Kummert, Michael and Bernier, Michel (2008) *Sub-hourly simulation of residential ground coupled heat pump systems.* Building Services Engineering Research and Technology, 29 (1). pp. 27-44. ISSN 0143-6244

Sub-hourly simulation of residential ground coupled heat pump systems

Michaël Kummert[1] and Michel Bernier[2]

AUTHOR DRAFT
Building Services Engineering Research Technology, vol. 29 (1), 2008, pp. 27–44
DOI: 10.1177/0143624407087286

[1] Energy Systems Research Unit – University of Strathclyde, Dept. of Mechanical Engineering, 75 Montrose street, Glasgow, G1 1XJ – United Kingdom

[2] École Polytechnique de Montréal, Dépt. De Génie Mécanique
CP 6079, Succ. Centre-Ville – Montréal, QC H3C 3A7 – Canada

ABSTRACT

Residential Ground Coupled Heat Pump systems are usually characterized by an ON/OFF behaviour of the heat pump with typical cycling frequencies of 1 to 4 cycles per hour. The ground loop fluid pump has the same ON/OFF behaviour and the borehole heat exchanger operates either in full flow or no flow conditions. Typical hourly simulations of GCHP systems use steady-state models for the heat pump and the borehole fluid (transient models being used for buildings and heat transfer in the ground). This paper reviews the models used in typical hourly simulations as well as transient models that are available and compares the results obtained using the two classes of models within the TRNSYS simulation environment. Both the long-term energy performance and the optimum system design are compared. It is shown that using steady-state models leads to an overestimation of the energy use that ranges from a few percents with oversized borehole heat exchangers to 75% for undersized exchangers. A simple Life Cycle Cost analysis shows that using steady-state models can lead to selecting a very different design than the one that would have been selected using dynamic models.

1. INTRODUCTION

Closed-loop Ground Coupled Heat Pump (GCHP) systems with vertical boreholes have achieved a fast-growing market penetration in North America and some European countries in recent years. Figure 1 represents the most common system design in North-America, which includes a water-to-air heat pump and one or more vertical boreholes with a closed fluid loop filled with water or an antifreeze solution.

Residential systems are usually designed according to simple rules-of-thumb (see e.g. NRCan-OEE, 2004) and several design tools based on monthly averages are available (see section 2 for more details). More recently, the desire to design ultra-low energy houses, e.g. net-zero energy homes, has resulted in more value being attached to integrated performance simulation of residential GCHP systems. Several integrated building performance simulation tools have the capability to simulate such systems, including TRNSYS (Klein et al., 2004). The models implemented in other integrated simulation tools often share the same basic assumptions.

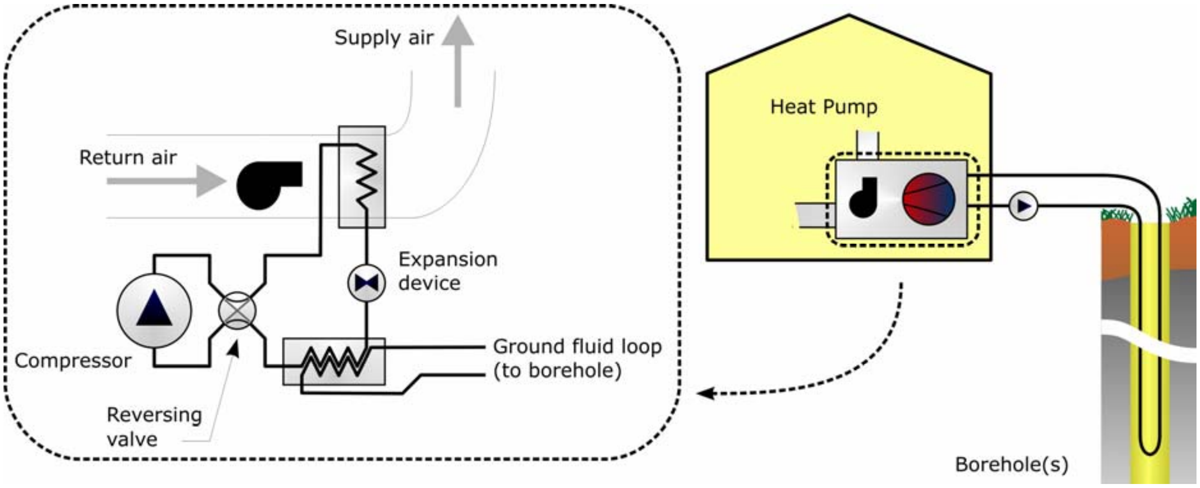

Figure 1: Typical GCHP system with forced-air heating and cooling

A general assumption in hourly simulations of buildings and GCHP systems is that transient effects can be neglected in some parts of the system: the heat pump, the geothermal fluid loop and the borehole itself. The aim of this paper is to investigate the importance of this assumption in designing and simulating the performance of residential GCHP systems.

Note: in the following, "Transient" or "Dynamic" is used as shorthand for models that include dynamics in the fluid loop, borehole and heat pump. "Steady-State" is used as shorthand for models that neglect those dynamics. It should be stressed that transient effects in the ground volume around the borehole and in the building structure are considered in both types of models.

Figure 2 and Figure 3 illustrate the differences between the two approaches compared in this paper using a simple step response in heating mode.Figure 2 shows the response of the ground inlet and outlet (return) temperatures just after the heat pump is switched ON. Ground and borehole parameters are provided in Table 2 (the borehole length is 55m). With the selected borehole diameter and flowrate, the traveling time for the fluid in the borehole is equal to 3 minutes.

The model that accounts for dynamics in the fluid loop and in the borehole ("Trn") predicts a slight increase in return temperature (as warmer water from the bottom of the borehole is pushed out) and then a smooth decrease in temperature in response to the energy removed by the heat pump. The model that does not take dynamics in the fluid loop and borehole into account ("SSt") predicts an instantaneous and much sharper drop in fluid temperatures, as the only modelled transient effect is the one that occurs in the ground volume, with a much longer time scale.

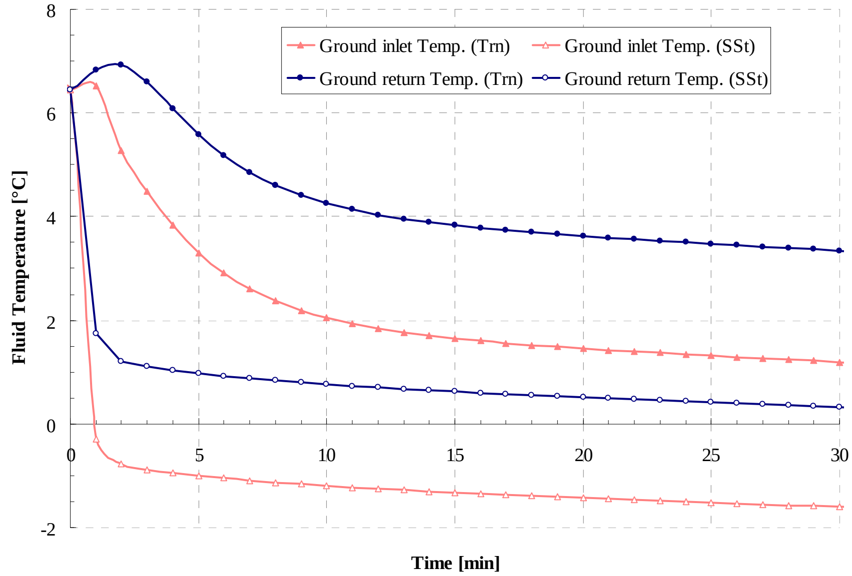

Figure 2: Ground temperatures response when heat pump is switched ON (transient and steady-state models)

Figure 3 shows the response of the heat pump itself. The transient model ("Trn") shows higher heat transfer rates and COP after a few minutes, because the fluid inlet temperature is higher. During the first few minutes, the dynamics included in the heat pump model itself result in a slower response with lower values of heat transfer rates and COP. It should be noted that plotted values represent the average over previous time step, according to the TRNSYS convention - e.g. the value at "1 min" is the average between t = 0 and t = 1 min).

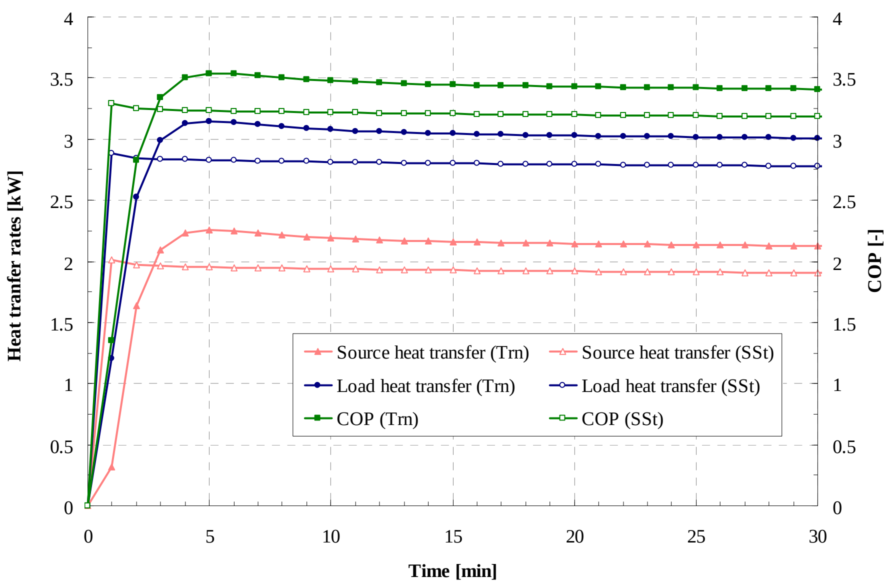

Figure 3: Short-term heat pump response after it is switched ON (transient and steady-state models)

The two Figures above show that the simulated performance using transient or steady-state models will be different: the capacity and COP of the heat pump is different, and the operation of high and low temperature limiters will be different (e.g. in the example chosen the heat pump might stop operating when the ground return temperature reaches 0°C, which happens after 1 hour with the steady-state model and after 5 hours with the transient model).
The difference in simulated long-term performance between both classes of models will depend on different factors: borehole design and flowrate, heat pump ON and OFF time, cycling frequency, ground temperature, etc. The aim of this paper is to quantify the difference in simulation results for a typical residential building in Montréal.

2. LITERATURE REVIEW

2.1 Heat pumps

The impact of ON/OFF cycling on the performance of heat pumps and air conditioners has been studied by many authors, both in commercial and residential buildings. Early studies adopted the "part-load factor" approach to account for the performance degradation during part-load operation. The part-load factor (PLF) is a correction coefficient applied to the steady-state performance of the machine ($0 \leq PLF \leq 1$). The concept of such a correction was later adopted in the ASHRAE standard concerning the method of testing the seasonal performance of heat pumps (ASHRAE, 1983). It was also implemented in many simulation programs, along with curve fits defining the relationship between the PLF and the part-load ratio (PLR) – see e.g. Henderson et al. (1999).

The PLR approach is very well suited to de-coupled simulations where the building load is calculated first and then passed to a different module calculating the performance of the HVAC plant. Several authors have proposed different models that make it easier to simulate the transient behaviour of both the building and the HVAC plant. MacArthur and Grald (1987) present a very detailed distributed model that allows tracking the refrigerant states in the machine during start-up phases. Such very detailed models are not well suited to integrated simulation tools because they involve a significant computing power and require numerous parameters that are only available for intensively tested machines. Goldschmidt et al. (1980) show that a simple first-order lumped capacitance model can reproduce the transient output of heat pumps and air-conditioners. Mulroy and Didion (1985) have refined the model in adding a second time constant in order to better reproduce the short-term behaviour of the refrigerant in the machine. The "one time constant" model was implemented in TRNSYS by Afjei and Wetter (1997) for an air-source heat pump (in heating mode only).

Another issue arising from ON/OFF cycling of heat pumps in cooling mode (or air conditioners) is the degradation of dehumidification performance. This phenomenon is mostly present in systems where the supply fan is continuously operated and Henderson and Rengarajan (1996) have proposed a model to take the latent performance degradation into account. The latent capacity degradation also occurs if the fan is cycled ON and OFF with the compressor, mainly because of a delay in the dehumidification at start-up. Katipamula and O'Neal (1991) measured the delay in a laboratory experiment and found values between 1.5 and 2 minutes. They also found that the latent capacity can actually be negative at start-up, depending on ON/OFF cycling times.

2.2 Ground heat exchangers

Ground coupled heat pump systems involve heat transfer processes in the ground that can take many years to reach a steady-periodic equilibrium. Because of that long time scale, most GCHP design programs take aggregated monthly building loads as an input and calculate the long-term performance of a borefield (e.g. NRCan, 1998). This procedure is well suited to large commercial systems with many boreholes. For these systems, the monthly maximum load and the yearly imbalance in ground loads are often the most important factors in designing the ground heat exchanger. Integrated building performance simulation tools such as TRNSYS (SEL, 2005) and EnergyPlus (UUIC and LBNL, 2006) are typically used with a time step of one hour and different models were developed to allow such "short time step" simulations (see e.g. Yavuzturk and Spitler, 1999, and Hellström, 1989). These models typically neglect the dynamics in the borehole and in the fluid or consider that the grout is part of the ground volume (and has the same properties).

Young (2004) presents a comprehensive review of existing ground heat exchanger models that take into account the transient behaviour of the fluid loop and the grout. Results obtained with classical models and a dynamic model are compared in hourly simulations of a peak-dominated building. It is shown that the dynamic model introduces a damping of the spikes in ground return temperature. Relatively large temperature differences are obtained (1.3°C) but the yearly energy performance is correctly estimated by the steady-state model.

A dynamic model was implemented in TRNSYS by Wetter and Huber (1997). The model only takes one borehole into account and it assumes a configuration with 2 U-pipes per borehole. It is not as widely used as the steady-state model known as the "Lund-DST" model ("Type 557", TESS, 2005). The latter is less computationally intensive and easier to set up, and most importantly it allows for multiple boreholes to be simulated.

2.3 Complete systems

The study by Young (2004) compares performance results for whole building simulations for different large, non-residential buildings. Small-scale residential applications such as single-family houses often present a marked ON/OFF behaviour and complex interactions between the system and controls (thermostat). Henderson (1992) presents a simulation study on such a system for a conventional (air-source) heat pump in cooling mode. The obtained results show the need to have a detailed model of the building (including all thermal capacitance effects), the thermostat and the heat pump in order to be able to quantify the cooling and dehumidification performance accurately.

3. SIMULATION CASE STUDY

3.1 Building and location

The case study is a typical single-family home for the Province of Québec, Canada. Architectural characteristics of the house are set according to "archetypes" that were developed to match statistical data for the existing housing stock. The selected archetypes (Gusdorf, 2001) were developed by National Resources Canada and implemented in the Residential Energy & Economic Simulator (REES). All houses have a conventional wood-frame structure and a heated basement. Houses are assumed to have a square floor plan and windows are equally distributed on the four external walls.

The thermal performance characteristics are selected to represent new houses built according to the R-2000 standard. The 90-percentile value from R-2000 houses built in 1997 is used for thermal insulation and air infiltration (Hamlin and Gusdorf, 1997). Internal gains are typical of a family of 4 (2 adults and 2 children) with a statistical presence of 50%. Standard weather data from the CWEC database (Numerical Logics. 1999) is used and the undisturbed ground temperature was obtained from the GS2000 software (NRCan, 1998). Table 1 lists the most important characteristics of the house and climate conditions selected for the case study. The heating load is about 6.5 MWh/y and the cooling load is about 4 MWh/y (the exact load depends on the controller behavior).

Table 1: Selected case study (house and climate)

House characteristics	
Heated area [m²]	215
Conditioned volume [m³]	525
Glazing fraction [%]	9.5
Thermal performance	
R ceiling [m²-K/W]	7.67
R external walls [m²-K/W]	6.02
R basement walls above gnd [m²-K/W]	4.67
R basement walls [m²-K/W]	4.30
Windows U-value* [W/m²-K]	1.90
Air change at 50 Pa [h^{-1}]	1.22
Gains, setpoints and schedules	
Sensible internal gains (occupants) [kWh/day]	2.4
Sensible nternal gains (other) [kWh/day]	20.0
Moisture gains (occupants) [kg/day]	2.4
Moisture gains (other) [kg/day]	3.5
Heating, day (7-23) [°C]	21
Heating, night (23-7) [°C]	18
Cooling (at all times) [°C]	25
Heating and cooling degree-days	
Heating DD, 18°C baseline [°C-day]	4694
Cooling DD, 10°C baseline [°C-day]	1222
*Windows are modelled using a detailed model that calculates the U-value depending on the conditions. Listed U-values are an approximation in nominal conditions and include the frame	

3.2 HVAC system

A ground-coupled water-to-air heat pump is used to heat and cool the house. Conditioned air is delivered to all the zones in the house and return air is fully re-circulated through the heat pump. Backup heat is provided by electric resistances located at the exhaust of the heat pump. Two nominal capacities are compared in the study, resp. 1 and 1.5 ton (resp. 3.5 and 5.3 kW of cooling). Performance data from an actual 1.5-ton machine is used with an appropriate scaling factor (ClimateMaster, 2006). Air and water (brine) flowrates are set according to the manufacturer's recommendation for maximum performance.

The heat pump is controlled by a thermostat in the main living space and the temperature in other zones (e.g. basement) can be quite different from the setpoint temperature. The two-stage thermostat is assumed to be especially designed for heat pumps: in heating mode, backup electric resistances are automatically controlled using a setpoint which is 0.5°C lower than the heat pump setpoint. During recovery from night setback, the setpoint is smoothly "ramped up" to avoid switching auxiliary heat ON unnecessarily. For the sake of simplicity, the thermostat allows for a constant recovery period of 3 hours (i.e. a recovery slope of 1°C/h).

Fresh air is provided by a dedicated Energy Recovery Ventilator (ERV) providing a constant air supply of 0.3 vol/h. The supply and exhaust fans are operated continuously. Performance characteristics were selected to match a typical residential unit using a permeable membrane plate exchanger: total fan power = 100 W, rated effectiveness = 70% sensible and 50% latent.

Table 2 lists some key parameters of the HVAC system. Finally, it should be noted that domestic hot water is assumed to be provided by an independent system and is not considered in this study.

Table 2: Key parameters of the HVAC system

Outside air ventilation (ERV)		
Flowrate [l/s]		44.6
Fan power (total for 2 fans) [W]		100
Rated sensible effectiveness [%]		70
Rated latent effectiveness [%]		50
Heat pump		
	1-ton	**1.5-ton**
Rated cooling capacity [kW]	3.63	5.45
Rated heating capacity [kW]	2.64	3.96
Air flowrate [l/s]	220	330
Water (brine) flowrate [l/s]	0.23	0.35
Backup electric resistance power [kW]	3.3	5.0

3.3 Ground heat exchanger

Physical properties for the ground and borehole filling (grout) are listed in Table 3. All systems use a single borehole equipped with 2 U-pipes. The pipes are assumed to be maintained against the wall of the borehole (e.g. by stretchers) and their internal diameter is adjusted to maintain the fluid velocity within recommended limits (0.6 to 1.2 m/s, NRCan,

1998) according to the rated flowrate of the heat pump. The depth of the borehole is a parameter of the study and is varied between 40 and 160 m.

Table 3: Ground and borehole thermal properties

Borehole	
Borehole diameter [m]	0.1524
Borehole length [m]	Variable [40;160]
U-pipes dimensions	Variable (see text)
Grout (fill) thermal conductivity [W/m-K]	0.73
Grout (fill) thermal capacity [kJ/m^3-K]	3900
Ground	
Ground thermal conductivity [W/m-K]	2.0
Ground thermal capacity [kJ/m^3-K]	2300
Ground thermal diffusivity [m²/day]	0.075
Yearly average surface temperature [°C]	6.4
Surface temperature amplitude [°C]	15.1
Geothermal gradient [°C/m]	0.018

4. MODELLING ASSUMPTIONS

The system is modelled in TRNSYS (Klein et al., 2004). The next sections provide some details on the existing models that were selected and on the new components that were developed.

4.1 Building

The building is simulated using TRNSYS Type 56, the standard multi-zone model included with the software. The component takes into account the thermal capacitance of all building elements and calculates the internal temperature and humidity response to external conditions and to the HVAC system. The building is modelled using 3 zones: basement, floors 1 and 2 (together), and unconditioned attic. The thermostat is assumed to respond to the air temperature in floors 1 and 2. The basement is conditioned but does not have a separate thermostat.

4.1.2 Basement heat losses

Type 56 itself does not include a detailed model for ground coupling. While using the undisturbed ground temperature with an additional thermal resistance can give acceptable results for some cases, we found that the influence of basement heat losses in a well-insulated R-2000 house was very significant and justified the use of a more detailed model. The 3-D ground model included in the TESS libraries ("Type 701", TESS, 2005) was used and coupled to Type 56.

4.1.3 Infiltration

Again, Type 56 does not calculate air infiltration rates internally, unless a complex multi-zone infiltration network is defined and an integrated coupling with COMIS is used. Simple solutions have been found to provide acceptable results for yearly energy use, e.g. using a constant infiltration rate equal to 1/20th of the measured infiltration at 50 Pa. In the case of well insulated houses, it is however advisable to use a more detailed model. A new TRNSYS component was developed to implement the AIM-2 model (ASHRAE, 2005), which has been extensively validated (see e.g. Walker and Wilson, 1998). The building is treated as one volume and the infiltration rate is calculated based on the wind speed, ambient temperature and building temperature. The calculated infiltration rate is provided as an input to the building model (Type 56).

4.2 Heat pump

4.2.1 Steady-state model
We used the geothermal heat pump model included in the TESS libraries ("Type 504", TESS, 2005). The component does not include a physical model; it interpolates the performance of the water-to-air heat pump from the performance map provided by manufacturers. This is the approach retained by most of existing building performance simulation tools. The model includes a backup electric heater. Some control logic to prevent operation outside of the intended range was added.

4.2.2 Dynamic model
A new TRNSYS component was developed by the authors to combine the steady-state performance interpolation typically used in simulation tools with a simple "one time constant" model to account for dynamics. The model interpolates a performance map to calculate the steady-state performance. A simple time constant profile is then applied to correct the steady-state performance. In the absence of manufacturer data, a time constant of one minute and a dehumidification delay of one minute were assumed (Henderson, 1992). The model also accounts for minimum ON and OFF times and high- and low-temperature protection according to manufacturer recommendations. The latent capacity is assumed to have the same profile as the sensible capacity with a time delay, i.e. a possible negative capacity (humidification) at start-up is not considered. The fan is cycled ON and OFF with the compressor.

4.3 Mechanical ventilation

The ERV unit is modelled using a steady-state constant effectiveness model (the flowrate through the ERV is assumed to be constant). Frost control is obtained by preheating the supply air and has a marginal effect on the energy use for ventilation.

4.4 Borehole and ground storage

4.4.1 Steady-state model
The "steady-state" model is the "Duct Ground Heat Storage" (DST) model developed at the university of Lund (Hellström, 1989), which is available in TRNSYS ("Type 557", TESS, 2005). The DST model calculates the ground temperature by spatially superposing the solutions to three sub-problems: the "global" heat transfer between the storage volume as a whole and the far-field, the "local" heat transfer occurring around the boreholes at a short time scale, and a "local steady-flux" heat transfer around the nearest pipe. The model uses numerical solutions for the "global" and "local" problems and an analytical solution for the "steady-flux" problem.

The DST model has been extensively validated and has become the reference ground storage model in TRNSYS. Its main limitation is that it was designed for densely packed ground heat exchangers for heat storage applications and it cannot cope with low-density, arbitrary patterns that are sometimes used in multiple-boreholes geothermal heat pump systems. This limitation is not a concern for the present paper since we are considering a single borehole. Another simplification of the model is that the thermal resistance between the fluid and the ground is assumed to be constant (see next section).

4.4.2 Dynamic model

The dynamic borehole model has been implemented in TRNSYS by Wetter and Huber (1997) and is known as "Type 451". The transient heat transfer in the borehole filling (grout) and in the ground around the borehole is solved using a finite difference method. The boundary conditions outside the detailed simulation volume are calculated using an analytical solution derived from the line source theory. The volume of heat transfer fluid is discretized in a number of nodes for which mass and energy balances are solved. The model calculates various equivalent thermal resistances between the fluid, grout and ground nodes. It is similar to the steady-state model in that the so-called "borehole resistance" (resistance between the fluid and the outside of the borehole) is only calculated once at the beginning of the simulation. This implies that design flowrate and fluid properties are used to calculate the equivalent resistances, instead of the actual (possibly time-varying) conditions. The main limitations of Type 451, in addition to the simplification that was just mentioned, are the impossibility to simulate several interacting boreholes and the fact that it automatically assumes a configuration with 2 U-pipes per borehole.

4.5 Thermostat

The thermostat plays a very important role in the studied system: the ON/OFF cycling frequency will have a significant impact on the heat pump performance and on the observed differences between the steady-state and dynamic models. The most restrictive assumption in this respect does not come from the thermostat model but from the building model: each zone is modelled by a fully-mixed air volume and the thermostat will react to the air temperature of that volume. The validity of simulated cycling times will be discussed in the results section.

The modelled thermostat is a conventional two-stage ON/OFF controller with hysteresis. The thermostat is assumed to have a 1°C dead band centered on the setpoints listed in Table 1, i.e. during the day heating is switched ON at 20.5°C and switched OFF at 21.5°C. The backup electric resistances (stage 2 heating) uses a setpoint that is 0.5°C lower than the first stage setpoint and a 1°C dead band (i.e. during the day it is switched ON at 20.0°C and OFF at 21°C).

The thermostat generates a smooth increase in the setpoint during recovery from night setback in order to avoid using backup heat unnecessarily. The ramp starts at 4 AM and ends at 7 AM. The setpoint for backup heating follows the same pattern so that it can be used if the heat pump is not able to achieve the increase in air temperature that is required to meet comfort conditions at the beginning of the "day" period. Figure 4 shows the setpoint profile (note that each system is switched ON when the air temperature falls 0.5°C lower than the respective setpoint and switched OFF when that setpoint is exceeded by 0.5°C).

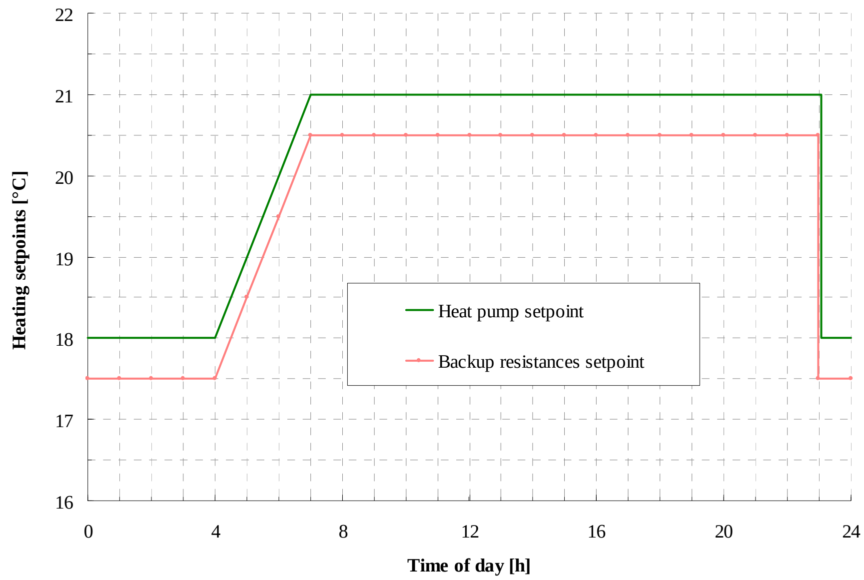

Figure 4: Heating setpoint profile

The setpoint profile for cooling is much simpler since a constant temperature is used and no backup cooling is available. The same dead band is used for cooling as for heating (0.5°C).

The heat pump fan is controlled with the compressor, i.e. it is switched OFF when neither cooling nor heating are requested. The geothermal fluid pump is controlled with the compressor as well, with the following exception: if the heat pump has stopped operating due to the ground temperature exceeding the acceptable range (too cold in heating, too warm in cooling), the circulating pump is left ON until the warning condition disappears.

4.6 Simulation time step and length

All simulations are carried on with a 3-min simulation time step. The steady-state models described here above are typically used with longer time steps (from one hour down to 15-min) but the ON/OFF nature of the controller and the heat pump model would cause large oscillations in the building temperature if long time step were used. This problem is frequently encountered in hourly simulations using TRNSYS Types 557 (DST model) and Type 504 (Heat pump). It is typically solved by either accepting larger dead bands or reducing the time step. We decided to use a consistent time step with all the models in order to isolate the effect of modelling assumptions for the heat pump and borehole from other effects that might be caused by larger time steps.

The simulated systems all have a single borehole and a small yearly imbalance in ground loads. This results in a relatively small year-to-year performance difference, so only 5 years of operation were simulated. The results of the 5^{th} year are presented in all cases.

5. RESULTS

5.1 Thermostat action and system behaviour

This section discusses simulation results obtained with dynamic models. The comparison with results from steady-state models is covered in the next sections.

Figure 5 shows two typical cold winter days with the 1.5-ton machine and a 120-m borehole (note that $\dot{Q}_{heat,tot}$ is equal to $\dot{Q}_{heat,hp}$ for all but 2 time steps). The ON/OFF action of the thermostat can clearly be seen, with an average cycling period of one hour during the day (30 min ON, 30 min OFF). This cycling time is longer than what is reported in (Manning et al., 2005) for a similarly sized R-2000 house equipped with a furnace. Several factors can explain the difference, such as the heating power, infiltration rate and the design of the ventilation system, or the location of the thermostat within the house. Modelling assumptions such as the coarse thermal zoning used in the building thermal model can also explain part of the differences. Further studies should investigate the effect of a refined zoning on the simulation results.

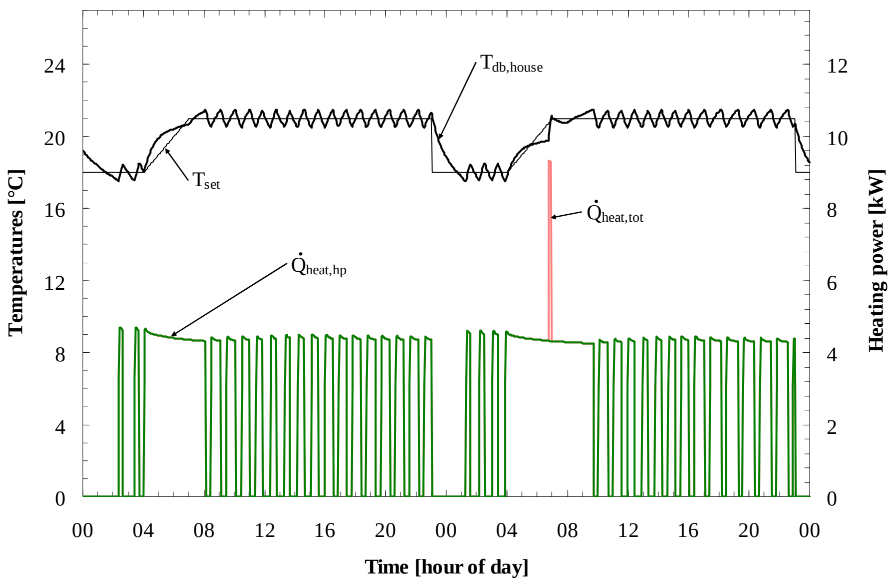

Figure 5: Typical cold winter days – 1.5-ton heat pump, 120-m borehole

The thermostat operation in recovery mode can also be seen in Figure 5. On the first day, the fixed 3-h preheat time implemented in the thermostat is long enough and the heat pump is able to bring the temperature back to 21°C at 7 AM. The second day is colder (-15°C at 5 AM versus -8°C for the first day) and the preheat time is insufficient. When the temperature falls 1°C lower than the setpoint, which occurs right before 7 AM, auxiliary heat is switched ON for a very short period of time.

Figure 6 shows the same cold days with a 1-ton heat pump and a 55-m borehole. Backup heat is switched ON much more often, because the heat pump is neither able to maintain the setpoint during the day nor to achieve the recovery slope expected by the thermostat. Furthermore, the borehole is undersized and the heat pump is switched OFF very often due to the ground return temperature reaching the lowest acceptable limit (-1.1°C). The machine is typically switched OFF for a few time steps (6-12 minutes), until the ground "recovers", and then ON again for a similar period of time. Auxiliary heat is used to maintain the room temperature at the desired level, either supplementing or replacing the heat pump.

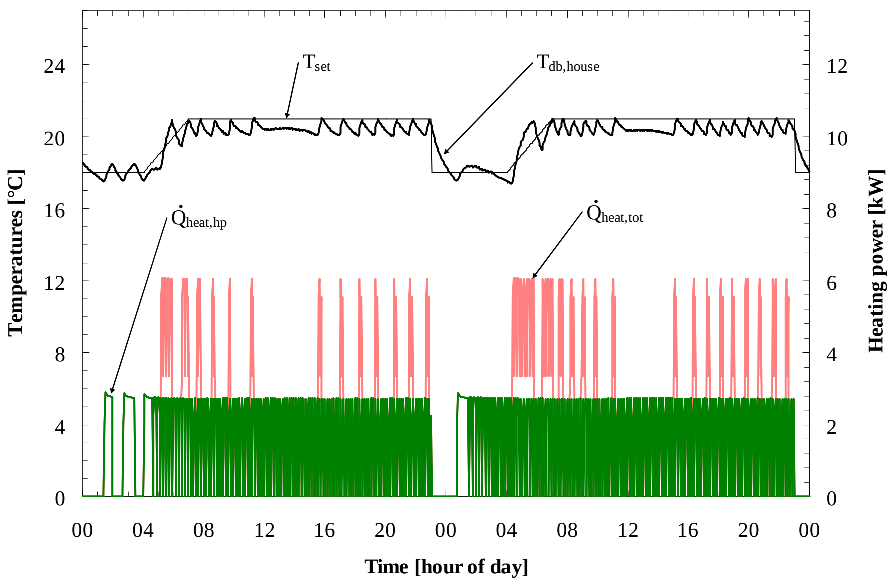

Figure 6: Typical cold winter days – 1.0-ton heat pump, 55-m borehole

The system behaviour on 3 consecutive hot summer days is illustrated in Figure 7 and Figure 8. With a 1.5-ton heat pump, the cooling capacity is sufficient and the system maintains the desired setpoint by cycling ON and OFF with typical ON times ranging from 15 to 45 minutes.

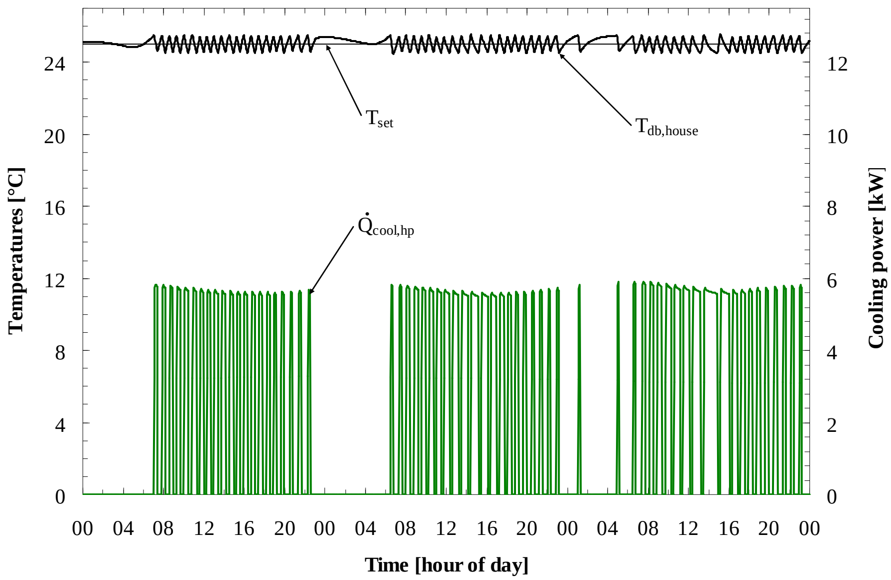

Figure 7: Typical hot summer days – 1.5-ton heat pump, 120-m borehole

The 1-ton heat pump is able to maintain the temperature within the desired range for the first two days but on the third day the temperature reaches 26°C while the heat pump operates continuously for most of the day. On the other hand, the undersized borehole does not affect the performance (the ground return temperature reaches a maximum of 34°C over the summer, while the upper limit of the heat pump operating range is 49°C).

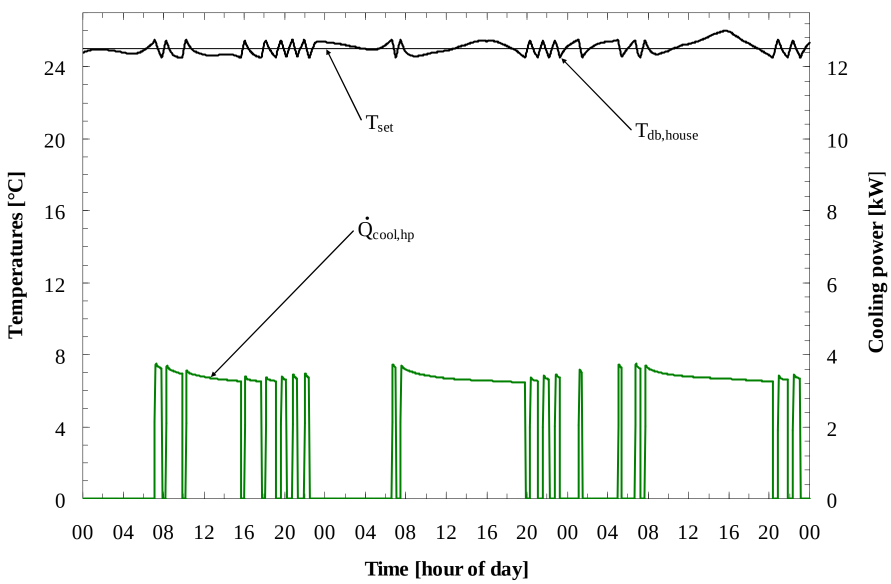

Figure 8: Typical hot summer days – 1.0-ton heat pump, 55-m borehole

5.2 Yearly performance

This section discusses the differences in simulation results when dynamic and steady-state models are used for the borehole and heat pump.

Table 4 presents the global performance for 8 systems (2 heat pump nominal capacities with 4 different borehole lengths). The first 3 columns provide the heat pump capacity and borehole length that were selected and the models that were used in each case. The next columns give the electrical energy used for cooling ($Q_{el,cool}$) and heating ($Q_{el,heat}$) and the respective COP (COP_{cool} and COP_{heat}). The cooling and heating load can be obtained by multiplying the corresponding Q_{el} and COP values. The loads vary slightly from case to case due to different controller oscillations and in some cases to insufficient capacity. Column 8 (Q_{el}) provides the total electricity used for heating and cooling and the last column is an overall COP calculated by dividing the net energy delivered to (in heating) or removed from (in cooling) the building by the total electricity use.

Table 4: yearly energy performance comparison

Heat pump	L_{Bore} [m]	Models	$Q_{el,cool}$ [kWh/y]	COP_{cool} [-]	$Q_{el,heat}$ [kWh/y]	COP_{heat} [-]	Q_{el} [kWh/y]	COP [-]
1-ton	40	Trn	1040	3.82	3170	1.88	4210	2.36
		SSt	1180	3.31	6290	1.02	7470	1.38
1-ton	55	Trn	850	4.79	2590	2.38	3430	2.98
		SSt	940	4.30	3130	1.87	4070	2.43
1-ton	80	Trn	710	5.82	2050	3.19	2770	3.85
		SSt	760	5.43	2090	3.10	2850	3.72
1-ton	110	Trn	650	6.40	1920	3.44	2580	4.17
		SSt	680	6.12	1940	3.40	2620	4.10
1.5-ton	60	Trn	850	4.75	2560	2.50	3410	3.06
		SSt	1020	3.91	3940	1.45	4960	1.95
1.5-ton	80	Trn	740	5.51	2140	3.11	2890	3.71
		SSt	840	4.84	2540	2.48	3380	3.07
1.5-ton	120	Trn	670	6.14	1860	3.69	2530	4.33
		SSt	700	5.90	1880	3.65	2580	4.26
1.5-ton	160	Trn	640	6.45	1790	3.84	2420	4.54
		SSt	650	6.39	1790	3.84	2440	4.52

When the borehole is long enough (≥80m for 1-ton and ≥120m for 1.5-ton), the simulated yearly electricity use for space conditioning (Q_{el}) are within 3 percents from each other and the difference can be considered as insignificant. However, the difference becomes very large for much shorter boreholes, reaching 75% for the 1-ton machine with a 40-m borehole and 45% for the 1.5-ton machine with a 60-m borehole. Differences are as large as 20% for intermediate cases within the range usually selected by practitioners (1.5-ton/80-m and 1-ton/55-m, i.e. about 180ft/ton or 16m/kW).

The largest discrepancies are observed in heating mode, which is explained by the behaviour illustrated in Figure 6: for shorter boreholes, the performance of the system is very sensitive to the ground return temperature when the latter is close to the lower limit of the heat pump operating range. Figure 2 shows that for typical cycling frequencies, the ground return temperature calculated by the transient model can be 4 or 5°C higher than the value calculated by the steady-state model. In extreme cases, such as the 1-ton/40-m case, the steady-state model predicts a yearly heating COP of 1.02, which means that the heat pump is almost never used to provide heating (electric resistances alone would have a COP of 1).

The influence of the heat pump dynamics on the results in Table 4 has been assessed by running the same simulations with the dynamic model of the borehole but a steady-state heat pump model. As expected, the performance of the heat pump increases slightly but the difference in yearly energy use is under 3% in all cases. The dehumidification load is also a few percents higher (<5% in all cases). The bulk of the difference between the steady-state and transient results can thus be attributed to borehole dynamics.

5.3 Optimal design

This section presents a simple Life Cycle Cost analysis of the simulated systems in order to assess the impact of the differences described above on the design process. The values selected for economic parameters and costs only serve to illustrate the case study and should be reviewed before being applied to other analyses. All costs are in Canadian dollars.

The Life Cycle Cost (LCC), or "net present worth", of a system is defined as (Duffie and Beckman, 1991):

$$\mathrm{LCC} = \mathrm{C_I} + \mathrm{C_{f,1}}\ \mathrm{PWF} \tag{1}$$

Where C_I is the initial cost, $C_{f,1}$ is the fuel cost for the first year and PWF is the present worth factor:

$$\mathrm{PWF} = \frac{1}{(\mathrm{d}-\mathrm{i})}\left[1-\left(\frac{1+\mathrm{i}}{1+\mathrm{d}}\right)^{\mathrm{N_y}}\right] \tag{2}$$

Where d is the market discount rate and i is the inflation rate for energy costs (N_Y is the number of years in the analysis). The selected parameters in this analysis are N_Y=25 years, d=0.07 and i=0.05, i.e. PWF = 18.8. The energy cost is taken as 0.06$/kWh.

The heat pump and borehole are the only parts of the system for which initial costs are considered. A cost of $2750 was assumed for the 1-ton heat pump and $4000 for the 1.5-ton machine. The borehole cost was estimated at $60 per m for deep wells, increasing to $100 per m for shorter wells. Table 5 presents the results of the LCC analysis using the energy use (Q_{el} in Table 4) in calculated with both the dynamic ("Trn") and steady-state ("SSt") models.

The results in Table 5 show that the discrepancies in calculated performance can have a significant impact on the optimal design. Keeping in mind the simplifications made in the LCC analysis, the steady-state models would lead to select a borehole depth of 80m with the 1-ton heat pump, while the dynamic models would lead to select a much smaller depth (40m). The analysis based on fully dynamic models also shows a small difference in LCC for the first 3 borehole depths with both heat pumps, while the results obtained with steady-state models show a much wider variation (up to 25%).

Table 5: Life-cycle cost comparison

System		Initial cost C_I [$]			Energy cost $C_{f,1}$ [$]		LCC [$]	
Heat pump	L_{Bore} [m]	Heat pump	Borehole	Total	Trn	SSt	Trn	SSt
1-ton	40	2750	4000	6750	250	450	11500	15200
1-ton	55	2750	5000	7750	205	245	11600	12300
1-ton	80	2750	6000	8750	165	170	11900	12000
1-ton	110	2750	7000	9750	155	155	12700	12700
1.5-ton	60	4000	5500	9500	205	300	13300	15100
1.5-ton	80	4000	6000	10000	175	205	13300	13800
1.5-ton	120	4000	7500	11500	150	155	14300	14400
1.5-ton	160	4000	9500	13500	145	145	16200	16200

5.4 Other benefits of using dynamic models

Using dynamic models brings other benefits than a more accurate yearly performance and design selection. Figure 9 shows the air temperature in the house and in the basement for two winter days, with both dynamic and steady-state models (1-ton heat pump, 40-m borehole). The dynamic models predict a moderate ground return temperature which allows the heat pump to operate for most of the time, in a way similar to what happens in Figure 5. The steady-state models, on the other hand, predict that the ground return temperature is too low for the heat pump to operate, which leads to a different operation pattern similar to the one in Figure 6: the heat pump cycles ON and OFF with a shorter cycling frequency corresponding to the "recovery time" for the ground and is supplemented or replaced by auxiliary heat.

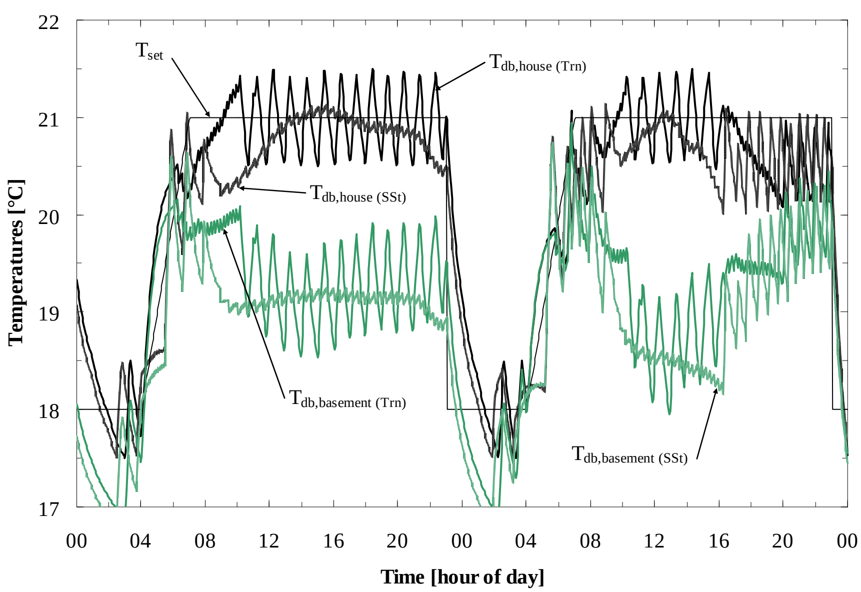

Figure 9: winter days – 1.0-ton heat pump, 40-m borehole – Dynamic and Steady-state models

The different modes of operation lead to different profiles of temperature in the house and in the basement. Even though the differences are relatively small, they could be significant in a thermal comfort study or if the controller action was studied in detail.

One example of controller-related difference between the two classes of models is the average temperature variation with different designs. Since the auxiliary heat is only turned ON when the temperature reaches 20.0°C (heat pump is turned ON at 20.5°C), the average house temperature will slightly decrease when the heat pump is operated less often (e.g. with an undersized ground heat exchanger). This will in turn result in a slight decrease in heating load. The magnitude of this phenomenon depends on the models that are used.

Table 6 shows the average house and basement temperatures for the two extreme borehole lengths with the 1-ton machine.

Table 6: average house and basement temperatures during the heating season

System		$T_{db,house}$ (average over heating season)		$T_{db,house}$ (average during day only)	
Heat pump	L_{Bore} [m]	Trn	SSt	Trn	SSt
1-ton	40	20.4	20.0	21.1	20.7
1-ton	110	20.6	20.6	21.3	21.3

System		$T_{db,basement}$ (average over heating season)		$T_{db,basement}$ (average during day only)	
Heat pump	L_{Bore} [m]	Trn	SSt	Trn	SSt
1-ton	40	18.3	17.8	18.5	18.0
1-ton	110	18.4	18.4	18.6	18.6

The steady-state models overestimate the change in temperature and calculate a lower heating load for shorter boreholes. The average temperature in the house, for example, decreases from 20.6 to 20.4°C with the dynamic models, but it decrease from 20.6 to 20.0°C with the steady-state models.

The assessment of control strategies may also require dynamic models. A good example is the behaviour of the heat pump when the ground return temperature reaches the lower acceptable limit for operation: the heat pump is switched OFF and the circulating pump is left ON until the ground return temperature reaches a higher value again. It is virtually impossible to simulate such a control strategy with the steady-state borehole model since the fluid temperature will oscillate between a very low and a very high value in one time step, independently of the length of that time step. Users traditionally cope with similar problems by artificially introducing or increasing controller dead bands or ignoring the control logic altogether, but having dynamic models for all components allow to simulate and possibly fine-tune such control strategies.

Finally, iterative solvers such as the one implemented in TRNSYS are known to have convergence issues in the presence of recyclic information loops without capacitance effects. This was confirmed in our simulations: in some cases a strong increase in the number of iterations would cancel the speed benefit that should have been obtained by switching from detailed, dynamic models to less computationally intensive steady-state models. For that reason, it is sometimes interesting to add simple dynamics to components, such as in the "one time constant" approach for the heat pump.

6. DISCUSSION AND CONCLUSIONS

This paper presents a simulation study of GCHP systems for a typical residential building in the Province of Québec. The aim of the study was to compare the results obtained with classical models used in TRNSYS to models that take dynamics into account in the heat pump, in the geothermal fluid loop and within the borehole. Classical TRNSYS models (SEL, 2004; and TESS, 2005) and models implemented in other integrated simulation tools typically neglect those dynamics.

The results show that dynamics in the heat pump itself have a very small impact on the simulated performance, with a slight increase in energy use and decrease in dehumidification load when dynamic models are used. Dynamics in the fluid loop and the borehole can have a much larger impact if the borehole depth is at the lower end of the range typically considered for GCHP systems. Steady-state models can lead to overestimating the energy use by as much as 75% in extreme cases, because they predict quick temperature drops in the ground return temperature that prevents the heat pump from operating in heating mode.

Adding dynamics to the heat pump, the fluid loop and the borehole has additional benefits. First, the temperature and humidity in the different zones of the building are simulated more accurately: the average temperature drop with an increased use of auxiliary heating can be quantified, and comfort can be assessed more accurately. The thermostat operation can also be optimized. Other control strategies can also be simulated and optimized, such as the heat pump operation when the ground return temperature is close to the lower limit of the operation range. Finally, adding dynamics to components promotes convergence with iterative solvers such as the one implemented in TRNSYS.

Several simplifications were made in this study and further work is required to confirm the findings reported in this paper using, among others, a finer zoning of the building model. A better handling of dehumidification capacity at startup would also allow to study the impact of dynamics on summer humidity levels in more details.

NOMENCLATURE

Variable	Units	Description
C_I	[$]	Investment cost
$C_{F,i}$	[$]	Fuel cost for year i
COP	[-]	Equipment global coefficient of performance (including electric resistances)
COP_{cool}	[-]	Equipment COP for cooling only
COP_{heat}	[-]	Equipment COP for heating only (heat pump + electric resistances)
d	[-]	Market discount rate (on a yearly basis)
i	[-]	Inflation rate (on a yearly basis)
L_{Bore}	[m]	Borehole depth
LCC	[$]	Life Cycle Cost
LCS	[$]	Life Cycle Savings
N_Y	[-]	Number of years in the LCS analysis
PWF	[-]	Present Worth Factor
Q_{cool}	[kWh]	Cooling load
$Q_{el,cool}$	[kWh]	Electricity use for cooling
Q_{heat}	[kWh]	Heating load
$Q_{el,heat}$	[kWh]	Electricity use for heating (heat pump + resistances)
Q_{load}	[kWh]	Space conditioning load
Q_{el}	[kWh]	Electricity use for space conditioning
$\dot{Q}_{cool,hp}$	[kW]	Useful cooling power from the heat pump evaporator (fan heat gain is included)
$\dot{Q}_{heat,hp}$	[kW]	Useful heating power from the heat pump condenser and fan
$\dot{Q}_{heat,tot}$	[kW]	Total heating power ($\dot{Q}_{heat,hp}$ + heat from backup electrical resistances)
$RH_{Basement}$	[%]	Percent relative humidity in the basement (all floors above ground)
RH_{House}	[%]	Percent relative humidity in the house (all floors above ground)
SSt	-	Shorthand for steady-state models
Trn	-	Shorthand for Transient (or dynamic) models
$T_{dbBasement}$	[°C]	Dry bulb (air) temperature in the basement
$T_{dbHouse}$	[°C]	Dry bulb (air) temperature in the house (all floors above ground)
T_{Set}	[°C]	Setpoint temperature for heating or cooling

REFERENCES

Afjei, T. and Wetter, M., 1997, TRNSYS Type: Compressor heat pump including frost and cycle losses, Version 1.1 - Model description and implementing into TRNSYS, Transsolar GmbH, Stuttgart, Germany.

ASHRAE, 1983, Standard 116-1983, Methods of testing seasonal efficiency of unitary air-conditioners and heatpumps. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc, Atlanta.

ClimateMaster, 2006, Engineering Design Guide: Tranquility 20™ single-stage (TS) series 1,4 to 6 ton horizontal and vertical earthpure™ units, p. 20 (Performance Data TS 018, ECM Motor).

Duffie, J.A. and Beckman, W.A., 1991 Solar Engineering of Thermal Processes, Wiley Interscience, NYC, NY, USA, second edition, ISBN 0-471-51056-4.

Eskilson, P., 1987. Thermal Analysis of Heat Extraction Boreholes. Ph.D. Thesis, Department of Mathematical Physics, University of Lund, Lund, Sweden.

Goldschmidt, V.W., Hart, G.H. and Reiner, R.C., 1980, A note on the degradation coefficient of a field tested heat pump-cooling and heating mode. ASHRAE Transactions, 86 (2), pp.368-375.

Gusdorf, J., 2001, EKO^10 Base Cases & Benchmarks (DRAFT). Natural Resources Canada - CANMET - Buildings Group.

Hamlin, T. and Gusdorf, J., 1997. Air tightness and energy efficiency of new conventional and R-2000 housing in Canada, 1997. Natural Resources Canada, CANMET - Buildings Group.

Hellström, G., 1989, Duct Ground Heat Storage Model, Manual for Computer Code, Department of Mathematical Physics, University of Lund, Sweden.

Henderson, H.I., 1992, Simulating combined thermostat, air conditioner, and building performance in a house, ASHRAE Transactions 98 (1), pp. 370-380.

Henderson, H.I. and Rengarajan, K., 1996, Model to predict the latent capacity of air conditioners and heat pumps at part-load conditions with constant fan operation, ASHRAE Transactions 102(1), 266-274.

Henderson, H., Huang, Y.J. and Parker, D., 1999, Residential equipment part load curves for use in DOE-2, Technical report LBNL-42175, Lawrence Berkeley National Laboratory, Berkeley, CA, USA.

Katipamula, S. and O'Neal, D.L., 1991, Performance degradation during on-off cycling of single-speed air conditioners and heat pumps: experimental results, ASHRAE Transactions 97(2), 331-339.

Klein, S. A., W.A. Beckman, J.W. Mitchell, J.A. Duffie, N.A. Duffie, T.L. Freeman, J.C. Mitchell, J.E. Braun, B.L. Evans, J.P. Kummer, R.E. Urban, A. Fiksel, J.W. Thornton, N.J. Blair, P.M. Williams, D.E. Bradley, T.P. McDowell, M. Kummert. 2004. TRNSYS 16 – A TRaNsient SYstem Simulation program, User manual. Solar Energy Laboratory, University of Wisconsin-Madison.

MacArthur, J.W., and Grald, E.W., 1987. Prediction of cyclic heat pump performance with a fully developed model and a comparison with experimental data. ASHRAE Transactions, 93 (2), pp. 1159-1172.

Manning, M.M., Swinton, M.C., Szadkowski, F., Gusdorf, J. and Ruest, K.., 2005, The Effects of Thermostat Setting on Seasonal Energy Consumption at the CCHT Research Facility (RR-191), Technical report, Institute for Research in Construction, National Research Council Canada, Ottawa, ON, Canada.

NRCan, 1998, GS2000 – Heat Exchanger Design Program, Version 2.0c. Natural Resources Canada, CANMET Technology Centre, Buildings Group, Ottawa, ON, Canada.

NRCan-OEE, 2004, Heating and cooling with a heat pump. Ground Source heat pumps, pp. 24-42. Natural Resources Canada, Office for Energy Efficiency, Ottawa, ON, Canada

Numerical Logics. 1999. Canadian Weather for Energy Calculations, Users Manual and CD-ROM. Environment Canada, Ottawa.

O'Neal, D. and Katipamula, S., 1991, Performance degradation during on-off cycling of single-speed air conditioners and heat pumps: model development and analysis, ASHRAE Transactions 97(2), 316-323.

SEL, 2005. TRNSYS 16 – A Transient System Simulation Program – Documentation set (9 Volumes). Version 16.00.0038. Solar Energy Laboratory, University of Wisconsin-Madison. Madison, WI. http://sel.me.wisc.edu/trnsys.

TESS, 2005 TESS Libraries Version 2.02, Reference Manuals (13 Volumes). Thermal Energy Systems Specialists, Madison, WI. http://tess-inc.com.

UIUC and LBNL, 2006, EnergyPlus Engineering Reference: The Reference to EnergyPlus Calculations, U.S. Department of Energy.

Wetter, M. and Huber, A., 1997, TRNSYS Type 451: Vertical Borehole Heat Exchanger EWS Model, Version 3.1 - Model description and implementing into TRNSYS, Transsolar GmbH, Stuttgart, Germany.

Yavuzturk, C., J.D. Spitler. 1999. A Short Time Step Response Factor Model for Vertical Ground Loop Heat Exchangers. ASHRAE Transactions. 105(2): 475-485.

Young, T.R., 2004, Development, Verification, and Design Analysis of the Borehole Fluid Thermal Mass Model for Approximating Short Term Borehole Thermal Response, Master's thesis, Oklahoma State University, Building and Environmental Thermal Systems Research Group.