OPTIMAL HOLDER CONFIGURATIONS FOR SUSPENDED GLASS PANELS

Fabio Auriemma and Roberto Aiello
Tallinn University of Technology, Department of Mechanical and Industrial Engineering,
Ehitajate tee 5, z.i.p.19086, Tallinn, Estonia.
email: fabio.auriemma@ttu.ee

Suspended glass panels are unframed windows supported by holders located peripherally and in proximity of the panel corners. These panels are made of glass or laminated glass and can be used, among other purposes, as noise barriers.

In this study, a two-step procedure is implemented in order to improve the acoustic performance of suspended panels. First, Young’s modulus and Poisson’s ratio of monolithic and laminated samples are extracted by comparing modal data from simulations, based on finite element method (FEM), and experimental analysis. Once the material properties have been obtained, an optimization procedure is implemented, aiming to find the optimal position of the holders that maximizes the sound transmission loss (TL) at low frequency range. This procedure is based on genetic algorithm (GA), which iteratively provides the position of the holders, hybrid modal FEM/statistical energy analysis (FEM/SEA), which is used to compute the TL of the panels.

Eight optimizations have been performed where the TL, averaged in the frequency range 20-300Hz, has been maximized for 8 different panel configurations, i.e. square (1m x 1m) and rectangular (2.5m x 0.8m) geometries, monolithic and laminated tempered glass, 4 and 6 holder set-ups.

Keywords: Laminated glass, Suspended glass panels, Windows, FEM/SEA, Genetic algorithm.

1. Introduction

Glass and laminated glass panels in frame-less configurations are widely utilized in modern architecture with both aesthetic and functional purposes. Among the wide variety of mounting solutions, recent trend has seen growing use of discrete holders, typically circularly shaped, placed nearby the edges. This configuration characterizes the so called suspended panels. The present paper elaborates on the effects of the position of the holders on the vibro-acoustic behaviour of these panels with the goal of finding the optimal positions which maximize the sound transmission loss (TL).

The TL of a partition is defined as a function of the transmission factor $\tau$, $TL = 10\log(1/\tau)$. $\tau$ is the ratio $(P_t/P_i)$ of the transmitted over the incident power and it is strictly related to the vibrational behaviour of the partition.

The study of the vibro-acoustic behaviour of a simple, perfectly limp plate, whose specific impedance comes exclusively from the inertia of its mass, dates back to the early ’40s of the previous century. In the work [1], the so called limp-wall mass-law was first formulated. In the study [2], the authors focussed on different combinations of impervious layers, air gaps and acoustical blankets, in
normal incidence. In [3], instead, a diffuse acoustic field was considered. Since then, a wide number of investigations have been carried out by taking into account different configurations of the partition [4, 5, 6, 7]. In the present work, a 2-step procedure is implemented, aiming at tackling the problem of the sound insulation of suspended glass panels at very low frequency range, which is believably the most critical range in the majority of the room acoustic applications ([8, 9, 10]). By following a classical procedure based on comparison between numerical and experimental modal data ([11]), some of the material properties of the panels, not known a-priori, are identified (see Fig. 1a). After, an optimization procedure based on GA and hybrid FEM/SEA ([12, 13]) has been implemented (see Fig. 1b) and the optimal position of the holders, which maximizes the TL averaged in 20-300 Hz frequency range, is provided.

Eight different configurations have been studied, i.e. panels with common square and rectangular geometries (sizes: 1m x 1m and 2.50m x 0.80m, respectively), provided with 4 and 6 holders, in monolithic and laminated (with intermediate film of polymeric Polyvinyl Butiral - PVB) structural arrangements.

2. Material characterization

For the characterization of the constitutive materials, the procedure shown in Fig. 1b is implemented. This procedure has been applied to monolithic and laminated samples (sizes: 50mm x 20mm x 8mm) provided with the material properties of the suspended glass panels object of the optimization process. The material properties of the FE model of the samples are updated iteratively until the numerical modal results match the experimental ones.

This approach provides linear elastic isotropic homogenized material properties also for the laminated sample enclosing the non linear elastic PVB interlayer. The isotropic homogenized material approximation is here justified by the fact that the global behaviour of the laminated sample will be utilized in the optimization process, resulting from the contribution of both constituting materials.

In both the experimental and the numerical approaches utilized for the material characterization, the samples are considered as in free-free conditions. In order to meet this condition during the tests, the physical samples have been held by a suspension system constituted by soft elastic bands designed to guarantee that the highest rigid mode frequency is less than 10 − 20% of first resonance frequency of the suspended structure ([14]).

The pick-picking method has been used to extract the modal parameters by means of roving hammer impact test ([15]). The equipment utilized consists of: a hammer AP Tech™ AU01 provided with force transducer; an ICP mono-axial accelerometer PCB™ 353B33; a dynamic signal analyser (National Instruments™ NIcDAQ 9174 and NI 9234); a PC based virtual instrument (LabVIEW™). The FRFs measurements have been obtained by sequentially impacting with the hammer 12 points...
Table 1: Material parameters extracted with indirect characterization based on modal analysis.

<table>
<thead>
<tr>
<th>Properties</th>
<th>Monolithic sample</th>
<th>Laminated sample</th>
</tr>
</thead>
<tbody>
<tr>
<td>Young’s Modulus</td>
<td>$6.7 \times 10^9$</td>
<td>$7.7 \times 10^9$</td>
</tr>
<tr>
<td>Density</td>
<td>2450</td>
<td>2490</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>0.25</td>
<td>0.4</td>
</tr>
</tbody>
</table>

Figure 2: Square and rectangular panel geometries, 4 and 6 support configurations.

As said before the mode shapes are dependent only by the geometry (physically they are stationary waves), so it is immediately possible to compare the numerical mode shapes with the experimental ones. The FEM solver utilized in this work for modal analysis is ANSYS™. The test object has been discretized by means of a mesh consisting of 906 nodes and 401 tetra elements. The outcomes of the material characterization process, i.e. the parameters found for the glass tempered panel and glass tempered panel with PVB, are shown in Table 1.

3. Optimization procedure

Once the material properties have been extracted, it has been possible to perform the second step of the procedure, i.e. the optimization process for the different panel configurations, as shown in Fig. 1b. The 8 panel configurations which have been optimized are listed in the section II. The results of the optimal solutions, in terms of averaged TL, are compared with the ones related to the standard configurations, with the mountings placed peripherally.

The optimization procedure has been implemented by means of of an external optimizer (Mode-Frontier™) coupled to a FEM solver (ANSYS™) and a hybrid FEM/SEA solver (VAOne™). The goal is to find the position of the holders which maximizes the averaged TL ($TL_{av}$) in the very low frequency range [20, 300] Hz, i.e. $TL_{av} = \int_{20}^{300} TL(f) df$. The external optimizer runs the GA, which varies the positions of the holders and provides new inputs for the FEM modal analysis. The modal results, in turn, are the input for the FEM/SEA acoustical model for the evaluation of the TL. As it will be shown, the strong influence of the mode shapes on the TL trend is determined by the resonance phenomenon between acoustic incident field and the structure panel.

The positions of the holders are chosen by the optimizer within certain geometrical limits, shown in Fig. 2. In the optimizations with 4 holders, the panel area is divided in four identical areas and a constraint is imposed to the optimizer so that each individual produced by the GA must have one holder in each of the four sub-areas. In the optimizations with 6 supports, the sub-areas are only two and each one contains three holders.

In the subsections containing the results of the optimizations, thus in Fig.s 3 and 5, sketches of the panel configurations are also included, where red squares refer to the holders in the standard configuration and green ones indicate the holders in the optimized configuration. The positions of the holders are also listed in tables embedded in the figures and expressed in form of relative coordinates, $(x_{rel}, y_{rel}) = (x/L_x, y/L_y)$, where $x$ and $y$ are the geometrical coordinates of the generic holder.
and $L_x$, $L_y$ are the horizontal and vertical sizes of the panel. For each case examined, a comparison between the $TL_{av}$ of the standard and the optimized configurations is given in terms of percentage variation, $\Delta TL_{av} = (TL_{av-opt} - TL_{av-std})/TL_{av-std}$.

### 3.1 Results of the optimization - Square panels

The results of the optimization process, including the TL curves of the optimal and the standard configurations are presented in this and in the following subsection. This subsection includes the

![Graph a](image_a.png)

**a)** Monolithic glass, square shape, 4 supports, 20-300 Hz.

![Graph b](image_b.png)

**b)** Monolithic glass, square shape, 6 supports, 20-300 Hz.

![Graph c](image_c.png)

**c)** Laminated glass, square shape, 4 supports, 20-300 Hz.

![Graph d](image_d.png)

**d)** Laminated glass, square shape, 6 supports, 20-300 Hz.

Figure 3: Standard and optimized configurations for square shaped panels.

The results of the standard and optimized configurations for square panels (see Fig. 3).

Regarding the standard configurations, it noticeable that a dip at the first resonant frequency is exhibited, which penalizes the acoustic behaviour of the panel. Moreover, the advantage of the laminated solutions is typically in the smoothing of the TL curve due to the increased damping. Higher
The optimal solutions are characterized by the presence of a TL peak following the dip of the first resonance mode. The positive effect of it is clearly noticeable in Fig. 3c and 3d (monolithic and laminated cases with 6 holders), where the peak almost overlaps the fall exhibited by the standard configurations. As a consequence, in this two cases the $TL_{av}$s provided by the optimal solutions overcome by 14.2% and 15% the performance of the standard configurations. In the case of Fig. 3b (monolithic with 4 holders) the improvement of the performance is reduced to 11.7% because the optimized solution provides two dips at 50 Hz and 70 Hz, other than the peak at 60 Hz.

It is interesting to notice that, in case of Fig. 3b (monolithic with 6 holders) the main advantage of the optimal configuration does not derive from the peak in correspondence of the first natural mode, but by the favourable trend of the TL from 70 Hz to 220 Hz. Similar behaviour is shown also in the configuration with laminated glass and 6 holders.

Regarding the geometrical positions of the holders, in the optimized configurations the constrains are moved towards the centre of the panel. In fact, in comparison with the standard case, the modal analysis shows that the central area of the optimized configurations, stiffened by the constraints, is almost unaffected by modal amplitudes at low frequencies. As a consequence, the transmission of the acoustic incidence power is largely reduced. Contrariwise, in the standard cases the whole area of the panel has non-zero modal amplitudes and it is involved in the sound transmission. In fact, in this case, the supports are placed peripherally providing a reduced stiffness for the central area. This can be seen in Fig. 4a and 4b, where the the first mode shapes of the standard and optimized configurations are shown. In the latter case a wide lobe in the central part of the panel is visible, while in the last case that lobe shaped mode is prevented by the presence of the supports.

3.2 Results of the optimization - Rectangular panels

In this study, the area of the rectangular panels is 2.24 times larger than that of the square panels, but the number of holders considered is still the same (4 and 6 holder cases). As a consequence the number of supports per unit area is reduced and the rigidity itself of the panels largely decreases. This situation negatively affects the behaviour at low frequencies where the increase of $TL_{av}$ with respect to the standard cases is about half than in case of optimized square panels. In fact, the maximum $\Delta TL_{av}$ is of 8.2% for the laminated, 6 holder case and the minimum is of 3.9% for the monolithic, 4
support case. These results are shown in Figure 5.

The reduced stiffness of the structures is obviously more sensitive for the 4 support cases. In any, case a clear effect of the reduced stiffness is the increase of modal density compared to the square cases. The first mode of the standard, 4 and 6 holder cases occurs at 8 and 24 Hz respectively. Naturally, the dip of TL at 8 Hz does not represent an acoustic problem since it occurs below the audible range and it can even be advantageous in the computation of the $TL_{av}$ since it is followed by a rise of TL. However, the optimized configuration has the supports placed towards the central part of the panel. This solution, similar to the one obtained for square panels with 4 supports, results in negligible modal amplitudes of the inner area at low frequencies. As previously mentioned, the advantages of the optimized configuration are limited because of the below 20 Hz 1st mode of the standard configuration and because, for higher natural frequencies, also the area within the perimeter delimited by the supports is not affected by non-zero modal amplitudes.

For the 6 holder cases, the advantage of the optimized configurations resides in the shift towards higher frequency of the first mode, which occurs at 51 Hz and positively affect the $TL_{av}$. In Figs. 4c and 4d the mode shapes at the 1st mode are depicted for both the standard and the optimized cases. Now, for the optimal solution, the location of the supports has not any more the goal of non zero modal amplitudes in the central area, but rather that of rising the modal frequencies.

Similar considerations can be done regarding the laminated panels, where the $\Delta TL_{av}$ is even higher, but the strategy followed by the optimizer and the configurations proposed are similar to the ones of the monolithic case.

A final note regards a common characteristics for the majority of the optimal solutions, in case of both square and rectangular panels. This is the presence of pairs of holders which are nearly parallel to each other and slightly oblique with the respect to the borders of the panel. This characteristic is more evident for the 4 holder cases. By analysing the mode shapes of the panels (see Fig. 4) it possible to understand that, by locating pairs of holders non parallel to the borders, it is more likely that the structure exhibits anti-symmetrical or, at least, non-symmetrical mode shapes. As known from literature, this fact can induce self-cancellation effects due to destructive interference or reduce constructive interference effects from different vibrating parts of the panel.

Despite the fact that 20-300 Hz is the most critical frequency range for acoustic insulation of windows, further studies are planned to find the positions of the holders which optimize the vibro-acoustic behaviour of suspended panels over a wider frequency range of excitation (e.g. 20-1000 Hz).

4. Conclusions

The requirement for suspended monolithic and laminated glass panels to effectively behave as noise barriers in room acoustic applications is addressed in this paper. A two-step procedure has been implemented aiming, first, at finding the material parameters of the panels (Young’s modulus and Poisson’s ratio) and thus at detecting the optimal positions of the holders which maximize the averaged sound transmission loss at the very low frequency range, 20-300 Hz.

The first step is based on iterative comparison between modal FE and experimental data. The second step uses a genetic algorithm to define the position of the holders and a hybrid modal FEM/SEA to compute the transmission loss.

8 different configurations have been analysed, including square and rectangular shaped panels, monolithic and laminated glass with PVB interlayer, 4 and 6 holder configurations. The TLs of the optimal configurations have been compared with those of the standard configurations where the supports are placed peripherally.

The optimization trends see the holders located centrally on the panels. In case of square panels, the configurations with centrally located holders allow reducing the effects of the first modes on the
transmission of the sound. With comparison to the standard configurations, the increase of $TL_{av}$ in the optimized configurations ranges from 11.7% to 14.2% for monolithic, 4 support and laminated 4 support cases respectively.

The increase of TL is lower in case of rectangular panels, ranging from 3.9% to 8.2% for monolithic 4 support and laminated 6 support configurations, respectively.

A common feature among almost all the optimized solution is the presence of pairs of supports which are nearly parallel to each other and slightly oblique with the respect to the borders of the panel. This is believably due to the acoustically advantageous effects of having non-symmetrical mode shapes of the panel.

Figure 5: Standard and optimized configurations for rectangular shaped panels in frequency range 20-300 Hz, see Table I: a) Case 9; b) Case 13; c) Case 11; d) Case 15.
5. Acknowledgements

This research has been supported by:
- Innovative Manufacturing Engineering Systems Competence Centre IMECC and Enterprise Estonia (EAS) co-financed by European Union Regional Development Fund, project EU48685.
- Estonian Centre of Excellence in Zero Energy and Resource Efficient Smart Buildings and Districts, ZEBE, grant 2014-2020.4.01.15-0016 funded by the European Regional Development Fund.

REFERENCES


