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Publisher's version / Version de l'éditeur:

<https://doi.org/10.1080/19401490903046785>

Journal of Building Performance Simulation, 2, 3, pp. 189-207, 2009-09-01

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NRCC-50866 #

Laouadi, A.

Sept. 1, 2009

A version of this document is published in / Une version de ce document se trouve dans:
Building Performance Simulation, 2, (3), pp. 189-207, September 01, 2009
DOI: <http://dx.doi.org/10.1080/19401490903046785>

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THERMAL PERFORMANCE MODELING OF COMPLEX FENESTRATION SYSTEMS

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ABSTRACT

Complex fenestration systems (CFS) have become standard elements in façade design of high performance buildings. They include, for example, shadings devices to control illumination, solar heat gains, glare, and view-out, and photovoltaic elements imbedded in glazing layers to produce electrical energy on site. However, current methodologies to evaluate the thermal performance of CFS are limited to a few products and types. This paper develops a general methodology to compute the thermal performance of CFS. The methodology assumes each system layer as porous with calculated effective radiation and thermal properties. A new thermal penetration length model was developed to account for the effects of porous layers on the convective film coefficients of adjacent gas spaces, and applied to various types of shading devices. This methodology is validated using the available measurement and CFD simulation results for the U-factor of double-glazed windows with between-pane and internal blinds.

KEYWORDS

Fenestration; window; glazing; shading; complex fenestration system

NOMONCLATURE

AB	: solar absorptance (decimals)
C	: constant, equation (16) (dimensionless)
c	: specific heat (J/kg.K)
D	: constant, equation (17) (dimensionless)
E	: black body emissive power (W/m^2)
F	: form factor (dimensionless)
Fr	: fullness ratio of drapes (dimensionless)
g	: gravitational coefficient (m/s^2)
H	: characteristic plate height (m)
h	: convective film coefficient of gas space ($\text{W/m}^2\text{K}$)
k	: thermal conductivity (W/mK)
L	: thickness (m)
m	: constant, equation (16) (dimensionless)
Nu	: Nusselt number (dimensionless)
q	: radiative flux density (W/m^2)
Ra	: Rayleigh number (dimensionless)
RF	: solar reflectance (decimals)
s	: spacing of slats, or length of drapery pleats (m)
SR	: surface area ratio (decimals)
T	: temperature (K)
TR	: solar transmittance (decimals)
v	: gas velocity (m/s)
w	: width of cavity, or slat (m)
z	: height of cavity (m)

Δx : thickness of a nodal control volume (m)

SUBSCRIPTS

a : gas space, or average

b : back surface, or blind layer

beam : beam solar radiation

c : convection

d : diffuse

dif : diffuse solar radiation

e : emission

eff : effective

eq : equivalent

f : front surface

i : layer node index, or incident

j : layer index

m : material

o : outgoing

r : radiation

s : screen layer

sol : solar

sm : slat material

GREEK SYMBOLS

α : slat angle (radians), or gas diffusivity (m^2/s), or absorption coefficient per unit length (m^{-1})

β : inclination angle (radians), or expansion coefficient of gas (K^{-1})

δ : reduced thermal penetration length (m)

- ε : emissivity (decimals), or emissivity coefficient per unit length (m^{-1})
- γ : thermal penetration length of a porous layer (m)
- κ : fibre density (dimensionless)
- η_{PV} : photovoltaic cell efficiency (decimals)
- ρ : density (kg/m^3), or long wave reflectance (decimals)
- τ : long wave transmittance (decimals)
- ν : gas viscosity (m^2/s)
- ϕ : openness factor (decimals)
- ω : porosity (decimals)

1 INTRODUCTION

Complex fenestration systems have become standard components in high performance building designs. CFS include, for example, varying types of shading devices integrated between glazing layers, or attached to the interior or exterior glazed façades to control natural illumination, solar heat gains, and view out, and photovoltaic elements imbedded in glazing layers to produce electrical energy on site. In some building applications with high indoor moisture contents such as restaurants and swimming pools, electrical-conductor transparent thin films are applied to indoor window surfaces to control moisture condensation and maintain view-out, and provide local secondary space heating. Although there has been significant advancement in the performance evaluation of CFS, predictions of the thermal performance of CFS remain a challenge to be addressed, particularly for systems with shading devices.

In the past decades, there has been significant work devoted to the evaluation of the optical, daylighting and energy performance of CFS incorporating shading devices. The ISO 15099 standard (ISO, 2003) presents a validated model to compute the optical and long-wave radiation characteristics of slat-type blinds. Other blind related work carried out further validation studies and developed improvement to the ISO blind model to account for the slat thickness, curvature and specularity (Rosenfeld et al., 2000; Breitenbach et al., 2001; Chantrasrisalai and Fisher, 2006; Collins and Jiang, 2008; Yohoda and Wright, 2005; 2004b). The IEA Task 27 (Kohl, 2006) carried out a comprehensive assessment of the solar optical and thermal performance of several product types of interior and exterior blinds and roller shades through measurement and computer simulation using the ISO 15099 standard and simple models. The IEA Task 34/43 (Loutzenhiser et al., 2007) carried out empirical validations of building-energy simulation tools for daylighting performance and thermal loads of interior and exterior Venetian blinds and shading screens. Most of the tested simulation tools used simple models for the prediction of the optical and thermal performance of the tested shading

devices. Recent work under the ASHRAE project (RP 1311) carried out experimental studies and developed semi-empirical optical models for several types of shading devices, including drapes, roller blinds and insect screens (Kotey et al., 2009a,b).

However, predictions of the thermal performance of CFS (e.g., U-factor) are at the early stage, particularly those related to the convection flows in open gas cavities adjacent to porous layers. Porous layers are transparent to thermal radiation, and permeable to fluid flow through their structure. Work under this area can be divided into four main approaches, namely: simple models, empirical-based models, CFD-based models, and a mix of models.

Simple thermal models of CFS are based on the thermal-resistance approach developed for simple fenestration systems, made up of essentially plain opaque glazing (Yahoda and Wright, 2004a; Wright, 2008). A one-dimensional conduction heat transfer model is used for glazing layers coupled with radiation and convection models at the layer boundary surfaces.

Convection models use existing correlations for the convective film coefficients in enclosed gas cavities or around tilted flat surfaces. Complex layers such as shading devices are treated as individual layers with effective radiation properties (emissivity and infra-red transmittance), but with assumed uniform layer temperature (no thermal resistance of shading layer). These simple models have been implemented in currently available fenestration tools such as WINDOW (LBNL, 2008) and WIS (WinDat, 2008). Recent research showed, however, that these simple models were not accurate for slat-type blinds (Yahoda and Wright, 2004a), and for diathermanous – thermally transparent- layers (Collins and Wright, 2006). Furthermore, these models cannot be applied to CFS layers with significant radiation absorption and emission within their media, such as, for example, shading devices and transparent or translucent insulation. In addition, these models do not account for energy production (e.g., electrical resistance heating) and conversion (e.g., photovoltaic) within layer media, which are getting more popular in today's building design.

Given the limitations of the simple thermal-resistance models, extensive work has been devoted to the experimental performance evaluation of CFS, and investigation of fluid flows in cavities of CFS incorporating shading devices. Earlier work by Grasso and Buchanan (1979) and Grasso et al. (1990) conducted measurement using the guarded hot plate method of the U-factor of single glazed windows combined with several product types of interior draperies and roller shadings. They found that sealing the cavities between shadings and window had significant effects in reducing the U-factor of windows. Fang (2000, 2001) conducted measurement using the hotbox method of the U-factor of single and double-glazed windows with interior curtains and Venetian blinds. Empirical relationships derived from the experiments were also developed for the U-factor of similar window and shade configurations. Garnet (1999) and Huang (2005) conducted laboratory measurement using the guarded heated plate apparatus of the U-factor of regular and low-e double-glazed windows with between pane metallic Venetian blinds. They found that the slat-tip-to-glazing spacing and slat angle positions had significant effect on the U-value of the window and blind system. The thermal bridging effect of the metallic blinds reached its maximum when the slat angles were horizontal, where the U-factor was higher than when slats were closed. To understand how the fluid flow behaves in cavities between window glazing and shadings, Machin et al. (1998), Collins et al. (2002b), and Naylor and Lai (2007) conducted interferometric experiments to visualize the temperature and air flow patterns in double glazed windows with internal and between pane metallic Venetian blinds. The local Nusselt number along the window height was measured for varying slat-tip-to-window glazing spacing. Cellular flows within cavities between slats were observed, particularly for low slat angle positions near the horizontal. Gas mass transfer through the blind layer was also observed. The studies concluded that the thermal conductivity of blinds and radiation heat transfer had strong effects on the local Nusselt number.

Parallel to the above experimental investigations, CFD computer simulations have been used for varying purposes: (1) to investigate the flow and

temperature patterns in gas cavities between glazing and shading layers, (2) to validate the simulation results with the measurement, and (3) to develop useful correlations for the convective film coefficient in cavities between glazing and shading layers. Laminar, two-dimensional flows were generally considered in the investigated work. Convective flows with and without direct radiation coupling in windows with internal Venetian blinds were addressed by several researchers, including Ye et al. (1999), Phillips et al. (2001), Collins et al. (2002a), Collins (2004), Shahid and Naylor (2005), and Naylor et al. (2006). The maximum effect of blinds on the heat transfer through the window and blind system was found when the blinds were closed, and placed near to the window glazing. The obtained simulation results for the temperature and flow patterns compared favourably with the available measurement of similar window and blind configurations. Correlations for the average Nusselt number of the window-blind cavity were also developed for various slat angles. Naylor and Collins (2005), Avedissian (2006), and Avedissian and Naylor (2007) addressed windows with between-pane Venetian blinds. Naylor and Collins (2005) employed a full CFD model (with direct radiation coupling) to simulate the experimental configuration of Garnet (1999). Overall, good comparison results were obtained. Avedissian (2006) employed a convection-only CFD model, and covered a broad range of flow governing parameters, including low and high values of the blind thermal conductivity, and glazing-to-blind spacing. Correlations for the average Nusselt number of the window cavity were developed for various slat angle positions. When the radiative heat transfer was post-coupled with the convective heat transfer, the obtained result for the U-factor of window compared favourably with the experimental results of Huang (2005). Avedissian and Naylor (2007), on the other hand, employed a full CFD model, and obtained comparable results with those of Avedissian (2007) and Huang (2005).

The CFD-based correlations for the convective film coefficient of cavities between glazing and shading layers have been successfully used in the thermal-resistance models as stated previously. Shahid (2003), Shahid and

Nayler (2005), and Nayler et al. (2006) applied this approach to windows with internal blinds, and Naylor and Collins (2005) and Avedissian and Naylor (2007) to windows with between-pane blinds. They found that the predictions of the U-factor of windows using such improved models compared well with the full CFD simulation, and deviated from the available measurement by about 10%. Recently, Wright et al. (2008) developed a simplified model to compute the film coefficient of a double glazed window cavity with between-pane metallic blinds. The blinds divide the window cavity into two sub cavities. Wright et al. (2008) used the existing cavity correlations (similar to those included in the ISO 15099 standard) to compute the film coefficients of the sub cavities based on a modified sub cavity width. The latter, which is larger than the true sub cavity width (equal to slat tip-to-glazing spacing), was found proportional to the slat width and cosine angle. The proportionality constant was determined by comparing the model predictions of U-factor with the measurement results of Huang (2005). The proposed model yielded exceptional results for the window cavity spacings of 17.78 mm and 25.4 mm. However, the model failed to accurately-predict the U-value of the window with the larger cavity spacing of 40 mm. Despite this drawback, the model of Wright et al. (2008) indicates an important conclusion that existing correlations for cavity film coefficients may be safely used to predict the thermal performance of window and shade systems without resorting to the time-consuming and computationally intensive CFD simulation. Furthermore, the model of Wright et al. (2008) allows more flexibility to address other combinations of shading types and tilted window configurations. Future work should address improvement to the model of Wright et al. (2008) for general applications.

It should be noted that the current advanced empirical and CFD-based thermal models are limited to single or double glazed windows with specific metallic blinds. They do not account for other physical and thermal properties of blinds such as slat spacing and thermal conductivity. Furthermore, they cannot be applied to other types of shading devices such as drapes, or to other window tilt configurations.

This paper addresses some of the previously identified issues in the thermal modelling of CFS. The aim is to develop a general methodology to compute the thermal performance of complex fenestration systems for implementation in fenestration computer programs.

2 OBJECTIVES

The specific objectives are:

- To revisit the current models of the heat transfer mechanisms through fenestration layers, and include the peculiarities of CFS such as imbedded elements for energy production or conversion, permeable layers for mass and radiation, and open gas spaces.
- To develop models to compute the effective radiation and thermal properties of permeable layers.
- To develop models to compute the convection film coefficients of gas spaces adjacent to porous layers.
- To validate and assess the methodology by comparing its predictions with the available measurement and simulation data.

3 OVERVIEW OF EXISTING FENESTRATION MODELS

Fenestration systems have undergone significant innovation, from simple systems, made up of essentially plain glass layers and enclosed sealed gas cavities, to far more complex systems. The latter may embody complex glazing layers with applied films, and shading devices with complex geometries. Heat transfer in such complex systems has, therefore, become more complex to handle. Glazing layers may be permeable to mass and thermal radiation, and cavity spaces may be sealed, or open so that cavity gas can move from one space to another through permeable layers, or through deliberate cavity openings. The flow in the cavity space may be forced (e.g., wind-driven), or natural, driven by buoyancy forces and temperature gradients (stack effect). The current methodology to compute the thermal performance of complex fenestration systems has, unfortunately,

inherited the methodology and assumptions of the simple fenestration systems. As outlined in the ISO 15099 standard (ISO, 2003), the current methodology treats glazing layers as thermally opaque layers, except in the thermal radiation energy balance where layers may be thermally transparent. Thermal radiation transport within a complex layer medium is not accounted for. Models for the convective heat transfer use existing correlations for heat transfer in sealed gas cavities, or around tilted flat surfaces. Correlations for the convective heat transfer in open cavities with permeable (porous) layers are not available. The ISO 15099 standard provides, however, a simple algorithm to calculate the convection film coefficient of a ventilated open cavity based on the film coefficient of a sealed cavity (h_{cavity}) and the average gas velocity (v_j) in cavity space. For buoyancy-driven flows, the gas velocity is calculated based on the piston flow model and the total pressure difference between the gas space and its connected environment. The film coefficient of a gas space (j) is, thus, given by the following relation:

$$h_{c,j} = 2 \cdot h_{\text{cavity},j} + 4 \cdot v_j \quad (1)$$

The ISO 15099 standard also suggests using equation (1) for forced cavity flows with known velocities.

Equation (1) was developed particularly for one-way cavity flows based on empirical and numerical data (Dick and Oversloot, 2003). Recent research also found that equation (1) yielded comparable results to the experimental and numerical data for mechanically ventilated double skin facades (Corgnati et al., 2007). However, care should be exercised when applying the equation for cavities where two-way flows might occur (Manz et al., 2004).

Regarding the estimation of the cavity film coefficient (h_{cavity}), there are several proposed methods, particularly, for glazing cavities encompassing slat-type blinds. The ISO 15099 model assumes the blinds divide the glazing cavity into two sub cavities and the cavity film coefficient is evaluated based on the width of the sub cavity (half of the glazing cavity if the blinds are placed in the center). This model does not account for the slat angle position and

geometrical characteristics. Other models use the slat-tip-to-glazing width to evaluate the cavity film coefficient. However, when compared with the experimental and CFD simulation data, these two models were found to underestimate or overestimate the thermal transmittance (U-factor) of windows (Yahoda and Wright, 2004a; Wright et al., 2008). Wright et al. (2008) recently proposed an improved model, termed the reduced-slat-width (RSW) model, in which the slat-tip to glazing width is calculated based on a 70% reduced slat width ($0.7w$). This model does not take into account the spacing and thermal conductivity of slats.

It should be added that there are no guidelines on how to apply the ISO ventilated cavity model for sealed glazing cavities encompassing porous shadings or slat-type blinds, despite the fact that most of the experimental work listed in this paper showed that there was a significant air flow through blind layers, particularly at the top and bottom edge sections, even when blinds are fully (but not ideally) closed (Naylor and Lai, 2007).

4 THE PROPOSED METHODOLOGY

The following sections present the details of a new methodology to compute the thermal performance of complex fenestration systems. The methodology starts by laying down the underlying equations for the heat transfer through complex glazing layers, and develops appropriate methods to handle the convective and radiative heat transfer in open cavities with layers permeable to radiation and mass.

4.1 ASSUMPTIONS

The following assumptions are considered.

- Each glazing layer is assumed solid and porous with calculated effective radiation and thermal properties to account for any convection and radiation effect in a layer medium.
- Heat transfer through a glazing layer medium is by one-dimensional conduction.

4.2 LAYER HEAT TRANSFER

Consider a multi-layer fenestration system consisting of (N) layers as shown in Figure 1. Layer 1 faces the exterior environment and layer (N) faces the interior environment. Each layer (j) is surrounded by gaseous spaces at its boundary surfaces. The layer exchanges heat with the adjacent environments by convection to the gaseous spaces and radiation to the adjacent layers. Heat is transported through the layer by conduction, and radiation. The transient energy balance on an elemental control volume at node (i) within a layer (j) is expressed by the following relation (Siegel and Howell, 2002):

$$\rho_{j,i} c_{j,i} \frac{\partial T_{j,i}}{\partial t} = \frac{\partial}{\partial x} \left(k_{j,i} \frac{\partial T_j}{\partial x} - q_{r,j,i} \right) + q_{sol,j,i} + q_{0,j,i} \quad (2)$$

where:

- $c_{j,i}$: effective specific heat at node i of layer j (J/kg.K)
- $k_{j,i}$: effective thermal conductivity at node i of layer j (W/mK)
- $T_{j,i}$: temperature at node i of layer j (K)
- $q_{r,j,i}$: net radiation flux per unit surface area at node i of layer j (W/m²)
- $q_{sol,j,i}$: absorbed solar radiation per unit volume at node i of layer j (W/m³)
- $q_{0,j,i}$: heat generation per unit volume at node i of layer j (W/m³)
- $\rho_{j,i}$: effective density at node i of layer j (kg/m³)

If the front (facing the exterior environment) and back (facing the interior environment) surfaces of layer j are both subject to the beam and diffuse incident solar radiation, the absorbed solar heat per unit volume at node i is expressed as follows:

$$q_{sol,j,i} = q_{f,sol,j,i} + q_{b,sol,j,i} \quad (3)$$

with:

$$q_{f,sol,j,i} = \alpha_{f,j,i} \cdot q_{f,beam} + \alpha_{f,d,j,i} \cdot q_{f,dif} - SR_{PV} \cdot \eta_{PV,f,j,i} \cdot TR_{f,1:j-1} (q_{f,beam} + q_{f,dif}) \quad (4)$$

$$q_{b,sol,j,i} = \alpha_{b,j,i} \cdot q_{b,beam} + \alpha_{b,d,j,i} \cdot q_{b,dif} - SR_{PV} \cdot \eta_{PV,b,j,i} \cdot TR_{b,N:j+1} (q_{b,beam} + q_{b,dif}) \quad (5)$$

where:

$q_{f,beam}$: beam solar radiation flux density incident on the front surface of fenestration system (W/m²)

$q_{b,beam}$: beam solar radiation flux density incident on the back surface of fenestration system (W/m²)

$q_{f,dif}$: diffuse solar radiation flux density incident on the front surface of fenestration system (W/m²)

$q_{b,dif}$: diffuse solar radiation flux density incident on the back surface of fenestration system (W/m²)

SR_{PV} : ratio of photovoltaic surface area to layer surface area (decimals)

$TR_{f,1:j-1}$: front solar transmittance of layer stack 1 to j-1 (decimals)

$TR_{b,N:j+1}$: back solar transmittance of layer stack N to j+1 (decimals)

$\alpha_{f,j,i}$: absorption coefficient per unit length of layer j at node i for the front incident beam solar radiation (m⁻¹)

$\alpha_{b,j,i}$: absorption coefficient per unit length of layer j at node i for the back incident beam solar radiation (m⁻¹)

$\alpha_{f,d,j,i}$: absorption coefficient per unit length of layer j at node i for the front incident diffuse solar radiation (m⁻¹)

$\alpha_{b,d,j,i}$: absorption coefficient per unit length of layer j at node i for the back incident diffuse solar radiation (m⁻¹)

$\eta_{PV,f,j,i}$: photovoltaic cell efficiency of layer j at node i to convert the front incident solar energy into electric energy (decimals)

$\eta_{PV,b,j,i}$: photovoltaic cell efficiency of layer j at node i to convert the back incident solar energy into electric energy (decimals)

The intermediate stack transmittance values in equations (4) and (5) are usually not standard outputs of a fenestration calculation program. However, they can be determined based on the standard outputs for the layer absorptances and fenestration system reflectances as follows:

$$TR_{f,1:j-1} = 1 - RF_f - \sum_{k=1}^{j-1} AB_{f,k} \quad (6)$$

$$TR_{b,N:j+1} = 1 - RF_b - \sum_{k=N}^{j+1} AB_{b,k} \quad (7)$$

where:

$AB_{f,j}$: front solar absorptance of layer j (decimals)

$AB_{b,j}$: back solar absorptance of layer j (decimals)

RF_f : front solar reflectance of fenestration system (decimals)

RF_b : back solar reflectance of fenestration system (decimals)

The nodal solar absorption coefficients ($\alpha_{j,i}$) may be calculated based on the layer solar absorptance (AB_j). In the absence of a photovoltaic energy conversion, one may assume that the solar radiation is uniformly absorbed along the layer thickness (L_j). In this case, the nodal solar absorptance coefficients are equal to the layer solar absorptance divided by its thickness ($\alpha_{j,i} = AB_j/L_j$). However, if the photovoltaic energy conversion is present, one may assume that the solar energy is absorbed only at the photovoltaic cell layers. In this case, the nodal absorptance coefficients are zero, except at the photovoltaic cells nodes. The nodal solar absorptance coefficient at the photovoltaic cell node is then equal to the layer solar absorptance divided by the total thickness of the nodal control volumes enclosing the photovoltaic layers ($\alpha_{j,i} = AB_j/\sum \Delta X_{PV,cell}$). For layers opaque to the solar radiation, the solar radiation is absorbed at the boundary surfaces. The nodal absorptance

coefficients are therefore zero, except at the boundary nodes where they are equal to the layer solar absorptance divided by the thickness ($\Delta x_{j,i}$) of a hypothetical control volume adjacent to the boundary surface ($\alpha_{j,i} = AB_j/\Delta x_{j,i}$).

The net radiation flux (q_r) is a complex quantity to calculate. It relates to the emission and absorption of thermal radiation within a layer medium. One detailed approach is to formulate the net radiation flux based on the radiation intensity distribution within a layer medium. This requires solving additional partial differential equations for the radiation transport in the medium (Siegel and Howell, 2002). This approach is possible, in particular, for media whose radiation properties (emission, absorption and scattering coefficients) and geometrical details are known. Another approach, which is well suited to complex fenestration systems, is to formulate the nodal net radiation flux in terms of the effective emissivity and absorptivity coefficients of a layer medium, its nodal emissive power and the incident radiation from all directions. In this approach, the layer medium is assigned nodal effective emissivity and absorptivity coefficients, which may be pre-calculated based on a detailed radiation model for the layer medium. The effective emissivity coefficients indicate the portion of the black body energy emitted per unit length, which exits from the layer boundary surfaces. Similarly, the nodal effective absorptivity coefficients indicate the portion of absorbed energy per unit length of radiation incident on the layer boundary surfaces. By virtue of the Kirchhoff's law (Siegel and Howell, 2002), the effective absorptivity and emissivity coefficients are equal. The effective radiation properties depend on the medium geometrical details, and may vary along the layer thickness. In this approach, the gradient of the radiation flux density is expressed as follows:

$$-\frac{\partial q_{r,j,i}}{\partial x} = \varepsilon_{f,j,i} \cdot q_{rff,j} + \varepsilon_{b,j,i} \cdot q_{rbi,j} - (\varepsilon_{f,j,i} + \varepsilon_{b,j,i}) \cdot E_{j,i} \quad (8)$$

where:

$E_{j,i}$: black body emissive power at node i of layer j (W/m^2)

$q_{rfi,j}$: radiation flux density incident on the front surface of layer j given by equation (13) (W/m²)

$q_{rbi,j}$: radiation flux density incident on the back surface of layer j given by equation (13) (W/m²)

$\varepsilon_{f,j,i}$: front effective emissivity coefficient per unit length at node i of layer j (m⁻¹)

$\varepsilon_{b,j,i}$: back effective emissivity coefficient per unit length at node i of layer j (m⁻¹)

The nodal emissivity (absorptivity) coefficients may be calculated based on the effective emissivities of a layer ($\varepsilon_{eff,j}$), which themselves may be available for layers with known geometrical details such as slat-type blind layers (ISO, 2003; Yahoda et al., 2004b), or screen type layers or drapes (see equations 21 to 23, and 32 to 33). For layers, which are fully transparent to thermal radiation, the emission and absorption of thermal radiation may occur along the layer thickness. In this case, the nodal emissivity coefficients may be assumed equal to the layer effective emissivity divided by its thickness ($\varepsilon_{j,i} = \varepsilon_{eff,j} / L_j$). For layers, which are weakly transparent or opaque to thermal radiation, the emission and absorption may be assumed to occur at the layer boundary surfaces. The nodal emissivity coefficients are, therefore, zero, except at the boundary nodes where they are equal to the layer effective emissivity divided by the thickness ($\Delta x_{j,i}$) of a hypothetical control volume adjacent to the boundary surface ($\varepsilon_{j,i} = \varepsilon_{eff,j} / \Delta x_{j,i}$).

To solve equation (2), the boundary node temperatures at $i = 1$ and $j = m$ have to be given. Applying the energy balance to a control volume encompassing the boundary nodes, one may obtain the following equations:

$$(\rho c)_{j,1} \frac{\partial T_{j,1}}{\partial t} \Delta x_1 = k_{j,1} \frac{\partial T_{j,1}}{\partial x} + h_{c,j-1} (T_{a,j-1} - T_{j,1}) + \left(q_{sol,j,1} + q_{0,j,1} - \frac{\partial q_{r,j,1}}{\partial x} \right) \Delta x_1 \quad (9)$$

$$(\rho c)_{j,m} \frac{\partial T_{j,m}}{\partial t} \Delta x_m = -k_{j,m} \frac{\partial T_{j,m}}{\partial x} + h_{c,j} (T_{a,j} - T_{j,m}) + \left(q_{sol,j,m} + q_{0,j,m} - \frac{\partial q_{r,j,m}}{\partial x} \right) \Delta x_m \quad (10)$$

where:

$h_{c,j-1}$: convection film coefficient of gas space (j) (W/m²K)

$h_{c,j}$: convection film coefficient of gas space (j-1) (W/m²K)

$T_{a,j-1}$: average temperature of gas space (j-1) (K)

$T_{a,j}$: average temperature of gas space (j) (K)

$\Delta x_{j,1}$: thickness of a control volume at boundary node i=1 of layer j (m)

$\Delta x_{j,m}$: thickness of a control volume at boundary node i=m of layer j (m)

Equations (2), (9) and (10) are non-linear and cannot be solved analytically. A suitable numerical method should therefore be used.

4.3 RADIATIVE HEAT TRANSFER

As previously mentioned in the assumptions, each layer is characterized by the effective radiation properties for transmittance (τ_{eff}), reflectance (ρ_{eff}), and emissivity or absorptance (ε_{eff}). Layers are assumed long enough so that the radiative contribution from the edge spacer sections can be neglected. This assumption holds for most fenestration systems, particularly when the calculation of the centre-glazing thermal conductance (or U-factor) is concerned. However, this assumption may break down in certain applications of double-skin facade systems where the order of magnitude of the thickness of the enclosed air spaces may be comparable to the layer lengths. In this case, a three surface (two layers and edge spacer section) radiative transfer should be considered together with a proper heat balance of the spacer sections.

Consider an isolated layer (j) as shown in Figure 1. The outgoing radiative fluxes from the front and back surfaces of layer are expressed as follows:

$$q_{rto,j} = \tau_{eff,b,j} \cdot q_{rbi,j} + \rho_{eff,f,j} q_{rfi,j} + q_{ef,j} \quad (11)$$

$$q_{rbo,j} = \tau_{eff,f,j} \cdot q_{rfi,j} + \rho_{eff,b,j} q_{rbi,j} + q_{eb,j} \quad (12)$$

The incident radiative fluxes on the front and back surfaces of layer are expressed as follows:

$$q_{rbi,j} = q_{rfo,j+1}, \quad q_{rfi,j} = q_{rbo,j-1} \quad (13)$$

where:

$q_{ef,j}$: emitted radiative flux density of layer (j) exiting from its front surface (W/m²)

$q_{eb,j}$: emitted radiative flux density of layer (j) exiting from its back surface (W/m²)

$q_{rfo,j}$: radiative flux density exiting from the front surface of layer (j) (W/m²)

$q_{rbo,j}$: radiative flux density exiting from the back surface of layer (j) (W/m²)

$q_{rfi,j}$: radiative flux density incident on the front surface of layer (j) (W/m²)

$q_{rbi,j}$: radiative flux density incident on the back surface of layer (j) (W/m²)

Note that the environment radiative fluxes $q_{rbo,j=0}$ and $q_{rfo,j=N+1}$ denote the incident fluxes from the exterior and interior environments, respectively.

The emitted radiative fluxes of the layer include the emission of radiation from the bulk layer medium. In a discrete nodal representation of the layer medium, the emission radiative fluxes are given by the following relations:

$$q_{ef,j} = \int_{L_j} \epsilon_{f,j}(\mathbf{x}) \cdot \mathbf{E}_j(\mathbf{x}) \cdot d\mathbf{x} = \sum_{i=1}^{i=m} \epsilon_{f,j,i} \cdot \Delta \mathbf{x}_i \cdot \mathbf{E}_{j,i} \quad (14)$$

$$q_{eb,j} = \int_{L_j} \epsilon_{b,j}(\mathbf{x}) \cdot \mathbf{E}_j(\mathbf{x}) \cdot d\mathbf{x} = \sum_{i=1}^{i=m} \epsilon_{b,j,i} \cdot \Delta \mathbf{x}_i \cdot \mathbf{E}_{j,i} \quad (15)$$

Equations (11) and (12) may be solved using a sequential iterative procedure, in which equation (11) is solved first for all layers (j = 1 to N), followed by equation (12). The process is repeated until convergence is reached.

Convergence is declared if the relative change in the nodal temperatures is less than a tolerance value.

4.4 CONVECTIVE HEAT TRANSFER IN OPEN GAS CAVITIES

A gas cavity between two consecutive layers is said to be open if one of the layers is permeable to gas, or/and one of the edges is deliberately un-sealed at the top, bottom or side surfaces. As mentioned before, the convective heat transfer in open cavities with layers permeable to radiation and mass is a complex phenomenon to predict as the convective effect of the gas cavity may extend to the media of the permeable layers and to other adjacent cavity spaces. To isolate the effect of the convective and radiative heat transfer in a layer medium from the convective heat transfer occurring in the adjacent gas space, each glazing layer is assumed to behave as a porous medium having equivalent effective radiation and thermal properties. The latter should be calculated for a given layer type. This assumption provided acceptable results for flows in air gaps encompassing shading devices (Safer et al., 2005; Zhang et al. 1991). The convective heat transfer in the gas cavity may be by natural or forced convection, or a combination of the two. In all cases, the model of the ISO standard 15099 for ventilated cavities (equation 1) is used in this work, but with a proper calculation of the cavity film coefficient (h_{cavity}) and gas velocity (v). The following sections present new models to compute these quantities (h_{cavity} and v) for open cavities bounded by porous layers.

4.4.1 Thermal Penetration Length Model For Natural Convection

The thermal penetration length model is used to compute the film coefficient of a gas cavity (h_{cavity}) enclosed between porous layers and subject to buoyancy driven flows. The thermal penetration length model stems from the fact that if natural convection is occurring in a cavity gas space, the convective effect may penetrate the porous layers up to a certain length (or depth). This phenomenon is known as *penetrative convection*, particularly in porous media (Bejan, 2004). For convective flows around surfaces, the thermal penetration length is equivalent to the thickness of the boundary layer as the latter separates between two zones: boundary layer zone with

significant temperature gradient and a stagnant zone far away from the surface. The convective heat transfer from the bounding layers to the cavity gas space is, therefore, a function of this thermal penetration length (Bejan, 2004).

To apply this concept to the problem of heat transfer in complex fenestration systems, consider a porous layer (j) of thickness L_p , forming a gas cavity with an adjacent layer (j+1). The layer (j) is initially at a uniform temperature T_j . If the adjacent layer (j+1) is maintained at a different temperature T_{j+1} , a buoyancy-induced flow may occur in the cavity, and the thermal convection effect may penetrate the medium of layer (j) up to a distance γ (see Figure 2). This convection effect may invalidate the use of correlations for cavity film coefficient based on the cavity thickness L_j . As the convective heat transfer in the gas cavity is a function of this penetration length (γ), one may, therefore, postulate that the cavity film coefficient (h_{cavity}) may be obtained using the existing correlations applied on a hypothetical cavity with a thickness linearly proportional to the thermal penetration length, namely $L_j + \delta$ (with $\delta \propto \gamma$). The thermal penetration length (γ), which is solely due to convection, depends on the geometrical and thermal characteristics of the layer medium (j), and the temperature difference ($T_{j+1} - T_j$). It should, however, be independent of the cavity spacing L_j . For highly porous layers such as shading layers (e.g., porosity ≈ 0.98 for blinds and draperies), convection in the cavities of the porous layer may be considered as convection in a homogenous medium. In this regard, if the front surface of layer (j+1) is moved to touch the back surface of layer j, then the thermal penetration length γ will be proportional to the thickness of the boundary layer of a vertical (or tilted) plate at temperature T_{j+1} adjacent to the gas space in the layer medium (j). The thickness of the boundary layer, itself, is a function of the Rayleigh number (Bejan, 2004). The distance δ will, therefore, take the following form:

$$\delta / H = C \cdot Ra_H^{-m} \quad (16)$$

where (C) and (m) are constants and H is a characteristic length of the porous layer, which are to be determined according to the flow and layer types. The exponent (m) in equation (16) may take the values 1/4 for laminar flows and 1/3 for turbulent flows (Bejan, 2004). Given the fact that the thickness of the boundary layer (γ) is also inversely proportional to the plate Nusselt number (Bejan, 2004), the constant (C) may, therefore, take the following form:

$$C = D \cdot Ra_H^m / Nu_H \quad (17)$$

where D is a constant to be determined, and Nu_H the Nusselt number for the convective heat transfer over a vertical (or tilted) plate with H as a characteristic height. Note that the ratio Ra_H^m / Nu_H in equation (17) should evaluate to a constant value, particularly for convection dominated flows.

For slat-type blind layers, the airflow in the blind layer is restricted in the cavity between slats. The characteristic height (H) is, therefore, equal to the slat spacing ($H = s$), and the Rayleigh number Ra_H , evaluated at the average temperature $(T_j + T_{j+1})/2$, is given by:

$$Ra_H = \frac{g \cdot \beta \cdot |T_{j+1} - T_j| H^3}{\alpha \cdot \nu} \quad (18)$$

where:

- g : gravitational coefficient (m/s^2)
- α : thermal diffusivity of cavity gas (m^2/s).
- β : thermal expansion coefficient of cavity gas (K^{-1}).
- ν : viscosity coefficient of cavity gas (m^2/s).

The proportionality constant (D) in equation (17) would depend on the flow type in the cavity between layers whether it is laminar or turbulent, and should be determined based on measurement or suitable CFD work. In the absence of such work, the constant D was estimated based on comparing the predicted (using the present models) and measured (using measurement by Huang et al., 2006) values of the U-factor for double glazed windows with

between-pane blinds (see details in the validation section below). For laminar flows within the vertical window cavities, the value of the D and C constants was found equal to 0.314 and 0.56, respectively.

In a similar way, equations (16) to (18) may be used for vertical slat blinds, or drapery layers, but with a characteristic height (H) set equal to the window height.

It should be noted that the penetration length model (δ) should be used only when the minimum cavity spacing L_j is finite and non-zero so that convection may take place in the cavity between layers. If both layers touch ($L_i = 0$), or if convection is not present in the cavity between layers (say when $Ra_{Li} \ll 1$), or if convection is not present in the cavity between slats (say when $Ra_H \ll 1$), δ should be set to zero. Further, the penetration length should take a maximum value not exceeding half of the thickness of the porous layer ($\delta \leq L_p/2$). For slat-type blind layers, the layer thickness (L_p) designates the horizontal projection of the slat width (including also its thickness portion). For drapery layers, the layer thickness (L_p) is given as shown in Figure 6.

The thermal penetration length (δ) should be calculated for each porous layer of a fenestration system. For example, If both cavity layers (j and j+1) are porous, the cavity thickness used to calculate the film coefficient (h_{cavity}) should be equal to $\delta_j + L_j + \delta_{j+1}$. Furthermore, although the constant D in equation (17) was estimated for laminar cavity flows, which mostly occur in fenestration systems, care should be exercised to apply this equation to turbulent, or transitional flows without further validation studies.

Finally, although the thermal penetration length model may theoretically be applied to any type of porous layers, one should bear in mind that the above analysis is limited to shading layers having high porosities (≈ 1). For other porous layers with low porosities such as, for example, layers with translucent insulation, equation (16) may still be used, but with different values for the constants m, and C, and with a proper formalism for the Rayleigh number in

porous media (more details on such formalism may be found in Bejan, 2004). This case is left for future research.

4.4.2 Glazing Sealed Cavities with Integrated Shading Devices

Glazing sealed cavities with integrated porous shading devices deserve particular attention. Airflows in such cavities differ from airflows in cavities without shadings. Experimental observations in cavities with blinds showed that air flows one-way in the sub cavities adjacent to a blind layer (see for example: Naylor and Lai, 2007). Air moves up along the hot glazing layer and moves down along the cold glazing layer. Due to the air temperature differential between the two sub cavities, air crosses the blind layer at the top and bottom edge sections and along the blind layer. The ISO ventilated cavity model may, thus, be used in such cases. Following the ISO monoculture, the top opening area may be set equal to the surface area of the edge-of-glazing section (with width = 6 mm) multiplied by the shading porosity, or for slat-type blinds, the surface area between the glazing bottom/top edge and the tip of the last slat, whichever is smaller. The front opening is set equal to the surface area of blind layer multiplied by its porosity.

It should be added that in most window and blind systems with narrow cavities (glazing cavity width comparable to blind slat width), the air velocity in the sub cavities adjacent to blind layers is so small that its effect on the film coefficient can be neglected in equation (1), particularly when blinds are open (slats horizontal). However, when blinds are closed, the air velocity in cavity may be significant, particularly when glazing layers feature low emissivity coatings to induce a significant stack effect in winter conditions (note that the stack effect is proportional to temperature differential and inversely proportional to the absolute temperatures of connected cavities; the lower the cavity temperatures the higher the stack effect).

4.4.3 Screen Flow Model for Forced Convection

Forced convection in cavities may occur using mechanical ventilation with known velocities, or unknown wind-driven ventilation. For cavities bounded by porous layers open to the exterior environment, the wind-induced air

velocity in the cavity may be obtained using the flow attenuation coefficient of the porous layers between the exterior environment and the cavity under consideration. For screen type layers (such as screen shadings and drapery sheets), the air velocity in a cavity may be obtained as follows (Miguel, 1998):

$$v_j = 0.89 \cdot \varphi^{1.25} \cdot v_{j-1}; \quad \text{valid for } 0.04 \leq \varphi \leq 0.9 \quad (19)$$

where v_j is the air velocity in cavity (j) sharing a porous layer with the front cavity (j-1) whose air velocity is v_{j-1} . Equation (19) was experimentally developed for screen shadings, but may be used for other shading devices in absence of any suitable information. If the wind-driven velocities are weak so that natural convection dominates, the ISO model for ventilated cavities (equation 1) should be used to calculate the air velocity in the cavity.

4.5 EFFECTIVE THERMAL AND RADIATIVE PROPERTIES OF SHADING LAYERS

Shading layers can be grouped in three generic types: perforated plain screens, slat-type blinds, and draperies (or pleated shadings). One common characteristic of all shading types is porosity, which is defined as the ratio of the void volume to the total volume of layer.

4.5.1 Screen Shadings

A screen-shading layer is characterized by its openness factor (φ) and the yarn (or fibre) density (κ). The openness factor is defined as the ratio of the void surface area to the total surface area of a screen, and the yarn density is defined as the ratio of the average width of a yarn section to the screen layer thickness. Figure 3 shows a schematic description of a screen layer. The openness factor is also equivalent to the screen porosity (ω). In general, the screen material (fabric) may transmit, reflect and absorb thermal radiation. Using the averaging theory of porous media, the effective thermal properties of a screen layer are given by the following relationship:

$$P_s = (1 - \varphi) \cdot P_m + \varphi \cdot P_a \quad (20)$$

where P stands for any given thermal property such as thermal conductivity (k), density (ρ), or the product of density and specific heat (ρc). The

subscripts (s, m, a) denote the screen layer, screen material (fabric) and air volume, respectively.

As for the effective long wave radiative properties of a screen layer, a proper radiation balance performed on a representative control volume as shown in Figure 3, yields the following relationships for the effective emissivity (absorptance), reflectance and transmittance:

$$\varepsilon_{eff,f} = (1 - \varphi) \cdot \varepsilon_{m,f} + \frac{4}{\kappa} \cdot \frac{(\sqrt{\varphi} - \varphi) \cdot F_{wf}}{1 + (1 - \varepsilon_{m,a})(2F_{wf} - 1)} \cdot \varepsilon_{m,a} \quad (21)$$

$$\rho_{eff,f} = (1 - \varphi) \cdot \rho_{m,f} + \frac{4}{\kappa} \cdot \frac{(\sqrt{\varphi} - \varphi) \cdot F_{wf}^2}{1 + (1 - \varepsilon_{m,a})(2F_{wf} - 1)} \cdot (1 - \varepsilon_{m,a}) \quad (22)$$

$$\tau_{eff,f} = 1 - \rho_{eff,f} - \varepsilon_{eff,f} \quad (23)$$

where:

F_{wf} : form factor of the vertical surface (wall) to the horizontal surface (floor) of the void section (dimensionless).

$\varepsilon_{eff,f}$: effective emissivity (absorptance) of the front screen surface (decimals)

$\varepsilon_{m,f}$: emissivity (absorptance) of the front surface of screen material (decimals)

$\varepsilon_{m,a}$: average emissivity (absorptance) of the front and back surfaces of screen material (decimals)

Similarly, the radiative properties for the back surface of a screen layer are obtained using equations (21) to (23) by simply replacing the index (f) by the index (b). The form factor F_{wf} is a function of the width of the void section (w_{void}) and the screen thickness (L_s), which are related to each others through the screen openness factor and yarn density as follows:

$$w_{void} = \frac{\sqrt{\varphi}}{1 - \sqrt{\varphi}} \kappa \cdot L_s \quad (24)$$

The void section may be approximated by a square cylinder with an average width = w_{void} and height = L_s . F_{wf} may, thus, be determined using the established relationships for the form factors of cylinder walls to its opening surface.

Figure 4 shows the profiles of the effective emissivity and infrared transmittance of a screen shading with an opaque fabric ($\tau_m = 0$, $\varepsilon_m = 0.9$) as a function of the openness factor. The effective emissivity and infrared transmittance vary non-linearly with the openness factor, except at high values of the openness factor ($\phi > 50\%$).

4.5.2 Slat-Type blinds

Figure 5 shows a slat-type blind layer at a tilted window position. The blind slats may be opaque or perforated (with openness factor ϕ_s). The radiative properties of slats may, thus, be given by the above screen model (equations 21 to 23).

The slats may enclose an open air space, particularly when they are fully open (slat angle $\alpha = 0$). If the blind layer is subject to a temperature difference at its front and back environments, a cellular flow may develop in the air space between slats (Machin et al., 1998; Naylor and Lai, 2007). This cavity flow may increase the heat transfer through the blind layer. Further, the environment air may flow through the porous blind layer, but its effect on heat transfer may be neglected due to low velocities and relatively high-pressure losses through the blind layer. To account for the effect of the slat cavity flow on heat transfer, the air cavity is assumed as a solid material with an equivalent thermal conductivity. Temperature gradients through a blind layer denote thus the average of the slat and enclosed air cavity temperatures. Using a representative control volume as shown in Figure 5, the effective thermal properties of a blind layer are calculated as follows:

$$k_b = (1 - \omega) \cdot k_{sm} + \omega \cdot k_{eq} \quad (25)$$

$$\rho_b = (1 - \omega) \cdot \rho_{sm} + \omega \cdot \rho_a \quad (26)$$

$$(\rho c)_b = (1 - \omega) \cdot (\rho c)_{sm} + \omega \cdot (\rho c)_a \quad (27)$$

where:

c_{sm} : specific heat of slat material (J/kg.K)

k_b : effective thermal conductivity of blind layer (W/mK)

k_{eq} : equivalent thermal conductivity of the air cavity between slats (W/mK)

k_{sm} : thermal conductivity of slat material (W/mK)

ρ_{sm} : density of slat material (kg/m³)

The porosity of a blind layer (ω) is equal to one minus the ratio of the slat material volume to the volume of a representative control volume as shown in Figure 5:

$$\omega = 1 - \frac{(1 - \varphi_s) L_s t_s}{L_p \cdot (s + t_s)} \quad (28)$$

where φ_s , L_s and t_s are openness factor, length and thickness of a slat, respectively. Equation (28) accounts for curved and thick slats, and therefore their effect on the thermal bridging is accounted for in the effective thermal conductivity of blinds.

The equivalent thermal conductivity (k_{eq}) of the cavity between slats (denoted by ABCD in Figure 5) may be determined based on the convection film coefficient of a rectangular cavity subject to temperature differential at its open boundary surfaces (AB and DC) and adiabatic conditions at slat surfaces (BC and AD) as follows:

$$k_{eq} = h_c(T_h - T_c, w, z_c, \beta_c) \cdot w \quad (29)$$

where:

h_c : film coefficient of between slats cavity (W/m²K).

T_h : temperature of the hot open surface of between slats cavity (K).

T_c : temperature of the cold open surface of between slats cavity (K).

w : thickness of between slats cavity (equal to the slat width) (m).

z_c : height of between slats cavity (m).

β_c : inclination angle of between slats cavity (radians).

The height and inclination angle of the between slats cavity are determined as follow:

$$z_c = s \cdot \cos \alpha \quad (30)$$

$$\beta_c = |\beta_w + \alpha|, \text{ if } \beta_c > \pi/2; \text{ otherwise, } \beta_c = \pi - |\beta_w + \alpha| \quad (31)$$

where:

s : spacing of slats (m).

α : tilt angle of slats (radians).

β_w : tilt angle of window (radians).

As for the effective radiative properties of a blind layer, Yahoda and Wright (2004b) presented details on the corresponding calculation procedures.

4.5.3 Drapes

Drapes are similar to screen shadings, except that the drapery fabric may be folded or pleated along the window width, hence forming open vertical pockets of air. In addition to the characteristics of a screen layer as mentioned before, drapes are characterized by the fullness factor (Fr), which indicates the ratio of the drapery fabric width to the window width. Fullness factors of 1.5 and 2 are typical values. The enclosed air pocket cavity may affect the heat transfer through the drapery layer if the latter is subject to temperature differential at its boundary environments. Further, the environment air may flow through the drapery porous layer, but its effect on heat transfer may be neglected due low velocities. Similar to a blind layer, flow within the air pocket cavities may be accounted for by an equivalent thermal conductivity. Equation (29) may, thus, be used for a drapery layer. Approximating the drapery configuration as a sequence of one-sided open rectangular cavity (Kotey et al., 2009a; Farber et al., 1963), as shown in

Figure 6, the cavity characteristics used to calculate the film coefficient in equation (29) are: cavity height (z_c) and tilt angle (β_c) are equal to the height and tilt angle of window, and the cavity thickness (w) is equal to the thickness of the drapery layer (L_p) minus the thickness of the drapery material fabric. Note that, the air pocket cavities are finite in the direction of the fenestration width, and, therefore, proper correlations to account for the effect of cavity width (s) should be used to compute the film coefficient in equation (29). With regard to the effective radiative properties of a drapery layer, a proper radiation heat balance over a representative control volume as shown in Figure 6, yields the following relationship:

$$\varepsilon_{eff,f} = \frac{1}{2} \varepsilon_{d,f} + \frac{1}{2} \frac{\varepsilon_{d,f} (1 - \rho_{d,b} F_{cc} + \tau_{d,f} \tau_{d,b} F_{cc}^3) + \varepsilon_{d,b} \cdot \tau_{d,b} F_{cc} (1 + F_{cc} - \rho_{d,f} F_{cc})}{(1 - \rho_{d,b} F_{cc})(1 - \rho_{d,f} F_{cc}) - \tau_{d,b} \tau_{d,f} F_{cc}^4} \quad (32)$$

$$\tau_{eff,f} = \frac{\tau_{d,f}}{2} \cdot (1 - F_{cc}) \frac{(1 - \rho_{d,b} F_{cc}) + (1 + F_{cc})(1 - \rho_{d,f} F_{cc}) + \tau_{d,f} F_{cc}^2 (\rho_{d,f} + \tau_{d,b} F_{cc})}{(1 - \rho_{d,f} F_{cc})(1 - \rho_{d,b} F_{cc}) - \tau_{d,f} \tau_{d,b} F_{cc}^4} \quad (33)$$

where:

F_{cc} : form factor of the pocket cavity surface to itself.

$\varepsilon_{d,f}, \varepsilon_{d,b}$: effective emissivity of the front and back surfaces of a drapery sheet.

$\rho_{d,f}, \rho_{d,b}$: long wave effective reflectances of the front and back surfaces of a drapery sheet.

$\tau_{d,f}, \tau_{d,b}$: long wave effective transmittance of the front and back surfaces of a drapery sheet.

The long-wave effective radiation properties of a drapery sheet are determined using the previous screen model (equations 21 to 23). The cavity form factor F_{cc} may be expressed as a function of the fullness factor as follows:

$$F_{cc} = 2 \frac{Fr - 1}{2Fr - 1} \quad (34)$$

Figure 7 shows the profiles of the effective emissivity and infrared transmittance of drapery layers with an opaque fabric ($\tau_m = 0$, $\varepsilon_m = 0.9$) as a function of the openness and the fullness factors.

5 MODEL VALIDATION AND ASSESSMENT

The above models are implemented in the research version of SkyVision (NRC, 2008), and validated using the available experimental measurement and CFD numerical calculations of centre of glass U-factor of windows with between-pane and internal Venetian blinds. The following sections provide details on the comparative studies.

5.1 EXPERIMENTAL VALIDATION

Huang (2005) and Huang et al. (2006) conducted laboratory measurements to compute the centre of glass U-factor of a double glazed window with integrated Venetian blinds using the Guarded Heated Copper Plate apparatus. The plate dimension were 635 mm x 635 mm (or 604 mm x 604 mm without the spacer section). Three sets of window cavity spacing were considered in the measurement: 17.78 mm, 25.4 mm, and 40 mm. The window was made up of 3 mm glass sheets with two high and low emissivity values (0.84 and 0.164 on the hot surface). The blinds were made of aluminium-alloy slats with the following characteristics: slat thickness = 0.2 mm, slat width $w = 14.79$ mm, slat spacing $s = 11.84$ mm, slat curvature height $d = 1.5$ mm, and slat emissivity of 0.792. To initiate heat flow through the window and blind system, the hot plate was maintained at a temperature of 30°C, and the cold plate took two temperature values of 20°C and 10°C. The heat flux was measured at the hot plate at three locations: centre and edge-end sections, each covering a surface area of 200 mm x 200 mm. The centre of glass U-factor was calculated based on the centre heat flux measurement and assumed values for the interior and exterior film coefficients of 8 W/m²K and 23 W/m²K, respectively. The window and blind system and measurement setup were also simulated using a CFD numerical method (Naylor and Collins, 2005; Avedissian and Naylor, 2007) to validate the prediction of the numerical model.

Figures 8 to 10 show a comparison between the measurement and CFD results and the current model for the three cavity gap spacing values, respectively. The predictions from the current model are in a very good agreement with both the measurement and CFD calculations for the regular and low-emissivity windows. Similar comparison results were also obtained for other temperature boundary conditions and cavity gap spacing values. The maximum difference of about 6% was observed for a window with a low-emissivity coating and large gap spacing (40 mm) when slats are almost closed (Figure 9). The current model provided better results than the simple model of Avedissian and Naylor (2007), which over-estimated the U-factor by about 10%. This simple model combines the one-dimensional heat transfer through the window panes, and the CFD-based correlations of the cavity film coefficient.

5.2 NUMERICAL VALIDATION

Naylor et al. (2006) employed a full CFD numerical simulation to compute the centre of glass U-factor of a double glazed window with internal Venetian blinds. Shahid (2003) also proposed a simple model, which combines a one-dimensional heat transfer through the window and blind layers, and CFD-based correlations for the cavity film coefficients. The modelled window was made up of two 3 mm regular glass sheets spaced 12.7 mm apart. The blinds were made of aluminium-alloy slats with the following characteristics: slat thermal conductivity = 123 W/mK, slat thickness = 0.16 mm, slat width $w = 25.4$ mm, slat spacing $s = 22.28$ mm, and slat curvature height $d = 1$ mm. The blinds were spaced from the window surface so that the slat-tip to glass distance was maintained at 27.5 mm, or 7.7 mm. The CFD calculations as well as the current model assumed that the air cavity between the window and blinds was open at the side, top and bottom sections. The current model uses the ISO model for a ventilated gap to compute the gap velocity in equation (1). The performance of the window and blind system was evaluated under the ASHRAE boundary conditions: Indoor temperature = 24°C, outdoor temperature = 32°C and wind speed = 3.5 m/s (with corresponding outdoor film coefficient of 16.77 W/m²K).

Figure 11 shows the comparison results for a window and blind system with a slat-tip to glass spacing = 27.5 mm, and slat emissivity values of $\varepsilon_b = 0.8$ and 0.1. The predictions from the current model are in good agreement with the CFD results, with a maximum difference less than 5%. The obtained results also reveal that internal regular blinds (with $\varepsilon_b = 0.8$) with open air cavities weakly affect the window U-factor with a maximum difference less than 17% obtained when slats are closed. Reducing the slat emissivity to $\varepsilon_b = 0.1$ may significantly reduce the window U-factor only when slats are closed (e.g., during nighttimes), with a maximum difference less than 40%.

5.3 EFFECT OF THERMAL CONDUCTIVITY OF BLINDS

Part of his CFD numerical study, Avedissian (2006) also reported on the effect of low slat thermal conductivity on the Nusselt number of a double glazed window with between-pane blinds. He found that the thermal bridging effect of blinds may be significantly reduced by up to 125% when using slats with low thermal conductivity (e.g., 0.4 W/mK) compared to metallic slats.

Unfortunately, these CFD results cannot directly be used to compute the U-factor of a window and blind system and comparing them with the current model. For general information and to show the significance of the current models, this section reports on the effect of the blind thermal conductivity on the U-factor of window and blind system. A typical size double glazed window (width = 0.6 m, height = 1.5 m) is considered for this purpose. The window is made up of 6 mm glass sheets spaced 20 mm apart. Venetian blinds are placed in the middle of the window air cavity, with the following characteristics: slat thickness = 0.2 mm, slat width $w = 12.7$ mm, slat spacing $s = 11$ mm, and slat emissivity = 0.8. Two values of slat thermal conductivity are considered $k_b = 120$ W/mK (metallic) and 0.2 W/mK (vinyl). The performance of the window and blind system is evaluated under the NFRC winter conditions (NFRC, 2004): Indoor temperature = 21°C, outdoor temperature = -18°C and wind speed = 5.5 m/s.

Figure 12 shows the effect of the blind thermal conductivity on the U-factor of regular and high performance windows with glass emissivity = 0.84 (high-e)

and 0.157 (low-e) on surface #3 of the interior pane. The model of Wright et al. (2008), the best model available so far, is also shown in the figure. Without the blinds, the U-factors are 2.65 W/m²K, and 1.84 W/m²K for the regular (high-e) and low-e windows, respectively. It is clear that the effect of low blind thermal conductivity on the U-factor is maximum (around 20% to 28% compared to metallic blinds) when slats are open, and minimum when slats are closed. Noteworthy is that the U-factor of a window and blind system increases with the angle of the low conductivity slats, being minimum when slats are open, particularly for low-e windows. This behaviour is reversed for regular window and blind systems with metallic slats, where the U-factor is minimum when slats are closed. This observation may be explained by the fact that, the effect of low slat thermal conductivity is more significant when thermal conduction dominates in the window air cavity (when slats are open) than when convection dominates (when slats are closed). This trend was also found in the CFD simulation results of Avedissian (2006). The impact of the proposed model is also clear when compared with the model of Wright et al. (2008), which does not take into account the slat spacing and thermal conductivity. For the metallic blinds, the predictions of the model of Wright et al. (2008) compare very well with the present model, except when blinds are almost closed. When blinds are almost closed (slat angle = 89°), the present model predicts that the airflow through the blind layer is significant and results in an increase of the cavity film coefficient by several percentages.

6 MODEL IMPLEMENTATION

To implement the forgoing models in fenestration computer simulation programs, the following algorithm should be used:

Step 1: Pre-calculate any constant property that does not vary with temperature, such as the projected thickness and the effective radiation properties (emissivity/absorptance and infrared transmittance) of each complex layer making up the fenestration system as per Section 4.5.

Step 2: Compute the following items:

- the effective thermal properties (thermal conductivity, specific heat and density) for each fenestration complex layer as per Section 4.5.
- the velocity for each ventilated gas space between layers using the model of the ISO Standard 15099 for natural ventilation, or equation (19) for gas spaces in contact with the exterior environment.
- the thermal penetration length (δ) using equations (16) and (17) for each gas space bounded by porous layers.
- the film coefficient of the gas cavity between layers (h_{cavity}) using existing correlations for enclosed and sealed cavities (for example, those of the ISO standard 15099), but by substituting the cavity thickness L with $L + \delta$.
- the film coefficient of the ventilated gas cavity using equation (1).

Step 3: evaluate the nodal temperatures of each fenestration layer and gas space, and the resulting convective and radiative heat exchanges as per Sections 4.2 and 4.3.

Step 4: Repeat steps 2 and 3 until convergence in layer temperatures is reached. Convergence is declared when the maximum relative change in the layer nodal temperatures is lower than a given tolerance.

7 CONCLUSION

This paper developed a general methodology to compute the thermal performance of complex fenestration systems employing open gas spaces and permeable layers, which may exchange mass and heat through radiation, convection and conduction to its surrounding environments, and elements for energy generation and conversion embedded in glazing layers. The models assume each system layer is porous with calculated effective radiation and thermal properties. The effective radiation properties (such as emissivity and infrared transmittance) and thermal properties (such as thermal conductivity, density and specific heat) of layers are calculated based on the geometrical and thermal characteristics of the layer medium and the effect of the convective heat transfer in the layer porous structure. A new thermal penetration length model was developed to account for the effect of a porous

layer on the convective heat transfer from its boundary surfaces to the adjacent gas spaces. The model was particularly applied to various types of shading devices. A validation study was carried out, in which the U-factor of double-glazed windows with between-pane and internal Venetian blinds were compared with the available laboratory measurement and CFD calculations. The comparison results showed that the model predictions were in good agreement with the measurement and CFD calculations. It should be stressed that, despite the good agreement, further validation work should be considered to cover other products and types of shading devices and complex layers, window tilt configurations, and boundary conditions, particularly the ones that trigger turbulent flows in cavities between shades and glazing.

ACKNOWLEDGMENT

This work was supported by the Institute for Research in Construction of the National Research Council Canada (NRC-IRC), CETC Buildings Group of Natural Resources Canada (NRCan), Canada Mortgage and Housing Corporation (CMHC), Ontario Power Authority, Hydro Quebec, Gas Metro Quebec, Pilkington USA, Prelco Inc., Talus Limited, and Lutron Electronics Corporation USA. The author is very grateful for their contributions and support.

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Figure 1 Schematic description of a multi-layer fenestration system. Note that gas spaces are treated differently than glazing layers.

Figure 2 Thermal penetration length of a porous layer

Figure 3 Schematic description of a screen layer

Figure 4 Profiles of the effective emissivity and infrared transmittance of opaque screen shadings.

Figure 5 Schematic description of a slat-type blind layer

Figure 6 Schematic description of a drapery layer

Figure 7 Profiles of the effective emissivity and infrared transmittance of drapery layers.

Figure 8 Result comparison for U-factor of double glazed windows with between-pane Venetian blinds for a window cavity spacing = 17.78 mm.

Figure 9 Result comparison for U-factor of double glazed windows with between-pane Venetian blinds for a window cavity spacing = 25.4 mm.

Figure 10 Result comparison for U-factor of double glazed windows with between-pane Venetian blinds for a window cavity spacing = 40 mm.

Figure 11 Result comparison for U-factor of double glazed windows with internal Venetian blinds for a slat-tip to glass spacing = 27.5 mm.

Figure 12 Effect of the blind thermal conductivity on the U-factor of double glazed windows with between-pane blinds.