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### **Engineering Structures**



# Point fixings in annealed and tempered glass structures: Modeling and optimization of bolted connections

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#### ABSTRACT

In the design of high load bearing elements made of tempered flat glass, connections cannot be avoided when large spans or high stiffness beams are considered. This paper investigates bolted connections in glass structures; the main objective is to determine the optimal joint. This work is performed through the determination of stress states due to both thermal tempering and in-plane loading. The modeling of the thermal tempering is performed with the FE software Abaqus and additional user subroutines. Experiments on industrial tempering line with specific set-up allow the determination of the air flow in the hole and then of the forced convection coefficients. The radiative heat transfer is also modeled numerically and the semi-transparency of glass in the near infrared is considered. In order to calculate residual stresses, the visco-elasticity of glass and the structural relaxation phenomena are taken into account. The computed stresses are checked against photo-elastic measurements. As various holes are considered, this study allows to determine the hole geometry for which the tempering process is the most effective. For the study of the consequences of in-plane loading, a large experimental campaign has been performed. The studied connection is derived from countersunk supports. The influences of different parameters as the hole geometry, the nature of the washer between glass and metallic connector as well as the glass-washer material friction coefficient were investigated. The modeling of these tests is performed with the FE software Abaqus. This modeling takes into account the ductility of the materials, the friction and the clearances between the parts. This modeling is validated thanks to failure stress measurements. The combination of modeling and experiments leads to identify optimal connection.

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#### 1. Introduction

Glass is a material that has been used for a long time in windows as a filling material and has much to offer in this regard due to its possibility to carry high compressive stresses.

For several years, there has been a trend in architecture to use glass not only as a part of the building envelope, but also as material for load bearing elements. This represents a special challenge because of the glass brittleness. In the most frequent cases, glass columns or beams are used. In the design of such structures made of tempered flat glass, connections cannot be avoided, especially when large spans or high stiffness beams are considered. The key differentiation in point bearing is done between glass panes fixed on their corner or edges and those fixed in drilled hole [1]. In the second case, loads are transferred via compound point-support or

\* Corresponding author. E-mail address: Fabrice.Bernard@insa-rennes.fr (F. Bernard). steel bolts to the glass hole. To avoid any contact between steel and glass a suitable layer or bushing material, such as aluminum or plastic, has to be applied.

Point-bearing in holes are also divided in types with a plate on each surface of the glass pane (raised head point fixture) and those with conical drillings (countersunk point fixture).

The use of point fixings with conical holes is interesting for several reasons. From an architectural point of view, the even glass surface is not disturbed by an additional plate and from a point of view of maintenance, an even surface is more convenient to clean.

A lot of structures all around the world present such connections; one of the main examples in France is the large facade of the "Cité des Sciences et de l'Industrie" at Paris [2]. Fig. 1 presents a schematic view of the used connection [2].

The two only ways for the design of a glass plate with such point-bearing is by means of 3D FE modeling and of extensive experiments in scale 1:1. For the FE modeling, the point-bearing itself as well as the surrounding area have to be modeled accurately to get close-to-reality results.



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Fig. 1. Schematic cross section of the studied connection.



Fig. 2a. Cross section of cylindrical holes a1 and a2.



Fig. 2b. Cross section of conical holes b1 and b2.

#### 2. Objectives of the study

The aim of this paper is to propose a complete modeling of such connections. The technology used is derived from the countersunk point fixture and from the one shown in Fig. 1.

Contrary to facades where loads (wind especially) are out of the plane, glass structures carry in-plane loadings. That is why only symmetrical geometries of holes in glass plates are considered in this work.

Another advantage of an accurate 3D FE modeling is the possible determination of the optimal connection. Then, particularly five different holes geometries are considered in this work. All of



Fig. 2c. Cross section of mean chamfer hole c1.

Table 1

The five different studied	geometries.
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Designation	$\Phi_{int}$ (mm)	$\Phi_{ext}$ (mm)
a1	38	40
a2	54	56
b1	24	40
b2	40	56
c1	30	40

them are presented in Figs. 2a–2c and Table 1. 19 mm thick Planilux<sup>®</sup>glass plates, produced by Saint-Gobain, are studied.

Besides, thermally tempered glass can also be required for some structures. Then the complete modeling and optimization of this kind of connections needs the determination of stress states due to both thermal tempering and metallic connector in-plane loadings. The present paper focuses on these two separated studies. The problem is solved in a numerical way and validated by means of experimental measurements.

A recent study [3] has concerned also numerical and experimental investigations on the stress distribution of bolted connections under In-Plane loads but, contrary to the present paper, these examinations are limited to drill holes with a cylindrical shape and to annealed glass. Some details also, especially on the connector and the interlayer shapes are different.

#### 3. FE computation of residual stresses in connection area

#### 3.1. Presentation of the thermo-mechanical computation

Previous analyses of glass tempering have been concerned with the calculation of residual stresses in infinite plates by means of a 1D modeling [4]. The computation of residual stresses in the vicinity of a straight edge (2D modeling) was carried out in [5] [5bis] and near holes in [6] or [7], but these previous analyses did not take into account, in an exhaustive way, the heat transfers occurring during the tempering process.

The presented contribution concerns the prediction of residual stresses, not only close to straight edges, but also in the vicinity of chamfered holes in 19 mm thick glass plates (3D modeling). A thermo-mechanical calculation is carried out with the Finite Element Method (FEM). Knowing both the mechanical behavior of glass and the temperature history in the whole plate during the tempering process, is then necessary.

The thermo-mechanical behavior of glass was widely studied in the literature. Narayanaswamy [4] proposed a model that includes both structural and viscous relaxation phenomena, and that considers glass as a thermorheologically simple material. The implementation of this model in the FE software Abaqus is described in [8]. The parameters of the model are provided in [9].

#### 3.2. Identification of heat transfers

The temperature history in the whole plate during the tempering process can be estimated while taking into account accurately the whole heat transfers. The analysis of the heat

The identified forced convection coefficients (in W/m<sup>2</sup>K).

Locations		First coefficient (first 220 s. of the process)	Second coefficient
Glass pane surface	Far away from hole (point 1 in Figs. 3a and 3b)	77	96
	Near to hole (point 2 in Figs. 3a and 3b)	72	75
Edge	(Point 3 in Figs. 3a and 3b)	62	72
Conical zones of holes	Geometry b1 (point 4 in Fig. 3a)	74	120
	Geometry b2 (point 4 in Fig. 3a)	78	132
	Geometry c1 (point 4 in Fig. 3a)	75	124
Cylindrical zones	Geometry a1 (point 5 in Fig. 3b)	58	68
	Geometry a2 (point 5 in Fig. 3b)	61	70



Fig. 3a. Locations of the identified convection coefficients for holes b1, b2 and c1.



Fig. 3b. Locations of the identified convection coefficients for holes a1 and a2.

equation reveals that heat transfers are of three types:

– First the thermal conduction appears when a temperature gradient exists inside the material. The conductive flux  $\Phi_{cond}$  (in W/m<sup>2</sup>) is function of the thermal gradient thanks to the thermal conductivity  $\lambda$  (in W/m/K) :

$$\Phi_{cond} = -\lambda(T) \operatorname{grad} T \tag{1}$$

For glass it has been found that  $\lambda$  varies linearly with temperature (*T* in K) as follows:

$$\lambda(T) = 0.975 + 8.5810^{-4}(T - 273) \tag{2}$$

- The cooling by air casts is modeled by a forced convection. The convective flux is given by:

$$\Phi_{conv} = h \left( T_{ext} - T_s \right) \tag{3}$$

h,  $T_{ext}$  and  $T_s$  denote, respectively, the convection coefficient, the exterior temperature and the glass surface temperature.

 In addition, because of the high temperature at the beginning of the tempering process, the modeling of the thermal radiation is necessary. Radiation is a complex phenomenon in glass which is a semi-transparent medium [10].

The convection coefficients in the different area of perforated plates (far away from edges, in the hole, on the straight edges ...) are identified using a hollow aluminum model representative of the external surfaces of a  $400 \times 400 \times 19 \text{ mm}^3$  holed glass plate. Each aluminum element (on the edge, on the plate surface, different faces of the hole) is isolated from the others thanks to PTFE washers. All of them are instrumented with thermocouples distributed everywhere on the perforated plate. The model is then submitted to real conditions of tempering (Securipoint <sup>®</sup> tempering on an industrial line of the Saint-Gobain Company) but heated to a temperature such as radiation is negligible. The temperature is recorded during the cooling thanks to thermocouples. The actual forced convection coefficients are identified with the resolution of the heat equation.

Table 2 gives the main values of the identified convection coefficients on the various area of the different plates (Figs. 3a and 3b) and for a Securipoint <sup>®</sup> tempering. Since two cooling steps are present in the industrial process, two values of the convection coefficients are given. For the cylindrical parts of holes b1 and b2, the coefficients obtained, respectively, for holes a1 and a2 are

considered. For the cylindrical part of hole c1, an intermediate value is used (first coefficient: 59  $W/m^2$  K; second coefficient: 69  $W/m^2$  K). Similarly, for the conical part of holes a1 and a2, the coefficients, respectively, obtained for holes b1 and b2 are taken into account.

All the values proved that air flows are more effective when conical surface are large.

For the modeling of thermal radiation in a semi-transparent medium like glass, a simplified model is used [8]. It consists in splitting the radiative flux in two fluxes which emanate from surfaces on one hand ( $\Phi_s$ ), and from the volume on the other hand ( $\Phi_v$ ). Thus, surface and volume emissivities of glass plates are defined in the following way:

- the surface emissivity ( $\varepsilon_{surf}$ ) is defined for the spectral field where glass is opaque,
- the volume emissivity ( $\varepsilon_{vol}$ ) is defined for the spectral field where glass is semi-transparent.

It is assumed that radiative transfers take place in a uniform way in all the volume. The fluxes are:

$$\Phi_s = 2\sigma[\varepsilon_{surf}(T_s)T_s^4 - \varepsilon_{surf}(T_{ext})T_{ext}^4]$$
(4)

and

$$\Phi_{v} = 2\sigma[\varepsilon_{vol}(t, T_{v})T_{v}^{4} - \varepsilon_{vol}(t, T_{ext})T_{ext}^{4}]$$
(5)

 $T_s$  is the surface temperature,  $T_{ext}$  the environment temperature,  $T_v$  is the mean temperature in the thickness (t) of the glass plate and  $\sigma$  the Stefan–Boltzmann coefficient. For the surface radiative flux, the factor 2 represents the heat exchanges of the two faces of the plate. For the volume radiative flux, the factor 2 is due to the heat exchanges with the two semi-spaces above and under the plate. It is then assumed that each point of the volume only exchanges radiative energy with outside, and not with the neighboring points within the volume. The surface and volume emissivities were numerically identified from experiments, they must be considered as apparent emissivities of flat glass [10]. The validity of this simplified model, especially near edges and holes, has been checked [8].

#### 3.3. Validation of the modeling and calculation of the residual stresses

Then the various heat transfers are identified, that allows an accurate prediction of residual stresses due to the thermal tempering of thick glass plate. The mesh used in this part of the study is axisymmetrical. A sensitivity analysis on the mesh fineness has been performed and has led to the optimal mesh visible in Fig. 4b. The mesh is regular and each finite element in the hole has an edge of 0.5 mm. The comparison between the residual stresses in perforated plates calculated by a FEM simulation and those obtained by means of photoelastic methods is very satisfactory particularly close to the edge and the hole, that is an important originality of this study [8].

Such a validation allows now the prediction of the optimal geometry for thermal tempering. The comparisons between the five studied geometry are performed according to four main criteria:

- the surface tangential stresses  $\sigma_{33}$  in the hole (see also Fig. 4a);

 Table 3

 Residual stresses calculation. Summary table of the predicted results.

Туре	Surface stre	ess $\sigma_{33}$ (MPa)		Membrane stress (for $x_1 = 0$ ) (MPa)	Neutral line position x <sub>1</sub> (mm)	Compression thickness (mm)
	Min.	Average	Max.			
a1	-106.1	-123	-149.8	-119.6	5.38	3.00
a2	-114.6	-127.7	-149.7	-125.1	6.08	3.35
b1	-133.1	-143.8	-157.6	-155.2	8.94	3.70
b2	-136.6	-147.9	-157.7	-155.6	9.49	4.00
c1	-119.2	-133.1	-142.4	-131.4	6.50	3.64



Fig. 4a. Coordinates definition.



**Fig. 4b.** Results of the FE modeling of the thermal tempering process. Definition of the compression thickness.

- the membrane stresses ( $\Sigma$ ) in the vicinity of the hole:  $\Sigma = \frac{1}{t} \int_{-t/2}^{t/2} (\sigma_{33}(x_2) \sigma_{11}(x_2)) dx_2$  (*t* is the thickness of the plate and  $x_2$  is the thickness coordinate, see Fig. 4a);
- the neutral line (where the membrane stresses are equal to zero): beyond it, a zone of integrated tension occurs, that may weaken the hole;
- the compression thickness in the vicinity of the hole: minimal distance between surface and core in extension (see Fig. 4b).

These comparisons are presented in Table 3. They show that among the five studied geometries, hole b2 is the most effective for thermal tempering reinforcement.

#### 3.4. Extrapolations to other geometries

Even if the convection coefficients are identified on five specific geometries, some extrapolations can be done. A sensitivity analysis on the length of the conical chamfer can then be performed. In this numerical study, the exterior diameter is kept constant whereas the interior one varies from 37 mm (no cylindrical part) to 52 mm (hole a2).

On Fig. 5, the tangential stress  $\sigma_{xx}$  along the hole is presented. On holes without cylindrical part, this stress is equal to zero on the symmetry plane (for s = 0 see Fig. 5). That represents a weakness which is not acceptable for structural applications.

The importance of a small cylindrical part is then put into evidence. An optimal geometry for thermal tempering can then be drawn: the cylindrical part has to represent about a quarter of the thickness.

# 4. In-plane loading of metallic connector: Experiments and FE modeling

The analysis of stresses in the vicinity of a chamfered hole in a glass plate loaded by a dowel-type connection is now presented. This analysis is both experimental and numerical and leads to the determination of the optimal point fixing.

#### 4.1. Description of the studied bolted connection

The metallic connector (shown in Fig. 6), derived from countersunk point fixture technology, is especially designed for this type of application (in plane loading via symmetrical holes).

In connection area, high local stress occurs at the edge of the hole. In steel construction local stress-peaks can be reduced by local plastification due to the plastic material behavior. Then, because of the brittle nature of glass, bolts have to be used with a ductile interface. Usually a soft metal cup is placed between the steel connector and the glass. In this study an aluminum ring is considered. It distributes uniformly the load into the glass over the contact area and enables to avoid a localized loading. Figs. 7 and 8



**Fig. 5.** Residual stresses modeling. Tangential stresses  $\sigma_{xx}$  along the hole chamfer (*s* corresponds to the curvilinear abscissa inside the hole including the cylindrical part). Then the junction between the cylindrical and the conical parts of the hole corresponds to the first local minimum (in absolute value). For example this junction corresponds to *s* = 7.5 mm for the hole with  $\phi_{ext}$  = 56 mm and  $\phi_{int}$  = 52 mm and *s* = 0 mm for the hole with  $\phi_{ext}$  = 56 mm and  $\phi_{int}$  = 37 mm (no cylindrical part). The second local minimum (in absolute value too) corresponds to the end of the hole (junction between hole and current zone) i.e. *s* = 10.33 mm for the hole with  $\phi_{ext}$  = 56 mm and  $\phi_{int}$  = 37 mm.



Fig. 6. Metallic connector used in this study.



Fig. 7. Aluminum washer for holes b and c.



Fig. 8. Aluminum washer for holes a.

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Main dimensions of the interlayer for holes b and c.	
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Туре	b1	b2	c1
d (mm)	36	50	36
D(mm)	40	56	40
<i>c</i> (mm)	8	8	5

present schematic cuts of this washer for, respectively, holes a and b/c configurations. Tables 4 and 5 give the main dimensions.

Besides, following some empirical and practical considerations, there is a difference of 1° between the relative inclination of the connector and the rings. A gap is generated between the two elements; this gap enables to avoid a too high stress concentration at the bottom of the chamfer and in the cylindrical part of the hole.

#### Table 5

Main dimensions of the interlayer for holes a.

Туре	a1	a2
A (mm) d (mm) D (mm)	32 37.9 40	48 53.9 56



Fig. 9a. Sketch of the test set-up.



Fig. 9b. Photo of the experimental campaign.

#### 4.2. Description of the experimental campaign

The used MTS testing machine has a loading capacity of 500 kN. Different  $350 \times 600 \text{ mm}^2$  glass plates are tested. The five hole geometries (Table 1) are considered. The perforated glass plate is glued to two metallic flasks that can rotate with the frame of the testing machine. The metallic connector is fixed to the horizontal cross-piece of the testing machine. The vertical upwards displacement rate is 0.5 mm/min (Figs. 9a and 9b).

More than 120 samples have been tested. The whole results are presented in Table 6 (experiments performed on annealed glass) and Table 7 (experiments on tempered glass).

Special attention is also paid on the initial torque applied to the bolt. Various values have been used and investigated. This initial prestressing is applied thanks to a torque wrench.

The deviation is quite low although glass is sensitive to surface flaws such as other brittle materials. Holes b1 and b2 allow the highest ultimate loadings. However, the diameter does not seem to play a crucial role. Figs. 10 and 11 show the evolution of the ultimate load according to initial torque for each king of holes and for both annealed and tempered glass.

Results of the experimental campaign on annealed glass. Ultimate loads according to initial torque and standard deviations.

Hole geometry	Initial torque (daN m)	Ultimate load (kN)	Exp. number (stand. deviation kN
Hole a1	2	12.2	4(1.5)
	3	13.2	1
	5	15.3	5 (2.1)
Hole a2	2	15.7	3 (1.9)
	5	11.4	6 (4.3)
	10	16.6	1
Hole b1	0	12.7	1
	1	23.3	5 (2.3)
	2	22.7	4 (4.3)
	>2.5	0	3
Hole b2	1	21.8	3(2.1)
	2	22	9 (2.5)
	5	11.9	1
	>5	0	2
Hole c1	2	24.4	2 (2.4)
	2.5	16.8	3 (3.3)
	3	20.5	5 (4.6)
	5	26	2 (4.7)

#### Table 7

Results of the experimental campaign on tempered glass. Ultimate loads according to initial torque and standard deviations.

Hole geometry	Initial torque (daN m)	Ultimate load (kN)	Exp. number (stand. deviation kN)
Hole a1	2	72.5	2 (3.8)
	5	72.4	6 (4.7)
	10	76.5	2 (4.9)
Hole a2	2	77	2(0)
	2.5	63	1
	5	77.9	4(9)
	>5	73	1
Hole b1	1	107	4(7.1)
	7.5	118.5	1
	10	115.7	1
Hole b2	2	112	4 (6.1)
	4	109	1
	10	85	1
Hole c1	1	86.5	2 (9.2)
	2.5	94.2	1
	7.5	89	1
	10	62	2 (2.9)

#### 4.3. Other analyses

Photoelasticity and fractography (post-mortem observation of the failure origin) are also performed during and after the tests.

#### 4.3.1. Fractography

At the end of each test, the failure origin has been investigated in order to estimate the failure stress by the measurement of the smooth fracture surface called mirror (see Fig. 12) by means of the following formula:

$$\sigma_{fail} = \frac{M}{\sqrt{r}} \tag{6}$$

where M is the mirror constant and r the mirror radius. The boundary of the smooth zone corresponds to a stress intensity factor equal to the crack branching stress intensity factor (noted Kcb).

In this study the value M = 1.85 MPa $\sqrt{m}$  is assumed [5]. It corresponds to a penny shape flaw located on a straight edge. The curvature radius of each hole of the present study is large enough to justify this assumption [11].

This method allows then to obtain a local stress state.



**Fig. 10.** Results of the experimental campaign on annealed glass. Ultimate load according to initial torque for each kind of hole.



**Fig. 11.** Results of the experimental campaign on tempered glass. Ultimate load according to initial torque for each kind of hole.



Fig. 12. Results of tests on annealed glass. Fracture origin and mirror zone.

#### 4.3.2. Photoelasticity

Photoelastic measurements have been carried out during tests with a polariscope in the zone located above the bolted connection. Thanks to these measurements, it is possible to follow the isochromatic fringes. The use of the 3D photoelasticity theory, as described in [12], has been necessary to achieve this goal. The methodology takes into account the change of the secondary principal directions in the thickness due to shear gradients in such complex geometries.

Then the images obtained during tests can be used for the validation of the FE modeling of these tests; they give information on stress states all around the hole.

**Fig. 13.** Validation of the FE modeling. Comparison between Experimental (left) and calculated (right) photoelastic fringes.

#### 4.4. Finite element modeling of tests

The FE software Abaqus is also used in this section. An attention is paid on the modeling of contacts. Two different contacts between deformable bodies are considered: on one hand between the steel connector and the aluminum rings, and on the other hand between the aluminum rings and the chamfer of the glass plate. Glass is supposed to be elastic, aluminum and steel are supposed to be elastic-plastic. The problem to solve is non-linear and threedimensional. Besides, friction and clearances between the different parts are taken into account. Surface-to-surface contact with small sliding option is used to define the contact relationship. The initial torque applied to the connector is introduced by prescribing a displacement to the connector towards the glass chamfer, this displacement is identified thanks to a photoelastic analysis. For the simulation of the mechanical tests performed on thermally tempered glass, previously calculated residual stresses are added to stresses induced by in-plane loading of the connector.

The validation of the modeling is performed firstly on the value of the failure stress and secondly on the photoelastic fringes around the connector. A local failure is assumed to occur in glass when the maximum principal stress reaches a critical value. Such a criterion is classical for brittle materials in which failure is due to a mode I fracture (fundamental fracture mode corresponding to crack opening). The failure stress estimated by the measurement of the mirror has been compared with the maximum principal stress given by simulation at the place where the failure took place during tests. The mean deviation between the measured and predicted failure stresses is less than 10% and 5% for, respectively, annealed and tempered glass. This is of the same order of magnitude than the precision of the stress measurements by fractography.

Besides, Fig. 13 shows an example of comparison between experimental and simulated photoelastic fringes for an in-plane loading equal to 20 kN and a large chamfer (on annealed glass). Very close-to-reality results enable to conclude that the FE modeling is validated.

4.5. Sensitivity analysis. Towards a determination of an optimal connection

#### 4.5.1. Initial torque applied to bolt

Fig. 14 presents the ultimate loads obtained on annealed glass according to the initial torque applied to the connector. Results are given only for large and mean chamfers.

Two contradictory effects are put into evidence not only for annealed glass but also for tempered glass, however it is more obvious for annealed glass. The first effect is harmful: the application of a high torque induces the immediate failure of



**Fig. 14.** Results of the experimental campaign on annealed glass. Ultimate load according to initial torque for large and average chamfer geometries.

Table 8

Results of numerical investigations. Maximum of the maximal principal stress (in MPa) for various initial torques and various hole geometries and for an in-plane loading equal to 50 kN.

Initial prestressing	0 daN m	1 daN m	2 daN m
Hole b1	46.5	45.3	42.9
Hole b2	38.2	37.2	34.7
Hole c1	41.7	40.1	36.1

samples. The minimal prestressing leading to immediate failure of glass varies with the hole geometry: it is equal to 2.5 daN m for hole b1 and 5 daN m for hole b2. Holes with small chamfer seem to be less sensitive to the harmful effect in this range of torque values. The second effect is beneficial. It is especially evident considering the low value of the ultimate load obtained for an initial torque equal to zero (see Fig. 14). This last result and conclusion is confirmed by the FE modeling. It has been shown that the torque induces a compression in the maximal principal direction of the inplane loading of the connector. Table 8 proves this last assumption. It shows the maximum of the calculated maximal principal stress for different values of initial torque (0, 1 and 2 daN m) and for a constant in-plane loading equal to 50 kN. This maximum decreases when initial torque increases.

These two opposite effects prove that an optimal value of the torque exists. Further research is needed to determine accurately this optimal value. It belongs to the range 1-3 daN m for annealed glass and is probably higher for tempered glass.

## 4.5.2. Influence of the glass–aluminum friction coefficient and of the interlayer material

Fig. 15 shows a post-mortem observation of a tested sample. Very clear marks can be observed on the conical zone of the hole and leading into the failure origin. Observations with Scanning Electron Microscope (SEM) and analyses with Energy Dispersive Spectrometer (EDS) reveal that these deposits are composed of aluminum (Fig. 16).

Then it seems that the interaction between the aluminum washer and the glass (a Hertz contact) excites a flaw of the glass surface. This contact is highly influenced by the friction coefficient between the two materials. That is why some tests with a lubrication of the glass–aluminum contact (application of a silicone joint without thickness) and with PTFE washer instead of aluminum washer have been performed in this study.

Tables 9 and 10 present the various results obtained on holes b and c and compare them to the results on the initial configuration of the connection. For tempered glass and lubricated contact only hole b1 has been tested.

The results are contradictory: whereas a lubrication and a PTFE washer are beneficial and increase the ultimate load on annealed



Fig. 15. Post-mortem observation of a tested sample (annealed glass).



Fig. 16. Aluminum deposit observed using SEM.

Results of experimental campaign. Ultimate loads (in kN) obtained with lubrication of the contact between glass and aluminum.

Hole	Glass type	Without lubrication	With lubrication	Comparison (%)
b1	Annealed glass	23.3	28.3	+21.5
	Tempered Glass	107	96	-10.3
b2	Annealed glass	22	28.9	+31.4
c1	Annealed glass	20	25.8	+28.75

#### Table 10

Results of experimental campaign. Ultimate loads (in kN) obtained with PTFE washer instead of aluminum washer.

Hole	Glass type	With aluminum washer	With PTFE washer	Comparison (%)
b1	Annealed glass	23.3	25.5	+9.4
	Tempered Glass	107	103.5	-3.3
b2	Annealed glass	22	30	+36.4
	Tempered Glass	112	142.75	+27.5
c1	Annealed glass	20	25	+25
	Tempered Glass	94.2	97.75	+3.6

glass, their effect on tempered glass is more debatable and depends on the hole geometry.

#### Table 11

Results of numerical investigations. Calculated maximal principal stress (in MPa) for various values of friction coefficient and for an in-plane loading equal to 50 kN.

Friction coefficient	Hole b1	Hole b2	Hole c1
0.01	51.7 (+26.7%)	37.6 (+15.7%)	39 (+29.6%)
0.05	49 (+20.1%)	36.5 (+12.3%)	37.8 (+25.6%)
0.1	45.3 (+11%)	34.7 (+6.8%)	33.2 (+10.3%)
0.15	40.8	32.5	30.1



Fig. 17. Cracking velocity according stress intensity factor [14], after [15].

For holes b1 and c1, the diminution of the friction coefficient decreases the excitation of the glass flaw by the washer but concentrates and then increases the maximal principal stress in a small area of the hole chamfer. This last result is confirmed by the FE modeling. Table 11 shows the calculated maximal principal stress for an in-plane load equal to 50 kN according to the friction coefficient. This table reveals that the increase of the maximal principal stress is not so important for hole b2.

For tempered glass, at high level of stress, the lubrication film is torn and the harmful effect is predominant. This effect is less important for hole b2, that is why the lubrication has no impact on this kind of geometry for tempered glass. PTFE washers are more resistant, then the two contradictory effects cancel out each other for holes b1 and c1 in tempered glass. For hole b2 the diminution of the flaw excitation becomes predominant and the ultimate load increases.

#### 4.5.3. Influence of the thermal tempering

The experimental campaign has shown that the strength of the tempered glass is higher than the sum of the annealed glass strength and the residual compressive strength (in absolute value). For hole b1, the mean ultimate load in annealed glass is 23 kN. The FE modeling enables to estimate the surface decompression load in tempered glass (70 kN) [13]. The mean ultimate load in tempered glass is 110 kN which is higher than 70 + 23 = 93 kN. It can then be established that the thermal tempering process causes a certain amount of crack healing.

Such a healing can be quantified using a subcritical cracking law which gives access to the radius of the flaw responsible of the failure. This calculation needs to remind some results on the propagation of cracks in glass. These results are illustrated by Fig. 17.

Instantaneous failure occurs when the elastic stress intensity factor reaches or exceeds the material toughness (or critical stress

	Service life and	load rate for	annealed and	tempered	glass
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	Annealed glass	Tempered glass	
t <sub>f</sub> (s)	127.2	111	
ở (MPa/s)	0.318	0.503	

intensity factor). This condition is called Irwin's fracture criterion and is expressed as following:

$$K_I \ge K_{Ic}.\tag{7}$$

A typical value for  $K_{lc}$  for soda lime glass is 0.75 MPa $\sqrt{m}$  [14]. Starting from this point there is an excess energy to drive the crack and the fracture becomes unstable. Under these conditions the crack propagates rapidly ("catastrophic" failure). The flow continues to accelerate and that finally causes the crack to bifurcate or branch along its front. This is observed as an abrupt branching [16]. The value of the stress intensity factor is then equal to another material parameter, K<sub>cb</sub>, called crack branching intensity factor. For soda lime glass a typical value of  $K_{cb}$  is 2.33 MPa $\sqrt{m}$  [14]. This value corresponds also to the mirror radius in fractography.

In glass a crack propagation occurs even if  $K_I \leq K_{Ic}$ . This is a consequence of a stress corrosion that causes flaws to grow subcritically in the presence of humidity.

The calculation of the crack healing due to thermal tempering needs to take into account all the crack propagation phenomena.

Considering the subcritical cracking Evans law it is possible to calculate the size of the surface flaw responsible of the failure [17]:

$$v = AK_{I}^{n} \tag{8}$$

with: A and n the parameters of the law (A = 0.124 and n =12.76 [14] for soda lime glass),  $K_I = \sigma Y \sqrt{a}$  the stress intensity factor, a the flaw size,  $\sigma$  the applied stress, Y the form factor and v the cracking velocity. Since  $v = \frac{da}{dt}$ , the Evans law leads to:

$$\frac{\mathrm{d}a}{\mathrm{d}t} = A \left(\sigma Y \sqrt{a}\right)^n. \tag{9}$$

Considering a constant loading rate,  $\sigma$  is equal to:

 $\sigma = \dot{\sigma} t_f$ 

where  $\dot{\sigma}$  denotes the loading rate and  $t_f$  the service life.

Then (9) leads to:

$$\frac{\mathrm{d}a}{a^{n/2}} = A\dot{\sigma}^n Y^n t_f^n \mathrm{d}t. \tag{11}$$

The integration of relation (11) gives the following expression of t<sub>f</sub>:

$$t_f^{n+1} = (n+1) \frac{\frac{2}{2-n} \left( a_c^{\frac{2-n}{2}} - a_0^{\frac{2-n}{2}} \right)}{A \dot{\sigma}^n Y^n}$$
(12)

where  $a_c$  is the flaw size leading to failure and  $a_0$  the initial flaw size.

Then  $(a_c - a_0)$  is the subcritical cracking quantity.

The Linear Elastic Fracture Mechanics applied to glass enables to write:

$$K_{lc} = \sigma Y \sqrt{a_c} \quad and$$

$$K_{cb} = \sigma Y \sqrt{r}.$$
(13)

As in Section 4.3.1, *r* represents the radius of the failure mirror. Then:

$$\frac{r}{a_c} = \left(\frac{K_{cb}}{K_{lc}}\right)^2 = 9.65\tag{14}$$

with the numerical values formerly given.

r can also be measured by optical microscopy after the test. The following numerical application is performed for plates with hole b1. For this kind of geometry, the mean value of the radius is 2.089 mm for annealed glass and 1.09 mm for tempered glass.  $a_c$ can then be evaluated.

Moreover, the analysis of load-displacement curves coupled with FE modeling enables to predict both service life and loading rate of the various glass plates. Some differences in loading rates between annealed and tempered glass exist and are due to the existence of air gaps between the various elements of the connection: the experimental set-up does not reach immediately its maximal stiffness. Since tempered glass is free of subcritical cracking when compressive residual stresses are not exceeded (in absolute value), the service life of glass plates has to be evaluated after this threshold. As previously said, the FE modeling described in Section 3.3 can be used to predict such surface decompression load. The mean value (over all prestressing values) is 70 kN [13]. Table 12 gives service life as well as load rate for both annealed and tempered glass.

The size  $a_0$  of the flaw responsible of the failure can then be calculated for, respectively, annealed and tempered glass:  $a_0 =$ 0.0632 mm and  $a_0 = 0.0302$  mm.

The flaw size for tempered glass is then smaller. That proves the crack healing effect of the thermal tempering. Thanks to this calculation this effect can even be quantified:

$$\frac{a_{0,\text{annealed glass}} - a_{0,\text{tempered glass}}}{a_{0,\text{annealed glass}}} = \frac{0.0632 - 0.0302}{0.0632}$$
$$= 0.522 \sim 50\%.$$

This effect was assumed in [18,19] but was never demonstrated and quantified as proposed. This result is important for building structures and might be taken into account into the design calculations.

#### 5. Conclusion

(10)

This study deals with the context of the structural glass and focuses on the structural connections between glass panes. The modeling of a dowel type connector, derived from countersunk support, inserted in a holed glass plate is developed with the Finite Element Method. A numerical model has then been developed for the prediction of the stress state in thermally tempered glass plates loaded by a metallic connector. It has been validated by comparisons with experimental results obtained locally (fractography) and in a global way (photoelasticity).

Various connection configurations have been considered, leading to the determination of an optimal point bearing. This study has shown that holes of type b2 (large diameter and large chamfer, with a cylindrical part equal to 1/6 of the plate thickness) present the best geometry among the whole studied ones. Indeed the thermal tempering reinforcement is the most effective and this kind of hole is the most resistant to the in-plane loading of the metallic connector. Besides, this specific geometry is the only one which benefits to the replacement of the aluminum washer by a plastic (PTFE) washer: a mean ultimate load equal to 143 kN has then been obtained during the experimental campaign.

Finally, the sensitivity study performed in both experimental and numerical ways enables to state some other conclusions. In the following essential results are summarized.

#### - Influence of the initial torque applied to the connector

This initial prestressing has a positive impact on the stress state since it induces a compressive stress in the maximal principal direction. However, this more pronounced contact leads also to a surface damage and then to an instantaneous breakage of some glass panes (especially for annealed glass). These two opposite effects prove that an optimal value exists. Further research is needed to determine accurately this optimal value. It depends on the hole shape and on the glass reinforcement.

– Influence of the friction

#### It seems that the contact between the aluminum washer and the glass excites a flaw of the glass surface. This contact is highly influenced by the friction coefficient. That is why a sensitivity study on this parameter has been performed. To examine the influence of the friction, different friction coefficients have been applied in the FE modeling. Besides, experiments with a lubrication of the contacts have also been carried out. Here two different phenomena seem to act in an opposite way: when decreasing the friction coefficient, the maximal principal stress increases but the excitation of the surface flaws decreases. Then the decrease of the friction is beneficial for annealed glass but seems to be harmful or without effect for tempered glass.

- Influence of the interlayer material

As it is sometimes practiced in glass structures, PTFE washers, instead of aluminum washers, have been considered in a few experiments. The conclusions on the influence of the interlayer material coincide with those on the influence of the friction coefficient. An increase of the ultimate load is obtained for annealed glass whereas no real impact, excepted for glass plates with hole b2, is put into evidence for tempered glass. The two opposite effects, as previously described for the influence of the friction, exist. Better results obtained for tempered glass are probably due to the fact that PTFE washers are more resistant than silicone joints.

– Influence of the thermal tempering

The thermal tempering has two beneficial effects. Not only it increases of course the glass strength because of the surface compression reinforcement but also this process seems to imply a certain amount of crack healing (about 50%). This crack healing has been supposed for a long time [18,19] without being clearly demonstrated.

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