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HEAT FLOW ACROSS THE AIR GAP BETWEEN TWO VERTICAL GLAZING GLASSES

Ling Man Tsang
Fisica Laboratorio, TIS, Block K, Macau University of Science and Technology, Taipa, Macau

ABSTRACT

The silicone sealant bite depth for any double- glass panel is obtained by complex stress analysis. We obtain the optimum air space between the outer and the inner layer of glass by considering the minimum heat transfer across the small air space. Such an air space ranges from h/133 to h/60. Radiation is not to be discussed here.

Keywords: Air gap determination, heat across a small gap.

INTRODUCTION

Whenever minimum heat transfer between indoor and outdoor is concerned, double glazing is a solution. In countries of extreme weathers, many glass curtain walls are double glazed with a small air gap between the outer and inner glass. Sometimes, the air space is filled up with inert gas such as argon in order to avoid moisture forming inside the enclosure. This paper consists of three topics:

1) Sealant bite depth in double glazing(bite is a term in glazing trade which refers to the cavity to be filled up with sealant)
2) Air gap between the two equal lites of vertical glass
3) Heat transfer across the air gap between two vertical glasses

The first two topics concern the structural aspects while the third topic is on heat transfer across the air gap. Generally, the content, discussion and presentation of the entire paper may not be a serious review of previous researches but rather a practical application. For instance, entropy changes across the air gap(space) [Honig 2011] are not included in this paper. Moreover, within such a small gap, we also neglect the velocity and temperature profiles as well as the skin friction[Nagendra-Tirunarayanan 1971, Singh and Paul 2006] Nevertheless, we provide very practical formulae for glass curtain wall designers and engineers.

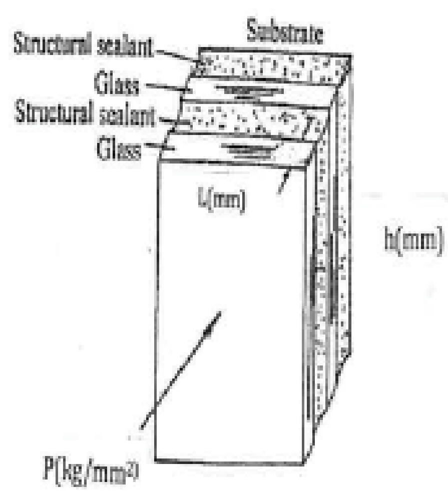
1. Sealant bite depth in double glazing

The whole double-glass panel is fabricated in the factory and then delivered to the construction site to be attached onto the aluminum frame vertically. The whole panel is mainly supported by structural sealant. This sealant is subjected to two forces, namely, the tensile one due to the external pressure acting perpendicularly to the panel plus the shearing one due to the weight of the panel. Failure of sealant will cause the whole panel falling onto the street and hence, selection of structural sealant is very important. From figure 1,

σ = tensile strength acting on unit length of sealant due to external pressure p (kg/mm^2)
τ= shear stress acting on unit length of sealant due to the weight of the two equal lites of glass
ρ = glass density
d = sealant bite depth (mm).

The resultant tensile stress can be written as:

$$\sigma_R = \frac{\sigma}{2} + \sqrt{\frac{\sigma^2}{2} + \tau^2} = \frac{phL}{2d(2h+2L)} + \sqrt{\left(\frac{phL}{2d(2h+2L)}\right)^2 + \left(\frac{2\rho thL}{d(2h+2L)}\right)^2} \quad (1)$$



Glazing (Double)

Fig. 1. Double Glazing (Double Glasses)

*Corresponding author email: lmtsang05@hotmail.com

As acceptable by all sealant manufacturers such as Dow Corning, Tremco, GE and etc, the maximum allowable tensile stress for structural sealant is $14x10^{-3} kg/mm^2$ (or 20 psi).

Hence, (1) can be rewritten as

$$14 \times 10^{-3} = \frac{phL}{4d(h+L)} + \frac{hL}{4d(h+L)}\sqrt{p^2+(4\rho t)^2} \quad (2)$$

The sealant bite depth is:

$$d = \frac{hL}{56 \times 10^{-3}(h+L)}\left[p + \sqrt{p^2+(4\rho t)^2}\right] \quad (3)$$

DISCUSSION

i) Since both σ and τ are not vectors, therefore, the resultant tensile stress $14x10^{-3} \neq (\sigma^2+\tau^2)^{1/2}$
ii) The resultant tensile stress can only be obtained from complex stress analysis or the Mohr's circle having the result of the form (1).
iii) For single glazing, the term $4\rho t$ can be neglected since $p > 4\rho t$

Eq (2) can then be reduced to, by setting $L \approx h$ where h is the longest length:

$$14x10^{-3} = ph/2d \quad (4)$$

which is the conventional formula adopted by the glazing industry for both two-side and 4-side single structural glazing.

2. Air gap between two equal lites of glass

As a common practice in double glazing, both glasses are identically of the same size. In order to find the optimum air gap δ (mm) between them, we shall start from the famous thin plate equation for the outer glass since it is directly under wind pressure. The inner glass is assumed to be firmly attached onto the frame

$$D\left(\frac{\partial^4 w}{\partial x^4} + \frac{2\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4}\right) = p \quad (5)$$

For 4-side simple support of the outer glass(Fig. 2), the solution of the above is

$$w = \sum_{n=1}^{\infty}\sum_{m=1}^{\infty} C_{mn} \sin\frac{m\pi x}{h} \sin\frac{n\pi y}{L} \quad (6)$$

where

w = deflection of the outer glass (mm) towards the inner glass,

$$C_{mn} = \frac{16p}{D\pi^6 mn}\left(\frac{h^2L^2}{m^2L^2+n^2h^2}\right)^2$$

P = wind pressure

D = plate rigidity= $\frac{Et^3}{12(1-\nu^2)} kg..mm$

E =Young's modulus of glass = $10^4 kg/mm^2$

t = glass thickness(mm),

ν = glass Poisson's ratio = 0.25

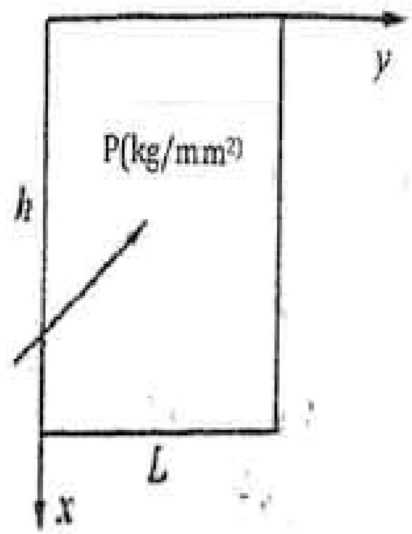


Fig. 2. 4 side simple supported glass lite subject to pressure p

For simple mode where m = n = 1, the maximum deflection at the centre x = h/2,
y = L/2 will become

$$\delta > w = \frac{180ph^4}{10^4 t^3 \pi^6}, where\ h > L.$$

Here we set L = h. (7)

Of course, the designed δ must be larger than the deflection w

where D = $\frac{10^4 t^3}{11.25}$, $C_{11} = \frac{16P \times 11.25}{10^4 t^3 \pi^6} \left(\frac{h^2L^2}{h^2+L^2}\right)^2$

Eq. (7) constrains the air gap between the two glasses so as to avoid glasses contact as well as the formation of Newton rings (Fig. 3). The latter is an aesthetic defect. Therefore, the designed δ in (7) must be larger than the deflection w.

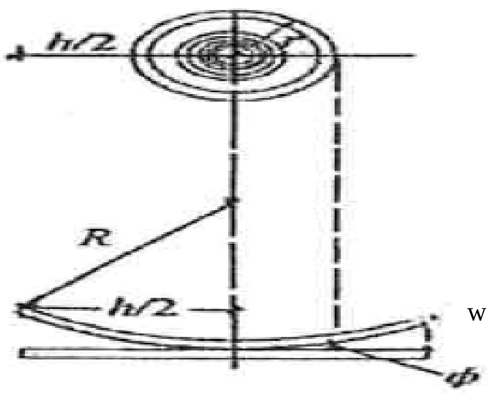


Fig. 3. Formation of Newton rings due to convex glass and plane glass contact.

If the fringes are produced in the air gap between a convex surface glass and a plane glass, the contour lines will be circular. The ring-shaped fringes thus produced were studied in detail by Newton. The value ϕ for any ring of radius r is the sagitta of the arc given by

$$\varphi = \frac{r^2}{2R} \tag{8}$$

The largest ϕ, of course, is

$$\phi = w = \frac{(h/2)^2}{2R}$$

Normally, rings or fringes concentrate themselves in the vicinity of the contact point. The above is just a reminder of how Newton rings are formed once the outer glass bends onto the inner glass.

Example

$p = 3$kPa = 300 $\times 10^{-6}$ kg/mm^2 ,
$h = 1400\ mm, t = 18\ mm$

By applying (7), the allowable air gap should not be less than 4mm, i.e.

$$\delta > w = 4mm$$

3 Heat Transfer across the air gap of two vertical glasses

Such a topic had been discussed extensively by ASHRAE and Jim Plavecsky (Ashrae, 1993; Plavecsky, 1991). Each tested insulated glass unit is dual pane of 900 mm x 1200 mm (height) x 3mm (thick). The environmental conditions are selected to conform to the Ashrae (1993). Winter design conditions are of 21°C interior and –17.8°C exterior and no solar radiation. It must be noted that in the consideration of heat flow through the windows, both convection and radiation need to be taken into account . However, no radiation heat transfer is required to find the optimum air gap δ in these tests [for radiation, see Rubin, 1982]. In fact, radiation can be reduced by suitable selection of glass emissivity. The window specimen to be tested is mounted onto a well-insulated test panel which separates a cold environment from a warm environment. Both temperature and rates of air flow is accurately controlled. Heat transfer occurs through the window from the warm side to the cold side. Fig.4 shows the profile of the overall U values being measured along the centre of the dual glass pane. Similar curves can also be obtained at different heights of the specimen. Ostrach (1972) showed that there is a linear relationship between h and δ.

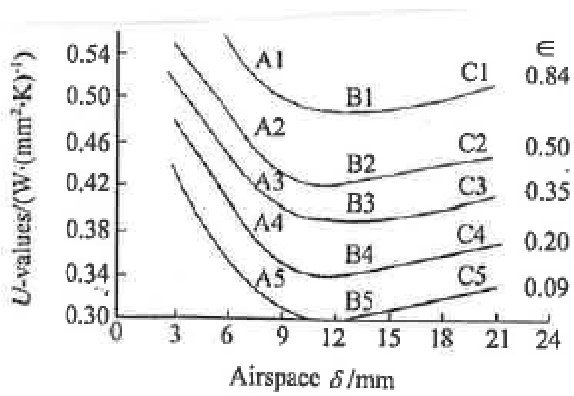


Fig. 4. U-values as a function of emissivity ε and air space δ.

DISCUSSION

i) Those curves from A's to B's are mainly convection governed by the heat transfer equation

$$q = \frac{\lambda_e \Delta T}{\delta}$$

where

q = heat across the airgap(kJ/($m^2 \cdot h$)),
ΔT = temperature gradient across the air gap ,
λ= heat transfer coefficient ($kJ / m.h.^oC$),
λ_e= heat transfer coefficient equivalent ($kJ / m.h.^oC$)
G = Grashof number.

The followings can be defined as [Ostrach, 1972]

$$\frac{\lambda_e}{\lambda} = 0.18G^{\frac{1}{4}}\left(\frac{\delta}{h}\right)^{\frac{1}{9}} \tag{9}$$

G = 2 $\times 10^4 \sim$ 2 $\times 10^5$ for laminar flow

$$\frac{\lambda_e}{\lambda} = 0.065\ G^{\frac{1}{3}} \left(\frac{\delta}{h}\right)^{\frac{1}{9}} \tag{10}$$

$G = 2x10^5 \, to \, 2x10^7$ for turbulence

ii) Those from B's to C's are mainly conduction governed by the same heat transfer equation as above but

$$\frac{\lambda_e}{\lambda} = 1 \tag{11}$$

G<2 $\times 10^3$(reasonable choice)

iii) Those in the vicinity of B's, say δ= 9mm to 20mm can be treated as transition from convection to conduction. Upon equating (9) and (11), we have

$$\left(\frac{h}{\delta}\right)^{\frac{1}{9}} = 0.18\ G^{\frac{1}{4}}\ for\ \lambda_e = \lambda,$$

$$for\ \delta = 9mm\ to\ 20mm$$

$$G = 5.87 \times 10^3\ to\ 8.38 \times 10^3$$

$$\frac{h}{\delta} = 60\ to\ 133 \tag{12}$$

Example

Taking the previous example of p = 300 × 10^{-6} kg/mm^2, L= 1400mm, t = 18m, h = 2300m, we obtain $\delta > w$= 4mm from (7) and δ = 16mm to 36mm from (12).

The value of δ can be chosen from 16mm to 36mm to achieve low U-value as well as to avoid glasses contact. For the selection of spacer, most manufacturers produce spacer width of δ= 18mm or 25mm. At least 4 pieces of spacer, placing along the sandwich edges, are used to keep the air gap distance constant. Structural silicone sealant can fill up the cavity along the perimeter without penetrating into the enclosure. The space between each spacer and the rim of the double glass panel is called the bite of the sealant.

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