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Simulations of combined solar thermal and heat pump systems for domestic hot water and space heating

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Abstract

The system combination of solar thermal collectors and heat pumps is a very attractive option for increasing the renewable energy usage at worldwide level for heating and domestic hot water preparation. In this work parallel and series combined solar and heat pump systems are analyzed within the IEA SHC Task44/ HPP Annex38 reference conditions for different buildings and a typical Central European climate. Three combined systems have been studied in detail: solar and air source heat pump, solar and ground source heat pump, and exclusively solar source heat pump in combination with an ice storage. Numerical calculations have been performed using two simulation platforms: TRNSYS-17 and Polysun-6®. Comparisons between the two simulation environments have been also provided. Moreover, a reference case without solar has been used to determine the potential efficiency benefits of using solar collectors compared to a system with a heat pump alone in the specified climate and for covering the specified heat load. Simulations presented in this work show that differences in system performance up to 4% can be expected between TRNSYS-17 and Polysun-6® for air source based systems, and higher discrepancies, up to 14% are obtained for ground source based systems. Comparisons between combined solar thermal and heat pump systems with their respective "heat pump only" reference solutions show that the absolute electricity savings of air source are usually higher compared to ground source based systems. Systems using large ice storages are able to reach seasonal performance factors in the range of 5, which is of the order of performance of combined solar and ground source heat pump systems.

Keywords: solar and heat pumps; simulation

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1. Introduction

The need for reducing fossil fuel dependency is pushing the combination of different technologies in order to increase the renewable energy usage at worldwide level. Among them, the combination of solar thermal and heat pumps is an attractive option for heating and domestic hot water preparation with a high share of local renewable energy use. Nevertheless, when combining the solar thermal technology with heat pumps several problems may occur that need to be avoided. The combination leads to a more complex system where poor design can lead to a significantly lower performance than expected. Control strategies and system hydraulics, in particular the combination of heat pumps with combi-storage tanks, can strongly influence the system performance and have been identified as a possible source of poor performance [1]. System analyses are of importance to provide recommendations for system design and standard solutions, which are key aspects to allow these combined technologies to spread worldwide.

In this work both parallel and series combined solar and heat pump systems are considered. Parallel systems have the advantage of being less complex than the series ones in terms of hydraulic connections and system control and therefore, parallel systems may be more robust and reliable. Nevertheless, some series systems based on solar assisted heat pumps with brine or ice storages are also of interest. A system with an ice storage can be an attractive alternative to a ground source heat pump when, for example, regulations forbid to drill a borehole. These systems can also be seen as an alternative to air source systems when efficiency or noise problems are of importance. Therefore, one type of these systems based on a large ice storage with immersed flat plate heat exchangers that can be de-iced is also studied in this work. Moreover, a reference case without solar is used to determine the potential benefits of using solar collectors compared to a system with a heat pump alone. In order to have the whole picture of the benefits of adding solar thermal to a heat pump, it is necessary to simulate the systems under several climates. This is out of the scope of the present work and has been presented in a separate paper [2].

Reference conditions defined 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" [3], known under the combined name Task44/Annex38 (T44/A38) are employed in order to analyze different systems combining solar and heat pumps under the same boundary conditions. Results are presented for three buildings and the typical Central European moderate climate of Strasbourg.

The need for reliability in the results when modeling different systems is a key aspect and since validation with experimental data for all systems is very time consuming, two simulation environments have been employed: TRNSYS-17 (http://www.trnsys.com) and Polysun-6® (http://www.polysun.ch). The simulation with both tools is not only useful to find inconsistencies in each simulation, but also it forces to analyze possible sources of difference and therefore helping to understand the different hypothesis used that may cause the discrepancies. Furthermore, it is also important to validate and analyze modeling tools used for planners and engineers since most of the people who will install these systems will not use TRNSYS, which more widely employed in the research community, and reliable and robust modeling tools are very important for a correct design of these systems.

2. Methodology

Simulations have been conducted using two simulation platforms: TRNSYS-17 (TN) and Polysun-6® (PS). The present TN simulations are carried out using the state of the art of the component models that have been validated separately in different works. The system configuration in TN is being extensively used and improved by several research institutes, most currently in the framework of the ongoing EU-FP7 project "MacSheep" (http://macsheep.spf.ch/).

The space heating (SH) loads have been introduced as a heat sink element in PS. The heating loads were previously calculated with TN building model Type-56. The domestic hot water (DHW) tapping profile is obtained from T44/A38 [4], the DHW set temperature is 45°C and the cold water temperature is 10°C.

Results have been obtained for three buildings SFH15, SFH45 and SFH100 of T44/A38 (see [4,5] for details) in Strasbourg. The three buildings represent a low (SFH15), medium (SFH45) and high (SFH100) building energy demand (Q_d), where SFH stands for Single Family House and the numbers, 15 for example, for the yearly energy demand in kWh/m^2 per building heated surface area in the city of Strasbourg. The buildings SFH15 and SFH45 have low temperature heat distribution systems ($T_{flow,max}$=35°C and $T_{return,max}$=30°C) and the building SFH100 has a higher temperature heat distribution system ($T_{flow,max}$=55°C and $T_{return,max}$=45°C).

The combined parallel systems consist of a combi-storage as a connecting component between the heat delivered from the heat pump, the solar thermal heat input, and the useful energy delivered to DHW or SH. The considered parallel systems are a combination of a Solar thermal system with an Air Source Heat Pump (SASHP) and a Solar thermal system with a Ground Source Heat Pump (SGSHP). In the series systems analyzed here, the solar energy can be directed to the combi-storage or to the heat pump, either directly or indirectly through an ice storage. Therefore, strictly speaking, the system concept is based on a combination of parallel and series. For simplifications reasons the concept of this system will be referred to as series. This system is labeled as a combined Solar and Ice storage Source Heat Pump (SISHP). In order to show the possible energy flows within the systems the energy flow chart of T44/A38 introduced by Frank et al. [6] is used in Fig.1 to present the scheme of the parallel SASHP (left) and the series SISHP (right) systems.

The combi-storage has separate connections for charging the storage DHW and SH zones by the heat pump. As recommended by Haller et al. [1], the return line from the storage to the heat pump in DHW charging mode is above the zone affected by SH operation, and the position of the sensor used for DHW charging control is well above the space heating zone of the combi-storage. For PS simulations it is important to have the return line from the storage to the heat pump in the DHW section at least one layer above the inlet of the heat pump to the SH section of the combi-storage. Otherwise some numerical mixing occurs due to the low and fixed number of control volumes (n_{cv}=12) included in the PS storage tank.

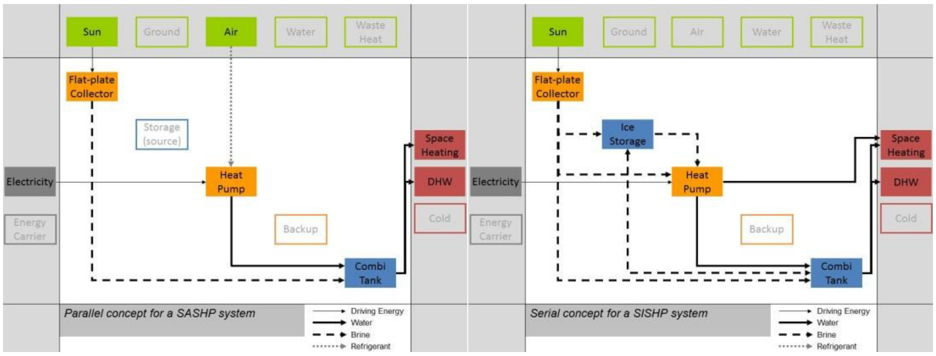

Fig. 1 Energy flow chart visualization scheme [4] for parallel SASHP (left) and SISHP (right).

The "heat pump only" reference system is defined similar as the combined system (same components) but without the solar part. Since systems using only heat pumps will most likely be installed without a combi-storage, the reference system is designed with two storage tanks; one 300 l tank for DHW and another 200 l tank for SH.

Polysun-6® has a very versatile user defined control function that allows implementing the control used in TN more easily. Therefore the same control strategies were implemented in the two simulation platforms.

All results presented using PS have been obtained from hourly results data with a validated post-processing tool. The post-processing has been used to recalculate some values as for example the electricity consumption of circulating pumps or control units for the heat pump and the solar thermal system. Moreover, user defined performance indicators, as the ones defined in section 2.1, were calculated from hourly values. For the electricity consumption of the control unit a constant value of 3 W during all the year has been assumed. Energy balances for every loop are also calculated to ensure proper use of values when computing the performance indicators. The same post-processing is also employed for TN simulations in order to avoid differences in the post-processing task.

For each simulation a platform independent check from T44A38 [7] has been performed to test validity of the results under the comparison within the same reference conditions. Moreover all simulations have been checked in terms of energy balances at a component and system level.

2.1. Performance indicators

In order to analyze the system performance and the improvements with respect to a reference system, several performance figures need to be defined. The main performance indicator for the system is the system's Seasonal Performance Factor calculated as described in [8] by:

$$SPF_{SHP+} = \frac{\int (\dot{Q}_{DHW} + \dot{Q}_{SH}) \cdot dt}{\int \dot{P}_{el,T} \cdot dt} = \frac{Q_{DHW} + Q_{SH}}{P_{el,T}} \tag{1}$$

where dt is the time step in [s]; $\dot{Q}$ is the heat load power in [W] and $\dot{P}_{el,T}$ the total electrical consumption power in [W]; Q is the yearly heat load energy and $P_{el,T}$ the total yearly electrical energy consumption. The subscripts SHP, DHW and SH stand for solar and heat pump, domestic hot water and space heating respectively. The total electricity consumption is calculated as:

$$\dot{P}_{el,T} = \dot{P}_{el,pu} + \dot{P}_{el,hp} + \dot{P}_{el,cu} + \dot{P}_{el,aux} \tag{2}$$

where the subscripts *pu*, *hp*, *cu* and *aux* refer to circulation pumps, heat pump, control unit and auxiliary 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 term $\dot{P}_{el,pu}$ used in this work include all circulation pumps of the system.

The seasonal performance factor of the heat pump alone is defined as:

$$SPF_{HP} = \frac{\int \dot{Q}_{con} \cdot dt}{\int \dot{P}_{el,hp} \cdot dt} \tag{3}$$

where $\dot{Q}_{con}$ is the heat delivered by the condenser of the heat pump. In order to compare results from one simulation of a particular system (SHP+) to the reference system (ref) the relative increase of SPF is used:

$$\Delta SPF_{SHP+} = \frac{SPF_{SHP+} - SPF_{ref}}{SPF_{ref}} \tag{4}$$

Another figure for the comparison between systems is the fractional solar electricity savings defined as:

$$f_{save,el} = 1 - \frac{(P_{el,T})_{SHP+}}{(P_{el,T})_{ref}} \tag{5}$$

The absolute electricity savings are calculated as follows:

$$P_{save,el} = (P_{el,T})_{ref} - (P_{el,T})_{SHP+} \tag{6}$$

3. Results

Three systems have been analyzed, two of them are parallel and one is in series. The considered parallel systems are a combination of a solar thermal system with an air source heat pump (SASHP) and a solar thermal system with a ground source heat pump (SGSHP). Their respective reference systems without the solar part are referred to as ASHP and GSHP. The last system analyzed is a series combination of a solar with a large ice storage as a source for the heat pump (SISHP).

The systems based on ice storage were not simulated with PS because of the special kind of ice storage model (see [9] for details) needed that is not currently implemented in PS. The “heat pump alone” systems are only calculated in PS because including a new system is much easier with this simulation tool. Despite of the fact that TN and PS can predict different system performance in absolute terms, the relative difference between two simulations, i.e. combined solar and heat pump and "heat pump alone" systems, within the same platform are not expected to be very different between TN and PS. Therefore the study with only one simulation platform should be enough in this case.

3.1. Analysis of parallel systems. Comparison between TRNSYS and Polysun-6®

A comparison task between TN and PS simulations for combined SASHP and SGSHP parallel systems has been undertaken to obtain insights into the capabilities and limitations of each modeling platform. Numerical results for all buildings and simulation environments are presented in Table 1. All this simulations are obtained using 15 m^2 of collector area (A_C).

The first two columns of the central section of Table 1 are used to present the amount of energy the heat pump provides to DHW, $Q_{HP,DHW}$ and to SH, $Q_{HP,SH}$. These terms are particularly important for the heat pump performance and mostly depend on control settings and positions of the inlet and outlet connections between the heat pump and the storage, as well as on the stratification capabilities of the storage [1]. For SFH15 and SFH45 the DHW section of the combi-storage is at higher temperatures compared to the SH section. However, this is not the case for SFH100 because of the high temperature (T_{flow}=55°C) of the distribution system (see section 2). For this reason, providing more energy at DHW level decrease the heat pump performance for SFH15 and SFH45, but it does not for SFH100. These two terms are quite similar between TN and PS for SFH15 and SFH45 for both SASHP and SGSHP, but not for SFH100.

In the right section of Table 1 relative differences between the two simulation tools are shown for the SPF of the heat pump and of the system performance as described in section 2.1. In this case TN simulations are considered to be the reference. PS tends to predict lower values of SPF_{HP} for both SASHP and SGSHP systems (see the sign of the ΔSPF_{HP}) with relative differences always below 3% for SASHP and below 13.5% for SGSHP systems. The SPF_{HP} of SASHP systems is quite similar between TN and PS. However for SGSHP systems, the discrepancies of heat pump alone performance are more significant. The main reason for this is thought to be the lower source (borehole) temperature predicted by PS when compared to TN. This is surprising since both platforms use the same ground heat exchanger model (EWS [10]). However, PS is using a newer version of the EWS model, while TN-EWS model is quite old. Nevertheless it would be surprising if the modifications of the newer version are the cause of such differences.

The differences in system performance ΔSPF_{SHP+} are below 4.6% for SASHP (quite good result) and below 13.5% for SGSHP systems.

Table 1. Results of SASHP and SGSHP for different building loads as simulation software: TRNYS-17 (TN) and Polysun-6® (PS).

System	Platform	A_c [m^2]	Building SFH	$Q_{HP,DHW}$ [MWh]	$Q_{HP,SH}$ [MWh]	Q_d [MWh]	$P_{el,T}$ [MWh]	SPF_{HP} [−]	SPF_{SHP+} [−]	ΔSPF_{HP} [%]	ΔSPF_{SHP+} [%]
SASHP	TN	15	15	0.57	2.14	4.55	1.15	2.87	3.83	-	-
SASHP	PS	15	15	0.55	2.02	4.65	1.05	2.80	4.00	-2.38	4.5
SASHP	TN	15	45	0.75	5.58	8.55	2.26	3.14	3.69	-	-
SASHP	PS	15	45	0.63	5.37	8.39	2.10	3.07	3.56	-2.22	-3.68
SASHP	TN	15	100	0.53	13.00	16.12	5.93	2.43	2.69	-	-
SASHP	PS	15	100	0.00	13.77	15.84	5.82	2.50	2.65	2.76	-1.44
SGSHP	TN	15	15	0.57	2.15	4.55	0.73	4.90	5.90	-	-
SGSHP	PS	15	15	0.60	2.09	4.64	0.72	4.44	5.73	-9.36	-2.97
SGSHP	TN	15	45	0.73	5.60	8.55	1.41	5.39	5.83	-	-
SGSHP	PS	15	45	0.72	5.51	8.45	1.37	5.04	5.06	-6.44	-13.31
SGSHP	TN	15	100	0.90	12.65	16.12	3.60	4.26	4.40	-	-
SGSHP	PS	15	100	0.00	13.90	15.81	3.96	3.69	3.85	-13.29	-12.34

The ΔSPF_{SHP+} depends on the SPF_{HP} and $P_{el,T}$. Therefore the difference in one term may be compensated by the other, see for example that ΔSPF_{SHP+} for SGSHP and SHF15 is lower than the SPF_{HP} for the same case.

In order to understand the possible source of differences, main yearly heat flows are presented for SASHP in Fig. 1(a) and for SGSHP in Fig. 1(b) for all building load demands. In Fig. 1 it can be seen that the solar yield is very similar between both simulation tools with a larger difference for the SFH100 building. SH demands are almost the same because a heat sink has been used in PS in order to meet reference system heating loads (see section 2). Storage losses are also in the same range of magnitude between both platforms. Piping losses in these simulations are lower in PS. Losses due to frosting and auxiliary heating of the heat pump are accounted in PS. Nevertheless, cycling and thermal losses are neglected in PS. This may be the reason the evaporator heat reported for TN is much higher than for PS. Consequently, the operating hours of the heat pump and of circulation pumps in the heat pump loop are also higher in TN. These heat pump losses are, however, more important in a series system where the source of the heat pump is the collector field (more collector area will be needed if heat pump losses would be accounted for in PS). In the case of SGSHP the higher evaporation needs are obtained from the ground so this will imply higher extraction from the ground. Heat pump losses are lower for GSHP systems than for ASHP as it can be seen in comparing this term from Fig. 1(a) and Fig. 1(b).

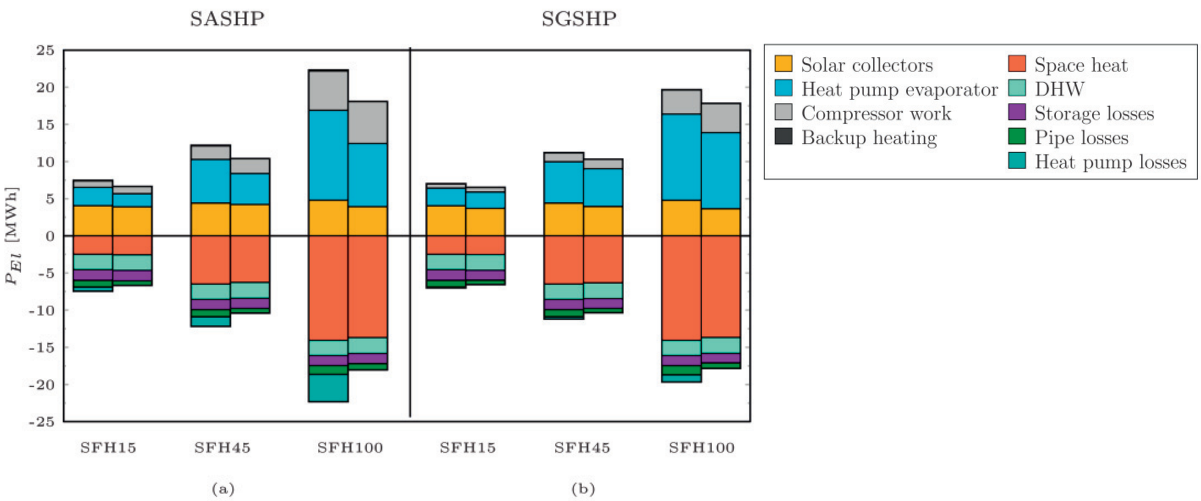

Fig. 2. Yearly energy balances comparison between TN (left bar) and PS (right bar) simulations for (a) SASGP and (b) SGSGP systems.

The electricity consumption distribution terms for SASHP SFH15 are shown in Fig.3 (left) for TN. Most of the energy consumed is from the heat pump compressor. In this case the backup heating is included as a heating rod in the heat pump unit to provide extra energy for DHW when needed. The ventilator is also part of the heat pump unit. Circulating pumps are in second position of importance. The controller units consumption are independent of the heating demand, therefore it is a relevant factor for SFH15 but not very important for SFH100. The controller unit and circulating pumps consumption losses share on total electricity demand decrease for higher energy demand buildings.

In Fig. 3(b) it can be observed that PS predicts lower values for the electricity consumption of pumps. This is the reason, besides the high share of the circulating pumps respect to the total, of having higher discrepancies for SFH15 in SASHP. In particular, the number of operating hours of the collector pump is constantly much lower in PS than in TN. The authors found that the outlet temperature of the collector in PS shows short term oscillations that lead to on/off cycling of the solar pump. The hourly averaged values for outlet temperature and power are similar to TN, but not the running time of the solar pump. In SASHP, as commented before, the number of operating hours of the heat pump is higher because of the neglected losses of the heat pump, this also affects the circulation pumps of the heat pump loop. In SGSHP the circulation pumps have a much closer value in terms of operating hours between both simulation tools because heat pump losses are less important. Nevertheless, the pumps consumption depends on nominal flow conditions and these are different for both simulation tools because of different manufacturers heat pump models used. Using mass flow rates that deviate substantially from the nominal conditions produced unrealistic results in PS and it is therefore not recommended.

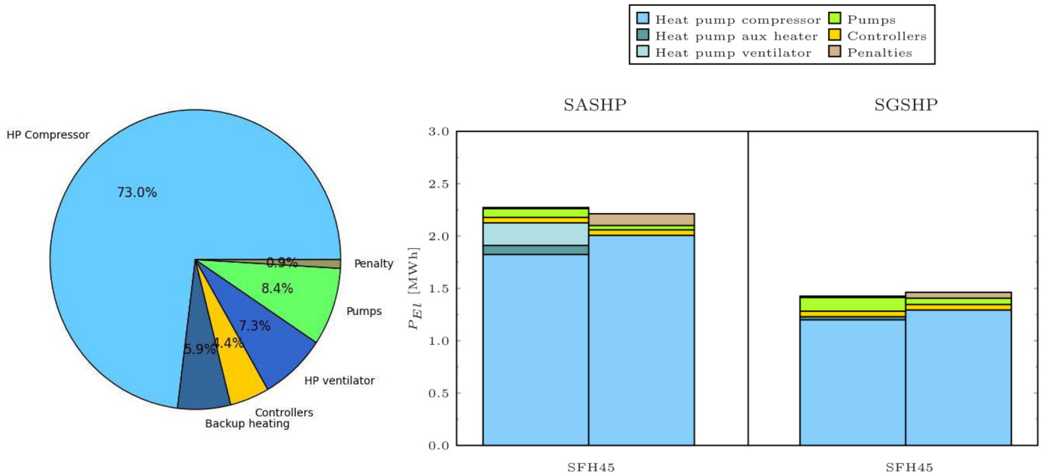

Fig. 3. Yearly electricity consumption distribution using TN for SASHP-SFH15 (left). Comparison between TN and PS for SFH45 and both SASHP and SGSHP systems (right).

3.2. Comparison of parallel systems against reference "heat pump alone" systems

The comparisons between coupled solar and heat pump systems against a reference system using only a heat pump, labeled as "heat pump alone" systems has been performed using PS.

Simulation results for the solar thermal and heat pump combinations SGSHP and SASHP, as well as for the "heat pump alone" systems GSHP and ASHP are presented in Table 2. In the right section of Table 2 the change of performance indicators (see section 2.1) with respect to the reference "heat pump only" system are shown.

Table 2. Results of (S)ASHP and (S)GSHP for different building loads.

System	Building SFH	$Q_{HP,DHW}$	$Q_{HP,SH}$	Q_d [MWh]	$P_{el,T}$ [MWh]	SPF_{HP} [−]	SPF_{SHP+} [−]	ΔSPF_{HP} [%]	ΔSPF_{SHP+} [%]	$f_{save,el}$ [%]	$P_{save,el}$ [MWh]
ASHP	15	2.57	2.66	4.64	1.78	3.23	2.61	-	-	-	-
SASHP	15	0.55	2.02	4.65	1.05	2.80	4.00	-13.34	53.13	40.86	0.73
ASHP	45	2.61	6.19	8.14	2.79	3.37	2.92	-	-	-	-
SASHP	45	0.63	5.37	8.39	2.10	3.07	3.56	-8.81	21.89	24.67	0.69
ASHP	100	2.61	14.21	15.95	6.93	2.58	2.30	-	-	-	-
SASHP	100	0.00	13.77	15.84	5.82	2.50	2.65	-3.04	15.18	16.05	1.11
GSHP	15	2.59	2.68	4.68	1.33	4.35	3.52	-	-	-	-
SGSHP	15	0.60	2.10	4.65	0.72	4.44	5.73	0.21	62.90	45.88	0.61
GSHP	45	2.60	6.55	8.55	2.00	4.91	4.27	-	-	-	-
SGSHP	45	0.72	5.51	8.45	1.37	5.04	5.06	2.00	18.41	31.43	0.63
GSHP	100	2.60	14.06	15.86	4.60	3.80	3.45	-	-	-	-
SGSHP	100	0.00	13.93	15.82	3.96	3.69	3.85	-2.87	11.70	13.84	0.64

For all simulations, the improvement in terms of fractional electricity savings $f_{save,el}$ is significant when the solar thermal system is added and it tends to be higher for low energy buildings and for SGSHP compared to SASHP systems.

For the ASHP (see upper part of Table 2), the heat pump performance (SPF_{HP}) decreases when the solar thermal system is added being a more important effect for low energy demand buildings. Despite of this, the SPF_{SHP+} increases because the solar thermal system has a much higher ratio of heat delivered to electricity consumed compared to the heat pump. As observed in [2], when the solar thermal system is added to an air source "heat pump alone system", two opposite effects in terms of performance can be observed in the behavior of the heat pump. On one hand the heat pump performance (SPF_{HP}) decreases because the solar thermal system covers part or all of the loads at times when the ambient temperature is moderately high, i.e. spring and summer periods, where also the performance of the air source heat pump is best. On the other hand, the performance of the heat pump alone (SPF_{HP}) slightly increases when the solar system is added because the solar thermal collectors cover some of the DHW loads at high temperature and therefore the heat pump works less time at high sink temperatures (see differences of $Q_{HP,DHW}$ between combined and "heat pump alone" systems). The share of DHW on the total heat demand also plays a role here. On a yearly basis the decrease of the heat pump alone performance for not working at the best periods is usually dominant and therefore the combined system SASHP, has a lower SPF_{HP} compared to the ASHP.

For the GSHP system (see bottom part of Table 2), the performance of the heat pump alone (SPF_{HP}) increases when the solar system is added for SFH15 and SFH45 because, as commented above, the heat pump works less time at high sink temperatures. However, for SFH100 this effect is not observed because of the high flow temperature of the heating distribution system and the lower share of DHW on the total heat demand.

The potential benefit in terms of $f_{save,el}$ is higher for SGSHP compared to SASHP systems. However the absolute electricity savings depend on the combination of the fractional savings $f_{save,el}$ and total electricity consumption of the reference "heat pump only" system. Since the latter is quite higher for ASHP than for GSHP, the absolute electricity savings of SASHP is higher than for SGSHP systems as can be observed in Table 2. In a study conducted in several cities around Europe [2] it has been shown that SASHP in general achieve higher $P_{save,el}$ compared to SGSHP systems.

3.3. Analysis of a SISHP system

In this section, an ice storage system based on immersed flat plate heat exchangers that can be de-iced is presented and simulated. The system concept has been explained by Philippen et al. [11] and the ice storage model description and validation has been provided by Carbonell et al. [9]. A short explanation of the ice system concept is provided hereafter.

When the heat pump extracts heat from the ice storage the growing ice layers on the heat exchanger decrease the overall heat transfer coefficient from the ice forming layer to the brine in the heat exchanger. As a result lower brine temperatures and heat pump performance are obtained. A strategy to prevent the effect of a decreasing overall heat transfer coefficient is to remove the ice layers periodically. The heat exchanger is de-iced before reaching too low brine temperatures by melting a small amount of ice that is in contact with the heat exchanger when the heat pump is switched off. When the melted ice thickness is large enough, the ice layers separate from the heat exchangers and due to buoyancy forces they are accumulated at the water surface of the ice storage.

Since ice storages can be considered as an alternative to ground source heat pump systems, the SGSHP system is used here a as reference. Systems based on large ice storages as the ones proposed here would not make sense for low energy buildings due to cost and space reasons, therefore only SFH45 and SFH100 are simulated in this case.

As an illustrative example of the behavior of the ice storage tank, a monthly energy balance plot (left axis) has been presented in Fig. 4 for SISHP-I20A20 (see Table 3 for nomenclature) case and building SFH45. The terms presented in the legend from top to bottom are: the heat input from the collector field (Q_{heat}), the gains (positive y-axis) and losses (negative y-axis) due to ice storage and ground exchange ($Q_{gain,loss}$), the heat released gains (positive y-axis) and accumulated (negative y-axis) in the ice storage in form of sensible heat ($Q_{release,acum}$), the energy of ice formation ($Q_{ice,form}$), the energy extracted from the heat pump (Q_{cool}), the energy used to melt the ice in the heat exchangers ($Q_{melt,hx}$), the energy used to melt de floating ice at the surface of the ice storage ($Q_{melt,floating}$).

Ice is formed from December to February when the solar energy is not able to balance the heat extraction of the heat pump and the ice storage is still not charged of sensible energy. Notice for example that in October and November, no ice is formed because the ice storage is full of sensible energy from the summer and therefore the heat released term of the ice storage is very high (see $Q_{release,acum}$ term on the positive y-axis of Fig. 4).

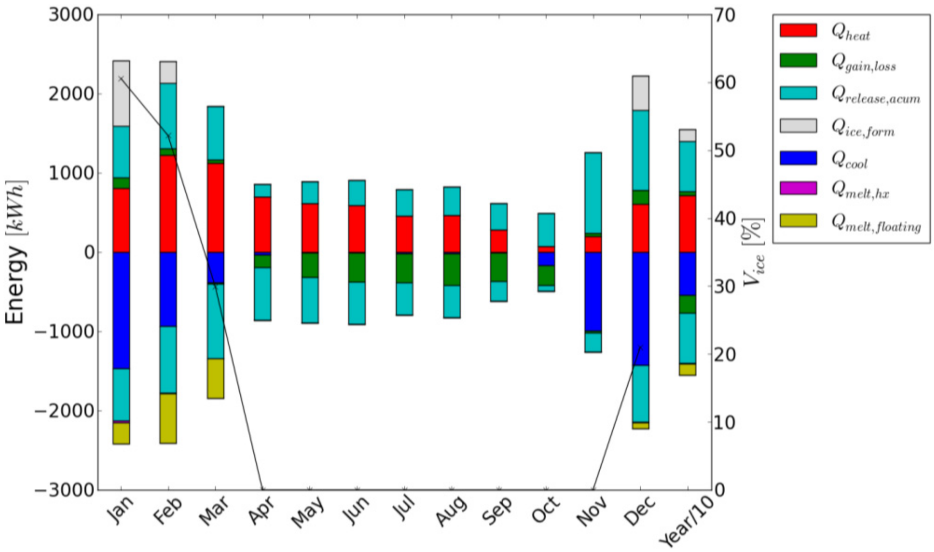

Fig. 4. Monthly and yearly energy balances of and ice storage for the SISHP-I20A25 and building SFH45 using TN.

The solar collector input (Q_{heat}) increase from November to February, where the maximum solar energy is used by the ice storage, and afterwards the solar energy decreases until October where the minimum solar energy input is found. The time of maximum usage of solar energy correspond the month of maximum ice melted. From January to March the solar input correlates well with the floating ice melting. In summer the ice storage is almost charged of sensible heat (average temperature in August is around 65°C), and during these months most of the solar energy is used for balancing the losses to the ground. From November to March the storage gains energy from the ground, and from April to October the storage losses energy to the ground.

Table 3. Results of several SISHP systems compared against SGSHP reference system for different building loads.

System	Building	V_{ice}	A_c	A_{unc}	Q_d	$P_{el,T}$	SPF_{HP}	SPF_{SHP+}	ΔSPF_{HP}	ΔSPF_{SHP+}
	SFH	[m^2]	[m^2]	[m^2]	MWh]	[MWh]	[−]	[−]	[%]	[%]
SGSHP	45	-	15	0	8.55	1.41	5.39	5.83	-	-
SISHP-I20A20	45	20	20	5	8.52	1.41	5.42	5.53	0.64	-8.31
SISHP-I20A30	45	20	30	5	8.52	1.26	5.61	5.90	4.12	1.13
SISHP-I25A15	45	25	15	5	8.52	1.52	5.35	5.01	-0.82	-14.17
SGSHP	100	-	15	0	16.12	3.60	4.26	4.40	-	-
SISHP-I30A45	100	30	45	5	16.06	2.98	4.57	5.10	7.40	15.93
SISHP-I40A30	100	40	30	5	16.05	3.19	4.53	4.78	6.47	8.76

The ratio between the maximum volume of ice and the volume of ice storage is shown as solid line in the right axis of Fig. 4 for each month. It can be observed that there is no ice in the storage from April to November. The maximum value of 60% is found in January. As a design criterion it is not allowed to have more that 70% of ice because, in this case, the ice layers may not be detached from the heat exchanger surface anymore and the evaporator temperature of the heat pump may be too low to run.

Results for different ice storage based systems and a SGSHP system used as reference here are presented in Table 3. Results presented with SISHP systems include 5 m^2 of uncovered collectors, mostly for de-icing reasons in cases where sun is not shining. For SFH45 (see upper part of Table 3) the system performance of a SGSHP system is very high, with a SPF_{SHP+} of 5.8. Using an ice storage of 25 m^3 with the same collector area as the SGSHP a lower SPF_{SHP+} of 5.01 compared to a SASGP is obtained. Increasing the collector area, from 15 to 30 m^2 (see SISHP-I20A30) the system performance increases to 5.9 and the volume of the storage can be reduced to 20 m^3 without ever having the storage at 70% of the ice capacity. With 20 m^3 of ice storage volume the collector area can be reduced until 20 m^2 with an SPF_{SHP+} of 5.53.

Results for SFH100 are shown in the bottom part of Table 3. Both SISHP simulations perform with higher efficiency compared to a SGSHP system under these specific conditions because the collector area is much higher for the SISHP systems. In this case reducing the storage volume from 40 to 30 m^3 and increasing the collector area from 30 to 45 m^2 also improves system efficiency reaching a SPF_{SHP+} of 5.1.

4. Conclusions

Three different combined solar thermal and heat pumps systems, using air source, ground source and ice source, (SASHP, SGSHP and SISHP systems respective) have been numerically investigated. Three buildings representing low, medium and high energy demand have been simulated using the reference conditions defined in T44/A38.

A detailed comparison between TRNSYS-17 and Polysun-6® has been performed for SASHP and SGSHP systems. In general terms, differences in heat pump and system seasonal performance factors up to 4% can be expected for SASHP systems and higher differences, up to 14%, are found in SGSHP systems.

The potential benefit has been studied by comparing the combined systems with their respective "heat pump only" reference solutions for air source and ground source based systems (ASHP and GSHP respectively). The system performance improvements of the combined systems are significant in all simulations. The fractional electricity savings are in general higher for SGSHP compared to SASHP systems. Nevertheless, the absolute electricity savings of SASHP are found to be usually higher compared to the SGSHP systems.

Ice source based systems are capable to reach system performances of the order of SGSHP systems. Increasing collector area between two SISHP simulations leads to a better system performance and it allows reducing the ice storage volume significantly.

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