A 2-D NUMERICAL MODEL FOR HEAT TRANSFER CALCULATIONS IN MULTIPANE WINDOWS

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SUMMARY

The objective of this research is the numerical determination of overall heat transfer coefficients (*U-Factor*), solar heat gain factors (*SHGF*), and surface temperature profiles of windows. The program takes account of the presence of the natural convection, the radiation between panes, and the thermal effects of the frame. Temperature profiles are useful for the calculation of thermal stresses in glazings and the prediction of condensation problems. Results are compared with measurements and the numerical calculations of others. In benchmark solution comparisons errors, are less than 10^{-2} %. Comparisons with the numerically determined heat flux through a cavity bounded by two sheets of glass and spacer bars, i.e., a simple window, differed by less than 1 %. In a comparison between simulated solar heat gain factors and solar calorimeter experimental results very good agreement is obtained for nine samples, with errors of 2.5% or less for seven samples, and errors of 12% and 5% for the other two.

INTRODUCTION

Twenty years ago most engineers used various hand calculation methods based on the so called *steady-state methods* to design for peak-load conditions in a building. The most important design consideration then was to ensure that any heating ventilating and air conditioning (HVAC) plant could meet the worst design peak load for heating and cooling. These conditions could be satisfactorily designed for using a overall heat transfer coefficient (*U-factor*) for the thermal conductance and a Solar Heat Gain Factor (*SHGF*), for peak solar gain. Nowadays engineers are interested in the optimization of a building's energy efficiency and thermal performance. Therefore, such approximations alone are no longer adequate. Also, with the development of new window technologies and materials, given the rise of so called *super windows* (Gilmore, 1986), even the determination of these parameters cannot be handled by current calculation procedures. The design of super windows or their elements, such as edge-seals, frames and glazing systems, cannot be done experimentally alone because experiments are quite expensive and time consuming, and the type and accuracy of measurement are limited by experimental capabilities. In addition, the development of measures for energy labeling schemes require that more detailed parameters be determined, e.g., shading coefficient, condensation resistance, solar heat gain. These needs, mentioned above, are leading to the development of new tools, both experimental and numerical, to quantify the heat transfer, to design new elements, and to develop new parameters in addition to the *U-factor* and *SHGF* for fenestration systems.

The objective of this research is the numerical determination of the overall heat transfer coefficient (*U-Factor*) and the solar heat gain factor (*SHGF*) to evaluate the thermal performance, and surface temperature profiles to assist in the prediction of condensation on windows. The program takes account of the presence of the natural convection, the radiation between panes and, the thermal effects of the frame. The developed numerical model, a non-orthogonal finite-volume code, provides information regarding energy flows within the window and temperature distributions in individual glazing and frame. The latter is useful for the calculation of thermal stresses in glazing and the prediction of condensation problems.

A detailed review and some basic information on the physics of the heat transfer in windows is presented in de Abreu (1996) which complements several well known and excellent reviews that have appeared in the literature concerning *U-Factor* calculation procedures (McCabe and Goss, 1987) and natural convection in enclosures (for instance, Ostrach, 1988; Wright and Sullivan, 1989; Fusegi and Hyun, 1994).

Despite recent advances in computer hardware, numerical simulation algorithms, and grid generation schemes, which have the potential capability to model the physical and geometrical complexities of windows, little work has been done to model a whole window, except for the works of Smith *et al.* (1993), Curcija and Goss (1994), and Wright and Sullivan (1994). Numerical modeling can provide detailed information (velocity and temperature distributions, stream line and heat flux fields, throughout an entire window) that is generally unavailable from experimental studies due to limitation in the experimental techniques currently available. A well defined comparison between numerical simulation and measurement can determine the level of numerical accuracy for a given calculation. Therefore, various aspects of the window performance that are not available from experimental measurements can be further studied. It is this unique capability of the computational tool that can most impact the design and thermal analysis of windows.

MATHEMATICAL MODEL

At the external surfaces of the window (both indoor and outdoor) convection and radiation heat transfer occur simultaneously, while in window cavities (frame and glazing cavities), depending on the intensity of the air movement, convection or radiation can prevail. In the solid portions of the window system (glazing layers, edge, and frame construction elements) and small enclosed air spaces, where air motion is suppressed, only conduction occurs. These components form the region of interest.

All the walls are stationary and impermeable; the no slip conditions exists at them. The glazing system is imperfectly transparent to radiation (i.e., some absorption of radiation may occur) and the frame components are opaque to radiation with convection, radiation and conduction energy exchanges with the surroundings occurring on window outside surfaces. The top and bottom walls are opaque to radiation. It is assumed that the confined fluid is Newtonian and that the Boussinesq approximation is valid. All surfaces are taken to be gray diffuse reflectors and emitters of long-wave radiation (may be specular in case of a closed rectangular cavity) and the confined gas is assumed to be radiatively passive. The radiation model is a two-band model: short and long wave. This implies that emission from the surfaces of the cavity is "long-wave" radiation and that solar radiation is "short-wave". If one or more of the glass panes are spectrally selective a spectral solar radiation model (short wave) may be used to account for sharp changes in the radiation properties with the wave length. The quations are written in terms of the primitive variables, namely, local velocity components, pressure and temperature. When viscous dissipation in the energy equation is neglected the governing differential equations for Newtonian and incompressible fluid, with constant properties except in the formulation of the buoyancy term, can be written in Cartesian coordinates as:

- **continuity**:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \quad (1)$$

- **momentum:**

in x:

$$\rho\frac{Du}{Dt}=-\frac{\partial p_k}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \quad (2)$$

in y:

$$\rho\frac{Dv}{Dt}=-\frac{\partial p_k}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+\rho g\beta\left(T-\bar{T}\right) \quad (3)$$

- **energy:**

$$\rho c_p\frac{DT}{Dt}=k\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)+\dot{S} \quad (4)$$

where, ρ, μ, β, g, c_p and k are the density, viscosity, coefficient of thermal expansion, gravity, constant-pressure specific heat and conductivity, respectively; t is the time; x, y are the Cartesian coordinates; u, v are the velocity components; T is the temperature; $\rho g\beta(T-\bar{T})$ is the buoyancy force/unit volume in the y direction where $\bar{T}=(T_h+T_c)/2$; T_h and T_c are the temperatures at the hot wall and cold wall, respectively; $\dot{S}$ is the source term, and the origin of the coordinates (x, y) is placed at the most lower left corner of the cavity with gravity in the $-y$ direction. p_k, the kinematic pressure, drives the flow. It is obtained from the following expression

$$p(y)=p_o-\int_0^y\bar{\rho}(y)g\,dy+p_k$$

- ***at all surfaces (fluid/solid interfaces)***

$$v=u=0 \quad (5)$$

On the solid portions of the window system heat transfer is governed only by the modified energy equation since the velocity components are zero, and therefore momentum and continuity equations no longer needed to be solved. The energy equation becomes:

$$\rho c_p\frac{\partial T}{\partial t}=k\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)+\dot{S} \quad (6)$$

The temperature boundary conditions are derived from the energy balance at the surfaces.

The long wave radiation heat transfer occurring in the glazing cavity can be described using the Radiosity Matrix Method for determining radiation exchange in an enclosure (Siegel and Howell, 1981). The incident long-wave radiative heat flux denoted by the superscript l, at any surface j of the cavity can be written as

$$q_j^l=\frac{(J_j^l-\varepsilon_j^l\sigma T_j^4)}{\rho r_j^l} \quad (7)$$

in which J_j^l is the radiosity:

$$J_k^l=\varepsilon_k^l\sigma T_k^4+\rho r_k^l\sum_{j=1}^{N}J_k^l E_{j-k}^l \quad (8)$$

where the $E_{j\text{-}k}^l$ is the specular exchange factor between j and k, which accounts for radiation going from surface j to surface k directly and by all possible specular (mirror-like) reflections and ρr_k^l is the diffuse reflectance of surface k. When the surfaces of the enclosure do not specularly reflect radiation, the specular exchange factors $E_{j\text{-}k}^l$ reduce to the usual view factor (configuration factor) $F^l_{j\text{-}k}$. Methods for calculating $E_{j\text{-}k}^l$ are given in advanced radiation texts, like Siegel and Howell (1981). For a rectangular cavity (bounded by plane surfaces), the view factor between any two points on the same surface is zero; this substantially reduces the number of terms on the right hand side of Equation (8).

Furthermore, since a two-band radiation model is assumed and all the emitted radiation is in the long-wave band, the short-wave component of radiosity, J^s, depends only on the geometry and the solar radiation. The short-wave radiosity for a double pane window is therefore,

$$J_j^s=\tau_j^s q^s+\rho r_j^s\sum_{k=1}^{N}J_k^s E_{k-j}^s \quad (9)$$

and at the others surfaces

$$J_j^s=\rho r_j^s\sum_{k=1}^{N}J_k^s E_{k-j}^s \quad (10)$$

The Edward's embedding technique was chosen as the model to calculate the short wave radiosities because is a simple way to determine the fraction of solar radiation that strike and is absorbed by the surface directly or by multiple reflections, in multi-glazed windows.

It follows from Equations (9) and (10) that the short-wave radiosities need only to be calculated once for a given geometry. The long wave component J^l_j, however, needs to be evaluated at each stage of the calculation so that the radiosity

$$J_j=J_j^s+J_j^l \quad (11)$$

The incident radiative heat fluxes q^l and q^s in the Equations above can then be calculated from the radiosity by

$$q_j^l = \frac{(J_j^l - \varepsilon_j^l \sigma T_j^4)}{\rho r_j^l}, \qquad q_j^s = \frac{J_j^s}{\rho r_j^s} \qquad (12)$$

The incident short-wave radiation q_s is externally imposed (from the sun) and the incident long-wave radiation σT_{eff}^4 is calculated from the externally imposed radiation temperature, which depends on the surroundings.

NUMERICAL MODEL

For window cavities the natural-convection phenomena (in both frame and glazing), as described by Equations (1), (2), (3), and (4), the momentum and energy equations are coupled through the body force term which depends on the temperature field. The solid portions (glazing layers, edge-seal, and frame), also depend on the temperature field as described by Equation (6). As a result, all the aforementioned equations must be solved simultaneously and hence represent an additional level of complexity to obtaining a solution.

A Finite Volume Method (FVM) (Patankar, 1980) using a collocated pressure arrangement on a non-orthogonal grid (Rhie, 1981) was used in this work. In this formulation the approximate equations are obtained through conservation balances of the conserved property (mass, momentum, enthalpy, etc.) in the elemental control volume. Heat transfer and fluid flow problems require the solution of general conservation equations (Patankar, 1980) of the form

$$\frac{\partial}{\partial t}(\rho\Phi)+\frac{\partial}{\partial x}(\rho u\Phi)+\frac{\partial}{\partial y}(\rho v\Phi)=\frac{\partial}{\partial x}\left(\Gamma^{\Phi}\frac{\partial\Phi}{\partial x}\right)+\frac{\partial}{\partial y}\left(\Gamma^{\Phi}\frac{\partial\Phi}{\partial y}\right)+\dot{S}^{\Phi} \qquad (13)$$

over some specified problem domain and with the appropriate boundary conditions for the solution variable which may represent any conserved quantity.

Grid Description. The aim of the finite volume method is to replace Equation (13) with a set of algebraic equations involving the values of Φ at a finite number of discrete control volumes and to preserve conservation throughout. The nodes are located inside the control volumes which collectively constitute the solution domain. For sake of simplicity Figure 1 represents one example for a window system (glazing, seal and frame) with a straightforward geometry. To handle the irregular shape of the window system, as shown in Figure 1, some control volumes are blocked off, so that the remaining active control volumes form the desired shape.

Radiation Exchange. Due to the non-existence of discretization points P for temperature at the internal boundaries of the cavities, the long-wave radiation exchange will be included in the energy equation as a source term of the control volumes adjacent to the boundary. Equation (11) can be put in the matrix formulation

$$[A] \bullet [q] = [T] \qquad (14)$$

and the irradiance q can be found by inverting the coefficient matrix A. The coefficient matrix depends only on the geometric configuration of the enclosure. Therefore the rate of energy generation per unit volume at any wall control volume is

$$\dot{S} = \frac{q}{\Delta x} \qquad (15)$$

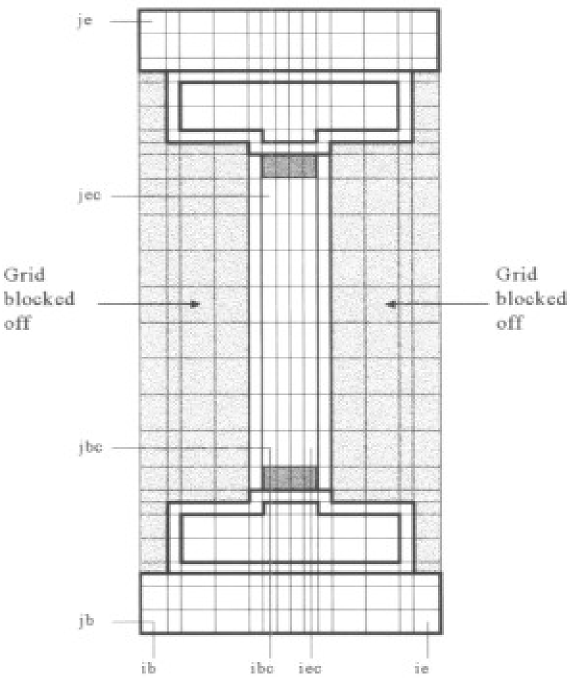


Figure 1: The Collocated Grid.

Also, the solar radiation transmitted to the cavities and absorbed by the walls is placed as a source term in Equation (15). The solar radiation absorbed by the outside surfaces of the window is placed as a source term for the semi-transparent medium in the centre control volume and for the opaque surfaces as flux.

Throughout the current study the grid aspect ratio will depend on the insulated glazing unit (IGU) being simulated, varying between 5 to 20 and the number of control volumes will be determined by a grid independence study for each case studied.

Solution Procedure. The numerical model can use three different schemes for modeling the convective and diffusive fluxes of energy and momentum at each control volume face (Patankar, 1980): first-order upwinding differencing scheme (UDS), central differencing scheme (CDS), and exponential differencing scheme (EDS). A non-staggered pressure grid arrangement was applied to the flow field. In the presence of multicellular flows in a thermally driven cavity CDS is used, because EDS damps the appearance of these multicellular flows for a fairly wide range of parameters (Drummond *et al.*, 1991).

There are two iteration loops. In the main iteration (or coefficient update loop), outer loop, the coefficients are evaluated from the most recent solution fields and the set of linear equations for temperature (T) is solved iteratively, using a line by line solver based on an Alternating Direction Implicit (ADI) procedure, along with an over relaxation procedure. Iterations are performed on the linear equations for T until a preset level of residual reduction (4 levels in all cases thereafter) or until a maximum number of iterations is reached. Within each coefficient update loop, an inner loop is called to solve the velocities equations, u and v, and the pressure equations, p, which corresponds directly to the SIMPLEC algorithm (Hutchinson and Raithby, 1986). The numerical solution of each of these variables is carried out using an ADI procedure, using the same criterion described for the temperature equations. An Additive Correction Method (Hutchinson and Raithby, 1986), ACM, block correction is applied to speed the solution for p. Also, the ACM algorithm can be used to accelerate convergence for T, u, v when necessary. At the end of each outer loop iteration, steady state convergence is verified by comparing the relative maximum change T, u and v between two consecutive iterations. Finally, the condition of steady state will only be satisfied if the error between the heat flow in and out of the window accomplish the following criteria:

$$\left|\frac{Q_{in} - Q_{out}}{Q_{out}}\right| \leq 10^{-5}$$

between two consecutive iterations and

$$Q_{in}^{n+1} - Q_{in}^{n} \leq 10^{-5}$$

for at least four consecutive iterations.

It is worthwhile to mention that the time step for solving natural convection problems depends on the grid size and the Rayleigh number because of the weak coupling between momentum and energy equation due to buoyancy, that affects in turn the delicate u, v and p coupling. If the time step is too small the evolution of the solution is prone to oscillations due to decoupling between the energy and momentum equations, and consequently the decoupling in u, v and p equations. The time step is obtained by multiplying the natural time scale (maximum time step allowed for numerical stability based on an explicit scheme) by a factor n, i.e.:

$$\Delta t = n\Delta t_{\max}$$

The energy balance is applied over the entire problem domain with the thermal boundary conditions applied at the outer edges. This step in the coefficient update loop generates a temperature solution for all control volumes. However, all velocities at locations outside the cavity are set to zero and the velocity boundary conditions, used in solving balances to conserve momentum and mass, are applied at the walls of the cavity. Thus, the natural convection velocity field in the cavity is generated for non-uniform temperature distributions at the cavity walls.

Application of the energy balance across the composite problem domain required care to ensure that the heat flux is correctly calculated at the surfaces between materials of different thermal conductivities. The conductive heat flux at the bordering faces is calculated using the harmonic mean of the neighboring conductivities according to the procedure outlined by Patankar (1980).

Assessment of the numerical method. The capabilities of the solution method to model windows were examined, with respect to accuracy, cost, and reliability. In order to fulfill this goal, code validation was divided into three stages, with increasing degree of complexity. Each stage was designed to examine and to validate a particular feature of the code. For the pure natural convection problem the accuracy and reliability is satisfactory to model natural convection alone in orthogonal and non-orthogonal grids. In comparisons with bench mark solutions, the errors are lower than 10^{-2}% and the ability to correctly model secondary and tertiary cells by perturbing the flow is demonstrated. For the conjugate problem (conduction coupled with natural convection) a set of five runs is made and the results are compared with Wright's work (1990) for cavity aspect ratio, A=40, width 0.0127 mm and $6 \times 10^{3} \leq Ra \leq 1.2 \times 10^{4}$. The same grid, same numerical scheme (CDS), and same convergence criteria are used by both programs. The results are compared and for all runs the errors are less than 1% for heat flux through the hot wall. Finally, for the interaction among surface radiation, natural convection and convection, which includes all modes of heat transfer present in a window system, surface-to-surface radiation exchange is shown to be accurately modeled. Differences are found in the third decimal place for the radiation heat flux when compared to Wright's code (Wright 1990). The solar heat gain for normal radiation were compared with the experimental data of Harrison and Wonderen (1994) for 9 window samples, with errors for the solar heat gain factor lower than 2.5% for three samples, 5.3% , 19% and 12% for the others. The results for this comparison are presented in the next section. The simulation procedure proves satisfactory to model the heat transfer and fluid dynamics in fenestration systems.

RESULTS

Solar Radiation Interaction. To validate the model in relation to the interaction of radiation, conduction and natural convection when solar radiation is present, no data was found in the literature showing local effects in the fluid flow and in the heat transfer mechanisms. Experimental and numerical results of solar heat gain factor (*SHGF*), an averaged quantity, for a set of nine window samples (Harrison and Wonderen, 1994) are therefore used to verify the model. The *SHGF* used together with the window's thermal transmittance (*U-factor*) permits comparative ranking of fenestration systems, calculating Energy Rating (ER) numbers, and determining the annual energy performance.

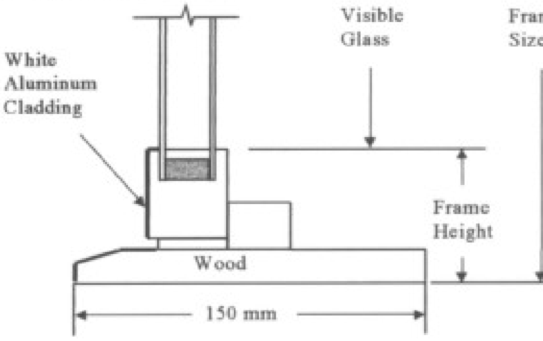
The nine window samples selected represent a broad range of insulating glazing systems that include clear and heat absorbing glass, reflective and spectrally selective coatings. All samples are specified to be double-glazed air-filled sash units fitting in a common wood casement frame, see Figure 2. Details of the nine glazing systems are given in Table 1.

Sash No.	Glazing System
S1	Clear / Clear
S2	Clear / Pyrolytic low-E, ε=0.2
S3	Clear / LoE-178, ε=0.08
S4	Clear / LoE Sun, ε=0.1
S5	LoE Sun-145, ε=0.1 / Clear
S6	Evergreen Tinted / Clear
S7	Reflective Metal SS-114 / Clear
S8	Clear / Low-E^2-171, ε=0.04,
S9	Evergreen Tinted / Pyrolytic Low-E, ε=0.2

Table 1: Description of Test Glazing Systems.

The *SHGF* measurements were performed at the CANMET Window Test Facility (Harrison and Wonderen, 1994) and the average measurement uncertainty was $\pm$ 4.9%. They did numerical simulations using the VISION/FRAME (EEL 1995, AGSL 1995) computer program package. Results for the nine windows are presented in Table 2.

The results shown in Table 2 indicate good agreement among measured results and those calculated values using FRAME/VISION and this model. They agree to about 2.5% or less for samples S1 to S5 and S9, 5.3% for S6, 19% for S7, and 12% for S8. The results of the model are consistently higher than those from FRAME/VISION. This can be explained by the approximation used by FRAME to calculate the *SHGF* (see Harrison and Wonderen, 1994). The values of center-glass *SHGF* calculated by the model are nearly the same as those calculated by VISION as expected because both programs use the Edwards' embedding technique to calculate *SHGF*.



Frame Size 1200 x 600 mm
Visible Glass 1045 x 445 mm
Frame Height 77.5 mm

Figure 2: Test Frame Sash Cross Section.

Sash No.	Solar Heat Gain Factor		
	Measured	VISION/FRAME	Model
S1	0.50 ± 0.02	0.49	0.495
S2	0.48 ± 0.02	0.46	0.468
S3	0.41 ± 0.02	0.40	0.404
S4	0.36 ± 0.02	0.35	0.356
S5	0.21 ± 0.02	0.21	0.212
S6	0.34 ± 0.02	0.31	0.322
S7	0.12 ± 0.02	0.09	0.097
S8	0.29 ± 0.02	0.32	0.325
S9	0.30 ± 0.02	0.29	0.296

Table 2: Summary of Results of Solar Heat Gain Factors.

Surface Temperature Profiles. This section is a brief description of a collaborative research project, including both measurement and simulation studies, aimed at determining the surface temperature of a set of insulated glazing units (IGU's). Duplicate sets of glazing units were provided to two measurement laboratories and their construction details were given to two simulation laboratories, one of which is the research group that the authors are part of. All participants in the study were asked to perform their portion of the research without knowledge of the results of the other investigators, i.e., a "blind study". A complete description of this collaboration among four laboratories is given in Sullivan *et al.* (1996).

The numbering scheme shown in Table 3 will be used throughout this section to identify the glazing units. The design options incorporated in these glazing units were selected to cover a range of edge-seal type, pane spacing, low-E coating, number of glazings that would reasonably test both the measurement and simulation techniques. These design options are of interest because each one is expected to affect the indoor surface temperature profile.

The boundary conditions used in the simulations are given in Table 4. The heat transfer coefficients include both long wave radiation and convective effects. No solar radiation is present. These boundary conditions approximate the American Society of Heating, Refrigerating and Air Conditioning (ASHRAE) winter design condition with a 6.7 m/s wind on the cold side and natural convection on the warm side but were chosen with the expectation that they would also approximate the conditions in the measurement laboratories.

IGU#	Glass Description	Pane Spacing(s), d	Spacer(s)
1	Clear double	12.7 mm	Foam
2	Clear double	12.7 mm	Aluminum
3	Clear double	6.4 mm	Foam
4	Clear double	19.1 mm	Foam
5	Low e double	12.7 mm	Foam
6	Clear triple	12.7 mm	Foam
7	Clear triple	6.4 mm	Foam

Table 3: Description of Blind Test Glazing Units.

Indoor Temp T_i	Indoor Heat Transfer Coefficient h_i	Outdoor Temp T_o	Outdoor Heat Transfer Coefficient h_o
21.1°C	8.3 W/(m²°C)	-17.8°C	30 W/(m²°C)

Table 4: Blind Test Glazing Unit Boundary Conditions.

The horizontal axis is vertical distance with zero corresponding to the bottom edge of the glazing unit and the top edge at 508 mm. The vertical axis records the temperature profile in degrees Celsius on the warm side (room side) of the glazing unit. This temperature profile represents any vertical profile along the glazing unit as long as the "side effects", i.e., 3-D effects, of the spacer, sealant, etc. are not influencing the temperature.

In Figure 3 it is relatively easy to sort out the different temperature profiles on the basis of their center-glass heat transfer characteristics. For example, in the mid-part of the graph, away from the top and bottom edge effects, the lowest performing unit is the double glazed 6.35 mm unit, and the best performing units are the clear triple unit with the largest spacings (two 12.7 mm air cavities) and the 12.7 mm low-E unit. The remaining units occupy a position consistent with intuition.

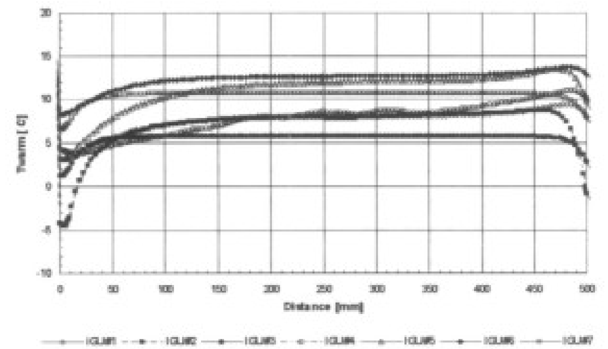


Figure 3: Warm Side Temperature Profile for All Seven Blind Test Glazing Units.

The comparison between simulations and experiments for all of the data sets show good agreement. It can be seen in de Abreu (1996), even without adjustment, that each set of curves falls within a band typically no wider than ±1°C. Exceptions occur in the vicinity of steep temperature gradients as seen near the top and bottom of the glazings where the observed temperature band can be as large as ±3°C. However, these discrepancies are misleading knowing that some spatial uncertainty exists regarding the location of measured profiles - a topic which is discussed in more detail in Sullivan *et al*. (1996). Figure 4 shows a comparison for a IGU #6, clear triple 12.7 mm foam spacer.

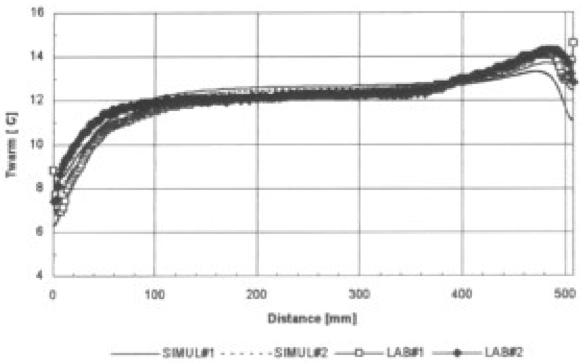


Figure 4: Temperature Profiles, IGU #6, Clear Triple 12.7 mm Foam Spacer.

CONCLUSIONS

The model presented here is a non-orthogonal grid, control-volume based, computational code developed specifically for the two-dimensional modelling of window. One strength of the code is its ability to model an entire window assembly (centre-glass, edge-glass, frame, and wall) simultaneously. This is in contrast to current industry practice of modelling the centre-glass region separately from the edge-glass/frame/wall region as practised by the Canadian Standards Association's (CSA) window energy rating (ER) procedure and the National Fenestration Rating Council procedure (de Abreu, 1996).

This model incorporates much of the appropriate physics, it yields detailed surface temperature profiles, it provides flexible boundary condition capabilities, and it readily admits modification as refinements are developed. For example, no assumptions concerning the location of edge-glass demarcation is needed; a detailed vertical temperature profile along the entire window is available for comparison with thermographic measurements; specification of an indoor window surface temperature or heat transfer coefficient can be replaced with room air convection; and, different algorithms modelling specular reflection of solar radiation can be tested.

The ability of this model to accurately determine local window temperatures makes it possible to be used to supplement experimental data in validating simpler, more user friendly, codes (EEL 1995, AGSL 1995).

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