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Building Research Note

DBR's Approach for Determining the Heat Transmission Characteristics of Windows

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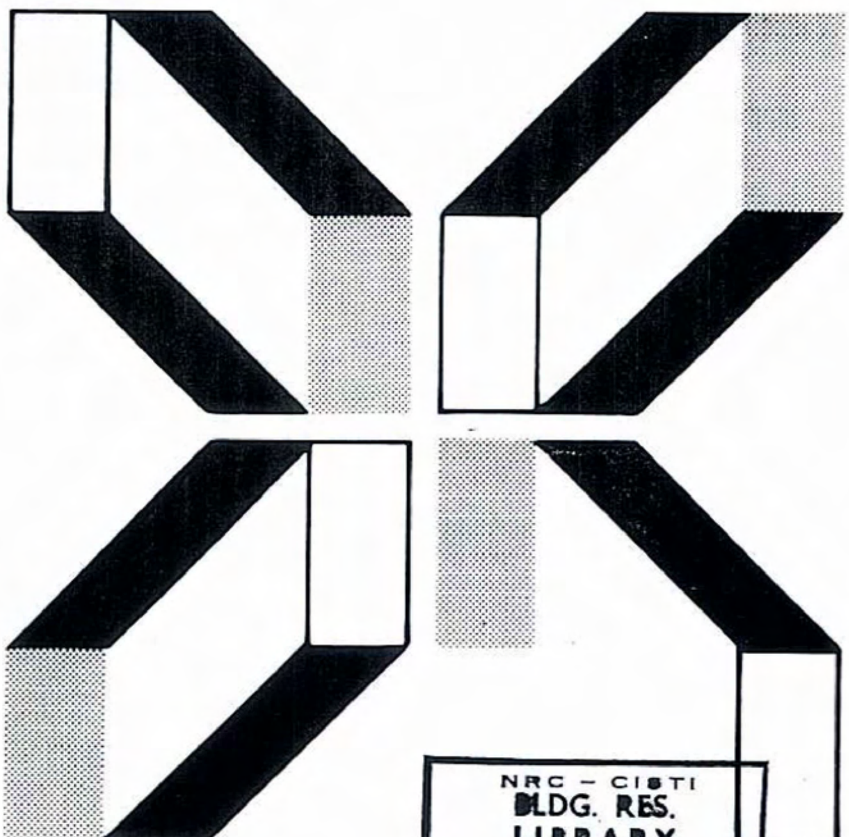
BRN 234



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DBR'S APPROACH FOR DETERMINING THE HEAT TRANSMISSION CHARACTERISTICS OF WINDOWS

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BRN 234
ISSN 0701-5232
Ottawa, November 1985

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ABSTRACT

Current standard hot box techniques for determining the thermal conductance of walls are not adequate for use with windows. This report discusses the problems with testing windows and presents the Division of Building Research window calorimeter designed to reduce the errors associated with determining calorimeter enclosure surface temperature and air temperature.

RÉSUMÉ

Les techniques de chambre chaude normalisées actuelles qui servent à déterminer la conductance thermique des murs ne conviennent pas dans le cas des fenêtres. Le présent rapport examine la question des essais de fenêtres et décrit le calorimètre pour fenêtres de la Division des recherches en bâtiment, appareil destiné à réduire les erreurs associées à la température de la surface et de l'atmosphère de l'enceinte calorimétrique.

INTRODUCTION

The Division of Building Research of the National Research Council of Canada (DBR/NRCC) has for many years supported an on-going program to study the thermal performance of windows. Early work concentrated on surface temperature performance as it relates to window condensation, but as energy costs have increased attention has been directed towards heat transmission characteristics. DBR, using a window calorimeter it developed for its environmental test facility, is working on a heat transmission or thermal resistance test for windows.

This report discusses the development of the DBR window calorimeter and the test procedure, in which is calculated an equivalent surface temperature that is used to determine the coefficient of heat transfer between inside and outside specimen surfaces. Experience in applying this procedure, using a calorimeter designed for walls, facilitated the design of a window calorimeter incorporating a uniform temperature baffle and convection heater. Calibration of the window calorimeter is discussed and an example calculation is given.

BACKGROUND

Information on the thermal performance of windows is required for two main reasons:

- to assist designers to predict heat losses,
- to provide a ranking of windows for thermal comparison.

For the first category the test would have to be complex and include factors affecting heat transmission, i.e., temperature, solar incidence, wind, room-side air movement, air leakage, etc. Such an evaluation would be very expensive. The second category, which is of immediate interest, is designed to provide a ranking of products in the laboratory that would be similar to the ranking expected in the field. The procedure could be standardized to ensure that the values from different facilities are comparable. It would allow for comparison of different windows, as well as assessment of thermal improvement in a particular window system.

For ranking purposes windows can be described by their U-value, which is the overall coefficient of heat transfer (air to air), or by their C-value, which is the coefficient of heat transfer between inside and outside surfaces. There are a number of test methods that claim to measure U-values of high conductance specimens such as windows: ASTM C236, Standard Test Method for Steady-State Thermal Performance of Building Assemblies by Means of a Guarded Hot Box; ASTM C976, Standard Test Method for Thermal Performance of Building Assemblies by Means of a Calibrated Hot Box; AAMA 1503.1, a method developed by the Architectural Aluminum Manufacturers Association. The two ASTM standards include a C-value calculation based on surface temperature measurements and area weighting. Because measurements of temperature on glass and area weighting are not very accurate, DBR is developing an alternative approach for establishing C-values for windows. This procedure involves calculating a room-side equivalent surface temperature for the specimen by means of the radiative and convective components of heat tranfer between calorimeter and specimen. A weather-side specimen surface temperature is also calculated using the specimen heat flow and the weather-side surface film resistance. The characteristics of the calorimeter and weather-side surface film must first be established by calibration tests (discussed later). This procedure was developed by Solvason et al. (1) in the early 1960s for determining the equivalent inside surface temperature of non-isotropic walls.

The benefits of the DBR procedure for calculating a C-value are:

1) it is possible to compare results from different laboratories. As exact repeatability of the air films produced by different test facilities is not required, the air films can be specified within broad limits. Emphasis is placed on calibration of equipment over a range of heat flows and specimen sizes to define the facility characteristics.

2) the uncertainty introduced by using thermocouples to measure glass and frame temperatures to be used in calculating an area-weighted average surface temperature is eliminated.

- Radiant heat exchange among various surfaces within the calorimeter and on the specimen surface is accounted for in order to increase accuracy of C-value determination.
- A design U-value used for product comparison can be calculated using standard values for the inside and outside air films.

THEORY

In summary (2), the approach involves measuring the total power supplied to the calorimeter and deducting the heat transfer through the mask (a support wall of known properties) to arrive at the heat transfer through the specimen. From the specimen heat transfer, using the relations for radiative and convective heat tranfer from the calorimeter to the specimen, the equivalent room-side surface temperature of the specimen can be calculated. The equivalent weather-side surface temperature is also calculated from the specimen heat transfer and the air film provided by the DBR wind machine. The thermal conductance, resistance, design thermal resistance, and design coefficient of heat transmission are then calculated.

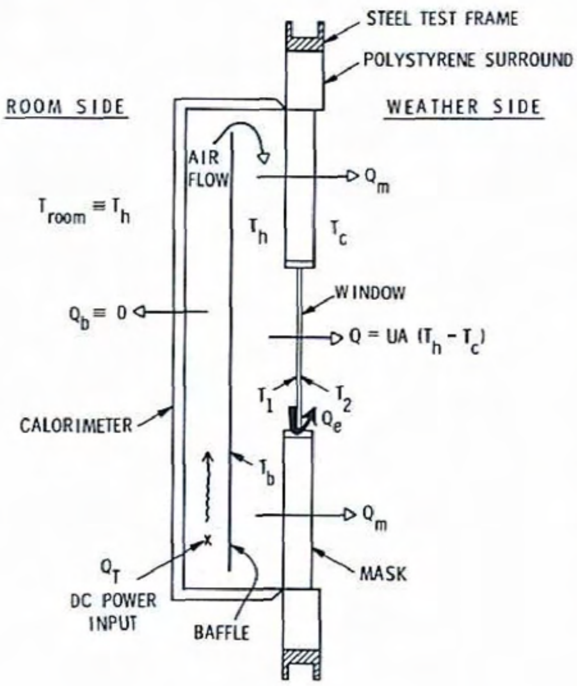

Figure 1. Calorimeter box

The thermal analysis of specimens mounted in a support wall (mask) placed between a weather-side and a room-side chamber is based on the heat balance in the calorimeter box (Figure 1).

$$Q_T = Q_b + Q_m + Q_e + Q_s \tag{1}$$

where Q_T = total power supplied to calorimeter
Q_b = heat flow through calorimeter box walls (controlled so as to be of negligible value)
Q_m = heat flow through the mask (function of surface temperatures and mask conductance)
Q_e = flanking loss around the edge of the specimen
Q_s = heat flow through the specimen.

Noting that Q_s is equal to the heat flow through the surface of the specimen, it may be expressed as

$$Q_s = Q_r + Q_c \tag{2}$$

where Q_r = radiant heat exchange between calorimeter surfaces and room-side surface of the specimen, and
Q_c = convective heat exchange between calorimeter air and room-side surface of the specimen.

In a grey enclosure the radiation exchange between two surfaces can be expressed as:

$$Q_r/A_1 = q_r = F_{1b} \cdot \sigma \cdot (T_b^4 - T_1^4) \tag{3}$$

where A_1 = area of surface 1
$_{1b}$ = overall interchange factor for radiation from surface 1 to surface b
$F\sigma$ = Stefan-Boltzman constant*
T_b = absolute temperature of surface b (calorimeter baffle)
T_1 = absolute temperature of surface 1 (specimen).

The convection component is approximated by

$$Q_c/A_1 = q_c = C \cdot (T_h - T_1)^B \tag{4}$$

where T_h = the absolute temperature of the air in the calorimeter box, and B and C are constants to be determined from calibration tests. Equations 3 and 4 can be approximated by:

$$q_r = h_r \cdot (T_b - T_1) \tag{5}$$

$$q_c = h_c \cdot (T_h - T_1) \tag{6}$$

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*for SI units $\sigma = 5.6703 \times 10^{-8}$, $W/(m^2 \cdot K^4)$

where $h_r = F_{1b} \cdot \sigma \cdot (T_b^2 + T_1^2) \cdot (T_b + T_1)$

$h_c = C \cdot (T_h - T_1)^{B-1}$

As the radiative and convective heat flow paths are parallel, the sum of the two coefficients, h_r and h_c, is the room-side combined heat transfer coefficient (air film) for the configuration under test, f_i. If T_h was equivalent to T_b, the total heat transfer to and through the specimen can be expressed as:

$$\frac{Q_s}{A_1} = q_s = f_i \cdot (T_h - T_1) \qquad (7)$$

The weather-side air film, f_o, is:

$$f_o = q_s/(T_2 - T_c) \qquad (8)$$

where T_2 = calculated weather-side surface temperature

T_c = weather-side air temperature.

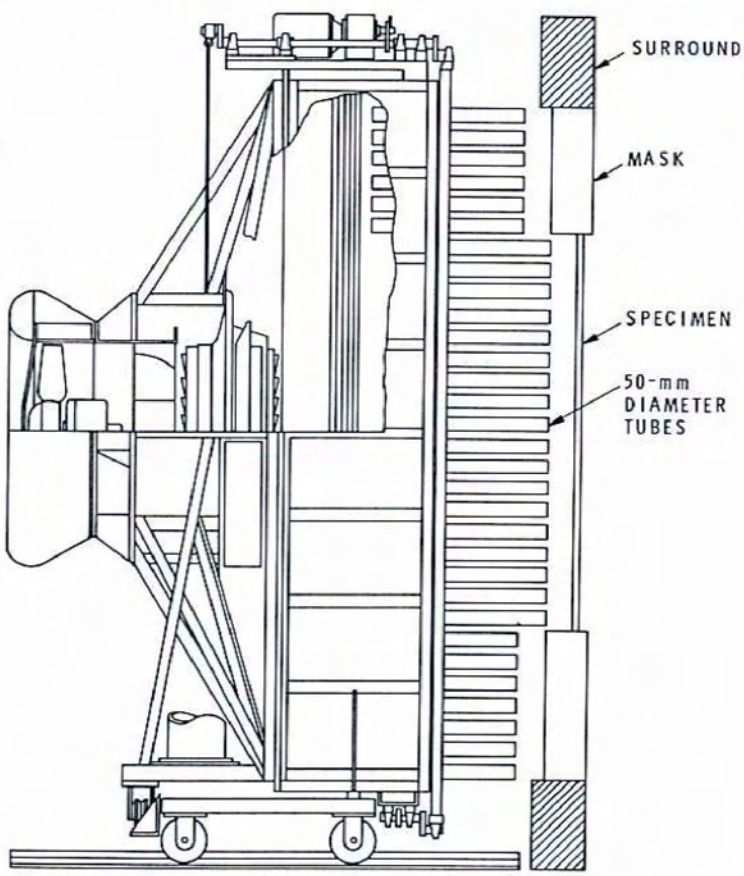

Figure 2. DBR wind machine

The above calculations are based on the assumption that over the range of weather-side temperatures to be used the DBR wind machine (Figure 2) generates a relatively constant f_o and the radiating temperatures are equivalent to T_c. The test specimen conductance, C, and resistance, R, are calculated by:

$$C = q_s/(T_1 - T_2)$$
$$R = 1/C \quad (9)$$

For comparison, a design thermal resistance value, R_D, can be obtained by the addition of standard surface film resistances, R_{fi} and R_{fo}, to the specimen thermal resistance value R:

$$R_D = R_{fi} + R + R_{fo} \quad (10)$$

The design U-value is then $1/R_D$.

PROBLEMS WITH CONVENTIONAL CALORIMETER FOR WINDOW TESTING

Using the above approach and the DBR calorimeter designed for walls, a test series was undertaken to evaluate the thermal performance of sealed glazing units. This indicated that the wall calorimeter design is not suitable for window testing. The main problem is its inability to maintain a uniform and definable calorimeter enclosure surface temperature and air temperature. The heat transfer through the window relative to the mask causes the calorimeter surfaces to be colder than T_h and both T_b and T_h to be non-uniform.

DBR WINDOW CALORIMETER

This calorimeter, designed specifically to measure the heat transmission characteristics of windows, has the following features:

- uniform temperature baffle to give a uniform T_b
- uniform temperature convection heater to improve the uniformity of T_h.

An evaporating/condensing panel using refrigerant R-12 was designed to provide a uniform temperature on the baffle (Figure 3). A tubular heater for boiling the refrigerant is contained in a pipe reservoir connected to the panels.

The uniform-temperature convection heater was also obtained by evaporating and condensing R-12 (Figure 4). It was constructed of two fin tubes connected to a copper pipe reservoir containing a second tubular heater of the same power rating, for simplicity, as that in the baffle. The constant-temperature baffle and convection heater were installed in the window calorimeter box (Figure 5).

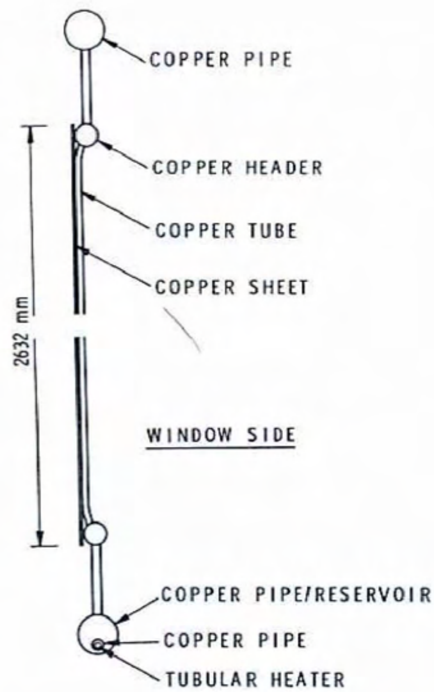

Figure 3. Uniform temperature baffle

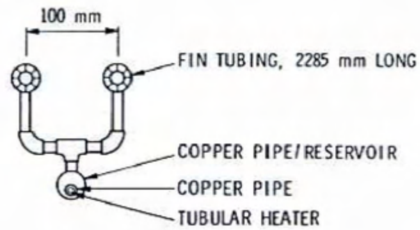

Figure 4. Uniform temperature convection heater

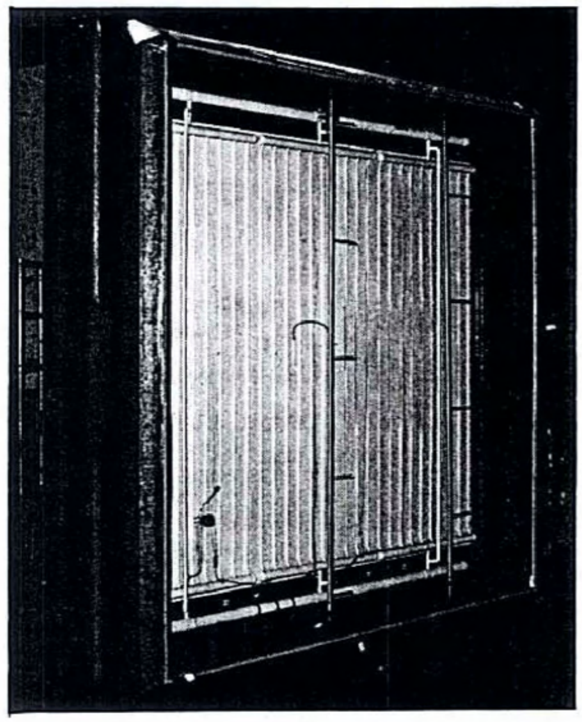

Figure 5. Window calorimeter

Calibration

Ability to measure the total power input to the calorimeter, Q_T, and the uniformity of the baffle and air temperature (T_b and T_h) have been reported (1). The final part of the calibration program now nearing completion is:

- to determine whether the radiation exchange between baffle and specimen can be expressed sufficiently accurately using F_{1b} derived from simplified equations.
- to determine B and C constants for a range of power inputs and specimens.

Two models are being considered for the radiation exchange between specimen and calorimeter enclosure. The first model is based on a simple, parallel heat transfer concept between the specimen and the constant temperature baffle. The second model assumes that there is an exchange among all surfaces of the enclosure and specimen. The latter is a more accurate model of the radiation exchange within the calorimeter enclosure, but it requires more detailed analysis.

For the first model, F_{1b} is represented by

$$F_{1b} = \frac{1}{\frac{1}{e_1} + \frac{A_1}{A_b}\left(\frac{1}{e_b} - 1\right)} \tag{11}$$

where e_1, e_b = emissivities of the specimen and baffle,
A_1, A_b = area of the specimen and baffle.

The second model uses programs developed by Mitalas and Stephenson (3) to calculate radiant energy interchange factors for the calorimeter enclosure. These factors are used in the following to calculate q_r:

$$q_r = Q_r/A_1 = \sigma \sum_{i=2}^{6} F_{1i} \cdot (T_i^4 - T_1^4) \tag{12}$$

and

$$h_r = \frac{\sigma \sum_{i=2}^{6} F_{1i}\,(T_i^4 - T_1^4)}{(T_b - T_1)}$$

where T_i = absolute temperature of surface i, and F_{1i} = interchange factor for surface 1 to surface i.

For calibration tests using the specimen shown in Figure 6, Q_s is determined from the following

$$Q_s = A_s \cdot C_c \cdot \Delta T_c \tag{13}$$

where A_s = area of calibration specimen
C_c = conductance of calibration specimen core
ΔT_c = measured temperature drop across the calibration specimen core.

The surface temperatures, T_1 and T_2, for the calibration tests are calculated from the following:

$$T_1 = T_1' + \frac{C_c}{C_g}\,\Delta T_c;\; T_2 = T_2' - \frac{C_c}{C_g}\,\Delta T_c \tag{14}$$

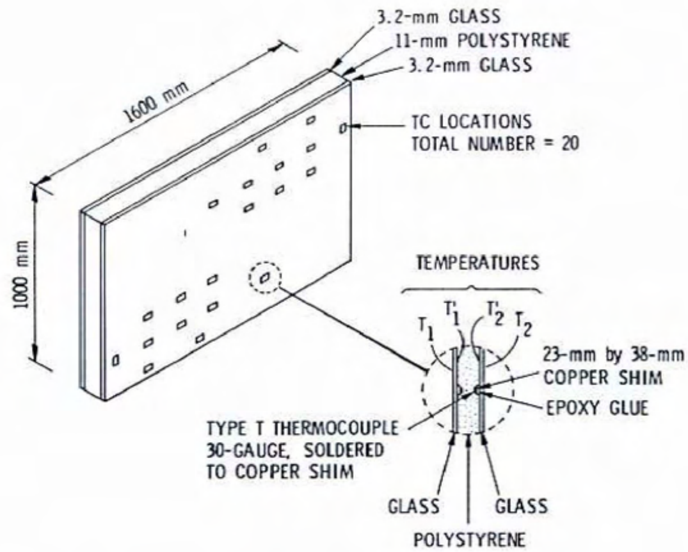

Figure 6. Calibration specimen

where T_1' = weighted average warm-side surface temperature of calibration specimen core (measured at the glass facing/core interface of calibration specimen)

T_2' = weighted average cold-side surface temperature of calibration specimen core (measured at the glass facing/core interface of calibration specimen)

C_g = conductance of the glass facing.

Using the values for Q_s and T_1 determined by Equations 13 and 14, Q_r can be calculated and Q_c established for each set of power input conditions using Equations 2, 3 and 4. The B and C constants for the convective heat transfer are established from a curve fit of log Q_c versus log $(T_h - T_1)$. f_o is obtained from Equation 8, using q_s, T_2 and T_c.

Q_e, the flanking loss around the edge of the calibration specimen, is not included in the Q_s value for calibration and characterization of the calorimeter box. For testing of windows, however, Q_s is determined by subtracting the mask loss, Q_m (calculated from surface temperature measurements and the conductance expression for the mask), from the total calorimeter power, Q_T. This means that the window specimen heat flow will include Q_e losses. It will be accounted for in the calculated equivalent surface temperature for the specimen, T_1, since a window with a high conductance frame will then have a lower T_1 than one with a low conductance frame. For comparative ranking this appears to be a reasonable compromise, in that Q_e cannot be adequately established for all combinations of window systems to be tested.

Example. Calorimeter characteristics derived for a particular test set-up are:

$$q_r = 0.76 \cdot \sigma \cdot (T_b^4 - T_1^4) \text{ W/m}^2 \qquad \text{(Eq. 3)}$$

$$q_c = 1.684\ (T_h - T_1)^{1.287}\ W/m^2 \qquad \text{(Eq. 4)}$$

$$f_o = 21.9\ W/(m^2 \cdot K)$$

Measured values:

$Q_T = 260$ W $\qquad T_c = 252.2$ K (-21.0°C)

$A_1 = 1.605\ m^2$ $\qquad T_h = 295.0$ K (-21.8°C)

$\qquad T_b = 295.4$ K (-22.2°C)

Calculations:

$Q_m = 57.8$ W

$Q_s = 260 - 57.8 = 202$ W

$q_s = Q_s/A_1 = 126\ W/m^2$

$$q_s = q_c + q_r = 126\ W/m^2 \qquad \text{(Eq. 2)}$$

$$q_r = 0.76 * 5.6703 * 10^{-8}\ (295.4^4 - T_1^4)\ W/m^2 \qquad \text{(Eq. 3)}$$

$$q_c = 1.684\ (295.0 - T_1)^{1.287}\ W/m^2 \qquad \text{(Eq. 4)}$$

Solving the above three equations for q_c, q_r and T_1 yields

$q_c = 59.2\ W/m^2$

$q_r = 66.8\ W/m^2$

$T_1 = 279.1$ K (5.9°C)

By Equations 5 and 6, therefore

$$h_r = q_r/(T_b - T_1) = 66.8/(295.4 - 279.1) = 4.10\ W/(m^2 \cdot K) \qquad \text{(Eq. 5)}$$

$$h_c = q_c/(T_h - T_1) = 59.2/(295.0 - 279.1) = 3.72\ W/(m^2 \cdot K) \qquad \text{(Eq. 6)}$$

so that $f_i = h_c + h_r = 7.82\ W/(m^2 \cdot K)$.

Using the weather-side air film, T_2 may be calculated by means of Equation 8:

$$T_2 = \frac{q_s}{f_o} + T_c = \frac{126.0}{21.9} + 252.2$$

$$T_2 = 258.0 \text{ K } (-15.2°\text{C})$$

The test specimen conductance or C-value can now be calculated by means of Equation 9:

$$\text{C-value} = q_s/(T_1 - T_2) = 126/(279.1 - 258.0)$$

$$\text{C-value} = 5.97 \text{ W}/(\text{m}^2 \cdot \text{K})$$

$$\text{R-value} = 1/C = 0.168 \text{ m}^2 \cdot \text{K/W}$$

For comparison, a design U-value, U_D, and design thermal resistance value, R_D, can be defined by adding standard ASHRAE air films and calculated as follows:

$$R_D = 0.12 + \text{R-Value} + 0.03 \qquad \text{(Eq. 10)}$$

$$= 0.318 \text{ m}^2 \cdot \text{K/W}$$

$$U_D = 1/R_D = 3.14 \text{ W}/(\text{m}^2 \cdot \text{K})$$

SUMMARY

Current standard hot box techniques for determining the thermal conductance and resistance of walls are not adequate for use with windows. Measuring glass and frame temperatures for use in calculating a weighted average surface temperature for the window is impractical. Determination of laboratory U-values for windows does not recognize that the surfaces of the calorimeter will be colder than the air temperature, so that a single air temperature cannot properly describe the heat transfer to the window.

The Division of Building Research earlier developed a procedure for determining the thermal conductance and resistance of non-isotropic walls that is now being applied to windows. A calorimeter designed specifically for windows reduces the errors associated with determining calorimeter enclosure surface temperature and air temperature.

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