Conference Proceedings

EuroSun 2014
Aix-les-Bains (France), 16 – 19 September 2014

Simulation of Combined Solar Thermal, Heat Pump, Ice Storage and Waste Water Heat Recovery Systems. Design Criteria and Parametric Studies.

Daniel Carbonell , Daniel Philippen, Martin Granzotto, Michel Y. Haller and Elimar Frank.

Institut für Solartechnik (SPF), Hochschule für Technik (HSR),
CH-8640 Rapperswil, Switzerland

Abstract

Combined solar thermal and heat pump systems have attracted interest in the last decade in order to increase the share of renewable energies in space heating and domestic hot water preparation applications. Among many system concepts that combine heat pumps and solar thermal collectors, ice storage based concepts are analyzed here. Waste water heat recovery (WWHR) is an interesting option in these systems becasue ice storages are operated with low temperatures most of the year. Almost any waste heat that is recovered in winter can be used to regenerate the ice storage, i.e. to melt the ice.

The goal of this paper is to provide design criteria for sizing these type of systems for single family houses demands and to analyze several WWHR integrations. System performance was investigated studying the effects of heat exchanger area in the ice storage, size and type of collector (covered and uncovered both with selective coating), size of ice storage and different waste water heat recovery devices.

Simulations presented here show that uncovered collectors perform better than covered collectors except for large sized systems where auxiliary electrical back-up is not necessary. When the auxiliary back-up is not needed, the use of direct heat from the ice storage to cover the heating demand is usually of benefit. Waste water heat recovery systems are found to significantly increase the system performance, especially when waste water can be stored in a tank. The system performance can increase from 4 to 20% if a waste water storage is used and around 2 % using a drainwater heat recovery system, depending on system design and building demands.

keywords: ice storage, solar thermal, heat pumps, waste water heat recovery, system simulation.

1. Introduction

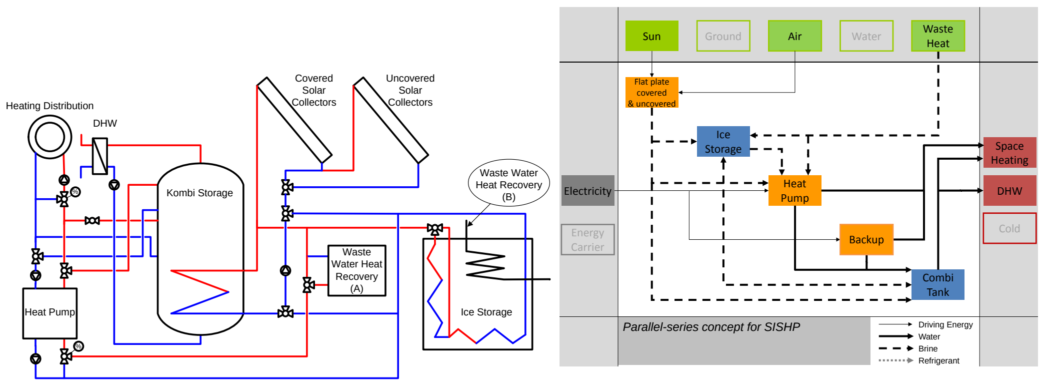
Research on combined solar thermal and heat pump systems has been carried out since many years. However, only in the last years a significant increase in the number of installed system has been found in European markets. Ruschenburg et al. (2013) surveyed 88 companies and found that most of them entered the market in recent years and a large share from those were located in Germany and Austria. A recent example of the research done in this topic is the already finished study conducted in the framework of the International Energy Agency (IEA), Solar Heating and Cooling programme (SHC Task 44) and Heat Pump programme (HPP Annex 38) “Solar and Heat Pumps”, known under the combined name Task44/Annex38 (http://task44.iea-shc.org/). The goal of this task was to assess performance and relevance of these systems, to provide a common definition of performance indicators, and to contribute to successful market penetration of these systems (Hadorn, 2012).

The motivations in using ice storages as a source for heat pumps are various: i) ice storages are an alternative to ground source heat pump (GSHP) systems when for example, regulations forbid to drill boreholes, ii) they can be installed in the basement of the building for cost reduction (no drilled holes) or when there is no ground space available, iii) they can also be seen as an alternative to air source heat pumps when efficiency or noise problems are of importance, iv) they have a higher storage capacity compare to sensible storages, v) they can be used to store surplus solar heat, especially in summer and at high temperatures and vi) they can be used as additional sensible storage.

From the system concept point of view combined solar thermal and heat pump systems can be classified as parallel, series, regenerative or a combination of them (Ruschenburg et al., 2013). In those systems the heat pump can have several sources, namely solar, ground, air and waste water. The ice storage is used in heating applications as a source for the heat pump and it is loaded by the solar field or by other heat sources, e.g. waste water heat. This field of application is sometimes referred in the industry as solar ice. The ice storage simulated here is a combined parallel and series system with solar energy, air and waste water as sources for the heat pump (see Fig. 1b).

2. System description and concept

The ice storage system employed here is based on immersed flat plate heat exchangers with a de-icing concept. The de-icing concept is used to improve the heat withdrawal of the heat exchangers and it was firstly presented in Philippen et al. (2012) and further explained in Carbonell et al. (2014). With this concept, using heat exchangers only at the bottom part of the storage, it is possible to reduce considerably the heat exchanger area needed. Considering that the ice layers are chaotically stored on the surface and that therefore some water will be present between the floating ice layers, it is estimated that around 70% of the storage can be iced. The system concept also considers wall heat exchangers (wall-hxs) along the height of the storage. These heat exchangers serve mainly to use the stored heat accumulated in summer periods in the upper part of the storage for space heating (SH) and domestic hot water (DHW) preparation in autumn. The heat exchangers located at the bottom of the storage cannot access the heat stored in the upper part of the storage directly. The transfer of the heat stored in summer from the ice to the warm storage (Q_{Cs-Ws}) is thereby only possible with wall-hxs. By means of control strategies, wall-hx are not allowed to ice.



(a) (b)

Fig. 1: (a) Simplified hydraulic scheme and (b) energy flow view, of a combined solar thermal and heat pump, ice storage and waste heat recovery system.

Waste water heat recovery (WWHR) systems are of special interest in this application due to the low temperature of the ice storage in winter and due to the fact that WWHR provides heat at times when other sources (solar, ambient air) may not have a significant contribution. Being able to recover the waste water heat is expected not only to increase the system performance, but also to obtain a system more independent on DHW usage. The daily usage of DHW is user dependent and system efficiencies may be highly affected by the DHW load profile. Therefore, with the possibility of recovering this heat, the system performance is expected to become less dependent on the amount of DHW consumption.

Heat recovery systems can be classified as retentive and not retentive and also as active or passive systems. With non retentive systems, the heat has to be used at the time when waste water is available. Instead, retentive systems are able to store the grey water (black water is not considered) to be used when needed. If the heat of the waste water can be used in a passive way, then there is no need to run an additional circulation pump. Two different WWHR systems have been analyzed in the present study: i) Gravity film heat exchanger (WW_{GFX}) and ii) Waste water storage (WW_{store}). A simplified hydraulic scheme and the energy flows of this system are shown in Fig. 1. The two systems studied here, WW_{GFX} and WW_{store} are located at position A in Fig. 1a.

2.1. *Gravity film heat exchanger (WW_{GFX})*

The GFX is a drainwater heat recovery system consisting of a pipe installed into a section of a vertical drain line and coils wrapped around it. Heat is transferred from the waste water passing through the large, central pipe to the brine which moves upwards through the coils on the outside of the pipe. The GFX is a non-retentive and active system, therefore the circulation pump in the evaporator loop needs to be activated when there is flow in the GFX in order to use all possible waste water heat.

2.2. *Waste water storage (WW_{store})*

A waste water storage of 130 l is located after the ice storage and before the heat pump as shown in Fig. 1a (position A). This is a retentive and active system. Thought being an active system, the circulating pump is not activated by the control due to the WWHR system unless de-icing is needed. With this system, the control only opens the valve of the WWHR loop when it is possible to obtain energy from the WW_{store}. If flow is circulating in the evaporator loop due to normal operation then, energy is provided to the system by the WW_{store}.

3. Methodology

Simulations were conducted with the simulation environment TRNSYS-17. Results have been obtained for two buildings, SFH45 and SFH100 following T44/A38 boundary conditions (see Dott et al. (2012) and Haller et al. (2012) for details) in Strasbourg (France). The two buildings represent medium (SFH45) and high (SFH100) building energy demand, where SFH stands for Single Family House and the numbers, 45 for example, for the yearly space heating energy demand in kWh/m^2 per building heated surface area in the city of Strasbourg. The building SFH15 has not been simulated because the system presented here is expected to be too expensive for very low energy demand buildings. The building SFH45 has a low temperature heat distribution systems (35/30 ^{o}C at nominal conditions for supply and return respectively) and the building SFH100 has a higher temperature heat distribution system (55/50 ^{o}C) because radiators were meant to be used. The domestic hot water (DHW) tapping profile is obtained from T44/A38 (Haller et al., 2012), the DHW set temperature is 45 ^{o}C and the cold water temperature is set to 10 ^{o}C.

A key aspect in the simulations presented here is the ice storage model which was developed at the Institut für Solartechnik (SPF). The mathematical formulation of the ice storage model was presented in Carbonell et al. (2013) along with a validation using laboratory measurements. The other models of the system have been validated separately in different works and by different institutes. In particular, the model employed for the GFX was developed and tested by Heinz et al. (2012). The system configuration in TRNSYS has been extensively used and improved during the last four years in the framework of national and international projects. Philippen et al. (2014) compared numerical results of the system simulated here with monitored data of a pilot plant in Jona (Switzerland) which was designed, installed and monitored by the Institut für Solartechnik (SPF). First simulations using T44/A38 reference conditions without WWHR were presented in Carbonell et al. (2014). The waste water profile used was obtained from Heinz et al. (2012) and it has been modified in order to be consistent with the T44/A38 DHW demand.

4. Performance indicators

The main performance indicator for the system is the Seasonal Performance Factor (SPF) calculated as described in Malenkovic et al. (2012):

$$SPF_{SHP+} = \frac{Q_{DHW} + Q_{SH}}{P_{el,T}} = \frac{Q_D}{P_{el,T}} \tag{1}$$

Q is the yearly heat load energy and $P_{el,T}$ the total yearly electrical energy consumption. The subscripts SHP, DHW, SH and D stand for solar and heat pump, domestic hot water, space heating and total demand respectively. The total electricity $P_{el,T}$ is calculated considering the consumption of the circulation pumps, heat pump, control unit, auxiliary ($P_{el,aux}$) and penalties (see Haller et al. (2012)). The symbol "+" in the $SHP+$ from Eq. 1 refers to the consideration of the heat distribution circulating pump in the electricity consumption. Therefore, the system performance indicator used in this work include all circulation pumps of the system and also all thermal losses/gains from storages and pipings.

5. Results

5.1. Heat exchanger area in the ice storage

One of the questions when designing an ice storage is to size the heat exchanger area, e.g. the heat extraction power. The installed extraction power should be related to a design value of ice thickness on the heat exchanger and to the heat pump evaporator power. In the simulations used here the heat pump has a nominal capacity of 6.5 kW for SHF45 with a COP of 4.8 (at B0/W35), which means an extraction power of the evaporator of 5.2 kW. A heat pump of 11 kW from the same manufacturer has been used for SFH100. In this case the extraction power from the evaporator is 8.8 kW with a COP of 4.9 (at B0/W35).

Results with varying heat exchanger area and ice storage volume are presented in Fig. 2 using covered (cov) collectors with and without the wall-hxs in the ice storage. In Fig. 2, simulations for SFH45 are obtained with 20 m^2 of collector area, while SFH100 results are presented using 35 m^2 . From results presented in Fig. 2a for SFH45, one can observe that for A_{hx} above approximately 23 m^2, the SPF_{SHP+} is more or less constant for most of the simulated cases. In the ice storage concept used here no more extraction power than the needed for the heat pump is necessary because with some heat exchangers at the bottom with *"de-icing"* capabilities it is possible to fill up most of the storage with ice. With steel flat plate heat exchangers, an extraction power of 200 W/m^2 with an inlet temperature of -3^{o}C, mass flow rate of 88 kg/(hm^2) and 1 cm of ice on the surface is estimated. Therefore, to use the full potential of the heat pump under this nominal conditions, 26 m^2 of heat exchanger area would be needed for the 6.5kW heat pump and 44 m^2 for the 11 kW heat pump considering. This theoretical value corresponds quite well with the simulated results of SFH45 (maximum SPF_{SHP+} in Fig. 2a and it will be used hereafter for SFH45 simulations. However, the choice of this nominal conditions, in particular the choice of 1 cm of ice thickness, depend on the system concept and on the size of components and cannot be extrapolated to other conditions and/or systems. For example, simulations for SFH100 show a quite different behavior. One might expect a needed area of 44 m^2 as deduced before, but for 40 and 50 m^3 a heat exchanger area of around 15 m^2 seems to be sufficient. For SFH100 and V_{ice} of 20 and 30 m^3, the theoretical value of 44 m^2 corresponds quite well with the A_{hx} that has the maximum SPF_{SHP+} (Fig. 2b) and the value of 40 m^2 will be used hereafter for all SFH100 simulations. Nevertheless, a more detailed study is needed to find a clear design criteria to determinate the A_{hx} needed since this value depends on the size of the other components. For this work we use the mentioned values of A_{hx} to ensure that results do not depend on this choice.

As explained in section 2, the main reason of having a wall-hx consists in being able to transfer the heat

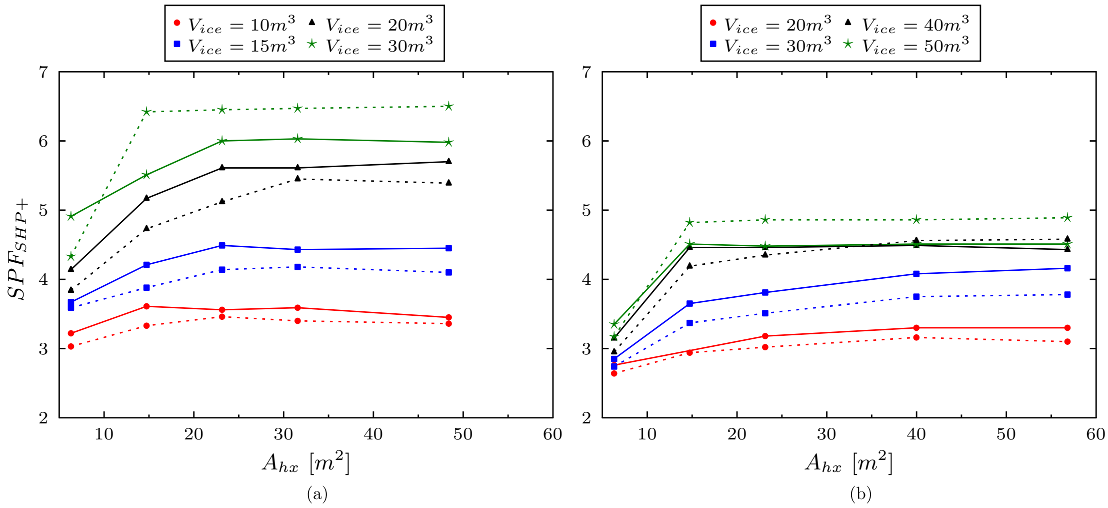


Fig. 2: System seasonal performance factor as function of heat exchanger area and ice storage volume for building (a) SFH45 and A_{cov}=20 m^2 and (b) SFH100 and A_{cov}=35 m^2. Solid lines are without wall-hx and dashed lines are with wall-hx.

stored at high temperature in the upper part of the ice storage to the warm storage for SH and DHW use (Q_{Cs-Ws}). This mechanism is mostly used in autumn when heat stored in summer in the ice storage can be used directly. In principle one might think that the consideration of a wall-hx should always be of benefit or in the worst case having little effect in the system performance, but simulations show a decrease of the system performance when wall-hxs are used for systems where the SPF_{SHP+} is below approximately 5.25 for SFH45 and below 4.5 for SFH100. The system performance for these systems is strongly related to the auxiliary heat ($P_{el,aux}$) needed to deliver the heat demand when the ice storage is full of ice and the heat pump evaporator temperature drops below the minimum allowed value (-8^{o}C has been used here as the minimum outlet brine temperature allowed). For systems with $SPF_{SHP+} > 5.25$ in SFH45, the auxiliary electrical back-up system is hardly needed. In those situations it is of benefit to use direct heat from the ice storage, i.e. simulations with wall-hx have higher SPF_{SHP+}. These results are in accordance with the findings from Haller and Frank. (2011), i.e. the use of direct heat is usually a better option than the indirect use through the heat pump. However, when the system is designed such that the back-up direct electric heating system ($P_{el,aux}$) is needed, then it is better to avoid transferring heat directly from the ice storage to the warm storage because this implies that the ice storage will freeze earlier or for longer time, compared to the case with $Q_{Cs-Ws} = 0$, and thus more direct electric heating is needed. Instead, it is a better option to keep the ice storage as warm as possible to delay and shorten the time when it is completely frozen. A way to confirm this statement consist of blocking the control program that allows to transfer heat directly from the ice storage to the warm storage when wall-hx are used, i.e. forcing Q_{Cs-Ws} to be 0. Simulations blocking this program have been carried out and differences in SPF_{SHP+} with and without wall-hx are in the order of 1%.

5.2. Collector type and area

In this section numerical studies are focused to decide which kind of collectors can be used, i.e covered (cov), uncovered (unc), both with selective coating, and which collector area should be installed. Several ice storage volumes are considered since they are related to the size of the collector field. For all simulations presented here wall-hx are not used and therefore the heat transfer mechanism (Q_{Cs-Ws}) is not relevant (see section 2 for details).

System performance is presented in Fig. 3 for building SFH45 using only covered (Fig. 3a or uncovered (Fig. 3b collectors. In Fig. 3a it can be observed that the SPF_{SHP+} range from 2 to 7 approximately, depending on collector area and ice storage volume. Another important fact observed is that the increase of SPF_{SHP+} per m^2 of collector area is higher when back-up electrical auxiliary heat $P_{el,aux}$ is needed (see dashed lines and right axes of Fig. 3). When $P_{el,aux}$ is close to zero, the increase of SPF_{SHP+} per m^2 of collector area is reduced. It can be seen that each SPF_{SHP+} line can be described by two lines with constant slope, one before, and one after the collector area that causes $P_{el,aux}$ to be approximately 0. Results using only uncovered collectors are shown in Fig. 3b. In these results it can be seen that the increase of SPF_{SHP+} per m^2 of collector area after $P_{el,aux} \sim 0$ is very small. This value corresponds to approximately an SPF_{SHP+} of 5. Once this system performance is achieved the increase of collector area or ice storage does not significantly increase the system efficiency.

For building SFH100 the SPF_{SHP+} range from 2 to 4.5 (see Fig. 4), much lower than that of the SFH45. The reason for this is the temperature setting of the heat distribution system (see section 3) which at nominal conditions are 55/50 °C for supply/return respectively. In order to show that high system performances are also possible for high energy demand buildings, a new building labelled SFH100* has been simulated which uses a radiant floor system (the thermal mass of the radiant floor has been subtracted from the building SFH100), and thus the temperatures of the heat distribution system are those used for SFH45, i.e. 30/35 °C for supply/return respectively. Simulations with SFH100* (see dashed-dot lines in Fig. 4 show that high system performance factors can be achieved for high energy demand buildings if a low temperature heating distribution system is used.

Differences of $P_{el,aux}$ from covered (Fig. 4a and uncovered (Fig. 4b collectors for SFH100 are quite significant. Using uncovered collectors the auxiliary back-up is not needed in many of the simulated cases while for covered collectors the $P_{el,aux}$ is significant in many of the presented results (see Fig. 4 dashed lines and right axes). This large difference between covered and uncovered is related to the fact that uncovered collectors are able to provide heat in series at higher temperatures for the evaporator and the heat pump needs to stop during lower periods for having too low evaporator temperatures.

For uncovered collectors, results are very stable when $P_{el,aux} \sim 0$, no matter which of the simulated collector areas and storage volume are employed. Therefore a proper sizing is of great importance.

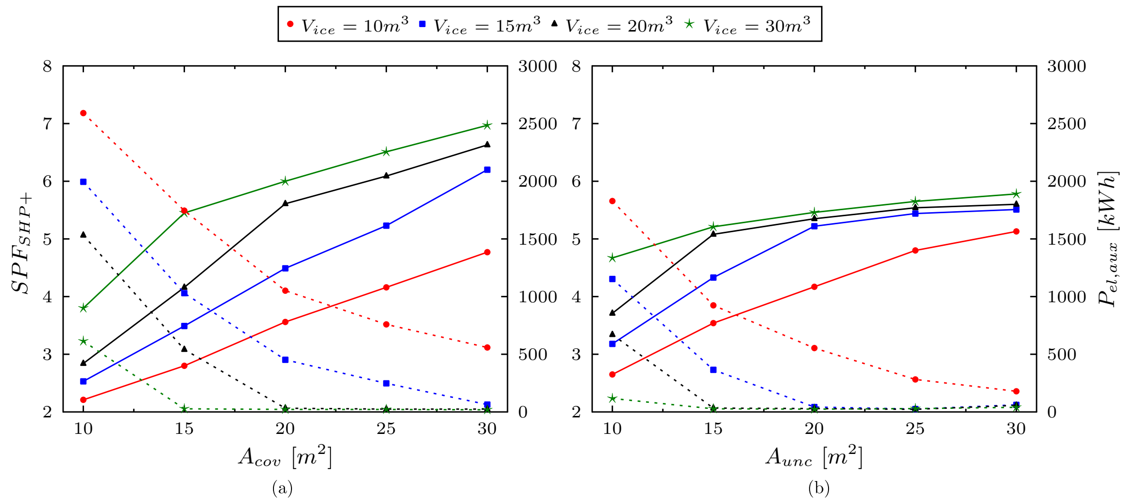


Fig. 3: System seasonal performance factor (solid lines and left axes) and yearly auxiliary energy (dashed lines and right axes) as function of ice storage volume and of covered (a) and uncovered (b) collector area for building SFH45.

In order to easily compare simulations between covered and uncovered collectors, results have been plotted in the same graph in Fig. 5a for SFH45 and in Fig. 5b for SFH100. From Fig. 5a one can see that for SPF_{SHP+} below 5.2 approximately, uncovered collectors perform better. For SFH100 (see Fig. 5b, this threshold is around 4. However, the increase of performance for SFH100 when covered collectors are used for high performances, is much smaller compared to the case of SFH45. Results for large systems where direct electic

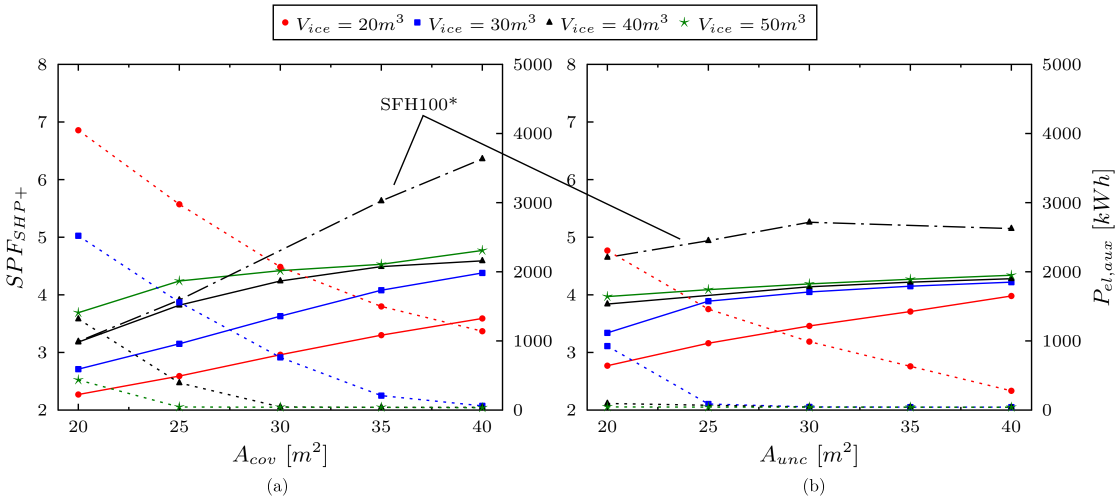


Fig. 4: System seasonal performance factor (solid lines and left axis) and yearly auxiliary energy (dashed lines and right axis) as function of ice storage volume and covered (a) and uncovered (b) collector area for building SFH100. Dashed dot lines are system performance results for SFH100* and 40 m^3 of ice storage.

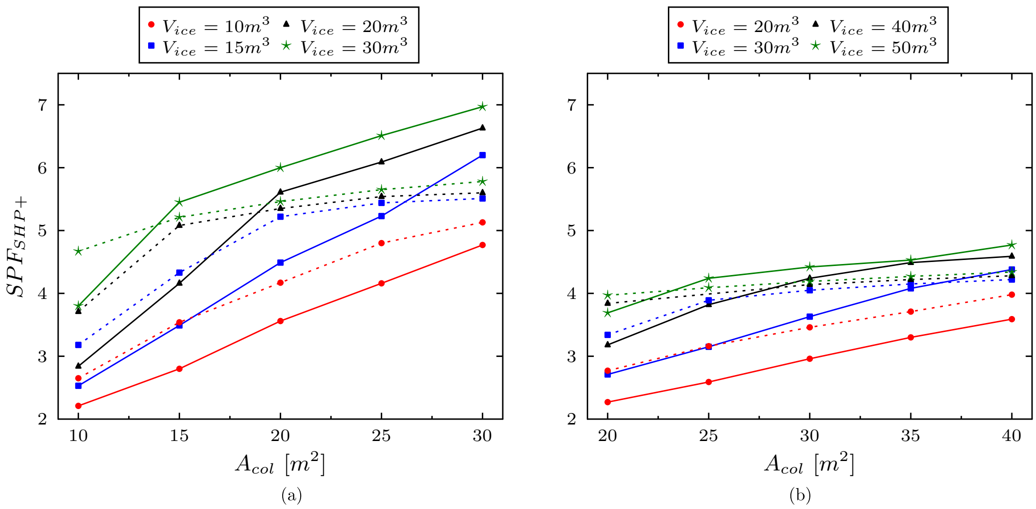


Fig. 5: System seasonal performance factor comparison between covered (solid line) and uncovered (dashed line) collectors as function of collector area and ice storage volume for building SFH45 (a) and SFH100 (b).

heating is not used can be improved if the control is able to switch the loading process from the ice to the warm storage is irradiation is high enought (these results are not shown in this paper).

5.3. Design criteria and definition of reference systems

Simulations presented above show that the system performance factors, SPF_{SHP+} range from 2 to 7 in SFH45 and from 2 to 4.5 in SFH100. The design criteria should be a compromise between system performance, installation cost (cost analyses is not provided here) and building restrictions (space available for ice storage and collector field). Ice storage systems can be seen as an alternative of ground source heat pump (GSHP) systems. Therefore one may consider the system performance of GSHP systems as a reference. Simulations using GSHP systems without any solar collector for SFH45 are in the order of 3.5−4 (see Haller et al. (2014) for example). Another criterion considered to be of importance is a system without the need to use any back-up. The main reason for the latter approach is to help avoiding overcharging the electricity net in very cold weeks in winter, specially in an expected future in which electrical demand will be higher, among other reasons, due to the increase of heat pump installations. With the latter criteria it is not expected to avoid installing the auxiliary system, but to not use it in years with *"normal"* climatic conditions. The back-up should always be installed for extremely cold years unless an oversized system, with its corresponding higher cost, is installed. Therefore two different criteria are considered here for design purposes:

- A system comparable in terms of performance to GSHP systems, i.e with $SPF_{SHP+} \sim 4$. This criteria is labeled as $Cr_{SPF \sim 4}$
- A system without electrical back-up, i.e. with $P_{el,aux} \simeq 0$. This criteria is labeled as Cr_{No-Aux}

Here it should be noted that, due to the higher temperatures of the heating distibution system, a GSHP sytem for SFH100 will most likely not achieve an SPF_{SHP+} of 4. Nevertheless, the criteria $Cr_{SPF \sim 4}$ will be used in this work for both SFH45 and SFH100.

Considering that two buildings have been simulated (SFH45 and SFH100), and that two design criterias are defined ($Cr_{SPF \sim 4}$ and Cr_{No-Aux}) , a total of four systems, described in Tab. 1 as Ref, are used as a reference. For systems designed following criteria $Cr_{SPF \sim 4}$, uncovered collectors are used. For the system without auxiliary back-up (Cr_{No-Aux}) covered collectors are used. The choice of collector type is based on results obtained in section 5.2. For SFH100 the two systems chosen acomplish both criteria $Cr_{SPF \sim 4}$ and Cr_{No-Aux}. Nevertheless, a number of two have been chosen for having a consistent way of presenting results and following the idea of uncovered for $Cr_{SPF \sim 4}$ and covered for Cr_{No-Aux}. For the cases defined in Tab. 1 the use of direct heat from the ice storage for SH or DHW is not improving the results (simulations were conducted to confirm it) and therefore the wall-hxs are not considered. The reason why using wall-hxs are not improving the performance when the system is designed with criteria Cr_{No-Aux} is because this system is chosen as the smallest system without the need of auxiliary back-up. When wall-hxs are used some auxiliary heat is needed for the reasons explained in section 5.1. For the system designed with $Cr_{SPF \sim 4}$ the choice of not using a wall-hx is clear after results shown in section 5.1.

5.4. Waste water heat recovery (WWHR)

Two WWHR systems described in section 2.1 and 2.2 respectively are studied here and simulations for the four reference cases defined in Tab. 1 as Ref are presented including WW_{GFX} and WW_{store}. For these simulations the increase in system efficiency ΔSPF_{SHP+} and the absolute savings $P_{save,el}$ respect to the reference system without a WWHR system (Ref) are presented in the last two columns. Comparing the two WWHR systems, it is clear that the WW_{store} is more efficient than the WW_{GFX}. In the WW_{store} the waste water fluid remains in the storage for large periods allowing a very good heat exchange, while in WW_{GFX} the heat transfer occurs in a short period (while waste water is flushing) and the global efficiency of the heat exchanger is clearly lower.

The system designed with criteria $Cr_{SPF \sim 4}$ in SFH45 has more benefit of using a WW_{store} system than the system using criteria Cr_{No-Aux}. Notice that the the increase in SPF_{SHP+} for building SFH45 by adding waste water use of type WW_{store} is 22.7% for $Cr_{SPF \sim 4}$ and 8.5% for Cr_{No-Aux}. The reason can be explained with the column of the auxiliary back-up $P_{el,aux}$. In systems sized with criteria $Cr_{SPF \sim 4}$, the $P_{el,aux}$ decrease when WWHR are used, specially using the WW_{store} concept. This means that, using WWHR ,the time period where the ice storage is completely frozen and thus, the time period where back-up is used, can be reduced or even avoided. Nevertheless, when criteria Cr_{No-Aux} is used, the back-up system does not need to run for the reference system (this is in fact the criteria used for designing the system). Therefore when using a WWHR system with the Cr_{No-Aux}, the system efficiency improves significantly but not in the same magnitude as in the case with criteria $Cr_{SPF \sim 4}$. As explained in section 5.2, the use of $P_{el,aux}$ greatly affects the system performance. The benefits of using a WW_{store} in SFH100 are not higher when using $Cr_{SPF \sim 4}$ compared to Cr_{No-Aux}, because as explained in section 5.3, for building SFH100 the two chosen reference systems fulfill both $Cr_{SPF\sim4}$ and Cr_{No-Aux} and therefore, none of them needs a significant amount of back-up. The benefits of using a GFX is around 2% in all simulations except for SFH100 and $Cr_{SPF\sim4}$ where little benefit has been found.

Tab. 1: Reference cases without WWHR (Ref) selected following the two design criteria for dimensioning defined in section 5.3. Simulations with WWHR from the selected reference cases (Ref) for SFH45 and SFH100.

System	Building SFH	Design Critera	A_c $[m^2]$	A_{unc} $[m^2]$	V_{ice} $[m^3]$	$P_{el,aux}$ $[MWh]$	$P_{el,T}$ $[MWh]$	SPF_{SHP+} $[-]$	ΔSPF_{SHP+} $[\%]$	$P_{el,save}$ $[MWh]$
Ref	45	$Cr_{SPF\sim4}$	0	15	15	0.37	1.90	4.33	-	-
WW_{GFX}	45	$Cr_{SPF\sim4}$	0	15	15	0.32	1.85	4.43	2.19	0.04
WW_{store}	45	$Cr_{SPF\sim4}$	0	15	15	0.03	1.53	5.32	22.75	0.36
Ref	45	Cr_{No-Aux}	20	0	20	0.03	1.45	5.61	-	-
WW_{GFX}	45	Cr_{No-Aux}	20	0	20	0.03	1.42	5.72	2.02	0.03
WW_{store}	45	Cr_{No-Aux}	20	0	20	0.03	1.33	6.08	8.55	0.12
Ref	100	$Cr_{SPF\sim4}$	0	25	30	0.04	3.97	3.89	-	-
WW_{GFX}	100	$Cr_{SPF\sim4}$	0	25	30	0.04	3.95	3.91	0.32	0.01
WW_{store}	100	$Cr_{SPF\sim4}$	0	25	30	0.03	3.81	4.05	3.95	0.15
Ref	100	Cr_{No-Aux}	30	0	40	0.03	3.66	4.21	-	-
WW_{GFX}	100	Cr_{No-Aux}	30	0	40	0.03	3.58	4.30	2.20	0.08
WW_{store}	100	Cr_{No-Aux}	30	0	40	0.03	3.50	4.40	4.55	0.16

6. Discussions and conclusions

Simulations show that SPF_{SHP+} can range from 2 up to 7 in SFH45 with ice storages from 10 to 30 m^3 and collector areas from 10 to 30 m^2, and from 2 to 4.5 in SFH100 with ice storages from 20 to 50 m^3 and collector areas from 20 to 40 m^2. The same range of performances observed in SFH45 can be achieved for SFH100* (building with low temperature heating distribution system) with the same sizes used for building SFH100. An advantage of ice storage systems is that a certain (and high) system performance can be reached with different combinations of collector area and ice storage volume being able to adapt to building size restrictions. Two strategies for designing ice storage based systems are suggested: i) sizing using the system performance of a ground source heat pump system as a reference ($Cr_{SPF\sim4}$) and ii) sizing the system without the need of electrical back-up (Cr_{No-Aux}).

In order to size the heat exchanger area in the proposed ice storage concept one needs to consider not only the heat pump extraction power but also the thickness of ice on the heat exchanger in order to define the nominal or design conditions. This ice thickness depends on the system design. In simulations for building SFH45, 23 m^2 of heat exchanger area which corresponds to an ice thickness of 1 cm, have been found to be a good value for many simulations. However this is only valid for the concept presented here were de-icing is possible. More studies need to be done in order to find a good criteria for the selection of the heat exchanger area.

In order to select the size and type of collectors one should first think about the system performance that needs to be achieved and the available space for ice storage and collector field. From an efficiency point of view covered collectors perform better at high SPF_{SHP+}, when the auxiliary does not need to run. In building SFH45 the threshold for covered collectors to perform better is relatively high, around $SPF_{SHP+} \sim 5.2$ and therefore, for most of the simulated systems uncovered collectors perform better. For building SFH100* (high temperature heat distribution system) covered collectors can be used to achieve very high SPF_{SHP+} as in the case of building SFH45. The sizes of the system should be selected properly, since the increase of collector area or ice storage may not significantly improve the system performance once a certain level of performance has been reached if direct heat is not used. In general, direct heat is only of benefit when direct electric back up is not used.

Waste water heat recovery systems can be of great benefit for system performance, specially if a waste water storage is used. System performance can increase from 8.5 to 22.7% for building SFH45 and by 4.5 %

for SFH100 respect to the case without WWHR when WW_{store} is used. Using a WW_{GFX} an increase of system performances in the range of 2 % can be expected.

Acknowledgments

The authors would like to thank the Swiss Federal Office of Energy (SFOE) for the financing support received under the project High-Ice.

References

Carbonell, D., Haller, M.Y., Philippen, D., Frank, E., 2014. Simulations of combined solar thermal and heat pump systems for domestic hot water and space heating. Energy Procedia 48, 524–534.

Carbonell, D., Philippen, D., Haller, M.Y., Frank, E., 2013. Development and validation of a mathematical model for ice storages with heat exchangers that can be de-iced, in: Proceedings of ISES Solar World Congress, Cancún, México.

Dott, R., Haller, M., Ruschenburg, J., Ochs, F., Bony, J., 2012. The reference framework for system simulation of the IEA SHC Task44/HPP Annex 38: Part B : Building and space heat load. Technical Report IEA-SHC Task44 Subtask C. www.iea-shc.org/task44.

Hadorn, J.C., 2012. IEA solar and heat pump systems - Solar heating and cooling Task44 and heat pump programme Annex 38. Energy Procedia 30, 125– 133.

Haller, M., Dott, R., Ruschenburg, J., Ochs, F., Bony, J., 2012. The reference framework for system simulation of the IEA SHC Task44/HPP Annex 38: Part A : General simulation boundary conditions. Technical Report IEA-SHC Task44 Subtask C. www.iea-shc.org/task44.

Haller, M.Y., Carbonell, D., Mojic, I., Winteler, C., Bertram, E., Bunea, M., Lerch, W., Ochs., F., 2014. Solar and heat pump systems – summary of simulation results of the IEA SHC Task 44/HPP Annex 38, in: Proceedings of the 11^{th} IEA Heat Pump Conference, Montreal, Canada.

Haller, M.Y., Frank., E., 2011. On the potential of using heat from solar thermal collectors for heat pump evaporators, in: Proceedings of ISES Solar World Congress, Kassel, Germany.

Heinz, A., Lerch, W., Breidler, J., Fink, C., Wagner, W., 2012. Wärmerückgewinnung aus Abwasser im Niedrigenergie- und Passivhaus: Potenzial und Konzepte in Kombination mit Solarthermie und Wärmepumpe - WRGpot. Technical Report im Auftrag des Bundesministeriums für Verkehr, Innovation und Technologie, Österreich. Technische Universität Graz, Institut für Wärmetechnik, AEE INTEC.

Malenkovic, I., Eicher, S., Bony, J., 2012. Definition of main system boundaries and performance figures for reporting on SHP systems. Technical Report IEA-SHC Task44 Subtask B. www.iea-shc.org/task44.

Philippen, D., Carbonell, D., Haller, M.Y., Frank, E., Brunold, S., 2014. Auslegung und Betrieb einer hocheffizienten Solarthermie-Wärmepumpen-heizung mit eisspeicher, in: 24. OTTI Symposium Thermische Solarenergie, Kloster Banz, Germany.

Philippen, D., Haller, M.Y., Logie, W., Thalmann, M., Brunold, S., Frank, E., 2012. Development of a heat exchanger that can be de-iced for the use in ice stores in solar thermal heat pump systems, in: Proceedings of Eurosun, Rijeka and Opatija, Croatia.

Ruschenburg, J., Herkel, S., Henning, H.M., 2013. A statistical analysis on market-available solar thermal heat pump systems. Solar Energy 95, 79–89.