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Heating systems in buildings — Method for calculation of system energy requirements and system efficiencies — Part 4-2: Space heating generation systems, heat pump systems

Heizsysteme in Gebäuden — Rechenmethode für den Energiebedarf und die Systemeffizienz — Teil 4-2: Heizsysteme mit Wärmepumpe

Systèmes de chauffage en batiment — Méthode de calculation des besoins énergétiques et d'efficacité des systèmes — Partie 4-2 : Systèmes de chauffage utilisant les pompes à chaleur

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Contents

Foreword

This document (prEN 15316-4-2:2006) has been prepared by Technical Committee CEN/TC 228 “Heating systems in buildings”, the secretariat of which is held by DS.

The subjects covered by CEN/TC 228 are the following:

- design of heating systems (water based, electrical etc.);
- installation of heating systems;
- commissioning of heating systems;
- instructions for operation, maintenance and use of heating systems;
- methods for calculation of the design heat loss and heat loads;
- methods for calculation of the energy performance of heating systems.

Heating systems also include the effect of attached systems such as hot water production systems.

All these standards are systems standards, i.e. they are based on requirements addressed to the system as a whole and not dealing with requirements to the products within the system.

Where possible, reference is made to other European or International Standards, a.o. product standards. However, use of products complying with relevant product standards is no guarantee of compliance with the system requirements.

The requirements are mainly expressed as functional requirements, i.e. requirements dealing with the function of the system and not specifying shape, material, dimensions or the like.

The guidelines describe ways to meet the requirements, but other ways to fulfil the functional requirements might be used if fulfilment can be proved.

Heating systems differ among the member countries due to climate, traditions and national regulations. In some cases requirements are given as classes so national or individual needs may be accommodated.

In cases where the standards contradict with national regulations, the latter should be followed.

Introduction

This standard is part of a series of standards on the methods for calculation of system energy requirements and system efficiencies. The framework for the calculation is described in the general part (prEN 15316-1).

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

This standard presents methods for calculation of the additional energy requirements of a heat generation system in order to meet the distribution subsystem demand. The calculation is based on the performance characteristics of the products given in product standards and on other characteristics required to evaluate the performance of the products as included in the system. Product data, e.g. heating capacity or COP of the heat pump, shall be determined according to European test methods. If no European methods exist, national methods can be used.

This method can be used for the following applications:

— judging compliance with regulations expressed in terms of energy targets;

— optimisation of the energy performance of a planned heat generation system, by applying the method to several possible options;

— assessing the effect of possible energy conservation measures on an existing heat generation system, by calculating of the energy use with and without the energy conservation measure.

Only the calculation method is normative. The user shall refer to other European Standards or to national documents for input data. Additional values necessary to complete the calculations are to be given in a national Annex, if no national annex is available, default values are given in an informative Annex where appropriate.

1 Scope

The standard covers heat pumps for space heating, heat pump water heaters (HPWH) and heat pumps with combined space heating and domestic hot water production in alternate or simultaneous operation, where the same heat pump delivers the heat to cover the space heating and domestic hot water heat requirement.

The scope of this part is to standardise the:

— required inputs;

— calculation methods;

— required outputs

for heat generation for space heating and domestic hot water production of the following heat pump systems, including control:

— electrically-driven vapour compression cycle (VCC) heat pumps;

— combustion engine-driven vapour compression cycle heat pumps;

— thermally-driven vapour absorption cycle (VAC) heat pumps,

using combinations of heat source and heat distribution listed in Table 1

Table 1 — Heat sources and heat distribution in the scope of this part

Heat source	Heat distribution
Outdoor air	Air
Exhaust-air	Water
Indirect ground source with brine distribution	Direct condensation of the refrigerant in the appliance (VRF)
Indirect ground source with water distribution	
Direct ground source (Direct expansion (DX))	
Surface water	
Ground water	

2 Normative references

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) 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, *Heat exchangers — Testing procedures for the determination of performance criteria of air/air and air/exhaust gas heat recovery plants*

EN 12309-2:2000, *Gas-fired absorption and adsorption air-conditioning and/or heat pump appliances with a net heat input not exceeding 70 kW*

EN ISO 7345, *Thermal insulation – Physical quantities and definitions (ISO 7345:1995)*

EN ISO 13790 *Thermal performance of buildings – Calculation of energy use for space heating (ISO 13790:2004)*

EN 14511:2004, *Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling*

CEN/TS 14825:2003, *Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling — Testing and rating at part load conditions*

prEN 15316-1 *Heating systems in buildings – Method for calculation of system energy requirements and system efficiencies – Part 1: General*

prEN15316-2-3, *Heating systems in buildings – Method for calculation of system energy requirements and system efficiencies – Part 2.3: Space heating distribution systems*

prEN15316-3-1, *Heating systems in buildings – Method for calculation of system energy requirements and system efficiencies – Part 3.1: Domestic hot water systems – characterisation of needs (tapping requirements)*

prEN15316-3-2, *Heating systems in buildings – Method for calculation of system energy requirements and system efficiencies – Part 3.2: Domestic hot water systems – distribution*

prEN15316-4-1, *Heating systems in buildings – Method for calculation of system energy requirements and system efficiencies – Part 4.1: Space heating generation systems, combustion systems*

prEN 15927-6, *Hygrothermal performance of buildings – Climatic data – Part 6: Calculation and presentation of accumulated time–temperature differences, for assessing energy use in space heating*

3 Terms and definitions

3.1 Definitions

For the purposes of the present standard, the definitions in EN ISO 7345:1995 and the following definitions apply:

3.1.1
alternate operation
production of heat energy for the space heating and domestic hot water system by a heat generator with double service by switching the heat generator either to the domestic hot water operation or the space heating operation

3.1.2
application rating conditions
mandatory rated conditions within the operating range of the unit that are published by the manufacturer or supplier

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

3.1.4
balance point temperature
temperature at which the heat pump heating capacity and the building heat load are equal

3.1.5
bin
a statistical temperature class (sometimes a class interval) for the outdoor air temperature, with the class limits expressed in a temperature unit

3.1.6
calculation period
time period considered for the calculation of the heat losses of the space heating and/or domestic hot water system

3.1.7
coefficient of performance COP
ratio of the heating capacity to the effective power input of the unit

3.1.8
cumulative frequency
frequency of the outdoor air temperature cumulated over all 1 K bins

3.1.9
cut-out period
time period in which the electricity supply to the heat pump is interrupt by the supplying utility.

3.1.10
effective power input
average power input of the unit within the defined interval of time obtained from:

- the power input for operation of the compressor or burner 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

3.1.11
electrically-driven heat pump
in the frame of this standard, electrically-driven heat pumps denote vapour compression cycle heat pumps, which incorporate a compressor that is driven by an electric motor

3.1.12
frequency
the (statistical) frequency of an event is the number of times the event occurred in the sample. The frequencies are often graphically represented in histograms. In the frame of this standard the frequency of the outdoor air temperature is evaluated based on a sample of hourly-averaged data for one year.

3.1.13
heat demand, building
heat to be delivered to the heated space to maintain the internal set-point temperature of the heated space

3.1.14
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 alternate or simultaneous combined operation

3.1.15
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.

3.1.16
heat transfer medium
any medium (water, air, etc.) 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

3.1.17
heated space
room or enclosure heated to a given set-point temperature

3.1.18
heating system heat losses, total
total of the heat losses from the heating system, including recoverable heat loss

3.1.19
heating capacity Φg
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

3.1.20
low temperature cut-out
temperature at which heat pump operation is stopped and the total heat requirements are covered by a back-up heater

3.1.21
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

3.1.22
part load operation
operation state of the heat pump, where the average load requirement of the distribution subsystem is below the average heating capacity of the heat pump in the calculation period

3.1.23
part load ratio
the ratio between the generated heat during the calculation period and the maximum possible output from the heat generator during the same calculation period

3.1.24
primary pump
pump mounted in the circuit containing the generator and hydraulic decoupling, e.g. a heating buffer storage in parallel configuration or a hydraulic distributor

3.1.25
produced heat
Heat produced by the generator subsystems, i.e. the heat produced to cover the energy requirement of the distribution subsystem and the generator subsystem heat losses for space heating and/or domestic hot water.

3.1.26
recoverable system heat loss
part of the system heat loss, from the space heating and domestic hot water system, which may be recovered to lower the heat demand for space heating

3.1.27
recovered system heat loss
part of the recoverable system heat loss which contributes to meet the heat demand of the space

3.1.28
seasonal performance factor SPF
in the frame of this standard the ratio of the total annual energy delivered to the distribution subsystem for space heating and/or domestic hot water to the total annual input of driving energy (electricity in case of electrically-driven heat pumps and fuel/heat in case of engine-driven heat pumps or absorption heat pumps) plus the total annual input of auxiliary energy.

3.1.29
simultaneous operation
simultaneous production of heat energy for the space heating and domestic hot water system by a heat generator with double service, e.g. by refrigerant desuperheating or condensate subcooling

3.1.30
standard rating condition
mandatory condition that is used for marking and for comparison or certification purposes

3.2 Symbols and abbreviations

For the purposes of this document, the following symbols and units (Table 2) and indices (Table 3) apply. Abbreviations are listed in Table 4:

Table 2 — Symbols and Units

Symbol	Name of quantity	Unit
ϕ	Thermal capacity or electrical power input	W
η	Efficiency	-
θ	Celsius temperature	°C
ρ	Density	kg/m^3
$\Delta\theta$	Temperature difference, - spread	K
Δp	Pressure drop	Pa
c	Specific heat capacity	J/(kg·K)
CHDH	Cumulative heating degree hours	Kh
COP	Coefficient of performance	W/W
E	Quantity of energy, electricity	J
f	Correction factor	-
FC	Load factor	-
G	Quantity of energy, fuel	J
HDH	Heating degree hour	Kh
k	Recovered fraction of auxiliary energy	-
$\dot{m}$	Mass flow rate	kg/s
n	Number of quantity	-
N	Cumulative number of quantity	-
p	Fraction of quantity	-
Q	Quantity of heat	J
SPF	Seasonal performance factor	-
t	Time, period of time	S
V	Volume	m^3
$\dot{V}$	Volume flow	m^3/s
W	Quantity of energy, auxiliary	J
w	Weighting factor	-

Table 3 — Indices

$\Delta\theta$	temperature corrected	es	storage values acc. to EN 255-3 phase 4	nom	nominal
θlower	lower temperature limit	f	flow	oa	outdoor, outdoor air
θupper	upper temperature limit	g	generation subsystem	OD	outdoor at design Conditions
amb	ambient	h	space heating	ON	running, in Operation
aux	auxiliary	hot	hot process side	op	operation, operation limit
avg	average	hp	heat pump	out	output from system
bp	balance point	hw	hot water	r	return
bu	back-up (heater)	i	referring to bin i	rd	recovered
C	Carnot	ID	indoor at design conditions	rl	recoverable
cap	lack of capacity	in	consumed by system	s	storage
co	cut-out	int	internal	sb	stand-by
cold	cold process side	j	counting variable	si	sink
combi	simultaneous operation	k	counting variable	sin	single (operation)
d	distribution subsystem	l	loss	so	source
D	at design conditions	ltc	low temperature cut-out	standard	acc. to standard testing
DHW	domestic hot water	max	maximum	t	total
eff	effective	n	nominal	w	water
eng	engine	nh	non recoverable		

The indices are separated by a comma.

Table 4 — Abbreviations

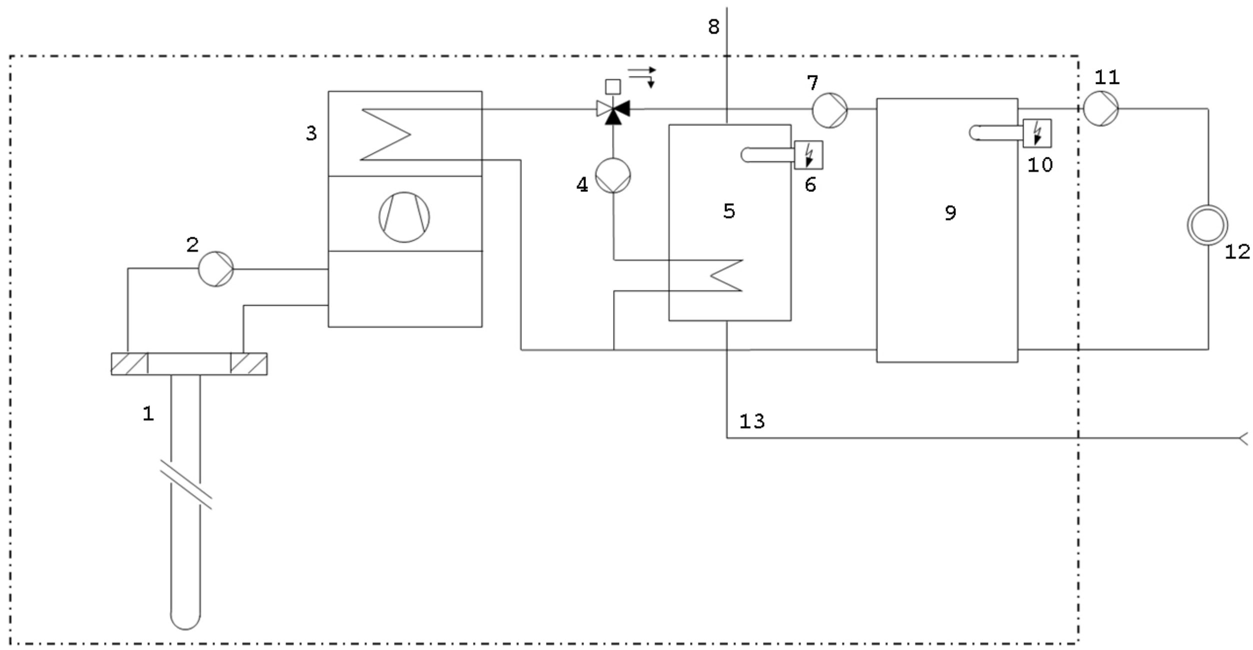
ATTD	Accumulated time-temperature difference
DHW	Domestic hot water
SH	Space heating
TTD	Time-temperature difference
VCC	Vapour compression cycle
VAC	Vapour absorption cycle

4 Principle of the method

4.1 Heat balance of generation sub-system

System boundary

The system boundary defines the components of the entire heating systems that are considered in this standard. For the heat pump generation subsystem the system boundary comprises the heat pump, the heat source system, attached internal and external storages and attached electrical back-up heaters. Auxiliary components connected to the generation subsystem are considered, as long as no transport energy is transferred to the distribution subsystem. For fuel back-up heaters the required back-up energy is determined in this part of prEN 15316, however, the efficiency calculation shall be accomplished according to the respective other parts of prEN 15316 (see chapter 4.6). The system boundary is depicted in Figure 1.



Key

1 heat source system (here: vertical borehole heat exchanger)
2 Source pump
3 Heat pump
4 DHW storage loading pump
5 DHW storage
6 DHW back-up heater
7 Primary pump
8 DHW hot water outlet
9 Heating buffer storage
10 Space heating back-up heater
11 Circulation pump space heating distribution system
12 Heat emission system
13 DHW cold water inlet

Figure 1 — System boundary of the generation subsystem

Physical factors taken into account:

The calculation method takes into account the following physical factors, which have an impact on the seasonal performance factor and thereby on the required energy input to meet the heat requirements of the distribution system

- type of generator (monovalent, bivalent)
- type of heat pump (driving energy (e.g. electricity or fuel), thermodynamic cycle (VCC, VAC))
- combination of heat source and sink (e.g. ground-to-water, air-to-air)
- space heating and domestic hot water energy requirements of the distribution subsystem
- effects of variation of source and sink temperature on heating capacity and COP according to standard product testing
- effects of compressor control in part load operation (ON-OFF, stepwise, variable speed units) as far as they are reflected in the heating capacity and COP according to standard testing or further test results on part load operation.
- auxiliary energy input needed to operate the generation subsystem not considered in standard testing of heating capacity and COP
- system heat losses due to space heating or DHW storage components including the connecting pipework
- location of the generation subsystem

Calculation structure:

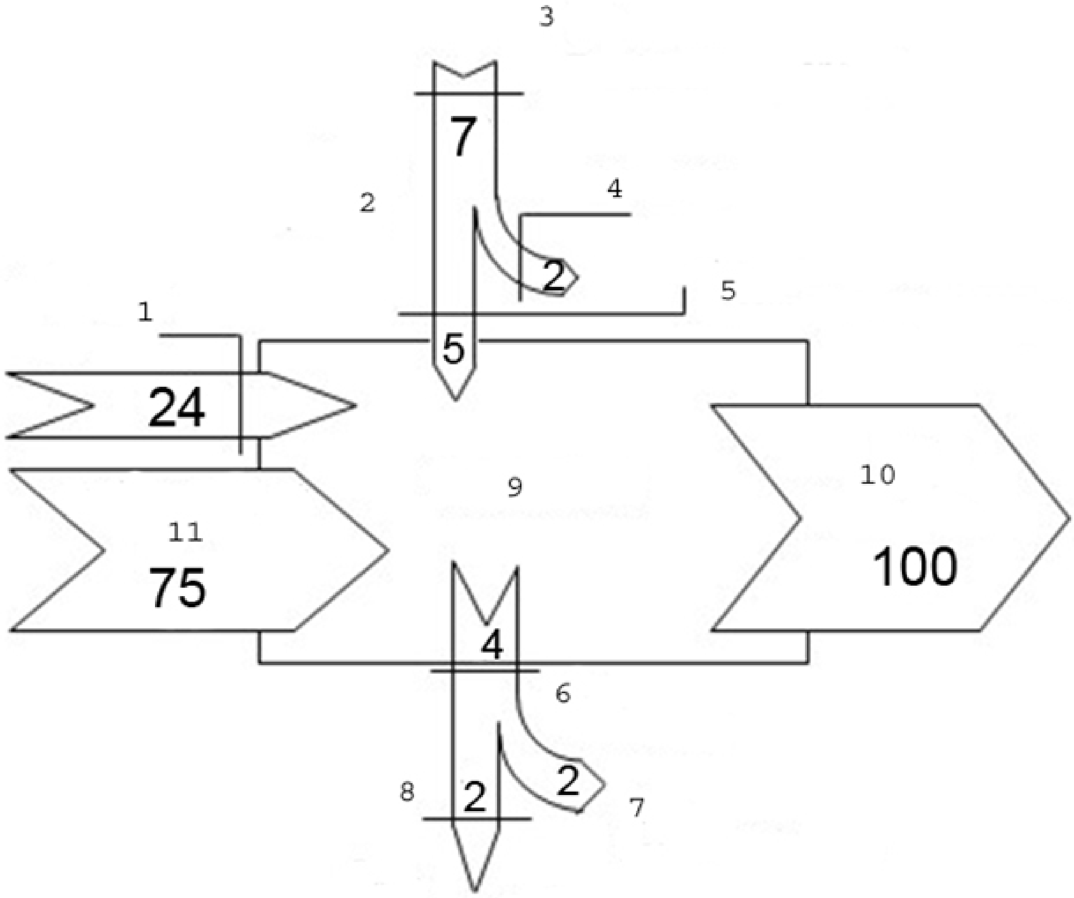
The calculation is performed considering the following input data:

- type, configuration and design of the generation subsystem;
- type of control of the generation subsystem;
- ambient conditions (outdoor air temperature, variation of source and sink temperature in the year)
- heat requirements for space heating and/or domestic hot water

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

- required energy input as electricity $E_{in,g}$ or driving energy $G_{in,g}$ (fuel, waste heat, solar heat) to meet the space heating and/or domestic hot water requirements
- total generator heat loss $Q_{l,g}$
- total recoverable generator heat losses $Q_{rl,g}$
- total required auxiliary energy W_g to operate the generation subsystem;

The following heat balance depicted in Figure 2 can be made for the generation subsystem (see Figure 1).



Key

1 Energy input to cover the heat requirement (e.g. electricity, fuel)
2 Auxiliary energy
3 Total auxiliary energy W_g
4 Unrecovered auxiliary energy $(1-k_{g,rd})\cdot W_g$
5 Recovered auxiliary energy $k_{g,rd}\cdot W_g$
6 Total losses $Q_{l,g}$
7 Recoverable losses $Q_{rl,g}$
8 Unrecoverable losses $Q_{nh,g}$
9 Generation subsystem
10 Heat requirement of the distribution subsystem $Q_{out,g} = Q_{in,d}$
11 Ambient heat $Q_{in,g}$ used as heat source of the heat pump

Figure 2 — Energy balance of generation subsystem

The numbers indicated in Figure 2 refer to the percentage of the energy flows to cover the distribution subsystem heat requirement (100 %). They are intended to give an idea of the size of the respective energy flows. The numbers vary dependent on the physical factors listed before. The numbers given in Figure 2 refer to an electrically-driven ground-source heat pump in monovalent space heating-only operation including a heating buffer storage.

4.2 Energy input needed to meet the heat requirements

4.2.1 Electrically-driven heat pumps

The energy balance for the electrically-driven generation subsystem is given by

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

eq. 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)
$Q_{l,g}$	losses of the generation subsystem	(J)
$Q_{in,g}$	ambient heat energy used as heat source of the heat pump	(J)
$k_{g,rd}$	recovered fraction of heat energy from the auxililaries	(-)
W_g	auxiliary energy input to operate the generation subsystem	(J)

The term $E_{in,g}$ is the electrical energy input to cover the heat requirement of the distribution subsystem. It comprises the electrical energy input to the heat pump and possibly installed electrical back-up heaters. Since for electrically-driven heat pumps the energy input to the heat pump is calculated based on the standard testing according to EN 14511, $E_{in,g}$ also includes the fractions of the auxiliary energies included in the COP. According to EN 14511 the auxiliary energies at the system limit of the heat pump are taken into account, i.e. the energy for control and safety devices during operation, the proportional energy input for pumps and fans to ensure the transport of the heat transfer media inside the unit as well as eventually energy for defrost operation and additional heating devices for the oil supply of the compressor (carter heating).

Thus, W_g only comprises the fractions not included in the COP according to the standard testing. $k_{g,rd}$ describes the fraction of auxiliary energy, which is recovered as thermal energy, e.g. for pumps where a fraction of the auxiliary energy is directly transferred to the heat transfer medium as thermal energy. This fraction is already contained in the COP according to EN 14511 for electrically-driven heat pumps, so $k_{g,rd} = 0$.

Heat losses $Q_{l,g}$ of the heat pump over the envelope are neglected unless heat loss values of the heat pump are known. For systems with integrated or external heating buffer or DHW hot water storage, generation subsystem losses in form of storage heat losses are considered including the connecting circulation pipes to the storage.

4.2.2 Engine-driven and absorption heat pumps

For engine-driven and absorption heat pumps, the energy balance for the generation subsystem is given by:

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

eq. 2

where

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

$G_{in,g}$ describes the driving energy input to cover the heat requirement of the distribution subsystem. For combustion engine-driven heat pumps, this driving energy is fuel, e.g. as diesel or natural gas. For thermally-driven absorption heat pumps, fuel-driven burners, but also solar energy or waste heat can be the driving energy input.

$Q_{out,g}$, the heat energy output of the generation subsystems equals the heat requirement of the distribution subsystem and contains all fractions of heat recovered from the engine or the flue gas of the engine, i.e.

recovered heat from the engine in entirely considered within the system boundary of the generation subsystem.

$k_{g,rd}$ gives the fraction of the auxiliary energy recovered as thermal energy and depends on the test method. As in the case of electrically-driven heat pumps $k_{g,rd} = 0$ if the recovered heat is already included in the COP.

4.3 Auxiliary energy W_g

Auxiliary energy is energy needed to operate the generation subsystem, e.g. the source pump or the control system of the generator. As for electrically-driven heat pumps heating capacity and COP in this standard are calculated on the basis of results from product testing, according to EN 14511, only the auxiliary energy not included in the test results, e.g. the power to overcome the external pressure drop and the power in stand-by operation, shall be considered in W_g.

Auxiliary energy is accounted to the generation subsystem as long as no transport energy is transferred to the distribution system. That means, in general the circulation pump is account to the distribution subsystem, unless a hydraulic decoupling exist. For a hydraulic decoupling between the generation and various distribution systems, e.g. by a heating buffer or domestic hot water storage in parallel configuration, the primary pump is accounted to the generation subsystem, as well.

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.

4.4 Recoverable, recovered and unrecoverable heat losses

The calculated losses are not necessarily lost. Parts of the losses are recoverable, and parts of these recoverable losses are actually recovered. The recovered losses are determined by the location of the generator and the utilisation factor (gain/loss ratio, see EN ISO 13790).

Recoverable heat losses $Q_{g,rl}$ are e.g. heat losses through the envelope of a generation subsystem, e.g. in form of storage losses when the storage is installed in the heated space. For a generation subsystem installed outside the heated space, however, the heat losses through the envelope of the generator are not recoverable. Flue gas losses of fuel engine-driven heat pumps are considered not recoverable, since all recovered flue gas losses inside the generation subsystem limits are contained in the heat output $Q_{out,g}$.

4.5 Calculation periods

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

However, an appropriate processing of the meteorological data may be used to carry out the calculation with monthly or hourly averaged values, if necessary.

NOTE Exactness of measured COP values according to EN 14511 for electrically-driven heat pumps are in the range of 5%. Comparison of a bin calculation described in chapter 5.2 on an annual basis and field monitoring values delivered an exactness of the calculation in the range of 6%. So, with the regard to the expense for the computation an annual or monthly approach seems sufficient.

4.6 Calculation by zones

A heating system may be split up in zones with different distribution systems. A separate circuit may be used for domestic hot water production.

Several heat generation systems may be available.

The total heat requirement of all the distribution subsystems shall equal the total heat output of the generation subsystems:

$$\sum_j Q_{out,g,j} = \sum_k Q_{in,d,k} \quad [J]$$

eq. 3

where

$Q_{out,g,j}$ heat energy requirement to be covered by generator j (J)

$Q_{in,d,k}$ heat energy requirement of distribution system k (J)

When more generation systems are available (multivalent system configuration), the total heat demand of the distribution system(s) $Q_{in,d,t}$ shall be distributed among the available generation systems and the calculation described in chapter 5 shall be performed independently for each generation system j on the basis of $Q_{out,g,j}$. This is accomplished in case of an installed back-up heater.

For intermittent heating, the requirements of EN ISO 13790 shall be considered. These are considered already in the calculation of the heat requirements according to the general part prEN 15316-1 or the emission part prEN 15316-2-1 of this standard, respectively.

4.7 Heat pumps with combined space heating and domestic hot water production

For combined operation of the heat pump for space heating and domestic hot water production, two kinds of operation modes can be distinguished, alternate and simultaneous operation.

In alternate operation the heat pump switches 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 1 with a domestic hot water storage in parallel. Domestic hot water operation is usually given priority, i.e. space heating operation is interrupted in case of domestic hot water heat demand.

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

— Desuperheating and/or condensate subcooling;

— cascade cycles with internal heat exchangers.

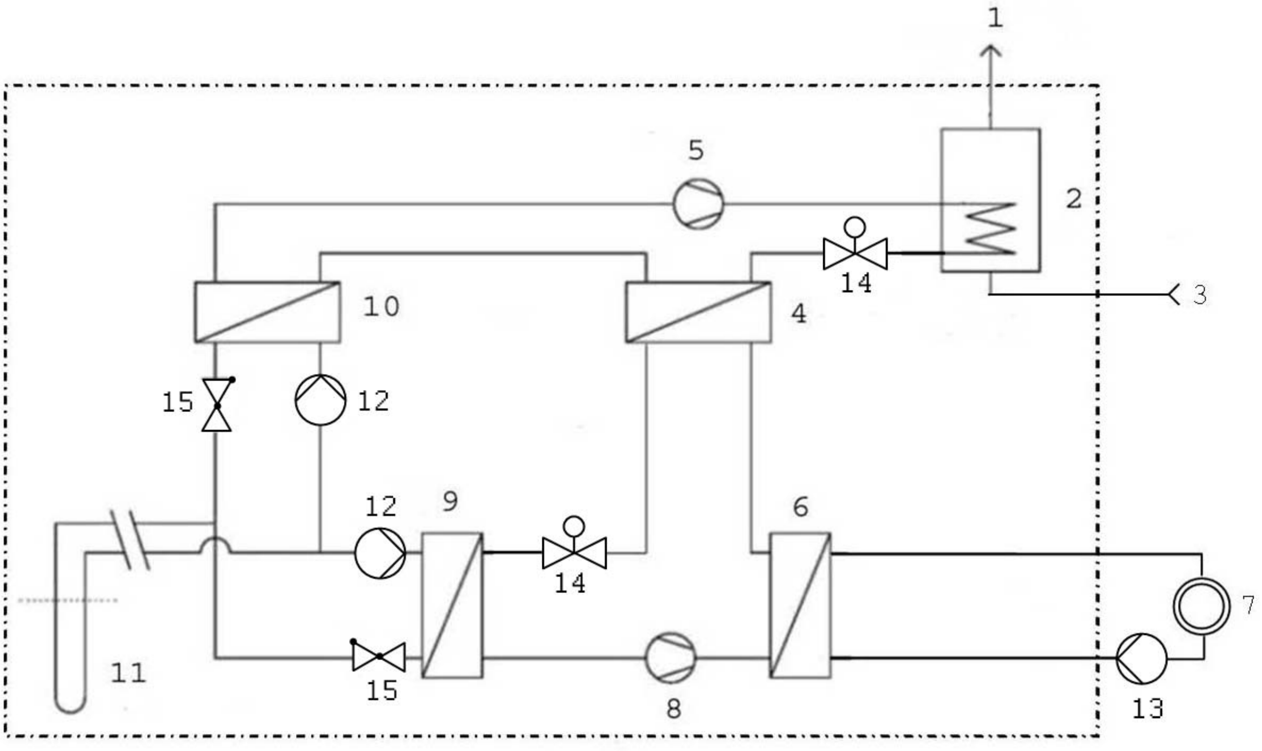
For these simultaneous concepts, space heating and domestic hot water requirements are covered at the same time. Figure 3 gives a sample of a hydraulic scheme of a simultaneous operating system using a cascade cycle with condensate subcooling.

For simultaneous system layout, 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 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 in operation, e.g. in the configuration shown in Figure 3, only the upper stage heat pump is in operation and the heat is extracted from the ground source (summer operation, no space heating demand)

— simultaneous operation:
both space heating and domestic hot water operation. For the configuration shown in Figure 3, both stages are in operation. The heat for the lower stage heat pump is taken from the ground source and 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)



Legend

1 Hot water outlet of the DHW system
2 Condenser of upper stage in DHW storage
3 Cold water inlet of the DHW system
4 Condensate subcooler
5 Compressor upper stage
6 Condenser lower stage
7 Heat emission system
8 Compressor lower stage
9 Evaporator
10 Heat exchanger to ground source upper stage
11 Vertical borehole ground heat-exchanger
12 Source pump
13 Circulation pump heating system
14 Expansion valve
15 Non-return valve

Figure 3— sample system with simultaneous operating heat pump with cascade cycle layout using condensate subcooling for domestic hot water production

The calculation used in this standard implies, that both the single operation modes and the simultaneous operation are tested according to standard testing, so heating capacity- and COP characteristic of all three respective operation modes are available. As heating capacity and COP characteristic of the simultaneous operation may differ significantly from the other two operation modes, these test results have to be available and taken into account.

5 Generation system calculation

In this standard, two performance calculation methods for the generation subsystem are described corresponding to different applications (simplified or detailed estimation). The two methods differ with respect to:

- required input data
- operation conditions taken into account
- calculation periods

The two methods and their field of application are shortly described in the following:

<u>Simplified seasonal performance method based on system typology, see 5.1 (tabulated values)</u>

For this method, the considered calculation period is the heating season. The performance is chosen from tabulated values for fixed performance classes of the heat pump, based on test results according to heat pump test standards, e.g. EN 14511 for electrically-driven heat pumps. The operation conditions (climate, design and operation of the heating system, heat source type) are based on typology of implementation characteristics and are not case specific. This method allows a country/region specific approach and requires a country/region specific national annex. Therefore, if there is no appropriate national annex available with the adapted values, this method cannot be used.

The tabulated values are in particular useful, if limited information on the generation subsystem exists, as it may be the case for existing buildings where for instance the COP of the heat pump has to be estimated.

<u>Detailed case specific calculation based on component efficiency data, see 5.2 (Bin-method)</u>

This method is also based on the test results according to heat pump test standards, e.g. EN 14511 for electrically-driven heat pumps, but supplementary data are needed in order to take into account the specific operation conditions of each individual installation. Therefore, the calculation period is split-up in bins dependent on the outdoor air temperature. The calculation is carried out for the corresponding bin operation conditions of the heat pump. The method shall be carried out with product data for the heating capacity and the COP. Example values given in the informative Annex F illustrate the data needed to perform the calculation.

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

The calculation method to be applied can be chosen dependent on the data available and the objectives of the user.

5.1 Simplified seasonal performance method based on system typology (system typology method)

5.1.1 Principle of the system typology method

This method assumes that:

- climatic conditions;
- design and operation of the heating system, including typical occupancy patterns of the relevant building sector;

— type of heat source

have been considered and incorporated in a procedure to convert standard test results of heat pump efficiency according to EN 14511 for electrically driven heat pumps into a seasonal performance factor (SPF) for the relevant building sector, e.g. domestic and non-domestic.

The steps within the simplified seasonal performance calculation procedure are:

(i) adapt test results for uniformity, taking account of the type of heat pump and the type of energy input;

(ii) adjust for seasonal performance at installed conditions, taking account of the climatic conditions, the design and operation of the heating system and the type of heat source;

(iii) deliver results (annual energy consumption, generation heat loss, auxiliary energy consumption, total recoverable generation heat loss, optionally SPF).

Thereby, the procedure allows for national characteristics of the relevant building sector.

For some heating systems, buffer storage vessels are applied to diminish heat pump cycling. These storage systems are considered to be part of the generation subsystem and their losses are taken into account in the generation subsystem, regardless if the storage vessels are an integral part of a specific heat pump and included in heat pump testing or are located external. For integral storages their losses may be included in the COP / SPF of the heat generation system depending on the testing applied. Storage systems for domestic hot water are also part of the generation subsystem.

In order to provide consistent values within this part of the standard, the tabulated values of the national annex shall be produced using the detailed method, e.g. the Bin-method described in 5.2 for the fixed boundary conditions of the different performance classes and building typologies, respectively. As the tabulated values are simplified values intended as conservative estimation of the energy input to the system, the tabulated values of the system typology method shall not deliver better values than the detailed calculation with the Bin-method.

5.1.2 Calculation procedure of the system typology method

Selection of appropriate seasonal performance

A seasonal performance factor is selected from the appropriate national annex on the basis of the following information:

— country/region (climate) in which the building is situated;

— building sector (residential building, non-residential building, industrial ,etc);

If there is no appropriate national annex, this method cannot be used. Annex E (informative) is an example of a national annex of tabulated values of seasonal performance factors (including consideration of a possibly installed back-up heater) for residential and non-residential buildings in the Netherlands.

Input information required for the simplified seasonal performance method

Input information for the procedure may consist of:

— heat pump function (space heating, domestic hot water production, combination);

— type of heat pump (electrically-driven, engine-driven, etc);

— type of energy input (electricity, natural gas, LPG, oil, etc);

— type of heat source;

- source pump or fan power;
- test results produced in accordance with standard tests, e.g. according to EN 14511 for electrically-driven heat pumps;
- heating capacity
- internal heating system storage included in efficiency tests (yes/no)
- internal domestic hot water storage characteristics (volume / dimensions, specific loss).

Output information obtained from the seasonal performance method

The output information for the procedure consists of:

- total annual energy consumption of the generation subsystem;
- total heat loss of the generation subsystem;
- auxiliary consumption;
- total recoverable losses of the generation subsystem;
- optionally, the seasonal performance factor;

5.2 Detailed case specific seasonal performance method based on component efficiency (Bin method)

5.2.1 Principle of the bin method

The required energy input $E_{in,g}$ according to eq. 1 to cover the heat requirement of the distribution subsystem, e.g. the electricity input for electrically-driven heat pumps, can be determined according to the equation

$$E_{in,g} = \sum_{i} \frac{Q_{out,g,i} + Q_{l,g,i}}{COP_i} \quad [J]$$

eq. 4

where

$E_{in,g}$	electrical energy input to cover the heat requirement of the distribution subsystem and generation subsystem losses	(J)
$Q_{out,g,i}$	heat energy requirement of distribution system for period of defined operation conditions	(J)
$Q_{l,g,i}$	heat losses of the generation subsystem for period of defined operation conditions	(J)
COP_i	coefficient of performance of the heat pump for period of constant operation conditions	(W/W)

taking thereby into account that the generation subsystem heat losses $Q_{l,g,i}$ have to be covered by the generator, as well.

However, as the heat pump heating capacity and COP strongly depend on the operating conditions, mainly on the source and sink temperature, the calculation can be performed for a number of i periods defined by the constant source and sink temperature conditions, and results are summed-up, which is expressed by the summation in eq. 4. Thus, to determine the required energy input basically the COP as well as the heat energy requirement and generation subsystem losses at the defined operating conditions have to be evaluated.

To evaluate the heat energy requirement of the distribution subsystem, the heat load for space heating and domestic hot water has to be known. If detailed information on the heat load is not available, e.g. if only monthly or annual values of the heat energy are given, the energy requirement dependent on the temperature operating conditions can be estimated by evaluating the outdoor air temperature.

Actually, the bin method is based on an evaluation of the cumulative frequency of the outdoor air temperature depicted in Figure 4. The annual frequency of the outdoor air temperature based on hourly averaged values is cumulated and divided into temperature intervals (bins), which are limited by an upper temperature θ_{upper} and a lower temperature θ_{lower}. Operation conditions of the bins are characterised by an operating point in the centre of each bin. For the calculation it is assumed that the operating point defines the operating conditions for the heat pump of the whole bin. The evaluation of the annual frequency and the cumulative annual frequency from hourly averaged data of an entire year is given in Annex A.

The temperature difference between the outdoor air temperature and the indoor design temperature defines a heating degree hour (also called time temperature difference (TTD) according to EN 15927-6 for a base temperature of the design indoor temperature, normally 20°C). It corresponds to the heat load for space heating. Therefore, the area under the cumulative frequency, the cumulative heating degree hours, corresponds to the energy requirement for space heating, since the temperature difference (corresponding to the heat load) is cumulated over the time. The cumulative heating degree hours (CHDH) are also called accumulated time temperature difference (ATTD) in EN 15927-6. Analogously, the DHW load depicted as constant daily profile in Figure 4 can be cumulated. Although DHW heat energy is not dependent on the outdoor temperature but may have a connection to the bin time, the operation conditions for the heat pump are relevant, as well. Summarising, the energy requirement for the operating conditions defined by the operating point can be characterised by the cumulative heating degree hours.

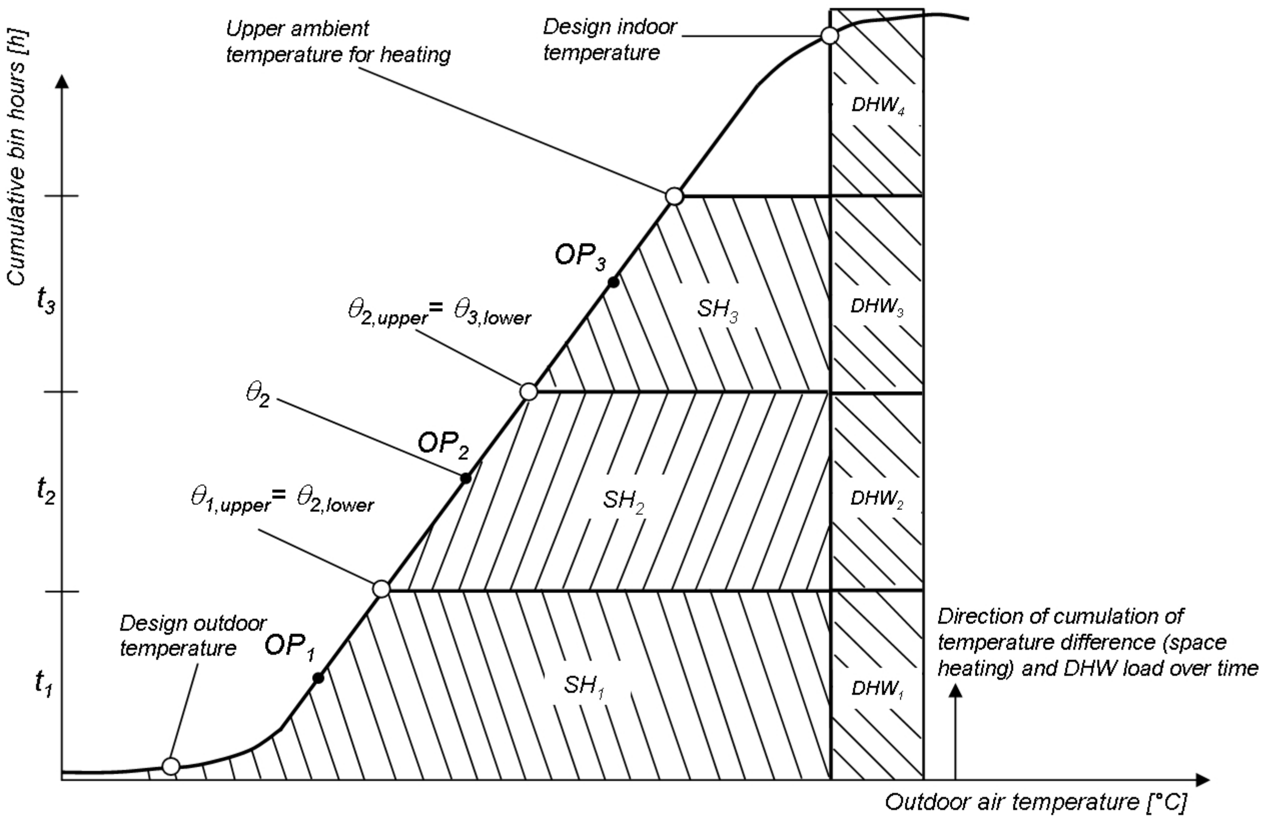


Figure 4 — Bin hours vs. outdoor air temperature – sample with 3 bins and constant daily DHW heat energy requirement

Another common way to depict the cumulative frequency is a clockwise 90° rotation called duration curve, which is shown in Figure 5 left hand side. Since this implies a negative temperature (y-) axis, it is depicted sometimes in a horizontally flipped diagram shown in Figure 5 right hand side. In the following, the cumulative frequency is depicted as in Figure 4 in line with the evaluation of frequency of the outdoor air temperature described in Annex A.

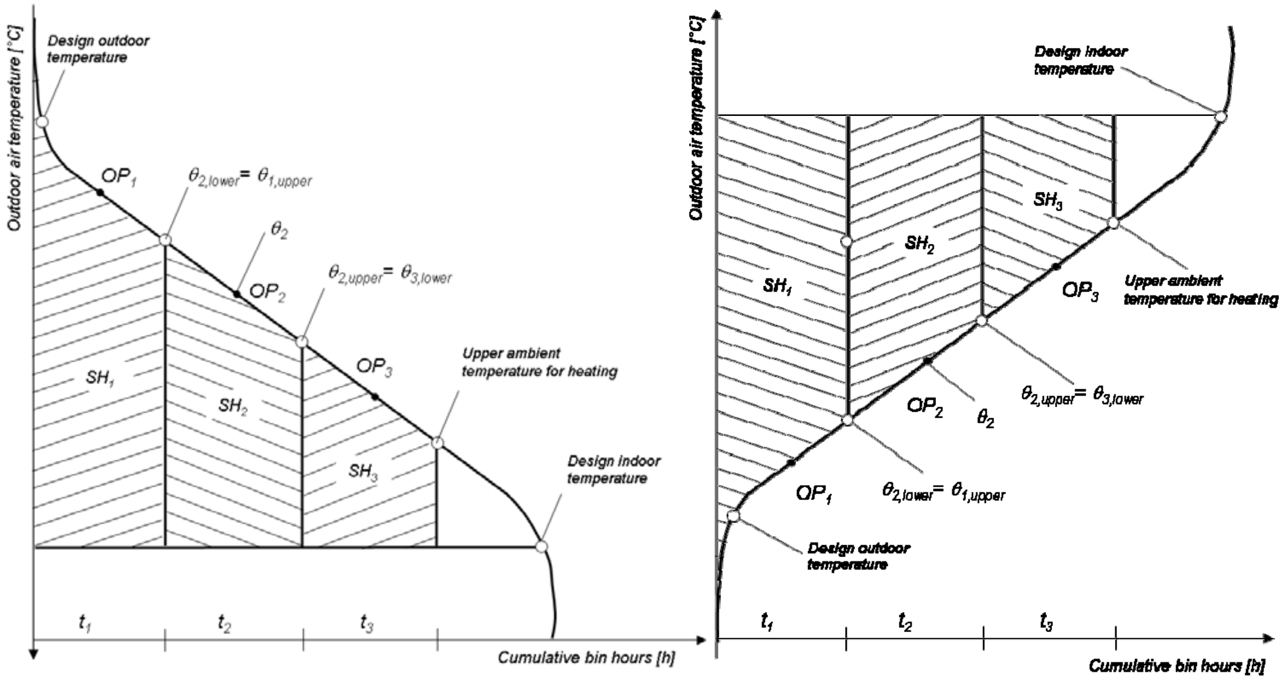


Figure 5 — outdoor air temperature vs. bin hours (duration curve) - sample with 3 bins SH-only

COP values, however, are normally only known at discrete test points based on standard product testing. The number of bins shall therefore be chosen dependent on the available information on the heat pump heating capacity and COP characteristic, e.g. according to EN 14511 for electrically-driven heat pumps. The operating points shall be chosen at the same source temperature as the test point and bin limits shall be set in the middle of two test points. In this way the number of bins corresponds to the number of test points with different source temperature from standard testing and an interpolation of the heat pump characteristic does only have to be accomplished to fit the sink temperature according to the respective flow temperature requirements at the operating point. Note that only in case of outdoor air source heat pumps the outdoor air temperature directly corresponds to the source temperature in the testing. For ground source heat pumps, for instance, the dependency of the source temperature on the outdoor temperature has to be considered to define the operating points, see also chap. 5.2.5.1.2. In case of mainly constant source temperatures, e.g. in case of ground water or exhaust air heat pumps, operating points shall be chosen at respective source temperatures to reflect the variation of the sink temperature.

If more detailed information on the heat pump exists, e.g. from manufacturer measurements according to the respective standard test procedures, more bins can be chosen to accommodate the available information of the heat pump characteristic. On the other hand, 1 K bins can be chosen, so that the heating capacity and the COP have to be interpolated for source temperatures between the known test points as described in chapter 5.2.5.1.2. Minimum number of bins corresponds to the different source temperatures defined by the test point of EN 14511 (standard and application rating) in order to consider the relevant impact on the characteristic, e.g. due to defrosting in case of outdoor air-to-water heat pumps. If not enough test points are available, the heat pump characteristic is interpolated, see chap. 5.2.5.1.2, or the exergetic efficiency method given in Annex C is to be applied.

The cumulative frequency is only dependent on the outdoor air temperature, and therefore does not take into account solar and internal gains. Even though the amount of energy is correct by using the heat energy requirement of the distribution subsystem according to prEN 15316-2-3, the redistribution of the energy to the bins depends also on the used gains (internal and solar). For existing buildings and newer standard houses the approximation with regard to the outdoor air temperature is quite good, while for new solar passive houses, it may get worse.

For a monthly calculation period, the cumulative frequency evaluated for a monthly data set is a good approximation of the redistribution of the solar and internal gains. Therefore, for a monthly calculation period, the cumulative frequency shall be calculated as the accumulated time temperature difference (ATTD) according to EN 15927-6 with a base temperature of the design indoor temperature, normally 20°C. This corresponds to the approach of the EN ISO 13790 for a monthly calculation period. For each month the calculation is accomplished for the bins chosen according to the available information on the heat pump characteristic.

For an annual calculation period, a correction of the redistribution to the bins can be made by using an upper temperature limit for heating dependent on the fraction of used solar and internal gains evaluated by the calculation according to EN ISO 13790. The upper temperature limit for heating can either be derived by the controller settings or based on the used gains and building type. The higher the fraction of used gains is the lower the heating limit is to be chosen. However, this is an approximation and for a significant quantity of used gains the method should be performed on a monthly basis.

For each bin, the heating capacity and the COP is evaluated from standard testing. The difference between the heat requirements and the heat energy delivered by the heat pump has to be supplied by the back-up heater in case of a bivalent system configuration. Losses associated to the generation subsystem in space heating and/or DHW operation and electricity input to auxiliaries are calculated, as well.

The total energy input in form of electricity, fuel or heat is determined by summing-up the results for each bin for the whole period of operation. Depending on the existence of a back-up system and its operation mode, supplied back-up energy is determined and summed-up, too, in order to calculate the overall energy consumption.

5.2.2 Input Data for the calculation with the bin method

Boundary conditions:

- meteorological data:
  - frequency of the outdoor air temperature of the site in 1 K resolution or hourly average values of the outdoor air temperature for an entire year (e.g. TRY or Meteonorm [1])
  - outdoor design temperature of the site

Space heating (SH) mode:

- indoor design temperature
- heat energy requirement of the space heating distribution subsystem according to prEN15316-2-3
- type and controller setting of the heat emission system (flow temperature of the heating system dependent on the outdoor air temperature, e.g. heating characteristic curve or characteristic of room thermostat), temperature spread at design conditions
- heat pump characteristics for heating capacity and COP according to product test standards (e.g. according to EN 14511 for electrically-driven heat pumps) and guaranteed temperature level that can be produced with the heat pump.
- results for part load operation, e.g. according to CEN/TS 14825 for electrically-driven heat pumps, if available
- for the simplified calculation method of the back-up energy, the balance point
- system configuration

    — installed back-up heater: operation mode, efficiency (fuel back-up heater acc. to prEN 15316-4-1)

    — installed heating buffer storage: stand-by loss value, flow temperature requirements

- power of auxiliary components (source pump, storage loading pump, primary pump, stand-by consumption)

Domestic hot water (DHW) mode:

- heat energy requirement of domestic hot water distribution subsystem according prEN15316-3-2
- temperature requirements of DHW operation: cold water inlet temperature (e.g. 15°C), DHW design temperature (e.g. 60°C)
- heat pump characteristics for DHW heating capacity and COP according to product test standards (e.g. according to EN 255-3 for electrically-driven heat pumps)
- set temperature for the energy delivery by the heat pump (e.g. at 55°C due to heat pump operating limit)
- parameters of the domestic hot water storage (stand-by loss value)
- installed back-up heater: operation mode, efficiency (fuel back-up heaters are calculated acc. to prEN 15316-4-1)

5.2.3 Calculation steps to be performed in the bin method

An overview of the calculation steps to be performed is listed below. A more detailed overview for different system configurations can be seen in the flow chart in Figure 6.

The individual steps are explained in detail in the remaining part of 5.2 as indicated. For each step, the description covers the different operation modes (space heating, domestic hot water) and the different types of heat pumps (electrically-driven, engine-driven, absorption).

A stepwise calculation example is given in Annex D.

Step 1: Determination of energy requirement of the single bins (see 5.2.4)

Step 2: Correction of steady state heating capacity/COP (e.g. EN 14511) for bin source and sink temperature operating conditions (see 5.2.5)

Step 3: If required correction of COP for part load operation (see 5.2.6)

Step 4: Calculation of generation subsystem heat losses (see 5.2.7)

Step 5: Determination of back-up energy of the single bins (see 5.2.8, simplified 5.2.8.2, detailed 5.2.9.3)

Step 6: Calculation of the running time of the heat pump in different operation modes (see 5.2.9)

Step 7: Calculation of auxiliary energy input (see 5.2.10)

Step 8: Calculation of recoverable generation subsystem losses (see 5.2.11)

Step 9: Calculation of total energy input to cover the requirements (see 5.2.12)

Step 10: Summary of required and optional output values (see chap. 5.2.13)

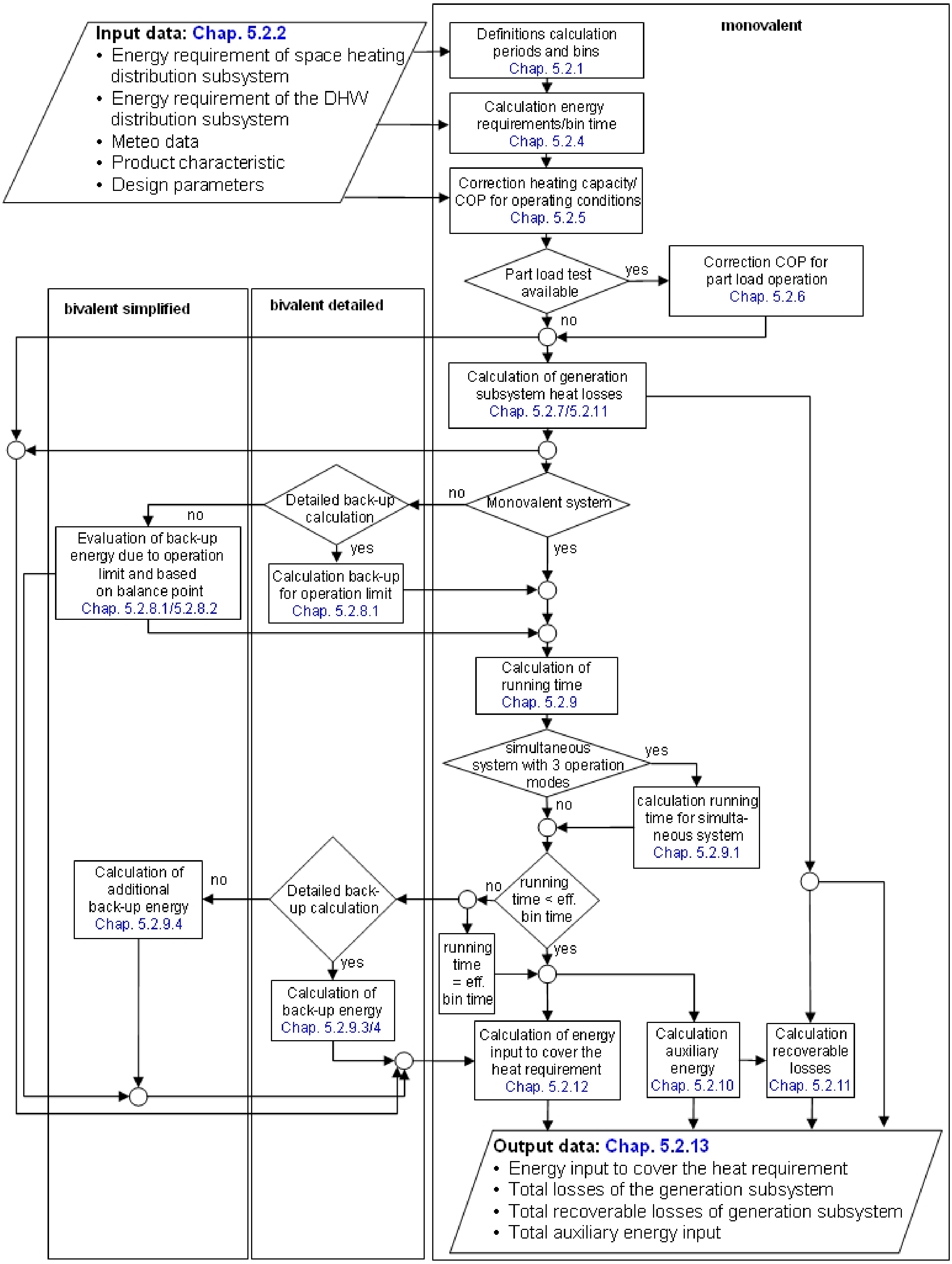


Figure 6 — Flow chart of the bin calculation method

5.2.4 Heat energy requirements for space heating and DHW mode for the bins

5.2.4.1 Space heating mode

The heat energy requirement for space heating of the distribution subsystem $Q_{d,in,h}$ is calculated according to the distribution part of this standard prEN 15316-2-3.

The heating requirement of bin i can be calculated by a weighting factor which is derived from evaluating the cumulative frequency of the outdoor air temperature by means of cumulative heating degree hours (CHDH). The evaluation of the cumulative heating degree hours from tables based on the hourly outdoor air temperature is described in Annex A.

The weighting factors are calculated by the equation

$$w_{h,i} = \frac{Q_{out,g,h,i}}{Q_{out,g,h}} = \frac{CHDH_{\theta upper,i} - CHDH_{\theta lower,i}}{CHDH_t} \qquad [-]$$

eq. 5

The space heating energy requirement of the respective bin is hence calculated by

$$Q_{out,g,h,i} = Q_{out,g,h} \cdot w_{h,i} \qquad [J]$$

eq. 6

where

$w_{h,i}$	weighting factor of the heat pump operation for space heating of bin i	(-)
$Q_{out,g,h,i}$	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)
$CHDH_{\theta upper,i}$	cumulative heating degree hours up upper temperature limit of bin i	(Kh)
$CHDH_{\theta lower,i}$	cumulative heating degree hours up lower temperature limit of bin i	(Kh)
$CHDH_t$	total cumulative heating degree hours up to upper temperature limit for space heating	(Kh)

The cumulative heating degree hours for the respective climatic regions shall be given in a national Annex or taken from national standardisation.

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

$$t_i = (N_{hours,\theta upper,i} - N_{hours,\theta lower,i}) \cdot 3600 s/h \qquad [s]$$

eq. 7

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

$$t_{i,eff} = t_i \cdot \frac{24h - t_{co}}{24h} \qquad [s]$$

eq. 8

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)

5.2.4.2 Domestic hot water mode

The heat energy requirement for domestic hot water of the distribution subsystem $Q_{d,in,DHW}$ is calculated according to prEN15316-3-2.

The domestic hot water heat requirement in bin i is calculated with the weighting factor for domestic hot water operation according to the equation

$$w_{DHW,i} = \frac{Q_{out,g,DHW,i}}{Q_{out,g,DHW}} = \frac{t_i}{t_t} \qquad [-]$$

eq. 9

and the DHW requirement in bin i follows according to the equation

$$Q_{out,g,DHW,i} = Q_{out,g,DHW} \cdot w_{DHW,i} \qquad [J]$$

eq. 10

where

$w_{DHW,i}$	weighting factor for DHW operation in bin i	(-)
$Q_{out,g,DHW,i}$	heat energy requirement of the domestic hot water distribution subsystem in bin i	(J)
$Q_{out,g,DHW}$	total heat energy requirement of the DHW distribution subsystem	(J)
t_i	bin time in bin i	(s)
t_t	total time of DHW operation (e.g. year round operation)	(s)

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.

5.2.5 Heating capacity and COP at full load

5.2.5.1 Space heating mode

The steady state heating capacity and COP are taken from standard test results of European test methods, e.g. according to EN 14511 for electrically-driven heat pumps. If European test standards are not available, e.g. for simultaneous operation, national methods shall be used. According to EN 14511, standard testing is performed at a standard rating point and several application rating conditions. Since the COP characteristic has the most significant impact on the heat pump performance, care shall be taken that COP-values are reliable. All available test points have to be taken into account, at least the test points prescribed by the standard testing (standard rating and application rating).

If national methods evaluate the heating capacity and the COP at different conditions as according to the test conditions of EN 14511, e.g. if the heating capacity and the COP value are related to the outlet temperature of both evaporator and condenser, this shall be stated clearly in the calculation report.

If flow conditions deviate between testing and operation the COP characteristic has to be corrected due to different temperature conditions at the condenser. The method for the correction is given in chapter 5.2.5.1.1.

To determine data for the whole range of the source and sink temperatures, linear inter- and extrapolation between the test points is applied both for the source and for the sink temperature, if necessary. Interpolation is performed between the temperatures of the two nearest test points. Extrapolation is performed by the nearest two points to the target point.

If, nevertheless, only one test point is available, correction for source and sink temperature can be done with the fixed exergetic efficiency approach described in Annex C instead of interpolating the data.

Some example values for the illustration of the data needed to accomplish the calculation for electrically-driven heat pumps are given in Annex F.2.

The source of the data shall be stated clearly in the calculation report (e.g. test data from standard test institutes, manufacturer data etc.). Preference shall be given to data from test institutes.

5.2.5.1.1 Correction of the COP characteristic for the temperature spread at the heat pump condenser

Evaluation of the COP dependency on the source and sink temperature is only correct, if the mass flow rate corresponds to the mass flow rate used during the standard testing, since otherwise, different temperature conditions exist at the heat pump condenser. Therefore, the temperature spread of the heat pump based on the mass flow rate defined by the design of the emission system has to be taken into account. Temperature spread and mass flow rate are linked by the equation

$$\Delta\theta = \frac{\phi_{hp}}{\dot{m}_w \cdot c_w} \qquad [K]$$

eq. 11

where

$\Delta\theta$	temperature spread on the condenser side of the heat pump	(K)
ϕ_{hp}	heating capacity of the heat pump	(W)
$\dot{m}_w$	mass flow rate of the heat transfer medium on the condenser side of the heat pump	(kg/s)
c_w	heat capacity of water	(J/(kg·K))

In case of the testing according to EN 14511 for electrically-driven heat pumps the temperature spread at the standard rating point is fixed to 5 K. With the temperature spread, the mass flow rate for the testing is determined and applied to all test points. Thus, the temperature spread during testing for the different operating points can be determined according to eq. 11. The temperature spread in operation can be determined by the mass flow in operation which is evaluated at outdoor design conditions.

If the temperature spread in testing and operation differs, the average temperature in the evaporator and condenser is different from the testing and therefore COP-values have to be corrected. The correction can be done based on the method of fixed exergetic efficiency given in Annex C according to the equation

$$COP_{\Delta\theta} = COP_{standard} \cdot \left[1 - \frac{\dfrac{\Delta\theta_{standard} - \Delta\theta_{op}}{2}}{\left\{ T_{si} - \dfrac{\Delta\theta_{standard}}{2} + \Delta T_{si} - (T_{so} - \Delta T_{so}) \right\}} \right] \qquad [W/W]$$

eq. 12

where

$COP_{\Delta\theta}$	COP corrected for a different temperature spread in testing and operation	(W/W)
$COP_{standard}$	COP derived from standard testing (e.g. according to EN 14511)	(W/W)

$\Delta\theta_{standard}$	temperature spread on the condenser side due to standard test conditions	(K)
$\Delta\theta_{op}$	temperature spread on the condenser side in operation due to the design of the heat emission system	(K)
T_{si}	sink temperature	(K)
ΔT_{si}	average temperature difference between heat transfer medium and refrigerant in condenser	(K)
T_{so}	source temperature	(K)
ΔT_{so}	average temperature between heat transfer medium and refrigerant difference in evaporator	(K)

The average temperature difference in the condenser and evaporator between the heat transfer medium and the refrigerant can be approximated by $\Delta T_{si}=\Delta T_{so}=4$ K for water based components. In the case of air based components $\Delta T_{si}=\Delta T_{so}=15$ K is set. However, it has to be secured that the minimum temperature difference between the heat transfer medium and the refrigerant is kept.

NOTE: Correction factor can be tabulated based on the combination of temperature spreads in testing and operation. The results of the correction according to eq. 12 correspond to the correction factors given in the tabulated values VDI 4650-1 [5] for air-to-water heat pumps and average temperature conditions e.g. the test point A2/W35.

5.2.5.1.2 Interpolation of heating capacity and COP for the temperature conditions

Based on the respective corrected characteristic of the $COP_{\Delta\theta}$ and the heating capacity, interpolation for the actual temperature conditions at the operating point of the respective bin is performed. The following source temperature applies for the respective type of heat pump:

- for an outdoor air heat pump, the source temperature is given by the outdoor air temperature based on the meteorological data of the site

- for a ground- or water-source heat pump, 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, values shall be given in a national annex. If a national annex is not available, an example profile for ground source heat pumps is given in Annex F.1.1.3 and a standard temperature for ground water in Annex F.1.1.4.

- for an exhaust-air heat pump 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 system (heating curve, room thermostat) or the temperature requirements of a possibly installed heating buffer storage. If the controller settings of the heating system is not known, typical controller settings for the heating curve for different kinds of heat emission systems are given in Annex B.1.1.

5.2.5.2 Domestic hot water mode

Electrically-driven domestic hot water heat pumps are tested as unitary systems, including the domestic hot water storage in the system boundary according to the standard EN 255-3. This standard testing delivers the COP-value for the extraction of domestic hot water, which is denoted COP_t in EN 255-3, at one standard test point, which depends on the type of the heat pump. For reasons of nomenclature, the COP_t of EN 255-3 is denoted $COP_{t,DHW}$ in this standard.

The $COP_{t,DHW}$ value is only valid for the extraction of domestic hot water and not for the loading of the storage without extraction of domestic hot water (stand-by operation), since the temperature conditions are different. However, the standard testing delivers an electrical power input to cover the storage losses denoted P_{es}, so electrical energy consumption to cover storage stand-by losses can be expressed by this value.

The sink temperature conditions of domestic hot water systems may change during the year. However, for calculation purposes the sink temperature conditions can be considered constant over the whole operation range as long as the draw-off temperature of the domestic hot water does not change much.

Due to varying source temperatures for the heat pumps operation, the operation period and thus COP values have to be corrected for these conditions. As only one standard test point depending on the type of heat pump is defined in EN 255-3, a temperature correction of the COP by interpolation is not possible. Therefore, a correction based on a fixed exergetic efficiency described in Annex C should be applied.

If no values according to EN 255-3 are available, the calculation for alternate operating systems is performed by evaluation of the space heating characteristic at an average DHW temperature calculated according to the equation

$$\theta_{w,avg} = f_{s,DHW} \cdot \theta_{op,hp} \qquad [°C]$$

eq. 13

where

$\theta_{w,avg}$	average hot water loading temperature	(°C)
$f_{s,DHW}$	temperature reduction factor for storage loading	(-)
$\theta_{op,hp}$	operation limit temperature of the heat pump (maximum hot water temperature, that can be reached with the heat pump operation)	(°C)

The temperature correction factor $f_{s,DHW}$ takes into account that the loading starts at lower temperatures than the maximum hot water temperature that can be reached by the heat pump operation (see also chapter 5.2.8.1) due to colder water at the storage heat exchanger. It increases during the loading to temperatures slightly above the maximum hot water temperature due to the required temperature difference for the heat transfer. Therefore, the average temperature for the loading is lower than the maximum hot water temperature that can be reached by the heat pump operation. Values of $f_{s,DHW}$ shall be given in a national Annex. If no national Annex is available, default values is given in Annex B.1.2

5.2.5.3 Engine-driven and absorption heat pumps

Steady state heating capacity and COP are taken from test results. The same considerations concerning the correction of temperature conditions as described in 5.2.5.1 and 5.2.5.2 apply depending on the test method used.

Example values of the required input data from testing for the space heating operation mode are given

- in Annex F.3.2.1 for air-to-water engine driven heat pumps
- in Annex F.3.2.2 for air-to-air engine-driven heat pumps
- in Annex F.4 for water-to-water absorption heat pumps

Example values for the domestic hot water operation mode are given

- In Annex F.5 for air-to-water engine-driven heat pumps

5.2.6 COP at part load operation

5.2.6.1 Space heating mode

Heat pumps with fixed speed compressor or fixed burner heat input for absorption heat pumps operate at part load operation by cycling between ON and OFF state. Therefore, at part load operation, losses due to cycling of the compressor occur and reduce the heating capacity and the COP of the heat pump.

Stepwise or continuously controlled variable capacity units, e.g. by means of an inverter for electrically-driven heat pumps or by modulation of burner heat input for absorption heat pumps, may have a better efficiency at part load. On the one hand, this may already be reflected in the full load values according to standard testing, e.g. EN 14511 for electrically-driven heat pumps, on the other hand, part load COP may to be more efficient.

However, for adequate system design, losses due to ON/OFF cycling are small. They are neglected in the frame of this calculation, unless they can be quantified by available test data on part load operation or national values given in a national Annex. Standard testing of part load operation for electrically-driven heat pumps is outlined in the Technical Specification CEN/TS 14825 for different types of compressor control. CEN/TS 14825 delivers the $COP_{50\%}$ which refers to a COP evaluated at 50% load.

If no values on part load operation are available, only the auxiliary stand-by consumption is taken into account which contributes to the degradation of the COP in part load operation.

Thus, if a part load correction is done at this place the stand-by auxiliary consumption calculated in chapter 5.2.10 must not be considered again.

In case of available measurements of the part load operation, the COP is interpolated to the respective part load condition in the bins that are characterised by a load factor corresponding to the part load ratio defined in the general part prEN 15316-1. It is calculated according to the eq. 14. The calculation has to be carried out only for bins with operating points above the balance point temperature.

$$FC = \frac{Q_{out,g,i}}{\phi_{hp,i} \cdot t_{i,eff}} \quad [-]$$

eq. 14

where

FC	load factor	(-)
$Q_{out,g,i}$	heat requirement of the distribution subsystem in bin i	(J)
$t_{i,eff}$	effective bin time in bin i	(s)
$\phi_{,hp,i}$	heating capacity of the heat pump in bin i	(W)

To accomplish the inter-/extrapolation, at least one part load test point has to be available, e.g. the $COP_{50\%}$. Then, the inter-/extrapolation can be performed between the full load COP and the part load COP as described in chapter 5.2.5.1.2.

5.2.6.2 Domestic hot water mode

For electrically-driven heat pumps start-up losses of the heat pump are already taken into account in the $COP_{t,DHW}$-value according to EN 255-3 due to the system testing.

5.2.6.3 Engine-driven and absorption heat pumps

For engine-driven and absorption heat pumps start-up losses shall be taken into account depending on the test procedure applied.

5.2.7 Heat losses through the generator envelope

5.2.7.1.1 Space heating mode

For heat pumps without an integrated storage in the same housing, the losses to the ambiance are neglected in the frame of this standard, unless national values are given for the envelope heat loss of the heat pump.

For engine-driven heat pumps, additionally the heat losses of the engine are considered. They shall be evaluated based on test results or manufacturer data. If no values are available the losses can be estimated by the efficiency of the engine and a possible fraction of recovered heat as for the CHP systems treated in prEN 15316-4-4. For the redistribution of the total heat losses to the bins or operation modes, if required, the bin time (for stand-by losses) and the running time (for operational losses) of the heat pump shall be evaluated.

A possibly internal or external heating buffer storage produces losses to the ambiance that can be calculated by a stand-by heat loss value for the bin i

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

eq. 15

Total storage losses of the heating buffer storage can be calculated by a summation over all bins

$$Q_{l,s,h,t} = \sum_{i=1}^{n_{bins}} Q_{l,s,h,i} \qquad [J]$$

eq. 16

where

$Q_{l,s,h,i}$	generation subsystem heat loss due to heating buffer storage loss to the ambiance 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_{sb}	stand-by heat loss	(kWh/d)
t_i	bin time	(s)
$Q_{l,s,h,t}$	generation subsystem heat loss to the ambiance due to heating buffer storage in bin i	(J)
n_{bins}	number of bins	(-)

If the stand-by heat loss from the storage vessel is not available default values are given in Annex B Table B 2.

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 flow- 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

$$\theta_{s,avg,i} = \frac{\theta_{f,i} + \theta_{r,i}}{2} \qquad [°C]$$

eq. 17

where

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

The flow temperature is evaluated according to the control of the heating system (heating curve, room thermostat) and the return temperature is calculated by interpolating the temperature spread of flow and return temperature between the design temperature spread at outdoor design temperature and $\Delta\theta=0$ at the indoor design temperature.

5.2.7.1.2 Domestic hot water mode

If data from storage testing are known the calculation of the losses of the domestic hot water storage shall be accomplished as for heating buffer storages according to eq. 15. 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 information is available, default values of the average DHW storage temperature are given in Annex B.1.3.

If no values on storage stand-by losses are available the calculation shall be carried out according to eq. 15 with values based on the volume of the storage vessel given in a national annex. If no national values are available default values are given in Annex B.1.4.

5.2.7.1.3 Heat losses of primary circulation pipes

Heat losses of the primary circulation pipes between the heat generator and the storage vessel are to be calculated according to the method given for the calculation of pipe heat losses in space heating distribution part prEN 15316-2-3 or the DHW generation part in prEN 15316-3-3 and added to the storage losses.

5.2.8 Calculation of back-up heater

Back-up energy can be required for two reasons: One reason is a temperature operating 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 5.2.8.1. On the other hand, there could be a multivalent design of the generator subsystem (see boundary conditions in chapter 4.6), 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 calculation of the back-up operation due to a lack of capacity a simplified and a detailed method are given.

The simplified method is based on the evaluation of the cumulative frequency and the balance point and, depending on the operation mode, the low temperature cut-out. It is described in chapter 5.2.8.2. The method assumes that the balance point is known and all influencing factors, e.g. power demand for space heating and DHW operation, cut-out times of the electricity supply etc.) have been taken into account.

For the detailed method a 1 K energy balance is accomplished for the range of lower source temperatures up to the temperature where no back-up energy is needed. It should be applied, if the balance point is not known or difficult to calculate, e.g. in systems with simultaneous operation, or if 1 K bins are chosen for the calculation anyway. The balance point is no longer an input, since it follows from the energy balance expressed by the required running time. The method is described in chapter 5.2.9.3 in connection with the evaluation of the running time.

5.2.8.1 Back-up energy due to the operation limit temperature of the heat pump

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

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

eq. 18

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)
$Q_{out,g,i}$	total heat energy requirement of the distribution subsystem	(J)
$\dot{m}_w$	mass flow rate of the heat transfer medium	(kg/s)
c_w	specific heat capacity of water	(J/(kg·K))
$\theta_{nom,i}$	nominal temperature requirement of the system	(°C)
$\theta_{op,hp}$	operation limit temperature of the heat pump (maximum 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:

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

eq. 19

where

$p_{bu,DHW,op,i}$	fraction of back-up energy due to the operation limit of the heat pump for DHW in bin i	(-)
$Q_{bu,DHW,op,i}$	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 (maximum 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.

5.2.8.2 Simplified back-up calculation: Back-up energy due to lack of capacity

Operation of back-up heater is determined by the system design criteria and can be characterised by the operation mode (alternate operation, parallel operation, partly parallel operation) and the respective temperatures, balance point temperature and, if required, low-temperature cut-out. By these temperatures, the energy fraction of the heat pump and back-up operation can be determined and energy consumption can be calculated. Evaluation of the area under the cumulative frequency can be done with tabulated cumulative heating degree hours of the site. An example for the evaluation of annual hourly averaged data from site measurements is given in Annex A, Table A 1.

5.2.8.2.1 Alternate operation mode of the back-up heater

In alternate operation mode of the back-up heater, the heat pump is switched-off at the balance point temperature, and only the back-up heater supplies the full heat energy requirement below the balance point. Figure 7 shows the areas in the cumulative annual frequency diagram of the outdoor air temperature, which correspond to the energy fractions. The area A_{BU} represents the energy fraction delivered by the back-up heater.

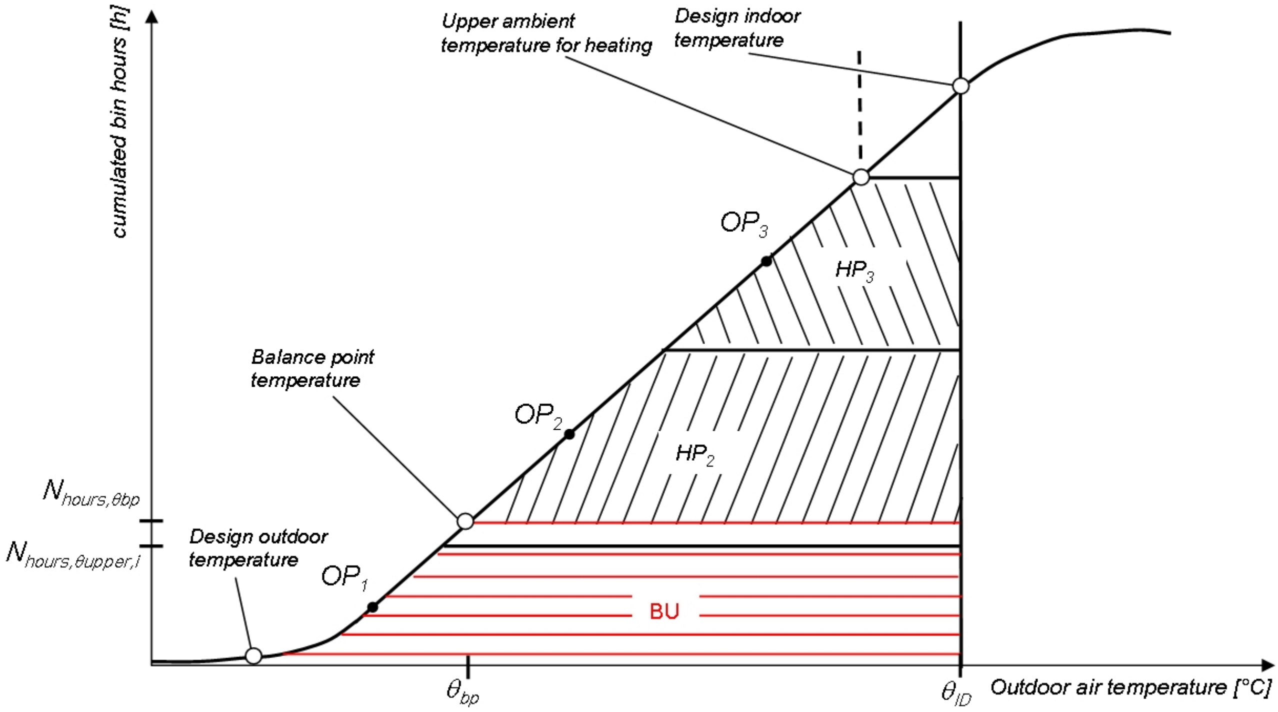


Figure 7 — Bin hours for alternate operation mode of the back-up heater – sample with 3 bins

If the balance points transcends the bin limit as in Figure 7, the fractions of the back-up heater for alternate operation in the lowest bin i and the subsequent bin i+1 are calculated

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

$$p_{bu,h,i} = \frac{A_{bu,i}}{A_i} = \frac{CHDH_{\theta upper,i}}{CHDH_{\theta upper,i} - CHDH_{\theta lower,i}} = \frac{CHDH_{\theta upper,i}}{CHDH_{\theta upper,i} - 0} = 1 \qquad [-]$$

eq. 20

$$p_{bu,h,i+1} = \frac{A_{bu,i+1}}{A_{i+1}} = \frac{CHDH_{\theta bp} - CHDH_{\theta lower,i+1}}{CHDH_{\theta upper,i+1} - CHDH_{\theta lower,i+1}} \qquad [-]$$

eq. 21

In case of a balance point below the bin limit the fraction is calculated according to the equation

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

$$p_{bu,h,i} = \frac{A_{bu}}{A_i} = \frac{CHDH_{\theta bp}}{CHDH_{\theta upper,i}} \qquad [-]$$

eq. 22

where

$p_{bu,h,i}$	fraction of space heating heat energy covered by the back-up heater in the lower bin i	(-)
$p_{bu,h,i+1}$	fraction of space heating heat energy covered by the back-up heater in the subsequent bin i+1	(-)
$A_{bu,i}$	fraction of the total area BU in Figure 7 in bin i	(Kh)
A_i	total area of bin i (between upper and lower temperature limit of bin i)	(Kh)

A_{i+1}	total bin area of the subsequent bin i+1	(Kh)
θ_{bp}	balance point temperature	(°C)
$\theta_{upper,i}$	upper temperature limit of bin i	(°C)
$CHDH_{\theta bp}$	cumulative heating degree hours up to the balance point θ_{bp}	(Kh)
$CHDH_{\theta lower,i}$	cumulative heating degree hours up to lower temperature limit of bin i $\theta_{lower,i}$	(Kh)
$CHDH_{\theta upper,i}$	cumulative heating degree hours up to upper temperature limit of bin i $\theta_{upper,i}$	(Kh)

5.2.8.2.2 Parallel operation mode of the back-up heater

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

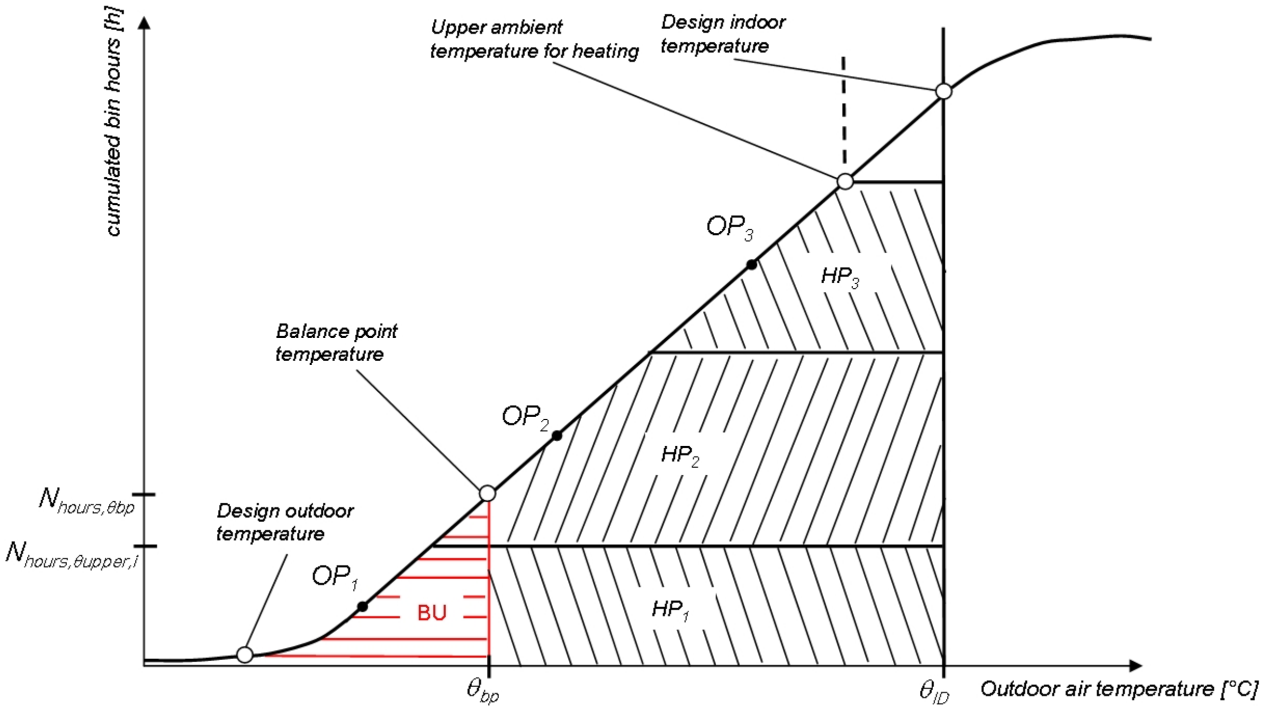


Figure 8 — Bin hours for parallel operation mode of the back-up heater – sample with 3 bins

Figure 8 shows the areas under the cumulative annual frequency of the outdoor air temperature, which correspond to the energy fractions. The area A_{BU} represents the energy fraction delivered by the back-up heater. The fraction of the back-up heater for parallel operation can be approximated by

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

$$p_{bu,h,i} = \frac{A_{bu,i}}{A_i} = \frac{CHDH_{\theta upper,i} - (\theta_{ID} - \theta_{bp}) \cdot N_{hours,\theta upper,i}}{CHDH_{\theta upper,i} - CHDH_{\theta lower,i}} \quad [-]$$

eq. 23

$$p_{bu,h,i+1} = \frac{A_{bu,i+1}}{A_{i+1}} = \frac{(CHDH_{\theta bp} - CHDH_{\theta upper,i}) - ((\theta_{ID} - \theta_{bp}) \cdot (N_{hours,\theta bp} - N_{hours,\theta upper,i})}{CHDH_{\theta upper,i+1} - CHDH_{\theta lower,i+1}} \quad [-]$$

eq. 24

In case of a balance point below the bin limit the fraction is calculated according to the equation

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

$$p_{bu,h,i} = \frac{A_{bu}}{A_i} = \frac{CHDH_{\theta bp} - (\theta_{ID} - \theta_{bp}) \cdot N_{hours,\theta bp}}{CHDH_{\theta upper,i} - CHDH_{\theta lower,i}} \quad [-]$$

eq. 25

where

$p_{bu,h,i}$	fraction of space heating heat energy covered by the back-up heater in the lower bin i	(-)
$p_{bu,h,i+1}$	fraction of space heating heat energy covered by the back-up heater in the subsequent bin i+1	(-)
$A_{bu,i}$	fraction of total area BU in Figure 8 in bin i	(Kh)
A_i	total area of bin i (between upper and lower temperature limit of bin i)	(Kh)
θ_{bp}	balance point temperature	(°C)
θ_{ID}	indoor design temperature	(°C)
$N_{hours,\theta bp}$	cumulated number of hours up to the balance point temperature	(h)
$CHDH_{\theta bp}$	cumulative heating degree hours up to the balance point θ_{bp}	(Kh)
$CHDH_{\theta lower,i}$	cumulative heating degree hours up to the lower temperature limit $\theta_{lower,i}$	(Kh)
$CHDH_{\theta lower,i}$	cumulative heating degree hours up to the upper temperature limit $\theta_{upper,i}$	(Kh)

NOTE: The vertical limit of A_{BU} in Figure 8 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 5.2.9.2 indicates if the approximation is not exact enough and correction is accomplished in eq. 41.

5.2.8.2.3 Partly parallel operation mode of the back-up heater

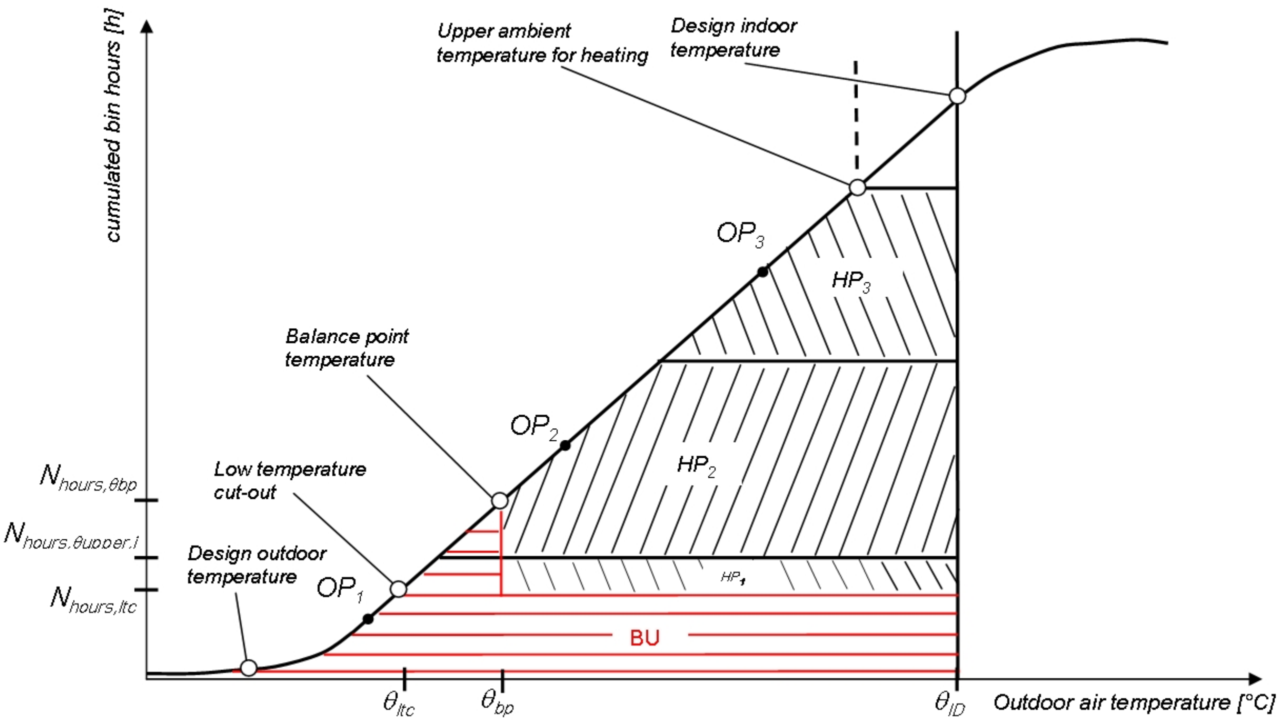


Figure 9 — Bin hours for partly parallel operation mode of the back-up heater – sample with 3 bins

In partly-parallel operation mode of the back-up heater, the heat pump is not switched-off at the balance point temperature, but 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 heat energy requirement of the space heating distribution subsystem. Figure 9 shows the areas under the cumulative annual frequency which correspond to the energetic fractions. The area A_{BU} represents the energy fraction delivered by the back-up heater.

The fraction of the back-up heater for partly parallel operation is approximated by

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

$$p_{bu,h,i} = \frac{A_{bu,i}}{A_i} = \frac{CHDH_{\theta upper,i} - (\theta_{ID} - \theta_{bp}) \cdot (N_{hours,\theta upper,i} - N_{hours,\theta ltc})}{CHDH_{\theta upper,i} - CHDH_{\theta lower,i}} \quad [-]$$

eq. 26

$$p_{bu,h,i+1} = \frac{A_{bu,i+1}}{A_{i+1}} = \frac{CHDH_{\theta bp} - CHDH_{\theta upper,i} - ((\theta_{ID} - \theta_{bp}) \cdot (N_{hours,\theta bp} - N_{hours,\theta upper,i})}{CHDH_{\theta upper,i+1} - CHDH_{\theta lower,i+1}} \quad [-]$$

eq. 27

In case of a balance point below the bin limit the fraction is calculated according to the equation

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

$$p_{bu,h,i} = \frac{A_{bu}}{A_i} = \frac{CHDH_{\theta bp} - (\theta_{ID} - \theta_{bp}) \cdot (N_{hours,\theta bp} - N_{hours,\theta ltc})}{CHDH_{\theta upper,i} - CHDH_{\theta lower,i}} \quad [-]$$

eq. 28

where

$p_{bu,h,i}$	fraction of space heating heat energy covered by the back-up heater in lower bin i	(-)
$p_{bu,h,i+1}$	fraction of space heating heat energy covered by the back-up heater in subsequent bin i+1	(-)
$A_{bu,i}$	fraction of area BU in Figure 9 in bin i	(Kh)
A_i	total area of bin i (between upper and lower temperature limit of bin i)	(Kh)
θ_{bp}	balance point temperature	(°C)
θ_{ID}	indoor design temperature	(°C)
θ_{ltc}	low-temperature cut-out temperature	(°C)
$N_{hours,\theta bp}$	cumulative number of hours up to the balance point temperature	(h)
$N_{hours,\theta upper,i}$	cumulative number of hours up to the upper temperature limit of bin i	(h)
$N_{hours,\theta ltc}$	cumulative number of hours up to the low-temperature cut-out	(h)
$CHDH_{\theta bp}$	cumulative heating degree hours up to the balance point temperature θ_{bp}	(Kh)
$CHDH_{\theta lower,i}$	cumulative heating degree hours up to the lower temperature limit $\theta_{lower,i}$	(Kh)
$CHDH_{\theta upper,i}$	cumulative heating degree hours up to the upper temperature limit $\theta_{upper,i}$	(Kh)

5.2.9 Running time of the heat pump

Running time of the heat pump depends on the heating capacity, given by the operation conditions, and on the heat requirement, given by the distribution subsystem. The running time can be calculated by the equation

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

eq. 29

where

$t_{ON,hp,i}$	running time of the heat pump in bin i	(s)
$Q_{hp,i}$	produced heat energy by the heat pump in bin i (heat energy requirement of the distribution subsystem and generation subsystem losses)	(J)
$\phi_{hp,i}$	heating capacity of the heat pump in bin i	(W)

The produced heat by 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. 30

where

$Q_{hp,i}$	produced heat energy by the heat pump in bin i (energy requirement of the distribution subsystem and generator losses)	(J)
$Q_{out,g,i}$	heat energy requirement of the distribution subsystem in bin i	(J)
$Q_{l,g,i}$	generator heat losses in bin i	(J)
$p_{bu,i}$	fraction of heat energy covered by the back-up heater	(-)

These equations can be applied for the different operation modes. Following items have to be considered.

Back-up calculation

If the simplified calculation of the back-up energy is applied, the fractions for space heating $p_{bu,i,h}$ and the fraction for domestic hot water operation due to the temperature operation limit of the heat pump $p_{bu,i,DHW,op}$ are known from the calculation in chapter 5.2.8 and are to be set in the equation.

If the detailed calculation of the back-up energy is applied, only the fraction of back-up energy for DHW due to the operation limit of the heat pump has to be taken into account, since the fraction of back-up energy due to a lack of capacity follows from the energy balance (see chapter 5.2.9.3). That means, only the fractions $p_{bu,op,i}$ due to the operation limit temperature are considered.

Operation mode

For heat pumps operating in SH-only mode or DHW-only mode the energy requirement is given by the actual space heating or domestic hot water heat requirement respectively, i.e. the energy requirement of the distribution subsystem and the generator losses.

For heat pumps operating alternately on the SH and DHW system total running time of the generator is determined by the sum of the space heating and domestic hot water heat energy requirements, produced at the respective heating capacity of the heat pump.

For heat pumps operating simultaneously for heat production for SH and DHW the running time has to be distinguished according to the state of operation. As the heat pump characteristic by simultaneous operation may differ significantly from the heat pump characteristic by the two single operation modes, the three following operation modes may have to be evaluated:

— space heating-only operation: running time is determined by the heat requirement of the space heating system and the respective characteristic of the heat pump in space heating-only mode;

— DHW-only operation: running time is determined by the domestic hot water requirement and the respective characteristic of the heat pump in DHW-only mode;

— simultaneous operation: running time is determined by the energy produced by 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. This is the case for instance in combined operating systems with desuperheating that work on a combi-storage for space heating and DHW. In this case only two characteristics, DHW-only and simultaneous combined, have to be taken into account and the time period of simultaneous operation is given by the heating period. The running time is evaluated based on these two characteristics.

Additional calculations of the energy fractions and the running times for systems with all three operation modes are given in chapter 5.2.9.1.

Heating capacity and COP of the respective operation mode shall be determined according to European test standards. If no European standards exist, national test methods should be used.

The total running time in the bin i can be calculated by the equation

$$t_{ON,hp,t,i} = t_{ON,hp,h,sin,i} + t_{ON,hp,DHW,sin,i} + t_{ON,hp,combi,i}$$ [s]

eq. 31

where

$t_{ON,hp,t,i}$	total running time of the heat pump in bin i	(s)
$t_{ON,hp,h,sin,i}$	running time in space heating-only operation in bin i	(s)
$t_{ON,hp,DHW,sin,i}$	running time in DHW-only operation in bin i	(s)
$t_{ON,hp,combi,i}$	running time in simultaneous operation in bin i	(s)

Depending on the type of system only some of different contributions exist, while the others are zero, e.g. the running time for space heating in DHW-only systems.

5.2.9.1 Additional calculations for simultaneous operating heat pumps with three operation modes

Principle

In the system depicted in Figure 3, for instance, all three operation modes occur, so the running time in the different operation modes has to be determined.

Since simultaneous operation only takes place in times of space heating and domestic hot water load, the running time is evaluated to characterise simultaneous operation. The maximum possible simultaneous operation is characterised by the minimum of required running time for space heating and DHW operation. Subsequently, the resulting maximum running time in simultaneous operation may then be corrected with a correction factor in order to take into account further controller impact.

After the estimation of the running time in simultaneous operation, the respective energies produced in simultaneous operation are calculated, and then the energy produced in SH-only and DHW-only can be determined by energy balances. As last step, the running time in SH-only and DHW-only operation is calculated based on these energies.

Since running time is related to produced energy, but storage losses of the DHW system may be expressed by the electricity input according to EN 255-3, the used energy to cover the heat requirement is calculated for the DHW system by subtracting the storage losses.

Calculation steps

The maximum running time in simultaneous operation is calculated by the equation

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

eq. 32

with the running time for DHW-operation is calculated with the heating capacity in simultaneous operation according to the equation

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

eq. 33

and analogously for space heating operation

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

eq. 34

where

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

The running time in simultaneous operation mode may also be influenced by the control and the load profiles. However, controller impact depends strongly on the setting and the system configuration and can be taken into account by a specific correction factor according to the equation

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

eq. 35

where

$t_{ON,hp,combi,i}$	running time in simultaneous operation in bin i	(s)
$t_{ON,hp,combi,max,i}$	maximum possible running time in simultaneous operation in bin i	(s)
f_{combi}	correction factor taking into account the impact of the control system 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, default values are given in B.1.6.	(-)

The DHW-energy and analogously the space heating energy produced in simultaneous operation is calculated by the equation

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

eq. 36

where

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

The rest of the heat energy is produced in SH-only and DHW-only operation and is determined by the equation for the respective operation modes

$$Q_{hp,sin,i} = Q_{hp,i} - Q_{hp,combi,i} \quad [J]$$

eq. 37

where

$Q_{hp,sin,i}$	produced heat by the heat pump in the respective single operation in bin i	(J)
$Q_{hp,i}$	produced heat energy by the heat pump in bin i	(J)
$Q_{hp,combi,i}$	produced heat energy by the heat pump in simultaneous operation in bin i	(J)

Since EN 255-3 gives the storage losses of the DHW storage in form of a electrical stand-by power input, the DHW heat energy requirement has to be evaluated by subtracting the storage losses. If no values according to EN 255-3 are available, the subtraction of the storage losses is not necessary.

The allocation of the DHW-storage losses to the single and simultaneous operation modes is done by $f_{combi.}$

So the DHW heat energy requirement in DHW-only and in simultaneous operation can be calculated by subtracting the storage losses according to the equation

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

eq. 38

where

$Q_{out,hp,DHW,sin,i}$	heat requirement of the DHW distribution subsystem 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 5.2.7.1.2)	(J)
f_{combi}	fraction of simultaneous operation	(-)
$p_{bu,DHW,i}$	fraction of DHW heat energy covered by the back-up heater in bin i	(-)

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

eq. 39

where

$Q_{out,hp,DHW,combi,i}$	heat requirement of the DHW distribution subsystem 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 5.2.7.1.2)	(J)
f_{combi}	fraction of simultaneous operation	(-)
$p_{bu,DHW,i}$	fraction of DHW heat energy covered by the back-up heater in bin i	(-)

The respective running time in SH-only and DHW-only operation modes are calculated according to eq. 29.

NOTE: Testing according to EN 255-3 does not deliver a heating capacity for the domestic hot water operation as an output. However, required data to evaluate an average heating capacity are provided by the testing in phase 2 of EN 255-3.

5.2.9.2 Boundary condition for the total 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

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

eq. 40

where

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

$t_{i,eff}$	effective bin time in bin i	(s)
$t_{ON,hp,h,sin,i}$	running time in space heating-only operation in bin i	(s)
$t_{ON,hp,DHW,sin,i}$	running time in DHW-only operation in bin i	(s)
$t_{ON,hp,combi,i}$	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 effective bin time is the running time and the missing back-up energy is calculated according to chapter 5.2.9.4.

5.2.9.3 Detailed back-up calculation: Back-up energy due to lack of capacity

The detailed evaluation of the back-up energy is based on the evaluation of the running time according to the boundary conditions given in chapter 5.2.9.2, but on the basis of 1 K bins. The comparison of the running time is accomplished, until the outdoor air temperature is reached, at which the effective time in the bin is longer than the required running time. The sample balance and the required calculations are summarised in Table 5. If the system is of alternate type, the running time in simultaneous operation is zero.

For the bins with a lack of running time, i.e. required running time is longer than the effective bin time, the heating capacity of the heat pump is not sufficient to cover the total requirement. The resulting back-up energy can be calculated based on the control strategy by the equation, i.e. the back-up heater either supplies heat to the space heating system or the DHW system as calculated in chapter 5.2.9.4.

Table 5 — Table containing the required calculation for the detailed determination of back-up energy

Outdoor air temperature (1 K bin)	Energy to be produced for SH $Q_{hp,h,i}$ acc. to eq. 30	Energy to be produced for DHW $Q_{hp,DHW,i}$ acc. to eq. 30	Heating capacity SH $\phi_{hp,h,sin}$ acc. to HP characteristic	Heating capacity DHW $\phi_{hp,DHW,sin}$ acc. to HP characteristic	Heating capacity SH combined $\phi_{hp,h,combi}$ acc. to HP characteristic	Heating capacity DHW combined $\phi_{hp,DHW,combi}$ acc. to HP characteristic	Running time for SH $t_{ON,hp,SH}$ acc. to eq. 29	Running time for DHW $t_{ON,hp,DHW}$ acc. to eq. 29	Running time combined $t_{ON,hp,combi}$ acc. to eq. 35	Total required running time $t_{ON,hp,t}$ acc. to eq. 31	Effective bin time (1 K bin) $t_{i,eff}$ acc. to eq. 8	Difference total running time to effective bin time	Required back-up energy $Q_{bu,cap,i}$ acc. to eq. 41
$\theta_{oa,min}$													
$\theta_{oa,min}$+ 1													
…													
Σ													Σ Back-up

5.2.9.4 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

$$Q_{bu,cap,i} = (t_{ON,hp,t,i} - t_{i,eff}) \cdot \phi_{hp,sin,i} \qquad [J]$$

eq. 41

where

$Q_{bu,cap,i}$	additional back-up energy due a lack of capacity	(J)
$t_{ON,hp,t,i}$	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

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

eq. 42

where

$p_{bu,i}$	fraction of heat energy covered by the back-up heater in bin i	(-)
$Q_{bu,op,i}$	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 (in case of simplified calculation)	(-)
$t_{ON,hp,t,i}$	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. 42.

5.2.10 Auxiliary energy

To calculate the auxiliary energy, the respective power of the auxiliary components has to be given as input. In heat pump systems auxiliary energy is basically used for pumps, fans, controls, additional oil supply heating (carter heating) and other electrical components like transformers.

The auxiliary energy is given by:

$$W_g = \sum_k \phi_{aux,g,k} \cdot t_{ON,aux,k} \qquad [J]$$

eq. 43

where

W_g	total auxiliary energy consumption	(J)
$\phi_{aux,g,k}$	electrical power of the auxiliary component k	(W)
$t_{ON,aux,k}$	relevant running or activation time of the respective auxiliary component k	(s)

Depending on the system configuration (e.g. with or without hydronic decoupling), the components described in chapter 4 are accounted to the generation subsystem.

Moreover, it has to be taken care, which auxiliary energies are already included in the COP values according to the standard testing of the heat pump, e.g. EN 14511 for electrically-driven heat pumps. For instance, in EN 14511 auxiliary energy for control during the running time of the generator is taken into account in the COP value. Correction shall be accomplished depending on the respective test standard.

The running time of the auxiliary components depend on the control of the generation subsystem.

— Source pump running time is normally linked to the running time of the heat pump evaluated in chapter 5.2.9 for the different operation modes.

— For primary pumps control depends on the installed systems, e.g. is linked to storage control in case of a heating buffer storage and thereby linked to the running time of the generator as well. In case of a hydronic distributor, primary pump may be switched-on periodically or even run through.

— Stand-by time can be calculated by the difference of the total activation time of the generator, e.g. the heating period for space heating operation, and the running time evaluated according to chapter 5.2.9. If a correction for part load operation of the COP is applied according to 5.2.6 the stand-by expenses are already considered and do not have to be considered here.

— For domestic hot water operation of electrically-driven heat pumps the storage loading pump is already entirely included in the $COP_{t,DHW}$ value according to EN 255-3 due to the system testing.

5.2.10.1 Engine-driven and absorption heat pumps

Depending on how the testing for engine-driven heat pumps and absorption heat pumps is accomplished for the operation modes space heating and DHW is accomplished, the respective part of auxiliary energy (e.g. pumps, fans for burners etc.) shall be considered.

5.2.11 Total losses and total recoverable heat loss of the generation subsystem

5.2.11.1 Recoverable heat losses from auxiliary consumption

Auxiliary energy is transformed partly to used energy and partly to heat losses.

Recoverable heat losses to the heat transfer medium are considered totally recovered

$$W_{g,rd} = \sum_k W_{g,k} \cdot k_{g,rd,k} \quad [J]$$

eq. 44

where

$W_{g,rd}$	totally recovered auxiliary energy	(J)
$W_{g,k}$	auxiliary energy consumption of the auxiliary component k	(J)
$k_{g,rd,k}$	fraction of auxiliary energy totally recovered as thermal energy	(-)

This fraction is already considered in the COP-value according to standard testing to EN 14511 for electrically-driven heat pumps, so $k_{g,rd} = 0$ for electrically-driven heat pump.

Heat losses of auxiliaries to the ambiance can be calculated according to the equation

$$W_{g,l} = \sum_k W_{g,k} \cdot p_{aux,g,k} \qquad [J]$$

eq. 45

and heat losses to the ambiance are assumed recoverable.
Recoverable heat losses can be calculated by a temperature reduction factor linked to location

$$W_{g,rl} = \sum_k W_{g,k} \cdot p_{aux,g,k} \cdot (1 - b_{g,k}) \qquad [J]$$

eq. 46

where

$W_{g,l}$	heat losses of auxiliary components to the ambiance	(J)
$W_{g,rl}$	recoverable heat losses of auxiliary components	(J)
$W_{g,k}$	auxiliary energy consumption of the auxiliary component k	(J)
$p_{aux,g,k}$	fraction of electrical energy transmitted to the ambiance. These values should be defined in a national annex. If no national values are specified, default values are given in Annex B.1.5.	(-)
$b_{g,k}$	temperature reduction factor for component k linked to location of the component. The values of b_g shall be given in a national annex. If no national values are specified, default values are given in Annex B.1.7.	(-)

5.2.11.1.1 Total generation subsystem losses

The total envelope heat losses of the generation subsystem can be obtained by a summation over the components, basically heat pump envelope losses, if considered, losses from the engine of engine-driven heat pumps, storage losses for the heating buffer and DHW storage, respectively, and losses of the connecting piping between generator and storage, according to the equation

$$Q_{l,g} = \sum_k Q_{l,g,k} + W_{g,l} \qquad [J]$$

eq. 47

where

$Q_{l,g}$	total generation subsystem heat losses to the ambiance	(J)
$Q_{l,g,k}$	heat losses to the ambiance of the generation subsystem component k	(J)
$W_{g,l}$	heat losses of auxiliary components to the ambiance	(J)

5.2.11.1.2 Recoverable heat losses due to generation subsystem envelope losses

Envelope losses are considered recoverable and can be calculated with a temperature reduction factor according to the equation

$$Q_{l,g,rl} = \sum_k Q_{l,g,k} \cdot (1 - b_{g,k}) \qquad [J]$$

eq. 48

where

$Q_{l,g,rl}$	recoverable heat losses of the generation subsystem	(J)
$Q_{l,g,k}$	heat losses to the ambiance of the generation subsystem component k	(J)
$b_{g,k}$	temperature reduction factor linked to location of the component k. The values should be given in a national annex. If no national values are specified, default values are given in Annex B.1.7.	(-)

5.2.11.1.3 Total recoverable heat losses of the generation subsystem

The total recoverable losses can be obtained by a summation of the generation subsystem envelope losses and the losses of auxiliary components to the ambiance according to the equation

$$Q_{l,g,rl,t} = Q_{l,g,rl} + W_{l,g,rl} \qquad [J]$$

eq. 49

where

$Q_{l,g,rl,t}$ total recoverable heat losses of the generation subsystem (J)

$Q_{l,g,rl}$ recoverable heat losses of the generation subsystem (J)

$W_{l,g,rl}$ recoverable heat losses of auxiliary components (J)

5.2.12 Calculation of total energy input

5.2.12.1 Electrically-driven heat pumps

5.2.12.1.1 Electricity input to the heat pump for space heating operation

The electricity input to the heat pump for space heating operation can be calculated by summing-up the electricity input of the respective bins according to:

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

eq. 50

where

$E_{in,hp,h}$ electrical energy input to the heat pump in space heating mode (J)

$Q_{hp,h,sin,i}$ produced energy by the heat pump in space heating-only operation in bin i (J)

$Q_{hp,h,combi.i}$ produced energy by the heat pump for space heating in simultaneous operation in bin i (J)

$COP_{h,sin,i}$ coefficient of performance in space heating-only operation at the operating point of bin i, taken as performance factor for the whole bin (W/W)

$COP_{h,combi,i}$ coefficient of performance of space heating in simultaneous operation at the operating point of bin i, taken as performance factor for the whole bin (W/W)

n_{bin} number of bins (-)

5.2.12.1.2 Electricity input to the heat pump for domestic hot water operation

The electricity input to the heat pump for domestic hot water operation can be calculated according to:

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

eq. 51

where

$E_{in,hp,DHW}$ electrical energy input to the heat pump in DHW mode (J)

$Q_{out,hp,DHW,sin,i}$ heat energy requirement of the DHW distribution subsystem in bin i covered by the heat pump in DHW-only operation (J)

$Q_{out,hp,DHW,combi,i}$ heat energy requirement of the DHW distribution subsystem in bin i covered by the heat pump in simultaneous operation (J)

$COP_{t,DHW,sin,i}$	coefficient of performance for the extraction of domestic hot water of bin i in DHW-only operation according to EN 255-3, taken as performance factor for the whole bin	(W/W)
$COP_{t,DHW,combi,i}$	coefficient of performance for the extraction of domestic hot water of bin i in simultaneous operation, taken as performance factor for the whole bin	(W/W)
P_{es}	electricity power input to cover storage losses according to EN 255-3 for DHW-only/simultaneous operation respectively	(W)
t_{DHW}	total time of all calculation periods in DHW-only/simultaneous operation respectively	(s)
n_{bin}	number of bins	(-)

The allocation of the total time to DHW-only and simultaneous DHW operation is done by the fraction of simultaneous operation as for the storage losses in chap. 5.2.9.1.

NOTE: If no values according to EN 255-3 are available, the calculation for alternate operating systems is accomplished in the same way as for the space heating operation mode. For COP-values to use see chap. 5.2.5.2.

5.2.12.2 Engine-driven and absorption heat pumps

The calculation of the energy input (fuel or waste heat, solar heat, respectively) to the generation subsystem depends on the applied test method for the COP characteristic with regard to considered heat recovery from the engine and the auxiliaries.

If a possible heat recovery from the engine cooling fluid and/or the engine flue gas and the auxiliaries is taken into account in the COP values, the driving energy can be calculated as for electrically-driven systems acc. to eq. 50. This approach corresponds to the system boundary given in chap. 4.2.2.

If the COP values only take into account the heat decoupled at the heat pump condenser, the produced heat based on the heat energy requirement of the distribution subsystem has to be reduced by the recovered energy from the engine and auxiliaries. The fuel or heat energy input to the engine-driven or absorption heat pump respectively can be calculated by the following equation which is applied for the operation modes SH and DHW and single and combined operation respectively, if required:

$$G_{in,hp} = \sum_{i=1}^{n_{bin}} \frac{Q_{hp,i} - Q_{eng,rd,i} - k_{g,rd} \cdot W_{g,i}}{COP_i} \quad [J]$$

eq. 52

where

$G_{in,hp}$	fuel or heat energy input to the engine-driven or absorption heat pump	(J)
$Q_{hp,i}$	produced energy of the heat pump in bin i	(J)
$Q_{eng,rd,i}$	recovered energy from the combustion engine in bin i (only engine-driven heat pumps)	(J)
$k_{g,rd}$	fraction of auxiliary energy recovered as thermal energy (depending on testing)	(-)
$W_{g,i}$	auxiliary energy consumption in bin i	(J)
COP_i	coefficient of performance at the operating point of bin i, taken as performance factor for the whole bin i for the respective operation mode	(W/W)
n_{bin}	number of bins	(-)

The recovered energy $Q_{eng,rd}$ shall be calculated based on test results or manufacturer data. If the net or the gross calorific value is to be used depends on which of these values is considered in the testing of the engine-driven heat pump. Definitions of other standards e.g. prEN15203 shall be taken into account as far as possible.

The redistribution of the recovered heat to the respective operation modes, if required, is to be evaluated for the individual system configuration based on installed components and controls.

5.2.12.3 Energy input to back-up system

5.2.12.3.1 Electrical back-up heater

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

eq. 53

where

$E_{in,bu}$	total 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)
$Q_{l,g,h,i}$	heat losses of the generation subsystem due to space heating operation in bin i	(J)
$p_{bu,h,i}$	fraction of SH heat energy covered by the back-up heater in bin i	(-)
$\eta_{bu,h}$	efficiency of the electrical back-up heater for space heating mode	(-)
$Q_{out,g,DHW,i}$	heat energy requirement of the DHW distribution subsystem in bin i	(J)
$Q_{l,g,DHW,i}$	heat losses of the generation subsystem due to DHW operation in bin i	(J)
$p_{bu,DHW,i}$	fraction of DHW heat energy covered by the back-up heater in bin i	(-)
$\eta_{bu,DHW}$	efficiency of the electrical back-up heater for DHW mode	(-)
n_{bin}	number of bins	(-)

5.2.12.3.2 Fuel back-up heater

Fuel back-up heaters are calculated in the same way as the electrical back-up heaters. However, the efficiency of the back-up heater shall be determined according to the respective standard of prEN 15316 depending on the type of back-up heater, i.e. prEN 15316-4-1 for combustion boiler back-up heaters.

5.2.12.4 Total driving and back-up energy input to cover the heat requirement

5.2.12.4.1 Electrically-driven generation subsystems

The total electrical energy input to cover the heat requirement is the sum of the single electrical energy inputs:

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

eq. 54

where

$E_{in,g}$	total electrical energy input to heat pump and back-up	(J)
$E_{in,hp,h}$	electrical energy input to the heat pump in space heating mode	(J)
$E_{in,hp,DHW}$	electrical energy input to the heat pump in DHW mode	(J)
$E_{in,bu}$	total electrical energy input to operate the back-up heater	(J)

5.2.12.4.2 Engine-driven and absorption heat pump generation subsystems

The total fuel or heat energy input is the sum of the single energy inputs

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

eq. 55

where

$G_{in,g}$	fuel or heat energy input to the engine-driven or absorption heat pump and back-up	(J)
$G_{in,hp,h}$	fuel or heat energy input to the heat pump in space heating mode	(J)
$G_{in,hp,DHW}$	fuel or heat energy input to the heat pump in DHW mode	(J)
$G_{in,bu}$	total fuel energy input to operate the back-up heater	(J)

5.2.12.5 Ambient heat used by the generation subsystem

The amount of ambient heat used for the produced heat energy of the heat pump to cover the space heating and/or DHW requirement and generation subsystem losses is calculated according to eq. 1, where the recovered auxiliary energy is to be set to $k_{g,rd} = 0$ for electrically-driven heat pumps tested according to EN 14511. For engine-driven and gas heat pumps, the factor $k_{g,rd}$ depends on the fraction taken into account during testing.

5.2.12.6 Seasonal Performance factor and expenditure factor of the generation subsystem

For the seasonal performance factor, two characteristic numbers can be defined based on different system boundaries, the heat pump as generator itself and the generation subsystem.

5.2.12.6.1 Electrically-driven generation subsystems

The total seasonal performance factor of the generation subsystem can be calculated according to the equation

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

eq. 56

where

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

and the total seasonal performance factor of the heat pump can be calculated according to the equation

$$SPF_{hp,t} = \frac{Q_{hp,h} + Q_{hp,DHW}}{E_{in,hp} + W_{so} + W_{sb}} \quad [-]$$

eq. 57

$Q_{hp,h}$	produced heat energy by the heat pump in space heating mode	(J)
$Q_{hp,DHW}$	produced heat energy by the heat pump in DHW mode	(J)
$E_{in,hp}$	Total electrical energy input to the heat pump to cover the heat requirement	(J)
W_{so}	auxiliary energy input for the source system	(J)
W_{sb}	auxiliary energy input to the heat pump in stand-by operation	(J)

By comparison of these two seasonal performance factors an assessment of the performance of the heat pump and the impact of generation subsystem losses can be made.

As the seasonal performance factor is the reciprocal value of the expenditure factor, it can also be calculated from:

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

eq. 58

where

e_g	expenditure factor for generation subsystem	(-)
$SPF_{g,t}$	total seasonal performance factor of generation subsystem	(-)

5.2.12.7 Total heat produced by the heat pump and the back-up heater

The total heat delivered by the back-up heater can be calculated according to the equation

$$Q_{bu} = \sum_{i=1}^{nbin} p_{bu,h,i} \cdot (Q_{out,g,hi} + Q_{l,g,hi}) + p_{bu,DHW,i} \cdot (Q_{out,g,DHW,i} + Q_{l,g,DHW,i}) \quad [J]$$

eq. 59

and the total heat produced by the heat pump by the equation

$$Q_{hp} = (Q_{out,g,h} + Q_{l,g,h}) + (Q_{out,g,DHW} + Q_{l,g,DHW}) - Q_{bu} \quad [J]$$

eq. 60

where

Q_{bu}	produced heat energy of the back-up heater	(J)
$p_{bu,h,i}$	fraction of space heating heat energy covered by the back-up heater in bin i	(-)
$Q_{out,g,h,i}$	heat energy requirement of the space heating distribution subsystem in bin i	(J)
$Q_{l,g,h,i}$	heat losses of the generation subsystem due to space heating operation in bin i	(J)
$p_{bu,DHW,i}$	fraction of DHW heat energy covered by the back-up heater in bin i	(-)
$Q_{out,g,DHW,i}$	heat energy requirement of the DHW distribution subsystem in bin i	(J)
$Q_{l,g,DHW,i}$	heat losses of the generation subsystem due to DHW operation in bin i	(J)
$Q_{out,g,h}$	total heat energy requirement of the space heating distribution subsystem	(J)
$Q_{l,g,h}$	heat losses of the generation subsystem due to space heating operation	(J)
$Q_{out,g,DHW}$	heat energy requirement of the DHW distribution subsystem	(J)
$Q_{l,g,DHW}$	heat losses of the generation subsystem due to DHW operation	(J)
n_{bin}	number of bins	(-)

5.2.13 Summary of output values

Required outputs

Energy input to cover the heat requirement of the distribution subsystems (chap. 5.2.12.4, eq. 54)

Total losses of the generation subsystem (chap. 5.2.11.1.1, eq. 47)

Total recoverable losses of the generation subsystem (chap. 5.2.11.1.3, eq. 49)

Total auxiliary energy input to the generation subsystem (chap. 5.2.10, eq. 43)

Optional outputs

Overall seasonal performance factor/expenditure factor of the generation subsystem (chap. 5.2.12.6, eq. 56, eq. 57, eq. 58)

Total used ambient heat (chap. 5.2.12.5, eq. 1)

Total heat produced by the the heat pump (chap. 5.2.12.7, eq. 60)

Total heat produced by the back-up heater (chap. 5.2.12.7, eq. 59)

Annex A
(informative)

Example of evaluation of meteorological data

For the calculations, the annual cumulative frequency of the outdoor air temperature of the site has to be evaluated. The procedure to derive the required tabulated values based on hourly values of the outdoor air temperature is given in this Annex.

If hourly-averaged values of the site are not available, the Software Meteonorm [1] can deliver hourly averaged values of site of the whole world based on a statistical evaluation.

Input data:

Hourly-averaged values of the outdoor air temperature of the site.

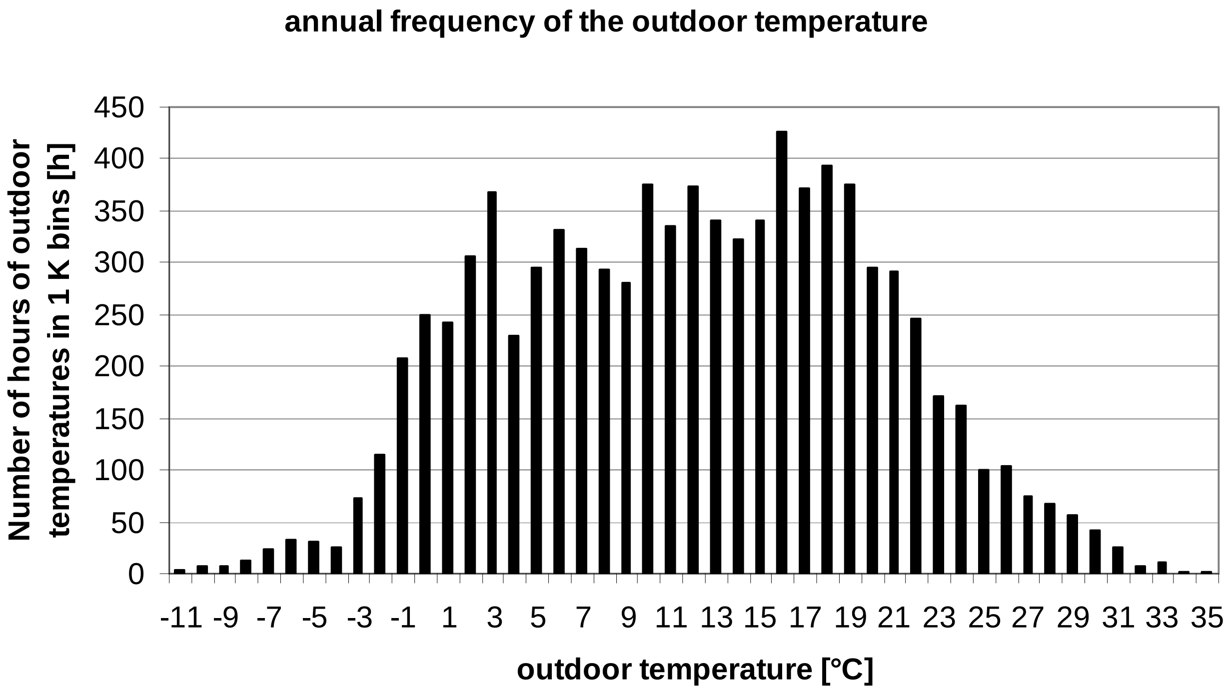


Figure A 1 — Annual frequency of the outdoor temperature

Step 1: Evaluation of the annual frequency:

The input data are sorted in temperature classes of 1 K bins beginning with the minimum outdoor air temperature.

The 1 K bin time corresponds to the time period (number of hours) in the respective 1 K bins.

The annual frequency is depicted in Figure A 1.

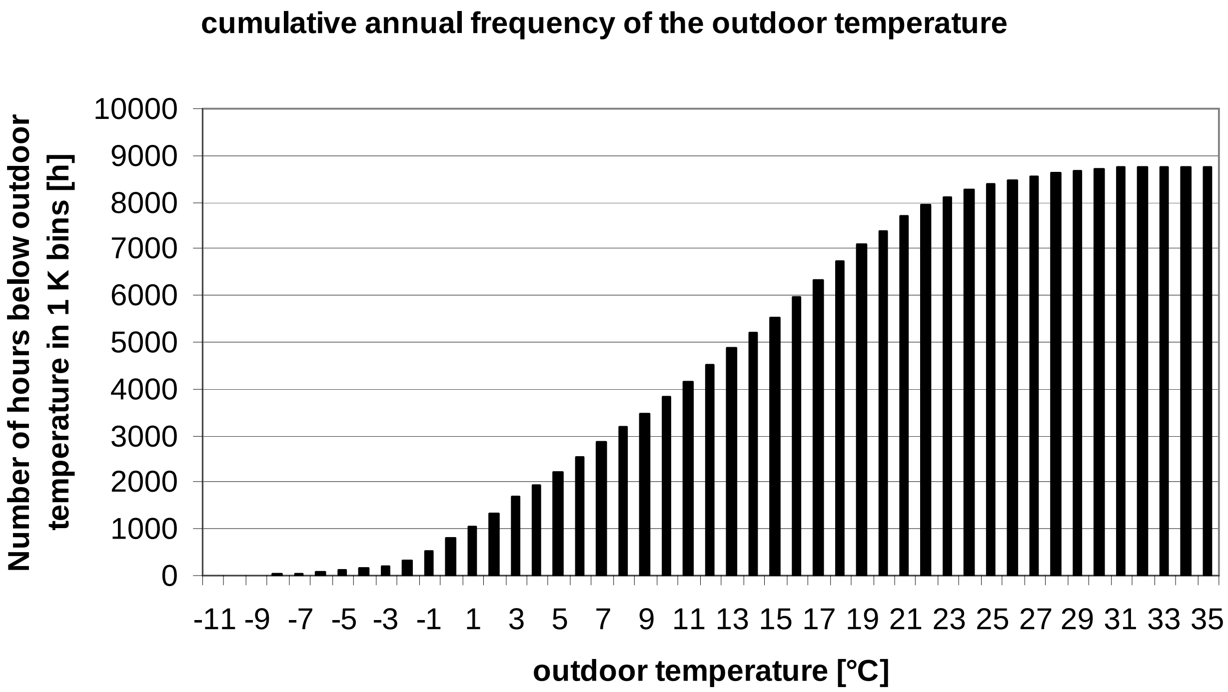


Figure A 2 — Cumulative annual frequency of the outdoor temperature

Step 2: Evaluation of cumulative annual frequency

The annual frequency of the outdoor temperature is summed-up bin-by-bin to derive the cumulative annual frequency:

$$N_k = \sum_{i=1}^{k} n_i \qquad [-]$$

eq. A 1

where

N	cumulated number of hours	(h)
k	number of the actual bin	(-)
n_i	number of hours for the actual bin	(h)
i	incremental variable for all bins from 1 to k	(-)

The cumulative frequency of the outdoor temperature is depicted in Figure A 2.

Step 3: Evaluation of heating degree hours

The heating degree hours for each bin can be derived from the respective temperature difference of bin temperature and indoor design temperature according to:

$$HDH_i = n_i \cdot (\theta_{ID} - \theta_{oa,i}) \qquad [Kh]$$

eq. A 2

where

HDH_i	heating degree hours for bin i	(Kh)
n_i	number of hours for bin i	(h)
θ_{ID}	indoor design temperature	(°C)
$\theta_{oa,i}$	outdoor air temperature for bin i	(°C)

Step 4: Evaluation of cumulative heating degree hours

The cumulative heating degree hours for a given bin, k, is calculated by summing up the heating degree hours for each bin 1 to k:

$$CHDH_{\theta k} = \sum_{i=1}^{k} HDH_i \qquad \text{[Kh]}$$

eq. A 3

where

$CHDH_{\theta k}$	cumulative heating degree hours up to temperature θ_k	(Kh)
HDH_i	heating degree hours for bin i	(Kh)
k	number of the actual bin	(-)
i	number bin i	(-)

With the cumulative heating degree hours, the weighting factors for the heat pump operation can be calculated according to the equations above.

Monthly calculation period

For a monthly calculation period the frequency and the cumulative frequency for each month are determined according to the above equations for a monthly data set.

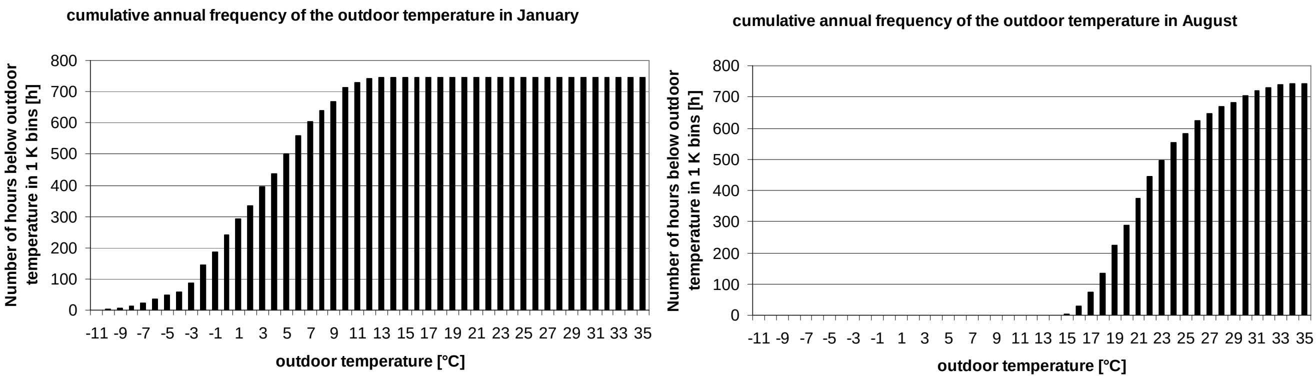


Figure A 3 — Monthly cumulative frequency of January (left) and August (right)

Table A 1— Evaluation of meteorological data of the site (based on hourly-averaged on site measurements in Gelterkinden, BL, Switzerland)

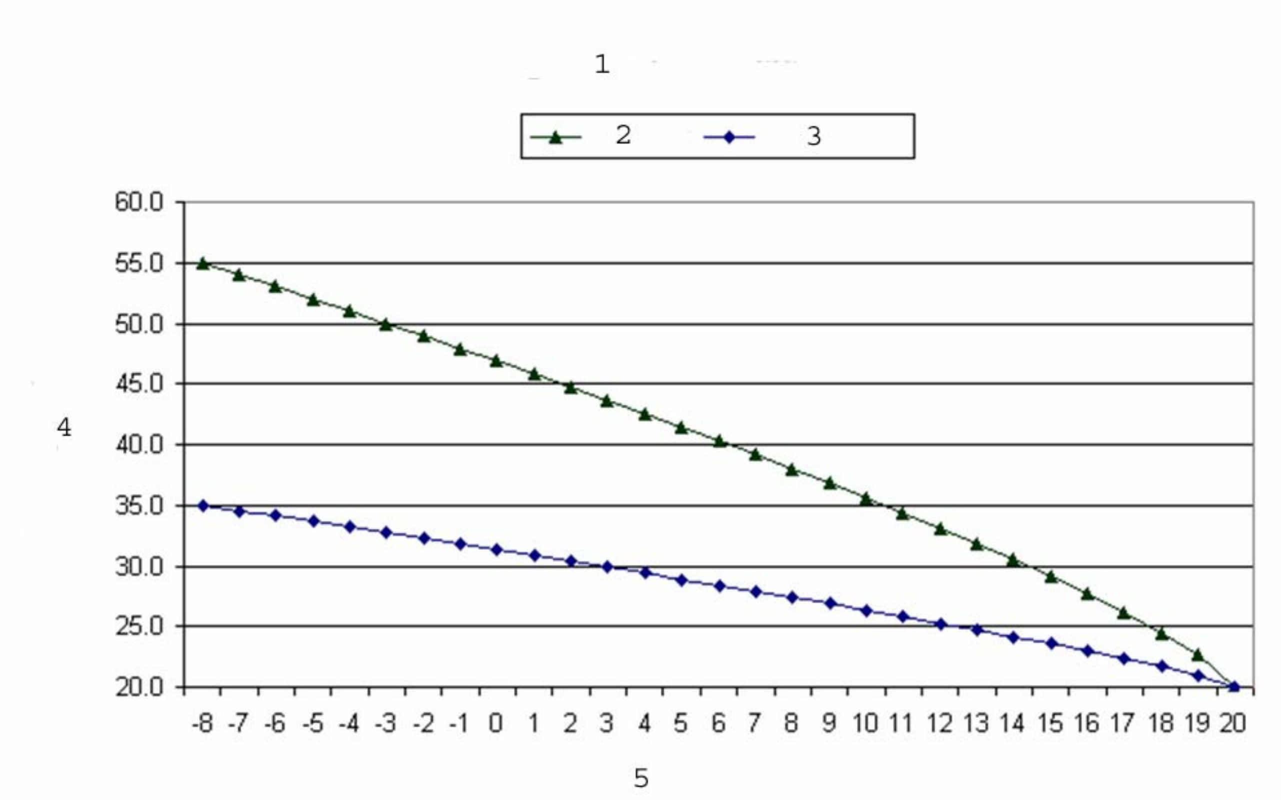
Bin	Bin time	cumulative bin time	HDH 20/15	Cumulative HDH 20/15	Weighting factor SH 1 K bins	Weighting factor DHW 1 K bins
-11	3	3	93	93	0.00	0.000
-10	7	10	210	303	0.00	0.001
-9	8	18	232	535	0.00	0.001
-8	12	30	336	871	0.00	0.001
-7	24	54	648	1519	0.01	0.003
-6	33	87	858	2377	0.01	0.004
-5	31	118	775	3152	0.01	0.004
-4	26	144	624	3776	0.01	0.003
-3	72	216	1656	5432	0.02	0.008
-2	114	330	2508	7940	0.03	0.013
-1	207	537	4347	12287	0.06	0.024
0	250	787	5000	17287	0.07	0.029
1	243	1030	4617	21904	0.06	0.028
2	306	1336	5508	27412	0.08	0.035
3	368	1704	6256	33668	0.09	0.042
4	230	1934	3680	37348	0.05	0.026
5	295	2229	4425	41773	0.06	0.034
6	331	2560	4634	46407	0.06	0.038
7	314	2874	4082	50489	0.06	0.036
8	293	3167	3516	54005	0.05	0.033
9	281	3448	3091	57096	0.04	0.032
10	376	3824	3760	60856	0.05	0.043
11	336	4160	3024	63880	0.04	0.038
12	373	4533	2984	66864	0.04	0.043
13	341	4874	2387	69251	0.03	0.039
14	322	5196	1932	71183	0.03	0.037
15	341	5537	1705	72888	0.02	0.039
16	426	5963	1704	74592		0.049
17	371	6334	1113	75705		0.042
18	393	6727	786	76491		0.045
19	376	7103	376	76867		0.043
20	295	7398	0	76867		0.034
21	291	7689	0	76867		0.033
22	246	7935	0	76867		0.028
23	171	8106	0	76867		0.020
24	163	8269	0	76867		0.019
25	100	8369	0	76867		0.011
26	103	8472	0	76867		0.012
27	74	8546	0	76867		0.008
28	67	8613	0	76867		0.008
29	56	8669	0	76867		0.006
30	42	8711	0	76867		0.005
31	26	8737	0	76867		0.003
32	8	8745	0	76867		0.001
33	11	8756	0	76867		0.001
34	2	8758	0	76867		0.000
35	2	8760	0	76867		0.000

Annex B
(informative)

Default values of parameters for the case specific seasonal performance method

B.1.1 Controller setting of flow temperature (heating characteristic curve)

In Figure B 1 typical controller settings and design temperature spreads based on different emission system are given.



Legend

1 Heating characteristic curve
2 Radiators
3 Floor heating
4 flow temperature [°C]
5 Outdoor air temperature [°C]

Figure B.1 —heating curves for radiators and floor heating system

The heating characteristic can be calculated according to the equation given in [7]

$$\theta_f = \theta_{ID} + \frac{\theta_{fD} - \theta_{rD}}{2} \cdot \frac{\theta_{ID} - \theta_{oa}}{\theta_{ID} - \theta_{OD}} + (\theta_{avg} - \theta_{ID}) \cdot (\frac{\theta_{ID} - \theta_o}{\theta_{ID} - \theta_{OD}})^{1/m} \quad [-]$$

eq. B 1

where

θ_f actual flow temperature (°C)

θ_{ID} indoor design temperature (in case of night setback average temperature) (°C)

$\theta_{f,D}$	flow temperature at design conditions	(°C)
$\theta_{r,D}$	return temperature at design conditions	(°C)
θ_{oa}	outdoor air temperature	(°C)
θ_{avg}	arithmetic average of flow and return temperature	(°C)
θ_{OD}	outdoor design temperature	(°C)
m	exponent to characterise the type of emission system (see Table B 1)	(-)

Table B 1 — Parameters of the heating curve for different heating emission systems

Type of emission system	Temperature combination (flow/return)	Exponent m
Radiators	55/45	1.2..1.4
Convectors	55/45	1.2..1.4
Floor heating	40/30	1.0

B.1.2 Temperature correction factor for DHW storage loading

The default value for the temperature correction factor for storage loading is $f_{s,DHW}$ = 0.95.

B.1.3 Average water temperature of DHW storages

The average DHW temperature of DHW storage is set to 90% of the hot water temperature at storage outlet

Table B 2 — Parameters of the heating curve for different heating emission systems

Hot water temperature at storage outlet [°C]	Average temperature of the stored hot water [°C]
60	54
55	49.5
50	45

B.1.4 Generator envelope

Losses of the heat pump are neglected, unless national values exist.

The default values for the maximum heat loss values of the storage are given in the Table B 3 (taken from the Swiss Energy directive [2]). The values refer to system with max. 2 water-filled pipe connections. For each further water-filled connection the values increase by 0.1 kWh/24 h up to a maximum of 0.3 kWh/24 h.

The conditions for the given values are

- An average water temperature of 65°C
- A surrounding temperature of the storage of 20°C
- No hot water draw-off
- System entirely filled with water

Table B 3 — Default values for maximum storage losses with an average water temperature of 65 C and an ambient temperature of 20 C without hot water draw-off [2]

Nominal storage volume [liters]	Maximum heat loss [kWh/24h]
30	0.75
50	0.9
80	1.1
100	1.3
120	1.4
150	1.6
200	2.1
300	2.6
400	3.1
500	3.5
600	3.8
700	4.1
800	4.3
900	4.5
1000	4.7
1200	4.8
1300	5.0
1500	5.1
2000	5.2

B.1.5 Generation subsystem auxiliaries

The part of the nominal electrical power $p_{aux,g}$ transmitted to the ambiance can be calculated

— For pumps and fans with the fraction

$$p_{aux,g} = 1 - \eta_{aux,g} - k_{rd,g}$$

The default value of the hydraulic efficiency $\eta_{aux,g}$ is 0.3 (defined in EN 14511 as pump or fan efficiency). $k_{rd,g}$ is set to 0.5, however, for the calculation of recovered losses it might already by included in the COP

— For other auxiliary components, i.e. auxiliary components without losses to the heat transfer medium, e.g. control

$$p_{aux,g} = 1$$

B.1.6 Factor f_{combi} for simultaneous operation

If no information is available, f_{combi} is set to 1 (controller impact neglected)

B.1.7 Temperature reduction factor linked to location

Table B 4—Default values of the temperature reduction factor

Generator location	Temperature reduction factor b
Inside heated space	0
Outside heated space	1

B.1.8 Efficiency values of the electrical back-up heater for space heating or DHW operation

$\eta_{bu} = 0.95$

Annex C
(informative)

Calculation method for source and sink temperature correction with fixed exergetic efficiency

The idea of the method is that the thermodynamic quality of the process stays constant over the whole operating range. Thermodynamic quality of a process can be expressed by the exergetic effiiciency as ratio between the real COP of the process and an ideal COP of the Carnot process. However, in real processes, the exergetic efficiency does not stay constant over the entire operating range, so the correction is only an approximation which shows good results near the standard test point. Accuracy deteriorates with increasing distance from the test point, and therefore the method is best suited for temperature correction where temperatures are not too far from the test point.

The exergetic efficiency can be calculated according to the equation

$$\eta_{ex} = \frac{COP}{COP_C} \qquad [-]$$

eq. C 1

where

η_{ex}	exergetic efficiency	(-)
COP	coefficient of performance	(W/W)
COP_C	Carnot COP	(W/W)

For electrically-driven heat pumps the Carnot COP is calculated according to the equation

$$COP_C = \frac{T_{hot}}{T_{hot} - T_{cold}} = \frac{\theta_{si} + 273.15}{\theta_{si} - \theta_{so}} \qquad [-]$$

eq. C 2

where

COP_C	Carnot COP	(-)
T_{hot}	temperature on the hot heat pump process side (sink)	(K)
T_{cold}	temperature on the cold heat pump process side (source)	(K)
θ_{si}	sink temperature of the heat pump	(°C)
θ_{so}	source temperature of the heat pump	(°C)

For thermally-driven heat pumps, e.g. absorption heat pumps, however, three temperature levels exist, the hot level of the generator heat input, the warm level of the used heat energy and the cold level of the heat source.
Thus, the Carnot COP is calculated according to the equation

$$COP_C = \frac{\dfrac{T_{in} - T_{cold}}{T_{in} \cdot T_{cold}}}{\dfrac{T_{hot} - T_{cold}}{T_{hot} \cdot T_{cold}}} = \frac{\theta_{si} + 273.15}{\theta_{in} + 273.15} \cdot \frac{\theta_{in} - \theta_{so}}{\theta_{si} - \theta_{so}} \qquad [-]$$

eq. C 3

where

COP_C	Carnot COP	(-)
T_{in}	temperature on the generation side (burner, boiler, heat exchanger)	(K)
T_{hot}	temperature on the hot heat pump process side (sink)	(K)
T_{cold}	temperature on the cold heat pump process side (source)	(K)
θ_{in}	temperature on the generation side (burner, boiler, heat exchanger)	(°C)
θ_{si}	sink temperature of the heat pump	(°C)
θ_{so}	source temperature of the heat pump	(°C)

Both source and sink temperature (and generator temperature in the case of thermally-driven heat pumps) can be considered by this approach. The advantage of the method is that only one test point is needed. In case of testing according to EN 255-3 or EN 12309-2, for instance, where only one test point is defined, an interpolation to correct the COP values for different source or sink temperature is not possible, but a correction with the exergetic efficiency is still applicable.

The effective COP for a different source or sink temperature can be calculated according to the equation

$$COP_{op} = COP_{standard} \cdot \frac{COP_{C,op}}{COP_{C,standard}} \quad [-]$$

eq. C 4

where

COP_{op}	COP due to temperature conditions in operation	(W/W)
$COP_{standard}$	COP due to standard test temperature conditions	(W/W)
$COP_{C,op}$	Carnot COP due to temperature conditions in operation	(-)
$COP_{C,standard}$	Carnot COP due to standard test conditions	(-)

For electrically-driven heat pumps, the correction factor f_T to take into account the impact of different temperatures can be calculated according to the following equation

— For air-to-water heat pumps or water-to-water heat pumps

$$f_T = \frac{COP_{C,op}}{COP_{C,standard}} = \frac{T_{si,out,op} \cdot (\theta_{si,out,standard} - \theta_{so,in,standard})}{T_{si,out,standard} \cdot (\theta_{si,out,op} - \theta_{so,in,op})} \quad [-]$$

eq. C 5

— For air-to-air heat pumps

$$f_T = \frac{COP_{C,op}}{COP_{C,standard}} = \frac{T_{si,in,op} \cdot (\theta_{si,in,standard} - \theta_{so,in,standard})}{T_{si,in,standard} \cdot (\theta_{si,in,op} - \theta_{so,in,op})} \quad [-]$$

eq. C 6

where

f_T	correction factor for temperature deviation from measured standard test point	(-)
$COP_{C,op}$	Carnot COP at source temperature in operation	(-)
$COP_{C,standard}$	Carnot COP at measured standard test point	(-)
$T_{si,out,op}$	outlet temperature on sink side in operation	(K)
$T_{si,out\ standard}$	outlet temperature on sink side at measured standard test point	(K)
$T_{si,in,standard}$	inlet temperature on sink side at measured standard test point	(K)
$\theta_{so,in,op}$	inlet temperature on source side in operation	(°C)

$\theta_{so,in,standard}$	inlet temperature on source side at measured standard test point	(°C)
$\theta_{si,out,op}$	outlet temperature on sink side in operation	(°C)
$\theta_{si,out,standard}$	outlet temperature on sink side at measured standard test point	(°C)
$\theta_{si,in,standard}$	inlet temperature on sink side at measured standard test point	(°C)

For thermally-driven heat pumps it is assumed that T_{in} is the same in both cases, so calculation is accomplished by the following equations

— For air-to-water heat pumps or water-to-water heat pumps

$$f_T = \frac{T_{si,out,op}}{T_{si,out,standard}} \cdot \frac{\theta_{in} - \theta_{so,in,op}}{\theta_{si,out,op} - \theta_{so,in,op}} \cdot \frac{\theta_{si,out,standard} - \theta_{so,in,standard}}{\theta_{in} - \theta_{so,in,stanard}} \quad [-]$$

eq. C 7

— For air-to-air heat pumps

$$f_T = \frac{T_{si,in,op}}{T_{si,in,standard}} \cdot \frac{\theta_{in} - \theta_{so,in,op}}{\theta_{si,in,op} - \theta_{so,in,op}} \cdot \frac{\theta_{si,in,standard} - \theta_{so,in,standard}}{\theta_{in} - \theta_{so,in,stanard}} \quad [-]$$

eq. C 8

where

$T_{so,in,op}$	inlet temperature on source side in operation	(K)
$T_{so,in,standard}$	inlet temperature on source side at measured standard test point	(K)
θ_{gen}	temperature on the generation side (burner, boiler, heat exchanger)	(°C)
$\theta_{si,out,op}$	outlet temperature on sink side in operation	(°C)
$\theta_{si,out,standard}$	outlet temperature on sink side at measured standard test point	(°C)
$\theta_{si,in,standard}$	inlet temperature on sink side at measured standard test point	(°C)
$\theta_{so,in,op}$	inlet temperature on source side in operation	(°C)
$\theta_{so,in,standard}$	inlet temperature on source side at measured standard test point	(°C)

Annex D
(informative)

Calculation example

D.1 Detailed calculation example

D.1.1 System configuration:

Simultaneous combined operating electrically-driven VCC brine-to-water heat pump with cascade cycle according to Figure 3 for space heating and domestic hot water production.

Bivalent design with electrical back-up heater, heating buffer storage in parallel, i.e. primary pump included in the system boundary, external DHW-storage, i.e. storage loading pump included in the system boundary.

Heat pump controlled with ON/OFF control.

D.1.2 Input Data for the calculation (according to chap. 5.2.2):

Table D 1 gives the boundary conditions of the site for the calculation. For the calculation the data of the outdoor air temperature and the respective evaluated data of Table A 1 are used. Table D 2 give the parameters of the space heating system.

Table D 1: Boundary conditions

Meteorological data	
Site (see Annex A)	Gelterkinden, BL, CH
Design outdoor temperature θ_{OD}[°C]	-8

Table D 2: Space heating operation mode

Input data space heating	
Space heating energy requirement $Q_{out,g,h}$ [kWh] (acc. to prEN 15316-2-3)	20158
Indoor design temperature θ_{ID} [°C]	20
Type of heat emission system (radiators/convectors/floor/wall/air)	Radiators
Exponent m of heat emission system [-]	1.2
Design flow temperature θ_{fD} (at outdoor design temperature θ_{OD}) [°C]	55
Temperature spread of SH system at design conditions ΔT_{OD} [K]	10
Upper temperature limit for heating θ_{tlh} [°C]	15
Cut-out time for the heat pump operation t_{co} [h/d]	3
Heating buffer storage (y/n)	Yes
Volume heating buffer storage $V_{s,h}$ [l]	400
Storage stand-by loss $Q_{s,sb}$ [kWh/24h]	2.3
Length of piping between generator and storage L_i [m]	10
Length specific heat loss U_{li} [W/(m·K)]	0.2
Temperature difference during storage testing $\theta_{s,sb}$ [°C]	40
Surrounding temperature of the storage θ_{sur} [°C]	15
Back-up heater (y/n)	Yes
Type of back-up heater (Electrical/Gas/Oil)	Electrical
Operation mode of back-up heater (alternative/parallel/partly parallel)	parallel
Efficiency of the back-up heater η_{bu}	0.95

Table D 3 gives the parameter for the DHW system. Table D 4 contains general parameters of the heat pumps, and Table D 5, Table D 6 and Table D 7 the characteristic of the heat pump for the different operation modes according to standard testing.

Table D 3: DHW operation mode

Input data DHW	
DHW energy requirement $Q_{out,g,DHW}$ [kWh] (acc. to EN 15316-3-2)	3300
Cold water inlet temperature θ_{cw} [°C]	15
Hot water temperature at storage outlet θ_{hw} [°C]	60
Volume DHW storage $V_{s,DHW}$ [l]	300
Factor simultaneous operation f_{combi} [-]	0.7
Back up heater has the same parameters as for space heating system	
Piping between generator and DHW storage has the same parameter as for the space heating system	

Table D 4: Heat pump

Heat pump	
Type of heat pump (A/W, A/A, B/W, W/W, DX/W, DX/A, DX/DC)	B/W
Type of DHW production (None/only/alternate/simultaneous)	simultaneous combined
Control of the heat pump (ON-OFF, step, variable speed)	ON-OFF
Operating limit $\theta_{op,hp}$ [°C]	55

Table D 5: Characteristic of the heat pump for space heating-only operation (EN 14511)

Characteristic heat pump			
Space heating-only (EN 14511)			
source temperature at test point θ_{so} [°C]	**-5**	**0**	**5**
sink temperature at first test point θ_{si} [°C]	**35**		
sink temperature at second test point θ_{si}[°C]	**45**		
COP at sink temperature 35°C [W/W]		4.5	5.2
COP at sink temperature 45°C [W/W]	3.0	3.5	4.0
heating capacity at sink temperature 35°C $\phi_{hp,h,sin}$ [kW]		9	10.3
heating capacity at sink temperature 45°C $\phi_{hp,h,sin}$ [kW]	7.7	8.7	9.9

Table D 6: Characteristic of the heat pump for space heating-only operation (EN 255-3)

DHW-only (EN 255-3)			
	-5	**0**	**5**
$COP_{t,DHW}$ (EN 255-3) [W/W]		2.36	
Power input for storage stand-by losses $P_{es,sin}$ [W] (EN 255-3)		62	
Heating capacity DHW only $\phi_{hp,DHW,sin}$[kW]	0.7	1.05	1.4

NOTE: Presently, EN 255-3 does not deliver a DHW heating capacity of the heat pump as an output. However, based on the test procedure an average heating capacity of the cycle performed in phase 2 can be calculated according to the equation without extension of the testing.

$$\phi_{hp,DHW} = \frac{Q_{out,DHW}}{t_t}$$

eq. D 1

At the time of writing EN 255-3 was under revision in CEN/TC 113/WG 10.

Table D 7: Characteristic of the heat pump for simultaneous operation (EN 255-3 during heating operation)

Simultaneous combined (EN 255-3 during heating operation)			
	-5	**0**	**5**
COP at sink temperature 35 of the heating system COP_{combi} [W/W]		4.09	4.67
COP at sink temperature 45 of the heating system COP_{combi} [W/W]	2.7	3.01	3.43
Heating capacity space heating combi at sink 35 $\phi_{hp,h,combi}$ [kW]		9.0	10.3
Heating capacity space heating combi at sink 45 $\phi_{hp,h,combi}$ [kW]	7.5	8.5	9.7
Heating capacity DHW combi at sink 35 $\phi_{hp,DHW,combi}$ [kW]		1.75	1.83
Heating capacity DHW combi at sink 45 $\phi_{hp,DHW,combi}$ [kW]	1.7	1.89	2.04
Power input for storage stand-by losses $P_{es,combi}$ [W] (EN 255-3)		36	

NOTE: Presently, there is no European test standard for simultaneous operating heat pump systems, thus national method shall be used until a European standard is available. This example has been calculated based on testing, where during the heating operation the DHW cycle according to EN 255-3 has been performed.

The COP in combined operation has been evaluated according to the equation

$$COP_{combi} = \frac{Q_{out,h} + Q_{out,DHW}}{E_{in}} \qquad [W/W]$$

eq. D 2

Table D 8 gives the parameters of the auxiliary components.

Table D 8: Power consumption of auxiliary components

Auxiliary components	
Power source pump P_{so} [W]	120
Power sink pump P_{si} [W]	54
Power storage loading pump P_s [W]	33
Power control and carter heating P_{OFF} [W]	10
Fraction of auxiliary energy input loss to the ambiance $p_{aux,g}$ [-]	0.2
Temperature reduction factor b_g [-]	1

D.1.3 Calculation

The calculation is accomplished step by step. For each step, the lowest bin is calculated by means of the respective equation of the normative part, for the other bins, the values are contained in a summarising table at the end of the step.

NOTE: Due to rounding of the results the exactness of the numbers can deviate slightly between the equations and the tables

D.1.3.1 Step 1: Calculation of the energy requirements in the bins

D.1.3.1.1 Definition of the bin distribution

- Operating points and number of bins have been chosen the same as for the test points of an air-to-water heat pump (3 bins for combined space heating/DHW, 1 bin for DHW-only), see also NOTE below.
- Lower limits and upper limits are calculated as approximate midpoint temperature between the operating points rounded to an integer value.
- The lower bin limit of the first bin is the lowest outdoor temperature which occurs in the meteo data set

— The upper bin limit of the last bin for space heating is the upper temperature limit for heating

NOTE: Since return temperatures from the ground heat exchanger depend on the characteristics of the ground, the test point temperatures are not directly related to an outdoor air temperature. Therefore, the same operating points as for the test points of air-to-water heat pumps have been chosen for this example, since these points are more or less evenly spread over the operating range. By the evaluation of the return temperature of the ground heat exchanger, this results in an operating point of the lowest bin 1 which is close to the test point B0. The operating point at 20°C outdoor air temperature is close to the test point B5. The two operating points inbetween have been set to reflect the change in the sink temperature due to the heating curve and have to be interpolated. Meteorological data given in Annex A have been used.

D.1.3.1.2 Calculation of bin time according to eq. 7

$$t_i = N_{hours,\theta upper,i} - N_{hours,\theta lower,i} = 330h - 0h = 330h$$

eq. D 3

D.1.3.1.3 Calculation of effective bin time according to eq. 8

$$t_{i,eff} = t_i \cdot \frac{24 - t_{co}}{24} = 330h \cdot \frac{24h - 3h}{24h} = 289h$$

eq. D 4

D.1.3.1.3.1 Calculation of weighting factors and energy requirements space heating acc. to eq. 5 and eq. 6 according to Table A 1

$$w_{h,i} = \frac{CHDH_{\theta upper,i} - CHDH_{\theta lower,i}}{CHDH_t} = \frac{CHDH_{\theta=(-2)} - CHDH_{\theta=(-11)}}{CHDH_{\theta=15}} \frac{7940Kh - 0Kh}{72888Kh} = 0.11 \qquad [-]$$

eq. D 5

$$Q_{out,g,h,i} = Q_{out,g,h} \cdot w_i = 20158kWh \cdot \frac{7940Kh}{72888Kh} = 2196kWh \qquad [kWh]$$

eq. D 6

D.1.3.1.3.2 Calculation of weighting factors and energy requirements DHW according to eq. 9 and eq. 10

$$w_{DHW,i} = \frac{t_i}{t_t} = \frac{330h}{8760h} = 0.04 \qquad [-]$$

eq. D 7

$$Q_{out,g,DHW,i} = Q_{out,g,DHW} \cdot \frac{n_{hours,i}}{n_{hours,t}} = 3300kWh \cdot \frac{330h}{8760h} = 124kWh \qquad [kWh]$$

eq. D 8

Table D 9 summarises the calculated data for all bins.

Table D 9: Summary of results step 1: Calculation of the energy requirements in the bins

Step 1: Energy requirements	**Bins space heating/DHW**			**DHW-only**	**Sum**
Bin number	Bin 1	Bin 2	Bin 3	Bin 4	
Operating points (chap. 5.2.1)	**-7**	**2**	**7**	**20**	
Lower bin limit (chap. 5.2.1)	-11	-2	4	15	
Upper bin limit (chap. 5.2.1)	-2	4	15	35	
bin time (eq. 7)	330	1604	3603	3223	**8760**
effective bin time (eq. 8)	289	1404	3153	2820	**7665**
weighting factor (eq. 5)	0.11	0.40	0.49		**1.00**
Energy requirement for SH (eq. 6)	2196	8133	9829		**20158**
weighting factor DHW (eq. 9)	0.04	0.18	0.41	0.37	**1.00**
Energy requirement for DHW (eq. 10)	124	604	1357	1214	**3300**

D.1.3.2 Step 2: Calculation of heating capacity and COP at full load (chap. 5.2.5)

D.1.3.2.1 Calculation of source and sink temperature at operating points

Calculation of source temperature (Polynomfit of sample profile Annex F.1.1.3, Figure F 1)

The source temperature, i.e. the return temperature from the ground heat exchanger is deduced from the profile given in Figure F 1 for the dependency of the return temperature of the ground heat exchanger on the outdoor air temperature according to the linear fit

$$\theta_{so,in} = 0.15 \cdot \theta_{oa} + 1.5 = 0.15 \cdot (-7°C) + 1.5 = 0.5°C$$

eq. D 9

Calculation of flow temperature at test points (heating curve Annex B, eq. B 1)

For the calculation of the flow temperature $\theta_f = \theta_{si,out}$, the heating curve is evaluated according to the equation

$$\theta_f = \theta_{ID} + \frac{\theta_{fD} - \theta_{rD}}{2} \cdot \frac{\theta_{ID} - \theta_{oa}}{\theta_{ID} - \theta_{OD}} + (\theta_{avg} - \theta_{ID}) \cdot \left(\frac{\theta_{ID} - \theta_{oa}}{\theta_{ID} - \theta_{OD}}\right)^{1/m}$$

eq. D 10

$$\theta_{si,out} = 20°C + \frac{55°C - 45°C}{2} \cdot \frac{20°C - (-7°C)}{20°C - (-8°C)} + (50°C - 20°C) \cdot \left(\frac{20°C - (-7°C)}{20°C - (-8°C)}\right)^{1/1.2} = 53.9°C$$

D.1.3.2.2 Interpolation of heating capacity and COP for temperature conditions

Linear interpolation of the heating capacity in direction of the source temperature according to chap. 5.2.5.1.2

$$\phi_{hp,h,sin(so,in,W35)} = \frac{\phi_{hp,h\,sin(B5/W35)} - \phi_{hp,h,sin(B0/W35)}}{5°C - 0°C} \cdot (\theta_{so,in} - 0°C) + \phi_{hp,h,sin(B0/W35)}$$
$$= \frac{10.3 - 9.0}{5°C} \cdot 0.5°C + 9.0kW = 9.1kW$$

eq. D 11

Linear interpolation of the heating capacity in direction of the sink temperature

$$\phi_{hp,h,sin(\theta so,in,\theta si,out)} = \frac{\phi_{hp,h,sin(\theta so,in/W45)} - \phi_{hp,h,sin(\theta so,in/W35)}}{45°C - 35°C} \cdot (\theta_{si,out} - 35°C) + \phi_{hp,h,sin(\theta so,in/W35)}$$
$$= \frac{8.8kW - 9.1kW}{10°C} \cdot 19°C + 9.1kW = 8.5kW$$

eq. D 12

D.1.3.2.3 Correction for temperature conditions during testing and operation

For different mass flow rates during standard testing and operation different temperature conditions at the condenser of the heat pump result, so the standard COP characteristic has to be corrected, incorporating the following calculations to determine the flow and temperature conditions during testing and operation.

Calculation of mass flow rate at standard rating conditions of EN 14511

EN 14511 defines a temperature spread on the condenser side of 5 K for the standard rating point (B0/W45) for radiator emission systems, thus the mass flow follows acc. to eq. 11 after rearranging for the mass flow rate

$$\dot{m}_{test} = \frac{\phi_{hp,h,sin(B0/W45)}}{\Delta\theta_{test} \cdot c_w} = \frac{8.7kW}{5K \cdot 4.182kJ/kg/K} = 0.42kg/s$$

eq. D 13

This mass flow of the standard rating point is used for all test points, so the respective temperature spread during testing at test points can be calculated by eq. 11

$$\Delta\theta_{standard} = \frac{\phi_{hp,h,sin(B-5/W45)}}{\dot{m}_{standard} \cdot c_w} = \frac{7.7kW}{0.42kg/s \cdot 4.182kJ/kg/K} = 4.4K$$

eq. D 14

Mass flow rate in operation (evaluated at design outdoor temperature θ_{OD}) acc. to eq. 11

$$\dot{m}_{eff} = \frac{\phi_{hp,h,sin(\theta OD,\theta si,out @ \theta OD)}}{\Delta\theta_{OD} \cdot c_w} = \frac{8.5kW}{10K \cdot 4.182kJ/kg/K} = 0.2kg/s$$

eq. D 15

Temperature spread with effective mass flow rate at test point acc. to eq. 11

$$\Delta\theta_{eff} = \frac{\phi_{hp,h,sin(B-5/W45)}}{\dot{m}_{eff} \cdot c_w} = \frac{7.7kW}{0.2kg/s \cdot 4.182kJ/kg/K} = 9.2K$$

eq. D 16

Correction for different temperature spreads during testing and in operation acc. to eq. 12

Since the condenser and the evaporator use brine and water respectively, both average temperature differences between refrigerant and heat transfer medium are set to ΔT_{si}=ΔT_{so} =4 K.

$$COP_{\Delta\theta} = COP_{standard} \cdot \left[1 - \frac{\frac{\Delta\theta_{standard} - \Delta\theta_{eff}}{2}}{(T_{si} - \frac{\Delta\theta_{standard}}{2} + 4[K] - (T_{so} - 4[K]))} \right]$$

eq. D 17

$$= COP_{(B-5,W45)} \cdot \left[1 - \frac{\frac{4.4K - 9.2K}{2}}{((273.15 + 45)K - \frac{4.4K}{2} + 4[K] - (273.15 + (-5))K - 4[K]))} \right]$$

$$= 3.0 \cdot 1.04 = 3.1$$

D.1.3.2.4 Interpolation of COP for space heating

Based on the corrected $COP_{\Delta\theta}$ characteristic the COP is linear interpolated in direction of the source temperature and the sink temperature in the same way as the heating characteristic.

D.1.3.2.5 COP and heating capacity for DHW-only operation

Since only one test point at B0 is delivered by testing according to EN 255-3 the $COP_{t,DHW}$ is corrected by the method of constant exergetic efficiency given in Annex C. Correction factors are calculated with the eq. C 5. The equation can also be applied for the correction of different sink temperatures in testing and operation, e.g. testing at 50°C hot water temperature and operation at 60°C. However, the method is only exact near the test point and exactness decreases with increasing distance to the test point.

$$f_T = \frac{COP_{C,op}}{COP_{C,standard}} = \frac{T_{si,out,op} \cdot (\theta_{si,out,standard} - \theta_{so,in,standard})}{T_{si,out,standard} \cdot (\theta_{si,out,op} - \theta_{so,in,op})} = \frac{(273.15 + 60)K \cdot (60 - 0)K}{(273.15 + 60)K \cdot (60 - 0.5)K} = 1.01$$

eq. D 18

The $COP_{t,DHW}$ for the operation conditions is thus calculated

$$COP_{t,DHW} = COP_{t,DHW,standard} \cdot f_T = 2.36 \cdot 1.01 = 2.38$$

eq. D 19

If only one test point is available, heating capacity for DHW cannot be corrected for changing temperature conditions. However, in this case, testing at three source temperatures was carried out, so the DHW heating capacity can be interpolated as done for the space heating.

D.1.3.2.6 COP and heating capacity for simultaneous operation

The COP and heating capacity for combined operation are calculated as done for the space heating mode. Since in this system the heating capacity in SH-only and simultaneous operation does not change significantly, the same correction factors for the correction of different temperature spreads can be used.

Due to the condensate subcooling, the DHW heating capacity is dependent on the temperature requirement of the space heating system and has to be interpolated based on the flow temperature of the space heating system.

NOTE: Lines in Table D 10 marked in italic are only required in case of simultaneous combined operating systems with three operation modes.

The DHW operation is much more efficient in simultaneous combined operation, thus it is important to consider all three operation modes. The higher values of the combined DHW heating capacity and combined

COP are due to the higher temperatures of the condensate subcooling in comparison to the ground source, i.e. the return temperature of the space heating system (see Figure 3).

Table D 10: Summary of results step 2: COP and heating capacity at full load

Step 2 COP and heating capacity at full load				
	Bin 1	Bin 2	Bin 3	Bin 4
	[-11..2]	[-2..4]	[4..15]	[15..35]
SH only				
Parameter source temperature	0.15	1.5		
Source temperature (Polynomfit, standard profile, chap. F.1.1.3)	0.5	1.8	2.6	4.5
Interpolation heating capacity for supply 35 (chap. 5.2.5.1.2)	9.1	9.5	9.7	10.2
Interpolation heating capacity for supply 45 (chap. 5.2.5.1.2)	8.8	9.1	9.3	9.8
Flow temperature at operating points (heating curve, Annex B.1.1)	54	44	38	
Heating capacity SH-only at supply temperature (chap. 5.2.5)	**8.5**	**9.2**	**9.6**	
mass flow at standard rating point acc. to EN 14511 (B0/W45)[kg/s]		0.42		
temperature spread at test points 35 [K] (eq. 11)		5.2	5.9	
temperature spread at test points 45 [K] (eq. 11)	4.4	5.0	5.7	
Source temperature at design conditions [°C] (chap. F.1.1.3)	0.3			
Flow temperature at design conditions [°C] (Annex B.1.1)	55			
heating capacity at design conditions source 35 [kW]	9.1			
heating capacity at design conditions source 45 [kW]	8.8			
heating capacity at design flow temperature [kW]	8.5			
mass flow in operation [kg/s] (eq. 11)	0.2			
temperature spread in operation 35 [K] (eq. 11)	9.3	10.6	12.2	
temperature spread in operation 45 [K] (eq. 11)	9.1	10.3	11.7	
correction factor for the temperature spread 35 [-] (eq. 12)	1.05	1.07	1.09	
correction factor for temperature spread 45 [-] (eq. 12)	1.04	1.05	1.07	
corrected COP value 35 [W/W]		4.8	5.7	
corrected COP value 45 [W/W]	3.1	3.6	4.3	
Interpolation COP source for sink temperature 35 (chap. 5.2.5.1)	4.8	5.1	5.2	5.6
Interpolation COP source for sink temperature 45 (chap. 5.2.5.1)	3.7	3.9	4.0	4.2
Interpolation of COP SH-only (sink temperature) (chap. 5.2.5.1)	**2.7**	**4.0**	**4.8**	
DHW-only				
Correction factor of the COP for DHW (Annex C, eq. C 5)	1.01	1.03	1.04	1.08
COP DHW-only	**2.4**	**2.4**	**2.5**	**2.6**
heating capacity for DHW-only (chap. 5.2.5.1)	**1.1**	**1.2**	**1.2**	**1.4**
Interpolation heating capacity DHW sink 35	*1.76*	*1.78*	*1.79*	
Interpolation heating capacity DHW sink 45	*1.85*	*1.89*	*1.91*	
heating capacity for DHW combined	***1.93***	***1.88***	***1.83***	
combi				
Interpolation heating capacity space heating combi for sink 35	*9.1*	*9.5*	*9.7*	
Interpolation heating capacity space heating combi for sink 45	*8.8*	*9.1*	*9.3*	
heating capacity SH combined at flow temperature	***8.5***	***9.2***	***9.6***	
corrected COP value 35 [W/W]	*3.8*	*4.4*	*5.1*	
corrected COP value 50 [W/W]	*3.1*	*3.5*	*4.1*	
Interpolation COP-combi source for sink 35	*4.4*	*4.6*	*4.7*	
Interpolation COP-combi source for sink 45	*3.6*	*3.7*	*3.8*	
Interpolation of COP-combi (flow temperature)	***2.8***	***3.8***	***4.4***	

D.1.3.3 Step 3: Correction for part load operation

Since for this example, no test data according to CEN/TS 14825 are available, cyclic losses are neglected and only the auxiliary energy during stand-by operation is used for the consideration of part load operation.

D.1.3.4 Step 4: Calculation of generation subsystem losses

Since no heat loss value for the heat pump envelope loss is given, the envelope losses of the heat pump are neglected.

D.1.3.4.1 Calculation for the heating buffer storage

<u>Average storage temperature heating buffer storage</u>

The average storage temperature $\theta_{s,avg,i}$ is approximated as average temperature of the flow and return temperature according to eq. 17

$$\theta_{s,avg,i} = \frac{(\theta_{si,out,i} + \theta_{r,i})}{2} = \frac{54°C + (54°C - 10°C)}{2} = 49°C$$

eq. D 20

The return temperature at the operating points is calculated by interpolation between the design temperature spread and the temperature spread 0 at the indoor temperature (chap. 5.2.7.1.1).

<u>Losses heating buffer storage</u>

$$Q_{l,s,h,i} = \frac{\theta_{s,avg,i} - \theta_{sur}}{\Delta\theta_{s,sb}} \cdot Q_{sb} \cdot t_i = \frac{(49-15)K}{45K} \cdot 2.3(kWh/24h) \cdot \frac{330h}{24h} = 23.9kWh$$

eq. D 21

<u>Losses of the piping heating buffer storage (chap. 5.2.7.1.3, according to equation A1 of prEN 15316-3-3)</u>

$$Q_{l,g,h,i} = \frac{1}{1000} \cdot U_{li} \cdot L_i \cdot (\theta_{si,out,i} - \theta_{sur}) \cdot t_{ON,hp,DHW} = \frac{1}{1000} \cdot U_{li} \cdot L_i \cdot (\theta_{si,out,i} - \theta_{sur}) \cdot \frac{Q_{out,g,h,i} + Q_{l,s,h,i}}{\phi_{hp,sin,i}}$$

$$= \frac{1}{1000} \cdot 0.2W/mK \cdot 10m \cdot (54-15)K \cdot \frac{(2196+24)kWh}{8.5kW} = 20.4kWh$$

eq. D 22

<u>Total losses space heating buffer storage</u>

$$Q_{l,g,h,i} = Q_{l,s,h,i} + Q_{l,g,p,i} = 23.9kWh + 20.4kWh = 44.3kWh$$

D.1.3.4.2 Calculation for the DHW storage

The losses of the DHW storage are calculated with the default value based on the storage size according to Table B 3. The maximum storage losses for a storage size of 300 l are 2.6 kWh/24h for a temperature difference of 45°C. Corrected losses for the temperature conditions can be calculated according to eq. 15.

The average storage temperature is evaluated the following: The heat exchanger is positioned at the lower third of the storage. Around the heat exchanger, an average temperature of 30°C and for the rest of the storage the required DHW temperature at storage outlet is assumed. Thus, the average storage temperature is calculated according to chap. 5.2.7.1.2

$$\theta_{s,avg,i} = 0.67 \cdot \theta_{hw} + 0.33 \cdot \theta_{s,hx,DHW} = (0.67 \cdot 60°C + 0.33 \cdot 30°C) = 50°C$$

eq. D 23

<u>Losses of the DHW storage according to eq. 15</u>

$$Q_{l,s,DHW,i} = \frac{\theta_{s,avg,i} - \theta_{sur}}{\Delta\theta_{s,sb}} \cdot Q_{sb} \cdot t_i = \frac{(50-15)K}{45K} \cdot 2.6(kWh/24h) \cdot \frac{330h}{24h} = 27.8kWh$$

eq. D 24

<u>Losses of the DHW piping between generator and storage</u>
<u>(chap. 5.2.7.1.3, according to equation A1 of prEN 15316-3-3)</u>

$$Q_{l,g,DHW} = \frac{1}{1000} \cdot U_{li} \cdot L_i \cdot (\theta_{hw} - \theta_{sur}) \cdot t_{ON,hp,DHW} = \frac{1}{1000} \cdot 0.2W/mK \cdot 10m \cdot (60-15)K \cdot \frac{(124+28)kWh}{1.1kW} = 12.4kWh$$

eq. D 25

<u>Total losses space DHW storage (chap. 5.2.7.1.3)</u>

$$Q_{l,g,DHW,i} = Q_{l,s,DHW,h,i} + Q_{l,s,DHW,i} = 27.8kWh + 12.4kWh = 40.2kWh$$

eq. D 26

D.1.3.4.3 Total storage envelope losses of the generation subsystem

$$Q_{l,g,i} = Q_{l,g,h} + Q_{l,g,DHW} = 44kWh + 40kWh = 84kWh$$

eq. D 27

Table D 11 gives a summary of the envelope losses of the generation subsystem.

NOTE: For the calculation of pipe losses, the running time for space heating and DHW operation has been estimated by single operation in order to avoid iterations.

Table D 11: Summary of results step 4: Calculation of the heat losses of the generation subsystem

Step 4 Calculation of envelope heat losses					
	Bin 1	Bin 2	Bin 3	Bin 4	Sum
	[-11..-2]	[-2..4]	[4..15]	[15..35]	
SH					
return temperature at operation points (chap. 5.2.7.1.1)	44	38	34		
average storage temperature heating buffer [°C] (eq. 17)	49	41	36		
storage losses heating buffer storage [kWh] (eq. 15)	24	88	160		**272**
Losses piping heating buffer [kWh] (prEN15316-3-3 eq. A1)	20	52	48		**120**
Total losses heating buffer [kWh] (chap. 5.2.7.1.3 and eq. 16)	44	140	208		**392**
DHW					
average storage temperature DHW (chap. 5.2.7.1.2) [°C]	50	50	50	50	
Losses piping DHW storage [kWh] (prEN15316-3.3 eq. A1)	12	57	122	98	**289**
Thermal losses DHW storage [kWh] (eq. 15)	28	136	304	272	**740**
Total losses DHW storage [kWh] (chap. 5.2.7.1.3 and eq. 16)	40	192	426	370	**1028**
Total thermal losses of the generator subsystem	**84**	**332**	**634**	**370**	**1420**

D.1.3.5 Step 5: Calculation of the back up energy

D.1.3.5.1 Back-up energy due to temperature operation limit of the heat pump

<u>Space heating</u>

Maximum system temperature requirements are below the operation limit of 55°C, so no back-up operation due to operation limit temperature is required in space heating mode.

<u>Domestic hot water</u>

The fraction of back-up heater operation for DHW due to the operation limit of the heat pump is calculated according to eq. 19

$$p_{bu,DHW,op,i} = \frac{Q_{bu,DHW,op,i}}{Q_{out,g,DHW}} = \frac{\theta_{hw} - \theta_{upper,hp}}{\theta_{hw} - \theta_{cw}} = \frac{(60-55)\mathrm{K}}{(60-15)\mathrm{K}} = 0.11$$

eq. D 28

D.1.3.5.2 Back-up energy due to lack of capacity of the heat pump

In this calculation example the detailed balance to determine the back-up energy due to a lack of capacity is applied, see chap. 5.2.9.3

For the detailed calculation of the back-up energy the running time is evaluated in 1 K bins, i.e. all energies, losses and heating capacities for the respective operation modes have to be determined in 1 K steps until no back-up energy is required anymore.

Table D 12 gives a shortened overview of the resulting table for the detailed calculation of the back-up energy.

For the bins with a lack of running time, the heating capacity of the heat pump is not sufficient to cover the total requirement. The resulting back-up energy can be calculated based on the control strategy. Here it is assumed that control supplies the back-up energy to the space heating system.

Table D 12: Detailed determination of back-up energy

Outdoor air temperature (1 K bin)	Energy to be produced for SH $Q_{hp,h}$ [kWh] (eq. 30)	Energy to be produced for DHW $Q_{hp,DHW}$ [kWh] (eq. 30)	Heating capacity SH $\phi_{hp,h,sin}$ [kW]	Heating capacity DHW $\phi_{hp,DHW,sin}$ [kW]	Heating capacity combined SH $\phi_{hp,combi,h}$ [kW]	Heating capacity combined DHW $\phi_{hp,combi,DHW,I}$ [kW]	Running time combined $t_{ON,hp,combi}$ [h] (eq. 32, eq. 35)	Running time SH-only $t_{ON,hp,h}$ [h] (eq. 29)	Running time DHW-only $t_{ON,hp,DHW}$ [h] (eq. 29)	Total required running time $t_{ON,hp,t}$ [h] (eq. 31)	Effective bin time (1 K bin) $t_{i,eff}$ [h] (eq. 8)	Difference required running time to effective running time	Required back-up energy (eq. 41)
-11	26	1	8.3	1.0	8.3	2.0	0.5	2.7	0.4	3.6	2.6	1.0	8
-10	59	3	8.3	1.1	8.3	1.9	1.1	6.0	0.9	8.0	6.1	1.9	16
-9	65	4	8.4	1.1	8.4	1.9	1.3	6.5	1.0	8.8	7.0	1.8	15
-8	95	5	8.5	1.1	8.5	1.9	1.9	9.3	1.5	12.7	10.5	2.2	18
-7	183	11	8.5	1.1	8.5	1.9	3.8	17.6	2.9	24.3	21.0	3.3	28
-6	242	15	8.6	1.1	8.6	1.9	5.3	22.8	4.0	32.1	28.9	3.2	28
-5	218	14	8.7	1.1	8.7	1.9	5.0	20.2	3.7	28.9	27.1	1.8	15
-4	176	11	8.7	1.1	8.7	1.9	4.2	15.9	3.1	23.2	22.8	0.4	4
-3	466	32	8.8	1.1	8.8	1.9	11.6	41.3	8.5	61.4	63.0	0.0	0
Σ													132

The resulting back-up energy for the 1 K bin is calculated according to eq. 41

$$Q_{bu,cap,i} = (t_{ON,hp,t,i} - t_{i,eff}) \cdot \phi_{hp,sin,i} = (3.6\text{h} - 2.6\text{h}) \cdot 8.3\text{kW} = 8.3\text{kWh}$$

eq. D 29

D.1.3.5.3 Fraction of back-up energy for space heating

The back-up energy can be summed up over the 1 K bins for the bin limits and the fraction of back-up energy in the respective bin can be calculated with the eq. 42

$$p_{bu,h,i} = \frac{Q_{bu,cap}}{Q_{out,g,h,i}} = \frac{132\text{kWh}}{2196\text{kWh}} = 0.06$$

eq. D 30

Table D 13 gives a summary of the fraction for back-up operation.

Table D 13: Summary of results step 5: Calculation of the back-up energy of the generation subsystem

Step 5 Fractions of back-up energy				
	Bin 1	Bin 2	Bin 3	Bin 4
	[-11..-2]	[-2..4]	[4..15]	[15..35]
Fraction back-up SH due to operation limit temperature (eq. 18)	0	0	0	0
Fraction back-up SH due to lack of capacity (eq. 41)	0.06	0	0	0
Total fraction back-up space heating (eq. 42**)**	**0.06**	**0.000**	**0.000**	**0.000**
Fraction back-up DHW due to operation limit temperature (eq. 19)	0.11	0.11	0.11	0.11
Fraction of back-up DHW due to lack of capacity (eq. 41)	0	0	0	0
Total fraction back-up DHW (eq. 42**)**	**0.11**	**0.11**	**0.11**	**0.11**

D.1.3.6 Step 6: Calculation of the running time for simultaneous operating systems with three operation modes

For simultaneous operation with three operation modes, SH-only, DHW-only and simultaneous SH/DHW, the running time in simultaneous operation is evaluated as comparison of the running time for both operation modes. The boundary condition for the time in simultaneous operation is the minimum of the running time due to the SH and DHW energy requirement.

Running time for DHW operation is thus calculated with the DHW heating capacity in simultaneous operation according to eq. 33

$$t_{ON,hp,DHW,i} = \frac{Q_{hp,DHW,i}}{\phi_{hp,DHW,combi,i}} = \frac{(Q_{out,DHW} + Q_{l,s,DHW}) \cdot (1 - p_{bu,DHW,op,i})}{\phi_{hp,DHW,combi,i}}$$
$$= \frac{(124\text{kWh} + 41\text{kWh}) \cdot (1 - 0.11)}{1.93\text{kW}} = 76\text{h}$$

eq. D 31

Analogous the running time for space heating operation is calculated according to eq. 34

$$t_{ON,hp,h,i} = \frac{Q_{hp,h,i}}{\phi_{hp,h,combi,i}} = \frac{(Q_{out,g,h} + Q_{l,s,h}) \cdot (1 - p_{bu,h,i})}{\phi_{hp,h,combi,i}} = \frac{(2196\text{kWh} + 44\text{kWh}) \cdot (1 - 0.06)}{8.5\text{kW}} = 247\text{h}$$

eq. D 32

And therefore the maximum running time in simultaneous operation follows according to eq. 32

$$t_{ON,hp,combi,max,i} = \min(t_{ON,hp,h,i}; t_{ON,hp,DHW,i}) = \min(247h; 76h) = 76h$$

eq. D 33

However, load shift and controller impact may reduce simultaneous operation which is considered in a correction factor f_{combi} depending on the applied control strategy. For the system in Figure 3 the factor has been evaluated for the particular system configuration to f_{combi} = 0.7, so the effective running time in simultaneous operation is calculated according to eq. 35

$$t_{ON,hp,combi,i} = t_{ON,hp,combi,max,i} \cdot f_{combi} = 76h \cdot 0.7 = 53h$$

eq. D 34

The DHW-energy produced in simultaneous operation is calculated according to eq. 36

$$Q_{hp,DHW,combi,i} = t_{ON,hp,combi,i} \cdot \phi_{hp,DHW,combi,i} = 53h \cdot 1.93kW = 102kWh$$

eq. D 35

The rest of the DHW-energy is produced in single operation and is determined according to eq. 37

$$Q_{hp,DHW,sin,i} = Q_{hp,DHW,i} - Q_{hp,DHW,combi,i} = 146kWh - 102kWh = 44kWh$$

eq. D 36

The space heating energy produced in simultaneous operation is calculated analogously according to eq. 36

$$Q_{hp,h,combi,i} = t_{ON,hp,combi,i} \cdot \phi_{hp,h,combi,i} = 53h \cdot 8.5kW = 451kWh$$

eq. D 37

and according to eq. 37

$$Q_{hp,h,sin,i} = Q_{hp,h,i} - Q_{hp,h,combi,i} = 2105kWh - 451kWh = 1654kWh$$

eq. D 38

The allocation of the storage losses to the single and simultaneous operation mode is done by the fraction of simultaneous operation which corresponds to f_{combi}.

Since EN 255-3 delivers the electricity input to cover the storage losses. Thus, the DHW energy requirement in DHW-only and in simultaneous operation can be calculated by subtracting the storage losses according to eq. 39

$$Q_{out,hp,DHW,combi,i} = Q_{hp,DHW,sin,i} - Q_{l,s,DHW,i} \cdot (1 - p_{bu,DHW,i}) \cdot f_{combi} = 102kWh - 40kWh \cdot (1 - 0.11) \cdot 0.7 = 77kWh$$

eq. D 39

and according to eq. 38

$$Q_{out,hp,DHW,sin,i} = Q_{hp,DHW,sin,i} - Q_{l,s,DHW,i} \cdot (1 - p_{bu,DHW,i}) \cdot (1 - f_{combi}) = 44kWh - 40kWh \cdot (1 - 0.11) \cdot (1 - 0.7) = 33kWh$$

eq. D 40

The running time in DHW-only operation can be calculated by eq. 29

$$t_{ON,hp,DHW,sin,i} = \frac{Q_{hp,DHW,sin,i}}{\phi_{hp,DHW,sin,i}} = \frac{44kWh}{1.1kW} = 40h$$

eq. D 41

And respectively the running time in space heating-only operation eq. 29

$$t_{ON,hp,h,sin,i} = \frac{Q_{hp,h,sin,i}}{\phi_{hp,h,sin,i}} = \frac{1654kWh}{8.5kW} = 195h$$

eq. D 42

The total running time can be calculated by the equation eq. 31

$$t_{ON,hp,t,i} = \min(t_{i,eff}\ ; t_{ON,hp,h,sin,i} + t_{ON,hp,DHW,i} + t_{ON,hp,combi,i})$$

$$= \min(289h;195h + 40h + 53h) = \min(289h;288h) = 288h$$

eq. D 43

The total running time of the heat pump is limited by the effective bin time. Due to the detailed calculation for the back-up energy, no correction due to the limit of running time has to be applied.

Table D 14 gives a summary of the running time and the produced energies in the different operation modes.

Table D 14: Summary of results step 6: Calculation of the running time of the generation subsystem

Step 6 Calculation of running time					
	Bin 1	Bin 2	Bin 3	Bin 4	Sum
	[-11..-2]	[-2..4]	[4..15]	[15..35]	
running time of the heat pump for SH [h] (eq. 33)	*247*	*902*	*1049*	*0*	***2198***
running time of the heat pump for DHW [h] (eq. 34)	*76*	*377*	*868*	*1032*	***2353***
max. running time combined (=min(SH;DHW)) [h] (eq. 32)	*76*	*377*	*868*	*1032*	
effective running time in combined operation [h] (eq. 35)	*53*	*264*	*607*	*0*	***925***
SH energy produced in combined operation [kWh] (eq. 36)	*451*	*2422*	*5814*	*0*	***8688***
Energy produced in SH-only operation [kWh] (eq. 37)	1654	5851	4223	0	**11728**
DHW energy produced in combined operation [kWh] (eq. 36)	*103*	*496*	*1110*	*0*	***1708***
DHW energy produced in DHW-only operation [kWh] (eq. 37)	44	212	476	1408	**2140**
running time for SH-only operation [h] (eq. 29)	194	638	442	0	**1274**
running time for DHW-only operation [h] (eq. 29)	40	181	387	1032	**1640**
fraction of combined operation [-]	*0.70*	*0.70*	*0.70*	*0.00*	
total running time without considering capacity [h] (eq. 31)	**288**	**1083**	**1437**	**1032**	**3839**
corrected total running time by back-up operation [h] (eq. 40)	288	1083	1437	1032	**3839**
storage losses in combined operation [kWh] (eq. 38	*25*	*120*	*265*	*0*	***410***
storage losses in DHW-only operation [kWh] (eq. 39)	*11*	*51*	*114*	*329*	***505***
DHW energy requirement DHW-combined [kWh] (eq. 38)	*77*	*376*	*845*	*0*	***1298***
DHW energy requirement DHW-only [kWh] (eq. 39)	*33*	*161*	*362*	*1079*	***1635***

NOTE: The rows in italic only occur for simultaneous operating systems

D.1.3.7 Step 7: Auxiliary energy consumption

Auxiliary energy consumption can be calculated according to eq. 43

$$W_g = \sum \phi_{aux,g} \cdot t_{ON,aux}$$

eq. D 44

The running time of the source pump is determined by the running time of the heat pump and energy consumption is calculated by eq. 43

$$W_{g,so} = \phi_{so} \cdot t_{ON,hp,t,i} = 120\text{W} \cdot 288\text{h} = 35\text{kWh}$$

eq. D 45

$$W_{g,sb} = \phi_{sb} \cdot (t_i - t_{ON,hp,t,i}) = 10\text{W} \cdot (330\text{h} - 288\text{h}) = 0.4\text{kWh}$$

eq. D 46

Primary pump in configurations with storage systems is linked to the generator operation, as well.

Stand-by operation is only accounted in time, when the generator is not running, i.e. bin time diminished by the time of generator operation.

Since values according to EN 255-3 are used, the storage loading pump for the DHW must not be considered, since it is included in the $COP_{t,DHW.}$ However, heat losses of the storage loading pump shall be considered (see Table D 16).

Table D 15 gives a summary of the calculation of auxiliary energies.

Table D 15: Summary of results step 7: Calculation of the auxiliary energy consumption

Step 7 Auxiliary energy input					
	Bin 1	Bin 2	Bin 3	Bin 4	Sum
	[-11..-2]	[-2..4]	[4..15]	[15..35]	
auxiliary source pump [kW] (eq. 43)	35	130	172	124	**461**
auxiliary primary pump [kWh] (eq. 43)	16	58	78	56	**207**
auxiliary stand-by [kWh] (eq. 43)	0.4	5	22	22	**49**
auxiliary DHW storage pump [kWh](incl.in COPt,DHW EN 255-3)	0	0	0	0	**0**
total auxiliary energy input [kWh] (eq. 43)	**51**	**193**	**272**	**202**	**717**

NOTE: For the power values given, it is assumed that these are the power values not taken into account by standard testing. For instance, for pumps the fraction to overcome the internal pressure drop in the evaporator is already taken into account. For electrically-driven heat pumps which are tested according to EN 14511 this can be taken into account by the equation

$$\phi_{int} = \frac{\Delta p \cdot \dot{V}}{\eta_p} = \frac{150\text{mbar} \cdot 2\text{m}^3/\text{h}}{0.3} = \frac{15000\text{N}/\text{m}^2 \cdot 2\text{m}^3/\text{h}}{3600\text{s}/\text{h} \cdot 0.3} = 27.8\text{W}$$

At the time of writing, pump efficiency in EN 14511 was fixed to 0.3.

The nominal power of the source pump would be 120 W + 28 W = 148 W

D.1.3.8 Step 8: Generation subsystem heat losses and recoverable generation subsystem heat losses

For mechanical auxiliary components like pumps and fans, where heat losses partly transferred to the heat transfer medium (considered entirely recovered), the losses to the ambiance are described by a default values of $p_{aux,g}$ = 0.2. For auxiliary components that produce heat (electrical device like controls or transformers, supplementary auxiliary heating devices), it is assumed, that the totally auxiliary energy input to these components is lost to the ambiance, e.g. the default values for $p_{aux,g}$ = 1.
Heat losses from auxiliary component k to the ambiance, e.g. the primary pump are calculated acc. to eq. 45

$$W_{g,l,k,i} = W_{g,k} \cdot p_{aux,g} = 16\text{kWh} \cdot 0.2 = 3\text{kWh}$$

eq. D 47

The total heat losses of the generation subsystem are calculated according to eq. 47

$$Q_{l,g,t,i} = \sum_k Q_{l,g,k,i} + W_{g,l,i} = (44\text{kWh} + 40\text{kWh}) + 12\text{kWh} = 96\text{kWh}$$

eq. D 48

It is assumed, that the generator (the heat pump) the storages (heating buffer storage and DHW-storage) and the auxiliary components are installed outside the heated space. Therefore, the temperature reduction factor is b_g = 1 for all components.

Heat losses from auxiliaries to the ambiance are considered recoverable and are calculated according to eq. 46

$$W_{g,rl,i} = W_g \cdot p_{aux,g} \cdot (1 - b_g) = 16\text{kWh} \cdot 0.2 \cdot (1-1) = 0\text{kWh}$$

eq. D 49

Heat losses through the generator envelope are considered recoverable and are calculated according to eq. 48

$$Q_{l,g,DHW,rl,i} = (Q_{l,s,h,i} + Q_{l,s.DHW,i}) \cdot (1 - b_g) = (44\text{kWh} + 40\text{kWh}) \cdot (1-1) = 0\text{kWh}$$

eq. D 50

Table D 16 gives a summary of the calculation of the total losses and the recoverable losses.

Table D 16: Summary of results step 8: Calculation total/recoverable losses generation subsystem

Step 8 Calculation of total/recoverable losses					
	Bin 1	Bin 2	Bin 3	Bin 4	Sum
	[-11..2]	[-2..4]	[4..15]	[15..35]	
Heat losses from source pump [kWh] (eq. 45)	7	26	34	25	**92**
Heat losses from primary pump [kWh] (eq. 45)	3	12	16	11	**41**
Heat losses from storage loading pump [kWh] (eq. 45)	2	7	9	7	**25**
Heat losses in stand-by operation [kWh] (eq. 45)	0.4	5.2	21.7	21.9	**49**
Total heat losses auxiliaries to ambiance [kWh] (eq. 45)	**12**	**50**	**81**	**65**	**208**
Total heat losses generation subsystem [kWh] (eq. 47)	**97**	**382**	**716**	**435**	**1630**
Recoverable from auxiliaries [kWh] (eq. 46)	0	0	0	0	**0**
Recoverable from envelope heat losses [kWh] (eq. 48)	0	0	0	0	**0**
Total recoverable losses [kWh] (eq. 49)	**0**	**0**	**0**	**0**	**0**

Step 9: Calculation of the energy input to the generation subsystem

5.2.13.1.1 Electricity input to the heat pump for space heating operation

The electricity input to the heat pump for space heating operation can be calculated by summing-up the electricity input of the respective bins according to eq. 50

$$E_{in,g,h} = \frac{Q_{hp,h,sin,i}}{COP_{sin,i}} + \frac{Q_{hp,h,combi,i}}{COP_{combi,i}} = \frac{1654kWh}{2.65} + \frac{451kWh}{2.85} = 624kWh + 158kWh = 782kWh$$

eq. D 51

5.2.13.1.2 Electricity input to the heat pump for domestic hot water operation

The electricity input to the heat pump for domestic hot water operation can be calculated according to eq. 51

$$E_{in,g,DHW} = \frac{Q_{out,hp,DHW,sin,i}}{COP_{t,DHW,sin,i}} + P_{es,sin,i} \cdot t_{DHW,sin,i} + \frac{Q_{out,hp,DHW,combi,i}}{COP_{t,DHW,combi,i}} + P_{es,combi,i} \cdot t_{DHW,combi,i}$$

eq. D 52

$$= \frac{33kWh}{2.4} + 0.062kW \cdot 330h \cdot (1-0.7) + \frac{77kWh}{2.85} + 0.036kW \cdot 0.7 \cdot 330h = 14kWh + 6kWh + 27kWh + 8kWh = 55kWh$$

Table D 17 gives a summary of the energy input to the heat pump to cover the heat requirement.

Table D 17: Summary of results step 9a: Calculation of the energy input to the heat pump

Step 9 Calculation of energy input to the heat pump					
	Bin 1	Bin 2	Bin 3	Bin 4	Sum
	[-11..-2]	[-2..4]	[4..15]	[15..35]	
electricity input to heat pump for SH-only [kWh] (eq. 50)	623	1468	876		**2967**
electricity input to heat pump for SH combined [kWh] (eq. 50)	*158*	*632*	*1307*		***2097***
electricity input to heat pump for DHW-only [kWh] (eq. 51)	14	66	147	423	**650**
electricity input to heat pump for DHW combined [kWh] (eq. 51)	*27*	*98*	*190*		***315***
electricity input to cover storage losses single [kWh] (eq. 51)	6	30	67	200	**303**
electricity input to cover storage losses combined [kWh] (eq. 51)	*8*	*40*	*91*	*0*	***140***
total electricity input to heat pump [kWh]	**837**	**2334**	**2678**	**623**	**6471**

5.2.13.1.3 Back-up heater

Energy input to the back-up heater is calculated according to eq. 53

$$E_{in,bu,i} = \frac{Q_{bu,h,cap,i}}{\eta_{bu,h}} + \frac{p_{bu,DHW,i} \cdot (Q_{out,g,DHW,i} + Q_{l,g,DHW,i})}{\eta_{bu,DHW}} = \frac{0.06 \cdot (2196 + 44)kWh}{0.95} + \frac{0.11 \cdot (124 + 40)kWh}{0.95}$$

$$= 141 \text{ kWh} + 19 \text{ kWh} = 160 \text{ kWh}$$

eq. D 53

Table D 18 gives a summary of the energy input to the back-up heater.

Table D 18: Summary of results step 9b: Calculation of the energy input to the back-up

Step 9 Calculation of the back-up heater					
	Bin 1	Bin 2	Bin 3	Bin 4	Sum
	[-11..-2]	[-2..4]	[4..15]	[15..35]	
back-up electricity space heating [kWh] (eq. 53)	142	0	0	0	**142**
back-up electricity DHW [kWh] (eq. 53)	19	93	209	185	**506**
Total electricity input back-up heater [kWh]	**161**	**93**	**209**	**185**	**648**
Total electricity input [kWh] (eq. 54)	**998**	**2428**	**2886**	**808**	**7120**

D.1.3.9 Step 10: Output values

D.1.3.9.1 Required output values

The required output values of the calculation are

- Energy input to cover the heat requirement $E_{in,g}$ (or $G_{in,g}$) according to eq. 54
  - $E_{in,g} = E_{in,hp} + E_{in,bu} = 6471$ kWh + 648 kWh = 7119 kWh
- Total losses of the generation subsystem $Q_{l,g}$ according to eq. 47
  - $Q_{l,g} = Q_{l,g,h} + Q_{l,g,DHW} + W_{g,l} = 392$ kWh + 1029 kWh + 208 kWh = 1629 kWh
- Recoverable losses of the generation subsystem $Q_{l,g,rl}$ according to eq. 49
  - $Q_{l,g,rl} = Q_{l,g,rl} + Q_{l,g,rl,aux} = 0$ kWh + 0 kWh = 0 kWh
- Total required auxiliary energy to operate the generation subsystem W_g according to eq. 43
  - $W_g = 717$ kWh

D.1.3.9.2 Optional output values

- Total used ambient heat $Q_{in,g}$ according to eq. 1
  - $Q_{in,g} = Q_{out,g} + Q_{l,g} - E_{in,g} - k_{g,rd} \cdot W_g = 20158\text{kWh} + 3300\text{kWh} + 1421\text{kWh} - 7119\text{kWh} = 17760\text{kWh}$
- Total heat energy delivered by the back-up heater Q_{bu} acc. to eq. 19 and eq. 42
  - $Q_{bu} = Q_{bu,DHWop} + Q_{bu,h,cap} = 0.11*(3300+1029)$ kWh + 132 kWh = 608 kWh
- Total heat energy produced by the heat pump system Q_{hp} acc. to eq. 30
  - $Q_{hp} = Q_{out,g,h} + Q_{l,g,h} + Q_{out,g,DHW} + Q_{l,g,DHW} - Q_{bu} = 20158$ kWh+3300 kWh+1421 kWh – 608 kWh

    = 24271 kWh
- Overall seasonal performance $SPF_{g,t}$ according to eq. 56
  - $SPF_{g,t} = (Q_{out,g,h}+Q_{out,g,DHW})/(E_{in,g}+W_g)$ = (20158 kWh+3300 kWh)/(7119 kWh+717 kWh) = 3.0
- Overall seasonal performance factor of the heat pump according to eq. 57
  - $SPF_{hp,t} = (Q_{hp})/(E_{in,hp}+W_{so}+W_{sb})$ = (24271 kWh)/(6471 kWh+416 kWh+49 kWh) = 3.5

Table D 19 gives a summary of required and optional output values of the calculation of the generation subsystem.

Table D 19: Summary of results step 10: output values of the generation subsystem

Step 10 Output values of the generation subsystem					
	Bin 1	Bin 2	Bin 3	Bin 4	Sum
	[-11.-2]	[-2..4]	[4..15]	[15..35]	
Energy input to cover the heat requirement $E_{in,g}$ [kWh] (eq. 54)	898	2324	2862	942	**7025**
Total losses of the generation $Q_{l,g}$ [kWh] (eq. 47)	97	382	716	435	**1630**
Total recoverable losses of generation$Q_{l,g,rl}$ [kWh] (eq. 49)	0	0	0	0	**0**
Total auxiliary energy input of generation Wg [kWh] (eq. 43)	50	194	272	201	**717**
optional output values					
Total heat delivered by the heat pump system Q_{hp}[kWh] (eq. 30)					**24271**
Total energy delivered by the back-up heater Q_{bu} [kWh] (eq. 42)					**608**
Total amount of used ambient heat $Q_{in,g}$ [kWh] (eq. 1)					**17760**
Overall SPF of the heat pump $SPF_{hp,t}$ [-] (eq. 57)					**3.5**
Overall SPF of the generation subsystem $SPF_{g,t}$ [-] (eq. 56)					**3.0**

Annex E
(informative)

Example for tabulated values of the system typology method as national Annex for the Netherlands

The following tables are an example of a possible national Annex containing output tables provided by Dutch standardisation Institut NEN for the Netherlands[4].

Seasonal efficiency and auxiliary energy consumption for heat pump installations in residential and non-residential buildings in the Netherlands

E.1 Scope

This procedure gives the calculation procedure to estimate the seasonal performance factor (SPF), gross seasonal efficiency, primary energy consumption and auxiliary energy consumption of heating installations with one or more heat pumps.

The procedure is developed for residential and non-residential buildings in The Netherlands.

E.2 References

All references to test data for electrically-driven heat pumps are to the European heat pump test standard EN 14511.

E.3 Heat pump seasonal performance

E.3.1 Residential buildings

For all heat pumps in a residential building SPF and seasonal efficiency are determined using the table below.

Table E 1 — Gross seasonal heat pump efficiency in residential buildings for single heat pump with index i.

Heat pump type / test results	**Gross seasonal heat pump efficiency ($\eta_{hp;i}$)**	
System design flow temperature $\theta_{flow;design}$	$\theta_{flow;design} \leq 35$ °C	$35 < \theta_{flow;design} \leq 45$ (°C)
First performance level for individual or collective electric heat pump, without any performance requirements, with heat source:		
— Soil	$3.8 \times \eta_{el}$ [a]	$3.4 \times \eta_{el}$ [a]
— Groundwater	$4.5 \times \eta_{el}$ [a]	$4.1 \times \eta_{el}$ [a]
— Outside air	$3.7 \times \eta_{el}$ [a]	$3.3 \times \eta_{el}$ [a]

Table E 2 — Gross seasonal heat pump efficiency in residential buildings for single heat pump with index i. (continued)

Heat pump type / test results	Gross seasonal heat pump efficiency ($\eta_{hp;i}$)	
System design flow temperature $\theta_{flow;design}$	$\theta_{flow;design} \leq 35$ °C	$35 < \theta_{flow;design} \leq 45$ (°C)
Second performance level for individual or collective electric heat pump, fulfilling performance requirements of Table E 3, with heat source:		
— Soil	$4.4 \times \eta_{el}$ [a]	$4.1 \times \eta_{el}$ [a]
— Ground water	$5.0 \times \eta_{el}$ [a]	$4.6 \times \eta_{el}$ [a]
— Outside air	$3.8 \times \eta_{el}$ [a]	$3.5 \times \eta_{el}$ [a]
Where:		
η_{el} efficiency of electricity generation;		
$\theta_{flow;design}$ Design flow temperature °C.		
[a] The result of this multiplication should be rounded down to a multiple of 0.025. The energy consumption of a source pump or fan is included in these figures.		

Table E 3 — Minimal required COP-values for second performance level, determined according to NEN- EN14511 for test conditions according to these standards.

Heat source	Test conditions according to EN 14511	Minimal COP according to EN 14511
Soil / water (brine / water)	(B0/W45) (B0/W35)	3.4 4.0
Ground water / water (water / water)	(W10/W45) (W10/W35)	4.2 5.1
Outside air / water (outside air / water)	(A7(6)/W45) (A7(6)/W35) (A-7(-8)/W45)	2.9 3.0 2.0
Where:		
A — Air as heat source, with its temperature level during test;		
(y) — Air as heat source, with its wet bulb temperature level during test;		
B — Soil as heat source, with its temperature level during test;		
W — Ground water as heat source, with its temperature level during test;		
B0/W35 — Brine as heat source, with brine inlet temperature at 0°C and a water distribution system with a flow temperature of 35°C. The temperature spread of the heat transfer medium over the condenser is 5 K according to EN 14511		

E.3.2 Non-residential buildings

For all heat pumps in non-residential buildings seasonal COP and efficiency is determined using the table below.

Table E 4 — Gross seasonal heat pump efficiency in non-residential buildings for single heat pump with index i.

System design flow temperature $\theta_{flow;design}$	Gross seasonal heat pump efficiency ($\eta_{hp;i}$)					
	$\theta_{flow;design}$<35°C		35°C≦$\theta_{flow;design}$<45°C		45°C≦$\theta_{flow;design}$<55°C	
Heat source	**EHP** [a]	**GMHP**	**EHP** [a]	**GMHP**	**EHP** [a]	**GMHP**
Soil / outside air	3.4 x η_{el}	1.6	3.1 x η_{el}	1.5	2.8 x η_{el}	1.4
Exhaust ventilation air	6.1 x η_{el}	2.6	5.1 x η_{el}	2.2	4.4 x η_{el}	2.0
Groundwater/aquifer	4.7 x η_{el}	2.1	4.2 x η_{el}	1.9	3.6 x η_{el}	1.8
Surface water	4.1 x η_{el}	1.9	3.7 x η_{el}	1.8	3.3 x η_{el}	1.7
Where:						
η_{el} efficiency of electricity generation;						
$\theta_{flow;design}$ system design flow temperature °C;						
EHP electric heat pump;						
GMHP gas engine driven heat pump.						
[a] The result of this multiplication should be round down to a multiple of 0.025. The energy consumption of a source pump or fan is included in these figures.						

E.4 Heat pump installation efficiency

For single heat pump installations and for heat pump installations with two or more heat pumps with equal efficiency, installation efficiency is equal to heat pump efficiency (heat pump index i = 1):

$$\eta_g = \eta_{hp,1}$$

For heat pump installations with two or more heat pumps or other heat generators with different efficiencies, the individual efficiencies are weighted.

First determine the ratio of the nominal capacity of the preferential heat pump to the nominal capacity of all heat generators:

$$ß_{heat} = \frac{P_{hg;pref}}{\Sigma P_{hg;i}}$$

where:

pref Index of the preferential heat pump;

$ß_{heat}$ is the ratio of the nominal capacity of the preferential heat pump to the nominal capacity of all heat generators;

$P_{hg;pref}$ is the total nominal capacity of the preferential heat pump (heat generator), with index i = pref, in kW;

$P_{hg;i}$ is the total nominal capacity of a heat generator with index i, in kW.

Then take the share of the preferential heat pump in the heat supply, f_{pref}, from Table E 5 for residential buildings or Table E 6 for non-residential buildings.

Table E 5 — Share of the total heat demand, generated with the preferential heat generator, f_{pref}, as function of the capacity ratio $ß_{heat}$ for residential buildings

$ß_{heat}$	f_{pref}		
Preferent heat generator	**Boiler or other heater**	**heat pump**	**co-generation**
From 0 to 0.1	0	0	0.15
From 0.1 to 0.2	0	0.48	0.45
From 0.2 to 0.3	0.5	0.79	0.60
From 0.3 to 0.4	0.8	0.93	0.60
From 0.4 to 0.6	1.0	0.97	0.60
From 0.6 to 0.8	1.0	0.98	0.60
Equal to or larger than 0.8	1.0	1.00	0.60

For intermediate values of $ß_{heat}$ the adjacent lower value has to be taken.

Table E 6 — Share of the total heat demand, generated with the preferential heat generator, fpref, as function of the capacity ratio ßheat for non-residential buildings

$ß_{verw}$	f_{pref}	
Preferent heat generator	**Heat pump, boiler, other heat generator**	**Co-generation**
From 0 to 0.05	0	0
From 0.05 to 0.1	0.25	0.25
From 0.1 to 0.2	0.48	0.48
From 0.2 to 0.3	0.79	0.6
From 0.3 to 0.4	0.93	0.6
From 0.4 to 0.6	0.97	0.6
From 0.6 to 0.8	0.98	0.6
Equal to or larger than 0.8	1.0	0.6

For intermediate values of $ß_{heat}$ the adjacent lower value has to be taken.

At last the heating installation efficiency is determined:

$$\eta_{g} = \frac{1}{\frac{(1 - f_{pref})}{\eta_{hg,npref}} + \frac{f_{pref}}{\eta_{hg;pref}}}$$

Where:

η_g is the generation efficiency of the heating installation;

f_{pref} is the year averaged fraction of the total heat supply, that is supplied by the preferential operated heat pump (heat generator);

$\eta_{hg;pref}$ is the seasonal efficiency of the preferential operated heat pump (heat generator);

$\eta_{hg;npref}$ is the seasonal efficiency of the remaining heat generators; in case of unequal efficiencies the average efficiency is determined, weighted with nominal loads.

$$\eta_{hg;npref} = \frac{\sum_{all\ i<>pref} \Sigma (\eta_{hg;i} * P_{hg;i})}{\sum_{all\ i<>pref} \Sigma P_{hg;i}}$$

E.5 Heat pump installation energy consumption

Primary energy consumption of the heat pump (heat generator) installation is given by:

$$Q_{in;g;prim} = \eta_g \times Q_{in;d}$$

Where:

$Q_{in;g;prim}$ Annual primary energy consumption of the heat generator installation.

$Q_{in;d}$ Annual heat demand of the building / distribution installation, to be fulfilled by the heat generator installation.

Recoverable losses are zero.

E.6 Heat pump installation auxiliary energy consumption

For heat pump installations in residential buildings additional primary energy consumption due to pumps and controls is given by the table below.

Table E 7 — Primary energy consumption due to pumps and controls for residential buildings

Heater type	**Conditions**	**$Q_{in;g;aux;el}$ [kWh]**
Heat generator pump	No pump control	2.2 * A
	Pump control	1.1 * A
System pump in collective heating installation	-	1.1 * A
Heat pump controls	-	0.88 * A

Where:

A Heated area of the building, in m^2

For heating installations in non-residential buildings additional primary energy consumption due to pumps, fans and controls is given by.

$$Q_{ing;;aux;e} = 8 * f * A / \eta_{el} \text{ [MJ]}$$

Factor f is found from the table below.

Table E 8 — Factor f for non-residential buildings

Conditions	**f**
No pumps in water circuits	0
If for more than 50% of the pump power, pumps are automatically off with the related heat generator or are frequency controlled.	0.5
All other situations	1.0

Converting to primary energy renders:

$$Q_{in;g;aux;prim} = Q_{in;g;aux;el} * 3.6 / \eta_{el} \text{ [MJ]}$$

Where

η_{el} Electricity generation gross efficiency; in [-]

Recoverable auxiliary energy losses are zero.

Annex F
(informative)

Example values for parameters to accomplish the case specific heat pump calculation method (bin method)

This annex contains examples for values needed to accomplish the case specific calculation method (bin method) to illustrate what kind of data is required. For the calculation values from standard testing are to be used.

F.1 Temperatures

F.1.1 Source temperatures

F.1.1.1 Outdoor air-to-water heat pumps

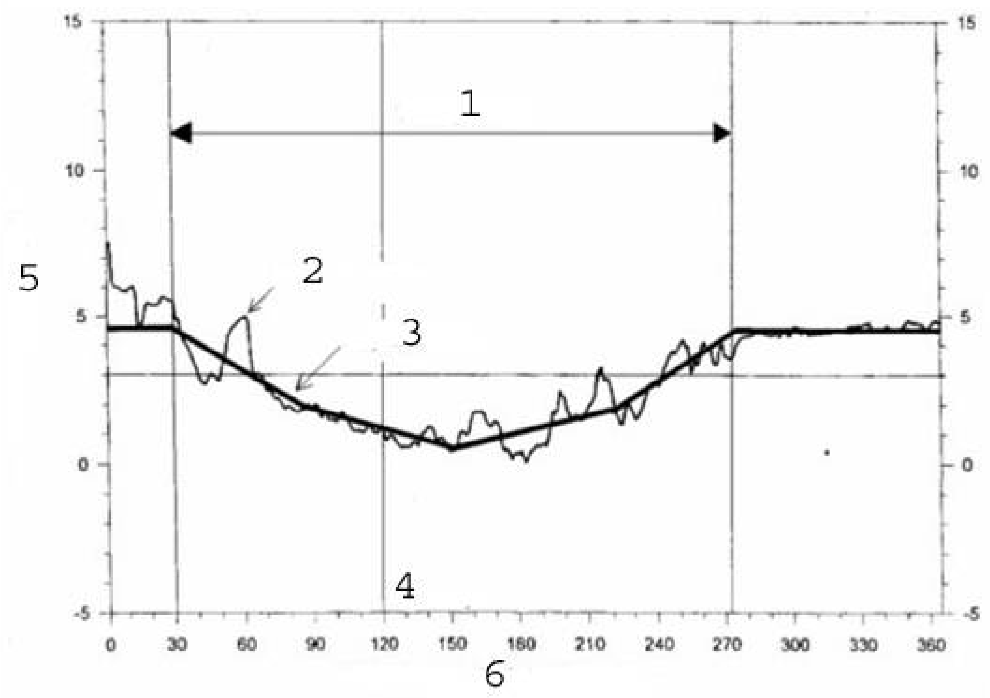
For air-to-water heat pumps, the source temperature corresponds to the outside air temperature

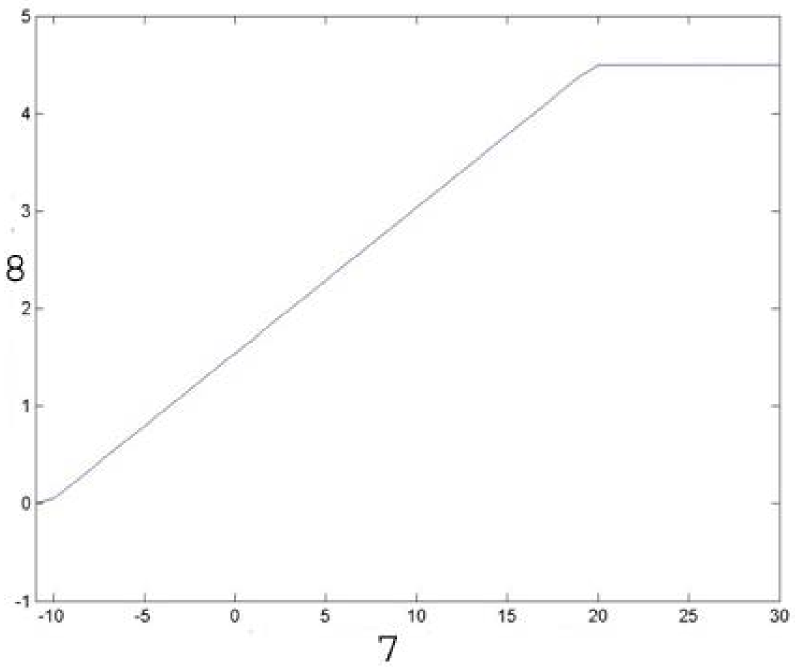
F.1.1.2 Exhaust-air-to-water heat pumps

For exhaust air-to-water heat pumps, the source temperature corresponds to the indoor design temperature

F.1.1.3 Brine-to-water heat pumps

A standard profile for the temperature during the heating period and a fit depending on the outdoor air temperature are given in [6].





Legend

1	Heating season	5	Source temperature [°C]
2	Simulation values	6	Time (days from the first September)
3	Approximation]	7	outdoor air temperature [°C]
4	1. January	8	Brine temperature [°C]

Figure F 1 — Standard profile for the return temperature of the ground heat exchanger of brine-to-water heat pump (left: temperature profile by simulation, right: fit)

Fit of the dependency of the source temperature on the outdoor air temperature

$$\theta_{\text{so,in}} = 0.15 \cdot \theta_{\text{oa}} + 1.5°\text{C} \tag{E 1}$$

where

$\theta_{so,in}$	source temperature at the evaporator inlet	[°C]
θ_{oa}	outdoor air temperature	[°C]

F.1.1.4 Ground water-to-water heat pumps

Ground water temperature is considered constant throughout the year and the default values for the ground water temperature is 10°C.

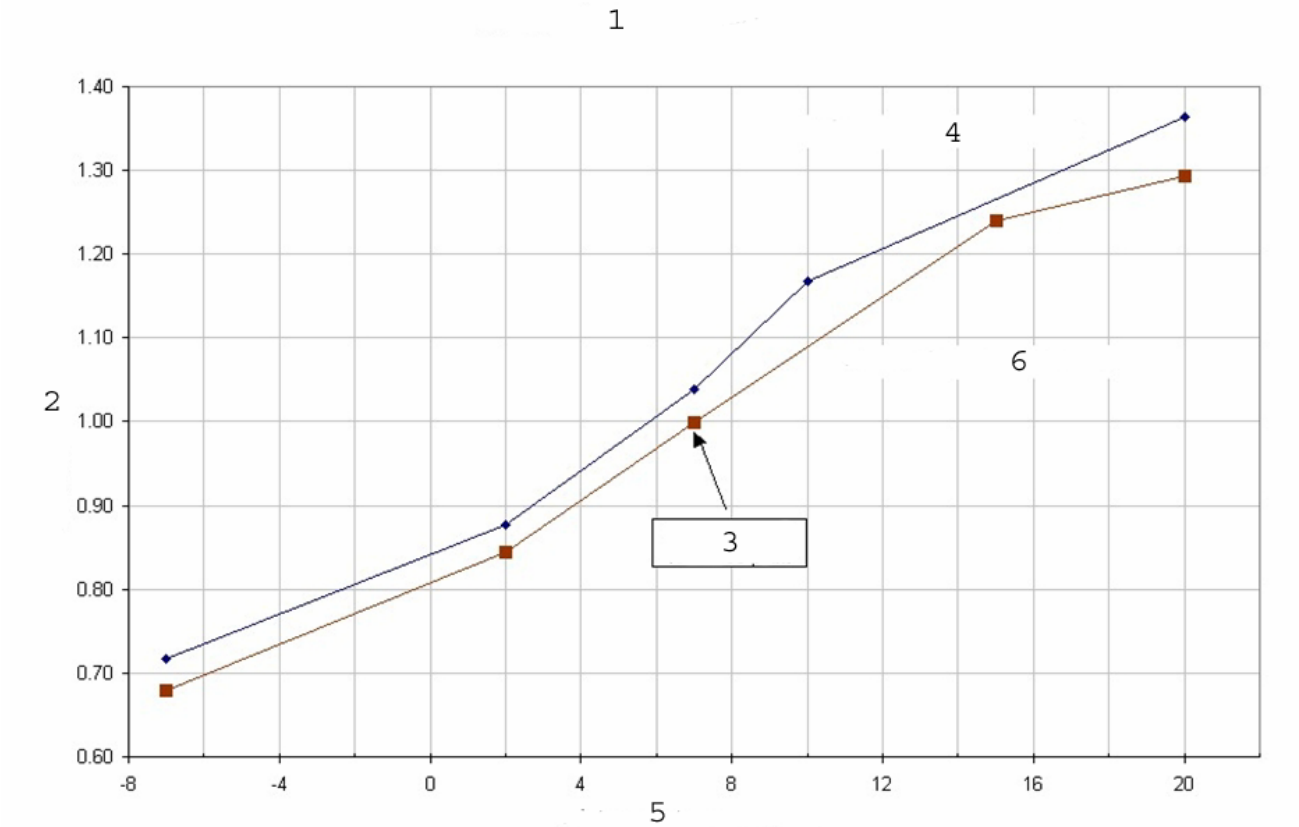
F.2 Example values for heating capacity and coefficient of performance for electrically driven heat pumps

F.2.1 Heating capacity

The following data are based on a statistical analysis of data provided by the Swiss national test centre WPZ (WPZ Bulletin [3]) now located at UAS Buchs. The data are based on testing according to the standard EN 255-2. EN 255-2 was replaced by EN 14511 in 2004 and different test conditions were introduced. However, at the time of writing, only few measurements according to EN 14511 were available and therefore the values according to the EN 255-2 are given as examples.

The relative heating capacity is the ratio of the heating capacity to the reference heating capacity, e.g. at the standard rating point of EN 14511, for air-to-water heat pump A7/W35, for brine-to-water heat pumps B0/W35 and for water-to-water heat pumps W10/W35.

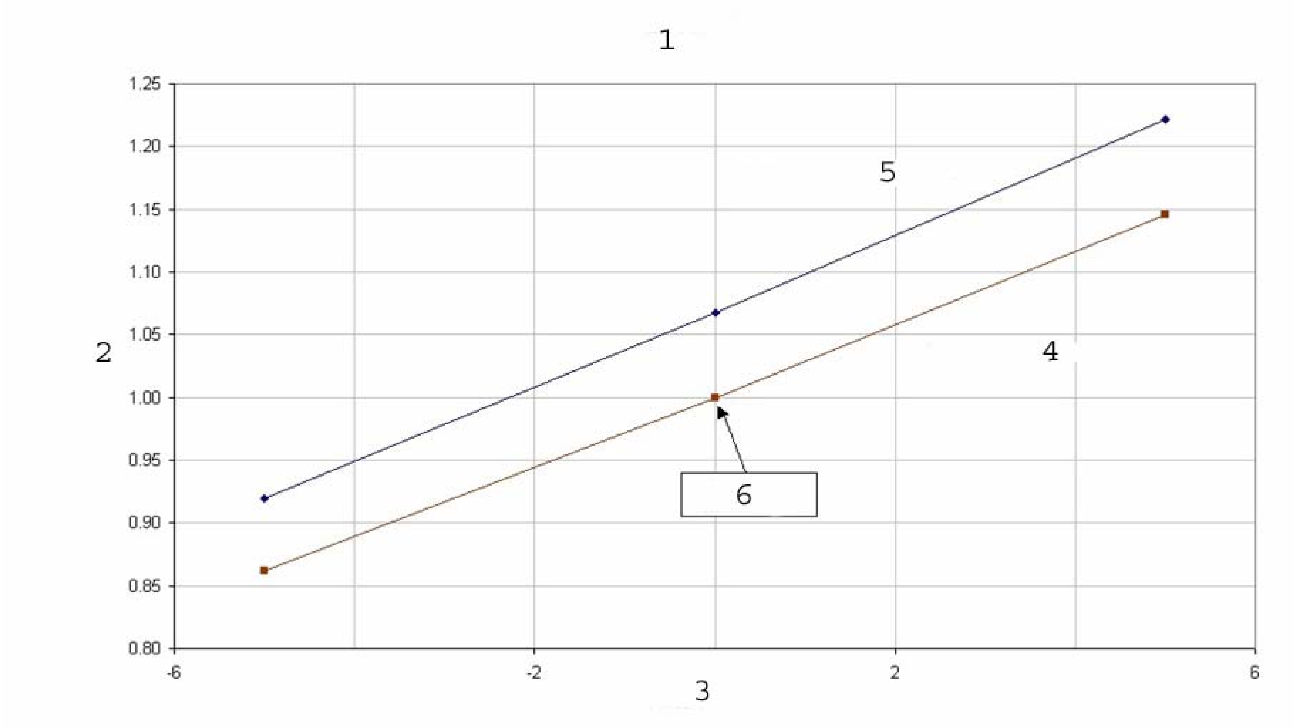
F.2.1.1 Air-to-water heat pumps



Relative heating capacity A/W heat pumps			
Legend	**Inlet source temperature**	**Outlet sink temperature**	
		35°C	**50°C**
1 Air-to-water electrically-driven heat pump			
2 Relative heating capacity	**-7°C**	0,72	0,68
3 Heating capacity reference point (A7/W35)	**2°C**	0,88	0,85
4 Outlet sink temperature: 40°C y = 0.00053x2 + 0.01665x + 0.8286	**7°C**	1,04	1,00
5 Outlet sink temperature: 50°C y = 0.00052x2 + 0.01715x + 0.77631	**10°C**	1,17	-
6 Outlet sink temperature: 55°C y = $0.00051x^2 + 0.01759x + 0.75004$	**15°C**	-	1,24
	20°C	1,36	1,29

Figure F 2 — Average heating capacity of air-to water heat pumps vs source and sink temperatures (reference: T1 according to EN 255-2)

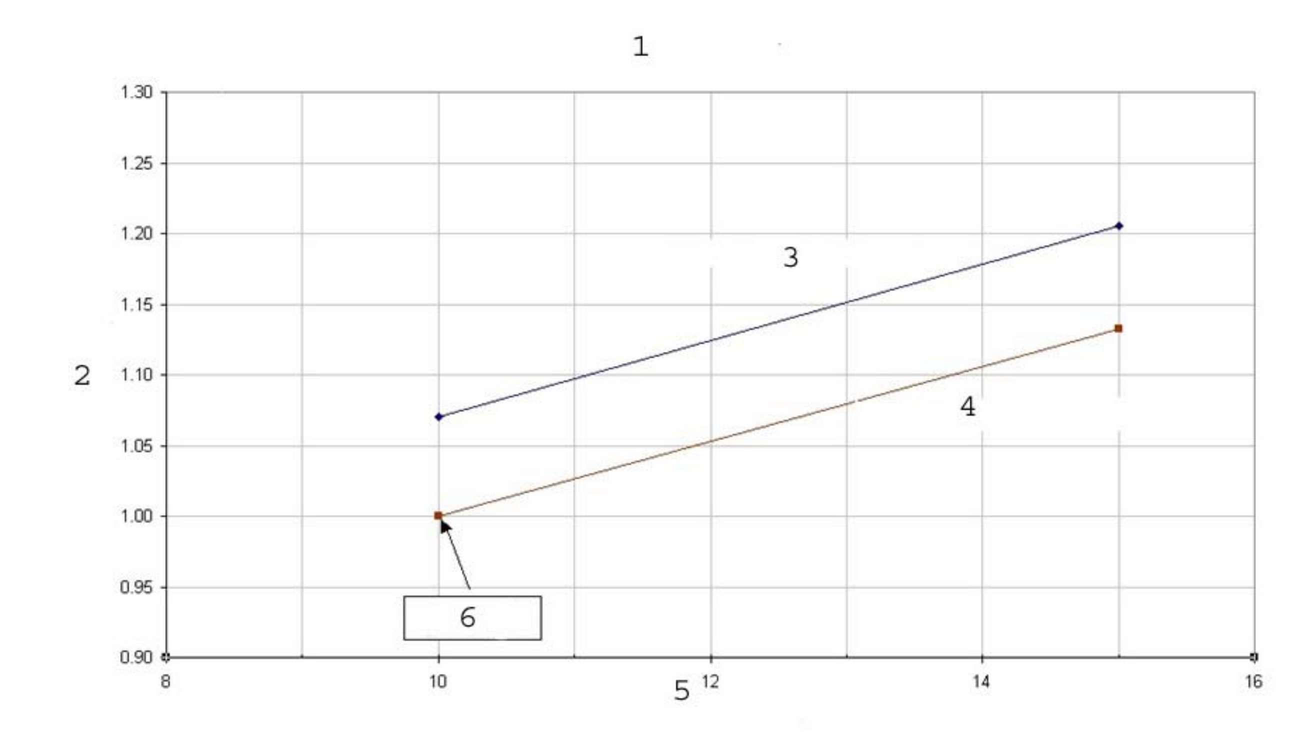
F.2.1.2 Brine-to-water heat pumps



Relative heating capacity B/W heat pumps				
Legend		Inlet source temperature	Outlet sink temperature	
			35°C	50°C
1	Brine-to-water electrically-driven heat pump			
2	Relative heating capacity	**-5°C**	0,92	0,86
3	Inlet source temperature [°C]	**0°C**	1.07	1,00
4	Outlet sink temperature: 50°C	**5°C**	1,22	1,15
5	Outlet sink temperature: 35°C			
6	Heating capacity reference point			

Figure F 3 — Average heating capacity of brine to water electrical heat pumps vs source and sink temperatures (reference point T1 according EN 255-2)

F.2.1.3 Water-to-water heat pumps



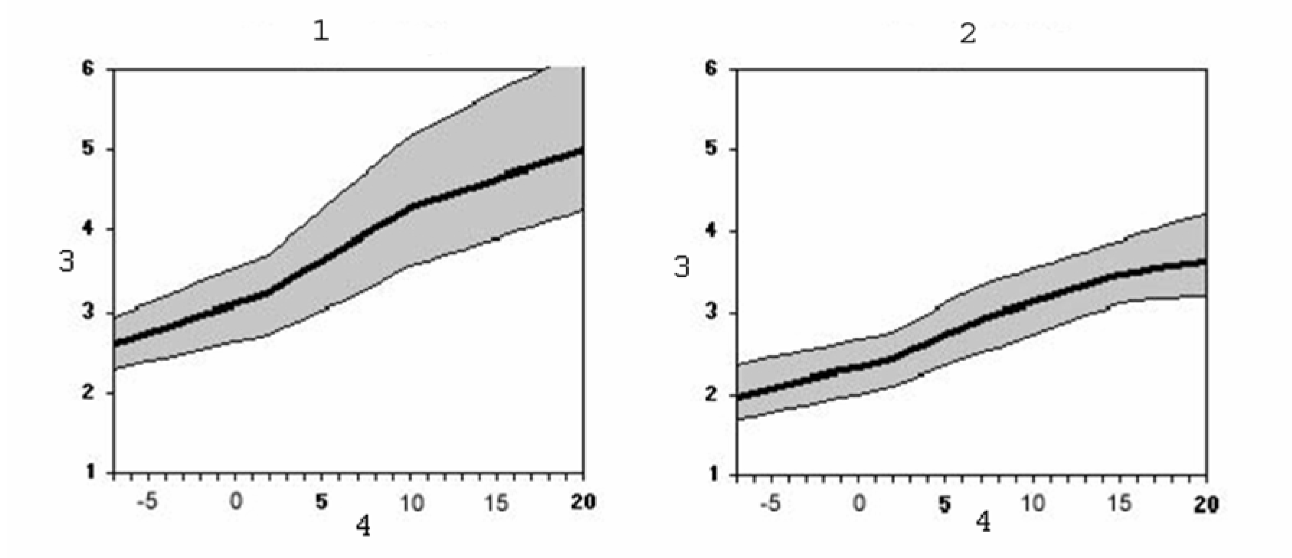
Relative heating capacity W/W heat pumps			
Legend	**Inlet source temperature**	**Outlet sink temperature**	
		35°C	**50°C**
1 Water-to-water electrically-driven heat pumps			
2 Relative heating capacity	**10°C**	1.07	1,00
3 Outlet sink temperature: 35°C	**15°C**	1,22	1,13
4 Outlet sink temperature: 50°C			
5 Inlet source temperature [°C]			
6 Heating capacity reference point			

Figure F 4 — Average heating capacity of water to water electrical heat pumps vs source and sink temperatures (standard rating point according to EN 255-2)

F.2.2 COP

The following data are based on a statistical analysis of data provided by the Swiss national test centre WPZ [3] according to EN 255-2. EN 255-2 was replaced by EN 14511 in 2004 and different test conditions were introduced, which deliver depending on the test point an about 5% lower COP-value. However, at the time of writing, only few measurements according to EN 14511 were available and therefore the values according to the EN 255-2 are given as examples.
Actual test results according to the actual test standards can be downloaded for instance at the Swiss heat pump test centre at the URL http://www.wpz.ch

F.2.2.1 Air-to-water heat pumps



Legend

1 Sink temperature θ_{si} = 35°C

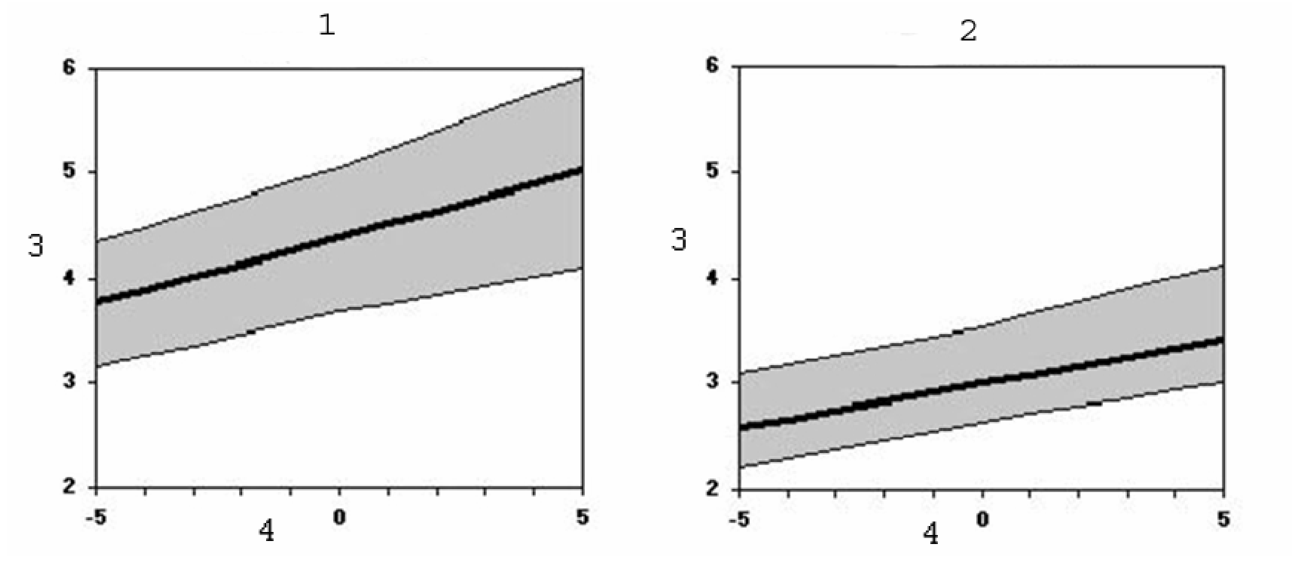
2 Sink temperature θ_{fsi}= 50°C

3 COP

4 Source temperature

Figure F 5 — COP-values of air-to water electrical heat pumps vs source temperatures (black line – average values, grey area – scatter band of the values)

F.2.2.2 Brine-to-water



Legend

1 Sink temperature θ_{si} = 35°C

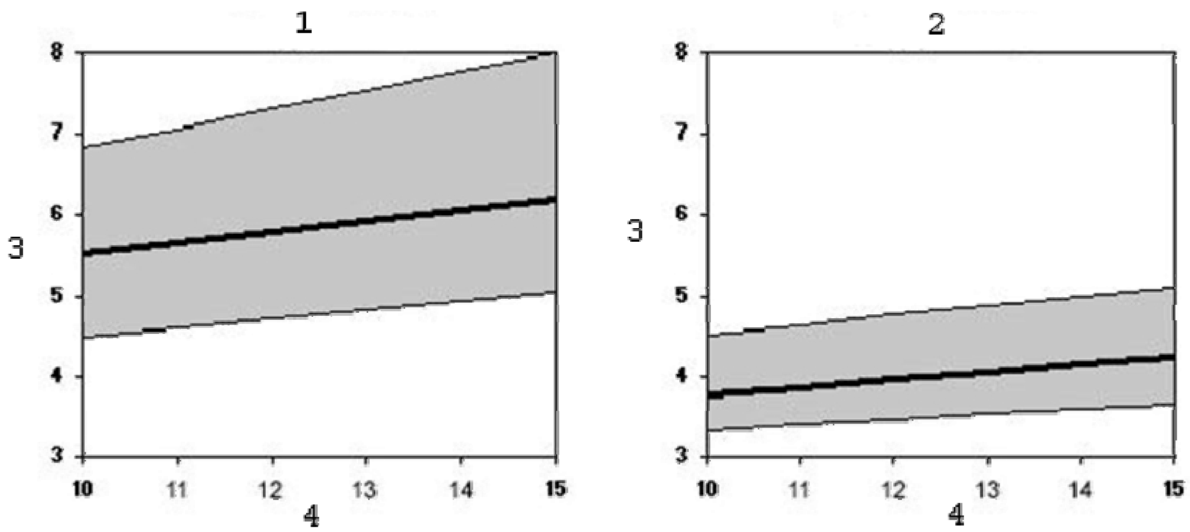
2 Sink temperature θ_{si} = 50°C

3 COP

4 Source temperature

Figure F 6 — COP-values of air-to water electrical heat pumps vs source temperatures (black line – average values, grey area – scatter band of the values)

F.2.2.3 Water-to-water



Legend

1 Sink temperature θ_{si} = 35°C

2 Sink temperature θ_{si} = 50°C

3 COP

4 Source temperature

Figure F 7 — COP-values of air-to water electrical heat pumps vs source temperatures (black line – average values, grey area – scatter band of the values)

F.3 Gas engine-driven heat pumps

F.3.1 Preface

Characteristics provided in this chapter are based on very few measurements (gas driven and absorption heat pumps).

They are only given as example values.

When carrying out the calculation care shall be taken to use as input data features provided by manufacturers under their responsibility.

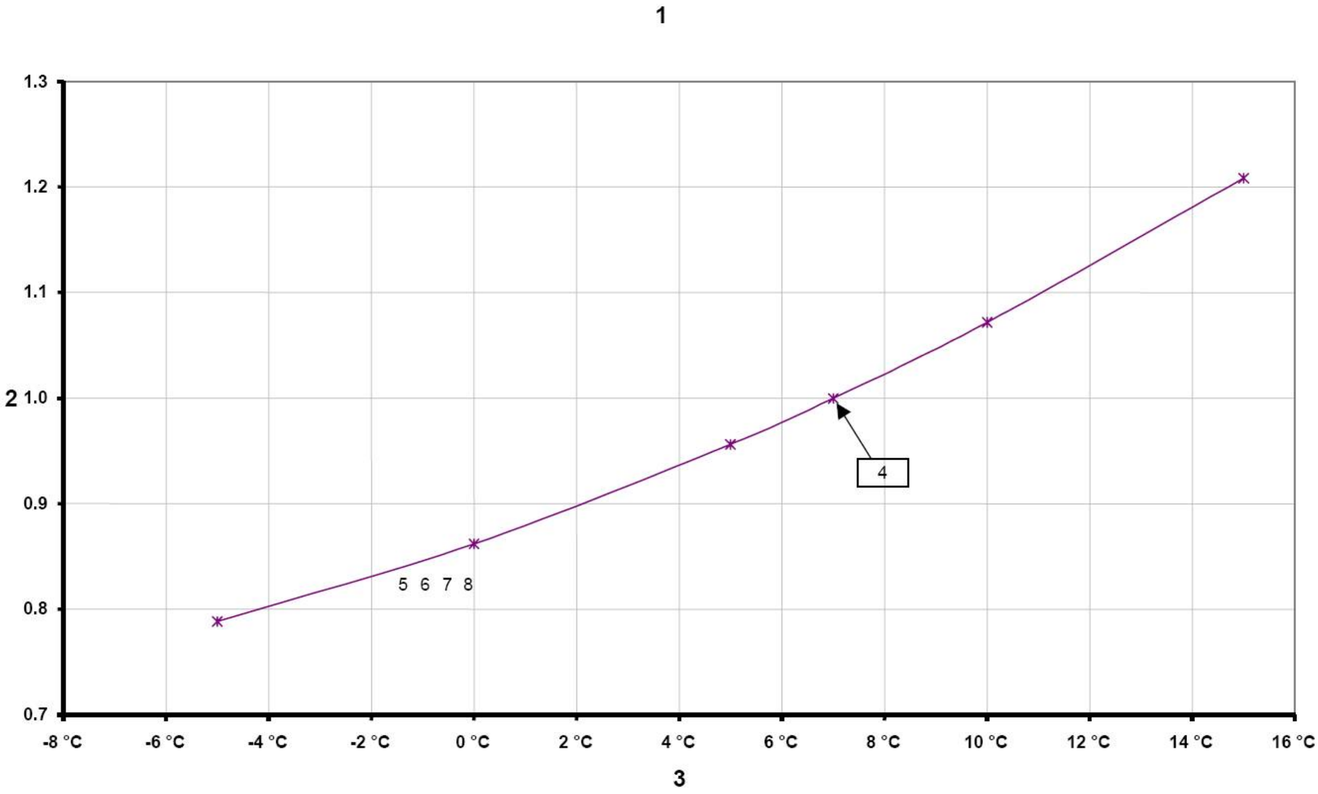
The following data are based on the average values of technical features provided by different manufacturers.

Note (*) Attention shall be paid that the COP values indicated in this annex are referring to the used energy (i.e. the energy delivered at the interface of the building).

F.3.2 Heating capacity

F.3.2.1 Air-to-water heat pumps

The data refer to the overall heating capacity (heat pump condenser and heat recovery from the engine).

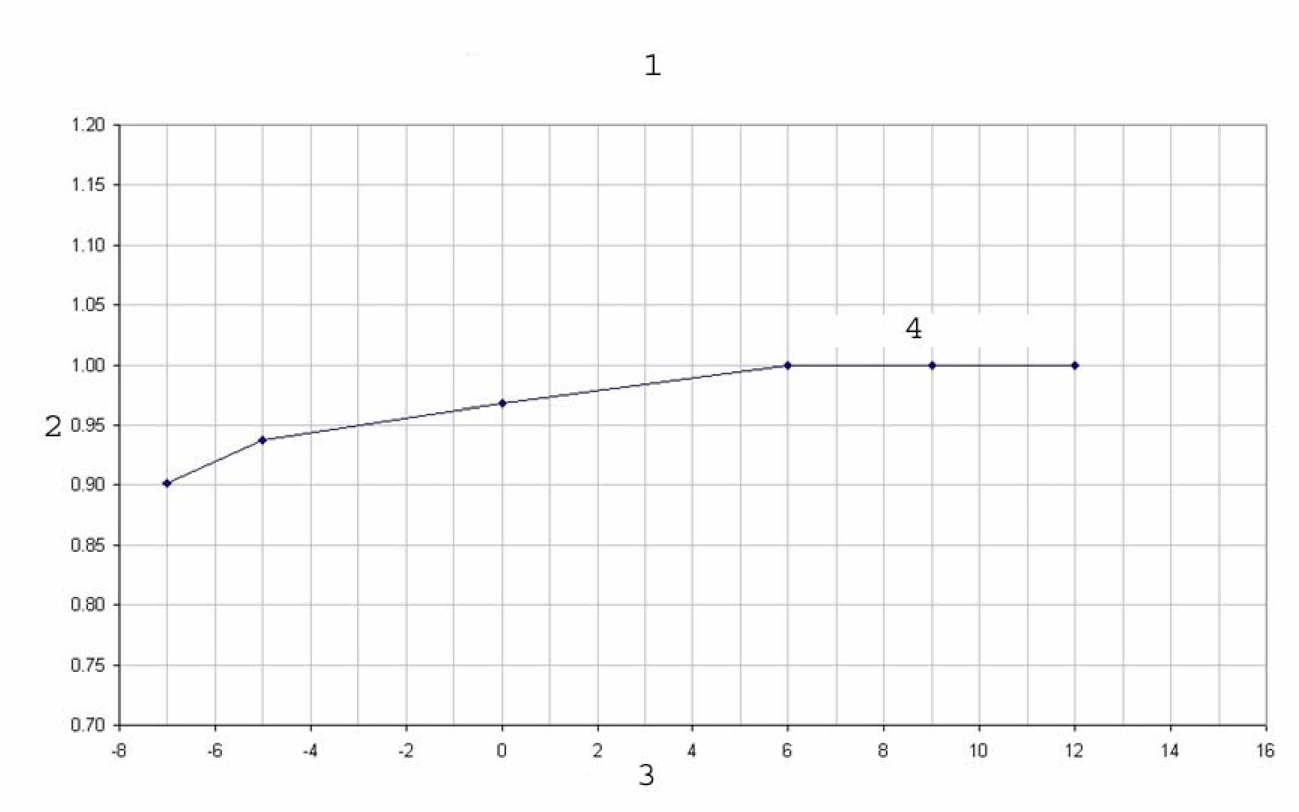


Relative heating capacity					
Legend	**Inlet source temperature**	**Outlet sink temperature**			
1 Air-to-water gas engine driven heat pump		**40°C**	**45°C**	**50°C**	**55°C**
2 Relative heating capacity	**-5 °C**	0,80	0,79	0.78	0.77
3 Inlet source temperature	**0 °C**	0,87	0,87	0.86	0.85
4 Heating capacity reference point: (7°C/45°C)	**5 °C**	0,96	0,96	0.96	0.95
5 Outlet sink temperature: 40°C	**7°C**	1,00	1,00	1,00	1,00
6 Outlet sink temperature: 45°C	**10 °C**	1,07	1,07	1,07	1,08
7 Outlet sink temperature: 50°C	**15 °C**	1.20	1,20	1,21	1,22
8 Outlet sink temperature: 55°C					
Polynom for all outlet sink temp. $y = 0.00042x^2 + 0.01679x + 0.8618$					

Figure F 8 — Average oveall heating capacity of air-to water gas engine driven heat pumps vs. source and sink temperatures

F.3.2.2 Air-to-air heat pumps

The data reflect the lower end of the market



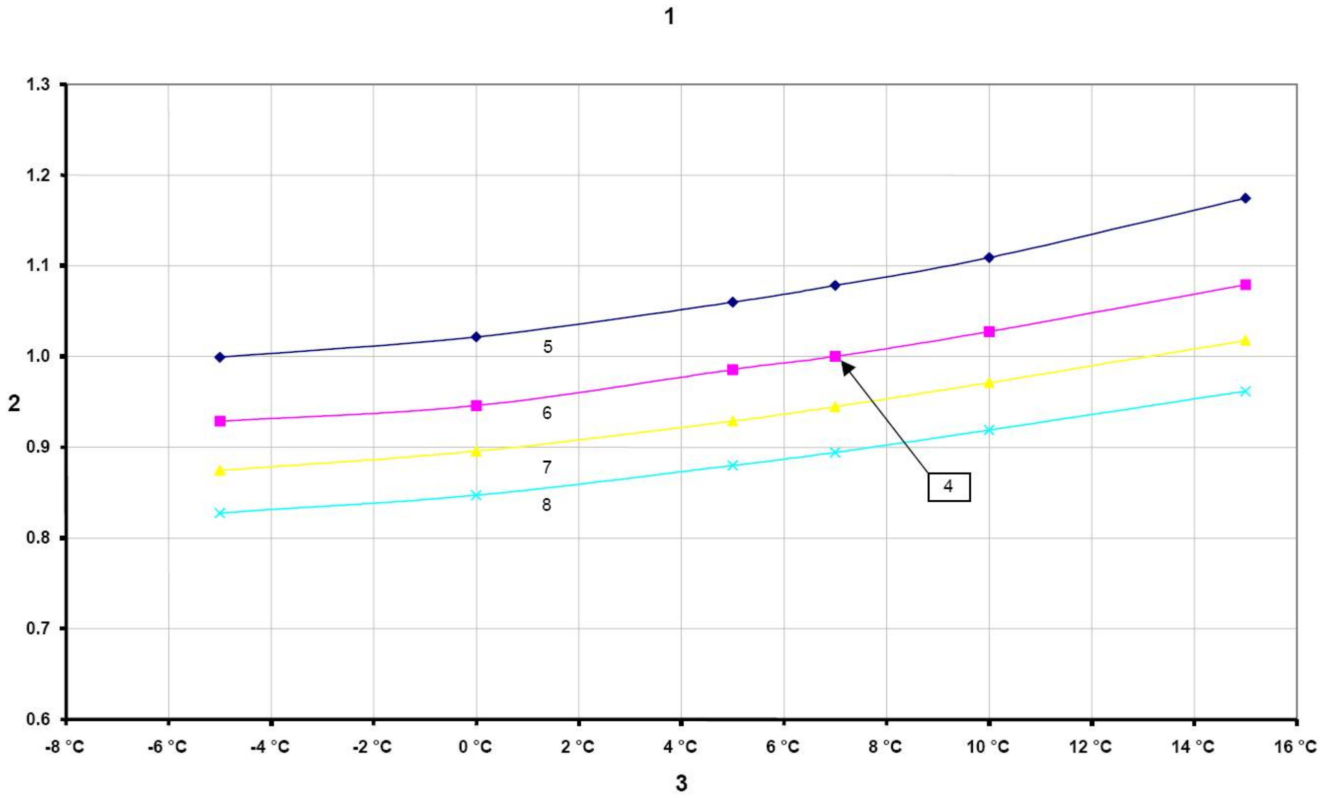
Relative heating capacity		
Legend	**Inlet source temperature**	**Outlet sink temperature**
1 Air-to-air gas engine driven heat pump		**20°C**
2 Relative heating capacity [kW]	**-7°C**	0,90
3 Inlet source temperature [°C]	**-5°C**	0,94
4 Outlet sink temperature: 20°C	**0°C**	0,97
	6°C	1,00
	9°C	1,00
	12°C	1,00

Figure F 9 — Heating capacity of air-to-air gas engine-driven heat pumps vs. source and sink temperatures

F.3.3 COP

Following data are based on the average values of technical features provided by manufacturers.

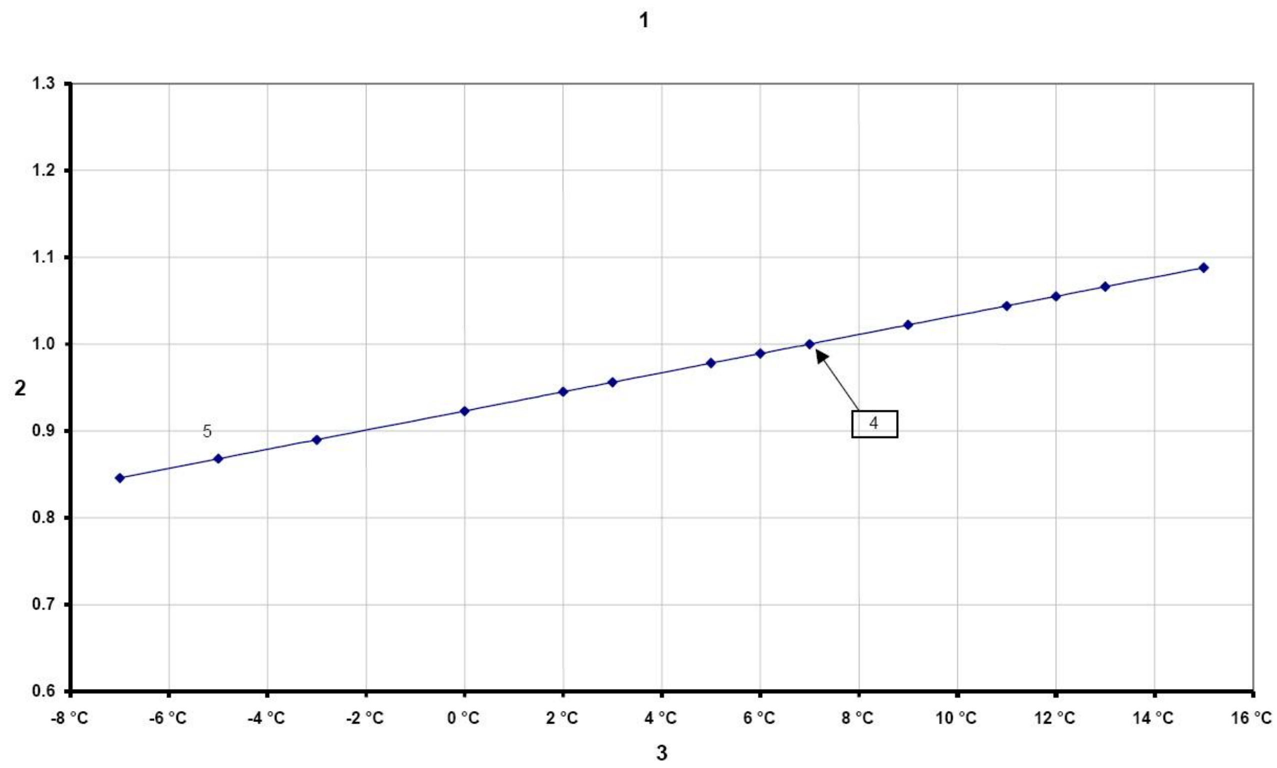
F.3.3.1 Air-to-water heat pumps



	Relative COP				
Legend	**Inlet source temperature**	**Outlet sink temperature**			
1 Air-to-water gas engine driven heat pump		**40°C**	**45°C**	**50°C**	**55°C**
2 Relative COP	**-5 °C**	1,00	0,93	0.87	0.83
3 Inlet source temperature	**0 °C**	1,02	0,95	0.90	0.85
4 COP reference point: (7°C/45°C)	**5 °C**	1,06	0,99	0.93	0.88
5 Outlet sink temperature: 40°C (y = 0.00027x^2 + 0.00605x ° 1.02207)	**7°C**	1,08	1,00	0.94	0.89
6 Outlet sink temperature: 45°C (y = 0.0002x_2 + 0.00564x ° 0.95026)	**10 °C**	1,11	1,03	0.97	0.92
7 Outlet sink temperature: 50°C (y = 0.00017x_2 + 0.00555x ° 0.89747)	**15 °C**	1,17	1,08	1,02	0.96
8 Outlet sink temperature: 55°C (y = 0.00015x^2 + 0.00533x° 0.84945)					

Figure F 10 — Typical COP of air-to water gas engine driven heat pumps vs source and sink temperatures

F.3.3.2 Air-to-air heat pumps



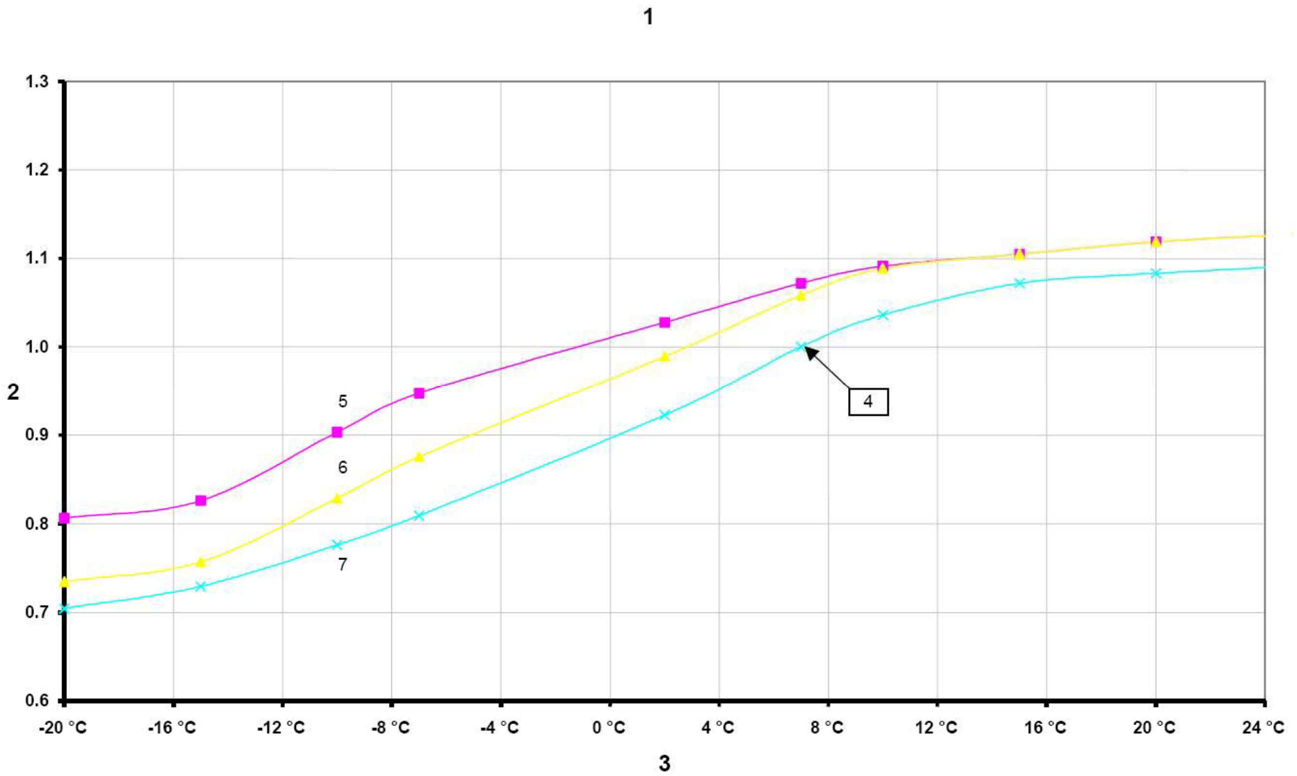
Relative COP		
Legend	**Inlet source temperature**	**Outlet sink temperature**
1 Air-to-air gas engine driven heat pump		**20°C**
2 Relative COP [W/W]	**-7°C**	0.85
3 Inlet source temperature [°C]	**-5°C**	0,87
4 COP reference point 7°C/20°C	**-3°C**	0,89
5 Outlet sink temperature: 20°C	**0°C**	0,92
(y = 0.00018x2 + 0.01106x + 0.92806)	**2°C**	0,94
	3°C	0,96
	5°C	0,98
	6°C	0,99
	7°C	1,00
	9°C	1,02
	11°C	1,04
	12°C	1,06
	13°C	1,07
	15°C	1,09

Figure F 11 — Typical COP of air-to air gas engine driven heat pumps vs source and sink temperatures

F.4 Absorption heat pumps

Typical relations of heating capacity and COP versus source and sink temperatures are given below for ammonia/water and water/lithium bromide Vapour Absorption Cycle (VAC) heat pumps. Relative heating capacity and relative COP show the same ratio, thus the below diagrams are valid for relative heating capacity and COP.

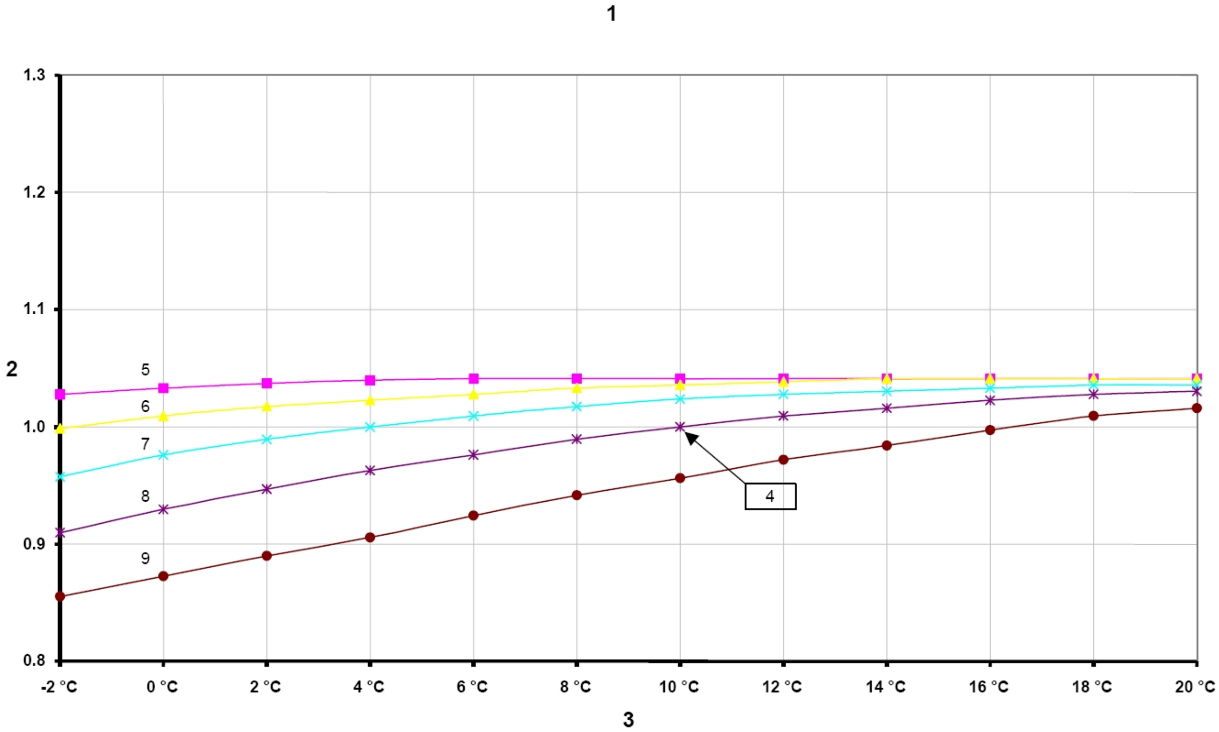
F.4.1.1 NH_3/H_20 heat pumps – outside air-to-water



Relative heating capacity and relative COP				
Legend	**Inlet source temperature**	**Inlet and outlet sink temperatures**		
1 Air-to-water NH_3/H_2O direct gas fired absorption heat pump		**20°C/30°C**	**35°C/45°C**	**40°C/50°C**
2 Relative heating capacity and relative COP	**-20 °C**	0,807	0,735	0,704
3 Inlet source temperature [°C]	**-15 °C**	0,826	0,757	0,729
4 Reference point heating capacity an COP: (7°C/50°C)	**-10°C**	0,903	0,829	0,776
5 Outlet sink temperature: 30°C	**-7°C**	0,948	0,876	0,809
6 Outlet sink temperature: 40°C	**2°C**	1,028	0.989	0,923
7 Outlet sink temperature: 50°C	**7°C**	1,072	1,058	1,0
	10°C	1,091	1,088	1,036
	15°C	1,105	1,105	1,072
	20°C	1,119	1,119	1,083
	25°C	1,127	1,127	1,091

Figure F 12 — Average relative heating capacity and COP of outside air-to water NH_3/H_20 absorption heat pumps vs source and sink temperatures

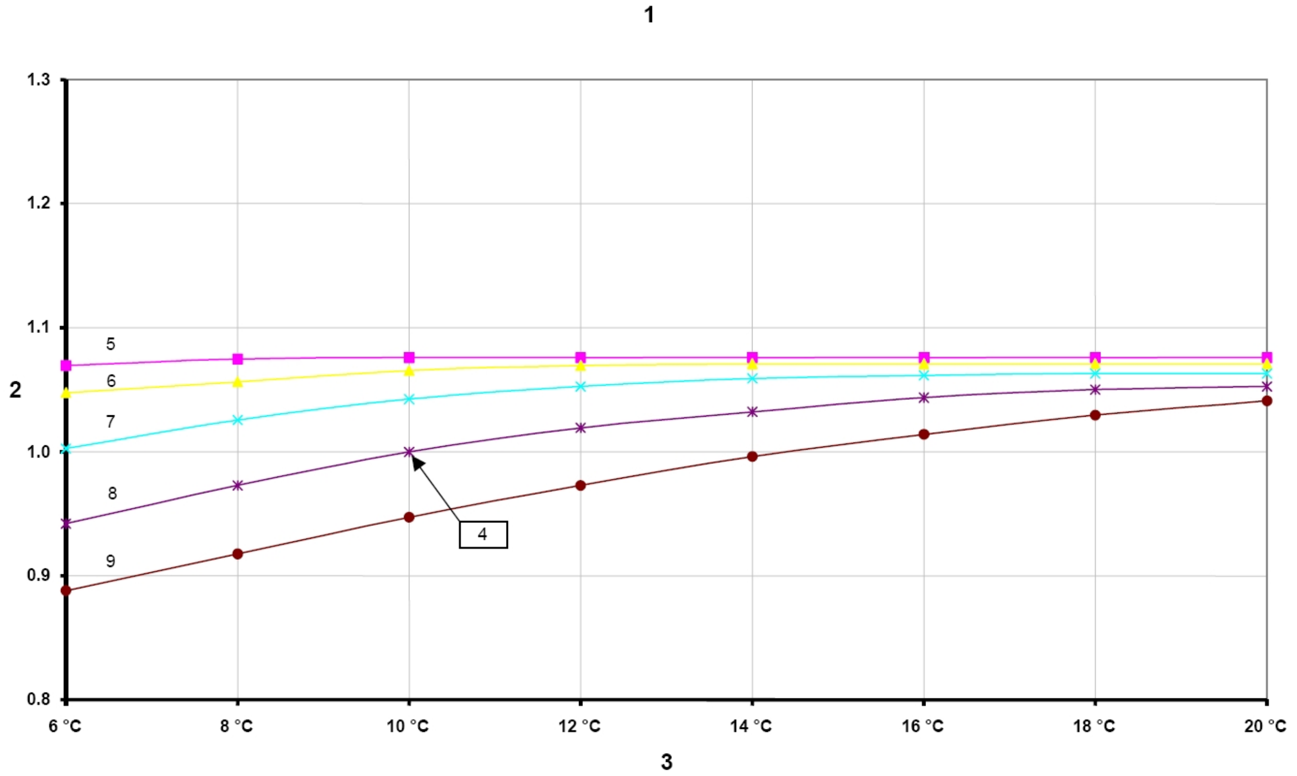
F.4.1.2 NH_3/H_20 heat pumps – brine-to-water



Relative heating capacity and relative COP							
Legend		**Inlet source tempe-rature**	**Inlet and outlet sink temperatures**				
			25°C/ 35°C	**30°C/ 40°C**	**40°C/ 50°C**	**40°C/ 50°C**	**45°C/ 55°C**
1	Brine-to-water NH_3/H_2O direct gas fired absorption heat pump						
2	Relative heating capacity and relative COP	**-2 °C**	1,028	0,999	0,958	0,910	0,855
3	Inlet source temperature [°C]	**0 °C**	1,033	1,009	0.976	0,930	0,873
4	Reference point heating capacity an COP: (10°C/50°C)	**2°C**	1,037	1,017	0,989	0,947	0,890
5	Outlet sink temperature: 35°C ($y = -0.00006x^2+0.00147x+1.03309$)	**4°C**	1,040	1,023	1,000	0,963	0,906
6	Outlet sink temperature: 40°C ($y = -0.00012x^2+0.00399x+1.00850$)	**6°C**	1,041	1,028	1,009	0,976	0,924
7	Outlet sink temperature: 45°C ($y = -0.00019x^2+0.00682x+0.97475$)	**8°C**	1,041	1,033	1,017	0,989	0,942
8	Outlet sink temperature: 50°C ($y = -0.0002x^2+0.00908x+0.92927$)	**10°C**	1,041	1,036	1,023	1,000	0,956
9	Outlet sink temperature: 55°C ($y = -0.00011x^2+0.00943x+0.87246$)	**12°C**	1,041	1,039	1,028	1,009	0,,972
		14°C	1,041	1,041	1,031	1,016	0,984
		16°C	1,041	1,041	1,033	1,023	0,997
		18°C	1,041	1,041	1,036	1,028	1,009
		20°C	1,041	1,041	1,036	1,031	1,016

Figure F 13 — Average relative heating capacity and COP of brine-to water NH_3/H_20 absorption heat pumps vs source and sink temperatures

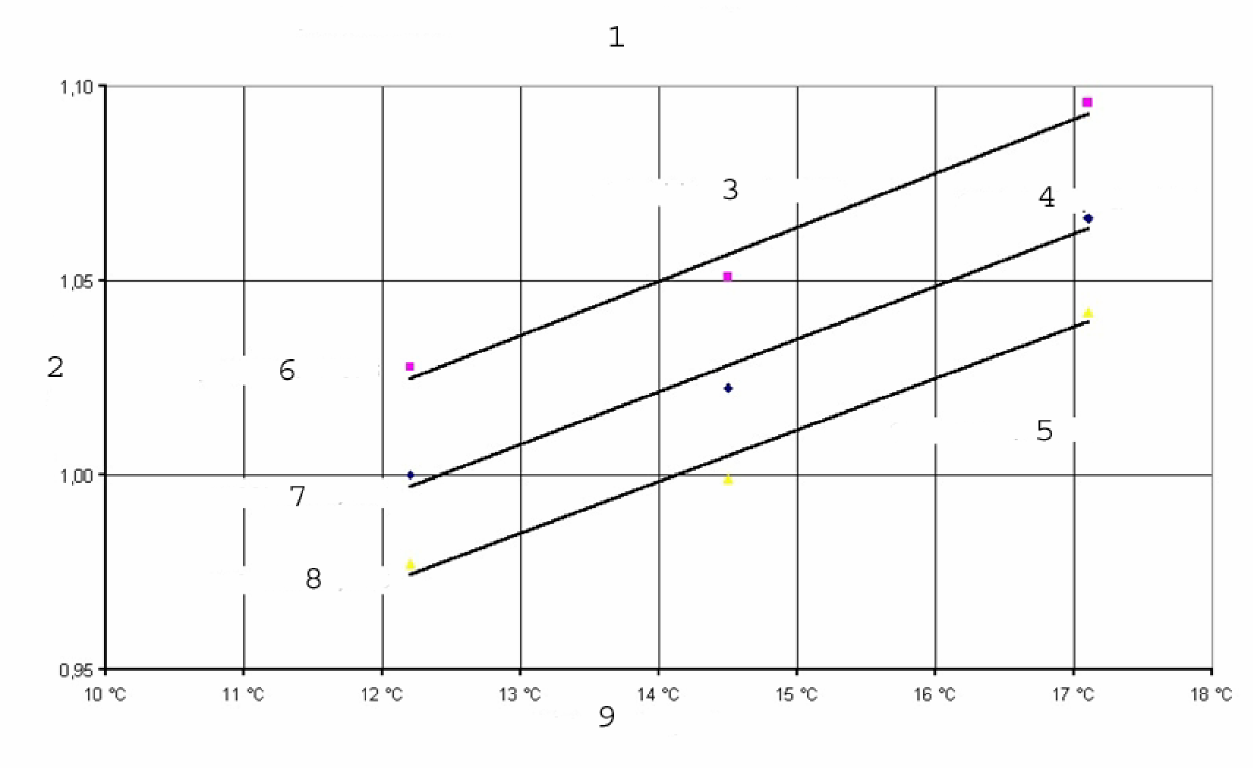
F.4.1.3 NH_3/H_20 heat pumps – water-to-water



Relative heating capacity and relative COP							
Legend		Inlet source tempe-rature	Inlet and outlet sink temperatures				
1	Water-to-water NH_3/H_2O direct gas fired absorption heat pump		25°C/ 35°C	30°C/ 40°C	40°C/ 50°C	40°C/ 50°C	45°C/ 55°C
2	Relative heating capacity and relative COP	6°C	1,069	1,071	1,003	0.942	0,888
3	Inlet source temperature [°C]	8°C	1,075	1,071	1,026	0,973	0,918
4	Reference point heating capacity an COP: (10°C/50°C)	10°C	1,076	1,071	1,042	1,000	0,947
5	Outlet sink temperature: 35°C ($y = -0.00007x^2+0.00210x+1.06078$)	12°C	1,076	1,071	1,053	1.019	0,973
6	Outlet sink temperature: 40°C ($y = -0.00023x^2+0.00746x+1.01198$)	14°C	1,076	1,069	1,059	1,032	0,996
7	Outlet sink temperature: 45°C ($y = -0.00048x^2+0.01656x+0.92272$)	16°C	1,076	1,066	1,062	1,044	1,014
8	Outlet sink temperature: 50°C ($y = -0.00006x^2+0.02330x+0.82474$)	18°C	1,076	1,057	1,063	1,050	1,030
9	Outlet sink temperature: 55°C ($y = -0.00041x^2+0.02180x+0.87246$)	20°C	1,076	1,048	1,063	1,053	1,041

Figure F 14 — Average heating capacity of water-to water NH_3/H_20 absorption heat pumps vs source and sink temperatures

F.4.1.4 H_20/LiBr heat pump



Relative heating capacity and relative COP					
Legend		**Inlet source temperature**	**Outlet sink temperature**		
			30°C	**34°C**	**37°C**
1	Water-to-water H_2O/LiBr direct fired absorption heat pump				
2	Relative COP	**12,2 °C**	1,028	1,000	0,977
3	Outlet sink temperature: 30°C (y = 0.00027x^2 + 0.00605x ° 1.02207)	**14,5°C**	1,051	1,022	0,999
4	Outlet sink temperature 34°C	**17,1°C°C**	1,096	1,066	1,042
5	y = 0.01393x + 0.85476				
6	Outlet sink temperature: 37°C				
7	y = 0.01355x + 0.8316				
8	y = 0.01325x + 0.8127				
9	Inlet source temperature [°C]				

Figure F 15 — Average heating capacity of water-to water H_20/LiBr absorption heat pumps vs. source and sink temperatures

F.5 Heat pumps with domestic hot water production (DHW)

F.5.1 Heating capacity of domestic hot water heat pumps

For the heating capacity of domestic hot water heat pumps in alternate operation, the heating capacity of the heating mode can be used as example value (see F.2.1) for the average loading temperature defined in equation eq. 13.

Bibliography

[1] Software Meteonorm 3.0, Meteotest, Switzerland (http://www.meteotest.ch)

[2] Swiss Energy directive EnV of 7. december 1998 (actual status on 30. November 2004), 730.01

[3] WPZ Bulletin, Wärmepumpen-Testzentrum Töss, July 2003, (download of actual WPZ Bulletin available on http://www.wpz.ch)

[4] Arie de Jong, Part 2 of NEN reaction to CEN/TC 228 N373 – Draft resolution C 112 (N374), Nederlands Normalisatie Instituut, Dec. 2003

[5] 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

[6] W. Eugster, Polydynamics Engineering Zürich, aus Bericht: Kostengünstige Niedrigtemperaurheizung mit Wärmepumpe: Phase 1 Machbarkeitsanalyse, Problemanalyse und Vorbereitung zum Bau eines Funktionsmusters. Forschungsprogramm Umgebungs-und Abwärme, Wärmekraftkopplung (UAW). Bundesamt für Energie (BFE),Bern, 1996

[7] SIA 384/2, Wärmeleistungsbedarf von Gebäuden, Swiss Engineers and Architects Association, 1982 (meanwhile replaced by SN 12831:2003)