ICS: 91.140.10

Geregistreerde Belgische norm

NBN EN 15316-4-2

1e uitg., november 2008

Normklasse: D 01

Verwarmingssystemen in gebouwen - Berekeningsmethode voor de systeemenergiebehoefte en het systeemrendement - Deel 4-2: Warmteopwekkers voor ruimteverwarming, warmtepompsystemen

Systèmes de chauffage dans les bâtiments - Méthode de calcul des besoins énergétiques et des rendements des systèmes - Partie 4-2 : Systèmes de génération de chauffage des locaux, systèmes de pompes à chaleur

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

Toelating tot publicatie: 26 november 2008

Deze Europese norm EN 15316-4-2:2008 heeft de status van een Belgische norm.

Deze Europese norm bestaat in drie officiële versies (Duits, Engels, Frans).

standards for .be

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Prijsgroep: 30

ICS: 91.140.10

norme belge enregistrée

NBN EN 15316-4-2

1e éd., novembre 2008

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Autorisation de publication: 26 novembre 2008

La présente norme européenne EN 15316-4-2:2008 a le statut d'une norme belge.

La présente norme européenne existe en trois versions officielles (allemand, anglais, français).

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Prix: groupe 30

EUROPEAN STANDARD

NORME EUROPÉENNE

EUROPÄISCHE NORM

EN 15316-4-2

June 2008

ICS 91.140.10

English Version

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

Systèmes de chauffage dans les bâtiments - Méthode de calcul des besoins énergétiques et des rendements des systèmes - Partie 4-2 : Systèmes de génération de chauffage des locaux, systèmes de pompes à chaleur

Heizungsanlagen in Gebäuden - Verfahren zur Berechnung der Energieanforderungen und Nutzungsgrade der Anlagen - Teil 4-2: Wärmeerzeugung für die Raumheizung, Wärmepumpensysteme

This European Standard was approved by CEN on 15 May 2008.

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EUROPÄISCHES KOMITEE FÜR NORMUNG

Management Centre: rue de Stassart, 36 B-1050 Brussels

Ref. No. EN 15316-4-2:2008: E

Contents

Foreword

This document (EN 15316-4-2:2008) has been prepared by Technical Committee CEN/TC 228 “Heating systems in buildings”, the secretariat of which is held by DS.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by December 2008, and conflicting national standards shall be withdrawn at the latest by December 2008.

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. CEN [and/or CENELEC] shall not be held responsible for identifying any or all such patent rights.

This document has been prepared under a mandate given to CEN by the European Commission and the European Free Trade Association (Mandate M/343), and supports essential requirements of EU Directive 2002/91/EC on the energy performance of buildings (EPBD). It forms part of a series of standards aimed at European harmonisation of the methodology for calculation of the energy performance of buildings. An overview of the whole set of standards is given in CEN/TR 15615 [13].

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.

EN 15316 *Heating systems in buildings — Method for calculation of system energy requirements and system efficiencies* consists of the following parts:

Part 1: General

Part 2-1: Space heating emission systems

Part 2-3: Space heating distribution systems

Part 3-1: Domestic hot water systems, characterisation of needs (tapping requirements)

Part 3-2: Domestic hot water systems, distribution

Part 3-3: Domestic hot water systems, generation

Part 4-1: Space heating generation systems, combustion systems (boilers)

Part 4-2: Space heating generation systems, heat pump systems

Part 4-3: Heat generation systems, thermal solar systems

Part 4-4: Heat generation systems, building-integrated cogeneration systems

Part 4-5: Space heating generation systems, the performance and quality of district heating and large volume systems

Part 4-6: Heat generation systems, photovoltaic systems

Part 4-7: Space heating generation systems, biomass combustion systems

According to the CEN/CENELEC Internal Regulations, the national standards organizations of the following countries are bound to implement this European Standard: Austria, Belgium, Bulgaria, Cyprus, Czech Republic, Denmark, Estonia, Finland, France, Germany, Greece, Hungary, Iceland, Ireland, Italy, Latvia, Lithuania, Luxembourg, Malta, Netherlands, Norway, Poland, Portugal, Romania, Slovakia, Slovenia, Spain, Sweden, Switzerland and United Kingdom.

Introduction

This European 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 (EN 15316-1 [9]).

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

This European Standard presents methods for calculation of the additional energy requirements of a heat generation sub-system in order to meet the distribution sub-system 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 sub-system, by applying the method to several possible options;

— assessing the effect of possible energy conservation measures on an existing heat generation sub-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

This European 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,

— resulting outputs,

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

— 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 as listed in Table 1.

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

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 — Test procedures for establishing performance of air to air and flue gases heat recovery devices*

EN 14511:2007 (all parts), *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 15203, *Energy performance of buildings — Application of calculation of energy use to existing buildings*

EN 15316-2-3, *Heating systems in buildings — Method for calculation of system energy requirements and system efficiencies — Part 2-3: Space heating distribution systems*

EN 15316-3-2, *Heating systems in buildings — Method for calculation of system energy requirements and system efficiencies — Part 3-2: Domestic hot water systems, distribution*

EN 15316-3-3, *Heating systems in buildings — Method for calculation of system energy requirements and system efficiencies — Part 3-3: Domestic hot water systems, generation*

EN 15316-4-1, *Heating systems in buildings — Method for calculation of system energy requirements and system efficiencies — Part 4-1: Space heating generation systems, combustion systems (boilers)*

EN ISO 7345:1995, *Thermal insulation — Physical quantities and definitions (ISO 7345:1987)*

EN ISO 13790 *Energy performance of buildings — Calculation of energy use for space heating and cooling (ISO 13790:2008)*

EN ISO 15927-6, *Hygrothermal performance of buildings — Calculation and presentation of climatic data — Part 6: Accumulated temperature differences (degree-days) (ISO 15927-6:2007)*

3 Terms, definitions, symbols and units

3.1 Terms and definitions

For the purposes of this document, the terms and definitions given in EN ISO 7345:1995 and the following 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
electrical energy used by technical building systems for heating, cooling, ventilation and/or domestic hot water to support energy transformation to satisfy energy needs

NOTE 1 This includes energy for fans, pumps, electronics, etc. Electrical energy input to the ventilation system for air transport and heat recovery is not considered as auxiliary energy, but as energy use for ventilation.

NOTE 2 In EN ISO 9488 [15], the energy used for pumps and valves is called "parasitic energy".

NOTE 3 In the frame of this standard, the driving energy input for electrically-driven heat pumps in the system boundary of the *COP* according to EN 14511 and an electrical back-up heater is not entitled auxiliary energy but only additional electrical input not considered in the *COP*.

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

3.1.6
building services
services provided by technical building systems and by appliances to provide indoor climate conditions, domestic hot water, illumination levels and other services related to the use of the building

3.1.7
calculation period
period of time over which the calculation is performed

NOTE The calculation period can be divided into a number of calculation steps.

3.1.8
calculation step
discrete time interval for the calculation of the energy needs and uses for heating, cooling, humidification and dehumidification

NOTE 1 Typical discrete time intervals are one hour, one month or one heating and/or cooling season, operating modes, and bins.

NOTE 2 In the frame of the bin method, calculation steps are based on outdoor temperature classes.

3.1.9
coefficient of performance *COP*
ratio of the heating capacity to the effective power input of the unit

3.1.10
cumulative frequency
frequency of the outdoor air temperature cumulated over all 1 K bins

3.1.11
cut-out period
time period in which the electricity supply to the heat pump is interrupted by the supplying utility

3.1.12
domestic hot water heating
process of heat supply to raise the temperature of the cold water to the intended delivery temperature

3.1.13
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;
- the proportional power input of the conveying devices (e.g. fans, pumps) for ensuring the transport of the heat transfer media inside the unit

3.1.14
electrically-driven heat pump
in the frame of this European Standard, electrically-driven heat pumps denote vapour compression cycle heat pumps, which incorporate a compressor that is driven by an electric motor

3.1.15
energy need for domestic hot water
heat to be delivered to the needed amount of domestic hot water to raise its temperature from the cold network temperature to the prefixed delivery temperature at the delivery point, not taking into account the technical building thermal systems

3.1.16
energy need for heating or cooling
heat to be delivered to or extracted from a conditioned space to maintain the intended temperature during a given period of time, not taking into account the technical building thermal systems

NOTE 1 The energy need is calculated and cannot easily be measured.

NOTE 2 The energy need can include additional heat transfer resulting from non-uniform temperature distribution and non-ideal temperature control if they are taken into account by increasing (decreasing) the effective temperature for heating (cooling) and not included in the heat transfer due to the heating (cooling) system.

3.1.17
energy use for space heating or cooling or domestic hot water
energy input to the heating, cooling or domestic hot water system to satisfy the energy need for heating, cooling (including dehumidification) or domestic hot water, respectively

NOTE If the technical building system serves several purposes (e.g. heating and domestic hot water) it can be difficult to split the energy use into that used for each purpose. It can be indicated as a combined quantity (e.g. energy need for space heating and domestic hot water).

3.1.18
frequency
(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 European Standard, the frequency of the outdoor air temperature is evaluated based on a sample of hourly-averaged data for one year

3.1.19
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

3.1.20
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

3.1.21
heat recovery
heat generated by a technical building system or linked to a building use (e.g. domestic hot water) which is utilised directly in the related system to lower the heat input and which would otherwise be wasted (e.g. preheating of the combustion air by flue gas heat exchanger)

3.1.22
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

3.1.23
heated space
room or enclosure which for the purposes of the calculation is assumed to be heated to a given set-point temperature or set-point temperatures

3.1.24
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.

3.1.25
heating or cooling season
period of the year during which a significant amount of energy for heating or cooling is needed

NOTE The season lengths are used to determine the operation period of technical systems.

3.1.26
internal temperature
arithmetic average of the air temperature and the mean radiant temperature at the centre of the occupied zone

NOTE This is the approximate operative temperature according to EN ISO 7726 [14].

3.1.27
low temperature cut-out
temperature at which heat pump operation is stopped and the total heat requirements are covered by a back-up heater

3.1.28
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

3.1.29
part load operation
operation state of the technical system (e.g. heat pump) where the actual load requirement is below the actual output capacity of the device

3.1.30
part load ratio
ratio between the generated heat during the calculation period and the maximum possible output from the heat generator during the same calculation period

3.1.31
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

3.1.32
produced heat
heat produced by the generator subsystems, i.e. the heat produced to cover the energy requirement of the distribution subsystem and the generation subsystem heat losses for space heating and/or domestic hot water

3.1.33
recoverable system thermal loss
part of the system thermal loss which can be recovered to lower either the energy need for heating or cooling or the energy use of the heating or cooling system

3.1.34
recovered system thermal loss
part of the recoverable system thermal loss which has been recovered to lower either the energy need for heating or cooling or the energy use of the heating or cooling system

3.1.35
seasonal performance factor *SPF*
in the frame of this European 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 combustion engine-driven heat pumps or absorption heat pumps) plus the total annual input of auxiliary energy

3.1.36
set-point temperature of a conditioned zone
internal (minimum intended) temperature, as fixed by the control system in normal heating mode, or internal (maximum intended) temperature, as fixed by the control system in normal cooling mode

3.1.37
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

3.1.38
space heating
process of heat supply for thermal comfort

3.1.39
standard rating condition
mandatory condition that is used for marking and for comparison or certification purposes

3.1.40
system thermal losses
thermal loss from a technical building system for heating, cooling, domestic hot water, humidification, dehumidification, ventilation or lighting that does not contribute to the useful output of the system

NOTE Thermal energy recovered directly in the subsystem is not considered as a system thermal loss but as heat recovery and is directly treated in the related system standard.

3.1.41
technical building system
technical equipment for heating, cooling, ventilation, domestic hot water, lighting and electricity production composed by sub-systems

NOTE 1 A technical building system can refer to one or to several building services (e.g. heating system, space heating and domestic hot water system).

NOTE 2 Electricity production can include cogeneration and photovoltaic systems.

3.1.42
technical bulding sub-system
part of a technical building system that performs a specific function (e.g. heat generation, heat distribution, heat emission)

3.2 Symbols and units

For the purposes of this document, the following symbols and units (Table 2), abbreviations (Table 3) and indices (Table 4) apply.

Table 2 — Symbols and units

Symbol	Name of quantity	Unit
ϕ	Thermal power, heating capacity, heat flow rate	W
η	Efficiency factor	-
θ	Celsius temperature	°C
ρ	Density	kg/m^3
$\Delta\theta$	Temperature difference, - spread	K
Δp	Pressure difference	Pa
b	Temperature reduction factor	-
c	Specific heat capacity	J/(kg·K)
DH	Degree hours	°Ch
COP	Coefficient of performance	W/W
COP_t	Coefficient of performance for the tapping of hot water	W/W
E	Quantity of energy, fuel	J
f	Factor (dimensionless)	-
β	Load factor	-
m'	Mass flow rate	kg/s
N	Number of items	-
k	Factor (fraction)	-
P	Power, electrical power	W
Q	Quantity of heat	J
SPF	Seasonal performance factor	-
t	Time, period of time	s
T	Thermodynamic temperature	K
V	Volume	m^3
V'	Volume flow rate	m^3/s
W	Auxiliary (electrical) energy	J

Table 3 — 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

Table 4 — Indices

$\Delta\theta$	temperature corrected	eng	engine	nrbl	non recoverable
θllim	lower temperature limit	es	storage values according to EN 255-3:1997, phase 4	on	running, in operation
θhlim	upper temperature limit	ex	exergetic	opr	operating, operation limit
amb	ambient	f	flow	out	output from subsystem
aux	auxiliary	gen	generation subsystem	p	pipe
avg	average	H	space heating	r	return
bal	balance point	hot	hot process side	rbl	recoverable
bu	back-up (heater)	ho	hour	rvd	recovered
cap	lack of capacity	hp	heat pump	st	storage
co	cut-out	i	internal	sby	stand-by
cold	cold process side	in	input to subsystem	sk	sink
combi	combined operation	j	index, referring to bin j	sngl	single (operation)
crnt	carnot	k	index	sc	source
dis	distribution subsystem	ls	loss	standard	according to standard testing
des	at design conditions	ltc	low temperature cut-out	tot	total
e	external	max	maximum	w	water, heat transfer medium
eff	effective	n	nominal	W	domestic hot water (DHW), DHW operation

NOTE The indices specifying the symbols in this standard are put in the following order:

— the first index represents the type of energy use (H = space heating, W = domestic hot water). If the equation can be applied for different energy uses by using the values of the respective operation mode, the first level index is omitted;

— the second index represents the subsystem or generator (gen = generation, dis = distribution, hp = heat pump, st = storage, etc.);

— the third index represents the type (ls= losses, gs = gains, in = input, etc.);

— other indices can be used for more details (rvd = recovered, rbl = recoverable, i = internal, etc.);

— a prefix n means non, rbl - recoverable, nrbl – non-recoverable.

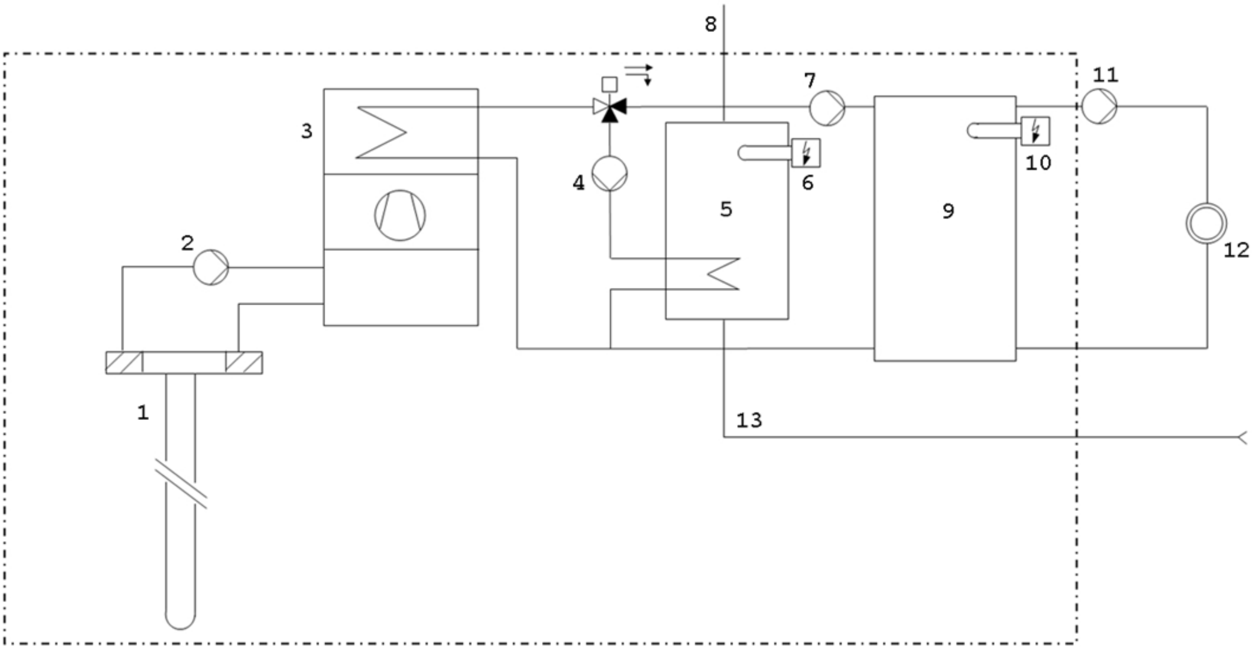
The indices are separated by a comma.

4 Principle of the method

4.1 Heat balance of generation subsystem

System boundary

The system boundary defines the components of the entire heating systems that are considered in this European 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 European Standard, however, the efficiency calculation shall be accomplished according to the appropriate other part(s) of EN 15316 (see 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 subsystem
12 heat emission subsystem
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 subsystems:

— type of generator configuration (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(s);

— 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 exist;

— auxiliary energy input needed to operate the generation subsystem not considered in standard testing of heating capacity and *COP*;

— system thermal losses due to space heating or domestic hot water 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

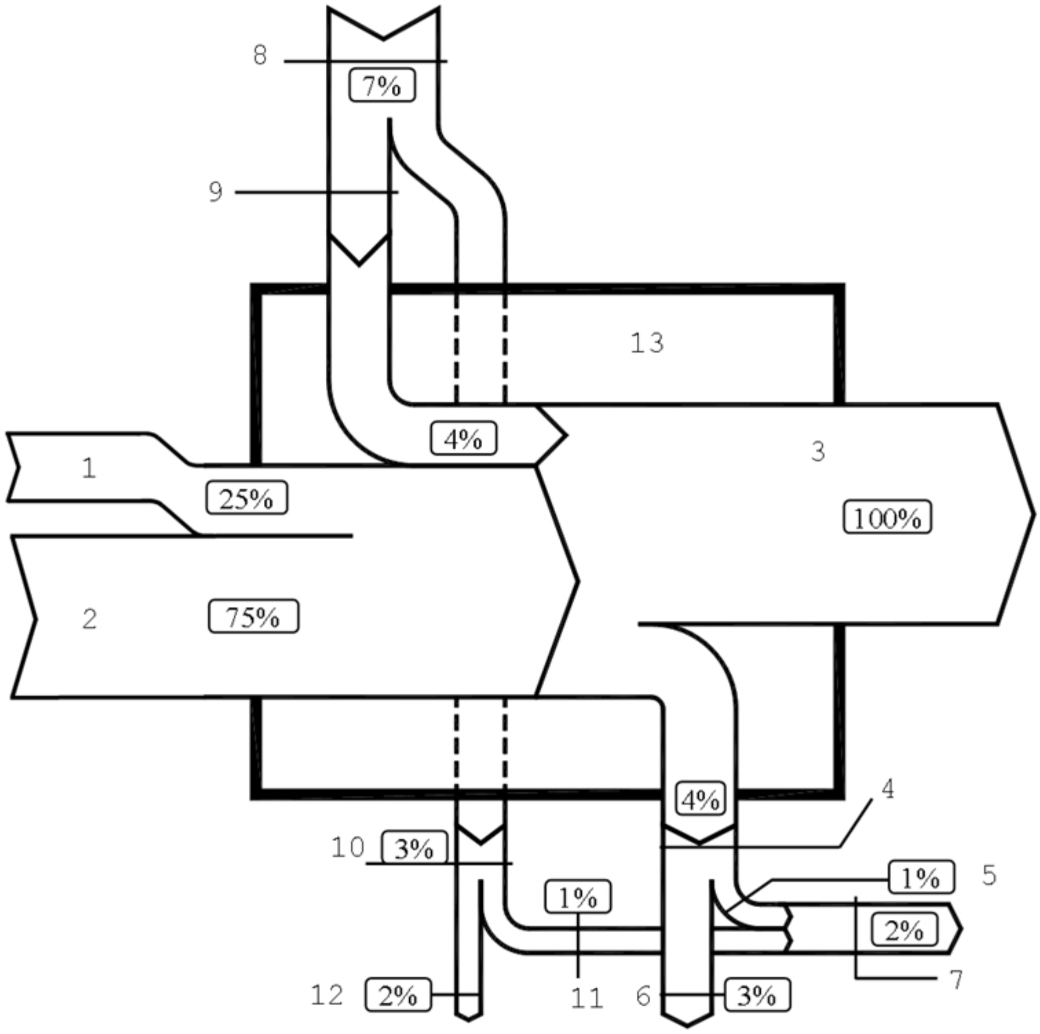
— required energy input as driving energy $E_{\mathrm{HW,gen,in}}$ (electricity, fuel, waste heat, solar heat) to meet the space heating and/or domestic hot water requirements;

— total generation subsystem thermal loss $Q_{\mathrm{HW,gen,ls,tot}}$;

— total recoverable generation subsystem thermal losses $Q_{\mathrm{HW,gen,ls,rbl,tot}}$;

— total required auxiliary energy $W_{\mathrm{HW,gen,aux}}$ to operate the generation subsystem.

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



Key

1 driving energy input to cover the heat requirement (e.g. electricity, fuel) $E_{\mathrm{HW,gen,in}}$

2 ambient heat used as heat source of the heat pump $Q_{\mathrm{HW,gen,in}}$

3 heat output of the generation subsystem corresponding to the heat requirement of the distribution subsystem(s) $Q_{\mathrm{HW,gen,out}} = Q_{\mathrm{HW,dis,in}}$

4 generation subsystem thermal losses $Q_{\mathrm{HW,gen,ls}}$

5 generation subsystem thermal loss (thermal part) recoverable for space heating $Q_{\mathrm{HW,gen,ls,rbl}}$

6 generation subsystem thermal loss (thermal part) non-recoverable $Q_{\mathrm{HW,gen,ls,nrbl}}$

7 generation subsystem thermal loss recoverable for space heating $Q_{\mathrm{HW,gen,ls,rbl,tot}}$

8 generation subsystem total auxiliary energy $W_{\mathrm{HW,gen,aux}}$

9 generation subsystem recovered auxiliary energy $Q_{\mathrm{HW,gen,aux,ls,rvd}}$

10 generation subsystem unrecovered auxiliary energy $Q_{\mathrm{HW,gen,aux,ls}}$

11 generation subsystem recoverable auxiliary energy $Q_{\mathrm{HW,gen,aux,ls,rbl}}$

12 generation subsystem non-recoverable auxiliary energy $Q_{\mathrm{HW,gen,aux,ls,nrbl}}$

13 generation subsystem

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

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

$$E_{\mathrm{HW,gen,in}} = Q_{\mathrm{HW,gen,out}} + Q_{\mathrm{HW,gen,ls}} - Q_{\mathrm{HW,gen,in}} - k_{\mathrm{gen,aux,ls,rvd}} \cdot W_{\mathrm{HW,gen,aux}} \quad \text{[J]} \qquad (1)$$

where

$E_{\mathrm{HW,gen,in}}$	is the driving electrical energy, fuel or heat input to cover the heat requirement of the distribution subsystem;	(J)
$Q_{\mathrm{HW,gen,out}}$	is the heat energy requirement of the distribution subsystems;	(J)
$Q_{\mathrm{HW,gen,ls}}$	is the thermal losses of the generation subsystem;	(J)
$Q_{\mathrm{HW,gen,in}}$	is the ambient heat energy used as heat source of the heat pump;	(J)
$k_{\mathrm{gen,aux,ls,rvd}}$	is the recovered fraction of auxiliary energy ;	(-)
$W_{\mathrm{HW,gen,aux}}$	is the auxiliary energy input to operate the generation subsystem.	(J)

In case of electrically-driven heat pumps:

— the term $E_{\mathrm{HW,gen,in}}$ 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, $E_{\mathrm{HW,gen,in}}$ also includes the fractions of the auxiliary energies included in the *COP*. According to EN 14511 the auxiliary energies at the system boundary 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 eventual energy for defrost operation and additional heating devices for the oil supply of the compressor (carter heating);

— thus, $W_{\mathrm{HW,gen,aux}}$ comprises only the fractions not included in the *COP* according to the EN 14511 standard testing. $k_{\mathrm{gen,aux,ls,rvd}}$ 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 $k_{\mathrm{gen,aux,ls,rvd}} = 0$;

— for thermal losses $Q_{\mathrm{HW,gen,ls}}$, the thermal losses of the heat pump over the envelope are neglected, unless heat loss values of the heat pump are known, e.g. given in a national annex. For systems with integrated or external heating buffer or domestic hot water storage, generation subsystem thermal losses in form of storage thermal losses and thermal losses from the connecting circulation pipes to the storage are considered.

In case of combustion engine-driven and absorption heat pumps:

— $E_{\mathrm{HW,gen,in}}$ 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 as well as solar energy or waste heat can be the driving energy input;

— $Q_{\mathrm{HW,gen,out}}$ is the heat energy output of the generation subsystems which is equal to the heat requirement of the distribution subsystems and contains all fractions of heat recovered from the engine or the flue gas of the engine, i.e. recovered heat from the engine is entirely considered within the system boundary of the generation subsystem;

— $k_{gen,aux,ls,rvd}$ gives the fraction of the auxiliary energy recovered as thermal energy and depends on the test method. The fraction $k_{gen,aux,ls,rvd} = 0$ if the recovered auxiliary energy is already included in the *COP*.

4.3 Auxiliary energy $W_{HW,gen,aux}$

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 European Standard are calculated on the basis of results from product testing, according to EN 14511, and 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 $W_{HW,gen,aux}$.

Auxiliary energy is accounted for in the generation subsystem as long as no transport energy is transferred to the distribution subsystem. Thus, in general the circulation pump is considered in the distribution subsystem, unless a hydraulic decoupling exists. For a hydraulic decoupling between the generation and various distribution subsystems, e.g. by a heating buffer or domestic hot water storage in parallel configuration, the primary pump is considered in 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

Not all of the calculated system thermal losses are necessarily lost. Some of the losses are recoverable, and part of the recoverable system thermal losses is 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 thermal losses $Q_{HW,gen,ls,rbl}$ 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 $Q_{HW,gen,out}$.

4.5 Calculation periods

Heat pump performance strongly depends on the operating 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 5.3 on an annual basis and field monitoring values showed an exactness of the calculation in the range of 6 %. So, with 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 into zones with different distribution subsystems. A separate circuit may be used for domestic hot water production.

Several heat generation subsystems may be available.

The total heat requirement of all the distribution subsystems, for instance of the space heating operation, shall equal the total heat output of the generation subsystems:

$$\sum_{j} Q_{\mathrm{H,gen,out,j}} = \sum_{k} Q_{\mathrm{H,dis,in,k}} \qquad [\mathrm{J}] \qquad (2)$$

where

$Q_{\mathrm{H,gen,out,,}}$ is the space heating heat energy requirement to be covered by generator j; (J)

$Q_{\mathrm{H,dis,in,k}}$ is the heat energy requirement of space heating distribution subsystem k. (J)

When more generators are available (multivalent system configuration), the total heat demand of the distribution subsystem(s) $\sum Q_{\mathrm{H,dis,in,k}}$ shall be distributed among the available generators and the calculation described in Clause 5 shall be performed independently for each generation subsystem j on the basis of $Q_{\mathrm{H,gen,out,j}}$. 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 EN 15316-1 [9] or EN 15316-2-1 [10], 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.

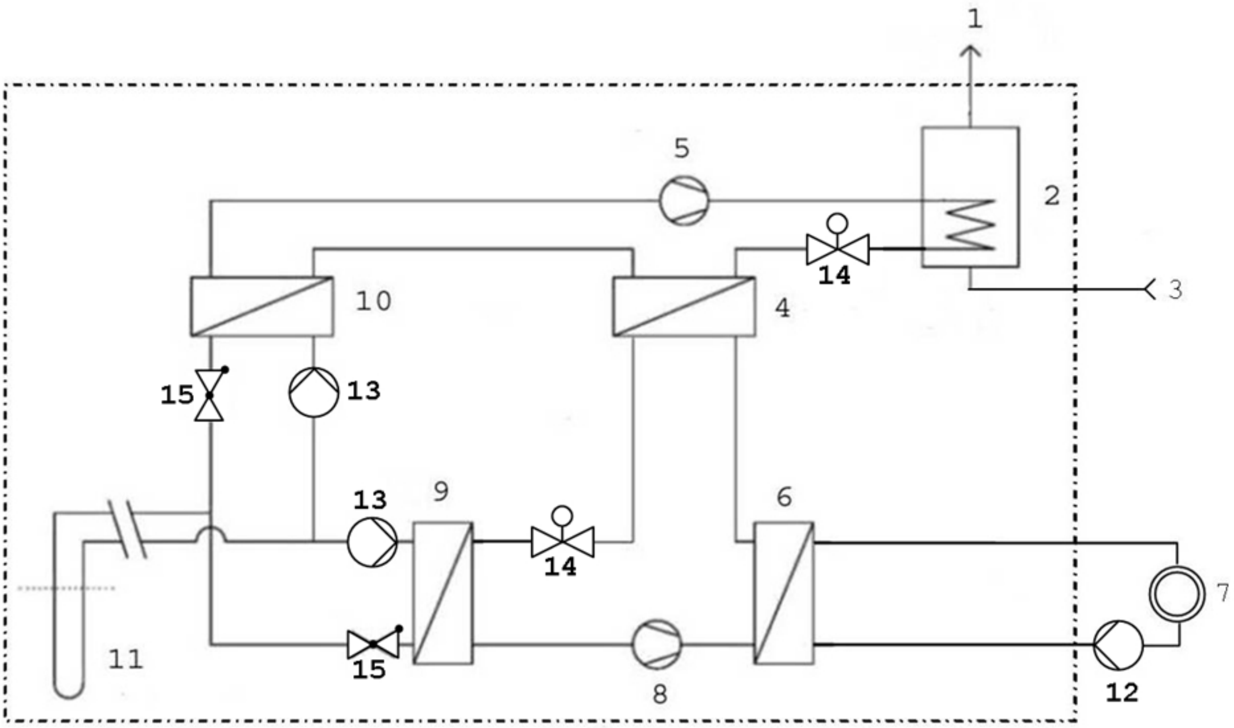
For these simultaneous concepts, space heating and domestic hot water requirements are covered at the same time. Figure 3 gives a sample of a hydraulic scheme of a simultaneous operating system using a cascade cycle with condensate subcooling.

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, domestic hot water 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).



Key

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 as evaporator of the upper stage
5 compressor upper stage
6 condenser lower stage
7 heat emission subsystem
8 compressor lower stage
9 evaporator lower stage
10 heat exchanger to ground source upper stage
11 vertical borehole ground heat-exchanger
12 circulation pump space heating distribution system
13 source pump
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 European Standard implies that both the single operation modes and the simultaneous operation are tested according to standard testing, so heating capacity- and *COP* characteristics of all three respective operation modes are available. As heating capacity- and *COP* characteristics 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 subsystem energy performance calculation

5.1 General

In this European 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;

— operating conditions taken into account;

— calculation periods.

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

Simplified seasonal performance method based on system typology, see 5.2 (tabulated values)

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 operating 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 may be the case for existing buildings, where for instance the *COP* of the heat pump has to be estimated.

Detailed case specific calculation based on component efficiency data, see 5.3 (Bin-method)

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 operating 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 operating conditions of the heat pump. The method shall be carried out with product data for the heating capacity and the *CO*P. 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 available data and the objectives of the user.

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

5.2.1 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 *COP*, e.g. 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 into account the type of heat pump and the type of energy input;

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

iii) deliver results (annual energy consumption, generation thermal loss, auxiliary energy, total recoverable generation thermal 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 subsystem 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 5.3 or any other method 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.

5.2.2 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 input to the generation subsystem;
- generation subsystem total thermal losses;
- generation subsystem auxiliary energy;
- generation subsystem thermal loss recoverable for space heating;
- optionally, the seasonal performance factor.

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

5.3.1 Principle of the bin method

The required energy input e.g. for the space heating operation mode $E_{\mathrm{H,gen,in}}$ according to Equation (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

$$E_{\mathrm{H,gen,in}} = \sum_{j} \frac{Q_{\mathrm{H,gen,out,j}} + Q_{\mathrm{H,gen,ls,j}}}{COP_{\mathrm{H,hp,j}}} \quad [\mathrm{J}] \qquad (3)$$

where

$E_{\mathrm{H,gen,in}}$ is the electrical energy input to cover the heat requirement of the space heating distribution subsystem; (J)

$Q_{\mathrm{H,gen,out,j}}$ is the heat energy requirement of the distribution subsystem in bin j (equal to heat output of the generation subsystem in bin j); (J)

$Q_{\mathrm{H,gen,ls,j}}$ is the thermal losses of the generation subsystem in bin j; (J)

$COP_{\mathrm{H,hp,j}}$ is the coefficient of performance of the heat pump for a period of constant operating conditions. (W/W)

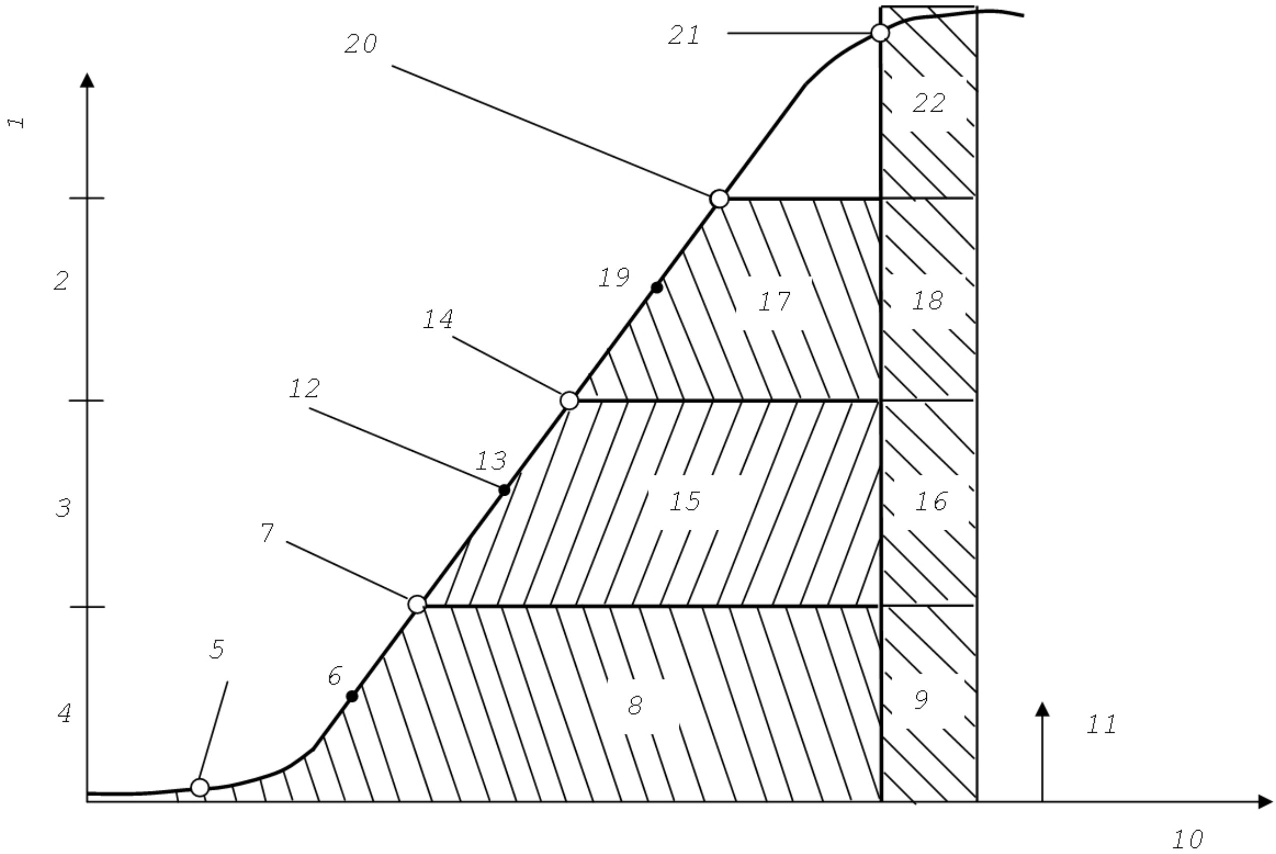
Equation (3) takes into account that the thermal losses of the generation subsystem $Q_{H,gen,ls,j}$ have to be covered by the generator as well.

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 j periods defined by constant source and sink temperature conditions, and results are summed-up, which is expressed by the summation in Equation (3). Thus, to determine the required energy input, basically the *COP* as well as the heat energy requirement and generation subsystem thermal 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 are 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_{hlim} and a lower temperature θ_{llim}. Operating 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 ISO 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 (DH_H) are also called accumulated time temperature difference (ATTD) in EN ISO 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 operating 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.

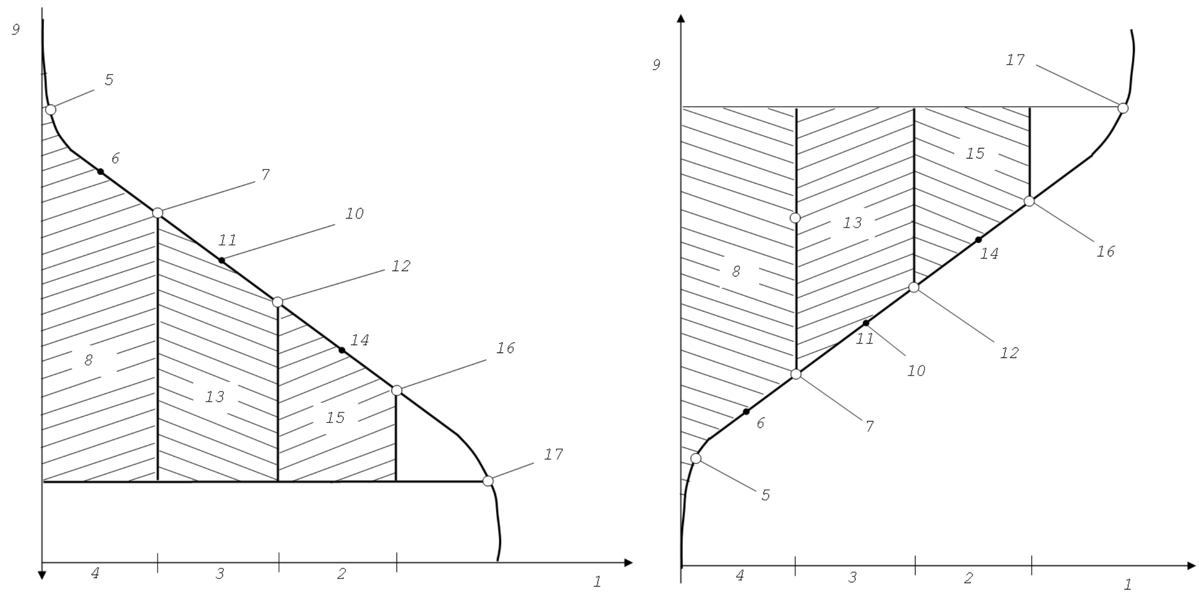


Key

1 cumulative bin hours [h]
2 t_3
3 t_2
4 t_1
5 design outdoor temperature
6 OP_1
7 $\theta_{1,hlim} = \theta_{2,llim}$
8 SH_1
9 DHW_1
10 outdoor air temperature [°C]
11 direction of cumulating temperature difference (space heating load) and DHW load over time
12 θ_2
13 OP_2
14 $\theta_{2,hlim} = \theta_{3,llim}$
15 SH_2
16 DHW_2
17 SH_3
18 DHW_3
19 OP_3
20 upper ambient temperature for space heating = $\theta_{3,hlim}$
21 design indoor temperature
22 DHW_4

Figure 4 — Bin hours vs outdoor air temperature – sample with 3 bins for space heating (SH) and constant daily domestic hot water (DHW) heat energy requirement (4 bins for DHW)

Another common way to depict the cumulative frequency is a 90° clockwise rotation called duration curve, which is shown in Figure 5 left hand side. Since this implies a negative temperature (y-) axis, a horizontally flipped diagram depicted in Figure 5 right hand side is found as well. 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.



Key

1 cumulative bin hours [h]

2 t_3

3 t_2

4 t_1

5 design outdoor temperature

6 OP_1

7 $\theta_{1,\text{hlim}} = \theta_{2,\text{llim}}$

8 SH_1

9 outdoor air temperature [°C]

10 θ_2

11 OP_2

12 $\theta_{2,\text{hlim}} = \theta_{3,\text{llim}}$

13 SH_2

14 OP_3

15 SH_3

16 upper ambient temperature for space heating = $\theta_{3,\text{hlim}}$

17 design indoor temperature

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

COP values, however, are normally only known at discrete test points based on standard product testing. The number of bins depends on the type of heat pump, the available information on the heat pump characteristic according to the standard testing and the calculation period.

Criteria for the choice of bins are:

— operating points shall be spread more or less evenly over the entire operation range;

— operating points shall be chosen at the test points as far as possible in order to include the available information on the heat pump characteristic (e.g. EN 14511) as exact as possible. Bin limits are to be set approximately in the middle between the operating points;

— number of bins shall reflect the changes in source and sink temperature. If both source and sink temperatures are constant over the whole operation range, one bin may be enough. In the case of big changes, more bins shall be chosen. For a monthly calculation period, less bins may be required than in an annual calculation.

On the other hand, 1 K bins can be chosen, so that the heating capacity and the *COP* is interpolated as described in 5.3.5.1.3. In general, number of bins shall at least correspond to the different source temperatures defined by the test points 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 more information is available, e.g. according to manufacturer information based on the standard testing, more bins can be chosen to accommodate the available information. If not enough test points are available, the heat pump characteristic is interpolated to the respective source/sink temperatures, see 5.3.5.1.3, or the exergetic efficiency method given in Annex C can be applied. Observe that the operating point temperature (corresponding to the outdoor air temperature) corresponds directly to the source temperature in the testing only in case of outdoor air source heat pumps. For ground-coupled heat pumps, for instance, the dependency of the source temperature on the outdoor air temperature has to be considered in order to determine the operating points, see 5.3.5.1.3.

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 EN 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 <u>monthly calculation period</u>, 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 ISO 15927-6 with a base temperature of the design indoor temperature, normally 20 °C. This corresponds to the approach of 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 <u>annual calculation period</u>, 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 be based on the used gains and building type. The higher the fraction of used gains is, the lower the heating limit has to be chosen. However, this is an approximation and a more exact redistribution is delivered by a monthly calculation.

For each bin, the heating capacity and the *COP* are 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. Storage and other generation subsystem thermal losses 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.

5.3.2 Input data for the calculation with the bin method

<u>Boundary conditions:</u>

- 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. test reference year TRY or Meteonorm [1]);
  - outdoor design temperature of the site.

<u>Space heating (SH) mode:</u>

- indoor design temperature;
- heat energy requirement of the space heating distribution subsystem according to EN 15316-2-3;
- type and controller setting of the heat emission subsystem (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, upper temperature limit for heating;

- 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 according to EN 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 to EN 15316-3-2;
- 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 according to EN 15316-4-1).

5.3.3 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 is given in the flow chart in Figure 6.

The individual steps are explained in detail in the remaining part of 5.3 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), if applicable. Additionally, for the back-up energy calculation, a simplified and a detailed method is given in connection with the calculation of the running time.

A stepwise calculation example is given in Annex D.

Step 1: Determination of energy requirement of the single bins (see 5.3.4);

Step 2: Correction of steady state heating capacity/*COP* (e.g. EN 14511) for bin source and sink temperature operating conditions (see 5.3.5);

Step 3: If required, correction of *COP* for part load operation (see 5.3.6);

Step 4: Calculation of generation subsystem thermal losses (see 5.3.7);

Step 5: Determination of back-up energy of the single bins (see 5.3.8, simplified in 5.3.8.3, detailed in 5.3.9.4);

Step 6: Calculation of the running time of the heat pump in different operation modes (see 5.3.9);

Step 7: Calculation of auxiliary energy input (see 5.3.10);

Step 8: Calculation of generation subsystem thermal loss recoverable for space heating (see 5.3.11);

Step 9: Calculation of the total driving energy input to cover the requirements (see 5.3.12);

Step 10: Summary of resulting and optional output values (see 5.3.13).

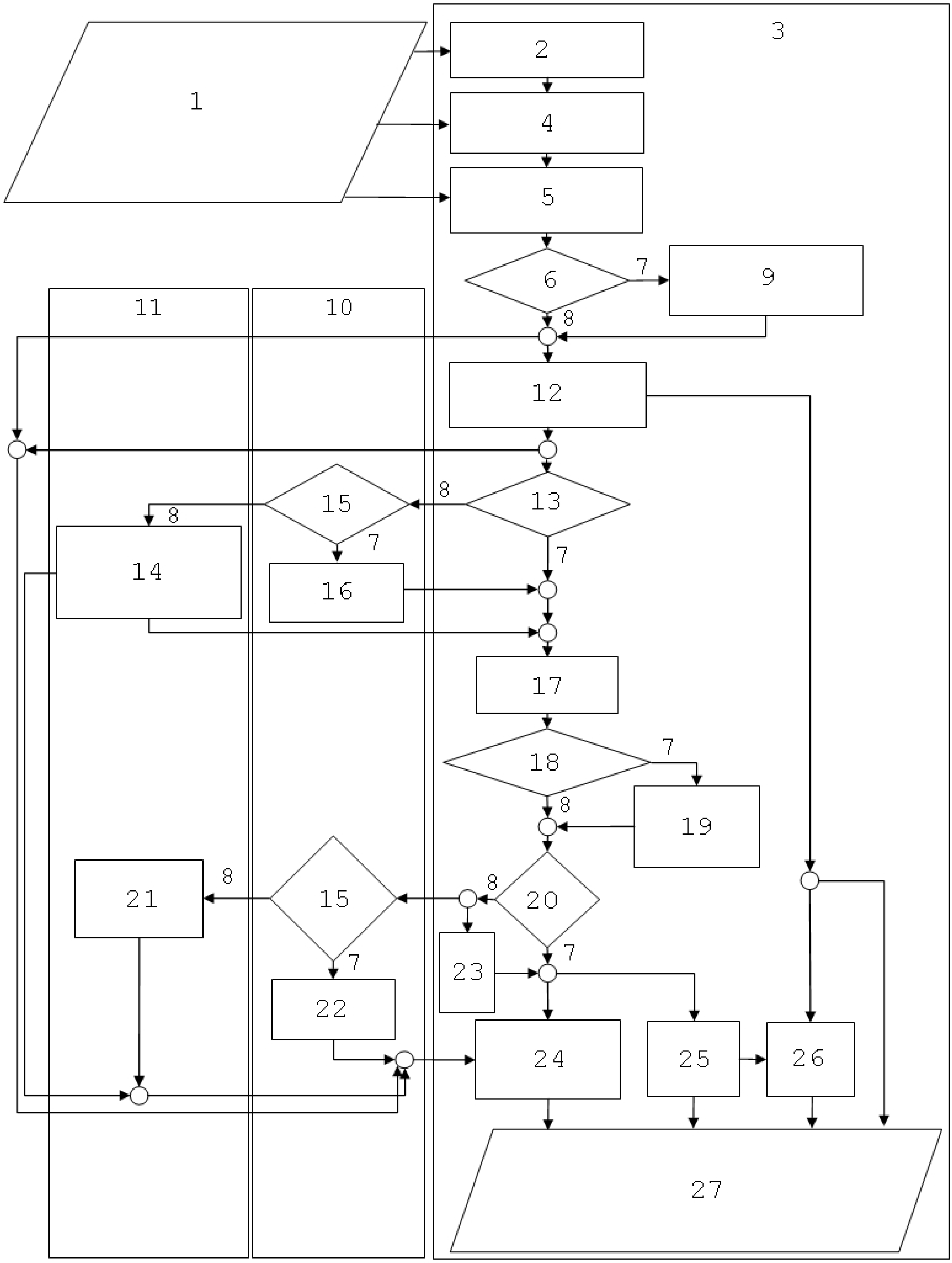


Figure 6 — Flow chart of the bin calculation method

Key

1 ***input data**, **see 5.3.2***
- energy requirement of space heating distribution subsystem
- energy requirement of the domestic hot water distribution subsystem
- meteo data
- product characteristics
- design parameters

2 definitions of calculation periods and bins
see 5.3.1

3 monovalent

4 calculation of energy requirements/bin time
see 5.3.4

5 correction of heating capacity/*COP* for operating conditions
see 5.3.5

6 part load test available

7 yes

8 no

9 correction of *COP* for part load operation
see 5.3.6

10 bivalent detailed

11 bivalent simplified

12 calculation of generation subsystem thermal losses
see 5.3.7

13 monovalent system

14 evaluation of back-up energy due to operation limit and based on balance point
see 5.3.8.1/2

15 detailed back-up energy calculation

16 calculation of back-up energy for operation limit
see 5.3.8.1/2

17 calculation of running time
see 5.3.9.1

18 simultaneous system with 3 operation modes

19 calculation of running time for simultaneous system
see 5.3.9.2

20 running time < eff. bin time

21 calculation of additional back-up energy
see 5.3.8.3

22 calculation of back-up energy
see 5.3.9.4/5

23 running time = eff. bin time

24 calculation of energy input to cover the heat requirement
see 5.3.12

25 calculation of auxiliary energy
see 5.3.10

26 calculation of thermal loss recoverable for space heating
see 5.3.11

27 ***output data: see 5.3.13***
- energy input to cover the heat requirement
- generation subsystem total thermal losses
- generation subsystem thermal loss recoverable for space heating
- total auxiliary energy input

5.3.4 Heat energy requirements for space heating and domestic hot water mode for the bins

5.3.4.1 Space heating mode

The heat energy requirement of the space heating distribution subsystem $Q_{H,dis,in}$ shall be calculated according to EN 15316-2-3.

The space heating requirement of bin j 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 (DH_H). 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

$$k_{H,j} = \frac{Q_{H,gen,out,j}}{Q_{H,gen,out}} = \frac{DH_{H,\theta hlim,j} - DH_{H,\theta llim,j}}{DH_{H,tot}} \quad [-] \quad (4)$$

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

$$Q_{H,gen,out,j} = k_{H,j} \cdot Q_{H,gen,out} \quad [J] \quad (5)$$

where

$k_{H,j}$	is the weighting factor of the heat pump operation for space heating of bin j;	(-)
$Q_{H,gen,out,j}$	is the heat energy requirement of the space heating distribution subsystem in bin j;	(J)
$Q_{H,gen,out}$	is the total heat energy requirement of the space heating distribution subsystem;	(J)
$DH_{H,\theta hlim,i}$	is the cumulative heating degree hours up to the upper temperature limit of bin j;	(°Ch)
$DH_{H,\theta llim,i}$	is the cumulative heating degree hours up to the lower temperature limit of bin j;	(°Ch)
$DH_{H,tot}$	is the total cumulative heating degree hours up to the upper temperature limit for space heating.	(°Ch)

The cumulative heating degree hours for the respective climatic regions shall be given in a national annex or taken from national standardisation. It is also possible to define weighting factors for a fixed bin scheme and standard locations in a national annex.

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

$$t_j = (N_{ho,\theta hlim,j} - N_{ho,\theta llim,j}) \cdot 3600 \quad [s] \quad (6)$$

where

t_j	is the time in bin j;	(s)
$N_{ho,\theta hlim,j}$	is the cumulative number of hours up to the upper temperature limit of bin j;	(h)
$N_{ho,\theta llim,j}$	is the cumulative number of hours up to the lower temperature limit of bin j.	(h)

A summation of all bin times t_i for space heating constitutes the heating season. Attention should be paid to national regulation on the heating season.

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 according to Equation (6) reduced by the cut-out time per day and is calculated by

$$t_{\mathrm{eff,j}} = t_{\mathrm{j}} \cdot \frac{24 - t_{\mathrm{co}}}{24} \qquad [s] \qquad (7)$$

where

$t_{\mathrm{eff,j}}$	is the effective bin time in bin j;	(s)
t_{j}	is the time in bin j;	(s)
t_{co}	is the cut-out hours per 24 hours (1 day).	(h/d)

5.3.4.2 Domestic hot water mode

The heat energy requirement of the domestic hot water distribution subsystem $Q_{\mathrm{W,dis,in}}$ shall calculated according to EN 15316-3-2.

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

$$k_{\mathrm{W,j}} = \frac{Q_{\mathrm{W,gen,out,j}}}{Q_{\mathrm{W,gen,out}}} = \frac{t_{\mathrm{j}}}{t_{\mathrm{tot}}} \qquad [-] \qquad (8)$$

and the domestic hot water energy requirement of bin j follows according to the equation

$$Q_{\mathrm{W,gen,out,j}} = k_{\mathrm{W,j}} \cdot Q_{\mathrm{W,gen,out}} \qquad [J] \qquad (9)$$

where

$k_{\mathrm{W,j}}$	is the weighting factor of the heat pump operation for DHW operation of bin j;	(-)
$Q_{\mathrm{W,gen,out,j}}$	is the heat energy requirement of the domestic hot water distribution subsystem in bin j;	(J)
$Q_{\mathrm{W,gen,out}}$	is the total heat energy requirement of the DHW distribution subsystem;	(J)
t_{j}	is the bin time in bin j;	(s)
t_{tot}	is the total time of DHW operation (e.g. year round operation).	(s)

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

5.3.5 Heating capacity and *COP* at full load

5.3.5.1 Space heating mode

5.3.5.1.1 General

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 shall 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 5.3.5.1.2.

To determine data for the whole range of 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, which is not possible in case of one test point. However, good results are only obtained near the test point.

Some example values for the illustration of the data needed to accomplish the calculation for electrically-driven heat pumps are given in F.3.

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.

5.3.5.1.2 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 subsystem, has to be taken into account. Temperature spread and mass flow rate are linked by the equation

$$\Delta\theta = \frac{\phi_{\mathrm{hp}}}{m'_{\mathrm{w}} \cdot c_{\mathrm{w}}} \qquad [\mathrm{K}] \qquad (10)$$

where

$\Delta\theta$ is the temperature spread on the condenser side of the heat pump; (K)

ϕ_{hp} is the heating capacity of the heat pump; (W)

m'_{w} is the mass flow rate of the heat transfer medium on the condenser side of the heat pump; (kg/s)

c_{w} is the heat capacity of heat transfer medium. (J/(kg·K))

In case of 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 Equation (10). 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 condenser during operation is different from during the testing and therefore *COP* values have to be corrected. The correction can be derived based on the method of fixed exergetic efficiency given in Annex C according to the equation

$$COP_{\Delta\theta} = COP_{\text{standard}} \cdot \left[1 - \frac{\dfrac{\Delta\theta_{\text{standard}} - \Delta\theta_{\text{opr}}}{2}}{\left\{ T_{\text{sk}} - \dfrac{\Delta\theta_{\text{standard}}}{2} + \Delta T_{\text{sk}} - \left(T_{\text{sc}} - \Delta T_{\text{sc}} \right) \right\}} \right] \quad \text{[W/W]} \qquad (11)$$

where

$COP_{\Delta\theta}$	is the *COP* corrected for a different temperature spread in testing and operation;	(W/W)
COP_{standard}	is the *COP* derived from standard testing (e.g. according to EN 14511);	(W/W)
$\Delta\theta_{\text{standard}}$	is the temperature spread on the condenser side due to standard test conditions;	(K)
$\Delta\theta_{\text{opr}}$	is the temperature spread on the condenser side in operation due to the design of the heat emission subsystem;	(K)
T_{sk}	is the sink temperature;	(K)
ΔT_{sk}	is the average temperature difference between heat transfer medium and refrigerant in condenser;	(K)
T_{sc}	is the source temperature;	(K)
ΔT_{sc}	is the average temperature difference between heat transfer medium and refrigerant in evaporator.	(K)

The average temperature difference in the condenser and evaporator between the heat transfer medium and the refrigerant can be approximated by $\Delta T_{\text{sk}} = \Delta T_{\text{sc}} = 4$ K for water based components. In the case of air based components $\Delta T_{\text{sk}} = \Delta T_{\text{sc}} = 15$ K is to be 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 Equation (11) 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.

5.3.5.1.3 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 <u>source temperature</u> 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 or the ground water inlet temperature 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 F.2.1.3 and an example temperature for ground water is given in F.2.1.4;

— 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, e.g. according to EN 308, shall be applied.

The actual <u>sink temperature</u> can be calculated according to:

— either 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 are not known, typical controller settings for the heating curve for different kinds of emission subsystems are given in B.1.

5.3.5.2 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 EN 255-3. This standard testing determines 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.

The COP_t 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 determines 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 pump operation, the operation period and thus *COP* values may 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. However, the method shall only be applied near the test point.

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 domestic hot water temperature calculated according to the equation

$$\theta_{\mathrm{W,avg}} = f_{\mathrm{W,st}} \cdot \theta_{\mathrm{hp,opr}} \qquad [°C] \qquad (12)$$

where

$\theta_{\mathrm{W,avg}}$	is the average hot water loading temperature;	(°C)
$f_{\mathrm{W,st}}$	is the correction factor for storage loading temperature;	(-)
$\theta_{\mathrm{hp,opr}}$	is the 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 $f_{\mathrm{W,st}}$ takes into account that the loading starts at lower temperatures than the maximum hot water temperature, which can be reached by the heat pump operation (see also 5.3.8.2), due to colder water at the storage heat exchanger. The hot water loading temperature 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, which can be reached by the heat pump operation. Values of $f_{\mathrm{W,st}}$ or the average loading temperature $\theta_{\mathrm{W,avg}}$ shall be given in a national annex. If no national annex is available, a default value is given in B.2.

5.3.5.3 Engine-driven and absorption heat pumps

Steady state heating capacity and *COP* are taken from test results. The same considerations concerning correction according to temperature conditions as described in 5.3.5.1 and 5.3.5.2 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:

— F.4.2.1 and F.4.3.2 for air-to-water gas engine driven heat pumps;

— F.4.2.2 and F.4.3.3 for air-to-air gas engine driven heat pumps;

— F.5.2 for air to water NH_3/H_2O gas absorption heat pumps;

— F.5.3 for NH_3/H_2O brine-to-water absorption heat pumps;

— F.5.4 for NH_3/H_2O water-to-water absorption heat pumps;

— F.5.5 for H_2O/LiBr water-to-water double effect gas absorption heat pumps.

5.3.6 *COP* at part load operation

5.3.6.1 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 (or the burner for absorption heat pumps) occur and may 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 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 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 stand-by auxiliary energy 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 energy calculated in 5.3.10 shall 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 EN 15316-1 [9]. It is calculated according to the equation

$$\beta_{\mathrm{j}} = \frac{Q_{\mathrm{HW,gen,out,j}}}{\phi_{\mathrm{hp,j}} \cdot t_{\mathrm{eff,j}}} \qquad [-] \qquad (13)$$

where

β_{j}	is the load factor in bin j;	(-)
$Q_{\mathrm{HW,gen,out,j}}$	is the heat energy requirement of the distribution subsystem in bin j;	(J)
$t_{\mathrm{eff,j}}$	is the effective bin time in bin j;	(s)
$\phi_{\mathrm{hp,j}}$	is the heating capacity of the heat pump in bin j.	(W)

To accomplish the interpolation, at least one part load test point has to be available, e.g. the $COP_{50\,\%}$ according to CEN/TS 14825. Then, the interpolation can be performed between the full load *COP* and the part load *COP* as described in 5.3.5.1.3.

5.3.6.2 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 value according to EN 255-3 due to the system testing.

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

5.3.7 Thermal losses through the generator envelope

5.3.7.1.1 Space heating mode

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

For engine-driven heat pumps, the thermal losses of the engine are considered. They shall be evaluated based on test results or manufacturer data. If no values are available, the thermal losses can be estimated by the efficiency of the engine and a possible fraction of recovered heat as for the CHP systems treated in EN 15316-4-4 [12]. For the redistribution of the thermal 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 installed internal or external heating buffer storage causes thermal losses to the ambiance that can be calculated by a stand-by thermal loss value for the bin j:

$$Q_{\mathrm{H,st,ls,j}} = \frac{\theta_{\mathrm{H,st,avg,j}} - \theta_{\mathrm{H,st,amb,j}}}{\Delta\theta_{\mathrm{st,sby}}} \cdot \frac{Q_{\mathrm{st,sby}} \cdot 1000 \cdot t_{\mathrm{j}}}{24} \qquad \text{[J]} \qquad (14)$$

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

$$Q_{\mathrm{H,st,ls,tot}} = \sum_{j=1}^{N_{bins}} Q_{\mathrm{H,st,ls,j}} \qquad \text{[J]} \qquad (15)$$

where

$Q_{\mathrm{H,st,ls,j}}$	is the thermal loss to the ambiance of the heating buffer storage in bin j;	(J)
$\theta_{\mathrm{H,st,avg,j}}$	is the average storage temperature of the heating buffer storage in bin j;	(°C)
$\theta_{\mathrm{H,st,amb,j}}$	is the ambient temperature at the storage location;	(°C)
$\Delta\theta_{\mathrm{st,sby}}$	is the temperature difference due to storage stand-by test conditions;	(K)
$Q_{\mathrm{st,sby}}$	is the stand-by heat loss due to storage stand-by test conditions;	(kWh/d)
t_{j}	is the bin time in bin j;	(s)
$Q_{\mathrm{H,st,ls,tot}}$	is the total storage thermal losses to the ambiance of the heating buffer storage;	(J)
N_{bins}	is the number of bins.	(-)

If the stand-by heat loss from the storage at test conditions is not available, default values are given in B.4.

The average storage temperature $\theta_{H,st,avg,j}$ is to be determined according to the storage control. If the storage is operated dependent on the temperature requirements of the heating system, it is approximated as average temperature of the flow- and return temperature of the space heating system, according to the equation

$$\theta_{H,st,avg,j} = \frac{\theta_{H,gen,f,j} - \theta_{H,dis,r,j}}{2} \quad [°C] \quad (16)$$

where

$\theta_{H,st,avg,j}$	is the average storage temperature of the heating buffer storage in bin j;	(°C)
$\theta_{H,gen,f,j}$	is the flow temperature of the space heating generation system in bin j;	(°C)
$\theta_{H,dis,r,j}$	is the return temperature from the space heating distribution system in bin j.	(°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.

5.3.7.1.2 Domestic hot water mode

If data from storage testing are known, the calculation of the storage thermal losses of the domestic hot water storage shall be accomplished as for space heating buffer storages according to Equations (14) and (15). The average storage temperature depends on the applied storage control, the position of the heat exchangers and the temperature sensors, etc. and shall be determined based on the product information. If no information is available, default values of the average domestic hot water storage temperature are given in B.3.

If no values on storage testing of stand-by heat losses are available, the calculation shall be carried out according to Equation (14) and (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 B.4.

5.3.7.1.3 Thermal losses of primary circulation pipes

Thermal losses of the primary circulation pipes between the heat generator and the storage vessel shall be calculated according to the method given for the calculation of thermal losses of pipes in EN 15316-2-3 or in EN 15316-3-3 and are added to the storage thermal losses.

5.3.8 Calculation of back-up heater

5.3.8.1 General

Back-up energy may 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 5.3.8.2. The other reason is in case of a multivalent design of the generator subsystem (see boundary conditions in 4.6), i.e. the heat pump is not designed for the total load. In this case, a fraction of back-up energy is required due to lack of heating capacity of the heat pump. For the calculation of the back-up operation due to lack of heating capacity of the heat pump, a simplified method 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 5.3.8.3. The method assumes that the balance point is known and all influencing factors, e.g. power demand for space heating and domestic hot water 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. This method should be applied if the balance point is not known or is difficult to calculate, e.g. for 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 5.3.9.4 in connection with evaluation of the running time.

5.3.8.2 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 this temperature limit are required, they cannot be produced by the heat pump alone, and reheating by a back-up heater to boost the temperature is required. Therefore, the fraction of back-up energy due to the operation limit of the heat pump can be calculated by

$$k_{\text{bu,opr,j}} = \frac{Q_{\text{bu,opr,j}}}{Q_{\text{gen,out,j}}} = \frac{m'_{\text{w}} \cdot c_{\text{w}} \cdot (\theta_{\text{n,j}} - \theta_{\text{hp,op}}) \cdot t_{\text{hp,on,j}}}{Q_{\text{gen,out,j}}} \quad [-] \qquad (17)$$

where

$k_{\text{bu,opr,j}}$	is the fraction of back-up energy due to the operation limit of the heat pump in bin j;	(-)
$Q_{\text{bu,opr,,j}}$	is the back-up heat energy due to the operation limit of the heat pump in bin j;	(J)
$Q_{\text{gen,out,j}}$	is the heat energy requirement of the distribution subsystem in bin j;	(J)
m'_{w}	is the mass flow rate of the heat transfer medium;	(kg/s)
c_{w}	is the specific heat capacity of the heat transfer medium;	(J/(kg·K))
$\theta_{\text{n,j}}$	is the nominal temperature requirement of the system in bin j;	(°C)
$\theta_{\text{hp,op}}$	is the operation limit temperature of the heat pump (maximum temperature that can be reached with the heat pump operation);	(°C)
$t_{\text{hp,on,j}}$	is the running time of the heat pump in bin j.	(s)

For space heating operation, the fraction $k_{\text{H,bu,opr,j}}$ usually does not occur, i.e. $k_{\text{H,bu,opr,j}} = 0$, since the design of the heat emission subsystem is usually adapted to required temperature levels below the operation limit of the heat pump.

For domestic hot water 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:

$$k_{\text{W,bu,opr,j}} = \frac{Q_{\text{W,bu,opr,j}}}{Q_{\text{W,gen,out,j}}} = \frac{\rho_{\text{w}} \cdot V_{\text{w,j}} \cdot c_{\text{w}} \cdot (\theta_{\text{w,out}} - \theta_{\text{hp,opr}})}{\rho_{\text{w}} \cdot V_{\text{w,j}} \cdot c_{\text{w}} \cdot (\theta_{\text{w,out}} - \theta_{\text{w,in}})} = \frac{(\theta_{\text{w,out}} - \theta_{\text{hp,opr}})}{(\theta_{\text{w,out}} - \theta_{\text{w,in}})} \quad [-] \qquad (18)$$

where

$k_{\text{W,bu,opr,j}}$	is the fraction of back-up energy in DHW mode due to the operation limit of the heat pump in bin j;	(-)
$Q_{\text{W,bu,opr,j}}$	is the back-up heat energy for DHW due to the operation limit of the heat pump in bin j;	(J)
$Q_{\text{W,gen,out,j}}$	is the heat energy requirement of the domestic hot water subsystem in bin j;	(J)

ρ_w	is the density of the heat transfer medium;	(kg/m^3)
$V_{w,j}$	is the volume of the hot water draw-off in bin j;	(m^3/s)
c_w	is the specific heat capacity of water;	(J/(kg·K))
$\theta_{W,out}$	is the temperature of the hot water at storage outlet;	(°C)
$\theta_{hp,opr}$	is the operation limit temperature of the heat pump (maximum temperature that can be reached with the heat pump operation);	(°C)
$\theta_{W,in}$	is the temperature of the cold water inlet.	(°C)

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

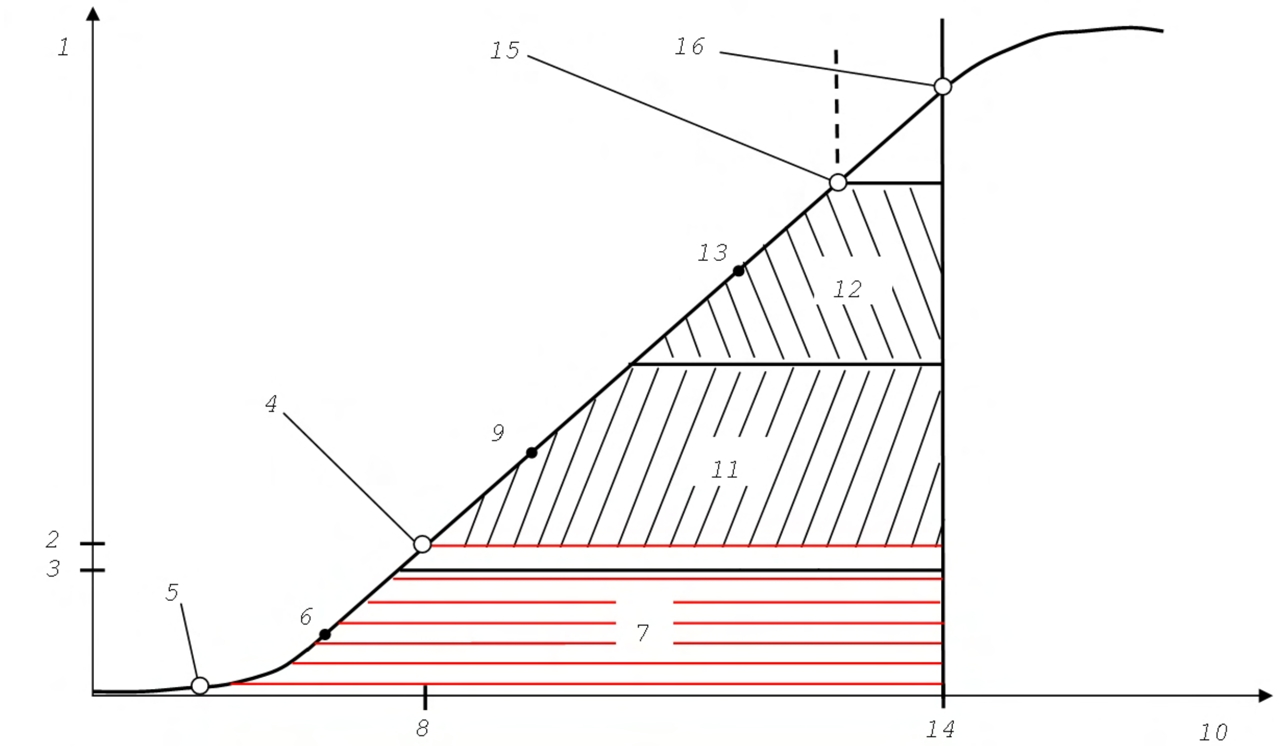
5.3.8.3 Simplified back-up calculation: Back-up energy due to lack of heating capacity of the heat pump

5.3.8.3.1 General

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 Table A.1.

5.3.8.3.2 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.



Key

1 cumulative bin hours

2 $N_{ho,\theta bal}$

3 $N_{ho,\theta hlim,j}$

4 balance point temperature

5 design outdoor temperature

6 OP_1

7 BU

8 θ_{bal}

9 OP_2

10 outdoor air temperature

11 HP_1

12 HP_2

13 OP_3

14 $\theta_{i,des}$

15 upper ambient temperature for heating

16 design indoor temperature

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

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

For $\theta_{bal} > \theta_{hlim,j}$

$$k_{H,bu,cap,j} = \frac{A_{bu,j}}{A_j} = \frac{DH_{H,\theta hlim,j}}{DH_{H,\theta hlim,j} - DH_{H,\theta llim,j}} = \frac{DH_{H,\theta hlim,j}}{DH_{H,\theta hlim,j} - 0} = 1 \quad [-] \quad (19)$$

$$k_{H,bu,cap,j+1} = \frac{A_{bu,j+1}}{A_{j+1}} = \frac{DH_{H,\theta bal} - DH_{H,\theta llim,j+1}}{DH_{H,\theta hlim,j+1} - DH_{H,\theta llim,j+1}} \quad [-] \quad (20)$$

In case of a balance point below the bin limit, the fraction of the back-up heater for alternate operation in the lowest bin j is calculated.

For $\theta_{bal} < \theta_{hlim,j}$

$$k_{H,bu,cap,j} = \frac{A_{bu,j}}{A_j} = \frac{DH_{H,\theta bal}}{DH_{H,\theta hlim,j}} \quad [-] \quad (21)$$

where

$k_{H,bu,cap,j}$ is the fraction of space heating heat energy covered by the back-up heater in the lower bin j; (-)

$k_{H,bu,cap,,j+1}$ is the fraction of space heating heat energy covered by the back-up heater in the subsequent bin j + 1; (-)

$A_{bu,j}$ is the fraction of the total area BU in Figure 7 in bin j; (°Ch)

$A_{bu,j+1}$ is the fraction of the total area BU in Figure 7 in bin j + 1; (°Ch)

A_j is the total area of bin j (between upper and lower temperature limit of bin j); (°Ch)

A_{j+1} is the total area of the subsequent bin j + 1; (°Ch)

θ_{bal} is the balance point temperature; (°C)

$\theta_{llim,j}$ is the lower temperature limit of bin j; (°C)

$\theta_{hlim,j}$ is the upper temperature limit of bin j; (°C)

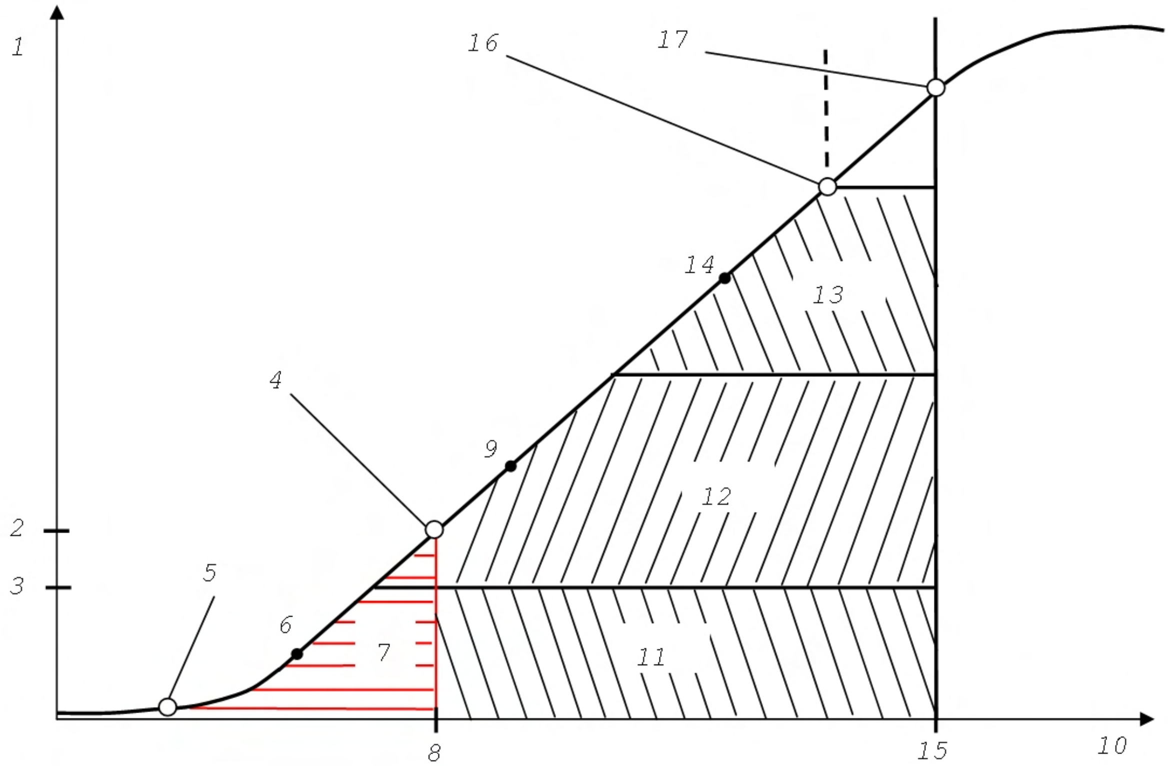
$DH_{H,\theta bal}$ is the cumulative heating degree hours up to the balance point temperature $\theta_{bal;}$ (°Ch)

$DH_{H,\theta llim,j}$ is the cumulative heating degree hours up to lower temperature limit of bin j $\theta_{llim,j;}$ (°Ch)

$DH_{H,\theta hlim,j}$ is the cumulative heating degree hours up to upper temperature limit of bin j $\theta_{hlim,j.}$ (°Ch)

5.3.8.3.3 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 of the energy requirement that the heat pump cannot deliver.



Key

1 cumulative bin hours [h]

2 $N_{ho,\theta bal}$

3 $N_{ho,\theta hlim,j}$

4 balance point temperature

5 design outdoor temperature

6 OP_1

7 BU

8 θ_{bal}

9 OP_2

10 outdoor air temperature [°C]

11 HP_1

12 HP_2

13 HP_3

14 OP_3

15 $\theta_{i,des}$

16 upper ambient temperature for heating

17 design indoor temperature

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

Figure 8 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. If the balance point transcends the bin limit as in Figure 8, the fractions of the back-up heater for parallel operation in the lowest bin j and the subsequent bin j + 1 are approximated.

For $\theta_{bal} > \theta_{hlim,j}$

$$k_{H,bu,cap,j} = \frac{A_{bu,j}}{A_j} = \frac{DH_{H,\theta hlim,j} - (\theta_{i,des} - \theta_{bal}) \cdot N_{ho,\theta hlim,j}}{DH_{H,\theta hlim,j}} \quad [-] \qquad (22)$$

$$k_{H,bu,cap,j+1} = \frac{A_{bu,j+1}}{A_{j+1}} = \frac{(DH_{H,\theta bal} - DH_{H,\theta hlim,j}) - (\theta_{i,des} - \theta_{bal}) \cdot (N_{ho,\theta bal} - N_{ho,\theta hlim,j})}{DH_{H,\theta hlim,j+1} - DH_{H,\theta llim,j+1}} \quad [-] \qquad (23)$$

In case of a balance point below the bin limit, the fraction of the back-up heater for parallel operation in the lowest bin j is approximated.

For $\theta_{bal} < \theta_{hlim,j}$

$$k_{H,bu,cap,j} = \frac{A_{bu,j}}{A_j} = \frac{DH_{H,\theta bal} - (\theta_{i,des} - \theta_{bal}) \cdot N_{ho,\theta bal}}{DH_{H,\theta hlim,j}} \quad [-] \qquad (24)$$

where

$k_{H,bu,cap,j}$ is the fraction of space heating heat energy covered by the back-up heater in the lower bin j; (-)

$k_{H,bu,cap,j+1}$ is the fraction of space heating heat energy covered by the back-up heater in the subsequent bin j + 1; (-)

$A_{bu,j}$ is the fraction of total area BU in Figure 8 in bin j; (°Ch)

$A_{bu,j+1}$ is the fraction of total area BU in Figure 8 in bin j + 1; (°Ch)

A_j is the total area of bin j (between upper and lower temperature limit of bin j); (°Ch)

A_{j+1} is the total area of bin j + 1; (°Ch)

θ_{bal} is the balance point temperature; (°C)

$\theta_{i,des}$ is the indoor design temperature; (°C)

$N_{ho,\theta bal}$ is the cumulative number of hours up to the balance point temperature; (h)

$N_{ho,\theta hlim,j}$ is the cumulative number of hours up to the upper temperature limit of bin j; (h)

$DH_{H,\theta bal}$ is the cumulative heating degree hours up to the balance point temperature $\theta_{bal;}$ (°Ch)

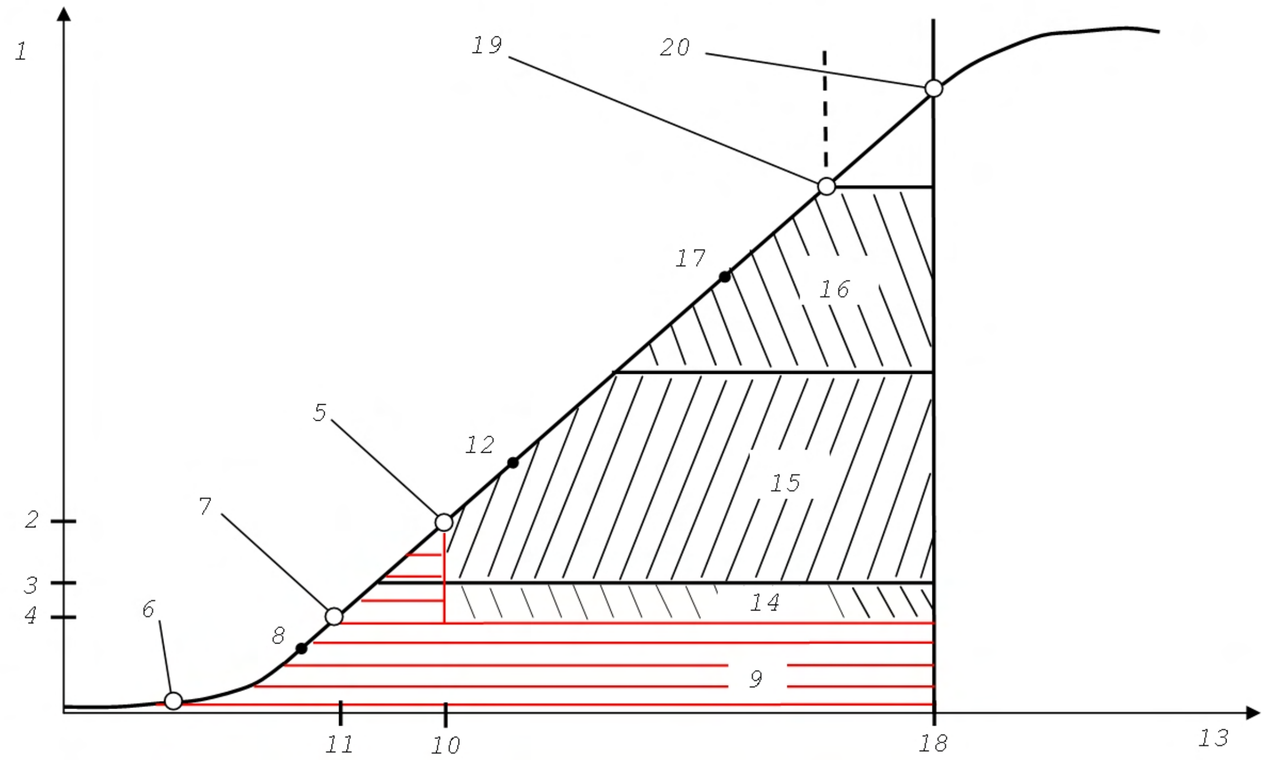
$DH_{H,\theta llim,j}$ is the cumulative heating degree hours up to the lower temperature limit of bin j $\theta_{llim,j;}$ (°Ch)

$DH_{H,\theta hlim,j}$ is the cumulative heating degree hours up to the upper temperature limit of bin j $\theta_{hlim,j.}$ (°Ch)

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 5.3.9.3 indicates whether the approximation is sufficiently exact or a correction is required according to Equation (40).

5.3.8.3.4 Partly parallel operation mode of the back-up heater

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 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.



Key

1 cumulative bin hours [h]
2 $N_{ho,\theta bal}$
3 $N_{ho,\theta hlim,j}$
4 $N_{ho,ltc}$
5 balance point temperature
6 design outdoor temperature
7 low temperature cut-out
8 OP_1
9 BU
10 θ_{bal}
11 θ_{tc}
12 OP_2
13 outdoor air temperature [°C]
14 HP_1
15 HP_2
16 HP_3
17 OP_3
18 $\theta_{i,des}$
19 upper ambient temperature for heating
20 design indoor temperature

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

If the balance point transcends the bin limit as in Figure 9, the fractions of the back-up heater for partly parallel operation in the lowest bin j and the subsequent bin j + 1 are approximated.

For $\theta_{bal} > \theta_{hlim,j}$

$$k_{H,bu,cap,j} = \frac{A_{bu,j}}{A_j} = \frac{DH_{H,\theta hlim,j} - (\theta_{i,des} - \theta_{\theta bal}) \cdot (N_{ho,\theta hlim,j} - N_{ho,\theta ltc})}{DH_{H,\theta hlim,j}} \quad [-] \qquad (25)$$

$$k_{H,bu,cap,j+1} = \frac{A_{bu,j+1}}{A_{j+1}} = \frac{(DH_{H,\theta bal} - DH_{H,\theta hlim,j}) - (\theta_{i,des} - \theta_{\theta bal}) \cdot (N_{ho,\theta bal} - N_{ho,\theta hlim,j})}{DH_{H,\theta hlim,j+1} - DH_{H,\theta llim,j+1}} \quad [-] \qquad (26)$$

In case of a balance point below the bin limit, the fraction of the back-up heater for partly parallel operation in the lowest bin j is approximated.

For $\theta_{bal} < \theta_{hlim,j}$

$$k_{H,bu,cap,j} = \frac{A_{bu,j}}{A_j} = \frac{DH_{H,\theta bal} - (\theta_{i,des} - \theta_{\theta bal}) \cdot (N_{ho,\theta bal} - N_{ho,\theta ltc})}{DH_{H,\theta hlim,j}} \quad [-] \quad (27)$$

where

$k_{H,bu,cap,j}$	is the fraction of space heating heat energy covered by the back-up heater in lower bin j;	(-)
$k_{H,bu,cap,j+1}$	is the fraction of space heating heat energy covered by the back-up heater in subsequent bin j + 1;	(-)
$A_{bu,j}$	is the fraction of total area BU in Figure 9 in bin j;	(°Ch)
$A_{bu,j+1}$	is the fraction of total area BU in Figure 9 in bin j + 1;	(°Ch)
A_j	is the total area of bin j (between upper and lower temperature limit of bin j);	(°Ch)
A_{j+1}	is the total area of bin j + 1;	(°Ch)
θ_{bal}	is the balance point temperature;	(°C)
$\theta_{i,des}$	is the indoor design temperature;	(°C)
θ_{ltc}	is the low-temperature cut-out temperature;	(°C)
$N_{ho,\theta bal}$	is the cumulative number of hours up to the balance point temperature;	(h)
$N_{ho,\theta hlim,j}$	is the cumulative number of hours up to the upper temperature limit of bin j;	(h)
$N_{ho,\theta ltc}$	is the cumulative number of hours up to the low-temperature cut-out;	(h)
$DH_{H,\theta bal}$	is the cumulative heating degree hours up to the balance point temperature θ_{bal};	(°Ch)
$DH_{H,\theta llim,j}$	is the cumulative heating degree hours up to the lower temperature limit of bin j $\theta_{llim,j}$;	(°Ch)
$DH_{H,\theta hlim,j}$	is the cumulative heating degree hours up to the upper temperature limit of bin j $\theta_{hlim,j}$.	(°Ch)

5.3.9 Running time of the heat pump

5.3.9.1 General

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

$$t_{hp,on,j} = \frac{Q_{hp,j}}{\phi_{hp,j}} \quad [s] \quad (28)$$

where

$t_{hp,on,j}$	is the running time of the heat pump in bin j;	(s)
$Q_{hp,j}$	is the produced heat energy by the heat pump in bin j	(J)
	(energy requirement of the distribution subsystem and generation subsystem thermal losses);	

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

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

$$Q_{hp,j} = (Q_{gen,out,j} + Q_{gen,ls,j})(1 - k_{bu,cap,j}) \quad [J] \qquad (29)$$

where

$Q_{hp,j}$ is the produced heat energy by the heat pump in bin j (J)

(energy requirement of the distribution subsystem and generation subsystem thermal losses);

$Q_{gen,out,j}$ is the heat energy requirement of the distribution subsystem in bin j; (J)

$Q_{gen,ls,j}$ is the generation subsystem thermal losses in bin j; (J)

$k_{bu,cap,j}$ is the fraction of heat energy covered by the back-up heater in bin j. (-)

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 $k_{H,bu,cap,j}$ and the fraction for domestic hot water operation due to the temperature operation limit of the heat pump $k_{W,bu,opr,j}$ are known from the calculation in 5.3.8 and are to be applied in Equation (29).

If the detailed calculation of the back-up energy is applied, only the fraction of back-up energy for domestic hot water operation due to the operation limit temperature of the heat pump has to be taken into account, since the fraction of back-up energy due to lack of heating capacity of the heat pump follows from the energy balance (see 5.3.9.4). That means only the fractions $k_{bu,opr,j}$ 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 heat pump 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 season. The running time is evaluated based on these two characteristics.

Additional calculations of the energy fractions and the running times for systems where all three operation modes occur are given in 5.3.9.2.

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

$$t_{\mathrm{hp,on,tot,j}} = t_{\mathrm{H,hp,on,sngl,j}} + t_{\mathrm{W,hp,on,sngl,j}} + t_{\mathrm{HW,hp,on,combi,j}} \quad \text{[s]} \qquad (30)$$

where

$t_{\mathrm{hp,on,tot,j}}$	is the total running time of the heat pump in bin j;	(s)
$t_{\mathrm{H,hp,on,sngl,j}}$	is the running time in space heating-only operation in bin j;	(s)
$t_{\mathrm{W,hp,on,sngl,j}}$	is the running time in DHW-only operation in bin j;	(s)
$t_{\mathrm{HW,hp,on,combi,j}}$	is the running time in simultaneous operation in bin j.	(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.

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

Principle

For the system depicted in Figure 3, for instance, all three operation modes occur, so the running time of 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 further controller impact into account.

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 a 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 net 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

$$t_{\mathrm{HW,hp,on,combi,max,j}} = \min(t_{\mathrm{H,hp,on,j}}, t_{\mathrm{W,hp,on,j}}) \quad \text{[s]} \qquad (31)$$

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

$$t_{\mathrm{W,hp,on,j}} = \frac{Q_{\mathrm{W,hp,j}}}{\phi_{\mathrm{W,hp,combi,j}}} \quad [s] \quad (32)$$

and analogously for space heating operation

$$t_{\mathrm{H,hp,on,j}} = \frac{Q_{\mathrm{H,hp,j}}}{\phi_{\mathrm{H,hp,combi,j}}} \quad [s] \quad (33)$$

where

$t_{\mathrm{HW,hp,on,combi,max,j}}$ is the maximum possible running time in simultaneous operation in bin j; (s)

$t_{\mathrm{W,hp,on,j}}$ is the running time in DHW operation in bin j; (s)

$Q_{\mathrm{W,hp,j}}$ is the produced heat energy by the heat pump for DHW in bin j; (J)

$\phi_{\mathrm{W,hp,combi,j}}$ is the DHW heating capacity of the heat pump in simultaneous operation in bin j; (W)

$t_{\mathrm{H,hp,on,j}}$ is the running time in space heating operation in bin j; (s)

$Q_{\mathrm{H,hp,j}}$ is the produced heat energy by the heat pump for space heating in bin j; (J)

$\phi_{\mathrm{H,hp,combi,j}}$ is the space heating heating capacity of the heat pump in simultaneous operation in bin j. (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

$$t_{\mathrm{HW,hp,on,combi,j}} = f_{\mathrm{combi}} \cdot t_{\mathrm{HW,hp,on,combi,max,j}} \quad [s] \quad (34)$$

where

$t_{\mathrm{HW,hp,on,combi,j}}$ is the running time in simultaneous operation in bin j; (s)

$t_{\mathrm{HW,hp,on,combi,max,j}}$ is the maximum possible running time in simultaneous operation in bin j; (s)

f_{combi} is the 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 national values are given, a default value is given in B.6.

The energy produced in simultaneous operation for domestic hot water and space heating, respectively, is calculated by the equation

$$Q_{\mathrm{hp,combi,j}} = \phi_{\mathrm{hp,combi,j}} \cdot t_{\mathrm{hp,on,combi,j}} \quad [J] \quad (35)$$

where

$Q_{\mathrm{hp,combi,j}}$ is the produced heat energy in simultaneous operation of the respective operation mode in bin j; (J)

$\phi_{\mathrm{hp,combi,j}}$ is the heat pump heating capacity in simultaneous operation of the respective operation mode in bin j; (W)

$t_{\mathrm{hp,on,combi,j}}$ is the running time in simultaneous operation in bin j. (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

$$Q_{\text{hp,sngl,j}} = Q_{\text{hp,j}} - Q_{\text{hp,combi,j}} \qquad [J] \qquad (36)$$

where

$Q_{\text{hp,sngl,j}}$	is the produced heat energy by the heat pump in the respective single operation in bin j;	(J)
$Q_{\text{hp,j}}$	is the produced heat energy by the heat pump in bin j;	(J)
$Q_{\text{hp,combi,j}}$	is the produced heat energy by the heat pump in simultaneous operation in bin j.	(J)

Since EN 255-3 gives the electricity to cover the heat losses of the DHW storage in form of an 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 operation can be calculated by subtracting the storage losses according to the equation

$$Q_{\text{W,hp,out,sngl,j}} = Q_{\text{W,hp,sngl,j}} - Q_{\text{W,st,ls,j}} \cdot (1 - k_{\text{W,bu,j}}) \cdot (1 - f_{\text{combi}}) \qquad [J] \qquad (37)$$

where

$Q_{\text{W,hp,out,sngl,j}}$	is the heat requirement of the DHW distribution subsystem covered by the heat pump in DHW-only operation in bin j;	(J)
$Q_{\text{W,hp,sngl,j}}$	is the produced DHW energy by the heat pump in DHW-only operation in bin j;	(J)
$Q_{\text{W,st,ls,j}}$	is the DHW storage losses in bin j (calculated according to 5.3.7.1.2);	(J)
f_{combi}	is the correction factor to take into account controller effect, corresponds to the fraction of simultaneous operation;	(-)
$k_{\text{W,bu,j}}$	is the fraction of DHW heat energy covered by the back-up heater in bin j.	(-)

Similarly for simultaneous operation, the DHW heat energy requirement can be calculated according to the equation

$$Q_{\text{W,hp,out,combi,j}} = Q_{\text{W,hp,combi,j}} - Q_{\text{W,st,ls,j}} \cdot (1 - k_{\text{W,bu,j}}) \cdot f_{\text{combi}} \qquad [J] \qquad (38)$$

where

$Q_{\text{W,hp,out,combi,j}}$	is the heat requirement of the DHW distribution subsystem covered by the heat pump in simultaneous operation in bin j;	(J)
$Q_{\text{W,hp,combi,j}}$	is the produced DHW energy by the heat pump in simultaneous operation in bin j;	(J)
$Q_{\text{W,st,ls,j}}$	is the DHW storage losses in bin j (calculated according to 5.3.7.1.2);	(J)
f_{combi}	is the correction factor to take into account controller effect, corresponds to the fraction of simultaneous operation;	(-)

$k_{W,bu,j}$ is the fraction of DHW heat energy covered by the back-up heater in bin j. (-)

The respective running time in SH-only and DHW-only operation modes are calculated according to Equation (28).

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 255-3:1997.

5.3.9.3 Boundary condition for the total running time

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

$$t_{hp,on,tot,j} = \min(t_{eff,j}, t_{H,hp,on,sgnl,j} + t_{W,hp,on,sgnl,j} + t_{HW,hp,on,combi,j}) \quad [s] \quad (39)$$

where

$t_{hp,on,tot,j}$ is the total running time of the heat pump in bin j; (s)

$t_{eff,j}$ is the effective bin time in bin j; (s)

$t_{H,hp,on,sngl,j}$ is the running time in space heating-only operation in bin j; (s)

$t_{W,hp,on,sngl,j}$ is the running time in DHW-only operation in bin j; (s)

$t_{HW,hp,on,combi,j}$ is the running time in simultaneous operation in bin j. (s)

If the calculated total running time is longer than the effective bin time, this is due to 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 5.3.9.5.

5.3.9.4 Detailed back-up calculation: Back-up energy due to lack of heating capacity of the heat pump

The detailed evaluation of the back-up energy is based on the evaluation of the running time according to the boundary conditions given in 5.3.9.3, 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 Equation (40), i.e. the back-up heater supplies heat either to the space heating system or to the domestic hot water system according to 5.3.9.5.

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

Outdoor air temperature θ_e(1 K bin)	Energy to be produced for SH $Q_{H,hp,j}$ according to Equation (29)	Energy to be produced for DHW $Q_{W,hp,j}$ according to Equation (29)	Heating capacity SH $\phi_{H,hp,sngl,j}$ according to heat pump charactерristic	Heating capacity DHW $\phi_{W,hp,sngl,j}$ according to heat pump characteristic	Heating capacity SH combined $\phi_{H,hp,combi,j}$ according to heat pump characteristic	Heating capacity DHW combined $\phi_{W,hp,combi,j}$ according to heat pump characteristic	Running time for SH $t_{H,hp,on}$ according to Equation (28)	Running time for DHW $t_{W,hp,on}$ according to Equation (28)	Running time combined $t_{hp,combi}$ according to Equation (34)	Total required running time $t_{hp,tot,j}$ according to Equation (30)	Effective bin time (1 K bin) $t_{eff,j}$ according to Equation (7)	Difference total running time to effective bin time	Required back-up energy $Q_{bu,cap,j}$ according to Equation (40)
$\theta_{e,min}$													
$\theta_{e,min} + 1$													
…													
Σ													Σ Back-up

5.3.9.5 Calculation of additional back-up energy due to lack of capacity

The additional back-up energy due to lack of heating capacity of the heat pump 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

$$Q_{bu,cap,j} = \phi_{hp,sngl,j} \cdot (t_{hp,on,tot,j} - t_{eff,j}) \qquad [J] \qquad (40)$$

where

$Q_{bu,cap,i}$	is the additional back-up energy due lack of heating capacity of the heat pump;	(J)
$t_{hp,on,tot,j}$	is the total (calculated) running time of the heat pump in bin j;	(s)
$t_{eff,j}$	is the effective bin time in bin j;	(s)
$\phi_{hp,sngl,j}$	is the heating capacity of the heat pump in the respective single operation mode.	(W)

The control strategy determines whether the back-up energy is supplied to the space heating system or the domestic hot water system. If the control strategy is not 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 domestic hot water system.

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

$$k_{\mathrm{bu,j}} = \frac{Q_{\mathrm{bu,opr,j}} + Q_{\mathrm{bu,cap,j}}}{Q_{\mathrm{gen,out,j}}} = k_{\mathrm{bu,opr,j}} + k_{\mathrm{bu,cap,j}} + \frac{(t_{\mathrm{hp,on,tot,j}} - t_{\mathrm{eff,j}}) \cdot \phi_{\mathrm{hp,sngl,j}}}{Q_{\mathrm{gen,out,j}}} \quad [-] \qquad (41)$$

where

$k_{\mathrm{bu,j}}$	is the fraction of heat energy covered by the back-up heater in the respective operation mode in bin j;	(-)
$Q_{\mathrm{bu,opr,j}}$	is the back-up energy due to operation limit temperature;	(J)
$Q_{\mathrm{bu,cap,j}}$	is the back-up energy due to lack of heating capacity of the heat pump;	(J)
$Q_{\mathrm{gen,out,j}}$	is the heat energy requirement of the distribution subsystem in bin j;	(J)
$k_{\mathrm{bu,opr,j}}$	is the fraction of back-up energy due to temperature operation limit;	(-)
$k_{\mathrm{bu,cap,j}}$	is the fraction of back-up energy due to lack of heating capacity of the heat pump (in case of simplified calculation);	(-)
$t_{\mathrm{hp,on,tot,j}}$	is the total (calculated) running time of the heat pump in bin j;	(s)
$t_{\mathrm{eff,j}}$	is the effective bin time in bin j;	(s)
$\phi_{\mathrm{hp,sngl,j}}$	is the 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 Equation (41).

The fraction $k_{\mathrm{bu,cap,j}}$ in Equation (41) only exists if the simplified back-up calculation according to 5.3.8.3 is applied. If the detailed back-up calculation according to 5.3.9.4 is applied, the fraction is contained in the lack of running time and $k_{\mathrm{bu,cap,j}} = 0$.

5.3.10 Auxiliary energy

5.3.10.1 General

To calculate the auxiliary energy, the respective power consumption of the auxiliary components has to be given as input. For 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:

$$W_{\mathrm{HW,gen,aux}} = \sum_{k} P_{\mathrm{gen,aux,k}} \cdot t_{\mathrm{gen,aux,on,k}} \quad [\mathrm{J}] \qquad (42)$$

where

$W_{\mathrm{HW,gen,aux}}$	is the total auxiliary energy;	(J)
$P_{\mathrm{gen,aux,k}}$	is the electrical power consumption of the auxiliary component k;	(W)
$t_{\mathrm{gen,aux,on,k}}$	is the 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 Clause 4 are considered in 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 heat pump is taken into account in the *COP* value. However, *COP* covers only the operation time, so stand-by operation has to be taken into account by the nominal power consumption of the auxiliary components and the stand-by time, unless a part load correction to the *COP* is applied according to 5.3.6.1. Correction shall be accomplished depending on the respective test standard.

The running time of the auxiliary components depends on the control of the generation subsystem:

- source pump running time is normally linked to the running time of the heat pump evaluated in 5.3.9 for the different operation modes;
- for primary pumps, control depends on the installed systems, e.g. 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 season for space heating operation, and the running time evaluated according to 5.3.9. If a correction for part load operation of the *COP* is applied according to 5.3.6, the stand-by auxiliary energy is already considered and does 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 value according to EN 255-3 due to the system testing.

5.3.10.2 Engine-driven and absorption heat pumps

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

5.3.11 Total thermal losses and recoverable thermal loss of the generation subsystem

5.3.11.1 Recoverable thermal loss from auxiliary components

Auxiliary energy is transformed partly to used energy and partly to thermal loss. Energy recovered as thermal energy transmitted to the heat transfer medium is considered totally recovered according to Equation (1) and is calculated by:

$$Q_{\mathrm{HW,gen,aux,rvd}} = \sum_k W_{\mathrm{gen,aux,k}} \cdot k_{\mathrm{gen,aux,ls,rvd,k}} \qquad \text{[J]} \qquad (43)$$

where

$Q_{\mathrm{HW,gen,aux,rvd}}$ is the totally recovered auxiliary energy; (J)

$W_{\mathrm{gen,aux,k}}$ is the auxiliary energy of the auxiliary component k; (J)

$k_{\mathrm{gen,aux,ls,rvd,k}}$ is the fraction of auxiliary energy of component k totally recovered as thermal energy. (-)

The fraction $k_{\mathrm{gen,aux,ls,rvd,k}}$ is already considered in the *COP* value according to standard testing of EN 14511 for electrically-driven heat pumps, and in this case $k_{\mathrm{gen,aux,ls,rvd,k}} = 0$.

Thermal loss to the ambiance of auxiliary components is calculated according to the equation

$$Q_{\mathrm{HW,gen,aux,ls}} = \sum_k W_{\mathrm{gen,aux,k}} \cdot k_{\mathrm{gen,aux,ls,k}} \qquad [\mathrm{J}] \qquad (44)$$

and is assumed recoverable according to a temperature reduction factor linked to location

$$Q_{\mathrm{HW,gen,aux,ls,rbl}} = \sum_k W_{\mathrm{gen,aux,k}} \cdot k_{\mathrm{gen,aux,ls,k}} \cdot (1 - b_{\mathrm{gen,aux,k}}) \qquad [\mathrm{J}] \qquad (45)$$

where

$Q_{\mathrm{HW,gen,aux,ls}}$ is the thermal loss to the ambiance of auxiliary components; (J)

$Q_{\mathrm{HW,gen,aux,ls,rbl}}$ is the recoverable thermal loss to the ambiance of auxiliary components; (J)

$W_{\mathrm{gen,aux,k}}$ is the auxiliary energy of the auxiliary component k; (J)

$k_{\mathrm{gen,aux,ls,k}}$ is the fraction of auxiliary energy of component k transmitted to the ambiance. (-) These values should be defined in a national annex. If no national values are specified, default values are given in B.5;

$b_{\mathrm{gen,aux,k}}$ is the temperature reduction factor for component k linked to location of the component. (-) The values of $b_{\mathrm{gen,aux,k}}$ shall be given in a national annex. If no national values are specified, default values are given in B.7.

5.3.11.2 Total generation subsystem thermal losses

The total envelope thermal 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.

The total thermal losses to the ambiance of the generation subsystem comprise envelope thermal losses and thermal losses of auxiliary components, according to the equation

$$Q_{\mathrm{HW,gen,ls,tot}} = Q_{\mathrm{HW,gen,ls}} + Q_{\mathrm{HW,gen,aux,ls}} = \sum_k Q_{\mathrm{gen,ls,k}} + Q_{\mathrm{HW,gen,aux,ls}} \qquad [\mathrm{J}] \qquad (46)$$

where

$Q_{\mathrm{HW,gen,ls,tot}}$ is the total generation subsystem thermal losses to the ambiance; (J)

$Q_{\mathrm{HW,gen,ls}}$ is the generation subsystem envelope thermal losses to the ambiance; (J)

$Q_{\mathrm{gen,ls,k}}$ is the envelope thermal losses to the ambiance of the generation subsystem component k; (J)

$Q_{\mathrm{HW,gen,aux,ls}}$ is the thermal losses to the ambiance of auxiliary components. (J)

5.3.11.3 Recoverable thermal loss due to generation subsystem envelope thermal losses

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

$$Q_{\mathrm{HW,gen,ls,rbl}} = \sum_{k} Q_{\mathrm{gen,ls,k}} \cdot (1 - b_{\mathrm{gen,k}}) \qquad [\mathrm{J}] \qquad (47)$$

where

$Q_{\mathrm{HW,gen,ls,rbl}}$ is the recoverable envelope thermal loss of the generation subsystem; (J)

$Q_{\mathrm{gen,ls,k}}$ is the envelope thermal losses to the ambiance of the generation subsystem component k; (J)

$b_{\mathrm{gen,k}}$ is the 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 B.7.

5.3.11.4 Total recoverable thermal loss of the generation subsystem

The total recoverable thermal loss of the generation subsystem comprises the recoverable envelope thermal loss and the recoverable thermal loss to the ambiance of auxiliary components according to the equation

$$Q_{\mathrm{HW,gen,ls,rbl,tot}} = Q_{\mathrm{HW,gen,ls,rbl}} + Q_{\mathrm{HW,gen,aux,ls,rbl}} \qquad [\mathrm{J}] \qquad (48)$$

where

$Q_{\mathrm{HW,gen,ls,rbl,tot}}$ is the total recoverable thermal loss of the generation subsystem; (J)

$Q_{\mathrm{HW,gen,ls,rbl}}$ is the recoverable envelope thermal loss of the generation subsystem; (J)

$Q_{\mathrm{HW,gen,aux,ls,rbl}}$ is the recoverable thermal loss of auxiliary components. (J)

5.3.12 Calculation of total energy input

5.3.12.1 Electrically-driven heat pumps

5.3.12.1.1 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:

$$E_{\mathrm{H,hp,in}} = \sum_{j=1}^{N_{\mathrm{bins}}} \frac{Q_{\mathrm{H,hp,sngl,j}}}{COP_{\mathrm{H,sngl,j}}} + \sum_{j=1}^{N_{\mathrm{bins}}} \frac{Q_{\mathrm{H,hp,combi,j}}}{COP_{\mathrm{H,combi,j}}} \qquad [\mathrm{J}] \qquad (49)$$

where

$E_{\mathrm{H,hp,in}}$ is the electrical energy input to the heat pump in space heating mode; (J)

$Q_{\mathrm{H,hp,sngl,j}}$ is the produced energy by the heat pump in space heating-only operation in bin j; (J)

$Q_{\mathrm{H,hp,combi,j}}$ is the produced energy by the heat pump for space heating in simultaneous operation in bin j; (J)

$COP_{\mathrm{H,sngl,j}}$ is the coefficient of performance in space heating-only operation at the operating point of bin j, taken as performance factor for the whole bin; (W/W)

$COP_{\text{H,combi,j}}$ is the coefficient of performance of space heating in simultaneous operation at the operating point of bin j, taken as performance factor for the whole bin; (W/W)

N_{bins} is the number of bins. (-)

5.3.12.1.2 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

$$E_{\text{W,hp,in}} = \sum_{j=1}^{N_{\text{bins}}} \frac{Q_{\text{W,gen,out,sngl,j}}}{COP_{\text{t,sngl,j}}} + P_{\text{es,sgnl}} \cdot t_{\text{W,sngl,tot}} + \sum_{j=1}^{N_{\text{bins}}} \frac{Q_{\text{W,gen,out,combi,j}}}{COP_{\text{t,combi,j}}} + P_{\text{es,combi}} \cdot t_{\text{W,combi,tot}} \quad \text{[J]} \quad (50)$$

where

$E_{\text{W,hp,in}}$ is the electrical energy input to the heat pump in DHW mode; (J)

$Q_{\text{W,gen,out,sngl,j}}$ is the heat energy requirement of the DHW distribution subsystem in bin j covered by the heat pump in DHW-only operation; (J)

$Q_{\text{W,gen,out,combi,j}}$ is the heat energy requirement of the DHW distribution subsystem in bin j covered by the heat pump in simultaneous operation; (J)

$COP_{\text{t,sngl,j}}$ is the coefficient of performance for the extraction of domestic hot water of bin j in DHW-only operation according to EN 255-3, taken as performance factor for the whole bin; (W/W)

$COP_{\text{t,combi,j}}$ is the coefficient of performance for the extraction of domestic hot water of bin j in simultaneous operation, taken as performance factor for the whole bin; (W/W)

$P_{\text{es,sgnl}}$ is the electricity power input to cover storage losses according to EN 255-3 for DHW-only operation; (W)

$P_{\text{es,combi}}$ is the electricity power input to cover storage losses according to EN 255-3 for simultaneous operation; (W)

$t_{\text{W,sngl,tot}}$ is the total time of all calculation periods in DHW-only operation; (s)

$t_{\text{W,combi,tot}}$ is the total time of all calculation periods in simultaneous operation; (s)

N_{bins} is the number of bins. (-)

Allocation of the total time to DHW-only operation and simultaneous operation is done by the fraction of simultaneous operation as for the storage losses in 5.3.9.2.

NOTE If no values according to EN 255-3 are available, the calculation is accomplished in the same way as for the space heating operation mode, but only for alternate operating systems. For *COP* values to use, see 5.3.5.2.

5.3.12.2 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 according to Equation (49). This approach corresponds to the system boundary given in 4.1.

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 space heating operation and domestic hot water operation, respectively, as well as for single operation and combined operation, respectively, if required:

$$E_{\mathrm{hp,in}} = \sum_{j=1}^{N_{\mathrm{bins}}} \frac{Q_{\mathrm{hp,j}} - Q_{\mathrm{eng,rvd,j}} - k_{\mathrm{gen,aux,ls,rvd}} \cdot W_{\mathrm{HW,gen,aux,j}}}{COP_{\mathrm{j}}} \quad [\mathrm{J}] \qquad (51)$$

where

$E_{\mathrm{hp,in}}$	is the fuel or heat energy input to the engine-driven or absorption heat pump in the respective operation mode;	(J)
$Q_{\mathrm{hp,j}}$	is the produced energy of the heat pump in bin j;	(J)
$Q_{\mathrm{eng,rvd,j}}$	is the recovered energy from the combustion engine in bin j (only engine-driven heat pumps);	(J)
$k_{\mathrm{gen,aux,ls,rvd}}$	is the fraction of auxiliary energy recovered as thermal energy (depending on testing);	(-)
$W_{\mathrm{HW,gen,aux,j}}$	is the auxiliary energy in bin j;	(J)
COP_{j}	is the coefficient of performance at the operating point of bin j, taken as performance factor for the whole bin for the respective operation mode;	(W/W)
N_{bin}	is the number of bins.	(-)

The recovered energy $Q_{\mathrm{eng,rvd}}$ shall be calculated based on test results or manufacturer data. Whether 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. prEN 15203 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.

5.3.12.3 Energy input to back-up system

5.3.12.3.1 Electrical back-up heater

$$E_{\mathrm{HW,bu,in}} = \sum_{j=1}^{N_{\mathrm{bins}}} \frac{(Q_{\mathrm{H,gen,out,j}} + Q_{\mathrm{H,gen,ls,j}}) \cdot k_{\mathrm{H,bu,j}}}{\eta_{\mathrm{H,bu}}} + \frac{(Q_{\mathrm{W,gen,out,j}} + Q_{\mathrm{W,gen,ls,j}}) \cdot k_{\mathrm{W,bu,j}}}{\eta_{\mathrm{W,bu}}} \quad [\mathrm{J}] \qquad (52)$$

where

$E_{\mathrm{HW,bu,in}}$	is the total electrical energy input to operate the back-up heater;	(J)
$Q_{\mathrm{H,gen,out,j}}$	is the heat energy requirement of the space heating distribution subsystem in bin j;	(J)
$Q_{\mathrm{H,gen,ls,j}}$	is the thermal losses of the generation subsystem due to space heating operation in bin j;	(J)

$k_{H,bu,j}$	is the fraction of space heating heat energy covered by the back-up heater in bin j;	(-)
$\eta_{H,bu}$	is the efficiency of the electrical back-up heater for space heating mode;	(-)
$Q_{W,gen,out,j}$	is the heat energy requirement of the DHW distribution subsystem in bin j;	(J)
$Q_{W,gen,ls,j}$	is the thermal losses of the generation subsystem due to DHW operation in bin j;	(J)
$k_{W,bu,j}$	is the fraction of DHW heat energy covered by the back-up heater in bin j;	(-)
$\eta_{W,bu}$	is the efficiency of the electrical back-up heater for DHW mode;	(-)
N_{bins}	is the number of bins.	(-)

The efficiency of the electrical back-up heater shall be given in a national annex or be determined based on the system configuration and the product. If no values are given, a default value is given in B.8.

5.3.12.3.2 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 EN 15316-4-1 for combustion boiler back-up heaters.

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

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

$$E_{HW,gen,in} = E_{H,hp,in} + E_{W,hp,in} + E_{HW,bu,in} \quad [J] \qquad (53)$$

where

$E_{HW,gen,in}$	is the total energy input to heat pump and back-up heater;	(J)
$E_{H,hp,in}$	is the energy input to the heat pump in space heating mode;	(J)
$E_{W,hp,\,in}$	is the energy input to the heat pump in DHW mode;	(J)
$E_{HW,bu,in}$	is the total energy input to operate the back-up heater.	(J)

5.3.12.5 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 domestic hot water requirement and generation subsystem losses is calculated according to Equation (1), where the recovered auxiliary energy is to be set to zero ($k_{gen,aux,ls,rvd} = 0$) for electrically-driven heat pumps tested according to EN 14511. For engine-driven and gas heat pumps, the factor $k_{gen,aux,ls,rvd}$ depends on the fraction taken into account during testing.

5.3.12.6 Seasonal performance factor and expenditure factor of the generation subsystem

5.3.12.6.1 General

For the seasonal performance factor, two characteristic values are defined based on different system boundaries, i.e. for the generation subsystem and for the heat pump as generator itself.

5.3.12.6.2 Electrically-driven generation subsystems

The total seasonal performance factor of the generation subsystem (incl. the heat pump and the electrical back-up heater) can be calculated according to the equation

$$SPF_{\mathrm{HW,gen}} = \frac{Q_{\mathrm{H,gen,out}} + Q_{\mathrm{W,gen,out}}}{E_{\mathrm{HW,gen,in}} + W_{\mathrm{HW,gen,aux}}} \quad [-] \qquad (54)$$

where

$SPF_{\mathrm{HW,gen}}$	is the total seasonal performance factor of generation subsystem;	(-)
$Q_{\mathrm{H,gen,out}}$	is the heat energy requirement of the space heating distribution subsystem;	(J)
$Q_{\mathrm{W,gen,out}}$	is the heat energy requirement of the domestic hot water distribution subsystem;	(J)
$E_{\mathrm{HW,gen,in}}$	is the total electrical energy input to heat pump and back-up heater;	(J)
$W_{\mathrm{HW,gen,aux}}$	is the auxiliary energy input to the generation subsystem.	(J)

The seasonal performance factor of the heat pump can be evaluated with regard to the produced heat characterising the performance of the heat pump as generator.

The heat pump performance with regard to produced heat is calculated according to the equation

$$SPF_{\mathrm{HW,hp}} = \frac{Q_{\mathrm{H,hp}} + Q_{\mathrm{W,hp}}}{E_{\mathrm{HW,hp,in}} + W_{\mathrm{gen,aux,sc}} + W_{\mathrm{gen,aux,sby}}} \quad [-] \qquad (55)$$

where

$SPF_{\mathrm{HW,hp}}$	is the seasonal performance factor of the heat pump;	(-)
$Q_{\mathrm{H,hp}}$	is the produced heat energy by the heat pump in space heating mode;	(J)
$Q_{\mathrm{W,hp}}$	is the produced heat energy by the heat pump in domestic hot water mode;	(J)
$E_{\mathrm{HW,hp,in}}$	is the total electrical energy input to the heat pump to cover the heat requirement;	(J)
$W_{\mathrm{gen,aux,sc}}$	is the auxiliary energy for the source system;	(J)
$W_{\mathrm{gen,aux,sby}}$	is the auxiliary energy for the heat pump in stand-by operation.	(J)

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

$$e_{\mathrm{HW,gen}} = \frac{1}{SPF_{\mathrm{HW,gen}}} \quad [-] \qquad (56)$$

where

$e_{\mathrm{HW,gen}}$	is the expenditure factor of generation subsystem;	(-)
$SPF_{\mathrm{HW,gen}}$	is the total seasonal performance factor of generation subsystem.	(-)

5.3.12.7 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

$$Q_{\mathrm{HW,bu,out}} = \sum_{j=1}^{N_{\mathrm{bins}}} (Q_{\mathrm{H,gen,out,j}} + Q_{\mathrm{H,gen,ls,j}}) \cdot k_{\mathrm{H,bu,j}} + (Q_{\mathrm{W,gen,out,j}} + Q_{\mathrm{W,gen,ls,j}}) \cdot k_{\mathrm{W,bu,j}} \quad [\mathrm{J}] \qquad (57)$$

and the total heat produced by the heat pump can be calculated according to the equation

$$Q_{\mathrm{HW,hp,out}} = \sum_{j=1}^{N_{\mathrm{bins}}} \left[(Q_{\mathrm{H,gen,out,j}} + Q_{\mathrm{H,gen,ls,j}}) + (Q_{\mathrm{W,gen,out,j}} + Q_{\mathrm{W,gen,ls,j}}) \right] - Q_{\mathrm{HW,bu,out}} \quad [\mathrm{J}] \qquad (58)$$

where

$Q_{\mathrm{HW,bu,out}}$	is the produced heat energy of the back-up heater;	(J)
$k_{\mathrm{H,bu,j}}$	is the fraction of space heating heat energy covered by the back-up heater in bin j;	(-)
$Q_{\mathrm{H,gen,out,j}}$	is the heat energy requirement of the space heating distribution subsystem in bin j;	(J)
$Q_{\mathrm{H,gen,ls,j}}$	are the thermal losses of the generation subsystem due to space heating operation in bin j;	(J)
$k_{\mathrm{W,bu,j}}$	is the fraction of DHW heat energy covered by the back-up heater in bin j;	(-)
$Q_{\mathrm{W,gen,out,j}}$	is the heat energy requirement of the DHW distribution subsystem in bin j;	(J)
$Q_{\mathrm{W,gen,ls,j}}$	are the thermal losses of the generation subsystem due to DHW operation in bin j;	(J)
$Q_{\mathrm{HW,hp,out}}$	is the produced heat energy by the heat pump;	(J)
N_{bins}	is the number of bins.	(-)

5.3.13 Summary of output values

Resulting outputs

— energy input to cover the heat requirement of the distribution subsystems (see 5.3.12.4, Equation (53));

— total thermal losses of the generation subsystem (see 5.3.11.2, Equation (46));

— total generation subsystem thermal losses recoverable for space heating (see 5.3.11.4, Equation (48));

— total auxiliary energy input to the generation subsystem (see 5.3.10, Equation (42)).

Optional outputs

— total heat produced by the back-up heater (see 5.3.12.7, Equation (57));

— total heat produced by the heat pump (see 5.3.12.7, Equation (58));

— overall seasonal performance factors/expenditure factor of the generation subsystem (see 5.3.12.6, Equation (54), Equation (55), Equation (56));

— totally used ambient heat (see 5.3.12.5, Equation (1)).

Annex A
(informative)

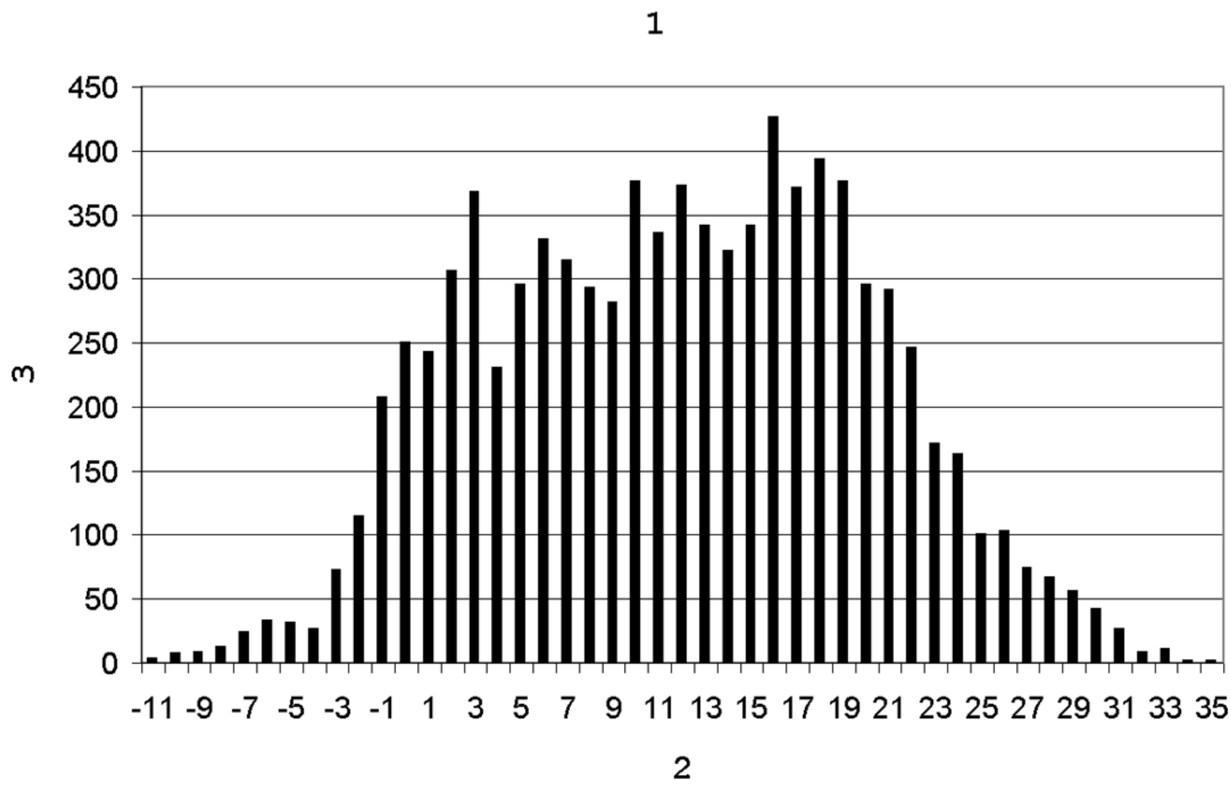
Example of evaluation of meteorological data

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, for instance the Software Meteonorm [1] can deliver hourly averaged values of site of the whole world based on a statistical evaluation.

<u>Input data:</u>

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



Key

1 annual frequency of the outdoor air temperature

2 outdoor air temperature [°C]

3 number of hours of outdoor air temperature in 1 K bins [h]

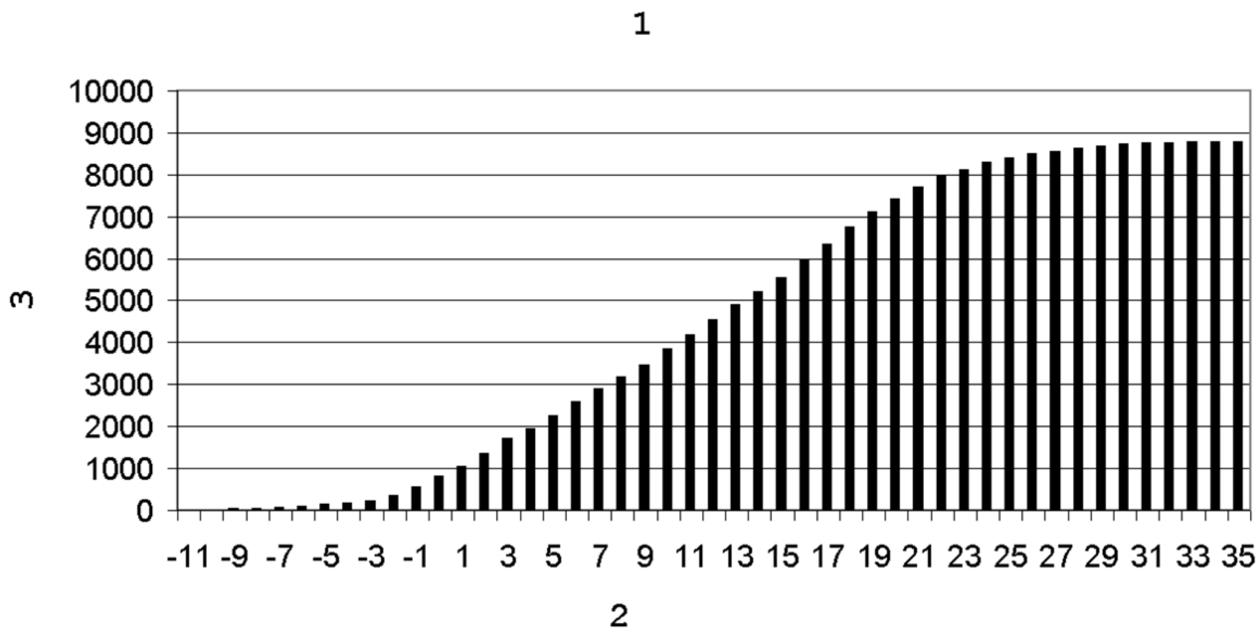
Figure A.1 — Annual frequency of the outdoor temperature

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.



Key

1 cumulative annual frequency of the outdoor air temperature

2 outdoor air temperature [°C]

3 cumulative number of hours below the respective outdoor air temperature [h]

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:

$$N_{\mathrm{ho,k}} = \sum_{j=1}^{k} N_{\mathrm{ho,j}} \qquad [-] \qquad (A.1)$$

where

$N_{\mathrm{ho,k}}$ is the cumulated number of hours up to bin k ($1 \leq k \leq N_{\mathrm{bins}}$); (h)

$N_{\mathrm{ho,j}}$ is the number of hours for the actual bin; (h)

j is the number of the actual bin; (-)

N_{bins} is the number of bins. (-)

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:

$$DH_{\mathrm{H,j}} = N_{\mathrm{j}} \cdot (\theta_{\mathrm{i,des}} - \theta_{\mathrm{e,j}}) \qquad [°\mathrm{Ch}] \qquad (A.2)$$

where

$DH_{\mathrm{H,j}}$	is the heating degree hours for bin j;	(°Ch)
N_{j}	is the number of hours for bin j;	(h)
$\theta_{\mathrm{i,des}}$	is the indoor design temperature;	(°C)
$\theta_{\mathrm{e,j}}$	is the outdoor air temperature for bin j.	(°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:

$$DH_{\mathrm{H,\theta j}} = \sum_{j=1}^{k} DH_{\mathrm{H,j}} \qquad [°\mathrm{Ch}] \qquad (A.3)$$

where

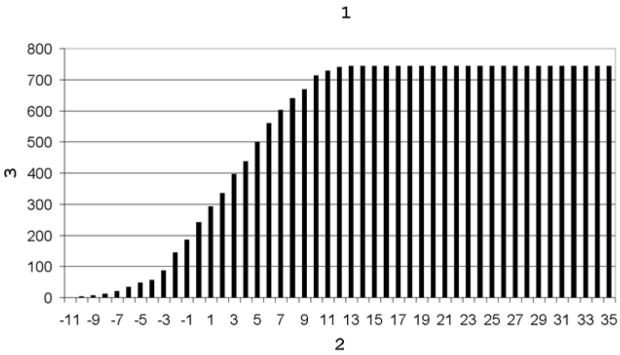
$DH_{\mathrm{H,\theta k}}$	is the cumulative heating degree hours up to temperature θ_k corresponding to bin k ($1 \leq k \leq N_{\mathrm{bins}}$);	(°Ch)
$DH_{\mathrm{H,j}}$	is the heating degree hours for bin j;	(°Ch)
j	is the number of the actual bin;	(-)
N_{bins}	is the number of bins.	(-)

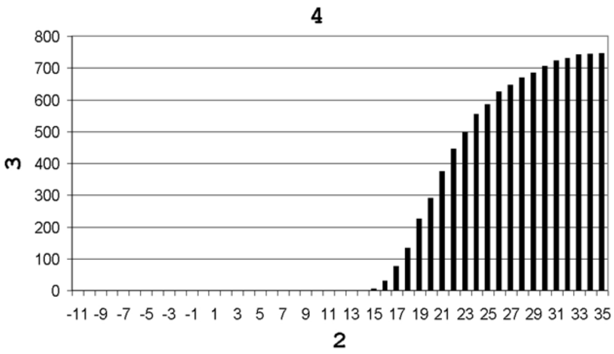
With the cumulative heating degree hours, the weighting factors for the heat pump operation can be calculated according to the equations above.

The calculations of step 1 through 4 are presented in Table A.1.

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. Samples for January and August are shown in Figure A.3.





Key

1 cumulative annual frequency of the outdoor air temperature in January

2 outdoor air temperature [°C]

3 cumulative number of hours below the respective outdoor air temperature [h]

4 cumulative annual frequency of the outdoor air temperature in August

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)

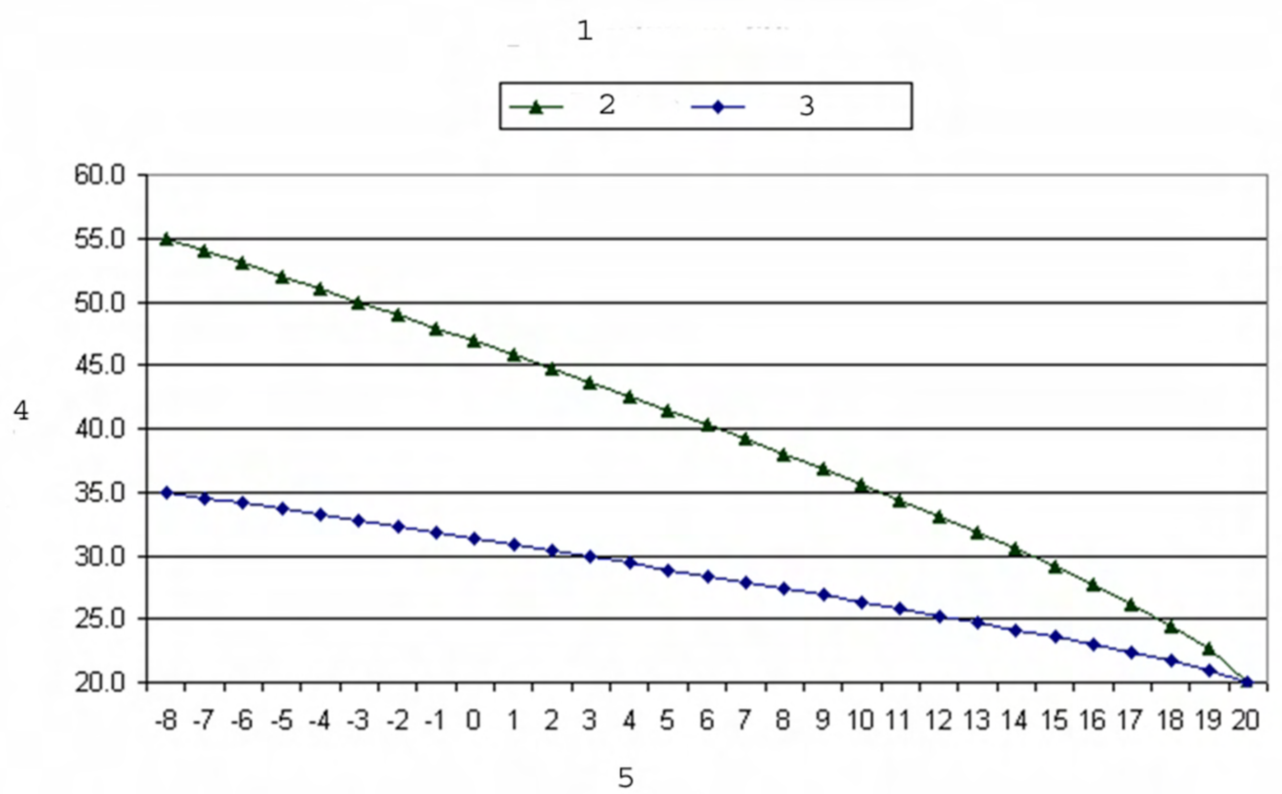
Bin	Bin time	Cumulative bin time	DH_H 20/15	Cumulative DH_H 20/15	Weighting factor for space heating mode (1 K bins)	Weighting factor for domestic hot water mode (1 K bins)
- 11	3	3	93	93	0,00	0,000
- 10	7	10	210	303	0,00	0,001
- 9	8	18	232	535	0,00	0,001
- 8	12	30	336	871	0,00	0,001
- 7	24	54	648	1 519	0,01	0,003
- 6	33	87	858	2 377	0,01	0,004
- 5	31	118	775	3 152	0,01	0,004
- 4	26	144	624	3 776	0,01	0,003
- 3	72	216	1 656	5 432	0,02	0,008
- 2	114	330	2 508	7 940	0,03	0,013
- 1	207	537	4 347	12 287	0,06	0,024
0	250	787	5 000	17 287	0,07	0,029
1	243	1 030	4 617	21 904	0,06	0,028
2	306	1 336	5 508	27 412	0,08	0,035
3	368	1 704	6 256	33 668	0,09	0,042
4	230	1 934	3 680	37 348	0,05	0,026
5	295	2 229	4 425	41 773	0,06	0,034
6	331	2 560	4 634	46 407	0,06	0,038
7	314	2 874	4 082	50 489	0,06	0,036
8	293	3 167	3 516	54 005	0,05	0,033
9	281	3 448	3 091	57 096	0,04	0,032
10	376	3 824	3 760	60 856	0,05	0,043
11	336	4 160	3 024	63 880	0,04	0,038
12	373	4 533	2 984	66 864	0,04	0,043
13	341	4 874	2 387	69 251	0,03	0,039
14	322	5 196	1 932	71 183	0,03	0,037
15	341	5 537	1 705	72 888	0,02	0,039
16	426	5 963	1 704	74 592		0,049
17	371	6 334	1 113	75 705		0,042
18	393	6 727	786	76 491		0,045
19	376	7 103	376	76 867		0,043
20	295	7 398	0	76 867		0,034
21	291	7 689	0	76 867		0,033
22	246	7 935	0	76 867		0,028
23	171	8 106	0	76 867		0,020
24	163	8 269	0	76 867		0,019
25	100	8 369	0	76 867		0,011
26	103	8 472	0	76 867		0,012
27	74	8 546	0	76 867		0,008
28	67	8 613	0	76 867		0,008
29	56	8 669	0	76 867		0,006
30	42	8 711	0	76 867		0,005
31	26	8 737	0	76 867		0,003
32	8	8 745	0	76 867		0,001
33	11	8 756	0	76 867		0,001
34	2	8 758	0	76 867		0,000
35	2	8 760	0	76 867		0,000

Annex B
(informative)

Default values of parameters for the case specific seasonal performance method

B.1 Controller setting of flow temperature (heating characteristic curve)

Typical controller settings and design temperature spreads based on different emission systems are given in Figure B.1.



Key

1 heating characteristic curve

2 radiators

3 floor heating system

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]

$$\theta_{\mathrm{H,gen,f}} = \theta_{\mathrm{i,des}} + \frac{\theta_{\mathrm{H,gen,fdes}} - \theta_{\mathrm{H,dis,r,des}}}{2} \cdot \frac{\theta_{\mathrm{i,des}} - \theta_{\mathrm{e}}}{\theta_{\mathrm{i,des}} - \theta_{\mathrm{e,des}}} + (\theta_{\mathrm{avg}} - \theta_{\mathrm{i,des}}) \cdot \left(\frac{\theta_{\mathrm{i,des}} - \theta_{\mathrm{e}}}{\theta_{\mathrm{i,des}} - \theta_{\mathrm{e,des}}}\right)^{1/n} \quad [°\mathrm{C}] \qquad \text{(B.1)}$$

where

$\theta_{\mathrm{H,gen,f}}$ is the actual flow temperature; (°C)

$\theta_{i,des}$ is the indoor design temperature (in case of night setback average temperature); (°C)

$\theta_{H,gen,f,des}$ is the flow temperature at design conditions; (°C)

$\theta_{H,dis,r,des}$ is the return temperature at design conditions; (°C)

θ_{e} is the outdoor air temperature; (°C)

θ_{avg} is the arithmetic average of flow and return temperature; (°C)

$\theta_{e,des}$ is the outdoor design temperature; (°C)

n is the 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 n
Radiators	55/45	1,2 to 1,4
Convectors	55/45	1,2 to 1,4
Floor heating	40/30	1,0

B.2 Temperature correction factor for domestic hot water storage loading

The default value for the temperature correction factor for storage loading is $f_{W,s}$ = 0,95.

B.3 Average water temperature of domestic hot water storages

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

Table B.2 — Average water temperature of domestic hot water storages

Hot water temperature at storage outlet [°C]	Average temperature of the stored hot water [°C]
60	54
55	49,5
50	45

B.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 systems 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 values given in Table B.3 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/24 h]
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
1 000	4,7
1 200	4,8
1 300	5,0
1 500	5,1
2 000	5,2

B.5 Generation subsystem auxiliaries

The part of the nominal electrical energy transmitted to the ambiance $k_{gen,aux,ls}$ can be calculated by:

— for pumps and fans:

$k_{gen,aux,ls} = 1 - \eta_{gen,aux} - k_{gen,aux,ls,rvd}$

The default value of the hydraulic efficiency $\eta_{gen,aux}$ is 0,3 (defined in EN 14511 as pump or fan efficiency). $k_{gen,aux,ls,rvd}$ is set to 0,5, however, for calculation of recovered losses it might already be included in the *COP*;

— for other auxiliary components, i.e. auxiliary components for which no energy is transferred to the heat transfer medium, e.g. control:

$k_{gen,aux,ls} = 1$.

B.6 Factor f_{combi} for simultaneous operation

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

B.7 Temperature reduction factor linked to location

Table B.4 — Default values of the temperature reduction factor

Generator location	Temperature reduction factor, b_{gen} and $b_{gen,aux}$
Inside heated space	0
Outside heated space	1

B.8 Efficiency value of the electrical back-up heater for space heating or DHW operation

If no information is available, the efficiency of the electrical back-up heater is set to η_{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

$$\eta_{\text{ex}} = \frac{COP}{COP_{\text{crnt}}} \quad [-] \quad \text{(C.1)}$$

where

η_{ex}	is the exergetic efficiency;	(-)
COP	is the coefficient of performance;	(W/W)
COP_{crnt}	is the coefficient of performance of the Carnot cycle Carnot *COP*.	(W/W)

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

$$COP_{\text{crnt}} = \frac{T_{\text{hot}}}{T_{\text{hot}} - T_{\text{cold}}} = \frac{\theta_{\text{sk}} + 273.15}{\theta_{\text{sk}} - \theta_{\text{sc}}} \quad [\text{W/W}] \quad \text{(C.2)}$$

where

COP_{crnt}	is the Carnot *COP*;	(-)
T_{hot}	is the temperature on the hot heat pump process side (sink);	(K)
T_{cold}	is the temperature on the cold heat pump process side (source);	(K)
θ_{sk}	is the sink temperature of the heat pump;	(°C)
θ_{sc}	is the 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

$$COP_{\mathrm{crnt}} = \frac{\dfrac{T_{\mathrm{gen,in}} - T_{\mathrm{cold}}}{T_{\mathrm{gen,in}} \cdot T_{\mathrm{cold}}}}{\dfrac{T_{\mathrm{hot}} - T_{\mathrm{cold}}}{T_{\mathrm{hot}} \cdot T_{\mathrm{cold}}}} = \frac{\theta_{\mathrm{sk}} + 273.15}{\theta_{\mathrm{gen,in}} + 273.15} \cdot \frac{\theta_{\mathrm{gen,in}} - \theta_{\mathrm{sc}}}{\theta_{\mathrm{sk}} - \theta_{\mathrm{sc}}} \quad \text{[W/W]} \qquad \text{(C.3)}$$

where

COP_{crnt}	is the Carnot *COP*;	(-)
$T_{\mathrm{gen,in}}$	is the temperature on the generation side (burner, boiler, heat exchanger);	(K)
T_{hot}	is the temperature on the hot heat pump process side (sink);	(K)
T_{cold}	is the temperature on the cold heat pump process side (source);	(K)
$\theta_{\mathrm{gen,in}}$	is the temperature on the generation side (burner, boiler, heat exchanger);	(°C)
θ_{sk}	is the sink temperature of the heat pump;	(°C)
θ_{sc}	is the 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 [8], 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

$$COP_{\mathrm{opr}} = COP_{\mathrm{standard}} \cdot \frac{COP_{\mathrm{crnt,opr}}}{COP_{\mathrm{crnt,standard}}} = COP_{\mathrm{standard}} \cdot f_{\mathrm{T}} \quad \text{[W/W]} \qquad \text{(C.4)}$$

where

COP_{opr}	is the *COP* due to temperature conditions in operation;	(W/W)
COP_{standard}	is the *COP* due to standard test temperature conditions;	(W/W)
$COP_{\mathrm{crnt,opr}}$	is the Carnot *COP* due to temperature conditions in operation;	(-)
$COP_{\mathrm{crnt,standard}}$	is the Carnot *COP* due to standard test conditions.	(-)

For electrically-driven heat pumps, the correction factor f_{T} – taking into account the impact of different temperatures – can be calculated according to the following equations:

— for air-to-water heat pumps or water-to-water heat pumps:

$$f_{\mathrm{T}} = \frac{COP_{\mathrm{crnt,opr}}}{COP_{\mathrm{crnt,standard}}} = \frac{T_{\mathrm{sk,out,opr}} \cdot (\theta_{\mathrm{sk,out,standard}} - \theta_{\mathrm{sc,in,standard}})}{T_{\mathrm{sk,out,standard}} \cdot (\theta_{\mathrm{sk,out,opr}} - \theta_{\mathrm{sc,in,opr}})} \quad \text{[-]} \qquad \text{(C.5)}$$

— for air-to-air heat pumps:

$$f_{\mathrm{T}} = \frac{COP_{\mathrm{crnt,opr}}}{COP_{\mathrm{crnt,standard}}} = \frac{T_{\mathrm{sk,in,opr}} \cdot (\theta_{\mathrm{sk,in,standard}} - \theta_{\mathrm{sc,in,standard}})}{T_{\mathrm{sk,in,standard}} \cdot (\theta_{\mathrm{sk,in,opr}} - \theta_{\mathrm{sc,in,opr}})} \quad \text{[-]} \qquad \text{(C.6)}$$

where

f_T	is the correction factor for temperature deviation from measured standard test point;	(-)
$COP_{crnt,opr}$	is the Carnot *COP* at source temperature in operation;	(-)
$COP_{crnt,standard}$	is the Carnot *COP* at measured standard test point;	(-)
$T_{sk,out,opr}$	is the outlet temperature on sink side in operation;	(K)
$T_{sk,out,standard}$	is the outlet temperature on sink side at measured standard test point;	(K)
$T_{sk,in,opr}$	is the inlet temperature on sink side in operation;	(K)
$T_{sk,in,standard}$	is the inlet temperature on sink side at measured standard test point;	(K)
$\theta_{sc,in,opr}$	is the inlet temperature on source side in operation;	(°C)
$\theta_{sc,in,standard}$	is the inlet temperature on source side at measured standard test point;	(°C)
$\theta_{sk,out,opr}$	is the outlet temperature on sink side in operation;	(°C)
$\theta_{sk,out,standard}$	is the outlet temperature on sink side at measured standard test point;	(°C)
$\theta_{sk,in,opr}$	is the inlet temperature on sink side in operation;	(°C)
$\theta_{sk,in,standard}$	is the inlet temperature on sink side at measured standard test point.	(°C)

For thermally-driven heat pumps it is assumed that $\theta_{gen,in}$ 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:

$$f_T = \frac{T_{sk,out,opr}}{T_{sk,out,standard}} \cdot \frac{\theta_{gen,in} - \theta_{sc,in,opr}}{\theta_{sk,out,opr} - \theta_{sc,in,opr}} \cdot \frac{\theta_{sk,out,standard} - \theta_{sc,in,standard}}{\theta_{gen,in} - \theta_{sc,in,standard}} \quad [-] \qquad (C.7)$$

— for air-to-air heat pumps:

$$f_T = \frac{T_{sk,in,opr}}{T_{sk,in,standard}} \cdot \frac{\theta_{gen,in} - \theta_{sc,in,opr}}{\theta_{sk,in,opr} - \theta_{sc,in,opr}} \cdot \frac{\theta_{sk,in,standard} - \theta_{sc,in,standard}}{\theta_{gen,in} - \theta_{sc,in,standard}} \quad [-] \qquad (C.8)$$

where

f_T	is the correction factor for temperature deviation from measured standard test point;	(-)
$T_{sk,out,opr}$	is the outlet temperature on sink side in operation;	(K)
$T_{sk,out,standard}$	is the outlet temperature on sink side at measured standard test point;	(K)
$T_{sk,in,opr}$	is the inlet temperature on sink side in operation;	(K)
$T_{sk,in,standard}$	is the inlet temperature on sink side at measured standard test point;	(K)
$\theta_{gen,in}$	is the temperature on the generation side (burner, boiler, heat exchanger);	(°C)
$\theta_{sk,out,opr}$	is the outlet temperature on sink side in operation;	(°C)

$\theta_{sk,out,standard}$ is the outlet temperature on sink side at measured standard test point; (°C)

$\theta_{sk,in,opr}$ is the inlet temperature on sink side in operation; (°C)

$\theta_{sk,in,standard}$ is the inlet temperature on sink side at measured standard test point; (°C)

$\theta_{sc,in,opr}$ is the inlet temperature on source side in operation; (°C)

$\theta_{sc,in,standard}$ is the 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 5.3.2)

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 gives the parameters of the space heating system.

Table D.1 — Boundary conditions

Meteorological data	
Site (see Table A.1)	Gelterkinden, BL, CH
Design outdoor temperature $\theta_{e,des}$ [°C]	-8

Table D.2 — Space heating operation mode

Input data space heating	
Space heating energy requirement $Q_{H,gen,out}$ [kWh] (according to EN 15316-2-3)	20 158
Indoor design temperature $\theta_{i,des}$ [°C]	20
Type of heat emission system (radiators/convectors/floor/wall/air)	Radiators
Exponent n of heat emission system [-]	1,2
Design flow temperature $\theta_{f,des}$ (at outdoor design temperature $\theta_{e,des}$) [°C]	55
Temperature spread of SH system at design conditions $\Delta\theta_{e,des}$ [K]	10
Upper outdoor 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 $V_{H,st,}$ [l]	400
Storage stand-by loss $Q_{st,sby}$ [kWh/24 h]	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 $\theta_{st,sby}$ [°C]	40
Surrounding temperature of the storage $\theta_{st,amb}$ [°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 parameters of the DHW system. Table D.4 contains general parameters of the heat pump, and Table D.5, Table D.6 and Table D.7 contain the characteristics 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 $Q_{W,gen,out}$ [kWh] (according to EN 15316-3-2)	3 300
Cold water inlet temperature $\theta_{W,st,in}$ [°C]	15
Hot water temperature at storage outlet $\theta_{W,st,out}$ [°C]	60
Volume DHW storage $V_{W,st}$ [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 parameters 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 $\theta_{hp,opr}$ [°C]	55

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

Space heating-only (EN 14511)			
Source temperature at test point θ_{sc} [°C]	**- 5**	**0**	**5**
Sink temperature at first test point θ_{sk} [°C]	**35**		
Sink temperature at second test point θ_{sk} [°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 $\phi_{H,hp,sngl}$ [kW]		9	10,3
Heating capacity at sink temperature 45 °C $\phi_{H,hp,sngl}$ kW]	7,7	8,7	9,9

Table D.6 — Characteristic of the heat pump for DHW-only operation (EN 255-3)

DHW-only (EN 255-3)			
Hot water reference temperature $\theta_{W,st,out}$ [°C]	55°C		
Source temperature at test point θ_{sc} [°C]	**- 5**	**0**	**5**
COP_t (EN 255-3) [W/W]		2,36	
Power input for storage stand-by losses $P_{es,sngl}$ [W] (EN 255-3)		62	
Heating capacity DHW-only $\phi_{W,hp,sngl}$ [kW]	0,7	1,05	1,4

NOTE 1 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.

$$\Phi_{W,hp} = \frac{Q_{W,gen,out}}{t_{tot}} \quad [W] \qquad (D.1)$$

NOTE 2 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)			
Source temperature at test point θ_{sc} [°C]	**-5**	**0**	**5**
COP at sink temperature 35 °C of the heating system COP_{combi} [W/W]		4,09	4,67
COP at sink temperature 45 °C of the heating system COP_{combi} [W/W]	2,7	3,01	3,43
Heating capacity space heating combi at sink temperature 35 °C $\phi_{H,hp,combi}$ [kW]		9,0	10,3
Heating capacity space heating combi at sink temperature 45 °C $\phi_{H,hp,combi}$ [kW]	7,5	8,5	9,7
Heating capacity DHW combi at sink temperature 35 °C $\phi_{W,hp,combi}$ [kW]		1,75	1,83
Heating capacity DHW combi at sink temperature 45 °C $\phi_{W,hp,combi}$ [kW]	1,7	1,89	2,04
Power input for storage stand-by losses $P_{es,combi}$ [W] (EN 255-3)		36	

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

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

$$COP_{combi} = \frac{Q_{HW,gen,out}}{E_{HW,gen,in}} \quad [W/W] \qquad (D.2)$$

Table D.8 gives the parameters of the auxiliary components.

Table D.8 — Power consumption of auxiliary components

Auxiliary components	
Power source pump P_{sc} [W]	120
Power sink pump P_{sk} [W]	54
Power storage loading pump P_{s} [W]	33
Power control and carter heating P_{sby} [W]	10
Fraction of auxiliary energy input loss to the ambiance $k_{gen,aux,ls}$ [-]	0,2
Temperature reduction factor $b_{gen,aux}$ [-]	1

D.1.3 Calculation

D.1.3.1 General

The calculation is accomplished step by step. For each step, calculation of the lowest bin is presented by means of the appropriate equations of the normative part of this standard, while the results of the calculations for the other bins are contained in a summarising table at the end of the step.

NOTE Due to rounding of the results, slight deviations between results presented by the equations and results presented in the summarising tables may occur.

D.1.3.2 Step 1: Calculation of the energy requirements in the bins

D.1.3.2.1 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 meteorological data set. The upper bin limit of the last bin for space heating is the upper outdoor 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 in between 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.1.3.2.2 Calculation of bin time according to Equation (6)

$$t_{\mathrm{j}} = (N_{\mathrm{ho,\theta hlim,j}} - N_{\mathrm{ho,\theta llim,j}}) = 330h - 0h = 330h \quad \text{[h]} \qquad \text{(D.3)}$$

D.1.3.2.3 Calculation of effective bin time according to Equation (7)

$$t_{\mathrm{eff,j}} = t_{\mathrm{eff,j}} \cdot \frac{24\,h - t_{\mathrm{co}}}{24\,h} = 330h \cdot \frac{24h - 3h}{24h} = 289h \quad \text{[h]} \qquad \text{(D.4)}$$

D.1.3.2.4 Calculation of weighting factors and heat energy requirements for space heating operation according to Equation (4), Equation (5) and Table A.1

$$k_{\mathrm{H,j}} = \frac{Q_{\mathrm{H,gen,out,j}}}{Q_{\mathrm{H,gen,out}}} = \frac{DH_{\mathrm{H,\theta\,hlim,j}} - DH_{\mathrm{H,\theta\,llim,j}}}{DH_{\mathrm{H,tot}}} = \frac{DH_{\mathrm{H,\theta\,hlim=-2°C}} - DH_{\mathrm{H,\theta\,llim=-11°C}}}{DH_{\mathrm{H,tot,\theta\,tlh=15°C}}} = \frac{7940°Ch - 0°Ch}{72888°Ch} = 0.11 \qquad \text{[h]} \qquad \text{(D.5)}$$

$$Q_{\mathrm{H,gen,out,j}} = k_{\mathrm{H,j}} \cdot Q_{\mathrm{H,gen,out}} = 20158kWh \cdot \frac{7940°Ch}{72888°Ch} = 2196kWh \qquad \text{[kWh]} \qquad \text{(D.6)}$$

D.1.3.2.5 Calculation of weighting factors and heat energy requirements for domestic hot water operation according to Equation (8) and Equation (9)

$$k_{\mathrm{W,j}} = \frac{t_{\mathrm{j}}}{t_{\mathrm{tot}}} = \frac{330h}{8760h} = 0.04 \qquad \text{[-]} \qquad \text{(D.7)}$$

$$Q_{\mathrm{W,gen,out,j}} = k_{\mathrm{W,j}} \cdot Q_{\mathrm{W,gen,out}} = 3300kWh \cdot \frac{330h}{8760h} = 124kWh \qquad \text{[kWh]} \qquad \text{(D.8)}$$

D.1.3.2.6 Calculation for all bins

Table D.9 summarises the calculation of energy requirements 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 [°C] (5.3.1)	**- 7**	**2**	**7**	**20**	
Lower bin limit [°C] (5.3.1)	- 11	- 2	4	15	
Upper bin limit [°C] (5.3.1)	- 2	4	15	35	
Bin time [h] (Equation (6))	330	1 604	3 603	3 223	**8 760**
Effective bin time [h] (Equation (7))	289	1 404	3 153	2 820	**7 665**
Weighting factor SH [-] (Equation (4))	0,11	0,40	0,49		**1,00**
Heat energy requirement for SH [kWh] (Equation (5))	2 196	8 133	9 829		**20 158**
Weighting factor DHW [-] (Equation (8))	0,04	0,18	0,41	0,37	**1,00**
Heat energy requirement for DHW [kWh] (Equation (9))	124	604	1 357	1 214	**3 300**

D.1.3.3 Step 2: Calculation of heating capacity and *COP* at full load (see 5.3.5)

D.1.3.3.1 Calculation of source and sink temperature at operating points

Calculation of source temperature (Polynomfit of sample profile, see F.2.1.3, 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.

$$\theta_{\text{so,in}} = 0.15 \cdot \theta_{\text{oa}} + 1.5 = 0.15 \cdot (-7°\text{C}) + 1.5 = 0.5°\text{C} \qquad [°\text{C}] \qquad (\text{D.9})$$

Calculation of flow temperature at test points (heating curve, see B.1, Equation (B.1)

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

$$\theta_{\text{H,gen,f}} = \theta_{\text{i,des}} + \frac{\theta_{\text{H,gen,fdes}} - \theta_{\text{H,dis,r,des}}}{2} \cdot \frac{\theta_{\text{i,des}} - \theta_{\text{e}}}{\theta_{\text{i,des}} - \theta_{\text{e,des}}} + (\theta_{\text{avg}} - \theta_{\text{i,des}}) \cdot (\frac{\theta_{\text{i,des}} - \theta_{\text{e}}}{\theta_{\text{i,des}} - \theta_{\text{e,des}}})^{1/n} \qquad [°\text{C}] \qquad (\text{D.10})$$

$$\theta_{\text{H,gen,f}} = 20°C + \frac{55°C - 45°C}{2} \cdot \frac{20°C - (-7°C)}{20°C - (-8°C)} + (50°C - 20°C) \cdot (\frac{20°C - (-7°C)}{20°C - (-8°C)})^{1/1.2} = 53.9°C \qquad [°\text{C}]$$

NOTE For the lower range of outdoor air temperatures below θ_e = - 8 °C, the required flow temperature exceeds the operation limit temperature of $\theta_{hp,opr}$ = 55 °C. In this case, back-up energy due to the operation limit would have to be considered. However, in this case, the back-up is already in operation due to lack of heating capacity of the heat pump in the lower temperature range (see Table D.12), implying that temperature requirement can be met.

D.1.3.3.2 Interpolation of heating capacity and *COP* for temperature conditions

Linear interpolation of the heating capacity in direction of the source temperature according to 5.3.5.1.3.

$$\begin{aligned}\phi_{\text{hp,h,sngl}_{(\theta\text{sc,in,W35})}} &= \frac{\phi_{\text{H,hp,sngl}_{(\text{B5/W35})}} - \phi_{\text{H,hp,sngl}_{(\text{B0/W35})}}}{5°C - 0°C} \cdot (\theta_{\text{sc,in}} - 0°C) + \phi_{\text{H,hp,sngl}_{(\text{B0/W35})}} \\ &= \frac{10.3 - 9.0}{5°\text{C}} \cdot 0.5°\text{C} + 9.0\text{kW} = 9.1\text{kW}\end{aligned} \qquad [\text{W}] \qquad (\text{D.11})$$

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

$$\begin{aligned}\phi_{\text{hp,h,sngl}_{(\theta\text{sc,in},\theta\text{sk,out})}} &= \frac{\phi_{\text{H,hp,sngl}_{(\theta\text{sc,in/W45})}} - \phi_{\text{H,hp,sngl}_{(\theta\text{sc,in/W35})}}}{45°C - 35°C} \cdot (\theta_{\text{sk,out}} - 0°C) + \phi_{\text{H,hp,sngl}_{(\theta\text{sc,in/W35})}} \\ &= \frac{8.8\text{kW} - 9.1\text{kW}}{10°\text{C}} \cdot 19°\text{C} + 9.1\text{kW} = 8.5\text{kW}\end{aligned} \qquad [\text{W}] \qquad (\text{D.12})$$

D.1.3.3.3 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 occur, and 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 according to Equation (10) after rearranging for the mass flow rate.

$$m'_{\text{standard}} = \frac{\phi_{\text{H,hp,sngl(B0/W45)}}}{\Delta\theta_{\text{standard}} \cdot c_{\text{w}}} = \frac{8.7kW}{5K \cdot 4.182kJ/kg/K} = 0.42kg/s \qquad \text{[kg/s]} \qquad \text{(D.13)}$$

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 Equation (10).

$$\Delta\theta_{\text{standard}} = \frac{\phi_{\text{H,hp,sngl(B-5/W45)}}}{m'_{\text{standard}} \cdot c_{\text{w}}} = \frac{7.7kW}{0.42kg/s \cdot 4.182kJ/kg/K} = 4.4K \qquad \text{[°C]} \qquad \text{(D.14)}$$

Mass flow rate in operation (evaluated at design outdoor temperature $\theta_{\text{e,des}}$) according to Equation (10)

$$m'_{\text{opr}} = \frac{\phi_{\text{H,hp,sngl}(\theta_{\text{e,des}},\theta_{\text{sk,out}})}}{\Delta\theta_{\text{e,des}} \cdot c_{\text{w}}} = \frac{8.5kW}{10K \cdot 4.182kJ/kg/K} = 0.2kg/s \qquad \text{[kg/s]} \qquad \text{(D.15)}$$

Temperature spread with effective mass flow rate at test point according to Equation (10)

$$\Delta\theta_{\text{opr}} = \frac{\phi_{\text{H,hp,sngl(B-5/W45)}}}{m'_{\text{opr}} \cdot c_{\text{w}}} = \frac{7.7kW}{0.2kg/s \cdot 4.182kJ/kg/K} = 9.2K \qquad \text{[°C]} \qquad \text{(D.16)}$$

Correction for different temperature spreads during testing and in operation according to Equation (11)

Since the condenser and the evaporator use brine and water respectively, both average temperature differences between refrigerant and heat transfer medium are set to $\Delta T_{\text{sk}} = \Delta T_{\text{sc}} = 4$ K.

$$COP_{\Delta\theta} = COP_{\text{standard}} \cdot \left[1 - \frac{\frac{\Delta\theta_{\text{standard}} - \Delta\theta_{\text{opr}}}{2}}{\left\{T_{\text{sk}} - \frac{\Delta\theta_{\text{standard}}}{2} + 4[K] - (T_{\text{sc}} - 4[K])\right\}}\right] \qquad \text{[W/W]} \qquad \text{(D.17)}$$

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

$$= 3.0 \cdot 1.04 = 3.1$$

D.1.3.3.4 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.1.3.3.5 *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 is corrected by the method of constant exergetic efficiency given in Annex C. Correction factors are calculated with the Equation (C.5). The equation can also be applied for the correction of different sink temperatures in testing and operation, e.g. testing at 55 °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.

$$f_{\mathrm{T}} = \frac{COP_{\mathrm{crnt,opr}}}{COP_{\mathrm{crnt,standard}}} = \frac{T_{\mathrm{sk,out,opr}} \cdot (\theta_{\mathrm{sk,out,standard}} - \theta_{\mathrm{sc,in,standard}})}{T_{\mathrm{sk,out,standard}} \cdot (\theta_{\mathrm{sk,out,opr}} - \theta_{\mathrm{sc,in,opr}})} \qquad [-] \qquad (D.18)$$

$$= \frac{(273.15 + 60)K \cdot (60 - 0)K}{(273.15 + 60)K \cdot (60 - 0.5)K} = 1.01$$

The COP_t for the operating conditions is thus calculated by the equation

$$COP_{\mathrm{t}} = COP_{\mathrm{t,standard}} \cdot f_{\mathrm{T}} = 2.36 \cdot 1.01 = 2.38 \qquad [W/W] \qquad (D.19)$$

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.1.3.3.6 *COP* and heating capacity for simultaneous operation

The *COP* and heating capacity for combined operation are calculated as done for the space heating mode. For this system, the heating capacity in SH-only and simultaneous operation does not change significantly, and 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.

D.1.3.3.7 Calculation for all bins

Table D.10 summarises the calculation of heating capacity and *COP* at full load for all bins.

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, i.e. the return temperature of the space heating system (see Figure 3) in comparison to the ground source.

Table D.10 — Summary of results step 2: Calculation of heating capacity and *COP* 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, F.2.1.3)	0,5	1,8	2,6	4,5
Interpolation heating capacity for supply 35 (5.3.5.1.3)	9,1	9,5	9,7	10,2
Interpolation heating capacity for supply 45 (5.3.5.1.3)	8,8	9,1	9,3	9,8
Flow temperature at operating points (heating curve, B.1)	54	44	38	
Heating capacity SH-only at supply temperature (5.3.5)	**8,5**	**9,2**	**9,6**	
Mass flow at standard rating point according to EN 14511 (B0/W45)[kg/s]		0,42		
Temperature spread at test points 35 [K] (Equation (10))		5,2	5,9	
Temperature spread at test points 45 [K] (Equation (10))	4,4	5,0	5,7	
Source temperature at design conditions [°C] (F.2.1.3)	0,3			
Flow temperature at design conditions [°C] (B.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] (Equation (10))	0,2			
Temperature spread in operation 35 [K] (Equation (10))	9,3	10,6	12,2	
Temperature spread in operation 45 [K] (Equation (10))	9,1	10,3	11,7	
Correction factor for the temperature spread 35 [-] (Equation (11))	1,05	1,07	1,09	
Correction factor for temperature spread 45 [-] (Equation (11))	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 (5.3.5.1.3)	4,8	5,1	5,2	5,6
Interpolation *COP* source for sink temperature 45 (5.3.5.1.3)	3,7	3,9	4,0	4,2
Interpolation of *COP* SH-only (sink temperature) (5.3.5.1.3)	**2,7**	**4,0**	**4,8**	
DHW-only				
Correction factor of the *COP* for DHW (Annex C, Equation (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 (5.3.5.1.3)	**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.1.3.4 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.1.3.5 Step 4: Calculation of generation subsystem envelope thermal losses

D.1.3.5.1 Calculation for heat pump envelope

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

D.1.3.5.2 Calculation for the heating buffer storage

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

The average storage temperature $\theta_{\mathrm{s,avg,i}}$ is approximated as average temperature of the flow and return temperature according to Equation (16)

$$\theta_{\mathrm{H,st,avg,j}} = \frac{\theta_{\mathrm{H,gen,f,j}} - \theta_{\mathrm{H,dis,r,j}}}{2} = \frac{54°C + (54°C - 10°C)}{2} = 49°C \qquad [°C] \qquad (D.20)$$

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 (see 5.3.7.1.1).

<u>Storage losses of the heating buffer storage according to Equation (14)</u>

$$Q_{\mathrm{H,st,ls,j}} = \frac{\theta_{\mathrm{H,st,avg,j}} - \theta_{\mathrm{H,st,amb,j}}}{\Delta\theta_{\mathrm{st,sby}}} \cdot \frac{Q_{\mathrm{st,sby}} \cdot 1000 \cdot t_{\mathrm{j}}}{24} \qquad [J] \qquad (D.21)$$

$$= \frac{(49-15)K}{45K} \cdot 2.3(kWh/24h) \cdot \frac{330h}{24h} = 23.9kWh$$

<u>Thermal losses of the piping between the generator and the heating buffer storage (see 5.3.7.1.3, according to Equation A.1 of EN 15316-3-3:2007)</u>

$$Q_{\mathrm{H,pi,ls,j}} = \frac{1}{1000} \cdot U_{\mathrm{lj}} \cdot L_{\mathrm{j}} \cdot (\theta_{\mathrm{sk,out,j}} - \theta_{\mathrm{gen,amb}}) \cdot t_{\mathrm{H,hp,on}} = \frac{1}{1000} \cdot U_{\mathrm{lj}} \cdot L_{\mathrm{j}} \cdot (\theta_{\mathrm{sk,out,j}} - \theta_{\mathrm{gen,amb}}) \cdot \frac{Q_{\mathrm{H,gen,out,j}} + Q_{\mathrm{H}}}{\phi_{\mathrm{H,hp,sngl,j}}} \qquad [kWh] \qquad (D.22)$$

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

<u>Total thermal losses of the space heating buffer storage</u>

$$Q_{\mathrm{H,gen,ls,j}} = Q_{\mathrm{H,st,ls,j}} + Q_{\mathrm{H,pi,ls,j}} \qquad [kWh] \qquad (D.23)$$

$$= 23.9kWh + 20.4kWh = 44.3kWh$$

D.1.3.5.3 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/24 h for a temperature difference of 45 °C. Corrected losses for the temperature conditions can be calculated according to Equation (14).

Average storage temperature in DHW storage

The average storage temperature is evaluated as follows: The heat exchanger is positioned at the lower third of the storage. Around the heat exchanger, the average temperature is 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 5.3.7.1.2.

$$\theta_{\mathrm{W,st,avg,j}} = 0.67 \cdot \theta_{\mathrm{W,st,out}} + 0.33 \cdot \theta_{\mathrm{W,st,hx}} \qquad [°C] \qquad (D.24)$$

$$= (0.67 \cdot 60°C + 0.33 \cdot 30°C) = 50°C$$

Storage losses of the DHW storage according to Equation (14)

$$Q_{\mathrm{W,st,ls,j}} = \frac{\theta_{\mathrm{W,st,avg,j}} - \theta_{\mathrm{W,st,amb,j}}}{\Delta\theta_{\mathrm{st,sby}}} \cdot \frac{Q_{\mathrm{st,sby}} \cdot 1000 \cdot t_{\mathrm{j}}}{24} \qquad [J] \qquad (D.25)$$

$$= \frac{(50-15)K}{45K} \cdot 2.6(kWh/24h) \cdot \frac{330h}{24h} = 27.8kWh$$

Thermal losses of the piping between the generator and the DHW storage (see 5.3.7.1.3, according to Equation A.1 of EN 15316-3-3:2007)

$$Q_{\mathrm{W,pi,ls,j}} = \frac{1}{1000} \cdot U_{\mathrm{lj}} \cdot L_{\mathrm{j}} \cdot (\theta_{\mathrm{W,st,out,j}} - \theta_{\mathrm{W,st,amb}}) \cdot \frac{Q_{\mathrm{W,gen,out,j}} + Q_{\mathrm{W,st,ls,j}}}{\phi_{\mathrm{W,hp,sngl,j}}} \qquad [kWh] \qquad (D.26)$$

$$= \frac{1}{1000} \cdot 0.2W/mK \cdot 10m \cdot (60-15)K \cdot \frac{(124+28)kWh}{1.1kW} = 12.4kWh$$

Total thermal losses of the DHW storage (see 5.3.7.1.3)

$$Q_{\mathrm{W,gen,ls,j}} = Q_{\mathrm{W,st,ls,j}} + Q_{\mathrm{W,pi,ls,j}} \qquad [kWh] \qquad (D\ 27)$$

$$= 27.8kWh + 12.4kWh = 40.2kWh$$

D.1.3.5.4 Total envelope thermal losses of the generation subsystem

$$Q_{\mathrm{HW,gen,ls,j}} = Q_{\mathrm{H,gen,ls,j}} + Q_{\mathrm{W,gen,ls,j}} \qquad [kWh] \qquad (D.28)$$

$$= 44kWh + 40kWh = 84kWh$$

D.1.3.5.5 Calculation for all bins

Table D.11 summarises the calculation of total envelope thermal losses for all bins.

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 envelope thermal 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 operating points (5.3.7.1.1)	44	38	34		
Average storage temperature heating buffer [°C] (Equation (16))	49	41	36		
Storage losses heating buffer storage [kWh] (Equation (14))	24	88	160		**272**
Thermal losses piping for heating buffer [kWh] (EN 15316-3-3:2007, Equation A.1)	20	52	48		**120**
Total thermal losses heating buffer [kWh] (5.3.7.1.3 and Equation (15))	44	140	208		**392**
DHW					
Average storage temperature DHW storage (5.3.7.1.2) [°C]	50	50	50	50	
Storage losses DHW storage [kWh] (Equation (14))	28	136	304	272	**740**
Thermal losses piping for DHW storage [kWh] (EN 15316-3-3:2007, Equation A.1)	12	57	122	98	**289**
Total thermal losses DHW storage [kWh] (5.3.7.1.3 and Equation (15))	40	192	426	370	**1 028**
Total envelope thermal losses of the generator subsystem	**84**	**332**	**634**	**370**	**1 420**

D.1.3.6 Step 5: Calculation of the back-up energy

D.1.3.6.1 Back-up energy due to temperature operation limit of the heat pump

Space heating

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.

Domestic hot water

The fraction of back-up heater operation for DHW due to the operation limit of the heat pump is calculated according to Equation (18).

$$k_{\mathrm{W,bu,opr,j}} = \frac{Q_{\mathrm{W,bu,opr,j}}}{Q_{\mathrm{W,gen,out,j}}} = \frac{(\theta_{\mathrm{w,out}} - \theta_{\mathrm{hp,op}\,r})}{(\theta_{\mathrm{w,out}} - \theta_{\mathrm{w,in}})} \quad [-] \qquad \text{(D.29)}$$

$$= \frac{(60-55)K}{(60-15)K} = 0.11$$

D.1.3.6.2 Back-up energy due to lack of heating capacity of the heat pump

For this calculation example, the detailed calculation method for determining the back-up energy due to lack of heating capacity of the heat pump is applied, see 5.3.9.4.

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 $Q_{H,hp}$ [kWh] (Equation (29))	Energy to be produced for DHW $Q_{W,hp}$ [kWh] (Equation (29))	Heating capacity SH $\phi_{H,hp,sngl}$ [kW]	Heating capacity DHW $\phi_{W,hp,sngl}$ [kW]	Heating capacity combined SH $\phi_{H,hp,combi}$ [kW]	Heating capacity combined DHW $\phi_{W,hp,combi}$ [kW]	Running time combined $t_{hp,on,combi}$ [h] (Equation (31), (34))	Running time SH-only $t_{H,hp,on}$ [h] (Equation (28))	Running time DHW-only $t_{W,hp,on}$ [h] (Equation (28))	Total required running time $t_{hp,on,tot}$ [h] (Equation (30))	Effective bin time (1 K bin) t_{eff} [h] (Equation (7))	Diff. between required running time and effective bin time [h]	Required back-up energy $Q_{bu,cap}$ [kWh] (Equation (40))
- 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 Equation (40)

$$Q_{bu,cap,j} = (t_{HW,hp,on,j} - t_{eff,j}) \cdot \phi_{hp,sngl,j} \qquad \text{[kWh]} \qquad (D.30)$$

$$= (3.6h - 2.6h) \cdot 8.3kW = 8.3kWh$$

D.1.3.6.3 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 Equation (41)

$$k_{H,bu,j} = \frac{Q_{H,bu,opr,j} + Q_{H,bu,cap,j}}{Q_{H,gen,out,j}} \qquad \text{[-]} \qquad (D.31)$$

$$= \frac{132kWh}{2196kWh} = 0.06$$

D.1.3.6.4 Calculation for all bins

Table D.13 summarises the calculation of back-up energy for all bins.

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 (Equation (17))	0	0	0	0
Fraction back-up SH due to lack of capacity (Equation (40))	0,06	0	0	0
Total fraction back-up space heating (Equation (41))	**0,06**	**0,000**	**0,000**	**0,000**
Fraction back-up DHW due to operation limit temperature (Equation (18))	0,11	0,11	0,11	0,11
Fraction of back-up DHW due to lack of capacity (Equation (40))	0	0	0	0
Total fraction back-up DHW (Equation (41))	**0,11**	**0,11**	**0,11**	**0,11**

D.1.3.7 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 requirements.

Running time for DHW operation is thus calculated with the DHW heating capacity in simultaneous operation according to Equation (32), taking into account also the thermal losses of the piping between the generator and the DHW-storage.

$$t_{\mathrm{W,hp,on,j}} = \frac{Q_{\mathrm{W,hp,j}}}{\phi_{\mathrm{W,hp,combi,j}}} = \frac{(Q_{\mathrm{W,gen,out,j}} + Q_{\mathrm{W,st,ls,j}} + Q_{\mathrm{W,pi,ls,j}}) \cdot (1 - k_{\mathrm{W,bu,opr,j}})}{\phi_{\mathrm{W,hp,combi,j}}} = \frac{(124kWh + 28kWh + 12kWh) \cdot (1 - 0.11)}{1.93kW} = 76h \quad \text{[s]} \quad \text{(D.32)}$$

Similarly, the running time for space heating operation is calculated with the SH heating capacity in simultaneous operation according to Equation (33), taking into account also the thermal losses of the piping between the generator and the heating buffer storage.

$$t_{\mathrm{H,hp,on,j}} = \frac{Q_{\mathrm{H,hp,j}}}{\phi_{\mathrm{H,hp,combi,j}}} = \frac{(Q_{\mathrm{H,gen,out,j}} + Q_{\mathrm{H,st,ls,j}} + Q_{\mathrm{H,pi,ls,j}}) \cdot (1 - k_{\mathrm{H,bu,cap,j}})}{\phi_{\mathrm{H,hp,combi,j}}} = \frac{(2196kWh + 24kWh + 20kWh) \cdot (1 - 0.06)}{8.5kW} = 247h \quad \text{[s]} \quad \text{(D.33)}$$

And therefore the maximum running time in simultaneous operation follows according to Equation (31).

$$t_{\mathrm{HW,hp,on,combi,max,j}} = \min(t_{\mathrm{H,hp,on,j}}, t_{\mathrm{W,hp,on,j}}) = \min(247h; 76h) = 76h \quad \text{[s]} \quad \text{(D.34)}$$

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 Equation (34).

$$t_{\text{hp,on,combi,j}} = f_{\text{combi}} \cdot t_{\text{hp,on,combi,max,j}} \qquad \text{[s]} \qquad \text{(D.35)}$$
$$= 76h \cdot 0.7 = 53h$$

The DHW-energy produced in simultaneous operation is calculated according to Equation (35)

$$Q_{\text{W,hp,combi,j}} = \phi_{\text{W,hp,combi,j}} \cdot t_{\text{W,hp,on,combi,j}} \qquad \text{[J]} \qquad \text{(D.36)}$$
$$= 53h \cdot 1.93kW = 102kWh$$

The rest of the DHW-energy is produced in single operation and is determined according to Equation (36)

$$Q_{\text{W,hp,sngl,j}} = Q_{\text{W,hp,j}} - Q_{\text{W,hp,combi,j}} \qquad \text{[J]} \qquad \text{(D.37)}$$
$$= 146kWh - 102kWh = 44kWh$$

The space heating energy produced in simultaneous operation is calculated analogously according to Equation (35)

$$Q_{\text{H,hp,combi,j}} = \phi_{\text{H,hp,combi,j}} \cdot t_{\text{H,hp,on,combi,j}} \qquad \text{[J]} \qquad \text{(D.38)}$$
$$= 53h \cdot 8.5kW = 451kWh$$

The rest of the space heating energy is produced in single operation and is determined according to Equation (36)

$$Q_{\text{H,hp,sngl,j}} = Q_{\text{H,hp,j}} - Q_{\text{H,hp,combi,j}} \qquad \text{[J]} \qquad \text{(D.39)}$$
$$= 2105kWh - 451kWh = 1654kWh$$

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, the DHW energy requirement in simultaneous and DHW-only operation, respectively, can be calculated by subtracting the storage losses, taking into account also the thermal losses of the piping between the generator and the DHW storage, according to Equation (38)

$$Q_{\text{W,hp,out,combi,j}} = Q_{\text{W,hp,combi,j}} - (Q_{\text{W,st,ls,j}} + Q_{\text{W,pi,ls,j}}) \cdot (1 - k_{\text{W,bu,j}}) \cdot f_{\text{combi}} \qquad \text{[J]} \qquad \text{(D.40)}$$
$$= 102kWh - (28 + 12)kWh \cdot (1 - 0.11) \cdot 0.7 = 77kWh$$

and according to Equation (37)

$$Q_{\text{W,hp,out,sngl,j}} = Q_{\text{W,hp,sngl,j}} - (Q_{\text{W,st,ls,j}} + Q_{\text{W,pi,ls,j}}) \cdot (1 - k_{\text{W,bu,j}}) \cdot (1 - f_{\text{combi}}) \qquad \text{[J]} \qquad \text{(D.41)}$$

$$= 44kWh - (28+12)kWh \cdot (1-0.11) \cdot (1-0.7) = 33kWh$$

The running time in DHW-only operation can be calculated by Equation (28)

$$t_{\text{W,hp,on,snglj}} = \frac{Q_{\text{W,hp,sngl,j}}}{\phi_{\text{W,hp,sngl,j}}} \quad \text{[s]} \qquad \text{(D.42)}$$

$$= \frac{44kWh}{1.1kW} = 40h$$

And respectively the running time in space heating-only operation by Equation (28)

$$t_{\text{H,hp,on,snglj}} = \frac{Q_{\text{H,hp,sngl,j}}}{\phi_{\text{H,hp,sngl,j}}} \quad \text{[s]} \qquad \text{(D.43)}$$

$$= \frac{1654kWh}{8.5kW} = 195h$$

The total running time can be calculated by Equation (30)

$$t_{\text{hp,on,tot,j}} = \min(t_{\text{eff,j}}; t_{\text{H,hp,on,sgnl,j}} + t_{\text{W,hp,on,sgnl,j}} + t_{\text{hp,on,combi,j}}) \quad \text{[s]} \qquad \text{(D.44)}$$

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

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 summarises the calculation of the running time and the produced energies in the different operation modes for all bins.

Table D.14 — Summary of results step 6: Calculation of the running time and produced energies 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] (Equation (33))	*247*	*902*	*1 049*	*0*	***2 198***
Running time of the heat pump for DHW [h] (Equation (32))	*76*	*377*	*868*	*1 032*	***2 353***
Max. running time combined (=min(SH;DHW)) [h] (Equation (31))	*76*	*377*	*868*	*1 032*	
Effective running time in combined operation [h] (Equation (34))	*53*	*264*	*607*	*0*	***925***
SH energy produced in combined operation [kWh] (Equation (35))	*451*	*2 422*	*5 814*	*0*	***8 688***
Energy produced in SH-only operation [kWh] (Equation (36))	1 654	5 851	4 223	0	**11 728**
DHW energy produced in combined operation [kWh] (Equation (35))	*102*	*496*	*1 110*	*0*	***1 708***
Energy produced in DHW-only operation [kWh] (Equation (36))	44	212	476	1 408	**2 140**
Running time for SH-only operation [h] (Equation (28))	195	638	442	0	**1 274**
Running time for DHW-only operation [h] (Equation (28))	40	181	387	1 032	**1 640**
Fraction of combined operation [-]	*0,70*	*0,70*	*0,70*	*0,00*	
Total running time without considering capacity [h] (Equation (30))	**288**	**1 083**	**1 437**	**1 032**	**3 839**
Corrected total running time by back-up operation [h] (Equation (39))	288	1 083	1 437	1 032	**3 839**
Storage losses in combined operation [kWh] (Equation (38))	*25*	*120*	*265*	*0*	***410***
Storage losses in DHW-only operation [kWh] (Equation (37))	*11*	*51*	*114*	*329*	***505***
DHW energy requirement combined operation [kWh] (Equation (38))	*77*	*376*	*845*	*0*	***1 298***
DHW energy requirement DHW-only [kWh] (Equation (37))	*33*	*161*	*362*	*1 079*	***1 635***

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

D.1.3.8 Step 7: Auxiliary energy

Auxiliary energy can be calculated according to Equation (42)

$$W_{\text{gen,aux}} = \sum_{k} P_{\text{gen,aux,k}} \cdot t_{\text{gen,aux,on,k}} \quad \text{[J]} \qquad \text{(D.45)}$$

The running time of the source pump is determined by the running time of the heat pump and auxiliary energy is calculated by Equation (42)

$$W_{\text{gen,aux,sc}} = P_{\text{gen,aux,sc}} \cdot t_{\text{gen,aux,sc}} = 120W \cdot 288h = 35kWh\,;$$

$$W_{\text{gen,aux,sby}} = P_{\text{gen,aux,sby}} \cdot (t_{\text{j}} - t_{\text{HW,hp,on}}) = 10W \cdot (330h - 288h) = 0.4kWh\,.$$

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

Stand-by operation is only considered during the time when the generator is not running, i.e. bin time reduced by the time of generator operation.

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

Table D.15 summarises the calculation of auxiliary energy for all bins.

Table D.15 — Summary of results step 7: Calculation of the auxiliary energy of the generation subsystem

Step 7: Auxiliary energy					
	Bin 1	Bin 2	Bin 3	Bin 4	Sum
	[- 11..- 2]	[- 2..4]	[4..15]	[15..35]	
Auxiliary energy source pump [kWh] (Equation (42))	35	130	172	124	**461**
Auxiliary energy primary pump [kWh] (Equation (42))	16	58	78	56	**207**
Auxiliary energy stand-by [kWh] (Equation (42))	0,4	5	22	22	**49**
Auxiliary energy DHW storage pump [kWh](incl. in COP_t EN 255-3)	0	0	0	0	**0**
Total auxiliary energy [kWh] (Equation (42))	**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 pumps, for instance, 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

$$\phi_{\mathrm{hp,i}} = \frac{\Delta p_{\mathrm{hp,i}} \cdot V'}{\eta_{\mathrm{gen,aux}}} = \frac{150 mbar \cdot 2m^3/h}{0.3} = \frac{15000 N/m^2 \cdot 2m^3/h}{3600 s/h \cdot 0.3} = 27.8W$$

At the time of writing, efficiency $\eta_{\mathrm{gen,aux}}$ was fixed to 0,3 in EN 14511. The nominal power of the source pump would be 120 W + 28 W = 148 W.

D.1.3.9 Step 8: Generation subsystem total thermal losses and recoverable thermal loss

For mechanical auxiliary components like pumps and fans where thermal losses are partly transferred to the heat transfer medium (considered entirely recovered), the losses to the ambiance are described by a default values of $k_{\mathrm{gen,aux,ls}}$ = 0,2. For auxiliary components that produce heat (electrical devices like controls or transformers and supplementary auxiliary heating devices), it is assumed that the total auxiliary energy is lost to the ambiance, e.g. the default value for $k_{\mathrm{gen,aux,ls}}$ = 1.

Thermal losses to the ambiance from auxiliary component k, e.g. the primary pump, are calculated according to Equation (44)

$$Q_{\mathrm{gen,aux,ls,k,j}} = W_{\mathrm{gen,aux,k,j}} \cdot k_{\mathrm{gen,aux,ls,k}} = 16kWh \cdot 0.2 = 3kWh \quad [\mathrm{J}] \qquad (\mathrm{D.46})$$

The total thermal losses of the generation subsystem are calculated according to Equation (46)

$$Q_{\mathrm{HW,gen,ls,tot,j}} = Q_{\mathrm{HW,gen,ls,j}} + Q_{\mathrm{HW,gen,aux,ls,j}} = \sum_k Q_{\mathrm{gen,ls,k,j}} + Q_{\mathrm{HW,gen,aux,ls,j}} = (44kWh + 40kWh) + 12kWh = 96kWh \quad [\mathrm{J}] \qquad (\mathrm{D.47})$$

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 $b_{\mathrm{gen}} = b_{\mathrm{gen,aux}}$ = 1 for all components.

Thermal losses to the ambiance from auxiliary component k, e.g. the primary pump, are considered recoverable and are calculated according to Equation (45)

$$Q_{\text{HW,gen,aux,ls,rbl,k,j}} = W_{\text{gen,aux,k,j}} \cdot k_{\text{gen,aux,ls,k}} \cdot (1 - b_{\text{gen,aux}}) \quad [\text{J}] \quad (\text{D.48})$$

$$= 16kWh \cdot 0.2 \cdot (1-1) = 0kWh$$

Thermal losses through the generator envelope are considered recoverable and are calculated according to Equation (47)

$$Q_{\text{HW,gen,ls,rbl,j}} = (Q_{\text{H,gen,ls,j}} + Q_{\text{W,gen,ls,j}}) \cdot (1 - b_{\text{gen}}) \quad [\text{J}] \quad (\text{D.49})$$

$$= (44kWh + 40kWh) \cdot (1-1) = 0kWh$$

Table D.16 summarises the calculation of the total thermal losses and the recoverable thermal loss for all bins.

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

Step 8: Calculation of total/recoverable thermal losses					
	Bin 1	Bin 2	Bin 3	Bin 4	Sum
	[- 11..- 2]	[- 2..4]	[4..15]	[15..35]	
Thermal losses from source pump [kWh] (Equation (44))	7	26	34	25	**92**
Thermal losses from primary pump [kWh] (Equation (44))	3	12	16	11	**41**
Thermal losses from storage loading pump [kWh] (Equation (44))	2	7	9	7	**25**
Thermal losses in stand-by operation [kWh] (Equation (44))	0,4	5,2	21,7	21,9	**49**
Total thermal losses auxiliaries to ambiance [kWh] (Equation (44))	**12**	**50**	**81**	**65**	**208**
Total thermal losses generation subsystem [kWh] (Equation (46))	**96**	**382**	**716**	**435**	**1 629**
Recoverable thermal loss from auxiliaries [kWh] (Equation (45))	0	0	0	0	**0**
Recoverable thermal loss through envelope [kWh] (Equation (47))	0	0	0	0	**0**
Total recoverable thermal loss [kWh] (Equation (48)**)**	**0**	**0**	**0**	**0**	**0**

D.1.3.10 Step 9: Calculation of the energy input to the generation subsystem

D.1.3.10.1 Electricity input to the heat pump

The electricity input to the heat pump for space heating can be calculated by summing up the electricity input for SH-only operation and for simultaneous operation according to Equation (49)

$$E_{\text{H,hp,in,j}} = \frac{Q_{\text{H,hp,sngl,j}}}{COP_{\text{H,sngl,j}}} + \frac{Q_{\text{H,hp,combi,j}}}{COP_{\text{H,combi,j}}} \quad [\text{J}] \quad (\text{D.50})$$

$$= \frac{1654kWh}{2.65} + \frac{451kWh}{2.85} = 624kWh + 158kWh = 782kWh$$

The electricity input to the heat pump for domestic hot water can be calculated by summing-up the electricity input for DHW-only operation and for simultaneous operation according to Equation (50)

$$E_{\mathrm{W,hp,in,j}} = \frac{Q_{\mathrm{W,gen,out,sngl,j}}}{COP_{\mathrm{t,sngl,j}}} + P_{\mathrm{es,sngl}} \cdot t_{\mathrm{W,sngl,tot,j}} + \frac{Q_{\mathrm{W,gen,out,combi,j}}}{COP_{\mathrm{t,combi,j}}} + P_{\mathrm{es,combi}} \cdot t_{\mathrm{W,combi,tot,j}} \quad [\mathrm{J}] \qquad (\mathrm{D.51})$$

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

Table D.17 summarises the calculation of energy input to the heat pump to cover the heat requirement for all bins.

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] (Equation (49))	624	1 468	876		**2 967**
Electricity input to heat pump for SH combined [kWh] (Equation (49))	*158*	*632*	*1 307*		***2 097***
Electricity input to heat pump for DHW-only [kWh] (Equation (50))	14	66	147	423	**650**
*Electricity input to heat pump for DHW combined [kWh] (*Equation (50)*)*	*27*	*98*	*190*		***315***
Electricity input to cover storage losses single [kWh] (Equation (50))	6	30	67	200	**303**
*Electricity input to cover storage losses combined [kWh] (*Equation (50)*)*	*8*	*40*	*91*	*0*	***140***
Total electricity input to heat pump [kWh]	**837**	**2 334**	**2 678**	**623**	**6 471**

D.1.3.10.2 Energy input to back-up heater

Electrical energy input to the back-up heater is calculated according to Equation (52)

$$E_{\mathrm{HW,bu,in,j}} = \frac{(Q_{\mathrm{H,gen,out,j}} + Q_{\mathrm{H,gen,ls,j}}) \cdot k_{\mathrm{H,bu,j}}}{\eta_{\mathrm{H,bu}}} + \frac{(Q_{\mathrm{W,gen,out,j}} + Q_{\mathrm{W,gen,ls,j}}) \cdot k_{\mathrm{W,bu,j}}}{\eta_{\mathrm{W,bu}}} \quad [\mathrm{J}] \qquad (\mathrm{D.52})$$

$$= \frac{0.06 \cdot (2196 + 44) kWh}{0.95} + \frac{0.11 \cdot (124 + 40) kWh}{0.95} = 141\ kWh + 19\ kWh = 160\ kWh$$

Table D.18 summarises the calculation of energy input to the back-up heater and total electrical energy input to cover the heat requirement for all bins.

Table D.18 — Summary of results step 9b:
Calculation of the energy input to the back-up heater and total electrical energy input

Step 9: Calculation of energy input to the back-up heater					
	Bin 1	Bin 2	Bin 3	Bin 4	Sum
	[- 11..-2]	[- 2..4]	[4..15]	[15..35]	
Electrical energy input to back-up heater for space heating [kWh] (Equation (52))	141	0	0	0	**141**
Electrical energy input to back-up heater for DHW [kWh] (Equation (52))	19	93	209	185	**506**
Total electrical energy input to back-up heater [kWh]	**160**	**93**	**209**	**185**	**648**
Total electrical energy input to cover the heat requirement [kWh] (Equation (53)**)**	**997**	**2 428**	**2 886**	**808**	**7 119**

D.1.3.11 Step 10: Output values

D.1.3.11.1 Resulting output values

The resulting output values of the calculation are:

— energy input to cover the heat requirement $E_{HW,gen,in}$ according to Equation (53)

$E_{HW,gen,in} = E_{HW,hp,in} + E_{HW,bu,in}$ = 6 471 kWh + 648 kWh = 7 119 kWh;

— total thermal losses of the generation subsystem $Q_{HW,gen,ls,tot}$ according to Equation (46)

$Q_{HW,gen,ls,tot} = Q_{H,gen,ls} + Q_{W,gen,ls} + Q_{HW,gen,aux,ls}$ = 392 kWh + 1 028 kWh + 208 kWh = 1 629 kWh;

— recoverable thermal loss of the generation subsystem $Q_{HW,gen,ls,rbl,tot}$ according to Equation (48)

$Q_{HW,gen,ls,rbl,tot} = Q_{HW,gen,ls,rbl} + Q_{HW,gen,aux,ls,rbl}$ = 0 kWh + 0 kWh = 0 kWh;

— total auxiliary energy to operate the generation subsystem $W_{HW,gen,aux}$ according to Equation (42)

$W_{HW,gen,aux}$ = 717 kWh.

D.1.3.11.2 Optional output values

— total heat energy produced by the back-up heater $Q_{HW,bu}$ according to Equation (52)

$Q_{HW,bu} = E_{HW,bu,in} \cdot \eta_{HW,bu} = Q_{W,bu,opr} + Q_{H,bu,cap}$

= 0,11 · (3 300 + 1 028) kWh + 132 kWh = 608 kWh;

— total heat energy produced by the heat pump system $Q_{HW,hp}$ according to Equation (29)

$Q_{HW,hp} = Q_{H,gen,out} + Q_{H,gen,ls} + Q_{W,gen,out} + Q_{W,gen,ls} - Q_{HW,bu}$

= 20 158 kWh + 392 kWh + 3 300 kWh + 1 028 kWh – 608 kWh = 2 4271 kWh;

— overall seasonal performance $SPF_{HW,gen}$ according to Equation (54)

$SPF_{HW,gen} = (Q_{H,gen,out} + Q_{W,gen,out})/(E_{HW,gen,in} + W_{HW,gen,aux})$

= (20 158 kWh + 3 300 kWh) / (7 119 kWh + 717 kWh) = 3,0;

— overall seasonal performance factor of the heat pump according to Equation (55)

$SPF_{HW,hp} = (Q_{HW,hp}) / (E_{HW,hp,in} + W_{gen,aux,sc} + W_{gen,aux,sby})$

= (24 271 kWh) / (6 471 kWh + 461 kWh + 49 kWh) = 3,5;

— total used ambient heat $Q_{HW,gen,in}$ according to Equation (1)

$Q_{HW,gen,in} = Q_{HW,gen,out} + Q_{HW,gen,ls} - E_{HW,gen,in} - k_{gen,aux,ls,rvd} \cdot W_{HW,gen,aux}$

= (20 158 kWh + 3 300 kWh) + (392 kWh + 1 028 kWh) – 7 119 kWh – 0 kWh = 17 760 kWh.

Table D.19 summarises the resulting and optional output values of the calculation of the generation subsystem.

Table D.19 — Summary of results step 10: Calculation of 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 $E_{HW,gen,in}$ [kWh] (Equation (53))	997	2428	2 886	808	**7 119**
Total thermal losses $Q_{HW,gen,ls,tot}$ [kWh] (Equation (46))	96	382	716	435	**1 629**
Total recoverable thermal loss $Q_{HW,gen,ls,rbl,tot}$ [kWh] (Equation (48))	0	0	0	0	**0**
Total auxiliary energy $W_{HW,gen,aux}$ [kWh] (Equation (42))	51	193	272	202	**717**
Optional output values					
Total heat energy produced by the back-up $Q_{HW,bu}$ [kWh] (Equation (52))					**608**
Total heat energy produced by the heat pump $Q_{HW,hp}$ [kWh] (Equation (29))					**24 271**
Overall SPF of the generation subsystem $SPF_{HW,gen}$ [-] (Equation (54))					**3,0**
Overall *SPF* of the heat pump $SPF_{HW,hp}$ [-] (Equation (55))					**3,5**
Total amount of used ambient heat $Q_{HW,gen,in}$ [kWh] (Equation (1))					**17 760**

D.2 Calculation example (spreadsheet format)

D.2.1 System configuration

Alternate combined operating electrically-driven VCC air-to-water heat pump according to Figure 1 for space heating and domestic hot water production.

Bivalent design with electrical back-up heater and DHW-storage, i.e. storage loading pump included in the system boundary.

D.2.2 Input data for the calculation (according to 5.3.2)

Table D.20 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.21 gives the parameters of the space heating system. Table D.22 gives the parameters of the DHW system. Table D.23 contains general parameters of the heat pump and Table D.24 contains the characteristic of the heat pump for the space heating-only operation mode according to the former test standard EN 255-2.

Table D.20 — Boundary conditions

A	**Meteorological data**	
A1	Site (see Table A.1)	Gelterkinden, BL, CH
A2	Design outdoor temperature $\theta_{e,des}$ [°C]	- 8

Table D.21 — Space heating operation mode

B	**Input data space heating**	
B1	Space heating energy requirement $Q_{H,gen,out}$ [kWh] (according to EN 15316-2-3)	10 960
B2	Indoor design temperature $\theta_{i,des}$ [°C]	20
B3	Type of heat emission system (radiators/convectors/floor/wall/air)	Floor heating
B4	Design flow temperature $\theta_{f,des}$ (at outdoor design temperature $\theta_{e,des}$) [°C]	28,5
B5	Temperature spread of SH system at design conditions $\Delta\theta_{e,des}$ [K]	3
B6	Upper outdoor temperature limit for heating θ_{tlh} [°C]	14
B7	Balance point for space heating operation θ_{bal} [°C]	- 5
B8	Cut-out time for the heat pump operation t_{co} [h/d]	0
B9	Heating buffer storage (y/n)	No
B10	Back-up heater (y/n)	Yes
B11	Type of back-up heater (electrical/gas/oil)	Electrical
B12	Operation mode of back-up heater (alternative/parallel/partly parallel)	parallel
B13	Efficiency of the back-up heater η_{bu}	1

Table D.22 — DHW operation mode

C	Input data DHW	
C1	DHW energy requirement $Q_{W,gen,out}$ [kWh] (according to EN 15316-3-2)	1 179
C2	Cold water inlet temperature $\theta_{W,st,in}$ [°C]	15
C3	Average hot water temperature at storage outlet $\theta_{W,st,out}$ [°C]	48,5
C4	Volume DHW storage $V_{W,st}$ [l]	200
C5	Storage stand-by loss $Q_{W,st,sby}$ [kWh/24 h]	4,2
C6	Temperature difference during storage testing $\theta_{W,st,sby}$ [°C]	40
C7	Ambient temperature of the storage $\theta_{W,st,amb}$ [°C]	15
	Piping losses included in storage stand-by loss value	
	Back-up heater is the same as for the space heating system	

Table D.23 — Heat pump

D	Heat pump	
D1	Type of heat pump	A/W
D2	Type of DHW production (none/only/alternate/simultaneous)	Alternate combined
D3	Control of the heat pump (ON-OFF, step, variable speed)	ON-OFF
D4	Operating limit $\theta_{hp,opr}$ [°C]	55

Table D.24 — Characteristic of the heat pump for space heating-only operation (EN 255-2)

E	Space heating-only (EN 255-2)				
E1	Source temperature at test point θ_{sc} [°C]	**- 7**	**2**	**7**	**20**
E2	Lower sink temperature at test points θ_{sk} [°C]	**35**			
E3	Upper sink temperature at test points θ_{sk} [°C]	**50**			
E4	*COP* at sink temperature 35 °C [W/W]	2,9	3,3	3,6	4,6
E5	*COP* at sink temperature 50 °C [W/W]	3,0	3,5	2,7	3,4
E6	Heating capacity at sink temperature 35 °C $\phi_{H,hp,sngl}$ [kW]	3,4	4,2	4,7	6,5
E7	Heating capacity at sink temperature 50 °C $\phi_{H,hp,sngl}$ kW]	7,7	8,7	9,9	5,7

Table D.25 contains the electrical power input to the auxiliary components and the parameters for the calculation of thermal losses of auxiliary components. The ventilator for the outdoor air source is included in the *COP* values given in Table D.24. The space heating circulation pump is accounted for in the distribution system, since no hydraulic decoupling between the generation and the distribution subsystem is installed.

Table D.25 — Power consumption of auxiliary components

F	Auxiliary components	
F1	Power storage loading pump P_s [W]	33
F2	Power control P_{sby} [W]	10
F3	Fraction of auxiliary energy pump loss to the ambiance $k_{gen,aux,ls}$ [-]	0,2
F4	Temperature reduction factor $b_{gen,aux}$ [-]	1

D.2.3 Calculation

The calculation is accomplished for three bins for space heating and 4 bins for DHW operation. The operating points correspond to the test points given in Table D.24.

For the back-up calculation, the simplified method is used based on the balance point given in Table D.21 and the parallel operation mode. Since design temperatures for both space heating and DHW are below the operation temperature limit of the heat pump, there is no back-up energy due to the operation limit. Since no test results according to EN 255-3 were available for the DHW operation and the heat pump system configuration is of alternate type, the DHW is calculated using the space heating characteristic evaluated at an average sink temperature of the storage loading (see 5.3.5.2).

Values from part load testing according to CEN/TS 14825 were not available either, so no explicit part load correction is done and only the stand-by auxiliary consumption is taken into account.

Table D.26 gives the calculation in spreadsheet format. In the second column, references to equations in the normative part of this standard and references within Table D.26 and to other tables of D.2 are given.

NOTE Due to rounding, the results may deviate slightly.

Table D.26 — Spreadsheet of the calculation according to the bin method

		Equation/reference	Bin 1	Bin 2	Bin 3	Bin 4	Total
Bin energies (see 5.3.4)							
G1	Operating points	according to test points	- 7	2	7	20	
G2	Lower limit	in the middle of test points	- 11	- 2	4	14	
G3	Upper limit	in the middle of test points	- 2	4	14	35	
G4	Weighting factor SH [-]	Equation (4)	0,11	0,41	0,48		**1,00**
G5	Space heating energy [kWh]	G4*B1	1 223	4 528	5 210		**10 960**
G6	Bin time [h]	Equation (6)	330	1 604	3 262	3 564	**8 760**
G7	Effective bin time [h]	Equation (7)	330	1 604	3 262	3 564	**8 760**
G8	Weighting factor DHW [-]	Equation (8)	0,04	0,18	0,37	0,41	**1,00**
G9	DHW energy [kWh]	G8*C1	44	216	439	480	**1 179**
Heat pump characteristic at operating points (see 5.3.5)							
G10	Mass flow test [kg/s]	Equation (10), from manufacturer	0,20				
G11	Mass flow operation [kg/s]	Equation (10)	0,27				
G12	Correction factor *COP* 35 °C [-]	Equation (11)	0,99	0,99	0,98		
G13	Correction factor *COP* 50 °C [-]	Equation (11)	1,00	0,99	0,99		
G14	*COP* corrected 35 °C [W/W]	G12 × E4	2,88	3,22	3,48		
G15	*COP* corrected 50 °C [W/W]	G13 × E5	2,01	2,27	2,68		
G16	Supply temperature [°C]	manufacturer, Equation (B.1)	28,5	26,5	25,2		
G17	*COP* SH [W/W]	interpolation G14, G15 to G16	3,25	3,76	4,00		
G18	Heating capacity SH [kW]	interpolation E6, E7 to G16	3,78	4,88	5,12		
G19	Average loading temperature [°C]	0,95 × C3 (according to B.2)	46				
G20	*COP* DHW [W/W]	interpolation G14, G15 to G19	2,26	2,55	2,93	3,72	
G21	Heating capacity at average loading temp [kW]	interpolation E6, E7 to G19	2,64	3,41	4,14	5,90	
Storage losses (see 5.3.7)							
G22	Average storage temperature [°C]	5.3.7.1.2	43				
G23	DHW storage losses [kWh]	Equation (14)	40	196	400	437	**1 073**
G24	DHW energy produced [kWh]	G9 + G23	85	412	839	916	**2 252**
Back-up energy (see 5.3.8.3, simplified method)							
G25	Fraction back-up energy [-]	Equation (24)	0,025	0	0		
G26	Back-up energy [kWh]	G5 × G25	31	0	0		
Running time (see 5.3.9)							
G27	Running time SH [h]	G5/G18	315	928	1 017		**2 259**
G28	Running time DHW [h]	(G9 + G23)/G21	32	121	202	155	**510**
G29	Total running time [h]	G27 + G28	347	1 049	1 219	155	
G30	Effective running time [h]	MIN(G29; G7)	330	1 049	1 219	155	**2 753**
G31	Additional back-up energy [kWh]	IF(G29 > G30); G18 × (G29 - G30); ELSE 0	64	0	0		**64**
G32	Back-up energy input [kWh]	(G26 + G31)/B13	95	0	0		**95**

Table D.26 — Spreadsheet of the calculation according to the bin method (*continued*)

		Equation/reference	**Bin 1**	**Bin 2**	**Bin 3**	**Bin 4**	**Total**
Auxiliary energy (see 5.3.10)							
G33	Auxiliary stand-by [kWh]	F2 × (G6 - G30)	0	6	20	34	**60**
G34	Auxiliary storage loading [kWh]	F1 × G28	1	4	7	5	**17**
Recoverable thermal losses (see 5.3.11)							
G35	Recoverable thermal losses from auxiliaries [kWh]	(G33 + G34 × F3) × (1 - F4)	0	0	0	0	**0**
G36	Recoverable thermal losses from DHW storage [kWh]	G23 × (1 - F4)	0	0	0	0	**0**
Total electrical driving energy (see 5.3.12)							
G37	Electrical energy input to the heat pump SH [kWh]	(G5 - G26 - G31)/G17	347	1 203	1 302		**2 852**
G38	Electrical energy input to the heat pump DHW [kWh]	(G9 + G23)/G20	38	162	286	246	**732**
G39	Total electrical energy input to the heat pump [kWh]	G37 + G38	384	1 365	1 588	246	**3 583**
Output values (see 5.3.13)							
G40	Electrical energy to cover heat requirement [kWh]	G39 + G32	479	1 365	1 588	246	**3 679**
G41	Total auxiliary energy [kWh]	G33 + G34	1	10	27	39	**77**
G42	Total thermal losses [kWh]	G23 + (G33 + G34 × F3)	40	203	421	472	**1 136**
G43	Total recoverable thermal losses [kWh]	G35 + G36	0	0	0	0	**0**
G44	Used ambient heat [kWh]	G5 + G9 + G23 - G40	828	3 576	4 460	670	**9 534**
G45	Heat energy covered by back-up heater [kWh]	G26 + G31	95	0	0	0	**95**
G46	SPF generation [-]	(G5 + G9)/(G40 + G41)					**3,23**
G47	SPF heat pump [-]	(G5 + G9 + G23 - G45)/(G33 + G39)					**3,60**

NOTE The calculation of the system has been compared to year-round field monitoring values. The difference in the seasonal performance factor between monitoring and calculation is about 6 %, which is in the range of the exactness of the *COP* values.

Annex E
(informative)

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

E.1 General

The following tables constitute an example of a possible national annex containing output tables provided by the Dutch Standardisation Institute 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.2 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.3 References

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

E.4 Heat pump seasonal performance

E.4.1 Residential buildings

For all heat pumps in a residential building, *SPF* and seasonal efficiency are determined using Table E.1, E.2 and E.3.

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

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]
NOTE For η_{el}, $\theta_{flow;design}$ and a, see Table E.2.		

Table E.2 — Gross seasonal heat pump efficiency in residential buildings for single heat pump with index i, second performance level

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} is the efficiency of electricity generation;		
$\theta_{flow;design}$ is the 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	is the air as heat source, with its temperature level during test;	
(y)	is the air as heat source, with its wet bulb temperature level during test;	
B	is the soil as heat source, with its temperature level during test;	
W	is the ground water as heat source, with its temperature level during test;	
B0/W35	is the 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.4.2 Non-residential buildings

For all heat pumps in non-residential buildings, seasonal *COP* and efficiency is determined using Table E.4.

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	$3{,}4 \times \eta_{el}$	1,6	$3{,}1 \times \eta_{el}$	1,5	$2{,}8 \times \eta_{el}$	1,4
Exhaust ventilation air	$6{,}1 \times \eta_{el}$	2,6	$5{,}1 \times \eta_{el}$	2,2	$4{,}4 \times \eta_{el}$	2,0
Groundwater / aquifer	$4{,}7 \times \eta_{el}$	2,1	$4{,}2 \times \eta_{el}$	1,9	$3{,}6 \times \eta_{el}$	1,8
Surface water	$4{,}1 \times \eta_{el}$	1,9	$3{,}7 \times \eta_{el}$	1,8	$3{,}3 \times \eta_{el}$	1,7

where

η_{el} is the efficiency of electricity generation;

$\theta_{flow;design}$ is the system design flow temperature °C;

EHP is the electric heat pump;

GMHP is the 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.5 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):

$$\eta_g = \eta_{hp,1} \qquad [-] \qquad (E.1)$$

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:

$$\beta_{heat} = \frac{P_{hg;pref}}{\Sigma P_{hg;i}} \qquad [-] \qquad (E.2)$$

where

pref is the index of the preferential heat pump;

β_{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}		
Preferential 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 $ß_{heat}$ for non-residential buildings

$ß_{heat}$	f_{pref}	
Preferential 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:

$$\eta_{\mathrm{g}} = \frac{1}{\frac{(1 - f_{\mathrm{pref}})}{\eta_{hg,npref}} + \frac{f_{\mathrm{pref}}}{\eta_{hg;\mathrm{pref}}}} \quad [-] \qquad (E.3)$$

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.6 Heat pump installation energy consumption

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

$$Q_{in;g;prim} = \eta_g \times Q_{in;d} \qquad [MJ] \qquad (E.4)$$

where

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

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

Recoverable losses are zero.

E.7 Heat pump installation auxiliary energy consumption

For heat pump installations in residential buildings, additional primary energy consumption due to pumps and controls is given in Table E.7.

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	$2{,}2 \times A$
	Pump control	$1{,}1 \times A$
System pump in collective heating installation	-	$1{,}1 \times A$
Heat pump controls	-	$0{,}88 \times A$
where		
A is the 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:

$$Q_{in;g;aux;el} = 8 \times f \times A \, / \, \eta_{el} \qquad [MJ] \qquad (E.5)$$

Factor *f* is given in Tabel E.8.

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:

$$Q_{in;g;aux;prim} = Q_{in;g;aux;el} \times 3{,}6 \, / \, \eta_{el} \qquad [MJ] \qquad (E.6)$$

where

η_{el} is the 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)

F.1 General

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. If no European Standard testing exists, national methods are to be used.

F.2 Temperatures

F.2.1 Source temperatures

F.2.1.1 Outdoor air-to-water heat pumps

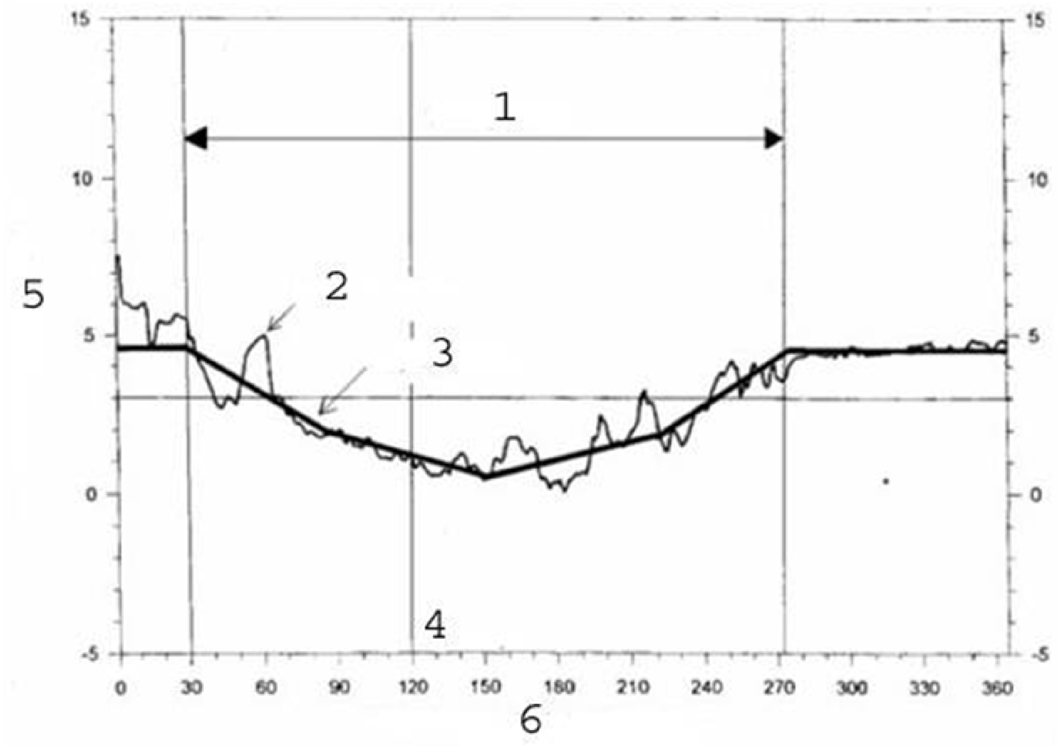
For air-to-water heat pumps, the source temperature corresponds to the outside air temperature.

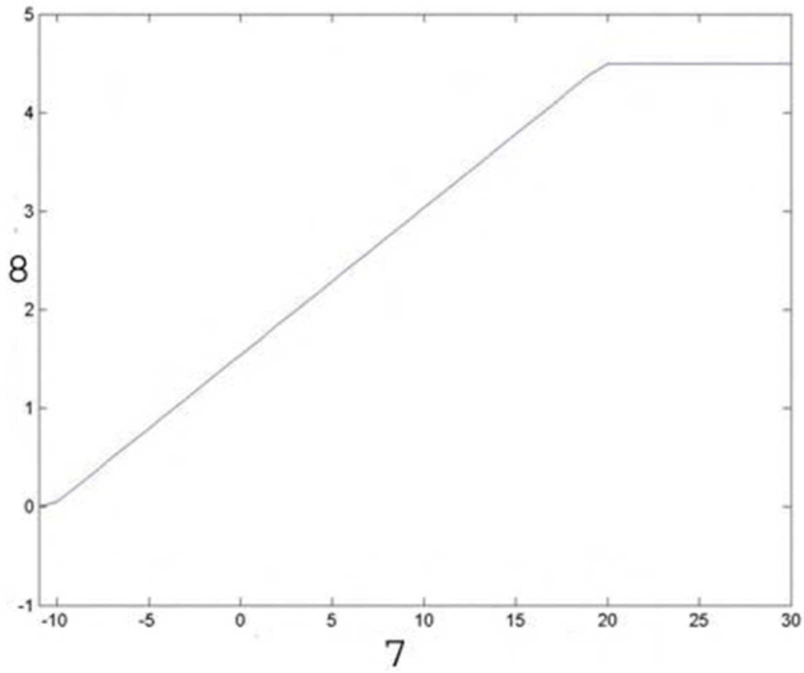
F.2.1.2 Exhaust-air-to-water heat pumps

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

F.2.1.3 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].





Key

1 heating season

2 simulation values

3 approximation]

4 1. January

5 source temperature [°C]

6 time (days from the first September)

7 outdoor air temperature [°C]

8 brine temperature [°C]

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

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

$$\theta_{\mathrm{sc,in}} = \max(0°C, \min(0.15 \cdot \theta_{\mathrm{e}} + 1.5°C, 4.5°C)) \qquad [°C] \qquad (F.1)$$

where

$\theta_{\mathrm{sc,in}}$	is the source temperature at the heat pump evaporator inlet;	[°C]
θ_{e}	is the outdoor air temperature.	[°C]

F.2.1.4 Ground water-to-water heat pumps

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

F.3 Example values for heating capacity and coefficient of performance for electrically driven heat pumps

F.3.1 General

The characteristics provided are only given as example values. When carrying out the calculation, care shall be taken to use – as input – data from standard testing according to EN 14511. If no test data from testing institutes are available, data of the manufacturers can be used for the calculation under their responsibility.

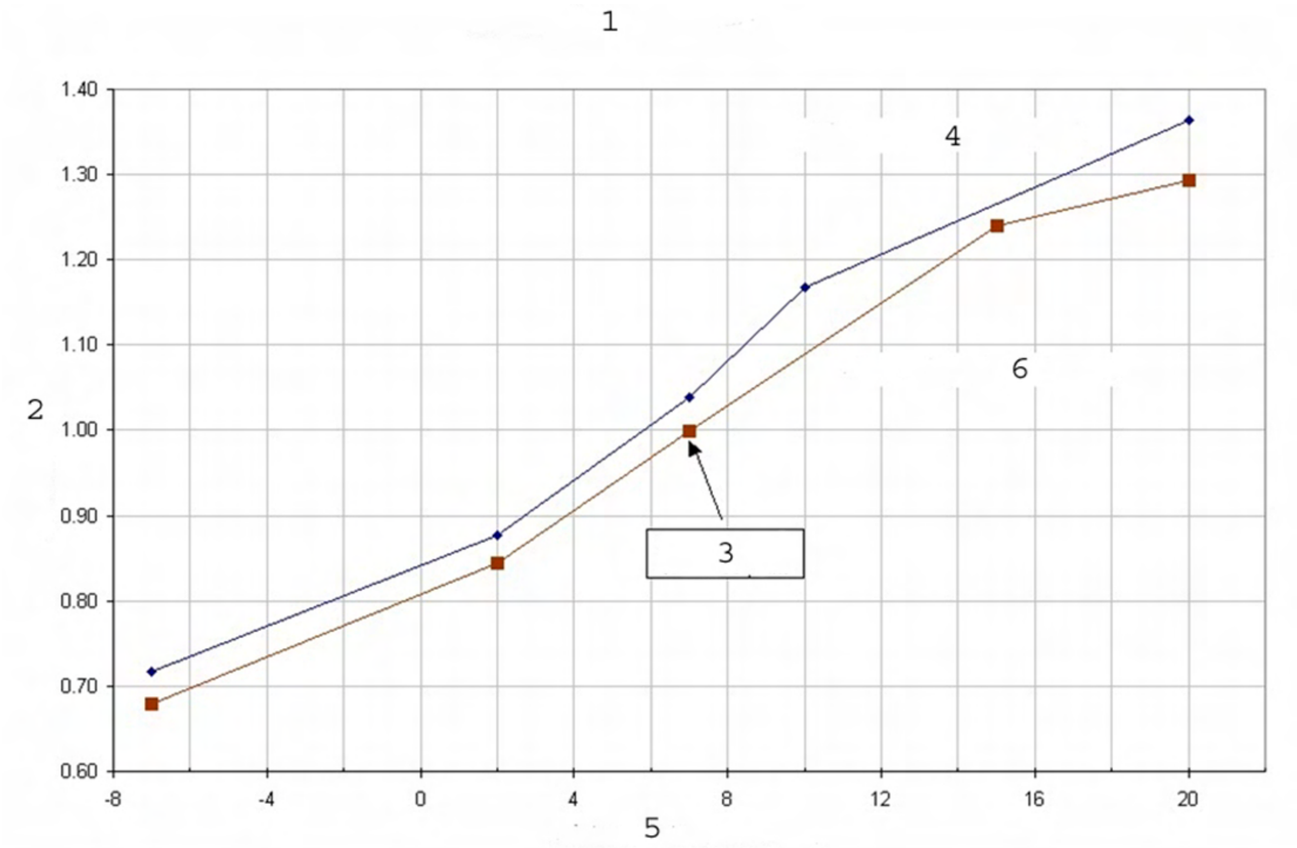
F.3.2 Heating capacity

F.3.2.1 General

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 A/W heat pump A7/W35, for B/W heat pumps B0/W35 and for W/W heat pumps W10/W35.

The following data are based on a statistical analysis of data provided by the Swiss national test centre WPZ (WPZ Bulletin [3]). 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 EN 255-2 are given as examples. Therefore, in the following figures, the reference point is set to T1 according to the former standard EN 255-2, corresponding to A7/W50, B0/W50 and W10/W50.

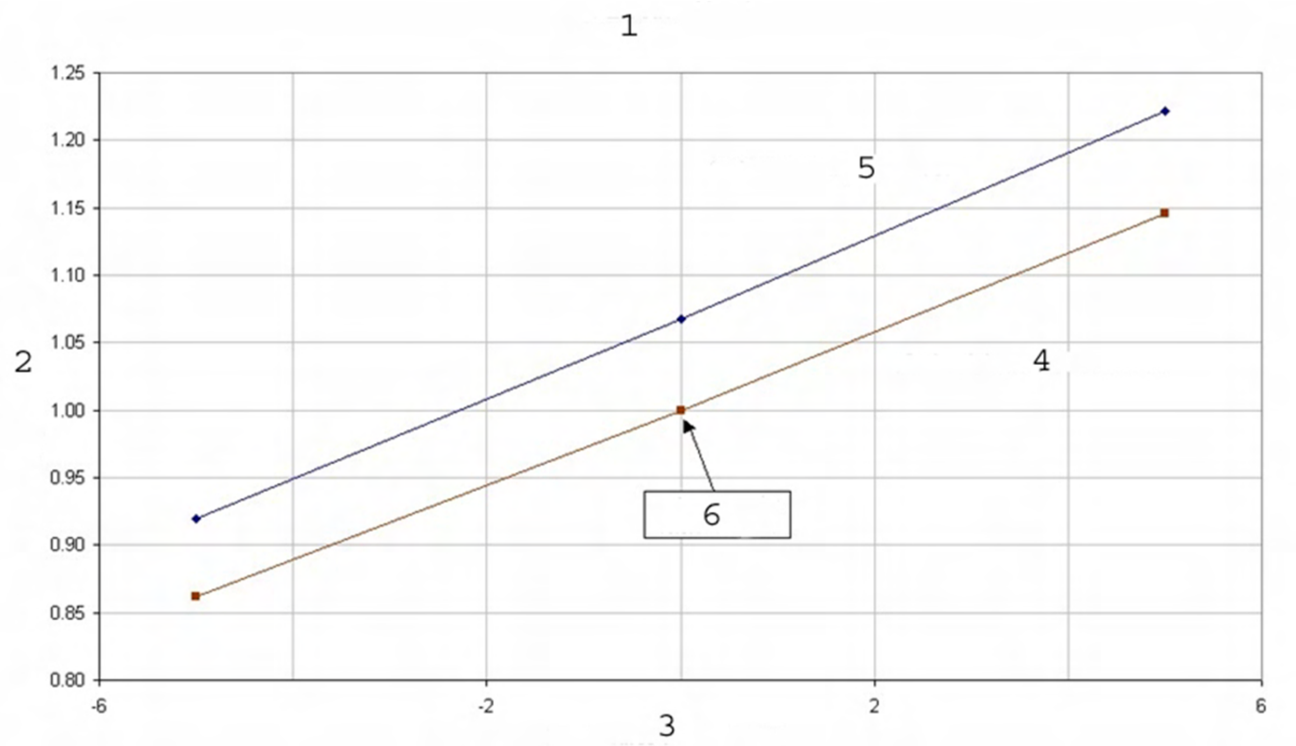
F.3.2.2 Air-to-water heat pumps



Relative heating capacity A/W heat pumps				
Key		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	**7 °C**	1,04	1,00
5	inlet source temperature [°C]	**10 °C**	1,17	-
6	outlet sink temperature: 50 °C	**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 point T1 according to EN 255-2)

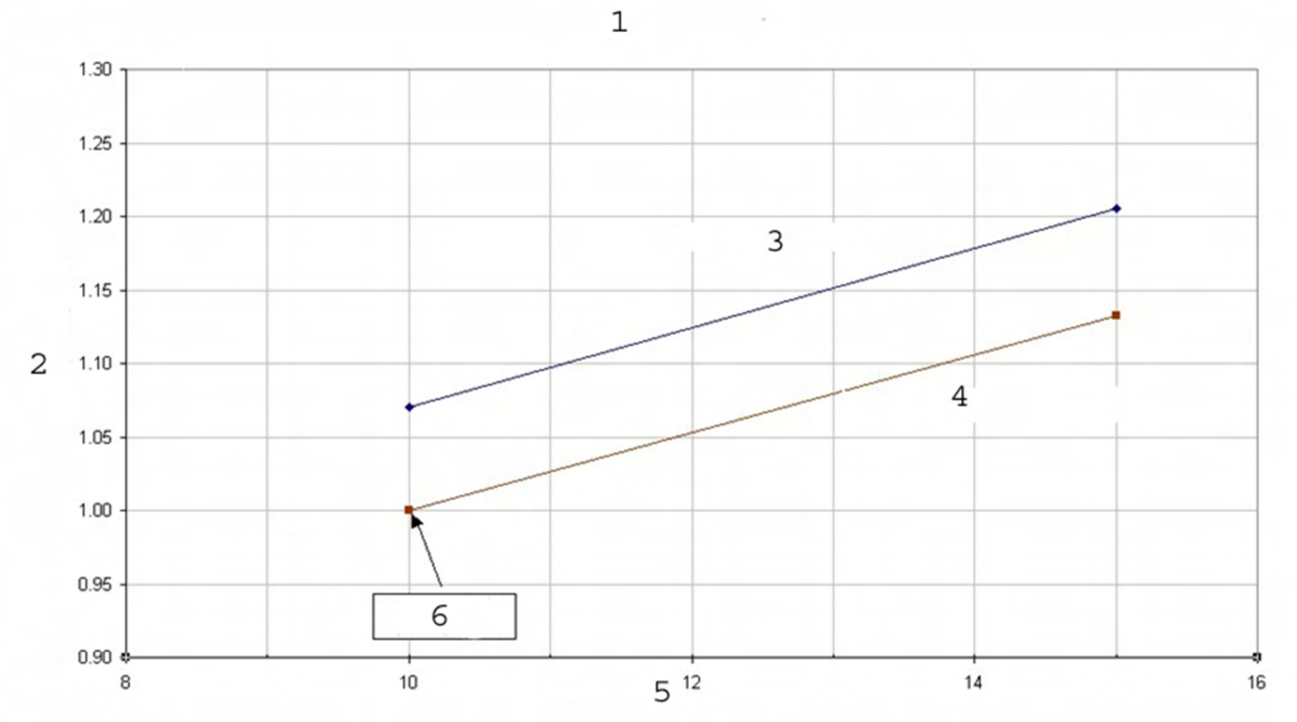
F.3.2.3 Brine-to-water heat pumps



Relative heating capacity B/W heat pumps				
Key		**Inlet source temperature**	**Outlet sink temperature**	
1	brine-to-water electrically-driven heat pump		**35 °C**	**50 °C**
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 to EN 255-2)

F.3.2.4 Water-to-water heat pumps



Relative heating capacity W/W heat pumps				
Key		**Inlet source temperature**	**Outlet sink temperature**	
1	water-to-water electrically-driven heat pumps		**35 °C**	**50 °C**
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)

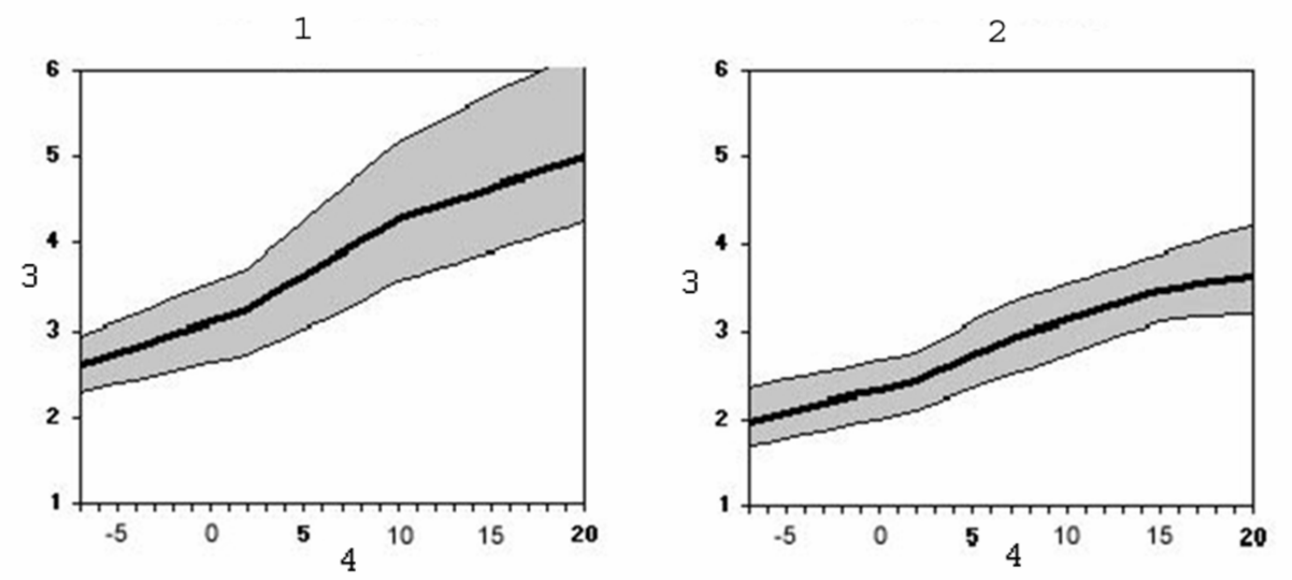
F.3.3 *COP*

F.3.3.1 General

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 – depending on the test point – determines 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.3.3.2 Air-to-water heat pumps



Key

1 sink temperature θ_{sk} = 35 °C

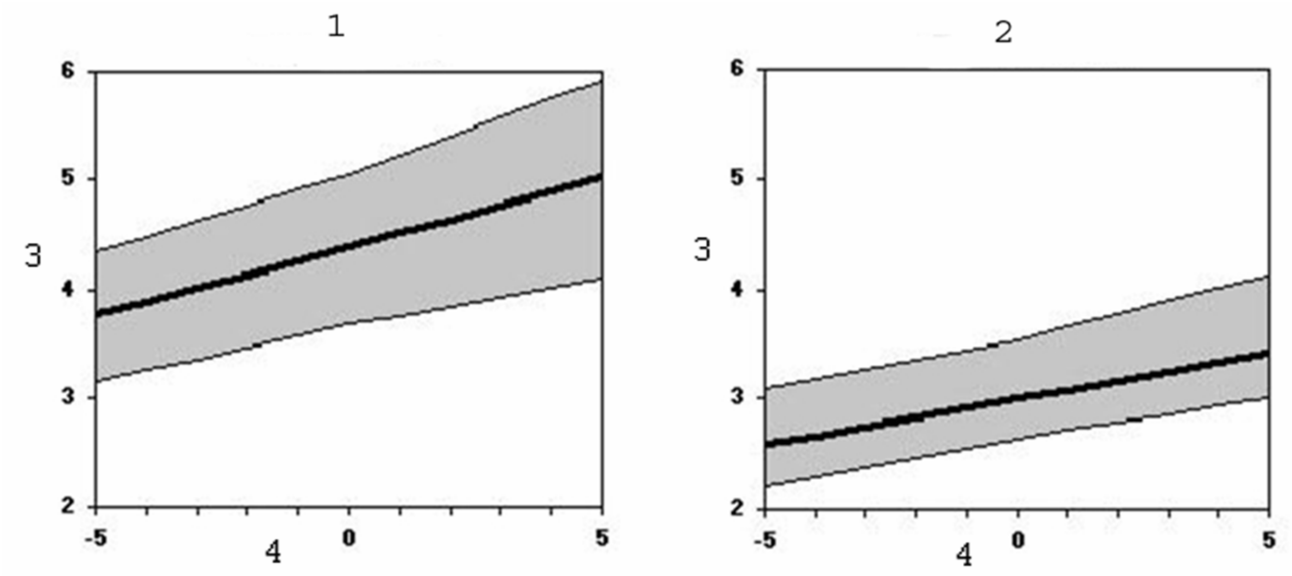
2 sink temperature θ_{sk}= 50 °C

3 *COP*

4 source temperature

Figure F.5 — *COP* values of air-to water electrically-driven heat pumps vs source temperatures (black line – average values, grey area – scatter band of the values)

F.3.3.3 Brine-to-water



Key

1 sink temperature θ_{sk} = 35 °C

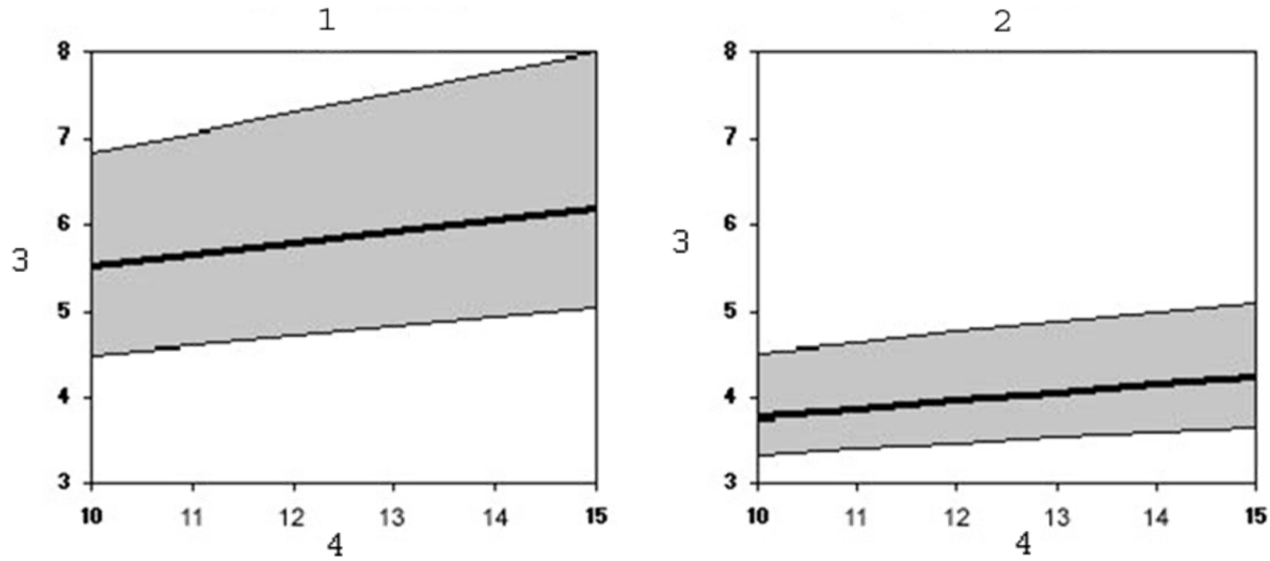
2 sink temperature θ_{sk} = 50 °C

3 *COP*

4 source temperature

Figure F.6 — *COP* values of brine-to water electrical heat pumps vs source temperatures (black line – average values, grey area – scatter band of the values)

F.3.3.4 Water-to-water



Key

1 sink temperature θ_{sk} = 35 °C

2 sink temperature θ_{sk} = 50 °C

3 *COP*

4 source temperature

Figure F.7 — *COP* values of water-to water electrical heat pumps vs source temperatures (black line – average values, grey area – scatter band of the values)

F.4 Gas engine-driven heat pumps

F.4.1 Preface

Characteristics provided here are based on very few measurements. 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.

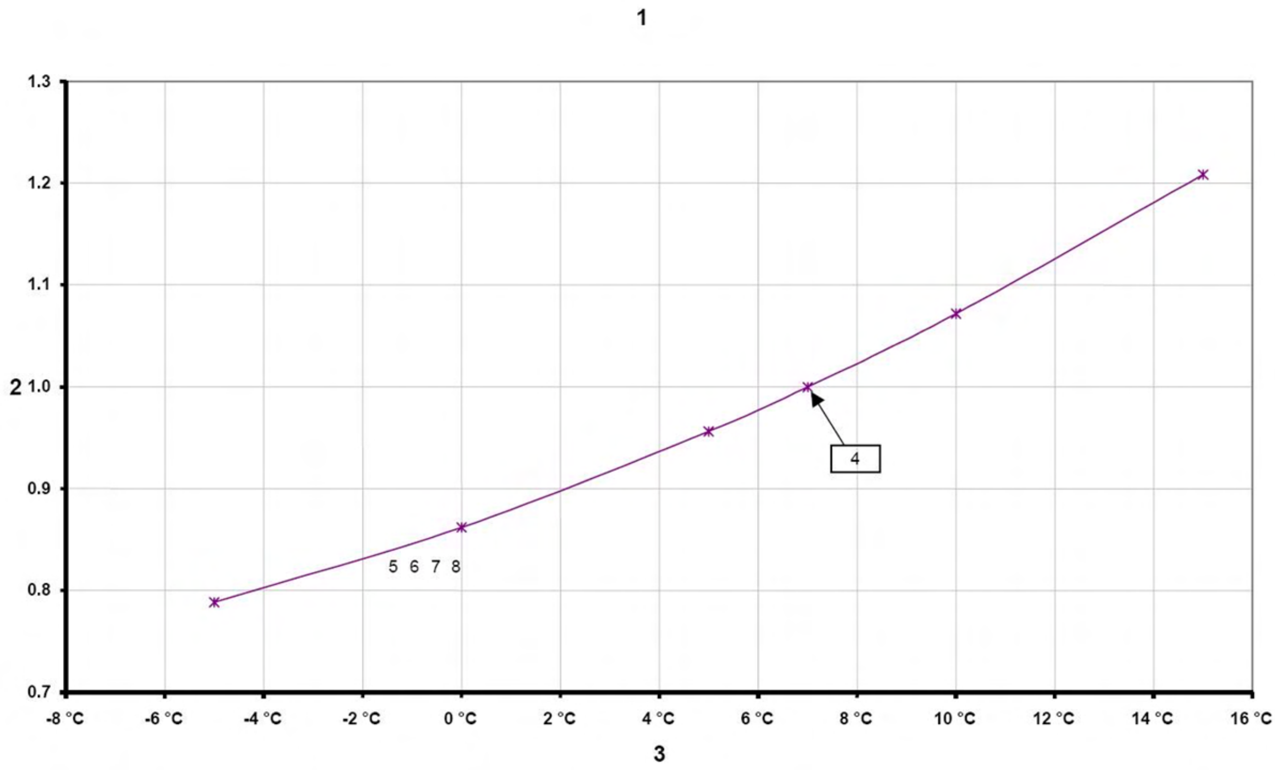
The following data are based on the average values of technical features provided by different manufacturers.

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.4.2 Heating capacity

F.4.2.1 Air-to-water heat pumps

The data refer to the overall heating capacity (heat pump condenser and heat recovery from the engine).

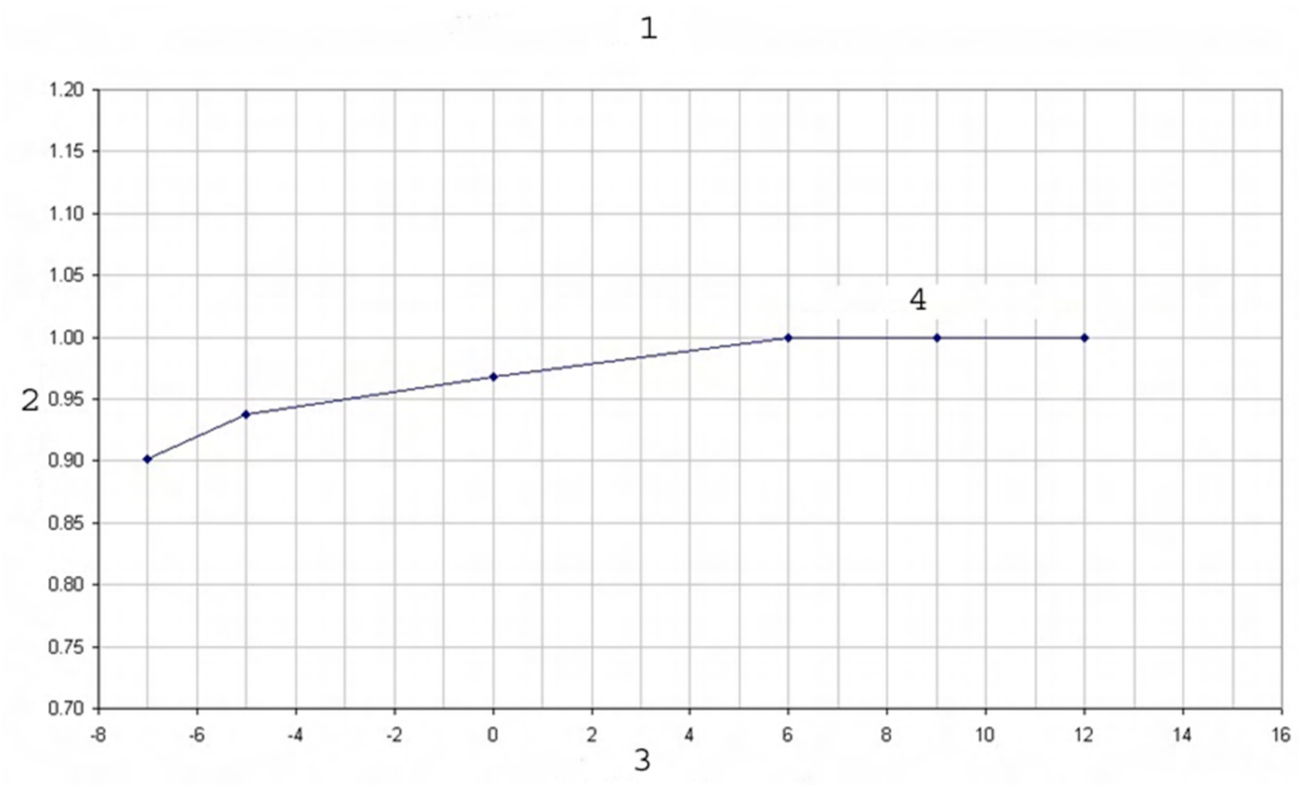


Relative heating capacity					
Key	**Inlet source temperature**	**Outlet sink temperature**			
1 air-to-water gas engine driven heat pump		**40 °C**	**45 °C**	**50 °C**	**55 °C**
2 relative heating capacity	**- 5 °C**	0,80	0,79	0,78	0,77
3 inlet source temperature	**0 °C**	0,87	0,87	0,86	0,85
4 heating 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 temperature $y = 0{,}00042\ x^2 + 0{,}01679\ x + 0{,}8618$					

Figure F.8 — Average overall heating capacity of air-to water gas engine driven heat pumps vs source and sink temperatures

F.4.2.2 Air-to-air heat pumps

The data reflect the lower end of the market.



Relative heating capacity		
Key	**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

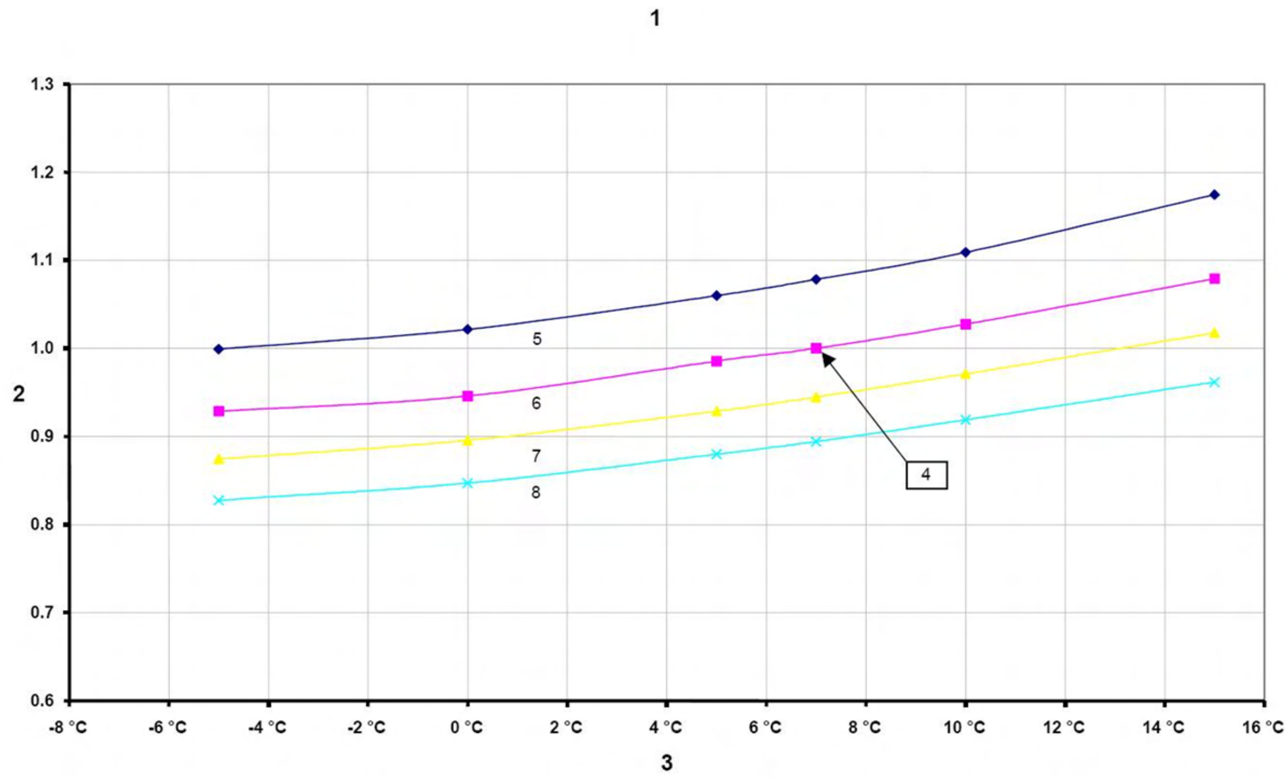
Figure F.9 — Heating capacity of air-to-air gas engine-driven heat pumps vs source and sink temperatures

F.4.3 *COP*

F.4.3.1 General

Following data are based on the average values of technical features provided by manufacturers.

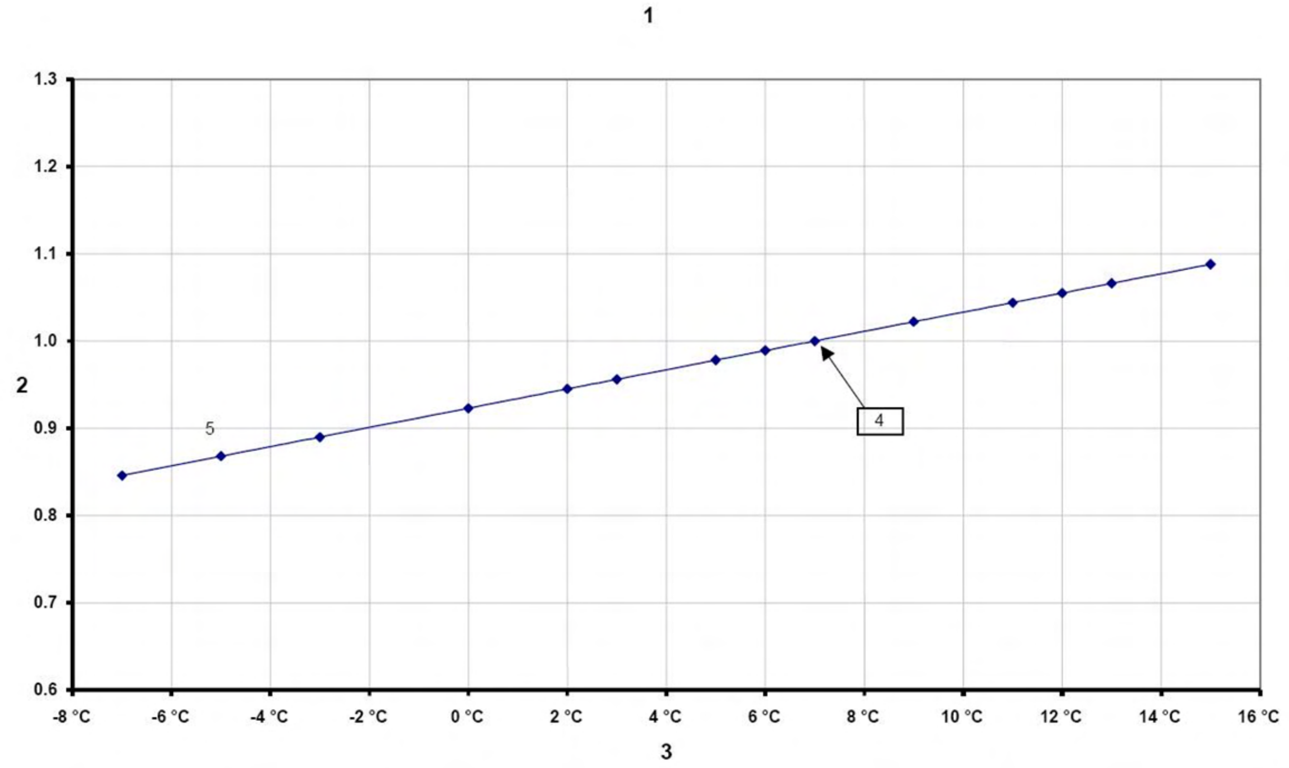
F.4.3.2 Air-to-water heat pumps



		Relative *COP*				
Key		**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 [°C]	**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 = 0,00027 x^2 + 0,00605 x 1,02207)	**7°C**	1,08	1,00	0,94	0,89
6	outlet sink temperature: 45 °C (y = 0,0002 x^2 + 0,00564 x 0,95026)	**10 °C**	1,11	1,03	0,97	0,92
7	outlet sink temperature: 50 °C (y = 0,00017 x^2 + 0,00555 x 0,89747)	**15 °C**	1,17	1,08	1,02	0,96
8	outlet sink temperature: 55 °C (y = 0,00015 x^2 + 0,00533 x 0,84945)					

Figure F.10 — Typical *COP* of air-to water gas engine driven heat pumps vs source and sink temperatures

F.4.3.3 Air-to-air heat pumps



Relative *COP*		
Key	**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 = 0,00018 x^2 + 0,01106 x + 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

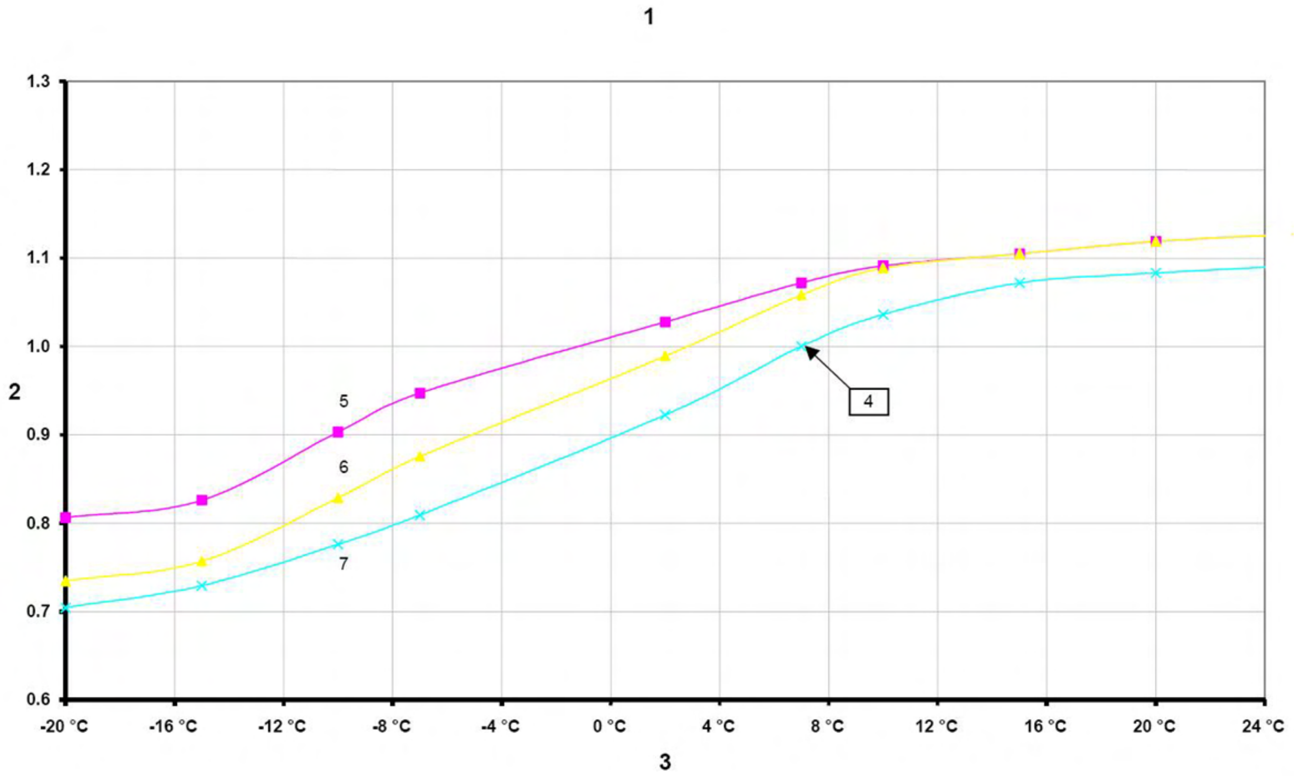
F.5 Absorption heat pumps

F.5.1 General

Characteristics provided here are based on very few measurements. 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.

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*. All values have been derived for heat pumps using gas as fuel.

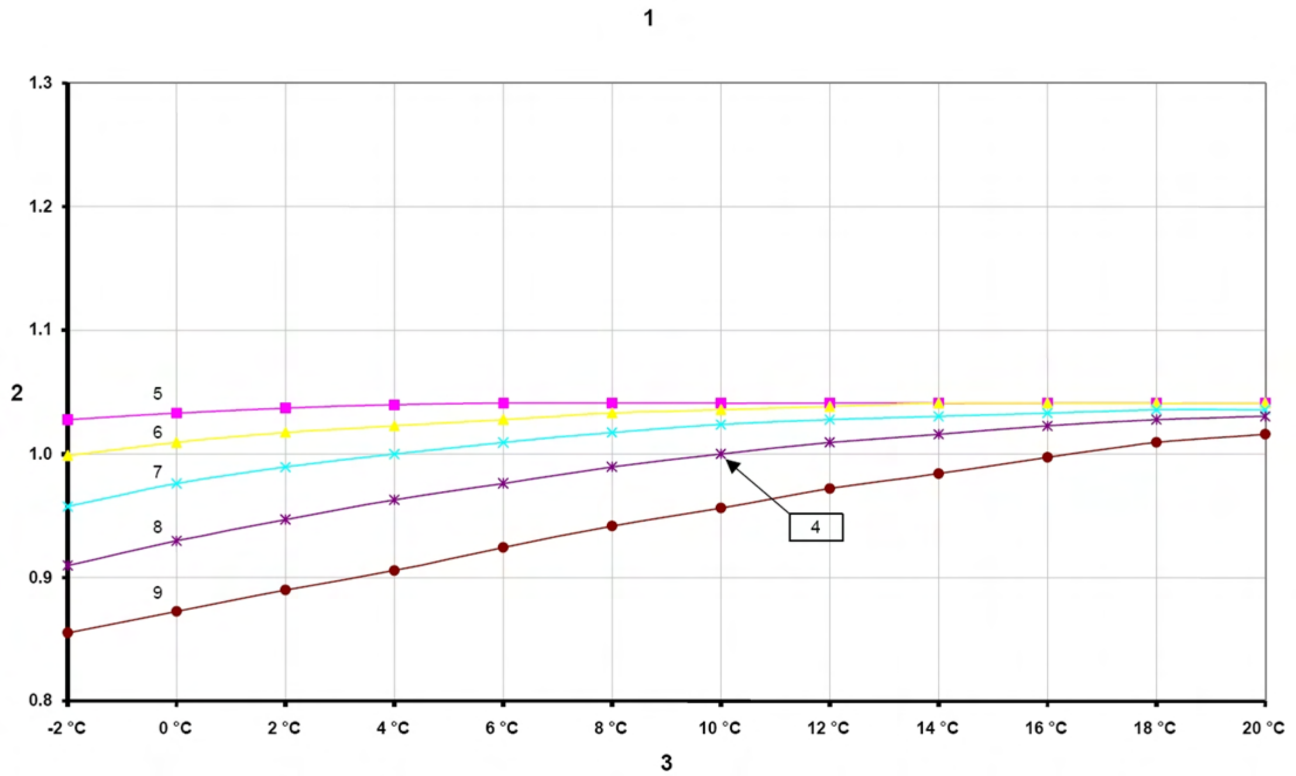
F.5.2 NH_3/H_20 heat pumps – outside air-to-water



Relative heating capacity and relative *COP*				
Key	**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**	**30 °C/40 °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

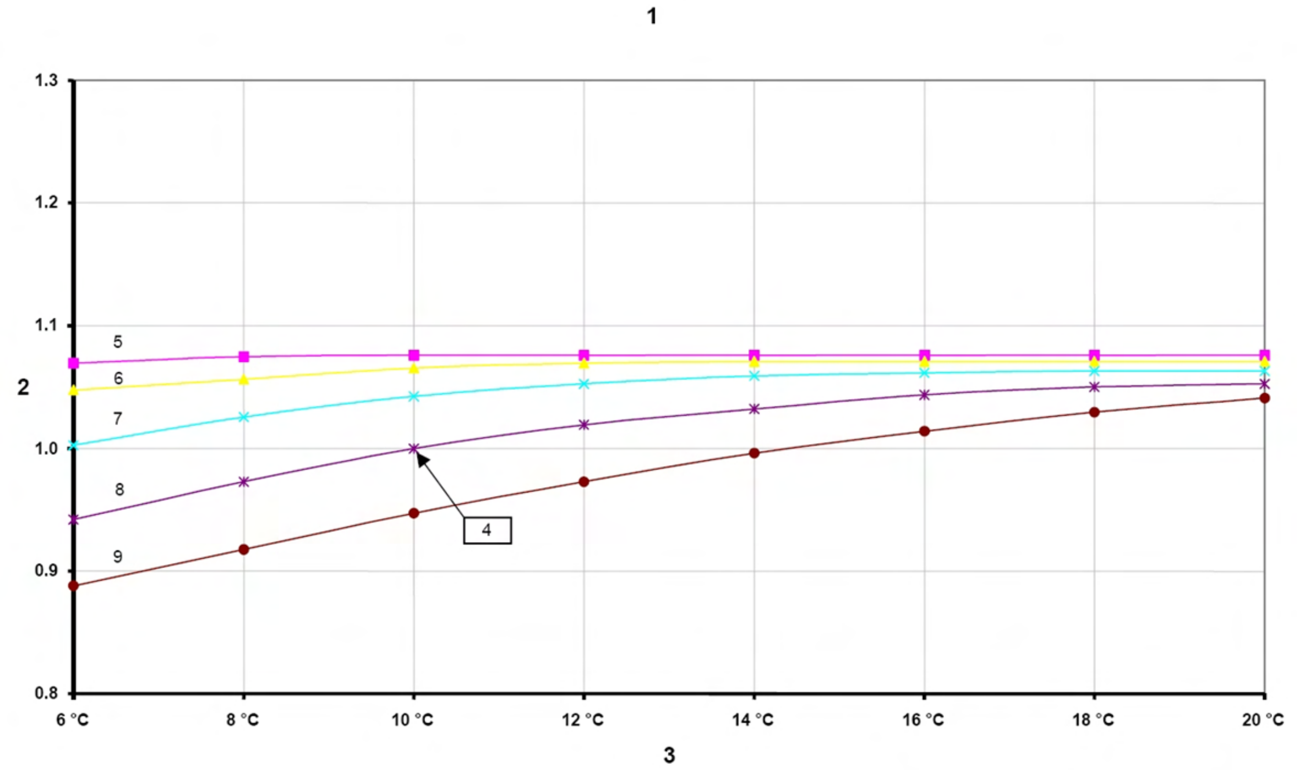
F.5.3 NH_3/H_20 heat pumps – brine-to-water



Relative heating capacity and relative *COP*							
Key		**Inlet source tempe-rature**	**Inlet and outlet sink temperatures**				
1	brine-to-water NH_3/H_2O direct gas fired absorption heat pump		**25 °C/ 35 °C**	**30 °C/ 40 °C**	**35 °C/ 45 °C**	**40 °C/ 50 °C**	**45 °C/ 55 °C**
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 ($y = - 0,00006\ x^2 + 0,00147\ x + 1,03309$)	**4 °C**	1,040	1,023	1,000	0,963	0,906
6	outlet sink temperature: 40 °C ($y = - 0,00012\ x^2 + 0,00399\ x + 1,00850$)	**6 °C**	1,041	1,028	1,009	0,976	0,924
7	outlet sink temperature: 45 °C ($y = - 0,00019\ x^2 + 0,00682\ x + 0,97475$)	**8 °C**	1,041	1,033	1,017	0,989	0,942
8	outlet sink temperature: 50 °C ($y = - 0,0002\ x^2 + 0,00908\ x + 0,92927$)	**10 °C**	1,041	1,036	1,023	1,000	0,956
9	outlet sink temperature: 55 °C ($y = - 0,00011\ x^2 + 0,00943\ x + 0,87246$)	**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

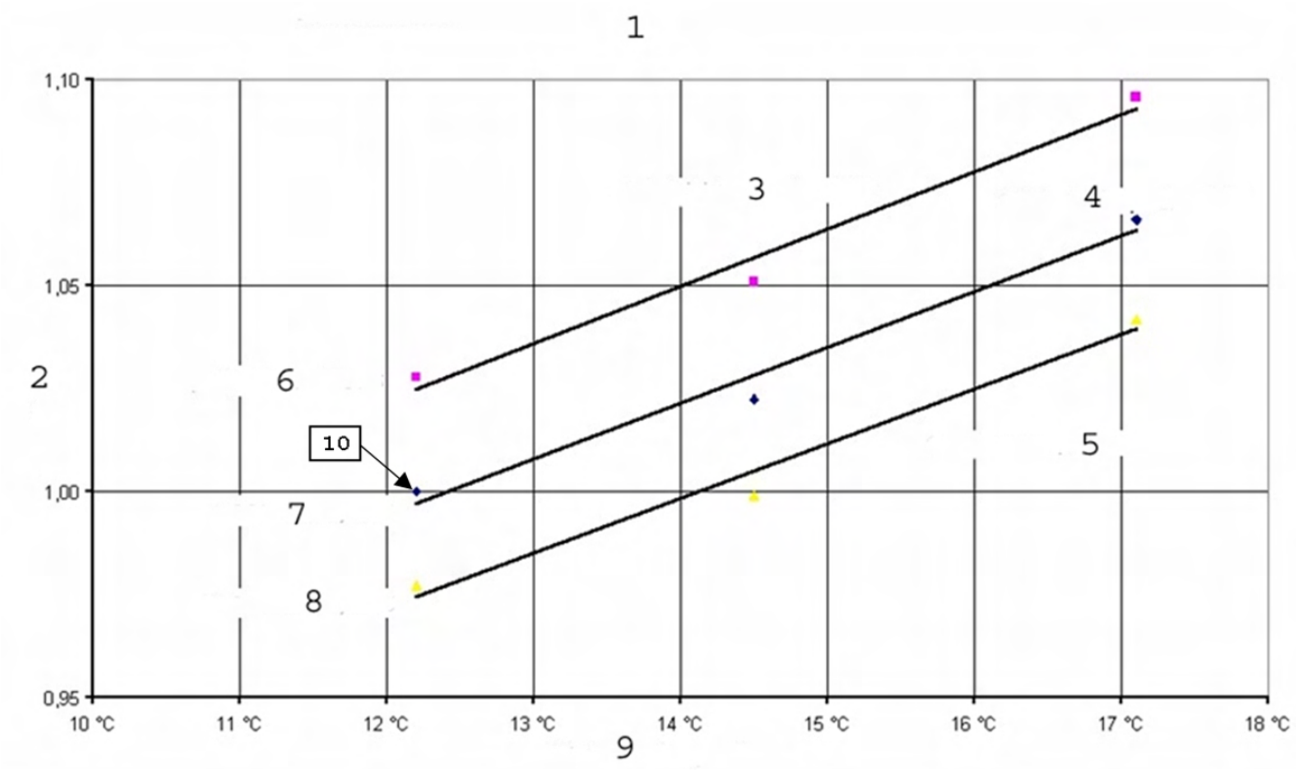
F.5.4 NH_3/H_20 heat pumps – water-to-water



Relative heating capacity and relative *COP*							
Key		**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**	**35 °C/ 45 °C**	**40 °C/ 50 °C**	**45 °C/ 55 °C**
2	relative heating capacity and relative *COP*	**6°C**	1,069	1,048	1,003	0.942	0,888
3	inlet source temperature [°C]	**8°C**	1,075	1,057	1,026	0,973	0,918
4	reference point heating capacity and *COP* (10 °C/50 °C)	**10°C**	1,076	1,066	1,042	1,000	0,947
5	outlet sink temperature: 35 °C ($y = - 0{,}00007\ x^2 + 0{,}00210\ x + 1{,}06078$)	**12°C**	1,076	1,069	1,053	1.019	0,973
6	outlet sink temperature: 40 °C ($y = - 0{,}00023\ x^2 + 0{,}00746\ x + 1{,}01198$)	**14°C**	1,076	1,071	1,059	1,032	0,996
7	outlet sink temperature: 45 °C $y = - 0{,}00048\ x^2 + 0{,}01656\ x + 0{,}92272$)	**16°C**	1,076	1,071	1,062	1,044	1,014
8	outlet sink temperature: 50 °C ($y = - 0{,}0006\ x^2 + 0{,}02330\ x + 0{,}82474$)	**18°C**	1,076	1,071	1,063	1,050	1,030
9	outlet sink temperature: 55 °C ($y = - 0{,}00041\ x^2 + 0{,}02180\ x + 0{,}87246$)	**20°C**	1,076	1,071	1,063	1,053	1,041

Figure F.14 — Average relative heating capacity and *COP* of water-to water NH_3/H_20 absorption heat pumps vs source and sink temperatures

F.5.5 H_20/LiBr double effect heat pumps



Relative heating capacity and relative *COP*				
Key	**Inlet source temperature**	**Outlet sink temperature**		
1 water-to-water H_2O/LiBr direct fired absorption heat pump		**30 °C**	**34 °C**	**37 °C**
2 relative heating capacity and relative COP	**12,2 °C**	1,028	1,000	0,977
3 outlet sink temperature: 30 °C	**14,5 °C**	1,051	1,022	0,999
4 outlet sink temperature 34 °C	**17,1 °C**	1,096	1,066	1,042
5 outlet sink temperature: 37 °C				
6 y = 0,01393 x + 0,85476				
7 y = 0,01355 x + 0,8316				
8 y = 0,01325 x + 0,8127				
9 inlet source temperature [°C]				
10 reference point heating capacity and *COP* (12,2 °C/34 °C)				

Figure F.15 — Average reletive heating capacity and *COP* of water-to water H_20/LiBr absorption heat pumps vs source and sink temperatures

F.6 Heat pumps with domestic hot water production (DHW) - 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 for the average loading temperature defined in Equation (12).

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