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Influence of hydraulics and control of thermal storage in solar assisted heat pump combisystems

Stefano Poppi[a,b] *, Chris Bales[a]

[a]Solar Energy Research Center (SERC), Dalarna University College, S-79188 Falun, Sweden
[b]Department of Energy Technology, KTH, SE-100 44 Stockholm, Sweden

Abstract

This paper studies the influence of hydraulics and control of thermal storage in systems combined with solar thermal and heat pump for the production of warm water and space heating in dwellings. A reference air source heat pump system with flat plate collectors connected to a combistore was defined and modeled together with the IEA SHC Task 44 / HPP Annex 38 (T44A38) "Solar and Heat Pump Systems" boundary conditions of Strasbourg climate and SFH45 building. Three and four pipe connections as well as use of internal and external heat exchangers for DHW preparation were investigated as well as sensor height for charging of the DHW zone in the store. The temperature in this zone was varied to ensure the same DHW comfort was achieved in all cases. The results show that the four pipe connection results in 9% improvement in SPF compared to three pipe and that the external heat exchanger for DHW preparation leads to a 2% improvement compared to the reference case. Additionally the sensor height for charging the DHW zone of the store should not be too low, otherwise system performance is adversely affected.

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Keywords: solar combisystems; heat pump; water storage; hydraulics and control

* Corresponding author. Tel.: +46 (0)23 778746; fax: +46 (0)23 778701.
E-mail address: spo@du.se

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1. Introduction

The practice of combining solar thermal with heat pump is widespread in the market of heat production for domestic uses. Many system configurations can be realized, but the most common developed by manufacturers is the parallel system. In parallel systems, both solar collector and heat pump either provide heat for space heating (SH) and domestic hot water (DHW) or for charging a hot store [1,2,3].

The use of thermal storage is necessary to accumulate the heat produced by solar collector and make it available to users when it is required. It can also be advantageous for operating the auxiliary heat source at optimal conditions or to reduce the number of on/off cycles. Thus, sizing and improving the efficiency of stratification become fundamental as shown in [4].

Heat pump performance depends on sink temperature, so changing the set point of hot water preparation in the store strongly affects its efficiency. Energy consumption increases up to 25% when sink temperature increases from 45 to 60°C at 8°C of evaporation temperature as derived from [5]. Hydraulics and control of heat pump also affect the energy performance of the whole system. In particular, significant energy savings can be achieved by turning-off the space heating distribution pump during DHW preparation as shown in [6].

The aim of the study was to analyze the influence of the three pipes and four pipes connections between heat pump and store as well as methods to charge the DHW zone of the store. In addition the influence of sizing the DHW heat exchanger has been studied for both external and internal solutions for DHW preparation as well as different positions of the temperature sensor in the upper volume of the store.

Costs of components were not considered, thus cost-effectiveness analysis has not been included. Consequently, annual electricity consumption and seasonal performance factor (SPF), defined similarly to [7], were used to compare results from different solutions.

This investigation has been carried out within the frame of the European Union's Seventh Framework Program FP7/2007-2011 in a project called MacSheep.

Nomenclature

DHW	domestic hot water
SH	space heating
SPF	seasonal performance factor
SHP	solar and heat pump system
Pel,Wel	electric power (kW) and energy (kWh)
pen	penalties
dist	distribution
SC	solar collector (loop)
PU	pumps
Ctrl	control
EH	electrical heating element (in the heat pump)
(T44A38)	IEA SHC Task 44 / HPP Annex 38

2. Methodology

This study has been carried out with the simulation platform TRNSYS 17 [8] with the boundary conditions defined for IEA SHC Task 44 / HPP Annex 38 (T44A38) "Solar and Heat Pump Systems" [9,10]. The climate of Strasbourg and building SFH45 were chosen for the study. Some key figures for these boundary conditions are given in Tab. 1.

Tab. 1. Key data for SH and DHW for Strasbourg climate according to T44/A38 boundaries

Building	Unit	SFH45
Supply temperature for SH	°C	35
Return temperature for SH	°C	30
Heat load	kWh/m^2	46
Annual space heating load	kWh	6434
Supply temperature for DHW	°C	The minimum temperature that has to be achieved is 45°C for all discharges except for the dishwashing (55°C).
Cold water temperature for DHW	°C	10
Annual DHW discharge energy	kWh	2075

For this study the outer system boundary was used, meaning that even the space heat distribution pump is included in the total electricity use. Penalty values were also included in the calculations and in addition were kept as constant as possible for all simulations to ensure that all system variations provided the same comfort level as well as supplied energy. In practice, the only variation in penalty value was for the supply of DHW, and so the set temperature for charging the store was varied so that the DHW penalty was always 0.85% ± 0.05% of the total DHW load (QDHW). Thus, seasonal performance factors ($SPF_{SHP+,pen}$) and total electricity use ($Wel_{,SHP+,pen}$) have been defined as following:

$$W_{el,SHP,+dist} = W_{el,HP} + W_{el,SC} + W_{el,\,EH} + W_{el,PU} + W_{el,Ctr} + W_{el,dist} \quad (1)$$

$$SPF_{SHP+pen} = \frac{\int (\dot{Q_{SH}} + \dot{Q_{DHW}}).dt}{\int (P_{el,SHP,+dist} + P_{el,DHW,pen} + P_{el,SH,pen}).dt} \quad (2)$$

A reference system was defined for this study based on a state of the art air source heat pump and parallel configuration. The parameter values for the models were derived from measurement data from specific component tests carried out at test institutes or, in the case of the heat pump, from the manufacturer. Details are given in the following chapter. The complete system model was not verified against measured data. Based on this reference system a number of variations were made and then simulated for a complete year. Results are only given for the complete year.

3. Reference system

3.1. Description

A schematic of the reference system is shown in Fig. 1. The system layout has been taken from systems already available on the market.

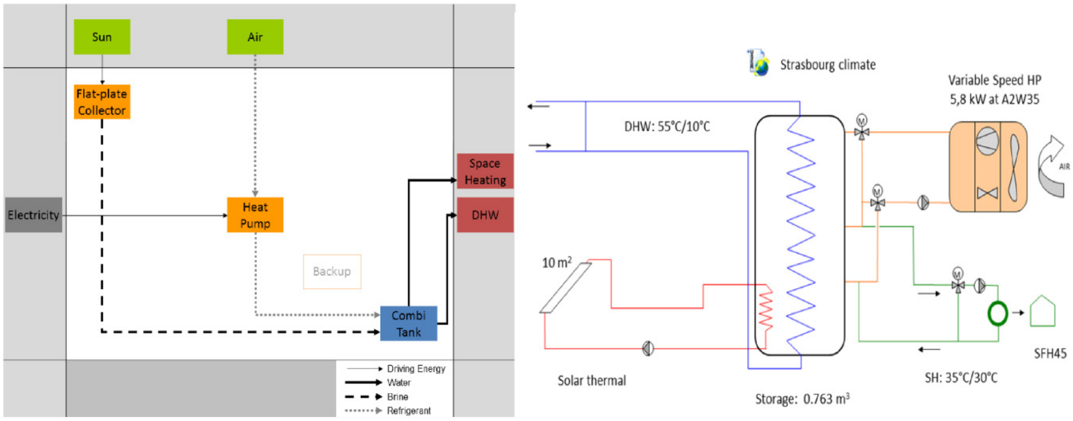

Fig. 1. Square view diagram (left) and scheme of the reference system (right).

It is a parallel system with the solar collectors that charge the hot water store via an internal heat exchanger and an air source heat pump that either charges the main store or serves the space heat load directly. Solar thermal consists of flat plate collectors that are tilted 45° and orientated to the south. The total absorber area for 4 modules is 9.28 m^2. More details are shown in [11]. The 750 liters water store [12] has a solar coil in the lower volume of the store and a stainless steel heat exchanger that covers the whole store for the preparation of DHW.

The air source heat pump is a R410A split unit coupled with a variable speed compressor. It has a COP of 3.5 at nominal conditions (air 2 °C/water 35 °C) [13]. In the simulation model a scale factor has been used to size the heat pump, so that it covers the maximum space heating load plus an extra 0.5 kW for charging for DHW preparation. The heating capacity of the heat pump is 5.8 kW at design conditions (air -12 °C/water 35 °C, defrosting at -2 °C).

The heat pump is connected to the store so that it charges either the upper volume for DHW preparation or the middle volume for space heating system, a so called four pipe connection. The connection can be switched via two 3-way-valves, so to connect either the heat pump or the space heating loop to the middle port and the return to the lowest of the three ports. Thus, a parallel configuration is realized to connect the heat pump and the space heating loop to the store. When the store is charged for space heating, some part of the flow goes via the space heating distribution system and the rest through the store in the amount depending of the operating conditions.

The starts and stops of the heat pump are controlled based on the temperature difference between the return temperature and the buffer storage temperature. The heat pump starts when the storage temperature drops below the set point minus a hysteresis. During running time, the heating capacity is adapted in order to reach the set point temperature according to the heating curve.

In DHW mode the control principle is the same, with the exception that the compressor always runs at full speed during the whole charging process and the temperature sensor is located in the upper part of the store. A single sensor is used of on/off, with a hysteresis of 4 K.

3.2. Modeling

Type 832 QDT multinode model [14] was used for the collector with parameters derived for the Viessmann Vitosol 200 collector based on testing according to EN 12975. Type 340 mutliport model [15] was used for the store with parameters derived from a test of the Viessmann store 340 M according to En 12977. The heat pump was modeled using Type 877 [16], which is a relatively new semi-physical model based on a calculation of the thermodynamic refrigerant cycle and the thermal properties of the used refrigerant. The parameter values were derived from measurement data for the Viessmann Vitocal 200-S variable speed air heat pump provided by the manufacturer and then scaled so that the capacity for the theoretical heat pump was just sufficient to meet the load for the SFH45 building in Strasbourg. The scaling included size of heat exchanges and compressor and resulted in a heat pump model with a capacity of 5 kW at design conditions and 8.5 kW at standard conditions A2W35. For variable speed operation, the heat pump heating capacity is adapted by a PI-controller in order to reach the set point temperature (flow temperature according to heating curve) in the flow line of the heat pump.

The electricity use of the pumps is not derived from the component models but is calculated separately in a set of equations using a nominal power for set conditions of pressure drop and flow rate and corrections depending on the actual flow rate and the (linear) dependency of the efficiency on the flow rate. Nominal efficiencies of 12% for the solar loop pump and 40% for the pumps used for charging the store from the heat pump were used. A fixed power of 15 W was used for the space heat distribution pump. The pump for DHW preparation for the variation with external heat exchanger was not modeled as the usage time is so small, resulting in insignificant electricity use compared to the overall energy balance of the system.

Pipes connecting the collector to the store and between store and heat pump are modeled explicitly using Type 31. The dimension of the pipes in the collector circuit was defined according to prCEN 12977-2:2007(E) [17] depending on the flow rate, as is the insulation standard. The insulation standard of the other pipes is defined using the same standard, but the pipe diameter was chosen according to thumb rules used by plumbers. Pipe runs were estimated for a standard installation resulting in 30 m piping in the collector loop (internal diameter 16 mm) and a total of 22 m for all the other pipes (internal diameter 25 mm). The U-value for the pipes was calculated theoretically based on the insulation level but corrected for nominal extra heat losses due to pipe connections of 0.12 W/m^2K (equivalent to a bare copper pipe of 0.1 m with diameter 0.035 m) and 0.24 for a component such as a valve (equivalent to a bare copper pipe of 0.2 m with diameter 0.035 m). In total 18 connections and 10 components were included.

The total pipe runs as well as number of connections and components were assumed to be the same for the variations with three and with four pipe connection between heat pump and store. For the case with the external heat exchanger for DHW preparation, an extra pipe model was added in the warm side loop between store and heat pump to estimate heat losses through the heat exchanger. The U-value for this pipe was calculated based on assumptions of [4]. It was assumed flow through this pipe was only during discharging.

4. Systems variations

The influence of hydraulics between store and heat pump on seasonal performance factor was studied.
The reference system has been compared with three other solutions. These solutions were:

- Three pipe connection and internal heat exchanger (3P_int) ;
- Three pipe connection and external heat exchanger (3P_ext);
- Four pipe connection and external heat exchanger (4P_ext).

The reference system is with this labeling scheme 4P_int. All four solutions are shown in Fig. 2.
The heat pump is connected to the store via three pipes, which can be used like four connections. During the preparation of DHW, the heat pump is connected to the store on the top and on the middle of the tank, respectively for charging and discharging. In case of the so-called three pipes connection, the return flow of the heat pump comes from the bottom of the tank instead. Thus, the heat pump has to heat up most of the entire volume as the existing hot water in the DHW zone is pushed down into the SH zone, while with four pipes connection, only the upper (DHW zone) is heated.

4.1. Influence of DHW set point temperature

The idea was to study the influence of the set point temperature of DHW on yearly energy consumption for the system “4P_int”.
The approach followed has been to keep a constant set point for all DHW draw offs, a slight change from those used in IEA SHC Task 44 / HPP Annex 38 (T44A38) “Solar and Heat Pump Systems”, where a few draw offs had higher discharge temperatures than the normal 45ºC. Following set point temperatures have been chosen: 40°C, 45°C, 50°C, 55°C and 60°C. Discharge volumes were adjusted in order to have the same energy discharge for the discharge as of the nominal case, and thus the total annual DHW load is the same in all cases. For each value, the

lowest turn OFF charge temperature was adjusted so that the DHW penalty was 0.85% ± 0.05%. This ensures the same comfort level is provided by the system in terms of how the supplied temperature deviates from the set point temperature. Hysteresis for DHW charging has been defined in the way that the ON-charge temperature is the OFF-charge temperature minus 4°C.

4.2. Influence of the size of External Heat Exchanger

The idea was to study the influence of sizing the external heat exchanger keeping constant its U-value. The UA-value of the DHW heat exchanger for the base case was estimated for typical discharge conditions (5300 W/K for SWEP Heat Exchanger B25Tx40) [18]. Pressure drops have been neglected and pipes in primary loop were kept the same as in the reference system. The UA-value is a constant and not dependent on flow rate or temperature.

The approach was to introduce a size factor "x" to vary the heat transfer area of the heat exchanger. Following values for "x" have been selected: 0.5, 0.75,1, 1.25 and 1.5. Hence, four different heat transfer area have been investigated from the smallest (50% less than the reference) to the biggest (50% more than the reference).

For each value, the lowest turn OFF charge temperature was adjusted so that that the DHW penalty was 0.85% ± 0.05%. Hysteresis has been kept the same.

4.3. Influence of the size of Internal Heat Exchanger

The approach that has been used is the same as previously, but the investigation has been carried out for the system "4P_int".

The influence of the sensor positioning in the upper volume of the tank was also studied. The approach used has been to select different heights of the temperature sensor in the DHW zone. Values chosen are shown in Fig. 3.

For each value, the lowest OFF charge temperature was adjusted so that the DHW penalty was 0.85% ± 0.05%. Hysteresis for DHW charging has been kept the same.

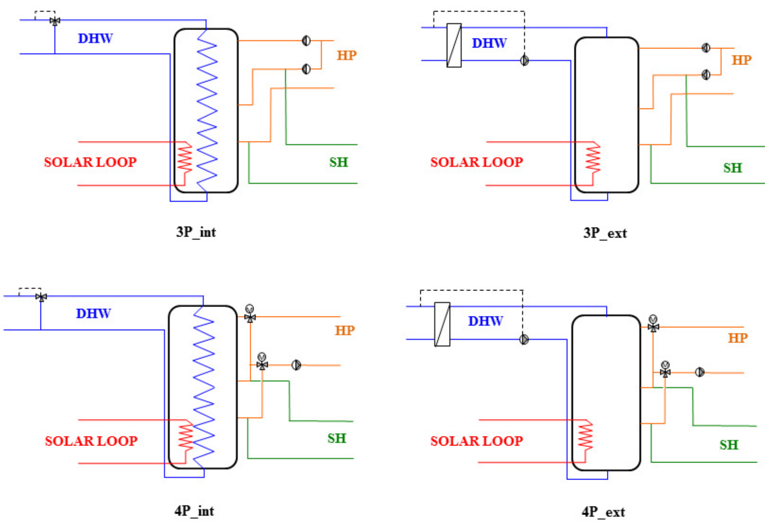

Fig. 2. System solutions with three pipe connection (top) and four pipe connection (bottom).

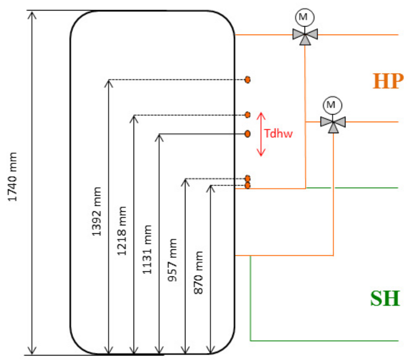

Fig. 3. System solutions with different positions for temperature sensor in the upper volume of the store

5. Systems variations

Fig. 4 shows both system results and the heat balances for the store in the reference system. The chart on the left shows that the heat pump compressor consumes ~2 MWh electric energy a year and 30% of the energy produced by the whole system results in thermal losses and auxiliary losses. The chart on the right shows that 54% of the total energy of the store is covered by solar and the thermal losses are 21% of the total heat balance for the store. Note that a significant part of the space heating load is delivered in variable speed mode directly from the heat pump and is thus not included in the heat balance for the store.

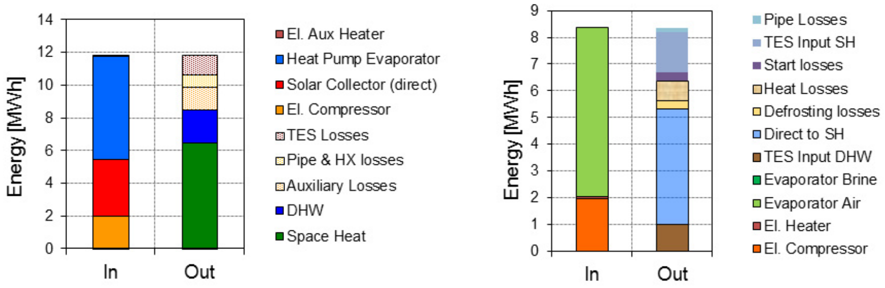

Fig. 4. Overall system results (left) and heat balance for the store (right) of the reference system

Fig. 5 shows the electricity use in the reference system. Compressor uses 77% whereas fans and controllers use almost the same amount, 9% and 7% respectively. Circulating pumps and auxiliary (electrical heater) use 7%.

The reference system has an $SPF_{SHP+,pen}$ of 3.26 and an annual electricity consume of 2.60 MWh. DHW penalties are 0.82% and 0.2% for space heating, relative to the total load for DHW and space heating respectively.

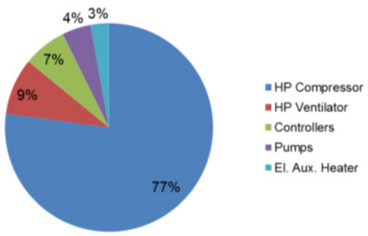

Fig. 5 Electricity use in the reference system

Fig. 6 shows results for the solutions shown in Fig. 2. “4P_ext” has 2% higher $SPF_{SHP+,pen}$ than “4P_int” and almost 8.7% higher than “3P_ext”. These results do not include heat losses from the external heat exchanger. A separate calculation for the base case values that included the heat losses from the external DHW heat exchanger showed a reduction in $SPF_{SHP+,pen}$ of 1 % (relative) based on UA-values from [4] and the assumption that flow on the store side is only during DHW discharge.

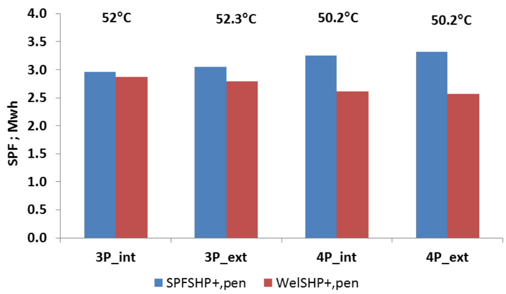

Fig. 6 Results of performance modelling for ASHP: $SPF_{SHP+,pen}$ (blue columns), $Wel_{SHP+,pen}$ (red columns) for DHW preparation

Fig. 7 shows the influence of DHW set temperature on both $SPF_{SHP+,pen}$ and $Wel_{SHP+,pen}$. $SPF_{SHP+,pen}$, which is plotted with blue diamonds, decreases from 3.52 at 40°C to 2.69 at 60°C. $Wel_{SHP+,pen}$, which is plotted with red squares, increases from 2.4 MWh at 40°C to 3.2 MWh at 60°C. The set temperature for charging the DHW zone in the store is indicated above each point, as it is in all similar figures. This value changes due to the methodology used, which ensures a DHW penalty of 0.85%.

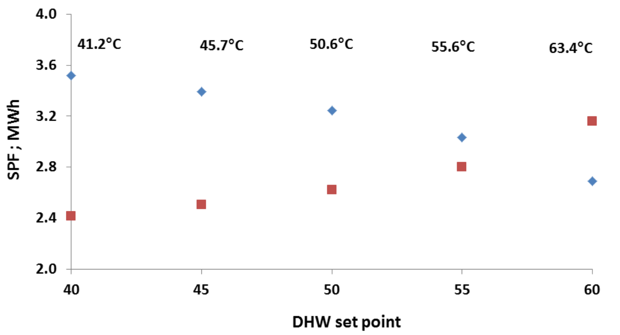

Fig. 7 Results of performance modelling for ASHP: $SPF_{SHP+,pen}$ (red square), $Wel_{SHP+,pen}$ (blue diamond) vs. temperature charging for DHW preparation

Fig. 8 (left and right) shows the influence of the area of the heat exchanger respectively for the external and the internal solution. In the first case, the best $SPF_{SHP+,pen}$ is achieved with the same value as the reference. In the second case, the best $SPF_{SHP+,pen}$ is achieved with 50% bigger area, although the difference is very small compared to the reference case.

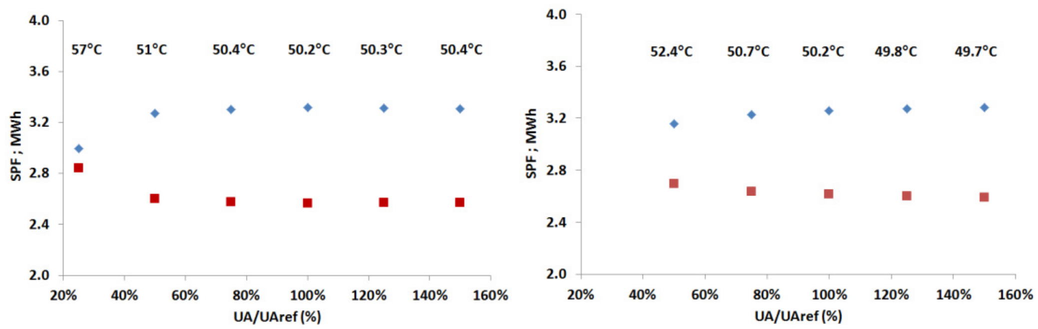

Fig. 8 (left) and (right) Results of performance modelling for ASHP: $SPF_{SHP+,pen}$ (blue diamond), $Wel_{SHP+,pen}$ (red square) as function of the increase of surface area for DHW HX: solution with external HX (left) and internal HX (right).

Fig. 9 shows results for different sensor positions. $SPF_{SHP+,pen}$ and $Wel_{SHP+,pen}$ are essentially constant above a height of 1.1 m.

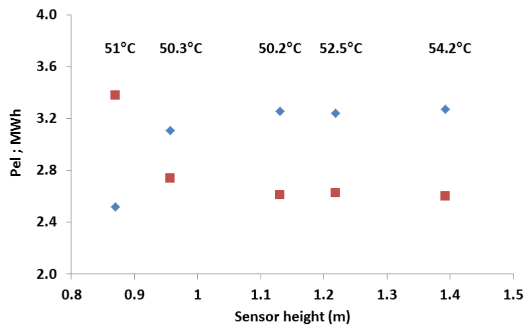

Fig. 9 Results of performance modelling for ASHP: $SPF_{SHP+,pen}$ (blue diamond), $Wel_{SHP+,pen}$ (red square) vs. height of DHW sensor

6. Discussion and Conclusions

The influence of hydraulics and control of thermal storage in a solar assisted heat pump combisystem has been analyzed. Single family house with a specific space heating load of 45 (kWh/m^2 y) for Strasbourg climate has been chosen as target building. Solution with internal heat exchanger and 4 pipes connections (4P_int) has been chosen as reference system because suggested and promoted by leading companies in the heating sector in Europe.

The reference system has 3.26 $SPF_{SHP+,pen}$ and consumes yearly 2.60 MWh electric energy, whereof more than 80% of it is used to run the compressor and the ventilator of the heat pump. Furthermore, 30% of the thermal energy produced by the complete system is wasted due to thermal and auxiliary losses. Thus, different system variations have been investigated and important results are highlighted and discussed:

1) Decreasing the set temperature reduces heat losses in the store, the pressure ratio in the heat pump (thus increasing COP) and allows solar to replace auxiliary heat at a lower temperature. Reducing by 5°C the DHW set point permits to decrease also the temperature at which the heat pump charge the store. Solution with 40°C DHW set and 41.2°C turn-OFF temperature achieves 3.52 $SPF_{SHP+,pen}$. Note that this result was achieved with the assumption that all DHW discharges were at 40ºC. In the T44A38 DHW profile some discharges are at 55ºC and thus no reduction of the charge temperature is possible.
2) The use of a fixed DHW penalty allows comparison for the same level of DHW comfort in all cases. However, the DHW charge turn-OFF temperature is always below the 55ºC supply temperature required by the DHW load profile in T44A38. Thus it is these high temperature discharges that result in a large

part of the penalty. A lower allowed value for the penalties would result in higher turn-OFF temperatures and higher electricity use in all cases.

3) Hydraulics between heat pump and store also affect the system performance. Solution with four pipes connections gives 9% better $SPF_{SHP+,pen}$ compared to the solution with three pipes because less volume of the store has to heat up during charge for DHW preparation. Solution with external heat exchanger for DHW preparation gives 2% better $SPF_{SHP+,pen}$ due to better stratification. Heat losses through the heat exchanger were not included in the results in the figures, but they were shown to not have a large impact on the results for the assumptions used. The effect of fluid streams at inlets has not been modeled.
4) DHW sensor height influences the performance of the system. If it is placed too low, then there is a negative interaction between the DHW and SH zones. This result is in agreement with those shown in [6] where a minimum distance of 20 cm is recommended.

Acknowledgements

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