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Thermal Performance of Windows, Doors and Shading Devices — Detailed Calculations

Performance Thermique de Vitrages, Portes et francais francais francais — Calculs detaillés

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Contents

Foreword

ISO (the International Organization for Standardization) is a world-wide 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 3.

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 International Standard may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

International Standard ISO 15099 was prepared by Technical Committee ISO/TC 163, *Thermal Insulation*,

— *Part [n]:*

— *Part [n+1]:*

Introduction

This standard describes a procedure to calculate indices of merit of many window and door products. The method provided in this standard will allow the user to determine total window and door product indices of merit: thermal transmittance (U_t), Total solar energy transmittance (g-value) and visible transmittance (τ_t).

Traditionally, windows have been characterised 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 in ISO 10292 (equivalent EN 673) (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. The approach taken in ISO/DIS 15099 is different. In this standard the energy balance equations are set up for every glazing layer taking into account the solar absorptance and actual temperatures. From these energy balance equations the temperatures of the individual layers and gaps are determined. The ISO/DIS 15099 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 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 in ISO/DIS 15099 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 standard is intended for use in computer programs. It was never intended as a "simplified calculation" procedure. Simplified methods are provided in other ISO standards. It is essential that these programs produce consistent values and that they are soundly based on a standard methodology. Although more complicated than the formulas used in the simplified standards, the formulas used in ISO/DIS 15099 are entirely appropriate for their intended use.

It is the intention of this working group to make use of other ISO standards which provide complementary information and data required for the execution of the algorithms involved. These ISO documents and other relevant documents are referenced herein.

Thermal Performance of Windows, Doors and Shading Devices — Detailed Calculations

1 Scope

This standard specifies detailed calculation procedures to determine 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 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 normative documents contain provisions, which through reference in this text, constitute provisions of this International Standard. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this International Standard are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.

ISO 6946: *Building components and elements -Thermal resistance and thermal transmittance - Calculation method.*

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 8990: *Thermal insulation - Determination of steady-state thermal resistance and related properties - Calibrated and guarded hot box.*

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

ISO 9251, *Thermal Insulation - Heat Transfer Vocabulary.*

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 and normal hemispherical solar irradiance for air mass 1.5.*

ISO/DIS 10077-1 *Thermal performance of windows, doors and shutters – Calculation of thermal transmittance - Part 1: Simplified method.*

ISO/DIS 10077-2: *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 flow and surface temperatures, Part 1. General calculation methods.*

ISO 10292: *Glass in building – Calculation of steady state U-values (thermal transmittance) of multiple glazing.*

ISO/DIS 12567: *Thermal performance of doors and windows – Determination of thermal transmittance by hot box method.*

ISO/CIE 10526: *CIE standard colorimetric illuminats.*

ISO/CIE 10527: CIE standard colorimetric observers.

3 Symbols and abbreviated terms

3.1 Symbols and units

Symbols and units used are in accordance with ISO 7345 and ISO 9288. The quantities, which are specific to this standard, are defined in table 3.2.

3.2 Symbols

Table 3.2 Symbols

Symbol	Definition	Units
A	area	m^2
	or aspect ratio	
C_p	specific heat at constant pressure	
d	thickness (d_g= thickness of glazing cavity)	m
$E_s(\lambda)$	solar spectral irradiance function (ISO 9845)	
$E_v(\lambda)$	colorimetric illuminance (CIE D65 function, ISO 10526)	
g	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 indoor conditioned space	
	or acceleration due to gravity	m^2/s
G	irradiance	W/m^2
	or parameter defined in Equation (45)	
h	film heat transfer coefficient	$W/(m^2·K)$
H	distance (height of glazing cavity)	m
$I_i^+(\lambda)$, $I_i^-(\lambda)$	spectral flux of radiant solar energy between i^{th} and $i+1^{th}$ glazing layers travelling in the external or i internal direction, respectively.	
I	total flux of incident solar radiation	W/m^2
J	radiosity	W/m^2
L	height of glazing cavity	m
U	thermal transmittance	$W/(m^2·K)$
l	length	m
n	number of glazings + 2	
$\hat{M}$	molecular mass	mole
Nu	Nusselt number	
P	pressure	Pa
q	density of heat flow rate	W/m^2
Q	heat flow rate	W
$\Re$	universal gas constant	
R	thermal resistance	$m^2·K/W$
Ra	Rayleigh number	
Ra_x	Rayleigh number based on length dimension x	
$R(\lambda)$	photopic response of the eye (ISO/CIE 10527)	
S_i	flux of absorbed solar radiation at i^{th} glazing layer	W/m^2
t	thickness	m
t_{perp}	largest dimension of frame cavity perpendicular to heat flow	m
T	temperature	K
V	free-stream air speed near window	m/s
x,y	dimensions in a Cartesian co-ordinate system	
Z	pressure loss factor	

α	absorptance	
β	thermal expansion coefficient of fill gas	K^{-1}
ΔT_i	temperature drop across i^{th} glazing cavity, $\Delta T_i = \lvert T_{f,i} - T_{b,i+1} \rvert$	
ε	total hemispherical emissivity	
θ	temperature	°C
	or angle	degree
σ	Stefan-Boltzmann constant, 5.6693×10^{-8}	$W/(m^2 \cdot K^4$
φ	function defined in equation (65)	
λ	thermal conductivity	W/(m·K)
	or wavelength	m
μ	dynamic viscosity	g/(m·s)
ρ	density	kg/m^3
	or specular reflectance: portion of incident radiation reflected such that the	
τ	transmittance	
Ψ	linear thermal transmittance	W/(m·K)

3.3 Subscripts

The subscripts indicated in table 3.3 are used.

Table 3.3 Subscripts

Abbreviation	Name
ave	average
air	air
b	backward
c	convection
cold	condition on the cold side
crit	critical
d	divider
de	divider edge glass
diff	diffuse
dir	direct
eff	effective
e	edge of glass
f	frame
f	front
fr	frame (using the alternate approach)
g	glass or vision portion
h	hot
h	horizontal
hot	condition on the hot (warm) side
in	indoor
j	counter
mix	mixture
n	counter
ne	environmental (outdoor)
ni	environmental (indoor)
out	outdoor
p	panel
r	radiation
r,m	mean radiant
red	reduced radiation
s	solar
s	source or sink

s	surface
surf	surface
t	total
v	vertical
x	at distance x
Ψ	perimeter
2D	coupling

4 Determination of total window and door system properties

This 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.

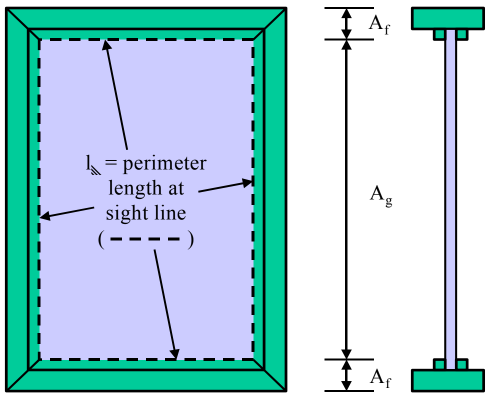


Figure 1 - Schematic diagram showing the window projected areas and vision perimeter

Clause 5 contains the procedure for calculating the required center-glass properties, U_g, g_g and τ_g . Clause 6 contains the procedure to calculate the corresponding frame properties, U_f and g_f , as well as the linear thermal transmittance, Ψ, which accounts for the interaction between frame and glazing or opaque panel. Clause 4 contains the procedure to calculate thermal transmittance, total solar transmittance and visible transmittance for the complete product. Clause 4.1 contains the procedure to calculate thermal transmittance. The effect of three-dimensional heat transfer in frames and glazing units is not considered. Clause 4.1.3 contains an alternate procedure to calculate edge of glass and frame thermal indices U_e, U_d, U_{de}, and U_{fr}, which are used in area-based calculations. Clause 7 contains the procedure for dealing with shading devices and ventilated windows. Clause 8 contains the procedure to determine and apply boundary conditions.

4.1 Thermal transmittance

The thermal transmittance of the fenestration product is:

$$U_t = \frac{\sum A_g U_g + \sum A_f U_f + \sum \ell_\psi \Psi}{A_t} \quad (1)$$

where, A_g and A_f are the projected vision area and frame area, respectively. The length of the vision area perimeter is l_Ψ. Ψ is linear thermal transmittance and can account 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. For example, several values of A_f must be used to sum the contributions of different values of U_f corresponding to sill, head, dividers and side jambs.

4.1.1 Glazed area thermal transmittance

The thermal transmittance can be found by simulating a single environmental condition involving indoor/outdoor temperature difference - with or without incident solar radiation. The thermal transmittance is the reciprocal of the thermal resistance.

$$U_g = \frac{1}{R_t} \quad (2)$$

R_t is found by summing the thermal resistances at the outdoor and indoor boundary, and thermal resistances of glazing cavities and glazing layers. See Figure 2.

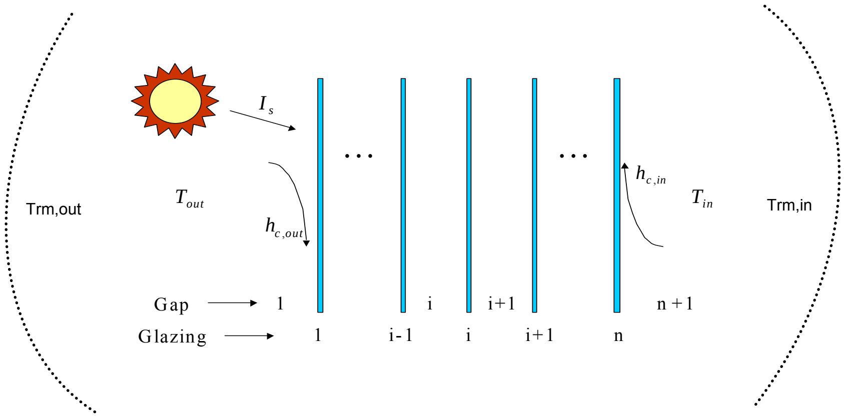


Figure 2 - Numbering system for glazing system layers

$$R_t = \sum_{i=1}^{n+1} R_i + \sum_{i=1}^{n} R_{g,i} \qquad (3)$$

where the thermal resistance of the i^{th} glazing is:

$$R_{g,i} = \frac{t_{g,i}}{\lambda_{g,i}} \qquad (4)$$

and the thermal resistance of the i^{th} space, where the 1st space is outdoor environment, the last space is indoor environment, and the spaces in between are glazing cavities, is:

$$R_i = \frac{T_{ni} - T_{ne}}{q_i} \qquad (5)$$

Where the environmental temperature is a weighted average of the ambient air temperature and the mean radiant temperature, T_{rm}, which is determined for outdoor and indoor environment in boundary conditions Clause 8.3.1.

T and T_{rm} are weighted, respectively, according to the relative magnitudes of the convective and radiative heat transfer coefficients (h_c and h_r) that exist between the surfaces of the fenestration system and the environment. Thus, T_n is:

$$T_n = \frac{h_c T_{air} + h_r T_{rm}}{h_c + h_r} \qquad (6)$$

where h_c and h_r are determined according to the procedure given in Clause 8.

4.1.2 Frame area/edge-glass thermal indices

In order to convert the results of a 2-D numerical analysis to thermal transmittances, it is necessary to record the rate of heat transfer from the indoor 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.

$$\Psi = L^{2D} - U_f l_f - U_g l_g \qquad (7)$$

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

$$U_f = \frac{L^{2D} - U_p \cdot l_p}{l_f} \qquad (8)$$

where L_p^{2D} is thermal coupling coefficient determined from the frame/foam insert system. U_p is the thermal transmittance of foam insert, l_p is the indoor side exposed length of foam insert (minimum 100 mm), l_f is the indoor side projected length of the frame section and l_g is the indoor side projected length of the glass section (see ISO 10077-2, figures C1 and C2, 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.

4.1.3 Alternate approach

An alternate 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_g, is divided into center-glass area, A_c, plus edge-glass area, A_e, and one additional thermal transmittance, U_e, is used to characterize the edge-glass area. If dividers are present then divider area, A_d and divider thermal transmittance, U_d are calculated, as well as corresponding divider edge area, A_{de} and thermal transmittance, U_{de}. The following formula shall be used to calculate the total thermal transmittance:

$$U_t = \frac{\sum U_c A_c + \sum U_{fr} A_f + \sum U_e A_e + \sum U_d A_d + \sum U_{de} A_{de}}{A_t} \tag{9}$$

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

$$U_{fr} = \frac{Q_{fr}}{l_f (T_{in} - T_{out})} \tag{10}$$

$$U_e = \frac{Q_e}{l_e (T_{in} - T_{out})} \tag{11}$$

and where l_f is projected length of frame area and l_e is the length of edge of glass area and is equal to 63.5 mm. These lengths are measured from indoor side. The quantities Q_{fr} and Q_e are heat flow rates through frame and edge-glass areas (indoor surfaces), respectively, including the effect of glass and spacer, and both are expressed per unit length of frame or edge-glass (i.e., W/m).

The summations included in equation (9) are used to account for the various sections of one particular component type. For example, 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.

NOTE 1 It should be noted that the two different approaches entail different definitions of frame thermal transmittance, denoted U_f and U_{fr}. The comparison of frame properties for two different products is only meaningful if the same calculation procedure has been used in both cases.

NOTE 2 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 three dimensional effects are neglected. This difference is more pronounced for smaller windows. The choice of l_e = 63.5 mm is made to reduce the discrepancy between the two alternate approaches.

4.2 Total solar energy transmittance

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

$$g_t = \frac{\sum g_g A_g + \sum g_f A_f}{A_t} \tag{12}$$

where, g_g and g_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.

4.2.1 Vision area total solar energy transmittance

The total solar energy transmittance can be determined for conditions involving internal/outdoor temperature difference and any level of incident solar radiation. It is found by summing the directly transmitted solar gain and the absorbed/redirected solar gain.

$$g_g = \tau_s + \sum_{i=1}^{n} \left\{ \frac{N_i S_i}{I_s} \right\} \tag{13}$$

N_i is the portion of S_i that finds its way to the indoor space (1).

$$N_i = \frac{\sum_{j=1}^{i} R_j + \sum_{j=1}^{i-1} R_{gl,j} + \frac{1}{2} R_{gl,i}}{R_{tot}} \tag{14}$$

NOTE R_j and R_{tot} are determined using the actual temperatures of the various glazing layers.

The relationship between S_i and I_s is given by equation (18).

For vented cavities, see Clause 7.

4.2.2 Frame total solar energy transmittance

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

$$g_f = \alpha_f \cdot \frac{U_f}{\frac{A_{surf}}{A_f} h_{out}} \tag{15}$$

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

NOTE If the alternate method of calculating U_t is being used, U_{fr} should be used instead of U_f in equation (13).

4.3 Visible transmittance

The visible transmittance of the total fenestration product is:

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

5 Vision area properties

NOTE 1 For glazing units only, ISO 9050 and ISO 10292 should be used.

NOTE 2 Clause 7 contains the extensions needed to model vented windows.

5.1 Glazing layer optics

5.1.1 Solar

The solar optical properties needed to describe the i^{th} glazing are (1) the front (outdoor side) spectral reflectance, $\rho_{f,i}(\lambda)$, (2) the back (indoor side) spectral reflectance, $\rho_{b,i}(\lambda)$, and (3) the spectral transmittance, $\tau_i(\lambda)$. See Figure 3.

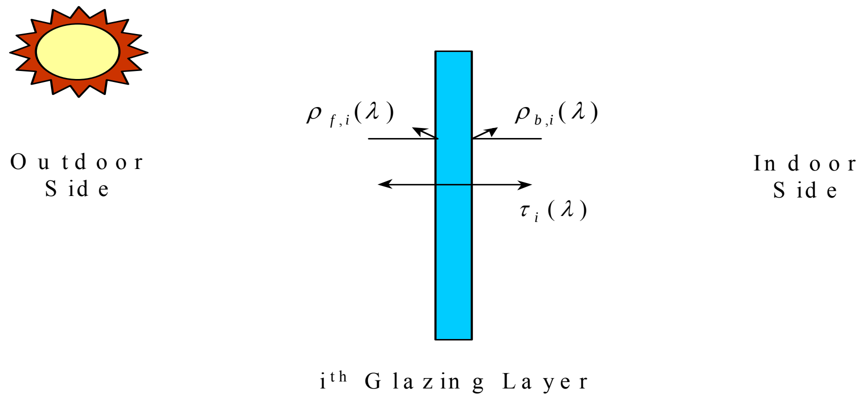


Figure 3 - Out-door and indoor spectral transmittance of a glazing layer

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

5.1.2 Long-wave

The long-wave optical properties needed to describe the i^{th} glazing are (1) the front (outdoor side) hemispheric emittance, $\varepsilon_{f,i}$, (2) the back (indoor side) hemispheric emittance, $\varepsilon_{b,i}$ and (3) the hemispheric-hemispheric transmittance, τ_i. These total optical properties apply to wavelengths from 5 to 40 µm.

The long-wave reflectance data are to be measured in accordance with ASTM E1585-93 (3) or prEN 12898. Values of the normal emittance resulting from this procedure are to be converted to hemispherical emittance using ASTM E1585-93 (3) or prEN 12898.The integration needed to convert measured spectral data to the required total longwave optical properties, $\varepsilon_{f,i}$, $\varepsilon_{b,i}$ and τ_i are also to be carried out in accordance with ASTM E 1585-93 (3) or prEN 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-emittance coating they are generally opaque in the infrared, so that τ_i=0 and hemispherical emittance can be calculated as in ASTM E1585-93 (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 (4).

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 provided in ISO 9050 or any other exact method. Depending on future modifications of ISO 9050, specific additions may be added to this standard covering the effects of optical properties of products (shading devices, diffusing panes, etc.) not adequately covered by ISO 9050.

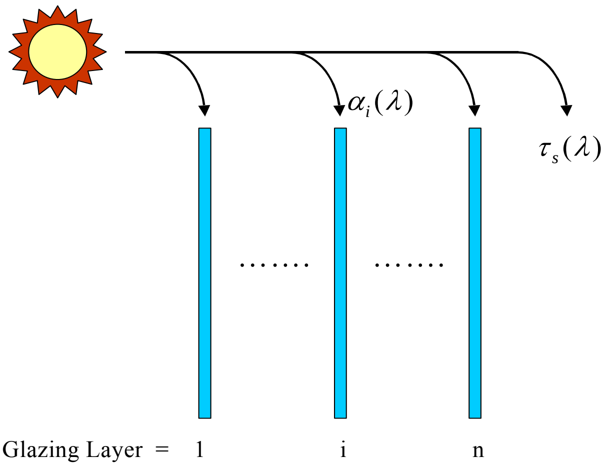


Figure 4 - Absorptance of the i^{th} glazing layer and solar spectral transmittance

Figure 4 shows how a window with *n* glazing layers together with the outdoor (*i*=0) and indoor (*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 $\alpha_i(\lambda)$ at the i^{th} glazing layer. Similarly, it is necessary to determine the solar spectral transmittance of the glazing system, $\tau_s(\lambda)$.

These quantities, $\alpha_i(\lambda)$ and $\tau_s(\lambda)$, 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 be exact (e.g., (5) is acceptable). The solution technique described by (5) is summarized in Annex A.

5.2.2 The solar spectrum

The spectral distribution of the incident solar radiation, $E(\lambda)$, is needed to calculate total optical properties and various total energy fluxes. Values of $E(\lambda)$ are reported at N_s values of λ (denoted here as $E(\lambda_j)$ and λ_j, respectively). Intermediate values of $E(\lambda)$ shall be found by linear interpolation of the tabulated values.

5.2.3 Absorbed amounts of solar radiation

The total flux of solar radiation absorbed at the i^{th} glazing layer, S_i, is determined by numerical integration over the solar spectrum according to equations (17) and (18).

$$A_i = \frac{\sum_{j=1}^{N_s-1} \alpha_i(\lambda_{j/j+1}) E_s(\lambda_{j/j+1}) \Delta\lambda_j}{\sum_{j=1}^{N_s-1} E_s(\lambda_{j/j+1}) \Delta\lambda_j} \qquad \Delta\lambda_j = \lambda_{j+1} - \lambda_j \tag{17}$$

$$S_i = A_i \cdot I_s \tag{18}$$

where $\alpha_i(\lambda_{j/j+1})$ is the value of α_i that is representative of the wavelength band from λ_j to λ_{j+1} and is given by,

$$\alpha_i(\lambda_{j/j+1}) = \tfrac{1}{2}\alpha_i(\lambda_j) + \tfrac{1}{2}\alpha_i(\lambda_{j+1}) \qquad (19)$$

and

$$E_s(\lambda_{j/j+1}) = \frac{E_s(\lambda_j) + E_s(\lambda_{j+1})}{2} \qquad (20)$$

Values of $E_s(\lambda)$ are given in ISO 9845.

5.2.4 Solar transmittance

The solar transmittance of the glazing system is:

$$\tau_s = \frac{\sum_{j=1}^{N_s-1} \tau_s(\lambda_{j/j+1}) E_s(\lambda_{j/j+1}) \Delta\lambda_j}{\sum_{j=1}^{N_s-1} E_s(\lambda_{j/j+1}) \Delta\lambda_j} \qquad \Delta\lambda_j = \lambda_{j+1} - \lambda_j \qquad (21)$$

where $E_s(\lambda_{j/j+1})$ is given by equation (20) and

$$\tau_s(\lambda_{j/j+1}) = \tfrac{1}{2}\tau_s(\lambda_j) + \tfrac{1}{2}\tau_s(\lambda_{j+1}) \qquad (22)$$

5.2.5 Visible transmittance

Visible transmittance, τ_v, is calculated using a weighting function that represents the photopic response of the eye, $R(\lambda)$. $R(\lambda_j)$ is tabulated for N_v values of λ_j. τ_v is given by:

$$\tau_v = \frac{\sum_{j=1}^{N_v-1} \tau_s(\lambda_{j/j+1}) E_v(\lambda_{j/j+1}) R(\lambda_{j/j+1}) \Delta\lambda_j}{\sum_{j=1}^{N_v-1} E_v(\lambda_{j/j+1}) R(\lambda_{j/j+1}) \Delta\lambda_j} \qquad \Delta\lambda_j = \lambda_{j+1} - \lambda_j \qquad (23)$$

where

$$R(\lambda_{j/j+1}) = \frac{R(\lambda_j) + R(\lambda_{j+1})}{2} \qquad (24)$$

$$E_v(\lambda_{j/j+1}) = \tfrac{1}{2}E_v(\lambda_j) + \tfrac{1}{2}E_v(\lambda_{j+1}) \qquad (25)$$

Values of $E_v(\lambda)$ are given in ISO 10526.

and $\tau_s(\lambda_{j/j+1})$ is given by equation (22).

5.3 Vision area heat transfer

5.3.1 Glazing layer energy balance

Radiative heat 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 on correlations based on experimental data.

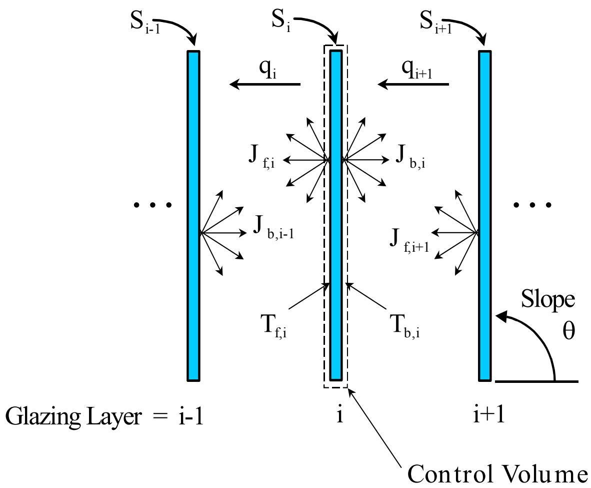


Figure 5 - Energy balance on the ith glazing layer

Figure 5 shows the ith glazing in a sloped multilayer array. The values of four variables are sought at each glazing. These are the temperatures of the outdoor and indoor facing surfaces, $T_{f,i}$ and $T_{b,i}$, plus the radiant fluxes leaving the front and back facing surfaces (i.e., the radiosities), $J_{f,i}$ and $J_{b,i}$. In terms of these variables q_i, is:

$$q_{\rm i} = h_{\rm c,i}\left[T_{\rm f,i} - T_{\rm b,i-1}\right] + J_{\rm f,i} - J_{\rm b,i-1} \tag{26}$$

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

$$q_{\rm i} = S_{\rm i} + q_{\rm i+1} \tag{27}$$

$$J_{\rm f,i} = \varepsilon_{\rm f,i}\sigma T_{\rm f,i}^4 + \tau_{\rm i} J_{\rm f,i+1} + \rho_{\rm f,i} J_{\rm b,i-1} \tag{28}$$

$$J_{\rm b,i} = \varepsilon_{\rm b,i}\sigma T_{\rm b,i}^4 + \tau_{\rm i} J_{\rm b,i-1} + \rho_{\rm b,i} J_{\rm f,i+1} \tag{29}$$

$$T_{\rm b,i} - T_{\rm f,i} = \frac{t_{\rm gl,i}}{2\lambda_{\rm gl,i}}\left[2q_{\rm i+1} + S_{\rm i}\right] \tag{30}$$

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

$$T_i(x) = \left[\frac{-S_i}{2k_{gl,i}t_{gl,i}}\right]x^2 + \left[\frac{T_{f,i} - T_{b,i}}{t_{gl,i}} + \frac{S_i}{2k_{gl,i}}\right]x + T_{b,i} \quad (31)$$

[DC: Hakim, note change in equation 31; first *S_i* term has "–" sign] where *x* is the distance from the indoor surface of the glazing.

Equation (27) describes an energy balance imposed at the surfaces of the i^{th} glazing. Equations (28) and (29) define the radiosities at the i^{th} glazing.

NOTE $\rho_{f,i} = 1 - \tau_i - \varepsilon_{f,i}$ and $\rho_{b,i} = 1 - \tau_i - \varepsilon_{b,i}$. The temperature difference across the i^{th} glazing is given by equation (30). It is assumed that the solar energy is absorbed uniformly through the thickness of the glazing. More detail regarding equations (26) through (31) is given by Wright (5).

5.3.2 Interaction with the environment

The effect of boundary conditions imposed by the environment on the window must be specified. The indoor and outdoor temperatures, $T_{f,n+1}$ and $T_{b,0}$, are:

$$T_{f,n+1} = T_{in} \quad (32)$$

$$T_{b,0} = T_{out} \quad (33)$$

The effect of long-wave irradiance at the glazing surfaces is included by setting

$$J_{f,n+1} = G_{g,in} \quad (34)$$ and $$J_{b,0} = G_{g,out} \quad (35)$$

where $G_{g,in}$ and $G_{g,out}$ are given by equations (133) and (131) in Section 8, respectively.

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

$$h_{c,n+1} = h_{c,in} \text{ and} \quad (36)$$

$$h_{c,1} = h_{c,out} \quad (37)$$

5.3.3 Convective heat transfer coefficient - glazing cavities

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

$$h_{c,i} = Nu_i \left(\frac{\lambda_{g,i}}{d_{g,i}}\right) \quad (38)$$

where $d_{g,i}$ is the thickness of the i^{th} fill gas layer (or pane spacing) and $\lambda_{g,i}$ 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, $A_{g,i}$, and the cavity slope, θ.

NOTE It should be recognised 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. 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 "g" subscripts for convenience):

$$Ra = \frac{\rho^2 d^3 g \beta C_p \Delta T}{\mu \lambda} \quad \text{(dimensionless)} \tag{39}$$

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

$$\beta = \frac{1}{T_m} \quad T_m = \text{fill gas mean temperature (K)} \tag{40}$$

The aspect ratio of the i^{th} fill gas cavity is:

$$A_{g,i} = \frac{H}{d_{g,i}} \tag{41}$$

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 following clauses. Each of these clauses pertains to one particular value, or range, of tilt angle, θ.

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

5.3.3.1 Cavities inclined at $0 \le \theta < 60°$ (7)

$$Nu_i = 1 + 1.44\left[1 - \frac{1708}{Ra\cos(\theta)}\right]^{\bullet}\left[1 - \frac{\left(1708\sin^{1.6}(1.8\theta)\right)}{Ra\cos(\theta)}\right] + \left[\left[\frac{Ra\cos(\theta)}{5830}\right]^{1/3} - 1\right]^{\bullet}$$

$$Ra < 10^5 \quad \text{and} \quad A_{g,i} > 20 \tag{42}$$

$$\text{where } [x]^{\bullet} = \left(x + |x|\right)/2 \tag{43}$$

5.3.3.2 Cavities inclined at $\theta = 60°$ (8)

$$Nu = \left[Nu_1, Nu_2\right]_{max} \tag{44}$$

$$\text{where } Nu_1 = \left[1 + \left[\frac{0.0936Ra^{0.314}}{1+G}\right]^7\right]^{1/7} \tag{45}$$

$$Nu_2 = \left[0.104 + \frac{0.175}{A_{g,i}}\right]Ra^{0.283} \tag{46}$$

$$G = \frac{0.5}{\left[1+\left[\frac{Ra}{3160}\right]^{20.6}\right]^{0.1}} \quad (47)$$

5.3.3.3 Cavities inclined at 60° < θ < 90° (8)

For layers inclined at angles between 60° and 90°, a straight line interpolation between the results of equations (42) and (46) is used. These equations are valid in the ranges of

$10^2 < Ra < 2\times10^7$ and $5 < A_{g,i} < 100$

5.3.3.4 Vertical cavities (9)

$$Nu = \left[Nu_1, Nu_2\right]_{max} \quad (48)$$

$$Nu_1 = 0{,}0673838 Ra^{1/3} \quad 5\times10^4 < Ra < 10^6 \quad (49)$$

$$Nu_1 = 0{,}028154 Ra^{0{,}4134} \quad 10^4 < Ra \le 5\times10^4 \quad (50)$$

$$Nu_1 = 1 + 1{,}7596678\times10^{-10} Ra^{2{,}2984755} \qquad Ra \le 10^4 \quad (51)$$

$$Nu_2 = 0{,}242\left[\frac{Ra}{A_{g,i}}\right]^{0{,}272} \quad (52)$$

[DC: Hakim, note editorial changes in equations 49-52. periods converted to comas]

5.3.3.5 Cavities inclined from 90° to 180° (10)

Gas layers contained in downward facing windows are modelled using:

$$Nu = 1 + \left[Nu_v - 1\right]\sin\theta \quad (53)$$

Nu_v is the Nusselt number for a vertical cavity given by equation (48).

5.3.3.6 Fill gas properties

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

$$\rho = \frac{P\hat{M}}{\Re T_m} \quad (54)$$

P= 101 300 Pa and T_m =293 K.

The specific heat capacity, C_p, and the transport properties μ and λ are evaluated using linear functions of temperature. For example, the viscosity can be expressed as:

$$\mu = a + bT_m \quad (55)$$

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

5.3.4 Properties of fill gas mixtures

The density, conductivity, viscosity and specific heat of gas mixtures can be calculated as a function of the corresponding properties of the individual constituents (11).

5.3.4.1 Molecular mass

$$\hat{M}_{mix} = \sum_{i=1}^{\nu} x_i \hat{M}_i \qquad (56)$$

where x_i is the mole fraction of the i^{th} gas component in a mixture of υ gases.

5.3.4.2 Density

$$\rho = \frac{P\hat{M}_{mix}}{\Re T_m} \qquad (57)$$

5.3.4.3 Specific heat

$$C_{p\,mix} = \frac{\hat{C}_{p\,mix}}{\hat{M}_{mix}} \qquad (58a)$$

$$\hat{C}_{p\,mix} = \sum_{i=1}^{\upsilon} x_i \hat{C}_{p,i} \qquad (58)$$

[DC: Hakim, note the change in equation 58 and addition of equation 58a]

where, the molar specific heat of the i^{th} gas is

$$\hat{C}_{p,i} = C_{p,i}\hat{M}_i \qquad (59)$$

5.3.4.4 Viscosity

$$\mu_{mix} = \sum_{i=1}^{\upsilon} \frac{\mu_i}{\left\{1 + \sum_{\substack{j=1\\ j\neq i}}^{\upsilon} \phi^{\mu}_{i,j} \frac{x_j}{x_i}\right\}} \qquad (60)$$

where

$$\phi^{\mu}_{i,j} = \frac{\left[1 + \left(\mu_i / \mu_j\right)^{1/2} \left(\hat{M}_j / \hat{M}_i\right)^{1/4}\right]^2}{2\sqrt{2}\left[1 + \left(\hat{M}_i / \hat{M}_j\right)\right]^{1/2}} \qquad (61)$$

5.3.4.5 Thermal conductivity

$$\lambda_{mix} = \lambda'_{mix} + \lambda''_{mix} \tag{62}$$

where λ' is the monatomic thermal conductivity and λ'' is included to account for additional energy moved by the diffusional transport of internal energy in polyatomic gases.

$$\lambda'_{mix} = \sum_{i=1}^{\upsilon} \frac{\lambda'_i}{\left\{1 + \sum_{\substack{j=1 \\ j \neq i}}^{\upsilon} \psi_{i,j} \frac{x_j}{x_i}\right\}} \tag{63}$$

and,

$$\psi_{i,j} = \frac{\left[1 + \left(\lambda'_i / \lambda'_j\right)^{1/2} \left(\hat{M}_i / \hat{M}_j\right)^{1/4}\right]^2}{2\sqrt{2}\left[1 + \left(\hat{M}_i / \hat{M}_j\right)\right]^{1/2}} \bullet \left[1 + 2.41 \frac{\left(\hat{M}_i - \hat{M}_j\right)\left(\hat{M}_i - 0.142\hat{M}_j\right)}{\left(\hat{M}_i + \hat{M}_j\right)^2}\right] \tag{64}$$

and,

$$\lambda''_{mix} = \sum_{i=1}^{\upsilon} \frac{\lambda''_i}{\left\{1 + \sum_{\substack{j=1 \\ j \neq i}}^{\upsilon} \phi^{\lambda}{}_{i,j} \frac{x_j}{x_i}\right\}} \tag{65}$$

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

$$\phi^{\lambda}{}_{i,j} = \frac{\left[1 + \left(\lambda'_i / \lambda'_j\right)^{1/2} \left(\hat{M}_i / \hat{M}_j\right)^{1/4}\right]^2}{2\sqrt{2}\left[1 + \left(\hat{M}_i / \hat{M}_j\right)\right]^{1/2}} \tag{66}$$

[DC: Hakim, note revisions to equations 60, 61, 65 and 66]

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

1. calculate λ'_i

$$\lambda'_i = \frac{15}{4} \frac{\Re}{\hat{M}_i} \mu_i \tag{67}$$

2. calculate λ''_i

$\lambda_i'' = \lambda_i - \lambda_i'$ λ_i is the conductivity of the i[th] fill gas component (Annex B) (68)

3. use λ_i' to calculate λ_{mix}'

4. use λ_i'' to calculate λ_{mix}''

5. $\lambda_{mix} = \lambda_{mix}' + \lambda_{mix}''$

6 Frame effects

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)

6.1 Area and lineal thermal transmittance

Frame areal thermal transmittance (Thermal transmittances) and lineal thermal transmittance (ψ-factor) shall be determined using two-dimensional (2-D) numerical modelling. This 2-D 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 2-D numerical analysis are provided in clauses 6.2 through 6.6 (also see ISO/DIS 10077, part 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 an energy flow between two specific nodes or across any given control volume face must 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}, must be specified. The numerical solver shall be able to generate the 2-D heat flow and temperature patterns that satisfy the governing equation. In Cartesian co-ordinates this equation is:

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} = -\frac{q'''_{source}}{\lambda} \tag{69}$$

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

$$q = -\lambda_{fe}\left(\frac{\partial T}{\partial x}e_x + \frac{\partial T}{\partial y}e_y\right) \tag{70}$$

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:

$$q = q_c + q_r + q_s \tag{71}$$

where, q_c is convective component, and q_r is radiative component of the density of heat flow rate, which shall be determined in accordance with clause 7.2 and 7.3 respectively. Quantity q_S is 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 can be ignored. Reinforcing or operating hardware that is essentially continuous, assembly screws or bolts that extend from the indoor to the outdoor side or bridge a thermal break, including incompletely de-bridged thermal break, shall be included in the model. These thermal-bridging elements may be modelled with 3-D calculational tools when available, otherwise they shall be modelled using the procedure outlined below (34):

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

$$\lambda_{eff} = F_b \cdot \lambda_b + (1 - F_b)\lambda_n$$

where:

s = size of thermal bridging element (e.g., size of a bolt head)

d = spacing of thermal bridging elements

λ_b = conductivity of thermal bridging material

λ_n = 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 $F_b \leq 1\%$, ignore thermal bridge

b) If $1\% < F_b \leq 5\%$, model using the above method providing that $\lambda_b > 10 \cdot \lambda_n$

c) If $F_b > 5\%$, always model using the above method.

Components in the window assembly that are compressed or deformed from their original shape once installed in the window (e.g. weather-stripping) shall be modeled in the compressed or deformed configuration. Adjustments to the dimensions of the geometric model are allowed only if they have no significant influence to the calculation (ISO 10211-1, 1996). More specifically, 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 1-4 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 1-4 below (See Figure 6).

1) The thickness of the representation (t) is equal to the average thickness +/- manufacturing tolerances.

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 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 x conductivity shall be replaced by web thickness x conductivity x (cos(γ) + sin(γ)) 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 (t) +/- manufacturing tolerances.

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 indoor and outdoor boundaries shall follow the frame profile. In the case where there are exterior and interior open frame profile cavities, the external and/or internal boundary conditions should only be applied to a depth that has the same dimension as the cavity width (12). For the remainder of the open frame profile cavity, an equivalent thermal conductivity determined in the same manner similar to that which is specified for internal enclosed frame cavities (see Clause 6.6).

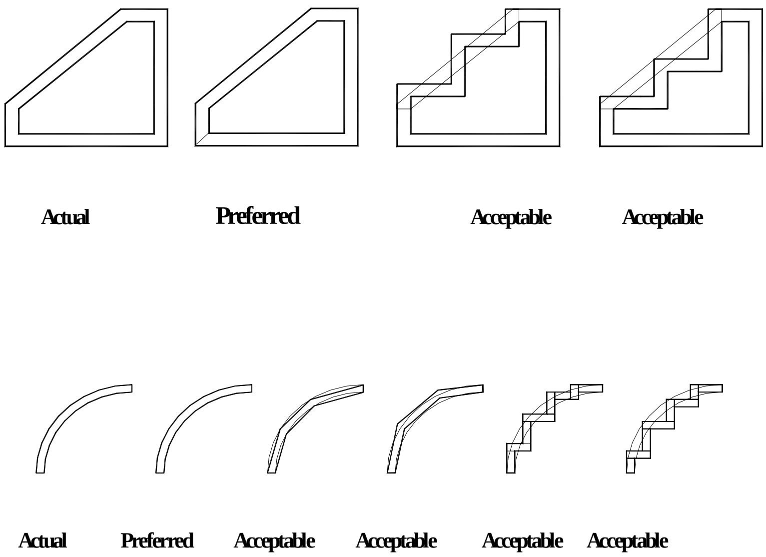


Figure 6 - Examples of possible approximations of the actual cross section

6.3.2 Meshing

The two-dimensional (2-D) 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 2-D heat transfer equation given in Clause 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:

1. 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 2-D 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 (1996) 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 criteria, 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 criteria using error estimation methods such as the one given in (2) below.

2. Energy Error Norm (13 and 14) applied so that the calculated frame/edge thermal transmittance is within 1% of the thermal transmittance determined with an ideal mesh.

3. 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 the i^{th} cavity:

$$\lambda_{\mathrm{eff,i}} = q_{\mathrm{i}} \left[\frac{d_{\mathrm{g,i}}}{T_{\mathrm{f,i}} - T_{\mathrm{b,i-1}}} \right] \quad (72)$$

6.6 Effective Conductivity – Unventilated Frame Cavities

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.

$$\lambda_{\mathrm{eff}} = (h_{\mathrm{c}} + h_{\mathrm{r}}) \cdot d \quad (73)$$

where

λ_{eff} is the effective conductivity;

h_c is the convective heat transfer coefficient;

h_r is the radiative heat transfer coefficient (h_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_c, is calculated from the Nusselt number, *Nu*, which can be determined from various correlations, depending on aspect ratio, orientation and direction of heat flow.

$$h_{\mathrm{c}} = Nu \frac{\lambda_{\mathrm{air}}}{d} \quad (74)$$

[DC: Hakim, note change in equation 74]

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

6.6.1 Heat flow downward

$$Nu = 1{,}0 \tag{75}$$

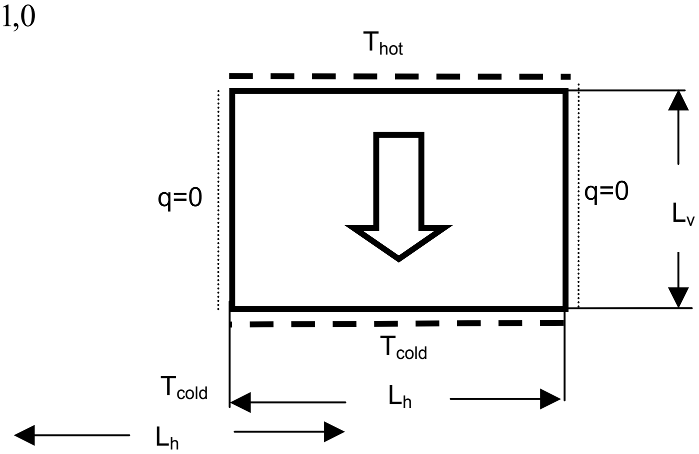


Figure f1 – Illustration of Rectangular Frame Cavity Downward Flow Direction

6.6.2 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.

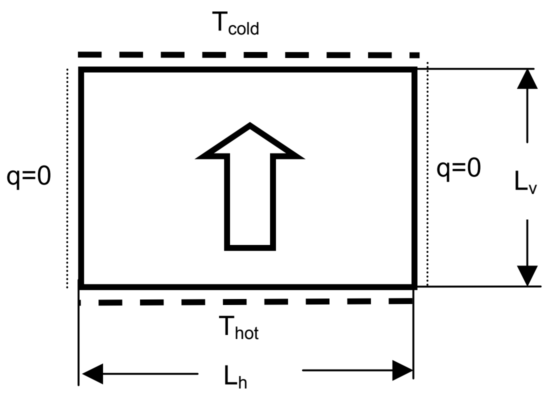


Figure f2 – Illustration of Rectangular Frame Cavity Upward Flow Direction

a) for $\frac{L_\mathrm{v}}{L_\mathrm{h}} \leq 1$ convection is restricted by wall friction, and

$$Nu = 1{,}0 \tag{76}$$

b) for $1 < \frac{L_\mathrm{v}}{L_\mathrm{h}} \leq 5$ the Nusselt number is calculated according to the method given by (15).

$$Nu = 1 + \left[1 - \frac{Ra_{\text{crit}}}{Ra}\right]^{+} \left[k1 + 2(k2)^{1-\ln k2}\right] + \left[\left(\frac{Ra}{5380}\right)^{1/3} - 1\right]^{+} \left[1 - e^{-0.95\left(\left(\frac{Ra_{\text{crit}}}{Ra}\right)^{1/3} - 1\right)^{+}}\right] \qquad (77)$$

where

$$k1 = 1.40 \quad k2 = \frac{Ra^{1/3}}{450,5} \quad [x]^{+} = \frac{x + |x|}{2}$$

and Ra_{crit} is a critical Rayleigh number, which is found by least squares regression of tabulated values (15)

$$Ra_{\text{crit}} = e^{\left(0.721\frac{L_h}{L_v}\right) + 7,46}$$

Ra is the Rayleigh number for the air cavity:

$$Ra = \frac{\rho_{\text{air}}^{2} L_{v}^{3} g \beta C_{p,air} (T_{\text{hot}} - T_{\text{cold}})}{\mu_{\text{air}} \lambda_{\text{air}}}$$

c) for $\frac{L_v}{L_h} > 5$ the Nusselt number is:(7):

$$Nu = 1 + 1,44\left[1 - \frac{1708}{Ra}\right]^{+} + \left[\left(\frac{Ra}{5830}\right)^{1/3} - 1\right]^{+} \qquad (78)$$

6.6.3 Horizontal heat flow

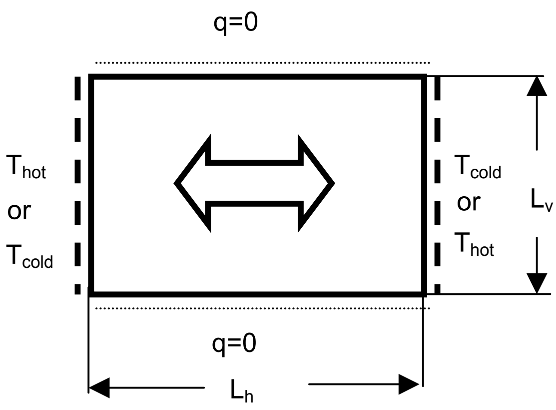


Figure f3 – Illustration of Rectangular Frame Cavity Horizontal Flow Direction

a) for $\frac{L_v}{L_h} < \frac{1}{2}$ the Nusselt number is: (15)

$$Nu = 1 + \left\{ \left(2{,}756 \cdot 10^{-6} Ra^{2} \left(\frac{L_{v}}{L_{h}} \right)^{8} \right)^{-0{,}386} + \left(0{,}623 Ra^{1/5} \left(\frac{L_{h}}{L_{v}} \right)^{2/5} \right)^{-0.386} \right\}^{-2.59} \tag{79}$$

where Ra is Raleigh number and is defined as:

$$Ra = \frac{\rho_{air}^{2} L_{h}^{3} g \beta C_{p,air} (T_{hot} - T_{cold})}{\mu_{air} \lambda_{air}}$$

b) for $\frac{L_v}{L_h} > 5$ the following correlation, also from (15), gives Nu as the maximum of:

$$Nu_{ct} = \left\{ 1 + \left[\frac{\left(0{,}104 Ra^{0{,}293}\right)}{\left(1 + \left(\frac{6310}{Ra}\right)^{1{,}36}\right)} \right]^{3} \right\}^{1/3} \tag{80}$$

$$Nu_{l} = 0{,}242 \left(Ra \frac{L_{h}}{L_{v}} \right)^{0{,}273} \tag{81}$$

$$Nu_{t} = 0{,}0605 Ra^{1/3} \tag{82}$$

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 (80) to (82) in clause 6.6.3.b shall be used.

The temperatures T_{hot} and T_{cold} are not known in advance, so it is necessary to estimate them. From previous experience it is recommended to apply T_{hot}=10°C and T_{cold}=0°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_{hot}-T_{cold} 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.

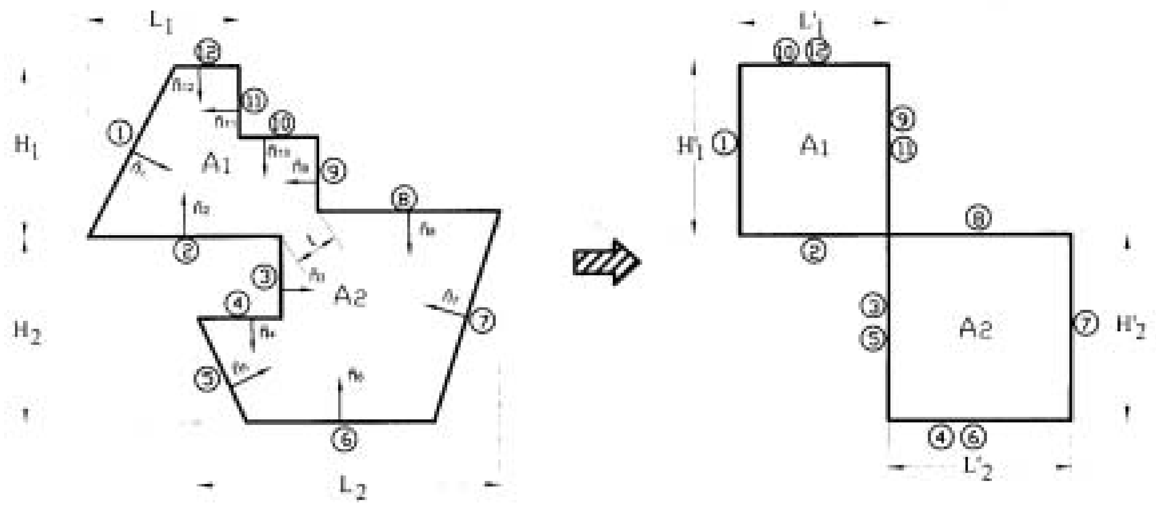
For unventilated irregularly shaped frame cavity, the geometry shall be converted into equivalent rectangular cavity according to the procedure in ISO/DIS 10077-2 (see also figure f4). For these cavities, the following procedure shall be used to determine which surfaces belong to vertical and horizontal surfaces of equivalent rectangular cavity (see also figure f5). If the shortest distance between two opposite surfaces is smaller than 5 mm then the frame cavity shall be split at this "throat" region.

a) any surface whose normal is between 270° and 45° a is left vertical surface

b) any surface whose normal is between 45° and 135° a is bottom horizontal surface

c) any surface whose normal is between 135° and 225° a is right vertical surface

d) any surface whose normal is between 225° and 315° a is top horizontal surface



$$\frac{L_1}{H_1} = \frac{L_1'}{H_1'} \qquad t \leq 5mm$$

$$\frac{L_2}{H_2} = \frac{L_2'}{H_2'}$$

Figure f4. Illustration of the Treatment of Irregularly shaped frame cavities

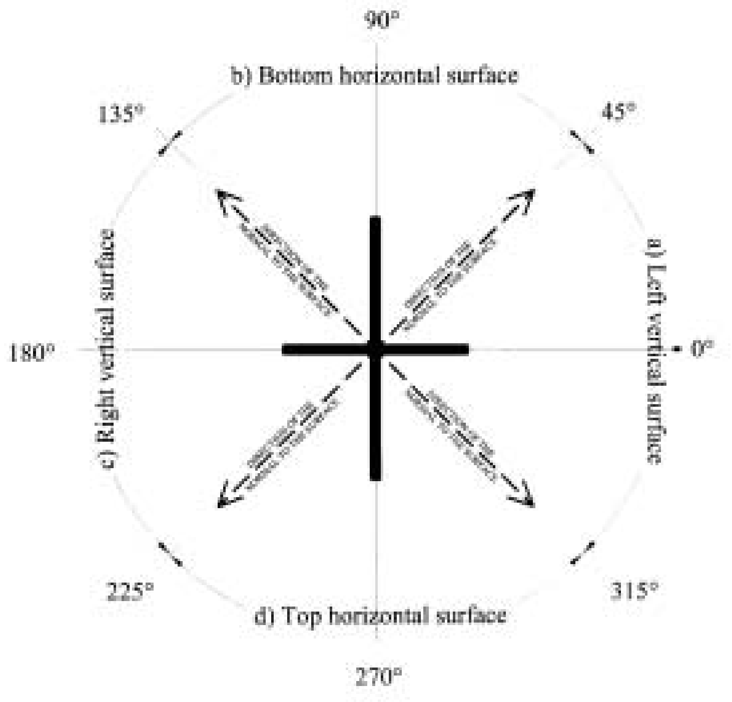


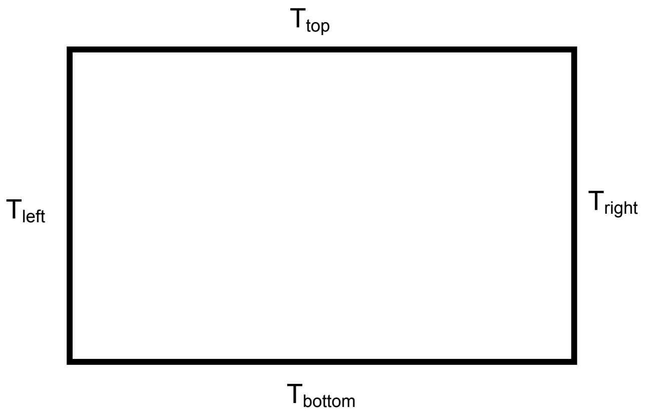
Figure f5. Illustration of how to select surface orientation for frame cavities-dashed lines indicate direction of the normal to surface with cut of angles at 45°, 135°, 225° and 315°.

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 f6):

a) heat flow is horizontal if the absolute value of the temperature difference between vertical cavity surfaces is larger than between horizontal the cavity surfaces,

b) heat flow is vertical heat flow up if absolute temperature difference between horizontal cavity surfaces is larger than between vertical cavity surfaces and temperature difference between the top horizontal cavity surface and bottom horizontal cavity surface is negative,

c) heat flow is vertical, heat flow down if absolute temperature difference between horizontal cavity surfaces is larger than between vertical cavity surfaces and temperature difference between the top horizontal cavity surface and bottom horizontal cavity surface is positive.



a) $\left|T_{right} - T_{left}\right| \geq \left|T_{top} - T_{bottom}\right|$ heat flow is horizontal;

b) $\left|T_{right} - T_{left}\right| < \left|T_{top} - T_{bottom}\right|$ and $T_{top} < T_{bottom}$ heat flow is vertical, heat flow up;

c) $\left|T_{right} - T_{left}\right| < \left|T_{top} - T_{bottom}\right|$ and $T_{top} > T_{bottom}$ heat flow is vertical, heat flow down.

Figure f6. Illustration of how to select heat flow direction

6.6.4 Radiant heat flow

The radiative heat transfer coefficient h_r shall be calculated using (16):

$$h_r = \frac{4\,\sigma\,T_{ave}^3}{\dfrac{1}{\varepsilon_{cold}} + \dfrac{1}{\varepsilon_{hot}} - 2 + \dfrac{1}{\dfrac{1}{2}\left(\left[(1+(\dfrac{L_h}{L_v})^2\right]^{\frac{1}{2}} - \dfrac{L_h}{L_v} + 1\right)}} \qquad \frac{W}{m^2 K} \tag{83}$$

where

$$T_{ave} = \frac{T_{cold} + T_{hot}}{2}$$

NOTE Above notation assumes radiant heat flow 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)

6.7 Ventilated air cavities and grooves

6.7.1 Slightly ventilated cavities and grooves with small cross section

Grooves with small cross sections (see Figure f6) at the external or internal surfaces of profiles and cavities connected to the exterior or interior 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 according to 6.6.

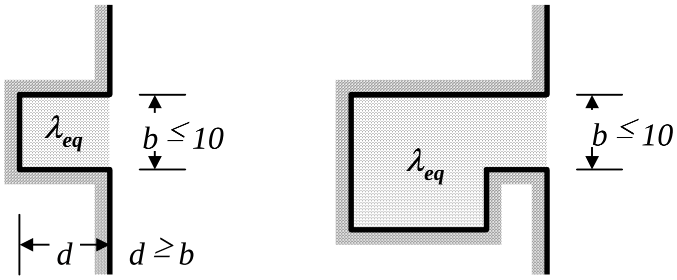


Figure f6: Examples for slightly ventilated cavities and grooves with small cross section

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_i and h_o, calculated according to section 8, are to be used at the developed indoor and outdoor surfaces, respectively.

In the case of a large cavity connected by a single slit and a developed surface exceeding the width of the slit by a factor of 10 the detailed radiation model shall be used for radiative portion of a surface heat transfer coefficient.

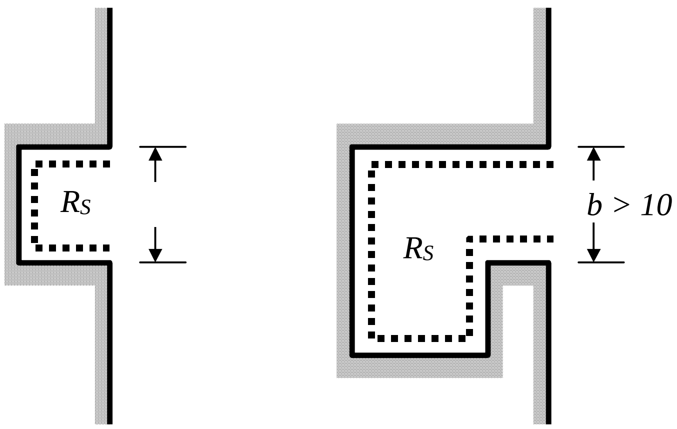


Figure 5: Examples for well ventilated cavities and grooves

7 Shading devices

7.1 Definitions

7.1.1 Introduction

This 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 this clause 7.

Information on calculation procedures and measurement techniques on shading devices can be found in references such as (17,18,19,20,21,22,23,24,and 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.

NOTE 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, with limited thermal-optical interaction.

Although there is no sharp cut distinction between the two types of shadings, the extra-fenestrial types of shadings 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).

This standard does not deal with the extra-fenestrial type of shading.

7.1.2 Principle of the calculations

The thermal-optical interaction of a layer type of shading devices 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.

But 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.

But 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 have to be translated into one-dimensional numbers. This is in particular the case for the optical properties. For instance, 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 absorptance of the layer.

7.2 Optical properties; 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:

- a 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.

NOTE 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:

For transmittance, for beam radiation, for each angle of incidence:

$\tau_{dir,dir}(\lambda_j)$ direct to direct transmittance

$\tau_{dir,dif}(\lambda_j)$ direct to diffuse transmittance

for diffuse radiation:

$\tau_{dif,dif}(\lambda_j)$ diffuse to diffuse transmittance

Similarly for the reflectance:

for beam radiation, for each angle of incidence:

$\rho_{dir,dir}(\lambda_j)$ direct to direct reflectance

$\rho_{dir,dif}(\lambda_j)$ direct to diffuse reflectance

for diffuse radiation:

$\rho_{dif,dif}(\lambda_j)$ diffuse to diffuse reflectance

and finally, for the absorptance:

$$\alpha_{dir}(\lambda_j) = (1 - \tau_{dir,dir}(\lambda_j) - \rho_{dir,dir}(\lambda_j) - \tau_{dir,dif}(\lambda_j) - \rho_{dir,dif}(\lambda_j))$$

$$\alpha_{dif}(\lambda_j) = (1 - \tau_{dif,dif}(\lambda_j) - \rho_{dif,dif}(\lambda_j))$$

Resulting amendments to the equations in clause 5:

For a window system incorporating layer type of shading devices the optical equations given in clause 5 remain the same, with the following extension:

1) each spectral flux equation in clause 5.2 shall be split into three: a 'dir,dir', 'dir,dif' and 'dif,dif' flux, with corresponding transmittance τ and reflectance ρ.

In the sum of the spectral fluxes the 'dir,dir', 'dir,dif' and 'dif,dif' parts shall be summed.

2) The transmittance shall be split, similar to the reflectance, into a forward and a backward value.

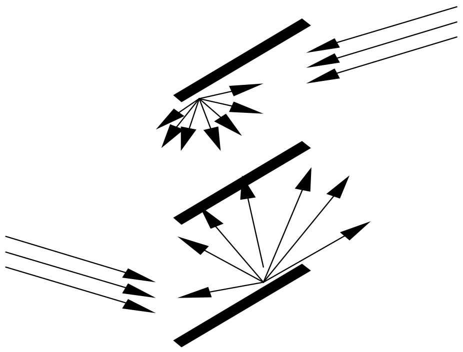
3) The sum of $\tau_{dir,dir}$ and $\tau_{dir,dif}$ is equal to the direct to hemispherical transmittance $\tau_{dir,h}$; similarly for the reflectance.

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

NOTE 1 There is no existing standard for the *measurement* of these optical properties. Until such 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 next clause provides a *calculation* method.

NOTE 2 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 $\tau_{dif,dif}$ and $\rho_{dif,dif}$ values are required: consequently, in that case the values for normal incidence provide insufficient information.

NOTE 3 Due to the redirection of the radiation the forward transmittance is not necessarily equal to the backward transmittance, as illustrated in Figure 7.



$\tau_{f,dif,dif} > \tau_{b,dif,dif}$

Figure 7 - 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 8).

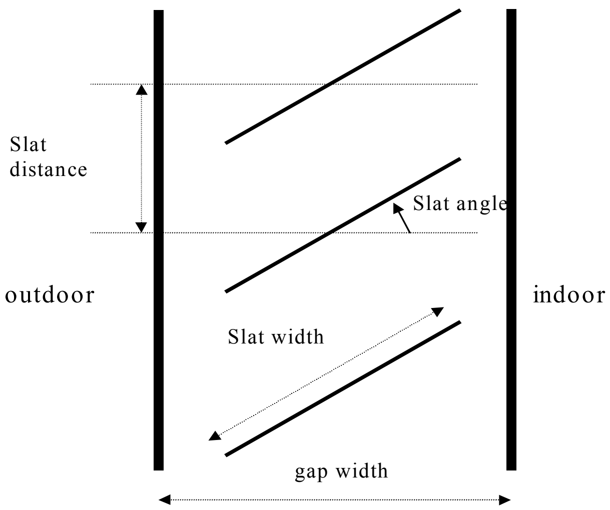


Figure 8 - Slat geometry

The air permeability can also be determined as function of slat geometry and position.

7.3.2 Optical properties

This clause gives the procedure to calculate the solar optical properties of a slat type of shading device provided:

- that the slats are non-specular reflecting;
- effects of the window edges may be neglected (two-dimensional situation, with each slat the same irradiation);

The procedure is to consider two adjacent slats and to subdivide the slats into five equal parts (see figure 9):

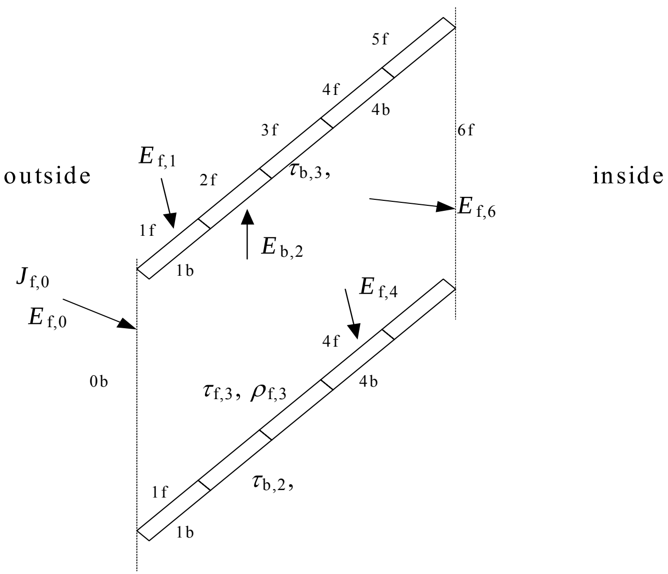


Figure 9 - Discretization used in the model

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 wave length 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 neglected.

Equations:

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

For each layer f,i and b,i, with i from 0 to n (here: n=6) and for each spectral interval λ_j, ($\lambda \rightarrow \lambda+\Delta\lambda$):

$$E_{f,i} = \sum_k \left\{ \left(P_{f,k} + \tau_{b,k} \right) E_{f,k} F_{f,k \longrightarrow f,i} + \left(\rho_{b,k} + \tau_{f,k} \right) E_{b,k} F_{b,k \longrightarrow f,i} \right\} \quad (84)$$

$$E_{b,i} = \sum_k \left\{ \left(P_{b,k} + \tau_{f,k} \right) E_{b,k} F_{b,k \longrightarrow b,i} + \left(\rho_{f,k} + \tau_{b,k} \right) E_{f,k} F_{f,k \longrightarrow b,i} \right\} \quad (85)$$

Where

E_k is the irradiation on surface *k*

$F_{p \rightarrow q}$ is the view or shape factor from surface p to surface q.

where

$E_{f,0} = J_0(\lambda_j)$

$E_{b,n} = J_n(\lambda_j) = 0$

where

J_0 is the radiosity from the outdoor environment (incident solar radiation)

J_n is the radiosity from the internal environment (room reflection)

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 for instance (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_0 :

$$\tau_{dif,dif}(\lambda_j) = E_{f,n}(\lambda_j) / J_o(\lambda_j) \quad (86)$$

Similarly, for the diffuse/diffuse reflection coefficient.

$$\rho_{dif,dif}(\lambda_j) = E_{b,o}(\lambda_j) / J_o(\lambda_j) \quad (87)$$

Direct-direct transmission and reflection:

By straightforward geometric calculation from the angle and aspect ratio of the slats (see figure 10) the beam radiation which passes the slats without touching can be calculated for given angle of incidence ϕ .

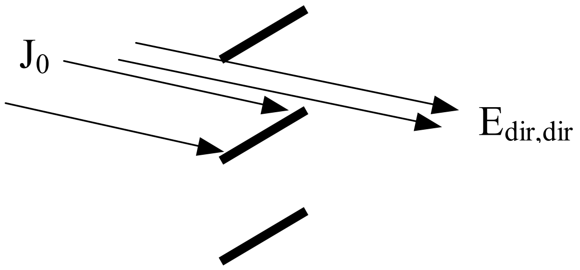


Figure 10 - Direct-direct transmission

This part of the transmission is wavelength independent.

This is the direct/direct transmission: $E_{dir,dir}$.(ϕ)

Consequently, the direct-direct transmittance for incidence angle ϕ amounts:

$$\tau_{\mathrm{dir,dir}}(\phi) = E_{\mathrm{dir,dir}}(\lambda_{\mathrm{j}}, \phi) / J_{\mathrm{o}}(\lambda_{\mathrm{j}}, \phi) \tag{88}$$

for any wavelength λ_j.

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

$$\rho_{dir,dir}(\phi) = 0 \tag{89}$$

Direct-diffuse transmission and reflection:

Firstly: calculate, for the given angle of incidence ϕ, which parts of the shading device *k* are directly irradiated by $J_{f,0}$ (see figure 11).

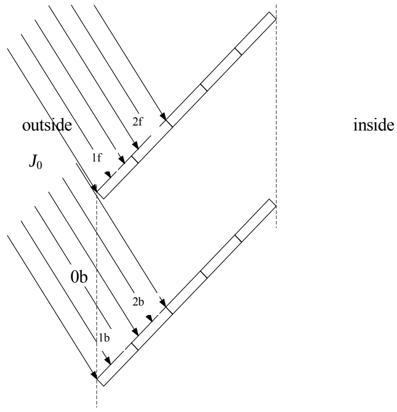


Figure 11 - Directly irradiated parts of the shading device

The view factors between the incident radiation J_0 and those directly irradiated parts *k* are:

$F_{f,0 \to f,k}$ = 1

$F_{f,0 \to b,k}$ = 1

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

$F_{f,0 \to b,n}$ = 0 and $F_{b,0 \to f,n}$ = 0

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

$$\tau_{\text{dir,dif}}(\lambda_j, \phi) = E_{f,7}(\lambda_j, \phi) / J_o(\lambda_j, \phi) \quad (90)$$

$$\rho_{\text{dir,dif}}(\lambda_j, \phi) = E_{b,7}(\lambda_j, \phi) / J_o(\lambda_j, \phi) \quad (91)$$

Absorptance:

That part which is not transmitted, nor reflected, is the part which is absorbed in the slats. Per wavelength band:

$$\alpha_{\text{dir}}(\lambda_j) = \left(1 - \tau_{\text{dir,dir}}(\lambda_j) - \rho_{\text{dir,dir}}(\lambda_j) - \tau_{\text{dir,dif}}(\lambda_j) - \rho_{\text{dir,dif}}(\lambda_j)\right) \quad (92)$$

$$\alpha_{\text{dif}}(\lambda_j) = \left(1 - \tau_{\text{dif,dif}}(\lambda_j) - \rho_{\text{dif,dif}}(\lambda_j)\right)$$

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.

NOTE The *normal* emissivity of the surfaces may be measured according to prEN 12898. There is no existing standard for the measurement of the *hemispherical* emissivity of opaque materials. Usually an emissometer is used for this purpose.

7.4 Ventilation

7.4.1 General

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

7.4.2 Main heat balance equations

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 clause 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 5. 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 standard for measurement of these properties. Until this testing standard is available, the calculation method of this clause shall be considered as provisional and is provided for information purposes only.

Equations:

Non-vented gap:

For the non-vented case (clause 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: $q_{c,i} = h_{c,i}(T_{f\,i} - T_{b\,i+1})$, is split into two parts (see figure 12), with the mean temperature of the air in the gap as variable:

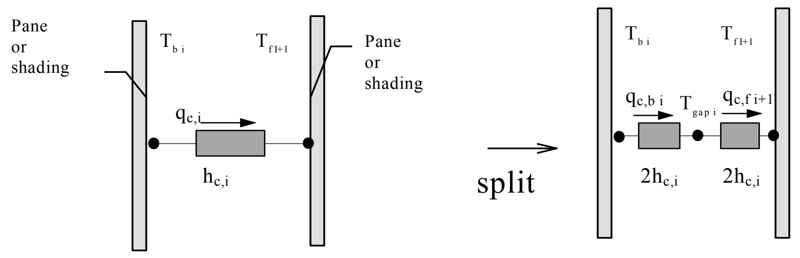


Figure 12 - Split convective heat transfer across gap; non-vented gap

$$q_{c,f\_i} = 2h_{c,i}(T_{f\_i} - T_{gap\_i}) = q_{c,b\_i+1} = 2h_{c,i}(T_{gap\_i} - T_{b\_i+1}) \quad (93)$$

where

q_{c,f_i} is the convective heat transfer from the one surface to the gap, in (W/m^2);

q_{c,b_i+1} is the convective heat transfer from the gap to the other surface, in (W/m^2);

$h_{c,i}$ is the surface-to-surface heat transfer coefficient by conduction/convection for non-vented cavities, given by the equations in clause 5.3, in (W/(m^{2}K))

T_{gap_i} is the equivalent mean temperature of the air in the cavity *i*, given in the next clause, in (°C)

T_{f_i} is the temperature of the surface of layer (pane, film or shading) *i*, facing the cavity *i*, see clause 5.3, in (°C);

T_{b_i+1} is the temperature of the surface of layer (pane, film or shading) *i+1*, facing the cavity *i*, see clause 5.3, in (°C).

Ventilated gap:

Amendments to equations in clause 5.3:

In a ventilated gap, due to the air movement, the convective heat exchange coefficient is increased (see figure 13).

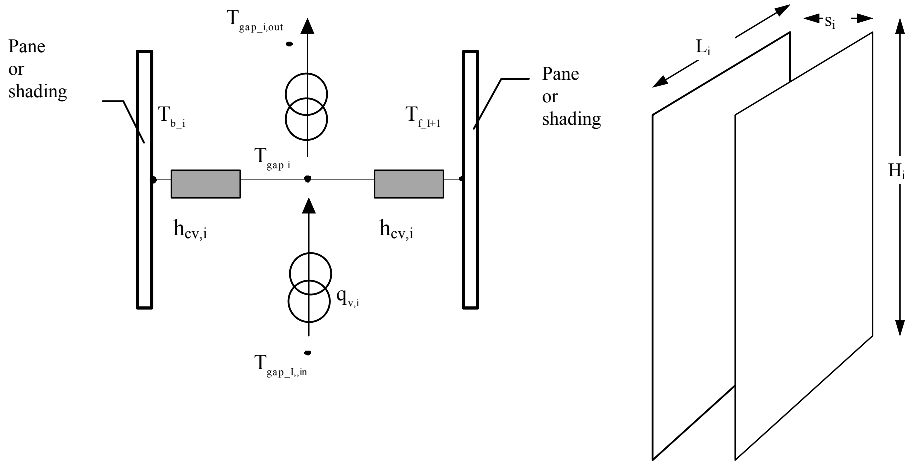


Figure 13 - Model of mean air- and outlet temperature and main dimensions

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

$$q_{c,b\_i} = h_{cv,i}(T_{b\_i} - T_{gap\_i}) \quad q_{c,f\_i+1} = h_{cv,i}(T_{gap\_i} - T_{f\_i+1}) \tag{94}$$

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

$$h_{cv,i} = 2h_{c,i} + 4v_i \tag{95}$$

where

q_{c,b_i} is the convective heat transfer from the one surface to the gap, in (W/m^2);

q_{c,f_i+1} is the convective heat transfer from the gap to the other surface, in (W/m^2);

$h_{cv,i}$ is the surface-to-air heat transfer coefficient by conduction/convection for vented cavities, given by eq. 7.12, in (W/(m^{2}K));

$h_{c,i}$ is the surface-to-surface heat transfer coefficient by conduction/convection for non-vented cavities, given by the equations in clause 5.3, in (W/(m^{2}K));

v_i is the mean air velocity in the gap, see clause 7.4.4, in (m/s);

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

T_{gap_i} is the equivalent mean temperature of the air in the cavity *i*, given by eq. (90) in the next clause, in (°C)

T_{b_i} is the temperature of the surface of layer (pane, film or shading) *i*, facing the cavity *i*, see clause 5.3, in (°C);

T_{f_i+1} is the temperature of the surface of layer (pane, film or shading) *i+1*, facing the cavity *i*, see clause 5.3, in (°C).

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

Amendments to equations in clause 5.3:

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

$$q_{\mathrm{v,i}} = \rho_{\mathrm{i}} \cdot c_{\mathrm{p}} \cdot \varphi_{\mathrm{v,i}} (T_{\mathrm{gap\_i,in}} - T_{\mathrm{gap\_i,out}}) / (H_{\mathrm{i}} \times L_{\mathrm{i}}) \qquad (96)$$

NB: with the equations in the next clause it can be shown that this equation is equal to: $q_{\mathrm{v,i}} = q_{\mathrm{cv,b\_i}} + q_{\mathrm{cv,f\_i+1}}$

NOTE The heat transfer is normalised to 1 m^2 of (the transparent part of) the window area.

where

$q_{v,i}$ is the heat transfer to the gap by ventilation, in (W/m^2);

ρ_i is the density of the air in cavity *k* at temperature T_{gap_i} , in (kg/m^3);

c_p is the specific thermal capacity of air, in (J/(kg⋅K) (i.c.: 1008);

$\varphi_{v,i}$ is the air flow rate in cavity *i*, see clause 7.4.4, in (m^3/s);

$T_{gap\_i,in}$ is the temperature at the inlet of the gap, in (°C);

The value of $T_{gap\_i,in}$ depends on where the air comes from: either the indoor or outdoor air temperature or the outlet temperature $T_{gap\_k,out}$ of the gap *k* with which the gap *i* exchanges air;

$T_{gap\_i,out}$ is the temperature at the outlet of the gap, see eq. 7.18 in next clause, in (°C);

L_i is the length of the cavity *i*, see Figure 13, in (m);

H_i is the height of the cavity *i*, see Figure 13, in (m).

Heat transfer to indoor environment

Extension of equations of clause 5:

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

Following the convention from clause 5, with *i* = *n* is the indoor environment:

for all cavities *k* with air flow to the indoor environment n:

$$q_{\mathrm{v,n}} = \sum_{\mathrm{i}} \rho_{\mathrm{i}} \cdot c_{\mathrm{p}} \cdot \varphi_{\mathrm{v,i}} (T_{\mathrm{gap\_i,out}} - T_{\mathrm{airn}}) / (H_{\mathrm{i}} \cdot L_{\mathrm{i}}) \qquad (97)$$

NOTE The heat transfer is normalised to 1 m^2 of (the transparent part of) the window area.

where

ρ_i is the density of the air in cavity *i*at temperature T_{gap_i} , in (kg/m^3);

c_p is the specific thermal capacity of air, in (J/(kg⋅K) (i.c.: 1008);

$\varphi_{v,i}$ is the air flow rate in cavity *i* , see clause 7.4.4, in (m^3/s);

$T_{gap\_i,out}$ is the temperature of the air at the outlet of the gap from where the air originates, see eq. 7.18 in next clause, in (°C);

$T_{air,n}$ is the temperature of the air at the indoor environment, in (°C);

L_i is the length of the cavity i, see Figure 13, in (m);

H_i is the height of the cavity i, see Figure 13, in (m).

7.4.3 Temperatures in the cavity

Assuming the mean velocity of the air in the space is known (see next clause), the temperature profile and the heat flow may 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 14).

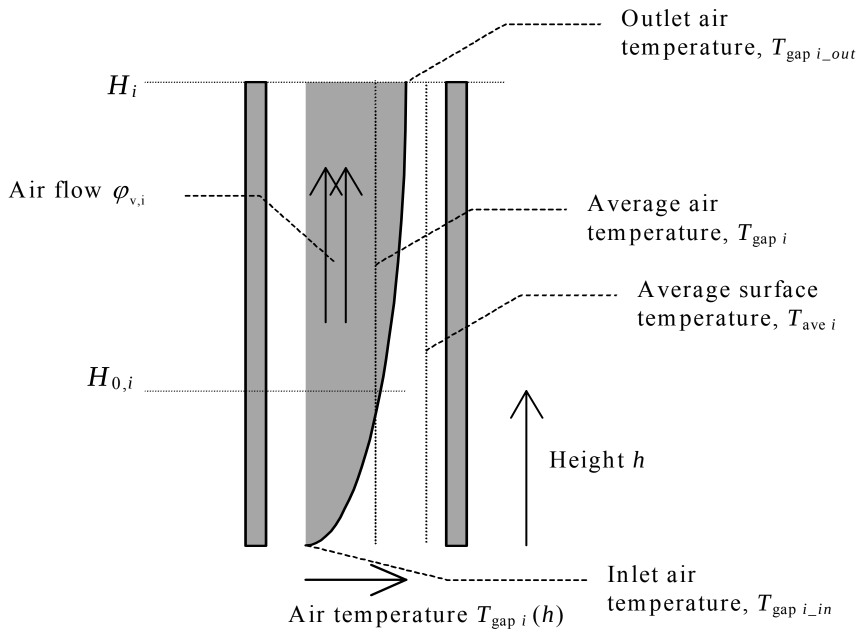


Figure 14 - Air flow in the gap of a window system

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

$$T_{gap\_i}(h) = T_{av,i} - (T_{av,i} - T_{gap\_i,in}) \cdot e^{-h/H_{0,i}} \quad (98)$$

where

$T_{gap\_i}(h)$ is the temperature of the air in gap *i* at position *h*, in (m);

$H_{0,i}$ is the characteristic height (temperature penetration length), see eq. (97), in (m);

$T_{gap\_i,in}$ is the temperature of the incoming air in gap *i*, in (°C);

T_{av_i} is the average temperature of the surfaces of layers *i* and *i+1*, given by equation:

$$T_{av\_i} = (T_{b\_i} + T_{f\_i+1})/2 \tag{99}$$

where

T_{av_i} is the average temperature of the surfaces of layers *i* and *i+1*, in (C);

T_{b_i} is the temperature of the surface of layer (pane, film or shading) *i*, facing the cavity *i*, see clause 5.3, in (C);

T_{f_i+1} is the temperature of the surface of layer (pane, film or shading) *i+1*, facing the cavity *i*, see clause 5.3, in (C).

The characteristic height of the temperature profile is defined by:

$$H_{0,i} = \frac{\rho_i \cdot c_p \cdot s_i}{2 \cdot h_{cv,i}} \cdot V_i \tag{100}$$

where

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

ρ_i is the density of the air at temperature T_{gap_j} , in (kg/m^3)

c_p is the specific heat capacity, in (J/(kgK)) (i.c.: 1008)

s_i is the width of the cavity *i*, in (m);

v_i is the mean velocity of the air flow in the cavity *i*, see clause 7.4.4, in (m/s);

$h_{cv,j}$ is the heat transfer coefficient for ventilated cavities, see eq. (92) in clause 7.4.2, in (W/(m^{2}K)).

The leaving air temperature is given by:

$$T_{gap\_i,out} = T_{av\_i} - \left(T_{av\_i} - T_{gap\_i,in}\right) \cdot e^{-H_i / H_{0,i}} \tag{101}$$

where

$T_{gap\_i,out}$ is the temperature of the air at the outlet of the gap *i*, in (C)

T_{av_i} is the average temperature of the surfaces of layers *i* and *i-1*, given by eq. (96), in (C);

$T_{gap\_i,in}$ is the temperature of the incoming air in the cavity *i*;

$H_{0,i}$ is the characteristic height (temperature penetration length), given by eq. (97), in (m);

H_i is the height of the space *i*, in (m).

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

$$T_{\text{gap\_i}} = \frac{1}{H_\text{i}} \int_0^H T_{\text{gap\_i}}(h) \cdot dh = T_{\text{av\_i}} - \frac{H_{0,\text{i}}}{H_\text{i}} \left(T_{\text{gap\_i,out}} - T_{\text{gap\_i,in}} \right) \tag{102}$$

where

T_{gap_i} is the equivalent mean temperature of the air in the cavity *i*, in (°C);

$T_{gap\_i,out}$ is the temperature of the air at the outlet of the gap *i*, in (°C);

$T_{gap\_i,in}$ is the temperature of the incoming air in gap *i*, in (°C);

T_{av_i} is the average temperature of the surfaces of layers *i* and *i+1*, given by eq. (96), in (°C);

$H_{0,i}$ is the characteristic height (temperature penetration length), given by eq. (97), in (m);

H_i is the height of the space *i*, in (m).

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:

$$v_\text{i} = \frac{\varphi_{\text{v,i}}}{s_\text{i} \cdot L_\text{i}} \tag{103}$$

where

v_i is the mean velocity of the air flow in the cavity *i*, in (m/s);

s_i the width of the cavity *i*, in (m);

L_i is the length of the cavity *i*, see Figure 13, in (m);

$\varphi_{v,i}$ air flow rate in cavity *i*, in (m^3/s);

NOTE $\varphi_{v,i}$ is the air flow rate for the whole area, *not* normalised per m^2.

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 (CFD modelling).

For conservative design calculations one may treat the cavity flow as forced convection (see clause 7.4.4.1), with the value for the air velocity v_i set to extreme low respectively extreme high values respectively, 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 15).

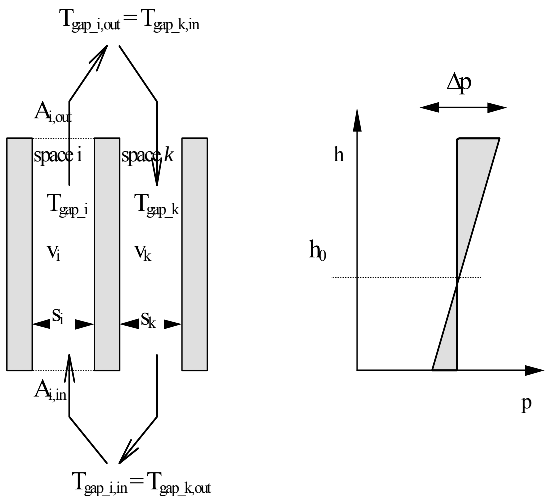


Figure 15 - Schematic presentation of the stack effect. The height of the neutral zone h_0 depends on the flow resistances of the inlet and outlet openings

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

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 (eq. 93). The driving pressure difference Δp_T may be written approximately as:

$$\Delta p_{T,i,k} = \rho_0 \cdot T_0 \cdot g \cdot H_i \cdot \cos\varphi_i \cdot \frac{(T_{gap\_i} - T_{gap\_k})}{T_{gap\_i} \cdot T_{gap\_k}} \qquad (104)$$

where

$\Delta p_{T,i,k}$ is the driving pressure difference between space *i* and space *k*, in Pa;

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

T_{gap_i} is the equivalent (mean) temperature of the air in the space *i*, see eq (99), K;

T_{gap_k} is the equivalent temperature of the connected space, which may be another gap *k* or the indoor or outdoor environment, K**;**

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

ρ_0 is the density of the air at temperature T_0, in (kg/m^3);

g is the gravity constant = 9.81 (m/s^2);

T_0 is reference temperature, (e.g.) T_0 = 283 K.

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 law):

$$\Delta p_{B,i} = \frac{\rho_i}{2} v_i^{\ 2} \tag{105}$$

Steady laminar flow (Hagen-Poiseuille law):

$$\Delta p_{HP,i} = 8 \cdot \mu_i \cdot \frac{H_i}{S_i^{\ 2}} V_i \tag{106}$$

Pressure loss in the inlet and outlet openings:

$$\Delta p_{Z,i} = \frac{P_i}{2}\left(Z_{in,i} + Z_{out,i}\right) \tag{107}$$

where

$\Delta p_{B,i}$ is the pressure loss B in space *i*, in Pa;

$\Delta p_{HP,i}$ is the pressure loss HP in space *i*, in Pa;

v_i is the mean velocity of the air flow in the cavity *i*, to be solved with eq. (108), in (m/s); (same for *k*);

μ_i is the viscosity of the air at temperature T_{gap_j} , in (Pa⋅s);

ρ_i is the density of the air at temperature T_{gap_j} , in (kg/m^3);

H_i is the height of the space *i,* in (m);

s_i the width of the cavity *i*, in (m);

Z_i the pressure loss factors *Z* of cavity *i*, according to eq.(109)below;

The same equations apply to space *k*, where $v_k = v_i \cdot s_i / s_k$.

But 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.

where

$\Delta p_{Z,i,k}$ is the pressure loss Z between space *i* and *k*, in Pa.

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:

$$\Delta p_{T,i,k} = \Delta p_{B,i} + \Delta p_{HP,i} + \Delta p_{Z,i,k} + \Delta p_{B,k} + \Delta p_{HP,k} \tag{108}$$

where

$\Delta p_{T,i,k}$ is the driving pressure difference between space *i* and space *k*, according to eq. 7.21, in Pa;

$\Delta p_{B,i}$ is the pressure loss B in space *i*, according to equation (102), in Pa;

$\Delta p_{HP,i}$ is the pressure loss HP in space *i*, according toequation (103), in Pa;

$\Delta p_{Z,i,k}$ is the pressure loss Z between space *i* and *k*, according toequation (104), in Pa;

$\Delta p_{B,k}$ is the pressure loss B in space *k*, according to equation (102), in Pa;

$\Delta p_{HP,k}$ is the pressure loss HP in space *k*, according to equation (103), in Pa.

Pressure loss factors

The pressure loss factors *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 16) according to:

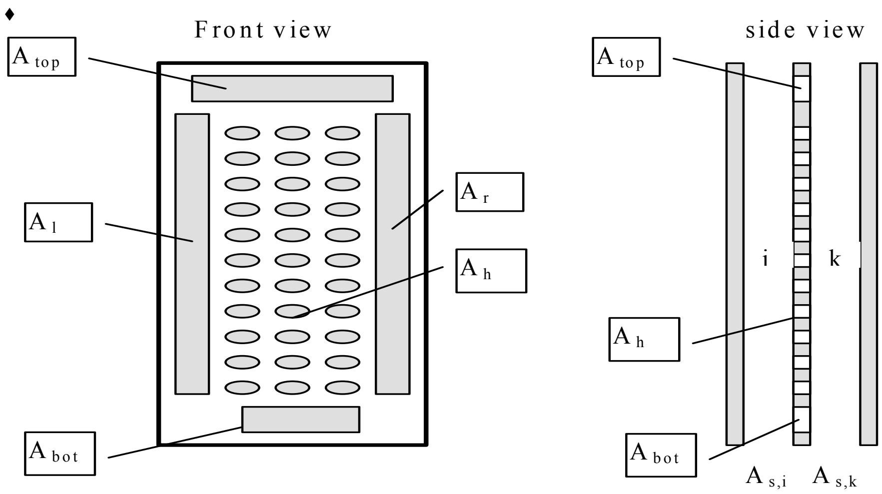


Figure 16 - Openings in a ventilated gap

$$Z_{in} = \left(\frac{A_s}{0{,}66 \cdot A_{eq,in}} - 1 \right)^2 \quad Z_{out} = \left(\frac{A_s}{0{,}6 \cdot A_{eq,out}} - 1 \right)^2 \qquad (109)$$

where

$A_{s,i}$ is the cross section of the space *i*; $A_{s,i} = s_i \times L_i$;

s_i is the width of the cavity *i*, in (m);

L_i is the length of the cavity *i*, in (m);

$A_{eq,in,i}$ is the equivalent inlet opening area of the cavity *i*, according to equation (110), in m^2;

$A_{eq,out,i}$ is the equivalent outlet opening area of the cavity *i*, according to equation (110), in m^2.

If the temperature T_{gap_i} (resp. T_{gap_k}) of the cavity *i* respectively *k* is higher than the temperature of the connected space *k* respectively *i* :

$$A_{\text{eq,in}} = A_{\text{bot}} + \frac{1}{2} \cdot \frac{A_{\text{top}}}{A_{\text{bot}} + A_{\text{top}}} (A_{\text{l}} + A_{\text{r}} + A_{\text{h}}) \qquad A_{\text{eq,out}} = A_{\text{top}} + \frac{1}{2} \cdot \frac{A_{\text{bot}}}{A_{\text{bot}} + A_{\text{top}}} (A_{\text{l}} + A_{\text{r}} + A_{\text{h}}) \tag{110}$$

Otherwise:

$$A_{\text{eq,out}} = A_{\text{bot}} + \frac{1}{2} \cdot \frac{A_{\text{top}}}{A_{\text{bot}} + A_{\text{top}}} (A_{\text{l}} + A_{\text{r}} + A_{\text{h}}) \qquad A_{\text{eq,in}} = A_{\text{top}} + \frac{1}{2} \cdot \frac{A_{\text{bot}}}{A_{\text{bot}} + A_{\text{top}}} (A_{\text{l}} + A_{\text{r}} + A_{\text{h}}) \tag{111}$$

where

A_s is the cross section of the space, in m^2;

A_{bot} is the area of the bottom opening, in m^2;

A_{top} is the area of the top opening, in m^2;

A_h is the total area of the holes in the surface (homogeneously distributed holes), in m^2;

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

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

(side openings homogeneous from top to bottom)

all for both *i* and *k*.

NOTE All these areas are total areas for (the transparent part of) the window, *not* normalised to values per m^2 area as the heat fluxes.

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 with an appropriate value for the equivalent air permeability of the surface, A_h. An appropriate value is to be determined on the basis of experiments or calculations (CFD modelling).

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

7.5 Total solar energy transmittance and thermal transmittance

Due to the non-linearity, for instance due to thermally induced air circulation, the total solar energy transmittance and the thermal transmittance change with local temperatures. The local temperatures change with incident solar radiation and absorption. Consequently, if incident solar radiation enters the equations, in order to calculate the total solar energy transmittance it must be recognised that the thermal transmittance also changes simultaneously. This makes it virtually impossible to separate the two effects.

By definition the thermal transmittance is assumed to be the thermal transmittance without solar radiation. As a consequence, the total solar energy transmittance is calculated as the difference in net heat loss between the situation with and without solar radiation.

In equation:

$$U = \frac{q_{\mathrm{in,no\_sun}}}{T_{\mathrm{i}} - T_{\mathrm{e}}} \qquad (112)$$

$$g = \frac{q_{\mathrm{in}} - q_{\mathrm{in,no\_sun}}}{I_{\mathrm{s}}} \qquad (113)$$

where

g is the total solar energy transmittance, dimensionless;

q_{in} is the total heat transfer through the transparent part of the window, resulting from the overall heat balance, in W/(m^{2}K);

$q_{in, no sun}$ is the total heat transfer through the transparent part of the window, without solar radiation, resulting from the overall heat balance, in W/(m^{2}K);

I_s is the intensity of incident solar radiation, in W/(m^2).

8 Boundary conditions

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 (i.e., the environment) to which the product is exposed.

The boundary conditions consist of:

a) Indoor and outdoor air temperatures, T_{in} and T_{out}, respectively.

b) Indoor and outdoor surface convective heat transfer coefficients, $h_{c,in}$ and $h_{c,out}$, respectively.

c) Solar spectral irradiance distribution, *E(λ)*, and a function describing the photopic response of the eye, *R(λ)*. Both *E(λ)* and *R(λ)* 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 interpolati*on.*

d) The longwave irradiance on the outdoor and indoor glazing surfaces, $G_{g,out}$ and $G_{g,in}$, respectively as well as the longwave irradiance at the outdoor and indoor frame surfaces, $G_{f,out}$ and $G_{f,in}$, respectively. It is assumed that outdoor longwave irradiance depends on the clearness of the sky factor, f_{clr}.

8.1 Reference boundary conditions

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

$E_s(\lambda)$ = ISO 9845 (hemispherical solar spectral irradiance tabulated at N_s values of λ)

$E_v(\lambda)$ = ISO 10526 (colorimetric illuminance tabulated at N_{sv} values of λ)

$R(\lambda)$ = ISO/CIE 10527, CIE Standard Colorimetric Observer (photopic response for the 2° observer tabulated at N_v values of λ)

8.1.1 Winter conditions

T_{in} = 20°C

T_{out} = 0°C

$h_{c,in}$ = 3,6 W/m^{2}K

$h_{c,out}$ = 20 W/m^{2}K

$T_{r,m}$=T_{out}

I_s=300 W/m^2

8.1.2 Summer conditions

T_{in} = 25°C

T_{out} = 30°C

$h_{c,in}$ = 2,5 W/m^{2}K

$h_{c,out}$ = 8 W/m^{2}K

$T_{r,m}$=T_{out}

I_s=500 W/m^2

8.2 Convective heat transfer

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 indoor and outdoor boundary surfaces is extremely difficult and can only be done by careful measurements and computer simulation. From these reasons, surface heat transfer coefficient correlations had been developed and are given in the following clauses.

8.2.1 Convective heat transfer coefficient - indoor side

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

$$q_{c,in} = h_{c,in}\left(T_{s,in} - T_{in}\right) \qquad (114)$$

where $T_{s,in}$ is used here to denote the temperature of any indoor fenestration surface (i.e., $T_{b,n}$ or the temperature of the indoor frame surface). The convective heat transfer coefficient, $h_{c,in}$ is determined from heat transfer correlations given in following clauses.

8.2.1.1 Heat transfer by natural convection

The natural convection heat transfer coefficient for the indoor side, $h_{c,in}$ is determined in terms of the Nusselt number, *Nu*.

$$h_{c,in} = Nu\left(\frac{\lambda}{H}\right) \qquad (115)$$

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 whole fenestration system, Ra_H.

$$Ra_{\mathrm{H}} = \frac{\rho^2 H^3 g C_{\mathrm{p}} \left| T_{\mathrm{b,n}} - T_{\mathrm{in}} \right|}{T_{\mathrm{m,f}} \mu \lambda} \quad \text{(dimensionless)} \tag{116}$$

where the various fluid properties are those of air evaluated at the mean film temperature:
$T_{\mathrm{m,f}} = T_{\mathrm{in}} + \frac{1}{4}\left(T_{\mathrm{b,n}} - T_{\mathrm{in}}\right)$

Correlations to quantify the indoor side convective heat transfer coefficient for still air (27)are presented in the following clauses.

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

Each of the following clauses pertains to one particular value, or range, of tilt angle, θ. Note that this categorization, as a function of θ, is based on the assumption that the indoor environment is warmer than the indoor glazing surface (i.e., $T_{in}>T_{b,n}$). If the reverse is true ($T_{\mathrm{in}}<T_{b,n}$) it is necessary to seek the appropriate correlation on the basis of the complement of the tilt angle, 180°- θ, instead of θ and to then substitute 180°-θ instead of θ when the calculation is carried out.

a) Windows inclined from 0° to 15° (0°≤θ<15°)

$$Nu_{\mathrm{in}} = 0{,}13 Ra_{\mathrm{H}}^{1/3} \tag{117}$$

b) Windows inclined from 15° to 90° (15°≤θ≤90°)

$$Nu_{\mathrm{in}} = 0{,}56\left(Ra_{\mathrm{H}} \sin\theta\right)^{1/4} \quad Ra_H \le Ra_c \tag{118}$$

$$Nu_{\mathrm{in}} = 0{,}13\left(Ra_{\mathrm{H}}^{1/3} - Ra_{\mathrm{c}}^{1/3}\right) + \left(0{,}56 Ra_{\mathrm{c}} \sin\theta\right)^{1/4} \quad Ra_H \ge Ra_c \tag{119}$$

$$Ra_{\mathrm{c}} = 2{,}5 \times 10^5 \left(\frac{e^{0{,}72\theta}}{\sin\theta}\right)^{1/5} \quad \theta \text{ in degrees} \tag{120}$$

c) Windows inclined from 90° to 179° (90°<θ≤179°)

$$Nu_{\mathrm{in}} = 0{,}56(Ra_{\mathrm{H}} \sin\theta)^{1/4} \quad 10^5 \le Ra_H \sin\theta \le 10^{11} \tag{121}$$

d) Windows inclined from 179° to 180° (179°<θ≤180°)

$$Nu_1 = 0{,}58 Ra_{\mathrm{H}}^{1/5} \quad Ra_H \le 10^{11} \tag{122}$$

[DC: Hakim, note revisions of equations 119, 120, and 121]

8.2.1.2 Forced convection (any tilt)

The following relation is to be used for the case of forced air flow on the indoor side of a fenestration system (ISO 6946):

$$h_{\mathrm{c,in}} = 4 + 4V \quad W/m^2k \qquad (V \text{ in m/sec}) \qquad (1$$

8.2.2 Convective heat transfer coefficient - outdoor side

The convective heat transfer on the outdoor side primarily occurs by forced convection. The density of convective heat flow on the outdoor boundary is defined as: $q_{\mathrm{c,out}} = h_{\mathrm{c,out}}\left(T_{\mathrm{s,out}} - T_{\mathrm{out}}\right)$

where $T_{s,out}$ is used here to denote the temperature of any outdoor fenestration surface (i.e., $T_{f,1}$ or the temperature of the outdoor frame surface).

The following is the calculational procedure 28) of the convective heat transfer coefficient on the outside surface derived from the experimental work of Ito, 29)

Input:

V= wind velocity [m/s]

θ = wind direction (angle measured clockwise from north - see figure 17.)

ε = wall azimuth (positive degrees westward from south and negative eastward - see

figure 17)

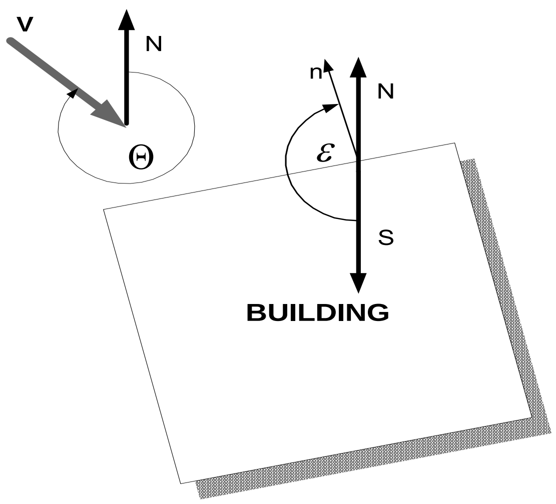


Figure 17 - Determination of wind direction and wall azimuth

n = wall normal direction

N = north

S = south

Output:

u = air velocity near the outside surface [m/s]

$h_{c,out}$ = convective heat transfer coefficient [W/m^{2}K]

Calculation sequence:

(1) Calculate wind direction relative to the wall surface γ:

$\gamma = \varepsilon + 180 - \theta$

If $|\gamma| > 180$ then $\gamma = 360 - |\gamma|$

If $-45 \leq |\gamma| \leq 45$ the surface is windward and otherwise the surface is leeward.

(2) Calculate the air velocity near the outside surface *u*:

a. If the surface is windward:

$$u = 0{,}25v \qquad V > 2 \tag{124}$$

$$u = 0{,}5v \qquad V \leq 2 \tag{125}$$

b. If the surface is leeward:

$$u = 0{,}3 + 0{,}05v \tag{126}$$

(3) Calculate the outside convective heat transfer coefficient:

$$h_{c,out} = 4{,}7 + 7{,}6v \tag{127}$$

8.2.2.1 Heat transfer by natural convection

The natural convection heat transfer coefficient for the outdoor side, $h_{c,out}$, is determined in terms of the Nusselt number, *Nu*.

$$h_{c,out} = Nu\left(\frac{\lambda}{H}\right) \tag{128}$$

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 cavity, Ra_H.

$$Ra_H = \frac{\rho^2 H^3 g C_p \left|T_{s,out} - T_{out}\right|}{T_{m,f}\,\mu\lambda} \qquad \text{(dimensionless)} \tag{129}$$

where the various fluid properties are those of air evaluated at the mean film temperature: $T_{m,f} = T_{out} + \frac{1}{4}\left(T_{s,out} - T_{out}\right)$

Correlations to quantify the outdoor side convective heat transfer coefficient are identical to the ones for indoor side (27) and are presented in clause 8.2.1.1.

NOTE The tilt angle θ, under standard conditions, needs to be replaced by its complement angle 180°- θ.

8.2.2.2 Forced convection (any tilt)

The following relation is to be used for the case of forced air flow on the indoor side of a fenestration system (ISO 6946):

$$h_{\mathrm{c,out}} = 4 + 4V \quad W/m^2k \qquad (V \text{ in } m/s) \tag{130}$$

NOTE WG2 is considering a variety of correlations to quantify forced convection heat transfer for use at indoor and/or outdoor surfaces.

8.3 Longwave radiation heat transfer

8.3.1 Mean radiant temperature

Outdoor 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).

$$G_{\mathrm{out}} = \sigma T_{\mathrm{rm,out}}^{\;4} \tag{131}$$

For field conditions, outdoor irradiance can be defined through the use of outdoor mean radiant temperature, $T_{rm,out}$:

It is assumed that outdoor fenestration surfaces are irradiated by the outdoor surfaces and the sky vault which 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 outdoor air temperature. The mean radiant outdoor temperature can then be defined as:

$$T_{\mathrm{rm,out}} = \left(\frac{\left(F_{\mathrm{ground}} + (1 - f_{\mathrm{clr}})F_{\mathrm{sky}}\right)\sigma T_{\mathrm{out}}^{\;4} + f_{\mathrm{clr}} F_{\mathrm{sky}} J_{\mathrm{sky}}}{\sigma} \right)^{1/4} \tag{132}$$

where F_{ground} and F_{sky} are view factors form the outdoor 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.

$F_{ground} = 1 - F_{sky}$

$$F_{sky} = \frac{1 + \cos\theta}{2}$$

J_{sky} is determined using the emissivity and temperature of the clear sky.

$$J_{sky} = \varepsilon_{sky} \sigma T_{sky}^{\;4}$$

Alternatively, if actual sky data is unavailable, model from Swinbank [?] can be used:

$$J_{sky} = \varepsilon_{sky} \sigma T_{out}^{\;4}$$

$$\varepsilon_{sky} = \frac{R_{sky}}{\sigma T_{out}^{\;4}}$$

$$R_{sky} = 5{,}31x10^{-13}T^{6}$$

Indoor irradiance is defined as:

$$G_{\mathrm{in}} = \sigma T_{\mathrm{rm,in}}^{4} \qquad (133a)$$

where, $T_{rm,in}$ is determined from temperatures and shape factors of surrounding indoor surfaces.

If it is assumed that indoor fenestration surfaces are irradiated only by the indoor room surfaces, which are treated as a large enclosure existing at the indoor air temperature. Indoor irradiance then becomes:

$$G_{\mathrm{in}} = \sigma T_{\mathrm{rm,in}}^{4}$$

(133)NOTE 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.

8.3.2 Detailed Radiation Heat Transfer Calculation

For fenestration systems whose ratio of total to projected boundary surface area on outdoor side is greater than 1,25 are called projecting, or 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.

The net radiation heat transfer on fenestration boundaries, q_r, of non-planar products shall be calculated using procedure outlined in section 8.3.2.1 or using the alternative procedure given in section 8.3.2.2.

8.3.2.1 Two-Dimensional Element To Element View Factor Based Radiation Heat Transfer Calculation

The emissivity of both indoor and outdoor 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 differ appreciably so, using Kirchoffs law:

$$q_{\mathrm{r,i}} = \varepsilon_{\mathrm{i}}\sigma T_{\mathrm{i}}^{\ 4} - \varepsilon_{\mathrm{i}} G_{\mathrm{i}} \qquad (134)$$

where, G_i is irradiance at surface i from all other surfaces.

$$G_{\mathrm{i}} = \sum_{\mathrm{j}}^{\mathrm{N}} F_{\mathrm{i-j}} J_{\mathrm{j}} \qquad (135)$$

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

$$J_{\mathrm{j}} = \varepsilon_{\mathrm{j}}\sigma T_{\mathrm{j}}^{\ 4} + \rho_{\mathrm{j}} G_{\mathrm{j}} \qquad (136)$$

Assuming all surfaces are grey: $\rho_j = 1 - \varepsilon_j$. Substituting ρ_j and G_j and using subscript i for convenience, equation (135) becomes:

$$J_{\mathrm{i}} = \varepsilon_{\mathrm{i}}\sigma T_{\mathrm{i}}^{\ 4} + (1-\varepsilon_{\mathrm{i}})\sum_{\mathrm{j=1}}^{\mathrm{N}} F_{\mathrm{i-j}} J_{\mathrm{j}} \qquad (137)$$

The equation (137) represents a system of N linear algebraic equations for the N unknown radiosities, J_j, which are determined from the solution of this system of equations. The system of equation (137) when expressed in matrix form becomes:

$$[C]\{J\} = \{F\} \quad (138)$$

where

$$C_{ij} = \frac{\delta_{ij} - (1 - \varepsilon_i)F_{i-j}}{\varepsilon_i} \quad (139)$$

$$F_i = \sigma T_i^{\ 4} \quad (140)$$

T_i in equation (140) is known temperature from previous iteration *k,* (i.e., $T_i|^k$). For the first iteration, the values for T_i are initial guesses).

NOTE Temperatures are calculated from the solution to the conduction problem given by equation (67), while net radiation heat flux (see equation 134) is calculated using J_i values from equation (137) and linearized term T_i^4, by using the first two terms of its Taylor series expansion about $T_i|^k$.

$$T_i^{\ 4} = 4\left(T_i|^k\right)^3 T_i|^{k+1} - 3\left(T_i|^k\right)^4 \quad (141)$$

This procedure is repeated until the following condition is satisfied:

$$\frac{\left\| T|^{k+1} \right\| - \left\| T|^k \right\|}{\left\| T|^{k+1} \right\|} \le tol \quad (142)$$

where *tol* is solution tolerance, whose value is typically less than 10^{-3}. $\| \ \|$ denotes the norm or root mean square value of the temperature vector, as defined in (30).

View factors $F_{i\text{-}j}$ can be calculated using Hottel's cross-string rule (26)If the view between two radiating surfaces is obstructed by a third surface, the effect of this obstruction shall be included.

8.3.2.2 Three-Dimensional Macro Surface Isothermal Method

The U-factor for projecting products must be reduced because of the self-viewing nature of the window. The alternate method presented here can be used instead of the multi-element method prescribed in clause 8.3.2.1.

The indoor side of the projecting product is treated as a single surface with uniform temperature and emissivity ε_g. The ratio of indoor-side surface area to indoor-side-opening area is denoted A_s/A_p. The opening area, A_p, will be similar to, but marginally less than, the projected area of the product, A_t. Using grey enclosure analysis it can be shown that the ratio of radiant heat flux at the surface of the projecting product versus the radiant heat flux at the surface of a similar non-projecting product is given by the following expression. More detail is provided by (31).

$$F_{rad} = \frac{1}{1 + \varepsilon_g \left(\frac{A_s}{A_p} - 1 \right)} \quad (149)$$

The indoor surface emissivity shall be set equal to the emissivity of indoor glazing surface ($\varepsilon_g = \varepsilon_{b,n}$).

In the analysis of individual window glazing segments, the radiant exchange from the indoor glazing surface can most readily be reduced by substituting a reduced emissivity for the indoor surface, ε_{red}, in place of the true surface emissitivity, $\varepsilon_{b,n}$.

$$\varepsilon_{\mathrm{red}} = F_{\mathrm{rad}} \varepsilon_{\mathrm{b,n}} \tag{150}$$

Similarly, in the analysis of frame and sash sections, the emissivity of the indoor surfaces shall also be reduced using the factor F_{rad}.

8.3.3 Simplified Radiation Heat Transfer Calculation

For fenestration systems whose ratio of total to projected boundary surface area is less than 1,25, the view factor from the fenestration system surfaces to the outdoor or indoor environment can be considered to be equal to 1.0 (i.e., no self radiating). In addition, it is reasonable to linearize radiation heat transfer contribution and express it in terms of surface heat transfer coefficient h_r as shown in Clauses 8.3.3.1 and 8.3.3.2.

8.3.2.1 Indoor surfaces

All indoor surfaces are denoted by subscript “s,in”, including frame surfaces. Although it is customary to consider radiation heat transfer on glazing boundary surfaces in terms of radiosities, as shown in clause 5.3.1, the following equation can be used for simplified radiation heat transfer calculations on both glazing and frame surfaces.

$$q_{\mathrm{r,in}} = h_{\mathrm{r,in}} \left(T_{\mathrm{s,in}} - T_{\mathrm{in}} \right) \tag{143}$$

where

$$h_{\mathrm{r,in}} = \frac{\varepsilon_{\mathrm{s,in}} \sigma \left(T_{\mathrm{s,in}}{}^4 - T_{\mathrm{in}}{}^4 \right)}{T_{\mathrm{s,in}} - T_{\mathrm{in}}} \tag{144}$$

8.3.2.2 Outdoor surfaces

All outdoor surfaces are denoted here by subscript $_{f,1}$, including frame surfaces. As with indoor surfaces, outdoor glazing boundary surfaces are typically dealt with in terms of radiosities, but the following equation can be used for simplified radiation heat transfer calculations on both glazing and frame surfaces.

$$q_{\mathrm{r,out}} = h_{\mathrm{r,out}} \left(T_{\mathrm{s,out}} - T_{\mathrm{r,m}} \right) \tag{145}$$

where:

$$h_{\mathrm{r,out}} = \frac{\varepsilon_{\mathrm{s,out}} \sigma \left(T_{\mathrm{s,out}}{}^4 - T_{\mathrm{r,m}}{}^4 \right)}{T_{\mathrm{s,out}} - T_{\mathrm{r,m}}}$$

8.4 Combined convective and radiative heat transfer

$$q = h \left(T_{\mathrm{s,in}} - T_{\mathrm{ne}} \right) \tag{146}$$

where $h = h_{\mathrm{r}} + h_{\mathrm{c}}$

For boundary surface whose geometry is approximated using rules detailed in Clause 6.3.1, the following adjustment to the combined surface heat transfer coefficient shall be applied:

$$h_{\mathrm{adjusted}} = \frac{A_{\mathrm{real}}}{A_{\mathrm{approximated}}} h \tag{147}$$

8.5 Prescribed density of heat flow rate

The frame/wall interface shall be treated as adiabatic. Also see ISO/DIS 10077 -2.

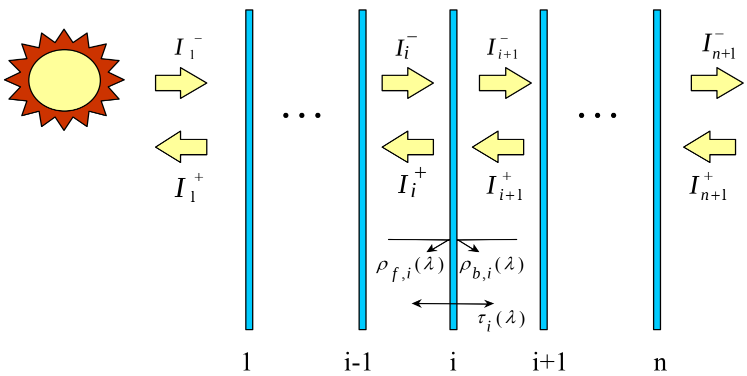
Even though approximate correlation in equation (13) is used for frame total solar energy transmittance calculation, the alternative method is to use density of heat flow boundary condition, which is equal to absorbed quantity of solar radiation on frame surfaces.

$$q_s = \alpha I_s \qquad (148)$$

Annex A
(informative)

Solution technique for the multi-layer solar optical model

A window with n glazing together with the outdoor (i=0) and indoor(i=n+1) spaces form an n+2 element array. The glazing system optical analysis can be carried out by considering the spectral fluxes of radiant energy flowing between the i-1th and ith glazings, $I_i^+(\lambda)$ and $I_i^-(\lambda)$. The + and - superscripts denote radiation flowing toward the outdoor and indoor side, respectively, as shown in Figure A.1.



Fígure A.1 - Analysis of solar flux in multi-layer glazing system

Equations (A.1) and (A.2) are applied while setting the reflectance and transmittance of the conditioned space to zero, $\rho_{\mathrm{f,n+1}}(\lambda)=0$ and $\tau_{\mathrm{n+1}}(\lambda)=0$.

$$I_i^+(\lambda)=\tau_i(\lambda)I_{i+1}^+(\lambda)+\rho_{\mathrm{f,i}}(\lambda)I_i^-(\lambda) \text{ i=1 to n+1} \quad \text{(A.1)}$$

$$I_i^-(\lambda)=\tau_{i-1}(\lambda)I_{i-1}^-(\lambda)+\rho_{\mathrm{b,i-1}}(\lambda)I_i^+(\lambda) \text{ i=2 to n+1} \quad \text{(A.2)}$$

It can be shown that the ratio of $I_i^+(\lambda)$ to $I_i^-(\lambda)$ is given by:

$$\frac{I_i^+(\lambda)}{I_i^-(\lambda)}=r_i(\lambda)=\rho_{\mathrm{f,i}}(\lambda)+\frac{\tau_i^2(\lambda)r_{i+1}(\lambda)}{1-\rho_{\mathrm{b,i}}(\lambda)r_{i+1}(\lambda)} \quad \text{(A.3)}$$

and the ratio of $I_{i-1}^-(\lambda)$ to $I_i^-(\lambda)$ is given by:

$$\frac{I^{-}_{i+1}(\lambda)}{I^{-}_{i}(\lambda)} = t_{i}(\lambda) = \frac{\tau_{i}(\lambda)}{1-\rho_{b,i}(\lambda)r_{i+1}(\lambda)} \qquad \text{(A.4)}$$

Edwards (32) gives the development of equations (A.3) and (A.4) by the net radiation method. Wright (5) uses ray tracing to reproduce this derivation.

Now all values of $I_i^-(\lambda)$ and $I_i^+(\lambda)$ can be found using the following steps. First, use equation (153)as a recursion relationship to calculate all values of $r_i(\lambda)$ by working from $r_{n+1}(\lambda) = \rho_{f,n+1}(\lambda) = 0$ to $r_1(\lambda)$. Second, use equation (A.4) to obtain $t_i(\lambda)$ from i=1 to i=n. The rest of the solution follows by first calculating the flux reflected from the glazing system to the outdoor side ($I_1^+(\lambda) = r_1(\lambda)I_1^-(\lambda)$) and then marching toward the indoor side from i=2 to i=1 calculating the remaining fluxes ($I_i^-(\lambda) = t_{i-1}(\lambda)I_{i-1}^-(\lambda)$ and $I_i^+(\lambda) = r_i(\lambda)I_i^-(\lambda)$).

Finally, the desired portions of incident radiation at each of the glazing layers are given by:

$$A_i(\lambda) = \frac{I_i^-(\lambda) - I_i^+(\lambda) + I_{i+1}^+(\lambda) - I_{i+1}^-(\lambda)}{I_1^-(\lambda)} \qquad \text{(A.5)}$$

and the portion transmitted to the conditioned space is,

$$\tau_s(\lambda) = \frac{I_{n+1}^-(\lambda)}{I_1^-(\lambda)} \qquad \text{(A.6)}$$

NOTE $I_1^-(\lambda)$ can be set equal to unity for the purpose of solving these equations.

Annex B
(normative)
Thermophysical fill gas property values

The following tables present linear equation coefficients with which the conductivity, viscosity and specific heat at constant pressure can be calculated for four (air, argon, krypton, xenon) glazing cavity fill gases. The least square linear curve fit equations were derived from thermophysical property data given in (33) Note that the molecular weights are listed in separate Table B.4.

The heat transfer calculations are based on the assumption that the fill gas is not an emitting/absorbing gas. Since SF_6 violates this condition it has not been presented in these tables.

Table B.1 - Thermal conductivity

Gas	Coefficient a W/(M·K)	Coefficient b W/(m·K²)	λ @ 0 ºC W/(m·K)
Air	$2{,}8733\times10^{-3}$	$7{,}76\times10^{-5}$	0,024
Argon	$2{,}2848\times10^{-3}$	$5{,}1486\times10^{-5}$	0,016
Krypton	$9{,}4429\times10^{-4}$	$2{,}8257\times10^{-5}$	0,0087
Xenon	$4{,}5381\times10^{-4}$	$1{,}7229\times10^{-5}$	0,0052

Where $\lambda = a+b*T(K)$

Table B.2 - Viscosity

Gas	Coefficient a N*s/m²	Coefficient b N*s/(m²·K)	μ @ 0 ºC N*s/m²
Air	$3{,}723\times10^{-6}$	$4{,}94\times10^{-8}$	$1{,}72\times10^{-5}$
Argon	$3{,}379\times10^{-6}$	$6{,}451\times10^{-8}$	$2{,}1\times10^{-5}$
Krypton	$2{,}213\times10^{-6}$	$7{,}777\times10^{-8}$	$2{,}4\times10^{-5}$
Xenon	$1{,}069\times10^{-6}$	$7{,}4143\times10^{-8}$	$2{,}1\times10^{-5}$

Where $\mu = a+b*T(K)$

Table B.3 - Specific heat at constant pressure

Gas	Coefficient a J/(kg·K)	Coefficient b J/(kg·K²)	Cp @ 0 ºC J/(kg*K)
Air	$9{,}693 \times 10^{2}$	$1{,}192 \times 10^{-2}$	972,5559
Argon	521,9285	0	521,9285
Krypton	248,0907	0	248,0917
Xenon	158,3397	0	158,3397

Where Cp = a+b*T(K)

Table B.4 - Molecular weights

Gas	kg/Kmol
Air	28,97
Argon	39,948
Krypton	83,80
Xenon	131,30

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>>>check use of italics instead of quotes, etc. (see BasicEn2.dot)!

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