Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 48 (2014) 370 – 379

Energy Procedia

SHC 2013, International Conference on Solar Heating and Cooling for Buildings and Industry
September 23-25, 2013, Freiburg, Germany

Simulation of a vertical ground heat exchanger as low temperature heat source for a closed adsorption seasonal storage of solar heat

Samuel Hennaut[a]*, Sébastien Thomas[a], Elisabeth Davin[a], Alexandre Skrylnyk[b], Marc Frère[b], Philippe André[a]

[a]*University of Liège, Building Energy Monitoring and Simulation, Avenue de Longwy 185, 6700 Arlon, Belgium*
[b]*University of Mons, Energy Research Cente,Boulevard Dolez 31, 7000 Mons, Belgium*

Abstract

This paper deals with the simulation of a vertical geothermal heat exchanger as low temperature heat source for a closed adsorption seasonal heat storage. The seasonal storage should allow reaching a nearly 100 % solar fraction for space heating of a "low energy" building". The selected adsorbent and adsorbate are respectively bromide strontium and water. The studied system, including the building and the ground exchanger, is simulated using the dynamic simulation software TRNSYS. Results show that expected performances are reached with a borehole of 100 m. The evaporation temperatures computed are really close to 0°C which might cause some problems. But an advanced research would maybe impose a deeper borehole to avoid cooling the ground on the long term.

Selection and peer review by the scientific conference committee of SHC 2013 under responsibility of PSE AG

Keywords: Seasonal storage; Geothermal heat exchanger; Solar combisystem; Bromide Strontium; Low temperature heat source

1. Introduction and background

Thermochemical or sorption seasonal heat storage is currently admitted as promising solution to increase drastically and efficiently the solar fraction of solar combisystems in temperate climate. This thermal storage

* Corresponding author. Tel.: +32-63-230-948; fax: +32-63-230-800 .
E-mail address: shennaut@ulg.ac.be

1876-6102 © 2014 The Authors. Published by Elsevier Ltd.
Selection and peer review by the scientific conference committee of SHC 2013 under responsibility of PSE AG
doi:10.1016/j.egypro.2014.02.043

technology is particularly suited with long-term storage because it offers high storage density and nearly no thermal loss. This paper deals with the adsorption reaction of water vapor by $SrBr_2$ in a closed system at low pressure. The working pair was selected according to a selection procedure described in [1, 2]. According to this selection and regarding to the storage density, the used working pair was considered as the best reactant allowing to reach a nearly 100 % solar fraction for the space heating of a “low energy” single-family house building.

In most cases, especially for closed adsorption systems, a low temperature heat source is necessary to vaporize the sorbate, when the storage is discharging. The evaporation temperature needed for space heating applications is generally around 5 to 10 °C. For Bromide Strontium, almost 2/3rd of the energy released by the hydration of the salt has to be provided by the low temperature source. In many research project (e.g. [3, 4]), this low temperature source is assumed to be provided by a vertical ground heat exchanger (GHX), receiving free energy from the ground. During the storage charging period, the GHX is also used as energy sink to dissipate the sorbate latent heat of condensation in the ground. In closed system the condensation of the sorbate and its storage in a tank as liquid is necessary to maintain the vapor pressure in the reactor below the equilibrium sorbent/sorbate. Again, the heat sink has to absorb around 2/3rd of the heat used for the dehydration reaction. Using the GHX for this purpose would allow to recharge (at least partially) the ground during summer. However this exchanger is rarely simulated or experimented and a (nearly) constant temperature is considered for the evaporation and the condensation of the sorbate.

Several papers highlight the importance of the cold source temperature on the performance of the sorption storage [3-5]. The working conditions of the GHX are slightly different from those met in combination with a classical heat pump, mainly concerning the lower temperature difference between the working fluid and the ground during discharging period. This paper tries to study more precisely, with simulations, the behavior of this low temperature source and the ability of the vertical GHX to provide the energy required for the storage process. The results allow also realizing a preliminary sizing of the exchanger.

Nomenclature

ΔH_r	heat of reaction [J]
H	efficiency [-]
C_p	thermal capacity of water [kJ/(kg.K)]
E_{backup}	annual final energy demand of backup heater [kWh]
E_{ref}	annual final energy demand of reference system boiler [kWh]
HX_{pinch}	pinch point difference temperature of the evapo/condenser heat exchanger [°C]
L_w	latent heat of evaporation of water [kJ/kg]
$\dot{m}_{GHX}$	flow rate of water in the borehole [kg/h]
Pmax	maximal power of the backup heater [kW]
q_a	flow rate of the vapour transferred between the tank and the reactor [kg/h]
Q_{backup}	annual energy load of backup heater [kWh]
$Q_{boiler,\ ref}$	annual energy load of the reference system boiler [kWh]
T_{cond}	condensation temperature [°C]
T_{evapo}	evaporation temperature [°C]
$T_{in\ GHX}$	Inlet temperature of the ground heat exchanger [°C]
$T_{out\ GHX}$	Outlet temperature of the ground heat exchanger [°C]

2. Simulated system

A vertical GHX is added to a complete solar combisystem integrating a closed adsorption seasonal storage. This exchanger is thermally linked to an evapo/condenser which is connected to the chemisorption storage reactor. This combisystem heating a “low energy” building was described and results were presented in previous works [6]. For these precedent simulations, the condensation and the evaporation temperatures were respectively assumed to be constantly equal to respectively 20 and 5 °C. TRNSYS is used to simulate dynamic behavior of the whole system.

2.1. Building energy need

The selected building is a recent wooden dwelling recently built in Belgium according to “low-energy” standards. Its total energy use for space heating is around 4200 kwh/year for a heated space of 100 m². This building allows an advantageous use of solar heat thanks to 40 m² of the roof facing South with a slope of 40 degrees. Heat emitters of the building are two heating slabs (one for each floor) supplied by water at 35°C. The temperature set point is respectively 20°C and 18°C for the ground floor and the first floor. The heating demand computed for domestic hot water (DHW) production is almost 2700 kWh/year.

2.2. Seasonal storage

The seasonal storage occurs through an adsorption closed system. The model used in simulations computes energy and mass balance considering some kinetic aspects. This model is based on the “detailed Modeling Approach” described in [7]. Parameters used in realized simulation correspond to the working pair $SrBr_2/H_2O$ according to the following reaction:

$$SrBr_2.6H_2O_{(s)} + \Delta H_r \rightleftharpoons SrBr_2.1H_2O_{(s)} + 5H_2O_{(g)} \qquad (1)$$

The closed adsorption storage is composed by a reactor integrating a heat exchanger. The reactor includes the totality of the adsorbent and is connected to a tank containing the adsorbate. Both are separated by a valve controlling the transfer of the adsorbate. A heat exchanger is included in the adsorbate tank. The system is presented at Fig.1. in summer (a) and winter (b) operation mode.

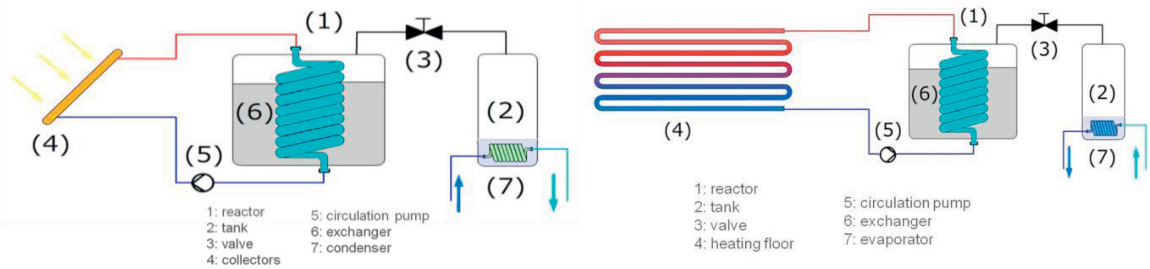

Fig. 1. Closed adsorption system (a) charging mode; (b) discharging mode.

During charging phase, heat supplied by solar collectors increases the temperature of the hydrated adsorbent leading to water vapor desorption. To avoid reaching the pressure equilibrium between water vapor and adsorbed water, the desorbed water has to be transferred to the tank (to continue the desorption reaction). This transfer occurs if the vapor pressure in the tank is lower than in the reactor. To achieve this lower vapor pressure in the tank, water vapor must be condensed in the tank. The higher the condensation temperature is the higher the temperature in the reactor has to be to achieve desorption. Therefore, to maintain relatively low temperature and water pressure in the tank, a cold sink is necessary to evacuate heat of condensation of water. In these simulations, the cold sink is the ground, which is heated through a vertical GHX.

During discharging phase, water stored in the tank has to be vaporized to react with the adsorbent. The heat of vaporization of the sorbate is supplied by the cold source through the GHX. The temperature reached at the evaporator will determine the maximal vapor pressure in the tank which has to be greater than in the reactor to ensure a flow of vapor to the latter. Finally, according to the adsorption thermodynamic equilibrium, the vapor pressure in the reactor will determine the maximal adsorption temperature which influences directly the space heating supply temperature. So, higher is the evaporation temperature, higher is the temperature in the reactor and in the heating loop.

2.3. Combisystem

The configuration of the combisytem (volume of the tank, collector area, mass of adsorbent) is based on results of [6].The simulated system is illustrated in Fig. 2. The solar collectors used are high performance flat plate collectors being developed for the Solautark Project. The expected performance of these collectors is :

- Optical efficiency - a_0 [-]: 0.80
- Linear loss coefficient – a_1 [W/(m².K)]: 1,57
- Quadratic loss coefficient – a_2 [W/(m².K²)]: 0,0072

The total area of collectors is 20 m². These collectors heat a 1,5 m³ tank used for space heating and DHW. The drawing is done at the top of the tank for both purposes and the DHW is produced using an external heat exchanger. The building is only heated from October 1st to April 30th. During this period, the temperature in the tank is limited to 95°C and the seasonal storage is only discharged. During the second part of the year ("storage period"), the building isn't heated and the temperature in the tank may not exceed 75°C. When this maximal temperature is reached in the tank, the excess solar heat is stored thermochemically.

Two auxiliary backup heaters are also simulated in the system. The first heater, integrated between the tank and the heating floor, is used to heat the supplied space heating water to 35°C if the temperature set point of the room isn't reached. The second backup heats the DHW to 45°C. And finally the thermochemical reactor (7000 kg of dehydrated $SrBr_2$) is used as backup heater during winter and is located just after the tank (before the first backup heater).

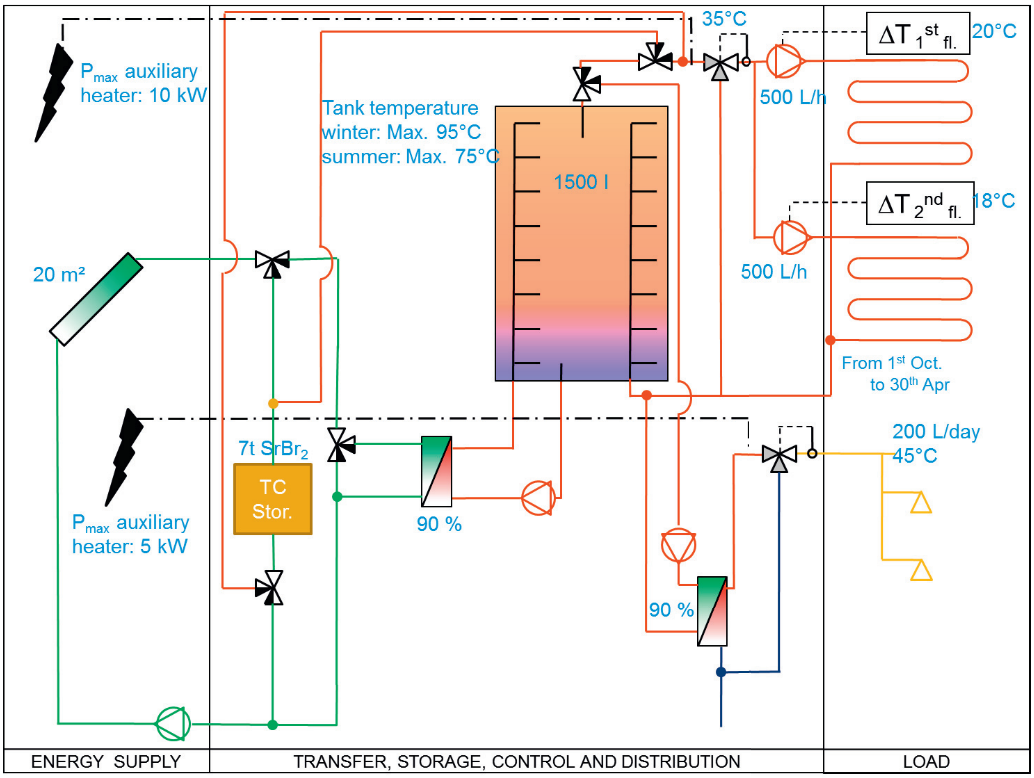

Fig. 2. Simulated combisystem integrating a seasonal thermochemical storage (TC Stor.).

2.4. Evaporator and low temperature source

The TRNSYS vertical GHX model used (Type 557a from TESS libraries) combines explicit finite difference method and analytical method to compute the temperature in the ground [8]. The parameters defined for this model are those of a classical double U-tube pipe of polyethylene and fill with pure water. A single borehole is considered with a thermal conductivity of 2 W/(m.K) for the filling material having a perfect contact with the pipe. The properties of the ground are assumed to be uniform:

- Thermal conductivity of the ground: 2 W/(m.K)
- Thermal capacity of the ground: 2000 kJ/m³.K

The initial temperature conditions of the ground are:

- Initial temperature of surface: 9.4 °C
- Initial thermal gradient: 0.01 °C/m

Fig. 3 illustrates the integration of the models of evaporator and GHX to the system. This exchanger is linked to a simple model of heat evapo/condenser computing the return temperature to the ground, using a constant flow rate pump. The pump works if some vapour is transferred between the reactor and the tank during the previous time step (for convergence improvement). According to the temperature of the fluid at the outlet of the GHX, the evapo/condenser model computes the return temperature to provide exactly the energy necessary to evapo/condensate the water vapour transferred during the time step before, according to the equations defined in Fig.3. This model computes also the temperatures of evaporation and condensation considering a heat exchanger pinch difference temperature of a few degrees. These both temperatures are transmitted to the model of the reactor which compute the vapour pressure in the evapo/condenser and the transfer of vapour for the current time step, as done previously for a constant temperature of evapo/condensation.

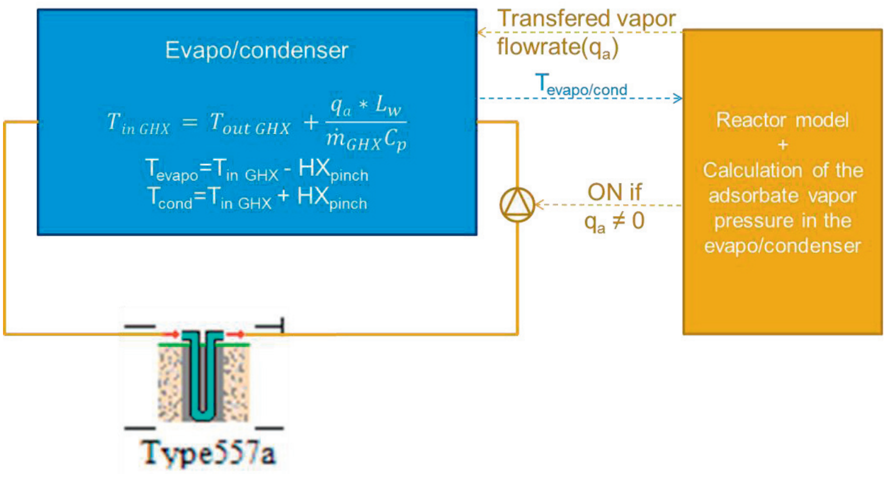

Fig. 3. Evapo/condenser model and links with the GHX and the reactor model.

3. Methodology

The complete system is simulated during one year modifying the values of 3 parameters related to the low temperature heat source: the borehole depth, the pinch point temperature difference of the evapo/condenser and the flow rate in the borehole. The simulated values are summed up in Table 1.

Table 1. Parameters tested.

Parameter	Minimum	Maximum	Step
Pinch point temperature difference [°C]	3	7	2
Borehole depth [m]	60	160	20
Flow rate in the borehole [kg/h]	700	1400	100

The solar fraction of the simulated configurations is computed thanks to the « fractional thermal energy savings » index ($f_{sav,therm}$) introduced in research results realised for Task 26 of the program IEA-SHC[9]:

$$f_{sav,therm} = 1 - \frac{E_{backup}}{E_{ref}} = 1 - \frac{\frac{Q_{backup}}{\eta_{backup}}}{\frac{Q_{boiler,ref}}{\eta_{boiler,ref}}} \quad (2)$$

This index doesn't consider parasitic consumption. It is computed for DHW production ($f_{sav,DHW}$) or for space heating only ($f_{sav,SH}$) but also for the complete system ($f_{sav,TOT}$). The reference consumptions ($Q_{boiler,\ ref}$) for DHW and space heating are respectively around 2700 and 4200 kWh. The efficiency (η) of the backup heater is assumed equal to the one of the reference boiler (85 %). This global indicator will be very useful to see the influence of selected parameters on the annual performance of the studied system. A more detailed analysis will also be carried out, regarding the GHX temperature levels and maximal heat transfers.

As a reference, Fig. 4 presents the f_{sav} values computed for the system simulated without GHX. Evapo/condensation temperatures are defined constant at respectively 5 and 20 °C.

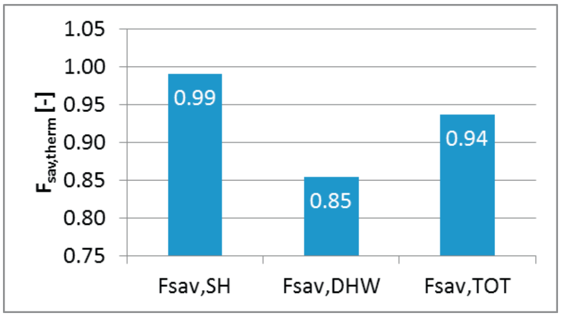

Fig. 4. $F_{sav,therm}$ computed for the reference system (without GHX – constant evepo/condensation temperature 5/20 °C)

4. Results and analysis

Graphical results presented about $f_{sav,therm}$ computation will focus only on $f_{sav,SH}$ because $f_{sav,TOT}$ results present similar curves moved downward and the influence of studied parameters on $f_{sav,DHW}$ is nearly negligible. Fig. 5 illustrates the $f_{sav,therm}$ computed for space heating with a pinch point difference temperature of 5°C for different borehole depths and flow rates. On this chart, the red line represents the solar fraction computed without GHX. Analysis of these results shows that in these conditions, a 100 m borehole provides performance similar to the reference system (red line) independently of the flow rate. Under 100 m, the flow rate has also nearly no influence on the performance and f_{sav} computed are really close to the reference: - 0.3 % for 80 m and -0.7 % for 60 m.

For a lower pinch point temperature difference of the evapo/condenser (3°C), all tested configurations offer similar performances. But, if the temperature difference increase until 7°C, $f_{sav,SH}$ are more sensitive to the borehole flow rate and depth (see Fig. 6). For a 60 m GHX, the solar fraction decreases of around 3% compared to the reference. In this case double the flow rate allows only gaining 0.5 %, which wouldn't be advantageous considering parasitic consumption. According to these observations, we assume that the sizing of the evapo/condenser and of the GHX are connected and should be done together.

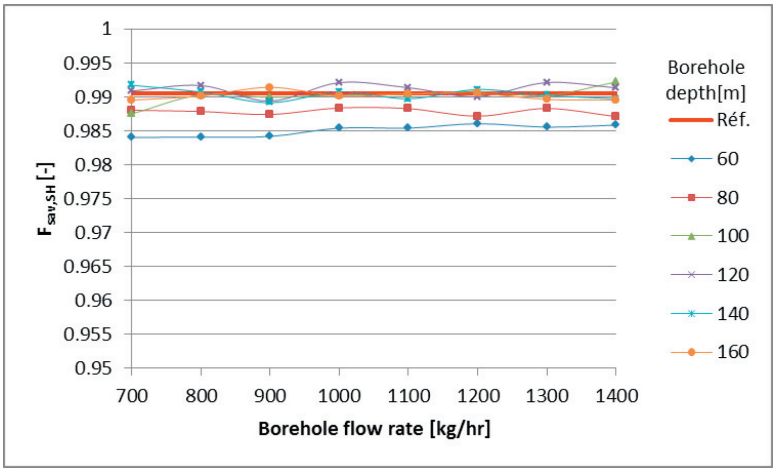

Fig. 5. Influence of the borehole depth and flow rate on the space heating solar fraction (HX_{pinch} = 5°C)

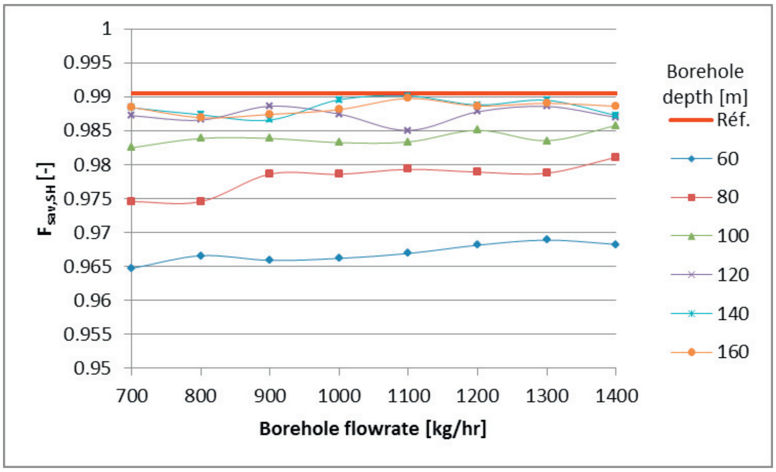

Fig. 6. Influence of the borehole depth and flow rate on the space heating solar fraction (HX_{pinch} = 7°C).

4.1. Influence of the borehole depth: 1000 kg/h flow rate – 5°C HX_{pinch}

According to hydraulic computation the flow rate in the U-tube pipe has to be greater than 400 kg/h to have a turbulent flow, which means 800 kg/h per borehole with two U-tube pipes. Considering a factor of safety, we will analyze the influence of the borehole depth with a flow rate of 1000 kg/h.

The inlet temperature of the GHX is directly related to the evapo/condensation temperature according to the equations defined in Fig. 3. During winter (see Fig. 7), the inlet temperatures computed are between 9,5 °C (in November) and 6 °C for a borehole of 100 m, which corresponds to an evaporation temperature of 1°C. Such a temperature is very low with water as working fluid (in the GHX pipe and in the adsorbate tank). If the GHX is 60 meters deeper, the minimal temperature computed is around 7.3 °C. Considering these temperatures it will be interesting to estimate the influence of glycol use as working fluid in the GHX. During summer (see Fig. 8.), the inlet temperature level is less critical for two reasons. Firstly, to avoid degradation of the GHX material, the inlet temperature is limited to 30°C but with a 100 m borehole, the inlet is under 25°C. Secondly, this period is less critical because of the intermittent solar availability which allows decreasing the condensation temperature during the night when there is no heat supplied. Furthermore, in simulations realized, only 3 months are necessary to charge the seasonal storage even though the storage period lasts 5 months. With a 160 m GHX, the inlet temperature is always under 20 °C.

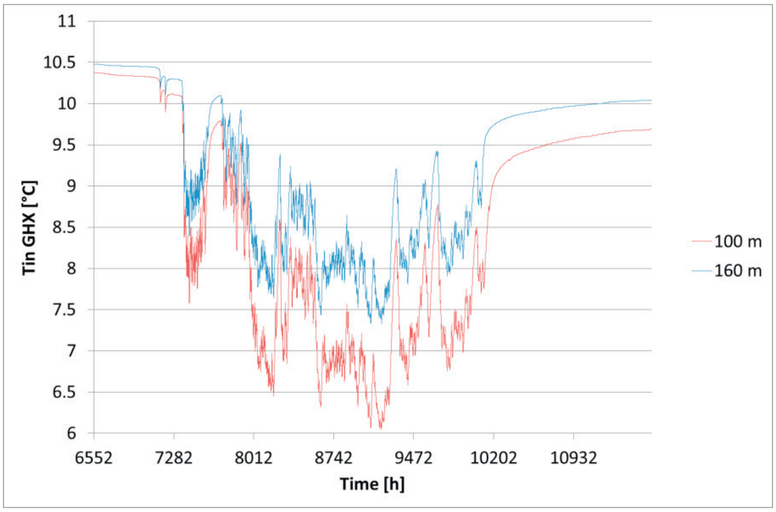

Fig. 7. GHX inlet temperature during discharging period.

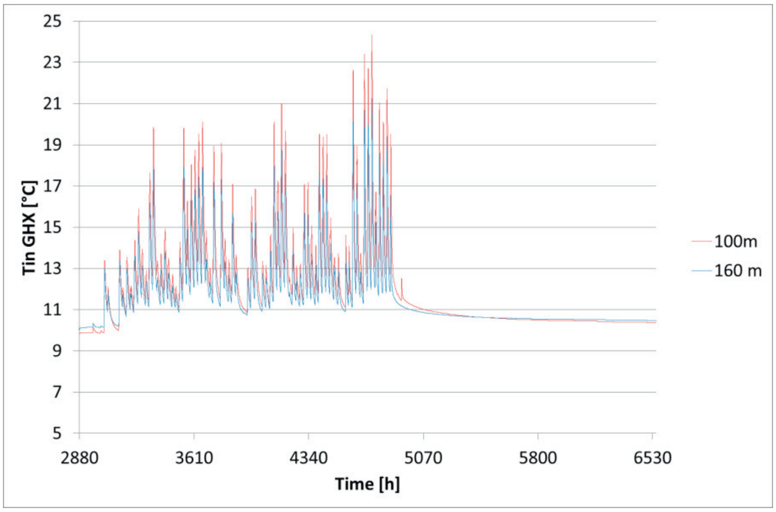

Fig. 8. GHX inlet temperature during charging period.

The maximal total heat transfer (Fig. 9) executed by the GHX increases with the borehole depth, which has a greater influence during the discharging period. A GHX of 100 m allows evacuating until 6.6 kW during summer and the ground will provide maximum 1.16 kW in winter. For a depth of 160 m instead of 100 m, the heat transfer raise of around 3% during charging phase and 9 % when discharging.

If this heat transfer is related to borehole depth (Fig. 10) these conclusions are reversed. Indeed the transfer per meter of borehole decreases with the length of the borehole, but this reduction is similar during both phases of the storage: around 35 % from 100 to 160 m depth. For a 100 m borehole the heat transfers per meter of depth are respectively around 65 W/m and 12 W/m in summer and winter.

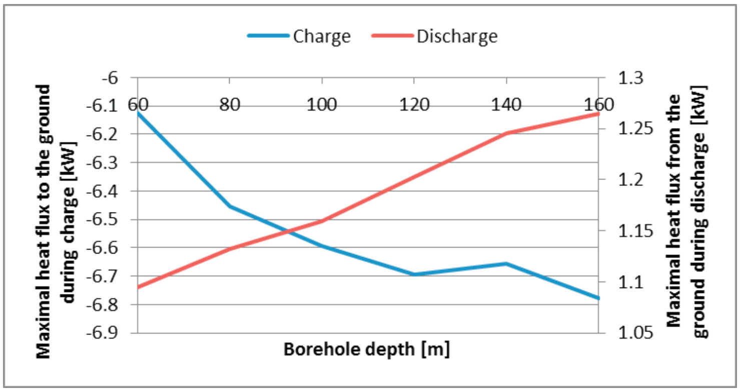

Fig. 9. Influence of the borehole depth on the maximal total heat transfer of the GHX (flow rate 1000 kg/h – HX_{pinch} 5°C).

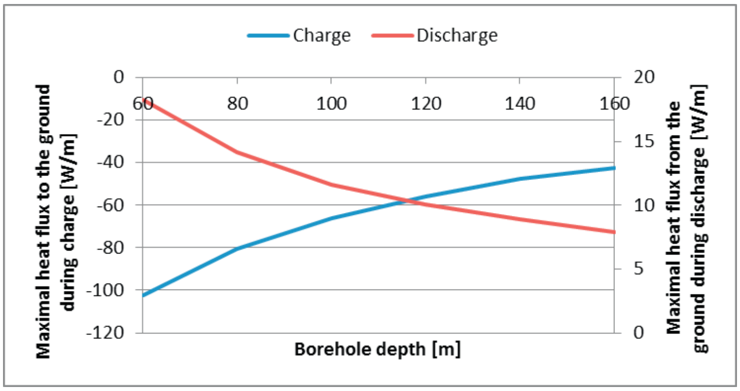

Fig. 10. Influence of the borehole depth on the maximal heat transfer of the GHX per meter of borehole (flow rate 1000 kg/h – HX_{pinch} 5°C).

During simulations, the working time of the pump integrated in the geothermal loop is computed. The total duration is around 5300 h of which around 2/3rd occur during the discharging period. Assuming an electrical power of 100 W for this pump, the total consumption would be 530 kWh. The working time decrease with the borehole depth: 5 % can be saved from 100 to 160 m.

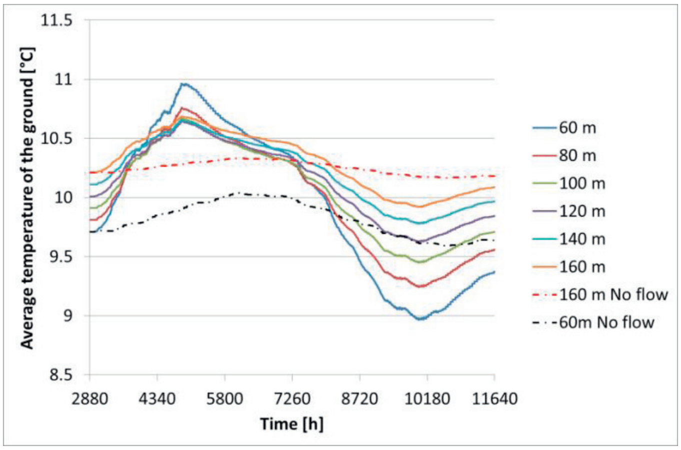

Fig. 11. Influence of the borehole depth on the average ground temperature.

Finally, the average temperature of the ground surrounding the borehole is studied (see Fig. 11). Sinusoidal curves are observed with higher amplitude when the GHX depth decreases. Results show that after one year, the average temperature of the ground is lower than at the start time for every curve. To establish reference curves, two simulations of the GHX are realized with same parameters as previously but without flow rate. The difference between these two simulations is the depths: 60 and 160 m. For the reference curve, the final temperature is nearly equal to the beginning temperature. Compared to these reference, at the end of one year, the average ground temperature decrease of respectively around 0,25 and 0,1 °C for 60 and 160 m. This observation highlights the importance of a long term simulation to study performance reduction after several years of use of the GHX. This impact should be broadened in future works.

5. Conclusion and outlook

The results presented show the ability of a 100 m geothermal heat exchanger to be used as cold source/sink for a closed adsorption seasonal storage for a “low energy” building. However, this borehole depth may cause some problems regarding the evaporation temperature (too close from 0°C). The influence of the depth on the ground temperature decrease has also to be studied deeply. The experimental validation of these results is planed.

Further work has also to focus on the parasitic consumption of this cold source. Indeed, the length of the pipe and the flow rate will influence directly the power consumption of the pump. The estimated annual consumption of the pump represents more than 10 % of the energy necessary for the building space heating.

The investment cost of such a borehole is also quite expensive. The GHX alone cost around 50 €/m (without working fluid, building connection and pump). As a conclusion, this cold source allows to use free energy but not freely.

Acknowledgements

Research presented in this paper is conducted in the SOLAUTARK project with the following partner’s: ESE, ArcelorMittal Liège R&D, Atelier architecture Ph. Jaspard, ULB, CTIB, M5, UMons, and ULg. This project is funded by the Plan Marshall of the Walloon Region.

References

[1] S. Hennaut, S. Thomas, P. Andre, E. Courbon, T. Berigot (Le), and M. Frère, "Pré-dimensionnement par simulations dynamiques d'un réacteur de stockage thermochimique assurant l'autonomie d'un système solaire combiné," presented at the Congrès Français de Thermique 2011 - Energie Solaire et Thermique, Perpignan, 2011.

[2] E. Courbon, A. Skrylnyk, S. Hennaut, P. Andre, and M. Frère, "Procédure simple pour estimer les capacités de stockage de chaleur des systèmes solide/gaz : Application au stockage d’énergie solaire dans les bâtiments," *Récents Progrès en Génie des Procédés,* vol. 101, 2011.

[3] N. Le Pierres, H. Liu, and L. Luo, "$CaCl_2/H_20$ absorption seasonal storage of solar heat," presented at the IC-SES 2011, Belfast, Ulster, 2011.

[4] G. Tanguy, P. Papillon, and C. Paulus, "Seasonal storage coupled to solar combisystem: dynamic simulations for process dimensioning," presented at the EuroSun 2010, Graz, Austria, 2010.

[5] K. E. N'Tsoukpoe, H. Liu, N. Le Pierrès, and L. Luo, "A review on long-term sorption solar energy storage," *Renewable and Sustainable Energy Reviews,* vol. 13, pp. 2385-2396, 2009.

[6] S. Hennaut, S. Thomas, E. Davin, A. Skrylnyk, M. Frère, and P. Andre, "Dynamic simulations of solar combisystems integrating a seasonal sorption storage: Influence of the combisystem configuration," presented at the EuroSun 2012 ISES-EUROPE Solar Conference, Rijeka, Croatia, 2012.

[7] A. Skrylnyk, E. Courbon, M. Frère, S. Hennaut, P. Andre, P. Sun, and G. Descy, "Modelling of a solar thermo-chemical system for energy storage in buildings," presented at the Innostock 2012 - The 12th International Conference on Energy Storage, Lleida, Spain, 2012.

[8] G. Hellström, "Heat Storage in the Ground - DUCT GROUND HEAT STORAGE MODEL - Manual for Computer Code," Department of Mathematical Physics - Univesity of Lund1989.

[9] T. Letz, "Validation and background information on the FSC procedure," IEA SHC2002.