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Heat pump system with uncovered and free ventilated covered collectors in combination with a small ice storage

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Abstract

In the past years research and development for heat pumps in combination with solar thermal collectors for preparation of domestic hot water and space heating increased. In this work it is analyzed how low solar irradiation and ambient air can be used as heat sources. Three different systems were modeled and simulated in TRNSYS. One of the systems is a basic parallel solar and air source heat pump combination which is sold today on the market. The other two systems use solar collectors in combination with an ice storage as the only heat source of the heat pump. One of the variants uses unglazed selective coated absorbers and the other variant uses covered collectors with controlled natural ventilation on both sides of the absorber. All systems have been sized to have the same costs for the end consumer. The heat load corresponded to a single family house that was simulated in six different climates. The simulation results show that unglazed collectors and a brine heat pump in combination with a 400 liter ice storage can reach a better performance than the reference. The natural ventilated collector didn't show significant advantages compared to the unglazed collectors or the reference system.

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Keywords: Selective Unglazed Collectors; Heat Pump; Natural Convection; Ice Storage; Serial Solar and Heat Pump Combination

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1. Introduction

In Europe, heating systems based on heat pumps are very popular for new buildings but also for the replacement of old heating systems. However, the application of the two most common heat pump types – air source and ground source – is subject to a number of restrictions. For example at some places the drilling of boreholes is forbidden and an air source heat pump can be unfavorable because of noise emissions. In such situations, a brine source heat pump can be combined with uncovered collectors in combination with a small ice storage. In this case, the use of uncovered collectors in comparison to covered collectors is advantageous because the ambient air can be used as a heat source when there is no or only low solar irradiation. A disadvantage of the uncovered collectors is that the direct heat contribution at higher temperatures is significantly lower. To solve this problem, a new collector design was investigated in this work, which combines the advantages of covered and uncovered collectors. Keller et al. [1] and the system SOLAERA, developed by Consolar [2] used forced convection between the absorber and the glazing of covered collectors in order to use the ambient air as a heat source if the collector was operated below the ambient air temperature. In this work we are presenting simulation results for a covered collector with natural ventilation. The collector-design is similar to a standard glazed collector but includes the possibility to use controllable natural ventilation between absorber and insulation as well as between absorber and glazing (Figure 3).

In this paper, three systems are compared with each other: A reference air source solar heat pump system, a system which uses a small ice storage and unglazed collectors as only heat source (**Unglazed**) and a system which uses the new concept of natural ventilated collectors in combination with a small ice storage as only heat source for the heat pump (**Ventilated**).

Nomenclature

a_1	Linear collector heat loss coefficient	[W/(mK)]
a_2	Quadratic collector heat loss coefficient	[W/(m^{2}K)]
A2W35	Air Temperature 2°C, Water Temperature 35°C	
B0W35	Brine Temperature 0°C, Water Temperature 35°C	
COP	Coefficient of Performance	[-]
DHW	Domestic Hot Water	
EES	Engineering Equation Solver	
P	Power	[W]
$\dot{Q}$	Heat transfer rate	[W]
SFH	Single Family House	
SPF	Seasonal Performance Factor	[-]
T44A38	IEA SHC Task 44/HPP Annex38	
U	Heat transfer coefficient	[W/(m^{2}K)]
UA	Heat transfer coefficient – area product	[W/K]
W	Work	[MWh]
η_0	Collector optical efficiency	[-]

Subscript

el	Electricity
HP	Heat Pump
PCM	Phase Change Material (Ice Storage)
pen	Penalties
SH	Space Heating
SHP	Solar and Heat Pump

2. Methods

2.1. General information

All simulations were performed with TRNSYS 17. The boundary conditions were based on the IEA SHC Task 44/ HPP Annex38 (T44A38) [3]. These boundary conditions were slightly adapted in order to include different climates and a more realistic DHW profile, which was calculated according to Jordan & Vajen [4]. Detailed information can be found in Mojic et al. [5]. Three different buildings were simulated according to the building definitions of T44A38. For the reference climate of Strasbourg their space heat demand is 15 kWh/(m^{2}a) (SFH15), 45 kWh/(m^{2}a) (SFH45), and 100 kWh/(m^{2}a), respectively. SFH45 was simulated for all climates, for Zurich SFH15 was simulated in addition, and for Carcassonne SFH100 was simulated in addition. Table 1 summarizes the climates, the corresponding heat loads and the domestic hot water demands.

Table 1: Summary of the simulated climates and their corresponding space heating and domestic hot water demand.

Location, Country Code	Space Heating Demand [kWh/(m^{2}a)]	Domestic Hot Water Demand [kWh/a]	Mean ambient Temperature [°C]
Zurich, SFH 45, CH	56.4	3038	9.1
Zurich, SFH 15, CH	21.6	3038	9.1
Wurzburg, DE	57.9	3038	9.1
Helsinki, FIN	93.3	3343	5.5
Carcassonne, SFH45, FR	23.2	2691	13.2
Carcassonne, SFH100, FR	61.6	2691	13.2
Davos, CH	79.6	3571	2.8
Graz, A	46.3	2913	10.7

2.2. Reference system

As a reference a system was chosen that is currently available on the market with the difference that the simulated one has an external DHW module. Figure 1 shows its hydraulic design. The collector field consist of 10 m^2 standard glazed flat plate collectors (a_1 = 3.95 W/(m^{2}K), a_2 = 0.0122 W/(m^{2}K^2), η_0 = 0.793, based on aperture area) with an inclination of 45°, orientated to the south. The heat storage (750 liters water) is equipped with an internal coil heat exchanger for the solar input (U = 312 W/(m^{2}K)). For the domestic hot water supply an external heat exchanger is used, which is simulated without heat losses (UA = 5333 W/K). The reference system includes an air-source heat pump that was simulated with Type 877[6] and had a thermal power of 8 kW and a COP of 3.5 at A2W35. The heat distribution system (floor heating for SFH15 and SFH45) was simulated with design flow and return temperatures of 35 °C / 30 °C (40°C/35°C for Davos and Helsinki). For Carcassonne SFH100 additional simulations were done with floor heating (35/30 °C) and radiator heating (55/45 °C).

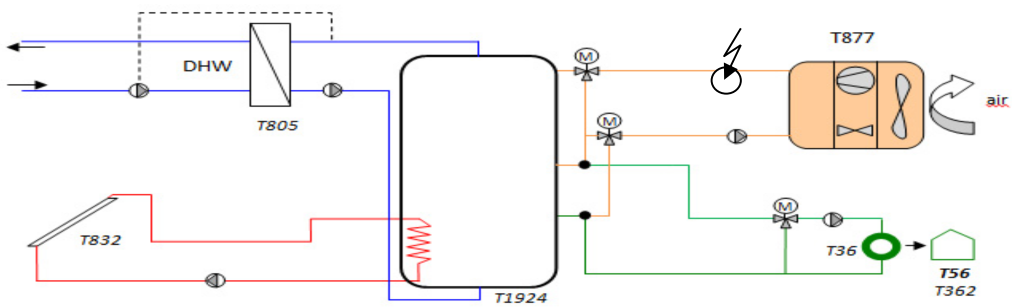


Figure 1: Simplified hydraulic scheme of the reference system; Txy = TRNSYS Type number xy

2.3. Alternative systems

The alternative systems (***Unglazed*** and ***Ventilated***) use different collectors, a different heat pump ('brine' source) and additionally an ice storage. The corresponding hydraulic scheme can be seen in Figure 2. The air source heat pump is replaced by a brine-source heat pump with a thermal power of 8 kW and a COP of 4.65 at B0W35, and is optimized for low brine temperatures. The ice storage has a volume of 400 liters, is equipped with a coiled pipe heat exchanger (diameter 20 mm, 30 mm distance between the pipes) and was simulated with Type 843 [7].

For the *Unglazed* system, the collector field was replaced by selective unglazed absorbers (Type 202 [8]) with a total area of 18 m^2 (b0 = 0.01 s/m, b1 = 9 W/(m^2K), b2 = 3.7675 J/(kgK), η_0 = 0.954, based on aperture area) inclined at 45° and a south orientation. The lower limit of the brine temperature was set to –16 °C. Below this temperature the heat pump stops running and the backup heater is used instead of the heat pump. Brine temperatures above +15 °C were avoided by mixing with the return brine flow from the evaporator.

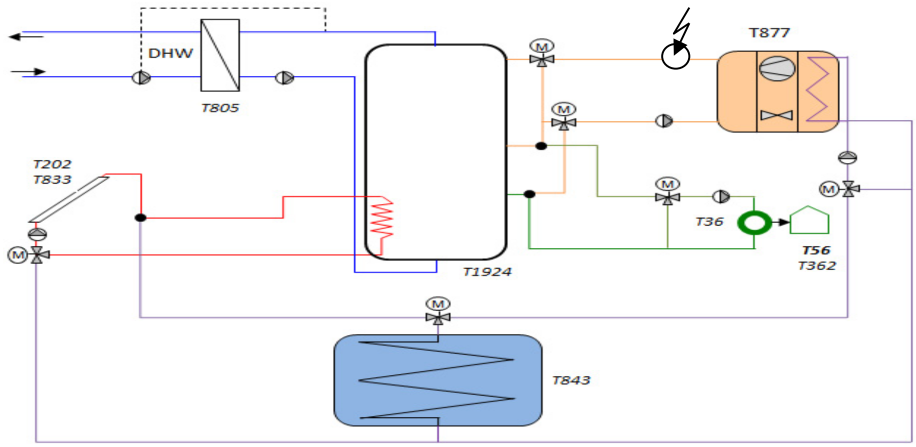
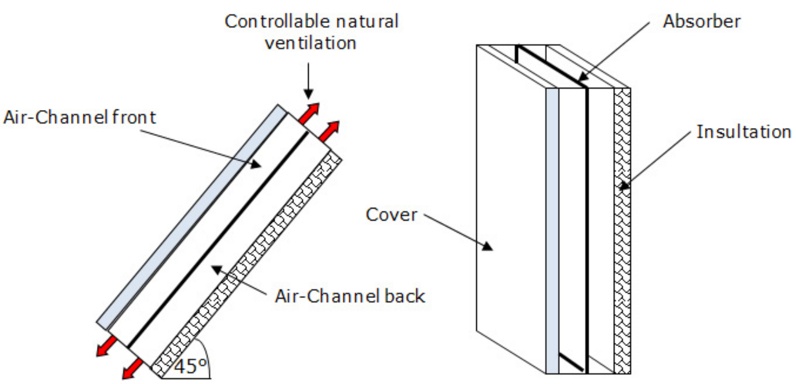


Figure 2: Simplified hydraulic scheme of both alternative systems; Txy = TRNSYS Type number xy

The heat pump primarily uses the collectors as its heat source. If the collector outlet temperature gets too low, the heat pump switches to use the ice storage as a heat source. The first priority of the collectors is to directly charge the hot storage. Only if the required temperature cannot be reached, the collectors are heating the ice storage up to a maximum temperature of 20 °C. In case the heat pump power output is too low, an electric backup heater switches on, which is placed downstream of the heat pump outlet. For all simulations a penalty value of max. 2% was allowed. This penalty value punishes for not meeting the defined comfort criteria, i.e. when the required temperatures for space heating and domestic hot water are not reached (for details see [9]).

The *Ventilated* system has the same hydraulic scheme as shown in Figure 2. The only difference to the *Unglazed* system is that another collector design is used (Figure 3). This design is basically a standard flat plate collector with passively opening channels that allow a natural convection between the absorber and the glazing as well as between the absorber and the insulation. Thus, ambient air can be used as a heat source for the heat pump at times of low (or no) solar irradiation, where the direct use of collector heat for the hot storage would be inefficient or impossible.



Parameters (closed/open)

$a_1 = 3.95/11$ W/(m^{2}K)

$a_2 = 0.0122/0.0265$ W/(m^{2}K^2)

$\eta_0 = 0.793/0.793$

Figure 3: Design of the naturally ventilated collector

The TRNSYS collector model Type 832 [10] was modified to include the natural ventilation feature. The collector efficiency parameters for the open air channels were calculated in EES (Engineering Equation Solver) based on equations from the Kolektor 2.2 software [11]. The natural convective heat transfer is based on the theory of Klan [12]. A possible influence of wind on the air flow in the channels was not taken into account. For this first study of the potential of this new concept, it was assumed that the mechanism for passively opening and closing the ventilation channels would work smoothly. However, this mechanism has not been developed yet.

A fair comparison of the performance of the three different collector and system concepts can only be done if the investment costs for all three systems are equal. Therefore, the collector areas of the alternative systems have been sized in order to reach the same estimated investment cost as for the reference. Thus, due to the less expensive collector design, the *Unglazed* and the *Ventilated* system have collector areas of 18 m^2 and 14 m^2 instead of 10 m^2 (reference system).

2.4. Analysis criteria

Table 2 shows the parameters which are used to compare the systems whit each other. The main parameter is the seasonal performance factor which is the ratio of all energy gains contributed to the system divided by the total electricity consumption of all system components including the space heat distribution pump and penalties for comfort losses.

Table 2: Summary of the rating parameters used for the comparison of the results

Parameter	**Unit**	**Description**	**Boundaries**
$SPF_{SHP+,pen}$	-	Seasonal performance factor	Complete system (with space heating pump (+) and penalties)
$Q_{solar,tot}$	MWh	Collector gain	Collector field without pipe losses
$W_{el,\ SHP+}$	MWh	Electric demand of the complete system	Complete system, with space heating pump
$W_{el.Backup}$	MWh	Electric demand of the backup heater	Backup heater

The SPF of the system including penalties is defined according to T44A38:

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

3. Results and discussion

Figure 4-6 show the comparison of the results for the 6 different climates. All shown energy values are given in MWh.

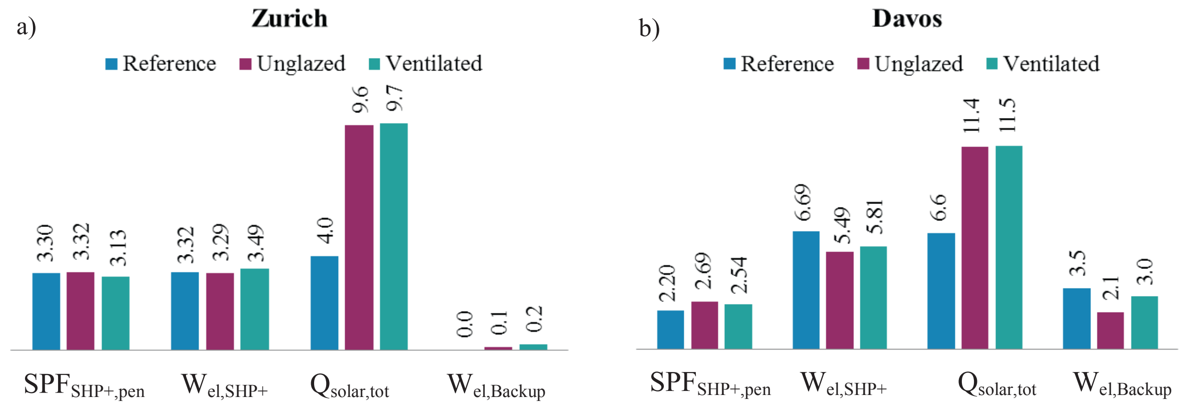


Figure 4: a) Comparison of the systems for SFH45 Zurich (Switzerland), b) Comparison of the systems for SFH45 Davos (Switzerland)

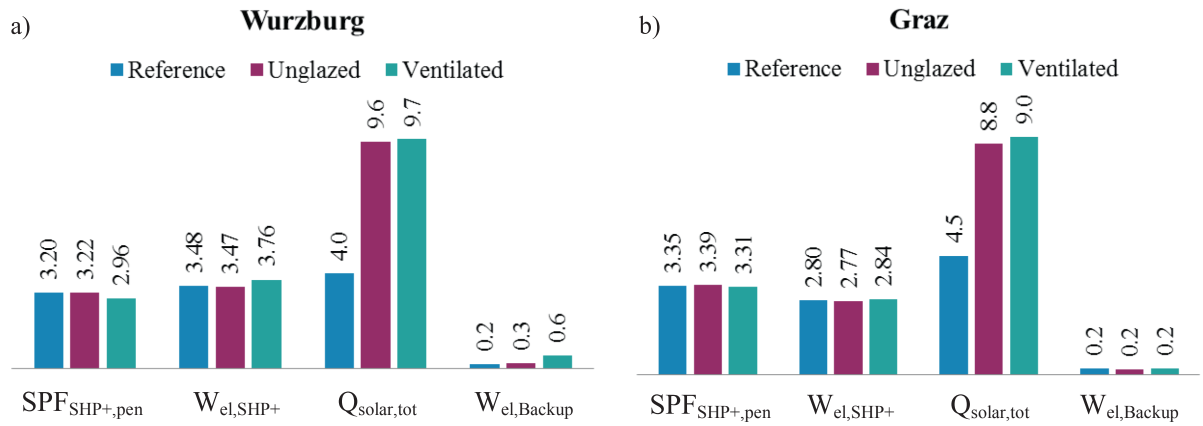


Figure 5: a) Comparison of the systems for SFH45 Wurzburg (Germany), b) Comparison of the systems for SFH45 Graz (Austria)

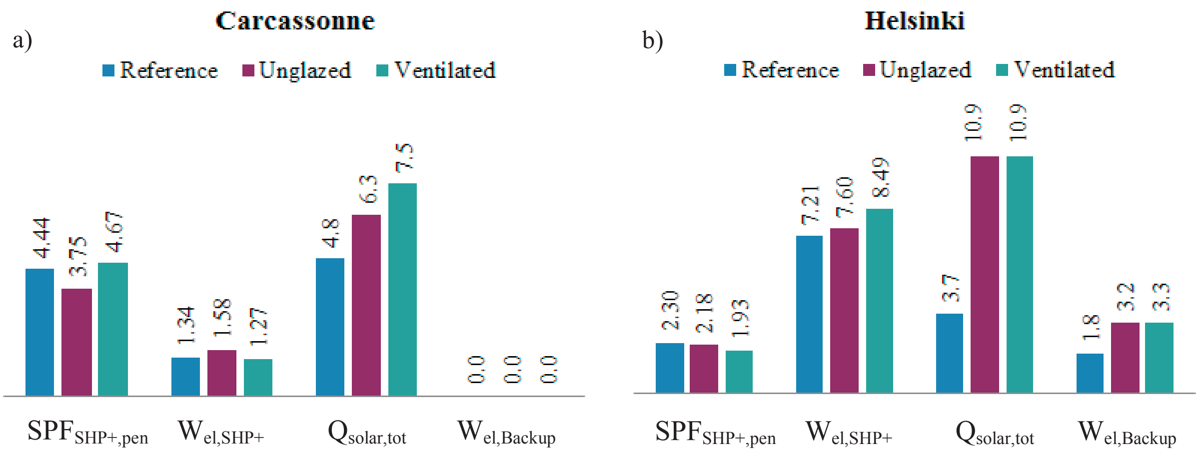


Figure 6: a) Comparison of the systems for SFH45 Carcassonne (France), b) Comparison of the systems for SFH45 Helsinki (Finland)

The simulation results for the climates of Zurich, Wurzburg and Graz show no significant performance difference between the reference and the *Unglazed* system. The electricity consumption is similar for both system types for these middle European climates. The collector gain of the unglazed collectors is about 2.4 times higher compared to the glazed collectors. This factor is lower in a climate with more sunny days, which is the case for Carcassonne, Graz and Davos. In Figure 5b it can be seen that the solar gain ratio between unglazed and glazed collectors decreases to 1.95 for Graz and 1.05 for Carcassonne (Figure 6a). For Carcassonne, also the SPF of the unglazed system is much lower (-15.5%) compared to the reference. For Davos, a cold climate with a very low ratio of DHW/SH, the *Unglazed* system performs +22% (SPF) better compared to the reference. Only for this climate also Davos (Figure 4b) the *Ventilated* system shows a better performance (-13% of W_{el}) than the reference. Further, for Carcassonne, the simulation results show a significant increase of the solar gain for the ventilated collector compared to the unglazed collector (+19%).

The el. backup heater has a large influence on the results for the cold climates Helsinki and Davos. For these climates, the chosen heat pump which is optimized for middle European climates is not an optimal choice. The penalties for Helsinki with the *Ventilated* system could not be kept within the limits of 2 %, and therefore this system is not a valid solution for this case.

Figure 7 and 8 compare the reference and the *Unglazed* system for different buildings in Zurich and Carcassonne in combination with different flow and return temperatures of the space heat distribution system. The simulations show the effect of the ratio of DHW/SH on the SPF of the system and on the electric demand. For Zurich, the performance difference between the *Unglazed* and the reference system is small regardless of the building. For Carcassonne, however, the difference is influenced by the ratio of DHW/SH. Moreover, the difference between the two system concepts is dependent on the flow and return temperature of the heat distribution. For SFH45 with design space heat distribution temperatures of 35/30 °C (Figure 8a) the reference system performs about 16% better than the *Unglazed* system, but for the same space heat demand with heat distribution at 55/45 °C (Figure 8b) the performance difference is only 8%. The SPF of the heat pump is about 10% higher for the *Unglazed* system (SFH45 - 35/30) compared to the reference. For SFH45 - 55/45 the heat pump of the *Unglazed* system performs about 20% better. For Carcassonne SFH100 with heat distribution at 35/30 °C, SPF_{SHP+} differs only little, but the SPF of the heat pump of the *Unglazed* system is 16% better than for the reference system. However, for the same building with higher space heating distribution temperatures of 55/45 °C, SPF_{SHP+} of the Unglazed system is 8% higher than for the reference, and the heat pump SPF is 29% higher (see Figure 8d). For both climates it can be seen that the difference in the total solar gain ($Q_{solar,tot}$) for the different types of collectors varies a lot depending on the building. However, the direct solar contribution of the unglazed and the standard collector ($Q_{solar\ to\ storage}$) show no significant differences between the two buildings.

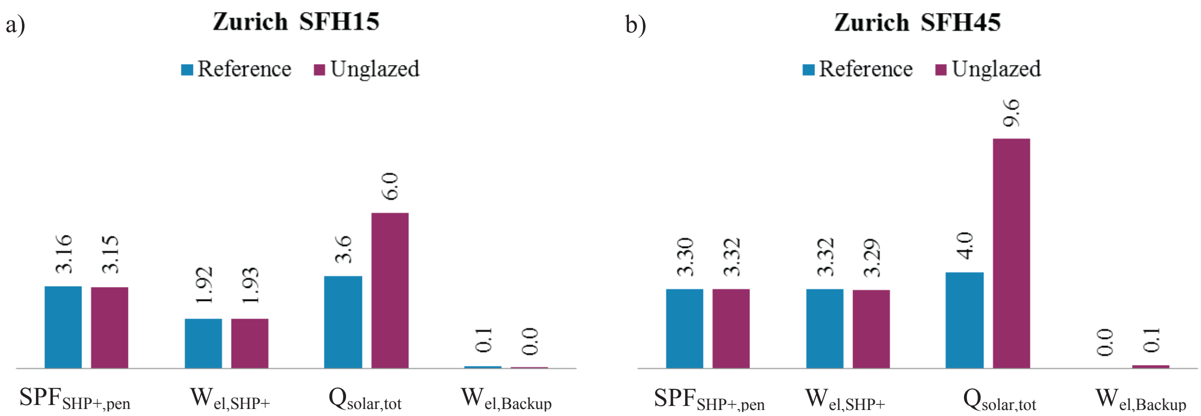


Figure 7: Comparison of simulation results between SFH15 (a) and SFH45 (b) for Zurich, both with 35°C flow temperature and 30°C return temperature for the heating system

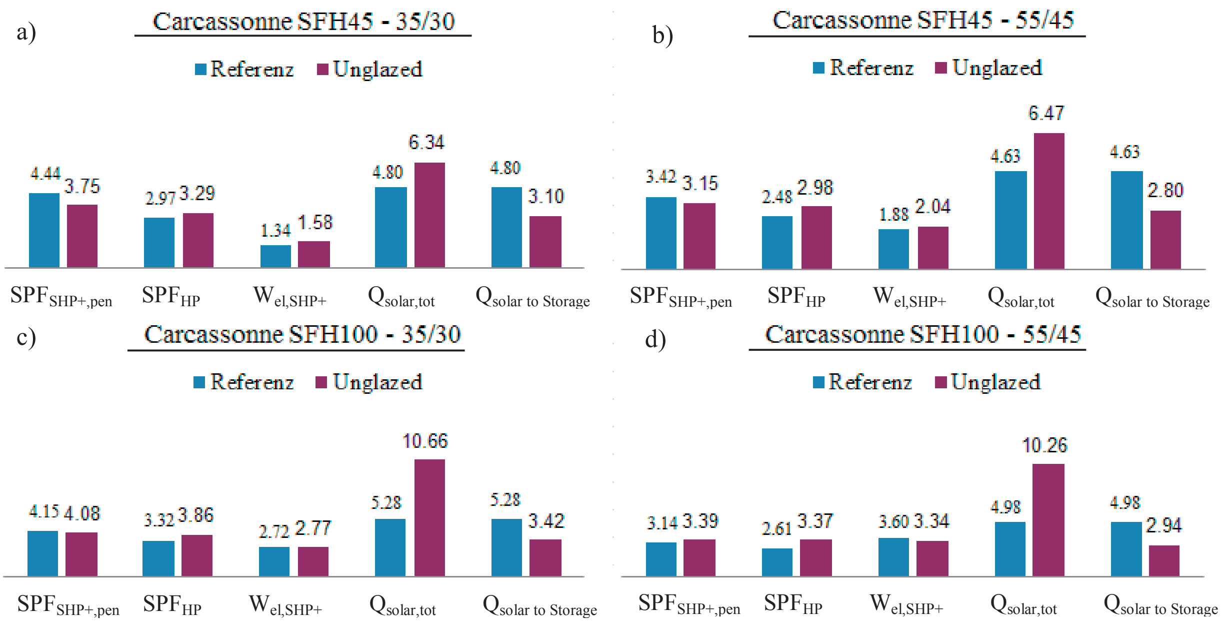


Figure 8: Simulation results for the Carcassonne climate for different building heat loads (SFH100 and SFH45) and for different flow and return temperatures of the heating system (35/30 and 55/45).

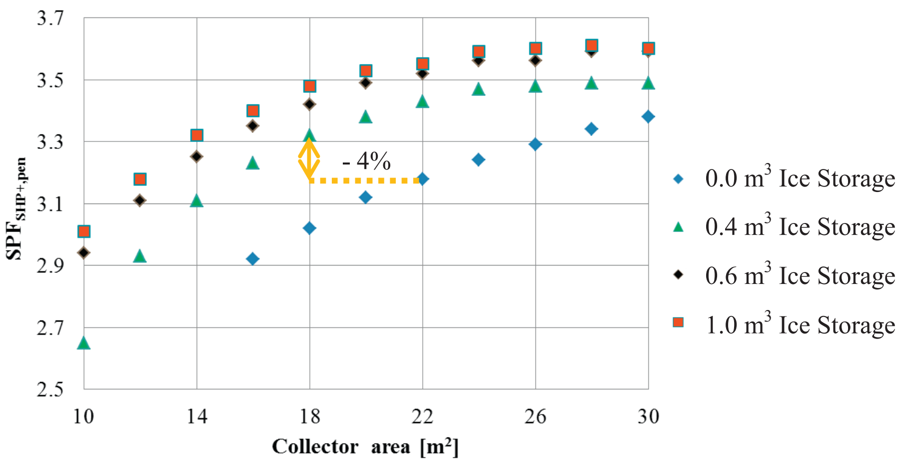


Figure 9: Influence of the ice storage volume and of the collector area on the $SPF_{SHP+,pen}$ for the *Unglazed* system and SFH45 Zurich

Figure 9 shows the influence of the collector area on the $SPF_{SHP+,pen}$ for different volumes of the ice storage. One can clearly see that the $SPF_{SHP+,pen}$ increases rapidly with increasing collector area. But it can also be seen that there exists an optimal size for the collector area. Another outcome of these results is that increasing the ice storage volume improves $SPF_{SHP+,pen}$ only to a limited extent. The bigger the collector area, the less a higher ice storage volume improves the SPF of the system. The blue squares show the performance of the unglazed system without ice storage. The SPF of such a system is considerably lower. Using a smaller ice storage would allow to install more collector area for the same overall system price. However, investing in 4 m^2 of additional unglazed collectors (total collector area 22 m^2) instead of investing the same amount into a 0.4 m^3 ice storage leads to a performance decrease of 4 % (shown by yellow arrows in Figure 9)

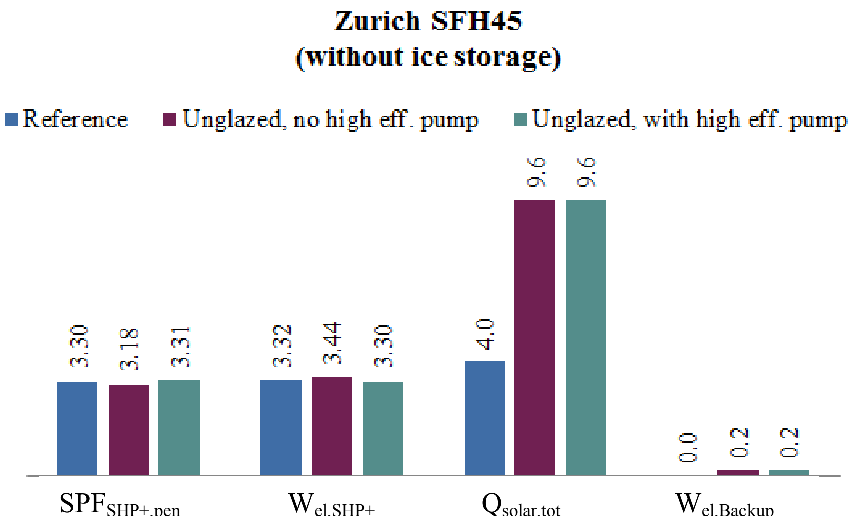


Figure 10: Comparison between Reference System and two *Unglazed* systems without ice storage, Zurich SFH 45, unglazed collector area 22 m^2

Due to the fact that for an *Unglazed* system without ice storage the electricity consumption of the solar pump may be very important, further simulations were done in order to see the difference between a system with a standard solar pump (η = 0.12) and with a high efficiency pump (η = 0.4). The results are shown in Figure 10. As an additional benchmark the results of the reference system are plotted, too. For both *Unglazed* systems the 0.4 m^3 ice storage was spared and for the spared cost 4 m^2 additional collector area (total 22 m^2) were added. A higher efficiency of the solar pump clearly increases the performance of the system without ice storage. The SPF_{SHP+} changes from 3.18 to 3.31 which is an improvement of about 4%. This shows how important the efficiency of the pump in such a system can be. The *Unglazed* system without ice storage can perform as well as the reference system, if the efficiency of the solar pump is high. For the future (2015) high efficiency pumps will be mandatory anyway in order to meet the European rules of energy efficiency.

It has to be mentioned that for the unglazed collectors not all effects were taken into account in this study: For example, the loss of selectivity of the surface when the collector is covered by water droplets from condensation of moisture from the air [13] or the ice formation that may occur when the circulating fluid is at temperatures below 0 °C were not modeled. Also the modeling of the ventilated collector did not take into account the influence of wind, thereby possibly underestimating the air source gains of this system. Another point is the cost estimation of such a collector, which cannot be done easily because there are no covered collectors on the market which use natural ventilation.

4. Conclusions

The comparison of the simulated systems shows that their performance strongly depends on the climate. For none of the systems it can be claimed that it is the best in all cases. Still, in many cases, the *Unglazed* system leads to a good performance compared with the reference, making it a good alternative. Only for Carcassonne its performance is worse for buildings with low temperature floor heating systems – which has significantly lower space heating demand than 61.6 kWh/(m^{2}a) (SFH100 Carcassonne). The results show that the *Unglazed* system is more likely to be better than the reference the higher the flow and return temperatures of the space heat distribution are. This confirms the results of the theoretical analysis presented in Haller et al. [14], who concluded that serial collector heat use (for the evaporator of the heat pump) is more advantageous compared to parallel collector heat use when the temperatures of the heat demand are higher.

Further it can be said that the size of the ice storage becomes more important if the collector area is small (less than 18 m^2). For larger collector fields (above 18 m^2) the benefit of a larger ice storage has to be put in contrast with the additional costs.

Furthermore, the simulation results indicate that the *Unglazed* system can work well even without an ice storage if the standard solar pump is exchanged for a high efficiency pump. This makes the *Unglazed* system more flexible: If the available roof area for solar collectors is large enough, the ice storage may be omitted. On the other hand, if there is enough space in the technical room and the orientation of the building is not optimal, a larger ice storage can be the favorable solution. Moreover, it has to be considered that the simulations are simplified – icing and snow coverage of the collectors was not taken in to account – what means that the ice storage solutions could be underestimated with these simulations.

The results show that selective unglazed collectors in combination with a small ice storage can lead to a good and reasonable seasonal performance factor, which lies in the same range or is even better than the SPF of the state of the art air source heat pump combined with standard glazed collectors. Additional benefits of the *Unglazed* system compared to the reference air source heat pump are that noise emissions and outdoor air heat exchanger units can be avoided. In the case of the natural ventilated collectors the benefit in some specific circumstances cannot overweigh the uncertainties and the low performance for the main field of application.

Acknowledgements

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