distribution | Gas Piping (RTU)<br>Ducting |
| Control | Thermostat<br>CCHP Control<br>Pump Control |
| Boreholes | Boreholes (Drilling & Piping)<br>Glycol<br>Pump |
| Accessories | Roof Curb<br>Economizer<br>LP Kit (Whitehorse) |

### Utility Costs

Current utility rate structures were included for all regions for both natural gas (propane for Whitehorse) and electricity. These rates included all current daily/monthly charges, tier/time of use structures, riders, and demand charges, based on local utility rate structures (Nova Scotia Hydro 2017, Heritage Gas 2017, Hydro Quebec 2017, Energir 2017, Toronto Hydro 2017, Enbridge 2017, BC Hydro 2017, Fortis 2017, Arctic Energy Alliance 2017).

A brief summary of these rates is provided in Table 7. Per unit energy costs are provided for illustration only, and are based on an average monthly electricity consumption of 4,000 kWh, and natural gas consumption of 50 $m^3$. Realistic tiered/time of use structures are applied in the simulations and lifecycle analysis.

Escalation rates are derived from the National Energy Board (NEB, 2016), and are summarized in Table 8. Natural gas rates in Halifax fluctuate greatly year to year. As such, the escalation rate for this region is based on the Canadian average for natural gas.

*Table 7: Utility rate structures*

| | Halifax | Montreal | Toronto | Vancouver | Whitehorse |
|---|---|---|---|---|---|
| Daily Fixed Elec. ($/day) | $0.00 | $0.41 | $1.15 | $0.33 | $0.00 |
| Electricity Rate ($/kWh) | $0.09 | $0.10 | $0.15 | $0.11 | $0.13 |
| Electricity Demand ($/kW) | $10.50 | $0.00 | $0.00 | $0.00 | $7.39 |
| Daily Fixed NG ($/day) | $0.72 | $1.80 | $2.30 | $0.82 | $0.00 |
| NG Rate ($/GJ) | $17.58 | $16.07 | $7.03 | $5.82 | $41.52 |

*Table 8: Utility escalation rates for selected regions*

| | Halifax | Montreal | Toronto | Vancouver | Whitehorse |
|---|---|---|---|---|---|
| Electricity | 0.10% | 0.10% | 0.60% | 0.30% | 3.00% |
| Natural Gas | 0.40% | 0.90% | 0.70% | 0.20% | 3.00% |


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# SIMULATION METHODOLOGY

This section presents the simulation methodology used for this study. Special attention is given to appropriately modelling the part load performance of each system.

## Building Model Simulations

Building heating and cooling loads were obtained using EnergyPLUS and the appropriate CWEC weather file at a time step of 1 hr. Calculated building heating and cooling loads (sensible and latent) were then imported into TRNSYS via a text reader, and combined with TRNSYS Type 690 to simulate building thermal behaviour.

## Mechanical System Simulations

All system simulations were performed in TRNSY because of the flexibility offered in modelling various HVAC systems. Each integration was simulated on an annual basis using a time step of approximately 5 minutes in order to appropriately model control decisions and avoid simulation convergence errors.

One key focus during system model development was appropriately modelling the part load behaviour of all major mechanical equipment (fuel-fired heating, DX cooling, heat pumps, fans, pumps). The ability to appropriately assess this behaviour is not native to many TRNSYS components. As such, a special procedure was derived based on the work of Kegel *et al.* (2014), which consisted of four main steps:

i. Using a PID controller (Type 23) to estimate the required heating/cooling energy over a single time step. (Fan and pump part loads are based on the heating/cool equipment served)

ii. Normalizing the control signal from the PID to a value between 0, 1. The normalized value estimates the part load ratio (PLR).

iii. Applying the product of equipment capacity and PLR to the airstream using Type 693.

iv. Calculating the energy use at part load using PLR and appropriate part load curve.

The procedure is shown visually in Figure 3.

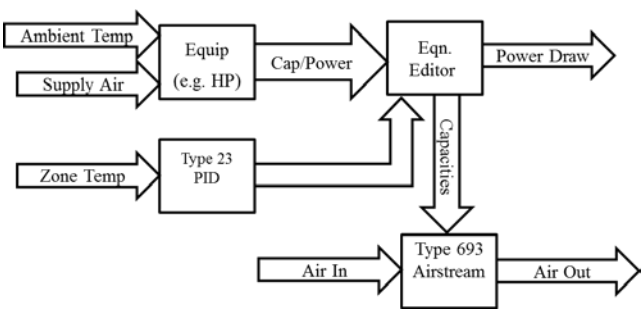


*Figure 3: PLR simulation methodology (Kegel et al, 2014)*

A summary of TRNSYS components, performance and PLR data, is provided in Table 9.

*Table 9: Mechanical system simulation parameters*

| Equip. | Type | Performance | PLR |
|---|---|---|---|
| Burner | 929 | York 2017b | NECB |
| DX Cool | 964 | York 2017a | NECB |
| ASHP | 966 | NECB | NECB |
| CCHP | 1024 | MSB. 2017 | MSB. 2017 |
| GSHP | 919 | CM 2017 | Hong & Liu 2010 |

The part load performance of each heat pump system in heating mode is also provided below in Figure 4. The part load factor (PLF) represents a ratio of energy use at current part load to energy use at full load conditions.

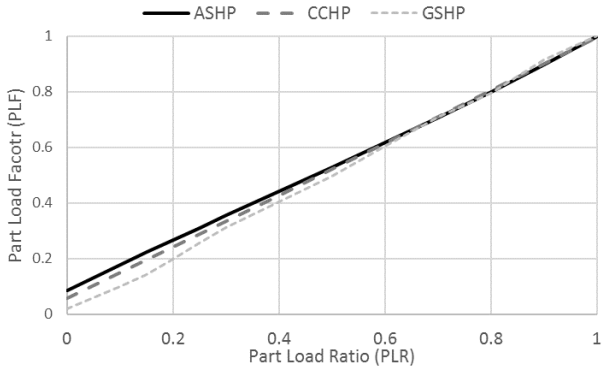


*Figure 4: Part load performance for HP systems*

## Heat Pump Modelling and Control

The thermal output of the heat pump is controlled using the PLR value calculated using the procedure above. At each time step, the boundary conditions for the heat pump (outdoor air temperature (ASHP, CCHP), entering water temperature (GSHP), and return air temperature(All systems)) are also checked to ensure they are within the limits defined by each equipment manufacturer.

All heat pumps are modelled using a data-driven approach, either following a performance curve from the NECB (ASHP), or using a lookup table of performance from the manufacturer (CCHP and GSHP). This approach neglects some transient details of HP operations, including the defrost cycle. While it is possible to model defrost outside the TRNSY HP models, defrost strategies are often proprietary to the heat pump in question, and are rarely provided in detail by manufacturers. As such, they have been neglected for this initial study. A report prepared for the US DOE shows that neglecting defrost will have a limited impact on seasonal COPs in cold climates (Williamson and Aldrich, 2015). Future work will study the impact of defrost cycles in greater depth.


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# RESULTS

Results are presented from both the energy and lifecycle cost perspectives. A brief discussion section also compares results with those from previous studies.

## Energy Performance

Annual energy performance for each system is provided in Table 10 for Halifax and Vancouver. Both cities represent maritime climates in Canada, with more moderate winters and cooler summers.

Each heat pump offers a clear reduction in the energy use for space heating and cooling, with these savings ranging from 29% to 41%. In both cities, the GSHP results in the most significant energy use reductions, primarily because of warmer and more stable heat pump source temperatures during the winter months. Interestingly, the overall energy performance of the ASHP and CCHP are very similar in both cities. Although the CCHP is able to offer modest reductions in heating energy use, these savings are offset by higher energy use in cooling mode. This primarily relates to the lower cooling COP of the CCHP units and their higher energy use at lower part loads.

Table 11 presents the energy performance of each heat pump system in Montreal and Toronto. Both cities have relatively cold winter design conditions, while also having significant total and latent cooling loads in the summer months.

As before, all heat pump systems offer strong energy savings, with the GSHP system representing the most efficient option. CCHP performance is notable in Montreal, offering over a 1,000 kWh reduction in energy use for heating. However, these savings are diminished by the reduced performance of these units in cooling mode. In a building with high internal loadings (such as the office in this study), the CCHP has less of an opportunity to operate in heating mode, and is unable to overcome its lower cooling efficiency on an annual basis.

*Table 10: Energy performance in Halifax and Vancouver*

| | **Halifax** | | | | **Vancouver** | | | |
|---|---|---|---|---|---|---|---|---|
| | Base Case | ASHP | CCHP | GSHP | Base Case | ASHP | CCHP | GSHP |
| Heating (kWh) | 13,433 | 6,115 | 4,963 | 3,488 | 7,146 | 2,734 | 2,191 | 1,744 |
| Cooling (kWh) | 1,481 | 1,376 | 1,722 | 1,504 | 1,462 | 1,348 | 1,887 | 1,529 |
| DHW (kWh) | 10,383 | 10,383 | 10,383 | 10,383 | 9,876 | 9,876 | 9,876 | 9,876 |
| Fans/Pumps (kWh) | 6,080 | 6,871 | 7,081 | 7,321 | 5,888 | 5,930 | 6,215 | 6,270 |
| Light/Equip (kWh) | 35,614 | 35,614 | 35,614 | 35,614 | 35,623 | 35,623 | 35,623 | 35,623 |
| Total (kWh) | 66,992 | 60,359 | 59,763 | 58,311 | 59,995 | 55,511 | 55,791 | 55,042 |
| Total Savings (kWh) | -- | 6,633 | 7,228 | 8,680 | -- | 4,484 | 4,204 | 4,953 |
| Savings H/C (%) | -- | 32% | 34% | 41% | -- | 31% | 29% | 34% |

*Table 11: Energy performance in Montreal and Toronto*

| | **Montreal** | | | | **Toronto** | | | |
|---|---|---|---|---|---|---|---|---|
| | Base Case | ASHP | CCHP | GSHP | Base Case | ASHP | CCHP | GSHP |
| Heating (kWh) | 17,438 | 9,306 | 8,279 | 4,757 | 16,385 | 6,983 | 6,282 | 4,322 |
| Cooling (kWh) | 2,894 | 2,754 | 3,769 | 3,443 | 2,608 | 2,511 | 3,508 | 3,042 |
| DHW (kWh) | 10,437 | 10,437 | 10,437 | 10,437 | 10,260 | 10,260 | 10,260 | 10,260 |
| Fans/Pumps (kWh) | 8,329 | 9,199 | 9,473 | 9,914 | 7,611 | 8,528 | 9,101 | 9,144 |
| Light/Equip (kWh) | 35,584 | 35,584 | 35,584 | 35,584 | 35,617 | 35,617 | 35,617 | 35,617 |
| Total (kWh) | 74,683 | 67,281 | 67,542 | 64,136 | 72,481 | 63,898 | 64,768 | 62,384 |
| Total Savings (kWh) | -- | 7,402 | 7,141 | 10,547 | -- | 8,583 | 7,713 | 10,097 |
| Savings H/C (%) | -- | 26% | 25% | 37% | -- | 32% | 29% | 38% |


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*Table 12: Energy performance in Whitehorse*

| | Base Case | ASHP | CCHP | GSHP |
|---|---|---|---|---|
| Heat (kWh) | 32,974 | 22,297 | 17,745 | 5,874 |
| Cool (kWh) | 769 | 729 | 1,101 | 398 |
| DHW (kWh) | 11,653 | 11,653 | 11,653 | 11,653 |
| Fn/Pm (kWh) | 8,609 | 8,719 | 9,148 | 13,787 |
| L/E (kWh) | 35,586 | 35,586 | 35,586 | 35,586 |
| Total (kWh) | 89,591 | 78,983 | 75,232 | 67,297 |
| Total Savings | -- | 10,608 | 14,539 | 22,293 |
| Savings H/C | -- | 25% | 34% | 53% |

Table 12 summarizes energy performance in Whitehorse. While all heat pump systems offer energy savings versus the base case, the value of the CCHP and GSHP is now far more evident, as these systems are able to deliver consistent heating to the building throughout the winter months. This is particularly true for the GSHP system, where heating operations are able to continue without restriction despite the cold outdoor air temperatures in Whitehorse.

It should be noted that the low cooling energy use for the GSHP in Whitehorse is not driven by improved system efficiencies, but instead by a lower limit on the entering fluid temperature to the heat pump. There is a period of time during the warmer months when entering fluid temperatures from the boreholes are below the specified minimums provided by the manufacturer. In these cases, the heat pump does not operate and no significant energy use is recorded.

Readers may also note the relatively high-energy use for fans and pumps. This is primarily driven by two factors. First, the fan schedule from the NECB requires continuous fan operations from 6AM to 8 PM, Monday to Friday. The units examined in this study are also constant volume units, and as such do not have a lower fan setting when heating or cooling is not required during scheduled hours. Secondly, since in all cases the rooftop equipment is the primary heating/cooling source, fans must also operate as needed during the nighttime and weekends in order to maintain adequate space temperature, further increasing energy use.

## Lifecycle Cost Performance

Lifecycle results are summarized in Table 13 (Halifax and Vancouver), Table 14 (Montreal and Toronto) and Table 15 (Whitehorse). It is clear that in all regions, the proposed heat pump integrations are unable to compete with the economics of the conventional RTU, with no heat pump demonstrating lower lifecycle costs over the 15-year period. Although the energy analysis demonstrated the significant energy savings that the heat pump integrations can have, shifting from natural gas to electricity for space heating consistently results in increased utility costs for the building owner in nearly all cases. In fact, only the GSHP in Montreal and Whitehorse offers some level of lifecycle utility cost savings, although these savings are outweighed by the significant capital investments required. The combination of these differences in utility rates, along with the added complexity of the heat pump integration (e.g., the need for auxiliary systems, more controls) present a challenging set of circumstances that are difficult to overcome given current natural gas prices in Canada.

The impact of the differing ratios between natural gas and electricity rates is evident when comparing utility costs for different regions. Overall increases in lifecycle utility costs are smallest in Montreal, where natural gas and electricity per unit costs are closest. Utility cost increases are greatest in Halifax and Whitehorse, where electricity billing includes an additional portion for the electrical demand. Since demand is clearly higher with electrically based heat pump systems, this further increases the cost of system operations in comparison to the conventional RTU.

## Discussion

It is important to note that the results presented in this study are for a single building type. Although not applicable for the small office, larger buildings with increased electricity demand could result in a shift in utility rate structure (e.g., small to medium commercial rate), which is likely to change the trends described. Additionally, different building types, especially those with lower internal gains and higher ventilation requirements, are likely to increase heat pump loading in the heating season, and lead to further energy savings (and, potentially, utility cost savings, depending on the region) versus the base case.

While not examined in this study, switching space heating from natural gas to electricity also has the potential to reduce greenhouse gas emissions in regions where the electricity grid is comprised primarily of low-emission or renewable energy source (e.g., the grid in Québec).

The results presented in this paper are significantly different from those shown for a large office in Tamasauskas *et al.* (2014). The main reason for this discrepancy can be attributed to the method of integration for the heat pump systems. In this study, each heat pump is integrated as a single zone rooftop system in order to maintain similarity with the base case and offer a like-for-like alternative for the building owner. As such, each heat pump integration is be sized for the full envelope and ventilation loads at design conditions, which often results in the units operating at lower part loads (and thus, less efficiently) for the majority of the year.


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In the Tamasauskas *et al.* study of heat pumps for a large office building, each heat pump system was integrated as a ductless system, with indoor units in each thermal zone. The indoor units of the heat pump were sized solely for the envelope loads of the building, which for an office are more likely to be driven by internal loading and occupancy. As such, these units are better matched to the loading of the zone, and are able to operate at higher part loads and thus more efficiently. This type of integration also results in better comfort conditions for occupants, as heating/cooling is tailored to the individual zone rather than being based on a single thermostat for the whole building. Furthermore, by separating heating/cooling and ventilation, significant energy savings were also obtained from lower fan energy use. Future work will examine the sensitivity of the results in this study to changes in CCHP sizing.

*Table 13: Lifecycle costs in Halifax and Vancouver*

| | **Halifax** | | | | **Vancouver** | | | |
|---|---|---|---|---|---|---|---|---|
| | Base Case | ASHP | CCHP | GSHP | Base Case | ASHP | CCHP | GSHP |
| Capital ($) | $30,912 | $38,318 | $52,809 | $67,207 | $31,999 | $37,654 | $54,353 | $51,320 |
| LCC Elec. ($) | $86,020 | $122,049 | $106,323 | $100,704 | $66,829 | $70,870 | $71,294 | $70,157 |
| LCC NG ($) | $22,626 | $11,758 | $11,758 | $11,758 | $8,248 | $6,361 | $6,361 | $6,361 |
| LCC Maint. ($) | $3,356 | $3,356 | $3,356 | $3,356 | $3,356 | $3,356 | $3,356 | $3,356 |
| LCC Tot. ($) | $142,914 | $175,481 | $174,247 | $183,026 | $110,433 | $118,241 | $135,365 | $131,194 |
| LCC Utility Save ($) | -- | -$25,162 | -$9,436 | -$3,817 | -- | -$2,154 | -$2,579 | -$1,441 |
| LCC Tot Save ($) | -- | NA | NA | NA | -- | NA | NA | NA |

*Table 14: Lifecycle costs in Montreal and Toronto*

| | **Montreal** | | | | **Toronto** | | | |
|---|---|---|---|---|---|---|---|---|
| | Base Case | ASHP | CCHP | GSHP | Base Case | ASHP | CCHP | GSHP |
| Capital ($) | $37,977 | $46,274 | $59,680 | $72,658 | $42,276 | $49,758 | $64,453 | $88,648 |
| LCC Elec. ($) | $59,070 | $71,338 | $71,657 | $67,494 | $87,606 | $99,544 | $101,474 | $97,546 |
| LCC NG ($) | $25,945 | $15,178 | $15,178 | $15,178 | $19,993 | $14,457 | $14,457 | $14,457 |
| LCC Maint. ($) | $3,356 | $3,356 | $3,356 | $3,356 | $3,356 | $3,356 | $3,356 | $3,356 |
| LCC Tot. ($) | $126,348 | $136,145 | $149,871 | $158,686 | $153,231 | $167,115 | $183,740 | $204,007 |
| LCC Utility Save ($) | -- | -$1,500 | -$1,819 | $2,344 | -- | -$6,402 | -$8,332 | -$4,404 |
| LCC Tot Save ($) | -- | NA | NA | NA | -- | NA | NA | NA |

*Table 15: Lifecycle costs in Whitehorse*

| | Base | ASHP | CCHP | GSHP |
|---|---|---|---|---|
| Capital ($) | $32,007 | $42,833 | $74,394 | $292,715 |
| LCC Elec. ($) | $117,497 | $223,675 | $214,708 | $157,408 |
| LCC NG ($) | $104,272 | $27,228 | $27,228 | $27,227 |
| LCC Maint. ($) | $3,356 | $3,356 | $3,356 | $3,356 |
| LCC Tot ($) | $257,132 | $297,091 | $319,687 | $480,707 |
| LCC Utility Save ($) | -- | -$29,134 | -$20,168 | $37,133 |
| LCC Tot Save ($) | -- | NA | NA | NA |


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*Table 16: Impact of carbon tax/escalation rates*

| | Toronto | | | | Vancouver | | | |
|---|---|---|---|---|---|---|---|---|
| | Base Case | ASHP | CCHP | GSHP | Base Case | ASHP | CCHP | GSHP |
| LCC Elec. | $87,606 | $99,544 | $101,474 | $97,546 | $66,829 | $70,870 | $71,294 | $70,157 |
| LCC NG | $21,870 | $15,180 | $15,180 | $15,180 | $9,403 | $7,033 | $7,029 | $7,033 |
| LCC Total | $155,109 | $167,838 | $184,459 | $204,730 | $111,588 | $118,913 | $136,032 | $131,866 |
| Esc. Req'd (%) | -- | 8 | 10 | 6 | -- | 7 | 9 | 5 |

## SENSITIVITY ANALYSIS

In order to better assess the sensitivity of the results to future energy pricing trends, results have been re-examined with a carbon tax of approximately $1.50/GJ. The results of this analysis are presented in Table 16, along with the required natural gas escalation rate needed for each system to show *utility cost savings* over the lifecycle of the unit. It is clear from the results that even with a carbon tax, it is still difficult for the heat pump integrations to achieve utility or total cost savings versus the base case. Escalation rates to achieve *utility cost savings* range from 5-6% for GSHP systems to 9-10% for the CCHP, which are substantial in comparison to current natural gas market trends in North America.

As a further examination, a hybrid heat pump/conventional RTU is also studied, which would contain an ASHP along with an auxiliary natural gas burner. Above a certain cut-off temperature, the heat pump would operate to meet the heating demands of the building. Below the cut-off temperature, the heat pump would be turned off and the natural gas burner would operate instead. Assuming that this notional unit would have the same capital cost as a conventional RTU, the hybrid unit would allow the building owner to avoid the cost of installing baseboards, while also shifting auxiliary energy use to less costly natural gas. This hybrid system would also provide the building owner with a measure of protection against fluctuations in both natural gas and electricity prices, and could represent a good entry point for heat pump technologies into the RTU market.

Sample results for the Halifax region are summarized in Table 17. A cut-off temperature of 7°C is selected such that utility costs equal those of the base case, with this value likely to vary depending on the heat pump used, current utility rates, and climate. It can be seen that the proposed system is able to reduce heating and cooling energy use by over 300 kWh while maintaining nearly identical utility and lifecycle costs from the base case.

*Table 17: Performance of hybrid system*

| | Base Case | Hybrid RTU |
|---|---|---|
| Heat/Cool (kWh) | 14,914 | 14,584 |
| LCC Utility | $108,650 | $108,400 |

# CONCLUSIONS

This paper has examined the potential impact of replacing conventional fuel-fired rooftop units with heat pumps for small office buildings in Canada. To conduct the analysis, an energy model of a small office building was developed for five regions in Canada. These building models served as a base case for detailed TRNSYS simulations of three heat pump systems: (i) a conventional air-source heat pump, (ii) a cold-climate air-source heat pump, and (iii) a ground-source heat pump. Results from these annual simulations were used as part of a lifecycle cost analysis to determine the preferred system in each region.

Each heat pump system demonstrated strong energy use reductions for heating and cooling, with savings ranging from 25% to 53%. However, these savings often did not result in corresponding reductions in lifecycle utility costs, primarily due to the significantly higher price of electricity in comparison to natural gas. A sensitivity analysis was conducted to examine the impact of a carbon tax and future natural gas escalation rates, with results further highlighting the challenges faced by heat pump systems in light of current utility rates. One potential medium-term solution is a hybrid RTU/ASHP. Such a unit could offer energy savings while protecting the building owner against future fluctuations in natural gas and electricity prices.

It is important to keep in mind that the results presented above are for a single building type, and system integration method. Future work will examine the performance of heat pump systems into buildings with different loading (retail, restaurants), in addition to examining the sensitivity of the results to different sizes of CCHP system.

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