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Thermal performance of buildings - Calculation of internal temperatures of a room in summer without mechanical cooling - General criteria and validation procedures (ISO 13791:2004)

Performance thermique des bâtiments - Calcul des températures intérieures en été d'un local sans dispositif de refroidissement - Critères généraux et méthodes de calcul (ISO 13791:2004)

Wärmetechnisches Verhalten von Gebäuden - Sommerliche Raumtemperaturen bei Gebäuden ohne Anlagentechnik - Allgemeine Kriterien und Validierungsverfahren (ISO 13791:2004)

This European Standard was approved by CEN on 7 June 2004.

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EUROPEAN COMMITTEE FOR STANDARDIZATION
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Contents

Foreword

This document (EN ISO 13791:2004) has been prepared by Technical Committee CEN/TC 89 "Thermal performance of buildings and building components", the secretariat of which is held by SIS, in collaboration with Technical Committee ISO/TC 163, “Thermal performance and energy use in the built environment”, Subcommittee SC 2, "Calculation methods".

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of an identical text or by endorsement, at the latest by March 2005, and conflicting national standards shall be withdrawn at the latest by March 2005.

This standard is one of a series of standards on calculation methods for the design and the evaluation of the thermal performance of buildings and building components.

This document includes a Bibliography.

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

Introduction

This document is intended for use by specialists to develop and/or validate methods for the hourly calculation of the internal temperatures of a single room.

Examples of application of such methods include:

a) assessing the risk of internal overheating;

b) optimizing aspects of building design (building thermal mass, solar protection, ventilation rate, etc.) to provide thermal comfort conditions;

c) assessing whether a building requires mechanical cooling.

Criteria for building performance are not included. They can be considered at national level. This standard can also be used as a reference to develop more simplified methods for the above and similar applications.

1 Scope

This document specifies the assumptions, boundary conditions, equations and validation tests for a calculation procedure, under transient hourly conditions, of the internal temperatures (air and operative) during the warm period, of a single room without any cooling/heating equipment in operation. No specific numerical techniques are imposed by this document. Validation tests are included in Clause 7. An example of a solution technique is given in Annex A.

This document does not contain sufficient information for defining a procedure able to determine the internal conditions of special zones such as attached sun spaces, atria, indirect passive solar components (Trombe walls, solar panels) and zones in which the solar radiation may pass through the room. For such situations different assumptions and more detailed solution models are needed (see Bibliography).

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 410, *Glass in building – Determination of luminous and solar characteristics of glazing.*

EN ISO 6946, *Building components and building elements – Thermal resistance and thermal transmittance – Calculation method (ISO 6946:1996).*

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

EN ISO 9251:1995, *Thermal insulation – Heat transfer conditions and properties of materials – Vocabulary (ISO 9251:1987).*

EN ISO 9288:1996, *Thermal insulation – Heat transfer by radiation – Physical quantities and definitions (ISO 9288:1989).*

EN ISO 9346:1996, *Thermal insulation – Mass transfer – Physical quantities and definitions (ISO 9346:1987).*

EN ISO 10077-1, *Thermal performance of windows, doors and shutters – Calculation of thermal transmittance – Part 1: Simplified method (ISO 10077-1:2000).*

EN ISO 10077-2, *Thermal performance of windows, doors and shutters – Calculation of thermal transmittance – Part 2: Numerical method for frames (ISO 10077-2:2003).*

EN ISO 13370, *Thermal performance of buildings – Heat transfer via the ground – Calculation methods (ISO 13370:1998).*

3 Terms, definitions, symbols and units

3.1 Terms and definitions

For the purposes of this document, the terms and definitions given in EN ISO 73451995, EN ISO 9251:1995, EN ISO 9288:1996 and EN ISO 9346:1996 and the following apply.

3.1.1
internal environment
closed space delimited from the external environment or adjacent spaces by the building fabric

3.1.2
room element
wall, roof, ceiling, floor, door or window that separates the internal environment from the external environment or an adjacent space

3.1.3
room air
air of the internal environment

3.1.4
internal air temperature
temperature of the room air

3.1.5
internal surface temperature
temperature of the internal surface of a building element

3.1.6
mean radiant temperature
uniform surface temperature of an enclosure with which an occupant would exchange the same amount of radiant heat as with the actual non-uniform enclosure

3.1.7
operative temperature
uniform temperature of an enclosure with which an occupant would exchange the same amount of heat by radiation plus convection as with the actual non-uniform environment

3.2 Symbols and units

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

Symbol	Quantity	Unit
A	area	m^2
A_s	sunlit area	m^2
C	heat capacity	J/K
F	view factor	-
I	intensity of solar radiation	W/ m^2
R	thermal resistance	m^2·K/W
T	thermodynamic temperature	K
U	thermal transmittance	W/(m^2·K)
V	volume	m^3
a	thermal diffusivity	m^2/s
c	specific heat capacity	J/(kg·K)
c_d	coefficient of discharge	-
c_p	specific heat capacity of air at constant pressure	J/(kg·K)
d	thickness	m
f_d	solar distribution factor	-
f_{ic}	internal convective factor	-
f_s	sunlit factor	-
f_{sa}	solar to air factor	-
f_{sl}	solar loss factor	-
g	heat flow rate per volume	W/m^3
h	surface coefficient of heat transfer	W/(m^2·K)
l	length	m
m	mass	kg
q_a	mass air flow rate	kg/s
p	pressure	Pa
q	density of heat flow rate	W/m^2
t	time	s
v	velocity	m/s
x,y,z	co-ordinates	m
Λ	thermal conductance of air layer	W/(m^2·K)
Φ	heat flow rate	W
J	total radiosity	W/m^2
α	solar absorptance	-
ε	total hemispherical emissivity	-
θ	Celsius temperature	°C
λ	thermal conductivity	W/(m·K)
μ	dynamic viscosity	kg/(m·s)
ρ	solar reflectance	-

3.3 Subscripts

a	air	cd	conduction
b	building	ec	external ceiling
c	convection	ef	external floor
D	direct solar radiation	eq	equivalent
d	diffuse solar radiation	ic	internal ceiling
e	external	if	internal floor
g	ground	il	inlet section
i	internal	lr	long-wave radiation
l	leaving the section	mr	mean radiant
n	normal to surface	op	operative
r	radiation	sa	solar to air
s	surface	sk	sky
t	time	sr	short wave radiation
v	ventilation	va	ventilation through air cavity

4 Determination of internal temperatures

4.1 Assumptions

The evaluation of the internal temperature of a room involves the solution of a system of equations of the transient heat and mass transfers between the external and internal environment through the opaque and transparent elements bounding the room envelope. The procedures given in this document allow the user to determine the time dependent temperature of each component, including the internal air. Accepted assumptions for the calculation of the internal temperatures of a single room under transient conditions in absence of any cooling plant are:

- the air temperature is uniform throughout the room;
- the various surfaces of the room elements are isothermal;
- the thermophysical properties of the materials composing the room elements are time independent;
- the heat conduction through the room elements (excluding to the ground) is assumed to be one-dimensional;
- the heat conduction to the ground through room elements is treated by an equivalent one-dimensional heat flow rate according to EN ISO 13370;
- the effect of thermal bridges are generally neglected, but if they are considered their heat storage contribution is neglected;
- air spaces are treated as air layers bounded by two isothermal and parallel surfaces;
- convective heat transfer coefficients: at the external surface they depend on the wind velocity and direction, at the internal surface they depend on the direction of the heat flow;
- the long-wave radiative heat flow rate at the external surfaces of the room elements is related to a time-independent heat transfer coefficient;
- the external radiant environment (sky excluded) is at the external air temperature (see 4.5.4.1);
- the distribution of solar radiation within the room is time-independent;
- the dimensions of each element are measured inside the room;
- the mean radiant temperature is calculated by weighting the various internal surface temperatures according to the relevant areas;

— the operative temperature is the average between the internal air temperature and the mean surface temperature.

4.2 Evaluation of the relevant temperatures

4.2.1 Internal air temperature

The air temperature of a room, at any given time, is obtained by solving Equation (1), where heat flow rates to room air are taken as positive:

$$\sum_{j=1}^{N}(Aq_{c,i})_j + \Phi_v + \Phi_{i,c} + \Phi_{sa} + \Phi_{va} = c_a\, m_{a,i}\frac{\partial \theta_{a,i}}{\partial t} \tag{1}$$

where

- N is the number of internal surfaces delimiting the internal air;
- A is the area of each building element;
- $q_{c,i}$ is the density of the heat flow rate by convection (see 4.5.2.2);
- Φ_v is the heat flow rate by ventilation (see 4.5.6);
- $\Phi_{i,c}$ is the convective part of heat flow rate due to internal sources (see 4.5.5);
- Φ_{sa} is the solar to air heat flow rate (see 4.5.3.4);
- Φ_{va} is the heat flow rate due to the air entering the room through air layers within the elements bounding the room;
- c_a is the specific heat capacity of air;
- $m_{a,i}$ is the mass of the internal air;
- $\theta_{a,i}$ is the temperature of the internal air;
- t is the time

NOTE Because of the very small value of the term ($c_a\, m_{a,i}$) the right-hand side of Equation (1) can be assumed to be zero.

4.2.2 Internal surface temperature

The internal surface temperature at element j is obtained by solving Equation (2), where heat flow rates to the internal surface, except $q_{c,j}$, are taken as positive:

$$q_{lr,j} + q_{sr,j} + q_{c,j} + q_{cd,j} + \Phi_{i,r} / (\sum_{j=1}^{N} A_j) = 0 \tag{2}$$

where

- q_{lr} is the density of heat flow rate due to long-wave radiation exchanged with other internal surfaces (see 4.5.4.2);
- q_{sr} is the density of heat flow rate due to the absorbed short-wave radiation (see 4.5.3.2);
- q_c is the density of heat flow rate released to room air by convection (see 4.5.2.2);

q_{cd} is the density of heat flow rate by conduction (see 4.5.1);

$\Phi_{I,r}$ is the heat flow rate due to the radiative component of internal gains (see 4.5.5);

N is the number of surfaces delimiting the internal air;

A_j is the area of room element *j*.

4.2.3 Surface delimiting two solid layers

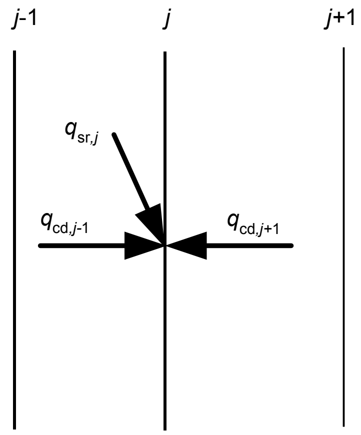


Figure 1 - Surface delimiting two layers

The temperature at surface *j* delimiting two layers in an element (Figure 1) is obtained by solving Equation (3):

$$q_{cd,j-1} + q_{cd,j+1} + q_{sr,j} = 0 \tag{3}$$

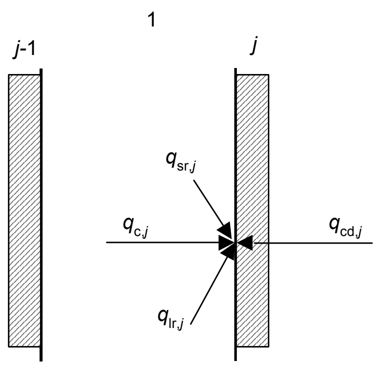
where

$q_{cd,j-1}$ is the density of heat flow rate by conduction from the *j*-1 surface (see 4.5.1);

$q_{cd,j+1}$ is the density of heat flow rate by conduction from the *j*+1 surface (see 4.5.1);

$q_{sr,j}$ is the density of heat flow rate due to the solar radiation absorbed by the surface *j*.

4.2.4 Surface of an air layer



Key
1 Air layer

Figure 2 - Surface delimiting an air layer

The temperature at surface *j* of an air layer (Figure 2) is obtained by solving Equation (4):

$$q_{c,j} + q_{lr,j} + q_{cd,j} + q_{sr,j} = 0 \qquad (4)$$

where

q_c is the density of the total heat flow rate released to the air layer (see 4.5.2);

q_{lr} is the density of the heat flow rate received by long-wave radiation across the air layer (see 4.5.4);

q_{cd} is the density of the heat flow by conduction (see 4.5.1);

q_{sr} is the density of heat flow rate absorbed due to an external source (e.g. solar radiation).

4.2.5 External surface of a room element

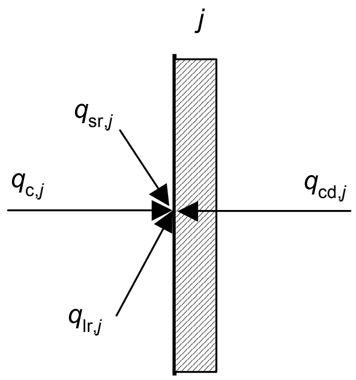


Figure 3 - External surface of an element

The temperature at surface *j* of a room element (Figure 3) is obtained by solving Equation (5):

$$q_{lr,j} + q_{sr,j} + q_{c,j} + q_{cd,j} = 0 \qquad (5)$$

where

q_{lr} is the density of heat flow rate by long-wave radiation at the surface (see 4.5.4.1);

q_{sr} is the density of heat flow rate due to the short-wave radiation absorbed by the surface (see 4.5.3.1);

q_c is the density of heat flow rate by convection with the air (see 4.5.2.2);

q_{cd} is the density of the conduction heat flow rate (see 4.5.1).

4.2.6 Relevant temperatures for special construction elements

4.2.6.1 Ceiling below an attic

The ceiling, the air space and the roof are considered as a single horizontal element with one-dimensional heat flow. The air space is considered as an air layer, treated in 4.5.2.3 and 4.5.2.4.

4.2.6.2 Floor on ground

The floor and the soil are considered as a single horizontal element with the heat flow treated according to EN ISO 13370. Boundary conditions are specified in 4.4.3.

4.2.6.3 Floor over cellar

The cellar is treated as an adjacent room according to EN ISO 13370. Boundary conditions are specified in 4.4.3.

4.2.6.4 Floor over crawl space

The floor, the crawl space and the soil are considered as a single horizontal element with the heat flow treated according to EN ISO 13370. The crawl space is treated as a ventilated air layer. Boundary conditions are specified in 4.4.5.

4.2.6.5 Glazed element

A glazed element is composed of a number of planes (glazing panes and possibly blinds) which are in thermal equilibrium with one another. The evaluation of temperatures of each plane is made using the following assumptions:

- the heat storage effects in the various planes are neglected;
- the heat flow by convection through the air layers between each pane is calculated according to 4.5.2.3 and 4.5.2.4;
- the density of heat flow rate due to the long-wave radiation between the various planes is calculated according to 4.5.4.3;
- the density of heat flow rate due to the short-wave radiation absorbed by each plane is treated as a source term.

4.3 Room thermal balance

In each equation of 4.2, the time dependent heat flow rates shall be expressed in terms of operators which relate the heat flow rate at the internal surface of each element to the temperature at the internal and external surface, and that of the internal air, by using suitable mathematical models of the heat transfer processes. The temperature of the internal air, together with the temperature of the different surfaces, shall be determined by solving the global equation system at each time step considered. A general expression of the equation system is:

$$\begin{pmatrix} \Pi_{1,1} & \Pi_{1,2} & \Pi_{1,N} & \Pi_{1,N+1} \\ \Pi_{2,1} & \Pi_{2,2} & \Pi_{2,N} & \Pi_{2,N+1} \\ \Pi_{N,1} & \Pi_{N,2} & \Pi_{N,N} & \Pi_{N,N+1} \\ \Pi_{N+1,1} & \Pi_{N+1,2} & \Pi_{N+1,N} & \Pi_{N+1,N+,1} \end{pmatrix} \cdot \begin{pmatrix} \theta_{\mathrm{is},1} \\ \theta_{\mathrm{is},2} \\ \theta_{\mathrm{is},N} \\ \theta_{\mathrm{a}} \end{pmatrix} = \begin{pmatrix} \Gamma_1 \\ \Gamma_2 \\ \Gamma_N \\ \Gamma_{N+1} \end{pmatrix} \tag{6}$$

where

N is the number of elements bounding the room corresponding to the internal surfaces delimiting the internal air;

Π are the coefficients of the unknown temperatures (θ) (from 1 to N relating to the internal surfaces, N + 1 relating to the internal air);

Γ are the coefficients of the known terms (from 1 to N relating to the internal surfaces, N +1 relating to the internal air);

θ are the unknown temperatures (from 1 to N relating to the internal surfaces, N +1 relating to the internal air).

The "Π" and "Γ" terms are obtained by rewriting Equation (1) and Equation (2) in order to separate the unknown parameters (air temperature at the given time t for Equation (1) and the internal surface temperature for each component at the given time t for Equation (2)) from the known parameters. The form of these equations depends on the solution technique adopted.

4.4 Boundary conditions

4.4.1 Single room

A single room model requires the knowledge of the conditions of adjacent rooms. The two following situations are considered:

— adjacent room with the same conditions (similar rooms);

— adjacent room with defined internal conditions.

If boundary conditions are very different from the above, the simple room model specified in this document shall not be used and it is necessary to calculate the real boundary conditions by a multi-room model able to take account of the heat transfer between the different rooms. This may be achieved by:

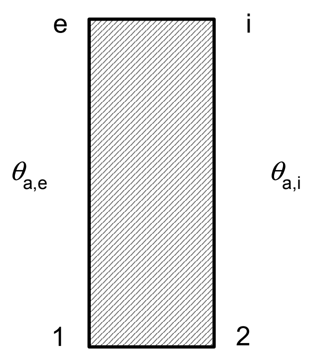
a) simultaneous solution of the global system equations for all rooms, or

b) iterative procedure by considering, as boundary conditions for each room, the temperatures determined at the previous time step.

4.4.2 Similar rooms

4.4.2.1 Partition (vertical) wall

Referring to Figure 4, the following boundary conditions are considered:



Key

1 Similar
2 Internal

Figure 4 - Partition vertical wall

$$
\begin{aligned}
\theta_{a,e} &= \theta_{a,i} \\
q_{sr,e} &= q_{sr,i} \\
q_{lr,e} &= q_{lr,i} \\
h_{c,e} &= h_{c,i}
\end{aligned}
\qquad (7)
$$

where

$\theta_{a,e}$ is the air temperature of the adjacent room;

$\theta_{a,i}$ is the air temperature of the room;

$q_{sr,e}$ is the density of heat flow rate due to absorbed short-wave radiation at the external surface of the wall;

$q_{lr,e}$ is the density of the heat flow rate by long-wave radiation exchanged with the other surfaces of the adjacent room;

$q_{sr,i}$ is the density of heat flow rate due to absorbed short-wave radiation at the internal surface of the wall (see 4.5.3.2);

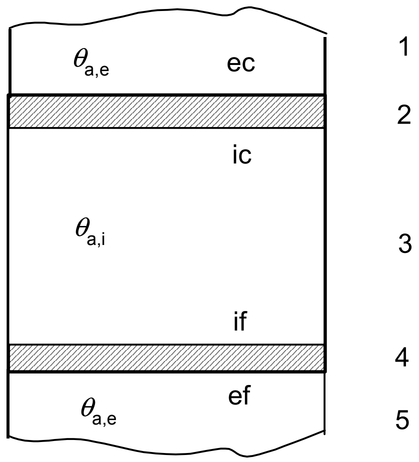
$q_{lr,i}$ is the density of the heat flow rate received by long-wave radiation at the internal surface of the wall from the other internal surfaces (see 4.5.4.2);

$h_{c,e}$ is the convective heat transfer coefficient at the external surface;

$h_{c,i}$ is the convective heat transfer coefficient at the internal surface (see Table 1).

4.4.2.2 Ceiling/floor

Referring to Figure 5, the following boundary conditions are considered:



Key

1 Similar room
2 Ceiling
3 Room
4 Floor
5 Similar room

Figure 5 - Ceiling/floor adjacent to similar rooms

$$\theta_{a,e} = \theta_{a,i}$$

$$q_{sr,ec} = q_{sr,if}$$

$$q_{lr,ec} = q_{lr,if}$$

$$q_{sr,ef} = q_{sr,ic} \tag{8}$$

$$q_{lr,ef} = q_{lr,ic}$$

$$h_{c,ec} = h_{c,if}$$

$$h_{c,ef} = h_{c,ic}$$

where

$\theta_{a,e}$ is the air temperature of the adjacent room;

$\theta_{a,i}$ is the air temperature of the room;

$q_{sr,ec}$ is the density of heat flow rate due to absorbed short-wave radiation at the external surface of the ceiling;

$q_{sr,ic}$ is the density of heat flow rate due to absorbed short-wave radiation at the internal surface of the ceiling (see 4.5.3.2);

$q_{sr,ef}$ is the density of heat flow rate due to absorbed short-wave radiation at the external surface of the floor;

$q_{sr,if}$ is the density of heat flow rate due to absorbed short-wave radiation at the internal surface of the floor (see 4.5.3.2);

$q_{lr,ef}$ is the density of the heat flow rate by long-wave radiation by the external surface of the floor with the other external surfaces;

$q_{lr,if}$ is the density of the heat flow rate by long-wave radiation by the internal surface of the floor with the other internal surfaces (see 4.5.4.2);

$q_{lr,ec}$ is the density of the heat flow rate by long-wave radiation from the external surface of the ceiling to the other external surfaces;

$q_{lr,ic}$ is the density of the heat flow rate by long-wave radiation from the internal surface of the ceiling to the other internal surfaces (see 4.5.4.2);

$h_{c,ec}$ is the convective heat transfer coefficient at the external surface of the ceiling;

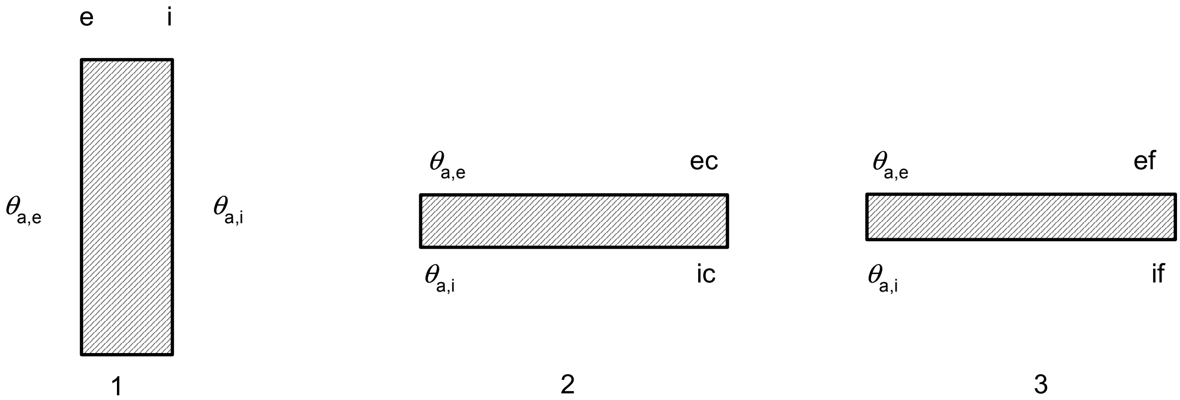
$h_{c,if}$ is the convective heat transfer coefficient at the internal surface of the floor (see Table 1);

$h_{c,ef}$ is the convective heat transfer coefficient at the external surface of the floor;

$h_{c,ic}$ is the convective heat transfer coefficient at the internal surface of the ceiling (see Table 1).

4.4.3 Adjacent room with defined value of the air temperature

For each component of the envelope (see Figure 6) the following boundary conditions are considered:



Key

1 Wall
2 Ceiling
3 Floor

Figure 6 - Wall, ceiling and floor adjacent to room with defined internal conditions

$$\theta_{a,e} = \theta_{a,d}$$

$$q_{sr,e} = 0$$

$$h_{c,e} = h_{c,i}$$

$$h_{c,ec} = h_{c,if} \qquad (9)$$

$$h_{c,ef} = h_{c,ic}$$

where

$\theta_{a,d}$ is the air temperature of the adjacent room;

$q_{sr,e}$ is the density of heat flow rate due to absorbed short-wave radiation at the external surface;

$h_{c,e}$ is the convective heat transfer coefficient at the external surface of the vertical wall;

$h_{c,i}$ is the convective heat transfer coefficient at the internal surface of the vertical wall (see Table 1);

$h_{c,ec}$ is the convective heat transfer coefficient at the external surface of the ceiling;

$h_{c,if}$ is the convective heat transfer coefficient at the internal surface of the floor (see Table 1);

$h_{c,ef}$ is the convective heat transfer coefficient at the external surface of the floor;

$h_{c,ic}$ is the convective heat transfer coefficient at the internal surface of the ceiling (see Table 1).

4.4.4 Floor on ground

The heat transfer between the room and the external environment through the ground is calculated as the sum of a steady state component and a monthly variable component as specified in EN ISO 13370. The monthly variable component is treated as one-dimensional and perpendicular to the floor surface. The calculation procedure shall combine this heat flow rate with the thermal storage of the floor construction together with a 0,5 m thick layer of soil beneath it.

NOTE One acceptable way is to introduce a monthly varying boundary temperature at a depth of 0,5 m beneath the floor construction. This boundary temperature is defined so that the heat flow rate from the room air at its mean monthly temperature to the boundary layer equals the heat flow rate to and through the ground calculated according to EN ISO 13370.

4.4.5 Cellar or crawl space

A cellar is considered as an adjacent room with fixed air temperature (see 4.4.2). A crawl space is treated as a floor on the ground according to EN ISO 13370. Boundary conditions are as specified in 4.4.3.

4.4.6 Ceiling below attic

According to the assumptions of 4.2.6.1, the boundary conditions are represented by:

$\theta_{a,e}$ is the external air temperature;

$q_{sr,e}$ is defined by Equation (17) in 4.5.3.1;

$q_{lr,e}$ is defined by Equation (24) in 4.5.4.1.

4.5 Terms in the thermal balance equations

4.5.1 Heat conduction through components

For elements with constant thermal conductivity and specific heat capacity, the density of heat flow by conduction is governed by the following equations:

$$q_n = -\lambda\left(\frac{\partial\theta}{\partial n}\right) \tag{10}$$

$$\lambda\left(\frac{\partial^2\theta}{\partial x^2}+\frac{\partial^2\theta}{\partial y^2}+\frac{\partial^2\theta}{\partial z^2}\right)+g = c\rho\frac{\partial\theta}{\partial t} \tag{11}$$

where

θ is the temperature of the component (in direction of the heat flow) at the time t;

q is the density of heat flow rate in direction n;

λ is the thermal conductivity of the medium;

c is the specific heat capacity of the medium;

ρ is the density of the medium;

g is the heat source term (heat flow rate per volume);

x,y,z are co-ordinates.

These equations may be solved by any appropriate procedure which provides results in accordance with the validation procedure given in Clause 7.

NOTE A suitable procedure is described in Annex A.

4.5.2 Convective heat transfer

4.5.2.1 General

Convective heat transfer occurs at the boundary surfaces of each building element and through air layers.

4.5.2.2 Convective heat flow rate at the surfaces of an element

The density of convective heat flow rate at the internal and external surface of element is given by:

$$q_c = h_c (\theta_s - \theta_a) \tag{12}$$

where

h_c is the convective heat transfer coefficient of the surface;

θ_s is the surface temperature;

θ_a is the air temperature.

At the external surface the values of the convective heat transfer coefficient $h_{c,e}$, is given by:

$$h_{c,e} = 4 + 4v \tag{13}$$

where

v is the wind velocity near the surface.

The wind velocity near the surface, v, depends on the climatic data of the locality and on the envelope characteristics. Unless otherwise specified, the value of 1 m/s shall be used. The values of the convective heat transfer coefficient at the internal surface, $h_{c,i}$, are given in Table 1.

Table 1 - Convective heat transfer coefficient at the internal surface

Vertical wall W/(m²·K)	**Heat flow upwards** W/(m²·K)	**Heat flow downwards** W/(m²·K)
2,5	5,0	0,7
NOTE The values in Table 1 were determined using the equations given in EN ISO 6946 for the following conditions: - temperature difference ($\theta_{s,i}$ - $\theta_{a,i}$) < 10 K; - surface hydraulic diameter = 4,5 m (4 × area/perimeter).		

The air temperature required in Equation (12) is:

— for internal surfaces: the room air temperature;

— for external surfaces: the conditions given in Table 2.

Table 2 - Air temperature

Buildings element	Air temperature conditions
External wall, roof	External air temperature
Partition wall, ceiling and roof to similar room	Internal air temperature
Partition wall, ceiling and roof to adjacent room with different conditions	Air temperature of the adjacent room
Floor on ground	Mean monthly external air temperature
Floor on cellar	Temperature of the cellar

4.5.2.3 Convective heat transfer through unventilated air layers

The density of convective heat flow rate through an unventilated air layer, q_c, is given by:

$$q_c = \Lambda_a \Delta\theta \tag{14}$$

where

$\Delta\theta$ is the temperature difference between the surfaces delimiting the layer;

Λ_a is the thermal conductance of the air layer.

The thermal conductance of an unventilated air layer is calculated according to:

— EN ISO 6946 between opaque surfaces;

— EN ISO 10077-1 between transparent surfaces.

NOTE For transparent surfaces, the thermal conductance of an unventilated air layer can be calculated assuming the following reference conditions:

— air density: 1,139 kg/m^3

— dynamic viscosity: 1,861 × 10^{-5} kg/(m·s)

— thermal conductivity: 0,0264 W/(m·K)

— specific heat capacity: 1008 J/(kg·K)

— thermodynamic temperature: 300 K

— temperature difference: 5 K.

Table 3 gives some values of thermal conductance, Λ_a, for vertical and horizontal unventilated air layers between transparent components. For other thicknesses, thermal conductances may be derived by interpolation.

Table 3 - Thermal convective conductance of unventilated air layers

Air layer thickness	**Vertical air layer**	**Horizontal air layer**	
m	Thermal conductance Λ_a W/(m^2·K)	Heat flow upwards Λ_a W/(m^2·K)	Heat flow downwards
		Thermal conductance Λ_a W/(m^2·K)	Thermal conductance Λ_a W/(m^2·K)
0,01	2,65	2,06	2,06
0,05	1,16	1,71	0,41
0,10	1,29	1,50	0,21
0,20	1,42	1,48	0,10

4.5.2.4 Convective heat transfer through ventilated air layer

The convective heat flow rate through a ventilated air layer, Φ_{va}, depends on the air flow rate in the air layer. The heat flow rates to be considered are:

a) the convective heat flow rate, Φ_{va}, due to air passing through the air layer and into the room, given by:

$$\Phi_{va} = m_{a,v}\, c_a (\theta_l - \theta_{a,i}) \tag{15}$$

where

$m_{a,v}$ is the mass air flow through the air layer;

θ_l is the temperature of the air leaving the layer;

b) the convective heat flow rate, $\Phi_{c,j}$, between surfaces and air, given by:

$$\Phi_{c,j} = h_a A_c (\theta_j - \theta_{eq})$$

$$\Phi_{c,j+1} = h_a A_c (\theta_{j+1} - \theta_{eq}) \tag{16}$$

where

A_c is the area of the surface in contact with the air layer;

h_a is the convective heat transfer coefficient for ventilated layers;

θ_{eq} is the equivalent temperature of the air in the layer;

j and j + 1 refer to the surfaces delimiting the air layer.

NOTE A procedure for determining the parameters in Equations (15) and (16) is given in Annex B.

4.5.3 Short-wave radiation heat transfers

4.5.3.1 Short-wave radiation heat transfer at the external surface of opaque element

The density of short-wave radiation heat flow rate at the external surface of an opaque element is given by:

$$q_{sr,e} = \alpha_{sr}(f_s I_D + I_d) \tag{17}$$

where

α_{sr} is the solar absorptance;

f_s is the sunlit factor;

I_D is the direct component of the solar radiation reaching the surface;

I_d is the diffuse component of the solar radiation reaching the surface.

The values of solar absorptance of external opaque surfaces, α_{sr}, depend on the characteristics of the external surface of the element. Table 4 gives values of the solar absorptance as a function of the colour of the external surface that may be used when no specific values are available.

Table 4 - Solar absorptance of external opaque surfaces

	Light colour	Intermediate colour	Dark colour
α_{sr}	0,3	0,6	0,9

The values of the direct, I_D, and diffuse, I_d, components of the solar radiation reaching the differently oriented surfaces may be derived from national data.

The sunlit factor, f_s, is given by:

$$f_s = \frac{A_s}{A} \tag{18}$$

A_s is the sunlit area of the wall (defined in 4.5.3.5);

A is the total area of the wall.

4.5.3.2 Short-wave radiation heat transfer at the internal surface of opaque elements

The density of heat flow rate by short-wave radiation absorbed at the internal surface of an opaque element is given by:

$$q_{sri} = (1 - f_{sa})(1 - f_{sl})(\Phi_{sr,D} + \Phi_{sr,d}) f_d \tag{19}$$

where

f_{sa} is the solar to air factor of the room;

f_{sl} is the solar loss factor of the room;

f_d is the distribution factor of the solar radiation at the internal surface of the element;

$\Phi_{sr,D}$ is the heat flow rate due to the direct component of solar radiation entering the room;

$\Phi_{sr,d}$ is the heat flow rate due of the diffuse component of solar radiation entering the room.

The heat flow rates due to the direct and diffuse components of the solar radiation, entering the room, are given by Equations (20) and (21), respectively:

$$\Phi_{sr,D} = \sum_{j=1}^{J} (I_D \tau_D A_s)_j \tag{20}$$

$$\Phi_{sr,d} = \sum_{j=1}^{J} (I_d \tau_d A)_j \tag{21}$$

where

J is the number of glazing components;

I_D is the direct component of solar radiation reaching the external surface of the system j;

I_d is the diffuse component of solar radiation reaching the external surface of the glazing system j;

τ_D is the direct solar transmittance of the glazing system;

τ_d is the diffuse solar transmittance of the glazing system;

A_s is the sunlit area of the glazing (see 4.5.3.5);

A is the glazing area.

The direct and diffuse solar transmittance of each glazing system τ_D and τ_d shall be determined according to EN 410. If no values are available, τ_D and τ_d shall be calculated at the normal incident angle.

Solar to air factor

The solar to air factor, f_{sa}, is the fraction of solar heat entering the room through the glazing which is immediately transferred to the internal air. This fraction depends on the quantity of internal items with very low thermal capacity such as carpets and furniture. It is assumed to be time independent.

Solar loss factor

The solar loss factor, f_{sl}, is the fraction of the solar radiation entering the room which is reflected back to the external environment. It depends on the geometrical characteristics and solar properties of the glazing system, the exposure of the glazing, the solar angles and the room geometry and colour of the surfaces. It is assumed to be time independent.

Distribution factors

The distribution factors, f_d, define the amount of the direct solar radiation absorbed per area at the different internal surfaces of the walls, ceiling, floor, etc. They depend on the solar angles, the geometrical dimensions of glazing and room, the short-wave reflectance of components, and the furniture and furnishings. It is assumed to be time independent.

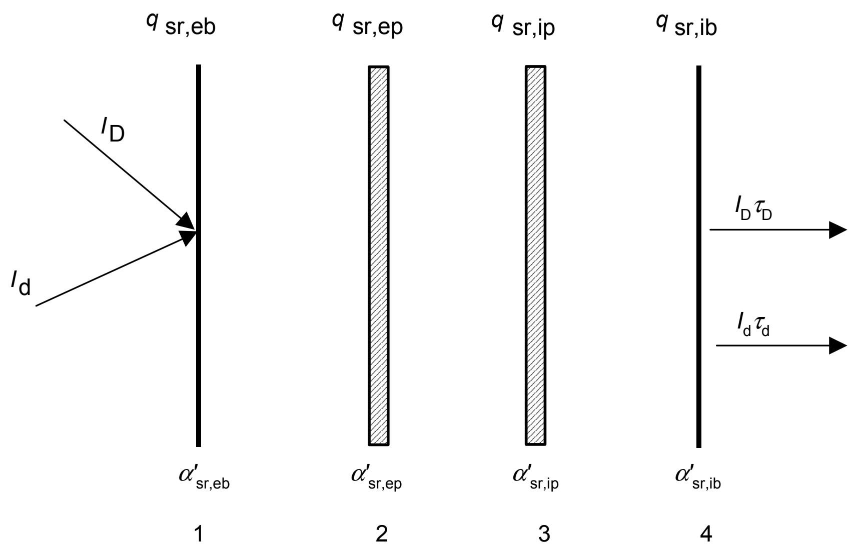
NOTE Unless otherwise specified, values of f_{sa}, f_{sl} and f_d given in informative Annex G can be used.

4.5.3.3 Short-wave radiation heat flow rate for transparent elements (including blinds and curtains)

Transparent elements partially transmit, reflect and absorb the solar radiation impinging on their external surface. For multi-layered glazed elements (e.g. double pane window with internal and/or external blind) the following solar parameters are required:

— direct and diffuse solar energy transmittance of the system, τ_D and τ_d;

— equivalent direct and diffuse solar energy absorptance of each component of the glazed element, α'_{sr}.



Key

1 External blind (eb)
2 External pane (ep)
3 Internal pane (ip)
4 Internal blind (ib)

Figure 7 – Double pane with internal and external blinds

These solar parameters shall be calculated according to EN 410 taking into account the solar energy transparence and reflectance of each component. If no data are available for various incident angles, values of these parameters at the normal incident angle can be taken as safe values. The density of heat flow rate for element *j* of the glazing system due to the absorbed solar radiation is then determined as:

$$q_{\mathrm{sr},j} = \alpha'_{\mathrm{sr},j}(I_\mathrm{D} f_\mathrm{s} + I_\mathrm{d}) \qquad (22)$$

where

j is the element of the glazing system;

α'_{sr} is the equivalent solar absorptance.

If curtains or venetian blinds are present, the following situations can occur:

a) curtain/blind completely closed;

b) curtain/blind not completely closed.

In case a) the glazing component and the curtain/blind are treated as a single envelope component having appropriate solar coefficients.

In case b) two different components shall be considered:

— the portion of glazing area not covered by the curtain/blind, comprising the glazing component only;

— the portion of glazing area covered by the curtain/blind, treated as in case a).

4.5.3.4 Solar to air heat flow rate

The solar to air heat flow rate, Φ_{sra}, is the heat flow rate due to solar radiation, entering through the glazing system, directly transferred to the internal air. It is given by:

$$\Phi_{\mathrm{sa}} = f_{\mathrm{sa}}(1 - f_{\mathrm{sl}}) \cdot (\Phi_{\mathrm{sr,D}} + \Phi_{\mathrm{sr,d}}) \qquad (23)$$

The parameters in Equation (23) are the same as in Equation (19).

4.5.3.5 Sunlit area of room element

When external obstructions are present, the area of an element can be partially shaded. Obstructions considered in this document are: horizontal overhangs, side fins, window set back and surrounding constructions. A reduction factor may be added for the diffuse radiation in Equations (16) and (19). For the sake of safety, it may be assumed that:

— the effect of shading is only to reduce the direct solar component;

— the effect of mutual reflections is negligible.

The evaluation of the sunlit area may be obtained by any appropriate procedure that provides results in accordance with the validation tests given in Clause 7.

NOTE A suitable procedure is described in Annex C.

4.5.4 Long-wave radiation heat transfer

4.5.4.1 Heat flow rate at the external surface

According to the assumptions of 4.1, the density of net long-wave radiant heat flow rate received by an external surface, $q_{\mathrm{lr,e}}$, is given by:

$$q_{\mathrm{lr,e}} = h_{\mathrm{lr,e}}(\theta_{\mathrm{a,e}} - \theta_{\mathrm{s,e}}) - q_{\mathrm{sk}} \tag{24}$$

where

$h_{\mathrm{lr,e}}$ is the long-wave radiative heat transfer coefficient;

$\theta_{\mathrm{a,e}}$ is the external air temperature;

$\theta_{\mathrm{s,e}}$ is the external surface temperature;

q_{sk} is the correction for the long-wave radiation exchanges from the wall to the sky.

Using thermodynamic temperatures ($T = \theta + 273{,}15$), the value of $h_{\mathrm{lr,e}}$ is approximated by:

$$h_{\mathrm{lr,e}} = 4\varepsilon\sigma\left(\frac{T_{\mathrm{a,e}} + T_{\mathrm{s,e}}}{2}\right)^3 \tag{25}$$

where

ε is the long-wave emissivity of the surface;

σ is the Stefan-Boltzmann constant;

$T_{\mathrm{a,e}}$ is the external air temperature;

$T_{\mathrm{s,e}}$ is the surface temperature.

According to the assumptions of 4.1, the calculations shall be made with a fixed value of $h_{\mathrm{lr,e}}$.

NOTE 1 The terms of Equation (24) can be calculated with the following conditions:

- emissivity of the external surface ε $l_{\mathrm{lr,e}}$ = 0,93;

- reference temperature $T_{\mathrm{m}} = \dfrac{T_{\mathrm{a,e}} + T_{\mathrm{s,e}}}{2} = 303$ K

Under these conditions, the value of $h_{\mathrm{lr,e}}$, for external surfaces, is 5,5 W/(m$^2\cdot$ K).

The correction for the long-wave radiation emitted from the element to the sky, q_{sk}, is given by:

$$q_{\mathrm{sk}} = F_{\mathrm{sk}}\, \varepsilon\, \sigma\left(\frac{T_{\mathrm{a,e}} + T_{\mathrm{sk}}}{2}\right)^3 \left(T_{\mathrm{a,e}} - T_{\mathrm{sk}}\right) \tag{26}$$

where

F_{sk} is the view factor from the element with the sky (solid angle divided by 2π);

$T_{\mathrm{a,e}}$ is the external air temperature;

T_{sk} is the temperature of the sky.

The temperature of the sky depends on the characteristics of the atmosphere and its vapour content. Values of these parameters are fixed at national level.

NOTE 2 If no information is available, the procedure of Annex E can be used for determining the various parameters. Equation (F.2) (valid for clear sky) can be used for determining the temperature of the sky; Equation (F.5) can be used for determining the correction term q_{sk}.

4.5.4.2 Heat flow rate at the internal surface

The density of net long-wave radiant heat flow rate, q_{lr}, received by the internal surface *j*, is given by:

$$q_{lr,j} = \sum_{k=1}^{N} (F_{j,k} J_{lr,k}) - J_{lr,j} \tag{27}$$

where

N is the number of surfaces delimiting the environment;

$F_{j,k}$ is the view factor from surface *j* to surface *k*;

$J_{lr,j}$ is the long-wave radiosity of the surface *j*;

$J_{lr,k}$ is the long-wave radiosity of the surface *k*.

$F_{j,k}$ is the mean value, over surface *j*, of a solid angle over which surface *k* is seen from a point on surface *j*, divided by 2π.

The long-wave radiosity of a surface is the total density of heat flow rate emitted and reflected by this surface, all surfaces being here considered as grey bodies. Thus the long-wave radiosity of surface *j* is:

$$J_{lr,j} = \rho_j \sum_{k=1}^{N} (F_{j,k} J_{lr,k}) + \varepsilon_j \sigma T_j^{\ 4} \tag{28}$$

where

ρ is the long-wave radiative reflectance;

ε is the long-wave radiative emittance;

σ is the Stefan-Boltzmann constant.

In order to calculate the long-wave radiant heat flow exchanged by the *N* different inside surfaces, the radiosity J_{lr} for each surface shall be first determined by solving the *N* simultaneous equations. The solution of Equation (27) shall be carried out for the various surfaces bounding the room.

Equation (27) may be solved by any appropriate procedure which provides results in accordance with the validation procedure given in Clause 7.

NOTE A suitable procedure is described in Annex F.

4.5.4.3 Air layers

The density of long-wave radiative heat transfer through air layers is given by:

$$q_{lr} = \Lambda_r \, \Delta\theta \tag{29}$$

where

$\Delta\theta$ is the temperature difference between the surfaces delimiting the air layer;

Λ_{lr} is the long-wave radiative conductance of the air.

The long-wave radiative conductance is given in:

— EN ISO 10077-2 for an air layer between glazing surfaces;

— EN ISO 6946 for an air layer between opaque layers.

For ordinary surfaces the following values are considered:

— opaque surface (ε = 0,93) Λ_{lr} = 5,0 W/(m^2·K);

— transparent surfaces (ε = 0,837) Λ_{lr} = 4,4 W/(m^2·K);

— opaque surface (ε = 0,93) and transparent surface (ε = 0,837) Λ_{lr} = 4,6 W/(m^2·K).

4.5.5 Internal gains

Internal gains usually derive from lighting, equipment and occupants. The relevant heat flow rate includes a convective component, $\Phi_{i,c}$, and a long-wave radiative component, $\Phi_{i,r}$, which are respectively included in Equations (1) and (2). The total radiative component, $\Phi_{i,r}$, is supposed to be uniformly distributed on all the internal surfaces bounding the room, including windows.

NOTE Unless otherwise specified, the heat flow rate values given in Annex H for internal sources can be used.

4.5.6 Heat flow due to ventilation

4.5.6.1 General

The net heat flow rate to the room air due to natural and mechanical ventilation is calculated as:

$$\Phi_v = c_a q_a (\theta_{il} - \theta_{a,i}) \tag{30}$$

where

c_a is the specific heat capacity of the inlet air;

q_a is the mass air flow rate;

θ_{il} is the inlet air temperature;

$\theta_{a,i}$ is the internal air temperature.

The inlet air temperature depends on its source (e.g. external air or adjacent room). The mass air flow rate results from natural and/or forced ventilation.

4.5.6.2 Natural ventilation

The air flow rate due to natural ventilation depends on the dimensions and type of cracks and openings (including doors and windows), the temperature difference, the wind speed and its direction. It is given by:

$$m_a = c_d \rho A \left(\frac{2 \Delta p}{\rho} \right)^n \tag{31}$$

where

c_d is the coefficient of discharge;

A is the area of the opening;

Δp is the pressure difference between internal and external environments;

ρ is the density of the air;

n is a coefficient between 0,5 and 1.

The values of coefficients c_d, n and area A depend on the position and flow characteristics of all openings. It is assumed that the air in the room is well mixed.

NOTE Annex J gives a suitable procedure to evaluate the volumetric air flow rate by natural ventilation.

4.5.6.3 Mechanical ventilation

The mass air flow rate due to forced ventilation depends on the characteristics of the supply/exhaust system.

5 Procedure for carrying out calculations

5.1 General

This document is used for calculating the internal temperatures (air and internal surfaces) for the intended purposes given in Clause 1. It should, however, be noted that these "predicted internal temperatures" are determined on the basis of a set of assumptions described in various clauses of this document. Using a different set of assumptions, e.g. boundary conditions or surface coefficients, will lead to a different set of "predicted internal temperatures". Furthermore, the calculations for determining "predicted internal temperatures" shall be carried out according to the procedure and values for the design climate data, design geometric and thermophysical characteristics of the room, and its components, and the design internal gains as defined below.

5.2 Design climatic data

5.2.1 General

The calculation method uses time varying values of various climatic data. The time period and the type of climatic data used affect the calculation of internal temperatures. The design climatic data may be given in a national annex to this document according to the following methods:

— long period design climatic data;

— design warm sequence.

5.2.2 Long-period design climatic data

If the internal temperatures are determined from a long period of climatic data, the hourly values of the following are required for the whole of period considered: the value of direct and diffuse solar radiation components on the various facades, the external air temperature, the wind velocity and its prevailing direction. Climatic data may be given in a national annex to this document.

5.2.3 Design warm sequence

Such a sequence of several days shall be chosen so as to be representative of the hottest conditions for a given climate. Because the hottest internal conditions do not necessarily coincide with the warmest external conditions, it may be necessary to consider the different climatic situations for several months. For each month, the design warm sequence shall be defined by:

— hourly values of direct and diffuse (sky and ground reflected) components of solar radiation on the various facades;

— hourly values of the external air temperature;

— hourly values of the wind velocity and its prevailing direction.

NOTE Annex D gives a procedure for deriving the design warm sequences.

The starting internal conditions are calculated by using the average monthly values of:

— hourly values of direct and diffuse (sky and ground reflected) components of solar radiation on the various facades;

— hourly values of the external air temperature;

— mean monthly value of the wind velocity and its prevailing direction.

5.3 Geometrical and thermophysical characteristics of room elements

The appropriate boundary conditions are given in 4.4. For each element the following properties shall be provided:

a) geometrical properties: width and height (or length and width) of each component measured from the internal surfaces;

b) thermophysical properties: thermal conductivity, density and specific heat of each layer of layered elements and thermal resistance of air gaps and window/blind thermal characteristics.

5.4 Design internal gains

Internal gains are generally due to occupants, equipment and lights. These contributions vary with occupant behaviour. Design internal gains may be given in a national annex to this document.

NOTE Annex H gives data which can be used unless other data are available.

5.5 Design occupant behaviour

Occupant behaviour greatly modifies the internal conditions of a room. The occupants influence the amount of internal heat sources, natural ventilation, and blind arrangements. The reference behaviour to introduce in the "design" calculation may be defined from national data. For a design calculation, the time of opening and closing of a window, the area of opening and the type of blind, shall be fixed.

Design occupant behaviour may be given in a national annex to this document.

5.6 Calculation procedure

5.6.1 General

The internal temperatures are determined by applying the calculation procedure defined in 4.4 and 4.5, considering the climatic data (see 5.2), the geometrical and thermophysical characteristics of the room elements (see 5.3), the internal gains (see 5.4), the occupant behaviour (see 5.5), and the starting conditions defined below. The calculation procedure involves the two consecutive steps:

a) evaluation of the starting conditions;

b) evaluation of the design internal conditions.

5.6.2 Definition of the starting conditions

If the long-period design climatic data are used, the calculation shall be carried out for at least two weeks and the resulting conditions shall be used as the starting conditions. If the design warm sequence is used, the starting conditions are obtained by repeating the calculation given in 4.2 and 4.3 for the mean monthly values of the climatic data defined in 5.2.3, until the predicted internal air temperature over two consecutive days differs by less than 0,01 K. These conditions are then used as the starting conditions.

5.6.3 Prediction of the internal temperatures

The internal temperatures are determined using, as initial conditions, the internal conditions evaluated in 5.6.1. For long-period design climatic data, the internal temperatures are determined hour by hour by considering climatic data defined in 5.2.2. For the design warm sequence, the internal temperatures are determined by iterating the procedures in 4.2 and 4.3, using the climatic data corresponding to the design warm sequence. The number of days depends on the characteristics of the local climatic data. The hourly values of the internal temperature shall be calculated as the average over each hour.

6 Report of the calculation

The calculation report shall include the input data adopted and the results of the calculation.

a) Input data:

— climatic data (hourly values of the external air temperature and solar radiation intensity);

— building characteristics: description of the building and of the rooms investigated;

— volume of room;

— for each element bounding the room:

opaque elements: area, exposure, thermophysical properties of each layer;

glazed elements: area, exposure, thermophysical and solar characteristics of each glazed element.

Local clock time shall be used for all time-dependent input data with the exception of climatic data. If the climatic data time convention is different from local clock time, the difference shall be reported.

b) Results:

— hourly values of the air ventilation flow rate (number of changes per hour);

— hourly values of the heat flow rate for internal sources (watts per square metre of floor area);

— hourly values of air temperature and mean radiant temperature.

The predicted temperatures shall be reported for the calculation period and not for the pre-conditioning period.

7 Validation procedures

7.1 Introduction

This document does not impose any specific numerical technique for the calculation of internal temperatures of a single room. The annexes give procedures for the calculation of the different parameters necessary for determining the internal temperature, according to the assumptions included in this document.

Any existing or new numerical solution which claims conformity with this document shall be validated with the tests in this section and be in agreement with the procedures and assumptions.

The results provided by any numerical solution model shall be within the range indicated for each test. The validation procedures refer both to each relevant heat transfer process and to the whole solution model.

NOTE The check of existing or new solution models can be made by the producers of the numerical solution models as well as the producers of computer programs.

7.2 Validation of heat transfer processes

7.2.1 General

Checks shall be made of the following processes:

a) heat conduction through opaque elements;

b) internal long-wave radiation heat exchanges;

c) evaluation of short-wave radiation heat transfer (calculation of shaded area of a window due to external obstructions).

7.2.2 Heat conduction through opaque elements

This procedure requires the evaluation of the internal air temperature of the room specified below at several time intervals.

a) Characteristics of the room:

— internal dimensions 1 m × 1 m × 1 m;

— all elements, including ceiling and floor, are identical with the same boundary conditions;

— the short-wave radiative heat transfer is assumed to be zero;

— the air flow rate due to the ventilation is assumed to be zero;

— the internal convective heat transfer coefficient of each element, including ceiling and floor, is $h_{c,i}$ = 2,5 W/(m^2·K);

— the external convective heat transfer coefficient of each element, including ceiling and floor, is $h_{c,e}$ = 8 W/(m^2·K);

— the emissivities of the internal and external surface of each element, including ceiling and floor, are assumed to be zero (the long-wave radiative heat transfer on the internal and external surfaces are assumed to be zero);

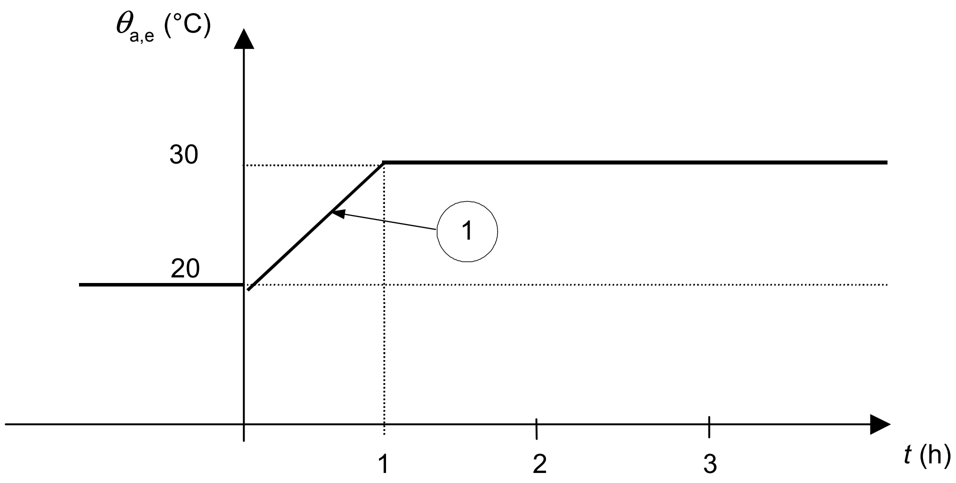
— the thermal capacity of the room air is assumed to be zero.

b) Boundary conditions:

— the external air temperature is variable according to Figure 8:

$t \leq 0$: $\theta_{a,e}$ = 20 °C; for $0 < t \leq 1$ (hour): linear variation of the external temperature $\theta_{a,e}$ from 20 °C to 30 °C; $t > 1$ h: $\theta_{a,e}$ = 30 °C;

the internal air temperature $\theta_{a,i}$ is constant at 20 °C for $t \leq 0$.



Key

1 Linear variation of the external temperature

Figure 8 - Variation of the external air temperature

Data to be calculated: the internal air temperature shall be determined after the following times:

a) 2 h; b) 6 h; c) 12 h; d) 24 h; e) 120 h

Tests shall be conducted for the room elements given in Table 5.

Table 5 - Characteristics of the room elements

Test No.	Thickness d m	Thermal conductivity λ W/(m·K)	Density ρ kg/ m³	Specific heat capacity c kJ/(kg·K)
1	0,20	1,2	2000	1,0
2[a]	0,10	0,04	50	1,0
3[a]	0,20	1,2	2000	1,0
	0,10	0,04	50	1,0
	0,005	0,14	800	1,5
4[a]	0,005	0,14	800	1,5
	0,10	0,04	50	1,0
	0,20	1,2	2000	1,0

[a] material layers are listed starting from the external side of the element

For each test, the differences between the values of the internal air temperature, for each time considered, shall be less than 0,5 K from the values given in Table 6.

Table 6 - Reference values of the internal air temperature, in °C

Test	Time				
	2 h	6 h	12 h	24 h	120 h
1	20,04	21,26	23,48	26,37	30,00
2	25,09	29,63	30,00	30,00	30,00
3	20,00	20,26	21,67	24,90	29,95
4	20,00	20,06	20,25	20,63	23,17

7.2.3 Internal long-wave radiation exchanges

The validation procedure is based on the calculation, in steady-state conditions, of the internal air temperature of various rooms having opaque walls, and adjacent to spaces with different external air temperatures. Referring to the various tests below indicated, the thermal conductance of surface No.2 (external wall) is Λ = 5,0 W/(m^2·K); the thermal conductance of the other walls (partition vertical walls, ceiling and floor) is Λ = 1,0 W/(m^2·K). The room geometries for the various tests are given in Table 7, and Figures 9a, 9b, 9c and 9d.

Table 7 - Room geometry

Test No.	Surface No.1 Partition vertical wall m^2	Surface No.2 External vertical wall m^2	Surface No. 3a)+3b)+3c) Partition vertical wall m^2	Surface No.4 Ceiling m^2	Surface No.5 Floor m^2	Volume m^3
1	1	1	2	1	1	1
2	18	12	30	24	24	72
3	9	90	99	90	90	270
4	18	6	36	24	24	72

The boundary conditions considered in the various tests are given in Table 8.

Table 8 - Boundary conditions

Surface	External air temperature $\theta_{a,e}$ °C	Surface coefficient of heat transfer			Total hemispherical emissivity ε
		$h_{lr,e}$ W/(m^2·K)	$h_{c,e}$ W/(m^2·K)	$h_{c,i}$ W/(m^2·K)	
Surface No.2	30	5,5	8	2,5	0,9
Other surfaces	20	5,5	8	2,5	0,9

The internal surface of wall No. 2 of each geometry absorbs a short-wave radiative density of heat flow rate equal to 100 W/m^2, time-independent. For other surfaces, the short-wave radiative density of heat flow rate is zero. For the test No.4 the surface No.2 (4 m × 1,5 m) is the external wall.

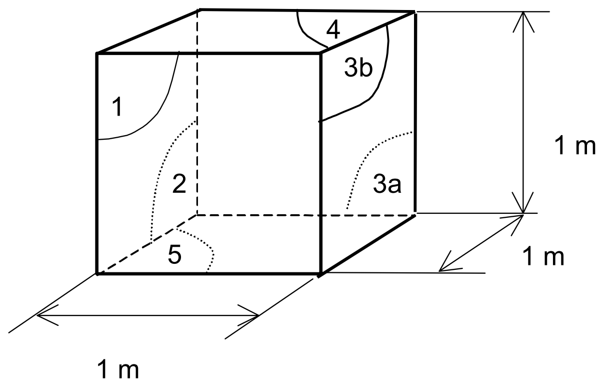


Figure 9a - Test No. 1: cubic geometry (1 m × 1 m × 1 m)

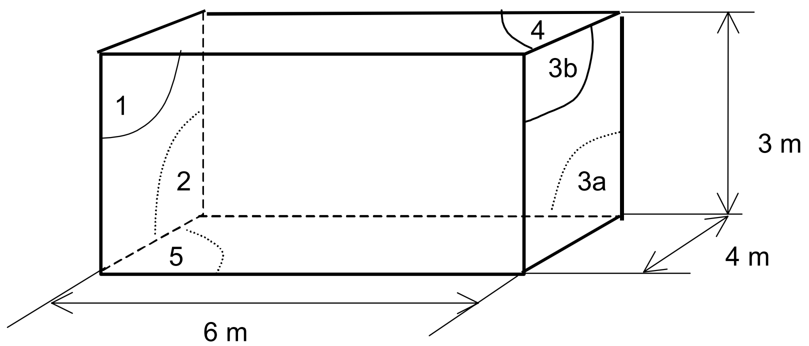


Figure 9b - Test No. 2: non-cubic geometry (3 m × 6 m × 4 m)

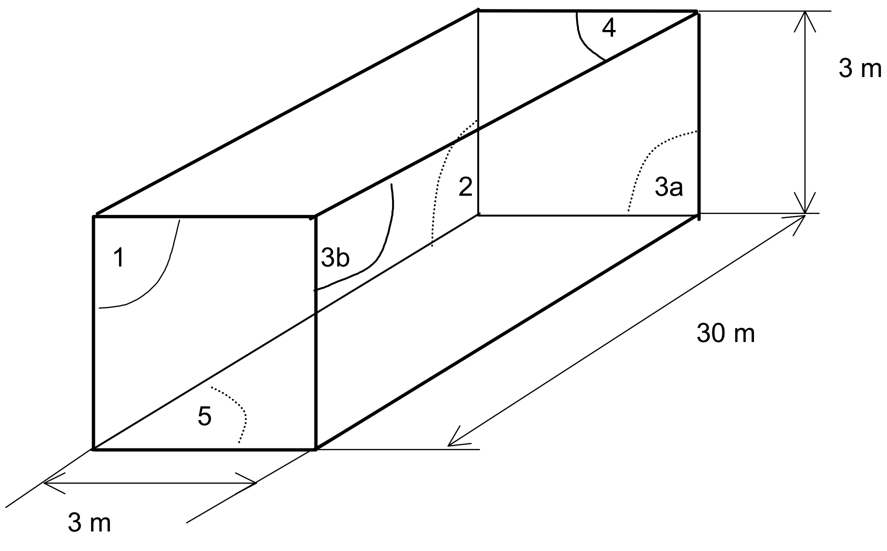


Figure 9c - Test No. 3: non-cubic geometry (3 m × 30 m × 3 m)

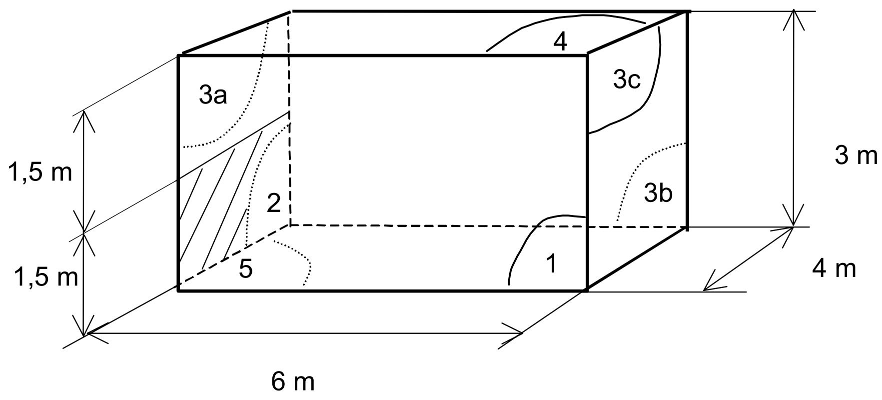


Figure 9d - Test No. 4: configuration (6 m × 4 m × 3 m) with a seventh surface

The values obtained shall not differ by more than 0,5 K from those in Table 9.

Table 9 - Internal air temperature, in °C

	Test No. 1	**Test No. 2**	**Test No. 3**	**Test No 4**
Result	34,4	30,4	38,5	25,5

7.2.4 Sunlit area of a window due to external obstructions

The validation procedure requires the evaluation of the sunlit factor f_s defined in Equation (18) as the ratio between the sunlit area of the plane surface and its total area, for the following tests:

Test No. 1: South orientation - see Figure 10a: overhang;

Test No. 2: South orientation - see Figure 10b: side fins;

Test No. 3: South orientation - see Figure 10c: overhang + side fins;

Test No. 4: South orientation - see Figure 10d: external obstruction;

Test No. 5: East orientation - see Figure 10e: overhang + side fins;

Test No. 6: East orientation - see Figure 10f: external obstruction.

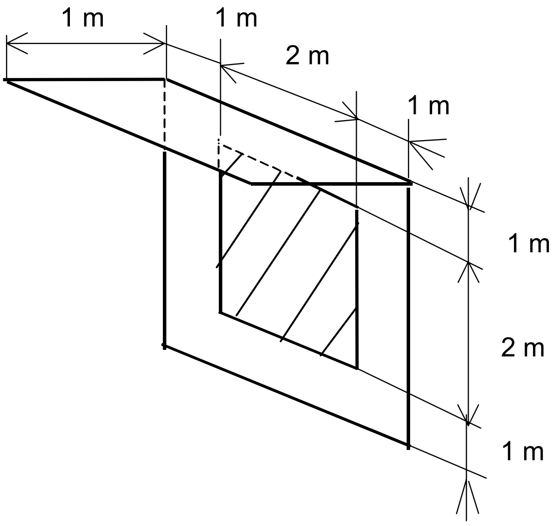


Figure 10a - Test No. 1: South orientation

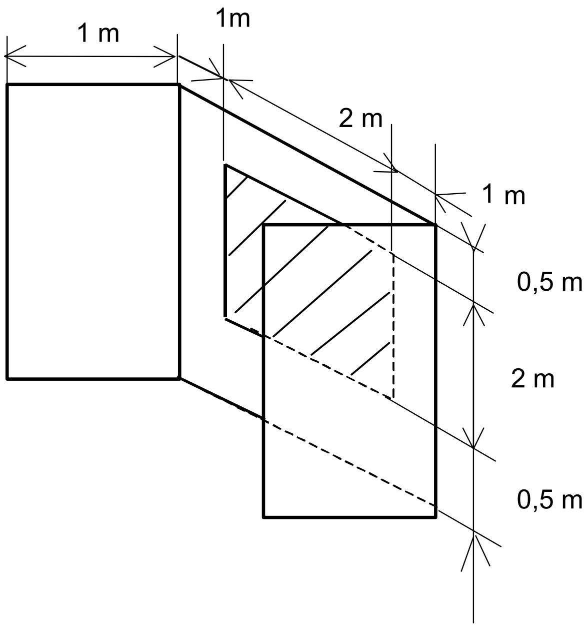


Figure 10b - Test No. 2: South orientation

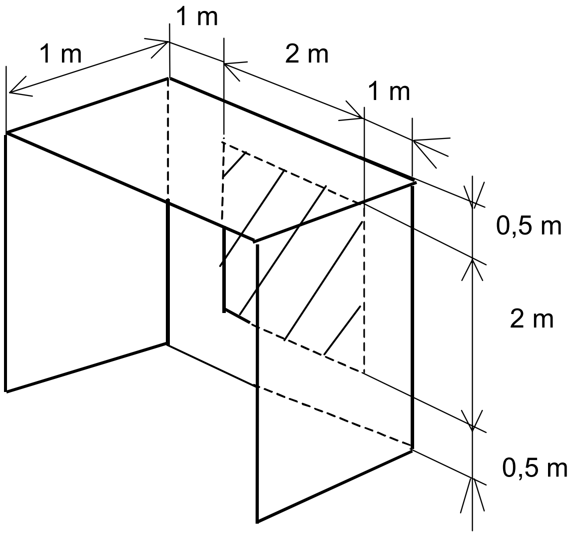


Figure 10c - Test No. 3: South orientation

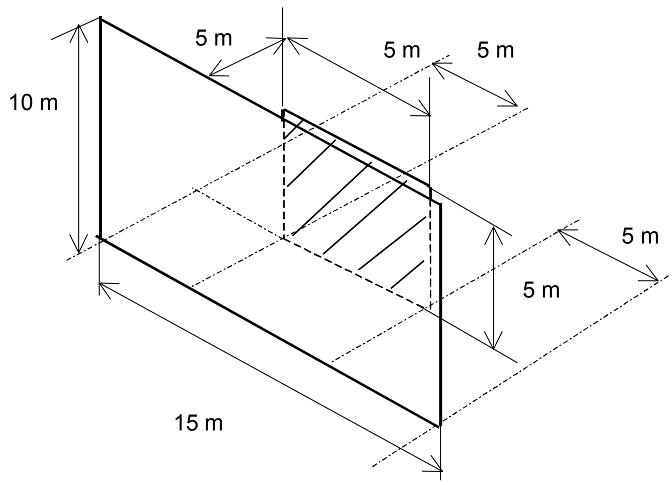


Figure 10d - Test No. 4: South orientation

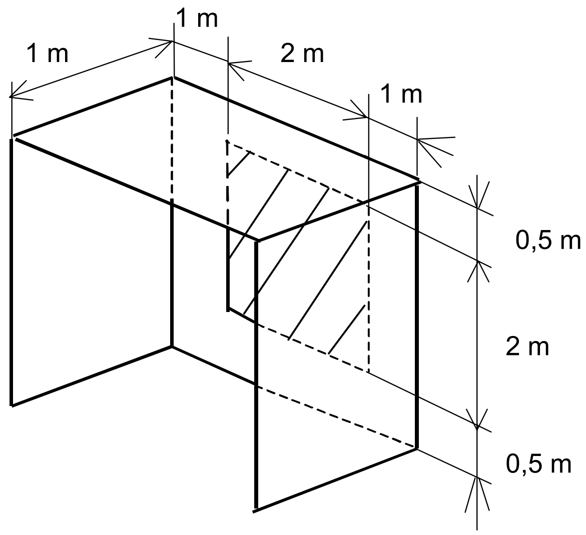


Figure 10e - Test No. 5: East orientation

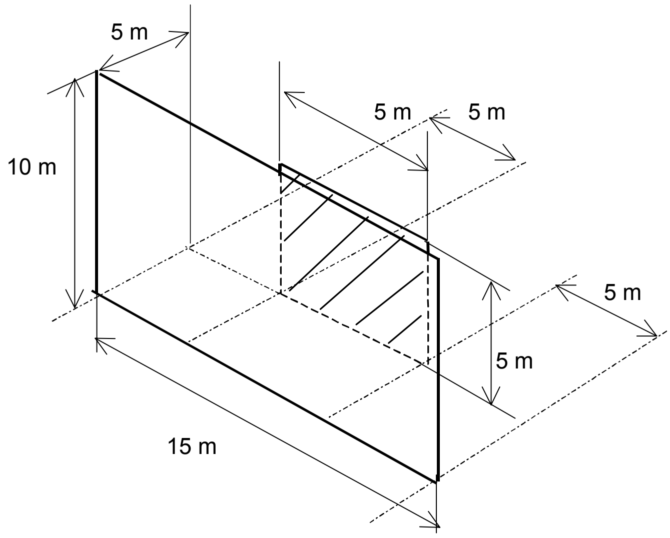


Figure 10f - Test No. 6: East orientation

The hourly solar angles to be used for the tests are determined at each 30 min and reported in Table 10.

Table 10 - Hourly solar angles

Hour	**Solar altitude angle** degrees	**Solar azimuth angle** degrees	**Hour**	**Solar altitude angle** degrees	**Solar azimuth angle** degrees
4: 30	2,9	120,6	8 : 30	40,16	74,97
5: 00	7,25	115,06	9 : 00	44,73	67,82
5: 30	11,70	109,61	9 : 30	49,05	59,80
6: 00	16,31	104,22	10 : 00	53,02	50,64
6: 30	21,04	98,79	10 : 30	56,4	40,10
7: 00	25,83	93,26	11 : 00	59,15	28,01
7: 30	30,65	87,54	11 : 30	60,91	14,46
8: 00	35,44	81,49	12 : 00	61,51	0,0

The solar azimuth angle of Table 10 is eastwards from South, i.e. 0° at south, positive to the east and negative to the west.

For each test, the sunlit factor f_s shall be determined at the following times:

7:00 a.m.; 7:30 a.m.; 8:00 a.m.; 8:30 a.m.; 9:00 a.m.; 9:30 a.m.; 10:00 a.m.; 10:30 a.m.; 11:00 a.m.; 11:30 a.m.; 12:00 noon.

The validation requires to check the values of the sunlit factor at each hour or at each half hour according to the characteristics of the solution model adopted. Each value of f_s shall not differ from those in Table 11 by more than 0,05. Negative values are not consistent.

Table 11 - Value of the sunlit factor, f_s, for various cases

Hour	Test No. 1	Test No. 2	Test No. 3	Test No. 4	Test No. 5	Test No. 6
7: 00	0,00	0,00	0,00	0,00	1,00	0,00
7: 30	0,66	0,34	0,00	1,00	0,95	0,00
8: 00	0,53	0,47	0,00	1,00	0,89	0,00
8: 30	0,38	0,62	0,00	1,00	0,81	0,00
9: 00	0,24	0,76	0,00	1,00	0,71	0,07
9: 30	0,19	0,88	0,07	1,00	0,58	0,33
10: 00	0,21	0,97	0,18	1,00	0,39	0,72
10: 30	0,26	1,00	0,26	0,97	0,07	1,00
11: 00	0,30	1,00	0,30	0,90	0,00	1,00
11: 30	0,32	1,00	0,32	0,86	0,00	1,00
12: 00	0,33	1,00	0,33	0,84	0,00	1,00

7.3 Validation procedure for the whole calculation method

7.3.1 General

Whole model validation considers the calculation of the operative temperature under cyclic conditions for several cases indicated below, and the comparisons of these values with those included in Table 21 and Table 22. According to the assumptions of 4.1, the operative temperature is calculated as the average of the internal air temperature and the mean radiant temperature. According to the assumptions of this document, the operative temperature is calculated as the average of the internal air temperature $\theta_{a,i}$ and the mean radiant temperature θ_{mr} of the internal surface of the room elements, calculated as:

$$\theta_{mr} = [\sum_{j=1}^{N}(\theta_{s,i}A)_j]/(\sum_{j=1}^{N}A_j) \tag{32}$$

where

N is the number of surfaces delimiting the internal space;

$\theta_{s,i}$ is the internal surface temperature;

A_j is the area of surface j.

The geometrical characteristics of the rooms are given in Table 12.

Table 12 - Room data

Component	Geometry A	Geometry B
Area (m^2):		
External opaque wall	6,58	3,08
Window	3,50	7,00
Partition wall (left)	15,40	15,40
(right)	15,40	15,40
(back)	10,08	10,08
Floor	19,80	19,80
Ceiling	19,80	19,80
Volume (m^3)	55,44	55,44

For determining the thermal capacity of the internal air, the following parameters are used:

— specific heat capacity of air: 1008 J/(kg·K)

— air density: 1,139 kg/m^3

7.3.2 Geometry for the test rooms

The various test cases refer to two different geometries (Figure 11 and Figure 12) located in two zones with three different types of envelope and ventilation.

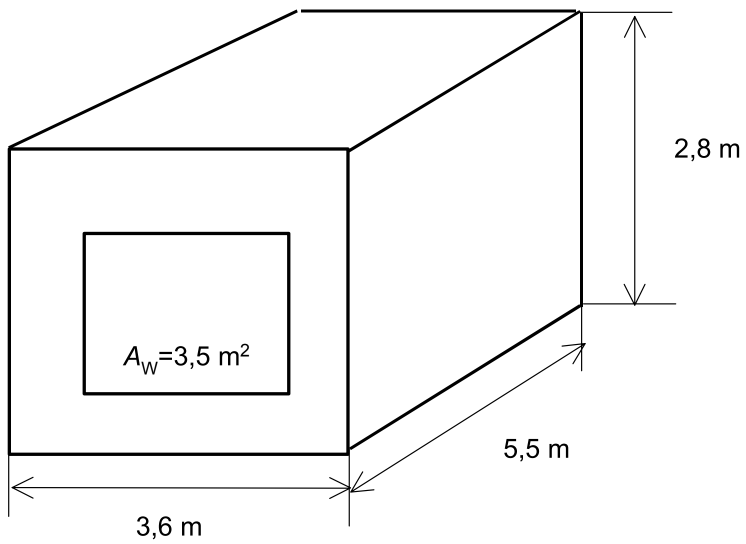


Figure 11 - Geometry A

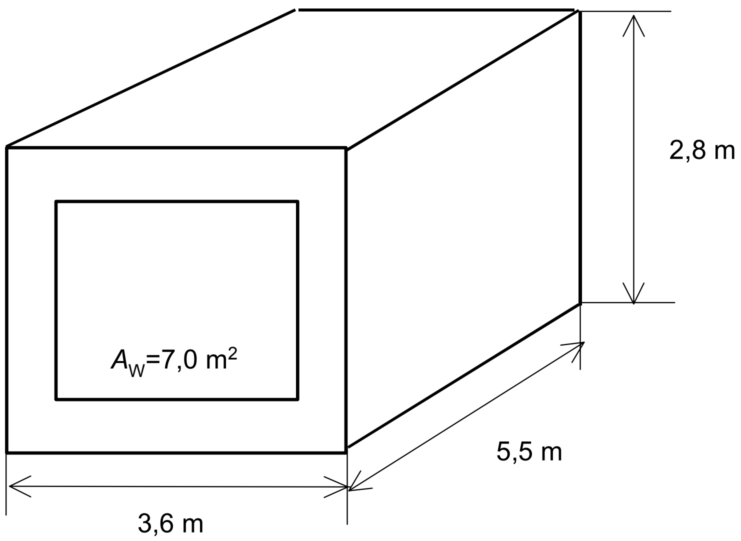


Figure 12 - Geometry B

7.3.3 Thermophysical properties of opaque walls

The thermophysical characteristics of the walls, ceiling and floor are given in Table 13.

7.3.4 Properties of glazing

The solar characteristics of the glass panes composing the glazing panes and the external shade are given in Table 14. This table shall be used for computer programs that need these data to calculate the values in Figure 13 and Figure 14.

Window specifications are given in Figures 13 and 14; the thermal resistance of window pane glass is assumed to be zero.

Table 13 - Thermophysical properties of the opaque components

Structure	s m	λ W/(m·K)	ρ kg/m^3	c_p kJ/(kg·K)
Type no.1 (external wall)				
Outer layer	0,115	0,99	1 800	0,85
Insulating layer	0,06	0,04	30	0,85
Masonry	0,175	0,79	1 600	0,85
Internal plastering	0,015	0,70	1 400	0,85
Type no.2 (internal wall)				
Gypsum plaster	0,012	0,21	900	0,85
Insulating layer	0,10	0,04	30	0,85
Gypsum plaster	0,012	0,21	900	0,85
Type no.3 (ceiling/floor)				
Plastic covering	0,004	0,23	1 500	1,5
Cement floor	0,06	1,40	2 000	0,85
Insulating layer	0,04	0,04	50	0,85
Concrete	0,18	2,10	2 400	0,85
Type no.4 (ceiling/floor)				
Plastic covering	0,004	0,23	1 500	1,5
Cement floor	0,06	1,40	2 000	0,85
Insulating layer	0,04	0,04	50	0,85
Concrete	0,18	2,10	2 400	0,85
Insulating layer	0,10	0,04	50	0,85
Acoustic board	0,02	0,06	400	0,84
Type no.5 (roof)				
External layer	0,004	0,23	1 500	1,3
Insulating layer	0,08	0,04	50	0,85
Concrete	0,20	2,1	2 400	0,85

Table 14 - Solar characteristics of the glazed element and the shade for all incident angles

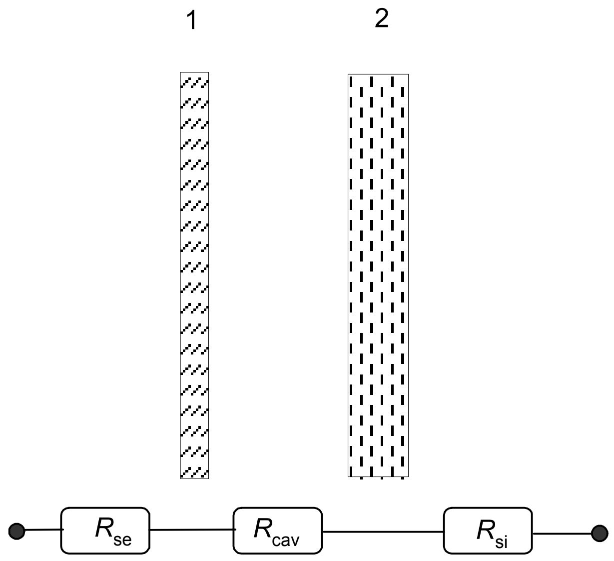
Component	τ_n	ρ_n
Pane	0,84	0,08
Shade	0,2	0,50

External, cavity and internal thermal resistances

R_{se} = 0,074 $m^2 \cdot K/ W$

R_{cav} = 0,08 $m^2 \cdot K/ W$

R_{si} = 0,125 $m^2 \cdot K/ W$



Key

1 External shade (or blind)
2 Pane

Figure 13 - Single pane window

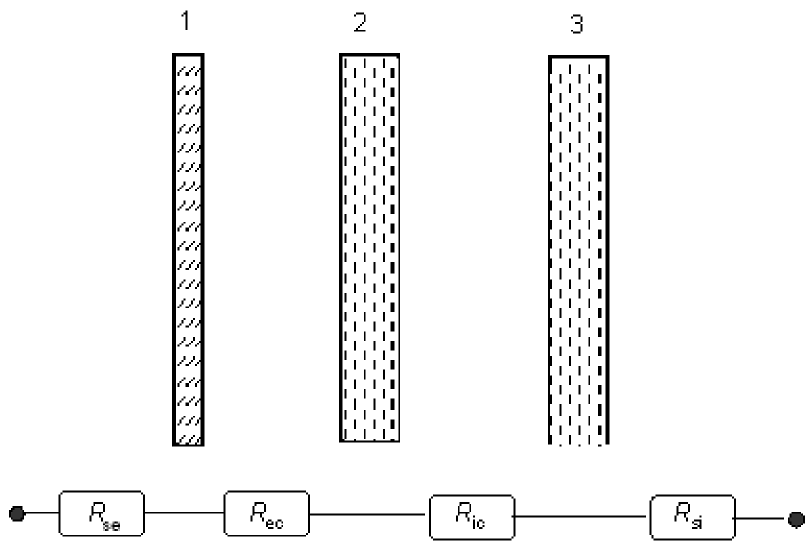
External, cavity and internal thermal resistances

R_{se} = 0,074 $m^2 \cdot K/ W$

R_{ec} = 0,080 $m^2 \cdot K/ W$

R_{ic} = 0,173 $m^2 \cdot K/ W$

R_{si} = 0,125 $m^2 \cdot K/ W$



Key

1 External shade (or blind)
2 External pane
3 Internal pane

Figure 14 - Double pane glazing with external shading device

7.3.5 Solar parameters

The solar parameters to be used are the following:

solar to air factor f_{sa} = 0,10;

solar loss factor f_{sl} = 0,00;

solar distribution factor:
floor $f_d = 0,5$;
ceiling $f_d = 0,1$;

total vertical walls (excluding windows) $f_d = 0,4$;

ceiling $f_d = 0,1$;

total vertical walls (excluding window) $f_d = 0,4$;

solar absorptance of all wall surfaces $\alpha_{sr} = 0,6$;

solar absorptance of the roof $\alpha_{sr} = 0,9$.

7.3.6 Boundary conditions

External convective heat transfer coefficient $h_{c,e} = 8,0$ W/(m^2·K);

Internal convective heat transfer coefficient $h_{c,i} = 2,5$ W/(m^2·K) (horizontal heat flow);

$h_{c,i} = 5,0$ W/(m^2·K) (upward heat flow);

$h_{c,i} = 0,7$ W/(m^2·K) (downward heat flow);

Radiative heat transfer coefficient $h_{r,e} = 5,5$ W/(m^2·K) (all surfaces).

(valid for $\varepsilon = 0,93$ and $T_{mr} = 303$ K)

The climatic data are given:

- solar radiation:
  Table 15a – Latitude 40° N is for geometry A;
  Table 15b – Latitude 52° N is for geometry B;

- external air temperature:
  Table 16a and Figure 15 for geometry A;
  Table 16b and Figure 16 for geometry B.

Table 15a - Solar radiation components for geometry A

Hour	Latitude 40° N					
	Horizontal			Vertical West		
	direct W/m²	direct W/m²	direct W/m²	direct W/m²	direct W/m²	direct W/m²
4	0	0	0	0	0	0
5	1	3	0	0	2	0
6	106	62	0	0	45	17
7	278	91	0	0	78	37
8	452	105	0	0	103	56
9	606	112	0	0	122	72
10	725	117	0	0	137	84
11	801	119	0	0	145	92
12	827	120	0	0	160	95
13	801	119	0	209	172	92
14	725	117	0	396	180	84
15	606	112	0	539	181	72
16	452	105	0	616	172	56
17	278	91	0	595	146	37
18	106	62	0	418	93	17
19	1	3	0	17	3	0
20	0	0	0	0	0	0

Table 15b - Solar radiation components for geometry B

Hour	Latitude 52° N					
	Horizontal			Vertical West		
	direct W/m²	direct W/m²	direct W/m²	direct W/m²	direct W/m²	direct W/m²
4	0	0	0	0	0	0
5	35	34	0	0	15	7
6	153	73	0	0	33	23
7	295	93	0	0	42	39
8	435	104	0	0	47	54
9	558	110	0	0	50	67
10	654	114	0	0	51	77
11	714	116	0	0	52	83
12	735	117	0	0	64	85
13	714	116	0	204	78	83
14	654	114	0	387	94	77
15	558	110	0	529	107	67
16	435	104	0	609	115	54
17	295	93	0	606	111	39
18	153	73	0	492	89	23
19	35	34	0	223	41	7
20	0	0	0	0	0	0

Table 16a - External air temperature for geometry A

Hour	θ_{ao} °C	Hour	θ_{ao} °C	Hour	θ_{ao} °C	Hour	θ_{ao} °C
1	23,6	7	22,8	13	32,7	19	29,9
2	23,0	8	23,9	14	33,6	20	28,4
3	22,5	9	25,8	15	34,0	21	27,0
4	22,1	10	27,3	16	33,6	22	25,8
5	22,0	11	29,3	17	32,8	23	24,9
6	22,2	12	31,2	18	31,5	24	24,2

Table 16b - External air temperature for geometry B

Hour	θ_{ao} °C	Hour	θ_{ao} °C	Hour	θ_{ao} °C	Hour	θ_{ao} °C
1	14,1	7	13,1	13	26,2	19	22,6
2	13,3	8	14,6	14	27,5	20	20,5
3	12,6	9	16,6	15	28,0	21	18,7
4	12,2	10	19,0	16	27,5	22	17,1
5	12,0	11	21,8	17	26,4	23	15,8
6	12,3	12	24,3	18	24,6	24	14,9

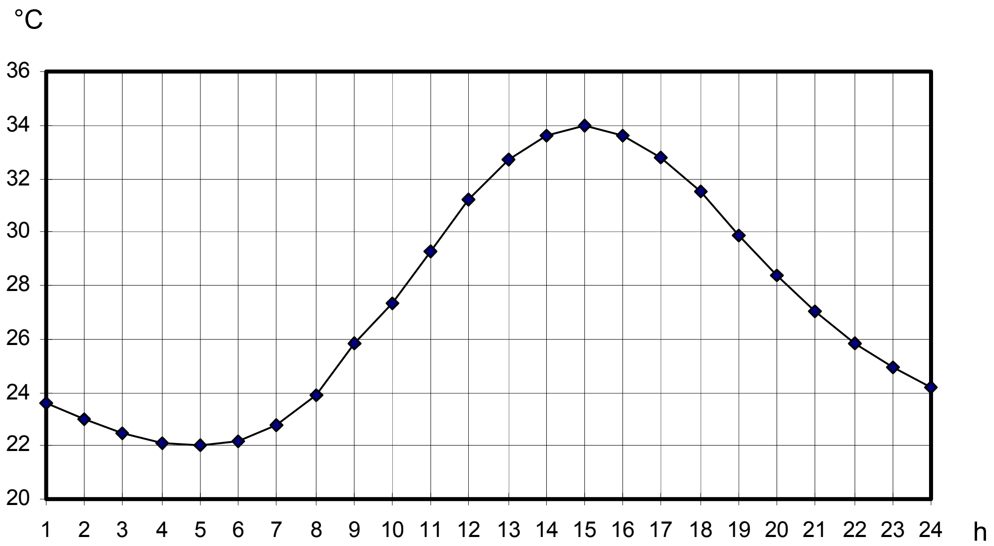


Figure 15 - External air temperature for latitude 40°, in °C

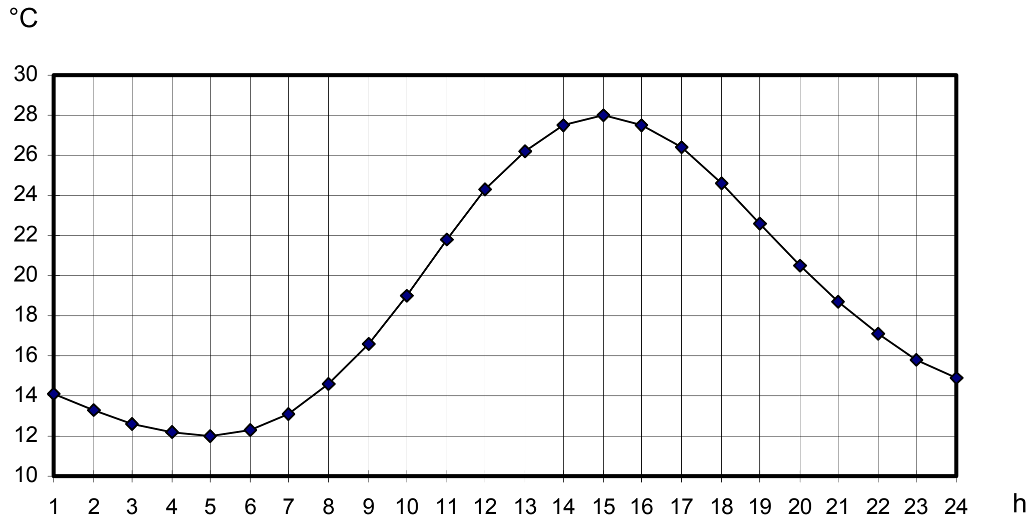


Figure 16 - External air temperature for latitude 52°, in °C

The values of the temperatures and solar radiation reported in the previous tables correspond to instantaneous values at each hour. If time intervals less than one hour are considered, the solar radiation at each time step shall be determined by linear interpolation between the previous and subsequent hour (see the example in Figure 17).

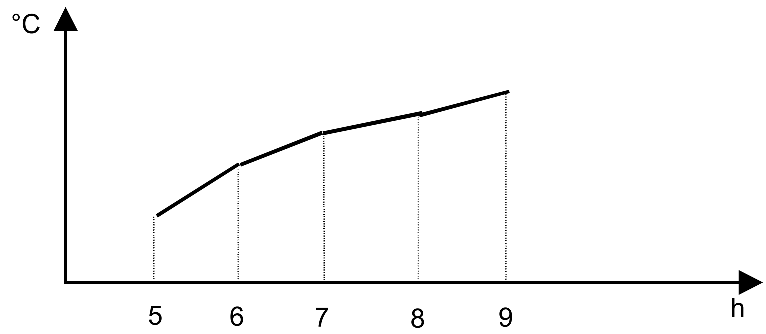


Figure 17 - Example of interpolation

7.3.7 Internal energy sources

The total heat flow rate due to internal sources expressed in watts per square metre of floor area is given in Table 17 and in Figure 18. The heat flow is transferred to the room by convection and radiation in equal proportions (50 % for each).

Table 17 - Total heat flow rate due to internal sources per floor area

Hour	Φ_i W/m^2	Hour	Φ_i W/m^2	Hour	Φ_i W/m^2	Hour	Φ_i W/m^2
0 to1	0	6 to 7	0	12 to 13	10	18 to19	15
1 to 2	0	7 to 8	1	13 to 14	10	19 to 20	15
2 to 3	0	8 to 9	1	14 to15	10	20 to 21	15
3 to 4	0	9 to10	1	15 to16	1	21 to 22	15
4 to 5	0	10 to11	1	16 to17	1	22 to 23	10
5 to 6	0	11 to 12	10	17 to 18	1	23 to 24	0

The daily total value of the internal gains is 117 Wh/m^2.

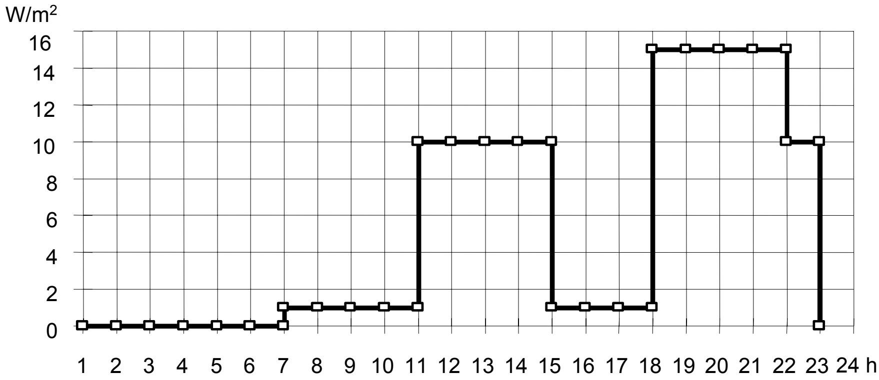


Figure 18 - Internal gains

7.3.8 Ventilation

Three different ventilation patterns are considered (Table 18):

a) air changes/h equal to 1 h^{-1}, constant;

b) air changes/h equal to 0,5 h^{-1}, constant from 6:00 a.m. to 18:00 p.m. (inclusive) - other hours air changes/h equal to 10 h^{-1}, constant;

c) air changes/h equal to 10 h^{-1}, constant.

Table 18 - Air changes per hour

Hour	a	b	c	Hour	a	b	c
0 to 1	1	10	10	12 to 13	1	0,5	10
1 to 2	1	10	10	13 to 14	1	0,5	10
2 to 3	1	10	10	14 to 15	1	0,5	10
3 to 4	1	10	10	15 to 16	1	0,5	10
4 to 5	1	10	10	16 to 17	1	0,5	10
5 to 6	1	10	10	17 to 18	1	0,5	10
6 to 7	1	0,5	10	18 to 19	1	10	10
7 to 8	1	0,5	10	19 to 20	1	10	10
8 to 9	1	0,5	10	20 to 21	1	10	10
9 to 10	1	0,5	10	21 to 22	1	10	10
10 to 11	1	0,5	10	22 to 23	1	10	10
11 to 12	1	0,5	10	23 to 24	1	10	10

7.3.9 Descriptions of the validation tests

The following tests are considered.

For geometry A (see Figure 11), three tests shall be carried out as shown in Table 19, where the types of elements are identified by the numbers used in Table 13 and the letters used in Figure 13.

Single pane glazing with external shade completely closed exposed to West.

Area of external opaque wall 6,58 m^2

Area of window 3,50 m^2

Table 19 - Test cases A

Test No.	External opaque wall	Glazing	Partition vertical wall to similar room	Ceiling to similar room	Floor to similar room	Roof
A.1	1	Single	2	4	4	-
A.2	1	Single	2	3	3	-
A.3	1	Single	2	5	3	5

For geometry B (see Figure 12), three tests shall be carried out as shown in Table 20, where the types of elements are identified by the numbers used in Table 13 and the letters used in Figure 15.

Double pane glazing with external shade completely closed exposed to West.

Area of external opaque wall 3,08 m^2

Area of window 7,00 m^2

Table 20 - Test cases B

Test No.	External opaque wall	Glazing	Partition vertical wall to similar room	Ceiling to similar room	Floor to similar room	Roof
B.1	1	double	2	4	4	-
B.2	1	double	2	3	3	-
B.3	1	double	2	5	3	5

7.3.10 Results of validations

For each test the following data, determined in cyclic conditions, shall be calculated:

— daily maximum value of the operative temperature $\theta_{op,max}$

— daily average value of the operative temperature $\theta_{op,av}$

— daily minimum value of the operative temperature $\theta_{op,min}$

For each case the comparison with the values reported in Table 21 for geometry A and Table 22 for geometry B, shall give a difference of less than 0,5 K.

Table 21 - Operative temperature for geometry A

Test	Ventilation	$\theta_{op,max}$ °C	$\theta_{op,ave}$ °C	$\theta_{op,min}$ °C
	a)	38,7	35,9	33,6
A.1	b)	34,1	29,4	25,5
	c)	33,5	29,0	25,4
	a)	37,6	35,9	34,4
A.2	b)	32,2	29,5	26,5
	c)	32,4	29,1	26,4
	a)	40,8	38,7	37,1
A.3	b)	35,4	31,6	28,0
	c)	33,8	30,3	27,4

Table 22 - Operative temperature for geometry B

Test	Ventilation	$\theta_{op,max}$ °C	$\theta_{op,ave}$ °C	$\theta_{op,min}$ °C
	a)	35,9	30,5	27,2
B.1	b)	29,9	21,3	16,4
	c)	28,1	21,5	16,2
	a)	33,7	30,8	28,5
B.2	b)	26,7	22,2	17,9
	c)	26,4	21,7	17,7
	a)	36,0	32,7	30,3
B.3	b)	29,6	24,2	19,2
	c)	27,7	22,7	18,6

NOTE More detailed information is given in Annex K.

Annex A
(informative)

Example of solution technique

A.1 Introduction

This annex gives a numerical method for the calculation of the internal temperature of a room in the absence of any cooling plant based on an implicit empirical technique.

A.2 Basic assumptions for the calculation method

The method is based on the assumptions given in 4.1 of this document. Further assumptions on the physical phenomena and on the solution techniques, in order to obtain a simpler energy balance, are introduced. They are:

- energy flows by long-wave radiation to/from each surface are calculated by reference to the effective mean radiant temperatures of the other surfaces, the latter being calculated by area-weighting. The fictitious Mean Radiant Temperature Node is considered as the central node of the network of all long-wave radiative heat transfers;

- the air temperature difference influencing the air flow by infiltration and/or ventilation is assumed to be constant during the time interval.

A.3 Calculation procedure

A.3.1 General

The method operates as follows :

a) it subdivides the system into nodes placed at pre-selected points of interest throughout the system to be simulated;

b) for each node in turn and in terms of all surrounding nodes which are in thermal contact, the governing differential equation is replaced by an implicit numerical approximation, based on the fact that half the sum of the net heat flow rate at the start and finish of any finite time increment is equated with the total rate of change in internal energy in the region represented by the node in question;

c) the resulting set of equations (one for each node) is solved simultaneously at each time step.

A.3.2 Evaluation of the temperature of each enclosure component

A.3.2.1 General

According to Clause 4 sets of equations are written for all nodes characterizing the component. For an opaque wall, in which thermal storage effects occur, a subdivision according to the finite difference technique is introduced. To achieve more accurate computations, homogeneous elements can be subdivided and specified in parts. For transparent components, in which thermal storage effects can be neglected, nodes are chosen according to the characteristics of the component.

For each node (from n = 1, at the external surface, to $n = N$, at the internal surface) the equation, deriving from the thermal balance at time $t + \Delta t$, is written in the general form:

$$K_{1,n}\theta_{n-1,t+\Delta t} + K_{2,n}\theta_{n,t+\Delta t} + K_{3,n}\theta_{n+1,t+\Delta t} = D_n \qquad \text{(A.1)}$$

where factors K depend on the characteristics of the element, and D depends on the state of the system. The temperature of node n at time $t + \Delta t$ is calculated as:

$$\theta_{N,t+\Delta t} = C_N \tag{A.2}$$

$$\theta_{n,t+\Delta t} = C_n - \frac{K_{3,n}\,\theta_{n+1,t+\Delta t}}{B_n} \tag{A.3}$$

where the coefficients C_n (n = 1, N), B_n (n = 1, N) and D_n (n = 1, N) are determined as follows:

$$B_n = K_{2,n} - \frac{K_{1,n}K_{3,n-1}}{B_{n-1}} \tag{A.4}$$

with $B_1 = K_{2,n}$

$$C_n = \frac{D_n - K_{1,n}C_{n-1}}{B_n} \tag{A.5}$$

with $C_1 = \dfrac{D_1}{K_{2,1}}$

The values B_n and C_n are determined with increasing subscript n for all nodes. The required temperature distribution is computed with decreasing subscript n. The nodes at the external surface (n = 1) and the internal surface (n = N) are included within the wall.

A.3.2.2 Coefficients in Equation (A.1)

A.3.2.2.1 Opaque components

With respect to the conflicting requirements imposed by the considerations of good accuracy as well as high speed and low cost of computation, the subdivision of the component into different nodes should be carefully considered. Three nodes per element (layer), with one node placed at each boundary of the element and one at the centre plane, is the minimum nodal representation. To achieve more accurate computations, homogeneous elements can be divided into more parts as follows:

a) limit the distance between two subsequent nodes (Δx) as a function of the thermal conductivity of the layer:

if λ < 0,5 W/(m·K) Δx_{max} = 0,02 m;

if λ > 0,5 W/(m·K) Δx_{max} = 0,03 m;

b) the time increment Δt is determined as: $\Delta t = \Delta x^2/(2a)$

where $a = \lambda/(\rho c)$ is the thermal diffusivity, in m²/s.

c) for multilayered constructions, the distance between two subsequent nodes should be calculated for each layer separately. The time increment to be used for the calculation corresponds to the maximum calculated for each layer.

NOTE This nodal representation scheme implies that multilayered construction nodes situated at the boundary between different homogeneous elements represent mixed thermal property regions. Furthermore the internal and external surface nodes, which represent convective, radiative and conductive heat exchanges, have associated thermal capacities equal to one half of the surface layer capacity.

For each node type of interest the factors K and D are derived as follows:

a) Node at the centre of homogeneous elements

$K_{1,n}$ = - F_n $K_{2,n}$ = 2 (1 + F_n) $K_{3,n}$ = -F_n (A.6)

$$D_n = \mathrm{F_n}\,\theta_{n-1,t} + 2\ (1 - \mathrm{F}_n)\theta_{n,t} - \mathrm{F_n}\,\theta_{n+1,t} + B_n\,(q_{n,t+\Delta t} + q_{n,t}) \quad (A.7)$$

with

$$\mathrm{F}_n = \frac{\lambda_n\ \Delta\ t}{\rho_n c_n \Delta\, x^2}$$

(A.8)

$$\mathrm{B}_n = \frac{\Delta t}{\rho_n c_n\, \Delta\, x_n}$$

where

Δt is the time increment;

c is the specific heat of the region;

Δx is the thickness of the region represented by node n;

λ is the thermal conductivity of the element;

ρ is the density of the element;

q is the density of heat flow rate generated within the region.

b) Node situated at the boundary between two different homogeneous elements

$K_{1,n}$ = - F_{n-1} ; $K_{2,n}$ = (2 + F_{n-1} + F_{n+1}) ; $K_{3,n}$ = - F_{n+1} (A.9)

$$D_n = \mathrm{F}_{n-1}\theta_{n-1,t} + (2 - F_{n-1} - F_{n+1})\theta_{n,t} + F_n + 1\theta_{n+1,t} + B(q_{n,t+\Delta t} + q_{n,t}) \quad (A.10)$$

where suffices n-1 and n+1 refer to the homogeneous elements on either side of boundary node n.

$$F_{n-1} = \frac{\lambda_{n-1}\,\Delta t}{\Delta\, x_{n-1}[\rho c\,\Delta\, x/2)_{n-1} + (\rho c\,\Delta\, x/2)_{n+1}]}$$

(A.11)

$$F_{n+1} = \frac{\lambda_{n+1}\,\Delta t}{\Delta\, x_{n+1}[\rho\, c\,\Delta\, x/2)_{n-1} + (\rho\ c\,\Delta\, x/2)_{n+1}]}$$

$$B_n = \frac{\Delta t}{\Delta\, x_{n+1}\,[(\rho c\,\Delta\, x/2)_{n-1} + (\rho c\,\Delta\, x/2)_{n+1}]} \quad (A.12)$$

c) Node situated at the boundary between a homogeneous element and an air layer

The node n-1 is on the opposite side of the air layer.

$K_{1,n}$ = - F_a $K_{2,n}$ = (3/4+ F_a + F_n) $K_{3,n}$= 1/4 - F_a (A.13)

$$D_n = F_a\ \theta_{n-1,t} + (3/4 - \mathrm{F}_a - F_n)\theta_{1,t} + (1/4 + F_n)\theta_{n,t} + B_n(q_{n,t+\Delta t} + q_{n,t}) \quad \text{(A.14)}$$

with

$$F_\mathrm{a} = \frac{(\Lambda_\mathrm{a} + \Lambda_\mathrm{lr})\Delta t}{\rho_n c_n \Delta x} \qquad F_n = \frac{2\lambda_n \Delta t}{\rho_n c_n \Delta x^2} \qquad B_n = \frac{2h_t \Delta t}{\rho_n c_n \Delta x_n} \quad \text{(A.15)}$$

where

$\Delta x$is the distance between two subsequent nodes;

Λ_lr is the long wave radiative conductance of the cavity;

Λ_a is the air conductance;

Subscript l refers to air layer.

In the Equation (A.15) the air conductance is determined by using the values of Table 3 or Table 4 or for an unventilated air cavity, or by $q_{\mathrm{c,n}}/(\theta_{n\text{-}1} - \theta_n)$ (see annex B) for a ventilated air cavity; the radiative conductance of the cavity is calculated according to 4.5.4.3, Equation (29).

d) Node at the external surfaces of a room element (*n* = 1)

$$K_{1,1} = 0 \qquad K_{2,1} = 3/4 + F_1 + F_\mathrm{e} \qquad K_{3,1} = 1/4 - F_1 \quad \text{(A.16)}$$

$$D_1 = (3/4 - F_\mathrm{e} - \mathrm{F}_1)\theta_{1,t} + F_\mathrm{e}\ (\theta_{\mathrm{ae},t+\Delta t} + \theta_{\mathrm{ae},t}) + (1/4 + F_\mathrm{i})\theta_{2,t} + B_1(q_{1,t+\Delta t} + q_{1,t}) \quad \text{(A.17)}$$

with $F_\mathrm{e} = h_\mathrm{c,e}\ \Delta t/(\rho_1\ c_1\ \Delta x_1)$ and $q_1 = q_\mathrm{sre} - q_\mathrm{er}$

where

θ_ae is the air temperature of the adjacent space;

$h_\mathrm{c,e}$ is the external convective surface coefficient of heat transfer (see Equation 14);

q_sre is the density of total solar heat flow rate absorbed at the external surface;

q_er is the density of long wave radiative heat flow rate to the vault sky.

e) Node at the internal surfaces of a room element (*n* = *N*)

$$K_{1,N} = ¼ - F_\mathrm{N} \qquad K_{2,N} = (\ 3/4 + F_N + F\mathrm{i}) \qquad K_{3,N} = 0 \quad \text{(A.18)}$$

$$D_N = (1/4 + \mathrm{F}_N)\theta_{N-1,t} + (3/4 - F_e - F_N)\theta_{N,t} + F_\mathrm{e}\ (\theta_{\mathrm{ai},t+\Delta t} + \theta_{\mathrm{ai},t}) + B_N\ (q_{N,t+\Delta t} + q_{N,t}) \quad \text{(A.19)}$$

with $F_\mathrm{i} = h_\mathrm{c,i}\ \Delta t/(\rho_N\ c_N \Delta x_N)$ and $q_N = q_\mathrm{lri} - q_\mathrm{sri}$

where

θ_ai is the room air temperature;

$h_\mathrm{c,i}$ is the internal convective surface coefficient of heat transfer (see Table 1);

q_lri is the density of net heat flow rate due to long-wave radiation received by the internal surface (see Annex E);

q_sri is the density of heat flow rate due to short wave radiation absorbed by the internal surface (see Equation (19)).

A.3.2.2.2 Transparent components

The following equations refer to a transparent component formed by a double pane of glass. In this component, neglecting the thickness of each glass pane, the following nodes are present:

— node at the external pane (external surface);

— node at the internal pane (internal surface);

— node in the air cavity between the panes.

This situation can be extended to consider internal and external blinds. One single pane without any blind can be considered by assuming the value of the heat transfer coefficient between each pane very high.

a) Node (n = 1) located at the external surface

$$K_{1,1} = 0 \qquad K_{2,1} = h_{c,e} + \Lambda t \qquad K_{3,1} = - \Lambda t \tag{A.20}$$

$$D_1 = h_{c,e}\, \theta_{e,t+\Delta t} + (q_{sr,e} - q_{er})\, t+\Delta t \tag{A.21}$$

node (n = N) located at the internal surface

$$K_{1,N} = - \Lambda_t \qquad K_{2,N} = h_{c,i} + \Lambda_t \qquad K_{3,N} = 0 \tag{A.22}$$

$$D_N = h_{c,i}\, \theta_{a,t+1} + (q_{lr,i} + q_{sr,i})\, t+1 \tag{A.23}$$

where

$h_{c,e}$ is the external convective heat transfer coefficient (from Equation (13));

$h_{c,i}$ is the internal convective heat transfer coefficient (from Table 1);

Λ_t is the sum of the gas conductance and the radiative conductance of the cavity;

q_{sr} is the density of the short-wave radiant heat flow absorbed by the surface;

q_{lr} is the density of the long-wave radiant heat flow absorbed by the surface.

b) Node n located between and two air layers

$$K_{1,n} = - \Lambda_{t,n-1} \qquad K_{2,n} = \Lambda_{t,n+1} + \Lambda_{t,n-1} \qquad K_{3,n} = - \Lambda_{t,n+1} \tag{A.24}$$

$$D_n = q_{n,t+\Delta t} \tag{A.25}$$

where

Λ_t is the sum of the air conductance and the long-wave radiative conductance Λ;

q_n is the density of the radiant heat flow absorbed by the surface.

A.3.2.3 Air temperature

The air temperature is determined by solving the following equation:

$$\sum_{j=1}^{N} K_{1,j}\theta_{N,t+\Delta t,j} + K_{2,a}\theta_{a,t+\Delta t} = D_a \tag{A.26}$$

with

$$K_{2a} = 2 + [\sum_{j=1}^{N}(h_{c,i,j}A_j) + (c_a\, q_n + c_a q_m)]/C_a$$

$$\begin{aligned} D_a &= [\sum_{j=1}^{N}(h_{c,i,j}\, A_j\, \theta_{N,j})/C_{a,j} + (\Phi_{a,t+\Delta t} + \Phi_{a,t})/C_{a,j}] + \\ &+ \left\{ 2 - [(\sum_{j=1}^{N}(h_{c,i,j}A_j) + c_a\, q_n + c_a\, q_m)/C_{a,j}] \right\} \theta_{a,i,t} + \\ &+ [c_a\, q_n\, (\theta_{ae,t+\Delta t} + \theta_{ae,t}) + c_a\, q_m(\theta_{v,t+\Delta t} + \theta_{v,t})/C_{a,j}] \end{aligned} \qquad \text{(A.27)}$$

where

- N is the number of enclosure surfaces;
- $h_{c,i,j}$ is the convection coefficient at each internal surface j;
- A is the area of each surface at each internal surface j;
- q_n is the mass air flow rate by natural ventilation;
- q_m is the mass forced air flow rate by mechanical ventilation;
- c_a is the specific heat capacity of the air;
- $\theta_{a,e}$ is the external air temperature;
- $\theta_{a,i}$ is the internal air temperature;
- θ_v is the temperature of the mechanically supplied air;

$$C_a = \frac{\rho_a c_a V}{\Delta t} \qquad \text{(A.28)}$$

where

- ρ_a is the density of air;
- V is the enclosure volume;
- Δt is the time increment.

The terms Φ_a are summations of the convective energy exchanges with the enclosure air at any time t and $t + \Delta t$. In general form this is given by:

$$\Phi_a = \Phi_{i,c} + \Phi_{air} + \Phi_{sa} \qquad \text{(A.29)}$$

where

- $\Phi_{i,c}$ is the heat flow rate due to the convective component of internal gains;
- Φ_{air} is the heat flow rate due to the air passing through internal air cavities;
- Φ_{sa} is the heat flow rate due to the convective part of solar radiation flow entering through the window.

The heat flow rate due to the convective component of internal gains acting at the enclosure air point (room air) is given by:

$$\Phi_{i,c} = \sum_{j=1}^{m} \Phi_{i,j}\, f_{i,c,j} \qquad \text{(A.30)}$$

where

m is the number of different sources;

Φ_{i} is the heat flow rate due to each internal source j;

$f_{\mathrm{i,c}}$ is the fraction of the total heat flow rate due to each internal source j.

The data of internal gains in annex H can be considered.

The mass air flow by natural ventilation can be calculated according to the procedure in annex H.

The convective part of the short-wave radiation entering through the glazing system, is given by Equation (22):

$$\Phi_{\mathrm{sra}} = f_{\mathrm{sa}}\ (1 - f_{\mathrm{lf}})\ (\Phi_{\mathrm{sr,D}} + \Phi_{\mathrm{sr,d}}) \qquad \text{(A.31)}$$

A.4 Room thermal balance

Equation (A.1), written for all internal surfaces, with Equation (A.26) written for the internal air, are simultaneously solved. From the solution of the system equations, the internal surface temperature of each enclosure component and the internal air temperature are determined. The required temperature distribution within each component is determined by using Equation (A.2) with decreasing subscript n.

Annex B
(informative)

Convective heat transfer through ventilated air layer

B.1 Introduction

The procedure presented in this annex allows the user to determine the convective heat transfer through a vertical and externally ventilated air layer. An example is the air gap between a glass pane and a curtain where openings are present around the curtain and the curtain itself may be partially permeable to the air flow.

B.2 Convective heat transfer for a vertical air layer

For a vertical ventilated air layer the convective heat transfer is:

— with the environment due to the air flow through the air cavity;

— with the boundary surfaces.

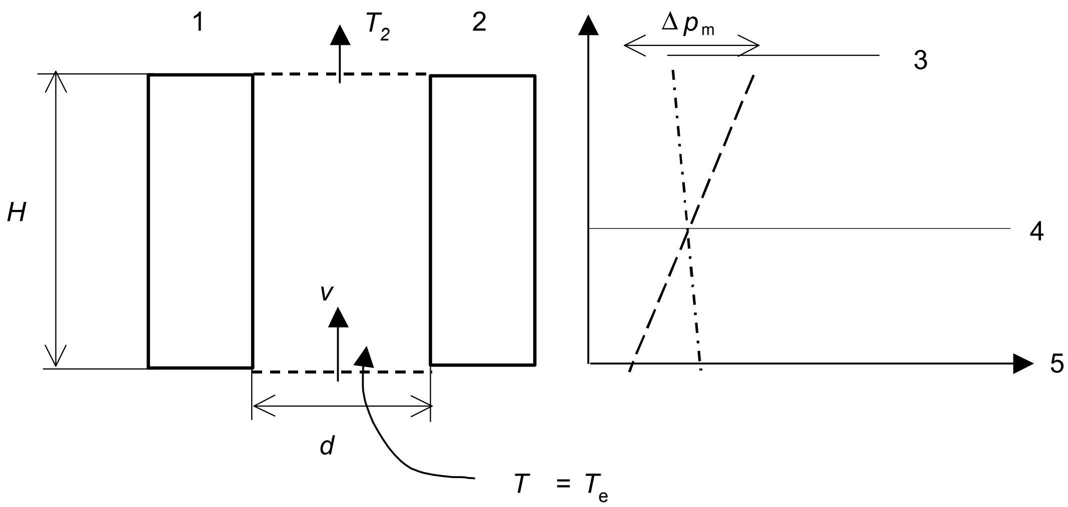
These require the evaluation of the air velocity within the cavity.

B 2.1 General

B 2.1.1 Evaluation of the outlet velocity of the vertical air layer

The mean velocity of the air in the space is principally caused by the stack effect. Convection caused by wind pressure is neglected.

The mean air velocity depends on the driving pressure difference, Δp_m, and on the total pressure loss in the ventilated air space itself (Figure B.1).



Key

1 Layer n - 1
2 Layer n
3 Top
4 Neutral plane
5 Bottom

Figure B.1 - Example of vertical air layer: the height of the neutral plane depends on the flow resistances of the inlet and outlet openings

The driving pressure difference results from the temperature difference between the air layer and the air of the environment. It is written as:

$$\Delta p_{\mathrm{m}} = \rho_0 \ g \ T_0 \, H (T_{\mathrm{eq}} - T_{\mathrm{ext}}) / (T_{\mathrm{eq}} \ T_{\mathrm{ext}}) \tag{B.1}$$

with

$$T_{\mathrm{eq}} = T_{\mathrm{av}} - E_{\mathrm{r}} (T_{\mathrm{av}} - T_{\mathrm{ext}})(1 - \mathrm{e}^{-1/E_{\mathrm{r}}}) \tag{B.2}$$

$$T_{\mathrm{av}} = (T_{n-1} + T_n)/2 \tag{B.3}$$

$$E_{\mathrm{r}} = [(\rho \ c_p \ d_{\mathrm{a}})/(2 \ h_{\mathrm{a}} \ H)] \ v \tag{B.4}$$

$$h_{\mathrm{a}} = 2 \ h_{\mathrm{g}} + c_{\mathrm{v}} \ v \tag{B.5}$$

where

Δp_{m}	is the driving pressure difference, in Pa;
ρ_0	is the density of the air at the temperature T_0, in kg/m^3;
H	is the height of the element, in m;
T_0	is the reference temperature;
T_{eq}	is the equivalent temperature of the air in the layer, in K;
T_{e}	is the temperature of the environment, external or internal air, in K;
T_{av}	is the average of temperatures of the surfaces delimiting the air layer, in K;
$T_{n\text{-}1}$, T_n	are the temperature of the layer n - 1 and n;
E_{r}	is a ventilation parameter;
c_p	is the specific heat of the air at the constant pressure, in J/(kg·K);
d	is the width of the air layer between the delimiting surfaces, in m;
v	is the mean velocity in the ventilated air layer, in m/s;
h_{g}	is the convective heat transfer coefficient for closed spaces, in W/(m^2·K);
c_v	is the velocity coefficient, in W·s/(m^3·K);
g	is the acceleration due to gravity (constant) 9,81 m/s^2.

The coefficient h_{g} can be calculated with the following equation:

$$h_{\mathrm{g}} = [\lambda \ A \ (Ra)^n]/s \tag{B.6}$$

where, for vertical spaces: A = 0,035; n = 0,38 and the Rayleigh number is:

$$Ra = \frac{g \, s^3 \, c_p \ \rho^2 \left| T_j - T_{j+1} \right|}{\mu \, \lambda T_{\mathrm{av}}} \tag{B.7}$$

where

μ	is the dynamic viscosity of the gas in the air layer, in kg/(m·s);
λ	is the thermal conductivity of the gas in the air layer, in W/(m·K);
ρ	is the density of the gas in the air layer.

The velocity coefficient c_v can be assumed equal to 4 W·s/(m^3·K).

The total pressure loss Δp_T in the ventilated air layer is calculated as :

$$\Delta p_{\mathrm{T}} = \Delta p_v + \Delta p_{\mathrm{d}} + \Delta p_{\mathrm{io}} \tag{B.8}$$

where

Δp_v is the kinetic pressure loss, in Pa;

Δp_d is the pressure loss due to the steady laminar flow, in Pa;

Δp_{io} is the pressure loss in the top and bottom openings, in Pa.

The terms in Equation (B.8) are given by:

$$\Delta p_v = \frac{1}{2}\rho\ v^2 \tag{B.9}$$

$$\Delta p_{\mathrm{d}} = 8\ \mu\ \frac{H}{s^2}\ v \tag{B.10}$$

$$\Delta p_{\mathrm{io}} = \frac{1}{2}\ \rho\ v^2\ (Z_1 + Z_2) \tag{B.11}$$

where

μ is the dynamic viscosity of the air, in kg/(m·s);

v is the velocity of the air within the cavity, in m/s;

H is the height of the cavity, in m;

d is the width of the cavity, in m;

Z_1, Z_2 are parameters relevant for the top and bottom openings.

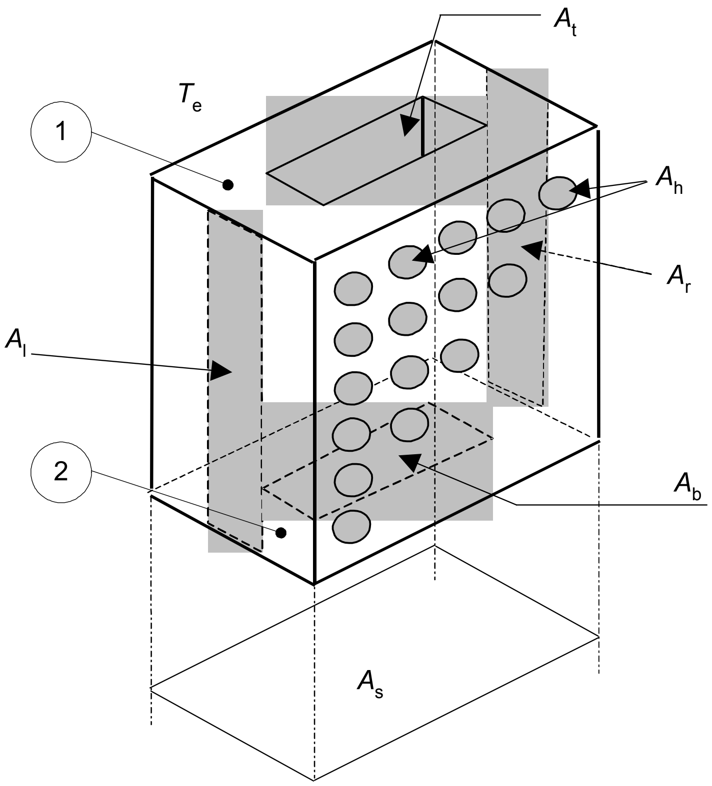
The parameters Z_1 and Z_2, according to Figure B.2, are given by:

$$Z_1 = \left(\frac{A_{\mathrm{s}}}{0{,}6\ A'_1} - 1\right)^2 \tag{B.12}$$

$$Z_2 = \left(\frac{A_{\mathrm{s}}}{0{,}6 A'_2} - 1\right)^2 \tag{B.13}$$

$$A'_1 = A_{\mathrm{b}} + \frac{A_{\mathrm{l}} + A_{\mathrm{r}} + A_{\mathrm{h}}}{4} \tag{B.14}$$

$$A'_2 = A_{\mathrm{t}} + \frac{A_{\mathrm{l}} + A_{\mathrm{r}} + A_{\mathrm{h}}}{4} \tag{B.15}$$



Key

1 Top
2 Bottom

Figure B.2 - Example of ventilated cavity

where

A_s the projected area of the considered system, in m^2;

A_t area of the top opening, in m^2;

A_b area of the bottom opening, in m^2;

A_l area of the left side opening, in m^2;

A_r area of the right side opening, in m^2;

A_h total area of the homogeneous distributed holes in the blind (only one side), in m^2.

The velocity v in the ventilated air layer is obtained by solving the following:

$$\Delta p_m = \Delta p_T \tag{B.16}$$

B.2.1.2 Evaluation of the convective heat transfers for a vertical air layer

The convective heat flow rates Φ are calculated as:

— due to the air motion:

$$\Phi_V = m_V \; c_p \; (T_{out} - T_{in}) \tag{B.17}$$

where

m_V is the mass air flow through the air layer, in kg/s;

T_{out} is the temperature of the air leaving the layer, in K;

T_{in} is the temperature of the air entering the air layer, in K.

with

$$q_a = r \, v \, A_a \tag{B.18}$$

where A_a is the area of the cross section of the air flow.

$$T_{out} = T_{av} - \left[T_{av} - T_{eq}\right] e^{-1/E_r} \tag{B.19}$$

— due to the convective heat exchanges with the surfaces n-1 and n delimiting the air layer:

$$\Phi_{c,n-1} = h_c \; A_c (T_{n-1} - T_{eq}) \tag{B.20}$$

$$\Phi_{c,n} = h_c \; A_c (T_n - T_{eq}) \tag{B.21}$$

where A_c is the area of the adjacent surfaces.

B.3 Convective heat transfer for an external horizontal air layer

B.3.1 General

A similar calculation for an external horizontal air layer is made (Figure B.3). In this case, the driving pressure difference Δp_w derives from the wind velocity as follows:

$$\Delta p_w = \Delta C_{pr} \frac{1}{2} \rho \, v^2 \tag{B.22}$$

where

ΔC_{pr} is the difference in the pressure coefficients between the inlet and the outlet of the air layer;

v is the wind velocity.

ΔC_{pr} assumes values changing from 0 (no openings perpendicular to the wind direction) to 1.

The total pressure losses Δp_T are calculated according to Equation (B.8). The velocity v in the ventilated air layer is obtained by solving the following:

$$\Delta p_w = \Delta p_T \tag{B.23}$$

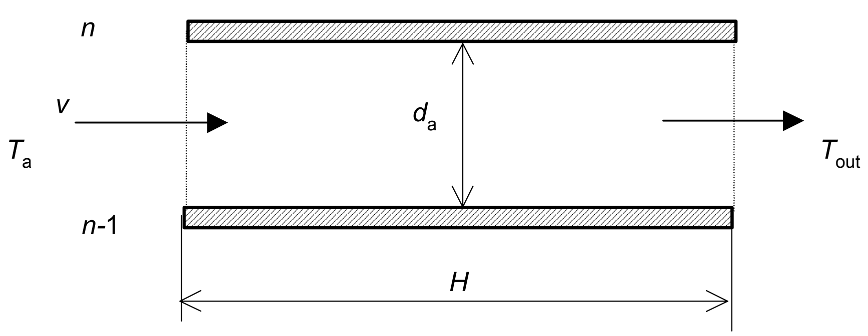


Figure B.3 - Example of horizontal air layer

B.3.2 Evaluation of the convective heat transfer for horizontal air layer

The convection heat transfers are calculated as in B.2.1.2:

— due to the air motion Φ_v, according to Equation (B.17);

— due to the convective heat exchanges with the surfaces n - 1 and n delimiting the air layer, Φ_c, according to Equation (B.20) and Equation (B.21).

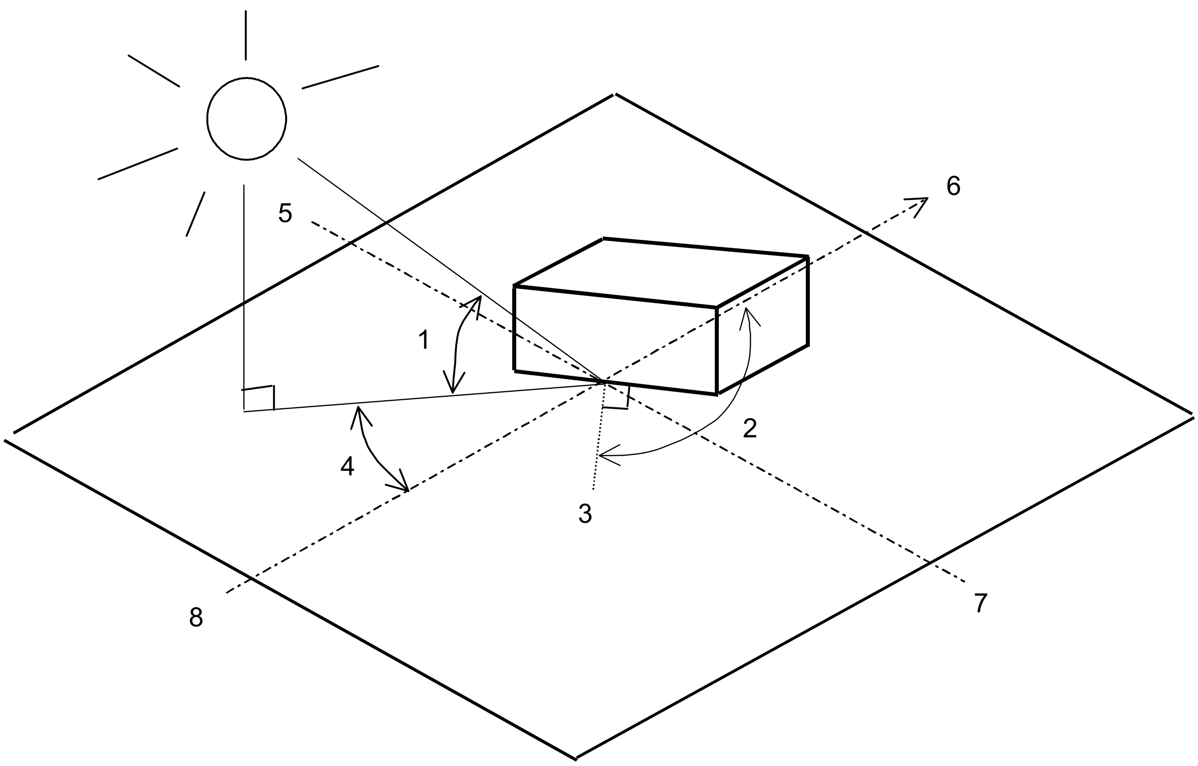
Annex C
(informative)

Shading due to overhangs and side fins

C.1 Introduction

This annex includes a procedure for determining the shaded area of a building element when obstructions due to an overhang and side fin are present.

The equations recognize wall solar azimuth angle γ and solar altitude β as known angles (Figure C.1). Other dimensions for a window of length *L* and height *H* area shown in Figure C.2 for an overhang and Figure C.3 for a side fin. Various shapes of shadows on a window are shown in Figure C.4. Shadows for an overhang, for an overhang with a vertical end projection, and for side fins, are separated to simplify the equations and their calculation procedure. After all types of shadows have been established, the procedure will then add up all the shadow areas for use in the final solar contribution.



Key

1 Solar altitude angle β
2 Wall azimuth angle ω
3 Normal to vertical surface
4 Solar azimuth angle γ
5 West
6 North
7 East
8 South

Figure C.1 - Solar azimuth angle γ and solar altitude β

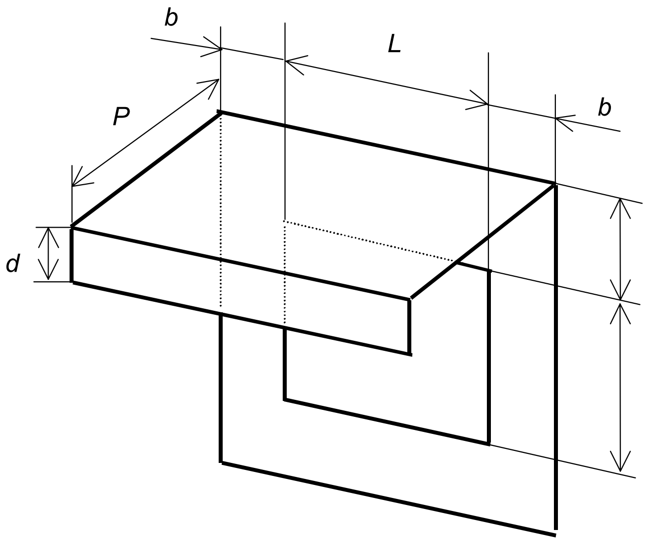


Figure C.2 - Overhang, main dimensions

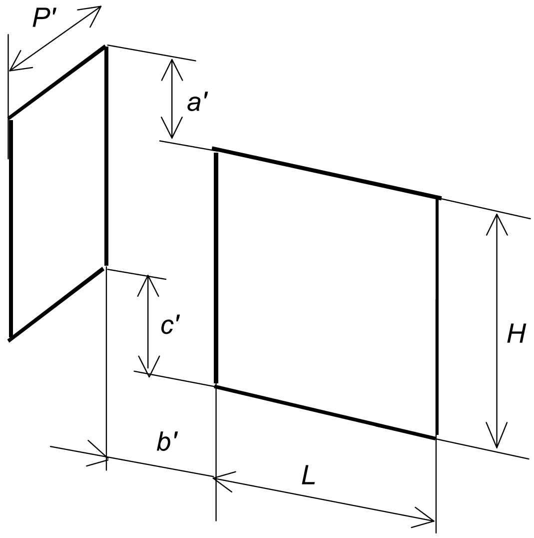


Figure C.3 - Side fin, main dimensions

In this annex the following calculations are introduced:

a) calculation path for overhang;

b) calculation path for vertical projection and the end of the overhang;

c) calculation path for side fin;

d) calculation path for sunlit area due to short side fin.

C.2 Calculation path for overhang

The width of the overhang is assumed to be equal to or greater than that of the window. The overhang can be located at the head of the window, or at distance a above the head of the window. Figure C.4 shows a typical shadow cast by an overhang. Aside from the fixed physical dimensions of the window arrangement *L*, *H*, *a* and *b*, all equations are expressed in terms of the shadow depth *T* and the shadow offset *M* cast by overhang, where:

$$T = P\frac{\tan\beta}{\cos\gamma} \qquad \text{(C.1)}$$
$$M = P\tan\gamma$$

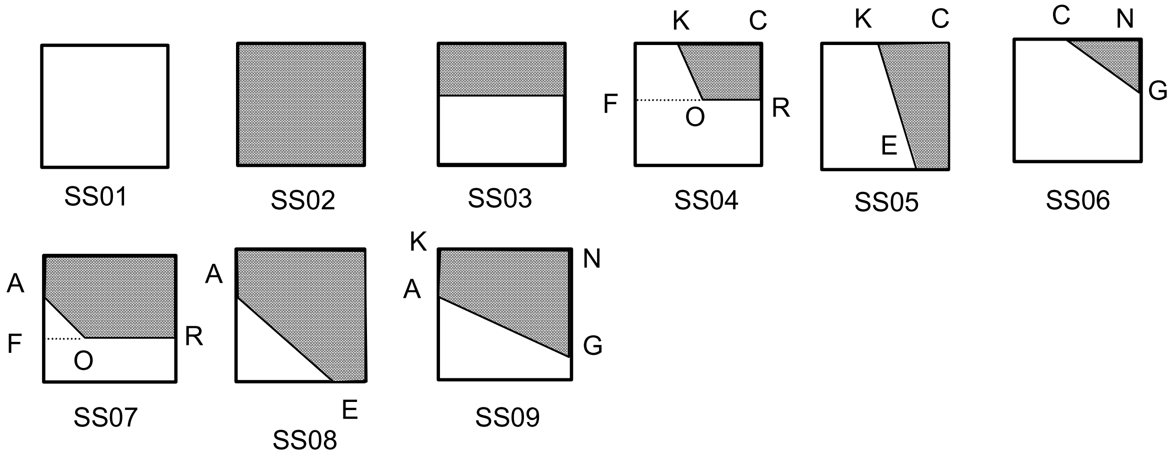


Figure C.4 - Shadow shapes of overhangs

The areas of the shadow shapes cast by an overhang, reported in Figure C.4 from SS01 to SS09, are as follows:

$$A_{01} = 0$$

$$A_{02} = H\ L$$

$$A_{03} = L\ (T - a)$$

$$A_{04} = (T - a)[L + b - \frac{M(1 + \frac{a}{T})}{2}]$$

$$A_{05} = H\left\{L - [(a + \frac{H}{2})\ \frac{M}{T}] + b\right\}$$

$$A_{06} = \left\{[(L + b)\ \frac{T}{M}] - a\right\}^2\ (\frac{M}{2\ T}) \qquad \text{(C.2)}$$

$$A_{07} = [(T - \mathrm{a})\ L] - \left[(M - b)^2\ \frac{T}{2\ M}\right]$$

$$A_{08} = H\ L - \left\{\left[(H + a)\ \frac{M}{T}\right] - b\right\}^2\ \frac{T}{2\ M}$$

$$A_{09} = L\ [(b + \frac{L}{2})\ \frac{T}{M} - a]$$

C.3 Calculation path for vertical projection at the end of the overhang

The width of the vertical projection is assumed to be equal to that of the overhangs. The area equations for the shadow shapes cast by a vertical projection at the end of an overhang, showed in Figure C.5 from SSV1 to SSV9, are as follows:

$$A_{v1} = 0$$
$$A_{v2} = H L$$
$$A_{v3} = L\ (T + d - a)$$
$$A_{v4} = H\ \ (L + b - M)$$
$$A_{v5} = (T + d - a)(L + b - M)$$
$$A_{v6} = L\ \ d \qquad \text{(C.3)}$$
$$A_{v7} = L\ \ (H + a - T)$$
$$A_{v8} = (L + b - M)\ \ d$$
$$A_{v9} = (L + b - M)\ (H + a - T)$$

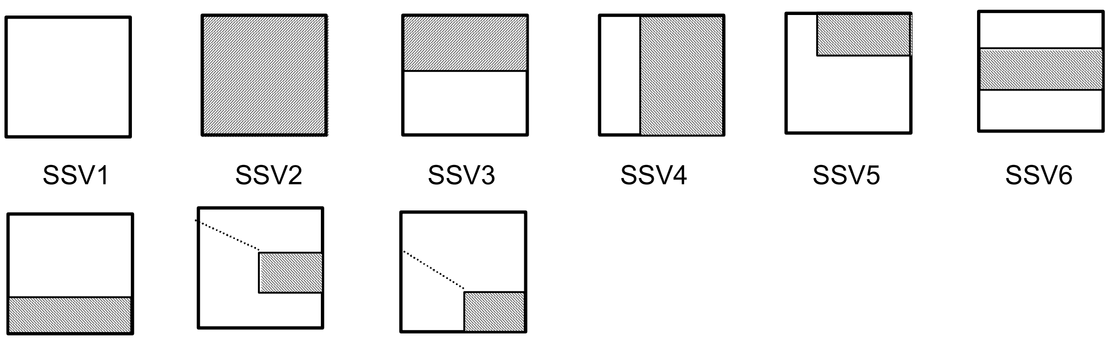


Figure C.5 - Shadow shapes of vertical projections at the end of overhangs

C.4 Calculation path for side fin

In order to include shadows cast by a sun hood (external shading device consisting of an overhang and side fins covering only the upper part of the window) or by the side of a balcony, fins are assumed to have any length greater than, less than or equal to the height of a window. They can be located at the window jamb, or a distance *b*' away from window jamb (Figure C.3). Dimension a' is considered positive when the fin extends above the window and negative when it stops below the head of the window. Dimension c' is considered to be zero in this part of calculation. Any sunlit area, resulting from a short fin that stops above the window sill level, will be deducted in the fourth part of the calculation. The calculation path for the side fin is very similar to that of the overhang. The area equations for the shadow shapes cast by the side fins, shown in Figure C.6 from SSF1 to SSF9, are the following:

$$
\begin{aligned}
A_{F1} &= 0 \\
A_{F2} &= H\ L \\
A_{F3} &= H\ (M'-b') \\
A_{F4} &= (M'-b')\ \left\{(H+a') - \left[\frac{T'\ (1+\frac{b'}{M'})}{2}\right]\right\} \\
A_{F5} &= L\ \left\{H - \left[(b'+\frac{L}{2})\ T'/M'\right] + a'\right\} \\
A_{F6} &= \left\{\left[(H+a')M'/T'\right] - b'\right\}^2\ \left(\frac{T'}{2M'}\right) \\
A_{F7} &= \left[(M'-b')\ H\right] - \left[(T'-a')^2\ \frac{M'}{2T'}\right] \\
A_{F8} &= H\ L - \left\{\left[(L+b')\ \frac{T'}{M'}\right] - a'\right\}^2\ \frac{M'}{2T'} \\
A_{F9} &= H\ \left[(a'+\frac{H}{2})\ \frac{M'}{T'} - b'\right]
\end{aligned}
\tag{C.4}
$$

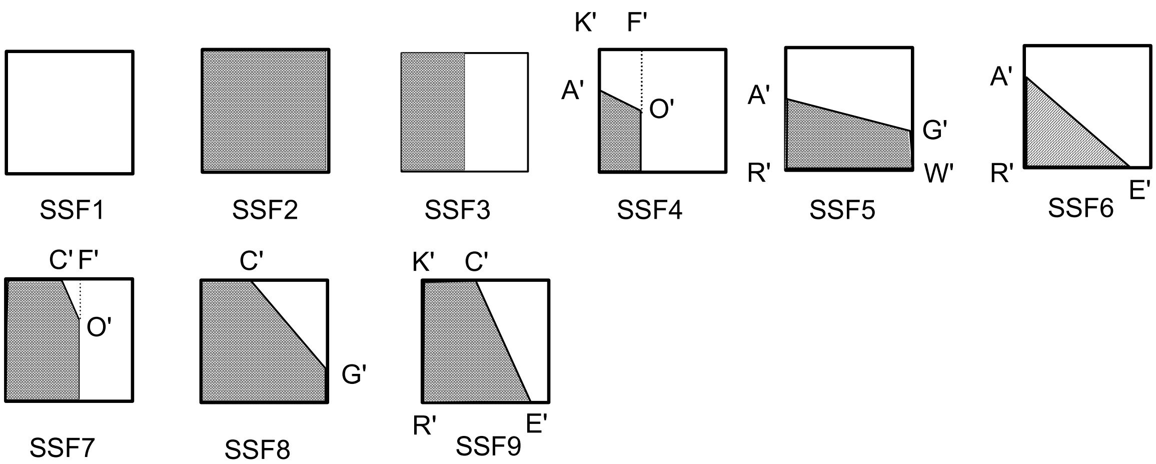


Figure C.6 - Shadow shapes of side fins

It should be noted that these equations have the same form as the equations for the overhang with variables *T*, *M*, *H*, *L*, *a* and *b* replaced by *M*', *T*', *L*, *H*, *a*' and *b*' respectively.

C.5 Calculation of the sunlit area due to short side fin

The shadow area equations for the various situations shown in Figure C.7 from SF1 to SF4 are the following:

$$A_{SF1} = 0$$

$$A_{SF2} = -(M'-b')\left\{c' - [T' \frac{(1+b'/M')}{2}]\right\}$$

$$A_{SF3} = -L \left\{c' - \left[(b'+L/2)\frac{T'}{M'}\right]\right\} \tag{C.5}$$

$$A_{SF4} = -[c' - (b' \frac{T'}{M'})]^2 \frac{M'}{2\,T'}$$

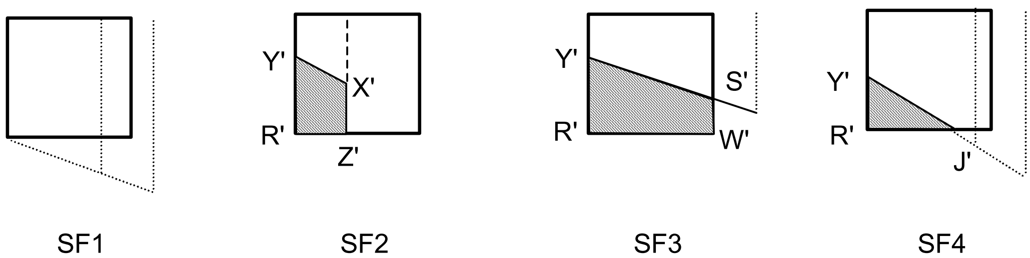


Figure C.7- Shapes of sunlit area due to short side fins

The minus signs indicate that these areas, if they occur, should be deducted from the previously accumulated shadow areas. The accumulation process is now necessary when different solutions are present. The algorithm, guided by a series of control statements, enters the shadow evaluation process and follows a single path through the flow chart. Along each of the path segments, the algorithm calculates and accumulates a particular set of shadow areas cast by overhang, vertical projection and side fin T' and shadow for use in the solar contribution for calculation. The shading devices considered in this algorithm are those of the most commonly used shapes. Other shapes can be introduced by expanding it.

C.6 External obstruction

When external obstructions are present, the following procedure can be used to determine the shaded area. This procedure is approximate and it is applicable when the wall or window is relatively small in comparison to the shading object. The wall or window is considered either completely shaded or completely in the sun. A partially shaded wall or window can be considered as either completely shaded or completely in the sun depending on the location of the wall reference point. Figure C.8 shows a typical wall-shading object relationship. It should be noted that the reference point may be located at any point on the wall or window. Locating the reference point at the top of the wall or window, as shown in the elevation in Figure C.8 is slightly conservative as compared to the reference point located at the centre of the wall or window. For the application of the procedure the following data are required:

— azimuth shaded angles α_1 ; α_2 (right + , left -);

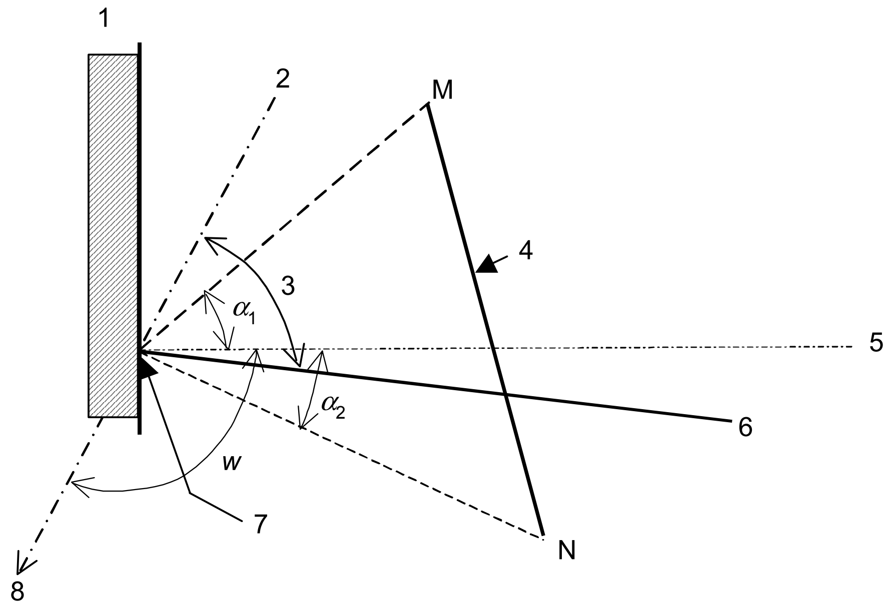
— altitude shadow limit angles δ_1 ; δ_2

— wall azimuth angle ω

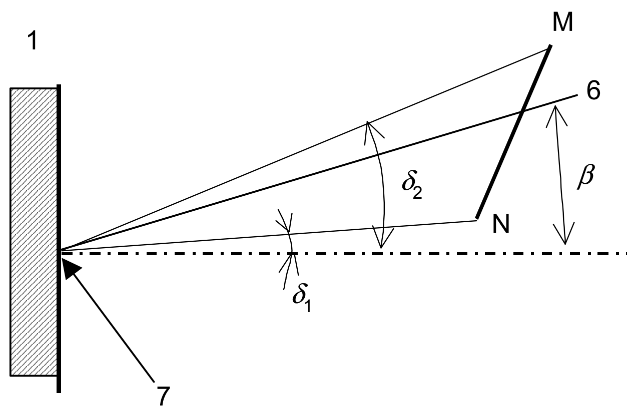
— solar azimuth angle γ

— solar altitude angle β

— limiting points of obstruction M, N



a) Plan



b) Elevation

Key

1 Wall
2 South
3 Solar azimuth angle γ
4 Shading object
5 Normal to the wall
6 Sun ray
7 Reference point
8 North

Figure C.8 - Reference points and angles

This procedure determines whether the wall is sunlit or shaded.

For the given position of the sun:

a) determine the wall-solar azimuth angle $\chi = \gamma - \omega$,

b) if $\chi < \alpha_1$ or $\chi > \alpha_2$ the window is in sun;

c) $\alpha_2 > \chi > \alpha_1$ $A = \chi - \alpha_1$; $B = \delta_1 + A\ (\delta_2 - \delta_1)$;

d) if $\beta > B$ the wall is in sun; otherwise it is in shade.

C.7 Sunlit factor

The area equations in C.2 to C.6 determine the amount of the shaded area A_{sh} of the component.

The sunlit area is given by:

$$A_s = A - A_{sh} \tag{C.6}$$

The sunlit factor f_s is determined as:

$$f_s = 1 - A_s/A \tag{C.7}$$

where

A_s is the shaded area;

A is the total area.

Annex D
(informative)

Design climatic data in the warm season

Design warm weather conditions (cloudless sky) occur on relatively few days in summer. If solar radiation produces a large proportion of heat flow, it is essential to calculate the internal temperature for several months separately. As a consequence, design climatic data should be generally defined for all months. In the absence of a standard[1] on climatic data, the following characteristics of the design climatic data for design temperature calculations can be used.

1) From a long-term data set, identify for each month and for each year the hottest 4-day spell. These are defined as a non-overlapping period of 4 days with the highest average of daily mean temperature.

2) For the chosen month identify discrete 4-day spells with an average period of one per year. This would mean looking at daily temperature data for each day of the chosen month and selecting, for example, the hottest 20 spells from a 20-year data set.

3) Exclude 50 % of these spells on the basis of low average temperature (those 4-day average of daily mean temperatures which lie below the median, where the 4-day averages are rounded to the nearest 0,5 °C).

4) Exclude 50 % of remaining spells on the basis of high diurnal range (those whose 4-day average of daily temperature range which lie above the median; where the 4-day averages are rounded to the nearest 0,5 °C).

This will produce a final selection of 4-day spells of critical conditions with a return period of 4 years.

5) Calculate the hourly averages of temperature and radiation for each hour in each day in the selected spells and take these as design values.

6) Calculate the averages of daily mean wind speed for each day in the selected spells and take these as design values for the corresponding day in the 4-day sequence.

[1] Under preparation within CEN/ TC 89

Annex E
(informative)

Calculation of the internal long-wave radiation exchanges in buildings

E.1 Introduction

This annex gives a simplified procedure for evaluating the internal long-wave radiation exchanges in buildings.

E.2 Limits of application

The procedure can be applied for situations where

— the emissivities of the various surfaces are similar;

— the geometry of the room is simple (rectangular).

For special situations it should be referred to the solution techniques reported in scientific literature (see Bibliography).

E.3 Calculation procedure

The total net long-wave radiation at the n^{th} internal surface is calculated by assuming that each surface n radiates to a fictitious surface f, which has the following characteristics:

- area $A = \sum_{n=1}^{N} (A)_n$ (N = number of the internal surfaces) (E.1)

- emissivity $\varepsilon_f = \sum_{n=1}^{N} (\varepsilon_n A_n) / A_f$ (E.2)

- temperature $T_f = \sum_{n=1}^{N} (A_n \varepsilon_n T_n) / (\varepsilon_f A_f)$ (E.3)

The density of radiant heat flow rate $q_{\mathrm{lri,k}}$ from a surface k to the others is:

$$q_{\mathrm{lri},k} = (q'_{\mathrm{lri},k} - q_{\mathrm{bal}}) \quad \text{(E.4)}$$

where

$$q'_{\mathrm{lri},k} = \sigma F_{\mathrm{if}} (T_f{}^4 - T_k{}^4) \quad \text{(E.5)}$$

$$F_{k,f} = (1 + \frac{1-\varepsilon_k}{\varepsilon_k} + \frac{A_k(1-\varepsilon_k)}{A_f \varepsilon_f}) \quad \text{(E.6)}$$

$$q_{\mathrm{bal}} = \sum_{k=1}^{N} (q'_{\mathrm{lri},k} A_k) / A_f \quad \text{(E.7)}$$

Annex F
(informative)

External radiative long-wave heat transfer coefficients

F.1 Introduction

This annex gives the terms influencing the long-wave radiation exchanges at the external surfaces and provides a procedure for the evaluation of the external long-wave heat transfer coefficient.

F.2 Terms and calculation procedure

Long-wave radiation is emitted by all solids, fluids and most gaseous molecules. External building elements exchange radiation with the sky, the ground, surrounding buildings and plants. For the long-wave radiation balance the view factor F between each environmental radiator and the surface is very important. Assuming that all surrounding radiators are black bodies and the surface considered has an emissivity ε, the net long-wave radiant flux density $q_{\mathrm{lr,e}}$ for an external surface is calculated as:

$$q_{\mathrm{lr,e}} = \varepsilon\sigma(F_{\mathrm{sk}}T_{\mathrm{sk}}{}^4 + F_{\mathrm{b}}T_{\mathrm{b}}{}^4 + F_{\mathrm{g}}T_{\mathrm{g}}{}^4 - T_{\mathrm{es}}{}^4) \quad \text{(F.1)}$$

where

F_{sk} is the view factor with the sky;

F_{b} is the view factor with other buildings;

F_{g} is the view factor with the ground;

T_{sk} is the absolute temperature of the sky;

T_{b} is the absolute temperature of other buildings;

T_{g} is the absolute temperature of the ground;

T_{es} is the external surface temperature of the wall;

ε is the emissivity of the surface;

σ is the Stefan-Boltzmann constant.

View factors depend on the orientation of the surface and on the type of the environments. Table F.1 gives typical view factors for different situations.

Table F.1 - View factors for external environments

Type of "environment"	Vertical surface		
	F_{sk}	F_{b}	F_{g}
City centre	0,33	0,34	0,33
Suburban area	0,41	0,18	0,41
Rural area	0,45	0,10	0,45
Horizontal surface (roof)			
	F_{sk}	F_{b}	F_{g}
All environments	1,00	0,00	0,00

The temperatures included in Equation (F.1) are approximated as follows:

$$T_{sk} = \left[9{,}36 \ x \ 10^{-6} \ (T_{a,e})^{6} \ \right]^{0{,}25} \quad \text{(F.2)}$$

$$T_{b} = T_{a,e}$$

$$T_{g} = T_{a,e}$$

where $T_{a,e}$ is the external air temperature, in kelvins.

Equation (F.1) can be written as:

$$q_{lr,e} = h_{r,e} \ (T_{a,e} - T_{es}) + q_{e,r} \quad \text{(F.3)}$$

with

$$h_{r,e} = 4 \ \varepsilon \ \sigma \ T_{m}^{3} \quad \text{(F.4)}$$

$$q_{e,r} = F'_{sk} \ \varepsilon \ \sigma \ (T_{sk}^{4} - T_{a,e}^{4}) \quad \text{(F.5)}$$

$$T_{m} = (T_{a,e} + T_{s,e}) / 2 \quad \text{(F.6)}$$

Annex G
(informative)

Solar factors

G.1 Introduction

This annex gives the procedures for determining the solar factors introduced in 4.5.3.2 of the document.

G.2 Solar to air factor

The solar to air factor, f_{sa} , is the fraction of the solar heat entering through the glazing and immediately transferred to the internal air. This fraction depends on the quantity of internal items with very low thermal capacity, such as carpets and furniture. Typical values of the solar to air factor are given in Table G.1 according to the amount of furniture.

Table G.1 - Solar to air factor f_{sa}

Amount of furniture	Solar to air factor f_{sa}
No furniture	0
Small amount of furniture	0,10
Large amount of furniture	0,20

G.3 Distribution factors

The distribution factors, f_{df}, are the fraction of the direct solar radiation absorbed per area of the different internal surfaces of the walls, ceiling, floor, etc. They depend on the solar angles, the geometrical dimensions of glazing and the room, the short-wave reflectance of components, and the furniture and furnishings. It is assumed to be time independent. Typical values of the distribution factors are given in Table G.2 as a function of the area-weighted average reflectance, ρ_m, and the area of the envelope elements.

Table G.2 - Distribution factors

Room reflectance	Floor	Vertical walls	Ceiling	Glazing
$\rho \geq 0,7$	$1/A_T$	$1/A_T$	$1/A_T$	$1/A_T$
$0,3 \leq \rho < 0,7$	$0,5/A_f$	$0,4/A_{wa}$	$0,1/A_c$	0,0
$\rho < 0,3$	$0,6/A_f$	$0,35/A_{wa}$	$0,05/A_c$	0,0

where

A_f is the floor area, in m^2;

A_{wa} is the area of all (external and internal) vertical opaque walls, in m^2;

A_c is the ceiling/roof area, in m^2;

A_T is the total envelope area excluding glazing, in m^2.

The area weighted average reflectance is given by:

$$\rho_m = \frac{\sum_{j=1}^{N} A_j \ \rho_j}{\sum_{j=1}^{N} A_j} \tag{G.1}$$

G.4 Solar loss factor

The solar loss factor, f_{lf} , is the fraction of the solar radiation entering through the glazing and reflected back to the external environment. It depends on the geometrical characteristics and solar properties of the glazing system, the exposure of the window, the solar angles and the room colour and geometry. It is assumed to be time independent. A detailed and exact calculation method of the short wave radiation back to the window, is much too complicated for the purposes of this document. It is nevertheless possible to apply here a simplified approach considering the following assumptions:

— the view factors for the various internal surfaces are not considered;

— the solar reflection factor for all surfaces is the same;

— the solar transmittance of the glazing system refers to the direct solar radiation component.

The solar loss factor is given by:

$$f_{lf} = \frac{\tau A_w}{A_t (1 - \rho_m)} \tag{G.2}$$

where

τ is the solar transmittance of the glazing system;

A_w is the window area;

A_t is the room overall surface area;

ρ_m average solar reflection coefficient of room surfaces (defined in Equation (G.1)).

Examples of solar loss factors for different room geometries and solar transmittance of glazing components are given in Table G.3 for a room having an average solar reflection of 0,5 and height 2,7 m.

Table G.3 – Examples of solar loss factors

τ	Glazing system	Window area / Floor area		
		large > 0,375	medium 0,25	small < 0,125
0,8	Clear glazing	0,12	0,08	0,04
0,5	Clear glazing with external shade	0,07	0,05	0,02

Annex H
(informative)

Internal gains

H.1 Introduction

This annex gives typical values of heat flow due to internal energy sources for residential and non residential buildings.

H.2 Residential building

Table H.1 gives the values of the total heat flow rate due to internal sources for different rooms of residential.

Table H.1 - Heat flow rate per floor area

Hour	Kitchen W/m^2	Dining room W/m^2	Bedroom W/m^2	Bathroom W/m^2
1	5	0	5	0
2	5	0	5	0
3	5	0	5	0
4	5	0	5	0
5	5	0	5	0
6	5	0	5	0
7	10	1	2	3
8	10	1	2	3
9	7	1	2	3
10	7	1	0	1
11	7	10	0	1
12	10	10	0	1
13	15	10	0	1
14	15	10	0	1
15	10	1	0	1
16	5	1	0	1
17	5	1	0	1
18	15	15	0	1
19	15	15	0	3
20	15	15	0	3
21	10	15	0	3
22	5	10	2	3
23	5	0	5	0
24	5	0	5	0

H.3 Non residential building

For a non residential building the effect of the internal sources can be very important. In the following Tables suggested values of the heat gain attributable to people, lighting and office equipment, are given.

Table H.2 - Sensible metabolic heat gains of people in offices

Description	**Total heat gain** W per person	**Radiant heat gain** W per person
Drawing office	100	55
High technology office	80	50
Executives office	80	50
Computer suite	80	50
Conference room	80	50
Circulation area	93	37
Toilet	93	37
Restaurant	107	43

Table H.3 - Heat gains from office lighting system

System	**Heat gain for floor area** W per m^2 floor area	**Radiant fraction of the total heat gain** W per m^2 floor area
General office	15	6,8
Drawing office	22	9,9
High technology office	9	4,0
Executives office	12	5,4
Computer suite	9	4,0
Conference room	9	4,0
Circulation area	5	2,3
Toilet	7	3,2

Table H.4 - Heat released from office equipment

Description	m^2 per person	Total heat gain W/m^2	Radiant fraction of the total heat gain W/m^2
General office	6	16	4,9
Drawing office	6	7	2,1
High technology office	6	35	8,4
Executives office	33	7	1,9
Computer suite	20	350	77,0
Conference room	2	5	0,0
Circulation area	2	5	0,0

Annex J
(informative)

Air ventilation

J.1 Introduction

This annex gives a procedure for the evaluation of the air flow rate due to natural ventilation through the openings in building. In the absence of a standard giving a detailed model based on the air mass balance, this annex can be used.

J.2 Calculation procedure

J.2.1 General

The amount of air flowing into the building depends upon the pressure difference between the internal and the external environments and also on the resistance that any openings give to the flow of air. The pressure difference is produced by the action of the wind flow around the building and by the difference in the density of the internal and external air. The aerodynamics of the air flowing are complex but, by deeming openings to be classifiable within two general types, it is possible to specify simple formulae to relate the flow rate to the pressure difference. These categories are:

a) cracks, or small openings with a typical dimension less than approximately 10 mm;

b) openings with a typical dimension larger than approximately 10 mm.

J.2.2 Cracks and small openings

For cracks the ventilation rate is given as:

$$m = \rho \; k \, l \, (\Delta p)^n \qquad \text{(J.1)}$$

where

ρ is the density of the air;

k is the crack coefficient;

l is the length of the crack, in metres;

Δp is the applied pressure difference, in Pa.

Table J.1 gives a range of values of k for the cracks formed around the openings lights of closed windows, which can be used when national values are not available. A suitable value for n is 0,67. The pressure difference between internal and external environments is obtained by adding the thermal buoyancy Δp_T and the wind contribution Δp_w.

$$\Delta p = \Delta p_T + \Delta p_w \qquad \text{(J.2)}$$

The thermal buoyancy Δp_T contribution is given by:

$$\Delta p_T = (\Delta T \, \rho \;\; g \, H_1 \, / \, T_m \,)^{0,5} \qquad \text{(J.3)}$$

where

ΔT is the temperature difference between internal and external environments;

T_m is the reference temperature (300 K);

g is the acceleration due to gravity (= 9,81 m/s^2);

ρ is the air density;

H_1 is the height between high and low level.

The wind contribution Δp_w is given by:

$$\Delta p_w = \rho \, v_r^2 / 2 \tag{J.4}$$

where

ρ is the air density;

v_r is the wind reference velocity.

Table J.1 - Values of *k* for windows (in litres per second and per metre of crack length for an applied pressure difference of 1 Pa)

Window type	Value of *k*	
	average	**range**
Sliding	0,08	0,02 to 0,30
Pivoted	0,21	0,06 to 0,80
Pivoted (weather stripped)	0,08	0,005 to 0,20

J.2.3 Large openings

J.2.3.1 General

The air flow rate can be evaluated by the approximate formula:

$$m = \rho \, c_d \, A \, (2 \, \Delta p / \rho)^{0,5} \tag{J.5}$$

where

c_d is the coefficient of discharge;

A is the area of opening;

Δp is the pressure difference;

ρ is the air density.

Values of the coefficient c_d and area A depend on the position and flow characteristics of all openings. It is usual to assign a value to the discharge coefficient corresponding to that for a sharp-edged orifice, taken here as 0,61. The value of A for other types of opening then becomes the equivalent area associated with that particular opening. Pressure generated by forces of wind and temperature difference produces a movement of air through these openings governed by the fact that the total flow of air into the circulation space equals the outgoing flow rate. Because of the impossibility to obtain a detailed solution for existing buildings, the solution is given for simple cases.

Figure J.2 shows a simple two dimensional representation of a building with no internal divisions and therefore consisting of a single cell with openings as shown, i.e. two (A_1 and A_3) at high level, and two (A_2 and A_4) at low level. According to the wind and temperature situation the air flow by natural ventilation is determined as follows. Figure J.3 shows a similar representation for a building without any internal partition and with openings on one side and one level only.

J.2.3.2 Simple buildings: openings on two facades

J.2.3.2.1 Wind only

$$m_\mathrm{w} = c_\mathrm{d}\, A_\mathrm{w}\, v_\mathrm{r}\, (\Delta c_p)^{0,5} \qquad \text{(J.6)}$$

and

$$v_\mathrm{r} = K\, z^a\, v_\mathrm{f} \qquad \text{(J.7)}$$

where

K is a coefficient as given in Table J.3;

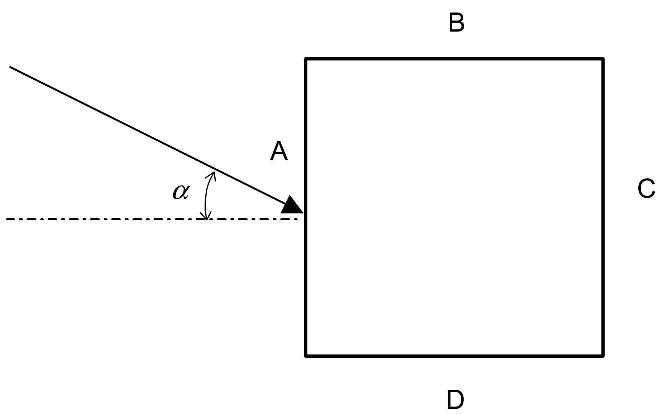
a is an exponent as given in Table J.3;

z is the height of the building, in metres;

v_f is the free wind velocity, in m/s.

$$\frac{1}{A_\mathrm{w}^2} = \frac{1}{(A_1 + A_2)^2} + \frac{1}{(A_3 + A_4)^2} \qquad \text{(J.8)}$$

where Δc_p is the applied differential mean pressure coefficient calculated as difference between the mean surface pressure coefficients, derived from Table J.2 for the opposite walls perpendicular to the wind direction.



Key

α Wind angle

A, B, C, and D state the position of the external surfaces of a room in plan view.

Figure J.1 - Number of surfaces with respect to the wind direction

Table J.2 - Coefficient c_p as a function of the wind angle and building proportions

Building height ratio h/w	Building plan ratio l/w	Wind angle α	c_p surfaces A	B	C	D
(h/w)<1/2	1<(l/w)≤3/2	0	0,7	-0,2	-0,5	-0,5
		90	-0,5	-0,5	0,7	-0,2
	3/2</(l/w)<4	0	0,7	-0,2	-0,6	-0,6
		90	-0,5	-0,5	0,7	-0,1
1/2≤(h/w)<3/2	1<(l/w)≤3/2	0	0,7	-0,2	-0,6	-0,6
		90	-0,6	-0,6	0,7	-0,2
	3/2<(l/w)<4	0	0,7	-0,3	-0,7	-0,7
		90	-0,5	-0,5	0,7	0,1
3/2≤(h/w)<6	1< (l/w) ≤3/2	0	0,8	-0,2	-0,8	-0,8
		90	-0,8	-0,8	0,8	-0,2
	3/2< (l/w) <4	0	0,7	-0,4	-0,7	-0,7
		90	- 0,5	- 0,5	0,8	- 0,1

with h height of the building; l length of the building; w depth of the building

Table J.3 - Coefficient *K* and exponent *a*

Terrain	*K*	*a*
Open flat country	0,68	0,17
Country with scattered wind breaks	0,52	0,20
Sub-urban	0,35	0,25
City centre	0,21	0,33

J.2.3.2.2 Temperature difference only

$$m_T = c_d\, \rho\, A_T\, (2\, \Delta\theta\, g\, H_1 / T_m)^{0,5} \tag{J.9}$$

$$\frac{1}{A_T^2} = \frac{1}{(A_1 + A_2)^2} + \frac{1}{(A_3 + A_4)^2} \tag{J.10}$$

where

c_d is the coefficient of discharge;

m_T is the mass flow rate due to temperature difference;

$\Delta\theta$ is the temperature difference between internal and external environments;

T_m is the reference temperature (300 K);

g is the acceleration due to gravity (= 9,81 m/s^2);

H_1 is the height between high and low level.

J.2.3.2.3 Wind and temperature difference together

$$m = m_T \quad \text{for} \quad v_f /(\Delta T)^{0,5} < (0,26\, (A_T / A_w)^{0,5}\, (H_1/ \Delta c_p)^{0,5})$$

$$m = m_w \quad \text{for} \quad v_f /(\Delta T)^{0,5} > (0,26\, (A_T / A_w)^{0,5}\, (H_1/ \Delta c_p)^{0,5}) \tag{J.11}$$

J.2.3.3 Openings in one wall only

J.2.3.3.1 Wind

$$m_w = 0,025\, A\, v_r \tag{J.12}$$

where v_r is defined by Equation (J.7).

J.2.3.3.2 Temperature differences with openings at two levels

$$m_T = c_d\, \rho\, A\, \left(\frac{2\, \Delta\theta\, g\, H_1}{T_m}\right)^{0,5} \tag{J.13}$$

where

A_T is determined from Equation (J.10);

c_d is the coefficient of discharge.

J.2.3.3.3 Temperature difference with openings at one level only

$$m_T = c_d\, \rho \frac{A}{3} \left(\frac{\Delta\theta\, g\, H}{T_m}\right)^{0,5} \tag{J.14}$$

where H is the opening height.

J.3 Example of calculation of natural ventilation rates for simple building

J.3.1 General

Consider a building consisting of a single undivided space 25 m long, 10 m wide and 8 m high. Given that the building is situated in an open suburban area, calculate:

a) the natural ventilation rate due to wind;

b) the ventilation rate due to the effect of a temperature difference of 6 K in the absence of the wind.

There are no ventilation openings on the shorter walls. On each of the longer walls there are openings of 2,5 m^2 at low level, and 5,0 m^2 at high level, separated by a vertical distance of 6,0 m, and evenly distributed along the length of the wall.

J.3.2 Wind

Determination of the pressure coefficient difference

Building height ratio = 0,8

Building plan ratio = 2,5

Thus, from Table J.2, the difference in mean surface pressure coefficients at the two long sides A and C of the building for a perpendicular wind is:

$\Delta c_p = 0{,}7 - (-0{,}3) = 1{,}0$

Determination of v_r

From climatic data table $v_r = 3{,}75$ m/s

terrain: country with scattered wind breaks :

From Table J.3: $K = 0{,}52$; $a = 0{,}20$

Thus, from Equation (J.7), using the building height of 8 m, $v_r = 3{,}75 \times 0{,}52\ (8)^{0{,}2} = 2{,}96$ m/s

Determination of A_w

From Equation (J.8):

$1/(A_w)^2 = 1/(5{,}0 + 2{,}5)^2 + 1/(5{,}0 + 2{,}5)^2$

$A_w = 5{,}3\ m^2$

Determination of ventilation rate

Using Equation (J.6):

$m_w = (0{,}61 \times 5{,}3 \times 2{,}96 \times 1)^{0{,}5} = 3{,}09\ m^3$ /s

The volume of the building is $V = 25 \times 10 \times 8 = 2000\ m^3$

The air change rate is $3600 \times 3{,}09 / 2000 = 5{,}57$ air changes/hour

J.3.3 Temperature difference

From the information given:

Temperature difference $\Delta\theta = 6$ K

Height between openings $H_1 = 6{,}0$ m

Thus from Equation (J.10):

$$1/(A_T)^2 = (1/(2{,}5 + 2{,}5)^2) + (1/(5{,}0 + 5{,}0)^2)$$

$$A_T = 4{,}47 \text{ m}^2$$

Taking $T_m = 300$ K and $\rho = 1{,}2$ kg/m^3

$$m_T = 0{,}61 \times 1{,}2 \, \frac{4{,}47}{3} \left(\frac{6 \times 9{,}8 \times 6}{300}\right)^{0{,}5} = 1{,}183 \text{ m}^3/\text{s}$$

The air change rate is:

$$3600 \times 1{,}183 / 2000 = 2{,}13 \text{ air changes/hour}$$

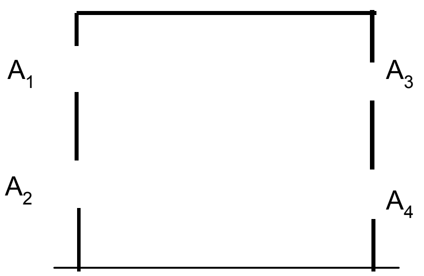


Figure J.2 - Openings on two sides

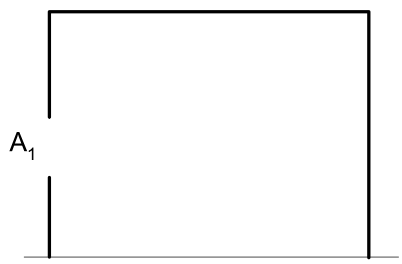


Figure J.3 - Opening on one side

Annex K
(informative)

Detailed results of the validation tests considered in the "whole validation model" procedure

This annex gives the hourly values of the operative temperature for the various tests considered in the "whole calculation method" included in 7.3.

Table K.1 - Geometry A: Hourly values of the operative temperature

Hours	A1a	A1b	A1c	A2a	A2b	A2c	A3a	A3b	A3c
0 to 1	35,6	27,8	27,6	35,5	28,1	28,0	38,4	29,8	29,1
1 to 2	35,1	27,2	27,0	35,3	27,7	27,5	38,1	29,3	28,7
2 to 3	34,7	26,6	26,5	35,0	27,3	27,2	37,8	28,9	28,3
3 to 4	34,4	26,1	26,0	34,8	26,9	26,8	37,6	28,5	27,9
4 to 5	34,0	25,7	25,6	34,6	26,6	26,5	37,3	28,2	27,6
5 to 6	33,8	25,5	25,4	34,5	26,5	26,4	37,2	28,0	27,4
6 to 7	33,6	26,6	25,3	34,4	27,8	26,4	37,1	29,7	27,4
7 to 8	33,6	27,1	25,6	34,5	28,4	26,6	37,1	30,4	27,6
8 to 9	33,7	27,5	26,0	34,6	28,7	27,1	37,1	30,9	28,0
9 to 10	33,8	27,9	26,7	34,7	29,1	27,7	37,3	31,2	28,6
10 to 11	34,0	28,3	27,7	34,9	29,4	28,4	37,5	31,6	29,4
11 to 12	34,7	29,2	29,0	35,5	30,0	29,5	38,0	32,4	30,5
12 to 13	35,5	30,2	30,4	36,1	30,5	30,4	38,6	33,1	31,6
13 to 14	36,5	31,3	31,7	36,7	31,1	31,3	39,4	33,8	32,6
14 to 15	37,6	32,6	32,8	37,3	31,7	32,0	40,2	34,6	33,3
15 to 16	38,2	33,3	33,3	37,5	32,0	32,3	40,5	35,0	33,7
16 to 17	38,6	33,8	33,5	37,6	32,2	32,4	40,7	35,3	33,8
17 to 18	38,6	34,1	33,1	37,6	32,2	32,1	40,7	35,4	33,6
18 to 19	38,7	33,0	32,6	37,6	31,9	31,7	40,8	34,2	33,1
19 to 20	38,1	31,9	31,6	37,2	31,2	31,0	40,3	33,2	32,3
20 to 21	37,7	31,0	30,8	37,0	30,5	30,4	40,0	32,5	31,7
21 to 22	37,4	30,3	30,1	36,8	30,0	29,8	39,8	31,8	31,1
22 to 23	37,0	29,5	29,3	36,5	29,3	29,2	39,5	31,2	30,4
23 to 24	36,2	28,5	28,4	35,9	28,6	28,5	38,8	30,4	29,7
Average	35,9	29,4	29,0	36,0	29,5	29,1	38,7	31,6	30,3

Table K.1.2 - Geometry B: Hourly values of the operative temperature

Hours	B1a	B1b	B1c	B2a	B2b	B2c	B3a	B3b	B3c
0 to 1	30,3	19,7	19,4	30,1	20,2	20,0	32,1	21,7	21,0
1 to 2	29,6	18,8	18,6	29,7	19,6	19,4	31,7	21,0	20,4
2 to 3	28,9	18,0	17,8	29,4	19,0	18,8	31,3	20,4	19,8
3 to 4	28,3	17,3	17,1	29,0	18,5	18,3	30,9	19,9	19,3
4 to 5	27,8	16,7	16,5	28,8	18,1	17,9	30,6	19,4	18,9
5 to 6	27,4	16,4	16,2	28,6	17,9	17,7	30,4	19,2	18,6
6 to 7	27,2	17,8	16,2	28,5	19,8	17,8	30,3	21,4	18,7
7 to 8	27,1	18,5	16,5	28,6	20,5	18,1	30,3	22,3	19,0
8 to 9	27,2	19,0	17,1	28,7	21,0	18,7	30,4	22,9	19,5
9 to 10	27,4	19,6	18,1	28,9	21,5	19,5	30,6	23,4	20,4
10 to 11	27,7	20,3	19,4	29,2	21,9	20,6	30,9	23,9	21,4
11 to 12	28,6	21,4	21,3	29,9	22,7	22,0	31,6	24,9	23,0
12 to 13	29,6	22,8	23,2	30,7	23,4	23,3	32,4	25,8	24,4
13 to 14	31,2	24,6	25,1	31,7	24,4	24,6	33,6	26,9	25,7
14 to 15	33,0	26,6	26,8	32,6	25,4	25,7	34,6	28,0	26,9
15 to 16	34,2	28,0	27,7	32,2	26,0	26,2	35,2	28,7	27,5
16 to 17	35,2	29,2	28,1	33,6	26,5	26,4	35,7	29,3	27,7
17 to 18	35,6	29,9	27,8	33,7	26,7	26,2	35,9	29,6	27,5
18 to 19	35,8	28,1	27,2	33,7	26,1	25,7	36,0	28,2	27,0
19 to 20	34,9	26,3	25,7	33,1	24,9	24,6	35,4	26,8	25,8
20 to 21	33,8	24,6	24,2	32,4	23,7	23,5	34,6	25,5	24,6
21 to 22	33,0	23,3	23,0	32,0	22,8	22,6	34,1	24,5	23,7
22 to 23	32,2	22,1	21,8	31,5	21,9	21,6	33,5	23,5	22,8
23 to 24	31,2	20,8	20,5	30,7	20,9	20,7	32,7	22,5	21,8
Average	**30,5**	**21,3**	**21,5**	**30,8**	**22,2**	**21,7**	**32,7**	**24,2**	**22,7**

Annex ZA
(normative)

Normative references to international publications with their corresponding European publications

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.

Year	Publication	Title	EN	Year	Title
–	ISO 6946	Building components and building elements – Thermal resistance and thermal transmittance – Calculation method	EN ISO 6946	-	Building components and building elements – Thermal resistance and thermal transmittance – Calculation method (ISO 6946:1996)
	ISO 7345	Thermal insulation – Physical quantities and definitions	EN ISO 7345	–	Thermal insulation – Physical quantities and definitions (ISO 7345:1987)
-	ISO 9288	Thermal insulation – Heat transfer by radiation – Physical quantities and definitions	EN ISO 9288	-	Thermal insulation – Heat transfer by radiation – Physical quantities and definitions (ISO 9288:1989)
-	ISO 9346	Thermal insulation – Mass transfer – Physical quantities and definitions	ISO 9346	-	Thermal insulation – Mass transfer – Physical quantities and definitions (ISO 9346:1987)
-	ISO 9251	Thermal insulation – Heat transfer conditions and properties of materials – Vocabulary	EN ISO 9251	-	Thermal insulation – Heat transfer conditions and properties of materials – Vocabulary (ISO 9251:1987)
-	ISO 10077-1	Thermal performance of windows, doors and shutters – Calculation of thermal transmittance – Part 1: Simplified method	EN ISO 10077-1	-	Thermal performance of windows, doors and shutters – Calculation of thermal transmittance – Part 1: Simplified method (ISO 10077-1:2000)
-	ISO 10077-2	Thermal performance of windows, doors and shutters – Calculation of thermal transmittance – Part 2: Numerical method for frames	EN ISO 10077-2[1]	-	Thermal performance of windows, doors and shutters – Calculation of thermal transmittance – Part 2: Numerical method for frames (ISO 10077-2:2003)
-	ISO 13370	Thermal performance of buildings – Heat transfer via the ground – Calculation methods	EN ISO 13370	-	Thermal performance of buildings – Heat transfer via the ground – Calculation methods (ISO 13370:1998)

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