

Annex 28

Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating

Part 1. Proposals for calculation method and test procedure

Final Report

Operating Agent: Switzerland



IEA Heat Pump Programme Report

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For further information about the IEA Heat Pump Programme and for inquiries on heat pump issues in general contact the IEA Heat Pump Centre at the following address:

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IEA HPP Annex 28

Final Report IEA HPP Annex 28

Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating

Part 1: Proposals for calculation method and test procedure

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IEA HPP Annex 28

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Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating

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Information on IEA HPP Annex 28

<http://www.annex28.net>

Preface

IEA HPP Annex 28 entitled “Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating” has been started to develop a calculation method for the Seasonal Performance Factor (SPF) of combined operating heat pump systems.

Since a calculation requires a characteristic of the component efficiency (COP) and the heating capacity, an adapted test procedure has been developed in parallel to enable performing the calculation.

Consequently, IEA HPP Annex 28 had the objective to deliver

- An easy-to-use calculation method (“hand calculation”) to calculate the Seasonal Performance Factor of combined operating heat pump systems without extensive computational expense
- An adequate test procedure to provide the required input data of the heat pump system with a minimum of testing expense

Following these objectives the final report of the IEA HPP Annex 28 is structured in two parts

Part 1: Proposals for calculation method and test procedure

The first part comprises an executive summary of the IEA HPP Annex 28 project and the two essential results, namely the proposal for the calculation method and the proposal for the test procedure formatted in the layout of the template for draft standards of the European standardisation organisation CEN.

Part 2: Project description and standard documentation

The second part comprises the background of the project, the national contributions, and introduction to the present situation in standardisation and systems on the markets, performed evaluations and the discussion and decision of the approaches, which led to the calculation and test procedure in part 1. Further in-depth information can be found in the country reports on Task 1 and Task 2&3 of the participating countries which are referenced in the report.

Executive Summary IEA HPP Annex 28

Motivation for IEA HPP Annex 28

Which space heating and domestic hot water (DHW) system is the best for my application?

Even though this question appears fundamental, it is still not easy to answer, since standardised characteristic numbers with regard to the energy consumption - or better the standardised calculation methods to derive these numbers - for the comparison of heating systems are still missing, in particular on an international basis.

On this background IEA HPP Annex 28 entitled "Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating" was launched in January 2003 in the framework of the Heat Pump Programme (HPP) of the International Energy Agency (IEA) with nine participating countries: AT, CA, CH, DE, FR, JP, NO, SE, USA. The objective of the Annex is to fill this gap and develop comprehensive test procedures to calculate the seasonal performance factor (SPF) of combined operating heat pumps as a recommendation to international standardisation committees.

The scope of the Annex is restricted to combined heat pump systems including heat pump compact units for low energy houses, but the characteristic number Seasonal Performance Factor is suited to perform comparisons to other heating systems, too. Actually, the SPF is the basis for a calculation of other characteristic numbers like primary energy consumption or CO₂-emission, which for instance is to be used for heating system comparison in the framework of the EU Directive on the Energy Performance of Buildings (EPBD).

Especially new system developments using combined operation are not covered by existing standard testing or calculation methods. Thus, the application of the test procedures and the calculation methods developed in the framework of IEA HPP Annex 28 is manifold:

- Manufacturers have regulations for providing precise and uniform technical data
- On the other hand, system layouts can be compared in the design process
- Moreover, energy labels or building standards can be based upon the SPF.
- Hence, a uniform testing and calculation is necessary to overcome trade barriers and enhance consumer confidence.
- Last but not least a better notion of the performance can stimulate system improvements.

Task 1: State-of-the-art survey (markets and standardisation)

The work in the IEA HPP Annex 28 started with a characterisation of the markets of combined operating heat pump systems as well as an enquiry of the systems under development in the participating countries. Furthermore, the status-quo in the respective standardisation both on the international level of the European, American and Japanese standardisation and on the national level of the participating countries was analysed.

Results revealed that European markets are dominated by alternate combined operating systems, i.e. switching the heat pump to either the space heating or DHW function. However, simultaneous combined operating heat pumps, i.e. heat production for both functions at the same time, e.g. by desuperheating and/or condensate subcooling, can also be found on the markets. In Sweden, Norway, France and Austria, different system configurations using desuperheating for the domestic hot water production are on the market. Further simultaneous operating systems are under development.

European standardisation comprises standards for the testing of the space heating-only operation and the DHW-only operation. However, combined operation is not yet treated in the standards. A common calculation approach for the seasonal performance factor on the European level does not exist, either, but is presently under development in the framework of the EPBD.

American markets are dominated by simultaneous combined operating systems using desuperheating basically by air-to-air heat pump systems. Standards on the testing and calculation of the space heating-only and the DHW-only operation exist as well as a standard for testing and calculation of the simultaneous operation restricted to the system configuration of A/A-heat pumps with desuperheater.

In Japan, an alternate combined operating heat pump system for DHW and a floor heating system has recently been introduced in the market. However, standards for the combined operation do not exist, yet, only standards for the space heating-only and the DHW-only operation are in use.

Results of the survey (Task 1) were presented in detail in an Interim report delivered to the Executive Committee of the IEA Heat Pump Program in March 2004.

Conclusions of the results

Based on these results it was decided to cover the combined operating systems by an extension of the existing standards.

- Concerning the system boundary the heat generator as well as attached storages and back-up heaters are included, since many integrated systems contain these components in one casing. Concerning the calculation scheme according to the EPBD, this corresponds to the system boundary “generator” and “storage”.
- For the testing, a black box approach was chosen, i.e. only values on the system boundary are measured, in order to facilitate and unify the handling of different system layouts. Highly integrated systems are to be tested in system testing, while for other systems a component based testing can be applied.
- For the calculation a bin-methodology was chosen, since it is the best compromise between exactness of the results and computational expense and is already introduced in the American standardisation for the calculation of the space heating-only operation and in national guidelines in some European countries.

Task 2: Test procedure

Testing of different combined operating heat pump systems in both alternate configuration and simultaneous configuration according to existing standards and standard proposals have been performed. Feedback of the evaluation of these actual standards in operation and the proposed ones will be transferred to the respective working groups of the standardisation committee.

Evaluation of the test procedure confirmed, that alternate combined operation can be covered by testing of space heating-only and DHW-only operation according to existing standards, since the heat pump is either in the one or the other of the two operation modes. Therefore, the heat pump characteristic does not change for alternate combined operation with regard to the single operation modes and no additional testing is required for alternate combined operating systems.

Based on these experiences with the standards the extension for combined operation has been accomplished by introducing a third test for the combined operation mode by performing the cycle of the DHW-only testing during operation of the heating system. As outlined before, the combined testing is only required for simultaneous combined operating systems, since alternate operation can be covered by testing the single operation modes. Evaluation of the combined testing, however, is not that simple as in space heating-only mode and DHW-only mode, since allocation of electrical energy input to the different operation modes is not obvious. On the other hand, calculation shall yield an overall seasonal performance factor of the heat pump system. Thus, combined operation can also be treated as an additional operation mode with an own characteristic, which significantly simplifies the evaluation of the combined testing.

Task 3: Calculation method

As stated before, the calculation shall be performed on the basis of a bin-methodology. The principle of a bin method is depicted in Fig. 1. The cumulative annual frequency of the outside temperature is divided into temperature classes (temperature bins). In the centre of each bin, an operating point is evaluated with regard to the heat pump operation at these specific ambient conditions, which is determined by the standard component testing according to the respective standard for the space heating-only operation. The operating point is considered to characterise the heat pump operation of the whole bin. The areas of each bin correspond to heating degree hours, and consequently characterises the energy consumption in the bin.

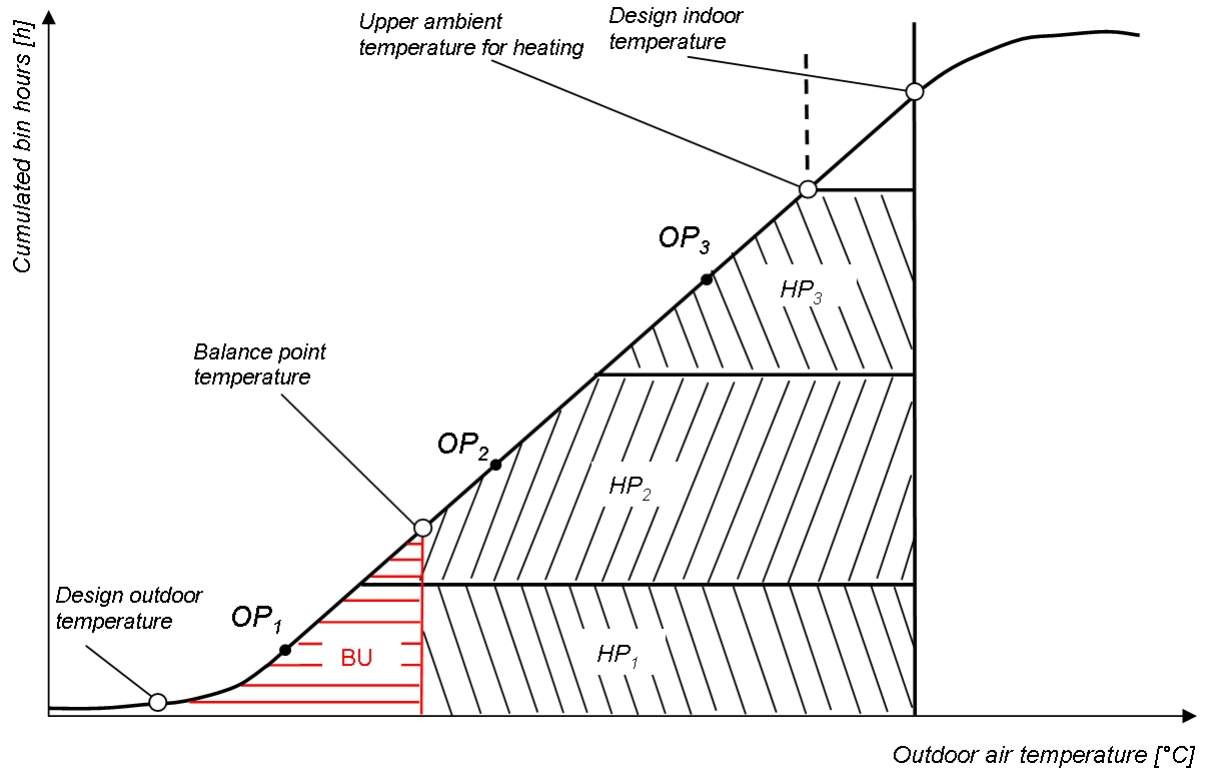


Fig. 1: Principle of the bin-methodology

Thus, a weighting of the operating conditions with this energy fraction of the bin and a subsequent summation of all bins delivers the seasonal performance. Electrical back-up heaters are also considered by an evaluation of the respective area in the cumulative frequency diagram, in Fig. 1 the area BU.

For the domestic hot water operation, a similar calculation can be performed based on standardised testing results of DHW testing, e.g. for Europe according to EN 255-3. The overall seasonal performance can be calculated by weighting the single seasonal performance factors with the annual energy consumption in heating and domestic hot water operation, respectively. Thus, the alternate combined operation can be covered in this way, since these systems are either in heating mode or in domestic hot water mode and consequently, heat pump characteristic is defined by the respective operation.

In simultaneous combined operation, however, the heat pump characteristic may change significantly between the single and the combined operation mode. Thus, a third operation mode, the combined operation has been introduced based on the test procedure already described. The estimation, how much combined operation takes place in the bin is done by an evaluation of the running time of each mode, which is determined by the respective heating capacity of the heat pump and the energy requirement in the bin.

The overall seasonal performance can be calculated by the weighting of three operation modes, i.e. space heating-only, domestic hot water-only and combined operation.

Obviously, the calculation method implies some simplifications in order to keep the calculation simple. One shortcoming with regard to low and ultra-low energy house may be the redistribution of the energy requirement to the bins, that is only dependent on the heating degree hours and thereby on the outside temperature, while in low-energy dwellings the solar gains may have a higher impact on the energy distribution. Another shortcoming is that in the present proposal an upper temperature for heating is an input value, while in the present building energy calculation no heating limit is calculated anymore. Nevertheless, controller settings normally contain this value in their parameter set.

Moreover, a further shortcoming is that controller layout and controller setting can only be partly taken into account, with the consequence that the calculation may require default factors characterising typical control schemes.

Validation of the calculation method and the test procedure

To validate the **test procedure** for simultaneous operation the test procedure has been applied to a simultaneous combined operating exhaust-air heat pump. It was approved that proposed test points are sufficient for a complete characteristic of the heat pump systems. The test delivered representative and reliable values. However, some test points require long testing time, which seems to be inevitable to enable a fair comparison. Moreover, it could also occur that the test procedure has to be adapted to special configurations of heat pumps in order to make a fair comparison between different system configurations.

To validate the **calculation method** with real data, field monitoring of pilot plants with combined operating heat pump systems were launched in the framework of the IEA HPP Annex 28. The evaluation is still in progress, so not all results could be integrated in the report despite an extension of the Annex time schedule. Nevertheless, the contained results give a first impression of the exactness of the method and the impact of different approaches.

Three direct expansion ground source heat pumps were evaluated for the heating operation. A compact unit and a ground source brine-to-water heat pump have been evaluated for the space heating and DHW operation. Differences between the calculation and the measured values are in the range of $\pm 6\%$ for the seasonal performance factor. Considering the simplifications in the approach these values are satisfactory and show the applicability of the method.

Not all types of system configuration were tested in the Annex nor were monitored in field measurements. Actually, no field measurement with simultaneous combined systems could have been evaluated, which is a task for further project work.

Implementation of results

The implementation of the results of the IEA HPP Annex 28 has already begun in the framework of the revision of standards in connection with the European Building Performance Directive (EPBD), which will deliver a set of calculation standards for domestic and non-domestic heating/DHW systems and shall come into operation in January 2006. The actual calculation approach of IEA HPP Annex 28 has been implemented in the heat pump draft of prEN 15316 part 4-2, and could be updated with the final results during the public enquiry which took place until the end of March 2006.

Concerning the test procedure, results will be transferred to the respective working group of the European standardisation organisation CEN as soon as it starts working.

Implementation of proposed testing and calculation for compact units is not yet clear, since presently there are no CEN working groups committed to these systems. However, the market share of these systems is growing continuously and there is a strong need for the manufacturers to provide uniform testing data for calculation of compact units in building regulations.

Germany has integrated a slightly changed calculation method of the IEA HPP Annex 28 on a monthly basis in the national standard DIN V 18599, which was finished with the deadline January 2006 with the implementation of the EPBD on the national level.

Conclusions

IEA HPP Annex 28 has delivered a calculation method for the Seasonal Performance Factor for combined operating heat pump systems including proposals for heat pump compact units. An adapted test procedure has been developed in parallel in order to deliver the necessary input data for the calculation.

First validation of the calculation method shows satisfactory results. Five systems comprising direct expansion, brine-to-water and a compact unit have been compared to field results within the Annex.

Results of the Annex 28 for the calculation have already been implemented in upcoming standards in the framework of the EPBD. Implementation of results on testing will follow as soon as respective CEN working groups will take up their work. Even though discussion of the proposals will take place in the future a uniform calculation method backed-up by the adequate testing seems realistic. In this respect, the objective of the IEA HPP Annex 28 has been achieved at least on the European level.

Concerning the international standardisation, the calculation approaches are quite similar, since all methods are based on the bin method. However, testing is still different due to the different systems on the market. Hence, to introduce internationally common standards, mainly the testing standards have to be unified, e.g. on ISO level. IEA HPP Annex 28 has delivered an evaluation of the relevant testing standards in Europe, North-America and Japan and can thus accelerate this process.

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A PROPOSAL FOR THE CALCULATION METHOD

Results of the IEA HPP Annex 28 with regard to the calculation method have been continuously transferred to the respective working group of the European standardisation organisation CEN. They have been integrated in a draft standard prEN15316 part 4-2 (formerly numbered prEN14335), which was sent to public enquiry for the standards in the framework of the EU Directive on the Energy Performance of Buildings (EPBD). After the standard was blocked some time for changes since it had already passed the internal enquiry of the Technical Committee, it has been sent out to public enquiry in October 2005.

Due to this situation, the following chapter contains two subchapters, an extract of the original draft that was sent to public enquiry with modifications based on Annex 28 decisions and on comments of the participants in order to update it to the proposals of the IEA HPP Annex 28. Basic comments received in public enquiry have been updated in the text and some general comments from public enquiry are discussed in part 2 chapter 3.4 of the final report. Extension of the Annex 28 will be introduced during the update to formal vote.

Moreover, the CEN draft standard contains as well parts that have not been in the scope of the IEA HPP Annex 28 and thus cannot be a proposal of the Annex, e.g. gas heat pumps, absorption heat pumps and other calculation methods like a typology method. These parts have been omitted, so the proposal deals with heat pump with electrically-driven compressor calculated case specific with the bin method.

A.1 Proposal of Annex 28 for the calculation method modified from CEN draft

A.1.1 Scope

The standard is part of the method for calculation of system energy requirements and system efficiencies prEN 15316 (formerly prEN14335). The framework for the calculation is described in the general part (prEN15316-1) (formerly prEN 14335-1) of the standard.

The energy performance can be assessed by giving either the heat generation system efficiencies or the heat generation system losses due to the system configuration.

The scope of this specific part is to standardise

- the required inputs
- the required outputs
- the calculation method

of the heat generation of heat pump systems with vapour compression cycle and electrically-driven compressor including its control using the following combination of heat sources and heat distribution

- outside air-to-water
- outside air-to-air
- exhaust air-to-water
- brine-to-water
- water-to-water
- direct expansion-to-water

The calculation refers to heat pumps for space heating and domestic hot water.

A.1.2 Normative references

This standard incorporates by dated or undated reference, provisions from other publications. These normative references are cited at the appropriate places in the text and the publications are listed hereafter. For dated references, subsequent amendments to or revisions of any of these publications apply to this standard only when incorporated in it by amendment or revision. For undated references the latest edition of the publication referred to applies.

EN 255-2 (replaced by EN 14511 in 2004)	1997	Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors – Heating mode, Part 2: Testing and requirements for marking for space heating units
EN 255-3	1997	Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors – Heating mode, Part 3: Testing and requirements for marking for sanitary hot water units–
EN ISO 7345	1996	Thermal insulation – Physical quantities and definitions
EN ISO 13790	2004	Thermal Performance of Buildings – Calculation of Building Energy Demand for Heating
prEN15316	2005	Heating systems in buildings. Method for calculation of system energy requirements and system efficiencies.
EN 14511	2004	Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling

A.1.3 Terms and definitions

A.1.3.1 Definitions

For the purpose of the present standard, the definitions in **ISO 7345** and the following definitions apply:

A.1.3.1.1 Alternate combined operation

Heat pump system layout, where the heat pump produces heat for the space heating system and the domestic hot water system by switching between the heating system and the domestic hot water system. Thus, heat is delivered either to the space heating or to the domestic hot water system

A.1.3.1.2 Application rating conditions

Mandatory rated conditions within the operating range of the unit that are published by the manufacturer or supplier.

A.1.3.1.3 Auxiliary energy

Energy required for the operation of any equipment (pumps, controls, carter heating) which is needed to fulfil the heat generation function.

A.1.3.1.4 Balance point temperature

Temperature at which the heat pump heating capacity and the building heat load are equal

A.1.3.1.5 Calculation period

Time period considered for the calculation of the heat losses of the space heating or domestic hot water system.

A.1.3.1.6 Coefficient of performance COP

Ratio of the heating capacity to the effective power input of the unit according to EN 14511.

A.1.3.1.7 Combined operation

Heat pump system layout that allows to produce heat for the space heating and the DHW system.

A.1.3.1.8 Cut-out period

Time period in which the electricity supply to the heat pump is interrupt by the supplying utility.

A.1.3.1.9 Effective power input

Average electrical power input of the unit within the defined interval of time obtained from:

- the power input for operation of the compressor and any power input for defrosting;
- the power input for all control and safety devices of the unit and;
- the proportional power input of the conveying devices (e.g. fans, pumps) for ensuring the transport of the heat transfer media inside the unit.

A.1.3.1.10 Heat demand, building

Heat to be delivered to the heated space to maintain the internal set-point temperature of the heated space.

A.1.3.1.11 Heat generator with double service

Heat generator which supplies energy to two different systems, e.g. the space heating system and the domestic hot water system in combined operation.

A.1.3.1.12 Heat pump

Unitary or split-type assemblies designed as a unit to transfer heat. It includes a vapour compression refrigeration system or a refrigerant/sorbent pair to transfer heat from the source by means of electrical or thermal energy at a high temperature to the heat sink.

A.1.3.1.13 Heat transfer medium

Any medium (water, air) used for the transfer of the heat without change of state. It can be:

- the fluid cooled by the evaporator ;
- the fluid heated by the condenser ;
- the fluid circulating in the heat recovery heat exchanger

A.1.3.1.14 Heated space

Room or enclosure heated to a given set-point temperature.

A.1.3.1.15 Heating capacity ϕ_{hp}

Heat given off by the unit to the heat transfer medium per unit of time.

NOTE: If heat is removed from the indoor heat exchanger for defrosting, it is taken into account.

A.1.3.1.16 Heating degree hour

Difference in temperature between the outdoor hourly mean temperature and a given design temperature of the building space.

A.1.3.1.17 Heating system heat losses, total

Total of the heat losses from the heating system, including recoverable heat loss.

A.1.3.1.18 Low temperature cut out

Temperature, at that heat pump operation is stopped and the total heat requirements are covered by back-up heater.

A.1.3.1.19 Operating range

Range indicated by the manufacturer and limited by the upper and lower limits of use (e.g. temperatures, air humidity, voltage) within which the unit is deemed to be fit for use and has the characteristics published by the manufacturer

A.1.3.1.20 Primary pump

The primary pump is the pump mounted in the circuit containing the generator and hydraulic decoupling in case of an installed hydraulic decoupling between the generation and the distribution subsystem, e.g. a heating storage in parallel configuration or a hydraulic distributor.

A.1.3.1.21 Recoverable heat loss

Part of the heat loss, from the space and domestic hot water system which can be recovered to lower the space heat use.

A.1.3.1.22 Recovered heat loss

Part of the recoverable heat loss which contributes to meet the heat requirement of the space.

A.1.3.1.23 Standard rating condition

Mandatory condition that is used for marking and for comparison or certification purposes.

A.1.3.1.24 Steady state

State of a system in which movement of matter or energy phenomena are taking place, when the various physical phenomena are independent of time.

A.1.3.1.25 Simultaneous combined operation

Operation mode or heat pump system layout where energy for space heating and domestic hot water heating is produced simultaneously. Frequent system layouts of the simultaneous operation are refrigerant desuperheaters and condensate subcoolers.

A.1.3.2 Symbols and abbreviations

For the purpose of the standard, the following symbols, units (**Table A 1**) and indices (**Table A 2**) apply:

Table A 1 - Symbols and Units

Symbol	Name of quantity	Unit
ϕ	Temperature change coefficient	-
ϕ	Thermal capacity	W
η	Efficiency	-
θ	Celsius temperature	°C
ρ	Density	kg/m ³
$\Delta\theta$	Temperature difference	K
Δp	Pressure drop	Pa
A	Area, surface	m ²
b	Temperature reduction factor	-
c	Specific heat capacity	J/(kg·K)
COP	Coefficient of performance	W/W
E	Quantity of energy, electricity	J
e	Expenditure factor	-
f	Correction factor	-
HDH	Cumulative heating degree hours	Kh
k	Fraction of recovered auxiliary energy	-
n	Number of quantity	-
N	Cumulative number of quantity	-
P	Electrical Power	
p	Fraction of quantity	-
Q	Quantity of heat	J
SPF	Seasonal performance factor	-
t	Time, period of time	s
U	Heat loss coefficient (U-value)	W/(m ² ·K)
\dot{V}	Volume flow	m ³ /s
W	Quantity of energy, auxiliary	J
w	Weighting factor	-

Table A 2 - Indices

<i>amb</i>	ambient	<i>hours</i>	number of hours	<i>out</i>	output from system
<i>aux</i>	auxiliary	<i>hp</i>	heat pump	<i>ra</i>	return air
<i>avg</i>	average	<i>hr</i>	heat recovery	<i>rd</i>	recovered
<i>bin</i>	referring to number of bins	<i>i</i>	referring to bin i	<i>rl</i>	recoverable
<i>bp</i>	balance point	<i>ID</i>	indoor at design conditions	<i>s</i>	storage
<i>bu</i>	back-up heater	<i>in</i>	consumed by system	<i>sa</i>	supply air
<i>cap</i>	lack of capacity	<i>int</i>	internal	<i>sb</i>	in stand-by operation
<i>combi</i>	combined operation	<i>L</i>	loss	<i>si</i>	sink
<i>d</i>	distribution	<i>lower</i>	lower limit	<i>sin</i>	single (operation)
<i>DHW</i>	domestic hot water	<i>ltc</i>	low temperature cut-out	<i>so</i>	source
<i>ea</i>	exhaust air	<i>max</i>	maximum	<i>t</i>	total
<i>eff</i>	effective	<i>nom</i>	nominal	<i>upper</i>	upper limit
<i>es</i>	stand-by acc. EN 255-3	<i>nr</i>	non recoverable	<i>w</i>	water
<i>ext</i>	external	<i>oa</i>	<i>outdoor air</i>	<i>x</i>	absolute water content of moist air
<i>g</i>	generation	<i>ON</i>	running	$\Delta\theta$	referring to temperature difference
<i>h</i>	heating	<i>op</i>	operation, operation limit		

A.1.4 Principle of the method

A.1.4.1 Heat balance of generation subsystem

The method covers heat pumps for heating, heat pump water heaters and heat pumps with combined space heating and domestic hot water production, where the same heat pump delivers the space heating and hot water heat requirement.

In the case of alternate operating heat pumps, the heat pump is switched from the space heating system to the hot water system. In simultaneous operating heat pumps, space heating and domestic hot water heat is supplied at the same time. Simultaneous operating heat pumps may have different heating capacity and COP characteristics in heating-only, DHW-only and simultaneous operation mode, which has to be taken into account in the calculation method.

Physical factors taken into account:

The calculation method takes into account the following physical factors that have an impact on the seasonal performance factor and thereby the required electrical energy input to meet the heat requirements of the respective space heating and /or DHW distribution subsystem

- type of generator (air-to-water, liquid-to-water, air-to-air)
- space heating and domestic hot water heat requirement
- effects of variation of source and sink temperature on heating capacity and COP
- auxiliary input needed to operate the generator not considered in standard testing of heating capacity and COP acc. to EN 14511
- system losses of the generation subsystem (heating buffer storage, DHW storage)

Calculation structure:

The calculation is performed considering the following input data

- type of the heat pump (air-to-water, liquid-to-water etc.)
- system configuration (integrated domestic hot water production, operating mode of the back-up system)
- ambient conditions (outdoor air temperature of the site)

Based on these input data, the following output data are calculated

- required electrical energy $E_{in,g}$ to meet the space heating and/or domestic hot water heat requirements
- total generator heat loss $Q_{l,g,t}$
- required auxiliary energy W_g to operate the generator
- total recoverable generator heat losses $Q_{l,g,rl}$

The following heat balance can be made for the generation subsystem (Figure A 1)

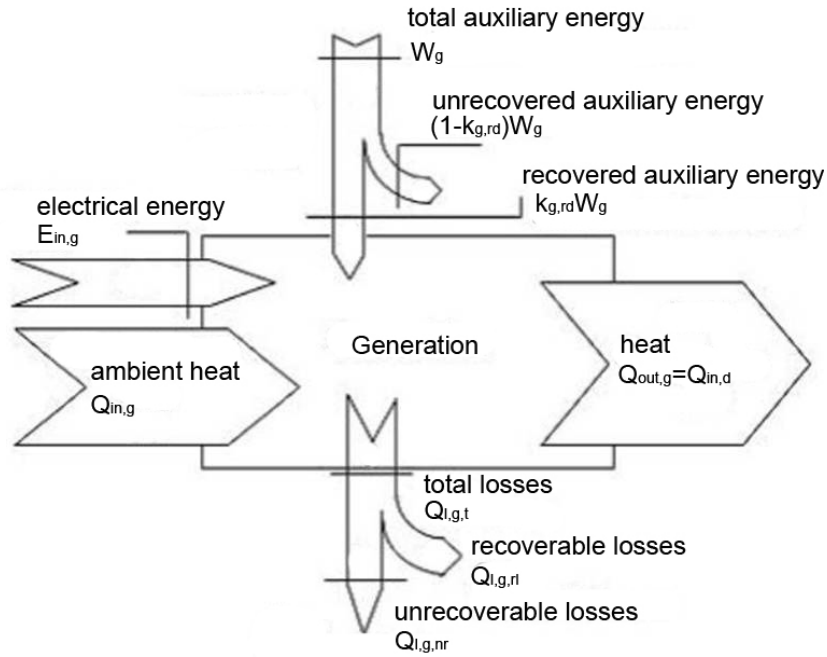


Figure A 1 - Energy balance of generation subsystem with calculated energy flows (electrically-driven heat pump)

A.1.4.2 Energy input needed to meet the heat requirements

The energy balance for the generation subsystem is:

$$E_{in,g} = Q_{out,g} + Q_{l,g,t} - Q_{in,g} - k_{g,rd} \cdot W_g \quad [J]$$

eq. A 1

where

$E_{in,g}$	electrical energy input to cover the heat requirement of the distribution subsystem	(J)
$Q_{out,g}$	heat energy requirement of the distribution subsystem	(J)
$Q_{in,g}$	ambient heat energy used as heat source by the heat pump	(J)
$Q_{l,g,t}$	total losses of the generation subsystem	(J)
$k_{g,rd}$	recovered thermal energy fraction from the auxiliaries	(-)
W_g	auxiliary energy input to operate the generation subsystem	(J)

The term $E_{in,g}$ refers to the energy to operate the heat pump according to the testing in EN 14511 (formerly EN 255-2). This standard considers partly the auxiliary energy for the source and sink pump, namely the fraction to overcome the internal pressure drop in the heat pump evaporator and condenser respectively as well as the energy for the control, defrosting and an eventually installed carter heating.

Total heat losses $Q_{l,g,t}$ mainly refer to the losses of storages that may be included in the system boundary.

$k_{g,rd}$ describes the fraction of auxiliary energy, that is recovered as thermal energy, e.g. in case of pumps where a fraction of the auxiliary energy is directly transferred as thermal energy to the heating water. This fraction is included in the COP value from standard testing according to EN 14511.

A.1.4.3 Auxiliary energy W_g

Auxiliary energy is energy needed to operate the generation subsystem, e.g. for the source pump or the control of the generation subsystem. As heat pump heating capacity and COP in this standard is calculated on the basis of results from component testing according to EN 14511 (formerly EN 255), only the auxiliary energy not included in the test results, e.g. the power to overcome the external pressure drop, is considered in W_g .

Auxiliary energy is accounted to the generation subsystem as long as no transport energy is transferred to the distribution system.

In the case of a heat pump, basically the source pump has to be considered in the generation subsystem.

In case a hydraulic decoupling between the generation and various distribution systems, e.g. by a storage in parallel configuration, the extra primary pump for loading of the heating buffer storage is accounted as well in the generation subsystem. In this case the power to overcome the external pressure drop has to be taken into account. If no primary pump is considered, since there is no hydraulic decoupling between the generation and distribution subsystem, the COP-values have to be corrected for the internal pressure drop, which is included in the COP-values by the standard testing.

A.1.4.4 Recoverable, recovered and unrecoverable heat losses

The heat losses of the generation subsystem that are calculated are not necessarily lost. They are partly recoverable losses, and a part of these recoverable losses is actually recovered. The part of the recovered losses is determined by the localisation of the generation subsystem and the utilisation factor (gain/loss ratio, see EN ISO 13790).

Recoverable generation subsystem heat losses $Q_{l,g,rl}$ are for instance heat losses through the envelope of a generation subsystem that is installed in the heated space. However, heat losses through the envelope of a generation subsystem are non recoverable, if installed outside the heated space.

A.1.4.5 Calculation periods

Heat pump performance strongly depends on the operating conditions, i.e. the source and the sink temperature. As source and sink temperatures vary over the operation period, the heat pump performance is evaluated at periods of defined source and sink temperatures. Thus, calculation periods are not oriented at the time scale, i.e. monthly values, but on the frequency of the outdoor air temperature.

However, the calculation can be performed as monthly calculation, if the method described in chapter 5 is accomplished for every month.

A.1.4.6 Heat pumps with combined space heating and hot water production

For combined operation of the heat pump two types of system configuration can be distinguished, alternate and simultaneous operation.

Alternate operation switches the heat pump operation from the space heating system to the domestic hot water system in case of domestic hot water demand, e.g. in the system configuration shown in Figure A 2 with a domestic hot water storage in parallel. The domestic hot water operation has a priority, i.e. space heating operation is interrupted in case of domestic hot water heat requirement.

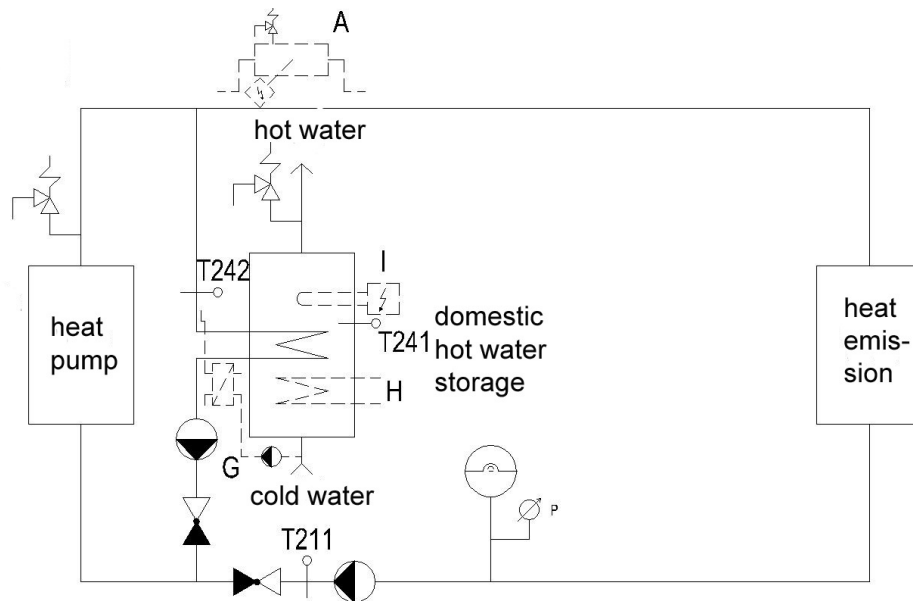


Figure A 2 - Heat pump with alternate hot water production by switching the heat pump from the heating system to the domestic hot water system

Newer concepts of heat pumps aim at improving the heat pump cycle to achieve better overall performance by using temperature adapted heat extraction by means of

- desuperheating
- condensate subcooling
- cascade cycles with internal heat exchangers

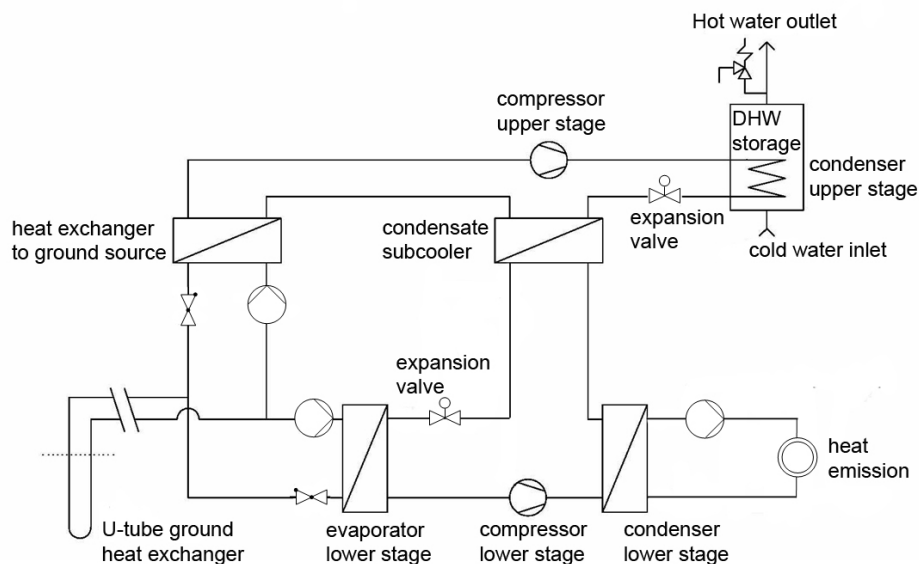


Figure A 3 - Heat pump with cascade cycle using condensate subcooling for hot water production

In these concepts space heating and hot water requirements are covered at the same time. Figure A 3 shows a hydraulic scheme of a simultaneous operating system using a cascade cycle with condensate subcooling for the DHW production.

In simultaneous operation three operation modes have to be distinguished:

- space heating-only operation:
Only the space heating system is in operation, e.g. in the configuration shown in Figure A 3, only the lower stage heat pump is in operation (winter time, DHW storage entirely loaded)
- domestic hot water-only operation:
Only the domestic hot water system is operated, e.g. in the configuration of Figure A 3, only the upper stage heat pump is operated and the heat is extracted from the ground source (summer operation, no heating)
- simultaneous operation:
Both space heating and domestic hot water operation is accomplished at the same time. For the configuration shown in Figure A 3, both stages are operated. The heat for the lower stage heat pump is taken from the ground source, the heat for the upper stage heat pump is taken from the condensate subcooling of the lower stage heat pump (winter operation, DHW storage partly unloaded)

As heating capacity and COP characteristic of the simultaneous operation may differ significantly from the other operation modes, more test results have to be available and taken into account. Adequate test procedures for different types of combined operating heat pumps are given in part B.

Depending on the system configuration, not necessarily all three operation modes occur like in the system depicted in Figure A 3. It might be the case that only simultaneous operation in wintertime and DHW-only operation in summertime occur, i.e. not in every simultaneous operating system configuration three operation modes have to be considered.

A.1.5 Generation subsystem calculation

Performance calculation methods of the heat generation corresponding to different applications (simplified/detailed estimation of seasonal performance) differ in

- in the data used
- the operating conditions taken into account
- the calculation period

In this report a detailed case specific method is treated. The field of application is shortly described in the following

- Detailed case specific seasonal performance method based on component efficiency (bin-method)

The method is based on the test results according to heat pump testing standard EN 14511 or the former EN 255, but complementary data are needed in order to take into account the specific operating conditions of each individual installation. Therefore, the heating period is split-up in bins dependent on the outside temperature. The calculation is carried out for the corresponding operating conditions of the heat pump.

As site specific outdoor air temperatures and specific test results for an individual heat pump are considered, the method is suited to prove the compliance with building regulations.

A.1.5.1 Principle of the case specific heat pump seasonal performance method based on efficiency data

As heat pump efficiency strongly depends on the operating conditions, i.e. source and sink temperature, the calculation is performed for periods defined by the source and sink temperature. The source and sink temperature level has the most significant impact on heat pump performance.

The calculation method is based on an evaluation of the outdoor air temperature. The annual frequency, which is derived from hourly average values of the outdoor air temperature, is divided into temperature intervals (bins), which are limited by an upper temperature θ_{upper} and a lower temperature θ_{lower} . Operating conditions of the bin are characterised by operating points in the centre of each bin (Figure A 4). The method assumes that the operating point defines the operating condition of the heat pump for the whole bin.

For each bin, the heating capacity and the COP are evaluated from standard test measurements. The difference between the heat requirements and the output energy of the heat pump has to be supplied by a back-up heater. System losses and electrical energy input to auxiliaries are calculated for each bin, too.

Summation of the energy input of the single bins is performed to receive the total electrical energy input for the whole period of operation and the seasonal performance factor of the generation subsystem, respectively. Depending on the existence of a back-up heater and its operating mode, supplied back-up energy is determined and summed-up to receive the overall energy consumption.

Cumulative frequency corresponds to the losses of the building, i.e. solar and internal gains are not reflected by the distribution. The gains are taken into account by using the net energy requirement of the space heating distribution subsystem. However, the redistribution to the bins might differ due to the used gains. Therefore, an upper temperature limit for heating is applied to correct the redistribution of the bins. The upper temperature limit for heating is set according to the controller settings, or, in case no values are known, based on the used solar and internal gains. The higher the fraction of used gains (solar passive houses), the lower the temperature limit for heating is to be chosen.

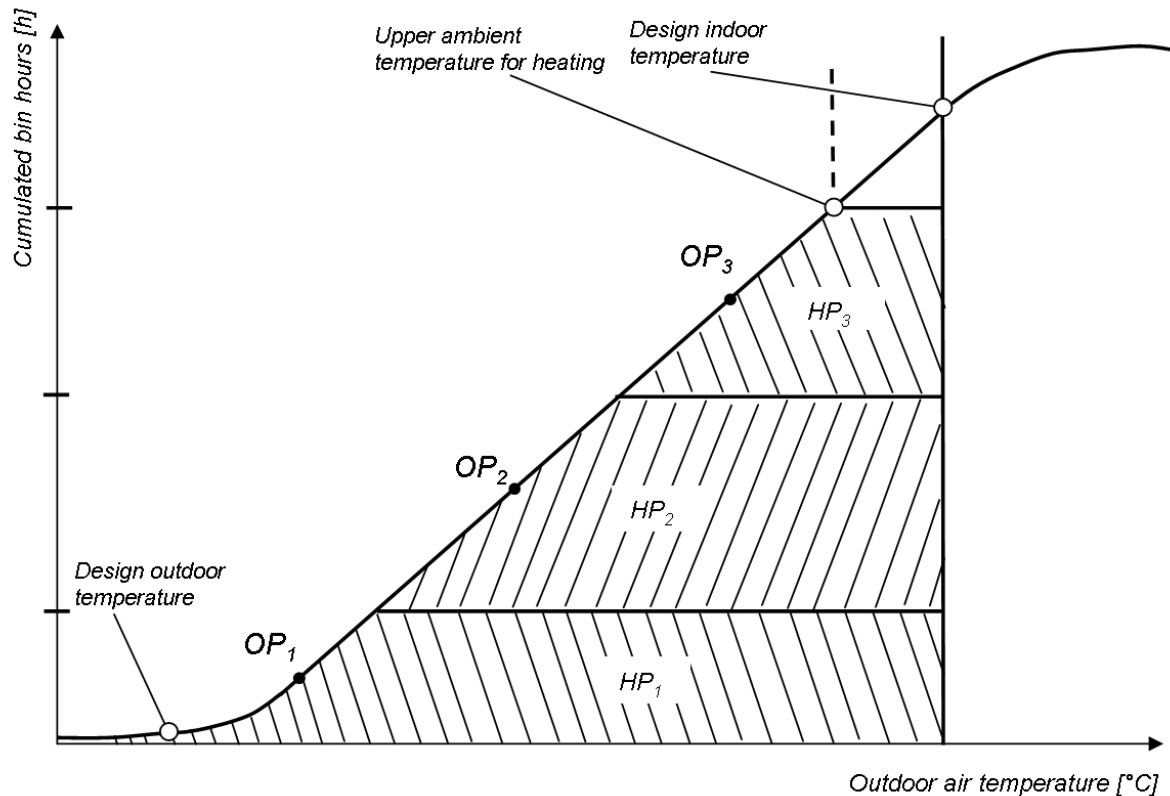


Figure A 4 - Bin hours vs. ambient dry bulb temperature (HP – heat pump, OP - operating point)

A.1.5.2 Input Data for the calculation

A.1.5.2.1 Space heating mode:

- Hourly-averaged data of outdoor temperature of the site (e.g. test reference year or data evaluated by Meteoronorm [A 1]), outdoor design temperature
- Heat energy requirement of the space heating distribution subsystem acc. to prEN 15316-2-3
- Controller setting of the heat emission system (heating characteristic curve = supply temperature of the heating system dependent on the outdoor temperature, return temperature at design conditions, upper temperature limit for heating)
- Heat pump characteristic for heating capacity and COP according to EN 14511 (or testing results according to the former EN 255-2)
- System configuration
 - back-up system operation mode, balance point temperature, low temperature cut-out
 - installed heating buffer storage and heat loss values (e.g. heat loss in 24 h)
- Auxiliary components
 - Nominal power of source pump, storage loading pump or circulation pump, stand by power input, respectively

A.1.5.2.2 Domestic hot water mode:

- Domestic hot water energy requirement according to the domestic hot water part of this standard prEN 15316-3-2
- Cold water inlet temperature (e.g. 15°C), DHW design temperature (e.g. 60°C), operating limit temperature of the heat pump (e.g. 55°C)
- DHW heat pump characteristic according to EN 255-3
- Parameters of the domestic hot water storage (heat loss value of the DHW storage)

A.1.5.3 Calculation steps to be performed

In this paragraph an overview of the calculation steps to be performed is given.

The single steps are explained in detail in the following chapters. Each subchapter contains the description for the respective operating modes (space heating, domestic hot water).

Step 1: Determination of energy requirement in the single bins (**chapter A.1.5.4**)

Step 2: Determination of back-up energy in the single bins (**chapter A.1.5.5**)

Step 3: If required, correction of steady state heating capacity / COP (EN 14511) for bin source and sink temperature (**chapter A.1.5.6**)

Step 4: Calculation of the running time of the heat pump in different modes (**chapter A.1.5.7**)

Step 5: Calculation of auxiliary energy input (**chapter A.1.5.8**)

Step 6: Calculation of generation subsystem heat losses
and recoverable generator losses (**chapter A.1.5.9**)

Step 7: Calculation of total energy input to cover the requirements, optional calculation of seasonal performance factor (**chapter A.1.5.10**)

A.1.5.4 Space heating and domestic hot water requirement for the calculation periods

A.1.5.4.1 Space heating mode

The heat requirement of the space heating distribution subsystem in bin i can be calculated with a weighting factor, which is derived by evaluating the frequency of the outside dry bulb temperature by means of heating degree hours (HDH).

The weighting factors are calculated by the equation

$$w_{h,i} = \frac{HDH_{\theta_{upper,i}} - HDH_{\theta_{lower,i}}}{HDH_{\theta_{tth}}} \quad [-] \quad \text{eq. A 2}$$

where

$w_{h,i}$	weighting factor for space heating in bin i	(-)
$HDH_{\theta_{tth}}$	total heating degree hours up to upper temperature limit for heating	(Kh)
$HDH_{\theta_{upper,i}}$	heating degree hours up to the temperature at upper limit of bin i	(Kh)
$HDH_{\theta_{lower,i}}$	heating degree hours up to the temperature at lower limit of bin i	(Kh)

The heating degree hours for the respective climatic regions shall be given in a national annex or taken from national standardisation. The national standards shall at least specify one site for which comparative calculation can be done.

The total space heating requirement is given by the distribution subsystem $Q_{d,in,h}$ and calculated in the distribution part of this standard prEN 15316-2-3.

The energy requirement in the respective bin is calculated by the equation

$$Q_{out,g,h,i} = Q_{out,g,h} \cdot w_{h,i} \quad [J] \quad \text{eq. A 3}$$

where

$Q_{out,g,h,i}$	heat energy requirement of the space heating distribution subsystem in bin i	(J)
$Q_{out,g,h}$	total heat energy requirement of the space heating distribution subsystem	(J)
$w_{h,i}$	weighting factor for space heating in bin i	(-)

The bin time is calculated as difference of the cumulative time at the upper and lower bin limit according to the equation

$$t_i = (N_{hours,\theta_{upper,i}} - N_{hours,\theta_{lower,i}}) \cdot 3600s/h \quad [s] \quad \text{eq. A 4}$$

where

t_i	time in bin i	(s)
$N_{hours,\theta_{upper,i}}$	cumulative number of hours up to the upper temperature limit of bin i	(h)
$N_{hours,\theta_{lower,i}}$	cumulative number of hours up to the lower temperature limit of bin i	(h)

A summation of all bin times t_i for space heating delivers the heating period.

However, for the heat pump operation there may be time restrictions, so that not the entire bin time is available for the heat pump operation, e.g. a possible cut-out time of the electricity supply on the background of particular tariff structures for heat pumps by the utility. Thus, the effective bin time is the time in the bin diminished by the cut-out time per day and is calculated

$$t_{i,eff} = t_i \cdot \frac{24h - t_{co}}{24h} \quad [s] \quad \text{eq. A 5}$$

where

$t_{i,eff}$	effective bin time in bin i	(s)
t_i	time in bin i	(s)
t_{co}	cut-out hours per 24 hours (1 day)	(h/d)

A.1.5.4.2 Domestic hot water mode

The total domestic hot water heating requirement is given by the distribution subsystem $Q_{d,in,DHW}$ and calculated in the domestic hot water part of this standard prEN 15316-3-2.

The DHW heat requirement in bin i can be calculated by a daily constant consumption using the bin time in bin i. It can be calculated by the equation

$$Q_{out,g,DHW,i} = Q_{out,g,DHW} \cdot \frac{N_{hours,i}}{N_{hours,t}} = Q_{out,g,DHW} \cdot w_{DHW,i} \quad [J] \quad \text{eq. A 6}$$

where

$Q_{out,g,DHW,i}$	domestic hot water requirement of the distribution subsystem in bin i	(J)
$Q_{out,g,DHW}$	total domestic hot water requirement of the distribution subsystem	(J)
$N_{hours,i}$	cumulated hours in bin i	(h)
$N_{hours,t}$	total hours of DHW operation (e.g. 8760 for year round operation)	(h)
$w_{DHW,i}$	weighting factor for DHW operation in bin i	(-)

NOTE Instead of a daily constant DHW consumption expressed by the bin time, a profile of the DHW consumption dependent on the outdoor air temperature can be considered.

A.1.5.5 Calculation for back-up heating

Back-up energy can be required for two reasons: One reason is a temperature operation limit of the heat pump, i.e. the temperature that can be reached with the heat pump is restricted to a maximum value. This fraction of back-up energy is treated in chapter A.1.5.5.1. On the other hand, there could be a multivalent design of the generator subsystem, i.e. the heat pump is not designed for the total load. Then, a fraction of back-up energy is required due to a lack of heating capacity of the heat pump. For the space heating system an simplified calculation of the back-up operation due to a lack of capacity is given in chap. A.1.5.5.2.

A.1.5.5.1 Back-up energy due to the operation limit temperature

Depending on the refrigerant and the heat pump internal cycle, the maximum temperature level that can be produced with the heat pump is restricted by an operation limit. If temperatures above a certain temperature are required they cannot be produced by the heat pump but have to be reheated by a back-up heater. Therefore, the fraction of back-up energy due to the operation limit of the heat pump can be calculated

$$p_{bu,op,i} = \frac{Q_{bu,op,i}}{Q_{out,g,i}} = \frac{\dot{m}_w \cdot c_w \cdot (\theta_{nom,i} - \theta_{op,hp}) \cdot t_{ON,hp,i}}{Q_{out,g,i}} \quad [-] \quad \text{eq. A 7}$$

where

$p_{bu,op,i}$	fraction of back-up energy due to the operation limit of the heat pump in bin i	(-)
$Q_{bu,op,i}$	Back-up heat energy due to the operation limit of the heat pump in bin i	(J)
$Q_{out,g,i}$	heat energy requirement of the distribution subsystem in bin i	(J)
\dot{m}_w	mass flow rate of the heat transfer medium	(kg/s)
c_w	specific heat capacity of the heat transfer medium	(J/(kg·K))
$\theta_{nom,i}$	nominal temperature requirement of the system	(°C)
$\theta_{op,hp}$	operation limit temperature of the heat pump (max. temperature, that can be reached with the heat pump operation)	(°C)
$t_{ON,hp,i}$	running time of the heat pump	(s)

For the space heating operation, the fraction $p_{bu,op,h,i}$ usually does not occur, i.e. $p_{bu,op,h,i} = 0$, since the design of the heat emission system is usually adapted to required temperature levels below the operation limit of the heat pump.

For DHW operation higher temperatures than the operation limit may be required so that the heat pump delivers the heat up to the operation limit temperature, e.g. 55°C, and the additional temperature requirement, e.g. up to 60°C, is supplied by the back-up heater. The fraction of back-up heat energy supplied to the domestic hot water system is given by

$$p_{bu,op,DHW,i} = \frac{Q_{bu,op,DHW,i}}{Q_{out,g,DHW,i}} = \frac{\rho_w \cdot V_w \cdot c_w \cdot (\theta_{hw} - \theta_{op,hp})}{\rho_w \cdot V_w \cdot c_w \cdot (\theta_{hw} - \theta_{cw})} = \frac{\theta_{hw} - \theta_{op,hp}}{\theta_{hw} - \theta_{cw}} \quad [-] \quad \text{eq. A 8}$$

where

$p_{bu,op,DHW,i}$	fraction of back-up energy due to the operation limit of the heat pump for DHW in bin i	(-)
$Q_{bu,op,DHW,i}$	DHW back-up heat energy due to the operation limit of the heat pump in bin i	(J)
$Q_{out,g,DHW,i}$	heat energy requirement of the domestic hot water subsystem in bin i	(J)

ρ_w	density of water	(kg/m ³)
V_w	volume of the hot water draw-off	(m ³ /s)
c_w	specific heat capacity of water	(J/(kg·K))
θ_{hw}	temperature of the hot water at storage outlet	(°C)
$\theta_{op, hp}$	operation limit temperature of the heat pump (max. temperature, that can be reached with the heat pump operation)	(°C)
θ_{cw}	temperature of the cold water inlet	(°C)

The operation limit temperature shall be taken from manufacturer data or evaluated based on the applied refrigerant.

NOTE: Depending on the system configuration back-up heat for DHW may also be used for covering a lack of capacity. The amount of back-up needed can be evaluated by the required running time of the heat pump according to chap A.1.5.7.3. The fraction due to a lack of capacity is added to the fraction of the back-up energy due to the temperature limit of the heat pump.

A.1.5.5.2 Back up energy due to a lack of capacity - heating mode

Operation of back-up heater is determined by the system design criteria and can be characterised by the operating mode (alternate operation, parallel operation, partly-parallel operation) and the respective temperatures, balance point temperature and eventually heat pump low-temperature cut-out. By these temperatures the energy fraction of the heat pump and back-up operation can be determined by the area under the cumulative frequency and energy consumption can thus be calculated.

A.1.5.5.2.1 Alternate operating mode of the back-up heater

In alternate operating mode of the back-up heater, the heat pump generator is switched-off at the balance point temperature, and only the back-up heating supplies the full heat energy requirement below the balance point.

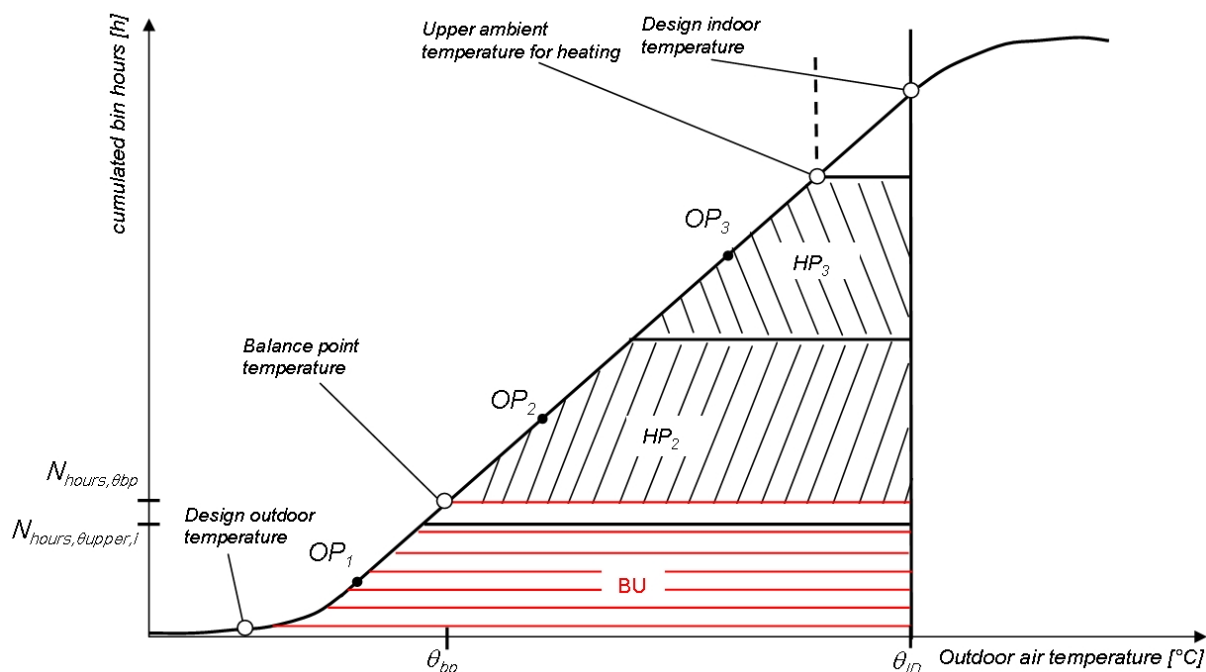


Figure A 5 – Fraction of back-up heat for alternate operating mode
(HP – heat pump, OP – operating point, BU – back-up)

Figure A 5 shows the areas under cumulative annual frequency of the outdoor air temperature, which correspond to the energy fractions. The area BU represents the energy fraction delivered by the back-up heater. The fraction of the back-up heat for alternate operation can be calculated by the equation

For $\theta_{bp} > \theta_{upper,i}$

$$p_{bu,h,i} = \frac{A_{bu,i}}{A_i} = \frac{HDH_{\theta_{upper,i}}}{HDH_{\theta_{upper,i}} - HDH_{\theta_{lower,i}}} = \frac{HDH_{\theta_{upper,i}}}{HDH_{\theta_{upper,i}}} = 1 \quad [-] \quad \text{eq. A 9}$$

$$p_{bu,h,i+1} = \frac{A_{bu,i+1}}{A_{i+1}} = \frac{HDH_{\theta_{bp}} - HDH_{\theta_{upper,i}}}{HDH_{\theta_{upper,i+1}} - HDH_{\theta_{lower,i+1}}} \quad [-] \quad \text{eq. A 10}$$

and for $\theta_{bp} < \theta_{upper,i}$

$$p_{bu,h,i} = \frac{A_{bu,i}}{A_i} = \frac{HDH_{\theta_{bp}}}{HDH_{\theta_{upper,i}}} \quad [-] \quad \text{eq. A 11}$$

where

$p_{bu,h,i}$	fraction of the space heating energy covered by the back-up heater	(-)
$A_{bu,i}$	area BU in bin i acc. to Figure A 5	(Kh)
$A_{bu,i+1}$	area BU in subsequent bin i+1 acc. to Figure A 5	(Kh)
A_i	total area of bin I (between upper and lower temperature limit)	(Kh)
HDH_{θ}	cumulated heating degree hours up to the respective temperature θ	(Kh)

A.1.5.5.2.2 Parallel operating mode of the back-up heater

In parallel operating mode of the back-up heater, the heat pump generator is not switched-off at the balance point temperature, but contributes to cover the energy requirement. The back-up heater only supplies the part that the heat pump cannot deliver due to the lack of heating capacity.

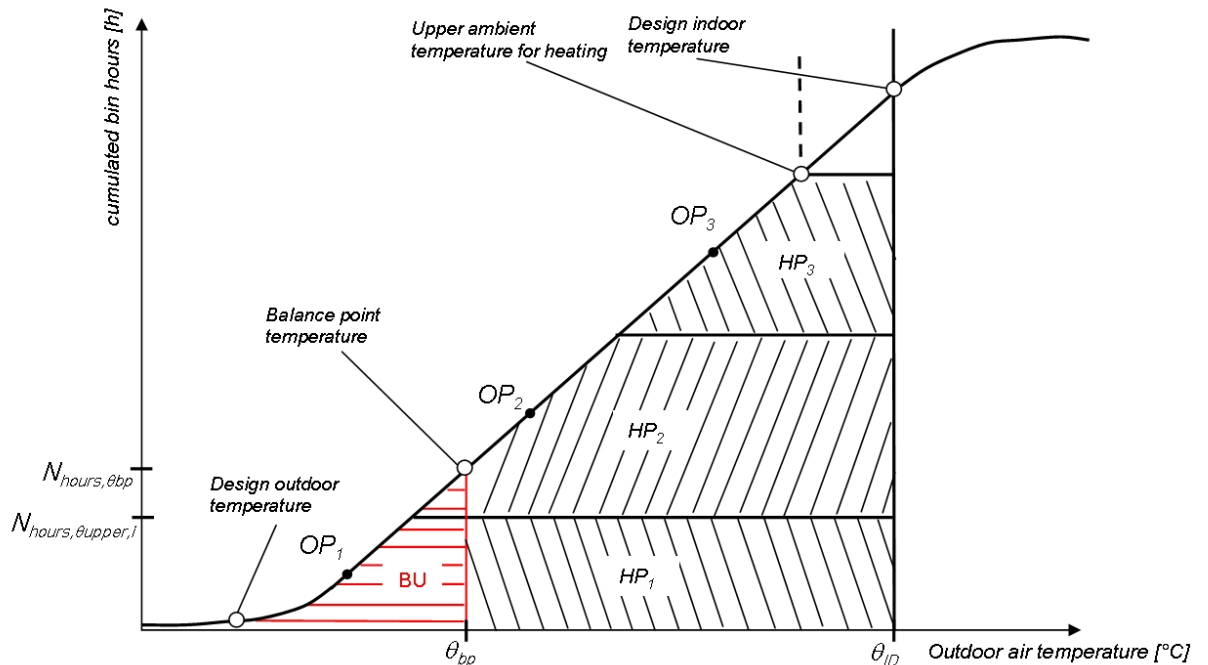


Figure A 6 – Fraction of back-up heat for parallel operating mode (HP – heat pump, OP – operating point, BU – back-up)

Figure A 6 shows the areas under cumulative annual frequency of the outdoor air temperature, which correspond to the energy fractions. The area BU approximates the energetic fraction delivered by the back-up heater. The fraction of the back-up heat for parallel operation can be approximated by the equation

For $\theta_{bp} > \theta_{upper,i}$

$$p_{bu,h,i} = \frac{A_{bu,i}}{A_i} = \frac{HDH_{\theta_{upper,i}} - (\theta_{ID} - \theta_{bp}) \cdot N_{hours,\theta_{upper,i}}}{HDH_{\theta_{upper,i}} - HDH_{\theta_{lower,i}}} \quad [-] \quad \text{eq. A 12}$$

$$p_{bu,h,i+1} = \frac{A_{bu,i+1}}{A_{i+1}} = \frac{(HDH_{\theta_{bp}} - HDH_{\theta_{lower,i+1}}) - ((\theta_{ID} - \theta_{bp}) \cdot (N_{hours,\theta_{bp}} - N_{hours,\theta_{lower,i+1}}))}{HDH_{\theta_{upper,i+1}} - HDH_{\theta_{lower,i+1}}} \quad [-] \quad \text{eq. A 13}$$

For $\theta_{bp} < \theta_{upper,i}$

$$p_{bu,h,i} = \frac{A_{bu}}{A_i} = \frac{HDH_{\theta_{bp}} - (\theta_{ID} - \theta_{bp}) \cdot N_{hours,\theta_{bp}}}{HDH_{\theta_{upper,i}} - HDH_{\theta_{lower,i}}} \quad [-] \quad \text{eq. A 14}$$

where

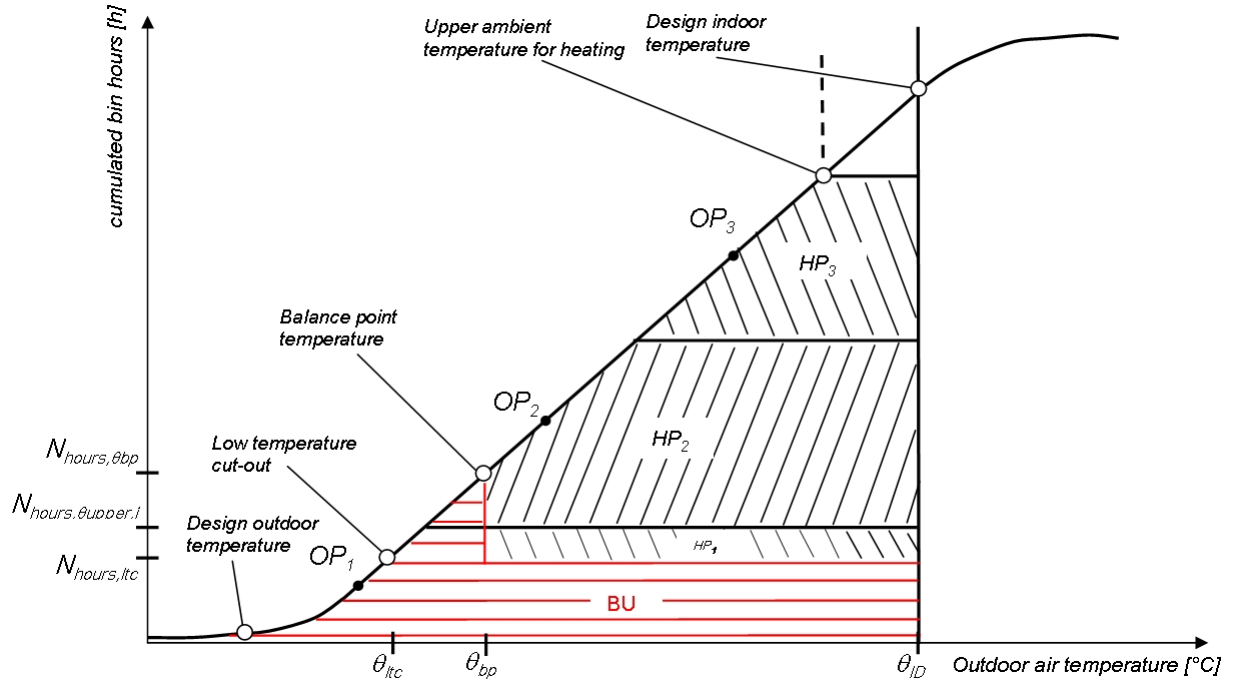
$p_{bu,h,i}$	fraction of the space heating energy covered back-up heater in bin i	(-)
$A_{bu,i}$	area BU in bin i acc. to Figure A 6	(Kh)
$A_{bu,i+1}$	area BU in bin i+1 acc. to Figure A 6	(Kh)
A_i	total area of bin i	(Kh)
θ_{bp}	balance point temperature	(°C)
θ_{ID}	indoor design temperature	(°C)
$N_{hours,\theta}$	cumulated number of hours up to the respective temperature θ	(h)
HDH_{θ}	cumulated heating degree hours up to the respective temperature θ	(Kh)

NOTE The vertical limit of A_{BU} in Figure A 6 is an approximation, since the heating capacity of the heat pump is not constant and decreases with decreasing source temperature, thus the line is inclined to higher temperatures of the outdoor air. For high balance points and air-source heat pumps the inclination gets stronger and may lead to higher back-up fractions. However, the boundary condition for the running time given in A.1.5.7.2 indicates if the approximation is not exact enough and correction is accomplished in eq. A 30.

A.1.5.5.2.3 Partly parallel operating mode of the back-up heater

In partly parallel operating mode of the back-up heater, the heat pump generator is not switched-off at the balance point temperature and runs up to the low-temperature cut-out, where the heat pump is switched-off and only the back-up heater is operated to supply the total heating requirement. Figure A 7 shows the areas under the cumulative annual frequency of the outdoor air temperature, which correspond to the energy fractions.

The area BU represents the energy fraction delivered by the back-up heater. The fraction of the back-up heat for partly parallel operation can be approximated by the equation



**Figure A 7 – Fraction of back-up heat for partly parallel operating mode
(HP – heat pump, OP – operating point, BU – back-up)**

For $\theta_{bp} > \theta_{upper,i}$

$$p_{bu,h,i} = \frac{A_{bu,i}}{A_i} = \frac{HDH_{\theta_{upper,i}} - (\theta_{ID} - \theta_{bp}) \cdot (N_{hours,\theta_{upper,i}} - N_{hours,\theta_{ltc}})}{HDH_{\theta_{upper,i}} - HDH_{\theta_{lower,i}}} \quad [-] \quad \text{eq. A 15}$$

$$p_{bu,h,i+1} = \frac{A_{bu,i+1}}{A_{i+1}} = \frac{HDH_{\theta_{bp}} - HDH_{\theta_{lower,i+1}} - ((\theta_{ID} - \theta_{bp}) \cdot (N_{hours,\theta_{bp}} - N_{hours,\theta_{lower,i+1}}))}{HDH_{\theta_{upper,i+1}} - HDH_{\theta_{lower,i+1}}} \quad [-] \quad \text{eq. A 16}$$

For $\theta_{bp} < \theta_{upper,i}$

$$p_{bu,h,i} = \frac{A_{bu}}{A_i} = \frac{HDH_{\theta_{bp}} - (\theta_{ID} - \theta_{bp}) \cdot (N_{hours,\theta_{bp}} - N_{hours,\theta_{ltc}})}{HDH_{\theta_{upper,i}}} \quad [-] \quad \text{eq. A 17}$$

where

$p_{bu,h,i}$	fraction of the space heating energy covered by the back-up generator in bin i	(-)
A_{bu}	area BU in Figure A 7	(Kh)
$A_{bu,i+1}$	area BU in bin i+1 acc. to Figure A 7	(Kh)
A_i	total area of bin i	(Kh)
θ_{bp}	balance point temperature	(°C)
θ_{ID}	indoor design temperature	(°C)
θ_{ltc}	low-temperature cut-out temperature	(°C)
$N_{hours,\theta}$	cumulated number of hours up to the respective temperature θ	(h)
HDH_{θ}	cumulated heating degree hours up to the respective temperature θ	(Kh)

A.1.5.6 Heating capacity and COP at full load (steady-state operation)

A.1.5.6.1 Heating mode

The steady state heating capacity and COP are taken from standard test results according to EN 14511 (or older values according to the former standard EN 255-2). In EN 14511 standard testing is performed at standard rating conditions and several application rating conditions. Since the COP characteristic has the most significant impact on the heat pump performance it shall be taken care, that COP-values are reliable. All available test points have to be taken into account.

To receive data for the whole range of the outdoor temperature linear inter- and extrapolation between the test points is applied both for the source as for the sink temperature. Interpolation is performed between the temperatures of the two nearest test points. Extrapolation is performed using the two nearest test points to the target point.

To receive an adequate exactness by interpolation, all available test points have to be taken into account, at least the test points prescribed by the standard testing acc. to EN 14511 (standard rating and application rating).

If not sufficient data is available from test institutes, manufacturer data measured according to EN 14511 can be used.

The source of the data shall be stated clearly in the calculation report

A.1.5.6.1.1 Temperature conditions for the interpolation of heating capacity and COP

Based on the respective measurements of the heating capacity and the COP according to EN 14511 (formerly EN 255-2), the interpolation for the actual temperature conditions at the operating point of the respective bin is performed. As stated above, interpolation is performed between the temperatures of the two nearest test points.

- In the case of outdoor air heat pump the source temperature is given by the outdoor air temperature.
- In the case of ground- or water-source heat pumps, the return temperature of the ground-loop or water-loop heat exchanger has to be used, respectively. As ground and water temperature depend on the site, they should be given in a national annex.
- In case of exhaust-air heat pumps without heat recovery, the source temperature corresponds to the indoor temperature, in case of an installed heat recovery, either combined test results of the heat pump and the heat recovery shall be used or an evaluation of the inlet temperature by the temperature change coefficient of the heat recovery according to EN 308 shall be applied.

The actual sink temperature can be calculated according to the controller settings of the heating systems. If the controller setting of the heating system is not known, typical controller settings for the sink temperature dependent on the outdoor air temperature (heating characteristic curve) for different kinds of heat emission systems are given in national building regulations.

A.1.5.6.2 Correction of COP for supply temperature

The mean bin sink temperature of the heat emission system is dependent on the heating characteristic. The evaluation of the COP is done by linear interpolation between the testing point according to EN 14511 (see paragraph A.1.5.6.1).

However, the assessment of the COP is only correct, if the volume flow rate in operation corresponds to the volume flow rate during the testing according to EN 14511, e.g. the temperature difference on the water side of the condenser has to be the same. Thus, the dimensioning of the heat emission system is to be taken into account. If different temperature differences occur between operation and standard testing, the temperature condition in the condenser is different and therefore, the COP has to be corrected according to the equation

$$\text{COP}_{\Delta\theta} = \text{COP} \cdot f_{\Delta\theta} \quad [\text{W/W}]$$

eq. A 18

where

$COP_{\Delta\theta}$	corrected COP for different temperature differences during testing and operation	(W/W)
COP	COP according to the testing according to EN 14511	(W/W)
$f_{\Delta\theta}$	correction factor according to Table A3	(-)

Table A 3 Correction factor $f_{\Delta\theta}$ for different temperature conditions at condenser (taken from [A 2])

$\Delta\theta_{si}$ in operation [K]	Temperature difference at condenser $\Delta\theta_{si}$ at measurement according EN 14511 [K]												
	3	4	5	6	7	8	9	10	11	12	13	14	15
3	1,000	0,990	0,980	0,969	0,959	0,949	0,939	0,928	0,918	0,908	0,898	0,887	0,877
4	1,010	1,000	0,990	0,980	0,969	0,959	0,949	0,939	0,928	0,918	0,908	0,898	0,887
5	1,020	1,010	1,000	0,990	0,980	0,969	0,959	0,949	0,939	0,928	0,918	0,908	0,898
6	1,031	1,020	1,010	1,000	0,990	0,980	0,969	0,959	0,949	0,939	0,928	0,918	0,908
7	1,041	1,031	1,020	1,010	1,000	0,990	0,980	0,969	0,959	0,949	0,939	0,928	0,918
8	1,051	1,041	1,031	1,020	1,010	1,000	0,990	0,980	0,969	0,959	0,949	0,939	0,928
9	1,061	1,051	1,041	1,031	1,020	1,010	1,000	0,990	0,980	0,969	0,959	0,949	0,939
10	1,072	1,061	1,051	1,041	1,031	1,020	1,010	1,000	0,990	0,980	0,969	0,959	0,949

A.1.5.6.3 Domestic hot water mode

Heat pumps for domestic hot water operation are tested as unitary systems including the domestic hot water storage in the system boundary according to the standard EN 255-3. The standard delivers the COP-value for the extraction of domestic hot water, called COP_t , in one standard test point. For reasons of nomenclature, the COP_t derived in EN 255-3 is denoted $COP_{t,DHW}$ in this standard.

This $COP_{t,DHW}$ value is only valid for the extraction of domestic hot water and not for the reheating of the storage without extraction of domestic hot water (stand-by operation), since the inlet temperature to the condenser and thus the COP of the heat pump changes significantly, if stand-by losses are considered.

However, the standard delivers an electrical power input to cover the storage losses, i.e. electrical energy consumption to cover stand-by losses can be expressed by this value.

The sink temperature conditions of the domestic hot water system may change during the year. However, for calculation purposes the sink temperature can be considered constant over the whole operating range as long as tapping temperature of the domestic hot water does not change much.

Due to varying source temperatures, the operation period and thus COP values have to be corrected for these conditions. As only one standard test point is defined in EN 255-3 a temperature correction of the COP shall be accomplished using an exergetic approach given in [A 4] or information of the heating characteristic at the respective temperatures.

If no values acc. to EN 255-3 are available, alternate combined operating system can be calculated by evaluating the heat pump characteristic at an average loading temperature of the DHW operation.

A.1.5.6.4 COP at part load operation

A.1.5.6.4.1 Heating mode

Heat pumps with fixed speed compressor operate in part load operation by cycling between on and off state. Therefore, in part load operation losses due to cycling of the compressor occur and reduce the COP of the heat pump. However, for adequate system design losses due to cycling are small and degradation of the COP is basically due to stand-by power and defrosting during off period. Cycling losses are considered negligible in the frame of this calculation. Additional stand-by input during off-periods is taken into account by the power input in A.1.5.8.1.3.

However, in this calculation approach effects of part load operation of an eventually elevated supply temperature are neglected.

A.1.5.6.4.2 Domestic hot water mode

Eventual start-up losses of the heat pump are taken into account in the $COP_{t,DHW}$ -value according to EN 255-3 and therefore no correction is required.

A.1.5.7 Operation time of the heat pump

Operation time of the heat pump depends on the heating capacity, given by the operating conditions, and the heat requirement, given by the distribution subsystem. It can be calculated by the equation

$$t_{ON,g,i} = \frac{Q_{hp,i}}{\phi_{hp,i}} \quad [s]$$

eq. A 19

where

$t_{ON,g,i}$ running time of the heat pump in bin i (s)

$Q_{hp,i}$ produced heat by the heat pump in bin i (J)
(requirement of the distribution subsystem and generation subsystem heat losses)

$\phi_{hp,i}$ heating capacity of the heat pump in bin i (W)

The produced heat of the heat pump can be calculated by the equation

$$Q_{hp,i} = (Q_{out,g,i} + Q_{l,g,i}) \cdot (1 - p_{bu,i}) \quad [J]$$

eq. A 20

where

$Q_{hp,i}$ produced heat by the heat pump in bin i (J)
(requirement of the distribution subsystem and generation subsystem heat losses)

$Q_{out,g,i}$ requirement of the distribution subsystem in bin i (J)

$Q_{l,g,i}$ generation subsystem losses in bin i (calculated in chapter A.1.5.9.2) (J)

$p_{bu,i}$ energy fraction covered by the back-up in bin i (-)

These equations can be applied to the different operation modes, since operation time of the heat pump depends on the produced heat energy, i.e. heat energy requirement and generation subsystem losses, and the operating mode of the heat generator has to be taken into account.

heat pumps operating in space heating-only or in domestic hot water-only mode

The energy is given by the actual produced energy for space heating or domestic hot water heat requirement respectively.

heat pumps operating alternately on space heating and domestic hot water generation

Total running time of the generator is determined by the sum of the produced energy for space heating and domestic hot water energy requirements, produced at the respective heating capacity of the heat pump according to the temperature conditions.

heat pumps operating simultaneously for space heating and domestic hot water generation

In case of simultaneous operating heat pumps, space heating and domestic hot water energy can be produced at the same time, so the running time has to be distinguished according to the state of operation. As the heat pump characteristic in simultaneous operation may differ significantly from the heat pump characteristic in single operation modes, adequate testing results for the characteristic in simultaneous operation have to be applied. Therefore, the three following operation modes have to be evaluated

- heating-only operation: running time is determined by the produced heat for the space heating system and the characteristic of the heat pump for space heating-only mode.
- domestic hot water-only operation: running time is determined by the produced heat for the DHW system and the DHW characteristic of the heat pump in DHW-only mode.
- simultaneous operating mode: running time is determined by the energy produced in simultaneous operation. The heating capacity of the heat pump in simultaneous operation has to be applied.

However, depending on the system configuration, not all three operation modes may occur in simultaneous operating systems. There are system configurations, for instance, where only simultaneous operation takes place in wintertime, so no space heating-only operation occurs. In this case only two characteristics have to be taken into account and the fraction of simultaneous operation is given by the given by the heating period.

Nevertheless, in the system depicted in Figure A 3 all three operation modes occur, so for the calculation the fraction of operation in the respective operation mode has to be determined.

A.1.5.7.1 Additional calculation for simultaneous operating heat pumps with three operation modes

The maximum possible simultaneous operation is characterised by the minimum of the running time for space heating and DHW operation, i.e. by the equation

$$t_{\text{ON,g,combi,max},i} = \min(t_{\text{ON,g,DHW},i}; t_{\text{ON,g,h},i}) \quad [\text{s}]$$

eq. A 21

where

$t_{\text{ON,g,h},i}$ running time to cover the space heating requirement and heat losses in bin i (s)

$t_{\text{ON,g,DHW},i}$ running time to cover the DHW requirement and DHW heat losses in bin i (s)

$t_{\text{ON,g,combi,max},i}$ max. possible running time in simultaneous operation in bin i (s)

The respective running time is calculated according to the equation

$$t_{\text{ON,g,DHW},i} = \frac{Q_{\text{hp,DHW},i}}{\phi_{\text{hp,DHW,combi},i}} \quad [\text{s}]$$

eq. A 22

where

$t_{ON,g,DHW,i}$	running time in DHW operation in bin i	(s)
$Q_{hp,DHW,i}$	produced DHW energy by the heat pump in bin i	(J)
$\phi_{hp,DHW,combi,i}$	DHW heating capacity of the heat pump in simultaneous operation	(W)

The same equation can be applied for the space heating mode.

The running time in simultaneous operation mode may also be influenced by the control and the load profile. The control system may have the larger impact, since controller setting defines the time periods, where simultaneous operation takes place. However, this depends strongly on the system configuration and can only be taken into account by a specific correction factor, leading to the equation

$$t_{ON,g,combi,i} = f_{combi} \cdot t_{ON,g,combi,max,i} \quad [s]$$

eq. A 23

where

$t_{ON,g,combi,i}$	running time in simultaneous operation in bin i	(s)
$t_{ON,g,combi,max,i}$	maximum possible running time in simultaneous operation	(s)
f_{combi}	correction factor taking into account the impact of the control system and the shift in the demand. Adequate factors for typical controller setting shall be given in a national Annex based on a specific evaluation of the system configuration. If no values are given, $f_{combi} = 1$ shall be set (i.e. influence of control system neglected)	(-)

The energy produced in simultaneous operation is calculated by the equation

$$Q_{hp,DHW,combi,i} = \phi_{hp,DHW,combi,i} \cdot t_{ON,g,combi,i} \quad [J]$$

eq. A 24

where

$Q_{hp,DHW,combi,i}$	produced DHW energy by the heat pump in simultaneous operation in bin i	(J)
$\phi_{hp,DHW,combi,i}$	DHW heating capacity of the heat pump in simultaneous operation	(W)
$t_{ON,g,combi,i}$	running time in simultaneous operation in bin i	(s)

The rest of the energy requirement is produced in single operation and can be determined by the following equation

$$Q_{hp,DHW,sin,i} = Q_{hp,DHW,i} - \phi_{hp,DHW,combi,i} \cdot t_{ON,g,combi,i} \quad [J]$$

eq. A 25

where

$Q_{hp,DHW,sin,i}$	produced energy by the heat pump in DHW-only operation in bin i	(J)
$Q_{hp,DHW,i}$	produced DHW energy by the heat pump in bin i	(J)
$\phi_{hp,DHW,combi,i}$	DHW heating capacity of the heat pump in simultaneous operation	(W)
$t_{ON,g,combi,i}$	running time in simultaneous operation in bin i	(s)

The allocation of the storage losses to the different operation modes is done by the fraction of simultaneous operation, which is equal to f_{combi} .

The used energy in DHW-only and in simultaneous operation can be calculated by subtracting the storage losses according to the equation

$$Q_{out, hp, DHW, sin, i} = Q_{hp, DHW, sin, i} - Q_{l, s, DHW, i} \cdot (1 - p_{bu, DHW, i}) \cdot (1 - f_{combi, i}) \quad [J]$$

eq. A 26

where

$Q_{out, hp, DHW, sin, i}$	DHW requirement covered by the heat pump in DHW-only operation in bin i	(J)
$Q_{hp, DHW, sin, i}$	produced DHW energy by the heat pump in DHW-only operation in bin i	(J)
$Q_{l, s, DHW, i}$	DHW storage losses in bin i (calculated in A.1.5.9.2.1)	(J)
$p_{bu, DHW, i}$	fraction of DHW energy covered by the back-up in bin i	(-)
$f_{combi, i}$	fraction of simultaneous operation in bin i	(-)

$$Q_{out, hp, DHW, combi, i} = Q_{hp, DHW, combi, i} - Q_{l, s, DHW, i} \cdot (1 - p_{bu, DHW, i}) \cdot f_{combi, i} \quad [J]$$

eq. A 27

where

$Q_{out, hp, DHW, combi, i}$	DHW requirement covered by the heat pump in simultaneous operation in bin i	(J)
$Q_{hp, DHW, combi, i}$	produced DHW energy by the heat pump in simultaneous operation in bin i	(J)
$Q_{l, s, DHW, i}$	DHW storage losses in bin i (calculated in A.1.5.9.2.2)	(J)
$p_{bu, DHW, i}$	fraction of DHW energy covered by the back-up in bin i	(-)
$f_{combi, i}$	fraction of simultaneous operation in bin i	(-)

The running time in space heating-only and DHW-only operation can be calculated by the equation

$$t_{ON, g, sin, i} = \frac{Q_{hp, sin, i}}{\phi_{hp, sin, i}} \quad [s]$$

eq. A 28

where

$t_{ON, g, sin, i}$	running time in space heating or DHW-only operation, respectively	(s)
$Q_{hp, sin, i}$	heat requirement and generator losses in the respective operation mode	(J)
$\phi_{hp, sin, i}$	heating capacity of the heat pump in the respective operation mode	(W)

NOTE Up to now EN 255-3 does not deliver an heating capacity for the domestic hot water operation, thus it is proposed as modification of EN 255-3 to provide the needed heating capacity as proposed in the test procedure in part B, chapter 4.2.

A.1.5.7.2 Boundary condition for the running time

The total running time must not be longer than the effective bin time, thus the total running time has to fulfil the boundary condition

$$t_{ON, g, t, i} = \min(t_{i, eff} ; t_{ON, g, h, sin, i} + t_{ON, g, DHW, sin, i} + t_{ON, g, combi, i}) \quad [s]$$

eq. A 29

where

$t_{ON, g, t, i}$	total running time of the heat pump in bin i	(s)
$t_{i, eff}$	effective bin time in bin i	(s)

$t_{ON,g,h,sin,i}$	running time in space heating-only operation in bin i	(s)
$t_{ON,g,DHW,sin,i}$	running time in DHW-only operation in bin i	(s)
$t_{ON,g,combi,i}$	running time in simultaneous operation in bin i	(s)

If the calculated total running time is longer than the effective bin time, this is due to a lack of heating capacity of the heat pump. In this case the running time is equal to the effective bin time and the missing back-up energy is calculated according to chapter A.1.5.7.3.

A.1.5.7.3 Calculation of additional back-up energy due to lack of capacity

The additional back-up energy due to a lack of capacity is calculated by multiplying the missing running time with the heating capacity of the heat pump in SH-only or DHW-only operation according to the equation

$$Q_{bu, cap, i} = (t_{ON, g, t, i} - t_{i, eff}) \cdot \phi_{hp, sin, i} \quad [J] \quad \text{eq. A 30}$$

where

$Q_{bu, cap, i}$	additional back-up energy due a lack of capacity	(J)
$t_{ON, g, t, i}$	total running time of the heat pump in bin i	(s)
$t_{i, eff}$	effective bin time in bin i	(s)
$\phi_{hp, sin, i}$	heating capacity of the heat pump in the respective single operation mode	(W)

The control strategy determines if the back-up energy is supplied to the space heating or the domestic hot water systems. If no control strategy is known, it is assumed, that the back-up heater supplies 50% of the back-up energy to the space heating system and 50% of the back-up energy to the DHW system.

The total fraction of back-up energy can be calculated according to the equation

$$p_{bu, i} = \frac{Q_{bu, op, i} + Q_{bu, cap, i}}{Q_{out, g, i}} = p_{bu, op, i} + p_{bu, cap, i} + \frac{(t_{ON, g, t, i} - t_{i, eff}) \cdot \phi_{hp, sin, i}}{Q_{out, g, i}} \quad [-] \quad \text{eq. A 31}$$

where

$p_{bu, i}$	fraction of heat energy covered by the back-up heater in bin i SH or DHW mode respectively	(-)
$Q_{bu, op, i}$	Back-up energy due to operation limit temperature	(J)
$Q_{bu, cap, i}$	Back-up energy due to lack of capacity of the heat pump	(J)
$Q_{out, g, i}$	heat energy requirement of the distribution subsystem in bin i	(J)
$p_{bu, op, i}$	fraction of back-up energy due to temperature operation limit	(-)
$p_{bu, cap, i}$	fraction of back-up energy due to lack of capacity (delivered by chap. A.1.5.5.2)	(-)
$t_{ON, g, t, i}$	total (required) running time of the heat pump in bin i	(s)
$t_{i, eff}$	effective bin time in bin i	(s)
$\phi_{hp, sin, i}$	heating capacity of the heat pump in single operation	(W)

To derive the fraction of back-up energy for the respective operation modes, the respective values (energy, heating capacity) for the operation mode have to be set in eq. A 31. The respective fractions due to the operation limit temperature of the heat pump and due to the lack of capacity have to be added for the respective operation modes. For the space heating mode, there may be already a fraction of back-up energy delivered by the approximation made in chap. A.1.5.5.2, which is considered in eq. A 31, by the term $p_{bu, cap, i}$.

A.1.5.8 Auxiliary energy

A.1.5.8.1 Heating mode

Auxiliary input to operate the heat pump depends on the type of heat pump (e.g. air-to-water, liquid-to-water) and the system configuration (with or without storage, combined domestic hot water production etc.). Auxiliary energy is required e.g. to operate the source pump of the heat pump.

Circulation pumps are accounted to the generation system, in case of a hydraulic decoupling of the generation and the distribution subsystems, e.g. by a storage in parallel or a hydraulic distributor. In case of no hydraulic decoupling the circulation pump of the heat sink is accounted to the distribution part.

Operation of the source pump is associated with the heat pump generator operation during operation time t_{ON} .

Operation of the primary pump is associated with the system configuration and the control strategy. It is either running through the whole activation time of the generator, i.e. the heating period, or it is related to the running time of the generator, if a corresponding control strategy is applied.

In case that the system contains a heating buffer storage in parallel, the operation of the primary pump is related to the running time of the heat pump t_{ON} .

Calculation is performed according to the following equation.

$$W_g = P_{aux,g} \cdot t_{ON,aux} \quad [J]$$

eq. A 32

where

W_g auxiliary energy (J)

$P_{aux,g}$ power of the auxiliary component (W)

$t_{ON,aux}$ running time of the respective auxiliary component (s)

In case of pump energies of the source and the sink pump, the parts that are already included in the EN 14511 (formerly EN 255) have to be corrected by the equation

$$P_{int} = \frac{\Delta p \cdot \dot{V}}{\eta_{aux}} \quad [W]$$

eq. A 33

where

P_{int} power to overcome the internal pressure drop in the respective component (W)

Δp pressure drop in the respective part of the heat pump (evaporator, condenser) – (Pa)
this value should be taken from measurements according to EN 14511 or manufacturer data

\dot{V} rated flow rate of the respective component – this values should be taken from measurements of EN 14511 or manufacturer data (m³/s)

η_{aux} efficiency of the auxiliary component (-)

EN 14511 sets a fixed pump efficiency η_{aux} of 0.3.

NOTE If the test is made according to the proposed test procedure in part B chapter B.4.1, the correction should be made using a measured pump efficiency or 0.3 if the heat pump has been tested with the pumps. If the test has been made without the pumps, an efficiency of 0.1 should be used, thereby encouraging to include the pumps in the test.

So the resulting power is calculated by the equation

$$P_{aux,g} = P_{aux,nom} - P_{int} \quad [W]$$

eq. A 34

where

$P_{aux,g}$	power of the auxiliary component	(W)
$P_{aux,nom}$	nominal power of the component	(W)
P_{int}	internal power due to the pressure drop in the component (condenser, evaporator) used for correction	(W)

A.1.5.8.1.1 Air-to-water heat pump

Air-to-water heat pumps are tested as unitary components, so that the auxiliary energy for fans at the source side is already taken into account during testing according to EN 14511. Thus, only the external power of the sink side has to be considered for air-to-water heat pumps in case of a hydraulic decoupling.

A.1.5.8.1.2 Liquid-to-water heat pump

In the case of liquid-to-water heat pumps the source pump is only partly considered in the standard testing values, thus the remaining electrical energy input of the source pump has to be taken into account. As operation of the source pump is associated with the generator, energy input can be calculated by eq. A 32 to eq. A 34, where $t_{ON,aux}$ is to be set as the heat pump running time $t_{ON,g}$ as calculated in eq. A 19.

A.1.5.8.1.3 Energy consumption of additional auxiliary systems

Control and an eventually installed carter heating are activated during the whole heating period. However, during the operating time of the heat pump, the energy for the control system is already included in the COP values according to EN 14511. Thus, the control system has to be taken into account only for the stand-by operation of the heat pump.

$$W_{g,sb} = P_{sb} \cdot t_{sb} \quad [J]$$

eq. A 35

where

$W_{g,sb}$	auxiliary energy consumption in stand-by operation	(J)
P_{sb}	nominal power consumption in stand-by operation	(W)
t_{sb}	stand-by period of the heat pump operation	(s)

The stand-by time t_{sb} differs for the operating modes:

- In space heating-only system configurations the stand-by time can be evaluated by the heating period minus the operating time of the heat pump for space heating.
- In DHW-only system configurations (heat pump water heater), the stand-by time can be evaluated by the whole year diminished by the operating time of the heat pump for domestic hot water production.
- In system configurations for both heating and domestic hot water the stand-by time can be evaluated as the whole year diminished by the total operating time of the heat pump, e.g. the sum for space heating operating time and domestic hot water operation time. An allocation of the stand-by operation to the single operating modes can be done as for the single system configurations.

A.1.5.8.2 Domestic hot water mode

As the storage loading pump is already included in the $COP_{t,DHW}$ -value, only the source pump has to be taken into account. The source pump is associated to the running time of the generator and can be calculated to eq. A 32 to eq. A 34, where $t_{ON,aux}$ is to be set as the generator running time $t_{ON,g}$ as calculated in eq. A 19.

A.1.5.8.3 Calculation of auxiliary energy input

Total auxiliary input in the bin i is calculated by summing up the single parts depending on the type of generator and the system configuration according to the equation

$$W_g = W_{g, so} + W_{g, si} + W_{g, sb} \quad [J]$$

eq. A 36

where

W_g	auxiliary energy consumption	(J)
$W_{g,so}$	auxiliary energy for source pump	(J)
$W_{g,si}$	auxiliary energy for primary pump (in case of a hydraulic decoupling)	(J)
$W_{g,sb}$	auxiliary energy for stand-by operation	(J)

A.1.5.9 Heat generator loss and recoverable heat generator loss

A.1.5.9.1 Auxiliary consumption

Depending on the component auxiliary energy may be partly transformed to mechanical energy and partly to thermal energy. The fraction of the used mechanical energy is given by the component efficiency (η_{aux}).

The thermal fraction of the auxiliary energy transmitted to the heat transfer medium (e.g. water, brine in source cycle) is assumed to be totally recovered and is already considered in the COP value according to EN 14511.

The losses to the ambience can be calculated

$$W_{l,g} = W_g \cdot p_{aux,g} \quad [J]$$

eq. A 37

where

$W_{l,g}$	auxiliary energy as heat losses to the ambience	(J)
W_g	auxiliary energy consumption of the generator	(J)
$p_{aux,g}$	fraction of heat loss of auxiliary components to the ambience. This value should be defined in a national Annex.	(-)

Losses to the heated space are assumed recoverable and can be calculated by

$$W_{l,g,rl} = W_g \cdot p_{aux,g} \cdot (1 - b) \quad [J]$$

eq. A 38

where

$W_{l,g,rl}$	recoverable auxiliary energy losses to the ambience	(J)
W_g	auxiliary energy consumption of the generator	(J)
$p_{aux,g}$	fraction of heat loss of auxiliary components to the ambience. This value should be given in a national Annex.	(-)
b	temperature reduction factor linked to location. This value should be given in a national Annex.	(-)

A.1.5.9.2 Generator heat losses

A.1.5.9.2.1 Heating mode

In case of a heat pump without attached storages, the losses to the ambience are negligible, since a heat pump normally is delivered in an insulated housing. Thus, for heating operation no envelope losses are taken into account in case no heating buffer storage is installed.

A possibly installed heating buffer storage, however, produces losses, that might be recoverable depending on the location of the storage. The losses to the ambience can be calculated by

$$Q_{l,s,h,i} = \frac{\theta_{s,avg,i} - \theta_{amb}}{\Delta\theta_{s, sb}} \cdot \frac{Q_{s, sb} \cdot 1000 \cdot t_i}{24} \quad [J]$$

eq. A 39

where

$Q_{l,s,h,i}$	generation subsystem heat loss due to heating buffer storage loss to the ambience in bin i	(J)
$\theta_{s,avg,i}$	average storage temperature in bin i	(°C)
θ_{amb}	ambient temperature at the storage location	(°C)
$\Delta\theta_{s, sb}$	temperature difference due to storage stand-by test conditions	(K)
$Q_{s, sb}$	stand-by heat loss	(kWh/d)
t_i	bin time	(s)
n_{bins}	number of bins	(-)

If no values of the storage losses are available, upper limits for the storage losses depending on the storage volume are to be used, e.g. as given in [A 3].

The average storage temperature $\theta_{s,avg,i}$ is to be determined according to the storage control. It is approximated as average temperature of the supply- and return temperature of the space heating system, if the storage is operated dependent on the temperature requirements of the heating system according to the equation

$$\theta_{s,avg,i} = \frac{\theta_{si,i} + \theta_{ri,i}}{2} \quad [^{\circ}C]$$

eq. A 40

where

$\theta_{s,avg,i}$	average storage temperature of the heating buffer storage	(°C)
$\theta_{si,i}$	supply temperature of the heating system in bin i	(°C)
$\theta_{ri,i}$	return temperature of the heating system in bin i	(°C)

The return temperature is evaluated based on the return temperature at design conditions.

Storage envelope losses are considered recoverable. Recoverable storage losses can be calculated according to the equation

$$Q_{l,g,h,rl} = Q_{l,s,h} \cdot (1 - b) \quad [J]$$

eq. A 41

where

$Q_{l,g,h,rl}$	recoverable losses of the generation subsystem	(J)
$Q_{l,s,h}$	heat losses of the heating buffer storage	(J)
b	temperature reduction factor linked to location. The values of b should be defined in a national Annex.	(-)

A.1.5.9.2.2 Domestic hot water mode

Losses of the domestic hot water storage shall be evaluated in the same way as for the space heating mode for an average DHW storage temperature. The average storage temperature depends on the applied storage control, the position of the heat exchangers and the temperature sensors, etc.. It shall be determined based on the product information. If no values are available, the average storage temperature can be estimated by 90% hot water reference temperature.

Storage envelope losses are considered recoverable. Recoverable storage losses can be calculated according to the equation

$$Q_{l,g,DHW,rl} = Q_{l,s,DHW} \cdot (1 - b) \quad [J] \quad \text{eq. A 42}$$

where

$Q_{l,g,DHW,rl}$	recoverable losses of the generator	(J)
$Q_{l,s,DHW}$	heat losses of the generator by DHW storage losses	(J)
b	temperature reduction factor linked to location. This value should be given in a national annex.	(-)

A.1.5.9.3 Total generator losses and total recoverable losses

The total heat losses of the generation subsystem can be obtained by a summation according to the equation

$$Q_{l,g,t} = Q_{l,s,h} + Q_{l,s,DHW} + W_{l,g} \quad [J] \quad \text{eq. A 43}$$

where

$Q_{l,g,t}$	total generation and storage subsystem losses	(J)
$Q_{l,s,h}$	generation and storage subsystem losses for heating	(J)
$Q_{l,s,DHW}$	generation and storage subsystem losses for DHW	(J)
$W_{l,g}$	total auxiliary energy losses to the ambience	(J)

The total recoverable losses can be obtained by a summation according to the equation

$$Q_{l,g,rl} = Q_{l,g,h,rl} + Q_{l,g,DHW,rl} + W_{l,g,rl} \quad [J] \quad \text{eq. A 44}$$

where

$Q_{l,g,rl}$	total recoverable generation and storage subsystem losses	(J)
$Q_{l,g,h,rl}$	recoverable generation and storage subsystem losses of space heating	(J)
$Q_{l,g,DHW,rl}$	recoverable generation and storage subsystem losses of DHW	(J)
$W_{l,g,rl}$	recoverable heat losses of auxiliary energy	(J)

A.1.5.10 Calculation of total energy input

A.1.5.10.1 Electrical energy input to the heat pump for heating operation

The electrical energy input to the heat pump can be calculated by summing-up the electrical energy input of the respective bins according to the equation

$$E_{in,g,h} = \sum_{i=1}^{n_{bin}} \frac{Q_{hp,h,sin,i}}{COP_{sin,i}} + \sum_{i=1}^{n_{bin}} \frac{Q_{hp,h,combi,i}}{COP_{combi,i}} \quad [J]$$

eq. A 45

where

$E_{in,g,h}$	electrical energy input to cover the heat requirement of the SH distribution subsystem	(J)
$Q_{hp,h,sin,i}$	produced energy of the heat pump in space heating-only operation in bin i	(J)
$Q_{hp,h,combi,i}$	produced energy of the heat pump in simultaneous operation in bin i	(J)
$COP_{sin,i}$	coefficient of performance for single operation at operating point in the respective bin, taken as performance factor for the whole bin	(-)
$COP_{combi,i}$	coefficient of performance for simultaneous operation at operating point in the respective bin, taken as performance factor for the whole bin	(-)
n_{bin}	number of bins	(-)

A.1.5.10.2 Electrical energy input to the heat pump for DHW operation

The electrical energy input for DHW operation can be calculated according to the equation

$$E_{in,g,DHW} = \sum_{i=1}^{n_{bin}} \left(\frac{Q_{out,hp,DHW,sin,i}}{COP_{t,DHW,sin,i}} + P_{es,sin,i} \cdot t_{sin} \right) + \sum_{i=1}^{n_{bin}} \left(\frac{Q_{out,hp,DHW,combi,i}}{COP_{t,DHW,combi,i}} + P_{es,combi,i} \cdot t_{combi} \right) \quad [J]$$

eq. A 46

where

$E_{in,g,DHW}$	electrical energy input to cover the heat requirement of the DHW distribution subsystem	(J)
$Q_{out,hp,DHW,sin,i}$	heat energy requirement covered by the heat pump in DHW-only operation	(J)
$Q_{out,hp,DHW,combi,i}$	heat energy requirement covered by the heat pump in simultaneous operation	(J)
$COP_{t,DHW,sin,i}$	coefficient of performance for the extraction of domestic hot water in DHW-only operation according to EN 255-3 taken as performance factor for the whole bin	(-)
$COP_{t,DHW,combi,i}$	coefficient of performance for the extraction of domestic hot water in simultaneous operation taken as performance factor for the whole bin	(-)
$P_{es,i}$	electrical power input to cover storage losses in single or simultaneous operation according to EN 255-3	(W)
t	time in simultaneous or DHW-only operation in bin i	(s)
n_{bin}	number of bins	(-)

The repartition of the total time in bin i simultaneous and DHW-only operation is done as for the storage losses (see chap. A.1.5.7.1)

If no values acc. to EN 255-3 are available, the DHW calculation is performed in the same way as the space heating mode.

A.1.5.10.3 Electrical energy input to back-up heater

$$E_{in,g,bu} = \sum_{i=1}^{n_{bin}} \frac{(Q_{out,g,h,i} + Q_{l,g,s,h,i}) \cdot p_{bu,h,i}}{\eta_{bu,h}} + \frac{(Q_{out,g,DHW,i} + Q_{l,g,s,DHW,i}) \cdot p_{bu,DHW,i}}{\eta_{bu,DHW}} \quad [J]$$

eq. A 47

where

$E_{in,g,bu}$	electrical energy input to operate the back-up heater	(J)
$Q_{out,g,h,i}$	heat energy requirement of the space heating distribution subsystem in bin i	(J)
$Q_{l,g,s,h,i}$	heat losses of heating buffer storage in bin i	(J)
$p_{bu,h,i}$	fraction of heat energy for space heating covered by back-up heater in bin i	(-)
$Q_{out,g,DHW,i}$	heat energy requirement of the DHW distribution subsystem in bin i	(J)
$Q_{l,g,s,DHW,i}$	heat losses of the DHW storage in bin i	(J)
$p_{bu,DHW,i}$	fraction of DHW heat energy covered by back-up heater in bin i	(-)
$\eta_{bu,h}$	efficiency of the electrical back-up heating for space heating	(-)
$\eta_{bu,DHW}$	efficiency of the electrical back-up heating for DHW production	(-)

A.1.5.10.4 Total energy input to the generator to cover the heat requirement

The total electrical energy input is calculated as sum according to the equation

$$E_{in,g} = E_{in,g,h} + E_{in,g,DHW} + E_{in,g,bu} \quad [J]$$

eq. A 48

where

$E_{in,g}$	total electrical energy input to cover the heat requirement of the distribution subsystem	(J)
$E_{in,g,h}$	electrical energy input to operate the heat pump in heating mode	(J)
$E_{in,g,DHW}$	electrical energy input to operate the heat pump in DHW mode	(J)
$E_{in,g,bu}$	electrical back-up energy input	(J)

A.1.5.10.5 Seasonal Performance factor of the generation subsystem

The seasonal performance factor can be calculated for the single operation, e.g. heating and domestic hot water and combined to an overall seasonal performance, or directly calculated as an overall seasonal performance.

Overall seasonal performance is calculated according to the equation

$$SPF_{g,t} = \frac{Q_{out,g,h} + Q_{out,g,DHW}}{E_{in,g} + W_g} \quad [-]$$

eq. A 49

where

$SPF_{g,t}$	total seasonal performance factor of generation subsystem	(-)
$Q_{out,g,h}$	space heating energy requirement of the distribution subsystem	(J)
$Q_{out,g,DHW}$	DHW energy requirement of the distribution subsystem	(J)
$E_{in,g}$	total electrical energy input	(J)
W_g	total auxiliary energy input	(J)

As the seasonal performance factor is the reciprocal value of the expenditure factor it can be also calculated by the equation

$$e_g = \frac{1}{SPF_{g,t}} \quad [-]$$

eq. A 50

where:

e_g	expenditure factor for the generator	(-)
$SPF_{g,t}$	total seasonal performance factor	(-)

A.1.5.11 Output of the calculation

Based on the input data in chap. A.1.5.2, the following output data are calculated

- required electrical energy $E_{in,g}$ to meet the heating and/or domestic hot water requirements (chapter **A.1.5.10.3**)
- total generator heat loss $Q_{l,g,t}$ (chapter **A.1.5.9.3**)
- required auxiliary energy W_g to operate the generation subsystem (chapter **A.1.5.8.3**)
- total recoverable generation subsystem heat losses $Q_{l,g,rl}$ (chapter **A.1.5.9.3**)
- seasonal performance factor $SPF_{g,t}$ (chapter **A.1.5.10.5**)

Bibliography

- [A 1] Software Meteonorm, Version 5.1, Meteotest 2005, Information on www.meteotest.ch
- [A 2] VDI 4650, Blatt 1: Calculation of heat pumps short procedure for the calculation of the seasonal performance factor (SPF) of heat pumps - Electric heat pumps for room heating, VDI Verlag, Düsseldorf, 2003
- [A 3] Swiss Energy directive EnV of 7. december 1998 (actual status on 30. November 2004), 730.01
- [A 4] Afjei et al., Low cost low temperature heating with heat pumps, Part 4: Technical Handbook, Final report of SFOE research project, Dec. 2000

A.2 Extensions of the calculation proposal

Due to the increasing market relevance of compact heat pump units with ventilation heat recovery the calculation approach of the bin-method has been extend to cover compact units.

However, due to the structure of the European standardisation, it will probably not be possible to integrate the extension in the framework of prEN15316, since ventilation is treated in another Technical Committee of CEN and currently, there is no work item for compact units. Nevertheless, the approach can be given as recommendation for a future extension of the calculation method.

A.2.1 Extension for compact units

A.2.1.1 Assumption for the calculation of compact units

As for the calculation of the heat pump certain assumptions and simplifications are made in order to keep the calculation simple.

To calculate the heat recovery, it is a necessary assumption to postulate the operation time of the heat recovery in order to avoid iterations.

- The heat recovery is postulated to run through the entire heating period.
This assumption seems realistic, since in wintertime, normally no window-airing is applied to minimize ventilation energy losses.
- The control system is optimised in that way, that the heat recovery is operated and reduces the heating energy requirement. For the rest of heat requirement the heat pump is applied to the limit of the heating capacity and only the rest is covered by the electrical back-up heating.

These two assumptions reflect the usual operation of compact units.

A.2.1.2 Calculation steps

In compact units a heat pump is connected to the heat recovery unit of the ventilation system. The heat pump supplies heat to both the heating and the DHW system. The combination of a heat pump and a heat recovery unit has the following consequences:

- The inlet temperature of the heat pump is influenced by the outlet temperature of the heat recovery unit
- The heating requirement of the house is reduced by the heat energy recovery of parts of the ventilation losses by the heat recovery unit, thus the heat pump has to deliver less energy in heating mode.

Component characteristic of the heat recovery unit has to be delivered by testing of the heat recovery unit. An adequate test procedure is contained in the proposal in chapter B, paragraph B.4.4.1. The modified scheme of the bin method taking into account the reduction in the heating requirement is shown in Figure A 8. It illustrates how the heat recovery unit is accounted.

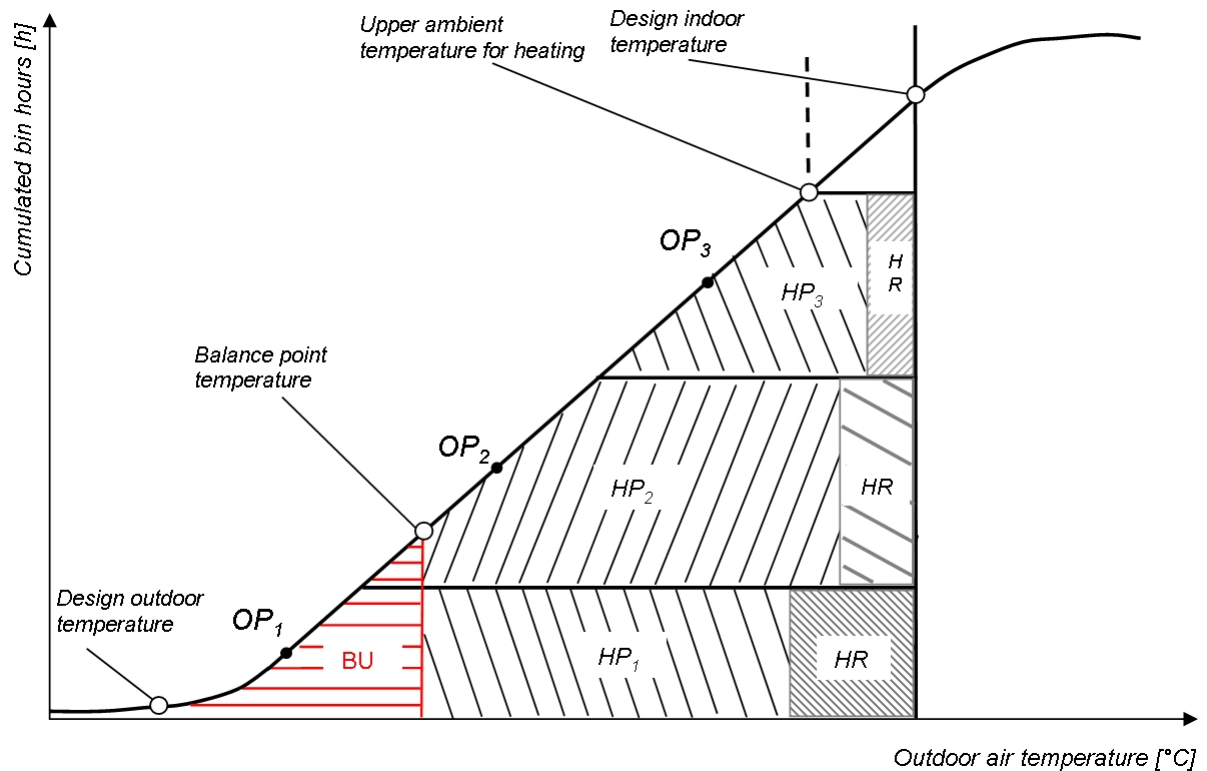


Figure A 8 - Extended bin-method for heat pump compact units

The quantity of the heat reduction and the respective electrical energy input for this reduction by the heat recovery unit has to be determined by the test procedure. The used value is the temperature change coefficient.

It can be defined for the supply air side and the return air side of the heat recovery unit by the following equation:

$$\Phi_{oa} = \frac{\theta_{sa} - \theta_{oa}}{\theta_{ra} - \theta_{oa}} \quad \Phi_{ra} = \frac{\theta_{ra} - \theta_{ea}}{\theta_{ra} - \theta_{oa}} \quad [-]$$

eq. A 51

where

Φ_{oa}	temperature change coefficient on the air inlet side (air flow from outside to building)	(-)
Φ_{ra}	temperature change coefficient on the air outlet side (air flow from building to outside)	(-)
θ_{ea}	air temperature of exhaust air exiting the heat recovery unit	(°C)
θ_{sa}	air temperature of supply air exiting the heat recovery unit	(°C)
θ_{oa}	outdoor air temperature	(°C)
θ_{ra}	air temperature of return air from the building (exiting the building/entering the HR)	(°C)

The electrical energy consumption of the heat recovery unit can be evaluated by multiplying the running time and the fan power. A corresponding kind of efficiency, called the electro-thermal amplification factor, is the ratio between the enthalpy flows recovered by the heat recovery unit to the electrical energy input for the heat recovery unit.

To consider the heat recovery unit, the following calculation steps have to be integrated in the calculation methods of chapter A.1. The energy recovered by the heat recovery can be described by the enthalpy flow entering the building. It can be calculated by the equation

$$Q_{hr,i} = \rho_{a,i} \cdot \dot{V}_{sa,i} \cdot c_{p1+x} \cdot \Phi_{oa} \cdot (HDH_{\theta upper,i} - HDH_{\theta lower,i}) \quad [J] \quad \text{eq. A 52}$$

where

$Q_{hr,i}$	heat energy recovered by the heat recovery unit in bin i	(J)
$\rho_{a,i}$	density of the air at temperature conditions in bin i	(kg/m ³)
$\dot{V}_{sa,i}$	volume flow rate of the supply air in bin i	(m ³ /s)
c_{p1+x}	specific heat capacity of moist air at conditions of the operating point	(J/(kg·K))
Φ_{oa}	temperature change coefficient for outside air to supply air	(-)
$HDH_{\theta upper,i}$	heating degree hours cumulated up to the upper temperature limit of bin i	(Kh)
$HDH_{\theta lower,i}$	heating degree hours cumulated up to the lower temperature limit of bin i	(Kh)

The recovered heat energy by the heat recovery unit has to be taken into account in the calculation of the weighting factor for the heat pump operation

$$p_{hr,i} = \frac{Q_{hr,i}}{Q_{out,g,h}} \quad [-] \quad \text{eq. A 53}$$

where

$p_{hr,i}$	weighting factor for the operation of the heat recovery unit in bin i	(-)
$Q_{out,g,h}$	heat requirement of the space heating distribution subsystem	(J)
$Q_{hr,i}$	heat energy recovered by the heat recovery unit in bin i	(J)

The space heating requirement to be covered by the heat pump in bin i is calculated according to eq. A 3. Therefore, the fraction of energy to be covered by the heat pump is calculated according to the equation

$$w_{hp,i} = \frac{HDH_{\theta upper,i} - HDH_{\theta lower,i}}{HDH_{\theta tlh}} - p_{bu,h,i} - p_{hr,i} \quad [-] \quad \text{eq. A 54}$$

where

$w_{hp,i}$	weighting factor for the heat pump	(-)
$HDH_{\theta upper,i}$	cumulated heating degree hours up to the upper temperature limit of bin i	(Kh)
$HDH_{\theta lower,i}$	cumulated heating degree hours up to the lower temperature limit of bin i	(Kh)
$HDH_{\theta tlh}$	total heating degree hours up to the upper temperature limit for heating	(Kh)
$p_{bu,h,i}$	energy fraction of the heating requirement delivered by the back-up heating in bin i	(-)
$p_{hr,i}$	energy fraction of heating requirement delivered by the heat recovery unit in bin i	(-)

If the test is made as combined test of the heat recovery unit and heat pump according to the proposed test procedure in part B, chapter B.4.4.1.3, the impact on the temperature at the inlet of the heat pump mentioned in the beginning of this chapter is already included.

If testing is performed separately for heat recovery and the heat pump unit, the impact can be considered depending on the type of heat pump.

In case of a heat pump using only the outlet air exiting the heat recovery unit, the temperature at the inlet of the heat pump can be calculated using the temperature change coefficient from return to exhaust air according the equation

$$\theta_{ea,i} = \theta_{ra,i} - \Phi_{ra,i}(\theta_{ra,i} - \theta_{oa,i}) \quad [^{\circ}\text{C}]$$

eq. A 55

where

$\theta_{ea,i}$	temperature of the outlet air of the heat recovery unit	($^{\circ}\text{C}$)
$\Phi_{ra,i}$	temperature change coefficient for return air to exhaust air in bin i	(-)
$\theta_{ra,i}$	indoor air temperature	($^{\circ}\text{C}$)
$\theta_{oa,i}$	outside air temperature	($^{\circ}\text{C}$)

In case of a heat pump using a mix of outlet air of the heat recovery unit and outdoor air, the temperature conditions at the inlet of the heat pump can be calculated for a mix of outdoor air and the outlet air of the heat recovery unit by an enthalpy balance according to the equation

$$\theta_{in,hp} = \frac{\dot{m}_{ea} \cdot c_{p,1+x} \cdot \theta_{ea,i} + \dot{m}_{oa} \cdot c_{p,1+x} \cdot \theta_{oa,i}}{\dot{m}_{ea} c_{p,1+x} + \dot{m}_{oa} c_{p,1+x}} \quad [^{\circ}\text{C}]$$

eq. A 56

where

$\theta_{in,hp}$	inlet temperature of the heat pump evaporator	($^{\circ}\text{C}$)
$\theta_{ea,i}$	outlet air temperature of the heat recovery unit in bin i	($^{\circ}\text{C}$)
$\theta_{oa,i}$	outdoor air temperature in bin i	($^{\circ}\text{C}$)
\dot{m}_{ea}	mass flow of the exhaust air exiting the heat recovery unit	(kg/s)
\dot{m}_{oa}	mass flow of the outside air	(kg/s)
$c_{p,1+x}$	specific heat capacity of moist air	(J/(kg·K))

With this temperature conditions at the inlet of the heat pump the COP characteristic can be evaluated, i.e. the heat pump characteristic of the test of single components can be corrected for the proposed combined test.

Since there are compact units, where the heat pump operation influences the operation of the heat recovery, the combined testing according to the proposed test procedure for compact units described in paragraph B.4.4.1 in chapter B is advised.

To evaluate the system performance, the heat recovery operation has to be taken into account, as well. The electricity consumption of the heat recovery operation is defined by the equation

$$E_{hr} = \sum_i P_{hr} \cdot t_{ON,hr,i} \quad [\text{J}]$$

eq. A 57

where

E_{hr}	electricity consumption of the heat recovery fans	(J)
P_{hr}	nominal power consumption of heat recovery	(W)
$t_{ON,hr,i}$	operation time of the heat recovery in bin i	(s)

Concerning the DHW operation, there is no reduction of the domestic hot water requirement. However, impacts on inlet temperature of the heat pump have to be taken into account for the evaluation of the DHW COP, which can be done by eq. A 55 and eq. A 56, if necessary, i.e. in case that testing is not combined testing of the heat recovery unit and heat pump.

Consequently, the calculation of the seasonal performance is to be accomplished according to the equation

$$SPF_{cu} = \frac{Q_{out,g,h} + Q_{out,g,DHW} + Q_{hr}}{E_{in,g,h} + E_{in,g,DHW} + E_{in,g,bu} + E_{hr} + W_g} \quad [-]$$

eq. A 58

where

SPF_{cu}	seasonal performance factor of the compact unit	(-)
$Q_{out,g,h}$	space heating requirement	(J)
$Q_{out,g,DHW}$	domestic hot water requirement	(J)
Q_{hr}	energy recovered by the ventilation system	(J)
$E_{in,g,h}$	electricity input to cover the space heating requirement	(J)
$E_{in,g,DHW}$	electricity input to cover the DHW requirement	(J)
$E_{in,g,bu}$	electricity input for the operation of the back-up heater	(J)
E_{hr}	electricity input to operate the heat recovery unit	(J)
W_g	auxiliary energy input to operate the generator	(J)

B PROPOSAL FOR THE TEST PROCEDURE

B.1 Scope

This proposal is to specify additional test conditions and testing requirements for heat pump systems with combined space heating and domestic hot water production.

Alternate operating systems with switching of the heat pump to either pure space heating or pure domestic hot water operation shall be tested according to the existing standards for the single operation modes, for Europe basically EN 14511 and EN 255-3.

Simultaneous operating heat pump systems with heat production for the space heating and the domestic hot water system at the same time require additional testing, since the heat pump characteristic in simultaneous combined operation changes significantly. This additional testing is the scope of this proposal.

B.1.1 Types of heat sources/system types

The heat sources

- Outside air
- Exhaust air
- Rock/ground/water

and the heat pump types

- Brine-to-water
- Air-to-water
- Direct expansion-to-water
- Heat pump compact units (outside air and/or exhaust air – to water and/or air)

are covered by this proposal.

B.2 Normative references

This standard incorporates by dated or undated references provisions from other publications. These normative references are cited at the appropriate places in the text and the publications are listed hereafter. For dated references, subsequent amendments to or revisions of any of these publications apply to this standard only when incorporated in it by amendment or revision. For undated references the latest edition of the publication referred to applies.

EN 255-3	1997	Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors – Heating mode, Part 3: Testing and requirements for marking for sanitary hot water units
EN 308	1997	Heat exchangers- Test procedures for establishing performance of air to air and flue gases heat recovery devices
EN 13141-7	2004	Ventilation for buildings - Performance testing of components/ products for residential ventilation - Part 7: Performance testing of a mechanical supply and exhaust ventilation units (including heat recovery) for mechanical ventilation systems intended for single family dwellings
EN 14511	2004	Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling
NT VVS 021	1983	Heat recovery unit: external leakage

B.3 Terms and definitions

In addition to definitions specified in the following the terms and definitions given in EN 14511-1:2004 and EN 255-3:1997 apply for the purposes of this proposal.

B.3.1 Definitions

B.3.1.1 Alternate combined operation

Heat pump system layout, where the heat pump produces heat for the space heating system and the domestic hot water system by switching between the heating system and the domestic hot water system. Thus, heat is delivered either to the space heating or to the domestic hot water system.

B.3.1.2 Combined operation

Heat pump system layout that allows producing heat for the space heating system and the DHW system

B.3.1.3 Compact unit

Unitary assembly containing a heat pump unit, a heat recovery unit and a domestic hot water storage in one common casing.

B.3.1.4 Desuperheating

Heat extraction from the heat pump cycle between the discharge of the compressor and the inlet of the condenser heat exchanger. The heat is extracted from the hot refrigerant gas leaving the compressor until reaching the saturation point at the compressor discharge pressure. The amount of heat decoupled mainly cools down the refrigerant gas, whereby condensing heat plays a minor role.

B.3.1.5 Direct expansion

In the context of this proposal ground-coupled heat pump systems that do not use an intermediate cycle to extract the heat from the ground but evaporates the refrigerant in the ground.

B.3.1.6 DHW-only operation

See single operation

B.3.1.7 Domestic hot water

Domestic hot water in the context of this proposal is water of a minimum temperature of 40°C.

B.3.1.8 Simultaneous combined operation

Operation mode or heat pump system layout, where energy for space heating and domestic hot water heating is produced simultaneously, i.e. at the same time. Frequent system layouts for simultaneous operation are refrigerant desuperheaters or condensate subcoolers

B.3.1.9 Single operation

Operation mode or heat pump system layout, where heat is delivered exclusively for one use, e.g. for space heating (denoted as well as space heating-only operation) or for domestic hot water generation (denoted as well as DHW-only operation)

B.3.1.10 Space heating-only operation

See single operation

B.3.1.11 Subcooling

Heat extraction from the heat pump cycle between the exit of the condenser and the inlet of the expansion device. The heat is extracted from the subcooled refrigerant liquid below bubble point temperature.

B.3.2 symbols and abbreviations

Table B 1 - Symbols and Units

Symbol	Name of quantity	Unit
ϕ	Temperature change coefficient	–
ϕ	Heating capacity	W
θ	Celsius temperature	°C
η	Efficiency	-
COP	Coefficient of performance for heating	W/W
E	Quantity of energy, electricity	J
ETV	Electro thermal amplification	-
H	Enthalpy	kJ
P	Electrical power	W
Q	Quantity of energy, heat	J
\dot{V}	Volume flow rate	m ³ /s
t	Time, period of time	s
Δp	Pressure difference	Pa or kPa

Table B 2 - Indices

a	Air	hp	Heat pump
b	Brine	hr	Heat recovery
bu	Back-up	in	Inlet to system
combi	Combined operation	oa	Outside air
cu	Compact unit	out	Outlet from system
DHW	Domestic hot water	ra	Return air
ea	Exhaust air	sa	Supply air
g	Ground	t	Total
h	Space heating	w	Water
H	Heating according to EN 14511-1	wb	Wet bulb
H,hs	Space heating including supplementary heating (not corrected to 0 kPa for pump and fan efficiencies)		

B.4 Test procedures

Testing of combined operating heat pump systems requires additional testing to that defined for the space heating-only or the domestic hot water (DHW)-only operation.

While in alternate combined operating systems the heat pump either works on the space heating systems or DHW system, testing can be covered by the testing according to the respective single operation mode testing. However, amendments and modifications outlined in chapter 4.1 for the space heating-only testing and in chapter 4.2 for the DHW-only testing shall be considered during testing of the single operation modes.

In simultaneous combined operation, however, the space heating and the DHW energy is produced at the same time by heat extraction at different points of the internal heat pump cycle. Therefore, the heat pump characteristic changes significantly and additional testing to cover this combined operation mode is required as described in chapter 4.3.

B.4.1 Space heating-only operation

The space heating-only mode shall be tested as described in EN 14511 with the following changes.

Setting of flow rate

Instead of two mass flow rates dependent on the type of distribution system (radiators, floor heating) only one mass flow rate shall be used during testing. Setting of the mass flow follows the standard rating points of the test conditions given in chapter 5.

Leakage testing

Additional leakage testing shall be carried out for exhaust-air heat pump units according to the Nordtest-method NT VVS 021. If the leakage exceeds 4% of the nominal air flow specified by the manufacturer at an external pressure difference of 250 Pa or the leakage of the unit exceeds 5% of the air flow at test point no. 1 given in Table B 6 the unit must be sealed, until the leakage is below the maximal permissible level.

Pump and fan power

The electrical power input to pumps and fans shall be taken into account as described in EN 14511. Preferably, accredited test data (ISO standard) of the pump or fan efficiency respectively shall be used for the calculation. Otherwise, if the pump and/or fan is not included in the test, the default value of the efficiency of the power of pumps and fans of $\eta = 0.1$ shall be applied for the calculation instead of $\eta = 0.3$ as described in EN 14511. If the pump and/or fan is included in the test the default value of $\eta = 0.3$ shall be used as described in EN 14511.

B.4.2 DHW-only operation

The DHW-only operation shall be tested as described in EN 255-3 with the following changes/ amendments.

Tapping pattern of the domestic hot water operation

The tapping cycle according to EN 255-3 shall be changed to four draw-offs instead of two. Tapping is extended to more draw-offs, if the difference in energy content of two following draw-off exceeds 5%. The volume of the draw-offs shall be determined by $\frac{1}{4}$ of the energy content of the storage at a tapping water reference temperature of 60°C, i.e. if the domestic hot water temperature level is lower, the volume of the draw-offs is higher.

Flow rate during domestic hot water tapplings

The flow rate of the tapping is fixed according to the storage testing to 1 storage volume/hour, at minimum 10 l/min, which equals 0.167 l/s.

Additional evaluation of an average heating capacity for the DHW operation

An average DHW heating capacity is calculated by the equation

$$\phi_{\text{DHW}} = \frac{Q_{\text{DHW}}}{t_t} \quad [\text{W}]$$

eq. B 1

where

ϕ_{DHW}	calculated average heating capacity for DHW operation	(W)
Q_{DHW}	heat energy extracted with the DHW according to EN 255-3 phase 2	(J)
t_t	time of cycle according to EN 255-3 phase 2	(s)

The heating capacity is required to estimate the running time of the heat pump system for the production of DHW heating.

Cold water inlet temperature

The temperature of the incoming cold water is set to 10°C as defined in Mandate M/324 [B 1]

B.4.2.1 Reduced testing for DHW operation mode

If enough information on the storage losses according to accredited testing, e.g. from storage testing, exist, these values can be used to evaluate the storage losses. In this case, the testing according to EN 255-3 can be shortened by omitting the phase 4 of EN 255-3 for the storage stand-by parameters. The necessary COP value to calculate the electrical energy input due to thermal storage losses shall be derived using the controller setting for reheating, the reference hot water temperature delivered by EN 255-3 and the heating characteristic. COP for the reference temperature and (U·A)-values for the mean temperature of the set values shall be used in the calculation.

B.4.3 Combined operation

B.4.3.1 Black box testing

Due to a manifold of possible system configurations the testing shall be carried out as a black box test-ing, i.e. only measured values at the system boundary are taken into account for the evaluation of the testing. The system boundary for this black box is depicted in Figure B 1.

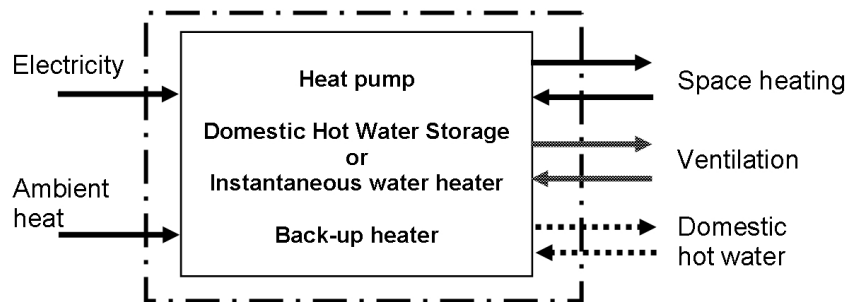


Figure B 1 - System boundary for the black box testing

According to Figure B 1 the system boundary for the black box testing comprises the heat pump and the domestic hot water storage or an instantaneous water heater respectively as well as installed back-up heaters.

B.4.3.2 Test procedure

Testing for combined operation shall be carried out by performing the DHW-cycle prescribed by the DHW-only testing according to EN 255-3 during the space heating-only steady-state testing according to the EN 14511, including the above mentioned modification in chapter 4.1 and chapter 4.2.

The combined testing is carried out after the steady state space heating-only test according to EN 14511 with the same settings concerning mass flow rates and heating water temperatures. Before

initiating the test, it has to be secured, that the maximum temperature in the storage is 30°C due to the environmental temperature restriction in EN 14511, e.g. by a large draw-off. If necessary, the compressor can be switched off.

The data to be recorded during the testing are the same as the corresponding data to be recorded in EN 14511 and EN 255-3.

The COP for combined operation mode shall be evaluated according to the equation

$$\text{COP}_{\text{combi}} = \frac{Q_h + Q_{\text{DHW}}}{P_t \cdot t_t} \quad [-] \quad \text{eq. B 2}$$

where

$\text{COP}_{\text{combi}}$	COP of the heat pump in simultaneous operation	(-)
Q_h	heat energy supplied to the heating systems during testing period	(J)
Q_{DHW}	heat supplied to the DHW heating according to EN 255-3 phase 2	(J)
P_t	average total power consumption of the heat pump during the testing cycle	(W)
t_t	time of the testing cycle	(s)

If due to the system configuration the testing is performed including the supplementary heater, the COP is to be calculated according to eq. B 6.

Moreover, the average heating and DHW heating capacity during the combined test shall be evaluated according to EN 14511 and EN 255-3 extended according to chapter B.4 respectively.

For the evaluation of the pump or fan power respectively the modifications of chapter B.4.1 apply.

B.4.3.3 Test apparatus

The testing is to be performed on the same test apparatus as specified EN 14511 and EN 255-3 and for ventilation testing as specified in EN 13141-7.

B.4.3.4 Uncertainties of measurements

The uncertainties of measurements given in EN 14511 and EN 255-3 apply for the combined testing, as well.

B.4.4 Test procedure for particular types of heat pump systems

B.4.4.1 Compact units

Testing of compact units comprises test procedures for

- Thermodynamic testing
- Leakage testing (including filter leakage testing)
- Air flow testing (pressure curve)
- Acoustic testing
- Handling/maintenance/safety
- Hygienic examinations

In the framework of this proposal, only the thermodynamic testing is described.

Testing is performed in four operation modes

- heat recovery-only operation
- combined heat recovery and heat pump operation for space heating
- heat pump operation for DHW-only

B.4.4.1.1 Test apparatus

The test apparatus is depicted in Figure B 2. The measured quantities are given in Table B 3

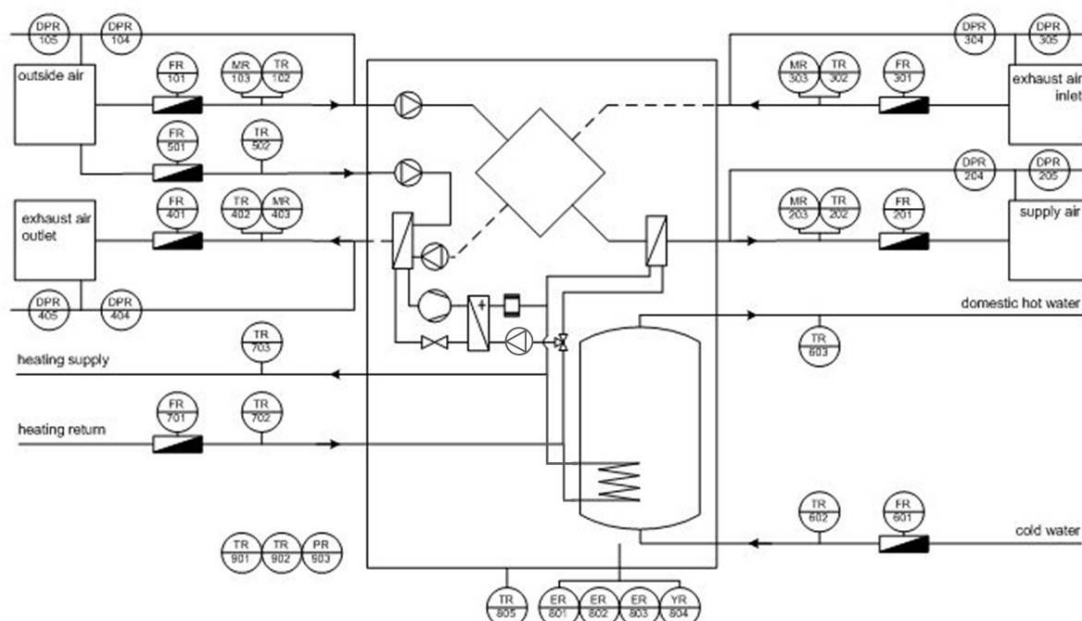
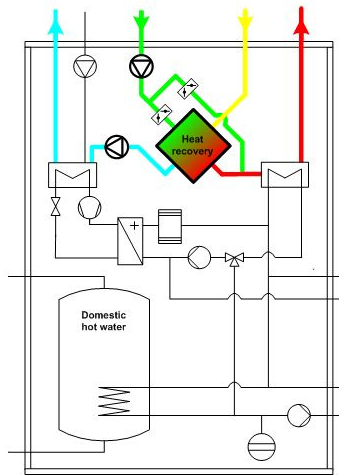


Figure B 2 - Test apparatus with position of measurement sensors

Table B 3 - Measured quantities

Item	Measured quantity	Item	Measured quantity
FR101	outside-air volume flow rate	TR302	return-air temperature
TR102	outside-air temperature	MR303	return-air humidity
MR103	outside-air humidity	DPR304	return-air pressure drop before unit
DPR104	outside-air pressure drop before unit	DPR305	return-air pressure difference compensating chamber / surroundings
DPR105	outside-air pressure difference compensating chamber / surroundings	FR301	return-air volume flow rate
FR201	supply-air volume flow rate	FR401	exhaust-air volume flow rate
TR202	supply-air temperature	TR402	exhaust-air temperature
MR203	supply-air humidity	MR403	exhaust-air humidity
DPR204	supply-air pressure drop after unit	DPR404	exhaust-air pressure drop after unit
DPR205	supply-air pressure difference compensating chamber / surroundings	DPR405	exhaust-air pressure difference compensating chamber / surroundings
FR501	outside-air volume flow rate for HP	ER801	power consumption of unit
TR502	outside-air temperature for HP	ER802	power consumption of HP
FR601	cold-water flow rate	ER803	monitoring of input voltage
TR602	cold-water temperature	YR804	condensate removal
TR603	domestic hot-water temperature	TR805	surface temperature of unit
FR701	flow rate of hot-water heating	TR901	Temperature of surrounding air
TR702	heating-return temperature	TR902	surface temperature of walls
TR703	heating-supply temperature	PR903	air pressure

B.4.4.1.2 Measurements of the heat recovery-only operation



During testing of the heat recovery, only the ventilators of the heat recovery are operated and the heat pump and the electrical back-up heating is switched-off, as depicted in Figure B 3. The DHW-storage has the temperature of its environment ($\pm 5K$). The heat recovery is tested at different flow rates dependent on the application, which are determined in the air flow testing.

Data to be recorded:

- Temperatures (all flows)
- Humidities (all flows)
- Volume flow (all flows)
- Pressure difference
- Atmospheric pressure
- Electrical values (power, voltage, current)

Figure B 3 - Testing of heat recovery

Test results:

- Temperature change coefficient supply air/return air

$$\Phi_{oa} = \frac{\theta_{sa} - \theta_{oa}}{\theta_{ra} - \theta_{oa}} \quad \Phi_{ra} = \frac{\theta_{ra} - \theta_{ea}}{\theta_{ra} - \theta_{oa}} \quad [-]$$

eq. B 3

where

Φ_{oa}	temperature change coefficient (from outside air to building)	(-)
Φ_{ra}	temperature change coefficient (from building to outside air)	(-)
θ_{sa}	temperature of the supply air after the heat recovery	(°C)
θ_{ra}	temperature of the return air from the heated space	(°C)
θ_{oa}	temperature of the outdoor air	(°C)
θ_{ea}	temperature of the exhaust air	(°C)

- Electro-thermal amplification

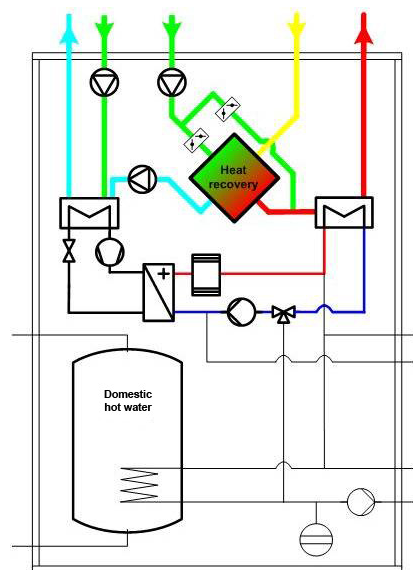
$$ETV = \frac{H_{sa} - H_{oa}}{E_{hr}} \quad [-]$$

eq. B 4

where

ETV	electro thermal amplification	(-)
H_{sa}	enthalpy of the supply air	(J)
H_{oa}	enthalpy of the outdoor air	(J)
E_{hr}	electrical energy input to the heat recovery unit	(J)

B.4.4.1.3 Measurements of heat recovery and heat pump operation for space heating



For the testing of the heat pump for space heating operation, the heat pump is operated in combination with the heat recovery, as shown in Figure B 4. The testing is conducted according to EN14511. The heat rejection from the heat pump is only used for space heating. The domestic hot water storage is loaded to 60°C. During the test there are no domestic hot water draw-offs. There is no heat transfer from or to the DHW-storage system. Electrical back-up heaters are switched-off.

Data to be recorded:

- Temperatures (all flows)
- Humidities (all flows)
- Volume flow (all flows)
- Pressure difference
- Atmospheric pressure
- Electrical values (power, voltage, current)

Figure B 4 Heat pump testing for space heating operation

Test results:

- Output capacity of the heat pump (according to EN 14511)
- COP of the heat pump (according to EN 14511)

Optionally, the COP can be evaluated for the entire unit using the entering outside air and the exhaust air conditions according to the system boundary in presented in Figure B 5

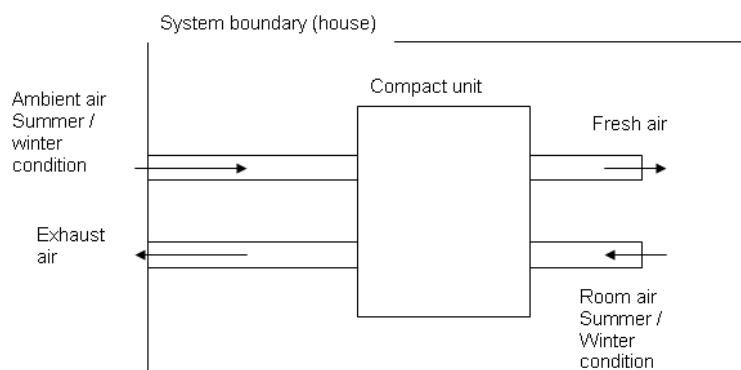


Figure B 5 - System boundary for the evaluation of the COP for compact units

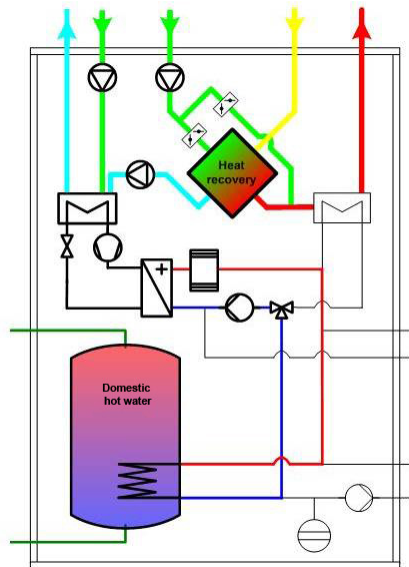
Respectively, the overall COP of the combined heat pump/heat recovery operation shall be calculated according to the

$$\text{COP}_{\text{cu}} = \frac{H_{\text{oa}} - H_{\text{ea}} + E_{\text{cu}}}{E_{\text{cu}}} \quad [-] \quad \text{eq. B 5}$$

where

COP_{cu}	COP of the compact unit	(-)
H_{oa}	enthalpy of the outdoor air	(J)
H_{ea}	enthalpy of the exhaust air	(J)
E_{cu}	total energy consumption of the compact unit	(J)

B.4.4.1.4 Measurements of the heat pump operation for domestic hot water heating



For the testing of the heat pump for DHW operation, the heat pump is operated in combination with the heat recovery (winter case) as depicted in Figure B 6 as well as without the heat recovery (summer case). The testing is performed according to EN 255-3 with the modifications outlined in chapter B.4.2. The storage is initially loaded with water of 20°C. All back-up heaters are switched-off.

Values to be monitored:

- Temperatures
- Humidities
- Volume flow rates air
- Volume flow rate water
- Pressure difference
- Atmospheric pressure
- Electrical values (power, voltage, current)

Figure B 6 Heat pump testing in DHW operation

The evaluation of the domestic hot water mode is performed according to the EN 255-3.

B.4.4.2 Special testing requirements for transcritical processes (e.g. CO₂-refrigerant)

Heat pumps with transcritical processes, in particular with CO₂-refrigerant have some particularities concerning their operation due the properties of the refrigerant. Consequently, typical operation conditions are different and should be reflected in the testing. Combined operating integrated CO₂ heat pump systems are basically operated with floor heating systems at supply temperatures as low as possible.

B.4.4.2.1 System testing

As CO₂-systems are highly integrated and performance depends strongly on the interaction of the heat pump and the storage, integrated CO₂ heat pump shall be tested as system according to the system boundary shown in Figure B 1.

B.4.4.2.2 Modification of test points

Since the performance decrease notably with high return temperature of the heating system, CO₂ systems will not be operated above return temperatures of the heating system above 40°C. Thus, test points should take into account this fact by limiting the temperature to a supply temperature of 40°C.

B.4.4.3 Simultaneously operating exhaust-air heat pumps

Testing of exhaust-air heat pumps comprises test procedures for

- Thermodynamic testing
- Leakage testing
- Air flow testing (pressure curve)
- Water flow testing (pressure curve)
- Acoustic testing
- Handling/maintenance/safety

In the framework of this proposal, only the thermodynamic testing is described.

Testing is performed in three operation modes

- heating-only operation
- DHW-only operation (summer operation mode)
- simultaneously combined operation (for space and domestic hot water heating)

B.4.4.3.1 Setting of air flow rate and air pressure

The air flow rate and air pressures are set for the nominal test point No. 1 in Figure B 7. The flow rate is adjusted during heat pump operation in order to achieve the specified temperature difference on the air side. In case of adjustable speed of the fan (number of revolutions), the lowest setting giving an external pressure difference over the test object of at least 100 Pa ($\Delta p_{e,a}$). The air pressure at sensor PR104 shall be at least 100 Pa higher compared to the one at sensor PR103 (see Figure B 7). The auxiliary fans shall be controlled in order to obtain an air pressure in location at sensor PR102 and PR104 is 0 ± 30 Pa (e). However, the static pressure difference between at PR102 and PR104 shall not exceed 0 ± 10 Pa.

For test point No 4 and 5, the air flow is changed and in case of adjustable speed of the fan (number of revolutions), the lowest setting giving an external pressure difference over the test object of at least 100 Pa ($\Delta p_{e,a}$). The auxiliary fans shall be controlled as above.

Air flows and air pressures are referred to 20°C at the current atmospheric pressure.

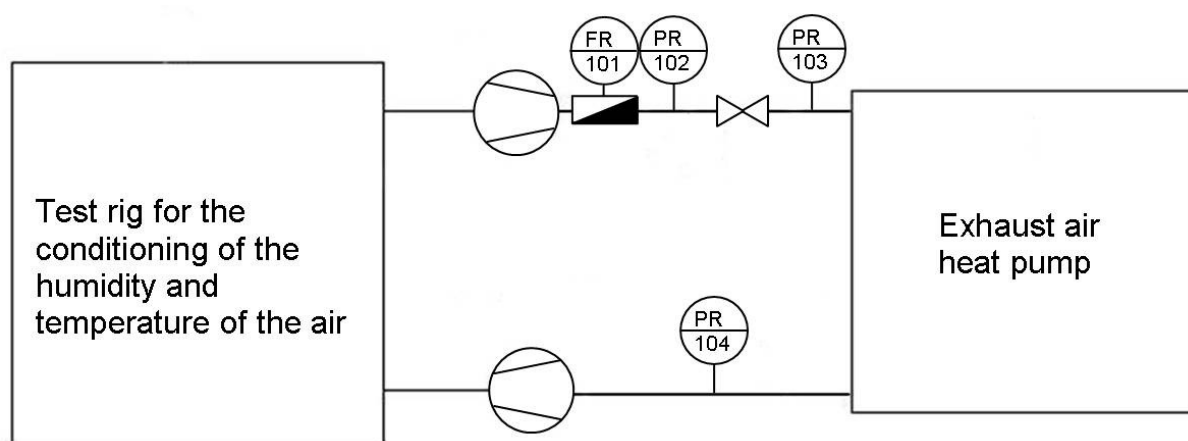


Figure B 7 - Setting of air flow rate and air pressure

Table B 4 - measured quantities for setting of air flow rate and air pressure

Item	Measured quantity	Item	Measured quantity
FR101	Exhaust air flow rate	DPR102	Static pressure after auxiliary fan
DPR103	Static pressure at entrance of exhaust air heat pump	DPR104	Static pressure at outlet of exhaust air heat pump

B.4.4.3.2 Setting of heating water flow rate and external pressure difference

The heating water flow rate and external pressure difference are set for the nominal test point No. 1 in Figure B 7. Measured quantities are given in Table B 4. The flow rate is adjusted during heat pump operation in order to achieve the specified temperature difference on the heating water side. In case of adjustable speed of the pump (number of revolutions), the lowest setting giving an external pressure difference over the test object of at least 0 kPa ($\Delta p_{e,w}$).

The pressure in the heating water system shall be at least 100 kPa (e) on the suction side of the heating water pump in order to avoid formation of gas bubbles in the heating water system, which can affect the accuracy of the test procedure. Prior to testing all liquid systems shall be carefully de-aired.

B.4.4.3.3 Heating-only operation

Heating-only tests shall be performed at the test points no. 1 – 7, specified in Table B 6. During these tests any supplementary heater shall be switched off. For heat pumps with a built-in storage tank, see Figure B 7, the test period shall be lengthened to 24 hours consisting of a 12-hour stabilisation (equilibrium) period followed by a 12-hour measurement period (data collection). During frosting

conditions, i.e. the evaporating temperature is below zero degrees Celsius, the measurements are made for a whole number of cycles. A cycle is defined as the period between a defrost initiation to the next initiation of a defrosting. If this period is shorter than 8 hours then measurements shall be performed during at least three complete cycles, resulting in at least 12 hours measuring period. If the period is between 8 and 12 hours measurements are made for two cycles, and if the period is longer than 12 hours the measurements are continued for maximum 24 hours. The temperature of the high-temperature heat transfer media entering the heat pump shall be kept at the specified value during these frosting and defrosting cycles.

During the test point no.10 (see Table B 6), the supplementary heating (e.g. an immersion heater) shall be switched on. In test points where the supplementary heater is allowed to switch on the requirement regarding maximal allowed deviation of the heating water temperature out from the heat pump according to EN 14511 does not apply, but larger deviations are accepted.

B.4.4.3.4 DHW-only operation

The testing is performed according to EN 255-3 with the modifications outlined in chapter 4, test point No. 11 in Table B 6. The test is performed in “summer operation”, e.g. the supplementary heating shall (if possible) be switched-off.

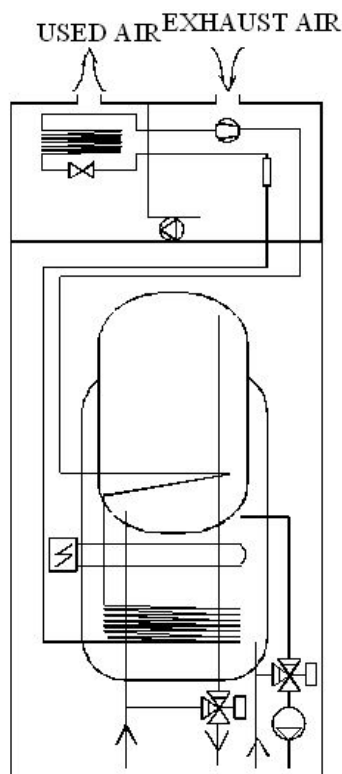


Figure B 8 - Simultaneous operating exhaust air heat pump

B.4.4.3.5 Simultaneously combined operation

The testing is performed as described in section B.4.3.2, which involves a combination of EN 14511 and EN255-3 with some changes. The test points no. 8 and 9 are performed. Since the heat source for an exhaust-air heat pump is limited by the ventilation air flow and the whole capacity of the heat pump is used for space heating in the prescribed test procedure, the supplementary heating system shall be allowed to switch on.

In EN255-3 the start and stop of the evaluation period is defined by start and stop of the compressor. However, during the simultaneously combined testing start and stop of the evaluation period is defined by start and stop of the supplementary heating system. Hence, the simultaneously combined test is considered as terminated at the first shut-off of the supplementary heating system following the fourth (or last) draw-off in the modified EN 255-3 cycle, phase 2.

In addition to the COP_{combi} for the combined operation mode, defined in eq. B 2, a COP accounting for the heat pump system, excluding the back-up heating, shall be evaluated according to the equation below.

$$\text{COP}_{\text{combi,hps}} = \frac{Q_h + Q_{\text{DHW}} - P_{\text{bu}} \cdot t_t}{(P_t - P_{\text{bu}}) \cdot t_t} \quad [-] \quad \text{eq. B 6}$$

where

COP _{combi,hps}	COP of the heat pump system, excluding the supplementary heating, in simultaneous operation	(-)
Q _h	heat energy supplied to the heating systems during testing period	(J)
Q _{DHW}	heat supplied to the DHW heating according to EN 255-3 phase 2	(J)
P _t	average total power consumption of the heat pump system (including back-up heating) during the testing cycle	(W)
P _{bu}	average power consumption of the supplementary heating during the testing cycle	(W)
t _t	time of the testing cycle according to EN 255-3 phase 2	(s)

The high degree of integration of the domestic hot water heating system (built-in storage tank) and the space heating system (mantle) results in simultaneously combined operation during the whole heating season, e.g. no space heating-only operation. Therefore, the average combined heating and DHW heating capacity during the combined test shall be evaluated according to the equation below.

$$\phi_{\text{hp,combi}} = \frac{Q_{\text{DHW}} + Q_h - P_{\text{bu}} \cdot t_t}{t_t} \quad [\text{W}] \quad \text{eq. B 7}$$

where

φ _{hp,combi}	calculated heating capacity for simultaneous operation	(W)
Q _{DHW}	heat extracted with the DHW according to EN 255-3 phase 2	(J)
Q _h	heat energy supplied to the heating systems during testing period	(J)
P _{bu}	average power consumption of the supplementary heating during the testing cycle	(W)
t _t	time of the testing cycle according to EN 255-3 phase 2	(s)

B.5 Test conditions

Testing of simultaneous operating systems requires testing of the single operation modes, i.e. the heating-only and the DHW-only operation as well as additional testing for the combined operation. The proposal for the test procedure for the testing of the single and combined operation modes is outlined in chapter B.4.

Resulting test points for single and combined operation modes are given in the following chapters

Table B 5 - Test points for outdoor air-source heat pumps

Table B 6 - Test points for exhaust-air (source) heat pumps

Table B 7 - Test points for ground-source heat pumps

Table B 8 - Test points for direct expansion heat pumps

Table B 9 - Test points for compact units

Table B 10 - Test points for compact units with installed ground-to-air heat exchanger

Table B 11 - Test points for CO₂-heat pumps

B.5.1 Air-to-water systems

B.5.1.1 Outdoor-air-to-water

Table B 5 - Test points for outdoor air-source heat pumps

Test point no.	Outdoor heat exchanger		Indoor heat exchanger		Operation mode	Comment
	Inlet dry bulb temperature $\theta_{oa,in}$ (°C)	Inlet wet bulb temperature $\theta_{wb,a,in}$ (°C)	Inlet temperature $\theta_{w,in}$ (°C)	Outlet temperature $\theta_{w,out}$ (°C)		
1	7	6	40	45	Heating-only	EN14511 rating
2	2	1	a	45	Heating-only	EN14511 application
3	-7	-8	a	45	Heating-only	EN14511 application
4	-15	-	a	45	Heating-only	EN14511 application
5	7	6	a	55	Heating-only	EN14511 application
6	-7	-8	a	55	Heating-only	EN 14511 application
7	7	6	a	35	Heating-only	EN 14511 modified
8	2	1	a	35	Heating-only	EN 14511 modified
9	20	12	a	45	Heating-only	EN 14511 modified
10	-15	-	a	35	Heating-only	EN 14511 modified
11	7	6	a	45	Combined	EN14511 + EN255-3 modified
12	7	6	a	35	Combined	EN14511 + EN255-3 modified
13	7	6	-	-	DHW-only	EN255-3 modified

a The test is performed at the flow rate obtained during the test at standard rating conditions (test point no. 1)

B.5.1.2 Exhaust air-to-water

Table B 6 - Test points for exhaust-air (source) heat pumps

Test point no.	Exhaust-air		Indoor heat exchanger		Heating demand/Heating capacity $\frac{P_{H,hs}}{P_H}$	Operation mode
	Inlet dry bulb (wet bulb) temperature $\theta_{a,in} (\theta_{wb,a,in}) (^{\circ}\text{C})$	Exhaust-air flow rate $\dot{V}_a (\text{m}^3/\text{s})$	Inlet temperature $\theta_{w,in} (^{\circ}\text{C})$	Outlet temperature $\theta_{w,out} (^{\circ}\text{C})$		
1	20(12)	$\dot{V}_{a,n}^a$	40	45	1	Heating-only, rating
2	20(12)	$\dot{V}_{a,n}$	b	35	1	Heating-only
3	20(12)	$\dot{V}_{a,n}$	b	55	1	Heating-only
4	20(12)	$1.5 \cdot \dot{V}_{a,n}$	b	45	1	Heating-only
5	20(12)	$0.75 \cdot \dot{V}_{a,n}^c$	b	45	1	Heating-only
6	15(10)	$\dot{V}_{a,n}$	b	45	1	Heating-only
7	20(15)	$\dot{V}_{a,n}$	b	45	1	Heating-only
8	20(12)	$\dot{V}_{a,n}$	b	45	1	Combined
9	20(12)	$\dot{V}_{a,n}$	b	35	1	Combined
10d	20(12)	$\dot{V}_{a,n}$	b	50	2	Heating-only
11	20(12)	$\dot{V}_{a,n}$	-	-		DHW

a The exhaust-air flow rate shall be set so that the temperature difference on the air side is 18 K at test point no. 1. The same air flow rate is used in all the test point except for no. 4 and 5.

b The test is performed at the flow rate obtained during the test at standard rating conditions (test point no. 1)

c In case the $0.75 \cdot \dot{V}_{a,n}$ is a lower value then the minimum air flow rate defined by the manufacturer, the air flow shall be set to the minimum value.

d This test point is only performed for simultaneously heating heat pumps (see.Figure B 8)

B.5.2 Brine-to-water systems

Table B 7 - Test points for ground-source heat pumps

Test point no.	Outdoor heat exchanger		Indoor heat exchanger		Operation mode	Comment
	Inlet temperature $\theta_{b,in}$ (°C)	Outlet temperature $\theta_{b,out}$ (°C)	Inlet temperature $\theta_{w,in}$ (°C)	Outlet temperature $\theta_{w,out}$ (°C)		
1	0	-3	40	45	Heating-only	EN14511, rating
2	5	a	a	45	Heating-only	EN14511, application
3	-5	a	a	45	Heating-only	EN14511, application
4	0	a	a	55	Heating-only	EN14511, application
5	0	a	a	35	Heating-only	EN14511, modified
6	5	a	a	35	Heating-only	EN 14511, modified
7	0	a	a	35	Combined	EN14511 + EN255-3 modified
8	0	a	a	45	Combined	EN14511 + EN255-3 modified
9	0	a	-	-	DHW-only	EN255-3 modified

a The test is performed at the flow rate obtained during the test at standard rating conditions (test point no. 1)

B.5.3 Direct expansion heat pumps

Table B 8 - Test points for direct expansion heat pumps

Test point no.	Ground temperature	Indoor heat exchanger		Heating mode	Comment
	Ground temperature θ_g (°C)	Inlet temperature $\theta_{w,in}$ (°C)	Outlet temperature $\theta_{w,out}$ (°C)		
1	4	30	35	Space	EN255, rating
2	-1	a	35	Space	EN255, application
3	10	a	35	Space	EN255, application
4	-1	a	50	Space	EN255, application
5	4	a	50	Space	EN255
6	10	a	50	Space	EN255
7	4	-	-	DHW	EN 255-3
8	15	-	-	DHW	EN 255-3

B.5.4 Compact units

Table B 9 - Test points for compact units

	Outdoor air temperature (°C)		Exhaust air temperature (°C)		Heating		DHW	Comment
	Inlet dry bulb temperature $\theta_{oa,in}$ (°C)	Inlet wet bulb temperature $\theta_{wb,in}$ (°C)	Inlet dry bulb temperature $\theta_{ea,in}$ (°C)	Inlet wet bulb temperature $\theta_{wb,ea,in}$ (°C)	Inlet temperature $\theta_{w,in}$ (°C)	Outlet temperature $\theta_{w,out}$ (°C)	Outlet temperature $\theta_{DHW,out}$ (°C)	
1	7	6	20	12	40	45	60	Heating
2	-15	-	20	7*	a	45	60	Heating, optional
3	-7	-8	20	7	a	45	60	Heating
4	2	1	20	10	a	45	60	Heating
5	-7	-8	20	7	a	35	60	Heating
6	7	6	20	12	a	35	60	Heating
7	15	10	20	14	a	35	60	Heating optional
8	7	6	20	12	-	-	60	DHW
a The test is performed at the flow rate obtained during the test at standard rating conditions (test point no. 1)								

* The wet bulb temperature is based on a humidity gain of 2.5 g/kg_{dry air} per outside temperature step based on the given rating point 20(12) in EN 14511

The testing is carried out for different volume flow rates dependent on the application determined in the air-flow testing for space heating.

If the unit is installed with a ground-to-air heat exchanger, relevant temperatures after the ground-to-air heat exchanger are given in Table B 10 for conditions of the Swiss Middleland. For other regions the temperatures have to be verified and eventually be adapted.

Table B 10 - Test points for compact units with installed ground-to-air heat exchanger

Outdoor air temperature (°C)				Exhaust air temperature (°C)		Heating		DHW	Comment
	Dry bulb outside temperature	Inlet wet bulb temperature	Inlet temperature after ground-to-air-heat exchanger	Inlet dry bulb temperature	Inlet wet bulb temperature	Inlet temperature	Outlet temperature	Outlet temperature	
	θ_{oa} (°C)	$\theta_{wb,in}$ (°C)	$\theta_{oa,in}$ (°C)	$\theta_{ea,in}$ (°C)	$\theta_{wb,ea,in}$ (°C)	$\theta_{w,in}$ (°C)	$\theta_{w,out}$ (°C)	$\theta_{w,out}$ (°C)	
1	7	6	7	20	12	40	45	60	Heating
2	-15	-	-7	20	7*	a	45	60	Heating, optional
3	-7	-8	2	20	7	a	45	60	Heating
4	2	1	5	20	10	a	45	60	Heating
5	-7	-8	2	20	7	a	35	60	Heating
6	7	6	7	20	12	a	35	60	Heating
7	15	10	...	20	14	a	35	60	Heating optional
8	7	6	7	20	12	-	-	60	DHW
a The test is performed at the flow rate obtained during the test at standard rating conditions (test point no. 1)									

* The wet bulb temperature is based on a humidity gain of 2.5 g/kg_{dry air} per outside temperature step based on the given rating point 20(12) in EN 14511

The testing is carried out for different volume flow rates dependent on the application determined in the air-flow testing for space heating.

B.5.5 CO₂ heat pumps

Table B 11 - Test points for CO₂-heat pumps

Test point no.	Outdoor heat exchanger		Indoor heat exchanger		Heating mode	Comment
	Inlet temperature $\theta_{b,in}$ (°C)	Outlet temperature $\theta_{b,out}$ (°C)	Inlet temperature $\theta_{w,in}$ (°C)	Outlet temperature $\theta_{w,out}$ (°C)		
1	0	-3	35	40	Space	EN14511, rating
2	5	a	a	40	Space	EN14511, application
3	-5	a	a	40	Space	EN14511, application
4	0	a	a	35	Space	EN14511, application
5	5	a	a	35	Space	EN14511, application
6	0	a	a	35	Combined	EN14511 + EN255-3 modified
7	0	a	a	30	Combined	EN14511 + EN255-3 modified
8	0	a	-	-	DHW	EN255-3 modified
a The test is performed at the flow rate obtained during the test at standard rating conditions (test point no. 1)						

Remark:

For a comparison of different types of heat pumps a fixed DHW outlet temperature should be considered. See discussion in part 2 of the report, chapter 3.5.4.2.

Bibliography

- [B 1] EU Mandate M/324
EU Mandate to CEN and CENELEC for the elaboration and adoption of measurement standards for household appliances – water heaters, hot water storage appliances and water heating, Sept. 2002

CONCLUSION

IEA HPP Annex 28 was started with the objective of enabling the assessment of the seasonal performance factor of different heat pump system solutions for fulfilling the requirements for space heating and domestic hot water. This performance evaluation should serve as well to compare heat pump systems to other heating systems for space heating and domestic hot water production. Actually, there is a variety of different heating systems on the market, but there is a lack of standardised, internationally uniform methods to compare energy performance in order to reward and promote environmentally sound technologies on the background of climate protection.

Thus, IEA HPP Annex 28 has delivered easy-to-use calculation methods and the necessary test procedures to provide the required input of the product characteristics for the calculation. The scope were heat pump systems with alternate or simultaneous combined production of space heating and domestic hot water, which are becoming more and more interesting due to actual trends to highly insulated buildings and low-temperature heating systems.

This part 1 of the final report presents two proposals for standards, one for the calculation method and one for the test procedures. For the calculation, the whole method is presented, since there is no common calculation method on European or international level. Since the testing is based on existing standards, only the extensions for combined operating systems have been integrated in the proposal. The existing standards are referenced for the operation modes already covered.

Actually, the present situation in Europe is favourable, since both calculation and testing standards for space heating and domestic hot water systems are under revision.

In the framework of the EU-Directive on the Energy Performance of Buildings (EPBD), a set of calculation standards for different types of space heating and domestic hot water systems are under development. Thus, there is a strong need for calculation methods for different generators and the other components of heating systems. Therefore, results of IEA HPP Annex 28 have continuously been transferred to the respective working group of the European standardisation organisation CEN. Parts of the proposal are already implemented in a CEN draft standard. After the 6-month public enquiry, result of the Annex 28 are implemented for the formal vote of the draft. On this background a common calculation method for Europe can be available soon.

Testing standards are revised on the European level to harmonise the test methods for different domestic hot water appliances on the background of labelling. For the testing of heat pumps, there are already common standards each for space heating and domestic hot water production, but newer systems with combined operation are not covered. Therefore, the Annex started with an assessment of the existing standards. Feedback and recommendation based on the results will be given to the respective standardisation working group. Subsequently, IEA HPP Annex 28 developed the missing test procedures to cover also the combined operation based on the existing standards, so it is easy to coherently integrate these extensions in the existing standardisation. However, the respective working groups of CEN took up work recently, so results will be transferred to the standardization in the near future.

Due to the present situation in Europe and due to the majority of European countries participating in the IEA HPP Annex 28, the two presented proposals refer to the European standard. However, during the project North-American and Japanese approaches for testing and calculation have also been analysed and discussed. Part 2 of this final report documents these discussions of the different approaches and the reasons that led to the two implemented methods in these two proposals.

Results of the evaluation show that the methods currently used in North-America and Japan are not so different from the European ones, but refer to systems used on the respective markets. In particular concerning calculation all methods are based on the bin-methodology. Concerning testing, there are some differences, which have to be unified in international committees like ISO.

IEA HPP Annex 28 has delivered a detailed analysis of the available methods for testing and calculation of combined heat pump systems and a fruitful exchange of expert knowledge. The common objective will help to develop internationally uniform standards for product certification and quality labelling.

ACKNOWLEDGEMENT

The authors would like to thank the Swiss Federal Office of Energy for funding and supporting the project, in particular the research programme manager Prof. Dr. Thomas Kopp for advising and his support as Swiss alternate delegate in the ExCo of the IEA Heat Pump Programme.

It has to be emphasized that the IEA HPP Annex 28 is a team work and the results presented in this report are based on the effort and contribution of the different member countries.

Hence, respect and thanks are expressed to all participants of the IEA HPP Annex 28 for the valuable contributions and for the constructive discussion and co-operation.

Special thanks are expressed to

- the Austrian National team for the layout and maintenance of the IEA HPP Annex 28 website at the URL <http://www.annex28.net> as participant's fee of the Annex
- the Executive Committee of the Heat Pump Programme for the opportunity to present the final results on a workshop in the framework of the 8th International Heat Pump Conference in Las Vegas in May 2005
- the IEA Heat Pump Centre for the promotion for this workshop
- the American National team for the organisational support before and during the workshop

Annex 28

Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating

Part 2. Project documentation

Final Report

Operating Agent: Switzerland



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The IEA

The IEA was established in 1974 within the framework of the Organization for Economic Cooperation and Development (OECD) to implement an International Energy Programme. A basic aim of the IEA is to foster cooperation among the IEA participating countries to increase energy security through energy conservation, development of alternative energy sources, new energy technology and research and development (R&D). This is achieved, in part, through a programme of energy technology and R&D collaboration, currently within the framework of over 40 Implementing Agreements.

The IEA Heat Pump Programme

The Implementing Agreement for a Programme of Research, Development, Demonstration and Promotion of Heat Pumping Technologies (IA) forms the legal basis for the IEA Heat Pump Programme. Signatories of the IA are either governments or organizations designated by their respective governments to conduct programmes in the field of energy conservation.

Under the IA collaborative tasks or “Annexes” in the field of heat pumps are undertaken. These tasks are conducted on a cost-sharing and/or task-sharing basis by the participating countries. An Annex is in general coordinated by one country which acts as the Operating Agent (manager). Annexes have specific topics and workplans and operate for a specified period, usually several years. The objectives vary from information exchange to the development and implementation of technology. This report presents the results of one Annex. The Programme is governed by an Executive Committee, which monitors existing projects and identifies new areas where collaborative effort may be beneficial.

The IEA Heat Pump Centre

A central role within the IEA Heat Pump Programme is played by the IEA Heat Pump Centre (HPC). Consistent with the overall objective of the IA the HPC seeks to advance and disseminate knowledge about heat pumps, and promote their use wherever appropriate. Activities of the HPC include the production of a quarterly newsletter and the webpage, the organization of workshops, an inquiry service and a promotion programme. The HPC also publishes selected results from other Annexes, and this publication is one result of this activity.

For further information about the IEA Heat Pump Programme and for inquiries on heat pump issues in general contact the IEA Heat Pump Centre at the following address:

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IEA HPP Annex 28

Final Report IEA HPP Annex 28

Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating

Part 2: Project documentation

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In the framework of the
Heat Pump Programme (HPP) of the
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Impressum

IEA HPP Annex 28

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Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating

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Preface

The part 2 of the final report of IEA HPP Annex 28 entitled “Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating” contains the project background, an overview of the national contributions as well as the considerations and discussions that led to the approaches incorporated in the proposals of the test procedure and the calculation method, that are described in part 1 of this report.

Focus of part 2 of the report are the results of Task 2, the development of the test procedures, and Task 3, the development of the calculation method.

Therefore, part 2 of the report is structured in

- an **introduction** to give the background of the work and the facts about the 34-month period of work as well as an introduction of the participants and the national projects.
- a **summary of results of Task 1** state-of-the-art survey which has been documented in detail in the Interim report [2]. Included results have been selected under the criteria to provide the required information to understand the following results of Task 2 and Task 3, which are in focus of this report.
- the **core section of the results of Task 2 and Task 3** to document and explain the background for the chosen approaches for testing and calculation in the two proposals contained in part 1 of this report.
- a **conclusion**, where essential results are summarised.

Summary

IEA HPP Annex 28 entitled “Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating” was launched in January 2003 in the framework of the Heat Pump Programme (HPP) of the International Energy Agency (IEA) with nine participating countries: Austria, Canada, Switzerland, Germany, France, Japan, Norway, Sweden and the USA. The objective of the Annex was to fill the gap of internationally uniform calculation methods to deliver the seasonal performance factor (SPF) of heat pump systems.

The SPF is the key figure for the comparison of different system solutions and for further calculations of primary energy consumption or carbon dioxide emissions, which may be commonly used in the framework of building regulations, energy guidelines or energy labels in the future. In order to perform the calculation, adequate and internationally uniform test procedures are needed to deliver the component characteristics of the heat pump system, in particular including testing and calculation for new system designs that are not covered by the existing standards. Part 1 of the final report contains two proposals, each one for the calculation method and the test procedure. These results of the IEA HPP Annex 28 are intended as a recommendation to international standardisation committees.

This part 2 of the final report is to document the project background, the investigations and discussions that led to the approaches outlined in the part 1. The Annex started with a survey of combined operating heat pump systems on the market of the participating countries and the state-of-the-art in the respective standardisation in Europe, North-America and Japan. As a result, it was agreed, that testing of combined operation shall be based on the existing standards and that calculation shall be based on the bin-method that is already introduced in standards. Based on the results of the market survey the system boundary was defined to include the heat pump and attached storage and back-up systems.

Due to the variety of different system configurations a black-box approach was chosen for the testing, i.e. only values at the system boundary shall be used for testing. Highly integrated systems, which are delivered as a unit shall be tested as system. For systems, which are not necessarily delivered as a unit, component based testing is possible. Standards for the testing of single operation, i.e. space heating-only operation and domestic hot water (DHW)-only operation already exist. It was found, that for alternate operating systems, i.e. systems which switch the heat pump from the space heating-only operation to a DHW-only operation, no additional testing is required, since the characteristics correspond to that in the single operation mode. However, for simultaneous operating systems, which extract heat for the space-heating and DHW operation at the same time from the heat pump cycle, the heat pump characteristic changes in comparison to space heating-only and DHW-only modes. Thus, the existing test procedures are extended to combined operation by performing a combined test, where the domestic hot water tapping cycle is performed during the operation of the heating system. The resulting characteristic is used to calculate a third operation mode of simultaneous operation.

Consequently, the bin-method has also been extended to include the combined operation mode. The quantity of energy produced in combined operation is evaluated by the running time of the system in the respective operation modes, which can be deduced from the energy requirement and the respective heating capacity. The seasonal performance factor is calculated by weighting the performance characteristic from testing with the respective delivered energy amounts. Furthermore, the approach has been extended to compact units by including a ventilation heat recovery.

First validation of the approaches was accomplished in the timeframe of the IEA HPP Annex 28. Testing with the proposed method proved to deliver reliable and reproducible results. However, for a fair comparison, some adjustments based on the system configurations may be required. Calculation results of the heating operation and alternate combined systems have been compared to results of field monitoring for ground-source heat pumps and a compact unit and showed a deviation in the range of $\pm 6\%$. Validation for other system types on the market is a continuous future task.

Results of the IEA HPP Annex 28 have already been integrated in the standardisation process. Results for the calculation method have been introduced in the draft standards prEN15316 of the European standardisation organisation CEN in the framework of the European Energy Performance Building Directive (EPBD) and have been discussed in Europe during a public enquiry of the draft. Results on the testing have been transferred to the standardisation after the respective CEN working groups took-up work.

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1 INTRODUCTION TO THE IEA HPP ANNEX 28

The IEA HPP Annex 28 “Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating” was initiated by Switzerland in the Heat Pump Programme (HPP) of the International Energy Agency (IEA) and started its work in January 2003. Switzerland provided the Operating Agent and project management (Institute of Energy in Building of University of Applied Sciences Northwestern Switzerland (<http://www.fhnw.ch>) in charge of the Swiss Federal Office of Energy (<http://www.bfe.admin.ch>). The project time of the IEA HPP Annex 28 was 34 month, and the final report was presented to the HPP Executive Committee (ExCo) in November 2005.

1.1 Background and Motivation

Motivation for the work carried out in the framework of IEA HPP Annex 28 are new system developments which are supposed to show an increase of energy performance due to improved heat pump cycles, but are currently not covered by the standardisation.

Moreover, at least in Europe, there is currently on the one hand a strong need for calculation methods in the framework of the implementation of the Directive on the Energy Performance of Buildings (EPBD) (see details below in chapter 1.1.2.1) and, on the other hand, an ongoing revision and harmonisation of test methods for different types of domestic hot water (DHW) household appliances in the context of energy labelling (see details below in chapter 1.1.2.2)

1.1.1 System Developments in the Building Sector

Due to a restriction of the energy consumption of newly-built residential dwellings in national directives or building regulations during the Nineties, the share of the energy demand for domestic hot water heating in relation to the overall heating energy demand of the building is growing continuously. In Germany, for example, ultra-low energy houses according to the “passive house standard” (<http://www.passiv.de>) must not require more than 15 kWh/(m²a) for space heating. Therefore, the domestic hot water heat requirement can constitute up to 50% of the total heat requirement. Thus, heat pump systems which can produce both space heating and domestic hot water with the same heat pump in combined operation are becoming more and more interesting.

New system developments using an enhanced heat pump cycle layout with internal heat exchange or heat decoupling on different temperature levels are usually not covered by existing standard testing or calculation methods. However, in the context described above, a combined generation of space heating and domestic hot water with the same heat pump is attractive, as efficiency gains are expected by the coupling. So, integrated systems providing different building services like space heating, domestic hot water production, ventilation or cooling are currently introduced in the market.

For the rating of these new systems testing and calculation is required for different fields of application:

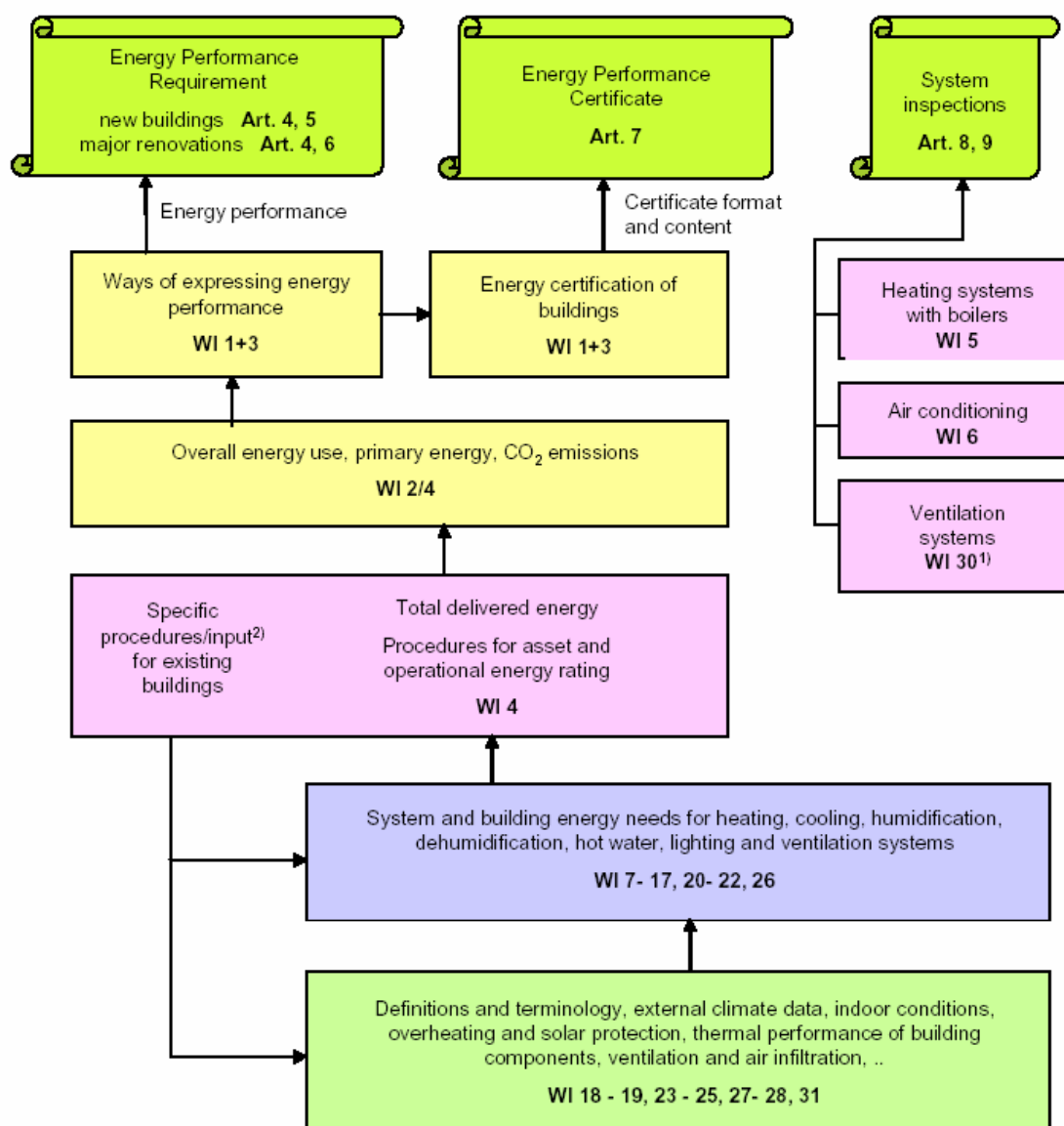
- **Manufacturers** need guidelines, how to test their components in compliance with other systems and which data have to be provided.
- **Planning engineers** need reliable efficiency data as an input for a seasonal performance comparison of systems, which is the basis for further cost analysis and assessment of environmental impact.
- **Product labelling** is presently changing to make product rating no longer dependent on instantaneous efficiency values, but on seasonal performance values, which seem to be a better metric for energy saving potentials and environmentally sound technologies.
- **Consumers** are interested in the performance and quality of the product with an emphasis on energy costs and on environmental issues, e.g. expressed as primary energy consumption.
- **Policy makers and consultants** need adequate key figures to define target values in regulations and directives.

Summarising, comprehensive and uniform test procedures and subsequent calculation methods are needed by various users to compare different heating systems and enable a fair and transparent competition of systems on the market.

1.1.2 European Standardisation

European standardisation in the field of systems for space heating and domestic hot water production is presently driven by the implementation of two directives, which are shortly described in the following paragraphs.

1.1.2.1 Directive on the Energy Performance of Buildings



1): Not explicitly mentioned in the Directive

2): Unless already covered by WI 7-28

Fig. 1: Calculation methodology in the framework of the EPBD (source [1])
WI is the abbreviation for Work item.

In December 2002 the European Commission (EC) published a Directive on the Energy Performance of Buildings [3], often shortly entitled as Energy Performance Buildings Directive (EPBD). The general objective of the EPBD is to transfer best practice examples found in the building sector of single countries of the European Union (EU) to the other member countries of the EU. Thereby, methods to keep climate protection objectives in terms of a reduction of CO₂-emissions as laid down in the Kyoto Protocol shall be spread over the entire EU. More information on the EPBD can be found at the URL <http://www.buildingsplatform.org>

Fig. 1 gives an overview of the calculation methodology outlined in the EPBD. The instruments described in the directive to achieve this objective are threefold:

- Energy performance requirements for new buildings and major renovations
- Introduction of a building energy performance certificate, often referred to as building energy passport, where both the performance of the building and the installed building technology is assessed and displayed, similar to other product labels already in operation in the EU.
- Guidelines and directives for heating system inspections and maintenance

The implementation of the directive comprises a revision of existing European buildings standards like the calculation of the building energy demand according to EN ISO 13790 [4], which is updated to include cooling demands as well as a set of updated or new standards covering the building technologies. Implementing the EPBD would enable a uniform comparison of different heating systems covering solar energy, biomass, heat pumps, boilers and cogeneration. The outline of the standardisation work is given in an umbrella document [1] of the European standardisation organisation CEN. The standards in the framework of the EPBD have come into operation by January 2006. Moreover, standards for the design of heating and domestic hot water systems are in preparation. However, these are not directly covered by the EPBD.

1.1.2.2 Mandate M/324 on Water Heating Systems

The mandate M/324 [5] to CEN and CENELEC aims to elaborate and adopt measurement standards for household domestic hot water appliances in order to derive comparable testing results for labelling purposes, i.e. testing standards for different hot water producing systems including heat pumps are to be harmonised. Therefore, five European tapping profiles have been defined and are included in the mandate and a general revision of testing standards has started. The working group committed to the revision of the European heat pump standard for domestic hot water systems EN 255-3 [6] has had its start-up meeting in the end of April 2006.

1.2 Scope and Objectives

The **scope** of the IEA HPP Annex 28 was originally dedicated to heat pump systems with combined space heating and domestic hot water (DHW) production. In simultaneous combined systems the latter one is produced basically by means of

- desuperheating
- subcooling of the condensate
- cascade hot water heat pump
- extracting the heat from the supply line of the heating system

Due to the present developments of compact units especially designed for low to ultra low energy houses, which simultaneously cover the requirements for heating, DHW and ventilation and often provide an option for cooling operation in summertime, too, the scope of the Annex 28 has been extended to

- air distribution systems
- combined space cooling/DHW and space heating/DHW production (year-round comfort systems).

The **objectives** of IEA HPP Annex 28 according to the legal text [7] are twofold:

- to establish test procedures which deliver the necessary data to calculate the overall seasonal performance factor of combined operating heat pump systems with a minimum requirement of testing equipment and testing time
- to work out a simple method to calculate the seasonal performance factor for these heat pump systems

Combined operation can basically be distinguished in two types, alternate and simultaneous combined operation

- in alternate combined operation, the heat pump works either on the space heating or on the DHW heating, i.e. the heat pump operation is switched between these two operation modes. Fig. 2 gives an example of a system layout for alternate combined operation.

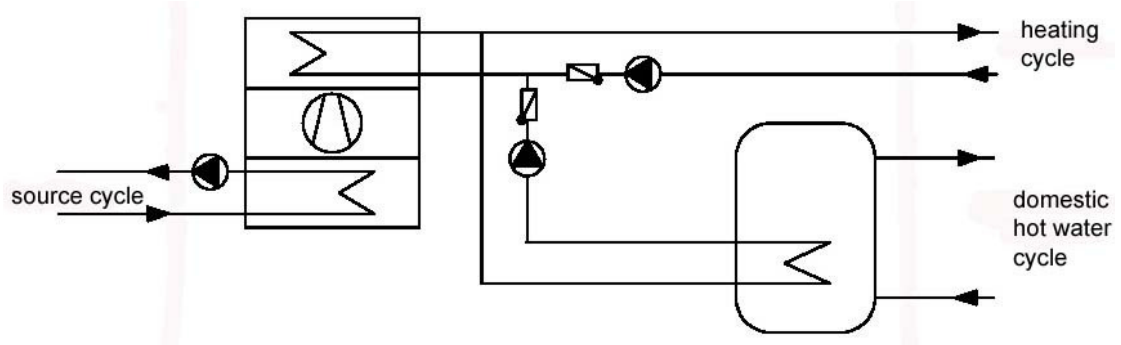


Fig. 2: Example of a system layout for an alternate combined operating heat pump system (source [9])

- in simultaneous combined operation, the space heating and domestic hot water energy is produced at the same time. Methods to realise this simultaneous operation are given under the keyword scope and are described in the following paragraph.

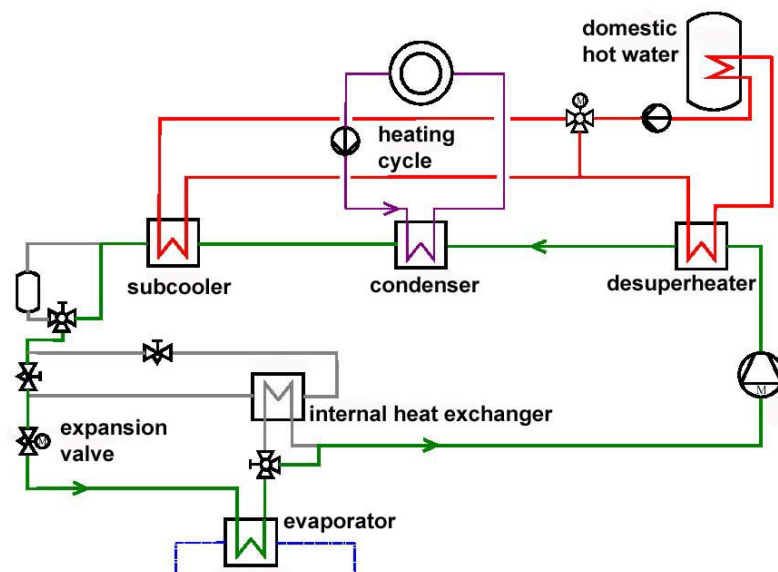


Fig. 3: Example of a system layout for a simultaneous combined operating heat pump system (source TU Dresden [8])

1.2.1 Cycle improvements used in simultaneous combined operating systems

Newer system developments use improved heat pump cycles with internal heat exchange and heat extraction by more than one heat exchanger on different temperature levels as depicted in Fig. 3. The three major mechanisms in cycles with single stage to extract heat on different temperature levels are

- **Desuperheating of the hot compressed gas after the compressor**

To extract heat from the heat pump cycle on a temperature level above the condensation temperature desuperheating of the hot compressed gas can be applied by placing a second heat exchanger, a desuperheater, between the compressor outlet and the inlet of the condenser. In conventional heat pumps, the desuperheating of the compressed gas takes place in the condenser. However, if there is a need for higher temperatures, e.g. for the domestic hot water production, heat on higher temperature level can be provided quite efficiently, since compressor power input is defined by the pressure level, which corresponds to the condensation temperature that is determined by the requirement of the heating system. So, by desuperheating, heat on a higher temperature level can be provided without raising the pressure level.

- **Subcooling of the condensate after the condenser**

A further heat extraction after the condenser on a temperature level 5 – 20 K lower than the condensation temperature can be used for preheating purposes or a source temperature in multi-stage heat pumps. The subcooling has two positive effects on the cycle efficiency: on the one hand, the expansion is more efficient because of the lower temperature level (reduction of exergetic losses), on the other hand, the subcooling increases the heating capacity without increasing the compressor power.

- **Internal heat exchange**

The internal heat exchange transfers heat from the warm condensate to the suction gas. It increases the suction gas temperature before the compressor and thereby the outlet temperature of the compressed gas at the compressor outlet. In this way, the potential of a heat extraction by desuperheating can be increased by higher mean temperature level for the supplied heat in the desuperheater. However, the efficiency gains from internal heat exchange depends on the properties of the refrigerant, since a higher compressor inlet temperature causes more electrical power input to the compressor. So, depending on the properties of the refrigerant, the one or the other effect is dominant. Moreover, the internal heat exchange is limited by a maximum compressor outlet temperature [8].

In conventional single stage heat pumps with only one heat exchanger on the sink side (condenser), the three effects of desuperheating, condensation and subcooling take place in this single heat exchanger. Thereby, the different temperature levels can normally not be used for different applications like space heating or domestic hot water.

Some system layouts use a combination with three parts of the heat exchanger, as well, as for instance in the case of a tripartite gas cooler in the Norwegian system depicted in chapter 1.4.3.7.

1.3 Project structure

The project was carried out task-shared between the participating countries, whereby the expenses for the operating agent are cost-shared. The project has been structured in three tasks, referring to the objectives of the Annex:

- **Task 1: Systems to be investigated**

Task 1 comprises a survey of existing heat pump systems on the market and systems under development as well as a characterisation of the state in standardisation. Moreover, the data sources for the calculation were analysed. The survey gave a system definition for the further investigation and the missing items in the present standardisation to cover the combined operating systems.

- **Task 2: Development of test procedures**

Development of test procedures based on the existing standardisation to cover the combined operation.

- **Task 3: Development of calculation methods**

Development of methods to calculate the seasonal performance factor of combined operating heat pump systems based on the test procedures in Task 2.

Since Task 2, the test procedure, and Task 3, the calculation method, have strong interdependencies they are worked out in parallel.

The Interim Report delivered in March 2004 to the HPP ExCo covers in detail the result of Task 1. This report gives only details of Task 1 as far as they are required for the understanding of the text.

On the 4th working meeting, the deadline for the final report was postponed to November 2005 in order to integrate further results of an evaluation of field monitoring (described in chapter 3.3) and experiences with the proposed test procedure for simultaneous operating heat pumps (described in chapter 3.6).

1.4 Participants and projects

1.4.1 Participants

Since the kick-off meeting in March 2003, participants from 10 countries (Austria (AT), Canada (CA), Switzerland (CH), Germany (DE), France (FR), Japan (JP), Norway (NO), Sweden (SE), United Kingdom (UK), USA (US)) worked in the IEA HPP Annex 28.

Due to misunderstandings concerning the funding of the Annex participants fee, the UK had to withdraw the participation in the IEA HPP Annex 28 in the end of 2004.

1.4.2 Work accomplished

In the IEA HPP Annex 28, three main activities have been carried out: the testing of combined operating systems according to existing and proposed standards, the field measurements of pilot plants and the calculation of the systems. Tab. 1 gives an overview of the focus of the single national projects accomplished by the participants of the IEA HPP Annex 28.

Tab. 1: National projects accomplished in the framework of IEA HPP Annex 28

	A/W	B/W	desuperheating	subcooling	Remark
AT		X	X		Direct expansion systems
CA		X	X		System with heating, DHW, air-conditioning
CH	X	X		X	Retrofit heat pump, extension to compact units, ventilation
DE	X				compact units, ventilation
FR		X	X		Part load operation
JP	X				CO ₂ systems
NO		X	X		CO ₂ , propane systems
SE	X	X	X		Test acc. to various standards
US			X		Experience and Evaluation of existing standard ASHRAE 137

In the single tasks the following work has been done:

1.4.2.1 Testing

- Testing of 5 alternate combined operating heat pumps (incl. DX, CO₂) (AT, CH, JP, SE)
- Testing of 6 simultaneous combined operating heat pumps (incl. CO₂) (AT, CA, CH, FR, NO, SE)
- Testing of compact units (DE, CH)
- Testing at part load operation (FR)

1.4.2.2 Field measurements

- Field measurements of 4 ground-coupled direct expansion heat pump systems (AT)
- Field monitoring of 5 compact units (CH, DE)

1.4.2.3 Calculation

- Bin-method for combined operation (alternate, simultaneous) (CH, SE)
- Simplified calculation for CO₂-systems with simultaneous combined operation (NO)
- Calculation approach for compact units (incl. ventilation) (CH, DE)

1.4.3 National teams and national projects

This chapter describes the national teams and the national contributions to IEA HPP Annex 28 in more detail country by country.

1.4.3.1 National team leaders

The national team leaders are the contact persons for the contribution of each country.

In the following the contact information of the national team leaders is given:

Operating Agent	Carsten Wemhöner	carsten.wemhoener@fhnw.ch
Austria	Heinrich Huber	heinrich.huber@arsenal.ac.at
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France	Dominique Hantz	dominique.hantz@cetiat.fr
Germany	Bernd Hafner	drhf@viessmann.com
Japan	Eiji Hihara	hihara@k.u-tokyo.ac.jp
Norway	Jørn Stene	jorn.stene@sintef.no
Sweden	Monica Axéll	monica.axell@sp.se
Switzerland	Thomas Afjei	thomas.afjei@fhnw.ch
USA	John J. Tomlinson	tomlinsonjj@ornl.gov

1.4.3.2 Austria

The Austrian participant has been arsenal research (<http://www.arsenal.ac.at>), Vienna. arsenal research is the National Austrian heat pump test centre.

In Austria, direct expansion heat pump systems which are not covered by the present European standardisation are very popular. Therefore, the focus of the Austrian contribution has been the development of a test procedure for direct expansion heat pump systems among others systems with water heating by desuperheater. Different kinds of storages with and without desuperheater have been tested according to the existing standards EN 255-2 [10] and EN 255-3 [6]: A standard storage (alternate mode, condenser in storage), a storage with fresh water system (alternate mode, switch of the heat pump from space heating system to storage), a combined storage with flat-plate desuperheating heat exchanger and a combined storage with desuperheating lance.

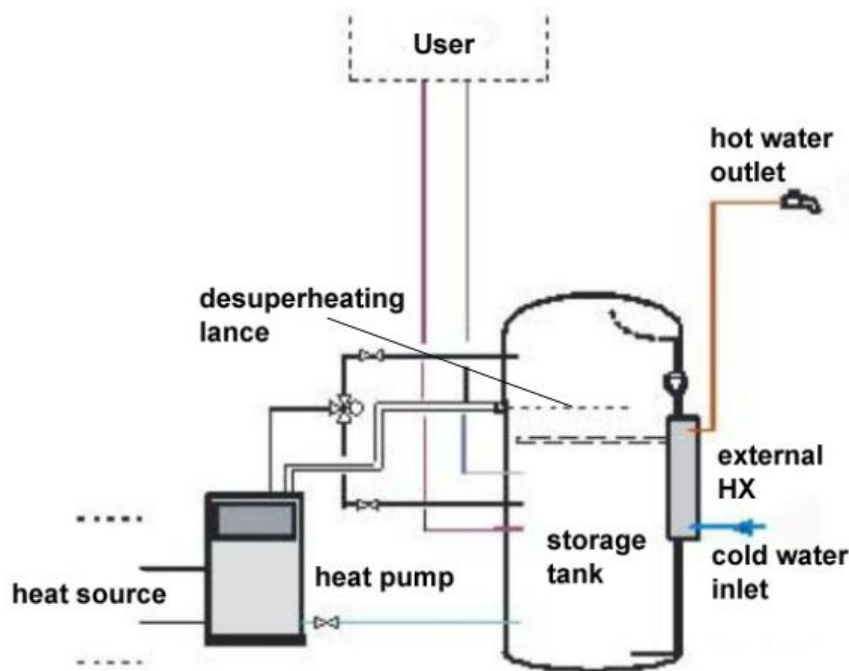


Fig. 4: System configuration with storage desuperheating lance [2]

Fig. 4 shows a sketch of the system with desuperheating lance. The storage tank is separated in two halves by a plate. The desuperheating lance operates on the upper part of the storage used for domestic hot water operation. The lower part of the storage serves as heating buffer storage for the heating system. In case of demand, a switching valve can transfer heating water to the upper part of the storage for domestic hot water use. The advantage of the desuperheating is the higher temperature in the storage at low condensation pressure of the heating system. The fraction of the heat decoupling by the desuperheating lance is about 12%. Depending on the temperature level of the upper part of the storage, the heat decoupling by desuperheating is reduced (self control effect) and more heating capacity is available for the space heating system.

Heat of DHW heating is extracted by means of an external heat exchanger on the DHW outlet side (indirect system), which has advantages concerning calcination and security.

Field monitoring of heat pumps have taken place in the application of single family houses.

Based on the characteristics delivered by the test rig standard testing, Austria has accomplished preliminary comparison of the calculation results to the field measurements.

1.4.3.3 Canada

Canada was represented by the research institute LTE which belongs to the large Canadian Utility Hydro Quebec (<http://www.ireq.ca>).

Canada performed test rig measurements of a ground-source heat pump which provides combined space heating by radiant floor heating and DHW production in winter operation, and air-conditioning and DHW production in summer operation. A test rig has been developed and a prototype heat pump has been tested. The system layout of the prototype heat pump is shown in Fig. 5. The heat decoupling of the heating system is of direct condensation type.

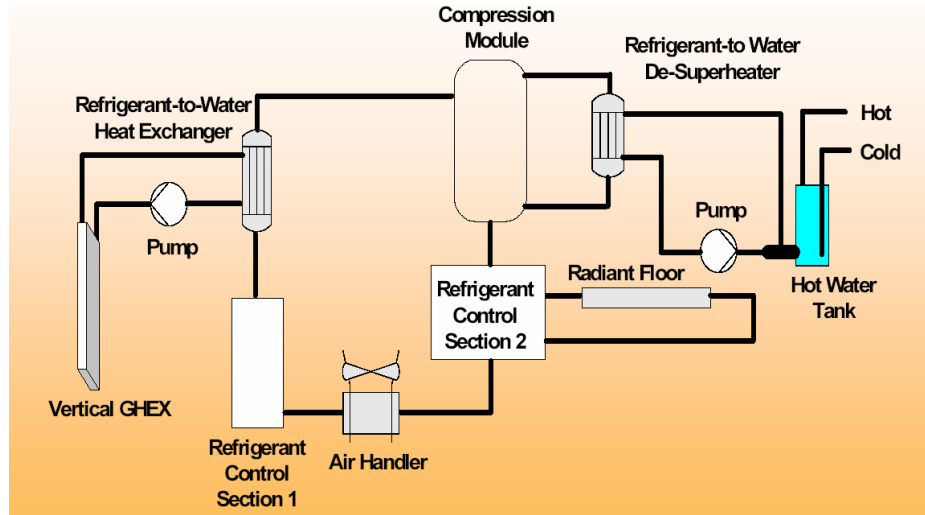


Fig. 5: System layout of the Canadian simultaneous brine-to-water heat pump with desuperheating for air-conditioning/DHW production and space heating/DHW production (source [34])

Both a proposal for a test procedure and a calculation method based on the American standard ASHRAE 137:2001 [20] were developed. The basic equations for the calculation of the SPF according to ASHRAE 137 are given in chapter 2.3. However, ASHRAE 137 only covers air-to-air heat pumps with desuperheater, so the standard had to be extended to the conditions of ground-source heat pumps. Details on the extension are given in chapter 3.2.7 for the calculation and chapter 3.5.7 for the testing.

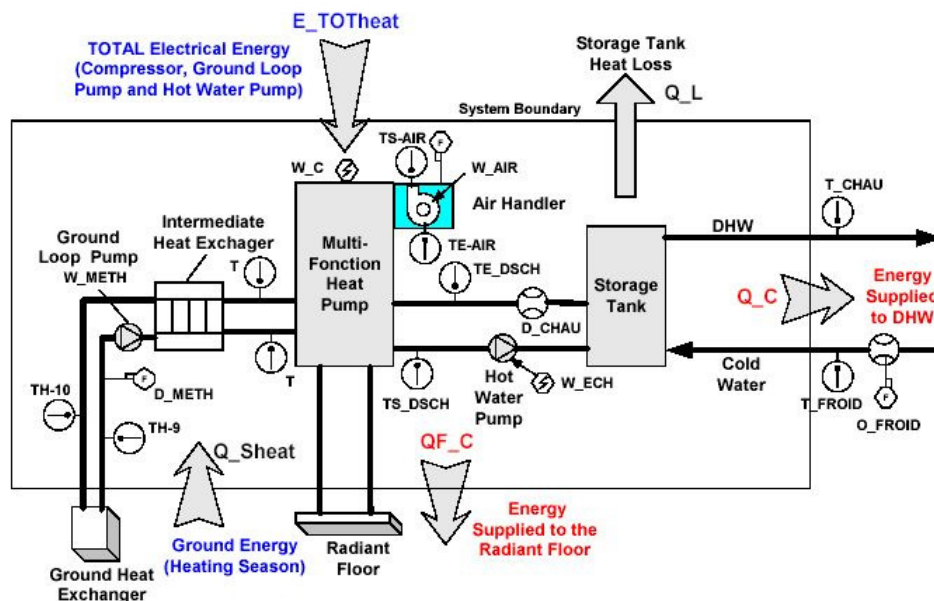


Fig. 6: Sketch of the test rig for the simultaneous brine-to-water heat pump with sensors and energy flows for combined heating operation (source [34])

1.4.3.4 France

For France one of the biggest utilities in Europe, Electricité de France (EDF) (<http://www.edf.fr>) and the national French heat pump test centre CETIAT (<http://www.cetiat.fr>) have participated.

CETIAT has accomplished measurements of a combined brine-to-water heat pump using desuperheating of the compressed refrigerant. Fig. 7 depicts the layout of the system. The system has the following operation modes: space heating only and combined space heating and DHW production by desuperheating. In addition, an electrical back-up heater is used, if there is no space heating, i.e. during nighttime and if the maximum water temperature in the storage tank is less than 60°C.

The testing of the system has been performed according to existing standards and standards being currently discussed or introduced: The space heating-only mode according to EN 255-2 [10], CEN/TS 14825 [12] for part load operation, EN 255-3 [6] and M/324 with the European reference tapping cycles N° 2 [5]. In conclusion, a test procedure for these types of heat pump systems is proposed.

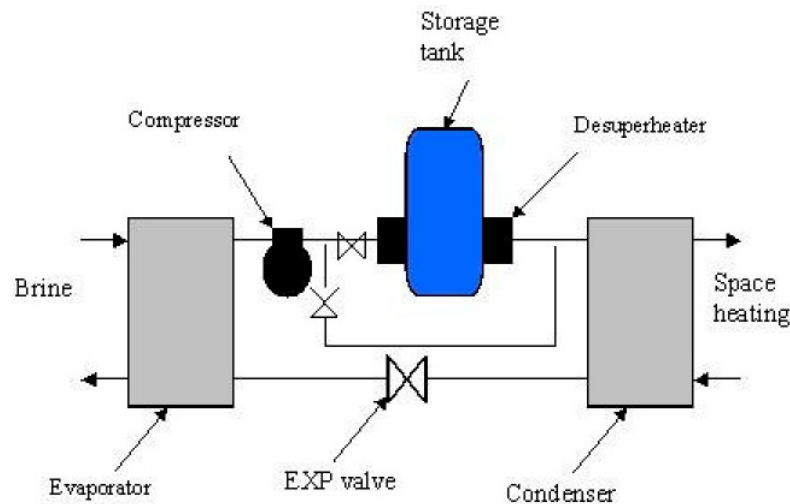


Fig. 7: System layout of the French simultaneous brine-to-water heat pump (source [13])

EDF has performed tests for part load operation of an air-to-water heat pump in heating and in cooling mode at different load ratios [5% to 100%], different capacities of the distribution/emission system [11, 22, 33 l/kW] and different settings of the hysteresis controller. Delivered results shall serve as recommendation for the calculation method. Fig. 8 shows the layout of the test rig for the part load testing.

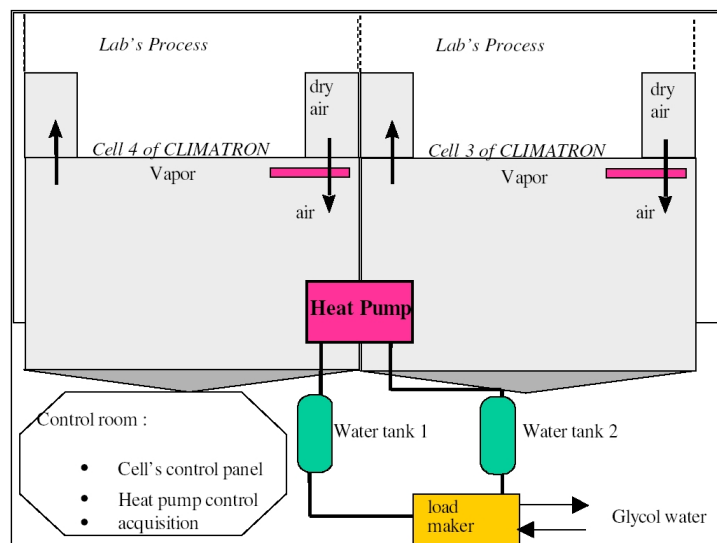


Fig. 8: Layout of the French test-rig for the testing of part-load operation of an air-to-water heat pump (source [13])

1.4.3.5 Germany

Germany was represented by the manufacturer Viessmann (<http://www.viessmann.com>), one of the biggest European manufacturers of heating and building technologies.

Viessmann introduced a compact unit for the application in ultra-low energy houses in the market in 2003. Fig. 9 presents the principle layout of a compact unit. The core element is a heat pump unit, which is coupled with a heat recovery system. The heat pump delivers heat to the space heating system and to the integrated DHW storage. For peak load, an electrical back-up heating delivers the required energy. An optional integration of a solar collector is possible for some systems, as well. Some units can also be used for cooling purposes by reverse operation of the heat pump.

Viessmann produces several modular systems, which are used for low energy houses. Thus, Viessmann is interested to have calculation methods to assess the seasonal performance and compare it to other heating systems. Hence, the project contribution of Germany refers to the development of a calculation method to be applied for these German passive house units. These methods shall be able to compare heat pump with other heating systems, e.g. gas boilers, for instance based on the European provisional standard prEN 15316 [16]. In the framework of a revision in the German standardisation, Viessmann has introduced the bin approach discussed in the Annex on a monthly basis on the national level.

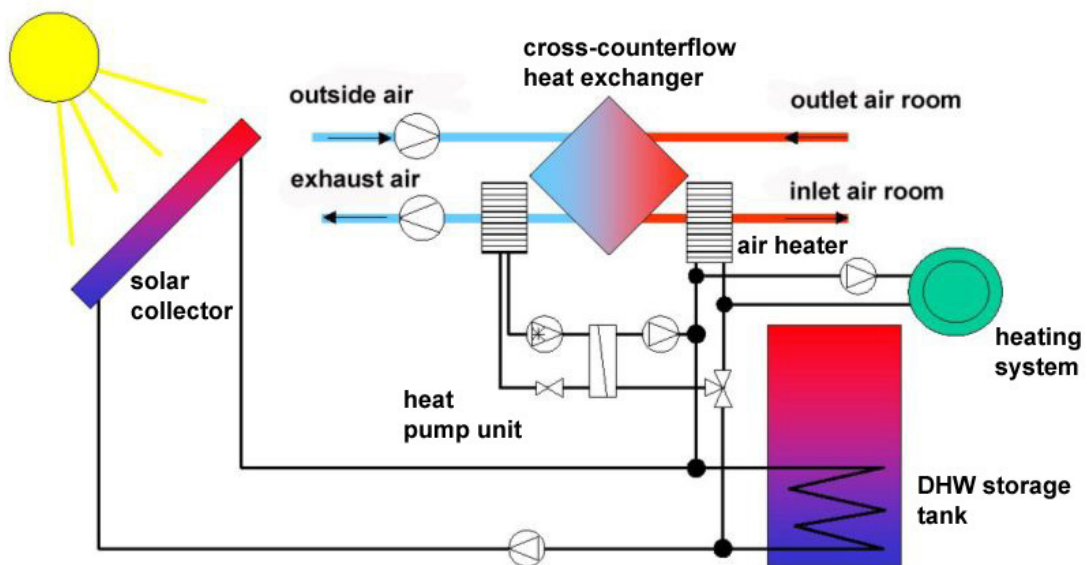


Fig. 9: Principle sketch of a compact unit (source [2])

Furthermore, Viessmann tries to reduce testing expenses by a dynamic heat pump testing based on parameter identification. This has been applied successfully for the determination of the COP (heating mode, EN 255-2) in a field test unit.

In the testing of solar components and systems, dynamic testing has been successfully introduced in the standardisation (see EN 12977 [35]). Viessmann tries to adapt these approaches for the heat pump testing, which is described in chapter 3.7.

1.4.3.6 Japan

Japan was represented by a national team with numerous participants. The national team leader was the University of Tokyo. The Heat Pump and Thermal Storage Technology Center of Japan (HPTCJ) (<http://www.hptcj.or.jp>) has organised a task force of the companies DENSO Co., Toshiba Carrier, Daikin Industries, Mitsubishi Electric, Hitachi Air Conditioning Systems, Hitachi Home and Life Solutions, Sanyo Electric and Matsushita Electric Industries, which are active in the Japanese heat pump market. Moreover, the utilities TEPCO (Tokyo Electric Power Company), Kansai Electric Power Company and Chubu Electric Power Company participated in the project work of IEA HPP Annex 28.

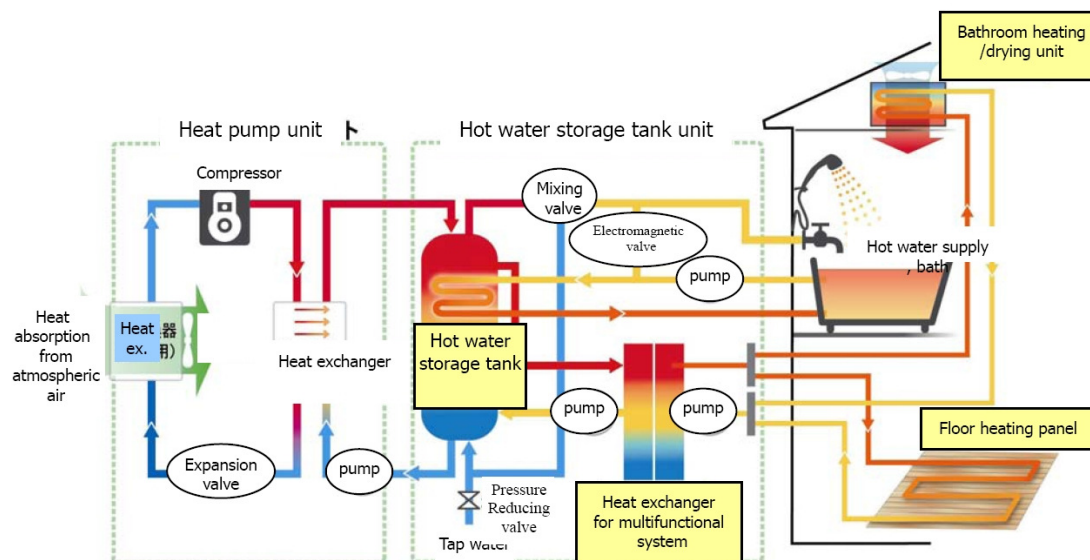


Fig. 10: System layout of the Japanese CO₂ heat pump with combined DHW and SH(source [21])

In Japan, heat pump water heaters using CO₂ have been introduced in the market in 2002. Therefore, the focus of Japan has been the development of adequate test procedures and calculation methods for these systems. Japan has performed test rig measurements of this CO₂-system with combined floor heating and possible additional heat use for radiator heating or drying. The hot water-only operation has been tested according to the Japanese Standard JRA 4050:2004 [23]. Testing has been performed for three seasons (winter, summer, intermediate) with fixed heat load and tapping profiles for the respective seasons. There is one test point per season, and tests are performed for the operation modes DHW-only and combined SH/DHW. By defining tapping patterns, transient effects due to the respective load profile are included in the test results.

Based on this testing approach of JRA 4050:2004 [23] Japan has worked out a calculation method with a "typical day approach" for the respective seasons described in chapter 3.2.4.1. The results of the testing for the respective seasons are weighted with a number of days in the season for all three operation modes to receive the seasonal performance factor of the heat pump operation.



Working fluid for the heat pump	Carbon dioxide
Tank capacity	460 L
Power-supply voltage	Single-phase 200 V
Heating capacity	6.0 kW
Electric power input	1.4 kW
Hot-water temperature in the tank	Approximately between 65°C and 90°C (automatically controlled in response to hot-water consumption)
Hot-water temperature at the time of use	Approximately between 35°C and 50°C (in 1°C intervals), 60°C
Maximum hot-water pressure	190 kPa (pressure reducing valve: 170 kPa)
Compressor	Variable-speed hermetic compressor
Size of the heat pump unit	640 mmH, 820 mmW, 300 mmD
Size of the hot-water storage unit	1890 mmH, 720 mmW, 800 mmD

Fig. 11: System for combined DHW/floor heating tested in Japan (source [21])

1.4.3.7 Norway

The Norwegian National Team, represented by SINTEF Energy Research (<http://www.sintef.no>), focused on CO₂ heat pump systems, which are favourable for combined operation. The report *“Residential CO₂ Heat Pumps for Combined Space Heating and Hot Water Heating – System Design, Test Procedures and Calculation of SPF”* [33] by J. Stene provides, among other things, an overview of possible designs and presents recommended test conditions and test procedures for B/W CO₂ heat pumps for combined space heating and hot water heating.

The results from the theoretical and experimental PhD study *“Residential CO₂ Heat Pump System for Combined Space Heating and Hot Water Heating”* [15] by J. Stene have also been evaluated in the context of the IEA HPP Annex 28. Fig. 12 shows the principle layout (left) and the prototype (right) of the analysed CO₂ heat pump system. The heat pump unit is equipped with a so-called tripartite gas cooler for preheating of DHW (A), low-temperature space heating (B) and reheating of DHW (C). The heat rejection from the tripartite CO₂ gas cooler can be compared with the heat rejection from a conventional heat pump using a condensate subcooler, a condenser and a desuperheater. The extensive experimental results have been used to evaluate the COP at varying operating conditions, compare the system performance with state-of-the-art B/W heat pumps and derive special testing requirements for the testing of heat pumps using CO₂. A simplified seasonal performance calculation method using fixed source and sink temperature has also been evaluated.

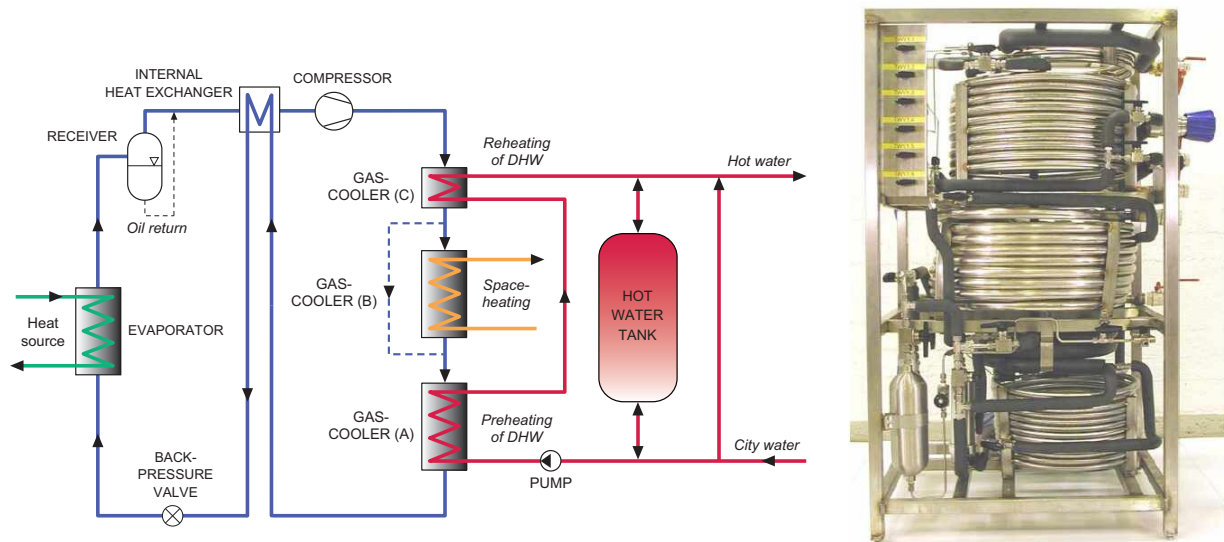


Fig. 12: B/W CO₂ heat pump for preheating of DHW, space heating and reheating of DHW measured in the Norwegian project: hydraulic scheme (left) prototype (right) (source [33])

Norway has also performed test rig measurements for a combined propane heat pump unit using a desuperheater for hot water heating. The results are presented in the report *“Testing of a Residential Brine-to-water Propane Heat Pump Unit for Combined Space Heating and Hot Water Heating”* [32].

The Norwegian participation in IEA HPP 28 has been financed by the Norwegian company Shecco Technology (<http://www.shecco.com>).

1.4.3.8 Sweden

SP (<http://www.sp.se>), the Swedish National Testing and Research Institute was the national team leader of Sweden. Chalmers University of Technology as well as the manufacturers Thermia Värme AB, NIBE AB and IVT AB were members of the national team.

Ground-source (B/W) heat pumps are the most frequently sold heat pump type in Sweden. Therefore, the Swedish project was concentrated on this heat pump type in the beginning. After defining the boundary of the system that should be included in the proposed test and calculation method, the first draft of a test procedure was outlined. Thereafter, an alternately operating ground-source heat pump system depicted in Fig. 13 left side was measured extensively according to the different European standards EN 14511 [11], EN 255-2 [10] and EN 255-3 [6] and according to Mandate M/324 [5]. The draft test procedure was also evaluated experimentally.

Due to the Swedish building regulation exhaust-air heat pumps are also popular in Sweden. Systems with combined space and domestic hot water heating are common for this market segment. Fig. 13 right side presents a simultaneous combined configuration of the exhaust-air heat pump system. In Sweden, a national test method for single-family houses has been in use since the end of the Eighties. However, this method was limited to Swedish circumstances regarding climate and regulations, and also to a certain range of house size. In the later part of the Swedish project, the integration of test procedures and calculation methods for exhaust air systems was performed, as well. To start with, this was based on the analysis of existing measurements.

Conclusions from the experimental results have been introduced in the proposal of a test procedure described in chapter 3.5.2.2, which is outlined for ground-source, outside air-source and exhaust air-source heat pumps and further developed to an adequate calculation method described in chapter 3.2.4.2 based on the definitions of the Annex 28, see chapter 2.5. Finally, parts of the proposed test procedure were evaluated experimentally with an exhaust-air heat pump in simultaneous combined operation.

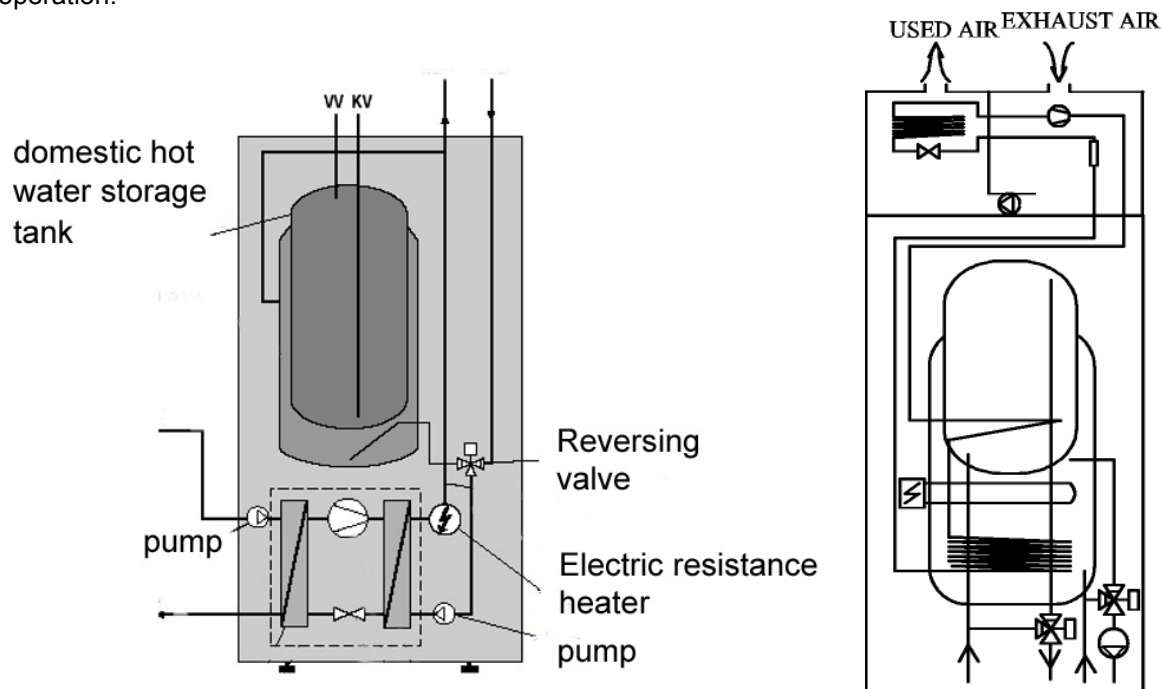


Fig. 13: Sketches of the alternate combined ground-source heat pump (left) and the simultaneous combined exhaust-air-to-water heat pump (right) measured in the Swedish project (source [17])

1.4.3.9 Switzerland

Besides the project management as operating agent, the Swiss contribution referred both to the development of a test procedure and a calculation method. In Switzerland, one research focus has been the development of so-called “retrofit heat pumps” for the replacement of boilers in existing buildings. These research activities led to the development of a cascade heat pump using condensate subcooling for domestic hot water heating. Fig. 14 shows the layout of the so-called Swiss Retrofit Heat Pump (SRHP). Testing of two combined operating heat pumps, the above simultaneous operating heat pump and an alternate operating heat pump, was performed at the Swiss national heat pump test centre WPZ Töss (<http://www.wpz.ch>) [9]. A respective calculation method called FHBB-method [14] has been proposed and discussed in the IEA HPP Annex 28. The calculation method in connection with this testing is based on the bin-methodology described in chapter 3.2.4.3.

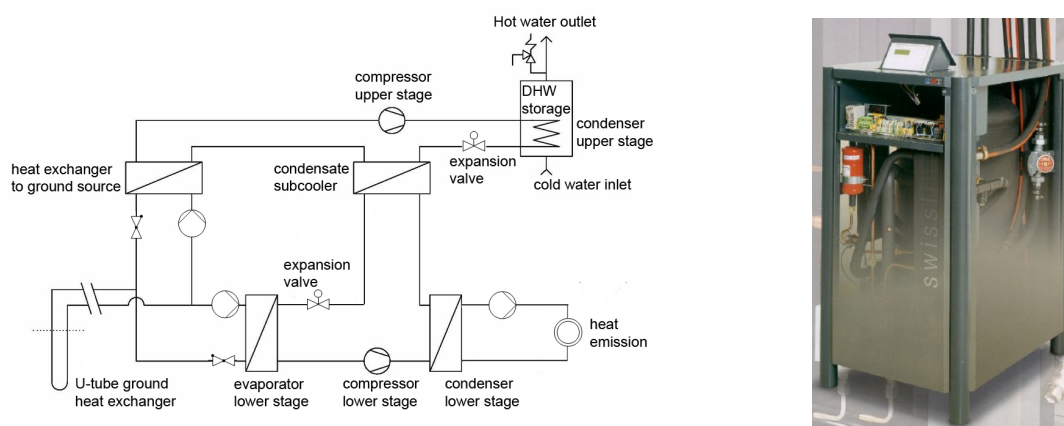


Fig. 14: Principle sketch of the Swiss simultaneous B/W heat pump for simultaneous space heating and DHW heating with condensate subcooling (source [2])

In a follow-up project, the test procedure and the calculation method were extended to compact units for low and ultra-low energy houses, which include a ventilation heat recovery system. Optionally, cooling in summer and solar energy input are sometimes possible. The extension is carried out in co-operation with the HTA Lucerne, which operates an accredited ventilation test rig. At the HTA Lucerne, the test procedure for compact units was developed in co-ordination with the extension of the calculation carried out at the FHNW. To validate the calculation approach, field monitoring of two compact units was carried out in parallel. Two pilot plants, one according to the Swiss MINERGIE® standard, corresponding to the space heating requirement of about 50 kWh/(m²a) and one according to the Swiss MINERGIE-P® standard (<http://www.minergie.ch>), corresponding to a space heating energy requirement of about 15 kWh/(m²a), which is similar to the German passive house standard, have been measured in detail to gain experience with field behaviour of these systems, prove and optimise the functionality of the compact units and validate the calculation approach with the measured data. Fig. 15 shows the pilot plant and a sketch of the used compact unit.

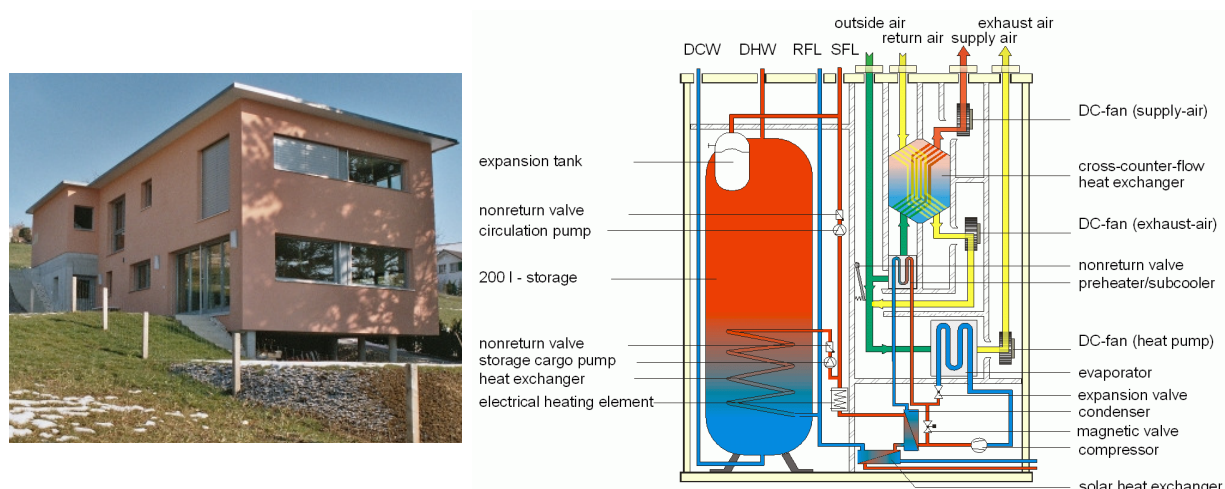


Fig. 15: Pilot plant in Gelterkinden, canton BL, and sketch of the used compact unit (source [37])

1.4.3.10 USA

In autumn 2003, the Department of Energy (DOE) mandated the work in the IEA HPP Annex 28 to the Oak Ridge National Laboratory (ORNL) (<http://www.ornl.gov>), which since has been the national team leader of the USA.

The USA is developing a combined operating heat pump with air conditioning/dehumidification and DHW-heating function in summer operation and space heating and DHW heating function in winter operation. A principle sketch of the system layout under investigation is shown in Fig. 16.

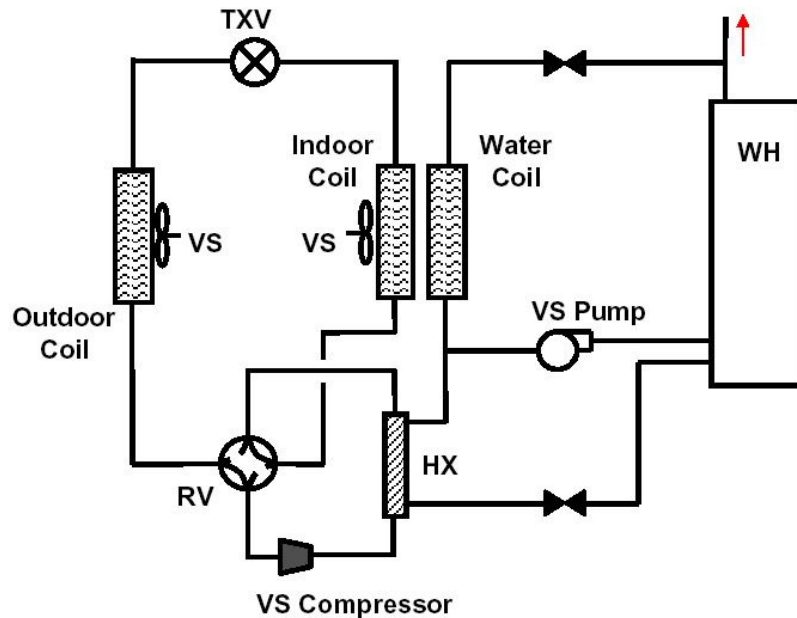


Fig. 16: Sketch of the US integrated heat pump for space heating, space cooling, dehumidification and DHW heating (source [31])

The integrated heat pump consists of a modulating compressor, two variable speed fans (components VS) and four heat exchangers, two air-to-refrigerant, one water-to-refrigerant and one air-to-water. By the air-to-water heat exchanger excess hot water that has been generated in the cooling and dehumidification operation can be used to condition the ventilation air. Both water heating and air conditioning can be done at the same time. Compressor, indoor fan and water pump speed are used to control both humidity and indoor temperature. Thereby, all space conditioning including ventilation and water heating load can be met. An advantage of the system layout with regard to other configurations is the possibility of covering as well pure ventilation operation, which only requires dehumidification.

The US has provided a list of relevant standards for testing and calculation, which are partly integrated in Tab. 4 in chapter 2.3.

1.5 Co-operation with external organisations

Since the results of the IEA HPP Annex 28 are intended as recommendations for respective international standardisation committees in order to implement the results in standards, the IEA HPP Annex 28 has a strong link to standardisation. For the continuous transfer of the results to the respective standardisation working groups on the testing of heat pump systems and the calculation of heating systems the following links to the respective CEN Technical Committees TC 113 and TC 228 have been established in the form of an external liaison, which basically guarantees the information exchange.

1.5.1 CEN/TC 113 - Heat pumps and air conditioning units

The main task of the Technical Committee CEN/TC 113 are standards for product testing. TC 113 is active in the revision of test procedures for air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling.

- CEN/TC 113/WG 8 developed the new test standard EN 14511 [11] replacing EN 255-2 [10]. The EN 14511 came into operation in Sept. 2004 and introduces some changes in the test procedure. Details are given in chapter 2.2.2.
- A new working group CEN/TC 113/WG 10 is in constitution to revise the EN 255-3 [6], the actual standard for testing of heat pump water heaters. Members of the national teams of France, Sweden and Switzerland are nominated for the new working group. Convenor of the working group will be the new head of the Swiss national test centre WPZ Töss, which recently moved to NTH Buchs. (<http://www.wpz.ch>)
- CEN/TC 113/WG 11 deals with the testing of direct expansion heat pumps. Results from the testing at arsenal research in the framework of the IEA HPP Annex 28 will be used in this working group.

On the Madrid working meeting in March 2003, TC 113 was informed about the activities of the IEA HPP Annex 28. The secretary of TC 113 was asked to create a liaison between TC 113 and the operating agent of the IEA HPP Annex 28 (decision 208). The external liaison has been accomplished in September 2003.

1.5.2 CEN/TC 228 - Heating systems in buildings

CEN/TC 228 is the Technical Committee (TC) active in the field of heating systems in buildings. Subjects covered by TC 228 are design, installation, commissioning, instruction for operation, maintenance and use of heating systems as well as calculation methods for the design heat load and the system performance.

- CEN/TC 228/WG 1 is to develop design criteria of water based heating systems. The national team leader of Switzerland in IEA HPP Annex 28 is participating in WG 1.
- CEN/TC 228/WG 4 is to develop calculation methods for heating systems in the framework of prEN 15316 (formerly prEN 14335) [16]. The Operating Agent of the IEA HPP Annex 28 is participating in the subgroup of WG 4 which deals with heat pump systems.
- Calculation of DHW systems is also treated in WG 4, so results of the IEA HPP Annex 28 on this subject can be directly transferred to the standardisation process, as well.

An external liaison to the CEN/TC 228 has been established on the TC 228 meeting in October 2004.

2 RESULTS OF TASK 1

The results of Task 1, the system and standardisation survey, has been extensively presented in an Interim Report [2]. In the following, basic results needed as background for the results of Task 2 and Task 3 are summarised.

2.1 Systems on the market and under development

2.1.1 Alternate operation mode

Europe

The most common systems for combined operating heat pumps nowadays are alternate operating systems. Alternate operating systems are wide-spread on the European market. The Swedish ground-source heat pump depicted in Fig. 13 is a common system with alternate combined operation. Moreover, the system configurations shown in Fig. 17 can be found.

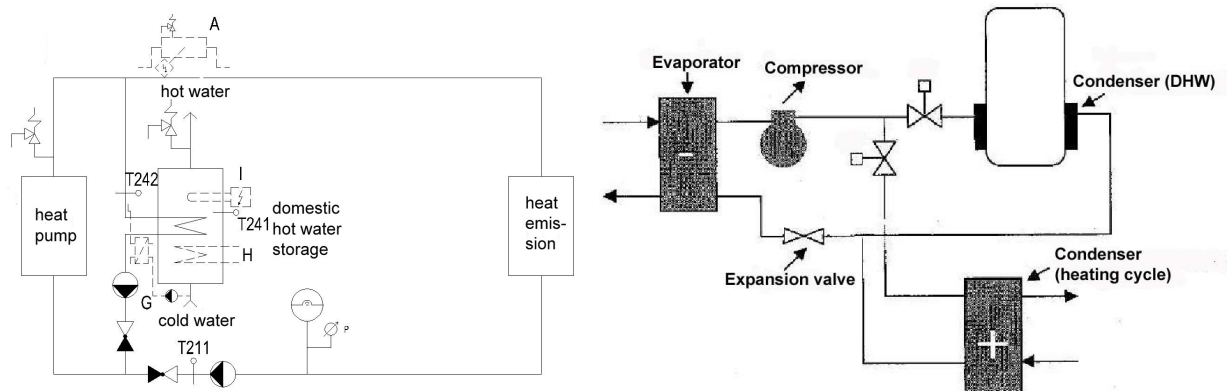


Fig. 17: Two examples of configurations of alternate operating systems found in Europe, see also Fig. 13 left

The system configuration can be distinguished as well by the type of storage and the hydraulic integration of the storage. Configurations found on the market include:

- Intermediate circuit with external refrigerant/water heat exchanger for the DHW circuit
- DHW water/water heat exchanger around the storage tank (double shell DHW storage tank)
- Refrigerant condenser inside the storage tank or around the storage
- Combined storage: a smaller DHW storage is integrated in a bigger heating buffer storage

Moreover, system configurations can be distinguished by the integration of storages in the heating cycle and the number of heating cycles connected to the heat pump. The most common configurations are

- systems without storage
- systems with serial storage in the return of the hydronic heating cycle
- systems with parallel heating buffer storage, optional with more than one heating cycle connected

2.1.2 Simultaneous operation mode

Besides the above described systems of the national project the following systems from Norway and Sweden have been encountered on the market.

Sweden

While in Sweden ground-source and outside air-source heat pumps are usually operated in alternate mode, the common configuration for exhaust-air heat pumps are actually a mantle storage tank, where a desuperheater heats the domestic hot water in the inner part, while a condenser is placed in the mantle of the storage tank water to work on both space heating and domestic hot water production.

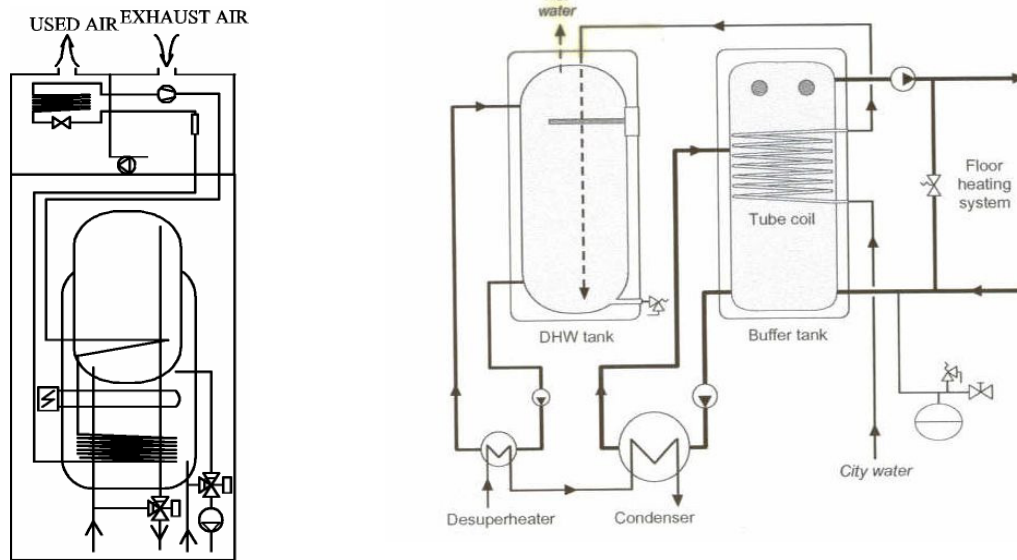


Fig. 18: Swedish desuperheating system using exhaust air (left) [17] and Norwegian ground-coupled desuperheater system (right)

Norway

In Norway many suppliers and manufacturers offer heat pump systems for simultaneous space heating and hot water heating. Fig. 18 (right side) shows a principle sketch of one possible system design. The condenser heat is used for space heating as well as for preheating of DHW by means of a tube coil mounted inside the buffer tank for the space heating system. A desuperheater is used to heat the DHW to the set-point temperature ($> 60^{\circ}\text{C}$). The main draw-back for heat pumps using only a desuperheater for DHW heating is, that heat can only be supplied from the heat pump unit to the DHW system as long as the compressor is running. The DHW production is therefore inevitably linked to the space heating demand of the house. With the alternative design in Fig. 18, where the condenser heat is used for preheating of DHW, the heat pump will be able to cover a larger share of the annual DHW heating demand.

North-America

Whereas in Europe combined operation is mostly of alternate configuration, North America has more simultaneous operation using desuperheating. Often, these systems are operated in summertime with combined cooling/air-conditioning and DHW production.

Japan

In Japan systems with combined floor heating and DHW production are in the market introduction. The typical configuration of these systems corresponds to the investigated system in the national Japanese project presented in Fig. 10, where the heat pump and the different heat users are decoupled by a storage tank that can provide the different required temperature levels.

2.2 International Standardisation

In this chapter existing international standards relevant for the scope of the IEA HPP Annex 28 are shortly presented, as far as the results of Task 2 and Task 3 are referring to the respective standards.

2.2.1 Test standards in Europe

Standards for testing of heat pumps are dealt with in the technical committee CEN/TC 113 of the European standardisation organisation CEN.

In Europe heat pumps are tested separately for the space heating and the domestic hot water mode. Testing for combined operation does not exist. Tab. 2 gives an overview of the most important standards for heat pump testing in Europe. EN 255-2 was replaced by EN 14511 in the end of 2004, and is now in operation. As an evaluation of EN 14511 in comparison to EN 255-2 has been made in the framework of the IEA HPP Annex 28, EN 255-2 is still presented in the Tab. 2.

Tab. 2: Overview of most important standards for the testing of heat pumps in Europe

European standards			
Designation	Title	Application	Output values/Remarks
EN 255-2:1997 (replaced by EN 14511 in the end of 2004)	Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors - Heating mode - Part 2: Testing and requirements for marking for space heating units	Test procedure Electrically driven Space heating-only operation	Heating capacity and COP at the test conditions: A/W: A-7/W50, A2/W35, A7/W50 B/W: B0/W35; B0/W50; B5/W50 W/W: W10/W35; W10/W50; W15/W50
EN 14511	Air conditioners, liquid chilling packages and heat pumps with electrically-driven compressors for space heating and cooling	Test procedure Electrically driven Space heating-only Space cooling-only	Heating capacity and COP in the test conditions according to Fig. 19, right hand side. Replaces EN 255-2, EN 814, EN 12055
EN 255-3:1997	Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors - Heating mode - Part 3: Testing and requirements for marking for sanitary hot water units	Test procedure Electrically driven DHW-only operation	COP _t for the tapping of DHW at test points according to Fig. 20 <ul style="list-style-type: none"> • Reference hot water temperature • Maximum volume in a single tapping • Heat-up time • Heat-up energy input • Stand-by power input
CEN/TS 14825	Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling - Testing and rating at part load conditions	Test procedure Fixed capacity units Step-controlled variable capacity units Continuous controlled variable capacity units	COP _{50%} EER _{50%} respectively

The different standards are described shortly in the following

2.2.2 EN 255-2 and EN 14511 (heating and cooling operation mode)

EN 14511 [11] is the European standard for the testing of heat pumps in space heating-only or cooling-only mode and replaces the former standard EN 255-2 [10]. The testing is performed as steady-state testing at defined test points. Principal testing results are the heating capacity and the COP at defined test points. COP-values contain the input power to pumps which is needed to overcome the internal pressure drop in the evaporator and condenser as well as additional electrical power for control and an eventually installed oil supply heating. Defrosting is considered in the testing

operation, as far as a defrosting occurs during the testing at the respective test points in case of air-source heat pumps. In EN 14511, the determination of the test mass flow rate has been changed. While formerly a manufacturer given mass flow rate was used, it is now determined by a given temperature difference over the evaporator/condenser at a standard rating point, and this determined mass flow is also used for the other so-called application rating points. Test points have also been changed. While only three test points were provided by EN 255-2, 10 test points are prescribed by EN 14511. Fig. 19 shows the test points according to EN 255-2 and EN 14511.

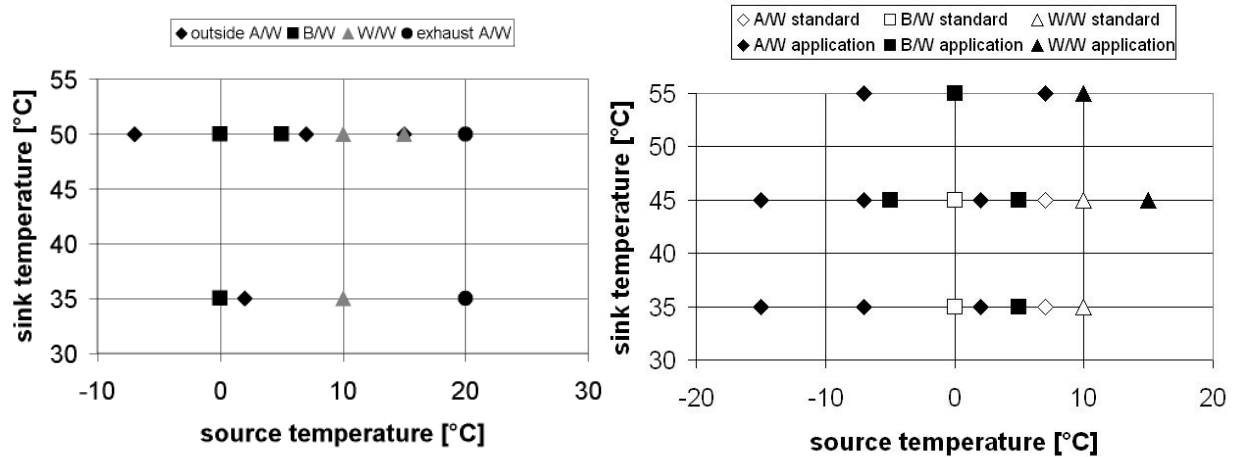


Fig. 19: Test points for different heat sources according to EN 255-2 (left) and EN 14511(right)

2.2.3 EN 255-3 (Domestic hot water operation mode)

The European standard for the DHW-mode of heat pumps uses a black-box system where testing includes the heat pump, the DHW storage, the internal storage loading pump as well as the input power to pumps to overcome the internal friction losses of the source heat exchanger.

In contrast to the steady-state testing of EN 255-2 or EN 14511, EN 255-3 evaluates a cycle.

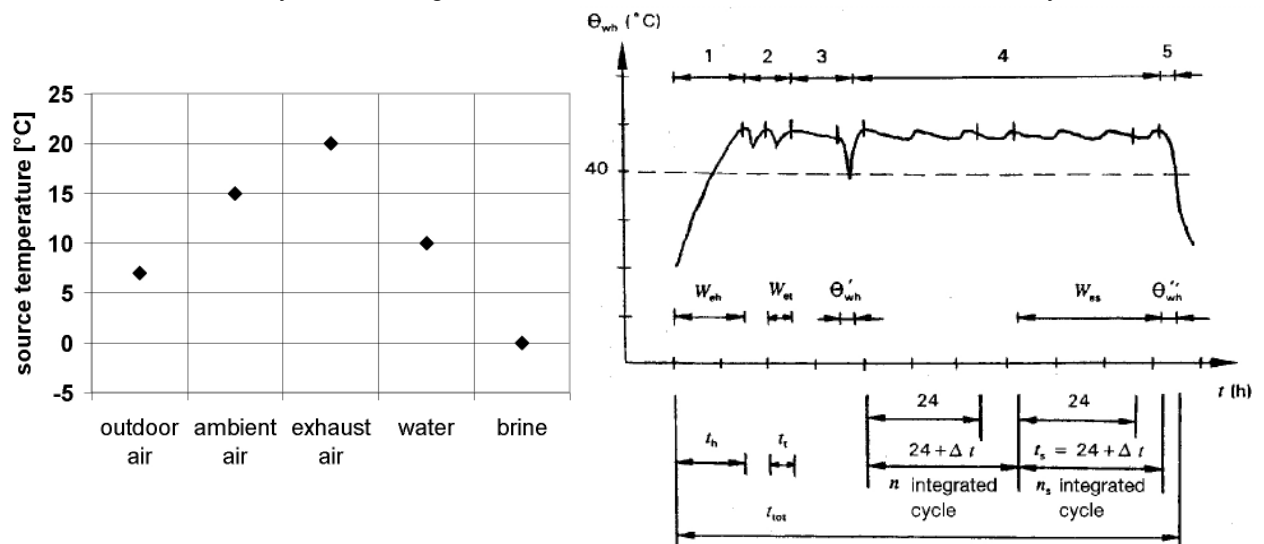


Fig. 20: Test points for different heat sources (left) and tapping pattern (right) according to EN 255-3

Thus, the COP-value is derived by averaged evaluation of the energy balance for the tapping cycle. For the determination of the COP, hot water draw-offs of half the storage volume are repeated, until the energy amount of the tapped hot water is within a 10% range.

Although comfortable to use, the major draw-back of EN 255-3 is a very long testing time of at least 4 days for a single test point, mainly due to the testing of storage stand-by losses.

Fig. 20 shows the test points (left) and the curves of the water temperature exiting the hot water storage tank. The testing cycle consists of five phases. The relevant phases for the determination of the COP-value for the extraction of domestic hot water, called COP_t in EN 255-3, are phase 2 and phase 4. The single phases are described shortly in the following.

Phase 1: Heating up the storage from equilibrium with the environment of 20°C until the controller switches-off the system.

Phase 2: Repeated tapings of half the storage volume. This procedure is repeated until the energy content in the extracted amount of hot water does not vary more than 10 % to the preceding. The last cycle is used to calculate a preliminary COP value according to the eq. 1

$$COP_{phase2} = \frac{Q_t}{W_{et}} \quad [-] \quad \text{eq. 1}$$

where

COP_{phase2}	preliminary COP value evaluated from phase 2 measurements	(-)
Q_t	energy content of the tapped sanitary hot water during the last cycle of phase 2	(J)
W_{et}	effective electrical energy input during the tapping period (phase 2)	(J)

The phase 2 COP contains the storage losses.

Phase 3/Phase 5: Hot water tapping until reaching a tap temperature of 40°C. This is used to evaluate the total capacity of the storage and the hot water reference temperature θ_{wr} .

Phase 4: The storage is loaded again and left in stand-by operation. The electrical energy input is recorded during this stand-by period. The energy input during phase 4 is used to calculate the stand-by power P_{es} by the equation

$$P_{es} = \frac{W_{es}}{t_s} \quad [W] \quad \text{eq. 2}$$

where

P_{es}	effective electrical power input during the standby period	(W)
W_{es}	effective electrical energy input during standby period	(J)
t_s	measuring time for standby power input	(s)

The value P_{es} is used to correct the preliminary COP-value of phase 2 for the storage stand-by losses to receive the COP_t -value according to the equation

$$COP_t = \frac{Q_t}{W_{et} - P_{es} \cdot t_t} \quad [-] \quad \text{eq. 3}$$

where

COP_t	COP for tapping sanitary hot water according to EN 255-3	(-)
Q_t	energy content of the tapped sanitary hot water during the last cycle of phase 2	(J)
W_{et}	effective electrical energy input during the tapping period (phase 2)	(J)
t_t	Duration of tapping and reheating time period of the evaluated draw off (phase 2)	(s)
P_{es}	effective electrical power input during the standby period	(W)

2.2.4 Mandate M/324

Mandate M/324 [5] on the harmonisation of test standards for household appliances contains 5 reference tapping cycles over 24 hours with differing total amount of tapped hot water.

Tab. 3 shows the tapping cycle N° 2 as an example. The flow-rate is distinguished in kitchen type draw-offs and other draw-offs. For the latter the specific flow rate (as given by the manufacturer) is applied. For the former the kitchen flow rate is applied as described in prEN 13203, namely at 2/3 of the specific flow rate.

Tab. 3: European tapping cycle N° 2 according to Mandate M/324 (taken from [5])

EU reference tapping cycle nr. 2						
	hr.min start	energy (kWh)	type	ΔT desired(K), to be achieved during tapping	min. ΔT (K), =start of counting useful energy	flow rate, S=specific rate, $R=$ $\frac{2}{3} \cdot S$
1	07.00	0,105	small		15	S
2	07.15	1,400	shower		30	S
3	07.30	0,105	small		15	S
4	08.01	0,105	small		15	S
5	08.15	0,105	small		15	S
6	08.30	0,105	small		15	S
7	08.45	0,105	small		15	S
8	09.00	0,105	small		15	S
9	09.30	0,105	small		15	S
10	10.30	0,105	floor	30	0	S
11	11.30	0,105	small		15	S
12	11.45	0,105	small		15	S
13	12.45	0,315	dishwash	45	0	R
14	14.30	0,105	small		15	S
15	15.30	0,105	small		15	S
16	16.30	0,105	small		15	S
17	18.00	0,105	small		15	S
18	18.15	0,105	clean		30	R
19	18.30	0,105	clean		30	R
20	19.00	0,105	small		15	S
21	20.30	0,735	dishwash	45	0	R
22	21.15	0,105	small		15	S
23	21.30	1,400	shower		30	S
total		5,845				

equivalent hot water litres at 60°C

100,2

2.2.5 CEN/TS 14825 (Part load operation)

No standards for part load operation testing of heat pumps are available in Europe. However, CEN/TC 113/WG 7 published a Technical Specification CEN/TS 14825 [12] dealing with a reduced capacity test. The test uses the same testing equipment as EN 255. In the scope are air conditioners, heat pumps and liquid chilling packages. Test procedures are described for fixed capacity units (ON/OFF operation) as well as for step or continuously controlled capacity units.

For fixed capacity units, the test is performed by operating the unit for 30 minutes with compressor ON and 30 minutes with compressor OFF. If the unit is not in the range of 40% to 60% of the steady-state heating capacity and the COP differs more than 5% from the rated COP, another test with proportionally changed ON and OFF times shall be performed.

For step control capacity units, the test results should be in a range of the heating capacity of 47% to 53%. If the values fall below 47% and the unit has another step higher than 50%, an additional test at this higher step shall be performed and vice versa. If there is no other step, then the test shall be performed with an adapted OFF period as in the case of the fixed capacity units.

For continuously controlled capacity units, the unit shall be operated in the range of 40% to 60% of the rated capacity. If the control does not permit this operation range, the test is performed in the same ways as for the step controlled units. The unit should operate continuously during the test. The only permitted exception is a defrosting cycle. Results should be in the range of 47% to 53%. If not, another test at a different operating point in the range of 40% to 60% should be performed. If the control does not allow that, the second test is performed as for step controlled units.

The standard delivers a $COP_{50\%}$ or an $EER_{50\%}$ respectively, which is to be specified in the test report and should be added to the technical data sheets of the manufacturer.

2.2.6 Calculation of seasonal performance in Europe

In Europe, standardisation in the building sector is currently in the process of harmonisation in the framework of the EU-Directive on Energy Performance of Buildings (EPBD), which is presented in chapter 1.1.1. The harmonisation refers both to building and building technologies as HVAC systems. Presently, a calculation method for the seasonal performance of heating systems on the European level is developed in CEN/TC 228/WG 4 in the framework of prEN 15316 [16], formerly denoted as prEN 14335. The number had to be changed in 2005 due to CEN rules. The standard covers the calculation of space heating and domestic hot water production. The general part prEN 15316-1 describes the basic calculation scheme applied to all systems. The methodology is shown in Fig. 21.

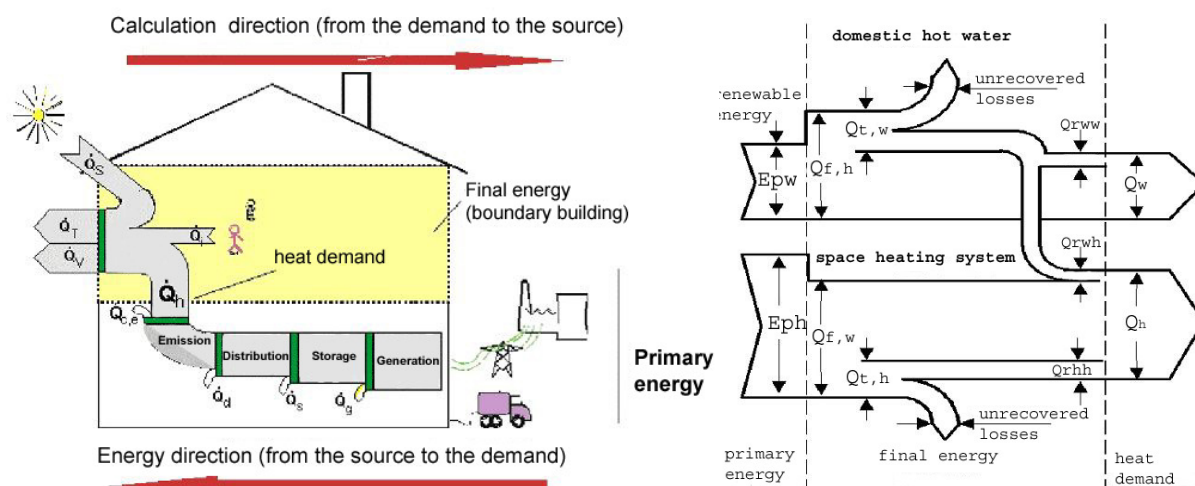


Fig. 21: Calculation methodology, system boundaries and energy balance of prEN 15316 (source [16])

The calculation direction is reverse to the direction of the energy flow, i.e. the calculation direction is from the consumer, the energy sink, to the generation and further converted to the primary energy. The final result of the calculation is the so called “expenditure factor”, the reciprocal value of the seasonal performance factor (SPF) based on primary energy. The conversion factors to transform “used energy” to “primary energy” are a matter of national Annexes to the general calculation scheme. The interface in the beginning of the calculation is the standard energy requirement delivered by the CEN/ISO standard EN ISO 13790 [4]. Then, the four subsystems “Emission”, “Distribution”, “Storage” and “Generation” are calculated. Each subsystem gets as input the energy requirement of the previous subsystems and delivers

- the energy input to cover this requirement
- the losses of the subsystems, distinguished in
 - total losses
 - recoverable losses
- required auxiliary energy.

Thus, energy input is distinguished in energy to cover the requirement and auxiliary energy needed for the operation of the system.

Losses are distinguished in “*recovered*”, “*recoverable*” and “*unrecoverable*” losses. System losses are not necessary lost. They can be recovered to lower the space heating requirement, if the component is installed in the heated space. Whether recoverable losses are recovered or not, is determined by the gain-loss ratio according to EN ISO 13790. The gain-loss factor gives a number, how much of the heat gains (internal gains by equipment and persons and solar gains) are really used for heating purpose based on the gain-loss ratio and time constant of the building. Recovered losses can be directly accounted in the subsystem to lower the space heating requirement of the following subsystem. This is for instance done with the amount of thermal losses of auxiliary components, which are transferred to the heat transfer medium. Recoverable losses are taken into account at the interface to the calculation of the heating requirement according to EN ISO 13790 respectively to lower the initial heating requirement. For instance the recoverable envelope losses of domestic hot water components are treated in that way, as shown in the energy balance in Fig. 21.

As an option on every stage of the single subsystems, the subsystem performance based on primary energy can be calculated to compare different systems.

Similar calculation approaches for cooling and ventilation shall be developed by the working groups of CEN/TC 89 for cooling and CEN/TC 156 for ventilation.

2.3 North American standardisation

In the following the basic equations for North American standardisation in the field of heat pumps by the US organisations ASHRAE (American Society of Heating, Refrigeration and Air-Conditioning Engineers) and ARI (Air Conditioning and Refrigeration Institute) as well as the Canadian CSA (Canadian Standards Association) are given. More details can be found in the Interim Report of IEA HPP Annex 28 [2] or in the Canadian Country report on Task 1 [43].

Basic metrics used for the performance characterisation of heat pumps are the HSPF (Heating Seasonal Performance Factor) and the SEER (Seasonal Energy Efficiency Ratio) in heating and cooling mode respectively. Tab. 4 gives an overview of the existing standards in the field of performance calculation for heat pumps in North America.

Calculation of the single mode is performed by a bin method according to ASHRAE 116 which applies a power balance for the heating seasonal performance according to the equation

$$\text{HSPF} = \frac{\sum_{j=1}^{18} n_j \cdot \text{BL}(t_j) \cdot F_{\text{def}}}{\sum_{j=1}^{18} n_j \text{HLF}(t_j) \delta(t_j) \dot{E}(t_j) / \text{PLF}(t_j) + \sum_{j=1}^{18} \text{RH}(t_j)} \quad [-] \quad \text{eq. 4}$$

where

HSPF	heating seasonal performance factor	(-)
t_j	temperature of the respective bin j	(°F)
$\text{BL}(t_j)$	building load of the temperature bin j	(Btu/h/1000)
F_{def}	defrost factor (derived from testing according to ARI 210/240)	(-)
HLF	heating load factor	(-)
$\delta(t_j)$	factor [0..1] to consider the fraction of monoenergetic operation	(-)
$\dot{E}(t_j)$	electrical power input	(W)
PLF	part load factor	(-)
RH	electrical input by a resistance heater in monoenergetic operation	(W)
n_j	fractional bin hours of the respective bin	(-)

Tab. 4: Overview of the most important standards for the testing and calculation of heat pumps in North America [2]

North American standards			
Designation	Title	Application	Output values/Remarks
ARI 210/240-2003	Unitary air conditioning and air source heat pump equipment	Test procedure and Calculation method for electrically-driven A/A heat pumps space heating-only/ Cooling-only operation	Heating capacity and COP in the testing conditions [°C]: Heating: A-8.3, A 1.7, A 8.3 Cooling: A 35, A 27, A 19.4 <ul style="list-style-type: none"> SEER for cooling operation HSPF for heating operation
ANSI/ASHRAE 116-1995	Methods of testing for rating seasonal efficiency of unitary air conditioners and heat pumps	Test procedure and calculation method for electrically-driven air-to-air heat pumps space heating-only/ Cooling-only operation	Same testing conditions as ARI 210/240 <ul style="list-style-type: none"> SEER for cooling operation HSPF for heating operation
ASHRAE 118.2-2002	Method of Testing for Rating Residential Water Heaters	Test procedure and calculation method Testing involves: First hour rating Simulated use test	Calculation of first hour rating for Type I and Type II water heaters Recovery efficiency for thermostatically controlled water heaters <ul style="list-style-type: none"> Daily water heater energy consumption Daily hot water energy content Energy factor
ASHRAE 124-2002	Methods of Testing for Rating Combination Space Heating and Water Heating Appliances	Test procedure and calculation method	Test procedures and calculation methods for combined space and water heating without a heat pump
ASHRAE 137-2001	Methods for testing for Efficiency of Space-Conditioning/Water Heater Appliances that include a desuperheater water heater	Test procedure and calculation method	Testing according to ASHRAE 116 Test points combined operation: Cooling&DHW [°C]: A 35, A 27.8 Heating&DHW [°C]: A -8.3, A 8.3 <ul style="list-style-type: none"> Combined seasonal performance factor for heating and DHW CPF_{hs} Combined seasonal performance factor for cooling and DHW CPF_{cs}
ARI 320-1998	Water source heat pumps	Test procedure	<ul style="list-style-type: none"> Cooling capacity; EER Heating capacity; COP
ARI 325-1998	Ground water source heat pumps	Test procedure	<ul style="list-style-type: none"> High and low temperature Cooling capacity; EER High and low temperature Heating capacity; COP
ARI 330-98	Ground Source Closed-Loop Heat Pumps	Test procedure	<ul style="list-style-type: none"> Cooling capacity; EER Heating capacity; COP
ARI 470-2001	Desuperheater / water heaters	Test procedure	Heat exchanger characteristics; <ul style="list-style-type: none"> Net capacity, pressure drops, temperatures, mass flows
ARI 870-2001	Direct Geoexchange Heat Pumps	Test procedure	<ul style="list-style-type: none"> Cooling capacity; EER Heating capacity; COP

The numerator describes the building load at each temperature bin j with a correction factor for the defrost control, if installed. The denominator describes the respective total electrical power input to cover the building load for each temperature bin j , the first term the fraction, that is required by the heat pump operation, the second term the power input to an installed back-up resistance heater. The output capacity and the electricity input are determined by linear interpolation between the test points. The heating capacity of the heat pump is only needed to calculate the heating load factor HLF of the bin.

The heat pump low temperature cut-out factor $\delta(t_j)$ characterises the monoenergetic operation. In the temperature range between the low cut-out temperature for the heat pump and the balance point temperature, $\delta(t_j)$ is set to 0.5, i.e. the heat pump and the back-up each contribute one half of the energy demand, below the low temperature cut-out, $\delta(t_j)$ is 0 and for temperature above the balance point, $\delta(t_j)$ is 1.

For the calculation the heating load factor is defined according to the equation

$$\begin{aligned} \text{HLF} &= \frac{\text{BL}(t_j)}{\dot{q}(t_j)}, & \text{if } \dot{q}(t_j) > \text{BL}(t_j) \\ \text{HLF} &= 1, & \text{if } \dot{q}(t_j) \leq \text{BL}(t_j) \end{aligned} \quad [-]$$

eq. 5

where

HLF	heating load factor	(-)
$\text{BL}(t_j)$	building load in temperature bin j	(Btu/h)
$\dot{q}(t_j)$	heating capacity of the heat pump in temperature bin j	(Btu/h)

Combined heating or cooling operation and DHW production

ASHRAE 137 combines the method of ASHRAE 116 with the testing and calculation of a desuperheater. The method of ASHRAE 137 treats as well simultaneous operation of either heating and domestic hot water production or cooling and domestic hot water production.

The definition of the SPF for combined cooling and hot water, called CPF_{cs} (combined performance factor space cooling and DHW season), is given by eq. 6

$$\text{CPF}_{\text{cs}} = \frac{\sum_{j=1}^8 \left(\frac{q(t_j)}{N} + \frac{q_w(t_j)}{N} \right)}{3.143 \left[\frac{\text{Btu/h}}{W} \right] \cdot \sum_{j=1}^8 \left(\frac{E(t_j)}{N} + \frac{ER(t_j)}{N} \right)}$$

eq. 6

and respectively for the combined heating and hot water season

$$\text{CPF}_{\text{hs}} = \frac{\sum_{j=1}^8 \left(\frac{q(t_j)}{N} + \frac{q_w(t_j)}{N} \right)}{3.143 \left[\frac{\text{Btu/h}}{W} \right] \cdot \sum_{j=1}^8 \left(\frac{E(t_j)}{N} + \frac{ER(t_j)}{N} + \frac{RH(t_j)}{N} \right)}$$

eq. 7

where

CPF_{cs}	for the combined space-cooling and water-heating season, the sum of the total space-cooling load and the useful portion of the total water-heating load divided by the total electrical energy consumed by the combined appliance (heat pump, electric water heater and, if applicable,	(-)
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	desuperheater water pump)	
CPF_{hs}	for the combined space-heating and water-heating season, the sum of the total space-heating load and the useful portion of the total water-heating load divided by the total electrical energy consumed by the combined appliance (heat pump, electric water heater and, if applicable, desuperheater water pump).	(-)
$\frac{q(t_j)}{N}$	for temperature bin j, the ratio of the total space cooling (or heating) provided to the total number of temperature bin hours in the space-cooling (or space-heating) and water-heating season.	(Btu)
t_j	temperature in bin j	(°F)
$\frac{q_w(t_j)}{N}$	for temperature bin j, the ratio of the total thermal energy associated with domestic hot water that is delivered to the consumer to the total number of temperature bin hours (tank stand-by losses are not included in this quantity)	(Btu)
$\frac{E(t_j)}{N}$	for temperature bin j, the ratio of the total electrical energy supplied to the heat pump (or air conditioner) and, if applicable, to the desuperheater water pump to the total number of temperature bin hours	(Wh)
$\frac{ER(t_j)}{N}$	for temperature bin j, the ratio of the total electrical energy supplied to the electric water heater to the total number of temperature bin hours.	(Wh)
$\frac{RH(t_j)}{N}$	for temperature bin j, the ratio of the total electrical energy used for resistive space heating to the total number of temperature bin hours (resistive heating is required when operating below the space-heating balance point).	(Wh)
j	for each climatic region, the total number of 5°F outdoor temperature bins having a nonzero entry for the fractional bin hours (i.e., $n_j / N > 0$) are given in the standard.	(-)
N	total number of temperature bin hours in the space-cooling or space-heating and water-heating season	(-)

The distribution of the total heating requirement to the part produced in combined operation is done by an evaluation of the actual heating and hot water load in the bin. The lower of both loads determines the partition of combined operation. Space heating-only operation is calculated according to ASHRAE 116 as described above.

For domestic hot water-only operation the seasonal performance factor is defined as

$$CPF_{ws} = EF \quad [-] \quad \text{eq. 8}$$

where

CPF_{ws}	seasonal performance factor for the domestic hot water-only operation	(-)
EF	energy factor to be determined according to ANSI/ASHRAE 118.2 – 2002	(-)

2.4 Japanese standardisation

Tab. 5 gives an overview of Japanese standards for the testing and calculation of heat pumps.

Japanese DHW-testing is performed according to the revised version of JRA 4050:2004. Details of the testing are given in connection with the Japanese proposal for the calculation method described in chapter 3.2.4.1

Tab. 5: Overview of the most important standards for testing and calculation of heat pumps in Japan [2]

Japanese standards			
Designation	Title	Application	Output values/Remarks
JIS 8615-1	Non-ducted air-conditioners and heat pumps – Testing and rating for performance	Test procedure	Rating for air-to-air heat pumps. Almost identical with ISO 5151
JIS 8615-2	Ducted air-conditioners and heat pumps – Testing and rating for performance	Test procedure	Rating for air-to-air heat pumps. Almost identical with ISO 13253
JRA 4048	Calculation method of annual power consumption for packaged air conditioners	Calculation method	Simplified modification from ARI 210/240
JRA 4049	Calculation method of annual power consumption for multi-split air conditioners	Calculation method	Simplified modification from ARI 210/240
JRA 4050:2004	Heat Pump Water Heaters Using Carbon Dioxide Refrigerant – Testing and Performance Requirements	Test procedure and calculation method	Rating of HPWH using CO ₂ refrigerant

2.5 Conclusion of the results of Task 1

In conclusion, the following decisions were made:

2.5.1 Testing

- Testing of simultaneous combined operation shall be treated by extending the existing standards (e.g. for Europe EN 255-3 and EN 14511)
- All testing shall follow a black-box approach, i.e. only results, that could be measured from outside the system shall be required. The system boundary for the black box testing is depicted in Fig. 22.
- Highly integrated systems, which are sold as a unit and the heat pump and the domestic hot water storage tank are integrated, shall be tested as system. In case of the combination of different components to a combined operating system, component-based results shall be used.

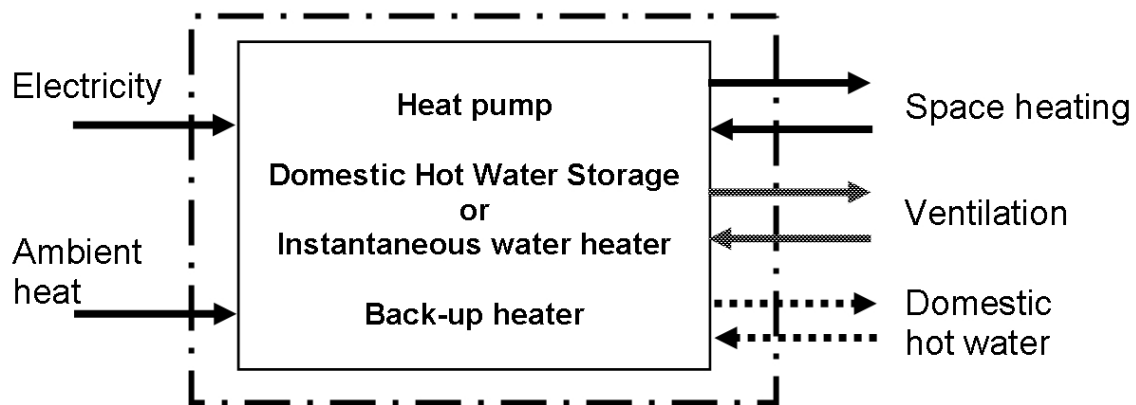


Fig. 22: System boundary for the black-box testing

2.5.2 Calculation

Calculation of the SPF shall be based on the bin-methodology (see description in chapter 3.2.1), which is already contained in several national European calculation methods and the American standardisation. The bin method seems the best compromise between an easy-to-use hand calculation method and the exactness of the results.

2.5.3 System boundary

The system boundary for the IEA HPP Annex 28 comprises the generator with attached storages, i.e. the heat pump unit, eventually installed back-up heaters and storages for domestic hot water or the heating system. This corresponds to the system boundary generator and storage in the EPBD calculation scheme as depicted in Fig. 23.

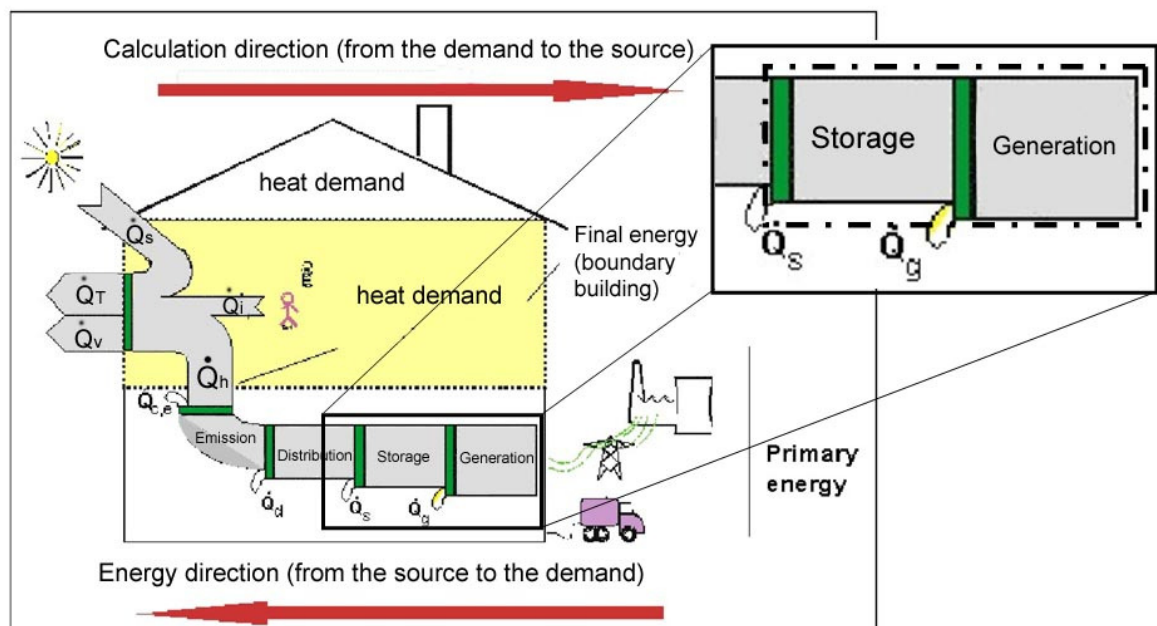


Fig. 23: System boundary for the systems covered by IEA HPP Annex 28 in the calculation scheme of the prEN 15316-1 (formerly prEN14335)

3 RESULTS TASK 2 AND TASK 3

3.1 Evaluation of actual and proposed European Standards

In this chapter, results of the evaluation of the existing and proposed European testing standards described in chapter 2.2.1 are presented. Proposed changes in conclusion of the evaluation are discussed in chapter 3.5 of the development of the test procedure.

Moreover, results from part load testing of an air-to-water heat pump are presented.

3.1.1 EN 14511 vs. EN 255-2

In 2004 the former standard EN 255-2 [10] for the testing of heat pumps in space heating-only mode has been replaced by EN 14511 [11], introducing some changes described in chapter 2.2.2.

Comparison of the two standards delivered the results that the COP-values of EN 14511 are in the same range as EN 255, but a bit lower due to the changes in the setting of the mass flow rate. However, EN 255-2 sometimes leads to unrealistic temperature conditions at the condenser side, so EN 14511 delivers more realistic values than EN 255-2 [17].

For different heat emission systems, e.g. floor heating or radiator heating, a different mass flow rate is determined according to EN 14511. Nevertheless, if the different mass flow rates are used at the same temperature conditions, the changes in the COP are very small. Accordingly, the standard could be simplified by using the same mass flow rate for different heat emission systems. Fig. 24 summarises the measured results of the comparison.

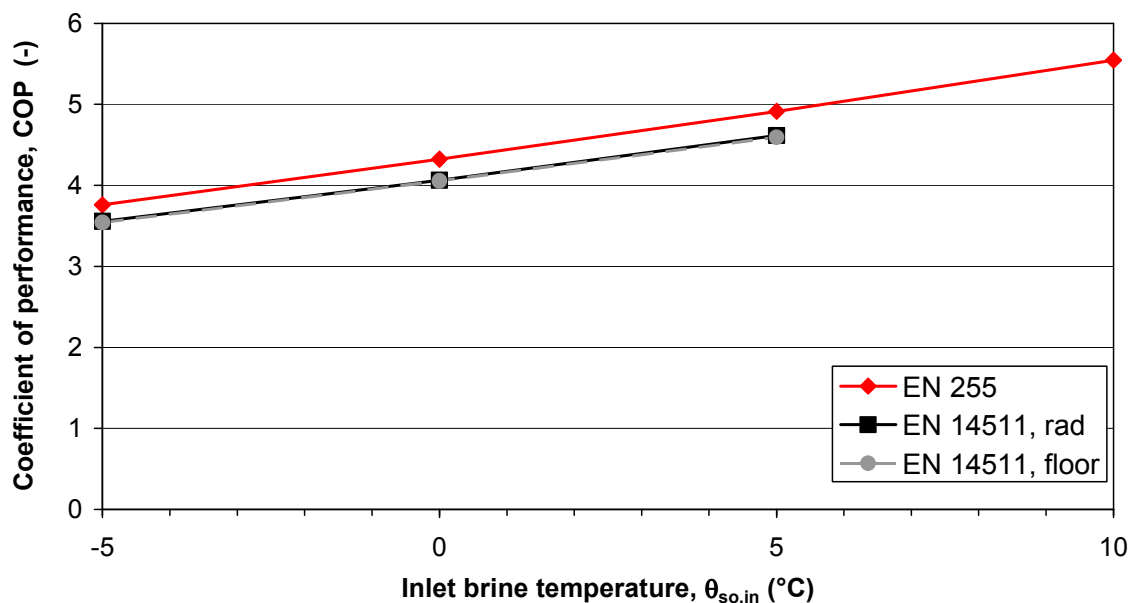


Fig. 24: Comparison of measured COPs according to EN 255-2 and EN 14511(source [17])

3.1.2 EN 255-3

EN 255-3 [6] is the actual standard for testing of heat pumps in DHW-only mode. However, it will soon be revised in connection with Mandate M/324 [5].

On the basis of the analysis of EN 255-3, in particular with regard to using this standard for combined operating systems, the following problems have been identified by the IEA HPP Annex 28 to be considered during a revision of the EN 255-3 in CEN/TC 113/WG 10. Proposals for modifications are given in the description of the test procedure in chapter 3.5.

- Storage stand-by losses

Determination of the storage stand-by losses is one of the crucial points concerning the standard, since the main problem refers to the long testing time which is mainly caused by this determination. The electricity input to cover storage stand-by losses are not absolutely necessary for the COP_t determination, since the change in COP_t is about 5% depending on the heat pump and thereby in the range of the measurement exactness [13]. A calculation based on the measurement results of the Swiss national project for a B/W heat pump [9] delivered a change in COP_t of only 2%, the measured Swedish heat pump, however, resulted in a change of 6.5%.

- Heating capacity in DHW mode

EN 255-3 does not deliver an heating capacity for the DHW operation, which is a necessary parameter, when calculating the running time for the determination of the fraction of combined operation [9].

- Domestic hot water tapping temperature

EN 255-3 does not fix an outlet temperature for the domestic hot water. Thus, measured values cannot be used directly to compare the performance using the COP-values. Therefore, it is to be discussed, if a revision should define a fixed outlet temperature for the domestic hot water, e.g. 60°C or introduce another characteristic number in order to have comparable values. On the other hand, comparison could be made by a seasonal performance calculation, so an extension is not absolutely necessary.

- Limited information on the heat pump characteristic

Due to the long testing time only one value for the COP_t is delivered depending on the type of the heat pump. Thus, an interpolation of the heat pump characteristic in the direction of changing source temperatures is not possible. However, the procedure to evaluate the COP_t itself does not take too much time, so it is to be discussed, if testing is to be extended to cover more test points, and if testing time of storage stand-by losses can be reduced (see first item of the list).

3.1.3 European tapping profiles defined in mandate M/324

In Mandate M/324 [5] five so called "EU reference cycles" for the testing of DHW appliances are defined to enable a uniform testing of DHW systems, see chapter 2.2.4.

An evaluation in comparison to EN 255-3 is given in the following.

- COP-values delivered by the tapping profile according to mandate M/324 tend to be lower. There could be different reasons for that: On the one hand differences are caused by different amounts of tapped water [13]. However, performing the test with EU reference cycle N° 3 where the energy content of the DHW tapplings is kept constant and only the number of tapplings is changed from 23 to 3, the EN 255-3 COP_t is about 20% higher [17]. Therefore, differences are likely to be caused by different conditions inside the storage (e.g. stratification, convection flows) that are caused by the different profiles. In the case of numerous small tapplings, the inlet temperature to the condenser (and thereby also the mean condensation temperature) tends to be higher [17], and consequently the COP_t is lower. Fig. 25 depicts the statements above.

In addition, there might be differences regarding the energy content/average temperature of the storage tank at start and stop of the evaluation period for the two cycles.

The same effect can be observed comparing a COP for the extraction of the domestic hot water (COP_t) and the COP during stand-by operation in the normal EN 255-3 cycle. The latter for reheating the storage stand-by losses is not determined in EN 255-3. Depending on the controller setting, the inlet temperature during stand-by operation is much higher than the inlet temperature after a tapping of half the storage volume, resulting in different mean temperatures in the course of the heating process and thereby leading to significantly lower COP-values for the reheating of the storage losses [9]. Actually, this is the reason why the electrical stand-by power input cannot be used to calculate directly the thermal storage losses and vice versa, since the COP is not known. In particular, it does not correspond to the COP_t .

- Despite partly small volumes (EU reference cycle N° 1), the reproducibility of COP-values is in the acceptable range of 5% [17], even though it may be problematic to secure the exactness of these small volumes [13].

The result of the comparison are presented in Fig. 25

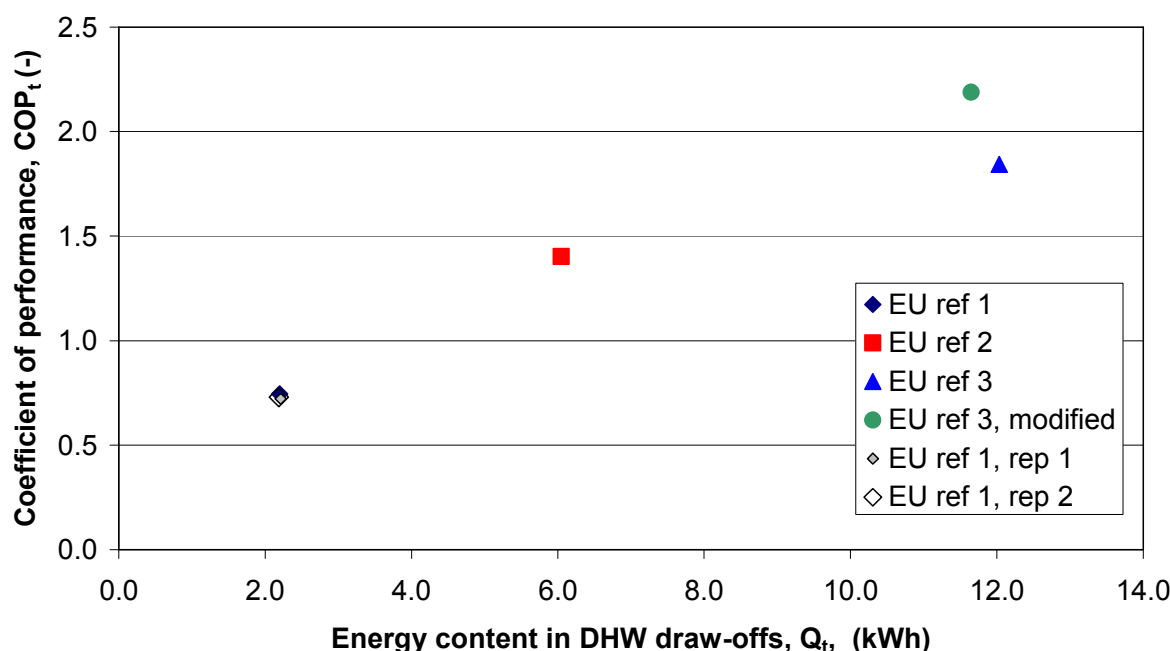


Fig. 25: Comparison of results for the COP_t according to EN 255-3 and M/324 (source [17])

Therefore, it is to be discussed, if measurements with tapping patterns containing 23 draw-offs are necessary and useful. While one opinion is that the tapping profiles will deliver more realistic values [13], the other opinion is that user behaviour varies a lot in different houses and in different countries anyway, so probably a simplification could be applied. However, if not a "profile" but "only cyclic tapping" is applied, the volume of the extracted water shall be reduced to e.g. one quarter of the storage volume, which seems to be a more realistic value [17]. Cyclic tappings have another big advantage: It is very important to secure that the storage has the same energy content in the beginning and in the end of the test. By cyclic tapping this can be assured, while this could be problematic with a profile without some conditioning time afterwards [17].

3.1.4 Part load operation

In France extensive testing of a heat pump in heating and cooling mode was carried out to quantify cyclic effects caused by part load operation. An air-to-water heat pump with a capacity of 9 kW was used in the investigation. The test rig is shown in Fig. 8 in chapter 1.4.3.4. The testing was carried out at EDF in the framework of two doctoral theses. The following results are taken from the presentation of the Purdue conference [18], [19].

Tests were carried out varying the capacity of the distribution system in the steps 11 l/kW, 21 l/kW and 31 l/kW as well as the setting of the return temperature hysteresis control in the steps $\Delta\theta = 1K$ and 3K. The results for the cooling mode of the heat pump shows that both the capacity and the setting of the hysteresis do only have a negligible impact on the part load performance of the heat pump. Results are depicted in Fig. 26.

Taking into account these results for the cooling mode, the presentation of the heating mode only shows the results at fixed inertia and fixed setting of the hysteresis controller. Declination of the heat pump efficiency is due to increasing impact of stand-by power consumption and expenses for the defrosting with decreasing heat load. In case of the tested heat pump, defrost control was only time-dependent, so defrosting did not take into account operating time of the heat pump. Correcting for these two impacts, the COP -values did not decline over the whole range of loads. So, in heating mode, the cycling effects are negligible, as presented in Fig. 27.

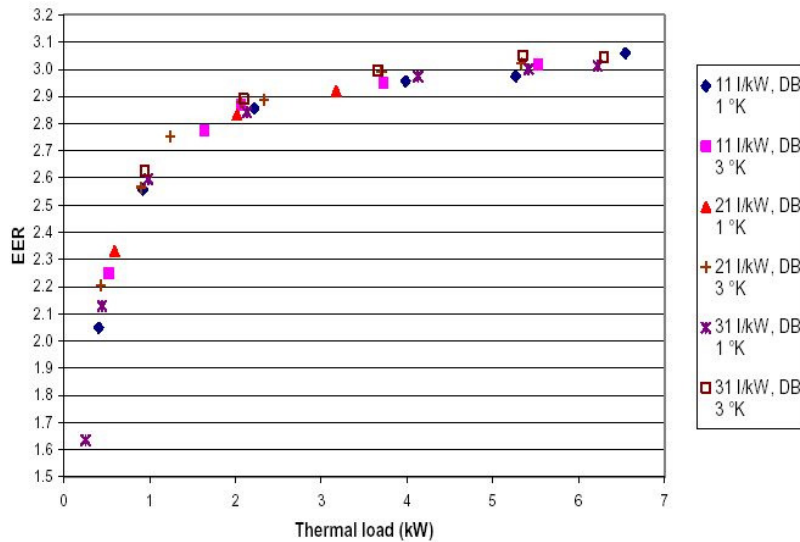


Fig. 26: Results of part load testing in cooling mode with different inertia and setting of the hysteresis return temperature controller (source [18], [19])

In conclusion the calculation method shall take into account:

- The stand-by power consumption in the OFF-state
This has already been implemented in the actual proposal of prEN15316 part 4.2 [16])
- The defrost control of the heat pump
However, it is difficult to take into account defrosting effects, if there is no information from the manufacturer or from testing on the control of the defrosting system. In consequence, if defrost losses have a significant effect on losses in the stand-by operation, testing to quantify this losses has to be provided.

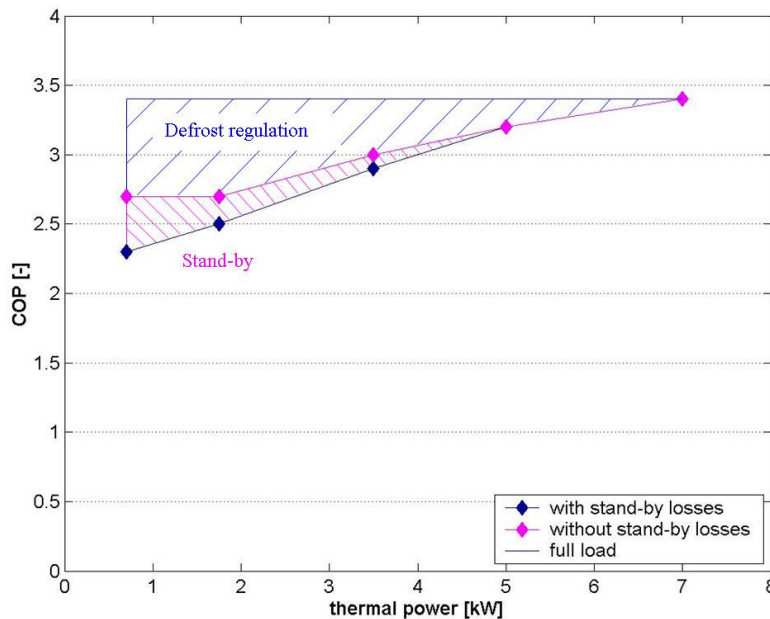


Fig. 27: Results for the heating mode divided into energy consumption for stand-by and defrosting (modified from source [18], [19])

However, there is possibly another effect of part load operation, which is not taken into account. Due to the cyclic control of the heat pump in comparison to the variable speed compressor, the heating water temperature may be higher to cover the space heating demand. Presently, a correction for this effect is not made.

3.2 Proposal for calculation method

According to the legal text [7] the objectives of the calculation method to be developed in the framework of this Annex are to be transparent, as far as possible easy-to-use, executable without extensive computation and capable of delivering a Seasonal Performance Factor (SPF) for space heating-only and DHW-only and combined operating heat pump systems. The results will in turn serve as a basis for a comparison of the heat pump systems among each other as well as comparison with other types of heating systems. Thereby, advantages of systems layout, e.g. in terms of primary energy consumption or carbon dioxide emission for defined boundary conditions shall be promoted.

Since the testing is accomplished to deliver the needed characteristic performance data (basically heating capacity and COP) to carry out the calculation, testing has to be coherent to the calculation method. Therefore, in this paragraph, the calculation approaches are described first, and the proposal for the respective test procedure follows in the next chapter.

The participants of the IEA HPP Annex 28 agreed on a calculation method based on a bin method. Therefore, the bin method is described in the following chapter, as it has been worked out in the Swiss national project [14] as basis for an extension for combined operation. Afterwards, the discussion of different proposals of the national projects to extend the method for the combined operation is summarised and the decision taken on the 4th working meeting are outlined. The bin approach of the IEA HPP Annex 28 [14] has been implemented as a step-by-step calculation method in the European draft standard prEN15316 (formerly prEN 14335 [16]), which was sent out to public enquiry in the framework of the European standardisation of the EPBD [1] (see description in chapter 1.1.2.1). The updated version with comments from the Annex participants and with regard to the system boundary chosen in IEA HPP Annex 28 is given in part 1 chapter A of this report. Amendments following these final results of the IEA HPP Annex 28 can be integrated after the public inquiry of the CEN draft standard. In this chapter the backgrounds and the principles of the calculation approaches are described. Comments received recently which may require further discussions or investigations are described in chapter 3.4.

3.2.1 Principle of the bin method

The bin method is based on the cumulative annual frequency of the outside temperature which is one major impact on the space heating energy requirement. A typical cumulative annual frequency is depicted in Fig. 28 in the mathematical layout of cumulative frequencies.

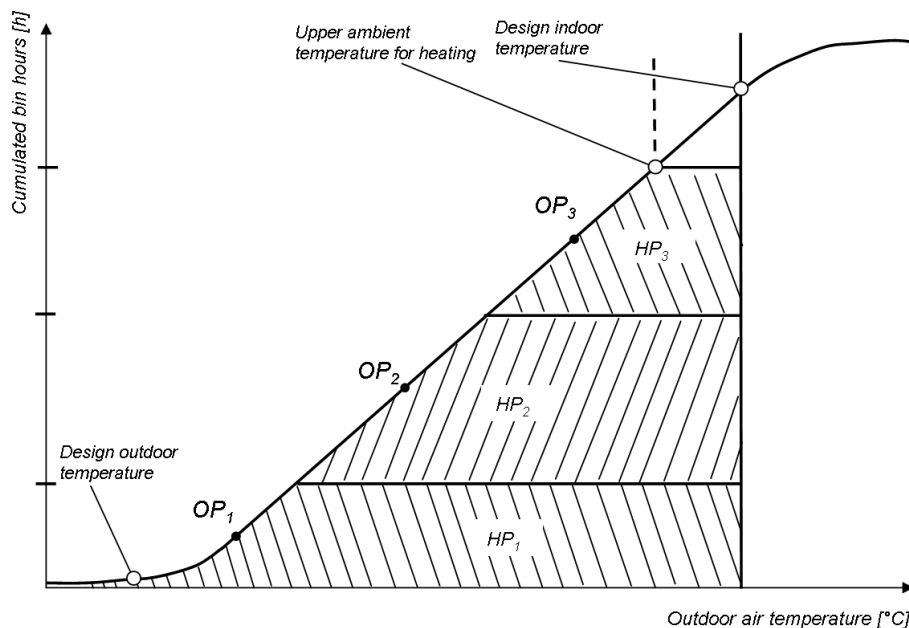


Fig. 28: Principle of bin-method on the basis of cumulative annual frequency of the site

Duration curves used in heating technologies are sometimes displayed in a 90° clockwise rotation. For the creation of the annual frequency based on hourly values of the outside temperature, details are given in the report of the Swiss national project [14].

The cumulative annual frequency of the outdoor dry-bulb temperature is divided in temperature classes called bins. The method assumes that operating conditions at the operating point are valid for the whole bin, i.e. the operating conditions characterise the mean operating conditions of the bin and therefore are set in the centre of the bin. The method does not prescribe the number of bins, however, three to five operating points are usually chosen for the space-heating operation. The number should be related to the available number of test points (see chapter 3.2.3).

The determination of the energetic fraction is done by the evaluation of the cumulative heating degree hours, derived from the hourly annual frequency of the ambient dry-bulb air temperature. They are dependent on the site, the design indoor temperature and the upper temperature limit for heating. The weighting factor is characterised by the energy requirement of the respective bin related to the total energy requirement, which can be derived by the heating degree hours according to the equation

$$w_{h,i} = \frac{Q_{out,g,h,i}}{Q_{out,g,h}} = \frac{HDH_{\theta,upper,i} - HDH_{\theta,lower,i}}{HDH_t} \quad [-] \quad \text{eq. 9}$$

where:

$w_{h,i}$	weighting factor for space heating operation	(-)
$Q_{out,g,h,i}$	space heating energy requirement in bin i	(J)
$Q_{out,g,h}$	total space heating energy requirement	(J)
HDH_t	total heating degree hours up to the upper temperature limit for heating	(Kh)
$HDH_{\theta,upper,i}$	heating degree hours to the upper temperature limit of the bin i	(Kh)
$HDH_{\theta,lower,i}$	heating degree hours to the lower temperature limit of the bin i	(Kh)

The performance factor of the respective bin can be derived by the equation

$$PF_i = \frac{Q_{out,g,h,i}}{\frac{(Q_{out,g,h,i} + Q_{l,s,h,i})}{COP_i} + W_{g,i}} \quad [-] \quad \text{eq. 10}$$

where

PF_i	performance factor	(-)
$Q_{out,g,h,i}$	space heating requirement in bin i	(J)
$Q_{l,s,h,i}$	storage losses of installed heating buffer storages	(J)
COP_i	coefficient of performance at operating point in bin i taken as performance factor for the bin	(W/W)
$W_{g,i}$	additional auxiliary energy input not considered in standard testing, e.g. for source pumping	(J)

The numerator in eq. 10 describes the thermal requirement in the bin and the denominator describes the required electrical energy input to produce this energy taking into account the following energies for every bin

- building energy requirement
- energy losses by storage(s)
- additional auxiliary energies for the system operation, which are not included in the standard COP values, e.g. for Europe according to EN 14511 or EN 255-3 respectively, e.g. for parts of the source and sink pumping and stand-by energy during OFF-periods

Energy losses by cyclic operation of the heat pump considered negligible in the frame of this calculation (see more details on French investigation in chapter 3.1.4 and chapter 3.2.2.5)

After the calculation of each bin a weighting with the weighting factors and summation of the single bins are performed to receive the seasonal performance factor of the heat pump according to the equation

$$SPF_{g,h,hp} = \frac{Q_{out,g,h}}{\sum_i \frac{Q_{out,g,h,i}}{PF_i}} = \frac{1}{\frac{Q_{out,g,h,i}/Q_{out,g,h}}{PF_i}} = \frac{1}{\sum_i \frac{w_{h,i}}{PF_i}} \quad [-] \quad \text{eq. 11}$$

where

$SPF_{g,h,hp}$	seasonal performance factor of the heat pump for space heating-only mode	(-)
$Q_{out,g,h}$	total space heating energy requirement	(J)
$Q_{out,g,h,i}$	space heating requirement in bin i	(J)
$w_{h,i}$	weighting factor for space heating operation	(-)
PF_i	performance factor in bin i	(-)

For monoenergetic or bivalent heat pump systems, a further weighting with energy fractions for heat pump and back-up operation is done to calculate the generation subsystem seasonal performance factor for the space heating operation mode $SPF_{g,h}$

$$SPF_{g,h} = \frac{Q_{out,g,h}}{\frac{Q_{out,g,h} \cdot (1 - p_{bu,h})}{SPF_{g,h,hp}} + \frac{Q_{out,g,h} \cdot p_{bu,h}}{\eta_{bu,h}}} \quad [-] \quad \text{eq. 12}$$

where

$SPF_{g,h}$	seasonal performance factor for space heating-only mode for bivalent systems	(-)
$Q_{out,g,h}$	total space heating energy requirement	(J)
$p_{bu,h}$	fraction of the produced energy for space heating covered by the back-up heater	(-)
$SPF_{g,h,hp}$	seasonal performance factor of the heat pump for space heating operation	(-)
$\eta_{bu,h}$	efficiency of the back-up generator in space heating operation	(-)

A similar calculation can be performed for the domestic hot water part of the system based on the testing results, e.g for Europe EN 255-3, see part 1 chapter A1. For heat pump water heaters only this hot water calculation is performed. The result of the hot water calculation is the seasonal performance factor for domestic hot water SPF_{DHW} . The two seasonal performance factors are combined by the energetic fractions in the same way as shown in eq. 12.

In the proposal given in part 1 chapter A 1, this calculation principle is adapted to the requirements of the EPBD calculation scheme described in chapter 2.2.6. Since the output in European standardisation is not an SPF but the energy input to the system to cover the space heating/DHW requirement (here electricity), the denominator of the above given seasonal performance factor is evaluated. Calculation of the single energies is also adapted to the standard to be consistent with standards for other heat generators, e. g. boilers.

3.2.2 Input data for the calculation method, interfaces to existing standardisation

Input values are required to evaluate the single terms of eq. 10. In the following the sources of input data and the interfaces to other standards are discussed.

3.2.2.1 Energy requirement

Heat requirements of the systems for space heating can basically be evaluated by the design heat load delivered by EN 12831 [25] or the annual energy requirement delivered by EN ISO 13790 [4]. This standard is currently revised in the framework of the EPBD. Since EN ISO 13790 is the basic standard in the framework of the EPBD, it is the basis for the input values used, namely the requirement of the distribution system (see description in chapter 2.2.6). Moreover, normally the energy value is the more realistic compared to the design heat load based on EN 12831, since internal and solar gains are taken into account, while the design heat load is actually a design value based on worst case conditions, which delivers higher annual energy values without correction with used gains. Following the framework of the EPBD, the space heating energy requirement would be the requirement distribution subsystem depicted in Fig. 21. The domestic hot water requirement will also be defined in the EPBD framework prEN15316 (former prEN14335).

Due to national regulations, the energy requirement may be evaluated differently e.g. by performing a power balance taking into account the transmission and ventilation losses as well as the internal gains. In Sweden, for instance, the heat load can be calculated by using the design outdoor temperature, the supply and return temperature at outdoor design conditions and annual use of energy and duration curve for the outdoor temperature as input data.

3.2.2.2 Meteorological data of the site and source temperature

The meteorological data needed for the calculation is basically the outdoor dry bulb air temperature, as far as possible in hourly resolution, which is the basis for cumulative annual frequency of the site. This data has to be provided in a national Annex of the standard. If national data are not available, the Software Meteororm [22] can be used, which contains world wide data generated on the basis of monthly values with statistical calculation approaches and contain data of meteorological stations, as well.

For **outside air-to-water heat pumps**, the outside air temperature directly corresponds to the source temperature and can be used to define the temperature conditions of the operating point. A wet bulb temperature can be coupled to the outside air conditions as done in EN 14511 [11], or duration curves can be used.

For **exhaust air-to-water heat pumps** without heat recovery, the indoor temperature, i.e. the source temperature for the heat pump is constant. However, the wet bulb temperature depends on the outside temperature and can be given as a correlation or can be calculated based on the humidity of the outdoor air, the ventilation flow rate and the moistening of the indoor air (from humans and various activities). Moisturing rates in the range of 3g/m^3 are given in [46]. For exhaust-A/W heat pumps in particular in combination with a heat recovery as used in compact units, additional testing and calculation are required to evaluate the temperature conditions for the heat pump, see chapter 3.2.6.

For **brine-to-water heat pumps** the characteristics of the ground have to be considered. Two possibilities have been discussed in Austria. On the one hand, a relation between the ground temperature and the outside air temperature and on the other hand a time based approach. Most of the participants of IEA HPP Annex 28 committed to B/W heat pump are in favour of a given relation between the ground temperature and the outside air temperature. Profiles based on national conditions are included in the Swiss [14] and French country reports. [13]. Approaches for the evaluation in Sweden are given in [17]. The data shall be given in a national Annex.

For **water-to-water heat pumps** the source temperature is usually set constant in the case of ground water. In the case of surface water, the conditions of the site have to be known and shall be given in a national Annex.

An interesting concept is implemented in Germany in the framework of DIN V 18599 [30], the national German implementation of the EPBD. All meteorological boundary conditions refer to one standard

site in the country, since the objective of the standard is the comparison of different heating system solutions and not the design of the system. Thereby, regional differences are eliminated. This concept seems very reasonable and is recommended by the IEA HPP Annex 28.

Proposal:

For standard calculations the meteorological boundary conditions shall be given for one standard site of each country in national Annexes. However, the method is not restricted by that.

3.2.2.3 Supply temperature of the space heating system

The supply temperature of the heating system is given by the controller setting e.g. in form of the heating characteristic curve, and can thus be related to the outside air temperature of the bin. Prescriptions of the heating characteristic curves are given for different types of heating systems in national standards and shall thus be given in national Annexes.

3.2.2.4 COP and heating capacity characteristic of the heat pump

The basic input values of the calculation to characterise the heat pump are the heating capacity and the COP of the heat pump. These values are delivered by standard testing, which is described in chapter 2.2 - 2.4. In fact, one task of the IEA HPP Annex 28 was the evaluation and improvement of present testing standard in coherence with the values needed by the calculation and the extension to cover combined operating heat pump systems.

Therefore, it is essential to restrict the calculation to the use of the standard values (see comment in chapter 3.4.3 and 3.4.4).

However, testing can only deliver COP and heating capacity data at discrete points of the whole operation range, namely the fixed test points. In case of little information of testing, e.g. for DHW testing in Europe according to EN 255-3, a correction of the COP has been done for changing operation conditions, mainly the temperatures. One approach is the approximation by a constant exergetic efficiency around the test points as described in [38].

In order to reduce testing expenses, it was decided in Annex 28, that missing values of the DHW characteristic shall be derived by interpolating the heating characteristic, see chapter 3.5.4.5.

3.2.2.5 Part load operation

Currently, the calculation method refers to results from testing according to CEN/TS 14825 [12]. However, test results of this technical specification are seldomly available. In this case default values derived by field measurements of an outside A/W heat pump by EDF, which have been analysed by J. Bernier [29], are given. However, results of the investigation made by EDF in the framework of IEA HPP Annex 28 do not approve this evaluation of J. Bernier so the values should be revised. Results of the French project described in chapter 3.1.4 delivered the basic result that the impact of cyclic operation of the heat pump is negligible. Thus, testing of part load operation for fixed step units is not necessary, if the stand-by power is determined, since this can separately be taken into account. Thus, the method given in part 1 has been revised with regard to part-load operation, since stand-by power input is already included. Nevertheless, possible effects on the supply temperature and thereby on the COP as stated in chapter 3.1.4 are not taken into account by this approach.

3.2.2.6 Parameters of attached storage systems

In case of installed heating buffer storage systems, parameters of the storage system concerning heat losses and temperature requirements have to be taken into account. Thus, the heat loss, i.e. (U·A)-value, and the temperature level of the storage have to be given.

In case of domestic hot water storages the stand-by power input to cover the storage losses is evaluated in the European standard EN 255-3. However, this standard delivers only the electrical power and not the thermal loss, see chapter 2.2.3 and 3.1.2. In the EPBD calculation scheme, the thermal losses are needed to evaluate the recoverable heat losses of the system, see chapter 2.2.6 for more details.

Nevertheless, it has been decided in IEA HPP Annex 28 to evaluate to the heating characteristic by means of the temperature settings of the DHW storage control to evaluate the COP, see chapter 3.5.4.5. Therefore, using interpolated values of the space heating characteristic the resulting COP can deliver the required thermal losses of the storage.

3.2.2.7 Nominal power of auxiliary systems

According to the system boundary given in chapter 2.5.3, the COP according to the European testing EN 14511 or EN 255-3 respectively does not include all auxiliary energies. Thus, additional auxiliary energy can be evaluated by the running time of the heat pump, or, in the case that the auxiliary component runs through, by the time of the bin.

3.2.3 Number and definition of bins

According to the different proposals for the combined operation, see next paragraph, described in the next chapter, the number and definitions of bins differs, as well.

Definition of bins according to the Japanese approach, which is based on seasons and typical days, is described in chapter 3.2.4.1.

The Swedish approach fixes the number of bins to five for the heating season and one for the DHW-only operation (summer period). Definition of bins shall be performed in the way, that the energy requirement in each bin represented by the area (see Fig. 28) is the same. Then, weighting of all bins is the same, but test points are no longer in the middle of the bins.

According to the Swiss approach, the position of the operating points shall correspond to the position of the test points. Number is not fixed, however, is defined by the test points. Bin limits are in the middle of the test points. For the summer operation one test points is proposed.

Proposal:

It seems reasonable to use as far as possible the values from the testing without interpolation of the source temperature. Therefore, a definition of the bins based on the test points and not on the energy requirement in the bin is recommended.

3.2.4 Approaches for the extension to include combined operation

Based on these principles of the described calculation method, different approaches to include the combined operation are shortly presented in this chapter to reflect the discussion and the decision of the IEA HPP Annex 28. The chosen approach is then described in more detail.

3.2.4.1 Evaluation of typical days for the bin (Japan)

This approach has been proposed by the Japanese National team. Details are contained in the Japanese country report on Task 2 and Task 3 [21].

The division of the total operation time is not made according to the outdoor air temperature, but ambient conditions are evaluated with regard to fixed seasonal classes, i.e. winter conditions, summer conditions, and intermediate. For each season typical days concerning temperature conditions as well as demand profiles are fixed, which correspond to the testing of the unit. By evaluating the sum of such typical days for the whole year, the seasonal performance is calculated.

Abstracting the method could be interpreted as bin approach, where higher ranking bins are chosen according to the season based on the heating load days. In the bins, a finer division is made by outside temperature to classes called warm day, average day and cold day, resulting in 5 bins altogether.

Tab. 6 gives the number of days corresponding to each season and the operation results of the system for Tokyo climatic conditions.

Summer is characterised by a DHW-only operation, intermediate season is characterised by DHW-only and combined DHW and space heating operation (warm day) and winter is characterised by combined DHW and space heating operation, divided into specific days (cold day and average day).

To reduce the testing expenses, a comparison has been made of three typical days (warm, average and cold) and only one average day. A comparison of the calculation result of these two approaches

Tab. 6: Determination of bins for Tokyo climatic conditions [21]

Season	Mode	Type of day	Number of days	Heat produced	Electric energy input to HP	Heat load	Total electric energy input
				MJ/day	kWh/day	MJ/day	kWh/day
Summer	WH	-	92	39.81	2.33	33.04	2.64
Intermediate	WH	-	123	54.83	3.61	45.66	3.93
	WH & FH	Warm day	30	85.73	8.39	56.29	9.04
Winter	WH & FH	Average day	108	115.81	11.60	80.49	12.49
	WH & FH	Cold day	12	145.70	17.07	98.92	19.31

delivered differences in the range of $\pm 5\%$, which seems good in relation to the reduced testing expense, so the 1-day method is preferred.

Summarising, the method facilitates the above given bin approach in two ways:

- by evaluating only two operation modes: DHW-only and combined SH/DHW, i.e. no single SH operation is tested anymore in combined operating systems, space heating is only taken into account in combination with the operation of the DHW system and occurs only dependent on the testing profiles.
- by reducing the number of bins by regarding only seasons and typical days with profiles for both space heating and domestic hot water operation: In a "1-day method", three bins are evaluated, a winter season with combined SH /DHW and an intermediate and summer season with DHW-only. In a "3-day method", the seasons are subdivided according to typical day cold, average and warm. A comparison shows that at the given boundary conditions of the calculation the "1 day method" is sufficient and delivers similar results to a "3-day method".

Respective characteristics of the heat pump system are evaluated by testing with both SH and DHW profiles for fixed temperature conditions. Details are given in the proposals for the test procedure in chapter 3.5.2.1

3.2.4.2 Evaluation of combined operation with regard to the ratio $Q_{out,g,h}/Q_{out,g,DHW}$

This approach for the calculation has been presented by the Swedish team in connection with the proposal for the test procedure described in chapter 3.5.2.2. Details are treated in the Swedish country report [17]. The main difference in the calculation to the above described method is the treatment of the operation modes.

In the Swedish approach the running time in different operation modes is no longer separated, but the heat pump characteristic is interpolated with regard to the ratio of $Q_{out,g,h}/Q_{out,g,DHW}$ in line with a respective evaluation of the test results based on the standards EN 14511 and EN 255-3.

Similar to the Japanese calculation approach, the space heating operation is no longer separated in single SH and combined SH and DHW, but the heating period is treated as period, where combined operation is possible, while in summer operation, exclusively DHW-only operation takes place.

However, the space heating-only operation in wintertime is taken into account by the ratio of the requirements $X = Q_{out,g,h}/Q_{out,g,DHW}$. Hence, in summertime, the ratio approaches zero (DHW-only operation and $Q_{out,g,h} = 0$), whereas in wintertime, depending on the boundary condition of the calculated system (e.g. type of building, characteristics of the DHW storage etc.), the ratio is in the range of $X = 2$ for ultra-low energy houses and about $X = 10$ for older buildings.

Consequently, for the calculation neither a splitting-up of space heating and DHW operation nor an evaluation of the running time is necessary [17]. However, testing has to deliver enough information on the heat pump characteristic in order to calculate the efficiency values and heating capacity values for the respective ratios X .

To perform the calculation, the interpolation of the heat pump characteristic is done in the following order:

- with regard to the ratio $Q_{out,g,h}/Q_{out,g,DHW}$
- with regard to the sink temperature
- with regard to the source temperature

The seasonal performance factor is calculated by a summation over all bins, as described in the above method.

3.2.4.3 Separation of operation modes for SH and DHW (Switzerland, France)

In this approach based on the method worked out by Switzerland [14], the single operation modes are treated independently of each other, i.e. the space heating-only operation is characterised by the test results for the space heating mode and the DHW-only operation is characterised by the heat pump characteristic in DHW mode. The heat pump characteristic is evaluated for the combined operation independent of the ratio $Q_{out,g,h}/Q_{out,g,DHW}$. The fraction of combined operation is evaluated by the running time in each operation mode, which is deduced from the heating capacity of the heat pump in the respective operation mode.

Excursus: Allocation of electricity input

Referring to a black box approach described in chapter 2.5.1 in the case of combined operation, an essential problem occurs for the evaluation: only the total consumption of electricity can be measured, but this total electricity input can hardly be allocated to the single functions space heating and domestic hot water [9]. Two possibilities arise to deal with this situation:

- Assumptions can be made to allocate the total electricity consumption which do not necessarily reflect the real physical conditions, but are reasonable to derive key figures for a comparison of different systems.
Two approaches were made to treat the simultaneous operation in two operation modes, one characteristic for combined SH and one characteristic for combined DHW:
 - In systems with desuperheater the space heating system gets only a part of the total heating capacity of the heat pump (only the fraction decoupled in the condenser), so the state of the space heating energy is approximated by the part load COP at this values. The part load efficiency is derived by testing with 50% load according to CEN/TS 14825 [12]. With this part load COP-value, the fraction of electricity input to the space heating system is calculated and the rest is accounted to the domestic hot water system [13].
However, what seems to be problematic is, that the heat pump is not in a part load operation as in the cyclic test according to CEN/TS 14825 with 50% ON and 50% OFF state of the compressor, so effects measured in the cyclic test may not characterise the combined operation at full load of space heating and domestic hot water. So, this is a "practical" approach to perform the allocation of electricity input to space heating and DHW operation.
 - The second approach is for systems using condensate subcooling and intends to have a useful and fair comparison to alternate combined operation. In alternate combined operation, the space heating operation is not influenced by the DHW operation, since the heat pump is switched and consequently both operation modes are independent. So, for the evaluation the COP of the space heating-only mode is kept constant, i.e. all the gains or losses in the energy consumption are accounted to the DHW part. This might not be the physical reality, but it enables an absolutely fair comparison with the alternate combined operating system. [9]
However, this approach is dependent on the system configuration, since in systems with desuperheating, COP in combined mode will change in comparison to the space heating-only operation, while in systems with condensate subcooling this is hardly the case.
- The key figure for the comparison is the overall SPF. Thus, the combined operation can be treated as a separate operation mode, and for the calculation, three different operation modes (space heating-only, DHW-only, combined heating-DHW) instead of two are considered.

The latter solution to treat the combined operation as a separate operation mode is the simpler and the more pragmatic one and thus may have advantages with regard to an easy-to-use calculation method. It is the basis of the following calculation approach for simultaneous operation. However, by doing so, the fraction of combined operation has to be known to perform an adequate weighting.

Decision:

Allocation of the total electricity input to the operation modes space heating / domestic hot water is not accomplished, but electricity input is accounted to the three operation modes space heating-only, DHW-only and combined SH/DHW

3.2.5 Discussion of the different approaches

In the following the basic pros and cons of the different approaches are summarised:

The Swedish and the Japanese approach use a simpler calculation, however, that may require a more extensive testing, since the fraction of combined operation is considered during testing. The Swiss approach requires only minimal testing, but the calculation becomes more extensive, since the running time of the system has to be evaluated to take into account the fraction of combined operation.

- Since testing is more costly than calculation, testing should be kept at minimum level.
- Alternate combined operation can be covered by weighting of the space heating-only and DHW-only operation modes, see chapter 3.5.1 Simultaneous combined operation has to be covered by additional testing for simultaneous operation.
- The Japanese proposal seems well suited for the investigated Japanese system, since the heat pump unit is decoupled from the different users by a heat storage. However, due to the profiles for both SH and DHW it might be difficult to characterise the combined operation in other systems, since depending on the profile little simultaneous operation may occur.
- Since there is no interpolation in the Japanese method testing has to be carried out for every operating point. Therefore, testing can become expensive, if there are many different operation modes of the system
- In the Swedish approach, no DHW-only operation is considered in wintertime. Depending on the characteristic for single and combined DHW operation, this could be quite a strong simplification, as characteristics may differ significantly. In intermediate season there might be quite a lot DHW-only operation as combined operation is limited by the operation of the space heating system. An example is the two stage cascade heat pump which is depicted in Fig. 14.

The Swiss retrofit heat pump is a cascade heat pump, where the lower stage is connected to the space heating system, while the upper part is connected to the DHW system. The lower and upper part can be operated independently. However, in simultaneous operation, i.e. if the lower and the upper stage operate at the same time, the upper stage uses condensate subcooling of the lower stage of as a heat source. In times of DHW-only operation, the heat is extracted from the ground source. Thus, the higher the supply temperature of the heating system is, the higher is the return temperature and thereby the source temperature for the upper stage of the heat pump.

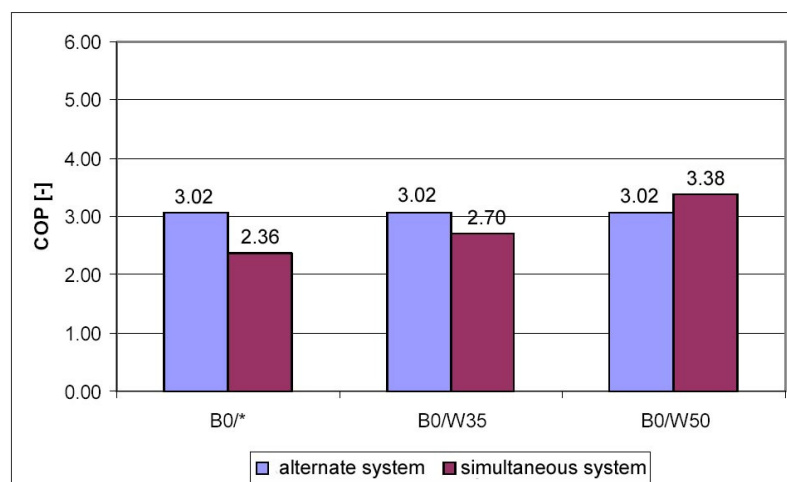


Fig. 29: Characteristic of the cascade heat pump and an alternate operating system ([9])

Fig. 29 shows the comparison of the COP characteristic of an alternate combined operating ground source heat pump and the Swiss retrofit heat pump [9]. While in case of the alternate combined operating system the COP is constant according to the expectations, the simultaneous combined operating system has a strong variation in the COP_t-characteristic, since source temperatures for the DHW operation vary considerably, namely from a COP_t in DHW-only operation of 2.4 (ground source) to a COP_t of 3.4 in combined operation at 50°C supply temperature of the space heating systems (condensate subcooling). If no DHW-only operation is considered, the simultaneous combined operating system performs better than the alternate combined operating system. However, even in wintertime, there will be DHW-only operation, since the combined operation depends on control as well as on the simultaneity of loads [14]. In intermediate season, simultaneous operation is even limited by the space heating system, since heating load decreases and DHW load stays constant, so that running time for DHW may be longer than the running time of the space heating system. Thus, for these system types with strongly varying characteristic the approach to evaluate only combined operation in wintertime seems to be a rather strong simplification and it appears, that the Swiss approach is the more general one. For the above reasons, the Swiss approach is chosen for the calculation method.

Proposal:

The calculation shall be carried out following the Swiss approach, since it is the more general approach to cover different system configurations, even though calculation might become more extensive due to the necessary evaluation of the running time of the heat pump in the different operation modes. On the other hand, it has thereby minimal testing requirements, since there is no need to consider the ratio of $Q_{out,g,h}/Q_{out,g,DHW}$.

3.2.5.1 Description of the extension of the calculation for combined operating systems

The chosen extension for combined operation is described in more detail in this chapter.

Depending on the system configuration, not all three operation modes space heating-only, DHW-only and simultaneous space heating and DHW may occur in simultaneous operating systems, e.g. the Swedish exhaust air heat pump with desuperheating depicted in Fig. 18 only runs in simultaneous operation in wintertime and in DHW-only operation in summertime. There are system configurations, for instance, where only simultaneous operation takes place in wintertime, so no space heating-only operation occurs. In this case only two characteristics have to be taken into account and the fraction of simultaneous operation is determined by the given heating period.

However, the example of the Swiss retrofit heat pump shows that there are also configurations, where three operation modes occur. Respective operation modes are evaluated by the running time, which is determined by the energy requirement divided by the heating capacity of the heat pump. Running time enables an evaluation of additional auxiliary energy requirement that is not considered in the standard testing according to EN 14511 and EN 255-3 [14].

In simultaneous combined operation, the heat pump is usually more efficient, so the maximum efficiency would be the maximum simultaneous combined operation. The maximum possible hot water production in simultaneous operation is characterised by the minimum of the running time of space heating and DHW, respectively, i.e.

- if $t_{ON,g,h,i} > t_{ON,g,DHW,i}$
by the maximum running time of the DHW, system $t_{ON,g,combi,max,i} = t_{ON,g,DHW,i}$ or
- if $t_{ON,g,h,i} < t_{ON,g,DHW,i}$
by the maximum running time of the space heating $t_{ON,g,combi,max,i} = t_{ON,g,h,i}$

where

$t_{ON,g,h,i}$	running time to cover the space heating requirement and losses in bin i	(s)
$t_{ON,g,DHW,i}$	running time to cover the DHW requirement and DHW losses in bin i	(s)
$t_{ON,g,combi,max,i}$	maximum possible running time in simultaneous operation in bin i	(s)

The running time in simultaneous operation mode may also be influenced by the control and the load profile. The control system may have the larger impact, since controller setting defines the time periods, where simultaneous operation takes place. However, this depends strongly on the system configuration and could only be taken into account by a specific correction factor. A possibility to take into account this impact is a correction factor according to the equation

$$t_{ON,g,combi,i} = f_{combi} \cdot t_{ON,g,combi,max,i} \quad [s] \quad \text{eq. 13}$$

where

$t_{ON,g,combi,i}$	running time in simultaneous operation in bin i	(s)
$t_{ON,g,combi,max,i}$	maximum possible running time in simultaneous operation	(s)
f_{combi}	correction factor taking into account the impact of the control system and the shift in the demand.	(-)

Adequate factors for typical controller settings could be given in a national Annex based on a specific evaluation of the system configuration.

In the Swiss national project [14], simulations of the extreme situations have been performed on the basis of the characteristic of the Swiss Retrofit Heat Pump depicted in Fig. 14 to evaluate the impact. As a result, the simultaneous operation varies from 30% to 60 % in wintertime leading to a variation in the SPF for a retrofit building of about 6% [14]. On this basis it seems also feasible to assume the best case for the combined operation. However, one has to bear in mind that these results are derived on the basis of a specific characteristic of a heat pump, so general statements are limited by this fact.

So, if no values are given, $f_{combi} = 1$ shall be set, which means that the control is optimised for the simultaneous operation.

The respective produced energy in the bin for simultaneous and the space heating-only and DHW-only operation can then be calculated by an energy balance

$$Q_{hp,DHW,combi,i} = t_{ON,g,combi,i} \cdot \phi_{hp,DHW,combi,i} \quad [J] \quad \text{eq. 14}$$

$$Q_{hp,DHW,sin,i} = Q_{hp,DHW,i} - Q_{hp,DHW,combi,i} \quad [J] \quad \text{eq. 15}$$

where

$Q_{hp,DHW,combi,i}$	produced DHW energy by the heat pump in simultaneous operation in bin i	(J)
$t_{ON,g,combi,i}$	running time in simultaneous operation in bin i	(s)
$\phi_{hp,DHW,combi,i}$	heating capacity of the heat pump for simultaneous DHW operation in bin i	(W)
$Q_{hp,DHW,sin,i}$	produced DHW energy in DHW-only operation in bin i	(J)
$Q_{hp,DHW,i}$	produced energy for DHW in bin i	(J)

The same equations apply for the space heating operation.

The used energy is calculated by subtracting the storage losses, which are proportional to the running time in simultaneous operation. For the calculation of the seasonal performance factor for simultaneous operation the equations eq. 11 and eq. 12 apply.

To calculate the overall seasonal performance factor, all three operation modes have to be weighted with the respective seasonal performance in each operation mode according to the equation

$$SPF_g = \frac{Q_{out,g,h} + Q_{out,g,DHW}}{\frac{Q_{out,g,h,sin}}{SPF_{g,h}} + \frac{Q_{out,g,DHW,sin}}{SPF_{g,DHW}} + \frac{Q_{out,g,h,combi} + Q_{out,g,DHW,combi}}{SPF_{g,combi}}} \quad [-] \quad \text{eq. 16}$$

where

SPF_g	overall seasonal performance factor of the generation subsystem	(-)
---------	---	-----

$Q_{out,g,h}$	space heating energy requirement	(J)
$Q_{out,g,DHW}$	DHW energy requirement	(J)
$Q_{out,g,h,sin}$	space heating energy requirement covered in space heating-only operation	(J)
$Q_{out,g,DHW,sin}$	DHW energy requirement covered in DHW-only operation	(J)
$Q_{out,g,h,combi}$	space heating requirement covered in simultaneous operation	(J)
$Q_{out,g,DHW,combi}$	DHW requirement covered in simultaneous operation	(J)
$SPF_{g,h}$	seasonal performance factor in space heating-only operation	(-)
$SPF_{g,DHW}$	seasonal performance factor in DHW-only operation	(-)
$SPF_{g,combi}$	seasonal performance factor in simultaneous operation	(-)

3.2.5.2 Assessment of the calculation method

Obviously, the calculation method implies some simplifications in order to keep the calculation simple.

One shortcoming with regard to low and ultra-low energy house may be the redistribution of the energy requirement to the bins is an approximation, since the cumulative frequency is dependent on the heating degree hours and thereby on the outdoor air temperature, while in low-energy dwellings the solar irradiation and internal gains may have a significant impact on the energy demand and distribution (see comment in chapter 3.4.1). However, the heating limit can be used to better approximate the distribution, see Swiss country report [37]. Note that the quantity of energy is taken into account correctly, but the redistribution to the bins is an approximation.

A further shortcoming is that controller layout and controller setting can only be partly taken into account, with the consequence that the calculation may require default factors to refer to standard control situations.

First validation to prove the validity of different approaches is described in chapter 3.3.

3.2.6 Extension of the approach to compact units

The calculation approach for compact units is given in part 1, chapter A 2. More details are given in the Swiss report [38]. In the following the basic assumptions of the calculation are presented.

In compact units a heat pump is connected to the heat recovery unit of the ventilation system. The heat pump supplies heat to both the space heating and the domestic hot water system. The combination of a heat pump and a heat recovery unit has the following consequences for the calculation:

- The inlet temperature of the heat pump is influenced by the outlet temperature of the heat recovery unit, i.e. depending on the layout of the system the total temperature fluctuation of the outdoor air is reduced.
- The space heating requirement of the house is reduced by the energy recovery of the mechanical ventilation system, thus the heat pump has to deliver less energy in space heating mode.

As a result, the heat recovery can be treated like an additional generator - even though there is no real heat generation, but only a reduction of the requirement by the recovery of ventilation losses.

Concerning the domestic hot water operation, the only influence of the heat recovery unit is a changing inlet temperature to the heat pump, since the domestic hot water requirement is not reduced as in case of the space heating operation.

The modified scheme of the bin method taking into account the reduction in the space heating requirement is shown in Fig. 30. It illustrates how the heat recovery unit is integrated in the calculation. A considerable share of the ventilation losses are recovered by the heat recovery unit, the heat pump covers the remaining requirement up to the capacity limit due to the design and the rest is covered by the electrical back-up heater.

3.2.6.1 Additional input data for compact units

For the calculation method, additional input data of the heat recovery unit are required

- Nominal total volume flow rate (heat recovery and heat pump in case of combination of exhaust air and outside air for the heat pump)
- Nominal flow rate for the heat recovery
- Nominal power consumption of the fans of the heat recovery

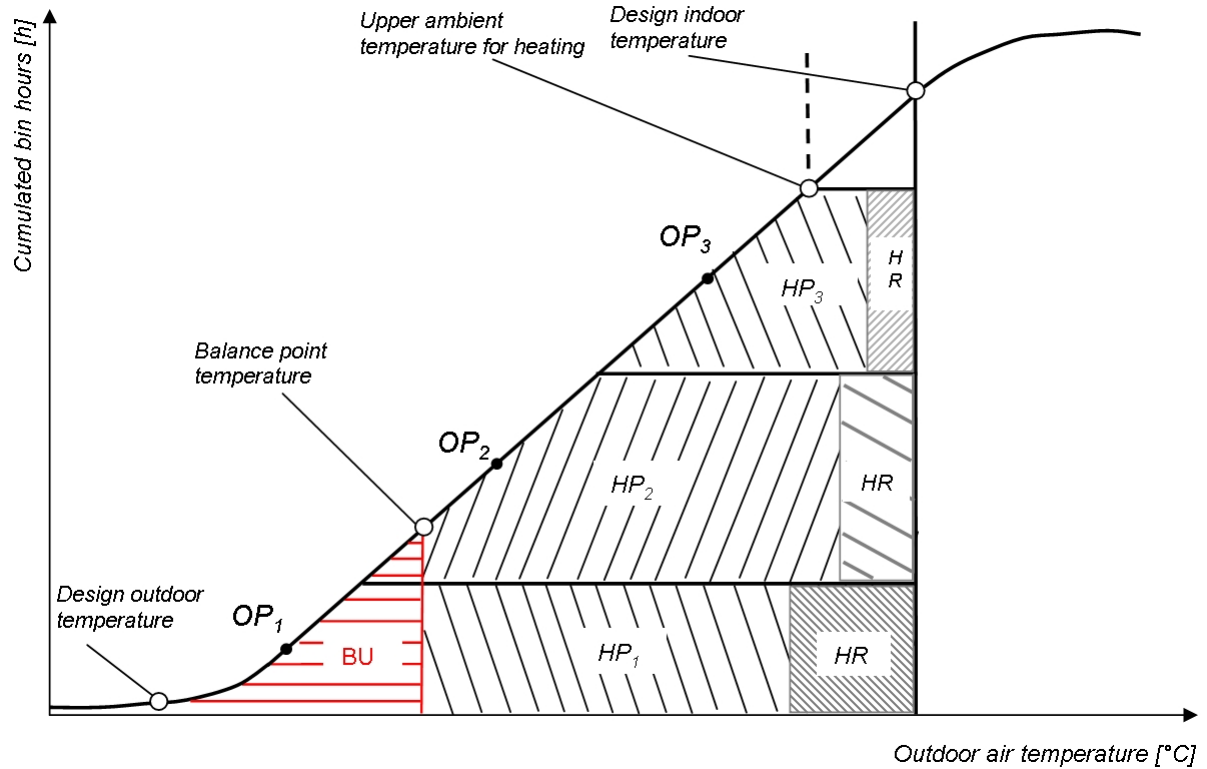


Fig. 30: Extended bin-method for heat pump compact units

3.2.6.2 Additional test values for compact units used for the calculation

In order to calculate the reduction of the space heating requirement, testing of the compact unit has to provide the component characteristic of the heat recovery. The entire thermodynamic part of the test procedure developed at HTA Lucerne is integrated in part 1, chapter B 4.4.1 and background on the testing are given in chapter 3.5.8.

For the calculation, the temperature change coefficients are used. It can be defined for the supply air side and the return air side of the heat recovery by the following equations:

$$\Phi_{oa} = \frac{\theta_{sa} - \theta_{oa}}{\theta_{ra} - \theta_{oa}} \quad \Phi_{ra} = \frac{\theta_{ra} - \theta_{ea}}{\theta_{ra} - \theta_{oa}} \quad [-]$$

eq. 17

where

Φ_{oa}	temperature change coefficient on the air inlet side (from outside to building)	(-)
Φ_{ra}	temperature change coefficient on the air outlet side (from building to outside)	(-)
θ_{ea}	air temperature of exhaust air exiting the heat recovery	(°C)
θ_{sa}	air temperature of supply air exiting the heat recovery	(°C)
θ_{oa}	outside air temperature	(°C)
θ_{ra}	air temperature of return air from the building (entering heat recovery)	(°C)

3.2.6.3 Assumptions for the calculation

The calculation applies some assumptions and simplifications.

Concerning the heat recovery, it is postulated, that energy losses should be minimized and therefore, the heat recovery runs through the entire heating period, thereby keeping ventilation losses at a minimum. This enables the calculation of the reduction of the heat requirement, since the running time is known. Moreover, it is postulated, that the efficiency of the heat recovery in the bin does not change. In the case of changing operation of the heat recovery, e.g. if heat pump operation has an impact on the quantity of energy recovery of the heat recovery unit, the calculation would have to be performed iteratively.

The running time also enables to evaluate the electricity consumption with regard to the fan power consumption.

If the test is performed according to the proposed procedure in part 1, chapter 4.4.1 no correction for the inlet temperature is required, since the test is done as combined test of the heat recovery operation and heat pump operation. If separate testing data of the heat pump and the heat recovery exist, the inlet temperature of the heat pump can be evaluate by the return air temperature change coefficient. Either the outlet temperature of the exhaust air from the heat recovery has to be taken (only ventilation air used by the heat pump) or a mixture of exhaust air and outside air. The temperature of the exhaust air can be calculated using the return air temperature change coefficient given in eq. 17.

3.2.7 Approach for calculation according ASHRAE 137 (Canada)

In the frame of the Canadian national project, the calculation method of ASHRAE 137 [20] for A/A heat pumps with desuperheater for combined cooling/DHW and combined space heating/DHW has been applied to a simultaneously operating ground-source heat pump. The ASHRAE approach uses also a bin-method with separation of the single operation modes.

The basic equations for the SPF calculation are given in eq. 4 - eq. 8 in chapter 2.3. More detailed information on American standardisation and testing and SPF calculation method can be found in the Interim Report of IEA HPP Annex 28 [2] or the Canadian country report [34].

For the calculation a correlation between ground temperature and ambient air temperature as depicted in Fig. 31 is proposed. Thus, the number of bins for A/A heat pumps have been reduced for B/W heat pumps to 4 bins.

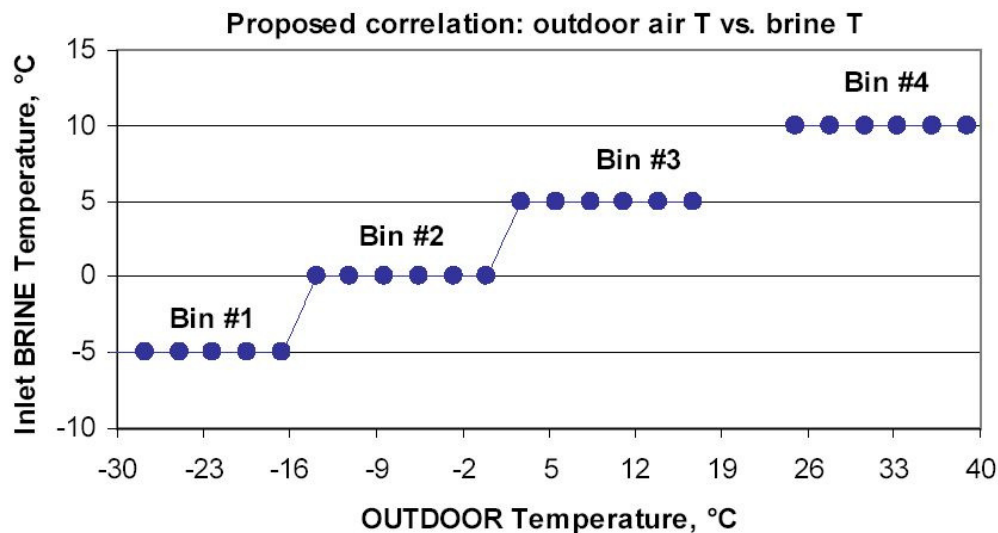


Fig. 31: Correlation between outdoor air temperature and brine temperature for the Canadian calculation (source [34])

The respective fractional bin hours for the climatic region V (Canada) are given in Fig. 32

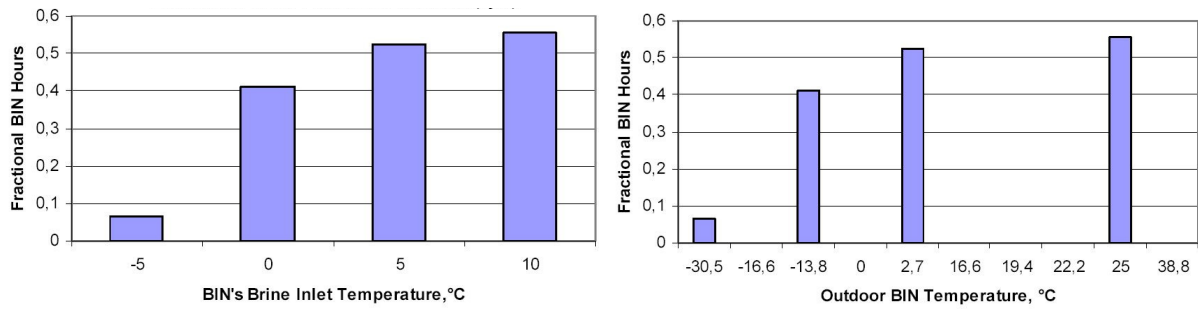


Fig. 32: Fractional bin hours for the brine-to-water combined heat pump for brine temperature (left) and outdoor air temperature (right) (source [34])

With the respective fractional bin hours the single fractions in eq. 4-8 are evaluated in order to calculate the seasonal performance factor.

3.3 Results of field measurements and validation of the calculation method

A principal task for the development of the calculation method and the respective test procedure is a comparison of the calculated results with measured data to get an idea of the exactness to be expected by the calculation. This validation on the basis of the field measurements is in progress. However, not all systems in field monitoring could have been evaluated due to the timeframe of IEA HPP Annex 28. Moreover, not all types of systems are monitored in the IEA HPP Annex 28, so further evaluation of the calculation method and the test procedure is a future task.

3.3.1 Validation for ground source direct expansion heat pump systems

3.3.1.1 Evaluation of the ground temperature

In the case of ground source heat pumps, one problem is the evaluation of the ground temperature as source temperature, since the bin method is related to a temperature basis, but the ground source temperature depends as well on the exhaustion of the ground and therefore the time of the year. A linear fit as approximation of the ground temperature based on the outside air temperature is presented in Fig. 33. The fit was accomplished based on measured ground temperature in different depths in the ground shown in Fig. 34. The comparison of the field monitoring and the calculation is based on this approximation of the ground temperature. In the following the results of the calculation are presented for the three systems with horizontal direct expansion [44].

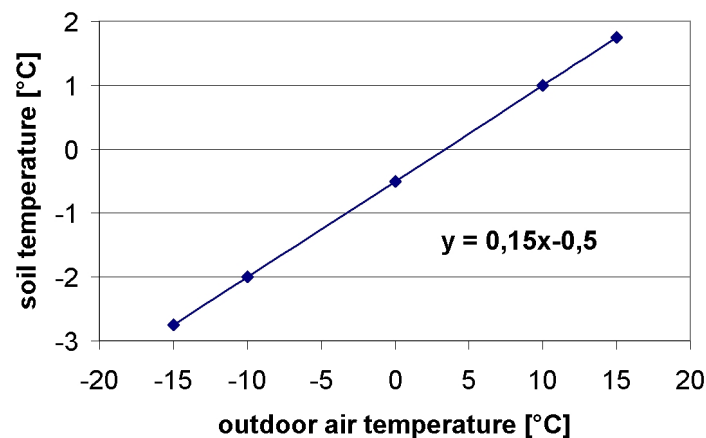


Fig. 33: Fitted ground temperature dependent on the outdoor air temperature (source [44])

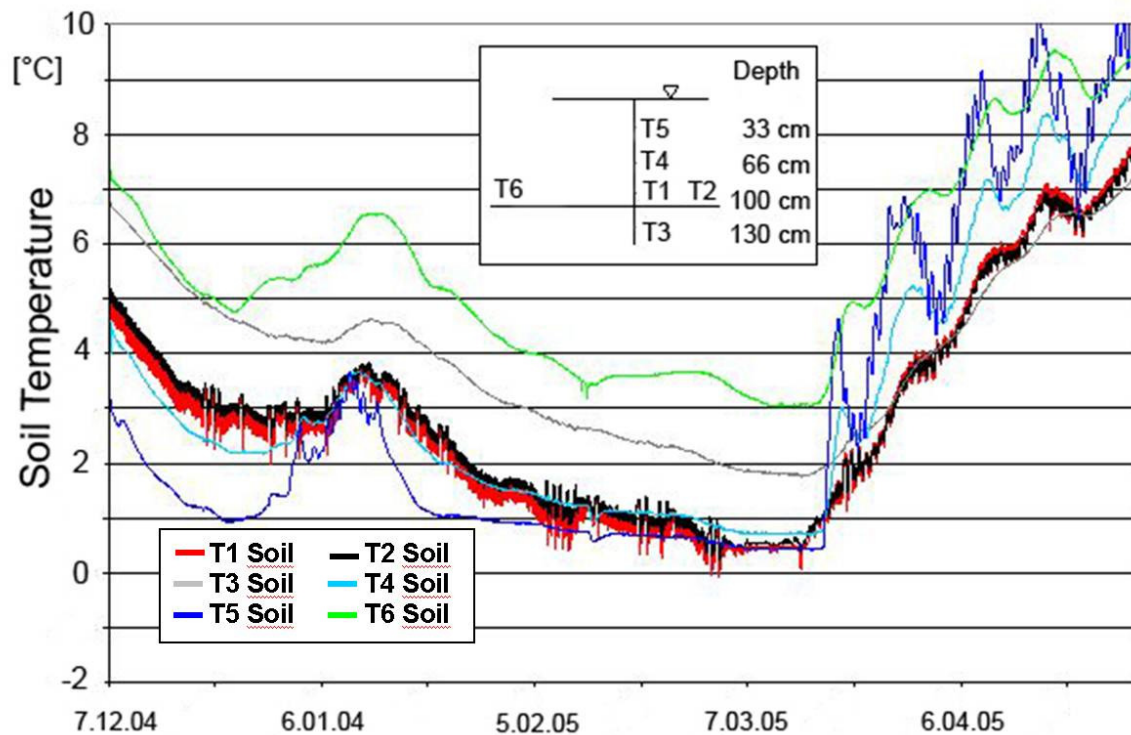


Fig. 34: Measured ground temperature (modified from [44])

3.3.1.2 Heat pump system 1 (single family house, 240m², horizontal DX heat pump system)

Evaluation period: November 03 – October 04

Tab. 7: Monitoring object 1 of the Austrian monitoring program

Design heat load	8.5 kW
Outdoor design temperature (θ_{OD})	-16°C
Indoor design temperature	22°C
Supply temperature at θ_{OD}	35°C
Heating limit	18°C
Supply temperature at heating limit	28°C

System 1 of the Austrian monitoring program is a single family house of an area of 240m² with a horizontal direct expansion system. Tab. 7 gives the design parameters of the house, Tab. 8 gives the measured characteristic of the system.

Tab. 8: Heat pump characteristic system 1

Source temperature [°C]	Supply temperature [°C]	Heating capacity [kW]	COP [-]
-1	35	9.3	4.2
4	35	10.6	4.4
10	35	11.8	5.2
-1	50	9.3	3.1
4	50	10.0	3.4
10	50	11.6	3.8

Austria has performed the calculation with 7 bins for the space heating mode. Fig. 35 gives the energy distribution to the bins.

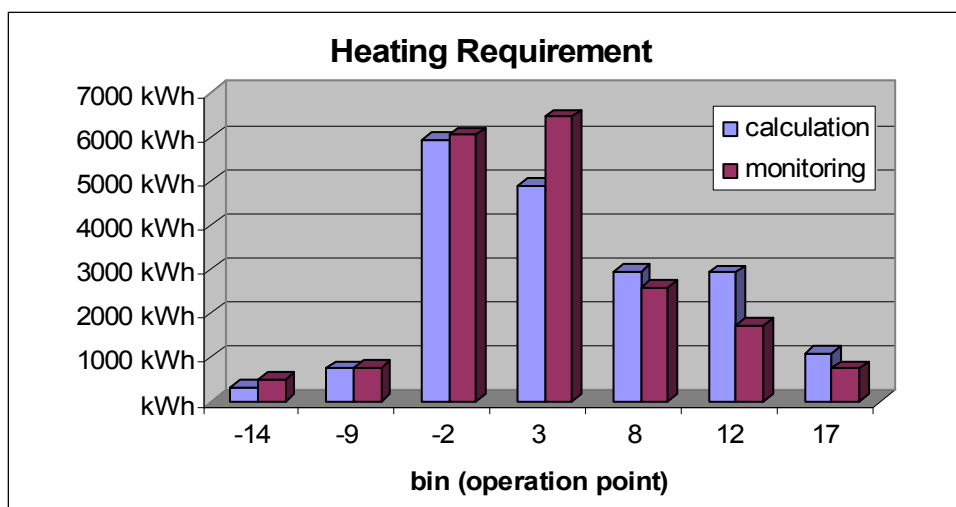


Fig. 35: Energy distribution of the bins in calculation and monitoring (source [44])

Tab. 9 gives the results of the calculation. The result of the calculation is quite exact, the deviation between the calculation and measurements is 3% for the seasonal performance. The calculation delivers higher values, which could be due to the redistribution of the space heating requirement which leads to higher energy amounts in the upper bins.

Tab. 9: Results system 1 (source [44])

	Monitoring result	Calculation result	Difference
Heating energy requirement [kWh]	18905	18905	Set input value
Electrical energy input [kWh]	4410	4280	2.9 %
Running time [h]	2063	2038	1.2%
SPF [-]	4.29	4.42	3 %

3.3.1.3 Heat pump system 2 (single family house, 260 m² horizontal DX heat pump system)

Evaluation period: October 03 – September 04

Design heat load	11.3 kW
Outdoor design temperature (θ_{OD})	-16°C
Indoor design temperature	22°C
Supply temperature at θ_{OD}	35°C
Heating limit	18°C
Supply temperature at heating limit	28°C



Fig. 36: Monitoring object 2 of the Austrian monitoring program

Monitoring object 2 is a single family house of an area of 212 m² with a 260 m² direct expansion system. Design parameters of the house are given in Fig. 36 and the tested heat pump characteristic is shown in Tab. 10.

Tab. 10: Heat pump characteristic system 2

Source temperature [°C]	Supply temperature [°C]	Heating capacity [kW]	COP [-]
-1	35	8.9	4.2
4	35	10.5	5.0
10	35	12.7	6.4
-1	50	8.5	2.8
4	50	9.9	3.3
10	50	11.5	3.8

Fig. 37 shows the energy distribution of the space heating requirement of the monitoring and the calculation for the chosen seven bins.

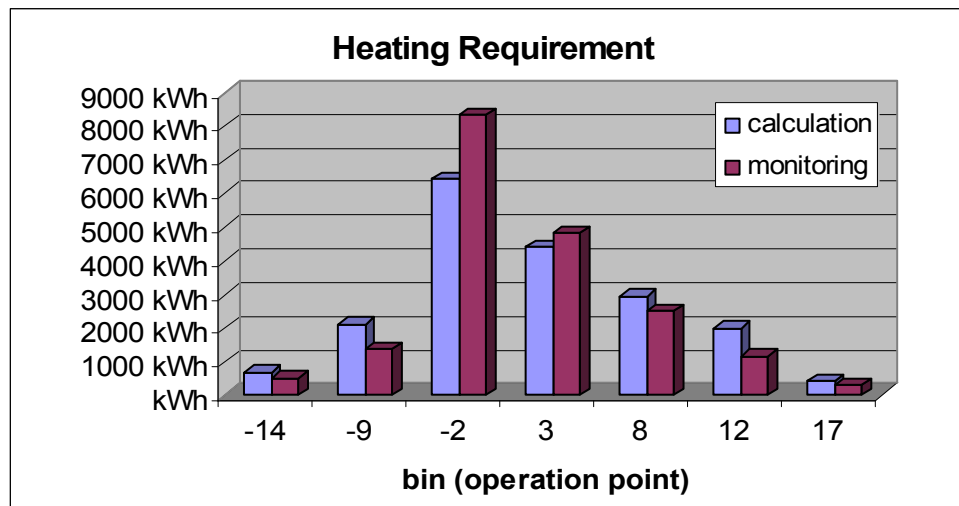


Fig. 37: Energy distribution of the bins in calculation and monitoring (source [44])

Results of System 2 are given in Tab. 11. In this case the calculation delivers a lower SPF-value, even though there is higher energy consumption in the upper bins and a lower in the middle. However, the consumption in the lower bins is higher, as well.

Tab. 11: Results system 2 (source [44])

	Monitoring result	Calculation result	Difference
Heating energy requirement [kWh]	19012	19012	Set input value
Electrical energy input [kWh]	3976	4208	5.8 %
Running time [h]	2819	1951	- 30.8 %
SPF [-]	4.79	4.52	- 5.6 %

What is striking is the high difference in running time of the heat pump.

3.3.1.4 Heat pump system 3 (single family house, 400m² horizontal DX heat pump system)

Monitoring object 3 is a single family house of an area of 320m² with 400m² direct expansion system. Details of the design are given in Fig. 38.

Evaluation period: November 04 – April 05

Design heat load	13.5 kW
Outdoor design temperature (θ_{OD})	-16°C
Indoor design temperature	22°C
Supply temperature at θ_{OD}	45°C
Heating limit	18°C
Supply temperature at heating limit	30°C



Fig. 38: Monitoring object 3 of the Austrian monitoring program

The monitoring of this system comprises only a period of six months, so not the entire heating period is covered. Measured characteristic is presented in Tab. 12.

Tab. 12: Heat pump characteristic system 3

Source temperature [°C]	Source temperature [°C]	Heating capacity [kW]	COP [-]
-1	35	13	4.1
4	35	16	5.0
10	35	17.7	5.5
-1	50	12.1	2.9
4	50	13.9	3.3
10	50	16.7	3.8

Since for system 3 the evaluation period comprises only the time from November to April, no distribution of the energies was evaluated. The results of the calculation are given in Tab. 13.

Tab. 13: Results system 3 (source [44])

	Monitoring result	Calculation result	Difference
Heating energy requirement [kWh]	12338	12338	Set input value
Electrical energy input [kWh]	3731	3162	-15.3 %
Running time [h]	960	1327	38.2 %
SPF [-]	3.31	3.90	17.8 %

3.3.1.5 Assessment of the validation for DX-heat pump systems

The evaluation of the calculation method for ground source heat pumps (DX-expansion systems) has shown that the calculated results vary in range of 3% to 18% compared to the monitored field measurements. Both a deviation to higher and to lower values of the seasonal performance was calculated.

The highest deviation was calculated for system 3 with 18%. However, system 3 has not been evaluated with the measured outside temperatures but with standard temperatures. The measured data of the ambient air temperature were not available at the time of the calculation, because they are not accessible before the total monitoring is terminated. Actually, the reason for this deviation is probably the comparison of field measurements of a strong winter with a calculation using average meteorological data for the ambient air temperature. Thus, this system is not a real validation but a comparison of average design data with field monitoring and should thus be excluded from the consideration. Taking into account this fact, the bin method delivers values in the range of ± 6 %.

3.3.2 Validation for compact unit heat pump system

3.3.2.1 Results from the field monitoring

The pilot plant in Gelterkinden, canton BL in Switzerland is a single-family house according to the Swiss MINERGIE standard (www.minergie.ch). Fig. 39 contains information on the design parameters of the pilot plant. The pilot plant is equipped with a heat pump compact unit.

Evaluation period: May 2004 – May 2005

Design heat load	4.1 kW
Outdoor design temperature (θ_{OD})	-8°C
Indoor design temperature	20°C
Supply temperature at θ_{OD}	29°C
Upper temperature limit for heating	14°C
Supply temperature at heating limit	23°C
Balance point	-5°C



Fig. 39: Single-family house in Gelterkinden, BL, equipped with the compact unit

The thermal and electrical energy consumption for the evaluated period is shown in Fig. 40.

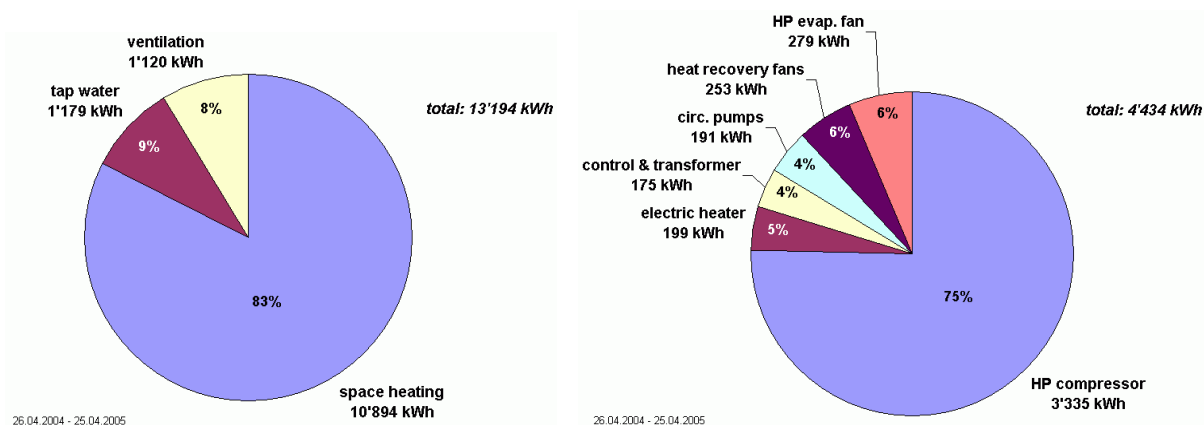


Fig. 40: Energy consumption of the pilot plant in Gelterkinden [37]

Concerning the heat energies, the space heating fraction is the dominant part, the DHW consumption is extremely low due the user behaviour (about half of the standard value of Swiss building regulation SIA 380/1) and therefore, the ratio between the two consumptions is not representative for this building standard.

The heat recovery unit reduces the space heating energy requirement by 8%. Concerning the electrical energy input to cover the heat requirement about 80% are consumed by the heat pump (75% compressor, 6% fan). The supplementary heating contributes only 5% to both space heating and domestic hot water. Heat recovery makes up 6% and the control of the compact unit contributes 4% to the total consumption, and the missing 4% are consumed by the pumps.

The evaluation of the SPF of the pilot plant has been performed for different system boundaries, the heat pump SPF-HP, the so-called „seasonal performance factor generator“ SPF-G, and the „seasonal performance factor system“ SPF-S. The three system boundaries are depicted in Fig. 41.

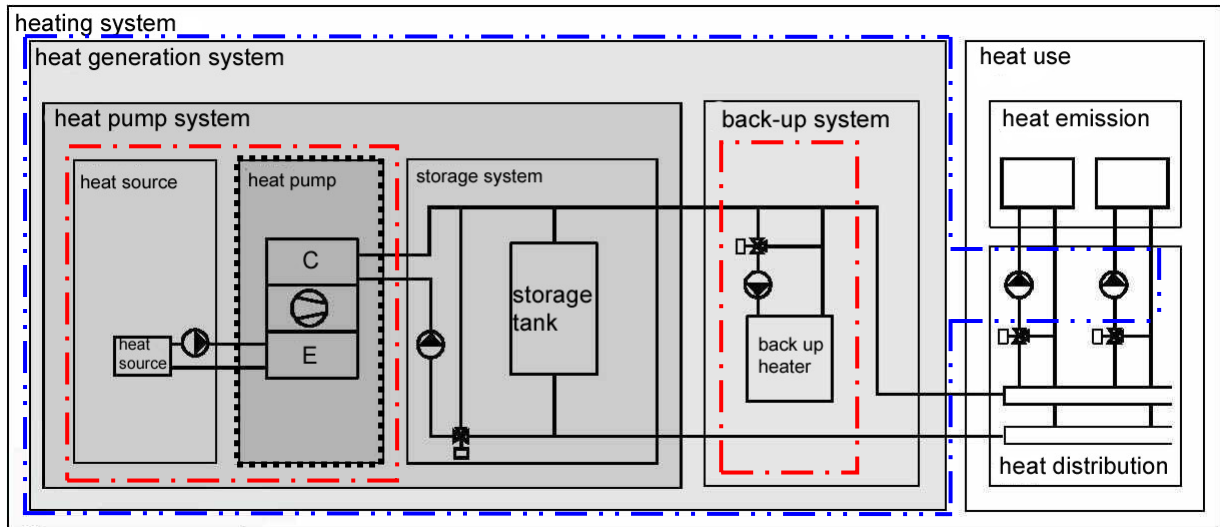


Fig. 41: System boundaries for the seasonal performance evaluation

The system boundary SPF-HP (dotted bold line in black) refers to the definition of EN 14511.

The seasonal performance factor generator (SPF-G) (dash-dotted line in red) includes all the different generators in the system boundary, i.e. in this case, the back-up heater is included in the system boundary which leads to the equation

$$\text{SPF - G} = \frac{Q_{\text{hp,h}} + Q_{\text{hp,DHW}} + Q_{\text{bu}}}{E_{\text{in,g,hp}} + E_{\text{in,g,bu}} + W_{\text{g}}} \quad [-] \quad \text{eq. 18}$$

where

SPF-G	seasonal performance factor “generator”	(-)
$Q_{\text{hp,h}}$	produced heat of the heat pump in space heating operation	(J)
$Q_{\text{hp,DHW}}$	produced heat of the heat pump in DHW operation	(J)
Q_{bu}	produced heat of the back-up heater in space heating and DHW	(J)
$E_{\text{in,g,hp}}$	electricity consumption of the heat pump	(J)
$E_{\text{in,g,bu}}$	electricity consumption of the electrical back-up heating	(J)
W_{g}	additional auxiliary electricity consumption of the source pump, control, carter heating	(J)

Thus, the SPF-G evaluates the produced heat of the system to cover the space heating requirement and the generator losses and is therefore well suited to compare different generators, e.g. heat pump and boiler. Since the heat recovery is not a generator, but only reduces the requirements, it is not included in this system boundary.

The seasonal performance factor system (SPF-S) (dash – double dotted line in blue) is related to used energy, so additional system losses are taken into account, e.g. storage losses and the pumping energies according to the equation

$$\text{SPF - S} = \frac{Q_{\text{out,g,h}} + Q_{\text{out,g,DHW}} + Q_{\text{hr}}}{E_{\text{in,g,hp}} + E_{\text{in,g,bu}} + E_{\text{hr}} + W_{\text{g}}} \quad [-] \quad \text{eq. 19}$$

where

SPF-S	seasonal performance factor “system”	(-)
$Q_{\text{out,g,h}}$	used heat of the building (space heating energy requirement)	(J)
$Q_{\text{out,g,DHW}}$	used heat of the DHW system (domestic hot water energy requirement)	(J)

Q_{hr}	heat energy recovered by the ventilation heat recovery system	(J)
$E_{in,g, hp}$	electricity consumption of the heat pump	(J)
$E_{in,g, bu}$	electricity consumption of the electrical back-up heating	(J)
E_{hr}	electricity consumption of the heat recovery unit	(J)
W_g	additional auxiliary electricity consumption of pumps, control, carter heating	(J)

In this system boundary, which refers to used energy, the heat recovery is included. Consequently, the SPF-S characterises the system performance and corresponds to the system boundary „generation and storage“ in Fig. 23.

Tab. 14 gives the performance key values of the field monitoring of the compact unit according to the different system boundaries for the winter and summer period. The value ETV describes the electro-thermal amplification of the heat recovery, i.e. the ratio of recovered heat for the supply air to electricity input for the operation. Φ_{oa} is the temperature change coefficient defined in eq. 17.

Tab. 14: Key values of the pilot plant Gelterkinden for the summer and winter period and the entire year [37]

HP compact unit Gelterkinden/BL	SPF-HP	SPF-G	SPF-SYS	ETV (weekly average)	Φ_{oa} (weekly average)
Summer	3.6	3.1	2.4	1.9..2.7	0.61..0.70
Winter	3.6	3.4	3.1	2.0..6.7	0.64..0.77
Year	3.6	3.4	3.0	1.9..6.7	0.61..0.77

The SPF-values are overall seasonal performance values for space heating and domestic hot water operation.

3.3.2.2 Comparison of calculated values to field measurements for the space heating

The calculation has been performed according to the proposed extension described in part 1 chapter A2 of the final report. Since no testing of the compact unit was performed in IEA HPP Annex 28, manufacturer data have been used to perform the calculation. Three bins were used for the calculation. Operating points are located at the test points according to the proposed method is part 1, chapter B 4.4.1.

Tab. 15: Manufacturer data of the heat pump according to test points of the proposed test procedure (see part 1, chapter B 5.4)

Source temperature [°C]	Supply temperature [°C]	Heating capacity [kW]	COP [-]
-7	35	3.36	2.90
7	35	4.66	3.54
-7	45	2.71	2.31
2	45	3.49	2.62
7	45	4.19	2.98

Tab. 15 gives the manufacturer data for the heat pump and Tab. 16 give the characteristic of the heat recovery unit.

The compact unit shown in the principle sketch in Fig. 9 has the particularity that the entering outside air flow is preheated in times of heat pump operation with an additional heat exchanger (subcooler), which takes the heat from condensate subcooling of the heat pump cycle. That means the characteristic of the heat recovery unit might change in times of heat pump operation due to this preheating. However, for the calculation, the impact can be neglected. The monitoring shows that the characteristic, e.g. the supply temperature of the air after the heat recovery does not change significantly with or without the preheater operation, since the preheated air causes a lower heat transfer in the heat recovery. Either the outside air is preheated by the subcooler or by the heat transfer in the heat recovery.

The calculation is based on a reduction of the total space heating energy requirement by the fraction delivered by the heat recovery in order to determine the space heating energy to be supplied by the

Tab. 16: Manufacturer data of the heat recovery according to test points of proposed test procedure (see part 1, chapter B 5.4)

Outside temperature [°C]	Indoor temperature [°C]	volume flow rate [m ³ /h]	Temperature change coefficient outside - supply air [-]	Temperature change coefficient return – exhaust air [-]
-7	20	101	0.8	0.64
2	20	101	0.84	0.75
7	20	101	0.84	0.81
-7	20	162	0.79	0.76
2	20	162	0.81	0.80
7	20	162	0.82	0.83

heat pump. This is only possible under the assumption, that the heat recovery runs through and that the characteristic of the heat recovery unit does not change in the bin. However, for the exactness required, the heat recovery unit can be calculated without iteration in every case, see above for the case of the subcooler.

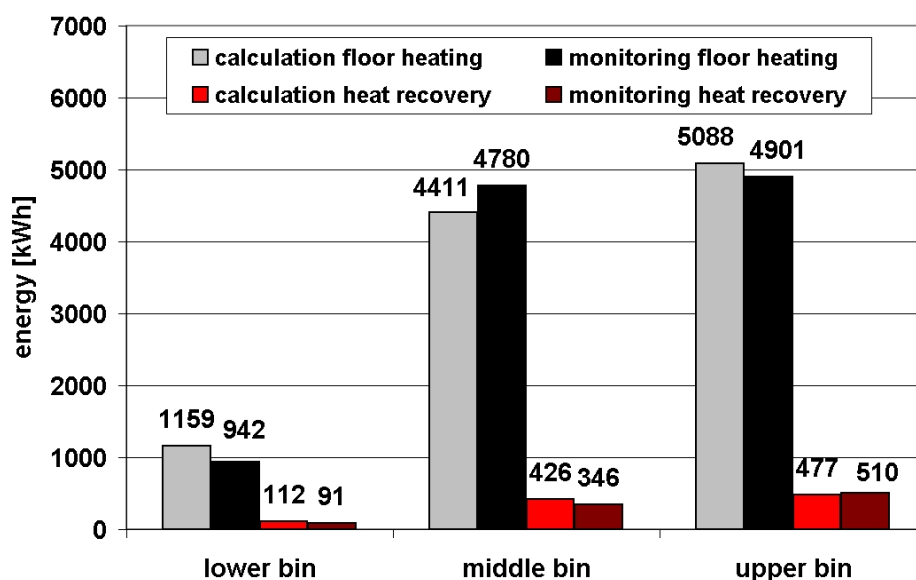


Fig. 42: Distribution of energies to the bins for the floor heating and the ventilation system

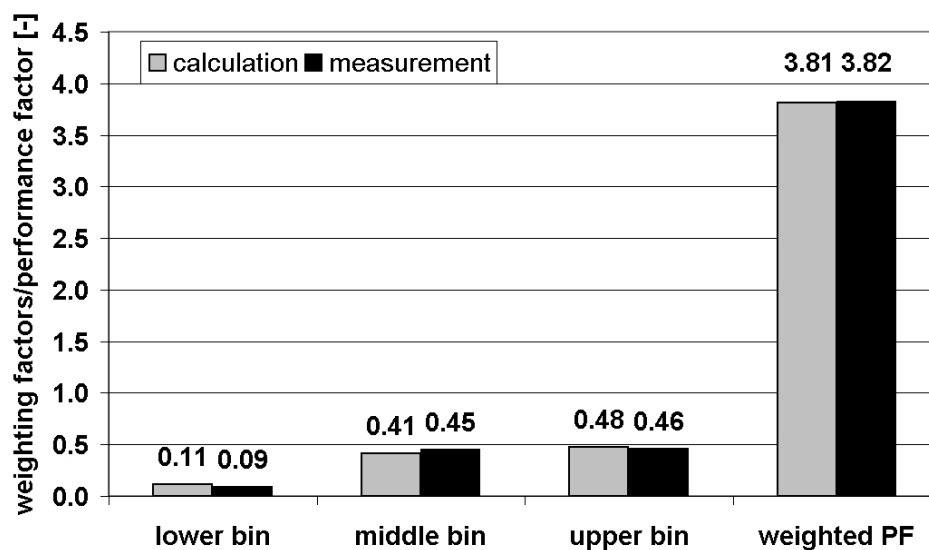


Fig. 43: Weighting factors derived by measured energies and according to the calculation

Fig. 42 shows the distribution of the space heating energy requirement covered by the floor heating and by the ventilation system heat recovery unit in the bins. Fig. 43 shows the respective weighting factors for the heat pump. For the calculation the weighting factors to redistribute the energy requirement to the bins were derived by the cumulative frequency of the outside air temperature, the monitoring values refer to the respective measured energy consumption. It becomes clear, that even though there are some differences in the calculated energy redistribution of the bins, the weighting factors only differ slightly and the weighted performance factor is almost the same, so the redistribution to the bins by means of the cumulative frequency delivers good results.

In Tab. 17 results of the comparison are shown for the three system boundaries depicted in Fig. 41. The calculation of the **heat recovery unit** shows a small difference between the calculation and the monitoring. This difference is basically due to the fluctuating mass flow rate of the heat recovery. For the calculation a mean mass flow has been taken. Moreover, temperature change coefficient is taken at the operating point.

The **SPF-HP** was calculated by weighting the COP characteristic according the weighting factors that are depicted in Fig. 43. Weighting with the different redistribution according to the calculation and the monitoring leads to almost the same PF-values. However, the weighted SPF-HP in the calculation differs from the COP evaluated of the real electricity consumption ($\text{SPF-HP} = Q_{\text{hp}}/E_{\text{hp}} = 3.70$), which is in the normal range of the exactness of COP measurements of about 5%.

The amount of **electrical back-up heating** differs quite a lot from the calculated one, but the absolute amount of back-up energy for space heating is small. The back-up energy has been evaluated by the

Tab. 17: Results of the calculation for the pilot plant Gelterkinden in space heating mode

	Monitoring (Reference)	Calculation	Difference
System boundary heat pump			
Space heating energy requirement [kWh]	11707	11707	Set input value
Reduction of space heating energy requirement by heat recovery [kWh]	947	1015	7.2 %
Evaluation time (heating period) [h]	5196	5196	Meteo data
Electrical energy heat pump [kWh]	2875	2795	-2.8 %
SPF-HP by weighting over bins	3.82	3.81	-0.2 %
SPF-HP monitoring	3.69	-	3.3 %
System boundary SPF-G			
Electrical back-up energy input [kWh]	137	33	-75.9 %
Running time of the heat pump [h]	2496	2202	-11.8 %
Auxiliary electrical energy for control (during stand-by) [kWh]	28	30	7.1 %
Electrical energy "generator" [kWh]	3040	2858	-6.0 %
Heat supplied to floor heating [kWh]	10760	10692	-0.6 %
SPF-G	3.54	3.74	5.6 %
System boundary SPF-S			
Electrical energy heat recovery (fans and control) [kWh]	208	212	1.9%
Auxiliary energy for circulation pump [kWh]	158	281 (121)	77.8% (-23.4 %)
Electrical energy "system" [kWh]	3406	3351 (3191)	-1.6 %, (-6.3 %)
SPF-S	3.44	3.49 (3.67)	1.5 %, (6.7)%

balance point temperature of -5°C for parallel operation. A more detailed energy balance would deliver a slightly higher value of 35 kWh. If the balance point would be higher, this absolute difference could be more significant, so a more detailed approach evaluating the required running time of the heat pump could be considered, see comment in chapter 3.4.2.

However, in this case, the reason could be traced to a difference of the measured heating capacity and the heating capacity according to the characteristic from testing in the lower temperature range. The characteristic showed slightly higher values than the measured one, so in the calculation, more heat can be covered by the heat pump and thus, less back-up energy is required than in the monitoring.

For the same reason of different heating capacity values the **running time of the heat pump** varies between monitoring and calculation. The different energy consumptions of the **control system** (stand-by energy) are due to the differences in running time, too.

Summing-up the electrical energy consumption for the **SPF-G**, the difference of the two values is 5.9%, which is basically due to the heat pump energy and the back-up energy.

Accordingly, the **SPF-G** differs about the same value with 5.6%, since the heat produced by the heat pump also differs a bit due to the heat recovery calculation.

Additionally, the heat delivered by the heat recovery and the electrical energy consumption of the circulation pump and the heat recovery has to be taken into account to derive the SPF-S.

Difference in the **electrical input to the heat recovery** is due to a slightly lower measured average consumption of the fans in the monitoring during the heating period compared to the nominal value. The energy value in Tab. 17 includes as well a fraction of the control for the entire heating period.

The real control of the **circulation pump** of the heating system is dependent on the outside temperature. The control strategy varies among manufacturers and is often not known, so it cannot be entirely taken into account. Thus, in order to keep the calculation mainly independent of the control strategy, the assumption is made for the calculation that the running time of the circulation pump corresponds to the running time of the heat pump (i.e. in the case of room thermostat, best case consideration) or the circulation pump runs through the entire heating period (in the case of heating curve, worst case consideration). Both values and the consequences for the resulting SPF are given in the Tab. 17.

Summarising, even though the single values differ in some cases, the SPF values for the space heating operation mode vary in the range of 7% which is in the range of the exactness of the measured COP-values from standard testing according to EN 14511.

3.3.2.3 Comparison of calculated values to field measurements for the DHW operation

Tab. 18 presents the results for the domestic hot water operation mode. The compact unit has an integrated domestic hot water storage. The heat pump operation is switched from the heating to the DHW system in case of demand, so the compact unit is of alternate combined operation type.

Since no measurements according to EN 255-3 were available, the heating characteristic has been evaluated at a yearly-average supply temperature for the heat pump derived from the measurements in order to take into account the interaction between the heat pump and the DHW storage. DHW storage losses, which are also derived from EN 255-3 testing, have been set to the measured values. The calculation has been performed using the same three operating points as for space heating and the fourth operating point at 20°C for the summer operation.

DHW consumption differs significantly in the single bins, i.e. the DHW consumption is not directly dependent on the bin time. However, for the validation, the time based redistribution of the total DHW heat requirement has been taken, so the redistribution differs from the monitored heat.

Nevertheless, the monitored and the calculated **SPF-HP** are almost equal.

The monitored **back-up energy input** could not be calculated due to the different heating capacity, which already produced less back-up energy for the space heating mode. Back-up energy due to and operation limit temperature of the heat pump was not found in the monitoring, since hot water temperature (max. 55°C) are below the operation limit. Moreover, about 25 kWh of the back-up operation is due to a large DHW load, where either a lack of capacity of the heat pump nor the operation limit of the heat pump is reached, but the back-up is switched-on after a time delay of 90 minutes to support the reheating of the DHW after large tapplings. Since these fractions are strongly control dependent, they cannot be taken into account in the calculation.

Tab. 18: Results of the calculation for the pilot plant Gelterkinden in domestic hot water mode

	Monitoring (Reference)	Calculation	Difference
System boundary heat pump			
DHW energy requirement [kWh]	1179	1179	Set input value
DHW storage losses [kWh]	1070	1070	Set input value
Evaluation time [h]	8760	8760	-
Heating period [h]	5196	5196	-
Electrical energy heat pump [kWh]	715	729	2.0%
SPF-HP by weighting over bins [-]	3.06	3.09	1.0 %
System boundary SPF-G			
Electrical back-up energy input [kWh]	63	0	-
Running time of the heat pump [h]	609	521	- 16.1%
Auxiliary stand-by electricity [kWh]	30	30	3.3 %
Electrical energy “generator” [kWh]	808	759	-6.1 %
SPF-G [-]	2.78	2.96	6.5%
System boundary SPF-S			
Auxiliary energy storage loading pump [kWh]	20	17	-14.5 %
Electrical energy “system” [kWh]	828	776	-6.3%
SPF-S [-]	1.42	1.52	7.0 %

Running time differs due to the detected difference in the heating capacity. However, the difference is so small, that stand-by electricity, which has been accounted to the DHW operation during summertime, is not affected.

For the **generator seasonal performance factor SPF-G** a difference of 6.5% results.

Auxiliary energy consumption for the loading of the storage is lower than in the measurements due to the difference in running time.

The SPF-S shows a difference of 7.0% between the calculated and the measured values.

Note that the small absolute value of the monitored SPF-S is due to very low DHW consumption and consequently the high fraction of storage stand-by losses, as can be concluded by the comparison of SPF-G and SPF-S value.

Tab. 19 gives the overall seasonal performance factor for the space heating and DHW operation, which is calculated by weighting the SPF with the respective energy fractions.

Tab. 19: Key values and comparison to monitored characteristic numbers of the pilot plant Gelterkinden equipped with a compact unit

HP compact unit Gelterkinden, BL	Monitoring (Reference)	Calculation	Difference
SPF-HP	3.59	3.66	1.9%
SPF-G	3.38	3.58	5.7%
SPF-S	3.04	3.12	2.7%

The comparison shows that maximum deviations of the overall seasonal performance of monitoring and calculation are in the range of 6%.

3.3.3 Validation for ground-source brine-to-water heat pump

The dwelling in Grafstal, canton Zurich, is equipped with a ground source brine-to-water heat pump. The system contains a DHW-storage in parallel connected to the heat pump as depicted in Fig. 17 left hand side, so the heat pump operates in alternate combined operation for space heating and DHW production. The system does not include a back-up heater, i.e. is operated monovalently. The field measurements took place from autumn 1998 until May 2000, the considered period for the validation is May 1999 to May 2000. The basic design parameters are summarised in Fig. 44. A detailed description of the result of the field monitoring is given in [39].

Evaluation period: May 1999 – May 2000

Design heat load	5.3 kW
Outdoor design temperature (θ_{OD})	-11°C*
Indoor design temperature	22°C
Supply temperature at θ_{OD}	34°C
Upper temperature limit for heating	15°C
Supply temperature at heating limit	30°C



Fig. 44: Design parameter of the NOAH system house in Grafstal, ZH (* lower value due to light construction, θ_{OD} of the site is -8°C)

Tab. 20 gives the characteristic of the heating capacity and the COP of the installed heat pump.

Tab. 20 Heat pump characteristic of the ground source heat pump in Grafstal, ZH, according to measurements of WPZ Töss [38]

	B-5/W35	B0/W35	B5/W35	B-5/W50	B0/W50	B5/W50
Heating capacity [kWh]	4.23	4.87	5.63	3.93	4.59	5.28
COP [-]	4.00	4.56	5.21	2.50	2.93	3.35

Fig. 45 shows the dependency of the brine temperature on the outside temperature at operated source pump in hourly resolution. To characterise the dependency, a quadratic fit depicted in Fig. 45 has been used.

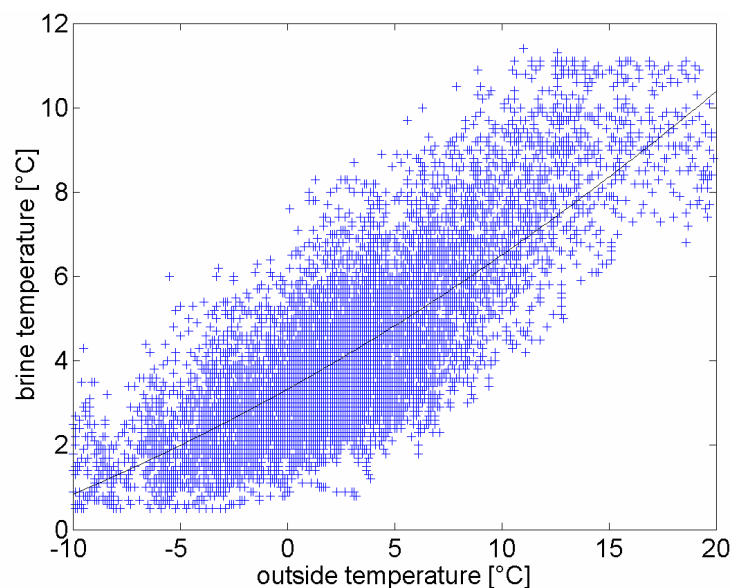


Fig. 45: Dependency of the brine inlet temperature on the outside air temperature

3.3.3.1 Comparison of calculated values to field monitoring for the space heating operation

For the validation, three operating points have been used. A comparison of the respective energy fractions of the bins measured in the monitoring and calculated based on the heating degree hours are presented in Fig. 46.

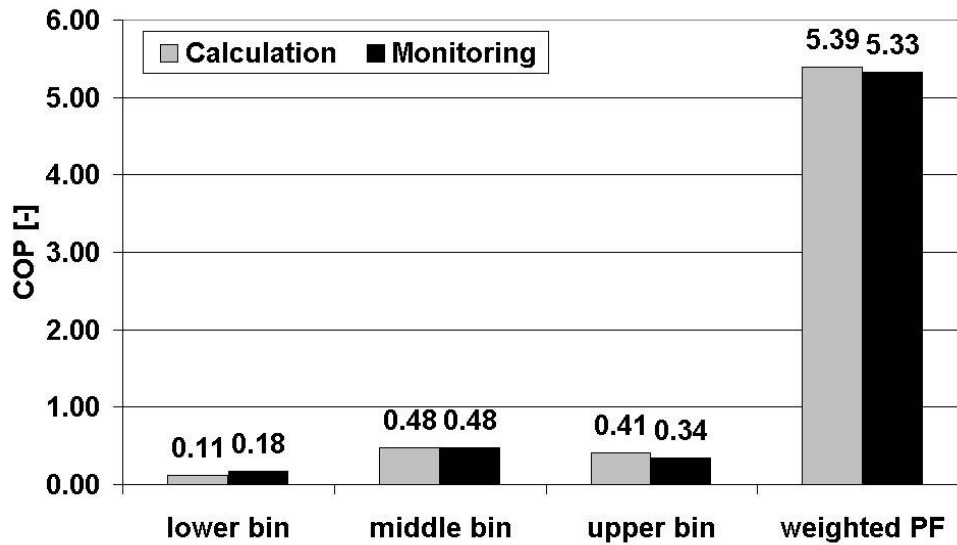


Fig. 46: Weighting factor in the bins and weighted performance factor in monitoring and calculation

Tab. 21 gives the results of the validation for the respective system boundaries SPF-HP, SPF-G and SPF-S for the heating operation.

Tab. 21: Results of the calculation for the pilot plant Grafstal in heating mode (* calculated value)

	Monitoring (Reference)	Calculation	Difference
System boundary heat pump			
Heating energy requirement [kWh]	10423	10423	Set input value
Losses distribution system [kWh]	336	336	Set input value
Evaluation time (heating period) [h]	5875*	5650	-3.8 %
Electrical energy for heat pump [kWh]	2030	1935	-4.7%
SPF-HP by weighting over bins	5.33	5.39	1.1 %
SPF-HP monitoring	5.13	-	5.1 %
System boundary SPF-G			
Running time of the heat pump [h]	2096	1873	-10.6 %
Energy for the source pump [kWh]	252	225	-10.7 %
Energy for control in stand-by [kWh]	53	53	- %
Electrical input "generator" [kWh]	2335	2213	-5.2 %
SPF-G	4.46	4.71	5.6 %
System boundary SPF-S			
Used heat for space heating [kWh]	10087	10087	Set input value
Energy for the circulation pump [kWh]	256	246	-3.9 %
Electrical energy "system" [kWh]	2591	2459	-5.1 %
SPF-S	3.89	4.10	5.4 %

Due to the monitoring, the system boundary includes the distribution losses of the system. The length of the heating period in the monitoring was evaluated by the running time of the heating circulation pump which runs through, thus it is slightly different to the calculation based on the evaluation of the outside temperature.

By weighting the COP-characteristic with the weighting factor of the cumulative annual frequency an averaged **SPF-HP** of 5.39 is derived. If weighting factors based on the measured energies are evaluated, the weighted SPF-HP is 5.33, so almost the same. However, the SPF-HP calculated as ratio of produced heat and consumed electricity delivers a value of 5.13, which means a difference of the measured characteristic and the operation of the heat pump of 5.1%. Reasons for the deviation are an uncertainty of the source temperature, which is fitted and thus is an averaged value, see Fig. 45, and the measurement exactness of the COP characteristic itself.

This tendency can be seen in the heating capacity, too, since the **running time** of the heat pump is longer in the monitoring, i.e. the average heating capacity is lower than expected by the characteristic. The difference in the running time also explains the difference in the electricity consumption of the source pump, since the operation of the source pump is coupled to the operation of the heat pump.

Control energy also differs slightly due to the different stand-by periods in the calculation and the monitoring, partly compensated by the different heating periods. Results for the system boundary SPF-G of the generator is 5.2% less electricity consumption and thus a 5.6% higher seasonal performance factor.

The **circulation pump** runs through the whole heating period, so the difference in the length of the heating period can be seen in the deviation of the electricity consumption of the pumps.

3.3.3.2 Comparison of calculated values to field monitoring for the DHW operation

Tab. 22 presents the results of the domestic hot water operation. The heating characteristic has been used for the DHW operation, too, since no measurements according to EN 255-3 were available. As in the calculation of Gelterkinden, for the domestic hot water operation 4 bins have been used. In the same way, a mean supply temperature has been evaluated from the measurements. Additional to the DHW storage losses, the losses in the distribution system are taken into account in the monitoring.

Tab. 22: Results of the calculation for the pilot plant Grafstal in domestic hot water mode

	Monitoring (Reference)	Calculation	Difference
System boundary heat pump			
Energy requirement [kWh]	2267	2267	Set input value
DHW storage losses [kWh]	368	368	Set input value
Losses of the distribution system [kWh]	394	394	Set input value
Electrical energy heat pump [kWh]	647	625	-3.4 %
SPF-HP	3.50	3.63	3.7 %
System boundary SPF-G			
Stand-by electricity for control [kWh]	34	38	-11.8 %
Running time of the heat pump [h]	481	416	-13.5 %
Energy for the source pump [kWh]	58	50	-13.8 %
Electrical energy "generator" [kWh]	739	713	-3.5 %
SPF-G	3.07	3.18	3.6 %
System boundary SPF-S			
Tapped DHW energy [kWh]	1505	1505	Set input value
Energy for the circulation pump for storage loading[kWh]	21	18	-14.3 %
Electrical energy "system" [kWh]	760	731	-3.8 %
SPF-S	1.98	2.06	4.0 %

The **SPF-HP** values for the DHW operation deviate 3.7% due to the same effects as in the heating mode. So does the **running time**, causing the different electricity consumption for the **source and the circulation pumps**. In the alternate combined DHW operation, the storage loading is carried out with the circulation pump, so the operation of the circulation pump is coupled to the running time of the heat pump, too.

Respective **SPF-G** values differ due to the different running time by 3.6 % and the **SPF-S** by 4.0%.

Tab. 23 gathers the overall seasonal performance of the monitoring and the calculation.

Tab. 23: Key values and comparison to monitored characteristic numbers of the pilot plant Grafstal equipped with a brine-to-water heat pump

B/W-Heat Pump Grafstal, ZH	Monitoring (Reference)	Calculation	Difference
SPF-HP	4.74	4.96	4.6%
SPF-G	4.13	4.34	5.1%
SPF-S	3.46	3.63	4.9%

3.3.4 Conclusion of the validation of the calculation of the SPF

Different combined operating heat pump systems for space heating and domestic hot water have been calculated and results have been compared to results from field monitoring performed in the IEA HPP Annex 28 and based on existing data of pilot plant measurements. Results are in the range of $\pm 6\%$, i.e. both a deviation to higher and lower than the monitored SPF-values are calculated.

In the case of ground-source heat pump, the source temperature has to be evaluated in dependency of the outside temperature, e.g. by fitting of measured or simulated data or based on typical profiles, thus temperature condition are less precise. On the other hand, the range of source temperatures is not that large as in case of air-to-water heat pumps. Thus, both types show a similar deviation between measured and calculated performances.

In the case of DHW-operation, an interaction between the heat pump and the storage exists and the average temperature for the DHW-production is influenced by the tapping volume and the behaviour of the storage. Thus, temperature conditions are less obvious. System testing gives more insight in this interaction. Since no values from the system testing were available for the monitored systems, values from the component-based testing of the space heating-only operation have been used for the calculation, which delivers feasible results for alternate operation, since temperature conditions have been approximated from the measured values. However, for simultaneous systems, probably a system testing will be required due to a more complex interaction between heat pump and storage.

Unfortunately, no evaluation of simultaneous combined operation could have been performed in the timeframe of the IEA HPP Annex 28 and no combination of testing according to the proposed test procedure and a subsequent calculation according to the proposed calculation method based on this tested characteristics could have been accomplished in the Annex 28. This will be a future task.

Summarising, the result of the calculation with a deviation of $\pm 6\%$ of seasonal performance is satisfactory with regard to the exactness of the measured COP-values, which hold uncertainties in the same range. Moreover, the calculation method implies certain assumptions and simplifications in order to keep the calculation simple in the sense of a "hand" calculation. Thus, the resulting exactness of the validation is in the range of an exactness to be expected by a "hand" calculation without extensive computation.

Shortcomings of the method are that control effects can hardly be taken into account in detail, which is shown in the comparison in case of the back-up energy and the auxiliary energy for pumps and fans, which may deviated from values calculated by the simplified approaches.

However, the different impacts on the performance can be evaluated and comparison under different boundary conditions can be accomplished.

3.3.5 Results for field monitoring of compact units

Germany has monitored two compact units, one installed in a single family house in Herzhausen and one installed in a flat in Gießen. Both buildings are built according to the German passive house standard (<http://www.passiv.de>). In the following, the two pilot plants and the results from the field monitoring are presented. A validation in form of comparison with calculated values has not been carried out in the timeframe of the IEA HPP Annex 28.

3.3.5.1 Pilot plant in Herzhausen

The pilot plant in Herzhausen, Hesse, Germany, is a single-family house in light-weight construction according to the German passive house standard.

Fig. 47 contains information on the parameters of the pilot plant. The pilot plant is equipped with the heat pump compact unit operating on an air-heating system and a vacuum tube collector.

Evaluation period:

Design heat load	10 W/m ²
Outdoor design temperature	-14°C
Heating curve (at θ_{OD} , at θ_{th})	Load dependent
Design heating energy demand	14 kWh/(m ² a)
Surface	180 m ²
Surface-volume ratio	0.4
Ground-to-air heat exchanger	2 x 30 m
Vacuum tube collector	3 m ²



Fig. 47: Single-family house in Herzhausen, equipped with the compact unit

Characteristics of the compact units are given in Tab. 24.

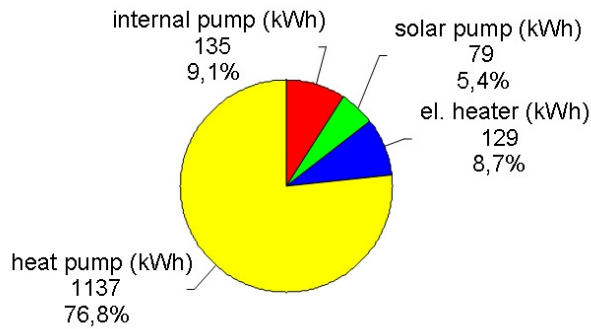
Tab. 24: Characteristic of the heat pump compact unit according to test points of the DIBt (total air flow rate (outside air and heat recovery) of 250 m³/h)

Mean air flow rate heat recovery [m ³ /h]	Outside air temperature [°C]	Supply temperature [°C]	Output capacity [kW]	COP [-]
89	-3	35	1.1	2.5
89	4	35	1.3	2.3
89	10	35	1.7	3.1
135	-3	35	1.5	2.7
135	4	35	1.5	2.8
135	10	35	1.9	3.3

The thermal and electrical energy consumption for space heating and DHW production during the evaluated period is shown in Fig. 48.

Concerning the heat energies, the fraction of the electrical back-up heater with 3.2% of the space heating and DHW is very small. Most of the heat is delivered by the heat pump, however, a solar fraction of about 23% contributes to the overall heating requirement.

electric energy consumption



heat delivered

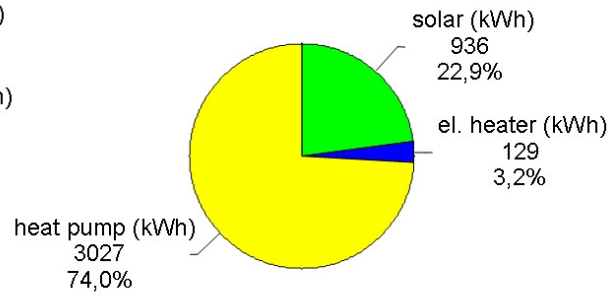


Fig. 48: Electric energy consumption and delivered heat of the pilot plant in Herzhausen

The overall SPF of the heat pump (heating and DHW) evaluated without the ventilation heat recovery is 2.7. Further characteristic number for different system boundaries are given in Tab. 25.

Tab. 25: characteristic numbers of the compact unit in Herzhausen, Hesse, Germany

HP compact unit Herzhausen	SPF-HP	SPF-G	SPF-SYS	ETV heat recovery	ETV solar system
Year	2.7	2.8	4.2	8	12

3.3.5.2 Pilot plant in Gießen

The pilot plant in Gießen, Hesse, Germany, is a flat in a multi-family house in standard construction according to the German passive house standard. Fig. 49 contains information on the design parameters of the pilot plant. The pilot plant is equipped with the heat pump compact unit Vitotres 343 operating on an air-heating system and a vacuum tube collector.

Evaluation period: May 2004 – May 2005

Design heat load	10 W/m ²
Design outdoor temperature θ_{OD}	-14°C
Heating curve (at θ_{OD} , at θ_{tth})	Load dependent
Supply temperature at θ_{OD}	Load dependent
Design Heating energy demand	15 kWh/(m ² a)
Surface	70 - 120 m ² /flat
Flat plate collector	~10 m ² /3 flats



Fig. 49: Flat in a multi-family dwelling in Gießen, equipped with the compact unit

Concerning the heat energies, the fraction of the electrical back-up heater is about 8% of the space heating and DHW is small. Most of the heat is delivered by the heat pump. The solar system contributes with about 9% to the overall heating requirement. The required electricity expense of only 41 kWh for the solar system is quite small leading to an electro-thermal amplification of about 7. The overall SPF of the heat pump evaluated without the ventilation heat recovery is 2.4. Further characteristic numbers for different system boundaries are given in Tab. 26.

Tab. 26: characteristic numbers of the compact unit in Gießen, Hesse, Germany

HP compact unit Gießen	SPF-HP	SPF-G	SPF-SYS	ETV heat recovery	ETV solar system
Year	2.5	2.1	2.9	8	6.9

The thermal and electrical energy consumption for space heating and DHW production during the evaluated period is shown in Fig. 50.

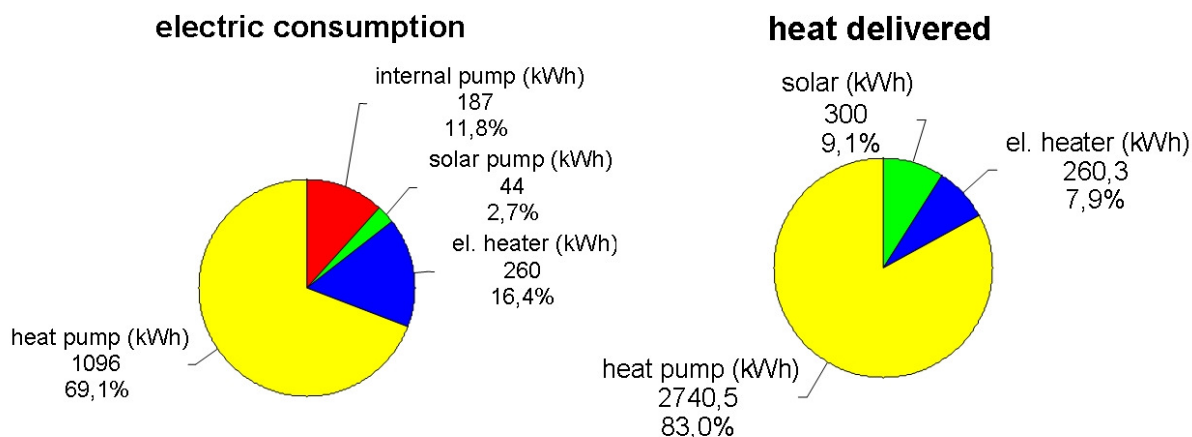


Fig. 50: Electric energy consumption and delivered heat of the pilot plant in Gießen

3.4 Discussion of comments on the part integrated in the CEN draft standard

The proposal of the calculation method has been already included in the CEN draft standard of prEN15316 part 4.2 in the timeframe of the IEA HPP Annex 28.

The CEN draft has been sent to a six-month public enquiry in the time of October 2005 – March 2006. This chapter gives the most important feedback from the public enquiry and includes as well comments from within the Annex 28 working group.

Most of the comments have already been updated in the proposal of IEA HPP Annex 28 for the calculation method, but other that require further investigation, can be updated while preparing the draft for the formal vote.

3.4.1 Use of heating degree hours and upper temperature limit for heating

A comment on the calculation method recently received refers to the utilization of heating degree hours and an upper temperature limit for heating, since in the calculation method for the building energy according to EN 832 or the follow-up standard EN ISO 13790, no upper temperature limit for heating is calculated anymore. So, the calculation methodology does not fit to the standard calculation.

The EN ISO 13790 calculates a monthly energy balance of the building resulting in monthly heat losses of the building due to transmission and ventilation and monthly heat gains due to internal (persons, lighting, equipment) and solar gains. For the utilization of the gains a utilization factor based

on a gain – loss ratio is calculated, taking into account the time constant of the building, i.e. the thermal mass of the building.

As a result of the monthly energy balance the heating requirement is available in a monthly resolution. It can be summed-up to an annual heating energy requirement of the building. Therefore, no upper temperature limit for heating is necessary for this calculation. In particular, the whole calculation is based on time-scale and not on temperature-scale (monthly values of the energy demand, which have no direct correlation to the outside temperature conditions).

Heating degree hours do not correspond to the calculation of the energy requirement, either, since the heating degree hours are directly related to the outside temperature and therefore do not take into account solar and internal gains. Therefore, they refer to a standard heat demand used for dimensioning of the heating capacity (power) of the heat generator, which, by definition, does not take into account gains, since the objective of the calculation is the design for the highest possible demand.

The system boundary in the framework of the EPBD shown in Fig. 23, which has also been used in the IEA HPP Annex 28, has as input value the requirement of the distribution system, which corresponds to a net heating requirement, so solar gain and internal gains are already subtracted. In fact, it corresponds to the building energy requirement according the EN ISO 13790 plus the losses of the emission and distribution system. Thus, the energy requirement is taken into account correctly. However, since heating degree hours are used for the redistribution of the requirement to the bins as an approximation, this approach may not take into account the redistribution of solar and internal gains correctly. This assumption may become the worse the higher the fraction of used solar and internal gains is, i.e. in low and ultra-low energy dwellings.

For an exact calculation, the heat demand, i.e. the thermal power dependent on the outside temperature would have to be known, since then an integration of the thermal power demand could be performed resulting in an exact distribution of the heating requirement for the heat pump. However, these values are not easy to evaluate. Actually, this power demand can hardly be determined by a "hand" calculation method for a standard, since solar and internal gains and their utilization would have to be calculated in high resolution.

Nevertheless, a correction of the cumulative frequency could be considered using information of the monthly calculated used gains of the building energy calculation. One approach for the correction is depicted in Fig. 51.

The monthly gains g delivered by EN ISO 13790 can be transformed in a temperature difference by the following approach:

The monthly losses l correspond to a temperature difference from indoor design temperature to outdoor air temperature. Therefore, the gains g correspond to a temperature difference, as well (see left part in Fig. 51), which can be subtracted from the loss temperature, i.e. the outdoor air temperature. The resulting temperature is called energy air temperature and is higher than the outdoor air temperature. Therefore, the effective heat demand d of the building can be considered as difference of transmission and ventilation losses and used solar and internal gains, i.e. as the temperature difference from indoor design temperature to energy air temperature. The assignment of the energy air temperature to the cumulative hours is accomplished by the monthly average temperature, i.e. the respective energy air temperature of the month is located at the cumulative number of hours of the monthly average temperature of the respective month. The resulting interpolated cumulative frequency could look like the line d in Fig. 51.

However, the resolution of the transmission and ventilation losses on the one hand and the used solar and internal gains on the other hand is monthly, and moreover, the monthly average temperature used as basis for the calculation is usually not lower than about 0°C for moderate climates. Therefore, the redistribution in the lower temperature range can often not be evaluated and the total resulting redistribution is limited by the monthly resolution.

Therefore, a closer evaluation is required to assess the error that is made by a redistribution according to the heating degree hours and the possible improvements e.g. by the described correction. Further information is contained in the Swiss country report [37].

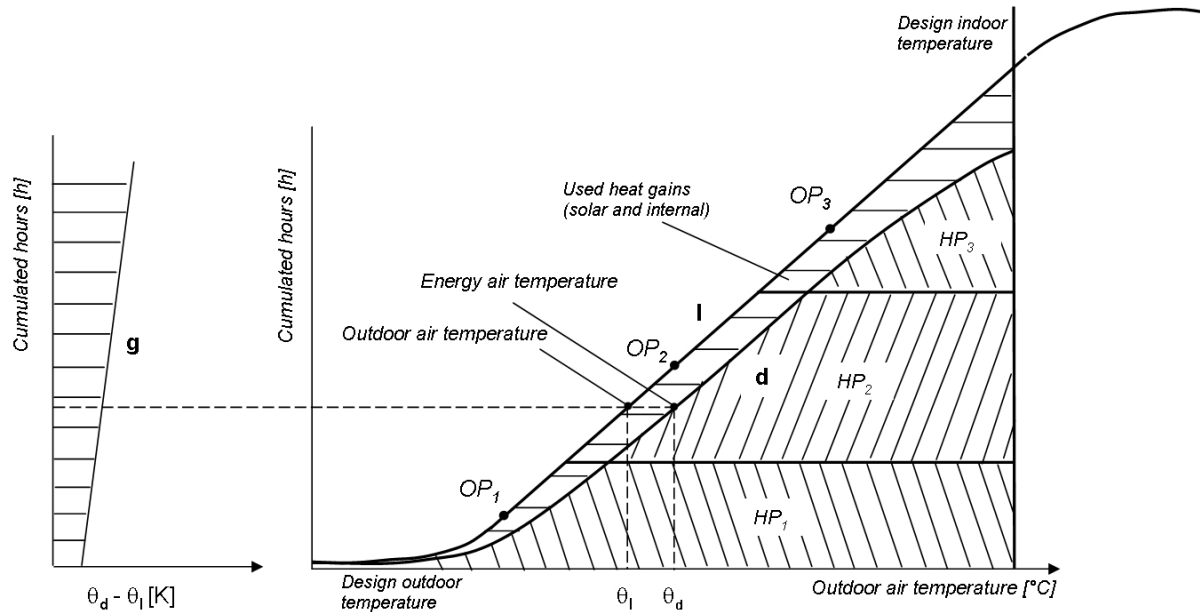


Fig. 51: Correction of cumulative annual frequency with information of the gains calculated in EN ISO 13790 (modified based on [36])

3.4.2 Calculation of the back-up heating

The calculation of the back-up heating implemented in the CEN draft considers the back-up operation based on the input of the balance point. Based on the balance point, the area under the cumulative frequency is evaluated depending on the operation mode of the back-up heating as described in part 1, chapter A 1.5.5. For the domestic hot water, only the back-up requirement due to a temperature limit for the operation of the heat pump is taken into account and not for a lack of heating capacity. This would refer to a constellation, where the heating capacity requirements of the heat pump for heating and domestic hot water are considered in the calculation of the balance point and the control has a domestic hot water preference, i.e. the heating capacity of the heat pump is used to cover the domestic hot water requirements and in case of a lack of capacity, the heating requirements are covered by back-up operation.

However, this approach may be too specific considering the wide range of possible system configurations and an operation of the back-up heater to cover a lack of capacity for the domestic hot water should be integrated.

This could be done, for instance, by an evaluation of the required running time of the heat pump to cover the space heating and domestic hot water requirement in the bin. If the required running time in the bin is longer than the available bin time, the back-up heater has to cover the missing heating capacity of the heat pump. The running time can be evaluated based on the energy requirement and the heating capacity at the respective conditions. To extend the exactness for the back-up calculation, it could be useful to evaluate this balance of the running time in 1 K steps based on the data of the cumulative frequency of the outside temperature until an outside temperature level is reached, at which the heat pump heating capacity is big enough to cover the requirements without back-up operation.

3.4.3 Use of default values

The implemented standard for the heat pump calculation contains default values, which have been proposed by further countries working in the CEN working group. In European standardisation the calculation shall be possible under all circumstances, even when no data of the heat pump are

available. However, default values should refer to lower performance level on the market. In this way, the calculation will deliver worse values than calculated with the measured values, so there is a motivation for the user to take the measured values, which will deliver better results.

A general comment and opinion is that the possibility to use default values undermines the use of correct values from laboratory measurements. If the thought behind this is to have different classes for different fields of use it should be clearly stated in the method description. Otherwise, it is questionable whether it is meaningful to develop a detailed test and calculation method if one can use default values instead. Actually, the objective of Annex 28 has been to deliver a transparent calculation method. Therefore, in part 1 of the report the use of default values has been minimized.

3.4.4 Use of typology method

Moreover, a simplified seasonal performance method based on system typologies has been integrated in the CEN draft standard. For the simplified calculation the considered period is the heating season. The performance is chosen from tabulated values for two performance classes of the heat pump based on test results according to the heat pump testing standard EN 14511. The operating conditions (climate, design and operation of the heating system, heat source type) are based on typology of implementation characteristics (climate, design and operation of the heating system, heat source type) and are not case specific, i.e. the evaluation of the seasonal performance factor only considers two classes of supply temperatures (35°C, 45°C), the type of heat pump and the two classes of COP-values. The method is thus limited by a country/region specific approach and requires country/region specific tabulated values.

Even though the method may fit quite well for a general market average of installed systems, it is not transparent, how values are calculated and boundary conditions are not comprehensive to cover different system configurations.

Proposal:

Calculation of heat pump systems shall be accomplished case specific according to the proposed calculation outlined in part 1 of the final report based on standard test results according to the proposed test procedures. This enables a transparent and fair comparison of different heat pump systems on the market and different boundary conditions.

3.4.5 Use of hourly values

The bin method was intended as a hand calculation method and is specified based on annual or monthly energy values as input. There was a demand to extend the scope to an hourly calculation. This is possible, however, looking at the results of the validation described in chapter 3.3 and taking into account that measured COP values have an exactness in the range of 5%-10%, a monthly or even an annual method seems sufficient with regard to the computational expense for an hourly method.

3.4.6 Application of the calculation method for variable speed units

In the calculation approach, part load losses due to cyclic operation in ON/OFF mode are considered small and are neglected, see part A1 chapter 1.5.6.4 and result of cyclic testing described in chapter 3.1.4. Therefore, the difference to variable speed units is basically the COP characteristic. Thus, if the unit is tested according to EN 14511, the characteristic can be applied in the same way to perform the calculation method, possibly with an additional input of the load factor to determine the adequate operating condition of the variable speed unit. The load factor can be evaluated by the ratio of the mean heat load in the bin and the design output capacity of the heat pump.

3.5 Proposal for test procedures

As described above, space heating-only operation and DHW-only operation modes can be tested by existing standards.

3.5.1 Testing of alternate combined operation

The approach for the alternate combined operation refers to the testing of the space heating-only and DHW-only modes. Comparison of the testing of combined operation and the testing of each single operation mode revealed that the weighting of the single COP-values with the energy requirements delivers good agreement with the corresponding COP in combined operation at the same ratio $X = Q_{out,g,h}/Q_{out,g,DHW}$ [17]. In Fig. 52 and Fig. 53 the measured and weighted values for different X ratios and different sink temperatures of the heating system are shown for an alternate operating brine-to-water heat pump. $COP_{hps,tot}$ refers to a COP of the heat pump system which is not corrected with pump and fan efficiencies, $COP_{t,s}$ refers to a COP for domestic hot water which includes the stand-by power

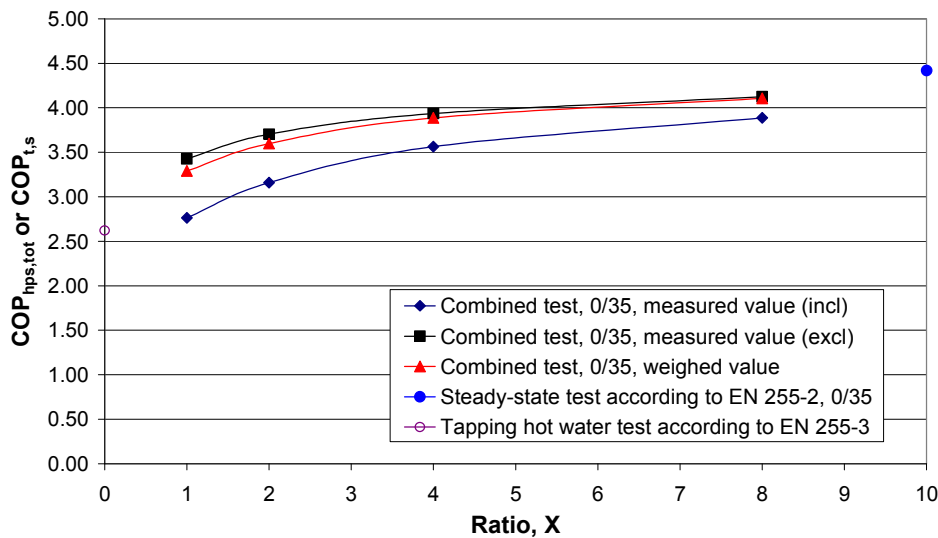


Fig. 52: Measured and calculated COP-values for alternate combined operating systems (source [17])

In conclusion, the testing of the space heating-only and DHW-only operation modes is sufficient for alternate combined operating systems, as already proposed in the calculation method described in chapter 3.2.4.3 [14].

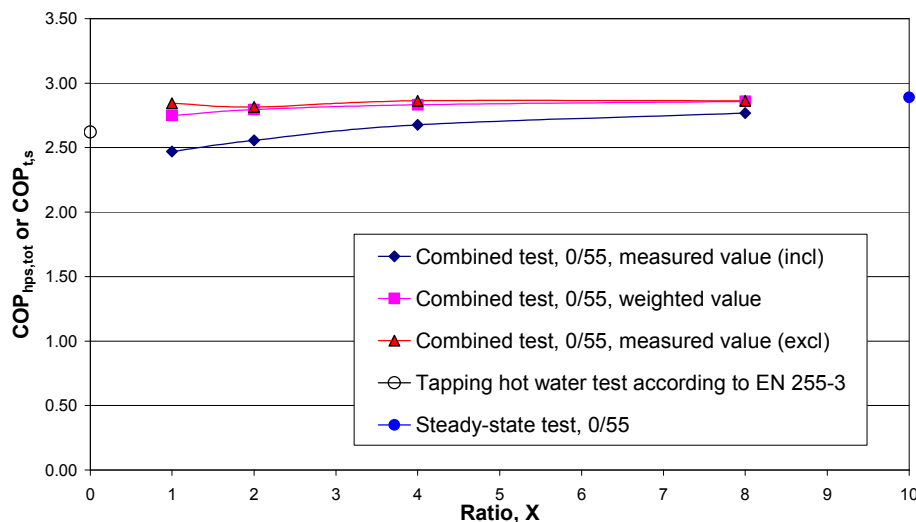


Fig. 53: Measured and calculated COP-values for alternate combined systems (source [17])

3.5.2 Testing of simultaneous combined operation

To treat the simultaneous combined operation the testing according to the standards for space heating and DHW is performed at the same time, i.e. the hot water tapplings according to the EN 255-3 are performed during the space heating operation. The test procedure and the test points should thus be based on the space heating-only and DHW-only operation modes.

Referring to a black-box configuration described in chapter 2.5.1, an essential problem occurs for the evaluation: only the total consumption of electricity can be measured, but this total electricity input can hardly be accounted to the space heating-only and DHW-only heating requirements. As described in chapter 3.2.4.3 this problem is solved by evaluating only a combined operation mode without allocation of the electrical energy input to the single building needs.

Furthermore, the above described experiences with the application of the DHW standards shall be considered and shortcomings are to be improved. The definition and evaluation of test points is determined by the chosen calculation method.

Like for the calculation method, the different proposals are presented shortly in the following and subsequently the resulting test procedure and the proposed changes to existing standards are described.

3.5.2.1 Test procedure with regard to the evaluation of characteristic days

This test procedure has been proposed by the Japanese team in compliance with the Japanese test standard JRA 4050:2004 [23].

If the proposal for the Japanese calculation method based on typical days described in chapter 3.2.4.1 is applied, the respective test conditions (temperatures, profiles etc.) are defined by the characteristic of the day. The tapping profiles of space heating and DHW used for the testing is given Fig. 54. Fig. 54 shows that the amount of DHW, e.g. the profile itself, is kept constant for the seasons. However, the cold water supply temperature changes between the seasons.

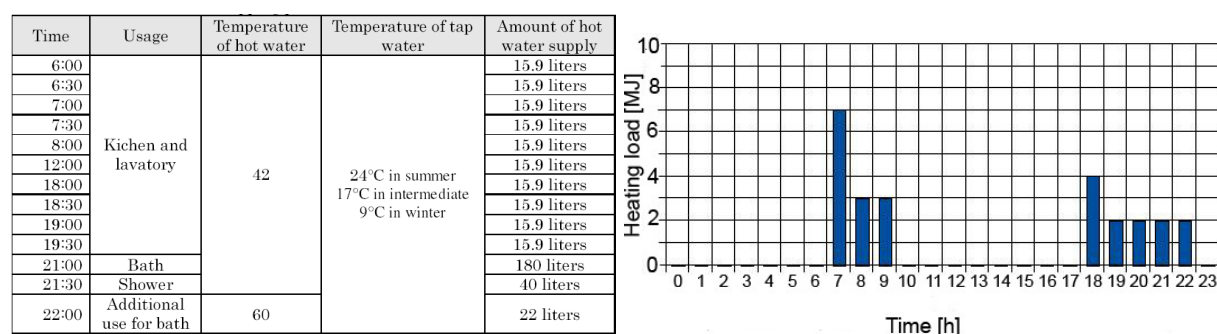


Fig. 54: Tapping profiles (left) and space heating load profile of an average day (right) of the Japanese proposal for the test procedure (source [23])

3.5.2.2 Evaluation of combined operation with regard to the ratio $X = Q_{out,g,h}/Q_{out,g,DHW}$

In this case testing has to cover different ratios of the demand for space heating and DHW at each of the respective test points, since an evaluation of the space heating-only mode in the calculation does not take place (see description in chapter 3.2.4.2).

Combined testing shall follow the testing in the single modes, but the DHW tapplings are accomplished during space heating operation.

In order to provide COP-values for the different ratios $X = Q_{out,g,h}/Q_{out,g,DHW}$, the testing shall be carried out until a ratio of 8 is reached. The evaluation of the recorded measurements is done for different time spans to receive the respective ratios $X = Q_{out,g,h}/Q_{out,g,DHW} = 1;2;4;8$. To reduce testing time, the test procedure depicted in Fig. 55 is proposed [17]. $P_{H, tot}$ refers to the total heating capacity of the heat pump for space heating, which is not corrected for pump or fan efficiencies, P_t to the thermal heating capacity of domestic hot water, $t_{w,in}$ and $t_{w,out}$ are the respective temperatures of the heating water for space heating at in-and outlet of the heat pump. Testing is performed continuously and the different ratios of DHW to SH consumption, indicated by the flags in Fig. 55 are evaluated afterwards. However, the total testing time depends on the energy content of the hot water, which depends on the storage size.

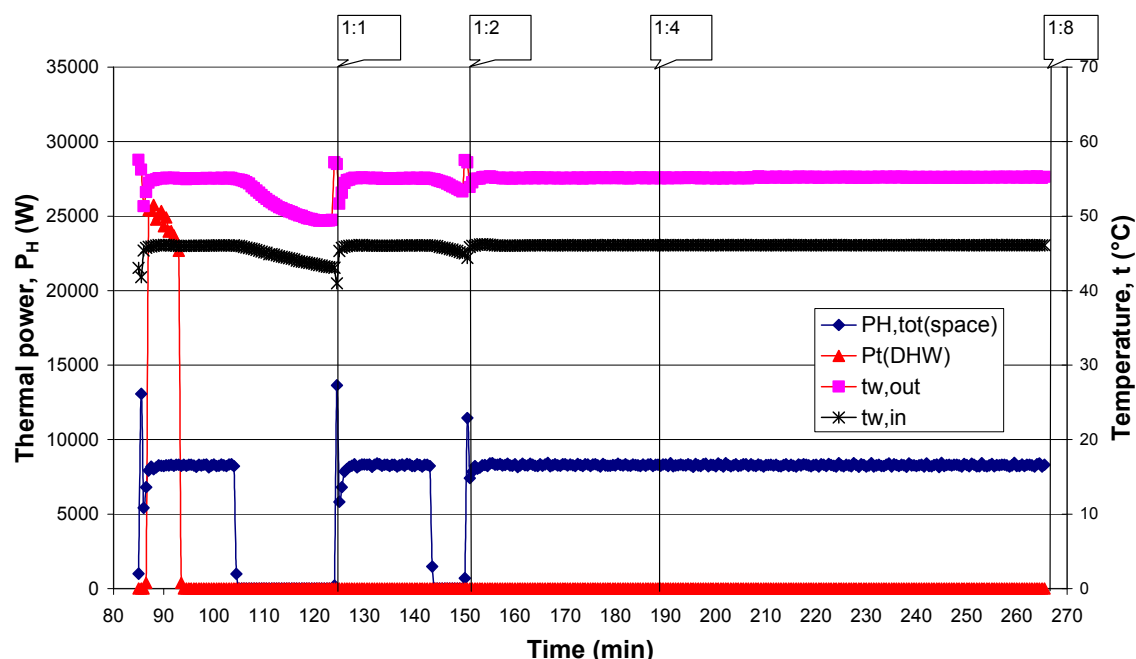


Fig. 55 Test cycle for simultaneous operation with evaluation of ratio $X = Q_{ou,g,h}/Q_{out,g,DHW}$ (source [17])

3.5.2.3 Test procedure for the separation of SH and DHW operation

If the calculation is performed implying the separation of the SH and DHW mode, additional testing for the simultaneous operation of the standards is required in the same way as in the proposed Swedish method, i.e. the hot water tapping pattern during space heating operation. However, the EN 255-3 cycle does not have to be changed, since running time is evaluated during the calculation to characterise the simultaneous operation.

3.5.2.4 Discussion of the different approaches for testing

Since the Swedish proposal is outlined quite detailed it is decided to follow the Swedish testing approach. However, due to the decision on the calculation method (see chapter 3.2.5), it is possible to shorten the testing time, since no mandatory testing of different ratios $X = Q_{out,g,h}/Q_{out,g,DHW}$ has to be considered, as the fraction of combined operation is determined by evaluating the running time.

Proposal:

Testing shall follow the test procedure described by Sweden [17], but regardless of the ratio $X = Q_{out,g,h}/Q_{out,g,DHW}$ to minimize the required testing time.

In the following the changes proposed by IEA HPP Annex 28 for existing and proposed standards and the extension for the combined operation are presented.

3.5.3 Proposed modification for testing in the heating mode (EN 14511)

3.5.3.1 Setting of the mass flow rate

In EN 14511 the mass flow rate is differentiated dependent on the type of heat emission system. However, Swedish experience described in chapter 3.1.1 above proves little influence on the COP. The mass flow is set at the test points according to Part 1, chapter B of the final report.

Proposal:

One mass flow rate is to be used for all types of heat emission systems at all operating conditions, i.e. heat source/sink temperature

For the setting of the mass flow, a rating point has to be fixed, so the rating points for floor heating at 35°C and the rating point for radiator heating at 45°C according to EN 14511-2 can be chosen. Arguments for the respective rating points are

- Radiator heating systems may have the larger market perspective due to the developments of retrofit heat pumps.
- However, a lot of heat pumps are used with floor heating due to favourable temperature conditions.
- Testing is performed at 35°C, 45°C and 55°C, so 45°C is in the centre of the test points.
- The “D-A-CH Gütesiegel”, a quality label of Germany, Austria and Switzerland, has chosen the rating point 35°C, which might be transferred to the European Heat Pump Association (EHPA) as basis for a European quality label.
- In the English version of EN 14511-2, only the rating point 45°C is given for exhaust air heat pumps. However, in the French and German version, the rating point at 35°C is included, as well. Thus, it is not clear, which version is the right and if the rating point at 35°C exists for exhaust air heat pumps.

Proposal:

The rating point at 45°C shall be used for setting the flow rate

3.5.3.2 Values for pump efficiency

In EN 14511 the efficiency of pumps and fans is set to 0.3 per definition. The Swedish team proposes to measure the liquid pump efficiency, and this measured efficiency shall be used for the evaluation of tested COP-values. This shall be done to take account of often poor pump efficiencies smaller than 0.1, which is by far lower than the default of 0.3 of EN 14511.

Discussion:

A measurement might be difficult, in particular in the case of the agreed black-box testing, since the electricity input might not be easily allocated to the pumps depending on the system configuration. Moreover, this topic of pump efficiencies has been discussed during the revision of the EN 14511, but without success. Nevertheless, IEA HPP Annex 28 agrees on the proposal of a lower default value for the pump efficiency of 0.1.

Proposal:

Pump efficiency may not be explicitly measured. However, a default value of 0.1 for the pump efficiency shall be introduced. It is also possible to replace the default value of 0.1, if either the pump efficiency is measured or if the manufacturer can deliver test data according to a respective test procedure.

3.5.3.3 Leakage testing for exhaust air heat pump

EN 14511 should be amended with leakage testing for exhaust air heat pumps, e.g. according to Nordtest procedure VVS 021, which is very essential in order to derive correct efficiency values from the EN 14511.

Proposal:

Leakage testing for exhaust air heat pump acc. to NT VVS 021 shall be included in the testing

3.5.4 Proposed modifications for the testing of the DHW mode (EN 255-3)

The following modifications for the domestic hot water test procedure according to EN 255-3 (see description in chapter 2.2.3) are proposed and partly necessary due to the extension to cover simultaneous combined operating systems [9].

3.5.4.1 Determination of an average heating capacity for the domestic hot water mode

By evaluation of the measurements of phase 2 of EN 255-3 an average heating capacity of the heat pump for DHW operation shall be calculated. This is just a further output value of the testing, which does not require changes in the test procedure. The heating capacity is to be calculated according to the equation

$$\phi_{\text{DHW}} = \frac{Q_t}{t_t} \quad [\text{W}]$$

eq. 20

where

ϕ_{DHW}	calculated heating capacity for DHW operation	(W)
Q_t	heat extraction by DHW tapping according to EN 255-3 phase 2	(J)
t_t	time of cycle according to EN 255-3 phase 2	(s)

Details are given in [9]. This heating capacity is necessary in case of the separation of SH and DHW operation mode in order to determine the time in the respective operation mode by evaluating the running time (see description of calculation method in chapter 3.2.4.3). Moreover, it enables the estimation of additional auxiliary energy not included in the standard testing.

Proposal:

Since no further testing is caused by determining an average heating capacity for DHW operation, and the value is necessary for the calculation method, an additional output value of EN 255-3 regarding the average heating capacity for DHW value shall be amended.

3.5.4.2 Definition of a DHW set temperature

EN 255-3 does not prescribe a DHW set temperature but manufacturer setting is used. Consequently, the delivered COP-values cannot be compared, since they might differ due to different DHW set temperature. To overcome this situation, there are basically two possibilities:

- Definition of a DHW set temperature:

With regard to legionella considerations, a prescribed DHW tapping temperature of 60°C could be a useful value. Manufacturers have to deliver their system for the testing with controller setting, which enable this required DHW temperature.

- Definition of a different characteristic value

This could be for instance an exergetic efficiency which takes into account the impact of the different DHW temperature levels [9].

Since the first option seems easier, as only one characteristic number for the heat pump operation, namely the COP_t, is delivered the idea to be discussed is to define a prescribed DHW tapping temperature.

Discussion:

- EN 255-3 actually does not prescribe a tapping temperature for DHW. On the other hand, a minimal temperature of 40°C is a lower limit to be entitled "domestic hot water". Thus, the temperature could be fixed to a required tapping temperature of 45°C, which is in compliance to other standards (boiler testing).

- This would be a disadvantage for CO₂ heat pumps, since this heat pump type can deliver easily DHW at a temperature of 60°C and above. With regard to the temperature requirement to avoid legionella, a temperature level of 60°C might be useful.
- Mandate M/324 has to be taken into account: Following the Mandate M/324, an energy content for the tapping at a reference temperature of 60°C shall be fixed (Attention: not a required tapping temperature!) and a minimal requirement to be entitled "domestic hot water" of 40°C as in EN 255-3 is left. Furthermore, a flow rate is taken from storage testing, which defines a minimum of 10 l/min (= 0.167 l/s) or 1 storage volume per hour (for larger storages).

It is proposed, not to fix the tapping temperature but to follow the Mandate M/324.

- Since it is not entirely clear, how the Mandate M/324 will be implemented, it might be necessary to change the tapping cycle to the prescribed EU reference cycles. However, a preference and recommendation of the IEA HPP Annex 28 is to simplify tapping profiles for storage systems by summing up small tapplings.

Proposal:

The energy content of the tapped hot water based on a reference temperature of 60°C shall be prescribed.

The temperature level to define "domestic hot water" is kept at 40°C according to EN 255 – 3 and Mandate M/324. However, upcoming legislation on temperature requirements with regard to legionella shall be considered and in case implemented for the prescription of a temperature.

The flow rate of the tapping shall be fixed according to the storage testing at 1 storage volume/hour, at minimum 10 l/min which equals 0.167 l/s.

3.5.4.3 Tapping pattern of domestic hot water

As it is not sure if and how mandate M/324 will be implemented it is proposed that the tapping of the DHW shall neither follow the original tapping pattern of EN 255-3 nor the tapping pattern in Mandate M/324 due to the following argumentation: Even though a sequence of a large number of tapplings at small mass flow rates can produce lower COP_t values than the two tapplings of half the storage volume, the European tapping profile are not capable to catch the user behaviour, since this differs very much between individuals and countries. On the other hand, the two tapping of half the storage volume represent a very large amount and is not very realistic.

So, an average between these two extremes is proposed that leads to a tapping pattern of four tapplings of a quarter of the storage. Moreover, it can be secured by cyclic tapping that energy content of the storage in the beginning and in the end of phase 2 of EN 255-3 testing period is the same without time- and cost-intensive stabilisation periods. Thus, a simplification of the European tapping cycles defined by Mandate M/324 for storage systems is proposed.

Proposal:

The cyclic extraction according to EN 255-3 should be kept, but the tapped volume should be changed to the more realistic value to simplify the tapping profile defined in Mandate M/324 for storage systems. The volume shall be determined by the energy content of the draw-off equivalent to ¼ of the energy content of the storage at a DHW reference temperature of 60°C. In addition, the energy content in the volume of the two last draw-offs shall not differ by more than 5 %

3.5.4.4 Electrical power input for stand-by losses

To reduce the testing time the following simplifications of the EN 255-3 are proposed:

- If values of the heat loss of the DHW storage are known, they shall be used for the calculation and testing of the electrical power input to cover the stand-by losses is not mandatory. However, if the client asks for the entire EN 255-3 test, it can be accomplished as prescribed, of course. The French team came to the same conclusion, as stated in chapter 3.1.2. To determine the electrical

energy input during stand-by a mean temperature for the storage reheating of stand-by losses shall be evaluated from the controller settings and the reference hot water temperature delivered by EN 255-3. With this mean temperature, the COP from the space heating characteristic is to be determined, and using this COP, the electrical energy to cover storage loss can be calculated.

- If no values of the heat loss of the storage are available or if the heat pump unit is highly integrated, e.g. in case of compact units or in case of CO₂ integrated heat pumps, the full testing according to EN 255-3 shall be performed once at average operating conditions.

Proposal:

Since the testing time seems to be the crucial point in the application of EN 255-3, information on the storage tank shall be used, if available. The necessary COP value to calculate the electrical power input due to thermal storage losses shall be derived using the controller setting for reheating, the reference hot water temperature delivered by EN 255-3 and the heating characteristic. The COP for the reference temperature and UA-values for the mean temperature of the set values shall be used in the calculation.

3.5.4.5 Extension of test points

It is proposed that phase 2 of EN 255-3, which is not so time-consuming, shall be performed for different source temperatures in order to derive a characteristic and therefore the possibility to interpolate the domestic hot water mode for different source temperatures. Concerning the test points, the prescription of EN 255-3 shall be adapted. If more than one test point is to be measured by repetition of phase 2, the source temperature should be adapted to the source temperature measured for the space heating-only mode.

Discussion:

Testing of one point in the DHW mode is already quite expensive. Therefore, an interpolation using information of the heating characteristic from the space heating, which is measured anyway, could be used instead of increasing the number of test points for DHW.

Proposal:

The testing acc. to EN 255-3 shall not be extended, but the characteristic from the space heating mode shall be used to derive the changes in COP of the heat pump caused by changing source temperatures.

3.5.4.6 Additional part load and defrost testing

In the course of the IEA HPP Annex 28, the necessity of additional part load and defrost testing has been discussed.

Discussion:

The evaluation of the French project shows only negligible effects on the COP by cycling (see chapter 3.1.4). Therefore, no additional testing for part load operation is proposed.

Defrosting is already covered in the EN 14511, and no necessity for additional testing of defrost losses is seen.

Proposal:

Additional testing for part load operation and defrosting is not required

3.5.4.7 Cold water temperature

Temperatures of incoming water shall be changed to 10°C which appears to be a more realistic value than the prescribed 15°C in EN 255-3. Moreover, Mandate M/324 also defines the cold water temperature of 10°C.

Proposal:

Change of the cold water inlet temperature from 15°C to 10°C in accordance with Mandate M/324

3.5.4.8 Test points

The test points for the simultaneous operation are based on the test points of the single operation modes. The test points for the complete testing for different heat pump types are given in part 1 chapter B 5 in the description of the extension of the test procedure for simultaneous operation.

3.5.5 Special requirements for testing of combined operating CO₂-systems

A so-called integrated B/W-heat pump using a tripartite gas cooler and heat decoupling by desuperheater, condenser and cooling down of the condensate according to Fig. 12 has been extensively measured by SINTEF Energy Research, Norway. The refrigerant in this heat pump is CO₂. The following conclusions for the testing of the systems could be drawn for this configuration of integrated heat pumps with CO₂-refrigerant [15].

3.5.5.1 Test points

Integrated brine-to-water and water-to-water CO₂ heat pumps are only applicable in low-temperature heat distribution systems, e.g. hydronic floor heating system, and it will not make much sense to test the systems at supply temperatures above 40°C.

Proposal:

Testing of CO₂-heat pumps shall be restricted to supply temperatures of the space heating system below 40°C

3.5.5.2 Test points

Consequently, typical test conditions at the sink temperature side could be:

Space heating mode:	supply/return temperature: 40°C/35°C, 35°C/30°C
DHW mode:	60°C
Combined mode:	A combination of the temperature regimes in the space heating and DHW modes

In addition, the heat pumps should be tested at two temperature levels for heat source side. According to EN 14511, the temperature level will be depending on whether it is an indirect (brine) system or a direct (water) system.

3.5.5.3 Source temperature, inlet temperature domestic hot water

The COP of the integrated CO₂ heat pump system is strongly influenced by the inlet water temperature for the DHW preheating gas cooler unit. This temperature is in turn determined by the initial cold water temperature and the thermodynamic losses in the single-shell storage tank (i.e. mixing of hot and cold water and conductive heat transfer). Therefore, the design of the tank is of crucial importance for the energy efficiency of the CO₂ heat pump system.

In order to simplify the test procedures and focus only on the heat pump unit, the tests could be carried out at a fixed inlet water temperature during operation in the DHW mode and the combined mode, e.g. 10°C. Another alternative is to test the CO₂ heat pumps at different inlet water temperatures, e.g. from 5°C to 20°C, but this will increase the extent and costs of the testing considerably.

Proposal:

In order to reduce testing expenses, testing in DHW and combined mode could be carried out at fixed inlet water temperature of 10°C in line with non CO₂ heat pumps

3.5.6 Special requirement of DX heat pump systems

The testing of direct expansion (DX) heat pump systems will be covered in the CEN working group CEN/TC 113/WG 11. The kick-off meeting for the working group has taken place in March 2005. Experiences of the testing of the DX-systems in the framework of IEA HPP Annex 28 will be discussed in the working group. Test points defined for testing direct expansion systems are contained in the test proposals in the part 1 of this report, chapter B 5.3. They are based on current testing at arsenal research. Further discussion will take place in the working group 11.

3.5.7 Test procedure for simultaneous B/W heat pump based on ASHRAE 137

The Canadian prototype of a simultaneous operating brine-to-water heat pump for space cooling & DHW and space heating & DHW depicted in Fig. 5 has been tested on a developed test rig based on ASHRAE 137 [20].

The test rig for the testing is shown in Fig. 6. Contrary to the IEA HPP Annex 28 approach described in chapter 2.5.1 a grey box concept for the testing has been applied, i.e. additional measurement points inside the system boundary were measured, namely refrigerant temperatures and pressures. However, this might be required only for the monitoring of the system.

Fig. 56 gives a flow chart of the proposed test procedure. The proposed test procedure is described in the Canadian country report in the following way:

"The test set-up imposes, that respective water valves are opened or closed in order to create a single flow path at the inlet and outlet of the desuperheater. The water flows from the storage tank at constant temperature during the test. The heat pump has to operate continuously in the desired simultaneous operation mode (space heating & DHW or space cooling & DHW, respectively). The desuperheater water pump has to be on, and the floor heating system or the air-conditioning fan coil is be regulated to keep the desired dry- and wet bulb indoor temperatures.

In a second step, constant-temperature water has to be supplied to the desuperheater inlet. The temperature of the water will be equal to the median inlet temperature determined from a simulated use test. During the data collection period, the inlet and outlet temperatures will remain within a tolerance of 0.5°C (1°F).

Since the water flow rate through the desuperheater is constant, the flow has to be adjusted to the nominal value reported by the manufacturer. The flow rate through the desuperheater will be maintained with a tolerance of 5% or 0.003 l/s (0.05 USGPM) whichever is higher during the data collection interval.

During the pre-test operation, the heat pump will operate in the selected mode for one hour or at equilibrium for half an hour at the specified conditions, whichever occurs later, before initiating data collection.

Except for the test tolerance variables, data recording will occur at ten-minute intervals. Recording will continue until four consecutive data sets are collected, while the test tolerances are met.

Space conditioning measurements and calculation in both modes (cooling & DHW and heating & DHW) will be measured in accordance with ASHRAE 116 [41].

The water heating capacity shall be determined from measuring the water flow rate and the temperature entering and leaving the desuperheater according to the equation

$$\phi_i = \dot{m}_w \cdot c_{pw} \cdot (T_{w,out} - T_{w,in}) \quad [W] \quad \text{eq. 21}$$

where

ϕ_i	water heating capacity of desuperheater water heating	(W)
\dot{m}_w	water mass flow rate	(kg/s)
c_{pw}	specific heat capacity of the water	(kJ/(kg·K))
$T_{w,out}$	outlet temperature of the water from the desuperheater	(°C)
$T_{w,in}$	inlet temperature of the water to the desuperheater"	(°C)

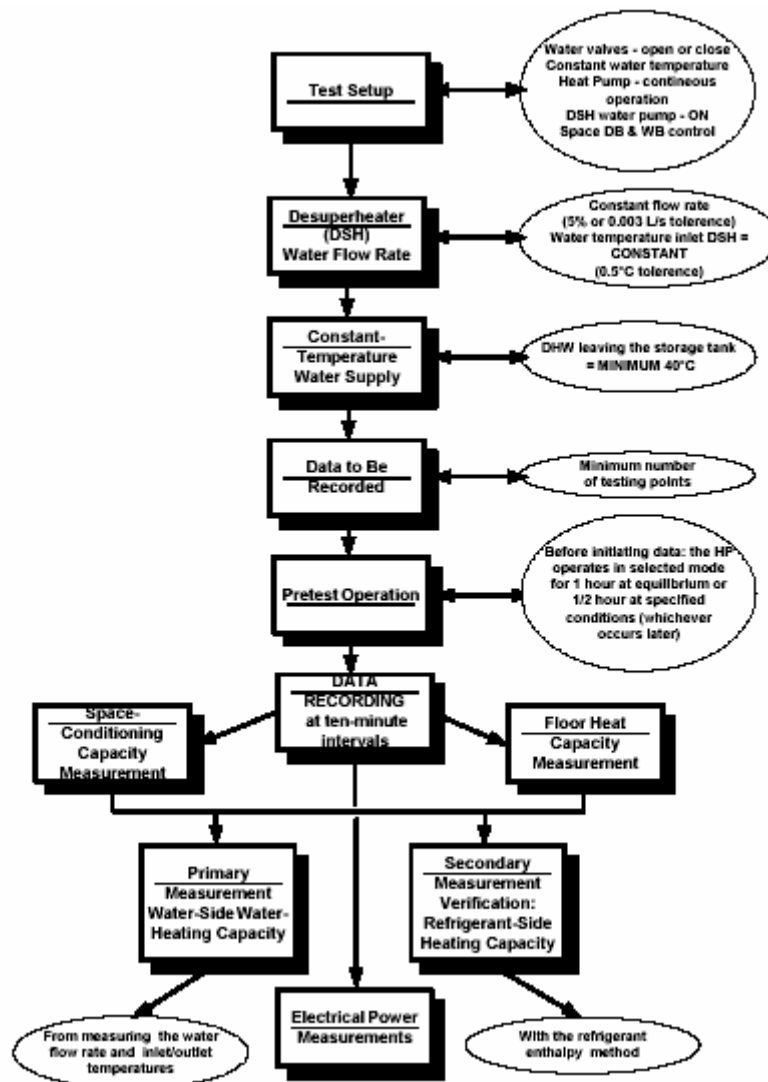


Fig. 56: Flow chart of the Canadian proposal for a test procedure

Four tests are proposed according to the operation modes at four selected outdoor temperature in accordance with the calculation approach described in chapter 3.2.7.

Tab. 27: Test points for the Canadian simultaneous ground-source heat pump (source [34])

TEST #	Inlet BRINE Temperature*	Brine Flow Rate	Refrigerant Direct Condensing Temperature (FLOOR)	Simultaneous Operating Mode (Conditions)	Minimum Water Tapping Temperature	Air Surrounding Indoor Unit (Dry bulb)
-	°C	L/s	°C	-	°C	°C
#1	- 5	Manufacturer	45	Floor HEAT&DHW (Low Temperature)	40	21.1
#2	0	Manufacturer	45	Floor HEAT&DHW (Standard)	40	21.1
#3	+5	Manufacturer	45	Floor HEAT&DHW (High Temperature)	40	21.1
#4	+ 10	Manufacturer	35	Air COOL&DHW	30	26.7

For at least one test, either a total energy balance or measurements on the refrigerant side shall be carried out for the desuperheater. In accordance with the black-box approach of IEA HPP Annex 28, the energy balance seems to be preferable.

3.5.8 Testing of compact units

Compact units are highly integrated heat pump systems coupled to a ventilation heat recovery. Thus, quite a lot of standards and national regulations that refer to the single components of a compact unit had to be taken into account. These standards, however, often have different test points. To make this situation even worse, some of the standards were under revision in the time of the IEA HPP Annex 28, e.g. EN 13141-7 on ventilation systems, or are to be revised soon, e.g. EN 255-3 for domestic hot water operation of heat pumps.

Tab. 28: Different test points of air heat exchangers according to different standards

Phys. Value	EN 308	EN 13141-7	EN 255/EN 14511	DIBt
Flow rate	V_n ; 0.66 V_n ; 1.5 V_n permutation			
Fresh air Temperature	5°C 5°C 5°C	same as EN 308	A_i 20/20 A_o 20/7	-3°C/80% 4°C/80% 10°C/80%
Exhaust air temperature	25°C {14°C} 25°C {18°C} 15°C {10°C}	Same as EN 308	A/W 20/50 A/W 20/35 A/A 20	21°C/36% 21°C/46% 21°C/56%

Tab. 28 gives a comparison of the test conditions contained in different standards. It is still an open question, which standard shall be the most relevant for compact units and thus define the test points. Switzerland opted for EN 14511, since the heat pump appears to be the core unit of the system.

3.5.8.1 Swiss approach for the testing of compact units

The test procedure described in part 1, chapter B 4.4.1 in connection with the test points in chapter B 5.4 is proposed in order to enable the inclusion of the heat recovery according to the calculation approach described in the final report part 1, chapter A 2.

Besides the thermodynamic testing the test procedure comprises

- Leakage testing (including filter leakage testing)
- Air flow testing (pressure curve)
- Acoustic testing
- Handling/maintenance/safety
- Hygienic examinations

Details on the entire testing are given in the test guideline of HTA Lucerne [40]. The following decisions were taken for the thermodynamic testing of the unit.

- The calculation of the heat recovery will be accomplished on the basis of the temperature change coefficient, since this value is easy to measure and therefore the most reliable value to characterize the heat recovery.
- For the calculation of the heat recovery in connection with the heat pump, test points in accordance with EN 14511 [11] are used. The test points delivered by the test procedure will comprise measurements of the single operation of the heat recovery unit and the combined operation of the heat recovery unit and the heat pump. These two cases are sufficient, since the heat recovery unit is postulated to run through the entire heating period. Consequently, the heating period can be characterised by the two measured operation modes.
- For the DHW operation, calculation will be based on test results according to EN 255-3 [10]. Since the method is time-consuming, the test procedure will only be performed for one point.

3.6 Validation of the proposed test procedure

Sweden has performed tests according to the proposed test procedure for an exhaust-air heat pump (EAHP) of the type shown in Fig. 13 right hand side. q_a denotes the volume flow rate of the exhaust-air.

Tab. 29: Proposed test conditions for an exhaust air heat pump (EAHP) [17]

Test point no.	Exhaust-air		Indoor heat exchanger		Heating demand/ Heating capacity $\frac{P_{H,hs}}{P_H}$	Heating mode
	Inlet dry bulb (wet bulb) temperature $\theta_{ea,in}(t_{wb,a,in})$ (°C)	Exhaust-air flow rate q_a (m ³ /h)	Inlet temperature $\theta_{w,in}$ (°C)	Outlet temperature $\theta_{w,out}$ (°C)		
1	20(12) $\theta_{dp}=4.95$	$q_{a,n}^a$	40	45	1	Space
2	20(12)	$q_{a,n}$	b	35	1	Space
3	20(12)	$q_{a,n}$	b	55	1	Space
4	20(12)	$1.5 \cdot q_{a,n}$	b	45	1	Space
5	20(12)	$0.75 \cdot q_{a,n}^c$	b	45	1	Space
6	15(10) $\theta_{dp}=4.95$	$q_{a,n}$	b	45	1	Space
7	20(15) $\theta_{dp}=11.66$	$q_{a,n}$	b	45	1	Space
8	20(12)	$q_{a,n}$	b	45	1	Combined
9	20(12)	$q_{a,n}$	b	35	1	Combined
10 ^d	20(12)	$q_{a,n}$	b	50	2	Space
11	20(12)	$q_{a,n}$	-	-	-	DHW EN255-3 mod
I	20(12)	$q_{a,n}$	-	-	-	DHW EN255-3
II	20(13) $\theta_{dp}=7.42$	$q_{a,n}^a$	40	45	1	Space
III	20(12)	$q_{a,n}$	e	35	$\cong 2$	Combined

a The exhaust-air flow rate shall be set so that the temperature difference on the air side is 18 K at test point no. 1.

b The test is performed at the flow rate obtained during the test at standard rating conditions (test point no. 1)

c In case the $0.75 \cdot q_{a,n}^a$ is a lower value then the minimum air flow rate defined by the manufacturer, the air flow shall be set to the minimum value.

d This test point is only performed for EAHP type 2 (see Fig. 13)

e The heating water flow rate was set at a value twice the one obtained in test point no. 1

3.6.1 Space heating-only operation

3.6.1.1 Wet bulb temperature

Tab. 30 shows the test points used to evaluate the impact of the wet bulb temperature. $P_H/P_{H,hs}$ denotes the ratio the heating capacity of the heat pump according to EN 14511 and the total heating capacity including the back-up heater. θ_{dp} refers to the dewpoint.

Tab. 30: Test points for evaluation of the wet bulb temperature [17]

Exhaust-air			Indoor heat exchanger			
Test point no.	Inlet dry bulb (wet bulb) temperature $\theta_{ea,in}(\theta_{wb,a,in})$ (°C)	Exhaust-air flow rate q_a (m ³ /h)	Inlet temperature $\theta_{w,in}$ (°C)	Outlet temperature $\theta_{w,out}$ (°C)	Heating demand/ Heating capacity $\frac{P_{H,hs}}{P_H}$	Heating mode
1	20(12), $\theta_{dp}=5$	$q_{a,n}^a$	40	45	1	Space
7	20(15), $\theta_{dp}=11.7$	$q_{a,n}$	b	45	1	Space
II	20(13), $\theta_{dp}=7.4$	$q_{a,n}^a$	40	45	1	Space

The three above test points were measured to characterise the influence of the wet bulb temperature.

Fig. 57 depicts the influence, which requires the measurement of two wet bulb temperatures due to the impact on both the COP and the heating capacity. The $COP_{hps,tot}$ refers to an uncorrected COP for the pumps. COP_{comp} denotes a COP, where only the compressor power input is taken into account. However, the curves are almost linear, so that a third point (test point II) inbetween is not required.

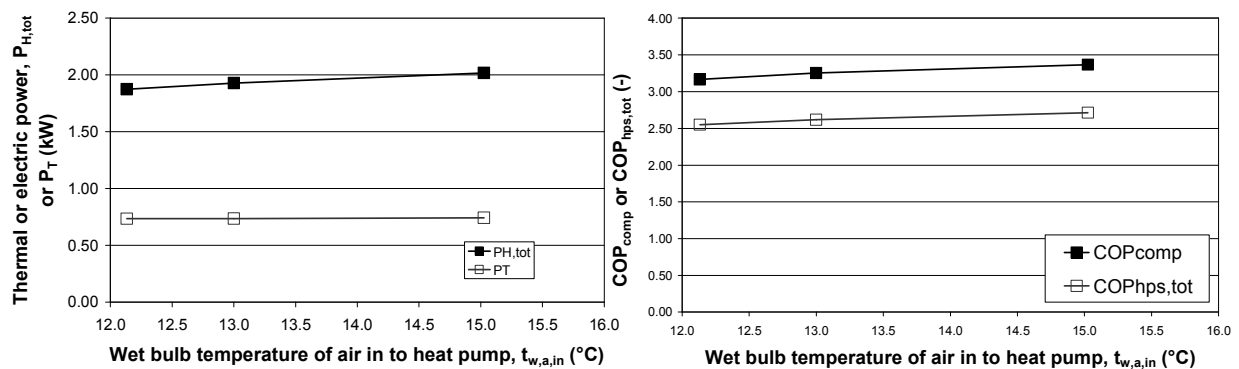


Fig. 57: Steady state testing dependent of the wet bulb temperature of EAHP type 2 (see Fig. 13)

3.6.1.2 Supply temperature

The supply temperature of the heating system has a strong impact on the COP and the heating capacity, as in the case of most heat pumps. Therefore, at least two test points at different supply temperatures should be measured, which is documented in Fig. 58.

Tab. 31: Test points to evaluate the impact of the supply temperature of the heating system

Exhaust-air			Indoor heat exchanger			
Test point no.	Inlet dry bulb (wet bulb) temperature $\theta_{ea,in}(\theta_{wb,a,in})$ (°C)	Exhaust-air flow rate q_a (m ³ /h)	Inlet temperature $\theta_{w,in}$ (°C)	Outlet temperature $\theta_{w,out}$ (°C)	Heating demand/ Heating capacity $\frac{P_{H,hs}}{P_H}$	Heating mode
1	20(12), $\theta_{dp}=4.95$	$q_{a,n}^a$	40	45	1	Space
2	20(12)	$q_{a,n}$	B	35	1	Space

To evaluate the impact of the supply temperature the test points 1 and 2 given in Tab. 31 are used.

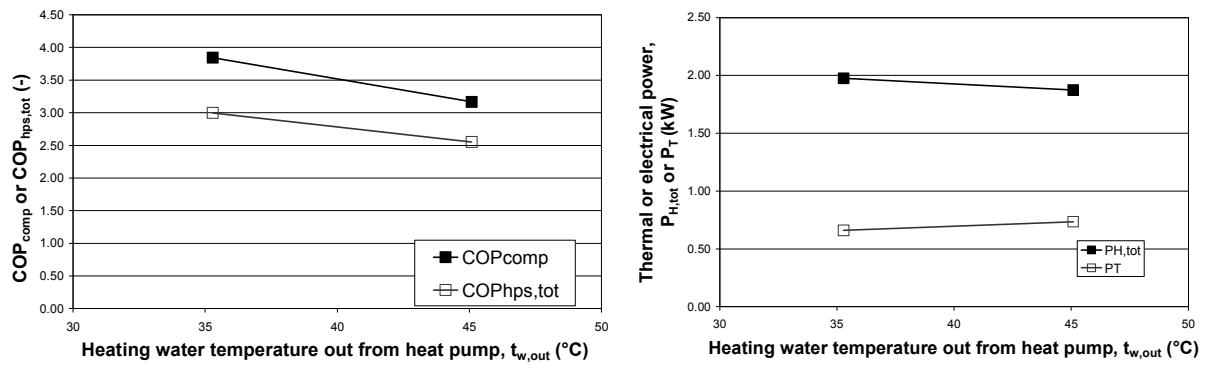


Fig. 58: Impact of the supply temperature of the heating system on COP and heating capacity

3.6.1.3 Volume flow rate

The volume flow rate has also an influence on the efficiency and the heating capacity, so at least two volume flow rates should be measured. Test points used to characterise the impact of the volume flow rate are given in Tab. 32.

Tab. 32: Test points to evaluate the impact of the volume flow rate [17]

Test point no.	Exhaust-air		Indoor heat exchanger		Heating demand/ Heating capacity $\frac{P_{H,hs}}{P_H}$	Heating mode
	Inlet dry bulb (wet bulb) temperature $\theta_{a,in}(\theta_{wb,a,in})$ (°C)	Exhaust-air flow rate q_a (m ³ /h)	Inlet temperature $\theta_{w,in}$ (°C)	Outlet temperature $\theta_{w,out}$ (°C)		
1	20(12), $\theta_{dp}=4.95$	$q_{a,n}^a$	40	45	1	Space
4	20(12)	$1.5 \cdot q_{a,n}$	b	45	1	Space

However, if the volume flow rate is lower in other operating points, probably the lower volume flow rates should be measured, too, since extrapolation is not recommended due to possible frosting with significantly changed results that are not reflected in an extrapolation. Test points 1 and 4 are used to evaluate the impact of the volume flow rate. Fig. 59 depicts the impact of the volume flow rate.

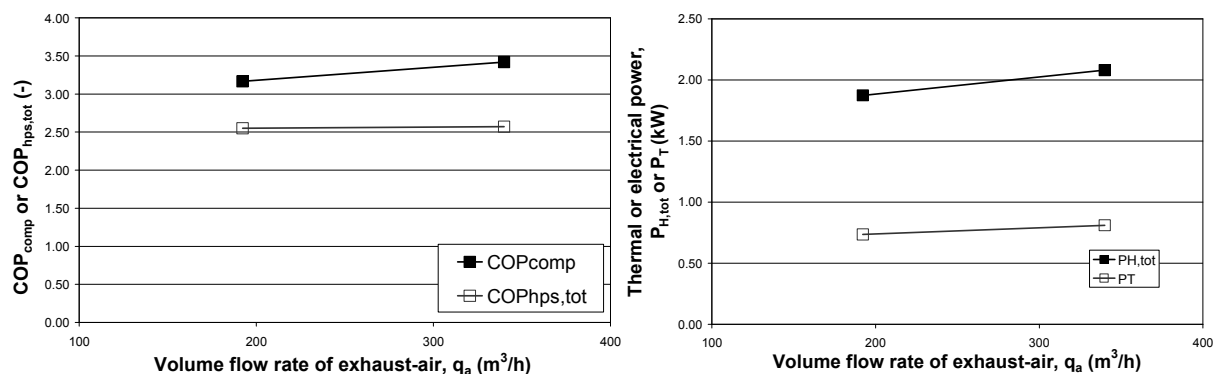


Fig. 59: Impact of different volume flow rates (source [17])

3.6.1.4 Back-up heater operation

To characterise the influence of back-up heater operation on the COP, a test with a double heat demand has been conducted. Test points used for the evaluation are given in Tab. 33

Tab. 33: Used test points for the evaluation of the back-up heater [17]

Test point no.	Exhaust-air		Indoor heat exchanger		Heating demand/ Heating capacity $\frac{P_{H,hs}}{P_H}$	Heating mode
	Inlet dry bulb (wet bulb) temperature $\theta_{ea,in}(\theta_{wb,a,in})$ (°C)	Exhaust-air flow rate q_a (m ³ /h)	Inlet temperature $\theta_{w,in}$ (°C)	Outlet temperature $\theta_{w,out}$ (°C)		
1	20(12) $\theta_{dp}=4.95$	$q_{a,n}^a$	40	45	1	Space
10 ^d	20(12)	$q_{a,n}$	b	50	2	Space

In case of this heat pump, no influence of the electrical back-up heater on the condensing temperature and thereby on the COP was detect, which documents a good thermal stratification in the storage tank. Fig. 60 shows the back-up heater impact.

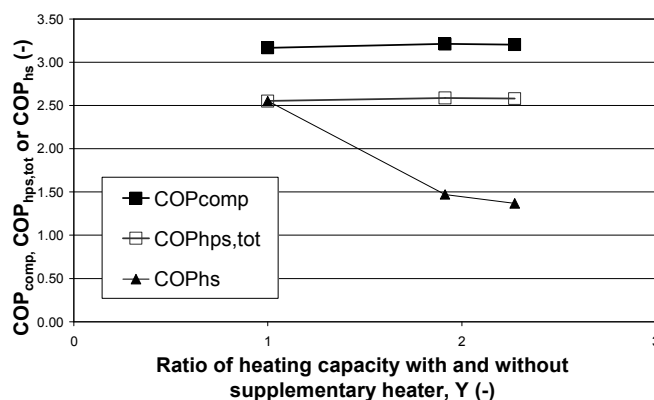


Fig. 60: Evaluation of the impact of the back-up heater (source [17])

3.6.2 DHW-only operation

In previous chapter 3.5 the proposed changes for the DHW-only testing are described. Thus, the EAHP type 2 (see Fig. 13) was tested according to the original EN 255-3 procedure and the EN 255-3 with the proposed modifications. Tab. 34 gives the test points 11 and I used to compare the proposed modified method with the original.

Tab. 34: Test points for the DHW-only testing (EAHP) [17]

Test point no.	Exhaust-air		Indoor heat exchanger		Heating demand/ Heating capacity $\frac{P_{H,hs}}{P_H}$	Heating mode
	Inlet dry bulb (wet bulb) temperature $\theta_{ea,in}(\theta_{wb,a,in})$ (°C)	Exhaust-air flow rate q_a (m ³ /h)	Inlet temperature $\theta_{w,in}$ (°C)	Outlet temperature $\theta_{w,out}$ (°C)		
11	20(12)	$q_{a,n}$	-	-	-	DHW EN255-3
I	20(12)	$q_{a,n}$	-	-	-	DHW EN255-3

Results gathered in Tab. 35 show that the original EN 255-3 procedure delivers higher COP_t values. This result is due to a lower mean condensation temperature in case of a larger draw-off, since more cold water enters in the storage, so the storage is colder and therefore the temperature entering the heat pump lower. The difference would even be larger, if the cold water temperature of the EN 255-3 were 10°C as in the modified testing see chapter

Moreover, the impact of the stand-by testing are quite high, which can be deduced to the electrical consumption of the fans, which are operated all the time and makes up to 39% of the stand-by consumption. Furthermore, it can be seen that in the last two draw-offs the deviation of the energy content is less the required 5%, in this case 0.4. However, for comparison reasons the fan power for neither the stand-by power input nor the COP values are corrected to 0 kPa.

Tab. 35: Results of the DHW only-operation according to the proposed modification and the original procedure according to EN 255-3 [17]

Test results	EN 255-3 modified	EN 255-3
COP_t	2.94	3.07
$COP_{t,s}$ (including stand-by power)	2.32	2.37
Deviation of draw-off volume for the two last draw-offs (%)	0.4	
Deviation of energy content for the two last draw-offs (%)	-	0.3

3.6.3 Combined operation

For the combined operation, two test points 8 and 9 are to be performed. For additional evaluation, the test point III has been performed, as well. Tab. 36 gives the three test conditions to evaluate the combined operation.

Tab. 36: Test points for the combined operation [17]

	Exhaust-air		Indoor heat exchanger			
Test point no.	Inlet dry bulb (wet bulb) temperature $\theta_{a,in}(\theta_{wb,a,in})$ (°C)	Exhaust-air flow rate q_a (m³/h)	Inlet temperature $\theta_{w,in}$ (°C)	Outlet temperature $\theta_{w,out}$ (°C)	Heating demand/ Heating capacity $\frac{P_{H,hs}}{P_H}$	Heating mode
8	20(12)	$q_{a,n}$	b	45	1	Combined
9	20(12)	$q_{a,n}$	b	35	1	Combined
III	20(12)	$q_{a,n}$	e	35	$\cong 2$	Combined

Fig. 61 shows the testing and Fig. 62 the evaluation of the COP at test point 8, i.e. at 45°C supply temperature. Contrary to the evaluation of brine-to-water heat pumps, the second heating period of the immersion back-up heater has to be taken into account, since otherwise COP-values higher than the space heating-only steady state test will result, which does not make sense.

Moreover, it can be seen in the COP evaluation that the values at lower ratios of the energy for space heating and for DHW operation are not representative values.

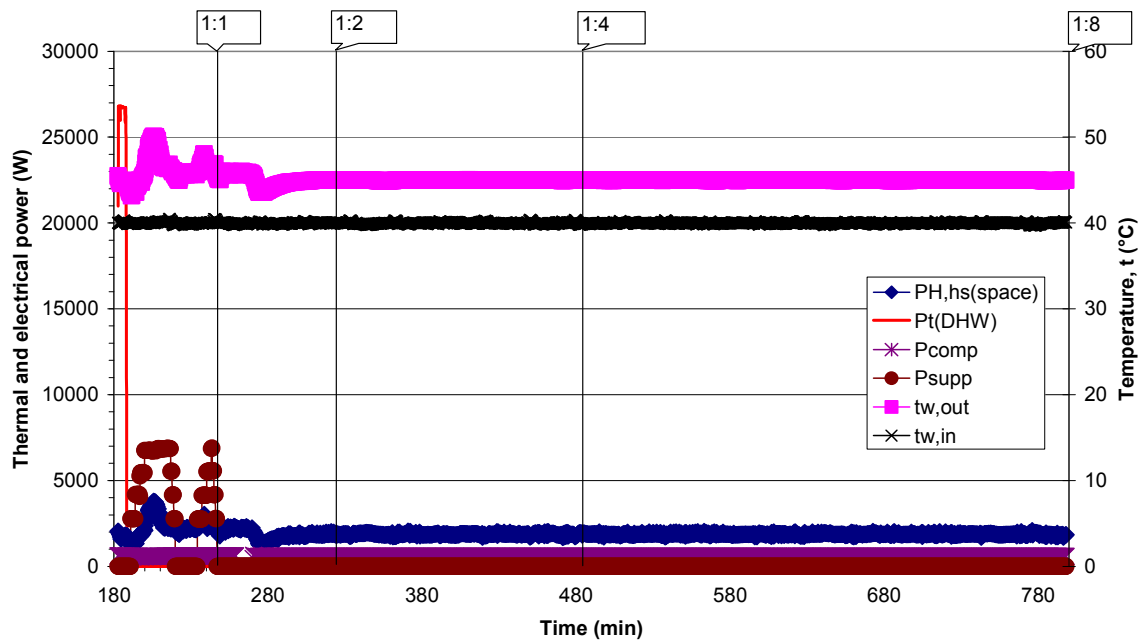


Fig. 61: Testing for combined operation at test point 8 [17]

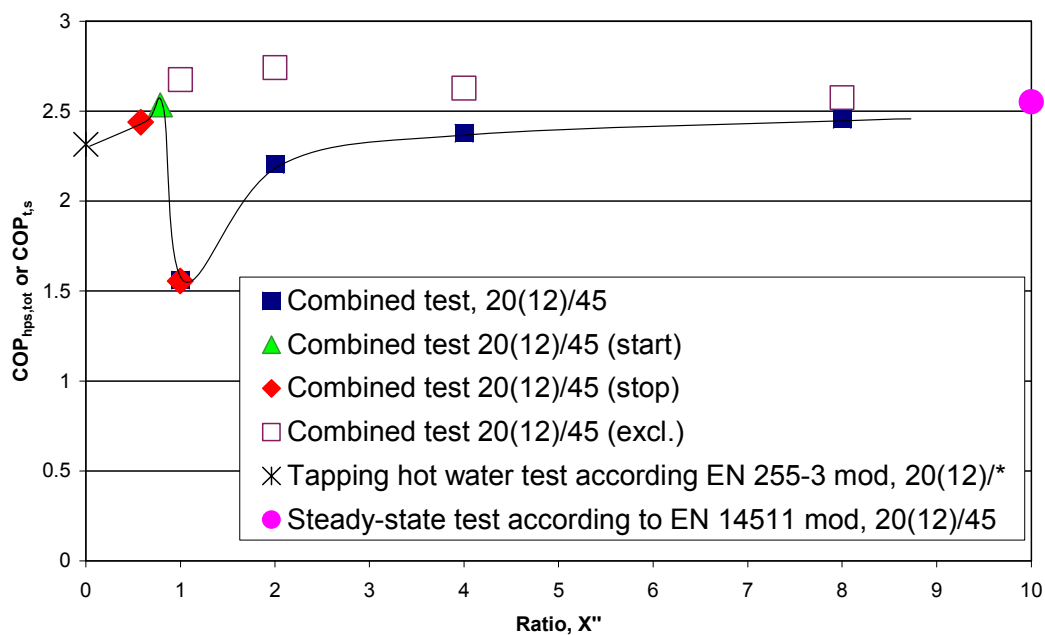


Fig. 62: COP for combined testing at test point 8 [17]

These results are confirmed at the other test point 9 at 35°C supply temperature. At this point, the operation of the supplementary heater has even a higher influence on the supply temperature.

Since the heating capacity at this point is near the balance point, the compressor switches on and off during the operation. Therefore, the values at low ratio X'' of space heating and DHW energy scatter even more than in case of 45°C supply temperatures. In addition, the compressor shut-offs results in a lower heating capacity of the heat pump. Fig. 63 shows the testing and Fig. 64 the evaluation of the COP.

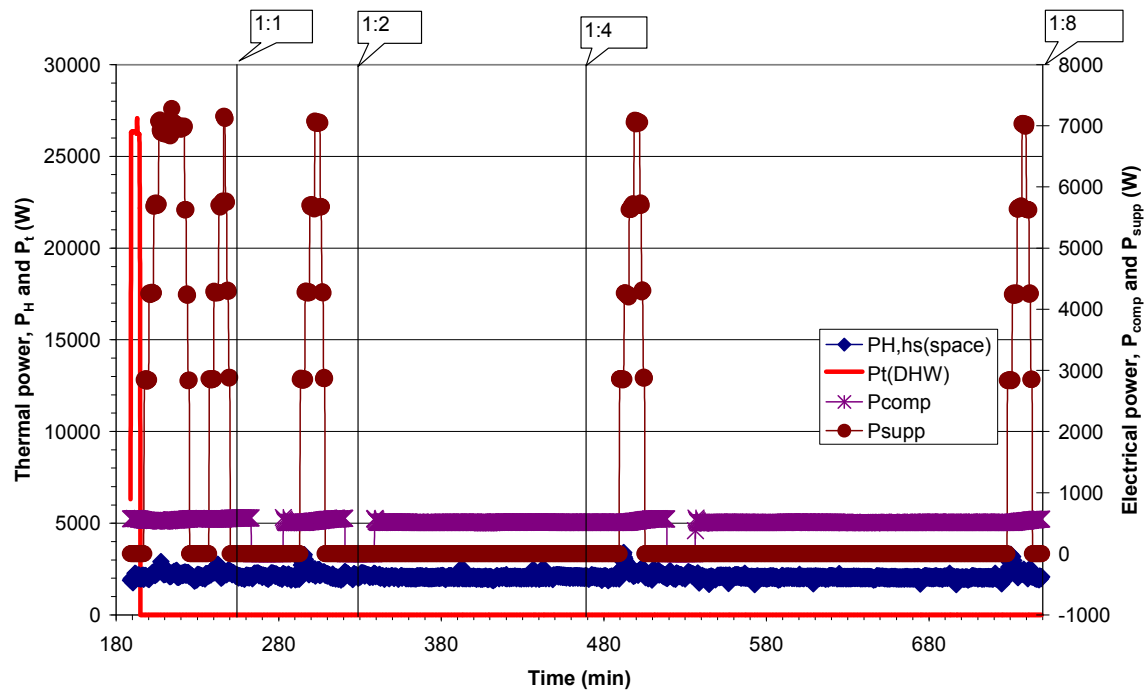


Fig. 63: Testing for combined testing at test point 9 [17]

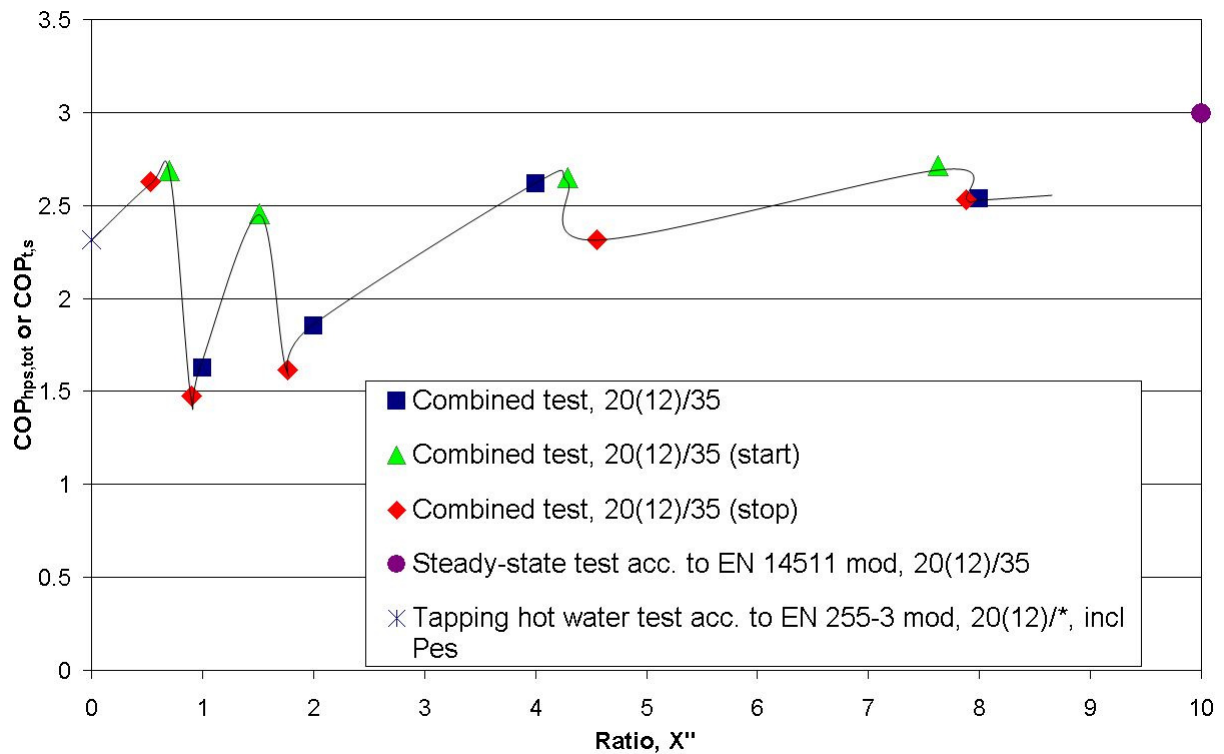


Fig. 64: COP for combined testing at test point 9 [17]

To avoid this operation at the balance point, the heating demand is doubled in the test point III at the same temperature conditions by increasing the heating water mass flow rate. Fig. 65 shows the results of the testing and Fig. 66 the evaluation of the COP.

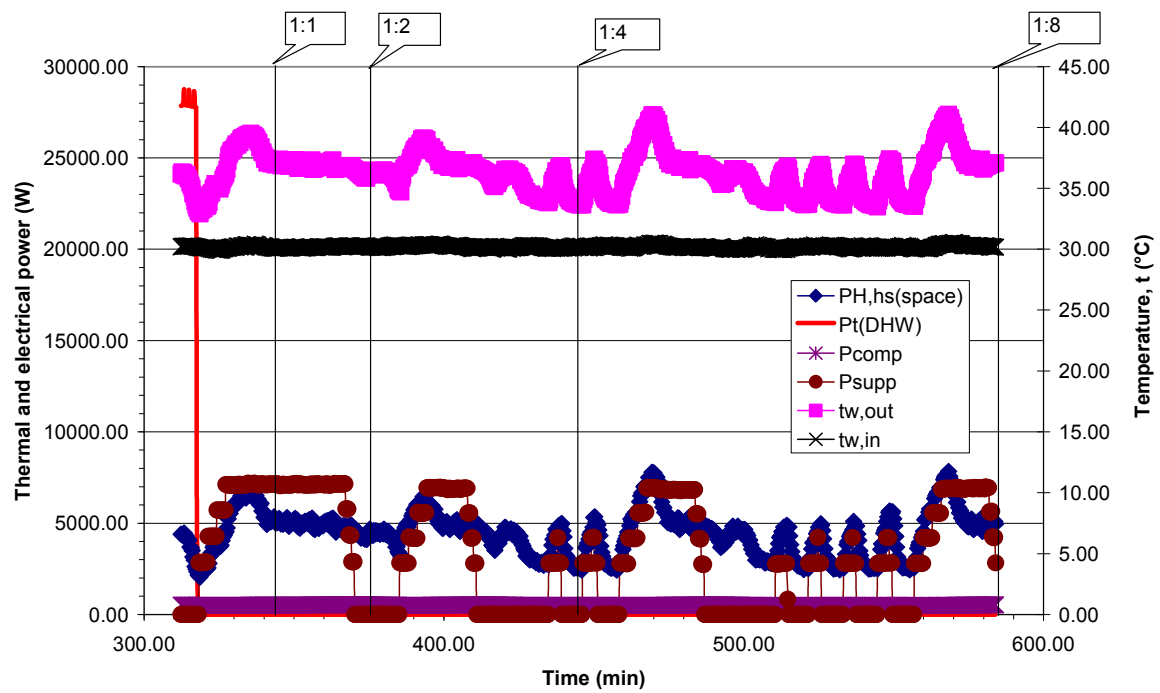


Fig. 65: Testing for combined testing at test point III [17]

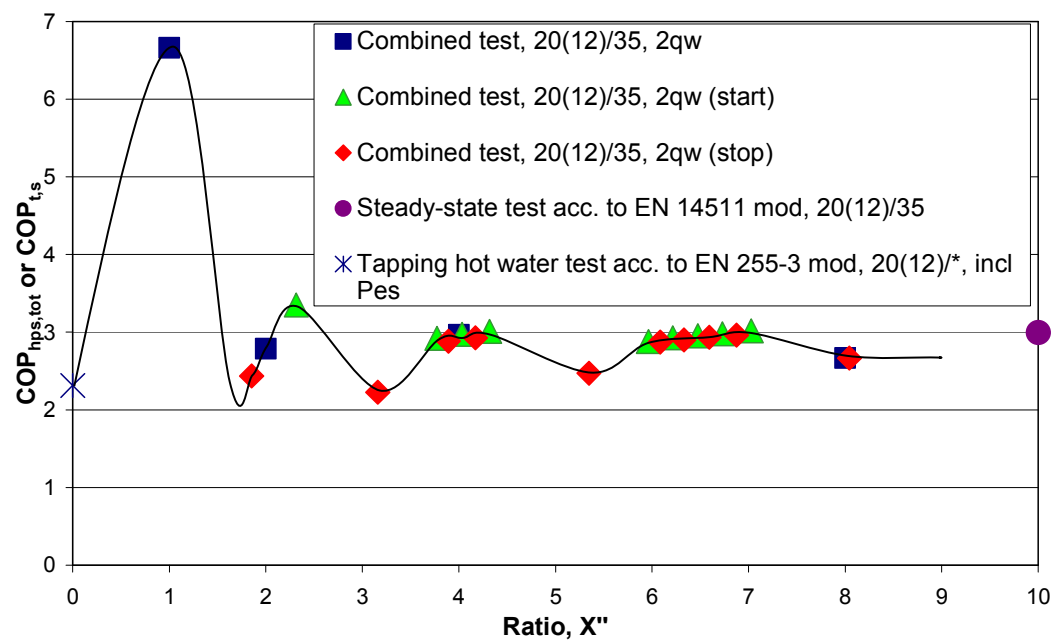


Fig. 66: Testing for combined testing at test point III [17]

For this operating condition, the on-off operation of the supplementary heater did not result in any compressor shut-offs and the heating capacity of the heat pump was unaffected.

3.6.4 Conclusions of the application of the proposed test procedure

- The results from the steady-state tests show that the number of test points according to Tab. 29 is necessary and as long as evaluation was performed, no more steady-state test points are necessary. However, the third test point for some parameters, e.g. heating water temperature (test point no. 3) and exhaust-air flow rate (test point no. 5), were not performed due to time and cost limits. Neither test point no. 6 was performed, which involved a lower dry bulb temperature of the exhaust-air compared to the other test points.
- The results from the draw-off tests show that the proposed test procedure result in representative and reliable values. The results also show that the stand-by losses might influence the COP value to a large extent and it is therefore important that this test is performed.
- The results from combined testing is necessary for simultaneously heating heat pumps, but the test procedure must be adapted to different heat pump types and design solutions in order to enable fair comparison of heat pumps of different types and design solutions. Using results from predetermined ratios X'' is a good solution for an alternating heating heat pump, but not for an EAHP type 2 depicted in Fig. 13 with simultaneous combined heating of heating water and DHW. The test procedure should result in one value for each operation condition (test point) in simultaneous operation according to the joint agreement in IEA HPP Annex 28 and two different alternatives are proposed. The first one is to stop the test after the first shut-off of the immersion heater following the draw-off. The second alternative is to stop the test after the first shut-off of the immersion heater occurring after a ratio of e.g. eight has been reached. The latter would involve addition of one or two test points in some situations.
- For test points where the supplementary heating, the immersion heater in this case, is allowed to switch-on, i.e. the test points no. 8, 9, 10 and III, the on and off operation of the immersion heater affects the heating water temperature out of the heat pump and the maximum allowed deviation must be increased compared to EN14511.
- Some of the test points are relatively time consuming. However, this is inevitable, since a long test period is necessary for guaranteeing reliable and representative test values.
- The proposed test procedure described in section 3.5 can deliver results that can be used as input data for calculation of SPF.

3.7 Approach for a dynamic test procedure

Germany has developed a dynamic test procedure as it is already introduced in the standardisation of testing of solar components like collectors and solar DHW-systems based on the method of parameter identification [42]. The method is described in the following. Respective measurements to perform the procedure are to be carried out with a compact unit.

3.7.1 Background

The testing of the domestic hot water operation according to EN 255-3 is time-consuming and thereby costly.

In the field of solar application dynamic testing methods have been successfully introduced in the standardisation that are capable to deliver the same component or system parameters with reduced testing time and testing expense. Moreover, broader boundary conditions can be used to derive the parameters (e.g. irradiation conditions in the case of solar collectors).

The method applied is a model-based approach, where the physical parameters of the model (the test parameters) are fitted from the measurements (parameter identification).

3.7.2 Model and calculation

The components of the system (above) and the model of the component with the respective parameters are given in Fig. 67.

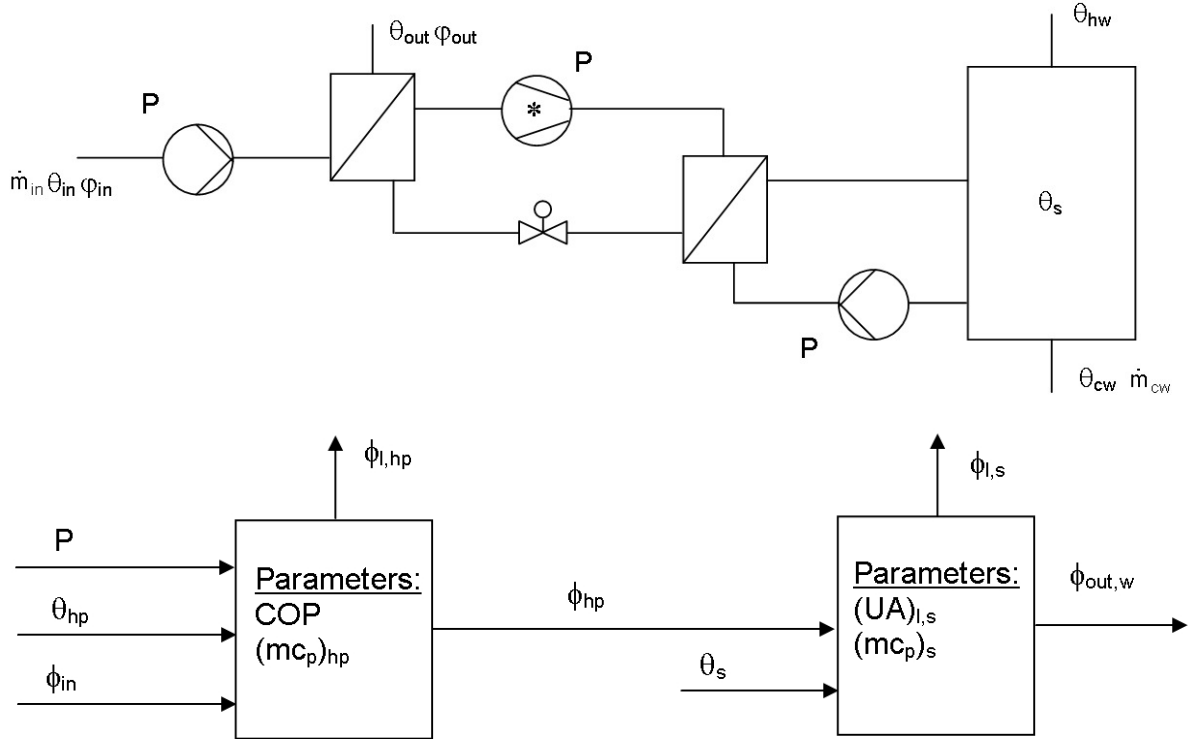


Fig. 67: System layout (above) and components in the model (below) for dynamic heat pump testing approach

The identification is based on the following assumptions and energy balances.

Assumption:

$$\theta_{hp} = \theta_s \quad [^{\circ}\text{C}] \quad \text{eq. 22}$$

where

θ_{hp} temperature of the heat pump $(^{\circ}\text{C})$

θ_s temperature of the storage $(^{\circ}\text{C})$

Balances:

Heat pump:

$$\phi_{hp} = \text{COP} \cdot P - (mc_p)_{hp} \cdot \dot{\theta}_{hp} = P + \phi_{in} - \phi_{l, hp} - (mc_p)_{hp} \cdot \dot{\theta}_s = f(\theta_{in}, \dot{m}_{in}, \phi_{in}, \theta_s) \quad \text{eq. 23}$$

where

ϕ_{hp} heating capacity of the heat pump (W)

COP coefficient of performance of the heat pump $(-)$

P electrical power input to the heat pump (W)

$(mc_p)_{hp}$ heat capacity of the heat pump (J/K)

θ_{hp}	temperature of the heat pump	(°C)
ϕ_{in}	thermal power extracted from the ambient source of the heat pump	(W)
$\phi_{l, hp}$	thermal loss of the heat pump	(W)
θ_s	temperature of the storage	(°C)

Storage:

$$(mc_p)_s \cdot \dot{\theta}_s = \phi_{hp} + \dot{m}_w \cdot c_w \cdot (\theta_{in,w} - \theta_{out,w}) - \phi_{l,s} \quad [W]$$

eq. 24

where

$(mc_p)_s$	heat capacity of the storage	(J/K)
$\dot{\theta}_s$	temperature change of the storage	(°C/s)
ϕ_{hp}	heating capacity of the heat pump	(W)
\dot{m}_w	massflow of water through the storage	(kg/s)
c_w	specific heat capacity of water	(J/(kg·K))
$\theta_{in,w}$	water temperature at storage inlet	(°C)
$\theta_{out,w}$	water temperature at storage outlet	(°C)
$\phi_{l,s}$	thermal loss of the storage	(W)

3.7.3 Required measurements to identify the parameters

Parameters of the storage:

The performed test to characterise the storage is similar to the testing performed for solar energy components that are proven to deliver reproducible results (see EN 12 977–3 [35]).

1. Preconditioning of the storage by cycling of hot water, water extraction of at least the whole storage volume
=> Determination of an average temperature and the energy content of the storage $(mc_p)_s$
2. Preconditioning of the storage by cycling of hot water, 24 h of cooling down in stand-by, water extraction of at least the whole storage volume (often 3 times the volume to secure total extraction of storage water)
=> measurement of the water temperature to determine the energy content of the storage after stand-by
3. Calculations:

$$\phi_{l,s} = (U \cdot A)_s \cdot (\theta_{env} - \theta_s) = (mc_p)_s \cdot \dot{\theta}_s \quad [W]$$

eq. 25

where

$\phi_{l,s}$	thermal losses of the storage	(W)
$(U \cdot A)_s$	heat loss of the storage	(W/K)
θ_{env}	temperature of storage environment	(°C)
θ_s	storage temperature	(°C)
$(m \cdot c_p)_s$	heat capacity of the storage	(J/K)
$\dot{\theta}_s$	temperature change of the storage	(°C/s)

$$UA = \left(\frac{m \cdot c_p}{\Delta t} \right) \cdot \ln \frac{\theta_s - \theta_{\text{env}} \big|_{t=0}}{\theta_s - \theta_{\text{env}} \big|_{t=\Delta t}} \quad [\text{W/K}]$$

eq. 26

where

U	heat loss coefficient	(W/(m ² ·K))
A	surface of the storage	(m ²)
θ _{env}	temperature of storage environment	(°C)
θ _s	storage temperature	(°C)
m	mass of storage water	(kg)
c _p	heat capacity of water	(J/(kg·K))
Δt	time difference	(s)

Eventually the cooling down curve or the (UA)-value are already available from storage testing.

Parameter of the heat pump

3 tests are carried out to determine the COP characteristic and the capacity of the heat pump. With the identified parameters, the respective values according the EN 255-3 can be derived from the model

A) Heating-up power:

=> Determination of the COP

Basis: Cold storage

Steps:

1. Starting with cold storage (θ_s < 20°C) and heating up to θ_s = θ_{used}
=> measurement of the storage temperature
2. Determination of storage energy (calculation)

B) Reheating power:

=> Determination of the thermal capacity of the heat pump, therefore hot storage in combination with cold heat pump

Basis: Hot storage

Steps:

1. storage losses for testing by cooling down the storage by cycling through an uninsulated pipe
2. reheating of the storage with the heat pump according to the controller setting, measurement of the storage temperature

Afterwards, a parameter identification of the COP characteristic and the thermal capacity of the heat pump can be done by these two measurements. Parameters of the heat pump are identified by optimisation in that way, that the temperature in the storage is fitted to the measured temperature.

C) Heating up of the storage after extraction of domestic hot water

=> check of identified parameters by reproducing the storage temperature

Basis: Hot storage

Steps:

1. Extraction of domestic hot water, water inlet < 20°C until compressor is switched on
2. measurement of storage temperature

3.8 Further work not covered under IEA HPP Annex 28

A key feature of the calculation method and the respective test procedure is a comparison of the calculated results with measured data to get an idea of the exactness of the testing and the calculation results as well as the impact of the different approaches applied. This comparison on the basis of the field measurements has started in Annex 28 and is still in progress. First results are presented in chapter 3.3. However, in the timeframe of IEA HPP Annex 28, not all results of field monitoring could be evaluated. Moreover, no simultaneous systems have been evaluated from field monitoring. It would be interesting to test a simultaneous system with the proposed method and have a field monitoring afterwards to perform the calculation and compare the results.

In any case further validation of the proposed testing and calculation and practical experience with the proposed methods will be a future task.

Furthermore, the following topics were not or not entirely covered by IEA HPP Annex 28, and are thus perspectives of further research work in connection with the work accomplished here:

- Testing for variable speed units, two step control as well as multi-compressor configurations
- Combined cooling/DHW operation, which is, for instance, to be covered by the EPBD, too
- Special testing requirements of natural refrigerants with transcritical cycle process, e.g. CO₂
- Certification under EU energy labelling directive (e.g. based on the SPF)

EU Mandate M/341 on Eco-Design of energy-using products

4 IMPLEMENTATION OF RESULTS

The results of the IEA HPP Annex 28 are intended to be implemented in the respective standards published by the international standardisation committees, as on level of European standardisation organisation CEN or on ISO level.

Thus, external liaisons have already been established with the respective Technical Committees of CEN, the TC 113 for the heat pump testing and the TC 228 for the calculation method of heating systems as described in chapter 1.5. As presently testing and calculation standards are under revision in Europe as described in chapter 1.1.2, there has been the opportunity of directly introducing the results in the standardisation process.

The calculation proposal has been introduced in the heat pump part of the standard prEN15316 (formerly prEN 14335) in the framework of the EPBD. This approach contains the described bin calculation procedure that has been presented in part 1 chapter A of the final report.

Updated results from the IEA HPP Annex 28 will be transferred after the six-month period for comments called “public enquiry” which was finished in March 2006, so the time schedule coincided with the completion of the IEA HPP Annex 28.

Concerning the working group for the revision of the heat pump DHW testing standard EN 255-3 after a long period of searching, the new head of the Swiss national test centre has been approved as Convenor for the working group. A start-up meeting of the working group took place in the end of April 2006. Transfer of results of the IEA HPP Annex 28 is simplified by the fact, that members of the national teams of France, Sweden and Switzerland are members of this working group, too.

Moreover, relevant results on the test of direct expansion heat pump systems accomplished by Austria will be used by the respective working group CEN/TC 113/WG 11, which has started working in the beginning of March 2005.

Last but not least, in the course of the revision of the national energy regulation, Germany has implemented a monthly-based bin-method based on the approach of IEA HPP Annex 28 in the national standard DIN V 18599.

5 CONCLUSION

IEA HPP Annex 28 had the objective of enabling the assessment of seasonal energy performance of different heat pump system solutions for fulfilling the requirements for space heating and domestic hot water. Actually, there is a variety of different heating systems on the market, but there is a lack of standardised, internationally uniform methods to compare energy performance in order to reward and promote environmentally sound technologies on the background of climate protection.

IEA HPP Annex 28 has delivered easy-to-use calculation methods and the necessary test procedures to deliver the required input data of the product characteristics for the calculation. The scope has been heat pump systems with alternate or simultaneous combined production of space heating and domestic hot water, which are becoming more and more interesting due to actual trends to highly insulated buildings and low-temperature combined operating space heating and DHW systems.

In this second part of the final report, the project background is presented. This comprises the description of the work accomplished in the framework of the IEA HPP Annex 28, in particular the national contributions. The Annex started with an analysis of the status quo of the systems on the market and the standardisation. A detailed documentation of results of the market and standardisation survey has been described in an Annex 28 Interim Report, concluding that the combined operation should be covered by an extension of existing standards.

The focus of this second part of the final report is to give an overview of the investigations, discussions and results, that led to the proposal for the testing and calculation presented in part 1. In that sense this part of the final report can be seen as documentation to the proposed test procedure and calculation method.

Moreover, since the proposals in part 1 are both based on European references due to the European background of most of the participant and the currently ongoing revision of European standards in the framework of the EPBD, the second part is to give the more international approach of North – American and Japanese testing and calculation, as well. The approach for the calculation and the testing in North America and Japan, which are presented in this report, indicate that there are not so big differences. Concerning calculation the bin method is a common basis to derive more international standards, e.g. on the ISO level. Since the data of the product characteristic which are necessary to perform the calculation are the same, testing can also be unified. Thus, remaining difference are mainly due to different system configurations and different user behaviour reflected e.g. in tapping profiles etc.. However, the black-box testing introduced in the IEA HPP Annex 28 can also be a common frame for uniform system testing for different system layouts. In that sense IEA HPP Annex 28 has prepared uniform testing and calculation standards by providing both information on the status-quo in the standardisation and an assessment of different approaches. Following work towards the uniform standards can be accomplished for instance on ISO level.

The proposed approaches have been tested in IEA HPP Annex 28. This process is not finished and will continue in the future to get more experience with the different applications.

Concerning the **test procedure**, proposed tests have been found to deliver reliable and reproducible results to perform the calculation. However, for a fair comparison some modifications of the testing may be required due to differing system configurations coming on the market.

Concerning the **calculation method**, calculation results have been compared to field monitoring results of the systems for ground source heat pumps and a compact unit. Results show a deviation in the seasonal performance in the range of $\pm 6\%$, which is satisfactory taking into account that simplifications have to be made in order to derive an easy-to-use method. However, further experiences with the proposed methods shall be gathered and is a continuous task.

Furthermore, not all aspects of the investigated systems could be covered, so there are tasks for further research in this field, e.g. for capacity controlled units or new refrigerants. In addition, systems currently tend to be more integrated to multi-functional systems – compact units have already been included in the Annex 28 work - so future extensions to calculation and testing will be necessary to follow the market developments.

6 ACKNOWLEDGEMENT

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It has to be emphasized that the IEA HPP Annex 28 is a team work and the results presented in this report are based on the effort and contribution of the different member countries.

Hence, respect and thanks are expressed to all participants of the IEA HPP Annex 28 for the valuable contributions and for the constructive discussion and co-operation.

Special thanks are expressed to

- the Austrian National team for the layout and maintenance of the IEA HPP Annex 28 website at the URL <http://www.annex28.net> as participant's fee of the Annex
- the Executive Committee of the Heat Pump Programme for the opportunity to present the final results on a workshop in the framework of the 8th International Heat Pump Conference in Las Vegas in May 2005
- the IEA Heat Pump Centre for the promotion for this workshop
- the American National team for the organisational support before and during the workshop

7 NOMENCLATURE

Variables

The nomenclature is derived from EN 12831 and prEN15316.

θ	Celsius temperature	°C
η	Efficiency	-
φ	Relative humidity	-
Φ	Temperature change coefficient	-
ϕ	Heating capacity, thermal power	W
Δ	Difference of two values	-
A	Surface area	m ²
COP	Coefficient of performance	W/W
c_p	Specific heat capacity	J/(kgK)
HDH	Heating degree hour	Kh
m	Mass	kg
\dot{m}	Mass flow rate	kg/s
P	Electrical power	W
PF	Performance factor	-
Q	Quantity of heat, energy	J
q	Volume flow rate	m ³ /h
SPF	Seasonal performance factor	-
U	Heat loss coefficient	W/(m ² K)
t	Time, period of time	S
W	Electrical energy	J
X	Ratio $Q_{out,g,h}/Q_{out,g,DHW}$	-
w	Weighting factor	-

Indices

aux	Auxiliary
bl	Building load (demand)
bu	Back-up
combi	In combined operation
comp	Compressor
cw	Cold water
ctrl	Control
cyc	Cyclic
d	Demand

Indices (continued)

DHW	Domestic hot water (mode)
dp	Dewpoint
EN 255-2	Refers to EN 255-2
EN 255-3	Refers to EN 255-3
ea	Exhaust air
env	Environment
g	Generation subsystem, Generator
H	Space heating according to EN 14511-1
H,hs	Space heating including supplementary heating (not corrected to 0 kPa for pump and fan efficiencies)
h	Space Heating (mode)
hp	Heat pump
hps,tot	Not corrected to 0 kPa for pump and fan efficiencies
hr	Heat recovery
i	Refers to bin i
in	Inlet to the system
l	Loss
lower	Lower limit
oa	Outside air
out	Outlet of the system (used energy)
ON	In operation
phase 2	Refers to phase 2 of EN 255-3 cycle
phase 4	Refers to phase 4 of EN 255-3 cycle
ra	Return air (building outlet)
s	Storage
sa	Supply air
sin	Single operation mode
SH	Space heating
so	Source
sys	System
t	Total
tot	Total
t,s	Domestic hot water including stand-by power input
upper	Upper limit
used	Used fraction of energy, used level of temperature
w	Water

Indices (continued)

wb	Wet bulb
0	From heat source

Nomenclature from ASHRAE/ARI standards

δ	Factor to consider fraction of back-up heating operation	-
BL	Building load	Btu/h/1000
COP_{cyc}	Cyclic coefficient of performance	(Btu/h)/W
CPF_{hs}	Combined performance factor heating season	Btu/Wh
CPF_{cs}	Combined performance factor	Btu/Wh
CPF_{ws}	Combined performance factor domestic hot water-only operation	Btu/Wh
E	Electrical energy input to heating	Wh
\dot{E}	Electrical power input	W
EF	Energy factor according to ASHRAE 118.2	-
ER	Electrical input to resistance water heater	W
F_{def}	Factor for correction of defrost losses	-
HSPF	Heating seasonal performance factor	Btu/Wh
HLF	Heating load factor	-
N	Total number of hours in a given season	h
η_j	Fractional bin hours	-
PLF	Part load factor	-
q	Total space conditioning provided	Btu
q_w	Hot water energy	Btu
\dot{q}	Heating capacity	Btu/h
RH	Resistance heating	kW
t_j	Temperature of bin j	°F

Nomenclature from EN 255-3

COP_t	COP for tapping sanitary hot water	-
P_{es}	effective electrical power input during the standby period	W
Q_t	heat energy extracted with the DHW during phase 2 of EN 255-3	kWh
t_s	measuring time for standby power input (phase 4)	s
t_t	Duration of tapping and reheating time period of the evaluated draw off (phase 2)	s
W_{es}	effective electrical energy input during standby period (phase 4)	kWh
W_{et}	effective electrical energy input during tapping period (phase 2)	kWh

Abbreviations / Websites

A/A	Air-to-Air heat pump
A/W	Air-to-water heat pump
ARI	Air-Conditioning and Refrigeration Institute (http://www.ari.org)
ASHRAE	American Society of heating, Refrigeration and Air-Conditioning Engineers (http://www.ashrae.org)
B/W	Brine-to-water heat pump
CEN	Comité Européen de Normalisation – European Standardisation Committee (http://www.cenorm.be)
COP	Coefficient of performance
CPF	Combined performance factor
CSA	Canadian Standard Association
DB	Dry bulb air temperature
DHW	Domestic hot water
DIBt	Deutsches Institut für Bautechnik (German Institute for building technologies) (http://www.dibt.de)
DIN	Deutsches Institut für Normung (German Standardisation Institute) (http://www.din.de)
DX	Direct expansion
EAHP	Exhaust-air heat pump
EN	Euronorm - European Standard
EUEB	European Union Eco labelling board
EPBD Directive)	EU Directive on the Energy Performance of Buildings (Energy Performance Building Directive)
JRA	Japanese Standard
HPWH	Heat pump water heater
HSPF	Heating seasonal performance factor
HPTCJ	Heat Pump and Thermal Storage Centre Japan (http://www.hptcj.or.jp/about_e)
NT	Nordtest (http://www.nordtest.org)
ORNL	Oak Ridge National Laboratory (http://www.ornl.gov)
prEN	Provisional European Standard
SEER	Seasonal Energy Efficiency Ratio
SH	Space heating
SPF	Seasonal performance factor
SPF-HP	Seasonal performance factor for the system boundary heat pump
SPF-G	Seasonal performance factor for the system boundary generator
SPF-S	Seasonal performance factor for the system boundary system
SP	Swedish National Test and Research Institute (http://www.sp.se)
WB	Wet bulb air temperature
WPZ	Wärmepumpentestzentrum (Swiss heat pump test centre (since 2004 at NTB Buchs) (http://www.wpz.ch)

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