QUANTIFYING THE ENERGY IMPACT OF A VARIABLE FLOW PUMP IN A GROUND-COUPLED HEAT PUMP SYSTEM

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ABSTRACT

This article presents the results of a case study aimed at quantifying the energy impact of replacing a constant-speed pump with a pump driven by a variable frequency drive in a ground-coupled heat pump (GCHP) system. The system is installed in a public high school located in St-Constant near Montréal.
Performance data collected from an on-site energy management system indicates that the annual energy consumption of the heat pumps represented 33% (63700 kWh) of the total energy consumption of the school while the circulating pump consumed 7.1% (13702 kWh). This performance data was used to validate the energy simulations which were performed using TRNSYS 15.
Simulations with variable-flow pumping indicate that pumping energy consumption is reduced by about 82% while the total (circulating pump + heat pumps) energy consumption is reduced by 18.5%.

INTRODUCTION

The 'École du Tournant' high school is an energy-efficient building. It consumes 79.2% less source energy than a typical high school built in accordance with the Model National Energy Code of Canada for Buildings (NRC-CNRC, 1997). In its category it is the most efficient in Quebec and the second in Canada. 'École du Tournant' is a two-storey 2682 m^2 building with classrooms, offices, meeting rooms, a library, a computer laboratory, a kitchen and a master hall that spans over two floors. The ground floor plan is illustrated in Figure 4. Approximately 220 students attend the school. During the evenings, week-ends, official holidays and the summer period there are no activities in the school.

Figure 1: **'École du Tournant' high school.**

During such periods, the mechanical system is out of operation if cooling is required or operates to maintain lower zone air temperature during heating periods. Classrooms are equipped with occupancy and luminosity sensors in order to reduce energy consumption. The school's architecture is designed in such a way so as to maximize natural light penetration. This design contributes to reduce the average lighting power and to lower the heating demand.
Based on utility bills, the total annual energy consumption of the school for 2003 is 192989 kWh. Energy consumption data obtained from the energy management system are shown in Figure 2 (covering the period from September 2003 to January 2004). About a third of the energy is used to heat/cool the building and 7.1% is used by the circulating pump. Compared to the heat pumps energy consumption the circulating pump energy consumption is relatively high. The objective of the present study is to simulate the impact of a variable flow pump on this system in order to reduce even further the energy consumption of the school.

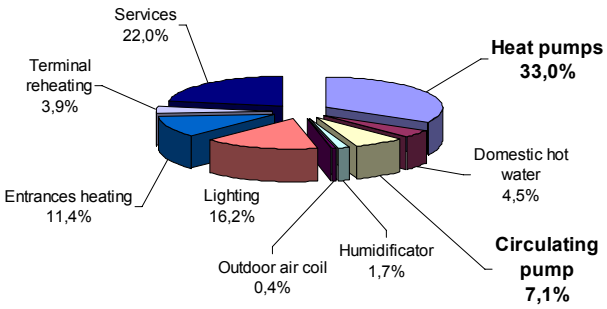

Figure 2: Breakdown of the energy consumption (from September 2003 to January 2004).

SYSTEM DESIGN

Figure 3 presents a schematic of the air handling unit. The outdoor air is pre-heated during daylight hours in winter (a bypass is provided for summer operation) by two solar walls (40 m^2 east-facing and 51 m^2 west-facing). These solar walls are identified by number 1 on Figure 3. Then, the preheated air encounters a sensible heat recovery exchanger, (number 2 on Figure 3), which transfers energy contained in the exhaust air to the outdoor air. After the heat recovery exchanger, the outdoor air is mixed with recirculating air (number 3 on Figure 3).

The correct mixture is controlled using a CO_2 sensor. Finally, a 10-ton 2 stages geothermal heat pump (number 4 on Figure 3) rises the air temperature to 22ºC. This heat pump provides pre-heated outdoor air to 24 local geothermal heat pump units. The locations of the ground floor heat pumps are shown on Figure 4. These heat pumps serve 13 individual thermal zones. They are linked to an internal reverse-return fluid loop.

The heat transfer fluid is composed of a water-methanol (20%) mixture. The loop is connected to a ground heat exchanger which consists of 18 vertical boreholes connected in parallel. A 5 horsepower constant-flow pump with a nominal flow rate of 8.86 L/s is used. This pump is in operation if at least one of the 25 geothermal heat pumps is working. The HVAC system and the lights are controlled by a centralized system which is remotely-accessed from the school district headquarters. There, experienced plant engineers and technicians monitor the system performance and can close on an "ad hoc" basis any of the HVAC equipment. This type of management contributes to the excellent energy performance of the school. However, the behaviour of the operator is difficult to predict and to simulate.

SIMULATION

Simulations are performed using TRNSYS 15 (Klein, 2000) with the IIsiBat interface. TRNSYS is an appropriate tool for transient energy simulations of thermal systems such as the one described in this article which involve "non-standard" systems. Simulations are performed using a one hour time step because some component models (heat pump, thermostat ….) have been built based on an one-hour interval. A brief description of the main components of the simulation model will now be presented.

WEATHER CONDITIONS

Since it was not possible to obtain actual measured weather data for the site it was decided to use the WYEC (Weather Year for Energy Calculations) meteorological data file for Montreal (ASHRAE, 1997).

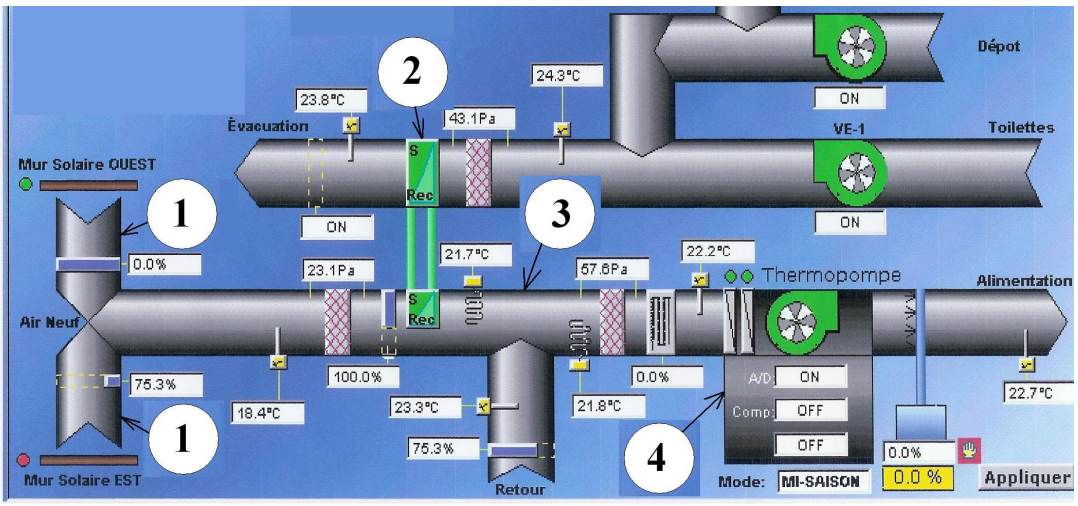

Figure 3: Outdoor air system

BUILDING MODEL

The thermal behaviour of the building is modeled with the TRNSYS multi-zone building component (TYPE 56). In order to use this component, a separate pre-processing program (PREBID), containing the building description, must first be executed. The architectural elements (external and internal walls, fenestration, roof, and slab) are described in details in PREBID using the existing library of materials and windows. The difference between the real thermal resistance of the architectural elements and the ones implemented in PREBID is less than 1% for the walls and roof and about 6% for the windows.

The building model is a non-geometrical balance model with one air node per zone, representing the thermal capacity of the zone. The thermal capacity is introduced in PREBID as an input for each zone. The model accounts for convective and radiative (longwave) heat flow from all inside surfaces, solar gains through windows, air infiltration and ventilation, internal heat gains by occupants, equipment and illumination. The building model used here has 23 different functional zones defined either by orientation, function, level of thermal gains or heat pump location.

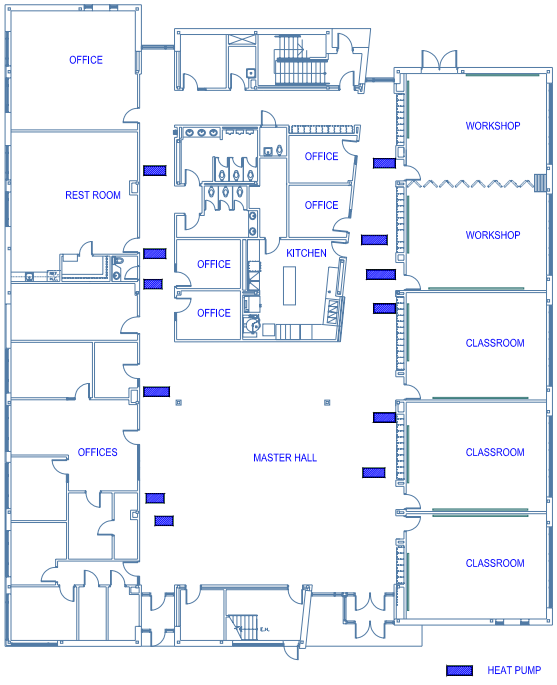

Figure 4: Ground floor plan layout including heat pump location.

The heat gains/losses through the foundation (slab-on-grade) are calculated using the method developed by Mitalas (1987). Air infiltration into the peripheral zones is based on a simple model proposed by ASHRAE (1977). This model evaluates the number of air changes per hour based on the wind speed and the difference between ambient and zone temperatures. Internal heat gains from the occupants and the equipment are determined using normalised parameters for occupancy (Hydro Quebec, 2004).

The average installed lighting power density is 8.7 W/m^2. This is about half the power of a typical high school. Such a low level is possible because natural lighting is maximized and high efficiency lights are used. Each zone has a lighting power density which is different from the average value. The real values, as determined from the engineering plans, are used for each zone.

OCCUPANCY, LIGHTING AND EQUIPMENT SCHEDULE

The occupancy of the school changes daily and seasonally. It is therefore convenient to generate a time-dependent forcing function which repeats cyclically in order to generate daily profiles.

In turn, these different daily patterns form an annual profile. The daily schedule is taken from Hydro-Québec (Hydro-Québec, 2004) and is modified to better match real operating conditions. The resulting daily profiles for the occupancy, lighting and equipment are shown on Figure 5. The annual schedule is created according to the official dates of school vacation. A total of five annual profiles were created depending on the daily profile they include or the type of the components they are linked to.

AIR HANDLING UNIT

Each component of the air handling unit described in Figure 3 has been modeled using specific in-house models. For a complete description of these models, readers are referred to the work of Iolova (2005).

GEOTHERMAL HEAT PUMP

The heat pump model created by Lemire (1999) is used in this study. This model is linked to a thermostat which calculates the time of operation in a given time step based on the zone load and heat pump capacity. The heat pump model uses the performance map approach with steady-state data. In this approach manufacturer's data are used to generate a series of correlations which determine heat pump capacity and energy consumption as a function of entering air and water temperatures, and air and water flow rates.

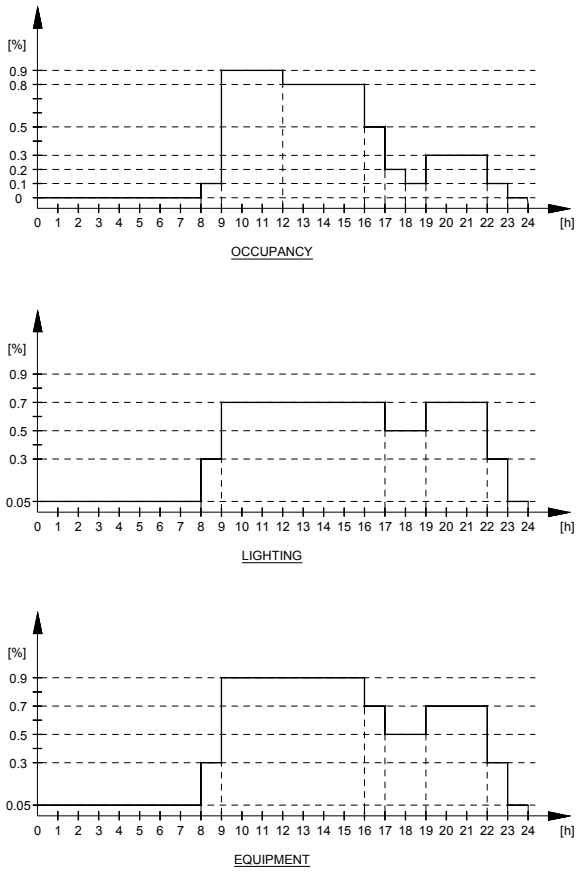

Figure 5: Daily profiles

Cycling losses are neglected. Thus, it is assumed that the heat pumps reach their steady-sate capacity instantaneously as soon as they are started.

THERMOSTAT

As mentioned earlier, the thermostat model calculates the time of operation of the heat pump during the user specified time step. A "temperature level" approach is used whereby the thermostat, the heat pump and the building interact until convergence at each time step. Given the zone temperature obtained from the building model, the thermostat sends an appropriate signal to the heat pump which supply/extract energy to/from the zone. The building model determines the new zone temperature. This process is repeated until two successive time of operation provided by the thermostat are within a given tolerance.

Thermostat settings in the simulations have to reflect the strategy used by the building operators. The zone temperatures (Tz) during the day and night occupancy, as well as during the school year and holidays are set differently. Thus, two profiles for each season (heating and cooling) are created taking into account night setback. Generally, the thermostat is set to maintain Tz > (22ºC+dead band) in winter and Tz < (26ºC-dead band) in summer from 9:00 a.m. to 6:00 p.m. during working days. For the remaining periods in winter, i.e. at night and during week-ends the zone temperature is maintained between 14ºC and 18ºC. In summer, when the building is unoccupied, the thermostat never calls for cooling and zone temperatures are free floating.

GROUND SOURCE HEAT EXCHANGER

The ground heat exchanger is modeled using the DST model (Hellström, 1996) which is part of the TESS library of TRNSYS (TESS, 2001). The temperature in the ground is calculated from three parts; a global temperature, a local solution, and a steady-flux solution. The global and local problems are solved with the use of an explicit finite difference method. The steady flux solution is obtained analytically. The temperature is then calculated using superposition methods.

As shown on Figure 6, the bore field installed at 'École du Tournant' is composed of 18 boreholes, 122 m deep and 150 mm in diameter. One U-tube (composed of two 32 mm HDPE pipes) is inserted in each borehole. The boreholes are grouted over their full depth. The distance between boreholes is 5.0m and they are positioned in a rectangular configuration (3×6). Other parameters are presented in Table 1.

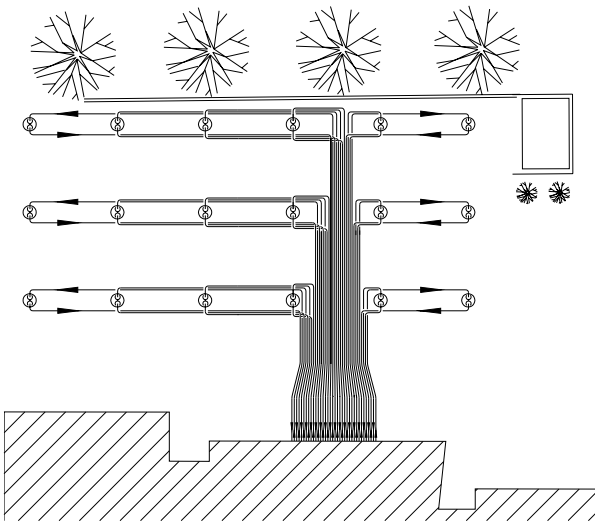

Figure 6: Bore field configuration at 'École du Tournant'. Each circle represents a borehole.

CIRCULATING PUMP

The geothermal system at 'École du Tournant' uses a constant speed circulating pump. As shown on Figure 7, there are actually 2 circulating pumps with only one

in operation at any given time; the second pump is used as backup in case of pump failure. The circulating pump is functioning if at least one heat pump is operating. In certain cases, this operation results in unnecessary high energy consumption for the circulating pump as flow passes through "idled" heat pumps.

Table 1: Parameters of the ground source heat exchanger

Parameters	Values
Storage volume	54900 m^3
Borehole depth	122 m
Number of boreholes	18
Borehole diameter	15 cm
Nominal Dia. of U-tube pipe	32 mm
Center-to-center half distance	5.1 cm
Pipe thermal conductivity	0.42 W/m.ºC
Grout (20% solids)	Bentonite Grout
Fill thermal conductivity	0.73 W/ m.ºC
Initial surface temperature	10 ºC
Number of ground layers	2
Layer material 1	Silty loam-saturated
Thermal cond. of layer 1	2.3 W/m.ºC
Heat capacity of layer 1	1652 kJ/m^3.K
Thickness of layer 1	1.8 m
Layer material 2	Granite-lower quartile
Thermal cond. of layer 2	2.8 W/ m.ºC
Heat capacity of layer 2	2014 kJ/m^3.K
Thickness of layer 2	152.4 m

Figure 7: Circulating pumps at 'École du Tournant'

A better approach is to vary pump flow according to the number of heat pumps in operation. This necessitates a regulation system which is presented schematically in Figure 8. This figure shows that the pump speed is regulated using a variable frequency drive (VFD) coupled to a differential pressure switch. The pump speed is adjusted so as to maintain a constant differential pressure across the supply and return lines of the circulating loop. This differential pressure corresponds to the pressure drop through the heat pumps at the nominal flow rate. When a heat pump unit is turned off, an on-off valve closes which stops the flow from entering this particular heat pump.

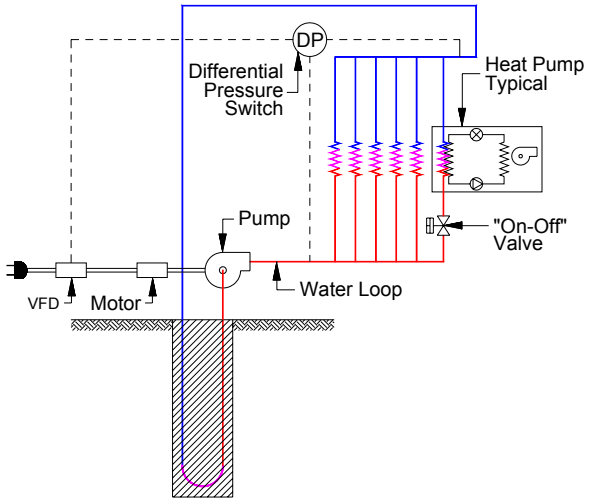

Figure 8: Schematic representation of the GSHP system with a variable-flow circulating pump

This results in an increase of the supply-return pressure difference. This pressure difference is sensed by the differential pressure switch which sends a signal to the VFD to reduce pump speed. Conversely, when the on-off valve opens, the differential pressure decreases and the differential pressure switch sends a signal to increase pump speed.

The operation of this variable flow system can be described using the classic head-flow graphical representation shown in Figure 9. As shown, the system curve intersects the pump curve at the nominal operating conditions when all heat pumps require flow. In the case of "École du Tournant" this point corresponds to a flow rate of 8.86 L/s. When on-off valves close, the operating point "slides" down on the system curve. The pump speed is adjusted so that the resulting pump curve intersects the new operating point. At the limit, when only a few heat pumps are operating the system curve tends towards the intersection of the system curve with the ordinate axis. This point corresponds to the differential switch setting, ΔP_s.

Other pressures of interest on this figure are ΔP_f, the friction head at the nominal operating conditions, and ΔP_t, the total head requirement at the nominal operating conditions

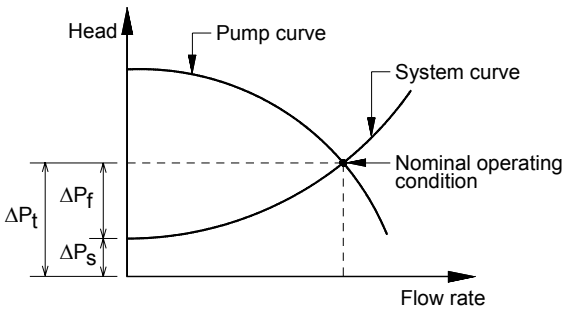

Figure 9: Typical system and pump curves

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A method has been proposed by Bernier and Lemire (1999) to model variable flow pumping system behaviour such as the one presented in Figure 8. They have derived non-dimensional pumping power curves ($P_{in}/P_{shaft,nominal}$) as a function of non-dimensional flow rate, $\dot{Q}_{nd}$, (= actual flow rate / nominal flow rate) for different pump characteristic curves and various values of the ratio X (= $\Delta P_S/ \Delta P_t$) and pump efficiency. Since the power at the nominal operating conditions, $P_{shaft,nominal}$, is a known quantity, the required input power, P_{in}, can be found from:

$$P_{in} = A \times P_{shaft,\ nominal} \qquad (1)$$

$$A=0.22464-0.50812\dot{Q}_{nd}+0.58266\dot{Q}_{nd}^2 + 0.7912\dot{Q}_{nd}^3 - 0.23834X+0.06207X^2 + 0.15208X^3+2.0553\dot{Q}_{nd}X-0.15778\dot{Q}_{nd}X^2 - 1.6652\dot{Q}_{nd}^2X-0.17377\dot{Q}_{nd}^2X^2 \qquad (2)$$

$$P_{shaft,nominal} = \frac{P_{theo}}{\eta_P} \qquad (3)$$

where P_{theo} is the theoretical pumping power and η_p is the nominal pump efficiency.

These equations have been implemented in a TRNSYS model (Lemire, 1999). This model is used in the present study. The nominal operating conditions of the circulating pump at "École du Tournant" are: ΔP_t = 21.8 m, $\dot{Q}$=8.86 L/s, η = 0.66, $P_{shaft,\ nominal}$ = 5 HP. The differential pressure switch setting is ΔP_s = 6.04 m; thus X = 0.28. In the case of variable flow simulations, the non dimensional flow rate varies from 0 to 1 according to the hourly flow rate requirements.

RESULTS

Given the relative complexity of the control strategy which involves many different thermostat set points and occupied/unoccupied periods, it was important to first establish that these control strategies were correctly implemented. This was accomplished by looking at the evolution of the zone air temperature for one of the 23 zones. This evolution is shown on Figure 10. A zoom on the second week of January (which starts on a Wednesday) is presented in Figure 11.

As shown on Figure 10, during the heating season (from 0h to 2500h and from 7000h to 8760h) the air temperature reaches the thermostat heating set point of 22°C when the building is occupied, and is maintained in the interval 14-18°C during the evenings, week-ends and holidays. During the mid-season (from 2500h to 4200h and from 5800h to 7000h) the heat pumps operate in cooling mode and the air temperature is between 22°C and 26°C.

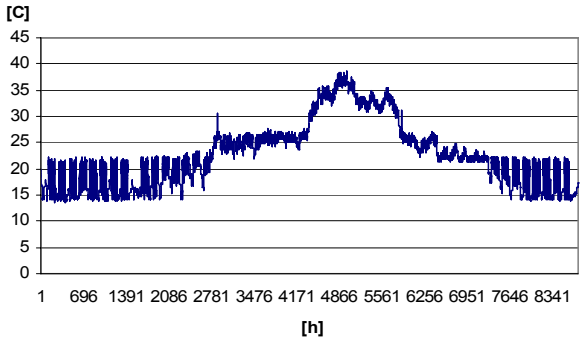

Figure 10: Evolution of a zone air temperature.

During the summer period (from 4200h to 5800h) there are no activities in the school and the heat pumps are not operating. During this period, the zone air temperature is free floating with fluctuations which depend on outdoor conditions. During this period, temperatures in excess of 35°C are obtained.

On Figure 11, the two different thermostat set points for the heating period are highlighted. The night setbacks are followed by the increase of the zone temperature to the desired daily temperature of 22°C. During he week-end (240h-288h), the air temperature is maintained between 14°C and 18°C.

These results show that the zone air temperatures control strategy is modeled correctly and corresponds to the actual building regulation.

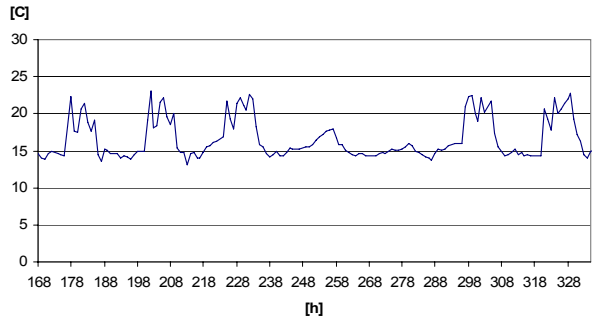

Figure 11: Evolution of the air zone temperature during the second week of January (which begins on a Wednesday).

Simulations were then compared to measured data taken from the building's energy management system. Figure 12 presents the results of a comparison between simulation and measured heat pump energy consumption. The comparison covers the period from September 2003 to January 2004. As shown, the maximum difference between the simulation results and the recorded data occurs in January (-22%) and the minimum difference in October (+1%) with an average of -8% for the studied period. Differences could probably have been reduced had actual meteorological data been used. However, as mentioned earlier, simulations were performed with a WYEC typical meteorological year.

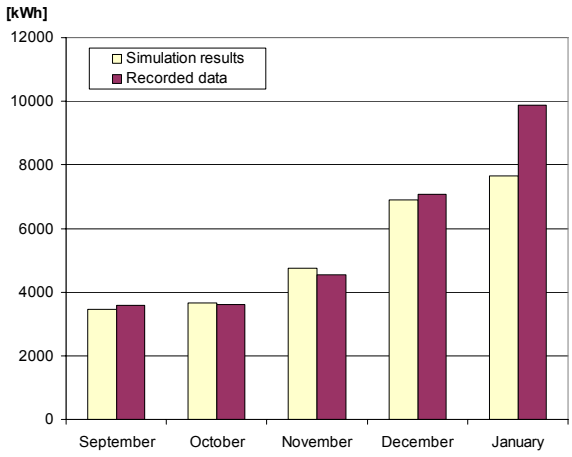

Figure 12: Comparison between simulated and recorded data for monthly heat pump energy consumption.

Constant flow. The constant flow energy consumption is shown on Figure 13. The differences between simulation results and collected data are more important than those recorded earlier for the heat pumps.

The minimum difference is -5% in January and the maximum is +36% in November. These discrepancies are essentially due to the control strategy applied to the circulating pump in order to reduce energy consumption. In January, at least one heat pump is functioning at any given time. Thus, the circulating pump is in operation most of the time and its energy consumption is relatively easy to predict as shown by the good agreement with the January data in Figure 13. During other months, the pump is often deactivated by the operator, especially at night. This behaviour is difficult to simulate and it is reflected in the relatively poor agreement between simulation and recorded data for the September to November period.

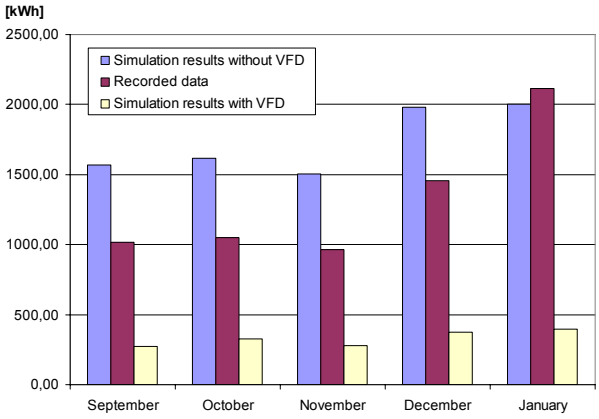

Figure 13: Circulating pump energy consumption

Table 2 presents a summary of energy consumption for the period covering the measured data. In the case of the constant flow pump, the ratio of the pump energy consumption to the heat pump energy consumption is 0.31 (17470/55490) for the simulation and 0.22 (13702/63686) for the measured data. Thus, simulations tend to overpredict the energy consumption of the circulating pump and underestimate the heat pump energy consumption. Nonetheless, simulations of the actual system were considered to be adequate. Then, the same simulation model was used to study what would be the energy impact of a variable flow pumping system.

Variable flow. Results of simulations with variable flow pumping are shown in Figures 13 and in Table 2 in conjunction with the constant flow results. Simulation results indicate that pumping energy

consumption is reduced by about 82% (from 17470 kWh to 3149 kWh) when variable-flow pumping is introduced. Variable flow pumping changes heat transfer in the geo-exchange system which, in turn, changes the inlet fluid temperature to the heat pumps (Iolova, 2005). However, as shown in Table 2, heat pump energy consumption is increased by only 1.5% (from 55492 kWh to 56324 kWh) when going from a constant flow pump to a variable flow scenario. Overall, simulation results show that the total (circulating pump + heat pumps) energy consumption is reduced by 18.5% (72962 vs 59473 kWh).

Table 2 Summary of the energy consumption for the recorded period.

Energy consumption (kWh)	Heat pumps	Circulating pump	Total
Recorded data	63686	13702	77388
Simulation results constant flow pump	55492	17470	72962
Simulation results variable flow pump	56324	3149	59473

CONCLUSION

This article presents the results of a case study aimed at quantifying the energy impact of replacing a constant-speed pump with a pump driven by a variable frequency drive in a ground-coupled heat pump (GCHP) system. The system is installed in a public high school located in St-Constant near Montréal.

Performance data collected from an on-site energy management is used to validate the energy simulations which were performed using TRNSYS 15. In the case of the constant flow pump, the ratio of the pump energy consumption to the heat pump energy consumption is 0.31 (17470/55490) for the simulation and 0.22 (13702/63686) for the measured data. Thus, simulations tend to over predict the energy consumption of the circulating pump and underestimate the heat pump energy consumption. These discrepancies are attributable to two factors. First, due to the unavailability of measured meteorological data, a WYEC typical meteorological file was used. Second, even though real building schedules were correctly implemented, building operation is often accomplished on an “ad hoc” basis by the building operator. This behaviour is difficult to implement in a simulation.

Simulations with variable-flow pumping indicate that pumping energy consumption is reduced by about 82% (from 17470 kWh to 3149 kWh). Variable flow pumping changes heat transfer in the geo-exchange system which, in turn, changes the inlet fluid temperature to the heat pumps. However, heat pump energy consumption is increased by only 1.5% (from 55492 kWh to 56324 kWh) when going from a constant flow pump to a variable flow scenario. Overall, simulation results show that the total (circulating pump + heat pumps) energy consumption is reduced by 18.5% (72962 vs 59473 kWh).

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