# SIMULATIONS ON SOLAR-ASSISTED HEAT PUMP HEATING SYSTEMS

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## ABSTRACT

Seit einigen Jahren finden sich verstärkt Heizungssysteme auf dem europäischen Markt, die sich durch die Kombination von Solarthermie und Wärmepumpe auszeichnen. Um wichtige Erkenntnisse bezüglich dieser Systeme zu gewinnen, wurden bereits öfters Simulationsstudien durchgeführt und veröffentlicht. Der vorliegende Beitrag behandelt hauptsächlich Fragestellungen zur Simulationsmethodik. Es wird insbesondere untersucht, welchen Einfluss das Ignorieren bzw. die Berücksichtigung von Direktverdampfung oder von Hilfsenergie hat. Dazu werden ein nicht-solares Referenz- und drei solar-unterstützte Wärmepumpensysteme simuliert, die ein Einfamilienhaus mit Raumwärme und Warmwasser versorgen. Die Ergebnisse werden stets vergleichend dargestellt und interpretiert.

During the last years, more and more heating systems can be found on the European market that are characterised by the combination of heat pumps and solar-thermal collectors. To analyse and to assess these systems, simulation studies have frequently been carried out. This paper focusses mainly on questions regarding the simulation methodology. In particular, it is examined how much influence can be expected from ignoring or respecting the influence of direct evaporation or auxiliary energy, respectively. This analysis is done by simulating a non-solar reference system and three solar-assisted heat pump systems, all meant to provide domestic hot water and space heating for a single-family house. The results are presented and interpreted comparatively.

## INTRODUCTION

### Background and Objective

In modern domestic heat pump systems, solar collectors are frequently incorporated for contributing to *domestic hot water* (DHW) generation or space heating. Beyond this well-established approach, today's market also features alternative systems in which a genuine interaction between solar collector and heat pump is intended. Because renewable energies are integrated more intensely in this way, these system are meant to achieve higher economical and ecological system efficiencies.

First approaches regarding such systems where carried out during the late 70s of the last century. At that time, the high complexity regarding hydraulics and control strategies inhibited any breakthrough, and the idea of solar-thermally assisted heat pumps vanished soon as did heat pump heating systems in general. During the last years, the idea was revitalised and heating systems combining heat pumps with solar-thermal technology entered the market in growing numbers. Approaches to classify manifold systems were conducted for example by [Henning, Miara 2009] and [Frank 2010].

Because combined solar heat pump systems have just recently passed the verge of being introduced to the market, no widely-accepted indicator has been developed yet to quantify the benefits of this combination. While there are standards to assess single heat pumps or collectors, no standards apply to systems whose performances depend strongly on the interaction and the implemented control strategy.

Simulations are frequently applied to analyse solar-assisted heat pumps or to carry out parameter variations, e.g. by [Citherlet et al. 2008], [Trinkl et al. 2009] or within the on-going Task 44 of the IEA Solar Heating and Cooling Programme, cf. [Hadorn 2010].

This paper focusses on methodology-oriented topics. The first objective refers to direct evaporation of heat pumps. Generally, direct-evaporating air-source heat pumps cannot be incorporated in systems using solar collectors as additional source. A brine that allows to transport energy from either the collector or the air source to the evaporator is used instead. However, this extra loop – or the additional heat exchanger, respectively – worsen the temperature level of the source and thus the efficiency of the heat pump. Therefore, an objective of this paper is to find out whether this effect is crucial to the overall efficiency or whether it can safely be ignored in simulations and planning.

The second objective is to set the benefits of solar-assisted heat pumps in relation to the auxiliary energy. As auxiliary energy we define the electricity consumed by components of the heating system apart from the heat pump itself, i.e. mainly circulation pumps. The question is whether the consumption, which depends on the system configuration, has an influence when several solar-assisted systems are compared.

Thirdly an finally, the DHW generation will be simulated in two different tank storage configurations. Again, the objective is to find out how much the results are influenced when systems are simulated whose efficiencies strongly depend on temperature levels.

Regarding the source of the heat pump, this paper is restricted to air. The reason is that, on the one hand, such systems are gaining higher and higher market share in Germany, while on the other hand, such sources are in most cases colder and thus less efficient than for instance ground sources. So, regarding the interaction with solar-thermal collectors, the highest potential is expected here.

**Examined Systems**

The systems analysed in this paper were labelled from I till IV. The first one does not incorporate any solar collectors and acts as a reference system, see Figure 1.

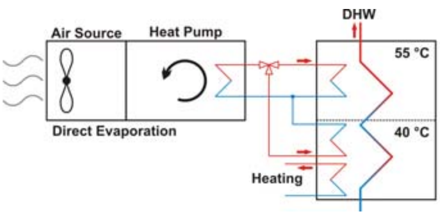


*Figure 1: System type I*

The remaining three systems are shown with growing complexity, starting with Figure 2, a configuration that is offered by many established companies on the German market.

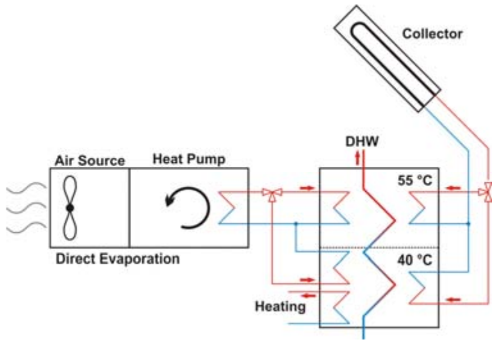


*Figure 2: System type II*

System type III is designed similarly, but an important difference is the introduction of a low-temperature buffer storage that acts as an alternative source for the heat pump, see Figure 3. Unfortunately, the necessary brine loop prevents direct evaporation as it was explained above. It is not disputed that such a system could theoretically be built with direct evaporation, i.e. the refrigerant flowing both through the air source and the buffer storage, but technical reasons are expected to avert this.

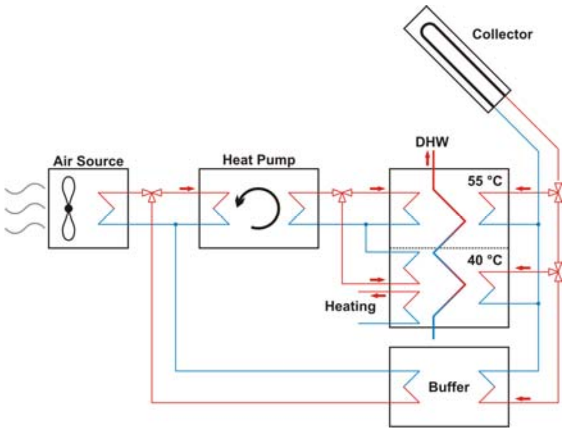


*Figure 3: System type III*

Figure 4 shows the last system, which is identical to type III regarding control and design except that the low-temperature does not contain water but a *phase-change material* (PCM), which is specified below.

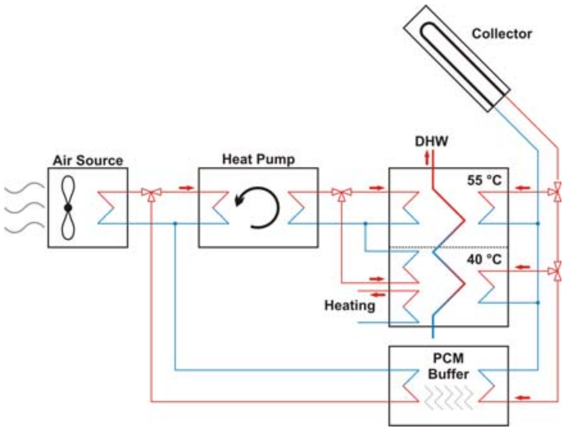


*Figure 4: System type IV*

### Methodology

For all systems, results will first be presented for idealised boundary conditions, i.e. without storage heat losses, with ideal heat transfer and without any auxiliary electricity consumption.

As the figures above show, no direct evaporation is applied in systems III and IV so that the low-temperature buffer storage can be linked to the evaporator of the heat pump. The additional heat exchanger required for this configuration is modelled with a constant efficiency of 85 %, which lowers the source temperature about 1 K. Afterwards, there will be a comparison with with these systems resimulated as if direct evaporation was applied, i.e. with the heat exchanger's efficiency set to 100 %.

To quantify the relation between solar gains and additional auxiliary energy required for these gains, all systems are resimulated. This time, a 60 W pump for the solar circuit is taken into account as well as pumps for the source-side circuit and the building-side circuit of the heat pump, each 40 W. Finally, the air-source fan is modelled to consume 80 W while operating.

The main indicator used for comparing the different systems is the *seasonal performance factor* (SPF) on a yearly basis. It is defined as the ratio between the heat delivered for space heating plus DHW during one year and the electrical energy consumed for heat pump operation ($SPF_{HP}$). If solar-thermal assistance is applied, it is taken into account by the system efficiency $SPF_{System}$. The $SPF_{System,aux}$ is used to include the auxiliary electricity consumption as defined above.

Within the scope of this paper, no cost analyses were carried out, neither for installations nor for operation. The reasons is – as it was explained above – that this paper is intended mainly to provide helpful data regarding methodology, i.e. to discuss the importancy of some simulation details.

The control algorithm for the solar collector is designed that its priority lies on DHW preparation. Contributing to space heating is the second priority. For system III and IV, the third priority is to heat the low-temperature buffer storage which acts as alternative source for the heat pump. This order of the priorities is derived from exergetic reasons. The theoretical background of heat pumps proves that they are inefficient when providing high-temperature energy, especially DHW. So, as long as the collector is capable, it is most sensible to use it for DHW preparation, while the low-temperature buffer is wisely chosen as last priority. Conditions to switch the collector's operational mode are based on the irradiation on the collector and the temperature difference between the respective storage and ambient air.

## SIMULATIONS

### Software

Within this paper, all simulations are carried out by using *IDA Indoor Climate and Energy* (IDA ICE) Version 4.0 Build 3.02 [Sahlin et al. 2004], and its equation-based modelling language NMF [Sahlin 1996]. Its similarity towards Modelica allows an easy exchange of models and know how. Furthermore, IDA ICE features a well-proven building model [Crawley et al. 2005] including a climate processor that can be linked to both buildings and collectors.

Unlike most simulation softwares, the length of each time step in IDA ICE is event-controlled, i.e. any time the solver registrates a larger change in any variable, a new time step begins. During night times, this can take up to one hour or, during dynamic processes like a storage giving a step response, up to a few seconds. The minimum and maximum time step length can be adjusted in the software settings.

### Building and Boundary Conditions

The virtual building depicted for these simulations is a single-family house located in Würzburg, Germany. Its architectural design including sizes, materials and thicknesses of walls and roof, size and properties of windows, air change rate and control of shading devices were taken from [Heimrath 2007]. From this report, which contains a detailed description of reference buildings, data referring to a building named SFH 60 – a single-family house with 60 kWh/m² heating demand – was selected. A similar approach was also chosen by [Citherlet et al. 2008]. Figure 5 shows the three-dimensional appearance and orientation.

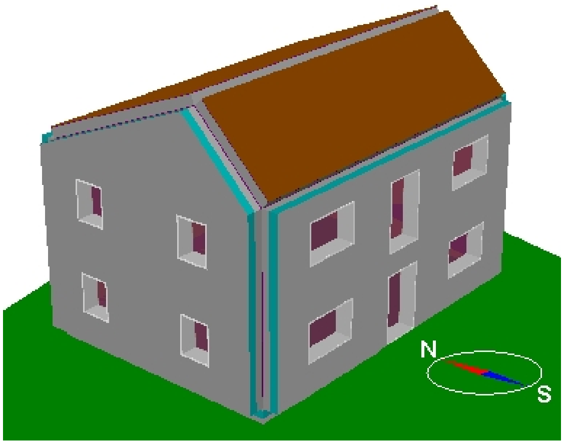


*Figure 5: Appearance of SFH 60 in IDA ICE*

In the following, the most important parameters of the simulated building are compiled.

- Location: Würzburg, Germany
- Weather data: Test Reference Year 13
- Living space: 140 m²
- Minimum room temperature: 20±0.5 °C, unreduced during night
- Heating demand: 8400 kWh/a, equivalent to 60 kWh/m²a
- DHW consumption: 2480 kWh/a at, equivalent to 17.7 kWh/m²a
- DHW storage temperature: 55±0.5 °C[1]
- Average occupancy: 2.4 persons, 100 W each, whereof 40 W as latent heat
- Average internal electricity loads: 200 W, equivalent to 12.5 kWh/m²a
- Air change rate: 0.4 1/h without heat recovery

Figure 6 shows variations of occupancy, internal loads and DHW consumption, which are modelled identically for every day of the year. The average values are identical to the definitions of [Heimrath 2007], whereas the variation was simplified.

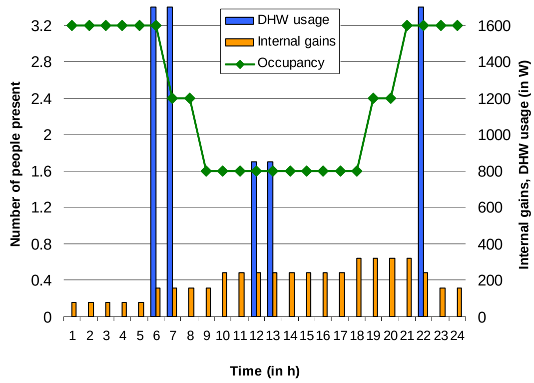


*Figure 6: Daily profile of occupancy, internal electricity gains and DHW usage*

The two floors of the building are simulated as one zone, whose IDA ICE plan is finally presented in Figure 7.

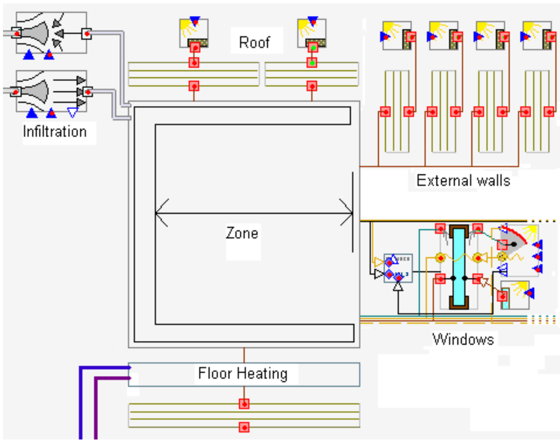


*Figure 7: Plan of SFH 60 in IDA ICE*

**Heating System Modelling**

The main components of the heating system to be modelled, namely solar collector, heat pump and storage, are described in the following.

The used collector is modelled according to [Isakson, Eriksson 1993], a model originally developed for TRNSYS and later adapted for IDA ICE. Its parameters are: 6 m² aperture area, tilted 40 °, oriented towards south, $\eta_0 = 0.8$, $a_1 = 3.5$ W/m²K, $a_2 = 0.02$W/m²K².

The fictive air-source heat pump is modelled according to two characteristic lines that are depicted in Figure 8. It is designed for monovalent operation and thus features a heating power of $P_{Heating}$ = 8 kW at A7/W35, while the respective COP is 4.2.

[1] The legionalla topic is ignored for simplicity's sake.

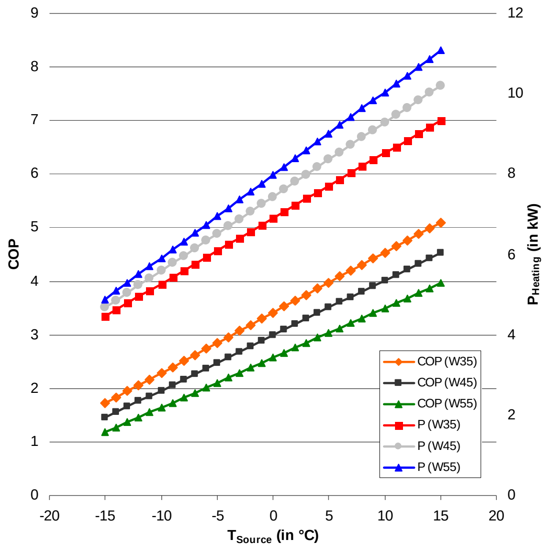


*Figure 8: Heat pump characteristics*

The storage provides DHW and space heating while it is fed by the heat pump and, if applicable, the solar collector as shown in Figure 2. In system III, it comprises three nodes, namely 300 l for DHW, 400 l for heating, and 300 l for low-temperature buffering.

The PCM buffer storage used in system IV is modelled with the phase change enthalpy of water, distributed by means of a differentiable function within a range of 5 K. The fission temperature is defined as 7.5 °C and the heat transfer is modelled to be ideal under all conditions.

## RESULTS AND DISCUSSION

### Without Auxiliary Energy

For the simulations ignoring auxiliary energy consumed by circulation pumps, Table 1 shows the results. All values are given with three digits. Considering the variety of estimated and sensitive boundary conditions, this should not be misunderstood. Such an accuracy can of course not exist in absolute measures, but the values to be compared are partly so similar to each other that an accuracy in relative terms is indispensible to see the effects.

*Table 1: Results ignoring auxiliary energy*

| SYSTEM | I | II | III | IV |
|---|---|---|---|---|
| $SPF_{HP}$ | 2.73 | 2.66 | 2.61 | 2.64 |
| $SPF_{System}$ | 2.73 | 3.15 | 3.09 | 3.12 |

For system I, the two values are identical because there is no collector and the heat pump is the only component of the system taken into account here.

In comparison, the system efficiency of system II is significantly higher because about 1800 kWh come as solar contribution, which is equivalent to a solar fraction of 16.3 %.

The direct solar contribution exists as well for system III, illustrated in Figure 9 as an orange bar. The figure shows the energy balance of the heating system. First divided into the useful energies, second according to the plants that supplied the energy, and third the energy balance of the heat pump in detail. It can be seen as yellow bar that the collector feeds 522 kWh to the low-temperature buffer tank. Nevertheless, the system efficiency remains unchanged.

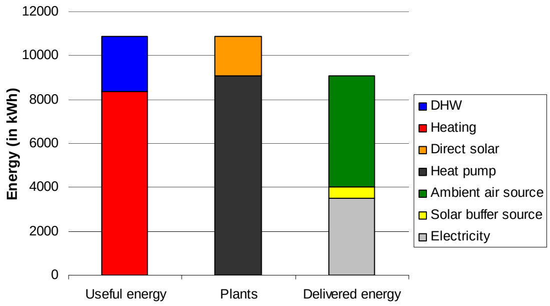


*Figure 9: Annual energy balance of system type III*

### Detailed Analysis

The reason is that direct evaporation is no longer applied. The details can be explained by interpreting Figure 10 where some results from an April day can be seen. No DHW is prepared here, which makes the explanation and interpretation easier while it has no influence on the key message.

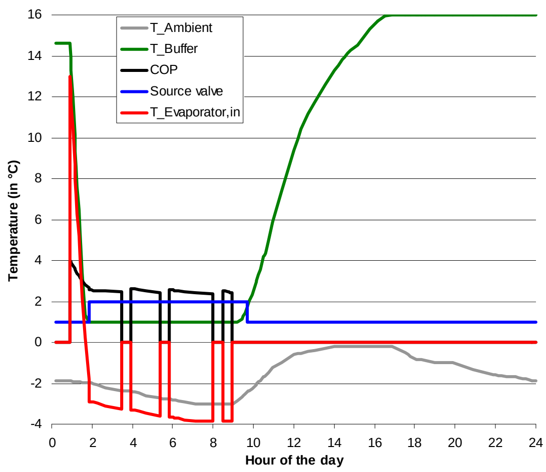


*Figure 10: Dynamic effects for system type III*

The ambient air temperature is below 0 °C for the whole day. At the beginning of the day, the buffer storage contains solar energy from the previous day. The selected source is therefore the buffer storage, which is indicated by position 1 of the valve that selects the source. The heat pump operation starts about 1 a.m. as the design room temperature has to be maintained day and night. Because the evaporator inlet temperature is close to the rather high buffer temperature, the *coefficient of performance* (COP) reaches a value of 4. When not operating, the COP is shown as 0.

Within less than one hour, the buffer storage temperature drops close to the freezing point, so the air source is selected instead – valve position 2. Immediately, the evaporator inlet temperature is lower than the ambient air temperature, and as a result the COP does not even reach 3. During the next hours, we see the heat pump in intermitting operation.

Around noon the buffer storage is heated up again. Though it was defined above that the collector priority is on direct heating, the specific irradiation on the collector is below 300 W/m² – unshown in the figure – which is considered too low for this purpose. No heat pump operation is required for the rest of the day due to irradiation and internal loads. As the buffer storage is heated up again by the end of the day, solar energy is shifted from this day to the next, again to serve as source.

All in all, it can be seen that the usage of the buffer storage has the potential to increase the average efficiency a lot. Unfortunately, it is designed so small that it is "emptied" far to soon. That fact that the air source is used predominantly is already evident from Figure 9. While the advantage of the system is obviously not fully utilised due to an insufficiently dimensioned collector, the dis-advantage comes in heavily. Going without direct evaporation means that the evaporator inlet temperature is about 1 K lower than it could be and has been for system I and II. The caused drop in efficiency is small, but accumulated over the year it is no wonder that the $SPF_{System}$ of system III is lower compared to system II.

### Direct Evaporation

For all systems are designed as if direct evaporation (index DE) is used, Table 2 shows the results. As no auxiliary energy is included, the values are to be compared with those of Table 1. The values of system I and II remain unchanged because direct evaporation is applied anyhow.

*Table 2: Simulating direct evaporation*

| SYSTEM | I | II | III | IV |
|---|---|---|---|---|
| $SPF_{HP,DE}$ | 2.73 | 2.66 | 2.69 | 2.71 |
| $SPF_{System,DE}$ | 2.73 | 3.15 | 3.19 | 3.21 |

Regarding system III and IV, some benefits can now be seen. They are marginal when compared to system II, but when compared to Table 1, the $SPF_{System}$ of system III increases by exactly 0.1.

These results are insofar important as they clearly show that simulating a system that is sensitive even to a small drop of the source temperature about 1 K can be easily and systematically misleading. In other words, as the complexity of such a system grows, the advantages have to be measured against the disadvantages that are brought.

### Phase Change Material

For system IV, 613 kWh are delivered to the PCM buffer tank. This is more compared to system III, but the effects are basically the same. Therefore, no significant increase in the efficiency can be seen.

### Auxiliary Energy

As we include the auxiliary energy consumed by pumps and air-source fan, the $SPF_{System,aux}$ are lower than the $SPF_{System}$ shown in Table 1. This could only be expected. Of greater interest is the fact that the relations between the efficiencies of each specific system type remains the same. So on the whole, the comparisons between the systems efficiencies remain unchanged.

*Table 3: Results including auxiliary energy*

| SYSTEM | I | II | III | IV |
|---|---|---|---|---|
| $SPF_{System,aux}$ | 2.58 | 2.96 | 2.92 | 2.94 |

By definition, the $SPF_{HP}$ is not affected by auxiliary energies and thus not shown here.

### Tank Storage Configurations

Figure 11 shows two possible configurations of the tank storage. As it was shown before, all simulations presented in this paper so far are carried out in configuration A.

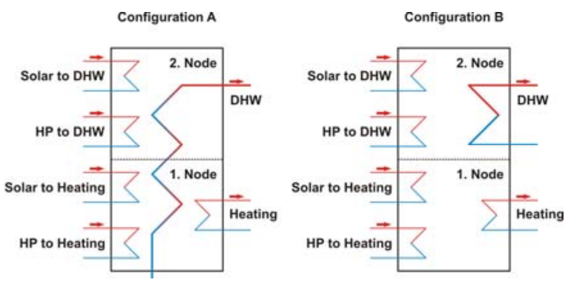


*Figure 11: Tank storage configurations*

For system II in configuration A, we get an $SPF_{System}$ of 3.15, as it is shown in Table 1. The corresponding value for configuration B is 2.92.

This can be explained by means of exergetic reasons, or more descriptively by the fact that a heat pump is always less efficient when the demanded supply-temperature is high. In configuration B, the DHW is completely prepared by means of such high-temperature heat, whereas in configuration A, it is pre-heated by less “valuable” heat.

Again, it is shown that simulations of systems that are sensitive to temperature levels depend on certain details. Moreover, in simulations of today’s building projects, the heating demand is mostly lower than the 60 kWh/m²a that were applied in this paper, especially when the passive house standard is applied. The comparision between configuration A and B would then be even more extreme due to the fact that the DHW production would dominate.

## CONCLUSION

In this paper, four solar-thermally assisted heat pump systems were defined and simulated. The focus of the simulations laid on 3 specific questions regarding the methodology. The results were analysed according to these questions.

The most important finding is that the introduction of a second source might reduce the system efficiency, whereas the opposite is intended. In this case, the second source was a solar-fed buffer storage, and its integration impeded the usage of direct evaporation. So, the average source temperature was lower than in the reference system, in which direct evaporation was applied.

This result is not meant to deny the potential of solar-assisted heat pump systems. Quite the contrary, positive effects of the interaction with the solar collector were shown indeed, but only for a small fraction of operation time. The main reason why the advantages were not able to compensate the disadvantages, much less to provide benefits, was that the collector was weakly dimensioned and thus not able to act as the main source of the heat pump.

Regarding the application of PCM, it is obvious that the system efficiency is in principal similar to a system using water as storage medium instead of PCM. If now the collector area is increased as mentioned above, the relation between these sytems’ efficiency can be expected to remain constant. Considering the fact that the PCM storage in this study was defined in a rather ideal way, the positive effects of using PCM seem to be low.

In a next step, the question could be answered whether auxiliary energy needed for the system pumps has to be taken into account. The auxiliary energy had no influence when the efficiencies of the systems were compared among each other. This finding is considered as valid as long as no pump dominates by means of its designed power.

Finally, an alternative configuration of the tank storage was modelled. Though it looked similar to the standard configuration at first glance, the second configuration caused the heat pump to increase its average supply temperature. This in turn resulted in a significantly lower system efficiency. If now the ratio between DHW and space heating is increased, the drop is expected even to be higher. Like the topic of direct evaporation, it was shown again that a system that is clearly sensitive to temperature levels has to be simulated carefully to get reliable results.

## ACKNOWLEDGEMENT

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