REVIEW ON TESTING AND RATING PROCEDURES FOR SOLAR THERMAL AND HEAT PUMP SYSTEMS AND COMPONENTS

Technical Report 5.1.2

AIT

Ivan Malenković

With contributions from:

CENER

Enric Mateu Serrats

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Contact Info

Address: Giefinggasse 2, 1210 Vienna, Austria
Tel.: +43 50 550 6350
Fax: +43 50 550 6679
Email: ivan.malenkovic@ait.ac.at

Quality Assurance in solar thermal heating and cooling technology – keeping track with recent and upcoming developments

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1 Introduction

A large number of integrated heat pump and solar thermal systems (SHP), mostly for heating and hot water preparation for residential applications, have recently been brought to the market in many European countries [1]. Although standardised testing and rating procedures, as well as different quality labels, are available for both single technologies, the combined systems are lacking both. Besides, the components optimised for an integrated operation i.e. the solar collector and the heat pump unit often do not meet the quality standards needed in the respective labelling scheme. This is mainly due to the fact, that the current standards have not been designed to cover many of the operation conditions, specific designs etc. which are found in combined systems. This fact can have a negative impact on the further development and marketing of solar and heat pump systems in the mid and long term. It is therefore very important for a sustainable development of this technology to include relevant operating conditions and specific features of the components into current standards and to consider developing new procedures for their performance evaluation, as well as the performance evaluation of the entire systems. Secondly, it is important to define the performance indicators for SHP systems to be able to compare the systems among themselves, as well as with other technologies.

Therefore, relevant standards were analysed to assess their applicability for component and system testing of SHP products. The review covered three groups of available national and international standards and guidelines:

- Standards and guidelines for testing and rating of electrically driven heat pumps;
- Standards for performance evaluation of solar thermal collectors;
- Standards for performance evaluation of solar thermal systems.

The following normative documents were collected and reviewed:

<u>Heat pumps</u>

- EN 14511-2 [2];
- ANSI/ASHRAE 37 [3];
- AHRI 320, 325, 330 [4-6];
- EN14825 [7];
- EN 16147 [8];
- VDI 4650-1 [9];
- EN 15316-4-2 [10];

<u>Solar thermal collectors</u>

- ISO 9806 [11-13];
- EN 12975-2 [14];
- ASHRAE 93 [15];

<u>Solar thermal systems</u>

- EN 12976-2 [16];
- EN 12977 [17-21].

2 Standards and guidelines for heat pumps

2.1 EN14511: Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling (check if UPDATE needed)

EN 14511 was drafted by the CEN/TC 113 “Heat Pumps and Air-Conditioning Units”. The current valid version is EN 14511:2011.

The scope of the standard includes electrically driven heat pumps and chillers for space heating and/or cooling. Heat transfer media on heat source and heat sink side can be air, water or brine.

The standard includes four parts:

- Part 1: Terms and definitions;
- Part 2: Test conditions;
- Part 3: Test methods;
- Part 4: Requirements.

EN 14511 defines test procedures for the rating of heat pumps and chillers under steady state conditions (except certain air-source units in heating mode) and at full capacity. The heating and/or cooling capacity of units for hydronic distribution systems is determined by measurement of the volume flow of the heat transfer medium and the inlet and outlet temperatures at the water or brine heat exchanger, taking into consideration the specific heat capacity and density of the heat transfer medium. For air-source units, either the air enthalpy method or the calorimeter room methods can be used. The electrical power input is measured directly.

The unit can be tested under one or more operating conditions depending on the manufacturer’s needs and the type of the unit. One standard rating condition comprises of one temperature and mass flow rate condition of the heat transfer fluid in the evaporator and one in the condenser. Additionally, the unit can be tested under defined application rating conditions, but these tests are not compulsory. Depending on the designation of the unit regarding the type of installation (indoors or outdoors) made by the manufacturers, different environmental conditions according to Table 1 must be applied.

Table 1: Environmental conditions for units according to the installation site

Unit type	indoor installation in °C dry bulb (wet bulb)	outdoor installation in °C dry bulb (wet bulb)
W/W, B/W	15 to 30	-
A/W with duct connection on the air inlet and outlet side	15 to 30	as inlet air temperatures
A/W without duct connection on the air inlet side	as inlet air temperatures	as inlet air temperatures
W/A, B/A with duct connection on the air inlet and air outlet side	15 to 30	-
W/A, B/A without duct connection on the air inlet and air outlet side	as inlet liquid temperatures	as inlet air temperatures
A/A with duct connection on the outdoor air inlet and outlet side	15 to 30	-
A/A without duct connection on the outdoor air inlet and outlet side	as inlet air temperatures	-
A/A with duct connection on the indoor air inlet and outlet side	-	as inlet air temperatures
W/W, B/W operating in cooling mode	-	25 to 35
W/W, B/W operating in heating mode	-	0 to 7

Rating conditions are given for a number of unit types:

- water-to-water, brine-to-water, water-to-brine and brine-to-brine units in heating and cooling mode;
- air-to-water and air-to-brine units in heating and cooling mode;
- liquid chilling packages for heat recovery condenser and with a remote condenser;
- basic, multiple circuit and modular air-cooled multisplit systems in the heating and cooling mode;
- modular heat recovery air-cooled multisplit systems;
- basic, multiple circuit and modular water-cooled multisplit systems in the heating and cooling mode.

Only the first two unit types will be discussed here since the others have low relevance for SHP systems.

The temperatures and the mass flow rates of liquid heat transfer media both for the heat source and for the heat sink are fixed for all rating conditions. The mass flow rate remains constant throughout the test. For air as heat transfer media, only the inlet dry bulb temperatures (and for some cases wet bulb temperatures) are fixed. In Table 2 through Table 5 an overview of all temperature levels is given:

Table 2: Outdoor heat exchanger temperatures in the heating mode

		Outdoor heat exchanger Inlet temperature	Outdoor heat exchanger Outlet temperature
Heating, standard rating conditions	water	10	7
	brine	0	-3
	outside air - dry (wet) bulb	7	6
	exhaust air - dry (wet) bulb	20	12
Heating, application rating conditions	water	15	*
	brine	5	*
	outside air - dry (wet) bulb	2 (1)	**
		-7 (-8)	**
		-15 (-)	**

Table 3: Indoor heat exchanger temperatures in the heating mode

		Indoor heat exchanger Inlet temperature	Indoor heat exchanger Outlet temperature
Heating, standard rating conditions	low temperatures	35	30
	medium temperatures	45	40
	high temperatures	55	47
	very high temperatures	65	50
Heating, application rating conditions	low temperatures	35	*
	medium temperatures	45	*
	high temperatures	55	*
	very high temperatures	65	*

Table 4: Outdoor heat exchanger temperatures in the cooling mode

		Outdoor heat exchnager Inlet temperature	Outdoor heat exchnager Outlet temperature
Cooling, standard rating conditions	brine and water - cooling tower	30	35
	brine and water - ground coupled	10	15
	air (for water and brine) - dry bulb	35	**
Heating, application rating conditions	air (for water - medium temperatures) -dry bulb	46	**
	air (for water - medium and low temperatures, brin)e - dry bulb	27	**

Table 5: Indoor heat exchanger temperatures in the cooling mode

		Indoor heat exchanger Inlet temperature	Indoor heat exchanger Outlet temperature
Cooling, standard rating conditions	water - medium temperatures	23	18
	water - low temperatures	12	7
	brine	0	-5

* The test is performed at the flow rate obtained during the test at the standard rating conditions; ** Not defined

The heating and the cooling capacities for steady state operation are calculated according to eq. 7 and eq. 8:

$$P_H = q \cdot \rho \cdot c_p \cdot \Delta T \qquad \text{eq. 1}$$

$$P_C = q \cdot \rho \cdot c_p \cdot \Delta T \qquad \text{eq. 2}$$

where

P_H	is the heating capacity;
P_C	is the cooling capacity;
q	is the volume flow rate of the heat transfer medium;
ρ	is the density of the heat transfer medium;
c_p	is the specific heat of the heat transfer fluid at constant pressure;
ΔT	is the difference between inlet and outlet temperatures of the heat transfer medium.

The measured power input includes the overall consumption of the unit including all fans and pumps which make an integral part of the unit. However, in case of integrated liquid pumps and ducted air units, only a part of the respective power input is taken into account which corresponds to the power needed to overcome the pressure drop over the internal heat exchanger (e.g. evaporator or condenser) or the air duct. On the other hand, if a liquid pump or a fan of a ducted unit is not integrated into the unit but needed for the operation, a certain power will be added to the overall power input which corresponds to the pressure drop over an integrated heat exchanger or the duct. Both can be calculated from equations 5 and 8, respectively. For units without duct connection, the entire fan power is included in the total power consumption.

Liquid pumps and fans dissipate one part of their electrical power input to the heat transfer media itself. This is also taken into account in the heat balance for the calculation of the Coefficient of Performance (COP, for heating mode) and Energy Efficiency Ratio (EER, for cooling mode) by adding or subtracting the same amount of power as in the case of the pressure drop consideration from the heating or cooling capacity.

The COP or the EER are thus calculated as follows:

a. if a liquid pump is integrated into the unit:

$$COP = \frac{P_H}{P_E - P_{LP,F}} \qquad \text{eq. 3}$$

$$EER = \frac{P_C}{P_E - P_{LP,F}} \qquad \text{eq. 4}$$

$$P_{LP,F} = \frac{q \times \Delta p_e}{\eta} \qquad \text{eq. 5}$$

b. if a liquid pump is not integrated into the unit:

$$COP = \frac{P_H}{P_E + P_{LP,F}} \quad \text{eq. 6}$$

$$EER = \frac{P_C}{P_E + P_{LP,F}} \quad \text{eq. 7}$$

$$P_{LP,F} = \frac{q \times (-\Delta p_i)}{\eta} \quad \text{eq. 8}$$

where:

$P_{LP,F}$ is the power of the liquid pump or the fan;

P_E is the total measured electricity input;

η is the assumed efficiency of the liquid pump or the fan;

q is the nominal heat transfer medium flow rate;

Δp_e is the measured external static pressure difference in the heat transfer medium;

Δp_i is the measured internal static pressure difference in the heat transfer medium.

The efficiency η of the fan is considered to be 0,3 by convention for all units. For liquid pumps, η is calculated according to the following formulae:

$$\eta = 0{,}0721 \cdot P_{hyd}^{0{,}3183}, \text{ if } P_{hyd} < 500 \text{ W} \quad \text{eq. 9}$$

$$\eta = 0{,}092 \cdot \ln\left(P_{hyd}\right) - 0{,}0403, \text{ if } P_{hyd} > 500 \text{ W} \quad \text{eq. 10}$$

For the measurement of heating and cooling capacity of water-to-water or brine-to-water units, as well as the cooling capacity of air-to-water units, the measurements are carried out in the steady state condition. This condition is defined as "*(…) when all the measured quantities remain constant without having to alter the set values, for a minimum duration of 1 h, with respect to the tolerances (…) Periodic fluctuations of measured quantities caused by the operation of regulation and control devices are permissible, on condition the mean value of such fluctuations does not exceed the permissible deviations listed (…)*". The tolerances and deviations refer to Table 6 in this document. The respective capacity is calculated as the average value from the recorded temperatures and volume flows.

The measurement procedure of cooling capacity for air-to-water and air-to-air units consists of three periods: preconditioning period, equilibrium period and data collection period.

In the preconditioning period the aimed test conditions should be reached and maintained for at least 10 minutes. This period should preferably end with a defrost cycle. If so, the temperatures and the water flow rates on the indoor heat exchanger should be set not before 20 minutes after the end of the defrost cycle.

In the equilibrium period, steady state conditions according to Table 6 should be maintained for at least one hour. If a defrost cycle occurs during this period, then a transient test procedure will be applied.

Table 6: Permissible deviations from the set values

Measured quantity	Permissible deviations from the set values	
	Arithmetic mean values	Individual measured values
Liquid		
inlet temperature	± 0,2 K	± 0,5 K
outlet temperature	± 0,3 K	± 0,6 K
volume flow	± 1 %	± 2,5 %
static pressure difference	-	± 10 %
Air		
inlet temperature	± 0,3 K	± 10 K
volume flow	± 5 %	± 10 %
static pressure difference	-	± 10 %
Refrigerant		
liquid temperature	± 1 K	± 2 K
saturated vapour temperature	± 0,5 K	± 1 K
Voltage	± 4 %	± 4 %

For the data collection period, two options are possible: steady state or transient test procedures. For the steady state tests a data collection period 70 minutes with a data sampling rate of at least 30 seconds is foreseen. This procedure applies if the value of the quantity *%ΔT* from equation 11 does not exceed 2,5 % during the first 35 minutes of the measurement. If it does exceed 2,5 %, the test must be carried out for transient conditions:

$$\%\Delta T = \left[\frac{\Delta T_i(\tau=0) - \Delta T_i(\tau)}{\Delta T_i(\tau=0)}\right] \cdot 100\,\% \qquad \text{eq. 11}$$

where:

$\Delta T_i(\tau=0)$ is the average temperature difference after first 5 minutes of the measurement;

$\Delta T_i(\tau)$ is the average temperature difference for the entire measurement period;

Table 7: Variations allowed in heating capacity tests when using the transient test procedure

Readings	Variation of values from specified test conditions			
	Arithmetical mean values		Individual readings	
	Interval H	Interval D	Interval H	Interval D
Air temperature entering indoor side				
dry bulb	± 0,6 K	± 1,5 K	± 1,0 K	2,5 K
Air temperature entering outdoor side				
dry bulb (for HX A>5 sqm, x2)	± 0,6 K	± 1,5 K	± 1,0 K	± 5,0 K
wet bulb	± 0,3 K	± 1,0 K	± 0,6 K	-
Inlet water temperature	± 0,2 K	-	± 0,5 K	-
Outlet water temperature	± 0,5 K	-	-	max ± 2 K

If $\%\Delta T$ dose exceed 2,5 % or a defrost cycle occurs during the equilibrium period or during the first 70 minutes of the data collection period, the transient test procedure and tolerances according to

Table 7 will apply. Values marked with H in

Table 7 are to be used for heating intervals, values marked with D for defrosting intervals. The data collection period is extended to either 3 hours or 3 complete defrosting cycles, whatever occurs first.

The heating capacity in this case is calculated as the average value from the recorded temperatures and volume flows for steady state tests.

Besides the rating tests, EN14511 also considers requirements for failure tests, operating range tests, freeze-up test, marking, technical report, technical data sheet and instructions.

Direct expansion heat pumps are covered by EN 15879-1 [22]. Nominal operating conditions for the indoor heat exchanger are comparable to those of EN 14511. Nominal heat source temperature (temperature of the brine bath in which the evaporator loops are submerged) is 4°C.

2.2 ANSI/ASHRAE 37-2009: Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment

ANSI/ASHRAE 37-2009 provides test methods for electrically-driven mechanical-compression unitary air conditioning and heat pumping units. The heat transfer medium on the indoor side must be air, while the outdoor coil can exchange heat with air, liquid, or via evaporative cooling.

The standard defines ways to evaluate the rated cooling or heating capacity of equipment. The efficiency of the equipment is *not* calculated in this standard.

To evaluate the steady state performance of a unit, five test methods are defined:

- Indoor air enthalpy method
- Outdoor air enthalpy method
- Compressor calibration method
- Refrigerant enthalpy method
- Outdoor liquid coil method

For units having less than 40 kW of cooling capacity, two of these methods must be used, and one must be the indoor air enthalpy method. For units of cooling capacity greater than 40 kW, only one method must be used, and it cannot be the outdoor air enthalpy method. Tables are provided for the necessary measurements to be taken for each of the five tests. In addition, acceptable instrumental uncertainties and data sampling intervals are defined.

Flow rates (of air, refrigerant, outdoor liquid, etc.) are all left to the control of the unitary equipment.

Defrost cycles (for heating equipment) are included in the performance of the unit. A complete cycle consists of a heating period and a defrost period, with the heating capacity calculated as the average over that entire period.

2.3 AHRI 320, 325, 330

This group of AHRI standards provide rating conditions for factory made, residential, commercial or industrial electrically driven heat pumps. The standards define provide classifications, definitions, testing, rating and performance requirements, as well as requirements for the marking and nameplate data for the following heat pump types:

- AHRI 320: Water-source heat pumps
- AHRI 325: Ground water-source heat pumps
- AHRI 330: Ground source closed-loop heat pumps

The tests should be performed according to the ANSI/ASHRAE 37 standard. The total cooling and/or heating capacities determined will be the average of the air-enthalpy method and the water coli method, as described in ANSI/ASHRAE 37.

The standard rating conditions are shown in

Table 8.

The standard rating tests are to be carried out at an indoor-side air quantity, delivered against the minimum specified external resistance, provided in a table as a function of the unit's capacity. Lower air-side quantities are allowed, if specified by the manufacturer.

Table 8: Standard rating conditions for AHRI 320, 325 and 330 standards

		Standard rating conditions							
		Cooling				Heating			
		inlet indoor unit	surrounding	outdoor heat exchanger		inlet indoor unit	surrounding	outdoor heat exchanger	
				inlet	outlet			inlet	outlet
AHRI 320		26,7	26,7	29,4	35,0	21,1	21,1	21,1	*
AHRI 325	high temperature	26,7	26,7	21,1	**	21,1	21,1	21,1	**
	low temperature	26,7	26,7	10,0	**	21,1	21,1	10,0	**
AHRI 330		26,7	26,7	25,0	**	21,1	21,1	0,0	*

* as obtained for standard rating conditions - cooling
** specified by the manufacturer

Capacity controlled heat pumps shall be rated at each step of the capacity reduction allowed by the controls. Special rating conditions are provided,

Table 9. Ratings at other operating conditions not specified by the standards may also be published as application rating conditions, as far as the defined test methods have been applied.

Performance requirements according to the standards include the following additional tests a unit has to pass:

- maximum operating conditions test (AHRI 320),
- low-temperature operation test for cooling (AHRI 320),
- insulation efficiency test for cooling (AHRI 320)
- functionality test (no interruptions in the operation and no damage to the equipment) (AHRI 320)

The operating conditions for the additional tests are given in

Table 9.

Table 9: Operating conditions for additional tests within AHRI 320, 325 and 330 standards

	Part-load rating¹ / maximum operating² / insulation efficiency test³ / low-temperature˜ conditions							
	Cooling				Heating			
	inlet indoor unit	surrounding	outdoor heat exchanger inlet	outdoor heat exchanger outlet	inlet indoor unit	surrounding	outdoor heat exchanger inlet	outdoor heat exchanger outlet
AHRI 320¹	26.7	-	23.9	*	21.1	21.1	23.9	*
AHRI 320²	35.0	35.0	35.0	*	26.7	26.7	32.3	*
AHRI 320˜	19.4	-	18.3	*	n.a.	n.a.	n.a.	n.a.
AHRI 320³	26.7	-	26.7	*	n.a.	n.a.	n.a.	n.a.
AHRI 325²	35.0	35.0	23.9	***	26.7	26.7	23.9	****
AHRI 325³	26.7	-	10.0	* (LT)	n.a.	n.a.	n.a.	n.a.
AHRI 330¹	26.7	-	21.1	**	21.1	21.1	5.0	*
AHRI 330²	35.0	35.0	37.8	*	26.7	26.7	23.9	*

* as obtained for standard rating conditions - cooling
** specified by the manufacturer
*** maintained at 75% of flow rate for high temperature cooling test
**** maintained at 110% of flow rate for high temperature heat test

The standards provide definitions for the efficiency figures of the heat pump unit:

- Coefficient of performance (COP) – the ratio of the heating capacity in watts to the power input in watts;
- Energy efficiency ratio (EER) – the ratio of the cooling capacity in Btu/h to the power input in watts.

The heating and cooling capacities are the net values, including the effects of circulating-fan heat, excluding supplementary resistance heat. The total power input includes the compressor, the fans and other items and controls included as a part of the model number. For ground-water source (AHRI 325) and ground source (AHRI 330) heat pumps, a pump penalty has to be considered.

The pump penalty for ground-water source heat pumps takes into account the power consumption of the water pump, according to equation 12:

$$PP = WF \cdot ((PP_B \cdot \Delta P) + 65) \qquad \text{eq. 12}$$

where:

PP is the total pumping penalty in watts;

WF is the water flow rate in liter per second;

PP_B is the basic pumping penalty, in kPa (values provided as a function of the water flow rate;

ΔP is the unit water pressure drop in kPa.

For ground source heat pumps (AHRI 330), the penalty for the water (or fluid) pump is 4.2 watts per l/s per kPa, which has to be added to the measured power input. This corresponds to about 25 % pump efficiency. The fluid used for all the tests shall be based on 15 % solution (weight) of sodium chloride in water.

2.4 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 current version EN 14825:2012 was drafted by the WG7 of the CEN TC113. It was published in April 2012. The revision of the document will, however, start immediately.

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: an average climate, one cold and one warm climate. Corresponding climate data is provided in an Annex to the document.

The standard covers air-to-air, water (brine)-to-air, air-to-water and water (brine)-to-water units. Only the latter two will be discussed.

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, which is well described in e.g. [23].

SEER and SCOP are calculated according to:

$$SEER = \frac{Q_{CE}}{\frac{Q_{CE}}{SEER_{on}} + H_{TO} \cdot P_{TO} + H_{SB} \cdot P_{SB} + H_{CK} \cdot P_{CK} + H_{OFF} \cdot P_{OFF}} \qquad \text{eq. 13}$$

$$SCOP = \frac{Q_{HE}}{\frac{Q_{HE}}{SEER_{on}} + H_{TO} \cdot P_{TO} + H_{SB} \cdot P_{SB} + H_{CK} \cdot P_{CK} + H_{OFF} \cdot P_{OFF}} \qquad \text{eq. 14}$$

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;

H_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. These periods are given in the informative Annex C of the standard, but only for air-to-air and water-to-air units. It is unclear however if the same values are to be used also for other unit types and, if not, which values are to be used. As the standard is currently under revision, this issue may be addressed in the next version of the document.

The $SEER_{on}$ and $SCOP_{on}$ are determined as follows:

$$SEER_{on} = \frac{\sum_{j=1}^{n} h_j \cdot P_C(T_j)}{\sum_{j=1}^{n} h_j \cdot \left(\frac{P_C(T_j)}{EER(T_j)} \right)} \qquad \text{eq. 15}$$

$$SCOP_{on} = \frac{\sum_{j=1}^{n} h_j \cdot P_H(T_j)}{\sum_{j=1}^{n} h_j \cdot \left(\frac{P_H(T_j) - elbu(T_j)}{COP(T_j)} + elbu(T_j) \right)} \qquad \text{eq. 16}$$

where

T_j is the bin temperature;

j is the bin number;

n is the amount of bins;

P_C is the cooling demand of the building for the corresponding temperature T_j;

P_H is the heating demand of the building for the corresponding temperature T_j;

h_j is the number of bin hours in which a certain temperature T_j occurs;

$EER(T_j)$ is the EER value of the unit for T_j;

$COP(T_j)$ is the COP value of the unit for T_j;

$elbu(T_j)$ is the capacity of an electrical heating back up unit for T_j.

If the unit cannot provide sufficient heating capacity for operation points below the bivalent point (part load condition F), the COP values are corrected by introducing an electrical backup heater with the COP of 1.

For the calculation of the $SEER_{on}$ and $SCOP_{on}$, a unit has to be tested for a certain number of part load conditions according to

Table 10 for cooling and Table 11 for heating. The part load ratios (PLR) given in the tables are calculated according to the following equation:

$$PLR = \frac{T_i - 16}{T_{design} - 16} \qquad \text{eq. 17}$$

The power consumption is measured by setting the thermostat to a value which triggers shutting down of the compressor. The auxiliary power consumption is then 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 after the B temperature condition test. Finally, the off mode test has to take place after the standby test by switching the unit into the off mode while remaining plugged.

The EER and COP values for the part load conditions A, B, C and D (also E and F) have to be provided. If measurements for these conditions are available, the EER and COP values for other temperature bins are generally interpolated between or extrapolated from these values. The cooling and heating demand are calculated from the respective full load value multiplied by the respective part load ratio of the corresponding bin.

Table 10: Part load conditions for the cooling mode

	Part load ratio [%]	Outdoor heat exchanger			Indoor heat exchanger		
		air-to-water	water (brine)-to-water		Fan coil		Radiant cooling
		dry bulb T [°C]	Cooling tower	Groound coupled	Fixed outlet	Variable outlet	
A	100	35	30 / 35	10 / 15	12 / 7	12 / 7	23 / 18
B	74	30	26 / *	10 / *	* / 7	* / 8,5	* / 18
C	47	25	22 / *	10 / *	* / 7	* / 10	* / 18
D	21	20	18 / *	10 / *	* / 7	* / 11,5	* / 18
* water flow rate as determined in the full load test (A)							

For load conditions above the condition A and below the condition D, the same values of EER as for A or D are used, respectively.

Table 11: Part load conditions for the heating mode (average climate)

	Part load ratio [%]	Outdoor heat exchanger			Indoor heat exchanger		
		air-to-water	water (brine)-to-water				
		dry (wet) bulb T [°C]	Ground water	Brine	Application	Fixed outlet	Variable outlet
A	88	-7 (-8)	10 / *	0 / *	low T	* / 35	* / 34
					medium T	* / 45	* / 43
					high T	* / 55	* / 52
B	54	2 (1)	10 / *	0 / *	low T	* / 35	* / 29
					medium T	* / 45	* / 36
					high T	* / 55	* / 43
C	35	7 (6)	10 / *	0 / *	low T	* / 35	* / 27
					medium T	* / 45	* / 33
					high T	* / 55	* / 38
D	15	12 (11)	10 / *	0 / *	low T	* / 35	* / 25
					medium T	* / 45	* / 29
					high T	* / 55	* / 33
E	f (TOL)	TOL			low T	* / 35	* / 34 - (-7-TOL) / (-7-2) x (34-29)
					medium T	* / 45	* / 43 - (-7-TOL) / (-7-2) x (43-36)
					high T	* / 55	* / 52 - (-7-TOL) / (-7-2) x (52-43)
F	f ($T_{bivalent}$)	$T_{bivalent}$			low T	* / 35	* / 34 - (-7-$T_{bivalent}$) / (-7-2) x (34-29)
					medium T	* / 45	* / 43 - (-7-$T_{bivalent}$) / (-7-2) x (43-36)
					high T	* / 55	* / 52 - (-7-$T_{bivalent}$) / (-7-2) x (52-43)
* water flow rate as determined in the full load test (A)							

The bivalent temperature $T_{bivalent}$ is to be set at 2°C or less for the average climate. The operational limit temperature TOL is the lowest outdoor temperature at which the heat pump can still deliver heating capacity and is stated by the manufacturer.

In case that a certain part load condition cannot be reached as stated in

Table 10 and Table 11, for example for units with on/off control, the following procedures are to be used (for air-to-water, water-to-water and brine-to-water units):

- Fixed capacity units: The EER or the COP are calculated from the following equations:

$$EER_{B,C,D} = EER_{DC} \cdot \left(\frac{CR}{C_c \cdot CR + (1 - C_c)} \right) \qquad \text{eq. 18}$$

$$COP_{A,B,C,D} = COP_{DC} \cdot \left(\frac{CR}{C_c \cdot CR + (1 - C_c)} \right) \qquad \text{eq. 19}$$

where

EER_{DC} is the EER corresponding to the declared capacity (DC) of the unit at the given temperature conditions for B, C or D;

COP_{DC} is the COP corresponding to the declared capacity (DC) of the unit at the given temperature conditions for B, C or D;

C_c is the degradation coefficient;

CR is the capacity ratio;

The degradation coefficient takes into account the electricity consumption of the unit while the compressor is switched off. It is calculated as

$$C_c = 1 - \frac{\text{measured power in compressor off state}}{\text{full capacity at part load condition}} \qquad \text{eq. 20}$$

If it cannot be determined, the default value of 0,9 should be used.

CR is equal to the heating or cooling demand over the declared capacity of the unit at the same temperature conditions.

The outlet temperatures of the indoor heat exchanger, as indicated in

Table 10 for cooling and Table 11 for heating should correspond to the time averaged outlet temperature according to equation 21:

$$t_{outlet,average} = t_{inlet,full\ load\ test} + (t_{outlet,full\ load\ test} - t_{inlet,full\ load\ test}) \cdot CR \qquad \text{eq. 21}$$

- **Staged capacity units**: The EER or the COP are calculated by the interpolation from the values on either side of the control step of the unit, if the given value (A, B, C or D) cannot be reached within ±3 %. If the smallest control step is higher than the required cooling or heating demand, the EER or the COP are calculated as for fixed capacity units.
- **Variable capacity control units**: The capacity for the given part load condition should be reached within ±5 % from the stated value. If this is not the case, then the same procedure as for the staged capacity units should be applied.

For fixed capacity units, an alternative test method for part load conditions is given. Thereafter, the test can be performed by obtaining the relevant temperature as a time averaged value over the testing period. It is however unclear which tolerances, data acquisition times etc. should be applied as such a test is not defined in EN14511.

The standard also defines measurement uncertainties related to the respective heat or cooling load, which are independent of the measurement uncertainties defined in EN 14511.

Finally, informative Annexes A and B give calculation examples for SEER, $SEER_{on}$, $SCOP_{on}$ and $SCOP_{net}$.

2.5 EN 16147: Heat pumps with electrically driven compressors - Testing and requirements for marking for domestic hot water units

The current version of the standard was issued in April 2011. It specifies methods for testing and rating of heat pumps connected to or including a domestic hot water storage tank. It superseded the standard EN 255-3.

Test conditions are similar to those of EN 14511, regarding the heat source and ambient temperatures for the heat pump,

Table 12.

Table 12: Test conditions within EN 16147:2011

Type of heat source	Temperature in °C (wet bulb)	Ambient temperature for heat pump in °C	Ambient temperature for storage tank in °C
Outside air, indoor installation	7 (6)	15 - 30	20
Outside air, outdoor installation	7 (6)	heat source temp.	20
Indoor air	15 (12)	heat source temp.	15
Exhaust air	20 (12)	15 - 30	20
Water	10 / 7	15 - 30	20
Brine	0 / -3	15 - 30	20
Direct evaporation	4	15 - 30	20

The uncertainties of measurement and the permissible deviations from the set values are also similar to EN 14511 and will not be given here.

The testing procedure includes the following tests:

- Heating up period – determination of the necessary time to heat up the storage from an initial state until the first turn-off of the compressor by the controls. The heating up time and the electricity consumption are measured;
- Determination of standby power input – power consumption in the standby mode is measured;
- Energy consumption and COP for reference tapping cycles – five tapping cycles according to the energy content of the hot water and type hot water usage are defined. The tapping cycles consist of a series of different types of delivery, which are provided as energy quantities, minimum temperature levels above the cold water temperature and hot water flow rates to be maintained. The consumed electrical energy is measured and corrected by the energy consumptions of fans or liquid pumps, similarly to the procedure described in EN 14511. The coefficient of performance is the ratio of the total useful heat delivered during the whole tapping cycle and the total (corrected) energy consumption during the tapping cycle.
- Determination of a reference hot water temperature and the maximum quantity of usable hot water in a single tapping – the reference hot water temperature is determined by measuring the outlet water temperature from the tank θ_{WH} after the compressor has switched off at the end of the last measurement period for the tapping cycles. The measurement lasts until the outlet temperature falls below 40°C, time t_{40}. The reference hot water temperature θ'_{WH} is calculated from equation. 22:

$$\theta'_{WH} = \frac{1}{t_{40}} \cdot \int_0^{t_{40}} \theta_{WH}(t) \cdot dt \qquad \text{eq. 22}$$

  The maximum amount of usable hot water is also determined for the reference temperature difference of 30 K.
- Temperature operating range test – the tests are performed with the minimal and the maximal heat source temperatures, indicated by the manufacturer.
- Safety tests – include shutting off the heat transfer medium flows, cut-off of the power supply and condensate draining.

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.

2.6 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

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. Due to differences in the practical annual temperature profile between the ground and ground water system on one hand and ambient air on the other, the guideline treats these two cases separately.

Heating application: The seasonal performance factor for space heating is calculated from equation 23:

$$\beta_h = \frac{\varepsilon_N \cdot F_\vartheta \cdot F_{\Delta\vartheta}}{F_P} \qquad \text{eq. 23}$$

where

ε_N is the COP of the heat pump according to EN 14511 or EN 255-3 for nominal conditions;

$F_{\Delta\vartheta}$ is the correction factor accounting for deviations in the temperature difference at the condenser between the measurement and operation;

F_ϑ is the correction factor accounting for different operating conditions;

F_P is the correction factor for the energy consumption of the heat source pump.

F_ϑ is given in table form as a function of the ground/water temperature and maximum supply temperature.

$F_{\Delta\vartheta}$ is given in table form as a function of the temperature difference at the condenser during laboratory measurement and at the design point of the heating system.

For F_P, recommendations for different source systems and capacity ranges are given. Separate recommendations are given for water source systems using intermediate heat exchangers in case the water quality is poor.

For air-source systems, the SPF is calculated from equation 24 (according to DIN V 4701-10 [24]):

$$\beta_h = (\varepsilon_{N1} \cdot F_{\vartheta 1} + \varepsilon_{N2} \cdot F_{\vartheta 2} + \varepsilon_{N3} \cdot F_{\vartheta 3}) \cdot F_{\Delta\vartheta} \qquad \text{eq. 24}$$

where

ε_{Ni} are the COPs of the heat pump according to EN 14511 or EN 255-3 for nominal conditions with different source temperatures: -7, 2 and 10°C;

$F_{\Delta\vartheta}$ is the correction factor accounting for deviations in the temperature difference at the condenser between the measurement and operation;

$F_{\vartheta i}$ are the correction factors accounting for different operating conditions at three different air temperatures, as stated above.

For $F_{\Delta\vartheta}$, the same tabular values apply as for ground coupled and water source systems.

Values for $F_{\vartheta i}$ are provided as tables for three different heating limit temperatures: 15, 12 and 10°C. The parameter is a function of the standard outdoor temperature and the maximum supply temperature.

DHW application: The seasonal performance factor for DHW application is denominated with β_W.

For ground and ground water source heat pumps, equation analogue to equation 24 is used. However, different ground or ground water dependent values for the correction factor F_ϑ are given in separate tables.

For ambient air-source heat pumps, an analogue procedure to the one described for the heating application is used. For heat pump systems using cellar air as heat source, the seasonal performance factor is calculated from equation 25:

$$\beta_h = \varepsilon_N \cdot F_1 \cdot 0{,}9 \qquad \text{eq. 25}$$

where

ε_N is the COPs of the heat pump according EN 255-3 for an air temperature of 15°C and water being heated form 15 to 50°C;

F_1 is the correction factor accounting for different hot water temperatures during laboratory measurement.

F_1 is given for water temperatures of 45, 50, 55, 60 and 65°C, linear interpolation for other values is allowed.

For bivalent operation, a table containing the demand coverage α by the heat pump is provided. It is a function of the bivalent point ϑ_{Biv} and the operation mode of the system (alternate, parallel, partly parallel).

The overall seasonal performance factor of the heat pump is calculated by weighting the energy demand for heating and domestic hot water:

$$\beta_{WP} = \frac{1}{x \cdot a / \beta_h + y \cdot a / \beta_w + 1 - \alpha} \qquad \text{eq. 26}$$

If heating or DHW is not provided by the heat pump, the respective term is not considered.

In the final chapter of the document, three calculation examples for three different heat sources are provided.

2.7 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 standard was elaborated by the CEN/TC 228 „Heating systems in buildings“. The current version was published in June 2008.

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). An overview of the considered heat sources and heat distribution systems is given in Table 13.

Table 13: Part load conditions for the heating mode (average climate)

heat source	heat distribution
air (outdoor and exhaust)	air
ground coupled (direct and indirect)	water
water (surface and ground)	direct condensation

The output data of the described calculations are:

- Driving energy of the system;
- Total thermal losses of the system;
- Total recoverable thermal losses of the system;
- Total auxiliary energy consumption.

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. To use this method, a national annex is needed.
- Calculation based on the temperature bin method, which is explained in the standard itself or e.g. in [23]

The standard gives only the calculation methods, in most cases it does not prescribe which input data to use; in some cases, however, recommendations are provided.

The calculation of the SPF with the temperature bin method is performed following the defined ten steps:

1. Determination of energy requirement of every single bin;
2. Correction of steady state heating capacity / COP for bin source and sink temperature operating conditions;
3. Correction of COP for part load operation, if required;
4. Calculation of generation subsystem thermal losses;
5. Determination of back-up energy requirements of the single bins;
6. Calculation of the running time of the heat pump in different operation modes;
7. Calculation of auxiliary energy input;
8. Calculation of generation subsystem thermal loss recoverable for space heating;
9. Calculation of the total driving energy input to cover the requirements;
10. Summary of resulting and optional output values.

The cumulative heating degree hours should be given in a national annex or available from national standards.

The heating energy demand of the heating distribution system should be calculated according to EN 15316-2-3 [25]. 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 <u>for the nominal capacity</u> should be determined according to a European standard. If possible, all relevant operation conditions should be considered or at least the operation conditions given in the standard. If the mass flows on the heat source or heat sink side differ from the design operating conditions, a correction by interpolation or extrapolation is possible.

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 exergetic efficiency can be performed and is described in an informative Annex.

Regarding the heat source, the following temperatures are to be used:

- For air-source heat pumps, the outside air temperature of the bin is to be used;
- For an exhaust air heat pump without heat recovery, the indoor temperature is the source temperature. If a heat recovery is included, combined test results for the heat pump and for the heat recovery unit can be used. Alternatively, an evaluation of the supply temperature according to the temperature variation coefficient of the heat recovery, e.g. according to EN 308 [26];
- For ground coupled or water heat pumps, values from national annexes or standards should be used. If none available, an example is given in an informative annex.

For the DHW, results from the measurements according to EN 255-3 [27] are to be used. Because of oscillating source temperatures, a correction has to be performed on the bases 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 mods.

Regarding <u>part load operation</u> 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.

<u>Total thermal losses</u> include the losses within the energy generation subsystem, thermal losses from all storages within the system as well as losses in the primary circulation pumps. These losses are accounted for both for the operation and the stand-by times. Some of the losses are recoverable, such as the losses to the heated ambient or the thermal losses of auxiliary components to heat transfer media. These recoverable losses are calculated and added to the energy output of the overall system.

If no storage is integrated in the heat pump casing, the <u>generator heat losses</u> for the heating operation are not considered if no national standards are available. For engine driven heat pumps, the thermal losses of the engine have to be calculated, but no specific method is given – only references to other standards and possible calculation methods.

If an internal or an external storage is part of the system, the losses to the ambient have to be calculated for every temperature bin. The stand-by heat losses are either given from the storage tests or standard values from an informative Annex are used. The mean storage temperature is obtained from the system control settings. If the temperature in the storage varies according to the heating load, the mean temperature is calculated. For the DHW storage, the same method is applied, only different temperatures according to the regulations have to be taken into account.

For the thermal losses of the primary circulation piping EN 15316-2-3 and EN 15316-3-3 [28] are to be used.

In polyvalent systems, the back-up heating is considered for two reasons:

- If temperatures in the distribution system are needed which are higher than the temperature operation limit of the heat pump;

- The heat pump was not dimensioned to cover the full heating and/or DHW load. In this case two calculation methods are described.

The simplified method is based on the evaluation of the cumulative temperature frequency and the bivalent or low temperature shut-off point. Energy fractions for the heat pump and for the back-up system are obtained. The input data for the calculation are the bivalent or the shut-off point and the operation mode – alternative, parallel or semi-parallel. In all cases, the fraction of the energy delivered by the back-up unit is calculated from the ratio of the area under the cumulative temperature frequency curve representing the energy which is not delivered by the heat pump to the total heating energy needs.

The detailed calculation is based on the evaluation of the running time for 1 K bins. The detailed calculation takes also into account also the specific controller settings.

The operation time of the heat pump per bin is calculated from the produced heating energy and the respective heat pump capacity for the operating condition within a single bin.

While the estimation of the running time is quite straight forward in cases where the heat pump produces heating energy and DHW in clearly separated operation cycles, it can be quite difficult to differentiate between these two running times for the simultaneous operation mode, in which both are produced at the same time. The maximum running time in the simultaneous mode can be calculated from the minimum running time needed in both modes. This time can be corrected for different controller settings. The respective energies produced in this operation mode are calculated. From an energy balance, the fractions of the heating and DHW energies can be calculated. Finally, from these energies, the respective running time can be estimated.

For the calculation of the auxiliary energy consumption, the energy consumption of all system components should be considered. Energy already included in the testing standards has to be taken into consideration.

The energy input to the heat pump is calculated as the sum of the energy inputs for every bin, derived from the delivered heat and the heat pump efficiency under the bin operating conditions. Similar calculation is performed for the energy input to the back-up unit.

Finally, two seasonal performance factors (SPF) can be calculated:

- SPF of the generation subsystem during operation, including the heat pump itself and the back-up heater
- SPF of the overall system, including all auxiliary energy consumptions (e.g. for the heat source system and for stand-by operation)

Comprehensive information on various calculation procedures, as well as default values for different parameters used for the calculations is provided in the Annexes.

3 Solar thermal collectors

3.1 ISO 9806: Test methods for solar collectors

The IS0 9806 standard consists of the following parts, under the general title test methods for solar collectors:

- Part 1: Thermal performance of glazed liquid heating collectors including pressure drop
- Part 2: Qualification test procedures
- Part 3: Thermal performance of unglazed liquid heating collectors (sensible heat transfer only) including pressure drop

The IS0 9806 Part 1 provides test methods and calculation procedures for determining the steady-state and quasi-steady-state thermal performance of solar collectors. It contains methods for conducting tests outdoors under natural solar irradiance and for conducting tests indoors under simulated solar irradiance. It is not applicable to tracking concentrating collectors, only gives some information in the annex for special biaxial incident angle modifiers of parabolic-trough concentrating collectors.

The testing conditions for the thermal performance are:

- Collector inlet temperature variation less than ± 0.1 K during the specified time before and during each test
- Minimum difference temperature of 1.5K
- Ambient temperature ta must vary less than ± 1.0 K
- G> 800 W.m-2 and DG no more than 3 %
- G measured with an incidence angle less than 30º
- Flow rate variation less than 1 %
- Wind speed between 2 and 4 m/s
- 4 equally spaced values of inlet fluid temperature
- The spectrum-weighted value of the transmittance-absorptance product at normal incidence must not differ more than 3% from the

value of the transmittance-absorptance product calculated using the standard spectrum.

The ISO 9806 Part 2 is dedicated to the durability tests and applies to all types of solar collectors, including integral collector storage systems but excluding tracking concentrating collectors. It describes the following testing procedures:

- Internal pressure tests for absorbers
- High-temperature resistance test
- Exposure test
- External thermal shock test
- Internal thermal shock test for liquid-heating collectors
- Rain penetration test
- Freezing test
- Impact resistance test (optional)
- Final inspection

The ISO 9806 Part 3 establishes methods for determining the thermal performance of unglazed liquid heating solar collectors. It contains methods for conducting tests outdoors under natural solar irradiance and simulated wind and for conducting tests indoors under simulated solar irradiance and wind. It is not applicable to those collectors in which a thermal storage unit is an integral part of the collector and those to collectors in which the heat transfer fluid can change phase, nor is it applicable to collectors affected by condensation of water vapour from the ambient air.

3.2 EN 12975-2: Thermal solar systems and components. Solar collectors. Test methods

The most commonly used methodology is the steady state according to EN 12975-2, Section 6.1, considered applicable to all stationary collectors, e.g., glazed, unglazed and evacuated tube collectors. It allows two different test methods to determine the thermal performance: the steady-state (SS) and the quasi-dynamic (QD).

This standard is not applicable to those collectors in which a thermal storage unit is an integral part of the collector and also is not applicable for qualification tests to tracking concentrating collectors, only the thermal performance test as given in clause 6.3 (quasi-dynamic testing) is applicable to most concentrating collector designs, from stationary non-imaging concentrators as CPCs to high concentrating tracking designs.

The collector is tested over its operating temperature range in order to determine its efficiency characteristic. Data points are obtained for at least four water inlet temperatures evenly spaced over the collector operating temperature range. Especially, one inlet temperature shall be selected such that the mean temperature in the collector lies within ± 3 K of the ambient air temperature, in order to obtain an accurate determination of η_0. At least four independent data points shall be obtained for each inlet temperature.

The following testing conditions must be considered for both testing methods:

- Minimum temperature difference of 1 K
- Ambient temperature ta must vary less than ± 1.0 K (indoor) or ± 1.5 K (outdoor)
- Flow rate variation less than 1 %
- 4 equally spaced values of inlet fluid temperature
- For the steady-state test method:
  - Collector inlet temperature variation less than ± 0.1ºC during specified period
  - DG no more than 50 Wm^{-2}
  - G> 700 $W.m^{-2}$ and G_d/G < 30 %
  - G measured with an incidence angle less than 20º or for an angle which doesn't change the incidence angle modifier more than 2 %
- For the steady-state test method:
  - Collector inlet temperature variation less than ± 1ºC during the specified period
  - G> 300 Wm^{-2}
  - G measured with an incidence angle from 60º to 20º or for an angle which doesn't change incidence angle modifier more than 2 %
- The efficiency curve model is defined for the steady state method as:

$$\frac{Q}{A} = F'(\tau\alpha)_{en} G_b - c_1(t_m - t_a) - c_2(t_m - t_a)^2 \quad \text{eq. 27}$$

And for the quasi-dynamic state method as:

$$\frac{Q}{A} = F'(\tau\alpha)_{en} K_{\theta_b}(\theta) G_b + F'(\tau\alpha)_{en} K_{\theta_d} G_d - c_1(t_m - t_a) - c_2(t_m - t_a)^2$$
$$- c_3 u(t_m - t_a) - c_4(E_L - \sigma T_a^4) - c_5 \, {dT_m}/{dt} - c_6 uG \quad \text{eq. 28}$$

The modelling of the long-wave irradiance dependence of the collector, is made in a similar way as described in the ISO 9806-3, for testing of unglazed collectors, but here it is treated as a heat loss term. The coefficients in the previous equation are explained below:

c_1 heat loss coefficient at $(t_m$-$t_a)$= 0 is modelled as $F'U_0$ [W/(m^2 K)]

c_2 temperature dependence of the heat losses, equal to $F'U_1$ [W/(m^2 K^2)]

c_3 wind speed dependence of the heat losses, equal to $F'U_u$ [J/(m^3 K)]

c_4 long-wave irradiance dependence of the heat losses, equal to F'_e [–]

c_5 effective thermal capacitance, equal to $(mC)_e$ [J/(m^2 K)]

c_6 wind dependence of the zero loss efficiency, a collector constant [s/m]

$K_{\theta d}$ incidence angle modifier for diffuse radiation, a collector constant [–]

$K_{\theta b}$ (θ) incidence angle modifier (IAM) for direct (beam) radiation [–]

For the steady-state test method when using a solar simulator and when testing collectors containing spectrally selective absorbers or covers, it is mandatory to check the effect of the difference in spectrum on the effective transmittance-absorptance product (ta) for the collector. This effect should be correct if the difference between the product *(ta)$_e$* under the simulator and under the optical air mass 1.5 solar radiation spectrum differs by more than ± 1 %.

The range for calculating this product is detailed (the measurement of the solar simulator spectral qualities shall be performed on the collector plane over the wavelength range of 0.3 mm to 3 mm and shall be determined in bandwidths of 0.1 mm or smaller. But it does not describe how to measure the optical properties. An outdoor check of the optical efficiency h_0 could be enough to control this effect.

According to this standard, the incidence angle modifier can be measured in steady state conditions with an incidence angle q ± 2.5º. The definition of the incidence angle modifier is in this case:

$$K(\theta) = \frac{\eta(\theta)}{\eta(0^\circ)} \qquad \text{eq. 29}$$

The measurement is usually done at an angle q = 50 ±2.5º. For conventional flat plate collectors, this angle will be sufficient. For some collectors with unusual optical performance characteristics, or if it is required for a system simulation, angles of 20°, 40°, 60° and others have to be determined.

It can also be performed by the quasi-dynamic test method obtaining the parameter b_0 assuming the following equation for the incidence angle modifier, mainly for the flat-plate collector, as described in e.g. ASHRAE 93-77 (see Chapter 3.3).

$$K_{\theta b}(\theta) = 1 - b_0\left[\left(\frac{1}{\cos\theta}\right) - 1\right] \qquad \text{eq. 30}$$

For the asymmetrical collectors there is no equation but it is required to measure at different incidence angles (at least 20º, 40º, 60º). For evacuated tubes of CPC collectors, the incidence angle modifier is measured on the longitudinal and transversal planes separately and assuming that: $K_\theta(\theta) = K_{\theta L} K_{\theta T}$ and using the following correlation:

$$\tan^2(\theta) = \tan^2(\theta_L) + \tan^2(\theta_T) \quad \text{eq. 31}$$

The standard durability testing procedure includes the following tests:

- Internal pressure
- High-temperature resistance
- Exposure
- External thermal
- Internal thermal
- Rain penetration
- Freeze resistance
- Internal pressure (retest)
- Mechanical load
- Impact resistance
- Final inspection

3.3 ASHRAE 93: Methods of Testing to Determine the Thermal Performance of Solar Collectors

This standard is for a steady state performance test and their thermal performance testing conditions are:

- Ambient temperature (ta) less than 30ºC;
- G > 790 Wm^{-2} and Gd no more than 20 %;
- The efficiency is defined as:

$$\eta_g = \frac{\int_{T_1}^{T_2} \dot{m} c_p (t_{f,e} - t_{f,i})\, dT}{A_g \int_{T_1}^{T_2} G\, dT} \quad \text{eq. 32}$$

This standard gives a procedure for determining the collector incident angle modifier for non-concentrating, stationary concentrating and for single-axis tracking collectors. It gives the following computation method for the collector incident angle modifier of non-concentrating collectors:

$$K_{\alpha\tau} = \frac{\eta_g}{(A_a/A_g) F_R (\tau\alpha)_{e,n}} \quad \text{eq. 33}$$

For concentrating collectors:

$$K_{\alpha\tau} = \frac{\eta_g}{(A_a/A_g)F_R[(\tau\alpha)_e\rho\gamma]_n} \qquad \text{eq. 34}$$

For a collector with isotropic behaviour:

$$K_\theta = 1 - b_0\left[\left(\frac{1}{\cos\theta}\right) - 1\right] \qquad \text{eq. 35}$$

4 Solar thermal systems

Systems and system components, other than collectors, are at very different stages in the standardization process. There is progress for systems in Europe, with European standards series covering factory made systems (EN12976) and custom built systems (EN 12977). The latest standards also include components, such as stores and controllers.

Factory made solar heating system are considered to be systems with one trade name, which are sold as a package and have a fixed system configuration. According to EN 12976, these systems are tested and assessed as a whole.

Custom built systems are subdivided into large and small systems:

- Large custom built systems are designed for a certain application. The choice of components, dimensioning and system configuration is in general unique;
- Small custom built systems are offered by a company as a prefabricated system design. All components and configurations are already specified.

The standards apply for the following system types:

- Factory Made Solar Heating Systems (EN 12976): Integral collector-storage systems for domestic hot water preparation, thermosyphon systems for domestic hot water preparation, Forced-circulation systems as batch product with fixed configuration for domestic hot water preparation;
- Custom Built Solar Heating Systems (EN 12977): Forced-circulation systems for hot water preparation and/or space heating, assembled using components and configurations described in a documentation file (mostly small systems), Uniquely designed and assembled systems for hot water preparation and/or space heating (mostly large systems).

4.1 EN 12976-2: Thermal solar systems and components - Factory made systems - Part 2: Test methods

A complete system test according to EN 12976-2 includes the following tests and inspections:

- freeze resistance;
- overheating protection;
- pressure resistance;
- water contamination;
- lightning protection;
- safety equipment;
- labelling;
- thermal performance characterisation;
- ability of solar-plus systems to cover the heat demand;
- reverse flow protection;
- electrical safety.

A short description of the standard within this document will be limited to the thermal performance characterisation.

Two test procedures for the assessment of the system's thermal performance can be used:

- CSTG (Complete System Testing Group) method can be used for "solar only" or "preheat sytems", as described in ISO 9459-2;
- DST (Dynamic System Testing) method can be applied to all types of systems, as described in ISO 9459-5.

Both methods are comparable and the conversion factors a and σ_a (equation 36) for different system types have been established [29]:

$$Q_{DST} = (a \pm \sigma_a)\, Q_{CSTG} \qquad \text{eq. 36}$$

The conversion values differ for different system configurations. However, parameter a ranges from 1,004 to 1,056 with standard deviation of less than 0,005 in all cases.

The performance indicators are specified only for DHW preparation. The following performance indicators (Table 14) have to be calculated for different system types using the reference conditions.

Table 14: Performance indicators for reporting

"solar-plus-supplementary"	"solar only" and "preheat"
net auxiliary energy demand, $Q_{aux,net}$	heat delivered by the solar system, Q_L
parasitic energy, Q_{par}	
Fractional energy saving f_{sav} (only in EN 12977)	solar fraction f_{sol} (as in EN ISO 9488)

4.1.1 CSTG Method – ISO 9459-2: Outdoor test methods for system performance characterization and yearly performance prediction of solar-only systems

The performance test for solar-only systems according to ISO 9459-2 is a 'black-box' procedure which produces a family of 'input-output' characteristics for a system. The test results may be used directly with daily mean values of local solar irradiation, ambient air temperature and cold-water temperature data to predict annual system performance. The current valid version of the standard is ISO 9459-2:1995.

The test procedure according to ISO 9459-2 includes a series of one-day outdoor tests of the complete system which are independent of each other. Only one draw-off at the end of the day is applied within every test. Every test begins with a preconditioned storage i.e. with a uniform, constant temperature at the entire volume. In total, the results of six measurements (days) have to be obtained: Four days with approximately the same values of the temperature difference between the average ambient air temperature and the temperature of the cold water inlet from the mains, with daily accumulated solar irradiation values evenly distributed over the range of 8 to 25 MJ/m^2; two measurements for days with the temperature difference between ambient air and mains water supply at least 9 K above or below the values from the first four days.

For the daily measurements, the system should operate 12 hours continuously, 6 hours before and 6 hours after the solar noon. At the end of the test, the water from the storage tank is drawn-off at a constant flowrate of 600 ± 50 l/h and a draw-off temperature profile over time is constructed.

The results from the tests consist of the daily energy input-output diagrams obtained from the equation 37:

$$Q = a_1 \cdot H + a_2 \cdot \left(t_{a(day)} - t_{main}\right) + a_3 \qquad \text{eq. 37}$$

The coefficients a_1, a_2 and a_3 are obtained using the least-square fitting method. Q is the net solar energy gain determined by the draw-off from the storage, as described. H represents the irradiation on the collector plane.

Additionally, tests to determine the degree of mixing in the storage tank during draw-offs, the heat loss coefficient of the storage tank and the overnight heat loss have to be performed. An optional test with an intermediate draw-off is also described.

Using the test results, which are presented in the form of system performance characteristics independent of the climatic conditions under which they were obtained, the method is able to predict long term performance of the system for various values of:

- solar irradiation;
- ambient air temperature;
- mains cold water temperature;
- load volume;

- hot water demand temperature.

The system performance is calculated for each day for given climatic data and the hot water consumption. It can take the effect of heat carry-over from one day to the other due to small amount of energy drawn-off, as well as overnight losses.

The following data are required for the calculation:

- Test data:
  - the input-output characteristics obtained from equation 37;
  - the draw-off temperature profiles for different irradiance ranges;
  - the mixing draw-off profile expressed as a function of volume;
  - the heat loss coefficient of the storage tank.
- Climatic data:
  - the daily solar irradiation on the collector plane;
  - the average ambient air temperature for the test period;
  - the average ambient air temperature during the night.
- System usage data:
  - the volume of the daily hot water consumption or the minimum useful temperature limit;
  - the cold water inlet temperature for each day.

The method is designed to deliver prediction about long term system performance with an accuracy of about ± 5 %.

4.1.2 DST Method – ISO 9459-5: System performance characterization by means of whole-system tests and computer simulation

ISO 9459-5 describes a procedure for dynamic testing of complete systems to determine system parameters for use in the “Dynamic System Testing Program” – proprietary software validated on a range of different systems. The currently valid version is ISO 9459-5:2007.

It specifies a method for outdoor laboratory testing of solar domestic hot-water (SDHW) systems which, however, can also be applied for in-situ tests and for indoor tests by specifying appropriate draw-off profiles and irradiance profiles for indoor measurements. The system performance is characterised by means of whole-system tests using a 'black-box' approach, i.e. no measurements on the system components or inside the system are necessary. As the CSTG, this method may be used with hourly values of local solar irradiation, ambient air temperature and cold-water temperature data to predict annual system performance. Implementation of the software requires substantial experience with the product application.

The theoretical model described in [30] is used to characterize SDHW system performance under transient operation. The identification of the parameters in the theoretical model is carried out by given parameter identification software. The program finds the set of parameters that gives the best fit between the theoretical model and the measured data.

The identified parameters are used for the prediction of the long-term system performance for the climatic and load conditions of the desired location, using the same model as for parameter identification. The system prediction part of the theoretical model requires hourly values of meteorological data and specific load data.

The method can be applied to the following systems:

- systems with forced circulation of fluid in the collector loop;
- thermosiphon systems;
- integral collector storage systems.

However, there are some limitations to the method regarding the following dimensions.

- the collector aperture area of the system is between 1 and 10 m^2;
- the storage capacity of the system is between 50 and 1 000 litres;
- The specific storage tank volume is between 10 and 200 litres per square metre of collector aperture area.

Further limitations stated in the scope of the standard include:

- the test procedure cannot be applied to systems containing more than one storage tank. This does not exclude preheat systems with a second tank in series. However, only the first tank is considered as part of the system being tested;

- systems with collectors having non-flat plate-type incident-angle characteristics can be tested if the irradiance in the data file(s) is multiplied by the measured incident-angle modifier prior to parameter identification. The same irradiance correction should, in this case, also be used during any performance predictions based on the identified parameters;
- The test procedure cannot be applied to integrated auxiliary solar systems, with a high proportion of the store heated concurrently by the auxiliary heater. The results of the tests are only valid if the contribution of the auxiliary heat source to the overall heat production of the system is less than 75 %.

The test procedure consists of three test sequences:

- **S-sol**: The test sequence contains a number of consecutive days of measurement with significant solar input. It consists of two specific daily operation conditions Test A and Test B. Test A with operating conditions to achieve high efficiencies with seven draw-offs per day, the draw-off volume depending on system volume and collector area. At least three valid days with cumulated solar irradiation of more than 12 MJ/m^2 are required. Test B is designed to acquire information on collector array performance at low efficiencies and about storage tank heat losses. It includes five draw-offs per day, with volume depending on the draw-off temperature. At least 3 valid days with cumulated solar irradiation larger than 12 or 15 MJ/m^2 are required, of which two must be consecutive.
- **S-store**: The sequence is designed to evaluate the overall storage losses. It requires two consecutive valid Test B days, after which the system is shaded for 36 to 48 h until the final draw-off. During this time, no draw-offs are made and the solar irradiance must be under 200 Wm^{-2}. At the end of the test, the storage is conditioned to the initial state.
- **S-aux**: The test is intended to determine the heat losses and the volume fraction of the auxiliary heated portion of the storage. The operation under low solar irradiation conditions and an integrated auxiliary heater is assessed. It contains four Test B days with draw-offs as in Test B and electrical heating at specified times.

After characterization of the solar system, the annual system output is calculated for different load volumes at reference locations.

4.2 EN 12977: Thermal solar systems and components. Custom built systems

EN 12977 establishes the requirements for the durability, reliability and safety of small and large custom-built solar thermal systems with liquid heat transfer fluid within the collector circuit, designed for heating and cooling of mainly residential buildings. Moreover, it defines the requirements for the design of larger custom-built systems.

The EN 12977 package of standards includes the following documents:

- EN 12977-1:2011 – Thermal solar systems and components. Custom built systems. Part 1: General requirements for solar water heaters and combisystems;
- EN 12977-2:2012 – Thermal solar systems and components. Custom built systems - Part 2: Test methods for solar water heaters and combisystems;
- EN 12977-3:2011 – Thermal solar systems and components - Custom built systems - Part 3: Performance test methods for solar water heater stores;
- EN 12977-4:2012 – Thermal solar systems and components - Custom built systems - Part 4: Performance test methods for solar combistores;
- EN 12977-5:2012 – Thermal solar systems and components - Custom built systems - Part 5: Performance test methods for control equipment

A short description of the performance evaluation method, as well as on-going assessment for an extension to SHP systems, are given in Technical Report 5.1.4 of the QAiST deliverable D5.1 [31].

5 Assessment of the standards regarding their application for SHP systems

5.1 Nomenclature and definition of performance indicators

The first step towards defining a uniform and harmonised nomenclature for different performance figures was to analyse currently available standards for testing and rating of heat pumps, solar thermal collectors and their respective systems. Table 15 and Table 16 give an overview of analysed standards and a short description of the performance figures and indicators defined within.

Regarding the performance figures for heat pumps, the following can be concluded:

- *COP – Coefficient of Performance*: In all reviewed standards used for the performance of the heat pump unit under steady-state (except defrosting sequence) operating conditions. However, the consideration of the liquid circulation pumps' influence on the energy inputs and outputs differ among the standards. It is generally used for heating applications only.
- *EER – Energy Efficiency Ratio*: The same like COP, but used for cooling applications in most of the European standards. In many US standards defined as Btu/hr of cooling energy per W electricity consumption.
- *SCOP – Seasonal Coefficient of Performance*: Used only in one standard to express the calculated heating efficiency of the heat pump unit only for an assumed climate, building load etc.
- *SEER – Seasonal Energy Efficiency Ratio*: Same like SCOP, but for cooling applications.
- *SPF – Seasonal Performance Factor*: In the VDI guideline used to express roughly the same efficiency as the SCOP – not taking into account the whole system but only the heat pump unit with some auxiliary energy. In European standards used as a figure to express the efficiency of the overall system, including all auxiliary energy inputs.

For solar thermal, four main performance indicators can be found:

- For *η – Collector Thermal Efficiency, f_{sol} – Solar Fraction* and *f_{sav} – Fractional Energy Savings*, the same definition was found in all analysed standards, according to Table 16.
- *Thermal Performance*: In the reviewed standards, no system performance figure similar to SPF for the heat pump systems was found. However, from the delivered heat and the parasitic energy, as defined in the reviewed standards, the efficiency of the system could be calculated in the same manner as for the heat pump systems.

Table 15: Overview of analysed standards for testing and rating of heat pumps

Heat Pump Standards		
Standard	**PF**	**Definition**
EN 14511	COP	The COP is defined as the ration of the heating output of the heat pump unit divided by the effective energy input to the unit for a steady state operating condition. Energy inputs and outputs are corrected by the pumping energy needed to overcome the pressure lossses on the heat exchangers inside the unit.
	EER	Same definition like the COP, used for cooling applications (useful energy is cooling).
EN 15879-1	COP / EER	Uses same definitions as EN 14511, applied on direct expansion heat pumps.
AHRI 320/325/330	COP	A ratio of the heating capacity, excluding supplementary resistance heat, to the power input for steady state operating conditions.
	EER	Same definition like the COP, used for cooling applications (useful energy is cooling).
ISO 13256-1	COP	Ratio of the net heating capacity to the effective power input of the equipment at steady state operating conditions. The power inputs and outputs are corrected in the same way as in EN 14511
	EER	Same definition like the COP, used for cooling applications (useful energy is cooling).
prEN 14825	SCOP	Ratio of the overal heating energy delivered over a one year time period to the total energy input to the system. It is a calculatory value obtained under certain assumptions regarding the heating load, climate, controls etc. The basis for the calculation are **unit** tests, e.g. according to EN 14511.
	SEER	Same as SCOP, but for cooling applications.
ASHRAE 116	HSPF	Ratio of the total heat delivered over the heating season (not exceeding 12 months) to the total energy input over the heating season. It is a calculatory value obtained under certain assumptions regarding the heating load, climate, controls etc. The basis for the calculation are **unit** tests.
	SEER	Ratio of the total heat removed during the normal period of usage for cooling (not exceeding 12 months) to the total energy input during the same period. Obtained same as the HSPF.
VDI 4650-2	SPF (β)	The ratio of the useful heat released in the course of one year over the electrical energy used to drive the compressor and the auxiliary drives. It is a calculatory figure based on the test results from EN14511. It does not take into account electricity consumption for e.g. ground water pump, heat pump off-mode etc.
EN 15316-4-2	SPF	The ratio of the overal energy output to the overal energy input (final energy) of the heat pump system for heating and DHW.

Table 16: Overview of analysed standards for testing and rating of solar thermal components and systems

Solar Thermal Standards		
Standard	**PF**	**Definition**
EN 12975-2	η	The Collector Thermal Efficiency is the ratio of the energy removed by the heat transfer fluid over a specified time period, to the product of a defined collector area (gross, absorber or aperture) and the solar irradiation incident on the collector for the same period, under steady or non-steady state conditions (according to ISO 9488).
ISO 9806	η	Same as EN 12975
ASHRAE 93	ηg	Collector Thermal Efficiency, defined as the actual collected useful energy to the solar energy intercepted by the collector gross area.
EN 12976, EN12977	fsol	Solar Fraction is the energy supplied by the solar part of a system divided by the total system load. The solar part of a system and any associated losses need to be specified, otherwise the solar fraction is not uniquely defined (according to ISO 9488).
	fsav	Fractional Energy Savings is the reduction of purchased energy achieved by the use of a solar heating system, calculated as 1 - [(auxiliary energy used by solar heating system)/(energy used by conventional heating system)] in which both systems are assumed to use the same kind of conventional energy to supply the user with the same heat quantity giving the same thermal comfort over a specified time period (according to ISO 9488).
	Thermal performance	The thermal performance is defined as a set of performance indicators. For solar systems without auxiliary energy sources, these are: The heat delivered by the solar heating system, QL; the solar fraction, fsol; the parasitic energy, Qpar, if any is available. For systems including auxiliary energy sources: The net auxiliary energy demand, Qaux, net; the fractional energy savings, fsav; The parasitic energy, Qpar.
ISO 9459	Thermal performance	Comparable definition to EN 12976 and EN 12977
EN 15316-4-3		Same nomenclature as in EN 12977.

5.2 Deficiencies of current standards and guidelines for the application on SHP systems

In Technical Report 5.1.3 of the QAiST deliverable D5.1 [32], a number of methods for testing and performance evaluation of SHP systems currently being developed by different institutes in Europe have been briefly described. There are currently two basic approaches:

- system oriented approach – the whole system or part of the system is tested with defined boundary conditions and the results are used either for parameter identification of the model and simulate the performance, or to extrapolate them to estimate the final energy usage;
- component oriented approach – component testing system simulation (CTSS) method.

Comparing the available method descriptions and existing normative documents with the available products on the market and considering the multitude of existing system configurations, the following can be stated:

<u>Heat pump standards and guidelines</u>

- The operating conditions for heat pump testing provided in EN 14511 and EN 15879-1 should be extended to cover higher temperature ranges which may occur in SHP systems;
- Additional heat sources with typical temperature levels should be added to the standards, e.g. direct evaporation in solar collectors or hybrid collectors;
- There are almost no procedures for transient testing of the heat pumps (defrosting is included in EN 14511, partly transient conditions within EN 16147 for DHW), a feature especially important for the component characterisation within the CTSS method;
- Test sequences for capacity controlled heat pumps are still not fully developed;
- The performance calculation methods described in EN 14825 and EN 15316-4-2 (temperature bin methods) are difficult to extend to SHP systems, mainly because of the storage effects (both on the heat source as well as on the heat sink sides) and different control concepts. However, some work has been done on this issue, e.g. [33], and further developments are expected within IEA SHC Task 44 / HPP Annex 38 and IEA HPP Annex 39.

<u>Standards for solar thermal collectors</u>

- Some of the collector types common in SHP systems are not included in the standards, e.g. hybrid collectors, PVT collectors (see Chapter 8 in [34] for current status on PVT test procedures) etc.

- Collectors with direct evaporation of the refrigerant are not covered by the standards;
- In SHP systems, collectors can operate under lower operating temperatures providing heat for heat source regeneration or directly to the heat pump evaporator. In cases of unglazed or hybrid collectors, a large portion of that heat may originate from the ambient air or condensation on the collector surface. These effects are currently not covered by the standards. However, in Chapter [TR5.1.5] a proposal for the extension of EN 12975 in this respect is presented.

Standards for solar thermal systems

- Currently, only domestic hot water preparation is covered by EN 12976 and EN 12977. The methods should be extended to heating applications;
- The heat pump could be seen as an auxiliary heat source for the solar combisystem for parallel SHP systems. However, more complex configurations need further investigations;
- The performance prediction methods apply currently only to a limited number of well-defined system configurations. However, as shown in Chapter [TR5.1.1], there is a large variety of configurations within SHP systems. Further work towards grouping of certain configurations into “families” is needed to simplify the testing and calculation procedure and limit the number of parameters and boundary conditions needed;
- The models provided in the standards do not apply for some special components. Moreover, appropriate heat pump models have to be added;
- The reference operating conditions for the “black box” methods are suitable only for specified system configurations. New operating conditions would have to be developed and validated;
- The methods do not account for multiple sources and sinks for each of the subsystems, nor for complex control algorithms;

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