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New Seismic Criteria For The Design Of Glass Window Shields To Reduce The Damages Of Flying Glass

Dr. Ali Al-Athary
College of Engineering
Al Mustansirya University

Dr. Saud A. Sultan
College of Engineering Al Mustansirya University

Abstract

Traffic vibrations, damaging windstorms, earthquakes, break-ins, and terrorism explosions cause extensive damage to buildings and property each year, and can even result in death and injury to building occupants and bystanders. Much of this destruction is due to a simple and preventable hazard: flying glass. The objective of this study was to find the most suitable shape, dimensions, thickness, and glass material properties of float glass sheets to be used to manufacture window shields to reduce the risk of death and injury by shattered and flying glass due to seismic, explosion, and traffic vibration waves. It was found that decreasing the dimensions of a certain geometrical shape of glass sheets and keeping all the other properties constant, not always increases the seismic equivalency factor (SEF). Decreasing the dimensions of the rectangular shape from (20x40 cm) to (15x25 cm) decreases the seismic equivalency factor (SEF) from 0.597 to 0.568. Also, decreasing the dimensions of the square shape from (15 cm) to (10 cm) decreases the seismic equivalency factor (SEF) from 2.837 to 1.107. Also, Increasing the thickness of the glass sheet and keeping all the other properties constant, not always increases the seismic equivalency factor (SEF). Increasing the thickness of the 15 cm square shape from 4 to 6 mm decreases the seismic equivalency factor (SEF) from 2.837 to 0.627. Increasing the thickness of the 15 cm square shape from 6 to 8 mm decreases the seismic equivalency factor (SEF) from 0.627 to 0.496. The effect of Young modulus and Poisson's ratio have little or no effect on the seismic equivalence factors (SEF) of the studied glass sheets. Guideline values for shape, dimensions, and thickness of float glass sheets were recommended for engineers involved in the design and manufacturing of window shields.

Key Words : Seismic Criteria, Flying Glass Damage, The Seismic Equivalency Factor, and Geometrical Shape of Glass Sheets.

الخلاصة

اهتزازات وسائط النقل و العواصف المدمرة و الزلازل الأرضية و حالات السطو والتفجيرات الإرهابية تسبب ضرر واسع للأبنية والممتلكات سنويا و تسبب الموت و الجروح لشاغلي الأبنية و المارة. معظم التدمير ناتج عن خطر بسيط ولا يمكن منعه إلا وهو الزجاج المتطاير.هدف الدراسة هو إيجاد الشكل الهندسي والابعاد والسمك وخواص مادة الزجاج الطافي المثلى التي يجب استخدامها لتصنيع الشبابيك الزجاجية لتقليل خطر الموت والجروح للزجاج المتحطم والمتطاير نتيجة الموجات الزلزالية و التفجيرات و اهتزازات وسائط النقل . طرق تحليلية وطريقة العناصر المحددة استخدمت لدراسة الاهتزاز ألانثنائي الحر للصفائح النحيفة. لقد وجد أن الشكل الهندسي الأمثل للألواح الزجاجية من وجهة نظر الاهتزاز الطبيعي والمعامل الزلزالي المكافئ هو اللوح الزجاجي المربع. وأن الشكل شبه المنحرف وشكل نصف الدائرة للألواح الزجاجية لها اقل اهتزاز طبيعي ومعامل زلزالي مكافئ بالمقارنة مع الإشكال الهندسية الأخرى. وأن الشكل المثلث و المثمن للألواح الزجاجية لها اقل اهتزاز طبيعي ومعامل زلزالي مكافئ بالمقارنة مع الشكل المربع والمستطيل. وأن زيادة السمك للألواح الزجاجية له تأثير قليل أو سلبي على المعامل الزلزالي المكافئ وهدا يتناقض مع الفكرة الشائعة لزيادة سمك الألواح الزجاجية لتقليل خطر التحطم. وأن معامل يونك ونسبة بواسون وطريقة تثبيت حافات الألواح الزجاجية لها تأثير قليل على المعامل الزلزالي المكافئ للألواح الزجاجية التي تمت دراستها.

1. Introduction

Glass is a solid material which is typically brittle and optically transparent. Float glass which is used to manufacture the window shield is a sheet of glass made by floating molten glass on a bed of molten metal, typically tin. This method gives the sheet uniform thickness and very flat surfaces. Most float glass is soda-lime glass, but relatively minor quantities of specialty borosilicate [(1)]. Modern windows are made from float glass which has a Young modulus of 50-90 GPa, Poisson ratio of 0.22-0.23, and a mass density of 2500 kg/m^3 [(1) & (2)]. Window shield safety and security can be achieved by holding the glass in its place even when it is shattered by vibration forces or flying debris. The glass window shield is usually made of thin plate sheets with different geometric shapes to satisfy mainly the architectural requirements as shown in Figure (1).

In practice, the thickness of these glass sheets is from 4 to 8 millimeters. The glass window shields are fixed to their surrounding frames by hardening cementing material. Recently, the common attitude is to reduce the dimensions and to increase the thickness of glass sheets to increase their resistance to shattering due to strong vibration created by traffic vibrations, seismic or explosion waves. The effect of the shapes, dimensions, thickness, and material properties of glass sheets is not known, and not mentioned in the literature up to the capacity of the authors knowledge. Therefore, this research was carried out to find the most suitable shapes, dimensions, thickness, and material properties of glass sheets to be used in the design of window shields to reduce the death and injury of building residents?"

1.1 Vibration of thin plates

Due to their potential applications, the vibration of thin plates with complex boundary conditions has received considerable attention by researchers. The free vibration of rectangular plates has been the subject of numerous studies. Free vibrations deal with some natural characteristics of plates. These natural vibrations occur at discrete frequencies, depending only on the geometry and material of the plate. Knowledge of the free vibration characteristics of thin elastic plates is important both for our general understanding of the fundamentals of the plate behavior and for industrial applications of plate structures. The natural frequencies of plate structures must be known to avoid the destructive effect of resonance with adjacent rotating or oscillating equipment (such as jet and reciprocating aircraft engines, electrical machinery, marine turbines and propulsors, etc.).

Since the free vibrations occur in the absence of all external forces we address the free vibration analysis of plates with the solution of homogeneous partial differential equations with homogeneous boundary conditions [(3)]. Dynamic loads may be created by moving vehicles, wind gusts, seismic disturbances, unbalanced machine vibrations, flight loads, sound, etc. Dynamic effects of time dependent loads on structures are studied in structural dynamics.

Figure (1): Different shapes of float glass sheets are used to manufacture window shields.

Structural dynamics deals with time-dependent motions of structures, primarily, with vibration of structures, and analyses of the internal forces associated with them. Thus, its objective is to determine the effect of vibrations on the performance of the structure. The dynamics of plates can be modeled mathematically by partial differential equations based on Newton's laws or by integral equations based on the considerations of virtual work. Damping effects are caused either by internal friction or by the surrounding media. Although structural damping is theoretically present in all plate vibrations, it has usually little or no effect on (a) the natural frequencies and (b) the steady-state amplitudes; consequently, it can be safely ignored in the initial treatment problem.

The free vibration, which occurs in the absence of applied loads but may be initiated by applying initial conditions to the plate (4). The free vibration deals with natural characteristics of the plates, and these natural vibrations occur at discrete frequencies, depending only on the geometry and material of the plates. Since the natural frequencies of free vibration can serve as an important index capable of describing the dynamic characteristics of plate structures, considerable attention is paid to their free vibration analysis. For simply supported, isotropic, rectangular plates of uniform thickness, the extension of Navier's method (4) yields the natural frequencies in a relatively simple manner.

Also, the forced vibration of such plates can be solved without too much difficulty by expressing the time dependency of the forcing function in trigonometric series. The extension of Levy's method [4] can yield, in a somewhat more complex way, solutions to free vibrations of rectangular plates having two opposite edges simply supported while the other two edges are free or fixed. Solution of forced vibrations of such plates, however, tends to be extremely tedious. Similar is the case of the free vibration analysis of rectangular and circular plates having all edges free or clamped. Energy methods are somewhat more general than the above discussed classical methods and yield remarkably accurate results even if the boundary conditions are more difficult [5]. Application of these methods can also be extended to plates of various shapes and of variables thickness. Their accuracy, however, depends to a large extent on a suitable selection of eigenfunctions.

This need to select appropriate shape functions that satisfy the geometric boundary conditions and approximate the modes of vibrations is the main disadvantage of energy methods. Methods have been introduced to facilitate the selection of suitable shape functions. When the only information sought is the lowest natural frequency of the plate, the use of Rayleigh's method [6], which often yields good accuracy, is highly recommended. A modification of Rayleigh's method by Morley [5] uses merely the dead load deflections of plates. In certain cases this approach can be extended to obtain natural frequencies pertinent to second and even to third modes as well. The resulting solutions, as all energy solutions, are upper bound.

The Ritz method [6] is based on the principle of minimum potential. This approach is more general than Rayleigh's solution techniques, since it can give information on frequencies and mode shapes of higher modes. The accuracy of the first and second modes is usually good. However, the accuracy for higher than second mode, in general, deteriorates progressively [5]. Russian scientists working on the theory of elasticity have significantly contributed to the development of energy methods. Like the method of Ritz, the energy based method developed by Galerkin [5] assumes the solution of modal shapes in the form of series, terms of which individually satisfy all boundary conditions and have fourth derivatives. Galerkin's approach applies the lateral displacements directly to the differential equation of motion, which leads to a simpler way of obtaining eigenvalues and eigenfunctions.

In the first step, the vibration problem of plates is reduced to the evaluation of definite integrals of simple functions selected in advance. Next, this integral transformation results in a system of homogeneous linear equations that has a nontrivial solution only when the system determinant vanishes. Thus, the final solution of a plate vibration problem is further reduced to the determination of eigenvalues and eigenvectors.Vlasov's method [5] eliminates (1) the problem of selecting shape functions and (2) the solution of the eigenvalue problem. This simplification is achieved by separating the variables and by introducing linear combinations of eigenfunctions of transverse vibrations of beams as shape functions. An additional advantage of this approach is that its accuracy does not deteriorate with higher modes, as is the case for Ritz's method and to a lesser extent for Galerkin's.

Although the required mathematical operations are relatively simple, they tend to be lengthy. Vlasov's method can be extended to cover forced vibrations of plates, but numerical methods are better suited to handle arbitrary plate geometry, boundary conditions and loadings.

Simple estimates for forced vibrations can be obtained by applying the dynamic load factor, based on circular frequencies of the plate. As is the case with the classical solutions of static plate problems, classical methods in plate dynamics serve one important purpose: that the results obtained can be used as benchmarks for all numerical methods (4). Many engineering structures are subject to excitation by unsteady forces of which the spectra extend far up the audio-frequency range (20 Hz–20 kHz).Examples include railway trains, cars, aircraft, ships, gas pipelines, buildings, industrial plant, space rockets, seismic, and explosion waves. The term 'high frequency' implies that the frequency range of concern extends to many times the fundamental natural frequency of the system; ultrasonic frequencies are not considered here. High-frequency vibration is of concern to engineers because of the potential for excessive sound transmission and for fatigue damage.

Not only is knowledge of natural frequencies and modes important from a design viewpoint (to avoid resonance conditions, for instance), but it is also the foundation for forced response calculations. The phenomenon, which occurs whenever the frequency of the excitation is close to the natural frequency of the system, is called beat. Figure (2) illustrates the general nature of this type of vibration. The dashed lines are the envelopes of the amplitudes. The period of the beat increases as the vibration approaches the resonance condition which represents a vibration with amplitude that indefinitely increases with time (5).

1.2 Seismic design requirements

Eurocode 8 (design code) (7) requires that the vibration mode shapes must represent a certain ratio of the total mass. In Eurocode 8, the requirement is that the total sum of strain (ε) in the three directions (x, y, and z) of all vibration mode shapes should be more or equal to 0.9 ($\varepsilon \geq 0.9$), and every mode shape should have a sum of strains larger than 5% in any direction in order to be included in the total sum.

This total sum is called the **seismic equivalence factor** and should be greater or equal to 0.9 for each structure or part of structure to satisfy the seismic requirements. The draft of Iraqi Seismic Design Code (8) recommended the use of proper design of structures to resist the loss of life and human injury.

Based on Eurocode 8, the vibration mode shapes generates equivalent static loads (for each vibration mode shape) which are then applied to the model in the static analysis. Then obtained strains due to the application of these internal forces of each mode shape are summed using the method described in the design code requirements. In order to explain this code requirement; assume that a seismic analysis was performed depending on the Eurocode 8 requirements (7).

First, calculate the frequencies of n vibration mode shapes of the studied structure (or a part of the structure). Second, create a table of calculate the seismic equivalence factor (which is the sum of strains in X, Y, Z directions of all vibration mode shapes studied) as shown in Table (1).

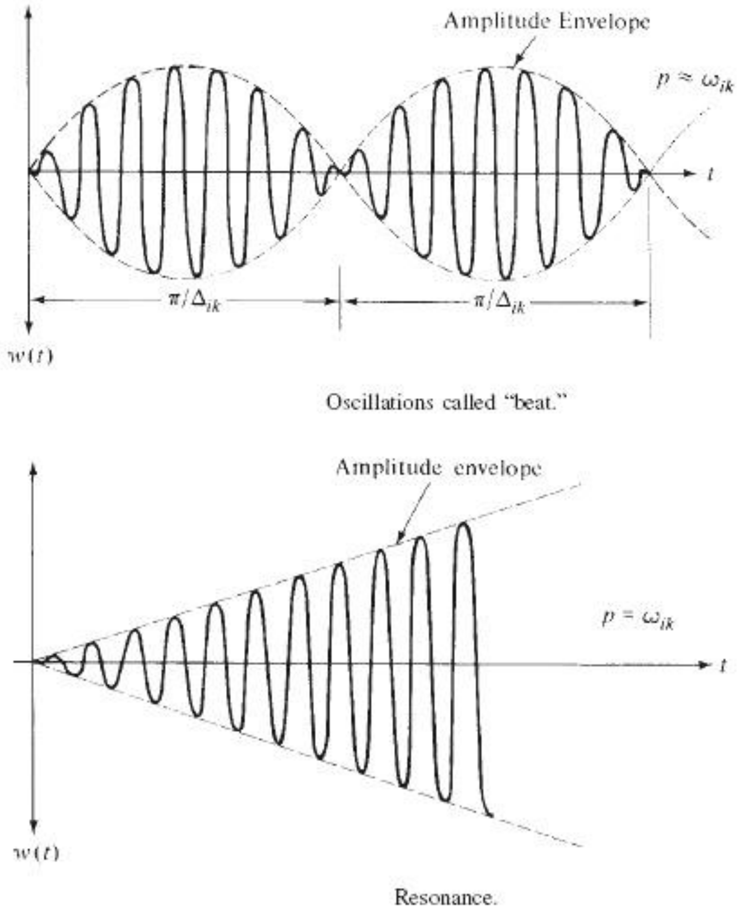


Figure (2): The general nature of resonance condition phenomenon [5].

Table (1): Eurocode 8 method to calculate the seismic equivalence factor [7].

Mode	F (Hz)	ε_X	ε_Y	ε_Z
1	49.56	0	0	0.525
2	108.80	0	0	0
3	109.41	0	0	0
4	193.48	0	0	0.002
5	193.81	0	0	0
6	208.27	0	0	0.149
7	304.91	0	0	0
8	305.98	0	0	0
9	340.43	0	0	0
9/9		**0**	**0**	**0.676**

1.3 Finite element method in dynamic analysis of plate vibration

The versatile Finite element (FEM) based on matrix displacement analysis is extremely well suited to the computerized solution of free vibration problems, associated with complex plate structures [(5)]. Computer solutions based on FEM are more general and flexible than those of other numerical techniques. For example, arbitrary geometry, loads and boundary conditions can be handled with ease. Computer programs, involving complete automation of dynamic analysis of plate, are readily available. A further advantage of the method is that it uses concepts familiar to structural engineers.

Although the derivation of suitable stiffness coefficients, assuring monotonic convergence of the solution to the exact number, is a difficult task, it needs only be done once; that is, the results are reusable. The use of kinematically consistent mass matrices, when high accuracy must be achieved in the vibration analysis, is considered to be mandatory. Disadvantages of finite element techniques in dynamic analysis of plates are (1) without a computer of fairly good capacity, the method is useless; (2) the accuracy of the solution of dynamic problems depends to a certain extent on the boundary conditions; (3) problems with convergence of the solution may occur and (4) operations with large matrices may create special problems [(5)]. The use of FEM is encouraged, provided that well tested stiffness and consistent mass matrices for discrete elements are readily available.The computer program accompanying Szilard book [(5)] is called the Win Plate Primer (WPP program system for static and dynamic analysis of plates.

The WPP is based on the widely used finite element analysis technique and is capable of handling even moderately sized plate problems encountered in everyday engineering practice. Because the versatile FEM was employed, this program system can treat a wide variety of boundary and loading conditions. The WPP system deals with the free vibration problems of plates. This program determines the first mode shape of the freely vibrating system along with the pertinent lowest circular frequency Comparison of these results with their analytically calculated counterparts shows very good agreement.

Again, when analytical solutions were not available, this comparison was based on results obtained by using the independent computer program system. In order to examine the accuracy of available analytical methods used to solve plate vibration problems and the accuracy of WPP computer program [(5)] the following procedure was followed:

1.3.1 Available plate vibration methods

1. Find the lowest circular frequency of a rectangular plate clamped on all edges using the Ritz method and check the result by the Galerkin method. (Figure 2):

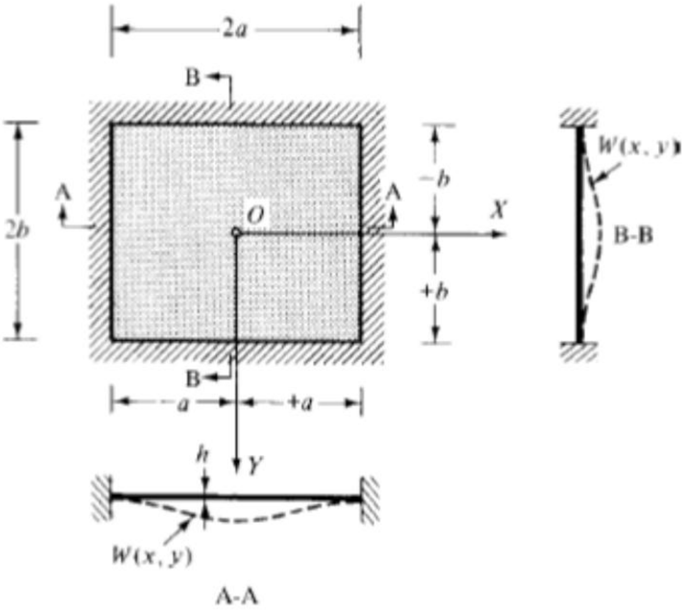


Figure (3): A rectangular plate clamped on all edges.

a. Ritz Method. In selecting a suitable infinite series expression of the deflection that satisfies the geometrical boundary conditions and closely approximates the shape of the first mode of vibration, we use [(5)]:

$$\omega_1 = 4\sqrt{2\left(\frac{1}{a^4} + \frac{2^8}{3^2\times 7^2}\cdot\frac{1}{a^2b^2} + \frac{1}{b^4}\right)}\cdot\sqrt{\frac{D}{\overline{m}}} \quad \text{------------------------------------ (1)}$$

For square plates (a=b):

$$\omega_1 = \frac{9.09}{a^2}\sqrt{\frac{D}{\overline{m}}} \quad \text{--- (2)}$$

where, ω_1 is the lowest circular frequency, D is the plate stiffness, $\overline{m}$ is the mass per unit area, and a, b are the plate dimensions.

This result agrees within 1.04% with the value obtained from more exact computation [(3)].

b. Galerkin's Method. The first modal shape is approximated; again, the variational equation becomes [5]:

$$\omega = 3\sqrt{\frac{7}{2}\left(\frac{1}{a^4} + \frac{4}{7}\frac{1}{a^2b^2} + \frac{1}{b^4}\right)}\sqrt{\frac{D}{\overline{m}}} \qquad (3)$$

where, ω is the lowest circular frequency.

Therefore, the lowest circular frequency for square plates (a=b) is:

$$\omega = \frac{9}{a^2}\sqrt{\frac{D}{\overline{m}}} \qquad (4)$$

The obtained result agrees almost completely with the exact solution of the problem [3]:

$$\omega_{\text{exact}} = \frac{8.9965}{a^2}\sqrt{\frac{D}{\overline{m}}} \qquad (5)$$

Notice that the second approximation of this frequency differs from the above value by 1.175% only. Using more additional terms, the ''exact'' value of the natural frequency may be obtained. The application of the fourth approximation to this problem for the square plate has shown that the refinement over the first approximation does not exceed 1.2%. Thus, the accuracy of the first approximation may be considered as quite satisfactory.

Then, the fundamental natural frequency of axisymmetric vibrations of a solid circular plate with radius R. (the first approximation) the plate is clamped along its boundary may be found as follows[4]:

$$\omega_{11} = \frac{10.33}{R^2}\sqrt{\frac{D}{\rho h}} \qquad (6)$$

where; ω_{11} is the lowest circular frequency, R is the plate radius, D is the plate stiffness, h is the plate thickness, and ρ is the plate mass density.

c. WPP's FEM Method : For a clamped plate shown in Figure (3) with linear distributed load, a comparison between the calculated moments in x and y directions (m_x and m_y) and the deflection in z direction (e_z) at points p_1, p_2, and p_3 using WPP program [5] and analytical solution [3] was carried out as shown in Table (2).

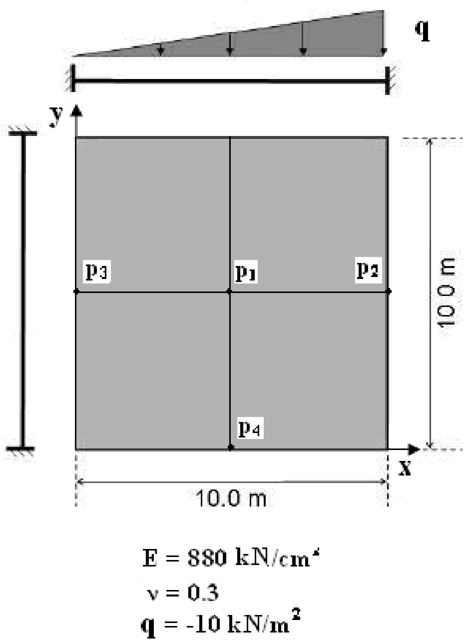


Figure (4): A square plate with clamped edges.

Table (2): A comparison between theory[3] and WPP computer program[5] results of example shown in Figure (2).

Results	Theory	WPP program	Error %
$m_{x,1}$ [kNm/m]	11.50	11.48	-0.17
$m_{y,1}$ [kNm/m]	11.50	11.48	-0.17
$e_{z,1}$ [m]	11.50	11.48	-0.17
$m_{x,2}$ [kNm/m]	33.40	33.23	-0.51
$m_{x,3}$ [kNm/m]	17.90	17.83	-0.39
$m_{y,4}$ [kNm/m]	25.70	25.53	-0.66

2. Objectives and Methodology

2.1 Objectives of the study

The main objective of this study is to determine the most suitable shape, dimensions, thickness, and material properties of glass sheets to be used to manufacture window shields with minimum risk of glass shattering and to reduce death and injury of building's residents. This objective can be achieved by investigating the effect of the following factors:

1. The effect of geometric shape of glass sheets on the natural vibration frequency and seismic equivalency factor.
2. The effect of dimensions of glass sheets on the natural vibration frequency and seismic equivalency factor.
3. The effect of thickness of glass sheets on the natural vibration frequency and seismic equivalency factor.
4. The effect of Young modulus of glass sheets on the natural vibration frequency and seismic equivalency factor.
5. The effect of Poisson ratio of glass sheets on the natural vibration frequency and seismic equivalency factor.

2.2 Methodology

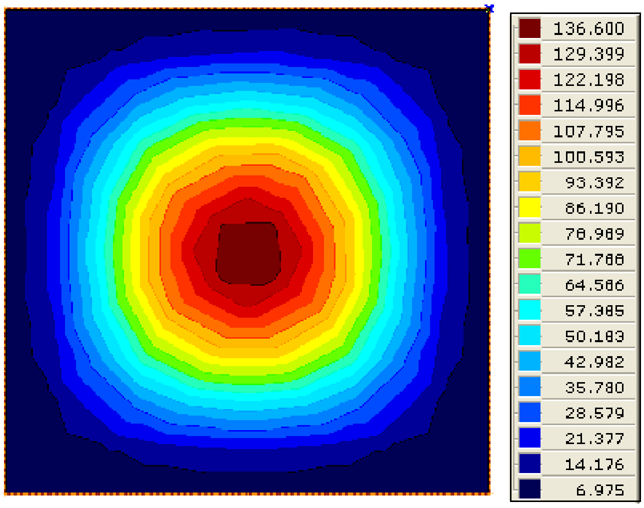
To investigate the effect of these factors, different shapes of glass sheets commonly used were considered, namely, squared, rectangular, triangular, trapezoidal, circular, half-circle, quarter-circle, and 8-sided polygon sheets. Different sizes of these shapes commonly used ranging from 10 cm to 120 cm were considered. Different thicknesses of glass sheets commonly used were considered, namely, 4mm, 6mm, and 8 mm. Two values of the Young modulus of float glass that used to manufacture the glass sheets were used, namely, 50 GPa and 70 GPa. Two values of the Poisson ratio of float glass that used to manufacture the glass sheets were used, namely, 0.22 and 0.23. The boundary conditions of the glass sheets were considered clamped (fixed) around the sheets which simulate the commonly used procedure of fixing these sheets within the window shields.

The mass density of float glass of 2500 kg/m^3 was considered in the analysis. The WPP computer program accompanying Szilard book [(5)] was used to calculate the lowest natural frequencies of different glass plates as shown in Table (3) and Table (4), nine mode shapes were studied for this purpose. The WPP calculated displacements in the three directions were used to calculate the strains in the three directions to find the seismic equivalence factors (samples are shown in Figure (5) and Figure (6)). Table (3) and Table (4) were prepared to summarize the calculated frequencies and seismic equivalency factor (SEF) of all studied plates due to natural vibration of mode shape1.

3. Results

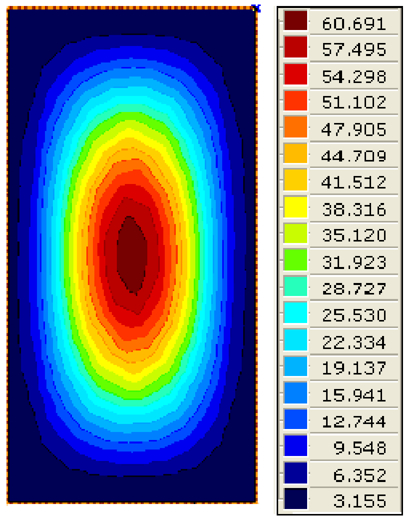
The objective of this study was to find the most suitable shape, dimensions, thickness, and material properties of float glass sheets to be used to manufacture window shields to reduce the risk of death and injury by shattered and flying glass due to traffic, seismic or explosion vibration waves. Comparing the frequencies and seismic equivalence factors of different shapes shown in Table (3) and Table (4), the following results were found:

1. Decreasing the dimensions of a certain geometrical shape of glass sheets and keeping all the other properties constant, not always increases the seismic equivalency factor (SEF). Decreasing the dimensions of the rectangular shape from (20x40 cm) to (15x25 cm) decreases the seismic equivalency factor (SEF) from 0.597 to 0.568. Also, decreasing the dimensions of the square shape from (15 cm) to (10 cm) decreases the seismic equivalency factor (SEF) from 2.837 to 1.107. Therefore, this finding contradict the commonly wide spread idea that decreasing the dimensions of the glass sheets increases their ability to resist vibrations.
2. Increasing the thickness of the glass sheet and keeping all the other properties constant, not always increases the seismic equivalency factor (SEF). Increasing the thickness of the 15 cm square shape from 4 to 6 mm decreases the seismic equivalency factor (SEF) from 2.837 to 0.627. Increasing the thickness of the 15 cm square shape from 6 to 8 mm decreases the seismic equivalency factor (SEF) from 0.627 to 0.496. Therefore, this finding contradict the commonly wide spread idea that decreasing the dimensions of the glass sheets increases their ability to resist vibrations.
3. For the range of dimensions studied, the trapezoidal and the 8-side polygon shapes did not achieve the minimum acceptable value of the seismic equivalency factor (SEF) of 0.9 as recommended by Eurocode 8[(7)].
4. For the range of dimensions studied of the half-circle shapes, only one size of 10 cm radius and thickness of 8 mm of the half-circle shape passed the requirements of Eurocode 8[(7)].
5. Increasing Young modulus of elasticity of the float glass material from 50 to 70 GPa and keeping all the other properties of the glass sheet constant, little or no difference was noticed on the results of frequencies and seismic equivalency factors (SEF's) shown in Table (3) and Table (4).
6. Increasing Poisson's ratio of the float glass material from 0.22 to 0.23 and keeping all the other properties of the glass sheet constant, little or no difference was noticed on the results of frequencies and seismic equivalency factors (SEF's) shown in Table (3) and Table (4).



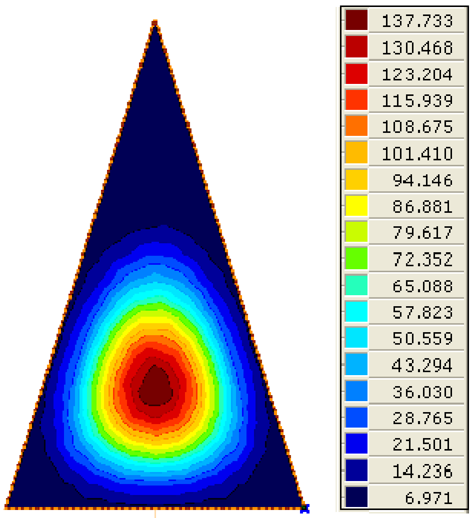
Square sheet a=18 cm, t= 8 mm

E= 50 GPa, μ=0.22



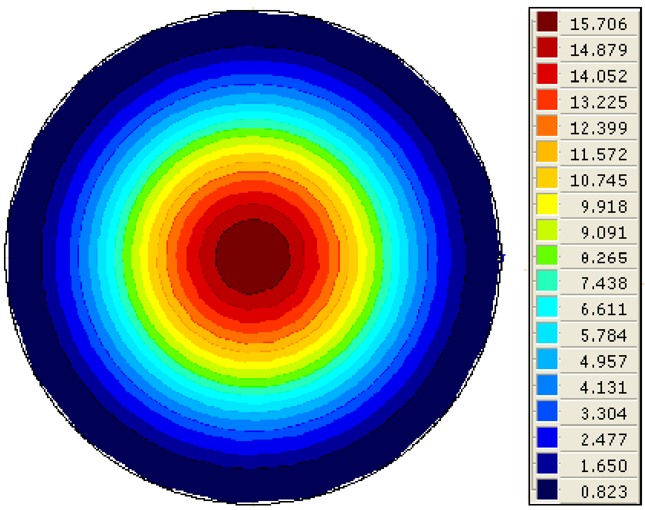
Rectangular sheet a=20 cm, b=40 cm

t= 8 mm, E= 50 GPa, μ=0.22



Triangular sheet a=15 cm, b= 25 cm

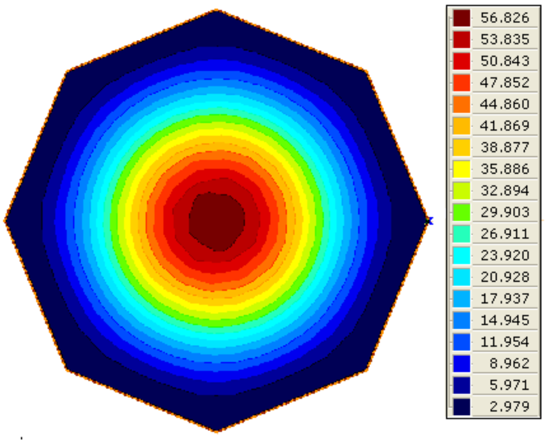
t= 8 mm, E= 50 GPa, μ=0.22



Circular sheet r=60 cm, t= 8 mm

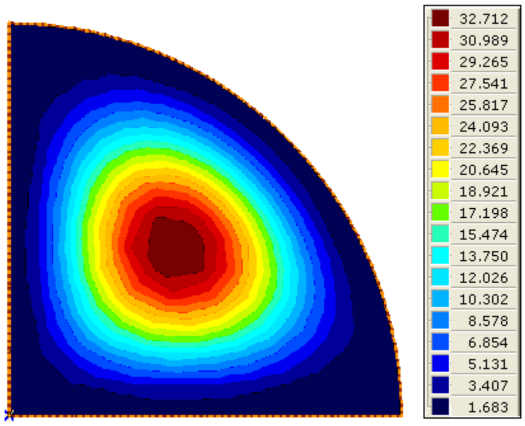
E= 50 GPa, μ=0.22

Figure (5): A sample of the calculated stains in the z direction of different clamped glass sheets due to natural vibration using mode shape1.



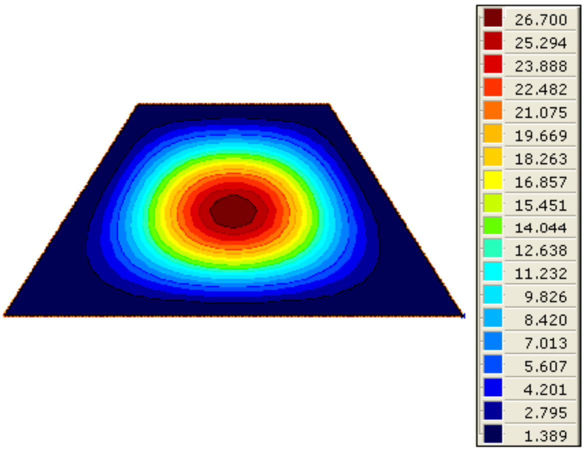
8-side Polygon sheet r=20 cm, t= 6 mm

E= 50 GPa, μ=0.22



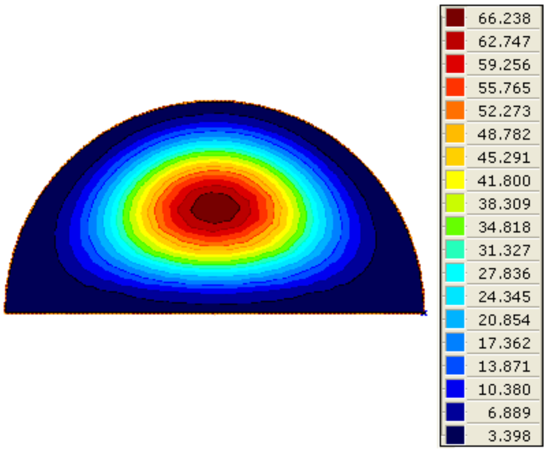
Quarter-circle sheet r=60 cm, t= 8 mm

E= 50 GPa, μ=0.22



Trapezoidal sheet a=15 cm, b= 25 cm

t= 8 mm, E= 50 GPa, μ=0.22



Half-circle sheet r=50 cm, t= 6 mm

E= 50 GPa, μ=0.22

Figure (6): A sample of the calculated stains in the z direction of different clamped glass sheets due to natural vibration using mode shape1.

4. Conclusions and Recommendations

4.1 Conclusions

The main conclusions of this research can be summarized as follows:

1. Increasing the thickness of floating glass sheet and keeping all the other properties and dimensions constant to increase its resistance to exerted vibrations is not always a true rule. This finding contradicts with the commonly popular idea that increasing the thickness of glass sheets reduces the shattering risk. Each case should be studied separately. For practical reasons, Table (5) and Table (6) were prepared to summarize the recommended shape, dimensions, and thickness of commonly used float glass sheets for engineers and technicians involved in the design and manufacturing of window shields.
2. Decreasing the dimensions of floating glass sheet and keeping all the other properties and dimensions constant to increase its resistance to exerted vibrations is not always a true rule. This finding contradicts with the commonly popular idea that decreasing the dimensions of glass sheets reduces the shattering risk. For practical reasons, Table (5) and Table (6) were prepared to summarize the recommended shape, dimensions, and thickness of commonly used float glass sheets for engineers and technicians involved in the design and manufacturing of window shields.
3. For the range of dimensions studied, the trapezoidal and the 8-side polygon shapes did not achieve the minimum acceptable value of the seismic equivalency factor (SEF) of 0.9 as recommended by Eurocode 8(7). Therefore, they are not suitable for the design and manufacturing of window shields. The same result is true for half-circle shape, in which only the sheet radius of 10 cm met the Eurocode 8(7) requirements.
4. Increasing Young modulus of elasticity (from 50 – 70 GPa) and Poisson's ratio (from 0.22 – 0.23) of glass sheet had little or no effect on the magnitude of frequencies and seismic equivalency factor (SEF) of the studied shapes and sizes shown in Table (3) and Table (4).

4.2 Recommendations

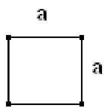
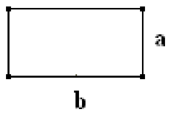
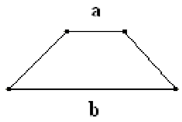
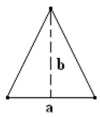
1. The results of this research are quiet important for both building occupants and manufacturers of glass window shields. After a comprehensive literature review, neither similar study nor results were found. These results hopefully will reduce the damaging effects of traffic vibrations, damaging windstorms, earthquakes, break-ins, and terrorism explosions due to flying glass.

2. It was noticed by the authors during their visits to different developed and developing countries that the sizes and thicknesses of glass sheets used for manufacturing window shields are not designed to reduce the effects of vibrations. Table (5) and Table (6) are considered as guidelines for practical purposes, different dimensions and properties of glass sheets should be analyzed for vibrations separately to reduce the risk of glass shattering and to reduce death and injury of building's residents.

3. Laboratory and field tests are required to enhance the results of this research.

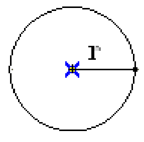
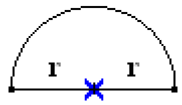
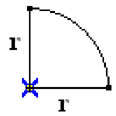
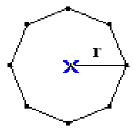
4. Future research is required to increase the resistance of glass sheets to exerted vibration by improving the method of fixing the glass sheets into the surrounding frame (boundary conditions).

Table (3): Lowest natural frequency (F) and seismic equivalency factor (SEF) of different glass sheets with different dimensions (a and b) and different thickness (t).

Glass Plate Properties			Glass Plate Shape							
			Square 		Rectangular 		Trapezoidal 		Triangular 	
a (cm)	b (cm)	t (mm)	F (Hz)	SEF*	F (Hz)	SEF*	F (Hz)	SEF*	F (Hz)	SEF*
50	120	4	143.64	0.625	127.65	0.551	167.94	0.59	285.14	0.453
20	40	4	840.66	0.66	557.24	0.597	371.12	0.627	1238.76	0.5
15	25	4	1395.84	*2.837*	106.69	0.568	2403.03	0.474	2299.52	0.543
10	30	4	2185.06	*1.107*	3255.61	*2.515*	938.48	0.605	3784.48	0.752
50	120	6	201.31	0.638	169.18	0.557	123.19	0.618	390.88	0.486
20	40	6	1194.8	0.68	807.37	0.606	540.72	0.638	1906.92	0.519
15	25	6	2032.65	0.627	1515.37	0.577	3309.46	0.49	3258.86	0.491
10	30	6	4491.22	*3.449*	3914.86	*3.834*	1318.51	0.633	5254.39	0.514
50	120	8	259.94	0.65	211.62	0.564	161.35	0.621	495.93	0.502
20	40	8	1548.75	0.526	1053.31	0.576	709.85	0.65	2359.78	*1.219*
15	25	8	2669.8	0.496	1912.63	*0.951*	4159.89	0.509	4155.08	0.51
10	30	8	5649.2	*4.824*	4549.66	*4.006*	1694.07	0.655	7028.93	*3.569*

Note: Bold, Italian, and underlined numbers highlight the acceptable SEF by Eurocode 8[7].

Table (4): Lowest natural frequency (F) and seismic equivalency factor (SEF) of different glass sheets with different dimensions and. thickness (t).

Glass Plate Properties		**Glass Plate Shape**							
		Circular Plates 		**Half Circle Plates** 		**Quarter Circle Plates** 		**8-side Polygon Plates** 	
R (cm)	**T (mm)**	**F (Hz)**	**SEF***	**F (Hz)**	**SEF***	**F (Hz)**	**SEF***	**F (Hz)**	**SEF***
60	4	24.9	0.676	151.8	0.569	130.21	0.621	34.1	0.668
50	4	146.55	0.669	105.61	0.598	193.89	0.61	73.36	0.628
20	4	233.7	0.654	402.85	0.601	1206.5	0.652	248.26	0.689
10	4	3903.71	***2.029***	2386.76	0.476	4555.05	***4.236***	1053.27	0.717
60	6	37.26	0.675	179.72	0.573	186.44	0.631	46.91	0.67
50	6	217.93	0.676	153.61	0.599	273.18	0.597	87.55	0.637
20	6	346.7	0.667	590.11	0.632	1676.06	0.483	368.04	0.694
10	6	5085.94	***4.458***	3484.58	0.488	6218.16	***4.572***	1506.21	0.547
60	8	49.56	0.676	210.57	0.578	242.51	0.643	59.95	0.673
50	8	288.43	0.684	201.04	0.602	352.85	0.616	103.87	0.647
20	8	457.68	0.68	774.79	0.615	2133.98	0.555	487.36	0.698
10	8	6827.74	***5.292***	4534.01	***4.045***	7659.65	***4.946***	1954.18	0.539

Note: Bold, Italian, and underlined numbers highlight the acceptable SEF by Eurocode 8(7).

Table (5): Recommended shapes, dimensions, and thicknesses of different glass sheets.

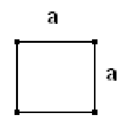
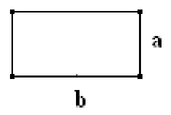
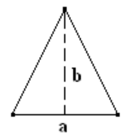
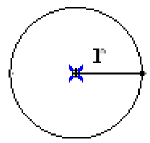
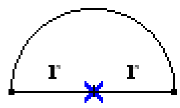
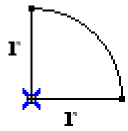
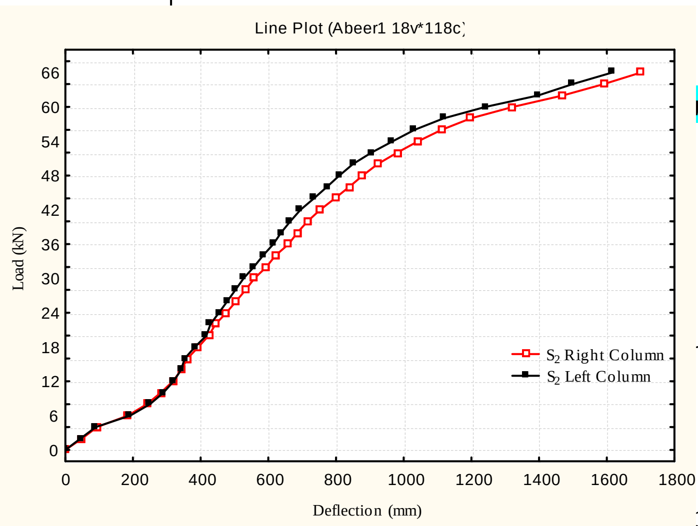
Glass Plate Properties			Glass Plate Shape					
			Square 		Rectangular 		Triangular 	
A (cm)	B (cm)	T (mm)	F (Hz)	SEF	F (Hz)	SEF	F (Hz)	SEF
15	25	4	1395.84	*2.837*	-	-	-	-
10	25	4	-	-	3350.91	*1.64*	-	-
10	30	4	2185.06	*1.107*	3255.61	*2.515*	-	-
10	40	4	-	-	3296.51	*1.292*	-	-
10	30	6	4491.22	*3.449*	3914.86	*3.834*	-	-
12.5	-	6	2909.48	*2.033*	-	-	-	-
10	40	6	-	-	3935.2	*1.632*	-	-
18.2	-	6	1430.87	*0.93*	-	-	-	-
20	40	8	-	-	-	-	2359.78	*1.219*
18.2	-	8	1854.64	*2.927*	-	-	-	-
15	25	8	-	-	1912.63	*0.951*	-	-
12.5	-	8	3770.29	*4.546*	-	-	-	-
10	30	8	5649.2	*4.824*	4549.66	*4.006*	7028.93	*3.569*

Table (6): Recommended shapes, dimensions, and thicknesses of different glass sheets

Glass Plate Properties		Glass Plate Shape					
		Circular Plates 		Half Circle Plates 		Quarter Circle Plates 	
R (cm)	**T (mm)**	**F (Hz)**	**SEF**	**F (Hz)**	**SEF**	**F (Hz)**	**SEF**
10	**4**	**3903.71**	***2.029***	**-**	**-**	**4555.05**	***4.236***
10	**6**	**5085.94**	***4.458***	**-**	**-**	**6218.16**	***4.572***
10	**8**	**6827.74**	***5.292***	**4534.01**	***4.045***	**7659.65**	***4.946***



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