Energy and Buildings 47 (2012) 600–611

Contents lists available at SciVerse ScienceDirect

Energy and Buildings

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

Performance prediction of a hybrid solar ground-source heat pump system

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ARTICLE INFO

Article history:
Received 27 October 2011
Received in revised form
22 December 2011
Accepted 24 December 2011

Keywords:
Hybrid solar ground-source heat pump (HSGSHP)
Solar assisted ground-source heat pump (SAGSHP)
Coefficient of performance (COP)
Control strategy
First-operation time
Load circulation pump control strategy

ABSTRACT

This paper presented a novel hybrid solar GSHPS (HSGSHPS) composed of a GSHPS and a solar assisted GSHPS (SAGSHPS) used in an office building for heating and cooling. The key issues of designing the HSGSHPS were introduced and a simulation model was developed in TRNSYS to predict the multi-year performance of this system. The simulated results showed that the proposed HSGSHPS was reasonably designed to resolve the ground temperature imbalance problem on an annual basis. The suitable control strategy for the solar collection and storage was found according to the coefficient of performance (COP) of SAGSHPS. The first-operation time has impact on the operation of SAGSHP and GSHPS. Injection of thermal energy into the borehole heat exchanger (BHE) or borehole thermal energy storage (BTES) before the heat extraction from BHE/BTES was favourable to the overall COP of system. It is very important to SAGSHPS to store sufficient solar thermal energy in the BTES because of its small spacing between boreholes. 32% of the electrical energy consumption in the HSGSHPS could be saved if the load circulation pump was turned off when no fan-coil was running, rather than always keeping it running at all times.

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1. Introduction

Ground-source heat pump systems (GSHPSs) have become popular around the world due to their higher heating and/or cooling coefficient of performance (COP) in comparison to regular air-source heat pump systems [1–3]. GSHP systems have been rapidly developing in Germany, USA, China, and elsewhere in recent years. In a GSHPS, the ground is commonly used as the heat source and/or sink in order to meet building's heating and/or cooling load requirements. However, the amount of heat injected into or drawn from the ground should be balanced on an annual basis to ensure that ground temperature does not change in the long-term. If this annual energy balance was not maintained, ground temperature might rise or drop, which could result in failure of GSHPSs. In the severely cold regions, heating requirement is much higher than that of cooling. An auxiliary heat source, such as a boiler or an electric heater, must be used to avert this annual energy imbalance. Solar energy, as a green and renewable energy, could be the ideal auxiliary energy source used in such system.

Several solar assisted ground-source heat pump systems (SAGSHPSs) used under various conditions appeared in literatures in recent years [4–8]. Generally, the solar thermal energy is stored in the ground during non-heating season, and extracted by a ground-source heat pump (GSHP) for heating in winter. SAGSHPSs are suitable for heating (water heating and/or space heating), but have limited application for space cooling [9,10]. It is difficult to design an exact solar system to complement the mismatch between heating load and cooling load, particularly for an office building, whose heating or cooling load is uncertain because of the number of occupants and the duration of occupancy. This paper presents a novel hybrid solar ground-source heat pump system (HSGSHPS) for heating and cooling of an office building. The performance of the overall HSGSHPS was predicted under the long-term multi-year manner.

2. Components and system

2.1. Building and system outline

The Energy Conservation Laboratory Centre (ECLC) is a research and office building as well as a renewable energy demonstration building is located in the new campus of Hebei University of Technology in Tianjin, China. The location of ECLC is 39.238°N and 117.066°E. ECLC has four stories above ground with a total floor area of 4953.4 m^2. There is a hollow space in the centre of the building from the second floor to the roof. The building is almost orientated in the north-south direction with a counter-clockwise rotation of 21° from north.

A HSGSHPS, consisting of a GSHPS and a SAGSHPS, was designed to meet the ECLC's heating and cooling requirement. The same solar

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doi:10.1016/j.enbuild.2011.12.035

Nomenclature

A_c	solar aperture area (m^2)
c	specific heat capacity (kJ/(kg K))
CLL	collection lower limit (°C)
COP	coefficient of performance
CUL	collection upper limit (°C)
ΔT	temperature difference (°C)
FOT	the first-operation time (h)
H_T	global radiation incident based on aperture area of the tilted collector surface (kJ/m^2)
M	mass flow rate (kg/s)
P	power (kW)
Q	thermal energy (kJ)
R	ratio
SF	storage factor
SLL	storage lower limit (°C)
SUL	storage upper limit (°C)
T	temperature (°C)
t	time (s)
W	electrical energy consumption (kJ)

Greek symbols

η_c	thermal efficiency of the collector based on aperture area (%)
η_{loss}	the loss rate (%)

Subscript

BTES	borehole thermal energy storage
C	cooling mode
H	heating mode
HL	heating load
HP	heat pump
L	load
S	source
SC	solar collector
SH	solar heating
ST	storage tank
c	cooling season
h	heating season
in	inlet
inj	injection
out	outlet
s	storage season
scp	storage circulation pump
sys	system
tot	total

collector system could supply the domestic hot water for students' shower facility and others throughout the year. The schematic of the HSGSHPS is shown in Fig. 1. Two GSHP units were used in the HSGSHPS. The first GSHP unit was used in a pure GSHPS to supply the entire building's cooling load requirement in the summer and partial heating load requirement in the winter. The second GSHP unit was used, coupled with a solar seasonal thermal storage system, in a SAGSHPS to supply the remaining heating load requirement of the building. A borehole heat exchanger (BHE) and a borehole thermal energy storage (BTES) were designed corresponding to two GSHP units. The BTES in SAGSHPS was very different from the BHE in GSHPS (to be discussed in later section).

In this system, the GSHPS was relatively simple while the SAGSHPS was more complex. As shown in Fig. 1, the SAGSHPS consisted of a solar collector (SC), a hot water tank (HWT), a solar circulation pump (P1), a borehole thermal energy storage (BTES), a heat pump (HP), a storage circulation pump or source circulation pump (P2), a load circulation pump (P3), fan-coils in the fourth floor (FC4) and valves. All the working fluid in the HSGSHPS was water. There were five loops in the SAGSHPS while there were two loops in the GSHPS. The water could be supplied through water treatment equipment (not shown in Fig. 1). All the loops could be controlled by motorized valves and pumps. Solar seasonal storage loop and the ground heat extraction loop had the same pump. But the pump circulated the fluid through the BTES with reverse direction controlled by the valves V3, V4, V8 and V9. During non-space-heating seasons (i.e., the cooling season (summer) and two shoulder seasons (spring and autumn)), solar thermal energy was stored in ground via the storage loop. Thus, the non-heating season is also called storage season for SAGSHPS. In winter, the heating is provided by the fan-coils with the hot water coming from solar storage tank if the temperature was high enough, or else with the hot water heated by the HP2 (using the stored solar thermal energy in ground). The SAGSHPS was designed as a direct system in order to improve the heat transfer efficiency. Water, as the working fluid in the whole HSGSHPS, could flow through the SC, the HWT, the BTES and heating distribution system. A serpentine tube heat exchanger was immersed in the storage tank. City water was heated by flowing through the serpentine tube heat exchanger for the production of domestic hot water.

2.2. Key issues of the HSGSHPS design

To design the HSGSHPS, some of the key issues should be considered as follows.

2.2.1. Determine the load of the GSHPS

Determining the load of the GSHPS is the first step to design a HSGSHPS. In order to keep the ground temperature unchanging on an annual basis, one must estimate the amount of heat extracted from and injected into the ground through the BHE. However, it is difficult, or even impossible, to design the heating and cooling system with an exact balance because of many uncertainties including the following: HP cannot always operate under the rated condition resulting in varying coefficient of performance of HP unit (COP_{HP}); building's heating or cooling load cannot be determined exactly, the air conditioning terminal unit can be turned on/off by occupants in different rooms; and extreme weather conditions will affect the building's heating and cooling load.

In order to keep the ground temperature unchanging on an annual basis, the heat extraction should be equal to the injection of the BHE. This can be estimated based on the COP_{HP} of GSHP unit in heating mode and cooling mode. The COP_{HP} in cooling mode is defined theoretically as,

$$COP_{HP,C} = \frac{Q_{L,C}}{W_{HP}} = \frac{Q_{L,C}}{Q_{S,C} - Q_{L,C}} \quad (1)$$

The COP_{HP} in heating mode is defined as,

$$COP_{HP,H} = \frac{Q_{L,H}}{W_{HP}} = \frac{Q_{L,H}}{Q_{L,H} - Q_{S,H}} \quad (2)$$

The ratio of heating load to cooling load on an annual basis can be defined as,

$$R_L = \frac{Q_{L,H}}{Q_{L,C}} \quad (3)$$

And the ratio of heat extraction to injection of the BHE on an annual basis can be defined as,

$$R_S = \frac{Q_{S,H}}{Q_{S,C}} \quad (4)$$

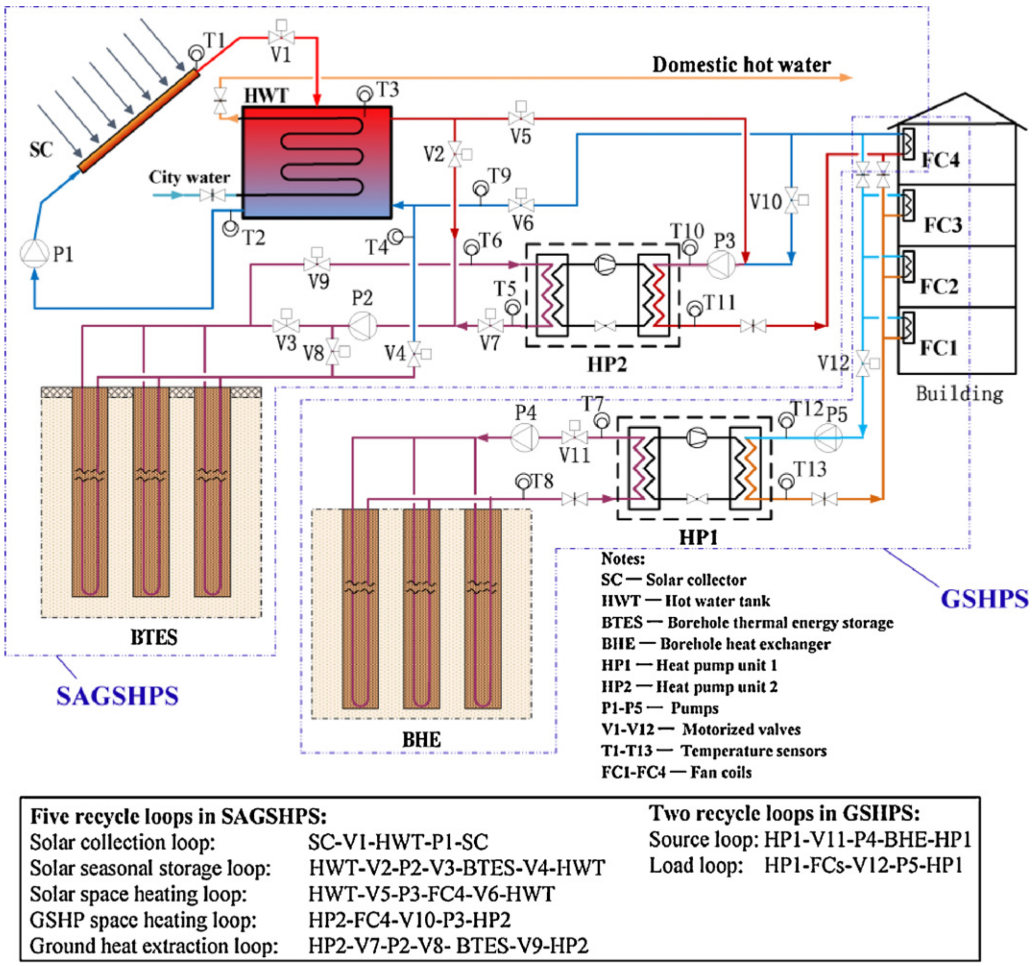

Fig. 1. Schematic diagram of the HSGSHPS.

Thus, the relationship of R_L and R_S is shown as,

$$R_L = \frac{COP_{HP,H}(COP_{HP,C}+1)}{COP_{HP,C}(COP_{HP,H}-1)} R_S \tag{5}$$

The rated COP_{HP} of the HP1 (model: PSRHH1201-Y manufactured by Climaveneta (China) company) is 4.53 in heating mode and 5.22 in cooling mode. To guarantee the heat extraction equal to the heat injection, i.e. R_s is 1 in Eq. (5), the ratio of heating load to cooling load on an annual basis should be,

$$R_L = \frac{COP_{HP,H}(COP_{HP,C}+1)}{COP_{HP,C}(COP_{HP,H}-1)} = \frac{4.53 \times (5.22+1)}{5.22 \times (4.53-1)} = 1.53$$

That is to say, the ground thermal energy can be kept balance on an annual basis (i.e., the cyclic temperatures of BHE can be repeated yearly) if the heating load is 1.53 times of the cooling load. Simulation results showed that ECLC's heating load of is 2.4 times of its cooling load. The auxiliary energy has to be used in this system for space heating. According to the load calculation, the sum of the heating load of the first, second and third floor is 1.56 times of the total cooling load. So, the heating load of the fourth floor is separated from the GSHPS, and supplied by the SAGSHPS. Furthermore, some fan-coils can be switched to connect with the GSHPS or SAGSHPS. This strategy may be able to minimize/eliminate the effect of the discrepancy between the actual load and the designed load of GSHPS, if the above fan-coils are switched between GSHPS and SAGSHPS in heating process based on the actual requirement.

2.2.2. Determine the solar collector area and the hot-water-tank volume

The performance of underground thermal storage of SAGSHPS depends strongly on the matching between the water tank volume and the area of solar collectors [11]. It is very important for the HSGSHPS's design to determine the solar collector area and the hot water tank volume. All the auxiliary heat is supplied by SAGSHPS. The space heating can be derived from three different ways: solar thermal energy in winter, seasonal thermal storage in ground extracted by heat pump, and electricity power conversion of the HP. There is a basic principle to determine the minimum solar collector area. That is, the solar thermal storage should maintain the ground temperature on an annual basis. It can be shown as an equation as following:

$$Q_{SH} + Q_{HP} \geq Q_{HL} \tag{6}$$

where, Q_{SH} is heating through solar thermal energy in heating season; Q_{HP} is heating through HP; Q_{HL} is total required heating load provided by the SAGSHPS. Q_{HL} can be obtained from the design standards manual or from simulation results. Q_{SH} and Q_{HP} can be calculated from the following equations.

$$Q_{SH} = A_c H_{T,h} \eta_{c,h} (1 - \eta_{loss,h}) \tag{7}$$

$$Q_{HP} = \frac{COP_{HP,h}}{COP_{HP,h}-1} A_c H_{T,s} \eta_{c,s} (1 - \eta_{loss,s}) \tag{8}$$

In the proposed HSGSHPS, a total of 280 m^2 of the evacuated tube solar collector area was determined using Eq. (6) and installed, accounting for the requirements of space heating, domestic hot water and some research activities.

The principle to determine the volume of the hot water tank is that the maximum temperature of the solar collector never exceeds 90 °C. The collector temperature is affected by the hot water, tank water temperature and the solar irradiation. The tank water temperature is also affected by the borehole storage temperature or space heating set-point temperature. In fact, the solar irradiation in heating season is much lower than that in storage season. The tank volume was designed in terms of thermal energy storage conditions. The storage turn-on temperature (the tank temperature) was assumed to be 50 °C, and the average heat transfer rate per unit borehole length during storage season was 50 W/m. The volume of the hot water tank can be determined based on heat gain of solar collector and heat storage in ground per day in storage season. The maximum local global irradiation per day in Tianjin is 32.5 MJ/m^2 (the peak irradiance is 1193 W/m^2 on collector surface). The volume of the hot water tank was designed as 20 m^3 as a result. The maximum possible temperature of the collector was to be 86 °C.

2.2.3. Design the BHE for GSHPS and BTES for SAGSHPS

The BHE/BTES always consists of several tens of high-density polyethylene U-pipes buried in vertical boreholes. Based on previous experience, the BHE for GSHPS was formed by 66 vertical boreholes with individual depth of 120 m and spacing of 4 m apart. The BTES for SAGSHPS should consist of boreholes with small spacing and shallow depth for the purpose of seasonal storage. The BTES was formed by 25 vertical boreholes with individual depth of 50 m and spacing of 2.5 m apart. The number of boreholes of the BHE/BTES should match the HP's capacity. To decide the number of boreholes of BHE or BTES, different operation characteristics between GSHP and SAGSHP should be considered. The former should be based on cooling load, whereas the latter on heating load. Another difference between the BHE and BTES was the connection relationship of boreholes. In the BHE, all boreholes were connected in parallel, however, in the BTES, there were 3 groups of boreholes connected in series with 8 or 9 boreholes connected in parallel in every group. The fluid was circulated through the boreholes in the following manner to optimize the overall system performance. The fluid, water, should circulate from centre to border of the storage volume during heat injection process, while circulating from border to centre during heat extraction process. Moreover, the top of the BTES was covered with polyurethane foam insulation of 48 mm thickness to reduce heat loss.

Other components such as fan-coils, circulation pumps, HP units, fresh air units, could be determined using the same principle when designing a conventional heating and cooling system. This has been introduced and covered in many existing manuals or literatures; therefore, it is unnecessary to give detail about them in this paper.

3. Simulation

3.1. Simulation program

To simulate the combined system with solar collector and ground-source heat pumps, TRNSYS 17 (Transient Systems Simulation Program) [12] has been used. A numerical model was created for the HSGSHPS. A simplified hydraulic scheme of HSGSHPS in TRNSYS is shown in Fig. 2. The building was simplified into four zones with one zone per floor but the walls and windows remained the same as the actual building using the component model Type56. Correspondingly, four fan-coil models (Type 928) were used in the system to simulate the dozens of fan-coils in four floors with the same flow rate and power. The GSHP unit was modelled using component Type 927. An evacuated tube solar collector model (Type 71) and a flat bottomed storage tank model (Type 531) were used for the solar collection system. The weather model of Type 109 was

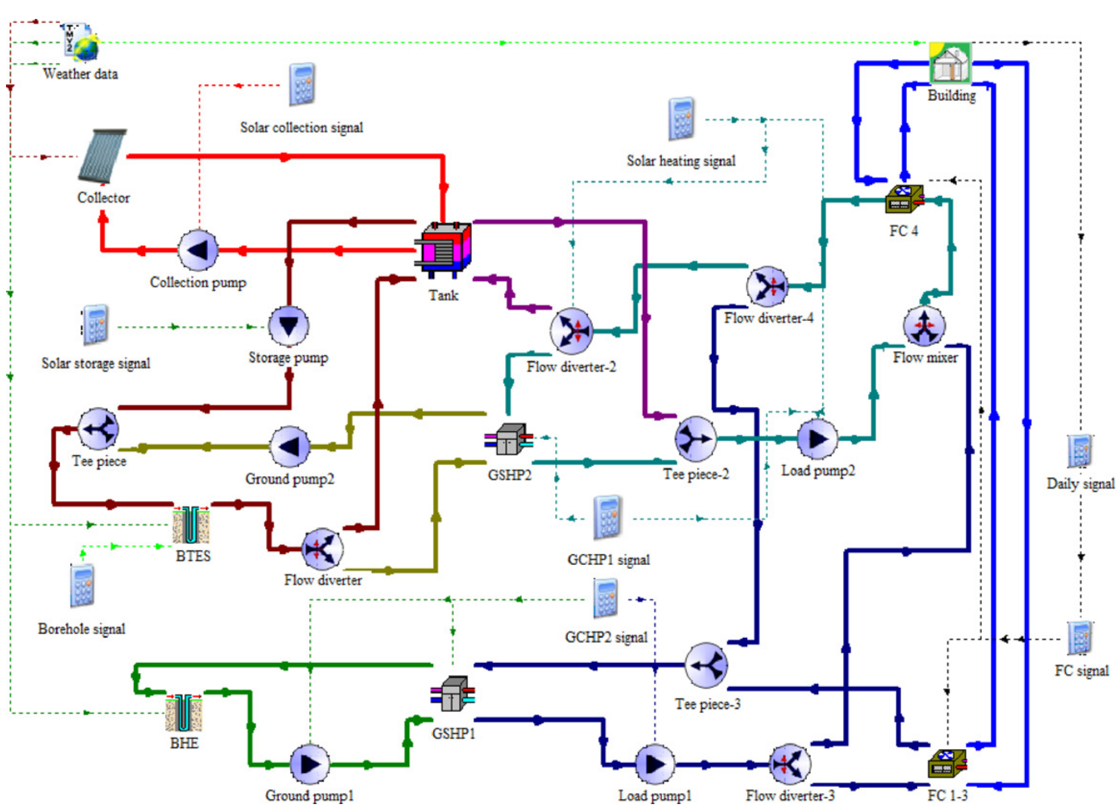

Fig. 2. Simplified hydraulic scheme of HSGSHPS in TRNSYS.

Table 1
The parameters of GSHPs in rated condition.

GSHP	HP in GSHPS	HP in SAGSHPS
GSHP type	PSRHH1201-Y	HRHH0252
The rated source entering/exiting temperature in cooling mode (°C)	25/30	–
The rated load entering/exiting temperature in cooling mode (°C)	12/7	–
The rated source entering/exiting temperature in heating mode (°C)	10/5	7/3.8
The rated load entering/exiting temperature in heating mode (°C)	40/45	40/45
The rated source flow rate/load flow rate in cooling mode (m^3/h)	65.6/55.6	–
The rated source flow rate/load flow rate in heating mode (m^3/h)	46.2/58.3	16.4/14.2
The rated cooling capacity/cooling power (kW)	323.2/61.9	–
The rated heating capacity/heating power (kW)	339.2/74.9	82.8/19.7

used. The weather data of Tianjin, China, were used in the simulation model. The domestic hot water was not been included in this simulation model. The solar collector with 280 m^2 of aperture area was used for seasonal storage and space heating.

3.2. HP model

Manufacturer's specifications of the GSHPs such as the flow rate, power, and capacity at the rated condition are shown in Table 1. The power and capacity of the GSHPs vary with flow rate, entering water temperature (EWT) of evaporator and EWT of condenser. The EWT of evaporator is the load EWT in cooling mode, while the EWT of condenser is the load EWT in heating mode. Two user-supplied data files were created as external files for Type 927 containing catalogue data for the capacity and power draw, based on the entering load and source temperatures. Cooling or heating capacity and power could then be read from those two external files during simulation.

3.3. BHE and BTES model

The component Type 557 was used to model the BHE and BTES. The soil thermal physical properties were set in the BHE and BTES models based on the experimental data. The thermal physical properties of soil changed with the depth because of the variation of the water content in and composition of soil with depth. The density, specific heat capacity and thermal conductivity of the soil varied with the soil depth are shown in Fig. 3. The limit of the number of layers was ten for the component Type 557. A simplified soil model was used in the simulation based on thermal conductivity with ten layers along depth direction. The undisturbed temperature distribution in the ground of 0–120 m depth could be plotted into three partitions: environmental impact layer (0–20 m below the surface), constant temperature layer (20–30 m of depth) and varying temperature layer (30–120 m). The undisturbed temperature in ground of 20–30 m depth was constant at 13.0 °C, while it increased at a rate of 0.3 °C/m with depth after 30 m deep. The average initial temperature of the ground was 14.1 °C between 0 and 120 m depth stage.

3.4. Simulation process

The studied building is a place for graduate students and faculty staff to conduct research work or study. The operating time of HSGSHP system was different in weekdays, weekends, and holidays. The distinction between daytime and night time schedules was considered as well. During daytime of the weekday, there were 130 occupants in the building, and the system operated from 8:00 a.m. to 6:00 p.m. normally. However, at night time of the weekday from 6:00 p.m. to 10:00 p.m., half of the number of daytime occupants in the building were considered, and the fan-coils were turned on or off according to the floor's occupancy. The system was turned off during other times from 10:00 p.m. to 8:00 a.m. The system operated according to half occupancy in the building during the weekends from 9:00 a.m. to 9:00 p.m., otherwise the system was turned off. Two holiday periods (January 25 to February 20 and July 22 to August 29) per year were modelled in this system too. The number of occupants and the operating time during the holidays were assumed to be same as weekends but the operating fan coils to be 1/3 of normal time.

Generally, the load circulation pump was turned on based on the schedule described above. In order to evaluate the energy-saving potential of this operating schedule, a contrasting case of an always-on case of the load circulation pump (i.e., never turn off during the heating season or cooling season) was simulated.

Each year was split into four seasons: heating season, cooling season and two shoulder seasons, as shown in Fig. 4. The heating season was set from 0:00 on November 15 to 0:00 on March 16 of the following year (i.e., a total of 121 days), while the corresponding simulation time was from 7632 h to 10,536 h (1776 h from 0:00 on January 1 of the following year) accordingly. The cooling season was set from 0:00 on May 20 to 0:00 on September 8 (111 days), while the corresponding simulation time was from 3336 h to 6000 h accordingly. The remaining time was for the two shoulder seasons respectively, with no heating and cooling.

The average temperature of BHE or BTES would vary with the first-operation time (FOT) of the HSGSHPS. The average ground temperature would increase or decrease according to the first operation time in winter or summer. In this paper, the FOT was at the end of the heating season (1776 h from 0:00 on January 1) in the base case and other cases if there was no special indication. The suitable FOT of the HSGSHPS was also studied through contrasting the results of three different FOTs. The schematic of simulation time and the three FOTs tested in this paper are shown in Fig. 4. For the SAGSHPS, its purpose is for heating only, so the end of its annual operation time was defined as the same as the end of the heating season.

This paper predicted the coefficient of performance of the systems (COP_{sys}) and of HP units (COP_{HP}) under various control strategies of the solar circulation pump and storage circulation pump. Several temperatures or temperature differences were used for controlling the solar collection loop and/or solar seasonal storage loop. The control signals were water temperature difference between solar collector outlet and inlet (ΔT_{SC}), temperature of tank outlet in the storage loop ($T_{ST,out}$), and temperature difference between tank outlet and tank inlet in the storage loop (ΔT_{ST}).

Temperature difference ΔT_{SC} and ΔT_{ST}, can be defined as

$$\Delta T_{SC} = T_{SC,out} - T_{SC,in} \tag{9}$$

$$\Delta T_{ST} = T_{ST,out} - T_{ST,in} \tag{10}$$

where $T_{SC,out}$, $T_{SC,in}$, $T_{ST,out}$ and $T_{ST,in}$ are represented by the temperature sensors T_1, T_2, T_3 and T_4 respectively as shown in Fig. 1.

The control strategies of the system can be studied through various operating limits of the pump. A pump would be turned on and operated between the operating limits. Four limits are defined for the control strategy: collection upper limit (CUL), collection lower limit (CLL), storage upper limit (SUL) and storage lower limit (SLL). CUL is the upper limit of ΔT_{SC}, while CLL is the lower limit of ΔT_{SC}. SUL is the upper limit of $T_{ST,out}$, while SLL is the lower limit of ΔT_{ST}.

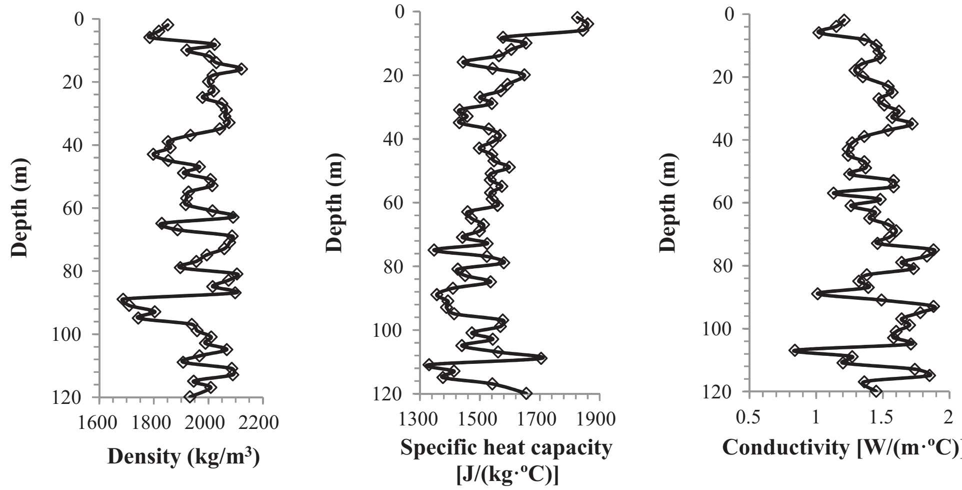

Fig. 3. Thermal physical parameters of the soil with different depths of boreholes.

The COP_{sys} is defined as,

$$COP_{sys} = \frac{Q_L}{W_{tot}} \tag{11}$$

Q_L can be calculated from the heating and/or cooling heat transfer from the fan-coils to indoor return air in a computation period. For the SAGSHPS, it is the total heating from solar and HP2. W_{tot} is the total electrical energy consumed in a computation period by all of the pumps, fans, compressors and so on. The computation periods can be one year, cooling season or heating season. For the SAGSHPS, the computation periods should be on a yearly basis even though it is used only for space heating. The electricity consumption in SAGSHPS for calculating the COP_{sys} should include that of the annual solar circulation pump (P1 in Fig. 1), the storage circulation pump (P2 in Fig. 1), the source circulation pump (P2 in Fig. 1), the load circulation pump (P3 in Fig. 1), the compressor of HP1 and the 4th-floor fan-coils. The COP_{sys} defined here is an average value in the computation period. It is not a power rate but an energy rate.

Similarly, the COP_{HP} is defined as,

$$COP_{HP} = \frac{Q_{L,HP}}{W_{HP}} \tag{12}$$

A base case was simulated based on CUL = 15 °C, CLL = 3 °C, SUL = 50 °C and SLL = 3 °C. Other cases were different from the base case with varying only one of the four control limits of CUL, CLL, SUL or SLL, while other parameters were kept constant as those in the base case when the suitable control strategies of the solar circulation pump and storage circulation pump were sought. The load circulation pump control strategy (LCPCS) and the FOT = 1776 h were used for the base case.

4. Results and discussion

4.1. Results of the base case

The average temperature of the BHE and BTES, COP_{HP}, COP_{sys}, heat loss and heat extraction of the BTES were studied. Figs. 5–7 illustrate the results of 15-year operating characteristics of the proposed HSGSHPS for the base case. The average temperature of the BHE or BTES refers to the average temperature of the ground in the storage volume. The ratio of heat loss to heat storage (LRS) and the ratio of heat extraction to heat storage (ERS) in BTES were defined to describe the characteristics of BTES. The heat loss of BTES is the total heat lost from the sides, bottom and top of the storage volume to the surrounding ground or the ambient. The heat extraction is the heat transferred from the source fluid to the HP2. The heat storage is the energy removed from the tank through the specified outlet port minus the energy added to the tank through the corresponding inlet port of the seasonal storage loop.

Fig. 5 illustrates that the average temperature of the BHE or BTES varies with the operating time annually in a periodical manner. The temperature of BTES increased annually because of the solar seasonal storage. The increase of the temperature in storage season was larger than the reduction in heating season of every year. The

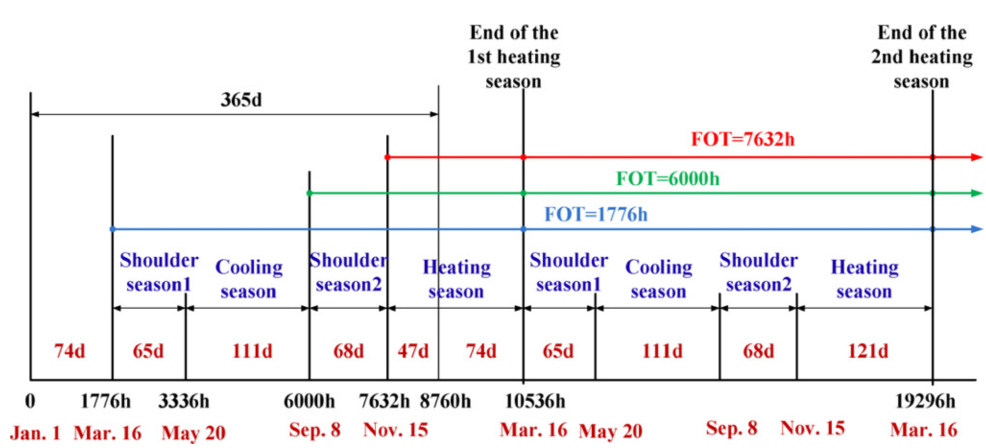

Fig. 4. The schematic of simulation time and FOT.

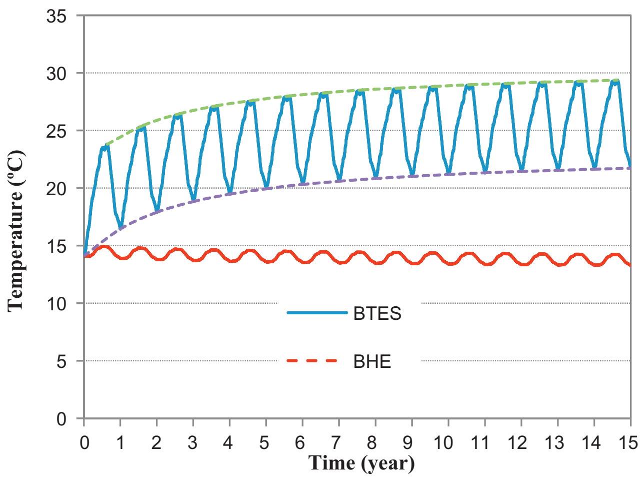

Fig. 5. Average temperature of the BHE and BTES for base case.

average temperature of the BTES increased by 9.7 °C in the storage season but decreased by 7.4 °C in the heating season during the first year. Thus the temperature of BTES increases by 2.3 °C after first year. Fig. 6 illustrates that the heat loss and heat extraction of the storage increased mostly in the first 5 years. It resulted in small internal energy variation of the BTES which would cause its temperature to increase annually. The ratio of internal energy variation to heat storage (1 – (LRS + ERS) in Fig. 6) was less than 10% of the heat storage after nine years. Thus the average temperature of the BTES increased at a rate of less than 0.2 °C a year after the first eight years.

Fig. 7 illustrates the annual COP variation with simulation time (year). The SAGSHPS operated only in heating mode. The GSHPS was responsible for the total cooling load of the building. So, the COP_{sys} of SAGSHPS represented heating mode only, while the COP_{sys} of GSHP in cooling mode was the same as that of HSGSHPS in cooling mode. The COPs of HSGSHPS shown in Fig. 7 represent whole year of heating and cooling modes. The COP_{HP} of SAGSHPS increased with the average temperature in BTES. The annual COP_{HP} of SAGSHPS and average temperature of BTES showed similar trend with the simulation time. COP_{sys} of SAGSHPS or HSGSHPS increased as a result of the increase of COP_{HP}. However, the range of increase was different. Annual COP_{HP} of SAGSHPS increased by 11.3% and the COP_{sys} of SAGSHPS improved by 7.5% after 15 years, while annual COP_{sys} of HSGSHPS increased by 1.3% for the same period. The increases were mainly due to the increase of temperature of BTES annually.

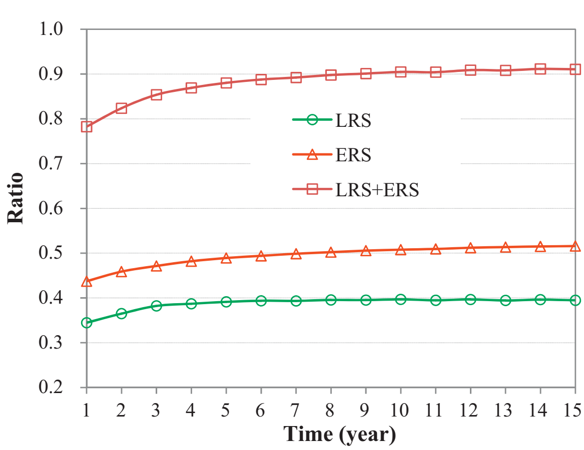

Fig. 6. Annual heat loss and heat extraction from the BTES for base case.

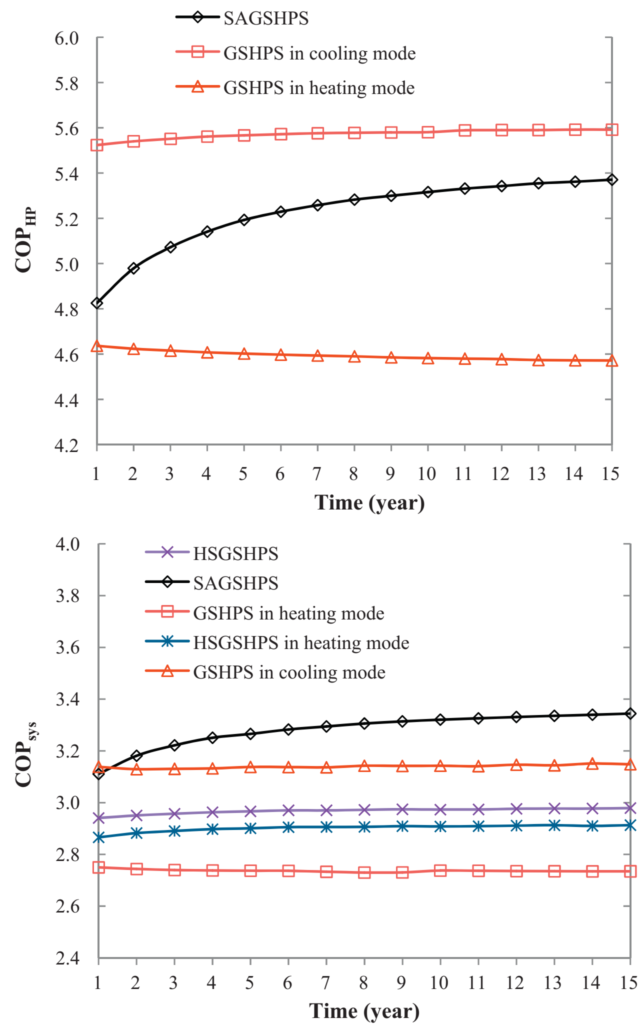

Fig. 7. Annual COP_{HP} and COP_{sys} variation with operating time for base case.

As mentioned above, the designed R_L was a little larger than it should be. There was a little more heat extraction from than injection into the BHE of GSHPS. The temperature of BHE reduced very slightly year over year (0.8 °C after 15 years) as shown in Fig. 6. The variation of the temperature of BHE in an annual period was small too (only 0.8 °C), due to the large spacing (4 m) between boreholes. The COP_{sys} of GSHPS hardly changed year over year. The results verified that the distribution of the heating load for GSHPS and SAGSHPS in the proposed HSGSHPS was acceptable. All in all, the proposed HSGSHPS had the capability of resolving the problem of the imbalance of extraction and injection as well as increasing the overall COP of the system.

4.2. Solar collection control strategy

Solar collection control strategy of the SAGSHPS has influence on the COP_{sys}, since the solar collection control strategy affects the useful energy capture and utilization of the collectors. The higher the CUL, the higher the heat loss in solar collector system is. Additionally, the less the useful energy capture of the collector, the less the storage thermal energy in the BTES is, resulting in lower COP_{HP} of SAGSHPS. On the other hand, too low CUL results in higher pumping electrical energy consumption needed to transfer solar gain from the collector to the water tank. An optimum CUL must exist. However it has not been systematically investigated in current research. Fig. 8 illustrates that the COP_{sys} of SAGSHPS varies with CUL or CLL.

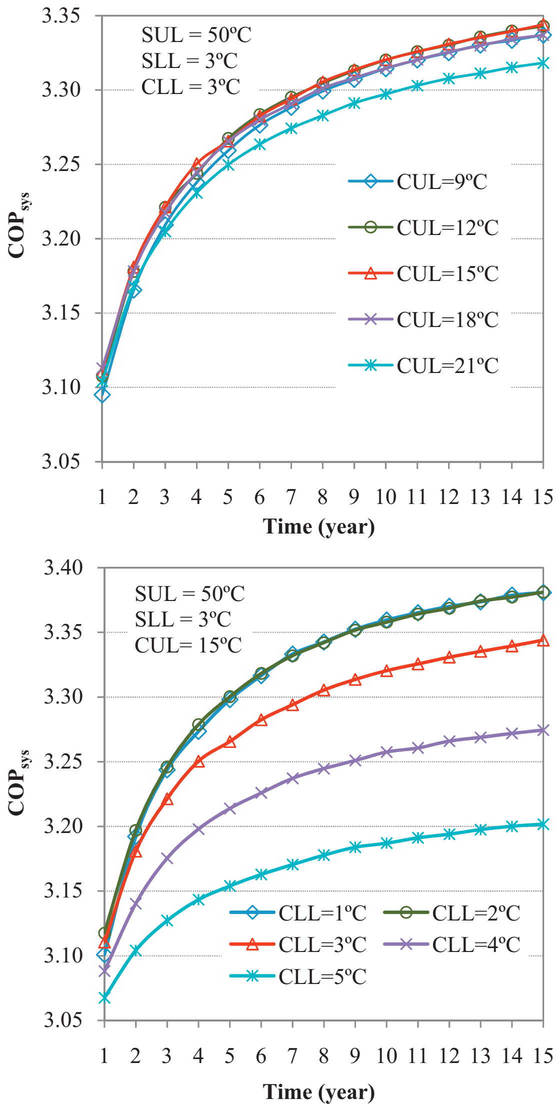

Fig. 8. Annual COP_{sys} of SAGSHPS variation with CUL or CLL.

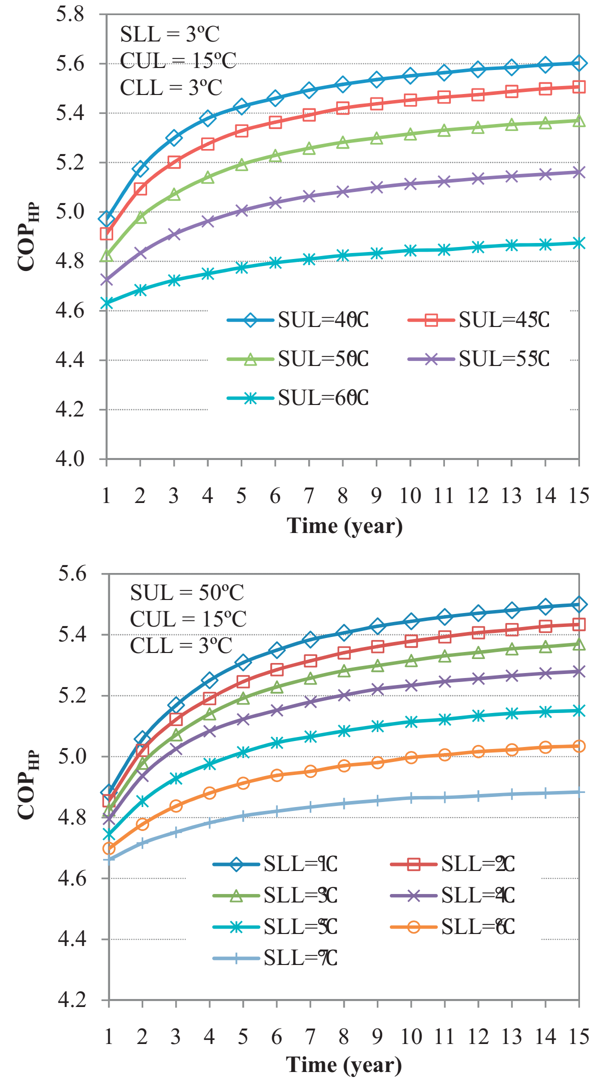

Fig. 9. COP_{HP} of SAGSHPS variation with SUL or SLL.

In these studied cases, all control strategies are same as those of the base case except the studied parameter of CUL or CLL. Fig. 8 shows that the suitable CUL is in the range of 12–18 °C according to the largest COP_{sys}, while the suitable CLL is in the range of 1–2 °C. The COP_{sys} is much more sensitive to CLL rather than CUL.

4.3. Solar storage control strategy

We defined a parameter called storage factor (SF). This is the ratio of heat injection, Q_{inj}, to the electrical energy consumption of the storage circulation pump, W_{scp}. It can be expressed in an equation as,

$$SF = \frac{Q_{inj}}{W_{scp}} \tag{13}$$

Obviously, SF affects the COP_{sys} of SAGSHPS positively. Q_{inj} and W_{scp} can be computed for a time period between t_1 and t_2 by the following equation,

$$Q_{inj} = \int_{t_1}^{t_2} cM(T_{BTES,in} - T_{BTES,out})\,dt = \int_{t_1}^{t_2} cM\,\Delta T_{BTES}\,dt \tag{14}$$

$$W_{scp} = \int_{t_1}^{t_2} P_{scp}\,dt \tag{15}$$

In the SAGSHPS, the specific heat capacity of water, c, the mass flow rate of water, M, and the power of storage circulation pump, P_{scp}, are regarded as constant. Thus, SF is in proportion to the temperature difference between the inlet and outlet of BTES, ΔT_{BTES}. Additionally, Q_{inj} is affected by the inlet temperature to BTES, $T_{BTES,in}$. The higher $T_{BTES,in}$ is, the higher the temperature difference between hot water and ground, thus the higher the Q_{inj}. If the heat loss of the pipes between the hot water tank and the inlet of BTES was ignored, $T_{BTES,in}$ would be regarded as the same as $T_{ST,out}$. So, high SUL or SLL results in high storage factor.

However, SUL or SLL affects the solar collector efficiency as well as the storage factor. High SUL or SLL results in low solar useful energy gain, low heat storage (heat injection) and low COP_{HP}. Fig. 9 shows the annual average COP_{HP} increases annually, but reduces as SUL or SLL increases. Lower COP_{HP} causes lower COP_{sys}, at the same time, higher SF results in higher COP_{sys}. The solar storage control strategy has influence on COP_{sys} similar to the solar collection control strategy does. There is a suitable SUL and SLL corresponding to the maximum COP_{sys}. Fig. 10 shows that the suitable SUL is about 50 °C, and SLL is about 5 °C.

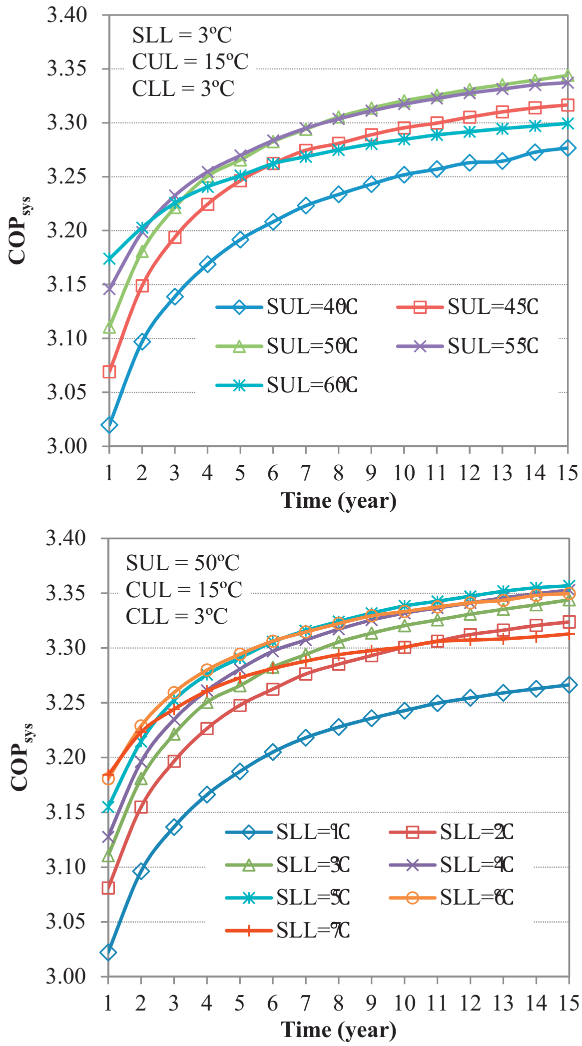

Fig. 10. COP$_{sys}$ of SAGSHPS variation with SUL or SLL.

4.4. Load circulation pump control strategy

In some cooling and heating systems, the load circulation pumps (LCPs) are operating under always-on conditions. It causes needless waste of electrical energy. In the proposed HSGSHPS, load circulation pump control strategy (LCPCS) was used. The load circulation pump (LCP) was turned off if no fan-coil was operating. The electrical energy consumption of the LCP would decrease and COP$_{sys}$ would increase under LCPCS. The energy saving rate of the total system is shown in Fig. 11 based on the LCP operating under LCPCS rather than always-on. If the LCPCS was used, about 32% of total electrical energy consumption could be saved in the proposed HSGSHPS. The potential electrical energy saving rate was only 15–17% for the SAGSHPS, while it was over 47% in the GSHPS in cooling mode.

4.5. The first-operation time

Since the ground temperature affects the COP of HP, the first-operation time (FOT) of the system will have an impact on the COP of HP, hence the COP of system. Three cases of FOT were studied: the heating season end time (1776 h), the cooling season end time (6000 h), and the heating season start time (7332 h).

The minimum temperature of the evaporator's outlet is determined based on the source fluid in order to protect the HP unit. The

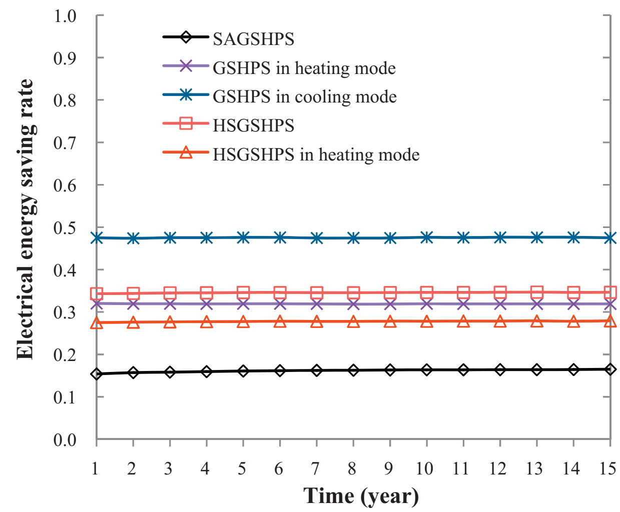

Fig. 11. Electrical energy saving rate in system, contrasting LCPCS with LCP always-on (SUL = 50 °C, SLL = 3 °C, CUL = 15 °C and CLL = 3 °C).

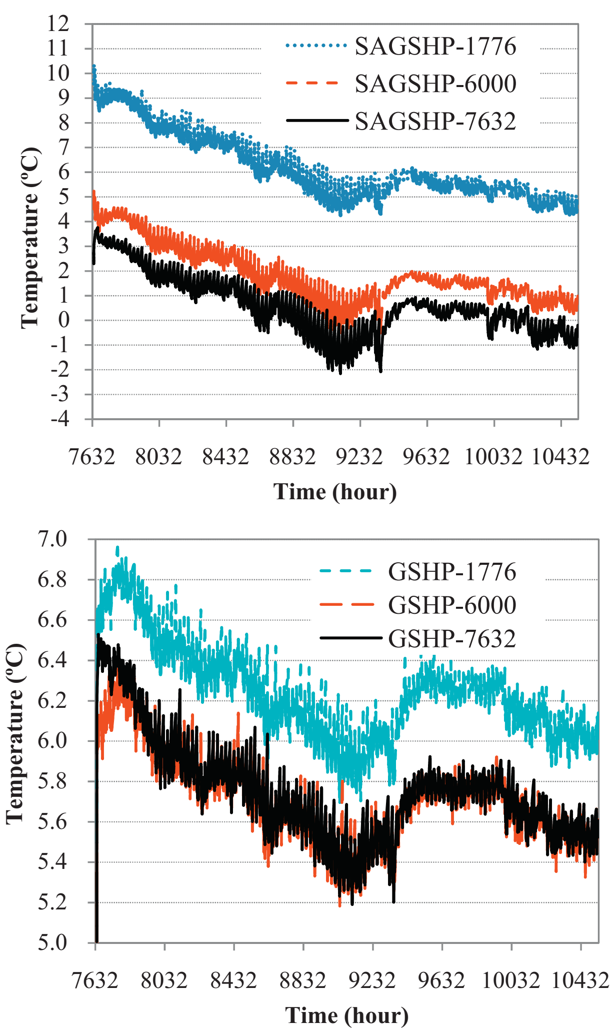

Fig. 12. Comparison of fluid temperatures out of evaporators of HP under different FOT.

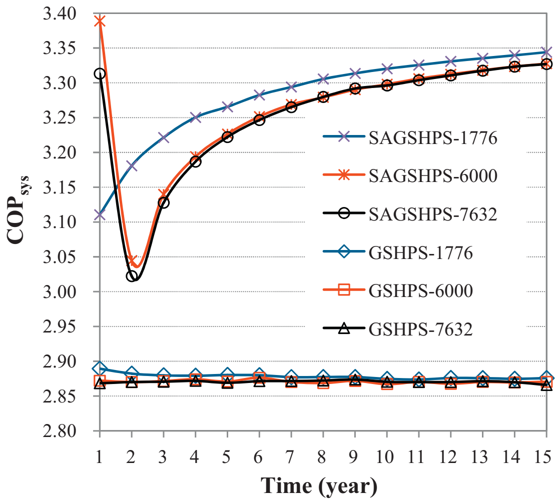

Fig. 13. COP_{sys} of SAGSHPS and GSHPS under different FOT.

fluid used in U-shape pipes of the boreholes of the proposed system is water. The minimum limited temperature is 5 °C based on the manufacturer's specification. The minimum temperature can drop to −8 °C, if the water is mixed with antifreeze (such as ethylene alcohol).

Fig. 12 shows the evaporator's outlet temperatures of SAGSHP and GSHP during the heating season of the first year. All the temperatures during the first heating season reach below the minimum limit of 5 °C for SAGSHP when the FOT is 7632 h or 6000 h. The minimum temperatures can even reach −2 °C if there is no previous heat storage from the solar loop. There are a few instances that the temperature drops below the limit of 5 °C for SAGSHP when the FOT is 1776 h. However, the evaporator's outlet temperatures are never lower than the 5 °C limit for GSHP. This is because of the large spacing of the boreholes (4 m) in the BHE of GSHPS comparing to the smaller spacing of the boreholes (2.5 m) in the BTES of SAGSHPS. Small spacing does not provide enough energy storage between boreholes, so the ground temperature drops quickly.

SAGSHPS's main purpose is to provide heating, so the end of the heating season means the end of the calculating year. Energy consumption of the first year is computed from the FOT to the end of the heating season, otherwise the following years are computed based on a whole year (365 d) (as shown in Fig. 4). So, the COP of SAGSHPS with FOT of 1776 h shown in Fig. 13 is very different for the first year compared to the rest when FOT is 6000 h or 7332 h. The COP_{sys} of SAGSHPS in the first year is large if it has no heat storage or partial heat storage because there is no or little solar collection and storage pump consumption. But the heat storage is necessary to guarantee the normal operation of SAGSHPS in the next heating season. GSHPS provides both heating and cooling, so the end of the computation year is a full 365 d (the beginning is also the end time).

Fig. 13 shows that the FOT has a minor influence on COP_{sys} of GSHPS or SAGSHPS. However, applying thermal injection before extraction, in general, enhances COP_{sys} based on multi-years stage. Heat extraction following heat injection (FOT = 1776 h) only saves 0.3% of electrical energy consumption in GSHPS compared to the case with FOT is 7632 h over the 15 year period. Similarly, in SAGSHPS, heat extraction following heat injection only saves 0.8% of electrical energy consumption.

In general, enough prior thermal storage prevents HP malfunctioning due to too low ground temperature and evaporator temperature in the case of small spacing of boreholes used in the BTES. Thermal injection to the ground before extraction is favourable to HP operation even for the GSHPS which has minor effect due to the FOT of the system.

4.6. Results of suitable case

According to the analysis above, a suitable case of the proposed HSGSHPS was simulated under the control strategy of SUL = 50 °C, SLL = 5 °C, CUL = 15 °C and CLL = 2 °C.

Fig. 14 shows that the average temperature in BTES increases with year. Ground temperature increases quickly in the first 5 years, and then increases gradually. Ground temperature only increases by 0.4 °C from the 15th to 25th year. The average temperature in BHE decreases gradually and very slowly. It decreases 1.0 °C after 25 years. This estimated change of soil temperature in BHE hardly affects the characteristic of GSHPS in the long run. Some literatures reported problem of operation in SAGSHPS for cooling and heating using only one borehole heat exchanger. Wang et al. [9] found that the heat storage capacity should be reduced appropriately to make the ground heat balance after 3 years of running. Rad et al. [10] also detected system malfunctioning in the cooling season because

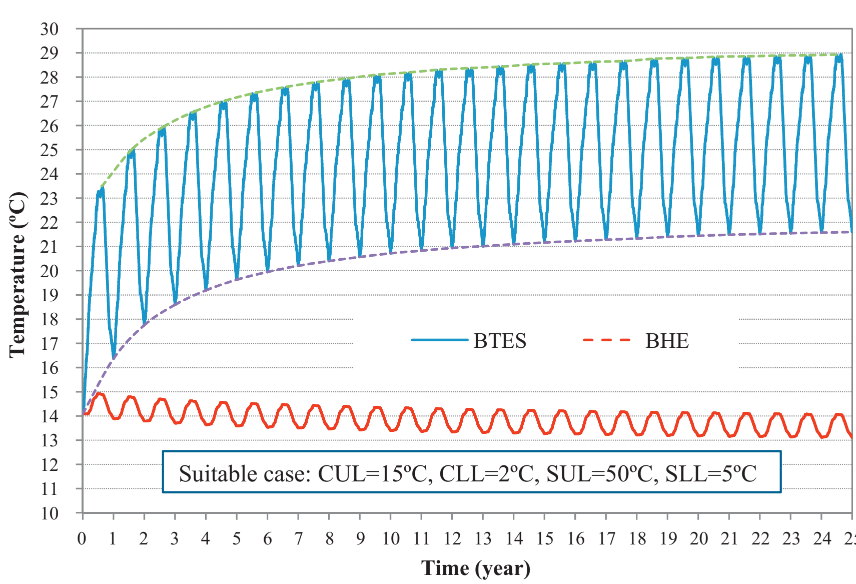

Fig. 14. Average temperatures BTES and BHE for the suitable case.

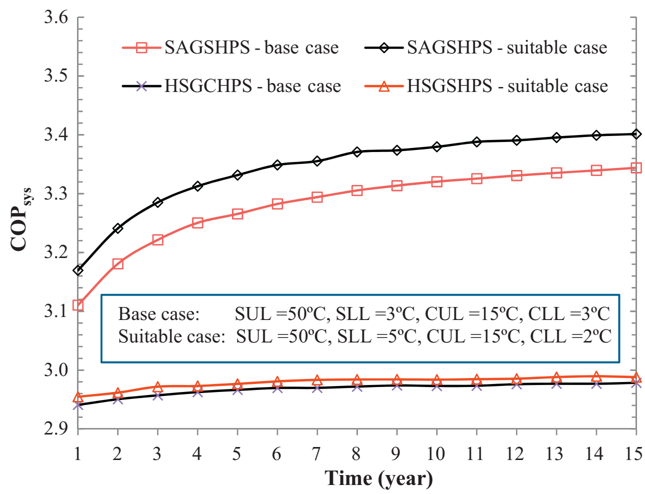

Fig. 15. Comparison of the COP_{sys} for the base case to that for the suitable case.

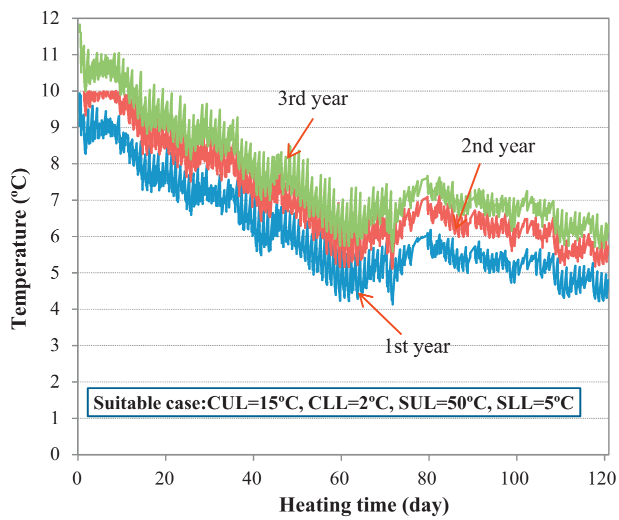

Fig. 16. Fluid temperatures out of evaporators of HP of SAGSHP for the suitable case.

the ground temperature was coming too high due to solar energy storage.

The results of the COP_{sys} for the suitable case are shown in Fig. 15. The COP_{sys} of SAGSHPS can increase from 3.42 to 3.17 and COP_{sys} of HSGSHPS can increase from 2.99 to 2.95 at the end of 25 years. The increase of the temperature in BTES results in very small increase of COP_{sys} of HSGSHPS (about 1.3% after 25 years), because the load of SAGSHPS is a small part (about 1/4) of the total load of HSGSHPS. The solar collection and storage control strategies have no effect on COP_{sys} of GSHPS, and the overall annual value is about 2.88.

The suitable case has better performance than the base case. Compared to the base case, the COP_{sys} of SAGSHPS for the suitable case increases from 1.7% to 2.0% per year, while the COP_{sys} of HSGSHPS increases from 0.3% to 0.5% per year. It should be noted that the suitable case is not an optimum case. Optimization research on control strategy will be carried out in the future as the next phase of the research project.

Fig. 16 shows that there are some instances when the outlet temperature of the evaporator are lower than 5 °C which is the limit of HP when water is used as the heat transfer fluid. However, the temperature increases year by year. The outlet temperature of the evaporator is higher than 5 °C during the entire heating season in the second year. The suitable case cannot absolutely resolve the problem of the temperature limit in the first year. However, the amount of time with evaporator exit temperature below 5 °C is only a small fraction of the entire heating season. Because the distribution systems of GSHP and SAGSHPS are connected together, the problem of temperature limit can be resolved. The GSHPS can provide total heating load when the HP of SAGSHPS cannot function due to low evaporator exit temperature. The heating load would then be supplemented by SAGSHPS in the following year to maintain the balance of the BHE of GSHPS.

5. Conclusions

The performance of a hybrid solar ground-source heat pump system was predicted by a simulation model developed in TRNSYS. The ground temperature, evaporator temperature, average annual coefficient of performance of heat pump (COP_{HP}) or system (COP_{sys}), and so on, were simulated, analyzed and discussed. Suitable operation control strategies have been assessed to provide directions for the operation of HSGSHPS. The results of simulation showed that it is very important to predict the long-term performance and the operation control strategies. The conclusions drawn are as follows:

(1) The solar collection and storage control strategy have significant effects on COP_{sys} of SAGSHPS. A suitable control strategy for the proposed HSGSHPS is determined, i.e. SUL = 50 °C, SLL = 5 °C, CUL = 15 °C and CLL = 2 °C. The COP_{sys} of the SAGSHPS and HSGSHPS can reach 3.42 and 2.99 from 3.17 and 2.95 at the end of 25 years respectively.
(2) The first-operation time (FOT) has major impact on the operation of SAGSHPS. The HP will not work due to low evaporator temperature if insufficient thermal energy is injected into the BTES before extraction. As the heat exchange fluid in the source (ground loop) of the SAGSHP is water, even though the FOT is from the end of the heating season, there are still a few occasions that the outlet water temperature from HP's evaporator is lower than the limit. In terms of enhancing the COP_{sys}, it is preferable to start heat injection into before heat extraction from boreholes for both SAGSHPS and GSHPS.
(3) For the proposed HSGSHPS, electrical energy demand of the system could be reduced by 32% if the load circulation pump is operated based on the fan-coils rather than always-on operation.
(4) The proposed HSGSHPS can be used in a building whose heating load is far larger than cooling load. The hybrid system of SAGSHPS and GSHPS can resolve the imbalance problem of the BHE in GSHPS and increases the COP_{sys} of SAGSHPS.

The parameters used for the model components such as BHE/BTES, solar collector, heat pump, and fan-coils correspond to that of the actual building itself. The accuracy of the simulation results should/and will be validated by the monitoring data obtained in the future.

Acknowledgments

The authors are grateful for the support provided by National Major Project of Scientific and Technical Supporting Programs of China during the 11th Five-year Plan Period (No. PEG200701), Key Scientific and Technical Support Program of Tianjin (No. 07ZCKFSF00400), and Natural Science Fund Program of Hebei (E2008000063). Our sincere thanks also go to Natural Science and

Engineering Research Council (NSERC) of Canada for their funding support through the Discovering Grant Program.

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