solarcombi+

General results of the virtual case study

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Identification of most promising markets and promotion of standardised system configurations for the market entry of small scale combined solar heating & cooling applications
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1 Introduction

In course of the virtual case study a large number of simulations have been carried out. For the selected chiller of each participating industry partner the following variations were carried out:

- Three locations (Strasbourg, Toulouse, Naples)
- Three types of buildings (Office, two residential buildings)
- Two systems of cold distribution (Fan coil and chilled ceiling)
- Two collector types (flat plate, evacuated tube)
- Two heat rejection technologies (wet cooler, hybrid cooler)
- Three storage volumes
- Five collector sizes

The office building was simulated in all three locations with fan coil only; the residential buildings were simulated in the two southern locations (Toulouse and Naples) only with both cold distribution systems.

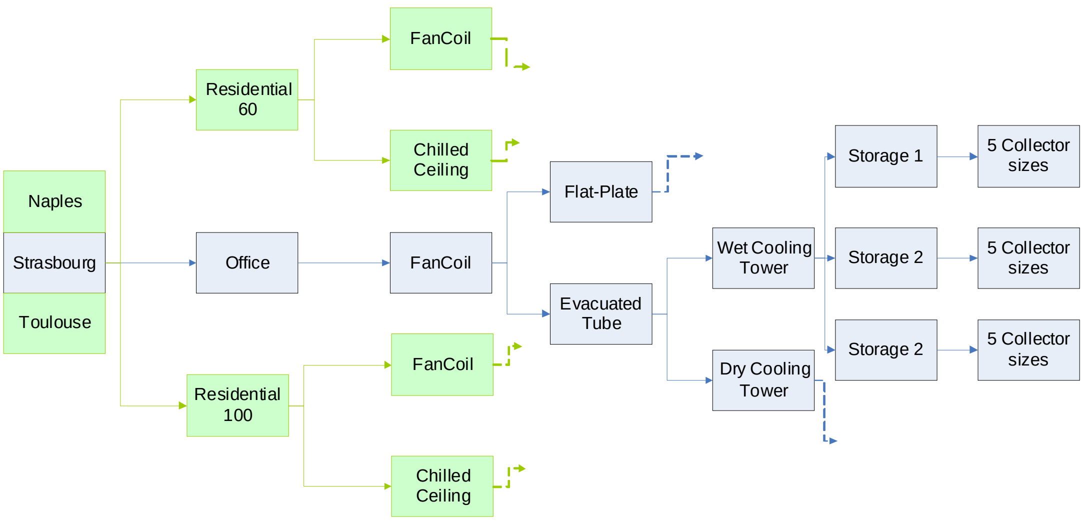

Figure 1: Overview of the variations of simulations for each chiller.

With all these variations the total number of simulations per chiller sums up to 660 (some manufacturers excluded certain configurations or analyzed additional variations).

For detailed information on the configurations used within the virtual case study please read the "Report on methodology for virtual case studies" available on the SolarCombi+ homepage.

1.1 General remarks

The objective of the virtual case study is challenging: standard system configurations for solar cooling applications shall be developed for five different chillers. The specific technologies of the chillers and the strategies of the respective industry partners lead to a number of compromises in order to facilitate a widely standardized configuration.

One compromise which was taken is the two basic hydraulic schemes with which the chillers are simulated. Subsequently it was aimed at treating all chillers according to one methodology – the heart of which is the *reference chilling capacity*. The introduction of this value shall guarantee that all chillers can meet the maximum cooling load under the same conditions.

The following differences in methodology between chillers exist and their effects were *not* quantified:

- Two basic hydraulic schemes were defined; each chiller is only simulated in one of them according to the preferences of the industrial partner.

- SOLution opted for a on/off control strategy for the solar system depending on the temperature difference between collector and storage; all other solar systems operate with variable speed pumps.

- SOLution markets their absorption chiller with cold storage, this was implemented for the virtual case study as well.

- ClimateWell provided a dynamic physical model for their chiller, all the other chillers are simulated by models based on steady-state conditions.

- Heat rejection units connected to Rotartica, Sonnenklima and Sortech chillers are operated by an individual strategy (variable fan speeds) in order to control the chilling capacity. The remaining systems work with an on/off strategy.

- The Sortech adsorption chiller allows for internal regulation of the chilling capacity by varying the sorption cycle duration. In the simulation it operates in power mode (max. power) when using solar energy and in eco mode (max. COP) when using fossil backup.

- Some chillers offer the heat pump mode for space heating in winter as well. However, for reasons of comparability of the study, this mode was not considered.

- The simulations were carried out with a fixed load file, so no building response could be considered. Therefore the loads (heating, DHW and cooling) had to be covered by a fossil backup if the solar heat was temporarily insufficient. Ideally the solar cooling system would be operated without backup allowing the building to exceed the set temperature occasionally.

- From the point of view of primary energy efficiency a compression chiller would be most feasible. In small scale systems this option is not implemented because of economical and conceptional reasons.

- Costs for energy (fossil fuel and electricity) are dependent on many factors (location, market development…) and highly fluctuating. Costs for chillers are in the actual stage of the market entrance unfavourably high.

As mentioned above a considerable limitation of the virtual case study consists in the fixed load file requiring boiler support for cooling. It is known that this fossil heat backup is not favourable for solar cooling applications. Besides, former experiences have shown that full load coverage may lead to an important decrease in primary energy savings and may have a sensible effect on the specific costs (operation costs as well as cost per saved primary energy). This effect has not been studied until now.

If a backup is indispensable, it could also consist of a cold backup with a compression chiller. This is not considered as small solar combi systems include already a boiler and so with using it in the cooling period as heat backup incur no further investment costs, whereas including a compression chiller would increase the investment costs of small solar combi systems disproportionally.

2 Development of simulations

2.1 Results of first simulation campaign

Initial simulations were carried out according to the report on the methodology of the virtual case study within this project. It turned out in the specification of appropriate package solutions that the system control could be optimised. This was done subsequently in the simulations using mainly improved control strategies. Thus, a large number of simulations were repeated. The implemented improvements are described in the following of this chapter. The results of the improved system are published in the online tool at the SolarCombi+ webpage and are used to determine the standard configurations in deliverable 4.1.

Two main issues have been envisaged for the improvements. Firstly there was a high fraction of fossil energy used for chiller operation resulting in a relatively low solar fraction for the cooling mode. Secondly the stagnation time of all simulated systems was much higher than in well dimensioned systems.

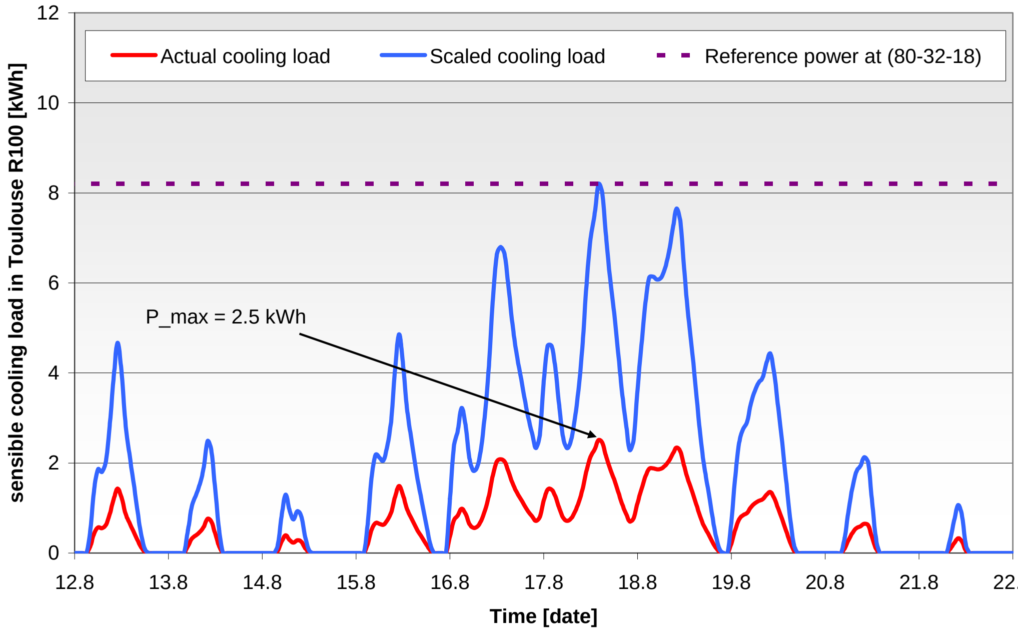

Figure 2: Graphic illustration of load file scaling: the red line indicates the sensible cooling load of a residential building (R100) in Toulouse in the period with the maximal cooling load (August). The blue line shows the same cooling load scaled to fit an exemplary reference chilling power at 80-32-18.

An important reason for these weaknesses was the fact that no building model but a fixed load was coupled to the simulation. Further limitations are consequences of the compromises taken in order to determine standard configurations. As described in part 1.1 of the "Report on the methodology of the virtual case study" the comparability between chillers with different

chilling capacities is guaranteed by scaling the load file to a reference chilling capacity at 80°C driving temperature divided by the maximum load at a specific location. This means that the size of the building is adapted for the chilling capacity of every chiller. Figure 2 visualizes load file scaling.

As there is no building model coupled to the simulation, the building cannot react to the system conditions. Subsequently, the system has to guarantee a temperature of 80°C to be able to always meet the maximum power even if this occurs only about one hour in the whole year (See figure 3 for frequency distribution of the cooling loads). So if the solar system provides a temperature of 70°C, the boiler is used to meet the required 80°C. Some chillers, especially adsorption chillers, allow much lower driving temperatures and could meet the actual required chilling capacity also with 70°C driving temperature. So the temperature rise from 70°C to 80°C by the boiler is not necessary. In real systems the boiler would be used only if the temperature in the building rises above the set values for comfort, i.e. if the solar system can not provide enough energy to meet the load.

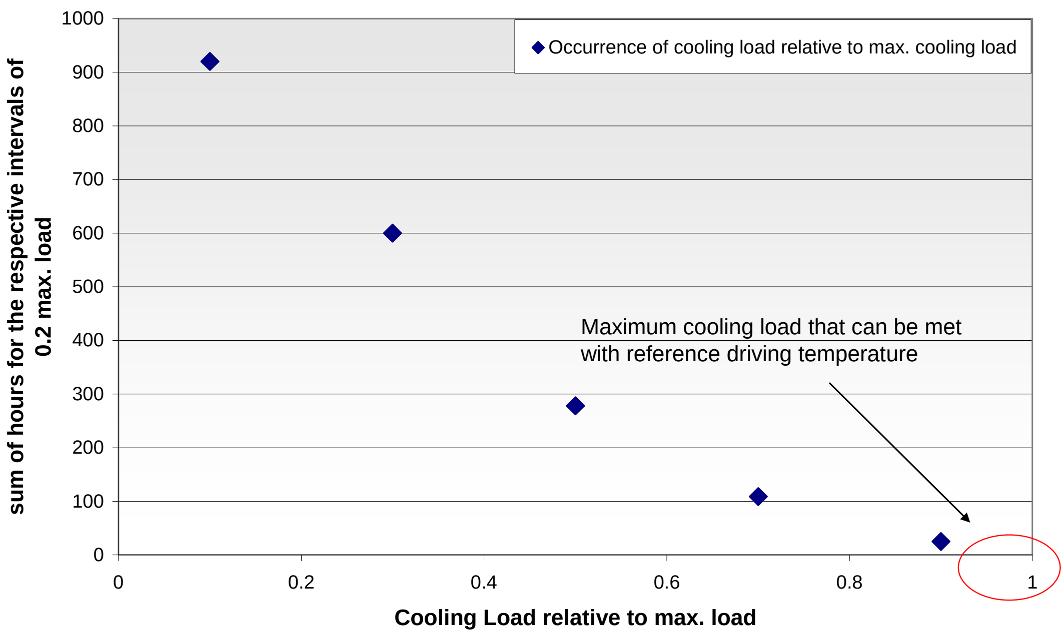

Figure 3: Frequency of occurrence from sensible cooling load relative to max. load at Toulouse, residential building R100.

The scaling of the load file also is an important reason for high stagnation times. As the scaling factor was calculated with a reference chilling capacity at 80°C driving temperature, the maximum load could be met at that temperature. Temperatures above 80°C could only be used by discontinuous chiller operation. Periods of low cooling loads and high irradiation quickly lead to a fully charged hot water storage tank and subsequent stagnation.

Another reason for high stagnation time is the maximum allowed storage temperature. This was set to 91°C. If higher temperatures are allowed, the capacity of the storage grows and stagnation time decreases. In the following three measures are shown which tackle the abovementioned weak points of the simulations.

2.1.1 Change in boiler operation

To take into account the possibility of some chillers to run with lower temperatures than 80°C, the initial control strategy has been changed in order to give solar energy the preference for chiller operation. The boiler is only used to power the chiller if:

a) The temperature provided by the solar system is below the minimum driving temperature of the respective chiller (+5K hysteresis).

b) The solar operated chiller was not able to cover the entire cooling load in the preceding time step. In that case the uncovered load is added to the actual load and subsequently covered by the heat backup. Thus it is met with a slight delay. Figure 4 illustrates the control strategy within 7 time steps (1.5 min in the virtual case study).

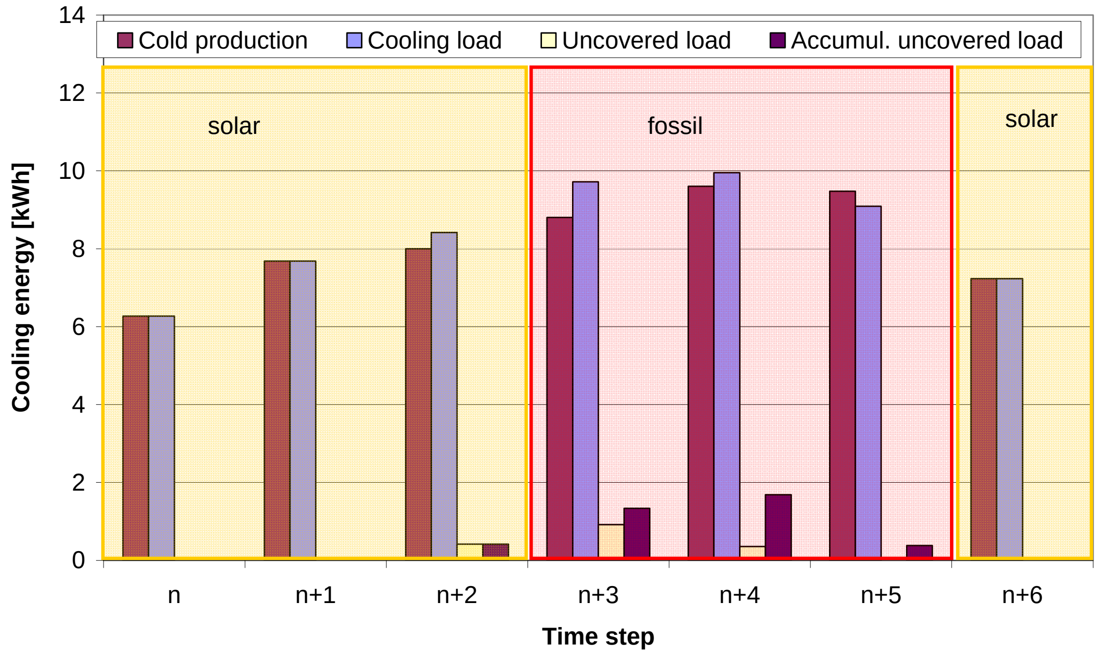

Figure 4: Simplified exemplification of case b) for boiler operation; during time steps n and n+1 the cooling load can be met by the controlled chiller in solar operation. In time step n+2 the demand is higher than production. Therefore the boiler is turned on in time step n+3 allowing for higher chilling capacity. It operates until the entire accumulated uncovered load is met in time step n+5. In step n+6 the system switches back to solar energy again.

The minimum driving temperatures of the simulated chillers are shown in table 2.

Type of chiller	Minimum driving temperature [°C]	Delta T to 80°C
ClimateWell	70	10
EAW	80	0
Rotartica	70	10
Sonnenklima	65	15
Sortech	65	15

Table 1: Minimum driving temperatures of simulated chiller models and temperature gap to 80°C for solar cooling.

With this solar preference strategy the solar fraction can be increased significantly and stagnation time during cooling time decreases. The simulation is closer to real control strategies with negligible losses of accuracy due to the fixed load file.

2.1.2 Change of maximum storage temperature

In the initial simulation the storage could only be charged up to 90°C since this is the maximum for many buffer storages. Since all chillers can deal with temperatures up to 95°C and offer maximum capacity under these conditions it was considered appropriate to increase the limit of the storage temperature to this value.

2.1.3 Change of load file scaling

In initial simulation the load file is scaled so it can be met with a reference driving temperature of 80°C. This means higher driving temperatures lead to on/off – operation since the cold production exceeds the cold demand.

After implementing the preceding measures and considering the frequency distribution shown in Figure 3 it was self-evident to determine the reference chilling capacity of each chiller with hot water temperatures of 95°C. This

further increases the temperature bands for solar operation as shown in Table 3.

Type of Chiller	Minimum driving temperature [°C]	Delta T to 95°C
ClimateWell	70	25
EAW	80	15
Rotartica	70	25
Sonnenklima	65	30
Sortech	65	30

Table 2: Minimum driving temperatures of simulated chiller types and temperature gap to 95°C for solar cooling.

2.2 Enhancement with applied measures

The improvement of solar fraction cooling and stagnation time of a sample configuration is shown in figure 5 and 6 for an exemplary chiller. The results are compared to the results from the initial simulations. The example configuration is the following:

Location: Toulouse (TOU)

Cold distribution: Chilled ceiling (CC)

Collector: Flat plate (FP)

Heat rejection: Wet cooling tower (WC)

Building standard: R100 (R100)

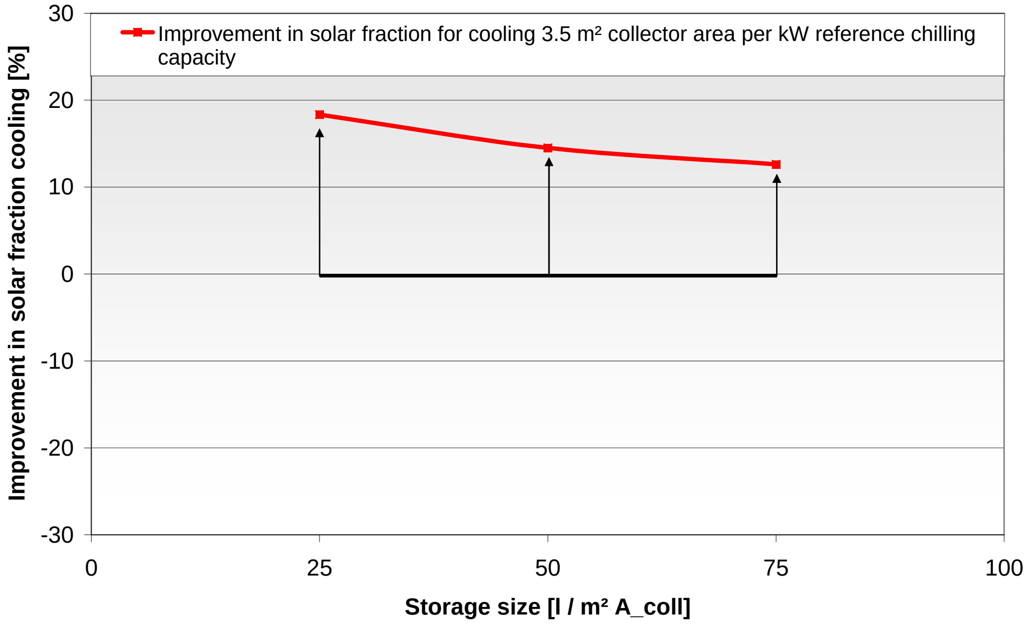

Figure 5: Improvement of solar fraction cooling from initial simulations (horizontal line through 0) to simulations with applied measures.

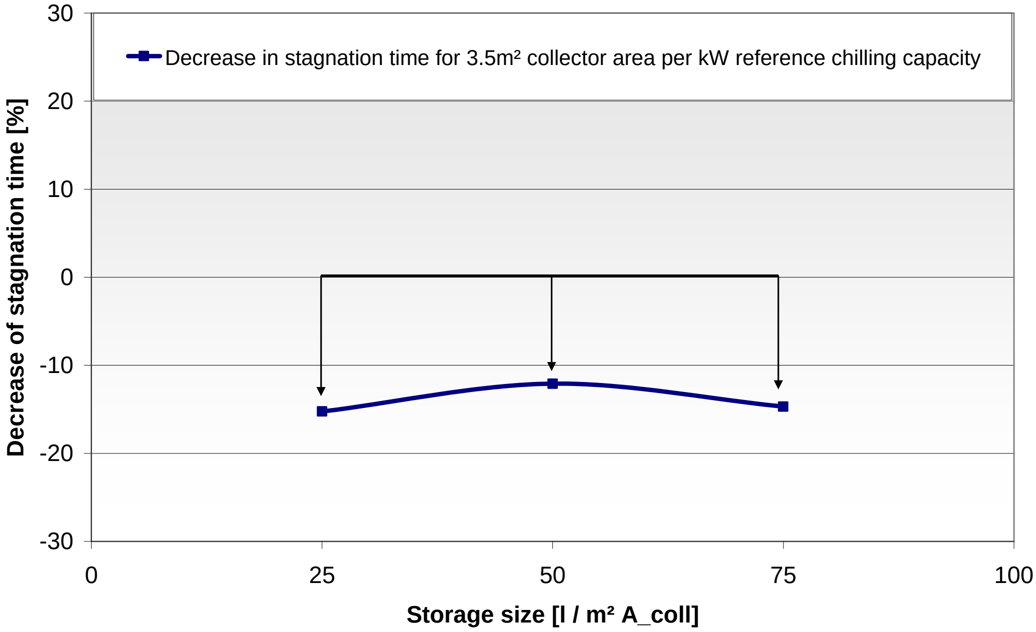

Figure 6: Decrease of stagnation time from initial simulation (horizontal line through 0) to simulation with applied measures.

3 Discussion of results

As the aim of the virtual case study is not to compare the results of the different chillers, but to show tendencies and best solutions for different configurations, the following figures show only relative figures. The reference system (100%) is always the system with a collector size of 3.5 m²/kW reference power (P_ref; medium size) and a storage size of 50 l/m² collector area (medium size), respectively for each chiller and configuration.

Therefore all of the following graphs obey this systematic: The continuous line shows the relative difference of the largest collector area (5 m²/kW P_ref) to the reference system as an average of all chillers. The dotted line represents the relative difference of the smallest collector area (2m²/kW P_ref) to the reference system, again as an average over all chillers. The (medium sized) reference system is indicated with a black dot. Thus the reader gets an impression of the relative influence of a differently sized system.

Figures 7 to 9 in section 3.1 indicate for one fixed configuration (Naples, chilled ceiling, flat plate collector, wet cooling, building standard R100) the influence of the collector area and storage volume these three (relative) performance figures:

- total solar fraction,
- specific gross collector yield
- costs per saved kWh primary energy.

The influence of the collector type on the same performance figures and the same configuration as in figure 7 to 9 is shown in the figures 10, 11 and 12. Here systems with evacuated tube collector are referenced to the medium sized system with flat plate collectors.

In contrast Figures 12, 13, 14 and 15 compare the results of a variety of system configurations separately for residential and office buildings.

The definition of all used performance figures is outlined in the “Report on the methodology of the virtual case study”, deliverable 3.2, section 3.1 of this project.

3.1 Influence of different system sizes

Figure 7 shows the relative total solar fraction. It increases with growing aperture areas and growing storage sizes. The best solution is closer to the reference system than the worst solution. The gain of the biggest system with respect to the reference is 30%, the loss of the smallest system 40%. This as well as the flattening curve with increasing storage volume illustrates that a larger sizing of the solar system does not infinitely lead to a significantly better system result. Considering the investment costs for the collector system a compromise between performance and costs must be sought.

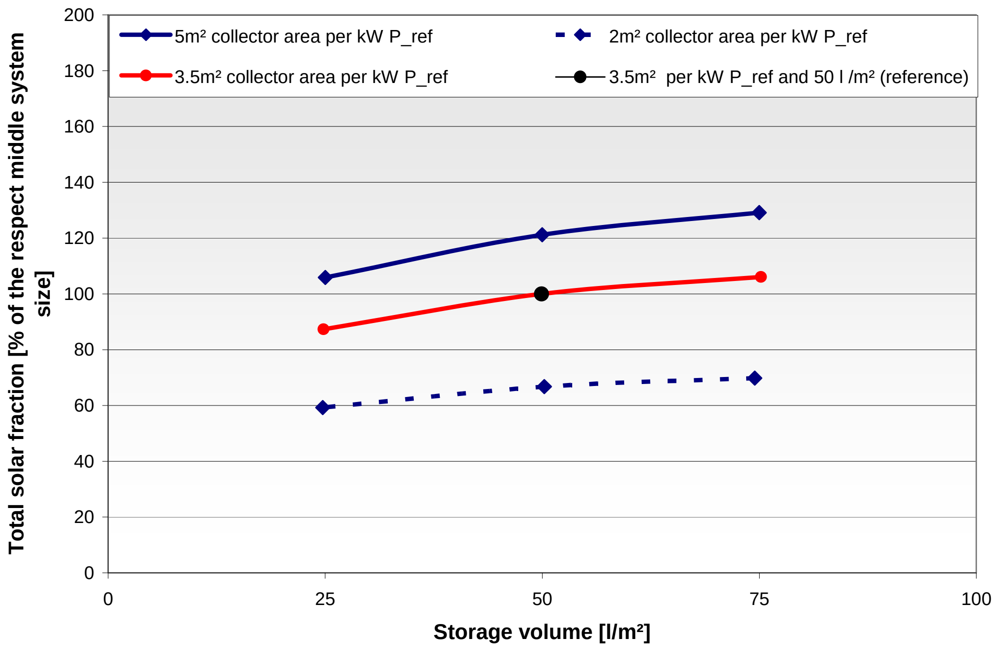

Figure 7: Deviation (average over different chillers) for total solar fraction with respect to the medium sized system (medium collector area and medium storage volume; black dot) for the system configuration Naples, chilled ceiling, flat plate collector, wet cooling tower, building standard R100. The average deviation is shown for 2, 3.5 and 5 m²/kW P_ref and 3 storage sizes.

In figure 8 the relative specific gross solar yield is presented (unit of absolute value: kWh/year/m²). It decreases with a larger collector area and smaller storage sizes. This reduction of solar yield is a consequence of lower collector efficiency at higher temperatures and prolonged stagnation periods when the storage is fully charged. The range with respect to the medium sized system is +30%/-25%. Again it becomes obvious that a change towards smaller collector areas has a larger influence. Reversely the additional yield of additional collector area for already large systems decreases.

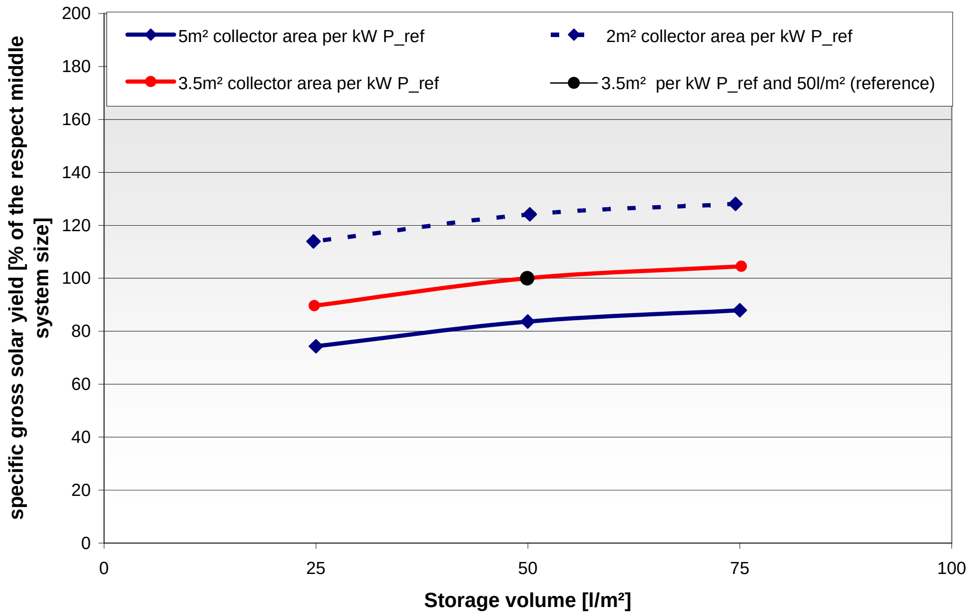

Figure 8: Deviation (average over different chillers) for specific gross solar yield with respect to the medium sized system (medium collector area and medium storage volume; black dot) for the system configuration Naples, chilled ceiling, flat plate collector, wet cooling tower, building standard R100. The average deviation is shown for 2, 3.5 and 5 m²/kW P_ref and 3 storage sizes.

Figure 9 shows the average deviation of the costs per saved kWh primary energy CPE. This evaluation number is defined as the cost of the primary energy saved by the solarcombi system in comparison to a conventional system with gas boiler and compression chiller. If positive it shows the additional cost to a conventional system per saved kWh primary energy. Thus it combines energetic and economic evaluation numbers. In the graph it can be seen, that the systems with big collector areas are close to the medium sized system, so the cost reduction from a medium sized to a large collector areas is comparably small. Contrarily the costs increase a lot from a medium system size to a smaller system size (up to 500%). The high values for 2 m² collector area per reference chilling capacity are result of high usage of the boiler resulting in much less primary energy saved whereas the costs of the total system do not decrease proportionally. Thus this value is an important performance indicator which helps to determine an appropriate compromise between system performance and investment cost (the collector system constitutes an important share of the total investment cost!). The minimum of the curve coincides with storage volumes above 75l/kW reference chilling capacity. This is a consequence of the boiler backup for cooling which is not favourable from the point of view of primary energy.

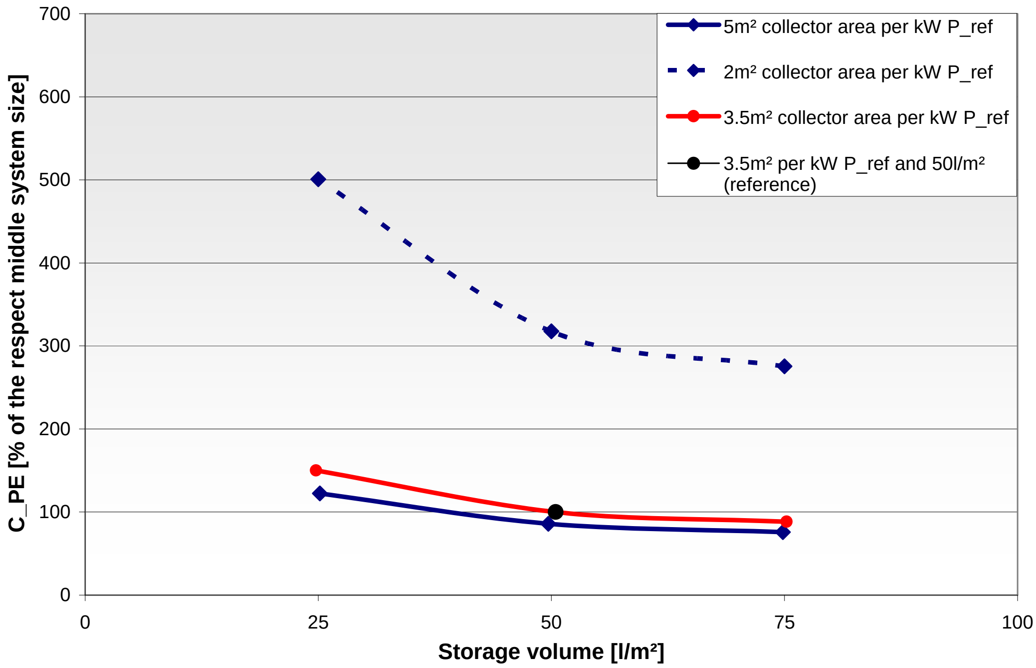

Figure 9: Deviation (average over different chillers) for cost of saved primary energy with respect to the medium sized system (medium collector area and medium storage volume; black dot) for the system configuration Naples, chilled ceiling, flat plate collector, wet cooling tower, building standard R100. The average deviation is shown for 2, 3.5 and 5 m²/kW P_ref and 3 storage sizes.

3.2 Influence of different collector types

Figures 10 and 11 illustrates that a higher total solar fraction as well as a higher specific gross solar yield can be achieved using evacuated tube (ET) instead of flat plate (FP) collectors, independently of the dimensioning of the collector array. The reason lies in the required high temperatures for solar cooling operation. At higher temperatures evacuated tube collectors are more efficient than flat plate absorbers. All configurations and system sizes are referenced to the indicated medium size system with flat plate collector.

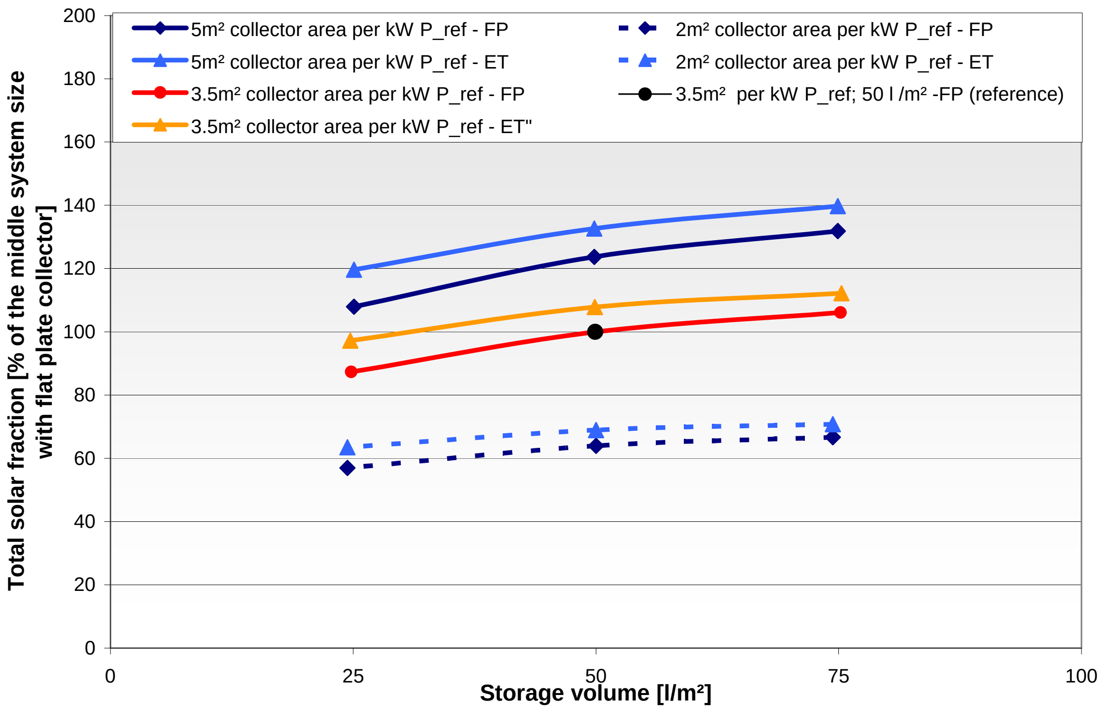

Figure 10: Deviation (average over different chillers) for total solar fraction with respect to the medium sized system with flat plate collector (medium collector area and medium storage volume; black dot) for the system configuration Naples, chilled ceiling, flat plate collector, wet cooling tower, building standard R100. The average deviation is shown for 2, 3.5 and 5 m²/kW P_ref and 3 storage sizes.

The difference in the costs for saved primary energy is strongly influenced by the specific investment costs for each technology. For the shown graph the investment costs for evacuated tube are twice the cost of flat plate technology. This results in costs for primary energy saving approximately 25 percentage points above those for flat plate collectors.

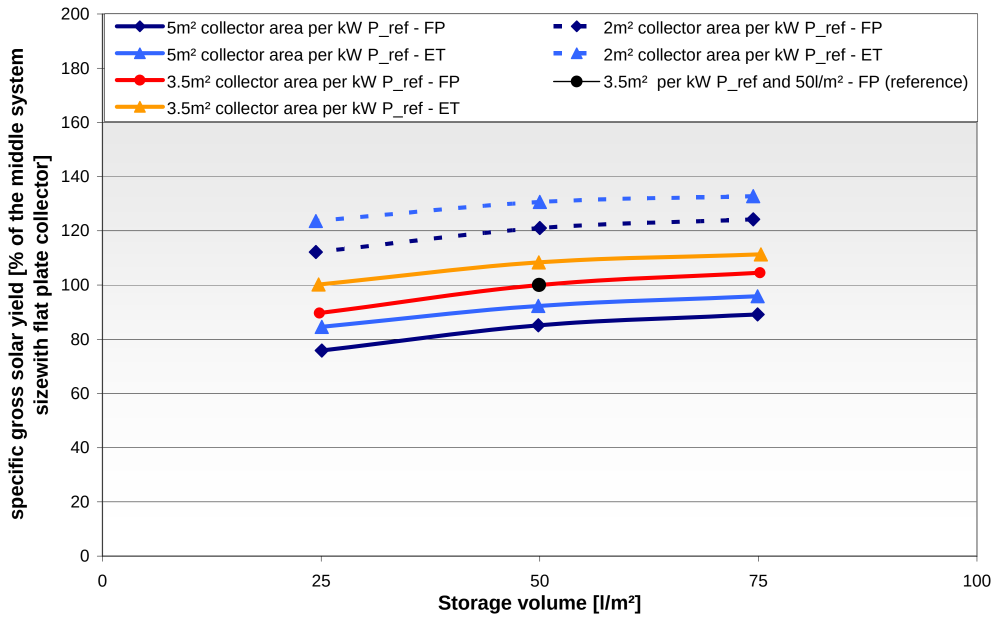

Figure 11: **Deviation from specific gross solar yield of the reference system with flat plate collector (solar medium sized system -> black dot) in the system configuration Naples, chilled ceiling, wet cooling, building standard R100 as an average over the chillers. The average deviation is shown for 2, 3.5 and 5 m²/kW P_ref, respectively 3 storage sizes and 2 collector types.**

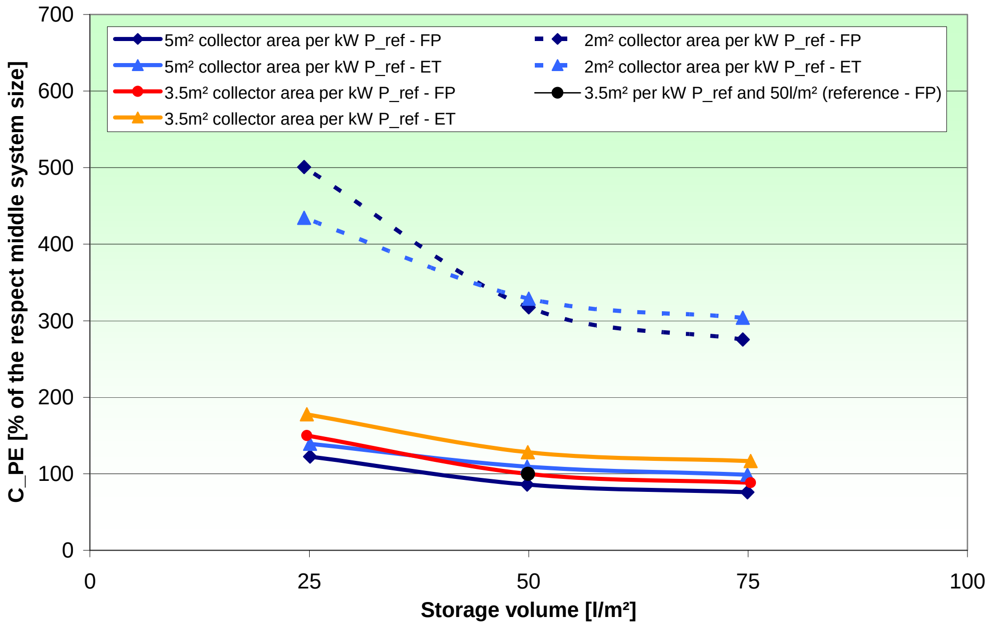

Figure 12: **Deviation from costs per kWh saved primary energy of the reference system with flat plate collector (solar medium sized system -> black dot) in the system configuration Naples, chilled ceiling, wet cooling, building standard R100 as an average over the chillers. The average deviation is shown for 2, 3.5 and 5 m²/kW P_ref, respectively 3 storage sizes and 2 collector types.**

3.3 Influence of different system configurations

In figures 13 and 14 the influence of the simulated variations (location, building standard, cold distribution system and heat rejection technology) is shown for residential buildings. The relative influence is rather uniform among all variations. In case of component variations this can be explained by the fact that the load file is adapted individually to each of the configurations.

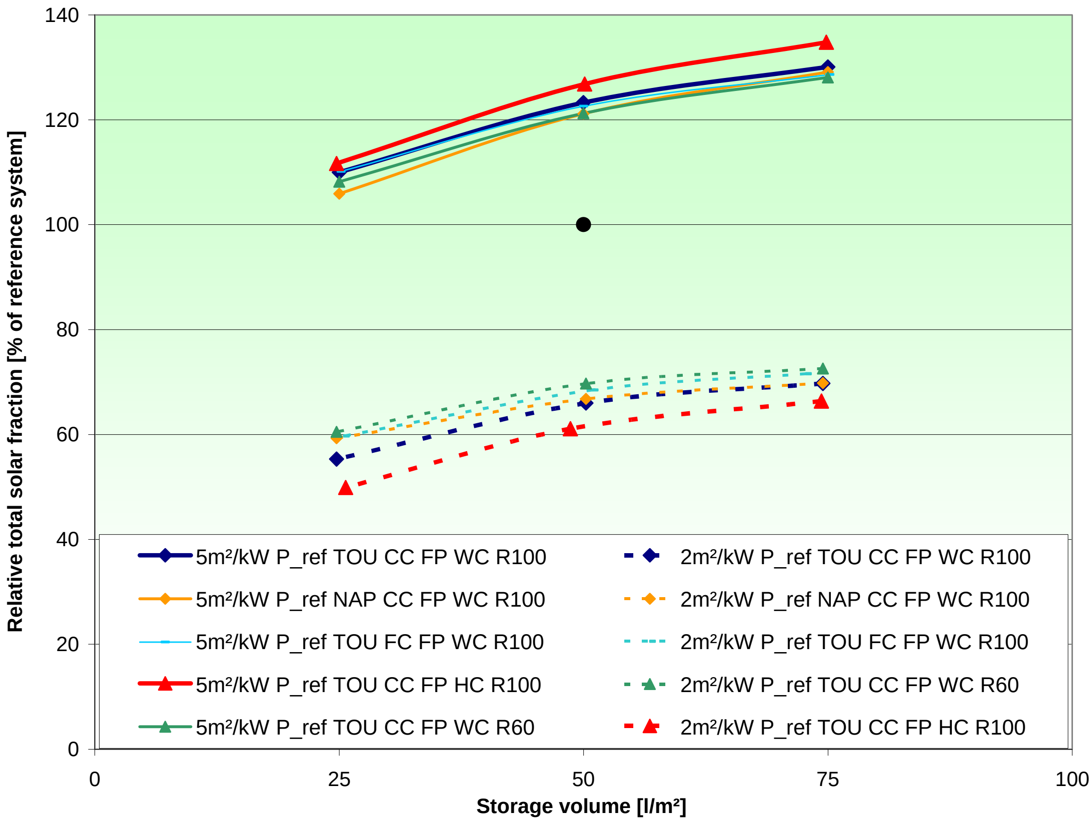

Figure 13: Deviation of the total solar fraction of various system configurations to the respective medium sized system with identical configuration (black dot) as an average over the chillers. The deviation is shown for 2 and 5 m²/kW P_ref, respectively 3 storage sizes. All shown configurations include a residential building.

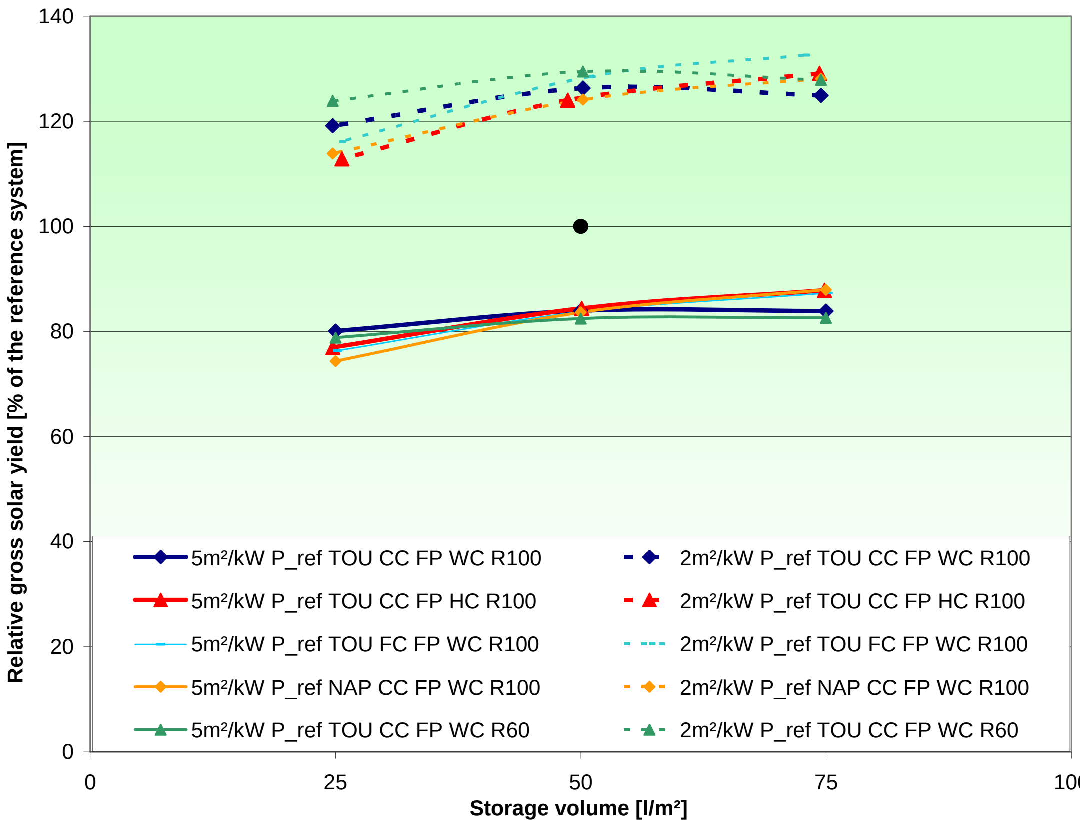

Figure 14: Deviation of the gross solar yield of various system configurations to the respective medium sized system with identical configuration (black dot) as an average over the chillers. The deviation is shown for 2 and 5 m²/kW P_ref, respectively 3 storage sizes. All shown configurations include a residential building.

In figures 15 and 16 the influence of the location on the total solar fraction and gross solar yield is presented for the office building. The differences in relative total solar fraction (figure 10) are noticeably higher than in all variations of the residential buildings. This is caused by three factors:

- the difference in the ratio of annual heating and cooling demand between the locations is much higher for office buildings than for the residential dwellings.

- The daily and weekly load profile of the office building is strongly influenced by setbacks for room temperature during nights and weekends.

- In the simulations the DHW demand of the office buildings is neglected.

These factors are also the reason for the more striking differences in gross solar yield.

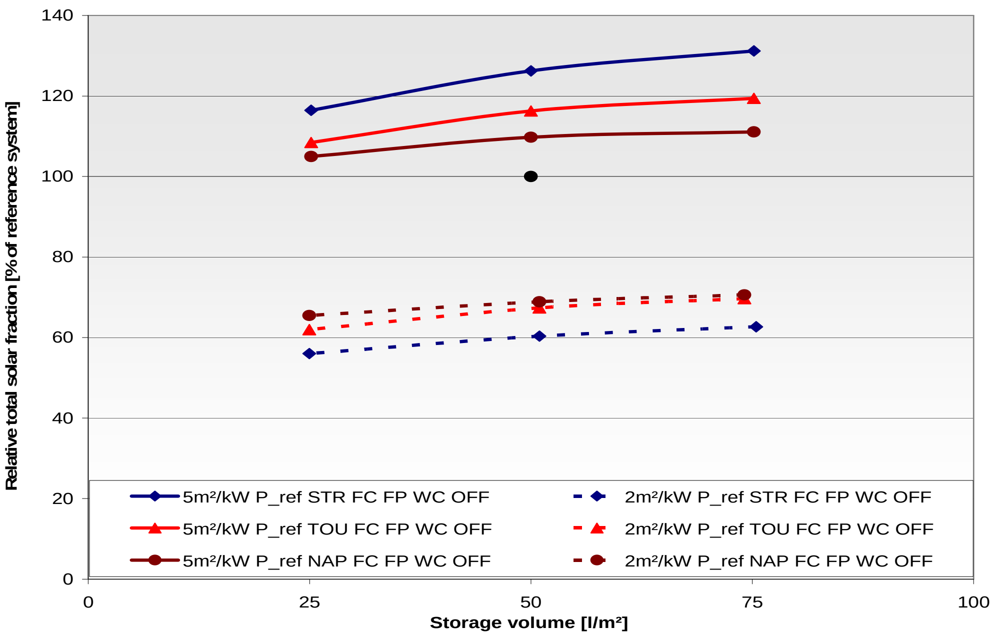

Figure 15: Deviation of the total solar fraction of various system configurations to the respective medium sized system with identical configuration (black dot) as an average over the chillers. The deviation is shown for 2 and 5 m²/kW P_ref, respectively 3 storage sizes. All shown configurations include an office building.

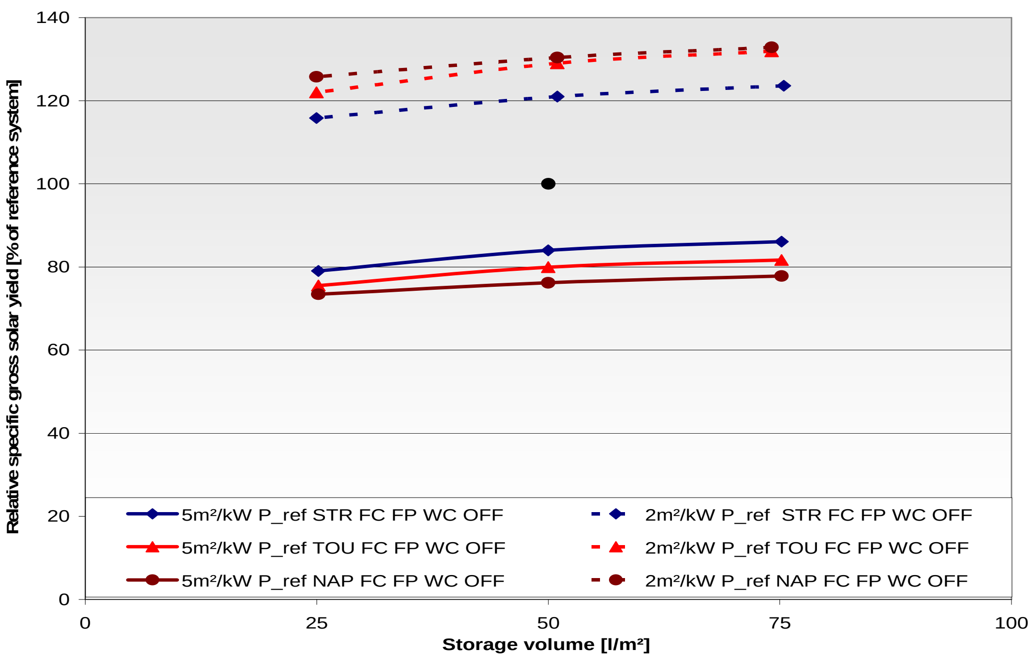

Figure 16: Deviation of the gross solar yield of various system configurations to the respective medium sized system with identical configuration (black dot) as an average over the chillers. The deviation is shown for 2 and 5 m²/kW P_ref, respectively 3 storage sizes. All shown configurations include a residential building.

4 Conclusion

In course of the virtual case study a large number a standardized methodology for the simulation of different chillers with a variety of system configurations was developed and improved. The results of the final simulations provide important information on system behaviour and system design for the selected standard cases. The best cases will be made available to the public in an online database on the SolarCombi+ homepage. Besides they are used to determine the standard system configurations in deliverable 4.1 of this project. An intensive analysis of the results is also the basis for the package solutions developed for each chiller in course of this project.

As general outcomes the following points shall be mentioned:

- Well-sized systems have a collector size of about 3.5 to 5 m²/kW reference chilling capacity and a hot storage volume of 50 to 75l/m² aperture area. In these ranges high total solar fractions can be obtained and the system operates close to the optimum in terms of primary energy savings and costs (see figure 9).

- The implemented improvements showed that an optimized control is essential. The presented measures proved an important positive effect. This indicates that an individual adaption of the system control to the chiller as a function of location, application and configuration offers further potential for improvement. Especially the control of pumps and the heat rejection fan must be studied.

- Evacuated tube collectors provide more useful energy at high temperature levels and thus facilitate higher chilling capacities (See figures 10 and 11, section 3.2) – at the cost of higher expenditures. Especially for those chillers with low minimum driving temperatures flat plate collectors may be favoured.

- Whenever acceptable a solar autonomous summer operation of the solar combi plus system is highly recommended. If a backup is necessary, biomass backup heater or a cold backup with a compression chiller driven with electricity from renewable energy sources should be chosen.

- Solar combi plus systems are most favourable in regions where no cooling demand occurs during the night and where the system has a high degree of utilization per year.

- Chilled ceiling systems are more favourable compared to fan coil systems in terms of chiller performance due to a higher temperature level in the chilled water circuit. However they are more expensive to install and often more difficult to use in reversible mode

In addition we believe that the other outcomes worth mentioning are:

- The static load type used for the simulations does not allow the inertia of the building to be considered and may lead to an underestimation of the cooling potential of the chillers.

- As the chillers work quite differently there are no "standard" conditions, each chiller has its own strength and weaknesses.

- The system setup has a significant influence on the efficiency of the system.

This report does **not** provide:

- a comparison of the different chillers

- a comparison of the two basic system configurations (system scheme C1 and E1, see “Report on methodology for virtual case study”).