Energy and Buildings 65 (2013) 1–9

Contents lists available at ScienceDirect

Energy and Buildings

journal homepage: www.elsevier.com/locate/enbuild

A descriptive and comparative analysis of three common control techniques for an on/off controlled Ground Source Heat Pump (GSHP) system

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ARTICLE INFO

Article history:
Received 23 July 2012
Received in revised form 3 May 2013
Accepted 4 May 2013

Keywords:
Heat pump
Control
Modeling
Simulation
Ground source
Regulation
Geothermal
Capacity

ABSTRACT

In the present paper, three common methods in order to control an on/off controlled Ground Source Heat Pump (GSHP) system called "Constant hysteresis", "Floating hysteresis", and "Degree–Minute" methods are comprehensively described. Then, the generic model already developed by the authors is used in order to do the dynamic simulation of the systems with three different control methods over a year and making the comparison between them. The results from annual modeling of the systems show that the mean temperature of the heating water supplied to the building for the system controlled with degree–minute method is always close to the required temperature, regardless of the climatic boundary conditions over a typical year, whereas, the average supply temperature for the system with constant hysteresis method is mostly higher or lower than the required temperature, depending on the boundary condition. Regarding the annual energy use, the degree–minute and constant hysteresis methods have the lowest and highest annual energy use respectively. Switching from constant hysteresis to floating hysteresis method, the annual energy use will become lower and the mean temperature of the heating water supplied to the building will be closer to the required one.

1. Terminology

In the present study, the temperature of heating water which leaves the condenser of the GSHP and is supplied to the building is called "supply temperature". Moreover, the temperature of the heating water that leaves the heating distribution system in the building and is supplied to the condenser is called "return temperature". Furthermore, the approach heat sink temperature to condenser (water side) is called "load side temperature" and the approach heat source temperature to evaporator (brine side) is called "source side temperature".

2. Introduction

Brine to water heat pump systems like Ground Source Heat Pumps (GSHPs) are one of the common and fast growing systems for heating the buildings in USA and European countries. According to Lund et al. [1], the worldwide installed capacity of GSHPs is about 33 GWt and the annual energy use of GSHPs is over 55 TWh. The number of countries with installation increased from 26 in 2000, to 33 in 2005, and to 43 in 2010. The equivalent number of installed 12 kW units (typical of US and Western European homes) is approximately 2.76 million. This is more than double the number of units reported for 2005, and four times the number reported for 2000 [1].

Only in Sweden, it is estimated that approximately 414 00 GSHPs have been installed providing about 4.5 GW of thermal capacity [2]. During 2011, brine to water heat pump systems in Sweden consumed about 4.5 TWh electricity in order to deliver about 13.5 TWh heat to buildings [2]. Furthermore, the market value of the houses has recently been reported to increase if equipped with an efficient GSHP system thus giving the GSHP systems an augmented status and a new role [3]. Therefore, any improvement in the efficiency of these systems can save a considerable amount of energy, gain a higher credibility for GSHPs in the market and reduce the greenhouse emissions to a large extent.

In order to improve the annual efficiencies of GSHP systems and reach the full thermodynamic potential of the systems, an appropriate capacity control technique must be developed and adjusted for each unique installation. Corberan et al. [4] recently developed a mathematical model capable of describing the performance of a ground source heat pump including the heat pump unit, a secondary ground loop circuit and a secondary building loop circuit. The model developed by Corberan et al. [4] focuses on the

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http://dx.doi.org/10.1016/j.enbuild.2013.05.006

sensitivity analysis and shows how control of building circuit variables such as set-point temperature and temperature bandwidth for both building hydronic circuit and building space affects overall system performance.

Mokhtar et al. [5] proposed an intelligent multi-agent building management system (MAS BMS) to improve the control implemented with a GSHP system. Simulation results show that the proposed intelligent MAS BMS is able to maximize the use of the GSHP effectively by profiling, predicting and coordinating its usage with other energy resources [5].

When capacity control issue is evaluated in a GSHP system, usually, the "conventional on/off controlled" heat pump system is compared with some other common control strategies such as variable capacity heat pumps such as the studies done by (Madani et al. [6–8]; Karlsson [9]; Karlsson and Fahlen [10]; Zhao et al. [11]). Some other studies put a lot of effort to suggest some new complex control strategies. For example, prognostic climatic control is suggested by (Bianchi et al. [12]; Sakellari et al. [13]) or fuzzy control is offered by Choi et al. [14]. However, in the literature, there are not so much detail about how the conventional on/off control method works, what are the influential parameters in this control method, and how different techniques and methods within the on/off control strategy might influence the control output and the system performance.

The present paper aims at description and evaluation of three common control techniques in order to control an on/off controlled GSHP system. Furthermore, annual modeling is carried out for the systems with these three control methods and the results are compared at different ranges of ambient temperatures.

3. Description of the control methods

Fig. 1 presents the schematics of the Ground Source Heat Pump studied in the present paper. Storage tank and domestic hot water usage are excluded in the present study.

In a brine to water heat pump systems connected to a hydronic heat distribution system such as radiator or floor heating, either supply temperature or return temperature (both were already defined in Section 1) are controlled via different control techniques. Usually, the controller receives the ambient temperature as the input; then, it calculates the temperature (either supply or return) that is thought to be the "required temperature" which should be obtained by the heat pump. The curve showing the required supply or return temperature based on the ambient temperature is usually called "heating curve". The heating curve can be drawn and used in the controller based on the Eqs. (1)–(6) [15].

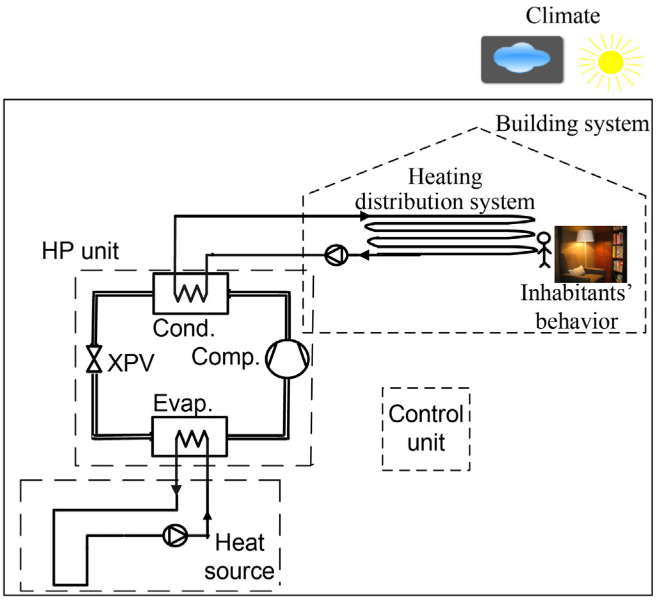

Fig. 1. The schematics of the Ground Source Heat Pump system studied in the present paper.

$$K_1 = \left| \frac{((T_{\text{supply}} - T_{\text{return}})/\ln((T_{\text{supply}} - T_{\text{indoor}})/(T_{\text{return}} - T_{\text{indoor}})))^{1+n}}{T_{\text{supply}} - T_{\text{return}}} \right|_{\text{DOT}} \quad (1)$$

$$K_2 = \left| \frac{T_{\text{indoor}} - T_{\text{amb}}}{T_{\text{supply}} - T_{\text{return}}} \right|_{\text{DOT}} \quad (2)$$

K_1 and K_2 are the constants which are calculated at Design Outdoor Temperature (DOT)[1] and n is the radiator expoenent.[2] DOT is assumed to be −20 °C for Stockholm. T_{indoor} is the indoor set-point temperature which is assumed to be 20 °C.

$$\alpha = K_2 \cdot \exp\left(\frac{\ln(K_1 \cdot (T_{\text{indoor}} - T_{\text{amb}})/K_2)}{1+n}\right) \quad (3)$$

$$\beta = \exp\left(\frac{T_{\text{indoor}} - T_{\text{amb}}}{\alpha}\right) \quad (4)$$

α and β are auxiliary variables that facilitate calculation of required supply and return temperatures according to Eqs. (5) and (6).

$$T_{\text{supply required}} = \frac{\beta \cdot (T_{\text{indoor}} - T_{\text{amb}}) + K_2 \cdot T_{\text{indoor}} \cdot (\beta - 1)}{K_2 \cdot (\beta - 1)} \quad (5)$$

$$T_{\text{return required}} = \frac{(T_{\text{indoor}} - T_{\text{amb}}) + K_2 \cdot T_{\text{indoor}} \cdot (\beta - 1)}{K_2 \cdot (\beta - 1)} \quad (6)$$

So basically, the controller in the GSHP system, as described in the present study, tries to keep the actual supply or return temperature as close as possible to the heating curve obtained from Eqs. (5) and (6). The following section describes three different techniques commonly used to keep the actual supply or return temperature close to the heating curve.

3.1. *Method A: using a constant hysteresis to control the return temperature*

In this method, the temperature of the heating water which leaves the heating distribution system called "return temperature" is the controlled parameter which is always compared with the heating curve (Eq. (6)). The hysteresis method allows the actual return temperature to divert from the required return temperature only to a certain extent within the lower and upper deadband as given in Eq. (7) and (8).

$$\text{start limit}_{\text{compressor}} = \text{Required return temperature} - \frac{\text{Hysteresis}}{2} \quad (7)$$

$$\text{stop limit}_{\text{compressor}} = \text{Required return temperature} + \frac{\text{Hysteresis}}{2} \quad (8)$$

That means if the hysteresis is assumed 5 K, the return temperature is allowed to swing maximum 2.5 K higher or lower than the required return temperature (Eq. (6)) before the heat pump starts or stops. When the return temperature is 2.5 K lower than required, the heat pump is turned on and the return temperature starts to

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[1] DOT is the lowest normally expected temperature in the area where the heat pump is installed.

[2] The radiator exponent represents the change in heat output of a radiator when the actual conditions differ from standard conditions.

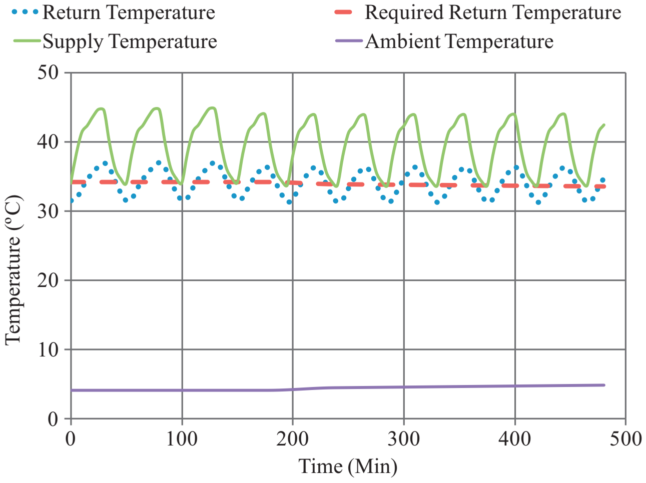

Fig. 2. An example of supply and return temperature variation for a GSHP system with constant hysteresis control method on return line. This graph is based on the simulation results.

increase again. When the return temperature is 2.5 K higher than required, the heat pump is turned off and the supply temperature stops from increasing.

Fig. 2 shows an example of how the supply and return temperature of the heating water vary when the heat pump is intermittently turned on and off. The dashed line shows the required return temperature calculated based on the heating curve. As shown in Fig. 2, the swing in the return temperature will be mostly within the upper and lower deadbands which were already set to ±2.5 K in the controller of this example. However, the temperature swing in the supply line will be higher than the return line; for example, the supply temperature shown in Fig. 2 varies between 35 °C and 45 °C.

In addition to a control algorithm that turns the heat pump on and off, the system needs a control strategy in order to operate the electrical auxiliary heater in the system. The method A which uses the constant hysteresis strategy to control the heat pump operation applies a combination of time-based and hysteresis control method for the electrical auxiliary heater operation. That means if the return temperature has stayed lower than start limit of compressor (Eq. (7)) for more than one hour since the last stop of the compressor, start and stop limit of auxiliary heater are used to turn on and off the electrical auxiliary heater, following Eqs. (9) and (10).

$$\text{start limit}_{\text{aux}} = \text{Required return temperature} - \frac{\text{Hysteresis}}{4} \quad (9)$$

$$\text{stop limit}_{\text{aux}} = \text{Required return temperature} + \frac{\text{Hysteresis}}{4} \quad (10)$$

Let us give an example: it is assumed that within method A as a common control method, the electrical auxiliary heater has three stages: 3 kW, 6 kW, and 9 kW. First, it should take one hour since the last stop of the compressor that the return temperature is lower than the start limit of the compressor (Eq. (7)). Then, the electrical auxiliary heater starts to be turned on or off based on the start and stop limit given in Eqs. (9) and (10). If the auxiliary heater is on for 10 min and the return temperature is still lower than the start limit, the second stage of the heater is tuned on (e.g. 6 kW). The auxiliary heater can also go to the third stage (9 kW) if the return temperature is still lower than the start limit for another 10 min.

For decreasing the electrical heater capacity, much shorter time is considered for response of the controller. For example, if the return temperature is higher than the stop limit, after couple of minutes, the electrical heater capacity decreases to lower stages (6 kW, 3 kW and then turning off the electrical heater).

3.2. Method B: degree–minute method to control supply temperature

In this method, the difference between the actual supply temperature and the required supply temperature is multiplied by the time in order to calculate a parameter called "degree–minute". The calculated degree–minute is summed over time and then used in order to control both heat pump and electrical auxiliary heater in the system. Considering the time as an input to the control unit, beside the temperature may give a better control over the energy delivered to the house, compared to solely temperature control. For example, the control algorithm can be as following:

- Turn on the heat pump when the sum is lower than −60 degree–minute.
- Turn off the heat pump when the sum goes back to 0 degree–minute.
- Turn on the first stage of the electrical auxiliary heater (3 kW) when the sum is lower than −600 degree–minute.
- Turn on the second stage of the electrical auxiliary heater (6 kW) when the sum is lower than −680 degree–minute.
- Turn on the third stage of the electrical auxiliary heater (9 kW) when the sum is lower than −760 degree–minute.
- Turn off the third stage of the electrical auxiliary heater (9 kW) when the supply temperature is 1 K higher than the required temperature.
- Turn off the second stage of the electrical auxiliary heater (6 kW) when the supply temperature is 2 K higher than the required temperature.
- Turn off the first stage of the electrical auxiliary heater (3 kW) when the supply temperature is 3 K higher than the required temperature.
- Reset the sum to zero whenever the supply temperature is 10 K higher than the required temperature.

Fig. 3 shows an example of how the supply temperature varies around the required temperature when the degree–minute method controls the operation of the heat pump. As seen in Fig. 3, the temperature swing in the supply temperature can be as high as 10 K. This high temperature swing might lead to a high temperature swing in the indoor temperature and might potentially deteriorate the comfort condition for the building inhabitants, particularly for light buildings with low thermal mass.

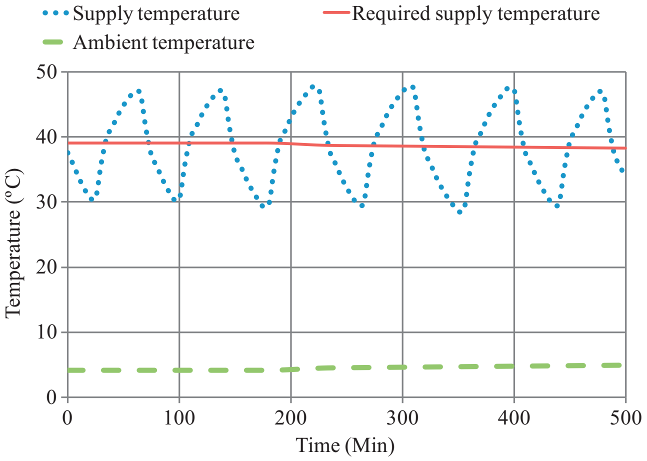

Fig. 3. An example of how the supply temperature changes when a degree–minute method is used to control the heat pump operation. This graph is based on simulation results.

3.3. Method C: using a floating hysteresis to control the return temperature

Similar to method A, the return temperature from the building is the controlled parameter which is compared to the required heating curve. However, differently from method A, method C does not use a constant hysteresis; instead, it uses a variable or floating hysteresis which changes based on change in the heat pump status (turning on or off). When the heat pump is turned on or off, the hysteresis increases to avoid too fast reaction of the control unit to the sudden temperature change; then, if the heat pump stays on or off, the hysteresis may gradually decrease until the next start/stop of the heat pump occur. Therefore, the floating hysteresis method is expected to have a better control over the water temperature compared to constant hysteresis method since the sensitivity of the control unit is adjusted to avoid too fast or too slow reaction to the changes in the system.

As an example, Eq. (11) is used to determine the hysteresis.

$$\text{Hysteresis}_{\text{floating}} = \frac{(\text{hysteresis}_{\max} - \text{hysteresis}_{\min})}{(t_h/\text{time factor}) + 1} + \text{hysteresis}_{\min} \tag{11}$$

t is the time (min) after the last change of heat pump status (being turned on or off). Time factor is the time which should be spent to have the floating hysteresis equal to the average of maximum and minimum hysteresis. For example, if the maximum hysteresis, the minimum hysteresis, and time factor are set to 20, 2, and 20 respectively, the floating hysteresis will be equal to the average hysteresis (11) after 20 min. The hysteresis obtained from Eq. (11) is used in Eqs. (7)–(10) in order to find the start and stop limit for both heat pump and electrical heater operation. When there is a demand for electrical auxiliary heater, the hysteresis will be locked to the maximum value.

Fig. 4 shows how the supply and return temperature of the heating water vary when the heat pump is intermittently turned on and off based on method C. The solid straight line shows the required return temperature calculated based on the heating curve. As shown in Fig. 4, the floating hysteresis (the dashed line in Fig. 4) varies between 16.4 and 8. So the swing in the return temperature (dotted line) will be mostly about 4 K. However, the temperature swing in the supply line will be considerably higher than the return line; for example, the supply temperature shown in Fig. 4 varies between 33 °C and 48 °C.

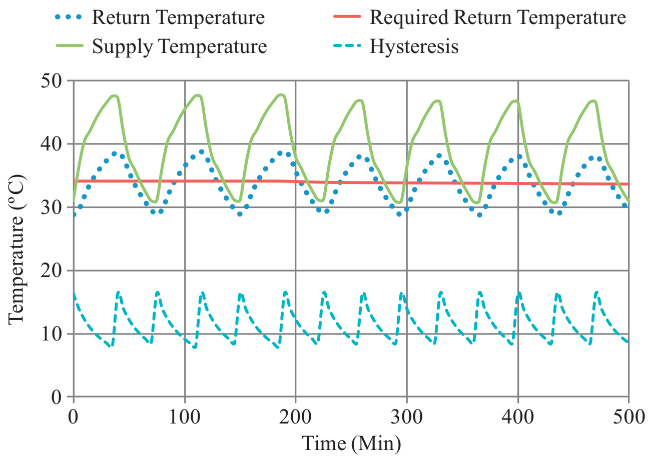

Fig. 4. An example of supply and return temperature variation for a GSHP system with floating hysteresis control method on the return line. This graph is based on the simulation results.

4. Methodology

In order to make a fair comparison between the annual performance of the GSHP system with method A, B, and C, the following parameters are essential:

- Not only heat pump unit model, but also all the other influential parameters within the system must be considered in the evaluation. For example, the performance of not only heat pump unit, but also the electrical auxiliary heater and liquid pumps in both source and sink sides should be evaluated over a year.
- The dynamic interaction between the heat pump unit, the building, the ground heat exchanger, and climate (including solar radiation, etc.) should be taken into account.
- Exactly the same boundary condition should be established for the HP system controlled with all the control techniques (method A, B, or C).

The generic model presented by Madani et al. [6,8] can satisfy the needs mentioned above; so the model is used in the present study in order to do the dynamic simulation of the systems with three different control methods over a year and making the comparison between them. A brief description of the sub-models is given in the following section.

4.1. The building model

The building is modeled by aid of TRNBUILD, an interface in TRNSYS (Klein 2005) in which the building descriptions and its thermal characteristics can be set. In this case, a single family house with 160 m^2 area in two floors located in Stockholm, Sweden is modeled. The heating load of the building at every time step is calculated based on the following parameters:

- The heat losses through the exterior walls, windows, floor and roof
- Infiltration losses
- Ventilation load
- Internal gains
- Solar gain

The construction materials for the walls, roof and floor were selected from the library embedded in TRNBUILD program. In this case, the exterior walls have a wooden lightweight frame construction with mineral wool as insulating material. The windows, added to the exterior walls, are double-glazed with one low-e coating and filled with air. The windows are equipped with internal shading which is applied when the zone temperature exceeds the upper limit zone temperature (20 °C). The floor construction is an externally insulated concrete slab on the ground. The position of the wall to the sun, the ambient temperature and wind velocity at every time and date of the year, and also the solar gains of the exterior walls and windows are all considered in calculation of the heat load at each time step.

Furthermore, the building model provides the opportunity to set the infiltration rate, ventilation rate, internal gains for each zone. In this case, the ventilation and infiltration rates for all the zones are set to 0.5 ACH (Air Change per Hour) and 0.1 ACH, respectively. Moreover, time schedules are applied for occupancy and appliance loads. That makes a variation in internal gain of every zone over a day. Table 1 presents a summary of the characteristics of the building which is modeled and used in the system modeling within the present study.

Table 1
A brief description of the buildings modeled in TRNSYS and used in the system modeling.

Building description	U values (W/m2°K)	Infiltration rate (ach)	Ventilation rate (ach)	Annual heating demand (kWh)	Heat capacitance (kJ/K)
Single family house	Wall: 0.2 Roof: 0.2 Floor: 0.2 Windows: 1.8 (frame + glass)	0.1	0.5	30 000	90 000

4.2. The on/off controlled (single speed) GSHP units equipped with electrical auxiliary heaters

A commercial on/off controlled heat pump unit currently used in single family houses within the European market is modeled. The model is a black box model that is able to process the following input data:

- Approach heat sink temperature to condenser (water side) called load side temperature (T_load).
- Approach heat source temperature to evaporator (brine side) called source side temperature (T_source).

The model outputs are the required compressor work and heating and cooling capacity of the HP unit.

4.3. The ground heat source (borehole)

The heat source for the heat pump in the present paper is the ground (crystalline rock) in which heat is exchanged with the bedrock by circulating a secondary fluid through a closed U-pipe loop in a vertical borehole. The secondary refrigerant transports the heat from the rock to the evaporator of the GSHP-unit.

The borehole model in the present study calculates the outlet temperature of the secondary fluid called "brine" from the borehole at each time step of simulation based on the inlet brine temperature to the borehole (°C), brine mass flow rate and properties, and some borehole characteristics such as:

- Rock thermal properties such as specific heat capacity (J/kg K), density (kg/m^3), thermal conductivity (W/m K).
- Borehole thermal resistance (K m/W), defined as the thermal resistance between the heat carrier fluid and the borehole wall.
- Geometrical properties such as borehole depth and diameter, pipe diameter, etc.

A summary of information about the ground source modeled within the present study is given as following:

- The borehole is 260 m deep, water filled and equipped with a U-tube heat exchanger.
- An aqueous solution of ethanol (20% ethanol by mass) is used as the secondary fluid.
- The pipes are made of polyethylene and have 40 mm external diameter and 2.4 mm thickness.

4.4. The liquid pumps

The liquid pump model estimates the required pump power based on the pressure drops in pipes, connections, and heat exchangers. There are two liquid pumps in the system presented in this study: one in the borehole (heat source side) and one in the heating distribution system (heat sink side). Both of them are constant speed pumps and a constant pump efficiency is assumed for them.

4.5. Climatic conditions

Stockholm, representing the Nordic countries' climatic condition, was selected as the location of all the systems. Therefore, the climatic data for Stockholm, obtained from Meteonorm database [16], is used for the annual simulation.

5. Results from the comparative analysis

An appropriate control of the supply or return temperature of the heating water is an important issue in an on/off controlled heat pumps systems. For example, the higher supply temperature than required leads to higher condensation temperature than required and consequently leads to lower heat pump COP. Fahlen [17] and Karlsson [9] discussed an example in which the difference between the cycle mean temperature and the on-mode mean temperature was 5.7 K. They assumed that this temperature difference can lead to 11% COP reduction in an on/off controlled heat pump (assuming the COP to change by 2%/K).

Furthermore, the variation of supply temperature might lead to change in the indoor air temperature which consequently affects the comfort condition. The indoor temperature swing or large deviation of indoor temperature from the set-point (e.g. 20 °C) might lead to thermal discomfort in the building inhabitants; in the case of having a room thermostat, the building inhabitants may react by manipulating the room thermostat and changing the room set-point temperature. In this case, the heating curve, i.e. the required supply or return temperature will change.

The following section gives a summary of the results for the supply temperature, indoor temperature, energy consumption, and seasonal performance factor in the systems controlled either by method A, B or C.

5.1. Supply temperature control when the heat pump operates intermittently

When the ambient temperature is higher than the balance point temperature,[3] the compressor of the on/off controlled heat pump is turned on and off intermittently. The balance point temperature for the building and heat pump unit selected for the present study is around −9 °C. If the ambient temperature is considerably higher than the balance point temperature, for example when the ambient temperature is higher than +8 °C, the heat pump cycle will have a short on-time in a cycle. Fig. 5 shows an example of how the supply and consequently indoor temperature change for degree–minute and constant hysteresis control methods when the ambient temperature changes between + 8 °C and + 10 °C. The black solid line shows the required temperature.

As shown in Fig. 5, the supply temperature for the degree–minute method can be 5 K lower or 11 K higher than the required temperature; whereas, the temperature swing for the constant hysteresis method is around ±6 K. Since the heat pump is over-dimensioned at this specific range of ambient temperature

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[3] The ambient temperature at which the heat capacity of heat pump is equal to the heat demand of the building is called "balance point temperature".

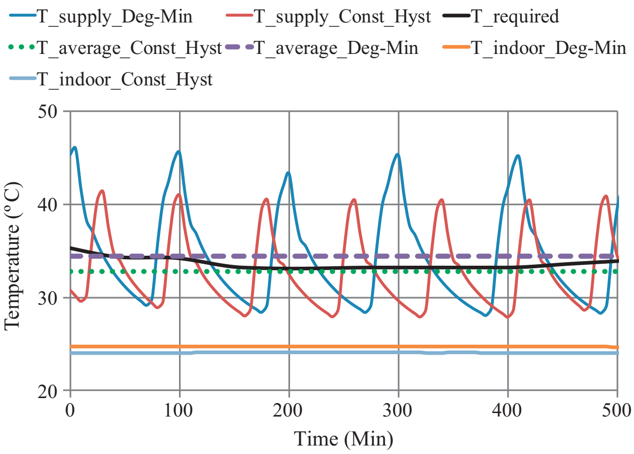

Fig. 5. Comparison between the supply temperature and indoor temperature for both method A (constant hysteresis) and B (degree–minute) methods when the ambient temperature changes between +8 °C and +10 °C.

(above +8 °C), for the HP system controlled with constant hysteresis method, the return temperature exceeds the upper limit shortly after the heat pump starts; then controller turns off the HP again; so the on-time for the system controlled with the constant hysteresis method is so short (around 15 min in Fig. 5) that the average supply temperature is slightly lower than the required temperature. However, for the system controlled with degree–minute method, the on-time will be long enough (around 25 min in Fig. 5) to keep the supply temperature very close to the required temperature.

As shown in Fig. 5, the indoor temperature for the system controlled with the constant hysteresis method is usually lower than the one for the system with degree–minute control. The lower indoor temperature for the hysteresis method is due to shorter on-time and lower average supply temperature, as mentioned before.

Furthermore, Fig. 6 shows an example of how the supply and consequently indoor temperature change for degree–minute and floating hysteresis control methods when the ambient temperature changes between +8 °C and +10 °C. In the floating hysteresis method, the controller has a higher hysteresis just shortly after the heat pump is turned on; so the heat pump stays on for a longer period (around 25 min in Fig. 6). Consequently, the supply and indoor temperature for the system controlled with floating hysteresis method is very close to the one with degree–minute method (Fig. 6).

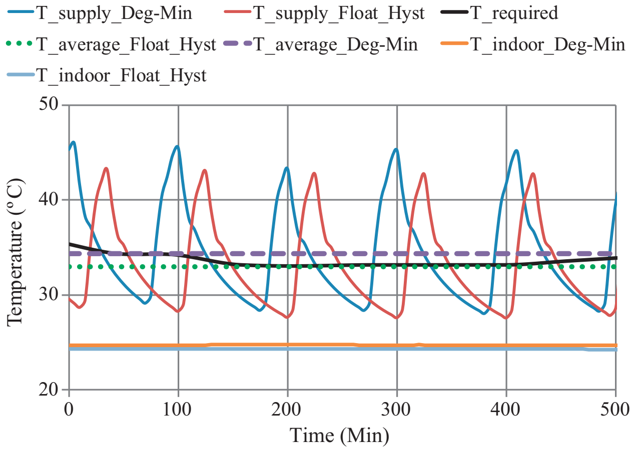

Fig. 6. Comparison between the supply temperature and indoor temperature for both method C (floating hysteresis) and B (degree–minute) methods when the ambient temperature changes between +8 °C and +10 °C.

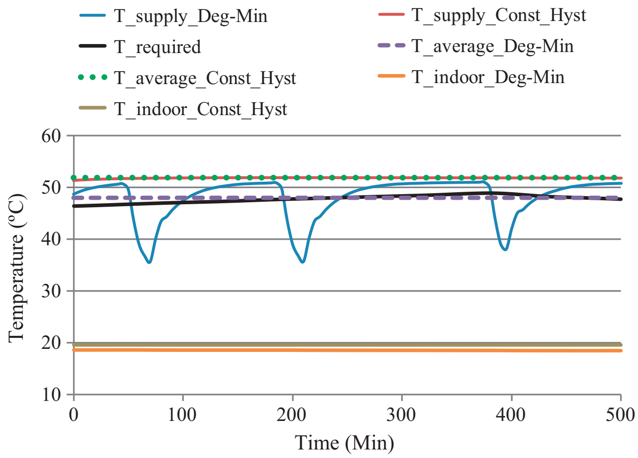

Fig. 7. Comparison between the supply and indoor temperatures of the systems controlled by method A or B when the ambient temperature between −6 °C and −4 °C.

When the ambient temperature is between 0 °C and +8 °C, the average supply temperatures for all three control methods are very close to each other. The oscillation of supply temperature is the least for the system controlled with method A (constant hysteresis method) compared to the two other control methods.

Apparently, when the ambient temperature decreases and becomes just slightly higher than the balance point temperature, which is −9 °C, each cycle's on-time increases. Fig. 7 shows an example of how the supply temperature changes for both constant hysteresis and degree–minute methods when the ambient temperature is between −6 °C and −4 °C and consequently the on-time of each cycle is relatively long.

Oppositely to the high ambient temperatures, when the ambient temperature is slightly higher than the balance point temperature, the on-time for the system controlled with hysteresis method, i.e. method A and C, will be so long (the system operates almost continuously) that the average supply temperature becomes higher than the required temperature. For example, the dotted line in the Fig. 8 shows that the average supply temperature for the system with constant hysteresis method is around 3.5 K higher than the required supply temperature.

However, for the system controlled with degree–minute method, the supply temperature is slightly higher than the required temperature for so long time that the sum of degree–minute value

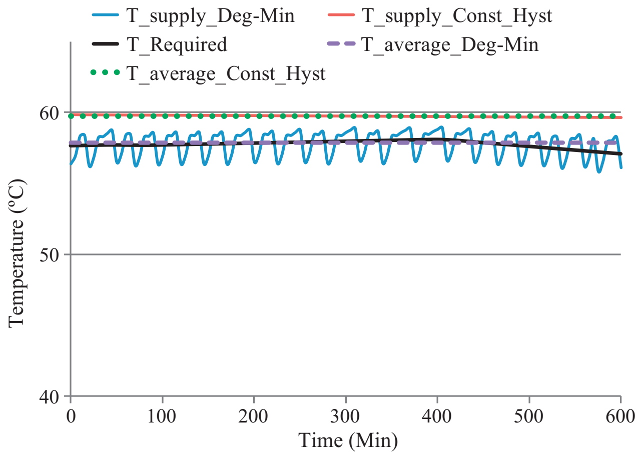

Fig. 8. Comparison between the supply temperatures for the system controlled with method A and B when the ambient temperature is below the balance point temperature.

exceeds zero and the heat pump is turned off. Turning off the heat pump for about 20 min gives an opportunity to the degree–minute controller to provide a supply temperature close to the required temperature at this range of ambient temperature.

5.2. Supply temperature control when the electrical auxiliary heater operates beside the heat pump

Fig. 8 shows how the supply temperature changes when both heat pump and electrical auxiliary heater operate together and they are controlled with method A and B. When the auxiliary heater works together with the heat pump, the degree–minute method (method B) usually changes the status of the electrical heater (turns it on or off) more often than hysteresis method. A very small deviation from the required temperature for a relatively long time makes the changes in degree–minute value high enough to turn the auxiliary heater on or off. This intermittent operation of electrical heater in the system with degree–minute method provides a supply temperature very close to the required temperature (Fig. 8); whereas the constant hysteresis method keeps the electrical heater on for a long time; so as seen in Fig. 8, keeping the auxiliary heater on for a long time provided a higher supply temperature than required for the constant hysteresis method.

For method C, as mentioned before, the hysteresis is locked to the maximum value (20 K) when the electrical auxiliary heater works; so when the electrical auxiliary heater is turned on, the controller keeps it on for a long time, even longer than the one controlled with method A. So the supply temperature for the system controlled with method C is sometime higher than the one with method A.

5.3. Supply temperature control: the results summary

The system modeling over a year shows how much the mean supply temperature is higher or lower than the average required temperature at different ranges of ambient temperature, when either method A (constant hysteresis method), method B (degree–minute method), or method C (floating hysteresis method) is applied as the control method (Fig. 9). As shown in Fig. 9, the mean supply temperature for the system controlled with degree–minute method is always close to the mean required supply temperature; whereas, the mean supply temperature for the system with constant hysteresis method is either higher (at low ambient temperatures) or slightly lower (at relatively higher ambient temperatures) than the required temperature. The reasons for this phenomenon were already discussed in Sections 5.1 and 5.2.

Regarding method C (floating hysteresis), the mean supply temperature is close to the required one when the ambient temperature is above the balance point temperature; whereas, the mean supply temperature is considerably higher than required when the ambient temperature is lower than the balance point temperature and auxiliary heater operates. The high hysteresis (20 K) is the cause of higher mean supply temperature in method C.

5.4. On and off cycle time

The typical cycle time is also an interesting parameter in the system control. Although the time length of each cycle is a parameter which strongly depends on the size of the heat pump, as a general conclusion, the results from the present study show that the degree–minute usually has a considerably longer on and off time for each cycle compared to constant hysteresis method.

For example, when method A (constant hysteresis method) is used for the control of the system and the ambient temperature is between +4 °C and +8 °C, the heat pump is usually on for 15–20 min and then it is turned off and stays off for about half an hour. After approximately half an hour being off, the heat pump is turned on again and works for another 15–20 min. However, when degree–minute is used as the control method at the same range of ambient temperature, the typical on and off time for each cycle are 30 and 65 min, respectively. The typical on and off time of each cycle for the floating hysteresis method is very similar to the degree–minute method.

5.5. Annual energy use and Seasonal Performance Factor (SPF)

Fig. 10 presents the annual energy use of the system components when either method A, B, or C is used to control the system capacity. As presented in Fig. 10, the annual energy use of the compressor is the lowest for the degree–minute method and the highest for constant hysteresis method. The annual energy use of compressor for the system controlled with degree–minute method is about 6% lower than the one with constant hysteresis method.

Degree–minute method has the lowest annual energy use of the electrical auxiliary heater (approximately 50% lower than constant hysteresis method) whereas floating hysteresis method has the highest use of auxiliary heater over a year. The high hysteresis for method C when the electrical heater operates leads to higher

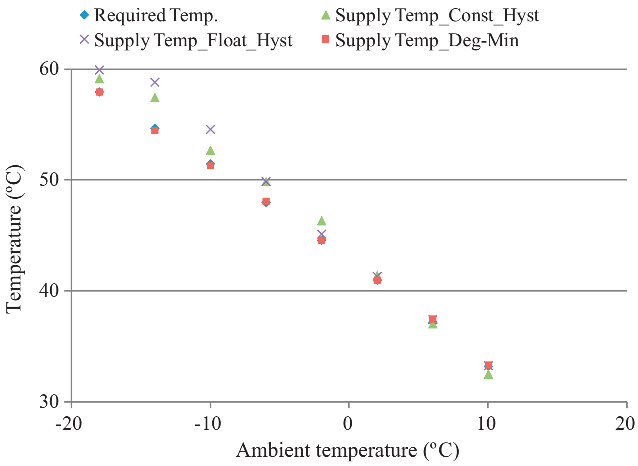

Fig. 9. How much on average the supply temperature is higher or lower than the required temperature when method A, B or C is used as the control method.

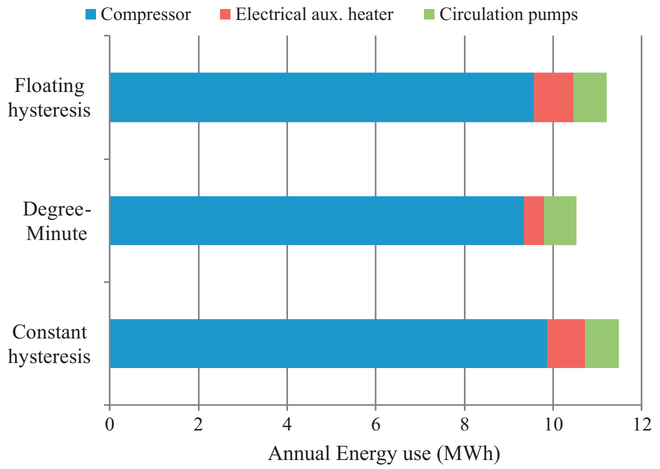

Fig. 10. Comparison between the annual energy consumption of different system components when the system uses either method A, B, or C as the control method.

energy use of the auxiliary heater. The annual energy use of the circulation pumps is almost the same for all three control methods.

Consequently, the lowest total energy use over a year belongs to the degree–minute method (9% lower than the constant hysteresis method). Since the heat provided to the building for all these three control methods are almost the same, the lowest total energy use over a year by the degree–minute method leads to the highest Seasonal Performance Factor (SPF) as well. The constant hysteresis method has the highest total energy use over a year that makes it a less favorable control option for on/off controlled heat pumps.

6. Conclusion

The present paper aims at describing and comparing three common methods in order to control an on/off controlled GSHP system. The generic model presented by Madani et al. [6,8] is used in order to simulate the performance of the heat pump system (including the heat pump, building, ground source, auxiliary heater, pumps etc.) over a year when either method A, B or C is used as the control method. The comparative analysis shows that

- When the ambient temperature is considerably higher than the balance point temperature, the on-time for each cycle of the system controlled with the constant hysteresis method is so short that the average supply temperature is lower than the required temperature. However, for the system controlled with degree–minute or floating hysteresis method, the relatively longer on-time of the heat pump at each cycle provides the supply temperature very close to the required temperature.
- When the ambient temperature is slightly higher than the balance point temperature, the system controlled with the hysteresis method operates almost continuously without any stop; so the average supply temperature for the hysteresis method is higher than the required supply temperature. However, the intermittent operation of heat pump controlled with degree–minute method still at such a low temperature leads to a supply temperature close to the required temperature.
- When the ambient temperature is lower than the balance point temperature, a high hysteresis used in method C gives a considerably higher supply temperature than required. Oppositely, the degree–minute method is shown as a robust method to control the operation of the electrical auxiliary heater.

Generally, concerning the temperature control of the heating water supplied to the building, the average supply temperature for the system controlled with degree–minute method is always close to the required supply temperature; whereas, the average supply temperature for the system with constant hysteresis method can be mostly higher or lower than the required temperature. All the three control methods yield a large oscillation in the supply temperature when the ambient temperature is relatively high. This might affect the thermal comfort of the building inhabitants if the thermal inertia of the building is relatively low.

Regarding the annual energy use, the degree–minute method has the lowest total energy use and constant hysteresis method has the highest total energy use over a year. Lower energy use by the compressor and electrical auxiliary heater leads to higher SPF for the degree–minute method compared to both constant and floating hysteresis methods.

Generally, the degree–minute method and floating hysteresis method (method B and C) described in the present study are shown to be better techniques to control the temperature of the heating water supplied to the building, compared to the constant hysteresis method (method A). The drawback of method C, the floating hysteresis method described in the present study is too high hysteresis when the electrical auxiliary heater operates.

As the final remark, it should be mentioned that the values for the control parameters used in the present study (such as -60 or -600 in degree–minute method) are given just as an example. Different values for the control parameters can yield different results such as different temperature oscillation though the general trend would be the same as shown here. Based on the results from the present study, it is generally recommended not to use constant control parameters in the controller (such as constant hysteresis) to avoid large supply temperature oscillation or large deviation from the required temperatures. One solution is to use the degree–minute method which uses dynamic control parameters. The control parameters can be continuously varied by the controller based on different inputs. The controller can always learn from the heat pump and the building system what the proper control parameters are. So the controller can change the parameters dynamically depending on some static inputs such as time constant of the building and some dynamic inputs such as time, ambient temperature and inhabitants' behavior.

Acknowledgement

This research is a part of a national research and development program in Sweden called "Effsysplus[4]". It is a four year program for applied R&D in Refrigeration and Heat Pump Technology financed by Swedish Energy Agency and several national and international industrial partners.

Appendix A. Supplementary data

Supplementary data associated with this article can be found, in the online version, at http://dx.doi.org/10.1016/j.enbuild.2013.05.006.

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