SOLAR AND HEAT PUMP SYSTEMS – SUMMARY OF SIMULATION RESULTS OF THE IEA SHC TASK 44/HPP ANNEX 38

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Abstract: Solar and heat pump heating systems have been simulated and analyzed by experts from different countries in Subtask C of the IEA SHC Task 44 / HPP Annex 38 "Solar and Heat Pump Systems" from 2010 to 2014. A summary of these simulation results is presented, compared and analyzed in this work, comprising systems where solar collectors provide heat in parallel to a heat pump, systems where solar heat is also or exclusively used for the evaporator of the heat pump, systems with ice storages, and systems with active solar regeneration of the ground source of the heat pump. Results show that the increase in overall system seasonal performance factor and electricity savings achieved by using solar thermal heat in combination with a heat pump depends to a large extent on the system concept, on the climate, and on the heat load that is to be served. For many system concepts small details are decisive for the system performance, among these are e.g. hydraulics and control of the heat pump integration into systems with combi-stores.

Key Words: heat pumps, solar thermal, combined systems

1 INTRODUCTION

The idea to combine heat pumps with solar thermal systems is not new. Already in 1955 Sporn & Ambrose presented a direct evaporation solar thermal collector that was used for a heat pump water heater. Similar designs are still built today, predominantly in warmer climates, e.g. in Portugal (Facão & Carvalho 2014) or Australia (Morrison et al. 2004), where there is enough sunshine in all seasons of the year and chances that the coefficient of performance (COP) of the heat pump can be increased compared to an air source system are high. However, this kind of concept and application is only one of numerous different possibilities to combine solar thermal systems with heat pumps. Within the IEA Solar Heating and Cooling Task 44 and Heat Pump Programme Annex 38 (T44A38), different designs of solar and heat pump systems have been investigated from 2010 to 2014. The investigations were in most cases restricted to the application of space heating and domestic hot water (DHW) for single family homes – with some exceptions that also included multifamily homes or cooling applications. Subtask C of T44A38 dealt with modeling and system simulations of solar and heat pump systems. Component models that are typically used for the simulation of solar and heat pump systems have been reviewed in Haller et al. (2012). It is important to notice that for the simulation of special solar and heat pump applications such as ground regeneration by solar collectors, collector operation below the ambient temperature, and heat source temperatures that are out of the range of normal heat pump operating conditions, care has to be taken to use the right component models, that may differ from the standard

component models used for normal applications, and the proper parameters for these models, that may not be available from standard test procedures.
This paper is presenting and reviewing results of simulations of different system concepts that were performed by different experts that participated in T44A38.

2 GENERAL SYSTEM CONCEPTS AND PERFORMANCE FIGURES

The possible configurations of solar and heat pump systems for heating and cooling are innumerable. Within T44A38 a common nomenclature and a common way of representation of these concepts in an energy flow chart has been developed and presented in Frank et al. (2010). The basic concepts used in this paper are:

- ***Parallel (P)***: in parallel system concepts the solar collectors and the heat pump are hydraulically connected in a parallel manner, i.e. it is not possible to use solar heat for the evaporator of the heat pump, and consequently the heat pump always uses another heat source, e.g. ambient air or a ground heat exchanger
- ***Series (S)***: in series system concepts the heat pump uses solar heat for the evaporator
- ***P/S:*** in a ***Parallel/Series*** concept, both ways of using collector heat are possible, either directly to serve the load or for storage on the hot side of the heat pump (P), or for the evaporator of the heat pump (S). The heat pump may in this case be a ***single source heat pump*** that uses only heat from the solar collectors, or a ***dual source heat pump*** that may also use an alternative heat source
- ***Regenerative (R)***: if the main source of the heat pump is a ground heat exchanger, solar heat can be used to regenerate the ground, thus increasing the temperature of the ground heat source
- ***P/R:*** in a ***Parallel/Regenerative*** system, both direct collector heat use as well as ground regeneration are possible

This list is not exhaustive, since there are other possible combinations of P, S, and R. Figures 1 - 2 give examples for some of these system concepts presented in the energy flow chart scheme of T44A38.

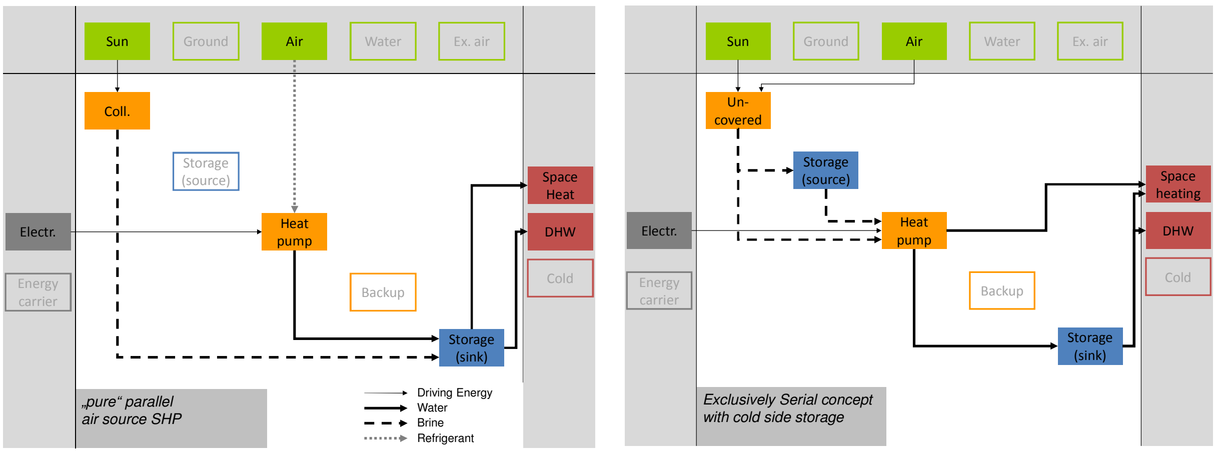

Figure 1: Energy flow charts of a parallel (P) system concept (left) and of a series (S) system concept with single source heat pump (right).

In order to be able to compare these systems, performance figures had to be defined that are based on the same system boundaries and assumptions. The seasonal performance factor (SPF) was thus defined by the ratio of useful heat delivered for space heating ($\dot{Q}_{SH}$) and DHW ($\dot{Q}_{DHW}$) to the total electricity consumption of all components of the system.

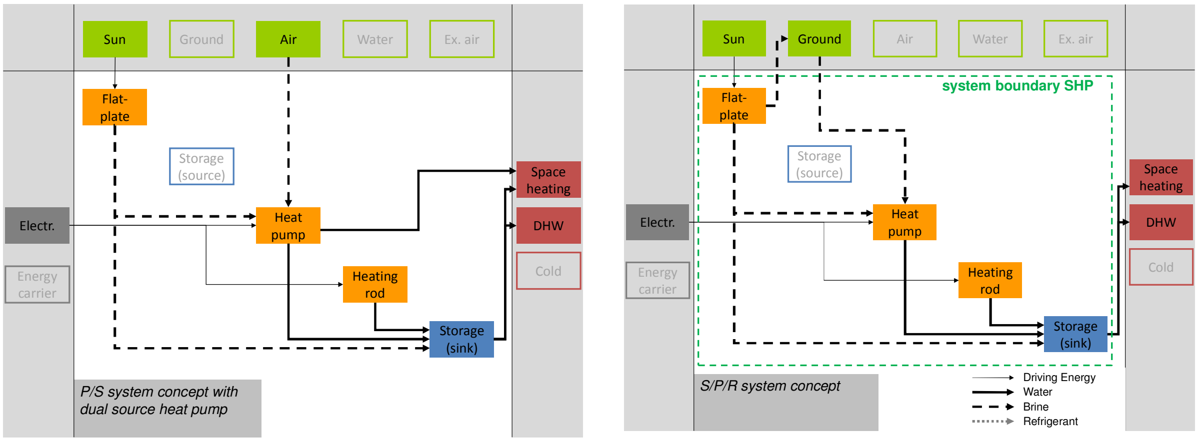

Figure 2: Energy flow charts of a P/S system concept with dual source heat pump (left) and system boundaries for the determination of the seasonal performance factor of the system based on an S/P/R concept (right).

The total electricity consumption includes all electric consumers that are needed to operate the system, either without ($P_{el,SHP}$) or with ($P_{el,SHP+}$) the pump of the hydronic space heat distribution[1]:

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

$$SPF_{SHP+} = \frac{\int \left(\dot{Q}_{SH} + \dot{Q}_{DHW} \right) \cdot dt}{\int P_{el,SHP+} \cdot dt} \tag{2}$$

The above definition of seasonal performance factors differs substantially from other definitions that have been used in heat pump system analysis. As an example, in field studies presented by Miara et al. (2011), four different seasonal performance factors were calculated, but none of them included the heat storage in the system boundaries. Other field studies presented by Erb et al. (2004) used three different performance factor definitions, two of these included the technical storage for space heating, but none of them included the DHW storage. Thus, performance factors presented according to the above definitions of T44A38 cannot be compared directly to these studies since they are lower than values where the useful heat is taken at the storage input. Exemplary simulation studies have shown that the difference in SPF evaluated before and after storage is:

- 0.2 to 0.5 for heat pump systems, and
- 0.4 to 1.0 for solar and heat pump systems where the storage is heated to higher temperatures in summer.

3 COMPARABLE BOUNDARY CONDITIONS

The SPF of solar and heat pump systems does not only depend on the system concept that is chosen and on the quality of its implementation, but to a large degree also on the boundary conditions of the climate and heat sources (temperatures of ambient air and ground, solar irradiation) on the one hand, and on the characteristics of the heat demand (temperature levels, distribution over the year) on the other hand. Whereas it is virtually impossible to reproduce boundary conditions in field testing, identical boundary conditions

[1] None of the Task participants presented results for systems with space heat distribution by air circulation.

may be applied in laboratory testing and in system simulation, which enables a direct comparison of the performance of different system concepts. For this reason, common boundary conditions for system simulations have been defined within T44A38. This includes the demand of space heat and domestic hot water as well as the climate and the properties of the ground and the ground source heat exchanger (GHX) (Dott et al. 2013; Haller et al. 2013a). Table 1 gives an overview of these boundary conditions for the climate of Strasbourg for three different space heating demands. SFH45 is a single family house (SFH) with a specific demand for space heating of 45 kWh/(m^{2}a). SFH15 is a passivehouse standard building with 15 kWh/(m^{2}a), whereas SFH100 has a demand of 100 kWh/(m^{2}a). The heated floor surface area of all buildings is 140 m^2. The demand for DHW (Q_{DHW}) is 2076 kWh/a for all buildings.

Table 1: Space heat and DHW load for the different buildings in Strasbourg.

Q_{SH} SFH15	2474	kWh/a, 35/30 °C
Q_{SH} SFH45	6476	kWh/a, 35/30 °C
Q_{SH} SFH100	14031	kWh/a, 55/45 °C
Q_{DHW}	2076	kWh/a

Within T44A38, the simulation platforms TRNSYS, Polysun, Matlab/Simulink, and IDA-ICE were used. The correct implementation of the boundary conditions on different platforms was determined by a platform independence check that included a comparison of monthly values for:

- solar irradiation on the horizontal and both direct and diffuse irradiation the 45° sloped south oriented surface
- heat demand for both DHW and space heating
- average ambient air temperature, humidity, and wind speed
- average temperature difference between the ambient and the fictive sky

The temperatures of space heat supply and returned were compared with cumulative energy curves that are shown in Figure 3. These curves show the quantity of heat (on the y-axis) that is provided with a supply / return temperature below a certain temperature level (on the x-axis).

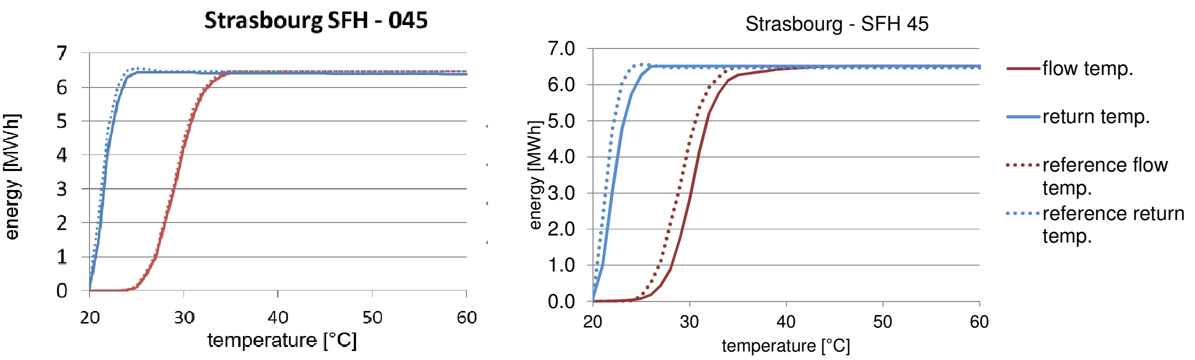

Figure 3: Comparison of space heat supply and return temperatures for different implementation of the T44A38 boundary conditions; left: implementation in Polysun by Zimmermann & Haller (2013); right: implementation in TRNSYS by Bertram (2013a).

An easy way of feignedly improving the system's energetic performance is to reduce the set temperature for DHW - that defines when the heat pump starts and stops charging the DHW storage tank - below the temperature that is needed to maintain a high comfort for the user. In order to prevent this, a minimum temperature level of 45 °C to the DHW distribution has been defined for most DHW draw-offs. If this temperature limit is not reached within a time step of simulation, the "missing energy" was counted as direct electric heating with a "penalty factor" of 1.5, i.e.:

$$P_{el,DHW,pen} = 1.5 \cdot \dot{m}_{DHW} \cdot cp_{wat} \cdot \max\left(\vartheta_{DHW,set} - \vartheta_{DHW};0\right) \qquad (3)$$

Simulations were discarded if the total penalties for DHW were larger than 2% of the total DHW demand, based on annual values. A similar approach was used for space heating if the simulated room temperatures dropped below 19.5 °C (Weiss 2003, pp.137–140).

4 OVERALL RESULTS FOR DIFFERENT SYSTEM CONCEPTS

In this section, SPF is used in general for $SPF_{SHP+,pen}$, where the subscript "pen" indicated that comfort penalties have been included in the calculation of the total electricity demand of the system and thus in SPF.

4.1 Strasbourg, SFH45

Figure 4 to Figure 6 show the seasonal performance factors for different system concepts with air, ground, or solar as the main heat source for the heat pump, for the SFH45 building in the climate of Strasbourg. The information on the left hand side is:

- ***ID:*** identification number for back-tracking of data
- ***Vst:*** total volume of all storage devices (i.e. also of possibly present cold storages at the source side of the heat pump) [m³]
- ***Acoll:*** total collector area [m²]
- ***type of collector:*** FP = flat plate, UC = unglazed absorber, SE = selective unglazed absorber, ET = evacuated tube
- ***classification:*** "Ref" = reference without solar; or according to the S/P/R notation presented in chapter 2.
- ***hp source:*** source(s) of the heat pump: A = air, G = ground, S = solar, GSR = ground, solar and space heating circuit

4.1.1 Air source systems

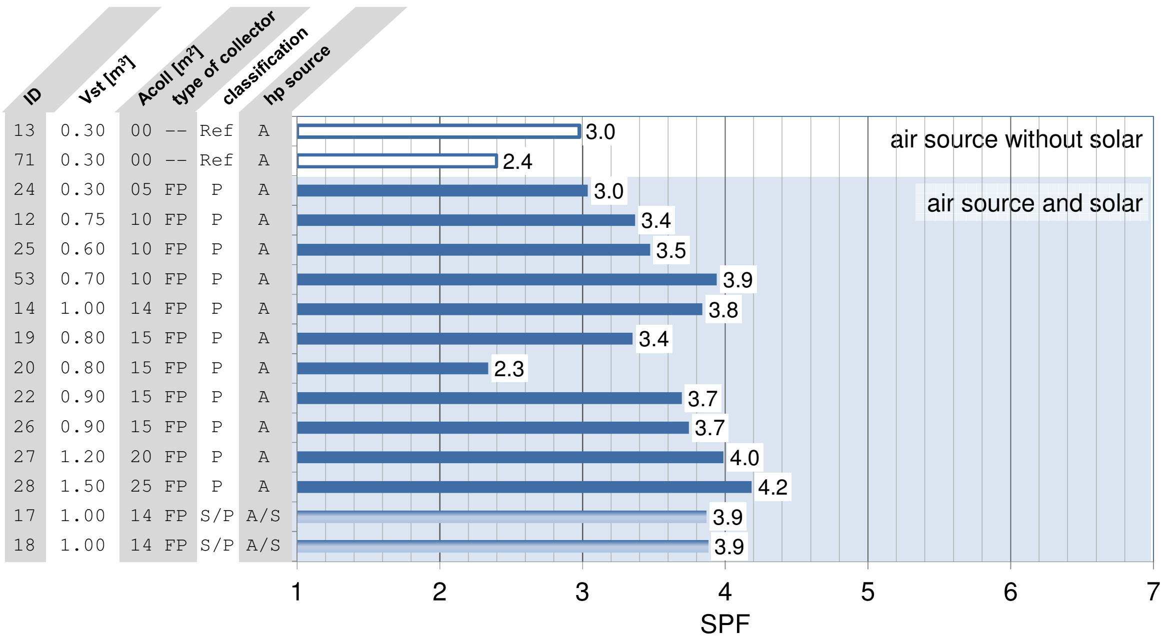

Figure 4: SPF of systems with air as the main heat source for the heat pump.

Simulations of air source systems without solar achieved SPF between 2.4 and 3.0. Simulations with solar achieved up to SPF 4.2 with large collector areas of 25 m², and up to 3.9 with smaller collector areas of 10 m². An outlier can be found with SPF 2.3 for a simulation where the hydraulics and control of combining the heat pump with a solar combi-storage was inappropriate. The same combination of components with a better solution for hydraulics and control achieved SPF 3.4. Series/Parallel system concepts (S/P) with dual source heat pumps do not seem to outperform the pure parallel system concepts according to the results of these simulation studies.

4.1.2 Ground source systems

Parallel ground source and solar system concepts reached SPF of 4.5 – 6.5 in different simulation studies. Systems without solar reached SPF 3.5 – 3.9. Ground regenerating systems with uncovered collectors do not seem to perform significantly better than the systems without solar collectors. Furthermore, the special system concept with a ground source heat pump for space heating and an additional heat pump for DHW that uses either solar collectors or the space heating circuit as a heat source (Citherlet et al. 2013) showed a performance that was similar as for a ground source heat pump without solar collectors.

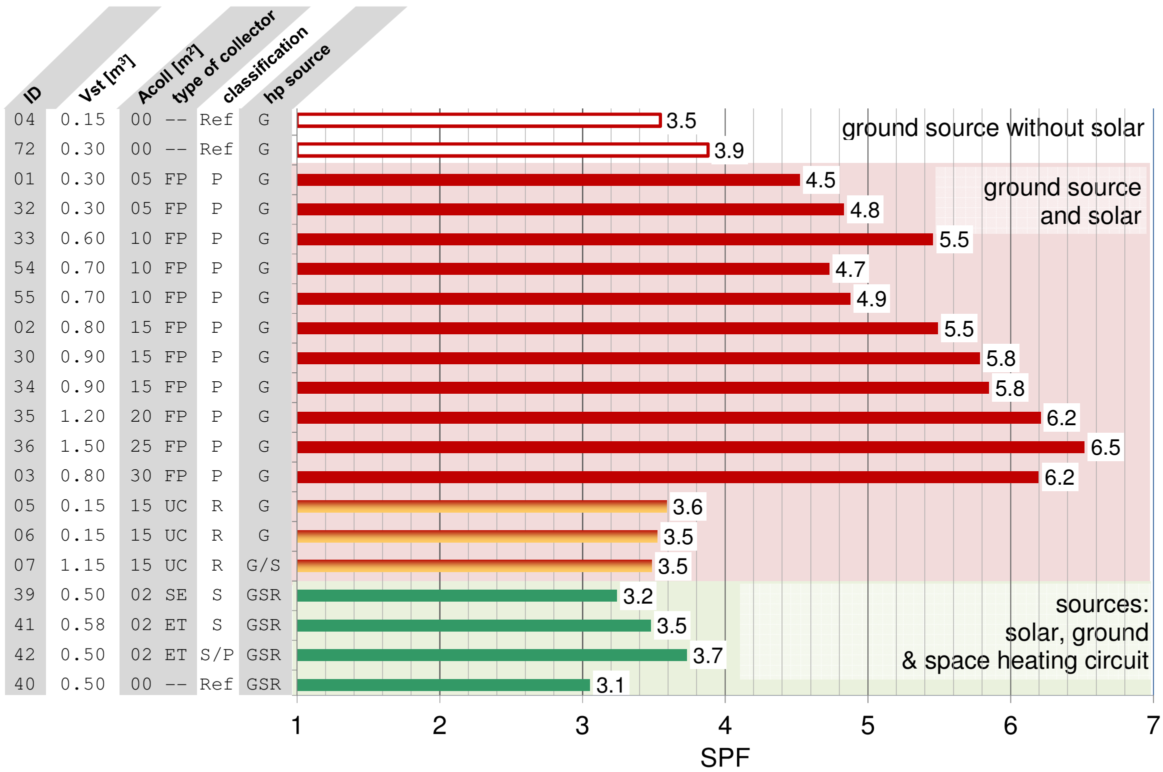

Figure 5: SPF of systems with ground as the main heat source for the heat pump.

4.1.3 "Solar only" source

The performance of systems with only solar collectors as a heat source was for most cases in the same range as for a parallel solar and air source system, or a system with only a ground source heat pump without solar collectors. One outlier is a system with SPF 5.3. This system also has a significantly higher cost of invest because it is equipped with an ice-storage of 20 m³ that is buried in the garden and a collector field that is composed of 20 m² flat plate collectors plus an additional 5 m² of unglazed selective absorbers. Another system with an ice storage was able to perform similar to ground source heat pump without solar collectors with 10 m³ of ice storage and 10 m² of uncovered collectors.

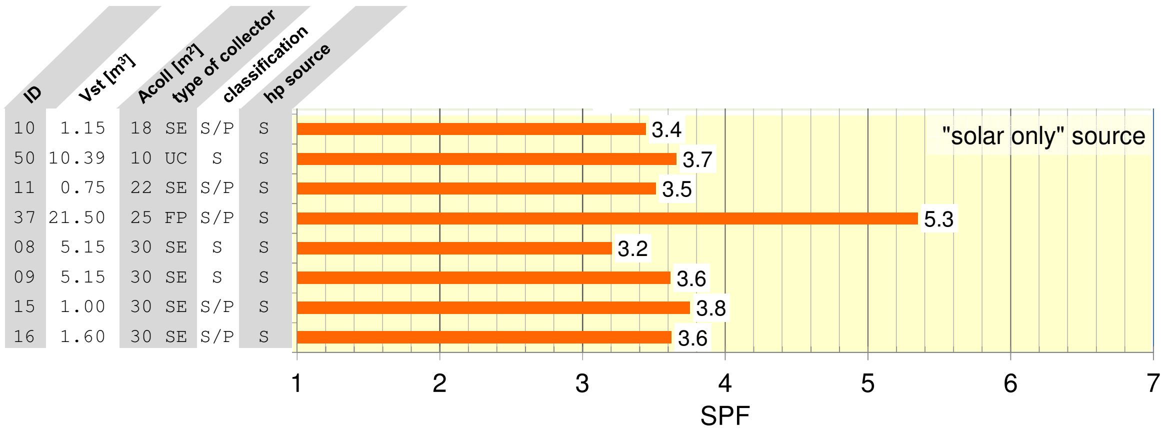

Figure 6: SPF of the solar and heat pump systems with solar heat as the only source for the heat pump.

4.1.4 All concepts

In general, the highest seasonal performance factors of up to 6.5 have been achieved with parallel ground source and solar combinations, followed by air source and "solar only" source systems. Figure 7 shows the SPF as a function of the area of the collector field. A general increase of SPF with collector area has only been found for systems with ground or air as the main heat sources, but not for systems with solar as the only heat source. Figure 8 shows that the electricity demand of all systems for SFH45 in Strasbourg was clearly dominated by the heat pump. If the electricity demand of the rest was significant, then this was usually the result of electric backup heating.

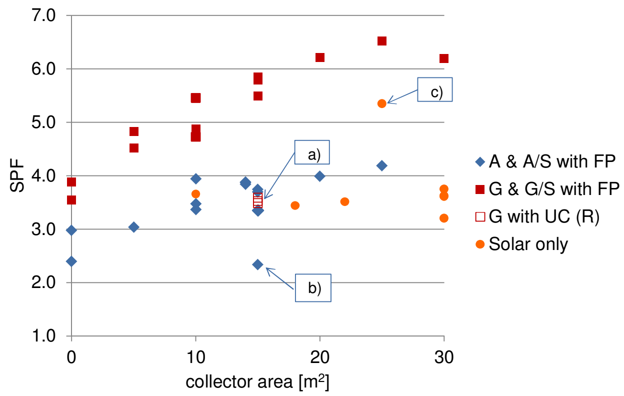

Figure 7: SPF vs collector area for different system concepts. Outliers are a) uncovered collectors for ground regeneration; b) inappropriate hydraulics and control, c) large ice-storage.

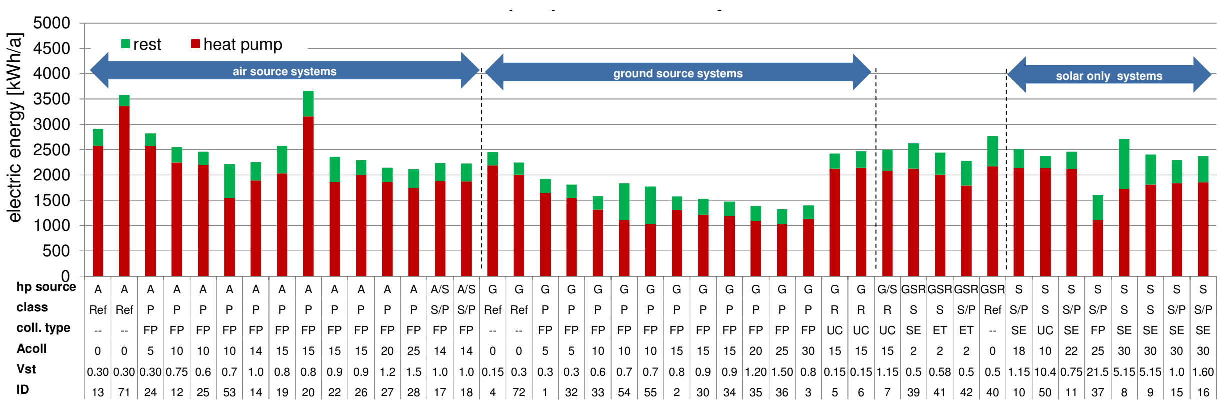

Figure 8: Electric energy demand split into heat pump and rest of the system.

4.2 Strasbourg SFH15 and SFH100

Similar results have been produced for buildings with lower and higher heat loads. In absolute terms electricity consumers that are additional to the heat pump are not more important for SFH15[2] than for SFH100, but in relative terms they are responsibly for a much higher share of the consumption and thus may have a large influence on the SPF of the system. Further, the significance of increasing SPF for SFH100 is entirely different than for SFH15. For example, increasing SPF_{SHP} from 3.0 to 4.0 saves 1340 kWh_{el}/a for SFH100, but only 380 kWh_{el}/a for SFH15.

4.3 Different climates

Some of the system concepts have also been simulated in different climates such as Helsinki or Davos. Some of these results will be available in the Task reports and handbook that are published in 2014.

5 INSIGHT INTO SOME OF THE SYSTEM CONCEPTS

5.1 Benefit of adding solar to a heat pump in parallel

The main advantage of parallel solar and heat pump systems compared to systems without solar is that the seasonal performance factor of the system can be increased, and thus electric energy can be saved. The increase of SPF, as well as the electric savings, depend on the type of heat pump, the type and area of the collectors, and also on the boundary conditions of the climate and the heat load. Figure 9 shows for air source (ASHP) and ground source (GSHP) systems that the results are quite dependent on the climatic conditions that are chosen, in particular on the solar resources that are available in winter.

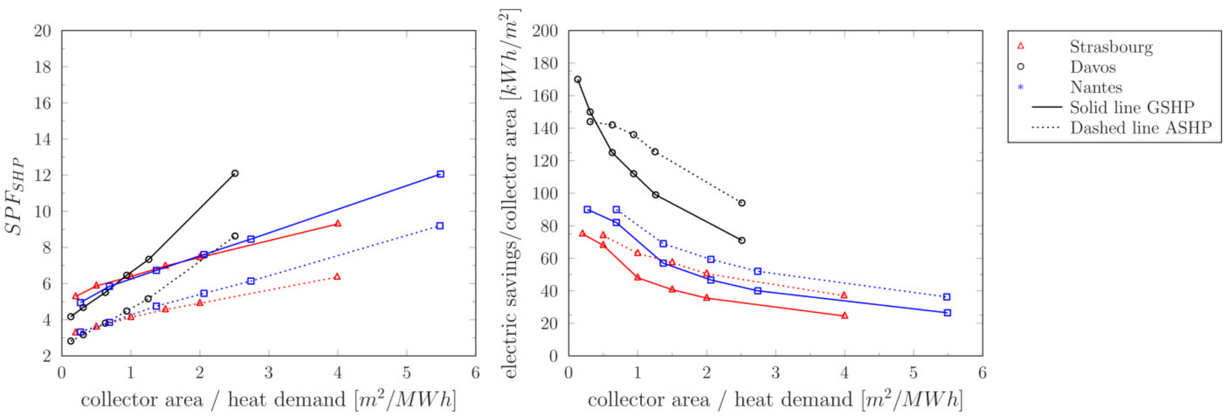

Figure 9: SPF the system and electric savings per m2 collector area for ground source and air source solar and heat pump systems in different climates and with collector areas of 2, 5, 10, 15, 20 and 40 m2 (recalculated from Carbonell et al. 2013).

5.2 Influence of space heat distribution

[2] without counting the air handling unit with heat recovery that is needed to reach this building standard.

Simulation results have shown that refraining from using thermostatic valves in solar and heat pump systems may lead to a higher seasonal performance factor. However, this does not mean that these systems consume less electric energy. The reason for this is that the building is overheated on days with passive solar gains, and the amount of heat supplied to the heat distribution is more than actually required in these periods. Since the heat demand increases more than the SPF, the total electricity consumption increases too (Figure 10).

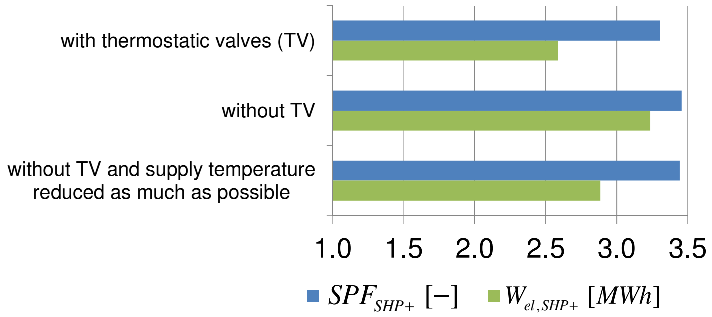

Figure 10: Performance of a solar and heat pump heating system with different control of space heat distribution.

5.3 Hydraulic integration and control of combi-storage charging

Solar combi-storages are used frequently in some European countries to store heat for DHW and for space heating in one device. Solar combi-storages are making use of natural stratification to maintain the uppermost part of the storage at the temperature for DHW, while the middle part may be at a temperature level that is 20 K lower for space heating with low temperature hydronic distribution systems, and the lowest third of the storage volume is used for DHW-preheating at around 10 – 30 °C.
The integration of combi-storages into systems with heat pumps deserves special attention, since insufficient stratification or poor hydraulic integration and control may lead to excessive charging of the DHW-zone by the heat pump and thus decrease the seasonal performance factor of the system dramatically. Based on annual system simulations (Haller et al. 2013b) and additional CFD investigations of storage charging with smaller timescales (Huggenberger 2013), the following recommendations can be given for the combination of solar-combistores with heat pumps:

1. The position of the DHW sensor for boiler charging control must be placed at a safe distance from the space heating zone of the storage:
    a. this distance is system-specific (it depends on the stratification capabilities of the storage),
    b. as a first guess, a minimum distance of 30 cm is recommended for a storage with a diameter of around 80 cm.
2. The return from the storage to the heat pump in DHW mode must be placed above the space heating zone of the storage.

Furthermore, it can be advantageous to bypass the storage when the heat pump runs in the space heating mode, and to set a time-window that allows for DHW charging by the heat pump only for few hours in the evening. The capabilities of the storage to maintain stratification should be tested with the relevant mass flow rates of charging and discharging that should not be surpassed in field installations.

5.4 Series concepts with dual source heat pumps

Series concepts with dual source heat pumps in general have not shown a better performance than their parallel solar and heat pump counterparts. Haller & Frank (2011) derived mathematically that using collector heat for the evaporator of the heat pump instead of using it directly in a dual source heat pump system only improves the system's performance if the following conditions are met:

$$\frac{\Delta COP_{hp}}{\left(COP_{hp,dir}-1\right)}\cdot\frac{\Delta\eta_{coll}}{\eta_{coll,dir}}>1 \qquad (4)$$

Where $COP_{hp,dir}$ is the COP of the heat pump using the other of the two heat sources (e.g. air, ground), ΔCOP_{hp} is the increase in COP that results from using solar heat for the heat pump instead of the other heat source, $\eta_{coll,dir}$ is the efficiency of the collector used directly at the temperature level of the heat demand, and $\Delta\eta_{coll}$ is the increase in efficiency that results from using the collector as a source for the heat pump instead. The condition presented in equation (4) is not easy to be met. E.g., by using solar heat in series instead of operating heat pump and solar collector in parallel, the COP of the heat pump would have to increase from 3.0 to 4.0, while the collector efficiency would have to multiply by a factor of 3 simultaneously. A threefold increase of collector efficiency requires a quite low efficiency to start with, which is typically only the case when the specific irradiance on the collector field was low, and thus little heat can be collected even if the collector efficiency is increased. In order to increase the COP of the heat pump from 3.0 to 4.0, the temperature that is provided by the collector must be substantially higher than the one of the alternate heat source, and the collector must deliver enough heat to match the demand of the heat pump.

5.5 "solar only" series or parallel/series concepts

Concepts with solar heat as the only heat source for the heat pump (single source heat pump) usually come along with uncovered collectors or special collector designs such as presented in Leibfried & Storck (2010) or Thissen (2011), that can use the ambient air as a heat source when there is no or only little solar irradiance. A cold side storage (e.g. an ice storage or a glycol storage) may be used to overcome periods of snow cover or extremely low ambient temperatures, before electric backup heating is replacing heat pump heating. These systems have the advantage that drilling boreholes can be avoided and no ventilated air source heat exchanger unit is needed. For many of the system concepts presented for this type of system research is still on-going, and a common agreement on the dimensioning of these systems has not been found yet. In particular, the behavior with snow cover or increasing ice layers on uncovered absorbers is often not reflected in simulation models and thus simulation results for these systems are to be taken with caution.

5.6 Regeneration

Ground regeneration for a properly dimensioned single borehole has not shown significant performance improvements in any of the simulation studies that were evaluated. Several studies have shown that the performance of systems with under-dimensioned boreholes can be increased to the level of a properly designed system (Kjellsson et al. 2010; Bertram 2013b; Ochs et al. 2014). This could be particularly interesting for retrofitting of existing boreholes, and eventually also for downsizing of borehole projects. Regeneration of borehole fields is a different topic that has not been studied within T44A38. The combination of PV/T collectors with ground regeneration has shown to increase PV yield by 4%, with the potential to increase it by 10% for hotter conditions - e.g. for roof integrated PV and low wind speeds (Bertram et al. 2012).

6 CONCLUSION

Within the IEA SHC Task 44 / HPP Annex 38, solar and heat pump systems have been analyzed systematically. Within Subtask C, boundary conditions for simulations were defined in combination with platform independence checks that allow for comparison of performance simulation results by different authors using different simulation platforms, based on harmonized definitions of the relevant performance figures. A selection of simulation results presented by the different Task participants shows the increase in seasonal performance factors and the electricity savings that can be achieved with parallel solar and heat pump combinations. The parallel/series concepts with dual source heat pump have not shown significant advantages over the parallel concepts. Parallel/series concepts that are based on solar heat only (single source heat pump) on the other hand have the advantage that they do not need ventilated air source evaporators or ground heat exchangers. They achieved similar performance as a parallel air source and solar combination or a ground source heat pump without solar. Higher seasonal performance factors were obtained for these concepts with higher effort, i.e. larger collector area and/or storage volume. Ground regeneration of a single borehole has only shown significant increase in performance for undersized boreholes. None of the series or regenerative system concepts was able to beat the performance of the best parallel ground source and solar combination. The influence of thermostatic valves on SPF and electricity demand has been shown, and design recommendations have been given for the integration of combi-storages with heat pumps.

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