

Simulation study of cascade heat pump for solar combisystems

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Abstract

This paper focuses on the study of cascade heat pump systems in combination with solar thermal for the production of hot water and space heating in single family houses with relatively high heating demand. The system concept was developed by Ratiotherm GmbH and simulated with TRNSYS 17. The basic cascade system uses the heat pump and solar collectors in parallel operation while a further development is the inclusion of an intermediate store that enables the possibility of serial/parallel operation and the use of low temperature solar heat. Parametric studies in terms of compressor size, refrigerant pair and size of intermediate heat exchanger were carried out for the optimization of the basic system. The system configurations were simulated for the complete year and compared to a reference of a solar thermal system combined with an air source heat pump. The results show ~13% savings in electricity use for all three cascade systems compared to the reference. However, the complexity of the systems is different and thus higher capital costs are expected.

“Key-words: cascade heat pump; solar combisystems; simulation study”

1. Introduction

The use of solar thermal combined with heat pumps for the production of domestic hot water (DHW) and space heating (SH) has recently increased in central Europe. The boom is related to higher efficiency of heat pumps compared to conventional heating systems and incentives due to high peak costs of electricity and environmental reasons. However, the coefficient of performance (COP) of a heat pump drops significantly when high pressure ratios occur, e.g. with large temperature lift between source and sink. Thus, it is possible to improve the system's performance by adopting modified refrigeration cycles (Pottker, 2013), e.g. a dual stage heat pump configured in cascade (Wang et al., 2005; Wu et al., 2012). The cascade system consists of two vapor-compression cycles connected via an intermediate heat exchanger (IHX). This solution gives more stable heating operations and higher water outlet temperatures (Jung et al., 2013), thus is suitable for applications with high supply temperatures. The main disadvantage is the limited range of working conditions due to the simultaneous running of the heat pumps. Therefore, by replacing the IHX with an intermediate tank, the system acquires flexibility meaning that three configurations can be realized: 1) the low stage heat pump charges the intermediate store; 2) the high stage heat pump discharges the intermediate store; 3) both heat pumps operate as a cascade via the intermediate heat exchanger. Moreover, the solar collector can provide heat at a low temperature that cannot be used in the combistore to this intermediate store. This connection is called solar indirect mode, while the solar direct mode occurs when the solar collector charges the combistore. The use of indirect mode may increase the coefficient of performance (COP) of the heat pump and the efficiency of solar collectors at the same time, but this is not a guarantee for a higher system performance. Haller and Frank (2011) studied the potential of using heat from solar thermal collectors for heat pump evaporator and one of the conclusions was that the indirect solar mode had a higher potential for improving the system performance for systems with large temperature lift between source and sink. The company Ratiotherm (Heizung + Solartechnik GmbH & Co. KG, Germany), has developed a system concept based on a cascade heat pump. The main focus is on the use with existing buildings where

heat loads can be relatively high.

The aim of the study was to analyze possible configurations of cascade heat pump in combination with solar thermal system for SH and DHW preparation in single family houses with relatively high heating demand. The scope was to compare three different configurations of cascade heat pump to the solution with a single stage. The solutions included an intermediate heat exchanger (CHP), an intermediate storage (CHPS) and an indirect solar mode (CHPSS). In addition the influence of the compressor size, the choice of refrigerant and the size of the intermediate heat exchanger had been investigated. Costs of components were not considered. Thus, annual electricity consumption and seasonal performance factor (SPF) were used for the comparison. The study has been carried out within the frame of the European Union's Seventh Framework Program FP7/2007-2011 in a project called MacSheep.

2. Methodology

This study has been carried out with TRNSYS 17 (Klein et al., 2011). The reference model was the same as in Poppi and Bales (2013), but with different boundary conditions. The climate of Zurich and building SFH100 (Dott et al., 2013; Haller et al. 2013) were chosen for the study. Some key figures for these boundary conditions are given in Table 1.

Tab. 1: Key figures for SH and DHW for SFH100 and the climate of Zurich

Building	Unit	SFH100
Design supply temperature for SH	°C	55
Design return temperature for SH	°C	45
Design load for heat distribution	kW	7.7
Annual space heating load	kWh	17224
Supply temperature for DHW	°C	45
Cold water temperature for DHW	°C	10
Annual DHW discharge energy	kWh	3038

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 functions were defined as in Haller (2013) and in addition were kept below the value of 1% for all simulations to ensure that all systems provided the same comfort level as well supplied energy. Thus, seasonal performance factor ($SPF_{SHP+,pen}$) and total electricity use ($W_{el,SHP+,pen}$) were defined as following:

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

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

An air source heat pump combisystem, defined within MacSheep (MacSheep, 2012-2015), was considered as reference. From the reference system, the single stage heat pump has been replaced by a cascade heat pump cycle (CHP). Then, parametric studies in terms of compressor size, refrigerant pair and size of intermediate heat exchanger were carried out. The optimized CHP was compared to other configurations with the same optimized cascade heat pump: cascade heat pump system with integrated storage (CHPS) and cascade heat pump system with integrated storage and indirect solar mode solar (CHPSS). All solutions were simulated for the complete year.

3. Reference system

3.1. Description

A schematic of the reference system is shown in Fig.1. The system layout is the same as the one described in Poppi and Bales(2013). 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 is 9.28 m². The 750 liters water store 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). 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 on 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 for on/off control, with a hysteresis of 4 K.

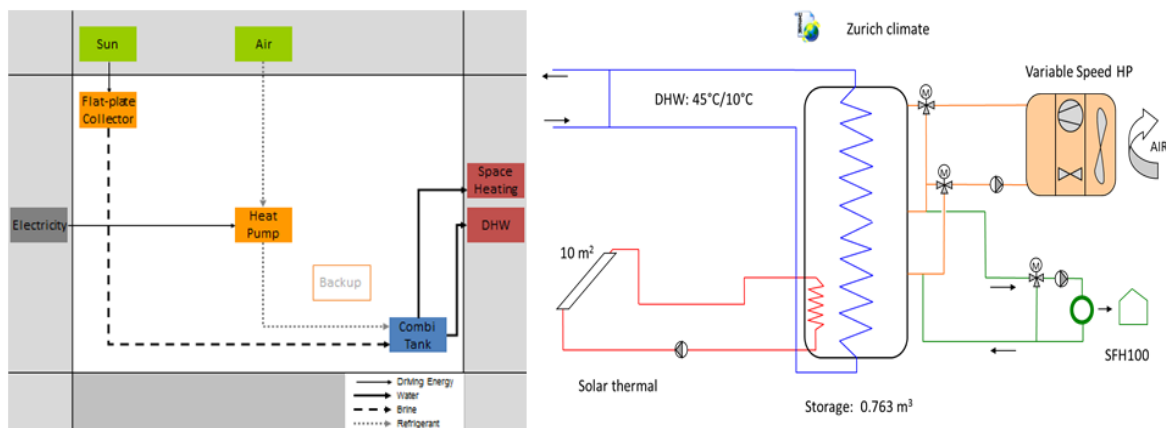


Fig. 1: Square view diagram (left) and scheme of the reference system (right)

3.2. Modeling

Type 832 QDT multinode model (Haller et al., 2012) was used for the collector with parameters derived for the Viessmann Vitosol 200 collector based on testing according to EN 12975-2 (2006). Type 340 multiport model (Drück and Pauschinger, 2006) was used for the store with parameters derived from a test of the Viessmann store 340 M according to EN 12977-2 (2012). The heat pump was modeled using Type 877 (Heinz and Haller, 2012), 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 heat pump can cover the maximum heat load plus an extra 0.5 kW for charging for DHW preparation. The heating capacity of the heat pump is 14.5 kW at standard conditions (A2W35, defrosting at -2 °C). 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 water pumps was not derived from the component models but was 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. Pipes connecting the collector to the store and between store and heat pump were modeled explicitly using Type 31. The dimension of the pipes in the collector circuit was defined according to prCEN 12977-2:2007(E) (CEN, 2007) depending on the flow rate, as is the insulation standard. The insulation standard of the other pipes was 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/m²K (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.

4. Cascade systems

4.1 Solution with intermediate heat exchanger (CHP)

The system concept consists of a cascade heat pump (CHP) coupled to the combistore that replaces the reference air source heat pump. As Type 877 has a heat exchanger for the condenser that models a single phase fluid on the sink side, the cascade heat pump was modelled with two Type 877 in series connected via a water loop. Fig. 2 shows the schematic of the two stage heat pump with the intermediate storage. The high stage (Type 877_WSHP) is coupled to the sink and the low stage (Type 877_ASHP) to the source bypassing the store via two 3-way-valves when system runs in cascade (refer to outer boundaries).

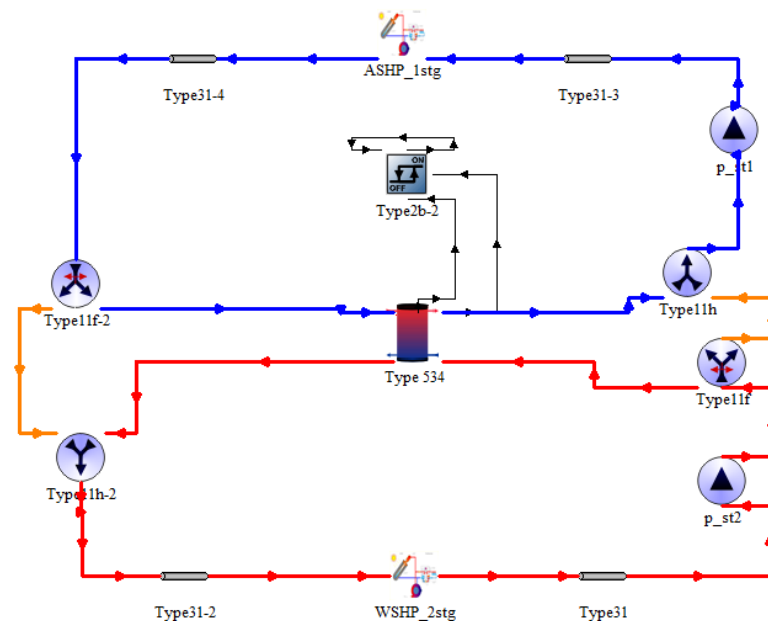


Fig. 2: Schematic of the two stage heat pump with intermediate storage in TRNSYS Studio

The heat losses through the pipes were defined negligible (1 kJ/(h·m²·K)) and the energy consumption for recirculation set to zero. The mass flow rate was 700 kg/h. The 3-way-valves were simulated by using two diverter valves (standard Type 11f) and two mixer valves (standard Type 11h). The main inputs and parameters of the heat pump models were kept the same as reference except for the compressor size, the refrigerant and the UA-value of the heat exchangers. Their optimization was achieved thanks to parametric studies. The heat losses of the compressor and the isentropic efficiency were modeled as function of the compressor frequency and the pressure ratio, similarly to the reference system.

Parametric studies

The compressors size for the refrigerant pairs: R410A/R134a, R407C/R134a, R410A/R404A, R290/R134a was optimized; the first termed refrigerant refers to the high stage and the second one refers to the low stage. These refrigerant pairs were chosen based on results from literature. The initial assumptions for the size of compressor were 4 and 7 m³/h respectively for low and high stage heat pump. Once the best combination of refrigerant pair and compressor size was defined, the optimal heat transfer coefficient for the intermediate HX was studied. The UA-values chosen for the parametric study were: 350, 700, 1050, 1400, 1750, 3500, 5250 and 7000 W/K (default value). $SPF_{SHP+,pen}$ and $Wel_{SHP+,pen}$ were used for the results comparison. Moreover, penalties were kept lower than 1% to ensure the same comfort level as well supplied energy.

4.2 Solution with intermediate storage (CHPS)

The system concept is based on the CHP with the addition of the integrated storage between the two heat

pump cycles (Fig. 3). The schematic is still the one shown of Fig. 2, but the two cycles are now decoupled. For modeling the intermediate storage, the nonstandard Type 534 was used. Details about the model can be found in Thornton et al. (2005). The store was via direct fluid pipes connected to the condenser of the low stage and to the evaporator of the high stage. The store was simulated as a fully mixed tank and no internal heat exchanger were considered. Thereby the store gets charged by the low stage and discharged by the high stage heat pump. The high stage has the same control function as for CHP. The low stage heat pump runs if the storage temperature is below a set temperature minus a dead band temperature of 5K until the set temperature is reached. This is controlled by the standard Type 2b.

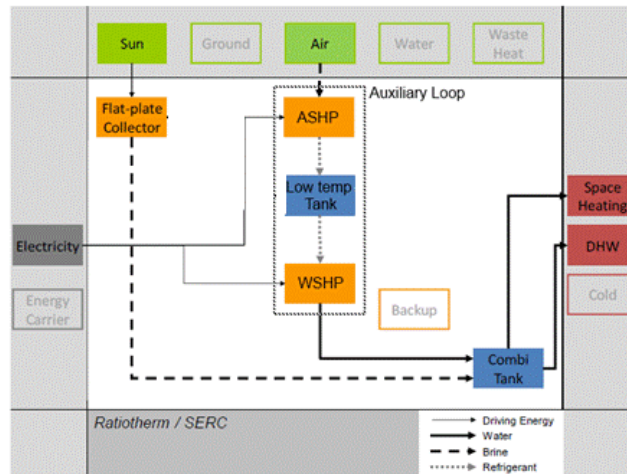


Fig. 3: Square view diagram of the CHPS

4.3 Solution with intermediate storage and indirect solar mode (CHPSS)

In CHPSS the main solar collector loop was also connected to the intermediate storage (Fig. 4) so that the system can use low temperature solar heat (indirect mode).

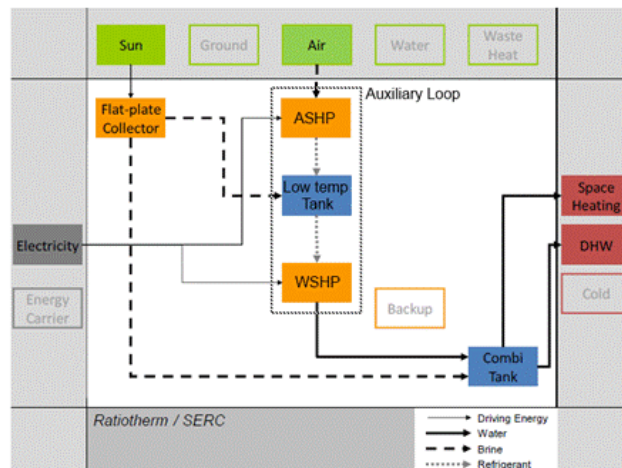


Fig. 4: Square view diagram of the CHPSS

The fluid flow switches between the combistore and the integrated store thanks to a 3-way-valve. The solar collector was coupled with the cold store by using a doubleport to avoid the heat resistance of a heat exchanger. Therefore, the intermediate store was filled with the collector fluid and the fluid loop changed to the glycol/water-mixture. The control strategy of the solar loop was changed accordingly to provide both indirect and direct modes. If the outlet temperature of the solar collector is higher than the set temperature of the intermediate storage (14°C) plus a dead band of 7°C the collectors charged the cold storage; the indirect mode is active as long as the collector outlet temperature is above the storage temperature plus 3°C . However, at the time when the solar collector can provide the combistore, this is the preferred mode of collector operation as suggested by Haller and Frank (2011). In order to set the priority to charge the combistore first, the integrated store is just provided with solar heat when the combistore solar control is

switched off.

5. Results and discussion

5.1 Reference system

Fig. 5 shows the annual electricity use of the compressor as a function of the outside ambient temperature (T_{amb}) and the inlet temperature to the condenser. The vast majority (84%) of the use of the heat pump occurs when the ambient temperature is between -5 and +5 °C and that this coincides mostly with inlet temperatures of 42 to 50 °C (71%). There is also a significant amount supplied at lower temperatures, 16% being used for inlet temperatures between 30 and 37 °C.

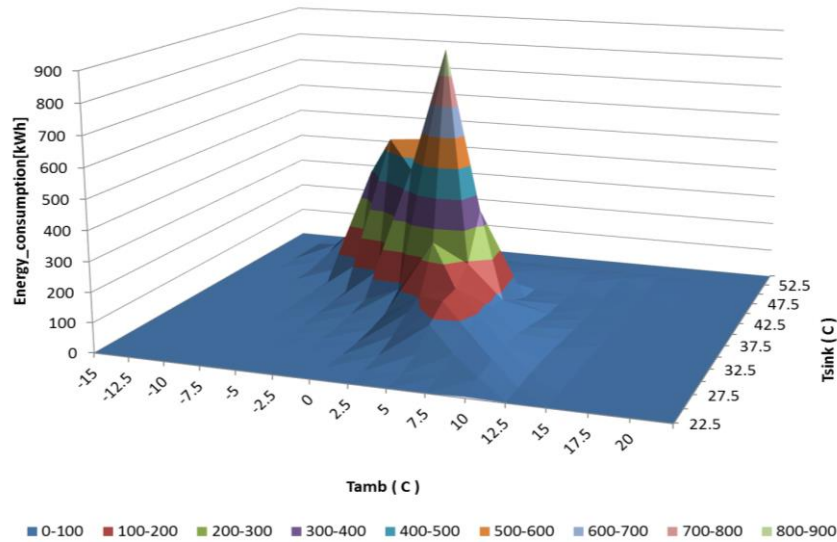


Fig. 5: Electricity use of the compressor in the reference system as a function of ambient temperature and inlet temperature to the condenser of the heat pump

Fig 6 shows the total electricity use, whereof 86% of the total share is consumed by the compressor and 8% by fans. Circulating pumps, electrical auxiliary heater and controllers use 2% each. The reference system has an $SPF_{SHP+pen}$ of 2.43 and an annual electricity consume of 8.34 MWh. DHW penalties are 0.2% and 0.0% for space heating, relative to the total load for DHW and space heating respectively.

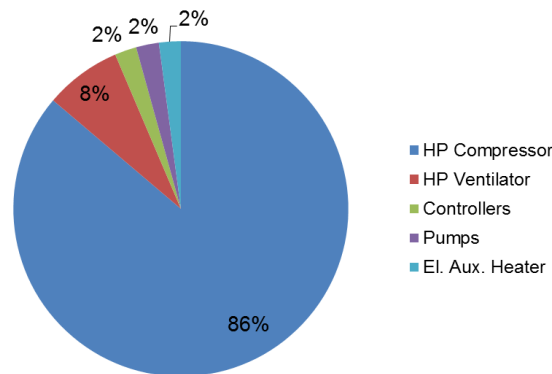


Fig. 6: Electricity use of the reference system

5.2 Parametric studies of CHP

Size of compressor

Results for R410A/ R134a are shown in Fig. 7. The blue line shows the function profile of $Wel_{SHP+pen}$ for different sizes of the compressors. The function has a minimum in correspondence of $V_{swept,ASHP} = 4 \text{ m}^3/\text{h}$. Vice versa for the $SPF_{SHP+pen}$ function, which is depicted in green. Red dots show the results with penalties higher than 1%.

For the case of the high stage, both $Wel_{SHP+pen}$ and $SPF_{SHP+pen}$ (orange and violet respectively) have a flatter profile. The best compressor combination with the refrigerant pair is $4 \text{ m}^3/\text{h}$ and $7 \text{ m}^3/\text{h}$ respectively for the

low and the high stage. In this configuration, the $SPF_{SHP+pen}$ is 2.76 and the $W_{el,SHP+pen}$ is 7.32 MWh.

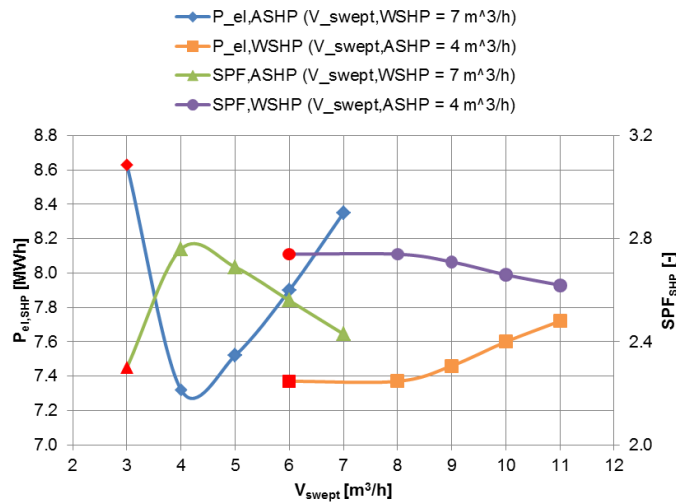


Fig. 7: Results of the parametric study with the compressor size for the refrigerant pair R410A/R134a. Red points show the results with penalties higher than 1%.

Refrigerant pair

The compressor sizing was done for each refrigerant pair. Table 2 shows the results for the optimized sizes of compressors for different refrigerant pairs. Best result is achieved with R410A/R134a. However, the results of R290/R134a are very close to the best, but with a bigger compressor size for the low stage cycle. Furthermore, this solution uses propane (R290) in the low stage, which is flammable; thus not suitable for domestic applications. The solution with the refrigerant pair R410A/R404A employees the smallest compressor sizes, but lower $SPF_{SHP+pen}$ compared to the best solution.

Table 1: Best results for the compressor size study with the different refrigerant pairs

	Unit	R410A/ R134a	R407C/ R134a	R410A/ R404A	R290/ R134a
$\dot{V}_{swept,ASHP}$	$[m^3/h]$	4	7	4	7
$\dot{V}_{swept,WSHP}$	$[m^3/h]$	7	7	4	7
SPF_{SHP}	$[-]$	2.76	2.71	2.64	2.75
$W_{el,SHP}$	$[MWh]$	7.32	7.47	7.64	7.34

Size of intermediate heat exchanger

Fig. 8 shows the results of the parametric study of the size of the intermediate HX.

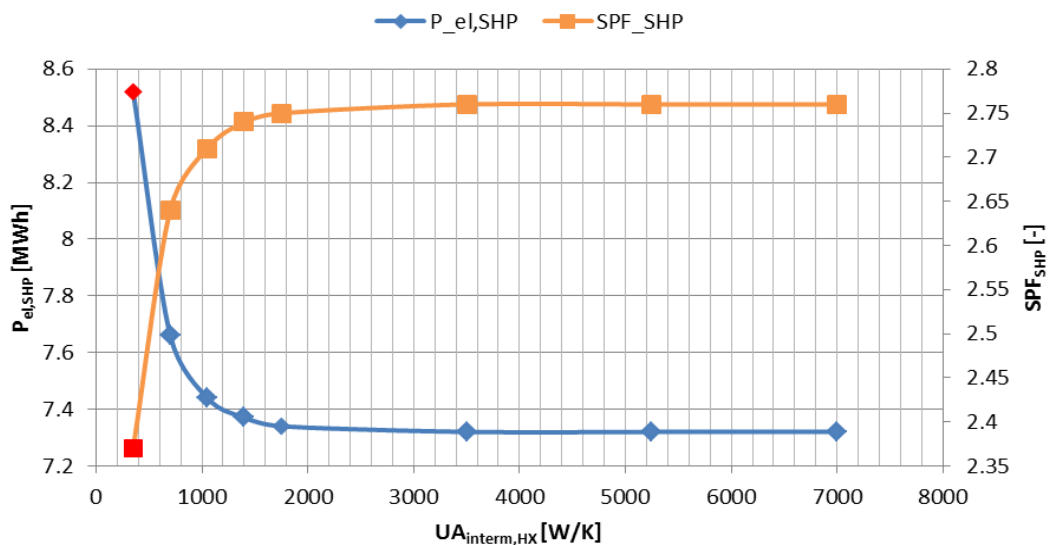


Fig. 8: Results of the parametric study with UA-value of the intermediate HX. Red points show the results with penalty higher than 1%.

In the beginning, the energy consumption of the system quickly decreases with an increasing UA-value, which then approached a minimum. The $SPF_{SHP+,pen}$ passes in the opposite way, it rises at first with increasing UA-value, which then approaches a maximum. In order to optimize the heat exchanger size, the UA-value should be chosen as small as possible while the energy consumption of the system should be near the minimum and the $SPF_{SHP+,pen}$ near the maximum. The solution with optimized UA-value of 1750 W/K has a $SPF_{SHP+,pen}$ of 2.75 and a $Wel_{SHP+,pen}$ of 7.34 MWh.

5.3 Comparison of the systems

Fig. 9 shows the results for one year of the four systems. Blue and red columns represent $SPF_{SHP+,pen}$ and $Wel_{SHP+,pen}$ respectively, while the green dots depict the reduction of electricity use compare to the reference case. All three cascade systems perform equally better than the reference. They have ~13% higher SPF and 12% lower electricity consumption. Thus, the main difference between the three cascade systems and the reference constitutes the cascade heat pump.

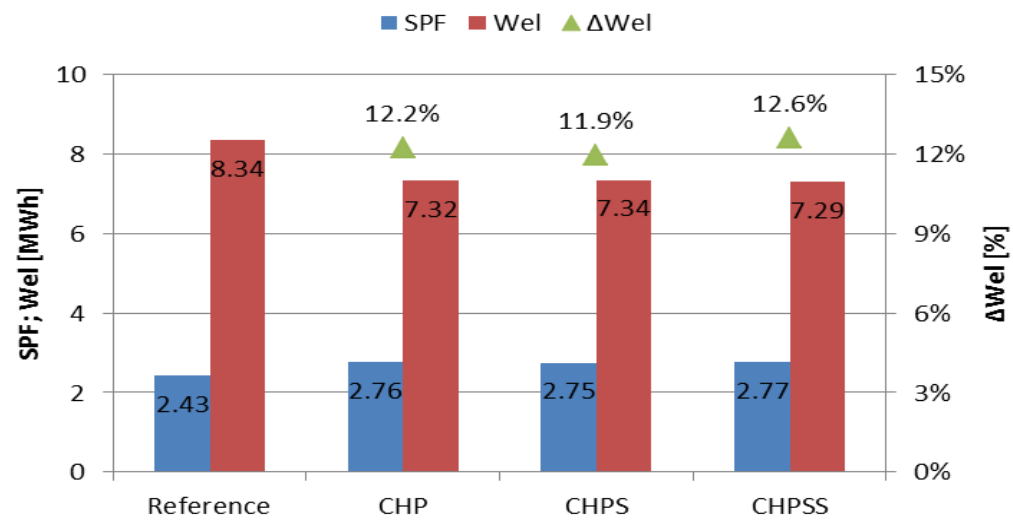


Fig. 9: Comparison of the SPF and electrical use of the systems

Fig. 10 shows the comparison of the energy balances, the condensation energy and the energy losses of the heat pump for the four systems. Losses are defined as percentage of the total energies out from the heat pump and count defrosting, starting and heat losses of compressors, pipes and intermediate storage to environment.

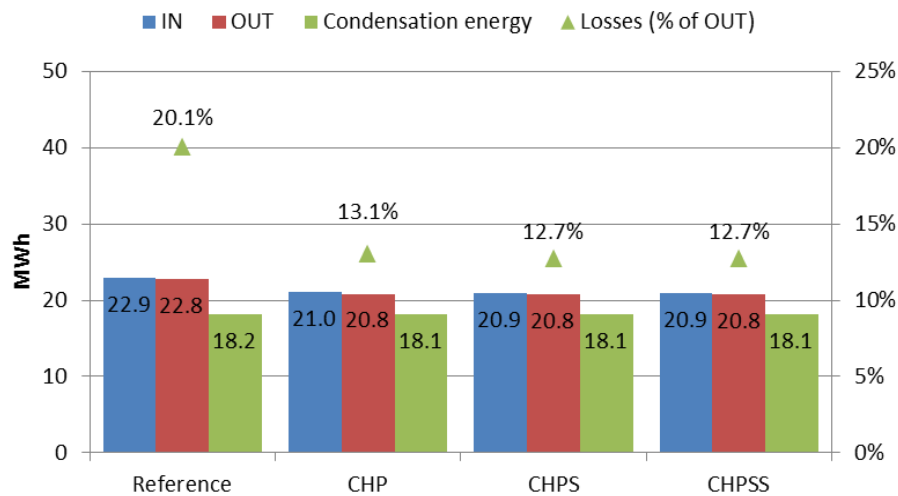


Fig. 10: Comparison of the energy balances and the energy losses of the heat pumps

The advantage of the cascade heat pump system is expressed mainly in the lower total pressure ratio and the

lower condensation temperature (discharge temperature of the high stage HP) that provides the SH and DHW. As a result the heat losses, which are a function of the pressure ratio and the difference between compressor and ambient temperature, are 2 MWh smaller. Furthermore, refrigerants in the cascade heat pump are individually adapted to the temperature levels in the auxiliary loop. Thus the refrigerant effect is greater and the heat pump operating at their individual optimized conditions. For example, this effect is also documented in Chua et al. (2010) and Wang et al. (2005). Consequently, the auxiliary loop of the reference system has a 1 MWh higher energy consumption compared to the cascade solution to provide the same condensation energy of around 18 MWh to SH and DHW. The compression efficiency for each stage is higher.

The comparison of the solar energy amounts of the systems for one year reveal that the indirect solar energy amount of 0.17 MWh is small compared to the direct solar energy amount of 3.49 MWh. Moreover, the indirect solar mode occurs just around 66 hours in the year. Consequently the indirect solar mode in the integrated storage has no big effect on the system performance.

The system concept based on the CHPSS has the highest $SPF_{SHP+pen}$ and lowest $Wel_{SHP+pen}$, but this solution is the most complex. The use with the intermediate storage implies higher capital costs and a complex control strategy for running the heat pumps in three different configurations. Furthermore, an extra sensor for the solar radiation is needed to calculate the outlet temperature of the collector when the solar indirect mode is on. This involves more cost for the radiation sensor and in the control, because of inaccuracies in the measurement. However, also using temperature sensors for a control strategy induce inaccuracies. The system concept based on CHP has 0.4% higher electrical consumption compared to CHPSS, but it has lower system complexity (the intermediate heat exchanger replaces the storage tank) and simpler control strategy. The main disadvantage is the limited range of working conditions due to the simultaneous running of the heat pumps.

6. Conclusion

The analysis of possible configurations of cascade heat pump for solar thermal combisystems has been carried out. Single family house with a specific space heating load of 123 (kWh/(m²·y)) for Zurich climate has been chosen as reference for the systems comparison. The solution with an air source heat pump in parallel with the solar collectors has been chosen as reference system because suggested and promoted by leading companies in the heating sector in Europe. The reference system has a $SPF_{SHP+pen}$ of 2.43 and an annual electricity consumption of 8.34 MWh, whereof most of it is used to run the compressor and ventilator of the heat pump. Three different heat pump configurations have been studied and compared to the reference: 1) cascade heat pump (CHP), cascade heat pump with an intermediate storage (CHPS), cascade heat pump with an intermediate storage and an indirect solar mode (CHPSS). The cascade system was optimized in terms of size of compressors, refrigerant pair and size of the intermediate heat exchanger for the CHP configuration. The solution with integrated storage and indirect solar mode achieves the best results with 12.6% of savings in total electric use. However, all three cascade systems perform more or less equally better than the reference, but the system complexity is different, thus capital costs are expected to be higher.

7. Acknowledgements

The research leading to these results has received funding from the European Union's Seventh Framework Program FP7/2007-2011 under grant agreement n° 282825 – Acronym MacSheep.

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