
**Thermal performance of windows, doors
and shading devices — Detailed
calculations**

*Performance thermique des fenêtres, portes et stores — Calculs
détaillés*

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Foreword

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

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

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

ISO 15099 was prepared by Technical Committee ISO/TC 163, *Thermal performance and energy use in the built environment*.

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Introduction

This International Standard describes a procedure for calculating indices of merit of many window and door products. The method provided in this International Standard allows the user to determine total window and door product indices of merit, viz thermal transmittance, total solar energy transmittance and visible light transmittance.

The procedures give the actual thermal performance of fenestration products for use in building energy analysis and for the evaluation of products in specific building applications. These procedures can also be used to produce data to compare products by using the standardized boundary conditions given either in this International Standard or taken from the appropriate International or National Standards (e.g., ISO 12567-1, ISO 10292, ISO 9050). This International Standard is also intended as a reference document for the description of models used in computer programs for detailed calculation of the thermal and optical transmission properties of window and door systems.

This International Standard gives detailed models for thermal and optical transmission in windows. These detailed models are necessary in many types of window to get agreement between calculations and tests.

Traditionally, windows have been characterized by separately calculating the “dark” or “night-time” thermal transmittance and the solar energy transmittance through the fenestration system. The thermal transmittance without the effect of solar radiation is calculated using the procedures given in ISO 10292 (for the vision portion) and the total solar energy transmittance, without taking into account the actual temperatures of the various panes, is obtained using ISO 9050. These calculations require the use of reference conditions that are not representative of actual conditions. In this International Standard the energy balance equations are set up for every glazing layer taking into account the solar absorption and actual temperatures. From these energy balance equations, the temperatures of the individual layers and gaps are determined. This is the only standard that takes into account these complex interactions. This more detailed analysis provides results that can then be expressed as thermal transmittance and τ_g -values and these values can differ from the results of simpler models.

Individual indices of merit obtained using fixed reference boundary conditions are useful for comparing products. However, the approach taken is the only way of calculating the energy performance of window systems for other environmental conditions including those conditions that may be encountered during hot box measurements.

Finally it must be emphasized that this International Standard is intended for use in computer programs. It was never intended as a “simplified calculation” procedure. Simplified methods are provided in other International Standards. It is essential that these programs produce consistent values and that they are based on a sound standard methodology. Although more complicated than the formulae used in the simplified standards, the formulae used in this International Standard are entirely appropriate for their intended use.

Thermal performance of windows, doors and shading devices — Detailed calculations

1 Scope

This International Standard specifies detailed calculation procedures for determining the thermal and optical transmission properties (e.g., thermal transmittance, total solar energy transmittance) of window and door systems based on the most up-to-date algorithms and methods, and the relevant solar and thermal properties of all components.

Products covered by this International Standard include windows and doors incorporating:

- a) single and multiple glazed fenestration products with or without solar reflective, low-emissivity coatings and suspended plastic films;
- b) glazing systems with pane spacing of any width containing gases or mixtures of gases;
- c) metallic or non-metallic spacers;
- d) frames of any material and design;
- e) fenestration products tilted at any angle;
- f) shading devices;
- g) projecting products.

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.

ISO 7345, *Thermal insulation — Physical quantities and definitions*

ISO 8301, *Thermal insulation — Determination of steady-state thermal resistance and related properties — Heat flow meter apparatus*

ISO 8302, *Thermal insulation — Determination of steady-state thermal resistance and related properties — Guarded hot plate apparatus*

ISO 9050, *Glass in building — Determination of light transmittance, solar direct transmittance, total solar energy transmittance, ultraviolet transmittance and related glazing factors*

ISO 9288, *Thermal insulation — Heat transfer by radiation — Physical quantities and definitions*

ISO 9845-1, *Solar energy — Reference solar spectral irradiance at the ground at different receiving conditions — Part 1: Direct normal and hemispherical solar irradiance for air mass 1,5*

ISO 10077-2:2003, *Thermal performance of windows, doors and shutters — Calculation of thermal transmittance — Part 2: Numerical method for frames*

ISO 10211-1, *Thermal bridges in building construction — Heat flows and surface temperatures, Part 1: General calculation methods*

ISO/CIE 10526:1999, *CIE standard Illuminants for colorimetry*

ISO/CIE 10527, *CIE standard colorimetric observers*

ISO 12567-1, *Thermal performance of windows and doors — Determination of thermal transmittance by hot box method — Part 1: Complete windows and doors*

EN 12898, *Glass in building — Determination of the emissivity*

3 Symbols

3.1 General

Symbols and units used are in accordance with ISO 7345 and ISO 9288. The terms, which are specific to this International Standard, are listed in Table 1.

3.2 Symbols and units

Table 1 — Terms with their symbols and units

Symbol	Term	Unit
A	area	m ²
A_i	portion of absorbed solar energy by the i th glazing layer	1
A_R	aspect ratio	1
b	width (breadth) of a groove or slit	mm
c_p	specific heat capacity at constant pressure	J/(kg·K)
d	thickness	m
d_g	thickness of glazing cavity	m
E	irradiance	W/m ²
$E_s(\lambda)$	solar spectral irradiance function (see ISO 9845-1)	1
$E_v(\lambda)$	colorimetric illuminance (CIE D65 function in ISO/CIE 10526:1999)	lx
g	acceleration due to gravity	m/s ²
G	parameter used in the calculation of convective heat transfer coefficients; see Equation (48)	1
h	surface coefficient of heat transfer	W/(m ² ·K)
H	height of glazing cavity	m
I	total density of heat flow rate of incident solar radiation	W/m ²
$I_i^+(\lambda)$ $I_i^-(\lambda)$	spectral heat flow rate of radiant solar energy between i th and $i + 1$ th glazing layers travelling in the external (+) or internal (-) direction	W

Table 1 (continued)

Symbol	Term	Unit
J	radiosity	W/m ²
l	length	m
\hat{M}	molecular mass	mole
N	number of glazings + 2	1
Nu	Nusselt number	1
P	pressure	Pa
q	density of heat flow rate	W/m ²
r	reflectance: portion of incident radiation reflected such that the angle of reflection is equal to the angle of incidence	1
R	thermal resistance	m ² ·K/W
$R(\lambda)$	photopic response of the eye (see ISO/CIE 10527)	
\mathcal{R}	universal gas constant	J/(kmol·K)
Ra	Rayleigh number	1
Ra_x	Rayleigh number based on length dimension x	1
S_i	density of heat flow rate of absorbed solar radiation at i th glazing layer	W/m ²
t_{perp}	largest dimension of frame cavity perpendicular to heat flow	m
T	thermodynamic temperature	K
ΔT_i	temperature drop across i th glazing cavity, $\Delta T_i = T_{f,i} - T_{b,i+1} $	K
u	air velocity near a surface	m/s
U	thermal transmittance	W/(m ² ·K)
v	free-stream air speed near window, mean air velocity in a gap	m/s
x, y	dimensions in a Cartesian co-ordinate system	1
Z	pressure loss factor	1
α	absorption	1
β	thermal expansion coefficient of fill gas	K ⁻¹
ε	total hemispherical emissivity	1
γ	angle	°
θ	temperature	°C
σ	Stefan-Boltzmann constant, $5,669\ 3 \times 10^{-8}$	W/(m ² ·K ⁴)
λ	thermal conductivity	W/(m·K)
λ_w	wavelength	m
μ	dynamic viscosity	Pa·s
ρ	density	kg/m ³
τ	transmittance	1
τ_S	total solar energy transmittance: the portion of radiant solar energy incident on the projected area of a fenestration product or component that becomes heat gain in the internal conditioned space	1
ϕ	parameter used in the calculation of viscosity and of thermal conductivity; see Equations (62) and (67)	1
φ	function used in the calculation of heat transfer; see Equation (112)	1
Φ	heat flow rate	W
Ψ	linear thermal transmittance	W/(m·K)

3.3 Subscripts

The subscripts given in Table 2 shall be applied.

Table 2 — Subscripts and meanings

Subscript	Meaning
ai	air
av	average
b	backward
bo	bottom of a gap
cc	condition on the cold side
cdv	conduction/convection (unvented)
cg	centre of glass
ch	condition on the hot (warm) side
cr	critical
cv	convection
de	divider edge glass
dif	diffuse
dir	direct
div	divider
eff	effective
eg	edge of glass
eq	equivalent
ex	external
f	frame
fr	frame (using the alternative approach)
ft	front
gv	glass or vision portion
ht	hot
hz	horizontal
<i>i</i>	counter
int	internal
inl	inlet of a gap
<i>j</i>	counter
m	mean
mix	mixture
<i>n</i>	counter
ne	environmental (external)
ni	environmental (internal)
out	outlet of a gap
p	panel
r	radiation or radiant
red	reduced radiation
s	surface
sc	source
sk	sink
sl	solar
t	total
tp	top of a gap
<i>v</i>	number of gases in a gas mixture
v	vertical
<i>z</i>	at distance <i>z</i>
Ψ	perimeter
2 <i>D</i>	coupling

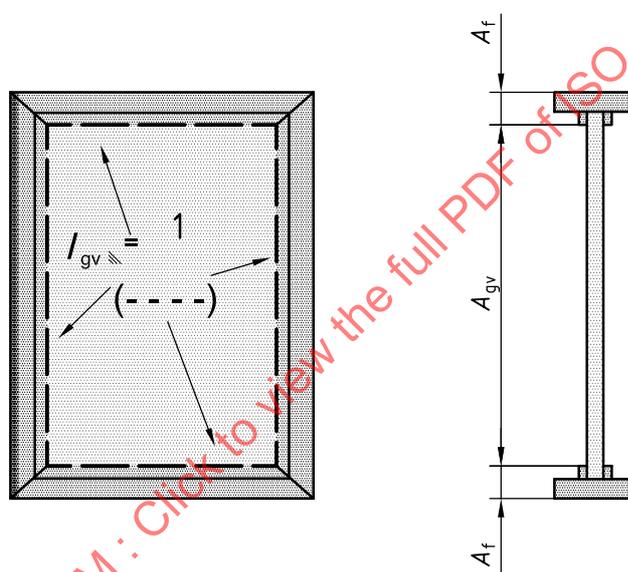
4 Determination of total window and door system properties

4.1 Thermal transmittance

4.1.1 General

This International Standard presents procedures by which detailed computations can be used to determine the thermal transmission properties of various product components, which are then used to determine the thermal transmission properties of the total product. Where national standards allow, test procedures may be used to determine component and total product properties.

The total properties for window and door products are calculated by combining the various component properties weighted by either their respective projected areas or visible perimeter. The total properties are each based on total projected area occupied by the product, A_t . The projected component areas and the visible perimeter are shown in Figure 1.



Key

1 perimeter length at sight line - - - - -

Figure 1 — Schematic diagram showing the window projected areas and vision perimeter

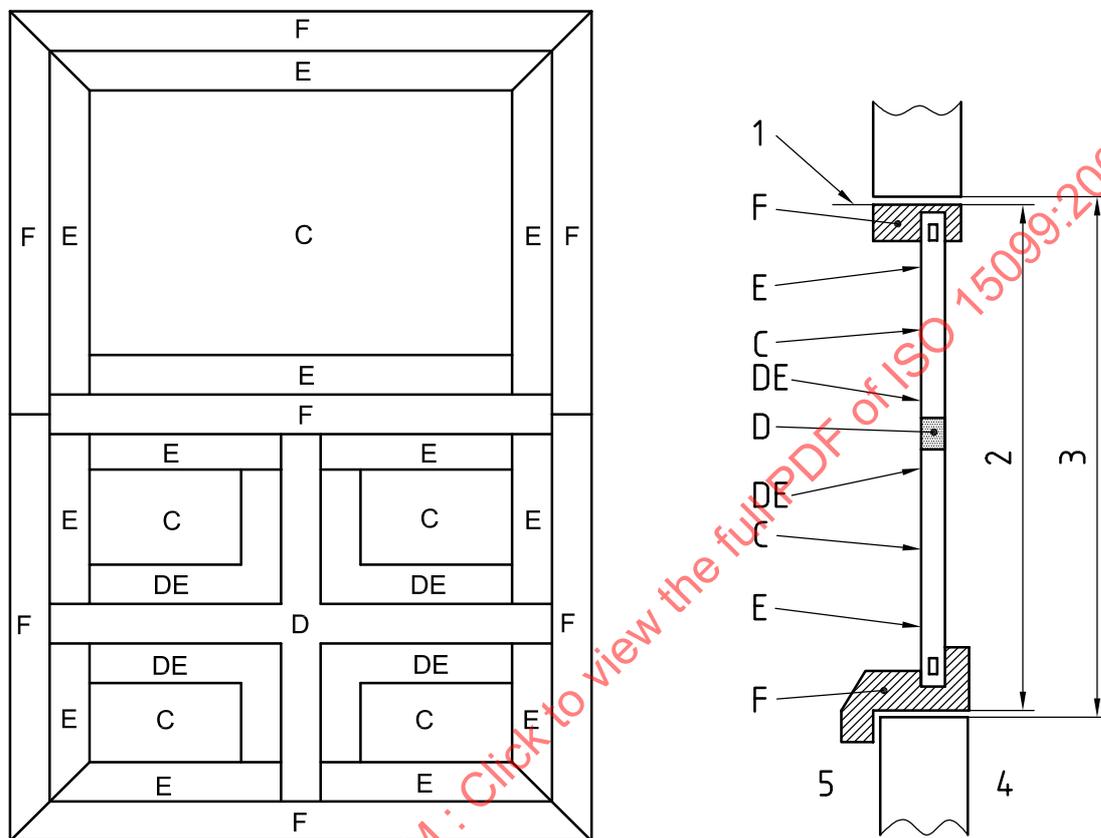
Clause 4 describes the procedure for calculating thermal transmittance, total solar transmittance and visible transmittance for the complete product. 4.1 describes the procedure for calculating thermal transmittance. The effect of three-dimensional heat transfer in frames and glazing units is not considered. 4.1.4 describes an alternative procedure for calculating edge of glass and frame thermal indices U_{de} , U_{eg} , U_t and U_{fr} , which are used in area-based calculations. Clause 5 describes the procedure for calculating the required centre-glass properties τ_{sgv} and τ_{gv} . Clause 6 describes the procedure for calculating the linear thermal transmittance, Ψ , which accounts for the interaction between frame and glazing or opaque panel. Clause 7 contains the procedure for dealing with shading devices and ventilated windows. Clause 8 describes the procedure for determining and applying boundary conditions. The thermal transmittance of the fenestration product is given by:

$$U_t = \frac{\sum A_{gv} U_{gv} + \sum A_f U_f + \sum l_\Psi \Psi}{A_t} \quad (1)$$

where A_{gv} and A_f are the projected vision area and frame area, respectively. The length of the vision area perimeter is l_Ψ , and Ψ is a linear thermal transmittance that accounts for the interaction between frame and glazing or the interaction between frame and opaque panel (e.g., a spandrel panel).

The summations included in Equation (1) are used to account for the various sections of one particular component type; e.g. several values of A_f are needed to sum the contributions of different values of U_f corresponding to sill, head, dividers and side jambs.

Figure 2 illustrates the division into components for the alternative approach described in 4.1.4, in which the edge-of-glass and divider-edge areas are 63,5 mm (2,5 in) wide. The sum of all component areas equals the total projected fenestration product area.



Key

C	Centre-of-glass	1	installation clearance
E	Edge-of-glass	2	projected area
F	Frame	3	rough opening
D	Divider	4	interior
DE	Divider-edge	5	exterior

Figure 2 — Centre-of-glass, edge-of-glass, divider, divider-edge, and frame areas for a typical fenestration product

4.1.2 Glazed area thermal transmittance

The thermal transmittance can be found by simulating a single environmental condition involving internal/external temperature difference, with or without incident solar radiation. Without solar radiation, the thermal transmittance is the reciprocal of the total thermal resistance.

$$U_{gv} = \frac{1}{R_t} \tag{2}$$

and when solar radiation is considered, then:

$$U_{gv} = \frac{q_{int}(I_s = 0)}{T_{ni} - T_{ne}} \tag{3}$$

where $q_{\text{int}}(I_s = 0)$ is the net density of heat flow rate through the window or door system to the internal environment for the specified conditions, but without incident solar radiation, in W/m^2 . The condition "without solar radiation" is used because all effects on the thermal resistances due to incident solar radiation are incorporated in the total solar energy transmittance or τ_{S} -value [see Equation (14)], and T_{ni} and T_{ne} are the environmental temperatures, as defined in Equation (7).

R_t is found by summing the thermal resistances at the external and internal boundaries, and thermal the resistances of glazing cavities and glazing layers. See Figure 3.

$$R_t = \frac{1}{h_{\text{ex}}} + \sum_{i=2}^n R_i + \sum_{i=1}^n R_{\text{gv},i} + \frac{1}{h_{\text{int}}} \quad (4)$$

where the thermal resistance of the i th glazing is:

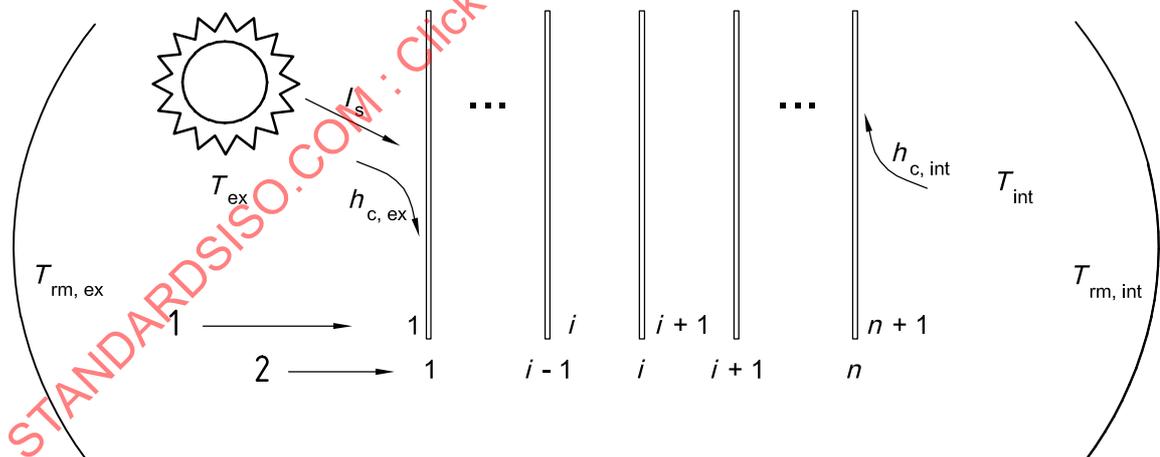
$$R_{\text{gv},i} = \frac{t_{\text{gv},i}}{\lambda_{\text{gv},i}} \quad (5)$$

and the thermal resistance of the i th space, where the first space is external environment, the last space is internal environment and the spaces in between are glazing cavities, (see Figure 3):

$$R_i = \frac{T_{\text{f},i} - T_{\text{b},i-1}}{q_i} \quad (6)$$

where $T_{\text{f},i}$, and $T_{\text{b},i-1}$ are the external and internal facing surface temperature of the i th glazing layer.

The environmental temperature [as defined in Equation (7)] is a weighted average of the ambient air temperature and the mean radiant temperature, T_{rm} , which is determined for external and internal environment boundary conditions (see boundary conditions in 8.4.1).



Key

- 1 gap
- 2 glazing

Figure 3 — Numbering system for glazing system layers

The environmental temperature, T_n , is:

$$T_n = \frac{h_{\text{cv}} T_{\text{ai}} + h_r T_{\text{rm}}}{h_{\text{cv}} + h_r} \quad (7)$$

where h_{cv} and h_r are determined according to the procedure given in Clause 8.

4.1.3 Frame area/edge-glass thermal indices

In order to convert the results of a two-dimensional numerical analysis to thermal transmittances, it is necessary to record the rate of heat transfer from the internal environment to the frame and edge-glass surfaces (in the absence of solar radiation). The linear thermal transmittance, Ψ , values and frame thermal transmittances shall be calculated according to the following equations.

$$\Psi = L^{2D} - U_f l_f - U_{gv} l_{gv} \quad (8)$$

where L^{2D} is thermal coupling coefficient determined from the actual fenestration system.

$$U_f = \frac{L_p^{2D} - U_p l_p}{l_f} \quad (9)$$

where

L_p^{2D} is thermal coupling coefficient determined from the frame/panel insert system;

U_p is the thermal transmittance of foam insert;

l_p is the internal side exposed length of foam insert (minimum 100 mm);

l_f is the internal side projected length of the frame section;

l_{gv} is the internal side projected length of the glass section (see Figures C.1 and C.2 of ISO 10077-2:2003, for further details on the definition of l_f and l_p).

The detailed procedure for determining L^{2D} is also given in ISO 10211-1.

4.1.4 Alternative approach (see Figure 2)

An alternative method is available for calculating frame thermal transmittance, U_{fr} . Using this method it is unnecessary to determine the linear thermal transmittance, Ψ . Instead, the glass area, A_{gv} , is divided into centre-glass area, A_c , plus edge-glass area, A_e , and one additional thermal transmittance, U_{eg} , is used to characterize the edge-glass area. If dividers are present then divider area, A_{div} and divider thermal transmittance, U_{div} are calculated, as well as corresponding divider edge area, A_{de} and thermal transmittance, U_{de} . The following equation shall be used to calculate the total thermal transmittance:

$$U_t = \frac{\sum U_{cg} A_c + \sum U_{fr} A_f + \sum U_{eg} A_e + \sum U_{div} A_{div} + \sum U_{de} A_{de}}{A_t} \quad (10)$$

where U_{fr} and U_{eg} can be determined from the following equations:

$$U_{fr} = \frac{\Phi_{fr}}{l_f (T_{ni} - T_{ne})} \quad (11)$$

$$U_{eg} = \frac{\Phi_{eg}}{l_{eg} (T_{ni} - T_{ne})} \quad (12)$$

and where l_f is projected length of frame area and l_{eg} is the length of edge of glass area and is equal to 63,5 mm. These lengths are measured on the internal side. The quantities Φ_{fr} and Φ_{eg} are heat flow rates through frame and edge-glass areas (internal surfaces), respectively, including the effect of glass and spacer, and both are expressed per length of frame or edge-glass. The calculations shall be performed for each combination of frame and glazing with different spacer bars.

The summations included in Equation (10) are used to account for the various sections of one particular component type; e.g., several values of A_f must be used to sum the contributions of different values of U_{fr} corresponding to sill, head and side jambs.

It should be noted that the two different approaches entail different definitions of frame thermal transmittance, denoted U_f and U_{fr} . The primary difference is that the U_{fr} includes the some of the heat transfer caused by the edge seal, whereas U_f does not. The comparison of frame properties for two different products is only meaningful if the same calculation procedure has been used in both cases.

The U_t values for windows calculated by the two methods may differ because of differences in the way frame and edge heat transfer is treated at the corners, particularly because the three dimensional effects are neglected. This difference is more pronounced for smaller windows. The choice of $l_{eg} = 63,5$ mm is made to reduce the discrepancy between the two alternative approaches.

4.2 Total solar energy transmittance

4.2.1 General

The total solar energy transmittance of the total fenestration product is:

$$\tau_s = \frac{\sum \tau_g A_g + \sum \tau_f A_f}{A_t} \quad (13)$$

where τ_g and τ_f are the individual total solar energy transmittance values of the vision area and frame area, respectively. The summations are included for the same reason that they appear in Equation (1) and shall be applied in the same manner to account for differing sections of one particular component type.

NOTE Equation (13) includes an assumption that the solar transmittance of the edge of glass is the same as that of the centre of glass area.

4.2.2 Vision area total solar energy transmittance

The total solar energy transmittance can be determined for conditions involving internal/external temperature difference and any level of incident solar radiation. It is found by calculating the difference between the net heat flow rate into the internal environment with and without incident solar radiation.

$$\tau_s = \frac{q_{int} - q_{int}(I_s = 0)}{I_s} \quad (14)$$

where

q_{int} is the net density of heat flow rate through the window or door system to the internal environment for the specified conditions, in W/m^2 ;

$q_{int}(I_s = 0)$ is the net density of heat flow rate through the window or door system to the internal environment for the specified conditions, but without incident solar radiation, in W/m^2 .

For the equivalent expression for U , see Equation (3).

The net density of heat flow rates, q_{int} and $q_{int}(I_s = 0)$ are calculated in 5.3.1 [Equation (27), for index $i = int$].

For a glazing assembly in which a shading device is involved, the amendments to the equations of 5.2 as given in 7.2 shall be applied.

4.2.3 Frame total solar energy transmittance

The frame total solar energy transmittance shall be calculated using the approximate equation:

$$\tau_f = \alpha_f \frac{U_f}{\frac{A_s}{A_f} h_{ex}} \tag{15}$$

where A_s is the developed surface area.

The external surface heat transfer coefficient (combined convective/radiative) at the frame, h_{ex} , is $h_{ex} = h_{cv,ex} + h_{r,ex}$.

If the alternative method of calculating U_t is being used, U_{fr} should be used instead of U_f in Equation (15).

More detailed two-dimensional or three-dimensional calculations, including the effects of off-normal solar radiation, shading, reflected solar radiation and solar radiation transmitted to the internal frame surfaces, can be performed in a manner analogous to Equation (14), and subject to boundary conditions given in 8.6.

4.3 Visible transmittance

The visible transmittance of the total fenestration product is:

$$\tau_t = \frac{\sum \tau_v A_{gv}}{A_t} \tag{16}$$

5 Vision area properties

5.1 Glazing layer optics

5.1.1 General

For glazing units only, the optical properties can be determined using ISO 9050. Clause 7 contains the extensions needed to model vented windows.

5.1.2 Solar

The solar optical properties needed to describe the i th glazing are: a) the front (external side) spectral reflectance, $r_{ft,i}(\lambda_w)$; b) the back (internal side) spectral reflectance, $r_{b,i}(\lambda_w)$; and c) the spectral transmittance, $\tau_i(\lambda_w)$. See Figure 4.

NOTE More information about $r_{ft,i}(\lambda_w)$, $r_{b,i}(\lambda_w)$ and $\tau_i(\lambda_w)$ can be found in [2].

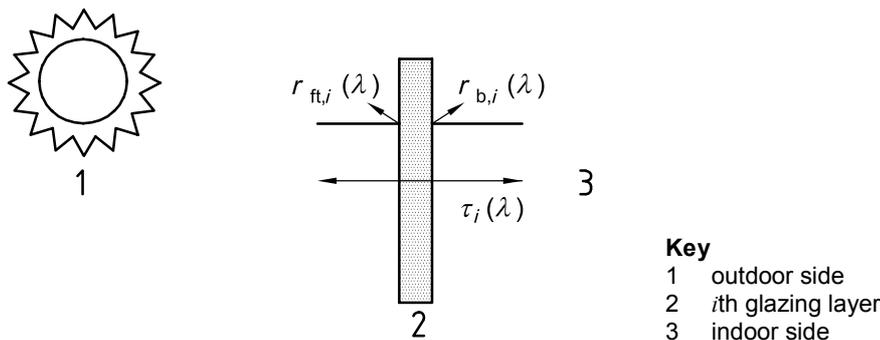


Figure 4 — Outdoor and internal spectral transmittance of a glazing layer

The solar optical data shall be measured in accordance with ISO 9050. Intermediate values of $r_{f,i}(\lambda_w)$, $r_{b,i}(\lambda_w)$ or $\tau_i(\lambda_w)$ are found by linear interpolation.

5.1.3 Long-wave

The long-wave optical properties needed to describe the i th glazing are: a) the front (external side) hemispheric emissivity, $\varepsilon_{ft,i}$; b) the back (internal side) hemispheric emissivity, $\varepsilon_{b,i}$; and c) the hemispheric-hemispheric transmittance, τ_i . These total optical properties apply to wavelengths from 5 μm to 50 μm .

The long-wave reflectance data shall be measured in accordance with EN 12898. Values of the normal emissivity resulting from this procedure shall be converted to hemispherical emissivity using the procedure described in [3] or in EN 12898. The integration needed to convert measured spectral data to the required total longwave optical properties, $\varepsilon_{ft,i}$, $\varepsilon_{b,i}$ and τ_i shall be carried out in accordance with [3] or EN 12898.

Some windows are constructed with suspended or stretched layers of thin plastic film between glass panes to make triple or quadruple glazing. When these layers are covered with a low-emissivity coating they are generally opaque in the infrared, so that $\tau_i = 0$ and hemispherical emissivity can be calculated as in [3]. For partially transparent films such as polyethylene terephthalate (PET), both specular transmittance and reflectance should be measured. Using the bulk model, the optical indices of the material can then be calculated and used to derive the hemispherical properties.

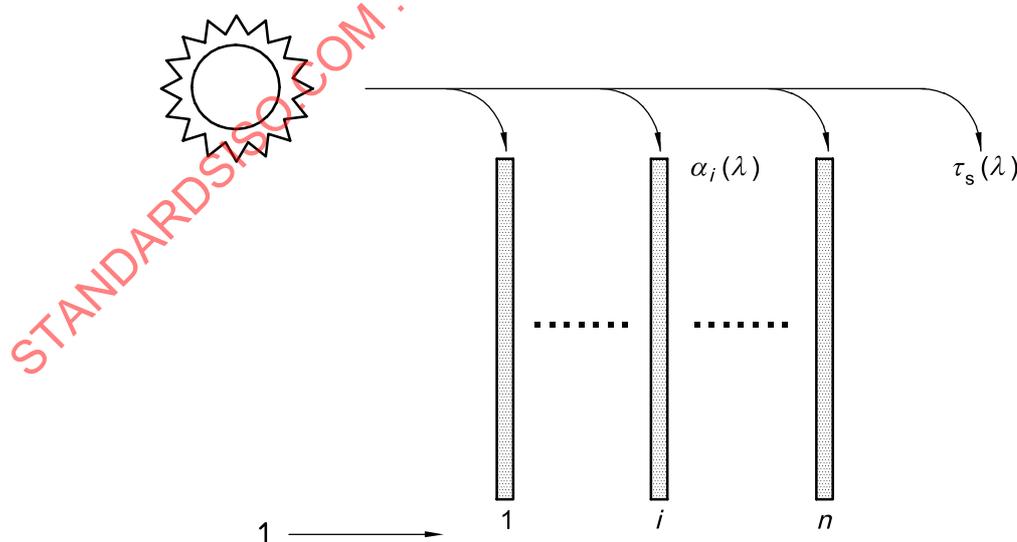
NOTE See [4] for more information.

5.2 Glazing system optics

5.2.1 Spectral quantities

The path of incident solar radiation within the various layers of the glazing system shall be modelled by the methods described in ISO 9050 or by any other exact method.

NOTE Depending on future modifications of ISO 9050, specific additions may be added to this International Standard covering the effects of optical properties of products (shading devices, diffusing panes, etc.) not adequately covered by ISO 9050.



Key

1 glazing layer

Figure 5 — Absorption of the i th glazing layer and solar spectral transmittance

Figure 5 shows how a window with n glazing layers together with the external ($i = 0$) and internal ($i = n + 1$) spaces can be treated as an $n + 2$ element array. It is necessary to determine the portion of incident solar radiation, at a given wavelength, that is absorbed at each of the glazing layers. This quantity is denoted $\alpha_i(\lambda_w)$ at the i th glazing layer. Similarly, it is necessary to determine the solar spectral transmittance of the glazing system, $\tau_s(\lambda_w)$.

These quantities, $\alpha_i(\lambda_w)$ and $\tau_s(\lambda_w)$, shall be calculated in accordance with ISO 9050 while setting the reflectance of the conditioned space to zero. Any other method that can be shown to provide an exact solution is acceptable.

NOTE The solution technique described in [5] is summarized in Annex A.

5.2.2 The solar spectrum

The spectral distribution of the incident solar radiation, $E(\lambda_w)$, is needed to calculate total optical properties and various total energy flow. Values of $E(\lambda_w)$ are reported at N_{sl} values of λ_w (denoted here as $E(\lambda_{wj})$ and λ_{wj} , respectively). Intermediate values of $E(\lambda_w)$ shall be found by linear interpolation of the tabulated values.

5.2.3 Absorbed amounts of solar radiation

The total flow rate of solar radiation absorbed at the i th glazing layer, S_i , is determined by numerical integration over the solar spectrum according to Equations (16), (17) and (18).

$$A_i = \frac{\sum_{j=1}^{N_{sl}-1} \alpha_i(\lambda_{w\ j\ j+1}) E_{sl}(\lambda_{w\ j\ j+1}) \Delta\lambda_{wj}}{\sum_{j=1}^{N_{sl}-1} E_{sl}(\lambda_{w\ j\ j+1}) \Delta\lambda_{wj}} \quad (17)$$

$$\Delta\lambda_{wj} = \lambda_{wj+1} - \lambda_{wj} \quad (18)$$

$$S_i = A_i \times I_{sl} \quad (19)$$

where A_i is the portion of the total incident solar radiation (on the glazing system) that is absorbed by the i th glazing layer, and $\alpha_i(\lambda_{w\ j\ j+1})$ is the value of α_i that is representative of the wavelength band from λ_{wj} to λ_{wj+1} and is given by

$$\alpha_i(\lambda_{w\ j\ j+1}) = \frac{1}{2} \alpha_i(\lambda_{w\ j}) + \frac{1}{2} \alpha_i(\lambda_{w\ j+1}) \quad (20)$$

and

$$E_s(\lambda_{w\ j\ j+1}) = \frac{E_s(\lambda_{w\ j}) + E_s(\lambda_{w\ j+1})}{2} \quad (21)$$

Values of $E_s(\lambda_w)$ are given in ISO 9845-1.

5.2.4 Solar transmittance

The solar transmittance of the glazing system is:

$$\tau_{\text{sl}} = \frac{\sum_{j=1}^{N_{\text{sl}}-1} \tau_{\text{sl}}(\lambda_{\text{w } j/j+1}) E_{\text{s}}(\lambda_{\text{w } j/j+1}) \Delta\lambda_{\text{w } j}}{\sum_{j=1}^{N_{\text{sl}}-1} E_{\text{s}}(\lambda_{\text{w } j/j+1}) \Delta\lambda_{\text{w } j}} \quad \Delta\lambda_{\text{w } j} = \lambda_{\text{w } j+1} - \lambda_{\text{w } j} \quad (22)$$

where $E_{\text{s}}(\lambda_{\text{w } j/j+1})$ is given by Equation (21) and

$$\tau_{\text{sl}}(\lambda_{\text{w } j/j+1}) = \frac{1}{2} \tau_{\text{sl}}(\lambda_{\text{w } j}) + \frac{1}{2} \tau_{\text{sl}}(\lambda_{\text{w } j+1}) \quad (23)$$

5.2.5 Visible transmittance

Visible transmittance, τ_{vs} , is calculated using a weighting function that represents the photopic response of the eye, $R(\lambda_{\text{w}})$. $R(\lambda_{\text{w}})$ is tabulated for N_{vs} values of $\lambda_{\text{w } j}$. τ_{vs} is given by:

$$\tau_{\text{vs}} = \frac{\sum_{j=1}^{N_{\text{vs}}-1} \tau_{\text{sl}}(\lambda_{\text{w } j/j+1}) E_{\text{vs}}(\lambda_{\text{w } j/j+1}) R(\lambda_{\text{w } j/j+1}) \Delta\lambda_{\text{w } j}}{\sum_{j=1}^{N_{\text{vs}}-1} E_{\text{vs}}(\lambda_{\text{w } j/j+1}) R(\lambda_{\text{w } j/j+1}) \Delta\lambda_{\text{w } j}} \quad \Delta\lambda_{\text{w } j} = \lambda_{\text{w } j+1} - \lambda_{\text{w } j} \quad (24)$$

where

$$R(\lambda_{\text{w } j/j+1}) = \frac{R(\lambda_{\text{w } j}) + R(\lambda_{\text{w } j+1})}{2} \quad (25)$$

$$E_{\text{vs}}(\lambda_{\text{w } j/j+1}) = \frac{1}{2} E_{\text{vs}}(\lambda_{\text{w } j}) + \frac{1}{2} E_{\text{vs}}(\lambda_{\text{w } j+1}) \quad (26)$$

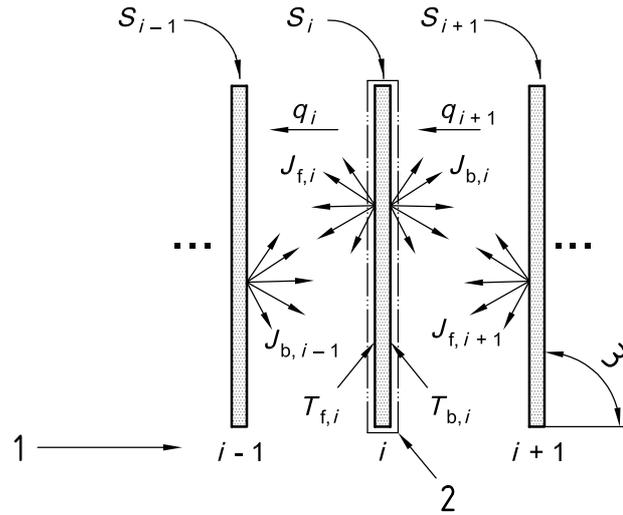
Values of $E_{\text{v}}(\lambda_{\text{w}})$ are given in ISO/CIE 10526.

and $\tau_{\text{sl}}(\lambda_{\text{w } j/j+1})$ is given by Equation (23).

5.3 Vision area heat transfer

5.3.1 Glazing layer energy balance

Longwave radiative exchange between glazing layers and conductive heat transfer within each glazing layer can be described using fundamental relations. Calculations dealing with convective heat transfer depend upon correlations based on experimental data.



- Key**
- 1 glazing layer
 - 2 control volume
 - 3 slope

Figure 6 — Energy balance on glazing layer *i*

Figure 6 shows the *i*th glazing in a sloped multilayer array. The values of four variables are sought at each glazing. These are the temperatures of the external and internal facing surfaces, $T_{ft,i}$ and $T_{b,i}$ plus the radiant heat leaving the front and back facing surfaces (i.e., the radiosities), $J_{ft,i}$ and $J_{b,i}$. In terms of these variables, q_i is:

$$q_i = h_{cv,i} (T_{ft,i} - T_{b,i-1}) + J_{ft,i} - J_{b,i-1} \tag{27}$$

The solution is generated by applying the following four equations at each glazing:

$$q_i = S_i + q_{i+1} \tag{28}$$

$$J_{ft,i} = \varepsilon_{ft,i} \sigma T_{ft,i}^4 + \tau_i J_{ft,i+1} + r_{ft,i} J_{b,i-1} \tag{29}$$

$$J_{b,i} = \varepsilon_{b,i} \sigma T_{b,i}^4 + \tau_i J_{b,i-1} + r_{b,i} J_{ft,i+1} \tag{30}$$

$$T_{b,i} - T_{ft,i} = \frac{t_{gv,i}}{2\lambda_{gv,i}} (2q_{i+1} + S_i) \tag{31}$$

A by-product of the analysis is the temperature profile through each glazing.

$$T_i(z) = \left(\frac{-S_i}{2\lambda_{gv,i} t_{gv,i}} \right) z^2 + \left(\frac{T_{ft,i} - T_{b,i}}{t_{gv,i}} + \frac{S_i}{2\lambda_{gv,i}} \right) z + T_{b,i} \tag{32}$$

where z is the distance from the internal surface of the glazing, positive in the direction of the external side.

Equation (28) describes an energy balance imposed at the surfaces of the *i*th glazing. Equations (29) and (30) define the radiosities at the *i*th glazing, and $r_{ft,i} = 1 - \tau_i - \varepsilon_{ft,i}$ and $r_{b,i} = 1 - \tau_i - \varepsilon_{b,i}$.

The temperature difference across the *i*th glazing is given by Equation (31). It is assumed that the solar energy is absorbed uniformly through the thickness of the glazing.

NOTE More details regarding Equation (27) to Equation (32) are given in References [5] and [35].

5.3.2 Interaction with the environment

The effect of boundary conditions imposed by the environment on the window shall be specified. The internal and external temperatures, $T_{ft,n+1}$ and $T_{b,0}$ are:

$$T_{ft,n+1} = T_{ai,int} \quad (33)$$

$$T_{b,0} = T_{ai,ex} \quad (34)$$

The effect of long-wave irradiance at internal and external glazing surfaces is included by setting

$$J_{ft,n+1} = E_{gv,int} \quad (35)$$

$$\text{and } J_{b,0} = E_{gv,ex} \quad (36)$$

where $E_{gv,int}$ and $E_{g,ex}$ are given by Equations (159) and (152) in Clause 8, respectively.

The effect of the convective heat transfer coefficients at the glazing surfaces is included by setting

$$h_{cv,n+1} = h_{cv,int} \quad (37)$$

$$h_{cv,1} = h_{cv,ex} \quad (38)$$

5.3.3 Convective heat transfer coefficient — glazing cavities

5.3.3.1 General

Convective heat transfer coefficients for the fill gas layers are determined in terms of the dimensionless Nusselt number, Nu_i :

$$h_{cv,i} = Nu_i \left(\frac{\lambda_{gv,i}}{d_{gv,i}} \right) \quad (39)$$

where $d_{gv,i}$ is the thickness of the fill gas layer (or pane spacing) i , and $\lambda_{gv,i}$ is the thermal conductivity of the fill gas. Nu_i is calculated using correlations based on experimental measurements of heat transfer across inclined air layers. Nu_i is a function of the Rayleigh number, Ra_i , the cavity aspect ratio, $A_{gv,i}$, and the cavity slope, γ .

It should be recognized that deflection of the panes in high aspect ratio cavities can occur. This deflection may increase or decrease the average cavity width, d . This deflection can be caused by changes in the cavity average temperature, changes in the cavity moisture content, nitrogen absorption by the desiccant or changes in the barometric pressure (due to elevation and/or weather changes) from the conditions during assembly.

NOTE Reference [6] discusses the effects of glass pane deflection and methods to estimate the change in the thermal transmittance due to this deflection.

The Rayleigh number can be expressed as (omitting the “ i ” and “ gv ” subscripts for convenience):

$$Ra = \frac{\rho^2 d^3 g \beta c_p \Delta T}{\mu \lambda} \quad (\text{dimensionless}) \quad (40)$$

Treating the fill gas as a perfect gas, the thermal expansion coefficient of the fill gas, β , is:

$$\beta = \frac{1}{T_m} \quad (41)$$

where T_m = fill gas mean temperature in kelvins.

The aspect ratio of the fill gas cavity i , is:

$$A_{gv,i} = \frac{H}{d_{gv,i}} \quad (42)$$

where H is the distance between the top and bottom of the fill gas cavity which is usually the same as the height of the window view area.

Correlation to quantify convective heat transfer across glazing cavities is presented in the 5.3.3.2 to 5.3.3.6. Each of these subclauses pertains to one particular value, or range, of tilt angle, γ .

This categorization, as a function of γ , is based on the assumption that the cavity is heated from the internal side (i.e., $T_{ft,i} > T_{b,i-1}$). If the reverse is true ($T_{ft,i} < T_{b,i-1}$), it is necessary to seek the appropriate correlation on the basis of the complement of the tilt angle, $180^\circ - \gamma$, instead of γ and to then substitute $180^\circ - \gamma$ instead of γ when the calculation is carried out.

$\gamma = 0$ is horizontal glazing, heat flow upwards

$\gamma = 90$ is vertical glazing, heat flow upwards

$\gamma = 180$ is horizontal glazing, heat flow downwards

5.3.3.2 Cavities inclined at $0 \leq \gamma < 60^\circ$

$$Nu_i = 1 + 1,44 \left(1 - \frac{1708}{Ra \cos(\gamma)} \right)^\bullet \left[1 - \frac{(1708 \sin^{1,6}(1,8\gamma))}{Ra \cos(\gamma)} \right] + \left[\left(\frac{Ra \cos(\gamma)}{5\,830} \right)^{1/3} - 1 \right]^\bullet \quad (43)$$

$$Ra < 10^5 \text{ and } A_{gv,i} > 20$$

$$\text{where } (X)^\bullet = \frac{(X + |X|)}{2} \quad (44)$$

NOTE For more details, see Reference [7].

5.3.3.3 Cavities inclined at $\gamma = 60^\circ$

$$Nu = (Nu_1, Nu_2)_{\max} \quad (45)$$

$$\text{where } Nu_1 = \left[1 + \left(\frac{0,093\,6 Ra^{0,314}}{1 + G} \right)^7 \right]^{1/7} \quad (46)$$

$$Nu_2 = \left(0,104 + \frac{0,175}{A_{g,i}} \right) Ra^{0,283} \quad (47)$$

$$G = \frac{0,5}{\left[1 + \left(\frac{Ra}{3\,160}\right)^{20,6}\right]^{0,1}} \quad (48)$$

NOTE For more details, see Reference [8].

5.3.3.4 Cavities inclined at $60^\circ < \gamma < 90^\circ$

For layers inclined at angles between 60° and 90° , a straight-line interpolation between the results of Equations (45) and (49) is used. These equations are valid in the ranges of:

$$10^2 < Ra < 2 \times 10^7 \text{ and } 5 < A_{gv,i} < 100$$

NOTE For more details, see Reference [8].

5.3.3.5 Vertical cavities

$$Nu = (Nu_1, Nu_2)_{\max} \quad (49)$$

$$Nu_1 = 0,067\,383\,8Ra^{1/3} \quad 5 \times 10^4 < Ra \quad (50)$$

$$Nu_1 = 0,028\,154Ra^{0,413\,4} \quad 10^4 < Ra \leq 5 \times 10^4 \quad (51)$$

$$Nu_1 = 1 + 1,759\,667\,8 \times 10^{-10} Ra^{2,298\,475\,5} \quad Ra \leq 10^4 \quad (52)$$

$$Nu_2 = 0,242 \left(\frac{Ra}{A_{gv,i}} \right)^{0,272} \quad (53)$$

NOTE For more details, see Reference [9].

5.3.3.6 Cavities inclined from 90° to 180°

Gas layers contained in downward facing windows are modelled using:

$$Nu = 1 + (Nu_v - 1) \sin \gamma \quad (54)$$

Nu_v is the Nusselt number for a vertical cavity given by Equation (49).

NOTE For more details, see Reference [10].

5.3.3.7 Fill gas properties

The density of fill gases in windows is calculated using the perfect gas law.

$$\rho = \frac{PM}{\Re T_m} \quad (55)$$

$$P = 101\,300 \text{ Pa and } T_m = 293 \text{ K.}$$

The specific heat capacity at constant pressure, c_p , and the transport properties μ and λ are evaluated using linear functions of temperature, e.g., the viscosity can be expressed as:

$$\mu = a + bT_m \quad (56)$$

Values of a and b coefficients appropriate for calculating c_p , μ and λ for a variety of fill gases are given in Annex B.

5.3.4 Properties of fill gas mixtures

5.3.4.1 General

The density, conductivity, viscosity and specific heat capacity of gas mixtures can be calculated as a function of the corresponding properties of the individual constituents.

NOTE For more details, see Reference [11].

5.3.4.2 Molecular mass

$$\hat{M}_{\text{mix}} = \sum_{i=1}^v m_i \hat{M}_i \quad (57)$$

where m_i is the mole fraction of the gas component i , in a mixture of gases.

5.3.4.3 Density

$$\rho = \frac{P \hat{M}_{\text{mix}}}{\Re T_m} \quad (58)$$

5.3.4.4 Specific heat

$$c_{p \text{ mix}} = \frac{\hat{c}_{p \text{ mix}}}{\hat{M}_{\text{mix}}} \quad (59)$$

where

$$\hat{c}_{p \text{ mix}} = \sum_{i=1}^v x_i \hat{c}_{p,i} \quad (60)$$

and the molar specific heat capacity of the i^{th} gas is

$$\hat{c}_{p,i} = c_{p,i} \hat{M}_i \quad (61)$$

5.3.4.5 Viscosity

$$\mu_{\text{mix}} = \frac{\sum_{i=1}^v \mu_i}{\left[1 + \sum_{\substack{j=1 \\ j \neq i}}^v \phi_{i,j}^{\mu} \frac{m_j}{m_i} \right]} \quad (62)$$

where

$$\phi_{i,j}^{\mu} = \frac{\left[1 + (\mu_i / \mu_j)^{1/2} (\hat{M}_j / \hat{M}_i)^{1/4} \right]^2}{2\sqrt{2} \left[1 + (\hat{M}_i / \hat{M}_j) \right]^{1/2}} \quad (63)$$

5.3.4.6 Thermal conductivity

$$\lambda_{\text{mix}} = \lambda'_{\text{mix}} + \lambda''_{\text{mix}} \quad (64)$$

where

λ' is the monatomic thermal conductivity;

λ'' is the additional energy moved by the diffusional transport of internal energy in polyatomic gases.

$$\lambda'_{\text{mix}} = \sum_{i=1}^v \frac{\lambda'_i}{\left\{ 1 + \sum_{\substack{j=1 \\ j \neq i}}^v \Psi_{i,j} \frac{m_j}{m_i} \right\}} \quad (65)$$

and

$$\Psi_{i,j} = \frac{\left[1 + (\lambda'_i / \lambda'_j)^{1/2} (\hat{M}_i / \hat{M}_j)^{1/4} \right]^2}{2\sqrt{2} \left[1 + (\hat{M}_i / \hat{M}_j) \right]^{1/2}} \times \left[1 + 2,41 \frac{(\hat{M}_i - \hat{M}_j)(\hat{M}_i - 0,142\hat{M}_j)}{(\hat{M}_i + \hat{M}_j)^2} \right] \quad (66)$$

and

$$\lambda''_{\text{mix}} = \sum_{i=1}^v \frac{\lambda''_i}{\left\{ 1 + \sum_{\substack{j=1 \\ j \neq i}}^v \phi_{i,j}^{\lambda} \frac{m_j}{m_i} \right\}} \quad (67)$$

where the previous expression for $\phi_{i,j}$ can also be written as

$$\phi_{i,j}^{\lambda} = \frac{\left[1 + (\lambda'_i / \lambda'_j)^{1/2} (\hat{M}_i / \hat{M}_j)^{1/4} \right]^2}{2\sqrt{2} \left[1 + (\hat{M}_i / \hat{M}_j) \right]^{1/2}} \quad (68)$$

To find λ_{mix} , use the following steps.

a) Calculate λ'_i

$$\lambda'_i = \frac{15}{4} \times \frac{\mathfrak{R}}{\hat{M}_i} \mu_i \quad (69)$$

b) Calculate λ''_i

$$\lambda''_i = \lambda_i - \lambda'_i$$

λ_i is the conductivity of the i th fill gas component (see Annex B) (70)

c) Use λ'_i to calculate λ'_{mix}

d) Use λ''_i to calculate λ''_{mix}

e) $\lambda_{\text{mix}} = \lambda'_{\text{mix}} + \lambda''_{\text{mix}}$ (71)

6 Frame effects

6.1 Area and linear thermal transmittance

Frame regions of the fenestration system consist of opaque areas that may or may not contain air cavities. Frames can be made from a variety of materials, but most common materials are wood, vinyl, aluminium and combinations of those (e.g., vinyl-clad wood).

Frame area thermal transmittance (thermal transmittances) and linear thermal transmittance, Ψ , shall be determined using two-dimensional numerical modelling. This two-dimensional analysis shall provide the rate of heat transfer through each unique frame section. See national standards for the required cross sections to be considered. Details regarding the required two dimensional numerical analysis are provided in 6.2 to 6.6 (also see ISO 10077-2).

6.2 Governing equations for calculating thermal transmittance

The governing equation shall be developed by imposing an energy balance describing steady-state heat transfer by conduction. The governing equation shall be discretized using a conservative formulation (i.e., the evaluation of energy flow between two specific nodes or across any given control volume face shall be done in a consistent manner throughout the analysis). The frame/edge-glass geometry and the corresponding thermal conductivity for each of the various materials, λ_{fe} , shall be specified. The numerical solver shall be able to generate the two-dimensional heat flow and temperature patterns that satisfy the governing equation. In Cartesian co-ordinates, this equation is:

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} = -\frac{q_{sc}'''}{\lambda} \quad (72)$$

where q_{sc}''' represents the internal heat generation in watts per square metre.

The density of heat flow rate, q , is conserved across any surface where two materials meet and is given by:

$$q = -\lambda \left(\frac{\partial T}{\partial x} e_x + \frac{\partial T}{\partial y} e_y \right) \quad (73)$$

where e_x and e_y are the components of the normal vector to the surface.

At the boundary, the density of heat flow rate, q , is equal to:

$$q = q_{cv} + q_r + q_{sc}(q_{sk}) \quad (74)$$

where q_{cv} is convective component, and q_r is radiative component of the density of heat flow rate, which shall be determined in accordance with 8.3 and 8.4 respectively. Quantity $q_{sc}(q_{sk})$ is the prescribed density of heat flow rate at the boundary (source or sink).

6.3 Geometric representation and meshing

6.3.1 Geometric representation

A two-dimensional representation or model of each frame, sash and edge-glazing assembly shall be made. The dimensions of all parts shall be the nominal values as given on the manufacturer's drawings, provided that these drawings are an actual representation of the fenestration product. Small radii and minor variations in material thicknesses due to manufacturing tolerances or strengthening/attachment requirements may be ignored.

Reinforcing or operating hardware that is essentially continuous, assembly screws or bolts that extend from the internal to the external side or bridge a thermal break, including any incompletely de-bridged thermal break, shall be included in the model. These thermal-bridging elements may be modelled with three-dimensional computational tools when available, otherwise they shall be modelled using the procedure outlined below:

NOTE For more details, see Reference [34].

Calculate the effective conductivity of thermal bridging elements (e.g., bolts, screws, etc.)

$$\lambda_{\text{eff}} = F_{\text{br}} \times \lambda_{\text{br}} + (1 - F_{\text{br}}) \lambda_{\text{o}} \quad (75)$$

where

$$F_{\text{br}} = s/w$$

s is the size of thermal bridging element (e.g., size of a bolt head);

w is the spacing of thermal bridging elements;

λ_{br} is the conductivity of thermal bridging material;

λ_{o} is the conductivity of the cross-section without the thermal bridge.

Use the following criteria to determine if it is necessary to apply the above procedure.

- a) If $F_{\text{br}} \leq 1\%$, ignore thermal bridge;
- b) If $1\% < F_{\text{br}} \leq 5\%$, model using the above method providing that $\lambda_{\text{br}} > 10 \lambda_{\text{o}}$;
- c) If $F_{\text{br}} > 5\%$, always model using the above method.

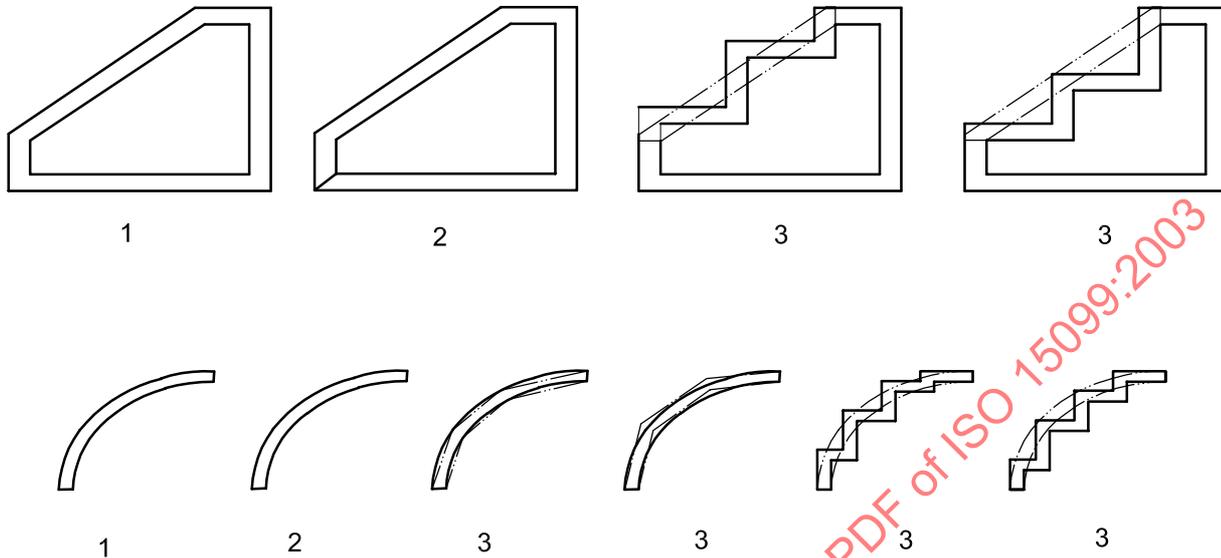
Components in the window assembly, which are compressed or deformed from their original shape once installed in the window (e.g. weather-stripping), shall be modelled in the compressed or deformed configuration. Adjustments to the dimensions of the geometric model are allowed only if they have no significant influence on the calculation (see ISO 10211-1).

Those segments of the actual cross section that are made up of vertical and horizontal surfaces shall be represented by similar straight lines that preserve the nominal thickness and relative position of the segment. Sloped lines shall be represented by a) similar sloped lines that preserve the nominal thickness and relative position of the segment or by b) a series of horizontal and vertical lines, which meet criteria 1 to 4 below. Curves shall be represented by a) similar curves that preserve the nominal thickness and relative position of the segment or by b) a series of horizontal, vertical, and sloped lines or a series of horizontal and vertical lines which meet criteria 1 to 4 below (see Figure 7).

- 1) The thickness of the representation, d , is equal to the average thickness.
- 2) All points on the represented line are within 5 mm of the actual line/curve. The averaged distance (for all points) between the represented line and the actual line/curve is less than 2,5 mm.
- 3) For conductive materials (materials where the conductivity is 10 times or more than that of any surrounding material), the path length shall be maintained to within 5%. If this condition is not possible, then the product of web thickness times conductivity shall be replaced by web thickness times conductivity times $(\cos \gamma + \sin \gamma)$, where γ is the angle of inclination of the sloped web. The same result is obtained whether γ is measured from the vertical or the horizontal reference.
- 4) When sloped materials are represented by a series of rectangles, the contact length between adjacent rectangles or polygons, l , is equal to the average actual thickness, d .

Some windows have nailing flanges that are used to help secure the window in the rough opening. If these flanges are intended to be covered up by the exterior cladding (e.g., siding or brick), the portion of the flange extending outside the rough opening shall be ignored.

In most cases, the internal and external boundaries shall follow the frame profile. In the case where there are exterior and interior open frame profile cavities (ventilated cavities and groves) the procedure described in 6.7 shall be followed.



- Key**
- 1 actual
 - 2 preferred
 - 3 acceptable

Figure 7 — Examples of possible approximations of the actual cross section

6.3.2 Meshing

The two-dimensional geometric model shall be divided or meshed into a series of small elements in order to provide an accurate representation of the heat flow patterns and temperature distributions. Mesh resolution shall be sufficient to ensure that the combined frame/edge thermal transmittance for each cross-section, obtained by solving the governing two-dimensional heat transfer equation given in 6.2, shall be within 1 % of the combined frame/edge thermal transmittance obtained from an ideal (i.e., infinitely fine) mesh. Acceptable meshing schemes include the following:

- a) **Successive refinements:** The governing heat transfer equation is solved for thermal transmittance for a given meshing arrangement. The mesh is made finer either uniformly or in regions of high two-dimensional heat flow and a new thermal transmittance determined. An extrapolation is made to the thermal transmittance with an infinite number of nodes. The mesh is fine enough when the calculated thermal transmittance is within 1 % of the extrapolated thermal transmittance.

NOTE This requirement is more stringent than that specified in ISO 10211-1, which requires that the number of subdivisions be doubled until the change in heat flow through the object is reduced to a prescribed tolerance. The more stringent criterion, specified above, is now possible with the increase of computing power. Finite element and finite volume methods, with unstructured (non-rectangular) meshes, can also meet this more stringent criterion using error estimation methods such as the one given in b) below.

- b) Energy error norm [13, 14] is applied so that the calculated frame/edge thermal transmittance is within 1 % of the thermal transmittance determined with an ideal mesh.
- c) Any other approach documented in refereed publications applied so that the calculated frame/edge thermal transmittance is within 1 % of the thermal transmittance determined with an ideal mesh.

6.4 Solid materials

The thermal conductivity values are usually taken from national standards. Where this is not the case the values listed in ISO 10077-2 may be used, but only if they directly match the materials used in the window construction. If neither of these sources is used, the thermal conductivity values are to be determined in accordance with ISO 8302 (guarded hot plate) or ISO 8301 (heat flow meter) at a mean temperature appropriate to national standards. It is assumed that all material thermal conductivity values are constant with respect to temperature.

The surface emissivity values of frame materials are usually taken from national standards. Where this is not the case surface emissivity values shall be determined in accordance with ISO 10077-2, but only if they directly match the materials used in the window construction.

6.5 Effective conductivity — Glazing cavities

Cavities shall be treated as if they contain an opaque solid with an effective conductivity. The effective conductivity of a given cavity shall be calculated using the results of the vision area analysis. At cavity i :

$$\lambda_{\text{eff},i} = q_i \left(\frac{d_{\text{gv},i}}{T_{\text{ft},i} - T_{\text{b},i-1}} \right) \quad (76)$$

6.6 Effective conductivity — Unventilated frame cavities

6.6.1 General

A frame cavity shall be treated as though it contains an opaque solid, which is assigned an effective conductivity. This effective conductivity accounts for both radiative and convective heat transfer and shall be determined as follows.

$$\lambda_{\text{eff}} = (h_{\text{cv}} + h_{\text{r}}) \times d \quad (77)$$

where

λ_{eff} is the effective conductivity;

h_{cv} is the convective heat transfer coefficient;

h_{r} is the radiative heat transfer coefficient ($h_{\text{r}} = 0$ in the case when detailed radiation procedure is used);

d is the thickness or width of the air cavity in the direction of heat flow.

The convective heat transfer coefficient, h_{cv} , is calculated from the Nusselt number, Nu , which can be determined from various correlations, depending on aspect ratio, orientation and direction of heat flow.

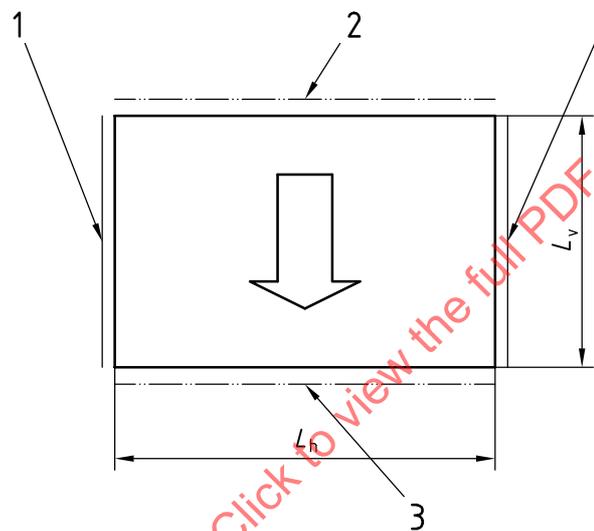
$$h_{cv} = Nu \frac{\lambda_{ai}}{d} \tag{78}$$

There are three different cases to be considered, depending on whether the heat flow is upward, downward, or horizontal.

6.6.2 Heat flow downward

$$Nu = 1,0 \tag{79}$$

See Figure 8.

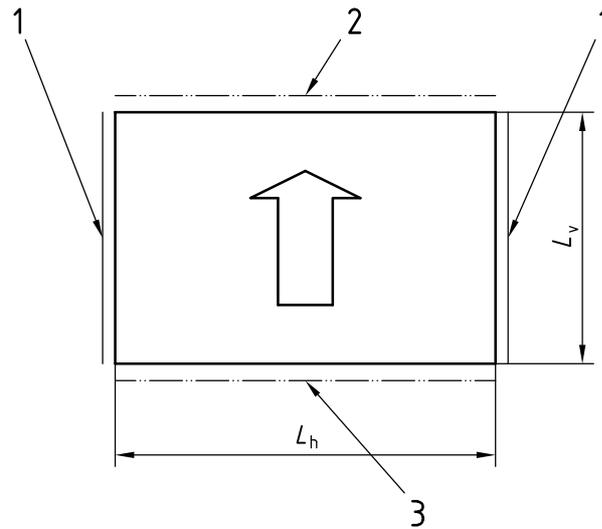


- Key**
- 1 $q = 0$
 - 2 T_{ch}
 - 3 T_{cc}

Figure 8 — Illustration of rectangular frame cavity downward flow direction

6.6.3 Heat flow upward

This situation is inherently unstable and will yield a Nusselt number that is dependent on the height-to-width aspect ratio, L_v/L_h , where L_v and L_h are the largest cavity dimensions in the vertical and horizontal directions, see Figure 9.



Key

- 1 $q = 0$
- 2 T_{cc}
- 3 T_{ch}

Figure 9 — Illustration of rectangular frame upward flow direction

- a) for $\frac{L_h}{L_v} \leq 1$ convection is restricted by wall friction, and

$$Nu = 1,0 \tag{80}$$

- b) for $1 < \frac{L_h}{L_v} \leq 5$ the Nusselt number is calculated according to the method given by

$$Nu = 1 + \left(1 - \frac{Ra_{cr}}{Ra} \right)^{\bullet} \left[k1 + 2(k2)^{1-\ln k2} \right] + \left[\left(\frac{Ra}{5830} \right)^{1/3} - 1 \right]^{\bullet} \left[1 - e^{-0,95 \left(\left(\frac{Ra_{cr}}{Ra} \right)^{1/3} - 1 \right)^{\bullet}} \right] \tag{81}$$

where

$$k1 = 1,40 \tag{82}$$

$$k2 = \frac{Ra^{1/3}}{450,5} \tag{83}$$

$$(X)^{\bullet} = \frac{X + |X|}{2} \tag{84}$$

Ra_{cr} is a critical Rayleigh number, which is found by least squares regression of tabulated values.

NOTE For more details, see Reference [15].

$$Ra_{cr} = e^{\left(0,721\frac{L_v}{L_h}\right) + 7,46} \quad (85)$$

Ra is the Rayleigh number for the air cavity:

$$Ra = \frac{\rho_{ai}^2 L_v^3 g \beta c_{p,ai} (T_{ch} - T_{cc})}{\mu_{ai} \lambda_{ai}} \quad (86)$$

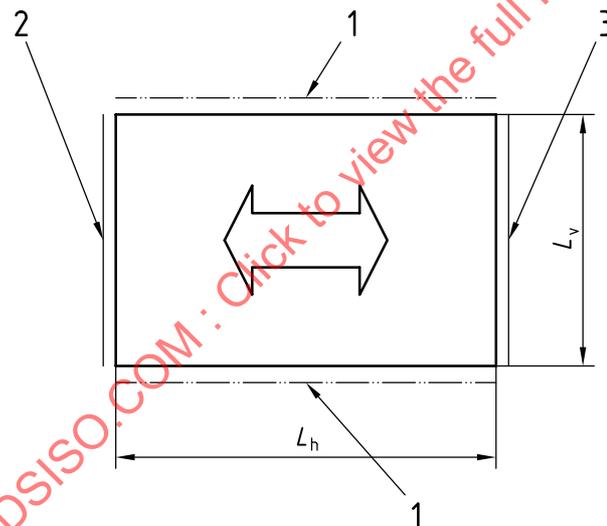
c) for $\frac{L_h}{L_v} > 5$ the Nusselt number is:

$$Nu = 1 + 1,44 \left(1 - \frac{1708}{Ra}\right)^{\bullet} + \left[\left(\frac{Ra}{5830}\right)^{1/3} - 1\right]^{\bullet} \quad (87)$$

NOTE For more details, see Reference [7].

6.6.4 Horizontal heat flow

See Figure 10.



Key

- 1 $q = 0$
- 2 T_{ch} or T_{cc}
- 3 T_{cc} or T_{ch}

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Figure 10 — Illustration of rectangular frame cavity horizontal flow direction

a) for $\frac{L_v}{L_h} < \frac{1}{2}$ the Nusselt number is:

$$Nu = 1 + \left\{ \left[2,756 \times 10^{-6} Ra^2 \left(\frac{L_v}{L_h}\right)^8 \right]^{-0,386} + \left[0,623 Ra^{1/5} \left(\frac{L_h}{L_v}\right)^{2/5} \right]^{-0,386} \right\}^{-2,59} \quad (88)$$

NOTE For more details, see Reference [15].

where Ra is Rayleigh number and is defined as:

$$Ra = \frac{\rho_{ai}^2 L_h^3 g \beta_{c,p,ai} (T_{ch} - T_{cc})}{\mu_{ai} \lambda_{ai}} \quad (89)$$

b) for $\frac{L_v}{L_h} > 5$ the following correlation, also the maximum $Nu [= (Nu_1, Nu_2, Nu_3)_{max}]$ is given as:

$$Nu_1 = \left\{ 1 + \frac{\left(0,104 Ra^{0,293} \right)^3}{\left[1 + \left(\frac{6\,310}{Ra} \right)^{1,36} \right]} \right\}^{1/3} \quad (90)$$

$$Nu_2 = 0,242 \left(Ra \frac{L_h}{L_v} \right)^{0,273} \quad (91)$$

$$Nu_3 = 0,060\,5 Ra^{1/3} \quad (92)$$

NOTE For more details, see Reference [9].

c) for $\frac{1}{2} < \frac{L_v}{L_h} < 5$ the Nusselt number is found using a linear interpolation between the endpoints of a) and b) above.

For jamb frame sections, frame cavities are oriented vertically and therefore the height of the cavity is in the direction normal to the plane of the cross section. For these cavities it is assumed that heat flow is always in horizontal direction with $L_v/L_h > 5$, and so correlations in Equations (90) to (92) in 6.6.4 b) shall be used.

The temperatures T_{ch} and T_{cc} are not known in advance, so it is necessary to estimate them. From previous experience it is recommended to apply $T_{ch} = 10 \text{ °C}$ and $T_{cc} = 0 \text{ °C}$. However, after the simulation is done, it is necessary to update these temperatures from the results of the previous run. This procedure shall be repeated until values of T_{ch} and T_{cc} from two consecutive runs are within 1 °C . Also, it is important to inspect the direction of heat flow after the initial run, because if the direction of the bulk of heat flow is different than initially specified, it will need to be corrected for the next run.

For an unventilated, irregularly shaped frame cavity, the geometry shall be converted into the equivalent of a rectangular cavity in accordance with the procedure in ISO 10077-2 (see also Figure 11). For these cavities, the following procedure shall be used to determine which surfaces belong to vertical and horizontal surfaces of equivalent rectangular cavity (see also Figure 12).

If the shortest distance between two opposite surfaces is smaller than 5 mm, then the frame cavity shall be split at this "throat" region. Also:

- a) any surface whose normal is between 315° and 45° is a left vertical surface;
- b) any surface whose normal is between 45° and 135° is a bottom horizontal surface;
- c) any surface whose normal is between 135° and 225° is a right vertical surface;
- d) any surface whose normal is between 225° and 315° is a top horizontal surface.

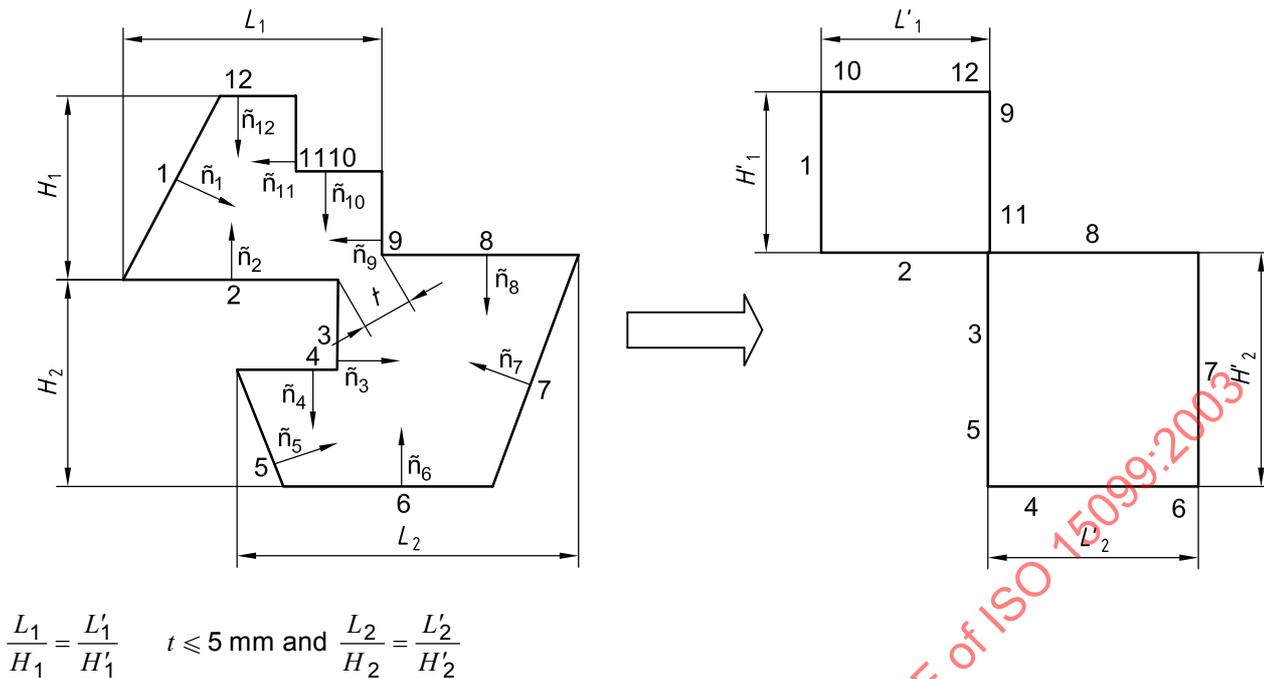


Figure 11 — Illustration of the treatment of irregularly shaped frame cavities

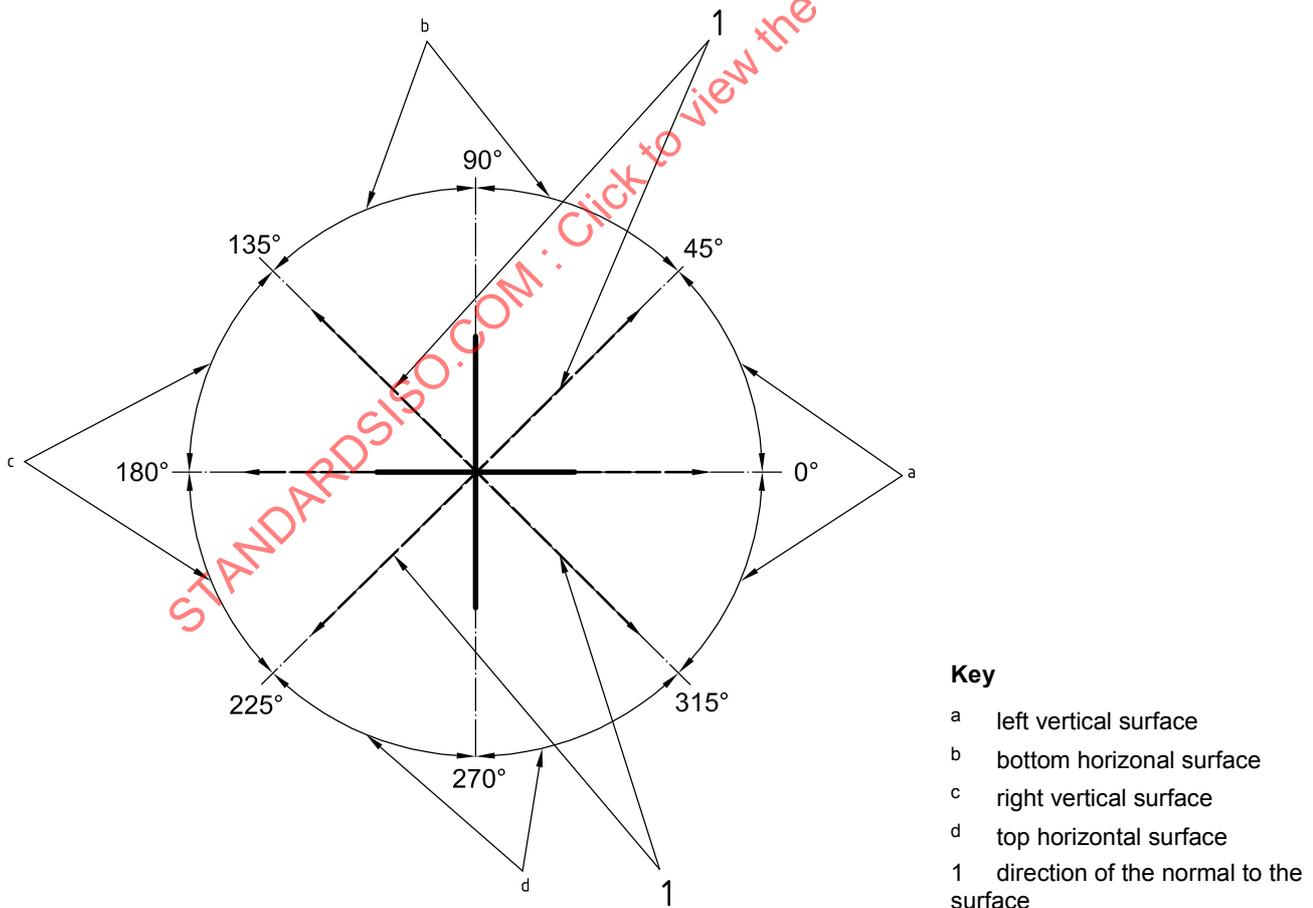
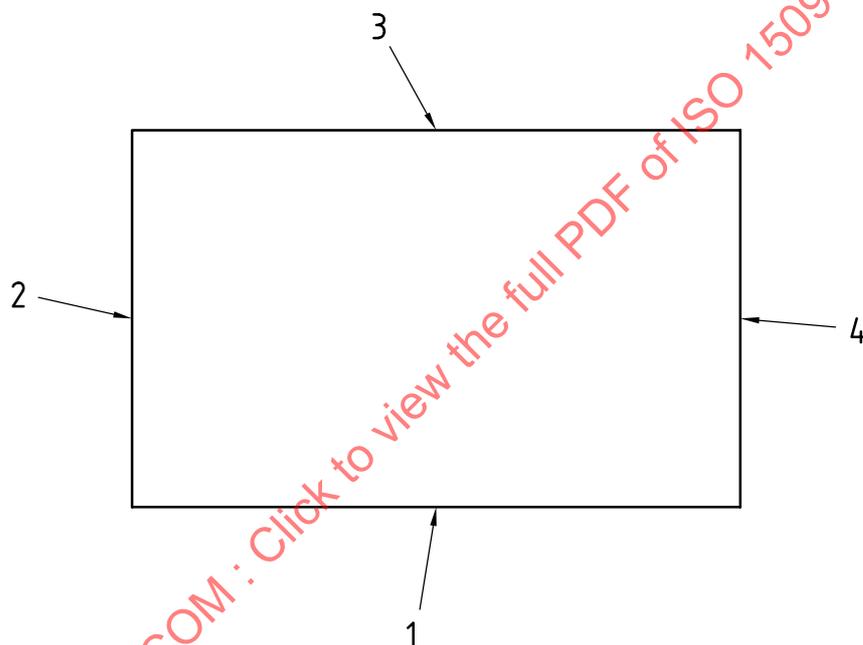


Figure 12 — Illustration of how to select surface orientation for frame cavities; dashed lines indicate direction of the normal to surface with cut off angles at 45°, 135°, 225° and 315°

Temperatures of equivalent vertical and horizontal surfaces shall be calculated as the mean of the surface temperatures according to the classification shown above. The direction of heat flow shall be determined from the temperature difference between vertical and horizontal surfaces of the equivalent cavity. The following rule shall be used (see also Figure 13):

- a) heat flow is horizontal if the absolute value of the temperature difference between vertical cavity surfaces is larger than between horizontal cavity surfaces;
- b) heat flow is vertical, heat flow upwards if the absolute temperature difference between horizontal cavity surfaces is larger than between vertical cavity surfaces and the temperature difference between the top horizontal cavity surface and bottom horizontal cavity surface is negative;
- c) heat flow is vertical, heat flow downwards if the absolute temperature difference between horizontal cavity surfaces is larger than between vertical cavity surfaces and the temperature difference between the top horizontal cavity surface and bottom horizontal cavity surface is positive.



Key

- 1 temperature of bottom surface, T_{bo}
- 2 temperature of left vertical surface, T_{lv}
- 3 temperature of top surface, T_{tp}
- 4 temperature of right vertical surface, T_{rv}

a) $|T_{rt} - T_{lf}| \geq |T_{tp} - T_{bo}|$



heat flow is horizontal;

b) $|T_{rt} - T_{lf}| < |T_{tp} - T_{bo}|$ and $T_{tp} < T_{bo}$



heat flow is vertical, heat flow up;

c) $|T_{rt} - T_{lf}| < |T_{tp} - T_{bo}|$ and $T_{tp} > T_{bo}$



heat flow is vertical, heat flow down.

Figure 13 — Illustration of how to select heat flow direction

6.6.5 Radiant heat flow

The radiative heat transfer coefficient h_r shall be calculated using:

$$h_r = \frac{4 \sigma T_{av}^3}{\frac{1}{\varepsilon_{cc}} + \frac{1}{\varepsilon_{ch}} - 2 + \frac{1}{\frac{1}{2} \left\{ \left[1 + \left(\frac{L_h}{L_v} \right)^2 \right]^{\frac{1}{2}} - \frac{L_h}{L_v} + 1 \right\}}}} \tag{93}$$

where $T_{av} = \frac{T_{cc} + T_{ch}}{2}$ (94)

The above notation assumes radiant heat flow to be in the horizontal direction. If the heat flow direction is vertical, then the inverse of the ratio L_h/L_v shall be used (i.e., L_v/L_h).

NOTE For more details, see Reference [16].

6.7 Ventilated air cavities and grooves

6.7.1 Slightly ventilated cavities and grooves with small cross section

Exposed grooves with small cross sections (see Figure 14) or cavities connected to the external or internal environments by a slit greater than 2 mm but not exceeding 10 mm are to be considered as slightly ventilated air cavities. The equivalent conductivity is twice that of an unventilated air cavity of the same size in accordance with 6.6. For cases where the slit is less than or equal to 2 mm, treat the cavity as completely enclosed in accordance with 6.6.

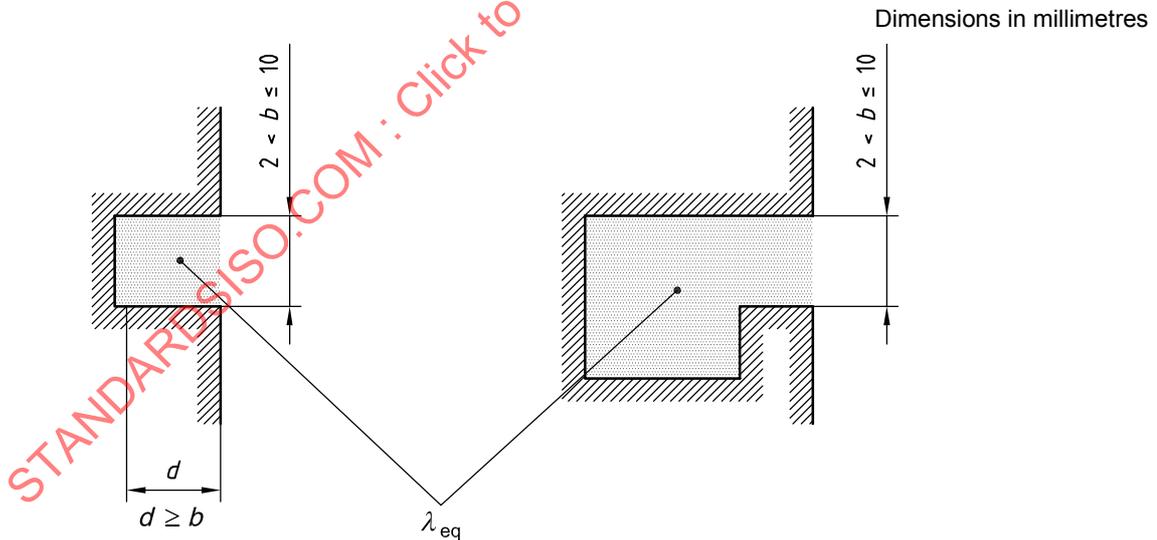


Figure 14 — Examples for slightly ventilated cavities and grooves with small cross section

6.7.2 Well-ventilated cavities and grooves with large cross section

In cases not covered by 6.6 and 6.7.1, in particular when the width b of a groove or of a slit connecting a cavity to the environment exceeds 10 mm, it is assumed that the whole surface is exposed to the environment. Therefore, the surface heat transfer coefficients, h_{inl} and h_{out} , calculated in accordance with Clause 8, shall be used at the developed internal and external surfaces, respectively.

In the case of a large cavity connected by a single slit and a developed cross sectional length exceeding the width of the slit by a factor of 5 (see Figure 15), the detailed radiation model (see 8.4.2) may be appropriate for determining the radiative portion of the surface heat transfer coefficient.

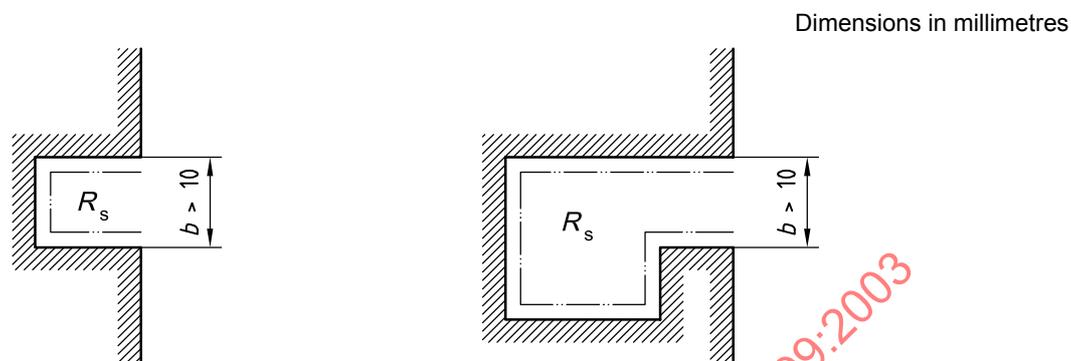


Figure 15 — Examples for well-ventilated cavities and grooves

7 Shading devices

7.1 Definitions

7.1.1 Introduction

Clause 7 provides the necessary equations for the effects of shading devices on the thermal and optical properties of a window system.

The scope is restricted to those kinds of shading devices which are or which may, by proper approximation, be treated as a layer parallel to the pane(s) of the window.

The introduction of shading devices in the model of the window system leads to modifications of the main thermal and optical equations. In order not to complicate the presentation of equations given in Clause 5, the necessary changes to those equations are not integrated in Clause 5 itself, but given as amendments in Clause 7.

NOTE Information on calculation procedures and measurement techniques on shading devices can be found in references [17 - 25]. In general, these references concern ongoing work.

The contents of this clause are based on the most up-to-date procedures, with simplifying approximations where needed due to practical limitations with respect to modelling and computational efforts and availability of product data.

Shading devices can be divided into two basic types:

- layer type of shadings, such as screens, curtains and venetian blinds which are located parallel to the pane(s), with intimate thermal-optical contact;
- extra-fenestrial type of shadings, such as awnings and overhangs, which are located less close to the panes. These types of shading may be regarded as part of the window's environment, because of the limited thermal interaction. They have mainly an effect on the temperature and radiation conditions outside the window. In specific cases, however, the conditions and properties of the window itself may also influence the condition of this "environment" (e.g. reflection of solar radiation, hot air pockets). These types of shading are outside the scope of this International Standard.

7.1.2 Principle of the calculations

The thermal-optical interaction of a layer type of shading device is, to a greater extent, similar to the panes and films. In this regard, the layer type of shading device may be defined in the model as a layer between two gaps. This thus-defined layer exchanges heat with the other components and/or the environment by conduction and convection and by thermal radiation. It also absorbs, reflects and transmits solar radiation. However, due to its porous structure (open weave, slats), the shading device is not only partially transmittant for solar radiation, but also for thermal (long wave) radiation. It shares this characteristic with some suspended thin films. This phenomenon is already covered in the equations by introducing in the equations transmittance for thermal radiation.

The shading device is usually also permeable for air, either due to its porous structure or due to openings at its perimeter. Air may cross the shading device and thus move from one gap to the other or from the environment into the gap behind the shading device and vice versa. This phenomenon has not been previously covered by the equations in the previous clauses and will therefore be introduced in this clause.

Because the layer type of shading device is modelled as a one-dimensional layer similar to a pane or film, the two- or three-dimensional characteristics shall be translated into one-dimensional numbers. This is in particular the case for the optical properties, e.g., the optical properties of a shading device are a function of the geometry of the device and the position in the assembly. To consider a slat type of shading device such as a venetian blind, information on the optical properties of the slat material, together with the geometry of the slats and their positions, is used to determine the overall transmittance, reflectance and absorption of the layer.

7.2 Optical properties

7.2.1 General

A particular characteristic of a shading device compared to “normal” glazings or films is that the incident solar radiation may change direction while being transmitted or reflected at the layer.

For the evaluation of thermal effects, the following approximation is considered to be sufficiently accurate.

Beam radiation transmitted or reflected by the solar shading device is considered to be split into two parts:

- an undisturbed part (specular transmission and reflection);
- a disturbed part.

The disturbed part is approximated as anisotropic diffuse (Lambertian).

Diffuse radiation transmitted or reflected by the solar shading device is assumed to remain diffuse.

An exact description of the way solar radiation travels through the system would require a full three-dimensional calculation using the full matrix of the transmission, absorption and forward and backward reflection for each angle of incidence at each component. For the evaluation of the spatial distribution of daylighting, this would be the necessary way to proceed.

Consequently the following solar properties of the solar shading device are required or transmittance, for beam radiation, for each angle of incidence:

$\tau_{\text{dir,dir}}(\lambda_{w_j})$ direct to direct transmittance;

$\tau_{\text{dir,dif}}(\lambda_{w_j})$ direct to diffuse transmittance.

For diffuse radiation:

$\tau_{\text{dif,dif}}(\lambda_{w_j})$ diffuse to diffuse transmittance.

Similarly for the reflectance, the following properties are required (for beam radiation, for each angle of incidence):

$r_{\text{dir,dir}}(\lambda_{\text{wj}})$ direct to direct reflectance;

$r_{\text{dir,dif}}(\lambda_{\text{wj}})$ direct to diffuse reflectance.

For diffuse radiation, the following properties are required:

$r_{\text{dif,dif}}(\lambda_{\text{wj}})$ diffuse to diffuse reflectance, and for the absorption:

$$\alpha_{\text{dir}}(\lambda_{\text{wj}}) = [1 - \tau_{\text{dir,dir}}(\lambda_{\text{wj}}) - r_{\text{dir,dir}}(\lambda_{\text{wj}}) - \tau_{\text{dir,dif}}(\lambda_{\text{wj}}) - r_{\text{dir,dif}}(\lambda_{\text{wj}})] \quad (95)$$

$$\alpha_{\text{dif}}(\lambda_{\text{wj}}) = [1 - \tau_{\text{dif,dif}}(\lambda_{\text{wj}}) - r_{\text{dif,dif}}(\lambda_{\text{wj}})] \quad (96)$$

7.2.2 Resulting amendments to the equations in 5.2 and 5.3

For a window system incorporating a layer type of shading device, the optical equations given in Clause 5 remain the same with the following extensions.

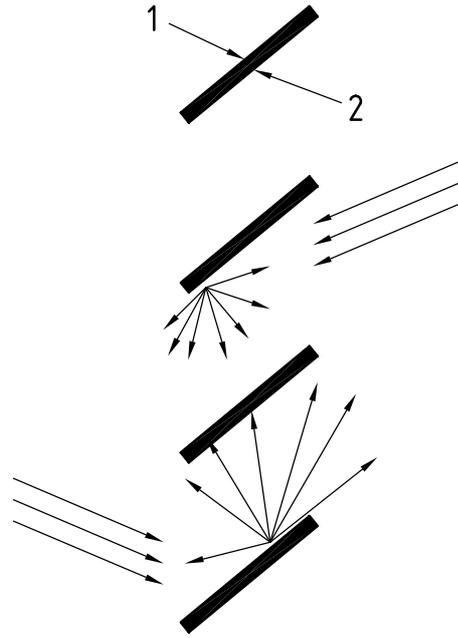
- Each spectral flow rate equation in 5.2 and 5.3 shall be split into three: a “dir,dir”, “dir,dif” and “dif,dif” heat flow, with corresponding transmittance τ and reflectance r . In the sum of the spectral heat flows: “dir,dir”, “dir,dif” and “dif,dif” parts shall be summed.
- The transmittance shall be split, similar to the reflectance, into a forward and a backward value.
- The sum of $\tau_{\text{dir,dir}}$ and $\tau_{\text{dir,dif}}$ is equal to the direct to hemispherical transmittance $\tau_{\text{dir,h}}$; similarly for the reflectance.

For slat type shadings, equations to calculate these properties are given in 7.3, on the basis of optical properties and the geometry of the slats.

NOTE There is no existing International Standard for the *measurement* of these optical properties. Until such a testing standard is available, the calculation method of this clause shall be considered as provisional and is provided for information purposes only, except for slat types of shading devices for which the following clause provides a *calculation* method.

Once a beam transmitting through or reflecting at a solar shading device is split into a direct and a diffuse part, the diffuse part continues its route through the system. This implies that even for normal incidence solar radiation, for all other panes, films and shading layers in the window, the $\tau_{\text{dif,dif}}$ and $r_{\text{dif,dif}}$ values are required; consequently, the values for normal incidence provide insufficient information.

Due to the redirection of the radiation, the forward transmittance is not necessarily equal to the backward transmittance, as illustrated in Figure 16.



Key

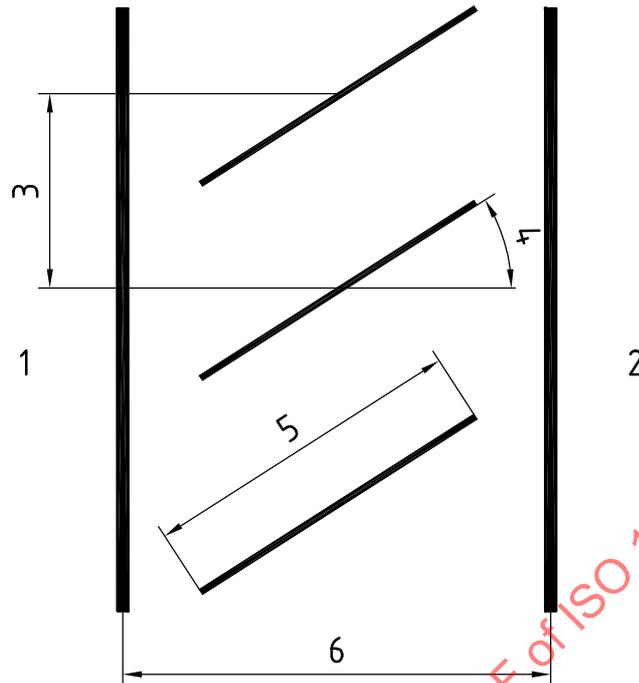
- 1 white
- 2 black

Figure 16 — Illustration of different values for forward and backward solar transmittance (slats with different colour at both surfaces)

7.3 Slat type of shading

7.3.1 General

For a shading device consisting of parallel slats, the optical properties can be determined as function of slat properties, geometry and position (see Figure 17). The air permeability can also be determined as function of slat geometry and position.

**Key**

- 1 outdoor
- 2 indoor
- 3 slat distance
- 4 slat angle
- 5 slat width
- 6 gap width

Figure 17 — Slat geometry

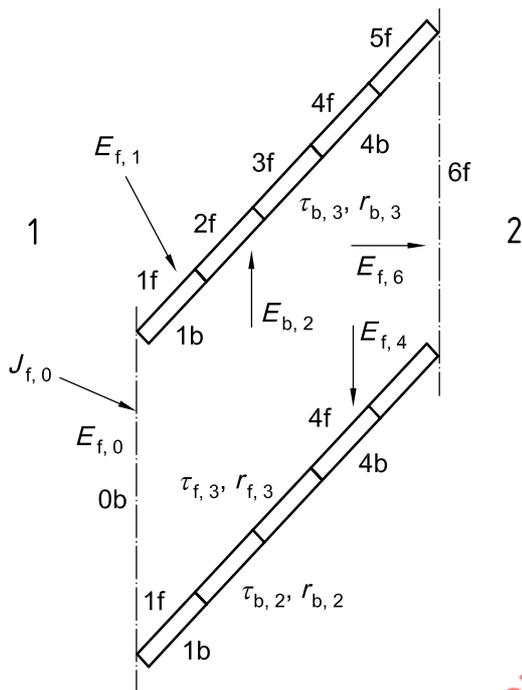
7.3.2 Optical properties

7.3.2.1 General

This subclause gives the procedure for calculating the solar optical properties of a slat type of shading device provided:

- that the slats are non-specular reflecting;
- that any effects of the window edges may be ignored.

The procedure is to consider two adjacent slats and to subdivide the slats into five equal parts (see Figure 18).



- Key**
- 1 outdoor
 - 2 indoor

Figure 18 — Discretization used in the model

Every slat is divided into five elements (the improvement of considering more elements is negligible). Notice that different properties can be assigned to every element, in particular to every side of the slat. The process described below has to be solved for every wavelength band required by the properties of the elements or by the rest of the transparent system where the shading device is installed.

Due to the assumption of non-specular reflection, a slight curving of the slats may be ignored.

7.3.2.2 Equations

These equations have a more general application, if the allocation of the layer numbers is generalized.

For each layer f,i and b,i , with i from 0 to n (here: $n = 6$) and for each spectral interval λ_{Wj} , ($\lambda_W \rightarrow \lambda_W + \Delta\lambda_W$):

$$E_{f,i} = \sum_k [(r_{f,k} + \tau_{b,k}) E_{f,k} F_{f,k \rightarrow f,i} + (r_{b,k} + \tau_{f,k}) E_{b,k} F_{b,k \rightarrow f,i}] \tag{97}$$

$$E_{b,i} = \sum_k [(r_{b,k} + \tau_{f,k}) E_{b,k} F_{b,k \rightarrow b,i} + (r_{f,k} + \tau_{b,k}) E_{f,k} F_{f,k \rightarrow b,i}] \tag{98}$$

where

E_k is the irradiance on surface k ;

$F_{p \rightarrow q}$ is the view or shape factor from surface p to surface q (e.g. b,k to b,i).

Also

$$E_{f,ex} = J_{ex}(\lambda_{w_j}) \quad (99)$$

$$E_{b,int} = J_{int}(\lambda_{w_j}) = 0 \quad (100)$$

where

J_{ex} is the radiosity from the external environment (incident solar radiation);

J_{int} is the radiosity from the internal environment (room reflection).

7.3.2.3 Diffuse-diffuse transmission and reflection

Due to the assumption of non-specular reflection, the values for the view factors $F_{p \rightarrow q}$ can be calculated by conventional view factor calculation methods for diffuse radiation exchange.

NOTE For calculation methods on view factors, see [26].

For diffuse incident radiation, the view factor between external environment and the other layers is also determined by the view factors for diffuse radiation exchange.

After solving the set of equations, the diffuse/diffuse transmission coefficient is found as the radiation reaching the internal environment $E_{f,n}$ ($n = 6$), divided by the incident solar radiation, J_{ex} :

$$\tau_{dif,dif}(\lambda_{w_j}) = E_{f,n}(\lambda_{w_j}) / J_{ex}(\lambda_{w_j}) \quad (101)$$

Similarly, for the diffuse/diffuse reflection coefficient:

$$r_{dif,dif}(\lambda_{w_j}) = E_{b,ex}(\lambda_{w_j}) / J_{ex}(\lambda_{w_j}) \quad (102)$$

7.3.2.4 Direct-direct transmission and reflection

By straightforward geometric calculation from the angle and aspect ratio of the slats (see Figure 19), the beam radiation which passes the slats without touching can be calculated for given angle of incidence γ .

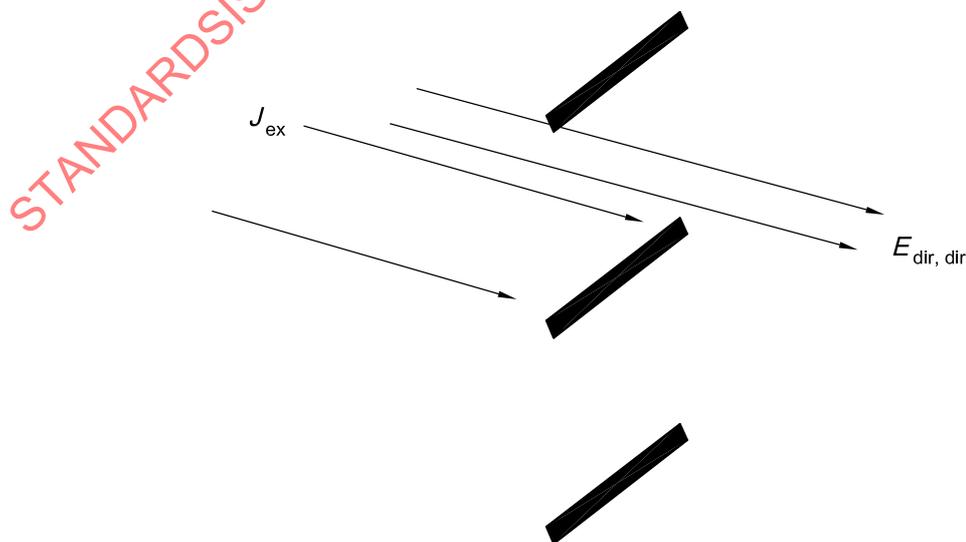


Figure 19 — Direct-direct transmission

This part of the transmission is wavelength-independent.

This is the direct/direct transmission: $E_{\text{dir,dir}}(\lambda_w)$

Consequently, the direct-direct transmittance for incidence angle γ amounts to:

$$\tau_{\text{dir,dir}}(\gamma) = E_{\text{dir,dir}}(\lambda_{w,j},\gamma) / J_{\text{ex}}(\lambda_{w,j},\gamma) \tag{103}$$

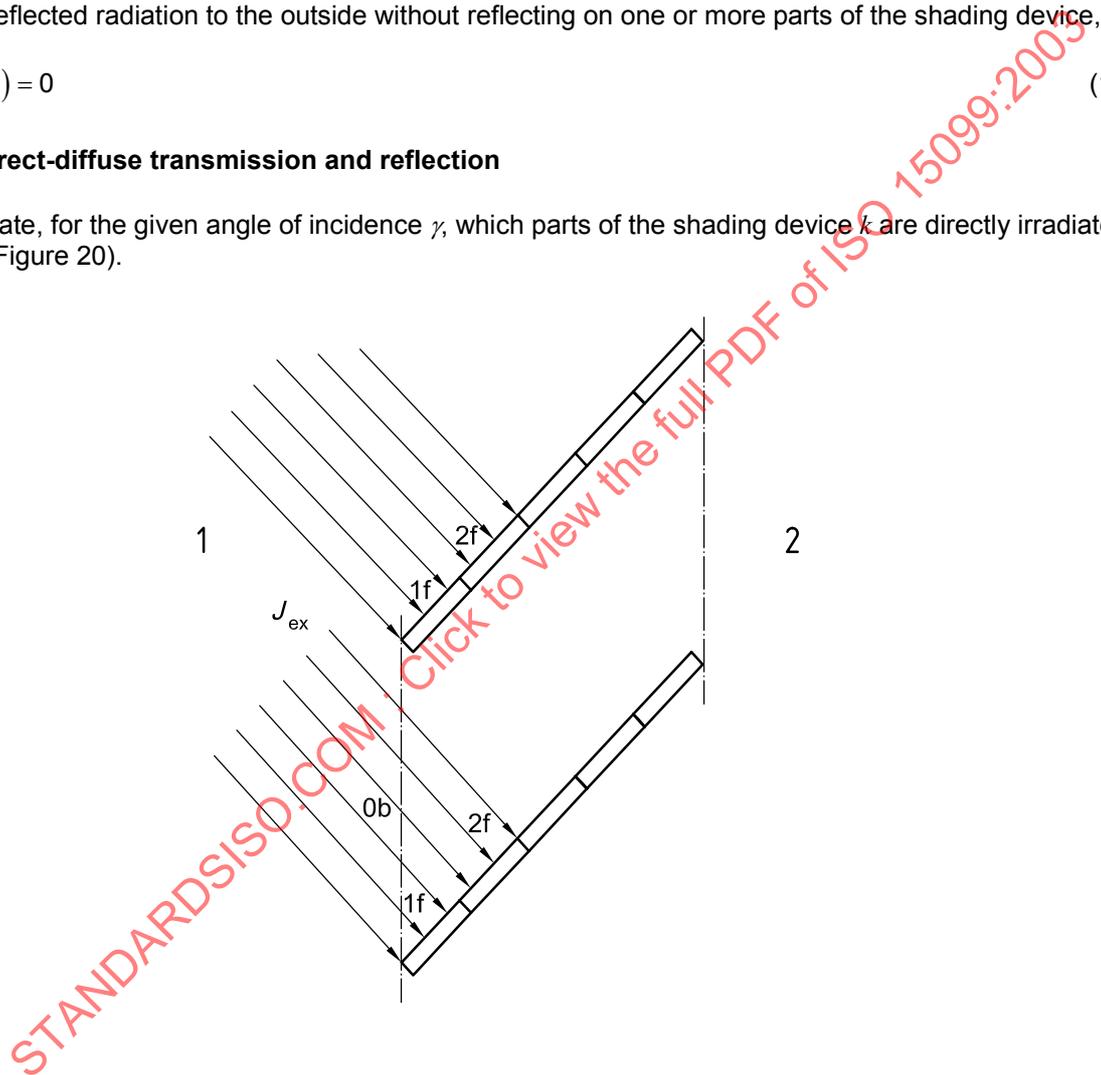
for any wavelength $\lambda_{w,j}$.

There is no reflected radiation to the outside without reflecting on one or more parts of the shading device, so:

$$r_{\text{dir,dir}}(\gamma) = 0 \tag{104}$$

7.3.2.5 Direct-diffuse transmission and reflection

Firstly, calculate, for the given angle of incidence γ , which parts of the shading device k are directly irradiated by $J_{f,\text{ex}}$ (see Figure 20).



- Key**
- 1 outdoor
 - 2 indoor

Figure 20 — Directly irradiated parts of the shading device

The view factors between the incident radiation J_{ex} and those directly irradiated parts k are:

$$F_{f,\text{ex} \rightarrow f,k} = 1$$

$$F_{f,\text{ex} \rightarrow b,k} = 1$$

The view factor between the internal and external environment is zero, to exclude the direct-direct transmittance:

$$F_{f,ex \rightarrow b,n} = 0 \text{ and } F_{b,ex \rightarrow f,n} = 0$$

After solving the set of equations we find the direct-diffuse transmission and reflection coefficients:

$$\tau_{\text{dir,dif}}(\lambda_{w,j},\gamma) = E_{f,n}(\lambda_{w,j},\gamma) / J_{\text{ex}}(\lambda_{w,j},\gamma) \quad (105)$$

$$r_{\text{dir,dif}}(\lambda_{w,j},\gamma) = E_{b,n}(\lambda_{w,j},\gamma) / J_{\text{ex}}(\lambda_{w,j},\gamma) \quad (106)$$

7.3.2.6 Absorption

That part which is neither transmitted, nor reflected, is the part which is absorbed in the slats. Per wavelength band:

$$\alpha_{\text{dir}}(\lambda_{w,j}) = \left\{ \left[1 - \tau_{\text{dir,dif}}(\lambda_{w,j}) - r_{\text{dir,dif}}(\lambda_{w,j}) - \tau_{\text{dir,dif}}(\lambda_{w,j}) - r_{\text{dir,dif}}(\lambda_{w,j}) \right] \right\} \quad (107)$$

$$\alpha_{\text{dif}}(\lambda_{w,j}) = \left\{ \left[1 - \tau_{\text{dif,dif}}(\lambda_{w,j}) - \rho_{\text{dif,dif}}(\lambda_{w,j}) \right] \right\} \quad (108)$$

7.3.2.7 Thermal transmittance and reflectance

The blinds are also semi-transparent for infrared (thermal) radiation. In order to obtain the IR transmittance and reflectance of the shading device for given (IR) slat properties, the same model is used as for the calculation of diffuse-diffuse transmission and reflection of solar radiation, replacing the slat's solar optical properties by its thermal radiation properties.

The *normal* emissivity of the surfaces can be measured in accordance with EN 12898. There is no existing standard for the measurement of the *hemispherical* emissivity of opaque materials. Usually an emissometer is used for this purpose.

Selected examples of calculated optical properties of a slat type of shading device as function of the slat properties and geometries are given in Annex C.

7.4 Ventilation

7.4.1 General

For a ventilated cavity, the set of equations given in 5.3 shall be extended in the way described in 7.4.2.

7.4.2 Main heat balance equations

7.4.2.1 Principle

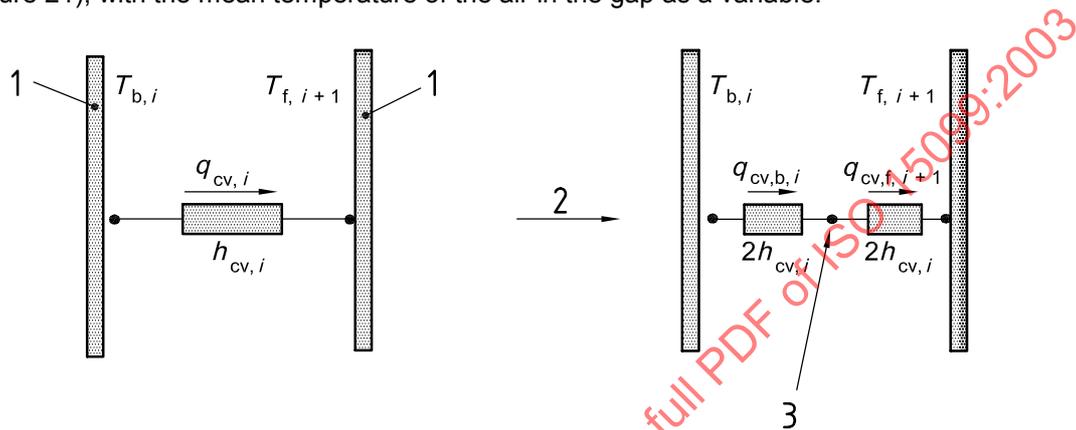
Air spaces may be connected to the exterior or interior environment or to other spaces. For a ventilated gap, the heat balance in the gap requires an extra term, the amount of heat supplied to or extracted from the gap air. This implies that it is no longer sufficient to describe, as in 5.3, the conductive/convective heat exchange in a gap as the heat transfer from one surface to the other. It is necessary to make a split between the conductive/convective heat transfer from one surface to the air and from the air to the other surface as illustrated in Figure 6. In the heat balance equations for the gap, the heat extracted from or supplied to the gap by ventilation is added to this air gap node.

The mean temperature of the air in the gap is given by equations for the heat exchange between the air flowing through the gap and the adjacent surfaces.

NOTE There is no existing International Standard for measurement of these properties. Until such a testing standard is available, the calculation method in 7.4.2 must be considered as provisional and is provided for information purposes only.

7.4.2.2 Equations — non-vented gap

For the non-vented case (5.3) the heat exchange by conduction/convection across a gap from one layer to the adjacent layer (pane, film or shading device) as given in 5.3.1: $q_{cv,i} = h_{cv,i}(T_{f,i} - T_{b,i+1})$, is split into two parts (see Figure 21), with the mean temperature of the air in the gap as a variable.



- Key**
- 1 pane or shading
 - 2 split
 - 3 $T_{gap,i}$

Figure 21 — Split convective heat transfer across non-vented gap

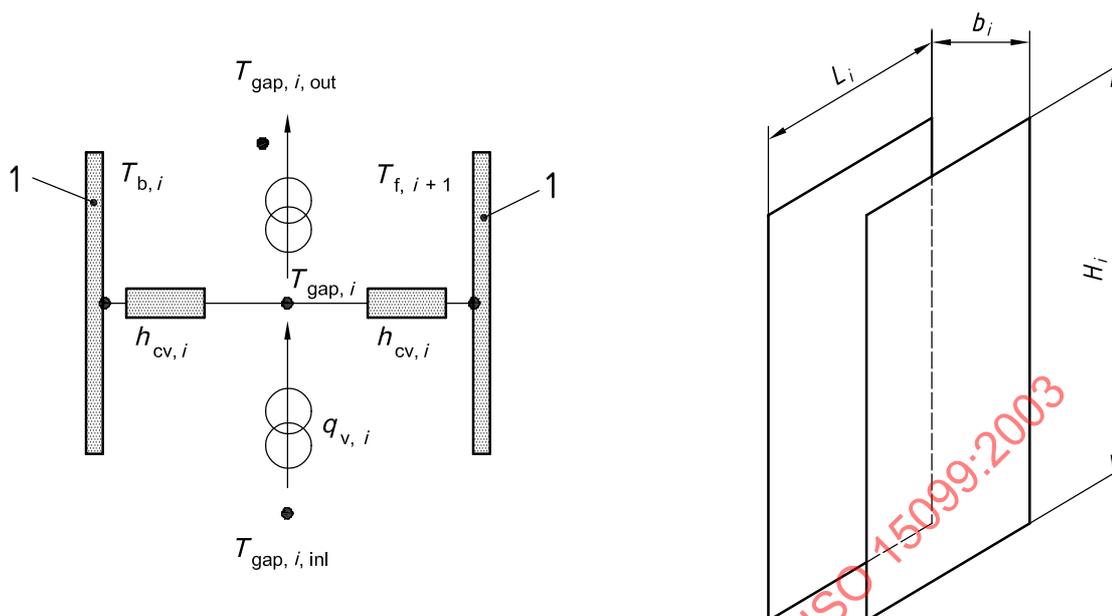
$$q_{cv,f,i} = 2h_{cdv,i}(T_{f,i} - T_{gap,i}) = q_{cv,b,i+1} = 2h_{cdv,i}(T_{gap,i} - T_{b,i+1}) \tag{109}$$

where

- $q_{cv,f,i}$ is the convective heat transfer from the one surface to the gap, in watts per square metre;
- $h_{cdv,i}$ is the surface-to-surface heat transfer coefficient by conduction/convection for non-vented cavities, given by the equations in 5.3, in watts per square metre kelvin;
- $T_{f,i}$ is the temperature of the surface of layer (pane, film or shading) i , facing cavity i , see 5.3, in kelvins;
- $T_{gap,i}$ is the equivalent mean temperature of the air in cavity i , given in Equation (119) below, in kelvins;
- $q_{cv,b,i+1}$ is the convective heat transfer from the gap to the other surface, in watts per square metre;
- $T_{b,i+1}$ is the temperature of the surface of layer (pane, film or shading) $i + 1$, facing the cavity i , see 5.3, in kelvins.

7.4.2.3 Ventilated gap

In a ventilated gap, due to the air movement, the convective heat exchange coefficient is increased (see Figure 22)


Key

1 pane or shading

Figure 22 — Model of mean air- and outlet temperature and main dimensions

 This increased coefficient is written as $h_{cv,i}$:

$$q_{cv,b,i} = h_{cdv,i}(T_{b,i} - T_{gap,i}), \text{ and } q_{cv,f,i+1} = h_{cv,i}(T_{gap,i} - T_{f,i+1}) \quad (110)$$

 with $h_{cv,i}$ given by the equation:

$$h_{cv,i} = 2h_{cdv,i} + 4V_i \quad (111)$$

where

 $q_{cv,b,i}$ is the convective heat transfer from the one surface to the gap, in watts per square metre;

 $h_{cv,i}$ is the surface-to-air heat transfer coefficient by conduction/convection for vented cavities, given by Equation (111), in watts per square metre kelvin;

 $q_{cv,f,i+1}$ is the convective heat transfer from the gap to the other surface, in watts per square metre;

 $h_{cdv,i}$ is the surface-to-surface heat transfer coefficient by conduction/convection for non-vented cavities, given by the equations in 5.3, in watts per square metre kelvin;

 V_i is the mean air velocity in the gap, see 7.4.4, in metres per second;

and with (same as for the non-vented case):

 $T_{gap,i}$ is the equivalent mean temperature of the air in cavity i , given by Equation (119) in 7.4.3, in kelvins;

 $T_{b,i}$ is the temperature of the surface of layer (pane, film or shading) i , facing cavity i , see 5.3, in kelvins;

$T_{f,i+1}$ is the temperature of the surface of layer (pane, film or shading) $i + 1$, facing cavity i , see 5.3, in kelvins.

For zero velocity, the equations for the ventilated gap reduce to the equations for the non-vented case.

Due to the ventilation, an extra term is added to the heat balance equations of the gap given in 5.3. The extra term is:

$$q_{vl,i} = \rho_i \times c_p \times \varphi_{vl,i} (T_{\text{gap},i,\text{inl}} - T_{\text{gap},i,\text{out}}) / (H_i L_i) \quad (112)$$

Equations (114) to (117) are formulated in such a manner to satisfy the following energy balance equation:

$$q_{vl,i} = q_{cv,b,i} + q_{cv,f,i+1} \quad (113)$$

where

$q_{vl,i}$ is the heat transfer to the gap by ventilation, in watts per square metre;

ρ_i is the density of the air in cavity k at temperature $T_{\text{gap},i}$, in kilograms per cubic metre;

c_p is the specific heat capacity of air, in joules per kilogram kelvin (i.c.: 1008);

$\varphi_{vl,i}$ is the air flow rate in cavity i , see 7.4.4, in cubic metres per second;

$T_{\text{gap},i,\text{inl}}$ is the temperature at the inlet of the gap, in kelvins.

The heat transfer is normalized to 1 m² of the aperture area.

The value of $T_{\text{gap},i,\text{inl}}$ depends on where the air comes from: either the internal or external air temperature or the outlet temperature $T_{\text{gap},k,\text{out}}$ of the gap k with which gap i exchanges air;

$T_{\text{gap},i,\text{out}}$ is the temperature at the outlet of the gap, see Equation (114), in kelvins;

L_i is the length of cavity i , see Figure 22, in metres;

H_i is the height of cavity i , see Figure 22, in metres.

7.4.2.4 Heat transfer to internal environment

The heat transfer to the internal environment shall be extended in a similar way with a term $q_{vl,k}$ for the heat transfer by ventilation by air coming from cavity k .

Following the convention from Clause 5, with $i = n$ for the internal environment, for all cavities k with air flow to the internal environment, n :

$$q_{vl,n} = \sum_i \rho_i \cdot c_p \cdot \varphi_{vl,i} (T_{\text{gap},i,\text{out}} - T_{\text{ai},n}) / (H_i \times L_i) \quad (114)$$

The heat transfer is normalized to 1 m² of the aperture area.

where

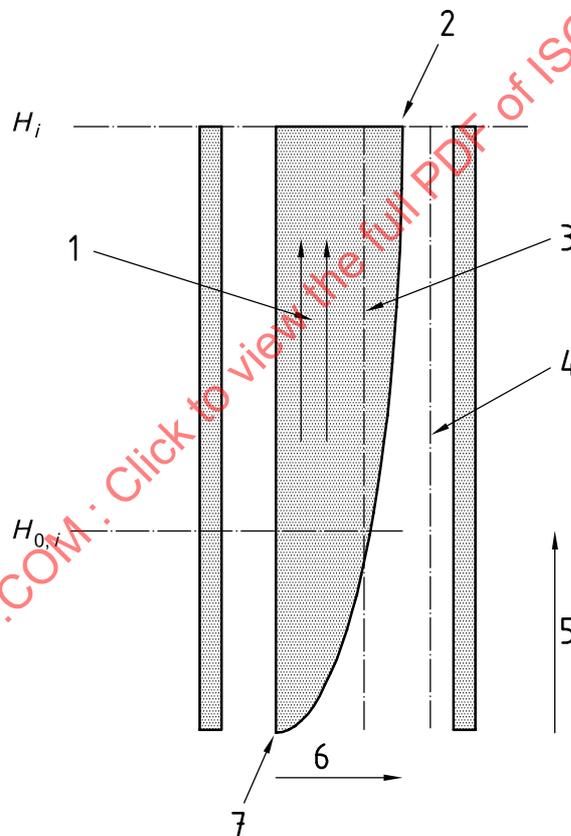
ρ_i is the density of the air in cavity i at temperature $T_{\text{gap},i}$, in kilograms per cubic metre;

c_p is the specific heat capacity of air, in joules per kilogram kelvin (i.c.: 1008);

- $\varphi_{Vl,i}$ is the air flow rate in cavity i , see 7.4.4, in cubic metres per second;
- $T_{\text{gap},i,\text{out}}$ is the temperature of the air at the outlet of the gap from where the air originates, see Equation (116), in kelvins;
- $T_{\text{ai},n}$ is the indoor air temperature, in kelvins;
- L_i is the length of cavity i , see Figure 22, in metres;
- H_i is the height of cavity i , see Figure 22, in metres.

7.4.3 Temperatures in the cavity

Assuming the mean velocity of the air in the space is known (see 7.4.4), the temperature profile and the heat flow can be calculated by a simple model. Due to the air flow through the space, the air temperature in the space varies with height (see Figure 23).



Key

- 1 air flow, $\varphi_{Vl,i}$
- 2 outlet air temperature, $T_{\text{gap},i,\text{out}}$
- 3 average air temperature, $T_{\text{gap},i}$
- 4 average surface temperature, $T_{\text{av},i}$
- 5 distance, x , from inlet
- 6 air temperature, $T_{\text{gap},i}(x)$
- 7 inlet air temperature, $T_{\text{gap},i,\text{inl}}$

Figure 23 — Air flow in the gap of a window system

The temperature profile depends on the air velocity in the space and the heat transfer coefficient to both layers. The air temperature profile in space i is given by:

$$T_{\text{gap},i}(x) = T_{\text{av},i} - (T_{\text{av},i} - T_{\text{gap},i,\text{in}}) e^{-x/H_{0,i}} \quad (115)$$

where

$T_{\text{gap},i}(x)$ is the temperature of the air in gap i at distance x from the inlet, in kelvins;

$H_{0,i}$ is the characteristic height (temperature penetration length), see Equation (117), in metres;

$T_{\text{gap},i,\text{in}}$ is the temperature of the incoming air in gap i , in kelvins;

$T_{\text{av},i}$ is the average temperature of the surfaces of layers i and $i + 1$, given by equation

$$T_{\text{av},i} = \frac{(T_{\text{b},i} + T_{\text{f},i+1})}{2} \quad (116)$$

where

$T_{\text{b},i}$ is the temperature of the surface of layer (pane, film or shading) i , facing cavity i , see 5.3, in kelvins;

$T_{\text{f},i+1}$ is the temperature of the surface of layer (pane, film or shading) $i + 1$, facing cavity i , see 5.3, in kelvins.

The characteristic height of the temperature profile is defined by:

$$H_{0,i} = \frac{\rho_i \times c_p \times b_i}{2h_{\text{cvl},i}} \times V_i \quad (117)$$

where

$H_{0,i}$ is the characteristic height (temperature penetration length), in metres;

ρ_i is the density of the air at temperature $T_{\text{gap},j}$, in kilograms per cubic metre;

c_p is the specific heat capacity, in joules per kilogram kelvin (i.c.: 1008);

b_i is the width of the cavity i , in metres;

$h_{\text{cvl},i}$ is the heat transfer coefficient for ventilated cavities, see Equation (111), in watts per square metre kelvin.

V_i is the mean velocity of the air flow in the cavity i , see 7.4.4, in metres per second;

The leaving air temperature is given by:

$$T_{\text{gap},i,\text{out}} = T_{\text{av},i} - (T_{\text{av},i} - T_{\text{gap},i,\text{inl}}) e^{-H_i/H_{0,i}} \quad (118)$$

where

$T_{\text{gap},i,\text{out}}$ is the temperature of the air at the outlet of gap i , in kelvins;

$T_{av,i}$ is the average temperature of the surfaces of layers i and $i - 1$, given by Equation (116), in kelvins;

$T_{gap,i,inl}$ is the temperature of the incoming air in the cavity i , in kelvins;

$H_{0,i}$ is the characteristic height (temperature penetration length), given by Equation (117), in metres;

H_i is the height of space i , in metres.

The thermal equivalent (average) temperature of the air in the space i is defined by:

$$T_{gap,i} = \frac{1}{H_i} \int_0^H T_{gap,i}(x) \times dx = T_{av,i} - \frac{H_{0,i}}{H_i} (T_{gap,i,out} - T_{gap,i,inl}) \quad (119)$$

where

$T_{gap,i}$ is the equivalent mean temperature of the air in the cavity i , in kelvins;

$T_{av,i}$ is the average temperature of the surfaces of layers i and $i + 1$, given by Equation (116), in kelvins;

$H_{0,i}$ is the characteristic height (temperature penetration length), given by Equation (117), in metres;

H_i is the height of space i , in metres.

$T_{gap,i,out}$ is the temperature of the air at the outlet of the gap i , in kelvins;

$T_{gap,i,inl}$ is the temperature of the incoming air in gap i , in kelvins;

7.4.4 Air flow and velocity

7.4.4.1 Forced ventilation

If the air flow within the air layer has a known value (for example due to mechanical ventilation), the equations given in 7.4.2 and 7.4.3 shall be applied as such, with the air velocity (m/s) given by:

$$V_i = \frac{\varphi_{vl,i}}{b_i \times L_i} \quad (120)$$

where

V_i is the mean velocity of the air flow in cavity i , in metres per second;

$\varphi_{vl,i}$ air flow rate in cavity i , in cubic metres per second;

b_i the width of cavity i , in metres;

L_i is the length of the cavity i , see Figure 22, in metres.

NOTE $\varphi_{vl,i}$ is the air flow rate for the whole area.

7.4.4.2 Wind-induced ventilation

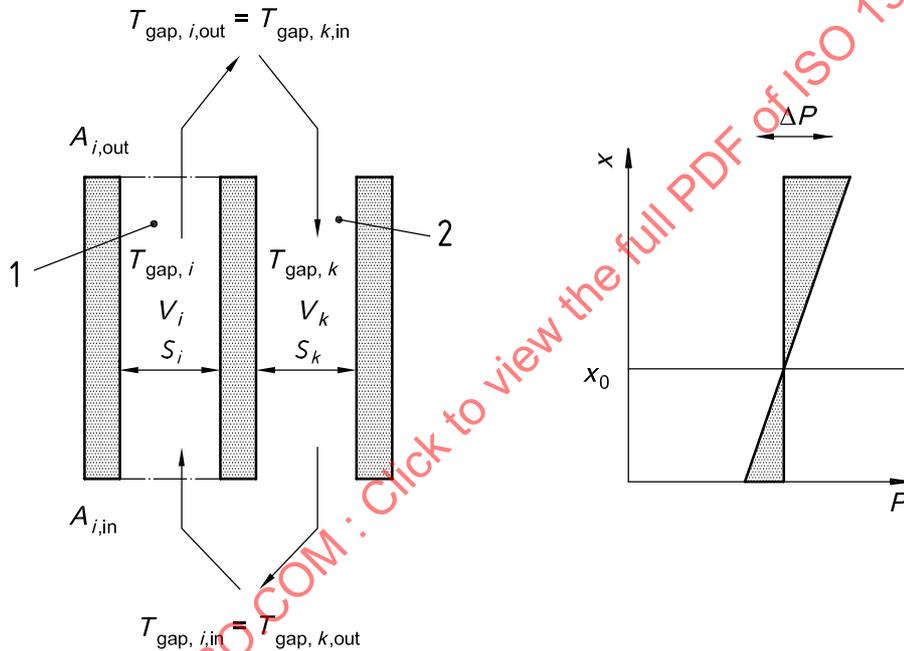
For external shading devices, the thermally-induced ventilation is mixed with wind-induced ventilation.

The heat exchange by ventilation between the shading and the next layer (pane) can be described on the basis of an appropriate value for the air flow or velocity. An appropriate value is to be determined on the basis of experiments or calculations [computational fluid dynamic (CFD) modelling].

For conservative design calculations, one may treat the cavity flow as forced convection (see 7.4.4.1), with the value for the air velocity V_i set to extreme low and extreme high values, thereby giving two values for the total solar energy transmittance.

7.4.4.3 Thermally-driven ventilation

The velocity of the air in the space, caused by the stack effect depends on the driving pressure difference and the resistance to the air flow of the openings and the space itself (see Figure 24).



- Key**
- 1 space, i
 - 2 space, k

Figure 24 — Schematic presentation of the stack effect

NOTE The height of the neutral zone x_0 depends on the flow resistances of the inlet and outlet openings

The air velocity is known by solving the set of equations given in 7.4.4.

The pressure difference results from a temperature difference between the space j and the connected space k , which is the exterior air, the interior air or another space. The temperature profile in the spaces is represented by the thermal equivalent temperature [Equation (119)]. The driving pressure difference Δp_T may be written approximately as:

$$\Delta P_{T,i,k} = \rho_0 \times T_0 \times g \times H_i \times |\cos \gamma_i| \times \frac{(T_{gap,i} - T_{gap,k})}{(T_{gap,i} \times T_{gap,k})} \quad (121)$$

where

$\Delta P_{T,i,k}$ is the driving pressure difference between space i and space k , in pascals;

ρ_0 is the density of air at temperature T_0 , in kilograms per cubic metre;

T_0 is reference temperature, e.g., $T_0 = 283$ K;

g is acceleration due to gravity = 9,81 metres per second per second;

γ_i is the tilt angle of space i in degrees from vertical;

H_i is the height of space i (same as space k), in metres;

$T_{\text{gap},i}$ is the equivalent (mean) temperature of the air in the space i , see Equation (119), in kelvins;

$T_{\text{gap},k}$ is the equivalent temperature of the connected space, which may be another gap k or the internal or external environment, in kelvins.

The air flow in the space is described as a pipe flow. Therefore, the following effects have to be taken into account.

Acceleration of the air to the velocity V (Bernouilli's equation):

$$\Delta P_{B,i} = \frac{\rho_i}{2} V_i^2 \quad (122)$$

Steady laminar flow (Hagen-Poiseuille law):

$$\Delta P_{HP,i} = 12 \cdot \mu_i \cdot \frac{H_i}{b_i^2} V_i \quad (123)$$

Pressure loss in the inlet and outlet openings:

$$\Delta P_{Z,i} = \frac{\rho_i}{2} \cdot V_i^2 (Z_{\text{inl},i} + Z_{\text{out},i}) \quad (124)$$

where

$\Delta P_{B,i}$ is the Bernouilli pressure loss in space i , in pascals;

ρ_i is the density of air at temperature $T_{\text{gap},j}$, in kilograms per cubic metre;

V_i is the mean velocity of air flow in cavity i , to be solved with Equation (120), in metres per second; (same for k);

$\Delta P_{HP,i}$ is the Hagen-Poiseuille pressure loss in space i , in pascals;

μ_i is the dynamic viscosity of air at temperature $T_{\text{gap},j}$, in pascal seconds;

H_i is the height of space i , in metres;

b_i is the width of cavity i , in metres;

Z_i is the pressure loss factors Z of cavity i , according to Equations (126) and (127).

The same equations apply to space k , where $V_k = V_i \cdot b_i / b_k$.

If the space k is the exterior or interior, $V_k = 0$ is assumed, in which case the pressure loss terms $\Delta P_{B,k}$ and $\Delta P_{HP,k}$ are zero as well as $\Delta P_{Z,k}$, where

$\Delta P_{Z,i,k}$ is the pressure loss Z between space i and k , in pascals.

The total pressure loss shall be equal to the driving pressure difference and this results in the velocities V_i and V_k by solving the equation:

$$\Delta P_{T,i,k} = \Delta P_{B,i} + \Delta P_{HP,i} + \Delta P_{Z,i} + \Delta P_{Z,k} + \Delta P_{B,k} + \Delta P_{HP,k} \quad (125)$$

where

$\Delta P_{T,i,k}$ is the driving pressure difference between space i and space k , according to Equation (121), in pascals;

$\Delta P_{B,i}$ is the Bernouilli pressure loss in space i , according to Equation (122), in pascals;

$\Delta P_{HP,i}$ is the Hagen-Poiseuille pressure loss in space i , according to Equation (123), in pascals;

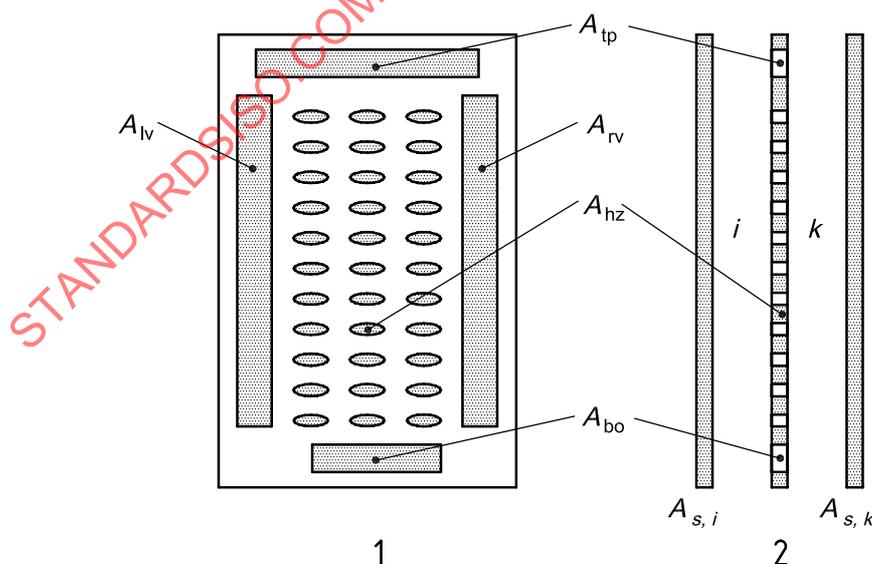
$\Delta P_{Z,i}$ is the pressure loss Z at the inlet and outlet of space i , according to Equation (124), in pascals;

$\Delta P_{z,k}$ the same as $\Delta P_{Z,i}$ but for space k ;

$\Delta P_{B,k}$ is the Bernouilli pressure loss in space k , according to Equation (122), in pascals;

$\Delta P_{HP,k}$ is the Hagen-Poiseuille pressure loss in space k , according to Equation (123), in pascals.

The pressure loss factor, Z , for openings may be estimated from the ratio of the equivalent area of an opening A_{eq} to the cross section of the space A_s (see Figure 25).



Key

- 1 front view
- 2 side view

Figure 25 — Openings in a ventilated gap

$$Z_{\text{inl}} = \left(\frac{A_{\text{s},i}}{0,6 \times A_{\text{eq,inl},i}} - 1 \right)^2 \quad (126)$$

and

$$Z_{\text{out}} = \left(\frac{A_{\text{s},i}}{0,6 \times A_{\text{eq,out},i}} - 1 \right)^2 \quad (127)$$

where

$A_{\text{s},i}$ is the cross section of space i ; $A_{\text{s},i} = b_i L_i$, in square metres;

b_i is the width of cavity i , in metres;

L_i is the length of cavity i , in metres;

$A_{\text{eq,inl},i}$ is the equivalent inlet opening area of cavity i , according to Equation (128) or (129), in square metres;

$A_{\text{eq,out},i}$ is the equivalent outlet opening area of the cavity i , according to Equation (128) or (129), in square metres.

If the temperature $T_{\text{gap},i}$ ($T_{\text{gap},k}$) of the cavity i (k) is higher than the temperature of the connected space k (i):

$$A_{\text{eq,inl}} = A_{\text{bo}} + \frac{1}{2} \frac{A_{\text{tp}}}{A_{\text{bo}} + A_{\text{tp}}} (A_{\text{lf}} + A_{\text{rt}} + A_{\text{ho}}) \quad A_{\text{eq,out}} = A_{\text{tp}} + \frac{1}{2} \frac{A_{\text{bo}}}{A_{\text{bo}} + A_{\text{tp}}} (A_{\text{lf}} + A_{\text{rt}} + A_{\text{ho}}) \quad (128)$$

Otherwise:

$$A_{\text{eq,out}} = A_{\text{bo}} + \frac{1}{2} \frac{A_{\text{tp}}}{A_{\text{bo}} + A_{\text{tp}}} (A_{\text{lf}} + A_{\text{rt}} + A_{\text{ho}}) \quad A_{\text{eq,inl}} = A_{\text{tp}} + \frac{1}{2} \frac{A_{\text{bo}}}{A_{\text{bo}} + A_{\text{tp}}} (A_{\text{lf}} + A_{\text{rt}} + A_{\text{ho}}) \quad (129)$$

where

A_{s} is the cross section of the space, in square metres;

A_{bo} is the area of the bottom opening, in square metres;

A_{tp} is the area of the top opening, in square metres;

A_{lf} is the area of the left side opening, in square metres;

A_{rt} is the area of the right side opening, in square metres;

A_{ho} is the total area of the holes in the surface (homogeneously distributed holes), in square metres.

It is assumed that the side openings are distributed evenly from top to bottom.

All these areas are total flow areas for the window (i.e., not normalized).

7.4.5 Gas-filled cavity with air circulation

In those cases of a closed cavity containing a gas mix and other component of the fenestration, e.g., an incorporated blind, the gas mix may flow from one side of the component (blind) to the other. In that case the equations given above remain valid, if "air" is replaced by "gas-mix", with the corresponding gas-mix properties.

7.4.6 Air permeability of slat types of shading devices

The air permeability of slat types of shading devices can be described using an appropriate value for the equivalent air permeability of the surface, A_{ho} . An appropriate value shall be determined on the basis of experiments or calculations [computational fluid dynamic (CFD) modelling].

For conservative design calculations, the value for the equivalent air permeability of the surface, A_{ho} , can be set to extreme low and extreme high values, thereby giving two extremes for the total solar energy transmittance.

7.5 Total solar energy transmittance and thermal transmittance

The thermal transmittance, U , and the total solar energy transmittance, τ_s , shall be calculated using Equations (2) and (14) respectively, applying the amendments to the equations in 5.2 as given in 7.2.

8 Boundary conditions

8.1 General

The various thermal properties can be determined using a standard calculation method but each will also be affected, to some extent, by the boundary conditions to which the product is exposed, i.e., the environment.

The boundary conditions consist of:

- internal and external air temperatures, T_{int} and T_{ex} , respectively;
- internal and external surface convective heat transfer coefficients, $h_{cv,int}$ and $h_{cv,ex}$, respectively;
- solar spectral irradiance distribution, $E(\lambda_w)$, and a function describing the photopic response of the eye, $R(\lambda_w)$. Both $E(\lambda_w)$ and $R(\lambda_w)$ consist of a set of function values listed for a set of discrete wavelength values. Function values at intermediate wavelengths can be found by linear interpolation;
- the longwave irradiance on the external and internal glazing surfaces, $G_{g,ex}$ and $G_{g,int}$, respectively, as well as the longwave irradiance at the external and internal frame surfaces, $G_{f,ex}$ and $G_{f,int}$, respectively. It is assumed that external longwave irradiance depends on the clearness of the sky factor, f_{clr} .

8.2 Reference boundary conditions

8.2.1 General

Unless a specific set of boundary conditions is of interest (e.g., to match test conditions, actual conditions or to satisfy a national standard), the following standard boundary conditions shall be used. In each case the following spectra shall be used.

$E_s(\lambda_w)$ = ISO 9845-1 (hemispherical solar spectral irradiance tabulated at N_s values of λ_w);

$E_v(\lambda_w)$ = ISO/CIE 10526 (colorimetric illuminance tabulated at N_{sv} values of λ_w);

$R(\lambda_w)$ = ISO/CIE 10527, (photopic response for the 2° observer tabulated at N_v values of λ_w).

8.2.2 Winter conditions

$$T_{\text{int}} = 20 \text{ °C}$$

$$T_{\text{ex}} = 0 \text{ °C}$$

$$h_{\text{cv,int}} = 3,6 \text{ W/(m}^2\cdot\text{K)}$$

$$h_{\text{cv,ex}} = 20 \text{ W/(m}^2\cdot\text{K)}$$

$$T_{\text{r,m}} = T_{\text{ex}}$$

$$I_{\text{s}} = 300 \text{ W/m}^2$$

8.2.3 Summer conditions

$$T_{\text{int}} = 25 \text{ °C}$$

$$T_{\text{ex}} = 30 \text{ °C}$$

$$h_{\text{cv,int}} = 2,5 \text{ W/(m}^2\cdot\text{K)}$$

$$h_{\text{cv,ex}} = 8 \text{ W/(m}^2\cdot\text{K)}$$

$$T_{\text{r,m}} = T_{\text{ex}}$$

$$I_{\text{s}} = 500 \text{ W/m}^2$$

8.3 Convective heat transfer

8.3.1 General

Convection heat transfer is energy transfer between a surface and a moving fluid. Heat is transferred by natural convection (i.e., convection driven by temperature gradient) when the air velocity is sufficiently small (i.e., less than 0,3 m/s). On the other hand, heat is transferred by forced and mixed convection for velocities above 0,3 m/s. Accurate determination of this convective heat transfer on both internal and external boundary surfaces is extremely difficult and can only be done by careful measurements and computer simulation. For these reasons, surface heat transfer coefficient correlations had been developed and are given in 8.3.2 and 8.3.3.

8.3.2 Convective heat transfer coefficient — internal side

8.3.2.1 General

The convective heat transfer on the internal side primarily occurs by natural convection, and rarely by mixed and forced convection. Standard boundary conditions assume natural convection on the internal side. The density of convective heat flow on the internal boundary is defined as:

$$q_{\text{cv,in}} = h_{\text{cv,in}}(T_{\text{s,in}} - T_{\text{in}}) \quad (130)$$

where $T_{\text{s,in}}$ is the temperature of any internal fenestration surface [i.e., $T_{\text{b,n}}$ (temperature of the internal glazing surface) or the temperature of the internal frame surface]. The convective heat transfer coefficient, $h_{\text{cv,in}}$ is determined from heat transfer correlations given in 8.3.2.2.

8.3.2.2 Heat transfer by natural convection

The natural convection heat transfer coefficient for the internal side, $h_{cv,int}$, is determined in terms of the Nusselt number, Nu .

$$h_{cv,int} = Nu \left(\frac{\lambda}{H} \right) \quad (131)$$

where λ is the thermal conductivity of air.

Nu is calculated as a function of the corresponding Rayleigh number based on the height, H , of the fenestration system, Ra_H

$$Ra_H = \frac{\rho^2 H^3 g C_p |T_{b,n} - T_{int}|}{T_{m,f} \mu \lambda} \quad (132)$$

where the various fluid properties are those of air evaluated at the mean film temperature:

$$T_{m,f} = T_{int} + \frac{1}{4}(T_{b,n} - T_{int}) \quad (133)$$

The internal convective heat transfer coefficient is a function of internal glazing layer surface temperature, $T_{b,n}$, for the case of natural convection so it is necessary to update the value of $h_{cv,int}$ as the solution of the glazed area heat transfer model proceeds.

Each of the following Equations (134) to (139) pertains to one particular value, or range, of tilt angle, γ . This categorization, as a function of γ , is based on the assumption that the internal environment is warmer than the internal glazing surface (i.e., $T_{int} > T_{b,n}$). If the reverse is true ($T_{int} < T_{b,n}$) it is necessary to seek the appropriate correlation on the basis of the complement of the tilt angle, $180^\circ - \gamma$, instead of γ , and to then substitute $180^\circ - \gamma$ instead of γ when the calculation is carried out.

- a) Windows inclined from 0° to 15° ($0^\circ \leq \gamma < 15^\circ$)

$$Nu_{int} = 0,13 Ra_H^{1/3} \quad (134)$$

- b) Windows inclined from 15° to 90° ($15^\circ \leq \gamma \leq 90^\circ$)

$$Nu_{int} = 0,56 (Ra_H \sin \gamma)^{1/4}; Ra_H \leq Ra_{cv} \quad (135)$$

$$Nu_{int} = 0,13 (Ra_H^{1/3} - Ra_{cv}^{1/3}) + 0,56 (Ra_{cv} \sin \gamma)^{1/4}; Ra_H > Ra_{cv} \quad (136)$$

$$Ra_{cv} = 2,5 \times 10^5 \left(\frac{e^{0,72\gamma}}{\sin \gamma} \right)^{1/5}; \gamma \text{ in degrees} \quad (137)$$

- c) Windows inclined from 90° to 179° ($90^\circ < \gamma \leq 179^\circ$)

$$Nu_{int} = 0,56 (Ra_H \sin \gamma)^{1/4}; 10^5 \leq Ra_H \sin \gamma < 10^{11} \quad (138)$$

- d) Windows inclined from 179° to 180° ($179^\circ < \gamma \leq 180^\circ$)

$$Nu_{int} = 0,58 Ra_H^{1/5}; Ra_H \leq 10^{11} \quad (139)$$

NOTE For more information on the convective heat transfer coefficient of still air see [27].

8.3.2.3 Forced convection (any tilt)

The following relation is to be used for the case of forced air flow on the internal side of a fenestration system.

$$h_{cv,int} = 4 + 4V_s \quad (140)$$

where V_s is free stream velocity near the fenestration surfaces in metres per second.

NOTE Equation (140) is taken from ISO 6946^[37].

8.3.3 Convective heat transfer coefficient — external side

8.3.3.1 General

The convective heat transfer on the external side primarily occurs by forced convection. For those situations where natural convection occurs, see 8.3.3.5. The density of convective heat flow on the external boundary is defined as:

$$q_{cv,ex} = h_{cv,ex} (T_{s,ex} - T_{ex}) \quad (141)$$

where $T_{s,ex}$ is the temperature of any external fenestration surface (i.e., T_f or the temperature of the external frame surface).

8.3.3.2 Different applications

There are two different applications that need to be considered for fenestration outside convective heat transfer coefficient correlations:

- a) fenestration product comparisons (rating), see 8.3.3.3;
- b) real building (field situation) fenestration component annual energy analysis, see 8.3.3.4.

8.3.3.3 Convective heat transfer coefficient correlation for product comparison or rating

The following relation is to be used for forced convection on the external side of a fenestration system:

$$h_{cv,ex} = 4 + 4V_s \quad (142)$$

8.3.3.4 Real building fenestration component annual energy analysis

Windows are major factors in overall energy performance of the real buildings. In order to be able to estimate the contribution to a building's thermal balance, it is necessary to define heat transfer characteristics of glazing as a function of different climate variables. Forced convective heat transfer on the external side of the building occurs between the surface of the building and the surrounding air at a rate that is determined by several factors. These include the temperature difference between the surface and the air, the speed and direction of any air movement (wind) over the building, and the shape and roughness of the building surface. Since those factors are highly variable, an exact mathematical analysis of the external surface convective heat transfer is not possible at this time due to the difficulty in defining the external surface and the highly variable wind conditions.

NOTE For more information on this procedure, see Reference [28]. The correlations are based on the building experimental work given in Reference [29].

Calculation procedure:

$$h_{cv,ex} = 4,7 + 7,6 V_s \tag{143}$$

where

— if the surface is windward:

$$V_s = 0,25 V; V > 2 \text{ m/s} \tag{144}$$

$$V_s = 0,5; V \leq 2 \text{ m/s} \tag{145}$$

where V is the wind velocity measured at a height of 10 m above ground level and V_s is free stream velocity near the fenestration surfaces;

— if the surface is leeward:

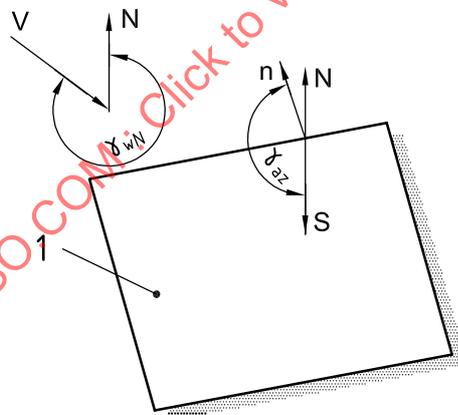
$$V_s = 0,3 + 0,05 V \tag{146}$$

To determine whether the surface is windward or leeward, calculate the wind direction, γ_w , relative to the wall surface (see Figure 26):

$$\gamma_w = \gamma_{az} + 180^\circ - \gamma_{wN} \tag{147}$$

If $|\gamma_w| > 180^\circ$, then $\gamma_w = 360^\circ - |\gamma_w|$

If $-45^\circ \leq |\gamma_w| \leq 45^\circ$, the surface is windward, otherwise the surface is leeward.



Key

- 1 building
- n wall normal direction
- γ_{wN} wind direction (angle measured clockwise from north)
- γ_{az} wall azimuth (positive degrees westward from south and negative eastward)
- N north
- S south

Figure 26 — Determination of wind direction and wall azimuth

8.3.3.5 Heat transfer by natural convection

The natural convection heat transfer coefficient for the external side, $h_{cv,ex}$, is determined in terms of the Nusselt number, Nu .

$$h_{cv,ex} = Nu \left(\frac{\lambda}{H} \right) \quad (148)$$

where λ is the thermal conductivity of air.

Nu is calculated as a function of the corresponding Rayleigh number based on the height, H , of the glazing, Ra_H .

$$Ra_H = \frac{\rho^2 H^3 g C_p |T_{s,ex} - T_{ex}|}{T_{m,f} \mu \lambda} \quad (149)$$

where the various fluid properties are those of air evaluated at the mean film temperature:

$$T_{m,f} = T_{ex} + \frac{1}{4}(T_{s,ex} - T_{ex}) \quad (150)$$

Correlations to quantify the external side convective heat transfer coefficient are identical to the ones for internal side and are presented in 8.3.2.1. The tilt angle γ needs to be replaced by its complement angle $180^\circ - \gamma$.

NOTE For more information, see [27].

8.4 Longwave radiation heat transfer

8.4.1 Mean radiant temperature

External mean radiant temperature will depend on the application, whether it is for field conditions or for product rating and comparison (i.e., controlled laboratory conditions). For field conditions, external irradiance can be defined through the use of external mean radiant temperature, $T_{rm,ex}$:

$$E_{ex} = \sigma T_{rm,ex}^4 \quad (151)$$

It is assumed that external fenestration surfaces are irradiated by the external surfaces and the sky vault consists of two areas, one cloudy and the other clear. The cloudy portion of the sky is treated as large enclosure surfaces existing at the external air temperature. The mean radiant external temperature can then be defined as:

$$T_{rm,out} = \left(\frac{(F_{gd} + (1 - f_{clr}) F_{sky}) \sigma T_{ex}^4 + f_{clr} F_{sky} J_{sky}}{\sigma} \right)^{1/4} \quad (152)$$

where F_{gd} and F_{sky} are view factors from the external surfaces of the fenestration system to the ground (i.e., the area below the horizon) and sky, respectively. The factor f_{clr} is the fraction of the sky that is clear.

$$F_{gd} = 1 - F_{sky} \quad (153)$$

$$F_{sky} = \frac{1 + \cos \gamma_w N}{2} \quad (154)$$

If the radiosity of the clear sky (J_{sky}) is known, it can be used directly in Equation (152).

NOTE Alternatively, if actual sky data are unavailable, the model from Reference [36] can be used.

$$J_{\text{sky}} = \varepsilon_{\text{sky}} \sigma T_{\text{ex}}^4 \quad (155)$$

$$\varepsilon_{\text{sky}} = \frac{R_{\text{sky}}}{\sigma T_{\text{ex}}^4} \quad (156)$$

$$R_{\text{sky}} = 5,31 \times 10^{-13} T^6 \quad (157)$$

Internal irradiance is defined as:

$$E_{\text{int}} = \sigma T_{\text{rm,int}}^4 \quad (158)$$

where $T_{\text{rm,int}}$ is determined from temperatures and shape factors of surrounding internal surfaces.

It is often assumed that internal fenestration surfaces are irradiated only by the internal room surfaces, which are treated as a large enclosure existing at the internal air temperature. Internal irradiance then becomes:

$$E_{\text{int}} = \sigma T_{\text{int}}^4 \quad (159)$$

The procedure outlined in this clause can be adapted to account for conditions that exist in a hot box test apparatus by determining the radiosities of the surfaces to which the window is exposed and the corresponding shape factors.

8.4.2 Detailed radiation heat transfer calculation

8.4.2.1 General

Fenestration systems whose ratio of total to projected boundary surface area on the internal/external side is greater than 1,25 are called non-planar fenestration systems. For these systems, individual fenestration surfaces (i.e., frame and glazing surfaces) are self-radiating and the assumption of large black body enclosure radiating at each fenestration surface with the view factor equal to 1,0 is invalid. This analysis may also be used for certain fenestration components, such as frame cavities and ventilated cavities and grooves.

The net radiation heat transfer on fenestration boundaries, q_r , of non-planar products shall be calculated using procedure outlined in 8.4.2.2 or the alternative procedure given in 8.4.2.3.

8.4.2.2 Two-dimensional element to element view factor based radiation heat transfer calculation

The emissivity of both internal and external environments is set to unity.

The net radiation heat transfer at any surface "i" is the difference between emitted radiation and absorbed portion of incident radiation. The temperatures of the surfaces do not appreciably differ so, using Kirchhoff's law:

$$q_{r,i} = \varepsilon_i \sigma T_i^4 - \varepsilon_i E_i \quad (160)$$

where E_i is irradiance at surface i from all other surfaces.

$$E_i = \sum_j^N F_{i-j} J_j \quad (161)$$

and F_{i-j} is the view factor from surface i to surface j . The radiosity of surface j , J_j , is given by:

$$J_j = \varepsilon_j \sigma T_j^4 + \rho_j E_j \quad (162)$$