Cost energetic analyses of ice storage heat exchangers in solar-ice systems.

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Abstract

The aim of the present paper is to analyze flat plates (FP) and capillary mats (CM) heat exchangers for solar-ice systems. The optimum heat exchanger area needed in the ice storage is based on energetic transient system simulations and on cost indicators. Solar-ice systems using ice storage volumes from 2 to 5 m^3 and selective uncovered collectors with 15 to 25 m^2 in the city of Zurich were able to achieve system performance factors SPF_{SHP+} ranging from 3.5 to 6 with both CM and FP in a single family house with 10 MWh yearly heating demand. Considering the cost of the system, only simulations with CM were able to achieve lower heat generation cost than that of a ground source heat pump (GSHP) with even higher SPF_{SHP+} (an SPF_{SHP+} of 4 was assumed for GSHP). For example a system with a collector area of 15 m^2 and 5 m^3 of ice storage volume can reach heat generation cost of 29 Rp./kWh, 0.5 Rp./kWh below the GSHP average reference cost, with an increase of SPF_{SHP+} of 20 % respect to the GSHP system. However, the targets can only be achieved using an appropriate heat exchanger area. The optimal heat exchanger area was found to be around 4 to 5 m^2 per m^3 of ice storage for CM and around 10 to 14 m^2/m^3 for FP. These heat exchanger ratios correspond to a distance between heat exchangers of around 12 - 17 cm for both CM and FP. These results were obtained assuming a conservative maximum ice fraction of 80 %.

Keywords: ice storage, heat exchangers, solar-ice.

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1. Introduction

One of the main priorities of the European Commision is to decrease the annual greenhouse gas (GHG) emissions in 2050 by 80% compared to the status of 1990 (Europen Commision, 2012). In order to reach this ambitious goal, the energy supply system needs to be decarbonised. The heating and cooling demands in Europe are responsible of 40 % of the total energy demand. Therefore, the increase of system efficiency as well as raising the share of renewable energy in the heating and cooling sector is necessary to mitigate the climate change by reducing the GHG emmissions. A promising example for a heating system with a high share of renewable energy is the combination of solar thermal and heat pump systems with ice storages, the so-called solar-ice systems. The interest in solar-ice systems is growing in central Europe, where climatic conditions are appropriate for this technology. One of the reasons for the market push of solar-ice systems is due to the regulations established for drilling boreholes. For this reason, solar-ice systems have been established as an alternative to ground source heat pump (GSHP) systems.

Some of the advantages of solar-ice systems respect to GSHP (Carbonell et al., 2016b, 2015) are: i) usually not restricted to regulations of water and soil ii) no need to regenerate the ground as in regions densely populated with boreholes iii) the ice storage is accessible, allowing for solving leakages or replace heat exchangers, iv) the ice storage can be installed in the cellar when no ground space is available v) it is a flexible system able to adapt to building size restrictions, i.e. the same system performance can be reached with different combinations of collector area and ice storage volume and vi) higher performance compared to GSHP can be achieved while being cost competitive if direct solar heat is used extensively. However, solar-ice systems have some disadvantages respect to GSHP such as: i) higher number of hydraulic components, ii) added complexity

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of the control, iii) higher installation cost if the same performance is desired, iv) degradation of performance when ice grows, v) need of a back-up system to cover peak demands after long cloudy and cold periods.

Some of the frequently asked questions and answers concerning solar-ice systems are (Carbonell et al., 2017a):

- *"How can a solar-ice system, which is based on a 0 °C temperature source for the heat pump, perform better than a GSHP system which has usually a higher temperature source?"* The main reason is that a solar-ice system can also use solar heat directly without the need of the heat pump. During times when solar heat is used directly, the system performance in terms of heat provided divided by electricity consumed can be up to fifty times higher compared to a GSHP system (only pumps consume energy). Moreover, since solar collectors are also used directly as a source for the heat pump, the source temperatures in solar-ice systems when the sun is shining can be considerably higher than in a GSHP.
- *"Whats is the use of the ice? Can it be spared?"* The key aspect of the solar-ice system concept relies on reducing the need of the storage volume by making use of the high latent heat of fusion released when ice is formed. Icing a specific quantity of water releases the same energy as cooling the same amount of water from 80 °C to 0 °C. Thus, although remaining always above the solidification temperature of water would lead to higher system performance, the required storage volume would be prohibitively large for this concept.

The principal idea of a solar-ice system is shown schematically in Fig. 1.

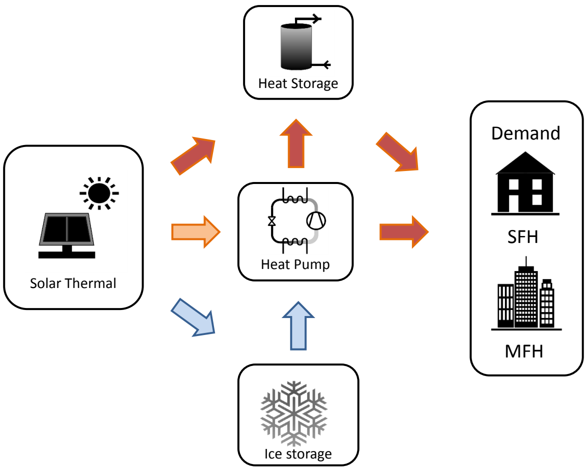

Fig. 1: Principle concept of solar-ice systems. The arrows show the heat fluxes that are on different temperature levels, i.e. red, orange and blue for high (>30 °C), medium (>10 °C) and cold (<10 °C) respectively.

Most of ice storages installed in Europe are based on ice-on-coil heat exchangers, and although other heat exchanger concepts exist on the market, their specific advantages and disadvantages remain unclear.

Several heat exchanger concepts for extracting the latent heat from water can be used. Each concept has to ensure that the ice layer on the specific heat exchanger reaches thicknesses that do not result in too high heat transfer resistance, and thus, in too low source temperatures for the heat pump. When ice grows on the surface of the heat exchanger the overall heat transfer coefficient of the heat exchanger decreases. If ice is not actively removed, the heat exchanger design has to ensure that the heat transfer capacity will be high enough at the maximum design mass ice fraction (ratio between mass of ice and total mass of water and ice) or ice thickness on the surface of the heat exchangers. For example, let's assume that flat heat exchanger plates covering all the height of the ice storage are installed with a distance of 10 cm between each heat exchanger plate. If the heat transfer capacity with 3 cm of ice on the surface is not high enough, the temperature of the heat transfer fluid will drop until the minimum temperature accepted by the heat pump is reached and the heat pump will be stopped for security reasons. If this occurs, the ice thickness can not exceed 3 cm and part of the latent storage capacity will be lost as 2 cm out of the theoretical maximum of 5 cm will not be used. Thus, the maximum ice fraction will never grow above 60% and the desired maximum accepted ice fraction may not be reached.

In principle, two main strategies exist for the design of heat exchangers for ice storages (Philippen et al., 2015) i) ice-on-hx and ii) free-of-ice-hx. In ice-on-hx, large heat exchanger areas, homogeneously distributed throughout the whole storage volume, are necessary. Heat exchangers such as coils, plates or capillary mats can be used. In free-of-ice-hx, several concepts such as ice slurries (Kauffeld et al., 2005) or de-icing, e.g by hot gas in ice harvesting system (ASHRAE, 2007) or by solar thermal collectors (Philippen et al., 2012) can be used. From all these concepts, only the ice-on-hx concepts are established in the solar and heat pump heating market. In this paper, the ice-on-coil and ice-on-plate concepts will be analyzed from an energetic and economic point of view.

2. Methodology

Dynamic system simulations are used in order to assess the heat exchanger area and heat exchanger type on a system level. The simulations have been conducted with the simulation environment TRNSYS-17 (Klein et al., 2010). The basic components to model a solar-ice system are: collectors, heat pump, ice storage, sensible thermal storage, building, climate and control. The ice storage model has been developed and validated in Carbonell et al. (2017b), the remaining component models were provided in Carbonell et al. (2016b), where a complete solar-ice system based on a de-icing concept was validated with monitored data of a pilot plant.

The time step of yearly simulations is set to 120 seconds. As a verification process several systematic checks are done for all simulations. Heat balances are checked in all individual components, hydraulic loops and also from the system perspective. The convergence criteria from TRNSYS is set to 5e-4, which allows to achieve heat imbalances always below 1.% with respect to the total heat demand. Iteration problems are also checked for all simulations and are always below 10 time steps per year simulated. In most cases iteration problems are in the order of 1 - 5 per year.

2.1. Hydraulic scheme of the simulated system

The hydraulic scheme of the complete solar-ice heating system is shown in Fig. 2. The main components of the heating system are the collector field, the combi-storage, the heat pump and the ice storage.

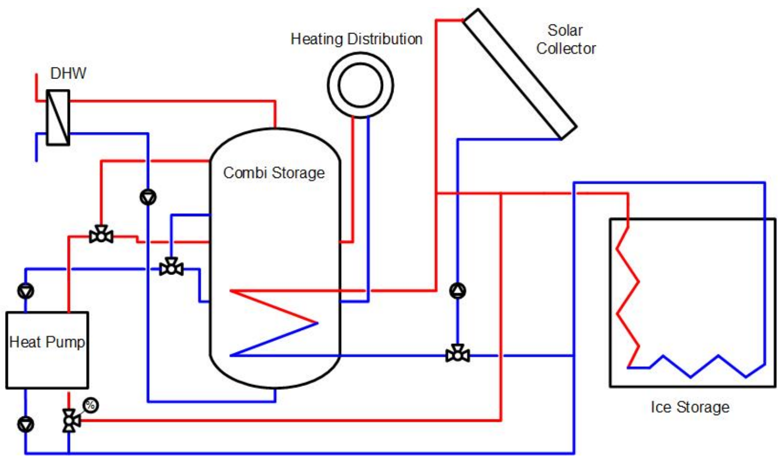

Fig. 2: Simplified hydraulic scheme of the analysed heating system.

The main energy source of the system is the solar irradiation. Some additional energy is extracted from the air, especially when uncovered collectors are used [1]. Part of the total solar irradiation is transformed by the collectors to useful heat for the system. This energy is transferred, either to the heat pump, to the ice storage, or to the combi-storage. When the heat pump is running, two operation modes are possible, depending on whether the collectors are able to provide energy or not: i) the heat pump uses the solar energy directly in a series operation mode, meaning that collectors are used as heat sources or ii) the heat pump uses only the ice storage as its source. With the hydraulic set-up proposed here, when the heat pump is running with solar energy as a source, the mass flow from the solar collector field is divided: one part goes to the heat pump and the other part goes to the ice storage. The split of the mass flow is basically controlled by each circulation pump, i.e. solar pump and heat pump evaporator pump. If these mass flows are not equal, its difference will flow to the ice storage. On these conditions, if more energy is available from the collector field than is needed by the heat pump, the ice storage is also loaded while the heat pump is running. On the contrary, if the collector output is lower than the heat pump needs, both, the ice storage and the collector field are used to provide heat to the evaporator of the heat pump. If none of these heat sources are available, which means that the ice storage is full of ice and the solar radiation and the ambient temperature are very low, the temperature of the heat pump evaporator drops below the minimum allowed value and the heat pump stops. In this case, a direct electric back-up is used. A method to reduce the need of back-up in winter times is to send the whole mas flow from the collector to the heat pump. The idea is to use the collectors even if the temperature is lower than 0 ^{o}C[2]. This can be achieved with the proposed hydraulic scheme by setting the mass flow from the collector equal to the one demanded by the heat pump. This tends to reduce the working temperature of the circuit and as a consequence decrease the heat pump COP. However, it reduces the times where the ice storage is used and thus the times where the direct electric back-up is used. Usually, reduction of the time use of the direct electric back-up is compensated by the loss of heat pump performance.

2.2. System control

The system part on the secondary side of the heat pump, i.e. the heating distribution and the DHW-preparation, and the heat pump itself are controlled in a standard way. The brine cycle on the primary side, on the other hand, needs some special operation modes to decide how to use the solar heat. The backup needs to be controlled too, such that it runs when there is a heating demand in the building and the ice storage is fully iced (no source available for the heat pump). Further, season-based priorities regarding the use of the solar heat are implemented. The solar-ice system has a global control with three main priorities in the following hierarchy:

1. Use of direct solar heat to provide the space heat to the building without switching on the heat pump.
2. Switching on the heat pump when not enough energy is available in the combi-storage in order to provide space heat. When heat pump is on, solar energy is prioritized as energy source.
3. Use solar heat to load the combi-storage and ice storage when the heat pump is off. Loading the combi-store is usually referred as direct solar heat. Direct solar heat is usually prioritized in spring, summer and autumn, but in winter the loading of the ice storage is prioritized.

The control has to cope with the fact that the ice storage is a cold sink for a long time during the year. Unless a logic is implemented, that switches to loading of the combi-storage when appropriate, the ice storage would be loaded predominantly. The control mode that actively stops the loading of the cold storage and tries to divert the solar heat on a higher temperature level to the combi-storage is called warm storage priority. If the warm storage priority is not active, the control mode cold storage priority is on, which loads the ice storage. For small sized systems regarding collector field and ice storage it is usually a better option to use a cold storage

[1] All results shown in this paper are obtained with selective uncovered collectors.
[2] In this case the ice storage would provide a higher temperature, i.e. around 0 ^{o}C.

priority in winter in order to avoid the time when the direct electric back-up is needed. As soon as there is no risk to fully ice the storage, then warm storage priority should be used.

2.3. Performance indicators

The main performance indicator for the systems is the System Performance Factor 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 energy demand and $P_{el,T}$ the total yearly electric 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 consumption is calculated as:

$$P_{el,T} = P_{el,pu} + P_{el,hp} + P_{el,cu} + P_{el,back-up} + P_{el,pen} \tag{2}$$

where the subscripts pu, hp, cu, aux and pen refer to circulation pumps, heat pump, control unit, back-up and penalties respectively. 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 includes all circulation pumps of the system and also all thermal losses/gains from storages and piping. Penalties for not providing the heating demand at the desired comfort temperature are calculated according to Haller et al. (2012). $P_{el,aux}$ is the energy used from the direct electric back-up system.

3. System energetic performance with varying heat exchanger type and area

Dynamic yearly system simulations have been carried out for different heat exchanger areas and for two types of heat exchangers, capillary mats (CM) and flat plates with stainless steel (FP-SS). Simulations have been performed for ice storage volumes of 3 m^3, 4 m^3 and 5 m^3 and collector areas of 15 m^2, 20 m^2 and 25 m^2. All simulations are carried out using a single family house building located in Zurich with approximately 9.5 MWh of total heating demand for SH and DWH.

Results for the SPF_{SHP+} are shown in Fig. 3 as a function of the ratio between the heat exchanger area and the ice storage volume and as a function of the distance between heat exchangers. The same distances between heat exchangers are used both for CM and FP-SS. However, since FP-SS have almost twice the area[3] compared to CM, results for FP-SS are shifted towards higher A_{hx}/V_{ice} axis. Both heat exchanger types are able to provide very high SPF_{SHP+}, up to 6 for the sizes used in these simulations. System performances above 4[4] are always achieved if the collector area is $\geq$ 20 m^2. For collector areas in the order of 15 m^2, an SPF_{SHP+} above 4 can be achieved with CM and V_{ice} = 4 m^3 and with FP-SS if V_{ice} = 5 m^3. Using a collector area of 15 m^2 and an ice storage volume of 5 m^3 both heat exchangers are able to provide an SPF_{SHP+} above 4.

Very high system performances, in the order of 5 to 6, can be achieved with different component sizes. These results confirm the flexibility of the system concept, i.e. the same SPF_{SHP+} can be obtained using different configurations of ice storage volume and collector area. This allows to achieve a specific SPF_{SHP+} even if there are restrictions such as small cellar capacity or available roof area. However, there is a limit on the lowest size of the collector field since the solar-ice system is based purely on solar energy as heat source. The system flexibility of solar-ice systems was discussed in Carbonell et al. (2014). Of particular interest are results with 4 m^3, since this storage volume could fit in many cellars of single family homes using two separated storages of 2 m^3 allowing to achieve an $SPF_{SHP+} \geq 5$ with collector areas of 20 m^2.

[3] The two faces of the flat plate heat exchanger are considered as heat exchanger area.
[4] As a reference GSHP is used with an SPF_{SHP+} of 4.

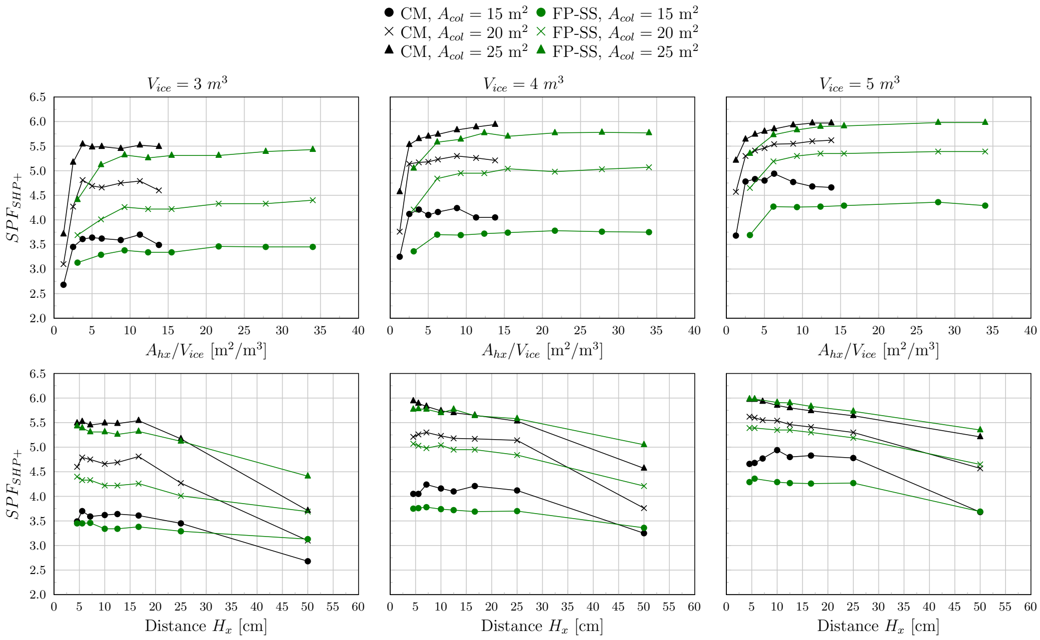

Fig. 3: Yearly system performance as function of (top) heat exchanger area divided by ice storage volume and (bottom) distance between heat exchangers for three ice storage volumes of (left) 2 m^3, (mid) 4 m^3 and (right) 5 m^3.

Regarding the heat exchanger type, the SPF_{SHP+} is usually higher for CM compared to FP-SS except for large distance between heat exchangers, e.g. $>$ 25 cm for V_{ice} =3 m^3. System performances for CM when distances between heat exchangers are small are higher because of the better heat transfer coefficient when icing at high ice fractions. The SPF_{SHP+} of CM is worse than that of FP-SS for cases where the distance between heat exchangers is way higher than that between the tubes in one CM. One can imagine a CM like a FP with empty spaces between tubes. Let's imagine the limit case where all tubes in one CM are in contact to each other. Under those circumstance CM would be worse than FP because of the higher resistance of polypropylene compared to stainless steel. When ice fills all the gaps between tubes, there is the added limitation that not all area is used due to the spaces between tubes.

Differences between FP-SS and CM are more prominent for low collector areas and low ice storage volumes. Results for 25 m^2 and 5 m^3 tend to an asymptotic solution where differences between CM and FP-SS are negligible. In those situations the direct electric back-up is not used because the system components are sized largely enough such that not all the latent heat capacity of the storage is necessary. The direct electric back-up is shown in Fig. 4 as a function of the ratio between A_{hx} and V_{ice}. Clearly, the increase of collector area and storage volume decrease the use of the direct electric back-up. The direct electric back-up is used exclusively in winter, when the ice storage if full (maximum ice fraction of 80 % has been assumed) and there is not enough solar energy to provide the heat for the evaporator of the heat pump. The use of the electric back-up is the dominant factor that influences the SPF_{SHP+}. In larger systems where the electric back-up is avoided, the system performance is quite independent of the heat exchanger area and less dependent on the collector area or ice storage volume. A SPF_{SHP+} value below 5.5 usually indicates the need of direct electric back-up.

In order to establish the optimum heat exchanger area in terms of SPF_{SHP+} it is of importance to investigate the maximum ice fraction achieved, as shown in Fig. 5. If the heat exchanger area is too low for the specific system design, some latent heat may not be used because the heat transfer coefficient will be too low at the maximum ice thickness achieved. This would lead to a situation where the heat pump would not be able to extract enough power from the ice storage while there is still liquid water present. For CM the maximum ice fraction of 80 % is reached approximately for ratios of A_{hx}/V_{ice} of 5-10 m^2/m^3. For FP the maximum ice fraction is reached at

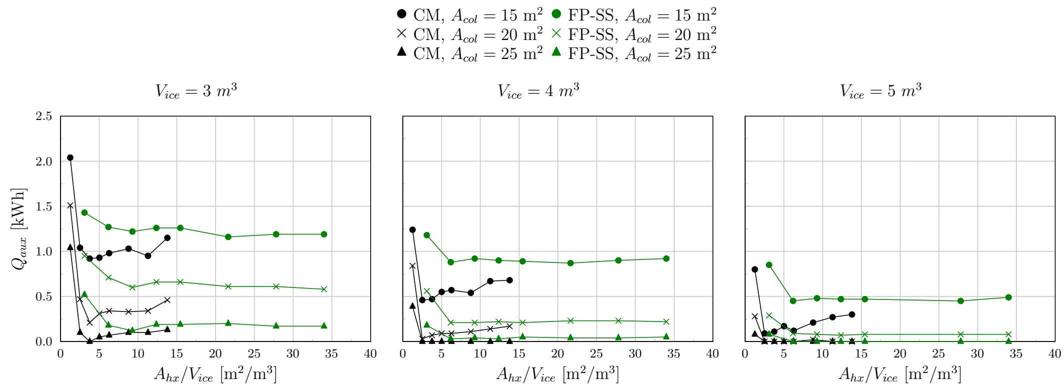

Fig. 4: Direct electric back-up as a function of heat exchanger area divided by ice storage volume for three ice storage volumes of (left) 2 m^3, (mid) 4 m^3 and (right) 5 m^3.

higher ratios of 10-15 m^2/m^3. For both cases these ratios correspond to a distance of around 7-12 cm between heat exchangers.

For some configurations, it is not possible to use all latent heat of the storage. For example for $V_{ice} = 5$ m^3 and $A_{col} = 25$ m^2, non of the heat exchangers setup can achieve an ice fraction of 80 %. This indicates that the system is oversized and either the ice storage or the collector area could be decreased and the system performance would not be penalized much. For example in Fig. 3 the SPF$_{\text{SHP+}}$ of $A_{col} = 25$ m^2 is almost the same if the storage volume is 4 m^3 or 5 m^3. Another way to make use of the latent heat and also to increase the SPF$_{\text{SHP+}}$ would be to give more priority to the warm storage loading.

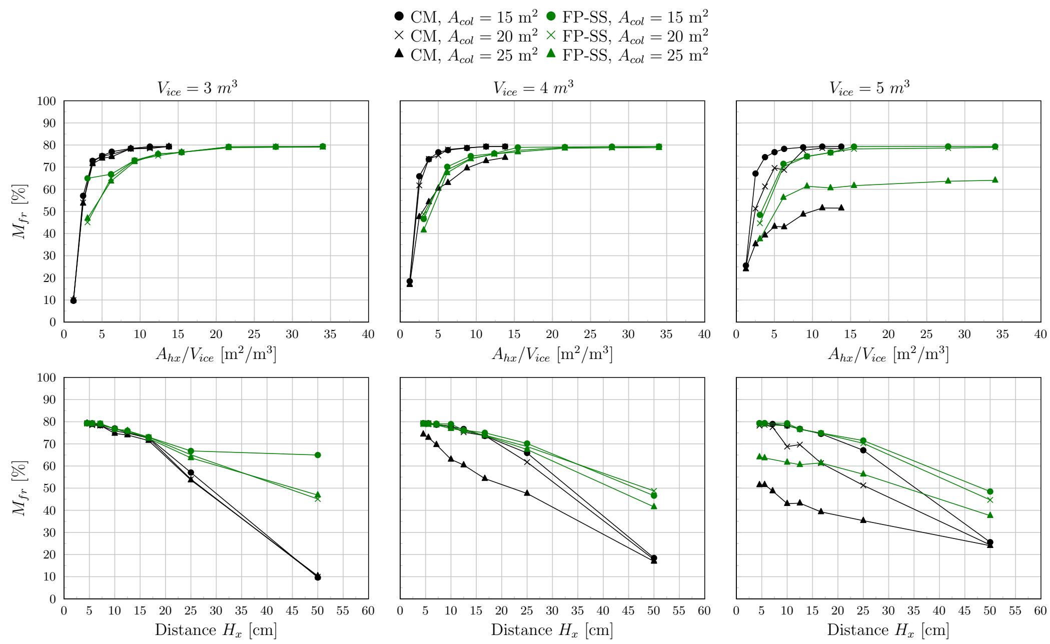

Fig. 5: Maximum yearly mass ice fraction as a function of (top) heat exchanger area divided by ice storage volume and (bottom) distance between heat exchangers for three ice storage volumes of (left) 2 m^3, (mid) 4 m^3 and (right) 5 m^3.

4. Cost analyses of solar-ice systems

In the section above the system performance in terms of energy has been provided. In this section, costs are taken into consideration in order to further evaluate different system designs. For all analyzed heating systems, investment costs and heat generation costs for prices of the Swiss energy market in 2016/2017 are calculated. The comparison of costs is used to find the optimum heat exchanger area considering both energetic system performance and a good value for the money.

The investment costs of the solar-ice systems are based on real costs that were gathered from several sources. Some data was obtained from the realization of two demonstration projects of solar-ice systems in Rapperswil-Jona, Switzerland. The data from the solar part was obtained from the SFOE project ReSoTech (Philippen et al., 2016). This data was used to derive cost functions per component. The cost functions are then used to calculate investment costs per specific system size simulated. All received cost functions were verified by a Swiss seller of heating systems in terms that they represent actual average market prices.

A ground source heat pump system is used as a reference to compare the system performance in terms of energetic efficiency and cost. Using a borehole length of 130 m for a single family house in the region of Zurich an SPF_{SHP+} around 4 is expected. Investment costs for GSHP are based on two offers of Swiss sellers for a GSHP system that supplies heat to a single family house (Causi, 2010). The cost functions were derived from average costs of the two offers. These two offers were used to estimate an uncertainty range on the heat generation cost. The average heat generation cost of a GSHP system based on the annuity is calculated as 39.6 ± 3.3 Rp./kWh.

For each system the present value of costs and the annuity are calculated following the methodology of VDI (2012) and Bangerter (1985) with some simplifications. Heat generation cost are calculated using the annuity factor. Main assumptions for the economic analysis are given in Table 1 and details for the calculation method were provided in Philippen et al. (2015). The electricity prices are taken from the price list of a regional Swiss utility and represent typical prices for small customers. The increase of electricity cost is based on the assessment of this utility. An interest rate of 2% is assumed, which represents an average interest used in credits for buying a house or for refurbishing it in Switzerland. An increase of the interest rate would shift all heat cost predictions to higher values, but differences between systems on the range of interest would be similar.

Tab. 1: Assumptions for calculation of heat generation costs.

Rate of interest	2.0 % p.a.
Analysis period	25 years
Yearly Maintenance	0.25 % of investment costs
Lifetime	25 years
Electricity costs (incl. VAT)	Fixed costs: 171 Fr. per year Variable costs: 0.13 Fr. per kWh
Increase of electricity costs	1 % p.a.

The system performance (top) and the heat generation cost (bottom) are shown in Fig. 6 as a function of the distance between heat exchangers for CM and FP-SS for $A_c = 15$ m^2 and all storage volumes analyzed in the section above. The SPF_{SHP+} is repeated here in order to compare directly graphs of SPF_{SHP+} and cost. The reference values for the GSHP system are included in all graphs. The objective is to achieve a solar-ice system with same or higher performance than GSHP and with similar heat generation costs. Therefore, the focus is on solar-ice systems with an $SPF_{SHP+} \geq 4$. It should be noted that cost calculations are based on offers for installations in single family houses that vary significantly among each other. Therefore, heat generation cost should be seen as orders of magnitude. To consider the large variation a 10% error band in the installation cost has been assumed for all cases. The uncertainty range of the heat generation cost of the GSHP is estimated as 39.6 ± 3.3 Rp./kWh which can be visualized in Fig. 6 and 7 as a gray shadow area.

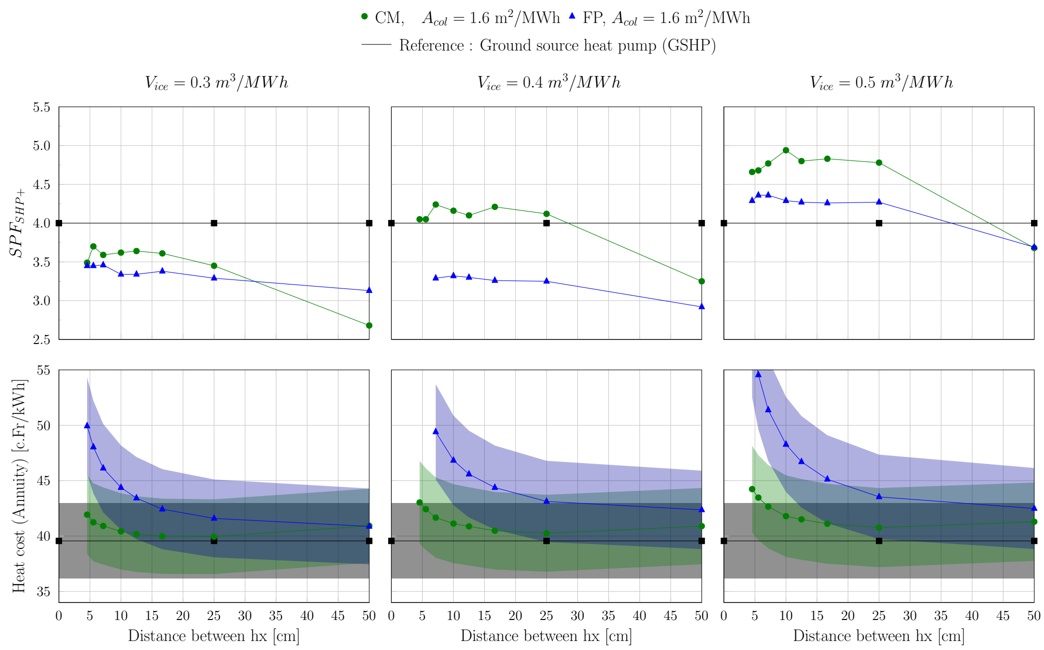

Fig. 6: System performance (up) and heat generation cost (bottom) as function of the distance between heat exchangers. Three ice storage volumes are used: (left) 3 m^3, (mid) 4 m^3 and (right) 5 m^3.

The heat generation cost calculated using the annuity factor are shown in bottom part of Fig. 6. There are two configurations where the $SPF_{SHP+} \geq 4$ and the heat generation cost are within the uncertainty range of the GSHP. Both configurations are using 15 m^2 of collectors, and 4 (CM) and 5 (CM and FP) m^3 storage volumes. These two system set-ups are of particular interest due to the cost comparable to GSHP and high efficiency. All systems using FP-SS show higher heat generation cost compared to CM. However, FP-SS are oxygen tight, and thus black steel can be used in the collector loop. In the other hand, CM made of polypropylene are usually installed with stainless steel pipes in the collector loop increasing its cost. Unfortunately, this cost difference has not been considered in the present calculation.

Using the cost function alone is not a good approach to decide on the heat exchanger area. For example, the lowest area seems to provide the cheapest system even when the energetic performance is relatively low. Thus, besides the cost function, the SPF_{SHP+} and also the maximum ice fraction achieved as shown in Fig. 5 should be used. Considering all these values it seems that the optimum heat exchanger area is in the order of 4-5 m^2/m^3 for CM and around 10-14 m^2/m^3 for FP-SS. Those ratios correspond to distances between heat exchangers of 12 - 17 cm for both heat exchanger types.

The heat generation cost along with the system performance are shown in Fig. 7 as a function of the collector area using a distance between heat exchangers of 16.6 cm. The heat generation costs increase linearly with collector area. The minimum of the cost function can not be observed with the simulated collector area.

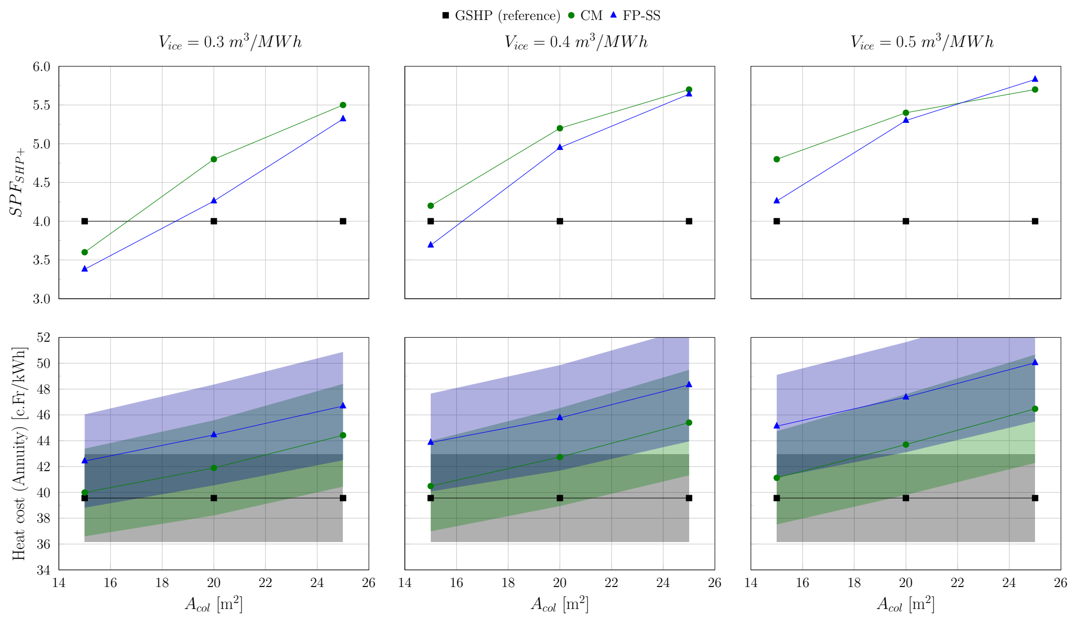

Fig. 7: System performance (up) and heat generation cost (bottom) as function of the collector area and ice storage volume for a distance of 16.6 cm between heat exchangers.

5. Conclusions

In this paper, the complete solar-ice system has been simulated with a well validated ice storage model from Carbonell et al. (2016a). From the system simulations the following conclusions can be drawn:

- A solar-ice system with ice storages volumes between 0.3 and 0.5 m^3/MWh (of total heating demand), and collector areas between 1.6 to 2.1 m^2/MWh can reach an SPF_{SHP+} in the range of 3.5 to 6 in Zurich for a single family home with 9.5 MWh total (SH + DHW) heating demand.
- Besides the sizes of the ice storage and collector field areas, the most relevant factor affecting the SPF_{SHP+} is the electric back-up necessary to run the system in winter periods. Control strategies should be focus on reducing it as a first target. When the electric back-up is minimized, control strategies should focus on the maximization of direct solar heat.
- Solar-ice systems can reach very high SPF_{SHP+} regardless of the heat exchanger type, but when the heat generation cost are included, capillary mats seem to be the most economic option. However, the heat exchanger area needs to be well design to reach these targets. An ice storage volume of 4 m^3 with capillary mats and 15 m^2 of collector area can reach an SPF_{SHP+} of 4.2, which represents an increase of SPF_{SHP+} of 5 % respect to GHSP with a similar heat generation cost.
- Considering both energetic and economic indicators, capillary mats are found to be the most promising solutions for solar-ice applications when compared to the other heat exchangers analysed along the project. Nevertheless, other factors could be used to choose other heat exchangers. For example, one may choose flat plates made of stainless steel instead of capillary mats due to the robustness of the heat exchanger, with a life time guarantee in the range of 25 years. Another reason to select stainless steel heat exchangers can be that they are oxygen tight and therefore an expensive stainless steel piping system in the collector loop can be avoided. The cost of the stainless steel piping system needed when plastic capillary mats are used could overcome the more expensive heat exchangers. However, this has not been considered in the cost calculations, where the piping system was the same for all heat exchangers.

- A strategy to reduce the direct electric back-up in small systems, e.g. using ice storages in the range of 0.3 - 0.4 m^3 per MWh of total heating demand and collectors areas in the range of 1.6 - 2.1 m^2/MWh, is to reduce the working temperature of the primary brine loop connecting the solar collectors, ice storage and heat pump. For those systems, priority should be set to use the collectors as primary source for the heat pump in winter, even that the working temperature decreases well below 0 ^{o}C. In those circumstances, the use of the ice storage would increase the instantaneous COP of the heat pump. However, it would increase the operating times of the direct electric back-up. The better instantaneous heat pump COP can hardly compensate the higher use of the electric back-up at a COP of 1. For larger systems, where the back-up is seldom used in winter, the control should give priority to the combi-storage. Those systems will have roughly an SPF_{SHP+} in the order of 5.5 or above. However, results form this paper suggest that only "small" sized systems with an SPF_{SHP+} around 4 - 4.5 could be cost competitive.

Acknowledgments

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