CombiSol project

Solar Combisystems Promotion and Standardisation

D3.1 : Comparison of test methods

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<th>Unit</th>
<th>Definition</th>
</tr>
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<tbody>
<tr>
<td>a₁</td>
<td>W/(m² K)</td>
<td>Heat transfer coefficient of the collector</td>
</tr>
<tr>
<td>a₂</td>
<td>W/(m² K²)</td>
<td>Temperature dependant heat transfer coefficient of the collector</td>
</tr>
<tr>
<td>A_c</td>
<td>m²</td>
<td>Collector area</td>
</tr>
<tr>
<td>C</td>
<td>kJ/(m² K)</td>
<td>Specific heat capacity of the collector</td>
</tr>
<tr>
<td>Csto</td>
<td>kJ/K</td>
<td>Thermal capacity of the whole store</td>
</tr>
<tr>
<td>d_off</td>
<td>day</td>
<td>Time-shift parameter</td>
</tr>
<tr>
<td>E₁2day</td>
<td>kWh</td>
<td>Energy demand / delivery of the 12 day test sequence</td>
</tr>
<tr>
<td>E_annual</td>
<td>kWh/a</td>
<td>Annual energy demand</td>
</tr>
<tr>
<td>E_aux</td>
<td>kWh/a</td>
<td>Primary energy demand</td>
</tr>
<tr>
<td>E_col</td>
<td>kWh/a</td>
<td>Annual global energy available in collector plane</td>
</tr>
<tr>
<td>E_glob,K</td>
<td>kWh/m²a</td>
<td>Annual global irradiation in collector plane</td>
</tr>
<tr>
<td>f_sav</td>
<td>%</td>
<td>Fractional energy saving</td>
</tr>
<tr>
<td>f_sol</td>
<td>%</td>
<td>Solar fraction</td>
</tr>
<tr>
<td>Q_aux,net</td>
<td>kWh/a</td>
<td>Net auxiliary heat demand of a solar combisystems</td>
</tr>
<tr>
<td>Q_aux</td>
<td>kWh/a</td>
<td>Auxiliary energy demand (including an annual boiler efficiency)</td>
</tr>
<tr>
<td>Q_col</td>
<td>kWh/a</td>
<td>Annual collector energy gain</td>
</tr>
<tr>
<td>Q_conv</td>
<td>kWh/a</td>
<td>Heat demand of a conventional (non solar) heating system (including an annual boiler efficiency)</td>
</tr>
<tr>
<td>Q_conv,net</td>
<td>kWh/a</td>
<td>Net Heat demand of a conventional (non solar) heating system</td>
</tr>
<tr>
<td>Q_CTSS</td>
<td>kWh/a</td>
<td>Heat demand / gain determined with the CTSS method</td>
</tr>
<tr>
<td>Q_SCSPT</td>
<td>kWh/a</td>
<td>Heat demand / gain determined with the SCSPT method</td>
</tr>
<tr>
<td>Q_d</td>
<td>kWh/a</td>
<td>Overall heat demand for space heating and domestic hot water</td>
</tr>
<tr>
<td>Q_d,shaw</td>
<td>kWh/a</td>
<td>Heat demand for domestic hot water preparation</td>
</tr>
<tr>
<td>Q_d,sh</td>
<td>kWh/a</td>
<td>Heat demand for space heating</td>
</tr>
<tr>
<td>Q_solar</td>
<td>kWh/a</td>
<td>Heat delivered by the solar collector into the store</td>
</tr>
<tr>
<td>Q_loss</td>
<td>kWh/a</td>
<td>Heat losses of the system</td>
</tr>
<tr>
<td>Q_store,conv</td>
<td>kWh/a</td>
<td>annual store heat losses of a reference (non solar) heating system</td>
</tr>
<tr>
<td>Q_ref</td>
<td>kWh/a</td>
<td>Heat demand of a reference system without solar</td>
</tr>
<tr>
<td>ΔT</td>
<td>K</td>
<td>Temperature difference</td>
</tr>
<tr>
<td>time</td>
<td>h</td>
<td>Hour of the year</td>
</tr>
<tr>
<td>UA_s,a</td>
<td>W/K</td>
<td>Heat loss rate of the store (standby)</td>
</tr>
<tr>
<td>V_sto</td>
<td>l</td>
<td>Effective storage volume</td>
</tr>
<tr>
<td>V_dhw,hx</td>
<td>l</td>
<td>Volume of the heat exchanger for domestic hot water preparation</td>
</tr>
<tr>
<td>V_s,hx</td>
<td>l</td>
<td>Volume of the solar heat exchanger</td>
</tr>
<tr>
<td>W_el</td>
<td>kWh/a</td>
<td>Parasitic electrical energy consumption</td>
</tr>
</tbody>
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### Greek Letters

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta_0$</td>
<td>-</td>
<td>Conversion factor of the collector</td>
</tr>
<tr>
<td>$\eta_{aux}$</td>
<td>%</td>
<td>Annual utilization rate of the boiler of the combsystem</td>
</tr>
<tr>
<td>$\eta_{conv}$</td>
<td>%</td>
<td>Annual utilization rate of the boiler of a conventional (non-solar) heating system</td>
</tr>
<tr>
<td>$\eta_{sys}$</td>
<td>%</td>
<td>System efficiency</td>
</tr>
<tr>
<td>$\vartheta$</td>
<td>°C</td>
<td>Temperature</td>
</tr>
<tr>
<td>$\vartheta_{average}$</td>
<td>°C</td>
<td>Yearly average cold water temperature</td>
</tr>
<tr>
<td>$\vartheta_{cw}$</td>
<td>°C</td>
<td>Cold water temperature</td>
</tr>
<tr>
<td>$\Delta \vartheta$</td>
<td>K</td>
<td>Average amplitude for seasonal variation in cold water temperature</td>
</tr>
</tbody>
</table>

### Abbreviation

<table>
<thead>
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<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>CCT</td>
<td>Concise Cycle Test</td>
</tr>
<tr>
<td>CTSS</td>
<td>Component Testing System Simulation</td>
</tr>
<tr>
<td>CSTG</td>
<td>Complete System Testing Group</td>
</tr>
<tr>
<td>DHW</td>
<td>Domestic hot water</td>
</tr>
<tr>
<td>DST</td>
<td>Dynamic System Test</td>
</tr>
<tr>
<td>IEA SHC</td>
<td>International Energy Agency, Solar Heating and Cooling Programme</td>
</tr>
<tr>
<td>LDL</td>
<td>Load discharge loop</td>
</tr>
<tr>
<td>SCS</td>
<td>Solar combisystem</td>
</tr>
<tr>
<td>SCSPT</td>
<td>Short Cycle System Performance Test</td>
</tr>
<tr>
<td>SSTG</td>
<td>Solar Storage Testing Group</td>
</tr>
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1 Introduction

The main objective of WP 3 is the comparison of two different methods for the performance testing of solar combisystems. The first method is the so-called CTSS method (CTSS: Component testing – system simulation) specified in the standard CEN/TS 12977-2. It is based on a component orientated testing approach and an annual system simulation to obtain the annual performance of the system.

The second method, the short cycle system performance test (SCSPT), uses a global approach. It is based on the CCT – approach (CCT: concise cycle test) but has been further developed by CEA/INES, France. The SCSPT method is based on a physical test of the whole system (except the collector field) and an extrapolation of the test results to obtain the annual performance of the system. For this testing approach no standard has been elaborated yet.

Within the CombiSol project three different designs of solar combisystems have been tested according to these test methods. The testing according to the CTSS method has been performed at the Institute of Thermodynamics and Thermal Engineering (ITW) at the University of Stuttgart, Germany, the testing according to the SCSPT method has been performed at the Institute National de l’Energie Solaire (INES), Chambery, France. For the comparison of the test methods, two of the three systems have been tested according to both test methods.

In this report the two test methods for testing solar combisystems will be described in detail. For the solar combisystems tested, the results of the thermal performance test determined according to the CTSS method and SCSPT method will be presented and discussed.

2 Component Testing - System Simulation (CTSS)

The CTSS method consists on a physical test of the main components of the solar combisystems and on an annual system simulation in order to obtain the annual thermal system performance. In Fig. 2.1 the principal structure of the CTSS method is depicted:
The water store or combistore is tested according to EN 12977-3:2008 or CEN/TS 12977-4:2010 respectively, the controller according to CEN/TS 12977-5:2010 and the collector according to EN 12975-2:2006. Within the test the parameters characterizing the thermal performance of the components are determined. Based on these parameters the thermal performance of the complete system is predicted by using a component based system simulation program such as TRNSYS. The annual system simulation can be carried out for different reference conditions such as meteorological data and load profiles.

The range of application of the CTSS method is very flexible due to its component-oriented testing approach. Hence it is possible to apply the CTSS method on nearly every system configuration of solar combisystems.

The standard CEN/TS 12977 is currently under development and it is expected that in 2012 it will become the status of a European standard. In Table 2.1 an overview of the CEN/TC 312 standards is given. The standard EN 12976 describes requirements and test methods for factory made solar thermal systems. This standard is only applicable to solar domestic hot water systems. The thermal performance of the system is determined according to EN 12976-2:2006 by applying the DST-method (DST = Dynamic System Test, ISO 9459-5:2007) or by using the CSTG-method (CSTG = Complete System Testing Group, ISO 9459-2:1995).

In the following sections a short description of the test methods defined in the standard EN 12977-2 is given.

Table 2.1: Titles of the CEN/TC 312 standard

<table>
<thead>
<tr>
<th>Number</th>
<th>Title “Thermal solar systems and components-….”</th>
</tr>
</thead>
<tbody>
<tr>
<td>EN 12975-1</td>
<td>Collectors. Part 1: General Requirements</td>
</tr>
<tr>
<td>EN 12975-2</td>
<td>Collectors – Part 2: Test Methods</td>
</tr>
<tr>
<td>EN 12976-1</td>
<td>Factory Made Systems. Part 1: General Requirements</td>
</tr>
<tr>
<td>EN 12976-2</td>
<td>Factory Made Systems. Part 2: Test Methods</td>
</tr>
<tr>
<td>CEN/TS 12977-1</td>
<td>Custom Built Systems. Part 1: General Requirements for solar water heaters and combisystems</td>
</tr>
<tr>
<td>EN 12977-3</td>
<td>Custom Built Systems. Part 3: Performance test methods for solar water heater stores</td>
</tr>
<tr>
<td>CEN/TS 12977-5</td>
<td>Custom built systems. Part 5: Performance test methods for control equipment</td>
</tr>
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</table>
2.1 Collector test according to EN 12975

The collector test is performed according to the European standard EN 12975. In the scope of this standard are covered and uncovered collectors which operate with a fluid as heat transfer medium.

In the first part of the standard EN 12975-1, requirements for solar thermal collectors regarding safety, reliability and durability are specified. Their objective is to ensure that the systems operate reliably, even under extreme conditions such as heavy snow or wind loads or extended stagnation periods during the summer. In addition, also the documentation of the system and the installation and operation manuals have to fulfil certain requirements in order to assure a correct installation and operation by the installer and owner, respectively.

In the second part of the standard, EN 12975-2, the test procedures for reliability and durability testing as well as for the determination of the thermal performance are described. The aim of performance testing of solar collectors is the determination of characteristic parameters which allow the description of the thermal behaviour of the collector. The knowledge of these parameters is essential for the prediction of the yearly energy output of the collector itself or of a complete thermal solar system. The required measurement data may be gathered either by outdoor tests or by using a solar simulator (see Fig. 2.2).

![Outdoor test facilities](image1)

![Solar simulator](image2)

Fig. 2.2: Outdoor test facilities of the “Research and Test Centre for Solar Thermal Systems” (TZS) at ITW, University of Stuttgart (left)
Solar simulator at the ITW, University of Stuttgart (right)

For the characterisation of the thermal performance, the EN 12975-2 permits a collector test under steady-state conditions and under quasi-dynamic conditions. During the steady state test, all boundary conditions such as solar radiance, ambient temperature and collector inlet temperature shall be constant. After recording data points over a representative range of operating conditions, the collector efficiency curve can be determined by means of multi-linear regression using the least square method.

During the quasi dynamic test the boundary conditions have to vary. Based on a series of measurements, specific collector parameters are determined. The quasi dynamic test method allows the determination of additional parameters such as the heat capacity of the collector and the incident angle modifier coefficient in addition to the efficiency curve.

The advantage of the quasi-dynamic method is that it allows for a much wider range of test conditions. This leads to easier and less expensive tests, especially for places with varying climate conditions such
as in Northern and Middle Europe. Furthermore, the quasi-dynamic test method offers a much more complete characterisation of the collector and a much wider range of collectors can be tested (FISCHER et al 2001).

2.1.1 Description of the collector test facility at ITW

At ITW, the solar collectors can be tested in an outdoor or indoor test facility. The outdoor test facility allows the testing of six collectors in parallel with a collector area of up to 6 m² each. Two solar simulators allow an indoor testing of solar collectors with a maximum area of 10 m² and a maximum irradiation of 1200 W/m². The hydraulic set up, the measurement equipment and the data recording is in accordance with the standard EN 12975-2.

2.2 Testing of hot water stores and solar combistores according to EN 12977 – 3 and CEN/TS 12977 – 4

In the standard EN 12977-3:2008 the thermal performance test method for hot water stores is described. The thermal performance test method for solar combistores is described in the technical specification CEN/TS 12977-4:2010. This test method is mainly based on the EN 12977-3 with a few additional elements.

With regard to the main aspect of the standard - the determination of parameters for the description of the thermal performance of the stores - two completely different test and evaluation procedures are described in EN 12977-3. They can be called 'classical' and 'advanced' procedure (see Fig. 2.3).

With the classical test and evaluation procedure the most elementary store parameters can be determined. The method for the determination of the thermal capacity and the heat loss rate is based on the work carried out by the European Solar Storage Testing Group (SSTG) at the end of the eighties (VISSER et. al. 1991). The method for the determination of the heat transfer rate of immersed heat exchangers was developed by J. M. Suter (SUTER et al. 1987).

The advanced procedure allows a more detailed characterisation of the thermal behaviour of the store by using a numerical model of the store. The parameters of this model are determined by means of parameter identification on the basis of measured data of several test sequences.

The two procedures differ in both the measured data (test sequences) and the evaluation procedure. With regard to the physical test of the store, the two procedures are based on measured data which are gained during different test sequences. It is important to mention that both procedures are not based on measurements inside the store. The data required for the classical procedure are gained by
measurements on a store test facility. The data for the evaluation with the advanced procedure can be
determined either from tests carried out on a store test facility or gained during a system test according
to ISO 9459-5 (Dynamic System Test, DST) with additional sensors in the collector loop.

With regard to the evaluation, the main advantage of the **classical** method is the transparent evaluation
procedure which is based on analytical energy and power balances. With this method it is possible to
determine the thermal capacity and the heat loss rate of the store as well as the heat transfer rate of
immersed heat exchangers. The main advantage of the classical method is that the determination of
store parameters is possible without using a numerical model of the store. However, since the dynamic
behaviour and thermal stratification effects cannot be assessed with this method, a detailed
characterisation of the store is not possible.

With the **advanced** procedure the thermal behaviour of the store can be characterised in detail by
using a numerical model of the store such as the MULTIPORT store model for TRNSYS (DRÜCK
2000). The parameters of this model are determined by means of parameter identifications on the basis
of measured data of several test sequences. This advanced test procedure was mainly developed by
ITW within several research projects such as e.g. IEA SHC, Task 14 (DRÜCK et. al. 1997). (IEA
is more flexible and even cheaper than the classical one. Furthermore a more detailed description and
characterisation of the store is possible.

### 2.2.1 Description of the store test facility from ITW

At ITW two test facilities located in climate-controlled test laboratories exist in order to perform the
store test according to EN 12977-3 and CEN/TS 12977-4. The test facilities consist of four modules:
Two modules are used for charging and two modules are used for discharging the store. The charge
modules can be used for charging with constant temperature or constant thermal power (up to 22 kW).
The discharge modules are supplied with cold water from a 2000 litres buffer store.

### 2.3 Testing of controllers according to CEN/TS 12977-5

Testing the function of controllers and temperature sensors is described within the Technical
Specification CEN/TS 12977-5:2010. The test procedure is divided into a mandatory test and an
optional part. Within the mandatory test the capability of temperature sensors to resist high
temperatures as well as possible differences of the sensor’s accuracy before and after the exposure is
determined. In addition a visual inspection of all sensor parts that had been exposed to the high
temperatures is demanded.

The optional test encompasses the following tests:

- Testing of the functioning of differential thermostats with the objective to investigate starting
  and stopping of pumps or switching of valves depending on the temperature levels within the
  system.
- Comparison of all indications and functions with the guidance delivered by the manufacturer.
- Documentation of the sensitivity of the starting and stopping differentials within a certain
  range of the nominal mains voltage (e. g. 230 V –10 % to 230 V +6 %).
2.3.1 Description of the controller test facility from ITW

In order to perform the test sequences according to CEN/TS 12977-5, the test facility consists of an input/output-emulator which is connected to the sensor and the output terminals of the controller to be tested. For communication, the emulator is connected to a PC via a serial port. The PC is equipped with specific software, providing temperature profiles through the emulator to the relevant sensor terminals of the controller. At the same time the emulator transfers the response of the controller to the PC.

For each single step of a temperature profile that is transferred to the controller by means of the emulator, the status of all outputs, whether active or inactive is detected and transferred back to the PC. In parallel to the temperature profile the corresponding response of the controller is stored in a data file. In case of controllers featuring variable mass flows, e.g. by pulsing the circulation pump, a pump installed in a hydraulic circuit is connected to the particular output. To adjust the pressure drop of the hydraulic circuit according to the real pressure drop of the collector loop and to flush the device the circuit is equipped with several valves. In addition, manometers and a magnetic-inductive flow meter are mounted.

2.4 Long-term performance prediction

The annual thermal performance of a solar combisystem is determined by modelling the system with a detailed transient simulation programme (e.g. TRNSYS). The component models and the model parameters for collector and store used in the system simulation have to be respectively the same as for the characterization of the collector (according to EN 12975-2) and for the combistore (according to CEN/TS 12977-4). The control concept of the system tested according to CEN/TS 12977-5 is implemented.

The annual thermal performance of the system is calculated for reference conditions according to CEN/TS 12977-2. For four reference climates (Stockholm, Wurzburg, Davos, Athens) the heating load is defined in load file in which the flow and return temperature and the mass flow rate is given in an hourly time step. The heating load is representative for the heat demand of a typical one family house in the corresponding country. The heat demand for domestic hot water (indicated in volume / day) is calculated by using a seasonal and climate dependant cold water temperature. The draw-off is performed 6 hours after solar noon at a (mixing) temperature of 45 °C.

The performance indicators characterizing the thermal performance of a solar combisystem are:

- heat demand for domestic hot water $Q_{d,hw}$ in kWh/a
- heat demand for space heating $Q_{d,sh}$ in kWh/a
- net auxiliary energy demand $Q_{aux,net}$ in kWh/a
- parasitic energy demand $W_{par}$ in kWh/a
- fractional energy saving $f_{sav}$ in %
- solar fraction $f_{sol}$ in %
- system efficiency $\eta_{sys}$ in %

The fractional energy savings compares the energy demand of a conventional (non solar heating) system to the auxiliary energy demand of the solar thermal system and is calculated with equation 2.1:
\[ f_{\text{sol}} = \frac{Q_{\text{conv}} - Q_{\text{aux}}}{Q_{\text{conv}}} \cdot 100 \]

In the equation is:

- \( Q_{\text{conv}} \) the energy demand in kWh/a of a conventional (non solar) heating system taken into account an annual utilization factor of a conventional boiler \( \eta_{\text{conv}} = 0.75 \) and annual heat losses of a reference store \( Q_{\text{ref,conv}} \).

\[ Q_{\text{conv}} = \frac{Q_d + Q_{\text{loss,ref}}}{\eta_{\text{conv}}} \]

\( Q_d \) is the overall heat demand for space heating and domestic hot water (\( Q_{d,sh} + Q_{d,hw} \)) in kWh/a.

- \( Q_{\text{aux}} \) the auxiliary energy demand in kWh/a taken into account an annual utilization factor of the auxiliary boiler of \( \eta_{\text{aux}} = 0.75 \)

\[ Q_{\text{aux}} = \frac{Q_{\text{aux,net}}}{\eta_{\text{aux}}} \]

The solar fraction is defined as the energy supplied by the solar part of the system (\( Q_l \) in kWh/a) divided by the total system load (\( Q_d \) in kWh/a) and is calculated with equation 2.4:

\[ f_{\text{sol}} = \frac{Q_l}{Q_d} \cdot 100 \]

The system efficiency \( \eta_{\text{sys}} \) is calculated with equation 2.5:

\[ \eta_{\text{sys}} = \frac{f_{\text{sol}} \cdot Q_{\text{conv,net}}}{E_{\text{glob,K}} \cdot A_c} \cdot 100 \]

In the equations the following abbreviations are used:

- \( E_{\text{glob,K}} \): Global irradiation in collector plane in kWh/(m² a)
- \( A_c \): Collector area in m²

3 Short Cycle System Performance Test

*Main Authors: M. Albaric, A. Leconte (INES CEA)*

3.1 Description of the SCSPT method and elaboration of the test sequence used

The short cycle system performance test (SCSPT) is based on the CCT method (BALES 2000; HALLER et. al 2007; NARON et al. 2002, VOGELSANGER 2002). The complete system (excluding the collector field) is set up in an indoor test facility. The building is simulated on-line so that the heat supply is controlled via the controller of the solar combisystem.
The testing consists of a 32 hours preconditioning phase, a 12 days core phase and an 8 hours discharge of the store (cf. Table 3.1).

Table 3.1: Description of the test phase of the short cycle system performance test

<table>
<thead>
<tr>
<th>Number</th>
<th>Phase</th>
<th>Duration</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Initial conditioning</td>
<td></td>
<td>Conditioning of the storage to 20 °C (without solar and auxiliary energy)</td>
</tr>
<tr>
<td>2</td>
<td>Primary conditioning</td>
<td>8</td>
<td>Upper and lower part of the storage has to be brought to reasonable temperatures. Upper part is heated to the auxiliary set point temperature.</td>
</tr>
<tr>
<td>3</td>
<td>Secondary conditioning</td>
<td>24</td>
<td>Final conditioning with the simulation of one winter day. It permits to bring the storage to an energy level which corresponds to the last day of the core phase.</td>
</tr>
<tr>
<td>4</td>
<td>Core phase</td>
<td>288</td>
<td>12 test sequence days with climate and load simulation</td>
</tr>
<tr>
<td>5</td>
<td>Final discharge</td>
<td>8</td>
<td>Discharge of the store</td>
</tr>
</tbody>
</table>

The 12 days of the core phase of the testing represent the average weather and load conditions of a whole year. During the elaboration phase of the SCSPT, weather data have been obtained in an iterative optimization process in which the predicted annual values of a reference system, evaluated from a 12 day simulation, are compared with the values of an annual system simulation (cf. Fig. 3.1), (ALBARIC 2008). The predicted results from the 12 day simulation are extrapolated to a complete year. The extrapolated results have to correspond to the results of the annual simulation in terms of space heating demand, domestic hot water demand and internal energy of the store. The annual performance of the system is predicted by extrapolating the 12 days test results to a complete year.
For the determination of the 12 day weather data file, the annual system simulation and the 12 day system simulation were performed for the reference solar combisystems used in the IEA SHC task 32 under the following boundary conditions:

- Building SFH60, space heating demand of 60 kWh/a at the location of Zurich
- Climate: Zurich
- Domestic hot water demand: 200 l/day
- Collector area: 12 m$^2$
- Store volume: 600 l

The selected days for the Zurich test sequences are listed in the following table.

<table>
<thead>
<tr>
<th>Day of the 12 day test sequence</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
<th>12</th>
</tr>
</thead>
<tbody>
<tr>
<td>Selected day of the annual weather data file</td>
<td>20</td>
<td>57</td>
<td>298</td>
<td>116</td>
<td>198</td>
<td>228</td>
<td>244</td>
<td>259</td>
<td>134</td>
<td>57</td>
<td>26</td>
<td>328</td>
</tr>
</tbody>
</table>

In a next step, further simulations, 12 day simulation with the 12 day weather data file (cf. Table 3.2) and annual system simulation with the annual weather data file, have been performed for different configurations of solar combisystems and for different building types.
The reference solar combisystem of the IEA SHC program Task 32 with different collector sizes and store volumes

- One commercial solar combisystem storing heat within a water store
- One commercial solar combisystem using the thermal mass of the building as heat storage
- Different buildings with various heating loads of 30 kWh/m² (referred as SFH30) and 100 kWh/m² (referred as SFH100) in Zurich conditions.

The extrapolated results of the 12 day simulation have been compared to the results of the annual system simulation. In Fig. 3.2 the prediction of the primary energy used by the boiler from the 12 day test sequence is plotted against the values obtained from the annual system simulation. The deviation between both values is in the range of +/- 10%.

Once the test sequence elaborated and validated on various SCS, the same test sequence is used for all the SCS and is a fixed boundary condition of the SCSPT method.

![Graph showing comparison of primary energy consumption](image)

**Fig. 3.2:** Comparison of the prediction of the primary energy used by the boiler obtained from the annual system simulation and the 12 day system simulation for different boundary conditions.

### 3.2 Description of the semi virtual test facility from INES

The test facility allows a thermal performance test of various heating or cooling systems for different applications such as single family house, small industry, small tertiary or small collective housing by creating a semi-virtual environment around the tested system.

It mainly consists of:

- A central heating room able to supply hot and cold water on two distribution loops
• Test modules of 25 kW able to reproduce the desired dynamical thermal loads within these range of temperatures
• Test module of 50 kW able to simulate domestic hot water draw off

3.2.1 The central heating room

The central heating room is mainly composed of:

• One 150 kW chiller able to produce cold water at -12 °C connected to one 5000 l water store in order to stabilize the cold temperature in the distribution loop.
• One 54 kW electrical heater able to produce hot water at 180°C.

3.2.2 The 25 kW test module

Four 25 kW test modules are available and are able to emulate both the energy supply and the load depending on the configuration of the installation and the period of the test sequence. An example: the collector loop can be energy supplier during the days and consumer during the recooling mode. In summer when the storage reaches its maximum available temperature the collectors can be used to cool the store.

The modules are connected to the hot and cold water loop and draw energy depending on their needs.

The characteristics of the 25kW modules are:

• Temperature range: from -10 to +170 °C
• Flow rate range: from 100 l/h to 3600 l/h
• The maximal power of 25kW with a temperature difference of 10 °C between the primary and secondary loop

3.2.3 The 50 kW test module

The 50 kW module is dedicated to the domestic hot water draw off. It has one part for the fresh water preparation and one part for the domestic hot water draw off.

The temperature of the fresh water is prepared according to the set point temperature desired. The fresh water is stored in a 100 litres water store. The configuration covers temperature range from 2 °C to 25 °C. The domestic hot water draw off can be varied from 1 l/min to 64 l/min.

3.2.4 The control / command of the test facility

The communication between the modules and the supervision is realized using Ethernet network. Concerning the electronic equipment, National Instruments Compact Field Point modules allow:

• Temperature recording (PT100)
• Flow rate recording (electromagnetic or coriolis flow meter depending on the module)
• The control and the command of the pump and the valves of each module
• Measurement of the electricity consumption of the auxiliary of the tested system (pumps, …)
• Measurement of the electricity consumption of the auxiliary heater
• Emulation of the ambient and room temperature sensors with a variable resistance box

All these equipments are controlled with a LabView software which allows:

• Data transfer between the equipments
• Control of the TRNSYS software
• Real time visualization of the test sequence
• Real time calculation of all energy fluxes

![Diagram of the supervision system]

**3.2.5 Summary of the characteristics of the INES test facility**

The main characteristics of the INES test facility are summarized in Table 3.3:

<table>
<thead>
<tr>
<th>Table 3.3: Summary of the characteristic of the INES test facility</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Unit</strong></td>
</tr>
<tr>
<td>Number of load / discharge loops (LDL)</td>
</tr>
<tr>
<td>LDL outlet temperature</td>
</tr>
<tr>
<td>LDL flow rate</td>
</tr>
<tr>
<td>LDL heating power</td>
</tr>
<tr>
<td>LDL cooling power</td>
</tr>
<tr>
<td>DHW draw off</td>
</tr>
<tr>
<td>Fresh water temperature</td>
</tr>
</tbody>
</table>
Three commercial systems have been tested according to the CTSS method. The generic hydraulic system concept of the solar combisystems according to (THÜR 2009) is:

- Solar combisystems 1: B1 – Immersed DHW heat exchanger, auxiliary as return flow increase
- Solar combisystems 2: A1 – Tank in Tank, auxiliary as return flow increase
- Solar combisystems 3: C2 – external DHW preparation, heat store as buffer store

The solar combisystems 1 and 2 have also been tested according to the SCSPT method (cf. chapter 5). The solar combisystems 3 has only been tested according to the CTSS method.

The thermal performance of the solar combisystems is characterized by the following parameters:

- fractional energy saving \( f_{\text{sav}} \) in % (cf. equation 2.1).
- solar fraction \( f_{\text{sol}} \) in % (cf. equation 2.4)
- system efficiency \( \eta_{\text{sys}} \) in % (cf. equation 2.5)
4.1 Reference condition for the thermal performance characterization

The annual energy gain of the combisystems is calculated for reference weather data and standardized load profiles (hot water draw off and heating load). According to CEN/TS 12977-2, Annex A, the annual system simulation was performed for the following boundary conditions:

- **Building:**
  One family house with 128 m² heated floor area
- **Location:** Wurzburg
- **Collector area oriented south, 45 ° tilted**
- **20 m of total piping length in the collector circuit (10 m each)**
  piping located indoor
- **Hot water demand:** 200 l/d at 45 °C (2945 kWh/a)
  seasonal dependant cold water temperature of 10 °C +/- 3 °C (cf. equation 4.1)
- **Heating load:** 9090 kWh/a (70 kWh/(m² a))
  max. flow / return temperature: 50 °C / 30 °C
  night shut-off from 23\(^{00}\) - 5\(^{00}\)
- **Ambient temperature of the store:** 15 °C
- **Heat demand of a conventional (non solar) heating system:**
  12679 kWh/a (including heat losses of a conventional store of 644 kWh/a)

The seasonal dependant cold water temperature \( T_{cw} \) is calculated according to equation 4.1

\[
T_{cw} = \theta_{\text{average}} + \Delta \theta_{\text{amplit}} \cdot \sin \left( \frac{360 \times \text{time} + \left( 273.25 + d_{\text{off}} \right) \times 24}{8760} \right)
\]

In the equation is:

- \( \theta_{\text{average}} = 10.0 \, ^\circ \text{C} \): yearly average cold water temperature in \(^\circ \text{C}\)
- \( \Delta \theta_{\text{amplit}} = 3.0 \, ^\circ \text{C} \): average amplitude for seasonal variation in K
- \( \text{time} \): hour of the year in h
- \( d_{\text{off}} = 60 \text{ days} \): time-shift parameter in days

4.2 Solar combisystem 1

The solar combisystem consists of six flat plate collector modules with a total aperture area of 13.96 m² and a combistore with a nominal volume of 950 l. The domestic hot water preparation is realized by an immersed heat exchanger. The upper part of the combistore is kept at a minimum temperature of 50 °C. An external auxiliary boiler is heating up the upper volume of the combistore in times with low solar irradiation. Depending on the space heating return temperature and store temperature the combistore is bypassed or space heating flow is preheated by solar heated water in the water store and finally heated to the set temperature by the auxiliary heater.

The solar heat is transferred to the water in the combistore by an internal heat exchanger at the bottom part of the store.
The control strategy is based on a temperature difference.

- **Collector loop**
  - the collector loop pump turns on if $\Delta T = T_4 - T_1 > 5$ K
  - the collector loop pump turns off if $\Delta T = T_4 - T_1 < 3$ K or if $T_1 \geq 80$ °C
  - the flow rate in the collector loop is controlled in dependency of the temperature difference $\Delta T = T_4 - T_1$ and is set between 15… 50 kg/(m² h)

- **Auxiliary heating for domestic hot water preparation**
  - the auxiliary heater is turned on if $T_3 < 50$ °C
  - the auxiliary heater is turned off if $T_3 > 55$ °C
  - the maximum flow temperature is 70 °C

- **Auxiliary heating for room heating**
  - the space heating flow is preheated by solar heated water in the combistore if the temperature difference between the storage temperature ($T_2$) and the space heating return flow temperature ($T_3$) is greater than the switch on temperature difference of 5 K (return flow increase):
    - $T_2 - T_3 > 5$ K
  - the space heating flow is directly heated by the auxiliary boiler if:
    - $T_2 - T_3 < 3$ K

### 4.2.1 Test results

According to the testing standards described in chapter 2 the following main characteristic parameters of the different components (store, collector) were determined:

- **Collector parameters**
  - Conversion factor of the collector:
    - $\eta_0 = 0.791$
— Heat transfer coefficient of the collector:
  \( a_1 = 3.94 \text{ W/(m}^2\text{ K)} \)
— Temperature dependant heat transfer coefficient of the collector:
  \( a_2 = 0.012 \text{ W/(m}^2\text{ K}^2) \)
— Specific heat capacity of the collector:
  \( C = 5.35 \text{ kJ/(m}^2\text{ K)} \)

- Store parameters
  — Effective storage volume
    \( V_{sto} = 924 \text{ l} \)
  — Volume of the solar heat exchanger
    \( V_{s,hx} = 14.0 \text{ l} \)
  — Volume of the immersed heat exchanger for domestic hot water preparation
    \( V_{dhw,hx} = 33 \text{ l} \)
  — Thermal capacity of the whole store
    \( C_{sto} = 4046.0 \text{ kJ/K} \)
  — Heat loss rate of the storage (standby)
    \( U_{a,s} = 3.84 \text{ W/K} \)

According to CEN/TS 12977-5 the functionality of the controller has been tested.

With the parameters determined by the component test a numerical model of the combisystem is set up in TRNSYS. The thermal behaviour of the solar combisystem is simulated and the annual energy gain of the system is calculated for the reference weather data and load profile (cf. chapter 4.1). The following main results have been obtained:

Table 4.1: Main results of the thermal performance prediction for the solar combisystem 1 using the CTSS method

<table>
<thead>
<tr>
<th>Running time collector pump [h]</th>
<th>Stagnation [h]</th>
<th>Collector energy gain [kWh/a]</th>
<th>Collector energy gain</th>
<th>Heat losses</th>
<th>Auxiliary heat demand [kWh/a]</th>
</tr>
</thead>
<tbody>
<tr>
<td>500</td>
<td>278</td>
<td>4301</td>
<td>415</td>
<td>1256</td>
<td>9444</td>
</tr>
</tbody>
</table>

According to equation 2.1, 2.4 and 2.5 the following values are obtained for characterizing the thermal performance:

Table 4.2 Thermal performance characterization of the solar combisystem 1

<table>
<thead>
<tr>
<th>( f_{sav} ) [%]</th>
<th>( f_{sol} ) [%]</th>
<th>( \eta_{sys} ) [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>25.5</td>
<td>21.5</td>
<td>18.8</td>
</tr>
</tbody>
</table>
4.3 Solar combisystem 2

The solar combisystem 2 (see Fig. 4.2) consists of five flat plate collector modules with a total aperture area of 11.3 m² and a combistore with a nominal volume of 750 l. In the combistore an internal tank is integrated for heating up the domestic hot water by the surrounding water. The upper part of the store is kept at a minimum temperature of 55 °C. An external auxiliary boiler is heating up the upper part in times of low solar irradiation.

The solar heat is transferred to the water in the combistore by an internal heat exchanger at the bottom of the store. The solar supported room heating is based on the principle of a return flow increase. If the return temperature of the room heating \( T_4 \) is below the store temperature \( T_3 \), the return flow is going through to the store (return flow increase to use heat from solar). If necessary the return flow is then heated up by the auxiliary heater via the hydraulic switch.

![System design of the solar combisystem 2](image)

The control strategy of the solar combisystem is based on a temperature difference control:

- **Collector loop**
  - the collector loop pump turns on if \( \Delta T = T_1 - T_2 > 8 \text{ K} \)
  - the collector loop pump turns off if \( \Delta T = T_1 - T_2 < 4 \text{ K} \)
  - the flow rate in the collector loop is 250 l/h (constant)

- **Auxiliary heating for domestic hot water preparation**
  - the auxiliary heater is turned on if \( T_{SF1} < 55 \text{ °C} \)
— the auxiliary heater is turned off if $T_{SF1} > 60 \, ^\circ C$
— the maximum flow temperature is $80 \, ^\circ C$

• Auxiliary heating for room heating
— The return flow is passing through the store if $T_3 - T_4 > 6 \, K$
— The return flow is directly heated up by the auxiliary heater if $T_3 - T_4 < 3 \, K$

4.3.1 Test results

According to the testing standards described in chapter 2 the following main characteristic parameters of the different components (store, collector) were determined:

• Collector parameters
  — Conversion factor of the collector:
    $\eta_0 = 0.811$
  — Heat transfer coefficient of the collector:
    $a_1 = 3.623 \, W/(m^2 \cdot K)$
  — Temperature dependant heat transfer coefficient of the collector:
    $a_2 = 0.0146 \, W/(m^2 \cdot K^2)$
  — Specific heat capacity of the collector:
    $C = 4.3 \, kJ/(m^2 \cdot K)$

• Store parameters
  — Volume of outer tank
    $V_{sto} = 402.0 \, l$
  — Volume of the solar heat exchanger
    $V_{s,hx} = 14.0 \, l$
  — Volume of the immersed heat exchanger for domestic hot water preparation
    $V_{dhw,hx} = 364.7 \, l$
  — Thermal capacity of the whole store
    $C_{sto} = 3142.5 \, kJ/K$
  — Heat loss rate of the storage (during standby)
    $U_{A_{s,a}} = 3.16 \, W/K$

According to CEN/TS 12977-5 the functionality of the controller has been tested.

Using the parameters determined by the component test a numerical model of the combisystem is set up in TRNSYS. The thermal behaviour of the solar combisystem is simulated and the annual energy gain of the system is calculated for the reference weather data and load profile (cf. chapter 4.1). The following main results have been obtained:

Table 4.3: Main results of the thermal performance prediction for the solar combisystem 2 using the CTSS method

<table>
<thead>
<tr>
<th>Running time collector pump [h]</th>
<th>Stagnation [h]</th>
<th>Collector energy gain [kWh/a]</th>
<th>Collector energy gain collector loop [kWh/a]</th>
<th>Collector energy gain store [kWh/a]</th>
<th>Auxiliary heat demand [kWh/a]</th>
</tr>
</thead>
<tbody>
<tr>
<td>942</td>
<td>650</td>
<td>3373</td>
<td>328</td>
<td>814</td>
<td>9818</td>
</tr>
</tbody>
</table>
According to equation 2.1, 2.4 and 2.5 the following values are obtained for characterizing the thermal performance:

Table 4.4 Thermal performance characterization of the solar combisystem 2

<table>
<thead>
<tr>
<th>$f_{sav}$ [%]</th>
<th>$f_{sol}$ [%]</th>
<th>$\eta_{sys}$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>22.6</td>
<td>18.4</td>
<td>20.6</td>
</tr>
</tbody>
</table>

### 4.4 Solar combisystem 3

The solar combisystem 3 (see Fig. 4.3) consists of six flat plate collector modules with a total aperture area of 14.1 m² and a combistore with a nominal volume of 933 l.

The upper part of the store is kept at a minimum temperature of 60 °C by an external auxiliary boiler.

The domestic hot water is prepared by an external domestic hot water heat exchanger. The space heating circuit is only connected to the combistore. The solar heat is transferred to the water in the combistore by an external heat exchanger.

![Fig. 4.3: System design of the solar combisystem 3](image)

The control strategy is based on a temperature difference.

- **Collector loop**
  - the collector loop pump turns on if $\Delta T = T_C - T_{SP2} > 5$ K
  - the collector loop pump turns off if $\Delta T = T_C - T_{SP2} < 3$ K or if $T_{SP1} \geq 95$ °C
  - the flow rate in the collector loop is 40 kg/ (m² h)

- **Auxiliary heating for domestic hot water preparation**
  - the auxiliary heater is turned on if $T_{D2} < 65$ °C
  - the auxiliary heater is turned off if $T_{D2} > 60$ °C
  - the maximum flow temperature is 67 °C
• Auxiliary heating for space heating
  — the auxiliary heater is turned on if $T_{SP2} < T_{SH,FL} - 5$ K
  — the auxiliary heater is turned off if $T_{SP2} > T_{SH,FL}$ °C
  — the maximum flow temperature is 67 °C

4.4.1 Test results

According to the testing standards described in chapter 2 the following main characteristic of the different components (store, collector) were determined:

• Collector parameters
  — Conversion factor of the collector:
    $\eta_0 = 0.842$
  — Heat transfer coefficient of the collector:
    $a_1 = 3.818$ W/(m$^2$ K)
  — Temperature dependant heat transfer coefficient of the collector:
    $a_2 = 0.018$ W/(m$^2$K$^2$)
  — Specific heat capacity of the collector:
    $C = 9.537$ kJ/(m$^2$ K)

• Store parameters
  — Effective storage volume of the combistore
    $V_{sto} = 933$ l
  — Thermal capacity of the whole store
    $C_{sto} = 3877.4$ kJ/K
  — Heat loss rate of the storage (standby)
    $U_{A,s,a} = 4.73$ W/K

According to CEN/TS 12977-5 the functionality of the controller has been tested.

Using the parameters determined by the component test a numerical model of the combisystem is set up in TRNSYS. The thermal behaviour of the solar combisystem is simulated and the annual energy gain of the system is calculated for the reference weather data and load profile (cf. chapter 4.1). The following main results have been obtained:

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>772</td>
<td>91</td>
<td>3563</td>
<td>426</td>
<td>1991</td>
<td>10425</td>
</tr>
</tbody>
</table>

According to equation 2.1, 2.4 and 2.5 the following values are obtained for characterizing the thermal performance:

<table>
<thead>
<tr>
<th>$f_{sav}$</th>
<th>$f_{sol}$</th>
<th>$\eta_{sys}$</th>
</tr>
</thead>
</table>
5 Test results of the SCSPT method

Three commercial solar combisystems have been tested according to the global test method. The generic hydraulic system concept of the solar combisystems according to (THÜR 2010) is:

- Solar combisystems 1: B1 – Immersed DHW heat exchanger, auxiliary as return flow increase
- Solar combisystems 2: A1 – Tank in Tank, auxiliary as return flow increase
- Solar combisystems 3: C2 – External DHW unit, auxiliary charging heat storage

The solar combisystems 1 and 2 have also been tested according to the CTSS method (cf. chapter 4). The solar combisystem 3 has only been tested according to the SCSPT method.

5.1 Reference condition for testing

For all experimental tests realized within this project, the boundary conditions used are the following:

- Building: SFH60 from IEA SHC Task 32
- Climate: Zurich
- Collector area: 16.1 m²
- Characteristics of the collector used:
  - $\eta_0 = 0.77$
  - $a_1 = 3.478 \text{ W/m}^2\text{K}$
  - $a_2 = 0.015 \text{ W/m}^2\text{K}^2$
  - $\text{IAM} = 0.96$
  - Specific heat capacity of the collector: $C = 7.422 \text{ kJ/(m}^2\text{K)}$
- Domestic hot water consumption: 200 l/day
- Cold water temperature according to equation 4.1 with:
  - $d_{\text{average}} = 9.7 ^\circ \text{C}$ yearly average cold water temperature in °C
  - $\Delta d_{\text{amplit}} = 6.3 \text{ K}$ average amplitude for seasonal variation in K
  - time:
  - $d_{\text{off}} = 60$ time-shift parameter in days

5.2 Solar combisystem 1

The design of the solar combisystem 1 is described in chapter 4.2.

During the SCSPT the heat delivered by the collector field is determined by simulating the collectors with the parameter specified in chapter 5.1. The collector area is set to 16.1 m². The collector loop control strategy is the same as in the CTSS method:
Collector loop control strategy:

- the collector loop pump turns on if $\Delta T = T_4 - T_1 > 5 \, \text{K}$
- The collector loop pump turns off if $\Delta T = T_4 - T_1 < 3 \, \text{K}$ or if $T_1 \geq 80 \, ^\circ\text{C}$
- The flow rate in the collector loop is controlled in dependency of the temperature difference $\Delta T = T_4 - T_1$ and is set between 15…50 kg/(m² h)

The generic hydraulic system concept of the solar combisystem tested is shown in Fig. 5.1. According to the figure for each hydraulic loop the flow and return temperature and the flow rate have been measured for calculating the heat quantity. In addition, the primary energy used by the gas boiler has been recorded.

![System design of the solar combisystem 1 including the measuring points.](image)

The following parameters for controlling have been set during the test sequence:

- Room temperature: 20 °C
- Draw-off temperature of DHW: 45 °C
- Cold water temperature: cf. chapter 5.1
- Set temperature of the backup volume for DHW: 60.0 °C
- Return flow increase start-up and shut-off temperature difference: $\Delta T_{on} = 6.0 \, \text{K}$, $\Delta T_{off} = 3 \, \text{K}$
- The heating curve depends on the room and ambient temperatures. The ratio of dependency is:
  - 62 % room temperature dependant
  - 38 % ambient temperature dependant

The heating curve used during the test is shown in the following Fig. 5.12.
Fig. 5.2: Heating curve used for the test of the solar combisystem 1

5.2.1 Test results

Two test sequences have been performed. In Table 5.1 the measurement results are displayed. The following abbreviations are used:

- $Q_{d,sh}$: Heat demand for space heating in kWh
- $Q_{d,hw}$: Heat demand for hot water preparation in kWh
- $Q_{solar}$: Heat delivered by the solar collector into the store in kWh
- $Q_{aux,net}$: Net auxiliary heat demand in kWh
- $E_{aux}$: Primary energy demand in kWh
- $W_{el}$: Parasitic energy consumption in kWh
- $Q_{loss}$: Heat losses of the system in kWh

The index “12 day” indicates that the results are obtained within the 12 day test of the SCTP test method.

<table>
<thead>
<tr>
<th>Test</th>
<th>$Q_{d,sh,12day}$ [kWh]</th>
<th>$Q_{d,hw,12day}$ [kWh]</th>
<th>$Q_{solar,12day}$ [kWh]</th>
<th>$Q_{aux,12day}$ [kWh]</th>
<th>$E_{aux,12day}$ [kWh]</th>
<th>$W_{el,12day}$ [kWh]</th>
<th>$Q_{loss,12day}$ [kWh]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test 1</td>
<td>295.9</td>
<td>102.0</td>
<td>158.6</td>
<td>314.4</td>
<td>335.6</td>
<td>37.5</td>
<td>75.1</td>
</tr>
<tr>
<td>Test 2</td>
<td>290.2</td>
<td>101.8</td>
<td>147.9</td>
<td>301.4</td>
<td>340</td>
<td>37.1</td>
<td>57.3</td>
</tr>
<tr>
<td>Difference</td>
<td>1.9 %</td>
<td>0.2 %</td>
<td>6.7 %</td>
<td>4.1 %</td>
<td>-1.3 %</td>
<td>1.1 %</td>
<td>23.7 %</td>
</tr>
</tbody>
</table>

The heat losses of the system are calculated by an energy balance around the system (cf. equation 5.1)

$$Q_{loss} = Q_{solar} + Q_{aux,net} - (Q_{d,sh} + Q_{d,hw})$$

5.1
Between the two test sequences, differences are noticeable concerning the heat delivered by the solar collector circuit, the heat demand for space heating and the auxiliary heat demand. It can be explained by the fact that “test 2” has been stopped during approximately 2 hours and restarted. It happened during the day number 7 of the test sequence. When the test has been restarted at day number 6 the building (TYPE 56) in the TRNSYS 16 simulation has been initialized with the room temperature before the test sequence stopped. Furthermore, the two test sequences have not been done in the exactly same experimental conditions.

The comparatively high value of the system heat losses on the hydraulic side of the system can be explained by:

- the heat losses of the store
- the heat losses of the hydraulic pipes

During the test sequence the testing room is maintained at a constant temperature of $T = 18.6 \, ^\circ C$

For the global approach the annual results are extrapolated from the 12 days test sequence. The relation used to extrapolate the results is calculated according to equation 5.2:

$$E_{\text{annual}} = \frac{E_{\text{days}}}{12} \cdot 365$$

### Table 5.2: Annual extrapolated test results for the solar combisystem 1

<table>
<thead>
<tr>
<th></th>
<th>$Q_{\text{sh}}$ [kWh]</th>
<th>$Q_{\text{hw}}$ [kWh]</th>
<th>$Q_{\text{solar}}$ [kWh]</th>
<th>$Q_{\text{aux}}$ [kWh]</th>
<th>$E_{\text{aux}}$ [kWh]</th>
<th>$W_{\text{el}}$ [kWh]</th>
<th>$Q_{\text{loss}}$ [kWh]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test 1</td>
<td>9000</td>
<td>3103</td>
<td>4824</td>
<td>9563</td>
<td>10208</td>
<td>1141</td>
<td>2284</td>
</tr>
<tr>
<td>Test 2</td>
<td>8827</td>
<td>3096</td>
<td>4499</td>
<td>9168</td>
<td>10342</td>
<td>1129</td>
<td>1743</td>
</tr>
<tr>
<td>Difference</td>
<td>1.9 %</td>
<td>0.2 %</td>
<td>6.7 %</td>
<td>4.1 %</td>
<td>-1.3 %</td>
<td>1.1 %</td>
<td>23.7 %</td>
</tr>
</tbody>
</table>

### 5.3 Solar combisystem 2

The design of the solar combisystems 2 is described in chapter 4.3.

The generic hydraulic system concept of the solar combisystem tested is shown in figure Fig. 5.3. According to the figure for each hydraulic loop the flow and return temperature and the flow rate have been measured for calculating the heat quantity. In addition, the primary energy used by the gas boiler has been recorded.
For simulating the collector field during the test sequence of the SCSPT the collector has been modelled with the parameter specified in chapter 4.1. The collector area has been set to 16.1 m². Furthermore, the same control strategy as for the CTSS method has been implemented in the simulation software for testing.

- Collector loop control strategy:
  - The collector loop pump turns on if $\Delta T = T_1 - T_2 > 8$ K
  - The collector loop pump turns off if $\Delta T = T_1 - T_2 < 4$ K
    or if $T_2 \geq 60$ °C
  - The flow rate in the collector loop is 250 l/h

The following controller parameters have been set during the test sequence:

- Room temperature: 20 °C
- Draw-off temperature of DHW: 45 °C
- Cold water temperature: cf. chapter 5.1
- Set temperature of the backup volume for DHW: 60.0 °C
- Return flow increase start-up and shut-off temperature difference: $\Delta T_{on} = 6$ K, $\Delta T_{off} = 3$ K
- Heating curve depends only on the ambient temperature (see figure 5.4).
5.3.1 Test results

Two test sequences have been performed. The measurement results are displayed in Table 5.3. The abbreviations used are explained in chapter 5.2.1.

<table>
<thead>
<tr>
<th></th>
<th>$Q_{\text{d,sh,12day}}$</th>
<th>$Q_{\text{d,hw,12day}}$</th>
<th>$Q_{\text{solar,12day}}$</th>
<th>$Q_{\text{aux,12day}}$</th>
<th>$E_{\text{aux,12day}}$</th>
<th>$W_{\text{el,12day}}$</th>
<th>$Q_{\text{loss,12day}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test 1</td>
<td>437.2</td>
<td>98.7</td>
<td>161.7</td>
<td>425.9</td>
<td>472.9</td>
<td>35.8</td>
<td>51.7</td>
</tr>
<tr>
<td>Test 2</td>
<td>437.9</td>
<td>98.9</td>
<td>163.5</td>
<td>423.8</td>
<td>470.8</td>
<td>35.6</td>
<td>50.5</td>
</tr>
<tr>
<td>Difference</td>
<td>-0.2 %</td>
<td>-0.2 %</td>
<td>-1.1 %</td>
<td>0.5 %</td>
<td>0.4 %</td>
<td>0.6 %</td>
<td>2.3 %</td>
</tr>
</tbody>
</table>

A good reproducibility exists between the two test sequences. The energy consumptions in this test are higher than for the previous test for the solar combisystem 1. The reason for this is that the heating curve is not adapted very well to the system and building. As a consequence the mean room temperature during the test sequence is around 21.8 °C which is higher than the 20 °C demanded.

The system heat losses on the hydraulic side of the system are around 51 kWh. Like for the previous test, it can be explained by:

- The heat losses of the storage
- The heat losses of the hydraulic pipes

In Fig. 5.5 infrared images of the piping, combistore and interconnections of the solar combisystem are displayed. It can be seen that despite insulations high temperatures are reached especially at the connections between the components.

Fig 5.5.a and Fig 5.5.c show high surface temperatures on the connections between the DHW part of the store and the hydraulic pipe from the boiler.

Fig 5.5.b gives an impression on the heat losses around the hydraulic separator between the boiler and the heating loop.
For the global approach the annual results are extrapolated from the 12 days test sequence with equation 5.2.

Table 5.4: Annual extrapolated test results for the solar combsystem 2

<table>
<thead>
<tr>
<th></th>
<th>Q_{sh}</th>
<th>Q_{hw}</th>
<th>Q_{solar}</th>
<th>Q_{aux}</th>
<th>E_{aux}</th>
<th>W_{el}</th>
<th>Q_{loss}</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test 1</td>
<td>13298</td>
<td>3002</td>
<td>4918</td>
<td>12955</td>
<td>14384</td>
<td>1089</td>
<td>1573</td>
</tr>
<tr>
<td>Test 2</td>
<td>13320</td>
<td>3008</td>
<td>4973</td>
<td>12891</td>
<td>14320</td>
<td>1083</td>
<td>1536</td>
</tr>
<tr>
<td>Difference</td>
<td>-0.2 %</td>
<td>-0.2 %</td>
<td>-1.1 %</td>
<td>0.5 %</td>
<td>0.4 %</td>
<td>0.6 %</td>
<td>2.3 %</td>
</tr>
</tbody>
</table>

### 5.4 Solar combsystem 4

The solar combsystem 4 has only been tested according to the SCSPT method. The generic hydraulic system concept of the solar combsystem tested is shown in figure Fig. 5.6. It consists of an 800 l combistore with an external unit for hot water preparation and a collector area of 16.1 m².
Fig. 5.6: System design of the solar combisystem 3 including the measuring points

The external DHW unit is fed from the auxiliary volume in the top of the water store. This volume needs to be kept at a sufficient high temperature by the auxiliary heater, if the input of solar energy is not sufficient. The space heating loop is connected to the water store which operates as a hydraulic switch. The space heating loop takes it power from the middle part of the combistore. Due to the inertia effect of the water in the store, the auxiliary heater is operating mostly at its maximum power.

According to Fig. 5.6 for each hydraulic loop the flow and return temperature and the flow rate have been measured for calculating the heat quantity. The boiler was not included in the solar combisystem. For the test a standard boiler with an efficiency of 85% was used.

During the SCSPT the heat delivered by the collector field is determined by simulating the collectors with the parameters specified in chapter 5.1. For simulating the collector field during the tests the following control strategy has been implemented in the testing software:

- Collector loop control strategy:
  - Collector circuit start-up temperature difference: 5 K
  - Collector circuit switch off temperature difference: 3 K
  - The flow rate of the solar loop is controlled by the solar loop controller

The following controller parameters have been set during the test sequence:

- Room temperature: 20 °C
- Draw-off temperature of DHW: 45 °C
- Cold water temperature: cf. chapter 5.1
- Set temperature of the backup volume for DHW: 60.0 °C
- The heating curve depends only on the ambient temperature. Based on the graph in Fig. 5.7 the slope used for the heating curve during the test sequences is 0.7. The slope of 0.7 is chosen in order to have a regulated room temperature of around 20°C.
5.4.1 Test results

Two test sequences have been performed. The measurement results are displayed in Table 5.3. The abbreviations used are explained in chapter 5.2.1.

Table 5.5: Measurement results for the 12-day test sequence for the solar combisystem 2

<table>
<thead>
<tr>
<th></th>
<th>$Q_{\text{d,sh,12day}}$ [kWh]</th>
<th>$Q_{\text{d,hw,12day}}$ [kWh]</th>
<th>$Q_{\text{Solar,12day}}$ [kWh]</th>
<th>$Q_{\text{aux,12day}}$ [kWh]</th>
<th>$E_{\text{aux,12day}}$ [kWh]</th>
<th>$W_{\text{el,12day}}$ [kWh]</th>
<th>$Q_{\text{loss,12day}}$ [kWh]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test 1</td>
<td>363.4</td>
<td>89.9</td>
<td>198.7</td>
<td>393.3</td>
<td>462.7</td>
<td>23.2</td>
<td>138.7</td>
</tr>
<tr>
<td>Test 2</td>
<td>366.8</td>
<td>88.2</td>
<td>191.6</td>
<td>399.5</td>
<td>470</td>
<td>22.7</td>
<td>136.1</td>
</tr>
<tr>
<td>Difference</td>
<td>-0.9%</td>
<td>1.9%</td>
<td>3.6%</td>
<td>-1.6%</td>
<td>-1.6%</td>
<td>2.2%</td>
<td>1.9%</td>
</tr>
</tbody>
</table>

The system heat losses are two times greater than for the two other solar combisystem tests. For a more detailed analysis infrared images of the system installed have been taken during operation (cf. Fig. 5.8). From these images the high value of the heat losses of the system can be explained. The main reasons are:

- The storage heat losses are very high due to the fact that the insulation is not well adjusted to the store (Fig. 5.8 c)
- Unused hydraulic connections of the store are not well insulated (Fig. 5.8 b)
- The hydraulic connections between the hydraulic units and the water store are not well insulated (Fig. 5.8 a)
For the global approach the annual results are extrapolated from the 12 days test sequence according to equation 5.2.

Table 5.6: Annual extrapolated test results for the solar combisystem 3

<table>
<thead>
<tr>
<th></th>
<th>(Q_{\text{d.sh}}) [kWh]</th>
<th>(Q_{\text{d,hw}}) [kWh]</th>
<th>(Q_{\text{Solar}}) [kWh]</th>
<th>(Q_{\text{aux}}) [kWh]</th>
<th>(E_{\text{aux}}) [kWh]</th>
<th>(W_{\text{el}}) [kWh]</th>
<th>(Q_{\text{loss}}) [kWh]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test 1</td>
<td>11053</td>
<td>2735</td>
<td>6044</td>
<td>11963</td>
<td>14074</td>
<td>706</td>
<td>4219</td>
</tr>
<tr>
<td>Test 2</td>
<td>11157</td>
<td>2683</td>
<td>5828</td>
<td>12152</td>
<td>14296</td>
<td>691</td>
<td>4140</td>
</tr>
<tr>
<td>Difference</td>
<td>-0.9%</td>
<td>1.9%</td>
<td>3.6%</td>
<td>-1.6%</td>
<td>-1.6%</td>
<td>2.2%</td>
<td>1.9%</td>
</tr>
</tbody>
</table>

Fig. 5.8: Thermal image of the store and the piping of the solar combisystem during operation
6 Comparison of the thermal performance characterization of solar combisystem using the CTSS and the SCSPT method

Two approaches for testing of solar combisystems, the CTSS method and the SCSPT method, have been presented. For the comparison of the two test methods, two solar combisystems (solar combisystem 1 and solar combisystem 2) have been tested according to both, the CTSS method and the SCSPT method (cf. chapter 4 and 5). In this chapter the test results obtained by these test methods are compared.

6.1 Comparison of test results of the CTSS method and SCSPT method obtained by physical testing of two solar combisystems

For the comparison of the two test methods the annual system simulation of the CTSS method has to be adapted to the boundary condition of the SCSPT method. The simulation is performed for the climate of Zurich with the heating load of the SFH60 buildings. The collector is modelled with the parameters of the SCSPT method instead of the parameters determined according to EN 12975-2. The collector area is set to 16.1 m² for both combisystems. In the CTSS method, the combistore is modelled by using the parameters determined according to CEN/TS 12977-4 and the control strategy described in chapter 4.2 and chapter 4.3 is implemented. In Table 6.1 a summary of the boundary conditions used for the annual system simulation in the CTSS method is given.

Table 6.1: Boundary conditions for laboratory testing

<table>
<thead>
<tr>
<th>Main characteristic</th>
<th>One family house, 140 m² heated floor area (SFH 60)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Location</td>
<td>Zurich</td>
</tr>
<tr>
<td>Collector area</td>
<td>16.1 m²</td>
</tr>
<tr>
<td>Collector orientation</td>
<td>South, 45° tilted</td>
</tr>
<tr>
<td>Store volume</td>
<td>System 1: 950 l</td>
</tr>
<tr>
<td></td>
<td>System 2: 750 l</td>
</tr>
<tr>
<td>Hot water demand</td>
<td>200 l/d at 45 °C at 6 pm (3000 kWh/a)</td>
</tr>
<tr>
<td></td>
<td>seasonal dependant cold water temperature of 9.7 °C +/- 6.3 K (cf. equation 4.1)</td>
</tr>
<tr>
<td>Heating load</td>
<td>8540 kWh/a, equivalent to 61 kWh/(m²a) for Zurich climate</td>
</tr>
<tr>
<td></td>
<td>Flow/ return temperature: 40 °C / 35 °C</td>
</tr>
<tr>
<td>Control strategy</td>
<td>System 1: cf. chapter 4.2</td>
</tr>
<tr>
<td></td>
<td>System 2: cf. chapter 4.3</td>
</tr>
<tr>
<td>Auxiliary boiler type</td>
<td>Gas heater</td>
</tr>
<tr>
<td>Collector piping</td>
<td>piping length of the return and flow line: 10 m each, indoor; pipe insulation according to 12977-2, Annex A2.</td>
</tr>
</tbody>
</table>
Ambient temperature of the store  15 °C

Heat demand of a conventional
(non solar) heating system:  12184 kWh/a

In Table 6.2 the load for domestic hot water and space heating determined with the CTSS method and the SCSPT method are listed.

<table>
<thead>
<tr>
<th>System</th>
<th>Test method</th>
<th>$Q_{d,sh}$ [kWh/a]</th>
<th>$Q_{d,hw}$ [kWh/a]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar combisystem 1</td>
<td>SCSPT</td>
<td>9000</td>
<td>3103</td>
</tr>
<tr>
<td></td>
<td>CTSS</td>
<td>8540</td>
<td>3000</td>
</tr>
<tr>
<td>Solar combisystem 2</td>
<td>SCSPT</td>
<td>13298</td>
<td>3002</td>
</tr>
<tr>
<td></td>
<td>CTSS</td>
<td>8540</td>
<td>3000</td>
</tr>
</tbody>
</table>

There is a high deviation between the two methods in the heating load. The main reasons for the deviations in the heat demand for space heating are:

- In the CTSS method the heating load is simulated by a load file in which the mass flow rate and the flow and return temperature of the space heating circuit is defined. This ensures that in each system simulated the same load is applied. In the SCSPT method the heating controller of the system directly regulates the heat supply for space heating. According to the control strategy of the system, the controller automatically adapts the flow temperature and the mass flow rate of the space heating circuit. If the control strategy is not well adapted a higher (or lower) heating load is obtained in conjunction with a higher (or lower) room temperature in the building simulation. In the solar combisystem 2 the control strategy implemented was solely ambient temperature dependant. Hence, no feedback from the actual room temperature is given to the controller. This explains the much higher heating load than the one required. The mean temperature in the building was 21.8 °C instead of 20.0 °C (cf. chapter 5.3.1).

- In the CTSS method no heat losses of pipes and components (valves, pumps) from the store to the space heating or domestic hot water circuit or the auxiliary heater are included. Only heat losses of pipes in the collector circuit are considered. In the SCSPT method as the complete system is setup in the test facility the heat losses of pipes between the different components, and of the components themselves of the system are taken into account. (cf. 5.3.1 and 5.4.1).

The deviation in the heat demand for domestic hot water in the CTSS method and the SCSPT can be explained by:

- Differences in the hot water and cold water temperature during a draw-off.
  In the CTSS method the draw-off temperature is set to 45 °C and the cold water temperature is seasonal dependant according to equation 4.1. In the SCSPT the same set temperatures are implemented in the testing software. However, the measurements in the SCSPT show that the draw-off temperature varies between 44 °C and 52 °C with a mean temperature of 48 °C. Reason for this is that the thermostatic mixing valve is
installed in the system test set up. Furthermore, due to cold water stagnation between two draw-offs, the cold water temperature is higher than the set temperature, especially for low flow rates (lower than 100 kg/h). As the draw-off is time-controlled a defined hot water volume is drawn and not a defined energy quantity.

As the heat demand in the SCSPT method is much higher than requested a direct comparison of the SCSPT results with the results obtained with the CTSS method is not possible. Hence, in a next step the system simulation of the CTSS method has been further adapted to the boundary condition of the SCST test.

### 6.2 Adaption of the CTSS method to the boundary condition of the SCSPT method

For the comparison of the two test methods, the boundary conditions in the annual system simulation of the CTSS method are further adapted to the laboratory test of the SCSPT method. For the determination of the heating load a building model (TYPE 56) of the SFH60 building is implemented in the simulation software. The corresponding flow temperature and flow rate to cover the load are regulated directly by the controller of the solar combisystem. This time, the control strategy is adapted according to chapter 5.2 and chapter 5.3.

In addition, in the simulation software, the piping between the components have been adapted to the real system set up during the SCSPT by means of piping length, piping insulation and heat loss coefficients of the piping insulation.

In the flow chart in Fig. 6.1 the methodology for the comparison is shown. The nomenclature “Sim\textsubscript{CTSS}” is introduced to differ from the CTSS where the annual system simulation is performed with the reference boundary condition according to EN 12977-2, Annex A.

The comparison of the test methods is performed on two stages:

1. Comparing the results of the 12 day test
2. Comparing the annual results
**Comparison of the 12 day test sequence:**

In a first step, the system simulation based on the Sim\textsubscript{CTSS} is performed only for the 12 days of the SCSPT method. The 12 day weather data file of the SCSPT method was used as input file instead of the annual weather data file.

The results of the 12 day simulation are compared to the experimental results of the SCSPT. The quality criteria are the deviation in:

- Space heating demand $Q_{d,sh}$
- Domestic hot water demand $Q_{d,hw}$
- Heat delivered by the solar collector into the store $Q_{solar}$
- The auxiliary energy demand $Q_{aux,net}$
- The heat losses of the system $Q_{loss}$

**Comparison of the annual results**

In a next step the annual results are compared. Now, for the annual system simulation of the Sim\textsubscript{CTSS} method the annual weather data file is used.

In the SCSPT method the experimental results are extrapolated to a year according to equation 5.2.

### 6.2.1 *Comparison of the 12 day test results*

In Table 6.3 the main results of the 12 day test sequence for the solar combisystem 1 and 2 are presented. Listed are the heat demands, heat gains and heat losses in the different circuits calculated by the 12 day simulation of the Sim\textsubscript{CTSS} and measured during the SCSPT.
Table 6.3: Energies on the different loops for the 12 day test sequence determined by the 12-day simulation of the SimCTSS and the measurement results of the SCSPT method.

<table>
<thead>
<tr>
<th>Test method</th>
<th>$Q_{d,sh,12day}$ [kWh]</th>
<th>$Q_{d,hw,12day}$ [kWh]</th>
<th>$Q_{solar,12day}$ [kWh]</th>
<th>$Q_{aux,net,12day}$ [kWh]</th>
<th>$Q_{loss,12day}$ [kWh]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar Combisystem 1</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>SCSPT</td>
<td>295.9</td>
<td>102.0</td>
<td>158.6</td>
<td>314.4</td>
<td>75.1</td>
</tr>
<tr>
<td>SimCTSS</td>
<td>288.9</td>
<td>108.0</td>
<td>152.5</td>
<td>296.0</td>
<td>51.6</td>
</tr>
<tr>
<td>$\epsilon_{rel}$</td>
<td>-2.4 %</td>
<td>5.6 %</td>
<td>-4.0 %</td>
<td>-6.2 %</td>
<td>-45.5 %</td>
</tr>
<tr>
<td>Solar Combisystem 2</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>SCSP</td>
<td>437.9</td>
<td>98.7</td>
<td>161.7</td>
<td>425.9</td>
<td>51.7</td>
</tr>
<tr>
<td>SimCTSS</td>
<td>473.6</td>
<td>104.1</td>
<td>170.1</td>
<td>433.5</td>
<td>25.9</td>
</tr>
<tr>
<td>$\epsilon_{rel}$</td>
<td>7.5 %</td>
<td>5.2 %</td>
<td>4.9 %</td>
<td>1.8 %</td>
<td>-99.6 %</td>
</tr>
</tbody>
</table>

In Table 6.3 the relative error $\epsilon_{rel}$ is calculated according to equation 6.1:

$$\epsilon_{rel} = \frac{Q_{i,CTSS} - Q_{i,SCSPT}}{Q_{i,CTSS}} \cdot 100$$

with:

- $Q_{i,SCSPT}$: heat demand / gain obtained with the SCSPT method in kWh
- $Q_{i,SimCTSS}$: heat demand / gain obtained with the SimCTSS method in kWh

The relative error of the heat demand for space heating for the solar combisystem 1 is $\epsilon_{rel} = -2.4 \%$, for the solar combisystem 2 $\epsilon_{rel} = 7.5 \%$. Again, the deviations can be explained by disparities between the physical test of the SCSPT and the simulation of the SimCTSS:

- The control strategy of the heating controller in the SimCTSS simulation was implemented according to the heating curve described in chapter 5.2 (solar combisystem 1) and chapter 5.3 (solar combisystem 2).
  In the SCSPT the heating controller is part of the system test. The heating curve of the controller might differ from those implemented in the SimCTSS simulation software.
- In the SimCTSS simulation additional pipe losses have been implemented. These pipe losses might still differ from those of the real test set up.

Concerning the deviation in the heat demand for hot water preparation, the same reason as described in chapter 6.1 are valid: In the SCSPT the hot and cold water temperature during a draw-off vary from the set temperature. This results in a higher or smaller heat quantity withdrawn from the store than demanded.

High deviations between the two approaches exist for the system heat losses. The following reason is responsible for the deviations:

- A higher heat demand in the SCSPT method which results in higher heat losses in the different circuits
- Higher heat losses of the store and the pipes in the SCSPT than in the simulation of the CTSS method
During the physical test of the SCSPT method high heat losses occur in the auxiliary heating circuit. This is mainly due to a hydraulic separator which is installed between the boiler circuit and the space heating circuit. During the test set up the boiler pump was running most of the time which causes relatively high temperatures in the auxiliary heating circuit and hence comparatively high heat losses.

In the Sim\textsubscript{CTSS} simulation no losses in the auxiliary heating loop are considered.

### 6.2.2 Comparison of the annual results

In Table 6.4 the annual results for the solar combisystem 1 and 2 are presented. Listed are the heat demands, heat gains and heat losses in the different circuits calculated by the annual simulation of the Sim\textsubscript{CTSS} test method and extrapolated from the 12 day test sequence of the SCSPT method. In addition, the overall heat demand of the system and the overall heat delivered to the system are listed. The overall heat demand $Q_d$ (kWh/a) is calculated according to equation 6.2 the total heat delivered $Q_{\text{delivered}}$ (kWh/a) is determined according to equation 6.3.

$$Q_d = Q_{d,sh} + Q_{d,hw} \quad \text{6.2}$$

$$Q_{\text{delivered}} = Q_{\text{solar}} + Q_{\text{aux.net}} \quad \text{6.3}$$

In the last row of Table 6.4, the fractional energy saving $f_{\text{sav}}$ is presented. For the calculation of the fractional energy saving, the reference heat demand is calculated according to equation 6.4.

$$Q_{\text{ref}} = Q_d + Q_{\text{conv}} \quad \text{6.4}$$

with the reference heat losses of a conventional system of $Q_{\text{conv}} = 644$ kWh/a $Q_{\text{ref}} = 644$ kWh/a.

<table>
<thead>
<tr>
<th>Test method</th>
<th>$Q_{d,sh}$ [kWh/a]</th>
<th>$Q_{d,hw}$ [kWh/a]</th>
<th>$Q_{\text{solar}}$ [kWh/a]</th>
<th>$Q_{\text{aux.net}}$ [kWh/a]</th>
<th>$Q_d$ [kWh/a]</th>
<th>$Q_{\text{delivered}}$ [kWh/a]</th>
<th>$Q_{\text{loss}}$ [kWh/a]</th>
<th>$f_{\text{sav}}$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar Combisystem 1</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>SCSPT</td>
<td>9000</td>
<td>3103</td>
<td>4824</td>
<td>9563</td>
<td>12103</td>
<td>14387</td>
<td>2284</td>
<td>25.0</td>
</tr>
<tr>
<td>Sim\textsubscript{CTSS}</td>
<td>8957</td>
<td>3287</td>
<td>4287</td>
<td>9376</td>
<td>12224</td>
<td>13663</td>
<td>1419</td>
<td>27.2</td>
</tr>
<tr>
<td>$\varepsilon_{\text{ref}}$</td>
<td>-0.5 %</td>
<td>5.6 %</td>
<td>-12.5 %</td>
<td>-2.0 %</td>
<td>1.0 %</td>
<td>-5.3 %</td>
<td>-61.0 %</td>
<td>8.1 %</td>
</tr>
<tr>
<td>Solar Combisystem 2</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>SCSPT</td>
<td>13298</td>
<td>3002</td>
<td>4918</td>
<td>12954</td>
<td>16300</td>
<td>17873</td>
<td>1573</td>
<td>23.5</td>
</tr>
<tr>
<td>Sim\textsubscript{CTSS}</td>
<td>13603</td>
<td>3168</td>
<td>5139</td>
<td>12435</td>
<td>16771</td>
<td>17574</td>
<td>803</td>
<td>28.6</td>
</tr>
<tr>
<td>$\varepsilon_{\text{ref}}$</td>
<td>2.2 %</td>
<td>5.2 %</td>
<td>4.3 %</td>
<td>-4.2 %</td>
<td>2.8 %</td>
<td>-1.7 %</td>
<td>-95.9</td>
<td>17.8 %</td>
</tr>
</tbody>
</table>

There is a good agreement in the annual space heating demand determined with the two test methods with a maximum deviation of 2.2 % for the solar combisystem 2.

In the SCSPT the heat demand for domestic hot water preparation is lower than in the Sim\textsubscript{CTSS} test method. An explanation for the deviation is given in the previous chapter 6.2.1.
A high deviation in the heat losses of the system (store and piping) between the two test methods exist. An explanation is given in chapter 6.2.1.

The fractional energy savings determined with the SCSPT method is lower than the fractional energy savings determined with the CTSS method. The absolute difference is 2.2 % for the solar combisystem 1 and 5.1 % for the solar combisystem 2. In the SCSPT method both combisystems have to provide a higher heat quantity than in the SimCTSS. The main reason for this is that the heat losses determined with the SCSPT are higher than the heat losses determined with the CTSS. These heat losses can be partly compensated by a higher solar gain especially during summer. The greater part of the heat losses have to be compensated by the auxiliary heater. The increase in auxiliary energy demand reduces the fractional energy savings.

In Fig. 6.3 and Fig. 6.2 the energy demand and energy gain calculated by the SCTP method and the SimCTSS method are depicted for the two combisystems. In the figures, $E_{col}$ (kWh) is the annual energy gain in collector plane, determined according to equation 6.5:

$$E_{col} = A_c \cdot E_{glob,k}$$

![Fig. 6.2 Energy demand and energy gain of the solar combisystem 1 predicted by the SCSPT method and the SimCTSS method.](image)

$$E_{col} = A_c \cdot E_{glob,k}$$
7  Performance prediction for different boundary conditions

Often, customers and manufactures are interested in getting information of the thermal performance of a solar combisystem under different boundary conditions.

With the CTSS method a thermal performance prediction for any boundary conditions can be performed by adapting the annual system simulation.

In the SCSPT method the annual performance is obtained by extrapolating the 12 day test results to a complete year. Hence, the performance prediction is in only possible for the boundary conditions, for which the physical testing of the solar combisystem has been performed. To overcome this limitation, within the CombiSol project it is analyzed if an extrapolating of the SCSPT results to other boundary conditions is possible based on the standard EN 15316-4-3:2007. The quality of this extrapolation is verified by comparing the results extrapolated from the SCSPT method to the results of the annual system simulation of the CTSS method. This time, the annual system simulation of the CTSS method is performed based on the boundary conditions described in the standard CEN/TS 12977-2.

The extrapolation of test results (SCSPT) and the annual system simulation (SimCTSS) are performed for two different climates: Wurzburg and Stockholm.

The annual space heating load for these locations and the building related to the space heating load is specified in the standard CEN/TS 12977-2, Annex A4. The reference conditions for the performance prediction are summarized in Table 7.1.
Table 7.1: Boundary conditions for laboratory testing

<table>
<thead>
<tr>
<th>Climate</th>
<th>Wurzburg</th>
<th>Stockholm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Main characteristic</td>
<td>Single family house, 128 m² heated floor area;</td>
<td>One family house, 140 m² heated floor area</td>
</tr>
<tr>
<td>Space heating demand</td>
<td>9090 kWh/a</td>
<td>14960 kWh/a</td>
</tr>
<tr>
<td>Hot water demand</td>
<td>110 l/d, 200 l/d</td>
<td>seasonal dependant cold water temperature:</td>
</tr>
<tr>
<td></td>
<td>10.0 +/- 3.0 °C</td>
<td>8.5 +/- 6.4 °C</td>
</tr>
<tr>
<td>Heat demand for hot water preparation</td>
<td>110 l/d: 1620 kWh/a</td>
<td>110 l/d: 1689 kWh/a</td>
</tr>
<tr>
<td></td>
<td>200 l/d: 2945 kWh/a</td>
<td>200 l/d: 3071 kWh/a</td>
</tr>
<tr>
<td>Hot water temperature</td>
<td>45 °C</td>
<td>45 °C</td>
</tr>
</tbody>
</table>

7.1 Performance prediction with the CTSS method

The annual system simulation is performed with the boundary conditions defined in Table 7.1. The collector is simulated with the parameters of the SCSPT method instead of the parameters determined according to EN 12975-2 of the CTSS method. The collector area is set to 16.1 m² for both combisystems. The combistore is simulated by using the parameters determined according to CEN/TS 12977-4 (cf. chapter 4.2 and chapter 4.3). The control strategy described in chapter 4.2 and chapter 4.3 is implemented.

In Table 7.2 the main results are listed. In the table, $Q_{\text{col}}$ [kWh/a] is the annual energy gain of the collectors, $Q_{\text{solar}}$ [kWh/a] the annual heat delivered by the solar collector into the store, $Q_{\text{aux,net}}$ [kWh/a] the auxiliary energy demand, $Q_{\text{loss}}$ [kWh/a] the heat losses of the system and $f_{\text{sav}}$ the fractional energy saving.

Table 7.2: Annual energy quantities for the different circuits determined by annual system simulations using the CTSS method.

<table>
<thead>
<tr>
<th>Climate / Hot water demand</th>
<th>$Q_{\text{col}}$ [kWh/a]</th>
<th>$Q_{\text{solar}}$ [kWh/a]</th>
<th>$Q_{\text{aux,net}}$ [kWh/a]</th>
<th>$Q_{\text{loss}}$ [kWh/a]</th>
<th>$f_{\text{sav}}$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar CombiSystem 1</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Stockholm</td>
<td>110 l/d</td>
<td>3509</td>
<td>3256</td>
<td>14269</td>
<td>864</td>
</tr>
<tr>
<td></td>
<td>200 l/d</td>
<td>3846</td>
<td>3593</td>
<td>15281</td>
<td>822</td>
</tr>
<tr>
<td>Wurzburg</td>
<td>110 l/d</td>
<td>3111</td>
<td>2862</td>
<td>2862</td>
<td>866</td>
</tr>
<tr>
<td></td>
<td>200 l/d</td>
<td>3537</td>
<td>3285</td>
<td>9593</td>
<td>823</td>
</tr>
<tr>
<td>Solar CombiSystem 2</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Stockholm</td>
<td>110 l/d</td>
<td>4412</td>
<td>3924</td>
<td>13970</td>
<td>1248</td>
</tr>
<tr>
<td></td>
<td>200 l/d</td>
<td>4882</td>
<td>4380</td>
<td>14819</td>
<td>1175</td>
</tr>
<tr>
<td>Wurzburg</td>
<td>110 l/d</td>
<td>4036</td>
<td>3563</td>
<td>8449</td>
<td>1307</td>
</tr>
<tr>
<td></td>
<td>200 l/d</td>
<td>4542</td>
<td>4050</td>
<td>9208</td>
<td>2938</td>
</tr>
</tbody>
</table>
7.2 Performance prediction based on the SCSPT method and EN 15316-4-3

For an extrapolation of the SCSPT results to other boundary conditions the COMBI-EN tool developed within the CombiSol project is used (NIELSEN 2010, BALES et al 2010). The COMBI-EN tool is a simple excel tool which implements the standard calculation method EN 15316-4-3 for solar combisystems, as defined for the Energy Performance of Buildings Directive (EPBD).

Based on the experimental results from the SCSPT method the COMBI-EN tool has been further adjusted. Instead of an annual value of the heat demand, a monthly load profile including the heat losses from the hydraulic distribution loop was implemented in the software. The monthly load profile is extrapolated from the 12 day test of the SCSPT method. In a next step, the parameter of the solar loop efficiency (heat exchanger factor) $\eta_{\text{loop}}$ and the solar loop losses $U_{\text{loop}}$ have been adjusted so that the results of the COMBI-EN tool meets the results of the SCSPT method in terms of the solar energy gains and the auxiliary energy demand.

The following parameters of the solar loop efficiency and the solar loop losses have been used:

<table>
<thead>
<tr>
<th>System</th>
<th>$\eta_{\text{loop}}$</th>
<th>$U_{\text{loop}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar combisystem 1</td>
<td>0.7</td>
<td>22</td>
</tr>
<tr>
<td>Solar combisystem 2</td>
<td>0.75</td>
<td>13.1</td>
</tr>
</tbody>
</table>

As already described in BALES et al 2010 the standard calculation method EN 15316-4-3 overestimates the solar contribution if the “typical values” of the collector circuit parameters, the solar loop loss rate and the solar loop efficiency, are chosen. By adapting these two input parameters, the solar energy gain can be adapted. To reduce the solar gain, relatively high values of the “solar loop loss rate” and relatively low values of the “solar loop efficiencies” have to be chosen.

In Table 7.4 the results of the SCSPT and the results of the COMBI-EN tool with the adjusted parameters are given. The error varies between 1.8 % (primary energy consumption) to 5.4 % (auxiliary heat demand). The fractional energy saving determined with the COMBI-EN tool is less than determined with the SCSPT method. The absolute deviation in the fractional energy saving is 2.8 % (solar combisystem 1) and 2.7 % (solar combisystem 2).

<table>
<thead>
<tr>
<th></th>
<th>$Q_{\text{solar}}$ [kWh]</th>
<th>$E_{\text{aux}}$ [kWh]</th>
<th>$Q_{\text{aux,net}}$ [kWh]</th>
<th>$f_{\text{sav}}$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar combisystem 1</td>
<td>COMBI-EN</td>
<td>4433</td>
<td>10167</td>
<td>9659</td>
</tr>
<tr>
<td>SCSPT method</td>
<td>4499</td>
<td>10342</td>
<td>9168</td>
<td>27.0</td>
</tr>
<tr>
<td>Difference (%)</td>
<td>1.5</td>
<td>1.7</td>
<td>-5.4</td>
<td>10.4</td>
</tr>
</tbody>
</table>

Solar Combisystem 2
With these adjustments, the COMBI-EN tool was used to extrapolate the test results of the SCSPT method to different boundary conditions (climate, load). In Table 7.5 the main results of the extrapolation are listed for the different boundary conditions.

Table 7.5: Annual heat delivered by the solar combisystem, auxiliary heat demand and fractional energy savings calculated with the COMBI-EN tool for the solar combisystem 1 and 2.

<table>
<thead>
<tr>
<th>Climate / Hot water demand</th>
<th>Q_{solar} [kWh]</th>
<th>Q_{aux,net} [kWh]</th>
<th>f_{sav} [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar Combisystem 1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Stockholm</td>
<td>110 l/d</td>
<td>3085</td>
<td>14181</td>
</tr>
<tr>
<td></td>
<td>200 l/d</td>
<td>3640</td>
<td>15080</td>
</tr>
<tr>
<td>Wurzburg</td>
<td>110 l/d</td>
<td>3000</td>
<td>8336</td>
</tr>
<tr>
<td></td>
<td>200 l/d</td>
<td>3622</td>
<td>9058</td>
</tr>
<tr>
<td>Solar Combisystem 2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Stockholm</td>
<td>110 l/d</td>
<td>2615</td>
<td>14629</td>
</tr>
<tr>
<td></td>
<td>200 l/d</td>
<td>3134</td>
<td>15564</td>
</tr>
<tr>
<td>Wurzburg</td>
<td>110 l/d</td>
<td>2459</td>
<td>8835</td>
</tr>
<tr>
<td></td>
<td>200 l/d</td>
<td>3016</td>
<td>9620</td>
</tr>
</tbody>
</table>

7.3 Comparison of the CTSS method and the extrapolated SCSPT method

Fig. 7.1 and Fig. 7.2 show the fractional energy savings of the solar combisystem 1 and 2 for different climates (Wurzburg and Stockholm) and daily hot water consumptions (110 l/d and 200 l/d). The values are determined with the annual system simulation of the CTSS method (blue bars) and extrapolated from the SCSPT method using the COMBI-EN tool (red bars). The absolute deviation between these two methods is less than 5 % for both systems.

A further analysis has shown that the deviation increases with increasing annual solar irradiation in collector plane. For Athens and Davos the extrapolated results from the SCSPT differ significantly from those of the CTSS method.
Fig. 7.1: Fractional energy savings of the solar combisystem 1 for different climates (Stockholm, Wurzburg) and daily hot water consumptions (110 l/d, 200 l/d) determined with the annual system simulation of the CTSS method (blue bars) and extrapolated from the test results of the SCSPT test method using the COMBI-EN tool (red bars).

Fig. 7.2: Fractional energy savings of the solar combisystem 2 for different climates (Stockholm, Wurzburg) and daily hot water consumptions (110 l/d, 200 l/d) determined with the annual system simulation of the CTSS method (blue bars) and extrapolated from the test results of the SCSPT test method using the COMBI-EN tool (red bars).

8 Comparison between SCSPT and on site monitoring

Main Authors: T. LETZ, (INES Education), M. Albaric (INES-CEA)

Some systems tested according to the SCSPT method have also been evaluated within the in situ monitoring (cf. "WP4: In-situ determination of primary energy savings"). Table 8.1 shows the main parameters of three different systems that have been evaluated in laboratory according to SCSPT method, and where on site monitoring results are available. For manufacturer 1, two monitoring results
are available, but for one of them, the really installed hydraulic scheme differs from the one tested in the laboratory, because an additional domestic hot water store has been installed after the immersed heat exchanger in the main water store. Therefore it has not been considered for the comparison.

Table 8.1: Comparison between indicators coming from laboratory tests and from on site monitoring

<table>
<thead>
<tr>
<th>Location of auxiliary energy evaluation</th>
<th>Solar collector area</th>
<th>System type</th>
<th>Climate</th>
<th>Space heating load (kWh)</th>
<th>DHW load (kWh)</th>
<th>Total load (kWh)</th>
<th>FSC</th>
<th>Fsav</th>
</tr>
</thead>
<tbody>
<tr>
<td>Auxiliary heater inlet</td>
<td>Manufacturer 1</td>
<td>B1</td>
<td>Zürich</td>
<td>8914</td>
<td>3099</td>
<td>12013</td>
<td>0.52</td>
<td>30%</td>
</tr>
<tr>
<td>Auxiliary heater inlet</td>
<td>Manufacturer 2</td>
<td>C1</td>
<td>Graz</td>
<td>11105</td>
<td>2709</td>
<td>13814</td>
<td>0.54</td>
<td>20%</td>
</tr>
<tr>
<td>Auxiliary heater inlet</td>
<td>Manufacturer 3</td>
<td>A1</td>
<td>Stuttgart</td>
<td>13309</td>
<td>3005</td>
<td>16314</td>
<td>0.53</td>
<td>28%</td>
</tr>
<tr>
<td>Auxiliary heater outlet</td>
<td>Manufacturer 1</td>
<td>B1</td>
<td>Lyon</td>
<td>11971</td>
<td>1601</td>
<td>13572</td>
<td>0.38</td>
<td>26%</td>
</tr>
<tr>
<td>Auxiliary heater inlet</td>
<td>Manufacturer 2</td>
<td>C1</td>
<td>Graz</td>
<td>9161</td>
<td>381</td>
<td>9543</td>
<td>0.59</td>
<td>21%</td>
</tr>
<tr>
<td>Auxiliary heater outlet</td>
<td>Manufacturer 3</td>
<td>A1</td>
<td>Stuttgart</td>
<td>13353</td>
<td>1745</td>
<td>15098</td>
<td>0.42</td>
<td>20%</td>
</tr>
</tbody>
</table>

Climate of the locations of monitored systems are continental climates, similar to the Zürich climate used for testing. For manufacturers 1 and 3, real loads are similar to test loads. System 2 has a smaller load compared to the test load.

Figure 8.1 shows the fractional energy savings as a function of the FSC value (LETZ, 2001), for laboratory and on site results. Following observations can be made:

- Results of onsite measurements are consistent with those obtained from laboratory test, because for each manufacturer, the line between both points has a similar slope compared to the range of properly working systems.
- For systems of manufacturers 1 and 3, real points have smaller values for FSC and Fractional Energy Savings, mainly linked to smaller collector areas, since loads and irradiation available are similar.
• For systems of manufacturer 2, it is the opposite: the real point has higher values for FSC and Fractional Energy Savings, linked simultaneously to slightly larger collector areas, but mainly to a smaller load.

This comparison shows a good correlation between the results coming from the SCSPT method in the lab and those coming from in situ monitoring, what in fact is a proof of the validity of the FSC approach used to evaluate the thermal efficiency of solar combisystems.

9 Summary and conclusion

In this report two test methods for performance testing of solar combisystems, a component orientated approach (CTSS) and a system testing approach (SCSPT) are presented and compared.

The CTSS method is based on a physical test of the main components and on an annual simulation of the entire system to determine the annual performance of the combisystem. The test method is defined in the standard series CEN/TS 12977 and well accepted within Europe and beyond.

The SCSPT method, based on the CCT – approach and further developed by CEA/INES, France, is based on a physical test of the entire system except the collectors. The annual performance of the combisystem is obtained by extrapolating the results of this short term test sequence to a complete year. For this testing approach no official standard is available up to now. However, within the CombiSol project it has been demonstrated that a good correlation exists between laboratory testing and in situ monitoring.

In the SCSPT method the focus is set on the performance of the whole solar heating system. A central element of the testing is to take into account the overall control strategy - and hence the interaction of the different hydraulic circuits. In addition, as for the physical test the entire system is installed on the test facility, heat losses of pipes and components are taken into account in a realistic way. During the physical test the energy gains from the collector and the heat demand for space heating are simulated online by using a simulation model of the collector and the building. According to the control strategy of the heating system the mass flow rate and flow temperature in the collector circuit, space heating circuit and auxiliary heater circuit are directly adapted.

The CTSS method sets its focus on the performance of the solar part of the heating system. In the system simulation, the heat demand for space heating is defined in a load file with the corresponding flow and return temperatures and mass flow rates. This ensures that in each system simulation the same heat demand at the same temperature level is applied.

For comparing of the auxiliary energy demand of the solar combisystem with the reference system within the CombiSol project the same annual efficiency of the auxiliary boiler of 75 % is assumed. Heat losses of pipes which also arise in the conventional system e.g. in the auxiliary heater circuit and space heating circuit are not taken into account.

Due to the different focus of testing a direct comparison of the test results of the CTSS method and SCSPT method is not possible yet (cf. chapter 6.1). However, when the system simulation of the CTSS method is adapted to the SCSPT method comparable results between the two approaches are obtained (cf. chapter 6.2).
With the CTSS method the thermal performance of the solar combisystem can be predicted for any arbitrary boundary conditions. An extrapolation of the test results from the SCSPT method with the COMBI-EN tool shows good agreement for the locations of Stockholm and Wurzburg but not for locations with much higher solar irradiations such as Athens or Davos.

At present the CTSS method is more advanced. One main advantage it the flexibility especially with regard

- to test complete system families with varying sizes of collector areas and / or store volumes
- to test solar combisystems which are individually adapted to a specific system set up
- to carry out performance predictions for other climates, heating loads or collectors

On the other hand, the SCSPT method is well adapted:

- to test prefabricated systems
- to take into account the thermal losses of the system (pipes) in a more realistic way
- to take into account in detail the overall control strategy of the system combined with the interaction of different hydraulic components

Future work should concentrate on a further development and validation of the SCSPT method. It has to be ensured that the test method allows a direct comparison of the thermal performance of different combisystems. This is only possible if for each system tested the annual heat demand for space heating and domestic hot water is similar. Another important aspect is to be able to extrapolate the test results to other boundary conditions such as heat loads, climates and collector orientation and sizes. Additional work is thus required to adjust EN 15316-4-3 to the specific needs of the SCSPT method or to develop extrapolating procedures for the SCSPT method (on-going work at INES).

The CTSS method is very suitable to determine the thermal performance of the solar part of a heating system. However, with the growing complexity of the system design and with the growing importance of an energy efficient operation of the whole heating system the control unit is gaining relevance. This can only be ensured if the control strategy of the different hydraulic circuits of a solar combisystem is optimal adjusted to each other. Here, future work should concentrate on the development of more advanced controller testing procedures or even the integration of the real control unit into the dynamic computer simulation of the entire solar heating system. A first step towards this direction are the test procedures for multi-function controllers already specified in the new standard CEN/TS 12977-5:2010 (PETER et al, 2008).

Concerning the testing of solar combisystems two parallel existing performance test methods might in general be an option for the future. The same situation already exists for solar domestic hot water systems with regard to the CTSS method and the DST (cf. ISO 9459-5) or CSTG (cf. ISO 9459-2) method respectively. The main advantage of two test methods for the same product category is the option to apply for a specific test the most appropriate and cost effective method. However, in order to implement the SCSPT method as a standardised performance test method it has to be further developed with regard to the aspects mentioned above and applied and validated for a broad spectrum of different system concepts.
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The in this document mentioned European standards are available from: Beuth Verlag GmbH, Burggrafenstrasse 6, D-10787 Berlin, internet: www.beuth.de or from other national standardisation bodies (see: www.cenorm.be/catweb/27.160.htm)