

## Deliverable B1 - Report on system characterization in the laboratory

Date: 26.11.2015

---

By Diego Menegon<sup>1</sup>, Roberto Fedrizzi<sup>1</sup>

<sup>1</sup>Institution EURAC  
Address viale Druso 1, I-39100 Bolzano  
e-mail [diego.menegon@eurac.edu](mailto:diego.menegon@eurac.edu)  
e-mail [roberto.fedrizzi@eurac.edu](mailto:roberto.fedrizzi@eurac.edu)

## IEA Solar Heating and Cooling Program

The Solar Heating and Cooling Programme was founded in 1977 as one of the first multilateral technology initiatives ("Implementing Agreements") of the International Energy Agency. Its mission is *"to enhance collective knowledge and application of solar heating and cooling through international collaboration to reach the goal set in the vision of solar thermal energy meeting 50% of low temperature heating and cooling demand by 2050."*

The member countries of the Programme collaborate on projects (referred to as "Tasks") in the field of research, development, demonstration (RD&D), and test methods for solar thermal energy and solar buildings.

A total of 53 such projects have been initiated to-date, 39 of which have been completed. Research topics include:

- ▲ Solar Space Heating and Water Heating (Tasks 14, 19, 26, 44)
- ▲ Solar Cooling (Tasks 25, 38, 48, 53)
- ▲ Solar Heat for Industrial or Agricultural Processes (Tasks 29, 33, 49)
- ▲ Solar District Heating (Tasks 7, 45)
- ▲ Solar Buildings/Architecture/Urban Planning (Tasks 8, 11, 12, 13, 20, 22, 23, 28, 37, 40, 41, 47, 51, 52)
- ▲ Solar Thermal & PV (Tasks 16, 35)
- ▲ Daylighting/Lighting (Tasks 21, 31, 50)
- ▲ Materials/Components for Solar Heating and Cooling (Tasks 2, 3, 6, 10, 18, 27, 39)
- ▲ Standards, Certification, and Test Methods (Tasks 14, 24, 34, 43)
- ▲ Resource Assessment (Tasks 1, 4, 5, 9, 17, 36, 46)
- ▲ Storage of Solar Heat (Tasks 7, 32, 42)

In addition to the project work, there are special activities:

- SHC International Conference on Solar Heating and Cooling for Buildings and Industry
- Solar Heat Worldwide – annual statistics publication
- Memorandum of Understanding with solar thermal trade organizations
- Workshops and conferences

### Country Members

Australia	Germany	Singapore
Austria	Finland	South Africa
Belgium	France	Spain
China	Italy	Sweden
Canada	Mexico	Switzerland
Denmark	Netherlands	Turkey
European Commission	Norway	United Kingdom
	Portugal	United States

### Sponsor Members

European Copper Institute	Gulf Organization for Research and Development
ECREEE	RCREEE

## Current Tasks & Working Group:

Task 42	<i>Compact Thermal Energy Storage</i>
Task 43	<i>Solar Rating and Certification Procedures</i>
Task 45	<i>Large Systems: Solar Heating/Cooling Systems, Seasonal Storages, Heat Pumps</i>
Task 46	<i>Solar Resource Assessment and Forecasting</i>
Task 47	<i>Renovation of Non-Residential Buildings Towards Sustainable Standards</i>
Task 48	<i>Quality Assurance and Support Measures for Solar Cooling</i>
Task 49	<i>Solar Process Heat for Production and Advanced Applications</i>
Task 50	<i>Advanced Lighting Solutions for Retrofitting Buildings</i>
Task 51	<i>Solar Energy in Urban Planning</i>
Task 52	<i>Solar Energy and Energy Economics in Urban Environments</i>
Task 53	<i>New Generation Solar Cooling and Heating (PV or Solar Thermally Driven Systems)</i>
Task 54	<i>Price Reduction of Solar Thermal Systems</i>

## Completed Tasks:

Task 1	<i>Investigation of the Performance of Solar Heating and Cooling Systems</i>
Task 2	<i>Coordination of Solar Heating and Cooling R&amp;D</i>
Task 3	<i>Performance Testing of Solar Collectors</i>
Task 4	<i>Development of an Insolation Handbook and Instrument Package</i>
Task 5	<i>Use of Existing Meteorological Information for Solar Energy Application</i>
Task 6	<i>Performance of Solar Systems Using Evacuated Collectors</i>
Task 7	<i>Central Solar Heating Plants with Seasonal Storage</i>
Task 8	<i>Passive and Hybrid Solar Low Energy Buildings</i>
Task 9	<i>Solar Radiation and Pyranometry Studies</i>
Task 10	<i>Solar Materials R&amp;D</i>
Task 11	<i>Passive and Hybrid Solar Commercial Buildings</i>
Task 12	<i>Building Energy Analysis and Design Tools for Solar Applications</i>
Task 13	<i>Advanced Solar Low Energy Buildings</i>
Task 14	<i>Advanced Active Solar Energy Systems</i>
Task 16	<i>Photovoltaics in Buildings</i>
Task 17	<i>Measuring and Modeling Spectral Radiation</i>
Task 18	<i>Advanced Glazing and Associated Materials for Solar and Building Applications</i>
Task 19	<i>Solar Air Systems</i>
Task 20	<i>Solar Energy in Building Renovation</i>
Task 21	<i>Daylight in Buildings</i>
Task 22	<i>Building Energy Analysis Tools</i>
Task 23	<i>Optimization of Solar Energy Use in Large Buildings</i>
Task 24	<i>Solar Procurement</i>
Task 25	<i>Solar Assisted Air Conditioning of Buildings</i>
Task 26	<i>Solar Combisystems</i>
Task 27	<i>Performance of Solar Facade Components</i>
Task 28	<i>Solar Sustainable Housing</i>
Task 29	<i>Solar Crop Drying</i>
Task 31	<i>Daylighting Buildings in the 21st Century</i>
Task 32	<i>Advanced Storage Concepts for Solar and Low Energy Buildings</i>
Task 33	<i>Solar Heat for Industrial Processes</i>
Task 34	<i>Testing and Validation of Building Energy Simulation Tools</i>
Task 35	<i>PV/Thermal Solar Systems</i>
Task 36	<i>Solar Resource Knowledge Management</i>
Task 37	<i>Advanced Housing Renovation with Solar &amp; Conservation</i>
Task 38	<i>Solar Thermal Cooling and Air Conditioning</i>
Task 39	<i>Polymeric Materials for Solar Thermal Applications</i>
Task 40	<i>Towards Net Zero Energy Solar Buildings</i>
Task 41	<i>Solar Energy and Architecture</i>
Task 44	<i>Solar and Heat Pump Systems</i>

## Completed Working Groups:

*CSHPSS; ISOLDE; Materials in Solar Thermal Collectors; Evaluation of Task 13 Houses; Daylight Research*

## Contents

Nomenclature, Acronyms and Abbreviation .....	5
1 Introduction .....	6
2 Solar Heating and Cooling Systems' Evaluation .....	7
2.1 Performance definition .....	7
3 Mass, Heat and Electricity Fluxes Assessment .....	9
3.1 Review of Available Standards in the Heat Pumps Sector .....	9
3.2 Review of Available Standards in the Solar Thermal Sector .....	14
3.3 Critical Analysis of the Available Standards .....	15
3.4 Review of Available System Test Procedures not Included in Standards .....	16
4 New whole system dynamic procedure .....	19
4.1 Climate selection .....	19
4.2 Building selection .....	20
4.3 Definition of load file .....	20
4.4 Boundary conditions selection .....	20
4.5 Boundary conditions emulation in the laboratory .....	22
4.6 Test execution and data analysis .....	27
5 Case studies analysis .....	29
5.1 Bolzano sequence – 6 days .....	29
5.2 Zurich sequence – 6 days .....	39
6 Conclusions .....	43
Bibliography .....	44

## Nomenclature, Acronyms and Abbreviation

<b>Nomenclature</b>		
$T$	Temperature	[°C]
$\Delta T$	temperature difference	[K]
$\dot{m}$	mass flow	[kg/s]
$c_p$	Specific heat	[J/kgK]
$E$	Thermal Energy	[kWh]
$\dot{Q}$	Thermal Power	[kW]
COP	Coefficient of performance	[-]
EER	Energy efficiency ratio	[-]
SCOP	Seasonal COP	[-]
SEER	Seasonal EER	[-]
SPF	Seasonal performance factor	[-]
$\eta_{col}$	Solar collector efficiency	[-]
$f_{sav}$	Fractional energy savings	[-]
$\alpha$	Thermal diffusivity	[m <sup>2</sup> /s]
$k$	Thermal conductivity	[W/m <sup>2</sup> K]
$\mu$	Dynamic viscosity	[Kg/m s]
$\nu$	Kinematic viscosity	[m <sup>2</sup> /s]
$c_p$	Specific thermal capacity at constant pressure	[J/Kg K]
<b>Subscripts</b>		
cond	condenser	
evap	evaporator	
in	inlet	
out	Outlet	
<b>Abbreviation</b>		
DHW	Domestic hot water	
SH	Space heating	
SC	Space cooling	

# 1 Introduction

The performance of a heating and cooling system is strongly affected by the way the single components are integrated together and by the boundary conditions, which the system is subject to.

This is mostly true for systems driven by a number of different energy sources and setup with a complex control strategy. In these cases, the dynamics of the system have to be taken into account, in order to perform a reliable system characterization.

The implementation of a dynamic laboratory tests procedure allows to evaluate the performance considering those aspects. Different procedures are currently under development by different research institutes [1] but their implementation is still debatable.

The aim of a dynamic test procedure is to assess the system performance when operating under real-like boundary conditions. To develop a suitable procedure, some requirements are defined in order to reliably evaluate the system performance:

- The test should represent the behaviour of the system in a real installation.
- The result should represent the seasonal performance.
- The result must be accurate and reliable.
- The test must be easy to perform and cost effective to be attractive for industry.
- The procedure should be reproducible for different systems, climates and loads.

In this document, different dynamic test approaches are reviewed versus standardised stationary test methods. In addition, a test procedure newly developed at EURAC is presented and compared to the other.

The latter results are reported with respect to a solar assisted heat pump system, while to-date the test has not been proved on a solar cooling system. Nonetheless, the degree of complication of the heating and cooling system presented here is comparable with a solar cooling one. Moreover the test procedure has been assessed on a single adsorption chiller operating in a real solar cooling plant, showing positive results system [2,3]. This suggests that the procedure could be wholly extended to solar heating and cooling systems.

## 2 Solar Heating and Cooling Systems' Evaluation

The survey of the available standards in the fields of heat pumps and solar heating systems rating tests carried out in the IEA HPP Annex 34 and IEA SHC Task44/ HPP Annex38 ended in a number of suitable procedures.

With regards to those related to HPs testing however, most refer to electrically driven ones and do not take into consideration their possible use within complex hybrid systems. Since many aspects regarding application, operating conditions and measurement procedures are similar or equal to those of the thermally driven heat pumps, they are anyhow discussed here.

On the other hand, most of the standards referring to solar thermal systems either tackle single components testing again (solar collectors, storages) or do not contemplate the connection of the solar heating system with a heat pump (nor compression or sorption).

The survey covered ISO, CEN, ANSI/ASHRAE, AHRI, JRAIA/JSA, JRA, VDI and DIN documents regarding testing and evaluation methods. As the market is continuously changing, so are the standards: some are under revision at the moment, therefore the following description might result in lacking or even erroneous/outdated information.

### 2.1 Performance definition

In order to express the effectiveness of a heat pump or a system using a heat pump, three levels of performance figures have been defined:

- COP and EER as the unit effectiveness at nominal rating conditions under steady-state operation
- SCOP and SEER for the assessment of the unit performance under defined, time dependent rating conditions over a certain period of time
- SPF for the assessment of the system performance under defined, time dependent rating conditions over a certain period of time.

Since here systems are tackled, the last performance figure will be mainly considered and standards/test procedures for its elaboration will be addressed. However, interesting pieces of information can be also derived from standards regulating the evaluation of the SCOP and SEER for single heat pumps; therefore, also some of those will be reported here.

The **Seasonal Performance Factor** can be generally defined as the ratio of useful energy output to energy input with respect to a given system boundary  $i$ :

$$SPF_i = \frac{\sum Q_{i,out}}{\sum E_{i,in}} \quad (1)$$

However, especially in hybrid systems, different types of end energies are used for system operation: besides the electrical energy, gas, oil, biomass or heat from the district heating network or waste heat from an industrial process might be used. As different types of energies have in general different specific exergy content and also different economic values and environmental impact, they should be evaluated separately. Practically, for a system with both thermal and electric energy inputs, a thermal and an electrical SPF have to be provided separately:

$$SPF_{i,th} = \frac{\sum Q_{i,out}}{\sum Q_{i,in}} \quad (2)$$

$$SPF_{i,el} = \frac{\sum Q_{i,out}}{\sum W_{el,i,in}} \quad (3)$$

Although the SPF provides in most cases a good figure to estimate the quality of the system under given operating conditions, the **Primary Energy Ratio (PER)** gives more in-depth information under the economic or environmental point of view. It is defined as the ratio of the useful energy output to the primary energy input to the system boundary. To be able to calculate it, certain primary energy factors for every type of energy input have to be provided and agreed upon.

$$PER_i = \frac{\sum Q_{i,out}}{\sum \left( \frac{W_{el,i,in}}{\varepsilon_{el}} + \frac{Q_{i,in}}{\varepsilon_{in}} \right)} \quad (4)$$

Depending on the aim of the calculation, the primary energy can be defined as “overall” (e.g. for the analysis of the economic aspects) or “non-renewable only” (e.g. to estimate net emissions). The primary energy factors  $\varepsilon_i$  depend on the location of the system, time of the year and on local policies. However, some generalized values are given in the national Annexes of the EN 15316 or in EN 15603:2008. If substituted with emission factors (e.g. expressed in kgCO<sub>2,eq</sub> per kWh energy) or energy price (e.g. expressed in monetary unit per kWh energy), the equivalent CO<sub>2</sub> emissions or the energy costs of the system over the considered period of time can be obtained, respectively.

Beside these two key figures, a number of other performance indicators might be of interest in solar heating and cooling systems, depending on the target group: renewable energy ratio, solar fraction, fractional energy saving, global warming potential, etc. These are extensively discussed in report delivered as the end result of the activity B7 (“Collection of criteria to quantify the quality and cost competitiveness for solar cooling systems”). Therefore, they are no further discussed here.

Despite the specific technical figure selected to evaluate the system/subsystem performance, three main topics have to be addressed when the characterization is pursued:

1. System (or subsystem or component) **boundaries** have to be **defined**
2. Mass, heat and electricity fluxes crossing the boundaries with incoming and outgoing direction have to be identified
3. Mentioned mass, heat and electricity fluxes have to be measured, with respect to meaningful operation conditions and timeframes.

With regards to the first two issues, much work has been performed within the IEA SHC Task38 and further proposals are reported in the next chapters with the purpose of rationalizing the nomenclature and system boundaries definition/illustration.

On the contrary, a quite confused condition can be at present noticed with respect to the third point: a standard is far from being elaborated when complex hybrid systems are to be evaluated; on the hand, even though the scientific community is widely engaged in defining laboratory test procedures for the estimation of systems’ performance, visions on methods to be used and research directions largely diverge.



### 3 Mass, Heat and Electricity Fluxes Assessment

Mass, heat and electricity fluxes crossing the defined boundaries have to be measured, with respect to **meaningful system working conditions and timeframes**.

In the following paragraphs, suitable standards and guidelines, and state of the art test procedures are reported and discussed.

#### 3.1 Review of Available Standards in the Heat Pumps Sector

Table 1 gives an overview of the standards assessed within the works of the IEA HPP Annex34. The review on the suitable standers in the heat pumps and solar thermal collectors sectors is based on the comprehensive analysis performed by Malenkovic in the framework of IEA SHC Task44/ HPP Annex38.

*Table 1 - Overview of suitable standards and guidelines from IEA HPP Annex34*

Standard for the assessment of the SCOP/SEER		
AHRI 560	Absorption water chilling and water heating packages	2000
ANSI/ASHRAEE 182	Method of testing absorption water chilling and water heating packages	2008
EN 16147	HP with electrically driven compressor – testing and requirements for marking for DHW units	2011
EN 14825	Air conditioners, liquid chilling packages and HP with electrically driven compressor for SH and SC- Testing and rating at part load conditions and calculation of seasonal performance	2012-UR
EN 12309	Gas-fired absorption and adsorption air conditioning and/or HP appliance with a net heat input not exceeding 70kW – Part 2: Rational use of energy	2012-UR
Standard for the assessment of the SPF		
VDI 4650-1	Calculation of HP- Simplified method for the calculation of the SPF of HP- electric HP for SH and SHW	2009
VDI 4650-2	Simplified method for the calculation of the annual coefficient of performance and the annual utilization ration of sorption heat pumps- gas HP for SH and DHW	UD
EN 15314-4-2	Heating systems in buildings- Method for calculation of system energy requirements and system efficiencies – Part 4-2 SH generation systems, HP systems.	2008-UR

### 3.1.1 EN14825: Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling – Testing and rating at part load conditions and calculation of seasonal performance

The aim of the standard is to give a basis for the comparison of heat pumps, chilling packages and air conditioners on the basis of the Seasonal Energy Efficiency Ratio (SEER) for cooling and Seasonal Coefficient of Performance (SCOP) for heating applications. It provides a description of the calculation method and the part load conditions for three different climates: one average climate, one cold and one warm climate.

The standard covers air-to-air, water (brine)-to-air, air-to-water and water (brine)-to-water units. For the detailed rating conditions and test methods EN 14511 Parts 2 and 3 are used. The calculation is based on the temperature bin method. SEER and SCOP are calculated according to:

$$SEER = \frac{Q_{CE}}{\frac{Q_{CE}}{SEER_{on}} + H_{TO} \cdot P_{TO} + H_{SB} \cdot P_{SB} + H_{CK} \cdot P_{CK} + H_{OFF} \cdot P_{OFF}} \quad (5)$$

$$SCOP = \frac{Q_{HE}}{\frac{Q_{HE}}{SCOP_{on}} + H_{TO} \cdot P_{TO} + H_{SB} \cdot P_{SB} + H_{CK} \cdot P_{CK} + H_{OFF} \cdot P_{OFF}} \quad (6)$$

Where:

- $SEER_{on}$  is the seasonal efficiency of the unit in active cooling mode;
- $SCOP_{on}$  is the seasonal efficiency of the unit in active heating mode;
- $Q_{CE}$  is the reference annual cooling demand;
- $Q_{HE}$  is the reference annual heating demand;
- $H_i$  is the number of hours the unit is considered to work in the modes indicated by the indices;
- $P_i$  is the electricity consumption during the modes indicated by the indices.

Indices:

- $TO$  thermostat off mode;
- $SB$  standby mode;
- $CK$  crankcase heater mode;
- $OFF$  off mode.

The reference annual heating and cooling demands are obtained from the respective design load multiplied with the equivalent heating or cooling periods in hours. For the calculation of the  $SEER_{on}$  and  $SCOP_{on}$ , a unit has to be tested for a certain number of stated part load conditions.

The power consumption is measured by setting the thermostat to a value which triggers shutting down of the compressor. The auxiliary power consumption is also measured. For the measurement of the power consumption in the standby mode, the unit is stopped by the control device and the power measured. For the crankcase heater power consumption it is only stated, that the measurement has to last for 8 hours. Finally, the off mode test has to take place after the standby test by switching the unit into the off mode while remaining plugged.

### 3.1.2 EN16147: Heat pumps with electrically driven compressors - Testing and requirements for marking for domestic hot water units

It specifies methods for testing and rating of heat pumps connected to or including a domestic hot water storage tank. The testing procedure includes the following tests:

- Heating up period
- Determination of standby power input
- Energy consumption and COP for reference tapping cycles
- Determination of a reference hot water temperature and the maximum quantity of usable hot water in a single tapping
- Temperature operating range test.
- Safety tests.

The efficiency figure defined by the standard,  $COP_{DHW}$ , is determined for different, non-stationary operating conditions and thus does not correspond to the definitions of the COP given in other standards. Furthermore, the system boundary includes the hot water storage, thus the storage losses are also included in the energy balance.

### 3.1.3 ARI 560: Standard for Absorption Water Chilling and Water Heating Packages

This standard applies to water/LiB (both steam- and hot water-fired), indirectly-fired double effect chillers (both steam- and hot water-fired) and directly-fired double effect chiller/heaters. It does not apply to machines that are air-cooled, exhaust gas-fired or used only for heating.

The test procedure provides a definition with tolerances for steady state operation. Once the machine is in steady state, three sets of data are taken at a minimum of 5 minute intervals. The data taken is enough to establish the cooling produced, the driving energy provided to the machine, and in addition an energy balance to verify the accuracy of measurements.

There are three sets of conditions for which data is to be taken: (1) full load “standard rating conditions,” (2) full load “application rating conditions” and (3) part load conditions. The standard conditions are precisely defined, while the application conditions allow the manufacturer to choose from within a range of values for temperatures and flow rates.

Part load is to be evaluated at 100%, 75%, 50%, and 25% of the full load capacity. The results can be represented in any one or more of the following three ways:

- At standard rating conditions, a weighted average (representing typical building loads) is calculated from the four part load fractions to obtain the integrated part load value (IPLV)
- At application rating conditions, the same weighted average can be calculated to obtain the non-standard part load value (NPLV).
- In the event that a unit cannot operate down to 25% part load, a cyclic degradation factor is applied to represent the part loads below the limit of the machine.

### **3.1.4 ANSI/ASHRAE 182-2008: Method of Testing Absorption Water-Chilling and Water-Heating Packages**

This standard prescribes a method of testing absorption water-chilling and water-heating packages to verify capacity and thermal energy input requirements at a specific set of operating conditions. This standard applies to absorption packages used to chill and/or heat water and testing that will occur where proper instrumentation and load stability can be provided. This standard is not intended for testing typical field installations. The ANSI/ASHRAE 182 standard is a method of test (MOT) standard meant to be used in conjunction with a rating procedure such as ARI 560.

The standard applies only to water-cooled units. It applies to chillers using water/LiBr, ammonia/water, and other working fluids, both single- and double-effect. The chiller can be direct-fired by natural gas, LP gas, oil, or other fuel; or it can be indirectly fired by steam, hot water, a hot gas stream, or other hot fluids. It covers three modes of operation: cooling-only, heating-only, and combined cooling and heating.

Test data is to be taken at steady state conditions. Tolerances are defined for establishing steady state. Once steady state is established, three sets of data are recorded at a minimum of 5-minute intervals, and within a maximum 45 minute period.

### **3.1.5 EN 12309-2: Gas-fired absorption and adsorption air-conditioning and/or heat pump appliances with a net heat input not exceeding 70 kW. Rational use of energy**

It defines test methods for the determination of the Gas Utilization Efficiency (GUE) of gas driven adsorption or absorption heat pumps in heating and cooling mode. This performance figure is assessed at the full capacity and at steady state conditions. Therefore, energy consumption of auxiliaries and the degradation effect due to part load operation are not taking into accounts.

### **3.1.6 VDI 4650-1: Calculation of heat pumps - Simplified method for the calculation of the seasonal performance factor of heat pumps - Electric heat pumps for space heating and domestic hot water**

The VDI 4650-1 “describes an easy, yet sufficiently exact, method for the calculation of the energy efficiency, which takes into account all influence quantities of technical relevance”. The currently applicable version (March 2009) expresses the efficiency of the heat pump in terms of the seasonal performance factor, not as annual effort figure as previous versions. The guideline applies to electrically driven heat pumps for heating and/or domestic hot water (DHW) production up to 100 kW heating capacity. Heat sources covered by the guideline are ground water, ground (both boreholes and horizontal ground heat exchangers) and air. Only water-based central heating system is considered on the heat sink side.

The performance of the heat pump is calculated for heating and DHW applications separately and weighted according to the respective contribution to the annual energy demand. Both the seasonal performance factors are calculated starting from the rated COP values assessed according to EN 14511; the COPs are then manipulated by means of a number of correction factors, accounting for different heat rejection typical temperatures, operating condition, etc.

### 3.1.7 VDI 4650-2: Simplified method for the calculation of the annual coefficient of performance and the annual utilization ratio of sorption heat pumps - Gas heat pumps for space heating and domestic hot water

The scope of VDI 4650-2 is to define a method to estimate seasonal performance figures of a gas fired thermally driven heat pump based on measurements under part load laboratory conditions. Currently, VDI 4650-2 has the status of a pre-norm. It is defined for monovalent gas fired sorption heat pumps up to a heating power of 70kW. As ambient heat sources ground water, boreholes, air and solar radiation gained by a solar collector are considered. The heat is used for domestic hot water preparation and space heating.

Basically, two seasonal performance figures are defined. The annual use efficiency  $\eta_N$  is defined as the produced heat per consumed fuel. The annual heating figure  $\zeta$ , however, is defined as the amount of produced heat per amount of consumed fuel and electricity. Fuel and electricity are weighted equally, thus they are added without correction.

The calculation of the seasonal use efficiency and the annual heating figure is based on the temperature bin method. This means that the use efficiencies and heating figures are calculated from the measured performance in laboratory for several part load conditions. The average of these values is taken as a seasonal value. Based on DIN 4702-8, the part loads are 13%, 30%, 39%, 48% and 63% of full load heating power.

The assumption is that in part loads the volume flows are kept constant, thus part load is defined as a reduction of the heating loop inlet and outlet temperature.

Both gas and electricity consumption should be measured during the tests. The liquid pump for the heating distribution system is considered only through the pressure loss through the heat pump unit (EN 14511).

### 3.1.8 EN 15316-4-2: Heating systems in buildings – Method for calculation of system energy requirements and system efficiencies – Part 4.2: Space heating generations systems, heat pump systems

The scope of the standard covers both heating and DHW heat pumps, in alternate or simultaneous operation. The heat pumps can be driven electrically, with a combustion engine or thermally (absorption only).

The EN 15316-4-2 describes two different methods for the calculation of the SPF, which differ regarding the needed input data, the considered operating conditions and the calculation periods:

- Simplified method based on the system typology, which delivers the SPF for the heating season. The input parameters are taken from the tables and do not take into consideration the specific configuration of the system.
- Calculation based on the temperature bin method, which is explained in the standard itself.

The cumulative heating degree hours should be given in a national annex or available from national standards. The energy demand for each bin is calculated using a weighting factor calculation based on the heating degree hours for every bin. The domestic hot water demand is also calculated using weighting factors, similar to the heating energy demand.

The heating capacity and the COP for the nominal capacity should be determined according to a European standard. Also, in order to cover the whole range of heat source and heat sink temperatures, the COP values should be interpolated or extrapolated from the measured values. If the COP for only one operating condition is available, a correction for both heat source and heat sink based on the constant exergy efficiency can be performed and is described in an informative Annex.

For the DHW, results from the measurements according to EN 255-3 are to be used. Because of oscillating source temperatures, a correction has to be performed on the basis of constant exergy efficiency, same as for the heating operation mode. If no data from the tests are available, an average DHW charge temperature can be calculated.

Finally, the overall COP is interpolated from the test data for the heating and DHW operation modes. For engine driven heat pumps and absorption heat pumps no reference to applicable test methods is given. It is however stated, that the same corrections regarding operating conditions apply.

**Regarding part load operation the standard states, that the losses due to the on-off operation are negligible.** They are not considered in the calculation, except if considered in the tests which yielded the input data. For the off mode, only the auxiliary energy consumption is regarded. If part load data are available from other standards, e.g. EN 14825, the COP for each operating condition (every bin) should be interpolated and a load factor is to be calculated. For DHW operation, the start-up losses are already considered in the EN 255-3. For engine driven and absorption heat pumps, the start-up losses have to be considered in the test standards.

## 3.2 Review of Available Standards in the Solar Thermal Sector

### 3.2.1 EN 12977

It describes the procedure to assess the performance of “custom built” systems through the Component Test System Simulation (CTSS) method. According to it, some parameters are determined through tests carried out for each single component. The performance of the whole system is predicted using a simulation program (TRNSYS).

### 3.2.2 ISO 9459

ISO 9459-2: through the Complete System Testing Group (CSTG), it is applicable to solar system without auxiliary heating. This test method uses a series of one-day outdoor tests and a "black box" procedure that produces a family of "input-output" correlation equations. The system characterization is obtained by the determination of:

- Input-Output diagram;
- Draw-off temperature profile;
- Tank overnight heat losses coefficient.

This information is needed in order to obtain Long Term Performance Prediction (LTPP) of the system for one load pattern.

ISO 9459-5: through the Dynamic System test (DST) some parameters are assessed and are used to predict the annual system performance. This latter passage is obtained with a specific computer program that uses hourly values of local solar irradiation, ambient air temperature and cold-water temperature.

### 3.2.3 EN 12975

It allows to assess the collector performance in steady-state or quasi-dynamic conditions. It is not applicable for tracking concentrating collectors or when the storage unit is integrated with the collector.

The testing conditions are different compared to the previously standard. The main features that are assessed are collector output power and collector instantaneous efficiency.

### 3.2.4 EN 12976

It is applied to describe the reliability and performance tests for “factory made” systems. Reliability test consists into verifying the resistance of these systems to mechanical loads, thermal shocks, freezing, etc. For what concerns the performance assessment the two procedure of ISO 9459-2 and ISO 9459-5 can be applied.

### 3.3 Critical Analysis of the Available Standards

As can be seen in the short review reported above, there are a number of lacks in the available standards for testing components/systems performance:

- A consistent definition of the performance figures is missing (e.g. in some cases the electricity consumption include the proportional consumption of auxiliary devices, in others it doesn't); thermal energy consumption has the same weight as electricity consumption;
- There is lack of consistent nomenclature regarding same performance figures;
- Not all the technologies and/or applications are covered in all standards;
- With exception to the EN 12309, the European standards about TDHP are related only to direct fired machines;
- Standard for TDHP don't have separate figures for thermal and electrical efficiencies. In some cases the electricity consumption is added to the thermal energy consumption in other cases is not done;
- A clear definition of test conditions and procedures for discontinuous machines is not available;
- Tests of large systems are not accounted for;
- Most of times, tests are carried out under stationary conditions. Even when part-loads and ,in general, different working modes are considered, transitory behaviour, therefore inertial effects, is never accounted for;
- European testing procedures do not provide any accurate information on testing under part load conditions and quite different methods are promoted. The selection of part loads percentages and weight/duration is fully questionable: clear indications how the selection was made are not given most of the time.
- The control of the component/system is never taken into consideration
- Long term performance-decay is never considered
- A method for the calculation of the SPF for an entire complex hybrid systems (e.g. solar cooling systems including storages and backup) has not been defined yet.

**In general then, it might be concluded that 1) the available standards lack of consistent and agreed performance figures, 2) laboratory tests are quite complicated and test conditions are somehow questionable, 3) inertial effects, control and long time performance are never considered.**

### 3.4 Review of Available System Test Procedures not Included in Standards

As already stated, moving from these considerations, the scientific community is trying to “bridge the gap” in the field of systems testing. Different approaches are here reported and briefly discussed. The main methodologies at present available are:

- **Bin** method
- **CTSS** method (Component Testing/System Simulation)
- **ACDC** method (Annual Calculation Direct Comparison)
- **SCSPT** method (Short Cycle System Performance Testing)
- **CCT** method (Concise Cycle Testing)

#### 3.4.1 Bin Method

The Bin Method is based on the EN 14825 and EN 12309: at present it is well suited for testing single components with boundary conditions that can be decided depending on the location. The component performance is obtained by stationary testing at full and partial load. The integration of the single stationary performance over a range of different boundary conditions, which establish the operation range (for the specific location) gives the seasonal behaviour. It is an easy though accurate method when single components are considered. The use with respect to hybrid systems is far more complicated since control and dynamic effects cannot be considered; therefore the interactions among the system’s components cannot be acknowledged. Correction/Interaction factors should be defined, which identification might result in a tricky task.

#### 3.4.2 CTSS Method

A step forward towards considering the integration of components into a system is the Component Testing/System Simulation method. It is based on the TS 12977 and relies on the validation of numerical models, through data acquired during stationary tests of single components. A wide span of components’ working conditions is inspected during tests. The seasonal performance is obtained through numerical simulation of the entire system.

The range of application of the CTSS method is very flexible due to its component oriented testing approach. The system control is fully considered; however, losses and inertial effects are hardly accounted for, since most of the numerical models available (e.g. for solar collectors, pipes and heat exchangers) are stationary ones. Un-stationary tests should be carried out to develop full dynamic models.

#### 3.4.3 ACDC Method

The ACDC test method is on the ISO 9459 and has been developed by SERC for a combi store test and is used for predicting a long term performance after a short term measurement. Within this test procedure, the system is completely set up and operated for a specific number of days, according to a predefined test sequence; the first days are used to achieve a thermal conditioning inside the system. The central days simulate “typical” days representative of winter, spring, summer and fall working conditions. During the test, all the heat fluxes are recorded. The data are used in a “parameter identification” procedure applied to the numerical models used for the simulation of the entire system. The calculation is then used for predicting the seasonal performance.



Here, both control and dynamic behaviour are taken into consideration during the test. Still the test sequence (temperatures and mass flows) is defined by following questionable criteria. Moreover, the method reaches its limits when different climate conditions and demands for space heating and domestic hot water have to be investigated. The test results are only valid for weather conditions that correspond to the test weather conditions and similar heat loads.

The first issue (test sequence) should undergo a systematization procedure to agree on standardized series; the latter could be easily solved by simply defining standardized locations and loads (sized on the basis of the dimension of the hybrid system).

The main disadvantage of this procedure (as much as for the CTSS method) lies in the long time need for accomplishing the parameter identification phase with regards to all system's components.

### 3.4.4 SCSPT and CCT methods

SCSPT (originally developed by CEA-INES) and CCT (originally developed by SPF) methods [1] rely on the characterization of the system operating as a whole under “quasi-real” boundary conditions, reproduced in a laboratory. The boundary conditions are reproduced on-line connecting thermo-regulators in the laboratory to a dynamic simulation tool. Some selected/representative days are emulated in the laboratory.

With regard to these methods, only “small” systems can be analysed in a laboratory. However, monitoring of installed systems could be regarded as a special SCSPT/CCT method, carried out under real boundary conditions.

These methods look like the most “advanced” ones, since the entire system is tested considering dynamic effects, losses and control. Due to the high capacities of the storages used in the SHC plant (compared to a Solar Combi system for example), the inertial effect of the system becomes an important parameter, and it should be investigated.

Still some questions remain open like the selection of the representative days and the necessity of simulation tools: in this case however, the simulation activity is certainly limited to the assessment of the boundary conditions (building loads and weather conditions).

The main differences between these two methods are given in Table 2Table 8:

*Table 2 – Differences between the SCSPT and the CCT methods*

	SCSPT	CCT
Weather conditions	A specific procedure has been developed to produce the twelve days of the test sequence. This procedure can be applied to various climates. Up to now, Zurich, Stockholm and Barcelona are available.	The test sequence is only available for Zurich climate.
Seasonal performance calculation	The annual calculation is made by a simple extrapolation of the test results. A procedure based on neural network is under development in order to identify a simple dynamic model which could be used for every climate and load needed.	Parameters of a detailed model are identified based on the test results. This model is then used to predict annual performance for the selected climate and load, but also for every climate and load needed.

Most of the comments reported in the previous chapter hold to this procedure too. The main advantage of the SCSPT method with respect to the previous is due to lack of the parameter identification phase. In this case however, the selection of the representative days and the extrapolation of the seasonal performance out of a test based on them might be critical.

## 4 New whole system dynamic procedure

Figure 1 shows the configuration of the procedure and can be described with the following phases:

1. Selection of the climate (paragraph 4.1).
2. Selection of the building (paragraph 4.2).
3. Definition of the load file with a simulation of the building coupled to the climate (paragraph 4.3).
4. Selection of the boundary conditions (paragraph 4.4).
5. Emulation of the boundary conditions in the laboratory (paragraph 4.5).
6. Execution of the test and analysis of the data (Paragraph 4.6).



Figure 1 Test procedure.

### 4.1 Climate selection

A weather data file is needed to define the boundary conditions. A test reference year (TRY) is obtained with Meteonorm defining a weather data file with a one minute resolution (extrapolation from the hourly weather data with Type 109). This high resolution weather data is significant for the emulation of the components with low thermal capacity (e.g. dry cooler) and to achieve realistic transient variations of the boundary conditions.

The procedure is applicable to different climates. Figure 2 shows the annual average temperature all over the world. The selection of standard climates is important for the comparison between different systems. In fact, two tests made with reference to different climate zones cannot be compared. As an example the EN 14825 foresees 3 regions for the heating and 1 for cooling.

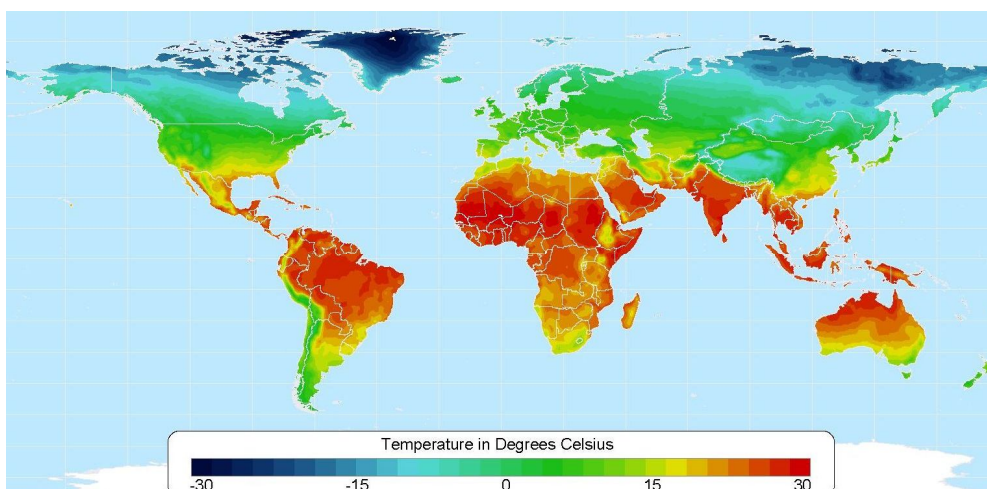


Figure 2 Average annual temperature of the world regions.

The procedure presented has been used to study the same system into two climates. The first one is Bolzano (Italy) because the climate is characterized by hot summers and very cold winters and thus there are high cooling and heating loads. The second is Zurich since it has been used as a reference in the other methodologies reported. The average annual temperature of Bolzano is 12°C while for Zurich is 9 °C.

## 4.2 Building selection

By coupling the climate to a specific building one obtains the load which has to be covered by the heating and cooling system.

At now, the building selected for the explanation of the procedure is a single family house with 2 floors. The internal area is 180 m<sup>2</sup>, the external wall components are bricks, plaster, and 10 cm of EPS for insulation and the windows are double layer with internal air interspace (the transmittance of external wall is about 0.27 W/m<sup>2</sup>K).

With the climate of Bolzano, the heating demand is 54 kWh/m<sup>2</sup>y and the cooling demand of 20 kWh/m<sup>2</sup>y while with the climate of Zurich the space heating load is 72.2 kWh/m<sup>2</sup>y and the space cooling load is low (2.2 kWh/m<sup>2</sup>y). The DHW demand is calculated around 20 kWh/m<sup>2</sup>y

## 4.3 Definition of load file

This building is modelled with TrnBuild and simulated with Type 56 included into the TRNSYS libraries, providing heating and cooling load profiles.

The domestic hot water load profile is defined statistically with the program DHWcalc developed within the IEA SHC Task 26 [7].

Figure 3 shows the space heating and cooling annual load files, and the daily domestic hot water file.

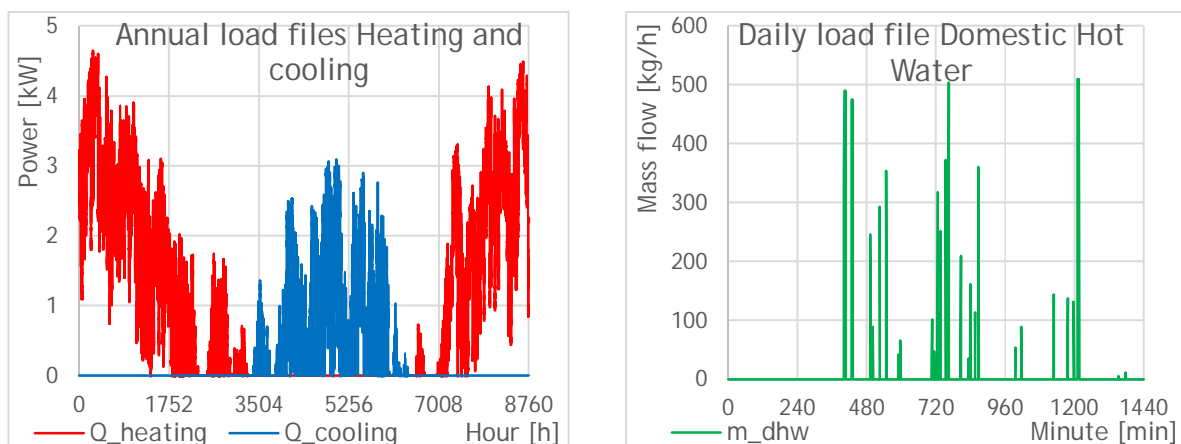


Figure 3 Example of annual Space heating and cooling loads and daily domestic hot water load.

## 4.4 Boundary conditions selection

After having defined the boundary conditions, these have to be selected to create a short test sequence. To perform the selection the “Event” was defined. In the procedure developed for the dynamic test of components [2,3], the “event” was defined as the continuous period of working of the component. In this case, the Event corresponds to one day as in most of the other procedures.

The procedure has to evaluate the system annual performance testing only a few events caught within the yearly operation.

This is a delicate stage of the procedure and there is much discussion underway around it and different approaches are used in different research centres. This is because each day is characterised by its own irradiance profile, temperature profile, humidity profile, load profile etc., and the erroneous selection of few reference days can bring to large errors in the assessment of the system performance.

This procedure adopts an objective method for the selection of the days which is based on clustering mathematical approach. Clustering means grouping a set of objects in such a way that objects in the same group (cluster) are more similar to each other than those in other groups. It creates “N” groups of days. The parameter “N” depends on the wished duration of the test chosen (N clusters = N days duration of test).

The clusters outline is calculated by minimizing, for every cluster, the distance between the cluster points. From the clusters it is identified the cluster center called “centroid”. The reference days to be selected are the closest to the centroids in the respective clusters. They are called “medoids” as indicated in Figure 4.

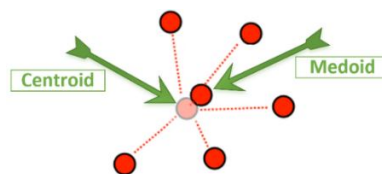


Figure 4 Centroid and medoid definition.

The method adopted uses a 2D classification of days: the coordinates are defined with the average temperature and the total horizontal irradiation.

Figure 5 shows the selected days as a function of external temperature and global horizontal irradiation. It can be noticed that the days are quite distributed during the year.

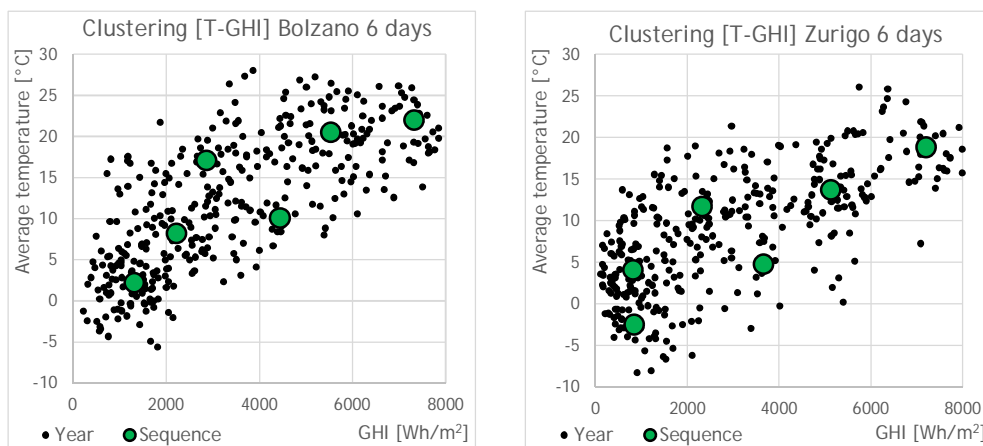


Figure 5 Identification of 6 day selected. A) Bolzano B) Zurich.

## 4.5 Boundary conditions emulation in the laboratory

The boundary conditions influence the thermal system in many ways. In particular, the air temperature and humidity, the solar irradiance and other parameters related to the weather influence the building heating and cooling demands. The air temperature and the irradiation also influences the performances of a thermal system components like heat pumps and solar thermal collectors.

The laboratory infrastructure must emulate the effects of this boundary conditions on a generic thermal system some components being not installed in the laboratory (i.e. the building, the solar collectors, etc.).

Figure 6 shows an example of the system under analysis and the physical boundaries imposed to it in the lab. The part of the system installed in the laboratory is represented by the grey area. The components outside the boundary are emulated:

- Solar panels.
- External condensing/evaporating unit of the reversible heat pump.
- Domestic hot water system.
- Distribution system.

To be a representation of realistic working conditions, the system has to be installed in the laboratory with the same configuration used in the real installation. The laboratory does not have to influence the internal control of the system. Instead this has to evolve according to the manufacturer strategies set up.

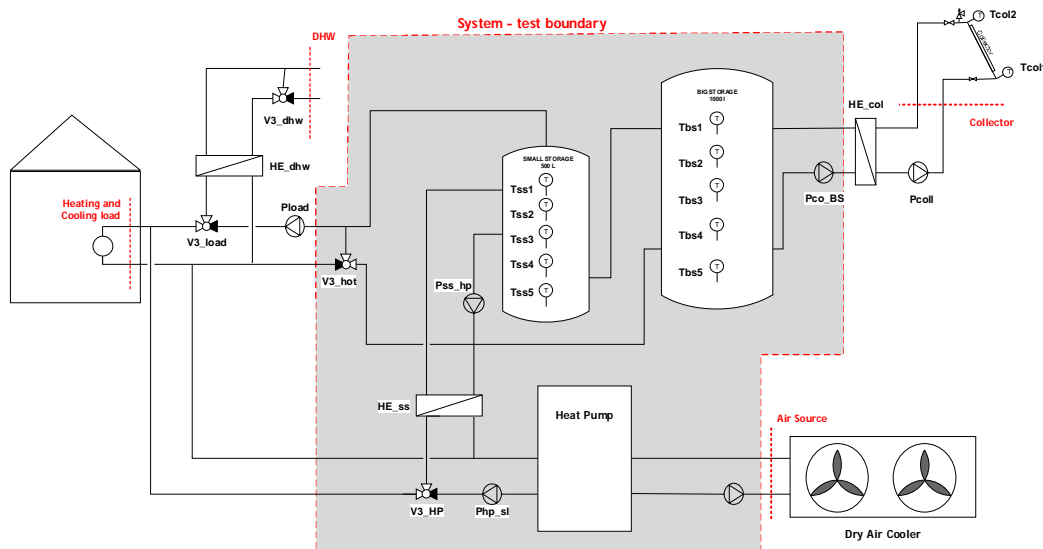


Figure 6 System boundary example.

Figure 7 shows the detail of the test method and how the data is exchanged between emulated components and system. The emulated components are performed with “concentrated parameter” models. These are used to calculate the set points imposed to the laboratory conditioning devices in the following way:

- at each time step, the outputs from the tested system are measured; these data together with the time-dependent weather conditions are inputted to the component subroutine;

- the component subroutine uses these values to calculate the response of each emulated device. These become the set points to the control subroutine;
- the laboratory controllers impose the set points as input to the tested system.

The emulation of each component is presented in the following paragraphs.

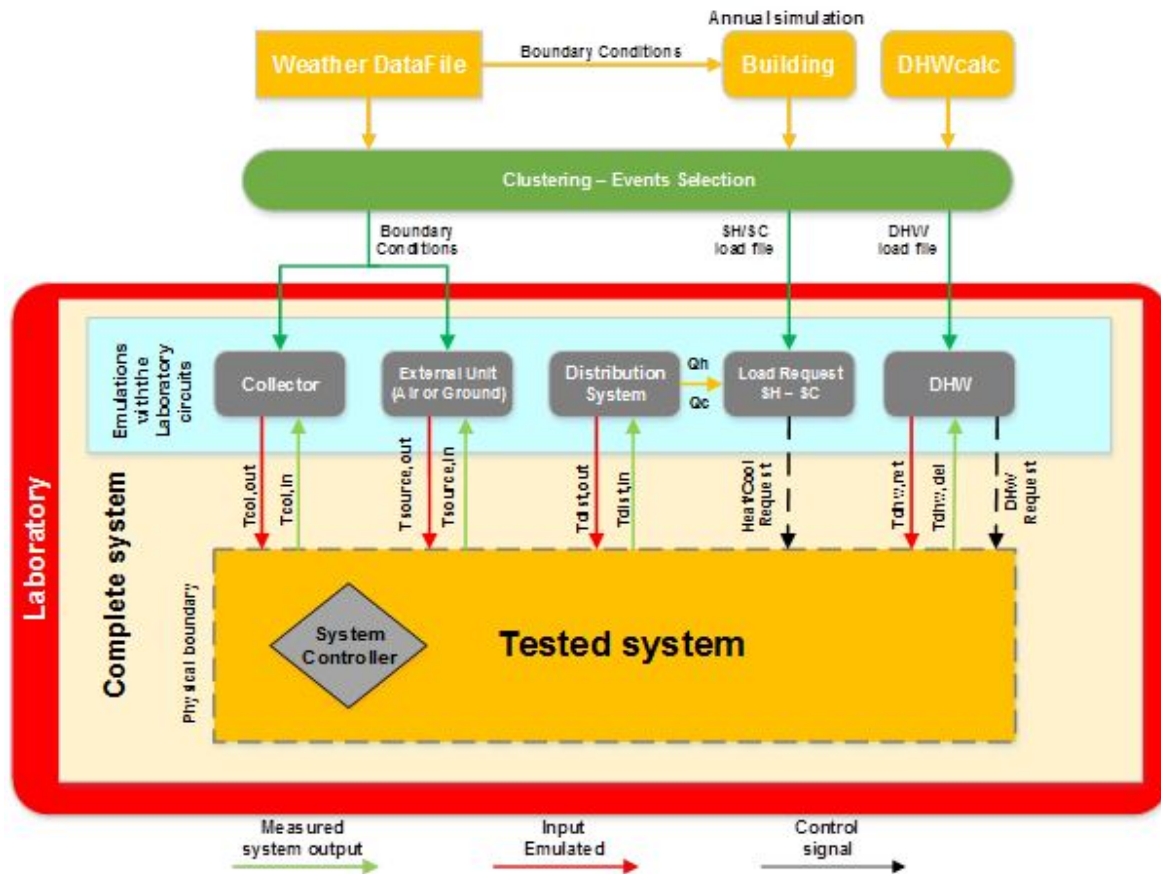


Figure 7 Schematization of the test method.

#### 4.5.1 Solar thermal collectors emulation

The collector field is not part of the tested system because it is difficult to install it physically and to achieve reproducible conditions in terms of ambient temperatures and irradiation. The collector output power and temperatures are reached with a dedicated circuit that uses a thermo-regulator. For the emulation of the collector field, the model requires the data in Table 3. This information is provided by the collector test certificate according to the reference standard (e.g. EN 12975-2).

Table 3 Collector parameters.

Number of collector modules		
Hydraulic configuration of the solar field		
Gross area of collector	A	[m <sup>2</sup> ]
Zero loss efficiency	$\eta_0$	[-]
linear heat loss coefficient	$a_1$	[W/m <sup>2</sup> K]
quadratic heat loss coefficient	$a_2$	[W/m <sup>2</sup> K <sup>2</sup> ]
specific heat capacitance of the collector	C	[kJ/m <sup>2</sup> K]

For each time step the collector efficiency is assessed with a quadratic correlation with respect to the reduced temperature difference  $T_m^*$ :

$$\eta = \eta_0 - a_1 \cdot T_m^* - a_2 \cdot IT_{col} \cdot T_m^{*2} \quad (1)$$

$$T_m^* = \frac{T_m - T_a}{IT_{col}} \quad (2)$$

When the collector circuit is activated, the outlet temperature is calculated from the inlet temperature and the total irradiance imposed on the collector surface by the weather file.

$$T_{out,col} = T_{in,col} + \frac{A_{col} \cdot IT_{col}}{\dot{m}_{col} \cdot cp} \quad (3)$$

To introduce an inertia effect, a moving average is applied to the outlet temperature.

#### 4.5.2 Distribution system emulation and heating & cooling load files

The space heating and cooling needs are defined from a fixed-load file as indicated in the paragraph 4.5, while only the behaviour of the water distribution system is emulated by means of a concentrated parameters model. The advantage of this is that a common, fixed load allows to compare between different systems. Furthermore, it avoids to emulate the entire building emulation that is more complicated than other emulations.

Since a real-time simulation is not performed, the internal air temperature of the building is unknown. Consequently, the exact behaviour of the thermostatic valves cannot be reproduced. Therefore, in order to take into account the effects of the discontinuous operation of the system, a different approach is needed compared to approaches which perform the emulation of the whole building.

The system activation is based on energetic considerations on the load. The equations (4), (5), (6), (7) are used to describe the space heating and cooling behaviour and these equations can be explained through Figure 8. At time “Start count” ( $\square_0$ ), a counter starts to count the cumulative energy of the load file (building load). When it reaches the “energy limit  $\Delta E_1$ ”, the system is activated at the time  $\square_1$  eq. (4). This “energy limit” is represented by the area under the red curve filled with orange dotted lines.



After the activation, the calculation of the cumulative load continues eq. (5), while the energy given by the system starts to be counted eq. (6). When these two energies are equal to  $\square_2$ , the system is deactivated eq. (7). This balance is represented by the area under the red curve filled with the red dotted lines (eq. (5)) and the area highlighted under the green line (eq. (6)). The red and green areas are equal. When those two areas become equal, the load request turns off and the counter is restarted. The blue triangular line is the cumulative count of energy debt which increases when the heat pump is turned off and decreases when it is turned on until “payment” of the debt. This calculation represents the effect of control of internal temperature with a thermostat.

$$\int_{\theta_0}^{\theta_1} \dot{Q}_{load} d\theta = \Delta E_{0-1} = \Delta E_1 \quad (4)$$

$$\int_{\theta_0}^{\theta_2} (\dot{Q}_{load}) d\theta = \Delta E_{0-1} + \Delta E_{1-2} \quad (5)$$

$$\int_{\theta_1}^{\theta_2} (\dot{Q}_{sys}) d\theta = -(\Delta E_{0-1} + \Delta E_{1-2}) \quad (6)$$

$$\int_{\theta_0}^{\theta_2} (\dot{Q}_{load} + \dot{Q}_{sys}) d\theta = 0 \quad (7)$$

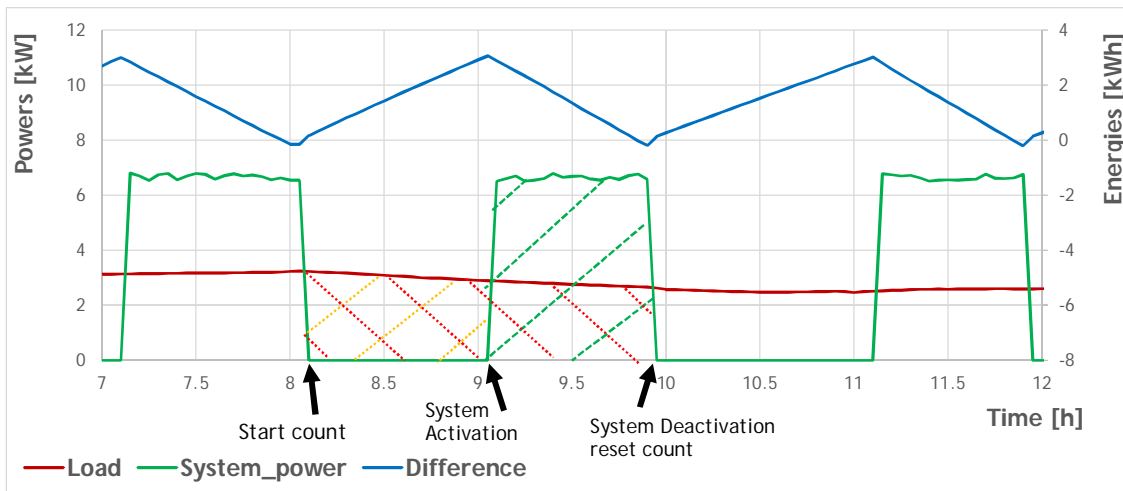


Figure 8 Heating and cooling load File. Activation principle.

As a reference for the heat distribution a radiant panels system is taken. The behaviour of the distribution system is modelled with simplified equations. These do not consider the inertial effects on the system; however in needed, the models could be improved to account for the inertia of both the slab and the building. As it is discussed by Haberl et al. [8] this assumption is necessary to avoid problems for the repeatability of the results of the short test sequence. This is due to the fact that the heat delivered on one day could be consumed in the next days of the sequence because of the thermal inertia typical of this distribution system.

Figure 9 shows the emulation principle. When the system is activated, it sets a flow and a temperature according to its control. From the measurement of the mass flow ( $m_{sys}$ ) and temperature ( $T_{out,sys}$ ), the emulation calculates the heat delivered to the building and the consequent return temperature.

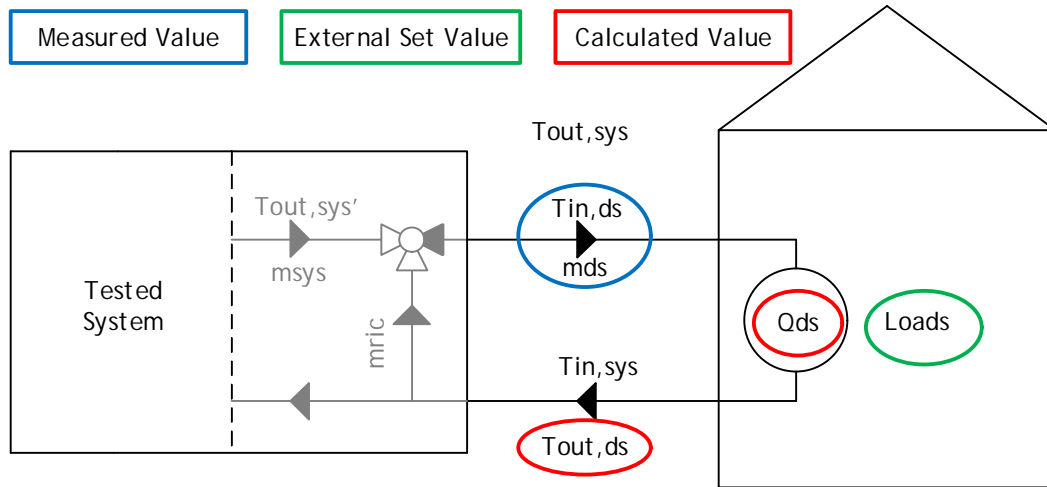


Figure 9 Distribution system emulation.

The thermal power of the radiant panel is calculated as a function of the delivery temperature. Four different equations have been defined for the two floors and the heating and cooling operations, being this validate through numerical simulations and monitoring data.

The difference between the simulation and the simplified emulation's energy calculations is lower than 3%.

### 4.5.3 Domestic hot water emulation

The annual profile of DHW has been defined in advance with a statistical profile using the program DHWcalc developed within the IEA SHC Task 26. The total annual energy consumption is 2550 kWh of useful heat. This "useful heat" is calculated with a hot water set-point of 40°C.

The annual consumption corresponds to a daily consumption of 7 kWh. From the annual sequence, a single day with consumption of 7 kWh was selected. To this consumption, an additional 4 kWh were added for the pre extraction heat loss: the hot water trapped in the pipeline at the end of extraction represents a heat loss and the plant has to provide this loss. The same profile is used in all selected events/days.

A dedicated circuit in the laboratory is used to reject the equivalent useful heat in order to get the return temperature from the measured supply water temperature. The heat to be rejected is defined by the DHW file:

$$\dot{Q}_{DHW} = \dot{m}_{dhw} \cdot cp \cdot \Delta T = \dot{m}_{dhw} \cdot cp \cdot (T_{hot,dhw} - T_{cold,dhw}) \quad (8)$$

The return temperature is calculated as consequence of the delivery temperature and the fixed draw-off:

$$T_{ret,dhw} = T_{del,dhw} - \frac{\dot{Q}_{DHW}}{\dot{m}_{dhw,sys} \cdot cp} \quad (9)$$

#### 4.5.4 External air unit emulation

Usual practice is to install the air units in a climatic chamber that reproduce the ambient condition of the external air units. If a climatic chamber is not available, also this is to be emulated.

The thermal power and the electric consumption are calculated as a function of the air temperature ( $T_{amb}$ ) and the inlet temperature ( $T_{in,dc}$ ). Two different equations were defined for the working mode (heat rejection and heat source).

$$\dot{Q}_{DC,source} = 1.3987 - 0.6416 \cdot T_{in,dc} + 0.622 \cdot T_{amb} \quad (10)$$

$$\dot{Q}_{DC,rejection} = -(19.1493 - 1.6325 \cdot T_{in,dc} + 1.0396 \cdot T_{amb}) \quad (11)$$

An empiric equation, that relates the electric consumption to the heat extracted/rejected from/to the air, was used to evaluate the electrical consumptions of the fans

$$\dot{W}_{el,fan} = k \cdot \dot{Q}_{th} \quad (12)$$

In Equation (12), it is important to distinguish whether the operation condition is heat extraction or rejection. The thermal power exchanged by the dry-cooler  $\dot{Q}_{th}$  is the thermal power transferred from the air to the evaporator if the heat pump works in heating mode, or it is the heat rejected from the condenser to the air if the heat pump works in cooling mode. The coefficient  $k$  is equal to 0.03 or 0.01, respectively.

Clearly these values refer to one specific air-unit model and brand. As per the solar collectors, the operational features of this components shall be known by the manufacturer, if the device cannot be tested directly.

#### 4.6 Test execution and data analysis

The test starts with two preconditioning phases, the test bench brings the storages to a predetermined temperature; after that, the last 24 hours event of the test series is performed as the second preconditioning phase. After that, the core sequence starts, lasting  $N$  days.

Figure 10 and Figure 11 shows an example of the profile file of external temperature, irradiance and loads as input to the test bench for a 6 days sequence.

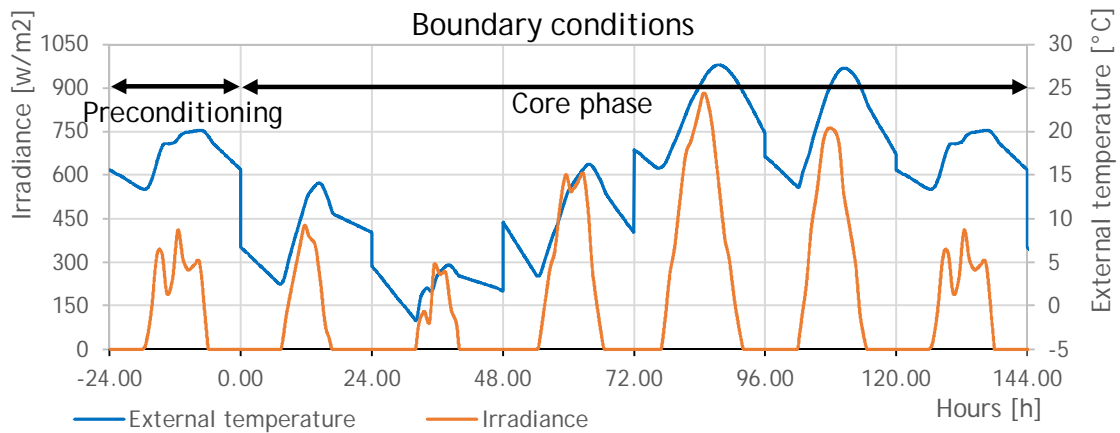


Figure 10 - Boundary conditions, preconditioning and core phase.

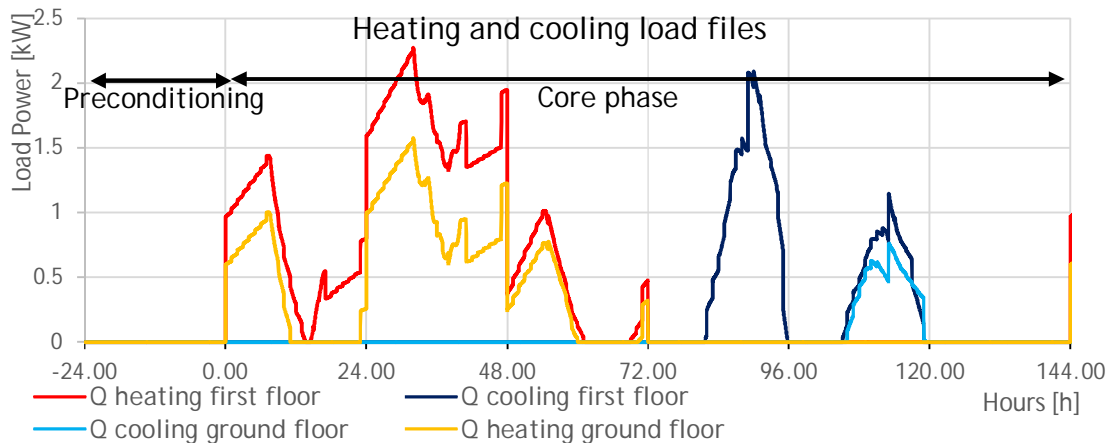


Figure 11 - Load file, preconditioning and core phase.

The last phase of the procedure after execution of tests concerns the analysis of the results.

Each data (temperatures, mass flows, positions of the valves, electrical powers) are acquired with a time step of 5 seconds. The analysis script calculates for each day of the sequence:

- Heating cooling and DHW loads
- Electrical powers and other system supply energies like natural gas, LPG, biomass etc.
- Performance ratio/figures - COP EER and SPF.

## 5 Case studies analysis

The system studied (Figure 6) is a solar assisted heat pump system, consisting of:

- An electrically driven water to water compression heat pump connected to an external air-unit
- Two storages connected in series with a volume of 500 and 1000 litres, respectively.
- Solar collectors (emulated) in the range of 8 to 16 m<sup>2</sup>
- The hydraulic distribution and the controller.

The heat pump included in the tested system is described in Table 4. This model is an electric driven water to water compression heat pump which uses the refrigerant R-410 A as working fluid. The installed compressor is a scroll-type compressor. An electrical resistance as backup system it is not installed in this unit. The capacity of model results oversized with respect to the instantaneous loads that it has to cover. This lead to short and frequent activations of the heat pump especially in cooling mode.

*Table 4: Main features of the heat pump that is included in the system.*

Compression Heat Pumps	
Heating capacity	9,42 kW <sub>th</sub>
COP	5,1
(EN14511:2013; 30/35°C-10/7°C)	
Cooling capacity	10,7 kW <sub>th</sub>
EER	5,2
(EN14511:2013; 23/18°C-30/35°C)	
Working fluid	R-410 A
Backup system	No
Compressor type	Scroll without inverter

### 5.1 Bolzano sequence – 6 days

The first sequence was defined with a 2D clustering approach selecting 6 days in the climate of Bolzano. The system has 16 m<sup>2</sup> of solar collectors and the load is defined with the building described previously.

Figure 12 shows the boundary conditions defined with the clustering. The last day of the sequence is used as preconditioning (as indicated with the orange box). The primary vertical axis indicates the temperature while the secondary vertical axis indicates the irradiance (on horizontal and on collector surface).

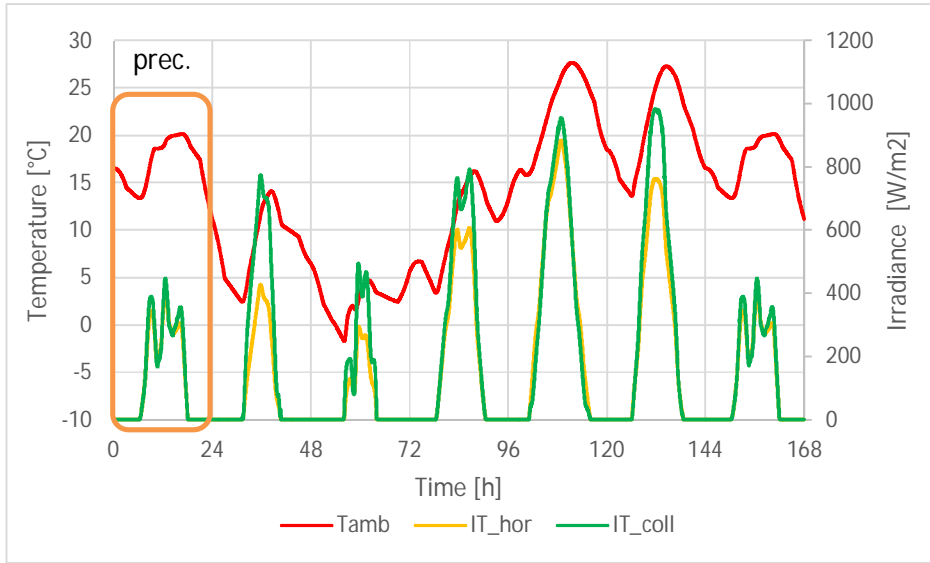


Figure 12 Temperature and irradiance boundary conditions in the six days sequence of Bolzano.

The sequence has been tested in the laboratory and the results compared to those obtained with a numerical simulation of the system.

### 5.1.1 Sequence results

Figure 13 shows the electrical consumption of the system evaluated with the test and with a simulation of the system during the different days of test. The simulation can be regarded as a CTSS method analysis.

In the figure it can be seen that the electrical energy is almost the same with the exception of days 4 and 6. The reason can be identified because of an inconsistency between test and simulation in the space cooling, explained with Figure 14.

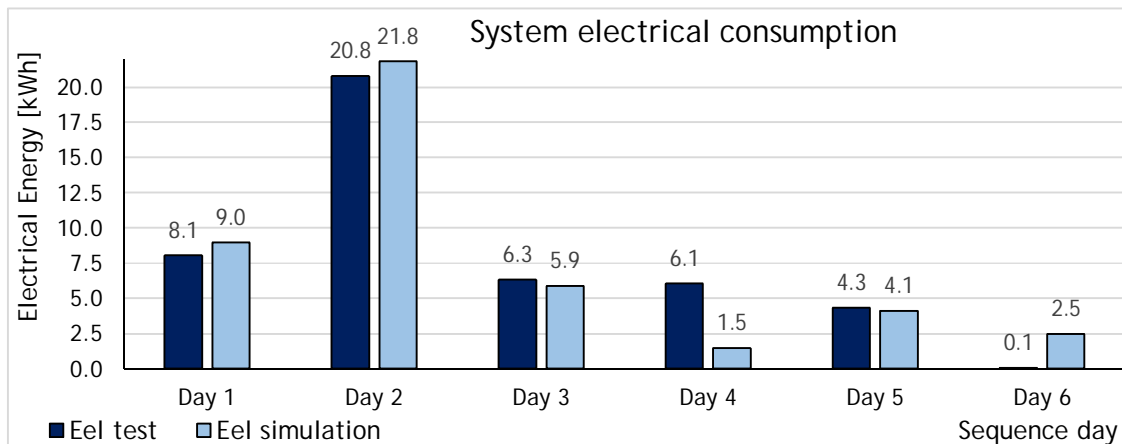


Figure 13 Comparison of electric consumption during the test with the simulation.

The high consumption in the day 2 can be explained by Figure 12 and Figure 14. This is the coldest day with very low irradiation and therefore is the day with the highest load. The consequence is that the system has to cover a high load with low coefficient of performance.

Figure 14 presents the space heating and space cooling energy use and electric consumption divided for the different days of test. In the first three days the building requires heating load, during day 4 and 5 requires cooling load and in the last day it does not require any space load.

In the figure, it is possible to notice that the cooling load is distributed differently between test and simulation. The inertia of the distribution system in the simulation is higher compared to the one modelled in the emulation: the effect is that during the simulation the load required during the day 4 it is satisfied partially in this day and partially during day 5. At the same time, the load of day 5 is partially satisfied during day 6. Instead, during the test the load is satisfied during within the specific day. If this is not effectively accounted for, the annual consumption extrapolated incorporates a certain amount of uncertainty.

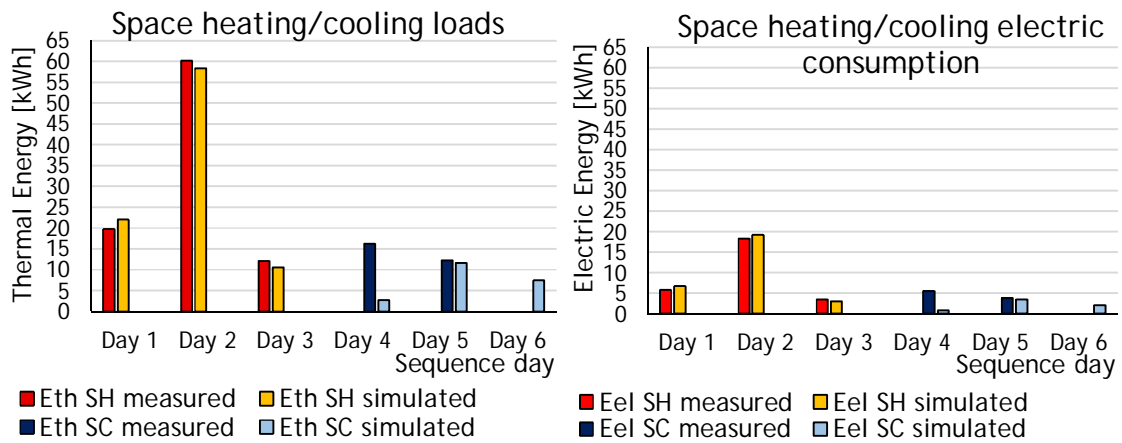


Figure 14 Space heating/cooling load and electric consumption.

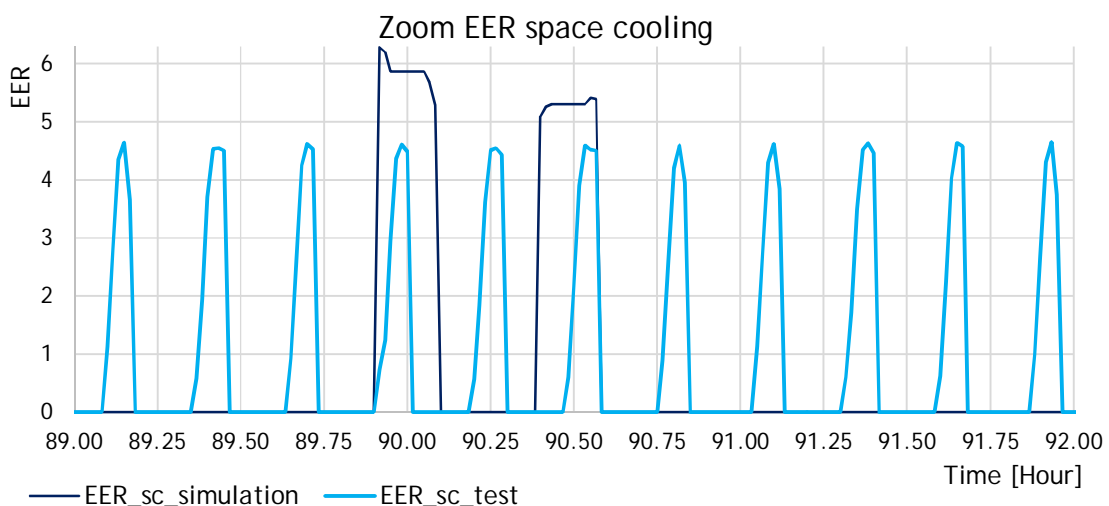


Figure 15 EER & COP in a zoom of the fourth test day.

In addition, when we look at time distribution of cooling performance we see a large difference between simulation and test (see Figure 15). During the test, the heat pump was activated 11 times for the space cooling scheme while the simulation only 2 times. This behaviour is explained with the lower inertia of the distribution system and the heat pump size (oversized with respect to the loads). The first cause (low inertia of distribution system) is a lack of the test method that has to be improved as stated already, while the second one (size of heat pump) is a lack of the tested system.

The consequence is also visible in terms of frequency distribution of EER in Figure 16. The test has a lower peak of efficiency in space cooling ( $EER_{test} = 4.5$  instead  $EER_{simulated} = 5.8$ ).

Conversely with respect to the simulation however, the test can catch a second peak at  $EER = 0.5$ , which is caused by the heat pump transients. The simulation does not highlight this effect because the model is made with a stationary performance map.

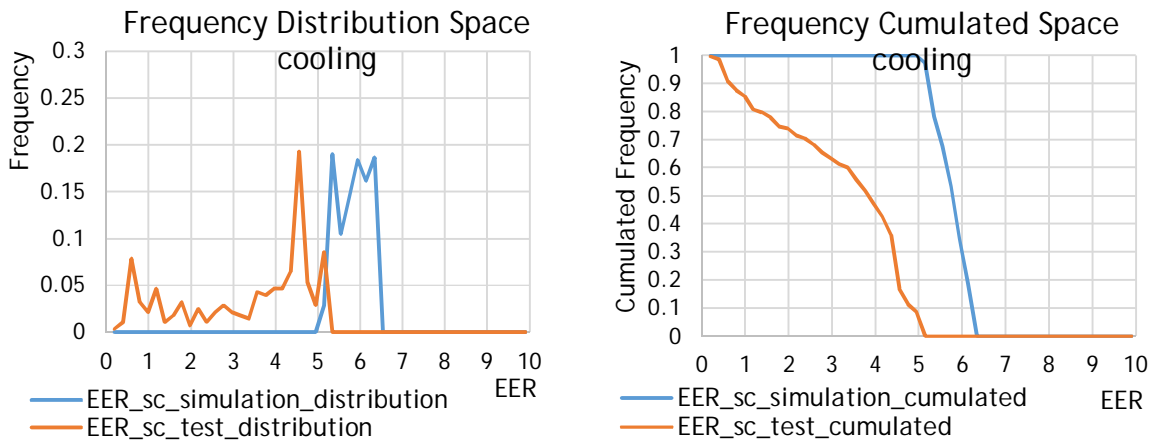


Figure 16 - Space cooling frequency and cumulated distribution.

Looking at the other loads -Space Heating and Domestic hot water (Figure 17 and Figure 18) -, the frequency analysis shows a good correlation between test and simulation.

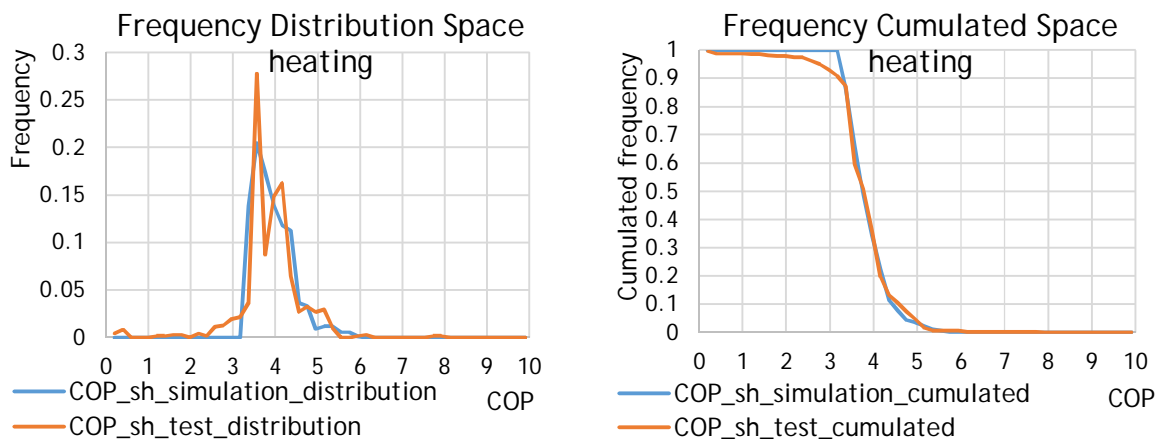


Figure 17 - Space heating frequency and cumulated distribution.



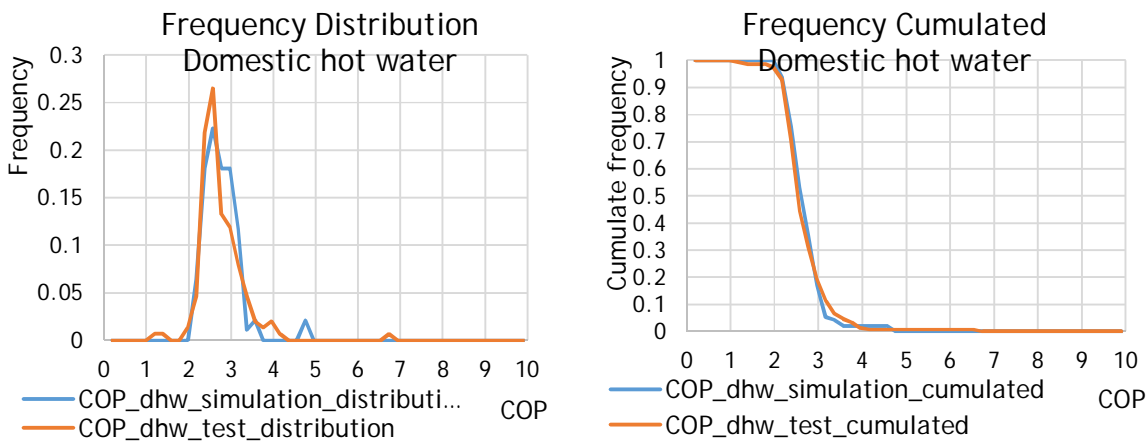


Figure 18 Domestic Hot Water frequency and cumulated distribution.

Figure 19 shows the domestic hot water load and its electrical consumption. These values are close to the simulation during all days.

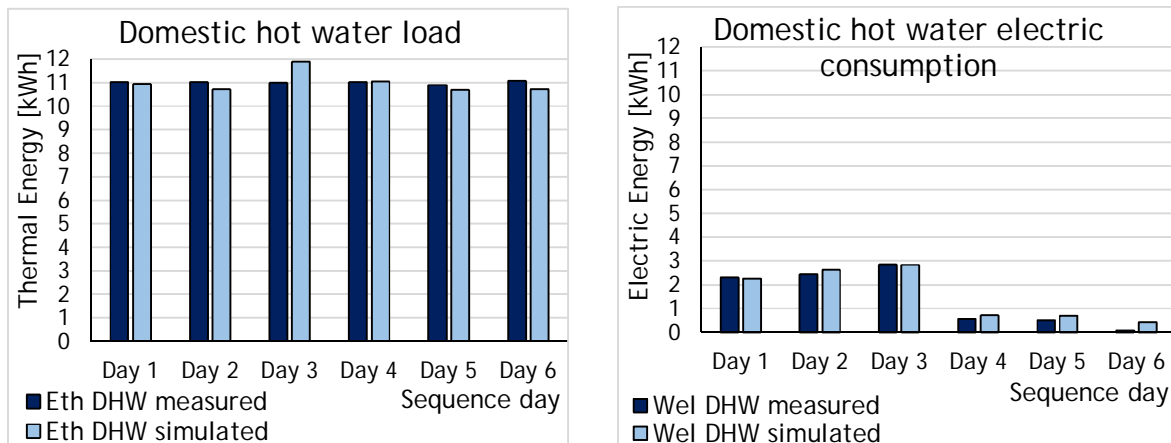


Figure 19 Domestic hot water load and electric consumption.

Figure 20 shows the DHW extraction counting of time and energy extracted, discretized by four temperature ranges. The figure presents a daily average condition. The delivery temperature is below 40°C for a large time (about 0.81 h compared to the 0.45 h of “useful energy”). In terms of energy extracted, 7kWh/day is delivered at 40 °C (useful energy) and only 4kWh/day is delivered at lower temperature as a pre extraction (non-useful energy).

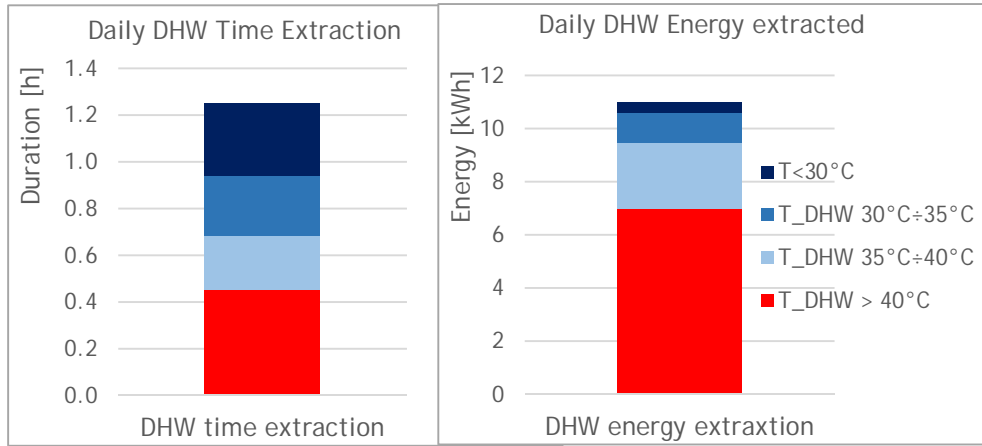


Figure 20 Domestic hot water duration and energy extracted.

From the loads and the electric consumption, the performance factors can be calculated. Figure 21 shows the comparison of the total performance factor ( $PF_{tot}$ ) obtained in the test and in the simulation. The difference between day 4 and day 6 is explained by the redistribution of the space cooling load shown above. The sequence SPF evaluated with the test is 4.2 while with the simulation is 4.0 (5% difference).

The PF is higher when the contribution of the solar source to the load is higher. The daily PF depends on the fraction of SH, SC and DHW loads respect the total load and from which sources satisfy these load. The DHW load is mainly covered by solar, therefore during day 6, the PF reaches the maximum value (in the test).

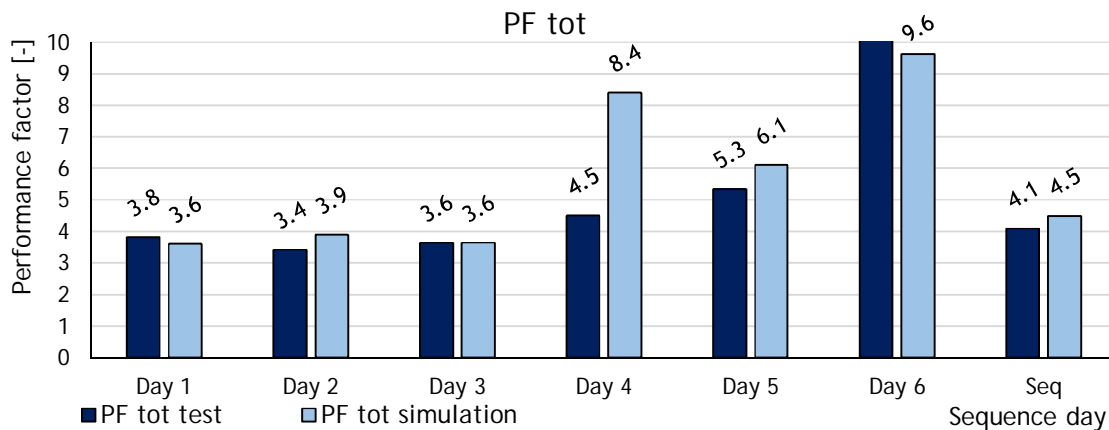


Figure 21 Seasonal performance factors.

Regarding the solar collector harvest, Figure 22 shows that the simulation collects much more energy compared to the emulation. The difference is about 23 kWh for the whole sequence (simulation 213.7 and emulated 190.5). This difference is mainly due to the day 2 and day 6 of the sequence. The different behaviour during these two days can be analysed considering the efficiency and the activation of the scheme of the collector. During day 2 and day 3, in the simulation, the collector is activated discontinuously and for longer time.

Figure 23 shows the collector efficiency during the whole sequence. The test and simulations have almost the same efficiency.

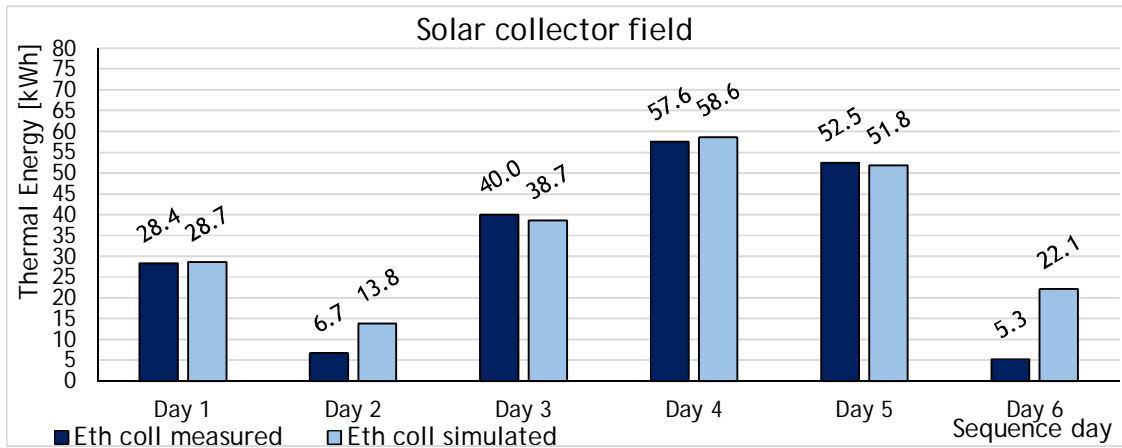


Figure 22 Solar collectors energy.

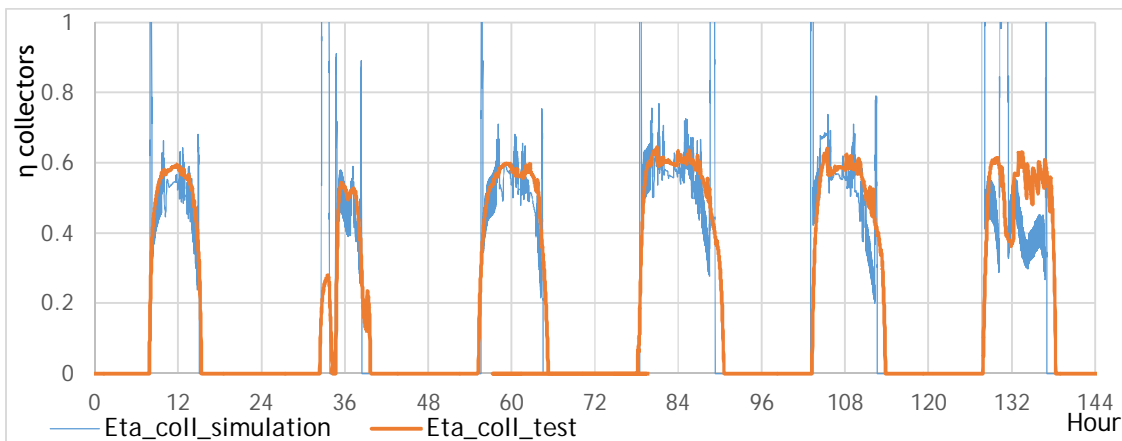


Figure 23 Solar collectors efficiency.

The frequency and cumulated distributions of solar collector efficiency are reported in Figure 24; the graph considers the six days sequence and shows how the simulation has similar behaviour compared to the test.

Figure 25 and Figure 26 compare the storages temperature measured in the test and simulated. This comparison is useful for validate the simulation model and explain some unexpected behaviours. Looking to the big storage (Figure 25), the temperature development is similar with exception of the temperature at the top of the storage when the collector is activated. The same effect is visible also in the small storage (Figure 26) when the heat pump is activated.

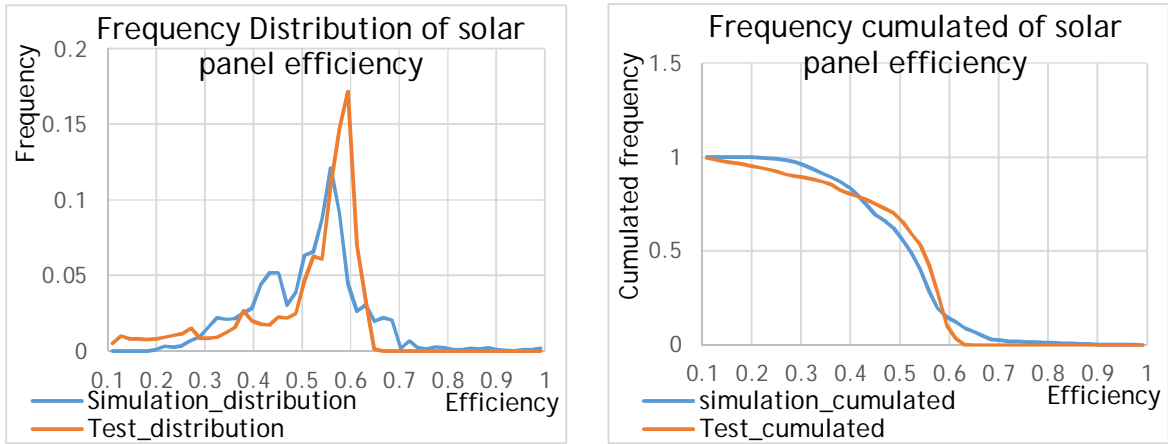


Figure 24 Solar collectors efficiency frequency.

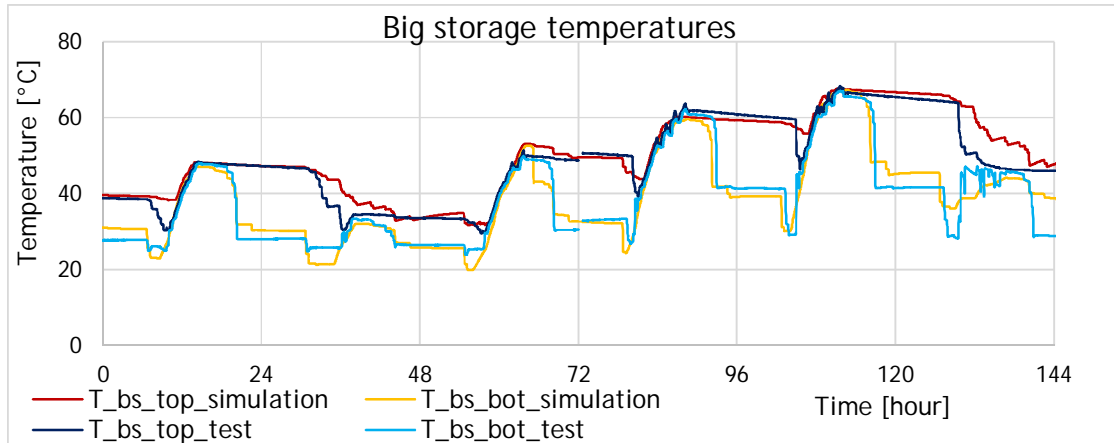


Figure 25 Big storage top and bot temperatures.

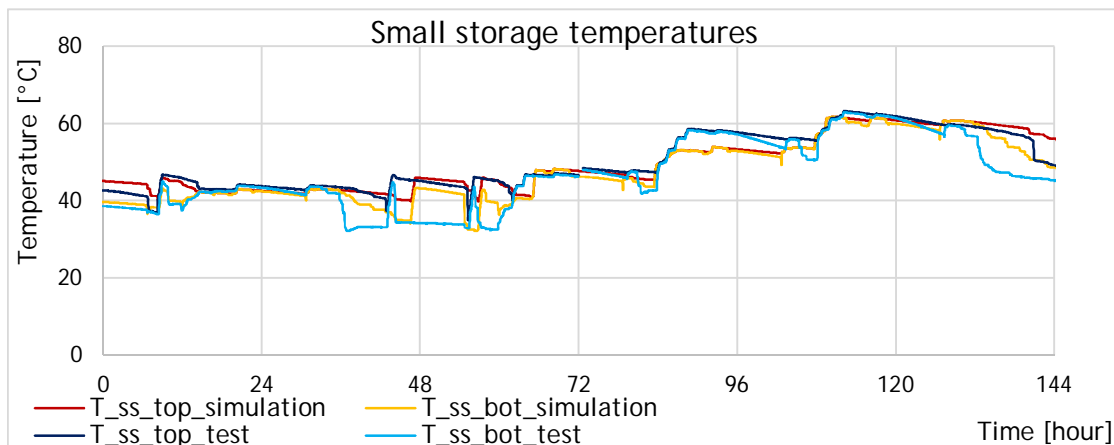


Figure 26 Small storage top and bot temperatures

Looking in detail to this effect for the small storage in Figure 27, we can see that in the winter days tested there is a consistent storage destratification (green arrows) caused by the in the initial phase of the storage charge with the heat pump. Before the heat pump reaching steady state conditions, a mass flow colder than the storage water is recirculated. This mixing causes the top storage temperature to fall. This inefficient behaviour suggests on the one hand that the simulation cannot catch properly this effect and, from the practical point of view, that a stratification separator should be integrated in the small storage for improve the system performance.

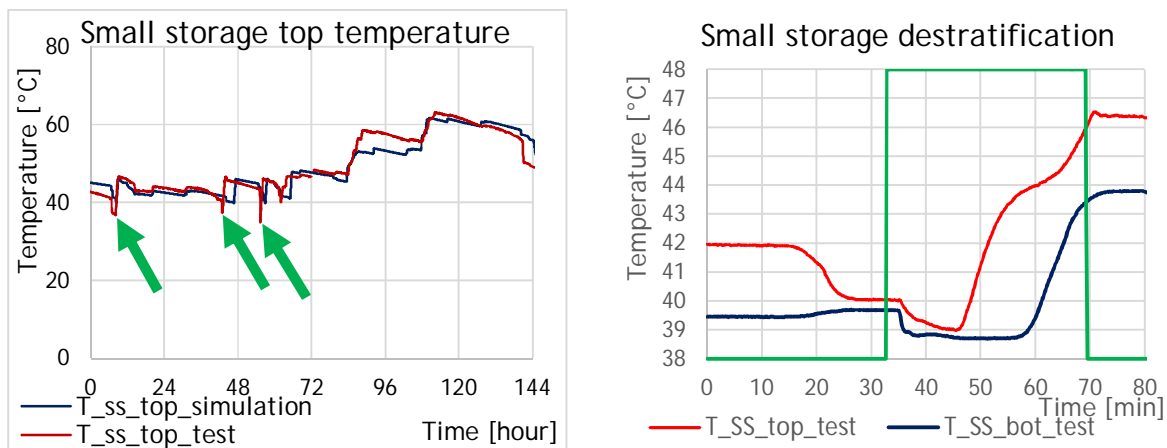


Figure 27 Stratification in the small storage.

### 5.1.2 Annual extrapolation

The goal of the procedure is not only to find the performance of the sequence, but also to evaluate the performance over a whole reference year. This is done by weighing the daily energy with the cluster size of each day.

In this case the sizes of the clusters are shown in Table 5, together with the annual main results of the test and of a yearly simulation. The seasonal performance factors and the solar fractions are indicated in the Table 6. Figure 28 to Figure 30 show graphically the table results, showing that the simulation and the test are quite similar.

Table 5 Cluster energy evaluation.

	Cluster 1	Cluster 2	Cluster 3	Cluster 4	Cluster 5	Cluster 6	Test	Simulation
Size	65	93	36	35	75	61	-	-
Eth DHW [kWh]	716.30	1024.86	396.00	386.05	817.50	675.88	<b>4016.6</b>	<b>3754.3</b>
Wel DHW [kWh]	149.50	226.92	102.24	19.60	37.50	5.49	<b>541.3</b>	<b>597.7</b>
Eth SH [kWh]	1283.10	5594.88	434.88	0.00	0.00	0.00	<b>7312.9</b>	<b>8363.1</b>
Wel SH [kWh]	373.75	1705.62	125.64	0.00	0.00	0.00	<b>2205.0</b>	<b>2272.3</b>
Eth SC [kWh]	0.00	0.00	0.00	569.10	918.75	0.00	<b>1487.9</b>	<b>1865.0</b>
Wel SC [kWh]	0.00	0.00	0.00	192.85	287.25	0.00	<b>480.1</b>	<b>450.1</b>

Eth coll [kWh]	1843.40	626.82	1438.20	2016.00	3937.50	322.69	<b>10184.6</b>	<b>11848.6</b>
----------------	---------	--------	---------	---------	---------	--------	----------------	----------------

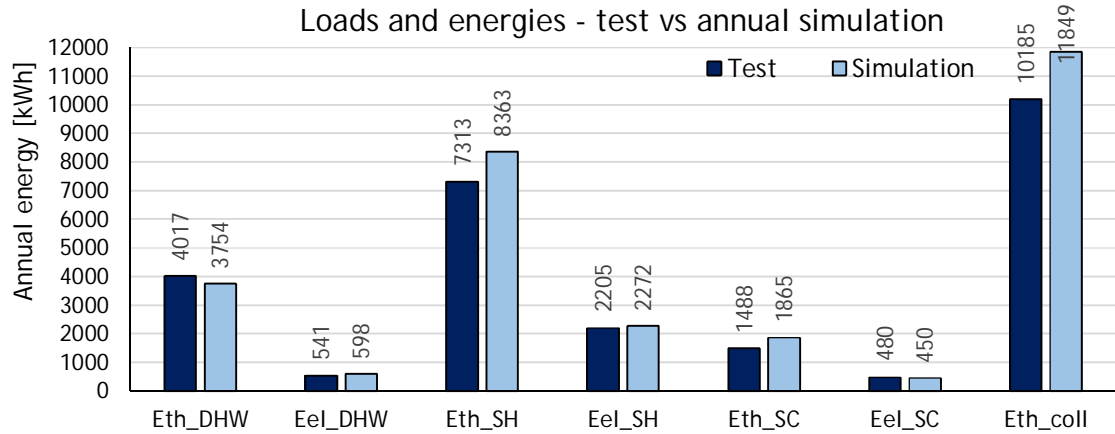


Figure 28 Load and energies test and annual simulation.

The performance factors for space heating and cooling in the test are a slightly smaller than in the simulation, while the performance factor of DHW is slightly higher. The SF obtained for the domestic hot water is 0.7, while is nearly 0 for solar heating in both cases.

Table 6 Seasonal performance factors and solar fractions.

	Test	Simulation
SPF_DHW	7.42	6.28
SPF_SH	3.32	3.68
SPF_SC	3.10	4.14
SPF_tot	3.97	4.28
SF_DHW	0.68	0.64
SF_SH	0.00	0.05

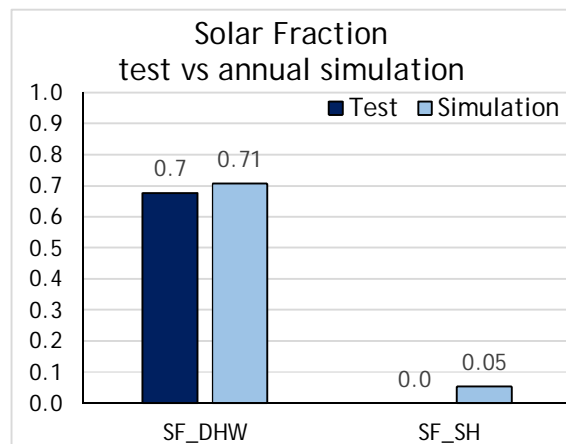
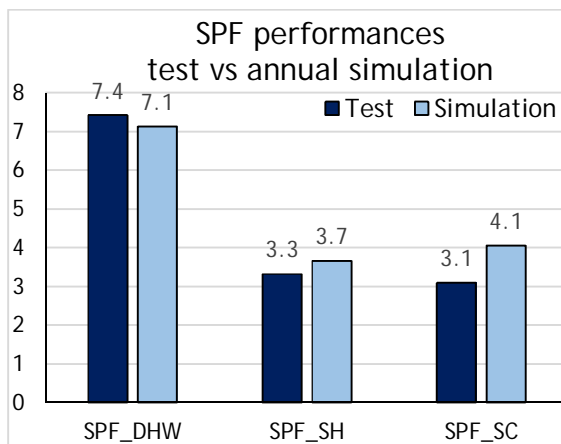


Figure 29 - Seasonal performance factors and solar fractions for heating, cooling and domestic hot water

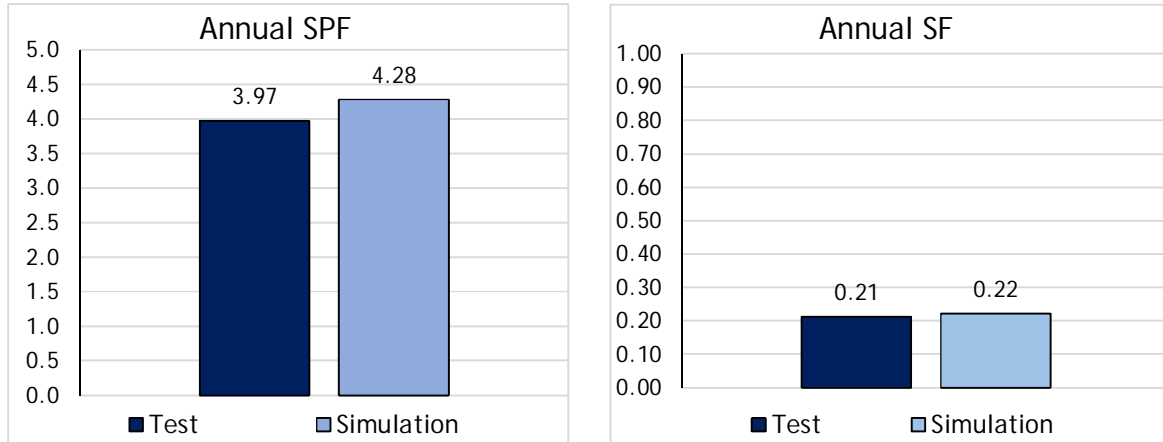


Figure 30 – Overall system seasonal performance factor and solar fraction

## 5.2 Zurich sequence – 6 days

The second complete test is performed with the Zurich climate. Figure 31 shows the boundary conditions selected. The difference with the sequence of Bolzano is that it is a colder climate and there is no space cooling request.

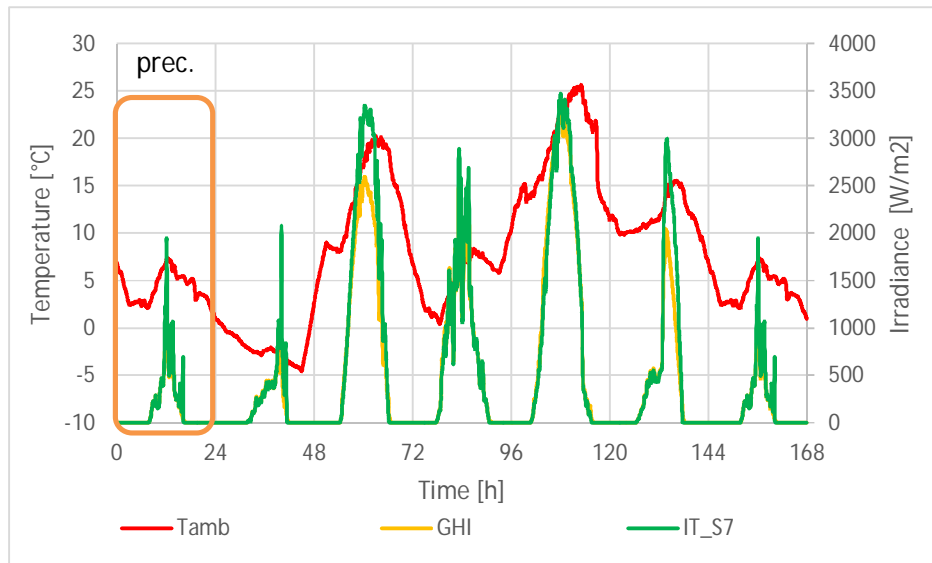


Figure 31 Temperature and irradiance boundary conditions in the six days sequence of Zurich.

### 5.2.1 Sequence results

Figure 32 shows the electric consumptions. The test consumption is few lower than that obtained through simulation, the motivation being explained with the following figures.

Figure 33 presents the comparison of heating load between the test and the simulation. Part of the space heating in day 2 is covered during day 3 and a small part of day 4 heating load is covered in day 5 which has no load request. This behaviour has already been highlighted.

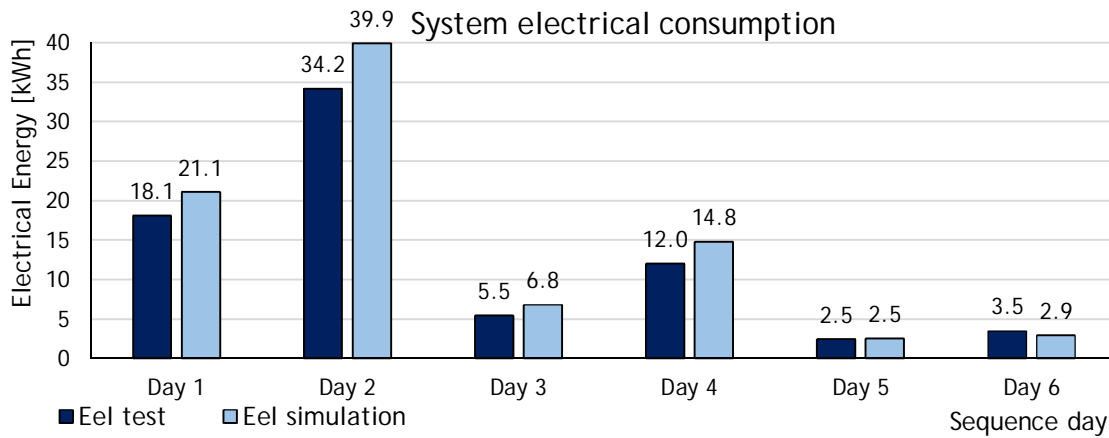


Figure 32 Electric consumption.

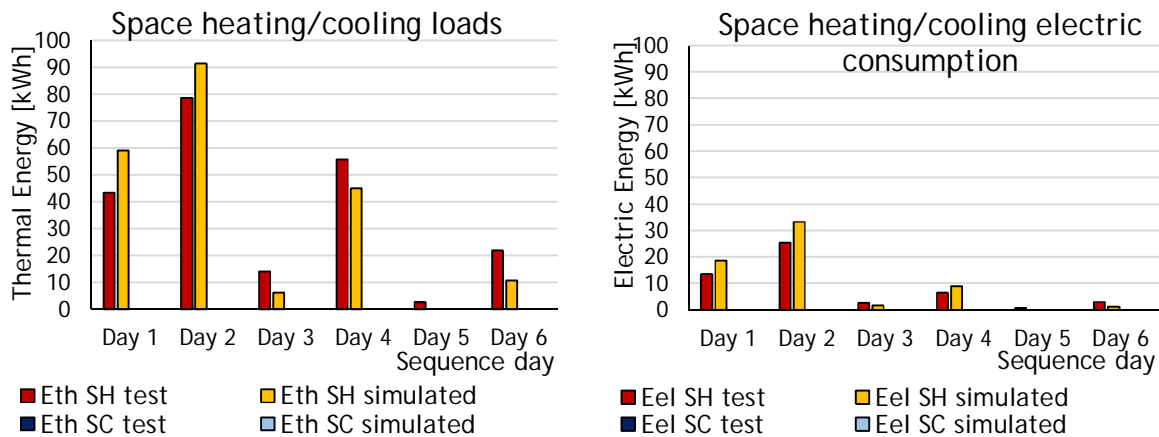


Figure 33 Space heating thermal and electric energy.

Figure 34 shows the solar panels collected energy. The day 1 and day 2 have collected less energy in the test than the one simulated. These are days with low irradiance and the transients have a predominant effect in the test when the irradiance is low as indicated also in the sequence of Bolzano.

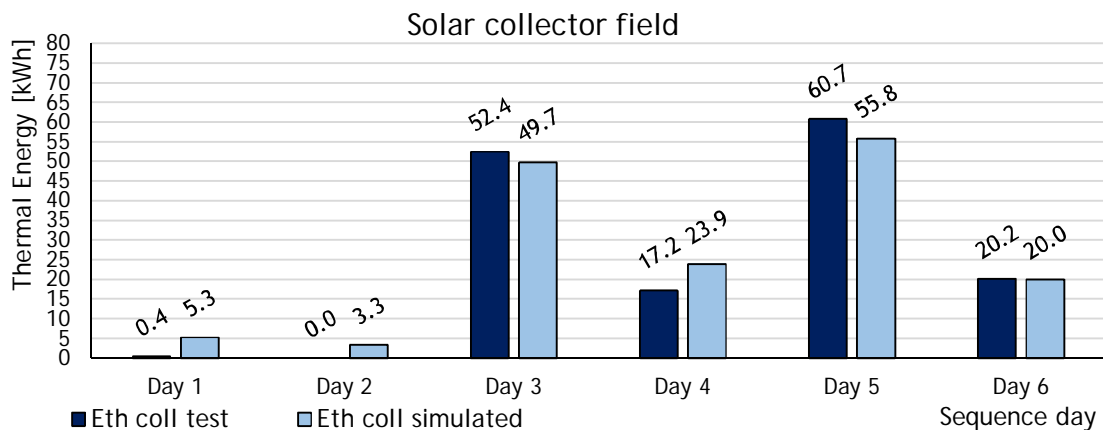


Figure 34 Solar collectors energy.



In the following figure the SPF is higher in the test: the difference between test and simulated SPF is about 0.5. The higher performance factor obtained during the 4 and 5 is related to the higher solar space heating contribution. The difference during day 3 and 6 is related to low thermal loads encountered, therefore again to the higher SF contribution for domestic hot water preparation.

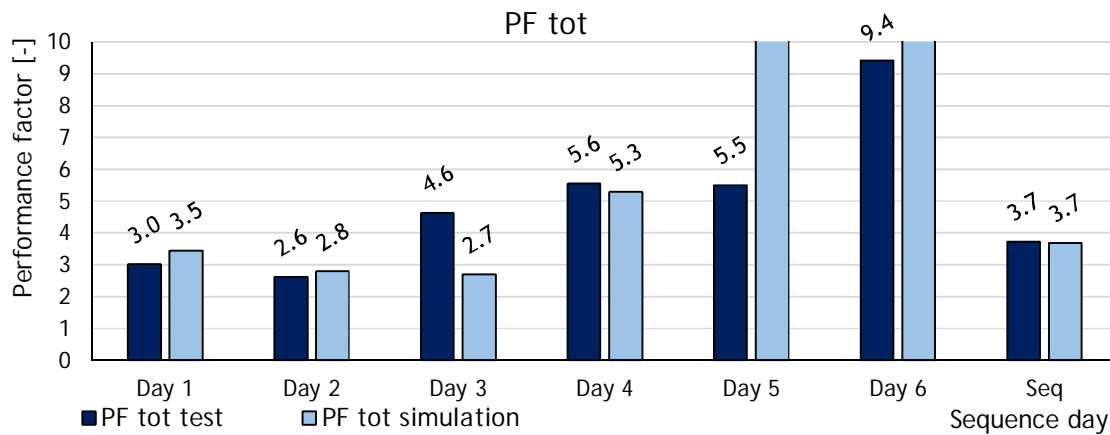


Figure 35 Performance figures test vs simulated sequence.

### 5.2.2 Annual extrapolation

Table 7 shows the annual energies calculated from the sequence days weighed with the cluster size while Table 8 shows the seasonal performance factors and the solar fractions. The domestic hot water SF in the simulation is two times higher compared to test even if the solar fraction is higher in test. On the other hand, the seasonal performance factor for space heating of the test is only 17% lower than simulation. The motivation is the low COP of heat pump obtained during day 2 and day 3.

Table 7 Cluster energy evaluation.

	Cluster 1	Cluster 2	Cluster 3	Cluster 4	Cluster 5	Cluster 6	Test	Simulation
Size	83	47	68	31	50	86	-	-
Eth DHW [kWh]	919	520	751	341	549	933	<b>4012</b>	<b>3716</b>
Wel DHW [kWh]	389	417	201	171	89	43	<b>1310</b>	<b>798</b>
Eth SH [kWh]	3603	3693	962	1727	132	1878	<b>11995</b>	<b>11477</b>
Wel SH [kWh]	1112	1189	170	201	35	255	<b>2962</b>	<b>3185</b>
Eth SC [kWh]	0	0	0	0	0	0	<b>0</b>	<b>331</b>
Wel SC [kWh]	0	0	0	0	0	0	<b>0</b>	<b>79</b>
Eth coll [kWh]	33	0	3563	533	3036	1733	<b>8898</b>	<b>10095</b>

Table 8 Seasonal performance factors and solar fractions.

	Test	Simulation
SPF_DHW	3.06	4.70
SPF_SH	4.05	3.60
SPF_SC	N.D.	4.20
SPF_tot	3.75	3.82
SF_DHW	0.46	0.84
SF_SH	0.22	0.08

Figure 36 shows graphically the results presented in the previous table.

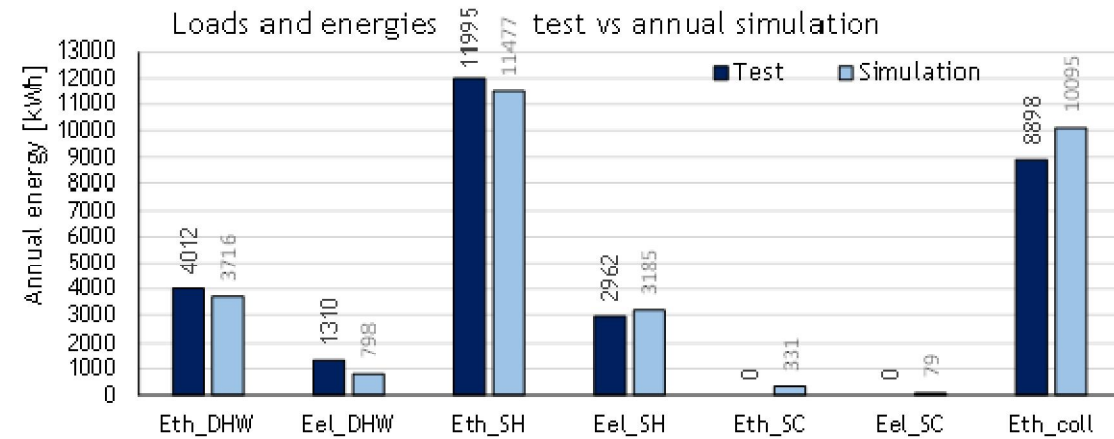


Figure 36 Loads and energies, test and annual simulation.

## 6 Conclusions

Solar heating and cooling systems are characterized by fully dynamic working conditions, therefore their performance assessment is not trivial. The actual standards present limits of application since not all the technologies available on the market are covered, and most of them are component oriented only.

Whole system dynamic test procedures are not coded yet, although a large portion of the research community is being developing test approaches in this direction.

A number of whole system dynamic test procedures have been assessed and compared to a new approach developed by EURAC.

In the latter case, a short sequence is defined with a mathematical approach that allows to easily select the reference test days. The procedure foresees the application of a common load file to the test of different systems in order to allow comparison on a common base. The components that cannot be installed in the laboratory are emulated with simplified concentrated parameters equations.

The test results for the climates of Bolzano and Zurich are here reported for a complex heating and cooling system.

The evaluation of seasonal performance directly extrapolated from the test results have been compared to the ones obtained with an annual simulation of the system. The six day sequences presents values of seasonal performance factor and solar fraction with a small deviation with respect to the annual simulated ones. The total seasonal performance factor for the two climates deviates from the annual simulation about 0.2, while the solar fraction deviates from the annual simulation of about 1 %.

Looking more in detail into the time series and frequency distributions of performance, the tests highlight some limits of the tested system that in the simulation are not identified, since numerical models do not consider capacitance effects. Incidentally, the calibration of inertial effects in numerical models of e.g. storage tanks and heat pumps (both compression and sorption) is a time consuming and complicated task. Inertial effects in cooling season are only verified through real-like dynamic tests.

On the other hand, further work is needed to improve the concentrated parameters models used in the presented test method. This is mostly through with respect to the capacitance effects related to the heating distribution system and to the whole building.

In addition to the comparison among different whole system dynamic test methods, it is demonstrated here how strongly the performance of a heating and cooling system is affected by time varying boundary conditions and by the control strategies implemented. As such, whole system tests seem to be necessary when the performance assessment of complex systems is tackled; the performance characterisation obtained as the summation of the single components' operation might eventually result in substantial unpredictable under- / over-estimations.

This should be considered when new rating standards and labelling procedures for heating and cooling systems are developed in the future.

## Bibliography

- [1] M.Y. Haller, R. Haberl, T. Persson, C. Bales, P. Kovacs, D. Chèze, et al., Dynamic whole system testing of combined renewable heating systems – The current state of the art, *Energy Build.* 66 (2013) 667–677. doi:10.1016/j.enbuild.2013.07.052.
- [2] D. Menegon, A. Vittoriosi, R. Fedrizzi, A new test procedure for the dynamic laboratory characterization of thermal systems and their components, *Energy Build.* 84 (2014) 182–192. doi:10.1016/j.enbuild.2014.07.085.
- [3] A. Vittoriosi, R. Fedrizzi, D. Menegon, Evaluation of Dynamic Operation Effects for a Heat Pump in a Solar Combi-plus System, in: *International Solar Energy Society*, 2015: pp. 1–10. doi:10.18086/eurosun.2014.03.28.
- [4] T. Letz, Validation and background information on the FSC procedure, 2002. [http://archive.iea-shc.org/publications/downloads/task26-validation\\_and\\_background\\_information.pdf](http://archive.iea-shc.org/publications/downloads/task26-validation_and_background_information.pdf).
- [5] A. Napolitano, W. Sparber, A. Thür, P. Finocchiaro, B. Nocke, Monitoring Procedure for Solar Cooling Systems, 2011. [http://iea-shc-task38.org/reports/Task38-SubtaskA-B\\_report\\_D-B3b\\_D-A3a\\_monitoring%20procedure%20V6.pdf/view](http://iea-shc-task38.org/reports/Task38-SubtaskA-B_report_D-B3b_D-A3a_monitoring%20procedure%20V6.pdf/view).
- [6] L. Siso, O. Gavalda, Formula of Cost-Analysis Tool, 2007.
- [7] U. Jordan, K. Vajen, DHWcalc: Program to generate domestic hot water profiles with statistical means for user defined conditions, in: *ISES Sol. World Congr.*, 2005. <http://solar-publikationen.umwelt-uni-kassel.de/uploads/2005%20ISES-SWC%20Jordan%20und%20Vajen%20Program%20to%20Generate%20Domestic%20Hot%20Water%20Profiles%20with%20Statistical%20Means%20for%20User%20Defined%20Conditions.pdf> (accessed March 5, 2015).
- [8] R. Haberl, M.Y. Haller, P. Papillon, D. Chèze, T. Persson, C. Bales, Testing of combined heating systems for small houses: Improved procedures for whole system test methods, (2015). [http://www.macsheep.spf.ch/fileadmin/user\\_upload/macsheep/dokumente/MacSheep\\_D2\\_3\\_Improved\\_Procedures\\_for\\_whole\\_system\\_testing\\_150211\\_final.pdf](http://www.macsheep.spf.ch/fileadmin/user_upload/macsheep/dokumente/MacSheep_D2_3_Improved_Procedures_for_whole_system_testing_150211_final.pdf) (accessed March 19, 2015).