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Ventilation for buildings — Calculation of room temperatures and of load and energy for buildings with room conditioning systems

Lüftung von Gebäuden — Berechnung der Raumtemperaturen, der Last und Energie von Gebäuden mit Klimaanlagen

Ventilation des bâtiments — Calcul de la température des pièces, de la charge et de l'énergie pour les bâtiments équipés de système de conditionnement d'air

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Fax: (+43 1) 213 00-818
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Air-conditioning engineering

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Ventilation for buildings - Calculation of room temperatures and of load and energy for buildings with room conditioning systems

Systèmes de ventilation des bâtiments - Calcul de la température des pièces, de la charge et de l'énergie pour les bâtiments équipés de système de climatisation

Lüftung von Gebäuden - Berechnung der Raumtemperaturen, der Last und Energie für Gebäuden mit Klimaanlagen

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Contents

Page

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Licensed to Sang Hoon Lee. ANSI store order # X_92429. Downloaded 11/11/2008 11:05 PM. Single user license only. Copying and networking prohibited

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Foreword

This document (EN 15243:2007) has been prepared by Technical Committee CEN/TC 156 “Ventilation for buildings”, the secretariat of which is held by BSI.

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of an identical text or by endorsement, at the latest by February 2008, and conflicting national standards shall be withdrawn at the latest by February 2008.

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

Attention is drawn to the need for observance of EU Directives transposed into national legal requirements. Existing national regulations with or without reference to national standards, may restrict for the time being the implementation of the European Standards mentioned in this report.

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Introduction

Clauses 13 and 14 of this standard deal with the calculation of the energy demand of HVAC systems, specifically in connection with the Energy Performance Rating in connection with the Energy Performance of Buildings Directive. The calculation of the building energy demand that has to be met is dealt with in prEN ISO 13790 "Thermal Performance of Buildings – Calculation of energy use for space heating and cooling" – this information is input data for the procedures addressed in this standard. Calculation methods satisfying this standard may also be used for other purposes, (for example, for system sizing). These are covered by Clauses 1 to 12 of the standard. Users of calculation methods should exercise care in ensuring that the need for appropriate modifications are considered and, if necessary, implemented for other applications.

The standard has an unusual large portion of informative annexes in terms of number and size. This is due to the fact that the area covered by this standard is highly dependent on the system solutions, which exist in a large number of variations, and therefore many issues can only be shown in an exemplary way and experts in the different countries would not agree in putting this generally in normative way. Also, due to the different approaches taken in the different countries for the implementation of the EPBD, different solutions should be possible in parallel and the normative part can only be general. Nevertheless, the standard intends to give room for documentation of specific solutions, in order to provide information in enough depth, to make common parts and difference transparent for possible closer harmonisation in future.

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1 Scope

The scope of this European Standard is:

- To define the procedure how the calculation methods to determine the temperatures, sensible loads and energy demands for the rooms shall be used in the design process.

- To describe the calculation methods to determine the latent room cooling and heating load, the building heating, cooling, humidification and dehumidification loads and the system heating, cooling, humidification and dehumidification loads.

- To define the general approach for the calculation of the overall energy performance of buildings with room conditioning systems.

- To describe one or more simplified calculation methods for the system energy requirements of specific system types, based on the building energy demand result from prEN ISO 13790, and to define their field of application.

A general framework standard is given which imposes an hourly calculation for all cases which cannot be covered by simplified methods, and gives requirements on what has to be taken into account. Input and output data are defined.

The target audience of this standard is twofold:

- Designers of HVAC systems, which are given an overview of the design process with the relevant references to the different involved standards (Clauses 5 to 12).

- Developers of regulations and tools, which find requirements for calculation procedures to be used for the energy requirements according to the EPBD (Clauses 13 and 14).

The idea followed by this standard is, that for the detailed approach one single calculation method is used for the different room related purposes such as room temperature calculation, room cooling and heating load calculation, and room energy calculation. This means, for the building type envisaged (buildings with room conditioning systems) it is an alternative to simplified calculation methods such as heating load according to EN 12831 and heating energy according to prEN ISO 13790. This standard does not describe any detailed methods for the sensible room based calculations. For this it refers to the relevant standards EN ISO 13791, EN ISO 13792, EN 15255 and EN 15265.

This standard specifies simplified methods and describes the necessary functionality of methods for the calculation of standardized annual energy consumption by systems providing temperature control, mechanical ventilation and humidity control in existing and new buildings. For brevity, these are described as HVAC systems. These systems may provide any or all of these services, including heating, cooling, air filtration, humidification or dehumidification. For the air side calculations of air based systems it refers to EN 15241. Systems providing heating but no other services are covered by EN 15316. These boundaries are, however, not kept strictly in the informative annexes, because some of the shown example calculations follow a holistic approach and this separation is therefore not always possible.

The standard specifically relates to demand calculations needed for Energy Performance Rating in connection with the Energy Performance of Buildings Directive.

These installations may include:

- Emission, distribution, storage and generation for cooling.

- Emission, distribution and heat exchanger for heating if these functions are performed using an air conditioning system; all heating functions performed by direct heating or using water as a heat transport medium are treated in other standards.

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The calculation of cooling and heating energy demand within buildings is dealt with by prEN ISO 13790 and is a required input. This standard only addresses these issues to the extent that HVAC systems have an influence on the loads.

The boundaries and relations between the covered areas are shown in Figure 1.

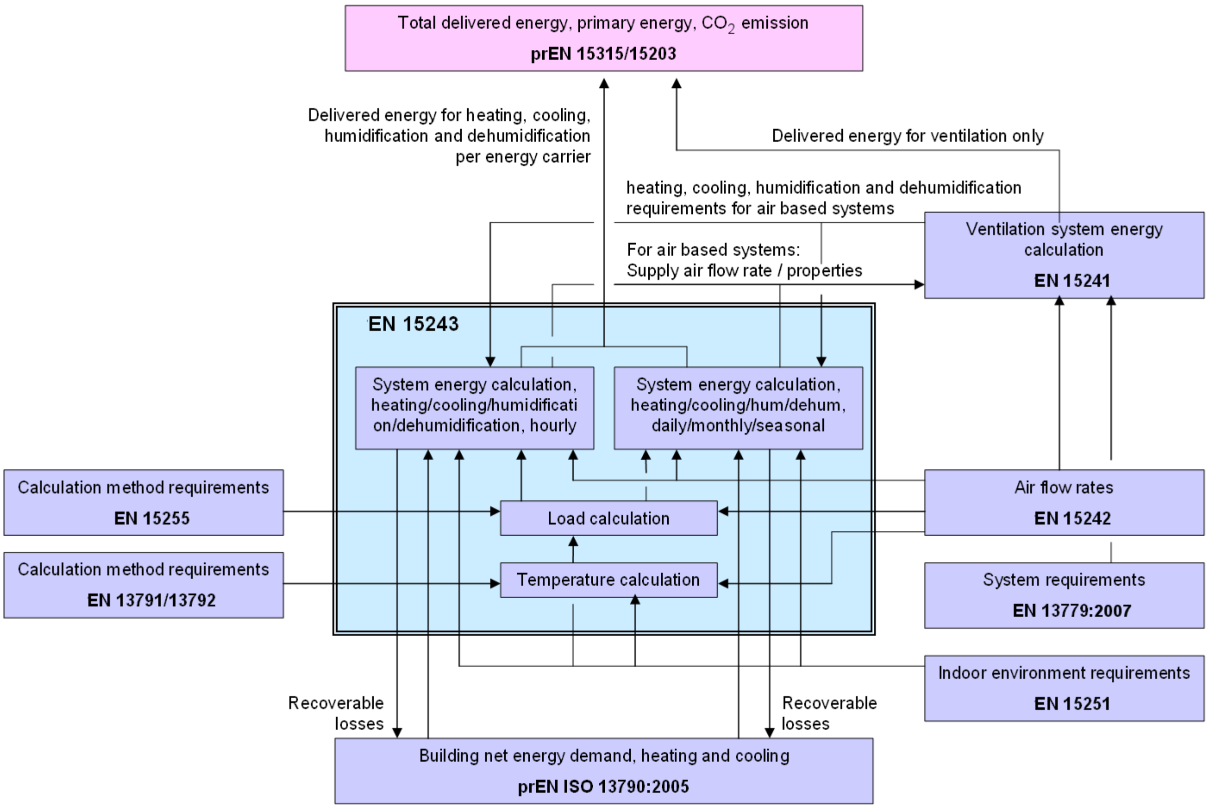

Figure 1 — Chart showing the relations to other standards related to the EPBD

2 Normative references

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

EN 13779, *Ventilation for non-residential buildings — Performance requirements for ventilation and room-conditioning systems*

EN 15026, *Hygrothermal performance of building components and building elements — Assessment of moisture transfer by numerical simulation*

EN 15241, *Ventilation for buildings — Calculation methods for energy losses due to ventilation and infiltration in commercial buildings*

EN 15242:2007, *Ventilation for buildings — Calculation methods for the determination of air flow rates in buildings including infiltration*

EN 15251, *Indoor environmental input parameters for design and assessment of energy performance of buildings addressing indoor air quality, thermal environment, lighting and acoustics*

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EN 15255:2007, *Thermal performance of buildings — Sensible room cooling load calculation — General criteria and validation procedures*

EN 15316-2-1, *Heating systems in buildings — Method for calculation of system energy requirements and system efficiencies — Part 2-1: Space heating emission systems*

EN 15377-3, *Heating systems in buildings — Design of embedded water based surface heating and cooling systems — Part 3: Optimizing for use of renewable energy sources*

prEN ISO 13790, *Energy performance of buildings — Calculation of energy use for space heating and cooling (ISO/DIS 13790:2005)*

EN ISO 13792, *Thermal performance of buildings — Calculation of internal temperatures of a room in summer without mechanical cooling — Simplified methods (ISO 13792:2005)*

prEN ISO 15927-2, *Hygrothermal performance of buildings — Calculation and presentation of climatic data — Part 2: Hourly data for design cooling load (ISO/DIS 15927-2:2007)*

EN ISO 15927-4, *Hygrothermal performance of buildings — Calculation and presentation of climatic data — Part 4: Hourly data for assessing the annual energy use for heating and cooling (ISO 15927-4:2005)*

3 Terms and definitions

For the purposes of this document, the following terms and definitions apply.

3.1
room
enclosed space or part of an enclosed space

3.2
room heating load
daily profile of the energy flow rate which must be added to a room under design conditions in order to keep its comfort conditions within a defined range

3.3
room cooling load
daily profile of the energy flow rate which must be extracted from a room under design conditions in order to keep its comfort conditions within a defined range

3.4
room sensible cooling load
daily profile of the energy flow rate which must be extracted from a room under design conditions in order to keep its temperature (air temperature or operative temperature) within a defined range

3.5
room latent cooling load
daily profile of the energy flow rate which must be extracted from a room under design conditions in order to keep its humidity below a defined limit

3.6
basic room sensible cooling load
daily profile of the energy flow rate which must be extracted from a room under design conditions in order to keep its air temperature at a constant value

3.7
room latent heating load
daily profile of the energy flow rate which must be added to a room under design conditions in order to keep its humidity above a defined limit

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3.8
room conditioning system
system able to keep a comfort conditions in a room within a defined range

NOTE Air conditioning as well as surface based radiative systems are included.

3.9
zone
group of rooms forming part of a building, assigned to a system

3.10
zone cooling load
daily profile of the energy flowrate to be extracted from a zone

NOTE It is calculated by superposition of the room cooling load profiles

3.11
zone heating load
daily profile of the energy flowrate to be added to a zone for heating purposes

NOTE It is calculated by superposition of the room heating load profiles

3.12
zone humidification load
daily profile of the energy flowrate to be added to a zone for humidification purposes

NOTE It is calculated by superposition of the room humidification load profiles

3.13
system
set of HVAC components which provides heating, cooling, humidification and dehumidification energy to a zone in order to meet the comfort conditions in the rooms

NOTE The system boundaries are at the emission/extraction of heat and/or conditioned air to the rooms, the envelope of the system (leakage and/or heat transfer) and the energy delivered to the system in form of fuel and/or electricity. Intermediate boundaries may be necessary for calculation purposes

3.14
system cooling load
daily profile of the energy flowrate to be extracted from a system under design conditions taking into account the system impact

3.15
system heating load
daily profile of the energy flowrate to be added to a system under design conditions which meets the zone heating load, taking into account the system impact

3.16
system cooling capacity
maximum heat extraction flowrate of a system under specified conditions

3.17
system heating capacity
maximum heat addition flowrate of a system under specified conditions

3.18
room cooling energy demand
energy amount to be extracted from the room in order to keep its comfort conditions within a defined range throughout the year under typical meteorological conditions

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3.19
room heating energy demand
energy amount to be added to the room in order to keep its comfort conditions within a defined range throughout the year under typical meteorological conditions

3.20
system cooling energy demand
energy amount to be extracted from the system in order to keep its intended operating conditions throughout the year under typical meteorological conditions

3.21
system heating energy demand
energy amount to be added to the system in order to keep its intended operating conditions throughout the year under typical meteorological conditions

3.22
HVAC system
system providing temperature control, mechanical ventilation and humidity control in a building

3.23
turn over temperature
outside temperature when no heating or cooling demand exists within a building, also known as the ‘free temperature’ of the building

Definitions of system types are given in Table 1, Table 2 and Annex C.

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4 Symbols and abbreviations

NOTE The symbols of Annex E and Annex F are not taken into account in the lists below, since they origin from different national documents and do not follow the same rule.

Table 1 — Symbols and units

symbol	Name	Unit
A	Area	m^2
a	Humidity production	kg/s
c	Specific heat capacity	J $kg^{-1}K^{-1}$
E	Delivered energy or energyware	J
EIR	Energy input ratio	-
EER	Energy efficiency ratio	-
F	Frequency	-
f	Fraction or factor	-
f_w	Weighting factor	-
Q	Thermal energy	J
q_m	Mass flow rate	$kg.s^{-1}$
W	Auxiliary electrical energy	J
x	Absolute humidity	kg/kg dry air
θ	Temperature	°C
θ_{ii}	initial internal temperature	°C
$\delta\theta_{vs}$	spatial variation of temperature	K
$\delta\theta_{vt}$	control accuracy	K
ρ	Density	kg/m^3

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Table 2 — Indices

symbol	Name
a	Air
aux	auxiliary
c	Cooling
d	Distribution
dem	Demand
e	Emission
entr	Entering
f	Floor
g	Generation
h	Heating
hum	Humidifier
i	Internal
ii	Initial internal
loss	Losses
Ia	interaction
in	input to subsystem
prim	Primary
r	Refrigerant
recirc	Recirculation
s	Storage
sat	Saturation
sea	Seasonal
sens	Sensible
start	At starting time
tot	Total
out	output of subsystem
w	Water
(E)	Electricity – indicating the fuel type for delivered energy
(G)	Gas – indicating the fuel type for delivered energy
(O)	Oil – indicating the fuel type for delivered energy

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5 General approach

The general approach for the calculation procedure of a building with a room conditioning system is shown in Figure 2.

The first step is to define whether a humidity control is necessary or not. If so, there is a cooling and/or humidification requirement, and the latent room cooling and heating load has to be calculated.

If no humidity control is required, it has to be decided whether cooling is necessary or not. An optional step for this is a room temperature calculation. It can be done for some specifically chosen rooms.

If no cooling is required, the project is out of the scope of this standard and can be treated according to the heating related standards EN 12831, prEN ISO 13790 and the different parts of EN 15316.

If cooling is required, a next optional step is a basic room cooling load calculation. This optional calculation is done with constant comfort conditions and gives a basic value which can serve for comparison with different system solution values. It can give help in the system and control choice decision.

If an envisaged system is not adequate to meet the basic room heating and cooling loads, modifications may be considered on the building envelope and construction.

After the choice of the system, an appropriate room cooling and heating load calculation shall be performed. This is the base for sizing the room based cooling equipment such as air volume flow rate, chilled ceiling power, fan coil power, radiator power, embedded system power etc. It generally consists of three parts:

— Room sensible cooling load.

— Room cooling load due to room based ventilation.

— Room latent cooling load.

The load due to room based ventilation only occurs and has to be taken into account, when there is room based ventilation such as natural ventilation through windows or vents or room based mechanical ventilation.

The latent room cooling load has been taken into account in the beginning, when the humidity of a room has to be controlled. But the latent gains have an influence on the local or central equipment also in the other cases (uncontrolled dehumidification by cooling coils).

The zone load calculation, which is done by superposition of the room load profiles, gives then the base for the sizing of central cooling equipment.

Economic and/or energy considerations may be the reason for an equipment sizing which does not cover all the load cases. In this case, a further room temperature calculation may be performed to show the consequences of this undersizing.

With a chosen and sized central equipment, yearly energy calculations can be performed in order to optimise the system. Iterations may include building modifications, a new system and control choice and/or a new equipment sizing.

An outline for a “best procedure” for this general approach is given in Annex A.

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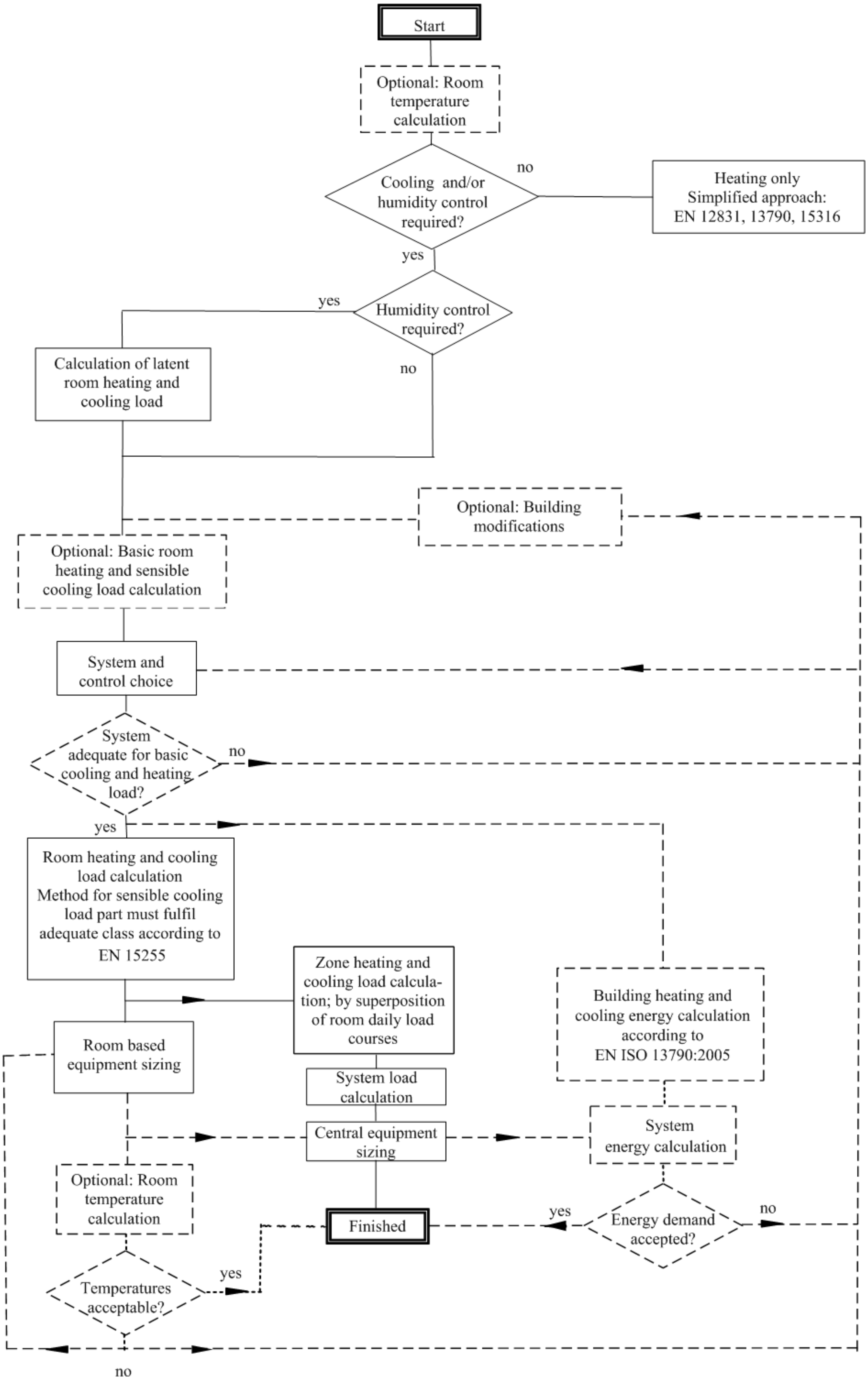

Figure 2 — Flow chart for general approach (all dotted steps and loops are optional)

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6 Room temperature calculation without room conditioning system

6.1 Choice of rooms

Typical rooms representing different building areas shall be chosen.

Annex B gives a proposal for a possible procedure to choose rooms.

6.2 Calculation method

The method used to perform the room temperature calculation shall comply with the requirements given in EN ISO 13792. Compliance with EN ISO 13791 can be decided on a national basis.

6.3 Boundary conditions

6.3.1 Climatic data

The climatic data used for the room temperature calculation shall be calculated and presented according to prEN ISO 15927-2 or EN ISO 15927-4. The choice of the type of data and the duration of the period considered is defined on a national basis.

6.3.2 Internal loads

The assumptions on internal loads shall be determined on a national basis. The following values shall be considered on an hourly basis for the whole period considered:

— Lighting.

— Occupants.

— Equipment.

In the absence of prescribed values they shall be agreed with the customer.

6.3.3 Window opening

The effect of openable windows shall be calculated according to EN 15242.

6.3.4 Acceptable comfort conditions

The level of temperatures considered as acceptable and the tolerable frequency of exceeding are defined on a national basis. Requirements are given in EN 15251.

7 Room cooling load calculation

7.1 Basic sensible room cooling load calculation

The basic room cooling load calculation is performed under constant comfort conditions for a convective system with unlimited maximum cooling power. The used calculation method shall fulfil the requirements of class A1 defined in EN 15255.

7.2 System dependent sensible room cooling load calculation

The room cooling load calculation shall be performed by using a method which fulfils the requirements of the appropriate class according to EN 15255:2007, Tables 4 and 5.

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7.3 Latent room cooling load calculation

In the most general case, the room based latent load is determined by a full moisture balance of the room, including effects of moisture transport in building components and furniture. In this case the calculation is equal to the calculation of the room humidification and dehumidification energy demand for a certain time step, described in 13.2.

A simplified approach is to neglect storage effects in building components and furniture, which leads to a pure room air moisture balance. This depends on:

— The water mass entering and leaving the room (input)

— The humidity internal gains water (input)

A part of the load may be covered by room based equipment. This can be taken into account according to Annex H.

7.4 Boundary conditions

7.4.1 Definition of room conditions (temperature, humidity, tolerances)

Assumptions on room conditions are defined on a national basis in accordance with EN 15251.

7.4.2 Climatic data

The climatic data used for the cooling load calculation shall be calculated and presented according to prEN ISO 15927-2.

7.4.3 Internal loads

General requirements on information on internal loads are given in EN 13779.

7.4.4 Ventilation rates

The air flow rates for ventilation purposes are defined according to EN 15242.

8 Room heating load calculation

8.1 Calculation procedure

The same calculation procedure as for the system dependent room cooling load calculation shall be used.

8.2 Climatic data

The climatic data used for the cooling load calculation shall be calculated and presented according to EN ISO 15927-4.

8.3 Ventilation rates

The air flow rates for ventilation purposes are defined according to EN 15242.

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9 Room based equipment sizing

Room based equipment used to meet the room heating and cooling load (fan coils, chilled ceilings, split chillers etc., but also supply air flow rates and conditions) shall be sized according to common engineering knowledge and manufacturer's information. The influence of the equipment size on the system energy consumption, e.g. due to operation conditions and/or part load behaviour, shall be considered.

10 Zone load calculation

The zone cooling and heating load is found by superposition of the daily profiles of the room cooling and heating loads. It is not adequate to add up non coincident hourly peak load values.

11 System heating and cooling load calculation

No specific method can be given for the system heating and cooling load. It is highly dependent on the chosen system. It shall be performed according to common engineering rules, taking into consideration all the relevant processes. For air based systems this generally involves psychrometric calculations.

12 Central system equipment sizing

Central equipment used to meet the system heating and cooling load (air handling units, chillers etc.) shall be sized according to common engineering knowledge and manufacturer's information. The influence of the equipment size on the system energy consumption, e.g. due to operation conditions and/or part load behaviour, shall be considered.

13 Room and building energy calculation

13.1 General

The calculation of cooling and heating energy demand within buildings is described in prEN ISO 13790. The result of this calculation is an input for the system energy demand calculation described in Clause 14. Building related issues are only addressed to the extent that HVAC systems have an influence on the building energy demand.

13.2 Humidification and dehumidification energy demand

In the most general case, the room based latent energy demand in a certain time step is determined by a full moisture balance of the room, including effects of moisture transport in building components and furniture. This is a complex calculation, the mechanisms of which are described in EN 15026. In many cases, the calculation of the room latent load can be simplified. Two approaches are possible:

— The storage effect in building components and furniture is neglected. The latent load is then determined by a pure room air moisture balance, This leads to an overestimation of the room and exhaust air humidity. The humidity balance depends on:

  — The water mass entering and leaving the room (input)

  — The humidity internal gains water (input)

  — The condensation on the coil of room based equipment (output)

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Energy demand due to humidification and dehumidification for local emitters may occur intentionally or as a side effect of cooling. A calculation method is given in Annex H.

— The storage effect in building components and furniture as well as the moisture sources in the room are neglected and the moisture content of the exhaust air is assumed to be equal to the one of the supply air. This leads to an underestimation of the room and exhaust air humidity and can be desired in cases with equipment, the performance of which depends on the exhaust air humidity, and for which the simplified calculation approach shall be on the safe side.

13.3 Relation to system energy calculation methods

According to prEN ISO 13790, building energy demand calculation methods are divided into detailed and simplified categories.

System behaviour calculation methods can be divided in hourly, monthly, seasonal, and annual categories. The main distinction is between hourly methods and methods using larger time steps. Table 3 shows a classification of combinations of calculation methods.

Table 3 — Classification of building vs system calculation methods

<table>
<tr><td></td><td></td><th colspan="2">System calculation</th></tr>
<tr><td></td><td></td><td>Hourly</td><td>Monthly, seasonal, annual</td></tr>
<tr><td rowspan="2">Building Calculation</td><td>Hourly</td><td>BhSh</td><td>BhSm</td></tr>
<tr><td>Monthly, seasonal</td><td>BmSh</td><td>BmSm</td></tr>
</table>

Type BhSh

This configuration makes it possible to take into account hourly interactions between building behaviour and system behaviour. It is the case for example for VAV system where the air flow depends on the cooling demand, or for latent load which are difficult to calculate on a monthly basis.

Type BmSh

In this case the system behaviour is calculated based on hourly values before the building calculation is performed. This can be done when the system behaviour is not dependent on the building behaviour. It can be for example for systems reacting mainly to outdoor climate (for example combination of outdoor air temperature and humidity). Some assumptions can also be used e.g. relating for example indoor temperature to outdoor temperatures.

Type BmSm and BhSm

In this case the system behaviour is calculated by averaged monthly, seasonal or annual values, using in general statistical analysis based on hourly calculation for typical climates, configurations etc.. It can also be done directly if the system is simple enough to neglect the interaction with both outdoor climate and the building behaviour.

A commonly used category of system calculation method is based on the frequency distribution of hourly outdoor air temperature and/or humidities. This can be combined with either hourly or monthly/seasonal building calculations, thus belong to category BhSh or BmSh.

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14 System energy calculation

14.1 General approach

14.1.1 System structure and boundaries

The general structure of the system and calculation is given in Figure 3. Heating and cooling systems have in general a parallel structure. Heating and cooling function may be combined in emitter and distribution components (ducts, fan coil units) and also in generation (split systems for heating and cooling; reversible cooling machines). In most systems distribution of heat and cold is performed by both water and air. Heat and cold are transferred from water to air using heat exchangers in the air conditioning plant or at the emitters, using for instance induction units, terminal reheat units or radiators. The structure is worked out in more detail for different systems in 14.1.2.

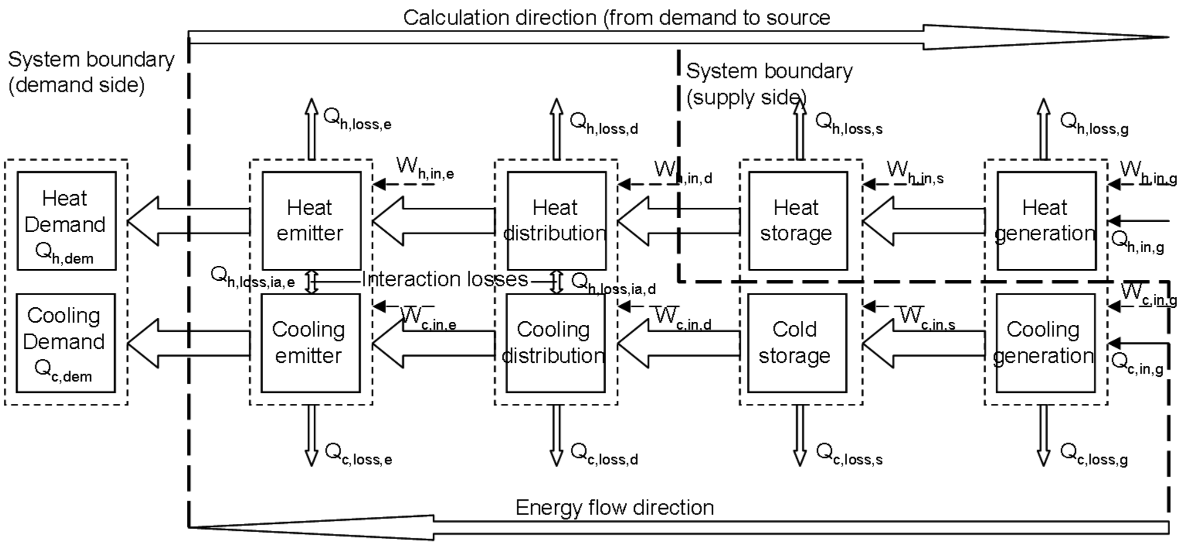

Figure 3 — General HVAC system structure and energy flows The system boundary includes contributions from EN 15241 in case of all air systems, see 14.2.4.2

System boundaries for the scope of this standard are indicated in the figure. For the demand side the system boundary is between the emitters and the demand; for cooling the system boundary is at the right side of the cooling generation. For heating the system boundary may be at different places, depending on system type. Heat generation and storage are always outside the system boundary (indicated). Also hot water distribution and emitter (radiator) components are outside the system boundary. Only the air-conditioning emitter and distribution components are within the system boundary. For practical reasons also interaction losses between all heat and cold distribution and emitter components are within the scope of the standard. In Annex C boundary layers are worked out in more detail.

14.1.2 Energy calculation structure

Each sub-system may have losses and auxiliary power use. Heat and cold generators consume fuel or power. Emitter and distribution components may also have interaction losses, due to simultaneous heating and cooling, mixing of heat and cold flows, etc.

Over a given period the delivered energy $Q_{h,in,g}$ for heating and $Q_{c,in,g}$ for cooling is given by:

$$Q_{h,in,g} = Q_{h,dem} + Q_{h,loss,e} + Q_{h,loss,ia,e} + Q_{h,loss,d} + Q_{h,loss,ia,d} + Q_{h,loss,s} + Q_{h,loss,g} \quad (1)$$

and:

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$$Q_{\mathrm{c,g,in}} = Q_{\mathrm{c,dem}} + Q_{\mathrm{c,loss,e}} + Q_{\mathrm{c,loss,ia,e}} + Q_{\mathrm{c,loss,d}} + Q_{\mathrm{c,loss,ia,d}} + Q_{\mathrm{c,loss,s}} + Q_{\mathrm{c,loss,g}} \quad (2)$$

In some installations, different energyware may be used for heating or cooling generation, for example:

— Heating installation using electricity for a compressor heat pump and gas for additional boilers.

— Cooling installation using gas for a sorption cooler and electricity for a compressor cooler.

In these cases the delivered energy shall be calculated separately for the different energywares, for example $Q_{\mathrm{h,in,g}}(E)$, $Q_{\mathrm{h,in,g}}(G)$, $Q_{\mathrm{h,in,g}}(O)$ for delivered energy using electricity, gas or oil.

Total auxiliary electricity consumption $W_{\mathrm{h,in,tot}}$ for heating and $W_{\mathrm{c,in,tot}}$ for cooling is given by:

$$W_{\mathrm{h,in,tot}} = W_{\mathrm{h,in,e}} + W_{\mathrm{h,in,d}} + W_{\mathrm{h,in,s}} + W_{\mathrm{h,in,g}}$$

and:

$$W_{\mathrm{c,in,tot}} = W_{\mathrm{c,in,e}} + W_{\mathrm{c,in,d}} + W_{\mathrm{c,in,s}} + W_{\mathrm{c,in,g}}$$

where:

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$Q_{h,in,g}$	delivered energy (fuel consumption) for heating in Joule
$Q_{h,dem}$	building heat demand in Joule
$Q_{h,loss,e}$	emitter heat loss in Joule
$Q_{h,loss,ia,e}$	emitter heat loss, due to interaction with cooling system, in Joule
$Q_{h,loss,d}$	distribution heat loss in Joule
$Q_{h,loss,ia,d}$	distribution heat loss, due to interaction with cooling system, in Joule
$Q_{h,loss,s}$	storage heat loss in Joule
$Q_{h,loss,g}$	generation heat loss in Joule
$Q_{c,in,g}$	delivered energy (fuel consumption) for cooling in Joule
$Q_{c,dem}$	building cooling demand in Joule
$Q_{c,loss,e}$	emitter cold loss in Joule
$Q_{c,loss,ia,e}$	emitter cold loss, due to interaction with heating system, in Joule
$Q_{c,loss,d}$	distribution cold loss in Joule
$Q_{c,loss,ia,d}$	distribution cold loss, due to interaction with heating system, in Joule
$Q_{c,loss,s}$	storage cold loss in Joule
$Q_{c,loss,g}$	generation cold loss in Joule
$W_{h,in,tot}$	total auxiliary energy consumption for heating in Joule
$W_{h,in,e}$	heat emitter auxiliary energy consumption in Joule
$W_{h,in,d}$	heat distribution auxiliary energy consumption in Joule
$W_{h,in,s}$	heat storage auxiliary energy consumption in Joule
$W_{h,in,g}$	heat generation auxiliary energy consumption in Joule
$W_{c,in,tot}$	total auxiliary energy consumption for cooling in Joule
$W_{c,in,e}$	cooling emitter auxiliary energy consumption in Joule
$W_{c,in,d}$	cooling distribution auxiliary energy consumption in Joule
$W_{c,in,s}$	cold storage auxiliary energy consumption in Joule
$W_{c,in,g}$	cooling generation auxiliary energy consumption in Joule

In Figure 4 the basic energy flow scheme for a subsystem is given, including the energy flow symbols.

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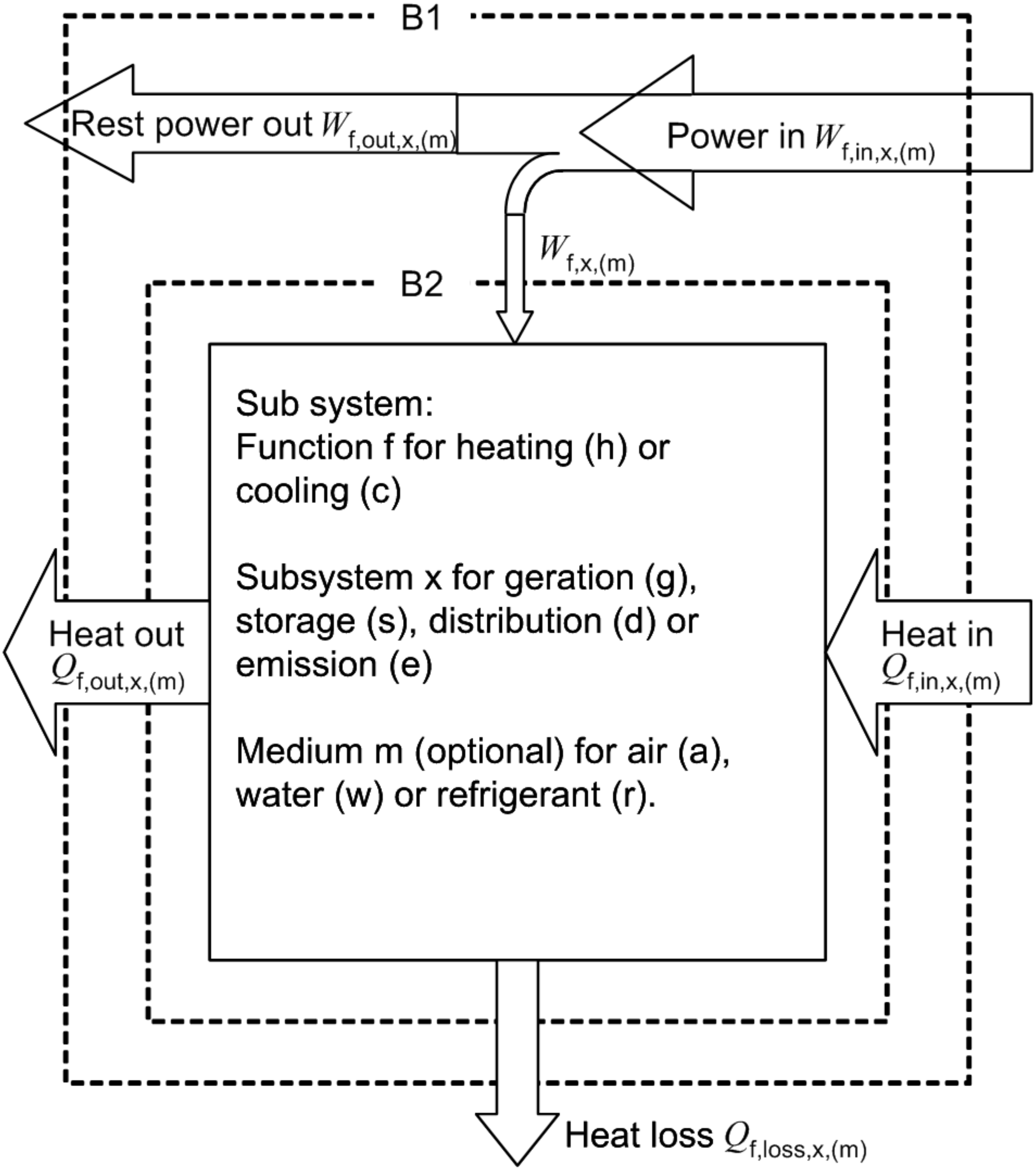

Figure 4 — Subsystem energy flows

This information can be passed to prEN 15203 and prEN 15315 for conversion to primary energy and CO_2 emission and for assessment or further calculations.

14.1.3 Calculation methods

The system energy calculation can be done simplified or detailed.

— Simplified or implicit methods operate with longer periods of time – typically monthly or daily – and use parametric descriptions of HVAC system operation, rather than represent the actions of

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HVAC systems directly. As a result, they generally require fewer calculations to be carried out when applied to a building. However, the performance of many air-conditioning (and some heating) systems is non-linear or discontinuous with load or outdoor conditions. In these situations, the development of adequate parametric descriptions may be a considerable task in the development of a simplified or implicit procedure.

— Detailed or explicit methods aim to provide algorithms that directly reflect the timestep by timestep operation of each important component and each important mechanism within the system. Commonly, an energy calculation is carried out using hourly timesteps that represent a complete year[1]. Procedures that represent a year by a lesser number of days, but carry out an explicit calculation for those days are here considered to be detailed or explicit methods.

This standard allows different routes to determine system energy performance:

— For both simplified and detailed methods an overview of required functionality of calculation methods is given.
Calculation methods require a kind of verification to show the required functionality is available. In Annex F the EDR-method is described that might be used for this verification.
Calculation methods satisfying this functionality required by this standard may also be used for other purposes, (for example, for system sizing). They are the content of the Clause 1 to 12.
Users of calculation methods should exercise care in ensuring that the need for appropriate modifications are considered and, if necessary, implemented for other applications.

— For the simplified method several calculation procedures for energy consumption and system losses are given, requiring different levels of knowledge about the installation parameters. Alternative methods are allowed.

— For the detailed method little calculation procedures can be given due to the fact that there is a large variety of systems and variations. This clause now only contains procedures to asses the effect of the emitter on the sensible and latent load.

Before going through these clauses first a system overview is given below to illustrate the basic structure and system boundaries for different HVAC systems.

14.1.4 HVAC System Overview

In Table 4a an overview of HVAC systems is given. In Table 5 these systems are described in more detail, both in a function table and in system schemes, including an indication of system boundaries for the components treated in this standard. Table 4b gives an overview of ventilation-only systems. The calculation of the latter is covered by EN 15241.

[1] Many air-conditioning (and heating) system components have characteristic timescales shorter than this so, strictly speaking, hourly timesteps include some generalizations that we characterize as implicit.

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Table 4a — HVAC system overview

Code	System name	**Does not always include heating provision and may be used with separate heating system** (including room heaters which may be within terminals)	Does not include integral provision of ventilation and may be used with separate (cooled) ventilation system
A	**All-air systems**		
A1	Single duct system (including multi-zone)		
A2	Dual duct system		
A3	Single duct, Terminal reheat		
A4	Constant Volume (with separate heating)	✔	
A5	Variable Air Volume (with separate heating)	✔	
B	**Water-based systems**		
B1	Fan coil system, 2-pipe	✔	✔
B2	Fan coil system, 3-pipe		✔
B3	Fan coil system, 4-pipe		✔
B4	Induction system, 2-pipe non change over	✔	
B5	Induction system, 2-pipe change over		
B6	Induction system, 3-pipe		
B7	Induction system, 4-pipe		
B8	Two-pipe radiant cooling panels (including chilled ceilings and passive chilled beams)	✔	✔
B9	Four-pipe radiant cooling panels (including chilled ceilings and passive chilled beams)		✔
B10	Embedded cooling system (floors, walls or ceilings)		✔
B11	Active beam ceiling system	✔	✔
B12	Heat pump loop system	□	✔ □
C	**Packaged Air Conditioning Units**		
C1	Room units (including single duct units)	✔	✔
C2	Direct expansion single split system	✔	✔
C3	Direct expansion multi split system (including variable refrigerant flow systems)	✔	✔

Some systems usually provide heating or ventilation within an integrated system. Others (identified in the columns) normally require separate provision of these services.

In some cases (notably A5, B4 and C1) the separate heating may take the form of electric heaters within room terminals.

Where ventilation is provided separately, this is considered in this standard as an additional HVAC system and a calculation procedure shall be capable of handling both types of system (and multi-system buildings) satisfactorily.

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Table 4b - Ventilation Systems Overview

Code	System name	Heat recovery possible from exhaust air	Use of regenerative energy possible[a]	Air heating possible
E	**Exhaust air system (exhaust air fan assisted)**			
E1	Central fan [b] without heat pump			
E2	Central fan [b] with heat pump	✔		
E3	Single room[c] fan without heat pump			
BAL	**Balanced system (supply air and exhaust air fan assisted)**			
BAL1	Central system [b] without heat recovery		✔	✔
BAL2	Central system [b] with heat exchanger	✔	✔	✔
BAL3	Central system [b] with heat pump	✔	✔	✔
BAL4	Central system [b] with heat exchanger and heat pump	✔	✔	✔
BAL5	Single room[c] unit without heat recovery			✔
BAL6	Single room[c] unit with heat exchanger	✔		✔
S	**Supply air system (supply air fan assisted)**			
S1	Central system[b] without regenerative energy			✔
S2	Central system[b] with regenerative energy		✔	✔
S3	Single room unit[c] without regenerative energy			✔

[a] for example ground to air heat exchanger or solar air collector

[b] systems for dwellings or residential buildings

[c] fans or units for single rooms

14.2 Required functionality of detailed and simplified calculation methods

14.2.1 General

In this clause the required functionality for both detailed and simplified calculation methods including the heating and cooling demand calculation are discussed. General principles and requirements are listed in 14.2.2 . The following section, 14.2.3 :addresses verification of procedures and 14.2.4 summarises a number of simplified procedures for handling some of the mechanisms that need to be addressed. These are described in more detail in informative annexes or in other standards.

14.2.2 General principles and reporting of procedure

— The system is modelled using models for all subsystems. In Figure 4 the basic energy flow scheme for a subsystem is given. The basic system structures are given in the “system overview”.

— Using a detailed method requires semi-dynamic models, often static component models assuming average values for all variables like temperatures and flows. In reality the variable values and controls may be more dynamic so in the model losses due to these dynamics should be taken care of.

— Table 5 shows which mechanisms are important for which HVAC system types. Energy calculation methods should address all the mechanisms that are relevant to the system types being considered. The documentation accompanying each calculation method shall report how each mechanism is represented. Different degrees of calculation complexity will be appropriate for different applications. Annex D contains recommendations for appropriate levels of treatment and refers to other relevant annexes and standards.

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14.2.3 Verification of building and HVAC system calculation methods

14.2.3.1 General

This section describes a procedure for “verifying” whether or not a particular calculation procedure is appropriate to a particular application. In effect, it is a process of sequential “filtering” of options. Calculation methods can be eliminated if they fail to meet particular criteria. Surviving the process does not guarantee satisfactory performance under all possible conditions, although it does reduce the likelihood of unsatisfactory performance.

If sufficient reference data are available the Energy Diagnosis Reference (EDR) described in Annex F (informative) can be used.

14.2.3.2 Background

A vast number of calculation methods for predicting the energy performance of buildings is available. They differ in scope, in complexity of calculation and in the quantity of input data required. In practice, even methods of similar scope and complexity often yield different results even when identical input data are used.

At a national level, a single calculation method may be demanded. In this case the procedure can be used as part of the selection process for that method. Alternatively, a number of different methods may be acceptable, in which case the procedure can be used as part of the acceptance procedure.

This standard covers only calculations relating to HVAC systems. In practice, integrated methods that combine calculation of building energy demand and HVAC system energy consumption will often be used. The principles described below are equally applicable in this case.

For the Energy Performance Certification of Buildings, key requirements are:

- Credibility. The mechanisms should be technically sound and the results of the calculation shall be plausible.
- Discrimination. The rating should be better for systems and buildings that are clearly more energy-efficient.
- Repeatability. Different users should get essentially the same results for the same building.
- Transparency. It should be straightforward to check the data used and the execution of the procedure.

For design purposes – and especially for system sizing, over-heating risk and high-cost decisions – additional factors come into play. In particular the implications of uncertainty and any bias in the results will need to be considered in conjunction with a risk assessment and the selection of appropriate safety margins.

14.2.3.3 Procedure

Step 1. Define required characteristics and check that the method appears to provide them

The first step is to define the necessary and desirable characteristics of the calculation methods. This includes requirements related to:

- Output: for example, whether both heating and cooling consumptions are required; whether the output is annual energy demand or design load (and under what conditions).
- Scope: for example, what types of system can be covered.
- Applicability: for example the requirements for Energy Performance Certification noted above.

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Different priorities may exist for different classes of building. The Dutch EDR process illustrates one way of structuring this aspect of the process.

Possible calculation methods should be inspected to check that they do not lack any of these essential features. This check can be carried out by inspection of user guides, theoretical background papers etc but it is preferable to also apply each method to a few simple buildings. It is difficult to assess many of the non-technical requirements without such initial application testing. At this step, it is not usually possible to test accuracy – though the cause of obviously improbable results should be investigated.

Table 5 lists important technical features that affect the energy consumption of different types of HVAC system. Some calculation procedures may not adequately handle all these features, in which case their application should be restricted to applications that exclude the possible use of systems for which the absent features are important.

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Table 5a — Important technical features that affect the energy consumption of different types of HVAC system

Mechanism	System type																			
	All-air systems					Water-based systems												Packaged Air Conditoners		
	A1	A2	A3	A4	A5	B1	B2	B3	B4	B5	B6	B7	B8	B9	B10	B11	B12	C1	C2	C3
Within-room mechanisms																				
Room heat balance and temperature	Part of building demand calculation – not HVAC system calculation																			
Room moisture balance and moisture content	Part of building demand calculation – not HVAC system calculation																			
Definition of zones and ability to combine room demands into zonal demands	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔		✔	✔
Combination of room conditions into zonal return air state	✔	✔	✔	✔	✔				✔	✔	✔	✔								
Contribution to room demands from separate ventilation / base cooling system						✔	✔	✔					✔	✔	✔	✔	✔	✔	✔	✔
Contribution to room demands from heat gains or losses from pipes and ducts	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔				
Impact of proportional band on energy supplied	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔
Impact of dead band on energy suppplied	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔

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Mechanism	**System type**																			
	All-air systems					**Water-based systems**												**Packaged Air Conditoners**		
	A1	**A2**	**A3**	**A4**	**A5**	**B1**	**B2**	**B3**	**B4**	**B5**	**B6**	**B7**	**B8**	**B9**	**B10**	**B11**	**B12**	**C1**	**C2**	**C3**
Effect of open-loop control or averaging of sensors	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔		✔	✔
Effect of absence of interlock between heating and cooling					✔		✔		✔				✔		✔	✔		✔	✔	✔
Distribution: terminal issues																				
Energy penalties from hot/cold mixing or reheat systems		✔	✔		✔	✔		✔	✔		✔									
Terminal auxiliary energy.					✔	✔	✔	✔								✔	✔	✔	✔	✔
Effect of sensible heat ratio of terminal *(and risk of condensation)*						✔	✔	✔					✔	✔	✔	✔	✔	✔	✔	✔
Lack of local time control	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔				✔
Heat gains and losses from pipes and ducts *Includes AHUs and other air-handling components*	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔			

Mechanism	**System type**																			
	All-air systems					**Water-based systems**												**Packaged Air Conditoners**		
	A1	**A2**	**A3**	**A4**	**A5**	**B1**	**B2**	**B3**	**B4**	**B5**	**B6**	**B7**	**B8**	**B9**	**B10**	**B11**	**B12**	**C1**	**C2**	**C3**
Duct system air leakage *Includes AHUs and other air-handling components*	▯	✔	✔	✔	✔				✔	✔	✔	✔								
Refrigerant pipework heat losses																	✔	✔	✔	✔
Fan and pump energy pickup	✔▯	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔			
Heat recovery provision	✔▯	✔	✔	✔	✔															
Distribution systems: operation																				
Latent demand calculation at central (zonal) plant *(includes dewpoint control plus reheat)*	✔▯	✔	✔	✔	✔															
Adiabatic spray cooling	✔▯	✔	✔	✔	✔															
Additional demands produced by hot deck:cold deck mixing systems	✔▯	✔																		
Impact of mixing of return water temperature in 3-pipe systems							✔				✔									

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Mechanism	**System type**																			
	All-air systems					**Water-based systems**												**Packaged Air Conditoners**		
	A1	**A2**	**A3**	**A4**	**A5**	**B1**	**B2**	**B3**	**B4**	**B5**	**B6**	**B7**	**B8**	**B9**	**B10**	**B11**	**B12**	**C1**	**C2**	**C3**
Wastage due to changeover in 2-pipe systems						✔				✔			✔							
Impact of variable ventilation air recirculation *Typically CO2 controlled – total air flow unchanged*	✔	✔	✔	✔	✔															
Impact of air-side free cooling	✔	✔	✔	✔	✔															
Distribution systems: auxiliary energy																				
Auxiliary energy use by fans and pumps (other than in terminals)	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔			
Cold and heat generation																				
Cold generator (chiller) part-load performance (including multiple installations and over-sizing)	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔
Water-side free cooling						✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔			
Thermosyphon operation	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔			

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Mechanism	**System type**																			
	All-air systems					**Water-based systems**												**Packaged Air Conditoners**		
	A1	**A2**	**A3**	**A4**	**A5**	**B1**	**B2**	**B3**	**B4**	**B5**	**B6**	**B7**	**B8**	**B9**	**B10**	**B11**	**B12**	**C1**	**C2**	**C3**
Impact on chiller performance of central heat rejection equipment *Includes cooling towers, dry coolers etc. Included in overall performance of packaged systems*	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔			
Auxiliary energy use by central heat rejection equipment *Included in overall performance of packaged systems*	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔			
Heat generator (boiler) part-load performance (including multiple installations and over-sizing)	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔
Auxiliary energy use by heat generators *Includes gas boosters, fuel pumps etc Included in overall performance of packaged systems*	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔			
Energy use for humidification	✔	✔	✔	✔	✔															

Mechanism	System type																			
	All-air systems					Water-based systems												Packaged Air Conditoners		
	A1	A2	A3	A4	A5	B1	B2	B3	B4	B5	B6	B7	B8	B9	B10	B11	B12	C1	C2	C3
Bivalent systems *Includes boiler + CHP; condensing boiler + non-condensing boiler; heat pump + top-up; evaporative cooling + chiller…*	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔	✔

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Table 5b - Important technical features that effect the energy consumption of different types of ventilation systems

Mechanism	System type											
	Exhaust air system			Balanced system						Supply air system		
	E1	E2	E3	BAL1	BAL2	BAL3	BAL4	BAL5	BAL6	S1	S2	S3
Room heat balance and temperature	Part of building demand calculation											
Definition of zones and ability to combine room demands into zonal demands	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
Combination of room conditions into zonal return air state		✓			✓	✓	✓		✓			
Impact of heat recovery from exhaust air and use for supply air preheating					✓	✓	✓		✓			
Impact of use regenerative energy					✓	✓	✓				✓	
Impact of air heating				✓	✓	✓	✓	✓	✓	✓	✓	✓
Impact of demand-dependent air flow	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
Impact of control	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
Contribution to room demands from heat gains or losses from ducts	✓	✓		✓	✓	✓	✓			✓	✓	
Contribution to room demands from heat losses from storage		✓				✓	✓					
Contribution to room demands from heat gains or losses from generation and air heating		✓		✓	✓	✓	✓	✓	✓	✓	✓	✓
Auxiliary energy use by fans	✓	✓		✓	✓	✓	✓			✓	✓	
Heat gains and losses from ducts	✓	✓		✓	✓	✓	✓			✓	✓	
Duct system air leakage	✓	✓		✓	✓	✓	✓			✓	✓	
Auxiliary energy use by pumps		✓				✓	✓					
Heat losses from storage		✓				✓	✓					
Auxiliary energy use by fans and control	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
Energy use by heat pump		✓				✓	✓					
Energy use by air heating				✓	✓	✓	✓	✓	✓	✓	✓	✓
Energy use by defrosting operation					✓	✓	✓		✓			
Heat losses from heat pump		✓				✓	✓					
Heat losses from heating				✓	✓	✓	✓	✓	✓	✓	✓	✓
Heat pump part-load performance		✓				✓	✓					

Step 2. Verification of calculation accuracy

There are several techniques for verifying calculation methods, none of which can completely guarantee accuracy in all conditions. The main ones are shown in Table 6. The application of verification techniques is decided on a national basis. If a verification technique is used, the results obtained shall be stated.

Table 6 — Verification techniques

Technique	Method	Strengths	Weaknesses
Theory checking	Equations are checked	* Ensures that appropriate theory has been applied.	* Requires expert knowledge and judgement about appropriate theory
Code checking	Computer code is checked line by line	* Ensures that equations have been correctly implemented * Ensures that data are correctly processed	* Only possible by experts * Only possible for open programs *Slow and time consuming *Only practicable for simple algorithms
Analytic tests	Predictions for simple situations are compared with analytic solutions	* Tests correct processing by input interface, program, and output interface * Exact answer known, so absolute accuracy can be determined	* There are very few situations for which analytic solutions can be produced * Analytic solutions not possible for building-like spaces * May be difficult to isolate testable algorithms from program * Cannot test whole program
Inter-model comparisons	Prediction are compared with those from other methods supplied with equivalent data	* Relatively easy to undertake and, in principle, can cover any building situation * Can test whole program * By carefully sequencing tests, sensitivity to design changes can be compared	* Tests for consistency between models rather than for absolute accuracy. If one model is out of line with the remainder, this may indicate poor performance – alternatively, it may indicate more accurate modelling
Empirical validation	Prediction are compared to detailed measurements in real buildings (Predictions should be made without knowledge of actual performance – that is, blind)	* In principle, the most powerful validation technique as it compares prediction with reality	* Difficult to carry out convincingly because it is very difficult and time-consuming to obtain measurements of adequate accuracy. In practice, this means that data are from unoccupied simple buildings.
Statistical validation	Predictions for many buildings are compared to energy consumption measurements	* The most directly relevant verification for existing buildings	* There are many uncontrolled variables, especially in occupied buildings, so the verification is of a combination of calculation method and data set. * The results are specific to the combination of building type of the sample, * In practice, there are large variations that are not explained by the calculation method

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Depending on the application, verification may be in terms of quantitative values (typically energy consumptions) or in terms of a ranking of different situations.

At present it is not possible to provide a useful range of analytic, inter-model or empirical tests for complete HVAC systems.

14.2.4 Calculation procedures: Information in other standards

14.2.4.1 General

Calculation procedures for some of the mechanisms included in Table 5 are already addressed in other CEN Standards. Informative examples procedures for other mechanisms are contained in annexes to this standard.

14.2.4.2 All air systems

For air based systems not controlled to react on the room loads (such as constant volume flow rate systems or purely time dependent multi stage systems with outdoor reset supply temperature control) the calculation method is defined in EN 15241.

If the system is designed to be controlled according to the room/building heating/cooling demand (e.g. VAV systems or room dependent supply temperature control), the room/building load dependent variables are calculated from the room energy balance as follows.

Two possibilities of room or zone temperature control are considered:

1) air supply temperature control: A local coil situated in the supply air flow is controlled by the room sensor. In this case the air flow rate is defined at the AHU level. In the heating season, the airflow will be in general higher than the one required for hygienic purposes. A recirculation air ratio can then be associated to it in order to save energy.

2) air volume control: A local grille controls the air flow rate. In this case the supply air temperature is defined at the AHU level. The local supply air flow rate cannot be less than the one required for hygienic purposes, and will generally be higher. A procedure to calculate it is given in EN 15242:2007 (Annex C).

Once the heating demand or the cooling demand of the room is known:

For case 1: In a heating situation and as the local coil has no impact on the AHU behaviour, it is possible to consider the coil as a standard emitter directly situated in the room.

In a cooling situation and if the AHU is only devoted to one zone, the supply air set point temperature is modified according to the cooling demand. If not, the calculation method requires to be adapted.

For case 2: the local required air flow rate is calculated according to the cooling demand.

The calculation of all air side processes for the system energy requirements is then done according to EN 15241, taking into account the respective values for the room/building load dependent parameters.

In order to allocate the system energy demand to the purposes of a system, it may be adequate to make the distinction between the "basic system" which provides ventilation only, and a (fictitious) "peak system", which only covers the room conditioning purposes. In that case, the calculation is done twice: once for the "basic system" and once for the complete system as described above. The difference between the two is the "peak system" energy demand. An example of such an approach is in E.2.

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14.2.4.3 Separate heating systems

If the heating demand of the building is covered by a separate system independent from the cooling and ventilation system (e.g. radiator or floor heating system), the heating system calculation is performed according to EN 15316-2-1 or EN 15377-3.

14.2.4.4 Night ventilation (mechanical or natural)

The calculation of the air flow rates for natural ventilation is performed according to EN 15242. These are considered in the building energy demand calculations according to prEN ISO 13790.

14.2.4.5 Cooling systems embedded in construction components

The energy performance of the embedded system itself is defined by EN 15377-3. The distribution losses of the system connecting to and serving the embedded system can be treated according to Annex J and Annex K.

14.3 Simplified system losses and energy demand calculation methods

14.3.1 General remarks

In simplified methods component losses or efficiencies are given for each component or for groups of components. The calculation direction of Figure 3 is followed: emission, distribution, storage and generation. The methods presented here are not prescriptive but are optional. Other methods may be used if the required functionality is taken into account.

14.3.2 Emission losses

14.3.2.1 Basis of the method

The internal temperature is affected by:

— the spatial variation due to the stratification, depending on the emission system,

— the temporal variation depending on the capacity of the control device to assure a homogeneous and constant temperature.

The internal temperature θ_i, taking into account the emission system, is calculated by the following formula:

$$\theta_i = \theta_{ii} + \delta\theta_{vs} + \delta\theta_{vt}$$

With:

θ_{ii} initial internal temperature,

$\delta\theta_{vs}$ spatial variation of temperature,

$\delta\theta_{vt}$ control accuracy.

These internal temperatures are used in the calculations instead of the initial room temperature.

Remarks:

The spatial and temporal variations of the temperature can depend on the thermal load. In the absence of sufficient data, these variations are not taken into account in this version.

Example values for emission losses are given in Annex G.

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14.3.3 Emission auxiliary energy demand calculation

For some room based equipment, a calculation method for the auxiliary energy demand is described in Annex L.

14.3.4 Calculation of cold water distribution

A possible method for the calculation of distribution losses is given in Annex K.

The principles for the calculation of the auxiliary energy use by pumps etc. are described in Annex J.

14.3.5 Humidification and dehumidification energy demand

The energy demand for humidification and dehumidification occurring in the air handling unit is treated in EN 15241.

A calculation method for the auxiliary energy for humidification is given in J.4.3

14.3.6 Cold generation and chiller energy performance

The calculation of cold generator and chiller energy performance is treated in Annex I. A calculation method for the auxiliary energy demand of heat rejection equipment is given in Annex M.

14.3.7 Example calculation procedures

Different methods are possible. 3 examples are given in Annex E.

14.4 Detailed system losses and energy demand calculation method

14.4.1 General remarks

No complete method can yet be given due to the number of systems and variations. All methods shall take into account the functional requirements discussed in 14.2.

14.4.2 Climatic data

The climatic data used for the energy calculation shall be calculated and presented according to EN ISO 15927-4.

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Annex A
(informative)

Best procedure for design process

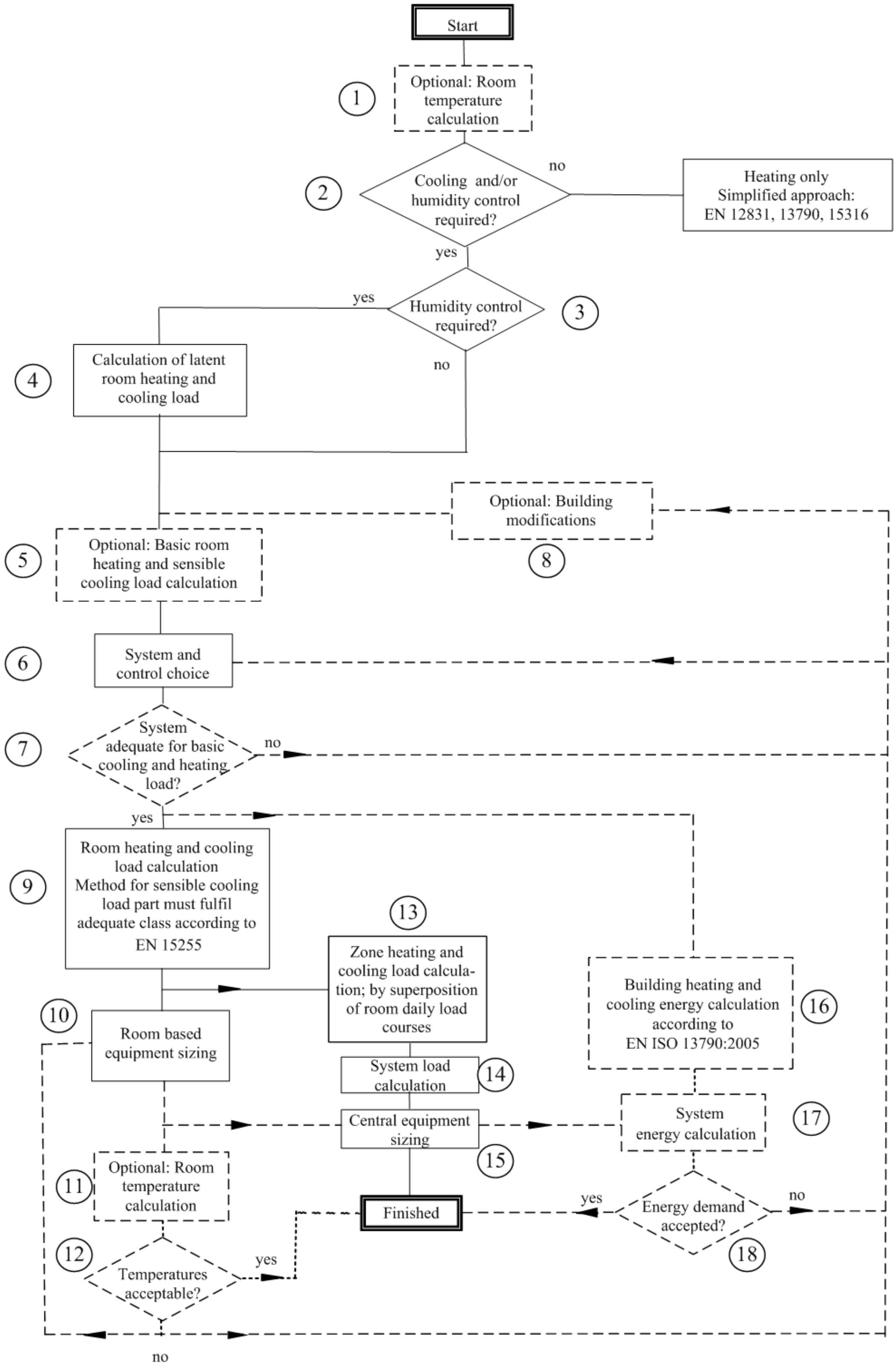

Figure A.1 — Flowchart with numbers for explanations

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The recommended "best procedure" involves the following steps, which are related to the numbers given in Figure A1.

1) Perform room temperature calculation according to Clause 6 for selected rooms.

2) Decide whether either cooling or humidity control or both are required

3) If 1 = "yes": Define range of humidity in rooms. Calculation of latent heating and cooling load according to 7.2

4) Perform basic cooling load calculation according to 7.2

5) Choose system type and control strategy.

6) Decide whether chosen system type and control strategy is adequate for basic load. If "no": ->8, or redo choice of system type and control (-> 6).

7) Propose modifications on building envelope and structure to responsible people, and redo steps 4 to 6

8) Perform system dependent heating and cooling load calculation according to 7.3

9) Size room based equipment: such as air volume flow rate, chilled ceiling power, fan coil power, radiator power, embedded system power etc.

10) Perform room temperature calculation in case of intentionally undersized equipment.

11) Decide whether temperatures acceptable. If "no" redo room based equipment sizing (12a -> 10) or propose building envelope and structure modifications (12b -> 8)

12) Perform zone load calculations according to Clause 10.

13) Perform system load calculation according to Clause 11.

14) Size central equipment, may be influenced by room based equipment sizing.

15) Perform building energy demand calculation according to Clause 13

16) Perform system energy demand calculation according to Clause 14

17) Decide whether energy demand acceptable. If "no": Propose building envelope and structure modifications (-> 8) or redo choice of system type and control (-> 6)

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Annex B
(informative)

Proposed procedure for choice or typical rooms for temperature calculation

— Divide the building into the lowest possible number of areas with spaces of the same or enough similar operation and construction, and with the same orientation;

— Pick from these areas those with a ratio of more than 10 % of the total building area for which air conditioning is envisaged;

— Pick from these areas all those with orientations in the sectors between E - SE and SW - W;

— If an area found is equal to a space, the modelled zone is this space;

— if an area consists of several spaces (the more frequent case), the modelled zone is the one of these spaces, which represents best the whole group of spaces.

— Areas with orientations out of the sectors defined above have to be treated in the same way,

if overheating occurs in the zones mentioned above,

if there is a group with harder operational conditions (higher internal gains).

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Annex C
(informative)

System overview

Index	System name	Heat/cold genera-tion	Air distribution system						Central air conditioning		Water distribution system	Central water conditioning		Emission		Decentral heat/cold generation & distribution
		central/ decentral	Number of supply ducts	Air quantity	Pres-sure	Velo-city	Cen-tral retour air	Air recir-cula-tion	Summer / cooling control	Winter / heating control	Type	Cold water / control	Warm water / control	Appa-ratus	Local control	
A	**All-air systems**															
A1	Single duct system (including multi-zone)	C	1	>Ventilation	HP/LP	H/L	yes	yes	Cooling, central control	Heating, central control	2 pipe ww	6-12?	Heating curve	Anemo-stat & radiator (opt)	Thermo-static (rad)	---
A2	Dual duct system	C	2	>Ventila-tion	HP/LP	H/L	yes	yes	Cooling (control?)	(Pre-) heating (control?)	2 pipe ww	6-12?	Heating curve	Anemo-stat & radiator (opt)	Thermostatic mixing chamber & thermostatic (rad)	---
A3	Single duct, Terminal reheat	C	1	>Ventila-tion	LP	L	yes	yes	Cooling, fixed setpoint.	Prehea-ting, fixed setpoint	2 pipe ww	---	Heating curve	Convec-tor	Thermo-static	---
A4	Constant Volume (with seperate heating)	C	1	>Ventila-tion	HP/LP	H/L	yes	yes	Cooling, fixed setpoint.	Prehea-ting, fixed setpoint	2 pipe ww	6-12?	Heating curve	VAV unit & radiator (opt)	Thermo-static (rad)	---
A5	Variable Air Volume (with seperate heating)	C	1	>Ventila-tion	HP/LP	H/L	yes	yes	Cooling, fixed setpoint.	Prehea-ting, fixed setpoint	2 pipe ww	6-12?	Heating curve	VAV unit & radiator (opt)	Flow & thermo-static (rad)	---
B	**Water-based systems**															
B1	Fan coil system, 2-pipe	C	1	Ventila-tion	LP	L	yes	Op-tion	Cooling, fixed setpoint.	Prehea-ting, fixed setpoint	2 pipe cw (S), ww (W)	6-12 ?	Heating curve	Fan coil unit	Thermo-static	---

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Index	System name	Heat/cold genera-tion	Air distribution system						Central air conditioning		Water distribution system	Central water conditioning		Emission		Decentral heat/cold generation & distribution
		central/ decentral	Number of supply ducts	Air quantity	Pres-sure	Velo-city	Cen-tral retour air	Air recir-cula-tion	Summer / cooling control	Winter / heating control	Type	Cold water / control	Warm water / control	Appa-ratus	Local control	
B2	Fan coil system, 3-pipe	C	1	Ventila-tion	LP	L	yes	Op-tion	Cooling, fixed setpoint.	Pre-heating, fixed setpoint	1 pipe ww, 1 pipe cw, common return ww/cw	6-12?	Heating curve	Fan coil unit	Thermo-static (2x)	---
B3	Fan coil system, 4-pipe	C	1	Ventila-tion	LP	L	yes	Op-tion	Cooling, fixed setpoint.	Pre-heating, fixed setpoint	2 pipe ww, 2 pipe cw	6-12?	Heating curve	Fan coil unit	Thermo-static (2x)	---
B4	Induction system, 2-pipe non change over	C	1	Ventila-tion	HP	H	yes	Op-tion	Pre-cooling, curve	Heating, heating curve	2 pipe cw	6-12?	Heating curve	Induction unit	Thermo-static	---
B5	Induction system, 2-pipe change over	C	1	Ventila-tion	HP	H	yes	Op-tion	Cooling, fixed setpoint.	Pre-heating, fixed setpoint.	2 pipe cw (S), ww (W)	6-12?	Heating curve	Induction unit	Thermo-static	---
B6	Induction system, 3-pipe	C	1	Ventila-tion	HP	H	yes	Op-tion	Cooling, fixed setpoint.	Pre-heating, fixed setpoint	1 pipe ww, 1 pipe cw, 1 common return ww/cw	6-12?	Heating curve	Induction unit	Thermo-static (2x)	---
B7	Induction system, 4-pipe	C	1	Ventila-tion	HP	H	yes	Op-tion	Cooling, fixed setpoint.	Pre-heating, fixed setpoint	2 pipe ww, 2 pipe cw	6-12?	Heating curve	Induction unit	Thermo-static (2x)	---
B8	Two-pipe radiant cooling panels (including chilled ceilings) and passive chilled beam	C	1	Ventila-tion	HP	H	yes	Op-tion	Cooling, fixed setpoint.	Pre-heating, fixed setpoint.	2 pipe cw (S), ww (W)	Cooling curve or fixed 6-12 for air cooling	Heating curve	Active ceiling for heating/ cooling	Thermo-static	---
B9	Four-pipe radiant cooling panels (including chilled ceilings) and passive chilled beams	C	1	Ventila-tion	HP	H	yes	Op-tion	Cooling, fixed setpoint.	Pre-heating, fixed setpoint.	2 pipe ww, 2 pipe cw	Cooling curve or fixed; 6-12 for air cooling?	Heating curve	Active ceiling for heating/cooling	Thermo-static (2x)	---

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Index	System name	Heat/cold genera-tion	Air distribution system						Central air conditioning		Water distribution system	Central water conditioning		Emission		Decentral heat/cold generation & distribution
		central/ decentral	Number of supply ducts	Air quantity	Pres-sure	Velo-city	Cen-tral retour air	Air recir-cula-tion	Summer / cooling control	Winter / heating control	Type	Cold water / control	Warm water / control	Appa-ratus	Local control	
B 10	Embedded systems	C	1	Ventila-tion	LP	L	yes	no	Air: precooling, fixed setpoint; Water: central control	Air: preheating, fixed setpoint; Water: central control	2 pipe cw/hw	Cooling curve or intermit-tent	Heating curve	Embed-ded pipes for heating/ cooling	no	---
B 11	Active beam ceiling system	C	1	Ventila-tion	LP	L	yes	no	Air: precooling, fixed setpoint; Water: central control	Air: preheating, fixed setpoint; Water: central control	2 pipe cw/hw	Cooling curve or intermit-tent	Heating curve	Active ceiling for heating/ cooling plus fan	Thermo-static	---
B12	Heat pump loop system	C / D	1	Ventila-tion	LP	L	yes	no	Pre-cooling, fixed setpoint.	Pre-heating, fixed setpoint.	---	6-12?	Heating curve	Anemo-stat & DX internal unit	Thermo-static control of DX unit	rev. heat pump with water distribution system
C	**Packaged Air Conditioners**															
C1	Room units & single duct ventilation	C / D	1	Ventila-tion	LP	L	yes	no	Pre-cooling, fixed setpoint.	Pre-heating, fixed setpoint.	---	6-12?	Heating curve	Anemostat & packaged unit	Thermo-static control of DX unit	Room unit with chiller or rev. heat pump
C2	Single split system & single duct ventilation	C / D	1	Ventila-tion	LP	L	yes	no	Pre-cooling, fixed setpoint.	Pre-heating, fixed setpoint.	---	6-12?	Heating curve	Anemostat & DX-internal unit	Thermo-static control of DX unit	DX exter-nal unit (rev. heat pump) & coolant pipes
C3	Multi-split system & single duct ventilation	C / D	1	Ventil-ation	LP	L	yes	no	Pre-cooling, fixed setpoint.	Pre-heating, fixed setpoint.	---	6-12?	Heating curve	Anemo-stat & DX internal unit	Thermsotatic control of DX unit	DX exter-nal unit (rev. heat pump) & coolant pipes

HVAC system structures are worked out below with distinction between water and air energy flows and an indication of system boundaries for the system components treated in this standard. Please note that a *functional* distinction between heat and cold distribution by water doesn't imply a *physical* distinction, as for instance in 2- and 3-pipe systems. The calculation of many of the shown functions are treated in other standards, especially in EN 15241 for all central air handling equipment.

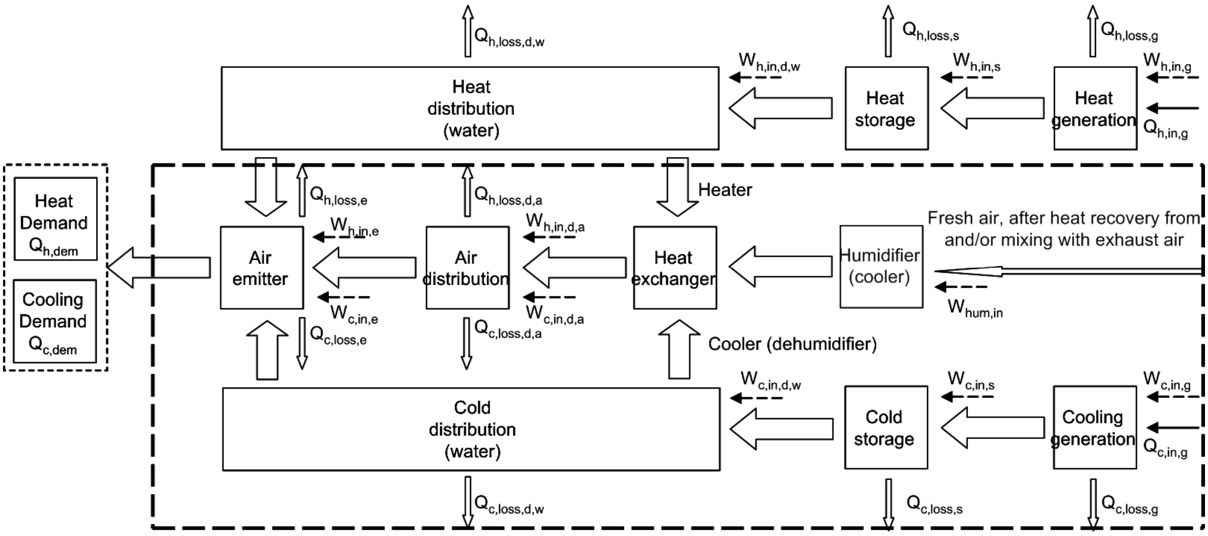

Figure C.1 — Fan coil and induction, 2-pipe - change over, 3- or 4-pipe systems

Many traditional air-conditioning systems have both pre-conditioning of the air in the air conditioning plant and final conditioning at the emitters, using fan coil units or induction units. Fan coil and induction, 3- and 4-pipe systems allow simultaneous final heating in one room and cooling in another room. 3-pipe systems suffer from heat/cold "destruction" due to mixing return flows of heat and cold water. 2-pipe induction - change over system allows either heating or cooling. In all these systems interaction losses may occur.

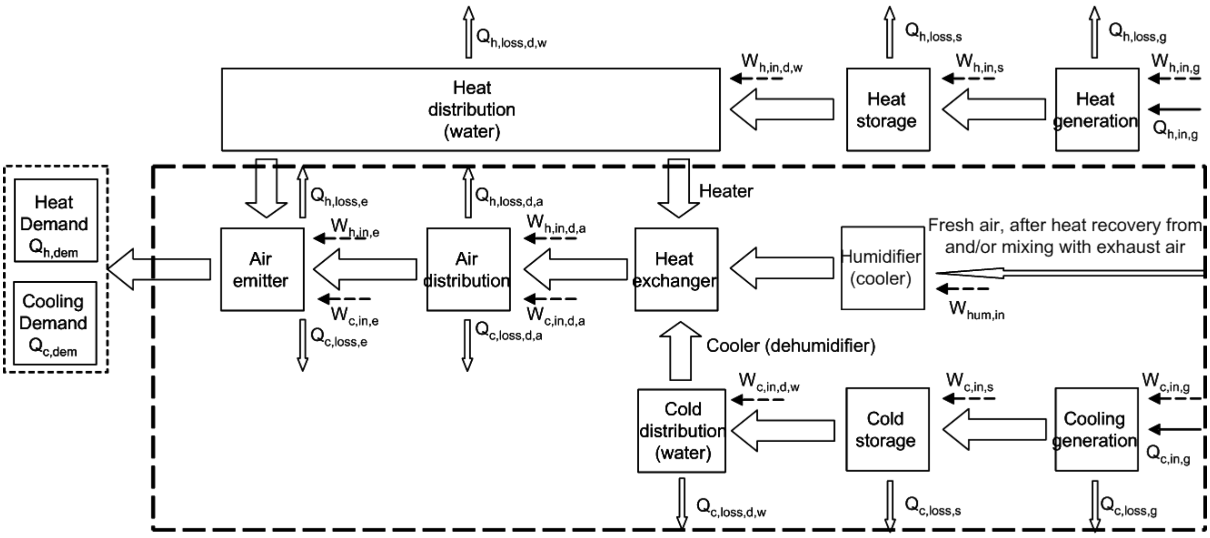

Figure C.2 — Terminal reheat

The terminal reheat system allows only final heating. Here also interaction losses may occur due to pre-cooling and final heating simultaneous.

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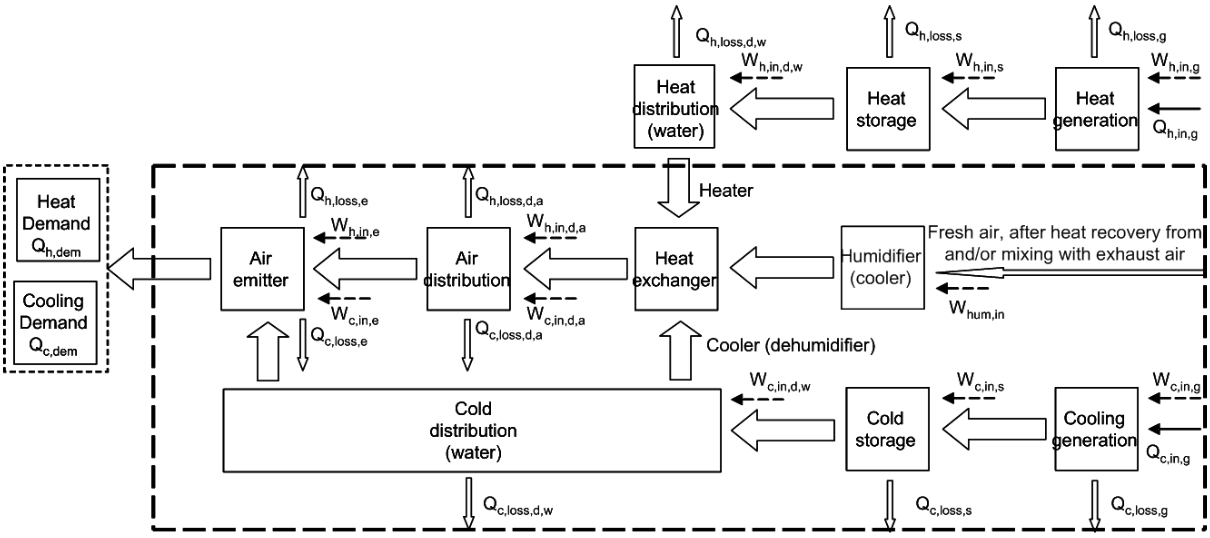

Figure C.3 — Induction 2-pipe - non change over system

The induction 2-pipe - non change over system follows a reverse approach to terminal reheat and allows only final cooling. Here also interaction losses may occur due to pre-heating and final cooling simultaneous.

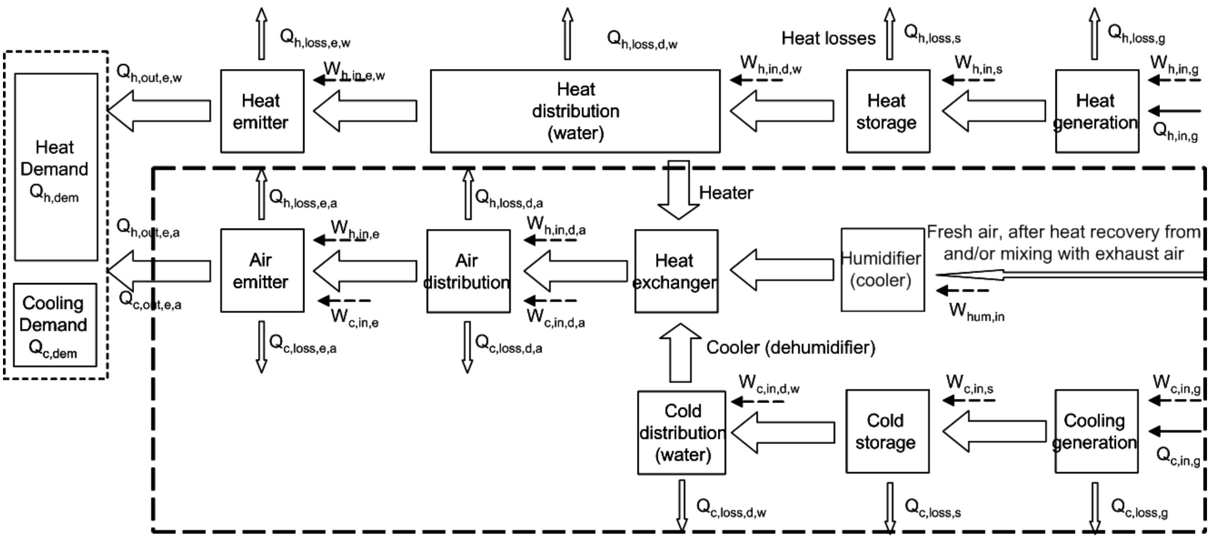

Figure C.4 — Single duct, Constant Volume or Variable air volume systems

Single duct, constant volume and VAV systems are pre-conditioned all-air systems with optional radiators for final heating. VAV systems allow some room-control using flow (volume) control.

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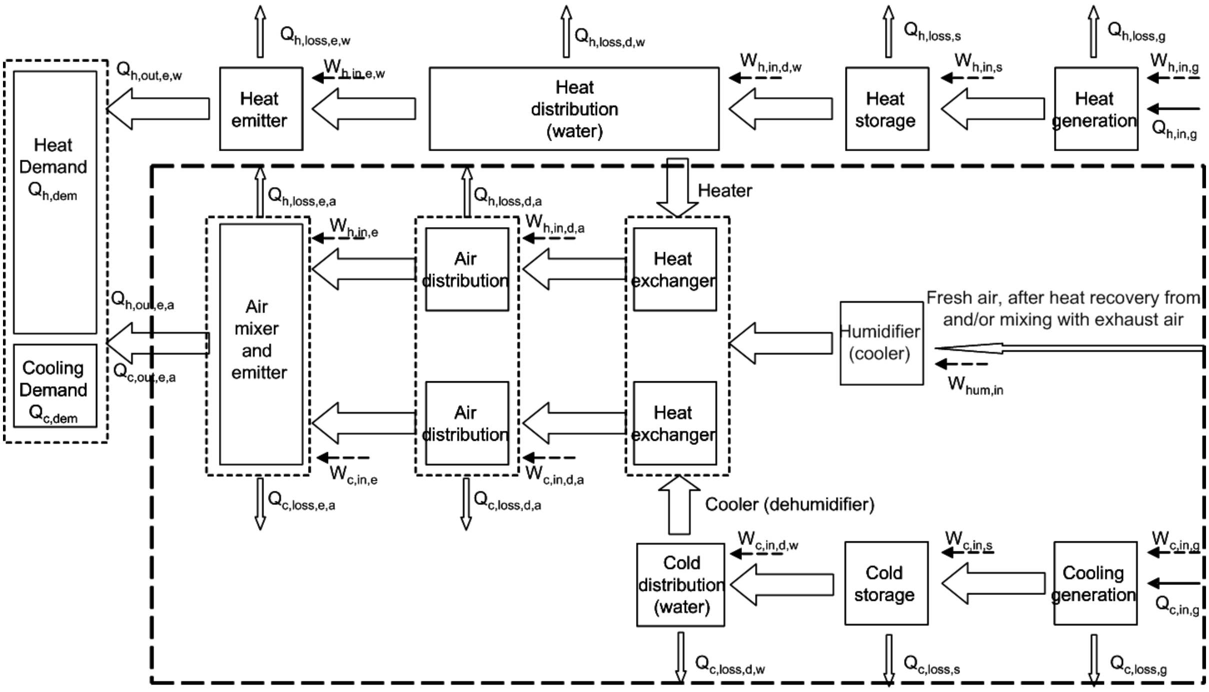

Figure C.5 — Dual duct system

Dual duct systems are all-air systems allowing room-control by mixing pre-cooled and pre-heated air. Radiators are optional.

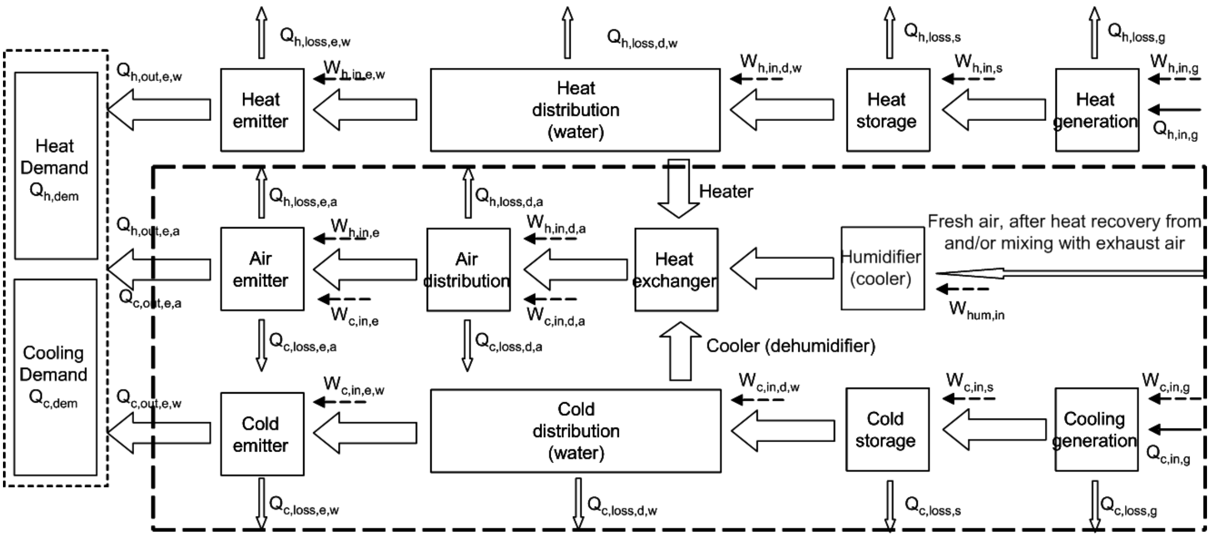

Figure C.6 — 2-pipe - change over / 4-pipe ceiling systems or embedded systems & single duct ventilation system

2-pipe - change over and 4-pipe ceiling systems behave much like 2-pipe induction - change over and 4-pipe induction systems. Here the heat and cold emitter are not using the air system for final heating and cooling but a ceiling system for direct heating and cooling the room.

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Embedded systems are integrated in building construction. Due to the inertia these systems can be operated asynchronicly with the load, e.g. at night time when better heat sink conditions apply. In addition, these systems are designed for very low temperature differences, and therefore have also a limited capacity, but for many office applications sufficient. They work with the condition that the temperature can rise (within the comfort range) during occupation time, i.e. is not constant. EN 15377-3 defines the emission calculation of those systems, the boundary is the water distribution up to the embedded pipes.

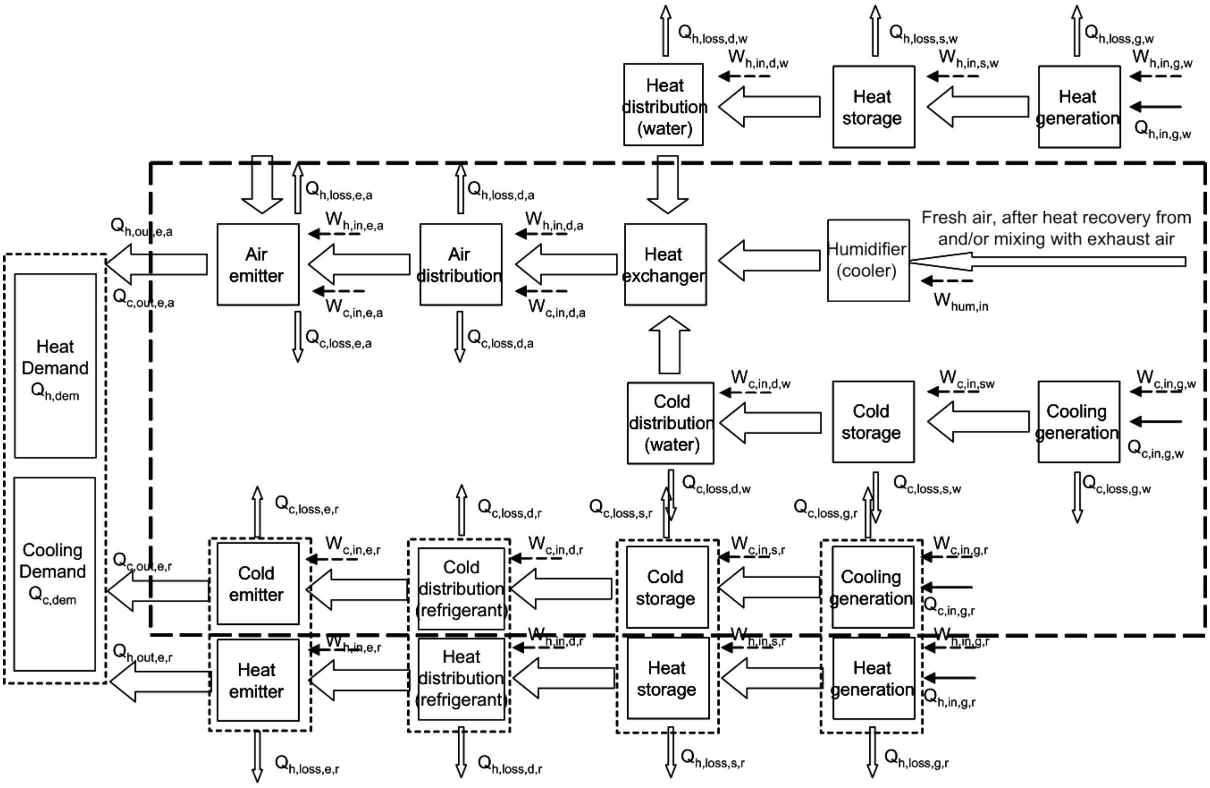

Figure C.7 — Packaged air conditoners including single / multi split systems, room units & single duct ventilation system

Splits systems and room units are systems for room heating and cooling working independent of the ventilation system. Heat and cold distribution are done by refrigerant.

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Annex D
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Schematic relationship between HVAC system energy procedure, building energy demand calculations, data and outputs

HVAC system energy calculation methods should include the following mechanisms. Not all mechanisms are important for all types of system. Information on this is given in Table 5 in the main text.

This table only deals with HVAC system features and not building energy demand calculations, even for integrated building and system models.

Mechanism	Importance	Detailed calculation methods		Simplified calculation methods		Information exchange	
		Procedure	Comment	Procedure	Comment	Imported to system model	Exported from system model
Within-room mechanisms							
Room heat balance and temperature	Essential	***CALCULATED BY BUILDING ENERGY DEMAND MODEL*** *(which also defines the incidental gains, target temperatures, required ventilation rates, occupancy periods etc)*				Room sensible heat demands. Room temperatures. Set points. Ventilation requirements. **Frequency distributions of joint hourly values of demands, temperatures and moisture contents are required.**	Contributions to room demands from system (see table below for details) exported to building energy demand model.
Room moisture balance and moisture content	Essential	***CALCULATED BY BUILDING ENERGY DEMAND MODEL*** *(which also defines the incidental latent gains and target moisture contents, if any)*				Room latent demands. Room moisture contents. Set points. **Frequency distributions of joint hourly values of energy demand temperatures and moisture contents are required.**	Contributions to room energy demand from systems (see table below for details) exported to building energy demand model.

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Mechanism	Importance	Detailed calculation methods		Simplified calculation methods		Information exchange	
		Procedure	Comment	Procedure	Comment	Imported to system model	Exported from system model
Definition of zones	Essential	***ASSUMED TO BE WITHIN BUILDING ENERGY DEMAND MODEL***				Collections of rooms that comprise HVAC zones.	
Combination of room demands into zonal demands	Essential	Explicitly calculated for each time step		Room demands combined using pretabulated diversity allowances	May require considerable previous calculation to determine tabulated values		
Combination of room conditions into zonal return air state	Essential	Explicitly calculated for each time step		Room demands combined using pretabulated diversity allowances			
Contribution to room energy demand from separate ventilation / base cooling system	Essential	Explicitly calculated for each time step	Supply temperature or moisture content may be scheduled against outdoor conditions	See ventilation standard. May be summarised by pretabulated values	May require considerable previous calculation to determine tabulated values		Cooling (or heating) and moisture provided to room exported to building energy demand model.
Contribution to room demand from heat gains or losses from pipes and ducts	Highly desirable	Explicitly calculated for each time step.	Need to distinguish between modulated and fixed temperature circuits. See EN 15316-2-3 for heating pipework	Summarised by pretabulated values, (expressed, for example, as proportion of demand)	Need to distinguish between modulated and fixed temperature circuits. See EN 15316-2-3 for heating pipework		Heat gains or losses to room exported to building energy demand model
Impact of proportional band on energy output	Desirable	Explicitly calculated for each time step.		Summarised by pretabulated values or by resetting monthly (say) setpoints		Uses demands from building energy demand model	Resulting temperatures may be exported to indoor environment assessment procedures
Impact of dead band on energy output	Desirable	Explicitly calculated for each time step.		Possibly summarized by pretabulated values	Perhaps ignored	Uses demands from building energy demand model	Resulting temperatures may be exported to indoor environment assessment procedures

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Mechanism	Importance	Detailed calculation methods		Simplified calculation methods		Information exchange	
		Procedure	Comment	Procedure	Comment	Imported to system model	Exported from system model
Effect of open-loop control or averaging of sensors	Highly desirable	Explicitly calculated for each time step.		Nominal penalty or possibly summarized by pretabulated values	Perhaps ignored, in which case the limitations of the method should be clearly stated	Uses demands and zone information from building energy demand model	Resulting temperatures may be exported to indoor environment assessment procedures
Effect of absence of interlock between heating and cooling	Highly desirable	Explicitly calculated for each time step.		Nominal penalty or possibly summarized by pretabulated values	If ignored, which is undesirable, the limitations of the method should be clearly stated	Uses demands from building energy demand model	
Distribution: terminal issues							
Energy penalties from hot/cold mixing or reheat systems	Essential *(these systems may be rarely used but the energy penalties are high)*	Explicitly calculated for each time step		Nominal penalty or possibly summarized by pretabulated values	If ignored, which is undesirable, the limitations of the method should be clearly stated		
Terminal auxiliary energy. *Includes heat pump terminals for water-loop systems, fans in split system units and fan-coils*	Essential	Explicitly calculated for each time step		Nominal allowance or possibly summarized by pretabulated values		Preferably uses equipment-specific values, so possible link to equipment standards	
Effect of sensible heat ratio of terminal	Essential	Explicitly calculated for each time step		Nominal allowance or possibly summarized by pretabulated values *See ventilation standard??*		Preferably uses equipment-specific values, so possible link to equipment standards	

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Mechanism	Importance	Detailed calculation methods		Simplified calculation methods		Information exchange	
		Procedure	Comment	Procedure	Comment	Imported to system model	Exported from system model
Lack of local time control	Highly desirable	Explicitly considered for each time step		Nominal penalty or possibly summarized by pretabulated values	If ignored, which is undesirable, the limitations of the method should be clearly stated	Uses occupancy periods from building energy demand model	
Distribution system: heat balance							
Heat gains and losses from pipes and ducts *Includes AHUs and other air-handling components*	Desirable	Explicitly considered for each time step	May include the impact on building heat balance (within room effects already accounted earlier in table). May distinguish between modulated and fixed temperature circuits. See EN 15316-2-3 for heating pipework	Nominal allowance or possibly summarized by pretabulated values	See EN 15316-2-3 for heating pipework. Perhaps ignored, in which case the limitations of the method should be clearly stated	Preferably uses equipment-specific values, so possible link to equipment standards	
Duct system air leakage *Includes AHUs and other air-handling components*	Essential If energy modulates with demand can take as proportional, else fixed rate	Explicitly considered for each time step	May include the impact on building heat balance (within-room effects are already accounted earlier in table). May distinguish between modulated and fixed temperature circuits. See ventilation standards	Nominal allowance or possibly summarized by pretabulated values	See ventilation standards for guidance and default values. If ignored, which is undesirable, the limitations of the method should be clearly stated	Preferably uses equipment-specific values, so possible link to equipment standards	

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Mechanism	Importance	Detailed calculation methods		Simplified calculation methods		Information exchange	
		Procedure	**Comment**	**Procedure**	**Comment**	**Imported to system model**	**Exported from system model**
Refrigerant pipework losses	Desirable (*Inefficiency due to heat gains and losses and to reduced flow rates)*	To be decided	Probably has to be included within equipment specification?	To be decided	Probably has to be included within equipment specification?	Preferably uses equipment-specific values, so possible link to equipment standards	
Fan and pump energy pickup	Highly desirable *Especially for fans*	Explicitly considered for each time step	See ventilation standards. Fan and pump demands have to be calculated first, and fan efficiencies defined!	Nominal allowance or possibly summarized by pretabulated values	See ventilation standards. Fan and pump demands have to be calculated first, and fan efficiencies defined!	Preferably uses equipment-specific values, so possible link to equipment standards	
Heat recovery provision	Essential	Explicitly considered for each time step		Summarized by pretabulated values	See ventilation standards. May require considerable previous calculation to determine tabulated values	Preferably uses equipment-specific values, so possible link to equipment standards	
Distribution system : operation							
Latent energy demand calculation at central (zonal) plant *(includes dewpoint cooling plus reheat)*	Essential	Explicitly considered for each time step	Should include the impact of control strategy, notably chilled water temperature modulation	Preferably use “binned” demand and weather data. Perhaps use pretabulated values	May require considerable previous calculation to determine tabulated or binned values. *See ventilation standards?*	Preferably uses equipment-specific values, so possible link to equipment standards	

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Mechanism	Importance	Detailed calculation methods		Simplified calculation methods		Information exchange	
		Procedure	Comment	Procedure	Comment	Imported to system model	Exported from system model
Adiabatic spay cooling	Desirable	Explicitly considered for each time step		Preferably use “binned” demand and weather data. Perhaps use pretabulated values	May require considerable previous calculation to determine tabulated or binned values. *See ventilation standards?*		
Additional energy demands produced by hot deck:cold deck mixing systems	Essential *(these systems may be rarely used but the energy penalties are high)*	Explicitly considered for each time step	Should include the impact of control strategy, notably deck temperature scheduling	Nominal penalty or possibly summarized by pretabulated values	If ignored, which is undesirable, the limitations of the method should be clearly stated		
Impact of mixing of return water temperature in 3-pipe systems	Desirable	Preferably explicitly considered for each time step		Nominal penalty or possibly summarized by pretabulated values	Perhaps ignored, in which case the limitations of the method should be clearly stated		
Wastage due to changeover in 2-pipe systems	Desirable	Preferably explicitly considered for each time step.	Partially an operational issue, so difficult to model explicitly	Nominal penalty	Perhaps ignored, in which case the limitations of the method should be clearly stated		
Impact of variable ventilation air recirculation *Typically CO2 controlled – total air flow unchanged*	Highly desirable	Explicitly considered for each time step		Summarized by pretabulated values	See ventilation standards. May require considerable previous calculation to determine tabulated values	Uses occupancy information from building energy demand model	

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Mechanism	Importance	Detailed calculation methods		Simplified calculation methods		Information exchange	
		Procedure	Comment	Procedure	Comment	Imported to system model	Exported from system model
Impact of air-side free cooling	Essential	Explicitly considered for each time step		Preferably use "binned" demand and weather data. Perhaps use pretabulated values	May require considerable previous calculation to determine tabulated or binned values	Uses weather data, probably most conveniently from building energy demand model	
Distribution system: auxiliary energy							
Auxiliary energy use by fans and pumps	Essential for fans, desirable for pumps	Explicitly considered for each time step	Should include ability to allow for variable flow systems	Summarized by pretabulated values	Should include ability to allow for variable flow systems, which may require considerable previous calculation to determine tabulated values	Preferably uses equipment-specific values, so possible link to equipment standards. Uses occupancy information from building energy demand model	
Cold and heat generation							
Cold generator (chiller) part-load performance *Requirements apply to each member of multiple installations*	Essential	Explicitly considered for each time step	*Separate text outlines options*	Preferably use "binned" demand and weather data. Perhaps use pretabulated values (eg seasonal performance values).	May require considerable previous calculation to determine tabulated values. Should include sequenced operation	Preferably uses equipment-specific values, so possible link to equipment standards.	
Water-side free-cooling	Highly desirable	Explicitly considered for each time step		Preferably use "binned" demand and weather data. Perhaps use pretabulated values	May require considerable previous calculation to determine tabulated values If ignored, which is undesirable, the limitations of the method should be clearly stated		

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Mechanism	Importance	Detailed calculation methods		Simplified calculation methods		Information exchange	
		Procedure	Comment	Procedure	Comment	Imported to system model	Exported from system model
Thermosyphon operation	Highly desirable	Explicitly considered for each time step		Summarized by pretabulated values	May require considerable previous calculation to determine tabulated values Perhaps ignored, in which case the limitations of the method should be clearly stated		
Impact on cold generator (chiller) performance of heat rejection equipment *Includes cooling towers, dry coolers etc*	Essential	Preferably explicitly considered for each time step		Nominal allowance or possibly summarized by pretabulated values	May be absorbed into cold generator (chiller) performance figures	Preferably uses equipment-specific values, so possible link to equipment standards.	
Auxiliary energy use by heat rejection equipment	Essential	Preferably explicitly considered for each time step	May be absorbed into heat rejection performance figures	Nominal allowance or possibly summarized by pretabulated values	May be absorbed into cold generator (chiller) performance figures	Preferably uses equipment-specific values, so possible link to equipment standards	
Heat generator (boiler) part-load performance. *Requirements apply to each member of multiple installations*	Essential	Preferably explicitly considered for each time step	Some types of heat generator (eg heat pumps) will require relatively sophisticated treatment	Summarized by pretabulated values (eg seasonal performance values)	See EN 15316-4 . Should include sequenced operation Some types of heat generator (eg heat pumps) will require relatively sophisticated treatment	Preferably uses equipment-specific values, so possible link to equipment standards	

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Mechanism	Importance	Detailed calculation methods		Simplified calculation methods		Information exchange	
		Procedure	Comment	Procedure	Comment	Imported to system model	Exported from system model
Auxiliary energy use by heat generators *Includes gas boosters, fuel pumps, etc.*	Highly desirable	Preferably explicitly considered for each time step		Nominal allowance or possibly summarized by pretabulated values	See EN 15316-4 standards. Perhaps ignored, in which case the limitations of the method should be clearly stated		
Energy use for humidification	Highly desirable	Preferably explicitly considered for each time step		Nominal allowance or possibly summarized by pretabulated values	Perhaps ignored, in which case the limitations of the method should be clearly stated	Preferably uses equipment-specific values, so possible link to equipment standards	
Bivalent systems *Includes boiler + CHP, condensing boiler + non-condensing boiler, heat pump + top-up, evaporative cooling + chiller......*	Highly desirable	Explicitly considered for each time step		Preferably use “binned” demand and weather data. Perhaps use pretabulated values,	See EN 15316-4. May require considerable previous calculation to determine tabulated or binned values	Preferably uses equipment-specific values, so possible link to equipment standards	

Annex E
(informative)

Example simplified system losses and energy demand calculation methods

E.1 Example 1 (Dutch proposal)

E.1.1 Emission losses

Emission losses are combined with distribution losses.

E.1.2 Distribution losses

E.1.2.1 General remarks

The efficiency for the distribution of heat and cold is a measure for the waste of energy because of simultaneous heating and cooling in an energy sector and the not usable heat and cold losses from ducts and pipes within an energy sector.

The method given here stems from the Dutch EPN. If the method is suitable for other countries other (national) data in the tables might be needed. If other simplified methods are required a more flexible structure is possible to include different optional methods.

E.1.2.2 Calculation rule

Determine for all indoor climate control systems the distribution efficiency for heat distribution, $\eta_{\text{distr;heat}}$, based upon the waste factor and the proportion of heat demand to cold demand with:

$$\eta_{\text{distr;heat}} = \frac{1,0}{1,0 + a_{\text{heat}} + f_{\text{waste}} / f_{\text{dem;heat}}} \tag{E.1}$$

Determine for all systems with cooling the distribution efficiency for cold distribution, $\eta_{\text{distr;cool}}$, based upon the waste factor and the proportion of the heat demand to the cold demand with:

$$\eta_{\text{distr;cool}} = \frac{1,0}{1,0 + a_{\text{cool}} + f_{\text{waste}} / f_{\text{dem;cool}}} \tag{E.2}$$

In which:

$f_{\text{dem;heat}}$ is the fraction of the heat demand with regard to the total demand of heat and cold, determined according to E.1.2.3;

$f_{\text{dem;cool}}$ is the fraction of the cold demand with regard to the total demand of heat and cold, determined according to E.1.2.3;

f_{waste} is the factor for waste of energy because of simultaneous cooling and heating, determined according to E.1.2.4;

a_{heat} is the factor for the pipe losses, duct losses and temperature control of the distribution system for heating, determined according to E.1.2.4;

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a_{cool}	is the factor for the pipe losses, duct losses and temperature control of the distribution system for comfort cooling, determined according to E.1.2.4;

Distribution efficiency should be rounded down to two decimal places.

The distribution losses are calculated by:

$$Q_{loss;heat;e\&d} = Q_{dem;heat;room} \cdot (1 - \eta_{distr;heat}) / \eta_{distr;heat} \tag{E.3}$$

$$Q_{loss;cold;e\&d} = Q_{dem;cold;room} \cdot (1 - \eta_{distr;cold}) / \eta_{distr;cold} \tag{E.4}$$

Where:

$Q_{dem;heat;room}$	is the heat demand at space level per year, in MJ;
$Q_{dem;cool;room}$	is the cooling demand per year, in MJ.

E.1.2.3 Fractions heat demand and cooling demand

Determine for the energy sector the fraction of the yearly heat demand with regard to the sum of the heating demand and the cooling demand, according to:

$$f_{dem;heat} = \max\left(\frac{Q_{dem;heat;room}}{Q_{dem;heat;room} + Q_{dem;cool;room}};0.1\right) \tag{E.5}$$

In which:

$$Q_{dem;heat;room} = Q_{dem;heat;1;room} + Q_{dem;heat;2;room} + ... + Q_{dem;heat;12;room} \tag{E.6}$$

$$Q_{dem;cold;room} = Q_{dem;cold;1;room} + Q_{dem;cold;2;room} + ... + Q_{dem;cold;12;room} \tag{E.7}$$

Determine for the considered energy sector the ratio of the yearly cooling demand to the sum of the heating demand and the cooling demand, according to:

$$f_{dem;cool} = \max[(1 - f_{dem;heat});0,1] \tag{E.8}$$

Where:

$f_{dem;heat}$	is the fraction of the heat demand with regard to the total demand of heating and cooling;
$f_{dem;cool}$	is the fraction of the cold demand with regard to the total demand of heating and cooling;
$Q_{dem;heat;room}$	is the heat demand at space level per year, in MJ;
$Q_{dem;heat;1,2,..;room}$	is the heat demand in month 1,2,... at space level, in MJ;
$Q_{dem;cool;room}$	is the cooling demand per year, in MJ.
$Q_{dem;cold;1,2,..;room}$	is the cold demand in month 1,2,... at space level, in MJ;

E.1.2.4 Waste factor and distribution loss factors

Adopt for all systems the factors f_{waste}, a_{heat} and a_{cool} from the table below.

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Not all effects are already expressed in this table. For instance: the higher losses of 3-pipe systems (B2, B6) are not visible compared to 2-pipe and 4-pipe systems (B1, B3, B4, B5, B7).

Table E.1 — Waste factors, f_{waste}, and distribution losses a_{heat} and a_{cool} for heating respectively cooling in the case of central generation

System code	Heat distribution by	Cold distribution by	Individual heating control	Waste factor f_{waste} (-) *	Weighing factor pipe and duct losses	
					Heating a_{heat} (-)*	Cooling a_{cool} (-)
No airco system	water or water and air	not available	yes	0	0,08	-
			no	0	a1	-
		water	yes	f2	0,08	0,01
A1, A2, A4 (all with radiator) A3, A5		air	yes	0	0,08	0
			no	0	a3	0
B1, B2, B3, B5, B6, B7, B8, B9, B10		water and air	yes	f4	0,08	0,01
No airco system	air	not available	yes	0	0	-
			no	0	a5	-
		water	yes	f6	0	0,01
A2 (without radiator)		air	yes	0.4	0	0
A1, A4 (all without radiator)			no	0	a7	0
B4		water and air	yes	f8	0	0,01
C1, C2, C3	central air + decentral direct heating	central air + decentral direct cooling	yes	0	0	0

NOTE 1 The system code is given in 14.1.4.

NOTE 2 Individual heating control means that on space level the flow or the temperature of the supplied heat distribution medium can be controlled by a thermostat per room. Individual heating control per space can occur with for instance thermostatic radiator valves or by thermostatic controlled air valves in air systems. An air conditioning unit serving just one space is also considered as an individual heating control.

The values of the parameters f* en a* are presented in tables below. They are depending on the applied heating curve for air distributed by the central air handling unit:

— Conventional heating curves.

— Energetic optimized heating curve or heating curve with local control.

The effect of an energetic optimized heating curve depends on climatic data. For The Netherlands the definition of an energetic optimized heating curve is given in guideline ISSO 68. Other national standards or guidelines on this subject might be used here to distinguish between both tables.

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Table E.2 — f_{waste} and a_{heat} for a conventional heating curve

θ_{TO} (ºC)	f2=f4 (-)	f6=f8 (-)	a1=a3 (-)	a5=a7 (-)
≤6	0,29	0,55	0,33	0,48
7	0,21	0,45	0,33	0,48
8	0,15	0,35	0,33	0,48
9	0,10	0,20	0,33	0,48
10	0,07	0,16	0,33	0,48
11	0,04	0,10	0,33	0,48
12[a]	0,03	0,08	0,25	0,36
13	0,02	0,06	0,18	0,24
14	0,01	0,03	0,14	0,17
15	0	0,01	0,10	0,10
≥16	0	0	0,08	0,04

[a] The reference office building which was used to determine the system efficiencies has a turn over temperature of 12 ˚C.

The values of a1 = a3 and a5 = a7 at turn over temperatures lower than 11 °C are set equal to those at 11 °C. These situations will not often occur with existing buildings in practice. However new buildings can have a lower turn over temperature.

Table E.3 — f_{waste} and a_{heat} for energetic optimized heating curve [ISSO 68] or a heating curve with local control.

θ_{TO} (ºC)	f2=f4 (-)	f6=f8 (-)	a1=a3 (-)	a5=a7 (-)
≤6	0,07	0,16	0,08	0,04
7	0,04	0,10	0,08	0,04
8	0,03	0,08	0,08	0,04
9	0,02	0,06	0,08	0,04
10	0,01	0,03	0,08	0,04
11	0	0,01	0,08	0,04
≥12 [a]	0	0	0,08	0,04

[a] When the turn over temperature is higher than 12 ˚C it is possible to use an energetically optimized heating curve without negative effects to thermal comfort or condensation on air ducts. When the turn over temperature is lower than 12 ˚C the optimized curve has to be adjusted so no negative effects will occur. Consequently this adjustment will have the effect of lower system efficiency.

The values a1 = a3 and a5 = a7 at turn over temperatures lower than 9 are, as an assumption, chosen equal to the values at 9 °C. These values can be determined more accurately.

The *turn over temperature* is the outside temperature where no heating or cooling demand exists within the building, also known as the ‘free temperature’ of the building. That means no heating or cooling needs to be supplied to the ventilation air.

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The turnover temperature is calculated by:

$$\theta_{\mathrm{TO}} = \frac{T_{\mathrm{in}} + T_{\mathrm{in;cool}}}{2} - \frac{\left(Q_{\mathrm{intern;ann}} + Q_{\mathrm{solar;t;ann}}\right) \times \dfrac{f_{\mathrm{u;avg}}}{n_{\mathrm{m;ann}} \times t_{\mathrm{m}}}}{H_{\mathrm{tr}} + H_{\mathrm{vent}}} \quad \text{(E.9)}$$

With:

θ_{TO}	turn over temperature in [°C]
T_{in}	day averaged room temperature in [°C]
$T_{\mathrm{in;cool}}$	day averaged room temperature for cooling demand in [°C]
$Q_{\mathrm{intern;ann}}$	annual heat gain by internal heat production in [MJ]
$Q_{\mathrm{solar;t;ann}}$	annual heat gain by radiation of the sun through transparent surfaces (windows) in [MJ]
$f_{\mathrm{u;avg}}$	average utility factor for heat gains (=0,64) in [-]
t_{m}	time within a month (=2,63) in [Ms]
$n_{\mathrm{m;ann}}$	number of months within a year (=12) in [-]
H_{tr}	specific heat losses by transmission in [W/K]
H_{vent}	specific heat losses by ventilation in [W/K]

The turn over temperature is always determined over a period of exactly 12 months.

E.1.3 Storage losses

Storage losses are combined with generation losses.

E.1.4 Generation efficiency and energy consumption

E.1.4.1 General approach

For the simplified method only one average generation efficiency is determined, that can be applied for all months. This generation efficiency is for gas fired cooling machines equal to the Seasonal Energy Efficiency Ratio (SEER) and for electrical cooling machines equal to SEER multiplied with the Efficiency of power generation.

If a system is equipped with one or identical cooling machines the efficiency is determined according to E.1.4.2. If two or more different cooling machines are applied first the separate efficiencies are determined followed by the calculation of the combined efficiency according to E.1.4.3.

E.1.4.2 Single cold generation efficiency

The annual efficiency of single cooling machines can be determined in three ways:

— fixed efficiencies;

— efficiencies depending on full-load data;

— efficiencies depending on full- and part-load data.

Fixed efficiencies

For single cooling machines the fixed efficiencies are listed below.

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Table E.4 - Fixed efficiency values for single cooling machines

Cooling machine type and heat sink	Efficiency $\eta_{gen;cool}$
Compression cooler / outside air	$2{,}25 \cdot \eta_{pg}$
Compression cooler / soil heat exchanger or groundwater	$5{,}0 \cdot \eta_{pg}$
Absorption cooler / outside air	$1{,}0 \cdot \eta_{th}$
Free cooling / soil heat exchanger or groundwater	$12{,}0 \cdot \eta_{pg}$

With:

η_{pg} Efficiency of power generation in units

η_{th} Efficiency of heat generation in units

Efficiencies depending on full-load data

If for compression cooling machines full load data are available, in accordance with EN 14511, take the SEER equal to the test COP at the following test conditions:

for devices that extract heat from air, in accordance with EN 14511:

— heat rejected to air: A35(24)/A27(19);

— heat rejected to water: W30/A27(19).

for devices that extract heat from water, in accordance with EN 14511:

— heat rejected to air: A35(24)/W7;

— heat rejected to water: W30/W7.

The generation efficiency is equal to the SEER, multiplied by power generation.

Efficiencies depending on full- and part-load data

An example calculation method for the seasonal efficiency of cold generators is given in Annex I.

E.1.4.3 Multiple cooler generation efficiency

After determining the efficiency of the different coolers the combined efficiency is determined from the formula:

$$\eta_{gen;cool} = 1 / \{ \alpha_{pref} / \eta_{gen;cool;pref} + (1 - \alpha_{pref}) / \eta_{gen;cool;nonpref} \} \tag{E.10}$$

With:

α_{pref} Part of the preferred cooler in total cold demand in units

$\eta_{gen;cool;pref}$ Efficiency of preferred cooler in units

$\eta_{gen;cool;nonpref}$ Efficiency of non preferred cooler in units

The part α_{pref} is read from Table E.5.

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Table E.5 - Preferred cooler demand as a function of preferred cooler power

Part of preferred cooler in total power β_{pref}	Part of preferred cooler in total demand α_{pref}
<= 0,1	0,1
0,1 – 0,2	0,2
0,2 – 0,3	0,5
0,3 – 0,5	0,8
> 0,5	1,0

$$\beta_{pref} = P_{cool;pref} / P_{cool;tot} \quad \text{(E.11)}$$

With:

β_{pref}	Part of the preferred cooler in total cooling power in units
$P_{cool;pref}$	Power of preferred cooler in kW
$P_{cool;tot}$	Power of non preferred cooler in kW

E.1.5 HVAC system annual energy consumption

The results of the different components are combined in two figures:

The heat demand of all rooms in a zone or system and the emitter/distribution losses determine the heat demand to be fulfilled by the heat generator:

$$Q_{dem;heat;system} = \Sigma\,(Q_{dem;heat;room} + Q_{loss;heat;e\&d}) \quad \text{(E.12)}$$

The cooling demand of all rooms in a zone or system, the emitter/distribution losses and the cold generation efficiency determine the primary energy demand of the HAVC system:

$$Q_{dem;cold;system} = \Sigma\,(Q_{dem;cold;room} + Q_{loss;cold;e\&d}) / \eta_{gen;cold} \quad \text{(E.13)}$$

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E.2 Example 2 (German proposal)

E.2.1 Scope

This proceeding describes the calculation of energy demand of air-conditioning systems (AC-systems) with a part of outdoor air. The task of AC-system is the guarantee of constant room air exchanges (basic system).

The energy demand is composed of the parts heating, cooling, humidifying and de-humidifying up to the supply air condition and the transport of air.

Characteristics of a basic system are, that the air volume flow and the supply air temperature are predetermined independent of the thermal loads in the building zone.

Frequently the basic system is combined with a peak system for compensation of cooling loads. The calculation of the additional energy demand for the peak system must go separate. There are two cases to distinguish:

1) The peak system is served by a second energy medium without outside air.
   Example:

— cooling ceilings

— air recirculation units

— cooling coils in an induction unit.

The energy demand of the peak system is calculated acc. to the method of monthly balancing of building (advanced method of EN 832 or prEN ISO 13790).

2) The peak system is created from an increase of outside air volume flow.
   Example: - Variable Air Volume system (VAV-system).

The method of this special case is still being revised. The aim is to adapt the proceeding for the basic system by including the results from monthly balancing of building (advanced method of EN 832 or prEN ISO 13790).

The most frequent AC-systems can be created from a combination of basic and peak system.

In special cases alternative methods like VDI guideline 2067-21 or computer simulations may be used, if the conditions correspond with the principles of this proceeding.

E.2.2 Method

The calculation method is based on the following steps.

1) The climate zones have been condensed to minimize calculation, computation and data handling.

2) For the most frequent AC-systems an extensive matrix of variants has been created based on components and operation mode.

3) For each variant the energy demand has been calculated in detail for a basic case by hourly simulation. The results are stored as specific guide values in a database or table.

4) The user has to choose one of the typical systems of the matrix and read the specific guide values from the database or table.

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5) The specific guide values have to be adjusted by a limited number of input values.

6) The expenditure for air transport is to be calculated using the respective physical equations.

7) The inputs „supply air volume flow“ and “supply air temperature“ are also inputs for the balancing of building (EN 832 or prEN ISO 13790). If the supply air temperature is below the room air temperature, there will be a negative heat-source with constant potential. The building balance will show as results the energy demand of the peak system required.

Figure E.1 shows the summary of procedure.

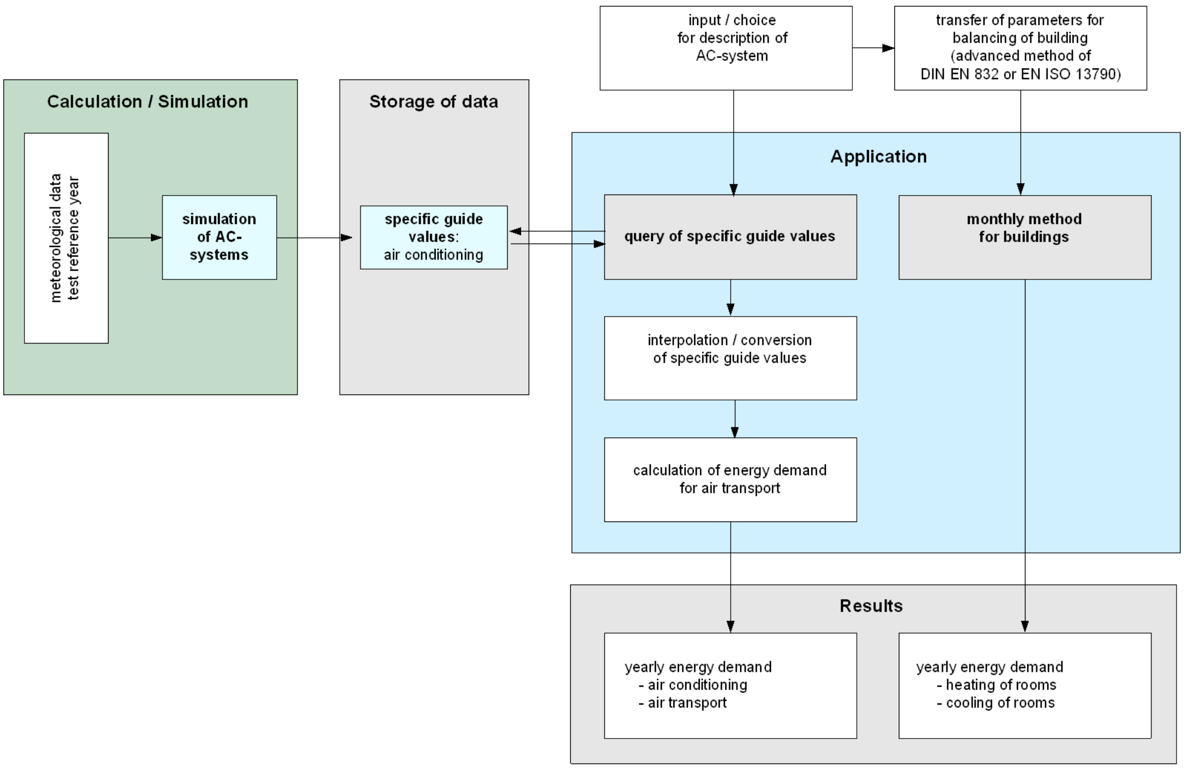

Figure E.1 — Proceeding of calculation method

E.2.3 Application for the territory of federal republic of Germany

E.2.3.1 Meteorological data

An evaluation of meteorological data for Germany has shown, that one set of representative weather data for the whole of territory will be precise enough.

The representative weather data record is the TRY 05 with the DWD-station “Würzburg”.

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Table E.6 — Monthly values of meterological data for TRY-zone 05

month	air temperature [°C]			absolute humidity x [g/kg]		
	mean value	minimum	maximum	mean value	minimum	maximum
Jan	- 1,3	- 12,5	9,2	3,0	1,1	6,1
Feb	0,6	- 16,9	7,8	3,4	0,8	5,5
Mar	4,1	- 4,8	12,5	4,0	1,6	6,7
Apr	9,5	- 3,2	22,6	4,5	2,0	7,4
May	12,9	1,5	25,0	6,7	3,4	11,7
Jun	15,7	6,4	27,7	8,0	4,4	13,9
Jul	18,0	8,8	31,2	9,1	5,9	13,1
Aug	18,3	7,1	31,4	9,5	5,3	13,9
Sep	14,4	2,4	32,6	8,1	4,6	12,7
Oct	9,1	- 1,4	21,5	6,3	3,3	10,0
Nov	4,7	- 5,1	18,4	4,7	2,4	9,5
Dec	1,3	- 12,1	12,0	3,7	1,2	7,2

E.2.3.2 Matrix of characteristics

The matrix of AC-systems is the result from a combination of the following characteristics in Table E.7.

Table E.7 — Characteristics of systems and process guiding variants

Code	request of humidity	Code	type of humidifyier	Code	type of energy recovery	Code	energy recovery index
0	no	0	adiabatic washer constant	0	no	0	45 %
1	with tolerance	1	adiabatic washer adjustable	1	only heat transfer	1	60 %
2	with tolerance	2	steam evaporator	2	transfer of heat and moisture	2	75 %

For the characteristics there are the following definitions.

E.2.3.3 Request of humidity

1) The setpoints of supply air humidity are <u>not</u> free to choose.

2) If a request for supply air humidity is required, this will apply for humidifying and for dehumidifying.

3) If acceptable the range of supply air humidity („with tolerance") will be in the range of 6,0 g/kg ≤ x_{zu} ≤ 11,0 g/kg (steam content in supply air).

4) If no range is acceptable („without tolerance") the setpoint of supply air humidity is x_{zu} = 8,0 g/kg (steam content in supply air).

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E.2.3.4 Types of humidifier

1) Adiabatic washers are divided in adjustable (humidification index = variable) and not adjustable (humidification index = constant). The operation mode of non adjustable washers will be by dew-point control.

2) For steam air humidifiers the different ways of generating steam (thermal or electrical) will be taken into account by a separate calculation of primary energy. This is not part of this proceeding.

E.2.3.5 Types of energy recovery

1) Energy recovery systems are to be distinguished in systems with and without (only sensible heat) moisture recovery.

2) It is assumed, that the energy recovery system is adjustable.

E.2.3.6 Energy recovery index

1) It is assumed the heat recovery index (definition: VDI 2071) is constant. Impacts from condensation or freezing are to be neglected.

2) For heat recovery systems with moisture recovery it is assumed, that the heat recovery index is the same as the moisture recovery index. Therefore it is possible to use the procedure also for AC-systems with recirculated air.

Not all of 3^3 combinations of characteristics in Table E.7 are practical. Table E.8 shows the 46 combinations, which make sense.

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Table E.8 — Combinations of characteristics for basic systems

		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
request humidity	no	x	x	x	x																	
	with tolerance					x	x	x	x	x	x	x	x	x	x	x	x	x	x	x	x	x
	without tolerance																					
type of humidifie	adiabatic washer constant					x	x	x	x	x	x	x										
	adiabatic washer ajustable												x	x	x	x	x	x	x			
	steam evaporat																			x	x	x
type of energy recovery	no	x				x							x							x		
	only transfer		x	x	x		x	x	x					x	x	x					x	x
	transfer heat and moisture									x	x	x					x	x	x			
energy recovery index	45 %		x				x			x				x			x				x	
	60 %			x				x			x				x			x				x
	75 %				x				x			x				x			x			

		22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46
request humidity	no																									
	with tolerance	x	X	x	x																					
	without tolerance					x	x	x	x	x	x	x	x	x	x	x	x	x	x	x	x	x	x	x	x	x
type of humidifie	adiabatic washer constant					x	x	x	x	x	x	x														
	adiabatic washer ajustable												x	x	x	x	x	x	x							
	steam evaporator	x	X	x	x															x	x	x	x	x	x	x
type of energy recovery	no					x							x							x						
	only heat-transfer	x					x	x	x					x	x	x					x	x	x			
	transfer of heat and moisture		X	x	x					x	x	x					x	x	x					x	x	x
energy recovery index	45 %		X				x			x				x			x				x			x		
	60 %			x				x			x				x			x				x			x	
	75 %	x			x				x			x				x			x				x			x

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E.2.4 Specific guide values

E.2.4.1 General

For all variants of Table E.8 the specific guide values are shown in Table E.9.

The specific guide values apply to the basic case:

— supply air temperature = 18 °C

— daily hours of operation = 12 h

— yearly days of operation = 365 d

and are related to an air volume flow of 1 m^3/h.

Table E.9 contains the following values:

q_i specific value for energy demand [Wh/(m^3/h)]

$g_{i,u}$ gradient of specific energy demand for supply air temperature < 18°C, in Wh/(K m3/h)

$g_{i,o}$ gradient of specific energy demand for supply air temperature > 18°C, in Wh/(K m3/h)

Indices i:

H heating

St steam

C cooling

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Table E.9 - Specific energy index values for a whole year

Based on Test-Reference Year 05 – Würzburg ,Germany

Variant number	Humidity requirement			Type of humidifier			Type of heat recovery			Recovered heat coefficient			Energy index values for $\theta_{V,mech}$ = 18 °C; $t_{V,mech}$ = 12 h; $d_{V,mech}$ = 365 d whole year						
	none	with toler-ance	no toleranc e	evapo-ration, not con-trollable	evapor-ation, controll-able	steam humidifie r	none	heat only	heat and moisture	45 %	60 %	75 %	heat			steam	cooling		
													$q_{H,18°C,12h}$	$g_{H,u}$	$g_{H,o}$	$q_{St,12h}$	$q_{C,18°C,12h}$	$g_{C,u}$	$g_{C,o,}$
													$\frac{Wh}{m^3/h}$	$\frac{Wh}{K \cdot m^3/h}$	$\frac{Wh}{K \cdot m^3/h}$	$\frac{Wh}{m^3/h}$	$\frac{Wh}{m^3/h}$	$\frac{Wh}{K \cdot m^3/h}$	$\frac{Wh}{K \cdot m^3/h}$
1	x						x						11 369	952	1 120	–	1 951	750	320
2	x							x		x			3 340	611	952	–	1 923	747	316
3	x							x			x		1179	275	809	–	1 913	747	316
4	x							x				x	51	12	450	–	1 903	747	316
5		x		x			x						16 241	1 003	1 170	–	2 751	667	235
6		x		x				x		x			9 820	839	1 125	–	2 707	668	232
7		x		x				x			x		8 823	824	1 052	–	2 696	668	232
8		x		x				x				x	8 394	827	968	–	2 685	668	232
9		x		x					x	x			8 942	842	1 051	–	2 710	652	230
10		x		x					x		x		8 439	826	965	–	2 714	644	230
11		x		x					x			x	8 414	826	878	–	2 700	640	229
12		x			x		x						15 479	969	1 165	–	1 970	703	246
13		x			x			x		x			7.338	729	1 028	–	1 929	703	240
14		x			x			x			x		5 011	622	928	–	1 918	703	240
15		x			x			x				x	2 978	488	791	–	1 908	703	239
16		x			x				x	x			5 471	692	1 017	–	1 901	674	241
17		x			x				x		x		2 615	493	898	–	1 882	664	240
18		x			x				x			x	428	102	654	–	1 863	654	240
19		x				x	x						11 144	959	1 161	4 472	2 015	737	257
20		x				x		x		x			3 351	596	950	4 453	2 000	604	255
21		x				x		x			x		1 222	234	805	4 453	1 990	604	255
22		x				x		x				x	273	9	405	4 479	1 981	604	255

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Variant number	Humidity requirement			Type of humidifier			Type of heat recovery			Recovered heat coefficient			Energy index values for $\theta_{V,mech}$ = 18 °C; $t_{V,mech}$ = 12 h; $d_{V,mech}$ = 365 d whole year						
	none	with toler-ance	no toleranc e	evapo-ration, not con-trollable	evapor-ation, controll-able	steam humidifie r	none	heat only	heat and moisture	45 %	60 %	75 %	heat			steam	cooling		
													$q_{H,18°C,12h}$	$g_{H,u}$	$g_{H,o}$	$q_{St,12h}$	$q_{C,18°C,12h}$	$g_{C,u}$	$g_{C,o,}$
													$\frac{Wh}{m^3/h}$	$\frac{Wh}{K \cdot m^3/h}$	$\frac{Wh}{K \cdot m^3/h}$	$\frac{Wh}{m^3/h}$	$\frac{Wh}{m^3/h}$	$\frac{Wh}{K \cdot m^3/h}$	$\frac{Wh}{K \cdot m^3/h}$
23		x				x			x	x			3 434	604	956	2 488	1 986	601	255
24		x				x			x		x		1 319	260	815	1 902	1 970	596	254
25		x				x			x			x	261	11	450	1 557	1 955	592	254
26			x	x			x						22 403	1 405	1 367	–	5 654	19	28
27			x	x				x		x			14 858	1 419	1 381	–	5 606	21	20
28			x	x				x			x		12 649	1 428	1 390	–	5 595	21	20
29			x	x				x				x	10 692	1 438	1 400	–	5 585	21	20
30			x	x					x	x			11 438	1 429	1 390	–	5 157	19	18
31			x	x					x		x		8 948	1 444	1 406	–	5 000	18	18
32			x	x					x			x	8 312	1 454	1 415	–	4 845	18	17
33			x		x		x						21 807	1 329	1 337	–	5 002	113	64
34			x		x			x		x			13 663	1 249	1 325	–	4 981	109	61
35			x		x			x			x		11 105	1 204	1 311	–	4 971	109	61
36			x		x			x				x	8 667	1 140	1 286	–	4 961	109	61
37			x		x				x	x			9 653	1 175	1 307	–	4 550	104	59
38			x		x				x		x		5 951	1 008	1 257	–	4 392	103	58
39			x		x				x			x	2 829	574	1 087	–	4 237	103	57
40			x			x	x						12 604	1 265	1 318	9 658	5 333	201	109
41			x			x		x		x			4 837	930	1 234	9 633	5 282	203	109
42			x			x		x			x		2 777	561	1 108	9 632	5 272	203	109
43			x			x		x				x	1 965	368	703	9 632	5 261	203	109
44			x			x			x	x			5 041	952	1 239	5 577	4 879	201	108
45			x			x			x		x		2 990	609	1 127	4 450	4 736	200	107
46			x			x			x			x	1 991	374	780	3 886	4 596	200	93

E.2.4.2 Specific component operating time

The utilization times of individual components of the system used for thermal conditioning of air in HVAC systems are required for evaluating the energy demands of building service technical auxiliaries.

Table E.10 shows the index values for determining the annual utilization times of the individual components. In these tables, the relative component operating time $t_{i,\mathrm{r}}$ represents the quotient of the annual operating time of the component and the annual HVAC system operating time. It is permissible to carry out linear interpolation between the values given in Table E.9 in order to obtain values for other recovered heat coefficients.

Conversion of the relative component operating times for freely-selectable supply-air temperature set-points $\vartheta_{\mathrm{V,mech}}$ and system operating times $t_{\mathrm{v,mech}}$ is carried out within the following limits:

8 h ≤ $t_{\mathrm{V,mech}}$ ≤ 24 h

14 °C ≤ $\vartheta_{\mathrm{V,mech}}$ ≤ 22 °C

using the following equations:

$$t_{\mathrm{H,r}} = t_{\mathrm{H,r,14°,12\,h}} + \left(t_{\mathrm{H,r,22°,12\,h}} - t_{\mathrm{H,r,14°,12\,h}}\right)\frac{\left(\vartheta_{\mathrm{V,mech}} - 14\,°\mathrm{C}\right)}{8\,\mathrm{K}} \quad \text{(E.14)}$$

$$t_{\mathrm{C,r}} = \left(t_{\mathrm{C,r,14°,12\,h}} + \left(t_{\mathrm{C,r,22°,12\,h}} - t_{\mathrm{C,r,14°,12\,h}}\right)\frac{\left(\vartheta_{\mathrm{V,mech}} - 14\,°\mathrm{C}\right)}{8\,\mathrm{K}}\right) f_{\mathrm{h,C}} \quad \text{(E.15)}$$

$$t_{\mathrm{St,r}} = t_{\mathrm{St,r,14°,12\,h}} = t_{\mathrm{St,r,22°,12\,h}} = \mathrm{constant} \quad \text{(E.16)}$$

$$t_{\mathrm{WRG,r}} = t_{\mathrm{WRG,r,14°,12\,h}} = t_{\mathrm{WRG,r,22°,12\,h}} = \mathrm{constant} \quad \text{(E.17)}$$

$$t_{\mathrm{VB,r}} = t_{\mathrm{VB,r,14°,12\,h}} = t_{\mathrm{VB,r,22°,12\,h}} = \mathrm{constant} \quad \text{(E.18)}$$

Where:

$f_{\mathrm{H,c}}$ is the correction factor to account for the daily operating time used to calculate the net energy demand for cooling as defined in 7.2.

The annual component operating time t_{i} is calculated from the system operating time and the relative component operating time:

$$t_{\mathrm{i}} = t_{\mathrm{i,r}}\, t_{\mathrm{V,mech}}\, d_{\mathrm{V,mech}} \quad \text{(E.19)}$$

Where the index i can represent the following:

H heater;

C cooler;

St steam humidifier;

WRG heat recovery device;

VB evaporation humidifier.

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Table E.10 - Annual relative component utilization times

Variant number	$t_{i,14°C,12h}$ for $t_{v,mech}$ = 12 h and $\theta_{V,mech}$ 14 °C					$t_{i,r,22°C,12h}$ for $t_{v,mech}$ = 12 h and $\theta_{V,mech}$ = 22 °C				
	Heater	Steam humidifier	Cooler	Heat recovery	Evaporation humidifier	Heater	Steam-humidifier	Cooler	Heat recovery	Evaporation humidifier
	$t_{H,r,14°C,12h}$	$t_{St,r,14°C,12h}$	$t_{C,r,14°C,12h}$	$t_{WRG,r,14°C,12h}$	$t_{VB,r,14°C,12h}$	$t_{H,r,22°C,12h}$	$t_{St,r,22°C,12h}$	$t_{C,r,22°C,12h}$	$t_{WRG,r,22°C,12h}$	$t_{VB,r,22°C,12h}$
	h	h	h	h	h	h	h	h	h	h
1	0,58	–	0,38	-	–	0,83	–	0,15	–	–
2	0,25	–	0,37	0,63	–	0,76	–	0,13	0,89	–
3	0,04	–	0,37	0,63	–	0,71	–	0,13	0,89	–
4	–	–	0,38	0,63	–	0,59	–	0,13	0,89	–
5	0,64	–	0,43	–	–	0,86	–	0,25	–	–
6	0,57	–	0,42	0,56	0,57	0,87	–	0,24	0,65	0,57
7	0,57	–	0,42	0,56	0,57	0,84	–	0,22	0,65	0,57
8	0,57	–	0,43	0,56	0,57	0,75	–	0,24	0,65	0,57
9	0,57	–	0,43	0,56	0,57	0,81	–	0,25	0,79	0,57
10	0,57	–	0,43	0,56	0,57	0,76	–	0,24	0,79	0,57
11	0,57	–	0,43	0,56	0,57	0,67	–	0,25	0,79	0,57
12	0,59	–	0,36	–	0,57	0,86	–	0,15	–	0,57
13	0,42	–	0,36	0,64	0,57	0,80	–	0,14	0,87	0,57
14	0,34	–	0,36	0,64	0,57	0,74	–	0,14	0,87	0,57
15	0,21	–	0,36	0,64	0,57	0,64	–	0,14	0,87	0,57
16	0,38	–	0,36	0,66	0,57	0,80	–	0,14	0,89	0,57
17	0,17	–	0,36	0,66	0,57	0,74	–	0,14	0,89	0,57
18	0,01	–	0,36	0,66	0,57	0,64	–	0,14	0,89	0,57
19	0,57	0,56	0,38	–	–	0,86	0,56	0,16	–	–
20	0,26	0,56	0,38	0,62	–	0,79	0,56	0,14	0,86	–
21	0,06	0,56	0,38	0,62	–	0,74	0,56	0,14	0,86	–
22	0,04	0,56	0,38	0,62	–	0,62	0,56	0,15	0,86	–
23	0,27	0,56	0,38	0,62	–	0,79	0,56	0,15	0,86	–

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Variant number	$t_{i,14°C,12h}$ for $t_{v,mech}$ = 12 h and $\theta_{V,mech}$ 14 °C					$t_{i,r,22°C,12h}$ for $t_{v,mech}$ = 12 h and $\theta_{V,mech}$ = 22 °C				
	Heater	Steam humidifier	Cooler	Heat recovery	Evaporation humidifier	Heater	Steam-humidifier	Cooler	Heat recovery	Evaporation humidifier
	$t_{H,r,14°C,12h}$	$t_{St,r,14°C,12h}$	$t_{C,r,14°C,12h}$	$t_{WRG,r,14°C,12h}$	$t_{VB,r,14°C,12h}$	$t_{H,r,22°C,12h}$	$t_{St,r,22°C,12h}$	$t_{C,r,22°C,12h}$	$t_{WRG,r,22°C,12h}$	$t_{VB,r,22°C,12h}$
	h	h	h	h	h	h	h	h	h	h
24	0,07	0,56	0,38	0,62	–	0,74	0,56	0,15	0,86	–
25	0,03	0,56	0,38	0,62	–	0,62	0,56	0,15	0,86	–
26	0,75	–	0,34	–	0,75	1,00	–	0,34	–	0,75
27	1,00	–	0,34	0,66	0,75	1,00	–	0,34	0,66	0,75
28	1,00	–	0,34	0,66	0,75	1,00	–	0,34	0,66	0,75
29	1,00	–	0,34	0,66	0,75	1,00	–	0,34	0,66	0,75
30	1,00	–	0,35	0,78	0,75	1,00	–	0,35	0,78	0,75
31	1,00	–	0,35	0,78	0,75	1,00	–	0,35	0,78	0,75
32	1,00	–	0,35	0,78	0,75	1,00	–	0,34	0,78	0,75
33	0,64	–	0,31	-	0,75	0,98	–	0,26	–	0,75
34	0,81	–	0,32	0,69	0,75	0,97	–	0,26	0,75	0,75
35	0,76	–	0,32	0,69	0,75	0,96	–	0,26	0,75	0,75
36	0,69	–	0,32	0,69	0,75	0,95	–	0,27	0,75	0,75
37	0,73	–	0,32	0,81	0,75	0,96	–	0,27	0,87	0,75
38	0,57	–	0,32	0,81	0,75	0,95	–	0,27	0,87	0,75
39	0,27	–	0,32	0,81	0,75	0,90	–	0,27	0,87	0,75
40	0,81	0,74	0,39	–	–	0,95	0,74	0,28	–	–
41	0,44	0,74	0,39	0,61	–	0,92	0,74	0,28	0,73	–
42	0,27	0,74	0,39	0,61	–	0,89	0,74	0,28	0,73	–
43	0,25	0,74	0,39	0,61	–	0,78	0,74	0,28	0,73	–
44	0,47	0,74	0,40	0,68	–	0,92	0,74	0,28	0,80	–
45	0,28	0,74	0,40	0,68	–	0,90	0,74	0,28	0,80	–
46	0,25	0,74	0,40	0,68	–	0,81	0,74	0,37	0,80	–

E.2.5 Energy demand for air transport

The electrical power of fans can be calculated:

$$P_{\mathrm{el}} = \frac{\dot{V}_{\mathrm{L}} \Delta p_{\mathrm{ges}}}{\eta_{\mathrm{ges}}} \tag{E.20}$$

where: $\dot{V}_{\mathrm{L}}$ air volume flow

Δp_{ges} difference of pressure in ducts and air handling units

η_{ges} efficiency of fan, motor, transmission system.

It is assumed, that the electrical energy demand of the fan will be transformed totally into heat and transferred to the air flow. The heat dissipation of fan participates to heating up the supply air. The increase in supply air temperature Δt_{ZU} can be calculated:

$$\Delta t_{\mathrm{ZU}} = \frac{\Delta p_{\mathrm{ges,Zu}}}{\varphi c_{\mathrm{p,L}} \eta_{\mathrm{ges},ZU}} . \tag{E.21}$$

If a heat recovery system is available, the energy demand of the exhaust fan will rise the supply air temperature too.

The typical data for the specific guide values are:

— $\Delta p_{\mathrm{ges,zu}}$ = 1,200 Pa

— $\Delta p_{\mathrm{ges,AB}}$ = 800 Pa

— η_{ges} = 0,70.

The result is an increase of supply air temperature Δt_{ZU} = 1,4 K. This increase is already included in the specific guide values. For other supply air fan data it's possible to adjust the heating and cooling demand (see E.2.6.1).

No consideration will be given to other exhaust air fan data.

The energy demand of air transport Q_{F} can be calculated:

$$Q_{\mathrm{F}} = (P_{\mathrm{el},ZU} + P_{\mathrm{el},AB}) \ \ z_{\mathrm{h}} \ \ z_{\mathrm{d}} \tag{E.22}$$

for z_{h}, z_{d} see E.2.6.2.

E.2.6 Conversion and calculation of specific values

E.2.6.1 Conversion for selectable supply air temperature setpoints

The specific guide values are related to a constant supply air temperature of t_{zu} = 18 °C. Within the range of definition:

14 °C ≤ t_{zu} ≤ 22 °C

it's possible to convert the specific values for supply air temperatures different from 18 °C, which are free selectable.

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The conversion is done by linear interpolation by use of the gradients g in Table A.1, which are defined according to Equation (E.23).

$$g = \frac{|\Delta q_i|}{|\Delta t_{zu}|} = \frac{|q_{i,t_{ZU}} - q_{i,18°C}|}{|t_{zu} - 18°C|} \qquad \text{(E.23)}$$

Gradients are shown only for specific values of heating and cooling. Because of the non-linearity it has to be distinguished between the direction of change above or below 18 °C.

The conversion follows the Equation (E.24) to (E.27).

$$q_W = q_{W,18\,°C} - g_{W,o}\,(t_{zu} - 18\,°C) \text{ for } 18\,°C \le t_{zu} \le 22\,°C \qquad \text{(E.24)}$$

$$q_W = q_{W,18\,°C} + g_{W,u}\,(t_{zu} - 18\,°C) \text{ for } 14\,°C \le t_{zu} \le 18\,°C \qquad \text{(E.25)}$$

$$q_K = q_{K,18\,°C} + g_{K,o}\,(t_{zu} - 18\,°C) \text{ for } 18\,°C \le t_{zu} \le 22\,°C \qquad \text{(E.26)}$$

$$q_K = q_{K,18\,°C} - g_{K,u}\,(t_{zu} - 18\,°C) \text{ for } 14\,°C \le t_{zu} \le 18\,°C \qquad \text{(E.27)}$$

The index of gradients g are:

W heating

K cooling

o change of supply air temperature above 18 °C

u change of supply air temperature below 18 °C.

The conversion is applicable for constant supply air temperature. But a conversion is also applicable, with some restrictions, for supply air temperature, which are controlled proportional to the outside air temperature.

In this case the calculation of heating demand is done with winter-setpoint for supply air temperature and the demand of cooling with summer-setpoint. In each case there will be a conservative result.

The more efficient the heat recovery system the less important is the impact of the exact rise of supply air temperature and the exact change-over.

E.2.6.2 Conversion for selectable operation time

The basic case of the specific guide values refer to an operation time daily from 06:00 h – 18:00 h and 365 days per year.

The daily operation time z_h and the yearly operation days z_d may be adjusted with following restrictions.

— range of definition z_h: $8\text{ h} \le z_h \le 24\text{ h}$

— position of operation time: 12:00 h ± 0,5 z_h

— the operation day z_d are placed equally of the year.

The conversion of the specific values can be calculated using Equations (E.28) to (E.30). With the new correction factors $f_{h,i}$ it's possible to consider the asymmetry of weather by day and by night.

$$q_H = q_{H,12\,h}\,\frac{z_h}{12}\,f_{h,W}\,\frac{z_d}{365} \qquad \text{(E.28)}$$

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$$q_{\mathrm{C}} = q_{C,12\,\mathrm{h}} \frac{z_{\mathrm{h}}}{12} f_{\mathrm{h,K}} \frac{z_{\mathrm{d}}}{365} \tag{E.29}$$

$$q_{\mathrm{St}} = q_{\mathrm{St,12h}} \frac{z_{\mathrm{h}}}{12} f_{\mathrm{h,D}} \frac{z_{\mathrm{d}}}{365} \tag{E.30}$$

where:

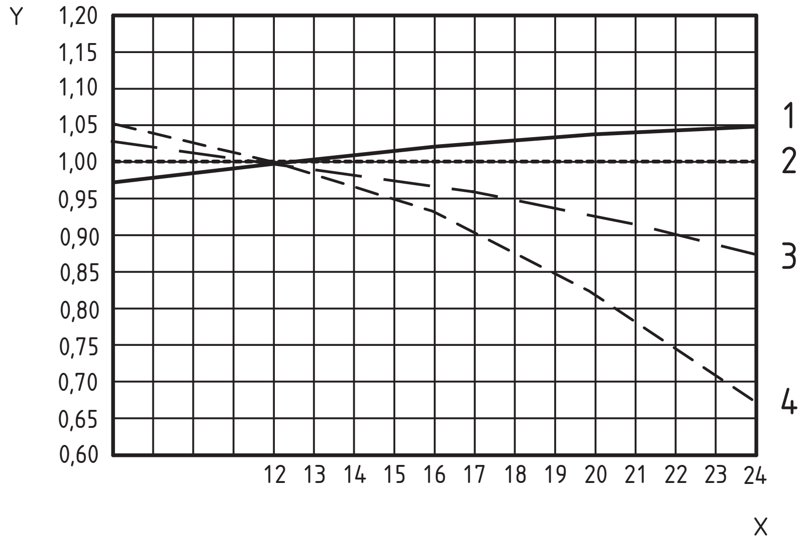
q_{H}, q_{C},q_{St}: specific values for heating, cooling, steam for selectable operation time

$q_{\mathrm{H,12h}}$, $q_{\mathrm{C,12h}}$, $q_{\mathrm{St,12h}}$: specific values for heating, cooling, steam for basic operation time z_{h} = 12 h (according to Table E.9)

z_{h}: daily operation hours [h]

z_{d}: yearly operation days [d]

$f_{\mathrm{h,W}}$, $f_{\mathrm{h,K}}$, $f_{\mathrm{h,D}}$: correction factor for weather-asymmetry (heating, cooling, steam) according to Figure E.2



Key

X operation time of ac-system [h/d]

Y correction factor[-]

1 heating $f_{\mathrm{h,W}}$

2 steam $f_{\mathrm{h,d}}$

3 cooling $f_{\mathrm{h,k}}$ "request of humidity without tolerance"

4 cooling $f_{\mathrm{h,k}}$ "no request of humidity with tolerance"

Figure E.2 — Correction factors $f_{\mathrm{h,i}}$ for consideration of daily operation time

E.2.6.3 Conversion for selectable energy recovery coefficients

If the energy recovery index Φ is selected between the standard values 0 %, 45 %, 60 % and 75 %, a linear interpolation is permissible.

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In this case the conversion of the specific values q_i' and q_i'' can be calculated separately for the next lower value Φ' and the next higher value Φ'' as per section E.2.6.1 and E.2.6.2.

In the next step the specific values q_i for heat, cold and steam can be interpolated between the values q_i' and q_i''.

$$q_i = q'_i + \frac{q''_i - q'_i}{\Phi'' - \Phi'}(\Phi - \Phi') \qquad \text{(E2.31)}$$

E.2.6.4 Calculation of energy demand

The specific values are related of the basic case with supply air volume flow of 1 m³/h. The calculation of the energy demands Q_i can be calculated with actual air volume flow as per Equation (E.32)

$$Q_i = q_i \dot{V}_L \qquad \text{(E.32)}$$

where:

Q_i energy demand of heating, cooling and steam generation [Wh]

q_i specific value of heating, cooling, steam [Wh/(m³/h)]

$\dot{V}_L$ volume flow of supply air [m³/h].

E.2.6.5 Algorithm

Table E.11 — Algorithm

1.	**Classification of system and choice of characteristics:** 1. request of humidity 2. type / operation mode of humidifying 3. type of energy recovery **Input of:** 1. energy recovery index Φ 2. supply air temperature t_{ZU} 3. operation time z_h, z_d 4. air volume flow $\dot{V}_L$ 5. fan data η_{ges}, $\Delta p_{ges,ZU}$, $\Delta p_{ges,AB}$	
2.	**Calculation of the energy expenditure of air transport** Correction of supply air temperature in consideration of fan power (step 5)	Equations: (E.20), (E.21), (E.22)
3.	**choice of higher and lower energy recovery index Φ' und Φ''** with Φ' < Φ Φ'' ≥ Φ with Φ'; Φ'' in [0; 0,45; 0,60; 0,75]	
4.	**Determination of specific values** $q'_i = f(\Phi')$ and $q''_i = f(\Phi'')$ from table / database with index *i*: W heating K cooling D steam	Table E.12

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5.	**Conversion for supply air temperature,** if $t_{ZU} \neq 18$ °C through linear interpolation with gradients for q'_i and q''_i with index *i*: W heating K cooling	Equations: (E.24) – (E.27) Table E.12
6.	**Conversion of operation time, if** $z_h \neq 12$ h or $z_d \neq 365$ d for q'_i and q''_i with index *i*: W heating K cooling D steam	Equations: (E.28) – (E.30) Figure E.2
7.	**Calculation of q_i for really energy recovery effectiveness Φ** through linear interpolation between q'_i and q''_i with index i: W heating K cooling D steam	Equation (E.31)
8.	**Determination of q_i with really air flow** with index i: W heating K cooling D steam	Equation (E.32)

E.2.7 Example

The example is an outdoor air handling unit for a hospital with:

— integrated circuit recovery unit

— steam evaporator

— requests of humidity in high quality.

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Table E.12 - Example results

1	Inputs				
		heat recovery index Φ	38 %		
		setpoint of supply air temperature t_{zu}	16		°C
		daily operation time z_h	24		h
		yearly operation time z_d	350		h
		supply air volume	30.000		m^3/h
		efficiency of the fan	65 %		
		pressure difference of supply/exhaust air fan Δp_{ges}	1.400	1.000	Pa
2	Calculation of air transport				
		fan power		30,8	kW
		increase of supply air temperature		0,4	K
		energy demand		258.462	KWh
3	Required variants for interpolation: 41 (Φ'' = 45 %) and 40 (Φ'' = 0 %)				
			Φ''	Φ''	
		heat recovery index	0 %	45 %	
4	Specific guide values		q'	q"	
		heating	12.442	4.679	$Wh/(m^3/h)$
		steam	9.650	9.648	$Wh/(m^3/h)$
		cooling	5.116	5.082	$Wh/(m^3/h)$
5	Supply air temperature				
		supply air temperature including fan-heating	15,60	15,60	°C
		$q_{W,u}$	1.259	923	$Wh/(Km^3/h)$
		$q_{k,u}$	217	222	$Wh/(Km^3/h)$
	Conversion of specific values		q'	q"	
		heating	9.422	2.465	$Wh/(m^3/h)$
		cooling	5.637	5.615	$Wh/(m^3/h)$
6	Operation time				
		correction factor $f_{h,H}$	1,050	1,050	-
		correction factor $f_{h,D}$	1,000	1,000	-
		correction factor $f_{h,K}$	0,875	0,875	-
	Conversion of specific values				
		heating	18.972	4.963	$Wh/(m^3/h)$
		steam	18.507	18.503	$Wh/(m^3/h)$
		cooling	9.459	9.422	$Wh/(m^3/h)$
7	Interpolation for the actual energy recovery index				
		heating		7.142	$Wh/(m^3/h)$
		steam		18.504	$Wh/(m^3/h)$
		cooling		9.427	$Wh/(m^3/h)$
8	Energy demand for actual air volume				
		demand of heating		214.272	kWh
		demand of steam		555.108	kWh
		demand of cooling		282.824	kWh
		demand of electricity		258.462	kWh

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E.3 Example 3: Monthly HVAC system cooling energy calculations using degree-day methods

E.3.1 Theory

E.3.1.1 General

A cooling degree-day calculation method must account for a number of factors that are not significant for heating degree-day applications. These are:

1) The presence of latent loads.

2) The possibility of high fresh air loads.

3) The use of heat recovery, whether sensible only or sensible + latent.

4) Variable air flows.

5) The wide variety of cooling systems in use.

6) The temperature dependence of chiller coefficient of performance.

The key to a cooling energy assessment lies in the definition of base temperature. For cooling, this needs to be defined specifically for each different type of cooling system. The definitions of three types of system are presented here. The same principle can be adopted to determine the base temperature for other system types.

E.3.1.2 All air systems

ASHRAE [2] defines cooling degree-day base temperature in the same way as for heating. However, with an air conditioned building the outdoor temperature will exceed the internal temperature for much of the time the system is operating (although not necessarily all of the time depending on the magnitude of the gains). Under these high outdoor temperature conditions the term Q_G/U' has little physical significance in terms of the building energy balance; in any case this term has no relationship to the fresh air or latent components of the cooling demand. It follows that the usual definition of base temperature cannot be used as a reliable definition of base temperature for all-air cooling systems.

In this case we can define the base temperature with reference to the cooling coil, which can be achieved by considering the coil energy balance, [3]. Consider the 100 % fresh air system. The energy extracted from the air is given by

$$Q_E = \dot{m}c_p(\theta_{ao} - \theta_c) + \dot{m}h_{fg}(g_o - g_c) \tag{E.33}$$

where Q_E is the rate of heat removal from the air, *m* is the mass flow rate of the air, c_p is the specific heat of air, h_{fg} is the enthalpy of vaporisation of water, θ_{ao} is the outside air temperature, θ_c is the off coil air temperature, and g_o and g_s are the outside and off coil moisture contents respectively. This load can be broken down further into its components

$$Q_E = Q_{fabric} + Q_{solar} + Q_I + Q_{fan} + Q_{f.a.,S} + Q_{f.a.,L} + Q_L \tag{E.34}$$

Where Q_{solar}, Q_I, and Q_L are the solar, internal sensible, and room latent loads respectively. Q_{fabric} is the fabric gain given by

$$Q_{fabric} = U'(\theta_{eo} - \theta_{ai}) \tag{E.35}$$

Where θ_{eo} is the sol-air temperature. This can be simplified by assuming the opaque fabric gains are small compared to the glazing conduction, in which case θ_{eo} can be replaced by the air temperature,

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θ_{ao}. If this is taken as the mean air temperature during the occupied hours, denoted θ'_{ao}, this will capture at least some of these gains to the space. (Using the mean 24 h air temperature would cancel these gains, and suggest no cooling load at all from this component). $Q_{f.a.,S}$ is the net sensible heat extracted from the fresh air, given by

$$Q_{\mathrm{f.a.,S}} = \dot{m}c_{\mathrm{p}}\left(\theta_{\mathrm{ao}} - \theta_{\mathrm{ai}}\right) \tag{E.36}$$

Where $\dot{m}$ is the mass flow rate of air and c_p is the specific heat of air. $Q_{f.a.,L}$ is the fresh air latent load, which can be combined with the room latent load to give the latent heat extracted from the air

$$Q_{\mathrm{f.a.,L}} + Q_{\mathrm{L}} = \dot{m}h_{\mathrm{fg}}(g_{\mathrm{o}} - g_{\mathrm{s}}) \tag{E.37}$$

Q_{fan} is the heat imparted to the air by the fan

$$Q_{\mathrm{fan}} = \frac{\dot{v}\,\Delta P}{\eta_{\mathrm{fan}}} = \dot{m}\,c_{\mathrm{p}}\left(\theta_{s} - \theta_{\mathrm{c}}\right) \tag{E.38}$$

where $\dot{v}$ is the volume flow rate of air, ΔP is the pressure rise across the fan, η_{fan} is the fan efficiency and θ_s is the supply air temperature.

Of these loads the fabric gain (which may be a loss in certain circumstances) is likely to be reasonably small, especially in the UK where ambient temperatures will be close to the indoor set point temperature for much of the time. This makes the assumptions about how to treat this load that follow relatively unimportant.

The latent load can be treated as an equivalent sensible load, and the moisture difference converted to a fictitious temperature difference in air temperature across the coil as follows

$$\dot{m}h_{\mathrm{fg}}\left(g_{\mathrm{o}} - g_{\mathrm{s}}\right) = \dot{m}c_{\mathrm{p}}\Delta\theta'_{\mathrm{L}} \tag{E.39}$$

where $\Delta\theta'_L$ is the fictitious latent temperature difference. Rearranging Equation 39 gives

$$\Delta\theta'_{\mathrm{L}} = \frac{h_{\mathrm{fg}}}{C_{\mathrm{p}}}(g_{\mathrm{o}} - g_{\mathrm{s}}) \tag{E.40}$$

Putting in typical values for h_{fg} and c_p of 2450 kJ kg^{-1} and 1.02 kJ kg^{-1}K^{-1} respectively gives

$$\Delta\theta'_{\mathrm{L}} = 2400\left(g_{\mathrm{o}} - g_{\mathrm{s}}\right) \tag{E.41}$$

All of these component loads can be combined and expressed in terms of temperature differences, i.e. each load is defined as the air temperature drop across the coil associated with that load:

$$Q_{\mathrm{E}} = \dot{m}c_{\mathrm{p}}\left[(\theta_{\mathrm{ao}} - \theta_{ai}) + \frac{\dot{v}\Delta P}{\dot{m}c_{\mathrm{p}}\eta_{\mathrm{fan}}} + \frac{Q_{\mathrm{s}}}{\dot{m}c_{\mathrm{p}}} + \frac{U'}{\dot{m}c_{\mathrm{p}}}(\theta'_{\mathrm{ao}} - \theta_{\mathrm{ai}}) + 2400(g_{\mathrm{o}} - g_{\mathrm{s}})\right] \tag{E.42}$$

In terms of energy extracted over time this can be re-expressed as an integral:

$$\int Q_{\mathrm{E}}dt = \dot{m}c_{\mathrm{p}}\int\left\{\theta_{\mathrm{ao}} - \left[\theta_{\mathrm{ai}} - \frac{\dot{v}\Delta P}{\dot{m}c_{\mathrm{p}}\eta_{\mathrm{fan}}} - \frac{Q_S}{\dot{m}c_{\mathrm{p}}} - \frac{U'}{\dot{m}c_{\mathrm{p}}}(\theta'_{\mathrm{ao}} - \theta_{\mathrm{ai}}) - 2400\left(g_{\mathrm{o}} - g_{\mathrm{s}}\right)\right]\right\}dt \tag{E.43}$$

The term inside the square bracket is equivalent to the base temperature. More simply stated, the cooling base temperature is the off-coil air temperature (required to deal with the sensible loads) *minus* the fictitious latent temperature difference.

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The integral on the right hand side of Equation (E.43) is the cooling degree-day total. In order to find the monthly cooling degree-days average monthly values of the variables in the square brackets can be used. In the case of moisture, the average difference can be found from a form of Hitchin's [4] formula

$$\overline{\left(g_\mathrm{o}-g_\mathrm{s}\right)}=\frac{\bar{g}_\mathrm{o}-g_\mathrm{s}}{1-e^{-k\left(\bar{g}_\mathrm{o}-g_\mathrm{s}\right)}} \tag{E.44}$$

where g_o is the mean monthly outdoor moisture content, and k is found from

$$k=\frac{2{,}5}{\sigma_\mathrm{go}} \tag{E.45}$$

where σ_go is the standard deviation of outdoor monthly moisture content. For London k lies in the region 1220 to 2300 [3] with a mean value of 1700. Note that $\Delta\theta'_\mathrm{L}$ is relatively insensitive to changes in k.

It is possible to extend the concept to account for intermittent operation (with thermal capacity effects) and the use of heat recovery.

E.3.1.3 Thermal capacity effects

The issue of thermal capacity and intermittent operation is highly complex, but it is possible to provide a simplified model to attempt to take some account of this. Gains to the space will be absorbed by internal surfaces before becoming an apparent cooling load. Depending on the thermal capacity of the exposed mass the load on the cooling system can be mitigated if these gains can be stored and effectively released outside of occupancy hours. This is the principle of night-time cooling where the building fabric warms up slowly during the day (staying below the room temperature), and only releases the heat when the ambient air temperature falls below the fabric temperature. These effects can be accounted for by assuming an exponential rate of cooling, yielding

$$\Delta\theta_\mathrm{i}=e^{-\frac{t_3-t_1}{\tau}}\left(\theta_\mathrm{sp}-\theta_\mathrm{ao}\right) \tag{E.46}$$

Where $\Delta\theta_\mathrm{i}$ is the change in the temperature of the building fabric, t_3-t_1 is the unoccupied period. In this case θ_ao should be the average overnight outdoor air temperature. This change in fabric temperature can be multiplied by the thermal capacity of the structure, and divided by 24 × 3600 to give the average rate of gain that will be absorbed by the structure, Q_C, during the day:

$$Q_\mathrm{C}=\frac{C\ \Delta\theta_\mathrm{i}}{24\times3600} \tag{E.47}$$

Q_C will have the opposite sign to the gains as it is mitigating the load on the plant. It can be incorporated into the base temperature expression (the square brackets of Equation E.43) as follows:

$$\theta_\mathrm{b}=\theta_\mathrm{ai}-\frac{\dot{v}\Delta P}{\dot{m}c_\mathrm{p}\eta_\mathrm{fan}}-\frac{Q_\mathrm{S}}{\dot{m}c_\mathrm{p}}-\frac{U'}{\dot{m}c_\mathrm{p}}\left(\theta'_\mathrm{ao}-\theta_\mathrm{ai}\right)-2400\left(g_\mathrm{o}-g_\mathrm{s}\right)+\frac{Q_C}{\dot{m}c_\mathrm{p}} \tag{E.48}$$

E.3.1.4 Heat recovery

Heat recovery in air conditioning systems can be either sensible only or sensible plus latent recovery. Sensible only heat recovery includes plate heat exchangers, run around coils and heat pipes; sensible plus latent systems employ air recirculation or hygroscopic thermal wheels.

The effectiveness, ε, of a counter flow heat exchanger is constant and is given by

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$$\varepsilon = \frac{(\theta_{\mathrm{ao}} - \theta_{\mathrm{m}})}{(\theta_{\mathrm{ao}} - \theta_{\mathrm{r}})} \tag{E.49}$$

Where θ_r is the return air temperature (normally the room air temperature plus a temperature rise for fan and duct gains) and the on-coil air temperature, θ_m, is therefore

$$\theta_{\mathrm{m}} = \theta_{\mathrm{ao}} - \varepsilon(\theta_{\mathrm{ao}} - \theta_{\mathrm{r}}) \tag{E.50}$$

where only positive values of (θ_{ao}-θ_r) are valid. From a practical point of view this presents a complication to the degree-day problem as θ_{ao} is variable, and this should be incorporated into the degree-day integral as follows

$$D_{\mathrm{c}} = \int (\theta_{\mathrm{ao}} - \varepsilon(\theta_{\mathrm{ao}} - \theta_{\mathrm{r}}) - \theta_{\mathrm{b}}) dt \tag{E.51}$$

In theory latent heat recovery should be treated in the same way, in which case the on-coil moisture content, x_m, is given by

$$x_m = x_o - \varepsilon(x_o - x_r) \tag{E.52}$$

The temperature rise due to gains defined in Equation E.41 becomes modified such that

$$\Delta\theta'_L = 2400(x_m - x_s) \tag{E.53}$$

The mean monthly moisture difference of Equation E.44 becomes

$$\overline{(x_m - x_s)} = \frac{\bar{x}_o - x_s}{1 - e^{-k(\bar{x}_o - x_s)}} - \frac{\varepsilon(\bar{x}_o - x_r)}{1 - e^{-k(\bar{x}_o - x_r)}} \tag{E.54}$$

Thus the latent heat recovery effects can be embedded into the expression for the base temperature. This is a pragmatic solution, justified by the fact that latent loads are generally a small fraction of the total load. Note that for all air systems the effectiveness is related to the fresh air fraction, FAF, by

$$\varepsilon = 1 - FAF \tag{E.55}$$

with

$$FAF = \frac{\dot{m}_{\mathrm{FA}}}{\dot{m}_{\mathrm{FA}} + m_{\mathrm{R}}} \tag{E.56}$$

where m_{FA} is the mass flow rate of the fresh air and m_R is the mass flow rate of the room return air.

A full worked example of an all-air energy calculation is shown in E.3.2.

E.3.1.5 Fan coil systems

The cooling coil of a central air handling unit supplying treated fresh air to distributed fan coil units (that deal with the majority of the local gains) largely deals with the fresh air load, but can also provide latent cooling. Where the latent gains are dealt with by the fan coils (i.e. the fan coils are run wet), the approach described in E.3.1.2 can be used by amalgamating the fan coils into one virtual coil and using average gains for the building. However, where the coils are run dry but the central coil provides latent cooling, the two components should be considered separately. The energy balance for the central coil, Q_{cc}, is given by

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$$Q_{\mathrm{cc}} = \dot{m}_{\mathrm{FA}} c_{\mathrm{p}} \left(\theta_{\mathrm{ao}} - \theta_{\mathrm{c}} + \Delta\theta'_{\mathrm{L}}\right) \tag{E.57}$$

where m_{FA} is the mass flow rate of fresh air. The fan coil load, Q_{FC}, can be separated into the fresh air component and the room air component.

$$Q_{\mathrm{FC}} = \dot{m}_{\mathrm{FA}} c_{\mathrm{p}} \left(\theta_{\mathrm{c}} - \theta_{\mathrm{s}}\right) + \dot{m}_{\mathrm{R}} c_{\mathrm{p}} \left(\theta_{\mathrm{r}} - \theta_{\mathrm{s}}\right) \tag{E.58}$$

where θ_{r} is the room air temperature and θ_{s} is the supply air temperature leaving the fan coil. θ_{s} is a function of all the sensible gains to the space as defined in 1.1 (it has not been shown in its component parts here for clarity). If the energy demand is expressed in terms of the total system mass flow rate (i.e. the sum of fresh and return air flows) we want an expression such that

$$\int Q_{\mathrm{E}} dt = \dot{m}_{\mathrm{T}} c_{\mathrm{p}} \int \left(\theta_{\mathrm{ao}} - \theta_{\mathrm{b}}\right) dt \tag{E.59}$$

where m_{T} is the total mass flow rate, (m_{FA}+m_{R}), it is possible to combine Equations E.57 and E.58 to show that

$$\theta_{\mathrm{b}} = \left[\theta_{\mathrm{s}} + \frac{\dot{m}_{\mathrm{R}}}{\dot{m}_{\mathrm{T}}}\left(\theta_{\mathrm{o}} - \theta_{\mathrm{r}}\right) - \frac{\dot{m}_{\mathrm{FA}}}{\dot{m}_{\mathrm{T}}} \Delta\theta'_{\mathrm{L}}\right] \tag{E.60}$$

Note that this expression for the base temperature is related to the *total mass flow rate of the system*.

E.3.1.6 Chilled beams and ceilings

Chilled beams and ceilings, whether active or passive, will not experience latent loads; these systems are designed to deal only with sensible loads to prevent condensation problems in the occupied space. These systems can be dealt with in the same way as the heating system. The energy balance on the cooling element will be equal to the sensible gains minus the losses from the space

$$Q_C = Q_G - U'\left(\theta_i - \theta_o\right) \tag{E.61}$$

At zero load the gains equal the losses, and this occurs when the outdoor temperature equals the base temperature.

$$\left(\theta_i - \theta_b\right) = \frac{Q_G}{U'} \tag{E.62}$$

Which leads to precisely the same definition of base temperature as the heating case, but the temperatures within the degree-day integral are reversed. Thus the energy demand will be

$$\int Q_E dt = U' \int \left(\theta_{ao} - \theta_b\right) dt \tag{E.63}$$

From this simple energy balance it is possible to draw inferences that may help in building energy diagnostics. For example when $\theta_{\mathrm{o}} = \theta_{\mathrm{i}}$ (i.e. there are no losses) then load will be equal to the gains, or $Q_{\mathrm{C}}=Q_{\mathrm{G}}$.

E.3.2 Worked example

E.3.2.1 All-air cooling

E.3.2.1.1 General

For a centralised all-air air conditioning system, monthly energy estimates can be carried out according to the method set out in E.3.1.2. This section presents examples for both full fresh air and heat recovery using air recirculation. The recirculation method can also be used for heat recovery

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systems employing sensible and latent recovery (for example hygroscopic thermal wheels), and can be adapted for sensible-only recovery by ignoring the latent recovery component. Using degree-day methods for heat recovery systems suffers from an inability to explicitly vary the amount of heat recovery according to prevailing conditions; bin methods are able to account for the control of (for example) damper positions dependent upon outdoor temperature or enthalpy. However, the methodology set out here does implicitly account for this heat recovery control and any errors due to lack of explicit rigour are likely to be small. Unlike the heating example, uncertainty analysis has not been conducted on the cooling methodology, and caution is urged when interpreting the results.

E.3.2.1.2 Example 1 Full fresh air

Table E.13 shows the required inputs. In its simplest form the base temperature is the off-coil dry-bulb temperature minus the fictitious latent temperature difference. The off-coil temperature is determined by all of the sensible gains that the system is required to deal with: solar and internal gains; fabric gains; fan gains; sensible fresh air load. These can be mitigated by overnight cooling effects, which can be incorporated into the base temperature. In addition is the latent load which is dealt with by the fictitious temperature rise derived in Equations (E3.7) to (E3.9). Each of these components are shown in Equation (E3.16).

Table E.13 - Input for air-cooling estimate

Inputs		
Inside set point	θ_{ai}	22°C
Room moisture content	x_i	0,0084kg/kg
Supply temp	$\theta_{s\ design}$	14°C
Fan pressure	Δp	1500Pa
Fan efficiency	η_{fan}	0,6
Design sensible gains	$Q_{S\ design}$	200kW
Monthly average sensible gains	Q_S	125kW
Heat carrying capacity of air	mc_p	25,0kW/K
Fabric heat loss coefficient	AU	2,5kW/K
Monthly mean outside temp (day)	$\theta_{ao\ day}$	23°C
Monthly mean outside temp (night)	$\theta_{ao\ night}$	16°C
Mean monthly outside temp (overall)	θ_{ao}	18,5
Fabric thermal capacity	$\rho c_p V$	250000kJ/K
Unoccupied period	Δt_{unocc}	16h
time constant	τ	27,78h
Overnight cooling	$\Delta\theta_i$	3,37K
Fabric cooling availability	Q_C	9,76kW
Supply moisture content	x_s	0,0083kg/kg
Monthly mean outside moisture	x_o	0,009kg/kg
Constant (moisture)	$k\ (x)$	1700
Mean positive moisture difference	$(x_o - x_s)$ bar	0,00101kg/kg
Fictitious latent rise	$2400(x_o - x_s)$	2,414560712K
Days in the month	N	31
Constant (dry bulb)	k	0,71
Chiller COP	COP	3
Cost of electricity	c_{el}	0,05p/kWh
CO_2 factor	f_{CO2}	0,43kg/kWh

Note the heat carrying capacity of the air or 25 kWK^{-1} has been calculated from the design conditions of 200 kW sensible gain to the space and a room to supply air temperature difference of (22-14).

Taking the components of Equation E.48 in turn:

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Fan temperature rise, from Equation E.35

$$(\theta_s - \theta_c) = \frac{20{,}4 \times 1{,}5}{25 \times 0{,}6} = 2\ \mathrm{K}$$

Sensible gains to the space (solar, people, lights and machines)

$$\frac{Q_s}{\dot{m}c_p} = \frac{125}{25} = 5K$$

Day time fabric gain from Equation (E.38) and dividing by mc_p

$$\frac{U'}{\dot{m}\,c_p}(\theta_{ao}' - \theta_{ai}) = \frac{2{,}5}{25}(23 - 22) = 0{,}1\ \mathrm{K}$$

Mitigation due to overnight cooling, combining Equations (E.46) and (E.47) and dividing by mc_p

$$\frac{Q_C}{\dot{m}\,c_p} = \frac{250 \times 10^3}{24 \times 3\,600} \times \frac{e^{-\frac{16}{27{,}8}} \times (22 - 16)}{25} = -0{,}4\ \mathrm{K}$$

Fictitious latent component, combining Equations (E.41) and (E.44)

$$\overline{\Delta\theta_L'} = 2\,400 \times \frac{0{,}009 - 0{,}008\,4}{1 - e^{-1700 \times (0{,}009 - 0{,}008\,4)}} = 2{,}4\ \mathrm{K}$$

Putting all of these together (Equation (E.48) now gives the base temperature

θ_b = 22 – 2 – 5 – 0,1 – 2,4 +0,4 = 12,9°C

Degree-days can now be calculated from a modified form of Hitchin's formula, equation, that simply swaps the positions of the temperatures (i.e. now using θ_o-θ_b).

$$D_m = \frac{31 \times (18{,}5 - 12{,}9)}{1 - e^{-0{,}71 \times (18{,}5 - 12{,}9)}} = 177\ \mathrm{K} - day$$

Since these degree-days are the effective summation of temperature difference across the coil, these are multiplied by the mass flow and specific heat of the air, mc_p, and then by 24 to give kWh. This is divided by the *COP* of the chiller to get fuel consumption

$$F = \frac{24 \times 25 \times 177}{3{,}0} = 35\,400\ \mathrm{kWh}$$

The cost is

35400 × 0,05 = £1770

and related CO_2 emissions

35400 × 0,43/1000 = 15,2 tonnes

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E.3.2.1.3 Sensible and latent heat recovery

This is a slight extension to example 1. If we consider a re-circulation system sensible and latent heat will be recovered in proportion to the amount of room air mixing with the fresh air. This is normally expressed in terms of the fresh air fraction, FAF, or effectiveness, ε (which is actually 1-FAF). Thus for a FAF of 0,25 the effectiveness would be 0,75. This can be incorporated into Equations (E.51) and (E.54) to deal with the influence of heat recovery. Equation (E.54) will reduce the latent load on the coil according to how often g_o exceeds g_r; the form of Hitchin's formula accounts for the distribution of g_o that enables this. Thus the reduced fictitious latent temperature rise is now

$$\Delta\theta_L' = 2\,400 \times \left(\frac{0{,}009 - 0{,}008\,3}{1 - e^{-1700 \times (0{,}009 - 0{,}008\,3)}} - \frac{0{,}75 \times (0{,}009 - 0{,}008\,4)}{1 - e^{-1700 \times (0{,}009 - 0{,}008\,4)}} \right) = 0{,}73\ \text{K}$$

This gives a revised base temperature of

θ_b = 22 – 2 – 5 – 0,1 – 0,73 + 0,4 = 14,6 °C

When using Hitchin's formula Equation (E.51) is dealt with in the same way as the latent component above. If working with daily or hourly outdoor temperatures the individual differences (θ_o – θ_b) and (θ_o – θ_r) would be determined separately and summed. Using Hitchin's formula the solution is found by

$$D_m = 31 \times \left(\frac{18{,}5 - 14{,}6}{1 - e^{-0{,}71 \times (18{,}5 - 14{,}6)}} - \frac{0{,}75 \times (18{,}5 - 22)}{1 - e^{-0{,}71 \times (18{,}5 - 22)}} \right) = 122\ \text{K - d}ay$$

Giving the energy consumption

$$F = \frac{24 \times 25 \times 122}{3{,}0} = 24\,400\ \text{kWh}$$

Comparing this with the solution in example 1 shows the savings due to heat recovery. It is possible to conduct the analysis for sensible-only recovery by ignoring the adjustment for latent load, i.e. the base temperature in example 1 would be entered into the degree-day calculation of example 2.

Note that for fan coils (see E.3.1.3) the effects of heat recovery are bound up in the base temperature calculation (Equation (E.60)). The problem with fan coils is that they re-circulate air over the cooling coil irrespective of outdoor conditions, and they cannot make use of free cooling of fresh air systems.

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Annex F
(informative)

EDR Verification of building and installation calculation methods

F.1 Introduction

F.1.1 General

THE Energy Diagnosis Reference (EDR) is a methodology to verify the performance of calculation procedures. This annex describes this methodology to analyse energy calculation procedures. This methodology should at least include the following aspects:

- Thermal characteristics of the building, heating installation and hot water supply, air-conditioning installation, ventilation systems and/or natural ventilation, lighting installations, position and orientation of building, passive solar systems, shading devices and the indoor climatic conditions.
- Take into account the positive effects of natural lighting, installed renewable energy sources, electricity produced by CHP and/or collective heating and cooling systems.
- Adopt the categories of buildings as classified.

Different methodologies may be applied at national level provided these are in accordance with the requirements laid down in the EPBD.

F.1.2 Brief description of EDR

By means of EDR, calculation methods are evaluated concerning comprehensiveness and reliability with regard to calculating energy consumption. First to be decided is whether or not the model (with regard to the alleged functions) provides all necessary data for calculating the remaining energy consumption, in other words: the energy consumption after the energy saving measures have been taken. For this particular purpose two tools have been developed: a functionality matrix and masks.

Subsequently, the results of the model are tested with regard to reliability by means of representative cases (energy as well as economic cases). If the results of a method are based on complete information and fall within certain ranges of reference values, the method is entitled for a quality certificate (attest).

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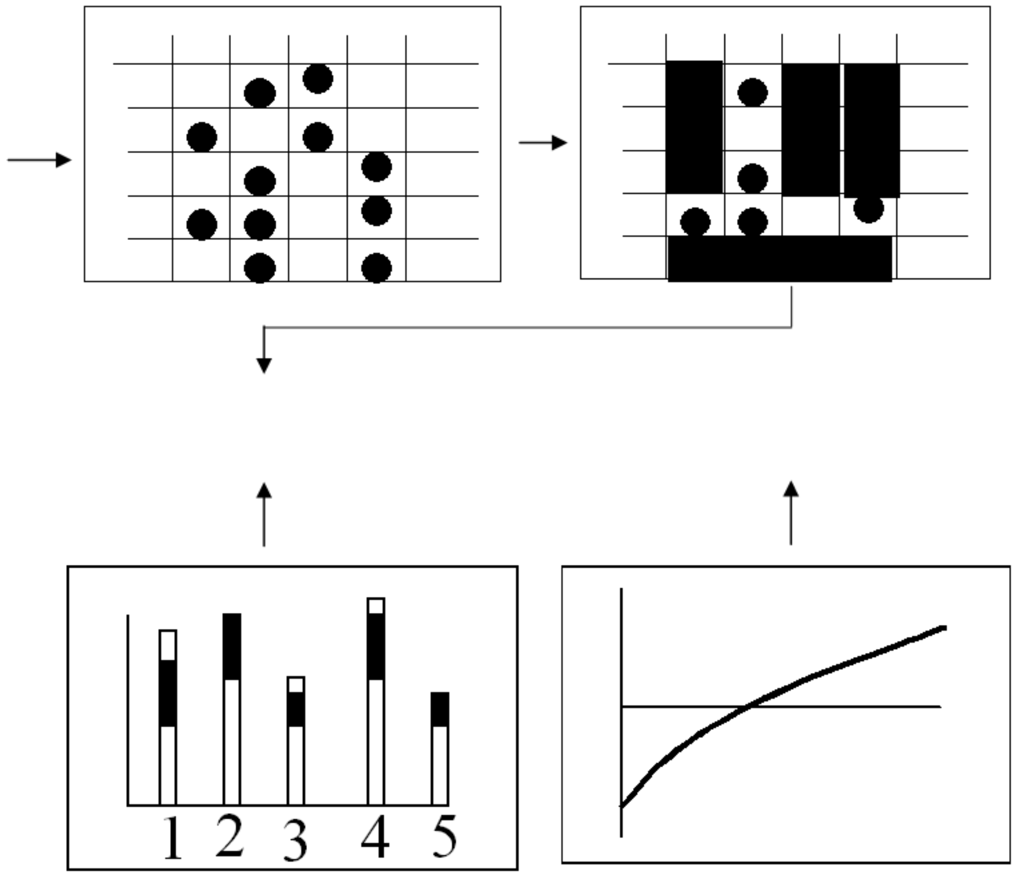

Key (numbering is not related to the figure for Energy cases, i.e. key 7)

1 functionality matrix

2 masks

3 input energy calculation method

4 system boundaries EDR

5 testing procedure

6 EDR-validated method

7 energy cases

8 economic cases

Figure F.1 - Scheme: How does EDR work

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The EDR uses the following elements:

- The functionality matrix. This is the verification framework for deciding whether the method can carry out the alleged functions. These include the scope (type of buildings), relevant energy flows related to the building and installations with regard to heating, cooling, and lighting, as well as the various physical entities related to the different energy flows.
- The masks. These include those energy flows per type of building, which contribute substantially to the energy performance. Masks and functionality matrix brought together make it possible to decide whether the method (on the basis of complete information) is able to pronounce upon the expected energy performance or possible energy savings.
- Energy cases. These include reference values applicable to various types of buildings by means of which the reliability of the calculation results of a method can be ascertained. These reference values consist of calculation results, measurement data or a mixture of both
- Economic cases. These provide reference values with regard to investment costs, interest rate, price of energy, maintenance costs, subsidies, etc., on the basis of which payback periods and net actual value can be determined.
- Testing procedure. Firstly, the comprehensiveness of the input data, and secondly the results are tested. Subsequently, these results are compared with the reference values from test cases and acceptable ranges. Finally, the EDR system decides whether the proposed calculation method satisfies the quality requirements.

F.1.3 How to obtain the attest (quality certificate)

It is intended that suppliers of calculation methods will have the opportunity to obtain a quality mark for their methods from an accredited certification body. The certificate will include an objective assessment with regard to the effectiveness of the proposed method based on the output of the testing procedure through EDR. For this purpose the test criteria of the EDR will be laid down in a so-called ASD (Assessment Directive for energy calculation methods for buildings) it will be possible to refer to this ASD with regard to tenders as well as legislation or other policy instruments.

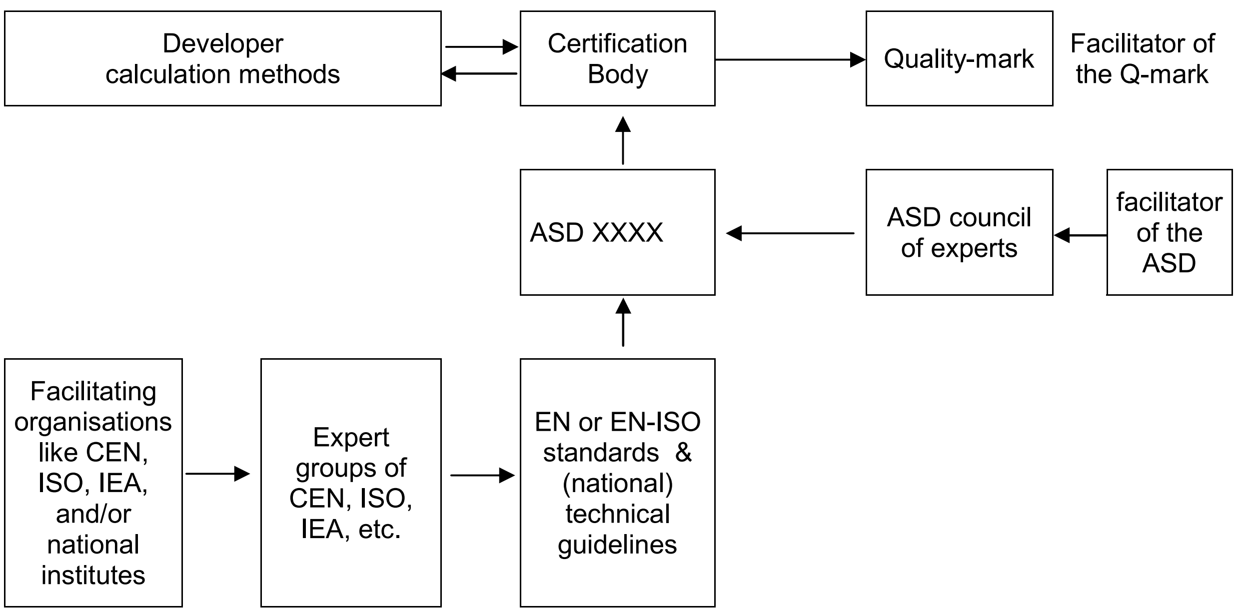

Figure F.2 - How to obtain a ADS certificate with the possible quality-mark

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F.1.4 Importance of EDR

The development of a standard methodology for the independent testing of existing calculation methods offers significant advantages to legislators and market parties such as clients, consultants, developers and suppliers of energy service companies.

- The use of reliable methods ultimately will lead to lower energy consumption or an increased use of renewable energy sources. By allowing the use of various calculation methods, the set-up of the construction industry and the actual design practice as well as initiatives therein are clearly taken into account. In the context of its own policies and legislation (laws, financial instruments) authorities can now refer to validated energy specification methods.
- Future building owners and operators are in a better position to taking well-founded investment decisions, based not only on investment costs but also on operating costs. They can demand the use of certified models when calling for tenders.
- Consultants may use their own methods, developed on the basis of their own specific position in the construction process and their own specific expertise.
- Developers of calculation methods will quickly improve and/or complete these systems if they deviate from the standard quality.
- Providers of energy services, finally, will have better sales arguments at their disposal.

With the introduction of a tool for testing existing calculation methods, those already disposing of a standard calculating method may use EDR to permit the use of equivalent methods. A method validated by EDR is implicitly in accordance with the requirements of the EPBD. The evaluation method based on European expertise and practices thereby directly promotes harmonisation. For companies in the supply industry, the construction and installation sector and/or energy sector, operating at an international level this might offer substantial advantages.

F.1.5 Relations: EDR, CEN, ISO, IEA HVAC BESTEST etc.

Only few relevant IEA-HVAC BESTEST cases are developed and available yet ([11],[12]). More cases (unfortunately not all) are under development within IEA Task 34/43 HVAC BESTEST [13]. When HVAC BESTEST cases come available also calculation methods that pass these HVAC BESTEST cases come available and these can be used to determine reference values to test cases which could be described in further to develop CEN Standards.

The EDR offers a complete set of cases for energy demand validation using IEA BESTEST and CEN and ISO Standards . Validation cases as defined in various EN Standards are defined on fewer aspects than the aspects considered in the EDR (example: ventilation, setpoint, window orientation). In standards like EN 15255, EN 15265, EN ISO 13791, EN ISO 13792 test cases are defined on combinations of different aspects.

F.2 Method description

F.2.1 General

The EDR based verification method contains:

1. functionality matrix (FM)
2. masks
3. reference/test cases
4. results of reference cases should fall within certain ranges

The testing procedure is given in Figure F.3. It is important it is checked the calculation method contains all relevant functions.

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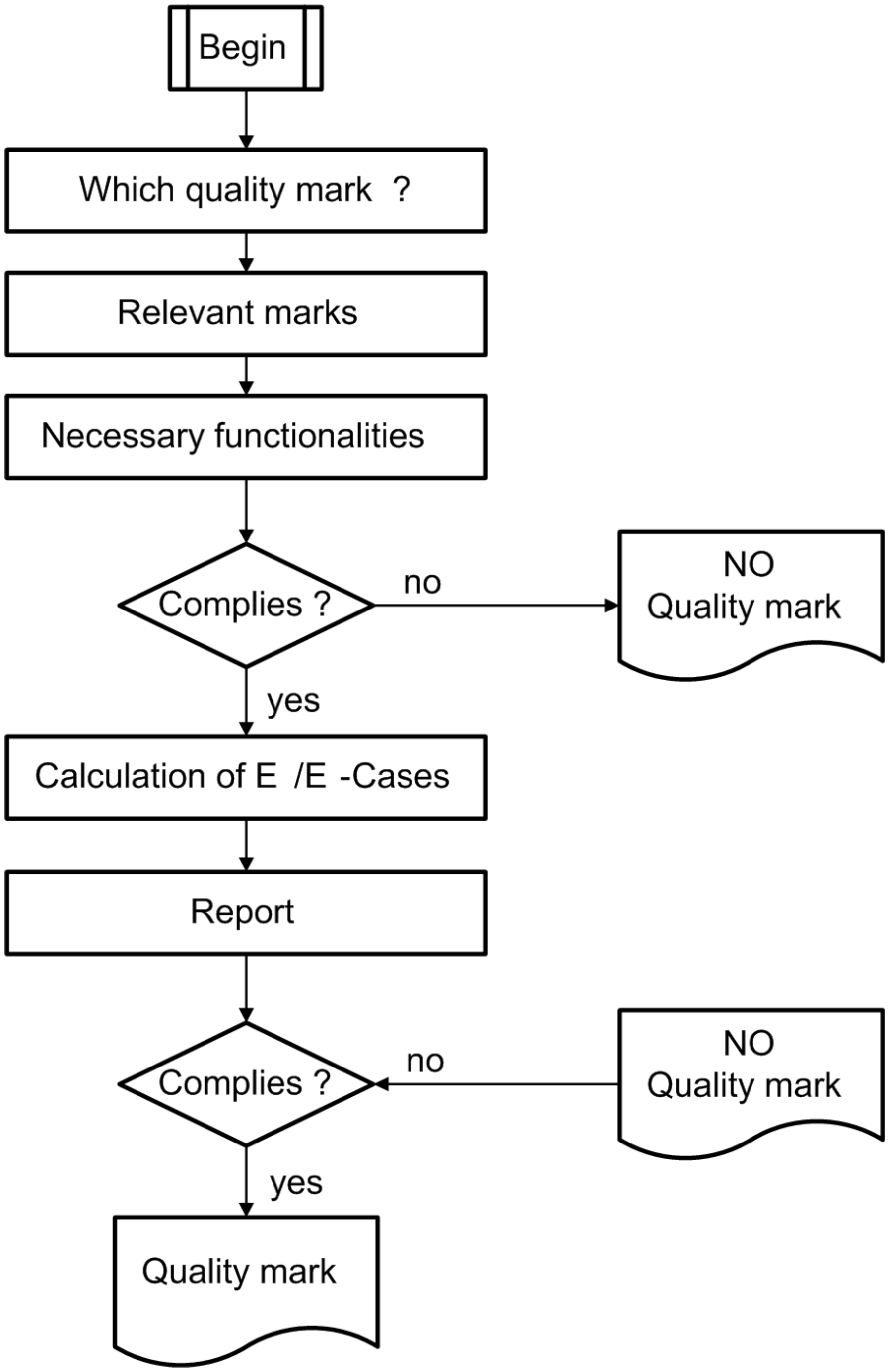

Figure F.3 - Testing procedure EDR

F.2.2 Functionality Matrix

The aim of the Functionality Matrix (FM) is to fix the functionalities of a calculation method for the energy consumption of a building. By completing the FM for a calculation method it is determined:

- for which building function(s) is the method appropriate;
- for which system function(s) is the method appropriate;

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- for which sub models is the method appropriate;
- which energy flows are taken into account;
- which parameters play a role;
- which method is used to determine these parameters

The FM has 3 levels:

- level 1, system functions (heating, cooling, humidification, dehumidification, warm tap water, lighting, transport of persons) and sub models (demand or load, supply, distribution and generation);
- level 2, energy flows;
- level 3, properties of parameters (not present/hard-coded/user given/calculated and constant/switched/controlled) per energy flow.

An example of the FM is given in F.2.6.

Within this informative annex an outline of mechanisms that influence the energy consumption of airconditioning systems is given. Using this relative global outline the FM can be used to be very specific in a more (general) structured way about the requirements of an applicable (to be tested/verified) calculation method.

F.2.3 Masks

Masks are used to specify a set of test cases that should be calculated (grouping/selection mechanism).

Within the certification procedure the masks are formally addressed.

In F.2.7 an example of a Mask is given. The relevant test cases are specified.

F.2.4 Reference/Test cases

Examples of an EDR reference/test cases have to be described. The described tests are part of the procedures which have to be performed according to Mask given above.

F.2.5 Results of reference cases should fall within certain ranges

Within the EDR, reference values for the energy demand cases are determined using a BESTEST passed computer code (these programs are assumed 'EDR-'reliable/accurate because they passed the BESTEST), see F.3. For energy demand of HVAC systems (generation) within the EDR so-called ***qualitative reference values***, are used, meaning the order of results of the different tests is given (example: test 3 has the lowest energy consumption, test 1 has an in between energy consumption and test 2 has the highest energy consumption) but not the reference results (including a bandwidth) of every separate test case (example: test 3 has an energy consumption in between x MJ and y MJ). Qualitative reference values are used because no reliable (= according to EDR approach: BESTEST approved) calculation methods are available to determine the reference values.

F.2.6 Example of the Functionality Matrix

Table F.1. Shows level 1 of the FM (system functions and submodels for all building functions)

Table F.2 gives an example of level 2 of the FM for the system function “Heating”

Table F.3 gives an example of level 3 of the FM for the energy flow “Transmission” of sub model “Demand”, that belongs to system function “Heating”

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Table F.1 - Level 1 of the FM (system functions and sub models for all building functions)

Page number	System function	Submodel	Building functions: All	Dwellings	Offices	Schools	Health care – clinical	Health care – non-clinical	Hotel, restaurants, bars	Shops	Meeting rooms	Sport	Accomodation	Prison	Dual functions	Further information on page
1.	Heating	1. Demand														1.1.
		2. Supply														1.2.
		3. Distribution														1.3.
		4. Generation														1.4.
2.	Cooling	1. Demand														2.1.
		2. Supply														2.2.
		3. Distribution														2.3.
		4. Generation														2.4.
3.	Humidification	1. Demand														3.1.
		2. Supply														3.2.
		3. Distribution														3.3.
		4. Generation														3.4.
4.	Dehumidification	1. Demand														4.1.
		2. Supply														4.2.
		3. Distribution														4.3.
		4. Generation														4.4.
5.	Hot tapwater	1. Demand														5.1.
		2. Supply														5.2.
		3. Distribution														5.3.
		4. Generation														5.4.
6.	Lighting	1. Demand														6.1.
		2. Supply														6.2.
		3. Distribution														6.3.
		4. Generation														6.4.
7.	Vertical transport of persons	1. Demand														7.1.
		2. Supply														7.2.
		3. Distribution														7.3.
		4. Generation														7.4.

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Table F.2 - Example of level 2 of the FM: for the system function "Heating"

Submodel	Energy flow	Give cross when present on level given:			Parameters see page:
		Room	Zone	Building	
1. Demand (heat balance)	Transmission losses				1.1.
	Ventilation losses				
	Infiltration losses				
	Internal heat production				
	External heat production (solar)				
	Storage of heat				
2. Supply	Heat supply				1.2.
	Heat losses				
	Heat production				
	Auxiliary energy				
3. Distribution (inside building)	Distribution-energy				1.3.
	Heat losses				
	Heat production				
	Heat recovery				
	Storage of heat				
	Auxiliary energy				
4. Generation	Boilers				1.4.
	Gas-driven heat pump				
	Electrical-driven heat pump				
	Absorption heat pump				
	Power co-generation				
	External supply of heat				
	Electrical heating				
	Solar				
	Short tem heat storage				
	Long tem heat storage				

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Table F.3 - Example of level 3 of the FM: for the energy flow "Transmission" of sub model "Demand", that belongs to system function "Heating"

Variables per energy flow	Not present	Properties								
		Fixed in method		User given to method				Calculated in method		According to standard …
		Constant	Switched	Constant	Array-values	Switched	Controlled	One time	Each time interval	
Transmission losses										
Vertical walls:										
- U-value										
- R-value										
- Surface heat transfer coef.										
- Cold bridges										
- Area										
Windows:										
Glass:										
- U-value										
- R-value										
- Surface heat transfer coef.										
- Cold bridges										
- Area										
Frame work:										
- U-value										
- R-value										
- Surface heat transfer coef.										
- Cold bridges										
- Area										
Floors:										
- U-value										
- R-value										
- Surface heat transfer coef.										
- Cold bridges										
- Area										
Roofs:										
- See floors, vertical walls										
(internal) walls:										
- See floors, vertical walls										
Conditions:										
- Indoor temperature										
- Outdoor temperature										
- Temperature next room										

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F.2.7 Example: Mask – generation of cold

Table F.4 - Selection of reference/test cases on level 2 of the Functionality Matrix, system function cooling

Submodel	Energy flow	Give cross when present on level given:			Parameters see page:
		Room	Zone	Building	
1. Demand (heat balance)	Transmission losses				2.1.
	Ventilation losses				
	Infiltration losses				
	Internal heat production				
	External heat production (solar)				
	Storage of heat/cold				
2. Supply	Cold supply				2.2.
	Cold losses				
	Cold production				
	Auxiliary energy				
3. Distribution (inside building)	Distribution-energy				2.3.
	Cold losses				
	Heat production				
	Cold recovery				
	Storage of cold				
	Auxiliary energy				
4. Generation	Electric Compression Cooler			A.2.4.1	2.4.
	Absorption Cooler				
	External supply of cold				
	Evaporative cooling				
	Short term storage				
	Long term storage in an Aquifer			A.2.4.1	
	DX cooler			A.2.4.1	

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F.3 EDR Calculation method for reference values

F.3.1 Inroduction

This clause describes the calculation method for reference values for EDR for single-family dwellings. The method first entails determining the centre of the reference area and then the size of the area surrounding it (bandwidth).

F.3.2 Centre of the reference area

BESTEST accepted computer codes, called attested computer codes, have been used for the calculations. These programmes have been validated in previous international studies. The test cases for EDR single-family dwellings were calculated using both programs. The centre of the reference area is determined by averaging the results from both calculation methods.

F.3.3 Structure of the reference area

The reference area surrounding the average result of the calculated test cases should take the following aspects into account:

1. The accuracy of the calculation results from : attested computer codes as compared to 'the actual situation'. This number is referred to as **a**.

2. The accuracy that is required or permitted in consulting practice. This number is referred to as **b**.

Re 1. This accuracy is determined by comparing the test case to one of the BESTEST cases. The dispersion of the calculation results found in that study for the case in question are also deemed applicable to the current calculations.

If the mutual difference between the attested computer codes results exceeds the dispersion in the BESTEST case, the difference between the attested computer codes results is considered indicative of the dispersion.

Re 2. The accuracy required from a consultancy point of view depends on the effect of a certain measure on energy consumption, taking the payback time into account. The criterion is that inadequate calculation methods should fall short. A fixed margin of error of 20 % is taken as a starting point.

F.3.4 Size of the reference area (bandwidth)

Accuracy numbers **a** and **b** are calculated in the following manner:

If **b** is more than 1.5 · **a**, the size of the reference area will be 2 · **b**.

If **b** is less than 1.5 · **a**, the size of the reference area will be 2 · (1.5 · **a**).

If the constructed reference area includes unrealistic values as a result of the construction method used, it should be adjusted manually. For instance, the constructed reference area should be adjusted if it includes values that are the result of a boiler yield exceeding 100%. Likewise, an energy-saving measure should never result in negative energy saving. In that case, too, the reference area is adjusted accordingly.

Please note: In some cases, the reference values have been adjusted somewhat to ensure that calculation results from a number of calculation methods fall within the reference area. However, the reference area will be limited in the future.

Figure F.4 is a visual representation of the calculation method for reference values.

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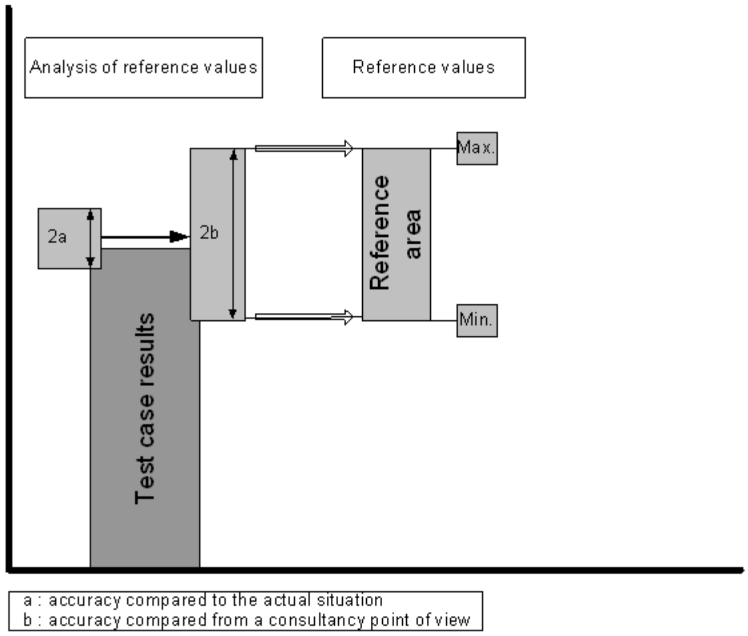

Figure F.4 - Structure of reference area

F.3.5 Examples

F.3.5.1 Example 1: A test case in which the effect of the U-value is assessed.

The difference between a case with a high U-value and one with a low U-value is 20 GJ.

a is estimated at 10 % (based on IEA work). Therefore, **a** = 2,0 GJ.

b is set at 20 %. This means that **b** = 20 % · 20 GJ = 4 GJ.

Is **b** > 1,5 · **a**? 4 > 1,5 · 2 = 3,0 GJ. The condition is met.

Consequently, the total width of the reference area equals 2 · **b** (= 8 GJ).

The minimum value equals (20-4 =) 16 GJ, the maximum value equals (20+4 =) 24 GJ.

F.3.5.2 Example 2: A test case in which the effect of the boiler yield is assessed.

The quotient for a test case with an HR107 boiler and a CR boiler equals 0,90.

a is estimated at 10 %. Consequently, **a** = 0,09.

b is set at 15 %, therefore **b** = 0,14.

Is **b** > 1.5 · **a**? 0,14 > 1.5 · 0.09 = 0,14. The condition is met.

The total width of the reference area equals 2 · **b** (= 0,27).

The minimum value equals 0,77, the maximum value 1,04.

The maximum value exceeds 1. This is unrealistic.

Consequently, the maximum value is manually set at 1.00.

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Annex G
(informative)

Example values for emission losses

G.1 Zones including several groups

When the rules leads to differentiate several groups, the calculation of the internal temperature must be done for every group. The internal temperature of the zone is obtained by pondering of internal temperatures of each group. The coefficients of weighting are the surfaces of every group.

The spatial variation depends on:

— the type of emitter,

— the ceiling height.

Table G.1 indicates the spatial variation class according to the type of emitter and the ceiling height.

Table G.1a — Spatial variations by type of heating emitter and the corresponding spatial variation class

Class of spatial variation	Emitter	Spatial variation for the local of less than 4 m under ceiling (K)	Supplementary spatial variation above 4 m of ceiling height (K/m)
A	Floor heating	0	0
B	Cassettes, tubes and radiate ceilings	0,5	0,2
C	Other emitters	0,5	0,4
Values are given as examples			

Table G.1b — Spatial variations by type of cooing emitter and the corresponding spatial variation class

Class of spatial variation	Emitter	Spatial variation for the local of less than 4 m under ceiling (K)	Supplementary spatial variation above 4 m of ceiling height (K/m)
A	Cooled ceilng	0	0
B	Other emitters	0	-0,2
C	Cooled floor	-0,4	-0,4
Values are given as examples			

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G.2 Control accuracy

The control accuracy depends on the emitter and the associated control system.

Table G.2 indicates for the different types of products the applicable standards and the relevant certifications

Table G.2 — Control accuracy

	Standard	**Certification**	**Control accuracy $\delta\theta_{vt}$ (K)**	
			heating	**cooling**
Direct electric emitter with built in controller	EN 60675	NF Performance category C	0,9	Not applicable
Thermostatic radiator valve	EN 215	CENCER	1,2	Not applicable
Individual zone control equipment	prEN 15500	EUBAC cert	Cah (defined in the standard and certified)	-Cac (defined in the standard and certified)
Other controller if emission can be totally stopped			1,8	-1,8
Other controllers if emission cannot be totally stopped			2	-2

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Annex H
(informative)

Calculation of latent energy demand

H.1 Presentation

This clause enables to calculate the indoor humidity pattern, and the related energy to dehumidify the air for a local system (humidifying and dehumidifying in central equipment are considered in EN 15241).

The humidity balance is defined in 13.2.

For the cooling coil condensation, a simple model is used based on the bypass factor. It assumes that one part of the air passing through the coil is not treated, and that the other part leaves at the dew point characteristics of the cooling coil temperature. This latter air flow is called here recirculated air flow.

The required recirculated air flow depends on the sensible cooling need and the cooling coil temperature. This one is assumed to equal to the cooled supply water+ 2K.

$$q_{\mathrm{m,recirc}} = \frac{Q_{\mathrm{sens}}}{c_{\mathrm{pa}}.(26-\theta_{sat})} \quad \text{(H.1)}$$

With:

$q_{\mathrm{m,recirc}}$	Required recirculated air mass flow rate	kg/s
Q_{sens}	Sensible cooling need	Wh
c_{pa}	Mass heat of dry air	J/kg K
θ_{sat}	Cooled coil temperature	°C

The behaviour of the local emitter depends on its control as follows:

Control 1. Non variable water flow (on/off; control on the fan).

In this case the cooled coil temperature is equal to the water inlet + 2K. Qmrecirc is calculated with Equation H.1.

Control 2. Variable water flow.

In this case m_{recirc} is calculated with the following formula

$$q_{\mathrm{m,recirc}} = A_{\mathrm{f}} \cdot 15 \cdot \rho_{\mathrm{ai}} \quad \text{[kg/s]}$$

With:

ρ_{ai} internal air density (kg/m3)

A_{f} floor area

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And θ_{sat} is calculated using Equation H.1.

NOTE The user will have only to define:

1) The cooled water temperature

2) The water control type

All other values are internally calculated when using an hourly based method.

H.2 Application for hourly calculation

On a given hour, 2 situations can occur:

1. At the start of the hour, the indoor humidity $w_{i,start}$ is higher than the saturation humidity for the cooling coil wisat. The condensation will occur, either during the whole hour either during one part of it.

2. At the beginning of the hour, the indoor humidity wistart is lower than the saturation humidity for the cooling coil wisat. The cooling coil will be dry at the beginning, but condensation can occur before the end of the hour.

For a duration of *t*,

$$x_{i,fin} = x_{i,start} + A\,(1 - e^{-B \cdot t}) \tag{H.2}$$

A and B are calculated as follows:

If the cooling coil is wet:

$$A_{wet} = (x_{entr} \cdot q_m + a_i + q_{m,recirc} \cdot x_{sat}) / (q_m + q_{m,recirc}) - x_{start}$$

$$B_{wet} = (q_m + q_{m,recirc}) / (V \cdot \rho_{ai})$$

If the cooling coil is dry,

$$A_{dry} = (x_{entr} \cdot q_m + a_i) / q_m - x_{start}$$

$$B_{dry} = q_m / (V \cdot \rho_{ai})$$

With:

x_i	indoor humidity	kg/kg dry air
x_{entr}	entering air humidity	kg/kg dry air
q_m	entering air flow	kg/s
x_{start}	indoor humidity at the beginning	kg/kg
a_i	internal humidity gains	kg/s
ρ_{ai}	Density of indoor air	kg/m^3
x_{sat}	saturated humidity at the cooling coil temperature	kg/kg

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Case 1. Cooling coil wet at the beginning (wiprev > wisat)

Check on possible transition

If the cooling coil was permanently wet, the value of the indoor air humidity at the end of the hour would be equal to:

$$x_{i,fin,check} = x_{i,start} + A_{hum} (1 - e^{-Bhum})$$

Case 1.1: If $x_{i,fin,check} > x_{i,sat}$; the cooling coil was permanently wet. The calculation is then:

$$x_{i,moy} = x_{i,start} + A_{hum} (1 - (1- e^{-Bhum}) / B_{hum})$$

$$Q_{lat} = 2501000 \cdot q_{m,recirc} (x_{i,moy} - x_{sat})$$

Case 1.2: If $x_{i,fin,check} < x_{i,sat}$, the cooling coil was dry at the end of the hour.

The time before cooling coil becomes dry (t_{hum}) is then equal to:

$$t_{hum} = - (\mathrm{Ln}(C_{hum})) / B_{hum}$$

With:

$$C_{hum} = 1 - (x_{i,sat} - x_{i,start}) / A_{hum}$$

The duration of the dry period is: $t_{dry} = 1 - t_{hum}$

$$x_{i,en,trans} = x_{i,start} + A_{hum} (1 - e^{-Bhum \cdot thum})$$

$$x_{i,moy,trans} = x_{i,start} + A_{hum} (1 - (1- e^{-Bhum \cdot thum}) / (B_{hum} \cdot t_{hum}))$$

$$Q_{lat} = 2501000 \cdot q_{m,recirc} (x_{i,moy} - x_{sat})$$

The final indoor room humidity is calculated by:

$$x_{i,end} = w_{i,f,sat} + A_{dry} (1 - e^{-Bdry \cdot Tdry})$$

Case 2. Cooling coil dry at the beginning ($w_{i,prev} < x_{i,sat}$)

Check on possible transition.

If the cooling coil was permanently dry, the value of the indoor air humidity at the end of the hour would be equal to

$$x_{i,fin,check} = x_{i,start} + A_{dry} (1 - e^{-Bdry})$$

Case 2.1 If $x_{i,fin,check} < x_{i,sat}$; the cooling coil was permanently dry. The calculation is then

$$x_{i,fin} = x_{i,start} + A_{dry} (1 - e^{-Bdry \cdot (1-thum)})$$

$$Q_{lat} = 0$$

Case 2.2 If $x_{i,fin,check} > x_{i,sat}$, the cooling coil was wet at the end of the hour

The time before cooling coil becomes wet (t_{dry}) is then equal to:

$$t_{dry} = - (\mathrm{Ln}(C_{dry})) / B_{dry}$$

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With:

C_{dry} = 1 - ($x_{i,sat}$ – $x_{i,start}$) / A_{dry}

t_{hum} = 1-t_{dry}

The calculation for the wet phase is:

$x_{i,moy,trans} = x_{i,start} + A_{hum}\,(1 - (1 - e^{-Bhum \cdot Tthum}) / (B_{hum} \cdot T_{hum})))$

$Q_{lat} = 2501000 \cdot q_{m,recirc}\,(x_{i,moy,trans} - x_{sat})$

The final indoor room humidity is calculated by:

$x_{i,fin} = x_{i,start} + A_{hum}\,(1 - e^{-Bhum \cdot Thum}))$

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Annex I
(informative)

Example Calculation of Seasonal Efficiency of Cold Generators and Chillers in Air Conditioning Systems

I.1 Introduction

For energy certification purposes, we need to be able to calculate the energy consumption of cold generators and chillers – along with other components of air-conditioning systems. In this annex the word “chiller” is used as a convenient shorthand for any cold generator used as part of an air-conditioning unit or system. This includes the “cold generator” within an air-conditioning package unit, for example.

It is convenient to describe the ratio between the annual cooling load placed on a chiller and its corresponding energy consumption as a “seasonal efficiency” or, more accurately “seasonal energy efficiency ratio”. However, this ratio is not simply a characteristic of the chiller, but also depends on a number of other factors that vary with the application.

Sometimes the installed cooling capacity will be greater than the peak cooling demand, and some systems will have more than one chiller, Seasonal efficiency calculations need to be able to accommodate both situations.

The “efficiency” of a chiller may vary significantly with the load placed on it and with the temperature to which heat is rejected (for air-cooled chillers, the ambient air temperature). For many chillers, the efficiency at part-load is less than at full load – though for others, the opposite is true. Generally, efficiency increases as heat rejection temperature falls. The two effects are usually of comparable size. For air-cooled equipment, the heat rejection temperature is closely linked to ambient air temperature but, for liquid-cooled equipment the temperature variations clearly depend on the source of the cooling fluid.

In some cases, the relationship between efficiency and these factors may be summarised as a continuous curve; in other cases there are discontinuities. The characteristic time constants of chillers and packaged air-conditioners are usually such that it is difficult to define efficiency meaningfully for periods much less than, say ten minutes while, for some equipment, the variations of performance with load can be difficult to represent over periods much in excess of an hour.

It follows that methods of representing the overall performance of an air-conditioning system must be subject to similar constraints. Indeed, some systems introduce additional discontinuities – for example, by switching to full fresh air and no chiller operation when outdoor temperatures are sufficiently low.

In principle, the chiller (or the complete air-conditioning system) can be modelled mathematically and its energy consumption calculated directly for each operating period (say, each hour). Alternatively, chiller energy consumption can be measured (or pre-calculated) over a range of operating conditions to produce a “performance map” that can be applied to each demand figure. Often, information is often not available in this degree of detail.

This annex sets out the theoretical background for this type of calculations, discusses how this can be applied with differing levels of information, and provides an illustrative example of its application.

I.2 Theory

I.2.1 The objective

To calculate the energy consumption of a chiller or set of chillers in an air-conditioning system gives some knowledge of:

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- the cooling demands placed on it (or them)
- the energy efficiency of the chillers at a number of part-load conditions

The same processes can be used to calculate the energy consumption of the complete system and can also be applied to heating systems.

I.2.2 Combination of load frequencies and part-load performance measurements

Over some period of time the cooling demand on a chiller is Q_C (kWh). During this period, energy E_C is used to meet this demand.

Efficiency is defined as Q_C/E_C. The inverse, Energy Input Ratio EIR is often more convenient to use.

Then $E_C = EIR \cdot Q_C$

Clearly, over some longer period of time the total consumption is simply the sum of the consumption during different time periods.[2]

$\Sigma E_C = \Sigma (EIR \cdot Q_C)$

and we can define an overall EIR as $\Sigma E_C / \Sigma Q_C = \Sigma (EIR \cdot Q_C) / \Sigma Q_C$

Note that, if we express the equation in terms of efficiency instead of EIR, the overall efficiency is the harmonic mean of the individual efficiencies. (That is the reciprocal of the sum of the reciprocals)

More generally, when Q_C is zero, there may still be an energy consumption. In this case EIR is infinite and efficiency zero (irrespective of the size of the no-load consumption). Denote such zero-load consumption as E_0.

$\Sigma E_C = \Sigma (EIR \cdot Q_C) + \Sigma E_0$

and overall EIR is $(\Sigma (EIR \cdot Q_C) + \Sigma E_0) / \Sigma Q_C$

EIR is a function of Q_C and of heat rejection temperature: for air-cooled chillers outdoor temperature.

This calculation can be carried out for each individual time step (say each hour), or the frequency of each level of demand (and temperature) during the period of interest may be determined. This is the basis of “bin analysis”. For example a “bin” might be defined as containing the number of hourly occurrences of a cooling load between 35 % and 40 % of the design cooling load of the building that are coincident with ambient air temperature between 24 °C and 26 °C.

Denote the frequency with which each condition occurs as F, and associate a value of EIR with each bin. Then

$\Sigma E_C = \Sigma (F \cdot EIR \cdot Q_C) + F \cdot E_0$

The frequencies and the demands may be further combined to generate *demand weightings*, $f_w = F \cdot Q_C$

$\Sigma E_C = \Sigma (f_w \cdot EIR) + F \cdot E_0$

---

[2] For simplicity subscripts denoting the range over which summations are made are not shown.

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I.2.3 Seasonal performance indices

Seasonal performance indices can be defined for individual chillers or for sets of chillers operating in sequence. The American Integrated Performance Load Value (IPLV) [2] and proposed Euopean Seasonal Energy Efficiency Ratio (ESEER) [6] are examples of indices for individual chillers.

More generally, an ***overall EIR***, can be calculated for any given period as:

$EIR_{sea} = (\Sigma\ (F \cdot EIR \cdot Q_C) + F \cdot E_0)\ /\ \Sigma\ Q_C$

or

$EIR_{sea} = (\Sigma\ (f_w \cdot EIR) + F \cdot E_0)\ /\ \Sigma\ Q_C$

The overall seasonal EER (SEER) is the reciprocal of this.

The ratio between SEER and full-load EER is sometimes known as the “mean partial-load factor”

PLV_{av} = SEER/EER

I.2.4 Calculation of representative EIRs

Each bin has an associated EIR value. However, each bin has a finite size (say from 45 % load to 55 % load) within which there may be significant variations of EIR –especially if the bin size is large.

Strictly speaking, the EIR for each bin should be calculated from the distribution of loads within the bin in the same way as the seasonal figure is calculated from the bin data. Pragmatically, it is rarely possible to do this, and it is necessary to select EIR by, for example, taking a value that represents the mid-point of the bin range, or the average of the two bounding conditions.

I.2.5 Multiple chillers

For systems with multiple chillers, a combined EIR value must be calculated for each bin, representing a combined EIR of all the operating chillers (and the load conditions on each – for example, one chiller at full output and a second one at 50 % of full output). Note that this is not obtained by averaging EERs (unless the harmonic average is used) but by adding consumptions and determining the combined total consumption. The later worked example illustrates this**.**

I.2.6 Calculations for systems

The same theory may be applied to complete systems. In this case, it is convenient to subdivide the energy into two classes: auxiliary energy W (kWh) that is used principally for energy transfer (fans and pumps) and for controls; and direct consumption, E (kWh) (used for the generation of heating or cooling by chillers, boilers and their ancillary equipment). Direct consumption depends not only on the on the efficiency of conversion of fuel or electricity to heat (or cold) but also on the efficiency of distribution of heat (or cold).

System EIR may be defined as the ratio between the total cooling (or heating) demands in the spaces being conditioned and the energy consumed by chillers or boilers. Auxiliary energy will be calculated separately (for example as a multiple of installed fan power and operating hours). For packaged units, EIR may more conveniently be defined to include both direct and auxiliary energy consumption.

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I.3 Practical application

I.3.1 Background

Ideally the theory would be applied to data that are specific to the building and system under consideration, to the actual or expected pattern of use, and to the local climate. This requires detailed information on chiller performance[3] over a wide range of conditions, and detailed estimates of building cooling demand – from detailed simulation, for example. Such detailed information is rarely available, and simplifications have to be made. This section describes how this may be done in ways that are consistent with various levels of available data.

Simplifications fall into two related areas: chiller information and bin definitions.

Simplifications inevitably reduce the resolution of the calculation. However, the basic data will always be uncertain to some degree, and high theoretical resolution does not necessarily mean more precise or reliable results. For many applications, consistency of approach will be more important than fine degrees of apparent accuracy. For energy performance certification purposes, standardized assumptions will be necessary.

Figure I.1 shows a hierarchy of simplifications. The choice depends on data availability and on the calculation effort that is thought justifiable for a particular purpose.

---

[3] Either from manufacturers' data or by detailed modelling. The characteristic time constants of chillers and packaged air-conditioners are usually such that it is difficult to define efficiency meaningfully for periods much less than, say ten minutes, while for some equipment, the variations of performance with load can be difficult to represent over periods much in excess of an hour.

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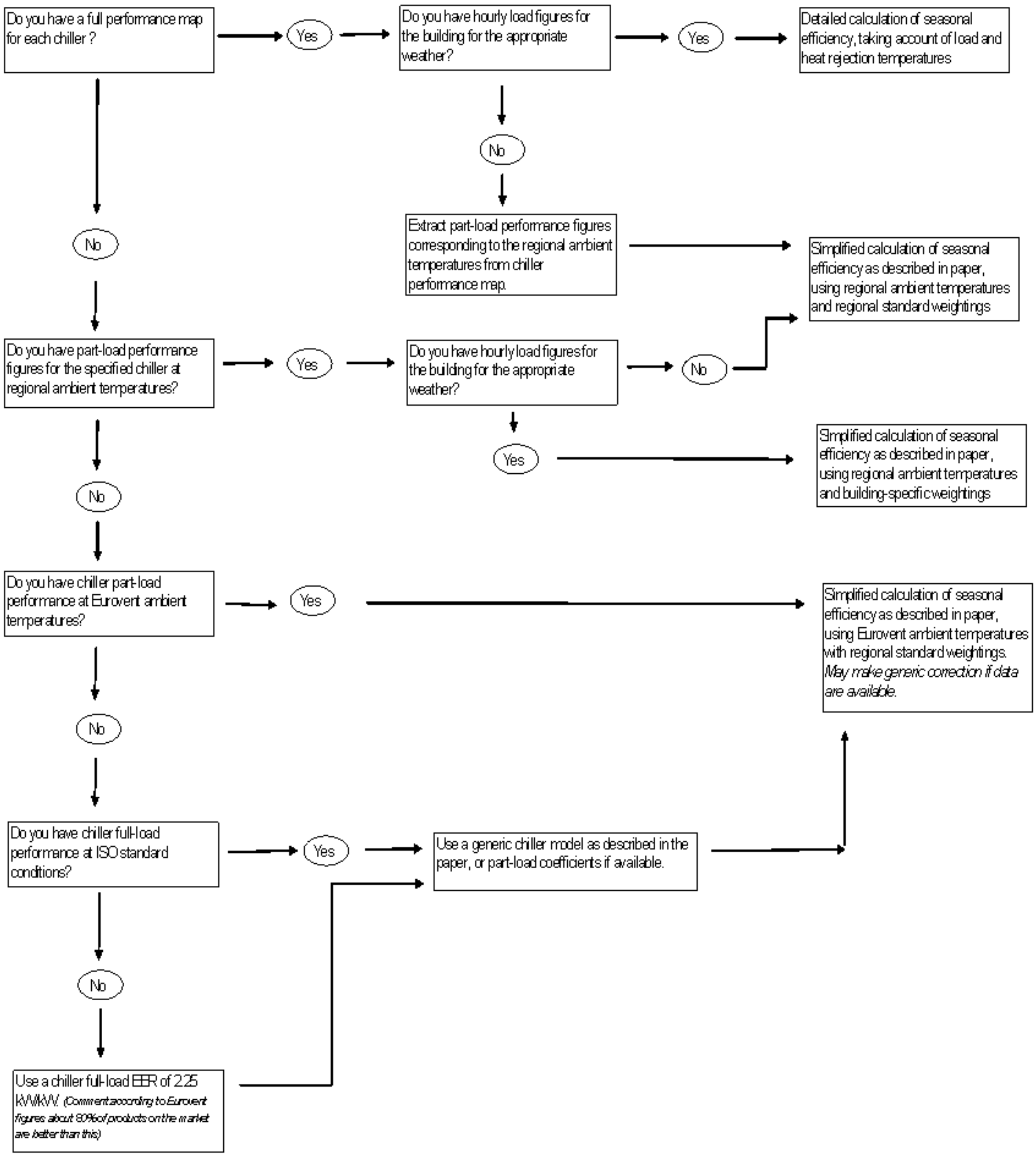

Figure I.1 - Hierarchy of simplifications

I.3.2 Simplification of load frequency data

When building cooling demands are calculated on an hourly basis, the production of hourly bins (of joint demand and ambient temperature) is straightforward. Alternatively the calculated hourly loads can be used directly with chiller performance data if this is available. For standardised energy calculations, standardised weather data are obviously required.

When cooling demands are calculated monthly or annually a standardised set of load frequencies (or demand weightings) can be used. These could be generated in several ways, of which computer simulations of characteristic buildings under standardised weather conditions are perhaps the most satisfactory. Clearly, these standardised demand frequencies will best match those of the building

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under consideration if the simulations are for a similar building with a similar air-conditioning system, similar pattern of use and located in a similar climate.

Since the efficiency of air-cooled chillers depends on ambient air temperature, the demand frequencies should ideally be accompanied by associated ambient air temperatures. Failing this, it may be possible to estimate ambient temperatures from the demand figures – the two variables are usually correlated, though imperfectly. The implications of inappropriate temperatures are discussed in the section dealing with chiller performance simplifications.

The EECCAC project [7] derived load frequency distributions from building energy simulations for a wide range of system types at three sites: London, Milan and Seville. As Roger Hitchin [8] points out, the loads placed on the chiller system can vary with the type of cooling system installed, as well as with weather and building design and use.

I.3.3 Approximation of chiller performance data

Ideally the chiller performance should be evaluated at each time step of the cooling demand calculation. This requires a full (and reliable) performance map for the chiller(s) that are installed or proposed. Some manufacturers are able to provide this information, but it is not always publicly available nor independently verified.

If a full chiller performance map does not exist, measurements under standard “full-load” conditions and some part-load conditions may be available.

The EECCAC project concluded that a minimum of four demand weightings (each associated with an ambient temperature representing the mean value associated with these load conditions) was necessary to distinguish between chillers of differing performance. The US IPLV rating also uses four part-load conditions. Values were determined by the EECAC project for a range of climates and systems. For a fan-coil system located in London, they were:

Table I.1 - Example demand ratings

Relative frequency of occurrence (% of operating hours)	Cooling demand as percentage of design load	Relative demand weighting (frequency of occurrence x proportional demand)	Mean ambient temperature associated with demand
60,8 %	25 %	42,3 %	16,1
34,9 %	50 %	48,5 %	20,1
4,2 %	75 %	8,7 %	24,6
0,2 %	100 %	0,5 %	27,6

There are several existing and proposed sources of chiller performance information under conditions other than full load standard test conditions. For example Eurovent is proposing to include part-load tests into its certification programme; CEN/TS 14825 describes part-load tests, as does Italian standard UNI 10963 [9]. ARI standard 550.590-98 [2] defines four part-load tests required for the ARI IPLV rating; and the EECCAC project proposes European tests equivalent to the ARI tests.

The particular temperature conditions prescribed for the part-load tests will not be appropriate for all climates and cannot apply to each individual chiller in a multi-chiller installation with the chillers operating in sequence. Manufacturers may be able to provide sufficient information to interpolate to appropriate national or regional temperatures, where these are known. If this is not possible, data relating to the “wrong” temperatures has to be accepted. A limited number of calculations on the impact of using ESEER temperatures – which reflect Southern European conditions – in the UK suggest that the error will be of the order of 5 % to 10 %. As it will always be in the same direction (better efficiencies, reflecting the lower summer temperatures in the UK), the impact on equipment

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selection choices is probably not great. (With the caveat that chillers with free-cooling capability might be under-valued).

Sometimes only full-load performance under standard test conditions is known. For many chillers and packaged units this information is available from Eurovent Certification, most readily accessed via www.ukepic.com. Under these conditions, part-load performance values have to be estimated using generic models.

There are several existing procedures for doing this:

The procedure previously used in the 1989 version of ARI standard 210/240 [2] (current ARI standards require part-load testing). This relates PLV (the ratio of part load EER to EER at design outout) defined earlier) to the "part load ratio" PLR (the ratio between the average load on the chiller to its design output) as:

$PLV = 1 – C_d.(1- PLR)$

C_d is to be specified by the manufacturer or taken by default as 0,25

Italian standard UNI 10963 uses a slightly more complex relationship:

$PLV = PLR/(a.PLR + b)$

This requires the knowledge of performance at two test conditions to determine the coefficients

DIN V18599-7 [10] tabulates values of PLV against PLR for seven types of air-cooled cold generator (including packaged units) and seven types of water-cooled chiller. It also provide correction factors for different flow and return operating temperatures and different outdoor temperatures (for air-cooled units).

It should be remembered that additional auxiliary energy will be required for heat rejection for water-cooled chillers.

When no information is known about the chiller (for example in existing systems or in the initial stages of design), it will be necessary to use a default value for full-load performance, and apply the generic part-load estimates mentioned above. From Eurovent certification data approximately 80% of chillers have EER values above 2,25, so this could be a suitable default assumption, An equivalent value for water-cooled chillers would be 3,40. (Additional auxiliary energy for heat rejection equipment will also be required for water-cooled chillers).

For packaged units, the boundary between energy label classes E and F is a suitable default. This is:

Air to air

Split and multi split: 2,40

Single-duct packaged units: 1,80

Other packaged units: 2,20

Water to air

Split and multi split: 2,50

Other packaged units: 3,20

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I.4 Illustrative example of estimation of seasonal EER

I.4.1 General

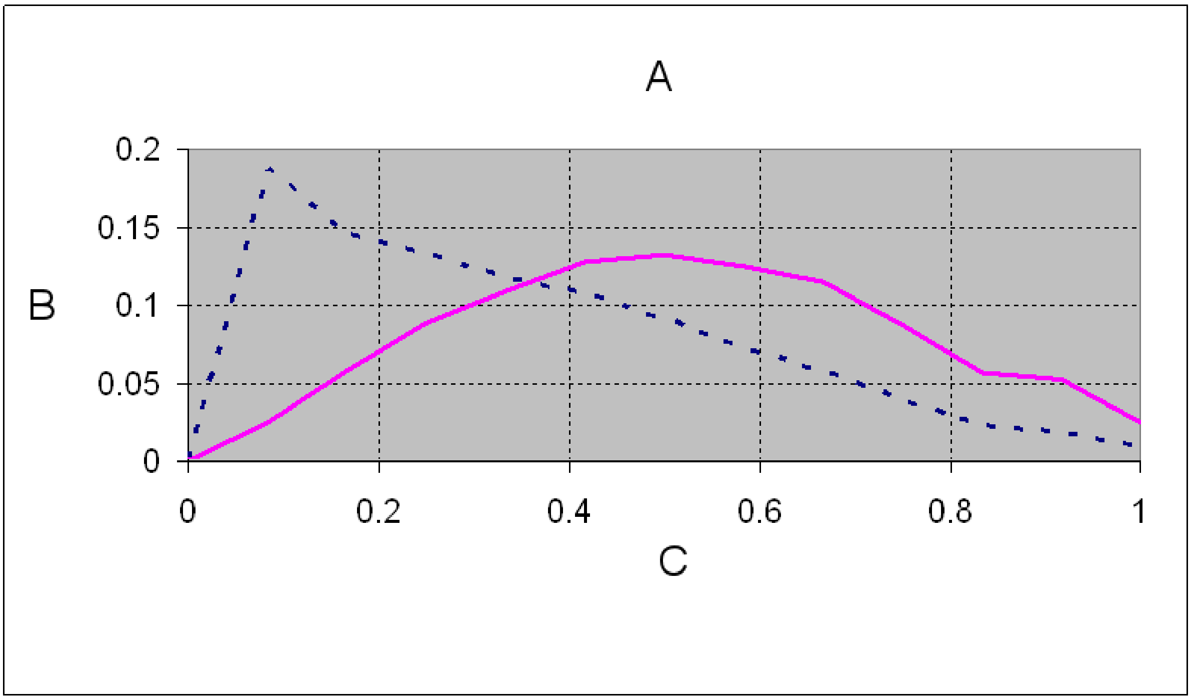
This example illustrates the processes of:

- determining load frequency distributions
- determining multi-chiller EER from data for single chillers
- mapping the chiller rating data onto the load frequency distribution
- estimating seasonal EER

The general principles illustrated here are applicable to any set of chiller or boiler part-load conditions and sizing. In practice it may be convenient to build the process into software such as a spreadsheet. Suggestions for data sources are made at the end of this section.

I.4.2 Load frequency distributions

Either from load calculations for a specific building, or a standard national or regional assumption, we have a *frequency* distribution of different fractional cooling loads (where 1 = the peak load). For energy calculations, we need to convert this into an energy demand distribution, by multiplying the frequency by the part-load fraction. This generates a *load-weighted* distribution. In the chart below, the frequency distribution for an office in the UK is shown as a broken line, and the load-weighted distribution as a solid line.



Key:
A Normalised load distributions
B Normalised frequency
C Fractional load
Frequency
Load weighted

Figure I.2 - Normalised load distribution

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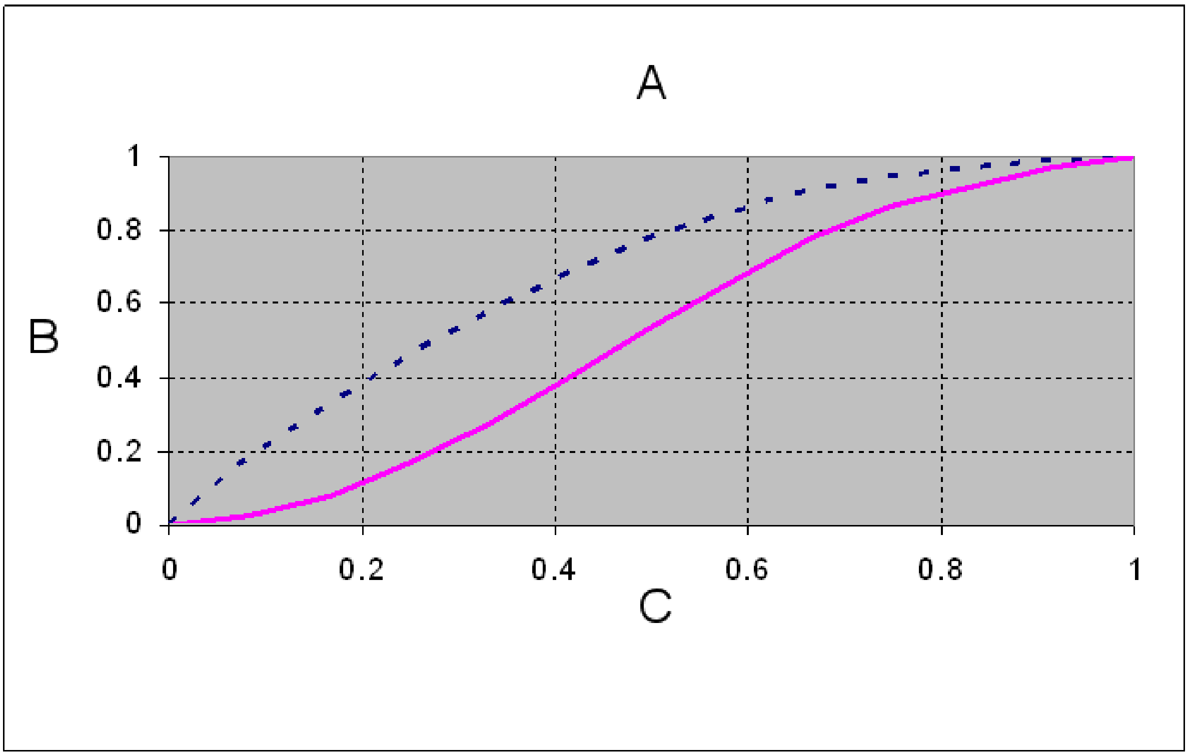
For later stages in the calculation, it is convenient to convert the load-weighted distribution into a cumulative distribution as shown below.



Key:
A Normalised cumulative load distributions
B Normalised frequency
C Fractional load
Frequency
Load weighted

Figure I.3 - Normalised Cumulative Load Distributions

I.4.3 Combined chiller performance

In the example, we assume two identical chillers, each capable of providing 75% of the peak load, operating in sequence, each with part-load performance values of

Table I.2

% part-load	EER
100	2,8
75	2,7
50	2,5
25	2,0

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With two chillers we therefore have eight combinations for which we can calculate an EER. [4] These are shown in the table below. Because the example system is oversized, three of the conditions are for loads in excess of the peak load.

When both chillers are operating, the combined EER is determined by dividing the relative demand on each chiller by the appropriate EER, summing the total consumption and dividing the result by the total demand.

Thus with 1 chiller operating at 100 % output and the other at 25 %, we have

Chiller 1 consumption = ½,8 = 0,357

Chiller 2 consumption = 0,25/2 = 0,125

Total consumption = 0,482   Combined EER = 1,25/0,482 = 2,59

Table I.3

Chiller 1 fractional load	Chiller 2 fractional load	Chiller 1 EER	Chiller 2 EER	Combined EER
0	0	N/A	N/A	N/A
0,25	0	2,0	N/A	2,00
0,5	0	2,5	N/A	2,50
0,75	0	2,7	N/A	2,70
1	0	2,8	N/A	2,80
1	0,25	2,8	2,0	2,59
1	0,5	2,8	2,5	2,69
1	0,75	2,8	2,7	2,76
1	1	2,8	2,8	2,80

I.4.4 Mapping the chiller ratings on to the load frequency

Each of these chiller rating points maps onto the building's load frequency distribution, as shown in Figure I.4 as solid vertical lines.

[4] If the chillers do not operate in sequence, the procedure has to be modified accordingly.

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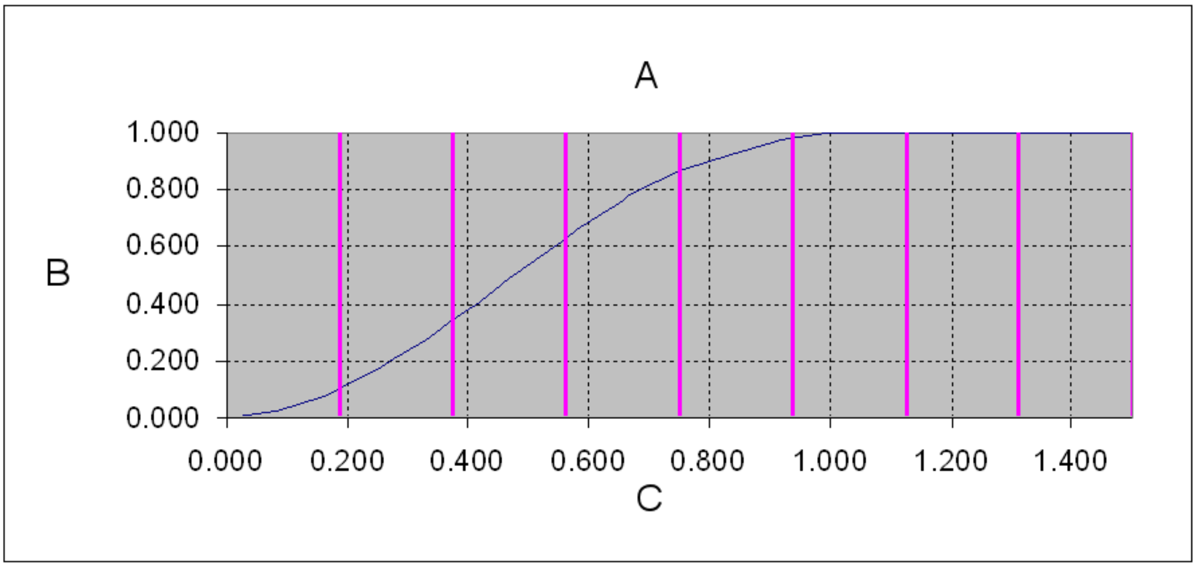
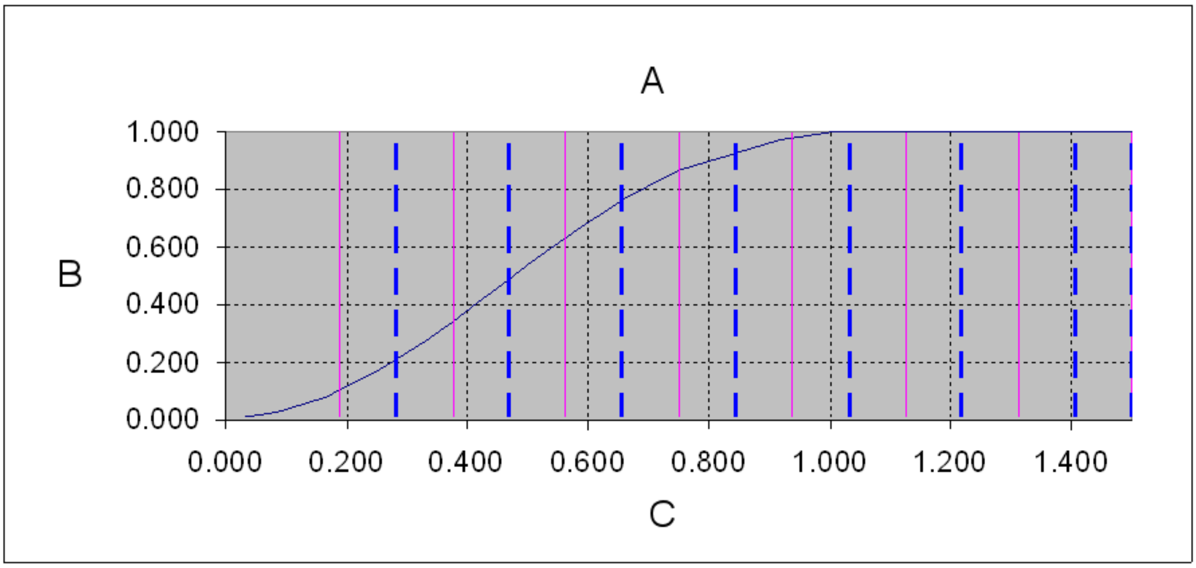


Key:
A Rating points
B Cumulative load frequency
C Fractional load

Figure I.4

We need to associate each rating point with some proportion of the cooling energy demand. We do this by first finding the *building* fractional load that is midway between each chiller rating point. This divides the frequency distribution into a number of bands, each of which contains one chiller rating point. (The lowest band has a lower limit of zero building load). This is illustrated in Figure I.5, where the band boundaries are shown as broken vertical lines.



Key:
A Energy bands
B Cumulative load frequency
C Fractional load

Figure I.5

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The weighting for each rating point is the difference of the cumulative loads at each of the two boundary conditions (It may be necessary to interpolate between known values of the load frequency distribution to determine these values).

We then have to divide each demand weighting by the appropriate EER to calculate the total consumption and can then derive the seasonal EER.

The process is illustrated in the table below.

Table I.4

Chiller 1 fractional load	Chiller 2 fractional load	Building fractional demand	Demand weighting for the demand level	Combined EER	Energy consumption
0	0	0	0	N/A	0
0,25	0	0,188	0,104	2,00	0,0520
0,5	0	0,375	0,238	2,50	0,0952
0,75	0	0,563	0,290	2,70	0,1074
1	0	0,750	0,234	2,80	0,0836
1	0,25	0,938	0,115	2,59	0,0444
1	0,5	1,125	0,019	2,69	0,0071
1	0,75	1,313	N/A	2,76	N/A
1	1	1,500	N/A	2,80	N/A

Total (normalised) energy consumption = 0,3897

Combined seasonal EER = 1/0,3897 = 2,57

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I.5 Example for calculated part-load-values

(From [10], for Test-Reference-Year 05, Würzburg, Germany)

The final energy demand of the compression refrigeration units is calculated, using EER and PLV, in the equation:

$$Q_{\mathrm{C,f,electr}} = \frac{Q_{\mathrm{C,outg,a}}}{EER \cdot PLV_{\mathrm{av}}} \tag{I.1}$$

$$EER \cdot PLV_{\mathrm{av}} = SEER$$

EER	is the rated energy efficiency ratio of cooling in kW/kW (see Tables I.8 to I.11);
PLV_{av}	is the mean part load value, (see Table I.12);
$SEER$	is the seasonal energy efficiency ratio, in kW/kW;
$Q_{\mathrm{C,f,electr}}$	is the final energy demand of the compression refrigeration unit (electrical), in kWh.
$Q_{\mathrm{C,outg,a}}$	is the annual refrigeration energy output, in kWh.

Table I.5 - Types of partial-load control of water-cooled compressor-type refrigeration units evaluated by the index-value method

Water-cooled compressor-type chillers, type of partial-load control	
(1)	Piston or scroll compressor with two-point control (ON/OFF operation)
(2)	Piston or scroll compressor with multi-stage control (at least four power control stages in multi-compressor combinations)
(3)	Piston compressors with individual cylinder shut-down
(4)	Piston or scroll compressors with hot-gas bypass control
(5)	Screw compressors with valve control
(6)	Turbo-compressors with inflow choke control

Table I.6 - Types of partial-load control of air-cooled compressor-type chillers evaluated by the index-value method

Air-cooled compressor-type chillers, type of partial-load control	
(A)	Piston or scroll compressor with two-point control and buffer storage (ON/OFF operation)
(B)	Piston or scroll compressor with multi-stage control (at least four power control stages in multi-compressor combinations)
(C)	Screw compressors with valve control

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Table I.7 - Types of partial-load control of air-cooled room air-conditioning systems evaluated by the index-value method

Air-cooled room air-conditioning systems, type of partial-load control	
(D)	Two-point control for single-zone system, pulsed (ON/OFF operation)
(E)	Two-point control for multiple-zone system, pulsed (ON/OFF operation) (where applicable with exhaust-chamber control or individual cylinder shutdown)
(F)	Continuously variable control for single-zone systems, for example frequency-controlled/pulsed, with electronically-controlled expansion valve (inverter control)
(G)	Continuously variable control for multiple-zone systems, for example frequency-controlled/pulsed, with electronically-controlled expansion valve (inverter control)

Table I.8 - Default values for rated energy efficiency ratio of cooling *EER* for water-cooled chillers
(based on an analysis of products on the market – approximately 60 % of products are more efficient than these figures)

Refrigerant	Cooling-water input / output temperatures, °C	Chilled-water output temperature, °C	Mean evaporation temperature, °C	Standard value of rated energy efficiency ratio of cooling *EER*		
				Normal power ranges		
				Piston and scroll compressors 10 kW to 1500 kW	Screw compressors 200 kW to 2000 kW	Turbo-compressors 500 kW to 8000 kW
R134a	27/33	6	0	4,0	4,5	5,2
		14	8	4,6	5,3	5,9
	40/45	6	0	3,1	2,9	4,1
		14	8	3,7	3,7	4,8
R407C	27/33	6	0	3,8	4,2	–
		14	8	4,4	4,9	–
	40/45	6	0	3,0	2,7	–
		14	8	3,6	3,3	–
R410A	27/33	6	0	3,6	–	–
		14	8	4,2	–	–
	40/45	6	0	2,8	–	–
		14	8	3,3	–	–
R717	27/33	6	0	–	4,6	–
		14	8	–	5,4	–
	40/45	6	0	–	3,1	–
		14	8	–	3,7	–
R22	27/33	6	0	4,1	4,6	5,1
		14	8	4,8	5,4	5,7
	40/45	6	0	3,2	3,0	4,1
		14	8	3,8	3,6	4,7

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Table I.9 - Default values for rated energy efficiency ratio of cooling *EER* for air-cooled compressor chillers

Refrigerant	Chilled-water output temperature °C	Mean evaporation temperature °C	Standard value of rated energy efficiency ratio of cooling *EER* Normal power ranges Piston and scroll compressors 10 kW to 1500 kW	Standard value of rated energy efficiency ratio of cooling *EER* Normal power ranges Screw compressors 200 kW to 2000 kW
R134a	6	0	2,8	3,0
	14	8	3,5	3,7
R407C	6	0	2,5	2,7
	14	8	3,2	3,4
R410A	6	0	2,4	–
	14	8	3,1	–
R717	6	0	–	3,2
	14	8	–	3,9
R22	6	0	2,9	3,1
	14	8	3,6	3,8

Table I.10: Default values for energy efficiency ratio of cooling of air-cooled room air-conditioning systems with <12 kW

Air-cooled room air-conditioning systems < 12 kW, in accordance with 2002/31/EC, rated energy efficiency ratio of cooling *EER*		
Type of unit	***EER***	**Type of partial-load control**
Compact air-conditioners, window-mounted or wall-mounted	2,6	pulsed (ON/OFF control) (D)
Split systems	2,7	pulsed (ON/OFF control) (D)
		frequency-controlled
Multi-split systems	2,9	pulsed (ON/OFF control) (E)
		frequency-controlled

Table I.11 - Default values for energy efficiency ratios of cooling of air-cooled room air-conditioning systems with >12 kW

Air-cooled room air-conditioning systems > 12 kW, rated energy efficiency ratio of cooling *EER*		
Type of unit	***EER***	**Type of partial-load control**
VRF systems with variable refrigerant mass flow	33,5	at least one frequency-controlled parallel? compressor (G)

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Table I.12 - Partial-load index values (types of usage: office buildings and comparable)

Compressor type: water-cooled	Operating mode: room conditioning / AHU cooling			Cooling water input to refrigeration unit constant: Cooling tower PLV_{AV} [-]	Cooling water input to refrigeration unit constant: Cooling tower $f_{R,VK}$ [-]	Cooling water input to refrigeration unit constant: Dry cooler PLV_{AV} [-]	Cooling water input to refrigeration unit constant: Dry cooler $f_{R,TK}$ [-]	Cooling water input to refrigeration unit variable: Cooling tower PLV_{AV} [-]	Cooling water input to refrigeration unit variable: Cooling tower $f_{R,VK}$ [-]	Cooling water input to refrigeration unit variable: Dry cooler PLV_{AV} [-]	Cooling water input to refrigeration unit variable: Dry cooler $f_{R,TK}$ [-]
(1)	Room conditioning			0.92	0.12	0.92	0.09				
	AHU cooling	Temperature and limited humidity control	no heat recovery	0.93	0.10	0.93	0.08				
			heat recovery	0.92	0.10	0.92	0.08				
		Full temperature and humidity control	no heat recovery	0.92	0.10	0.92	0.08				
			Sensible + latent HR	0.92	0.11	0.92	0.09				
(2)	Room conditioning			1.31	0.12	1.26	0.08	1.54	0.37	1.74	0.63
	AHU cooling	Temperature and limited humidity control	no heat recovery	1.33	0.09	1.27	0.08	1.57	0.29	1.75	0.70
			heat recovery	1.33	0.10	1.27	0.08	1.56	0.31	1.74	0.71
		Full temperature and humidity control	no heat recovery	1.33	0.10	1.27	0.08	1.56	0.30	1.75	0.65
			Sensible + latent HR	1.32	0.11	1.26	0.08	1.55	0.36	1.73	0.68
(3)	Room conditioning			0.82	0.13	0.79	0.09	0.96	0.40	1.09	0.65
	AHU cooling	Temperature and limited humidity control	no heat recovery	0.73	0.10	0.70	0.08	0.86	0.31	0.95	0.72
			heat recovery	0.75	0.11	0.72	0.08	0.88	0.34	0.99	0.73
		Full temperature and humidity control	no heat recovery	0.75	0.10	0.71	0.08	0.88	0.33	0.98	0.67
			Sensible + latent HR	0.77	0.12	0.74	0.09	0.91	0.38	1.02	0.71
(4)	Room conditioning			0.56	0.13	0.56	0.09				
	AHU cooling	Temperature and limited humidity control	no heat recovery	0.45	0.10	0.45	0.08				
			heat recovery	0.48	0.11	0.48	0.09				
		Full temperature and humidity control	no heat recovery	0.47	0.11	0.47	0.08				
			Sensible + latent HR	0.51	0.12	0.51	0.09				
(5)	Room conditioning			1.01	0.12	0.97	0.09	1.19	0.38	1.79	0.64
	AHU cooling	Temperature and limited humidity control	no heat recovery	0.87	0.09	0.83	0.08	1.02	0.30	1.53	0.72
			heat recovery	0.91	0.10	0.87	0.08	1.06	0.33	1.58	0.73
		Full temperature and humidity control	no heat recovery	0.90	0.10	0.86	0.08	1.05	0.32	1.58	0.67
			Sensible + latent HR	0.95	0.11	0.91	0.08	1.11	0.37	1.66	0.70
(6)	Room conditioning							1.21	0.38	1.37	0.64
	AHU cooling	Temperature and limited humidity control	no heat recovery					1.07	0.30	1.19	0.71
			heat recovery					1.11	0.32	1.23	0.73
		Full temperature and humidity control	no heat recovery					1.10	0.32	1.23	0.66
			Sensible + latent HR					1.15	0.37	1.29	0.70
(7)	Room conditioning			1.07	0.12	1.09	0.08	1.35	0.37	1.46	0.64
	AHU cooling	Temperature and limited humidity control	no heat recovery	0.99	0.09	0.95	0.08	1.28	0.30	1.29	0.73
			heat recovery	1.01	0.10	0.97	0.08	1.30	0.33	1.31	0.74
		Full temperature and humidity control	no heat recovery	1.01	0.10	0.96	0.08	1.29	0.31	1.31	0.67
			Sensible + latent HR	1.04	0.11	1.00	0.08	1.32	0.36	1.34	0.70

Compressor type: air-cooled	Operating mode: room conditioning / AHU cooling			PLV_{AV} [-]
(A)	Room conditioning			1.32
	AHU cooling	Temperature and limited humidity control	no heat recovery	1.37
			heat recovery	1.35
		Full temperature and humidity control	no heat recovery	1.36
			Sensible + latent HR	1.34
(B)	Room conditioning			1.43
	AHU cooling	Temperature and limited humidity control	no heat recovery	1.50
			heat recovery	1.48
		Full temperature and humidity control	no heat recovery	1.49
			Sensible + latent HR	1.46
(C)	Room conditioning			1.14
	AHU cooling	Temperature and limited humidity control	no heat recovery	1.09
			heat recovery	1.10
		Full temperature and humidity control	no heat recovery	1.09
			Sensible + latent HR	1.11
(D)	Room conditioning			1.24
	AHU cooling	Temperature and limited humidity control	no heat recovery	
			heat recovery	
		Full temperature and humidity control	no heat recovery	
			Sensible + latent HR	
(E)	Room conditioning			0.85
	AHU cooling	Temperature and limited humidity control	no heat recovery	
			heat recovery	
		Full temperature and humidity control	no heat recovery	
			Sensible + latent HR	
(F)	Room conditioning			1.37
	AHU cooling	Temperature and limited humidity control	no heat recovery	
			heat recovery	
		Full temperature and humidity control	no heat recovery	
			Sensible + latent HR	
(G)	Room conditioning			1.33
	AHU cooling	Temperature and limited humidity control	no heat recovery	1.21
			heat recovery	1.23
		Full temperature and humidity control	no heat recovery	1.23
			Sensible + latent HR	1.28

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Annex J
(informative)

Auxiliary energy for cooling-water and cold-water distribution

Auxiliary energy for cooling-water and cold-water distribution (Danish/German proposal)

J.1 Electrical energy demand

J.1.1 General

Table J.1 shows the essential parameters for electrical energy demands of cooling and cold-water networks.

Table J.1 - Essential parameters for electrical energy demands of cooling and cold-water networks

Parameter	Cold-water network	Cooling-water network
Cooling refrigeration	Rated power ($\dot{Q}_{C,outg}$) of refrigeration unit	Rated power ($\dot{Q}_{C,outg}$) of refrigeration unit
		Energy efficiency ratio *EER* or heat ratio of refrigeration unit
Cooling control	Distribution of parts of $\dot{Q}_{C,outg}$ among the refrigeration units and partial shut-down of individual evaporators	Distribution of parts of $\dot{Q}_{C,outg}$ among the refrigeration units and partial shut-down of individual condensers
	Control of refrigeration units in part-load operation	Control of refrigeration units in part-load operation
	Use of cooling storage systems	Control strategy for refrigeration unit, cooling tower fan and pump
	Minimum water quantity, evaporator	Minimum water quantities, condenser
Fluid drive	Temperature difference input/output	Temperature difference input/output
	Pressure drops – longest water paths – tubing cross-sections – fittings and valves, heat exchangers – hydraulic adjustment	Pressure drops – longest water paths – tubing cross-sections – fittings and valves, heat exchangers – design type (open or closed)
	Cooling media (e. g. water, glycol etc.)	Heat media (e. g. water, glycol etc.)
Pump selection	Design quality [a]	Design quality
	Pump type and efficiency	Pump type and efficiency
	Use of controlled or uncontrolled pumps	Use of controlled or uncontrolled pumps
	Type of pump control	Type of pump control

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Table J.1 - Essential parameters for electrical energy demands of cooling and cold-water networks *(continued)*

Distribution circuit design[b]	Assignment of refrigeration unit to distribution networks	Assignment of refrigeration unit to cooling tower
	Single-circuit or dual-circuit system (with primary and secondary circuits)	
	Distribution of user circuits corresponding to the cooling demand (branch or zone control)	
	Creating a hydraulic balance	
	Mixing and overflows in the distribution circuit	
Operating time / load profiles	Cooling load hours of the building	Refrigeration unit operating hours
	Pump shut-down when no cooling is required: – by spaces: HVAC and building zones – by periods: seasonal as well as night-time and weekend shut-down or reduction[c] of power	Pump shut-down when no cooling is required: – via the refrigeration unit Using the cooling tower for free convection cooling
	Integration into building automation	Integration into building automation
Power control in distribution circuit	Mass flow control with – two-way choke valves or – three-way diverter valves or – controlled pump	Mass flow control with – two-way choke valves or – controlled pump
	Temperature control by three-way additive mixing	Temperature control by three-way additive mixing
	Interaction between controls of sinks and pump	Interaction of controls of cooling tower and pump
Other auxiliary energy	Pumps for – air humidification – indirect evaporation cooling	Circulation spray pumps
	Pumps in KVS heat recovery systems	Frost-protection heating
	Pressure maintenance and degassing systems	Water preparation, dosing pumps
	Condensate pumps	
	Electromotoric valves, electrothermal valves	

a Deviation of the operating point from the design operating point.

b The system design affects the required pump power, the operating time and the pump control options.

c Intermittent operation

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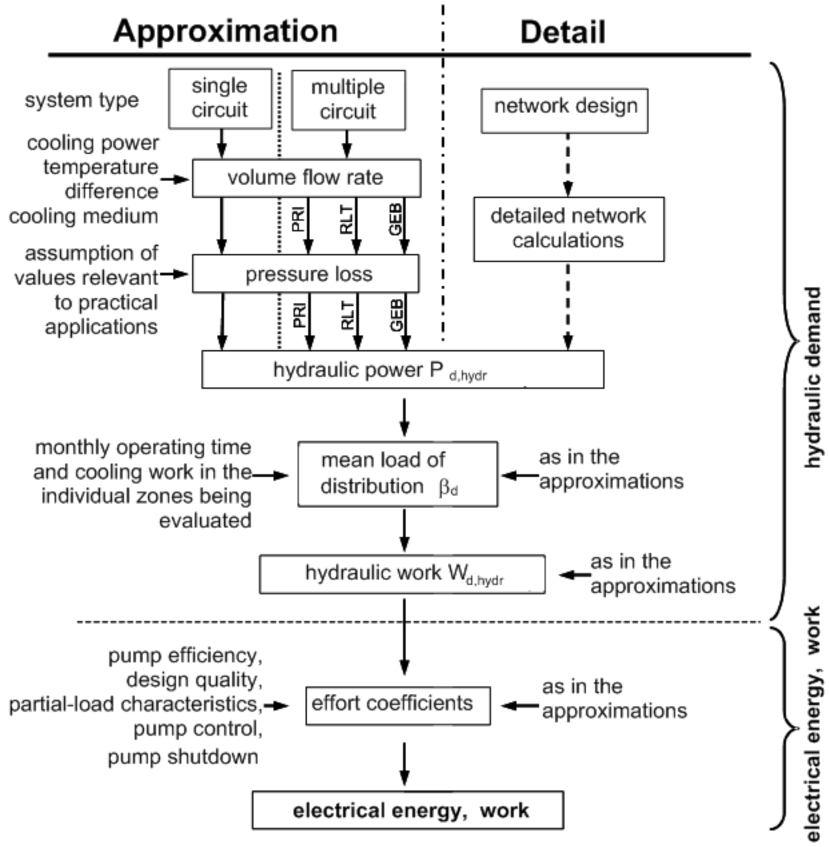

Key

PRI = primary circuit, RLT = heating, ventilation and air-conditioning system,
GEB = building cooling, index d = distribution

Figure J.1 - Procedure for calculating the pump energy demands for cooling-water and cold water

The calculation methods for determining the electrical energy demands must be taken into account for every pump distribution circuit in the system on the basis of the nominal cooling load of the calculation zones of the building or the HVAC system. In these calculations, the respective refrigeration or cooling power of the distribution circuit being evaluated as well as the operating time and the associated cooling work per time period are used.

The pump distribution circuits are designated “cooling supply units” in the following and are identified by an index “Z”. In many cases the cold-water distribution in the building zones is achieved by one cooling supply unit (Z) serving several zones (e. g. in primary and secondary circuits). Where this is the case, the cooling power values (e. g. $\dot{Q}_{\mathrm{C}}$ or $\dot{Q}_{\mathrm{C*}}$) and cooling work (e. g. Q_{C} or $Q_{\mathrm{C*}}$) of the individual zones served by the cooling supply unit (Z) must be added. The required pump operating time within the cooling supply unit (Z) is assumed to be equal to the maximum time during which cooling is required.

J.1.2 Electrical energy demand of distribution

The annual energy demand is determined as the sum of the energy demands over the periods under consideration. The time periods (e. g. annual balance method = 1 year, monthly balance method = 12 months) are indicated by the indexes “l”:

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$$Q_{\mathrm{Z,aux,d,a}} = \sum_{l=1}^{n} Q_{\mathrm{Z,aux,d,l}} \qquad \text{(J.1)}$$

where:

$Q_{\mathrm{Z,aux,d,a}}$ is the annual electrical energy demand of the cooling refrigeration unit in kWh;

$Q_{\mathrm{Z,aux,d,l}}$ is the annual electrical energy demand of the cooling refrigeration unit per time period, in kWh;

n is the number of corresponding time periods in one year.

The electrical energy demand for operating the cold-water and cooling-water distribution systems can be determined, in a simplified form, on the basis of the hydraulic requirements and a demand coefficient for evaluating the pump operation:

$$Q_{\mathrm{Z,aux,d,l}} = W_{\mathrm{d,hydr,l}} \cdot e_{\mathrm{d,l}} \qquad \text{(J.2)}$$

where:

$W_{\mathrm{d,hydr,l}}$ is the hydraulic energy demand per time period, in kWh;

$e_{\mathrm{d,l}}$ is the demand coefficient for distribution within the time period.

J.2 Hydraulic energy demand for distribution

J.2.1 General

The hydraulic energy demand of cold-water and cooling-water distribution systems within a specific time period is determined separately for every pump distribution circuit in the system on the basis of the hydraulic power at the design operating point, the pump operating time and the mean load of distribution within the time period.

$$W_{\mathrm{d,hydr,l}} = \frac{P_{\mathrm{d,hydr}}}{1\,000} \cdot t_{\mathrm{d,l}} \cdot \beta_{\mathrm{d,l}} \cdot f_{\mathrm{Abgl}} \qquad \text{(J.3)}$$

where:

$P_{\mathrm{d,hydr}}$ is the hydraulic power of the cooling refrigeration unit at the design operating point, in W;

$t_{\mathrm{d,l}}$ is the pump operating time within the time period, in h;

$\beta_{\mathrm{d,l}}$ is the mean load of distribution within the time period;

f_{Abgl} is the correction factor for hydraulic adjustment.

The hydraulic power at the design operating point is calculated by the equation:

$$P_{\mathrm{d,hydr}} = 1\,000 \cdot \Delta p_{\mathrm{Z}} \cdot \frac{\dot{V}_{\mathrm{Z}}}{3\,600\ \mathrm{s/h}} \qquad \text{(J.4)}$$

where:

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$\dot{V}_Z$ is the volume flow of the cooling refrigeration unit at the design operating point, in m³/h;

ΔP_Z is the pressure difference across the cooling refrigeration unit at the design operating point, in kPa.

The volume flow at the design operating point is calculated from the cooling power, from the temperature difference between input and output and from the material properties of the cooling medium used in the distribution circuit under consideration.

$$\dot{V}_Z = \frac{3\,600\,{}^{s}\!/_{h} \cdot \dot{Q}_Z}{\Delta\vartheta_{Z,cl} \cdot c_{cl} \cdot \rho_{cl}} \tag{J.5}$$

where:

$\dot{Q}_Z$ is the rated cooling or refrigerating power of the cooling refrigeration unit at the design operating point, in kW;

$\Delta\vartheta_{Z,cl}$ is the temperature difference (input / output) across the cooling refrigeration unit at the design operating point, in K;

c_{cl} is the specific heat of the cooling medium (coolant) in kJ/kg · K;

ρ_{cl} is the density of the cooling medium air, in kg/m^3.

In the case of water-cooled refrigeration units, the cooling power is calculated as follows:

$$\dot{Q}_Z = \dot{Q}_{C,outg} \cdot \left(1 + \frac{1}{EER}\right) \tag{J.6}$$

where:

$\dot{Q}_{C,outg}$ is the cooling power of the refrigeration unit, in kW;

EER is the cooling energy efficiency ratio of compression refrigerators, in the case of absorption refrigeration units, ξ is used.

The cooling work in the cooling circuit is calculated using the equation

$$Q_Z = Q_{C,outg} \cdot \left(1 + \frac{1}{SEER}\right)$$

where

$Q_{C,outg}$ is the cooling refrigeration output of the refrigeration unit (in the resp. month), in kWh;

SEER is the annual cooling energy efficiency ratio of compression refrigerators; in the case of absorption refrigeration units, ξ is used .

J.2.2 Pressure head at the design-rating operating point

The pressure difference (Δp) at the design operating point is a function of the resistance of the pipe network (with various individual resistances) and the supplementary component resistances. The pressure difference the pump is required to achieve is determined for the most unfavourable branch

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or the most unfavourable sink connected to the supply unit. In the case of known system parameters, the calculations are carried out in detail for the pipe network. As an alternative, realistic estimated values of the individual resistances are used for the approximation method.

$$\Delta p_{\mathrm{Z}} = R \cdot L_{\mathrm{max}} \cdot (1 + z) + \Delta p_{\mathrm{WÜE}} + \Delta p_{\mathrm{RV}} + \Delta p_{\mathrm{WÜV}} \tag{J.7}$$

where:

R is the pressure drop along the piping, in kPa/m;

L_{max} is the maximum pipe length in the distribution circuit, in m;

z is the share of the individual resistances in the pipe friction loss;

Δp_{Z} is the pressure difference across the cooling refrigeration unit at the design operating point, in kPa;

$\Delta p_{\mathrm{WÜE}}$ is the pressure difference across the heat exchangers in the generator, in kPa;

Δp_{RV} is the pressure difference across the control valves, in kPa;

$\Delta p_{\mathrm{WÜV}}$ is the pressure difference across the heat exchangers in the sink, in kPa.

J.2.3 Δp approximation values

The approximation values given below for pressure drops across individual components must be used in Equation (J.7) according to the design of the respective distribution circuit. The values given are guide values for use in planning. In all cases, a minimum and a maximum dimension value will be defined for design specifications, these being determined by the targeted investment and operating costs.

Friction losses in pipes depend on the length and type of distribution networks (network topology) as well as on the mean flow rates and the smoothness of the internal surfaces of the pipes. Approximation values which can be assumed for the specific pressure gradient R in kPa/m and the additional share of individual flow resistances z are shown in Table J.2:

Table J.2 - Pressure gradient R in kPa/m and additional share of individual flow resistances z in pipe networks.

Pressure gradient R, in kPa/m	0,25
Share of individual resistances z	0,3

The maximum length of piping in cooling circuits, primary circuits and distribution circuits of HVAC systems is a function of twice the distance between the cooling refrigeration unit and the respective heat transfer components.

For a rectangular building, the maximum pipe length of cold-water distribution circuits can be approximated using the external dimensions:

$$L_{\mathrm{max}} = 2 \cdot \left(L + \frac{B}{2} + h_{\mathrm{G}} \cdot n_{\mathrm{G}} + 10 \right) \tag{J.8}$$

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where:

- L is the length of the building, in m;
- B is the width of the building, in m;
- h_G is the average storey height, in m;
- n_G is the number of storeys.

Approximation values which can be assumed for pressure drops across components in distribution circuits are shown in Table J.3.

Table J.3 - Assumed values for pressure drops across components in distribution circuits

Components	Pressure drop in kPa	
Plate evaporator	$\Delta p_{WÜE}$	40
Tube evaporator		30
Condenser		45
Cooling tower, closed	Δp_{KT}	35
open		35
Water/water	$\Delta p_{WÜ}$	50
Hydraulic transfer	$\Delta p_{ÜG}$	5
No-return valve	Δp_{RSV}	5
Central air cooler	$\Delta p_{WÜV}$	35
Central air heater		20
Induction equipment		35
Chilled ceiling, cooling convector		35
Shut-off valve, OPEN/SHUT	Δp_{RV}	10
Choke valve, continuous control		see Equation (J.9)
Valve authority, a =		0,4
Three-way valve, deflection		10

The equation for determining the pressure drop across a continuous-control choke valve is:

$$\Delta p_{RV,stetig} = \frac{a}{(1-a)} \cdot \Delta p_{WÜV} \qquad (J.9)$$

These assumed values apply to water at 10 °C with $\nu \approx 1{,}5$ mm²/s. If cooling media of viscosity $\nu > 4$ mm²/s or Flo-Ice are used, then the pressure drops in the distribution networks and heat exchangers must be determined separately. Calculation of the pressure drops of branch control valves is only really possible in the course of detail planning.

J.2.4 Pump operating times

The demand times for air cooling in the HVAC system and the demand times for pumps which are operated at the same times as the refrigeration unit (recooling the condenser and the primary circuit

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pump in the evaporator circuit) are determined as monthly values for each utilization unit / zone. The building cooling demand times are determined using the balance procedure in. The operating times of the pumps in the cooling refrigeration units depend on the system design concept and may exceed the determined demand times for cooling in the refrigeration unit. This may be the case, for instance, if there are no technical provisions for demand-controlled switching of the pumps, or if an uninterrupted supply of cooling energy is required.

If several different utilization units/zones are served by one and the same distribution circuit, then the operating times of the pumps in this distribution circuit are based on the unit or zone with the maximum demand time. The respective cooling or refrigeration energy demand is the sum of the individual energy demands of the utilization units/zones.

Four operating modes can be distinguished. Operating mode 1 applies to fully demand-controlled operation, operating modes 2 to 4 apply to supply-controlled operation (see Table J.4 below).

Table J.4 - Operating modes

Operating modes		**Operating time**	**Minimum requirements for implementing the operating mode**
1	**Fully-automated, demand-controlled operation**	**td,l = tc*,op,l or tc,op,l**	**Pumps switched on and off in relation to the current cooling demand (e. g. via control processes or building automation system)**
2	**Seasonal shut-down, night-time and weekend shut-down**	$t_{\mathrm{d},l} = d_{\mathrm{m}} \cdot (24 - t_{\mathrm{Na}} - 0{,}15 \cdot t_{\mathrm{W}}$ $t_{\mathrm{d,l}} = t_{\mathrm{RLT\text{-}Betrieb}}$ **(for time period l = 1 month)**	**External switching on and off of the pumps by superimposed services (e. g. refrigeration units, time switches or building automation systems)**
3	**Seasonal shut-down in the months without cooling demand**	$t_{\mathrm{d},l} = d_{\mathrm{m}} \cdot 24$ **(for time period l = 1 month)**	**External switching on and off of the pumps (e. g. manually or automatically)**
4	**Year-round pump operation (even when no demand exists)**	**td,l = 8 760 h** **(for time period l = 1 year)**	—
		$t_{\mathrm{d},l} = d_{\mathrm{m}} \cdot 24$ **(for time period l = 1 month[a])**	

where:

tc*,l or tc,l is the time during which there is a cooling demand in the refrigeration unit, in h;

is the air cooling for HVAC system: the demand times tc*,op,l (see 5.4.3);

is the space cooling: the demand times tc,op,l.

tNa is the number of night-time shut-down hours per day, in h/d;

tWe is the number of shut-down hours per weekend, without night-time shut-down, in h/w;

dm is the number of days per month, in d/m.

a The resulting energy demand in the months without cooling demand must be determined for the minimum volume flow which has to be maintained in the distribution circuit (see Equation (J.11) with $\beta'_{\mathrm{d,l}} = 0$**).**

J.2.5 Mean distribution load

The mean load of distribution b_{D} is calculated for every pump distribution circuit in the system on the basis of the design-rating cooling power, the cooling work of the refrigeration unit and the operating time within the respective time period:

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$$\beta_{\text{d,l}} = \frac{Q_{\text{Z,outg,l}}}{\dot{Q}_{\text{Z}} \cdot t_{\text{d,l}}} \tag{J.10}$$

where

$\dot{Q}_{\text{Z}}$ is the rated cooling or refrigerating power of the cooling refrigeration unit at the design operating point, in kW;

$Q_{\text{Z,outg,l}}$ is the energy demand for cooling the refrigeration unit.

- The annual energy demand ($Q_{\text{c*,outg}}$) for HVAC cooling, monthly values are calculated as described in Annex K.
- The monthly energy demand ($Q_{\text{c,outg}}$) for space cooling is calculated as described in **K.2**.

If overflow devices (e. g. valves, pipes etc.) are installed in the distribution circuit under consideration[5)], then their effect on the mean hydraulic demand must be assessed using the following equation:

$\beta'_{\text{d,l}}$ in relation to the determined mean load

$$\beta_{\text{d,l}} = \beta'_{\text{d,l}} + \left(1 - \beta'_{\text{d,l}}\right) \cdot \frac{\dot{V}_{\text{Z,min}}}{\dot{V}_{\text{Z}}} \tag{J.11}$$

where:

$\dot{V}_{\text{Z,min}}$ is the minimum volume flow in the distribution circuit, in m³/h;

$\dot{V}_{\text{Z}}$ is the volume flow of the cooling refrigeration unit at the design operating point, in m³/h, as calculated by Equation (J.5).

The minimum volume flow must be calculated on the basis of the requirements of the cooling refrigeration unit or, where applicable, of the overpressure relief valve in the sink circuit [6)].
In the case of hydraulic separation of the primary circuit [7)] or the use of deflection valves in the sink circuit, a value of b_{D} = 1 can be assumed to apply to the associated supply pumps.

J.2.6 Correction factor f_{Abgl} for hydraulic adjustment

- For networks with hydraulic adjustment, factor f_{Abgl} = 1.
- For networks without hydraulic adjustment, factor f_{Abgl} = 1,25.

[5)] E. g. for ensuring a minimum water flow to the evaporator or for limiting the pressure drop across the sink or for ensuring an unbroken cooling supply to the network.

[6)] The respective function of the overflow valve is a function of the interaction of the system pressure drop, pump characteristics and the valve activation pressure.

[7)] E. g. via a hydraulic two-way valve or a storage device connected in parallel.

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J.3 Demand coefficients

J.3.1 General

The operating behaviour of the pumps installed in the system is evaluated using the demand coefficient for cold-water distribution within the time period:

$$e_{\mathrm{d,l}} = f_{\mathrm{e}} \cdot \left(C_{\mathrm{P1}} + C_{\mathrm{P2}} \cdot \beta_{\mathrm{d,l}}{}^{-1}\right) \tag{J.12}$$

where:

f_{e} is the efficiency factor of the pump;

C_{P1}, C_{P2} are the constants which take pump power control methods into consideration, as given in Table J.5.

By integrating these factors, the demand coefficient accounts for the essential factors affecting the annual electrical power consumption of a pump, as derived from the pump size, its efficiency and its partial-load and control characteristics.

J.3.2 Efficiency factor f_{e} of the pump

The efficiency factor takes into account the overall efficiency of the pump and is the product of the pump efficiency (coupling efficiency) and the efficiency of the driving motor. If the electrical power consumption of the pump at the design operating point is known, the following applies:

$$f_{\mathrm{e}} = \frac{P_{\mathrm{Pumpe}}}{P_{\mathrm{hydr}}} \tag{J.13}$$

where:

P_{Pumpe} is the electrical power consumption of the pump at the design-rating operating point.

If the pump data are not known, then the following assumption can be made:

$$f_{\mathrm{e}} = \left(1{,}25 + \left(\frac{200}{P_{\mathrm{d,hydr}}}\right)^{0{,}5}\right) \cdot f_{\mathrm{Adap}} \cdot b \tag{J.14}$$

where:

f_{Adap} is the correction factor for adaptation to the operating point;

b is the evaluation factor for the building category, existing buildings: b = 1,2, new construction: b = 1,0.

The assumption values for f_{e} from Equation (J.14) apply to:

— radial centrifugal pumps (with motors of efficiency class 1 at the pump's rated design operating point. If the pump is not operated at the design point or if the hydraulic part of the pump and the motor are not matched correctly, the values of f_{e} are worse);

— maximum pressure differences $\Delta p \leq 0{,}6$ bar for $P_{\mathrm{hydr}} < 0{,}2$ kW; $\Delta p \leq 1{,}5$ bar for $0{,}2\ \mathrm{kW} \leq P_{\mathrm{hydr}} \leq 0{,}5$ kW; or $\Delta p \leq 4{,}0$ bar for $P_{\mathrm{hydr}} > 0{,}5$ kW;

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— water at 20 °C with ν ≈ 1,0 mm²/s.

If cooling media of viscosity ν > 4 mm²/s or Flo-Ice are used, then the assumed values for f_e must be determined separately. For cooling media with 4 mm²/s ≤ ν ≤ 40 mm²/s, an adapted value of f_e can be calculated for the pump types and delivery heights mentioned above:

with f'_e as obtained by Equation (J.14)

$$f_e = f'_e \cdot \left(1 + \left(\frac{\nu^2_{cl}}{16 \cdot P_{d,hydr}}\right)^{0,4}\right) \qquad (J.15)$$

where:

ν_{cl} is the kinematic viscosity of the cooling medium, in mm²/s.

J.3.3 Correction factor f_{Adap} for adaptation

The adaptation correction factor accounts for the power consumption of the pump at the actual operating point. This takes into consideration the differences between the power consumption of the pump as installed and an optimally adapted pump at design operating point conditions. The actual operating point of the pump is always related to the design-rating volume flow. For pumps which cannot be adapted, there will be more or less large differences between the actual operating point and the design-rating operating point. An adaptable pump can normally be adjusted to achieve design-rating operating point values more closely.

— known / optimally adapted pump: f_{Adap} = 1,0

If the pump data are not known, then the following assumption can be made:

— for pumps not adapted at all: f_{Adap} = 1,2;

— for electronically adapted pumps: f_{Adap} = 1,05.

J.3.4 Pump power adaptation during operation

The energy demand can be reduced considerably by adjusting the pump power to match the hydraulic demand of distribution circuits with variable volume flow demands. The pump power consumption can be varied by:

— internal speed controls (e. g. Δp = constant or Δp = variable);

— external speed controls;

— partial switching on and off of pumps operated in parallel (e. g. dual pumps).

Table J.5 shows values for C_{P1} and C_{P2} as given in Equation (J.11). These coefficients account for pump power reduction in uncontrolled and controlled pump operation.

Table J.5 - Values of C_{P1} and C_{P2} in relation to the pump operating mode

Pump operating mode	Controlled	Uncontrolled
C_{P1}	0,25	0,85
C_{P2}	0,75	0,15

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J.3.5 Switching of individual pumps in parallel-pump installations

In distribution circuits with variable volume flow demands and flat system characteristic curves, the use of pumps connected in parallel (e. g. dual pumps) can achieve quite good partial-load adaptation. In such distribution circuits, switching individual pumps on and off in relation to the demand can reduce the energy demand considerably. In distribution circuits in which the values of β_d are lower than 0,7, the coefficients given in Table J.5 for controlled pump operation can be used to determine the demand coefficient for volume flow adaptation by switching parallel pumps on and off.

J.4 Other auxiliary energy demands (auxiliary drives)

J.4.1 Pump heating registers

Additional pumps installed in preheating and reheating registers are not dealt with in the present document.

J.4.2 Pumps and drives for heat recovery

J.4.2.1 Pumps in integrated fluid circulation systems

The annual electrical energy demand $Q_{hr,f,aux}$ of the pump in an integrated fluid circulation system can be calculated using a procedure analogous to that described in J.1.

As an alternative, the following simplified method can be applied for each utilization unit/zone of the respective HVAC system.

$$Q_{hr,f,aux,a} = P_{el,av,KVS} \cdot t_{WRG} \tag{J.16}$$

with

$P_{el,av,KVS} = V_{Außenluft} \cdot 0{,}03\ W/m^3/h$, for uncontrolled pumps

$P_{el,av,KVS} = V_{Außenluft} \cdot 0{,}015\ W/m^3/h$, for pumps with speed control

where:

$Q_{hr,f,aux,a}$ is the annual auxiliary energy demand for heat recovery;

t_{WRG} is the operating time of the heat recovery pump;

$V_{Außenluft}$ is the design-rating outdoor air volume flow of the heat recovery system.

J.4.2.2 Rotor drive

The annual auxiliary energy demand for driving the rotor of a rotary heat exchanger is calculated as follows:

$$Q_{hr,f,aux,a} = P_{el,av,rot} \cdot t_{WRG} \tag{J.17}$$

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Table J.6 - Electrical power for rotor drives

Design-rating outdoor-air volume flow of heat recovery system $V_{\text{Außenluft}}$ m³/h	**Effective electrical power, rotor drive** $P_{\text{el,av,rot}}$ W
up to 7 500	90
> 7 500 up to 25 000	180
> 25 000 up to 65 000	370
> 65 000	750

Rotor operating time for heat recovery as deduced in J.4.2.1.

J.4.2.3 Heat pumps

It is not possible to calculate heat pump balances by the simplified method. For systems with heat pumps, the simulation values must be determined individually as described in DIN V 18599-3, and the corresponding electrical energy demand must be calculated correspondingly. These methods are then applied to calculate $Q_{\text{hr,f,aux}}$.

J.4.3 Water humidifier pumps

The annual auxiliary energy demand $Q_{\text{mh,f,aux}}$ for water humidifier pumps is calculated on the basis of the following standard values for HVAC system operating times and the equation:

$$Q_{\text{mh,f,aux,a}} = V_{\text{Außenluft}} \cdot P_{\text{el,mh}} \cdot t_{\text{VB}} \cdot f_{\text{mh}} \quad \text{(J.18)}$$

where:

$P_{\text{el,mh}}$ is the specific power consumption of the pumps of the water humidifiers per m^3/h air volume flow (see Table J.6);

t_{VB} is the air humidification operating time calculated;

f_{mh} is the partial-load factor for humidification control;

$V_{\text{Außenluft}}$ is the design-rating outdoor-air volume flow of the humidification system.

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Table J.7 - Default values for water humidifiers (mean annual values)

	Control	Power $P_{el,mh}$ W/m³/h	Control factor f_{mh} 6 g/kg	Control factor f_{mh} 8 g/kg
Contact and drip humidifiers	uncontrolled and valve-controlled	0,01	1	1
Rotary spray humidifiers	uncontrolled	0,20	1	1
	Valve control	0,20	1	1
	on/off (proportionally)	0,20	0,35	0,50
	speed control	0,20	0,20	0,30
High-pressure humidifiers	speed control	0,04	0,35	0,50
Hybrid humidifiers	on/off	0,02	0,35	0,50

J.4.4 Electrical energy demand for central HVAC unit controls

Due to the wide variety of different systems and the strong interaction with the building automation and control systems, the electrical demand for central HVAC unit controls is not dealt with in this document.

J.5 Guideline for calculating the electrical energy demand of cooling-water and cold-water distribution systems

J.5.1 General

The effect of different system concepts on the energy demand of pumps for cold-water distribution is described below using a multiple-circuit system within an office building (see Figure J.2) as an example.

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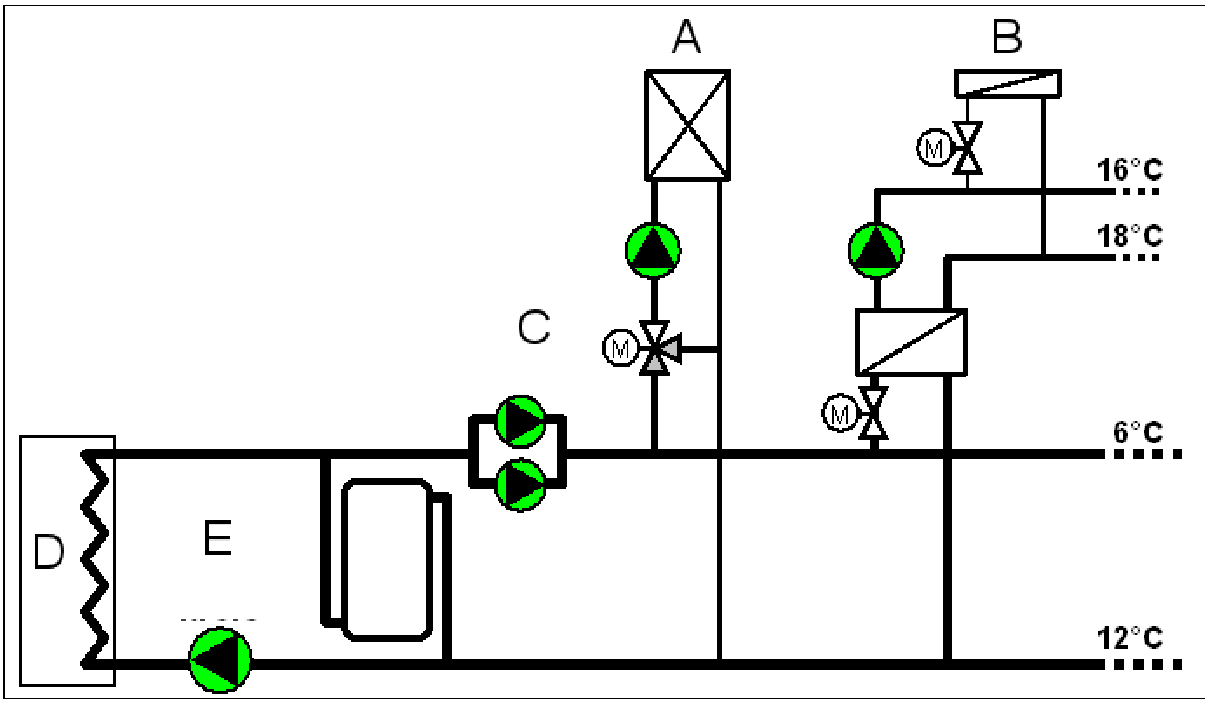

Key:

A Central HVAC System
B Space/building cooling
C Main distribution
D Evaporator
E Primary circuit

Figure J.2 - Example of a multi-circuit cold-water comfort air-conditioning system

J.5.2 Specific volume flow in the distribution circuit

The volume flow in the distribution circuits at the design-rating operating point is calculated using Equation (J.5). If water is used as a cooling medium, the following specific volume flows $V_{d,spez}$ per kW cooling power are obtained (see Table J.8).

Table J.8 - Specific volume flows

Distribution circuit	Temperature difference	$V_{d,spez}$ (m³/h)/kW
Primary circuit, main distribution, central HVAC system	$\Delta\vartheta_{Z,cl}$ = 6K	$4{,}0 \cdot 10^{-5}$
Building cooling	$\Delta\vartheta_{Z,cl}$ = 2K	$1{,}2 \cdot 10^{-4}$

J.5.3 Pressure head Δp_Z at the design-rating operating point

The pressure head at the design-rating operating point is a function of the resistance and length of the pipe network and the additional resistances of the individual components (valves, heat exchangers etc.). Depending on the size of the building and the design of the hydraulic networks, the pressure drops resulting for the same type of space environment control tasks can differ very strongly. The

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specification of undersized pipe networks when designing systems leads to lower investment costs but higher energy demands and vice-versa. Table J.9 shows typical values of pressure drops in various types of system.

Table J.9 - Pressure drops at design-rating operation point

Distribution circuit	**Pressure drops Δp_Z in the distribution circuit, kPa**		
	High resistance	**Medium resistance**	**Energy efficiency optimized**
Primary circuit	150	100	50
Main distribution	400	250	150
Central HVAC	250	150	100
Building cooling	400	300	200

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J.5.4 Annual pump operating times $\sum t_{d,l}$

The annual operating times of the distribution circuit pumps, which depend on the system concept, are shown in Table J.10, using an office building as an example.

Table J.10 - Annual pump operating times

Distribution circuit	Operating time[a] h per annum		
	Demand-controlled	Intermittent	Seasonal
Primary circuit	1250	–	–
Main distribution, central HVAC building cooling	1300	2200	5100

[a] All values are understood to be examples for an office building with corresponding system hydraulics (see Figure J.2) and are not to be considered as substitutes for normative calculations of the electrical energy demand of the pumps.

J.5.5 Specific electrical power of distribution

The specific electrical power of distribution can be used as an index value for the quality of the design of the cooling-water and cold-water distribution system. It is calculated as follows:

$$P_{\text{d,spez.,l}} = \frac{P_{\text{d,hydr}} \cdot f_{\text{e}}}{\dot{Q}_{\text{z}}} \qquad \text{(J.19)}$$

where:

$P_{\text{d,spez.,l}}$ is the specific electrical power of the distribution system, in W per kW cooling power;

$P_{\text{d,hydr}}$ is the hydraulic power of the system zone being evaluated, at the design-rating operating point, in W (see Equation (J4));

f_{e} is the pump efficiency factor (see Equations (J13) and (J14) or (J15));

$\dot{Q}_{\text{Z}}$ is the rated cooling or refrigerating power of the cooling refrigeration unit at the design-rating operating point, in kW (see Equation (J5) or (J6)).

J.5.6 Electrical energy demand of distribution

Using an office building with a cold-water distribution circuit as an example (see Figure J.2), the specific electrical energy demand of distribution can be estimated using the values given in Table J.9. The specific electrical energy demand is the annual energy consumption of the distribution circuit pumps in relation to the respective cooling power, it is expressed in (kWh/a)/kW. Taking various design and operating concepts (cases 1, 2 and 3) into consideration, the following values are obtained (see Table J.11).

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Table J.11 - Electrical energy demand of distribution

Characteristic	Case 1	Case 2[a]	Case 3[a]
Δp of pipe network	high resistance	medium resistance	energy efficiency optimized
Operating time	seasonal / intermittent	intermittent	demand-controlled
Design	no pump adaptation, no hydraulic adjustment overflow in the network	as frequently seen in practical implementations	optimum
Pump operating mode	uncontrolled	uncontrolled/controlled	fully controlled
	$\frac{Q_{\mathrm{Z,aux,d}}}{\dot{Q}_{\mathrm{Z}}},\ in\ \frac{(kWh\ per\ annum)}{\mathrm{kW}}$		
Primary circuit	16	5	2
Main distribution	59	12	5
Central HVAC	48	20	7
Building cooling	170	45	25

[a] All values are understood to be examples for an office building with corresponding system hydraulics (see Figure J.2) and should not be considered as substitutes for normative calculations of the electrical energy demand of the pumps.

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Annex K
(informative)

Thermal and dehumidification distribution losses in cooling systems

K.1 Cooling for the HVAC system

The general equation for calculating the usable energy demand for cooling of the HVAC system is:

$$Q_{c^*,outg} = Q_{c^*,b} + Q_{c^*,ce} + Q_{c^*,d} + Q_{c^*,s} \quad (K.1)$$

where:

$Q_{c^*,outg}$ is the generator cooling output for air-conditioning to the HVAC system;

$Q_{c^*,b}$ is the usable energy demand for the cooling coil;

$Q_{c^*,ce}$ is the cooling energy transfer, air-conditioning cooling for HVAC system;

$Q_{c^*,d}$ is the cooling energy of distribution, air-conditioning cooling for HVAC system;

$Q_{c^*,s}$ is the cooling energy storage, air-conditioning cooling for HVAC system.

If the simplified cooling calculation procedure is used, the individual components are evaluated as shown in Table K.1.

Cooling energy of transfer, AC cooling for HVAC system:

$$Q_{c^*,ce} = ((1 - \eta_{c^*,ce}) + (1 - \eta_{c^*,ce,sens}))\cdot Q_{c^*,b} \quad (K.2)$$

where:

$\eta_{c^*,ce}$ is the degree of utilization of cooling transfer for HVAC (constant);

$\eta_{c^*,ce,sens}$ is the sensible utilization factor of cooling transfer (HVAC). This takes into account undesirable dehumidification in existent air-cooling equipment.

Cooling energy of distribution, air-conditioning cooling for HVAC system:

$$Q_{c^*,d} = (1 - \eta_{c^*,d}) \cdot Q_{c^*,b} \quad (K.3)$$

where:

$\eta_{c^*,d}$ is the degree of utilization of the distribution;

$Q_{c^*,s} = 0$ cooling energy of storage, air-conditioning cooling for HVAC system (K.4)

In cases where combinations of air-cooling and dehumidification are integrated in the HVAC unit, or space cooling with cold-water systems (with a dehumidification option) or individual room air

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conditioners are used, the respective more unfavourable (lower) factor $\eta_{c*,ce,sens}$ of the space cooling and HVAC cooling must be applied once only for HVAC cooling (outdoor air). For space cooling, the factor $\eta_{c,ce,sens}$ = 1 must be applied in such cases.

Table K.1 - Factors for (mean annual value) cooling, HVAC system

Cooling system	$\eta_{c*,ce,sens}$			$\eta_{c*,ce}$	$\eta_{c*,d}$
Humidity demand	none	with tolerance	without tolerance		
Cold water 6/12	0,87	0,94	1	0,90	0,95 (pipes inside building) 0,90 (pipes outside building)
Cold water 14/18	1	1	1	0,90	0,95 (pipes inside building) 0,90 (pipes outside building)
Cold water 18/20	1	1	1	1,0	1,0
Direct evaporation	0,87	0,94	1	0,90	0,95 (pipes inside building) 0,90 (pipes outside building)
Free cooling via cooling tower	1	1	1	0,90 … 1	0,95 (pipes inside building) 0,90 (pipes outside building)

Other intermediate values for the cold-water input temperature must be interpolated.

If multiple generator circuits are involved, the refrigeration energy output for the HVAC cooling function must be calculated separately for each cooling generator system.

K.2 Cooling energy supply for space cooling

The general equation for calculating the usable energy demand for space cooling is:

$$Q_{c,outg} = Q_{c,b} + Q_{c,ce} + Q_{c,d} + Q_{c,s} \qquad (K.5)$$

where:

$Q_{c,outg}$ is the usable generator cooling output to the (room) space-cooling system;

$Q_{c,b}$ is the usable energy for space cooling;

$Q_{c,ce}$ is the cooling output of transfer for the space-cooling system;

$Q_{c,d}$ is the cooling output of distribution for the space-cooling system;

$Q_{c,s}$ is the cooling output of storage for the space-cooling system.

Within the framework of the simplified cooling calculation procedure, the individual components are evaluated as shown in Table K.2.

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Cooling loss during transfer for the cooling system:

$$Q_{c,ce} = ((1 - \eta_{c,ce}) + (1 - \eta_{c,ce,sens}))\cdot Q_{c,b} \qquad (K.6)$$

where:

$\eta_{c,ce}$ is the degree of utilization of transfer of cooling to the space-cooling system (see Table K.2);

$\eta_{c,ce,sens}$ is the sensible degree of utilization of transfer to the space-cooling system (see Table K.2). This takes into account the undesirable dehumidification in existent air- cooling equipment.

Cooling loss during distribution for the cooling system:

$$Q_{c,d} = (1 - \eta_{c,d}) \cdot Q_{c,b} \qquad (K.7)$$

where:

$\eta_{c,d}$ is the degree of utilization of the distribution (see Table K.2).

Cooling loss of storage for the cooling system:

$$Q_{c,s} = 0 \qquad (K.8)$$

Table K.2 - Factors (mean annual values) for space cooling

Cooling system	$\eta_{c,ce,sens}$	$\eta_{c,ce}$	$\eta_{c,d}$
Cold water 6/12	0,87	1,00	0,90
Cold water 8/14 (e. g. fan convector)	0,90	1,00	0,90
Cold water 14/18 (e. g. fan convector, induction)	1,00	1,00	1,00
Cold water 16/18 (e. g. cooling ceiling)	1,00	1,00	1,00
Cold water 18/20 (e. g. building component activation)	1,00	0,90	1,00
Direct evaporation	0,87	1,00	0,90 = 1 if it has already been accounted for in the machine

Other intermediate values for the cold-water input temperature must be interpolated.

If multiple generator circuits are involved, the refrigeration energy output for the space-cooling system must be calculated separately for each cooling generator system. The portions of internal foreign heat output from the cooling system are low and are not taken into account in the following.

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Annex L
(informative)

Auxiliary energy use by terminals

(Danish/German proposal)

L.1 Energy demand for space cooling – fans

The electrical energy demand $Q_{c,ce,aux}$ of secondary air fans for space cooling is calculated in relation to the type of device and the individual device designs. The specified energy demand refers to devices with multi-stage speed control (based on 1 000 h operation of the fan-operated convectors and 500 full cooling system utilization hours).

$$Q_{c,ce,aux} = f_{c,ce,aux} \cdot Q_{c,outg} \cdot t_{C,op}/1000\ \text{h} \tag{L.1}$$

where:

$f_{c,ce,aux}$ is the specific energy demand of the secondary air fans;

$Q_{c,outg}$ is the generator cooling output for space cooling (see Equation K.1);

$t_{C,op}$ is the space cooling demand time.

For other designs (e. g. precision air conditioners, climate-controlled cabinets etc.) the respective manufacturer's specifications must be used. The electrical power of the fans should be derived from the cooling power (nominal data) and their energy demand evaluated on the basis of 1 000 operating hours.

Table L.1 - Standard values for the specific energy demand of fans for space cooling

	Rated power kW/kW	$f_{c,ce,aux}$ kWh/kWh
Space air conditioners: DX internal units with air distribution via ducts and individual air vents	0,030	0,060
Space air conditioners: DX internal units with ceiling cassettes	0,020	0,040
Space air conditioners: DX internal units, wall and parapet-mounted units	0,020	0,040
Cold-water fan convectors, parapet and ceiling-mounted units cold water 6 °C	0,020	0,040
Cold-water fan convectors, parapet and ceiling-mounted units cold water 14 °C	0,035	0,070
Cold-water fan convectors, ceiling-mounted units with air distribution via ducts, cold water 14 °C	0,040	0,080

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Annex M
(informative)

Auxiliary energy demand, heat rejection

M.1 Calculation

The energy balance evaluation of heat rejection is carried out on the basis of the specific design-related electrical energy demand $q_{R,electr}$ of the heat rejection equipment and a mean utilization factor $f_{R,av}$ of the heat rejection system. If the rated heat rejection power and the mean operating time are taken into account, the final energy demand of the heat rejection system is expressed by the following equation:

$$Q_{C,f,R,electr} = \dot{Q}_{R,outg} \cdot q_{R,electr} \cdot f_{R,av} \cdot t_{R,op} \tag{M.1}$$

where:

$Q_{C,f,R,electr}$ is the final energy demand of the heat rejection system (electrical), in kWh.

$\dot{Q}_{R,outg}$ is the rated heat rejection power, in kW;

$q_{R,electr}$ is the specific electrical energy demand of the heat rejection system, in kW/kW;

$f_{R,av}$ is the mean utilization factor, heat rejection;

$t_{R,op}$ is the heat rejection operating time in hours.

with

$$\dot{Q}_{R,outg} = \dot{Q}_{c,outg} \cdot \left(1 + \frac{1}{EER}\right) \text{ for compressor-type refrigeration units} \tag{M.2}$$

or

$$\dot{Q}_{R,outg} = \dot{Q}_{c,outg} \cdot \left(1 + \frac{1}{\zeta}\right) \text{ for absorption chillers} \tag{M.3}$$

where:

$\dot{Q}_{c,outg}$ is the cooling power of the refrigerating unit, in kW;

EER is the energy efficiency ratio in kW/kW;

ζ is the rated heat ratio, in kW/kW.

When the index-value method is applied, a distinction is made between evaporative heat rejection (with open and with closed cooling-water circuits) and dry heat rejection systems. Table M.1. shows the specific electrical energy demand of heat rejection systems with and without additional silencers.

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Table M.1 - Specific electrical energy demand $q_{R,electr}$ of heat rejection systems

	Cooling tower or evaporative condenser (including water spray pumps)		Dry cooler
	Closed circuit	Open circuit	
	$q_{R,electr}$ kW/kW		
Without additional silencer (axial fan)	0,033	0,018	0,045
With additional silencer (radial fan)	0,040	0,021	—

The mean utilization factor $f_{R,av}$ for evaporative heat rejection systems can be taken from the tables in Annex I (Table I.12) as the factor $f_{R,VK}$ (applicable to both open and closed circuits), and the corresponding factor for dry heat rejection systems as factor $f_{R,TK}$ in the same tables for the respective type of building usage. In the case of deviating usages, a type of usage which corresponds to the planned utilization profile must be selected. If no other data are known, the part load values for personal offices – single occupant (usage type 1) must be used in the calculations. The values corresponding to the design-specification cooling-water temperature level as stated for evaluation of the refrigeration unit must be applied. The index values for heat rejection must be taken in relation to the cooling-water temperature control mode and in correlation to the *PLV* value of the respective refrigeration unit.

If only one room conditioning system or one HVAC system is supplied with cooling energy within one and the same type of usage, the utilization factor $f_{R,av}$ can be used directly for the energy evaluation calculations. Where environment control cooling is achieved in parallel by space-cooling and HVAC systems, the index values must be weighted according to the percentage contributions of the respective space-cooling and HVAC-cooling systems to the annual refrigeration energy output.

$$f_{R,av,n} = \frac{Q_{c,outg,a,n} \cdot f_{R,av} + Q_{c^*,outg,a,n} \cdot f_{R,av}}{Q_{C,outg,a,n}} \tag{M.4}$$

where:

$f_{R,av,n}$ is the mean utilization factor, heat rejection; depending on the type of usage;

$Q_{C,outg,a,n}$ is the annual usable refrigeration energy output per type of usage n, in kWh;

$Q_{\mathbf{c},outg,a,n}$ is the annual refrigeration energy output for cooling, room conditioning system per type of usage n, in kWh;

$Q_{\mathbf{c^*},outg,a,n}$ is the annual refrigeration energy output for cooling, HVAC system per type of usage n, in kWh.

The following equation applies to these terms:

$$Q_{C,outg,a,n} = Q_{c,outg,a,n} + Q_{c^*,outg,a,n} \tag{M.5}$$

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If only one type of usage occurs in a specific cooled sector, the utilization load factor $f_{R.av,n}$ can be used directly in the energy-related evaluation calculations. Where several types of usage occur in a specific cooled sector, the index values must be weighted according to the percentage contributions of the respective types of usage to the annual usable cooling energy output.

$$f_{R,av} = \frac{\sum_{1}^{n} Q_{C,outg,a,n} \cdot f_{R,av,n}}{Q_{C,outg,a}} \tag{M.6}$$

The heat rejection system operating time $t_{R,op}$ is assumed to be the maximum value of the annual total of the monthly cooling element demand times and/or of the monthly demand times of the room air conditioners.

$$t_{R,op,n} = Max\left(\sum_{1}^{12} t_{c^*,op,mth}; \sum_{1}^{12} t_{c,op,mth}\right)_n \tag{M.7}$$

where:

$t_{c^*,op,mth}$ is the monthly demand time of the cooling elements, in h (see E.2.4.2);

$t_{c,op,mth}$ is the monthly demand time of the room air-conditioning cooling, in h;

$t_{R,op,n}$ is the annual heat rejection system operating time in h, depending on the type of usage.

If there are several different types of usage in one and the same cooled sector, the operating time must be determined as the maximum value of the heat rejection system operating time of all types of usage.

$$t_{R,op} = Max(t_{R,op,n}) \tag{M.8}$$

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M.2 Partial-load index values of heat rejection systems

The utilization factor $f_{\mathrm{R,av}}$ of heat rejection systems of water-cooled refrigeration units must be calculated by applying Equation (M.9) to the mean value of the partial-load air-volume ratio of the heat rejection system. Since the overall efficiency of the fan decreases when running under partial-load conditions, only the square of the ratio of the fan power to air volume transported is used in the evaluation.

$$f_{\mathrm{R,av}} = \frac{\sum \left(\frac{\dot{V}_{\mathrm{L},i}}{\dot{V}_{\mathrm{L}}} \right)_i^2}{\sum\limits_{n=0,1}^{1} s_{\mathrm{n}}} \tag{M.9}$$

where:

$\left(\frac{\dot{V}_{\mathrm{L},i}}{\dot{V}_{\mathrm{L}}} \right)_i$ is the partial-load air-flow ratio of the heat rejection system in time interval i;

$f_{\mathrm{R,av}}$ is the utilization factor of the heat rejection system.

The partial-load air-flow ratio must be determined for every time interval i as a function of the cooling-water temperatures and the type of heat rejection, whereby Equation (M.10) is applied for evaporative heat rejection (VK) and Equation (M.13) for dry heat rejection (TK). In these calculations, the air-flow ratio must never be lower than the minimum limit of 25 %.

Cooling tower or evaporative condenser:

$$\left(\frac{\dot{V}_{\mathrm{L},i}}{\dot{V}_{\mathrm{L}}} \right)_{\mathrm{VK}} = \frac{\ln\left(1 - \frac{\eta_{\mathrm{V},i}}{c_{\mathrm{K}}}\right) \cdot l_{\mathrm{min},i}}{\ln\left(1 - \frac{\eta_{\mathrm{V}}}{c_{\mathrm{K}}}\right) \cdot l_{\mathrm{min}}} \quad \text{with} \quad \left(\frac{\dot{V}_{\mathrm{L},i}}{\dot{V}_{\mathrm{L}}} \right)_{\mathrm{VK}} \geq 0{,}25 \tag{M.10}$$

where:

η_V is the temperature ratio of evaporative heat rejection at the rated full power;

l_{min} is the relative minimum air volume flow at rated full power;

c_K is the cooling-tower constant of evaporative heat rejection systems;

$l_{\mathrm{min},i}$ is the relative minimum air volume flow in time interval i;

$\eta_{\mathrm{V},i}$ is the temperature ratio of the evaporative re-cooler within the time interval i.

The following parameter values must be used as standard values when evaluating heat rejection systems with the index-value method:

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- temperature ratio at rated full power η_V = 0,5 (cooling zone range 6 K, cooling difference limit 6 K);
- relative minimum air flow at rated full power l_{min} = 0,87;
- cooling-tower constant c_K = 0,8.

The temperature ratio $\eta_{V,i}$ per time interval i is calculated using Equation (M.11) and the relative minimum air flow $l_{min,i}$ per time interval i is calculated using Equation (M.12).

$$\eta_{V,i} = \frac{T_{W,1,i} - T_{W,2,i}}{T_{W,1,i} - T_{F,i}} \qquad \text{(M.11)}$$

$$l_{\text{min},i} = -0{,}0162 \cdot T_{F,i} + 1{,}2103 \qquad \text{(with } 10°\text{C} \leq T_{F,i} \leq 25°\text{C)} \qquad \text{(M.12)}$$

where:

$T_{W,1,i}$ is the cooling-water input temperature of the heat rejection system during time interval i, in °C;

$T_{W,2,i}$ is the cooling-water output temperature of the heat rejection system during time interval i, in °C;

$T_{F,i}$ is the wet-bulb temperature of outdoor-air during time interval i, in °C;

Dry coolers:

$$\left(\frac{\dot{V}_{L,i}}{\dot{V}_L}\right)_{TK} = \frac{\left(T_{W,1,i} - T_{W,2,i}\right)}{\left(T_{W,1,0} - T_{W,2,0}\right)} \cdot \frac{\left(T_{W,2,0} - T_{AU,0}\right)}{\left(T_{W,2,i} - T_{AU,i}\right)} \text{ with } \left(\frac{\dot{V}_{L,i}}{\dot{V}_L}\right)_{TK} \geq 0{,}25 \qquad \text{(M.13)}$$

where:

$T_{W,1,i}$ is the cooling-water input temperature of the heat rejection system during time interval i, in °C;

$T_{W,1,0}$ is the cooling-water input temperature of the heat rejection system when running at rated cooling power, in °C;

$T_{W,2,i}$ is the cooling-water output temperature of the heat rejection system during time interval i, in °C;

$T_{W,2,0}$ is the cooling-water output temperature of the heat rejection system when running at rated cooling power, in °C;

$T_{AU,0}$ is the outdoor-air temperature at which the rated power is defined, in C, ($T_{AU,0}$ = 32 °C);

$T_{AU,i}$ is the outdoor-air temperature during time interval i, in °C.

In detailed energy-balance studies, the operating time of the heat rejection system $t_{R,op}$ may differ from the value calculated by Equation (M.8). In such cases, Equation (M.14) should be applied to determine this time on the basis of the partial-load demand values s_n of all partial-load stages k_n :

$$t_R = \sum_{n=0,1}^{1} s_n \qquad \text{in h} \qquad \text{(M.14)}$$

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