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Arctic Climate Horizontal Ground-Coupled Heat Pump

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ABSTRACT

Heating buildings in isolated communities of northern Quebec is done by the combustion of fuel oil. Ground-source heat pump is one of the potential technologies to replace oil furnaces but the performance of heat pumps is unknown in arctic to subarctic climate. The ground thermal properties and temperature can have a major impact on the size of the required ground heat exchangers. The simulation of different ground-coupled heat pump systems for a residential size building with horizontal slinky or straight ground heat exchangers located in Kangiqsualujjuaq was therefore achieved to anticipate potential energy savings. Simulations were based on an inventory of geological data available for the area that helped to define the subsurface temperature and thermal properties. Trench length needed for the building having an annual heating energy demand of 15.9 MWh would be between 165 m and 260 m depending on the exact thermal conductivity of the ground. A simulation of the ground heat exchanger operating temperature showed that fluid can reach -13°C, which is lower than the conventional limit of -6.5°C of commercially available heat pumps. No energy savings have been found simulating horizontal ground heat exchangers with a regular heat pump having an electric compressor, even with heat recovery from a diesel engine that would activate the compressor. Simulations performed for an absorption heat pump operated with fuel resulted in viable energy savings. A system with an air-source absorption heat pump offered annual savings of 2 075 $ and 1 482 L of fuel oil, which represents 17.4 % of the fuel consumed by a conventional furnace. Simulations for a ground-source absorption heat pump system indicated savings of 4 702 $ and 3 358 L of fuel oil per year, about 39.4 % of the fuel consumed by a conventional furnace.

1. Introduction

Remote communities of the Nunavik territory in northern Quebec rely on fossil fuel to heat buildings. The public utility Hydro-Québec produces electricity from diesel generators and distribute this electricity with independent grids not connected to the hydro-electric network. Electricity form diesel generators is sold at 5.71 cents per kWh to residential customers, below the production cost ranging from 0.30 to more than 1.00 $ per kWh, with an average of 0.43 $ per kWh in 2010, depending on the latitude of the community (Hydro-Québec, 2011). Efforts to reduce greenhouse gas (GHG) emissions and improve energy utilization in the North have led to the adaptation of renewable energy technologies in this arctic to subarctic climate. One of the technologies that could be used to reduce GHG emissions are ground source heat pump (GSHP) systems.

Drilling equipment can be hardly available in the smaller communities of northern Quebec, reducing the feasibility of vertical ground heat exchangers (GHEs) installed in boreholes although such systems are known to be efficient. Horizontal GHEs can be easier to install when there is excavating machinery available. However, the ground at shallow depths is affected by the outside air temperature. This can cause lower GHE operating temperature, on one hand, and helps to restore the ground temperature under unbalanced building loads requiring heating, on the other hand.

The Cold Climate Housing Research Center (CCHRC), located in Fairbanks in Alaska, having 7509 heating degree days per year to keep 18 °C (7509 HDD_{18}), has been monitoring a horizontal ground-coupled heat pump system in this cold climate since 2013 (CCHRC, 2016b). The coefficient of performance (COP) of the heat pump has dropped from 3.6 to 2.9 between the months of December of the first and the third year. The ground remained at freezing point during winter due to the phase change of water contained in the sedimentary deposits (CCHRC, 2016a; Garber-Slaght et Peterson, 2017). This phase change effect has been further documented with numerical simulation validated with experimental data for a borehole in sandy deposits that remained at the freezing point for a long period of time when saturated, while it would drop at lower temperature when unsaturated (Eslami-nejad et Bernier, 2012). The higher thermal conductivity of saturated sand compared to dry sand could also be a factor explaining that phenomenon.

The operation of a horizontal ground-coupled heat pump system for a research station having the size of a residential building to be constructed in Kangiqsualujjuaq, Nunavik (8428 HDD_{18}; Figure 1), was modeled in this study. The objective of the work was to evaluate the energy savings that a ground-coupled heat pump system can potentially provide at this location and to illustrate the challenges associated to the design of such system in a northern arctic to subarctic climate. A slinky horizontal GHE coupled to a heat pump to meet part of a building heating demand was simulated analytically. The system operating cost was compared for different heating scenarios considering an oil furnace, a ground-coupled heat pump with the electric compressor driven by a either a diesel engine, an electric engine supplied by a generator set and a heat recovery system with a diesel engine drive. Systems heated by an air and ground source gas-absorption heat pumps was additionally considered for comparison purposes.

Figure 1: Map of Nunavik communities showing the location of Kangiqsualujjuaq.

2. Background information

2.1 Building description

The building that was studied is planned to be constructed in 2017. It will be used by the *Centre d'études nordique* (CEN) for research purposes and as accommodations for researchers. The overall dimensions are 15.8 m by 9.4 m and it sits on jacks. The building was modeled with the OpenStudio plugin for Sketchup and the OpenStudio program itself (Figure 2), while the building simulations were achieved with the EnergyPlus program. The heating set points are at 21°C during the day and 15.6°C during the night. The space types modeled were based on

189.1:2009 Midsise Apartment CZ4-8 and *189.1:2009 Office CZ4-8* schedules. The building envelope specifications are presented in Table 1.

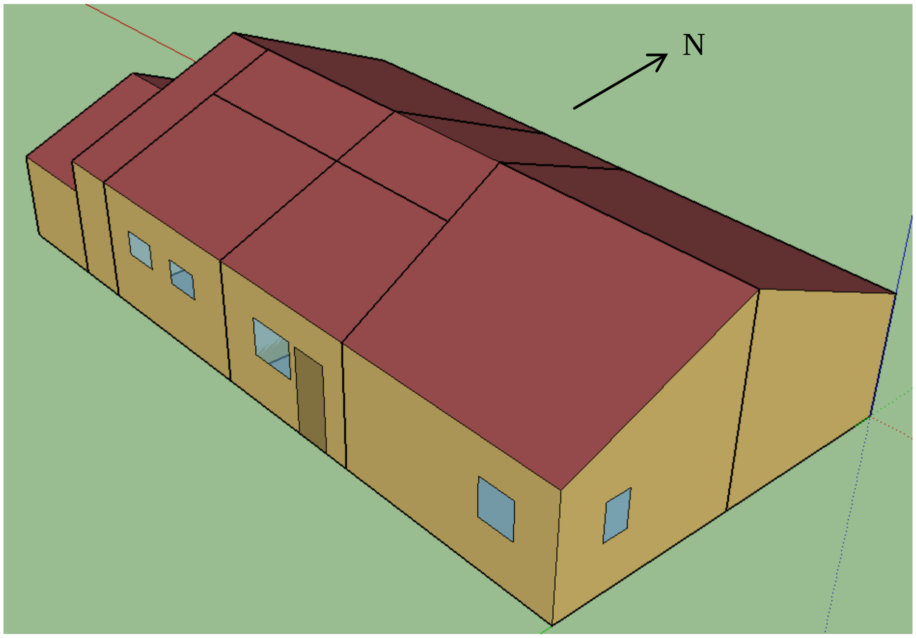

Figure 2: Building model on Sketchup.

Table 1: Building Envelope specifications.

	RSI (R-value) $[m^2\ K\ W^{-1}]$ $[ft^2\ °F\ hr\ BTU^{-1}]$	U-Value $[W\ m^{-2}\ K^{-1}]$
Outside walls	5 (28.4)	
Floor	6.7 (38.2)	
Roof	10.1 (57.6)	
Windows		U-1.8
Doors		U-2.0

The outside air is supplied by a 120 CFM heat recovery ventilator. A water to water heat pump distributes heat to each zone, which are equipped with gas reheat variable air volume (VAV) air terminals.

The first simulation was done with baseboard heating according to a weather file for Kuujjuaq (Figure 1), at 160 km to the West-Southwest from Kangiqsualujjuaq since there was no weather data available for this last location. Note that the mean annual air temperature in Kuujjuaq is slightly warmer than that in Kangiqsualujjuaq.

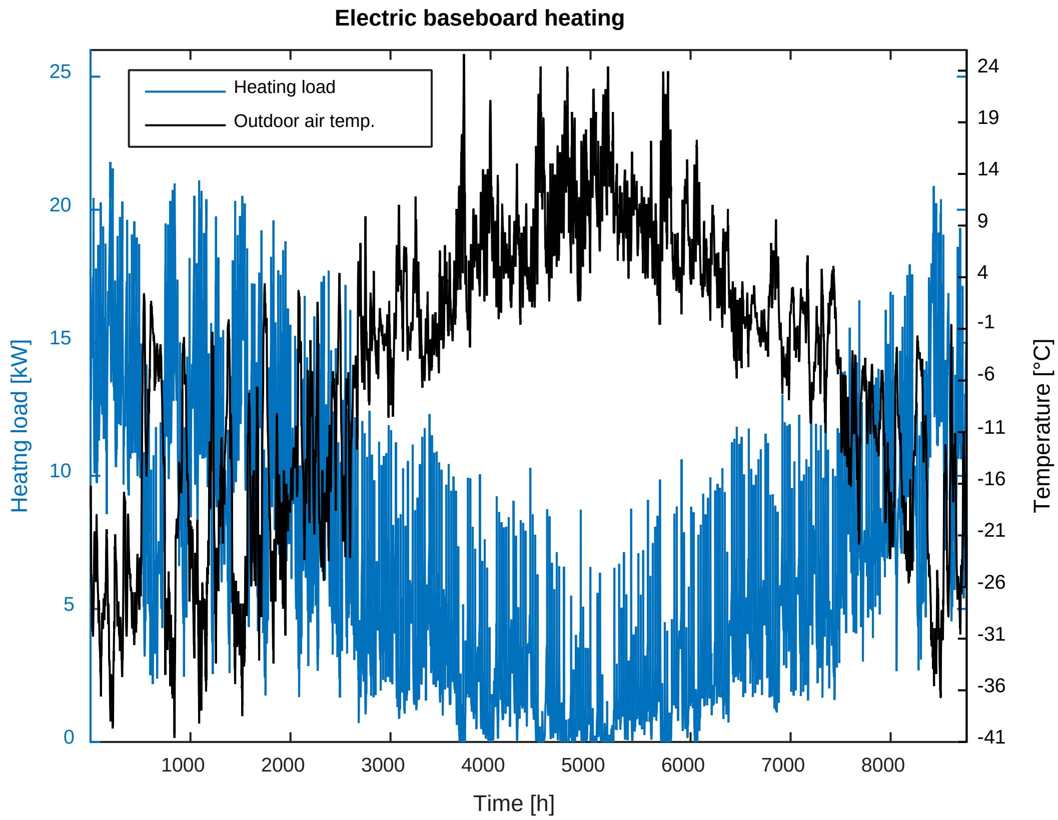

Figure 3: Heating loads and outside air temperature.

The outside air temperature reaches 25°C during summer and decreases to -41°C during the winter (Figure 3). It is observed that the outside air temperature is correlated with the building heating load: at -41°C the induced peak heating load is 24.9 kW and heating is required even during the warmest summer months. The baseboard heating energy required is 68 100 kWh annually.

The first step in sizing a geothermal heat pump system is to analyze the thermal loads. Each hourly heating demand was presented on a scatter plot (Figure 4, left) and placed in descending order (Figure 4, right). The loads depend on the outside air temperature. The higher value zones on the left plot are the first hours of the days, where the set point changes from 15.6°C to 21°C. The ascending order heating loads shows that 6.2 kW (25% of the peak heating load) could cover the heating demand for up to 62.2% of the year.

The performances of a heat pump are influenced by the entering fluid temperatures affected by the shallow ground temperatures under the influenced of the outside air. A detailed analysis of the ground thermal behavior was mandatory.

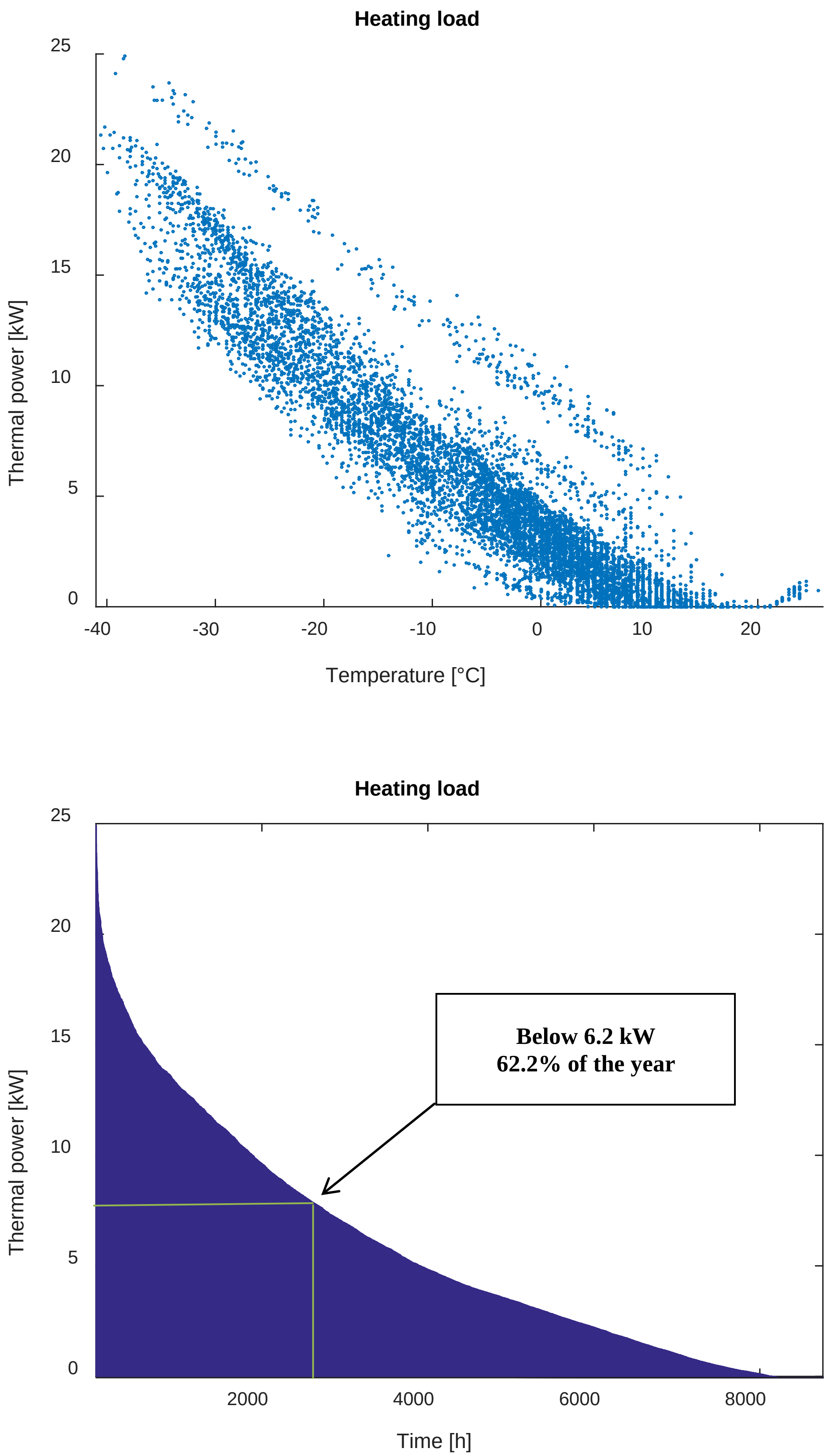

Figure 4 Building heating loads.

2.2 Ground thermal state and properties

The ground temperatures have been recorded at three kilometer west of the CEN building site. The air and ground temperature at depths from 0 to 2.4 m was actually measured in Kangiqsualujjuaq from 1988 to 1996 at *Terrasse marine* (KANGTMA location; Centre d'études nordiques, 2017). Data from 1998 were extracted from the existing databased and used to model the ground-coupled heat pump system (Figure 5).

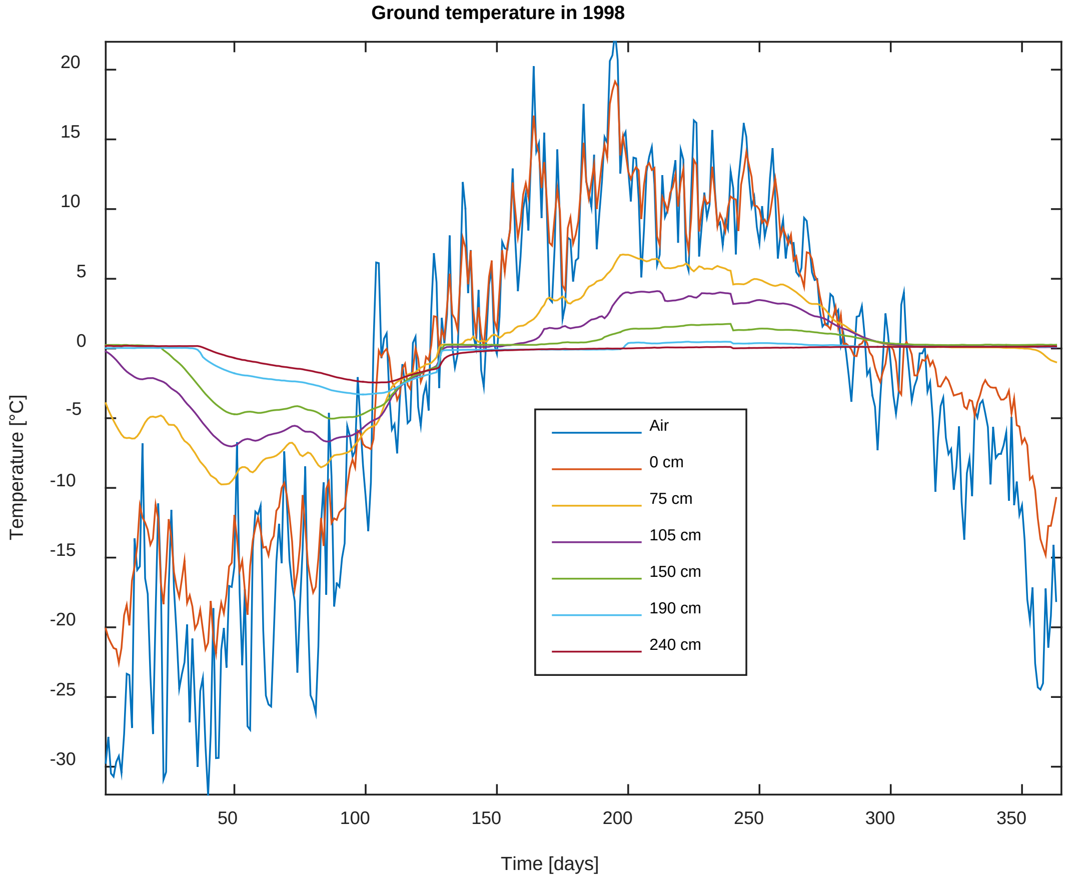

Figure 5 Ground temperature at different depths measured in 1998 at KANGTMA in Kangiqsualujjuaq (Centre d'études nordiques, 2017).

The air temperature amplitude ranges from -31°C to 21°C, while the ground temperature amplitude reduces with depth. The Kusuda model (Kusuda *et al.*, 1965; DoE, 2016) was fitted to the observed temperature curves and used in EnergyPlus to simulate the ground temperature. The ground temperature is determined in this case with:

$$T(z,t) = \overline{T}_s - \Delta T_s \cdot e^{-z\sqrt{\frac{\pi}{\alpha\tau}}} \cdot cos\left(\frac{2\pi t}{\alpha t} - \theta\right) \quad (1)$$

where, T (°C) is the ground temperature as a function of time and depth, T_s (°C) is the average annual ground surface temperature, ΔT_s (°C) is the annual amplitude of the ground surface temperature, Θ (days) is the phase shift or day of minimum surface temperature, α (m^2/d) is the thermal diffusivity of the ground and τ (365 d) is a time constant. The only ground property taken into account with this approach is the thermal diffusivity, which was deduce from the temperature measured at different depths.

The minimum Entering Water Temperature (EWT) of a heat pump is typically set to 0°C when designing a ground-coupled heat pump system in southern Canadian latitudes. The EWT limit of heat pumps is commonly -6.5°C (-20°F), reducing possible operations under low ground temperature common to northern territories. However, this limit can be overcome by designing custom heat pumps, for which heat exchanger material will sustain low operating temperature conditions. In this paper, the low temperature limit was not taken into account.

Superficial geological map of Quaternary deposits evidencing the presence of marine clay, remolded soil and gravel and sand geological units was available for the Kangiqsualujjuaq area (Carbonneau et al., 2015; Figure 6). At the location of the studied building, the sedimentary deposits are gravel and sand. The deposits were assumed to be wet to identify a range of thermal conductivity that can be associated to the type of sediments with respect to temperature (Farouki, 1981). The thermal conductivity for saturated gravel varies from 2.1 W m^{-1} K^{-1} to 3.2 W m^{-1} K^{-1} between -5°C and 7°C, respectively. A fine sand with a 60% saturation at -10°C and 10 °C has a the thermal conductivity of 1.0 W m^{-1} K^{-1} to 2.0 W m^{-1} K^{-1}. The density further influence the thermal properties of the sediments under freezing and unfreeze conditions (Farouki, 1981). A thermal conductivity of 2.5 W m^{-1} K^{-1}, based on Farouki's work, and a density of 1800 kg m^{-3} and a specific heat of 1555 J kg^{-1} K^{-1} was used in this study based on the average values of grey slightly silty sandy gravel and fine sand (saturated; Hamdhan et Clarke, 2010).

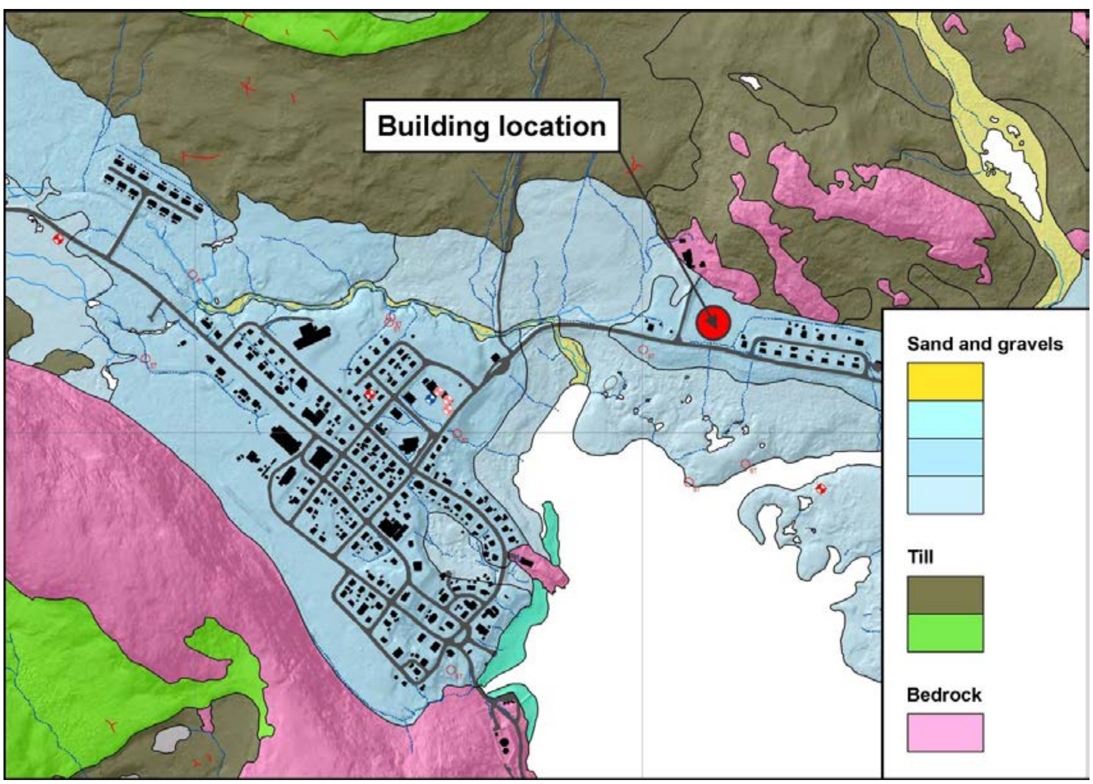

Figure 6: Simplified geological map of Kangiqsualujjuaq (Carbonneau et al., 2015).

3. Methodology

3.1 Sizing the GHE

Sizing of the horizontal GHE was done using the so called ASHRAE method for straight pipes buried in trenches. The ASHRAE method consist in using three heat pulses to represent the ground loads (Kavanaugh et Rafferty, 1997):

$$H = \frac{q_a R'_a + q_m R'_m + q_h \left(R'_p + R'_d\right)}{T_o + dT - T_f} \tag{2}$$

where q_a, q_m and q_d are the mean annual, the monthly and hourly block thermal loads, R'_a, R'_m, R'_b and R'_d are the annual, monthly, pipe and hourly bloc thermal resistance evaluated from G-Functions. dT is the mean temperature difference at the given depth, which is found from the Kusuda relation (Eq. 1).

These resistances were evaluated using the thermal response factors (G_h – functions) associated with a specific horizontal field arrangement (Bose *et al.*, 1985):

$$R'_a = \frac{G_h(Fo_f) - G_h(Fo_1)}{k_s} R'_m = \frac{G_h(Fo_1) - G_h(Fo_2)}{k_s} R'_h = \frac{G_h(Fo_2)}{k_s} \tag{3}$$

$$Fo_f = \frac{\alpha t_f}{r_p^2} Fo_1 = \frac{\alpha (t_f - t_1)}{r_p^2} Fo_2 = \frac{\alpha (t_f - t_2)}{r_p^2} \tag{4}$$

The G_h functions are associated with the horizontal arrangement (see Bose, et al 1985). The calculation was done with a single pipe in a single trench for simplicity but any one can be used. T_o, is the mean ground temperature in Eq. 2 and T_f the mean fluid temperature at the end of the calculation period:

$$T_f = \frac{T_{in} + T_{out}}{2} \tag{5}$$

$$\frac{q_h}{2 \cdot \dot{m} \cdot C_p} = \frac{T_{out} - T_{in}}{2} \tag{6}$$

The $\dot{m}$ is the fluid mass flowrate and Cp its specific heat. The pipe thermal resistance is associated with the conduction resistance of the pipe material and the convection resistance of the fluid:

$$R'_p = \frac{\ln\left(r_{po}/r_{pi}\right)}{2\pi k_{pipe}} + \frac{1}{2\pi r_{pi} h_{fluid}} \tag{7}$$

where r_{po} and r_{pi} are the outer and inner pipe radius, k_{pipe} is the thermal conductivity of the pipe and h_{fluid} the convection coefficient of the fluid. A slinky pipe configuration was selected over straight pipe because the slinky configuration can be used in restricted spaces (Xiong *et al.*, 2015). In theory, the ASHRAE equation can be used when the response factors from the horizontal arrangement are replaced by the G-function given by Xiong et al. However, these functions are dependent of the final configuration and an iterative procedure was used instead. Eq 2 was rearranged in order to evaluate the temperature at the exit of the field and the length of the field was iterated until the design temperature was found:

$$T_f = T_o + dT - \frac{q_a R'_a + q_m R'_m + q_h (R'_p + R'_d)}{H_p} \tag{8}$$

where again the expression of the soil resistances are similar to Eqs. 3 and 4 except that the G_h functions are replaced by the expressions given by Xiong et al.. In this case:

$$T_{out} = T_f + \frac{q_h}{2 \cdot \dot{m} \cdot C_p} \tag{9}$$

It should highlight that in Eq. 8 H_p is the total pipe length and not the total trench length. The total trench length is found according to the pitch and the ring diameter of the slinky coils.

The selected water-to-water heat pump used to size the GHE can supply 6.4 kW of heating capacity at the lowest modeled temperature with a COP of 2.

Sizing and simulation of the slinky heat exchanger was achieved with the parameters outlined in Table 2. The final outlet fluid temperature in the GHE was selected to be -14°C after five years, considering that a custom heat pump would be used. The energy simulation of a system was required as the next step to evaluate the energy consumption of such system and to evaluate the potential energy savings for the building operators.

Table 2: Ground heat exchanger parameters used for sizing calculation and simulation of the building heating system.

Design Flow Rate [$m^3\ s^{-1}$]	0.000568
Sediment Thermal Conductivity [$W\ m^{-1}\ K^{-1}$]	2.5
Sediment Density [$kg\ m^{-3}$]	1800
Sediment Specific Heat [$J\ kg^{-1}\cdot K^{-1}$]	1555
Pipe Thermal Conductivity [$W\ m^{-1}\cdot K^{-1}$]	0.4
Pipe Density [$kg\ m^{-3}$]	641
Pipe Specific Heat [$J\ kg^{-1}\cdot K^{-1}$]	2405
Pipe Outer Diameter [m]	0.0267
Pipe Thickness [m]	0.0048
Heat Exchanger Configuration	Horizontal
Coil Diameter [m]	0.8
Coil Pitch [m]	0.4
Trench Depth [m]	2
Trench Length [m]	165
Number of Trenches	1
Length of Simulation [years]	5

3.2 GSHP simulation

A detailed simulation of the GSHP operation was carried out using the OpenStudio and EnergyPlus programs. The slinky horizontal GHE model used by EnergyPlus is that proposed by

Xiong et al. (Xiong *et al.*, 2015), with simplifications to ensure fast computation of G-functions (DoE, 2016). Many output variables are available from the program such as heat transfer rate, inlet, outlet and average fluid temperature and mass flowrate.

The energy consumption and cost of four technological scenarios were initially compared: first an 80% efficient oil furnace, second a diesel engine driven heat pump (DEHP; NN *et al.*, 2011) with a 45 % efficiency motor, third the same as case two with a 80% efficiency heat recovery system and finally, four, an electric heat pump with electricity supplied from diesel generators assuming an efficiency of 33 % (Genset). The cost of fuel oil was assumed to be 1.40 $/L and that of electricity equal to 0.80$/kWh (Hydro-Québec, 2011). This is the reported price by Hydro-Québec in 2010. This price is representative of the current price in 2017 (since the historical hub price of diesel in 2010 is similar to 2017).

3.3 Absorption heat pump, air-source

An absorption heat pump is a device that does not need electricity to operate. It uses fuel combustion to heat or cool buildings. The process is more complex than a conventional heat pump (Figure 11), where the compressor is activated by an electric motor. A thermal compressor, enclosing and an absorber and a desorber, replaces the electric compressor in an absorption heat pump. The desorber needs a high temperature source, such as fuel combustion, and the absorber rejects heat at a medium range temperature (e.g. 50°C).

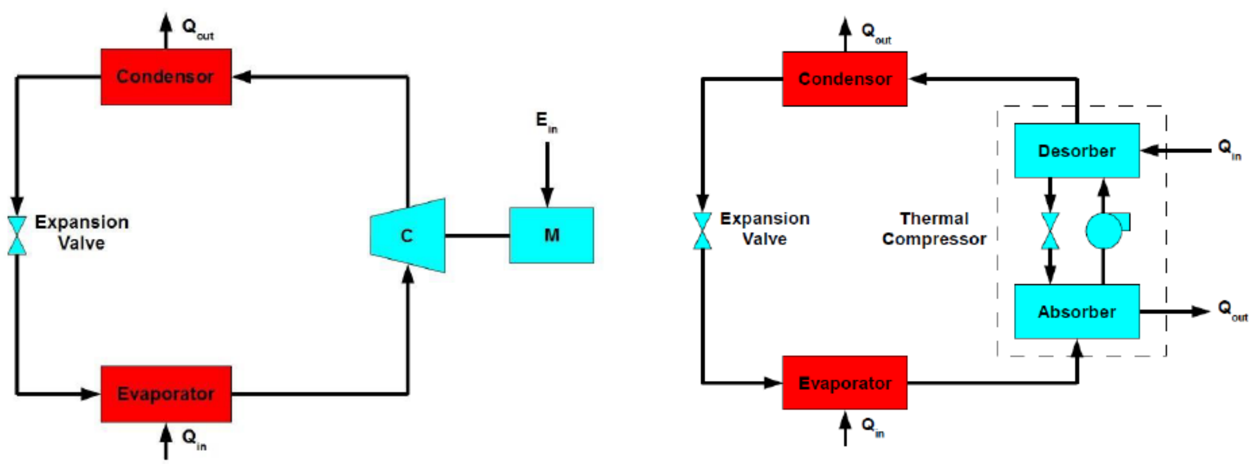

Figure 7: Schematic diagrams of an electric heat pump (left) and an absorption heat pump (right) (Garrabrant, 2015).

An air-source absorption heat pump for low temperature climate has been developed (Garrabrant, 2015). It can supply heat at ambient air temperature as low as -25 °C, with a COP of 1.2, and has a COP of 1.4 at 8.3 °C. It uses natural gas as an energy source for the desorber and the cost is about 4 500 $ while it can supply 23.3 kW at 8.3°C and 16.3 kW at -25°C, which is a good match for the building simulated in this study.

A Matlab model was developed in this study using the performance curves of the experimental setup reported by Garrabrant (2015). The air-source absorption heat pump and a furnace supplies heat to the building in this model. The absorption heat pump cannot fulfill the heating load when the temperature is lower than -25°C and the furnace turns on.

3.4 Absorption heat pump, ground-source

A ground-source absorption heat pump could have better performances than an air-source system by circulating the chilled water in a horizontal GHE rather than exchanging heat with the outside air. A TRNSYS model was developed in this study to couple the building loads, determined with EnergyPlus, to a horizontal GHE using an assembly editor simulating the performances of the absorption heat pump (Figure 8).

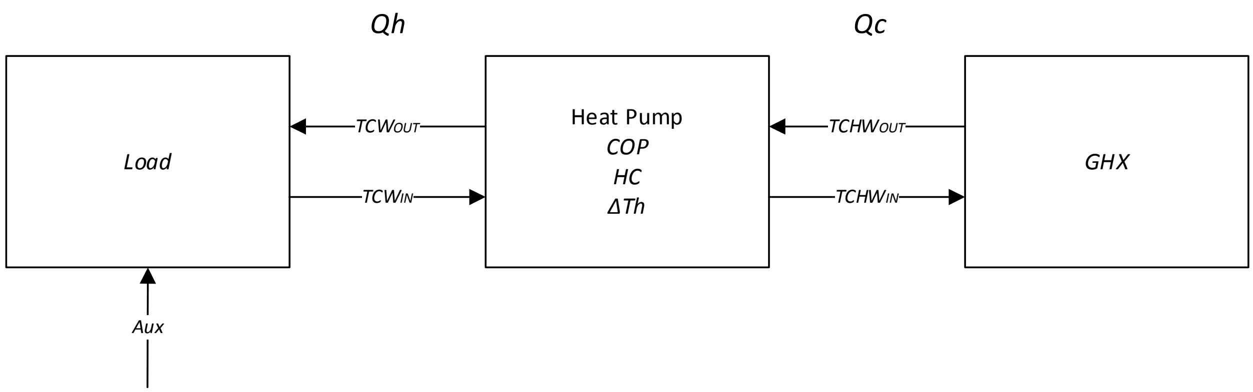

Figure 8: Schematic diagram of a ground-source absorption heat pump.

The performance of the heat pump was described with a 2nd order polynomial fit. The system variables are function of the chilled water inlet temperature or the GHE outlet ($TCHW_{IN}$):

$$COP = -0.000108126537853727 * TCHW_{IN}^2 + 0.00792336096429471 * TCHW_{IN} + 1.436850889176639 \quad (10)$$

$$\Delta Th = 0.00186656687156818 * TCHW_{IN}^2 + 0.156429080188268 * TCHW_{IN} + 10.9427947168472 \quad (11)$$

$$HC = -0.00508608396928548 * TCHW_{IN}^2 + 0.131801430472601 * TCHW_{IN} + 22.7665139689586 \quad (12)$$

The load side heat transfer rate (condenser, Qh) was calculated with the minimal building load value (Qb), the heat capacity of the heat pump (HC) and the hydronic temperature difference heat transfer rate ($Q\Delta Th$). The auxiliary heat supplied by the furnace (Aux) is the difference between the heat supplied by the heat pump and the building load:

$$Q\Delta Th = \dot{m} \cdot Cp \cdot \Delta Th \tag{13}$$

$$Qh = \min\,(Qb, HC, Q\Delta Th) \tag{14}$$

$$Aux = Qb - Qh \tag{15}$$

Then, the inlet cooling water temperature (TCW_{IN}) of the heat pump was evaluated with the design cooling water set point (TCW_{OUT}):

$$TCW_{IN} = TCW_{OUT} - \frac{\mathrm{Qh}}{\dot{m} \cdot Cp} \tag{16}$$

The source side heat transfer rate (evaporator, Qc) can then be evaluated with:

$$Qc = Qh - \frac{Qh}{COP} \tag{17}$$

The chilled water outlet temperature outlet was finally evaluated with:

$$TCHW_{OUT} = TCHW_{IN} - \frac{\mathrm{Qc}}{\dot{m} \cdot Cp} \tag{18}$$

4. Results

4.1 Ground temperature

The measured temperature profile at 1.05 m depth has similar amplitude to that simulated when using a diffusivity of 0.0072 m^2/day (Figure 9). Greater differences can be observed for measured and simulated temperature at 2.4 m depth. The main difference is an observed temperature of 0°C maintained for most part of the year while simulated temperatures are lower. This is due to the phase change of water from liquid to solid (before summer) and from solid to liquid (after summer). These differences have not been take into account when simulating the ground-coupled heat pump system and could cause an over evaluation of the energy consumption of the heat pump. Said differently, this phase change of water could be beneficial to the performances of a geothermal heat pump system.

Thermal diffusivity and especially thermal conductivity have an important role to play in the behavior of a GSHP system. The higher the thermal diffusivity, the shorter a GHE is required to reach a certain temperature after heat has been extracted. Water can affect the soil properties in a shallow horizontal GHE. Here it can be pointed out that ice, water in its solid form, has a much larger diffusivity than in its liquid form. Operating a GHE below the freezing point would be more efficient with more water there is in the subsurface.

4.2 GSHP sizing

Both ASHRAE method for straight horizontal GHE and Xiong method for slinky heat exchanger are affected by thermal conductivity of the ground (Table 3). The trench length doubles with the Slinky GHEs when the ground thermal conductivity is reduced from 3.5 to 1.5 W m^{-1} K^{-1}. The same trend is observed when sizing the straight pipe system.

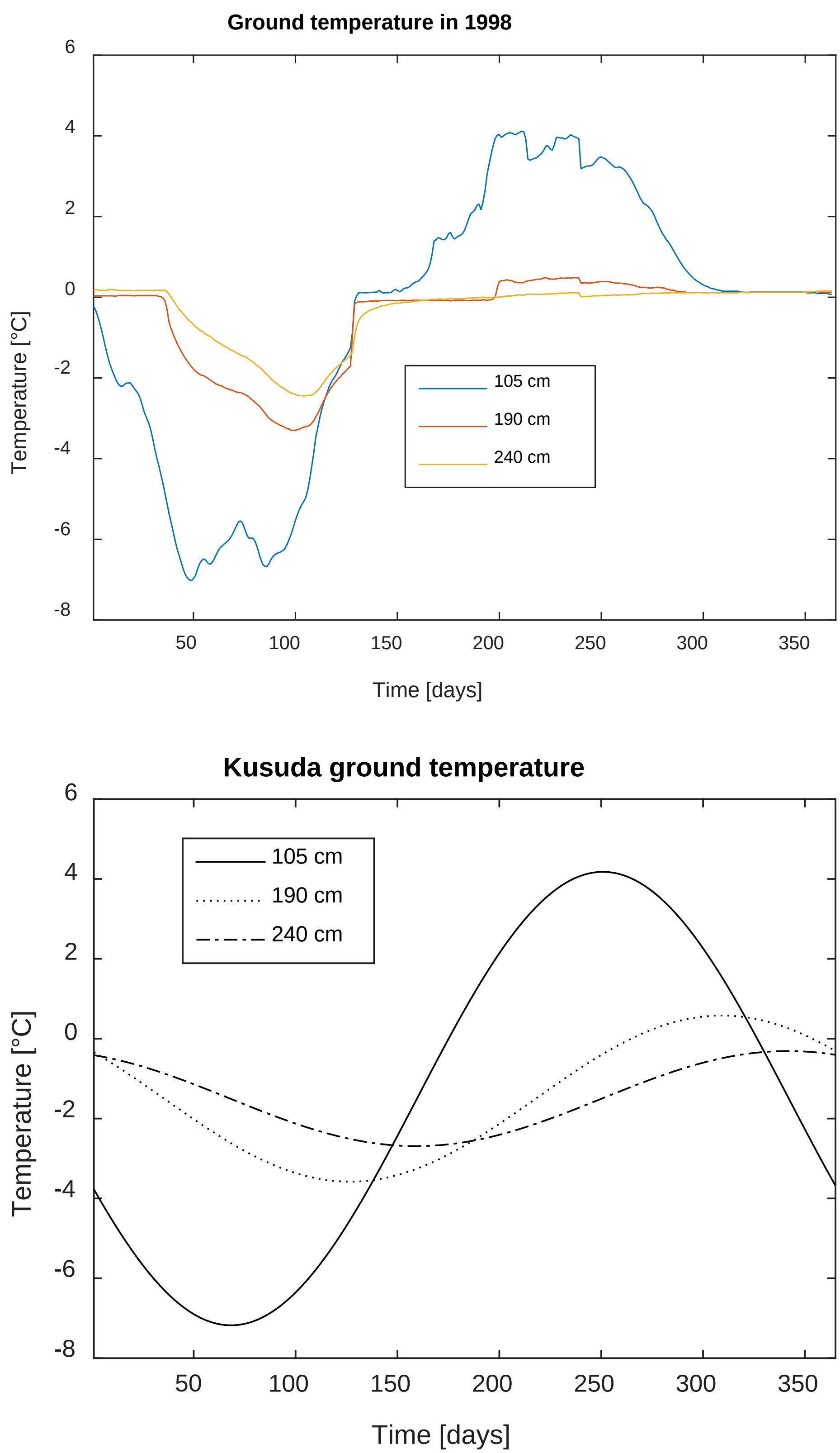

Figure 9: Measured (top) and simulated (bottom) ground temperature.

Table 3: Effect of thermal conductivity on trench length.

Thermal conductivity [W m^{-1} K^{-1}]	Ground heat exchanger length [m]		
	Slinky		Straight
	Trench	Pipe	Trench & pipe
1.5	260	1894	650
2.5	165	1203	433
3.5	130	947	338

4.3 GSHP simulations

The detailed energy simulation allows the fluid temperature to be evaluated at short time steps, hourly steps in this case (Figure 10), do determine energy savings provided by the heat pump system.

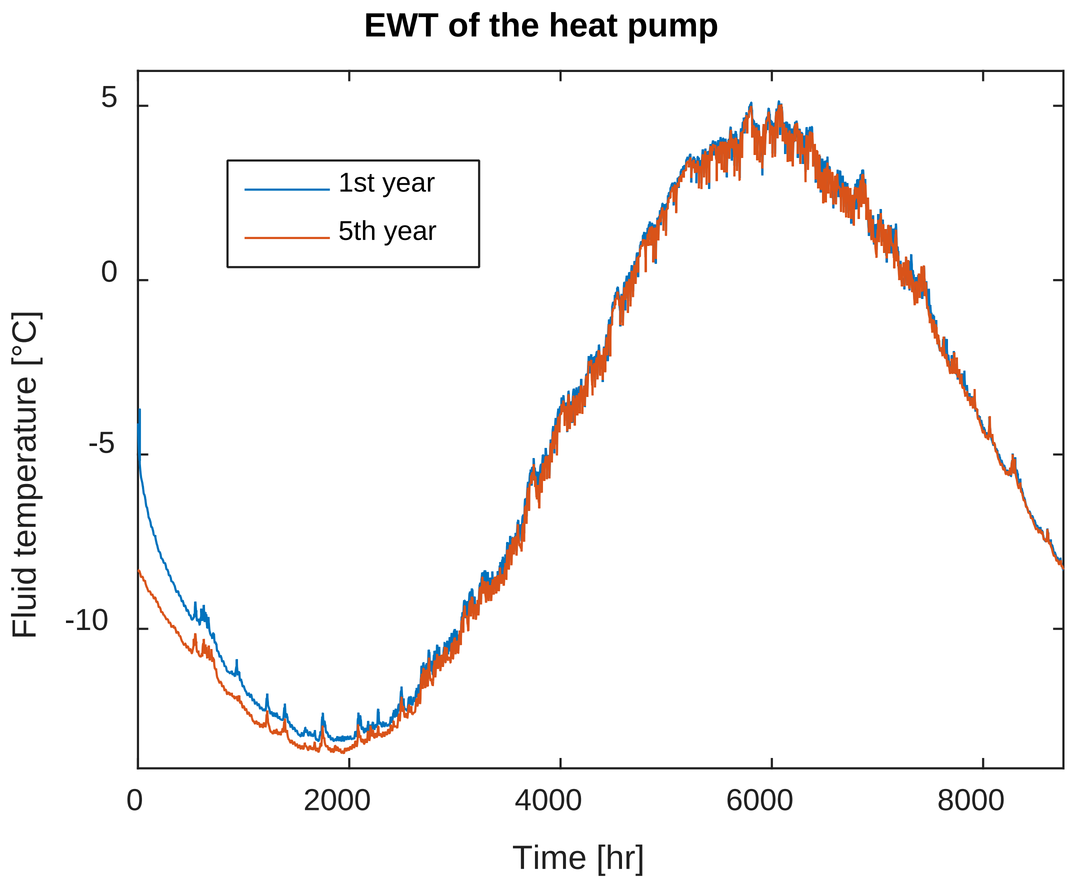

Figure 10: EWT of the heat pump.

The heat pump EWT for the first hours of the simulation starts close to undisturbed ground temperature at -3°C.The temperature is -8.3°C at the beginning of the second year. The minimum temperature of the first year is -13.2°C and the fifth year is -13.6°C. These low temperatures have an impact on the COP of the heat pump (Figure 11).

The simulated COP is on the low side of the performance curve, with an average of 3.3, stable over the five years of simulation. The trend line shows an increase of performances with higher EWT.

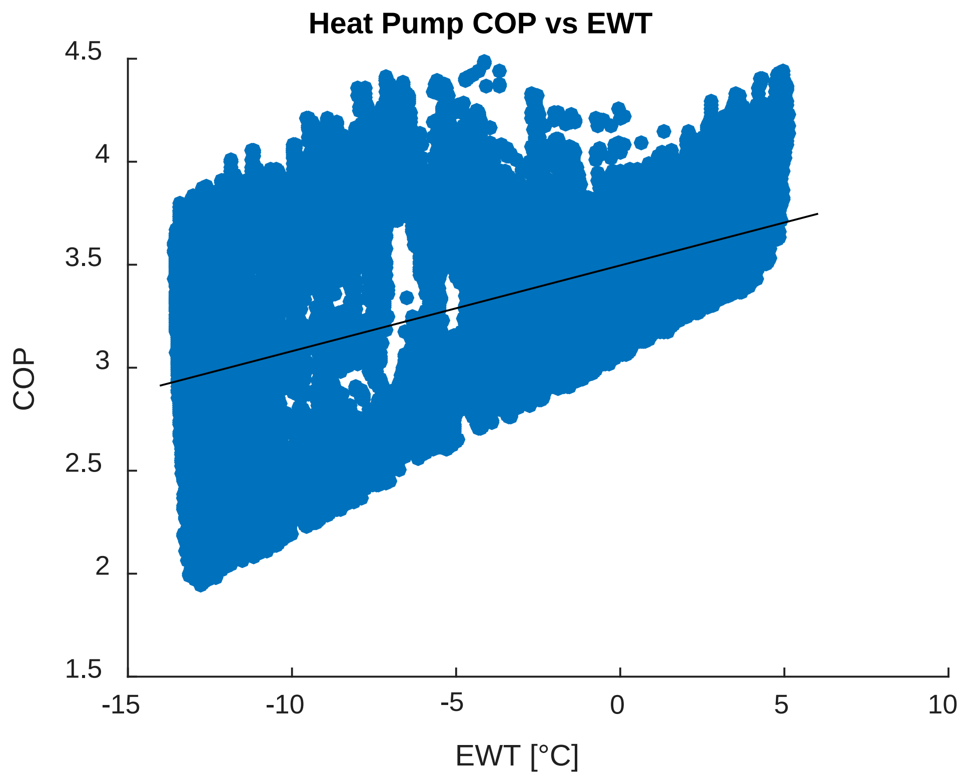

Figure 11: Heat pump COP vs EWT.

4.4 Absorption heat pump, air-source

The COP and the heating loads from the combined air-source absorbtion heat pump and furnace system were evaluated with the hourly simulation (Figure 12). The COP of the absorption heat pump was between 1.17 and 1.57. The furnace supplies heat, starting at a temperature as low as -20°C, when the heat pump cannot completely cover the heating loads of the building.

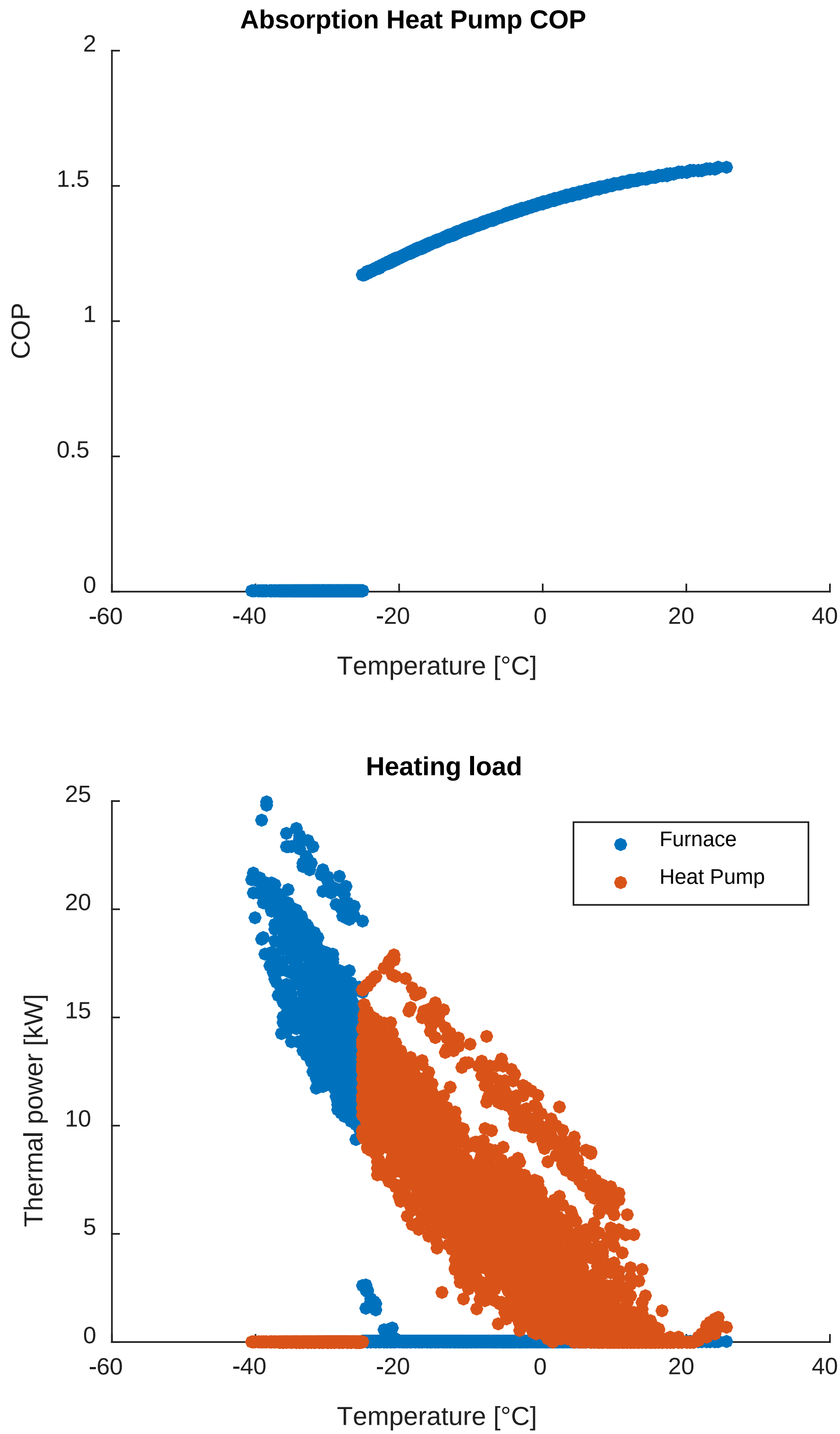

Figure 12: Air-source absorption heat pump COP (top) and heating load (bottom).

4.5 Absorption heat pump, ground-source

It was convenient that the experimental low temperature absorption heat pump (Garrabrant, 2015) was the perfect size for the simulated building. The heating requirements of the house were 68 100 kWh and the fuel oil furnace only had to supply 1182 kWh. The absorption heat pump ground-source maintained an average COP of 1.33.

4.6 Energy savings

The energy consumption and energy cost for the different system simulated were evaluated over a five years period and were similar for every years such that only one year results are presented (Table 4). The furnace requires less energy to heat the building and is cheaper to operate than the GSHP options, considering a diesel engine, a diesel engine with heat recovery and the compressor activated by electricity from diesel generators. The absorption heat pump systems, air or ground-source, consumed less energy than a fuel oil furnace. The energy and GHG emissions savings for a yearly period are presented in Table 5. No savings were obtained for GSHP systems when compared to a furnace even if when heat recovery system was added to the diesel engine. The annual energy bill for both the air-source and the ground-source absorption heat pumps are lower than the furnace. The air- and ground-source absorption heat pump simulation resulted in an annual saving of 2 075 $ and 4702 $ and 1 482 L and 3 358 L of fuel oil, respectively. This later solution is the most interesting scenario to reduce operating costs.

Table 4: Energy and cost of heating technologies (1 year).

	Fuel oil [L]			
Technology	**Heat pump**	**Furnace**	**Electricity [kWh]**	**Operating cost**
Oil furnace	0	8517	0	11,924 $
GSHP DEHP	8691	2340	0	15,444 $
GHSP Genset	0	2340	38956	34,441 $
GSHP DEHP HR	8691	620	0	13,034 $
Absorption Heat Pump Air-Source	4451	2584	0	9,849 $
Absorption Heat Pump Ground-Source	5040	119	0	7,223 $

5. Discussion and conclusions

Horizontal GSHP systems have been installed in cold climates. A system is in fact operating at the CCHRC in Fairbanks Alaska. This location has 7509 heating degree-days below 18°C compared to 8428 for Kangiqsualujjuaq. Fuel oil is the main source of heat for buildings at this later location and efforts should be made to reduce this energy consumption.

Table 5: Energy and cost savings (1 year).

Technology	Annual savings		
	Dollar ($)	Fuel [L]	GHG [tCO_2e]
Furnace			21.23
GSHP DEHP	-3,519	-2514	27.50
GSHP Genset	-22,517	-6525	37.49
GSHP DEHP HR	-1,110	-793	23.21
Absorption Heat Pump Air-Source	2,075	1482	17.53
Absorption Heat Pump Ground-Source	4,702	3358	12.86

A GSHP system was designed and simulated for a research center to be constructed at Kangiqsualujjuaq. Commercially available heat pumps commonly have a minimum entering water temperature of -6.5°C. Custom heat pumps would have to be design and built since the ground temperature can decrease as low as -10°C at a 2 m depth. The simulations performed did not took into account this low temperature limit of -6.5°C.

Measured ground temperature indicates that the ground freezes and thaws every year. This latent heat maintains ground temperature at 0°C for part of the year, which could result in better performances of horizontal GHEs. One limitation of the model is that it does not take into account the phase change of water in the ground such that simulation results can be considered conservative. The advantage of latent heat would only be available for the first year or so since heat is not injected back during summer as the heat pump is still in heating mode. The higher thermal conductivity of the ice water can additionally be considered as an advantage compared to the water liquid state.

The ground thermal conductivity is shown to have a major impact on the length of trenches needed for the slinky GHEs. A thermal conductivity of 3.5 W m^{-1} K^{-1} compared to 1.5 W m^{-1} K^{-1} resulted into trench length of 130 m and 260 m, respectively. The risk of an under sized system should be minimized by sampling the sedimentary deposits and measure the thermal conductivity in the laboratory.

The simulation showed a heat pump coefficient of performance between 2 and 4.5, with an average of 3.3. Four technologies were compared, which are an oil furnace, a geothermal heat pump operated by a diesel driven engine with and without heat recovery and a geothermal heat pump with electricity for the compressor supplied from a generator. The performances of the horizontal GSHP systems did not achieve energy savings over a conventional furnace. Fuel oil furnace would require 8 517 L of fuel oil annually at a cost of 11 924 $. An air-source absorption heat pump simulation showed more interesting results with 2 075 $ of savings per year and 3 358 L of fuel oil. The return on investment for this apparatus could be approximately 2.2 years considering an estimated unit price of 4 500 $ and neglecting shipping and installation costs. A ground-source absorption heat pump system should cost less than 23 510 $ and would lead to a return on investment of 5 years. Detailed simulations with on-site tests need to be conducted to validate the potential energy savings.

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