Engineering Structures 31 (2009) 946–955

Contents lists available at ScienceDirect

Engineering Structures

journal homepage: www.elsevier.com/locate/engstruct

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

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A R T I C L E   I N F O

Article history:
Received 30 May 2008
Received in revised form
1 September 2008
Accepted 1 December 2008
Available online 28 January 2009

Keywords:
Glass
Connections
In-plane loading
Thermal tempering

A B S T R A C T

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.

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 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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doi:10.1016/j.engstruct.2008.12.004

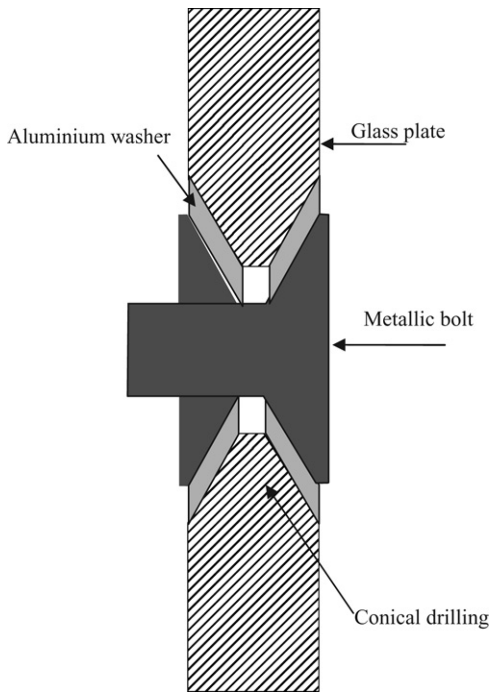

Fig. 1. Schematic cross section of the studied connection.

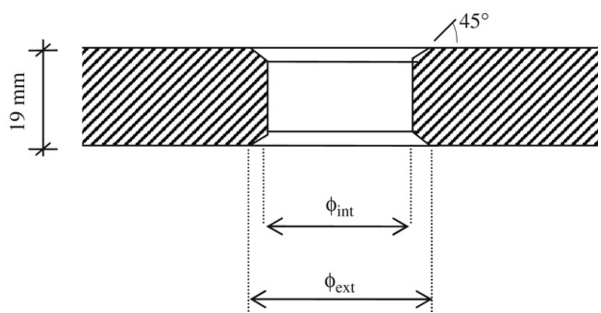

Fig. 2a. Cross section of cylindrical holes a1 and a2.

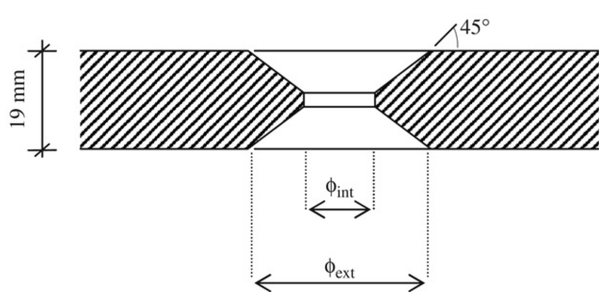

Fig. 2b. Cross section of conical holes b1 and b2.

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

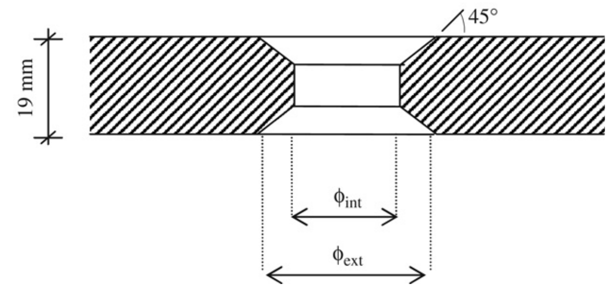

Fig. 2c. Cross section of mean chamfer hole c1.

Table 1
The five different studied geometries.

Designation	Φ_{int} (mm)	Φ_{ext} (mm)
a1	38	40
a2	54	56
b1	24	40
b2	40	56
c1	30	40

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

Table 2
The identified forced convection coefficients (in W/m^2K).

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

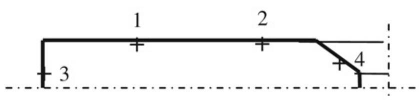

Fig. 3a. Locations of the identified convection coefficients for holes b1, b2 and c1.

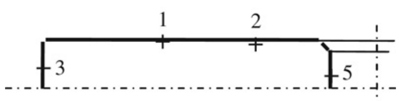

Fig. 3b. Locations of the identified convection coefficients for holes a1 and a2.

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 Φ_{cond} (in W/m^2) is function of the thermal gradient thanks to the thermal conductivity λ (in W/m/K) :

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

For glass it has been found that λ 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:

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

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 × 400 × 19 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 ®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 ®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 (Φ_s), and from the volume on the other hand (Φ_v). Thus, surface and volume emissivities of glass plates are defined in the following way:

- the surface emissivity (ε_{surf}) is defined for the spectral field where glass is opaque,
- the volume emissivity (ε_{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:

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

and

$$\Phi_v = 2\sigma[\varepsilon_{vol}(t, T_v)T_v^4 - \varepsilon_{\mathrm{vol}}(t, T_{ext})T_{ext}^4] \tag{5}$$

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 σ 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 σ_{33} in the hole (see also Fig. 4a);

Table 3
Residual stresses calculation. Summary table of the predicted results.

Type	Surface stress σ_{33} (MPa)			Membrane stress (for $x_1 = 0$) (MPa)	Neutral line position x_1 (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

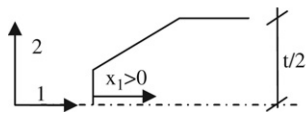

Fig. 4a. Coordinates definition.

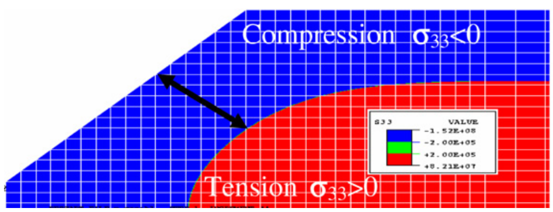

Fig. 4b. Results of the FE modeling of the thermal tempering process. Definition of the compression thickness.

– the membrane stresses (Σ) 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 σ_{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

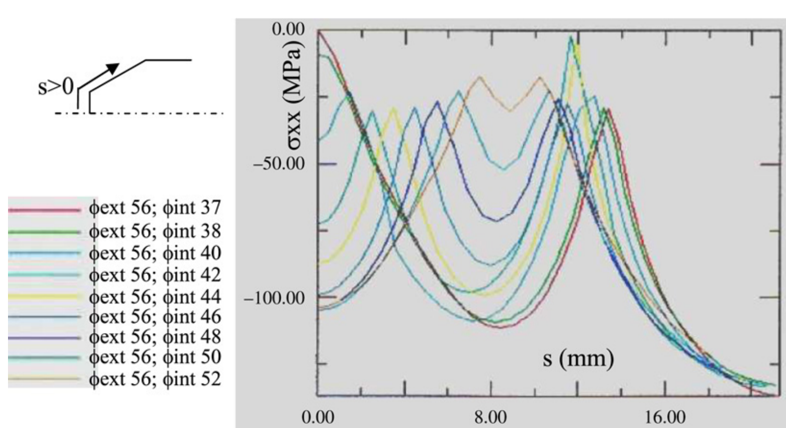

Fig. 5. Residual stresses modeling. Tangential stresses σ_{xx} along the hole chamfer (s corresponds to the curvilinear abscissa inside the hole including the cylindrical part). Then the junction between the cylindrical and the conical parts of the hole corresponds to the first local minimum (in absolute 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 $\phi_{int} = 52$ mm and $s = 13.43$ mm for the hole with $\phi_{ext} = 56$ mm and $\phi_{int} = 37$ mm.

Fig. 6. Metallic connector used in this study.

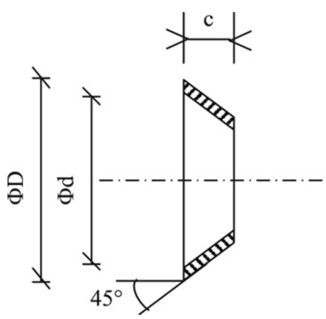

Fig. 7. Aluminum washer for holes b and c.

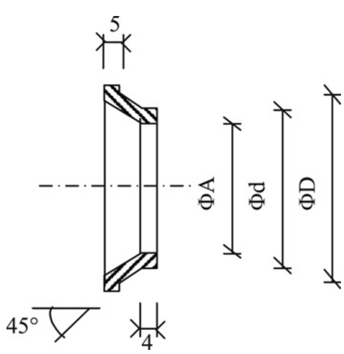

Fig. 8. Aluminum washer for holes a.

Table 4
Main dimensions of the interlayer for holes b and c.

Type	b1	b2	c1
d (mm)	36	50	36
D (mm)	40	56	40
c (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.

Type	a1	a2
A (mm)	32	48
d (mm)	37.9	53.9
D (mm)	40	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 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.