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

*Performance Thermique de Vitrages, Portes et francais francais francais — Calculs détaillés*

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## 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, total solar energy transmittance and visible light transmittance.

The procedures in ISO 15099 give the actual thermal performance of fenestration products for use in building energy analysis and for the evaluation of products in specific building applications. These procedures can also be used to produce data to compare products by using the standardised boundary conditions given either in this standard or taken from the appropriate international or national standards (e.g., EN ISO 12567, ISO 10292, ISO 9050). The standard is also intended as a reference document for description of models used in computer programs for detailed calculation of the thermal and optical transmission properties of window and door systems.

The standard gives detailed models for thermal and optical transmission in windows. These detailed models are necessary in many types of windows to get agreement between calculations and tests.

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 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. ISO 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 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 International Standards. It is essential that these programs produce consistent values and that they are based on a sound standard methodology. Although more complicated than the formulas used in the simplified standards, the formulas used in ISO 15099 are entirely appropriate for their intended use.



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

ISO 9288, *Thermal insulation - Heat transfer by radiation – Physical quantities and definitions.*

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

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

EN 12898, *Glass in buildings-Determination of the emissivity*<sup>1</sup>

ISO/CIE 10526, *CIE standard Illuminates for colorimetry.*

ISO/CIE 10527, *CIE standard colorimetric observers.*

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<sup>1</sup> To be published.

### 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 1.

#### 3.2 Symbols

Table 1 - Symbols

Symbol	Quantity	Unit
$A$	area	$m^2$
	or aspect ratio	
$A_i$	Portion of absorbed solar energy by the $i^{\text{th}}$ glazing layer	
$c_p$	specific heat capacity at constant pressure	$J/(kg \cdot K)$
$d$	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 internal 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	$W/(m^2 \cdot K)$
$H$	distance (height of glazing cavity)	$m$
$I_i^+(\lambda)$ , $I_i^-(\lambda)$	spectral heat flow rate of radiant solar energy between $i^{\text{th}}$ and $i+1^{\text{th}}$ glazing layers travelling in the external or $i$ internal direction, respectively.	
$I$	total density of heat flow rate of incident solar radiation	$W/m^2$
$J$	radiosity	$W/m^2$
$L$	height of glazing cavity	$m$
$l$	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
$\mathfrak{R}$	universal gas constant	$J/(kmol \cdot K)$
$R$	thermal resistance	$m^2 \cdot K/W$
$Ra$	Rayleigh number	
$Ra_x$	Rayleigh number based on length dimension $x$	
$R(\lambda)$	photopic response of the eye (ISO/CIE 10527)	
$S_i$	density of heat flow rate of absorbed solar radiation at $i^{\text{th}}$ glazing layer	$W/m^2$
$t$	thickness	$m$
$t_{\text{perp}}$	largest dimension of frame cavity perpendicular to heat flow	$m$
$T$	temperature	K
$u$	air velocity near a surface	$m/s$
$U$	thermal transmittance	$W/(m^2 \cdot K)$
$V$	free-stream air speed near window, mean air velocity in a gap	$m/s$
$x, y$	dimensions in a Cartesian co-ordinate system	

$Z$	pressure loss factor	
$\alpha$	absorptance	
$\beta$	thermal expansion coefficient of fill gas	$K^{-1}$
$\Delta T_i$	temperature drop across $i^{\text{th}}$ glazing cavity, $\Delta T_i =  T_{f,i} - T_{b,i+1} $	
$\varepsilon$	total hemispherical emissivity, or angle	
$\gamma$	angle	
$\theta$	temperature	$^{\circ}C$
	or angle	degree
$\sigma$	Stefan-Boltzmann constant, $5,6693 \times 10^{-8}$	$W/(m^2 \cdot K^4)$
$\phi$	Parameter defined in Equations (62 and 68)	
$\varphi$	function defined in Equation (97)	
$\lambda$	thermal conductivity	$W/(m \cdot K)$
	or wavelength	m
$\mu$	dynamic viscosity	$g/(m \cdot s)$
$\rho$	density	$kg/m^3$
	or specular reflectance: portion of incident radiation reflected such that the angle of reflection is equal to the angle of incidence	
$\tau$	transmittance	
$\Psi$	linear thermal transmittance	$W/(m \cdot K)$

### 3.3 Subscripts

The subscripts given in Table 2 are used.

**Table 2 - Subscripts**

Abbreviation	Name
ave	average
air	air
bot	bottom of a gap
b	backward
c	convection
cq	center of glass
cold	condition on the cold side
crit	critical
de	divider edge glass
dif	diffuse
dir	direct
div	divider
eff	effective
eq	edge of glass
eq	equivalent
f	front, or frame
fr	frame (using the alternate approach)
g	glass or vision portion
h	hot
h	horizontal
hot	condition on the hot (warm) side
in	internal, or inlet
:	...

m	mean
mix	mixture
n	counter
ne	environmental (external)
ni	environmental (internal)
out	external, or outlet of a gap
p	panel
r	radiation or radiant
red	reduced radiation
s	solar, source, sink or surface
surf	surface
t	total
top	top of a gap
v	vertical
x	at distance x
Ψ	perimeter
2D	coupling

## 4 Determination of total window and door system properties

### 4.1 Thermal transmittance

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-A.

**Figure 1 A- 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  $\tau_g$ . Clause 6 contains the procedure to calculate the corresponding frame properties,  $U_f$  and  $g_f$ , as well as the linear thermal transmittance,  $\Psi$ , 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. The thermal transmittance of the fenestration product is:

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

where,  $A_g$  and  $A_f$  are the projected vision area and frame area, respectively. The length of the vision area perimeter is  $l_\psi$ , and  $\Psi$  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. Also in Figure 1-B, the edge-of-glass and divider edge areas are 63,5 mm (2,5 in.) wide. The sum of all component areas equals the total projected fenestration product area.

**Figure 1-B: Center-of-glass, edge-of-glass, divider, divider-edge, and frame areas for a typical fenestration product.**

#### 4.1.1 Glazed area thermal transmittance

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

$$U_g = \frac{1}{R_t} \tag{2A}$$

and when solar radiation is considered, then:

$$U_g = \frac{q_{in}(I_s = 0)}{T_{ni} - T_{ne}} \tag{2B}$$

where  $q_{in}(I_s=0)$  is the net density of heat flow rate through the window or door system to the internal environment for the specified conditions, but without incident solar radiation, in  $W/m^2$ . The condition 'without solar radiation' realises that all effects on the thermal resistances due to incident solar radiation are incorporated in the total solar energy transmittance or g-value (see Equation (13)), and  $T_{ni}$ , and  $T_{ne}$  are the environmental temperatures, as defined in Equation (6).

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

$$R_t = \frac{1}{h_{out}} + \sum_{i=2}^n R_i + \sum_{i=1}^n R_{g,i} + \frac{1}{h_{in}} \tag{3}$$

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

$$R_{g,i} = \frac{t_{g,i}}{\lambda_{g,i}} \tag{4}$$

and the thermal resistance of the  $i^{th}$  space, where the first space is external environment, the last space is internal environment, and the spaces in between are glazing cavities, (see Figure 5)

$$R_i = \frac{T_{f,i} - T_{b,i-1}}{q_i} \tag{5}$$

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

The environmental temperature (as defined in Equation (6) below) is a weighted average of the ambient air temperature and the mean radiant temperature,  $T_{rm}$ , which is determined for external and internal environment boundary conditions (see boundary conditions in 8.4.1).

**Figure 2 - Numbering system for glazing system layers**

Thus, the environmental temperature,  $T_n$ , is:

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

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

#### 4.1.2 Frame area/edge-glass thermal indices

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

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

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

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

where

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

$U_p$  is the thermal transmittance of foam insert;

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

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

$l_g$  is the internal 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 (see Figure 1B above)

An alternate method is available for calculating frame thermal transmittance,  $U_{fr}$ . Using this method it is unnecessary to determine the linear thermal transmittance,  $\Psi$ . 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,  $U_{eg}$ , is used to characterize the edge-glass area. If dividers are present then divider area,  $A_{div}$  and divider thermal transmittance,  $U_{div}$  are calculated, as well as corresponding divider edge area,  $A_{de}$  and thermal transmittance,  $U_{de}$ . The following equation shall be used to calculate the total thermal transmittance:

$$U_t = \frac{\sum U_{cg} A_c + \sum U_{fr} A_f + \sum U_{eg} A_e + \sum U_{div} A_{div} + \sum U_{de} A_{de}}{A_t} \quad (9)$$

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

$$U_{fr} = \frac{Q_{fr}}{l_f (T_{ni} - T_{ne})} \quad (10)$$

$$U_{eg} = \frac{Q_{eg}}{l_{eg} (T_{ni} - T_{ne})} \quad (11)$$

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 internal side. The quantities  $Q_{fr}$  and  $Q_e$  are heat flow rates through frame and edge-glass areas (internal surfaces), respectively, including the effect of glass and spacer, and both are expressed per length of frame or edge-glass (i.e., W/m). The calculations will have to be performed for each combination of frame and glazing with different spacer bars.

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.

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

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

## 4.2 Total solar energy transmittance

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

$$g_t = \frac{\sum g_g A_g + \sum g_f A_f}{A_t} \quad (12)$$

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.

Equation (12) includes an assumption that the solar transmittance of the edge of glass is the same as that of the center of glass area.

### 4.2.1 Vision area total solar energy transmittance

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

$$g_g = \frac{q_{in} - q_{in}(I_s = 0)}{I_s} \quad (13)$$

where

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

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

For the equivalent expression for  $U$ , see Equation (2B).

The net density of heat flow rates,  $q_{in}$  and  $q_{in}(I_s=0)$  are calculated in 5.3.1 (Equation (26), for index  $i = in$ ).

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

#### 4.2.2 Frame total solar energy transmittance

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

$$g_f = \alpha_f \cdot \frac{U_f}{\frac{A_{surf}}{A_f} h_{out}} \quad (14)$$

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

If the alternate method of calculating  $U_i$  is being used,  $U_{fr}$  should be used instead of  $U_f$  in Equation (14).

More detailed two dimensional or three dimensional calculations, including the effects of off-normal solar radiation, shading, reflected solar radiation and solar radiation transmitted to the internal frame surfaces, can be performed in a manner analogous to Equation (13), and subject to boundary conditions given in 8.6.

### 4.3 Visible transmittance

The visible transmittance of the total fenestration product is:

$$\tau_t = \frac{\sum \tau_v A_g}{A_t} \quad (16)$$

## 5 Vision area properties

### 5.1 Glazing layer optics

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

### 5.1.1 Solar

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

Note: More information about  $\rho_{f,i}(\lambda)$ ,  $\rho_{b,i}(\lambda)$  and  $\tau_i(\lambda)$  can be found in [2].

**Figure 3 - Out-door and internal 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  $i^{\text{th}}$  glazing are (1) the front (external side) hemispheric emittance,  $\varepsilon_{f,i}$ , (2) the back (internal side) hemispheric emittance,  $\varepsilon_{b,i}$  and (3) the hemispheric-hemispheric transmittance,  $\tau_i$ . These total optical properties apply to wavelengths from 5 to 50  $\mu\text{m}$ .

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

Note: Also see ASTM E1585-93 [3].

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  $\tau_i=0$  and hemispherical emittance can be calculated as in [3]. For partially transparent films such as polyethylene terephthalate (PET) both specular transmittance and reflectance should be measured. Using the bulk model the optical indices of the material can then be calculated and used to derive the hemispherical properties.

Note: See [4] for more information.

## 5.2 Glazing system optics

### 5.2.1 Spectral quantities

The path of incident solar radiation within the various layers of the glazing system shall be modelled by the methods 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  $i^{\text{th}}$  glazing layer and solar spectral transmittance**

Figure 4 shows how a window with  $n$  glazing layers together with the external ( $i=0$ ) and internal ( $i=n+1$ ) spaces can be treated as an  $n+2$  element array. It is necessary to determine the portion of incident solar radiation, at a given wavelength, that is absorbed at each of the glazing layers. This quantity is denoted  $\alpha_i(\lambda)$  at the  $i^{\text{th}}$  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 provide an exact solution is acceptable.

Note: The solution technique described in [5] is summarised 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 flow. Values of  $E(\lambda)$  are reported at  $N_s$  values of  $\lambda$  (denoted here as  $E(\lambda_j)$  and  $\lambda_j$ , 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 flow rate of solar radiation absorbed at the  $i^{\text{th}}$  glazing layer,  $S_i$ , is determined by numerical integration over the solar spectrum according to Equations (16), (17) and (18).

$$A_i = \frac{\sum_{j=1}^{N_s-1} \alpha_i(\lambda_{j/j+1}) E_s(\lambda_{j/j+1}) \Delta\lambda_j}{\sum_{j=1}^{N_s-1} E_s(\lambda_{j/j+1}) \Delta\lambda_j} \quad (16)$$

$$\Delta\lambda_j = \lambda_{j+1} - \lambda_j \quad (17)$$

$$S_i = A_i \cdot I_s \quad (18)$$

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

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

and

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

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

### 5.2.4 Solar transmittance

The solar transmittance of the glazing system is:

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

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

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

### 5.2.5 Visible transmittance

Visible transmittance,  $\tau_v$ , 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  $\lambda_j$ .  $\tau_v$  is given by:

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

where

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

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

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

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

**Figure 5 - Energy balance on glazing layer i**

Figure 5 shows the  $i^{\text{th}}$  glazing in a sloped multilayer array. The values of four variables are sought at each glazing. These are the temperatures of the external and internal facing surfaces,  $T_{f,i}$  and  $T_{b,i}$ , plus the radiant heat 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:

$$q_i = h_{c,i} [T_{f,i} - T_{b,i-1}] + J_{f,i} - J_{b,i-1} \quad (26)$$

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

$$q_i = S_i + q_{i+1} \quad (27)$$

$$J_{f,i} = \varepsilon_{f,i} \sigma T_{f,i}^4 + \tau_i J_{f,i+1} + \rho_{f,i} J_{b,i-1} \quad (28)$$

$$J_{b,i} = \varepsilon_{b,i} \sigma T_{b,i}^4 + \tau_i J_{b,i-1} + \rho_{b,i} J_{f,i+1} \quad (29)$$

$$T_{b,i} - T_{f,i} = \frac{t_{g,i}}{2\lambda_{g,i}} [2q_{i+1} + S_i] \quad (30)$$

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

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

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

Equation (27) describes an energy balance imposed at the surfaces of the  $i^{\text{th}}$  glazing. Equations (28) and (29) define the radiosities at the  $i^{\text{th}}$  glazing, and  $\rho_{f,i} = 1 - \tau_i - \varepsilon_{f,i}$  and  $\rho_{b,i} = 1 - \tau_i - \varepsilon_{b,i}$ .

The temperature difference across the  $i^{\text{th}}$  glazing is given by Equation (30). It is assumed that the solar energy is absorbed uniformly through the thickness of the glazing.

Note: More detail regarding Equations (26) through Equation (31) is given by Wright [5] and Hollands, Wright and Granqvist [35].

### 5.3.2 Interaction with the environment

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

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

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

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

$$J_{f,n+1} = G_{g,\text{in}} \quad (34)$$

$$\text{and } J_{b,0} = G_{g,\text{out}} \quad (35)$$

where  $G_{g,\text{in}}$  and  $G_{g,\text{out}}$  are given by Equations (133) and (131) in clause 8, respectively.

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

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

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

### 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 fill gas layer (or pane spacing)  $i$ , 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,  $\theta$ .

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.

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

The Rayleigh number can be expressed as (omitting the "i" and "g" subscripts for convenience):

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

Treating the fill gas as a perfect gas the thermal expansion coefficient of the fill gas,  $\beta$ , is:

$$\beta = \frac{1}{T_m} \quad (40)$$

where  $T_m$  = fill gas mean temperature (K)

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

$$A_{g,i} = \frac{H}{d_{g,i}} \quad (41)$$

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,  $\theta$ .

This categorization, as a function of  $\theta$ , is based on the assumption that the cavity is heated from the internal 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,  $180^\circ - \theta$ , instead of  $\theta$  and to then substitute  $180^\circ - \theta$  instead of  $\theta$  when the calculation is carried out.

### 5.3.3.1 Cavities inclined at $0 \leq \theta < 60^\circ$

$$Nu_i = 1 + 1,44 \left[ 1 - \frac{1708}{Ra \cos(\theta)} \right]^* \left[ 1 - \frac{(1708 \sin^{1,6}(1,8\theta))}{Ra \cos(\theta)} \right] + \left[ \left[ \frac{Ra \cos(\theta)}{5830} \right]^{1/3} - 1 \right]^* \quad (42)$$

$Ra < 10^5$  and  $A_{g,i} > 20$

$$\text{where } [x]^* = (x + |x|) / 2 \quad (43)$$

Note: For more details, see Reference [7].

### 5.3.3.2 Cavities inclined at $\theta = 60^\circ$

$$Nu = [Nu_1, Nu_2]_{\max} \quad (44)$$

$$\text{where: } Nu_1 = \left[ 1 + \left[ \frac{0,0936 Ra^{0,314}}{1 + G} \right]^7 \right]^{\frac{1}{7}} \quad (45)$$

$$Nu_2 = \left[ 0,104 + \frac{0,175}{A_{g,i}} \right] Ra^{0,283} \quad (46)$$

$$G = \frac{0,5}{\left[ 1 + \left[ \frac{Ra}{3160} \right]^{20,6} \right]^{0,1}} \quad (47)$$

Note: For more details, see Reference [8].

### 5.3.3.3 Cavities inclined at $60^\circ < \theta < 90^\circ$

For layers inclined at angles between  $60^\circ$  and  $90^\circ$ , a straight-line interpolation between the results of Equations (44) and (48) is used. These equations are valid in the ranges of:

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

Note: For more details, see Reference [8].

### 5.3.3.4 Vertical cavities

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

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

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

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

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

Note: For more details, see Reference [9].

### 5.3.3.5 Cavities inclined from $90^\circ$ to $180^\circ$

Gas layers contained in downward facing windows are modelled using:

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

$Nu_V$  is the Nusselt number for a vertical cavity given by Equation (48).

Note: For more details, see Reference [10].

### 5.3.3.6 Fill gas properties

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

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

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

The specific heat capacity at constant pressure,  $c_p$ , and the transport properties  $\mu$  and  $\lambda$  are evaluated using linear functions of temperature. For example, the viscosity can be expressed as:

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

Values of a and b coefficients appropriate for calculating  $c_p$ ,  $\mu$  and  $\lambda$  for a variety of fill gases are given in Annex B.

### 5.3.4 Properties of fill gas mixtures

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

Note: For more details, see Reference [11].

#### 5.3.4.1 Molecular mass

$$\hat{M}_{\text{mix}} = \sum_{i=1}^v x_i \hat{M}_i \quad (56)$$

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

#### 5.3.4.2 Density

$$\rho = \frac{P\hat{M}_{\text{mix}}}{\mathfrak{R}T_m} \quad (57)$$

#### 5.3.4.3 Specific heat

$$c_{p\text{mix}} = \frac{\hat{c}_{p\text{mix}}}{\hat{M}_{\text{mix}}} \quad (58)$$

where 
$$\hat{c}_{p\text{mix}} = \sum_{i=1}^v x_i \hat{c}_{p,i} \quad (59)$$

and, the molar specific heat capacity of the  $i^{\text{th}}$  gas is

$$\hat{c}_{p,i} = c_{p,i} \hat{M}_i \quad (60)$$

#### 5.3.4.4 Viscosity

$$\mu_{mix} = \sum_{i=1}^v \frac{\mu_i}{\left[ 1 + \sum_{\substack{j=1 \\ j \neq i}}^v \phi_{i,j}^{\mu} \frac{x_j}{x_i} \right]} \quad (61)$$

where

$$\phi_{i,j}^{\mu} = \frac{\left[ 1 + \left( \frac{\mu_i}{\mu_j} \right)^{1/2} \left( \frac{\hat{M}_j}{\hat{M}_i} \right)^{1/4} \right]^2}{2\sqrt{2} \left[ 1 + \left( \frac{\hat{M}_i}{\hat{M}_j} \right) \right]^{1/2}} \quad (62)$$

#### 5.3.4.5 Thermal conductivity

$$\lambda_{mix} = \lambda'_{mix} + \lambda''_{mix} \quad (63)$$

where:

$\lambda'$  is the monatomic thermal conductivity

$\lambda''$  is the additional energy moved by the diffusional transport of internal energy in polyatomic gases.

$$\lambda'_{mix} = \sum_{i=1}^v \frac{\lambda'_i}{\left[ 1 + \sum_{\substack{j=1 \\ j \neq i}}^v \psi_{i,j} \frac{x_j}{x_i} \right]} \quad (64)$$

and,

$$\psi_{i,j} = \frac{\left[ 1 + \left( \frac{\lambda'_i}{\lambda'_j} \right)^{1/2} \left( \frac{\hat{M}_i}{\hat{M}_j} \right)^{1/4} \right]^2}{2\sqrt{2} \left[ 1 + \left( \frac{\hat{M}_i}{\hat{M}_j} \right) \right]^{1/2}} \cdot \left[ 1 + 2,41 \frac{(\hat{M}_i - \hat{M}_j)(\hat{M}_i - 0,142\hat{M}_j)}{(\hat{M}_i + \hat{M}_j)^2} \right] \quad (65)$$

and,

$$\lambda''_{\text{mix}} = \sum_{i=1}^v \frac{\lambda''_i}{\left\{ 1 + \sum_{\substack{j=1 \\ j \neq i}}^v \phi^{\lambda_{i,j}} \frac{x_j}{x_i} \right\}} \quad (66)$$

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

$$\phi^{\lambda_{i,j}} = \frac{\left[ 1 + \left( \lambda'_i / \lambda'_j \right)^{1/2} \left( \hat{M}_i / \hat{M}_j \right)^{1/4} \right]^2}{2\sqrt{2} \left[ 1 + \left( \hat{M}_i / \hat{M}_j \right) \right]^{1/2}} \quad (67)$$

To find  $\lambda_{\text{mix}}$ , use the following steps.

a- Calculate  $\lambda'_i$

$$\lambda'_i = \frac{15}{4} \frac{\Re}{\hat{M}_i} \mu_i \quad (68)$$

b- Calculate  $\lambda''_i$

$$\lambda''_i = \lambda_i - \lambda'_i \quad \lambda_i \text{ is the conductivity of the } i^{\text{th}} \text{ fill gas component (see Annex B)} \quad (69)$$

c- Use  $\lambda'_i$  to calculate  $\lambda'_{\text{mix}}$

d- Use  $\lambda''_i$  to calculate  $\lambda''_{\text{mix}}$

e-  $\lambda_{\text{mix}} = \lambda'_{\text{mix}} + \lambda''_{\text{mix}}$  (69-a)

## 6 Frame effects

### 6.1 Area and lineal thermal transmittance

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

Frame areal thermal transmittance (thermal transmittances) and lineal thermal transmittance ( $\Psi$ -factor) shall be determined using two-dimensional numerical modelling. This two-dimensional analysis shall provide the rate of heat transfer through each unique frame section. See National standards for the required cross sections to be considered. Details regarding the required two dimensional numerical analysis are provided in 6.2 through 6.6 (also see ISO/DIS 10077-2).

## 6.2 Governing equations for calculating thermal transmittance

The governing equation shall be developed by imposing an energy balance describing steady-state heat transfer by conduction. The governing equation shall be discretized using a conservative formulation (i.e., the evaluation of 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,  $\lambda_{fe}$ , must be specified. The numerical solver shall be able to generate the two dimensional heat flow and temperature patterns that satisfy the governing equation. In Cartesian co-ordinates this equation is:

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} = -\frac{q'''_{\text{source}}}{\lambda} \quad (70)$$

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

$$q = -\lambda_{fe} \left( \frac{\partial T}{\partial x} e_x + \frac{\partial T}{\partial y} e_y \right) \quad (71)$$

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:

$$q = q_c + q_r + q_s \quad (72)$$

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 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 may be ignored.

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

Note: For more details, see Reference [34].

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

$$\lambda_{eff} = F_b \cdot \lambda_b + (1 - F_b) \lambda_n \quad (72-a)$$

where

$$F_b = s/d$$

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

$d$  is the spacing of thermal bridging elements

$\lambda_b$  is the conductivity of thermal bridging material

$\lambda_n$  is the conductivity of the cross-section without the thermal bridge

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

- a) If  $F_b \leq 1\%$ , 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 modelled 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).

Those segments of the actual cross section that are made up of vertical and horizontal surfaces shall be represented by similar straight lines that preserve the nominal thickness and relative position of the segment. Sloped lines shall be represented by (a) similar sloped lines that preserve the nominal thickness and relative position of the segment or by (b) a series of horizontal and vertical lines, which meet criteria 1 to 4 below. Curves shall be represented by (a) similar curves that preserve the nominal thickness and relative position of the segment or by (b) a series of horizontal, vertical, and sloped lines or a series of horizontal and vertical lines which meet criteria 1 to 4 below (See Figure 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 times conductivity shall be replaced by web thickness times conductivity times  $(\cos(\gamma) + \sin(\gamma))$ , where  $\gamma$  is the angle of inclination of the sloped web. The same result is obtained whether  $\gamma$  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 internal and external boundaries shall follow the frame profile. In the case where there are exterior and interior open frame profile cavities (ventilated cavities and groves).

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

### 6.3.2 Meshing

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

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

This requirement is more stringent than that specified in ISO 10211-1: which requires that the number of subdivisions be doubled until the change in heat flow through the object is reduced to a prescribed tolerance. The more stringent 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 [b] below.

b. Energy error norm [13 and 14] is applied so that the calculated frame/edge thermal transmittance is within 1% of the thermal transmittance determined with an ideal mesh.

c. Any other approach documented in refereed publications applied so that the calculated frame/edge thermal transmittance is within 1% of the thermal transmittance determined with an ideal mesh.

## 6.4 Solid materials

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

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

## 6.5 Effective conductivity - Glazing cavities

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

$$\lambda_{\text{eff},i} = q_i \left[ \frac{d_{\text{g},i}}{T_{\text{f},i} - T_{\text{b},i-1}} \right] \quad (73)$$

## 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.

$$\lambda_{\text{eff}} = (h_c + h_r) \cdot d \quad (74)$$

where

$\lambda_{\text{eff}}$  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.

$$h_c = Nu \frac{\lambda_{\text{air}}}{d} \quad (75)$$

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**

$$Nu = 1,0 \quad (76)$$

See Figure 7.

**Figure 7 – 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, see Figure 8.

**Figure 8 – Illustration of rectangular frame upward flow direction**

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

$$Nu = 1,0 \quad (77)$$

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

$$Nu = 1 + \left[ 1 - \frac{Ra_{\text{crit}}}{Ra} \right]^+ \left[ k1 + 2(k2)^{1-\ln k2} \right] + \left[ \left( \frac{Ra}{5380} \right)^{\frac{1}{3}} - 1 \right]^+ \left[ 1 - e^{-0,95 \left( \left( \frac{Ra_{\text{crit}}}{Ra} \right)^{\frac{1}{3}} - 1 \right)^+} \right] \quad (78)$$

where

$$k1 = 1,40 \quad (78-a)$$

$$k2 = \frac{Ra^{\frac{1}{3}}}{450,5} \quad (78-b)$$

$$[x]^+ = \frac{x + |x|}{2} \quad (78-c)$$

$Ra_{\text{crit}}$  is a critical Rayleigh number, which is found by least squares regression of tabulated values.

Note: For more details, see Reference [15].

$$Ra_{\text{crit}} = e^{\left(0,721 \frac{L_h}{L_v}\right)^{+7,46}} \quad (78-d)$$

$Ra$  is the Rayleigh number for the air cavity:

$$Ra = \frac{\rho_{\text{air}}^2 L_v^3 g \beta C_{p,\text{air}} (T_{\text{hot}} - T_{\text{cold}})}{\mu_{\text{air}} \lambda_{\text{air}}} \quad (78-e)$$

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

$$Nu = 1 + 1,44 \left[1 - \frac{1708}{Ra}\right]^+ + \left[\left(\frac{Ra}{5830}\right)^{\frac{1}{3}} - 1\right]^+ \quad (79)$$

Note: For more details, see Reference [7].

### 6.6.3 Horizontal heat flow

See illustration in Figure 9.

**Figure 9 – Illustration of rectangular frame cavity horizontal flow direction**

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

$$Nu = 1 + \left\{ \left( 2,756 \cdot 10^{-6} Ra^2 \left(\frac{L_v}{L_h}\right)^8 \right)^{-0,386} + \left( 0,623 Ra^{\frac{1}{5}} \left(\frac{L_h}{L_v}\right)^{\frac{2}{5}} \right)^{-0,386} \right\}^{-2,59} \quad (80)$$

Note: For more details, see Reference [15].

where  $Ra$  is Raleigh number and is defined as:

$$Ra = \frac{\rho_{\text{air}}^2 L_h^3 g \beta C_{p,\text{air}} (T_{\text{hot}} - T_{\text{cold}})}{\mu_{\text{air}} \lambda_{\text{air}}} \quad (80\text{-a})$$

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

$$Nu_{\text{ct}} = \left\{ 1 + \left[ \frac{(0,104 Ra^{0,293})}{\left( 1 + \left( \frac{6310}{Ra} \right)^{1,36} \right)} \right]^3 \right\}^{1/3} \quad (81)$$

$$Nu_l = 0,242 \left( Ra \frac{L_h}{L_v} \right)^{0,273} \quad (82)$$

$$Nu_t = 0,0605 Ra^{1/3} \quad (83)$$

Note: For more details, see Reference [9].

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

For jamb frame sections, frame cavities are oriented vertically and therefore the height of the cavity is in the direction normal to the plane of the cross section. For these cavities it is assumed that heat flow is always in horizontal direction with  $L_v/L_h > 5$ , and so correlations in Equations (81) to (83) in 6.6.3.b shall be used.

The temperatures  $T_{\text{hot}}$  and  $T_{\text{cold}}$  are not known in advance, so it is necessary to estimate them. From previous experience it is recommended to apply  $T_{\text{hot}}=10^\circ\text{C}$  and  $T_{\text{cold}}=0^\circ\text{C}$ . However, after the simulation is done, it is necessary to update these temperatures from the results of the previous run. This procedure shall be repeated until values of  $T_{\text{hot}}-T_{\text{cold}}$  from two consecutive runs are within  $1^\circ\text{C}$ . 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 10077-2 (see also Figure 10). 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 11).

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

- a) any surface whose normal is between  $315^\circ$  and  $45^\circ$  a is left vertical surface
- b) any surface whose normal is between  $45^\circ$  and  $135^\circ$  a is bottom horizontal surface
- c) any surface whose normal is between  $135^\circ$  and  $225^\circ$  a is right vertical surface
- d) any surface whose normal is between  $225^\circ$  and  $315^\circ$  a is top horizontal surface

$$\frac{L_1}{H_1} = \frac{L_1'}{H_1'} \quad t \leq 5mm \quad \text{and} \quad \frac{L_2}{H_2} = \frac{L_2'}{H_2'}$$

Figure 10- Illustration of the treatment of irregularly shaped frame cavities

Figure 11- 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 12):

- heat flow is horizontal if the absolute value of the temperature difference between vertical cavity surfaces is larger than between horizontal the cavity surfaces;
- 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;
- 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.

Figure 12. 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:

$$h_r = \frac{4 \sigma T_{ave}^3}{\frac{1}{\varepsilon_{cold}} + \frac{1}{\varepsilon_{hot}} - 2 + \frac{1}{\frac{1}{2} \left[ \left( 1 + \left( \frac{L_h}{L_v} \right)^2 \right)^{\frac{1}{2}} - \frac{L_h}{L_v} + 1 \right]}} \quad \frac{W}{m^2 K} \quad (84)$$

where:

$$T_{ave} = \frac{T_{cold} + T_{hot}}{2} \quad (84-a)$$

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.,  $L_v/L_h$ ).

Note: For more details, see Reference [16].

## 6.7 Ventilated air cavities and grooves

### 6.7.1 Slightly ventilated cavities and grooves with small cross section

Exposed grooves with small cross sections (see Figure 13) or cavities connected to the external or internal environments by a slit greater than 2 mm but not exceeding 10 mm are to be considered as slightly ventilated air cavities. The equivalent conductivity is twice that of an unventilated air cavity of the same size according to 6.6. For cases where the slit is less than or equal to 2 mm, treat the cavity as completely enclosed in accordance with 6.6.

**Figure 13: 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,  $h_{in}$  and  $h_{out}$ , calculated according to 8, are to be used at the developed internal and external surfaces, respectively.

In the case of a large cavity connected by a single slit and a developed cross sectional length exceeding the width of the slit by a factor of 5 (see Figure 14), the detailed radiation model (see 8.4.2) may be appropriate for determining the radiative portion of the surface heat transfer coefficient.

**Figure 14- Examples for well ventilated cavities and grooves**

## 7 Shading devices

### 7.1 Definitions

#### 7.1.1 Introduction

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

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

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

Note: 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.

Shading devices can be divided into two basic types:

- layer type of shadings, such as screens, curtains and venetian blinds which are located parallel to the pane(s), with intimate thermal-optical contact;
- extra-fenestrial type of shadings, such as awnings and overhangs, which are located less close to the panes. These types of 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). These types of shadings are outside the scope of this standard.

### 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.

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

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

$\tau_{dir,dir}(\lambda_j)$  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, the following properties are required (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, the following properties are required:

$\rho_{dif,dif}(\lambda_j)$  diffuse to diffuse reflectance, and for the absorptance:

$$\alpha_{dir}(\lambda_j) = (1 - \tau_{dir,dir}(\lambda_j) - \rho_{dir,dir}(\lambda_j) - \tau_{dir,dif}(\lambda_j) - \rho_{dir,dif}(\lambda_j)) \quad (85-a)$$

$$\alpha_{dif}(\lambda_j) = (1 - \tau_{dif,dif}(\lambda_j) - \rho_{dif,dif}(\lambda_j)) \quad (85-b)$$

**Resulting amendments to the equations in 5.2 and 5.3:**

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

- 1) Each spectral flow rate equation in 5.2 and 5.3 shall be split into three: a 'dir,dir', 'dir,dif' and 'dif,dif' heat flow, with corresponding transmittance  $\tau$  and reflectance  $\rho$ . In the sum of the spectral heat flows: '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 following clause, on the basis of optical properties and the geometry of the slats.

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

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

Due to the redirection of the radiation the forward transmittance is not necessarily equal to the backward transmittance, as illustrated in Figure 15.

**Figure 15 - 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 16).

**Figure 16 - Slat geometry**

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

#### 7.3.2 Optical properties

##### 7.4.2.1 General

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 ignored.

The procedure is to consider two adjacent slats and to subdivide the slats into five equal parts (see Figure 17).

**Figure 17 - Discretization used in the model**

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

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

##### 7.4.2.2 Equations

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  $\lambda_j$ , ( $\lambda \rightarrow \lambda + \Delta\lambda$ ):

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

$$E_{b,i} = \sum_k \left\{ (\rho_{b,k} + \tau_{f,k}) E_{b,k} F_{b,k \rightarrow b,i} + (\rho_{f,k} + \tau_{b,k}) E_{f,k} F_{f,k \rightarrow b,i} \right\} \quad (86)$$

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.

$$E_{f,0} = J_0(\lambda_j) \quad (86-a)$$

$$E_{b,n} = J_n(\lambda_j) = 0 \quad (86-b)$$

$J_0$  is the radiosity from the external environment (incident solar radiation)

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

### 7.4.2.3 Diffuse-diffuse transmission and reflection

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

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

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

After solving the set of equations, the diffuse/diffuse transmission coefficient is found as the radiation reaching the internal environment  $E_{f,n}$  ( $n=6$ ), divided by the incident solar radiation,  $J_0$  :

$$\tau_{\text{dif,dif}}(\lambda_j) = E_{f,n}(\lambda_j) / J_0(\lambda_j) \quad (87)$$

Similarly, for the diffuse/diffuse reflection coefficient.

$$\rho_{\text{dif,dif}}(\lambda_j) = E_{b,o}(\lambda_j) / J_0(\lambda_j) \quad (88)$$

### 7.4.2.4 Direct-direct transmission and reflection

By straightforward geometric calculation from the angle and aspect ratio of the slats (see Figure 18) the beam radiation which passes the slats without touching can be calculated for given angle of incidence  $\phi$ .

**Figure 18 - Direct-direct transmission**

This part of the transmission is wavelength independent.

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

Consequently, the direct-direct transmittance for incidence angle  $\phi$  amounts:

$$\tau_{\text{dir,dir}}(\phi) = E_{\text{dir,dir}}(\lambda_j, \phi) / J_0(\lambda_j, \phi) \quad (89)$$

for any wavelength  $\lambda_j$ .

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

$$\rho_{\text{dir,dir}}(\phi) = 0 \quad (90)$$

#### 7.4.2.5 Direct-diffuse transmission and reflection:

Firstly: calculate, for the given angle of incidence  $\phi$ , which parts of the shading device  $k$  are directly irradiated by  $J_{f,0}$  (see Figure 19).

**Figure 19 - Directly irradiated parts of the shading device**

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

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

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

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

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

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

$$\tau_{\text{dir,dif}}(\lambda_j, \phi) = E_{f,n}(\lambda_j, \phi) / J_o(\lambda_j, \phi) \quad (91)$$

$$\rho_{\text{dir,dif}}(\lambda_j, \phi) = E_{b,n}(\lambda_j, \phi) / J_o(\lambda_j, \phi) \quad (92)$$

#### 7.4.2.6 Absorptance:

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

$$\alpha_{\text{dir}}(\lambda_j) = (1 - \tau_{\text{dir,dif}}(\lambda_j) - \rho_{\text{dir,dif}}(\lambda_j)) \quad (93)$$

$$\alpha_{\text{dif}}(\lambda_j) = (1 - \tau_{\text{dif,dif}}(\lambda_j) - \rho_{\text{dif,dif}}(\lambda_j)) \quad (93\text{-a})$$

#### 7.4.2.7 Thermal transmittance and reflectance:

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

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

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

### 7.4 Ventilation

#### 7.4.1 General

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

## 7.4.2 Main heat balance equations

### 7.4.2.1 Principle:

Air spaces may be connected to the exterior or interior environment or to other spaces. For a ventilated gap, the heat balance in the gap requires an extra term, the amount of heat supplied to or extracted from the gap air. This implies that it is no longer sufficient to describe, as in 5.3, the conductive/convective heat exchange in a gap as the heat transfer from one surface to the other. It is necessary to make a split between the conductive/convective heat transfer from one surface to the air and from the air to the other surface as illustrated in Figure 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 international standard for measurement of these properties. Until such a testing standard is available, the calculation method in 7.4.2 shall be considered as provisional and is provided for information purposes only.

### 7.4.2.2 Equations: non-vented gap

For the non-vented case (5.3) the heat exchange by conduction/convection across a gap from one layer to the adjacent layer (pane, film or shading device) as given in 5.3.1:  $q_{c,i} = h_{c,i}(T_{f,i} - T_{b,i+1})$ , is split into two parts (see Figure 20), with the mean temperature of the air in the gap as variable:

**Figure 20 - Split convective heat transfer across non-vented gap**

$$q_{c,f,i} = 2h_{c,i}(T_{f,i} - T_{\text{gap},i}) = q_{c,b,i+1} = 2h_{c,i}(T_{\text{gap},i} - T_{b,i+1}) \quad (94)$$

where

$q_{c,f,i}$  is the convective heat transfer from the one surface to the gap, in  $\text{W}/\text{m}^2$ ;

$q_{c,b,i+1}$  is the convective heat transfer from the gap to the other surface, in  $\text{W}/\text{m}^2$ ;

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

$T_{\text{gap},i}$  is the equivalent mean temperature of the air in cavity  $i$ , given Equation (103) below, in  $^{\circ}\text{C}$

$T_{f,i}$  is the temperature of the surface of layer (pane, film or shading)  $i$ , facing cavity  $i$ , see 5.3, in  $^{\circ}\text{C}$

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

### 7.4.2.3 Ventilated gap

In a ventilated gap, due to the air movement, the convective heat exchange coefficient is increased (see Figure 21).

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

$$q_{c,b,i} = h_{cv,i}(T_{b,i} - T_{\text{gap},i}), \text{ and } q_{c,f,i+1} = h_{cv,i}(T_{\text{gap},i} - T_{f,i+1}) \quad (95)$$

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

$$h_{cv,i} = 2h_{c,i} + 4V_i \quad (96)$$

**Figure 21 - Model of mean air- and outlet temperature and main dimensions**

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 Equation. 96, in  $W/(m^2 \cdot K)$

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

$V_i$  is the mean air velocity in the gap, see 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 cavity  $i$ , given by Equation (103) in the following clause, in  $^{\circ}C$

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

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

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

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

$$q_{v,i} = \rho_i \cdot c_p \cdot \varphi_{v,i} (T_{gap,i,in} - T_{gap,i,out}) / (H_i \times L_i) \quad (97)$$

Equations (98) to (103) are formulated in such a manner to satisfy the following energy balance equation:

$$q_{v,i} = q_{cv,b,i} + q_{cv,f,i+1} \quad (97-a)$$

where

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

$\rho_i$  is the density of the air in cavity  $k$  at temperature  $T_{gap,i}$ , in  $kg/m^3$

$c_p$  is the specific heat capacity of air, in  $J/(kg \cdot K)$  (i.c.: 1008);

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

$T_{gap,i,in}$  is the temperature at the inlet of the gap, in  $^{\circ}C$

The heat transfer is normalised to  $1 m^2$  of the aperture area.

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

$T_{\text{gap}_i,\text{out}}$  is the temperature at the outlet of the gap, see Equation (98), in °C

$L_i$  is the length of cavity  $i$ , see Figure 21, in m

$H_i$  is the height of cavity  $i$ , see Figure 21, in m.

#### 7.4.2.4 Heat transfer to internal environment

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

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

$$q_{v,n} = \sum_i \rho_i \cdot c_p \cdot \varphi_{v,i} (T_{\text{gap}_i,\text{out}} - T_{\text{air},n}) / (H_i \cdot L_i) \quad (98)$$

The heat transfer is normalised to 1 m<sup>2</sup> of the aperture area.

where

$\rho_i$  is the density of the air in cavity  $i$  at temperature  $T_{\text{gap}_i}$ , in kg/m<sup>3</sup>

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

$\varphi_{v,i}$  is the air flow rate in cavity  $i$ , see 7.4.4, in m<sup>3</sup>/s

$T_{\text{gap}_i,\text{out}}$  is the temperature of the air at the outlet of the gap from where the air originates, see Equation (102) in °C

$T_{\text{air},n}$  is the indoor air temperature, in °C

$L_i$  is the length of cavity  $i$ , see Figure 21, in m

$H_i$  is the height of cavity  $i$ , see Figure 21, in m.

#### 7.4.3 Temperatures in the cavity

Assuming the mean velocity of the air in the space is known (see 7.4.4), the temperature profile and the heat flow can be calculated by a simple model. Due to the air flow through the space, the air temperature in the space varies with height (see Figure 22).

#### Figure 22 - Air flow in the gap of a window system

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

$$T_{\text{gap}_i}(h) = T_{\text{av},i} - (T_{\text{av},i} - T_{\text{gap}_i,\text{in}}) \cdot e^{-h/H_{0,i}} \quad (99)$$

where

$T_{\text{gap},i}(h)$  is the temperature of the air in gap  $i$  at position  $h$ , in °C

$H_{0,i}$  is the characteristic height (temperature penetration length), see Equation 101, in m

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

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

$$T_{\text{av},i} = (T_{\text{b},i} + T_{\text{f},i+1})/2 \quad (100)$$

where

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

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

The characteristic height of the temperature profile is defined by:

$$H_{0,i} = \frac{\rho_i \cdot c_p \cdot s_i}{2 \cdot h_{\text{cv},i}} \cdot V_i \quad (101)$$

where

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

$\rho_i$  is the density of the air at temperature  $T_{\text{gap},j}$ , in kg/m<sup>3</sup>

$c_p$  is the specific heat capacity, in J/(kg.K) (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 7.4.4, in m/s

$h_{\text{cv},i}$  is the heat transfer coefficient for ventilated cavities, see Equation.(96) in 7.4.2, in W/(m<sup>2</sup>.K).

The leaving air temperature is given by:

$$T_{\text{gap},i,\text{out}} = T_{\text{av},i} - (T_{\text{av},i} - T_{\text{gap},i,\text{in}}) \cdot e^{-H_i/H_{0,i}} \quad (102)$$

where

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

$T_{\text{av},i}$  is the average temperature of the surfaces of layers  $i$  and  $i-1$ , given by Equation (100), in °C

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

$H_{0,i}$  is the characteristic height (temperature penetration length), given by Equation (101), in m

$H_i$  is the height of space  $i$ , in m.

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

$$T_{\text{gap}_i} = \frac{1}{H_i} \int_0^H T_{\text{gap}_i}(h) \cdot dh = T_{\text{av}_i} - \frac{H_{0,i}}{H_i} (T_{\text{gap}_i,\text{out}} - T_{\text{gap}_i,\text{in}}) \quad (103)$$

where

$T_{\text{gap}_i}$  is the equivalent mean temperature of the air in the cavity  $i$ , in °C

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

$T_{\text{gap}_i,\text{in}}$  is the temperature of the incoming air in gap  $i$ , in °C

$T_{\text{av}_i}$  is the average temperature of the surfaces of layers  $i$  and  $i+1$ , given by Equation (100), in °C

$H_{0,i}$  is the characteristic height (temperature penetration length), given by Equation (101), in m

$H_i$  is the height of 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:

$$V_i = \frac{\varphi_{v,i}}{s_i \cdot L_i} \quad (104)$$

where

$V_i$  is the mean velocity of the air flow in cavity  $i$ , in m/s

$s_i$  the width of cavity  $i$ , in m

$L_i$  is the length of the cavity  $i$ , see Figure 21, in m

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

Note:  $\varphi_{v,i}$  is the air flow rate for the whole area, in m<sup>2</sup>.

##### 7.4.4.2 Wind induced ventilation

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

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

For conservative design calculations one may treat the cavity flow as forced convection (see 7.4.4.1), with the value for the air velocity  $v_i$  set to extreme low 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 23).

**Figure 23 - Schematic presentation of the stack effect.**

Note: 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 7.4.4.

The pressure difference results from a temperature difference between the space  $j$  and the connected space  $k$ , which is the exterior air, the interior air or another space. The temperature profile in the spaces is represented by the thermal equivalent temperature (Equation (103)). The driving pressure difference  $\Delta p_T$  may be written approximately as:

$$\Delta p_{T,i,k} = \rho_0 \cdot T_0 \cdot g \cdot H_i \cdot \cos \varphi_i \cdot \frac{(T_{\text{gap},i} - T_{\text{gap},k})}{T_{\text{gap},i} \cdot T_{\text{gap},k}} \quad (105)$$

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 space  $i$  (same as space  $k$ ), in m;

$T_{\text{gap},i}$  is the equivalent (mean) temperature of the air in the space  $i$ , see Equation (103), in K;

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

$\varphi_i$  is the tilt angle of space  $i$  in degrees from vertical;

$\rho_0$  is the density of air at temperature  $T_0$ , in  $\text{kg/m}^3$ ;

$g$  is the gravity constant = 9,81  $\text{m/s}^2$ ;

$T_0$  is reference temperature, (e.g.)  $T_0 = 283 \text{ K}$ .

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 equation):

$$\Delta p_{B,i} = \frac{\rho_i}{2} V_i^2 \quad (106)$$

Steady laminar flow (Hagen-Poiseuille law):

$$\Delta p_{HP,i} = 12 \cdot \mu_i \cdot \frac{H_i}{S_i^2} V_i \quad (107)$$

Pressure loss in the inlet and outlet openings:

$$\Delta p_{Z,i} = \frac{\rho_i}{2} V_i^2 (Z_{\text{in},i} + Z_{\text{out},i}) \quad (108)$$

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 air flow in cavity  $i$ , to be solved with Equation (109), in m/s; (same for  $k$ );
- $\mu_i$  is the viscosity of air at temperature  $T_{gap,j}$ , in Pa·s;
- $\rho_i$  is the density of air at temperature  $T_{gap,j}$ , in kg/m<sup>3</sup>;
- $H_i$  is the height of space  $i$ , in m;
- $s_i$  the width of cavity  $i$ , in m;
- $Z_i$  the pressure loss factors  $Z$  of cavity  $i$ , according to Equation (110).

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

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

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

$$\Delta p_{T,i,k} = \Delta p_{B,i} + \Delta p_{HP,i} + \Delta p_{Z,i} + \Delta p_{Z,k} + \Delta p_{B,k} + \Delta p_{HP,k} \quad (109)$$

where

- $\Delta p_{T,i,k}$  is the driving pressure difference between space  $i$  and space  $k$ , according to Equation (105), in Pa
- $\Delta p_{B,i}$  is the pressure loss  $B$  in space  $i$ , according to Equation (106), in Pa;
- $\Delta p_{HP,i}$  is the pressure loss  $HP$  in space  $i$ , according to Equation (107), in Pa;
- $\Delta p_{Z,i}$  is the pressure loss  $Z$  at the inlet and outlet of space  $i$ , according to Equation (108), in Pa;
- $\Delta p_{Z,k}$  the same as  $\Delta p_{Z,i}$  but for space  $k$ ;
- $\Delta p_{B,k}$  is the pressure loss  $B$  in space  $k$ , according to Equation (106), in Pa;
- $\Delta p_{HP,k}$  is the pressure loss  $HP$  in space  $k$ , according to Equation (107), in Pa.

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 24) according to:

**Figure 24 - Openings in a ventilated gap**

$$Z_{in} = \left( \frac{A_s}{0,6 \cdot A_{eq,in}} - 1 \right)^2 \quad (110-a)$$

and

$$Z_{\text{out}} = \left( \frac{A_s}{0,6 \cdot A_{\text{eq,out}}} - 1 \right)^2 \quad (110)$$

where

$A_{s,i}$  is the cross section of space  $i$ ;  $A_{s,i} = s_i \times L_i$  in  $\text{m}^2$ ;

$s_i$  is the width of cavity  $i$ , in  $\text{m}$ ;

$L_i$  is the length of cavity  $i$ , in  $\text{m}$ ;

$A_{\text{eq,in},i}$  is the equivalent inlet opening area of cavity  $i$ , according to Equation (111 or 112), in  $\text{m}^2$ ;

$A_{\text{eq,out},i}$  is the equivalent outlet opening area of the cavity  $i$ , according to Equation (111 or 112), in  $\text{m}^2$ .

If the temperature  $T_{\text{gap}_i}$  (resp.  $T_{\text{gap}_k}$ ) of the cavity  $i$  respectively  $k$  is higher than the temperature of the connected space  $k$  respectively  $i$ :

$$A_{\text{eq,in}} = A_{\text{bot}} + \frac{1}{2} \cdot \frac{A_{\text{top}}}{A_{\text{bot}} + A_{\text{top}}} (A_l + A_r + A_h) \quad A_{\text{eq,out}} = A_{\text{top}} + \frac{1}{2} \cdot \frac{A_{\text{bot}}}{A_{\text{bot}} + A_{\text{top}}} (A_l + A_r + A_h) \quad (111)$$

Otherwise:

$$A_{\text{eq,out}} = A_{\text{bot}} + \frac{1}{2} \cdot \frac{A_{\text{top}}}{A_{\text{bot}} + A_{\text{top}}} (A_l + A_r + A_h) \quad A_{\text{eq,in}} = A_{\text{top}} + \frac{1}{2} \cdot \frac{A_{\text{bot}}}{A_{\text{bot}} + A_{\text{top}}} (A_l + A_r + A_h) \quad (112)$$

where

$A_s$  is the cross section of the space, in  $\text{m}^2$ ;

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

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

$A_h$  is the total area of the holes in the surface (homogeneously distributed holes), in  $\text{m}^2$ ;

$A_l$  is the area of the left side opening, in  $\text{m}^2$ ;

$A_r$  is the area of the right side opening, in  $\text{m}^2$ ;

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

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

#### 7.4.5 Gas filled cavity with air circulation

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

#### 7.4.6 Air permeability of slat types of shading devices

The air permeability of slat types of shading devices can be described 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 (computational fluid dynamic (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

The thermal transmittance,  $U$  and the total solar energy transmittance,  $g$  shall be calculated using Equations (2 and 13) respectively, applying the amendments to the equations in 5.2 as given in 7.2.

### 8 Boundary conditions

#### 8.1 General

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

The boundary conditions consist of:

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

#### 8.2 Reference boundary conditions

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

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

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

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

##### 8.2.1 Winter conditions

$$T_{in} = 20^\circ\text{C}$$

$$T_{out} = 0^\circ\text{C}$$

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

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

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

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

## 8.2.2 Summer conditions

$$T_{in} = 25^{\circ}\text{C}$$

$$T_{out} = 30^{\circ}\text{C}$$

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

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

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

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

## 8.3 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 internal and external 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.

### 8.3.1 Convective heat transfer coefficient - internal side

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

$$q_{c, in} = h_{c, in} (T_{s, in} - T_{in}) \quad (113)$$

where  $T_{s,in}$  is the temperature of any internal fenestration surface (i.e.,  $T_{b,n}$  or the temperature of the internal frame surface). The convective heat transfer coefficient,  $h_{c,in}$  is determined from heat transfer correlations given in 8.3.1.1.

#### 8.3.1.1 Heat transfer by natural convection

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

$$h_{c,in} = Nu \left( \frac{\lambda}{H} \right) \quad (114)$$

where  $\lambda$  is the thermal conductivity of air.

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

$$Ra_H = \frac{\rho^2 H^3 g C_p |T_{b,n} - T_{in}|}{T_{m,f} \mu \lambda} \quad (115)$$

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

$$T_{m,f} = T_{in} + \frac{1}{4} (T_{b,n} - T_{in}) \quad (116)$$

Correlations to quantify the internal side convective heat transfer coefficient for still air are presented in 8.3.1.2.

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

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

Each of the following Equations (177 to 122) pertains to one particular value, or range, of tilt angle,  $\theta$ . This categorization, as a function of  $\theta$ , is based on the assumption that the internal environment is warmer than the internal glazing surface (i.e.,  $T_{in} > T_{b,n}$ ). If the reverse is true ( $T_{in} < T_{b,n}$ ) it is necessary to seek the appropriate correlation on the basis of the complement of the tilt angle,  $180^\circ - \theta$ , instead of  $\theta$  and to then substitute  $180^\circ - \theta$  instead of  $\theta$  when the calculation is carried out.

a) Windows inclined from  $0^\circ$  to  $15^\circ$  ( $0^\circ \leq \theta < 15^\circ$ )

$$Nu_{in} = 0,13 Ra_H^{1/3} \quad (117)$$

b) Windows inclined from  $15^\circ$  to  $90^\circ$  ( $15^\circ \leq \theta \leq 90^\circ$ )

$$Nu_{in} = 0,56 (Ra_H \sin \theta)^{1/4}; \quad Ra_H \leq Ra_c \quad (118)$$

$$Nu_{in} = 0,13 (Ra_H^{1/3} - Ra_c^{1/3}) + 0,56 (Ra_c \sin \theta)^{1/4}; \quad Ra_H > Ra_c \quad (119)$$

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

c) Windows inclined from  $90^\circ$  to  $179^\circ$  ( $90^\circ < \theta \leq 179^\circ$ )

$$Nu_{in} = 0,56 (Ra_H \sin \theta)^{1/4}; \quad 10^5 \leq Ra_H \sin \theta < 10^{11} \quad (121)$$

d) Windows inclined from  $179^\circ$  to  $180^\circ$  ( $179^\circ < \theta \leq 180^\circ$ )

$$Nu_{in} = 0,58 Ra_H^{1/5}; \quad Ra_H \leq 10^{11} \quad (122)$$

### 8.3.1.2 Forced convection (any tilt)

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

$$h_{c,in} = 4 + 4V_s \quad W/(m^2 \cdot K) \quad (123)$$

where,  $V_s$  is free stream velocity near the fenestration surfaces in m/sec.

### 8.3.2 Convective heat transfer coefficient - external side

The convective heat transfer on the external side primarily occurs by forced convection. For those situations where natural convection occurs, see 8.3.2.4 below. The density of convective heat flow on the external boundary is defined as:

$$q_{c,out} = h_{c,out} (T_{s,out} - T_{out}) \quad (123-a)$$

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

### 8.3.2.1 Different applications

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

- a- Fenestration product comparisons (rating), see 8.3.2.2
- b- Real building fenestration (field situation) fenestration component annual energy analysis, see 8.3.2.3.

### 8.3.2.2 Convective heat transfer coefficient correlation for product comparison or rating

The following relation is to be used for forced convection on the external side of a fenestration system (see ISO 6946):

$$h_{c,out} = 4 + 4V_s \quad \text{W/(m}^2\cdot\text{K)} \quad (124)$$

where:  $V_s$  is in m/sec

### 8.3.2.3 Real building fenestration component annual energy analysis

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

Note: For more information on this procedure, see Kimura [28]. The correlations are based on the building experimental work of Ito, Kimura, and Oka [29].

Calculation procedure:

$$h_{c,out} = 4,7 + 7,6V_s \quad (124-C)$$

where:

If the surface is windward:

$$V_s = 0,25 \cdot V; \quad V > 2: \quad \text{m/s} \quad (125-A)$$

$$V_s = 0,5; \quad V \leq 2: \quad \text{m/s} \quad (125-B)$$

where  $V$  is the wind velocity measured at the tower and  $V_s$  is free stream velocity near the fenestration surfaces.

If the surface is leeward:

$$V_s = 0,3 + 0,05 \cdot V; \quad \text{m/s} \quad (126)$$

To determine whether the surface is windward or leeward, calculate the wind direction,  $\gamma$ , relative to the wall surface (see Figure 25):

$$\gamma = \varepsilon + 180^\circ - \theta \quad (127)$$

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

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

where:

$\theta$  = wind direction (angle measured clockwise from north - see Figure 25.)

$\varepsilon$  = wall azimuth (positive degrees westward from south and negative eastward - see Figure 25)

$n$  = wall normal direction

$N$  = north

$S$  = south

**Figure 25 - Determination of wind direction and wall azimuth**

### 8.3.2.4 Heat transfer by natural convection

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

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

where  $\lambda$  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,  $Ra_H$ .

$$Ra_H = \frac{\rho^2 H^3 g C_p |T_{s,out} - T_{out}|}{T_{m,f} \mu \lambda} \quad (129)$$

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

$$T_{m,f} = T_{out} + \frac{1}{4} (T_{s,out} - T_{out}) \quad (129-A)$$

Correlations to quantify the external side convective heat transfer coefficient are identical to the ones for internal side and are presented in 8.3.1.1. The tilt angle  $\theta$  needs to be replaced by its complement angle  $180^\circ - \theta$ .

Note: For more information, see [27].

## 8.4 Longwave radiation heat transfer

### 8.4.1 Mean radiant temperature

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

$$G_{out} = \sigma T_{rm,out}^4 \quad (130)$$

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

$$T_{\text{rm,out}} = \left( \frac{(F_{\text{ground}} + (1 - f_{\text{clr}})F_{\text{sky}})\sigma T_{\text{out}}^4 + f_{\text{clr}}F_{\text{sky}}J_{\text{sky}}}{\sigma} \right)^{1/4} \quad (131)$$

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

$$F_{\text{ground}} = 1 - F_{\text{sky}} \quad (131\text{-A})$$

$$F_{\text{sky}} = \frac{1 + \cos \theta}{2} \quad (131\text{-B})$$

If the radiosity of the clear sky ( $J_{\text{sky}}$ ) is known, it can be used directly in Equation (131).

Note: Alternatively, if actual sky data is unavailable, the model from Swinbank [36] can be used.

$$J_{\text{sky}} = \varepsilon_{\text{sky}} \sigma T_{\text{out}}^4 \quad (131\text{-C})$$

$$\varepsilon_{\text{sky}} = \frac{R_{\text{sky}}}{\sigma T_{\text{out}}^4} \quad (131\text{-D})$$

$$R_{\text{sky}} = 5,31 \times 10^{-13} T^6 \quad (131\text{-E})$$

Internal irradiance is defined as:

$$G_{\text{in}} = \sigma T_{\text{rm,in}}^4 \quad (132)$$

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

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

$$G_{\text{in}} = \sigma T_{\text{in}}^4 \quad (133)$$

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

## 8.4.2 Detailed radiation heat transfer calculation

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

The net radiation heat transfer on fenestration boundaries,  $q_r$ , of non-planar products shall be calculated using procedure outlined in 8.4.2.1 or the alternative procedure given in 8.4.2.2.

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

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

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

$$q_{r,i} = \varepsilon_i \sigma T_i^4 - \varepsilon_i G_i \quad (134)$$

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

$$G_i = \sum_j^N F_{i-j} J_j \quad (135)$$

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

$$J_j = \varepsilon_j \sigma T_j^4 + \rho_j G_j \quad (136)$$

Assuming all surfaces are grey:  $\rho_j = 1 - \varepsilon_j$ . Substituting  $\rho_j$  and  $G_j$  and using subscript i for convenience, Equation (136) becomes:

$$J_i = \varepsilon_i \sigma T_i^4 + (1 - \varepsilon_i) \sum_{j=1}^N F_{i-j} J_j \quad (137)$$

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.

Temperatures are calculated from the solution to the conduction problem given by Equation (70), while net radiation heat flow rate (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  $T_i|^{k}$ .

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

This procedure is repeated until the following condition is satisfied:

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

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.

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

Note: For more information on the norm or root mean square values of the temperature vector, see [30]. Also, for more information on the cross-string rule, see [26].

#### 8.4.2.2 Simplified three dimensional radiation method

The thermal transmittance of non-planer 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 8.4.2.1.

The emissivity of the internal surface is reduced by the factor given in Equation (143). The ratio of internal-side surface area to internal-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,  $A_t$ .

Note: More detail about this method is provided in Wright [31].

$$F_{\text{rad}} = \frac{1}{1 + \varepsilon_g \left( \frac{A_s}{A_p} - 1 \right)} \quad (143)$$

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

$$\varepsilon_{\text{red}} = F_{\text{rad}} \varepsilon_{\text{b,n}} \quad (144)$$

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

### 8.4.3 Simplified radiation heat transfer calculation

#### 8.4.3.1 Internal surfaces

All internal 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 5.3.1, the following equation can be used for simplified radiation heat transfer calculations on both glazing and frame surfaces:

$$q_{\text{r,in}} = h_{\text{r,in}} (T_{\text{s,in}} - T_{\text{rm,in}}) \quad (145)$$

where

$$h_{\text{r,in}} = \frac{\varepsilon_{\text{s,in}} \sigma (T_{\text{s,in}}^4 - T_{\text{rm,in}}^4)}{T_{\text{s,in}} - T_{\text{rm,in}}} \quad (146)$$

#### 8.4.3.2 External surfaces

All external surfaces are denoted here by subscript  $s_{\text{out}}$ , including frame surfaces. As with internal surfaces, external 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.

$$q_{\text{r,out}} = h_{\text{r,out}} (T_{\text{s,out}} - T_{\text{rm,out}}) \quad (147)$$

where

$$h_{r,out} = \frac{\varepsilon_{s,out} \sigma (T_{s,out}^4 - T_{rm,out}^4)}{T_{s,out} - T_{rm,out}}$$

### 8.5 Combined convective and radiative heat transfer

$$q = h(T_s - T_n) \quad (148)$$

$$\text{where } h = h_r + h_c \quad (148-A)$$

$T_s$  = surface temperatures

$T_n$  = environmental temperatures

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

$$h_{\text{adjusted}} = \frac{A_{\text{real}}}{A_{\text{approximated}}} h \quad (149)$$

### 8.6 Prescribed density of heat flow rate

The frame/wall interface shall be treated as adiabatic. Also see ISO10077 -2.

To calculate total solar energy transmittance of frames,  $g_f$  in analogy to Equation (13), the following boundary condition shall be used:

$$q_{in} = \alpha I_s \quad (150)$$

where:

$\alpha$  = is the absorptance of frame surfaces

## Annex A (informative)

### Solution technique for the multi-layer solar optical model

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

**Figure A.1 - Analysis of solar flow rate 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,  $\rho_{f,n+1}(\lambda) = 0$  and  $\tau_{n+1}(\lambda) = 0$ .

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

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

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

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

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

$$\frac{I_{i+1}^-(\lambda)}{I_i^-(\lambda)} = t_i(\lambda) = \frac{\tau_i(\lambda)}{1 - \rho_{b,i}(\lambda)r_{i+1}(\lambda)} \quad (\text{A.4})$$

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 density of heat flow rate reflected from the glazing system to the external side ( $I_i^+(\lambda) = r_i(\lambda)I_i^-(\lambda)$ ) and then marching toward the internal side from  $i=2$  to  $i=1$  calculating the remaining density of heat flow rates ( $I_i^-(\lambda) = t_{i-1}(\lambda)I_{i-1}^-(\lambda)$  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:

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

and the portion transmitted to the conditioned space is,

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

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

The approach used here can also be applied to the analysis of shading devices (i.e., clause 7) although the resulting equations will be more complex due to the conversion of beam solar radiation to diffuse solar radiation.

## Annex B (normative)

### Thermophysical fill gas property values

The following tables present linear equation coefficients with which the conductivity, viscosity and specific heat capacity 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 Touloukian [33].

The heat transfer calculations are based on the assumption that the fill gas is not an emitting/absorbing gas. Since SF<sub>6</sub> violates this condition it has not been presented in these tables.

**Table B.1 - Thermal conductivity**

Gas	Coefficient a	Coefficient b	$\lambda$ @ 0 °C	$\lambda$ @ 10 °C
	W/(m·K)	W/(m·K <sup>2</sup> )	W/(m·K)	W/(m·K)
Air	$2,873 \times 10^{-3}$	$7,760 \times 10^{-5}$	0,0241	0,0249
Argon	$2,285 \times 10^{-3}$	$5,149 \times 10^{-5}$	0,0163	0,0168
Krypton	$9,443 \times 10^{-4}$	$2,826 \times 10^{-5}$	0,0087	0,0090
Xenon	$4,538 \times 10^{-4}$	$1,723 \times 10^{-5}$	0,0052	0,0053

where  $\lambda = a + b \cdot T(K)$ , [W/m·K]

**Table B.2 – Viscosity**

Gas	Coefficient a	Coefficient b	$\mu$ @ 0 °C	$\mu$ @ 10 °C
	N·s/m <sup>2</sup>	N·s/(m <sup>2</sup> ·K)	N·s/m <sup>2</sup>	N·s/m <sup>2</sup>
Air	$3,723 \times 10^{-6}$	$4,94 \times 10^{-8}$	$1,722 \times 10^{-5}$	$1,771 \times 10^{-5}$
Argon	$3,379 \times 10^{-6}$	$6,451 \times 10^{-8}$	$2,100 \times 10^{-5}$	$2,165 \times 10^{-5}$
Krypton	$2,213 \times 10^{-6}$	$7,777 \times 10^{-8}$	$2,346 \times 10^{-5}$	$2,423 \times 10^{-5}$
Xenon	$1,069 \times 10^{-6}$	$7,414 \times 10^{-8}$	$2,132 \times 10^{-5}$	$2,206 \times 10^{-5}$

where  $\mu = a + b \cdot T(K)$  [kg/m·s]

Table B.3 - Specific heat at constant pressure

Gas	Coefficient a	Coefficient b	$c_p$ @ 0 °C	$c_p$ @ 10 °C
	J/(kg·K)	J/(kg·K <sup>2</sup> )	J/(kg·K)	J/(kg·K)
Air	1002,7370	$1,2324 \times 10^{-2}$	1006,1034	1006,2266
Argon	521,9285	0	521,9285	521,9285
Krypton	248,0907	0	248,0917	248,0917
Xenon	158,3397	0	158,3397	158,3397

where  $c_p = a + b \cdot T(K)$  J/(kg·K)

Table B.4 - Molecular weights

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

## Annex C (Informative)

### Examples of calculated values for optical properties of slat type of shading devices

This annex provides selected examples of calculated optical properties of a slat type of shading device as function of the slat properties and geometry, for different angles of incident solar radiation, applying the procedure given in 7.3.

#### Specifications (Figures C.1 and C.2):

##### Geometry:

slat distance: 12 mm

slat length: 16 mm

slat angle: 45° and/or 80°

##### Solar incidence angle:

0° and 60° from horizontal

##### Slat types:

opaque or translucent

white, pastel and/or dark coloured (outside and/or inside surfaces)

The optical properties of the materials of the different types of slats are given in table C.1.

Table C.1 – Optical properties of the slat materials

Slat	Long wave (IR) transmittance $\tau$	Long wave (IR) emissivity $\epsilon_h$	Solar transmittance $\tau$	Solar reflectance $\rho$
Opaque, white	0,0	0,90	0,0	0,70
Opaque, pastel	0,0	0,90	0,0	0,55
Opaque, dark	0,0	0,90	0,0	0,40
Translucent, white	0,40	0,55	0,40	0,50

Figure C.1 – Slat geometry and sun position

Figure C.2 – Slats angle 45° and 80°

#### Results:

The equations of 7.3 are used to calculate the optical properties of the shading device.

The optical properties are illustrated in Figures C.3 and C.4 and the results of the calculations are presented in Table C.2.

**Figure C.3 – Illustration of the direct-direct transmittance of the shading device**

**Figure C.4 – Illustration of the direct-diffuse and diffuse-diffuse transmittance and reflectance of the shading device**

Note that the optical properties for radiation from the inside are in general not the same as the optical properties for radiation from the outside. The optical properties for radiation from the inside are needed because of (multiple) reflections at the successive layers in the transparent system.

**Table C.2 - Selected examples of calculated optical properties of a slat type of shading device**

Product	Description									
A45	Venetian blinds with opaque white slats, slat angle 45°									
B45	Venetian blinds with opaque pastel colour slats, slats angle 45°									
C45	Venetian blinds with opaque light/dark slats, slats angle 45°									
C80	Venetian blinds with opaque light/dark slats, slats angle 80°									
D45	Venetian blinds with translucent white slats, slats angle 45°									
Property	A45	A45	B45	B45	C45	C45	C80	C80	D45	D45
<i>Geometry:</i>										
distance between slats (mm)	12		12		12		12		12	
slat length (mm)	16		16		16		16		16	
Slat angle, from horizontal (°)	45		45		45		80		45	
<i>Optical material properties:</i>										
IR transmittance slat	0,0		0,0		0,0		0,0		0,40	
IR emissivity slat external side	0,90		0,90		0,90		0,90		0,55	
IR emissivity slat internal side	0,90		0,90		0,90		0,90		0,55	
solar transmittance slat	0,0		0,0		0,0		0,0		0,40	
solar reflectance slat external side	0,70		0,55		0,70		0,70		0,50	
solar reflectance slat internal side	0,70		0,55		0,40		0,40		0,50	

Solar incidence angle from horizontal (°)	0	60	0	60	0	60	0	60	0	60
<i>Results:</i>										
Solar transmittance external side dir, dir ( $T_{dir,dir}$ ) <sub>external</sub>	0,057	0,0	0,057	0,000	0,057	0,000	0,000	0,000	0,057	0,000
Solar transmittance internal side dir, dir ( $T_{dir,dir}$ ) <sub>internal</sub>	0,057	0,310	0,057	0,310	0,057	0,310	0,000	0,088	0,057	0,310
Solar transmittance external side dir, dif ( $T_{dir,dif}$ ) <sub>external</sub>	0,141	0,073	0,090	0,047	0,096	0,051	0,012	0,005	0,373	0,277
Solar transmittance internal side dir, dif ( $T_{dir,dif}$ ) <sub>internal</sub>	0,141	0,288	0,090	0,216	0,076	0,271	0,011	0,027	0,373	0,306
Solar reflectance external side dir, dif ( $\rho_{dir,dif}$ ) <sub>external</sub>	0,394	0,558	0,295	0,430	0,371	0,544	0,622	0,678	0,418	0,567
Solar reflectance internal side dir, dif ( $\rho_{dir,dif}$ ) <sub>internal</sub>	0,394	0,103	0,295	0,066	0,216	0,070	0,356	0,273	0,418	0,273
Solar transmittance external side dif, dif ( $T_{dif,dif}$ ) <sub>external</sub>	0,332		0,294		0,291		0,038		0,495	
Solar transmittance internal side dif, dif ( $T_{dif,dif}$ ) <sub>internal</sub>	0,332		0,294		0,291		0,038		0,495	
Solar reflectance external side dif, dif ( $\rho_{dif,dif}$ ) <sub>external</sub>	0,345		0,260		0,323		0,604		0,380	
Solar reflectance internal side dif, dif ( $\rho_{dif,dif}$ ) <sub>internal</sub>	0,345		0,260		0,193		0,345		0,380	
IR transmittance external side ( $T_{IR}$ ) <sub>external</sub>	0,227		0,227		0,227		0,0245		0,385	
IR transmittance internal side ( $T_{IR}$ ) <sub>internal</sub>	0,227		0,227		0,227		0,0245		0,385	
IR emissivity external side ( $\epsilon$ ) <sub>external</sub>	0,729		0,729		0,729		0,89		0,536	
IR emissivity internal side ( $\epsilon$ ) <sub>internal</sub>	0,729		0,729		0,729		0,89		0,536	

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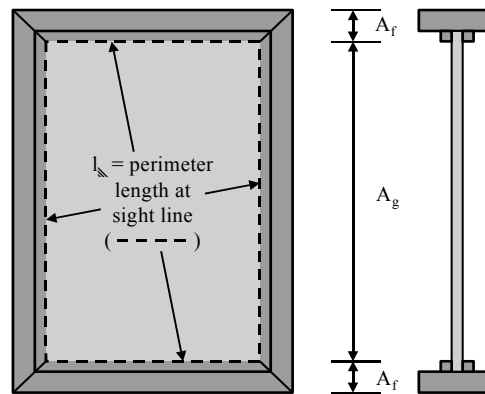
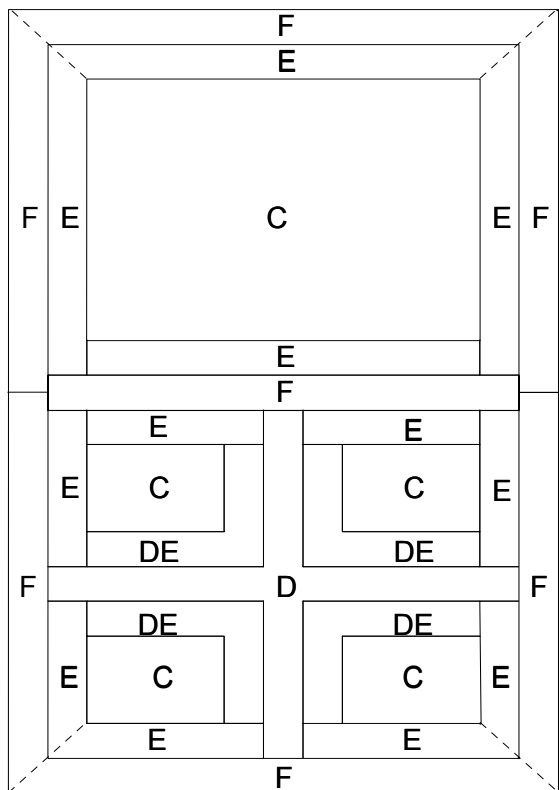


Figure 1 A- Schematic diagram showing the window projected areas and vision perimeter



Legend

- C Center-of-Glass
- E Edge-of-Glass
- F Frame
- D Divider
- DE Divider-Edge

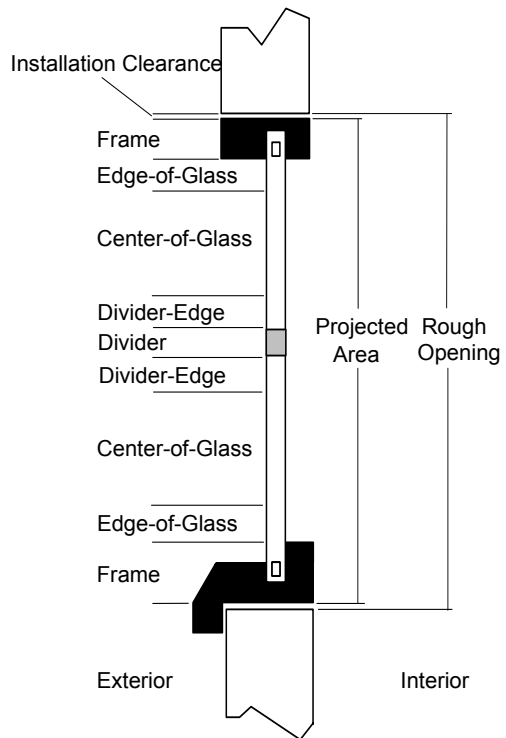


Figure 1-B: Center-of-glass, edge-of-glass, divider, divider-edge, and frame areas for a typical fenestration product.

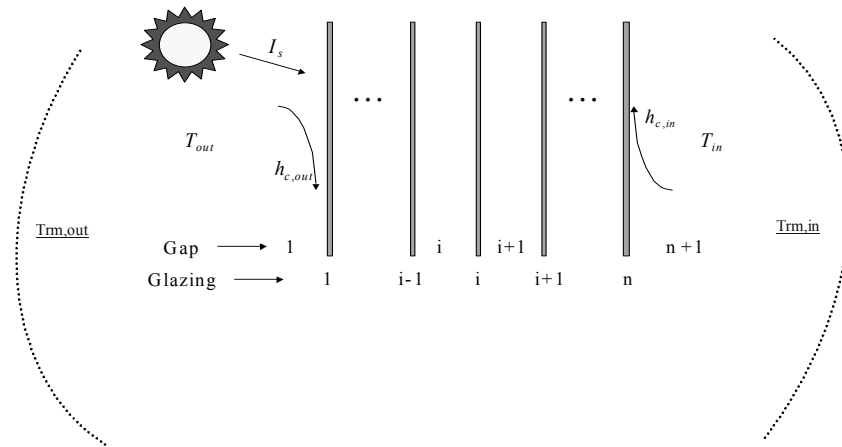


Figure 2 - Numbering system for glazing system layers

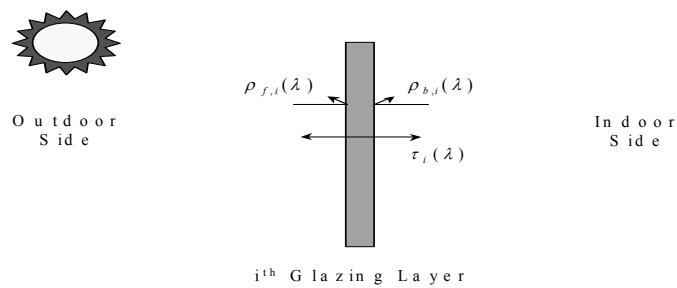


Figure 3 - Out-door and internal spectral transmittance of a glazing layer

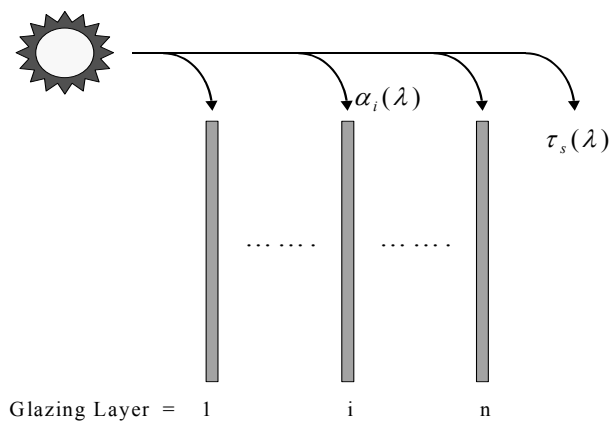


Figure 4 - Absorptance of the  $i^{\text{th}}$  glazing layer and solar spectral transmittance

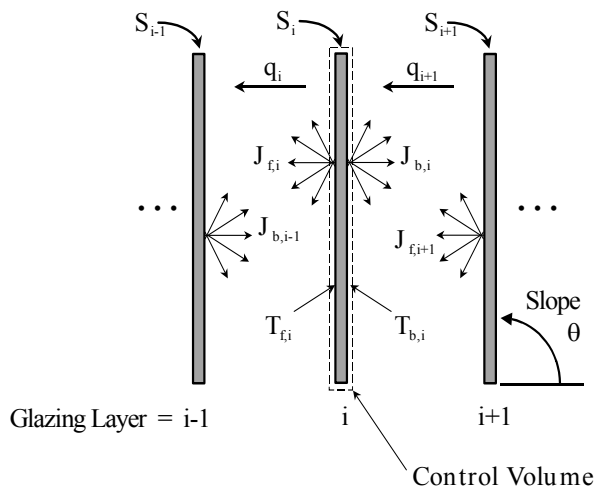


Figure 5 - Energy balance on the  $i^{\text{th}}$  glazing layer

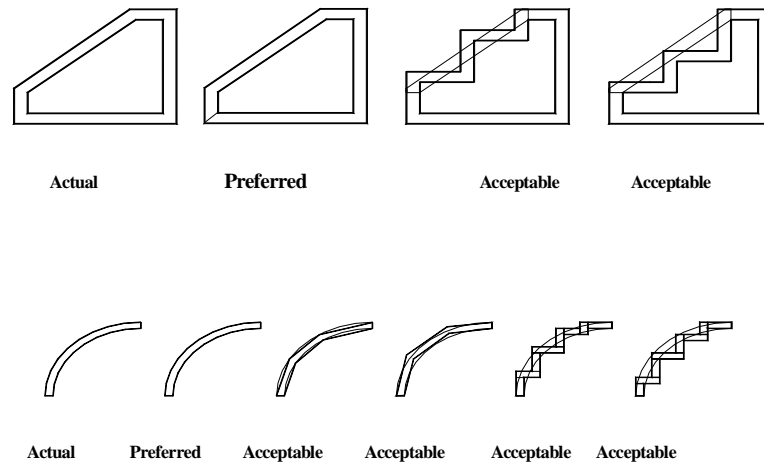


Figure 6 - Examples of possible approximations of the actual cross-section

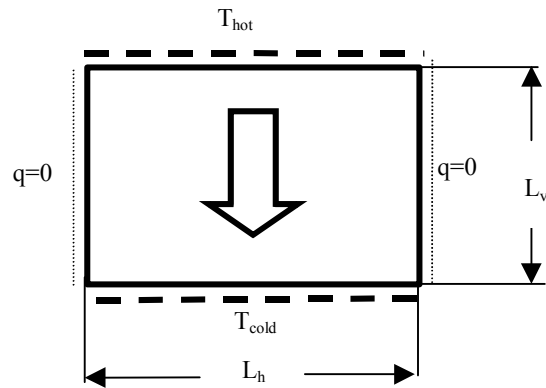


Figure 7 – Illustration of rectangular frame cavity downward flow direction

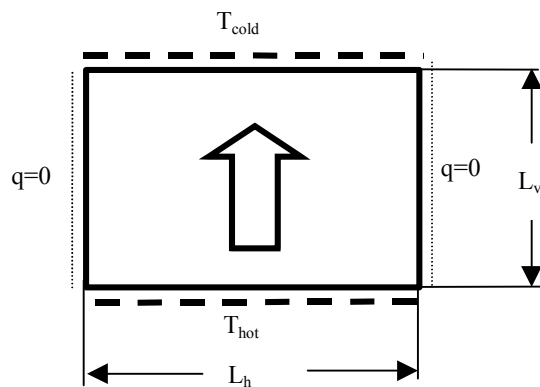


Figure 8 – Illustration of rectangular frame cavity upward flow direction

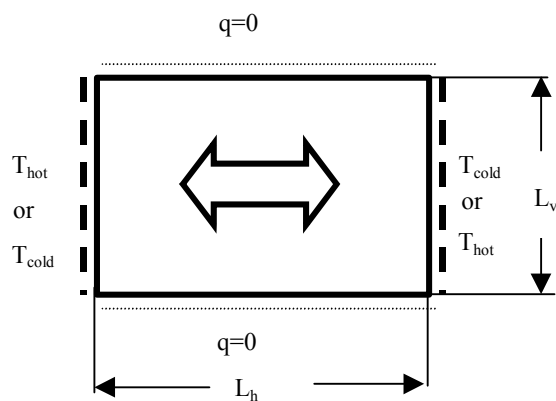
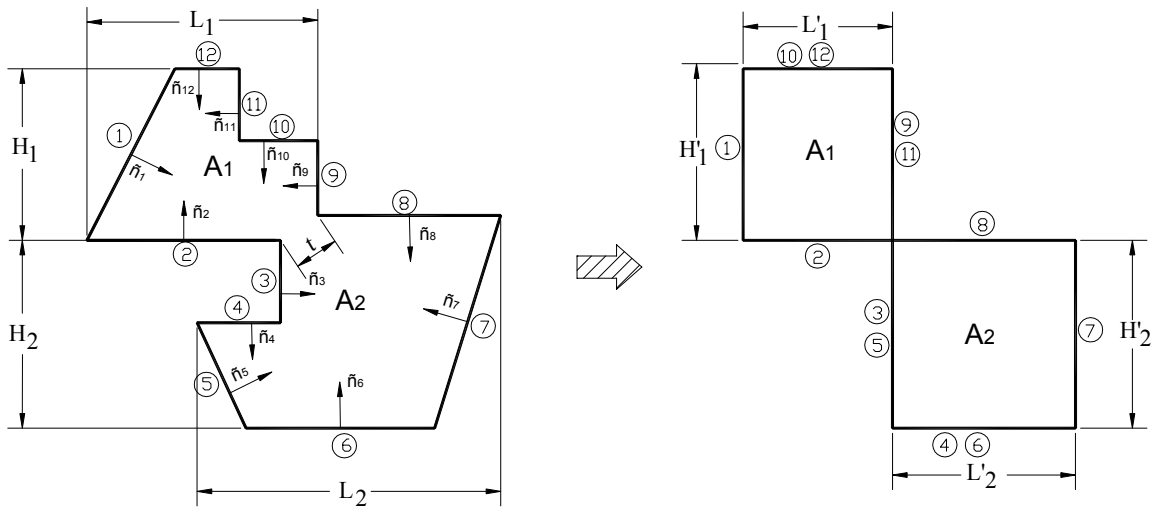


Figure 9 – Illustration of rectangular frame cavity horizontal flow direction



$$\frac{L_1}{H_1} = \frac{L'_1}{H'_1} \quad t \leq 5mm \quad \text{and} \quad \frac{L_2}{H_2} = \frac{L'_2}{H'_2}$$

Figure 10. Illustration of the treatment of irregularly shaped frame cavities

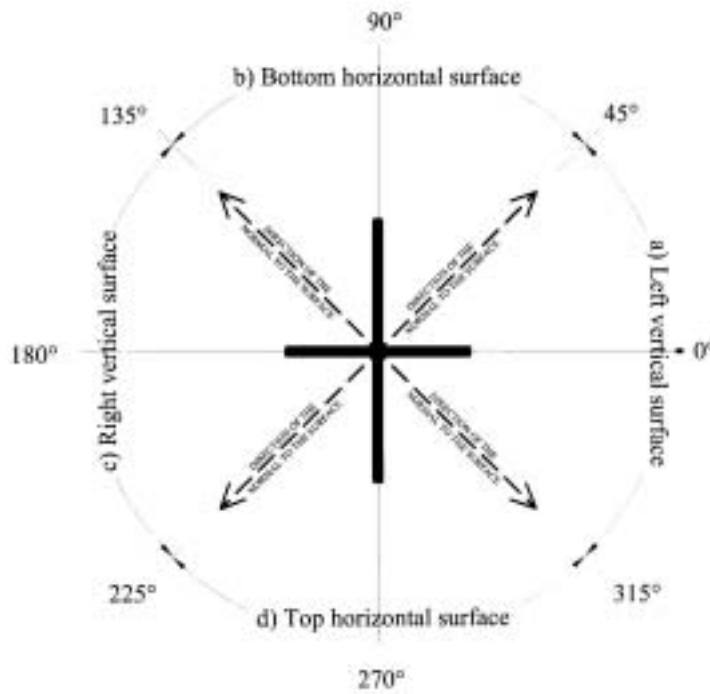
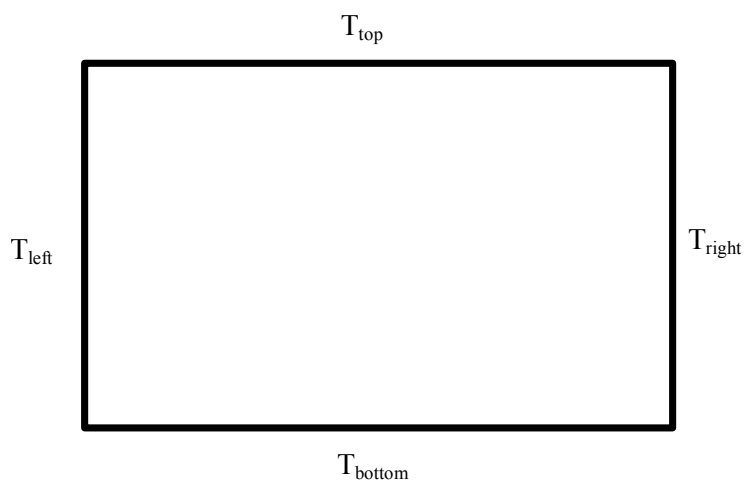



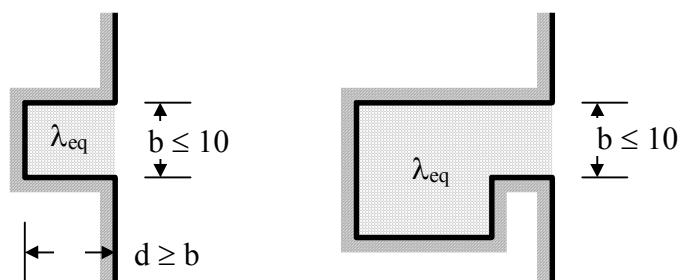


Figure 11. Illustration of how to select surface orientation for frame cavities-dashed lines indicate direction of the normal to surface with cut of angles at 45°, 135°, 225° and 315°.



- a)  $|T_{right} - T_{left}| \geq |T_{top} - T_{bottom}|$   heat flow is horizontal;
- b)  $|T_{right} - T_{left}| < |T_{top} - T_{bottom}|$  and  $T_{top} < T_{bottom}$   heat flow is vertical, heat flow up;
- c)  $|T_{right} - T_{left}| < |T_{top} - T_{bottom}|$  and  $T_{top} > T_{bottom}$   heat flow is vertical, heat flow down.

**Figure 12. Illustration of how to select heat flow direction**



**Figure 13: Examples for slightly ventilated cavities and grooves with small cross section**

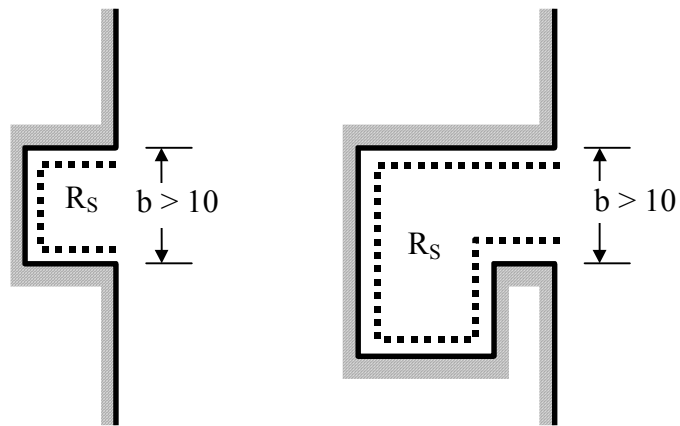


Figure 14: Examples for well ventilated cavities and grooves

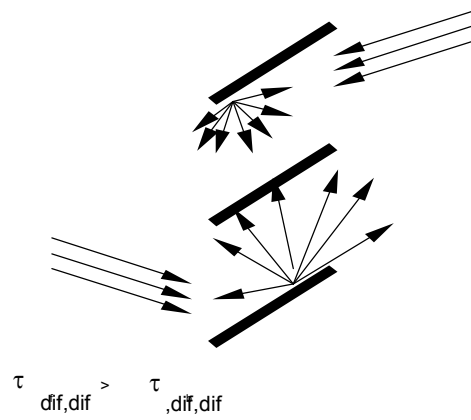


Figure 15 - Illustration of different values for forward and backward solar transmittance (slats with different colour at both surfaces)

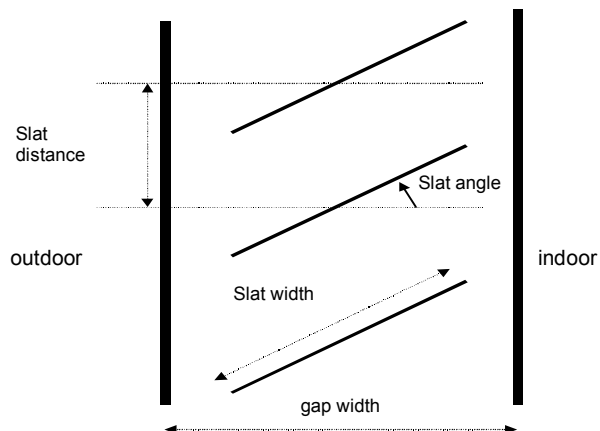


Figure 16 - Slat geometry

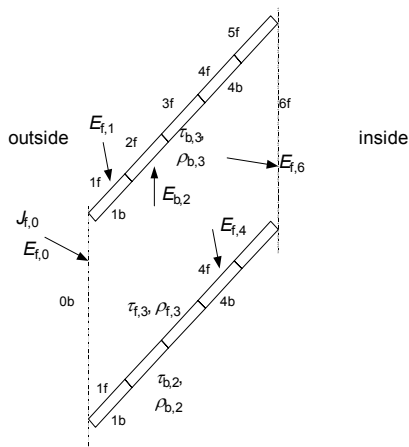


Figure 17 - Discretization used in the model

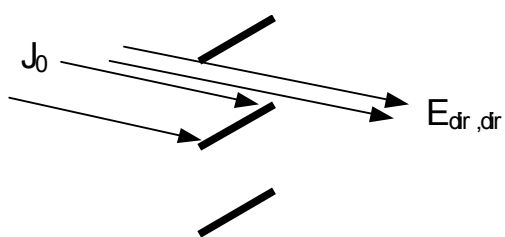


Figure 18 - Direct-direct transmission

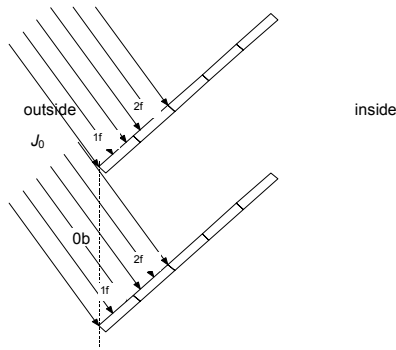


Figure 19 - Directly irradiated parts of the shading device

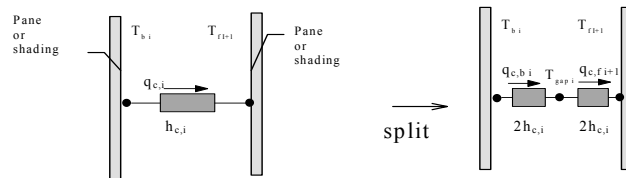


Figure 20 - Split convective heat transfer across gap; non-vented gap

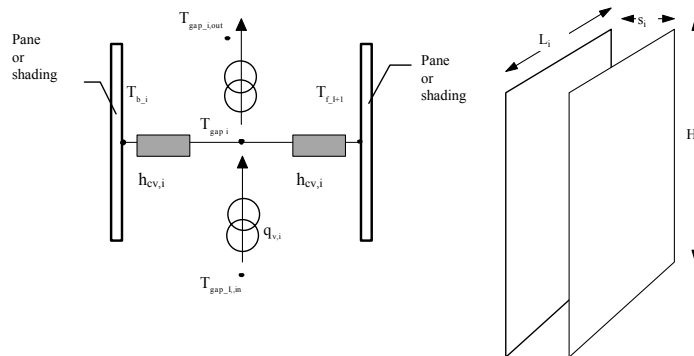


Figure 21 - Model of mean air- and outlet temperature and main dimensions

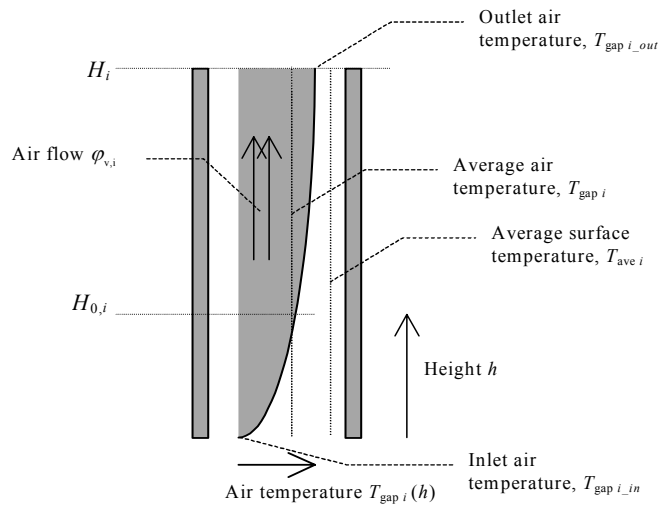


Figure 22 - Air flow in the gap of a window system

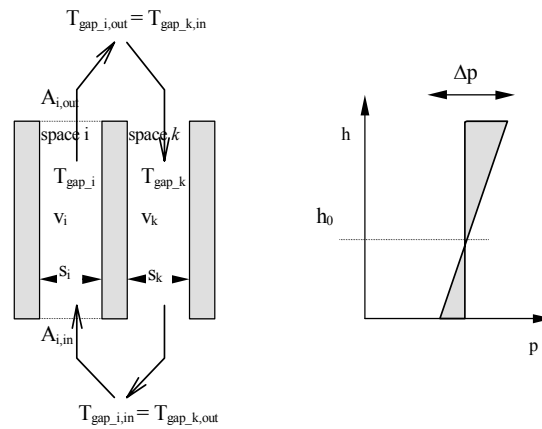


Figure 23 - Schematic presentation of the stack effect.

The height of the neutral zone  $h_0$  depends on the flow resistances of the inlet and outlet openings

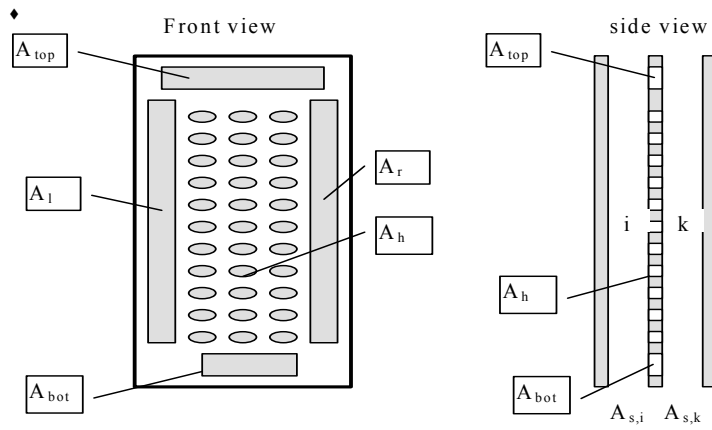


Figure 24 - Openings in a ventilated gap

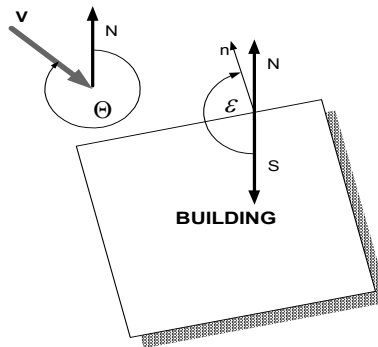


Figure 25 - Determination of wind direction and wall azimuth

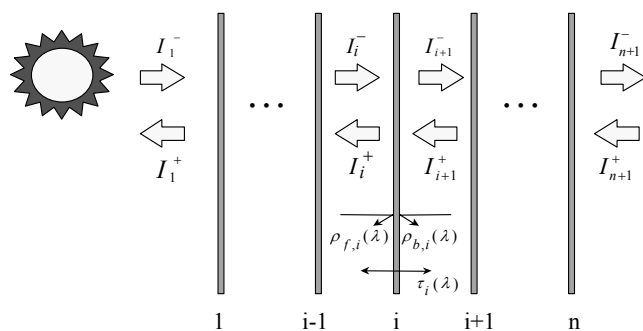


Figure A.1 - Analysis of solar density of heat flow rate in multi-layer glazing system

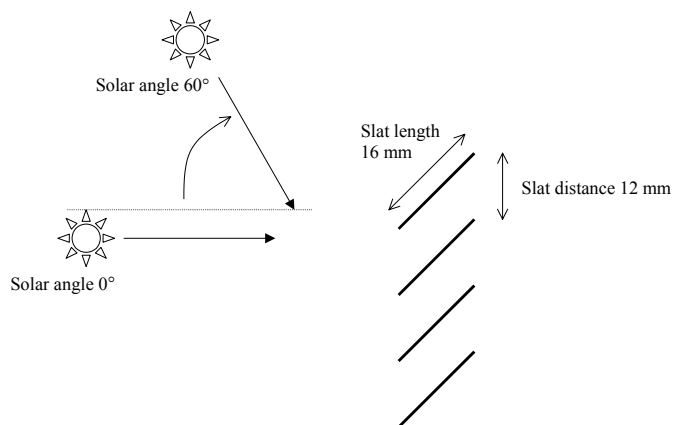


Figure C.1 – Slat geometry and sun position

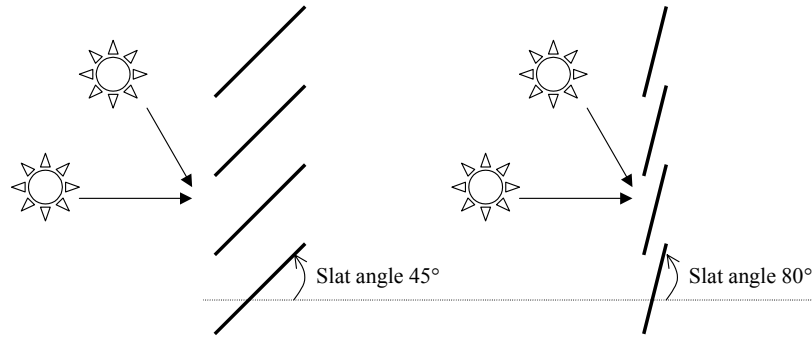


Figure C.2 – Slats angle 45° and 80°

Examples of direct-direct transmittance  $\tau_{dir,dir}$

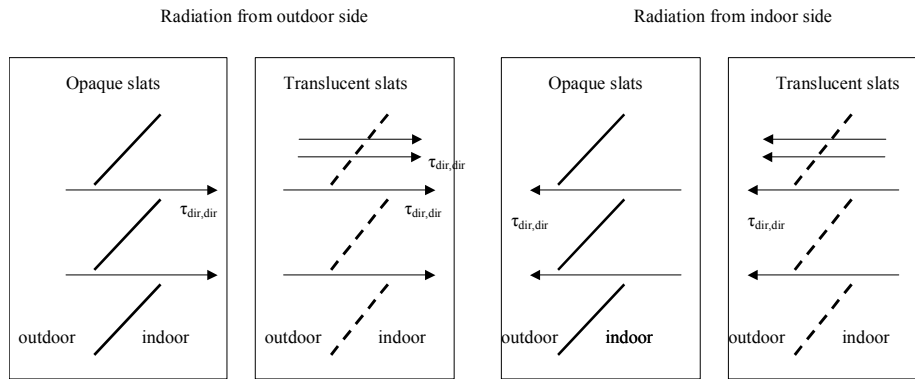


Figure C.3 – Illustration of the direct-direct transmittance of the shading device

Examples of direct-diffuse transmittance  $\tau_{dir,dif}$  and reflectance  $\rho_{dir,dif}$

Examples of diffuse-diffuse transmittance  $\tau_{dif,dif}$  and reflectance  $\rho_{dif,dif}$

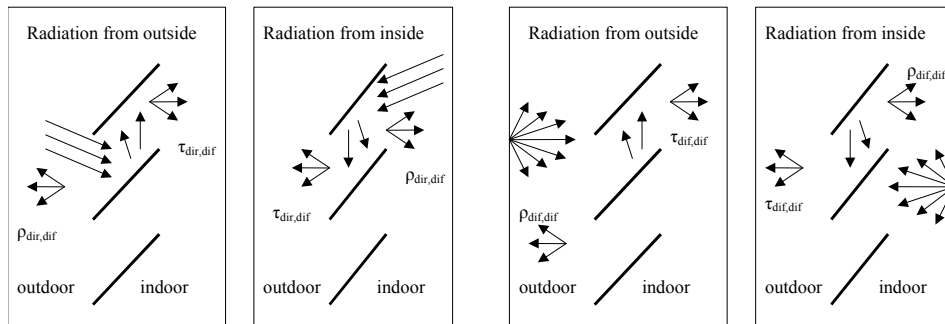


Figure C.4 – Illustration of the direct-diffuse and diffuse-diffuse transmittance and reflectance of the shading device