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Heating and cooling systems in buildings — Method for calculation of the system performance and system design for heat pump systems —

Part 2: **Energy calculation**

Systèmes de chauffage et de refroissement dans les bâtiments — Méthode de calcul de la performance du système et de la conception du système pour les systèmes de pompes à chaleur —

Partie 2: Calcul énergétique

Reference number
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ISO copyright office
Case postale 56 • CH-1211 Geneva 20
Tel. + 41 22 749 01 11
Fax + 41 22 749 09 47
E-mail copyright@iso.org
Web www.iso.org

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

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For an explanation on the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the WTO principles in the Technical Barriers to Trade (TBT) see the following URL: Foreword - Supplementary information

The committee responsible for this document is ISO/TC 205, *Building environment design*.

ISO 13612 consists of the following parts, under the general title *Heating and cooling systems in buildings — Method for calculation of the system performance and system design for heat pump systems*:

— *Part 1: Design and dimensioning*

— *Part 2: Energy calculation*

Introduction

This International Standard is a part of a series of standards on the methods for calculation of heating system energy requirements and heating and cooling system efficiencies.

— ISO 13612-1 deals with design and sizing of heat pump systems.

— ISO 13612-2 presents the energy calculation method.

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

This part of ISO 13612 presents methods for calculation of the additional energy requirements of a heat generation subsystem in order to meet the distribution subsystem demand. The calculation is based on the performance characteristics of the products given in product standards and on other characteristics required to evaluate the performance of the products as included in the system. Product data, e.g. heating capacity or COP of the heat pump, is determined according to products standards.

This method can be used for the following applications:

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

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

— assessing the effect of possible energy conservation measures on an existing heating/cooling generation subsystem, by calculating the energy use with and without the energy conservation measure.

Only the calculation method is normative. The user shall refer to other 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.

NOTE The results of this method can be used to assess the energy performance of the heating/cooling system when summing up the results over a period of calculation.

Heating and cooling systems in buildings — Method for calculation of the system performance and system design for heat pump systems —

Part 2: **Energy calculation**

1 Scope

This International Standard is applicable to heat pumps for space heating and cooling, heat pump water heaters (HPWH), and heat pumps with combined space heating and/or cooling and domestic hot water production, in alternate or simultaneous operation, where the same heat pump is used for space heating and domestic hot water heating.

This part of ISO 13612 provides a calculation method under steady conditions that corresponds to one calculation step.

The results of this calculation are incorporated in larger building models and take into account the influence of the external conditions and building control that influence the energy requirements for heating and cooling supplied by the heat pump system.

This part of ISO 13612 specifies the required inputs, calculation methods, and required outputs for output thermal power generation for space heating and cooling and domestic hot water production of the following heat pump systems, including control:

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

— combustion engine-driven vapour compression cycle heat pumps;

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

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

Table 1 — Heating/cooling sources and energy distribution

Source	Distribution
Outdoor air	Air
Exhaust-air	Water
Indirect ground source with brine distribution	Direct condensation/evaporation 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 documents, in whole or in part, are normatively referenced in this document and are indispensable for its application. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

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

ISO 13612-1, *Heating and cooling systems in buildings — Method for calculation of the system performance and system design for heat pump systems — Part 1: Design and dimensioning*

ISO 13675, *Heating systems in buildings — Method and design for calculation of the system energy performance — Combustion systems (boilers)*

ISO 13790, *Energy performance of buildings — Calculation of energy use for space heating and cooling*

ISO/TR 16344, *Energy performance of buildings — Common terms, definitions and symbols for the overall energy performance rating and certification*

3 Terms and definitions

For the purposes of this document, the terms and definitions in ISO 13612-1, EN ISO 7345:1995, and ISO/TR 16344 and the following apply.

3.1
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

3.2
application rating conditions
mandatory rated conditions within the operating range of the unit that are published by the manufacturer or supplier

3.3
auxiliary energy
electrical energy used by technical building systems for heating, cooling, ventilation, and/or domestic water to support energy transformation to satisfy energy needs

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

Note 2 to entry: In EN ISO 9488, the energy used for pumps and valves is called “parasitic energy”.

Note 3 to entry: In the frame of this part of ISO 13612, the driving energy input for electrically driven heat pumps in the system boundary of the COP and an electrical back-up heater is not considered auxiliary energy but only additional electrical input not considered in the COP.

3.4
balance point temperature
temperature at which the heat pump heating capacity and the building heat load are equal

3.5
bin
statistical temperature class (sometimes a class interval) for the outdoor air temperature

Note 1 to entry: The class limits are expressed in a temperature unit.

3.6
building service
service 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

3.7
calculation period
period of time over which the calculation is performed

Note 1 to entry: The calculation period can be divided into a number of calculation steps.

3.8
calculation step
discrete time interval for the calculation of the energy needs and uses for heating, cooling, humidification, and dehumidification

Note 1 to entry: Typical discrete time intervals are 1 h, 1 mon, or one heating and/or cooling season, operating modes, and bins.

Note 2 to entry: In the frame of the bin method, calculation steps are based on outdoor temperature classes.

3.9
coefficient of performance
COP
ratio of the heating/cooling capacity to the effective power input of the unit

3.10
cumulative frequency
frequency of the outdoor air temperature cumulated over all 1 K bins

3.11
cut-out period
time period in which the electricity supply to the heat pump is interrupted by the supplying utility

3.12
domestic hot water heating
process of heat supply to raise the temperature of the cold water to the intended delivery temperature

3.13
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, and

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

3.14
electrically driven heat pump
vapour compression cycle heat pump which incorporates a compressor driven by an electric motor

3.15
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

3.16
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 to entry: The energy need is calculated and cannot be easily measured.

Note 2 to entry: 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 are not included in the heat transfer due to the heating (cooling) system.

3.17
energy use for space heating or cooling or domestic hot water
energy input to the heating, cooling, or hot water system to satisfy the energy need for heating, cooling (including dehumidification), or hot water, respectively

Note 1 to entry: 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).

3.18
frequency
<statistical> number of times the event occurred in the sample

Note 1 to entry: The frequencies are often graphically represented in histograms. In the frame of this part of ISO 13612, the frequency of the outdoor air temperature is evaluated based on a sample of hourly averaged data for one year.

3.19
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

3.20
heat pump
unitary or split-type assemblies designed as a unit to transfer heat

Note 1 to entry: 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.

3.21
heat recovery
heat generated by a technical building system or linked to a building use (e.g. domestic hot water) which is utilized 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)

3.22
heat transfer medium
medium (water, air, etc.) used for the transfer of the heat without change of state

Note 1 to entry: The fluid cooled by the evaporator, the fluid heated by the condenser, and the fluid circulating in the heat recovery heat exchanger.

3.23
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

3.24
heating capacity
ϕ_g
heat given off by the unit to the heat transfer medium per unit of time

Note 1 to entry: If heat is removed from the indoor heat exchanger for defrosting, it is taken into account.

3.25
heating or cooling season
period of the year during which a significant amount of energy for heating or cooling is needed

Note 1 to entry: The season lengths are used to determine the operation period of technical systems.

3.26
internal temperature
arithmetic average of the air temperature and the mean radiant temperature at the centre of the occupied zone

Note 1 to entry: This is the approximate operative temperature according to ISO 7726.

3.27
low temperature cut-out
temperature at which heat pump operation is stopped and the total heat requirements are covered by a back-up heater

3.28
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

3.29
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

3.30
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

3.31
primary pump
pump mounted in the circuit containing the generator and hydraulic decoupling

EXAMPLE A heating buffer storage in parallel configuration or a hydronic distributor.

3.32
produced heat
heat produced by the generator subsystems

Note 1 to entry: In the context of this part of ISO 13612, this is 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.

3.33
recoverable system thermal loss
part of a 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

3.34
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

3.35
seasonal performance factor
SPF
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

3.35.1
cooling seasonal performance factor
CSPF
ratio of the total annual amount of heat that the equipment can remove from the indoor air when operated for cooling in active mode to the total annual amount of energy consumed by the equipment during the same period

3.35.2
heating seasonal performance factor
HSPF
ratio of the total annual amount of heat that the equipment, including make-up heat, can add to the indoor air when operated for heating in active mode to the total annual amount of energy consumed by the equipment during the same period

3.36
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

3.37
simultaneous operation during the heating period
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)

3.38
simultaneous operation during the cooling period
simultaneous production of output thermal power for the space cooling and domestic hot water system by a heat generator with double service (e.g. by refrigerant desuperheating or condensate subcooling)

3.39
space heating/cooling
process of heat supply for thermal comfort

3.40
standard rating condition
mandatory condition that is used for marking and for comparison or certification purposes

3.41
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 1 to entry: Thermal energy recovered directly in the subsystem is not considered as a system thermal loss but as heat recovery and directly treated in the related system standard.

3.42
technical building system
technical equipment for heating, cooling, ventilation, domestic hot water, lighting, and electricity production composed of subsystems

Note 1 to entry: A technical building system can refer to one or to several building services (e.g. heating system, heating and DHW system).

Note 2 to entry: Electricity production can include cogeneration and photovoltaic systems.

3.43
technical building subsystem
part of a technical building system that performs a specific function (e.g. heat generation, heat distribution, heat emission)

4 Symbols and abbreviated terms

For the purposes of this part of ISO 13612, the symbols and units in Table 2 and indices in Table 4 apply. Abbreviated terms are listed in Table 3.

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	Electrical (auxiliary) energy	J

Table 3 — Abbreviated terms

Abbreviation	Description
ATTD	Accumulated time-temperature difference
DHW	Domestic hot water
SH	Space heating

Table 3 *(continued)*

Abbreviation	Description
SC	Space cooling
TTD	Time-temperature difference
VCC	Vapour compression cycle
VAC	Vapour absorption cycle

Table 4 — Index

Δθ	temperature corrected	eng	engine	nrbl	non-recoverable
θllim	lower temperature limit	es	storage values acc. to EN 255-3, 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
C	space cooling	hp	heat pump	st	storage
cap	lack of capacity	Int	internal	sby	stand-by
co	cut-out	In	input to subsystem	sk	sink
cold	cold process side	j	index, referring to bin j	sngl	single (operation)
combi	combined operation	k	index	sc	source
crnt	Carnot	Ls	loss	standard	acc. to standard testing
dis	distribution subsystem	Ltc	low temperature cut-out	tot	total
des	at design conditions	max	maximum	w	water, heat transfer medium
Ext	external	n	nominal	W	domestic hot water (DHW),DHW operation
eff	effective				

NOTE The indices specifying the symbols in this part of ISO 13612 are put in the following order:

— the first index represents the type of energy use (H = space heating, W = domestic hot water). If the formula 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.

5 Principle of the method

5.1 Flowchart of the calculation method

Heat pump systems for heating and cooling can be independent or used as part of a system including other generators. Figure 1 explains how the information and output of the calculation are used in such multiple systems. In this case, the heat pump, including its integrated back-up system (if any), is considered as the priority generator.

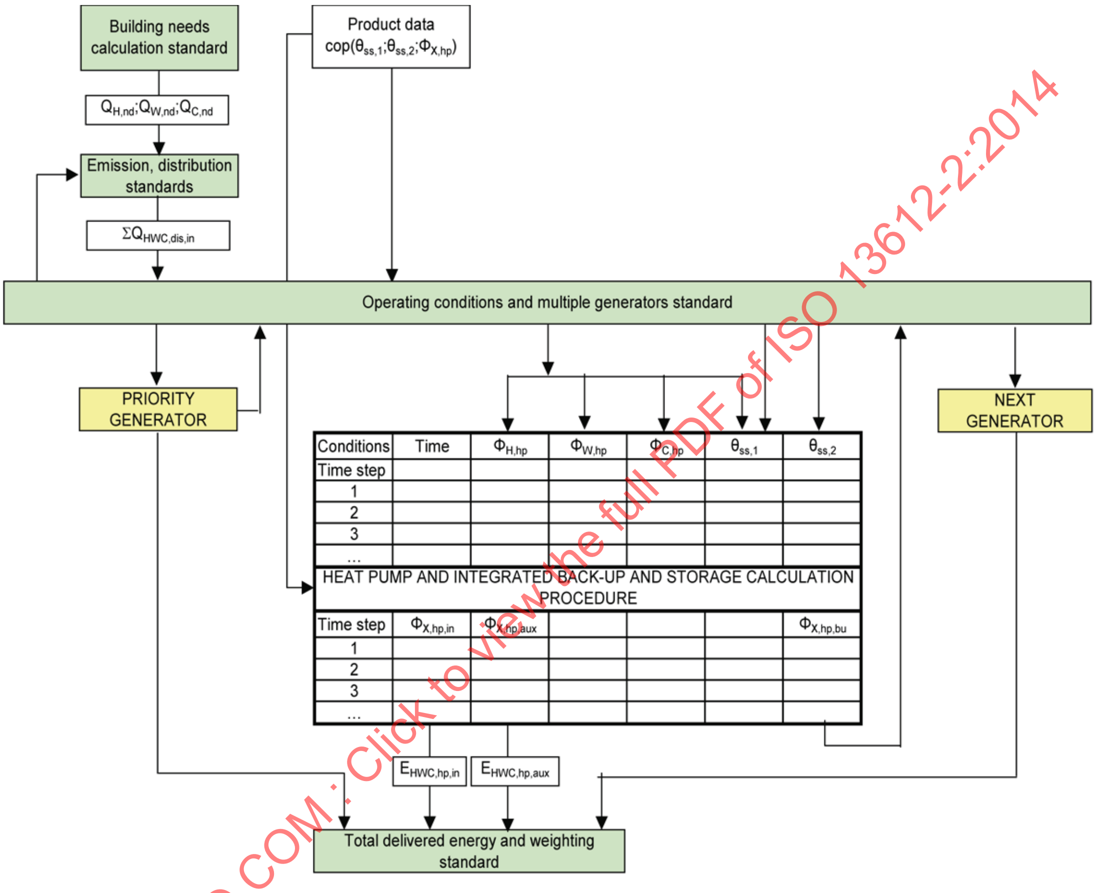


Figure 1 — Heat pump systems and interaction with other generators

The performance calculation method for the generation subsystem is described in the flowchart presented in Figure 2.

The method is based on calculating the amount of energies delivered to the heat pump system using tabulated values. Methods to establish the coefficient of performance (COP) according to the different heat pump system characteristics and available data are presented in Annexes A, B, C, D, and E.

The methodology is based on an hourly calculation as default time step for the calculation. The time step should be adapted according to the climatic data available and the accuracy required for the calculation.

An overview of the calculation steps to be performed is listed below. A more detailed overview for different system configurations can be seen in the flowchart in Figure 2.

The elementary calculation steps are explained in detail in the part of Clause 6 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 calculation, simplified and detailed methods are given in connection with the calculation of the running time.

Step 1: Collection of the input data (see 6.1)

Step 2: Identification of the time step for the calculation (see 6.2)

Step 3: Determination of energy requirements for heating and cooling periods (see 6.3)

Step 4: Construction of the energy delivered by the heat pump system depending on climatic conditions (see 6.4)

Step 5: Calculation of generation subsystem heat losses (see 6.4)

Step 6: Determination of back-up energy (see 6.5 for simplified; see 6.6.4 for detailed)

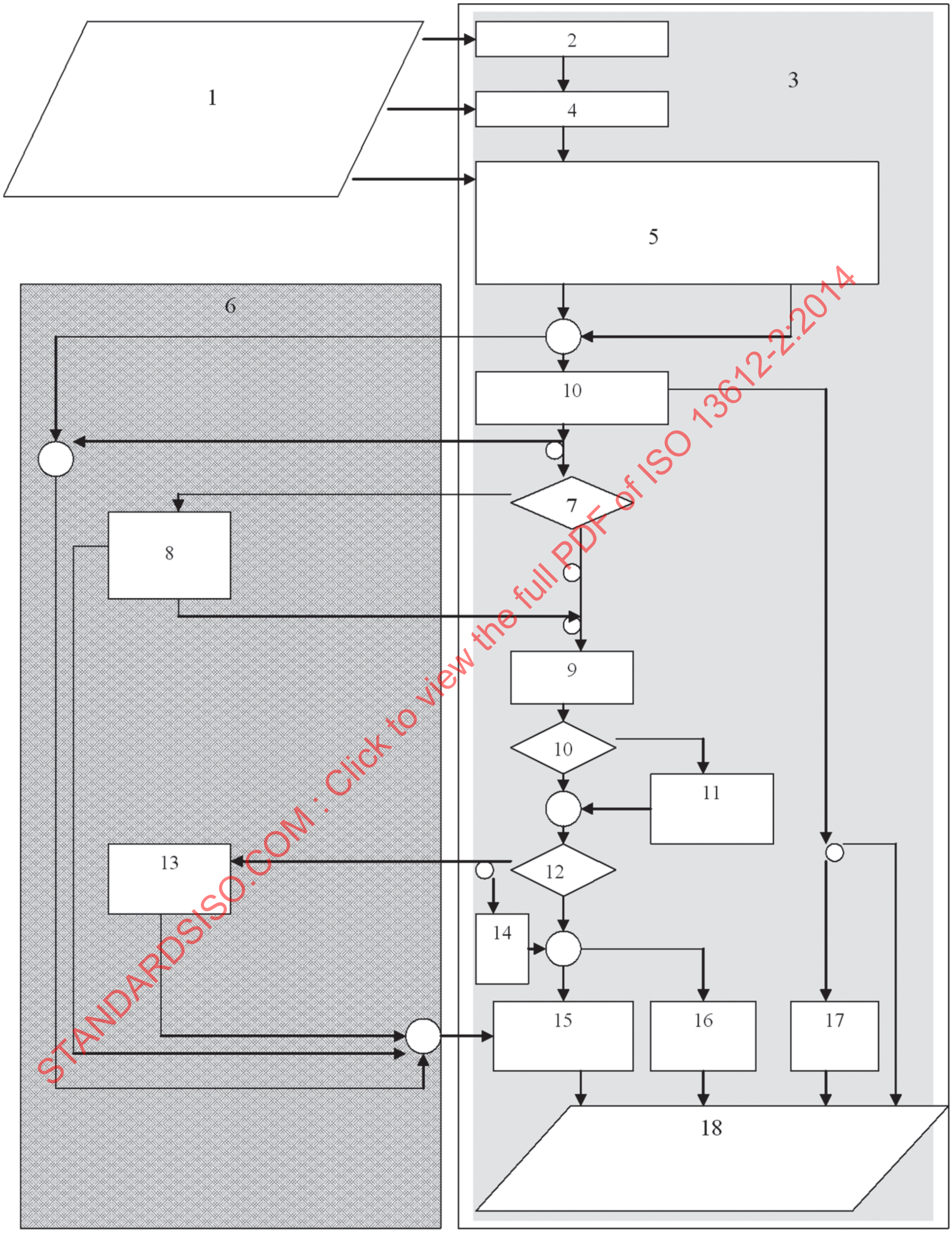
Step 7: Calculation of the running time of the heat pump in different operation modes (see 6.6)

Step 8: Calculation of auxiliary energy input (see 6.7)

Step 9: Calculation of recoverable generation subsystem losses (see 6.8)

Step 10: Calculation of the total driving energy input to cover the requirements (see 6.9)

Step 11: Summary of required and optional output values (see 6.10)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Key

1 input data (6.1):

— energy requirement of space heating distribution subsystem;

— energy requirement of the DHW distribution subsystem;

— meteorological data;

— product characteristic;

— design parameters.

2 definitions calculation period time step operation (6.2)

3 monovalent mode

4 calculation energy requirements for the step time (6.3)

5 tabulated value heating capacity/COP for operating conditions (6.4)

tabulated value cooling capacity/COP for operating conditions (6.4)

6 bivalent mode

7 calculation of generation subsystem heat losses

8 evaluation of back-up energy due to operation limit and based on balance point

9 calculation of running time

10 simultaneous system with three operation modes (Y/N)

11 calculation of operating time with simultaneous system with three operation modes

12 running time < time step for the calculation (Y/N)

13 calculation of back-up energy

14 running time = time step for the calculation

15 calculation of energy input to cover the heat requirement

16 calculation of auxiliary energy

17 calculation of recoverable losses

18 output data (6.10):

— energy input to cover the energy requirement;

— total losses of the generation subsystem;

— total recoverable losses of generation subsystem;

— total auxiliary energy input.

Figure 2 — Flowchart of the calculation method

The method is based on calculating the amount of energies delivered to the heat pump system using tabulated values. Methods to establish the tabulated values according to the different heat pump system characteristics are presented in Annexes A, B, C, D, and E.

The methodology is based on an hourly calculation as default time step for the calculation. The time step should be adapted according to the climatic data available and the accuracy required for the calculation.

5.2 System boundary

The system boundary defines the components of the entire heating systems that are considered in this part of ISO 13612. 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 included in the system boundary.

Distribution and emission systems are out of the system boundaries.

5.3 Physical factors

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 subsystem.

— 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)

— space cooling energy requirements of the distribution subsystem(s)

— effects of variation of source and sink temperature on heating and/or cooling capacity and COP according to standard product testing

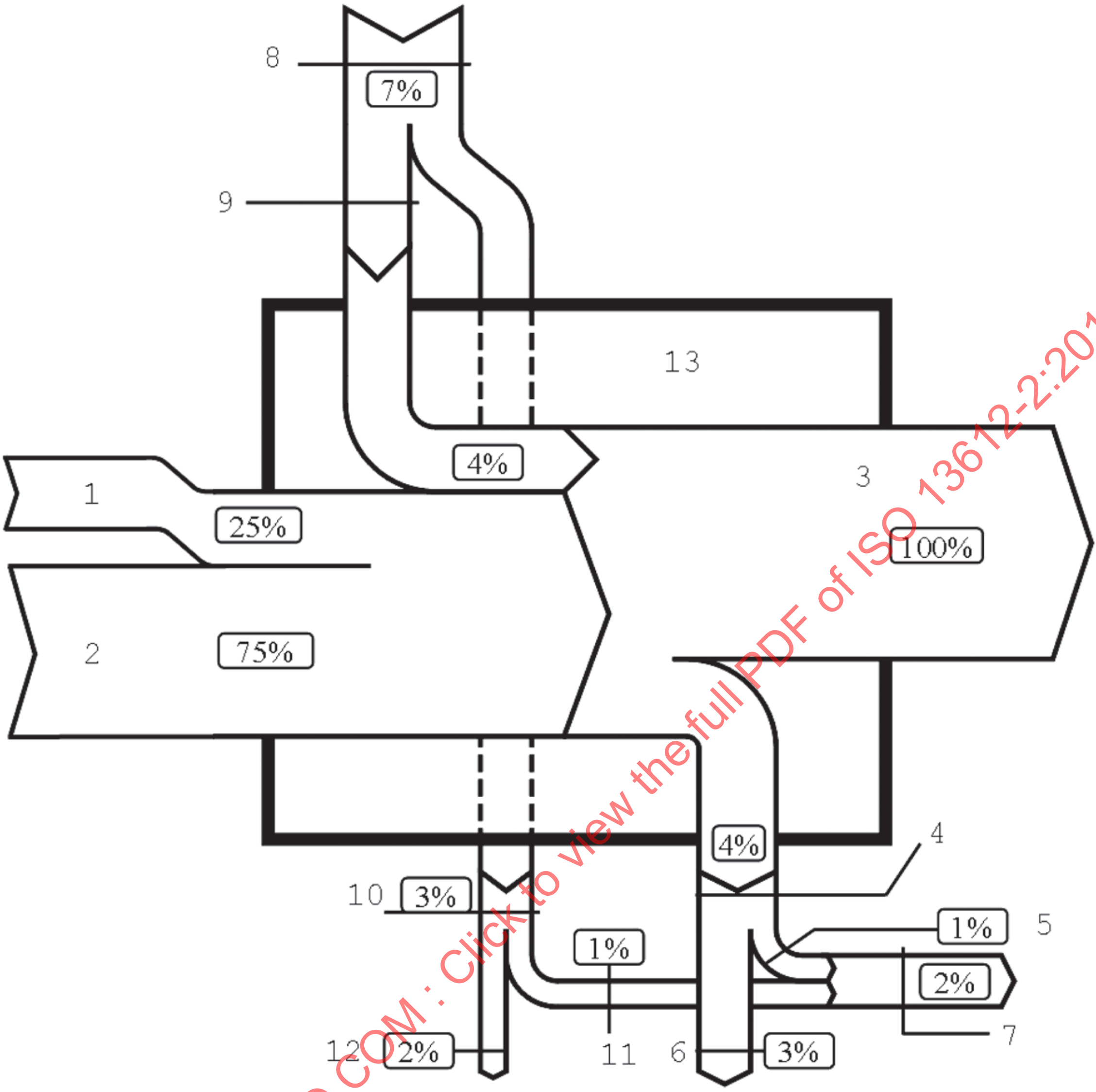
— effects of compressor control in part load operation (ON-OFF, stepwise, variable speed units) as far as they are reflected in the heating capacity and COP according to standard testing or further test results on part load operation exist

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

— system heat losses due to space heating or DHW storage components or space cooling, including the connecting pipework or ducts

— location of the generation subsystem

5.4 Schematization of the heat pump for heating and cooling



Key

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

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

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

4 generation subsystem heat losses, $Q_{HW,gen,ls,tot}$

5 recoverable generation subsystem heat losses, $Q_{H,gen,ls,rbl}$

6 non-recoverable generation subsystem heat losses, $Q_{HW,gen,ls,nrbl}$

7 total recoverable generation subsystem heat losses, $Q_{H,gen,ls,rbl,tot}$

8 auxiliary energy input, $W_{HW,gen,aux}$

9 recovered heat loss of auxiliary components, $Q_{HW,gen,aux,ls,rcv}$

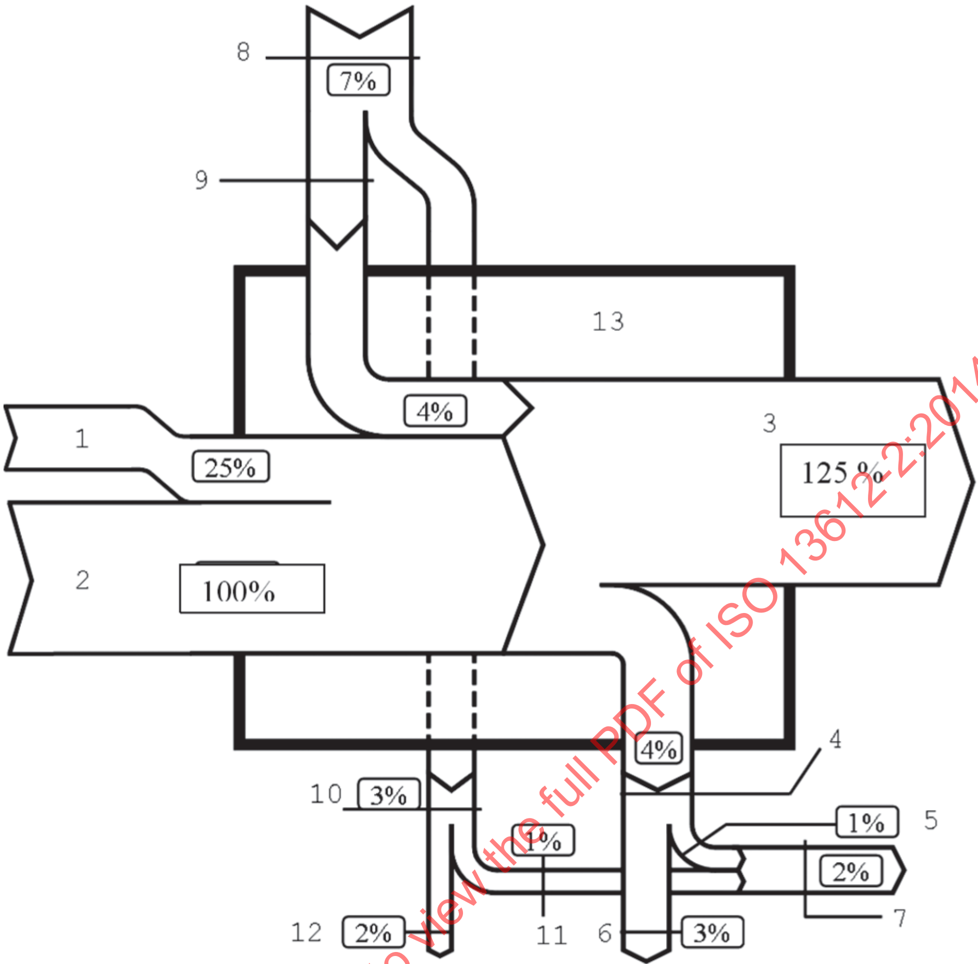
10 unrecovered heat loss of auxiliary components, $Q_{HW,gen,aux,ls}$

11 recoverable heat loss of auxiliary components, $Q_{H,gen,aux,ls,rbl}$

12 non-recoverable heat loss of auxiliary components, $Q_{HW,gen,aux,ls,nrbl}$

13 generation subsystem

Figure 3 — Example of energy balance of generation subsystem for heating



Key

1 driving energy input to cover the cooling requirement (e.g. electricity, fuel), $E_{CW,gen,in}$

2 extracted energy for cooling, $Q_{CW,gen,in}$

3 heat output of generation subsystem ($Q_{HW,gen,out} = Q_{HW,dis,in}$)

4 generation subsystem heat losses, $Q_{HW,gen,ls,tot}$

5 recoverable generation subsystem heat losses, $Q_{CW,gen,ls,rbl}$

6 non-recoverable generation subsystem heat losses, $Q_{HW,gen,ls,nrbl}$

7 total recoverable generation subsystem heat losses, $Q_{H,gen,ls,rbl,tot}$

8 auxiliary energy input, $W_{HW,gen,aux}$

9 recovered heat loss of auxiliary components, $Q_{HW,gen,aux,ls,rcv}$

10 unrecovered heat loss of auxiliary components, $Q_{HW,gen,aux,ls}$

11 recoverable heat loss of auxiliary components, $Q_{H,gen,aux,ls,rbl}$

12 non-recoverable heat loss of auxiliary components, $Q_{HW,gen,aux,ls,nrbl}$

13 generation subsystem

Figure 4 — Example of energy balance of generation subsystem for cooling

The numbers indicated in Figures 3 and 4 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 depending on the physical factors listed before. The numbers given in Figure 3 refer to an electrically driven ground-source heat pump in monovalent space heating-only operation, including buffer storage.

5.5 Input and output of the calculation method

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;

— cooling requirements for air-conditioned space.

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

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

— required energy input as driving energy, $E_{CW,gen,in}$, e.g. electricity, fuel, waste heat, solar heat, to meet the space cooling and/or domestic hot water requirements;

— total generation subsystem heat loss, $Q_{HW,gen,ls,tot}$;

— total recoverable generation subsystem heat losses, $Q_{H,gen,ls,rbl,tot}$;

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

5.6 Energy input needed to meet the heat requirements for electrically driven heat pumps

The energy balance for the electrically driven generation subsystem is given by Formula (1).

$$E_{HW,gen,in} = Q_{HW,gen,out} + Q_{HW,gen,ls,tot} - Q_{HW,gen,in} - k_{gen,aux,ls,rvd} \cdot W_{HW,gen,aux} \quad (1)$$

where

$E_{HW,gen,in}$ is the driving electrical energy, fuel or heat input to cover the heat requirement of the distribution subsystem (J);

$Q_{HW,gen,out}$ is the heat energy requirement of the distribution subsystems (J);

$Q_{HW,gen,ls,tot}$ is the total heat losses of the generation subsystem (J);

$Q_{HW,gen,in}$ is the ambient heat energy used as heat source of the heat pump (J);

$k_{gen,aux,ls,rvd}$ is the recovered fraction of heat energy from the auxiliaries (-);

$W_{HW,gen,aux}$ is the auxiliary energy input to operate the generation subsystem (J).

In case of electrically driven heat pumps, the term $E_{HW,gen,in}$ is the electrical energy input necessary for the heat pump system to cover the energy requirement of the distribution subsystem. It comprises the electrical energy input to the heat pump system and possibly installed back-up heaters.

NOTE For some standards, such as EN 14511, $E_{HW,gen,in}$ 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, eventually, energy for defrost operation and additional heating devices for the oil supply of the compressor (carter heating).

— Thus, $W_{HW,gen,aux}$ only comprises the fractions not included in the COP standard testing. $k_{gen,aux,ls,rvd}$ 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 $k_{gen,aux,ls,rvd} = 0$.

— For total heat losses, $Q_{HW,hp,ls,tot}$, the heat 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 DHW hot water storage, generation subsystem losses in the form of storage heat losses and losses of the connecting circulation pipes to the storage are considered.

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

— $E_{HW,gen,in}$ 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, not only fuel-driven burners but also solar energy or waste heat can be the driving energy input.

— $Q_{HW,gen,out}$, the heat energy output of the generation subsystems, is equal to the heat requirement of the distribution subsystem and contains all fractions of heat recovered from the engine or the flue gas of the engine, i.e. recovered heat from the engine 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 heat is already included in the COP.

5.7 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.

NOTE As for electrically driven heat pumps, heating capacity and COP in this part of ISO 13612 are calculated on the basis of results from product testing; according to EN 14511, only the auxiliary energy not included in the test results, e.g. the power to overcome the external pressure drop and the power in stand-by operation, are considered in $W_{HW,gen,aux}$.

Auxiliary energy is accounted to the generation subsystem as long as no transport energy is transferred to the distribution subsystem. That means, in general, the circulation pump is accounted to the distribution subsystem, unless hydraulic decoupling exists. For 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 accounted to the generation subsystem as well.

In this case, the power to overcome the external pressure drop has to be taken into account. If no primary pump is considered, since there is no hydraulic decoupling between the generation and distribution subsystem, the COP values have to be corrected for the internal pressure drop, which is included in the COP values by the standard testing.

5.8 Recoverable, recovered, and unrecoverable heat losses

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

Recoverable heat losses, $Q_{H,gen,ls,rbl}$, are, for example, heat losses through the envelope of a generation subsystem in the 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 $Q_{HW,dis,in}$.

5.9 Calculation periods

Heat pump performance strongly depends on the operating conditions, which are basically the source and the sink temperature. As source and sink temperatures vary over the heating and cooling periods and over the year, the heat pump performance shall be calculated accordingly with an adapted step period, in line with the climatic data used. As default, a step time of 1 h is considered in this part of ISO 13612.

The time step shall be adapted according to the climatic data available and the accuracy required for the calculation.

NOTE For some alternative methods, calculation periods are not oriented at the time scale, i.e. monthly values, but on the frequency of the outdoor air temperature (bin method).

5.10 Calculation by zones

A heating/cooling system may be split up in 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:

$$\sum_{j} Q_{\mathrm{H,gen,out,j}} = \sum_{k} Q_{\mathrm{H,dis,in,k}} \tag{2}$$

where

$Q_{\mathrm{H,gen,out,k}}$ 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) $Q_{\mathrm{H,dis,in,k}}$ shall be distributed among the available generators and the calculation described in Clause 6 shall be performed independently for each generation subsystem k on the basis of $Q_{\mathrm{H,gen,out,k}}$. This is accomplished in case of an installed back-up heater.

For intermittent heating, the requirements of ISO 13790 shall be considered.

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 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, and

— cascade cycles with internal heat exchangers.

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

For simultaneous system layout, three operation modes shall be distinguished:

— space heating-only (cooling-only) operation:

only the space heating system is in operation only the lower stage heat pump is in operation (winter time, DHW storage entirely loaded);

— domestic hot water-only operation:

only the domestic hot water system is in operation (summer operation, no space heating demand);

— simultaneous operation:

both space heating (or cooling) and domestic hot water operation. For the configuration shown in Figure 4, 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).

For combined cooling and domestic hot water, the calculation shall identify which priority use is considered. In most cases, priority is given to cooling use as DHW is a co-product. Back-up heater is provided the necessary complement to satisfy DHW production.

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

6 Generation subsystem calculation

6.1 Input data

Input information for the procedure can consist of the following:

— the type of building and sector;

— climatic data adapted to the local consideration;

— the operating condition (including typical occupancy patterns of the relevant building sector taken into account, comfort level, and room temperature/humidity);

— the heat pump function (space heating, domestic hot water production, space cooling, any combination of these);

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

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

— the type of heat source;

— the source pump or fan power;

— the test results produced in accordance with standard tests (e.g. EN 14511) for electrically driven heat pumps;

— the heating capacity, cooling capacity;

— whether performance data include the effect of an integrated storage, if any;

— integrated domestic hot water storage characteristics (volume/dimensions, specific loss);

— information about control of the heat pump system and priority given to the energy use (e.g. priority to domestic hot water during heating or cooling mode);

— the design and operation of the generation heat pump system, the calculation time–step, and the calculation period;

— characteristics of the integrated back up heater.

The calculation period shall be subdivided into three subperiods as the heat pump system is operating for

— space heating (and domestic hot water),

— space heating and space cooling alternatively,

— space cooling (and domestic hot water).

The climatic data shall be considered as constant for any calculation time step.

6.2 Energy requirements for space heating, space cooling, and DHW mode

6.2.1 Space heating and space cooling mode

The energy requirement of the space heating and space cooling distribution subsystem, $Q_{H,dis,in}$, is specified in 6.1.

If data provided are not in line with the time step calculation, then data are derived from data provided by

— using the sum of the heat energy requirement if the time step is greater than the time period between data, or

— using linear interpolation if the time step t_i is lower than the time period between bin scheme and standard locations in a national annex.

6.2.2 Domestic hot water mode

The heat energy requirement of the domestic hot water distribution subsystem $Q_{W,dis,in}$ is calculated according to 6.1.

Energy requirements are adapted to the time step accordingly with 6.3.

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

6.3 Tabulated values of the COP for heating and cooling at full load

The values of the thermal power provided to the output system are provided according to methods presented in Annexes A, B, and C.

6.4 Heat losses through the generator envelope

6.4.1 Space heating mode

For heat pumps without an integrated storage in the same housing, the losses to the ambiance are neglected in the frame of this part of ISO 13612, unless national values are given for the envelope heat loss of the heat pump.

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

Internal or external heating buffer storage produces losses to the ambient that can be calculated by a stand-by heat loss value for the step time t_j:

$$Q_{H,st,ls,j} = \frac{\theta_{H,st,avg,j} - \theta_{H,st,amb,j}}{\Delta\theta_{st,sby}} \cdot \frac{Q_{st,sby} \cdot 1000[W/kW] \cdot t_j}{24[h/d]} \tag{3}$$

where

$Q_{H,st,ls,j}$ is the heat loss of the heating buffer storage to the ambient in time t_j (J);

$\theta_{H,st,avg,j}$ is the average storage temperature of the heating buffer storage in time t_j (°C);

$\theta_{H,st,amb,j}$ is the ambient temperature at the storage location (°C);

$\Delta\theta_{st,sby}$ is the temperature difference due to storage stand-by test conditions (K);

$Q_{st,sby}$ is the stand-by heat loss due to storage stand-by test conditions (kWh/d);

t_j is the step time t_j (constant) (s).

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

The average storage temperature, $\theta_{H,st,avg,j}$, is to be determined according to the storage control. If the storage is operated depending on the temperature requirements of the heating system, it is approximated as the average temperature of the flow and return temperature of the space heating system, according to Formula (4).

$$\theta_{H,st,avg,j} = \frac{\theta_{H,gen,f,j} - \theta_{H,dis,r,j}}{2} \tag{4}$$

where

$\theta_{H,st,avg,j}$ is the average storage temperature of the heating buffer storage at time t_j (°C);

$\theta_{H,gen,f,j}$ is the flow temperature of the space heating generation system at time t_j (°C);

$\theta_{H,dis,r,j}$ is the return temperature from the space heating distribution system at time t_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.

The same principle applies for the cooling mode.

6.4.2 Domestic hot water mode

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

If no values on storage stand-by losses are available, the calculation shall be carried out according to Formula (3), with values based on the volume of the storage vessel given in a national annex. If no national values are available, default values are given in Annex B.

6.4.3 Cooling mode

For the cooling mode, the same principles as the one used for the heating mode apply.

— For heat pumps without an integrated storage in the same housing, the losses to the ambiance are neglected in the frame of this part of ISO 13612, unless national values are given for the envelope thermal losses 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 losses can be estimated by the efficiency of the engine and a possible fraction of recovered heat. For the redistribution of the total thermal losses to the bins or operation modes, if required, the stand-by losses for each step time and the running time (for operational losses) of the heat pump shall be evaluated.

Internal or external buffer or ice storage produces losses to the ambient that can be calculated by a stand-by heat loss value for the step time t_j in accordance with Formula (3).

6.4.4 Heat losses of primary circulation pipes

Heat losses of the primary circulation pipes between the heat generator and the storage vessel shall be added to the storage losses.

6.5 Calculation of back-up heater

6.5.1 General

Back-up energy can be required for two reasons.

— 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 6.5.2.

— Multivalent design of the generator subsystem (see boundary conditions in 5.10), i.e. the heat pump is not designed for the total load. Then, a fraction of back-up energy is required due to a lack of heating capacity of the heat pump.

For the calculation of the back-up operation due to a lack of capacity, a simplified and a detailed method are given.

The simplified method is based on the evaluation of the cumulative frequency and the balance point and, depending on the operation mode, the low temperature cut-out. It is described in 6.5.2. The method assumes that the balance point is known and all influencing factors (e.g. power demand for space heating and DHW operation, cut-out times of the electricity supply, etc.) have been taken into account.

For the detailed method, a 1 K energy balance is accomplished for the range of lower source temperatures up to the temperature where no back-up energy is needed. It should be applied, if the balance point is not known or difficult to calculate, e.g. in systems with simultaneous operation, or if 1 K bins are chosen for the calculation anyway. The balance point is no longer an input since it follows from the energy balance expressed by the required running time. The method is described in 6.6.4 in connection with the evaluation of the running time.

6.5.2 Back-up energy due to the operation limit temperature of the heat pump in heating mode

Depending on the refrigerant and the heat pump internal cycle, the maximum temperature level that can be produced with the heat pump is restricted by an operation limit. If temperatures above a certain temperature are required, they cannot be produced by the heat pump but have to be reheated by a back-up heater. Therefore, the fraction of back-up energy due to the operation limit of the heat pump can be calculated using Formula (5).

$$k_{\text{bu,opr,j}} = \frac{Q_{\text{bu,opr,j}}}{Q_{\text{gen,out,j}}} = \frac{m'_{\text{w}} \cdot c_{\text{w}} \cdot \left(\theta_{\text{n,j}} - \theta_{\text{hp,op}}\right) \cdot t_{\text{hp,on,j}}}{Q_{\text{gen,out,j}}} \tag{5}$$

where

$k_{\text{bu,opr,j}}$ is the fraction of back-up energy due to the operation limit of the heat pump in time t_{j} (-);

$Q_{\text{bu,opr,,j}}$ is the back-up heat energy due to the operation limit of the heat pump in time t_{j} (J);

$Q_{\text{gen,out,j}}$ is the heat energy requirement of the distribution subsystem in time t_{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 at time t_{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 at time t_{j} (s).

For the 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 DHW operation, higher temperatures than the operation limit may be required so that the heat pump delivers the heat up to the operation limit temperature, e.g. 55 °C, and the additional temperature requirement, e.g. up to 60 °C, is supplied by the back-up heater. The fraction of back-up heat energy supplied to the domestic hot water system is given by Formula (6).

$$k_{\mathrm{W,bu,opr,j}} = \frac{Q_{\mathrm{W,bu,opr,j}}}{Q_{\mathrm{W,gen,out,j}}} = \frac{\rho_{\mathrm{w}} \cdot V_{\mathrm{w,j}} \cdot c_{\mathrm{w}} \cdot \left(\theta_{\mathrm{w,out}} - \theta_{\mathrm{hp,opr}}\right)}{\rho_{\mathrm{w}} \cdot V_{\mathrm{w,j}} \cdot c_{\mathrm{w}} \cdot \left(\theta_{\mathrm{w,out}} - \theta_{\mathrm{w,in}}\right)} = \frac{\left(\theta_{\mathrm{w,out}} - \theta_{\mathrm{hp,opr}}\right)}{\left(\theta_{\mathrm{w,out}} - \theta_{\mathrm{w,in}}\right)} \tag{6}$$

where

$k_{\mathrm{W,bu,opr,j}}$ is the fraction of back-up energy in DHW mode due to the operation limit of the heat pump at time t_{j} (-);

$Q_{\mathrm{W,bu,opr,j}}$ is the DHW back-up heat energy due to the operation limit of the heat pump at time t_{j} (J);

$Q_{\mathrm{W,gen,out,j}}$ is the heat energy requirement of the domestic hot water subsystem at time t_{j} (J);

ρ_{W} is the density of the heat transfer medium (kg/m^3);

$V_{\mathrm{w,j}}$ is the volume of the hot water draw-off at time tj (m^3/s);

c_{W} is the specific heat capacity of water [J/(kg·K)];

$\theta_{\mathrm{W,out}}$ is the temperature of the hot water at storage outlet (°C);

$\theta_{\mathrm{hp,opr}}$ is the operation limit temperature of the heat pump (maximum temperature that can be reached with the heat pump operation) (°C);

$\theta_{\mathrm{W,in}}$ 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.

6.6 Running time of the heat pump

6.6.1 General

The running time of the heat pump depends on the heating/cooling capacity, given by the operating conditions, and on the heat requirement, given by the distribution subsystem. The running time can be calculated by Formula (7):

$$t_{\mathrm{hp,on,j}} = \frac{Q_{\mathrm{hp,j}}}{\phi_{\mathrm{hp,j}}} \tag{7}$$

where

$t_{\mathrm{hp,on,j}}$ is the running time of the heat pump at time t_{j} (s);

$Q_{\mathrm{hp,j}}$ is the produced heat energy by the heat pump at time t_{j} (heat energy requirement of the distribution subsystem and generation subsystem losses) (J);

$\phi_{\mathrm{hp,j}}$ is the heating (or cooling) capacity of the heat pump at time t_{j} (W).

The running time at step j, $t_{\mathrm{hp,on,j}}$, shall be considered for verification that this running time is lower than the time step calculation.

— If yes, calculate the part load factor, then the corrected COP.

— If no, link to the back-up heater (additional energy to be supplied) and HP at full load.

The produced heat by the heat pump can be calculated by Formula (8).

$$Q_{\mathrm{hp,j}} = \left(Q_{\mathrm{gen,out,j}} + Q_{\mathrm{gen,ls,j}}\right)\left(1 - k_{\mathrm{bu,cap,j}}\right) \tag{8}$$

where

$Q_{\mathrm{hp,j}}$ is the produced heat energy by the heat pump at time t_{j} (energy requirement of the distribution subsystem and generation subsystem losses) (J);

$Q_{\mathrm{gen,out,j}}$ is the heat energy requirement of the distribution subsystem at time t_{j} (J);

$Q_{\mathrm{gen,ls,j}}$ is the generation subsystem heat losses at time t_{j} (J);

$k_{\mathrm{bu,cap,j}}$ is the fraction of heat energy covered by the back-up heater at time t_{j} (-).

These formulae can be applied for the different operation modes. The following items shall be considered.

Back-up calculation

The fraction of back-up energy for DHW due to the operation limit of the heat pump has to be taken into account, since the fraction of back-up energy due to a lack of capacity follows from the energy balance (see 6.6.4). That means only the fractions $k_{\mathrm{bu,opr,j}}$ due to the operation limit temperature are considered.

Operation mode (heating and/or DHW)

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, with priority given to provide energy to the DHW system (default mode).

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 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, shall 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 6.6.2.

The total running time at time t_j can be calculated by Formula (9):

$$t_{hp,on,tot,j} = t_{H,hp,on,sngl,j} + t_{W,hp,on,sngl,j} + t_{HW,hp,on,combi,j} \quad (9)$$

where

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

Operation mode (cooling and/or DHW)

For heat pumps operating in SC-only mode, the energy requirement is given by the actual space cooling, i.e. the energy requirement of the distribution subsystem and the generator losses.

Running time calculation identical to the space heating only mode:

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

For heat pumps operating simultaneously for heat production for SC 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 operation modes may have to be evaluated.

NOTE Simultaneous operation of heat pump systems for space cooling and DHW is used in specific cases when the energy used for DHW is low versus the energy use for space cooling or when the heat pump system is used for pre-heating of the DHW.

First step: Identify operating conditions and energy requirements for both energy uses.

Calculate separately the running time for space cooling and domestic hot water.

— Identify the minimum value and check which energy output is reached for both energy requirements.

— Then calculate the energy left for the single mode operation.

— Check if the corresponding running time matches the time step calculation.

Second step: Calculate the additional running time necessary to complete the energy requirements.

If the calculated running time is higher than the available time step calculation, then identify the load for other generators (heating/cooling).

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

6.6.2.1 Principle

When all three operation modes occur, the running time in the different operation modes has to be determined.

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

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

NOTE 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.

6.6.2.2 Calculation steps

The maximum running time in simultaneous operation is calculated by Formula (10):

$$t_{\mathrm{HW,hp,on,combi,max,j}} = \min\left(t_{\mathrm{H,hp,on,j}}, t_{\mathrm{W,hp,on,j}}\right) \tag{10}$$

where the running time for DHW operation is calculated with the heating capacity in simultaneous operation according to Formula (11):

$$t_{\mathrm{W,hp,on,j}} = \frac{Q_{\mathrm{W,hp,j}}}{\phi_{\mathrm{W,hp,combi,j}}} \tag{11}$$

and, analogously, according to Formula (12) for space heating operation:

$$t_{\mathrm{H,hp,on,j}} = \frac{Q_{\mathrm{H,hp,j}}}{\phi_{\mathrm{H,hp,combi,j}}} \tag{12}$$

where

$t_{\mathrm{HW,hp,on,combi,max,j}}$	is the maximum possible running time in simultaneous operation at time t_j (s);
$t_{\mathrm{W,hp,on,j}}$	is the running time in DHW operation at time t_j (s);
$Q_{\mathrm{W,hp,j}}$	is the produced heat energy by the heat pump for DHW at time t_j (J);
$\phi_{\mathrm{W,hp,combi,j}}$	is the DHW heating capacity of the heat pump in simultaneous operation at time t_j (W);
$t_{\mathrm{H,hp,on,j}}$	is the running time in space heating operation at time t_j (s);
$Q_{\mathrm{H,hp,j}}$	is the produced heat energy by the heat pump for space heating at time t_j (J);
$\phi_{\mathrm{H,hp,combi,j}}$	is the space heating capacity of the heat pump in simultaneous operation at time t_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 Formula (13):

$$t_{\mathrm{HW,hp,on,combi,j}} = f_{\mathrm{combi}} \cdot t_{\mathrm{HW,hp,on,combi,max,j}} \tag{13}$$

where

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

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

f_{combi} is the correction factor, taking into account the impact of the control system (-).

NOTE Adequate factors for typical controller setting shall be given in a national annex based on a specific evaluation of the system configuration.

The DHW energy and, analogously, the space heating energy produced in simultaneous operation, is calculated by Formula (14):

$$Q_{\mathrm{hp,combi,j}} = \phi_{\mathrm{hp,combi,j}} \cdot t_{\mathrm{hp,on,combi,j}} \tag{14}$$

where

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

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

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

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

$$Q_{\mathrm{hp,sngl,j}} = Q_{\mathrm{hp,j}} - Q_{\mathrm{hp,combi,j}} \tag{15}$$

where

$Q_{\mathrm{hp,sngl,j}}$ is the produced heat by the heat pump in the respective single operation in time t_j (J);

$Q_{\mathrm{hp,j}}$ is the produced heat energy by the heat pump in time t_j (J);

$Q_{\mathrm{hp,combi,j}}$ is the produced heat energy by the heat pump in simultaneous operation in time t_j (J).

NOTE 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 is evaluated by subtracting the storage losses. If no values according to EN 255-3 are available, the subtraction of the storage losses is not necessary.

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

So, the DHW heat energy requirement in DHW-only and in simultaneous operation can be calculated by subtracting the storage losses according to Formula (16):

$$Q_{W,hp,out,sngl,j} = Q_{W,hp,sngl,j} - Q_{W,st,ls,j} \cdot \left(1 - k_{W,bu,j}\right) \cdot \left(1 - f_{combi}\right) \quad (16)$$

where

$Q_{W,hp,out,sngl,j}$	heat requirement of the DHW distribution subsystem covered by the heat pump in DHW-only operation at time t_j (J);
$Q_{W,hp,sngl,j}$	produced DHW energy by the heat pump in DHW-only operation at time t_j (J);
$Q_{W,st,ls,j}$	DHW storage losses at time t_j, calculated in 6.4.2 (J);
f_{combi}	correction factor to take into account controller effect, corresponds to the fraction of simultaneous operation (-);
$k_{W,bu,j}$	fraction of DHW heat energy covered by the back-up heater at time t_j (-).

$$Q_{W,hp,out,combi,j} = Q_{W,hp,combi,j} - Q_{W,st,ls,j} \cdot \left(1 - k_{W,bu,j}\right) \cdot f_{combi} \quad (17)$$

where

$Q_{W,hp,out,combi,j}$	is the heat requirement of the DHW distribution subsystem covered by the heat pump in simultaneous operation at time t_j (J);
$Q_{W,hp,combi,j}$	is the produced DHW energy by the heat pump in simultaneous operation at time t_j (J);
$Q_{W,st,ls,j}$	is the DHW storage losses at time t_j (calculated in 6.4.2) (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 at time t_j (-).

The respective running time in SH-only and DHW-only operation modes are calculated according to Formula (6).

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.

6.6.3 Boundary condition for the total running time

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

$$t_{\mathrm{hp,on,tot,j}} = \min\left(t_{\mathrm{eff,j}}, t_{\mathrm{H,hp,on,sgnl,j}} + t_{\mathrm{W,hp,on,sgnl,j}} + t_{\mathrm{HW,hp,on,combi,j}}\right) \tag{18}$$

where

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

If the calculated total running time is longer than the effective bin time, this is due to a lack of heating capacity of the heat pump. In this case, the effective bin time is the running time and the missing back-up energy is calculated according to 6.6.5.

The same principle applies to the cooling mode.

6.6.4 Back-up calculation: Back-up energy due to lack of capacity (for heating)

The detailed evaluation of the back-up energy is based on the evaluation of the running time according to the boundary conditions given in 6.6.3. The comparison of the running time is accomplished, until the outdoor air temperature is reached, at which the effective time for the time considered is longer than the required running time (step calculation time). The sample balance and the required calculations are summarized in Table 5. If the system is of alternate type, the running time in simultaneous operation is zero.

For the time steps with a lack of running time, i.e. required running time is longer than the effective step calculation 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 using Formula (19), i.e. the back-up heater either supplies heat to the space heating system or the DHW system as calculated in 6.6.5.

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

Label	Value for time t_j
Outdoor air temperature, θ_e(1 K t_j)	$\theta_{e,min}$ / $\theta_{e,min}$ + 1
Energy to be produced for SH, $Q_{H,hp,j}$, acc. to Formula (7)	
Energy to be produced for DHW, $Q_{W,hp,j}$, acc. to Formula (7)	
Heating capacity for SH, $\phi_{H,hp,sngl,j}$, acc. to HP characteristic	
Heating capacity for DHW, $\phi_{W,hp,sngl,j}$, acc. to HP characteristic	
Heating capacity for SH combined, $\phi_{H,hp,combi,j}$, acc. to HP characteristic	
Heating capacity for DHW combined, $\phi_{W,hp,combi,j}$, acc. to HP characteristic	
Running time for SH, $t_{H,hp,on}$, acc. to Formula (6)	
Running time for DHW, $t_{W,hp,on}$, acc. to Formula (6)	
Running time combined, $t_{hp,combi}$, acc. to Formula (12)	
Total required running time $t_{hp,tot,j}$ acc. to Formula (8)	
Theoretical time to cover energy requirement at time t_j, $t_{eff,j}$	
Difference total running time to effective bin time	
Required back-up energy, $Q_{bu,cap,j}$, acc. to Formula (18)	

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

The additional back-up energy due to a lack of capacity is calculated by multiplying the missing running time with the heating capacity of the heat pump in SH-only or DHW-only operation according to Formula (19):

$$Q_{bu,cap,j} = \phi_{hp,sngl,j} \cdot \left(t_{hp,on,tot,j} - t_{eff,j} \right) \tag{19}$$

where

$Q_{bu,cap,i}$ is the additional back-up energy due a lack of capacity (J);

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

$t_{eff,j}$ is the theoretical operating time at time t_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 if the back-up energy is supplied to the space heating or the domestic hot water systems. If no control strategy is known, it is assumed, that the back-up heater supplies 50 % of the back-up energy to the space heating system and 50 % of the back-up energy to the DHW system.

The total fraction of back-up energy can be calculated according to Formula (20):

$$k_{\mathrm{bu,j}} = \frac{Q_{\mathrm{bu,opr,j}} + Q_{\mathrm{bu,cap,j}}}{Q_{\mathrm{gen,out,j}}} = k_{\mathrm{bu,opr,j}} + k_{\mathrm{bu,cap,j}} + \frac{\left(t_{\mathrm{hp,on,tot,j}} - t_{\mathrm{eff,j}}\right) \cdot \phi_{\mathrm{hp,sngl,j}}}{Q_{\mathrm{gen,out,j}}} \quad (20)$$

where

$k_{\mathrm{bu,j}}$	is the fraction of heat energy covered by the back-up heater in the respective operation mode at time t_{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 capacity of the heat pump (J);
$Q_{\mathrm{gen,out,j}}$	is the heat energy requirement of the distribution subsystem at time t_{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 capacity (in case of simplified calculation) (-);
$t_{\mathrm{hp,on,tot,j}}$	is the total (required) running time of the heat pump at time t_{j} (s);
$t_{\mathrm{eff,j}}$	is the theoretical operating time at time t_{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 Formula (20).

The fraction $k_{\mathrm{bu,cap,j}}$ in Formula (20) only exists, if the simplified back-up calculation according to Formula (7) is applied. If the detailed back-up calculation according to 6.6.4 is applied, the fraction is contained in the lack of running time and $k_{\mathrm{bu,cap,j}} = 0$.

6.7 Auxiliary energy

6.7.1 General

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

The auxiliary energy is given by Formula (21):

$$W_{\mathrm{HW,gen,aux}} = \sum_{k} P_{\mathrm{gen,aux,k}} \cdot t_{\mathrm{gen,aux,on,k}} \quad (21)$$

where

$W_{\mathrm{HW,gen,aux}}$	is the total auxiliary energy consumption (J);
$P_{\mathrm{gen,aux,k}}$	is the electrical power 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).

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

— Source pump running time is normally linked to the running time of the heat pump evaluated in 6.6 for the different operation modes.

— For primary pumps, control depends on the installed systems (e.g. is linked to storage control in case of heating buffer storage and thereby linked to the running time of the generator as well). In case of a hydronic distributor, the primary pump can 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 6.6. If a correction for part load operation of the COP is applied according to Annex C, the stand-by expenses are already considered and do not have to be considered here.

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

The following rules apply for estimation of the operating time of the auxiliary:

— for variable speed HP, the relevant running corresponds to the time step calculation when the HP is operating;

— for on/off HP, the relevant running time corresponds to the total running time of the HP at any calculation step as calculated in 6.6, if linked to the on/off control of the HP; if not, the relevant running time corresponds to the time step calculation variable speed.

NOTE Extra time shall be added for pre-conditioning or post operation.

6.7.2 Engine-driven and absorption heat pumps

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

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

6.8.1 Recoverable heat losses from auxiliary consumption

Auxiliary energy is transformed partly to used energy and partly to heat losses as presented in Figure 2 and corresponds to energy flows numbered 10,11, and 12.

Recoverable heat losses to the heat transfer medium are considered totally recovered as considered in Formula (1).

$$Q_{\mathrm{HW,gen,aux,rvd}} = \sum_{k} W_{\mathrm{gen,aux,k}} \cdot k_{\mathrm{gen,aux,ls,rvd,k}} \tag{22}$$

where

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

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

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

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

Heat losses of auxiliaries to the ambiance can be calculated according to Formula (23):

$$Q_{\text{HW,gen,aux,ls}} = \sum_k W_{\text{gen,aux,k}} \cdot k_{\text{gen,aux,ls,k}} \tag{23}$$

and heat losses to the ambiance are assumed recoverable.

Recoverable heat losses can be calculated by a temperature reduction factor linked to location:

$$Q_{\text{H,gen,aux,ls,rbl}} = \sum_k W_{\text{gen,aux,k}} \cdot k_{\text{gen,aux,ls,k}} \cdot \left(1 - b_{\text{gen,aux,k}}\right) \tag{24}$$

where

$Q_{\text{HW,gen,aux,ls}}$ is the heat losses of auxiliary components to the ambiance (J);

$Q_{\text{H,gen,aux,ls,rbl}}$ is the recoverable heat losses of auxiliary components to the ambiance (J);

$W_{\text{gen,aux,k}}$ is the recoverable heat losses of auxiliary components (J);

$k_{\text{gen,aux,ls,k}}$ is the fraction of electrical energy transmitted to the ambiance (-);

NOTE These values should be defined in a national annex. If no national values are specified, default values are given in Annex A.

$b_{\text{gen,aux,k}}$ is the temperature reduction factor for component k linked to location of the component. (-)

NOTE The values of $b_{\text{gen,aux,k}}$ shall be given in a national annex. If no national values are specified, default values are given in Annex A.

6.8.1.1 Total generation subsystem losses

The total envelope heat losses of the generation subsystem can be obtained by a summation over the components, basically heat pump envelope losses, if considered, losses from the engine of engine-driven heat pumps, storage losses for the heating buffer and DHW storage, respectively, and losses of the connecting piping between generator and storage, according to Formula (25):

$$Q_{\text{HW,gen,ls,tot}} = \sum_k Q_{\text{gen,ls,k}} + Q_{\text{HW,gen,aux,ls}} \tag{25}$$

where

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

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

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

6.8.1.2 Recoverable heat losses due to generation subsystem envelope losses

Envelope losses are considered recoverable and can be calculated with a temperature reduction factor according to Formula (26):

$$Q_{\mathrm{H,gen,ls,rbl}} = \sum_{k} Q_{\mathrm{gen,ls,k}} \cdot \left(1 - b_{\mathrm{gen,k}}\right) \tag{26}$$

where

$Q_{\mathrm{H,gen,ls,rbl}}$	is the recoverable heat losses of the generation subsystem (J);
$Q_{\mathrm{gen,ls,k}}$	is the heat 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 (-). NOTE The values should be given in a national annex. If no national values are specified, default values are given in Annex A.

6.8.1.3 Total recoverable heat losses of the generation subsystem

The total recoverable losses can be obtained by a summation of the generation subsystem envelope losses and the losses of auxiliary components to the ambiance according to Formula (27).

$$Q_{\mathrm{H,gen,ls,rbl,tot}} = Q_{\mathrm{H,gen,ls,rbl}} + Q_{\mathrm{H,gen,aux,ls,rbl}} \tag{27}$$

where

$Q_{\mathrm{H,gen,ls,rbl,tot}}$	is the total recoverable heat losses of the generation subsystem (J);
$Q_{\mathrm{H,gen,ls,rbl}}$	is the recoverable heat losses of the generation subsystem (J);
$Q_{\mathrm{H,gen,aux,ls,rbl}}$	is the recoverable heat losses of auxiliary components (J).

6.9 Calculation of total energy input

6.9.1 General consideration for the calculation

The value of the COP is considered as constant for each time period t_{j}.

The same principle for calculation of the different energy flows applies for cooling operation.

6.9.2 Electrically driven heat pumps

6.9.2.1 Electricity input to the heat pump for space heating operation or space cooling 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 Formula (28):

$$E_{\mathrm{H,hp,in}} = \sum_{j=1}^{t_{\max}} \frac{Q_{\mathrm{H,hp,sngl,j}}}{COP_{\mathrm{H,sngl,j}}} + \sum_{j=1}^{t_{\max}} \frac{Q_{\mathrm{H,hp,combi,j}}}{COP_{\mathrm{H,combi,j}}} \tag{28}$$

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 energy produced by the heat pump in space heating-only operation at time t_{j} (J);
$Q_{\mathrm{H,hp,combi,j}}$	is the energy produced by the heat pump for space heating in simultaneous operation at time t_{j} (J);
$COP_{\mathrm{H,sngl,j}}$	is the coefficient of performance in space heating-only operation at the operating point conditions of time t_{j} (W/W);
$COP_{\mathrm{H,combi,j}}$	is the coefficient of performance of space heating in simultaneous operation at the operating point conditions of time t_{j} (W/W);
$t_{\max}$	is the duration of the calculation (-).

NOTE COP is considered as a constant value for each time period t_{j}.

The same principle applies for cooling operation.

6.9.2.2 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 Formula (29).

$$E_{\mathrm{W,hp,in}} = \sum_{j=1}^{t_{\max}} \frac{Q_{\mathrm{W,gen,out,sngl,j}}}{COP_{\mathrm{t,sngl,j}}} + P_{\mathrm{es,sgnl}} \cdot t_{\mathrm{W,sngl,tot}} + \sum_{j=1}^{t_{\max}} \frac{Q_{\mathrm{W,gen,out,combi,j}}}{COP_{\mathrm{t,combi,j}}} + P_{\mathrm{es,combi}} \cdot t_{\mathrm{W,combi,tot}} \tag{29}$$

where

$E_{\mathrm{W,hp,in}}$	is the electrical energy input to the heat pump in DHW mode (J);
$Q_{\mathrm{W,hp,out,sngl,j}}$	is the heat energy requirement of the DHW distribution subsystem at time t_{j} covered by the heat pump in DHW-only operation (J);
$Q_{\mathrm{W,\ hp,out,combi,j}}$	is the heat energy requirement of the DHW distribution subsystem at time t_{j} covered by the heat pump in simultaneous operation (J);
$COP_{\mathrm{t,sngl,j}}$	is the coefficient of performance for the extraction of domestic hot water of time t_{j} in DHW-only operation (W/W);
$COP_{\mathrm{t,combi,j}}$	is the coefficient of performance for the extraction of domestic hot water at time t_{i} in simultaneous operation (W/W);
P_{es}	is the electricity power input to cover storage losses for DHW (W);
t_{W}	is the total time of all calculation periods in DHW-only/simultaneous operation, respectively (s);
$t_{\max}$	is the duration of the calculation period (s).

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

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.

6.9.3 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 Formula (28). This approach corresponds to the system boundary given in 5.2.

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

$$E_{\text{hp,in}} = \sum_{j=1}^{N_{\text{bins}}} \frac{Q_{\text{hp,j}} - Q_{\text{eng,rvd,j}} - k_{\text{gen,aux,ls,rvd}} \cdot W_{\text{HW,gen,aux,j}}}{COP_{\text{j}}} \qquad (30)$$

where

$E_{\text{hp,in}}$ is the fuel or heat energy input to the engine-driven or absorption heat pump in the respective operation mode (J);

$Q_{\text{hp,j}}$ is the produced energy of the heat pump at time t_{j} (J);

$Q_{\text{eng,rvd,j}}$ is the recovered energy from the combustion engine in bin i (only engine-driven heat pumps) (J);

$k_{\text{gen,aux,ls,rvd}}$ is the fraction of auxiliary energy recovered as thermal energy (depending on testing) (-);

$W_{\text{HW,gen,aux,j}}$ is the auxiliary energy consumption at time t_{j} (J);

COP_{j} is the coefficient of performance at the operating point of time t_{j}, taken as performance factor for the respective operation mode (W/W);

t_{max} is the duration of the calculation (s).

The recovered energy $Q_{\text{eng,rvd}}$ shall be calculated based on test results or manufacturer data. If the net or the gross calorific value is to be used depends on which of these values is considered in the testing of the engine-driven heat pump.

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.

6.9.4 Energy input to back-up system

6.9.4.1 Electrical back-up heater

$$E_{\mathrm{HW,bu,in}} = \sum_{j=1}^{N_{\mathrm{bins}}} \frac{\left(Q_{\mathrm{H,gen,out,j}} + Q_{\mathrm{H,gen,ls,j}}\right) \cdot k_{\mathrm{H,bu,j}}}{\eta_{\mathrm{H,bu}}} + \frac{\left(Q_{\mathrm{W,gen,out,j}} + Q_{\mathrm{W,gen,ls,j}}\right) \cdot k_{\mathrm{W,bu,j}}}{\eta_{\mathrm{W,bu}}} \quad (31)$$

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 at time t_{j} (J);
$Q_{\mathrm{H,gen,ls,j}}$	is the heat losses of the generation subsystem due to space heating operation at time t_{j} (J);
$k_{\mathrm{H,bu,j}}$	is the fraction of SH heat energy covered by the back-up heater at time t_{j} (-);
$\eta_{\mathrm{H,bu}}$	is the efficiency of the electrical back-up heater for space heating mode (-);
$Q_{\mathrm{W,gen,out,j}}$	is the heat energy requirement of the DHW distribution subsystem at time t_{j} (J);
$Q_{\mathrm{W,gen,ls,j}}$	is the heat losses of the generation subsystem due to DHW operation at time t_{j} (J);
$k_{\mathrm{W,bu,j}}$	is the fraction of DHW heat energy covered by the back-up heater at time t_{j} (-);
$\eta_{\mathrm{bu,DHW}}$	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 heating shall be given in a national annex or determined based on the system configuration and the product.

6.9.4.2 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 ISO 13675 for combustion boilers.

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

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

$$E_{\mathrm{HW,gen,in}} = E_{\mathrm{H,hp,in}} + E_{\mathrm{W,hp,in}} + E_{\mathrm{HW,bu,in}} \quad (32)$$

where

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

6.9.6 Ambient heat used by the generation subsystem

The amount of ambient heat used for the produced heat energy of the heat pump to cover the space heating and/or DHW requirement and generation subsystem losses is calculated according to Formula (1), where the recovered auxiliary energy is to be set to $k_{\mathrm{gen,aux,ls,rvd}}$ = 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.

6.10 Summary of output values

6.10.1 Required outputs

— Energy input to cover the heat requirement of the distribution subsystems [6.9.5, Formula (31)]

— Total losses of the generation subsystem [6.8.1.1, Formula (24)]

— Total recoverable losses of the generation subsystem [6.8.1.3, Formula (26)]

— Total auxiliary energy input to the generation subsystem [6.7, Formula (20)]

— Total heat produced by the back-up heater

— Totally used ambient heat [6.9.6, Formula (1)]

6.10.2 Optional outputs

— Total heat produced by the heat pump

Annex A
(informative)

Construction of COP matrix with a single test result

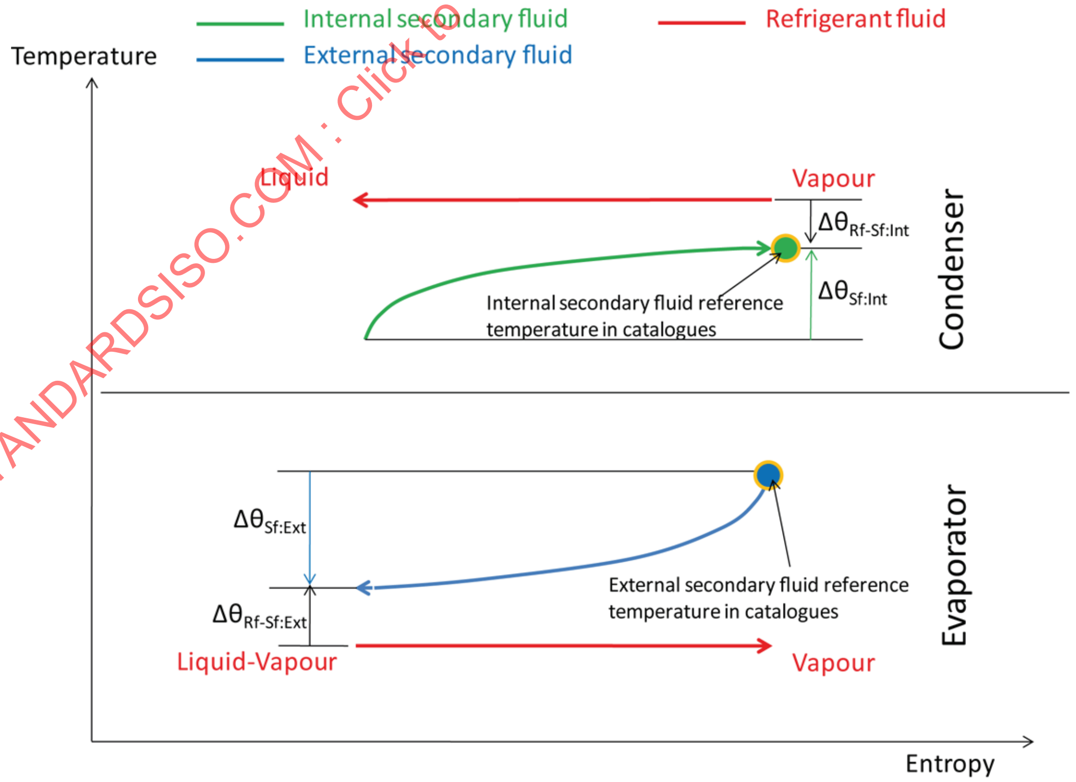
A.1 Introduction

The performances of heat pumps are usually referred to the following:

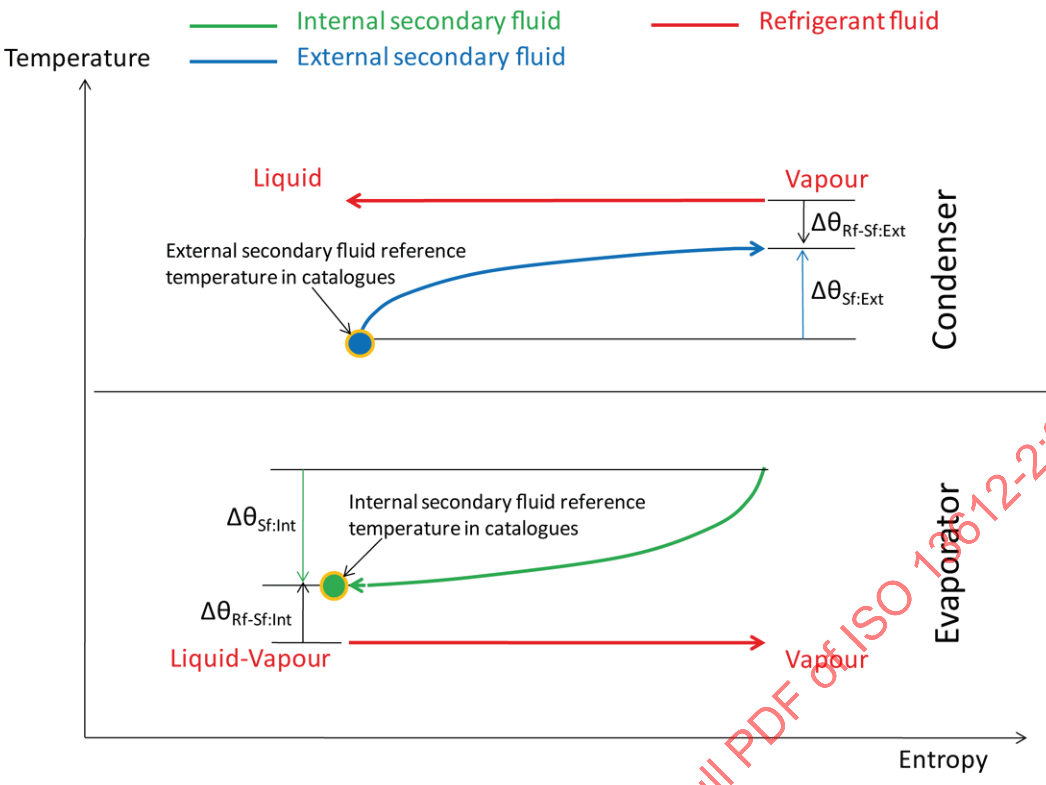
— the outlet temperature of the internal secondary fluid, $\theta_{\mathrm{Sf:i}}$;

— the inlet temperature of the external secondary fluid, $\theta_{\mathrm{Sf:e}}$.

The real thermodynamic inverse cycle involves the refrigerant fluid so refrigerant temperatures should be used as consistent references when the COP of heat pumps is calculated. But the calculation of actual refrigerant fluid temperatures is complex due to refrigerant fluid characteristics, compressor performances, and heat exchanger efficiencies in each specific condition. Typical temperature differences between the refrigerant fluid and the secondary fluids ($\Delta\theta_{\mathrm{Rf-Sf}}$) are used in the heat pump design stage. They usually depend on the secondary fluid characteristics and on the running mode. So the temperature differences between the refrigerant fluid and the secondary fluids, namely $\Delta\theta_{\mathrm{Rf-Sf:Int}}$ and $\Delta\theta_{\mathrm{Rf-Sf:Ext}}$ for the internal and external secondary fluids, respectively, can be defined at design conditions, depending on secondary fluids and the running mode. In Figure A.1 the aforementioned concept is clearly expressed, together with the most relevant temperature differences taking place in the evaporator and the condenser.



a) Heating case



b) Cooling case

Figure A.1 — Graphical explanation of temperature differences in the evaporator and the condenser

In typical evaporator and condenser design, the following values may be considered:

$$\Delta\theta_{\mathrm{Sf:Int}} = \begin{cases} 5\,\mathrm{K} \text{ if the internal secondary fluid is water} \\ 10\,\mathrm{K} \text{ if the internal secondary fluid is air} \end{cases} \tag{A.1}$$

$$\Delta\theta_{\mathrm{Sf:Ext}} = \begin{cases} 5\,\mathrm{K} \text{ if the external secondary fluid is water} \\ 10\,\mathrm{K} \text{ if the external secondary fluid is air} \end{cases} \tag{A.2}$$

$$\Delta\theta_{\mathrm{Rf-Sf:Int}} = 5\,\mathrm{K} \tag{A.3}$$

$$\Delta\theta_{\mathrm{Rf-Sf:Ext}} = 5\,\mathrm{K} \tag{A.4}$$

As a consequence, if one considers catalogue data as references for secondary fluid temperatures (i.e. the inlet temperature for the external secondary fluid and the outlet temperature for the internal secondary fluid), the average refrigerant temperatures at the evaporator and at the condenser can be calculated using Formula (A.5) and Formula (A.6) and Table A.1 as a first approximation for any operating condition.

$$\theta_{\mathrm{Rf-Sf:i}} = \theta_{\mathrm{Sf:Int}} + \Delta\theta_{\mathrm{Rf-Sf:Int}} \tag{A.5}$$

$$\theta_{\mathrm{Rf-Sf:e}} = \theta_{\mathrm{Sf:Ext}} + \Delta\theta_{\mathrm{Rf-Sf:Ext}} \tag{A.6}$$

Table A.1 — Temperature differences between the average and secondary fluid reference temperatures

Running mode	Secondary fluid kind	Refrigerant fluid — secondary fluid temperature difference K	
		Internal side ($\Delta\theta_{Rf-Sf:Int}$)	External side ($\Delta\theta_{Rf-Sf:Ext}$)
Winter	Air	+5	-15
	Water	+5	-10
Summer	Air	-5	+15
	Water	-5	+10

Table A.1 allows a simplified assumption of values for $\Delta\theta_{Rf-Sf:Int}$ and $\Delta\theta_{Rf-Sf:Ext}$. Further accuracy could be reached calculating $\Delta\theta_{Rf-Sf:Int}$ and $\Delta\theta_{Rf-Sf:Ext}$ basing on the evaporator and condenser capacities and on secondary fluids flow rates. But such an approach would require iterative calculations, so it is not considered in this part of ISO 13612 for the sake of simplicity.

Ideal inverse thermodynamic cycles can be assumed as references in the basic conceptualization of conventional heat pumps. From thermodynamics, the value of COP for ideal inverse thermodynamic cycles operating between the internal environment (at average temperature $\bar{\theta}_{Int}$) and the external one (at average temperature $\bar{\theta}_{e}$) comes from Formula (A.7).

$$COP^*_{\bar{\theta}_{Int},\bar{\theta}_{Ext}} = \frac{\bar{\theta}_{Int} + 273,15}{\left|\bar{\theta}_{Int} - \bar{\theta}_{Ext}\right|} \tag{A.7}$$

We assume that $\bar{\theta}_{Int}$ and $\bar{\theta}_{Ext}$ correspond to the reference temperatures of the refrigerant fluid, respectively, at the evaporator and at the condenser (in cooling mode) or at the condenser and at the evaporator (in heating mode). As a consequence, Formula (A.7) can be approximated as follows:

$$COP^*_{\theta_{Rf:Int},\theta_{Rf:Ext}} = \frac{\theta_{Rf:Int} + 273,15}{\left|\theta_{Rf:Int} - \theta_{Rf:Ext}\right|} \tag{A.8}$$

For convenience, Formula (A.8) should be linked to internal and external secondary fluid reference temperatures rather than to refrigerant temperatures, so the following formula can be easily obtained:

$$COP^*_{\theta_{Sf:Int},\theta_{Sf:Ext}} = \frac{\left(\theta_{Sf:Int} + \Delta\theta_{Rf-Sf:Int}\right) + 273,15}{\left|\left(\theta_{Sf:Int} + \Delta\theta_{Rfr-Sf:Int}\right) - \left(\theta_{Sf:Ext} + \Delta\theta_{Rf-Sf:Ext}\right)\right|} \tag{A.9}$$

The COP for ideal inverse thermodynamic cycles at the same operating temperatures, namely $COP^*_{\theta_{Sf:Int},\theta_{Sf:Ext}}$, can thus be calculated. If the actual COP is known for the real heat pump working at the same operating temperatures, namely $COP_{\theta_{Sf:Int},\theta_{Sf:Ext},f=1}$, then it is possible to define how worse the real inverse thermodynamic cycle is compared with the ideal one. That is defined through the so-called efficiency of the 2nd principle of thermodynamics, namely $\eta_{2ndP,\theta_{Sf:Int},\theta_{Sf:Ext}}$, defined through Formula (A.10).

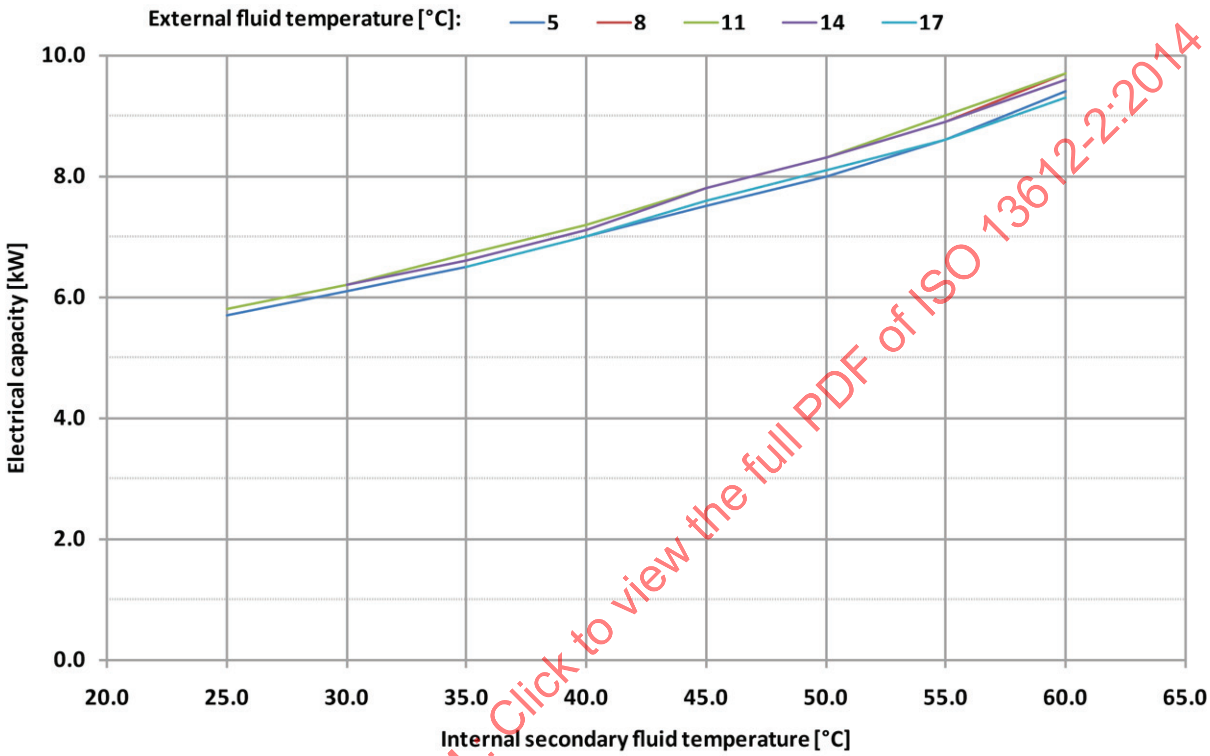
$$\eta_{2ndP,\theta_{Sf:Int},\theta_{Sf:Ext}} = \frac{COP_{\theta_{Sf:Int},\theta_{Sf:Ext},f=1}}{COP^*_{\theta_{Sf:Int},\theta_{Sf:Ext}}} \tag{A.10}$$

The meaning of the 2nd principle efficiency can be clearly explained in the following way: The higher the 2nd principle efficiency, the closer the performances of the actual heat pump at the considered operating conditions, compared with the ideal heat pump at the same operating conditions.

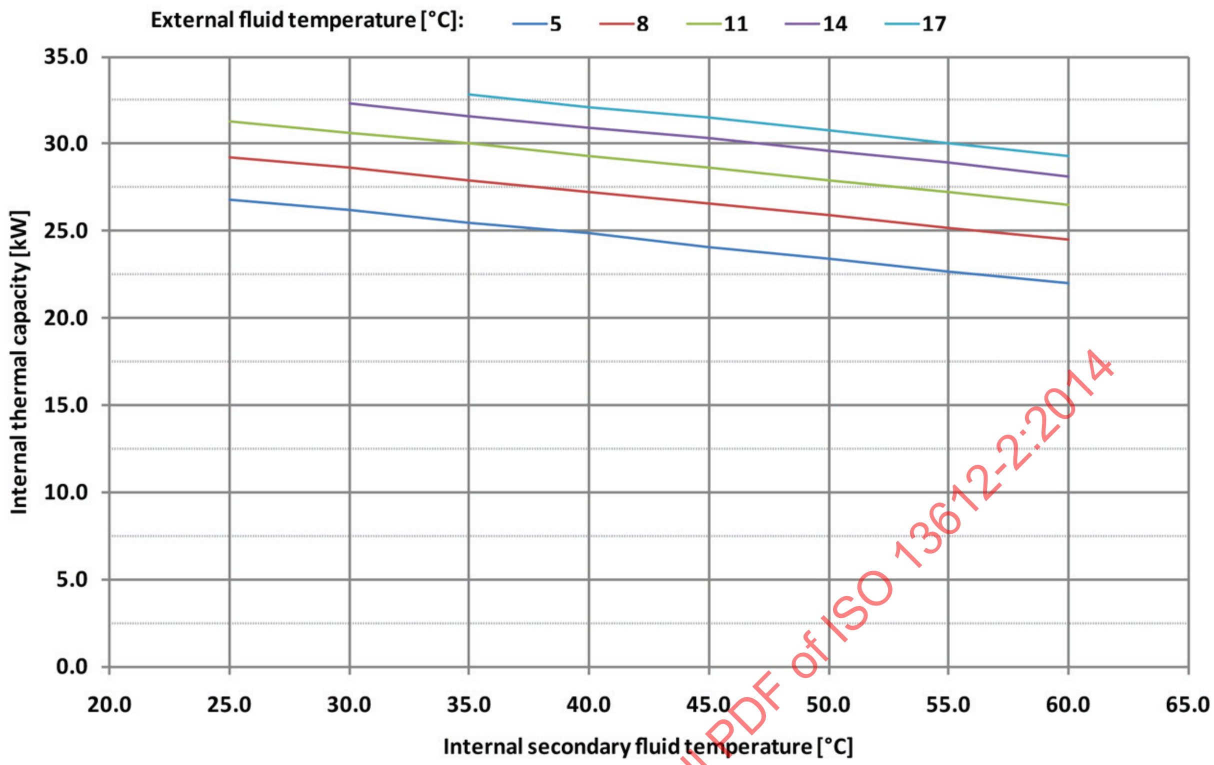
In the end, the examination of data declared by companies shows the following characteristics of heat pumps (as shown in Figure A.2):

a) As a first degree of approximation, the electrical capacity can be considered constant varying the secondary fluid temperature at the evaporator, if the secondary fluid temperature at the condenser is constant [Figure A.2 a)].

b) As a first degree of approximation, the condenser capacity can be considered locally constant varying the secondary fluid temperature at the condenser, if the secondary fluid temperature at the evaporator is constant [Figure A.2 b)].



a) Electrical capacity versus external and internal secondary fluid temperatures (heating mode)



b) Thermal capacity at the internal side versus external and internal secondary fluid temperatures (heating mode)

Figure A.2 — Electrical capacity and thermal capacity at the internal side versus external and internal secondary fluid temperatures (heating mode)

These two characteristics will be used to provide convenient starting values in the calculation procedures for the estimation of COP and capacities at any operating condition.

A.2 Detailed calculation procedure

A.2.1 Introduction and input data

The present calculation procedure is used to estimate COP and capacities at any operating condition. It gives reliable results, even starting from the minimum amount of data such as the performances at one nominal condition point. The use of more rated points provides better accuracy in calculations. At each rated point, the performances of the heat pump must be expressed in terms of the following magnitudes:

— running mode (heating or cooling);

— internal secondary fluid outlet temperature, $\theta_{\mathrm{Sf:Int}}$;

— external secondary fluid inlet temperature, $\theta_{\mathrm{Sf:Ext}}$;

— internal thermal capacity, P_{e};

— electrical capacity, P_{El} (for vapour compression heat pumps) or generator capacity, P_{Gen} (for absorption heat pumps).

A.2.2 Calculation procedure

A.2.2.1 General

In the following steps, the detailed calculation procedure for the estimation of COP and full capacities is explained. At first the nominal point(s) are characterized via the calculation of quantities to be used in the rest of the procedure. Then, from the values declared at the nominal point(s), the COP and capacities at full load operation can be calculated for any user-defined couple of internal and external secondary fluid temperatures ($\theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}$), within the declared range for operating conditions.

A.2.2.2 Characterization of rated points

A.2.2.2.1 Vapour compression heat pumps

At each point rated at full load, the following calculations apply.

— Calculation of the Carnot inverse cycle COP, basing on Table A.1:

$$COP^*_{\theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}} = \frac{\left(\theta_{\text{Sf:Int}} + \Delta\theta_{\text{Rf-Sf:Int}}\right) + 273{,}15}{\left|\left(\theta_{\text{Sf:Int}} + \Delta\theta_{\text{Rf-Sf:Int}}\right) - \left(\theta_{\text{Sf:Ext}} + \Delta\theta_{\text{Rf-Sf:Ext}}\right)\right|} \tag{A.11}$$

— Calculation of the actual rated full load COP:

$$COP_{\theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}, \text{f}=1} = \frac{P_{\text{Int}, \theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}, \text{f}=1}}{P_{\text{El}, \theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}, \text{f}=1}} \tag{A.12}$$

— Calculation of the thermal capacity at the external side:

$$P_{\text{Ext}, \theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}, \text{f}=1} = P_{\text{Int}, \theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}, \text{f}=1} - P_{\text{El}, \theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}, \text{f}=1} \quad \text{in heating mode} \tag{A.13}$$

$$P_{\text{Ext}, \theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}, \text{f}=1} = P_{\text{Int}, \theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}, \text{f}=1} + P_{\text{El}, \theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}, \text{f}=1} \quad \text{in cooling mode} \tag{A.14}$$

— Calculation of the 2nd principle of thermodynamics efficiency:

$$\eta_{\text{2ndP}, \theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}, \text{f}=1} = \frac{COP_{\theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}, \text{f}=1}}{COP^*_{\theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}}} \tag{A.15}$$

A.2.2.2.2 Absorption heat pumps

At each point rated at full load:

— Calculation of the Carnot inverse cycle COP, basing on Table A.1:

$$COP^*_{\bar{\theta}_{\text{Int}}, \bar{\theta}_{\text{Ext}}} = \frac{\bar{\theta}_{\text{Gen}} - \text{Max}\left(\theta_{\text{Sf:Int}} + \Delta\theta_{\text{Rf-Sf:Int}}, \theta_{\text{Sf:Ext}} + \Delta\theta_{\text{Rf-Sf:Ext}}\right)}{\left|\left(\theta_{\text{Sf:Int}} + \Delta\theta_{\text{Rf-Sf:Int}}\right) - \left(\theta_{\text{Sf:Ext}} + \Delta\theta_{\text{Rf-Sf:Ext}}\right)\right|} \cdot \frac{\theta_{\text{Sf:Int}} + \Delta\theta_{\text{Rf-Sf:Int}} + 273{,}15}{\bar{\theta}_{\text{Gen}} + 273{,}15}$$

(A.16)

— Calculation of the actual rated full load COP:

$$COP_{\theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}, \text{f}=1} = \frac{P_{\text{Int}, \theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}, \text{f}=1}}{P_{\text{Gen}, \theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}, \text{f}=1}} \tag{A.17}$$

— Calculation of the thermal capacity at the external side:

$$P_{\text{Ext},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}} = P_{\text{Int},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}} - P_{\text{Gen},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}} \quad \text{in heating mode} \tag{A.18}$$

$$P_{\text{Ext},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}} = P_{\text{Int},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}} + P_{\text{Gen},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}} \quad \text{in cooling mode} \tag{A.19}$$

— Calculation of the 2nd principle of thermodynamics efficiency:

$$\eta_{\text{2ndP},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}} = \frac{COP_{\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}}}{COP^*_{\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}}}} \tag{A.20}$$

A.2.2.3 Calculation of COP and capacities of heat pumps at full load

A.2.2.3.1 Vapour compression heat pumps

COP and capacities for any couple of temperature conditions ($\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}}$), at full load conditions, arise from an elaboration of the same values rated at one of the declared nominal points. In particular, the nominal point closest to the specific temperature conditions has to be identified. Any reference to the closest rated point is performed using parentheses and subscript "Ref". So $(\theta_{\text{Sf:Int}})_{\text{Ref}}$, $(\theta_{\text{Sf:Ext}})_{\text{Ref}}$, $(P_{\text{Int}})_{\text{Ref}}$, $(P_{\text{El}})_{\text{Ref}}$, $\left(COP^*_{\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}}}\right)_{\text{Ref}}$, $\left(COP_{\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}}\right)_{\text{Ref}}$, $\left(P_{\text{Ext},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}}\right)_{\text{Ref}}$, and $\left(\eta_{\text{2ndP},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}}\right)_{\text{Ref}}$ are known. The calculation procedure depends on the running mode.

— In heating mode:

— Calculation of ideal COP at temperatures $\theta_{\text{Sf:Int}}$ and $(\theta_{\text{Sf:Ext}})_{\text{Ref}}$:

$$COP^*_{\theta_{\text{Sf:Int}},(\theta_{\text{Sf:Ext}})_{\text{Ref}}} = \frac{\left(\theta_{\text{Sf:Int}} + \Delta\theta_{\text{Rf-Sf:Int}}\right) + 273{,}15}{\left|\left(\theta_{\text{Sf:Int}} + \Delta\theta_{\text{Rf-Sf:Int}}\right) - \left(\theta_{\text{Sf:Ext}} + \Delta\theta_{\text{Rf-Sf:Ext}}\right)_{\text{Ref}}\right|} \tag{A.21}$$

— Calculation of the electrical capacity:

$$P_{\text{El},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},1} = \frac{\left(P_{\text{Int},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}}\right)_{\text{Ref}}}{\left(\eta_{\text{2ndP},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}}\right)_{\text{Ref}} \cdot COP^*_{\theta_{\text{Sf:Int}},(\theta_{\text{Sf:Ext}})_{\text{Ref}}}} \tag{A.22}$$

— Calculation of ideal COP at temperatures $\theta_{\text{Sf:Int}}$ and $\theta_{\text{Sf:Ext}}$:

$$COP^*_{\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}}} = \frac{\left(\theta_{\text{Sf:Int}} + \Delta\theta_{\text{Rf-Sf:Int}}\right) + 273{,}15}{\left|\left(\theta_{\text{Sf:Int}} + \Delta\theta_{\text{Rf-Sf:Int}}\right) - \left(\theta_{\text{Sf:Ext}} + \Delta\theta_{\text{Rf-Sf:Ext}}\right)\right|} \tag{A.23}$$

— Calculation of the internal thermal capacity:

$$P_{\text{Int},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}} = P_{\text{El},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}} \cdot COP^*_{\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}}} \cdot \eta_{\text{2ndP},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}} \tag{A.24}$$

— Calculation of the external thermal capacity:

$$P_{\text{Ext},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}} = P_{\text{Int},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}} - P_{\text{El},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}} \tag{A.25}$$

— In cooling mode:

— Calculation of ideal COP at temperatures $(\theta_{\text{Sf:Int}})_{\text{Ref}}$ and $\theta_{\text{Sf:Ext}}$:

$$COP^*_{(\theta_{\text{Sf:Int}})_{\text{Ref}},\theta_{\text{Sf:Ext}}} = \frac{\left(\theta_{\text{Sf:Int}} + \Delta\theta_{\text{Rf-Sf:Int}}\right)_{\text{Ref}} + 273{,}15}{\left|\left(\theta_{\text{Sf:Int}} + \Delta\theta_{\text{Rf-Sf:Int}}\right)_{\text{Ref}} - \left(\theta_{\text{Sf:Ext}} + \Delta\theta_{\text{Rf-Sf:Ext}}\right)\right|} \tag{A.26}$$

— Calculation of the electrical capacity:

$$P_{\mathrm{El},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} = \frac{\left(P_{\mathrm{Ext},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}}\right)_{\mathrm{Ref}}}{1+\left(\eta_{\mathrm{2ndP},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}}}\right)_{\mathrm{Ref}}\cdot COP^{*}_{(\theta_{\mathrm{Sf:Int}})_{\mathrm{Ref}},\theta_{\mathrm{Sf:Ext}}}} \quad \text{(A.27)}$$

— Calculation of ideal COP at temperatures $\theta_{\mathrm{Sf:Int}}$ and $\theta_{\mathrm{Sf:Ext}}$:

$$COP^{*}_{\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}}} = \frac{\left(\theta_{\mathrm{Sf:Int}}+\Delta\theta_{\mathrm{Rf-Sf:Int}}\right)+273,15}{\left|\left(\theta_{\mathrm{Sf:Int}}+\Delta\theta_{\mathrm{Rf-Sf:Int}}\right)-\left(\theta_{\mathrm{Sf:Ext}}+\Delta\theta_{\mathrm{Rf-Sf:Ext}}\right)\right|} \quad \text{(A.28)}$$

— Calculation of the internal thermal capacity:

$$P_{\mathrm{Int},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} = P_{\mathrm{El},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}}\cdot COP^{*}_{\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}}}\cdot\eta_{\mathrm{2ndP},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}}} \quad \text{(A.29)}$$

— Calculation of the external thermal capacity:

$$P_{\mathrm{Ext},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} = P_{\mathrm{Int},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} + P_{\mathrm{El},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} \quad \text{(A.30)}$$

A.2.2.3.2 Absorption heat pumps

— Calculation of ideal COP at temperatures $\theta_{\mathrm{Sf:Int}}$ and $\theta_{\mathrm{Sf:Ext}}$:

$$COP^{*}_{\bar{\theta}_{\mathrm{Int}},\bar{\theta}_{\mathrm{Ext}}} = \frac{\bar{\theta}_{\mathrm{Gen}} - Max\left(\theta_{\mathrm{Sf:Int}}+\Delta\theta_{\mathrm{Rf\text{-}Sf:Int}},\theta_{\mathrm{Sf:Ext}}+\Delta\theta_{\mathrm{Rf\text{-}Sf:Ext}}\right)}{\left|\left(\theta_{\mathrm{Sf:Int}}+\Delta\theta_{\mathrm{Rf\text{-}Sf:Int}}\right)-\left(\theta_{\mathrm{Sf:Ext}}+\Delta\theta_{\mathrm{Rf\text{-}Sf:Ext}}\right)\right|}\cdot\frac{\theta_{\mathrm{Sf:Int}}+\Delta\theta_{\mathrm{Rf\text{-}Sf:Int}}+273,15}{\bar{\theta}_{\mathrm{Gen}}+273,15}$$

(A.31)

— Calculation of COP at temperatures $\theta_{\mathrm{Sf:Int}}$ and $\theta_{\mathrm{Sf:Ext}}$:

$$COP_{\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} = \left(\eta_{\mathrm{2ndP},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}}\right)_{\mathrm{Ref}}\cdot COP^{*}_{\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}}} \quad \text{(A.32)}$$

— Calculation of the generator capacity at temperatures $\theta_{\mathrm{Sf:Int}}$ and $\theta_{\mathrm{Sf:Ext}}$:

$$P_{\mathrm{Gen},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} = \left(P_{\mathrm{Gen}}\right)_{\mathrm{Ref}} \quad \text{(A.33)}$$

— Calculation of the internal thermal capacity at temperatures $\theta_{\mathrm{Sf:Int}}$ and $\theta_{\mathrm{Sf:Ext}}$:

$$P_{\mathrm{Int},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} = COP_{\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}}\cdot\left(P_{\mathrm{Gen}}\right)_{\mathrm{Ref}} \quad \text{(A.34)}$$

— Calculation of the external thermal capacity at temperatures $\theta_{\mathrm{Sf:Int}}$ and $\theta_{\mathrm{Sf:Ext}}$:

$$P_{\mathrm{Ext},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} = P_{\mathrm{Int},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} - P_{\mathrm{Gen},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} \text{ in heating mode}$$

(A.35)

$$P_{\mathrm{Ext},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} = P_{\mathrm{Int},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} + P_{\mathrm{Gen},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} \text{ in cooling mode}$$

(A.36)

A.3 Example of input data and output results

Together with this part of ISO 13612, four spreadsheets have been delivered. They show examples of application of the method in this part of ISO 13612 for two heat pumps, both used in heating and in cooling modes.

In Table A.2 and Figure A.3, examples of input and results are shown.

Table A.2 — Example of input and output for Method A — Cooling mode

GENERAL DESCRIPTION OF THE HEAT PUMP AND NOMINAL CONDITIONS AND CAPACITIES		
	RunningMode ("HEATING"/"COOLING")	COOLING
Secondary fluids	IntFluid ("WATER"/"AIR")	AIR
	ExtFluid ("WATER"/"AIR")	WATER
	TIntFluid_Outlet (Nominal conditions) [°C]	7.0
	TExtFluid_Inlet (Nominal conditions) [°C]	35.0
	IntCapacity (Nominal conditions) [kW]	4.7
	ElCapacity (Nominal conditions) [kW]	1.8
	COP (Nominal conditions) [-]	2.61
	COP_Ideal (Nominal conditions) [-]	6.40
	Eta2ndPrinc (Nominal conditions) [-]	0.41

ADDITIONAL RATED POINTS FOR PART LOAD CALCULATIONS				
Code [Alfa]	A	B	C	D
Part load ratio [%, ref. to Nom]	100.0	80.0	44.0	33.0
TIntFluid_Outlet [°C]	7.0	7.0	7.0	7.0
TExtFluid_Inlet [°C]	35.0	35.0	35.0	35.0
COP [-]	3.0	3.2	3.5	3.6
Part load ratio [%, ref. to actual]	100.0	80.0	44.0	33.0
COPFactor [-]	1.14	1.24	1.35	1.38
Use standard parzialization curve	YES			
Control	ON-OFF			

USER INQUIRIES (i.e.: example of results)								
$\theta_{s,i}$ [°C]	$\theta_{s,e}$ [°C]	$P_{Int,\theta s,i,\theta s,e,X\%}$ [kW]	Part Load Ratio X [%]	Correction factor for COP [-]	$COP_{100\%}$ [-]	COP [-]	$P_{El,\theta s,i,\theta s,e,X\%}$ [kW]	$P_{Ext,\theta s,i,\theta s,e,X\%}$ [kW]
6.0	30.0	5.0	106	1.02	2.87	2.93	1.71	3.29
8.0	36.0	4.0	83.4	0.94	2.62	2.46	1.62	2.38
12.0	40.0	3.0	58	0.83	2.66	2.22	1.35	1.65
6.0	32.0	2.0	42.9	0.76	2.73	2.06	0.97	1.03
12.0	36.0	3.0	55.9	0.82	2.93	2.42	1.24	1.76
15.0	38.0	2.0	34.8	0.71	3.04	2.14	0.93	1.07
6.0	25.0	1.0	20.6	0.58	3.29	1.9	0.53	0.47

In Table A.2, input areas are yellow shaded, whereas outputs are in the green area. In particular, the input areas are two: the upper one is used to provide data needed for full load COP and capacities calculation, whereas the lower area is used for calculations at part loads. It is clear that the amount of information needed by Method A for the achievement of full load COP and capacities at any temperature is really low and easy-to-find in any catalogue or data sheet. For instance, in the examples presented in the attached spreadsheets, just one nominal point is used for the calculation of COP and capacities at full load conditions. The rest of input and output areas will be described in the section regarding part load calculations.

TABLES FOR COMPARISON (used as references for method validation)

IntCapacity [kW]

DECLARED VALUES: Ext Temp [°C] \ Int Temp [°C]	6	7	8	10	12	15	16	18
25	5.0	5.2	5.4	5.7	6.1	6.6	6.7	7.1
30	4.8	5.0	5.1	5.4	5.8	6.3	6.4	6.8
35	4.5	4.7	4.8	5.1	5.5	5.9	6.0	6.4
40	4.3	4.4	4.5	4.8	5.1	5.6	6.0	-
43	4.1	4.2	4.3	4.6	4.9	-	-	-

CALCULATED VALUES: Ext Temp [°C] \ Int Temp [°C]	6	7	8	10	12	15	16	18
25	4.9	5.0	5.2	5.6	6.0	6.8	7.1	7.8
30	4.7	4.9	5.0	5.3	5.7	6.3	6.6	7.1
35	4.6	4.7	4.8	5.1	5.4	5.9	6.1	6.6
40	4.4	4.6	4.7	4.9	5.2	5.6	5.8	-
43	4.4	4.5	4.6	4.8	5.0	-	-	-

GAP: Ext Temp [°C] \ Int Temp [°C]	6	7	8	10	12	15	16	18
25	-4%	-4%	-3%	-2%	-1%	3%	6%	10%
30	-2%	-2%	-2%	-2%	-1%	1%	3%	5%
35	1%	0%	0%	-1%	-1%	0%	2%	2%
40	5%	4%	3%	2%	1%	0%	-4%	-
43	8%	6%	5%	3%	2%	-	-	-

COP [-]

DECLARED VALUES: Ext Temp [°C] \ Int Temp [°C]	6	7	8	10	12	15	16	18
25	3.29	3.36	3.45	3.58	3.70	3.88	3.94	4.06
30	2.89	2.95	3.01	3.12	3.24	3.38	3.44	3.55
35	2.51	2.56	2.61	2.71	2.79	2.94	2.98	3.07
40	2.15	2.18	2.23	2.31	2.40	2.52	2.86	-
43	1.93	1.98	2.02	2.10	2.18	-	-	-

CALCULATED VALUES: Ext Temp [°C] \ Int Temp [°C]	6	7	8	10	12	15	16	18
25	3.29	3.40	3.52	3.78	4.08	4.62	4.83	5.31
30	2.87	2.95	3.05	3.24	3.46	3.85	4.00	4.32
35	2.54	2.61	2.68	2.84	3.01	3.30	3.41	3.65
40	2.28	2.34	2.40	2.52	2.66	2.89	2.97	-
43	2.15	2.20	2.25	2.36	2.49	-	-	-

GAP: Ext Temp [°C] \ Int Temp [°C]	6	7	8	10	12	15	16	18
25	0%	1%	2%	6%	10%	19%	23%	31%
30	-1%	0%	1%	4%	7%	14%	16%	22%
35	1%	2%	3%	5%	8%	12%	14%	19%
40	6%	7%	7%	9%	11%	15%	4%	-
43	11%	11%	12%	13%	14%	-	-	-

ElCapacity [kW]

DECLARED VALUES: Ext Temp [°C] \ Int Temp [°C]	6	7	8	10	12	15	16	18
25	1.5	1.6	1.6	1.6	1.6	1.7	1.7	1.8
30	1.7	1.7	1.7	1.7	1.8	1.9	1.9	1.9
35	1.8	1.8	1.9	1.9	2.0	2.0	2.0	2.1
40	2.0	2.0	2.0	2.1	2.1	2.2	2.1	-
43	2.1	2.1	2.2	2.2	2.3	-	-	-

CALCULATED VALUES: Ext Temp [°C] \ Int Temp [°C]	6	7	8	10	12	15	16	18
25	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
30	1.6	1.6	1.6	1.6	1.6	1.6	1.6	1.6
35	1.8	1.8	1.8	1.8	1.8	1.8	1.8	1.8
40	1.9	1.9	1.9	1.9	1.9	1.9	1.9	-
43	2.0	2.0	2.0	2.0	2.0	-	-	-

GAP: Ext Temp [°C] \ Int Temp [°C]	6	7	8	10	12	15	16	18
25	-3%	-5%	-5%	-8%	-10%	-13%	-13%	-16%
30	-1%	-2%	-3%	-6%	-8%	-11%	-11%	-14%
35	-1%	-2%	-3%	-5%	-8%	-11%	-11%	-14%
40	-2%	-3%	-4%	-7%	-9%	-13%	-7%	-
43	-3%	-4%	-6%	-8%	-11%	-	-	-

ExtCapacity [kW]

DECLARED VALUES: Ext Temp [°C] \ Int Temp [°C]	6	7	8	10	12	15	16	18
25	6.6	6.8	6.9	7.3	7.7	8.3	8.4	8.9
30	6.5	6.6	6.8	7.2	7.5	8.1	8.2	8.7
35	6.4	6.5	6.7	7.0	7.4	8.0	8.1	8.5
40	6.2	6.4	6.6	6.9	7.3	7.9	8.1	-
43	6.2	6.3	6.5	6.9	7.2	-	-	-

CALCULATED VALUES: Ext Temp [°C] \ Int Temp [°C]	6	7	8	10	12	15	16	18
25	6.3	6.5	6.7	7.1	7.5	8.3	8.6	9.3
30	6.4	6.5	6.6	7.0	7.3	8.0	8.2	8.8
35	6.4	6.5	6.6	6.9	7.2	7.7	7.9	8.4
40	6.4	6.5	6.6	6.9	7.1	7.6	7.7	-
43	6.4	6.5	6.6	6.8	7.1	-	-	-

GAP: Ext Temp [°C] \ Int Temp [°C]	6	7	8	10	12	15	16	18
25	-4%	-4%	-4%	-4%	-3%	0%	2%	5%
30	-2%	-2%	-2%	-3%	-3%	-2%	0%	1%
35	0%	0%	-1%	-2%	-2%	-3%	-2%	-2%
40	3%	2%	1%	-1%	-2%	-4%	-5%	-
43	4%	3%	2%	0%	-2%	-	-	-

Figure A.3 — Example of tables provided by the manufacturer and tables built via Method A — Cooling mode

In Figure A.3, full load outputs from Method A are compared with data declared by the manufacturer. It is clear that, when outputs are desired for operating points not far from the given nominal point, the gap of Method A from rated performances is within 10 %. The results are especially good as regards electric capacity and COP. Moreover, the largest differences are shown when the temperature at the condenser side varies a lot from nominal conditions. So the results show good accuracy, even more if we consider that heat pumps are usually used in well-defined ranges of internal temperature (for instance, radiant panel heating, fan coil or radiator temperature ranges), and, basing on that specific range, manufacturers provide nominal data closer to each specific application needs.

The largest differences have been experienced in the case of defrost. In that case, Method A should be refined. The refinement could be performed in two ways:

— improving and making the calculation model more complex, in order to take into account defrost activation and maintain good accuracy with just one nominal point given by the manufacturer. The main variations in the model could consist of the following actions:

  — split of electrical capacity into compressor, defrost system, control system, pumps and fans electrical capacities, so that the COP of the cycle can be more consistently defined;

  — calculation of temperature differences between refrigerant and secondary fluids basing on instantaneous thermal capacities and fluid flows at evaporator and condenser;

— requiring one more nominal point, placed in the defrost activation temperature range, that can be used as a reference point when the heat pump works in defrost conditions.

The second option appears as the most affordable in order to limit the complexity of the method and maintain good calculation accuracy, whereas the first option is feasible for software and advanced modelling.

A.4 Advantages and disadvantages

Method A has the following advantages:

— input data consist of really few values, easy to collect even for old heat pumps;

— good calculation accuracy when the nominal rated point is in the middle of the actual operating conditions;

— the calculation procedure has a clear and consistent physics basis and does not consist of just mere coefficients adapted for convenience;

— the method can be improved *via* easy enhancements, not considered at this stage (in order to maximize easiness of calculations). The possible enhancements can be resumed in the following list:

  — split of electrical capacity into compressor, defrost system, control system, pumps and fans electrical capacities, so that the COP of the cycle can be more consistently defined;

  — calculation of temperature differences between refrigerant and secondary fluids basing on instantaneous thermal capacities and fluid flows at evaporator and condenser;

  — use of a larger number of nominal points (anyway limited in a small total number of 2 ÷ 4), especially when defrost has to be considered.

Method A has the following disadvantages:

— weak accuracy when both the following conditions take place:

  — just one nominal point is used;

  — large temperature differences between actual operating conditions and nominal conditions are involved, especially in the case of defrost operation.

Annex B
(informative)

Calculation of the COP based on interpolation of values

B.1 General

Annex B presents two approaches for calculation of the COP.

The first one is based on the interpolation from a reference value and using a set of coefficients based on the different technologies of heat pumps.

The second one presents a method that authorizes the calculation of the COP of on/off heat pumps at full load and partial loads, based on the definition of the second degree formula.

B.2 Construction of the reference matrix at full load capacity

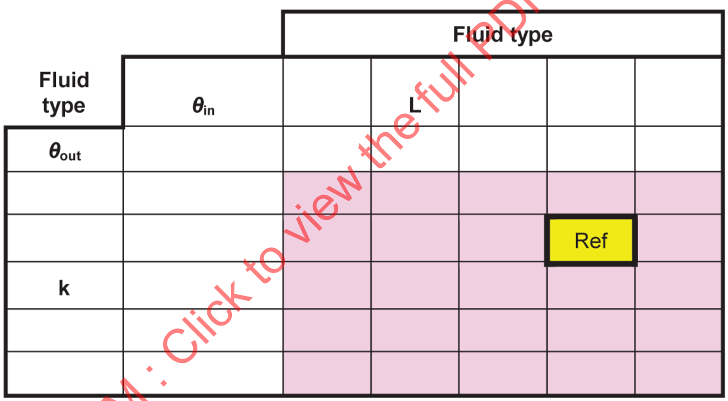


Figure B.1 — Basic matrix for determination of main values of COP at full load capacity

The matrix is built from a reference value (Example COP for θ_{in} = 7 °C and θ_{out} = 35 °C) and using a coefficient representing the evolution of the COP under external conditions and/or output temperature of the heat pump

$$COP\_k,l = C_{int}(k,ref) \cdot C_{out}(l,ref) \cdot COP_{ref} \qquad (B.1)$$

where

COP_k,l	is the coefficient of performance corresponding to conditions k (inlet temperature) and l (outlet temperature);
$C_{int}(k,ref)$	is the correction factor for values of temperature k and reference at the input of the heat pump;
$C_{out}(l,ref)$	is the correction factor for values of temperature l and reference at the output of the heat pump;
COP_{ref}	is the value of COP at reference conditions.

EXAMPLE Heat pump type: brine/water; reference value: COP at 7 °C/35 °C.

Table B.1 — Multiplication factors for brine/water heat pumps

Inlet temperature	Outlet temperature
C_{int} (2,7) = 0,9	C_{out} (25,35) = 1,1
C_{int} (12,7) = 1,1	C_{out} (45,35) = 0,8
C_{int} (17,7) = 1,2	C_{out} (55,35) = 0,64
	C_{out}(65, 35) = 0,51

B.3 Heat pump model

The value of the COP is established for a fixed value of the calculation period (typically 1 h).

B.3.1 Full load capacity with non-nominal conditions

B.3.1.1 Cooling mode

For this model, the heat pump system in cooling mode is characterized with the temperature of the sources at evaporator and condenser. It calculates the energy delivered to the system and the energy delivered to the cooling system. This model is based on two non-dimensional formulae in which polynomial characteristics are issued from the theoretical value of CO and specific product data.

The power of the compressor unit, Q_a , is obtained from Formula (B.2) as a function of the inlet (air) temperature of the condenser, and of the ration of the cooling power demand and energy used by the compressor at nominal conditions. The cooling power for non-nominal conditions is calculated based on the same principle.

$$\left(\frac{Q_{afl}}{Q_{ffl}}\right)=\left(\frac{Q_a}{Q_f}\right)_{nom}.\left(1+C_1.\Delta T+C_2.\Delta T^2\right) \quad (B.2)$$

where

Q_{afl} is the power delivered to the compressor at full load at non-nominal capacity;

Q_{ffl} is the cooling power demand at full load at non-nominal conditions;

$Q_{a,nom}$ is the power delivered to the compressor at full load at nominal capacity;

Q_{fnom} is the cooling power demand at full load at non-nominal conditions.

$$\Delta T=\left(\frac{T_{ext}}{T_{air,ent,h}}\right)-\left(\frac{T_{ext}}{T_{air,ent,h}}\right)_{nom} \quad (B.3)$$

where

T_{ext} is the external temperature at the inlet of the condenser;

$T_{air,ent,h}$ is the (humid) temperature at the inlet of the evaporator.

$$Q_{ffl}=Q_{fnom}\cdot\left[1+D_1.\left(T_{ext}-T_{ext,nom}\right)+D_2.\left(T_{air,ent,h}-T_{air,ent,h,nom}\right)\right] \quad (B.4)$$

C_i,D_i are the weighting factors (i = 1,2) used for the calculation of the power used at full capacity for non-nominal conditions. These weighting factors are obtained from product data (three values or more). The calculation procedure is presented in B.3.

B.3.1.2 Heating mode

The calculation of the power capacity for non-nominal conditions is expressed in Formulae (B.5), (B.6), and (B.7):

$$\left(\frac{Q_{\text{afl}}}{Q_{\text{cfl}}}\right)=\left(\frac{Q_{\text{a}}}{Q_{\text{c}}}\right)_{\text{n}}.\left(1+C_1.\Delta T+C_2.\Delta T^2\right) \tag{B.5}$$

$$\Delta T=\left(\frac{T_{\text{ext}}}{T_{\text{air,ent}}}\right)-\left(\frac{T_{\text{ext}}}{T_{\text{air,ent}}}\right)_{\text{n}} \tag{B.6}$$

$$Q_{\text{cfl}}=Q_{\text{cn}}\cdot\left[1+D_1.\left(T_{\text{ext}}-T_{\text{extn}}\right)+D_2.\left(T_{\text{air,ent}}-T_{\text{air,ent,n}}\right)\right] \tag{B.7}$$

where

Q_{c} is the power demand at the condenser;

Q_{afl} is the power used by the compressor at full capacity and non nominal conditions;

Q_{cfl} is the power demand at full load for non-nominal conditions;

$Q_{\text{a,nom}}$ is the power delivered to the compressor;

Q_{cnom} is the thermal power at nominal conditions;

$T_{\text{air,ent}}$ is the internal temperature at the inlet of the condenser.

C_{i},D_{i} are the weighting factors (i = 1,2) used for determination of the performance of the heat pump in heating mode at full load for non-nominal conditions. The definition of theses weighting factors is presenter in B.3.

If the condenser is located outside, Q_{c} is corrected with the coefficient C_{d} linked to the defrosting mode. The thermal capacity is corrected according to the following rules:

$$Q_{\text{c,cor}}=Q_{\text{c}} \text{ if } T_{\text{ext}}>2\,^{\circ}\text{C}$$

$$Q_{\text{c,cor}}=C_{\text{d}}.Q_{\text{c}} \text{ if } T_{ext}<2\,^{\circ}\text{C}$$

B.3.2 Model at part load for ON/OFF heat pumps

The model used for characterization of the heat pump is based on Formula (B.8).

$$\frac{Q_{\text{a}}}{Q_{\text{afl}}}=C_{\text{cp}}\cdot\left(\frac{Q_{\text{f}}}{Q_{\text{ffl}}}-1\right)+1 \tag{B.8}$$

with $\frac{Q_{\text{f}}}{Q_{\text{ffl}}}=\tau$ part load factor

C_{cp} is a parameter defined according to the type of compressor unit, the nature of fluid, and the temperatures at the condenser and at the evaporator.

The values of the COP in real conditions (part load and non-nominal conditions) are calculated as follows:

$$EER_{\text{réel}}=\frac{Q_{\text{f}}}{Q_{\text{a}}} \tag{B.9}$$

$$EER_{\text{nn}} = \frac{Q_{\text{ffl}}}{Q_{\text{afl}}} \tag{B.10}$$

The reduction factor at part load is calculated according to Formula (B.11):

$$K_{\text{cp}} = \left(\frac{EER_{\text{réel}}}{EER_{\text{nn}}}\right) = \left(\frac{\dfrac{Q_{\text{f}}}{Q_{\text{a}}}}{\dfrac{Q_{\text{ffl}}}{Q_{\text{afl}}}}\right) = \left(\tau.\frac{Q_{\text{afl}}}{Q_{\text{a}}}\right) = \left(\frac{\tau}{C_{\text{cp}} \cdot (\tau \; - 1) + 1}\right) \tag{B.11}$$

This expression shall be considered for any type of cooling unit. For HP, the expression of C_{cp} shall be replaced with the coefficient (1 – α), as α represents the ratio of the standby power with power delivered to the compressor at full load.

B.3.3 Calculation of the weighting factors used for assessment of the performance of ON/OFF heat pumps

B.3.3.1 Cooling mode

The calculation of the weighting factors is based on at least three sets of data.

Calculation is available for air or water.

For cooling mode, the temperatures at the evaporator and at the condenser are determined with

— the external air temperature or output water temperature, t_{sc}, and

— the output temperature of iced water or output air temperature (direct evaporative air cooler battery), t_{se}.

B.3.3.2 Calculation procedure for the weighting factor

The definition of t_{ec} is based on available product information:

— for air condenser, t_{ec} corresponds to ambient air (inlet air to the condenser);

— for water condenser, t_{ec} corresponds to the outlet temperature of the condenser.

The accuracy of the results of the model hardly depends on the choice of data used for characterization of these parameters. As a basis, it is preferable to choose three points with the following characteristics:

— point 1 shall correspond to nominal operation of the HP;

— point 2 shall correspond to the operating mode with the lowest value for Δt;

— point 3 shall correspond to the operating mode with the highest value for Δt.

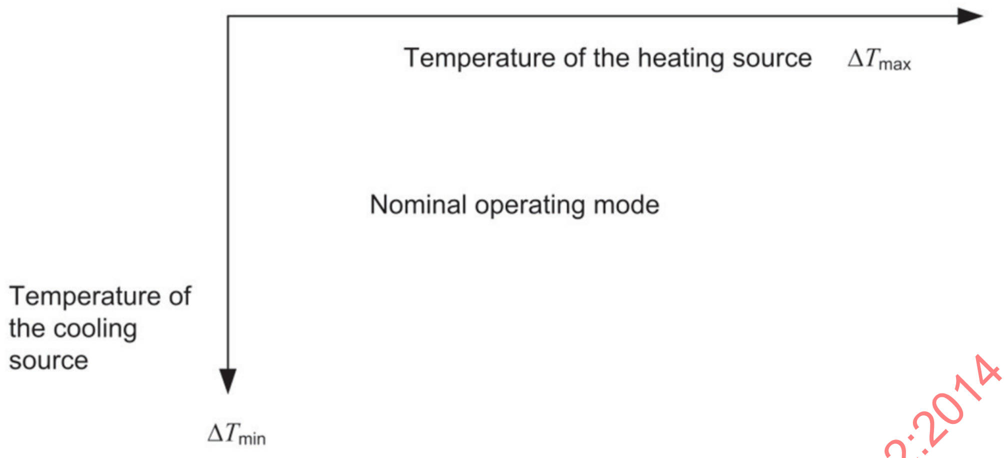


Figure B.2 — Position of the point used as a function of the temperatures of cooling source and heating sources

This procedure is preferred when the product data are available. Extrapolation increases the value systematic uncertainty.

$$\left(\frac{Q_{afl}}{Q_{ffl}}\right)=\left(\frac{Q_a}{Q_f}\right)_{nom}\cdot\left(1+C_1\cdot\Delta t+C_2\cdot\Delta t^2\right) \tag{B.12}$$

$$\text{with: } \Delta t=\left(\frac{t_{sc}}{t_{se}}\right)-\left(\frac{t_{sc}}{t_{se}}\right)_n \quad [\mathrm{K}] \tag{B.13}$$

$$Q_{ffl}=Q_{fn}\cdot\left[1+D1\cdot\left(t_{sc}-t_{scn}\right)+D_2\cdot\left(t_{se}-t_{sen}\right)\right] \tag{B.14}$$

The following parameters are used to determine the weighting factors:

$A=\dfrac{Q_{afl}\;Q_{fn}}{Q_{ffl}\;Q_{an}}$ $\quad$ $B=\Delta t$ $\quad$ $E=\Delta t^2$ $\quad$ $F=\dfrac{Q_{ffl}}{Q_{fn}}$

$G=t_{sc}-t_{scn}$ $\quad$ $H=t_{se}-t_{sen}$

Data set point 1 and 2 are linked with the following formulae:

$$A = 1 + C_1B + C_2E \tag{B.15}$$

$$F = 1 + D_1G + D_2H \tag{B.16}$$

$C_2=\dfrac{B_2A_1-B_1A_2+B_1-B_2}{B_2E_1-B_1E_2}$ $\quad$ $C_1=\dfrac{A_1-1-C_2E_1}{B_1}$

$D_2=\dfrac{G_2F_1-G_1F_2+G_1-G_2}{G_2H_1-G_1H_2}$ $\quad$ $D_1=\dfrac{F_1-1-D_2H_1}{G_1}$

Index i refers to set point data i.

Default value of the calculation model

Product data sometimes only consider the technical characteristics of the HP at nominal conditions. In this case, the necessary completion of the data set points.

In this case, the ratio of power based on the application of the formula of Carnot is used to establish default values:

$$C_1 = \left(\frac{Q_f}{Q_a}\right)_n \tag{B.17}$$

$$C_2 = 0 \tag{B.18}$$

B.3.3.3 Heating mode

Heating mode performances are calculated with the same procedure as the cooling mode.

Annex C
(informative)

Calculation of heat pump COP and thermal capacities at part load conditions

C.1 General

To estimate the value of COP at part load conditions, Method A and Method B walk the same path: The full load COP is multiplied by a proper part load factor, f_X, calculated basing on the actual part load ratio X, that expresses the degree of partialization, i.e. the ratio between the thermal capacity needed at the user side and the maximum capacity that the heat pump can deliver at the same temperatures of secondary fluids. With regard to vapour compression heat pumps, the method described in the present section accepts rated performances at any part load condition, even if it is suggested to use points required by EN 14825, for consistency among standards. If such data are not present, it uses formulae recommended by EN 14825.

C.2 Introduction and input data

In the following steps, the procedure to calculate the COP and average capacities at part load conditions is explained. In a few words, it is assumed that, at any couple of internal and external secondary fluid temperatures, the COP and capacities at part load conditions are tightly related to the ones at full load. That relation is easy and is expressed *via* a multiplier whose value depends on the degree of partialization. At any couple of internal and external secondary fluid temperatures, the degree of partialization is defined *via* the part load ratio, according to Formula C.1:

$$X = \frac{P_{\mathrm{Int},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=X}}}{P_{\mathrm{Int},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}}} [-] \tag{C.1}$$

where

$P_{\mathrm{Int},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=X}}$ is the average internal capacity needed at the user side, with secondary fluid temperatures $\theta_{\mathrm{Sf:Int}}$ and $\theta_{\mathrm{Sf:Ext}}$;

$P_{\mathrm{Int},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}}$ is the maximum internal capacity that can be expressed by the heat pump at secondary fluid temperatures $\theta_{\mathrm{Sf:Int}}$ and $\theta_{\mathrm{Sf:Ext}}$.

The aforementioned relation between the COP and capacities at part load and the ones at full load capacity consists in the following formulae:

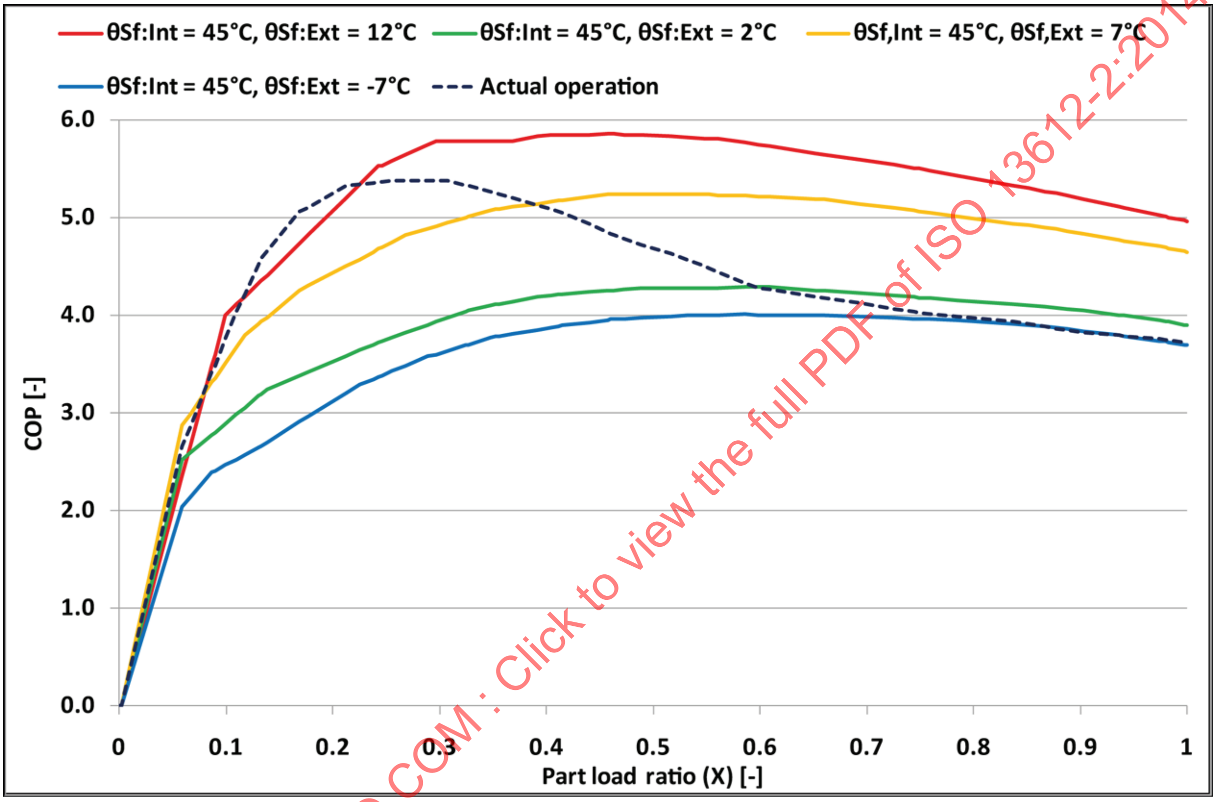
$$COP_{\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=X}} = COP_{\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=1}} \cdot f_X \tag{C.2}$$

$$P_{\mathrm{El},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=X}} = \frac{P_{\mathrm{Int},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=X}}}{COP_{\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=X}}} \tag{C.3}$$

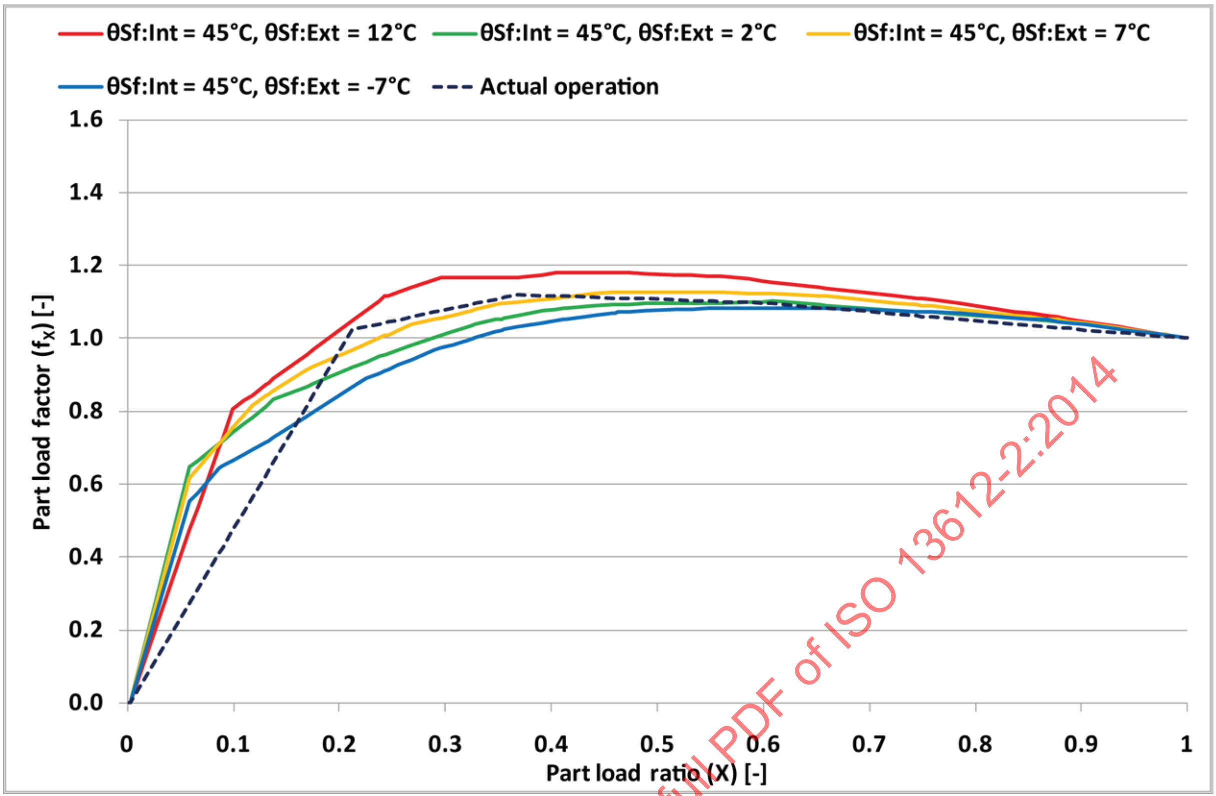
$$P_{\mathrm{Ext},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=X}} = P_{\mathrm{Int},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=X}} - P_{\mathrm{El},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=X}} \text{ in heating mode} \tag{C.4}$$

$$P_{\mathrm{Ext},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=X}} = P_{\mathrm{Int},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=X}} + P_{\mathrm{El},\theta_{\mathrm{Sf:Int}},\theta_{\mathrm{Sf:Ext}},\mathrm{f=X}} \text{ in cooling mode} \tag{C.5}$$

The concept that is the basis of this description of performances at part load conditions can be clearly explained by means of Figure C.1, where it can be noted how the relevant differences in COP trend can be efficaciously summarized by means of the part load factor (f_X), that is used to normalize part load COP curves at different internal and external secondary fluid temperatures. The part load factor curve is even more realistic when, for each part load ratio, the most realistic internal and external secondary fluid temperatures are considered so that an even better correspondence between the part load factor and the most probable couple of secondary fluid temperatures can be achieved.



a) COP versus part load capacity (expressed in terms of part load ratio X)



b) Part load factor versus part load capacity (expressed in terms of part load ratio X)

Figure C.1 — COP and part load factor versus part load capacity (expressed in terms of part load ratio X)

In brief, the profile of f_X versus X is needed. For that purpose, EN 14825 recommends that COP be rated at imposed part load values $X_{\text{Nom},A} = A_{\text{Nom}}$, $X_{\text{Nom},B} = B_{\text{Nom}}$, $X_{\text{Nom},C} = C_{\text{Nom}}$, and $X_{\text{Nom},D} = D_{\text{Nom}}$, with specified temperature couples ($\theta_{\text{Sf:Int}}, \theta_{\text{Sf:Ext}}$), where X_{Nom} means that the part load ratio is referred to the nominal capacity.

C.3 Detailed calculation procedure

C.3.1 Characterization of rated points

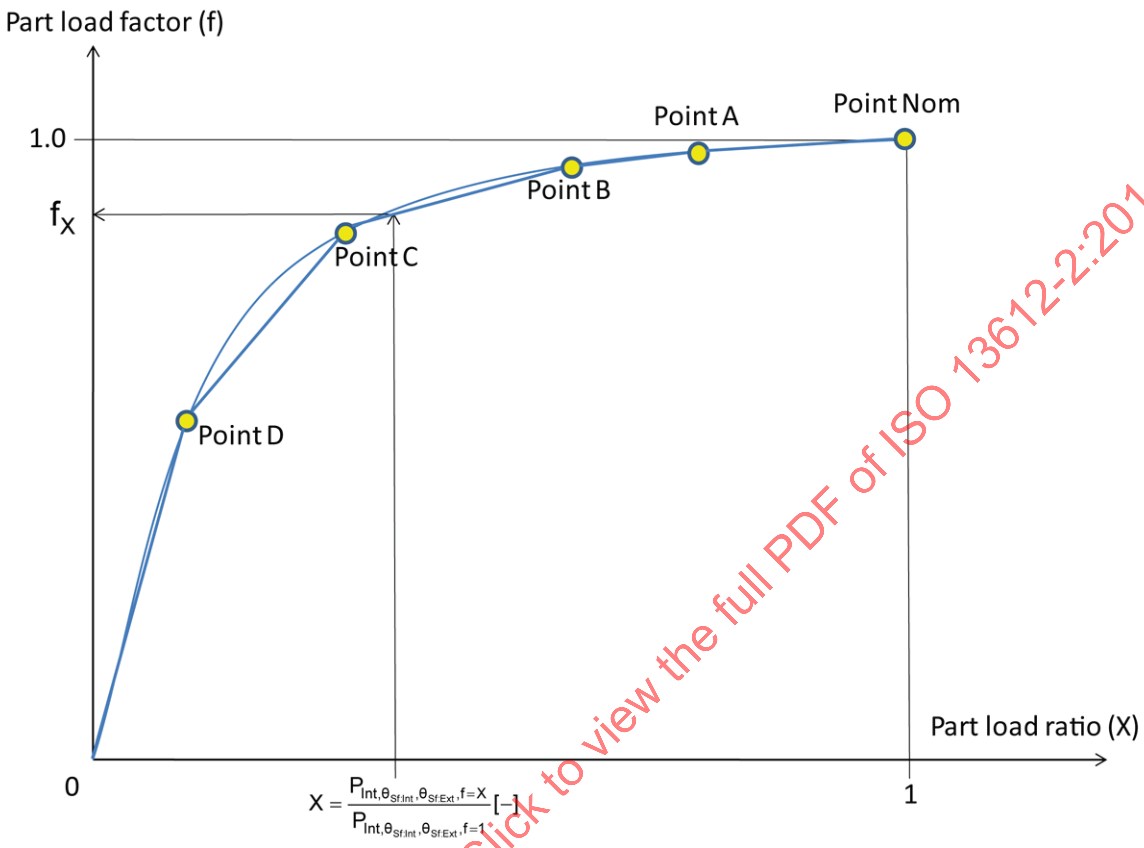
Particular attention has to be paid, since part load values A_{Nom}, B_{Nom}, C_{Nom}, and D_{Nom} are referred to nominal internal capacity $\left(P_{\text{Int},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}}\right)_{\text{Nom}}$ and not to $\left(P_{\text{Int},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}}\right)_{A \dots D}$. Hence, $X_A = A$, $X_B = B$, $X_C = C$, and $X_D = D$ values are derived, referred to full capacity at same $\theta_{\text{Sf:Int}}$ and $\theta_{\text{Sf:Ext}}$. In short, A, B, C, and D values are obtained, referred to the corresponding $\left(P_{\text{Int},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}}\right)_{A \dots D}$, and f_A, f_B, f_C, f_D, by the following correlations:

$$A = A_{\text{Nom}} \cdot \frac{\left(P_{\text{Int},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}}\right)_{\text{Nom}}}{\left(P_{\text{Int},\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}}\right)_A} \tag{C.6}$$

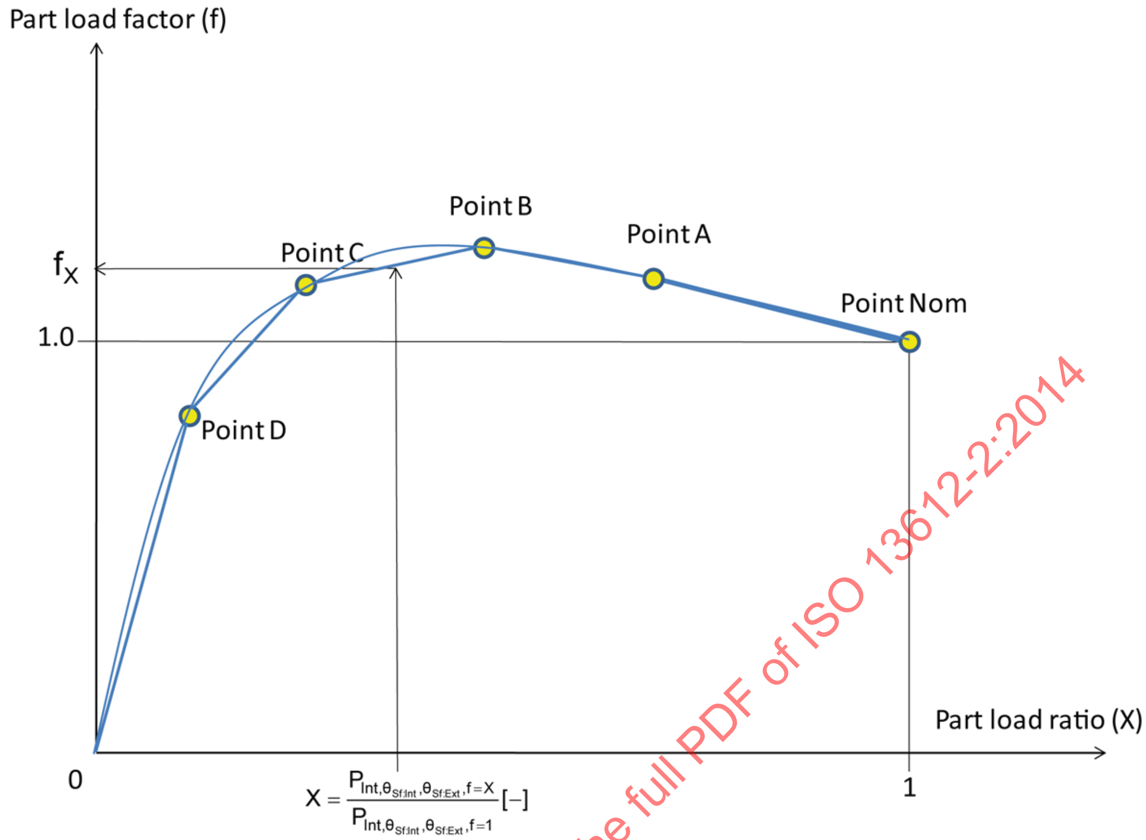
$$f_A = \frac{COP_{\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},A}}{\left(COP_{\theta_{\text{Sf:Int}},\theta_{\text{Sf:Ext}},\text{f=1}}\right)_A} \tag{C.7}$$

$$\begin{cases} B = ... \\ f_B = ... \end{cases}, \begin{cases} C = ... \\ f_C = ... \end{cases}, \begin{cases} D = ... \\ f_D = ... \end{cases}$$

The curve of pairs (A, f_A), (B, f_B), (C, f_C), and (D, f_D) is built. It will be used to calculate, *via* interpolation, the part load factor basing on the part load ratio, as explained in Figure C.2:



a) Typical correction factor curve for COP under part load conditions, for on-off compressors



b) Typical correction factor curve for COP under part load conditions, for inverter-driven compressors

Figure C.2 — Typical part load factor curves for COP under part load conditions, for on-off compressors and inverter-driven compressors

C.3.2 Calculation of COP and capacities of heat pumps at part load capacity

To sum up, when the curve in Figure C.2 is known, COP and capacities for any group of data ($\theta_{Sf:Int}$, $\theta_{Sf:Ext}$, $P_{Int,\theta_{Sf:Int},\theta_{Sf:Ext},f=X}$) can be calculated, after that the values of $P_{Int,\theta_{Sf:Int},\theta_{Sf:Ext},f=1}$ (maximum internal capacity that can be expressed by the heat pump at secondary fluid temperatures $\theta_{Sf:Int}$ and $\theta_{Sf:Ext}$) and $COP_{\theta_{Sf:Int},\theta_{Sf:Ext},f=1}$ have been calculated.

C.3.2.1 Vapour compression heat pumps

If part load points such as *A*, *B*, *C*, and *D* are not provided, then the following procedure has to be followed:

— If the heat pump has ON-OFF control, then:

— If the internal secondary fluid is water, from EN 14825:

$$f_X = \frac{X}{0{,}9 \cdot X + 0{,}1} \tag{C.8}$$

— If the internal secondary fluid is air, adapted from EN 14825:

$$f_X = \frac{X}{0{,}9 \cdot X + 0{,}1} \cdot \left[1 - 0{,}25 \cdot (1 - X)\right] \tag{C.9}$$

— If the heat pump has INVERTER control, then:

— If $X \geq 0{,}25$:

$$f_X = 1 \qquad \text{(C.10)}$$

— If $X < 0{,}25$:

If the internal secondary fluid is water, then, from prEN ISO 14825, Formula (1):

$$f_X = \frac{X \cdot \left(\frac{100}{25}\right)}{0{,}9 \cdot \left[X \cdot \left(\frac{100}{25}\right)\right] + 0{,}1}$$

If the internal secondary fluid is air, then, from Formula (2) in prEN ISO 14825 (in prEN ISO 14825, the formula is not complete, indeed):

$$f_{X\%} = \frac{X \cdot \left(\frac{100}{25}\right)}{0{,}9 \cdot \left[X \cdot \left(\frac{100}{25}\right)\right] + 0{,}1} \cdot \left\{1 - 0{,}25 \cdot \left[1 - X \cdot \left(\frac{100}{25}\right)\right]\right\}$$

NOTE Consider that the formula is a necessary simplification, when no data at part load conditions are given. This way, inverter heat pumps maintain anyway better COP at part load conditions compared with ON-OFF heat pumps. Under part load factor equal to 0,25 it is supposed that the heat pump works in ON-OFF mode, so the same part load factors as for ON-OFF control are used, but the considered modulation range in ON-OFF mode, for inverters, is assumed between 0,0 and 0,25, so the multiplier 100/25 is present.

C.3.2.2 Absorption heat pumps

The value of f_X comes from Tables C.1 and C.2, depending on the kind of heat pump control (on-off or modulating).

Table C.1 — f_X for on-off absorption heat pumps

X	0,10	0,20	0,30	0,40	0,50	0,60	0,70	0,80	0,90	1,00
f_X	0,68	0,77	0,84	0,89	0,92	0,95	0,97	0,99	1,00	1,00

Table C.2 — f_X for modulating absorption heat pumps

X	0,10	0,20	0,30	0,40	0,50	0,60	0,70	0,80	0,90	1,00
f_X	0,72	0,81	0,88	0,93	0,97	0,99	1,00	1,00	1,00	1,00

C.4 Example of input and output

In the spreadsheets delivered together with this part of ISO 13612, examples of application of the present method are shown for two heat pumps, both used in heating and in cooling modes.

In Figure C.3, examples of input and results are shown.

Additional rated points for part load calculations				
Cod [Alfa]	A	B	C	D
Part load ratio [%, ref, to Nom]	100,0	80,0	44,.0	33,0
TIntFluid_Outlet [%]	7,0	7,0	7,0	7,0
TExtFluid_Inlet [°C]	35,0	35,0	35,0	35,0
COP [-]	3,0	3,2	3,5	3,6
Part load ratio [%, ref. to actual]	100,0	80,34	44,19	33,14
COPFactor [-]	1,17	1,27	1,33	1,41
Use standard parzialization curve	YES			
Control	ON-OFF			

User Inquiries (i.e.: example of results)								
$\theta_{s,j}$ [°C]	$\theta_{s,e}$ [°C]	$P_{int,\theta si,\theta s,eX\%}$ [kW]	Part Load Ratio X [%]	Correction factor for COP [-]	COP100% [-]	COP [-]	$P_{El\theta si,\theta s,e,X\%}$ [kW]	$P_{Ext\theta si,\theta s,e,X\%}$ [kW]
6,0	30,0	5,0	104,17	1,01	2,89	2,93	1,7	3,3
8,0	36,0	4,0	83,86	0,94	2,53	2,38	1,7	2,3
11,0	40,0	2,0	40,16	0,74	2,35	1,74	1,1	0,9
6,0	32,0	2,0	42,59	0,75	2,73	2,06	1,0	1,0
12,0	36,0	3,0	55,70	0,82	2,71	2,23	1,3	1,7
15,0	38,0	2,0	34,79	0,70	2,68	1,89	1,1	0,9
6,0	25,0	1,0	19,84	0,57	3,29	1,88	0,5	0,5

Figure C.3 — Example of input and output for COP and capacities at part load conditions — cooling mode

In Figure C.3, the inputs needed for part load conditions are shown (yellow area), together with possible output (green area). At the end of these calculations, the heat pump behaviour can be predicted for any operating condition.

C.5 Conclusions

The present method is easy for use and calculations. Its accuracy depends on the accuracy of the method previously adopted for the estimation of full load COP and capacity.

Moreover, the method can be used starting from any group of points at part load conditions, even if it is recommended to use data rated at points suggested in EN 14825.

Annex D
(informative)

Model of heat pump in the BEST program

D.1 Concept of modelling of the equipment characteristics in the BEST program

The theme of the modelling of equipment characteristics is the compatibility between the characteristics unique to equipment, with dynamic characteristics taken into consideration, and the maintainability of such characteristics data. As for the maintenance after completion of programs, it is important how efficiently new types of high-efficiency equipment can be built in and the types of equipment of similar characteristics can be added without any mistake. For this purpose, equipment characteristics were basically static characteristics and the modelling was made easier.

As for the models of equipment characteristics, physical models are applied to that equipment that can be physically expressed, and regression formula models are applied to that equipment whose inherent characteristics are complicated. As for the regression formula models, equipment characteristics are classified into a couple of representative types and polynomial approximation is applied to them. The amount of inputs and outputs at the rated capacity of each piece of equipment and the power of auxiliaries and others are described separately in data s of equipment specifications, and the maintainability is improved by separating the models of characteristics and the tables of equipment specifications. This Annex describes the structure of equipment data, methodologies of modelling of representative equipment, and the results of calculations.

D.2 Structure of equipment data

As for general purpose equipment, applicable types are selected on the selection screen, and the rated values of respective equipment are inputted from the catalogues of manufacturers. In a stage, including designing stage, where specific types are not decided, it is decided to input the rated values of those equipment listed in the design standard of the Ministry of Land, Infrastructure, Transport, and Tourism.

As an example of the table of equipment specifications, one of the centrifugal chillers is shown in Table D.1. These values are set as fixed values at the time of initialization of element modules. As for other equipment, the model numbers of outdoor units, fuel consumptions (consumptions by fuel type) at rated operation, and characteristics for heating mode (rated output, rated input, rated temperature, and flow rate) are added in accordance with the types of equipment.

Table D.1 — Table of equipment specifications — Example of centrifugal chiller[a]

Centrifugal chiller	
1) Classification (1)	Centrifugal chiller
2) Classification (2)	For cooling only
3) Classification (3)	—
4) Name of manufacturer	Company A
5) Name of series	High-efficiency type (inverter)
6) Model number	***-800**
7) Refrigerant	HFC-134a
8) Characteristics at cooling	
Rated output (kW)	
Power consumption at rated operation	2,813
Main unit (kW)	505
Auxiliary unit (kW)	2
Rated temperature and flow rate	
Cold water	
Inlet temperature (°C)	12
Outlet temperature (°C)	7
Flow rate (m^3/h)	121
Pressure loss (Pa)	60
Cooling water	
Inlet temperature (°C)	32
Outlet temperature (°C)	37
Flow rate (m^3/h)	
Pressure loss (Pa)	148
9) Characteristic formula pattern index	49
10) Dynamic characteristics	C
Anti-repeat time (min)	15
Operation time of auxiliaries before start-up (min)	2
	4,5
Operation time of auxiliaries after shut-down (min)	—
11) Remarks	

[a] Some values are different from reality.

D.3 Equipment characteristics models

D.3.1 Heating and cooling equipment

D.3.1.1 Types of applicable heating and cooling equipment

The heating and cooling equipment to be developed are shown in Table D.2. Several types of characteristic formulae shown in Table D.2 are implemented for respective equipment. All of these characteristic formulae are regression formula models. They include those provided by manufacturers and those developed by WG on equipment characteristics based on data that have already been published. The characteristic formulae for the latter are developed by the formulation method of the commissioning (Cx) tool subcommittee of the Cx committee of this academic society, which was closed in fiscal 2007. All of these characteristic formulae allow for partial load characteristics, temperature characteristics of cooling medium, and others.

In this subclause, these models are described with a centrifugal chiller as a major example.

Table D.2 — Frame structure and status of formulation of heating and cooling equipment characteristics

Type			Outline
Centrifugal chiller	Standard unit	Vane control	Development from JRA data
	High-efficiency unit	Vane control	Data provided by manufacturers
		Inverter control	Data provided by manufacturers
		Ice storage	(Under review)
Air-source heat pump	Screw	Slide valve control	Data provided by manufacturers
		Inverter control	Data provided by manufacturers
	Scroll	Control of the number of compressors	Data provided by manufacturers
		Inverter control	(Under review)
	Ice storage		(Under review)
Water-cooled chiller	Screw	Slide valve control	Data provided by manufacturers
		Inverter control	Data provided by manufacturers
	Scroll	Control of the number of compressors	Data provided by manufacturers
	Ice storage		(Under review)
Absorption chiller	Direct-fired	Triple effect	Data provided by manufacturers
		Double effect	
		High year-round efficiency unit	(Under review)
	Steam-fired	Double effect	Data provided by manufacturers
	Hot water-fired	Single effect	Data provided by manufacturers
	Waste heat-fired	Triple effect	Already formulated by WG on co-generation
		Double effect	

D.3.1.2 Regression formula models of centrifugal chillers

The inputs and outputs of centrifugal chillers are shown in Figure D.1. For characteristic models, the data of Japan Refrigeration and Air Conditioning Industry Association and other data provided by manufactures are used, and the characteristic formulae of the following three types--standard unit (vane control), high-efficiency unit (vane control) and high-efficiency unit (inverter control)--are mounted. The performance of all types of equipment registered in the table of equipment specifications is calculated by either of these formulae.

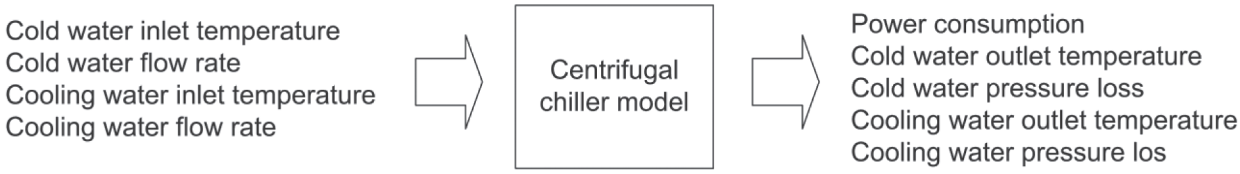


Figure D.1 — Input and output of centrifugal chiller model

Of these, the characteristic formula of the standard unit (vane control) is formulated by SWG on equipment characteristics based on the performance characteristic diagram of the association. Figure D.1 shows that the cooling capacity increases by up to 4 % when the cooling water temperature is low and the cold water temperature is high. With emphasis placed on this point for calculation, the maximum cooling capacity, R_{QMax} (%), is calculated first by Formula (D.1).

$$R_{\mathrm{QMax}} = 100 + 2(TW_{\mathrm{in}} - TW_{\mathrm{in\_S}}) - 0{,}5(TW_{\mathrm{Cin}} - TW_{\mathrm{Cin\_S}}) \tag{D.1}$$

(Provided, however, if $R_{\mathrm{QMax}} > 104$, $R_{\mathrm{QMax}} = 104$)

where

TW_{in} is the cold water inlet temperature (°C);

$TW_{\mathrm{in\_S}}$ is the rated temperature of cold water inlet (°C);

TW_{Cin} is the cooling water inlet temperature (°C);

$TW_{\mathrm{Cin\_S}}$ is the rated temperature of cooling water inlet (°C).

Then, power consumption W_{CR} (W) is calculated by Formula (D.2).

$$W_{\mathrm{CR}} = W_{\mathrm{CR\_S}} \cdot C_{\mathrm{Q\text{-}WC}} \cdot C_{\mathrm{WE}} + W_{\mathrm{CR\_A}}$$

$$C_{\mathrm{Q\text{-}WC}} = A_2 \cdot R_{\mathrm{QE}}^2 + A_1 \cdot R_{\mathrm{QE}} + A_0$$

$$\begin{pmatrix} A_0 \\ A_1 \\ A_2 \end{pmatrix} = \begin{pmatrix} A_{00} & A_{01} & A_{02} \\ A_{10} & A_{11} & A_{12} \\ A_{20} & A_{21} & A_{22} \end{pmatrix} \begin{pmatrix} 1 \\ TW_{\mathrm{Cin}} \\ TW_{\mathrm{Cin}}^{\ 2} \end{pmatrix} \tag{D.2}$$

where

$W_{\mathrm{CR\_S}}$ is the rated input of motor (W);

$C_{\mathrm{Q\text{-}WC}}$ is the transformation factor of load factor and cooling water temperature;

C_{WE} is the transformation factor of cooling water temperature;

$W_{\mathrm{CR\_A}}$ is the power of auxiliary unit (W);

R_{QE} is the load factor of chiller;

$A_{00} \sim A_{22}$ are the constants.

For reference, R_{WE} is calculated inside the model with the cold water inlet temperature and cold water flow rate of the input items.

The upper limit of cooling capacity is calculated by Formula (D.1), and the results of input of motor as calculated by Formula (D.2) are shown in Figure D.2.

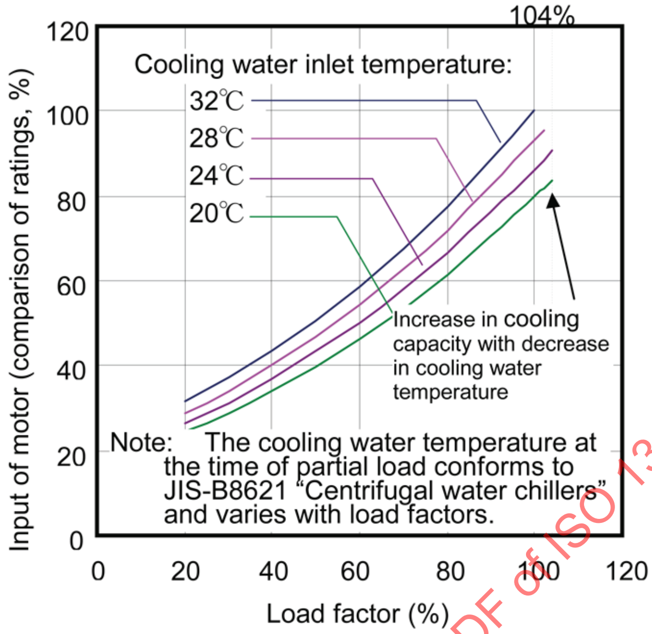
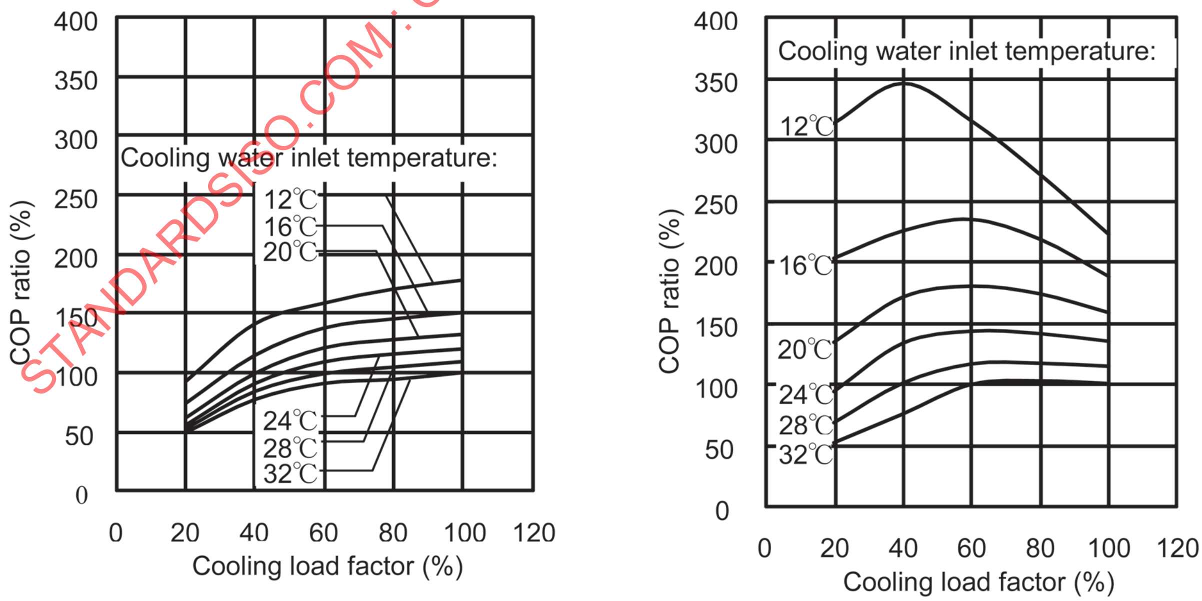


Figure D.2 — Centrifugal chiller characteristics A (fixed speed — standard) — Partial load characteristics by each cooling water temperature

In addition to the standard unit indicated in Figure D.2, the characteristics of high-efficiency units are shown in Figure D.3. Here, COP ratios are expressed with the rated point (load factor is 100 % and cooling water temperature is 32 °C) as the standard. In the case of centrifugal chillers, as the difference in characteristics due to types of refrigerants is small, HFCs and HCFCs are expressed by their common characteristics.



a) Fixed speed (vane control) unit

b) Inverter control unit

Figure D.3 — Characteristics of high-efficiency centrifugal chiller (example)

As for centrifugal chillers, the formulation of the standard unit, the high-efficiency unit, and the inverter-driven high-efficiency unit have been completed as shown in Table D.2.

D.3.1.3 Models of absorption water heater-chiller units

The inputs and outputs of absorption water heater-chiller units are shown in Figure D.4. As shown in Table D.2, they are classified into direct-fired and steam-fired to reflect the characteristics of respective types. As for direct-fired, the triple effect type of high COP is added. Moreover, it is made possible to cope with the characteristic formula of hot water-fired, which can be made use of for utilization of exhaust heat and others. The characteristic formula of waste heat-fired type is formulated on co-generation.

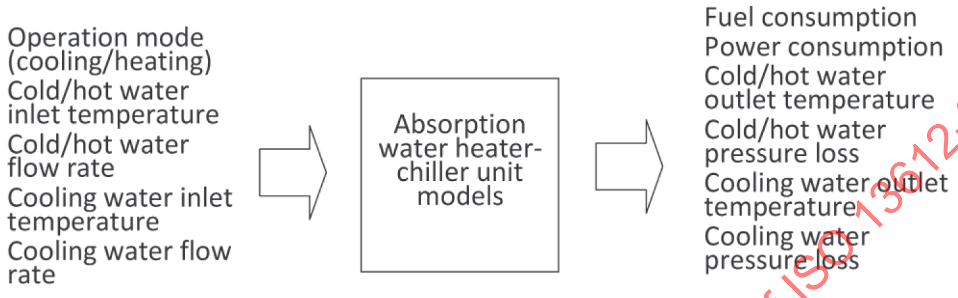


Figure D.4 — Inputs and outputs of absorption water heater-chiller unit models

As an example of equipment characteristics, the characteristics of the double effect steam-fired absorption chiller are shown in Figure D.5. The load factor of 100 % and cooling water temperature of 32 °C are considered as the standard points of steam consumption rate as is the case with Figure D.3. The partial load characteristics of the high-efficiency unit and that of the standard unit are almost the same and both can be expressed by the characteristics shown in Figure D.5. Moreover, the database is added with “high seasonal efficiency units” that are announced by Japanese manufacturers.

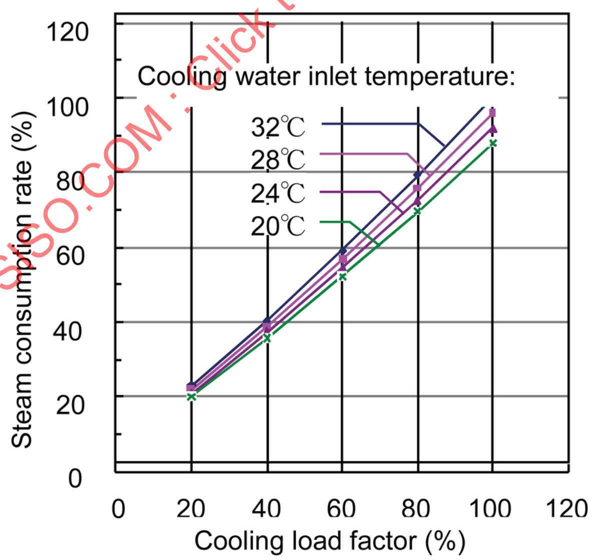


Figure D.5 — Characteristics of double effect steam-fired absorption chiller (example)

D.3.1.4 Air source heat pumps and water-cooled chillers

For air-cooled heat pumps, their types are expanded as shown in Table D.2, including the types of control of the number of compressors, which use scroll compressors. Moreover, efforts are made to cope with the types that spray water on outdoor units when the outdoor temperature rises. As for those types that have built-in cool/hot water variable flow pumps, adoption of a module structure that is composed of a heat pump and a pump is now under consideration. For air-cooled heat pumps for cooling only, the characteristics of air-cooled heat pumps at the time of cooling are used. It is also considered coping with ice storage, including water-cooled chillers.

The inputs and outputs of air-cooled heat pump models are shown in Figure D.6. The inputs and outputs of water-cooled chillers are almost the same as those of centrifugal chillers shown in Figure D.1.

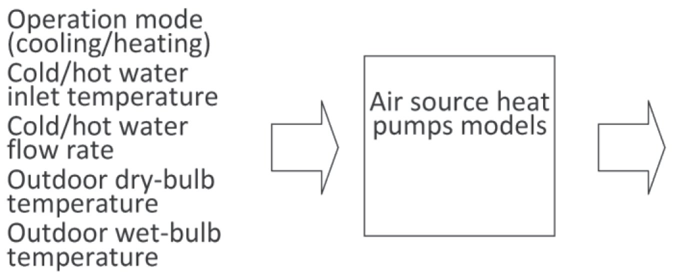


Figure D.6 — Inputs and outputs of air-cooled heat pumps models

Figure D.7 shows the partial load efficiency of air-source heat pumps (screw compressors, slide valve control type) at the time of cooling with the ratios that use the rated point (load factor is 100 % and outdoor dry-bulb temperature is 35 °C) as the standard. The characteristics of COP and the maximum capacity, which increase in proportion to the decrease in outdoor temperature, are reflected.

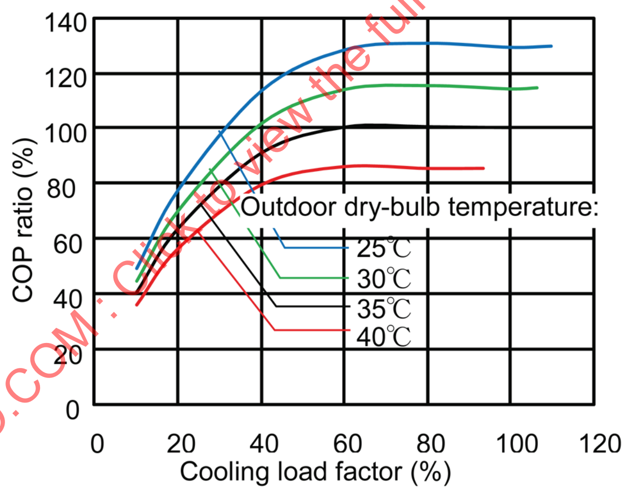


Figure D.7 — Characteristics of air source heat pumps at the time of cooling (example)

As for air-source heat pumps, the maximum cooling capacity is calculated based on outdoor temperature in accordance with Figure D.7, and actual outputs are calculated by the comparison with required cooling capacity that is decided by the conditions of cold water returned. COP ratios are provided as the functions of ratios of these outputs to the maximum cooling capacity.

D.3.2 Cooling tower

As for cooling towers, both open-type and closed-type are reviewed. As shown in Table D.3, the characteristics of a total of four types are adopted. The inputs and outputs of models are as shown in Figure D.10. They can cope with the changes in cooling water flow rates and the control of cooling water temperatures with three-way valves.

Table D.3 — Characteristic formulae adopted for cooling towers

Type of equipment	Name	Index	Method of production
Open type cooling tower	Specifications for 37,5 °C – 32,0 °C	A	Provided by the technical committee of Japan Cooling Tower Institute
	Specifications for 37,0 °C – 32,0 °C	B	
Closed type cooling tower	Specifications for 37,5 °C – 32,0 °C	C	
	Specifications for 37,0 °C – 32,0 °C	D	

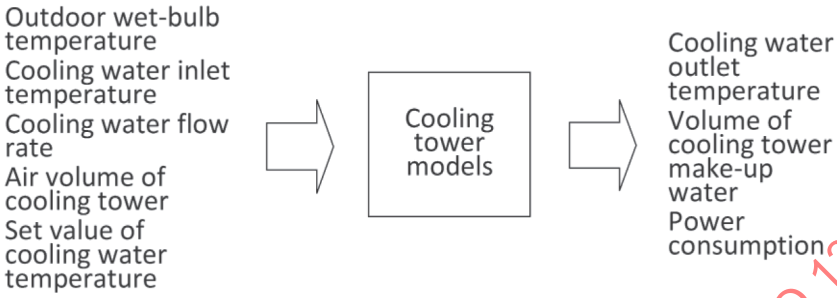


Figure D.8 — Inputs and outputs of cooling tower models

The calculation method of each characteristic is basically the same. The cooling water outlet temperature, TW_{out} (°C), is calculated by the following formulae.

Outlet temperature of main unit of cooling tower (outlet temperature without control by three-way valves):

$$TW_{out\_CT} = (a_1 \cdot TWB_{in}{}^2 + b_1 \cdot TWB_{in} + c_1) \cdot c_4 \cdot d \quad \text{(D.4)}$$

Outlet temperature with the control by three-way valves:

$$TW_{out} = TW_{out\_CT} \cdot \alpha + TW_{in\_CT} \cdot (1-\alpha) \quad \text{(D.5)}$$

And, a_1, b_1, and c_1 are expressed by Formula (D.6) and vary with each of the specifications shown in Table D.3.

$$\begin{pmatrix} a_1 \\ b_1 \\ c_1 \end{pmatrix} = \begin{pmatrix} a_{10} & a_{11} & a_{12} \\ b_{10} & b_{11} & b_{12} \\ c_{10} & c_{11} & c_{12} \end{pmatrix} \begin{pmatrix} 1 \\ TW_{in\_CT} \\ TW_{in\_CT}{}^2 \end{pmatrix} \quad \text{(D.6)}$$

where

$a_{10}, \ldots, c_{12}$ are the constants that depend on types;

c_4 is the function of cooling water flow ratio, cooling water inlet temperature, and outdoor wet-bulb temperature;

d is the function of air volume ratio of cooling tower, cooling water inlet temperature, and outdoor wet-bulb temperature;

TWB_{in} is the outdoor wet-bulb temperature;

α is the ratio of flow rate of three-way valves.

Examples of calculation results are shown in Figure D.11 and Figure D.12. Figure D.11 shows the relations between the wet-bulb temperature and outlet water temperature in the case where the cooling water inlet temperature changes, and Figure D.12 shows the relations between the wet-bulb temperature and outlet water temperature in the case where the cooling water flow rate changes. In either case, the outlet water temperature decreases in proportion to the decrease in the wet-bulb temperature. It also decreases together with the decrease of cooling water inlet temperature in Figure D.11 and the cooling water flow rate in Figure D.12. In either case, these results are the same as those of actual units, and reasonable results have been obtained.

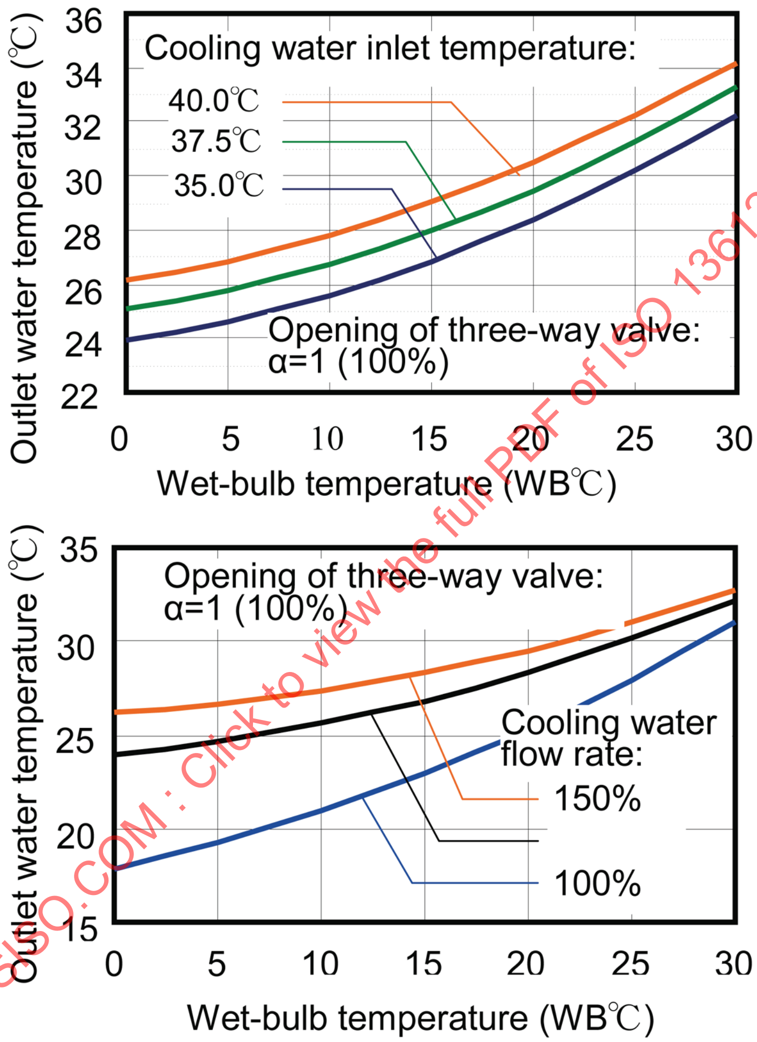


Figure D.9 — Characteristics of open-type cooling tower (specifications for 37,5 °C to 32,0 °C) at the time of change in cooling water flow rate

D.3.3 Variable refrigerant flow system

D.3.3.1 Policy to indicate characteristics

As for the equipment characteristics of multi-split-type air-conditioning systems as represented by variable refrigerant flow (VRF) systems, the standard types of EHP and GHP that are typical VRF systems have been formulated by using the models that handle indoor and outdoor units in one. The formulation of equipment characteristics is promoted by particularly taking the following three points into consideration:

a) expansion of types other than VRF systems of standard types;

b) development of common display format of characteristics data;

c) reflection of intermediate outputs and intermediate inputs for APF (annual performance factor) display.

D.3.3.2 Structure of the types of VRF systems

Table D.4 shows a list of various types of equipment subject to the formulation of equipment characteristics of packaged air-conditioners. For formulation, an overall frame is developed in consideration of (1) differences in use of buildings, (2) regional differences (specifications for cold climate areas), (3) classifications of loads assigned (fresh air treatment, etc.), (4) classifications of energy type (electricity, city gas, and kerosene), and (5) heat radiation systems (air cooling and water cooling).

D.3.3.3 Outline of characteristics models

Figure D.13 shows the outline of calculation models of VRF systems to be examined, and Table D.5 shows the types and descriptions of respective characteristic formulae in these models. The calculation models are composed of two or more indoor and outdoor units, and adopt the “system-integrated performance characteristics models” that provide the cooling and heating outputs and energy consumption that can be supplied by respective equipment under any given conditions and the amount of energy consumption as output information by inputting various pieces of information such as outdoor air conditions, conditions of air taken in indoors, rated specifications of equipment, length of refrigerant piping, difference of elevation between equipment, etc.

Table D.4 — Structure and status of frame of formulation of equipment characteristics of packaged air-conditioners

	Type		Outline
GHP	VRF systems	Switch from cooling to heating and *vice versa*	Partial review of characteristics
		Cooling and heating at the same time	(Under review)
	GHP with generator	Self-consumption type (switch from cooling to heating and *vice versa*)	Completion of development of approximation
		System interconnection type (switch from cooling to heating and *vice versa*)	(Under review)
EHP	VRF systems	Switch from cooling to heating and *vice versa*	Transformation of parts that reflect intermediate capacity
		Cooling and heating at the same time	(Under review)
		Support for use in cold climate areas	Formulation as an extension of form of characteristic formula of switch from cooling to heating and *vice versa*
		Switch from cooling to heating and *vice versa*	
		Water cooling	(Under review)
	VRF systems	Switch from cooling to heating and *vice versa*	Formulation with equipment characteristics of representative types of respective companies
		Support for use in cold climate areas	The same as above
	VRF systems	Switch from cooling to heating and *vice versa*	Formulation of equipment characteristics for air-conditioning in offices (for people)
			Refrigeration only type for electric rooms and ELV machine rooms
		Support for use in cold climate areas	For air-conditioning in office (for people)

GHP: Gas engine driven heat pump

EHP: Electric heat pump

KHP: Kerosene engine heat pump

Table D.4 *(continued)*

	Type		Outline
EHP	Outdoor air processing unit	Switch from cooling to heating and *vice versa*	Review of frame (classification of cases is now under consideration with the combined use of desiccant air-conditioners and total heat exchangers
		Cooling and heating at the same time	
		Support for use in cold climate areas	
		Water cooling	
	Ice storage VRF systems	Switch from cooling to heating and *vice versa*	Formulation by classification of cases at the time of thermal storage and non-thermal storage (indoor units are the same as those of VRF systems)
		Support for use in cold climate areas	(Under review)
KHP	VRF systems	Standard types	Formulation as is the case with GHP
		Support for use in cold climate areas	The same as above
GHP: Gas engine driven heat pump EHP: Electric heat pump KHP: Kerosene engine heat pump			

As for characteristic formulae, both motor-driven multi-split-type air-conditioners (hereinafter referred to as "EHP") and fuel-driven multi-split type air-conditioners (hereinafter referred to as "GHP") are expressed in almost the same form by the maximum cubic expressions of explanatory variables, each of which is independent, concerning indoor/outdoor units, at the time of cooling/heating, and cooling (heating) capacity/energy consumption, and formulated by giving coefficients from A* to N*. An example of the calculation results is shown in Figure D.14.

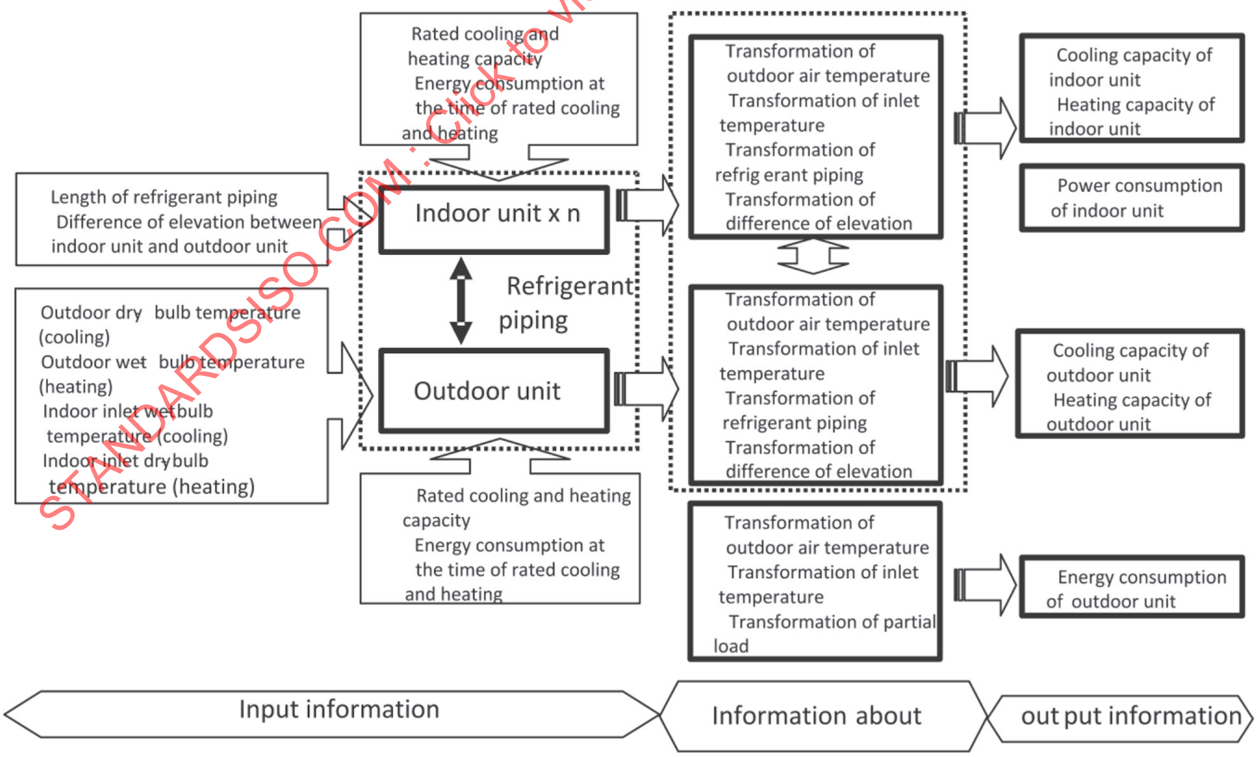


Figure D.10 — Outline of model of VRF system

Table D.5 — Calculation formulae of transformation factors to show equipment characteristics

Mode	Equipment	Objective variable	Adjustment coefficient No.	Adjustment coefficient Input parameter	Expressions of characteristics formula to calculate adjustment coefficients
Cooling	Indoor unit	Cooling capacity	1	DB_{OA}	COFcc_in_oa = A1 + A2 × DB_{OA} + A3 × DB_{OA}^2
			2	WB_{RA}	COFcc_in_ra =B1 + B2 × WB_{RA} + B3 × WB_{RA}^2 + B4 × WB_{RA}^3
			3	Lp	COFcc_in_lp = C1 + C2 × Lp
			4	Lh	COFcc_in_lh = D1 + D2 × Lh
	Outdoor unit		5	DB_{OA}	COFcc_out_oa = A1 + A2 × DB_{OA} + A3 × DB_{OA}^2
			6	WB_{RA}	COFcc_out_ra =B1 + B2 × WB_{RA} + B3 × WB_{RA}^2 + B4 × WB_{RA}^3
			7	Lp	COFcc_out_lp = C1 + C2 ×Lp
			8	Lh	COFcc_out_lh = D1 + D2 × Lh
		Energy consumption	9	R_C	COFce_out_rc = E1 + E2 × R_C + E3 × R_C^2
			10	DB_{OA}	COFce_out_oa = F1 + F2 × DB_{OA} + F3 × DB_{OA}^2
			11	WB_{RA}	COFce_out_ra =G1 + G2 × WB_{RA} + G3 × WB_{RA}^2 + G4 × WB_{RA}^3
Heating	Indoor unit	Heating capacity	12	WB_{OA}	COFhc_in_oa = H1+H2 × WB_{OA}+ H3 × WB_{OA}^2 + H4 × WB_{OA}^3
			13	DB_{RA}	COFhc_in_ra =I1 +I2 × DB_{RA} + I3 × DB_{RA}^2 + I4 × DB_{RA}^3
			14	Lp	COFhc_in_lp = J1 + J2 × Lp
			15	Lh	COFhc_in_lh = K1 + K2 × Lh
	Outdoor unit		16	WB_{OA}	COFhc_out_oa = H1 + H2 × WB_{OA} + H3 × WB_{OA}^2 + H4 × WB_{OA}^3
			17	DB_{RA}	COFhc_out_ra =I1 + I2 × DB_{RA}+I3 × DB_{RA}^2+I4 × DB_{RA}^3
			18	Lp	COFhc_out_lp = J1 + J2 × Lp
			19	Lh	COFhc_out_lh = K1 + K2 × Lh
		Energy consumption	20	R_C	COFhe_out_rc = L1 + L2 × R_C + L3 × R_C^2
			21	WB_{OA}	COFhe_out_oa = M1 + M2 × WB_{OA} + M3 × WB_{OA}^2 + M4 × WB_{OA}^3
			22	DB_{RA}	COFhe_out_ra =N1 + N2 × DB_{RA}+N3 × DB_{RA}^2

DBOA outdoor dry-bulb temperature

WB_{RA} indoor inlet wet-bulb temperature

WBOA outdoor wet-bulb temperature

DB_{RA} indoor inlet dry-bulb temperature

Lp length of refrigerant piping

Lh difference of elevation between indoor unit and outdoor unit

RC capacity, rated capacity (load ratio)

A1, A2, A3, ~N3 constants of characteristics formula

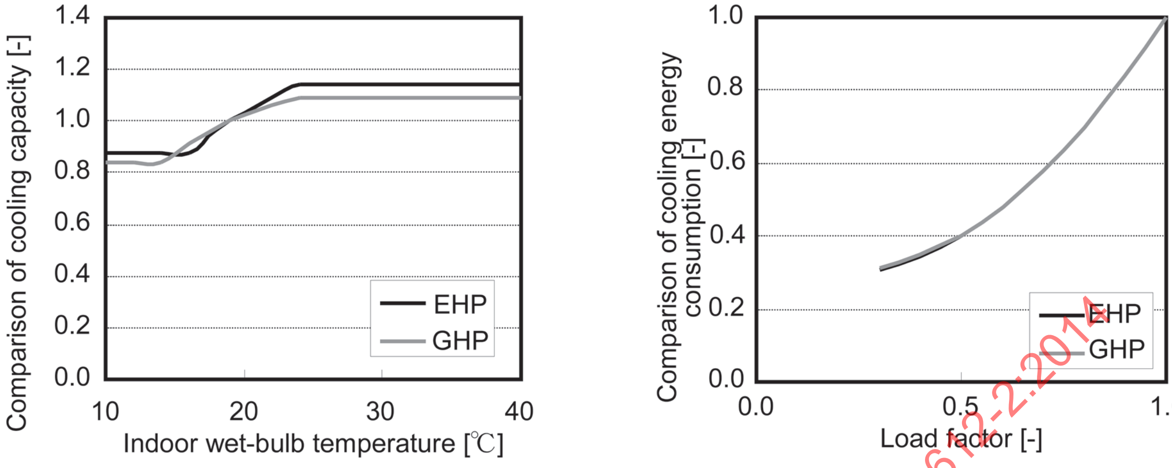


Figure D.11 — Equipment characteristics of VRF systems (example)

The improvement points of calculation models and major equipment characteristics shown in Table D.4 are described as follows.

D.3.3.4 Characteristics of calculation models of various packaged air-conditioners

D.3.3.4.1 Support for intermediate capacity and input display

In the past, only the data at the rated point of 100 % load factor were inputted in the partial load characteristics that represent inputs (energy consumption) and load factors (load processing capacity) to correct characteristic formulae of each equipment.

D.3.3.4.2 Development of common format to display characteristics data

On the assumption that there is a wide variety of equipment to be reviewed as shown in Table D.4 and the data of partial load characteristics unique to respective manufacturers would be supported in the future, the commonality of forms of approximate expressions that represent equipment characteristics is promoted as shown in Table D.6. As for approximate expressions, the scope of explanatory variables is divided into five sections in principle and expressed by approximation of cubic expression, including the support for discontinuous characteristics and support beyond the scope.

Table D.6 — Examples of common approximate expressions that represent equipment characteristics (examples of cooling)

Characteristics		Name of formula	Variable				Expressions of characteristics formula (A0~T3: constant)
			Name	Scope	Higher than	Less than	
Adjustment by room temperature	Adjustment of capacity	Kcti (WB)	WB: indoor wet-bulb temperature °C	Minimum	—	16 °C	$A3 \times WB^3 + A2 \times WB^2 + A1 \times WB + A0$
				Scope 1	16 °C	19 °C	$B3 \times WB^3 + B2 \times WB^2 + B1 \times WB + B0$
				Scope 2	19 °C	22 °C	$C3 \times WB^3 + C2 \times WB^2 + C1 \times WB + C0$
				Scope 3	22 °C	24 °C	$D3 \times WB^3 + D2 \times WB^2 + D1 \times WB + D0$
				Maximum	24 °C	—	$E3 \times WB^3 + E2 \times WB^2 + E1 \times WB + E0$
	Adjustment of input	Kcwti (WB)	WB: indoor wet-bulb temperature °C	Minimum	—	16 °C	$F3 \times WB^3 + F2 \times WB^2 + F1 \times WB + F0$
				Scope 1	16 °C	19 °C	$G3 \times WB^3 + G2 \times WB^2 + G1 \times WB + G0$
				Scope 2	19 °C	22 °C	$H3 \times WB^3 + H2 \times WB^2 + H1 \times WB + H0$
				Scope 3	22 °C	24 °C	$I3 \times WB^3 + I2 \times WB^2 + I1 \times WB + I0$
				Maximum	24 °C	—	$J3 \times WB^3 + J2 \times WB^2 + J1 \times WB + J0$
Adjustment by outdoor temperature	Adjustment of capacity	Kcta (WB)	DB: outdoor dry-bulb temperature °C	Minimum	—	−5 °C	$K3 \times WB^3 + K2 \times WB^2 + K1 \times WB + K0$
				Scope 1	−5 °C	15 °C	$L3 \times WB^3 + L2 \times WB^2 + L1 \times WB + L0$
				Scope 2	15 °C	25 °C	$M3 \times WB^3 + M2 \times WB^2 + M1 \times WB + M0$
				Scope 3	25 °C	43 °C	$N3 \times WB^3 + N2 \times WB^2 + N1 \times WB + N0$
				Maximum	43 °C	—	$O3 \times WB^3 + O2 \times WB^2 + O1 \times WB + O0$
	Adjustment of input	Kcwta (WB)	DB: outdoor dry-bulb temperature °C	Minimum	—	−5 °C	$P3 \times WB^3 + P2 \times WB^2 + P1 \times WB + P0$
				Scope 1	−5 °C	15 °C	$Q3 \times WB^3 + Q2 \times WB^2 + Q1 \times WB + Q0$
				Scope 2	15 °C	25 °C	$R3 \times WB^3 + R2 \times WB^2 + R1 \times WB + R0$
				Scope 3	25 °C	43 °C	$S3 \times WB^3 + S2 \times WB^2 + S1 \times WB + S0$
				Maximum	43 °C	—	$T3 \times WB^3 + T2 \times WB^2 + T1 \times WB + T0$

NOTE In addition to the items mentioned above, the length of piping, difference of elevation, and transformation of load factor are also formulated in the same manner.

D.3.3.5 Outline of characteristics of various packaged air-conditioners

D.3.3.5.1 VRF systems (types for use in cold climate areas)

The characteristics of VRF system types for use in cold climate areas are formulated by the same method as that for ordinary VRF systems that switch from cooling to heating and *vice versa*.

The types for use in cold climate areas are those equipment whose heating performance at the time of low outdoor air temperature is drastically improved by original technologies (adoption of new high-efficiency heat exchanger fins, for example) of respective companies. The types for use in cold climate areas have such characteristics that the tendencies of transformation factors of outdoor air temperature, outdoor unit capacity, and input values at the time of heating are different from those of ordinary types (see Figure D.13).

D.3.3.5.2 Air-conditioners for stores

The characteristics of representative types of air-conditioners for stores are formulated by the same method as that for ordinary VRF systems that switch from cooling to heating and *vice versa*. They have such characteristics that specific indoor units and outdoor units are combined together in advance as an integrated system and that there are many types. Their equipment characteristics show almost the same tendencies as those of VRF systems.

D.3.3.5.3 Characteristics of KHP (kerosene engine heat pump air-conditioner) VRF systems

As for the equipment characteristics of KHP, the two types (standard type and type for use in cold climate areas) are formulated based on GHP cooling/heating switching type VRF systems.

As for the transformation of capacity and input due to the outdoor air temperature at the time of heating, the equipment characteristics of the type for use in cold climate areas are reflected at outdoor wet-bulb temperature of 6 °C or lower, and it is differentiated from the standard type. Other characteristics are the same.

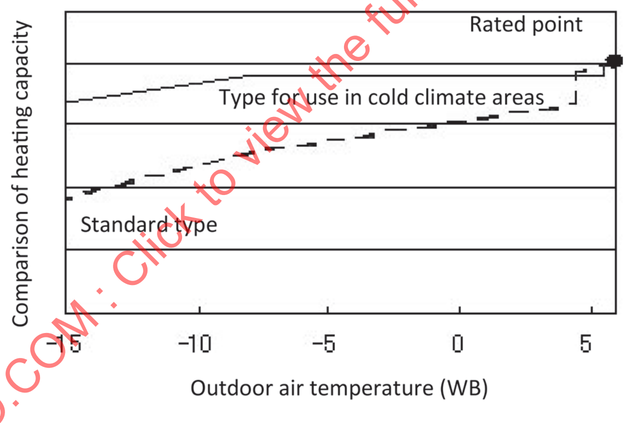


Figure D.12 — Comparison of heating characteristics of EHP VRF systems

D.3.4 Pumps and fans

D.3.4.1 Pumps

The pumps to be developed this time are two-pole and four-pole end suction centrifugal pumps. The inputs and outputs of models are shown in Figure D.13. The control system allows for operations with fixed speed and variable speed.

For characteristic models, data of pump characteristics are provided by, and the calculation formulae thereof are checked by, the Japan Society of Industrial Machinery Manufacturers.

a) The relations between the flow rate, GW_S, at rated capacity and the pump efficiency, EF_S, at rated capacity are formulated by JIS B 8313-91 (end suction centrifugal pumps).

b) The relations between the flow rate, GW, based on hearing data and characteristics of head, HW, and that between the flow rate, GW, and the characteristics of pump efficiency, EF, are formulated as the rate of change in rated values.

In the case of fixed speed, power consumption and heat quantity are calculated by the characteristic formula mentioned above. In the case of variable speed, necessary frequency is calculated inside, in addition to the characteristic formula mentioned above, and power consumption and heat quantity are calculated.

The calculation results of pump characteristics at the time of variable speed control by the above-mentioned model are shown in Figure D.14.

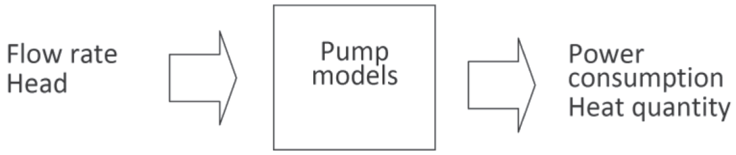


Figure D.13 — Inputs and outputs of pump models

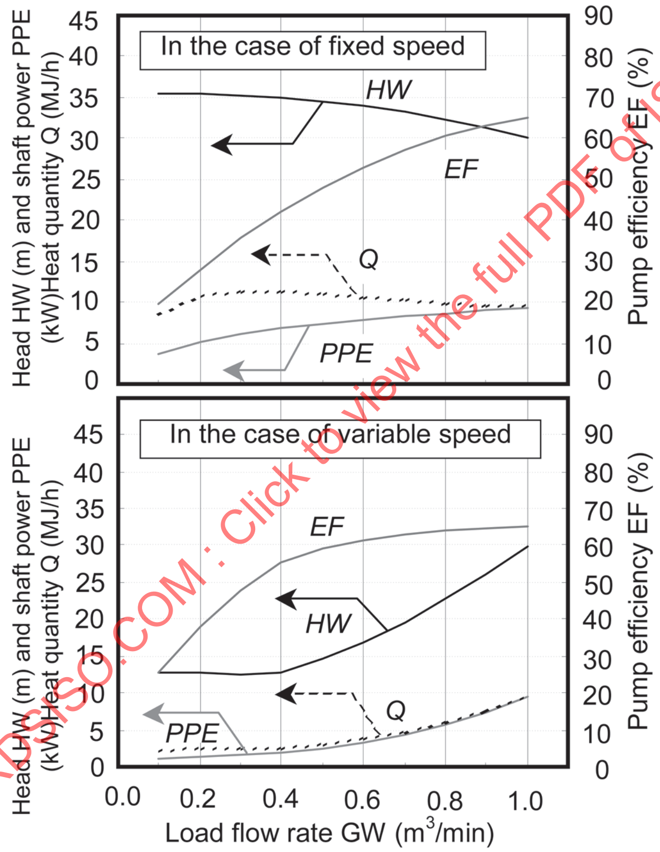


Figure D.14 — Characteristics of pump models (examples of rated flow rate of 1 m^3/min)

D.3.4.2 Fans

The fans to be developed this time are single-inlet type sirocco fans. The inputs and outputs of models are shown in Figure D.15. This time, the approximate expressions use simple models that do not make distinctions with the bearing numbers (# and No.) of fans to simplify the inputs at the time of program utilization. The control system allows for operations with fixed speed and variable speed.

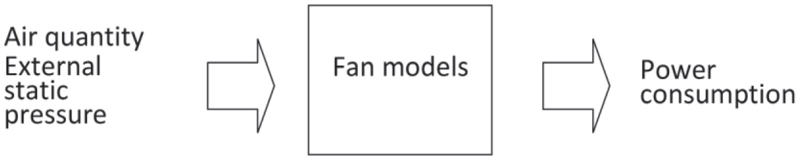


Figure D.15 — Inputs and outputs of fan models

The approximate expressions are developed by calculating the following relations based on the characteristics data of fans, which are obtained by conducting hearings of manufacturers.

$$FPAt = f_1(FGA,FPA) \tag{D.7}$$

$$FEF = f_2(FGA) \tag{D.8}$$

$$FPEEF = f_3(FGA) \tag{D.9}$$

where

FGA	is the air quantity at the time of operation (m^3/min);
FPA	is the external static pressure at the time of operation (Pa);
FPAt	is the external total pressure at the time of operation (Pa);
FEF	is the fan efficiency (%);
FPEEF	is the motor efficiency (%);
f_1, f_2, f_3	are the functions approximated by characteristics data.

They are formulated based on these formulae to calculate the shaft power of fans with *FPAt*, *FGA* and *FEF*, and moreover the power consumption with the shaft power of fans and *FPEEF*.

The calculation results of characteristics of fans by the models mentioned above are shown in Figure D.16. For reference, simple models are used as the approximate expressions for total pressure efficiency; this time, the characteristics of total pressure efficiency, etc. in JIS-B-8331 are not taken into consideration.

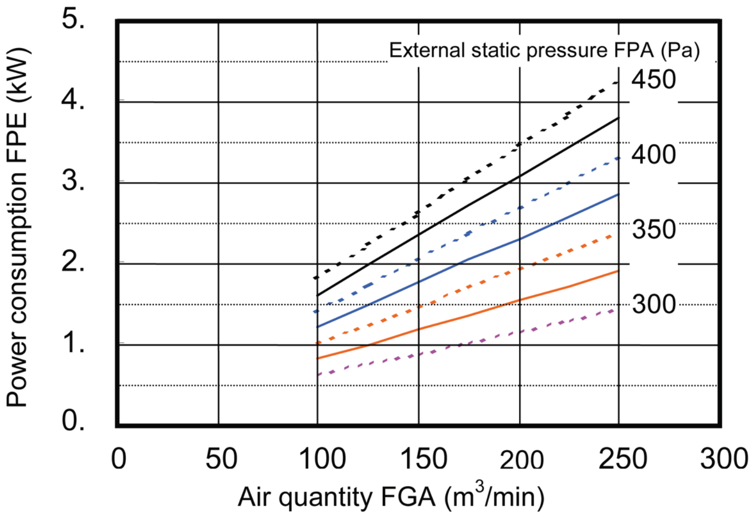


Figure D.16 — Characteristics of fan models (example)

D.3.5 Coils of air-conditioners

For coils of air-conditioners, plate fin coils are reviewed. As for calculation models, general methods to use heat transfer coefficients and wet surface coefficients, which are often used in coil selection calculations of air-conditioner manufacturers, are adopted. The parameters to decide the heat transfer coefficients and wet surface coefficients are found from product catalogues of manufacturers. The inputs and outputs of models are shown in Figure D.17.

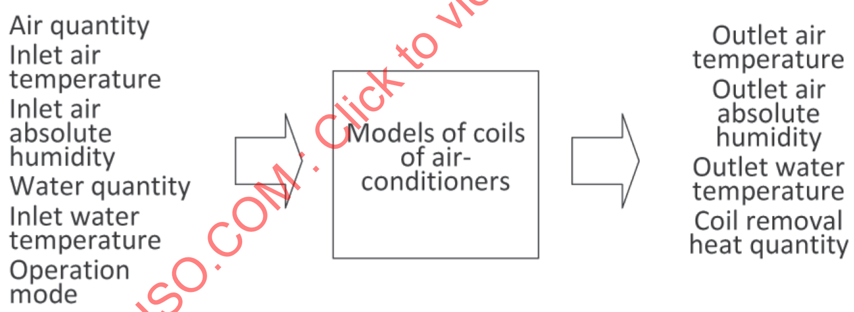


Figure D.17 — Inputs and outputs of models of coils of air-conditioners

In these models, first, the amount of coil heat extraction (sensible heat) is given as an assumed value, and the coil row numbers required for sensible heat treatment are calculated. Convergent calculations are made until the coil row numbers calculated agree with the coil row numbers set. Outlet conditions of water and air are calculated from the coil removal quantity (sensible heat) decided. The coil row numbers are calculated by the following formulae.

$$N_r = \frac{Q_t}{F_a \times K_f \times WSF \times MED} \tag{D.10}$$

$$\frac{1}{K_f} = \frac{1}{K_{fa} \times V_w^{K_{fb}}} + \frac{1}{K_{fc} \times V_a^{K_{fd}}} + r_0 \tag{D.11}$$

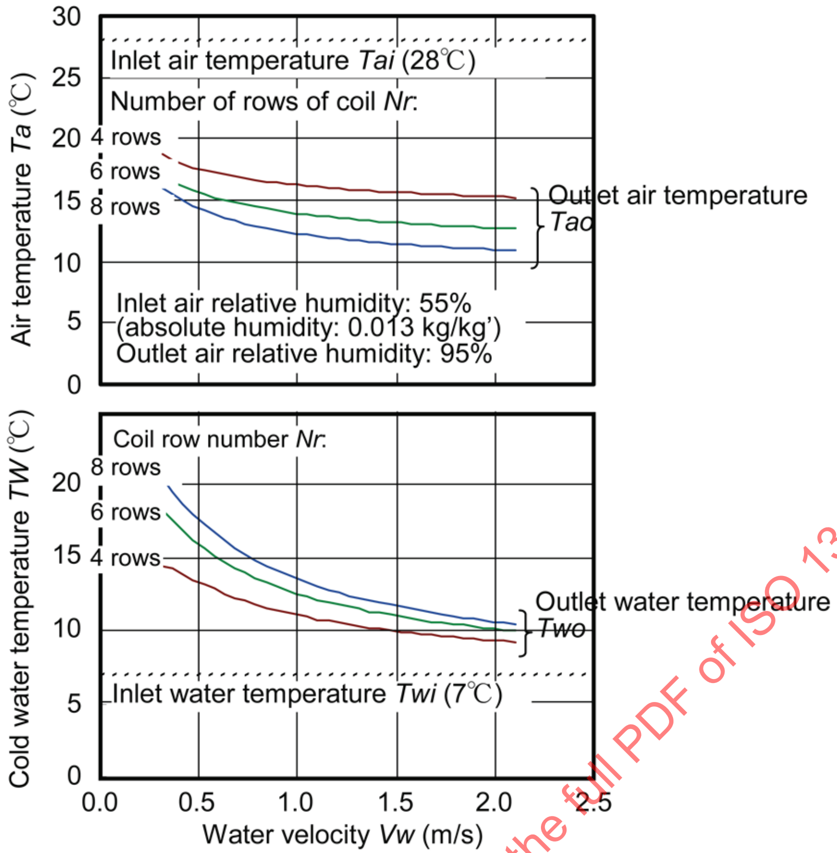
$$WSF = WS_a \times SHF^2 + WS_b \times SHF + WS_c \tag{D.12}$$

$$MED = \frac{(T_{ai} - T_{wo}) - (T_{ao} - T_{wi})}{\ln\left[(T_{ai} - T_{wo})/(T_{ao} - T_{wi})\right]} \quad (D.13)$$

where

N_r	is the number of rows of coil;
Q_t	is the amount of coil heat extraction (sensible heat);
F_a	is the front area;
K_f	is the heat transfer coefficient;
K_{fa}~K_{fd}	are the parameters to decide heat transfer coefficients;
r_0	is the thermal resistance of tube wall;
V_w	is the water velocity;
V_a	is the wind velocity;
WSF	is the wet surface coefficient;
SHF	is the sensible heat ratio;
WS_a~WS_c	are the parameters to decide wet surface coefficients;
MED	is the log-mean temperature difference;
T_{ai}	is the inlet air temperature;
T_{ao}	is the outlet air temperature;
T_{wi}	is the inlet water temperature;
T_{wo}	is the outlet water temperature;

The calculation results with models of coils of air-conditioners at the time of heating and cooling are shown in Figure D.18. As the water velocity of horizontal axis is proportional to water quantity, the temperature difference between cold water and hot water decreases in proportion to the increase in velocity.



NOTE Changes in outlet temperatures in proportion to water velocity in tube and number of rows of coil.

Figure D.18 — Characteristics of models of coils of air-conditioners (example of cooling coils)

D.3.6 Heat pump hot water supply system for business use

D.3.6.1 Calculation model for hot water supply system

Figure D.19 shows a calculation model for heat pump hot water supply system for business use. The subject of calculation is a centralized hot water supply system, and one set of heating and cooling equipment for hot water and hot water storage tank is provided to one circulation system. The demand for hot water supply and the point of use are set at one place for convenience, but the amount of hot water to be supplied represents the amount to be used simultaneously in the whole building. Figure D.20 shows a calculation model for commonly used combustion-type hot water boilers in the BEST. Different from the heat pump hot water supply system, calculation is conducted based on the following conditions:

a) hot water is returned to the upper part of hot water storage tank from the hot water return pipe on the secondary side;

b) make-up water is sent directly to the water heater from the lower part of hot water storage tank and is heated. If there is no make-up water and the temperature in the upper part of the hot water storage tank decreases, the water to be heated is sent from the upper part of hot water storage tank.

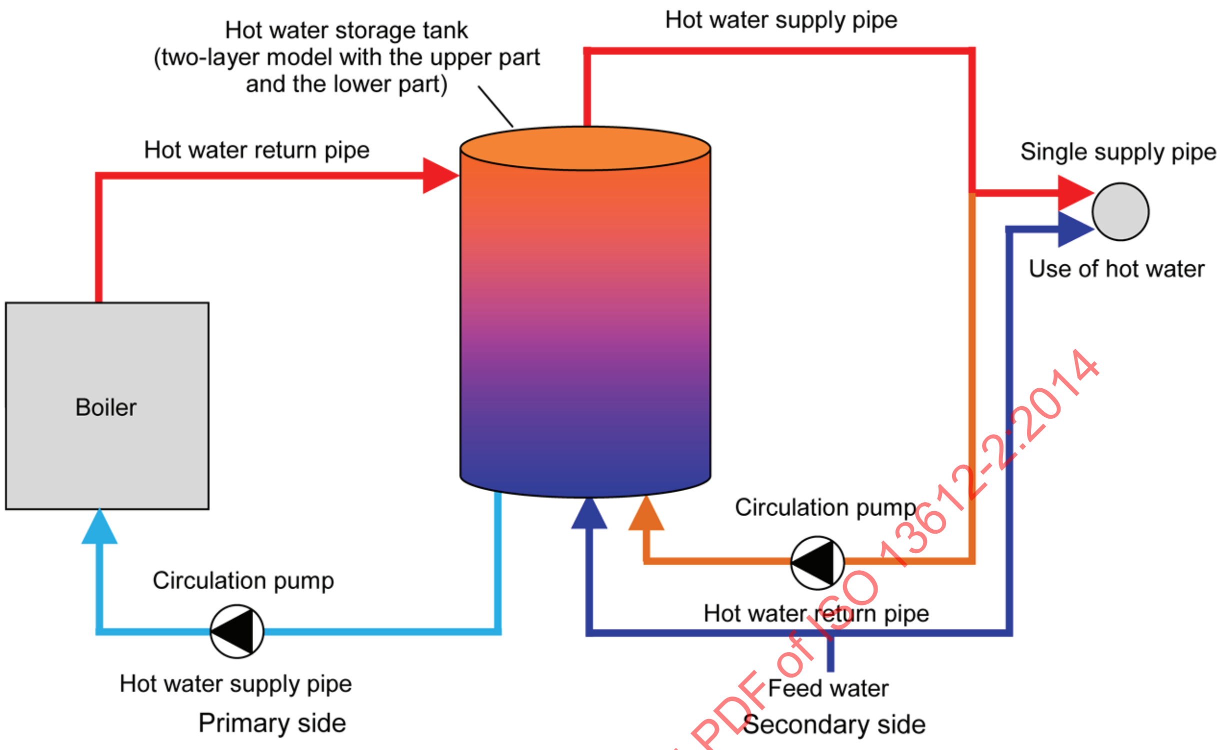


Figure D.19 — Calculation model for heat pump hot water supply system in the BEST

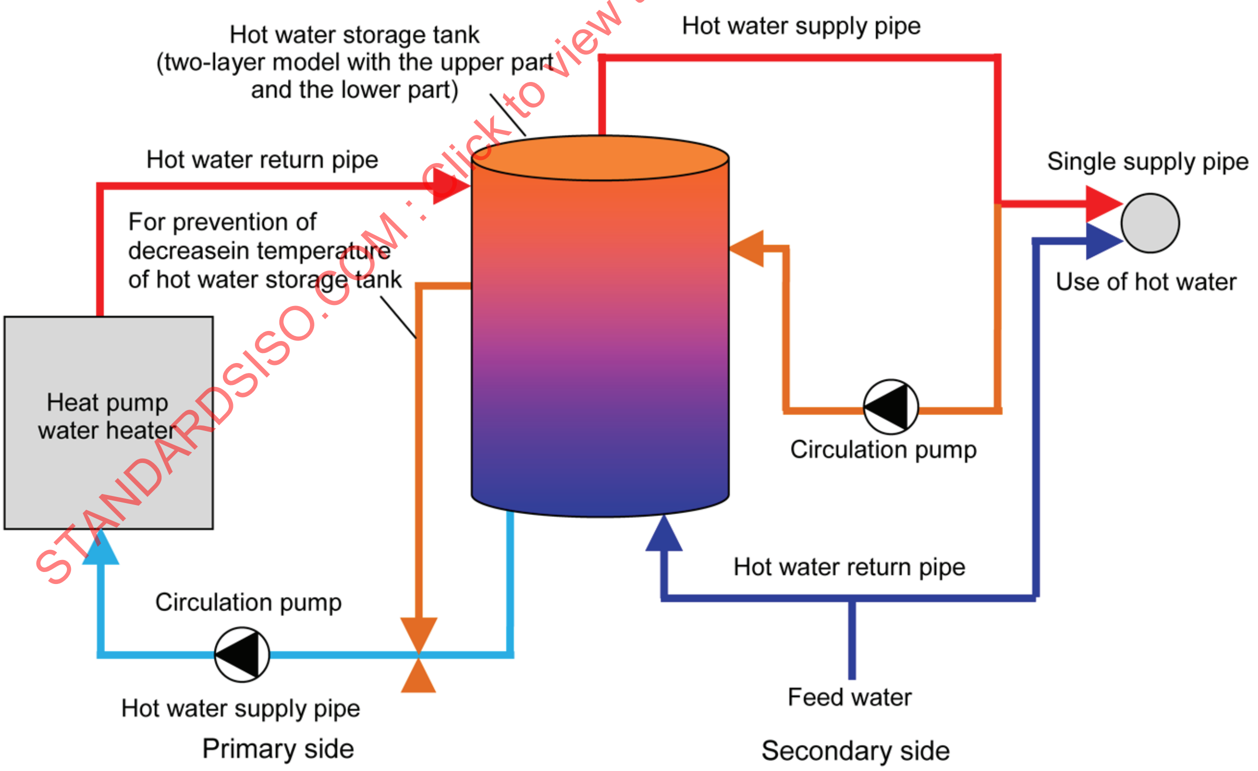


Figure D.20 — Calculation model for commonly used hot water boiler system in the BEST

D.3.6.2 Calculation model for hot water supply system calculation sequence

D.3.6.2.1 General

In the BEST, the pieces of equipment and the pipes that connect them together are defined as "calculation modules," and calculations on the whole system are conducted by delivering the results of calculation media (water volume and water temperature in this paper) between these modules based on the calculation sequence. The external definition to be made before calculation as well as each calculation module and the calculation sequence are shown as follows.

D.3.6.2.2 External definition

The calculation time-of-day data on outdoor air temperatures are acquired to calculate feed water temperatures and the efficiency ratios of water heaters.

The volume of hot water to be used and the pattern of hot water utilization are defined for the hot water demand to be calculated.

In addition to those defined above, the specifications of equipment and pipe modules (capacity, rated power consumption, thermal insulation specification, etc.) are defined.

D.3.6.2.3 Calculation sequence

Step 1: Calculation of feed water temperature

Step 2: Calculation to separate the feed water load and hot water load from the volume of hot water to be used

Step 3: Calculation on single supply pipe

Step 4: Calculation on the lower part of hot water storage tank

Step 5: Calculation on heat pump water heater

Step 6: Calculation on primary circulation pump for hot water supply

Step 7: Calculation of heat loss from hot water supply pipe on the primary side

Step 8: Calculation of heat loss from hot water return pipe on the primary side

Step 9: Calculation on the upper part of hot water storage tank

Step 10: Calculation on secondary circulation pump for hot water supply

Step 11: Calculation of heat loss from hot water supply pipe on the secondary side

Step 12: Calculation of heat loss from hot water return pipe on the secondary side

The calculation sequence mentioned above is conducted at every calculation time interval.

D.3.6.3 Description of calculation in each module

D.3.6.3.1 Calculation of feed water temperature

The temperature of feed water is calculated based on the outdoor air temperature by dividing the entire country of Japan into 12 regions and using "Regional Service Water Conversion Factors2." The outdoor air temperature is quoted from the Expanded AMeDAS Weather Data at every calculation time interval (10 min). However, as the temperature of feed water does not change as much as the outdoor air temperature does, the outdoor air temperature at 9:00 a.m. is used to calculate the representative temperature of feed water for the day.

D.3.6.3.2 Calculation to separate the feed water load and hot water load from the volume of hot water to be used

Figure D.21 shows the patterns of time-of-day use of hot water for typical applications. The volume of time-of-day use of hot water is calculated by multiplying this load ratio by the volume of hot water used per day.

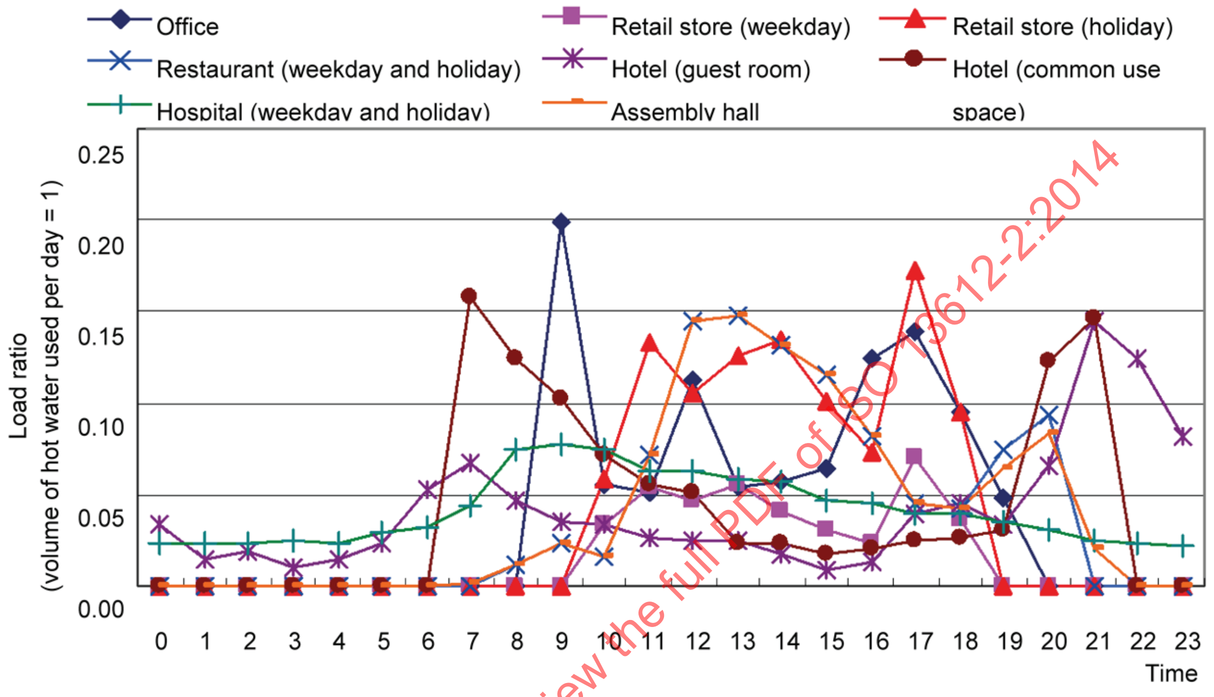


Figure D.21 — Example of patterns of hot water utilization for applications in each building

The feed water load and hot water load are calculated based on the feed water temperature calculated in Step 1 above, the example of hot water temperature at the outlet of hot water supply pipe on the secondary side: 58 °C" at the preceding calculation time of day, and the temperature of hot water used (example: 43 °C).

D.3.6.3.3 Calculation on single supply pipe

The volume of waste hot water is calculated based on the volume of water held in the single supply pipe and the frequency of disposal of waste hot water (twice a day, for example). And, it is added to the hot water load.

D.3.6.3.4 Calculation on the lower part of hot water storage tank

The volume of feed water to be supplied to the lower part of hot water storage tank is represented by the hot water load in Step 2 + the hot water load of single supply pipe in Step 3 (hereinafter referred to as "hot water load"). According to the load of hot water stored in the lower part of hot water storage tank and the water volume of primary pump for hot water supply × calculation time (the volume of water to be sent to heating and cooling equipment for hot water per calculation time), it is decided whether water is sent from the lower part or the upper part of hot water storage tank. If the hot water load cannot be disposed of within the calculation time, it is carried over to the next calculation time of day and the hot water load is accumulated.

Moreover, the amount of heat loss is calculated based on the ambient temperature of the storage tank and the state of thermal insulation.

D.3.6.3.5 Calculation on heat pump water heater

ON/OFF state of the water heater can be set by setting its schedule in advance. In general, however, the water heater is designed to automatically start its operation when the temperature difference between the “outlet temperature of hot water supply pipe on the primary side” and the “set temperature at the outlet of water heater (60 °C, for example)” reaches or exceeds the set value (5 °C, for example). If there is no temperature difference, its operation is turned OFF, and the outlet temperature of hot water supply pipe is the outlet temperature of the water heater. As for the outlet temperature of hot water supply pipe on the primary side, the value of the preceding calculation time of day is used.

For the input of equipment specifications of water heater, the rated heating capacity, rated power consumption, etc. are specified. For the equipment characteristics of heat pump water heater, those mentioned representatively in Figure D.22 are used. The characteristics are classified by the temperature of water that is supplied into the water heater (feed water temperature), and the partial load efficiency is presented by the relations between the outdoor air temperature and efficiency ratio (the ratio to rated efficiency).

The power consumption of heat pump water heater is calculated as follows:

1) heating capacity at a calculation time interval = water volume of primary pump for hot water supply × difference between outlet temperature and inlet temperature × operation time of water heater;

2) power consumption of water heater = heating capacity at a calculation time interval × [1/(rated efficiency × efficiency ratio)].

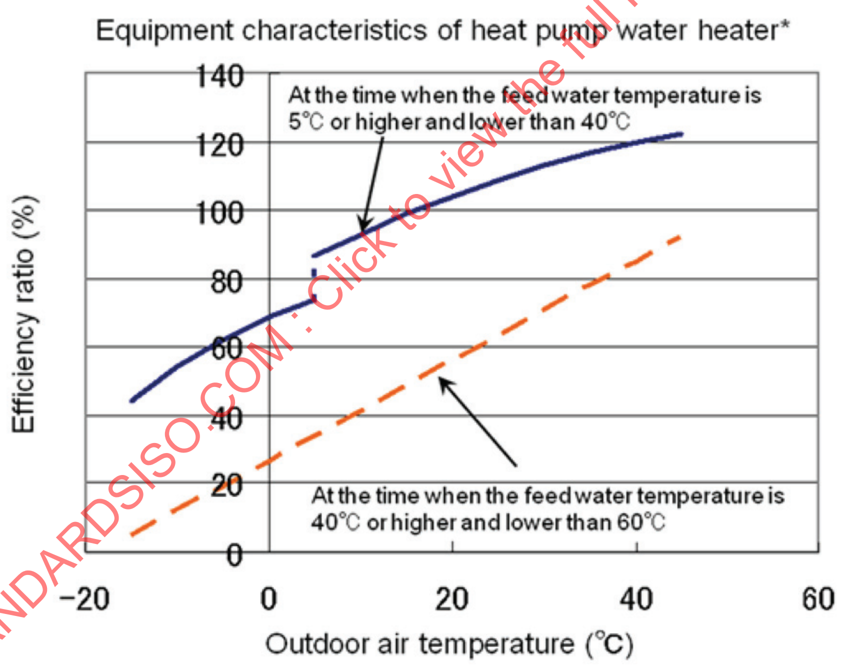


Figure D.22 — Equipment characteristics of heat pump water heater

D.3.6.3.6 Calculation on primary circulation pump for hot water supply

In line with ON/OFF state of operation of heat pump water heater, the volume of water flown and the amount of electricity consumed per calculation time are calculated based on the product of the rated water volume, rated amount of electricity consumed, and operation time as the inputted values of equipment specifications of the pump.

D.3.6.3.7 Calculation of heat loss from hot water supply pipe on the primary side

The average amount of heat loss per unit temperature difference is calculated based on the length, diameter, and the state of thermal insulation of the pipe. And, the outlet temperature of the pipe and the amount of heat lost from the pipe are calculated based on ambient temperature as well as the inlet temperature (temperature of the lower or upper part of hot water storage tank, for example) and flow rate of the hot water supply pipe on the primary side.

The heat loss from the pipe is calculated by the following formula (heat loss from other pipes and that from the hot water storage tank are also calculated based on the same concept):

$$T_{out} = T_{in} - (T_{in} - T_a) \times [1 - EXP\ (-C_t/3600) \times K_w/(C \times L/1000)] \tag{D.14}$$

where

T_{out} is the outlet temperature of the pipe (°C);

T_{in} is the inlet temperature of the pipe (°C);

T_a is the amount temperature of the pipe (°C);

C_t is the calculation time interval (s);

K_w is the average heat loss rate (W/°C);

C is the specific heat of water 1,167 (kg/W °C);

L is the volume of water in the pipe (g).

$$Q_P = 1{,}163 \times L/1\ 000 \times (T_{in} - T_{out}) \tag{D.15}$$

where

Q_P is the amount of heat lost from the pipe.

D.3.6.3.8 Calculation of heat loss from hot water return pipe on the primary side

As is the case with Step 7 above, the average amount of heat loss per unit temperature difference is calculated based on the length, diameter, and the state of thermal insulation of the pipe. The outlet temperature of the pipe (temperature in the upper part of hot water storage tank) and the amount of heat lost from the pipe are calculated based on ambient temperature as well as the inlet temperature (outlet temperature of water heater, 60 °C, for example) and flow rate of the hot water return pipe on the primary side.

D.3.6.3.9 Calculation on the upper part of hot water storage tank

The hot water heated by the water heater as a medium to be flown into the upper part of hot water storage tank and the hot water returned from the hot water secondary pipe of circulation system are mixed together and sent to the hot water secondary pipe of circulation system and to the supply pipe on the primary side to prevent the temperature of hot water storage tank from decreasing. The water supply in this case is calculated based on the volume of water and the balance of water temperature.

The total value of the water volume in the upper part of hot water storage tank and that in the lower part thereof is always equal to the effective volume of hot water stored in the hot water storage tank. To simplify the calculation, the water temperatures in the upper part and the lower part of hot water storage tank are calculated on the assumption that the hot water in the tank is completely mixed together (it is assumed that there is a temperature stratification between the upper part and the lower part).

Moreover, the amount of heat loss is calculated based on the ambient temperature and the state of thermal insulation of hot water storage tank.

D.3.6.3.10 Calculation on secondary circulation pump for hot water supply

The secondary circulation pump for hot water supply always operates when there is a hot water load. If the volume of hot water to be used is set at none and the secondary circulation pump for hot water supply is scheduled not to operate by setting, the operation of the pump is stopped and the calculation to reduce the loss of heat from pipes and the amount of electricity consumed by the pump can be conducted.

D.3.6.3.11 Calculation of heat loss from hot water supply pipe on the secondary side

As is the case with Step 7 above, the average amount of heat loss per unit temperature difference is calculated based on the length, diameter, and the state of thermal insulation of the pipe. The outlet temperature of the pipe (outlet temperature at the point of use of hot water) and the amount of heat lost from the pipe are calculated based on the ambient temperature as well as the outlet temperature (outlet temperature in the upper part of hot water storage tank, 60 °C, for example) and flow rate of the hot water supply pipe on the secondary side.

D.3.6.3.12 Calculation of heat loss from hot water return pipe on the secondary side

As is the case with Step 7 above, the average amount of heat loss per unit temperature difference is calculated based on the length, diameter, and the state of thermal insulation of the pipe. The outlet temperature of the pipe (temperature of water returning to the upper part of hot water storage tank) and the amount of heat lost from the pipe are calculated based on the ambient temperature as well as the outlet temperature (outlet temperature at the point of use of hot water, 58 °C, for example) and flow rate of the hot water return pipe on the secondary side.

D.3.7 Thermal storage

D.3.7.1 Concepts of thermal storage tank models

For the concepts of thermal storage tank models in BEST, Figure D.23 shows the concept of thermal storage tank of multi-connected mixing type, and Figure D.24 shows the concept of thermal storage tank model of temperature stratification type.

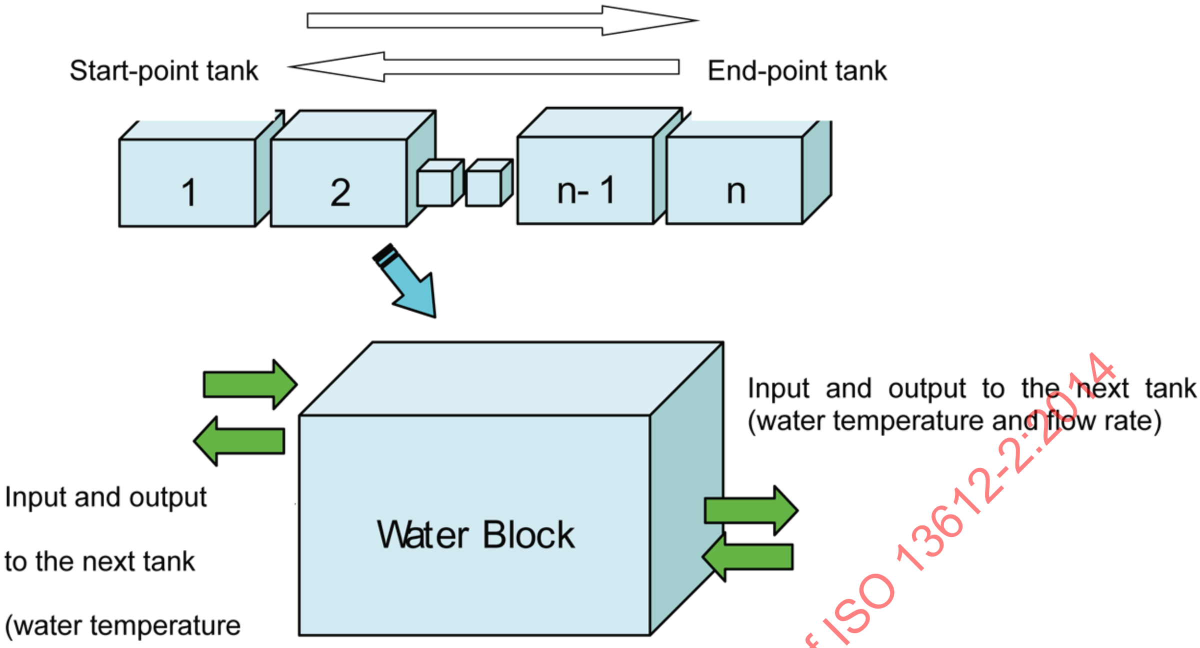


Figure D.23 — Concept of thermal storage tank of multi-connected mixing type

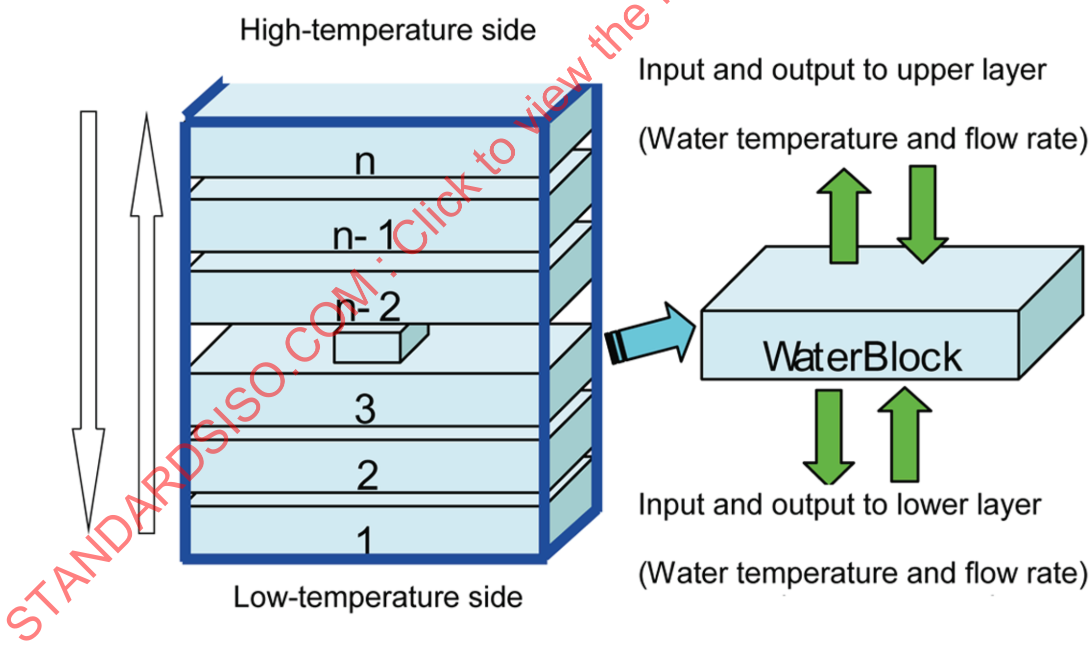


Figure D.24 — Concept of thermal storage tank model of temperature stratification type

In the case of the thermal storage tank of multi-connected mixing type, each tank is regarded as a water block and the water temperature is uniform. The concept is that these water blocks are coupled together, and there are inputs and outputs generated to the neighbouring tanks or to the primary and secondary sides. Therefore, the water temperature and inflow and outflow of each tank are the necessary information.

In the case of thermal storage tank model of temperature stratification type, it is considered that there are continuous temperature changes in the up-and-down direction in reality. Here, as shown in Figure D.24, the water temperature of one water block is considered uniform, but temperature stratification is formed by temperature differences caused by several water blocks piled up.

For both the thermal storage tank of multi-connected mixing type and the thermal storage tank model of temperature stratification type, the data of water temperature and flow rate are sent and received between the primary and secondary sides at both ends of water blocks.

D.3.7.2 Outline of thermal storage program in BEST

D.3.7.2.1 Current status of thermal storage program in BEST

The BEST programs include water thermal storage systems (multi-connected mixing type and temperature stratification type) and ice storage systems (on-site construction type). The programs for the water thermal storage system and ice storage system (on-site construction type) are mainly explained below.

D.3.7.2.2 Prospective systems and module structures

Figure D.25 shows the prospective basic water thermal storage air-conditioning system. Two or more heat sources can be installed on the primary side. At the time of thermal storage, all units are basically operated at full load but a module to control the number of units is also available for daytime following operation and in spring and fall. The inlet water temperature of heat source is controlled by a three-way valve. The pump on the secondary side can control variable flows. To prevent the temperature of return water to thermal storage tanks from lowering (at the time of cooling) in the case where there is a constant flow system such as FCU system and loads are small, it is also possible to select the secondary side water supply temperature control three-way valve to control the temperature of water supplied. The algorithm of TESEP-W is used mainly for the calculation of water temperatures in thermal storage tanks (item 17, see Table D.7). Such ideas are also reflected in other parts.

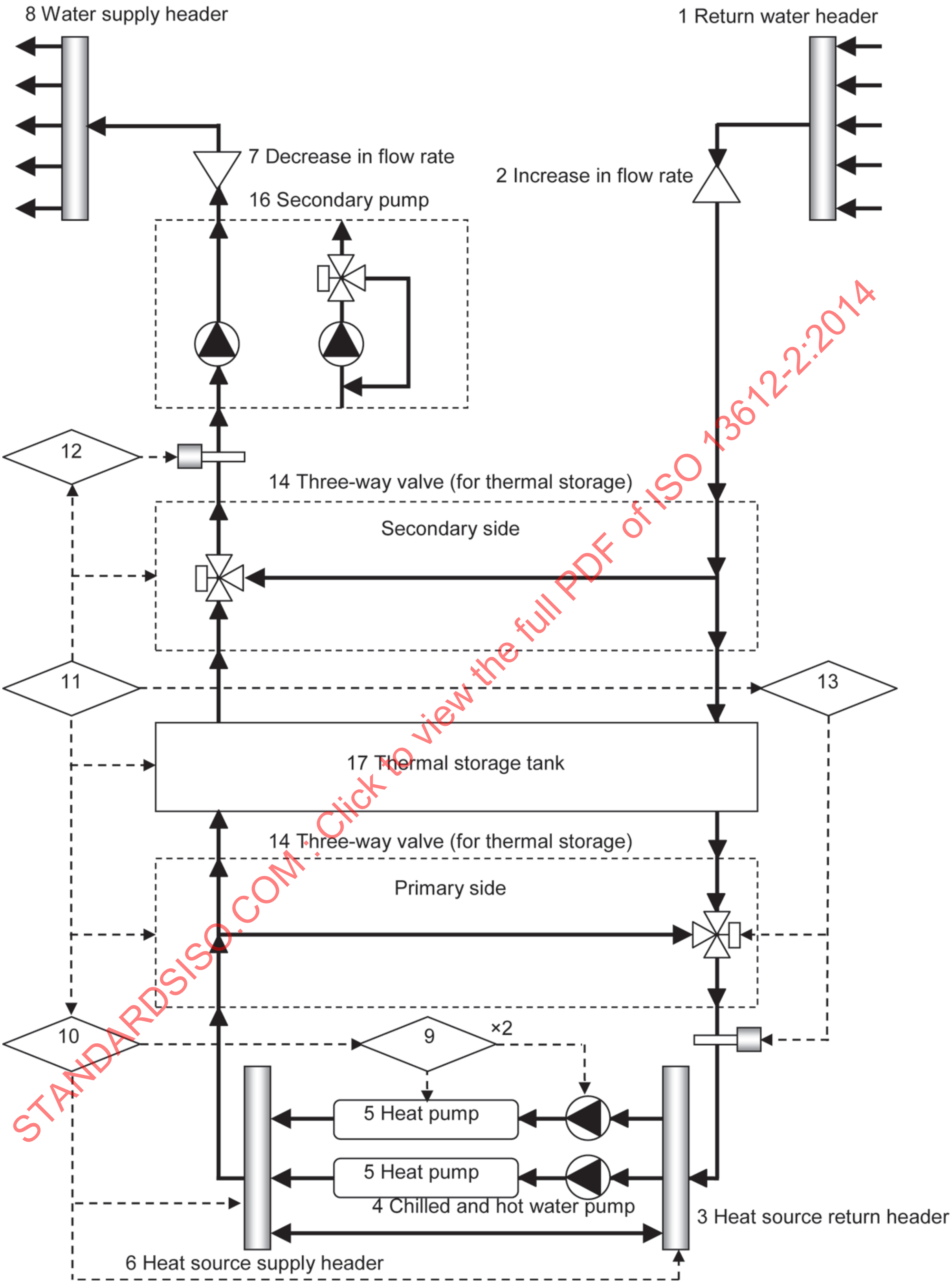


Figure D.25 — Prospective water thermal storage system and module structure

Figure D.26 shows the prospective ice storage system (on-site construction type). In the case of the ice storage system, the concept of thermal storage tank is similar to that of the water thermal storage tank. It can be considered that coils are installed in the water blocks that are shown in Figure D.23 and Figure D.24. In the ice storage system, ice is produced by a brine chiller in a thermal storage tank during