THE INFLUENCE OF EXTERNAL DHW MODULES ON THE YEARLY ENERGY CONSUMPTION OF SOLAR COMBISYSTEMS

Florian Ruesch, Elimar Frank

SPF Institut für Solartechnik, University for Applied Sciences Rapperswil (CH)

Abstract

The presence of external domestic hot water modules (DHWM) on the market is growing, amplified by stronger sanitary rules and an increasing popularity of large ‘solar’ storage tanks. In previous studies it has been shown that DHWM provide a potential for energy savings due to the large capacity of flat plate heat exchangers and thus the possibility of a good stratification in the storage tank. These results have been obtained numerically by modeling the parts and the control strategy of the investigated DHWM. In detailed measurements at SPF it has been shown that some modules have difficulties with the regulation of small flow rates, which result in temperature fluctuations and/or an increased DHW temperature. These effects lead to an increased primary return flow temperature, which has a negative influence on the system performance. A series of commercially available DHWM from six different manufacturers were tested extensively. Thereby three different influences on the yearly energy consumption were investigated separately; the electrical energy consumption, the additional heat losses of the module and additional system losses caused by high back flow temperatures. This paper focuses on the influence of the backflow temperature on the annual system performance based on measured values provided by a lookup table. The performance of the control mechanism was found to be most important for the energy saving potential of the tested modules. Based on simulated parameter variations a simplified method for the judgment of real DHWM was elaborated. This method allows predicting the influence of a DHWM to a reference solar combisystem by measuring the backflow temperature at only a few characteristic operation points.

1. Introduction

There is a large variety of technical solutions for providing solar domestic hot water (DHW) and space heat (SH) with so called solar combisystems. The method for preparing the DHW has a strong influence on the energy savings of the whole system. A comparison of the four most important principles (two separated tanks, tank in tank, immersed heat exchangers and external heat exchanger) was carried out by Pauschinger (1997). Following this simulation study, the largest potential for energy savings occurred using external heat exchangers, where the DHW is provided by a unit composed of the heat exchanger, a circulation pump and a control mechanism (the whole unit is called “domestic hot water module”, DHWM). Lorenz, et al. (1997) and Bales and Persson (2003) compared different DHWM for the use in Swedish combisystems and found the highest potential for modules with an electronically controlled variable speed pump. Since then, the demand for DHWM was growing and lot of different modules with variable speed pump and other new control strategies are on the market (Meyer 2009).

Detailed measurements of commercially available DWHM were carried out at SPF in order to develop a test procedure for the comparison of DHWM (Ruesch and Frank 2010). The procedure has been separated in the evaluation of four main performance indicators: power, comfort, efficiency and installation/maintenance. It was shown that several modules had difficulties in regulating the flow rate on the primary side of the heat exchanger during small tapping rates, which results in temperature fluctuations or an increased DHW temperature. The flow regulation strategy of the tested hydro-mechanic module was leading to a DHW temperature which changed as a function of the tapping rate and the temperature of the storage tank water. These effects affect the comfort of the provided DHW. But, as not only the DHW temperature but also the temperature of the back flow to the storage tank is affected, they have a negative influence on the stratification of the tank and the performance of the whole system. This paper is focused on the modelling and simulation of “real” DHWM with control problems and their influence on the performance of the whole system. Other energy relevant parameters, such as the electrical energy consumption or the additional heat losses of the module, are treated in a previous publication (Ruesch and Frank 2010).

2. Method

2.1. Experimental

An experimental set-up able to provide tapping rates up to 50 l/min was designed and installed. The use of a hot storage tank on the *primary sid*e (the side associated with the storage tank or the heat source) and a cold storage tank on the *secondary side* (the side associated with the DHW or the heat sink) allows controlling both input temperatures. A detailed overview of the test rig is given in (Ruesch and Frank 2010).

Experiments were carried out with two different set temperatures for the secondary outlet (the DHW temperature) T_{set}:

45°C	representing the minimal set temperature which is realistic for small systems with energy conscious users.
60°C	representing systems for which this temperature is needed because of regulations concerning DHW hygiene and legionella.

The secondary input temperature (cold water temperature) was fixed to 10 °C. The primary input temperature (temperature of the storage tank) was set to different values above the set temperature to investigate the behavior of the modules in different situations:

T_{set} + 5°C	Minimal realistic temperature difference.
T_{set} + 10°C	Typical temperature difference.
90 °C	Maximal realistic temperature.

Different tapping profiles were carried out to measure the behavior of the DHWM in the most relevant situations. The following tapping rates were defined for a tapping temperature of 42°C for different tapping types:

3 l/min	tapping rate for hand wash
7l/min	tapping rate for a small shower
14l/min	tapping rate for a large shower or bath

The tapping rate at the DHWM is adjusted to reach the above mentioned tapping rates at the tapping points after mixing the warm DHW with cold water of 10 °C to reach a temperature of 42°C. With this definition the secondary side flow rate at the DHWM depends on the reached temperature. When the output temperature of the DHW is 60°C, a flow rate of 4.48 l/min is needed at the module for a small shower.

The main tapping rates of 3,7 and 14 l/min were applied three times during five minutes. The mean in- and output temperatures were measured during the last minute of this tapping cycle. To characterize a DHWM the secondary side hot temperature (DHW temperature) and the "cold temperatures difference" ΔT_{cold} was measured this way. The latter denominates the difference between the primary side cold temperature (the back flow to the storage tank) and secondary side cold temperature (the cold water temperature), which was set to 10 ±1 °C in the experimental set up.

2.2. Simulation

Simulations were carried out in TRNSYS based on the simulation deck developed in the IEA SHC Task 32 and described in Heimrath and Haller (2007). The components with the following characteristics were used for the standard simulation:

DHWM: The DWHM was modeled based on a two dimensional lookup table which can be fed by measured data from each tested module. This model needs measured data of ΔT_{cold} and secondary side output temperatures for different secondary side flow rates and primary hot temperatures. The values between the measured points are interpolated. Temperatures are treated relatively to the secondary side cold temperature (the cold water temperature). Heat losses of the module are not taken into account in the model because they are estimated separately by another procedure. For each set temperature a lookup table was generated based on twelve measured operation points based on the settings described above (2.1). Additionally to the three tapping rates mentioned above, the largest tested tapping rate of 27 l/min was also added to the lookup table.

Storage Tank: $1m^3$ with a DHW part of 150-200l (Auxilliary heating on at T_{DHWset} +5°C off at T_{DHWset} +10°C), another 150 l are used as SH buffer volume (the temperature was set according to the heating curve), total heat loss 11.75 W/K, the solar charging is done with two immersed heat exchangers, the auxiliary heating is directly connected to the storage tank with two in- and outlet heights for DHW an SH charging.

Collector: 10m2 aperture area, heat loss coefficients: a1=3.311; a2=0.012; matched flow rate between 14 and 35 l/hm2.

Auxilliary Heating: modulating and condensing gas burner with 15.6 kW maximum power. Burner model and parameters in accordance to (M. Y. Haller et al 2011).

Building: different heating loads taken from the IEA SHC Task 32 standard model (Heimrath and Haller 2007).

DHW load profile: the load profile was created with the program DHWCalc (Jordan and Vajen) with steps of one minute and a total amount of 200l/day. The standard parameters of the program DHWCalc were use to create the load profile.

Location: Zürich

3. Results

3.1. General Findings

A set of six different commercially available DHWM were tested and ΔT_{cold} was measured for different tapping rates and primary side hot temperatures. ΔT_{cold} was dependent on the regulation type, the quality of the temperature regulation and the capacity of the used heat exchanger. Some different characteristics are illustrated in the following examples. The characteristic of a DHWM with a well adapted primary flow rate is illustrated in Fig. 1. The low temperature differences for small tapping rates are reached by the use of a relatively large heat exchanger (30 plates of $0.05m^2$). A module with a smaller heat exchanger (30 plates of $0.03m^2$) and some temperature fluctuations during small tapping rates is illustrated in Fig. 1Fig. 2. This small module is designed for single family use and is not able to reach the desired secondary side hot temperature (DHW temperature) for the largest illustrated flow rate of 27 l/min. That is also why the primary side cold temperature drops for 27 l/min (for the curve with T primhot =50° and with a smaller extent also the T prim hot = 55°C). The characteristic in Fig. 3 corresponds to a DHWM with a hydro mechanical primary flow regulation, which resulted in higher ΔT_{cold} for small tapping rates and low primary hot temperatures. There are different strategies to deal with the problem of controlling very small primary side flow rates. In the fourth example (Fig. 4) the characteristic of a DHWM is illustrated, which uses an optional 'comfort function' to reduce the temperature fluctuations. This function automatically switched on the integrated circulation pump, as soon as a draw off process is registered. The circulation flow rate adds to the tapping rate to create a larger secondary side flow rate, which is better to handle for the sensors and the microprocessor. But the gain in comfort induced an elevated cold temperature difference and thus has a negative influence on the energetic performance of the complete system.

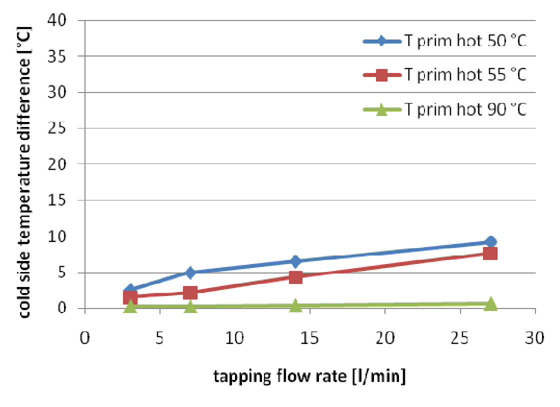

Fig. 1: Measured primary side back flow temperatures for different operating modes and a DHW set temperature of 45°C and a cold water temperature of 10 °C. This characteristic corresponds to a DHWM with a very well working speed control of the primary side pump and a large heat exchanger.

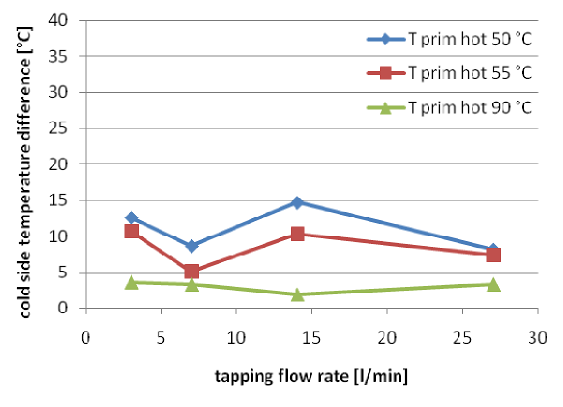

Fig. 2: Measured primary side back flow temperatures for different operating modes and a DHW set temperature of 45°C and a cold water temperature of 10 °C. This characteristic corresponds to a DHWM with a smaller heat exchanger and some fluctuations during small tapping rates.

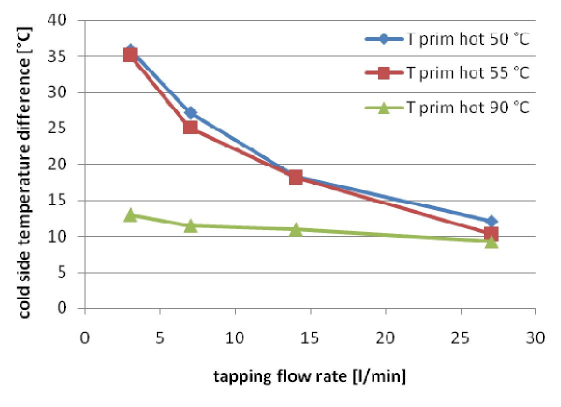

Fig. 3: Measured primary side back flow temperatures for different operating modes and a DHW set temperature of 45°C and a cold water temperature of 10 °C. This characteristic was measured for one DHWM with a mechanical primary flow control. The elevated temperature differences for small tapping rates and low temperatures on the primary side are caused by an elevated primary flow rate.

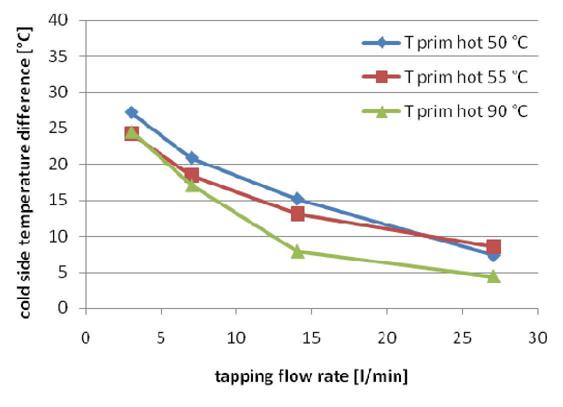

Fig. 4: Measured primary side back flow temperatures for different operating modes and a DHW set temperature of 45°C and a cold water temperature of 10 °C. This characteristic corresponds to a DHWM with the use of a 'comfort function' which switches on the circulation pump on the secondary side during small tapping rates to reduce fluctuations in the tapping temperature.

System simulations were carried out with the measured parameters and compared to the results with the same system, but with an 'ideal' DHWM with an infinite exchanger capacity, where the cold temperature difference is zero for all working points. The difference between the annual auxiliary energy use of the reference system with an ideal DWHM and a real DHWM is called 'additional annual energy consumption'. This number allows a comparison of different control strategies of DHWM and an estimation of the influence of control malfunction to the energy efficiency of the whole combisystem. For DHWM with large heat exchangers and a well adapted flow control (see Fig. 1), these additional losses were in the order of magnitude of 1 % of the total energy consumption for the DHW (see also Tab. 3 for exact numbers). The characteristics of the DHWM given in Fig. 2 results in additional losses of about 3% of the total DHW energy use. Strategies which result in strongly elevated temperature differences at the cold side as displayed in Fig. 3 and Fig. 4 cause an increased additional energy consumption of the combisystem in the range of 9..13% of the total DHW energy use. Changes in the additional annual energy consumption are mainly caused by enhanced storage tank heat losses (responsible for about 45% of the change) and decreased collector output (responsible for about 45% of the change).

3.2. Sensitivity to external parameters

The additional annual energy consumptions presented in the previous paragraph are only valuable for the system described in 2.2. To evaluate the sensitivity to of the findings to changes of the system, some system parameters were changed and the yearly additional annual energy consumption for two cold side temperature differences was simulated. For this reason three simulations were carried out for each system configuration; one with the theoretical ‘ideal’ DHWM, one with the a constant ΔT_{cold} of 10 °C and one with a ΔT_{cold} of 20°C for all characteristic working points in the lookup table. The additional annual energy consumption is plotted as a function of ΔT_{cold} in Fig. 5-8, were the influence of various parameter variations is highlighted.

Neither the exchange of the condensing gas burner with a conventional one, nor the increase of the collector aperture are from 10 m^2 to 15m^2 did affect the relative additional annual energy use caused by the DHWM significantly (see Fig. 5).

An elevated set point of the auxiliary heater and elevated heat losses of the storage tank did affect the influence of the DHWM characteristic. For both of these system configuration changes an elevated additional annual heat loss is observed (see Fig. 6). This tendency is relatively small, but especially for storage tanks with elevated thermal losses a low primary side cold temperature is important.

Different tapping profiles have a large effect on the total energy consumption of the system, and also on the relative additional annual energy use. In Fig. 7 the standard profile is compared to other tapping profiles as the profile with 6 min steps which was used for the Task 32 (Heimrath and Haller 2007), the profile proposed by the EU (EU 2010/30 2010) and another profile generated with the program DHWCalc (Jordan and Vajen), but with the double daily tapping amount of 400l.

Also the operation mode of the solar collectors is important system parameter. Especially for low flow systems a good stratification of the storage tank and a low back flow temperature from the DHWM is important. But also the standard system with a matched flow strategy behaves similar than the low flow system. By changing form low- to high-flow collector operation the influence of the cold temperature difference is dropping by about 30% (see Fig. 8).

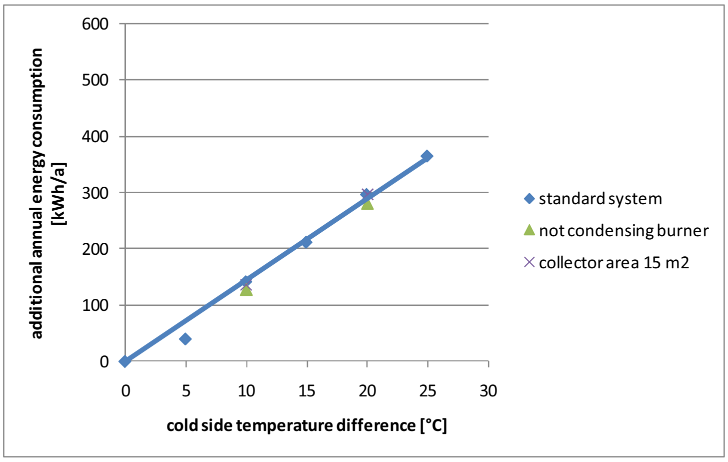

Fig. 5 The additional annual energy consumption of a solar combisystem is plotted as a function of the cold temperature difference of the DHWM. This parameter is almost independent on the collector size and the type of the auxiliary heating.

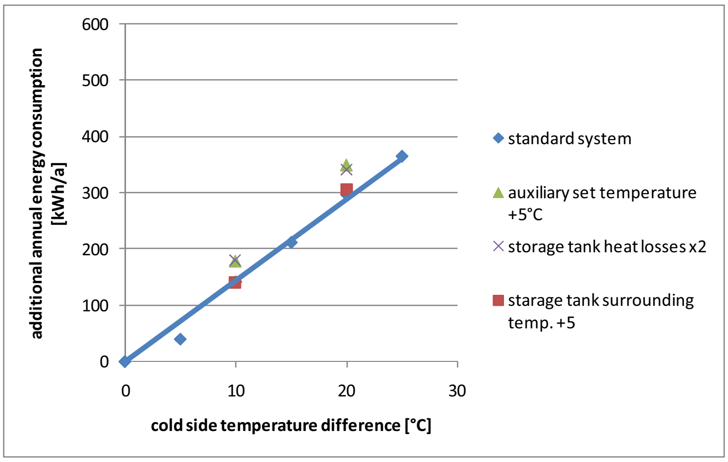

Fig. 6 The additional annual energy consumption of a solar combisystem is plotted as a function of the cold temperature difference of the DHWM. Changing the storage heat losses, the surrounding temperature does affect this measure. By doubling the storage tank heat losses it is increased by about 10%.

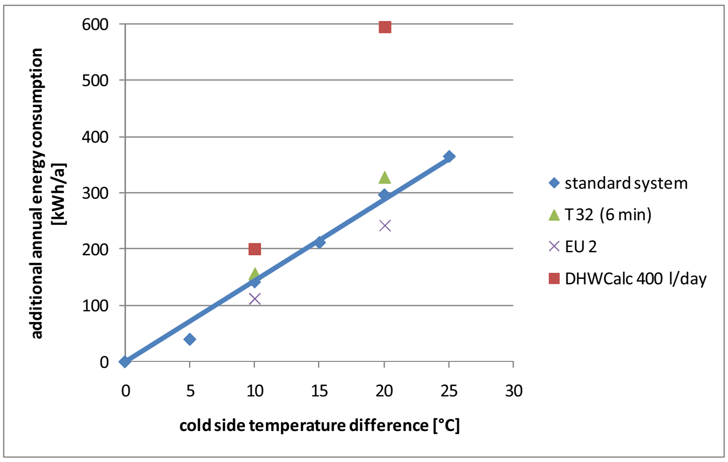

Fig. 7 The additional annual energy consumption of a solar combisystem is plotted as a function of the cold temperature difference of the DHWM. There is a large influence of the tapping profile and the DHW consumption to this parameter. By doubling the DHW consumption from 200l/d to 400l/d also the additional annual energy consumption increases by about this rate.

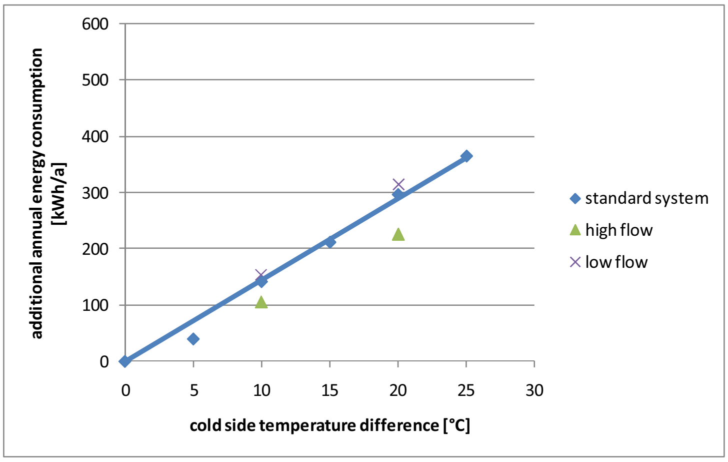

Fig. 8 The additional annual energy consumption of a solar combisystem is plotted as a function of the cold temperature difference of the DHWM. The simulated standard system with a matched flow strategy behaves similar than a low flow system (15 l/hm^2). The influence of the cold temperature difference is less important for high flow systems (35 l/hm^2).

3.3. Simplified method

In the former chapters, system simulations were described, which were used to estimate the additional annual energy consumption of a solar combisystem based on measured in- and output temperatures of real DHWM. However, the use of a system simulation is not desirable in a test method which should be as simple as possible. Yet, the information gained by the yearly system simulations is an important indicator for the performance of a DHWM and should be presented with the test method. Therefore, with the goal of a linearization of the problem, parameter variations were carried out. Based on the 'ideal' DHWM characteristic with no temperature difference on the cold side, each point of the lookup table (ΔT_{cold}) was varied separately between 0..25°C. With these imaginary DHWM characteristics, yearly simulations were carried out to quantify the importance of each operation point for the annual system performance. For each operation point the additional annual energy consumption was plotted as a function of ΔT_{cold} . The plots for a DHW set temperature of 45°C are given in Fig. 9. Even though there is a large difference in the energy use of the total system for the different simulated buildings (heat demand between 15 and 100 kWh/m^2a), the relative increase in the energy use caused by elevated cold temperature differences is similar for all building standards. The strongest influence on additional annual energy consumption can be seen at the working point ($T_{prim\ hot}$=55°C / V_{sec} = 7 l/min), where $T_{prim,hot}$ is the upper set point of the auxiliary heater in the system simulation and V_{sec} is at the closest point to the shower tapping rate in the selected tapping profile. This is the working point with the largest relevance for the energy use of the system. Other working points have only smaller or nearly no influence. Another conclusion from the graphs in Fig. 9 is that the additional annual energy use increases in a linear-like way with the cold temperature difference. In a strongly simplified approach, the results of the different building standards were merged an approximated by a linear fit. The slopes of these fitted straight lines going through zero are then used as linear factors for a simplified estimation of the additional annual energy use. The estimation is done according to the formula:

$Q_a=\Sigma_i F_i * \Delta T_{i,}$ (eq.1)

where i counts over all the working points of the lookup table. As a further simplification, the four most important working points were selected (for a DHW set temperature of 45°C) and the six most important working points for a DHW set temperature of 60°C. These working points are responsible for more than 90% of the additional annual energy consumption. By only using the most important working point a slight underestimation of the additional annual energy consumption is done. For this reason the linear factors of the selected working points where slightly modified to fit the total additional annual energy consumption, when the cold side temperature difference is elevated at all working points. The linear factors of the most important working points are given in Tab. 1 and Tab. 2. These factors are calculated for the above described combisystem and are dependent on the tapping profile, the system configuration and temperature set points of the system. An analysis of the sensitivity to these external parameters was described in chapter 3.2.

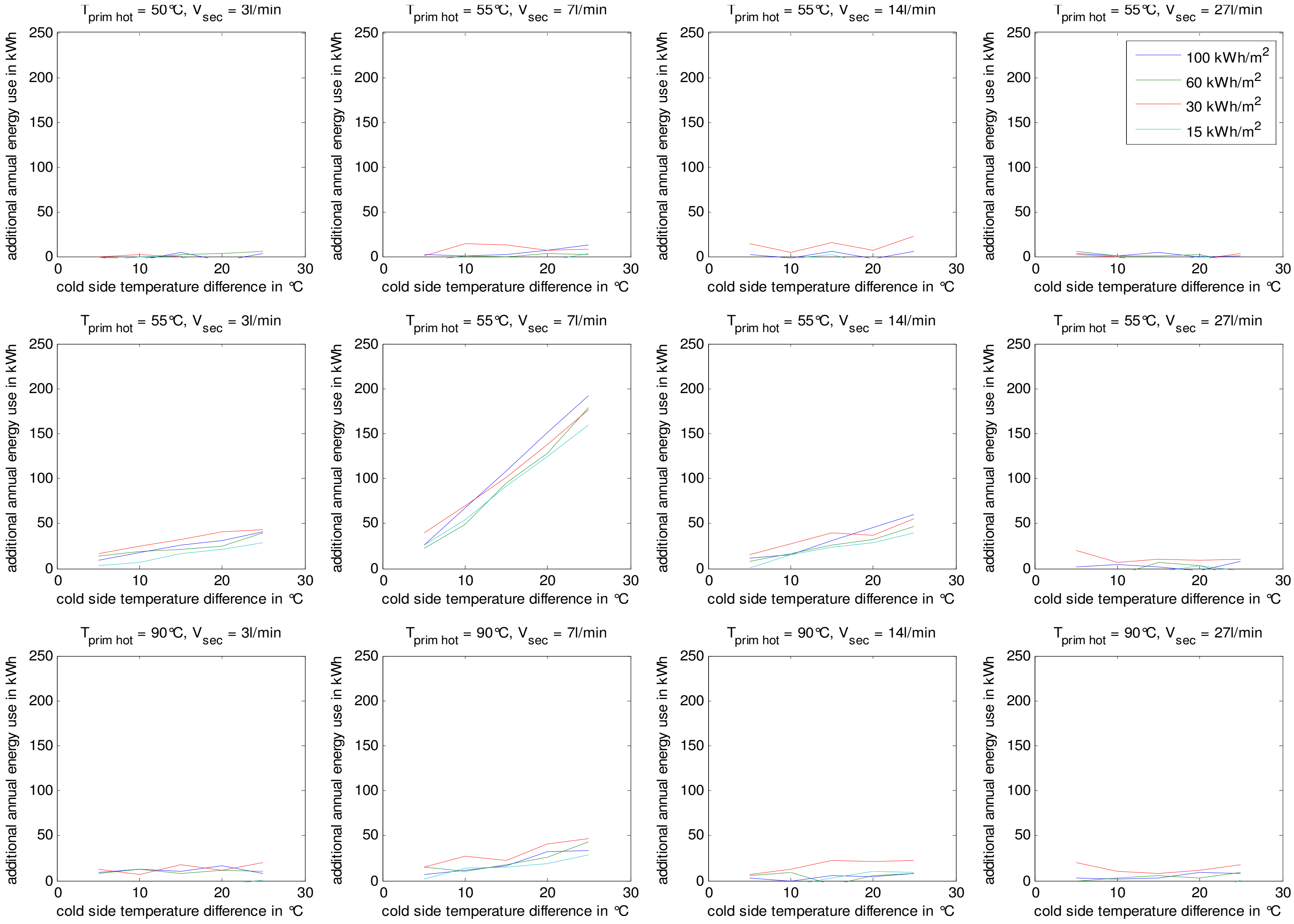

Fig. 9: The influence of elevated cold temperature differences at different working points to the additional annual energy consumption of a solar combisystem. Simulated results for four different building standards ranging from 15..100 kWh/m^2a are plotted as a function of the cold temperature difference.

Tab. 1 Linear factors for the simplified estimation of the additional annual energy use caused by elevated primary side hot temperatures for a DHWset temperature of 45 °C

Number i	$V_{sec42°C}$ [l/min]	$T_{primhot}$ [°C]	Factor $F45_i$ [kWh/(Ka)]
1	3	55	2.0
2	7	55	8.6
3	14	55	2.4
4	7	90	1.8

Tab. 2 Linear factors for the simplified estimation of the additional annual energy use caused by elevated primary side hot temperatures for a DHW set temperature of 60 °C

Number i	$V_{sec42°C}$ [l/min]	$T_{primhot}$ [°C]	Factor $F60_i$ [kWh/(Ka)]
1	3	70	1.1
2	7	70	5.1
3	14	70	1.4
4	3	90	1.0
5	7	90	4.4
6	14	90	1.1

This simplified method was tested with and compared to the measured data from the six tested DHWM. For this modules complete lookup tables were created and annual system simulations were carried out with the above described standard system. Tab. 3 shows a comparison between the results of these system simulations and the results from the simplified method. The accordance of the two methods is good for low additional annual energy consumptions, as reached by module 1. For the modules in the middle range (module 3, 4, 5 and 6a) the simplified method is overestimating the additional annual energy consumption in the range of 5..10 % (relative). Even for the modules causing an elevated additional energy consumption, as module 6b and 7, the accordance between the simplified method and the annual simulation are within a relative difference of 10%. Compared to the differences between the measured DHWM of more than a factor ten, the differences between simplified method and annual system simulation are still relatively small. For this reason the simplified method seems well suited to compare different tested DHWM while decreasing the effort significantly.

Tab. 3 Comparison of the additional annual energy use caused by elevated primary side cold temperature (backflow to the storage tank) evaluated once with a complete system simulation and once with the simplified Method described above.

Module	Flow control	Simulation [kWh/a]	Simplified Method [kWh/a]
1	Microcontroller	31	32
2	Microcontroller	67	72
3	Microcontroller	58	62
4	Microcontroller	92	96
5	Microcontroller	98	109
6a	Microcontroller	79	87
6b	Microcontroller	277	269
7	Hydromechanic	390	351

4. Conclusions and outlook

The primary flow control characteristic of the DHWM is an important parameter to the annual performance of solar combisystems. Different tested modules resulted in estimated additional annual energy consumptions in the range of 31..390 kWh/a. DHWM with large heat exchangers and a good working flow control can reach results close to theoretically ideal infinite heat exchangers. Elevated primary back flow temperatures were observed for a DHWM which uses the circulation pump for a comfort function and for a DHWM with a hydro mechanical regulation. In the first case the increase in comfort causes addition energy use of 9% of the total energy needed for DHW preparation only caused by the elevated primary side cold temperature. The additional energy losses of the circulation piping are not taken into account for this value. In the case of the hydro- mechanically controlled module (7) the additional annual losses are reaching more than 10% of the energy used for DHW preparation.

A simplified method was introduced to estimate the additional annual energy consumption based on a set of linear factors. The goal of this method is to quantify the influence of different measured characteristics of DHWM to the performance of a solar combisystem without the need of an annual system simulation. For a set of 6 commercially available DHWM the results of the simplified method were compared to detailed annual system simulations. The largest difference between the two methods was 10 % (relative) for additional annual energy consumption which differed between 31..390 kWh/a. As there is a large variety of possible system configuration and user comportment, which have a large influence on the system performance, such quantification is not valuable for all possible combisystems. But, as the influence of the different tested DHWM to the performance of the combisystem differed by more than a factor ten, the simplified method can be used to compare different DHWM and give estimation of the order of magnitude. Following the results presented in this paper, the simplified method will be used in the test procedure for DHWM that is currently being developed and tested by the authors.

5. Acknowledgements

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