Testing of combined heating systems for small houses: Improved procedures for whole system test methods

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MacSheep - New Materials and Control for a next generation of compact combined Solar and heat pump systems with boosted energetic and exergetic performance

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Executive Summary

Dynamic system test methods for heating systems were developed and applied by the institutes SERC and SP from Sweden, INES from France and SPF from Switzerland already before the MacSheep project started. These test methods followed the same principle: a complete heating system – including heat generators, storage, control etc., is installed on the test rig; the test rig software and hardware simulates and emulates the heat load for space heating and domestic hot water of a single family house, while the unit under test has to act autonomously to cover the heat demand during a representative test cycle.

Within the work package 2 of the MacSheep project these similar – but different – test methods were harmonized and improved. The work undertaken includes:

- Harmonization of the physical boundaries of the unit under test.
- Harmonization of the boundary conditions of climate and load.
- Definition of an approach to reach identical space heat load in combination with an autonomous control of the space heat distribution by the unit under test.
- Derivation and validation of new six day and a twelve day test profiles for direct extrapolation of test results.

The new harmonized test method combines the advantages of the different methods that existed before the MacSheep project. The new method is a benchmark test, which means that the load for space heating and domestic hot water preparation will be identical for all tested systems, and that the result is representative for the performance of the system over a whole year. Thus, no modelling and simulation of the tested system is needed in order to obtain the benchmark results for a yearly cycle. The method is thus also applicable to products for which simulation models are not available yet.

Some of the advantages of the new whole system test method and performance rating compared to the testing and energy rating of single components are:

- Interaction between the different components of a heating system, e.g. storage, solar collector circuit, heat pump, control, etc. are included and evaluated in this test.
- Dynamic effects are included and influence the result just as they influence the annual performance in the field.
- Heat losses are influencing the results in a more realistic way, since they are evaluated under "real installed" and representative part-load conditions rather than under single component steady state conditions.

The described method is also suited for the development process of new systems, where it replaces time-consuming and costly field testing with the advantage of a higher accuracy of the measured data (compared to the typically used measurement equipment in field tests) and identical, thus comparable boundary conditions. Thus, the method can be used for system optimization in the test bench under realistic operative conditions, i.e. under relevant operating environment in the lab.

This report describes the physical boundaries of the tested systems, as well as the test procedures and the requirements for both the unit under test and the test facility. The new six day and twelve day test profiles are also described as are the validation results.
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1 Introduction

Dynamic system test methods are an option to evaluate in the laboratory the performance of a heating system as a whole. Research institutes that applied such whole system test procedures until now are SERC and SP from Sweden, CEA INES from France and HSR SPF from Switzerland. Their test methods followed the same principle but they had been quite different in the details. Within the work package 2 (WP2) of the MacSheep project, the current state of the art of these test methods was summarized and published in Haller et al. (2013), and the test procedures were harmonized and further developed in order to create a more advanced common test method. Results from this process have also been published in Chèze et al. (2014).

Since this report D2.3 is the first public report of WP2 in MacSheep, it has been written with the aim to be a standalone document. Thus, some decisions and results reported in previous deliverables of this work package that were not public (D2.1, D2.2) may be repeated in this report. Furthermore, some of the results shown in the previous deliverables were revised in the meantime and are therefore updated with this report.
2 Purpose and general procedure of the test

The test shall show the behavior of a complete heating system under real-life conditions at different days of the year. Therefore, a complete heating system is installed on a test rig. The test rig emulates a single family house, including the space heat (SH) distribution system, the domestic hot water (DHW) draw offs, the solar collector field, and the environmental heat source of the heat pump. The system under test must completely act autonomously to cover the heat demand of the building and the draw-offs during a representative test cycle. This test cycle is composed of a number of consecutive test-days, each one representing a day of a real year.

The annual final energy consumption needed to cover the heat demand of the building is determined by extrapolation of the measured energy consumption during the test. This approach requires that the following conditions are met:

- The same amount of useful space heat energy and useful DHW energy must be delivered by all tested systems.
- The same level of comfort must be achieved (temperature levels of DHW and heated room).
- The difference in energy stored in the system at the beginning and at the end of the test sequence must be negligibly small.
- The extrapolation of the energy balance from the short test cycle with N days to the annual energy balance is possible by multiplication with the factor 365/N.

Thus, it is possible to provide the ranking of different systems in terms of the annual energetic performance of these systems based on the short test cycle of N days. The test is based on one climate and one definition of DHW and SH loads. The annual results that are obtained will a priori be valid for this one climate and load – just as a driving cycle for the determination of emission levels and fuel consumption in the automotive industry. The extrapolation to other climates and loads is a topic of further investigation in the future.

The final test result for the determination of the energetic performance of the systems is the auxiliary energy consumption (fuel and/or electricity). In the case of solar and heat pump systems, the only auxiliary energy consumed is the electricity. Therefore, the benchmark can be based on one of the following figures of merit:

- Total electricity consumption during the test sequence / whole year.
- Ratio of useful heat output divided by total electricity consumption (performance factor).
- Electric energy savings (fsav) compared to a reference system without solar thermal energy that is defined once and for all for a ground source heat pump system and for an air source heat pump system respectively.
3 Unit under test

A complete heating system is installed on the test bench and put into operation the same way as for an installation in the technical room of a detached house\(^1\). This chapter describes, which components are part of the tested system and how the controller of the tested system has to be parameterized. It also defines the boundaries and connections between the tested system and the test bench.

3.1 Physical boundaries

The physical boundaries of the tested system are shown in Figure 1. The following items are considered to be part of the tested system:

- the storage tank(s),
- the auxiliary heating unit(s),
- any additional devices needed to provide DHW to the DHW distribution line (e.g. external heat exchanger and pumps, passive mixing devices for scalding protection if needed)\(^2\),
- any additional devices for feeding the space heat distribution (e.g. space heat pump and temperature control mixing valve),
- the solar group (pumps, valves, controller),
- the connecting lines between the solar group and the solar collector field,
- any prefabricated parts that are necessary to connect the individual parts (e.g. switching valves, tee pieces and piping),
- the controller,
- any temperature sensors needed for the control of the system\(^3\) (including the ambient temperature sensor and the collector sensor). This also includes devices needed to mount the sensors (e.g. sensor pockets).

The solar collector field is not part of the tested system. The reason for this is that it is rather unpractical to put a 10 – 30 m\(^2\) solar collector field under a solar simulator in order to provide reproducible conditions for the irradiation on the collector field for all tested systems. Therefore, the solar collector field is simulated and emulated during the test. For this purpose, the collector has to be tested in advance using the standard testing procedure of EN ISO 9806. During the short cycle test, the collector is emulated in the test rig using a heat source and heat sink circuit that provides the actual realistic collector output power and temperature to the system.

In the case of ground source heat pumps the physical boundary between the tested system and the test bench is at the inlet and outlet of the boreholes' header. In the case of air source heat pumps the evaporator and air ventilator are parts of the tested system.

The total length of the connecting pipes between the solar group and the collector field is defined according to prEN 12977-2: 2007: 10 m each for the flow and return lines (e.g. 10 m

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\(^1\) At present, auxiliary heaters that are installed in the living room to provide heat directly to the ambient (e.g. a tiled stoves) cannot be tested with the harmonized test method.

\(^2\) DHW circulation is not considered to be a typical part of a system for a single family house and therefore not included or not activated if it is an option for the tested system.

\(^3\) Optionally, the ambient temperature sensor and the collector sensor can be emulated by a controllable electrical resistance.
of twin tube). The pipe diameter, insulation thickness and insulation material to be used for the collector circuit have to be clearly specified in the installation manual of the system.

Figure 1: Simplified hydraulic scheme showing the system boundaries where the energy balances are measured.

3.2 Controller

The aim of the whole system test method is to test heating systems under realistic operating conditions. This implies that the unit under test uses its own controller and control strategy. However, the control parameters have to be adjusted in a way that the heat demand of the emulated building will be met. Since the test sequence represents a short term test sequence, the following points have to be considered:

- Averaging of the ambient temperature for the determination of the heating season and the supply temperature set point for space heating has to be switched off or shortened to \( \leq 1 \) h.
- Heating season: The thresholds for start and stop of the heating season for the emulated building are 14 °C and 15 °C, respectively.

Heating curve: The flow rate in the heat distribution system may not be identical for all tested units because it is influenced by the characteristics and control of the space heat pump which is part of the tested system. Accordingly, the heating curve has to be adjusted individually. Figure 2 gives assistance for this purpose. It shows the supply temperature that is necessary to keep a room temperature of 20 °C (when the ambient temperature is -10 °C) versus the flow rate of the heat distribution system. Hence, the resulting flow rate of the heat distribution system with the pressure drop of the building has to be determined by this approach: operate the heat distribution pump with the setting that will be applied in the test and the given pressure drop of the heat distribution system (compare chapter 4.2.2: Pressure
drop for heat distribution system emulation). The flow rate that results with these settings corresponds to the flow rate that is used to determine the heating curve. Once the flow rate is known the heating curve can be adjusted to reach the required flow temperature.

Smart control strategies cannot be tested with the short term whole system test method. Therefore, those functions have to be switched off in the controller.

![Figure 2: Required supply temperature in the heat distribution system to meet the comfort requirements at ambient temperature of -10 °C.](image)

3.3 Interconnection between test rig and unit under test

At the system boundaries, the connection to the test rig that is provided from the test rig side contains an immersed temperature sensor and an effective heat trap of at least 14 cm effective height (distance t in Figure 3), made of stainless steel or other material of comparatively low conductivity (not steel or copper). The temperature sensor is placed as close as possible to the in/out-lets of the system under test, and the connections are insulated according to EN12828.

![Figure 3: Connection without heat rap (left), and with heat trap of effective height t (right).](image)
3.4 Fluids used

The properties of the fluids used in whole system testing must be known. This applies in particular to the heat capacity \( (c_p) \) and density as a function of the temperature and mixing ratio (concentration) for fluids that are mixtures (e.g., anti-freeze fluid in collector loop, brine solution for bore hole loop).

For ground source heat pumps and for collector circuits heat-carrier fluids may be a mixture of water and an anti-freeze additive and thus the properties depend on the mixing ratio. Therefore, the mixing ration must be checked before and after each test.
4 Test bench

The test facility emulates all consumers in a typical single family house in a realistic, thus dynamic way. In addition, the test bench provides the energy that is delivered by the source side of a heat pump and by the collector field.

4.1 General procedure for online simulation and emulation

Emulation can be based on load files or on real-time online simulations. In order to emulate the true response that is dependent to the behaviour of the tested system, real-time online simulations are used to calculate the behaviour of the solar collectors, the building with the heat distribution system, and the ground source heat exchanger.

The procedure for simulation and emulation can be described as follows:

• At the end of each simulation/emulation time step, measured values are passed from the test bench control software to the system simulation software.
• Based on these values, the simulation software is simulating the answer (response) of the emulated device for the next time step and returns the result to the test bench control software.
• During the next time step, the test bench control software controls the emulation of the simulated device, while the simulation software pauses and waits for the next input of measured values from the test bench control software.

The input data for the simulation is based on measurements that were acquired before the emulated time step. The actual values may deviate from those values during the current time step. In order to minimize the error resulting from this deviation, the time steps of the simulation shall not be larger than 2 min.

4.2 Pressure drop of the emulated components

4.2.1 Pressure drop for ground source emulation

The flow-dependent pressure drop curve (system resistance curve) of the ground heat source loop for the 1 x 123 m borehole with double-U pipe is estimated as shown in Figure 4. It can be calculated with Eq. 1 below:

\[ dp = 0.0882 \cdot \dot{V} + 0.215 \cdot \dot{V}^2 \]

Where

- \( dp \) bar, pressure drop, including header, bends etc.
- \( \dot{V} \) m³/h, volume flow rate of the entire ground source loop

The assumptions on which this calculation is based are shown in Annex C.

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4 It has to be noted that if the simulation program uses constants for the specific heat of fluids, the information exchange between the test bench control software and the simulation software must not be based on volume flow rate in combination with supply and return temperatures. Instead, transferred power, mass flow rate, and one temperature should be included in the information exchange.
4.2.2 Pressure drop for heat distribution system emulation

The flow-dependent pressure drop curve (system resistance curve) of the space heating circuit with all thermostatic valves open is estimated as shown in Figure 5 (see Figure 5). It can be calculated with Eq. 3 below:

\[
\Delta p = C_{pipe} \cdot \dot{V}^{1.8} + C_{valve} \cdot \dot{V}^2
\]

where

- \( \Delta p \) bar, pressure drop, with thermostatic valves open
- \( C_{pipe} = 6289 \) mbar/(kg/s)\(^{1.8}\), C-value for piping
- \( C_{valve} = 1800 \) mbar/(kg/s)\(^{2}\), C-value for the thermostatic valves
- \( \dot{V} \) m\(^3\)/h, volume flow rate of the space heat distribution system

More detailed calculations can be found in Appendix E: Pressure drop in the space heating circuit including thermostatic valves.
4.2.3 Pressure drop of the collector field emulation

While the space heat distribution and the borehole heat exchangers have to be identical to all units under test is the collector circuit part of the unit under test, although it is not installed completely (compare chapter 3.1: Physical boundaries). Therefore the calculation of pressure drop has to be done individually, taking into account the pressure drop of the modules (given in the test report) and the piping needed to connect the collectors to a collector field.

4.2.4 Adjusting the pressure drop

The electrical power consumption of a pump is dependent on the pressure loss that the pump has to overcome. For this reason, the pressure drop of the test rig has to be equal to the pressure drop of the component that should be emulated.

However, it is not possible to design a test bench with the same system resistant curve as the emulated component. Therefore, the pressure drop has to be adjusted for the relevant operating point.

The prerequisite is that the test bench has a lower pressure drop than the component to be emulated. To adjust the pressure drop of the emulator, the pump of the unit under test (e.g. the brine pump of an HP) has to be started. Thereafter, the pressure drop of the emulator must be increased by appropriate measures (e.g. a balancing valve) until the resulting combination of flow rate and pressure drop are on the system resistance curve of the respective component.

4.3 Test room

The room where the unit under test will be installed is conditioned to 20 °C ± 0.5 K. Stratified temperature profiles within the room must be avoided. If there is stratification (defined here as temperature difference of > 3 K between air at top and bottom of system), then the temperature must either be measured at several heights and averaged, or taken as an average sampling of different heights by an appropriate device.

4.4 Solar thermal collector field simulation and emulation

The collector simulation and emulation is based on the following pre-requisites:

- Collector performance data (Table 1) that are used for the simulation / emulation of the collector field must be available from tests according to EN ISO 9806 performed by a certified testing institute.
- The manufacturer must provide a drawing of the hydraulics of the collector field from which the following additional data for the collector field can be derived:
  - Number of collectors connected in series.
  - Number of collector series connected in parallel.
  - Additional piping for connections between the collectors and to the supply and return line, including their length, diameter and insulation thickness.
The collector models used for simulation and emulation shall be compatible with EN ISO 9806:2013, including incident angle modifiers as well as thermal capacity. The following rules are applied:

- For collectors connected in series, heat losses (i.e. the collector performance equation) shall be calculated separately for each collector element. Parallel collectors may be summed up to one element for the collector performance calculation.

- Heat losses of additional piping needed to connect the collector elements outside of the collector itself shall be added to the simulation if the total length of these pipings exceeds 1 m.

- If the specific mass flow in collector operation is lower than 50% of the tested specific mass flow at which the performance data has been measured a subtraction of 1% absolute is made on $\eta_0$, i.e. $\eta_0 = \eta_{0,\text{test}} - 0.01$.

- The pressure drop of the collector field is evaluated for the nominal mass flow rate of collector circuit operation during the test. The pressure drop calculation considers the pressure drop of the modules (given in the test report) and the piping needed to connect the collectors to a collector field and has to be measured at the in- and outlet of the emulator.

For collectors that are operated below the ambient temperature, condensation gains or effects of frosting may have to be considered in the simulation. Uncovered collectors may need additional data according to EN ISO 9806.

Table 1: Performance values for the simulation and emulation of the collector field.

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<tr>
<th>value</th>
<th>unit</th>
<th>explanation</th>
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<tr>
<td>A</td>
<td>m²</td>
<td>reference area of one collector element (usually the gross area)</td>
</tr>
<tr>
<td>$\eta_0$</td>
<td>-</td>
<td>zero loss efficiency (based on reference area)</td>
</tr>
<tr>
<td>$a_1$</td>
<td>W/(m²K)</td>
<td>linear heat loss coefficient (based on reference area)</td>
</tr>
<tr>
<td>$a_2$</td>
<td>W/(m²K²)</td>
<td>quadratic heat loss coefficient (based on reference area)</td>
</tr>
<tr>
<td>$C_{\text{eff}}$</td>
<td>kJ/(m²K)</td>
<td>specific heat capacitance of the collector (based on reference area)</td>
</tr>
<tr>
<td>IAM</td>
<td>-</td>
<td>incident angle modifier must be given as $b_0$ and $b_1$, as Ambrosetti-r or as table (bi-directional table in the case of vacuum tube collectors)</td>
</tr>
<tr>
<td>dp</td>
<td>bar</td>
<td>pressure drop of one collector element a)</td>
</tr>
<tr>
<td>mfr</td>
<td>kg/h</td>
<td>mass flow rate at which the collector has been tested and at which the pressure drop has been evaluated</td>
</tr>
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</table>

a) If the pressure drop has not been tested, the pressure drop will be estimated by the testing institute based on the parallel and serial tube lengths inside the collector and the number of elbows / bends in the collector. In the long run, this practice shall be replaced by a meaningful pressure drop test.

The collector sensor of the tested system may either be:

- Installed with good thermal contact and insulation at a point of the test rig hardware where the temperature corresponds to the simulated and emulated collector outlet temperature during the whole test, including standby.

- Replaced by a resistance emulator that provides the resistance that corresponds to the simulated collector outlet temperature during the whole test.
• Placed in a separate conditioning box that provides the simulated collector temperature to the sensor during the whole test.

During times of collector stagnation, the emulated collector loop temperature must not reach the simulated stagnation temperature but must remain at a temperature that is high enough to prevent the controller from turning the pump of the collector circuit on.

4.5 Building load simulation and emulation

4.5.1 General assumptions

The tested system is allowed to deliver heat to the building according to its own control strategy. For this purpose, the unit under test gets the actual ambient temperature of the test sequence via a conditioning box that provides the current ambient temperature where the temperature sensor has to be positioned, or alternatively by emulated resistance that corresponds to the sensor resistance at the simulated temperature (same procedures as listed for the collector temperature sensor).

The space heating load is a result of the climate and the definition of the building with its envelope, air exchange rate, as well as shading and natural ventilation assumptions. The building definition is based on the IEA SHC Task 44 / HPP Annex 38 SFH45 building described in Dott et al. (2013). The building simulation model is based on EN ISO 1379-2008 (simplified version for heating including dynamic effects). The parameters for this building model are given in Annex A. With the help of this real-time online building simulation, the response of the heat distribution system and the room temperature in the building is calculated and emulated. The simulation and emulation includes the pressure drop that is introduced by thermostatic valves and the return temperature of the heat distribution system. The thermostatic valves influence is emulated by a motorized valve that gradually closes and thus reduces the mass flow rate in the heat distribution circuit when the simulated room temperature is increasing from 20 (set point) to 22 °C.

4.5.2 Assuring identical space heat load

One challenge that was faced within the MacSheep project was to combine the advantages of identical space heat load for all tested systems with the request to let the tested system control the space heat distribution (mixing valve and space heating pump) as freely as possible. The previous test practice at the different institutes allowed to reach only one of these two targets, but not both within the same test method. Thus, it was e.g. not possible to perform a benchmark test (with identical heat load), while at the same time allowing the system to interrupt space heat distribution or allow a short-term under-coverage of space heat distribution during domestic hot water preparation.

The procedure described here is the solution that has been developed and tested within the MacSheep project. Since it combines the advantages of two entirely different approaches, it is called the "combined approach".

Pre-defined load

The heat that is delivered to the building during the test is compared on a daily basis to pre-defined 24 h load-files. The pre-defined load files represent the heat supply (energy target) that is necessary to keep the room temperature at 20 °C (compare Annex D).
Detecting and avoiding under-coverage

While the general room temperature set point is 20 °C, the tested system has to deliver enough heat to the building to keep the simulated room temperature above 19.5 °C at all times. If the simulated room temperature drops below this threshold of 19.5 °C or if the pre-defined heat load of the day is not met by the end of the day, the test has to be re-started with adapted control parameters (e.g. a steeper heating curve).

Avoiding excessive heat input to the building

A time-dependent maximum for heat input in excess of the pre-defined load is calculated as the maximum of 4 kWh and 10 % of the pre-defined load at the given time:

\[
\Delta Q_{SH,max}(\tau) = \text{MAX} \left[ 4 \text{kWh}; 0.1 \cdot Q_{SH,set}(\tau) \right]
\]

and the resulting value for \( \Delta Q_{SH,max} \) is gradually reduced to zero within the last three hours of each test-day. When the daily energy that is delivered to the building exceeds the maximum allowed heat delivery (current energy target plus tolerance), the heat supply will be restricted in order not to exceed the target. This is done by means of a motorized valve that either reduces or by-passes the mass flow rate in the emulated heat distribution system. When the heat load target of the day has been reached, the heat distribution system is idle (no flow or flow bypassing it).

This new approach to deal with space heat distribution was successfully tested during a whole system test at HSR SPF. Two valves are needed in the test bench for this purpose (compare Figure 6):

- One motorized valve (V1) that emulates the thermostatic valves and is used to restrict the heat supply to the building by reducing the flow rate.
- For heating systems that do not allow for closing the heat distribution completely (e.g. to avoid high pressure disruption of the compressor of a heat pump) a second valve (V2) is used for emulating an overflow valve that bypasses the heat distribution to the building (the three-way valve in Figure 6).

![Figure 6: Test bench valves for the combined approach.](image-url)
Figure 7 illustrates the progress of the test as measured on day 12 of a 12-day test sequence. It shows the energy target (dark red) and the target with the maximum allowed excess (blue). The difference between them is reduced to zero at the end of the day. In phase “A” of the test, the delivered energy is below the target with the maximum allowed excess. Hence, the motorized valve (V1) on the test bench emulates only the thermostatic valve (in the given example in Figure 7, the valve is closed by 10 – 20% during phase “A” which means that the room temperature is between 20.2 °C and 20.4 °C). In phase “B” of the test, the delivered energy exceeds the maximum allowed excess. Hence, the motorized valve restricts the mass flow further than it would be restricted according to a thermostatic valve only. When the final target of the day is reached (phase “C”), the valve closes completely and disables a further heat supply to the building.

4.6 Domestic hot water draw offs

The following devices are considered to be part of the tested system in order to be able to deliver domestic hot water:

- scalding protection (usually a passive tempering valve set to 52.5 °C) for systems without external heat exchangers
- any heat exchangers, pumps and controllers needed (e.g. for external DHW preparation)

The test rig emulates domestic hot water draw-offs according to a predefined load file. Therefore it has to fulfil several tasks at once:

- Conditioning of the temperature and flow rate of the mains water according to the predefined values.
- Controlling the mass flow rate of the draw off.
- Measuring the supply temperature, calculating the DHW energy, and comparing those values to the required values from the load file.
The flow rates given in the tapping profile are considered to be for a temperature of 40 °C at the point of tapping where the hot water is usually mixed with cold water to reach this temperature. If the supplied temperature from the tested system exceeds 40 °C (Thw in Figure 8), the influence of mixing at the point of tapping is emulated by reducing the flow rate by means of a passive or electronically controlled tempering valve that adds cold water in order to reach 40±1 °C (see Figure 8). As a result, the mass flow rate at the interface between the test-rig and the tested system may be lower than the mass flow rate defined in the tapping profile - just as it would be in a real case.

In order to assure a high level of DHW comfort, the temperature level, that has to be reached, has been defined. This temperature must be reached within a certain time in order to fulfil the comfort requirements. For small draw offs that are considered as unintended hot water tappings this temperature may be identical to the cold water temperature (i.e. no temperature requirement).

![Figure 8: Hydraulic scheme of the draw off loop showing the boundaries between the tested system and the test bench.](image)

The draw-off profiles used for MacSheep are based on statistical distributions of tappings (Jordan & Vajen 2005).

### 4.7 Auxiliary heating devices and energy consumption

By definition, all auxiliary heaters and backup heaters that are required to fulfil the comfort are part of the tested systems and are thus installed in the test bench. This may include:

---

5 unintended hot water tappings occur frequently in current installations with one-handle mixing valves at the point of tapping where the middle position of the handle is a position that mixes hot and cold.
- electric resistance heaters
- oil or gas burners
- pellet stoves or boilers
- heat pumps

Furthermore the electric demand of all controllers, pumps, valves, or other electricity consumers that are needed to operate the tested system must be monitored and evaluated.

As a general rule, all backup and auxiliary heating devices are installed and controlled as recommended by the manufacturer. Time-windows and temperature set points may be increased if needed in order to fulfil the comfort criteria of the test.

4.7.1 Oil or gas burners and pellet boilers

The consumption of oil or gas boilers must be measured continuously and with an accuracy of ±1 % for the total consumption in kg or Nm$^3$ over the whole test sequence. This consumption is multiplied with the upper heating value (gross calorific value) of the fuel that must be determined based on the actual fuel-mixture used during the test from a sample taken prior or during the test sequence, unless the value is known with sufficient accuracy and back-traceability from the supplier of the fuel (e.g. if pure methane from gas bottles is used for gas supply). For natural gas burning devices that are connected to a gas grid with changing gas composition, a volume proportional sampling is required in order to determine the average gross heating value of the consumed gas. For pellet boilers, a conditioning of the humidity of the fuel storage to 50 %rH is needed since the moisture of the wood pellets may adapt to the environment during the course of a test. These changes in moisture may bias the results from continuous weight measurements of the pellets reservoir by a scale (see e.g. also Konersmann et al. 2007).

The combustion air is taken from the laboratory that is conditioned to 20 °C ± 0.5 K, and the chimney draft is controlled to 10 Pa during the test.

4.7.2 Ground source heat pumps

The ground source heat pump is installed within the test bench (indoor units) or within an environment with the correct ambient temperature that can be expected for the particular device (e.g. direct expansion propane heat pumps for outdoor installation).

The physical boundary between the tested system and the test bench is at the inlet and outlet of the boreholes’ header that is simulated and emulated, where calibrated temperature sensors are installed. Between the heat pump inlet and outlet and the borehole header insulated tubes or pipes of 5 m length, an inner diameter of 25 mm and an insulation thickness of 15 mm are installed within an environment that is around 20 ± 2 °C.

The so called EWS model of Wetter & Huber (2007) is used for an online simulation of the borehole$^6$ with IEA SHC Task 44 / HPP Annex 38 parameters for the ground properties, and a borehole length of 1 x 123 m. During the test, the measured inlet temperature and mass flow is given to the simulation model and the return temperature from the simulation is then emulated by the test rig. Since the days of the test sequence represent a whole year within few days, and in reality the ground temperature varies significantly over the year, a method to account for this is required. Two methods have been derived and tested with good results.

---

$^6$ Further development of the test method could include the development of a more generic algorithm for this purpose.
AT SPF a conditioning of the simulated ground temperature before and between the test days has been performed. The simulation switches from the "decelerated real-time synchronous simulation" during a test day to the normal (faster) simulation speed for days of the meteorological year that are between the days that were selected for the test cycle. The load that is applied to the borehole during these days corresponds to a predefined load which is the same for all tested systems.

At SERC, no inter-day simulation was performed, rather a representative starting temperature for the borehole was derived by optimizing the initial conditioning period for the borehole model such that the test results could be extrapolated to annual results in the normal way.

4.7.3 Air source heat pumps

A climatic chamber is used to condition the ambient air temperature and humidity according to the climatic conditions that are also used for the simulation and emulation of the building and of the solar thermal collectors. The evaporator unit of the air source heat pump is placed in this climatic chamber. Thus, a realistic operation of the heat pump is ensured that contains, among other things, the defrosting of the evaporator. Care has to be taken that the air velocity in the climatic chamber is high enough to suppress short-circuits from the evaporator outlet to the inlet, and low enough to not significantly influence the air flow over the evaporator when the ventilator is on.

4.7.4 Accuracy of electricity consumption measurements

As long as the measured electricity consumption for a group of devices is lower than 10 % of the total measured end-energy consumption (including fuel energies), an accuracy of ± 3 % for the measurement of the electric energy consumption is acceptable. For devices with larger contribution – e.g. heat pumps - the accuracy should be within ± 1 %.
5 Boundary conditions for the climate and load

The test aims to give a representative result for the annual energy consumption to cover the heat demand of a single family house. The building definition can be found in chapter 4.5. In this chapter, the annual climate and the weather data for the short term test sequence that is applied during the test are described. Two new test sequences have been developed within the MacSheep project: A 12-day test sequence and a 6-day test sequence.

5.1 Annual climate and load

5.1.1 Weather data

For the simulation of solar thermal systems mainly hourly based weather data are used today. However, up to now it was unclear whether short time fluctuations of solar irradiation that are not present in hourly, artificial data, may have an influence on the benchmarking of systems. In order to analyse the impact of measured weather data with smaller time resolution on the result of solar thermal systems, a simulation study with six minute and hourly weather data from Zurich (Switzerland) has been performed in TRNSYS17 by Granzotto et al. (2014) for the MacSheep project. It was found that the impact of higher resolution weather data is only significant when simulating a system with an unrealistic low thermal capacity of the solar collector loop. For this reason, hourly weather data is used for the harmonized MacSheep test cycles.

The climate of Zurich was chosen for the whole system test. The data is taken from the TRNSYS extended weather data that uses hourly Meteonorm (2009) data.

5.1.2 DHW load

The DHW load profile was originally derived with the program DHWCalc developed by Jordan, with the theory described in Weiss (2003). The profiles have been simplified for the MacSheep project while maintaining a variation in flow rates and a large number of discharges with small flow rates in order to be realistic and to ensure that the systems tested with the method can supply for a wide range of conditions. The DHW load is summarized in Table 2.

<table>
<thead>
<tr>
<th>Table 2: Key data for the annual DHW-load.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total Volume (45-10 °C)</td>
</tr>
<tr>
<td>Average Number of tappings per day</td>
</tr>
<tr>
<td>Annual discharge energy for Zurich climate</td>
</tr>
<tr>
<td>Max Flow over TRNSYS time step</td>
</tr>
<tr>
<td>Min Flow over TRNSYS time step</td>
</tr>
<tr>
<td>Max Flow for UA-Calculation</td>
</tr>
<tr>
<td>Min Flow for UA-Calculation</td>
</tr>
</tbody>
</table>
5.2 Twelve day test sequence

5.2.1 Context

The aim of this work was to find an optimized sequence of 12 weather data days picked from 365 days annual data. The optimization process was defined according to the principle that:

- the performance results obtained for a tested system with the 12-day test sequence, shall be representative for the performance over the whole year;
- space heat load, DHW load, and auxiliary energy consumption can be extrapolated to annual results by the factor 365/12\(^7\);
- the chosen system used for the optimization process is the MacSheep reference solar and air source heat pump system with 10 m\(^2\) flat plate solar collector and 750 L buffer water storage.
- the first objective is to minimize the deviation between the values obtained by extrapolation from the short cycle test results and the annual simulation results for of the total electricity consumption of the whole solar and heat pump (SHP) system.

![Figure 9: Data exchange and energy flows between a real air source SHP system and the emulated environment during a test on a semi-virtual test bench. Qdhw: energy provided to DHW preparation - Wel: total electricity consumed by the system - Qsh: energy provided for space heating – QintSto: internal energy content of the thermal storage](image)

5.2.2 Methodology

The 12-day test sequence is a combination of a draw-off profile and a weather sequence applied to the system to be tested. For the evaluation of the 12-day test sequence, an additional initialization day before day 1 is necessary to initialize the system before starting to evaluate the energy balances of the core 12 days. This day 0 has the same profile as day 12.

---

\(^7\) This approach was applied previously by Albaric, M., Nowag, J., and Papillon, P. 2008 to solar combisystems combined with gas boiler.
The distribution of DHW draw-offs during the 12-day test sequence was adapted from the 12-days load file used previously at HSR SPF. One premise is to reach an identical state of energy stored within the storage after the test sequence as was present at the beginning of the test sequence. For this purpose, a relatively large amount of DHW energy needs to be drawn from day 9 to day 12. The original draw off profile was modified slightly and the total amount of the DHW load during the whole test was adjusted by +0.4% applied on every energy draw-off in order to meet the requirement that an extrapolation of the test results with the factor of 12/365 corresponds to the DHW energy drawn in the annual simulation. A comparison of the original (Ori) and the modified (Mod) profile is shown in Figure 10.

![Figure 10: Distribution summary of the original and modified draw-off profile for 12 days.](image)

The search of an optimal weather sequence for 1 + 12 days is achieved by a software tool that combines iterative TRNSYS simulations of the MacSheep reference system (see report D7.3) and post-processing of simulation results thanks to a custom Scilab automation and optimization module. The objective function that is minimized is shown below:

\[
\text{Eq. 4} \quad \min \sum_{i=1}^{12} (Q_{\text{SH,TS}}(i) - Q_{\text{SH,month}}(i))^2 + (Q_{\text{DHV,TS}}(i) - Q_{\text{DHV,month}}(i))^2 + (W_{\text{el,TS}}(i) - W_{\text{el,month}}(i))^2 + (Q_{\text{INTSTO,TS}}(i) - Q_{\text{INTSTO,month}}(i))^2
\]

The tool simulates once the reference system for 8760 h (one year) and calculates monthly energies for Qsh (space heat), Qdhw (domestic hot water), Wel (electricity consumption), QintSto (energy stored in the storage) as shown in Figure 9. The optimization process is started with an initial weather sequence of 1+12 days: these initial days are chosen to be close to the monthly average values of ambient air temperature (Tamb), total irradiation on horizontal plane (Ith) and total irradiation on collector plane (Itcoll) of the annual weather data. Then the algorithm simulates the 1+12 days of the test sequence: daily values for Qsh, Qdhw, Wel, QintSto are calculated from this simulation.
Then the tool calculates deviations between monthly averaged values and test sequence extrapolated (multiplied by 365/12) values and uses this deviation to iteratively search for days within the yearly weather database with modified characteristics for Tamb, Ith and Itcoll.

### 5.2.3 Resulting 12 day test sequence

The resulting optimized 12-day weather test sequence shows 1 % deviation between the 12-day extrapolated and the 365 day simulated total electricity consumption while the SH and DHW load deviations are below 1 %. An overview of this weather test sequence is represented in Figure 11.

![Figure 11: Overview of the optimized 12-days test sequence.](image)

Table 3: Daily overview of the 12-days weather test sequence averaged ambient temperature, sum of total horizontal irradiation and sum of total irradiation in the solar collector aperture plane.

<table>
<thead>
<tr>
<th>Day of the year</th>
<th>Tamb averaged [°C]</th>
<th>Ith daily sum [kWh/m²]</th>
<th>Ita daily sum [kWh/m²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>363</td>
<td>0.2</td>
<td>0.823</td>
<td>1.056</td>
</tr>
<tr>
<td>307</td>
<td>0.4</td>
<td>1.557</td>
<td>2.021</td>
</tr>
<tr>
<td>80</td>
<td>5.0</td>
<td>3.615</td>
<td>4.466</td>
</tr>
<tr>
<td>105</td>
<td>7.8</td>
<td>5.137</td>
<td>5.825</td>
</tr>
<tr>
<td>145</td>
<td>14.5</td>
<td>4.891</td>
<td>4.610</td>
</tr>
<tr>
<td>148</td>
<td>19.4</td>
<td>4.511</td>
<td>4.017</td>
</tr>
<tr>
<td>236</td>
<td>17.4</td>
<td>5.069</td>
<td>5.442</td>
</tr>
<tr>
<td>175</td>
<td>18.5</td>
<td>3.963</td>
<td>3.496</td>
</tr>
<tr>
<td>88</td>
<td>6.0</td>
<td>2.265</td>
<td>2.243</td>
</tr>
<tr>
<td>20</td>
<td>1.6</td>
<td>1.000</td>
<td>1.133</td>
</tr>
<tr>
<td>20</td>
<td>1.6</td>
<td>1.000</td>
<td>1.133</td>
</tr>
<tr>
<td>50</td>
<td>1.5</td>
<td>1.621</td>
<td>1.770</td>
</tr>
</tbody>
</table>

Tamb = ambient air temperature; Ith = total irradiation on horizontal plane; Ita = total irradiation on aperture area of the collector.
5.2.4 Validation

The representativeness of the resulting 12-day test sequence was investigated with a number of variations of the system configuration:

- so called “4-pipes” and “3-pipes” hydraulic scheme. The different numbers of pipe connections between the heat pump and the solar combistore respectively improves or decreases the overall system performance due to better or worse exergetic performance of the storage solution
- up to 50% increased solar collector area
- up to 33% higher thermal energy storage volume,
- up to 100% higher effective vertical thermal conductivity of the store for representing less stratification efficiency,
- ±5 % variation in the overall performance of the heat pump (changes in compressor and heat exchanger efficiencies).

The validation results (Table 4) show deviations between the annual and extrapolated values (V) below 4 % for total electricity consumed by the system and below 3 % for the seasonal performance factor of the whole system including the space heat distribution pump (SPF\textsubscript{SHP+}, as defined in report D7.3).
5.3 Six day test sequence

5.3.1 Context

One of the goals of the MacSheep project was to see if a six day test sequence can be found that fulfills the criteria of representativeness for the whole year’s energy balance just as well or with only a small decrease inaccuracy compared to a twelve days test cycle. Using six days instead of a twelve day test method reduces the time requirements for system testing and thus the cost and time-effort associated with testing.

5.3.2 Methodology

The starting point for this work is the test method for solar combistores developed by Bales (2004) and then further developed for testing of solar and pellet heating systems by Petterson et al. (2011), Persson et al. (2012) and Dalenbäck et al. (2011). The climatic conditions (air temperature and solar radiation) for the six days were adjusted using Genopt (Wetter, 2011) so that the solar gains, space heating demand and the systems total electricity demand were close to the simulated annual energy after multiplication with \(\frac{365}{6}\).

Eq. 3 below shows the objective function that was minimized.

\[
\text{Objective} = |\text{Wel}_{6\text{day}} - \text{Wel}_{\text{year}}| + |\text{Wcoll}_{6\text{day}} - \text{Wcoll}_{\text{year}}| + |\text{Wheat}_{6\text{day}} - \text{Wheat}_{\text{year}}|\]

\(\text{Wel}\) is the total electricity used, \(\text{Wcoll}\) is the energy delivered by the solar collector to the store and \(\text{Wheat}\) is the heat delivered to the floor heating system (space heating demand). The system used for the optimization was the MacSheep reference system with an air to...
water heat pump and the building was modelled using the ISO model 13790-2008 that was implemented into a TRNSYS Type by CEA INES (compare chapter 4.5 and Annex A). Besides the standard system also a system with a larger solar collector and store size were considered in the optimization work. The annual climate data file was the CH-Zuerich-SMA-66600.tm2 file.

The hot water load profile was based on a selection of the days used in the concise cycle test (CCT) method by SPF previously. The order of these days was reorganized by trial and error in combination with GenOpt optimization to achieve a low objective value. In the simulations two initial days (identical with day 5 and 6) were used for preconditioning of the system.

After the climate and DHW-load profiles were determined based on simulations with a solar and air source heat pump system, also a solar and ground source heat pump system was simulated. In order to meet the direct extrapolation criteria, the initial average ground temperature was adjusted until the extrapolated electricity demand from the six days was the same as for the annual simulation. Finally a parametric study was performed to investigate the deviation simulating different system variants.

### 5.3.3 Resulting six day sequence

The thermal capacity of the simulated floor heating system (that was based on the definitions of the IEA SHC Task 44 / Annex 38) had to be reduced to the capacity of a radiator in order to avoid a high time lag between energy being charged into the floor and energy being emitted from the floor to the building air. Without this adaption, a significant transfer of heat delivered on one day to heat consumed in subsequent days of the test sequence created problems for the repeatability of the results of the whole test sequence.

The order of the days in the test sequence was changed compared to the six day test by Bales according to Figure 14 so that both the start and the end of the test sequence are winter days, similar to the CCT and Short Cycle System Performance Test (SCSPT) test methods of CEA INES and SPF (Albaric et al., 2008; Vogelsanger, 2003; Haller & Vogelsanger, 2005; Haberl et al., 2008).

It was found that the DHW load profile was a key parameter to reduce the objective value. The spring and summer DHW-load has a strong influence on the possible solar gains of the system. In the last step of optimization only the electricity demand was included in the objective function and it was found that the objective value of systems with different sizes could be reduced by increasing the DHW-load during day 4 and reducing solar radiation during day 3. The final version of the DHW-load profile is presented in Figure 13. The final outdoor temperature and the solar radiation is presented in Figure 14. The solar radiation is reduced by about 2 % and the outdoor temperature is reduced by 1.8 °C compared to the original data file.
Figure 13: The hot water load file used in the six day test sequence.

Figure 14: The ambient temperature and solar radiation on collector plane for the six day test sequence.

Figure 15 shows the simulated ground temperatures for the reference ground source heat pump system during the six day test after the initial average ground temperature was optimised to 4.7 °C. Compared to the annual simulation, the ground temperatures during the summer days in the six day test are much lower.
5.3.4 Validation

The parametric studies presented in Figure 16 and Figure 17 show that the simulated electricity demands from the six day test are within 5% of the annual electricity demands for all the studied variations, except for the ground source heat pump system (Figure 17) with DHW sensor moved and for a 75% heat pump size where the electric heater enabled (in the reference system, the electric heater is disabled when the ambient temperature is above -7 °C).

Figure 16: Deviation in electricity demand between annual and six day simulation results for design variations of the air to water heat pump system.
Figure 17: Deviation in electricity demand between annual and six day simulation results for design variations of the ground source heat pump system.
6 Testing of smart control features

The term "smart control" is a popular selling argument that is used increasingly also for the control of HVAC systems or parts of these. As there is no generally agreed upon definition for smart control, the meaning differs from author to author. Within the MacSheep project, the following features have been identified as being possible candidates for making the control smarter than the ordinary control:

- **Self-learning algorithms**
- **Use of ICT** - using information from outside the heating system / information that is not available without additional sensors, or communication with the operator or owner by internet or mobile phones.
- **Automatic fault detection**
- **Forecasting** of heat load or availability of energetic resources and their price, possibly also including **model predictive control** (MPC)

Quite often, a combination of different smart control features is used to achieve a certain goal. For example, a forecast of the building load may also use data that has been obtained by ICT and/or may be based on self-learning algorithms for the prediction of the occupancy. Another example is fault detection based on the deviation of a forecast for solar yield from the thereafter measured values. Thus, a strict separation between different kinds of smart control features may not be possible.

In the following, the possibilities for including tests of smart control features into the whole system test methods are discussed.

### 6.1 Self-learning algorithms

A self-learning algorithm may be used in order to learn the behaviour of a building, the occupancy of the building or the DHW demand. The algorithm may detect patterns that follow a seasonal, weekly or intra-day scheme, or events that trigger a certain reaction (e.g. drop of outside temperature or pressure that precedes an increased demand for space heating).

Self-learning algorithms usually need several weeks in real time to adjust the parameters of a model. Therefore the short duration time of the performance test is not suitable to evaluate the performance of such procedures. Besides, the learned user behaviour would only be representative for one user behaviour boundary condition, and the robustness for other types of user behaviour or changing user behaviour cannot be tested within a short time or without additional tests.

Thus, self-learning algorithms are not suitable candidates to be tested with the short-time whole system test methods and must be switched off during whole system testing. However, it is imaginable to perform two system tests, one with "standard" control configuration, and one with a pre-defined learned pattern, in order to evaluate the difference between an untrained and an "ideally trained" system – although the relevance of such a comparative test may be questionable.

### 6.2 Use of ICT and variable electricity price information

For testing systems that use ICT features a communication with the control device using standard (free) protocol and database format or a format that the manufacturer is willing to
share with the testing institute is required. This requirement itself is not a problem and the hard- and software of the test-bench can be used to emulate the communication device that is outside of the tested system's boundaries. Thus, the following features may be included in the test procedures:

- **Communication of detected errors** or informative information about the heating system from the controller to the end user.

- **Communication of future electricity prices** (exact) to the tested system. The effect of (exact) electricity price information on the system behaviour may be tested by performing two system tests, one where the price information is available to the system, and one where it is not available. A comparison between the two test results may reveal the advantage of the price information for the owner in terms of a reduction in energy purchase cost. Thus, a new target function for the benchmarking is introduced with the cost of final energy consumption rather than the amount of final energy consumption (the final energy consumption may be higher when using price information). A profile for the electricity prices at different times during the test may be based on available data for a particular location of today that matches the weather data profile. However, this electricity price profile may not be representative for other countries of today, and even less so for the future.

- **Communication of remotely generated weather or electricity forecast** that may be subject to inaccuracies is not as straight-forward as the two items above and is dealt with more in detail in the section on forecasting.

Thus, testing for ICT features of the controller during a whole system test seems to be possible with some restrictions, possibly requiring a second test-run if the effect of ICT technology on the benchmark result shall be evaluated.

### 6.3 Automatic fault detection

Automatic fault detection may be tested by introducing faults into the system on purpose. For example, a sensor may be disconnected, or an installation error may be committed on purpose, in order to see if the system controller detects the fault. This would require first of all a list of the most relevant errors to be checked for and the definition of a procedure for the introduction of the error into the system under test. However, the errors may not be introduced into the system during the benchmark test as this would bias the results dramatically. Thus, although testing for fault detection seems to be possible, additional test sequence(s) or a second test-run would be needed in order to perform these tests.

### 6.4 Forecasting and model predictive control

Stochastic processes that influence the heat demand or the availability and price of energetic resources may be anticipated by forecasts.

Looking at stochastic processes that influence the heat demand for space heating and domestic hot water, weather and user behaviour (building occupancy and DHW consumption) are the most obvious. Some system controllers include strategies that anticipate those random processes to avoid overheating of the building by the system, for example when large passive solar gains are expected. Another example is the forecast of absence of the occupants that deviates from a static occupancy time schedule on a daily or
weekly base. This anticipation of the future weather and user behaviour (typically only 24 h to few days ahead) may be combined with deterministic models of the building to calculate the heating demand that the system shall provide in the near future. Thus, control values may be adapted in order to match the anticipated need for space heating and DHW consumption. In order to find the best control values, model predictive control (MPC) may be used. Herein, control parameters are optimized using model based simulations.

Forecasting may be based on self-learning algorithms for the user behaviour, availability of solar resource, etc.

Weather forecast and forecast of irregular electricity prices is often not implemented in the local control of the heating system itself, but obtained by internet or other ICT technology

In general, a short duration test of systems whose controller includes forecast is not straightforward for the following reasons:

- The reduction of a whole year weather sequence to a few test days is not creating representative transitions between the days. With other words, the forecast of a value for tomorrow on day two of the test would correspond to the following day in the meteorological year, which is not the day that follows in the test sequence.
- Forecasts that do neither rely on information from the outside (remote database server, ICT) nor on self-learning algorithms are hard to imagine, thus likely to be non-relevant.
- Forecasts that rely on self-learning algorithms are not suitable for being tested by whole system test methods (see section on self-learning algorithms above).
- For systems that are using remotely generated weather or electricity forecast from web servers the ICT and MPC section above equally applies. These forecasts are subject to uncertainties. The question arises whether the forecast that is communicated to the system in the test bench is a "correct" forecast or is also subject to inaccuracies that are similar to "real" forecasts. The creation of artificial inaccuracies for an already known future value (e.g. irradiation on the next day) is not an easy task, may require a large effort, and may not be representative for different sources of forecasts of today and the future. Thus, the introduction of such uncertainties into the already exactly known data is unlikely to be worth the effort.
7 Execution of the test

The manufacturer or a designated installer installs the system in the test room. After that, the controller is parameterized by the installer. For this purpose, some preliminary tests are necessary to check the rotation speed of the pumps, the heating curve of the controller and the accuracy and set temperatures of the domestic hot water mixing valves.

Before the test starts the storage tank of the unit under test has to be brought to reasonable temperatures. This may be achieved by enabling the auxiliary heater to heat the upper part of the tank to its auxiliary set temperature.

When the test has started, no adjustments of the controllers are allowed and thus the unit under test has to act autonomously to cover the heat demand for space heating and domestic hot water preparation. Figure 18 gives an overview of the schedule during the test.

The final 24 hours of the test sequence (day N of the N-days test sequence) are added as day 0 before the beginning of the core phase. This time is necessary to precondition the temperature in both, the installed storage tank and the simulated and emulated building and its slab.

The concept of the test is a real cycle. Thus runs the test after the conditioning phase seamless into the core phase and starts from the beginning again. The reason for this is to assure that the state of energy of the system at the beginning of the core-test sequence does not deviate significantly from the state at the end of the test sequence. When the energy consumption of the unit under test during the first day of the second run is identical to the energy consumption of the first day in the first run, one can assume that the energy stored in the thermal mass of the tank and in the slab of the emulated building at the beginning of the test sequence is identical to the energy that is stored in the end of the test sequence (phase “A”). When the energy consumption of the first day of the second run is not identical to the energy consumption of the first day in the first run, the test has to continue. The next chance to reach the end criteria is at the end of day two of the second run – than phase “B” has to be evaluated – or at the end of day three of the second run (phase “C”).

![Figure 18: Schedule of the test.](image-url)
8 Additional tests

The following tests are suggested in addition to the core test sequence:

- A storage tank heat loss test may be applied according to EN12977-3.
- Additional DHW comfort tests may be applied, e.g. based on EN13203.
- Additional tests for the evaluation of compactness and ease of installation that contain also criteria for the evaluation of fail-proofness (e.g. heat trapping and avoidance of unwanted natural convection) are proposed in Annex C. Included in these tests are:
  - **heat trap test**: during the storage heat loss test, the temperature at the point of connection of the system with the space heat flow and return line as well as the collector flow and return line (point between solar group and collector pipes) are measured and the temperature difference between this temperature and the room divided by the temperature difference between the storage tank and the room is reported. If this value is larger than 20 % (or 0.2), the system is lacking effective heat traps and these must be installed additionally on-site.
  - **unwanted circulation test**: during system standby, a pressure difference of ± 0.01 bar is applied by the test-bench to the space heating loop and to the solar loop. If the storage temperatures are affected by this measure, unwanted circulation is not prevented by the unit under test. The pressure difference must be applied with a holding time of 10 min at 10 mbar increments. The maximum pressure difference is 20 mbar in case of the building loop and 50 mbar in case of the collector loop.
9 Conclusion and outlook

The test method described within this report combines different whole system test methods that were applied at the different test institutes SERC, INES and SPF to a harmonized test method. For this purpose, both the boundary conditions for the test as well as the requirements of the test facility were defined.

Hourly Meteonorm data for the climate of Zurich were chosen for the whole system test. Starting from the annual climatic data-set, two new representative test sequences were developed: a twelve day sequence and a six day sequence.

The heating load of the building was determined and implemented based on ISO standards. Thus, the test is independent from existing commercial simulation software.

The physical boundaries of the unit under test were determined such that a fully autonomous operation of the unit under test is possible. Thus, a realistic operation of the system is ensured.

All of the existing whole system test methods had advantages and disadvantages. In the course of harmonization, the advantages of the different methods have been combined. Especially worth mentioning are:

- The new test method is a benchmark test. This means that the load for space heating and domestic hot water preparation is identical for all tested systems.
- The complete heating system can be tested. This includes all devices that are typically installed in the technical room and that are necessary to deliver domestic hot water and space heating, and it includes also all control devices.

The test method described in this report is ready to be used for whole system tests, but the development has not been completed yet. Pending issues are:

- A procedure to emulate a realistic pressure drop of thermostatic valves, based on the calculations shown in Annex C.2, Pressure drop in the space heating circuits including thermostatic valves.
- A procedure for the emulation of the ground source during the test sequence (two procedures have been demonstrated during the project, but there is no agreement on which of the methods should be used.

Three prototypes that have been developed within the MacSheep project will be tested in 2015. Thereby, both test sequences (6-day and 12-day test sequence) will be applied and their applicability and representativeness demonstrated.

Both test sequences have been developed with the aim to allow a direct extrapolation of the test results to annual performance figures. The simulations that were performed for the determination of the sequences showed that the extrapolation gives good results for a range of system variations. For the six day test sequence and ground source heat pump the maximum deviations were slightly larger. When the final developed MacSheep solar and heat pump systems will be tested with these methods, the measured data of the systems will be used to validate simulation models of the systems and the validated models will be used to calculate annual performance figures. These results will be compared with the extrapolated results of the short cycle tests and thus serve for further validation of the short cycle test sequences.
## 10 List of abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>fsav</td>
<td>electric energy savings</td>
</tr>
<tr>
<td>CCT</td>
<td>concise cycle test</td>
</tr>
<tr>
<td>DHW</td>
<td>domestic hot water</td>
</tr>
<tr>
<td>HP</td>
<td>heat pump</td>
</tr>
<tr>
<td>HVAC</td>
<td>heating, ventilating and air conditioning</td>
</tr>
<tr>
<td>ICT</td>
<td>information and communication technology</td>
</tr>
<tr>
<td>MPC</td>
<td>model predictive control</td>
</tr>
<tr>
<td>SCSPT</td>
<td>short cycle system performance test</td>
</tr>
<tr>
<td>SH</td>
<td>space heating</td>
</tr>
<tr>
<td>SHP</td>
<td>solar and heat pump</td>
</tr>
<tr>
<td>WP</td>
<td>work package</td>
</tr>
</tbody>
</table>
11 Bibliography


Annex A  Parameters for the ISO-building model

The parameters were determined according to IN ISO 13790:2008 and the description of the reference building SFH045 of Task 44 (Dott et al. 2012).

A.1 Parameter list

<table>
<thead>
<tr>
<th>Par</th>
<th>Value</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>29 050 000</td>
<td>J/K</td>
<td>$C_\text{m}$, thermal capacitance of zone (ISO 13790-2008: 12.3)</td>
</tr>
<tr>
<td>2</td>
<td>433</td>
<td>m$^2$</td>
<td>$A_\text{m}$, surface of the effective thermal mass (ISO 13790-2008: 12.2.2)</td>
</tr>
<tr>
<td>3</td>
<td>140</td>
<td>m$^2$</td>
<td>$A_\text{f}$, surface of the heated floor area (ISO 13790-2008: 6.4)</td>
</tr>
<tr>
<td>4</td>
<td>34.5</td>
<td>W/K</td>
<td>$H_{tr,w}$, window heat transfer coefficient (ISO 13790-2008 : ANNEX A)</td>
</tr>
<tr>
<td>5</td>
<td>0</td>
<td></td>
<td>Mode, 1 = coupling with external radiator (only part of solar gains to the building, rest to the soil surface, this mode cannot be found in ISO; 2 = ISO: all solar gains to the building</td>
</tr>
<tr>
<td>6</td>
<td>3.45</td>
<td>W/(m$^2$K)</td>
<td>$h_{\text{ls}}$, Heat transfer coefficient between the inner air and the star-node / wall surface (3.45 W/m$^2$.K according to the standard)</td>
</tr>
<tr>
<td>7</td>
<td>9.1</td>
<td>W/(m$^2$K)</td>
<td>$h_{\text{ls}}$, heat transfer coefficient between the thermal capacity and the inner surface / star-node (9.1 W/m$^2$.K according to the standard)</td>
</tr>
<tr>
<td>8</td>
<td>4.5</td>
<td></td>
<td>$X_{\text{r}}$, ratio between the surface of the inner walls and the heated floor (4.5 according to the standard)</td>
</tr>
<tr>
<td>9</td>
<td>4</td>
<td></td>
<td>$N_{\text{surf}}$, number of external surfaces for passive solar gains</td>
</tr>
</tbody>
</table>

A.2 Inputs to the model

- The time-dependent inputs for the temperature of the ambient air and for the temperature of the air supply are taken from the weather data.
- The power from the heat distribution system (or heat emission devices) is an output of the simulation of the heat distribution system.
- Internal gains (Eq. 4) are calculated with include gains from people present in the building and gains from equipment as well as the ground coupling losses as defined in Dott et al. (2012), and Haller et al. (2012). The ground coupling losses are included here for reasons of simplicity and because they were not foreseen in the ISO model.
- The ventilation heat loss coefficient (Eq. 5) is calculated based on the rate of infiltration of air $i_{\text{inf}}$. For the moment, $i_{\text{inf}} = 0.4$ is taken as a constant. However, for systems that tend to overheat the building above outside air temperature, $i_{\text{inf}}$ may be taken as a function of the room temperature just like in Task 44 / Annex 38 shown in (Dott et al. 2013), assuming that the occupants will open the window if the room temperature increases excessively and the outside air temperature is lower.
- The overall heat transfer coefficient of the outside walls and the roof is $H_{tr,\text{on}} = 67.55$ W/K.
- The insolation on the four surfaces (the building is oriented exactly north-south) is obtained from an annual simulation with the radiation processor. In the case of a 12-day test the insolation has to be pre-calculated with an annual simulation due to the dependency of the radiation processor of TRNSYS on the simulation time.
- The effective reception areas of the windows of the four surfaces are: $A_{\text{north}} = 1.43$ m$^2$; $A_{\text{south}} = 5.71$ m$^2$; $A_{\text{east}} = A_{\text{west}} = 1.90$ m$^2$. 
The non-shading factors are 1, and the IR-exchange coefficients are $14 \cdot A$ (i.e. $20 \text{ kJ/h}$, $80 \text{ kJ/h}$, $27 \text{ kJ/h}$, $27 \text{ kJ/h}$).

\begin{align*}
\text{Eq. 6} & \quad \phi_{\text{int}} = N_{\text{pers}} \cdot 100 [W] + \hat{Q}_{\text{gain,eq}} + \hat{Q}_{\text{hui,grd}} \\
\text{Eq. 7} & \quad H_{\text{ve}} = 0.34 \left[ W/(K \cdot m^3) \right] \cdot V_{\text{hui}} \cdot i_{\text{inf}} = 52.13 [W/K]
\end{align*}

### A.3 Calculation of ground coupling heat losses

Simplified ground coupling losses are calculated with Eq. 6 that is based on an approach presented in (ISO/DIS 13370 2005). Positive values of $\dot{Q}_{\text{ground}}$ correspond to heat gains from the ground, negative values to heat losses to the ground.

\[ \dot{Q}_{\text{ground}} = L_s \left( \bar{\theta}_s - \bar{\theta}_i \right) - L_{pl} \cdot \Delta \theta_i \cdot \cos \left( 2\pi \frac{t-t_{\text{shift}}-\alpha}{t_0} \right) - L_{pe} \cdot \Delta \theta_e \cdot \cos \left( 2\pi \frac{t-t_{\text{shift}}-\beta}{t_0} \right) \]

The values needed for calculating the time-dependent ground coupling heat losses according to Eq. 6 are presented in Table 5. The assumptions behind these values are shown in Table 6.

#### Table 5: Parameters for the calculation of ground coupling losses for SFH45 in Zürich.

<table>
<thead>
<tr>
<th>symbol</th>
<th>explanation</th>
<th>unit</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\bar{\theta}_s$</td>
<td>average outside temperature over the year (solar or snow influence on ground surface temperature may be included here)</td>
<td>°C</td>
<td>9.1</td>
</tr>
<tr>
<td>$\Delta \theta_s$</td>
<td>amplitude of sine-curve that approximates the outside temperature variation over one year</td>
<td>K</td>
<td>9.3</td>
</tr>
<tr>
<td>$\bar{\theta}_i$</td>
<td>average inside temperature over the year</td>
<td>°C</td>
<td>24.2</td>
</tr>
<tr>
<td>$\Delta \theta_i$</td>
<td>amplitude of sine-curve that approximates the inside temperature variation over one year</td>
<td>K</td>
<td>4.2</td>
</tr>
<tr>
<td>$t_{\text{shift}}$</td>
<td>time-shift of lowest temperature in the year based on the sine-curve that approximates the outside temperature over the year</td>
<td>h</td>
<td>488</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>additional time-shift for inside temperature variation</td>
<td>h</td>
<td>170</td>
</tr>
<tr>
<td>$\beta$</td>
<td>additional time-shift for external temperature variation</td>
<td>h</td>
<td>1019</td>
</tr>
<tr>
<td>$L_s$</td>
<td>steady state thermal coupling coefficient</td>
<td>W/K</td>
<td>10.6</td>
</tr>
<tr>
<td>$L_{pl}$</td>
<td>internal periodic thermal coupling coefficient</td>
<td>W/K</td>
<td>10.76</td>
</tr>
<tr>
<td>$L_{pe}$</td>
<td>external periodic thermal coupling coefficient</td>
<td>W/K</td>
<td>6.2</td>
</tr>
</tbody>
</table>
Table 6: Values from the IEA SHC Task 44 / Annex 38 used to derive the values shown in the previous table.

| symbol  | explanation                                | unit    | value
|---------|--------------------------------------------|---------|-------
| $\lambda_{grd}$ | conductivity of the ground                  |         | 2.0 W/mK |
| $\rho_{grd}$    | density of the ground                       |         | 2500 kg/m$^3$ |
| $c_{p,grd}$     | heat capacity of the ground                 |         | 0.8 kJ/kgK |
| $G_t$           | geothermal gradient                         |         | 0.025 K/m |
| $l_{floor}$     | length of the floor in contact with the ground |       | 10 m   |
| $b_{floor}$     | width of the floor in contact with the ground |       | 7 m    |
| $w$             | thickness of the walls around the floor in contact with the ground |   | 0.4 m |
| $R_{tot}$       | total thermal resistance of floor with inside and outside coefficients ($= R_{in} + R_f + R_{out}$) |   | 5.464 Km$^2$/W |
| $\psi_{gf}$     | Linear thermal transmittance associated with wall/floor junction |   | 0.0 W/mK |
Annex B  Dealing with measurement equipment inside the systems boundaries

If measurement equipment (e.g. flow meters or temperature sensors) are installed within the tested systems boundaries, the influence of these measurement devices has to be either negligible or compensated for.

In the case of pressure drop caused by flow meters or temperature sensors, including all additional installation needed to insert them into the system, the following procedure shall be applied:

The pressure drop of the respective measurement equipment at room temperature must be available or recorded for the range of mass flows that corresponds to the massflow in the loop that they are installed ($\Delta p_{\text{eq}}(\dot{m})$). After the installation of the flow and temperature measurement equipment, the pump(s) of the respective loop(s) are switched on and both flow and electric consumption of the pump(s) are recorded. The pressure head of the pump at the respective flow and electricity consumption is taken from the manufacturers pump specifications at the respective mass flow $\Delta p_{\text{pump}}(\dot{m}, P_{\text{el, pump}})$. The share of pressure drop caused by the measurement equipment will be determined for every time step of the system test as shown in Eq. 4.

$$f_{\text{equip}}(\dot{m}, P_{\text{el, pump}}) = \frac{\Delta p_{\text{eq}}(\dot{m})}{\Delta p_{\text{pump}}(\dot{m}, P_{\text{el, pump}})}$$  \hspace{1cm} \text{Eq. 9}$$

The share of the pressure drop caused by the measurement equipment ($f_{\text{equip}}$) shall never exceed 15%. For $f_{\text{equip}} < 5\%$, no corrections have to be applied. For $f_{\text{equip}}$ between 5% and 15%, the measured electricity consumption will be reduced at all times the pump is in operation. The amount of electric power to be subtracted at each timestep is given by Eq. 5.

$$\Delta P_{\text{el, pump}} = -f_{\text{on}} \cdot f_{\text{equip}} \cdot P_{\text{el, pump}}$$  \hspace{1cm} \text{Eq. 10}$$

where $f_{\text{on}}$ is 1 for pump on and 0 for pump off.

For pumps with variable speed / mass flows, a suitable number of values for $f_{\text{equip}}$ and $P_{\text{el, pump}}$ or a mass flow dependent function for $f_{\text{equip}}$ and $P_{\text{el, pump}}$ shall be used.
Annex C  Pressure drop in the space heating circuits including thermostatic valves

C.1 Pressure drop in the borehole heat exchanger

The pressure drop estimation shown in chapter 4.2.1, Pressure drop for ground source emulation, is based on the following assumptions (comp. Table 7):

<table>
<thead>
<tr>
<th>borehole field</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>number of boreholes</td>
<td>[-]</td>
<td>1</td>
</tr>
<tr>
<td>number of pipes per borehole (double-U)</td>
<td>[-]</td>
<td>2</td>
</tr>
<tr>
<td>depth of borehole</td>
<td>[m]</td>
<td>123</td>
</tr>
<tr>
<td>inner diameter of pipes</td>
<td>[m]</td>
<td>0.026</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>antifreeze</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>density</td>
<td>[kg/m³]</td>
<td>1037</td>
</tr>
<tr>
<td>reference temperature</td>
<td>[°C]</td>
<td>0</td>
</tr>
<tr>
<td>kinematic viscosity</td>
<td>[mm²/sec]</td>
<td>2.105</td>
</tr>
</tbody>
</table>

The pressure drop approximation has been obtained by fitting the average pressure drop of curves at 0 °C, 15 °C, and -10 °C that were derived with Darcy-Weissbach-equations. It is assumed that the additional pressure drop of headers, connections and bends is 100% of the pure pipe length pressure drop.

The pressure drop of the pipe lengths only is calculated according to Eq. 12:

\[
E \text{q. 11} \quad dp_{\text{borehole}} = \frac{2 \cdot L_{\text{borehole}} \cdot \lambda \cdot \rho \cdot u^2}{2 \cdot D}
\]

where

- \( L_{\text{borehole}} \) m, borehole length
- \( \lambda \) -, friction factor
- \( \rho \) kg/m³, density of the fluid
- \( u \) m/s, velocity of the fluid
- \( D \) m, inner diameter of the pipe

C.2 Pressure drop in the space heating circuits including thermostatic valves

This is a suggested definition for an arbitrary heating system that should be emulated by the test rig. For different boundary conditions of climate and or building, different values would need to be derived.
According to Roos (2002), the pressure drop of hydraulic loops in heating systems can be approximated by

\[ \Delta p = C \cdot \dot{V}^m \]

C can be taken as constant in a first approximation, m is an exponent between 1.8 (pipes) and 2.0 (valves). The total pressure drop of a valve and piping is therefore:

\[ \Delta p = C_{\text{pipe}} \cdot \dot{V}_{\text{pipe}}^m + C_{\text{valve}} \cdot \dot{V}_{\text{valve}}^m \]

with \( m_{\text{pipe}} = 1.8 \) and \( m_{\text{valve}} = 2.0 \)

To simplify the procedures, the many thermostatic valves that may be present in a heating system are replaced with one thermostatic valve for the whole loop – just as the building model is also only a one zone model with only one temperature that influences the valve position.

With known nominal volume flow of 600 kg/h, and a pressure drop for the case of the open thermostatic valve position that corresponds to 50 mbar for the thermostatic valve and 250 mbar for the hydraulic loop, \( C_{\text{pipe}} \) and \( C_{\text{v,open}} \) are determined as:

\[ C_{\text{pipe}} = 6289 \text{ mbar} / (\text{kg} / \text{s})^{1.8} \]
\[ C_{\text{v,nom}} = 1800 \text{ mbar} / (\text{kg} / \text{s})^{2} \]

From \( C_{\text{v,open}} \), the kvs-value (volume flow at 1 bar pressure difference) for the valve at fully open position) is determined as 2683 kg/s.

According to the relationship between kv-values and the valve position given in Roos (2002), the relationship for \( C_{\text{valve}} \) depending on the valve position can be obtained:

\[ C_{\text{valve}} = \frac{C_{\text{v,open}}}{\exp\left(n \cdot \{\text{STW} - 1\}\right)^2} \]

with \( n = 3.22 \) and STW the valve position (0 = closed, 1 = open).

For reasons of simplicity, mass flow rate has been used instead of volume flow rate, since the difference does not influence the results significantly.

The resulting pressure drop curves for different thermostatic valve positions are shown in Figure 19:
Figure 19: System resistance curve of the emulated heat distribution system for various positions of the thermostatic valve (Vpos).
Annex D  Pre-defined heating load

The pre-defined load files for the system test are shown in Figure 20.

Figure 20: Daily energy targets for the building load of the 12-day test sequence and the 6-day test sequence.
Annex E  Evaluation of compactness and ease of installation

Laboratory testing of combisystems as a whole is justified if:

- The system to be tested will be produced and installed many times.
- The system to be tested will be identical to the system that will be installed.

This is the case for small units destined for SFH and produced as fixed sets with a high degree of prefabrication (compactness). For the evaluation of compactness and ease of installation, the criteria shown in Table 8 are proposed. An indicator may be calculated based on these values that shows the degree of compactness, ease of installation and fail-proofness. In addition to this, the test report may contain information about the need of installing additional heat traps or no-return valves on site in the case that these are not included in the pre-fabricated units yet.

Table 8: Criteria for the evaluation of compactness, ease of installation, and fail-proofness.

**How many single pieces are delivered to the site of installation?**

Counted are: Storage Tank(s), Auxiliary heating units, expansion vessels (both for solar and for space heat/storage), Mixing valves, tee-pieces

Not counted: solar loop line for connection of the installation in the technical room to the collectors on the roof. One piece of connector each for solar (flow and return), space heat (flow and return), DHW (cold and hot) and Auxiliary (flow and return). A vessel for storing anti-freeze liquid before it is filled in. A compact pre-fabricated unit where no interconnections of pieces (hydraulic or electric) are necessary on-site is counted as one piece.

**How many man-hours are needed for installation, filling and commissioning?**

Counted from the point where the storage (and other pieces delivered)

**How many connections can be made wrong?**

All connections having the same diameter and no clear labeling by color-code or imprinted code on the pipe / insulation can be made wrong. The color codes must be clear without consultation of a handbook (e.g. red for hot and blue for cold on both sides of the connection). Possibility of connecting flow and return wrongly counts as 1 (not as two)

**How many of the following documents are present and complete at the time of installation?**

1. Installation Manual, 2. User’s Manual, 3. Maintainance Protocol. If present but not deemed complete count the respective document only as half present. If present but not useful at all do not count

**Is there an electrician needed to connect the power supply and/or sensors?**

The answer is YES if there is any AC equipment that does not have a power line that is at least 5 m and an AC-connector connected to it that fits the standard AC plugs of the respective country. The answer is also YES if a screw driver is needed on-site to connect electric parts including sensors, with the exception of ONE sensor for the collector temperature (must be low voltage). The screw driver for opening casings does not count

**In how many of the attached loops is unwanted circulation safely prevented?**
1. Collector loop: The answer is NO if external application of 0.03 bar in either direction leads to a detectable flow in the collector loop while the collector loop is not in operation.

2. Auxiliary heating loop: The answer is NO if the auxiliary heater is connected externally and the application of 0.02 bar in either direction leads to a detectable flow in the auxiliary heater unit. The answer is YES if the auxiliary heater is integrated into the store / within the same pre-fabricated unit as the store (no measurement needed).

3. Space heating loop: The answer is NO if the heating season is off / auxiliary heater is off and the application of 0.02 bar in either direction leads to a detectable flow in the auxiliary heater unit.

<table>
<thead>
<tr>
<th>Question</th>
<th>Answer</th>
</tr>
</thead>
<tbody>
<tr>
<td>How many additional sensors have to be connected to the controller on-site?</td>
<td>Counted are all sensors with the exception of the temperature sensors for the collector (1), the ambient (1) and the room (1).</td>
</tr>
<tr>
<td>How many additional sensors have to be placed on-site?</td>
<td>Counted are all sensors with the exception of the temperature sensors for the collector (1), the ambient (1) and the room (1).</td>
</tr>
<tr>
<td>Do the deliverable units fit through a standard door?</td>
<td>Enter 100% if they fit through a doorlight of 0.79 m width without removing any insulation or other things attached. Enter 50% if they fit only after removing parts (or all) of the insulation or other things attached. Enter 0% if they do not fit at all.</td>
</tr>
</tbody>
</table>

*a) may not be applicable for some countries like e.g. France where an electrician is required by national regulations.*