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**THE HARMONIZATION OF CANADIAN AND U.S.
WINDOW PROGRAMS AND STANDARDS**

DEPARTMENT OF ENERGY
GRANT # DE-FG36-94G010044

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FINAL REPORT

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THE HARMONIZATION OF CANADIAN AND U.S. WINDOW PROGRAMS AND STANDARDS

**DEPARTMENT OF ENERGY
GRANT # DE-FG36-94G010044**

**Comparison of U.S. and Canadian Thermal Performance Procedures,
Standards, and Accreditation and Certification Programs
for Fenestration Products**

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May 1, 1995

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THE HARMONIZATION OF U.S. AND CANADIAN WINDOW PROGRAMS AND STANDARDS

COMPARISON OF U.S. AND CANADIAN THERMAL PERFORMANCE PROCEDURES, STANDARDS, AND ACCREDITATION AND CERTIFICATION PROGRAMS FOR FENESTRATION PRODUCTS

MAY 1, 1995

INTRODUCTION

The window and door industry in North America have a strong interest in harmonizing U.S. and Canadian rating procedures and standards for predicting the thermal performance of their products. The costs and time associated with complying with the requirements in both countries are significant and could be reduced through reciprocity. The purpose of this paper is to compare the U.S. and Canadian rating programs. Joint efforts between the U.S. and Canada can use this information to focus on reconciling the differences and to work towards reciprocity.

The National Fenestration Rating Council (NFRC) in the U.S. and the Canadian Standards Association (CSA) in Canada have developed methods for predicting the energy performance of fenestration products. NFRC is mandated by the federal government to develop fair and accurate rating systems for fenestration products, along with a certification program. CSA's function is to develop standards for predicting the performance of products, and the Canadian Window and Door Manufacturers Association certifies products under the various standards.

The following identifies the applicable procedures and standards for both countries, and summarizes the major differences in the products that are covered, the sizes that are used, the U-factor and solar heat gain coefficient calculational and test methods, air leakage measurement, prediction of annual energy use, condensation resistance methods, reporting requirements, and accreditation and certification programs. The major differences between the systems are identified and recommendations are made to achieve reciprocity. The appendix contains a paper that is a synopsis of the findings from this study.

1. APPLICABLE NFRC PROCEDURES AND CSA STANDARDS

The National Fenestration Rating Council in the U.S. and the Canadian Standards Association in Canada have developed procedures and standard for rating the thermal performance of fenestration products. Table 1 lists them.

Table 1 - Identification of Procedures and Standards for Rating Energy Performance

Procedure/Standard	NFRC	CSA
Thermal (U-Factor) Calculation	NFRC 100-91, NFRC 100-94 (draft), NFRC 100-94: Attachment B (doors)	CSA A440.2-93 CSA A453 (doors)
Thermal (U-Factor) Test	NFRC 100-91 - Attachment A, NFRC 100-94 - Attachment A	CSA A440.2-93 CSA A453
Solar Heat Gain Calculation	NFRC 200-93	CSA A440.2-93 CSA A453
Solar Heat Gain Test	None currently	CSA A440.2-93 CSA A453
Measurement of Optical Properties	NFRC 300-93	ASHRAE Standard 74 (CSA A440.2-93)
Air Leakage	NFRC 400-94	CSA A440.2-93 CSA A453
Annual Energy Performance	NFRC 900-95 (draft)	CSA A440.2-93 CSA A453
Condensation Resistance	Under Development	CSA A440.1-90

Within each of the procedures and standards, computer programs, calculational methodologies, and test standards are referenced. These are discussed in detail under the individual topics.

2. PRODUCTS COVERED AND NOT COVERED

Table 2 lists all products which are and are not covered by the NFRC procedures and CSA standards presented in Table 1. Some of the products listed in Table 2 are sited as being covered in the documents given in Table 1; however, special conditions apply or working procedures are being developed currently.

Table 2 - Products Covered and Not Covered

Fenestration Product	NFRC	CSA	Fenestration Product	NFRC	CSA
Vertical Slider	YES	YES	Transoms ⁷	YES	YES
Horizontal Slider	YES	YES	Store Fronts	NO	NO
Casement	YES	YES	Transparent Portions of Curtainwalls ⁸	NO	NO
Projecting (Awning)	YES	YES	Opaque Portions of Curtainwalls ⁹	NO	NO
Fixed	YES	YES	Fixed Glazing Cast into Precast Concrete Panels ¹⁰	NO	NO
Single Door w/ Frame ¹	YES	YES	Storm Doors ¹¹	NO	NO
Double and Multiple Doors w/ Frames ²	YES	YES	Curved Glazing ¹²	NO	NO
Skylights and Sloped Glazing ³	YES	NO	Glass Block ¹³	NO	NO
Greenhouse / Garden ⁴	NO	NO	Garage Doors ¹⁴	NO	NO
Dual Action	YES	YES	Integral Shading Devices ¹⁵	YES	YES
Pivoted	YES	NO	Removable Shading Devices ¹⁶	NO	?
Combination and Composite Windows	YES	YES	Grilles ¹⁷	YES	YES
Bay or Bow Windows ⁵	YES	YES	True Divided Lites ¹⁸	YES	YES
Sidelites ⁶	YES	YES	Varying Glass Thicknesses ¹⁹	YES	NO

The special conditions and procedures include the following:

1. NFRC 100-94, Section B is the rating procedure for swinging and fixed doors and sidelites. The CSA standard is CSA A453.

-
2. NFRC 100-91 and 100-94 list an AA and a BB size for double doors. These sizes are used only for sliding doors which is what the CSA size is defined for. NFRC 100-94, Section B offers the option to rate swinging, double doors as single doors at the AA size (see below).
 3. NFRC 100-91 and 100-94 list skylights and sloped glazing as being covered. It was recently clarified that the rough opening shall be used for rating these products.
 4. NFRC 100-91 and 100-94 list greenhouse and garden windows as being covered but a working procedure has not been approved.
 5. NFRC 100-91 and 100-94 list bay and bow windows as being covered. Individual units within the product are rated separately, e.g. two casements mulled together are rated as individual casement units. CSA classifies bay and bow windows as composite and combination windows which shall be rated as individual units as in the NFRC procedure.
 6. Sidelites are now covered under NFRC 100-94 - Section B. CSA classifies window sidelites as fixed windows in A440.2, and door sidelites are addressed in CSA A453.
 7. Transoms are covered under NFRC 100-94. CSA classifies transoms as fixed windows in A440.2.
 8. NFRC is developing a procedure for rating curtainwalls and solariums.
 9. NFRC is planning to cover all curtainwall systems but at this time does not have an approved procedure.
 10. Fixed glazing cast into precast concrete walls is not explicitly addressed by NFRC.
 11. Storm doors are not explicitly excluded from NFRC 100-94 - Section B.
 12. Curved glass is not explicitly excluded from NFRC 100-91 or NFRC 100-94, but it is excluded from NFRC 200-94.
 13. Glassblock is included in NFRC 100-91 or NFRC 100-94, but it is excluded from NFRC 200-94.
 14. NFRC is developing a U-Factor procedure for rating garage doors.
 15. Integral shading devices are covered by NFRC 100-91 and 100-94 as well as CSA A440.2-93. For solar heat gain calculations the devices are retracted. For U-Factor calculations the CSA standard requires that the device be retracted to the position of maximum

viewing area. NFRC does not specify the position.

16. Removable shading devices are not covered by NFRC or CSA. NFRC has formed an attachment group to address such devices.

17. Default patterns for internal grilles are given in NFRC 100. NFRC 100 specifies a minimum of 12 inches between the grilles. NFRC 100 also states that if there is more than 1/8" gap between the glass and the grilles than the grilles do not need to be modelled. This has yet to be clarified in NFRC 200. CSA A440.2 is not clear on how to address grilles. It has been proposed to CSA that they adopt the method used in A453, which is slightly different from NFRC (see discussion on door sizes in the following section).

18. The default pattern for true-divided lites is given in NFRC 100 and is the same as that specified for internal grilles. NFRC 100 specifies a minimum of 12 inches between the dividers. CSA 440.2 is not clear on how to address grilles. It has been proposed to CSA that they adopt the method used in A453, which is slightly different from NFRC (see discussion on door sizes in the following section).

19. In NFRC 100-91, different glass thicknesses constitute individual glazing options. In CSA A440.2, glass thickness is not directly addressed; although, accepted practice is to model the glass thickness offered in the size being modelled. The difference in U-factor results can be significant for insulated glazing units with an overall width of 5/8" or less.

3. CORRESPONDANCE OF SIZES

Table 3 lists the model sizes specified in the NFRC and CSA documents. The most noticeable difference is that NFRC specifies two sizes: AA and BB, or residential and commercial (NFRC 100-94). CSA only specifies one size which corresponds to the NFRC AA size.

TABLE 3 - COMPARISON OF NFRC AND CSA STANDARD SIZES

Window Type	NFRC AA		NFRC BB		CSA ¹	
	Width	Height	Width	Height	Width	Height
Vertical Slider	36	60	48	72	36	60
Horizontal Slider	60	36	72	48	60	36
Casement	24	48	30	60	24	48
Projecting (Awning)	48	24	40	40	48	24
Fixed	48	48	48	72	48	48
Single Door w/ Frame ²	36	80			36	80
Double and Multiple Doors w/ Frames ³	72	82	80	80	72	82
Skylights and Sloped Glazing ⁴	24	48	48	48		
Greenhouse / Garden ⁵	60	36	72	48		
Dual Action	36	60	48	72	36	60
Pivoted ⁶	48	48	48	72		
Sidelites ⁷	14	80			48	48
Transoms ⁸	38	14			48	48

The differences between the types and sizes of fenestration products rated by the NFRC procedures and the CSA standards are tagged within Table 3 and are the following:

1. CSA sizes are given as ± 100 mm and are required for one size only. The NFRC AA size corresponds to the CSA size for all products rated by both systems.
2. CSA Standard A453 addresses swinging doors and is in draft form currently. The NFRC size is the nominal size of the door slab.

3. NFRC 100-91 lists an AA and a BB size for double doors. These sizes are used only for sliding doors which is what the CSA size is defined for. NFRC 100-94, Section B offers the option to rate swinging, double doors as single doors at the AA size.
4. Skylights and sloped glazing are not covered by the CSA standard.
5. Greenhouse and garden windows are not covered by the CSA standard and a working procedure has not been adopted by NFRC as of the end of 1994.
6. Pivoted windows are not covered by the CSA standard.
7. Door sidelites are covered in NFRC 100-94, Section B. The size is the nominal size. CSA rates window sidelites at the fixed window size in A440.2. CSA A453 specifies a door sidelite size and lite sizes.
8. Door transoms are covered in NFRC 100-94. CSA rates transoms at the fixed window size.

Door-lite sizes have been specified for swinging doors and sidelites in NFRC 100-94, Section B and CSA A453. These sizes refer to the daylighting opening. Door-lite sizes are shown in Table 4 and sidelite sizes are shown in Table 5.

Table 4 - Door Lite Sizes

Size	NFRC		CSA	
	width (in)	height (in)	width (mm)	height (mm)
None	-	-	-	-
1/4-Lite < 410 in ² CSA: 0.01-0.12 m ² (16-186 in ²)	22	19	550 (22 in)	200 (8 in)
1/2-Lite > 410 in ² , < 900 in ² CSA: 0.12-0.60 m ² (186-930 in ²)	22	41	550 (22 in)	900 (36 in)
3/4-Lite > 900 in ² , < 1100 in ²	22	50		
Full-Lite > 1100 in ² CSA: 0.60 m ² (930 in ²)	22	64	550 (22 in)	1600 (64 in)

Table 5 - Sidelite Lite Sizes

Size	NFRC		CSA	
	width (in)	height (in)	width (mm)	height (mm)
None	-	-	-	-
1/4-Lite < 65 in ² CSA: 0.01-0.044 m ² (16-68 in ²)	8	8	200 (8 in)	200 (8 in)
1/2-Lite > 65 in ² , < 350 in ² CSA: 0.044-0.22 m ² (68-340 in ²)	8	36	200 (8 in)	900 (36 in)
3/4-Lite > 350 in ² , < 460 in ²	8	50		
Full-Lite > 460 in ² CSA: 0.22 m ² (340 in ²)	8	64	200 (8 in)	1600 (64 in)

CSA A453 also specifies a grille pattern for door lites with grilles. The standard requires 2 vertical grille bars for all lite sizes. This is one more than what is required by NFRC 100-91 for a 22 inch width. CSA A453 requires no horizontal grilles for the minimal glazing case, 2 for the 1/2-lite, and 4 for the full-lite. This is comparable to the NFRC 100-91 requirements.

4. THERMAL (U-Factor) CALCULATIONAL METHODS

The NFRC U-Factor procedure and CSA U-Factor standard reference computer programs for the determination of fenestration U-Factor. The calculational methods in the NFRC procedures and the CSA standards are very similar. The programs are listed in Table 6.

Table 6 - Referenced Computer Programs

Calculation	NFRC	CSA
Center-of-glass, one-dimensional heat transfer	WINDOW 4.1	VISION 3
Frame and edge, two-dimensional heat transfer	FRAME 3.1	FRAME 3.1
Total Fenestration Properties	Approved area-weighted calculation. Can be done in WINDOW 4.1, spreadsheet, or by hand	Approved area-weighted calculation. Can be done on spreadsheet or by hand

The climatic conditions used by NFRC and CSA are identical: 0°F (-18°) outdoor temperature, 70°F (21°C) indoor temperature, and a 15 mph windspeed on the exterior which translates to an outside film coefficient of 5.1 Btu/hr-ft²-°F (30 W/m²-°C).

The only differences which arise in the calculational methods are those between the WINDOW and VISION programs. WINDOW and VISION use essentially the same approach to calculate the convective and radiative heat transfer across parallel planes. There is a minor difference in the conductive calculations which has no effect on U-Factors. The differences in the programs are listed in Table 7. Figure 1 compares the U-factors predicted by the two program for a double-pane glazing units with clear, 1/8" (3 mm) glass. The programs agree to within 2% over the range of gap thicknesses shown. For more detailed information see Wright (1994), Wright (1992), and Finlayson (1993).

Table 7 - Differences between WINDOW and VISION

Calculation	WINDOW	VISION	Difference
Interior and exterior film coefficients	same correlations	same correlations	none
Correlations for convective heat transfer between vertical glazing layers	ElSherbiny calculation at A=40	Wright (1991)	2% in Nusselt No.
Correlations for convective heat transfer between glazing layers at a tilt	same correlations	same correlations	none: except for tilt angles btw. 60° and 180° the results will differ slightly because they are function of the Nusselt No. calculated at 90°.
Gas properties	Linear curve fit to measured data for all	Ideal gas law @ 1 atm. for density; linear curve fit for all others	less than 1% difference in density
Optical Properties	Two-band and multi-band models, same optical data	Two-band model, same optical data. VISION 4 has multi-band model.	No effect on U-Factor

Fig. 1 Comparison of WINDOW 4.1 and VISION 3 U-Factors for a double-pane window with 1/8" clear glass and varying gap thicknesses.

5. THERMAL (U-Factor) TESTING METHODS

Thermal testing is required by the National Fenestration Rating Council to validate computer simulations. For each product line, the units with the best and worst case simulations are tested to validate the simulations. The results must agree to within $\pm 10\%$ or 0.04 Btu/hr-ft²-°F, whichever is greater. A product line is defined by operator type and frame type, although there are circumstances under which different operator types and frames can be validated by the same tests. The procedure by which fenestration products are tested for thermal performance is NFRC 100-91: Attachment A, Interim Standard Test Method for Measuring the Steady-State Thermal Transmittance of Fenestration Systems Using Hot Box Methods.

The Canadian Standards Association CSA A440.2-93 describes both measurement and calculation of U-Factors and either approach can be taken, but there is no validation requirement between tests and simulation. Manufacturers have chosen to use the calculation procedure rather than test units in almost all cases.

CSA references ASTM C1199-91, Test Method for Measuring the Steady-state Thermal Transmittance of Fenestration Systems Using Hot-Box Methods for thermal testing. ASTM C1199-91 is draft 10 of the NFRC 100-91: Attachment A document. The adopted version of NFRC 100-91: Attachment A is draft 11 of the document.

Essentially the two organizations are using the same test method. ASTM will update their test standards in 1996, and it is expected that the two organizations will be using the exact same document. There are slight differences in the outside and inside film coefficients which accounts for about a 1% difference in the results in highly conductive products (single-pane, aluminum) and less of an effect on low-conductivity products.

Table 8 - Comparison of Thermal Test Methods

CONDITION	NFRC	CSA	DIFFERENCE
Test	NFRC 100-91: Attachment A ASTM C236 ASTM C976	ASTM C1199 ASTM E1423	ASTM C1199 is draft 10 of NFRC 100-91: Attachment A. Cuurent NFRC document is draft 11.
Air Leakage	Seal on inside	Seal on inside	None
Outside Temperature	0 °F	-18°C±1 (0 °F±1)	None
Inside Temperature	70 °F	21 °C±1 (70 °F±1)	None
Windspeed	15 mph	6.7 m/s (15 mph)	None
Standardized Outside Film Coefficient	5.1 Btu/hr-ft²-F (29.0 W/m²-K)	30.0 W/m²-K (5.3 Btu/hr-ft²-F)	Slight
Standardized Inside Film Coefficient	Calculated	8.3 W/m²-K (1.46 Btu/hr-ft²-F)	The CSA standard fixes this value and ignores differences in surface temperatures and surface emittances.
Air Leakage	Seal unit with tape on inside to eliminate	Seal unit with tape on inside to eliminate	None
Solar Radiation	Exclude	Exclude	None
Wind Direction	Parallel or Perpendicular		
Sizes	AA or BB	AA (adjust to AA for reporting)	NFRC is more restrictive for validation reasons.

6. SOLAR HEAT GAIN CALCULATIONAL METHODS

NFRC 200-94 and CSA A440.2-93 and A453 reference computer programs for the determination of the fenestration solar heat gain coefficient. The calculational methods in the NFRC procedures and the CSA standards are very similar. The programs are listed in Table 9.

Table 9 - Referenced Computer Programs

Calculation	NFRC	CSA
Center-of-glass solar heat gain coefficient	WINDOW 4.1	VISION 3
Edge-of-Glass solar heat gain coefficient	Equal to center-of-glass value	Equal to center-of-glass value
Frame U-Factor (from which frame solar heat gain coefficient is calculated)	FRAME 3.1	FRAME 3.1
Total Fenestration Properties	Approved area-weighted calculation. Can be done in WINDOW 4.1, spreadsheet, or by hand	Approved area-weighted calculation. Can be done on spreadsheet or by hand

The NFRC and CSA procedures agree on the form of the calculation of the center-of-glass solar heat gain coefficient and on the calculation of the solar heat gain coefficient for a complete fenestration system. The solar heat gain coefficient for a fenestration system is the area-weighted summation of the solar heat gain coefficient of the center-of-glass, edge-of-glass, and frame elements. The solar heat gain coefficient (SHGC) of the center-of-glass area equals

$$SHGC = T_{sol} + A_{sol,i} * N_i$$

where T_{sol} is the solar transmittance of the glazing system, $A_{sol,i}$ is the absorbed solar radiation in the i th glazing layer, and N_i is the inward flowing fraction. The solar transmittance and absorptance are calculated from the optical properties of the individual glazing layers in a glazing system, and the inward flowing fraction is a function of the system U-Factor.

The differences which arise between the NFRC and CSA solar heat gain coefficient

calculational methods are those between the evaluation conditions, the WINDOW and VISION programs, and in the determination of the frame solar heat gain coefficient.

Evaluation Conditions

The primary difference in the evaluation conditions is that NFRC specifies summer environmental conditions for the calculations and CSA specifies winter conditions. Table 10 gives a detailed list of the evaluation conditions.

Table 10 - Differences in Evaluation Conditions

Condition	NFRC	CSA	Difference
Outdoor Air Temperature	89 °F	-18 °C (0 °F)	NFRC is ASHRAE summer condition. CSA is ASHRAE winter condition.
Indoor Air Temperature	75 °F	21 °C (70 °F)	NFRC is ASHRAE summer condition. CSA is ASHRAE winter condition.
Exterior Film Coefficient	7.5 mph wind	30.0 W/m ² -K 5.1 Btu/hr- ft ² -F	NFRC is ASHRAE summer condition. CSA is ASHRAE winter condition.
Interior Film Coefficient	Calculated	Calculated	Same calculation. Difference is in the air temperatures used in the calculation (see above).
Solar Radiation Intensity	248 Btu/hr-ft ²	783 W/m ² (248 Btu/hr-ft ²)	Same.
Angle of Incidence	0°	0°	Same.
Solar Optical Properties	NFRC 300-94	ASHRAE Standard 74	Same. WINDOW and VISION are using the same glass manufacturers' optical data and VISION 4 has a multi-band model.
Material Properties	WINDOW and FRAME	VISION and FRAME	Same.

Canada is using winter design conditions because heating loads dominate residential energy use for all of their climate zones. In the U.S., the dominant load varies with climate zone and energy cost. NFRC has chosen to rate SHGC under the worst-case summer conditions for cooling loads, and CSA has chosen to rate SHGC under the worst-case winter conditions for heating loads.

Assuming WINDOW and VISION would use the same optical properties and material properties for a given system, the differences in evaluation conditions change the center-of-glass solar heat gain coefficient by less than 0.02. NFRC values would be higher than CSA values. They are higher because the calculations are performed under summer conditions rather than winter conditions as with the CSA standard. The inward flowing fraction of absorbed solar radiation is higher in the summer than in the winter.

WINDOW and VISION

WINDOW and VISION use essentially the same approach to calculate the convective and radiative heat transfer across parallel planes. There is a minor difference in the conductive calculations which has no effect on U-Factors or solar heat gain coefficients. A significant difference between the two programs is the multi-band model in WINDOW. The currently referenced version of VISION in the CSA standard does not include a multi-band model; however, VISION 4 does have this capability and will be used in the CSA standard in the near future. The multi-band model calculates the total optical properties of a glazing system from spectral data for the individual glazing layers. The two-band model uses averaged properties for the solar and infrared spectra for the individual glazing layers. The difference in the results from the models are negligible with combinations of clear glass and other glazing, but become significant with combinations of more spectrally selective glazing. The differences in the programs are listed in Table 11. For more detailed information see Wright (1994), Wright (1992), and Finlayson (1993).

Table 11 - Differences between WINDOW and VISION

Calculation	WINDOW	VISION	Difference
Interior and exterior film coefficients	same correlations	same correlations	None (except in environmental conditions - see Table 10)
Correlations for convective heat transfer between vertical glazing layers	ElSherbiny calculation at $A=40$	Wright (1991)	2% in Nusselt No.
Correlations for convective heat transfer between glazing layers at a tilt	same correlations	same correlations	None. (see description under U-Factor calculation)
Gas properties	Linear curve fit to measured data for all	Ideal gas law @ 1 atm. for density; linear curve fit for all others	less than 1% difference in density
Air Mass	Air Mass 1.5	Air Mass 1.5	None. Both are using ASTM E891-87 (1987).
Glazing Optical Properties	NFRC 300	ASHRAE Standard 74	WINDOW and VISION using same data.
Total Optical Properties	Two-band and multi-band models, same optical data	Two-band model, same optical data	No effect on U-Factor. Some effect with Multi-band model on SHGC.

Frame Solar Heat Gain

The most significant difference between the NFRC Solar Heat Gain procedure and the CSA standard is the calculation of the frame solar heat gain coefficients. NFRC 200-93 defines the frame solar heat gain coefficient ($SHGC_{f-NFRC}$) as

$$SHGC_{f-NFRC} = k \cdot a \cdot U (\text{Btu/hr-ft}^2\text{-F}) / h$$

where k is an adjustment factor to account for differences between surface area and projected area, a is the absorptance of the frame, U is the frame U-Factor in $\text{Btu/hr-ft}^2\text{-F}$, and h is the exterior film coefficient ($2.94 \text{ Btu/hr-ft}^2\text{-F}$) which assumes a 7.5 mph wind. Note that the exterior film coefficient used in WINDOW 4 for the $SHGC$ calculation excludes radiative exchange. Currently k is assumed to be 1.0. The frame U-Factor is calculated under NFRC 100-91 winter conditions. All opaque elements (i.e. dividers) are treated the same way.

The CSA standard defines the frame solar heat gain coefficient ($SHGC_{f-CSA}$) as

$$SHGC_{f-CSA} = 0.0051 \cdot U (\text{W/m}^2\text{-C})$$

where U is the frame U-Factor in $\text{W/m}^2\text{-C}$. This equation also holds for dividers. Converting this equation to Inch-Pound units gives

$$SHGC_{f-CSA} (\text{i-p}) = 0.029 \cdot U (\text{Btu/hr-ft}^2\text{-F}).$$

Simplifying the $NFRC_f$ equation by assuming k equals 1.0 and that the average frame absorptance is 0.5 gives

$$SHGC_{f-NFRC} = 0.17 \cdot U (\text{Btu/hr-ft}^2\text{-F}).$$

The $NFRC_f$ equation calculates a $SHGC$ for the frame that is almost 6 times as great as that calculated by the CSA_f equation. For example, consider a wood window which has a frame U-Factor of $0.5 \text{ Btu/hr-ft}^2\text{-F}$. The NFRC procedure predicts a frame solar heat gain coefficient of 0.085, and the CSA standard predicts 0.0145. Almost half the difference is due to the fact the NFRC is using summer design conditions and CSA is using winter design conditions for the exterior film coefficient. Recall that the window exterior film coefficient under winter conditions is $5.1 \text{ Btu/hr-ft}^2\text{-F}$. Substituting this into the $SHGC_{f-NFRC}$ equation gives a value of $0.098 \cdot U$ for the $SHGC_{f-NFRC}$ as compared to $0.17 \cdot U$, assuming an absorptance of 0.5.

While the difference between the equations is considerable, the affect it has on the total product solar heat gain coefficient needs to be considered. A clear, double-glazed window has a center-of-glass solar heat gain coefficient of 0.8. Assuming the glazing accounts for 80% of

the window area, the solar heat gain coefficient for the window system is 0.66 using the NFRC procedure and 0.64 using the CSA standard (assuming a wood window with 0.5 for frame absorptance). This difference would be greater with a highly absorbing frame and lower with a less absorbing frame. Also, the lower the center-of-glass solar heat gain coefficient is, the less the difference is between the SHGC calculated under the two methods.

Combining the effects of the differences in the center-of-glass solar heat gain coefficient and the frame solar heat gain coefficient shows an even larger difference in the results. The worst case is a single-pane window with clear glass. WINDOW 4.1 predicts a center-of-glass solar heat gain coefficient of 0.86 and VISION predicts a center-of-glass solar heat gain coefficient of 0.85. Using the example above for the frame solar heat gain, the NFRC 200-93 methodology predicts a total window solar heat gain coefficient of 0.71 and the CSA A440.2-93 methodology predicts a total window solar heat gain coefficient of 0.68. The difference is 3%.

7. SOLAR HEAT GAIN TEST METHODS

Currently, there is no published standard for measuring solar heat gain coefficients. A draft procedure to measure fenestration solar heat gain coefficient has recently been submitted to NFRC (McCluney 1995). This procedure is based on the following references:

ASHRAE Standard 93-1986: Methods for Testing TO Determine the Thermal Performance of Solar Collectors

Harrison, S.J. and Dubrous, F.M., "Determination of Window Thermal Characteristics Using a Solar Simulator Based Test Method," ASHRAE Semi-annual Meeting, Atlanta, February 1990.

Bowen, R.P., "DBR's Approach for Determining the Heat Transmission Characteristics of Windows," National Research Council Canada, 1985

Harrison, S.J. and Barakat, S.A., "A Method of Comparing the Thermal Performance of Windows," ASHRAE Semi-Annual Meeting, Atlantic City, NJ, January, 1983.

ASHRAE Temperature Measurement Standard 41-91.

CSA A440.2 states that the solar heat gain coefficient can be determined by measurement using the Test Method for Determination of Fenestration Solar Heat Gain Coefficient Using Simulated Solar Irradiance. This method is based on the National Solar Test Facility in Mississauga, Ontario. The outdoor and indoor surface conductances must be obtained. The solar heat gain coefficient is reported using the an outdoor surface conductance of $30.0 \text{ W/m}^2\text{-C}$ ($5.1 \text{ Btu/hr-ft}^2\text{-F}$) and an indoor surface conductance of $8.3 \text{ W/m}^2\text{-C}$ ($1.46 \text{ Btu/hr-ft}^2\text{-F}$).

Solar heat gain coefficient test measurements are expensive and for homogeneous materials the calculational methods are typically used. For more complex systems, such as venetian blinds and glazing which redirect or scatters light, a measurement technique needs to be standardized to handle these products. An alternative is to modify a ray-tracing programs to calculate the solar heat gain through complex products. This would be less expensive and more accessible to industry.

8. AIR LEAKAGE

The National Fenestration Rating Council adopted NFRC 400-94 for determining fenestration product air leakage in April, 1995. The procedure uses ASTM Standard E283 as the test method. The Canadian Standards Association also requires that air leakage be measured in accordance with ASTM Standard E283. Table 12 lists the test conditions and differences between NFRC 400-94 and CSA A440 in applying ASTM E283.

TABLE 12 - DIFFERENCES IN AIR LEAKAGE TESTING

	NFRC 400-94	CSA A440	DIFFERENCE
Method	ASTM E283-94	ASTM E283	None.
Differential static test pressure	1.57 psf	75 Pa	None.
Temperature difference	None (as per ASTM E283)	None (as per ASTM E283)	None.
Correction to atmospheric conditions of 29.92 in. Hg, 50% RH, and 69.9 F	Yes		CSA does not specify correction.
Model sizes	Use AA size shown in Table 3 for NFRC	If not measured at Table 3 sizes for CSA, can adjust to these sizes using ratio of crack length	Specified sizes are the same. CSA allows for correction to the model sizes.
Reported rate of air leakage	cfm/ft ² and cfm/ft of operable crack length for operable windows	m ³ /h	Simply convert numbers. However, note difference in sizes discussed above.
Rounding	Measure to 2 decimal points; round up to first decimal	Report L_{75}/A_w to 2 decimal points	

The differences in the measurement of air leakage are inconsequential if measured to NFRC 400-94. These values can be converted to numbers that can be used in CSA A440. However, if a product's air leakage is measured according to CSA A440.2 specifications it may not qualify as having been measured to NFRC 400-94 because of possible size differences. If a product is tested to the size specified in Table 3 of Section 3 or is the closest product available to this size, then it is certifiable under NFRC 400-94. Otherwise, it is not.

Canada has minimum requirements for air leakage through fenestration products. For windows, the minimum requirement is an A2 air leakage classification of CAN/CSA-A440. For doors, the minimum requirement is an A2 air leakage classification of CAN/CGSB 82.1. Swinging doors must have an air leakage of less than 2.54 l/s/m² when tested in conformance with ASTM E 283, or it must be weatherstripped on all edges and have an added storm door. Pre-hung, insulated steel doors must conform to CAN/CGSB 82.5 "Insulated Doors." (CANMET 1994)

9. ANNUAL ENERGY USE

One of the goals of rating all of the aspects of fenestration product performance is to quantify the annual energy use attributable to the various products on the market. The National Fenestration Rating Council has a procedure under development, and the Canadian Standards Association is using the Energy Rating (ER). Both systems are rating systems. They do not assign a cost savings to the window or predict total energy savings. Table 13 summarizes the approaches.

TABLE 13 - ANNUAL ENERGY PERFORMANCE RATING

	NFRC	CSA
Standard or Procedure	NFRC 900-95 (draft)	CSA A440.2
Heating Value	FHR (Fenestration Heating Ratio)	ER (Energy Rating) - heating
Cooling Value	FCR (Fenestration Cooling Ratio)	ERc (Energy Rating) - cooling (draft)
Heating Equation	$FHR = 50.05 - 60.39*U + 10.02*SHGC - 6.00*AL + 11.68*U^2 + 10.00*SHGC^2 - .03*AL^2$ where AL is air leakage in CFM/ft	$ER = 72.20*SHGC - 21.90*U - 0.54*(L_{75}/A)*$ where L_{75} is the average window air leakage rate, m ³ /h
Cooling Equation	$FCR = 61.73 - 2.01*U - 71.30*SHGC - 0.81*AL - 1.34*U^2 - 9.64*SHGC^2 - .03*AL^2$ where AL is air leakage in CFM/ft	$ERc = 110.0*SHGC - 4.5*U - 0.07*L_{75}/A^{**}$
Simulation Program	DOE-2.1E	ENERPASS
Prototype Building	1-story home with Model Energy Code defaults (average insulation levels in wall)	Low-rise residential. Enerpass only used to predict weather dependence.
Weather	TMY Hourly Data. Initially performed for 6 locations. Is being extended to 30 locations	Hours during which air temperature less than 13°C for solar utilization. Averaged for 13 cities with degree-days between 4000 and 6200 °C-days.
Base Case for Comparison	Single-glazed, aluminum window	None.
Window Orientation	Equal distribution	Equal distribution
Window Air Leakage	0.65 cfm/ft is base case	Function of Air Leakage

*Based on average Canadian climate conditions. These are 72.20 for the off-angle incidence factor multiplied by the average solar radiation when heating is required, a 21.9°C outside-to-inside temperature difference, a 1.50 pressure factor, and 5823 heating hours. Refer to EEL

(1994) for climate-dependent equation.

**Based on average Canadian climate conditions for the cooling season. Cooling hours are 1464 °C-days. Refer to EEL (1994) for more information.

For more detailed information on the procedure being developed for the National Fenestration Rating Council refer to Crooks (1995). For more detailed information on Energy Rating (ER) see EEL (1994).

In terms of the relative performance of products, the ranking of a wide range of windows based on the heating ratings, FHR and ER, and cooling ratings, FCR and ER_c are shown in Table 2. The heating ratings rank the windows identically, while the cooling ratings are significantly different in how they rank the windows. The absolute values of the ratings are not close because the FHR and FCR values represent a percentage of heat and cooling energy savings respectively as compared to a home with single-pane, aluminum windows. ER and ER_c represent energy gain (loss if negative) through a window in W/m^2 .

Windows 11 and 12 in Table 2 are hypothetical windows. The purpose of them is to examine the ratings under limiting conditions. The best performing window possible in terms of heating energy savings would have $U=0.0$ and $SHGC=1.0$ (window #11). FHR predicts 72% home energy savings for this window and ER estimates a heat gain of 72 W/m^2 . FHR predicts 50% home energy savings from a window which has no heat loss, $U=0.0$, and no solar heat gain, $SHGC=0.0$ (window #12). ER predicts zero heat loss through the same window and no solar gains. (ER rates window #10 as superior to window #12 because of the solar heat gain. FHR predicts 44% home energy savings for window 10, which is less than what it predicts for window #12.) In terms of cooling energy, FCR predicts cooling energy savings of 62% for window #12 and ER predicts cooling energy use of 0 W/m^2 .

To compare the heating and cooling ratings on the same scale, FHR, ER, FCR and ER_c are normalized to represent the percentage of potential window energy savings. For FHR and FCR, the ratio of the percent energy savings for a particular product to that through the "best performing" product is calculated. ER and ER_c are converted by first calculating the energy savings as compared to a single-pane, aluminum window with clear glass (same as unit used for base case for FHR and FCR). Then the ratio of these energy savings to that through the "best performing" window are computed. Figures 2 through 9 compare the different rating systems for various scenarios.

In comparing FHR and ER, we found the following differences:

1. FHR predicts a greater percent of heating energy savings than the ER does for the ten different windows in Table 2 (Figure 2).

Fig. 2 FHR and ER compared on basis of potential energy savings from a window. Both values have been normalized. FHR has been normalized by energy savings of window 10. ER has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window 10.

Fig. 3 FCR and ERc compared on basis of potential energy savings from a window. Both values have been normalized. FCR has been normalized by energy savings of window 9. ER has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window 9.

Fig. 4 FHR and ER compared on basis of potential energy savings from a window. Both values have been normalized. FHR has been normalized by energy savings of window 10. ER has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window with $U=0.0$, $SHGC=1.0$.

Fig. 5 FCR and ERc compared on basis of potential energy savings from a window. Both values have been normalized. FCR has been normalized by energy savings of window 9. ER has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window with $U=0.0$, $SHGC=0.0$.

Fig. 6 FHR and ER compared on basis of potential energy savings from a window. Both values have been normalized. FHR has been normalized by energy savings of window 10. ER has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window with $U=0.0$, $SHGC=1.0$.

Fig. 7 FCR and ER_c compared on basis of potential energy savings from a window. Both values have been normalized. FCR has been normalized by energy savings of window 9. ER_c has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window with $U=0.0$, $SHGC=0.0$.

Fig. 8 FHR and ER compared on basis of potential energy savings from a window. Both values have been normalized. FHR has been normalized by energy savings of window 10. ER has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window with $U=0.0$, $SHGC=1.0$, and air leakage equal to 0.0.

Fig. 9 FCR and ER_c compared on basis of potential energy savings from a window. Both values have been normalized. FCR has been normalized by energy savings of window 9. ER_c has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window with $U=0.0$, $SHGC=0.0$, and air leakage equal to 0.0.

10. CONDENSATION RESISTANCE

Condensation resistance refers to a fenestration product's potential for forming condensation around the edge-of-glass area. The National Fenestration Rating Council has formed a subtask group to investigate measurement and calculation techniques, and the Canadian Standards Association specifies a procedure for measuring condensation resistance in CSA A440 (1990). NFRC's task group is researching the American Architectural Manufacturers Association (AAMA) standard: Condensation Resistance Factor (CRF) - AAMA 1503.1-88. CSA A440 (1990) references AAMA (1981): Voluntary test method for condensation resistance of windows, doors and glazed-wall sections.

Cardinal IG investigated the application of AAMA 1503.1-88 for the NFRC subtask group on condensation resistance. They found both advantages and disadvantages to the test. The advantages include the following: 1) it provides a relative ranking system, 2) the performance of products under a wide range of climatic conditions can be predicted, 3) a set of controlled environmental conditions are used, and 4) the test is valid for highly conductive fenestration products. The disadvantages to the test are: 1) it is not an absolute rating system, 2) infiltration is excluded which has a significant effect on condensation, 3) test performed under a constant static pressure, and 4) averaging of surface temperatures does not allow a consumer to distinguish between actual performance of products. The minutes of the subtask group meeting held in Montreal, Quebec (NFRC, March 1994) provide more detail on this topic.

There is strong interest in developing a calculational method for predicting condensation resistance. Enermodal Engineering, Ltd. has incorporated algorithms into the FRAME program to model condensation resistance. The condensation resistance model included in FRAME 4.0 is based on the work of John Wright (Wright 1995a). The use of this model, and comparison of simulation and test results are documented in the paper, "Numerical Prediction of Window Condensation Potential," by Alex McGowan (1995).

Wright (1995a, 1995b) developed a two-dimensional model to simulate the heat transfer through insulated glazing units. This model accounts for natural convection, conduction, and radiation exchange between the glazing layers. Wright's model differs from the one-dimensional heat transfer programs WINDOW and VISION in that it is a two-dimensional solution and includes the effects of the spacer assemblies separating the glazing layers.

Wright assumes isothermal planes in his model, as well as constant exterior film coefficients. The model predicts temperatures that are higher than the center-of-glass temperature at the head and lower than the center-of-glass temperature at the sill. Test results show the same pattern; however, the measured temperatures are 2°C to 5°C different from the predicted temperatures (McGowan 1995). The differences are due primarily to the assumptions of isothermal planes and constant film coefficients. Also, the results clearly show a dependence

on frame type. The larger temperature difference between predicted and measured results is seen with metal frames.

McGowan (1995) incorporated Wright's model into the FRAME program and compared the results with the existing FRAME program, finite-element analysis, and test measurements of surface temperatures. The existing FRAME program only accounts for conduction heat transfer. McGowan found good agreement between the finite-element analysis program and the FRAME version using Wright's model. Interesting to note is the strong effect frame type has on the measurements. Highly conductive metal frames show even lower temperatures than are predicted by the models. More than likely this is due to the conduction heat transfer between the glazing and frame. The lower surface temperature at the frame would also depress the local air temperature at the edge.

Wright (1995b) presents the simplified method for the two-dimensional model for condensation resistance that has been implemented in the VISION program. For a given glazing system, the one-dimensional solution is found first. The glazing system configuration and properties, along with the glazing temperatures are used in the two-dimensional solution of the velocity field. The glazing cavity is assumed to be a vertical, tall cavity with isothermal walls and adiabatic ends. The velocity field is determined from the balance of buoyant and viscous forces that exist through the center-of-glass region. The difference between the actual velocity field based on true wall temperatures from this simplified case is negligible. This process is performed for each glazing cavity.

11. REPORTING REQUIREMENTS

The National Fenestration Rating Council details reporting requirements in NFRC LAP 1-94: Laboratory Accreditation Program for simulation and test laboratories. The reporting requirements originally addressed U-Factor simulations and tests. The 1994 draft of the LAP includes total solar heat gain coefficients and center-of-glass visible transmittance as part of simulation reporting requirements. The testing requirements only address U-Factor. Air leakage reporting requirements will more than likely be addressed in the future.

The Canadian Standards Association specifies reporting requirements in Section 9 of CSA A440.2-93. These address testing and measurement reporting requirements for U-Factor, solar heat gain coefficient, air leakage rate, and Energy Rating.

In terms of U-Factor simulation reporting requirements, the two organizations require the same basic information: manufacturer, product specifics, assembly drawings and part drawings, computer simulation results for components and the total system. A few of the details differ, such as NFRC requires the calculated areas for the different components. The requirements for solar heat gain coefficients are essentially the same.

The National Fenestration Rating Council requires more information for their U-Factor testing reporting requirements than the Canadian Standards Association. NFRC requires that the testing laboratory report warm and cold side temperatures at a number of positions, while CSA only requires air temperatures at mid-height on either side of the specimen. Very few products are tested under CSA A440.2 for U-Factor because simulation-only is an option and no testing validation of the simulations is required. Whereas NFRC requires that the simulation results be validated with thermal tests, and therefore their testing reporting requirements are more stringent.

12. ACCREDITATION AND CERTIFICATION

Accreditation

The National Fenestration Rating Council describes the accreditation process for all simulation and testing laboratories in the LAP 1-94: Laboratory Accreditation Program. The Standards Council of Canada (SCC) accredits laboratories. They have an accreditation program for testing laboratories, and are developing a procedure for simulation laboratories. The program for testing laboratories is similar to NFRC's.

The National Fenestration Rating Council describes the accreditation process for all simulation and testing laboratories in the LAP 1-94: Laboratory Accreditation Program. To be accredited, a simulation laboratory must meet the following criteria:

- Laboratory Independence
- Technical Competence
- Simulation Laboratory Equipment and Operations Manual
- Testing Laboratory Equipment and Operations Manual
- Laboratory Quality Control Program and Manual
- Record Keeping Requirements
- Reporting Requirements
- Accreditation Workshop Requirements
- Fee Payment
- Participate in Annual Review and Proficiency Testing

Each of these criteria is described in the NFRC LAP 1-94 document.

One of the primary differences in the accreditation requirements is that the Standards Council of Canada specifies that a Canadian Professional Engineer must sign the work. This requirements is to ensure high quality simulation results and to provide a simple decreditation mechanism through the Professional Engineering Boards. NFRC has no such requirement.

Certification

The certification processes are very similar between the National Fenestration Rating Council and the Canadian Window and Door Manufacturers Association (CWDMA). The National Fenestration Rating Council established Independent Agencies to certify products under NFRC. The IA checks simulation and test results, identifies products which can be labelled by the manufacturer and thus certified under NFRC, provides NFRC with a list of manufacturers' certifiable products, and inspects window manufacturers' facilities. Note that for a product to be certified under the NFRC program, it must have an NFRC label.

In Canada, the certification of fenestration products is performed by the Canadian Window and Door Manufacturers Association . The CWDMA certifies products to CAN/CSA-A440 and A440.2. A manufacturer can apply to license each product line under the program. An audit agency inspects the manufacturer's facility twice annually. Certified products are labelled with temporary and permanent labels. This program will become widely used when the 1995 National Energy Code is adopted.

There are a number of programs which currently list certified fenestration products in Canada:

- All products listed under Ontario Hydro's Window Incentive Program are certified.
- The Power Smart Program has their own list of certified products which meet the Power Smart Program's ER requirements for different climates.
- The Canadian Window and Door Manufacturers Association has a certification program for all products which have been tested/simulated under CSA A440 (includes all window performance tests - air, wind, water, thermal, etc.).
- The Canadian Construction Materials Council has also logged all windows which have been tested/simulated under CSA A440. There is clearly overlap between some of the organizations.

13. SUMMARY OF FINDINGS

Rating Procedures and Standards

The National Fenestration Rating Council (NFRC) in the U.S. and the Canadian Standards Association (CSA) in Canada have developed methods for predicting the energy performance of fenestration products. NFRC is mandated by the federal government to develop fair and accurate rating systems for fenestration products, while CSA's function is to develop standards for predicting the performance of products.

Table 1 lists the applicable documents for each organization for predicting the U-factor, solar heat gain, air leakage, annual energy performance, and condensation resistance. NFRC and CSA have developed their programs independent of one another, although there is some overlap in the test procedures and calculational tools currently being used. The differences and similarities are summarized below.

Products

The National Fenestration Rating Council and the Canadian Standards Association have developed methods for addressing the energy performance of the following fenestration products:

Vertical Slider	Double and Multiple Doors w/ Frames
Horizontal Slider	Combination and Composite Windows
Casement	Dual Action Windows
Projecting (Awning)	Bay or Bow Windows
Fixed	Sidelites
Single Door w/ Frame	Transoms

Those products covered by National Fenestration Rating Council procedures but not by the Canadian Standards Association include the following:

Skylights and Sloped Glazing	Pivoted Windows
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Even though the National Fenestration Rating Council lists skylights as being covered there are still some issues that need to be resolved, such as whether to rate skylights in a vertical position or at a 20° slope. For now, skylights are rated in a vertical position.

Neither organization explicitly addresses the following products:

Store Fronts
Curtain Walls
Greenhouse and Garden Windows

Glass Block
Curved Glazing
Garage Doors
Storm Doors

Greenhouse and garden windows are listed in NFRC 100-91 as being covered, but a working procedure has yet to be established by which to rate these products. This is also true for the transparent portion of curtainwalls.

Attachments and glazing options which are addressed by both organizations include integral shading devices, internal grilles, true divided lites, and glass thickness. The performance of products with integral shading devices is measured with the devices retracted. NFRC 100 specifies a minimum of 12 inches between internal grilles or true divided lites. NFRC 100 also states that if there is more than 1/8" gap between the glass and the grilles than the grilles do not need to be modelled. This has yet to be clarified in NFRC 200. CSA A440.2 is not clear on how to address grilles. It has been proposed to CSA that they adopt the method used in A453 for swinging doors, which is slightly different from NFRC 100-91. CSA A453 requires 2 vertical grille bars for all lite sizes. This is one more than what is required by NFRC 100-91 for a 22 inch width. CSA A453 requires no horizontal grilles for the 1/4-lite case, 2 for the 1/2-lite, and 4 for the full-lite. These are comparable to the NFRC 100-91 requirements.

The National Fenestration Rating Council does not cover removable shading devices, and the Canadian Standards Association requires that the performance of products with such a device be measured with the device retracted.

In NFRC 100-91, different glass thicknesses constitute individual glazing options. In CSA A440.2, glass thickness is not directly addressed; although, accepted practice is to model the glass thickness offered in the size being modelled.

Sizes

While both organizations cover nearly the same products, they do not necessarily treat the products the same. For example, the National Fenestration Rating Council specifies two sizes at which products are rated and the Canadian Standards Association specifies one size. For almost all fenestration products which both organizations include, the smaller of the two NFRC sizes corresponds directly with the CSA size. The exceptions to this are sidelites and transoms. NFRC specifies a size of 38'x14' for transoms and CSA rates transoms at the fixed window size of 48'x48'. Currently, CSA also rates window sidelites at the fixed window size and door sidelites at 38'x14' in CSA A453.

There are also differences in the lite sizes for doors between the NFRC 100-95: Attachment B and CSA A453. NFRC calls out four lite sizes: 1/4-lite, 1/2-lite, 3/4-lite, and full lite. CSA only calls out three sizes: 1/4-lite, 1/2-lite, and full lite; and the 1/4-lite and 1/2-lite sizes are different from the NFRC sizes. The lite sizes for sidelites are the same.

Thermal U-factor Calculations

In calculating fenestration U-factors, the primary difference between the National Fenestration Rating Council's procedure NFRC 100-91 and the Canadian Standards Association standard CSA A440.2 is the program used to calculate the center-of-glass properties. NFRC 100-91 utilizes the WINDOW 4.1 program and CSA A440.2 utilizes the VISION 3 program currently. The two programs are nearly identical except for the algorithms to calculate convective heat transfer between vertical glazing layers and the calculation of gas properties. There is a difference in the approaches to calculate the optical properties, but this difference does not influence the U-factor calculations. Figure 1 shows that VISION 3 calculates slightly higher center-of-glass U-factors than WINDOW 4.1; the difference is less than 3%.

A subcommittee of the American Society of Heating, Refrigeration, and Air Conditioning Engineers is establishing a standard calculation methodology for determining center-of-glass U-factors. VISION currently conforms to the draft standard and WINDOW will be updated in the near future to conform with the standard.

Thermal (U-Factor) Testing Methods

The National Fenestration Rating Council requires validation of simulated U-factors with thermal tests. The Canadian Standards Association CSA A440.2-93 describes both measurement and calculation of U-Factors and either approach can be taken, but there is no validation requirement between tests and simulation. Manufacturers have chosen to use the calculation procedure rather than test units in almost all cases to determine U-factors under CSA A440.2.

CSA references ASTM C1199-91, Test Method for Measuring the Steady-state Thermal Transmittance of Fenestration Systems Using Hot-Box Methods for thermal testing. ASTM C1199-91 is draft 10 of the NFRC 100-91: Attachment A document. The adopted version of NFRC 100-91: Attachment A is draft 11 of the document.

Essentially the two organizations are using the same test method. ASTM will update their test standards in 1996, and it is expected that the two organizations will be using the exact same document. There are slight differences in the outside and inside film coefficients which accounts for about a 1% difference in the results in highly conductive products (single-pane, aluminum) and less of an effect on low-conductivity products.

Solar Heat Gain Calculations

The most significant differences between the National Fenestration Rating Council's rating procedures and the Canadian Standards Association's fenestration performance standards occur in the calculation of solar heat gain. The following lists the differences:

Evaluation Conditions

- NFRC 200-93 determines the solar heat gain coefficient under summer design conditions.
- CSA A440.2-93 determines the solar heat gain coefficient under winter design conditions.

Center-of-Glass Solar Heat Gain Coefficient

- NFRC 200-93 references WINDOW 4.1 to calculate the center-of-glass solar heat gain coefficient.
- CSA A440.2-93 references VISION to calculate the center-of-glass solar heat gain coefficient.

Total Optical Properties

- WINDOW 4.1 uses a multi-band model to calculate the total optical properties.
- VISION 3 uses a two-band model to calculate the total optical properties. VISION 4 has a multi-band model but this version has not been released to the public.

Frame Solar Heat Gain

- WINDOW 4.1 calculates the solar heat gain through frames as equal to the inward flowing fraction of the absorbed solar radiation.
- CSA A440.2-93 specifies the solar heat gain through frames as a function of frame U-factor alone. The equation is based on the results of multiple simulations. On the average, the solar heat gain through frames determined by the CSA standard is 1/6 of that predicted by WINDOW 4.1.

These differences result in as much as a 3% variation in the solar heat gain coefficient between the NFRC procedure and the CSA standard. The NFRC value is always greater than the CSA value.

SHGC Test

The National Fenestration Rating Council does not have a procedure for measuring solar heat gain coefficient currently. The Canadian Standards Association references the method based on work at the National Solar Test Facility in Mississauga, Ontario: Test Method for Determination of Fenestration Solar Heat Gain Coefficient Using Simulated Solar Irradiance.

The countries have been collaborating on developing test methods to make these measurements through research funded by the American Society of Heating, Refrigeration, and Air Conditioning Engineers. Hopefully this collaboration continues and a method is developed in parallel.

Air Leakage

The National Fenestration Rating Council's procedure for measuring air leakage, NFRC 400-94 and the Canadian Standards Association standard CSA A440.2 reference the ASTM standard E 283 as the test method. Differences will occur though because the size of the unit being tested under NFRC 400 and CSA A440.2 may not be the same. The CSA requirements are more flexible and units tested under NFRC 400 would meet the CSA requirements.

Annual Energy Performance

The National Fenestration Rating Council is developing a procedure for rating the annual energy performance of windows. The draft procedure currently determines a heating rating, FHR, and a cooling rating, FCR. The Canadian Standards Association uses the ER value in CSA A440.2, which is a heating load calculation only. A cooling rating has been proposed for adoption.

In terms of the relative performance of products, the ranking of a wide range of windows based on the heating ratings, FHR and ER, and cooling ratings, FCR and ER_c are shown in Table 14. The heating ratings rank the windows identically, while the cooling ratings are significantly different in how they rank the windows. The primary reason for the difference in the cooling ratings is the inclusion of natural ventilation in determination of the FCR equation.

The absolute values of the ratings are not close because the FHR and FCR values represent a percentage of heat and cooling energy savings respectively as compared to a home with single-pane, aluminum windows. ER and ER_c represent energy gain (loss if negative) through a window in W/m^2 .

The differences in the ratings are not so important because the calculation of FHR, FCR, ER and ER_c is very simple and could be performed by any simulation laboratory given the U-factor, SHGC, and air leakage for a product.

Condensation Resistance

The National Fenestration Rating Council and the Canadian Standards Association are evaluating test methods and developing calculational methods for predicting condensation on

fenestration products. There is a good chance that they will both reference the same test methods, but it appears that two different calculational methods will be developed. Enermodal Engineering, Ltd. (1995) has implemented a condensation resistance model into the FRAME program. This is accomplished through a link to the VISION program. Lawrence Berkeley Laboratory is currently developing a model and plans to complete the work later this year. The Lawrence Berkeley Laboratory model is not yet functional so a comparison could not be made.

Reporting

The National Fenestration Rating Council and the Canadian Standards Association require the same basic information for reporting results. A few of the details differ, such as NFRC requires the calculated areas for the different components. The requirements for solar heat gain coefficients are essentially the same.

Very few products are tested under CSA A440.2 for U-Factor because simulation-only is an option and no testing validation of the simulations is required. NFRC requires that the simulation results be validated with thermal tests, and therefore their testing reporting requirements are more stringent. NFRC requires that the testing laboratory report warm and cold side temperatures at a number of positions, while CSA only requires air temperatures at mid-height on either side of the specimen.

Accreditation and Certification

The National Fenestration Rating Council describes the accreditation process for all simulation and testing laboratories in the LAP 1-94: Laboratory Accreditation Program. The Standards Council of Canada (SCC) accredits laboratories. They have an accreditation program for testing laboratories, and are developing a procedure for simulation laboratories. The program for testing laboratories is similar to NFRC's.

One of the primary differences in the accreditation requirements is that the Standards Council of Canada specifies that a Canadian Professional Engineer must sign the work. This requirements is to ensure high quality simulation results and to provide a simple decreditation mechanism through the Professional Engineering Boards. NFRC has no such requirement.

The certification processes are very similar between the National Fenestration Rating Council and the Canadian Window and Door Manufacturers Association (CWDMA). The National Fenestration Rating Council established Independent Agencies to certify products under NFRC. The IA checks simulation and test results, identifies products which can be labelled by the manufacturer and thus certified under NFRC, provides NFRC with a list of manufacturers' certifiable products, and inspects window manufacturers' facilities. Note that for a product to

be certified under the NFRC program, it must have an NFRC label.

In Canada, the certification of fenestration products is performed by the Canadian Window and Door Manufacturers Association . The CWDMA certifies products to CAN/CSA-A440 and A440.2. A manufacturer can apply to license each product line under the program. An audit agency inspects the manufacturer's facility twice annually. Certified products are labelled with temporary and permanent labels. This program will become widely used when the 1995 National Energy Code is adopted.

14. HARMONIZATION

The goals of the National Fenestration Rating Council and the Canadian Standards Association in developing procedures and standards for rating the thermal performance of windows are very much the same. The following summarizes the major differences between the NFRC and Canadian programs and makes recommendations for working towards reciprocity between the programs:

FENESTRATION PRODUCTS AND SIZES

Both programs cover nearly the same products, which include all of the major product types on the market. NFRC specifies two sizes, and CSA specifies one size that matches the smaller of the NFRC sizes. A manufacturer who has certified their products to CSA A440 and wants to obtain NFRC certification could either report values at only one size or have the work performed at both sizes. There are other details that need to be addressed, such as grille patterns and door lite sizes.

U-FACTOR

The calculational tools for determining U-factor are currently nearly the same. The difference lies in validation of the simulation results. NFRC requires thermal testing for validation and CSA requires the signature of a P.E. Neither program appears to be changing their requirements, so reciprocity would have to be agreed upon to accept the results generated under either program.

Introduction of another modeling tool, similar to the FRAME program, will complicate reciprocity in this area. A thorough validation exercise needs to be performed with the new tool in order to promote reciprocity.

SOLAR HEAT GAIN COEFFICIENT

The calculational tools for determining SHGC are also essentially the same. The major difference is the climatic conditions under which the calculations are performed. NFRC uses summer conditions and CSA uses winter conditions. This results in NFRC solar heat gain coefficients being slightly higher than the CSA numbers, which in turn gives a better ER - annual energy rating. (ER is the only number reported on the CWDMA label.) If reciprocity is not achievable than an adjustment factor could be applied to NFRC SHGC's to arrive at an equivalent ER.

AIR INFILTRATION

The test methods are the same, although the test size requirements and the reporting of the values are different. A test laboratory could report both values and NFRC could adopt the size correction used by CSA to achieve reciprocity.

ANNUAL ENERGY PERFORMANCE

These are simple calculations that could be done for either program using the U-factor, SHGC, and air infiltration results.

CONDENSATION RESISTANCE

NFRC has yet to establish a procedure and CSA specifies a test method. Efforts should be made to coordinate the NFRC development process with CSA.

REPORTING

The reporting requirements differ very little.

ACCREDITATION AND CERTIFICATION

The accreditation and certification programs of the National Fenestration Rating Council and the Canadian Window and Door Manufacturers Association and the Standards Council of Canada are very similar. Both programs require inspections of test laboratories and manufacturers' facilities and product labels. The biggest differences lie in the fact that the reciprocity would have to be coordinated between NFRC and its Independent Agencies, and CWDMA and the Standards Council of Canada. There are also cost issues which need to be dealt with in order to have the various entities cooperate.

15. CONCLUSIONS

The question is will the National Fenestration Rating Council and the Canadian Window and Door Manufacturers Association certify products based upon their certification under the others program. Manufacturers have a strong interest in achieving reciprocity to minimize their product costs. This is also in the best interest of the consumer because manufacturer's costs will be passed onto the consumer. Also, there is no more energy to be saved by having to certify products under two systems that are nearly identical. Two systems only waste energy, time and money.

Nevertheless, to achieve reciprocity is not a simple task. This requires coordination and cooperation between the countries and the program managers of NFRC, CSA, and CWDMA. The differences are not necessarily insurmountable, but the implications of the solutions require thorough investigation.

Technical issues such as size differences, product definitions, simulation tools, verification of results, and differences in climatic conditions need to be addressed. However, program issues are more problematic. What are the benefits to NFRC, CSA, and CWDMA in achieving reciprocity? If reciprocity is achieved, the programs in the two countries will be competing for business. A window manufacturer can rate their products to CSA A440 for less money than it can to the NFRC program. Is it possible to establish a cost structure that will minimize this difference? This is the type of issue that needs to be resolved before the two programs even begin to solve the technical issues.

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APPENDIX

COMPARISON OF U.S. AND CANADIAN THERMAL PERFORMANCE EVALUATION PROGRAMS FOR FENESTRATION PRODUCTS

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INTRODUCTION

The window and door industry in North America have a strong interest in harmonizing U.S. and Canadian rating procedures and standards for predicting the thermal performance of their products. The costs and time associated with complying with the requirements in both countries are significant and could be reduced through reciprocity. The purpose of this paper is to compare the U.S. and Canadian rating programs. Joint efforts between the U.S. and Canada can use this information to focus on reconciling the differences and to work towards reciprocity.

The National Fenestration Rating Council (NFRC) in the U.S. and the Canadian Standards Association (CSA) in Canada have developed methods for predicting the energy performance of fenestration products. NFRC is mandated by the federal government to develop fair and accurate rating systems for fenestration products, along with a certification program. CSA's function is to develop standards for predicting the performance of products, and the Canadian Window and Door Manufacturers Association certifies products under the various standards.

The following identifies the applicable procedures and standards for both countries, and summarizes the major differences in the products that are covered, the sizes that are used, the U-factor and solar heat gain coefficient calculational and test methods, air leakage measurement, prediction of annual energy use, condensation resistance methods, reporting requirements, and accreditation and certification programs.

RATING PROCEDURES AND STANDARDS

Table 1 lists the applicable rating procedures and standards for NFRC and CSA for predicting the thermal performance of fenestration products. Within each of the procedures and standards, computer programs, calculational methodologies, and test standards are referenced. NFRC and CSA have developed their programs independent of one another, although there is some overlap in the test procedures and calculational tools currently being used.

TABLE 1 - APPLICABLE PROCEDURES AND STANDARDS

Procedure/Standard	NFRC	CSA
Thermal (U-Factor) Calculation	NFRC 100-91, NFRC 100-94 (draft), NFRC 100-94: Attachment B (doors)	CSA A440.2-93 CSA A453 (swinging door standard)
Thermal (U-Factor) Test	NFRC 100-91: Attachment A, NFRC 100-94: Attachment A	CSA A440.2-93 CSA A453
Solar Heat Gain Calculation	NFRC 200-93	CSA A440.2-93 CSA A453
Solar Heat Gain Test	None currently	CSA A440.2-93 CSA A453
Measurement of Optical Properties	NFRC 300-93	ASHRAE Standard 74 (CSA A440.2-93)
Air Leakage	NFRC 400-94	CSA A440.2-93 CSA A453
Annual Energy Performance	NFRC 900-95 (draft)	CSA A440.2-93 CSA A453
Condensation Resistance	Under Development	CSA A440.1-90

FENESTRATION PRODUCTS

The National Fenestration Rating Council and the Canadian Standards Association have methods for addressing the energy performance of the following fenestration products:

Vertical Slider	Double and Multiple Doors w/ Frames
Horizontal Slider	Combination and Composite Windows
Casement	Dual Action Windows
Projecting (Awning)	Bay or Bow Windows
Fixed	Sidelites
Single Door w/ Frame	Transoms

Those products covered by National Fenestration Rating Council procedures but not by the Canadian Standards Association include the following:

Skylights and Sloped Glazing Pivoted Windows

The National Fenestration Rating Council recently established that the rough opening shall be used to rate skylights. There are still some issues that need to be resolved, such as whether to rate skylights in a vertical position or at a 20° slope. For now, skylights are rated in a vertical position.

Neither organization explicitly addresses the following products:

Store Fronts
Greenhouse and Garden Windows
Glass Block
Garage Doors

Transparent Portion of Curtainwalls
Opaque Portion of Curtainwalls
Curved Glazing
Storm Doors

The National Fenestration Rating Council lists greenhouse and garden windows, curtainwalls, and garage doors, but the task groups have yet to establish working procedures to actually rate these products.

Attachments and glazing variations which are addressed by both organizations include integral shading devices, internal grilles, true divided lites, and glass thickness. Integral shading devices are rated in their retracted position by both organizations. NFRC 100 specifies a minimum of 12 inches between internal grilles or true divided lites. NFRC 100 also states that if there is more than 1/8" gap between the glass and the grilles then the grilles do not need to be modelled. This has yet to be clarified in NFRC 200. CSA A440.2 is not clear on how to address grilles. It has been proposed to CSA that they adopt standard grille patterns to cover all variations, which is similar to the method used in A453 for swinging doors. This method is slightly different from NFRC 100-91. CSA A453 requires 2 vertical grille bars for all lite sizes. This is one more than what is required by NFRC 100-91 for a 22 inch width. CSA A453 requires no horizontal grilles for the 1/4-lite case, 2 for the 1/2-lite, and 4 for the full-lite. These are comparable to the NFRC 100-91 requirements.

The National Fenestration Rating Council does not cover removable shading devices, and the Canadian Standards Association requires that the performance of products with such a device be measured with the device retracted.

In NFRC 100-91, different glass thicknesses constitute individual glazing options. To reduce the number of possible variations, NFRC allows glazing options to be grouped. A group of glazing options is represented by the worst performing glazing system in the group, and can vary in glass thickness, glass type, gap thickness, and gas fill. In CSA A440.2, glass thickness is not directly addressed, although accepted practice is to model the glass thickness offered in the size being modelled. This is the approach taken in CSA A453. This eliminates

many of the glazing system variations, and reduces confusion over unrealistic products being modelled.

SIZES

While both organizations cover nearly the same products, they do not necessarily treat the products the same. For example, the National Fenestration Rating Council specifies two sizes at which products are rated and the Canadian Standards Association specifies one size. For almost all fenestration products which both organizations include, the smaller of the two NFRC sizes corresponds directly with the CSA size. The exceptions to this are window sidelites and transoms.

NFRC specifies a size of 965 mm x 355 mm (38" x 14") for transoms and CSA rates transoms at the fixed window size of 1220 mm x 1220 mm (48" x 48"). CSA rates window sidelites at the fixed window size, and door sidelites at the same size as used by NFRC for sidelites. Under CSA A453, door sidelites receive the same rating as the equivalent door and are only modelled when there is not a matching door.

There are also differences in the lite sizes for doors specified in NFRC 100: Attachment B and CSA A453. NFRC calls out four lite sizes: 1/4-lite, 1/2-lite, 3/4-lite, and full lite. CSA only calls out three sizes: 1/4-lite, 1/2-lite, and full lite; and the 1/4-lite and 1/2-lite sizes are different from the NFRC sizes. The lite sizes for sidelites are the same.

THERMAL U-FACTOR CALCULATIONS

In calculating fenestration U-factors, the primary difference between the National Fenestration Rating Council's procedure NFRC 100 and the Canadian Standards Association standard CSA A440.2 is the program used to calculate the center-of-glass properties. NFRC 100-91 utilizes the WINDOW 4.1 program and CSA A440.2 utilizes the VISION 3 program currently. The two programs are nearly identical except for the algorithms to calculate convective heat transfer between vertical glazing layers and the calculation of gas properties. There is a difference in the approaches used to calculate the optical properties, but this difference does not influence the U-factor calculations. VISION 3 calculates slightly higher center-of-glass U-factors than WINDOW 4.1; the difference is less than 3%. For gap widths greater than 35 mm, WINDOW 4.1 is incorrect. VISION predicts a more correct value.

A subcommittee of the American Society of Heating, Refrigeration, and Air Conditioning Engineers is establishing a standard calculation methodology for determining center-of-glass U-factors. VISION currently conforms to the draft standard and WINDOW may be updated in the near future to conform with the standard.

THERMAL (U-FACTOR) TESTING METHODS

The National Fenestration Rating Council requires validation of simulated U-factors with thermal tests. The Canadian Standards Association CSA A440.2-93 describes both the measurement and calculation of U-Factors and either approach can be taken, but there is no validation requirement between tests and simulation. Manufacturers have chosen to use the calculation procedure rather than test units in almost all cases to determine U-factors under CSA A440.2.

CSA references ASTM C1199-91, Test Method for Measuring the Steady-state Thermal Transmittance of Fenestration Systems Using Hot-Box Methods for thermal testing. ASTM C1199-91 is draft 10 of the NFRC 100-91: Attachment A document. The adopted version of NFRC 100-91: Attachment A is draft 11 of the document.

Essentially the two organizations are using the same test method. ASTM will update their test standards in 1996, and it is expected that the two organizations will be using the exact same document. There are slight differences in the outside and inside film coefficients which accounts for about a 1% difference in the results in highly conductive products (single-pane, aluminum) and has less of an effect on low-conductivity products.

SOLAR HEAT GAIN COEFFICIENT CALCULATIONS

There are significant differences between the National Fenestration Rating Council's rating procedures and the Canadian Standards Association's fenestration performance standards for calculating solar heat gain coefficients. The following lists the differences:

Evaluation Conditions

- NFRC 200-93 determines the solar heat gain coefficient under summer design conditions.
- CSA A440.2-93 determines the solar heat gain coefficient under winter design conditions.

Center-of-Glass Solar Heat Gain Coefficient

- NFRC 200-93 references WINDOW 4.1 to calculate the center-of-glass solar heat gain coefficient.
- CSA A440.2-93 references VISION to calculate the center-of-glass solar heat gain coefficient.

Total Optical Properties

- WINDOW 4.1 uses a multi-band model to calculate the total optical properties.
- VISION 3 uses a two-band model to calculate the total optical properties. The latest version of VISION (VISION 4) has a multi-band model but has not been adopted into the standards.

Frame Solar Heat Gain

- WINDOW 4.1 calculates the solar heat gain through frames as equal to the inward flowing fraction of the absorbed solar radiation. The projected frame area is used along with an exterior film coefficient for a 7.5 mph wind (no radiation).
- CSA A440.2-93 specifies the solar heat gain through frames as a function of frame U-factor alone. The equation is based on the results of detailed simulations. On average, the solar heat gain through frames determined by the CSA standard is 1/6 of that predicted by WINDOW 4.1.

These differences result in as much as a 3% variation in the solar heat gain coefficient between the NFRC procedure and the CSA standard. The NFRC value is always greater than the CSA value. (Wright 1994)

SOLAR HEAT GAIN COEFFICIENT TEST METHODS

The National Fenestration Rating Council does not have a procedure for measuring solar heat gain coefficient currently. The Canadian Standards Association references the method based on work at the National Solar Test Facility in Mississauga, Ontario: Test Method for Determination of Fenestration Solar Heat Gain Coefficient Using Simulated Solar Irradiance. The countries have been collaborating on developing test methods to make these measurements through research funded by the American Society of Heating, Refrigeration, and Air Conditioning Engineers. This work may lead to a method acceptable to both organizations.

AIR LEAKAGE

The National Fenestration Rating Council's procedure for measuring air leakage, NFRC 400-94 and the Canadian Standards Association standard CSA A440.2 reference the ASTM standard E 283 as the test method. Differences will occur though because the size of the unit being tested under NFRC 400 and CSA A440.2 may not be the same. The CSA requirements are more flexible and units tested under NFRC 400 would meet the CSA requirements. The reverse is not true.

ANNUAL ENERGY PERFORMANCE

The National Fenestration Rating Council is developing a procedure for rating the annual energy performance of windows. The draft procedure currently determines a heating rating, FHR, and a cooling rating, FCR. The Canadian Standards Association uses the ER value in CSA A440.2, which is a heating load calculation only (EEL 1994a). A cooling rating, ER_c, has been developed but is not part of the standards (EEL 1993).

The most current forms of the regression equations for calculating FHR and FCR are as follows (Crooks 1995):

$$\text{FHR} = 50.05 - 60.39*U + 10.02*\text{SHGC} - 6.00*AL + 11.68*U^2 + 10.00*\text{SHGC}^2 - .03*AL^2 \quad (1)$$

$$\text{FCR} = 61.73 - 2.01*U - 71.30*\text{SHGC} - 0.81*AL - 1.34*U^2 - 9.64*\text{SHGC}^2 - .03*AL^2 \quad (2)$$

where AL is air leakage in CFM/ft. The regressions are based on the results for 6 climates and will be extended to 30 different climates.

The ER equation used in CSA A440.2-93 and the proposed ER_c equation are

$$\text{ER} = 72.20*\text{SHGC} - 21.90*U - 0.54*(L_{75}/A) \quad (3)$$

$$\text{ER}_c = 110.0*\text{SHGC} - 4.5*U - 0.07*L_{75}/A \quad (4)$$

where L₇₅ is the average window air leakage rate (m³/h) at a test pressure difference of 75 Pascals. The equations for ER and ER_c are based on average Canadian climate conditions. These are 72.20 for the off-angle incidence factor multiplied by the average solar radiation when heating is required, a 21.9°C outside-to-inside temperature difference, a 0.54 pressure factor. It is assumed that the hours during which air temperature less than 13°C that solar gains are utilizable. The values are the average for 13 cities with degree-days between 4000 and 6200 °C-days.

In terms of the relative performance of products, the ranking of a wide range of windows based on the heating ratings, FHR and ER, and cooling ratings, FCR and ER_c are shown in Table 2. The heating ratings rank the windows identically, while the cooling ratings are significantly different in how they rank the windows. The absolute values of the ratings are not close because the FHR and FCR values represent a percentage of heat and cooling energy savings respectively as compared to a home with single-pane, aluminum windows. ER and ER_c represent energy gain (loss if negative) through a window in W/m².

Windows 11 and 12 in Table 2 are hypothetical windows. The purpose of them is to examine the ratings under limiting conditions. The best performing window possible in terms of heating energy savings would have U=0.0 and SHGC=1.0 (window #11). FHR predicts 72% home energy savings for this window and ER estimates a heat gain of 72 W/m². FHR predicts 50% home energy savings from a window which has no heat loss, U=0.0, and no solar heat gain, SHGC=0.0 (window #12). ER predicts zero heat loss through the same window and no solar gains. (ER rates window #10 as superior to window #12 because of the solar heat gain. FHR predicts 44% home energy savings for window 10, which is less than what it predicts for window #12.) In terms of cooling energy, FCR predicts cooling energy savings of 62% for window #12 and ER predicts cooling energy use of 0 W/m².

To compare the heating and cooling ratings on the same scale, FHR, ER, FCR and ER_c are normalized to represent the percentage of potential window energy savings. For FHR and FCR, the ratio of the percent energy savings for a particular product to that through the "best

performing" product is calculated. ER and ER_c are converted by first calculating the energy savings as compared to a single-pane, aluminum window with clear glass (same as unit used for base case for FHR and FCR). Then the ratio of these energy savings to that through the "best performing" window are computed. Figures 1 through 8 compare the different rating systems for various scenarios.

**TABLE 2 HEATING AND COOLING
ANNUAL ENERGY PERFORMANCE RATINGS**

ID	U W/m ² -C (Btu/hr- ft ² -F)	SHGC	Air Leakage (cfm/ft)	FHR	ER	FCR	ER_c
1	7.6 (1.34)	0.78	0.65	0	-122	0	49.8
2	7.6 (1.34)	0.68	0.37	-1	-125	9	39.3
3	3.8 (0.67)	0.63	0.37	23	-44	13	51.2
4	3.8 (0.67)	0.53	0.37	21	-51	22	40.2
5	2.8 (0.49)	0.58	0.37	30	-25	17	50.4
6	2.8 (0.49)	0.48	0.37	28	-32	26	39.4
7	1.9 (0.33)	0.55	0.37	38	-7	20	51.3
8	1.7 (0.30)	0.44	0.37	37	-11	29	40
9	1.6 (0.29)	0.32	0.37	36	-18	38	27
10	0.9 (0.15)	0.37	0.37	44	3	34	36.2
11	0	1	0	70	72	-19	110
12	0	0	0	50	0	62	0

In comparing FHR and ER, we found the following differences:

1. FHR predicts a greater percent of heating energy savings than the ER does for the ten different windows in Table 2 (Figure 1).
2. FHR and ER show nearly the same dependency on U-factor (Figure 3).
3. FCR is not sensitive to changes in U-factor while ER_c is (Figures 2, 4 and 6). This is due to the fact that FCR accounts for natural ventilation and ER_c does not.
4. FHR and ER also show the same dependency on SHGC (Figure 5).

5. FCR and ER_c show the same dependency on SHGC (Figure 6), but again FCR does not vary with changes in U-factor.
6. The heat and cooling energy ratings show a weak dependency on air leakage, especially with the cooling ratings (Figures 7 and 8).

Fig. 1 FHR and ER compared on basis of potential energy savings from a window. Both values have been normalized. FHR has been normalized by energy savings of window 10. ER has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window 10.

Fig. 2 FCR and ER_c compared on basis of potential energy savings from a window. Both values have been normalized. FCR has been normalized by energy savings of window 9. ER_c has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window 9.

Fig. 3 FHR and ER compared on basis of potential energy savings from a window. Both values have been normalized. FHR has been normalized by energy savings of window 10. ER has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window with $U=0.0$, $SHGC=1.0$.

Fig. 4 FCR and ER_c compared on basis of potential energy savings from a window. Both values have been normalized. FCR has been normalized by energy savings of window 9. ER_c has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window with $U=0.0$, $SHGC=0.0$.

Fig. 5 FHR and ER compared on basis of potential energy savings from a window. Both values have been normalized. FHR has been normalized by energy savings of window 10. ER has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window with $U=0.0$, $SHGC=1.0$.

Fig. 6 FCR and ER_c compared on basis of potential energy savings from a window. Both values have been normalized. FCR has been normalized by energy savings of window 9. ER_c has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window with $U=0.0$, $SHGC=0.0$.

Fig. 7 FHR and ER compared on basis of potential energy savings from a window. Both values have been normalized. FHR has been normalized by energy savings of window 10. ER has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window with $U=0.0$, $SHGC=1.0$, and air leakage equal to 0.0.

Fig. 8 FCR and ER_c compared on basis of potential energy savings from a window. Both values have been normalized. FCR has been normalized by energy savings of window 9. ER_c has been normalized by calculating energy savings for all windows as compared to window 1, and dividing by the energy savings for window with $U=0.0$, $SHGC=0.0$, and air leakage equal to 0.0.

CONDENSATION RESISTANCE

The National Fenestration Rating Council is evaluating test methods and developing a calculational method for predicting condensation on fenestration products. The Canadian Standards Association has a test procedure for measuring condensation resistance (CSA-A440.1-M90, Section 11.12). A paper by Elmahdy entitled, "A Universal Approach to Laboratory Assessment of the Condensation Potential of Windows," describes the method.

As for calculational methodologies, two approaches are being developed currently. Enermodal Engineering, Ltd. (McGowan 1995, Wright 1995) has implemented a condensation resistance model into the FRAME program. This is accomplished through a link to the VISION program. Lawrence Berkeley Laboratory is currently developing a model and plans to complete the work later this year. The Lawrence Berkeley Laboratory model is not yet functional so a comparison could not be made.

REPORTING

The National Fenestration Rating Council and the Canadian Standards Association require the same basic information for reporting results. A few of the details differ, such as NFRC requires the calculated areas for the different components. The requirements for solar heat gain coefficients are essentially the same.

Very few products are tested under CSA A440.2 for U-Factor because simulation-only is an option and no testing validation of the simulations is required. NFRC requires that the simulation results be validated with thermal tests. NFRC requires that the testing laboratory report warm and cold side temperatures at a number of positions, while CSA only requires air temperatures at mid-height on either side of the specimen.

ACCREDITATION AND CERTIFICATION

The National Fenestration Rating Council describes the accreditation process for all simulation and testing laboratories in the LAP 1-94: Laboratory Accreditation Program. The Standards Council of Canada (SCC) accredits laboratories. They have an accreditation program for testing laboratories, and are developing a procedure for simulation laboratories. The program for testing laboratories is similar to NFRC's.

One of the primary differences in the accreditation requirements is that the Standards Council of Canada specifies that a Canadian Professional Engineer must sign the work. This requirements is to ensure high quality simulation results and to provide a simple decreditation mechanism through the Professional Engineering Boards. NFRC has no such requirement.

The certification processes are very similar between the National Fenestration Rating Council and the Canadian Window and Door Manufacturers Association (CWDMA). The National

Fenestration Rating Council established Independent Agencies to certify products under NFRC. The IA checks simulation and test results, identifies products which can be labelled by the manufacturer and thus certified under NFRC, provides NFRC with a list of manufacturers' certifiable products, and inspects window manufacturers' facilities. Note that for a product to be certified under the NFRC program, it must have an NFRC label.

In Canada, the certification of fenestration products is performed by the Canadian Window and Door Manufacturers Association . The CWDMA certifies products to CAN/CSA-A440 and A440.2. A manufacturer can apply to license each product line under the program. An audit agency inspects the manufacturer's facility twice annually. Certified products are labelled with temporary and permanent labels. This program will become widely used when the 1995 National Energy Code is adopted.

HARMONIZATION

The goals of the National Fenestration Rating Council and Canadian programs for rating the thermal performance of windows are very much the same. The following summarizes the significant difference between the NFRC procedures and CSA standards:

FENSTRATION PRODUCTS AND SIZES

Both programs cover nearly the same products, which include all of the major product types on the market. NFRC specifies two sizes, and CSA specifies one size that matches the smaller of the NFRC sizes. A manufacturer who has certified their products to CSA A440 and wants to obtain NFRC certification could either report values at only one size or have the work performed at both sizes. There are other details that will need to be addressed, such as grille patterns and door lite sizes.

U-FACTOR

The calculational tools for determining U-factor are essentially the same. The difference lies in validation of the simulation results. NFRC requires thermal testing for validation and CSA requires the signature of a P.E. Neither program appears to be changing their requirements, so reciprocity would have to be agreed upon to accept the results generated under either program.

SOLAR HEAT GAIN COEFFICIENT

The calculational tools for determining SHGC are also essentially the same. The major difference is the climatic conditions under which the calculations are performed. NFRC uses summer conditions and CSA uses winter conditions. This results in NFRC solar heat gain coefficients being slightly higher than the CSA numbers, which in turn gives a better ER - annual energy rating. (ER is the only number reported on the CWDMA label.) If reciprocity is not achievable than an adjustment factor could be applied to NFRC SHGC's to arrive at an equivalent ER.

AIR INFILTRATION

The test methods are the same, although the test size requirements and the reporting of the values are different. A test laboratory could report both values and NFRC could adopt the size correction used by CSA to achieve reciprocity.

ANNUAL ENERGY PERFORMANCE

These are simple calculations that could be done for either program using the U-factor, SHGC, and air infiltration results.

CONDENSATION RESISTANCE

NFRC has yet to establish a procedure and CSA specifies a test method. Efforts should be made to coordinate the NFRC development process with CSA.

REPORTING

The reporting requirements differ very little.

ACCREDITATION AND CERTIFICATION

The accreditation and certification programs of the National Fenestration Rating Council and the Canadian Window and Door Manufacturers Association and the Standards Council of Canada are very similar. Both programs require inspections of test laboratories and manufacturers' facilities and product labels.

CONCLUSIONS

The question is will the National Fenestration Rating Council and the Canadian Window and Door Manufacturers Association certify products based upon their certification under the others program. Manufacturers have a strong interest in achieving reciprocity to minimize their product costs. This is also in the best interest of the consumer because manufacturer's costs will be passed onto the consumer. Also, there is no more energy to be saved by having to certify products under two systems that are nearly identical. This only wastes more energy, time and money.

Nevertheless, to achieve reciprocity is not a simple task. This requires coordination and cooperation between the countries and the program managers of NFRC, CSA, and CWDMA. The differences are not necessarily insurmountable, but the implications of the solutions require thorough investigation.

Technical issues such as size differences, product definitions, verification of results, and difference in climatic conditions need to be addressed. Program issues are more problematic. What are the benefits to NFRC, CSA, and CWDMA in achieving reciprocity? If reciprocity is achieved, the programs in the two countries will be competing for business. A window manufacturer can rate their products to CSA A440 for less money than it costs to rate their products to NFRC procedures. Is it possible to establish a cost structure that will minimize

this difference? This is the type of issue that needs to be resolved before the two programs even begin to solve the technical issues.

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§1.0 Introduction

The thermal performance of a window is currently characterized by the window's thermal transmittance (U-factor) and its solar heat gain coefficient. The National Fenestration Rating Council (NFRC) has established a system for rating the thermal performance of windows. The U-factor is determined through computer simulation and validated with physical tests. The solar heat gain coefficient is determined for homogeneous products through computer simulation. Test methods exist for measuring solar heat gain through more complex products, although there is currently no standard.

Under the NFRC's rating program, a window must be simulated using the Window 4.1 and Frame 3.1 computer programs. There is some debate as to how accurately these computer programs simulate actual windows. This report addresses the differences in simulation styles and assumptions and what impact these differences have on the U-factor and solar heat gain coefficient. Section 2.0 covers center-of-glass modeling, section 3.0 covers spacer modeling, section 4.0 covers frame modeling, and section 5.0 concludes by weighing the relative importance of the assumptions discussed. The focus of this research is on U-factor. For a more detailed study of solar heat gain coefficients refer to Wright (1995).

This report also addresses the efficacy of various techniques, such as increasing glazing gap width or applying low-emittance coatings to interior glazing surfaces, at reducing total window U-factors.

§2.0 Effects of Center-of-Glass Modeling Assumptions in Window 4.1

The center-of-glass area is defined as the glazed area minus a strip 2.5 inches wide adjacent to the window frame. The center-of-glass area is modelled in Window 4.1. Details that are specified in the model include gap width between layers of glazing, glass thickness, gas fill between glazing layers, and glazing type.

§2.1 Glazing Gap Width

Large glazing gaps are generally associated with reduced U-factors because as the gap width increases, the heat flow path increases, so the resistance to heat flow is increased. At a certain width, however, natural convection becomes stronger and wider gaps actually have decreased resistance to heat flow. This relationship is illustrated in Figure 2.1. For an air-filled gap, the U-factor decreases steeply (by 0.16 Btu/h·ft²·°F between 0.18 and 0.5 inches) as the gap width increases until the gap width is a little more than 0.5 inches, when it begins to increase again. This same relationship occurs with other gas fills at a gap thickness specific to each gas. Wright (1995) has shown that the gap width has a negligible effect on solar heat gain coefficient (SHGC).

Figure 2.1 Effect of gap thickness on U-factor for a 0.125 inch glass, air-filled glazing unit.

For gap widths greater than 1.375 inches, the U-factor results in Window 4.1 are slightly higher than what is expected. It is recommended that for gap widths greater than 1.375 inches that the user model the gap at 1.375 inches in Window 4.1.

§2.2 Glass Thickness

One-dimensional heat transfer through glass may be described by the Fourier conduction equation:

$$Q = [k \cdot A \cdot (T_h - T_c)] / L, \quad (\text{Equation 1})$$

where Q is the rate of heat flow in Btu/h, k is the thermal conductivity in Btu·in/h·ft²·°F, A is the area of the material perpendicular to the direction of heat flow in ft², L is the length of the heat flow path through the material in feet, T_h is the temperature on the hot side of the material in °F, and T_c is the temperature on the cold side of the material in °F. When modeling a glazing unit in Window 4.1, all of the variables in Equation 1 are constant except for the length of the heat flow path (glass thickness). Clearly, the thicker the glass is, the slower the rate of heat transfer, Q . To illustrate this, a glazing unit with a 0.5 inch air gap is simulated with different glass thicknesses and the resulting center-of-glass U-factors are plotted in Figure 2.2. Note that the resulting U-factors only decrease from 0.494 to 0.476 Btu/h·ft²·°F (3.6%) when the glass thickness is increased from 0.088 to 0.32 inches.

Figure 2.2 Effect of glass thickness on U-factor for an air-filled, 0.5 inch gap, insulated glazing unit.

Since glazing optical properties are specific to glass thickness, the SHGC of a glazing system is strongly dependent on glass thickness.

§2.3 Low-Emittance Coatings

Another way of reducing the center-of-glass U-factor is to apply a low-emittance (low-E) coating to one of the glass layers. This reduces the amount of infrared radiation (heat) that is transmitted through the window. A glazing unit with a gap of 0.5 inches is simulated with 100% air and 97% argon for a range of emittances. Whether the coating is on the second or third surface has no effect on U-factor, although placement on the first or fourth surfaces does (see Table 2.1). The resulting U-factors are plotted in Figure 2.3. Note that this is a semi-log plot.

Figure 2.3 Effect of surface emittance on U-factor for a 0.5 inch gap glazing unit.

When modeling low-E coatings in Window 4.1, it is important to locate the coating on the correct surface. A glazing unit with a half inch air gap is modelled with a low-E coating ($e = 0.085$) on different surfaces. The results are presented in Table 2.1. Placing the low-E coating on the wrong surface can create as much as a 55% difference in center-of-glass U-factor, and as much as a 5% difference in solar heat gain coefficient. Low-E coatings should always be modeled on the second or third surfaces of an insulated glazing unit or the full effect of the coating on the U-factor will not be realized. Whether the low-E coating is on the second or the third surface has a negligible effect on U-Factor. However, a glazing unit with the low-E coating on the second surface has a lower solar heat gain coefficient than a glazing unit with the same coating on the third surface.

Table 2.1 Effect of placement of low-emittance coating on U-factor and SHGC.

Glazing Unit Description	Center-of-Glass U-Factor	Solar Heat Gain Coefficient
SG100(1)/AIR/CL	0.4858	0.59
SG100(2)/AIR/CL	0.3126	0.58
CL/AIR/SG100(3)	0.3127	0.64
CL/AIR/SG100(4)	0.3661	0.61

§2.4 Gas Fill

Glazing gaps may be filled with inert gas mixtures having conductivities less than that of air to further reduce the transfer of heat between glazing layers. The most common gas fill is argon. A few manufacturers are also using krypton.

To determine what relationship exists between the percentage of inert gas and the U-factor, a glazing unit with a 0.5 inch gap and a low-emittance coating on one surface is simulated with different percentages of argon. The resulting U-factors are plotted in Figure 2.4. A difference of 10% in argon fill results in about a 2% difference in the center-of-glass U-factor.

The single point labelled 100% argon in Figure 2.4 is the result of a discrepancy in the Window 4.1 program. The program allows the user to specify the mixture of gases, but gives a different answer if 0% air is specified in an air/argon mixture than what is assumed for pure argon. The difference between these two center-of-glass U-factors is about 1%.

Figure 2.3 compares an air-filled window with an argon-filled window while varying the surface emittance. For uncoated glass, the argon reduces the center-of-glass U-Factor by 0.02 Btu/h•ft²•°F. For coated glass with an emittance of 0.01, argon reduces the center-of-glass U-factor by 0.03 Btu/h•ft²•°F. The reduction is greater in low-E windows because conduction through the gas is the dominant mode of heat transfer. In uncoated windows, radiative heat transfer is dominant.

Figure 2.4 Effect of percent argon gas fill on U-factor for a 0.5 inch glazing gap and a low-emittance coating on one surface.

Figure 2.5 Effect of gap thickness on U-factor for a krypton-filled unit with a low-E coating ($e=.1$) on the second surface.

Figure 2.6 Effect of percent krypton gas fill on U-Factor for a unit with an 0.3 inch gap and a low-emittance coating ($e=.1$) on the second surface.

Figures 2.5 and 2.6 show the effect of gap thickness and percent gas fill on U-factor for a krypton-filled unit with a low-E coating on the second surface. The optimum gap width for a krypton-filled window is approximately 0.3 inches.

§2.5 Combined Effects

§2.5.1 Gap Width and Glass Thickness

When modeling a glazing unit of a fixed overall width, the gap width may vary depending on the glass thickness used. For example, a unit which is modelled at two different sizes may require thicker glass for structural reasons in the larger size. To determine what effect this has on the center-of-glass U-factor, glazing units in four overall widths were modelled with varying glass thicknesses and gap widths. The results are presented graphically in Figure 2.7. As long as the gap is greater than about 0.45 inches, the impact of gap width and glass thickness on U-factor is negligible (less than 2%). When the gap is less than 0.45 inches, the difference is as much as 11% for a change in gap thickness of 0.1 inches. The same results are shown in Figure 2.8 as a bar graph.

Figure 2.7 Effect of glass thickness on U-factor for four different overall glazing unit widths.

Figure 2.8 Effect of glass thickness on U-factor for four different overall glazing unit widths.

§2.5.2 Gap Thickness, Gas Fill and Coating Emittance

The effect that gap thickness has on center-of-glass U-factor is dependent on the composition of the gas mixture filling the gap. Figure 2.9 illustrates the differences in U-factor for three different gas mixtures, a low-E coating with an emittance of 0.10, and varying gap thicknesses. The effect gap thickness has on center-of-glass U-factor is also affected by the emittance of the surface coating.

Figure 2.9 Effect of gap thickness on U-factor for three different gas mixtures in a glazing unit with a low emittance coating ($e=0.1$).

Figure 2.10 Effect of gap thickness on U-factor for three different glazing units.

Figure 2.10 illustrates the effect on U-factor of gap thickness, gas fill, and coating emittance in combination. The lower the emittance the greater the effect is from varying the gap thickness. Again, this is because conduction through the gas fill dominates the heat transfer in low-E windows.

§2.6 Use of Spectral Data Files

When modeling glazing units in Window 4.1, files containing the spectral data for each glass type are available and should be used if reporting solar heat gain coefficient. The spectral data has no bearing on center-of-glass U-factors. To determine what effect omission of the spectral data has on the center-of-glass SHGC, several glazing units were simulated with and without the spectral data. Clear, bronze, green, and blue glass were combined with clear glass (Glazing Units 1-4), with SunGate100 Low-E glass (Glazing Units 5-8), with Heat Mirror and clear glass (Glazing Units 9-12), and with Heat Mirror and SunGate100 (Glazing Units 13-16). The results are compared in Table 2.2 and Figure 2.11. The effect of omitting the spectral data is to decrease the solar heat gain coefficient by as much as 23.8 percent.

Figure 2.11 Solar heat gain coefficients with and without spectral data.

Table 2.2 Effect of using spectral data on solar heat gain coefficient (SHGC).

#	Glazing Unit Description	SHGC		% Diff
		without spectral data	with spectral data	
1	CLR/.5" AIR/CLR	0.75	0.76	1.3
2	BRONZE/.5" AIR/CLR	0.62	0.62	0
3	GREEN/.5" AIR/CLR	0.50	0.51	2.0
4	AZURE/.5" AIR/CLR	0.46	0.46	0
5	CLR/.5" AIR/SG100	0.62	0.64	3.2
6	BRONZE/.5" AIR/SG100	0.50	0.51	2.0
7	GREEN/.5" AIR/SG100	0.40	0.43	7.5
8	AZURE/.5" AIR/SG100	0.36	0.40	11.1
9	CLR/.25" AIR/HM66/.25" AIR/CLR	0.36	0.37	2.8
10	BRONZE/.25" AIR/HM66/.25" AIR/CLR	0.30	0.30	0
11	GREEN/.25" AIR/HM66/.25" AIR/CLR	0.25	0.29	16.0
12	AZURE/.25" AIR/HM66/.25" AIR/CLR	0.23	0.28	21.7
13	CLR/.25" AIR/HM66/.25" AIR/SG100	0.32	0.35	9.4
14	BRONZE/.25" AIR/HM66/.25" AIR/SG100	0.27	0.29	7.4
15	GREEN/.25" AIR/HM66/.25" AIR/SG100	0.22	0.27	22.7
16	AZURE/.25" AIR/HM66/.25" AIR/SG100	0.21	0.26	23.8

§3.0 Spacer Effects on Edge-of-Glass and Frame Heat Transfer

The influence of varying the spacer effective conductivity on the heat transfer through the frame and edge-of-glass areas is discussed in this section. The results presented are taken from Reilly (1993) and are specific to fixed frames which have relatively short profiles (short frame height) as compared to other operator types. The influence of the spacer assembly on the heat transfer through the shorter frame is equal to or greater than it would be on a frame with a taller profile. The design of the frame and the position of the insulated glass unit in the frame will also affect the performance, although this is not considered here.

Arasteh (1989) analyzed the edge-of-glass heat transfer with respect to different spacer types, the position of the spacer relative to the sightline, and center-of-glass U-factors. The results show that as the center-of-glass U-factor decreases, the difference in edge-of-glass U-factors slowly increases between different spacer types. For example, the difference between the edge-of-glass U-factor with a single-seal aluminum spacer and a highly insulating spacer in a double-glazed unit with a center-of-glass U-factor of $0.50 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$ is $0.10 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$. In a double-glazed unit with a center-of-glass U-factor of $0.27 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$, the difference is $0.14 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$. The relationship between center-of-glass U-factor and edge-of-glass U-factor is not explored any further in this report.

Arasteh (1989) also investigated the impact of dropping the spacer below the sightline. By burying the glazing unit deeper into the sash or using a low-profile spacer, the heat transfer path from the exterior surface of the glass, through the spacer, and to the interior surface of the glass is effectively lengthened. This in turn reduces the heat loss through this area.

Fraser (1993) identified many of the differences between reported material conductivity values. Enermodal Engineering Limited (EEL, 1990) performed a sensitivity study to assess the impact of varying the material conductivities on window heat transfer. The results show that varying the conductivity of the elements which control the heat transfer through a window (the controlling resistance) can result in a difference of up to 25% in total window U-factor. This difference was found with a wood window in which the wood conductivity was varied by 40%. Nearly doubling the thermal conductivity of the thermal break in a thermally-broken aluminum window resulted in only a 4% difference in the combined frame and edge U-factor.

As with frames, the influence of variations in material conductivities on the heat transfer through a spacer assembly also depends on the controlling resistance. Fraser (1993) studied the difference between single-seal aluminum and dual-seal aluminum spacers. The disagreement between experimental and simulation results for single-seal spacers lead to the conclusion that the contact resistance at the glazing/spacer interface significantly reduces the heat loss. This interface is the controlling resistance in single-seal and dual-seal spacers. EEL (1990) took the analysis further and looked at the impacts from varying the spacer materials and dimensions on the edge-of-glass and frame heat transfer. Because the spacer assembly is not the controlling resistance in fenestration products, varying the material conductivities and dimensions over the

range of reported values has little effect on the total heat transfer through the edge-of-glass area and frame. The most significant influence is from varying the primary sealant thickness in dual-seal spacers and is explored further in this study.

Figure 3.1 Simulation model of the aluminum picture frame with a single-block spacer.

Figure 3.2 Simulation model of the vinyl picture frame with a two-block spacer model.

§3.1 Base Models

The base model for the study is a fixed frame with insulated glazing. The glazing system has an overall width of 0.75 inches, two lites of clear, double-strength glass, and a 0.51 inch air gap. Figures 3.1, 3.2, and 3.3 show the aluminum, vinyl, and wood fixed frame models respectively. In the figures, the left represents the outside and the right represents the inside. The fixed frame was chosen because of its relatively short profile (frame height). With a shorter frame profile, the influence of the spacer on the frame and edge-of-glass heat transfer is more pronounced. The aluminum frame has a frame height of 1.3 inches and the vinyl and wood frames have frame heights of 1.8 inches.

Figure 3.3 Simulation model of the wood picture frame with a detailed spacer model.

A spacer assembly consists of the spacer bar with primary and secondary sealants (Figures 3.4 and 3.5). In single-seal spacers, there is no secondary sealant and the primary sealant is behind the spacer bar. Dual seal spacers have a primary sealant between the glass and the spacer bar and the secondary sealant is behind the spacer bar. The difference between the two thermally-broken spacers is the thermal-break size and material. The spacer height is the distance from the bottom to the top of the spacer assembly. For this study, the bottom of the spacer assembly is flush with the bottom of the glass. Table 3.1 lists each of the spacers, the spacer height, the sealants, and the effective thermal conductivity. For more detailed information on the spacer assemblies see Elmahdy (1993), Fraser (1993), and Popular Science (1992).

Figure 3.4 Spacer Schematic of the Dual Seal Aluminum and Dual Seal C-Channel Spacers.

Figure 3.5 Spacer Schematic of the Thermally-Broken Spacer and the two-Block model for this spacer. The single-block model would simply be a rectangle the size of the two-block model.

Three different modeling approaches to the spacer were taken. The spacers were modeled as a single block (Figure 3.1), as two blocks (Figures 3.2 and 3.5), and in detail (Figures 3.3, 3.4, and 3.5). The effective thermal conductivities, k_{eff} of the spacers for the two-block and single-block models were determined by modeling the spacers in detail separate from the frame assembly. The effective conductivity is equal to the U-factor over a specified surface area multiplied by the overall width of the spacer assembly. The approach is described in detail in EEL (1993).

To account for the influence of the spacers on the heat transfer through the edge-of-glass and the frame areas, the simulation results are presented as U-factors for the edge-of-glass area, U_e , for the frame area, U_f , and for the combined edge-of-glass and frame areas, U_{e+f} (Figures 3.6, 3.7, and 3.8). The U-factor for the combined areas, U_{e+f} , best represents the influence a spacer can have on window heat transfer. Note that the center-of-glass U-factor, U_c , is held constant in this study at $0.49 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$.

The fixed window frame serves to show the maximum effect a spacer can have on frame and edge-of-glass heat transfer. The total window U-factor is omitted since the center-of-glass heat transfer is typically dominant in a fixed window and the impact of the spacer on total window heat transfer is therefore small. For frames with taller profiles, the influence on the edge-of-glass and frame heat transfer can be expected to be equal to or less than that shown here.

Table 3.1 Spacer Descriptions

Spacer	Spacer Bar Material	Primary Sealant	Secondary Sealant	Spacer Height (in)	k_{eff} Btu/h·ft ² ·°F
Single-Seal Aluminum	AL	Poly-sulphide	-	0.50	58
Dual Seal Aluminum	AL	PIB	Poly-sulphide	0.50	9.1
Dual Seal Aluminum	AL	PIB	Silicone	0.50	10.6
Low-Profile Dual Seal AL	AL	PIB	Poly-sulphide	0.43	9.8
Corrugated Strip	Butyl / AL	(Air)	-	0.43	4.1
Dual Seal C-Channel	Galv. Steel	PIB	Poly-sulphide	0.43	5.0
Dual Seal C-Channel	Stainless Steel	PIB	Poly-sulphide	0.43	2.3
Foam	Silicone	Urethane	-	0.50	1.4
TB Steel	Galv. Steel	PIB	Poly-sulphide	0.50	3.4
TB Steel	Galv. Steel	Urethane	-	0.50	2.2

KEY: AL=Aluminum; Galv. = Galvanized; k_{eff} = effective thermal conductivity for the single-block model; Spacer Height = distance from top of spacer to bottom of sealant behind the spacer; TB = thermally-broken.

Figure 3.6 Effect of spacer conductivity on U-factor for an aluminum fixed frame.

Figure 3.7 Effect of spacer conductivity on U-factor for a vinyl fixed frame.

Figure 3.8 Effect of spacer conductivity on U-factor for a wood fixed frame.

§3.2 Single- and Two-Block Models Compared to Detailed Model

In comparing U_{e+f} for the three modeling approaches, the single-block, two-block, and detailed models give identical results except for the thermally-broken spacers. For the thermally-broken spacers, the two-block model and the detailed model give the same results. For the vinyl frame only, the single-block model has a U_{e+f} that is 0.01 Btu/h·ft²·°F less than the two-block and detailed models.

§3.3 Variation in Material Properties

The influence of variations in material properties is seen with the dual-seal aluminum spacers and the C-channel spacers. With the dual-seal aluminum spacers, the sealants provide the controlling resistance. The conductivity of polyisobutylene is 1.66 Btu/h·ft²·°F. Figures 3.6, 3.7, and 3.8 show that significant increases in the conductivity of the primary sealant will not have a strong influence on U_{e+f} . However, decreasing the conductivity to 1.0 Btu·in/h·ft²·°F reduces k_{eff} of the dual seal aluminum spacer with a polysulphide secondary sealant to 6.5 Btu·in/h·ft²·°F. This reduces U_{e+f} by 0.01 Btu/h·ft²·°F - see Figures 3.6, 3.7, and 3.8. The conductance of the primary sealant is 83 Btu/h·ft²·°F. By reducing the conductivity to 1.0 Btu·in/h·ft²·°F, a conductance of 50 Btu/h·ft²·°F is obtained. This is equivalent to increasing the thickness of the primary sealant from 0.02 to 0.03 inches.

The difference in the secondary sealants of the two dual seal aluminum spacers - polysulphide has a thermal conductivity of $1.32 \text{ Btu}\cdot\text{in}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}$ and silicone has a conductivity of $2.5 \text{ Btu}\cdot\text{in}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}$ - results in a relatively small difference in k_{eff} (Table 3.1). Figures 3.6, 3.7, and 3.8 show that this also translates into a difference of $0.01 \text{ Btu}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}$. With respect to the spacer bar materials, galvanized steel has a thermal conductivity of $332 \text{ Btu}\cdot\text{in}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}$ and stainless steel has a thermal conductivity of $99 \text{ Btu}\cdot\text{in}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}$. The difference in $U_{\text{e+f}}$ for the two spacers is $.01 \text{ Btu}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}$.

§3.4 Variation in Effective Conductivity

Figures 3.6, 3.7, and 3.8 show the influence of varying the effective conductivity of a single-block model on the U-factors for each of the frame types for spacer heights of 0.50 and 0.43 inches. The figures clearly show that 0.5 inch spacer assemblies with effective conductivities less than $5 \text{ Btu}\cdot\text{in}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}$ have the strongest effect on the heat transfer through the frame and edge areas. Over the range of effective conductivities shown in the figures, $U_{\text{e+f}}$ differs by as much as $0.07 \text{ Btu}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}$ for the wood and vinyl frames, and $0.04 \text{ Btu}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}$ for the aluminum frame.

§4.0 Effects of Modeling Assumptions in Frame 3.1

The National Fenestration Rating Council (NFRC) requires the Frame 3.1 computer program to be used to model the thermal performance of window frames. Because the program uses a simple finite difference method to model window frames, there are some limitations that require special techniques to create an accurate model. These are issues that individual simulators may approach differently and it is important to determine how much impact on U-factors these differences might have. The issues that are addressed in this section are modeling exposed slanted surfaces, air cavities, thermal breaks, and aluminum frames.

NFRC does not require additional Frame 3.1 simulations to calculate solar heat gain coefficients. The solar heat gain coefficient for the frame is proportional to U_{frame} . The frame solar heat gain coefficient has a relatively small influence on the total window solar heat gain coefficient, so variations in U_{frame} are inconsequential. Refer to "Comparison of U.S. and Canadian Thermal Performance Procedures, Standards, and Accreditation and Certification Programs for Fenestration Products," (EEI 1995) for more detail.

§4.1 Exposed Sloped Surfaces

Since sloped surfaces cannot be directly specified in Frame 3.1, sloped sills are modelled using stepped surfaces. The boundary conditions must be modified to account for the resulting increase in outside surface area. The modified film coefficient is calculated according to

$$h_m = h_o(l_o/l_m), \quad \text{(Equation 2)}$$

where h_m is the modified film coefficient, h_o is the default film coefficient, l_m is the modified length (stepped) of exposed surface, and l_o is the true length (sloped) of exposed surface. The default film coefficient (h_o) for the outside surface (15 mph wind and 0°F) is 5.11 Btu/h•ft²•°F.

To quantify the effect of different modeling styles on U-factors, the sloped sill of a vinyl vertical slider is modelled using one, two, and four steps as shown in Figure 4.1.

Figure 4.1 Three styles of modeling a sloped sill.

For each model, the length of exposed surface used to model the sloped section is 2.000 inches. The length of the sloped section is 1.565 inches. Thus the modified film coefficient is 4.00 Btu/h·ft²·°F. The one step model is run using both the default and modified exterior film coefficients to determine the effect of neglecting the correction. The results of the four simulations are shown in Table 4.1.

Table 4.1 Effect on U-factors of number of steps used to model sloped surface in vinyl frame.

Vinyl Model	Conductance [Btu/h·ft ² ·°F]		
	U _{frame, Sill}	U _{edge, Sill}	U _{total, Window}
One Step, h = 5.11	0.4143	0.3061	0.319
One Step	0.4117	0.3060	0.318
Two Steps	0.3990	0.3055	0.318
Four Steps	0.3929	0.3053	0.318

The impact of neglecting to modify the exterior film coefficient is to decrease the frame conductance for the sill cross section by less than 1%. Between the one step and four step models with modified exterior film coefficients, U_{frame} changes almost 12%. When the sill is represented by multiple steps, the large air cavity is broken up into small rectangles. Breaking up air cavities decreases the overall conductance of heat and is discussed further in section 4.2.2. The NFRC guidelines require the effective conductivities of the smaller cavities to be adjusted to that of the equivalent larger rectangle. Also, inspection of the models shown in Figure 4.1 reveals that the models with more steps have a longer heat transfer path to the inside than the one step model. The relative importance of these two effects is established in section 4.2.3. The difference in U_{edge} between the three models is negligible.

Typically the sill is the only member with an exposed, sloped surface. If the sill and its edge-of-glass area account for 10% of the window area, then the modelling assumptions have less than a 1% impact on the total window U-factor.

To determine what effect these same modeling assumptions have on an aluminum window, the vinyl in the above model is replaced with aluminum. The results presented in Table 4.2 demonstrate that correcting the exterior film coefficient has a greater impact (slightly more than 1%) on the aluminum frame because of its greater conductivity, but it is still negligible for the whole window. Changing the number of steps in the sill has no significant impact on U-factors because the heat loss is through the aluminum and the air cavities provide no resistance to the heat flow.

Table 4.2 Effect on U-factors of number of steps used to model sloped surface in aluminum frame.

Aluminum Model	Conductance [Btu/h·ft ² ·°F]		
	U _{frame} (sill)	U _{edge} (sill)	U _{total} (window)
One Step, h = 5.11	1.7186	0.3972	0.709
One Step	1.6988	0.3962	0.708
Two Steps	1.6979	0.3962	0.708
Four Steps	1.6977	0.3960	0.708

§4.2 Air Cavities

§4.2.1 User-defined Air Cavities

The heat transfer through an air cavity in a frame model is estimated using an effective conductivity that accounts for conduction, convection, and radiation. The effective conductivity is calculated by the program using either user-specified temperatures and emittances or default conditions depending only on the emittances of the surfaces abutting the cavity.

To quantify the effects of using default air cavities, the air cavities in a single-hung vinyl window are modelled both using default conditions and specifying the actual cavity temperatures. For example, the five largest air cavities in the original, one step vinyl sill model were modified to use the temperatures calculated by running the model with the default conditions. Table 4.3 presents both the default (DF) and user-defined (UD) temperatures on the cold and warm sides of the five largest cavities, as well as the resulting conductivities. The difference in individual cavity conductivity is as much as 20% for the largest cavity. The effect on smaller cavities is as little as 0%.

Figure 4.2a shows the effective conductivities calculated using the default temperatures. Figure 4.2b shows the temperatures calculated using the default conductivities that were then used to define new effective conductivities. Figure 4.2c shows the resulting effective conductivities. Significant differences in effective conductivities are evident. The conductivities of the large air cavities in the other four cross sections were also redefined to determine the total effect on the window U-factor. Table 4.4 illustrates the effects on the sill and total window U-factors. While the general effect of redefining the air cavity conductivities with more accurate temperatures is to increase the U-factors, the change in total window U-factor is just 1.6%.

Table 4.3 Effect of using default air cavities on temperatures and effective conductivities.

Cavity	Temperature [$^{\circ}\text{F}$]				K_{eff} [Btu·in/h·ft 2 · $^{\circ}\text{F}$]		% Diff.
	Cold, DF	Cold, UD	Warm, DF	Warm, UD	DF	UD	
1	12.8	13.1	49.4	49.2	0.48	0.48	0
2	52.6	52.4	66.2	66.0	0.60	0.62	+3.2
3	15.8	16.6	46.5	45.7	1.05	1.18	+11.0
4	14.8	15.1	45.8	45.1	1.49	1.67	+10.8
5	2.5	2.4	14.5	16.0	1.28	1.07	-19.6

Figure 4.2 Modeling air cavities using default (a) and user-defined (b and c) conditions.

Table 4.4 Effect on U-factors of using default conditions for air cavities.

Model	Conductance [Btu/h·ft 2 · $^{\circ}\text{F}$]			
	U_{edge} (sill)	U_{frame} (sill)	U_{total} (sill)	U_{total} (window)
Default	0.3060	0.4117	0.3669	0.318
User-Defined	0.3062	0.4238	0.3739	0.323

§4.2.2 Breaking Up Large Air Cavities

Modeling sloped sills is discussed in section 4.1. It is noted that using several steps resulted in the air cavity in the sloped section of the sill being broken up into small rectangles. The relative dimensions and overall size of the air cavity dictates the amount of convective heat transfer occurring. In general, smaller air cavities have less convective heat transfer than larger ones and therefore have lower effective conductivities.

To quantify the effects of breaking up air cavities, an air cavity is modelled independently (not part of a window frame) as one, two, and three rectangles (see Figure 4.3). The temperature is defined as 10 °F on the cold side and 50 °F on the warm side. The emittance on both sides is taken to be 0.9. The overall dimensions of the cavity are 1.5 inches by 1.2 inches (width by height). The resulting total U-factors are summarized in Table 4.5. The effect of dividing a cavity into smaller rectangles is to reduce the conductance of the air pocket by as much as 57%.

a)	$e = 0.9$ $T = 10^{\circ}\text{F}$	$K = 1.80$	$e = 0.9$ $T = 50^{\circ}\text{F}$
b)	$e = 0.9$ $T = 10^{\circ}\text{F}$	$K = 1.36$ $K = 1.36$	$e = 0.9$ $T = 50^{\circ}\text{F}$
c)	$e = 0.9$ $T = 10^{\circ}\text{F}$	$K = 0.77$ $K = 0.77$ $K = 0.77$	$e = 0.9$ $T = 50^{\circ}\text{F}$

Figure 4.3 Air cavity modelled as one (a), two (b), and three (c) rectangles and resulting effective conductivities.

Table 4.5 Effect on U-factors of breaking up air cavities in the frame model.

Model	U_{total} [Btu/h·ft ² ·°F]	% Difference (from one cavity)
One Cavity (1.5" x 1.2")	1.20	0
Two Cavities (1.5" x 0.6" each)	0.91	24.2
Three Cavities (1.5" x 0.4" each)	0.52	56.7

To see the theoretical maximum effect that breaking up air cavities could have in an actual window simulation, the sloped vinyl sill is modeled with all of the air cavities broken up so that the effective conductivities are close to $0.3 \text{ Btu}\cdot\text{in}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}$. The U_{edge} for this model is $0.3045 \text{ Btu}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}$ and the U_{frame} is $0.2391 \text{ Btu}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}$. This represents a 70% decrease in sill frame U-factor. Extrapolating these results by noting that the frame is 28% of the total window area and assuming a similar difference could be achieved in other cross sections, breaking up large air cavities could have as much as a 20% effect on total window U-factor, although such an extreme case would never actually occur.

The NFRC requires this issue to be addressed in the following manner. An air cavity with a total area equal to the area of the whole cavity and width equal to the width of the whole cavity in the direction of heat flow must first be modelled to determine the correct conductivity. The smaller rectangles comprising the air pocket are assigned the same conductivity. The effect of making this adjustment in an actual sill model is illustrated in Table 4.6. The effective conductivity calculated in the "one step" model ($1.28 \text{ Btu}\cdot\text{in}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}$) is used in all of the air cavity rectangles comprising the sill cavity in the two and four step models.

Table 4.6 U-factors in sloped vinyl sill with user-defined conductivity (compare to Table 4.1).

Vinyl Model	Conductance [$\text{Btu}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}$]		
	U_{frame} (sill)	U_{edge} (sill)	U_{total} (window)
One Step	0.4117	0.3060	0.318
Two Steps	0.4041	0.3058	0.318
Four Steps	0.4044	0.3056	0.318

If the results in Table 4.6 are compared to those in Table 4.1, it becomes apparent that modeling a sloped sill with four steps produces a U_{frame} that is within 2% of the U_{frame} for the one step model if the effective conductivity of the air cavities is corrected. If default effective conductivities are used, the U_{frame} of the four step model differs from the U_{frame} of the single step model by up to 5%. Once the effective conductivities have been corrected, the residual difference in U_{frame} is due to the increased path length for heat flow in the multiple step models.

§4.3 Thermal Breaks

The following discussion of thermal break modeling is partially based on material presented by EEL (1992).

§4.3.1 Thermal Break Width

The controlling resistance in a thermally-broken aluminum window is the width of the thermal break. To determine the relationship between total window U-factor and thermal break width, a casement window with a thermally-broken aluminum frame was modelled with gaps of various widths (see Figure 4.4). The results of the comparison are presented in Table 4.7 and in Figure 4.5. As the thermal break width increases from 0.05 to 0.2 inches (400%), the total window U-factor decreases by nearly 7%. Increasing the width beyond 0.2 inches achieves lower U-factors but the difference for another 0.15 inches is only 4%.

(a) Gap = 0.100 inch

(b) Gap = 0.200 inch

(c) Gap = 0.400 inch

Figure 4.4 Casement window cross section with different thermal break widths.

Figure 4.5 Total window U-factor as a function of thermal break width.

Table 4.7 Effect of thermal break width on U-factors.

Width of Break (inches)	Conductance [Btu/h·ft ² ·°F]			
	U _{edge} (section)	U _{frame} (section)	U _{total} (section)	U _{total} (window)
0.050	0.5844	1.0239	0.7935	0.656
0.100	0.5738	0.9611	0.7581	0.637
0.150	0.5664	0.9150	0.7322	0.623
0.200	0.5605	0.8786	0.7118	0.611
0.250	0.5562	0.8496	0.6958	0.603
0.300	0.5523	0.8234	0.6813	0.595
0.325	0.5505	0.8112	0.6745	0.591
0.350	0.5488	0.8001	0.6683	0.588
0.375	0.5484	0.7935	0.6650	0.586
0.400	0.5458	0.7799	0.6572	0.581
0.450	0.5433	0.7624	0.6475	0.576
0.500	0.5410	0.7463	0.6387	0.571

Figure 4.6 Total window U-factor as a function of thermal break material conductivity.

§4.3.2 Thermal Break Conductivity

Another critical factor in modeling thermal breaks is the conductivity of the thermal break material. Using a gap width of 0.200 inches, the same casement window is modelled using various thermal break materials. The results of the comparison are presented in Table 4.8 and Figure 4.6. Decreasing the conductivity of the thermal break from 3.61 to 0.21 Btu·in/h·ft²·°F decreases the total window U-factor by 17%.

Table 4.8 Effect of thermal break conductivity on U-factors.

Conductivity of Thermal Break [Btu·in/h·ft ² ·°F]	Conductance [Btu/h·ft ² ·°F]			
	U _{edge} (section)	U _{frame} (section)	U _{total} (section)	U _{total} (window)
0.21 (polyurethane foam)	0.5217	0.6011	0.5595	0.528
1.18 (rigid vinyl)	0.5480	0.7902	0.6632	0.585
1.60 (fiberglass-reinforced nylon)	0.5547	0.8360	0.6885	0.599
2.15 (urethane)	0.5615	0.8809	0.7135	0.612
3.61 (high density polyethylene)	0.5730	0.9533	0.7539	0.634

§4.4 Interior Film Coefficients in Non Thermally-Broken Aluminum Frames

The controlling resistance in a non thermally-broken aluminum frame is the exposed surface area on the inside. The amount of heat transfer occurring at this surface is dictated by the surface area and the interior film coefficient. The interior film coefficient is calculated in Frame 3.1 using the following equation from the 1993 ASHRAE Fundamentals on page 27.4:

$$h_i = 0.27 \cdot (T_i - T_s)^{0.25} + [e_s \cdot \sigma \cdot (T_i^4 - T_s^4)] / (T_i - T_s), \quad (\text{Equation 3})$$

where h_i is the interior film coefficient in Btu/h·ft²·°F, T_i is the inside air temperature (70°F), T_s is the inside surface temperature, e_s is the inside surface emittance, and σ is the Stefan-Boltzmann constant (1.712E-9 Btu/h·ft²·°R⁴). The temperatures in the second part of the equation must be in degrees Rankine. An assumption must be made for the temperature and the emittance of the inside surface of the frame.

Frame 3.1 assumes a surface emittance of 0.84 and a surface temperature of 20 °F for an aluminum frame. To examine how accurate this assumption is and what effect it has on the total window U-factor, a single-hung window is modelled with four different values of emittance.

Since Frame 3.1 allows you only to specify the surface film coefficient and the air temperature at the boundary (70 °F), the emittance must be adjusted indirectly. An interior film coefficient is guessed, the Frame 3.1 model is run, and the resulting temperatures on the interior surface are examined. Another iteration is performed with a new interior film coefficient and the resulting temperatures are compared to the temperatures that would be required to produce the desired emittance. For example, to achieve a surface emittance of 0.84 (Frame's default emittance), the boundary is first given a film coefficient of 1.461 Btu/h·ft²·°F. After running Frame, the local temperature feature is used to find the temperature of the inside surface of the aluminum frame (34 °F). Putting this temperature and emittance into the above equation yields a film coefficient of 1.433 Btu/h·ft²·°F. Using a film coefficient of 1.433 Btu/h·ft²·°F resulted in convergence on the inside surface temperature. This same process was used to determine what emittance and surface temperature would be required to produce a film coefficient of 1.461 Btu/h·ft²·°F for this window. To determine what impact the assumption of surface emittance has on the total window U-factor, emittances of 0.5 and 0.2 were also simulated. The relationship between surface temperature and film coefficient is illustrated in Figure 4.7 for three values of emittance. Increasing the inside surface temperature by 50 °F decreases the film coefficient by 15%. The impact of emittance on total window U-factor is shown in Table 4.9 and Figure 4.8. Increasing the frame emittance from 0.2 to 0.87 increases the total window U-factor by 15%.

Figure 4.7 Relationship between inside surface temperature and inside film coefficient for three different surface emittances.

Table 4.9 Impact of frame emittance on total window U-factor.

Emittance of Aluminum	Interior Surface Temperature [°F]	Interior Film Coefficient [Btu/h·ft ² ·°F]	Total Window U-factor [Btu/h·ft ² ·°F]
[0.84]	[20]	1.461	0.7533
0.84	30	1.433	0.7491
0.87	34	1.461	0.7533
0.5	30	1.134	0.7018
0.2	27	0.8718	0.6554

[] indicates Frame default values. These values do not correlate with the given film coefficient for this window.

Figure 4.8 Total window U-factor as a function of frame emittance.

§4.5 Wide Glazing Cavities

Windows with storm panels can have a gap width of greater than one inch between the primary glazing unit and the storm panel. There are three potential mistakes that could be made in the simulations:

- 1) The gap width is greater than 1.375 inches and is modelled as so in Window 4.1. Any gap widths greater than 1.375 inches should be modelled at 1.375 inches as discussed in section 2.1.

For example, a window with a glazing gap of 3.83 inches, correctly modeled as 1.375 inches, has a center-of-glass U-factor of 0.51 Btu/(h•ft²•°F). If it is modeled with an air gap of 3.83 inches, it has a center-of-glass U-factor of 0.53 Btu/(h•ft²•°F). This is almost a 4% difference.

2) The gap is wider than it is tall and a correction is not made in the Frame 3.1 model. In the NFRC glazing cavity option, Frame 3.1 assumes that the direction of heat flow is parallel to the shortest side of the rectangle. This is not true when the glazing cavity is wider than it is tall, so the simulator must calculate the effective conductivity of the cavity and input it as a user-defined conductivity in the Frame 3.1 model. The effective conductivity is calculated as

$$k_{\text{eff}} = U_{\text{cog}} \cdot (T_{\text{out}} - T_{\text{in}}) \cdot \text{gap width} / (T_{\text{cold}} - T_{\text{warm}}) \quad (\text{Equation 4})$$

where U_{cog} is the center-of-glass U-factor from Window 4.1, T_{out} is the outside air temperature, T_{in} is the room air temperature, T_{cold} is the temperature of the cold-side surface bounding the gap, and T_{warm} is the temperature of the warm-side surface bounding the gap.

3) The pocket of air beneath the glazing cavity is modelled as a frame air cavity. It should be modelled as a glazing cavity since there is no physical separation.

To illustrate points 2 and 3 above, an aluminum horizontal slider with a storm panel separated from the primary glazing by 3.83 inches (see Figure 4.9) is modeled in six ways. First, Equation 4 is used to calculate an effective conductivity of 3.74 Btu•in/h•ft²•°F. This value is used to define rectangle 1 in models KEUD, KEFF, KENF, and KEAC. It is also used to define rectangles 2,3,4, and 5 in model KEFF. In model KEUD, rectangles 2,3,4, and 5 are defined using the "user-defined glazing cavity" option in Frame 3.1. The temperatures and center-of-glass U-factor from Window 4.1 are used and since the user specifies the direction of heat flow, the problem of inverting dimensions in a wide glazing cavity is avoided. This is the approach recommended in the NFRC Simulation Laboratory Workbook (NFRC, 1993). In model KENF, rectangles 2,3,4, and 5 are defined using the NFRC glazing cavity option in Frame 3.1, thus incurring the error discussed in point 2 above. In model KEAC, rectangles 2,3,4, and 5 are defined using the NFRC frame air cavity option in Frame 3.1, thus incurring the error discussed in point 3 above.

In model UDGC, all of the rectangles are defined using the "user-defined glazing cavity" option in Frame 3.1 with temperatures and center-of-glass U-factor from Window 4.1. In model NFRC, all of the rectangles are defined using the NFRC glazing cavity option in Frame 3.1, thus incurring the error discussed in point 2 above. The total window U-factors obtained by running these six models are presented in Table 4.10. Assuming model KEUD to be the most accurate, the percent difference of each other model from KEUD is calculated.

Study of the results presented in Table 4.10 suggest that there are three methods that are acceptable for modelling wide glazing gaps, represented by models KEUD, KEFF, and UDGC. These results reflect the fact that the errors described in points 1, 2, and 3 above are avoided in

Figure 4.9 Aluminum window with storm panel located 3.83 inches from primary glazing.

Table 4.10 Effect of defining wide glazing cavities in different ways on U-factor in aluminum window with storm panel 3.83 inches from primary glazing.

Model Name	Model Description	$U_{\text{total (window)}}$ Btu/h·ft ² ·°F	% Diff.
KEUD	$K_{\text{eff}} = 3.74 \text{ Btu}\cdot\text{in}/\text{h}\cdot\text{ft}^2\cdot\text{°F}$ in rectangle 1/ user-defined glazing cavities in rectangles 2,3,4,5	0.5252	0
KEFF	$K_{\text{eff}} = 3.74 \text{ Btu}\cdot\text{in}/\text{h}\cdot\text{ft}^2\cdot\text{°F}$ in all rectangles	0.5349	+1.8
KENF	$K_{\text{eff}} = 3.74 \text{ Btu}\cdot\text{in}/\text{h}\cdot\text{ft}^2\cdot\text{°F}$ in rectangle 1/ NFRC glazing cavities in rectangles 2,3,4, and 5	0.4803	-8.5
KEAC	$K_{\text{eff}} = 3.74 \text{ Btu}\cdot\text{in}/\text{h}\cdot\text{ft}^2\cdot\text{°F}$ in rectangle 1/ frame standard air cavities in rectangles 2,3,4, and 5	0.4831	-8.0
UDGC	User-defined glazing cavities in all rectangles using Window 4.1 temperatures	0.5223	-0.6
NFRC	NFRC glazing cavities in all rectangles using Window 4.1 temperatures and center-of-glass U-factor	0.4492	-14.5

these three approaches. Another words, wide glazing gaps could be modeled equivalently using either a user-calculated effective conductivity (Equation 4) or the temperatures and center-of-glass U-factor determined in Window 4.1. If one of these three approaches is not used, errors in total window U-factor of up to 14.5% could be realized.

§5.0 Conclusions

The intent of this report is to quantify and clarify the importance of differences in simulation styles and assumptions in using the Window 4.1 and Frame 3.1 computer programs to determine U-factors and solar heat gain coefficients of windows. The report addresses center-of-glass, spacer, and frame modeling in separate sections. The modeling differences discussed in this report include issues that must be decided by the simulator (using engineering judgement) due to lack of information or limitations of the computer program and mistakes due to user carelessness or errors in the computer program. The efficacy of various techniques at reducing total window U-factors is also addressed.

In most windows, the center-of-glass area accounts for an average of 50% of the window area. Therefore, differences in center-of-glass modelling potentially have the greatest impact on total window U-factors. Luckily, the center-of-glass area is more straight-forward to model than the frame and edge-of-glass areas.

The largest errors in center-of-glass U-factor occur at gap widths less than 0.438 inches and with misplacement of a low-emittance coating on an exposed glass surface. Placing the low-E coating on the wrong surface can create as much as a 55% difference in center-of-glass U-factor. For solar heat gain coefficient, the largest errors occur by not using spectral data files and misplacing coatings. Omitting the spectral data can decrease the solar heat gain coefficient by up to 23.8%. Placing the low-E coating on the wrong surface can create as much as a 5% difference in solar heat gain coefficient. Effects of gap width and gas fill are negligible on solar heat gain coefficients.

Although spacers are not the controlling resistance in fenestration products, they can have a significant impact on the total window U-factor. The greatest variation in combined edge and frame U-factor (U_{e+f}) is $0.07 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$ and occurs with a change in spacer effective conductivity. The effect of variation in effective conductivity on total window U-factors is between 0.0 and $0.05 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$, depending on the spacer type. Spacers with effective conductivities of less than $5 \text{ Btu}\cdot\text{in/h}\cdot\text{ft}^2\cdot^\circ\text{F}$ have the most significant impact on the heat transfer. Simplifying the spacer in the simulation model of the frame section will give the same U-factor as the detailed model. A difference of $0.01 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$ is found with the single-block model of the thermally-broken spacers. The uncertainty in material properties can result in variations in U_{e+f} of $0.01 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$. It is important to clearly identify the materials used in a spacer assembly because the performance can vary significantly, as seen with steel products. Differences in the performance of the material providing the controlling resistance have the strongest influence.

The frame area generally accounts for less than thirty percent of the window area, but some of the trickiest aspects of window modeling occur in the frame. Theoretically, breaking up large air cavities could have as much as a 20% effect on total window U-factor. Although such a case is extreme, it illustrates the importance of redefining the effective conductivities of broken-up air

cavities, especially in vinyl windows. A mistake in specifying the conductivity of the thermal break material could change the total window U-factor by as much as 17%. Modeling an aluminum window using default interior film coefficients could make a difference in total window U-factor of 15% if the window had a low-emittance surface on the inside. A wide glazing gap incorrectly modeled with NFRC glazing cavities could incur an error in total window U-factor of up to 14.5%. A mistake in modeling the thermal break width could produce a total window U-factor difference of nearly 7%. Other modeling issues discussed in the body of the report have less than a 5% impact on total window U-factors.

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**THE HARMONIZATION OF CANADIAN AND U.S.
WINDOW PROGRAMS AND STANDARDS**

**DEPARTMENT OF ENERGY
GRANT # DE-FG36-94G010044**

**TASK 6:
CONDENSATION RESISTANCE MODEL**

MARCH 22, 1995

**enermodal engineering, inc.
1554 Emerson Street
Denver, CO 80218
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March 22, 1995

Enermodal Engineering, Ltd. has incorporated algorithms into the FRAME program to model condensation resistance. The condensation resistance model included in FRAME 4.0 is based on the work of John Wright (Wright 1995a). The use of this model, and comparison of simulation and test results are documented in the paper, "Numerical Prediction of Window Condensation Potential," by Alex McGowan (1995) of Enermodal Engineering, Ltd. Mr. McGowan presented the paper at the January 1995 ASHRAE Meeting.

Wright (1995a, 1995b) developed a two-dimensional model to simulate the heat transfer through insulated glazing units. This model accounts for natural convection, conduction, and radiation exchange between the glazing layers. Wright's model differs from the one-dimensional heat transfer programs WINDOW and VISION in that it is a two-dimensional solution and includes the effects of the spacer assemblies separating the glazing layers.

Wright assumes isothermal planes in his model, as well as constant exterior film coefficients. The model predicts temperatures that are higher than the center-of-glass temperature at the head and lower than the center-of-glass temperature at the sill. Test results show the same pattern; however, the measured temperatures are 2°C to 5°C different from the predicted temperatures (McGowan 1995). The differences are due primarily to the assumptions of isothermal planes and constant film coefficients. Also, the results clearly show a dependence on frame type. The larger temperature difference between predicted and measured results is seen with metal frames.

McGowan (1995) incorporated Wright's model into the FRAME program and compared the results with the existing FRAME program, finite-element analysis, and test measurements of surface temperatures. The existing FRAME program only accounts for conduction heat transfer. McGowan found good agreement between the finite-element analysis program and the FRAME version using Wright's model. Interesting to note is the strong effect frame type has on the measurements. Highly conductive metal frames show even lower temperatures than

Condensation Resistance Model

are predicted by the models. More than likely this is due to the conduction heat transfer between the glazing and frame. The lower surface temperature at the frame would also depress the local air temperature at the edge.

Wright (1995b) presents the simplified method for the two-dimensional model for condensation resistance that has been implemented in the VISION program. For a given glazing system, the one-dimensional solution is found first. The glazing system configuration and properties, along with the glazing temperatures are used in the two-dimensional solution of the velocity field. The glazing cavity is assumed to be a vertical, tall cavity with isothermal walls and adiabatic ends. The velocity field is determined from the balance of buoyant and viscous forces that exist through the center-of-glass region. The difference between the actual velocity field based on true wall temperatures from this simplified case is negligible. This process is performed for each glazing cavity.

The National Fenestration Rating Council formed a Condensation Resistance task group to evaluate test and calculation methods for predicting interior temperatures around the perimeter of the glazing in a window. The work discussed above is available to the task group and was presented at the recent ASHRAE meeting.

To assist Lawrence Berkeley Laboratory in developing a condensation resistance model for the WINDOW program, we have attached the 3 papers mentioned above. The paper by Wright (95b) will be presented at the Window Innovations Conference in June, and provides more detail on implementing the algorithms. The computer code for the algorithms is not available for public release. Both the VISION model and the FRAME approach are proprietary.

NUMERICAL PREDICTION OF WINDOW CONDENSATION POTENTIAL

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INTRODUCTION

Although a substantial amount of effort has been expended to develop numerical methods for determining windows U-factors (EE 1993; Goss and Curcija 1994; Standaert 1985; CSA 1993a; NFRC 1991), there has been little work to date on using numerical methods to predict condensation potential. It is perhaps of direct interest to most ASHRAE members to determine heat loss and solar gains through windows as a precursor to sizing heating and cooling equipment, but condensation has long been recognized as an extremely important issue for consumers (and, consequently, for window manufacturers).

Moreover, building scientists recognize the link between condensation and increased energy consumption (due to latent loads), reduced occupant comfort and indoor air quality (from the presence of bacteria and mold), and structural damage (where accumulated condensation is absorbed by the building materials, thus reducing their structural stability). The National Fenestration Rating Council (NFRC) is developing a rating method for condensation potential in fenestration products as part of its mandate from the U.S. Department of Energy (DOE). A rating method would benefit from the use of simulation as a supplement to physical condensation resistance testing to reduce the cost and time required for implementation and increase the flexibility of the rating method.

This paper outlines one of the necessary components in the application of numerical methods for evaluating condensation in fenestration products. The theoretical approach and its practical application are discussed, as well as some comparisons between numerical prediction and physical test results for a sample of products.

THEORETICAL BACKGROUND

The first step in developing a numerical method to characterize condensation involves replicating the current test procedure. This will help to define the mechanisms at work for the controlled conditions in the laboratory. As there are well-documented results from laboratory testing available, we can use these data to guide the development of the numerical model.

The approach followed here is to compare results along the centerline of several products that were tested in an ear-

lier research project. This will not address concerns such as hardware or corner effects, but will provide an indication of whether numerical analysis can contribute to the prediction of condensation resistance potential. If it appears that a numerical model can adequately characterize the relatively controlled laboratory test situation, only then can we attempt to characterize other aspects of condensation (such as field performance, which is a much more difficult problem).

As Figure 1 shows, four main mechanisms affect the formation of condensation on a fenestration product in a test: the nature of the exterior surface convection; the nature of the interior surface convection; conductive heat transfer in the solid components of the window frame, spacer, and glazing systems; and convective and radiative heat transfer within the glazing cavity. We are not considering infiltration as a mechanism, as the current test procedure proposed by the NFRC calls for sealing and pressure-balancing the test specimen to remove infiltration from the evaluation. Test results may also include the effects of "wind-washing" (convection in cavities open to the exterior, caused by vorticity in the blower), which varies from lab to lab according to equipment, but this issue is beyond the scope of the current investigation.

Exterior and interior convective heat transfer is the subject of research by others (Yazdanian and Klems 1994; Cur-

Figure 1 Mechanisms affecting condensation resistance.

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cija and Goss 1993), and the conduction in the solid components of the window is well characterized (Carpenter and McGowan 1989; Reilly 1994; Wright et al. 1994), so we have chosen to concentrate on the convection and radiation within the glazing cavity.

Recent work at a Canadian university has produced a model for convective motion within the glazing cavity (Wright 1994), but that work assumed isothermal glazing surfaces and intentionally ignored the interaction between center-glass heat transfer and conduction in the spacer and frame assembly. This paper extends the work of the Canadian university in predicting condensation potential by including the two-dimensional effects in the edge-glass region, and considering the film coefficients and conduction in the solid components of the window assembly.

The method chosen was to integrate the university's convective model with the existing finite-volume calculation method used in FRAME (EE 1992a). FRAME is a two-dimensional, finite-volume calculation program used to determine total-window and window-component U-factors. The program is the standard program used to determine frame and edge-glass U-factors in Canada (CSA 1993a) and the United States (NFRC 1991).

FRAME is a conduction-only program and uses an effective conductivity to account for increased local heat transfer due to convection and radiation (see Carpenter and McGowan [1989]). This simplification is adequate for calculation of U-factors, as it accurately characterizes the overall heat transfer through a window or similar building assembly, but is not appropriate for local temperature calculations. By incorporating the university's convection model for the glazing cavity into FRAME, we can use the existing and well-documented techniques for analyzing window thermal performance (EE 1992b; CSA 1993b) to get a prediction of window condensation potential.

This method involves calculating the steady-state velocity field in the glazing cavity (assuming isothermal glazing surfaces and laminar flow, as described in Wright [1994]) and the temperature distribution across the cavity at the boundary between the edge-glass and center-glass regions—63.5 mm (2.5 in.) from the sightline.

The velocity field is used to determine the convective relationship between nodes in the glazing cavity in the finite-volume network and the temperature distribution is used to define boundary conditions at the edge-glass/center-glass interface. This accounts for energy convected into the edge-glass region from the center-glass region.

Radiative heat transfer across the glazing cavity is modeled by adding an energy source term equal to the radiative heat transfer calculated by VISION for the center-glass region to the cells on one side of the cavity and subtracting the same amount of energy from the cells on the other side of the cavity. This simulates radiative heat transfer from any point on one surface of the cavity to a point directly across the glazing cavity on the other side (i.e., the shape factor across

the cavity is 1.0, and we do not model radiative heat transfer from the inner glazing surface to the spacer or from the spacer to the outer glazing).

COMPARISON TO LAB TEST RESULTS

Having introduced a model for convection in the glazing cavity into the simulation method, it remains to be seen whether this produces any improvement in the capability of the simulation to predict condensation potential. To that end, the model was compared to results from physical testing at the NRCC and to a more complex computer model used by researchers at a U.S. university (FDI 1989) for five test specimens. The original testing at the NRCC and modeling at the U.S. university were described by Carpenter (1992) and EE (1993) for a previous ASHRAE research project. The specimens were selected to have a pronounced edge effect, as the intent was to determine the accuracy of the model in that region of the window. Therefore, these should be adequate samples to test the validity of the condensation resistance model. The specimens are

- a vinyl-frame horizontal slider that is 1,512 mm (59.5 in.) high and 902 mm (35.5 in.) wide with clear double glazing, air fill, and insulating spacers (the comparison was made on a vertical section through the center of the stationary panel);
- an aluminum-clad wood patio door measuring 2,038 mm (80.2 in.) high and 1,793 mm (70.6 in.) wide with clear double glazing, air fill, and a dual-seal aluminum spacer (on a vertical section through the center of the stationary panel);
- a wood, double-hung window measuring 1,835 mm (72.2 in.) high and 1,160 mm (45.7 in.) wide with a low-e ($\epsilon = 0.09$) coating on surface 3, argon fill, and a dual-seal aluminum spacer (compared on a vertical section through the center of the window);
- a vinyl-clad wood casement window measuring 1,518 mm (59.8 in.) high and 720 mm (28.3 in.) wide with clear double glazing, air fill, and a dual-seal aluminum spacer (compared on a vertical section through the center of the window); and
- an aluminum-frame horizontal slider measuring 904 mm (35.6 in.) high and 1,510 mm (59.4 in.) wide with clear double glazing, air fill, and a dual-seal aluminum spacer (on a vertical section through the center of the operator panel).

The specimens were tested in a guarded hot-box apparatus in accordance with the NRCC standard testing procedure (Bowen 1985). This procedure allows the tester to determine the film coefficients for the actual test specimen (rather than for a calibration panel), and the film coefficients determined for the test specimens are thus available for use in the simulation.

The specimens were sealed to prevent air infiltration and were tested at 21°C and -18°C air temperatures. The specimens were also simulated with a finite-element model at the U.S. university. Comparisons of interior surface temperatures from each method (including FRAME with and without the condensation model) are shown in Figures 2 through 6.

Figure 2 compares results for the vinyl horizontal slider, using four methods: physical testing (which provides measurements at six discrete points), finite-element analysis (labeled "FEM" in Figure 2), and FRAME with and without the convective model ("FRAME (conv)" and "FRAME (old)," respectively). Test and finite-element results show good agreement except at the top of the window ($y \approx 880$ mm), where the actual film coefficient in the test is lower than the average for the interior surface because the natural convective flow is not fully developed. This provides an indication of the magnitude of the error introduced by using an average film coefficient (derived from the test results) for the entire window and ignoring localized variations in convective and radiative surface heat transfer. It is evident that this issue should be addressed if simulation is to match testing in this region of the window.

The difference between the FRAME temperature prediction with and without the convective model is interesting. The temperature at the lower sightline ($y \approx 65$ mm) is 5.6°C colder when convection is modeled, and the temperature at the upper sightline ($y \approx 830$ mm) is 2.2°C warmer. This agrees with what would be expected when convection is included.

One point of interest in condensation resistance testing on the glass is 50 mm above the lower sightline ($y \approx 90$ mm in Figure 2), a specified test thermocouple location (AAMA 1981; CSA 1990). Figure 2 shows two measurements at this spot: the one labeled "Test" refers to a thermocouple attached to the specimen with masking tape, whereas the "Test B" thermocouple was attached with metal foil tape.

There has been some discussion regarding the appropriate attachment method for thermocouples, and this comparison highlights those problems. The metal foil tape provides better thermal contact between the glass surface and the thermocouple, so the reading is lower, but the metal surface of the tape does not accurately represent the radiative coupling to the climatic chamber. The paper tape has a surface emissivity similar to that of the glass, so it is more representative of the radiative heat transfer that would occur if the thermocouple were not present, but the paper tape does not provide thermal contact to the glass that is as good as the metal tape. The difference is small (1°C at $y \approx 90$ mm and 1.5°C at the center of the glass, or $y \approx 450$ mm), but the issue is not yet resolved. The "real" value is likely somewhere between these two readings.

As Figure 2 shows, the convective model is much closer to the measured temperature than the conduction-only model. Depending on which measured temperature the pre-

Figure 2 Vinyl horizontal slider—comparison of surface temperatures.

diction is compared against, the convective model is either 0.2°C or 1.2°C colder than measured temperatures and is 1°C colder than temperatures predicted by finite-element analysis at a point 25 mm (1 in.) above the lower sightline.

Figure 3 shows a similar comparison for the patio door specimen, which is quite tall. Again, the difference between finite-element analysis and test measurements is greatest at the top of the specimen, where convective flow is not developed (and where condensation resistance is not usually an issue). Again, the convective model predicts temperatures that are colder at the lower sightline (by 3.7°C) and warmer at the upper sightline (by 1.7°C) than the conduction-only model. In this case, the convection model agrees with measured values 50 mm (2 in.) above the lower sightline. Agreement between the convection model and the finite-element analysis is quite good, except in the region of the sill frame ($0 < y < 30$ mm).

Figure 4 shows a comparison for the wood double-hung window. This specimen was not analyzed using the finite-element method, so comparison is not available. The convec-

Figure 3 Alum-clad wood patio door—comparison of surface temperatures.

Figure 4 Wood double-hung window—comparison of surface temperatures.

tive model (labeled as “95% Ar” in Figure 4) shows good agreement to test on the frame and sash components. The agreement is poor at the center-of-glass; however, this anomaly can be easily explained. This product was simulated with 95% argon-gas fill and a .09 low-emissivity coating on surface 3, as per the manufacturer’s specifications. The test U-factor was suspiciously high, however, and was more representative of a window with no gas fill. It was hypothesized that the IGU may have lost its seal or that the unit was never gas filled, but there was no way of checking short of destructive testing (this was not an attractive option, however, as the unit was to be used for further validation testing).

When the window was simulated with no argon-gas fill, the total-product U-factor was very similar to the test result. Moreover, Figure 4 shows that the predicted surface temperatures (the line labeled “0% Ar”) show much better agreement with the test, especially at the upper-sash center-of-glass ($y \approx 1350$ mm), where the values differ by only 0.8°C . The difference between the convection model and the measured temperatures at the lower-sash center-of-glass ($y \approx 540$ mm) is probably due to differences in the inside film coefficient, as the product has an irregular geometry (the natural convection flow is interrupted by the meeting rail).

Figure 5 shows good agreement between finite-element analysis and the finite-volume convective model. Differences between simulation and test are larger on the frame, although this may be related to an error in the reported placement of the thermocouple on the sill cover. In an insulating (e.g., wood) frame, placement of the thermocouple can make a large difference in the measured temperature, as shown by the temperature gradient along the sill cover (a difference of roughly 19°C at $y \approx 55$ mm). It is also possible that “wind-washing” could be the cause of measured temperatures that are lower than expected, although that should not be a strong factor with this particular specimen, due to its weatherstripping and sash configuration.

Figure 5 Vinyl-clad casement window—comparison of surface temperatures.

Figure 6 Aluminum horizontal slider—comparison of surface temperatures.

Figure 6 shows the expected results for the frame and sash components: agreement between the convection model, finite-element analysis, and measurement everywhere except the head section, where the measured values are slightly warmer because of the undeveloped natural convection pattern in that region. Again, the convective model (“FRAME (conv)”) predicts temperatures that are colder than those predicted by the conduction-only model (“FRAME (old)”) at the bottom of the glazed area and warmer at the top, which is more in agreement with the finite-element model.

What is unusual in this case, however, is that the measured glass temperatures are much colder than the predicted results (by 3.6°C at center-glass and 4°C near the lower sightline, at $y \approx 90$ mm). There are three possible explanations for this difference: the center-glass temperatures may be depressed by the (extremely low) frame temperatures ($T_{\text{surface}} = -9^\circ\text{C}$ at the lower sightline) due to conduction in the glass; the characterization of radiative film coefficients in the simulation method does not account for radiation from the glass to the (extremely low-temperature) frame; or, pos-

sibly, the test method of determining average film coefficients does not account for the very large differences between local and average convective film coefficients for metal-frame windows, in which case standard temperature averaging may not be appropriate.

CONCLUSIONS

The convective model provides a great improvement in the ability of simulation to predict surface temperatures. Of the windows examined, simulation and test agree to within 2°C for nonmetal frames and to within 5°C for metal frames.

The differences are even smaller on the glass 50 mm above the lower sightline. There, we see a difference of less than 1°C for the nonmetal frames and 4.5°C for the metal-frame window. Given that the differences between test methods can be at least 1°C (depending on the type of tape used to mount the thermocouples), and that measured differences between testing agencies can be as much as 5°C (Elmahdy 1990), simulation appears to give reasonably accurate results.

Differences between test and simulation seem to be related to the characterization of the interior film coefficient, especially for metal-frame windows. One must be able to determine the variability of local film coefficients with respect to vertical position (as shown by the difference between test and simulation on the head sections of the specimens examined in this study), local geometry (corners and meeting rails), and radiative effects (local view factors between surfaces). There are other issues yet to be investigated ("wind-washing," contraction and warpage during testing, etc.), but these are beyond the scope of the current study.

Simulation can be used to double-check the validity of the test results (to determine, for example, whether a test specimen has lost its inert-gas fill).

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A TWO-DIMENSIONAL NUMERICAL MODEL FOR GLAZING SYSTEM THERMAL ANALYSIS

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ABSTRACT

A simple two-dimensional (2-D) numerical control volume formulation is presented that can be used to model heat transfer through a vertical insulated glazing unit. This model accounts for natural convection of the fill gas (including the effect of secondary cells), conduction within the solid materials, and radiant exchange between the various surfaces facing the fill-gas cavity. This model closely reproduces average and local heat transfer rates measured using a guarded heater plate apparatus. Simulations clearly show that fill-gas motion causes the minimum indoor surface temperature (during cold weather) to be located at the bottom edge of the indoor glazing. Calculated results were also used to gain insights into heat transfer patterns in glazing systems with various combinations of low-emissivity coatings, fill gases, and edge-seal designs.

INTRODUCTION

Windows play a major role in governing the energy consumption of both residential and commercial buildings. They account for a significant portion of the envelope heat transfer and virtually all of the solar gain. As a result, a significant effort has been made to design energy-efficient windows, and the advances made through the implementation of low-emissivity (low-e) coatings and substitute fill gases have transformed the industry.

It is common for sealed, insulated glazing units (IGUs) to exhibit condensation problems when operated in cold climates. Damaging accumulations of water, frost, and ice can form along the perimeter of the exposed surface of the indoor glazing because of the thermal short circuit caused by the edge seal. Condensation most readily forms along the bottom edge of the indoor glazing because this is where the conductive effect of the edge seal is augmented by local cooling caused by fill-gas motion.

The research described here consists of a combination of computer modeling and experimentation. Measurements of heat transfer rates through a variety of glazing systems showed that consistent differences could be found between the heat fluxes occurring near the top and bottom edges of glazing systems. These differences can be explained qualita-

tively by considering the motion of the fill gas but could only be quantified experimentally. Therefore, the goal of this undertaking was to formulate and test a computer model that accounts for the conduction, radiant exchange, and fill-gas motion within glazing systems. The comparison of calculated and measured results serves not only to validate the model but also demonstrates clearly that fill-gas convection in an IGU causes the minimum indoor glazing temperature to occur at the bottom edge of the glazing. Additional modeling can be used to predict condensation resistance values for windows that are in the design process or undergoing scrutiny for certification.

BACKGROUND

Heat transfer through many glazing systems has been measured using a guarded heater plate apparatus (Wright and Sullivan 1987a, 1987b, 1988). This apparatus consists primarily of two parallel copper plates (each 635 mm by 635 mm by 12.7 mm [25 in. by 25 in. by 0.5 in.]), positioned facing each other, that can be maintained at different but constant temperatures. The warm copper plate contains three guarded heater plates (each 200 mm by 200 mm [7.875 in. by 7.875 in.]) that lie in recesses located in a row across the center of the warm copper plate. They are embedded in such a way that the flat face of each heater plate is flush with the surface of the warm copper. Each heater plate is made of copper. Thus, the warm plate appears to be flat and continuous. It is possible to measure heat transfer that occurs over the face of each guarded heater plate.

IGUs tested in the guarded heater plate apparatus are built to the same face dimensions as the copper plates. Each unit is clamped between the copper plates. In order to promote good thermal contact, sheets of neoprene (rubber) are placed between the IGU and the copper plates. Measurements are usually made with the plates held vertically so that the three measurements of heat flux correspond to areas near the top, middle, and bottom of the IGU.

It was common for tests to be carried out with an interest only in the heat transfer rate measured at the middle heater plate because this provides a good measure of the center-glass U-factor. However, two interesting effects can be seen when the heat transfer rates measured at all three heater

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plates are examined. First, the heat flux at the top and bottom heater plates was consistently greater than the heat transfer at the middle heater plate. This happens because the low thermal resistance of the edge seals allows more heat transfer at the perimeter of the IGU than in the center-glass area. Second, the heat transfer measured at the bottom heater plate was always greater than the heat transfer at the top heater plate. Gas in the interpane cavity flows upward near the warm glazing and downward near the cold glazing. The descending gas becomes progressively colder. At the bottom of the cavity this cold fill gas turns and comes in direct contact with the bottom of the warm glazing as it starts its ascent. Thus, natural convection of the fill gas increases the heat transfer from the bottom heater plate but not from the top heater plate. It is this medium/low/high progression of heat transfer at the top/middle/bottom heater plates that was expected in results obtained from the numerical simulation of an IGU installed in the guarded heater plate apparatus.

THE COMPUTATIONAL MODEL

In order to enable a direct comparison between calculated and measured average and local heat transfer rates, a 2-D model was configured to simulate a glazing system under the conditions of a guarded heater plate test. Details of the problem domain and pertinent nomenclature are shown in Figure 1. The glazings are separated from the isothermal boundaries (i.e., the copper plates) by two sheets of neoprene rubber and the top and bottom ends of the problem domain are assumed to be adiabatic.

The glazing system model was developed by extending an existing and proven model (Wright 1990; Wright and Sullivan 1994) for natural convection of gas in a tall, vertical, rectangular cavity. The natural convection model was extended to solve for the temperature and heat flux distributions in double-pane glazing system assemblies. Modifications required to model this composite problem domain involved the additional consideration of conductive heat transfer in the solid sections (i.e., glazings and edge seals) and the exchange of energy by thermal radiation between the surfaces exposed to the interpane cavity. The following subsections describe the convection, conduction, and radiant exchange models. The description of the natural convection model is brief because detailed documentation can be found in the literature (Wright and Sullivan 1994).

Fill-Gas Convection

The natural convection model consists of a numerical (control volume) solution of the differential equations that describe the conservation of energy, horizontal and vertical momentum, and mass. The numerical algorithm used to solve these four equations, called SIMPLEC (Vandormaal and Raithby 1984), is an extension of the semi-implicit method for pressure-linked equations (SIMPLE), described

by Patankar (1980). SIMPLEC consists of an iterative "coefficient update loop" within which each of the four required conservation equations is solved in sequence. The only extension to the SIMPLEC procedure was the additive correction method (ACM) of Hutchinson and Raithby (1986), which was used to speed convergence.

The velocity and heat transfer solutions for fill-gas flow in a vertical window cavity are determined largely by the Rayleigh number (Ra). In the case of a fixed-pane spacing and a specified fill-gas composition, Ra is proportional to the temperature difference between the cavity walls. Ra typically does not exceed 12,000 under severe winter conditions in windows designed with optimum pane spacing. Various research studies have shown that in addition to the primary or base flow of fill gas that exists at low Ra , a stationary "cat's eye" pattern of secondary cells can be expected in the core of the flow when Ra exceeds a critical value (Wright and Sullivan 1989a). Bergholtz (1978) provided a means by which this critical Rayleigh number could be easily calculated and showed that cells can be expected for $Ra > 5,600$ in tall window cavities. The natural convection model used in this study incorporates a unique flow perturbation scheme that induces secondary cells when Ra is sufficiently high (Wright 1990; Wright and Sullivan 1994).

Solid Sections

The solid sections included in the problem domain consist of two sheets of neoprene, two sheets of glass, and two edge seals. Each of these three solids was treated as a homogeneous material.

The control volume arrangement was laid out to incorporate the glazing system materials and neoprene mats. The x-direction index, i , begins and ends at $i = ib$ and $i = ie$, respectively. The ib and ie control volumes correspond to the layers of neoprene. The glazings are located at $i = ib + 1$ and $i = ie - 1$. The cavity includes control volumes from $i = ibc = ib + 2$ to $i = iec = ie - 2$. The edge seals are uniformly split into the same number of horizontal divisions as the interpane cavity (i.e., 25 columns of control volumes). The grid aspect ratio in the cavity was set at $A_{grid} = 5$ and the same vertical subdivision used in the cavity was used for the glazing and neoprene materials adjacent to the cavity. The edge seals were divided into n_{seal} vertical divisions such that the size of the seal control volumes was about the same as that of the cavity control volumes (usually $n_{seal} = 5$). The same vertical divisions were used to set up control volumes in the glazings and neoprene adjacent to the edge seals. The y-direction index, j , runs from $j = jb$ to $j = je$ over the entire problem domain and from $je = jbc$ to $j = jec$ over the height of the cavity.

Some of the glazing systems tested contained edge seals built up from more than one component (i.e., spacer, sealants, etc.). The simulation model treated the edge seal as a single, homogeneous material. In order to be realistic, the thermal conductivity assigned to the fictitious seal material,

Figure 1 Two-dimensional glazing system thermal analysis geometry.

k_{seal} was based on thermal resistance measurements made using edge-seal samples tested in the guarded heater plate apparatus (Wright and Sullivan 1989b; Wright 1989, 1990).

The solution procedure used to model the composite problem was not complicated. One unique detail concerned the way in which the boundary conditions were applied. The energy balance was applied over the entire problem domain,

with the thermal boundary conditions (see Figure 1) applied at the outer edges. This step in the coefficient update loop generated a temperature solution for all control volumes. However, the velocity boundary conditions, used in solving balances to conserve momentum and mass, were applied at the walls of the cavity. Thus, the natural convection velocity field in the cavity was generated for nonuniform temperature distributions at the cavity walls and all velocities at locations outside the cavity were set to zero.

Application of the energy balance across the composite problem domain required care to ensure that the heat flux was correctly calculated at surfaces between materials of different thermal conductivity. Fortunately, such surfaces only exist where the velocity is zero; in these locations, book-keeping is only required to keep track of heat transfer by conduction. The conductive heat flux at such control volume faces was calculated according to the procedure outlined by Patankar (1980). For example, at a surface on the west wall of the interpane cavity (as shown in Figure 2), the surface temperature, T_{surf} is given by

$$T_{surf} = \frac{T_{ibc-1,j} \frac{k_g}{t_g} + T_{ibc,j} \frac{k}{\Delta x}}{\frac{k_g}{t_g} + \frac{k}{\Delta x}} \quad (1)$$

where t_g and Δx are the thicknesses of the glass and the fill-gas control volume adjacent to the glass surface, respectively, and k_g and k are the conductivities of the glass and fill gas, respectively. The corresponding surface heat flux, q_{surf} is given by

$$q_{surf} = 2 \frac{k_g}{t_g} (T_{ibc-1,j} - T_{surf}) = 2 \frac{k}{\Delta x} (T_{surf} - T_{ibc,j}) \quad (2)$$

THERMAL RADIATION MODEL

Figure 1 shows the fill-gas cavity. Each of the four cavity walls can be divided into finite surfaces—each of which emits, reflects, and absorbs thermal radiation. The incident radiative flux (or irradiance) at the p th surface is denoted H_p and can be expressed as

$$H_p = \frac{1}{A_p} \sum_{n \neq p} A_{nb} E_{nb-p} B_{nb} \quad (3)$$

where the summation is carried out for all of the "visible neighbor" wall surfaces (i.e., for all surfaces from which radiation can reach surface p). A_{nb} is the surface area of a

Figure 2 Surface heat flux and temperature at the cavity wall.

neighbor surface, E_{nb-p} is the geometric exchange factor from a neighbor surface to surface p , and B_{nb} is the radiative flux leaving (i.e., the radiosity at) the neighbor surface. Using the reciprocity relation,

$$A_{nb}E_{nb-p} = A_pE_{p-nb}, \quad (4)$$

Equation 3 can be simplified:

$$H_p = \sum_{nb} E_{p-nb} B_{nb}. \quad (5)$$

If all surfaces are treated as diffusely emitting and reflecting, then the exchange factors, E_{p-nb} , are equal to the more familiar geometric radiative shape factors, F_{p-nb} . However, it was thought possible that an appreciable portion of the radiation reflected from a glass surface would be reflected in a specular manner—especially if a low-e coating were present. It has been shown that subdivision of the hemispheric reflectance into specular and diffuse components,

coupled with the assumption of diffuse emission, lends itself to analytical treatment (Sparrow and Cess 1976). In accordance with this method, the reflectances of the vertical walls were split into specular and diffuse components as shown in Equation 6:

$$\rho = 1 - \epsilon = \rho^s + \rho^d. \quad (6)$$

Using this formulation for reflectance, B_{nb} can be expressed as the sum of emitted flux (surface temperature = $T_{surf,nb}$) and the diffusely reflected portion of the irradiance:

$$B_{nb} = \epsilon_{nb} \sigma T_{surf,nb}^4 + \rho_{nb}^d H_{nb}. \quad (7)$$

By combining Equations 5 and 7 and rearranging, one obtains the following:

$$H_p - \sum_{nb} E_{p-nb} \rho_{nb}^d H_{nb} = \sum_{nb} E_{p-nb} \epsilon_{nb} \sigma T_{surf,nb}^4. \quad (8)$$

Equation 8 can be expressed, for all surfaces of the cavity, in matrix format:

$$[A][H] = [T] \quad (9)$$

where the transpose of $[H]$ is given by

$$[H]^T = (H_1, H_2, H_3, \dots) \quad (10)$$

and the entries of the p th row of the coefficient matrix, $[A]$, are given by the left-hand side of Equation 8. The p th entry of the column vector, $[T]$, is given by the right-hand side of Equation 8. The solution to Equation 10 can be expressed in terms of the inverse of $[A]$:

$$[H] = [A]^{-1} [T]. \quad (11)$$

Notice that $[A]$ is a function of optical properties and geometry but does not depend on surface temperatures. It needs to be solved only once for a given cavity and selected set of surfaces. (In fact, the computer code does not invert the $[A]$ matrix, but reaches the same end by calculating its LU decomposition. Details can be found in chapter 2 of Press et al. [1986].) The $[T]$ vector is a function of the surface temperatures. Therefore, for a given distribution of surface temperatures, $[T]$ can be generated and the solution for $[H]$ using Equation 11 requires little computer time.

The exchange factors, E_{p-nb} , also need only be calculated once for a given geometry. Significant simplification in the calculation of E_{p-nb} follows from the fact that ρ^s is constant over all of the individual surface subdivisions that comprise each of the walls and from the assumption that ρ^s can be set to zero for the horizontal walls. Consider the calculation required if surface p is on the west wall of the cavity and the neighbor surface being considered is on the opposite wall. Figure 3 shows this geometry. It can be seen that E_{p-nb} is

Figure 3 Example of geometry for calculation of E_{p-nb} radiative exchange factor.

$$E_{p-nb} = F_{p-nb} + \sum_{i=1}^{\infty} (\rho_w^s)^i (\rho_E^s)^i F(2i)_{p-nb} \quad (12)$$

where ρ_w^s and ρ_E^s are the specular reflectance values for the west and east vertical walls, respectively. $F(2i)_{p-nb}$ is the geometric shape factor from surface p to the mirror image of surface nb after $2i$ reflections in the vertical walls. Each F_{p-nb} was computed using Hottel's crossed-string rule (Siegel and Howell 1972), and Equation 12 (or one of several similar equations) was applied by including the first 100 terms in the summation. Knowing that all radiation leaving any surface must eventually be absorbed or diffusely reflected at some surface, the accuracy of the calculated exchange factors was verified using the following check-sum:

$$\sum_{nb} E_{p-nb} (1 - \rho_{nb}^s) = 1 \quad (13)$$

for any surface, p .

If the fill-gas cavity is bounded by n gray surfaces on each vertical wall and m surfaces on each horizontal wall and their temperatures are taken as being fixed, then the applica-

tion of Equation 8 at each surface requires the solution of $2(n+m)$ simultaneous equations—yielding $2(n+m)$ values of H_p . This procedure also entails the calculation and storage of $2(n+m)^2$ values of E_{p-nb} . It was recognized that the computing requirements would be unnecessarily excessive if the surfaces specified for the radiation solution directly matched the control volume grid used for the convection and conduction portions of the analysis (i.e., $n \approx 250$, $m \approx 25$).

Fortunately, there is a large section away from the ends of the cavity (the center-glass section) where the temperatures of the vertical walls were expected to be isothermal. It was assumed that the center-glass portion of each glazing could be treated as a single surface for the purpose of applying Equation 8. These large center-glass surfaces were delimited according to the ASHRAE definition of center-glass area—exposed faces of finite control volumes starting more than 65 mm from the sight line were included in the center-glass surface. In the edge-glass portions of the cavity, the individual exposed faces of the control volumes were used in the analysis of radiation exchange. This approach reduced the number of H_p variables by a factor of about 4 and reduced the required storage for E_{p-nb} by a factor of more than 10.

Once the irradiance at each enclosure surface was known, the net flux of radiant energy into the surface, S_p , could be found:

$$S_p = \epsilon_p (H_p - \sigma T_{surf,p}^4) \quad (14)$$

S_p was included as a fixed energy source in the energy balance of the wall control volumes. The rate of energy generation per unit volume, S , at any wall control volume is

$$S = \frac{S_p}{\Delta n} \quad (15)$$

where Δn is the dimension of the control volume normal to the wall. The validity of the source terms was examined during each simulation by summing energy generation at the cavity wall control volumes. This total should be zero. S_p and S were updated at each iteration of the coefficient loop after the generation of a new temperature field.

SOLUTION CONVERGENCE

The 2-D glazing system simulation software was coded in a Fortran program called IGU-2D (Wright 1990). The SIMPEC algorithm used to generate a numerical solution is iterative in nature and the criterion for solution convergence in this case was based on the calculation of total heat transfer through the glazing system, Q . The IGU was subdivided into 29 columns of control volumes. Therefore, it was possible to calculate Q at 30 vertical cuts in the problem domain. Once the converged solution has been reached, these estimates of Q should be equal because adiabatic surfaces were pre-

scribed at the top and bottom boundaries. Q was calculated at the right-hand edge of the i th column of control volumes using

$$Q = Q_{rad} + \sum_{j=jb}^{je} q_{i,j} \Delta y_j \quad (16)$$

where

Q_{rad} = net amount of thermal radiation energy crossing the vertical cut from right to left (it can be determined by summing the rate of energy generation due to source terms at wall control volumes to the left of the vertical cut) and

$q_{i,j}$ = nonradiative, right-to-left heat flux through the right-hand face of the (i,j) th control volume. This heat flux, for vertical control volume faces within the cavity, is given by

$$q_{i,j} = \beta \frac{(T_{i+1,j} - T_{i,j})}{\Delta x} k - \rho C_p u_{i,j} \left\{ \left(\frac{1}{2} + \alpha \right) T_{i,j} + \left(\frac{1}{2} - \alpha \right) T_{i+1,j} \right\} \quad (17)$$

Note that α and β are upwinding coefficients calculated as a function of local fluid velocities for the purpose of evaluating the temperature and temperature gradient of the fill gas at each of the control volume faces. A good discussion of upwinding coefficients can be found in Patankar (1980) and details of how they were used in this particular study can be found in Wright (1990) and Wright and Sullivan (1994). In the case of control volume faces along the cavity walls, $q_{i,j}$ was calculated according to Equation 2.

During each execution of IGU-2D, the rate of heat transfer was calculated at the 30 vertical cuts and the maximum and minimum of these values, Q_{max} and Q_{min} , respectively, were noted. Iteration of the coefficient update loop was terminated when Q_{max} and Q_{min} differed by less than 0.1%.

PRELIMINARY RESULTS

Four preliminary simulation runs were carried out in order to examine the sensitivity of the solution to the specular/diffuse split chosen for the longwave reflectivities of the vertical cavity surfaces. Two glazing systems were modeled: a conventional double-glazed unit and a similar unit with a soft, low-e coating. Both glazing units had corrugated-strip edge seals and the fill gas was air. Two simulations were run for each glazing system. Highly specular reflectance properties were used in one simulation and purely diffuse reflectances were used in the other. In the specular runs, the glass surfaces were modeled with $\rho^s = 0.15$ and $\rho^d = 0.01$; the low-e surfaces were modeled with $\rho^s = 0.90$ and $\rho^d = 0.01$. In the diffuse runs, the glass surfaces were modeled with $\rho^s = 0.0$ and $\rho^d = 0.16$; the low-e surfaces were modeled with $\rho^s = 0.0$ and $\rho^d = 0.91$.

The output from each pair of simulations showed very little or no difference in all reported quantities. For example, the local heat fluxes calculated for the areas corresponding to the locations of the guarded heater plates differed at most in the fourth significant digit. Plotted heat flux and temperature profiles appeared to be identical. It was concluded that the diffuse/specular longwave reflection split at the glazing cavity surfaces has no significant bearing on heat transfer in glazing units. All surfaces were subsequently modeled as purely diffuse reflectors.

COMPARISON WITH MEASURED HEAT FLUX

Seven glazing systems were tested using the guarded heater plate apparatus and modeled using IGU-2D. All except one had low-e coatings. Two units had argon fill gas. The set of edge seals incorporated in the test units was purposely chosen to span a wide range of thermal resistance. They included the highly conductive conventional (aluminum spacer bar) edge seals, corrugated-strip edge seals¹ with intermediate thermal resistance, and spacers made from strips of rigid insulating foam. The various data needed to describe the glazing systems are listed in Table 1.

In order to simulate glazing systems installed in the guarded heater plate apparatus, the top and bottom edges of each glazing system were treated as adiabatic surfaces. When measurements were undertaken, these same surfaces were well insulated and it was assumed that any heat flux through the ends of the glazing system would be very small in relation to the heat flux between the glazings through the edge seals. Thus, measurement errors due to the nonadiabatic end condition were assumed to be negligible. However, this may not have been the case for measurements taken when highly insulating foam spacers were used. The same foam that was used for edge spacers was also used to insulate the edge of the test section. Therefore, even though the possibility of a calculation/measurement mismatch was foreseen, it was decided that the foam spacers should be simulated because of the insight to be gained by examining the heat flux patterns in systems with highly insulated edge seals and because of the chance that the nonadiabatic experimental boundary condition was unimportant or that its impact might be quantified.

Figure 4 shows a sample solution (glazing unit 3) of heat flux at the right-hand edge of the problem domain (i.e., at the vertical surface where $T = T_h$). This surface corresponds to the face of the hot copper plate in the guarded heater plate apparatus. The heat flux from the hot copper plate is shown as a function of distance from the bottom of the glazing unit. Glazing unit 3 was built with conventional edge seals that contain aluminum spacer bars. Figure 4 shows that the low thermal resistance of this edge seal results

¹This is a commercially available edge seal consisting of a corrugated aluminum spacer strip embedded in a strip of sealant.

TABLE 1
Calculated and Measured Glazing System Heat Flux Results

	ΔT_{pp} [°C]	l_z [mm]	w_{seal} [mm]	k_{seal} [W/ m·K]	t_n [mm]	t_g [mm]	ϵ_{lc}	ϵ_{seal}	q_{meas} [W/m ²] Top/Middle/Bottom	q_{calc} [W/m ²]
1)	18.88	12.13	9.53	0.52	3.175	3.94		0.827	96.9 95.3 105.7	90.1 89.4 96.7
	fill gas air, no coatings									
2)	19.39	12.41	9.53	0.53	3.175	2.84	0.096	0.827	49.0 47.0 58.9	45.1 44.1 55.0
	fill gas air									
3)	19.12	14.12	11.05	2.56	1.524	3.3	0.047	0.054	32.1 28.7 45.7	33.5 28.0 45.6
	fill gas argon									
4)	11.38	11.96	11.3	2.12	3.175	3.91	0.094	0.054	24.4 19.7 27.67	24.5 19.9 27.1
	fill gas argon									
5)	27.42	16.0	25.4	0.035	1.524	5.59	0.07	0.579	56.5 62.7 89.0	44.9 56.1 87.2
	fill gas air									
6)	27.34	13.0	25.4	0.035	1.524	5.59	0.07	0.579	60.8 61.2 83.7	50.2 59.6 82.6
	fill gas air									
7)	14.86	13.0	25.4	0.035	1.524	5.59	0.07	0.579	32.3 30.6 38.7	26.6 30.2 38.7
	fill gas air									

Notes: 1) Calibration of neoprene mats: $k_n=0.17$ W/m·K for 1.524 mm mats and $k_n=0.19$ W/m·K for 3.175 mm mats.

2) k_{seal} was based on seal dimensions and measured linear conductance. k_{lin} , values found in Wright and Sullivan (1989b). $k_{lin}=0.41$ W/m·K (corrugated strip seals) for units 1 and 2; $k_{lin}=2.0$ W/m·K (conventional seals) for units 3 and 4. Units 5, 6 and 7 had rigid insulating foam spacers without sealant and $k_{seal}=0.035$ W/m·K was taken directly from the manufacturer's literature.

3) Emissivities of seals, ϵ_{seal} , and low-e coatings, ϵ_{lc} , were measured using a Gier-Dunkel DB-100 Infrared Reflectometer.

in a very high heat flux (in excess of 600 W/m²) near the top and bottom edges of the glazing unit. Heat flux through the center-glass area is constant except for the small variation caused by the secondary cells. The asymmetry between the top and bottom edge-glass areas is a result of the fill-gas motion.

Figure 4 also shows three average values of the calculated heat flux corresponding to the locations of the guarded heater plates where heat transfer was experimentally metered. These results can be directly compared to measured heat flux results. The medium/low/high progression of heat flux found at the top/middle/bottom heater plates is typical of measured values taken for many glazing units tested using the guarded heater plate apparatus. The middle heater plate area provides a representative measure of center-glass heat

flux. The heat flux from the top and bottom heater plates is augmented by the thermal short circuit at the edge seal, and the motion of the fill gas further increases the heat flux from the bottom heater plate.

Two sets of heater-plate-area heat flux results are included in Table 1 for each glazing system—one set calculated and one set measured. Each set consists of heat flux data for the top, middle, and bottom heater plates. The same results are presented in bar-chart format in Figure 5.

Units 1 and 2

The first two glazing units for which results are shown in Figure 5 were built using corrugated-strip edge seals. The fill gas was air. Unit 1 had no coatings but unit 2 had a soft,

Figure 4 Calculated heat flux distribution at hot boundary and calculated average heat flux for heater plate areas (unit 3).

	CDG	DGLE	DGLE ² A	DGLEA	
Seals	cor'd	cor'd	conv1	conv1	
ϵ_z	12.1	12.4	14.1	12.0	[mm]
ΔT_{pp}	18.9	19.4	19.1	11.4	[K]
# cells	0	0	16	0	
Ra	3,355	4,145	7,490	2,623	
A	50.8	49.6	43.4	51.2

Figure 5a Calculated and measured heat flux results for top, middle, and bottom guarded heater plate areas (units 1, 2, 3, and 4).

low-e coating. Secondary cells were not expected and were not modeled for either of these glazing units. These units were tested at about the same temperature difference but unit 2 allowed about half as much heat transfer as unit 1 because of the low-e coating. The difference between heat flux at the center and bottom heater plates was greater for unit 2 because the low-e coating suppresses center-glass heat transfer but does not alter the thermal resistance of the edge seals.

The calculated heat flux results consistently underpredicted the measured data from units 1 and 2 by about 7%. This level of discrepancy is not unreasonable but the fact that it exists for both units and is independent of heater plate location suggests that improvement might be possible if the reason(s) for the difference were known. In addition to showing good agreement with the measured heat flux data, the trend in the calculated results closely follows the top/middle/bottom heater plate trend that was found in the measurements.

Units 3 and 4

Glazing units 3 and 4 were both built with conventional, aluminum spacer bar edge seals and both contained argon fill-gas. Unit 4 had a conventional soft, low-e coating. Unit 3 had a more recently developed soft, low-e coating with extremely low emissivity. Secondary cells were modeled in unit 3 but not in unit 4, which was tested at a lower temperature difference.

	DGLE	DGLE	DGLE	
Seals		Insulating Foam		
ϵ_z	16.0	13.0	13.0	[mm]
ΔT_{pp}	27.4	27.3	14.9	[K]
# cells	10	17	0	
Ra	15,955	8,437	4,638	
A	36.5	44.9	44.9

Figure 5b Calculated and measured heat flux results for top, middle, and bottom guarded heater plate areas (units 5, 6, and 7).

The agreement between calculated and measured heat flux in these glazing units was excellent. The heater plate heat flux was predicted to within 2% in five of the six cases. Arrows marked in Figure 5 indicate the difference in calculated heat flux resulting from the presence of secondary cells. The results for unit 3 show that Ra was low enough so that the secondary cells had little impact on the heat flux. The presence of cells augmented the heat flux at the top and middle heater plates by about 4% but caused little change (a very slight decrease) at the bottom heater plate.

Again, the customary top/middle/bottom trend in heat flux was observed because of the highly conductive edge seals. Note that the bottom heater plate heat flux was about 50% higher than the heat flux at the middle heater plate for units 3 and 4. The corresponding difference for units 1 and 2 was only about 25% because they were built with the less conductive corrugated-strip edge seals.

Unit 3 was tested with about the same temperature difference used to test units 1 and 2. By examining the heat flux at the center heater plate, it can be seen that the thermal resistance at the center of the test section was approximately doubled by the addition of the low-e coating. The subsequent use of argon fill-gas almost doubled the thermal resistance again.

Units 5, 6, and 7

Test units 5, 6, and 7 each contained a soft, low-e coating and had air as the fill gas. These three units did not incorporate commercial edge seals. Instead, the glazings were separated at their edges by strips of rigid, insulating foam. No sealant was used. Thus, the edge seal consisted of a single material with an extremely low thermal conductivity.

Units 5, 6, and 7 do not exhibit the top/middle/bottom trend in heat flux usually found in glazing systems because the low-conductivity spacers do not drive the heat flux up drastically at the top and bottom. Consequently, the heat flux at the top heater plate is about the same as, or may even be less than, the heat flux at the middle heater plate. It is interesting that the creation of cells in units 5 and 6 decreased the heat flux at the bottom heater plate.

The calculated and measured heat fluxes for unit 5 do not agree as well as the results for any of the other units. The heat flux measured at the center heater plate was 23% higher than the heat flux calculated without secondary cells. This discrepancy was cut in half after cells were introduced. The reason for the disparity can be understood by noting that the relatively large pane spacing and high temperature difference resulted in Ra being well in excess of the range of Ra for which the convection model was intended ($Ra \approx 16,000$, based on center-glass temperature difference across the cavity). On the basis of results shown in Wright and Sullivan (1994), it can be seen that the numerical convection model would underpredict the heat transfer rate by almost 20% at

$Ra = 16,000$ if no cells were present and that only half of the difference would be made up if cells were induced.

The simulation results for units 6 and 7 closely match the measured heat fluxes for the bottom and middle heater plates. In these locations, discrepancies were 3% or less. In contrast, the heat flux at the top heater plate was underpredicted by 21% in both units 6 and 7. Notice that the top heat flux was also underpredicted by about 25% for unit 5. This difficulty in calculating heat flux at the top heater plate exists only for units 5, 6, and 7—the units with highly insulating edge spacers. The specific reason for this is not clear but it is likely that this problem is linked to the adiabatic boundary assumption. Fortunately, the model does not exhibit the same difficulty when more conventional edge seals are used or at the bottom of the glazing system, where the heat flux and temperature profiles are of greatest interest.

HEAT FLUX IN THE GLAZING CAVITY

Having completed the comparison of calculated and measured heat fluxes at the problem boundary it is instructive to study the simulation results for details about heat flux within the glazing unit itself. Test units 1, 2, 3, and 7 were chosen for examination because they include glazing systems with and without a low-e coating, with and without argon fill-gas, and all three variations in edge-seal conductance. Units 1, 2, and 3 were tested at approximately the same temperature difference, facilitating the comparison of heat flux results. The simulation results for units 1, 2, 3, and 7 were assumed to be accurate on the basis of the comparisons presented in the previous section.

Figures 6, 7, 8, and 9 show the local x-direction heat flux, $q(y)$, plotted as a function of distance from the bottom of the glazing system for a vertical surface located at the hot wall of the cavity (i.e., $x = t_n + t_g + l_x$). Figures 6 through 9 show results for test units 1, 2, 3, and 7, respectively. Each figure includes two curves, with the higher curve showing total heat flux and the lower curve showing the radiative component. The hash marks on the lower curve delimit the center-glass portion of the glazing system as specified in the analysis of radiative exchange, which in all cases corresponds to the center-glass area defined by ASHRAE. The sight lines are located where the radiative heat flux becomes zero and the total heat flux rises sharply.

Figure 6 shows heat flux results for a conventional double-glazed IGU with corrugated-strip edge seals (unit 1). The heat flux is constant over the center-glass area, and radiative exchange constitutes the majority of the center-glass heat transfer. In the bottom edge-glass area, the heat flux increases because of the fill-gas motion. Heat flux through the edge seals peaks at more than 400 W/m^2 and is significantly higher than the heat flux through any portion of the sight area.

Unit 2 was similar to unit 1 except that a low-e coating was included in unit 2. These two units can be compared by

Figure 6 Heat flux component distributions along vertical cut at hot cavity wall (unit 1).

Figure 7 Heat flux component distributions along vertical cut at hot cavity wall (unit 2).

Figure 8 Heat flux component distributions along vertical cut at hot cavity wall (unit 3).

Figure 9 Heat flux component distributions along vertical cut at hot cavity wall (unit 7).

examining Figures 6 and 7. The pattern of heat flux is similar for units 1 and 2 but the total center-glass and edge-glass heat flux has been substantially reduced because the low-e coating has virtually eliminated the radiative heat flux in unit 2. In contrast, the peak edge-seal heat flux is greater in unit 2 (in excess of 500 W/m^2) than it was in unit 1. It is interesting that the increased edge-seal heat flux results from the presence of the low-e coating. The increase in edge-glass and center-glass thermal resistance imposes a larger temperature difference across the edge seals.

Units 2 and 3 can be compared by examining the results in Figures 7 and 8. The differences between these units include a switch from corrugated-strip to conventional edge seals and the presence of argon fill-gas instead of air. Again, the low-e coating limits the radiative heat flux to a small portion of the total. The secondary cells in unit 3 create a ripple in the heat flux distribution. The use of argon in unit 3 has reduced the center-glass and edge-glass heat flux. The use of conventional edge seals and the relatively high thermal resistance of the sight area have caused the peak heat flux in the edge seals to be in excess of $1,000 \text{ W/m}^2$.

Figure 9 shows heat flux results for unit 7. Unit 7 was similar to unit 2 except for the use of insulating foam instead of the corrugated-strip edge seals. Unit 7 was tested at a lower temperature difference ($\Delta T_{pp} = 14.9^\circ\text{C}$ versus 19.4°C). The center-glass thermal resistance of unit 7 was approximately the same as that for unit 2. The effect of the insulating foam spacer used in unit 7 is apparent in that the peak heat flux did not exceed 90 W/m^2 , which is an order of magnitude less than the peak heat flux found in unit 3. Away from the cavity wall, the edge-seal heat flux drops to almost the level of the center-glass heat flux. The asymmetry of the heat flux distribution caused by the fill-gas motion is highly pronounced in the results shown for unit 7.

Several observations can be made regarding results that are common to Figures 6 through 9. In each case the radiant heat flux distribution did not show a discontinuity at the junction of the edge-glass and center-glass areas. This result supports the validity of lumping the center-glass control volume surfaces together for the purpose of the radiative exchange model. The simulations also seem to give credence to the ASHRAE definition of center-glass area in that heat flux results were very close to being constant within the center-glass limits, but this may be misleading because of the isothermal boundary conditions imposed by the guarded heater plate apparatus. In all cases, the heat flux profiles were asymmetric, with the natural convection of the fill gas causing the bottom edge-glass heat flux to be significantly greater than the center-glass or top edge-glass heat flux. This asymmetry is most evident in units with highly insulating edge seals. This augmentation of heat flux at the bottom edge of the sight area corresponds to the location of the minimum indoor glazing temperature where the onset of condensation occurs during cold weather.

TEMPERATURE PROFILES AT THE WARM GLAZING

Temperature profiles, $T(y)$ versus y , have been produced for a plane at the hot vertical wall of the glazing cavity. This surface is the same vertical cut at which heat flux profiles were given in Figures 6 through 9. Figures 10 through 13 show temperature profiles for units 1, 2, 3, and 7 that correspond to Figures 6 through 9, respectively.

Figures 10, 11, and 12 show that the center-glass temperature drew progressively closer to the hot plate temperature as the center-glass thermal resistance of the glazing system increased. The same effect is apparent in windows exposed to the environment. Glazing systems with higher thermal resistance exhibit higher indoor glazing temperatures in cold weather and are less susceptible to condensation.

The location of the minimum indoor pane temperature is at (or very near) the bottom sight line. The extreme edges of the indoor glazing may exist at a lower temperature, but the glass outside the sight line is generally not exposed to the indoor air after being installed in a sash. Condensation forms first at the bottom sight line. The sight line temperature predictions resulting from simulation of the guarded heater plate configuration offer no quantitative information regarding windows exposed to more realistic conditions, but qualitative observations can be made.

Examining Figures 9 through 13, the minimum temperature at the bottom sight line was consistently lower than the temperature at the top sight line. This difference would not have been evident if the motion of the fill gas had not been modeled. The difference between the top and bottom sight line temperatures was less than 1°C in all four cases. This difference would be greater if the glazing system were exposed to the indoor environment rather than being held close to an isothermal plate.

In the three units constructed with commercial edge seals (see Figures 10, 11, and 12) the temperature of the warm glazing continued to drop along the edge seal below the bottom sight line. This is consistent with the heat flux profiles shown in Figures 6, 7, and 8, where the heat flux also was found to decrease across the face of the edge seal. Only in the glazing system with insulating foam spacers (shown in Figure 13) was an increase in temperature found below the bottom sight line.

CONCLUSIONS

A model has been formulated for the numerical simulation of heat transfer in glazing systems. This model includes the 2-D treatment of glazing and edge-seal conduction, the exchange of thermal radiation between the glazings, and natural convection of the fill gas (including secondary cells if needed). Simulation results pertain to both the center-glass and edge-glass areas of the glazing system.

Figure 10 Temperature distribution along vertical cut at hot cavity wall (unit 1).

Figure 11 Temperature distribution along vertical cut at hot cavity wall (unit 2).

Figure 12 Temperature distribution along vertical cut at hot cavity wall (unit 3).

Figure 13 Temperature distribution along vertical cut at hot cavity wall (unit 7).

Guarded heater plate measurements have been found to agree well with calculated heat flux values for a number of glazing systems with a wide variety of glazing system and edge-seal components. Simulation results have provided insights regarding the heat flux patterns and temperature profiles inside these glazing systems. The importance of fill-gas motion in determining the minimum indoor pane temperature has been demonstrated.

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A Simplified Method for the Numerical Condensation Resistance Analysis of Windows

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Introduction

Most modern windows contain a sealed, insulated glazing unit (IGU). The IGU consists of two panes of glass that are separated by an edge-seal. The edge-seal isolates the space between the glazings - thereby reducing the number of surfaces to be cleaned and creating an insulating cavity suitable for low-emissivity (low-e) coatings and/or low conductivity fill gases.

The edge-seal creates a thermal bridge at the perimeter of the IGU making this an area of increased thermal stress, high energy loss and the site of condensation in winter. The conductive nature of the edge-seal has been accentuated by the remarkable reductions in centre-glass U-factor made available over the last decade. Until recently few options were available to increase the thermal resistance of the edge-seal. A variety of edge-seals with significantly increased thermal resistance are now available but none offer more thermal resistance than the centre-glass portion of an IGU. The thermal resistance of the centre-glass cavity (low-e, 1/2 inch argon, $R=3$ hr-ft²F/BTU) can only be approached by the best rigid insulating foam ($R=2.5$ per half inch)^{1,2,3}.

The flow of fill gas within an IGU cavity contributes to the condensation problem at the bottom edge of the indoor glazing. In winter fill gas flows upward near the indoor glazing and downward near the outdoor glazing. The descending gas becomes progressively colder. At the bottom of the cavity this cold fill gas turns and approaches the indoor glazing where it starts its ascent. Thus, the glass near the bottom edge of the indoor glazing is cooled by the coldest fill gas.

Experimental results and detailed numerical simulation⁴ support the idea that fill gas motion contributes to the bottom-edge condensation problem. Clearly, any model attempting to quantify local heat transfer rates in these regions or any model

intended to determine the temperature distribution across the face of the glazing must account for both the natural convection of the fill gas and conduction within the edge-seal, glass and frame components.

None of the methods currently in widespread use for estimating edge-glass heat loss^{5,6,7} or the common numerical methods for analysing frame and edge-glass heat loss (e.g., reference 7) account for fill gas convection. Therefore, they cannot be used to determine the minimum indoor pane temperature needed to evaluate the condensation resistance of a window. Local variations in surface temperature can be found using very detailed numerical methods (e.g., references 4 and 8) but this approach is expensive in terms of operator time, CPU time and hardware required.

The method presented here provides a means by which a numerical, conduction-only, 2-D frame and edge-glass analysis can be extended to account for fill gas motion. This extension is not complicated and execution is no slower. Minimum indoor glazing temperature can be predicted with no additional effort and in a reasonable amount of time using a desktop computer.

The following sections provide details regarding this new procedure along with discussion regarding some of the simplifications that can safely be used in its application.

Numerical Analysis of Natural Convection

The control volume arrangement shown in Figure 1 is used to set up an analysis on a uniform, staggered grid. An energy balance applied to this control volume, in terms of the temperatures at the four neighbouring volumes and in the absence of energy sources (e.g., solar radiation), gives an expression for the temperature T_p .

$$a_p T_p = a_n T_n + a_e T_e + a_s T_s + a_w T_w \quad (1)$$

The coefficients (a_p , a_w , etc.) can be determined as a function of material properties, fluid velocities, control volume geometry and the assumed temperature profile between the nodes. Many sets of these coefficients must be generated - one set for each control volume in the problem domain. Almost all of the CPU time needed to arrive at a solution is consumed in the solution of the coefficient matrices. Very little time is used to generate the coefficients.

The form of Equation 1 shows that the temperature at any point is determined as a weighted average of the temperatures at surrounding locations. In order to examine

the effect of fluid motion consider the coefficient associated with the west side of the control volume, a_w .

$$a_w = k \frac{\Delta y}{\Delta x} + \frac{1}{2} \rho \cdot u_w \cdot \Delta y \cdot C_p \quad \text{(central difference)} \quad (2)$$

where k , ρ and C_p are the conductivity, density and specific heat of the fluid, respectively. Similar expressions are used for a_n , a_s and a_e . a_p is equal to the sum of the other four coefficients. See reference 9 for more detail.

Figure 1: Control Volume Arrangement Used for Energy Balance

Equation 2 provides some useful information. If a conduction problem were formulated (i.e., $u_w = u_e = v_n = v_s = 0$) using a square grid (i.e., $\Delta x = \Delta y$), T_p would be affected equally by all four neighbouring temperatures. In this case all four coefficients would be equal to $a_p/4$. However, if u_w , u_e , v_n and v_s are positive as shown in Figure 1 then a_w and a_s will be larger and a_e and a_n will be smaller. This makes sense because T_p will be affected strongly by the upstream temperatures, T_w and T_s , and only weakly by the temperatures at downstream locations.

The analysis of an entire window entails a variety of complications. In the glazing cavity, where convection takes place, four equations similar to Equation 1, corresponding to energy, x-momentum, y-momentum and mass balances, must be solved simultaneously for the entire problem domain. In the solid sections only conduction occurs and only the energy balance needs to be applied. The solution

yields temperatures, two sets of velocity components and the pressure field. Special attention must be taken to correctly account for changes in property values at surfaces where materials differ and at gas/solid interfaces. It is also necessary to include a model for radiant heat transfer in the window cavities.

A Simplified Approach

Recognizing that the detailed analysis of a full window is a major undertaking it is not surprising that conduction-only programs such as FRAME¹ have become very popular for the analysis of frame and edge-glass heat transfer. FRAME isolates the frame and edge-glass by assuming that the frame/wall and edge-glass/centre-glass (2.5 inches from the sight line) interfaces are adiabatic. FRAME also escapes the complexity of dealing with natural convection by treating the fill gas as an opaque solid that is assigned an "effective conductivity". This effective conductivity is quantified by a 1-D centre-glass heat transfer solution that can be provided by software such as VISION¹⁰ or WINDOW¹¹. It has been demonstrated that VISION or WINDOW, used in conjunction with FRAME, will give reliable estimates of overall window heat transfer.

A sample FRAME isotherm output plot is shown in Figure 2. It can be seen, by the near symmetry of the temperature distribution in the bottom of the glazing cavity, that the effect of fill gas motion has not been included in the analysis. In fact, FRAME cannot distinguish between the same cross-section used as a header or sill.

A conduction analysis, such as FRAME, can readily be extended to account for fill gas motion. It was shown in the previous section that a non-zero fluid velocity simply adds an extra term to the calculation of the various coefficients. The mechanics of solving for the temperature field remain unchanged, and no extra computational effort is required as long as the fill gas velocity field is already known.

A new capability has been included in the most recent version of the VISION software, VISION4, that allows it to estimate the fill gas velocity field. When a given glazing system is analysed the procedure starts in the conventional way by generating the 1-D centre-glass heat transfer solution. Next, the known pane spacing, fill gas properties and glazing surface temperatures are used to perform a 2-D numerical simulation of fill gas convection in the idealized case of a tall, vertical cavity with isothermal side walls and adiabatic ends¹. The wall temperatures of this idealized situation differ from the real situation only in the edge-glass section which constitutes a small portion of the cavity. It is assumed that the velocity field is

determined primarily by the balance of buoyant and viscous forces that exists through the centre-glass region and that the difference between this velocity field and the actual velocity field generated with the true wall temperatures can be neglected.

Once VISION4 has determined a solution for the entire cavity the velocity field for the edge-glass section of the cavity is stored on disk, along with the temperature profile across the edge-glass/centre-glass interface and the properties of the fill gas. The edge-glass/centre-glass-temperature profile is needed because it is not appropriate to

specify an adiabatic surface at this location when flow exists at the boundary. The temperature of the gas flowing into the edge-glass section must be known.

The entire process is repeated, once for each cavity in the glazing system, so that each disk file will contain data pertaining to one or more glazing cavities.

A new version of the FRAME program, FRAME 4.0, has been written and is capable of reading the VISION4 edge-glass velocity files. FRAME 4.0 can then use this information to generate a frame and edge-glass heat transfer solution that includes the effect of fill gas convection. Figure 3 shows an isotherm output plot produced by FRAME 4.0. The only difference between this plot and the plot shown in Figure 2 is the effect of fill gas convection. This is readily evident because the temperature field in the bottom of the glazing cavity has been skewed toward the warm indoor glazing.

Note that the temperature field of the fill gas, as seen in Figure 3 for example, as determined by FRAME 4.0 includes the effect of the more realistic, non-uniform wall (i.e., glazing and seal surface) temperatures. The temperature field generated by VISION4 is not stored and is not used in the FRAME 4.0 calculation. Therefore, small inaccuracies in the form of the velocity field generated by VISION4 can be expected to be of secondary importance.

Simplifications

The implementation of 2-D computational fluid dynamics (CFD) code in VISION 4 includes two simplifications.

Aspect Ratio: The coefficient for convective heat transfer across a vertical, gas filled, rectangular enclosure is known to be a function of the height-to-width aspect ratio, A , and the Rayleigh number, Ra ¹².

$$Ra = \frac{\rho^2 \ell^3 g C_p \Delta T}{\mu k T_m} \quad (3)$$

where, ℓ = cavity width (pane spacing)

ΔT = temperature difference between the vertical cavity walls

ρ , C_p , and μ = fill gas density, specific heat and viscosity

T_m = mean temperature (absolute) of the fill-gas

g = acceleration due to gravity

Accordingly, the velocity and temperature fields of the fill gas are also determined by Ra and A . A series of CFD runs was completed for the range of Ra of interest for IGU applications, $Ra \leq 12,000$, and for $10 \leq A \leq 100$. It was found that the flow/temperature fields were not affected by A for $A \geq 30$. Figure 4 illustrates this point. Figure 4 shows a plot of the velocity (vertical component, v) and temperature profiles at the centre-glass/edge-glass interface. The temperature data have been converted to a non-dimensional form using T_{ww} and T_{we} , the west and east wall temperatures. The velocity has been shown as a fraction of the maximum vertical velocity component, v_{max} . Both are shown as a function of x/l , the fraction of the distance across the cavity - from the west wall. Data are plotted at $Ra=10,000$ for two different aspect ratios: $A=30$ and $A=100$. The $A=30$ results are shown as data points which fall almost directly on the line that represents the $A=100$ data.

VISION4 takes advantage of this result to save computer memory and execution time. All VISION4 CFD runs use $A=30$ regardless of the specified cavity height.

Secondary Cells: It is known that the fill gas flow in tall cavities will include a cats-eye pattern of co-rotating secondary cells if the Rayleigh number is greater than about 5,700^{1,4,12,13,14}. Secondary cells are not generated by the VISION4 CFD procedure. There are two reasons for omitting the effect of secondary cells. First, it is possible but it is not practical to model window cavity flows with secondary cells on a desktop computer because the added computational effort increases the run time appreciably. Second, the presence of secondary cells increases the convective heat transfer rate across the centre-glass section of the cavity. In this area the cells create a homogenizing action that increases the temperature of the fill gas entering the bottom edge-glass section. The increased temperature of the fill gas in this location will increase the minimum indoor pane temperature. Therefore, by choosing to omit secondary cells, the resulting estimate of condensation resistance will correspond to a worst case situation.

Laminar Flow

The CFD model in VISION4 is based on the assumption that the fill gas flow is laminar and steady. This is true for over the full range of Ra of interest in IGU analysis: typically $Ra < 10,000$. At higher values of Ra the flow can become unsteady and eventually turbulent¹². This physical instability is also seen in the VISION4 CFD analysis making convergence become progressively slower for Ra from 10,000 to 20,000 and simulations with $Ra > 20,000$ are likely to result in floating point errors. This limitation does not detract from the value of the procedure because glazing systems that are associated with the potential for condensation problems will be double glazed sealed units with a near-optimum pane spacing corresponding to Rayleigh numbers well below 10,000 - regardless of fill gas type.

Conclusions

A method has been presented whereby a conduction-based 2-D frame and edge-glass numerical analysis can be extended to account for convective fill gas motion. The resulting glazing temperature profiles are useful for stress analysis and for evaluating condensation resistance. The method requires one initial CFD calculation to determine the fill gas flow pattern but adds very little complexity to the 2-D frame and edge-glass analysis. The frame and edge-glass calculation runs no more slowly

than it would have in dealing with a conduction-only problem and can still be run on a desktop computer.

Results from initial test runs using this method as implemented in VISION4 and FRAME4 are in good agreement with hot-box test results and more detailed simulations¹⁵.

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**THE HARMONIZATION OF CANADIAN AND U.S.
WINDOW PROGRAMS AND STANDARDS**

**COMPILATION OF ALL WINDOWS SIMULATED
AND TESTED FOR U.S. AND CANADIAN
VALIDATION PROJECTS**

December 13, 1994

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One of the most important issues facing the window community in North America is the harmonization of Canadian and US standards and simulation programs. There was support for this concept from all parties involved including the window industry, LBL and Canadian representatives.

This report identifies the 38 windows, including the 18 windows used for the Canada-US Phases 1 and 2 and ASHRAE Research Project, which have been simulated and tested for U.S. and Canadian validation projects. Also included are the simulation and test results for the windows. The details of these windows could be helpful in validating the WINDOW 5.0 program. To assist Lawrence Berkeley Laboratory in validation of WINDOW 5.0, we will provide the drawings for a few of the windows given in this report at Lawrence Berkeley Laboratory's request.

The list of windows is separated according to project. Under the US/Canada Joint Research Program two round-robins were performed and a total of 18 windows were tested and simulated. In addition, the Canadian Standards Association (CSA) sponsored a project which included 7 windows; 3 windows were tested and simulated for Queen's University; 3 windows were tested and simulated for the Flair Homes Project; and 7 windows were tested and simulated under the ASHRAE/NFRC project. We have not included the results for the seventh window in the CSA study because this window is used in continuing validation exercises.

Window Simulation and Test Results from
U.S. and Canadian Validation Projects

A. US/Canada Joint Research Program
A1. First Round Robin

ID	DESCRIPTION	U _{sim}	U _{test}	U _s /U _t
1	Aluminum, thermal break, fixed, sealed double glazing	3.37	3.55	0.95
2	Wood, casement, sealed double glazing	2.80	2.70	1.04
3	Aluminum, thermal break, fixed sealed double glazing	3.43	3.51	0.98
4	Aluminum, no thermal break, fixed, double glazing	3.53	3.75	0.94
5	Aluminum, no thermal break, fixed, single lazing	6.61	6.85	0.96
6	Aluminum, thermal break, fixed, double low-e glazing	3.10	3.12	0.99
7	Wood, casement, triple low-e glazing, 80% Kr, 20% Ar	1.44	1.54	0.94
8	Wood, Casement, sealed double glazing, 500 mm X 800 mm	2.70	2.44	1.11
9	Wood, casement, sealed double glazing, 700 mm X 800 mm	2.78	2.63	1.06
10	Wood, casement, sealed triple glazing, 500 mm X 800 mm	2.00	1.89	1.06
11	Wood, casement, sealed triple glazing, 700 mm X 1800 mm	1.96	1.92	1.02

A2. Second Round Robin

ID	DESCRIPTION	U _{sim}	U _{test}	U _s /U _t
1	Experimental urethane foam frame, fixed, sealed quad glazing (two lites of glass, two suspended films with e of 0.12, two steel spacers and one insulated spacer, 10% air and 90% Kr.	0.74	0.85	0.87
2	Wood, casement, quad glazing (as above)	1.08	1.20	0.90
3	Aluminum clad wood, casement, quad glazing (as above)	1.59	1.57	1.01
4	Aluminum clad wood, fixed, triple glazing with Swiggle Strip spacers and two low-e coatings (0.09), argon filled	1.11	1.24	0.90
5	Vinyl slider, sealed double glazing with aluminum spacer, low-e coating (0.12) argon filled, integral grills	2.26	2.06	1.10
6	Aluminum clad wood, awning, triple glazing with aluminum spacers, two low-e coatings, argon filled	1.76	1.82	0.97
7	Vinyl, tilt turn, sealed double glazing with aluminum spacers, one low-e coating (0.12)	2.07	2.08	1.00

B. CSA Study

ID	DESCRIPTION	U _{sim}	U _{test}	U _s /U _t
1	Aluminum, 4 vertical sliding single glazed aluminum sashes	3.00	3.23	0.93
2	Vinyl, 2 vertical sliding sealed double glazed sashes, low-e, argon, Superspacer	1.96	1.96	1.00
3	Wood, 12 fixed, sealed double glazing, low-e, argon, standard aluminum spacer	2.39	2.22	1.07
4	Wood, 12 fixed, sealed double glazing, low-e, argon, Swiggle Strip spacer	2.23	2.08	1.07
5	Wood, 12 fixed sealed double glazing, low-e, argon, Superspacer	1.98	2.04	0.97
6	Aluminum, 4 horizontal sliding single glazed sashes	3.05	3.03	1.01
7	Aluminum, 2 horizontal sliding sashes with sealed double glazing, low-e	x	y	0.95

C. Queen's University Study

ID	DESCRIPTION	U _{sim}	U _{test}	U _s /U _t
1	Aluminum, thermal break, casement, sealed double glazing	3.98	3.98	1.00
2	Wood, casement, sealed double glazing	2.81	3.09	0.91
3	Wood, casement, sealed triple glazing with low-e coatings	2.12	2.25	0.94

D. Flair Homes Project

ID	DESCRIPTION	U _{sim}	U _{test}	U _s /U _t
1	Wood, fixed, sealed triple glazing with low-e coatings	1.41	1.51	0.93
2	PVC coated wood, fixed, triple glazing with low-e coatings	1.67	1.66	1.01
3	Wood, fixed, double glazing with two HM	2.20	2.09	1.05

E. Current ASHRAE/NFRC project

ID	DESCRIPTION	U _{sim}	U _{test}	U _s /U _t
1	Vinyl, horizontal slider, sealed double glazing	2.98	3.02	0.99
2	Wood, double hung, sealed double glazing	2.21	2.36	0.94
3	Vinyl clad wood, casement, sealed double glazing	2.78	2.84	0.98
4	Aluminum clad wood, skylight, sealed double glazing	3.55	3.60	0.99
5	Glass fibre-reinforced plastic, voids filled with glass fibre insulation, casement, sealed, double glazing	2.60	2.88	0.99
6	Aluminum clad wood patio door, sealed double glazing	2.99	2.97	1.01
7	Aluminum, no thermal breaks, sealed double glazing	4.49	4.37	1.03

NOTES:

1. U-values are in units of W/(m² - °C).
2. Simulations performed by enermodal using the FRAME and WINDOW computer programs.
3. Tests performed by NRC.
4. Multiple glazing is air filled unless otherwise noted.
5. Spacers for sealed glazing are aluminum unless otherwise noted.
6. All simulations were made "blind" except for Series C.
7. U_{test} is average of NRC and ATI results after adjustment for areas for Series E.

WINDOW SCHEMATICS

FRAME REPRESENTATION FOR WINDOW #1

FRAME REPRESENTATION FOR WINDOW #2 (SILL)

FRAME REPRESENTATION FOR WINDOW #2 (HEAD/JAMB)

FRAME REPRESENTATION FOR WINDOW #4

FRAME REPRESENTATION FOR WINDOW #5

A1.5

FRAME REPRESENTATION FOR WINDOW #6

FRAME REPRESENTATION FOR WINDOW #7

HEAT FLOW PLOT

Heat Flow - Total : 15.06

(W/m depth) - Step: 15.06

FRAME 2.0

Date: 03-09-90

Time: 11:11:15

File: nrcdg

HEAT FLOW PLOT

Heat Flow - Total : 11.72

(W/m depth) - Step: 11.72

FRAME 2.0

Date: 03-09-90

Time: 11:00:04

File: nrctg

<p>Ar</p> <p>Andersen Corporation Bayport, Minnesota 55003</p>	
<p>DATE: 12/3/85</p>	<p>DRAWING NO. PSF-10186</p>

PERMA-SHIELD® FLEXIFRAME®
 WINDOW COMPONENTS
 WHITE OR TERRATONE

A2.3

TITLE		HORIZONTAL PICTURE	
W/REVEALS & NUMBERS		HURD	
DRAWING NO.	CSE510A	WOOD WINDOWS	
REVISION	-		
CUTSTOCK SIZE	-		
LIN. PART NO.	-		

A2.2

ORTECH.

Report No. 38-40390-W89-01

Loewen/Awning

6

Loewen Wood Awning Window

FIGURE 1
VERTICAL and HORIZONTAL CROSS - SECTIONS

A2.4

12/01/13

REPLA DOUBLE HUNG WINDOWS

DHW

WELDED HEAD, JAMBS AND SASHES; MECHANICALLY ASSEMBLED SILL; FULL SCREEN;
EQUAL GLASS SIZES.

PROFILE OPTIONS

FRAME	SASH
1226 H&J*	1306 residen.
1228 sloped sill	top
	1329
	commercial
	bottom

VERTICAL SECTION

DETAILS FIGURE 'C'

HEAD

DETAILS FIGURE 'D'

DETAILS FIGURE 'F'

HORIZONTAL SECTION

SILL

- Optional vinyl brickmould and aluminum sill for brick veneer construction.

DETAILS FIGURE 'E'

SCALE 1:2

A2.5

AWNING WINDOWS

R1A14

INSTALLATION DETAILS (BRICK VENEER) AND FRAME SECTIONS WITH SLOPED METAL CLADDING AND VINYL LINERS (SC/VL) - ALSO AVAILABLE WITH WOOD LINERS..

HEAD

VISIBLE
GLASS

FRAME SIZE

JAMB

VISIBLE
GLASS

FRAME SIZE

Hinge?

SILL

VISIBLE
GLASS

FRAME SIZE

SCALE 1:5

SCALE 1:2

Specifying

A2.6

pic to Scale

A2.7

AUG 17 '90 10:57 TO 15198840103

FROM MECHANICAL ENG.

T-611 P.04

FIGURE 1: FRAME REPRESENTATION OF WINDOW #8

FIGURE 2: FRAME REPRESENTATION OF WINDOW #9

FIGURE 3: FRAME REPRESENTATION OF WINDOW #10

FIGURE 5: FRAME REPRESENTATION OF WINDOW #12

FIGURE 6: FRAME REPRESENTATION OF WINDOW #13

FIGURE 4: FRAME REPRESENTATION OF WINDOW #11

L.3

Figure 1

Window 1

Head Drawing

Head FRAME Representation

B.1

Figure 2

Window 1

Sill Drawing

Sill FRAME Representation

Figure 3

Window 1

Left Upper-Jamb Drawing

Left Upper-Jamb FRAME Representation

Figure 4

Window 1

Left Lower-Jamb Drawing

Left Lower-Jamb FRAME Representation

Figure 5

Window 1

Right Upper-Jamb Drawing

Right Upper-Jamb FRAME Representation

Figure 6

Window 1

Right Lower-Jamb Drawing

Right Lower-Jamb FRAME Representation

Figure 7

Window 1

Meeting Rail Drawing

Meeting Rail FRAME Representation

Figure 8

Window 2

Head Drawing

Head FRAME Representation

B.2

Figure 9

Window 2

Sill Drawing

Sill FRAME Representation

Figure 10

Window 2

Upper-Jamb Drawing

Upper-Jamb FRAME Representation

Figure 12

Window 2

Meeting Rail Drawing

Meeting Rail FRAME Representation

Figure 11

Window 2

Lower-Jamb Drawing

Lower-Jamb FRAME Representation

Figure 13

Windows 3,4,5

Head/Sill/Jamb Drawing

Head/Sill/Jamb FRAME Representation

Figure 14

Windows 3,4,5

True Divided Lite Drawing

True Divided Lite FRAME Representation

Figure 15

Window 6

Left Head Drawing

Left Head FRAME Representation

R.C.

Figure 16

Window 6

Right Head Drawing

Right Head FRAME Representation

Figure 17

Window 6

Left Sill Drawing

Left Sill FRAME Representation

Figure 18

Window 6

Right Sill Drawing

Right Sill FRAME Representation

Figure 19

Window 6

Left Jamb Drawing

Left Jamb FRAME Representation

Figure 20

Window 6

Right Jamb Drawing

Right Jamb FRAME Representation

Figure 21

Window 6

Meeting Rail Drawing

Meeting Rail FRAME Representation

Figure 21

Window 6

Meeting Rail Drawing

Meeting Rail FRAME Representation

Figure 22

Window 7

Head Drawing

Head FRAME Representation

Figure 23

Window 7

Sill Drawing

Sill FRAME Representation

Figure 24

Window 7

Upper-Jamb Drawing

Upper-Jamb FRAME Representation

Figure 26

Window 7

Meeting Rail Drawing

Meeting Rail FRAME Representation

Figure 25

Window 7

Lower-Jamb Drawing

Lower-Jamb FRAME Representation

Manufacturer's Drawing of the Vertical Section - Vinyl Horizontal Slider

Manufacturer's Drawing of the Vertical Section - Wood Double-Hung E.2

Manufacturer's Drawing of the Vertical Section - Vinyl-Clad Casement

Manufacturer's Drawing of the Vertical Section - Aluminum-Clad Wood Skylight

Manufacturer's Drawing of the Vertical Section - Aluminum-Clad Wood
Patio Door

Manufacturer's Drawing of the Vertical Section - Aluminum Horizontal Slider