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Building environment design — Embedded radiant heating and cooling systems —

Part 4: Dimensioning and calculation of the dynamic heating and cooling capacity of Thermo Active Building Systems (TABS)

Conception de l'environnement des bâtiments — Systèmes intégrés de chauffage et de refroidissement par rayonnement —

Partie 4: Dimensionnement et calculs relatifs au chauffage adiabatique et à la puissance frigorifique pour systèmes d'éléments de construction thermoactifs (TABS)

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Foreword

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

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

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For an explanation of the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT), see www.iso.org/iso/foreword.html.

This document was prepared by Technical Committee ISO/TC 205, *Building environment design*, in collaboration with the European Committee for Standardization (CEN) Technical Committee CEN/TC 228, *Heating systems and water based cooling systems in buildings,* in accordance with the Agreement on technical cooperation between ISO and CEN (Vienna Agreement).

This second edition cancels and replaces the first edition (ISO 11855-4:2012), which has been technically revised.

The main changes compared to the previous edition are as follows:

— editorial corrections;

— picture redraws;

— updated Bibliography;

— improved wording.

A list of all parts in the ISO 11855 series can be found on the ISO website.

Any feedback or questions on this document should be directed to the user’s national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Introduction

The radiant heating and cooling system consists of heat emitting/absorbing, heat supply, distribution, and control systems. The ISO 11855 series deals with the embedded surface heating and cooling system that directly controls heat exchange within the space. It does not include the system equipment itself, such as heat source, distribution system and controller.

The ISO 11855 series addresses an embedded system that is integrated with the building structure. Therefore, the panel system with open air gap, which is not integrated with the building structure, is not covered by this series.

The ISO 11855 series is applicable to water-based embedded surface heating and cooling systems in buildings. The ISO 11855 series is applied to systems using not only water but also other fluids or electricity as a heating or cooling medium. The ISO 11855 series is not applicable for testing of systems. The methods do not apply to heated or chilled ceiling panels or beams.

The object of the ISO 11855 series is to provide criteria to effectively design embedded systems. To do this, it presents comfort criteria for the space served by embedded systems, heat output calculation, dimensioning, dynamic analysis, installation, control method of embedded systems, and input parameters for the energy calculations.

The ISO 11855 series consists of the following parts, under the general title *Building environment design — Embedded radiant heating and cooling systems*:

— Part 1: *Definitions, symbols, and comfort criteria*

— Part 2: *Determination of the design heating and cooling capacity*

— Part 3: *Design and dimensioning*

— Part 4: *Dimensioning and calculation of the dynamic heating and cooling capacity of Thermo Active Building Systems (TABS)*

— Part 5: *Installation*

— Part 6: *Control*

— Part 7: *Input parameters for the energy calculation*

ISO 11855-1 specifies the comfort criteria which should be considered in designing embedded radiant heating and cooling systems, since the main objective of the radiant heating and cooling system is to satisfy thermal comfort of the occupants. ISO 11855-2 provides steady-state calculation methods for determination of the heating and cooling capacity. ISO 11855-3 specifies design and dimensioning methods of radiant heating and cooling systems to ensure the heating and cooling capacity. ISO 11855-4, this document, provides a dimensioning and calculation method to design Thermo Active Building Systems (TABS) for energy saving purposes, since radiant heating and cooling systems can reduce energy consumption and heat source size by using renewable energy. ISO 11855-5 addresses the installation process for the system to operate as intended. ISO 11855-6 shows a proper control method of the radiant heating and cooling systems to ensure the maximum performance which was intended in the design stage when the system is actually being operated in a building. ISO 11855-7 presents a calculation method for input parameters to ISO 52031.

Building environment design — Embedded radiant heating and cooling systems —

Part 4: Dimensioning and calculation of the dynamic heating and cooling capacity of Thermo Active Building Systems (TABS)

1 Scope

This document allows the calculation of peak cooling capacity of Thermo Active Building Systems (TABS), based on heat gains, such as solar gains, internal heat gains, and ventilation, and the calculation of the cooling power demand on the water side, to be used to size the cooling system, as regards the chiller size, fluid flow rate, etc.

This document defines a detailed method aimed at the calculation of heating and cooling capacity in non-steady state conditions.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements 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.

ISO 11855-1, *Building environment design — Embedded radiant heating and cooling systems — Part 1: Definitions, symbols, and comfort criteria*

ISO 11855-2, *Building environment design — Embedded radiant heating and cooling systems — Part 2: Determination of the design heating and cooling capacity*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 11855-1 apply.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at https://www.electropedia.org/

4 Symbols

For the purposes of this document, the symbols in Table 1 apply.

Table 1 — Symbols

Symbol	Unit	Quantity
A_F	m^2	Area of the heating/cooling surface area
A_W	m^2	Total area of internal vertical walls (i.e. vertical walls, external façades excluded)
C	J/(m^2·K)	Specific thermal capacity of the thermal node under consideration

Table 1 *(continued)*

Symbol	Unit	Quantity
C_W	J/(m²·K)	Average specific thermal capacity of the internal walls
c_j	J/(kg·K)	Specific heat of the material constituting the *j*-th layer of the slab
c_{Wa}	J/(kg·K)	Specific heat of water
d_a	m	External diameter of the pipe
E_{Day}	kWh/m²	Specific daily energy gains
f_{rm}^h	-	Running mode (1 when the system is running; 0 when the system is switched off) in the *h*-th hour
f_s	-	Design safety factor
$F_{v\ F\text{-}C}$	-	View factor between the floor and the ceiling
$F_{v\ F\text{-}EW}$	-	View factor between the floor and the external walls
$F_{v\ F\text{-}W}$	-	View factor between the floor and the internal walls
$h_{A\text{-}C}$	W/(m²·K)	Convective heat transfer coefficient between the air and the ceiling
$h_{A\text{-}F}$	W/(m²·K)	Convective heat transfer coefficient between the air and the floor
$h_{A\text{-}W}$	W/(m²·K)	Convective heat transfer coefficient between the air and the internal walls
$h_{F\text{-}C}$	W/(m²·K)	Radiant heat transfer coefficient between the floor and the ceiling
$h_{F\text{-}W}$	W/(m²·K)	Radiant heat transfer coefficient between the floor and the internal walls
H_A	W/K	Heat transfer coefficient between the thermal node under consideration and the air thermal node ("A")
H_C	W/K	Heat transfer coefficient between the thermal node under consideration and the ceiling surface thermal node ("C")
H_{Cct}	W/K	Heat transfer coefficient between the thermal node under consideration and the circuit
H_{CondDn}	W/K	Heat transfer coefficient between the thermal node under consideration and the next one
H_{CondUp}	W/K	Heat transfer coefficient between the thermal node under consideration and the previous one
H_{Conv}	-	Fraction of internal convective heat gains acting on the thermal node under consideration
H_F	W/K	Heat transfer coefficient between the thermal node under consideration and the floor surface thermal node ("F")
H_I	W/K	Coefficient connected to the inertia contribution at the thermal node under consideration
H_{IWS}	W/K	Heat transfer coefficient between the thermal node under consideration and the internal wall surface thermal node ("IWS")
H_{Rad}	-	Fraction of total radiant heat gains impinging on the thermal node under consideration
h_t	W/(m²·K)	Total heat transfer coefficient (convection + radiation) between surface and space
J	-	Number of layers constituting the slab as a whole
J_1	-	Number of layers constituting the upper part of the slab
J_2	-	Number of layers constituting the lower part of the slab
L_R	m	Length of installed pipes
$\dot{m}_{H,sp}$	kg/(m²·s)	Specific water flow in the circuit, calculated on the area covered by the circuit
m_j	-	Number of partitions of the *j*-th layer of the slab
n	-	Actual number of iteration in iterative calculations
n_h	h	Number of operation hours of the circuit
n_{Max}	-	Maximum number of iterations allowed in iterative calculations

Table 1 *(continued)*

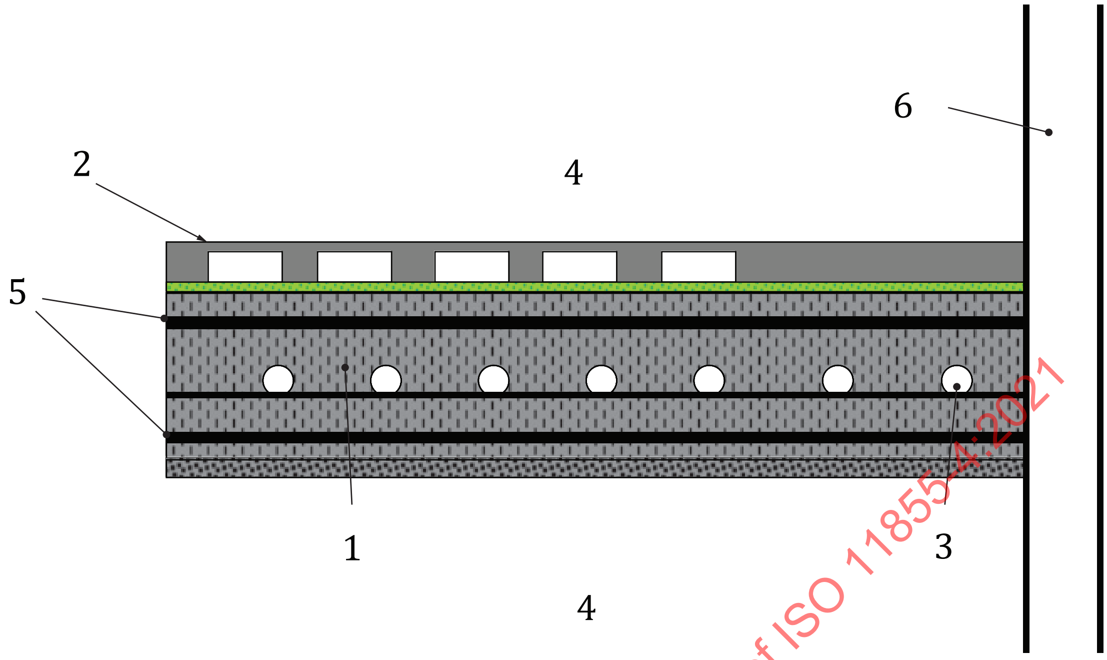
Symbol	Unit	Quantity
$P_{Cct}^{Max,h}$	W	Maximum cooling power reserved to the circuit under consideration in the *h*-th hour
$P_{Cct,Spec}^{Max}$	W/m^2	Maximum specific cooling power (per floor square metre)
q_i	W/m^2	Inward specific heat flux
q_u	W/m^2	Outward specific heat flux
Q_C^h	W	Heat flux impinging on the ceiling surface (“C”) in the *h*-th hour
Q_{Cct}^h	W	Heat flux extracted by the circuit in the *h*-th hour
Q_{Conv}^h	W	Total convective heat gains in the *h*-th hour
Q_F^h	W	Heat flux impinging on the floor surface (“F”) in the *h*-th hour
$Q_{IntConv}^h$	W	Internal convective heat gains in the *h*-th hour
Q_{IntRad}^h	W	Internal radiant heat gains in the *h*-th hour
Q_{IWS}^h	W	Heat flux impinging on the internal wall surface (“IWS”) in the *h*-th hour
$Q_{PrimAir}^h$	W	Primary air convective heat gains in the *h*-th hour
Q_{Rad}^h	W	Total radiant heat gains in the *h*-th hour
Q_{Sun}^h	W	Solar heat gains in the room in the *h*-th hour
Q_{Transm}^h	W	Transmission heat gains in the *h*-th hour
Q_W	W/m^2	Average specific cooling power
R	(m^2·K)/W	Generic thermal resistance
$R_{Add\ C}$	(m^2·K)/W	Additional thermal resistance covering the lower side of the slab
$R_{Add\ F}$	(m^2·K)/W	Additional thermal resistance covering the upper side of the slab
R_{int}	(m^2·K)/W	Internal thermal resistance of the slab conductive region
$R_{L,p}$	(m^2·K)/W	Conduction thermal resistance connecting the *p*-th thermal node with the boundary of the (p+1)-th thermal node
R_r	(m^2·K)/W	Pipe thickness thermal resistance
R_t	(m^2·K)/W	Circuit total thermal resistance
$R_{U,p}$	(m^2·K)/W	Conduction thermal resistance connecting the *p*-th thermal node with the boundary of the (p-1)-th thermal node
R_W	(m^2·K)/W	Wall surface thermal resistance
R_{Wa}	(m^2·K)/W	Water flow thermal resistance
R_x	(m^2·K)/W	Pipe level thermal resistance
R_z	(m^2·K)/W	Convection thermal resistance at the pipe inner side
s_r	m	Pipe wall thickness
s_1	M	Thickness of the upper part of the slab
s_2	m	Thickness of the lower part of the slab
W	m	Pipe spacing
δ_j	m	Thickness of the *j*-th layer of the slab
$\Delta\theta$	K	Generic temperature difference

Table 1 *(continued)*

Symbol	Unit	Quantity
$\Delta\theta_{Comfort}^{Max}$	K	Maximum operative temperature drift allowed for comfort conditions
Δt	s	Calculation time step
θ_{A}^{h}	°C	Temperature of the air thermal node (“A”) in the *h*-th hour
θ_{C}^{h}	°C	Temperature of the ceiling surface thermal node (“C”) in the *h*-th hour
$\Delta\theta_{Comf}^{Max}$	°C	Maximum operative temperature allowed for comfort conditions
$\theta_{Comf,Ref}$	°C	Maximum operative temperature allowed for comfort conditions in the reference case
θ_{F}^{h}	°C	Temperature of the floor surface thermal node (“F”) in the *h*-th hour
θ_{IW}^{h}	°C	Temperature of the core of the internal walls thermal node (“IW”) in the *h*-th hour
θ_{IWS}^{h}	°C	Temperature of the internal wall surface thermal node (“IWS”) in the *h*-th hour
θ_{MR}^{h}	°C	Room mean radiant temperature in the *h*-th hour
θ_{Op}^{h}	°C	Room operative temperature in the *h*-th hour
θ_{p}^{h}	°C	Temperature of the *p*-th thermal node in the *h*-th hour
θ_{PL}^{h}	°C	Temperature of the pipe level thermal node (“PL”) in the *h*-th hour
θ_{Slab}^{Av}	°C	Daily average temperature of the conductive region of the slab
$\theta_{Wa,In}^{h}$	°C	Water inlet actual temperature in the *h*-th hour
$\theta_{Wa,In}^{Setp,h}$	°C	Water inlet set-point temperature in the *h*-th hour
$\theta_{Wa,In,Ref}^{Setp}$	°C	Water inlet set-point temperature in the reference case
$\theta_{Wa,Out}^{h}$	°C	Water outlet temperature in the *h*-th hour
λ_{b}	W/(m·K)	Thermal conductivity of the material of the pipe embedded layer
λ_{j}	W/(m·K)	Thermal conductivity of the material constituting the *j*-th layer of the slab
λ_{r}	W/(m·K)	Thermal conductivity of the material constituting the pipe
ξ	K	Actual tolerance in iterative calculations
ξ_{Max}	K	Maximum tolerance allowed in iterative calculations
ρ_{j}	kg/m^3	Density of the material constituting the *j*-th layer of the slab
ω	various	Slope of correlation curves

5 The concept of thermally active building surfaces (TABS)

A thermally active building surface (TABS) is an embedded water-based surface heating and cooling system, where the pipe is embedded in the central concrete core of a building construction (see Figure 1).

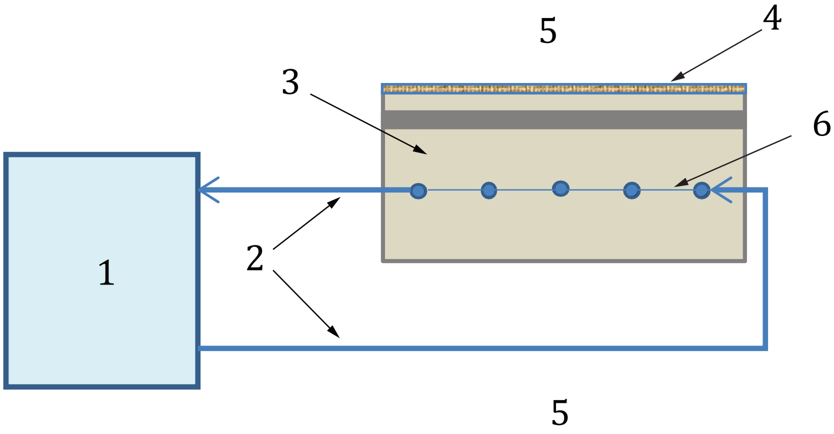


Key

1 concrete
2 floor
3 pipes
4 room
5 reinforcement
6 window

Figure 1 — Example of position of pipes in TABS

The building constructions embedding the pipe are usually the horizontal ones. As a consequence, in the following sections, floors and ceilings are usually referred to as active surfaces. Looking at a typical structure of a thermally active building surfaces (TABS), heat is removed by a cooling system (for instance, a chiller), connected to pipes embedded in the slab. The system can be divided into the elements shown in Figure 2.



Key

1 heating and cooling equipment

2 hydraulic circuit

3 slab including core layer with pipes

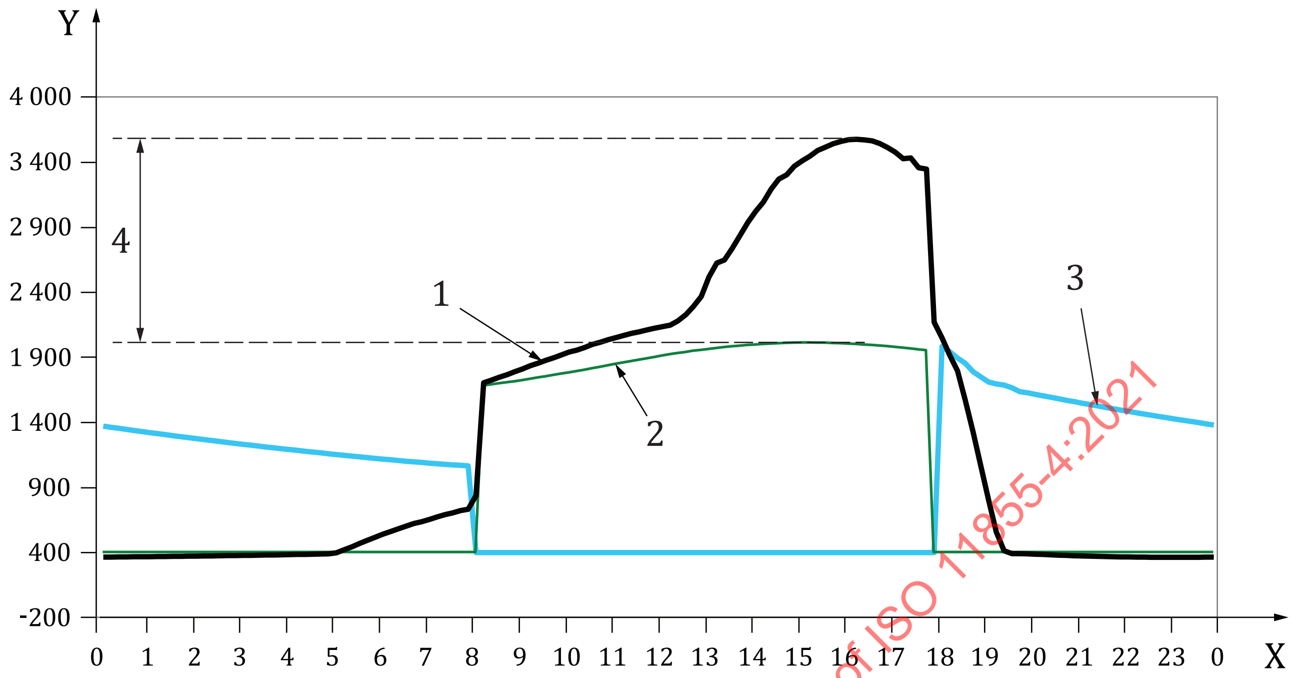
4 possible additional resistances (floor covering or suspended ceiling)

5 room below and room above

6 pipe level

Figure 2 — Simple scheme of a TABS

Thermally active surfaces exploit the high thermal inertia of the slab in order to perform the peak-shaving. The peak-shaving consists in reducing the peak in the required cooling power (see Figure 3), so that it is possible to cool the structures of the building during a period in which the occupants are absent (during night time, in office premises). This way the energy consumption can be reduced and a lower night time electricity rate can be used. At the same time a reduction in the size of heating and cooling system components (including the chiller) is possible.



Key

X time, h

Y cooling power, W

1 heat gain

2 cooling power needed for conditioning the ventilation air

3 cooling power needed on the water side

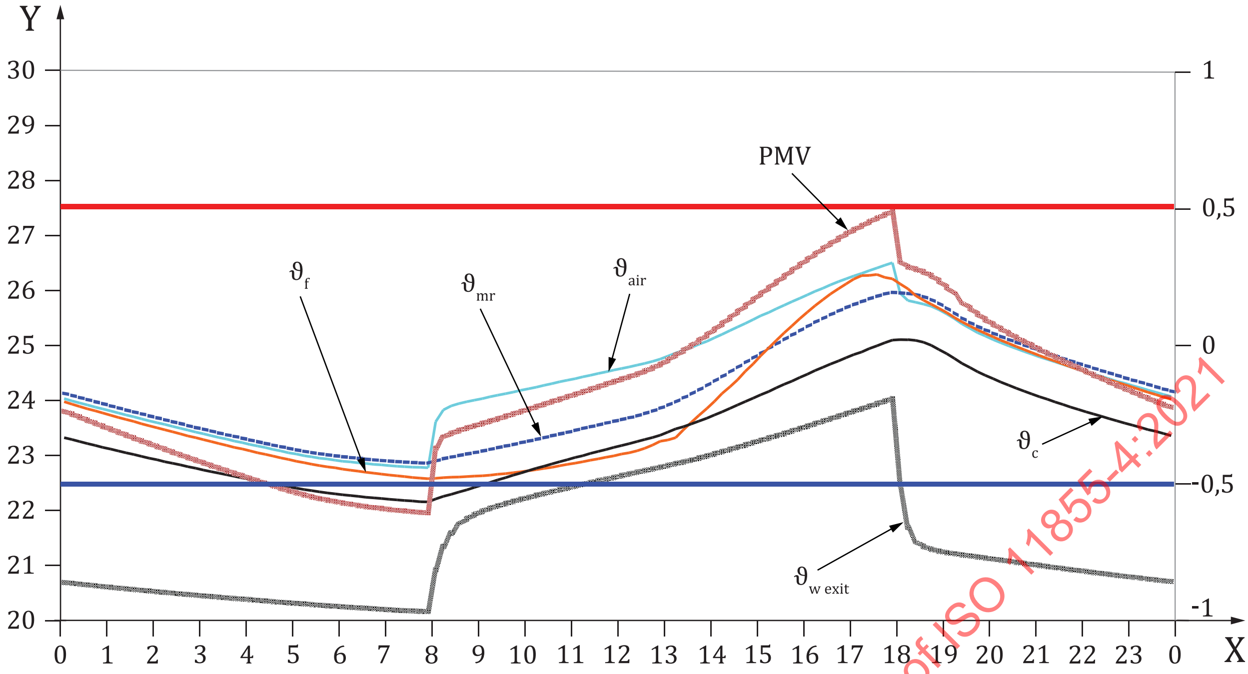
4 reduction of the required peak power

Figure 3 — Example of peak-shaving effect

TABS can be used both with natural and mechanical ventilation (depending on weather conditions). Mechanical ventilation with dehumidifying can be required depending on external climate and indoor humidity production. In the example in Figure 3, the required peak cooling power needed for dehumidifying the air during day time is sufficient to cool the slab during night time.

As regards the design of TABS, the planner needs to know if the capacity at a given water temperature is sufficient to keep the room temperature within a given comfort range. Moreover, the planner needs also to know the heat flux on the water side to be able to dimension the heat distribution system and the chiller and boiler. This document provides methods for both purposes.

When using TABS, the indoor temperature changes moderately during the day and the aim of a good TABS design is to maintain internal conditions within the range of comfort, i.e. −0,5 < PMV < 0,5, during the day, according to ISO 7730 (see Figure 4).



Key

X	time, h
Y	temperature, °C
PMV	predicted mean vote
θ_{air}	air temperature
θ_c	ceiling temperature
θ_{mr}	mean radiant temperature
θ_f	floor temperature
$\theta_{w\ exit}$	water return temperature

Figure 4 — Example of temperature profiles and PMV values vs. time

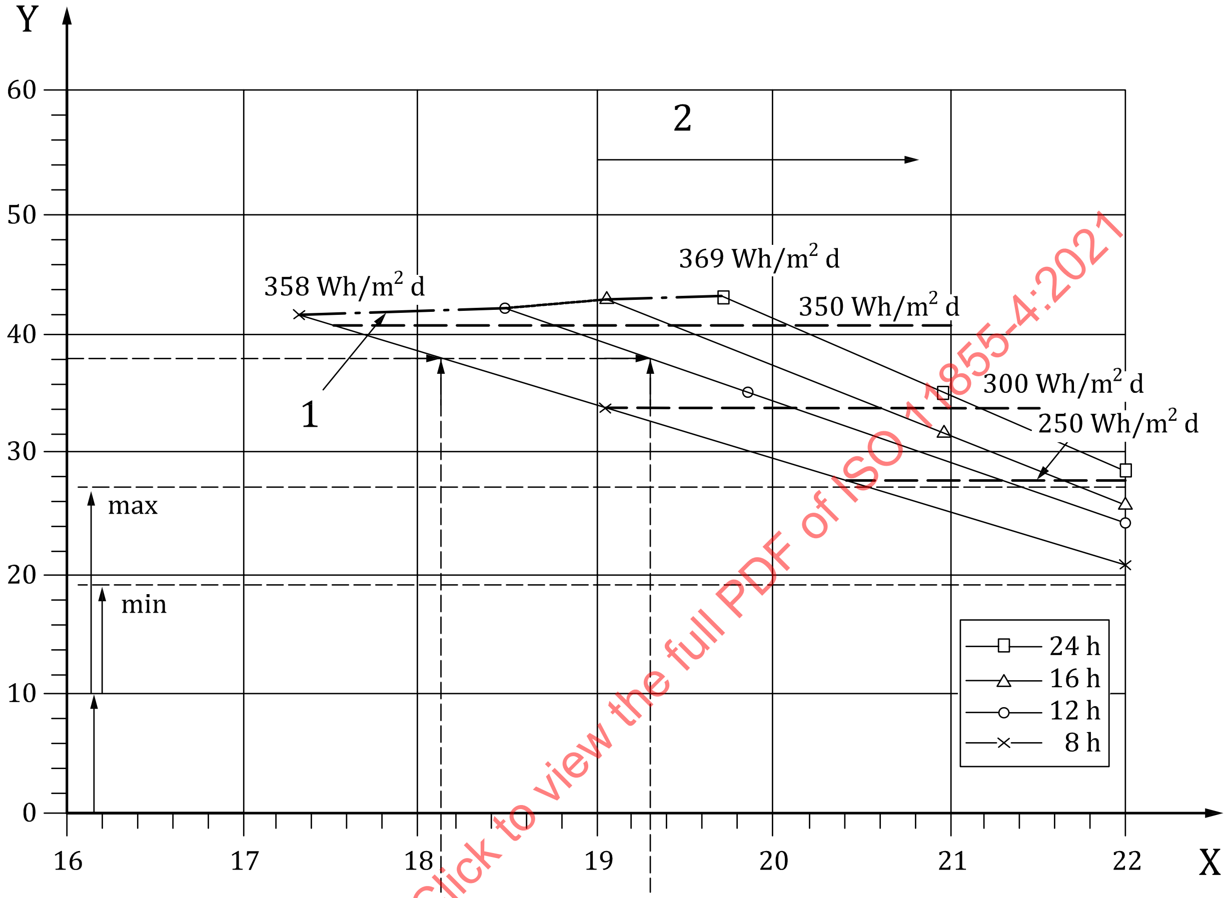
Some detailed building system calculation models have been developed to determine the heat exchanges under unsteady state conditions in a single room, the thermal and hygrometric balance of the room air, prediction of comfort conditions, check of condensation on surfaces, availability of control strategies and calculation of the incoming solar radiation. The use of such detailed calculation models is, however, limited due to the high amount of time needed for the simulations. The development of a more user-friendly tool is required. Such a tool is provided in this document and allows the simulation of TABS.

The diagrams in Figure 5 show an example of the relation between internal heat gains, water supply temperature, heat transfer on the room side, hours of operation and heat transfer on the water side. The diagrams refer to a concrete slab with raised floor (R = 0,45 (m²·K)/W) and an allowed room temperature range of 21 °C to 26 °C.

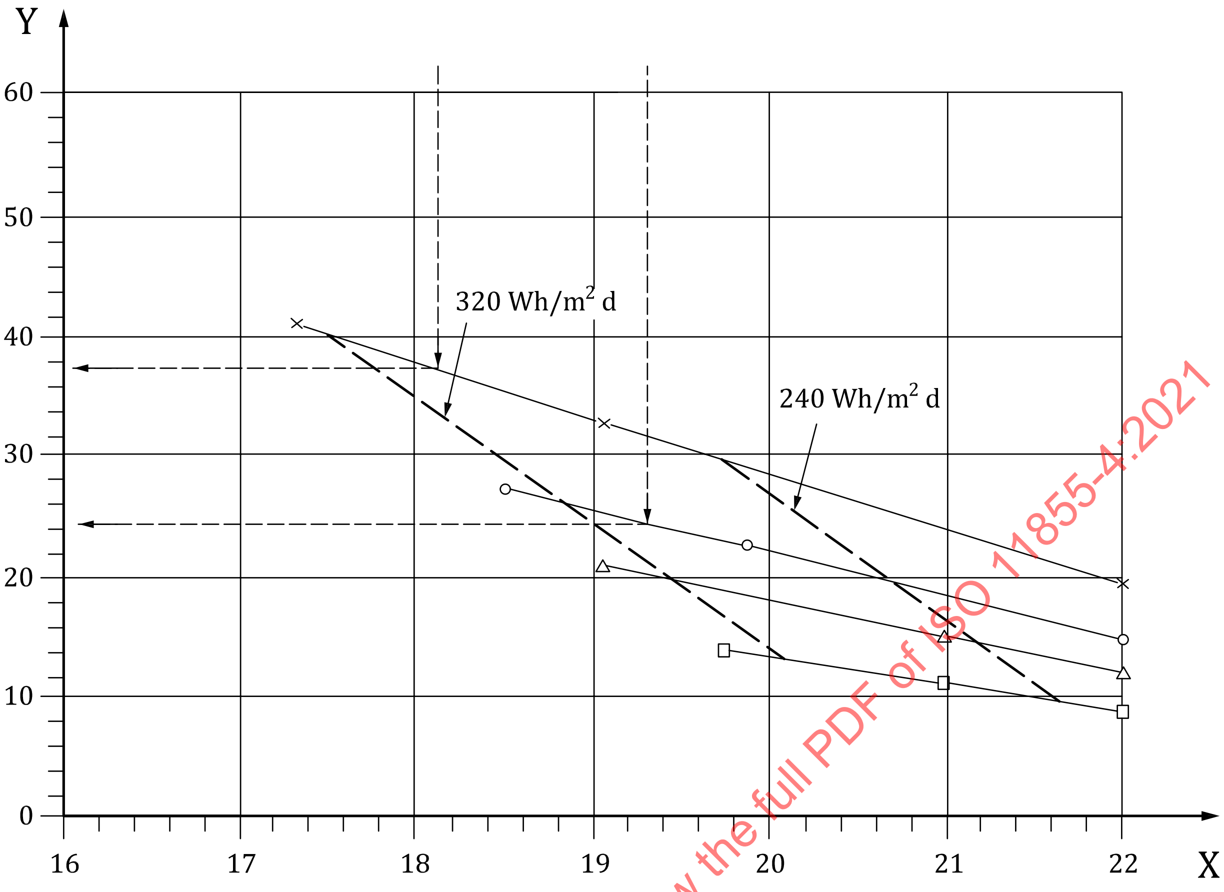
The upper diagram shows on the Y-axis the maximum permissible total heat gain in space (internal heat gains plus solar gains) [W/m²], and on the X-axis the required water supply temperature. The lines in the diagram correspond to different operation periods (8 h, 12 h, 16 h, and 24 h) and different maximum amounts of energy supplied per day [Wh/(m²·d)].

The lower diagram shows the cooling power [W/m²] required on the water side (to dimension the chiller) for TABS as a function of supply water temperature and operation time. Further, the amount of energy rejected per day is indicated [Wh/(m²·d)].

The example shows that, for a maximum internal heat gain of 38 W/m² and 8 h operation, a supply water temperature of 18,2 °C is required. If, instead, the system is in operation for 12 h, a supply water temperature of 19,3 °C is required. In total, the amount of energy rejected from the room is approximately 335 Wh/m^2 per day. In the same conditions, the required cooling power on the water side is 37 W/m^2 (for 8 h operation) and 25 W/m^2 (for 12 h operation) respectively. Thus, by 12 h operation, the chiller can be much smaller.



a)



b)

Key

X (upper diagram)	supply temperature tabs, °C
Y (upper diagram)	maximum total heat gain in space (W/m^2, floor area)
Y (lower diagram)	mean cooling power tabs (W/m^2, floor area)
1	maximurn temperature increase (21 °C - 26 °C)
2	self-regulating effect of slab

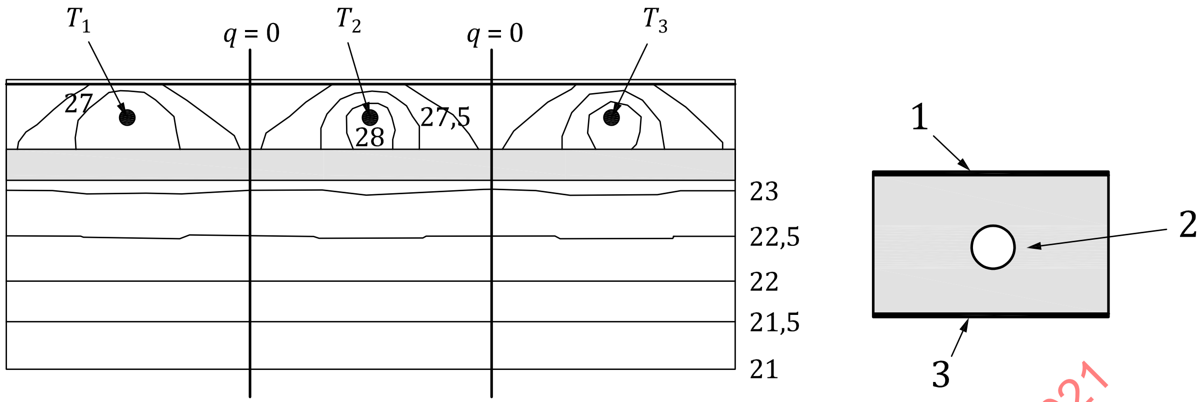
Figure 5 — Working principle of TABS

6 Calculation methods

6.1 General

TABS are systems with high thermal inertia. Therefore, for sizing chillers coupled with them, dynamic simulations shall be carried out. In principle, the solution of heat transfer inside structures with embedded pipes shall deal with 2-D calculations (see Figure 6). The calculation time required to consider the 2-D thermal field and the overall balance with the rest of the room is usually too high. Therefore, mathematical models in literature are usually based on a link between the pipe surface and the upper and lower surfaces (i.e. floor and ceiling).

One possibility to model radiant systems is to apply response factors to the pipe surface, upper surface and lower surface of the slab (see Figure 7). This way, the conduction heat transfer is defined via nine response factor series, that can be reduced to six response factor series, because of reciprocity rules.



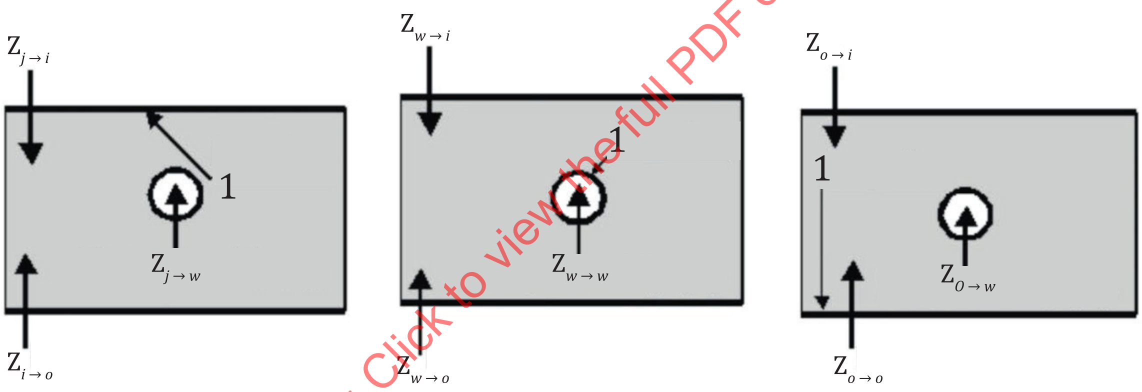
Key

1 upper surface

2 pipe surface

3 lower surface

Figure 6 — Heat transfer through structures containing pipes

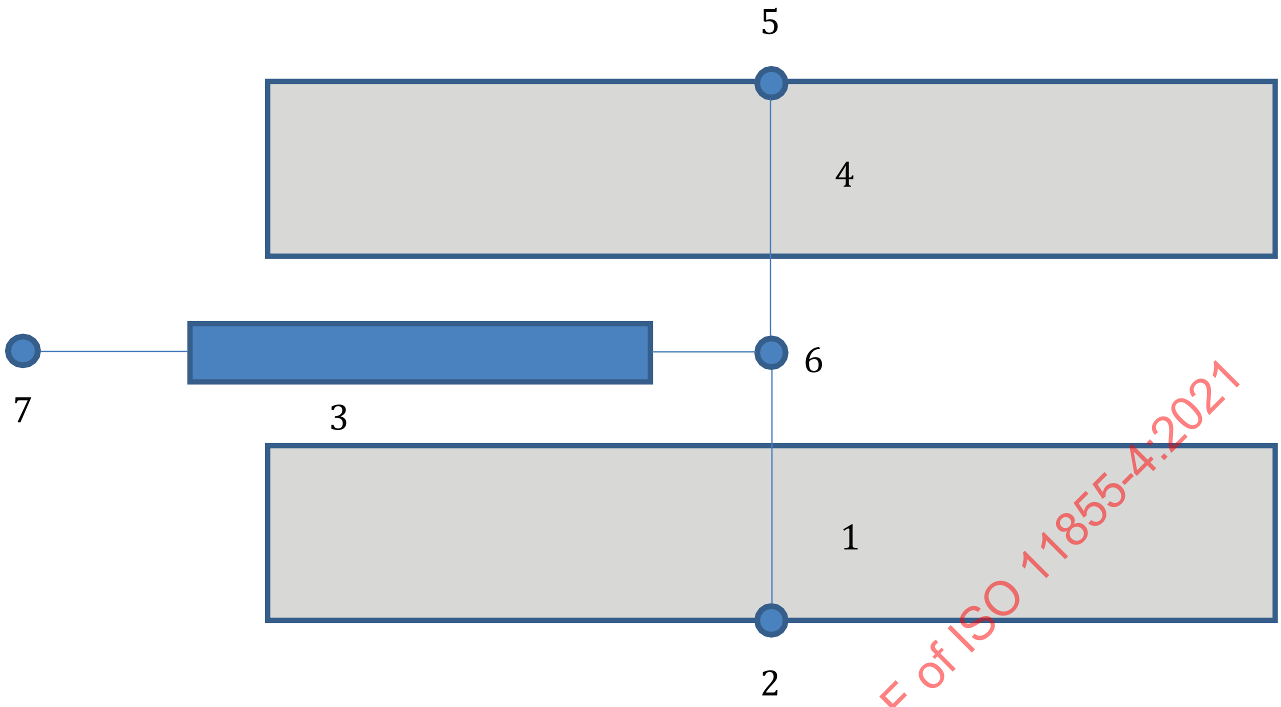


Key

1 impulse

Figure 7 — Transfer functions for building elements containing pipes

Another possibility is to consider a resistance between the external pipe surface and an equivalent core temperature at pipe level, which represents the average temperature along the axial plane of the pipes (see Figure 8). From the core level to upward and downward levels, a 1-D resistance-capacity network or 1-D response factor series (or transfer function) can be applied.



Key

1	lower part of the slab
2	lower surface temperature (ceiling)
3	circuit total thermal resistance
4	upper part of the slab
5	upper surface temperature (floor)
6	mean temperature at the pipe level
7	water supply temperature

Figure 8 — Simplified model for the conductive heat transfer in a structure containing pipes

In this document, the following calculation methods are presented.

— Rough-sizing method, based on a standard calculation of the cooling load (error: 20 % to 30 %). To be used starting from the knowledge of the daily heat gains in the room (see 6.2).

— Simplified method using diagrams for sizing, based on the knowledge of the total energy to be extracted daily to ensure comfort conditions (error: 15 % to 20 %). For details, see 6.3.

— Simplified model based on finite difference method (FDM) (error: 10 % to 15 %). It consists in detailed dynamic simulations predicting the heat transfers in the slab and even in the room via FDM. Based on the knowledge of the values of the variable cooling loads of the room during each hour of the day. For further details, see 6.4. Annex A describes simplified diagrams based on the simplified calculation method reported in 6.4,

— Detailed simulation models (error: 6 % to 10 %). It implies the overall dynamic simulation model for the radiant system and the room via detailed building-system simulation software (see 6.5).

6.2 Rough sizing method

The cooling system shall be sized via the following Formula (1):

$$P_{\text{Cct,Spec}}^{\text{Max}} = \frac{E_{\text{Day}}}{n_{\text{h}}} \cdot 1\ 000 \cdot f_{\text{s}} \tag{1}$$

where

$P_{\text{Cct,Spec}}^{\text{Max}}$	is the maximum specific cooling power (per floor square metre) in W/m^2;
E_{Day}	is the specific daily energy gains in kWh/m^2;
n_{h}	is the number of operation hours of the circuit in h;
f_{s}	is the safe design factor (greater than one, usually 1,15) in -.

For this purpose, E_{Day} shall be calculated in the following way:

— the hourly values of heat gains are calculated for the room under the design conditions and occupancy schedules, via an energy simulation tool or a proper method for the calculation of heat gains;

— E_{Day} is the sum of the 24 values of heat gains.

The heat gains calculation shall be carried out using an operative temperature 0,5 °C lower than the average operative temperature during occupancy hours, for the sake of safe design. As a consequence, if the room operative temperature drift during occupancy hours is 21,0 °C to 26,0 °C, then the room average operative temperature during occupancy hours is 23,5 °C, and the reference room operative temperature for the calculation of heat gains is 23,0 °C.

6.3 Simplified sizing by diagrams

In this case, the calculation of the heat gains shall be carried out by means of the value of the total cooling energy to be provided during the day in order to ensure comfort conditions at the average operative temperature (for instance, 23,0 °C). This method is based on the assumption that the entire thermally conductive part of the slab is maintained at an almost constant temperature during the whole day, due to its own thermal inertia and the thermal resistance dividing it from the rooms over and below. This average temperature of the slab is calculated by the method itself and is used to calculate the water supply temperature depending on the running time of the circuit.

The following magnitudes are involved in this method.

— E_{Day} : specific daily energy gains in the room during the design day. It consists of the sum of heat gains values acting during the whole design day, divided by the floor area, in kWh/m^2.

— $\theta_{\text{Comf}}^{\text{Max}}$: maximum operative room temperature allowed for comfort conditions, in °C.

— Orientation of the room: used to determine when the peak load in heat gains happens [east (morning), south (noon) or west (afternoon)].

— Number of active surfaces: distinguishes whether the slab works transferring heat both through the floor side and through the ceiling side or just through the ceiling side (see Figures 9, 10 and 11).

— n_{h} : number of operation hours of the circuit in h.

— R_{Int} : internal thermal resistance of the slab conductive region in (m^2·K)/W. It is the average thermal resistance that connects the conductive parts of the slab placed near the pipe level to the pipe level itself [see Formula (4)].

— θ_{Slab}^{Av}: daily average temperature of the conductive region of the slab in °C. It is a result of the present method and depends on the number of active surfaces (ceiling only, or ceiling and floor), the running mode (24 h or 8 h) and the shape of the internal load profile (lunch break or not) and room orientation. The average temperature of the slab is achieved through coefficients included in the method by Formula (2).

$$\theta_{Slab}^{Av} = \theta_{Comf}^{Max} + \omega \cdot E_{Day} \quad (2)$$

where ω is a coefficient, whose values are given in Tables 2 and 3.

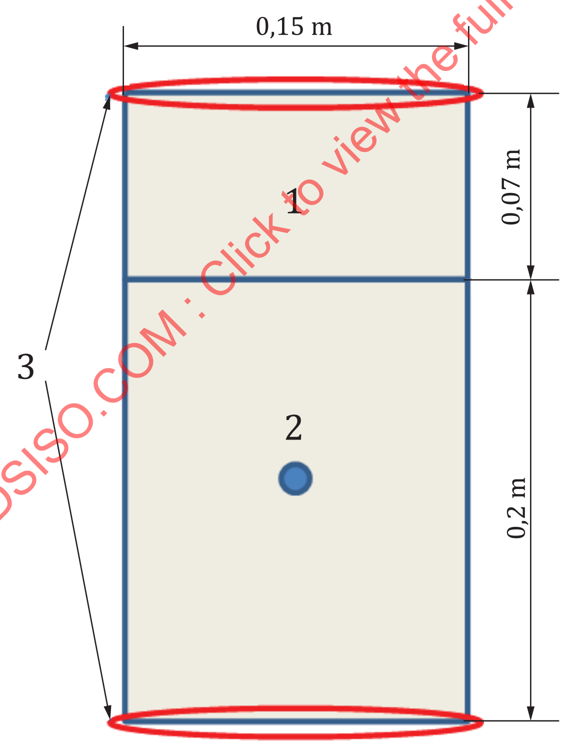
— R_t : circuit total thermal resistance, obtained via the resistance method (for further details, see ISO 11855-2) in (m^2·K)/W. This thermal resistance depends on the characteristics of the circuit, pipe, and conductive slab (see Figure 14).

— $\theta_{Wa,In}^{Setp,h}$: water supply temperature required for ensuring comfort conditions in °C.

It is obtained through Formula (3):

$$\theta_{Wa,In}^{Setp,h} = \theta_{Slab}^{Av} - \left(\frac{E_{Day} \cdot 1\,000}{h} \right) \cdot (R_{int} + R_t) \quad (3)$$

Dimensions in metres



Key

1 concrete

2 reinforced concrete

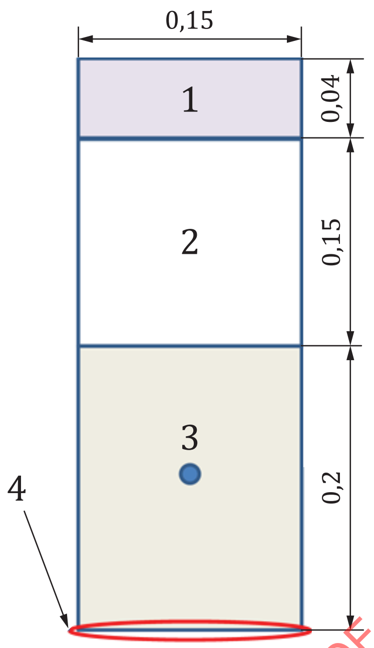
3 active surfaces

Conductive region: material 1 and material 2.

Number of active surfaces: 2.

Figure 9 — Example 1 — Conductive regions and numbers of active surfaces

Dimensions in metres



Key

1 wood

2 air

3 reinforced concrete

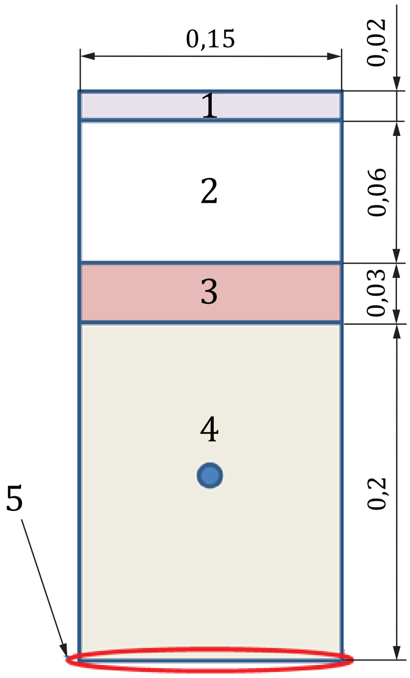
4 Active surface

Conductive region: material 3.

Number of active surfaces: 1.

Figure 10 — Example 2 — Conductive regions and numbers of active surfaces

Dimensions in metres



Key

1 wood

2 concrete

3 fibreglass

4 reinforced concrete

5 active surface

Conductive region: material 4.

Number of active surfaces: 1.

Figure 11 — Example 3 — Conductive regions and numbers of active surfaces

The internal thermal resistance of the slab conductive region (R_{int}) expressed in $(m^2 \cdot K)/W$ is the average thermal resistance that connects the conductive parts of the slab placed near the pipe level to the pipe level itself. Formula (4) describes how to calculate it.

$$R_{int} = \frac{\frac{R_{Up}}{2} \cdot \frac{R_{Down}}{2}}{\frac{R_{Up}}{2} + \frac{R_{Down}}{2}} \quad (4)$$

where

R_{Down} is the total thermal resistance of the lower part of the slab conductive region;

R_{int} is the internal thermal resistance of the slab conductive region;

R_{Up} is the total thermal resistance of the upper part of the slab conductive region.

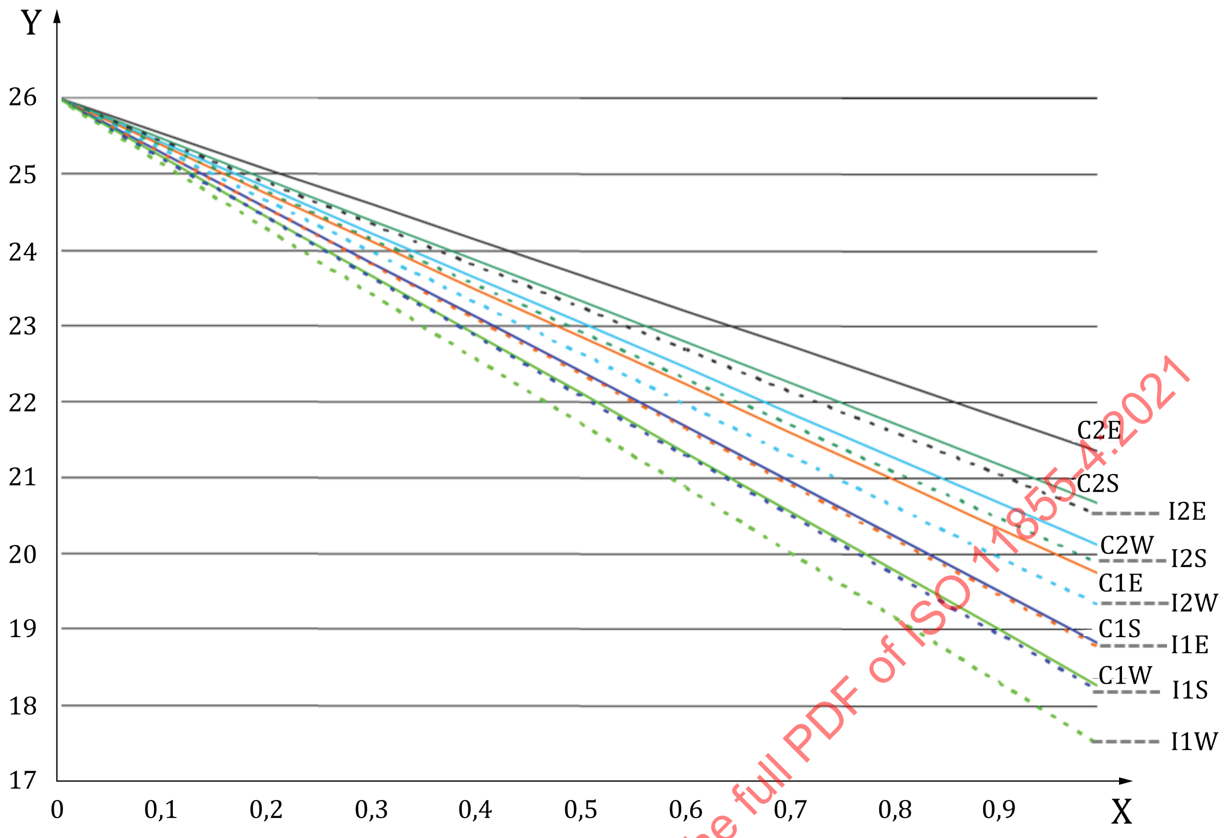
Table 2 — Constant internal heat gains from 8:00 to 18:00

Circuit running mode	Number of active surfaces	Orientation of the room		
		East (E)	South (S)	West (W)
		ω		
Continuous (24 h)	Floor and ceiling (C2)	-4,6 816	-5,3 696	-5,935
	Only ceiling (C1)	-6,3 022	-7,2 237	-7,7 982
Intermittent (8 h)	Floor and ceiling (I2)	-5,5 273	-6,1 701	-6,7 323
	Only ceiling (I1)	-7,2 853	-7,8 562	-8,5 791

Table 3 — Constant internal heat gains from 8:00 to 12:00 and from 14:00 to 18:00

Circuit running mode	Number of active surfaces	Orientation of the room		
		East (E)	South (S)	West (W)
		ω		
Continuous (24 h)	Floor and ceiling (C2)	-6,279	-7,1 094	-7,3 681
	Only ceiling (C1)	-7,9 663	-8,7 989	-8,7 455
Intermittent (8 h)	Floor and ceiling (I2)	-8,1 474	-8,758	-9,3 264
	Only ceiling (I1)	-10,029	-10,685	-10,967

By the choice of $\theta_{\text{Comf}}^{\text{Max}}$, it is possible to adapt the method to different maximum room operative temperatures, if the same maximum operative temperature drift allowed for comfort conditions is kept. Once $\theta_{\text{Comf}}^{\text{Max}}$ is defined, the tables can be summarized by diagrams. For example, if $\theta_{\text{Comf}}^{\text{Max}}$ = 26 °C, the diagram for constant internal heat gains from 8:00 to 18:00 is as given in Figure 12.



Key

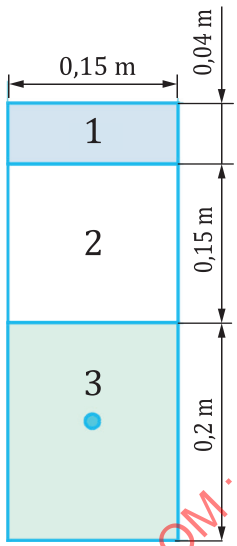
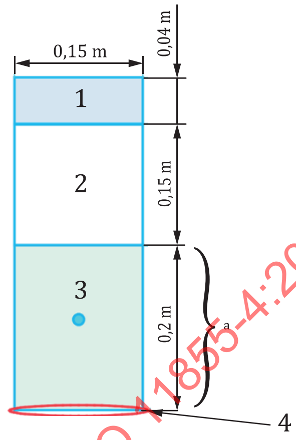
X E_{Day}, kWh/m²

Y θ_{slab}, °C

Figure 12 — Diagram for determining θ_{slab} as a function of the specific daily energy, exposure of the room (E = east, S = south, W = west), running mode of the circuit (C = continuous - 24 h, I = intermittent - 8 h), and number of active surfaces (1 or 2), in the case of constant internal heat gains during the day

Examples of calculation with input data and steps are presented in Table 4.

Table 4 — Example of TABS calculation

Input data	**Main calculation steps**
— E_{Day} : 0,6 kWh/m²; — Shape of thermal loads: no lunch break; — θ_{Comf}^{Max} : 26 °C; — Exposure of the room: south; — n_h: 24 h; — Thermal conductivity of the conductive region of the slab: 1,9 W/(m·K); — R_t: 0,07 (m²·K)/W; — Kind of floor:  **Key** 1 wood 2 air 3 reinforced concrete	Individuation of the conductive region and number of active surfaces:  **Key** 1 wood 2 air 3 reinforced concrete 4 active surface a Conductive region. — Calculation of R_{int}: — $R_{up} = R_{down} = 0,1/1,9 = 0,053$ (m²·K)/W — $R_{int} = 0,0265$ (m²·K)/W — Determination of ω (from Table 2): — 7,2 237 (m²·K)/kWh $\theta_{Slab}^{Av} = 26,0 - 7,2237 \cdot 0,6 = 21,7$ °C $\theta_{Wa,In}^{Setp} = 21,7 - \frac{0,6 \cdot (0,0265 + 0,07) \cdot 1000}{24} = 19,3$ °C

6.4 Simplified model based on FDM

The model is based on the calculation of the heat balance for each thermal node defined within the slab and the room. The slab and the room are divided into thermal nodes used to calculate the main heat fluxs taking place during the day. The temperature of each thermal node during the hour under consideration depends on the temperatures of the other thermal nodes during the same hour. As a consequence, the heat balances of all the thermal nodes would require the solution via a system of formulae, or an iterative solution. The last option is the one chosen in this document. As a consequence, most of the formulae regarding this method (see also Annex B) apply for each iteration executed in order to approach the final solution. The use of an iterative method requires the definition of four quantities:

n: actual number of the current iteration in -;

n_{Max}: maximum number of iterations allowed in -;

ξ: actual tolerance at the current iteration in K;

ξ_{Max}: maximum tolerance allowed in K.

The actual number of the current iteration and the actual tolerance at the current iteration are calculated at each iteration and compared with the maximum number of iterations and tolerance allowed respectively. In particular, if $\xi < \xi_{Max}$ and $n < n_{Max}$, then the solution has been found within the given conditions. Instead, if $n >= n_{Max}$, then the number of iterations performed has been too high and the solution has not reached the given accuracy. That would require a higher value of n_{Max} or ξ_{Max}, in case a lower degree in accuracy can be accepted.

6.4.1 Cooling system

As regards the cooling equipment, it is simulated via the following magnitudes:

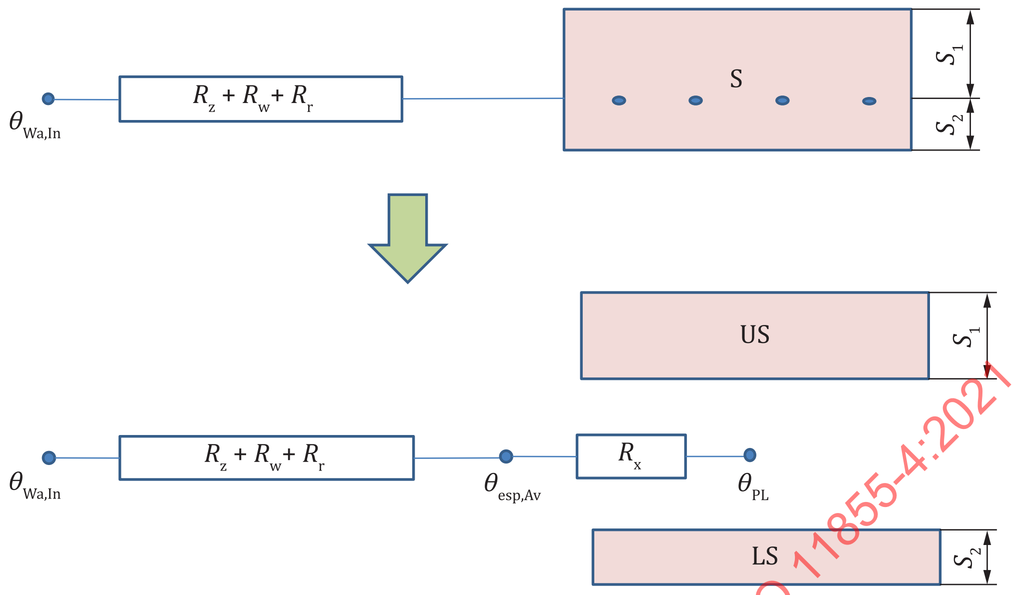
— $\theta_{Wa,In}^{Setp,h}$: water inlet set-point temperature in the h-th hour in °C;

— $P_{Cct}^{Max,h}$: maximum cooling power reserved to the circuit under consideration in the h-th hour in W.

The limited power of the cooling system shall be taken into account, since the chiller is able to keep a constant supply water temperature only when the heat flux extracted by the circuit is lower than the maximum cooling power expressed by the chiller. For further details, see Annex B. Additionally, Annex C provides a tutorial guide to assess the model and Annex D a computer program for TABS.

6.4.2 Hydraulic circuit and slab

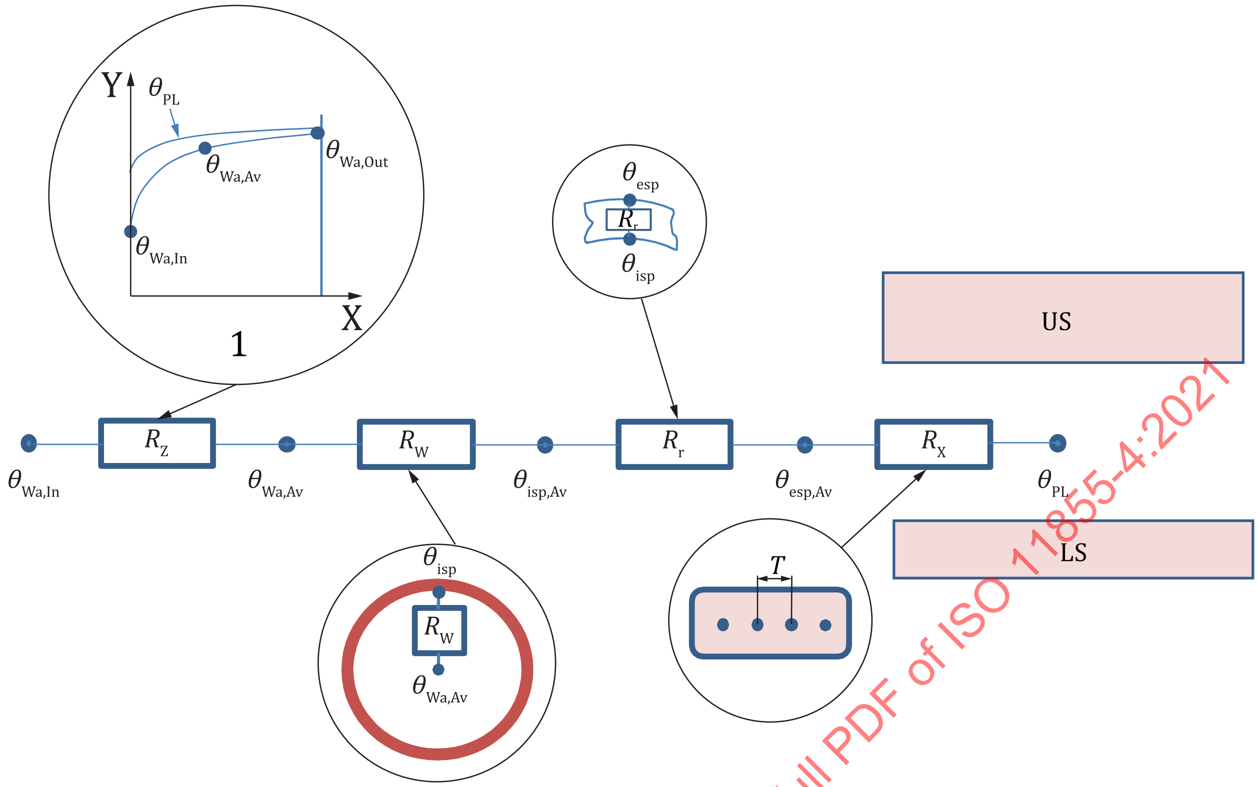
The resistance method (for further details, see ISO 11855-2) is applied. It sets up a straightforward relation, expressed in terms of resistances, between the water supply temperature and the average temperature at the pipe plane, θ_{PL} so that the slab can be split into two smaller slabs. In this way, the upper slab (which is above the pipe plane) and the lower slab (which is below the pipe plane) are considered separately (see Figures 13 and 14). Their thermal behaviour is analysed through an implicit finite-difference methods (FDM). Use Annex B for the calculation process.



Key

LS	lower part of the slab
R_r	pipe thickness thermal resistance
R_w	convection thermal resistance at the pipe inner side
R_x	pipe level thermal resistance
R_z	water flow thermal resistance
S	slab
S_1	thickness of the upper part of the slab
S_2	thickness of the lower part of the slab
US	upper part of the slab
$\theta_{esp,Av}$	average temperature at the outer side of the pipe
θ_{PL}	average temperature at the pipe level
$\theta_{Wa,In}$	water inlet temperature

Figure 13 — Concept of the resistance method



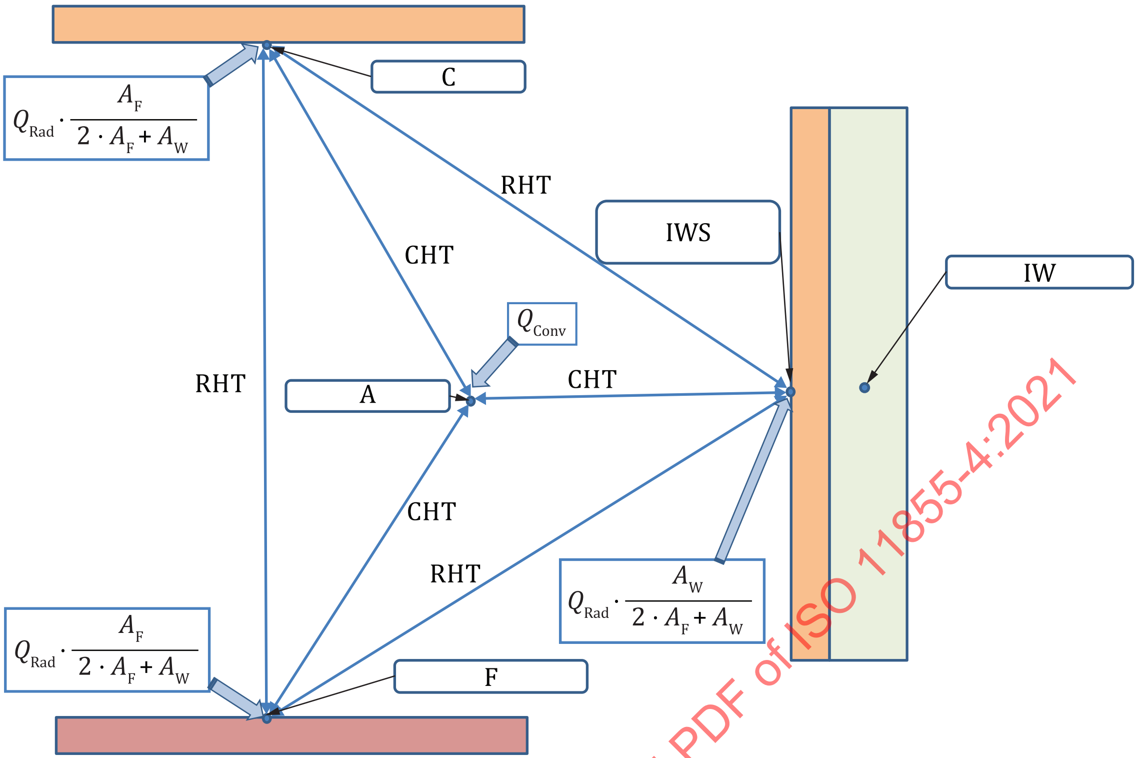
Key

L	length of installed pipes
LS	lower part of the slab
R_r	pipe thickness thermal resistance
R_W	convection thermal resistance at the pipe inner side
R_x	pipe level thermal resistance
R_z	water flow thermal resistance
T	pipe spacing
US	upper part of the slab
$\theta_{esp,Av}$	average temperature at the outer side of the pipe
$\theta_{isp,Av}$	average temperature at the inner side of the pipe
θ_{PL}	average temperature at the pipe level
$\theta_{Wa,Av}$	water average temperature
$\theta_{Wa,In}$	water inlet temperature
$\theta_{Wa,Out}$	water outlet temperature

Figure 14 — General scheme of the resistance method

6.4.3 Room

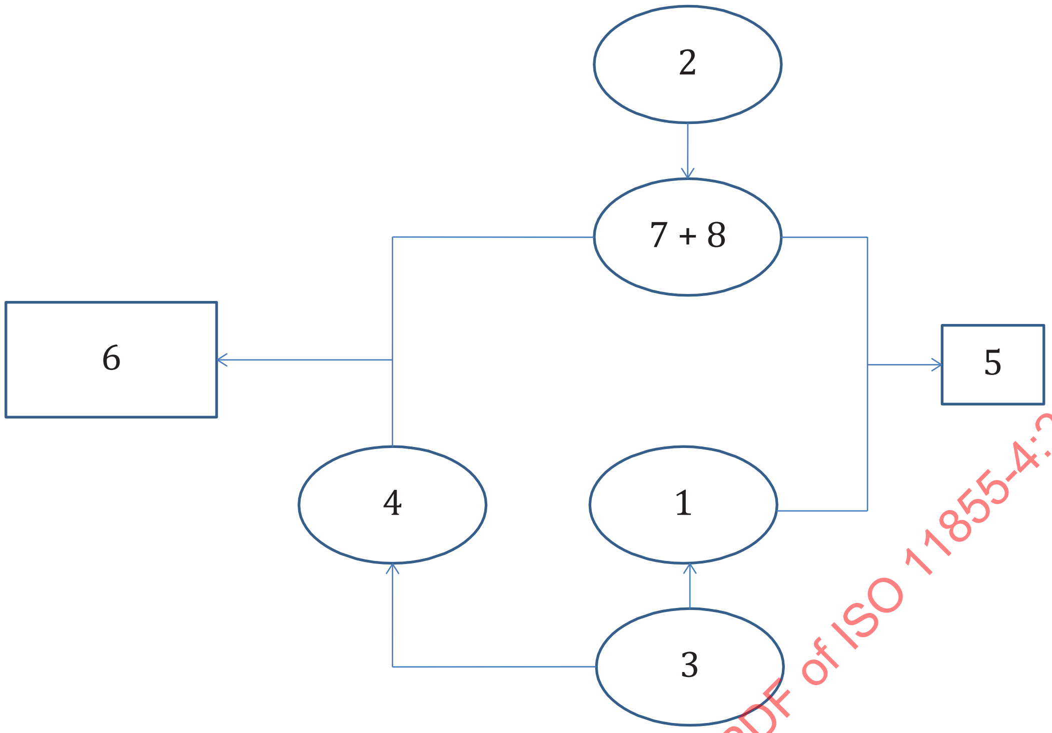
An air node is taken into account and connected with the upward and downward surface of the slab and with a fictitious thermal node at the wall surface. Two surfaces of the slab are connected to each other to take into account the radiation exchange between them, and finally each slab surface is connected to the wall surface node (see Figure 15). Moreover, hourly heat gains are distributed on air and surfaces, depending on their characteristics (see again Figure 15). The composition of heat gains is shown in Figure 16. For further details, see Annex B.



Key

A	thermal node representing the air in the room
C	thermal node representing the ceiling surface
CHT	convective heat transfer
F	thermal node representing the floor surface
IW	thermal node representing the internal walls
IWS	thermal node representing the internal wall surface
RHT	radiant heat transfer
Q_{Conv}	total convective heat gains
Q_{Rad}	total radiant heat gains

Figure 15 — Scheme of the thermal network representing the room



Key

1 convective internal heat gains
2 design weather conditions
3 internal heat gains
4 radiant internal heat gains
5 total convective heat gains
6 total radiant heat gains
7 solar gain
8 transmission through the external surfaces

Figure 16 — Heat loads acting in the room and how they take part in the calculations

6.4.4 Limits of the method

The following limitations shall be met:

— pipe spacing: from 0,15 m to 0,3 m;

— usual concrete slab structures have to be considered, λ = 1,15 - 2,00 W/(m·K), with upward additional materials, which might be acoustic insulation or raised floor. No discontinuous light fillings can be considered in the structures of the lower and upper slabs.

If these conditions are not fulfilled, a detailed simulation program has to be applied for dimensioning the TABS (see 6.5).

Under the above-mentioned conditions, a cooling load calculation or a simulation for a convective system can be carried out for an entire 24 h period and with an internal temperature equal to the average room operative temperature during the occupancy hours. The results of this calculation to be taken into account as input for the present simplified model are the solar heat gains and the heat fluxs into the room from the external surface.

6.5 Dynamic building simulation programs

For all cases which are not in the range of validation of the simplified methods, TABS calculations have to be carried out by means of a detailed dynamic building-system model.

These TABS calculations have to take into account the water flow into the pipes, the heat conduction between upward and downward surface of the slab and the pipe level, heat conduction of each wall, mutual radiation between internal surfaces, convection with air, and the thermal balance of the air.

Whenever results of TABS calculations are reported, the computer program applied shall be specified.

7 Effects of acoustic ceiling units on the cooling performance of TABS

Acoustic comfort, as well as thermal comfort, plays an important role for human well-being and productivity. TABS require large hard surfaces to be exposed, which could have a negative impact on the acoustic quality of indoor spaces. Free-hanging ceiling absorbers can be a solution for addressing acoustic concerns; however, they will affect the cooling performance of TABS when used in combination.

Different types of free-hanging ceiling absorbers (horizontal or vertical panels) can modify the cooling performance of the TABS and thermal comfort of the occupants.

The cooling performance of the TABS decreases when the ceiling surface coverage increases. The heat exchange between the room and the TABS is hindered when the celiling is covered with free-hanging horizontal sound absorbers.

The presence of the vertical baffles also has an effect on the cooling performance of TABS. The cooling performance decreases as the number of baffles increases. As a consequence of the reduction in cooling performance, the operative temperature in the occupied space increases.

For low sound absorption levels, the cooling performance of the TABS remains similar for horizontal and vertical panels. However, for higher sound absoption levels, horizontal panels have a higher influence on the cooling performance of the TABS than vertical baffles.

Horizontal sound absorbers can efficiently absorb a wider range of sound frequencies and require less absorptive material than vertical baffles for a given sound absorption; horizontal sound absorbers represent a more efficient solution when low sound absorption levels are required. When higher sound absorption levels are required, vertical baffles represent a better fit for purpose as they have a lower impact on the cooling performance of the TABS for equivalent sound absorption levels.

Sounds these panels need to be combined with wall-mounted acoustic units to achieve an optimal sound absorption in the full spectrum of frequencies. Vertical free hanging units, or baffles, are a significant alternative to be used in combination with TABS. Vertical baffles have a lower impact on the cooling performance of the TABS compared to horizontal panels for equivalent sound absorption levels.

Cold air stagnation in the plenum is the major problem for the convective heat exchange between the TABS and the room. The masking effect of the panels, especially horizontal sound absorbers, not only prevents stagnated cold air from mixing with the room air, but also degrades the cooling performance of the TABS.

8 Input for computer simulations of energy performance

To facilitate dynamic computer simulations of buildings with embedded radiant heating and cooling systems, the equivalent resistances between the heat conduction layer (pipe level) and the upward and downward surfaces can be used.

For type E, F, and G systems in ISO 11855-1, this resistance is directly calculated. Both the equivalent inward and outward resistance is calculated.

For type A, B, C and D systems (in ISO 11855-1 and the EN 1264 series) the equivalent resistance is calculated from the inward specific heat flux, q_i, and outward specific heat flux, q_u, taking into account the surface resistance according to this formula:

Equivalent resistance:

$$R = \Delta\theta/q - 1/h_t \quad (5)$$

where

$\Delta\theta$ is the heating and cooling medium temperature difference in K;

h_t is the total heat transfer coefficient (convection + radiation) between surface and space in W/(m^2·K).

Annex A
(informative)

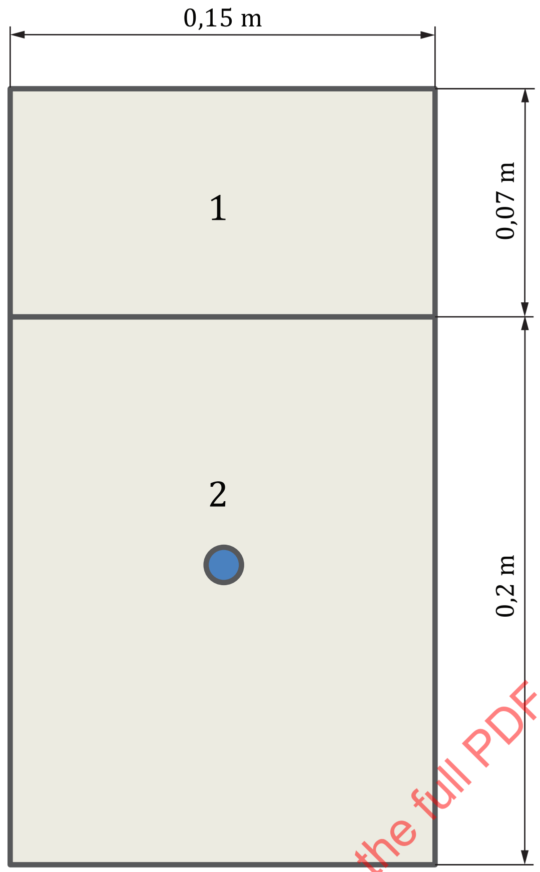
Simplified diagrams

Based on the simplified calculation method in 6.4, the following diagrams for design of a TABS have been developed. The diagram in Figure A.2 shows an example of the relation between internal heat gains, water supply temperature, heat transfer on the room side, hours of operation and heat transfer on the water side. The diagrams correspond to a concrete slab shown in Figure A.1 with a solid concrete floor, conductivity 1,2 W/(m·K), pipe spacing of 0,15 m and a permissible room temperature range of 21 °C to 26 °C.

The upper diagram in Figure A.2 shows on the Y-axis the maximum permissible total heat gain in space (internal gains plus solar gains) in W/m^2, and on the X-axis the required water supply temperature. The lines in the diagram correspond to different hours of operation (8 h, 12 h, and 24 h) and different daily energy gains in Wh/(m^2·d).

The lower diagram in Figure A.2 shows the cooling power in W/m^2 required on the water side (to size the chiller) for TABS as a function of water supply temperature and operation time. Further, the amount of energy rejected per day is indicated in Wh/(m^2·d).

The example shows that by a maximum internal heat gain of 48 W/m^2 and 8 h operation, a supply water temperature of 17,8 °C is required. If, instead, the system is in operation for 24 h, a supply water temperature of 21,3 °C is required. In total, the amount of energy rejected from the room is approximately. 460 Wh/m^2 per day. The required cooling power on the water side is for 8 h operation 58 W/m^2 and for 24 h operation only 20 W/m^2. Thus, for a 24 h operation, the chiller can be much smaller.

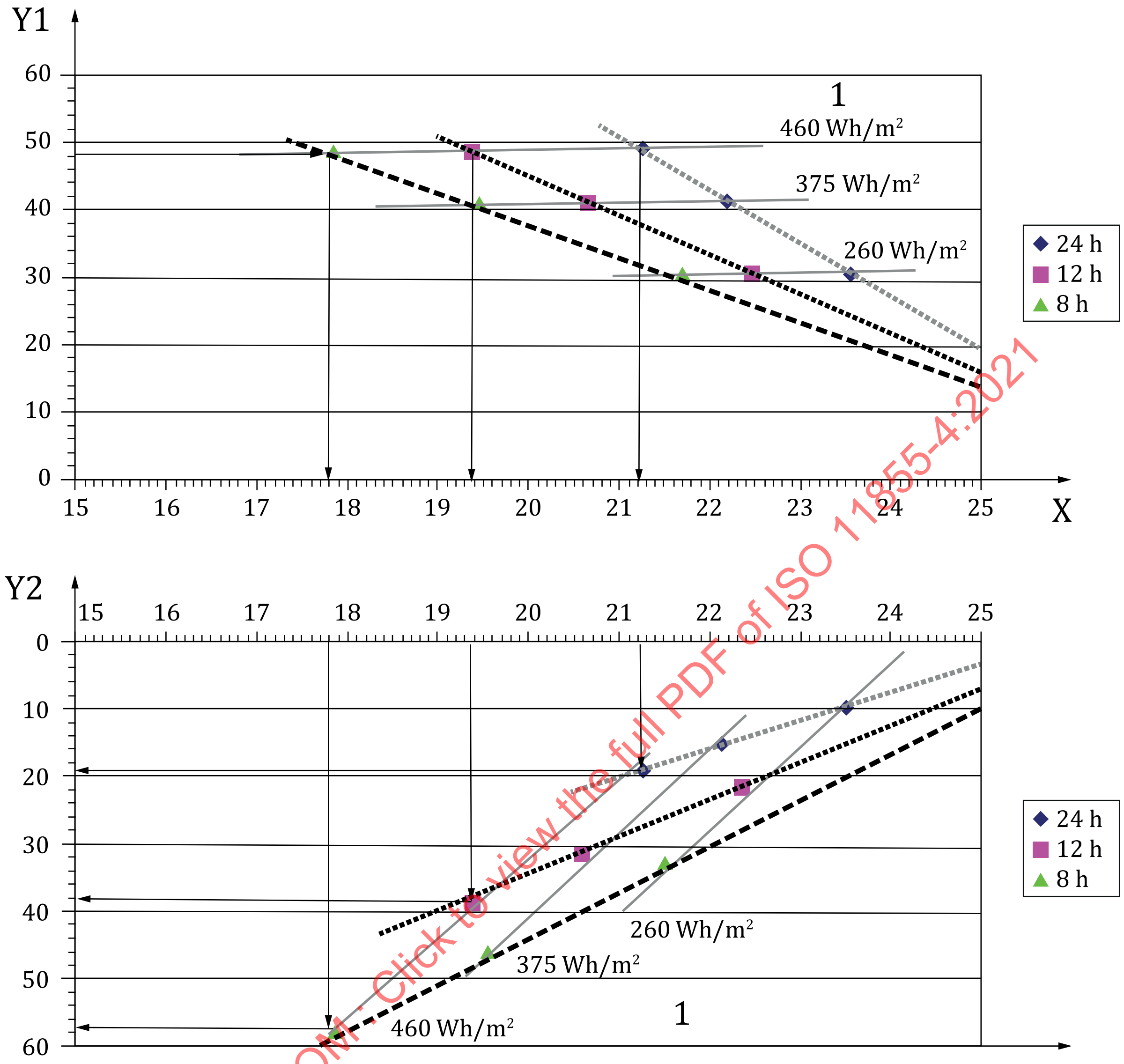


Key

1 concrete

2 reinforced concrete

Figure A.1 — Slab used in the simplified calculations



Key

X (upper diagram) supply water temperature, in °C

Y1 (upper diagram) maximum heat gain, in W/m²

Y2 (lower diagram) size of the chiller, in W/m²

1 overall heat gian

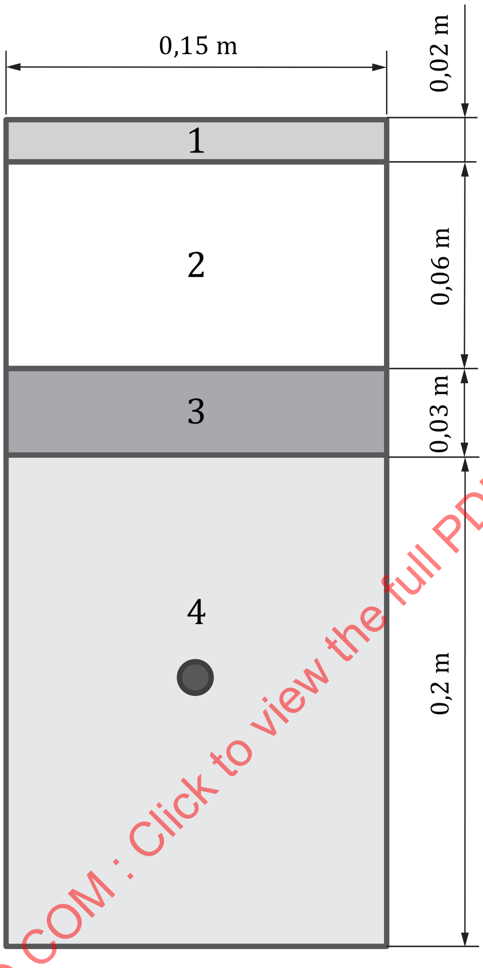
Figure A.2 — Simple diagrams showing the relation between heat gains in the room, lines for system running hours, supply water temperature θ_w, and energy removal on the water side

The diagrams in Figure A.4 correspond to a concrete slab shown in Figure A.3 with a solid concrete floor [conductivity 1,2 W/(m·K)], pipe spacing of 0,15 m and a permissible room temperature range of 21 °C to 26 °C.

The upper diagram in Figure A.4 shows on the Y-axis the maximum permissible total heat gain in space (internal gains plus solar gains) in W/m², and on the X-axis the required water supply temperature. The lines in the diagram correspond to different hours of operation (8 h, 12 h, and 24 h) and different daily energy gains in Wh/(m²·d).

The lower diagram in Figure A.4 shows the cooling power in W/m² required on the water side (to size the chiller) for TABS as a function of water supply temperature and operation time. Further, the amount of energy rejected per day is indicated in Wh/(m²·d).

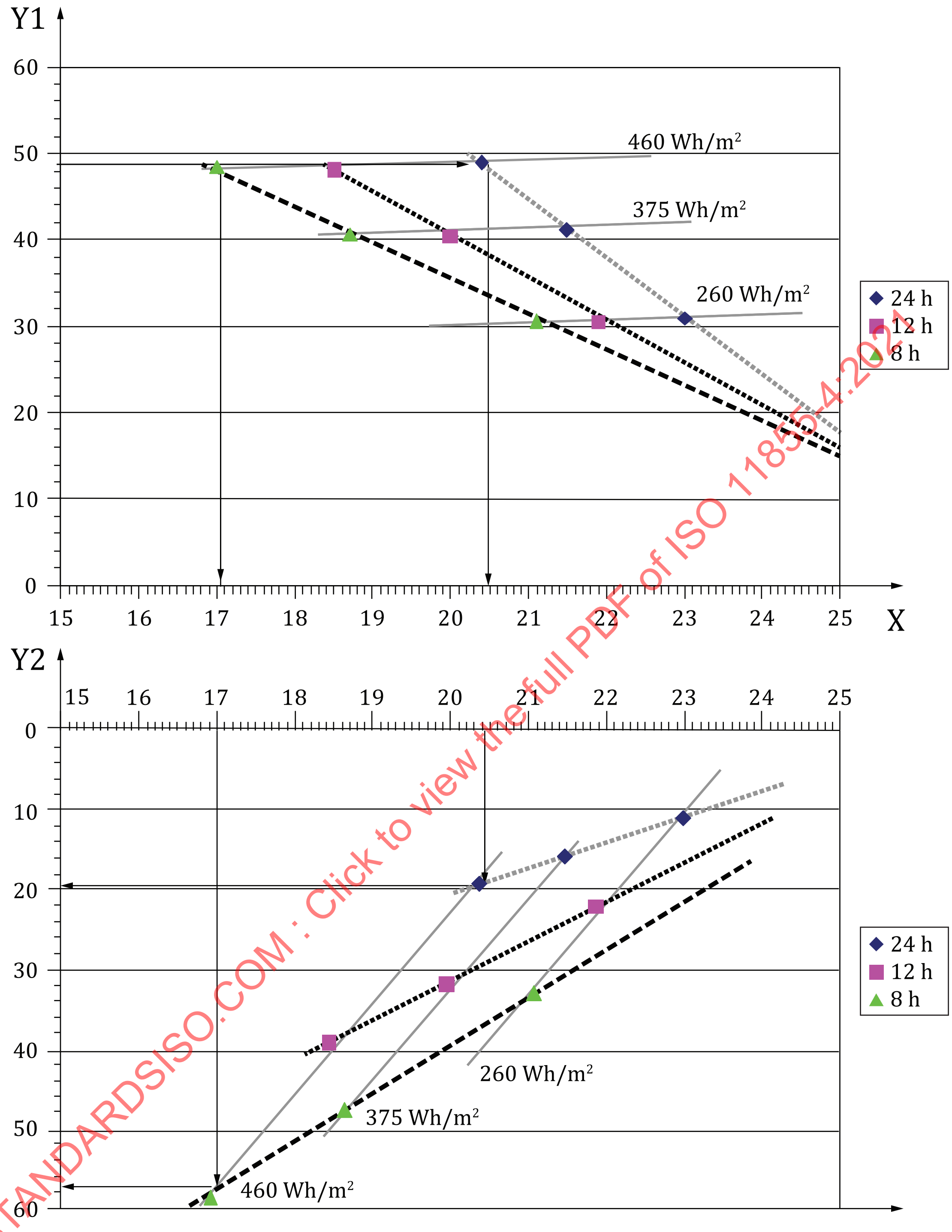
The example shows that by a maximum internal heat gain of 48 W/m^2 and 8 h operation, a supply water temperature of 17,0 °C is required. If, instead, the system is in operation for 24 h, a supply water temperature of 20,5 °C is required. In total, the amount of energy rejected from the room is approximately 460 Wh/m^2 per day. The required cooling power on the water side is for an 8 h operation 58 W/m^2 and for a 24 h operation only 20 W/m^2. Thus, for a 24 h operation, the chiller can be much smaller.



Key

1 wood
2 concrete
3 fibreglass
4 reinforced concrete

Figure A.3 — Slab used in the simplified calculations



Key

X (upper diagram) supply water temperature, °C

Y1 (upper diagram) Q_l, W/m²

Y2 (lower diagram) Q_w, W/m²

Figure A.4 — Simple diagrams showing the relation between heat gains in the room, lines for system running hours, supply water temperature θ_w, and energy removal on the water side

Diagrams like the ones in Figures A.2 and A.4 can be extended to different maximum operative temperatures allowed (but keeping the same room temperature drift). Such an extension can be performed by modifying the X-axis of both of the diagrams by the following formula:

$$\theta_{\mathrm{Wa,In}}^{\mathrm{Setp}} = \theta_{\mathrm{Wa,In,Ref}}^{\mathrm{Setp}} + \left(\theta_{\mathrm{Comf}}^{\mathrm{Max}} - \theta_{\mathrm{Comf,Ref}}^{\mathrm{Max}}\right) \tag{A.1}$$

where

$\theta_{\mathrm{Comf}}^{\mathrm{Max}}$ is the maximum allowable operative temperature in the case under consideration in °C;

$\theta_{\mathrm{Comf,Ref}}^{\mathrm{Max}}$ is the maximum allowable operative temperature in the reference case of the diagram in °C;

$\theta_{\mathrm{Wa,In}}^{\mathrm{Setp}}$ is the water inlet temperature in the case under consideration in °C;

$\theta_{\mathrm{Wa,In,Ref}}^{\mathrm{Setp}}$ is the water inlet temperature in the reference case of the diagram in °C.

Moreover, the second diagrams in Figures A.2 and A.4 can be substituted by the following formula, if no graphical support is needed:

$$Q_{\mathrm{w}} = \frac{E_{\mathrm{Day}}}{n_{\mathrm{h}}} \tag{A.2}$$

where

Q_{w} is the average specific cooling power of TABS [W/m²];

E_{Day} is the specific daily energy gains [Wh/m²];

n_{h} is the number of running hours [h].

As regards the heating capacity, dynamic calculations can also be carried out when looking at the possibility to use the system only for a part of the day. If steady state calculations are needed, this shall be done by using the resistance method introduced in ISO 11855-2.

Annex B
(normative)

Calculation method

B.1 Pipe level

R_t is the total thermal resistance [(m^2·K)/W] between the water supply temperature and the pipe level temperature, determined by the resistance method (for further details see ISO 11855-2). R_t can be calculated through Formula (B.1):

$$R_t = R_z + R_w + R_r + R_x \tag{B.1}$$

where

$$R_z = \frac{1}{2 \cdot \dot{m}_{H,sp} \cdot c_{Wa}}, \quad R_w = \frac{T^{0,13}}{8 \cdot \pi}\left(\frac{d_a - 2 \cdot s_R}{\dot{m}_{H,sp} \cdot L_R}\right)^{0,87}, \quad R_p = \frac{T \cdot \ln\left(\frac{d_a}{d_a - 2 \cdot s_R}\right)}{2 \cdot \pi \cdot \lambda_r}, \text{ and } R_x = \frac{T \cdot \ln\left(\frac{T}{\pi \cdot d_a}\right)}{2 \cdot \pi \cdot \lambda_b}$$

Two conditions shall be fulfilled for the application of these formulae:

— the formula for R_x is valid only if $s_1 / W > 0{,}3$; $s_2 / W > 0{,}3$; and $d_a/W < 0{,}2$;

— the formula for R_z is valid only if $\dot{m}_{H,sp} \cdot c_{Wa} \cdot \left(R_w + R_p + R_x\right) \geq \frac{1}{2}$.

If both conditions are fulfilled, Formula (B.1) can be applied.

B.2 Thermal nodes composing the slab and room

The slab is composed of $J = J_1 + J_2$ material layers, where J_1 is the number of layers constituting the upper part of the slab and J_2 is the number of layers constituting the lower one. As a consequence, J sets of physical properties (ρ_j, c_j, λ_j) shall be known. Besides, each layer has its own thickness, δ_j, thus, for geometrical consistency:

$$s_1 = \sum_{j=1}^{J_1} \delta_j \text{ and } s_2 = \sum_{j=J_1+1}^{J_1+J_2} \delta_j \tag{B.2}$$

For the calculations, each material layer is subdivided into a number of smaller divisions. For each material layer, the number of layers, m_j, into which it is divided for the calculations, shall be decided. Each division inherits the physical properties from the material layer which it belongs to. Each layer division constitutes a thermal node, that is a finite volume where a local heat balance is performed, in order to get temperatures and heat fluxs taking place within the slab and the room.

For a consistent description of the thermal behaviour of the slab and room, more thermal nodes shall be defined. All of the following thermal nodes shall be built up:

— thermal nodes representing the slab divisions (thermal nodes “I”, standing for “internal”);

— thermal node placed at the floor surface, over additional thermal resistances such as carpets, moquette, and raised floor (thermal node “F”, standing for “floor”);

— thermal node placed at the pipe level surface, connecting the slab with the water circuit (thermal node “PL”, standing for “pipe level”);

— thermal node placed at the ceiling surface, below additional thermal resistances such as suspended ceiling (thermal node “C”, standing for “ceiling”);

— thermal node placed at the surface of the internal walls, over the thermal resistance constituted by the internal covering, such as the plaster layer (thermal node “IWS”, standing for “internal wall surface”);

— thermal node placed inside the internal walls (thermal node “IW”, standing for “internal walls”);

— thermal node representing the air node (thermal node “A”, standing for “air”).

The thermal nodes should be enumerated according to the following the following rules:

— thermal node 1: thermal node “F”;

— thermal nodes 2 to $1+\sum_{1}^{J_1} m_j$: thermal nodes “I” representing the upper part of the slab;

— thermal node $2+\sum_{1}^{J_1} m_j$: thermal node “PL”;

— thermal nodes $3+\sum_{1}^{J_1} m_j$ to $2+\sum_{1}^{J_1+J_2} m_j$: thermal nodes “I” representing the lower part of the slab;

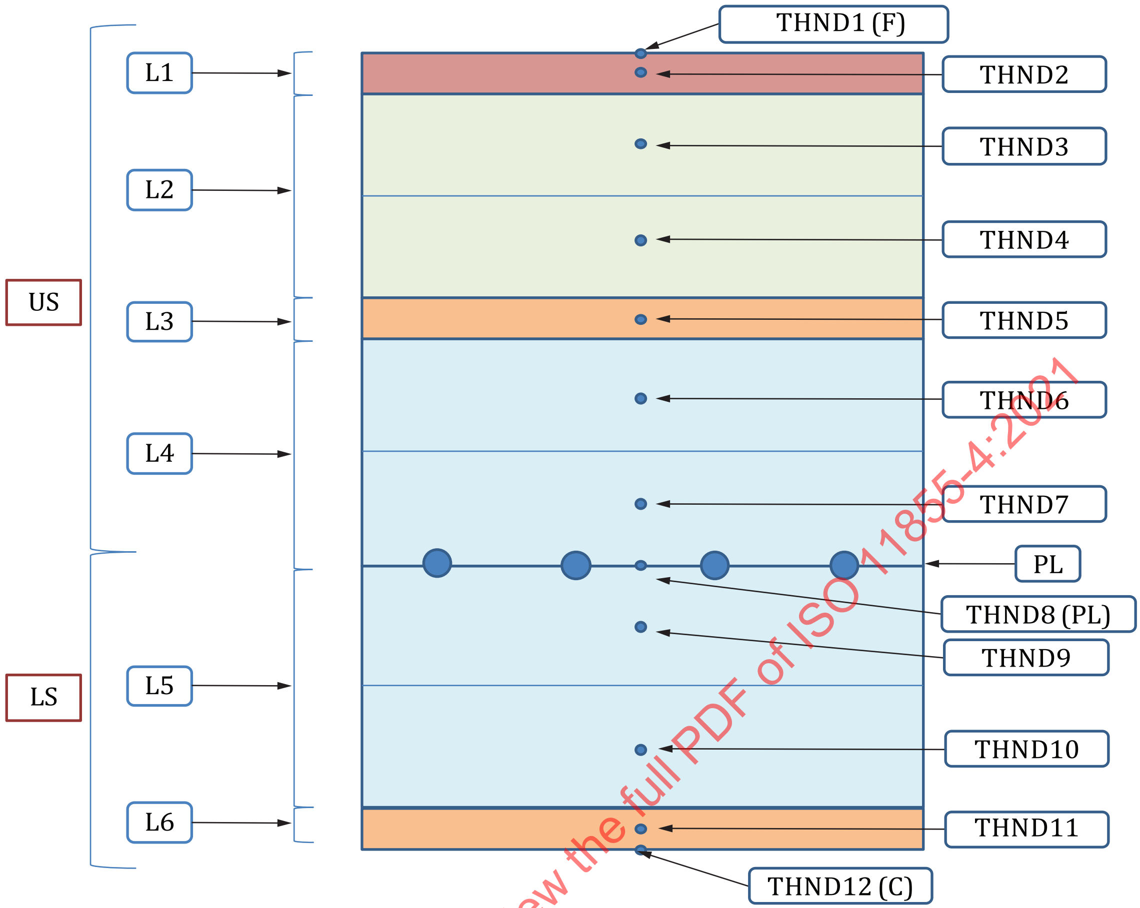
— thermal node $3+\sum_{1}^{J_1+J_2} m_j$: thermal node “C”;

— thermal node $4+\sum_{1}^{J_1+J_2} m_j$: thermal node “IWS”;

— thermal node $5+\sum_{1}^{J_1+J_2} m_{\mathrm{j}}$: thermal node “IW”;

— thermal node $6+\sum_{1}^{J_1+J_2} m_j$: thermal node “A”.

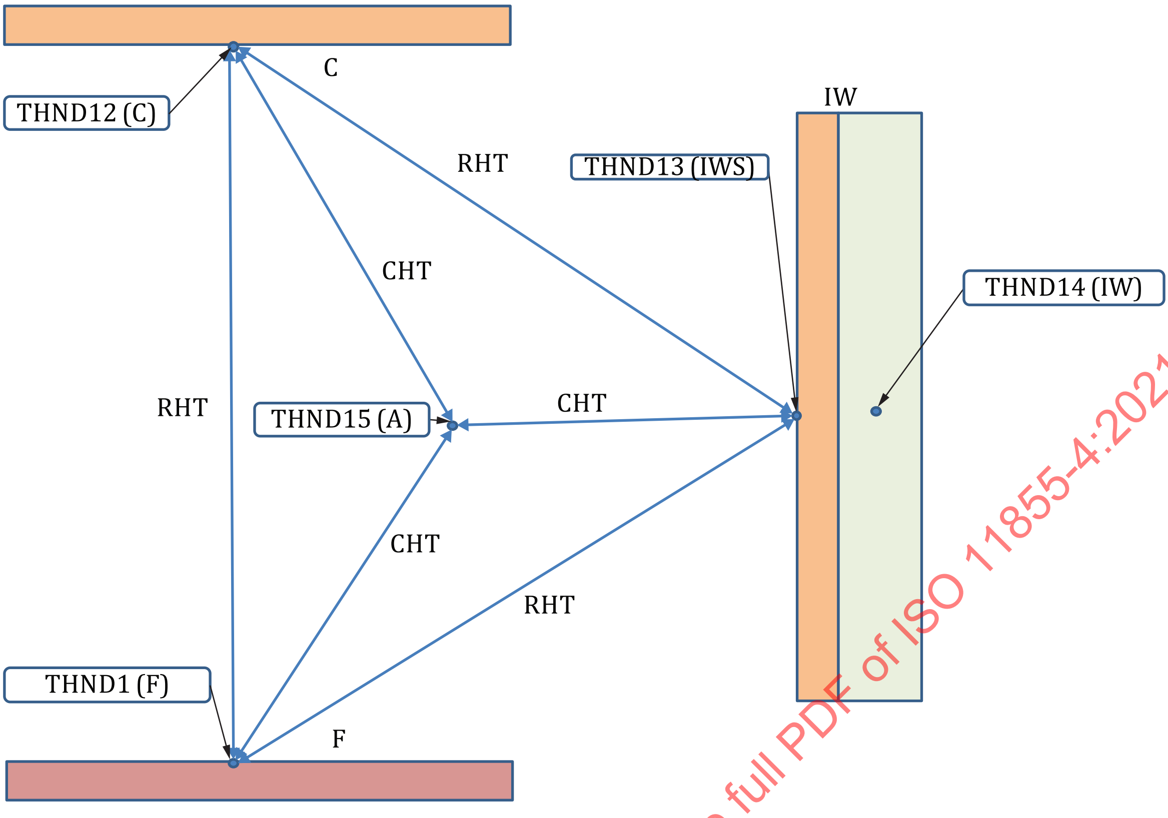
The Figures B.1 and B.2 summarize the thermal nodes mentioned above.



Key

C	ceiling
F	floor
L	layer
LS	lower part of the slab
PL	pipe level
THND	thermal node
US	upper part of the slab

Figure B.1 — Example of thermal nodes representing the slab



Key

A air
C ceiling
CHT convective heat transfer
F floor
IW internal walls
IWS internal wall surface
RHT radiant heat transfer
THND thermal node

Figure B.2 — Example of thermal nodes representing the room and correlated heat transfer connections

In order to perform heat transfer calculations, each thermal node is characterized by three main physical magnitudes:

— thermal inertia of the p-th thermal node:

for thermal nodes "I": $C_p = \rho_j \cdot c_j \cdot \frac{\delta_j}{m_j}$ (B.3)

for thermal nodes "IW": $C_p = C_W$ (B.4)

for thermal nodes "F", "PL", "C", "IWS", and "A": $C_p = 0$ (B.5)

— thermal resistance RU_p, which connects the p-th thermal node with the boundary of the previous thermal node:

for thermal nodes “I”: $RU_p = \dfrac{\delta_j}{2 \cdot m_j \cdot \lambda_j}$ (B.6)

for thermal nodes “C”: $RU_p = R_{\text{Add C}}$ (B.7)

for thermal nodes “IW”: $RU_p = R_{\text{Walls}}$ (B.8)

for thermal nodes “F”, “PL”, “IWS”, and “A”: $RU_p = 0$ (B.9)

— thermal resistance RL_p, which connects the p-th thermal node with the boundary of the next thermal node;

for thermal nodes “I”: $RL_p = \dfrac{\delta_j}{2 \cdot m_j \cdot \lambda_j}$ (B.20)

for thermal nodes “F”: $RL_p = R_{\text{Add F}}$ (B.11)

for thermal nodes “IWS”: $RL_p = R_{\text{Walls}}$ (B.12)

for thermal nodes “PL”, “C”, “IW”, and “A”: $RL_p = 0$ (B.13)

B.3 Calculations for the generic *h*-th hour

Further data required to perform the simulations are listed below:

— A_F : area of the heating/cooling surface [m²];

— A_W : total area of the internal vertical walls [m²];

— $F_{v\,F\text{-}C}$: view factor between the floor and the ceiling [-];

— $F_{v\,F\text{-}EW}$: view factor between the floor and the external walls [-];

— $h_{A\text{-}F}$: convective heat transfer coefficient between the air and the floor [W/(m²·K)];

— $h_{A\text{-}C}$: convective heat transfer coefficient between the air and the ceiling [W/(m²·K)];

— $h_{A\text{-}W}$: convective heat transfer coefficient between the air and the internal walls [W/(m²·K)];

— R_{AddC} : additional thermal resistance covering the lower side of the slab [(m²·K)/W];

— R_{AddF} : additional thermal resistance covering the upper side of the slab [(m²·K)/W];

— R_{Walls} : wall surface thermal resistance [(m²·K)/W]:

— C_W : average specific thermal capacity of the internal vertical walls [m²];

— R_t : circuit total thermal resistance [(m²·K)/W];

— $\dot{m}_{H,sp}$: specific design water flow in the circuit [kg/(s·m²)];

— c_{Wa} : specific heat of water [J/(kg·K)];

— f_{rm}^{h} : running mode for each h-th hour [-];

— $P_{\mathrm{Cct}}^{\mathrm{Max},h}$: maximum cooling power reserved to the circuit under examination for each h-th hour [W];

— $\theta_{\mathrm{Wa,In}}^{\mathrm{Setp},h}$: water inlet set-point temperature for each h-th hour in °C;

— Q_{IntConv}^{h} : internal convective heat gains for each h-th hour [W];

— Q_{IntRad}^{h} : internal radiant heat gains for each h-th hour [W];

— Q_{PrimAir}^{h} : primary air convective heat gains for each h-th hour [W];

— Q_{Sun}^{h} : solar heat gains in the room for each h-th hour [W];

— Q_{Transm}^{h} : transmission heat gains for each h-th hour [W];

— Δt : calculation time step [s]. For hourly time steps: $\Delta t = 3\,600$ s;

— n_{Max} : maximum number of iterations allowed [-];

— ξ_{Max} : maximum tolerance allowed [K].

The values of Q_{IntConv}^{h}, Q_{IntRad}^{h}, Q_{PrimAir}^{h}, Q_{Sun}^{h}, and Q_{Transm}^{h} shall be known for the whole day. Q_{IntConv}^{h}, Q_{IntRad}^{h} and Q_{PrimAir}^{h} depend on the people and the equipment in the room and on the possible air supply and infiltration, and can thus be estimated. Q_{Sun}^{h} and Q_{Transm}^{h} can be calculated by other software (through commercial software enabling calculation of the cooling loads of a room with a constant room temperature equal to the average room operative temperature during occupancy hours).

For every time step, the running strategy of the circuit f_{rm}^{h} shall be decided before the simulation is started, and the supply water temperature $\theta_{\mathrm{Wa,In}}^{\mathrm{Setp},h}$ is an input as well. These parameters are chosen by the designer and, by performing the simulation with different sets of parameters, it is possible to approach the best combination of running strategy of the circuit and supply water temperature.

The following shortcuts are useful in the following calculations:

$$F_{\mathrm{vF-W}} = 1 - F_{\mathrm{v\,F-EW}} - F_{\mathrm{v\,F-C}} \tag{B.14}$$

$$h_{\mathrm{F-W}} = 5{,}5 \cdot F_{\mathrm{v\,F-W}} \tag{B.15}$$

$$h_{\mathrm{F-C}} = 5{,}5 \cdot F_{\mathrm{v\,F-C}} \tag{B.16}$$

Formula (B.14) is expressed in -, Formulae (B.15) and (B.16) are expressed in W/(m²·K).

At each h-th hour, the following calculations shall be executed:

— calculation of the heat loads acting in the room:

$$Q_{\mathrm{Conv}}^{h} = 0{,}15 \cdot Q_{\mathrm{Transm}}^{h} + Q_{\mathrm{IntConv}}^{h} + Q_{\mathrm{PrimAir}}^{h} \tag{B.17}$$

$$Q_{\mathrm{Rad}}^{\mathrm{n}} = 0{,}85 \cdot Q_{\mathrm{Transm}}^{h} + Q_{\mathrm{IntRad}}^{h} + Q_{\mathrm{Sun}}^{h} \tag{B.18}$$

Formulae (B.17) and (B.18) are expressed in W.

From estimations performed on several detailed simulations, it was assumed that about 15 % of the heat gains passing through the external wall act on the room in a convective way, while the remaining 85 % can be considered a pure radiant heat load.

From this point on, iterations shall be performed, at each hour, in order to approach the solution of the thermal field in the room. The main points of the iterations are listed below.

— Determination of the water supply temperature. The inlet water temperature is calculated by taking into account the average heat flux extracted in the hour under consideration, according to the following formula (expressed in °C):

$$\theta_{\mathrm{Wa,In}}^{h} = \max\left(\theta_{\mathrm{Wa,in}}^{\mathrm{Setp},h}, \theta_{\mathrm{Wa,in}}^{h} + \frac{\left(Q_{\mathrm{Cct}}^{h} - P_{\mathrm{Cct}}^{\mathrm{Max},h}\right)}{\dot{m}_{\mathrm{H,sp}}^{h} \cdot A_{\mathrm{F}} \cdot c_{\mathrm{Wa}}}\right) \tag{B.19}$$

where

$\theta_{\mathrm{Wa,In}}^{\mathrm{Setp},h}$	is the water inlet set-point temperature in the h-th hour in °C;
$\theta_{\mathrm{Wa,In}}^{h}$	is the water inlet actual temperature in the h-th hour in °C;
Q_{Cct}^{h}	is the heat flux extracted by the circuit in the h-th hour in W;
$P_{\mathrm{Cct}}^{\mathrm{Max},h}$	is the maximum cooling power reserved to the circuit under consideration in the h-th hour [W];
$\dot{m}_{\mathrm{H,sp}}^{h}$	is the specific water mass flow in the circuit in the h-th hour in kg/(s·m^2);
A_{F}	is the area of the heating/cooling surface in m^2;
c_{Wa}	is the specific heat of the fluid flowing in the circuit in J/(kg·K).

— Calculation of the temperature of each thermal node, at the current iteration of the *h*-th hour.

Each *p*-th thermal node has peculiar characteristics that can be summarized in coefficients (“H”), to be used as shortcuts in the calculation of the energy balance for each thermal node:

H_{A}	is the coefficient summarizing the heat transfer coefficient between the *p*-th thermal node and thermal node “air” [W/K];
H_{IWS}	is the coefficient summarizing the heat transfer coefficient between the *p*-th thermal node and thermal node “internal wall surface” [W/K];
H_{F}	is the coefficient summarizing the heat transfer coefficient between the *p*-th thermal node and thermal node “floor” [W/K];
H_{C}	is the coefficient summarizing the heat transfer coefficient between the *p*-th thermal node and thermal node “ceiling” [W/K];
H_{Rad}	is the fraction of room radiant heat loads impinging on the *p*-th thermal node in -;
H_{Conv}	is the fraction of room convective heat loads acting on the *p*-th thermal node in -;
H_{CondUp}	is the coefficient summarizing the thermal conduction connection between the *p*-th thermal node and the previous thermal node [i.e. the (p-1)-th thermal node] in W/K;

H_{CondDn} is the coefficient summarizing the thermal conduction connection between the p-th thermal node and the next thermal node [i.e. the (p+1)-th thermal node] in W/K;

H_I is the coefficient summarizing the inertia contribution at the p-th thermal node in W/K;

H_{Cct} is the coefficient summarizing the heat transfer coefficient between the p-th thermal node and the water inlet temperature in W/K.

The coefficients seen above are specified for each thermal node, in Table B.1.

Table B.1 — Coefficients for the calculation of the temperature at each thermal node

	p-th node = node "F"	*p*-th node = node "I"	*p*-th node = node "PL"	*p*-th node = node "C"	*p*-th node = node "IWS"	*p*-th node = node "IW"	*p*-th node = node "A"
H_A	$h_{A-F} \cdot A_F$	0.	0.	$h_{A-C} \cdot A_F$	$h_{A-W} \cdot A_W$	0.	0.
H_{IWS}	$h_{F\text{-}W} \cdot A_F$	0.	0.	$h_{F\text{-}W} \cdot A_F$	0.	0.	$h_{A-W} \cdot A_W$
H_F	0.	0.	0.	$h_{F\text{-}C} \cdot A_F$	$h_{F\text{-}W} \cdot A_F$	0.	$h_{A-F} \cdot A_F$
H_C	$h_{F\text{-}C} \cdot A_F$	0.	0.	0.	$h_{F\text{-}W} \cdot A_F$	0.	$h_{A-C} \cdot A_F$
H_{Rad}	$\frac{A_F}{2 \cdot A_F + A_W}$	0.	0.	$\frac{A_F}{2 \cdot A_F + A_W}$	$\frac{A_W}{2 \cdot A_F + A_W}$	0.	0.
H_{Conv}	0.	0.	0.	0.	0.	0.	1.
H_{CondUp}	0.	$\frac{1}{RU_p + RD_{p-1}} \cdot A_F$	$\frac{1}{RU_p + RD_{p-1}} \cdot A_F$	$\frac{1}{RU_p + RD_{p-1}} \cdot A_F$	0.	$\frac{1}{RU_p + RD_{p-1}} \cdot A_W$	0.
H_{CondDn}	$\frac{1}{RD_p + RU_{p+1}} \cdot A_F$	$\frac{1}{RD_p + RU_{p+1}} \cdot A_F$	$\frac{1}{RD_p + RU_{p+1}} \cdot A_F$	0.	$\frac{1}{RD_p + RU_{p+1}} \cdot A_W$	0.	0.
H_I	0.	$\frac{C_p \cdot A_F}{\Delta t}$	0.	0.	0.	$\frac{C_p \cdot A_W}{\Delta t}$	0.
H_{Cct}	0.	0.	$\frac{1}{R_t} \cdot A_F$	0.	0.	0.	0.

By means of the coefficients seen above it is possible to calculate the temperature of each node at the end of the time step under consideration. At each iteration, the temperature of each thermal node, at the end of the time step under consideration, is calculated via the following formula (expressed in °C) :

$$\theta_p^h = \frac{\begin{pmatrix} H_{\text{Air}} \cdot \theta_{\text{A}}^h + H_{\text{IWS}} \cdot \theta_{\text{IWS}}^h + H_{\text{F}} \cdot \theta_{\text{F}}^h + H_{\text{C}} \cdot \theta_{\text{C}}^h + H_{\text{Rad}} \cdot Q_{\text{Rad}}^h + H_{\text{Conv}} \cdot Q_{\text{Conv}}^h + \\ + H_{\text{CondUp}} \cdot \theta_{p-1}^h + H_{\text{CondDn}} \cdot \theta_{p+1}^h + H_{\text{I}} \cdot \theta_p^{h-1} + H_{\text{Cct}} \cdot \theta_{\text{Wa,In}}^h \cdot f_{\text{rm}}^h \end{pmatrix}}{H_{\text{A}} + H_{\text{IWS}} + H_{\text{F}} + H_{\text{C}} + H_{\text{CondUp}} + H_{\text{CondDn}} + H_{\text{I}} + H_{\text{Cct}} \cdot f_{\text{rm}}^h} \quad \text{(B.20)}$$

The achieved temperatures θ_p^h are stored and compared with the ones calculated at the previous iteration ($\theta_p^{h'}$), in the following way:

— calculation of the actual tolerance at the current iteration: $\xi = \sum_p \left(\theta_p^h - \theta_p^{h'}\right)$ in K;

— comparison of the actual tolerance with the maximum tolerance allowed: $\xi < \xi_{\text{Max}}$.

If $\xi > \xi_{\text{Max}}$ and $n < n_{\text{Max}}$, then the required accuracy has not been reached and another iteration shall be executed.

If one more iteration more shall be executed, then Q_{Cct}^h is calculated via the following formula (expressed in W):

$$Q_{\text{Cct}}^h = \frac{\left(\theta_{\text{PL}}^h - \theta_{\text{Wa,In}}^h\right)}{R_{\text{t}}} \cdot f_{\text{rm}}^h \cdot A_{\text{F}} \quad \text{(B.21)}$$

Otherwise, the following quantities can be calculated and stored:

$$Q_{\text{F}}^h = h_{\text{A-F}} \cdot A_{\text{F}} \cdot \left(\theta_{\text{A}}^h - \theta_{\text{F}}^h\right) + h_{\text{F-C}} \cdot A_{\text{F}} \cdot \left(\theta_{\text{C}}^h - \theta_{\text{F}}^h\right) + h_{\text{F-W}} \cdot A_{\text{F}} \cdot \left(\theta_{\text{IWS}}^h - \theta_{\text{F}}^h\right) + \frac{A_{\text{F}}}{2 \cdot A_{\text{F}} + A_{\text{W}}} \cdot Q_{\text{Rad}}^h \quad \text{(B.22)}$$

$$Q_{\text{C}}^h = h_{\text{A-C}} \cdot A_{\text{F}} \cdot \left(\theta_{\text{A}}^h - \theta_{\text{C}}^h\right) + h_{\text{F-C}} \cdot A_{\text{F}} \cdot \left(\theta_{\text{F}}^h - \theta_{\text{C}}^h\right) + h_{\text{F-W}} \cdot A_{\text{F}} \cdot \left(\theta_{\text{IWS}}^h - \theta_{\text{C}}^h\right) + \frac{A_{\text{F}}}{2 \cdot A_{\text{F}} + A_{\text{W}}} \cdot Q_{\text{Rad}}^h \quad \text{(B.23)}$$

$$Q_{\text{IWS}}^h = h_{\text{A-W}} \cdot A_{\text{W}} \cdot \left(\theta_{\text{A}}^h - \theta_{\text{IWS}}^h\right) + h_{\text{F-W}} \cdot A_{\text{F}} \cdot \left(\theta_{\text{F}}^h - \theta_{\text{IWS}}^h\right) + h_{\text{F-W}} \cdot A_{\text{F}} \cdot \left(\theta_{\text{C}}^h - \theta_{\text{IWS}}^h\right) + \frac{A_{\text{W}}}{2 \cdot A_{\text{F}} + A_{\text{W}}} \cdot Q_{\text{Rad}}^h \quad \text{(B.24)}$$

$$\theta_{\text{MR}}^h = \frac{A_{\text{F}} \cdot \theta_{\text{F}}^h + A_{\text{F}} \cdot \theta_{\text{C}}^h + A_{\text{W}} \cdot \theta_{\text{IWS}}^h}{2 \cdot A_{\text{F}} + A_{\text{W}}} \quad \text{(B.25)}$$

$$\theta_{\text{Op}}^h = \frac{\theta_{\text{A}}^h + \theta_{\text{MR}}^h}{2} \quad \text{(B.26)}$$

$$\theta_{\text{Wa,Out}}^h = \theta_{\text{Wa,In}}^h + \frac{Q_{\text{Cct}}^h}{\dot{m}_{\text{H,sp}} \cdot A_{\text{F}} \cdot c_{\text{Wa}}} \quad \text{(B.27)}$$

Formulae (B.22) to (B.24) are expressed in W, Formulae (B.25) to (B.27) are expressed in °C.

B.4 Sizing of the system

The allowed range for the operative temperature of the room is usually 20 °C to 26 °C. If the room operative temperature is always in this range (or in any range of comfort temperatures chosen by the planner and agreeing with local or international standards), then the system is well sized. Otherwise the running strategy, the supply water temperature or the circuit characteristics have to be changed.

Annex C
(informative)

Tutorial guide for assessing the model

The values in Table C.1 will be used.

Table C.1 — Input data for slab, room, circuit, boundary conditions and results

Main slab input data	J_1	3	Input
	J_2	1	Input
	δ_1	0,02 m	Input
	m_1	2	Input
	λ_1	0,17 W/(m·K)	Input
	ρ_1	700 kg/m^3	Input
	c_1	2 300 J/(kg·K)	Input
	δ_2	0,07 m	Input
	m_2	3	Input
	λ_2	1,1 W/(m·K)	Input
	ρ_2	1 900 kg/m^3	Input
	c_2	850 J/(kg·K)	Input
	δ_3	0,1 m	Input
	m_3	4	Input
	λ_3	1,9 W/(m·K)	Input
	ρ_3	2 000 kg/m^3	Input
	c_3	880 J/(kg·K)	Input
	δ_4	0,1 m	Input
	m_4	4	Input
	λ_4	1,9 W/(m·K)	Input
	ρ_4	2 000 kg/m^3	Input
	c_4	880 J/(kg·K)	Input
Δt		3 600 s	Input
Main room input data	A_F	30 m^2	Input
	A_W	48 m^2	Input
	$h_{A\text{-}F}$	1,5 W/(m^2·K)	Input
	$h_{A\text{-}C}$	5,5 W/(m^2·K)	Input
	$h_{A\text{-}W}$	2,5 W/(m^2·K)	Input
	$F_{v\ F\text{-}EW}$	0,21	Input
	$F_{v\ F\text{-}C}$	0,35	Input
	R_{addF}	0,1 (m^2·K)/W	Input
	R_{addC}	0 (m^2·K)/W	Input
	R_{Walls}	0,05 (m^2·K)/W	Input
	C_W	25 600 J/(m^2·K)	Input

Table C.1 *(continued)*

Main circuit input data	R_t	0,073 (m²·K)/W	Input
	c_{Wa}	4 187 J/(kg·K)	Input
	$\dot{m}_{H,sp}$	0,01 kg/(m²·s)	Input
Hourly boundary conditions			
From 00:00 to 8:00	Q_{Conv}^{h}	30 W	Input
	Q_{Rad}^{h}	10 W	Input
	f_{rm}^{h}	1	Input
	$\theta_{Wa,In}^{Setp,h}$	20,0 °C	Input
	$P_{Cct}^{Max,h}$	1 000 W	Input
From 8:00 to 19:00	Q_{Conv}^{h}	400 W	Input
	Q_{Rad}^{h}	300 W	Input
	f_{rm}^{h}	0	Input
	$\theta_{Wa,In}^{Setp,h}$	20,0 °C	Input
	$P_{Cct}^{Max,h}$	0 W	Input
From 19:00 to 24:00	Q_{Conv}^{h}	150 W	Input
	Q_{Rad}^{h}	100 W	Input
	f_{rm}^{h}	1	Input
	$\theta_{Wa,In}^{Setp,h}$	20,0 °C	Input
	$P_{Cct}^{Max,h}$	0 W	Input

Main results

Time	θ_F^h [°C]	θ_C^h [°C]	θ_A^h [°C]	Q_F^h [W]	Q_C^h [W]	Q_{IWS}^h [W]	Q_{ICct}^h [W]
1	22,7	22,3	22,7	-1	109	-68	770
2	22,5	22,1	22,4	-8	94	-46	716
3	22,3	22,0	22,2	-11	85	-34	666
4	22,2	21,8	22,1	-12	79	-27	620
5	22,0	21,7	21,9	-12	75	-23	577
6	21,9	21,6	21,8	-12	71	-20	537
7	21,8	21,5	21,7	-11	68	-18	501
8	21,6	21,4	21,6	-10	66	-16	467
9	22,5	21,8	23,4	124	442	134	0
10	22,7	22,1	23,8	147	472	81	0
11	22,9	22,3	24,1	161	485	54	0
12	23,1	22,5	24,3	170	490	39	0
13	23,3	22,7	24,5	176	492	32	0

Table C.1 *(continued)*

14	23,5	22,9	24,7	179	493	28	0
15	23,6	23,1	24,9	181	493	26	0
16	23,8	23,3	25,0	182	493	25	0
17	24,0	23,4	25,2	183	493	24	0
18	24,1	23,6	25,4	184	493	24	0
19	24,3	23,8	25,5	184	492	23	0
20	23,8	23,5	24,3	89	246	-85	871
21	23,6	23,2	24,0	69	236	-55	915
22	23,4	23,0	23,8	55	234	-40	926
23	23,3	22,8	23,6	47	234	-32	878
24	23,1	22,6	23,4	42	234	-26	826