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Effects of Glass Plate Curvature on the U-Factor of Sealed Insulated Glazing Units

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ABSTRACT

This paper presents the results of a study aimed at quantifying the change in the U-factor caused by glass plate curvature in sealed, insulated glazing (IG) units. The curvature may be caused by a number of factors, two of which will be studied in this paper—barometric pressure and gas space temperature variations. In the first part of this paper, the equations governing glass plate deflections and heat transfer through IG units are briefly reviewed. Then, glass plate deflections and the resulting change in the U-factor of several IG units are examined for ASHRAE-type winter conditions. Yearly simulations were also performed for Montréal, Canada, and Toulouse, France, to examine the combined effects of exterior temperature, barometric pressure, and wind speed. These last results show that the U-factor of a particular IG unit (triple glazing, low-emissivity with air) may vary up to 5% above and 10% below the yearly average.

INTRODUCTION

Windows constitute an important part of the building envelope and accurate values of their U-factors (the inverse of thermal resistance) are essential for design and energy analysis of buildings. The *1993 ASHRAE Handbook—Fundamentals* (ASHRAE 1993) provides overall coefficients of heat transmission (commonly known as U-factors) for most commercially available windows. The center-of-glass U-factors reported by ASHRAE are based on the assumption that the glass plates are flat and parallel to each other. In reality, the glass plates are never truly parallel because of naturally occurring pressure differentials that tend to bend the plates. Notwithstanding the preliminary study of Bourret et al. (1995), apparently no work has been reported in the literature on the change in the U-factors caused by glass plate curvature.

A schematic of an insulated glazing (IG) unit is shown in Figure 1. These units are usually inserted into a frame to form

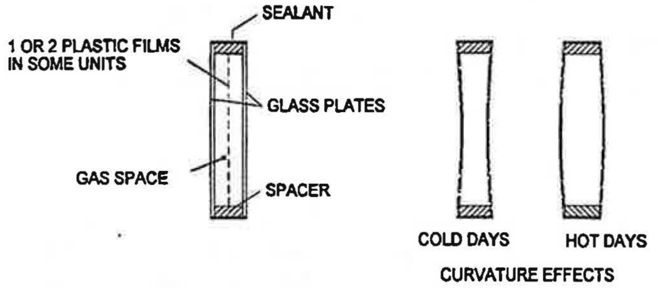

Figure 1 *Schematic of an insulated glazing (IG) unit.*

a window unit. Typically, an IG unit is composed of two plates of glass separated by a spacer and filled with either dry air or argon. The initial gas pressure and temperature are usually the atmospheric pressure and ambient temperature prevailing during fabrication. A sealant is fixed around the perimeter to provide a hermetic seal. In some units, one or two low-emissivity (low-e) plastic films are inserted in the gas space to increase thermal resistance. A small hole in the plastic film equalizes the pressure on both sides of the film. Typically, the thickness of the glass plates ranges from 3 to 8 mm (1/8 to 3/8 in.) and the gas space thickness varies from 6 to 25 mm (1/4 to 1 in.).

During cold winter days, the gas temperature inside the sealed unit will decrease below the initial filling temperature. Thus, the internal gas pressure will decrease and the panes of glass will be subjected to a pressure differential, under which the glass will bend inward, as indicated in Figure 1. Conversely, on hot days the glass will bend outward. In addition to this temperature effect, glass deflection is also affected by varying barometric pressure. Other factors, such as nitrogen adsorption by the desiccant, initial plate curvature during fabrication, wind pressure, and temperature nonuniformity of the plates, may also influence glass plate deflection. However, this paper is only concerned with effects of temperature and barometric pressure.

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Aside from the unpleasant visual distortion of reflected objects (Patenaude 1991), glass plate curvature changes the center-of-glass U-factor of the IG unit. The objective of this paper is to quantify these changes. The changes in the U-factor resulting from glass plate deflections are first evaluated for several IG units under ASHRAE-type winter conditions. Then, the yearly variation of the U-factor is evaluated for a particular IG unit for Montréal, Canada, and Toulouse, France. Before presenting these results, the governing equations and the solution methodology will be briefly described.

GOVERNING EQUATIONS

The present study is concerned with changes in the U-factor of IG units caused by glass plate curvature, as shown in Figure 1. The glass plates composing the IG unit are assumed to be initially flat and parallel to each other.

Mechanical Aspects

Deflections are calculated using thin plate theory (Timoshenko and Woinowsky-Krieger 1959). For a rectangular plate, simply supported and uniformly loaded with a pressure P, the deflection w at a location x,y is given by

$$w(x, y) = \frac{16P}{\pi^6 D} \sum_{m = 1, 3, 5, \ldots} \sum_{n = 1, 3, 5, \ldots} \frac{\sin\frac{m\pi x}{a}\sin\frac{n\pi y}{b}}{mn\left(\left(\frac{m}{a}\right)^2 + \left(\frac{n}{b}\right)^2\right)^2} \text{ with } D = \frac{Et^3}{12(1-\nu^2)} \tag{1}$$

where a and b represent the dimensions of the plate, E is Young's modulus ($E = 7.2 \times 10^{10}$ Pa [10.5×10^6 psi] for glass), t is the plate thickness (m [in.]), and ν is Poisson's ratio ($\nu = 0.22$ for glass). The maximum deflection is obtained at the center for $x = a/2$ and $y = b/2$ and the mean deflection, w_{mean}, is given by

$$w_{mean} = \frac{16P}{\pi^6 D} \sum_{m = 1, 3, 5, \ldots} \sum_{n = 1, 3, 5, \ldots} \frac{4}{m^2 n^2 \pi^2 \left(\left(\frac{m}{a}\right)^2 + \left(\frac{n}{b}\right)^2\right)^2}. \tag{2}$$

The series in Equations 1 and 2 converge rapidly and the first two to three terms in each summation are usually sufficient for an accurate determination of w. It is important to note that Equations 1 and 2 are strictly valid for small deflections (about the value of the plate thickness, Solvason [1974]). The value of P is given by

$$P = P_{baro} - P_{final} \tag{3}$$

where P_{baro} is the barometric pressure and P_{final} is the final gas space pressure. The value of P_{final} is obtained by assuming that the gas inside the IG unit behaves as a perfect gas. Thus,

$$\frac{P_{final}V_{final}}{T_{final}} = \frac{P_{init}V_{init}}{T_{init}} \text{ with } V_{final} = V_{init} - ab(w_{mean,1} + w_{mean,2}) \tag{4}$$

where P, T, and V represent pressure, absolute temperature, and volume, respectively. The subscripts *init* and *final* refer to initial and final gas space conditions, respectively. The mean deflections of glass plates 1 and 2 are given by $w_{mean,1}$ and $w_{mean,2}$, respectively. Equations 1 through 4 constitute a set of coupled equations and T_{final} must be determined to solve for w.

Thermal Aspects

Heat transfer in a differentially heated gas cavity consists of simultaneously occurring radiative and convective heat transfer. These processes are represented by two thermal resistances in parallel in Figure 2, with h_c and h_r representing the convective and equivalent radiative heat transfer coefficients, respectively. Convective heat transfer in a gas-filled cavity with a large aspect ratio has been the subject of a number of publications (e.g., ElSherbiny et al. [1982a]; Wright and Sullivan [1989]). Therefore, only a brief discussion of the key elements will be presented here.

Convective Heat Transfer The Nusselt number, Nu, is used to quantify convective heat transfer in a cavity. The value of Nu represents the ratio of heat transfer across the fluid in the convective regime over heat transfer in the purely conductive regime. For a given value of Nu, h_c is given by

$$h_c = \frac{\text{Nu } k_{gas}}{L}. \tag{5}$$

The value of Nu is dependent on the Rayleigh number, Ra ($= g\beta L^3 (T_2 - T_3)/\nu\alpha$); the Prandtl number, Pr; the vertical aspect ratio, b/L; and the horizontal aspect ratio, a/L. The influence of this last parameter is negligible for large horizontal aspect ratios ($a/L > 5$), as in most windows. Furthermore, for large horizontal and vertical aspect ratios, the thermal boundary conditions along the perimeter are unimportant (ElSherbiny

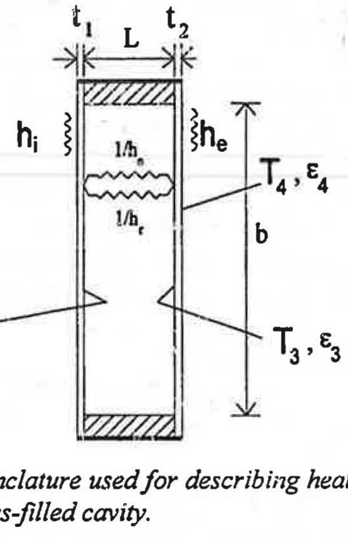


Figure 2 *Nomenclature used for describing heat transfer in a gas-filled cavity.*

et al. 1982b). In this work the detailed experimental correlations of ElSherbiny et al. (appendix in ElSherbiny et al. [1982a]) have been used to obtain values of Nu as a function of the vertical aspect ratio. These correlations were obtained using air as the heat transfer fluid (Pr = 0.71). However, the Prandtl number for most gases is close to the one for air, so these correlations can be used for other gases. These empirical correlations were obtained for two parallel flat plates and are reportedly accurate to within ±5% (Wright 1995).

The various flow regimes in a cavity have been described elsewhere (Wright and Sullivan 1989) and will only be briefly reviewed here. When the temperature difference across the cavity is small (small Ra), heat transfer across the cavity takes place primarily by conduction. The temperature profile is linear and this condition is referred to as the "conduction regime," with Nu essentially equal to 1.0. At higher Ra, heat transfer is augmented by bulk movement of the fluid from the hot to the cold side. This is the boundary layer regime; in that case, heat transfer takes place more by convection in the boundary layers and less by conduction in the core.

The correlations of ElSherbiny et al. are not strictly valid when the walls of the cavity are not flat. To account for plate curvature, a reduced gas space thickness, L_r, was used as the characteristic length. The value of L_r is simply given by

$$L_r = L - (w_{mean,1} + w_{mean,2}) \tag{6}$$

where subscripts 1 and 2 refer to plates 1 and 2, respectively. Thus, calculations are made as if the original rectangular cavity ($L \times a \times b$) was replaced by a new rectangular cavity ($L_r \times a \times b$). This assumption is questionable when there are large plate curvatures, as the walls of the pinched cavity might influence the convection current. The assumption becomes reasonable when plate curvature is small and it is certainly acceptable in the conduction regime, where the convection currents are not significant.

Radiative Heat Transfer The equivalent radiative heat transfer coefficient between the two plates is obtained here by assuming uniform plate temperatures. The plates are large enough so that a view factor of unity can be assumed. Under these assumptions and with reference to the nomenclature presented in Figure 2, h_r is simply given by

$$h_r = \sigma(T_2^2 + T_3^2)\frac{T_2 + T_3}{\frac{1}{\varepsilon_2} + \frac{1}{\varepsilon_3} - 1}. \tag{7}$$

Finally, the center-of-glass U-factor of the IG unit, U_{cg}, is given by

$$U_{cg} = \left(\frac{1}{h_i} + \frac{t_1}{k_{v,1}} + \frac{t_2}{k_{v,2}} + \frac{1}{h_c + h_r} + \frac{1}{h_e}\right)^{-1} \tag{8}$$

where h_i and h_e are the indoor and outdoor surface heat transfer coefficients, respectively, and k_v is the glass thermal conductivity. Typically, in this work, the values of h_i and h_e are the standard values of 8.29 and 29 W/m²·°C (1.46 and 5.11 Btu/h·ft²·°F) used by ASHRAE (1993). In the latter portion of this paper, the effects of wind speed will be investigated. For these cases, the value of h_e is obtained by adding the convective, $h_{e,c}$, and radiative, $h_{e,r}$, parts. These values are given by ASHRAE (1993):

$$\begin{aligned} h_{e,c} &= 7.2(V)^{0.78} \text{ for } 5 < V < 30 \text{ m/s} (16.4 < V < 98.4 \text{ ft/s}) \\ &= 5.62 + 3.9V \text{ for } V < 5 \text{ m/s } (V < 16.4 \text{ ft/s}) \\ h_{e,r} &= \varepsilon_4\sigma(T_4^4 - T_{ext}^4)/(T_4 - T_{ext}) \end{aligned} \tag{9}$$

where V is the wind speed.

Each term in Equation 8 contributes more or less to the overall value of U_{cg}. For example, in a standard IG unit (ε = 0.84 on all surfaces), the contributions, in terms of thermal resistance, of each term (from left to right on the right side of Equation 8) to the value of U_{cg} are: 35%, 1%, 1%, 55%, and 8%, respectively. Glass plate curvature will mainly affect the value of h_c and indirectly affects the value of h_r.

Solution Methodology

Equations 2 through 8 need to be solved simultaneously to obtain w and U, the two quantities of interest. An iterative solution method, using a commercially available equation solver, was used to solve this set of coupled equations. Fluid properties were evaluated at the mean gas space temperature, T_{final}, by taking the average temperature of the two walls (T_2 and T_3) forming the cavity.

RESULTS

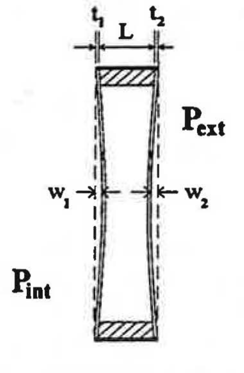
Preliminary Considerations

The thermal and mechanical portions of the solution methodology were first checked against published results. For the mechanical portion, Solvason's results (Solvason 1974) were used. The description of the geometry and the results of this comparison are presented in Table 1. It should be noted that in this case the IG unit has two glass plates of different thicknesses and that the difference between the internal (room) pressure and the external pressure is due to an added wind pressure. The final gas space temperature was fixed at −12.2°C (10°F), thus thermal calculations were not required. As shown in Table 1, the agreement between both sets of results is good, with a maximum difference of 2.7%.

The center-of-glass U-factors found in Table 5 of chapter 27 of *ASHRAE Fundamentals* (ASHRAE 1993) were used to validate the thermal portion of the solution methodology. For this verification, the mechanical portion of the solution procedure was deactivated. The results of this validation are presented in Table 2. Three different types of glazings were examined (the identification numbers correspond to those found in Table 5 of chapter 27 of the 1993 *ASHRAE Fundamentals*): a standard double glazing unit (no. 17), an air-filled triple glazing with a low-e film (no. 45), and an argon-filled triple glazing with a low-e film (no. 47). As shown in Table 2, the differences between the present results and those of

TABLE 1 Comparison Between the Results of the Present Study and Those of Solvason (1974)

$a =$ 0.762 m (width, 30 in.)
$b =$ 0.914 m (height, 36 in.)
$t_1 =$ 0.00476 m (3/16 in.)
$t_2 =$ 0.00635 m (1/4 in.)
$L =$ 0.0122 m (1/2 in.)
$P_{init} =$ 99.26 kPa (14.4 psia)
$P_{int} =$ 102.66 kPa (14.890 psia)
$P_{ext} =$ 102.84 kPa (14.915 psia)
$T_{init} =$ 21.1°C (70°F)
$T_{final} =$ −12.2°C (10°F)



	Solvason (1974)	Present Work	Diff.
P_{final}	101.705 kPa (14.751 psia)	101.736 kPa (14.756 psia)	0.03%
w_1	1.331 mm (0.0524 in.)	1.296 mm (0.051 in.)	2.6%
w_2	2.667 mm (0.105 in.)	2.594 mm (0.102 in.)	2.7%

ASHRAE are small and are probably due to differences in gas properties.

The results presented in Tables 1 and 2 indicate that the mechanical and thermal calculations developed for the present work compare favorabley with data found in the literature.

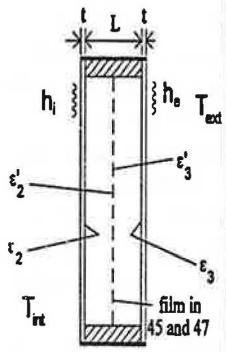
ASHRAE-Type Winter Conditions

In this section calculations are performed for ASHRAE winter conditions. The interior and exterior temperatures are 21°C (70°F) and −18°C (0°F), respectively, and the remaining operating conditions are presented in Table 3. A fixed window with a width of 1.2 m (4 ft) and a height of 1.8 m (6 ft) was selected as the base case. This corresponds to an NFRC BB-type window (ASHRAE 1993). Three different IG units were examined. These units correspond to IG units 17, 45, and 47 as described in Table 2.

A three-dimensional representation of glass plate deflection (not to scale) is presented in Figure 3 for IG units 17 and 45 (the deflection of type 47 is almost identical to the one obtained for type 45) and Table 3 presents the calculation results. First, as shown in Table 3, the Nusselt number was equal to unity in all three cases, indicating the presence of the conduction regime in the cavity. The maximum glass plate deflections are 0.61, 1.12, and 1.09 mm (0.024, 0.044, and 0.043 in.) for IG unit types 17, 45, and 47, respectively, while the mean deflections are 0.26, 0.48, and 0.46 mm (0.010, 0.019, and 0.018 in.), respectively. The percentage of gas space thickness reduction $(L_r - L)/L$ are 8.1%, 7.5%, and 7.3%, respectively. Finally, Table 3 presents the U-factors as calculated with and without deflections. As shown in this table, the difference between these two values ranges from 4.4% to 5.8%. Thus, glass plate curvature can have a relatively significant impact on the U-factors.

TABLE 2 Comparison Between the U-Factors (Without Deflections) Obtained by ASHRAE and Those Obtained in the Present Work

$a =$ 1.0 m (39.4 in.)
$b =$ 1.0 m (39.4 in.)
$t =$ 0.00318 m (1/8 in.)
(The film has a negligible thickness)
$h_i =$ 8.29 W/m²·°C (1.46 Btu/h·ft²·°F)
$h_e =$ 29 W/m²·°C (5.11 Btu/h ft²·°F)
$k_v =$ 0.917 W/m·K (6.35 Btu·in./h·ft²·°F)
$T_{int} =$ 21°C (70°F)
$T_{ext} =$ −18°C (0°F)



ASHRAE No. (Table 5, Chapter 27, 1993)	U-Factor ASHRAE (W/m²·°C) [Btu/h·ft²·°F]	U-Factor Present Work (W/m²·°C) [Btu/h·ft²·°F]	Diff.
DG-air No. 17, Double Glazing L = 6.4 mm (1/4 in.) $\varepsilon_2 = 0.1$, $\varepsilon_3 = 0.84$ gas = air	2.44 [0.430]	2.49 [0.439]	2.0%
TGLe-air No. 45, Triple Glazing (2 glass, 1 film) L = 12.7 mm (1/2 in.) (divided in 2 equal spaces) $\varepsilon_2 = 0.84$, $\varepsilon'_2 = 0.01$, $\varepsilon_3 = 0.1$, $\varepsilon'_3 = 0.84$ gas = argon	1.53 [0.270]	1.57 [0.277]	2.6%
TGLe-argon No. 47, Triple Glazing (2 glass, 1 film) L = 12.7 mm (1/2 in.) (divided in 2 equal spaces) $\varepsilon_2 = 0.84$, $\varepsilon'_2 = 0.01$, $\varepsilon_3 = 0.1$, $\varepsilon'_3 = 0.84$ gas = argon	1.19 [0.210]	1.21 [0.213]	1.5%

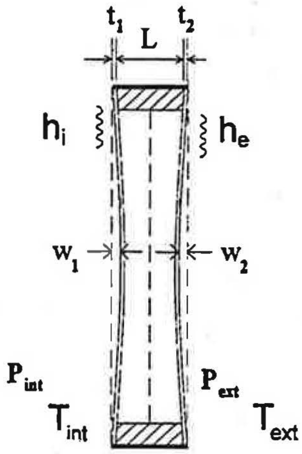
Effect of Exterior Temperature

Aside from IG unit 45, which has already been described, this section includes results for IG units 6 and 18. These units represent air-filled double-glazing units (L = 12.7 mm [1/2 in.]) with (no. 18) and without (no. 6) a low-emissivity surface.

Figure 4 shows the variation of the U-factor as a function of exterior temperature, T_{ext}. The corresponding variation of gas space thickness reduction as a function of T_{ext} (not shown in Figure 4) is linear and is approximately equal to 10% at −30°C (−22°F) and 0% at 20°C (68°F) for all three units. For IG units 6 and 18, glass plate deflection has a weak impact on the U-factor, with a maximum difference of 0.8% and 1.7%, respectively, as shown in Figure 4. In fact, the variation of the U-factor caused by air property variations

TABLE 3 Glass Plate Deflections and U-Factors for Three Types of IG Units

a =	1.2 m (4 ft)
b =	1.8 m (6 ft)
t_1 =	3.175 mm (1/8 in.)
t_2 =	3.175 mm (1/8 in.)
P_{init} =	101.3 kPa (14.69 psia)
P_{int} =	101.3 kPa (14.69 psia)
P_{ext} =	101.3 kPa (14.69 psia)
T_{init} =	21°C (70°F)
T_{int} =	21°C (70°F)
T_{ext}	−18°C (0°F)
h_i =	8.29 W/m²·°C (1.46 Btu/h·ft²·°F)
h_e =	29 W/m²·°C (5.11 Btu/h·ft²·°F)



ASHRAE No. (see Table 2)	$w_1 = w_2$ (mm)[in.]	$w_{mean,1} = w_{mean,2}$ (mm)[in.]	Gas Space Thickness Reduction (%) $(L_r - L)/L$	Nu	U-Factors (W/m²·°C) [Btu/h·ft²·°F] Without Deflection	With Deflection	Difference
17	0.61 [0.024]	0.26 [0.010]	8.1%	1.00	2.49 [0.439]	2.60 [0.459]	4.4%
45	1.12 [0.044]	0.48 [0.019]	7.5%	1.00	1.57 [0.277]	1.66 [0.293]	5.7%
47	1.09 [0.043]	0.46 [0.018]	7.3%	1.00	1.21 [0.213]	1.28 [0.226]	5.8%

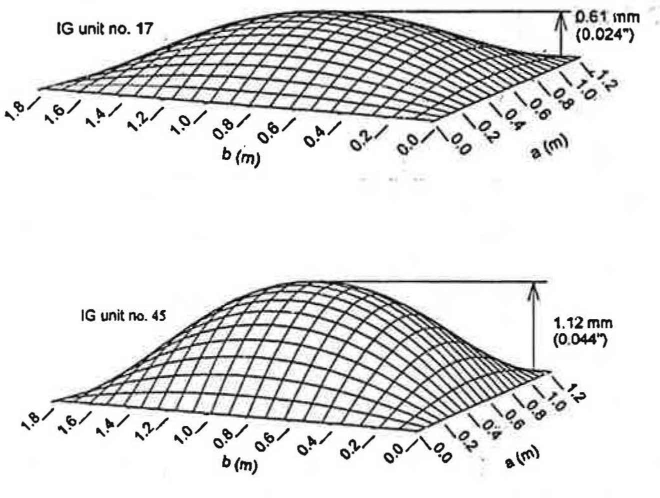


Figure 3 Three-dimensional representation of glass plate deflections for two types of IG units (not to scale).

with temperature is greater than the variation caused by glass plate curvature. For example, the decrease in the U-factor for IG unit 18 without deflections (dotted line) is approximately 10% from −30°C (−22°F) to 20°C (68°F), which is greater than the 0.8% and 1.7% values quoted above.

The results obtained with these two units should, however, be interpreted with caution. As shown in Figure 4, for low exterior temperatures the value of Nu is greater than 1, indicating the presence of a significant convection current inside the cavity. As was stated earlier, glass plate curvature will certainly affect the value of Nu and, consequently, the U-factor. Before any firm conclusions can be made for these IG units, more research is needed to ascertain the influence of glass plate deflection on the convection current inside a gas-filled cavity.

Nonetheless, it is worthwhile to examine why glass plate curvature does not significantly affect the U-factors for IG units 6 and 18. The reason is that at low temperatures the value of Ra decreases because L decreases due to plate curvature. With a reduction of Ra, the value of Nu as determined using the relationships of ElSherbiny et al. (1982a) will also decrease. As it turns out, the relative decrease in the value of Nu is similar to the relative decrease in the value of L. According to Equation 5, the resulting value of h_c does not change significantly and, consequently, the U-factor is not affected.

As shown in Figure 4, the variation of the U-factor for IG unit 45 (air-filled triple glazing with a low-e film) is different. First, the value of Nu was equal to one over the full range of exterior temperatures. Consequently, these results do not suffer from the uncertainty in the value of Nu experienced by units 6 and 18. Results show that for T_{ext} = −30°C (−22°F), there is a 6.7% difference in the values of the U-factor with and without deflections. It is also interesting to note that when glass plate deflections are considered (solid line in the top portion of Figure 4), the variation of the U-factor as a function of temperature is small—about 1.5% for −30°C (−22°F) < T_{ext} < 20°C (−68°F). This is because two effects that almost counterbalance are competing here. First, glass plate curvature tends to

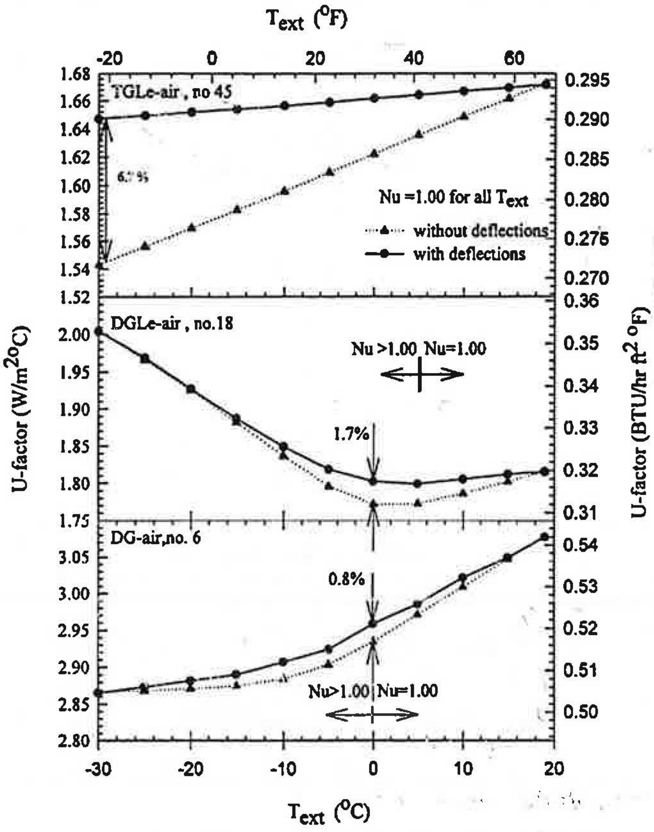


Figure 4 *Variation of the U-factor of exterior temperature (T_{ext}) for three types of IG units.*

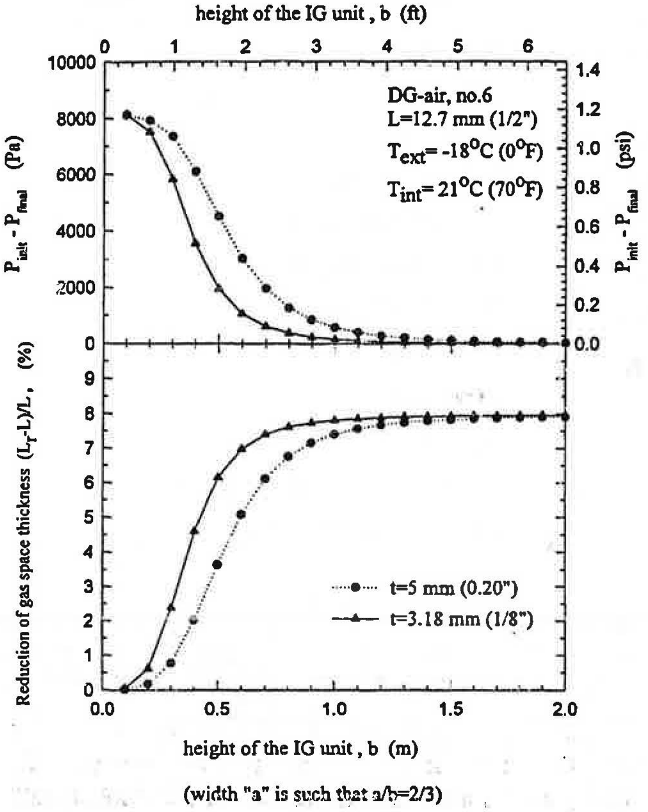


Figure 5 *Reduction in gas space thickness as a function of IG unit dimensions.*

increase U as T_{ext} is reduced. Second, the changes in air properties when T_{ext} is reduced tend to decrease U.

Effects of Different IG Unit Dimensions

The results presented so far were for a 1.2 × 1.8 m (4 × 6 ft) IG unit. It is interesting to look at the effects of various IG unit dimensions. Figure 5 shows the reduction of gas space thickness as a function of IG unit dimensions for two different glass plate thicknesses. A constant width/height ratio of 2/3 was selected and the other parameters are the same as those described in Table 2. The corresponding differences between the initial and final gas space pressures are shown in the top portion of the figure.

As expected, glass plate deflections are small for small IG units. For example, the reduction in gas space thickness is less than 1% for a 20 × 30 cm (8 × 12 in.) unit for both glass plate thicknesses. However, when b is increased to more than 30 cm (12 in.), glass plate deflection increases sharply and reaches a plateau at $b \sim 1.0$ m (3 ft), where the reduction in gas space thickness is approximately 8%. This may seem surprising considering that w_{mean} is a function of b^4 (Equation 2); one would think that the reduction in gap space thickness would continue to rise as b increases. This would be true for a constant value of P ($P_{init} - P_{final}$). However, P decreases as b increases, as shown in the top portion of the figure. The 8% reduction in gap space thickness observed for $b > 1.0$ m (3 ft) is simply the ratio T_{final}/T_{init}. In other words, for large values of b, a state of mechanical equilibrium is reached where $P_{final} \approx P_{init}$ and V_{final}/V_{init} (or L_f/L) is given by T_{final}/T_{init}.

Furthermore, Figure 5 shows that the ratio L_f/L is approximately independent of glass thickness (as long as both plates composing the IG unit are of the same thickness). These last observations are important, as they imply that the mechanical and thermal aspects can be decoupled, allowing one to evaluate T_{final} based on a gas space reduction of 8% and then proceed to evaluate the U-factor.

For smaller windows, the observation made in the preceding paragraph is not applicable. As indicated in Figure 5, for $b < 1.0$ m (3 ft) the reduction in gas space thickness is dependent on the glass plate thickness, with the thicker glass plates experiencing smaller glass plate deflections. For example, for $b = 0.5$ m (1.6 ft) the reduction in gas space thickness for $t = 5$ mm (0.20 in.) is approximately half the value calculated for $t = 3.18$ mm (1/8 in.).

It is noteworthy to mention that for a fixed IG unit width and height, glass plate deflection increases with increasing values of L. However, the reduction in gas space thickness remains essentially constant (Bourret et al 1995).

Effects of Varying Barometric Pressure

Mean glass plate deflections as a function of barometric pressure are presented in Figure 6 for two filling pressures. For this case, $T_{final} = T_{init}$, thus only pressure effects are considered. The range of barometric pressures represents the

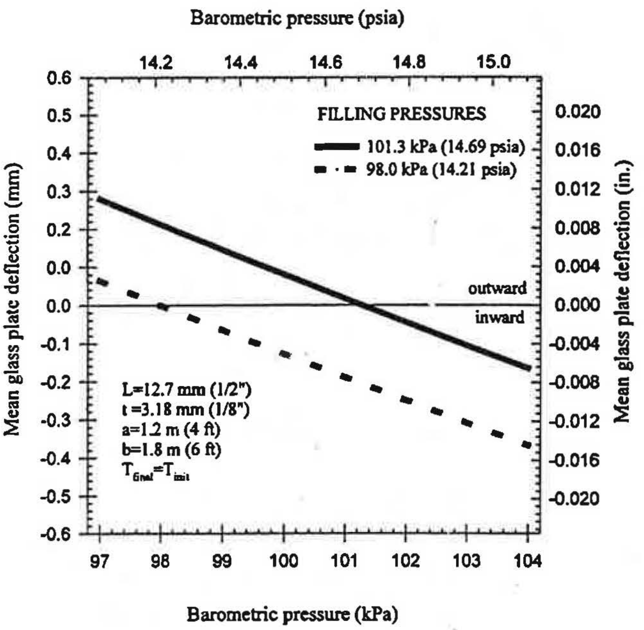


Figure 6 *Mean glass plate deflections as a function of barometric pressure for two filling pressures.*

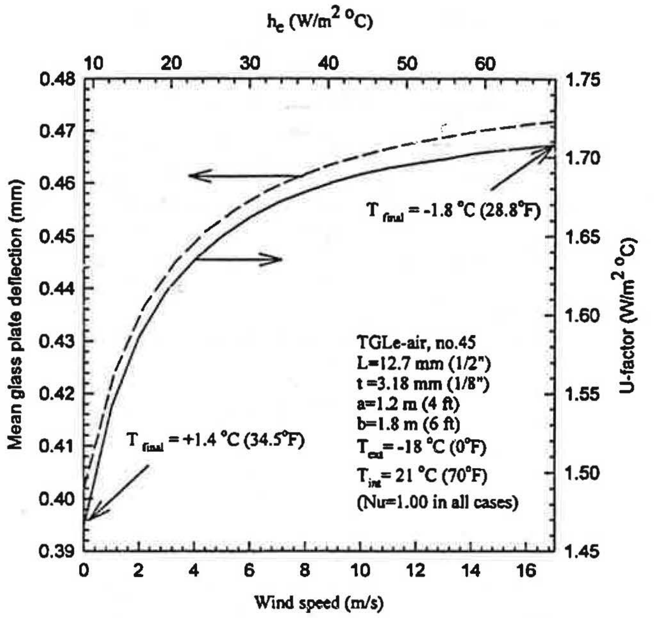


Figure 7 *The effects of wind speed on mean glass plate deflection and U-factor for IG unit type 45. (To obtain: Btu/h·ft²·°F, divide W/m²·°C by 5.678; ft/s, multiply m/s by 3.28; inches, divide mm by 25.4.)*

actual range that can be expected for a typical year in Montréal. In the worst case, with a filling pressure of 98.0 kPa (14.21 psia) and a barometric pressure of 104.0 kPa (15.08 psia), the mean glass plate deflection (inward) is close to 0.37 mm (0.014 in.). This value is similar to the mean glass plate deflection caused by temperature as reported in Table 3 for IG units 45 and 47.

Effects of Varying Wind Speed

Wind speed affects the value of the outside surface heat transfer coefficient and, consequently, the value of the U-factor. The effects of wind speed are presented in Figure 7, where mean glass plate deflection and U-factor variations have been plotted as a function of wind speed (bottom axis) and the corresponding outside film coefficient, h_e (top axis). The value of h_e has been calculated according to Equation 9. As shown in Figure 7, the effects of wind speed on the U-factor are significant, as the U-factor increases by approximately 17% for a wind speed increase from 0 to 17 m/s (56 ft/s). Most of this increase in the U-factor is due to the increase of h_e, which directly affects the value of U (Equation 8). In addition, an increase of h_e decreases the value of T_{final}. As shown in Figure 7, T_{final} equals 1.4°C (34.5°F) and −1.8°C (28.8°F) for wind speeds of 0 and 17 m/s (56 ft/s), respectively. This decrease in the value of T_{final} increases mean glass plate deflection from about 0.40 to 0.47 mm, as shown in Figure 7. However, the impact of this deflection on the U-factor is minimum because, as was stated earlier in relation to Figure 4, glass plate deflection and changes in air properties tend to counterbalance each other.

Yearly Variation of the U-Factor

So far, this paper has examined the individual effects of T_{ext}, P_{baro}, and wind speed. In this last series of results, which are presented in Figures 8 and 9, yearly simulations are performed for Montréal (Figure 8) and Toulouse (Figure 9) to examine the combined effects of these three factors. Typical meteorological year (TMY) weather files were used to obtain hourly values of T_{ext}, P_{baro}, and wind speed. To reduce the number of data presented in Figure 8, simulations were performed every 10 hours starting with hour number 1 on January 1. Thus, only 876 data points are presented in Figure 8. For Toulouse, only 730 data points (two per day) are presented. A triple-glazed, air-filled, low-e unit (IG unit 45) was selected for these simulations. In each case Nu was equal to unity, indicating the presence of the conduction regime.

As shown in Figures 8 and 9, the yearly averaged values of the U-factors for Montréal and Toulouse are 1.63 W/m²·°C (0.287 Btu/h·ft²·°F) and 1.62 W/m²·°C (0.285 Btu/h·ft²·°F), respectively. This close agreement seems to indicate that the yearly averaged values of the U-factor are almost independent of the climate. The U-factor varies widely during the year, with values 5% above and 10% below the average value for both cities. It can also be seen that the spread of the data above and below the average can be considered to be time independent, i.e., summer variations of the U-factor above and below the average are almost identical to the winter variations.

The U-factor reported by ASHRAE and presented in Figures 8 and 9 is for winter conditions. This value may not be adequate for heating load calculations, as it does not represent a worst case value. In fact, as shown in Figures 8 and 9, the

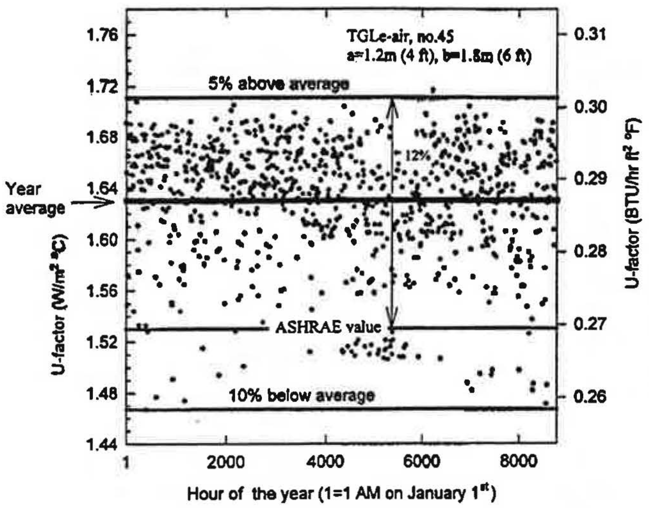


Figure 8 *Yearly variation of the U-factor for Montréal (Canada) for simultaneously varying* T_{ext}, P_{baro}, *and wind speed. The initial filling pressure and temperature are 101.3 kPa (14.69 psia) and 21°C (70°F), respectively.*

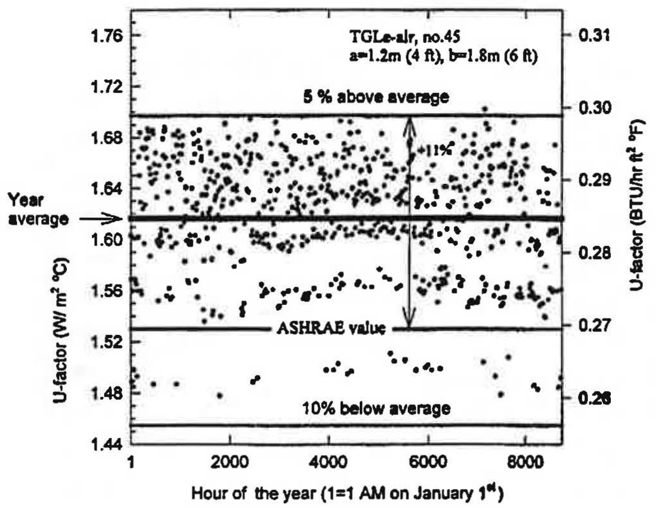


Figure 9 *Yearly variation of the U-factor for Toulouse (France) for simultaneously varying* T_{ext}, P_{baro}, *and wind speed. The initial filling pressure and temperature are 101.3 kPa (14.69 psia) and 21°C (70°F), respectively.*

ASHRAE value underestimates the worst case (5% above average) by 11% and 12% for Toulouse and Montréal, respectively.

CONCLUSION

This study has examined the effects of glass plate curvature on the U-factor of insulated glazing units. This was accomplished by solving the governing coupled mechanical and thermal equations. It was shown that glass plate curvature can have a relatively significant impact on the value of the U-factor. For example, for ASHRAE's typical winter conditions (T_{ext} = −18°C [0°F]), gas space thickness reduction can reach 7.3% with a corresponding decrease of 5.8% in the U-factor for a triple-glazed, air-filled, low-e unit with a total gap spacing of 12.7 mm (1/2 in.) (IG unit 45 in chapter 27 of *ASHRAE Fundamentals* [1993]). The effects of simultaneous variations of exterior temperature, barometric pressure, and wind speed were also examined for Montréal and Toulouse for a year and for IG unit 45. These last results indicate that the U-factor may vary up to 5% above and 10% below the yearly average. Furthermore, the ASHRAE-tabulated values may underestimate heating load calculations by as much as 12% when glass plate deflection is not taken into account. This study concentrated on IG units with fairly small gap spacing operating in the conduction regime. More research is needed to ascertain the influence of glass plate curvature on the convection current occurring inside gas-filled cavities with larger gap spacing.

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