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Domestic Heat Pump System With Solar Thermal Collectors as Heat Source and Annual Ice Storage

Andreas Gschwend
andreas.gschwend@ntb.ch

Stefan S. Bertsch

DOMESTIC HEAT PUMP system with solar thermal collectors as heat source and annual ice storage

Andreas GSCHWEND[1]*, Stefan S. BERTSCH[2]

[1] Interstate University of Applied Sciences of Technology NTB
Institute for Energy Systems, Werdenbergstrasse 4, 9471 Buchs, SWITZERLAND
Phone: (+41 (0)81 755 34 19), Fax: (+41 (0)81 755 54 34), e-mail: andreas.gschwend@ntb.ch

[2] Interstate University of Applied Sciences of Technology NTB
Institute for Energy Systems, Werdenbergstrasse 4, 9471 Buchs, SWITZERLAND
Phone: (+41 (0)81 755 34 69), Fax: (+41 (0)81 755 54 34), e-mail: stefan.bertsch@ntb.ch

ABSTRACT

Aim of this paper is a theoretical and experimental investigation of a domestic heating system with heat pump using solar thermal collectors as the only heat source. In the system described, the heat pump uses an ice-water tank as annual storage system taking advantage of the phase change at 0°C. Energy is supplied to the storage system using low temperature solar thermal collectors. Low temperatures inside the solar collector lead to an increased annual yield. The thermal collectors can also be used to directly heat domestic hot water or to supply heat to the buildings heating system. Benefit of this system should be a significantly increased performance compared to a standard air to water heat pump while at the same time reducing its noise level. Additionally no drilling for geothermal probes is needed and no permission to use ground water is required. A theoretical model was built for this system and successfully validated using the field measurements.

The field unit still had to fight several problems (evaporator too small, problems with control system) which lead to a very low overall seasonal efficiency of 1.43. But the measurement results could still be used to validate the heat pump and building model. Starting from the lessons learnt in the field, the system was dimensioned correctly and simulated accordingly. The seasonal efficiency achieved in the simulation increased to 2.4 which is comparable to air source heat pumps. It is expected that further modifications of the system could result in seasonal performance factors of 3.0-4.0. While this would be a fairly good performance and the noise pollution of the heat pump would be lower compared to air source heat pumps, the investment costs needed would grow considerably.

1. INTRODUCTION

Heat pumps are used more and more in domestic heating systems. There are three common types of heat pumps, they differ in the heat source they use.

-**Ground- or lake water heat pumps** use underground water or water from a lake as a heat source. The sources have a constant, relatively high temperature during the whole year; therefore, the seasonal performance factor is high and reaches from 3.8 to 5.0 (*Eschmann, 2011*). The drawback of the system is that it cannot be applied everywhere and permission of the local authorities is always needed.

-**Heat pumps with geothermal probes** take the medium-deep ground (up until 300 m) as a heat source. Since the temperatures in the ground increase by about 3°C per 100 meters in depth, deeper probes result in a heat source at a higher temperature. Good seasonal performance factors from 3.5 to 4.5 can be achieved (*Eschmann, 2011, Erb et al. 2004*). Drawbacks of the system are relatively high investment costs for drilling the probe and the fact that these drilling operations are not possible at every place (cities).

-**Air source heat pumps** use the ambient air as heat source. Advantages of the systems are the low investment costs as well as the fact that they could be built everywhere. Due to the fact, that the system performs worst when most heat is needed the annual performance factor is relatively low; typical values lie in between 2.1 and 3.2 (*Eschmann, 2011, Erb et al. 2004*). Another disadvantage is the unwanted noise produced by the fan; this is an issue in Switzerland that has led to more stringent regulations especially in urban areas.

In this paper a new system approach is under investigation. The new system will be applicable for most climates and should result in a better performance than air source heat pumps. As the only energy source of the system, a thermal solar collector is used. During hot weather conditions, e.g. in summer and parts of spring/fall, the collector supplies heat directly to the domestic hot water tank or the heating system. If the desired water temperature for hot water heating or room heating cannot be reached, or if there is surplus heat in summer, the heat from the solar collector is stored in a large tank. This water tank is fabricated as an ice-water tank that uses the latent energy of water as annual storage. This water tank is used by the heat pump as its only energy source. The ice-water tank acts as a PCM (phase changing material) thermal storage. Due to the fact that the temperature of the ice-water storage is lower than the ground temperature, no insulating layer for the large tank is needed. An electrical emergency heating system can be used as a backup solution for the heat pump system during long bad weather periods.

A big advantage of the new system is that the heat source temperature is fairly high during the whole year. This results in higher evaporating temperatures and therefore better efficiency. Additionally, the heat source (solar collector) can be easily installed on most buildings and usually does not need special permissions. Finally, solar collectors do not pollute the surroundings with noise.

Several heating systems similar to the one described here have been proposed lately. *Frank et. al (2010)* and *Sparber et al. (2010)* show a good overview of possible combinations between heat pumps and solar collectors. The second paper also publishes results on known measurements on combinations of solar-thermal and heat pump systems. *Haller and Frank (2011)* are investigating the performance difference between direct and indirect use of the heat from the solar collectors combined with the heat pump. A slightly different system, where the heat from the solar collectors is only used as second source has been published by *Loose et al. 2011*. No comparison of the system using a ground probe as heat storage and the system with air as second heat source has been published by them, since only one system has been measured yet. Finally, a quite similar system, to the one presented here is published by *Lerch et. al (2011)*. In their system besides the solar collector also waste water is used as energy source. Another major difference is the use of an intermediate circuit for the heat source, unlike the current system that uses direct evaporation. There are also studies on large scale systems combining heat pump and solar collector like *Ozgener and Hepbasli (2005)*.

In the underlying study measurements are carried out on a prototype system over a whole heating period. The tested system was designed and built by the small Swiss heat pump manufacturer Kneco. The authors of this study designed and installed the measurement setup. Based on the layout of the prototype system a numerical model was derived and validated using the measurement results.

On the basis of the simulations the major mistakes of the current prototype will be identified and optimizations will be proposed, which can be implemented in the system before the next heating period.

2. Measurements & Results

The building equipped with the heating system is an old, traditional farm house. The building originally consists of two parts: the residential house and the barn. The residential part has been renovated during the last few years and the barn has been converted into a second apartment. Figure 1 shows a schematic of the heating system.

Detailed description of the heating system

Two heat exchangers are installed in the ice-water tank. One heat exchanger contains a brine-water mixture (green lines) that also flows through the solar collector. The other heat exchanger is the evaporator of the heat pump and contains the refrigerant. Basic idea of this arrangement was to increase the evaporating temperature due to no intermediate circuit between storage tank and heat pump. As we can see in Figure 1, the solar thermal circuit (green lines) is also connected to the domestic hot water storage (left tank) and the heating system (right tank). As soon as the temperature of the solar collectors is above the domestic hot water or heating system direct heating is used. If the brine temperature of the solar collector is between 0°C and the minimal heating temperature, the energy from the solar collectors is stored in the ice-water tank. The heat pump circuit itself (brown lines) is a single stage unit with variable speed drive for capacity control. It is used both for residential heating and domestic hot water.

The most important heating system and building parameters are:

- Solar collector, glazed flat panels, 16 m^2, Ice-water storage: 6m^3
- Domestic hot water tank and heating system tank: 500 l each
- Compressor: Bock HGX34e/315, reciprocating compressor with variable speed drive
- Evaporator: stainless steel tube with 18 mm outer diameter and 48 m length
- Heating capacity of the heat pump: 5 – 13 kW
- The heating load in winter calculated according to the Swiss standard SIA 384.201 is 13 kW
- Heated area of the building; 402 m^2, heavy building construction with floor heating

DEFROSTING MODE OF THE SOLAR COLLECTORS

Figure 1 shows the overall schematic of the heating system. If the sun begins to shine, the water in the collector heats up. A pump circulates the water in the collectors and valve V_1 closes the loop of the cycle in order not to transport cold water into the heating system. If a small area of the collectors is snow free, this function can be used to melt the snow on all collectors. Once all collectors are free of snow the temperature will rise fast and the valve V_1 opens in order to lead the energy into the building system.

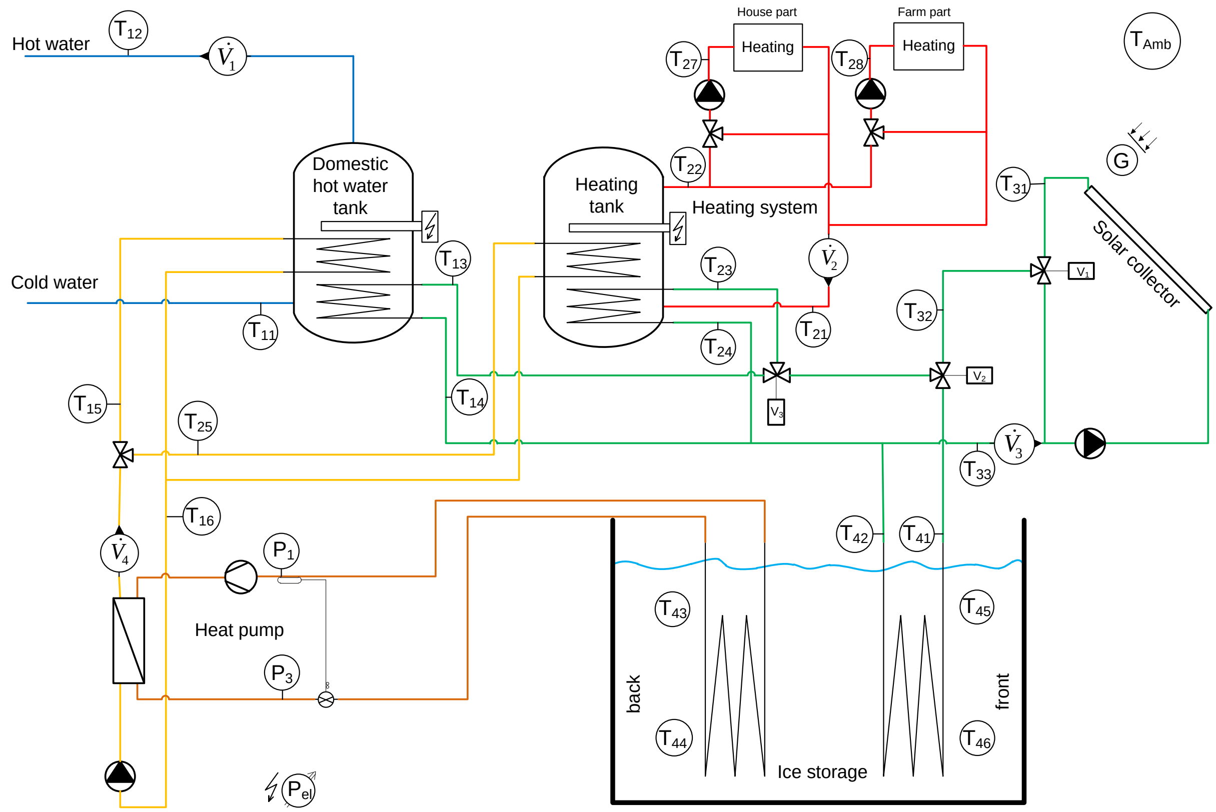

Figure 1: Heating system in the field measurement

2.1 Instrumentation and measurements

The following parameters are measured in the prototype system: temperature, volume flow rate, pressure, electrical power, solar irradiation and position of the valves. The locations of all measured parameters are shown in Figure 1. The temperature sensors are thermocouples of the type K with an uncertainty of ± 0.5°C. To measure the volume flow rate inductive flow sensors (ifm SM6004, ifm SM7004) with an uncertainty of 2% of the measured value were used. Pressure is measured using a piezoresistive transmitter (PAA-21Y) from Keller with an uncertainty of 0.5% (full scale). The electrical power is measured by three true rms powermeters (ITL 100 from Infratrek) which show an uncertainty of 0.2%. An ESTI-Sensor is used to measure the solar radiation. The position from each valve can be measured using a low voltage signal. As data acquisition unit a CompactRIO system from NI was used. The data acquisition software saves the data at regular intervals of 15s on a USB stick. Approximately once a month the data is collected manually and evaluated. In this way errors in the data acquisition can be found in time. The measurements started at the beginning of December 2011. Due to some corrections to the overall system (fine adjustment of the heat pump installation) only measurements from January until Mid-April could be used.

2.2 Measurement results

After measuring and evaluating one heating period (December to April) two big weaknesses of the current prototype system can be identified: Firstly, the ice-water tank with a volume of 6 m^3 is too small. Especially during long cold weather periods the heat storage is insufficient. This fact was amplified by an especially cold month of February during which no solar radiation was available to heat up the tank. Therefore there are long stretches of time where the heat pump is stopped and the electrical emergency heating is turned on. For the given measurement period this fact has reduced the overall system performance significantly.

Secondly, the size of the evaporator is too small. This leads to a large temperature difference between evaporation temperature and ice-water tank and reduces the COP. Besides the insufficient heat transfer of the evaporator, also the thickness of the accumulated ice on the heat exchanger increases rapidly and reduces the thermal conductivity. Evaporating temperatures of down to -20°C were leading on one hand to low efficiency and insufficient heating capacity, which resulted in even longer run times of the backup heating system. These two facts will be addressed in the near future to achieve a better overall efficiency starting from the next heating season.

In Table 1 the measurement results from January 5th, 2012 until April 12th, 2012 are summarized. The current prototype system has a seasonal performance factor of only 1.43 (including electrical reheat) due to the aforementioned problems. In order to calculate the overall energy consumption the following equation was used:

$$COP = \frac{\dot{V}*(T_1*\rho(T_1)*cp(T_1)-T_2*\rho(T_2)*cp(T_2))}{\dot{Q}_{el}} \quad (1)$$

Table 1: Measurement results from the heating period 2012

	Period 5th Jan 2012 until 12th April 2012
Electrical Energy (heat pump and electrical reheat)	9849 kWh
Energy used for domestic hot water	5437 kWh
Energy used for heating	8674 kWh
Energy from the thermal collector	2449 kWh
Resulting seasonal performance (Jan-April)	1.43

Even so the system is not performing as intended the measurements still can be used to establish and validate a simulation model and suggest changes to the system to improve its performance.

3. Simulation Model

For a better understanding of the solar heat pump system and for the purpose of further optimization, a mathematical model of the system was implemented in Matlab. The simulation determines the seasonal performance factor based on different input parameters specified below. In the beginning of the simulation all subroutines are initialized. Those subroutines represent the different components of the solar heat pump system. The subroutines are called by the main routine subsequently one after the other for each time step until the simulation time has been reached. The system simulation builds on a forward Euler method with fixed step size. At the end of the (annual) simulation all stored values for temperatures, power and pressure are used to analyze the data.

3.1 Subroutines

-**Domestic hot water consumption**: Several different standard consumption profiles for the domestic hot water consumption are generated by DHWcalc (*Jordan & Vajen, 2003*). Different profiles for workdays and weekend are used.
-**Solar radiation and ambient temperature**: The weather data as input for the simulation code are provided from Meteonorm. Currently only the climate data of the Rhein valley, Switzerland is implemented, but the code can be extended in the future to investigate the system under different climatic conditions. The yield on the solar thermal collector is calculated from the climate data regarding the different orientation of the panels.

-**Controller of the solar cycle**: The controller receives the temperature information from the collector, status of the ice-water tank, the temperature of the heating tank and the domestic hot water tank as well as the solar radiation. Based on these values, the energy flow and operation of the system is controlled.
-**Solar thermal collector**: The solar radiation will heat up the water inside the collector. The water in turn transports the energy into one of the three storage tanks of the system. Heat gain and loses to the ambient are calculated as follows (*Kartchenko, 2004*)

$$\dot{Q}_{water} = A_{collector} G_{radiation} \eta \quad (2)$$

$$\eta = \eta_0 - a_1 T_{m*} - a_2 G_{radiation} {T_{m*}}^2 \quad (3)$$

$$T_{m*} = \frac{T_{collector} - T_{ambient}}{G_{radiation}} \quad (4)$$

with the values of $\eta_0 = 0.8$; $a_1 = 3.60\ \frac{W}{m^2K}$; $a_2 = 0.0110\ \frac{W}{m^2K}$ as found in the database of the solar collector test-center SPF (www.spf.ch).

-**Connecting water pipes**: For the connection from the collector to the water tank and from the heat pump to the water tank a pipe model was used. Losses to the ambient were calculated using a simple thermal resistance model with given ambient temperatures. Pumping power is considered.
-**Controller of the heat pump cycle**: The heating capacity of the variable speed heat pump is controlled based on the temperatures inside the water tanks. The controller also turns the heat pump off during electrical cutoff-times.
-**Heat pump**: With a simplified heat pump model lookup tables for efficiency and heating capacity were generated using the software EES (*Klein, 2011*). This model assumes a standard compressor model using isentropic and volumetric efficiencies and ε-NTU models for the heat exchangers. Input parameters are the condensing and evaporating temperatures as well as the water return temperature of the heat sink, which influences the subcooling and therefore the performance. The basic model has been developed in another concerning hot water supply using heat pumps (*Vetsch et al., 2011*). The refrigerant used for the cycle is R134a. Based on subcooling, condensing and evaporating temperatures, the COP (Coefficient of Performance) and the heating capacity are calculated and stored in lookup tables which then are used in Matlab.
-**Ice-water tank**: The ice-water storage tank is cooled down by the evaporator of the heat pump and heated up by the water flow from the solar collectors and to a small amount by the energy flow from the ground surrounding the tank. If the water temperature declines below 0°C, ice will start to build up (6). Energy gains from the solar collector will reverse this process and therefore melt the existing ice or heat up the water contents of the tank. It is assumed that the ice grows and melts homogenously around the tubes of the evaporator. Conventional heat transfer correlations for forced convective boiling (inside the evaporator tube) and natural convection (outside of the tube) are used in the model. Thermal resistances of the tube-wall and ice layer are considered

$$\Delta T_{ice\ tank} = \Delta \dot{Q}_{liquid} \frac{\Delta t}{V_{tank}\, \rho_{water}\, cp_{water}} \quad (5)$$

$$\Delta m_{ice} = \Delta \dot{Q}_{frozen} \frac{\Delta t}{H_{sf\ ice}} \quad (6)$$

-**Hot water storage tank with internal heat exchanger**: The volume of the tank and the heat exchanger are divided into n horizontal elements. For each element a volume and energy balance is calculated. For the free convection at different densities an approach from Yih (1965) was used as shown in figure 2. More details for the model can be found in *Vetsch et al., 2011*.

$$\dot{Q}_{e[i]} = \left(\dot{Q}_{cond[i]} + \dot{Q}_{conv[i]} + \dot{Q}_{he[i]} + \dot{H}_{if[i]} + \dot{H}_{ef[i]}\right) - \left(\dot{Q}_{loses[i]} + \dot{Q}_{cond[i-1]} + \dot{Q}_{conv[i-1]} + \dot{H}_{if[i-1]}\right) \quad (7)$$

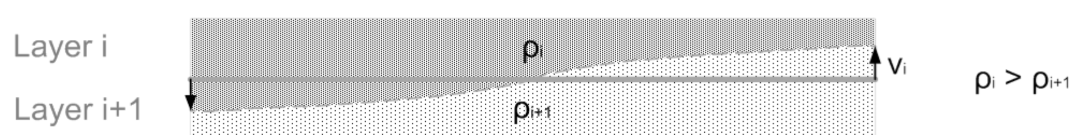

Figure 2: Free convection inside the storage tank in cross-sectional view

-**Space heating**: In this subroutine the losses of the building to the ambient and the internal gains are considered. The internal gains considered consist of solar radiation through the windows and the heat dissipated from the residents and the floor heating. The window size is assumed in relation to the living space according to the typical Swiss building style

$$A_{Windows} = 4\sqrt{A_{living\ space}}h * 20\% \tag{9}$$

with 15% of the window area facing North, 35% facing south and 25% each facing West and East.

3.2 General

For every simulation step the temperature of the ambient condition, as well as the temperatures from the last simulation step, are used to calculate heat and energy flows. The functions for density, specific heat, conductivity, and viscosity were generated using EES and then simplified with polynomials for the implementation in Matlab.

One of the most critical parameters for the stability of the simulation code is the step time. If the volume elements are small and the volumetric flow is high, the step time must be very short, in order to allow convergence. Hence, the simulation code has been optimized by adapting the minimum time step for a certain geometry before conducting the annual simulation.

The simulation is always started at midnight (New year). Therefore the start values can be easily set in the initialization process. The solar thermal collector and the piping to the roof is cold (ambient temperature). The heat pump is turned off. Therefore the interior piping and heating system is set at room temperature. The domestic hot water tank on the other hand is filled with 56-60°C hot water, as designed by the control strategy. Also, the heating tank is warm. Its temperature lies 2 K higher than the temperature for the heating system, which is also set by the overall control strategy. The conditions for the ice-water tank can be chosen according to the demand. Reasonable estimates can be usually taken from the simulations carried out previous to the current one. Since an annual simulation is carried out, the start values show only little influence on the overall results.

3.3 Verification and field measurements

Figure 3 shows simulation results and measured values of Supply and evaporating temperature during a period of 48 hours. The comparison of these values has been chosen as one parameter for the validation, since they are very sensitive to deviations between model and measurement. The start values for initializing the simulation were given from the measurement. The simulated values are matching the measured values with an average temperature difference of 1.67°C. The measured evaporating temperature is slightly higher compared to the simulated evaporating temperature. The difference in evaporating temperature is most likely caused by small deviations in the geometry of the actual heat exchanger and its model. The heat transfer correlations for natural convection do not consider the exact shape of the heat exchanger and its surroundings. As we can see, the time difference of heat pump operation between measurement and simulation is small, which means that the building model represents the actual system very well. Typical errors in efficiency and heat flow calculation are 9.6% and 8.1%, which is low given the fact, that user behavior influences the measurement results. Due to the good agreement between model and measurement, the simulation code can be used for a detailed investigation of the system and for parametric studies.

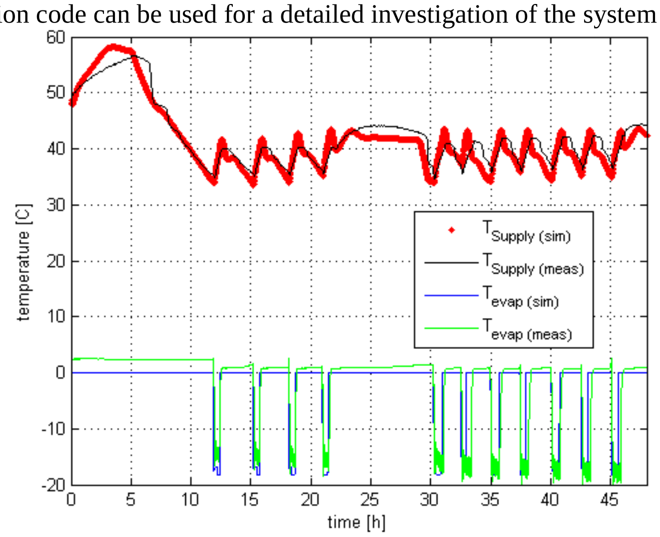

Figure 3: Comparison of measured and simulated evaporating and supply temperature during a period of 48 hours.

As we can already see from these first results, the evaporation temperature is low compared to the temperature of the ice. This large temperature difference can be explained by the fact that the evaporator in the field installation is undersized, resulting from a significant increase of needed heating capacity after completion of the heating system.

4. Simulation results

In order to show the overall efficiency of a correctly dimensioned system, the following assumptions were taken: the investigated building is similar in build and size compared to the building used for the field measurements. Its heated area is 400 m^2 with a maximum heat load of 27 W/m^2. The heating system on the other hand has been resized by the authors to improve efficiency and reliability. The evaporator has been doubled in length to 100 m of 18 mm tube and the ice water storage size was increased to 50 m^3, so that seasonal storage effect can be used better.

Figure 4 shows the energy harvested from the solar collector. The first, very narrow bar represents the solar irradiation onto the roof area for each month. The second narrow bar represents how much energy actually is transferred into the water. The following, wider bar shows the solar energy that was actually harvested by the solar collectors. The small differences between second and third bars represent the losses of the hydronic system. As we can see in the wide bar, most of the energy harvested from the collectors is actually used for the ice storage system, due to the low temperatures achieved at the collector. Especially in winter the minimum temperature for direct heating and hot water supply cannot be achieved in the collector very often. But it is still possible to use some of the solar energy for the ice-water storage. During summer, almost no heat is needed for heating and the domestic hot water requirements can be fulfilled easily. Therefore, a large part of the solar energy can be stored in the ice water tank. The annual energy harvest of the solar collector reaches 811 kWh/m^2. The typically annual energy harvest for hot water systems under the given climatic conditions of Switzerland is 400-550 kWh/m^2. The overall efficiency of the solar thermal collector therefore can be raised from typically 30-50 % to approximately 70% due to the lower temperatures in the collector.

Figure 5 shows the amount of domestic hot water heated, using the heat pump and directly by the solar collector. The slim pink bar shows the amount of domestic hot water consumed. The difference in height accounts for heat losses of the hot water tank and piping. We can see from the figure that the domestic hot water demand during summer and winter does not vary considerably. We can also see that during summer time, more than 60% of the water can be directly heated by the solar collector and only a smaller part is heated using the heat pump. In winter, when there is only little solar irradiation, the relation between heat produced by the solar panels and using the heat pump switches. Comparing December and January, we can see that even though January is the colder month, more heat can be used directly from the solar panels. This can be explained by the usually very sunny month of January, while there is more fog in November and December.

Figure 6 shows the monthly energy generation for heating and domestic hot water, produced by the heat pump only. As we can see, the domestic hot water demand is small compared to the overall heating demand. The trend of this graph shows the exact opposite behaviour as the solar thermal yield, which explains the large ice-water tank needed for the system. The pink bars in figure 6 account for the energy supply with the difference in height accounting for heat losses again.

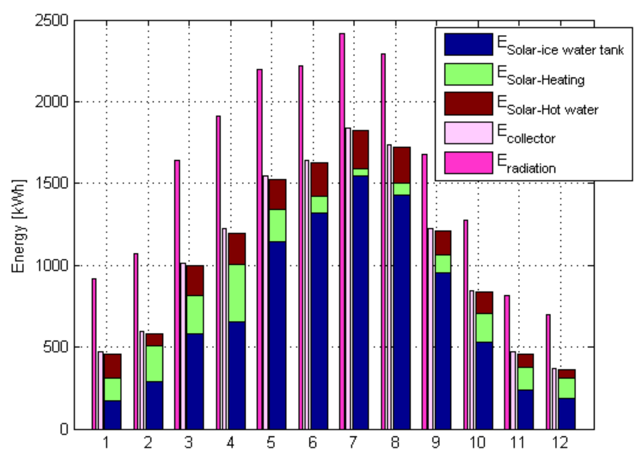

Figure 4: Energy harvested from the solar collector; the different colors show how the energy is used.

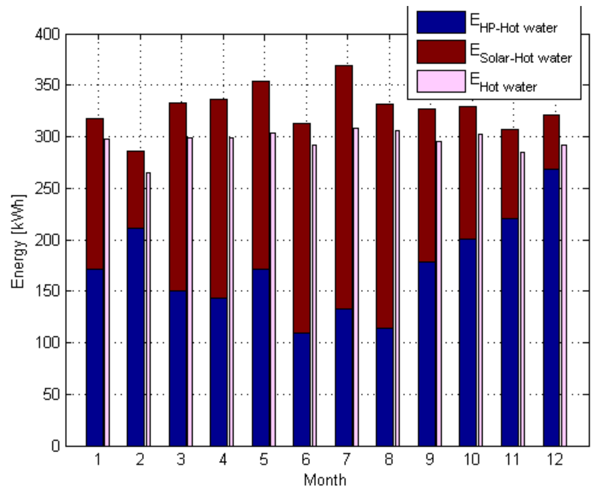

Figure 5: Domestic hot water heated using the heat pump (E_{HP}) and the solar collector (E_{Solar}); Pink bar: domestic hot water usage.

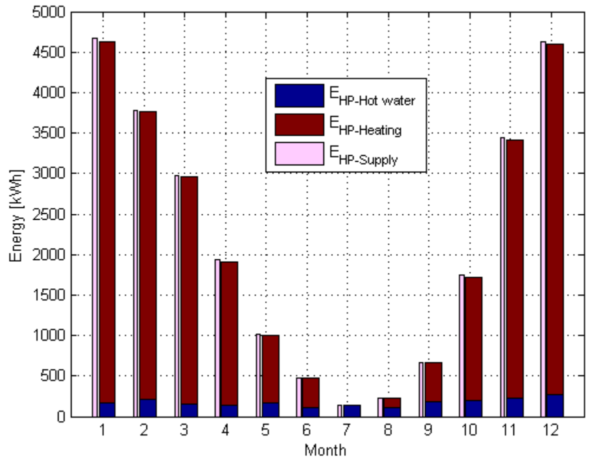

Figure 6: Proportion of energy production from the heat pump for heating and domestic hot water.

The energy balance of the ice-water storage is presented in figure 7. As expected, there are small gains and a large demand in winter, while the solar collectors yield more energy in summer. Overall the energy demand is larger than the solar gains during summer. The size of the solar panels is therefore too low for the energy demand of the building. Since for the given building there is no additional space for solar thermal collectors, an alternative heat source would be needed. Possibilities are an air-to water heat exchanger operating in summer, or preferably the recovery of waste heat from the building.

Figure 8 shows the evaporating and ambient temperature of the system as well as the thickness of the ice accumulated around the evaporator tubes. We can see that the temperature difference between evaporation and ice water is still very large. This is due to the very thick ice layer that insulates the tubes. These results lead to the conclusion, that a direct evaporating system as proposed in this building is not an optimal solution, unless the ice could be removed regularly from the evaporator. Possibilities for ice removal are either mechanical means, or using the heat from the solar collectors to partially melt the ice on the evaporator which then floats atop of the ice water storage. Systems like this would need careful design.

The ice layer in December is larger than the layer in January and the ice also does not completely melt in summer. These two facts lead also to the conclusion, that the heat gain from the collector is too small for the heat demand of the building. The overall seasonal efficiency achieved with the system presented in the simulation would be at 2.1 for domestic hot water heating only. For heating and domestic hot water combined, the efficiency is at 2.4. These values are approximately in the area of a conventional air source heat pumps, but low for the amount of investment cost for the overall system.

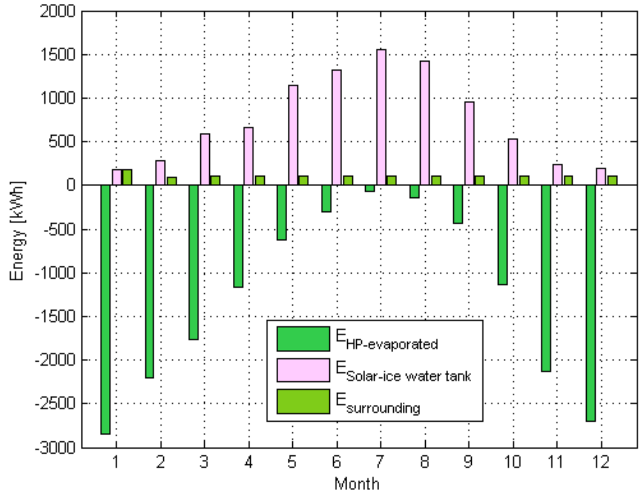

Figure 7: Energy balance over the ice water tank.

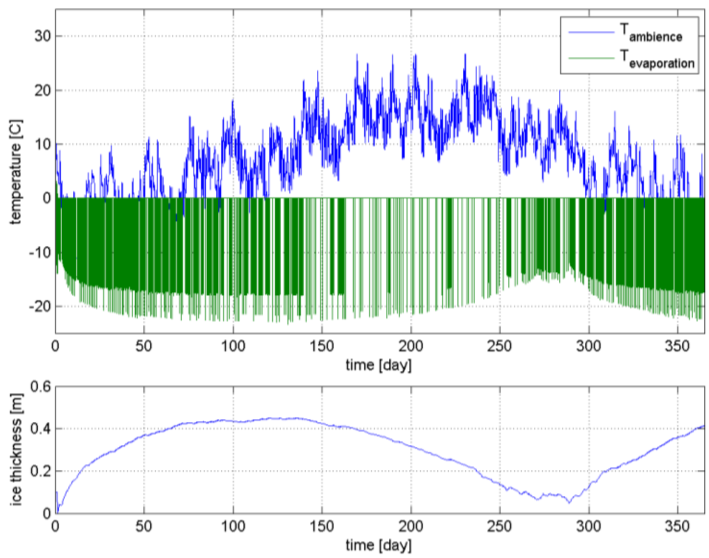

Figure 8: Top: Ambient and evaporating temperature;
Bottom: Thickness of the ice layer around the evaporator.

The efficiency of the system could be further optimized with larger solar panels and a defrosting mechanism for the evaporator. In this way seasonal efficiencies of approximately 3 should be realistic. The authors would also suggest replacing the direct expansion system of the evaporator with an intermediate circuit. Advantages would be far lower first costs for the tubing in the ice-tank and therefore larger possible surface areas of the heat exchanger. This would lead to lower ice thicknesses. Secondly, also the defrosting of the ice-water tank heat exchanger could be realized easier.

5. CONCLUSIONS

A heat pump system with solar panels as heat source and an ice-water tank as annual storage was investigated in the field. Subsequently, the system was modeled and successfully validated against the measurement results. The field unit has been found to suffer from several problems, resulting from a significant adjustment in the building heating demand after completion of the heating system. Therefore the whole system is undersized. Especially the evaporator is too small which leads to a very large ice buildup and low evaporating temperatures which in turn reduces the seasonal performance. After a very cold winter, an overall efficiency of 1.4 was measured.

In the simulation the system was resized in order to improve efficiency. The maximum heat load was set to approximately 90W per meter length of the evaporator and the ice-water storage tank was enlarged as well. Without changing anything else in the system a seasonal performance of 2.4 for domestic hot water and heating combined was found. This factor is in the same area as respective air source heat pumps. Improvements of the control strategy and with respect to defrosting of the evaporator would help to further increase the performance. Seasonal efficiencies of 3.0 to 4.0 seem plausible in the given climatic zone. Systems optimized in such a way would then provide heat to the building in a comparable way of geothermal heat pumps with respect to noise and efficiency. Further usage in the field will then mostly depend on first costs and reliability.

NOMENCLATURE

Symbols

A	area, [m^2]
$G_{radiaion}$	global radiation, [W/m^2]
$\dot{Q}$	heat transfer rate, [W]
$\dot{H}$	enthalpy transfer rate, [W]
a_1	solar coefficient 1, [W/m^2-K]
a_2	solar coefficient 2, [W/m^2-K^2]
T_{m*}	solar factor, [K-m^2/W]
ΔT	temperature difference, [K]
Δt	step time size, [s]
V	volume, [m^3]
cp	specific heat, [J/kg-K]
Δm	mass difference, [kg]
$H_{sf\ ice}$	specific heat of fusion, [J/kg]
v	velocity, [m/s]
g	gravity, [m/s^2]
h	room height, [m]

Greek

η	efficiency, [-]
η_0	conversion factor, [-]
ρ	density, [kg/m^3]

Subscripts

e	volume element of the water tank
water	water
collector	collector
ambient	ambient
ice tank	ice-water tank
liquid	liquid
frozen	frozen
cond	conduction
conv	convection
he	heat exchanger
if	internal flow
ef	external flow
losses	losses to the ambient

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