



ELSEVIER

available at www.sciencedirect.comjournal homepage: www.elsevier.com/locate/ijrefrig

Capacity control in ground source heat pump systems part II: Comparative analysis between on/off controlled and variable capacity systems

Hatef Madani*, Joachim Claesson, Per Lundqvist

Department of Energy Technology, Division of Applied Thermodynamics and Refrigeration, Royal Institute of Technology, SE-100 44 Stockholm, Sweden

ARTICLE INFO

Article history:

Received 23 March 2010
 Received in revised form
 4 May 2011
 Accepted 16 May 2011
 Available online 26 May 2011

Keywords:

Heat pump
 Control
 Modelling
 Simulation
 Variable speed

ABSTRACT

In the present paper, as the second part of two, modeling and simulation was carried out for a Ground Source Heat Pump (GSHP) system in the presence of all the most important interacting sub-systems such as building, ground heat source, electrical auxiliary heater, and the heat pump unit in order to make a fair and comprehensive comparison between the annual performance of on/off controlled and variable capacity systems. The annual modeling showed that dimensioning of the on/off controlled GSHP based on the peak heat demand of the building plays a significant role when the two control strategies are compared: if the on/off controlled GSHP is dimensioned to cover only 55% of the peak heat demand of the building, the electrical auxiliary, which then covers about 10% of the annual heating demand of the building, makes the SPF of the on/off controlled GSHP to be lower than the one of the variable speed system. On the contrary, when the on/off controlled system is dimensioned to cover more than 65% of the building's peak heat demand, i.e. more than 95% of the annual heat demand of the building, there is no considerable difference between the SPFs of the on/off controlled and variable capacity systems.

© 2011 Elsevier Ltd and IIR. All rights reserved.

Régulation de la puissance des pompes à chaleur sol-eau. Partie II : analyse comparative des systèmes régulés de façon tout ou rien et des systèmes à puissance variable

Mots clés : Pompe à chaleur ; Contrôle ; Modélisation ; Simulation ; Vitesse variable

* Corresponding author. Tel.: +46 (0)8 790 86 53; fax: +46 (0)8 20 30 07.

E-mail address: hatef.madani@energy.kth.se (H. Madani).

0140-7007/\$ – see front matter © 2011 Elsevier Ltd and IIR. All rights reserved.

doi:10.1016/j.ijrefrig.2011.05.012

1. 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. As of 2005, there are over a million units installed worldwide providing about 15 GW of thermal capacity (Rybach, 2005) and the annual global increase has been more than 10% over the last ten years (Lund et al., 2004). It is estimated that approximately 300,000 brine to water heat pumps have been installed in Sweden providing about 3 GW of thermal capacity (Forsen and Nowacki, 2011). Furthermore, each year, about 27,000 new GSHP units are installed in Sweden (Ibid). Therefore, any improvement in the efficiency of these systems can lead to saving a considerable amount of energy and reducing the greenhouse gas emissions to a large extent.

Capacity control in GSHPs is one of the techniques having a potential for efficiency improvement.

The conventional on/off control is one of the most common methods for capacity control in GSHPs. In on/off controlled heat pump (HP) units, the compressors are switched on or off by aid of a control algorithm that estimates the discrepancy between building heat demand and heat pump capacity. However, there are some alternative methods suggested by the other researchers which might improve the system efficiency. A variable speed heat pump is one of those methods. Generally, in a variable speed heat pump unit, the speed of the inverter-driven compressor is modulated by aid of control algorithm so that the heat output from the heat pump unit offsets the load of the building; consequently, the discrepancy between the heat supply the heat demand (in the building) can be decreased.

However, there are some issues which have not yet been clarified:

- How does the COP (Coefficient Of Performance) of a variable capacity heat pump system change when the climatic conditions vary over a year?
- What are the conditions under which the variable capacity heat pump units yield a better performance than on/off controlled ones?
- How does the elimination of auxiliary electrical heater in variable capacity heat pump systems influence the yearly seasonal performance factor, SPF, for the heat pump system?

Some researchers have investigated both on/off controlled and variable capacity systems. For example, Zhao et al. (2003) did a comparison among on/off and variable capacity controls, in addition to two other methods, for GSHPs and concluded that changing the compressor speed is the preferred method. However, Karlsson and Fahlen (2007) concluded from their research that despite improved performance at part load (as was already shown by Tassou and Qureshi, 1996), the variable-speed controlled heat pump did not improve the annual efficiency compared to the intermittently operated heat pump. Karlsson (2007) suggested that it is mainly due to the inefficiencies of the inverter and the electrical motor of the compressor and the need for control of pumps used in the heating and ground collector systems (Karlsson, 2007). The

experimental study by Cuevas and Lebrun (2009) showed that in the variable speed scroll compressor, the inverter efficiency varies between 95% and 98% when the compressor electrical power varies between 1.5 and 6.5 kW. It was also found that the additional electrical motor losses induced by the presence of the inverter are negligible (Cuevas and Lebrun, 2009).

The presents paper aims at making a fair comparison between the annual performance of an on/off and variable capacity GSHP systems, using the generic model developed by Madani et al. (2011). The unique features of the current study are as following:

- Not only heat pump unit model, but also all the other relevant sub-models such as the building, the ground heat exchanger, and climate are implemented in detail; so it is feasible to study the influence of any given parameter on the whole system.
- All the dynamic interactions among the system components are considered, using a quasi static model;
- Exactly the same boundary condition is established for both the on/off and the variable capacity systems when they are compared to each other.
- The inverter and motor losses are taken into account in the model of the variable speed GSHP unit;
- The influence of the compressor speed on the isentropic efficiency of compressor is considered;

2. Methodology

The annual performance of the two following control strategies is evaluated by modeling all the systems over a year:

- GSHP systems (four different sizes) with on/off controlled compressors and single speed pumps in both source and load sides
- A GSHP system with a variable speed compressor and single speed pumps in both source and load sides

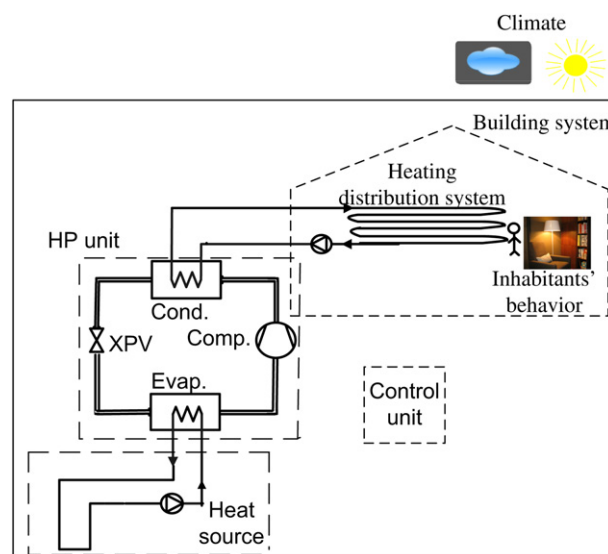


Fig. 1 – The schematics of the systems whose annual performances are analyzed in the present study.

Table 1 – A brief description of the buildings modelled in TRNSYS and used in the system modelling.

| Building description | U values (W/m ² ·K) | Infiltration rate (ach) | Ventilation rate (ach) | Annual heating load (kWh) | Heating peak load (kW) |
|----------------------------------|--|-------------------------|------------------------|---------------------------|------------------------|
| Single family house (large size) | Wall: 0.3 Roof: 0.2 Floor: 0.2 Windows: 1.8 (frame + glass) | 0.1 | 0.5 | 33050 | 14 |

Fig. 1 shows the schematic of the systems studied in the present paper. The system not only covers the heat pump unit consisting of compressor, expansion valve, condenser and evaporator, but also building as the heat load, ground as the heat source, and pumps. The system also includes the electrical auxiliary heater, required when the heat pump unit is not designed to cover the peak heat demand of the building. The system in the present study excludes the storage tank.

The building as the heat load, the ground heat exchanger as the heat source, the boundary condition such as the climatic condition, all are established to be exactly the same for both control strategies; so the strategies can be compared in a fair way.

3. Sub-model description

The general information about the sub-models is already provided by Madani et al. (2011). The detailed information about the sub-models made for the cases in the present paper (two systems mentioned in section 2) are:

3.1. The building

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 large house located in Stockholm, Sweden, with a peak load of 14 kW and annual heat demand of 33,000 kWh is modeled. As presented by Madani et al. (2011), 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 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. A summary of information about the building model can be found in Table 1.

Fig. 2 also shows variation of the building heat load in different ambient temperatures. As it can be seen from Fig. 2, there is a large variation in different building heat demands at the same ambient temperature. This stems from variation of solar radiation, internal gain, wind speed, and also, significantly, from the variation of ambient conditions at the previous hours. For example, a sudden change in the ambient temperature does not change the building heat demand instantly due to the high thermal mass of the building.

3.2. The on/off controlled (single speed) GSHP unit equipped with an electrical auxiliary heater

A typical on/off controlled heat pump unit currently used in Swedish single family houses has been modeled. As presented by Madani et al. (2011), all the sub-models including the single speed compressor, the evaporator, and condenser are modeled individually and connected together to form the

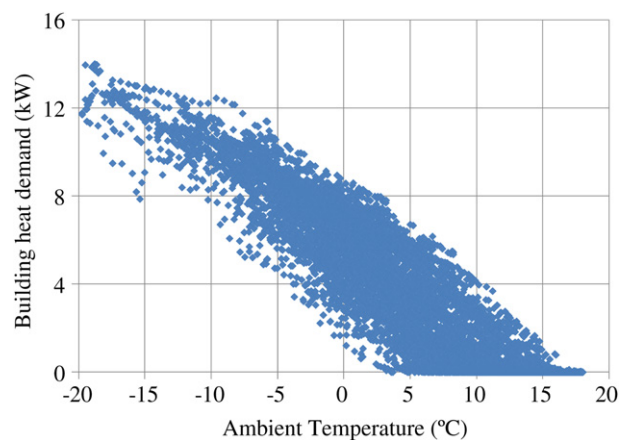


Fig. 2 – The building heat demand (kW) at different ambient temperatures.

Table 2 – A brief size description of the single speed GSHP units which are modelled in the present paper.

| | Heat capacity @ source side/load temp of 0/35 °C (kW) | COP @ source side/load temp of 0/35 °C | Approximate balance point Temperature (°C) | Heat pump heat capacity/building peak heat demand (%) | % of annual heat demand covered by HP | % of annual heat demand covered by electrical auxiliary heater |
|--------|---|--|--|---|---------------------------------------|--|
| Unit A | 7.7 | 2.98 | –5 | 55 | 90.4 | 9.9 |
| Unit B | 9.2 | 3.04 | –8 | 66 | 96.4 | 3.6 |
| Unit C | 12 | 2.97 | –13 | 86 | 99.6 | 0.4 |
| Unit D | 13.2 | 2.94 | –17 | 94 | 99.94 | 0.06 |

HP-unit model. Engineering Equation Solver (Klein, 2003) is selected as the simulation tool for the HP unit modeling. The main function of this model is to process the input data given from the building (load side temperatures), borehole models (source side temperatures), and climate (ambient temperatures) by finding the required compressor work, heating and cooling capacity of the unit at every time step of the simulation based on the given input.

Usually, HP units with single speed compressors in Sweden are dimensioned to cover about 55–70% of the peak heat demand i.e. the building heat demand at the Design Outdoor Temperature. The designed HP units typically cover 85–98% of the total heating energy demand over a year (Forsén, 2010). In the present study, as presented in Table 2, four HP units with different sizes (with the heat capacity ranges from 7.7 kW to 13.2 kW at source/load temperature of 0 °C/35 °C) are selected. According to the heat load of the building located in Stockholm, these units cover from 90 to 99.9% of the total heating energy demand over a year.

Fig. 3 presents the relationship between the heat capacities of on/off controlled HP units, used in the present study, and the heat demand of the building, described in 3.1, at different ambient temperature. In this figure, the source side temperature is assumed constant at 0 °C, contrary to the variable heat source temperature in the actual modeled system. Furthermore, the load side temperature varies based on the variation in ambient temperature in order to meet the heating demand of the building at different ambient conditions.

As shown in Fig. 3, the point where the two curves (heat pump heat capacity and building heat demand curves) cross,

often denoted as “the balance point”, the capacity of the HP unit is equal to the heat demand of the building. The ambient temperature at which the balance point occurs is called “balance point temperature”. The balance point temperatures for the particular combination of the building (already described in 3.1) and the HP-units in the present study are given in Table 2. On the right hand side of the balance point (between the heat capacity line and building heat demand line where the ambient temperature is higher than balance point temperature), the capacity of the HP unit is higher than the heat demand and the compressor will turn on and off intermittently (the compressor cycling region). Oppositely, on the left side of the balance point (between the heat capacity line and building heat demand line where the ambient temperature is lower than balance point temperature), the heat demand of the building is higher than the HP unit can deliver and the electrical auxiliary heater is activated to compensate the deficit.

It should be mentioned that in the real case, the building heat demand does not change linearly by variation in the ambient temperature (as was indicated in Fig. 3) because of the solar insolation, the thermal inertia of the building, and user behavior. Thus Fig. 3 and also Fig. 4 only depict an estimation of the building heat demand at different ambient temperatures; whereas, in the modeling process, all the influential parameters on the building heat demand including the solar insolation and thermal inertia are considered.

Additional information about the units’ characteristics and performances are presented in the Table 2.

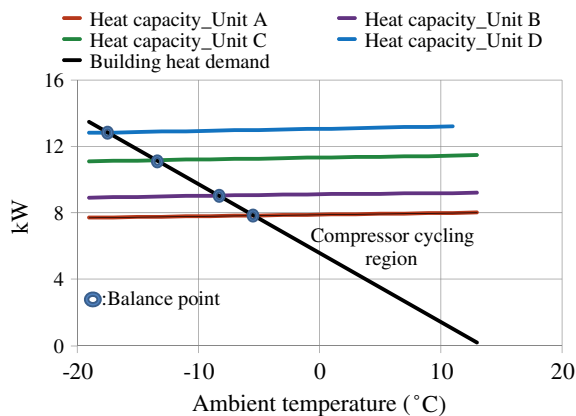


Fig. 3 – Principal building heat demand and the heat capacities of single speed HPs (unit A,B,C, and D) at different ambient temperatures (a rough estimation).

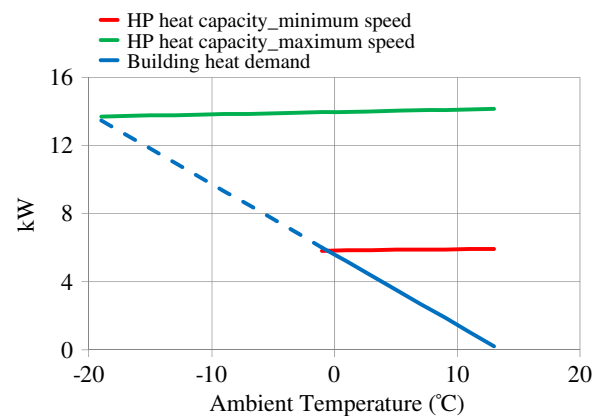


Fig. 4 – The building heat demand and variable speed Hp heat capacity at different ambient temperature (a rough estimation).

3.3. The variable speed GSHP unit

The same GSHP unit as unit B (described in Table 2) has been modeled; however, a frequency inverter is retrofitted to the unit B in order to change the compressor speed and consequently the unit heat capacity. Similarly to the on/off controlled case, all the components including the compressor (but this time, variable speed one), the evaporator, condenser and expansion valve are modeled individually and connected together to form the HP unit model (Madani et al., 2011).

Fig. 4 depicts the heat capacity of the variable speed GSHP unit at minimum and maximum compressor speeds, alongside the heating demand of the building. As shown in Fig. 4, for the variable speed HP unit, in spite of decreasing the compressor speed down to the minimum frequency (30 Hz), there is still compressor cycling region; however, the balance point temperature for the variable speed HP unit is about -1 which is 4K higher than the balance point temperature of the smallest on/off controlled HP unit (unit A). That means by decreasing the compressor speed from 50 Hz (single speed case) to 30 Hz (variable speed case), the cycling region is shifted to the right hand side and becomes smaller. As it is shown with the dashed blue line in Fig. 4, when the ambient temperature is below -1 °C, the compressor speed varies between the minimum and maximum in order to provide as much heat as needed for the building.

Table 3 presents the heat capacities and COPs of the variable speed GSHP unit at three compressor speed: 30 Hz which is the minimum compressor speed, 50 Hz which is the speed that the compressor is designed for, and 90 Hz which is the maximum speed of the compressor. As it can be seen from Tables 2 and 3, the single speed unit shows higher performance at 50 Hz compared with the variable speed unit due to the avoidance of inverter losses.

3.4. 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. A summary of information about the modeled ground source for both studied systems is given as following:

Table 3 – The heat capacities and COPs of the variable speed GSHP unit modelled in the present paper at three different compressor frequencies.

| | Heat capacity @ source side/load temp of 0/35 °C (kW) | COP @ source side/load temp of 0/35 °C |
|----------------------------------|---|--|
| Minimum compressor speed (30 Hz) | 6.1 | 3.2 |
| Medium compressor speed (50 Hz) | 9.2 | 2.85 |
| Maximum compressor speed (90Hz) | 14.4 | 2.33 |

- The borehole is 200 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.
- The working fluid volume flow rate is 2 m³ hr⁻¹

3.5. The liquid pumps

The liquid pump model estimates the required pump power based on the pressure drops in pipes, connections, and heat exchangers. Constant pump efficiency is assumed for the single speed pumps. As it is shown in the system layout (Fig. 1), there are two liquid pumps in the system, one in the heating distribution system of the building (load side of the heat pump) with the rated power of 40 W, and one in the borehole side with the rated power of 240 W. The first pump is always on during the heating season; however, the second pump is only in operation during the on time of the HP unit.

3.6. Climatic conditions

Stockholm, representing the Nordic countries' climatic condition, was selected as the location of both systems. Therefore, the climatic data for Stockholm, obtained from Meteotest (Meteotest, 2009) database, is used for the annual simulation.

4. Results from system modeling

At this stage, all the sub-models are combined together to form the models of both variable speed and on/off heat pump system. The detailed information about the system modeling was already presented by Madani et al. (2011). The time step for modeling was set to 5 min.

4.1. The unit COP in different temperature ranges

In order to make a comparison between the performance of the single speed and variable speed HP units over a year, their performance are analyzed in different ranges of ambient temperature. Before presenting the results for different temperature ranges, it is worthy to clarify some points:

- The temperature ranges given below are approximate values and strongly depend on the design of the units. Different sizes of the units or buildings may yield completely different ranges of temperatures.
- The COPs shown in Fig. 5 to Fig. 10 are the unit's COPs including the auxiliary heater, but not the power consumed by the liquid pumps; whereas, the calculated seasonal performance factors in the next section includes the pumps energy consumptions.
- In the calculation of the system COP, the cycling losses of the compressor are neglected according to IEA HPP Annex 28 (Wemhoener and Afjei, 2006).

4.1.1. Ambient temperature is above +12 °C

When the ambient temperature is above 12 °C, the system is mostly turned off because there is no considerable heat

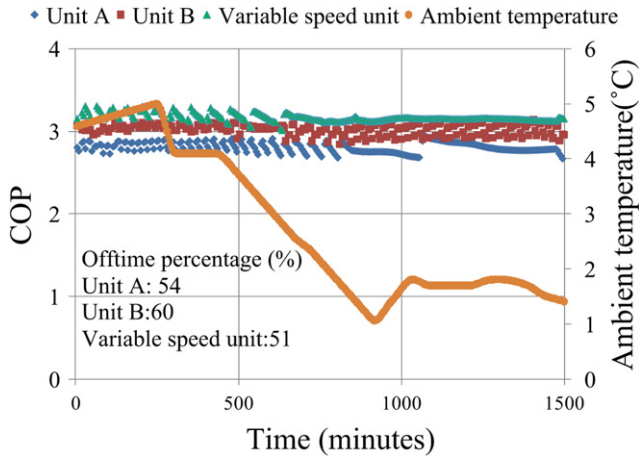


Fig. 5 – Dynamic COP of the variable speed and single speed GSHP systems (system A, B) when the ambient temperature is between +1 °C and +5 °C.

demand of the building. When the ambient temperature is between 12 °C and 20 °C, internal and solar gains of the building are usually enough to compensate for the building heat losses.

4.1.2. Ambient temperature is between -4 °C and $+12$ °C

In this temperature range (-4 °C_+12 °C), the on/off controlled heat pumps (unit A, B, C, and D) mostly work intermittently. That means that in this temperature range, usually, the heat demand of the building is still lower than the heat capacity of the on/off controlled units; The variable speed unit either operates intermittently (more often when the ambient temperature is above 0 °C) or continuously changing the frequency (mostly from 30 Hz to 50 Hz) to meet the building heating demand.

Figs. 5 and 6 show examples of the COP changes in the on/off and variable speed heat pump systems when the ambient temperature is between +1 °C and +5 °C (Fig. 5) and between -4 °C and +1 °C (Fig. 6). The abscissa in Figs. 5 and 6 represent the time (in minutes) which is selected out of the annual modeling of the system to show a typical day with this range of ambient temperature. The ordinate on the left and right hand sides of the figures represent system COP and ambient temperature (°C) respectively.

As shown in Figs. 5 and 6, when the performance of the on/off controlled heat pump units (unit A, B, C or D) are compared with the one of variable speed unit in this temperature range, there is no general conclusion to draw: the variable speed unit at low frequency can have a higher COP than the single speed unit¹ or contrarily, at many points in this temperature range, on/off controlled units can show a better performance than variable speed unit due to the inverter, motor or compressor losses.

Due to the fact that the variable speed unit at 30 Hz yields lower heat capacity than the on/off units do at 50 Hz, the off time period for the variable speed unit is shorter than the

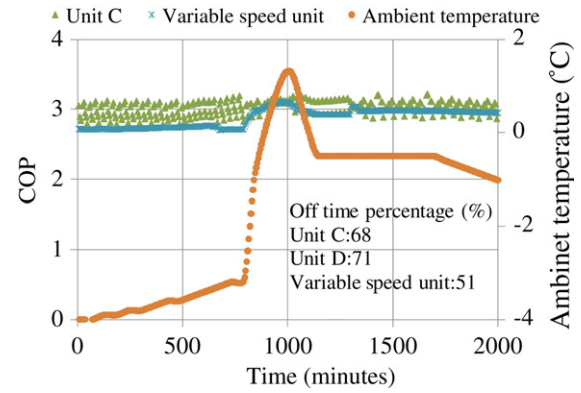


Fig. 6 – Dynamic COP of the variable speed and single speed GSHP systems (system C) when the ambient temperature is between -4 °C and $+1$ °C.

ones for the single speed units. As presented in Figs. 5 and 6, the off time percentages (the ratio between the off time to total time) of the variable speed unit is 51% due to the capability to slow the compressor speed down to 30 Hz. Oppositely, unit C and unit D have a high off time percentage (68 and 71%) which means they are too large to be used in this building.

4.1.3. When the ambient temperature is between -12 °C and -4 °C

In this temperature range (-12 °C_ -4 °C), unit A, unit B, and variable capacity HP unit work continuously whereas unit C and unit D still operate intermittently. Furthermore, the electrical auxiliary heater in unit A operates almost continuously and in unit B, it is turned on and off intermittently. As it is shown in Fig. 7, in the on/off controlled HP units, the auxiliary heater has a stepwise operation due to the heat demand of the building.

Fig. 7 shows how the COP of unit A decreases considerably at this low ambient temperature (the ambient temperature lower than the balance point temperature) due to operation of the electrical auxiliary heater. For the variable speed unit, lowering the ambient temperature from -4 °C to -10 °C

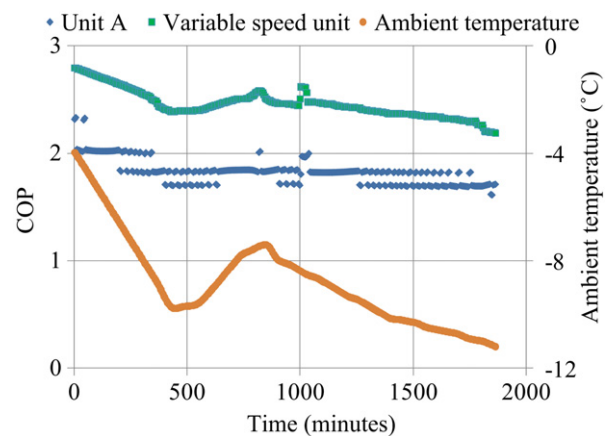


Fig. 7 – Dynamic COP of both variable speed and single speed GSHP systems (system A) when the ambient temperature is between -12 °C and -4 °C.

¹ That can be caused by increasing the COP by decreasing the heat pump frequency.

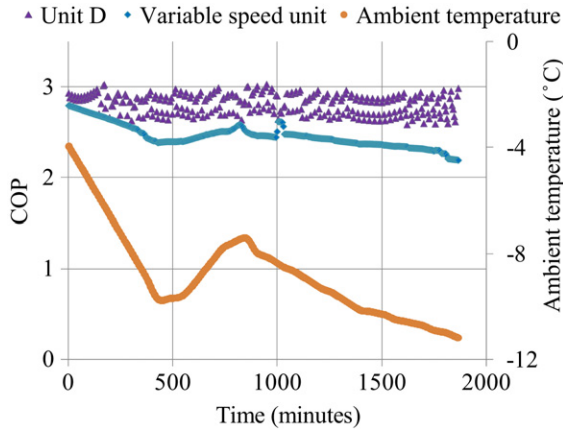


Fig. 8 – Dynamic COP of both variable speed and single speed GSHP systems (system D) when the ambient temperature is between $-12\text{ }^{\circ}\text{C}$ and $-4\text{ }^{\circ}\text{C}$.

reduces the COP from 2.8 to 2.4. This stems from increasing the compressor speed.

As shown in Fig. 8, in unit D, when the auxiliary heater is shut off, the COP of the single speed system is higher than the one for variable speed system. The slight change in the COP for the variable speed system after about 1000 min, shown in both Figs. 7 and 8, is due to the activities of the building inhabitants which reduces the heat demand of the building. That causes a decrease of compressor speed and an increase of unit COP.

4.1.4. When the ambient temperature is below $-12\text{ }^{\circ}\text{C}$

The variable capacity unit and all the single speed units except unit D work continuously when the temperature is lower than $-12\text{ }^{\circ}\text{C}$; the electrical auxiliary heater in the units A and B also operate in a stepwise manner; therefore, the system COP for unit A and B would be decreased considerably. In the variable capacity system, the compressor speed varies usually between 60 Hz and 90 Hz in order to satisfy the heat demand. As may be seen from Fig. 9, the variable capacity system has higher efficiency than unit A due to avoiding the electrical auxiliary heater;

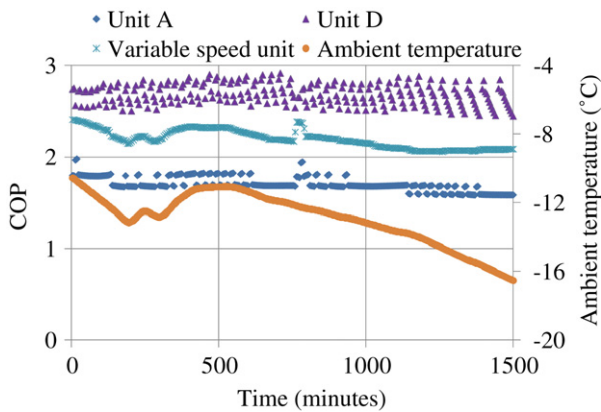


Fig. 9 – Dynamic COP of both variable speed and single speed GSHP systems (systems A and D) when the ambient temperature is between $-18\text{ }^{\circ}\text{C}$ and $-11\text{ }^{\circ}\text{C}$.

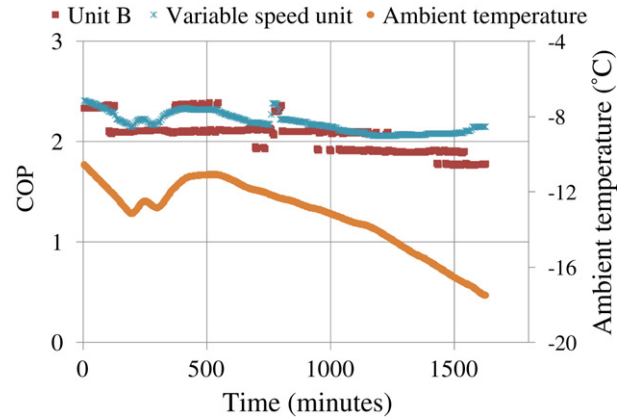


Fig. 10 – Dynamic COP of both variable speed and single speed GSHP systems (systems B and C) when the ambient temperature is between $-18\text{ }^{\circ}\text{C}$ and $-11\text{ }^{\circ}\text{C}$.

Furthermore, the modeling results show that there is no significant difference between the COP of system B, system C, and the variable speed system. As it is shown in Fig. 10 for system B, the frequent use of electrical auxiliary heater in system B (or system C) can make the COP of the on/off controlled system be higher or lower than the COP of the variable speed system.

4.2. The annual performance of the system

Fig. 11 presents the annual energy use of compressors, electrical auxiliary heater, and pumps for both single speed and variable speed systems. Unit D has the highest annual compressor energy consumption and the lowest pump energy consumption which shows this unit is obviously oversized. After unit D, the variable speed system has the highest compressor energy use because the variable speed compressor covers all the heat demand without any need for the electrical auxiliary heater.

Since the on-time period of the variable speed unit is higher than the on-time of the unit A, B, and C, the brine side pump works for longer time over a year and the pumps energy

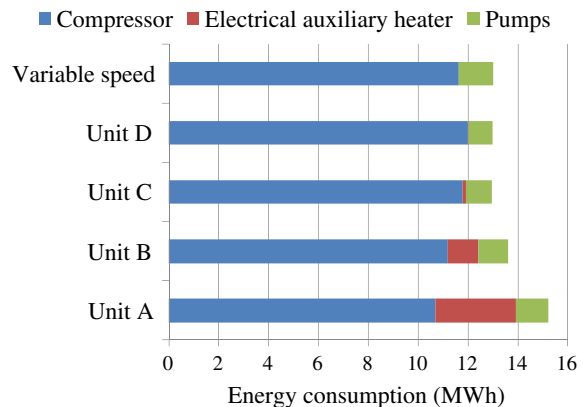


Fig. 11 – The annual energy use of system components: variable speed and on/off controlled systems.

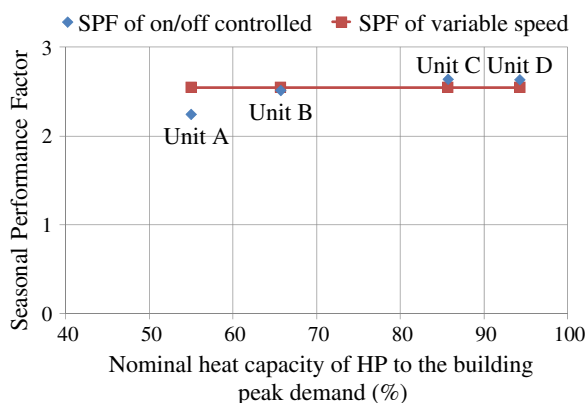


Fig. 12 – Seasonal Performance Factor (SPF) of on/off controlled systems versus the ratio of HP heat capacity to the building heat demand (%). The constant line represents the SPF of the variable speed system which is independent of the X axis.

consumption in the variable speed system is higher than the one for system A, B, and C.

Finally, it can be seen from Fig. 11 that the unit A and unit B have higher total energy consumption than variable speed system over a year due to high annual energy consumption by their electrical auxiliary heaters.

Fig. 12 shows the Seasonal Performance Factor (SPF) of the on/off controlled HP systems versus the one of the variable speed HP system. The abscissa represents the ratio of nominal heat capacity of the on/off controlled units (i.e. the heat capacity @ 0 °C/35 °C) to the building heat demand in percentage (%). This ratio is an important parameter when a heat pump is dimensioned for a specific building. The ordinate represents SPF of the system including the annual energy consumptions of the compressor, pumps, and the electrical auxiliary heater (in the on/off controlled cases). Since there is only one variable speed system in the present study and that system covers 100% of the peak heating demand of the building, the SPF of the variable speed system in Fig. 12 is shown as the constant value (2.54).

As shown in Fig. 12, the SPF of unit A is about 13% lower than the SPF of the variable speed system whereas the SPF of system B, C, and D are almost the same as the one of variable speed system. This shows the significant role of electrical auxiliary heater in the on/off controlled system when they are compared with the variable speed systems. The larger use of the electrical auxiliary heater, the higher total energy use of the on/off controlled system would be. This suggests that a well-dimensioned single speed heat pump unit which can cover about 96% of the annual heat demand of the building (such as unit B) may make the SPF of the system very close to SPF of the variable speed system. Clearly, the economic constraints limit the size of a single speed HP unit to some extent. That means the economic constraints may make the system B the best option in the present study even though the SPF of system C and D are slightly higher than the SPF of system B.

It should be mentioned that contrary to the variable capacity system, on/off controlled systems would suffer from

higher oscillation in the temperature of the supplied heating water to the building and consequently, high indoor temperature swing. That happens particularly for system C and D.

5. Conclusion

The current paper compares the annual performance of the on/off controlled and inverter-driven variable capacity heat pump systems, using the generic model developed by Madani et al. (2011). The liquid pumps used in both systems are constant speed pumps. Both systems are modeled in details and the results of the modeling for the system components are validated with the experimental data from in-situ field measurements (Madani et al., 2011).

The results from the annual modeling for both systems indicate:

- When the annual performance of on/off controlled GSHP system is compared with the one of the variable speed GSHP system, it must be noticed that dimensioning of the on/off controlled GSHP based on the peak heat demand of the building plays a significant role: if the on/off controlled GSHP is dimensioned to cover only 55% of the peak heat demand of the building, the important role of the electrical auxiliary heater (covering about 10% of the annual heating demand of the building) makes the SPF of the on/off controlled GSHP be lower than the one of the variable speed system. On the contrary, when the on/off controlled system is dimensioned to cover more than 65% of the building's peak heat demand, i.e. more than 95% of the annual heat demand of the building, there is no considerable difference between the SPFs of the on/off controlled and variable capacity systems.
- The energy use by liquid pumps in the variable speed GSHP system can be 5–30% higher compared to the single speed system. Compared to on/off controlled system, the variable speed system operates for a longer time during a year which makes the pump in the ground heat exchanger work for a longer time as well.

As a final remark, in order to obtain a higher efficiency for on/off controlled systems, it is important to dimension them in a way that electrical auxiliary heater covers less than 5% of annual heat demand. Furthermore, in order to increase the annual efficiency of the existing on/off controlled systems, it is conditionally² possible to retrofit an inverter to the current single speed HP systems, already installed in the buildings. When the ambient temperature is above the balance point of the system, the HP system works with the same conditions as before, i.e. with the inverter disconnected. However, when the ambient temperature is lower than the balance point and there is a need for the auxiliary heater, the inverter can be connected to the system and increase the compressor speed. Therefore,

² It must be technically allowed by the compressor manufacturer.

the need for electrical auxiliary heater which deteriorates the system COP is eliminated and the system yields a better annual performance.

Acknowledgements

This research is a part of a national research and development program in Sweden called “Effsys2³”. 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.

REFERENCES

- Cuevas, C., Lebrun, J., 2009. Testing and modelling of a variable speed scroll compressor. *Appl. Therm. Eng.* 29, 469–478.
- Forsen, M., Nowacki, J.E., 2011. Swedish heat pump Association, SVEP Personal Communication.
- Karlsson, F., 2007. Capacity Control of Residential Heat Pump Heating System. Department of Energy and Environment. Göteborg, Chalmers University of Technology. Doctoral thesis.
- Karlsson, F., Fahlen, P., 2007. Capacity-controlled ground source heat pumps in hydronic heating systems. *Int. J. Refrigeration* 30, 221–229.
- Klein, S.A., 2003. EES: Engineering Equation Solver, Professional Version. F-Chart Software, Middleton, WI. <http://www.fchart.com> Website.
- Klein, S.A., et al., 2005. TRNSYS Version. 16. Solar Energy Laboratory. University of Wisconsin-Madison. <<http://sel.me.wisc.edu/trnsys>> Website.
- Lund, J., Sanner, B., et al., 2004. Geothermal (Ground Source) heat pumps, a world overview. *Geo-Heat Centre Quarterly Bulletin*, Klmath Falls, Oregon: Oregon Institute of Technology 25 (3), 1–10.
- Madani, H., Claesson, J., Lundqvist, P., 2011. Capacity control in ground source heat pump systems- Part I: modelling and simulation. *Int. J. Refrigeration* 34 (6), 1338–1347.
- Meteotest, 2009. Meteonorm Version 6.1. Bern, Switzerland, Website. <http://www.meteonorm.com/pages/en/meteonorm.php>.
- Qureshi, T.Q., Tassou, S.A., 1996. Variable-speed capacity control in refrigeration systems. *Appl. Therm. Eng.* 16 (2), 103–113.
- Rybach, L., 2005. The Advance of Geothermal Heat Pumps Worldwide. Newsletter IEA Heat Pump Centre (23/4).
- Wemhoener, C., Afjei, T., 2006. Test procedure and seasonal performance calculation for Residential heat pumps with combined Space and Domestic Hot water heating final report IEA HPP Annex 28, Muttenz, Switzerland.
- Zhao, L., Zhao, L., et al., 2003. Theoretical and basic experimental analysis on load adjustment of geothermal heat pump systems. *Energy Convers. Manage.* 44, 1–9.

³ www.effsys2.se.