NOISE INSULATION OF LIGHTWEIGHT ELEMENTS

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ABSTRACT

The growing number of sources of noise (road, railways, etc.) within urban agglomerations and their surrounding neighborhoods has risen in the past few years. The consequent steady increase in noise levels has led to deteriorating standards of acoustic comfort in dwellings. In the case of the lightweight elements comprising the façade of a building this is quite important.

Acoustic comfort inside dwellings can be improved in several ways, particularly through sound insulation of the construction elements used in the façade of a building. The least efficient separation elements, generally, are glazed openings.

The main purpose of this work, therefore, is to develop prompt and reliable methodologies that permit: (1) prediction of acoustic insulation provided by glass alone, where the variables are the area of the glass pane, its thickness, the number of panes and the thickness(es) of the air chamber(s), and (2) characterization of the acoustic behavior of the glass in the frame, bearing in mind the type of frame.

This work is complemented by laboratory experiments and the results correlated with the critical and resonance frequencies of the glass panel itself, and the resonance frequencies of the testing rooms.

1. INTRODUCTION

The past few years has seen the development and increasing application of light structural systems in the construction of buildings. In some cases, façades and interior divisions are often made from lightweight construction elements, such as glazed components. This work is concerned with the study of the insulation provided by such elements.

The insulation conferred by a separation element depends fundamentally on its mass, frequency and angle of incidence of sound, on the number of panels and the dimensions of the air chambers (in the case of multiple elements), the rigidity of the element and its area.

This work has been developed in two distinct phases. A first phase was concerned only with the glass panel (or panels), where there were variations in its area, thickness, number of panels and thickness of the air chamber. A second phase studied the same type of glass panel incorporated into two different kinds of frame.

This study first presents a group of models for predicting insulation, followed by a description of laboratory tests. The experimental results are then compared with those of the predictive models.

2. ACOUSTIC INSULATION

The transmission of sound energy in a separation element proceeds by the vibration of the element, with the mass and sound frequency being relevant variables. As the mass of the element increase, so does insulation, as a result of increasing forces of inertia. When the frequency of incident sound on an element that maintains the same mass is increased, the vibration power of the element decreases and greater dissipation of sound energy is observed, in addition to the consequent rise in acoustic insulation.

Besides these two variables, there are others that may affect the acoustic insulation of a separation element. These include the angle of incidence of the waves, the existence of weak points in the insulation, rigidity, damping of the element and, in the case of multiple elements, the number of panels and their individual characteristics and separation. In an actual situation, the transmission of sound between two contiguous rooms depends not only on the separation elements, but also on the connections between the surrounding elements, and on the way in which propagation proceeds inside the emitting and receptor rooms. In this process the vibration modes themselves, within each of the rooms, determine the manner of propagation.

The mathematical description of the phenomena involved in acoustic insulation is thus very complex. Studies such as these are usually conducted with variations in only a limited number of the variables in question. This results in a set of simplified insulation predictive models. Some of these models are described below since, even though they are simplified, the nevertheless allow us to understand the acoustic phenomena involved.

2.1. Models Predicting Insulation in Single Elements

Taking a simple separation element to comprise a group of juxtaposed masses with the potential for independent displacement, and assuming damping forces to be null, sound insulation increases in a form that is close to linear, with rises of 6 dB for each doubling of the mass or for each doubling of the sound frequency. This variation follows a law, known as the Law of Theoretic Mass or the Law of Theoretical Frequency.

However, an element's rigidity and damping affect its vibration mode, making a greater transmission of sound energy possible in the natural vibration frequencies of the element. Experimental analysis shows that average growth in insulation per duplication of mass is generally less than 6 dB. In reality, different authors have defined quite different average insulation curves, but always with values lower than 6 dB for a doubling in mass.

In this study, the prediction of insulation has been made using a mixed model, based on the model suggested by Meisser [4], which takes an experimental law of mass, with an increase of 4 dB per doubling of mass and an insulation against 500 Hz of 40 dB per doubling of frequency. To this last, dips in insulation in the natural vibration frequencies were applied and, for frequencies outside these zones, the law of theoretical frequency was adjusted, with an inclination of 6 dB per eighth, as shown diagrammatically in Figure 1.

There are two kinds of phenomena in single elements that can cause significant dips in insulation. One occurs through the transversal movement of the panel in pure flexion, generally at low frequencies. The other is due to the longitudinal movement of plane bending waves along the panel, occurring usually at higher frequencies, and is more relevant to thin panels that have a large area.

2.1.1. Eigenfrequencies due to Transverse Vibration of the Panel in Pure Flexion

Any flat element, by virtue of its density, the module of elasticity, the Poisson coefficient, its thickness, length and width and manner of support, presents a group of eigenfrequencies. In the specific case of glass panes, the eigenfrequencies' properties, relative to the pane's transversal movement by pure flexion, can be determined from a study of thin and finite plates, if we consider, as is usual, that their thickness, relative to their length and width, is negligible.

Taking a thin and finite plate with a simple support all round, the frequency properties through transversal flexion, in Hz, can be obtained from Eq.1,

$$f_{nm} = \frac{\pi}{2} \cdot \left(\frac{n^2}{a^2} + \frac{m^2}{b^2} \right) \cdot \sqrt{\frac{D}{\rho . h}} \tag{1}$$

where n and a are respectively the number of the eigen mode and the dimension (m) according to x; m and b the number of the vibration mode and the dimension (m) according to y; ρ the mass by unit of volume of the material which the plate is made of (Kg/m^3); h the thickness of the plate (m) and D the rigidity of the plaque (kN.m).

The number of eigen modes obtainable from Eq. 1 is theoretically infinite. However, the modes which significantly influence acoustic insulation are the first vibration modes.

2.1.2. Eigenfrequencies due to Longitudinal Movement of Plane Bending Waves

If a plate can behave as if it were infinite, it is useful to investigate the conditions in which plane harmonic waves can be propagated.

If we take the propagation velocity of the longitudinal flexion waves along the plate as equal to the speed of the sound projected on the plane of the panel, we get Eq.2.,

$$\omega = \left(\frac{c}{\sin \phi} \right)^2 \cdot \sqrt{\frac{\rho . h}{D}} \tag{2}$$

where ω is the angular frequency (rad.), c is the velocity of sound propagation in the air (m/s) and ϕ is the incidence angle of the sound relative to a direction perpendicular to the element.

The critical frequency (f_c) is designated as being that which corresponds to a speed of the sound is tangent (ϕ=90), from which we get:

$$f_c = \frac{c^2}{1,8138. h} \cdot \sqrt{\frac{\rho .(1 - \nu^2)}{E}} \tag{3}$$

where υ is the Poisson coefficient of the material and E the Young module (N/m^2).

2.2. Models Predicting Insulation in Double Elements

The sound insulation of a single separation element may increase if the element is divided into two panels separated by an air chamber. However, a double separation element displays other dips of insulation, especially in the resonance frequencies of the mass and air chamber units and in the resonance frequencies arising from successive reflections (for stationary waves) in the air chamber. In addition, dips occur in the critical frequencies of each panel, and eventually in the first Eigenfrequencies due to transversal vibration, through panel flexion, should these present values that are not very low.

As with the single elements, and in this case more obviously, the application of analytical models to double elements usually leads to results that are very different from experimental ones. In these conditions, for such elements, a predictive method based on experimental analysis is also used. It is founded on a frequency law with a 6 dB inclination per eighth, to which insulation dips are applied in their various natural vibration frequencies and, for zones outside these frequencies, an adjustment is made to a law with an 8 dB inclination per eighth, as shown in Figure 1.

In this method the experimental law of mass is equal to that referring to single elements with the same total mass with the addition of a constant (Dif2), which depends fundamentally on the thickness of the air chamber and on the mass of each panel. In prevailing situations this increment is approximately 4 dB, for small air chambers of the order of 2 to 4 cm and panels with large, and similar masses.

2.2.1. Eigenfrequencies due to Movement of the Mass-air-mass Unit

The calculation of resonance frequencies for the mass-air chamber unit can be done by studying the natural frequency of the system constituted by the panel with mass *m1*, air chamber with thickness *d* (to which rigidity K corresponds) and the panel with mass *m2*. If the rigidity and damping of the panels are disregarded, as is the case with single panels, the problem is simplified and the determination of the resonance frequency (in rad/s) can be done by means of Eq. 4.

$$\omega = \sqrt{K.\frac{m_1 + m_2}{m_1 . m_2}} \tag{4}$$

For normal humidity conditions and at a temperature of 20ºC, and for most situations, with diffuse incidence, an approximate frequency resonance (in Hz) can be obtained from Eq. 5.

$$f_{res} = 84.\sqrt{\frac{1}{d}.\left(\frac{1}{m1} + \frac{1}{m2}\right)} \tag{5}$$

Where the sound frequency incident on the double element is lower than the resonance frequency of the unit, the air chamber has a negligible effect and the element functions in a manner similar to that of a single element with the same total mass. However, if the frequency of the sound incident on a double element is higher than the resonance frequency, the air chamber absorbs part of the sound energy and the index of sound reduction is greater than that observed in a single element having the same mass. In these conditions it is important that this natural frequency has relatively low values, generally below 100 Hz, which, in lightweight panels and in normal glass, are possible for quite thick air chambers.

2.2.2. Eigenfrequencies due to Multiple Reflections in the Air Chamber

Successive reflections may occur inside the air chamber in the transmission of sound by a double element with two parallel panels, giving rise to stationary waves. This phenomenon arises when the thickness of the air chamber is equal to a value that is a multiple of half the length of the wave (n.λ/2), causing the appearance of n resonance frequencies, obtained by Eq. 6.

$$f_1 = \frac{c}{2.d} \quad , \; f_2 = 2.\frac{c}{2.d} \quad , \ldots , \; f_k = k.\frac{c}{2.d} \tag{6}$$

Thus, for the first resonance frequency to be outside the zone sensitive to the human ear, the distance between the panels should be small. However, for lightweight panels with small air chambers, the dips in insulation due to resonance of the unit may be significant, being more important, generally, that those caused by multiple reflections within the air chamber.

2.3. Predictive Insulation Models for triple elements

The index of sound reduction for a triple separation element, like that observed for double elements relative to singles ones, is usually higher than the corresponding one for double elements with the same mass. Indeed, for most current situations, the greater the number of panels, separated by air chambers or by other materials arranged in the form of a sandwich, the greater the expected sound insulation. As has been done for double elements, this is presented here diagrammatically, in a simplified form, to permit prediction of the insulation curve for triple elements. This outline is like that given for double elements, but with a greater number of dips in natural frequencies and an experimental frequency law of 10 dB per eighth, as shown in Figure 1.

The natural frequencies involved in the sound transmission in triple elements are of the same type as those found in the case of double elements, and may be calculated by the formulae already indicated, with the exception of the unit's resonance frequency. In this case there are two resonance frequencies for the unit mass *m1*, air chamber with thickness *d1*, panel with mass *m2*, air chamber with thickness *d2* and panel with mass *m3*, and these can be determined by means of Eq. 7.

$$[m_1 . m_2 . m_3].(\omega^2)^2 - [K_1 . m_3 .(m_1 + m_2) + K_2 . m_1 .(m_2 + m_3)].(\omega^2) + [K_1 . K_2 .(m_1 + m_2 + m_3)] = 0 \qquad (7)$$

with $K_1 = \frac{\rho . c^2}{d_1}$ *e* $K_2 = \frac{\rho . c^2}{d_2}$

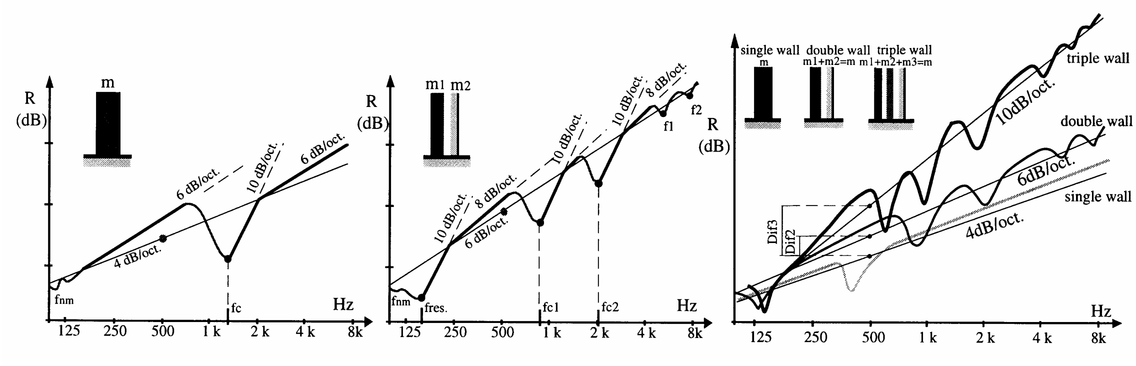


Fig. 1 – Insulation curves predicted for single, double and triple elements

2.4. Eigenfrequencies of a Parallelepiped Chamber

The precise characterisation of a sound field which is established in an enclosed space, in the presence of a sound source, is not an easy task. It requires complex physical-mathematical treatment and involves variables which are difficult to quantify, such as the characteristics of sound energy dissipation in the air and the surrounding medium, and those of the sound sources.

There are three distinct ways of analysing this type of problem. First, there is a rigorous approach, is based on wave theory, from which it is possible to determine, among other aspects, the vibration frequency properties of enclosed spaces, which determine the form of sound propagation. The second is based on the geometric theory of spaces, and introduces major simplifications and is generally only valid for very high frequencies or large spaces where the diffuse field is practically non-existent. A third way, also simplified, is based on statistical theory, and can be applied to most enclosed spaces, but it does not always lead to very accurate results, especially for very low frequencies. In this study, we only used wave theory, which made it possible to determine the eigen mode of the chambers.

The vibration properties of an enclosed space, owing to the formation of stationary waves, depends on the shape and geometrical dimensions of the elements of the surroundings as well as of the boundary conditions throughout the entire surroundings. Taking a parallelepiped chamber, with dimensions L_x, L_y and L_z, the equation of equilibrium which governs the waves of sound pressure in the chamber (Helmholtz Equation) leads to Eq. 5, which allows us to obtain the eigenfrequencies of the chamber,

$$f_{nmp} = \frac{c}{2} \cdot \sqrt{\frac{n^2}{L_x^2} + \frac{m^2}{L_y^2} + \frac{p^2}{L_z^2}} \qquad (8)$$

where c is the velocity of sound propagation in the air (m/s); L_x, L_y and L_z are the dimensions of the chamber (m); e, n, m and p are the whole numbers (0, 1, 2, etc) which represent the no. of the frequency property according to x, y and z respectively.

If we analyse Eq. 5, it can be seen that the first eigenfrequencies for large spaces are registered for very low frequencies, while for small spaces these frequencies are high.

3. EXPERIMENTAL

3.1. Preparation of the Chambers

The experimental work envisaged the study of the behaviour of glazed openings when subjected to variations in acoustic pressure. This work had to be carried out in acoustic conditions which would permit control of the variables in play. To achieve this, it would be desirable to have acoustic chambers specially prepared for this purpose. Given that such conditions did not exist, the laboratory work was begun by adapting an existing chamber so that it had the minimum conditions necessary for the work. The first phase consisted of construction work to build two contiguous chambers which would guarantee a high degree of acoustic insulation (if we disregard the opening for testing samples). In a second phase, some absorbent elements were placed in the receiving chamber, as a way of diminishing the reverberation of that chamber.

3.2. Construction of the Chain of Measurement

The detection by experiment of the eigenfrequencies properties was possible by capturing the vibration of the glass and the sound in both test chambers. The chain of measurement used consisted of three channels for capturing data: one channel to capture the accelerations of the glass, and two which originated from two microphones, one in the emitting chamber, and the other in the receiving chamber (measuring simultaneously).

The chain of measurement is shown in Fig. 2, schematically and in a simplified form.

3.3. Experimental Conditions

After complete calibration of the chain of measurement, the test samples were positioned, and the points of capture of the vibrations of the glass and the sound pressure in each chamber selected. The standardised sound source was then placed in the emitting chamber. Neither the sound nor the vibrations could be measured correctly if the position of the transducers and the excitation source were not chosen with great care. For example, an accelerometer placed close to the centre of the glass does not detect the vibration mode properties of transversal vibration, occurring through flexion of the plates, for which this point does not move.

The whole system of capturing sound and vibrations, as well as the position of the transducers and the source, is shown diagrammatically in Fig. 2. The capture of sound pressure was done simultaneously in both chambers, at ten positions distributed around the interior of each chamber.

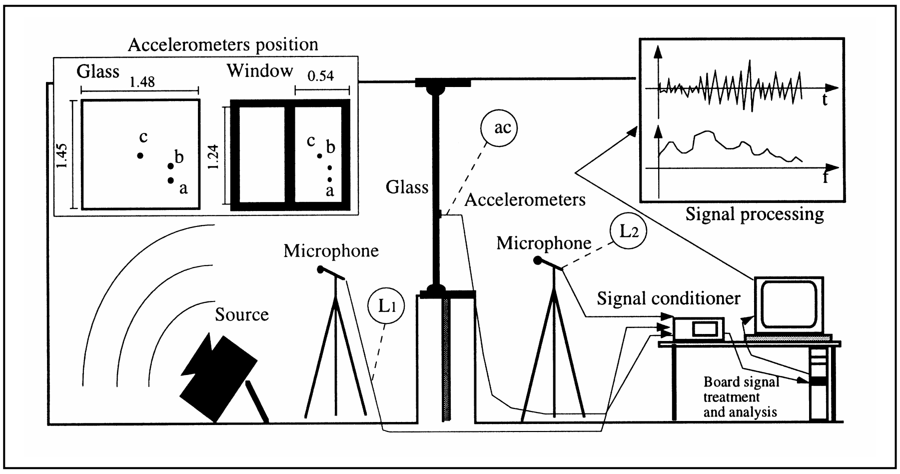


Fig. 2 - Diagram of the rooms and the acquisition system used

The results obtained from the system of capture shown in Fig. 2 were first processed using the program "VIEWDAC", with the Fourier Fast Transformation application (FFT) which allowed the data to pass to the frequency domain. Later we used programs developed for dividing the spectrum into frequency bands and determining a mean curve of acoustic insulation.

3.4. Type of Test Samples

This work looks at two distinct types of tests: one in which only the glass panels, single or multiple, are tested, and the other in which the same kind of glass is tested, but inside two different window frames.

In the tests on glass without frames, two types of glass were used: one 4mm thick and the other 8mm thick, combined with different forms and different air chambers, as shown in Table 1.

Table 1 – Glazed Solutions Tested (without frame)

Type of Glazed Solution	Sample	8mm Thickness	4mm Thickness	4mm Thickness	Air Chamber 1 (in mm)	Air Chamber 2 (in mm)
Single	1	-	+	-	-	-
	2	+	-	-	-	-
Double	3	+	+	-	2,5	-
	4	+	+	-	10	-
	5	+	+	-	20	-
	6	+	+	-	35	-
	7	+	+	-	50	-
	8	+	+	-	100	-
	9	+	+	-	200	-
	10	-	+	+	10	-
	11	-	+	+	50	-
Triple	12	+	+	+	10	10
	13	+	+	+	100	10
	14	+	+	+	100	50

- Not used; **+** used

The tests on glass inside frames were carried out on two kinds of frames, both with single glass panels 8mm thick. With this type of test we were only attempting to observe the dips of insulation introduced by the frame itself. The first solution consisted of using a "good quality" frame with very good seals, and with two opening panes and central reinforcement. The second solution, frankly worse in terms of acoustics, comprised a frame with two sliding panes, like most of those used in buildings in Portugal today.

4. RELEVANT RESULTS

The acoustic behaviour of the glazed solutions tested in the laboratory depended on the way in which the sound propagated, in both the room where the sound was emitted and the room where it was received. In this way the dynamic modal parameters defining the dynamic behaviour of both rooms were characterized before measuring the acoustic insulation. This analysis was based on the determination of the eigen modes of vibration, both analytically and experimentally.

The presentation of the results of acoustic insulation gives those resulting from the experimental studies and those expected from the simplified models described in point 2, simultaneously, on the same graph. These graphs also show the acceleration spectra relative to the movement of the panels (at the central point of each panel), as well as indicating the eigenfrequencies for each solution tested, calculated analytically. This kind of presentation makes it possible to compare the predicted insulation with that obtained experimentally. It also localizes the dips of insulation resulting from eigenfrequencies of the panels and of the rooms.

Figure 3 gives the insulation curves (predicted and experimental), the critical frequency (calcualted analytically) and the vibration of the panel at its central point for single glazed solution, 8mm thick.

Analysis of Figure 3 enables us to verify a displacement between the expected results and the experimental ones. It shows a global experimental insulation about 3 dB smaller and a slope of the frequency law, outside the zones where the dips in insulation were bigger, close to 5 dB/octave.

For the several double glazed solutions tested, we only present here the results of the two different solutions with 8+4mm glass: one with a 10mm air chamber (Figure 4) and the other with 100mm air chamber (Figure 5).

Fig. 3 – Insulation and vibration curves in single 8mm glazed solution

Fig. 4 – Insulation and vibration curves in double 8+(10)+4 mm glazed solution

Analysis of Figure 4 enables us to see another displacement between the experimental and theoretical results, especially for low frequencies. In the case of the second solution, the dislocation between the results is much smaller, with an experimental insulation higher than the predicted one being observed at medium frequency levels.

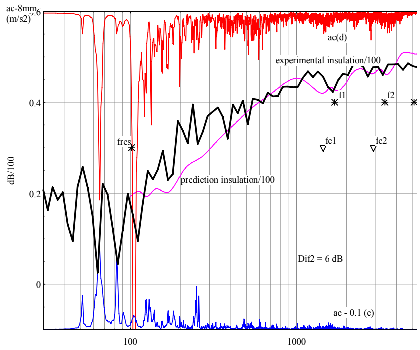


Fig. 5 – Insulation and vibration curves in double 8+(100)+4 mm glazed solution

Fig. 6 – Global insulation indices in double 8+4mm glazed solutions, with different air chambers

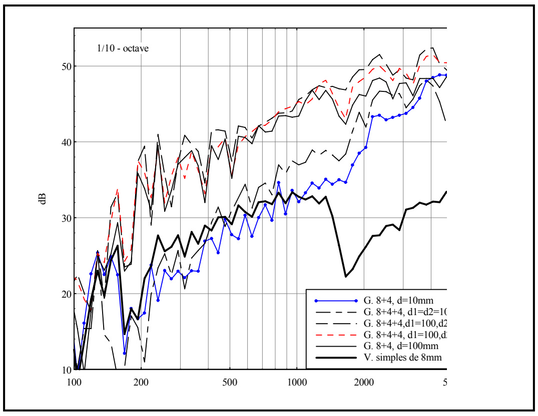
As can be verified from the analysis of Figure 6, there is a range of air chamber thicknesses where insulation is minimal. Curiously, these are the dimensions currently used in air chambers of glazed solutions. Global insulation improves as the thicknesses decrease significantly or increase to values close to or greater than 50mm.

Fig. 7 – Insulation and vibration curves in triple 8+(10)+4+(10)+4 mm glazed solution

Fig. 8 – Insulation and vibration curves in triple 8+(100)+4+(10)+4 mm glazed soltuion

Regarding the triple glazed solutions tested, only the results of the different triple panels 8+4+4 with two air chambers, 10mm thick (Figure 7), and the solution with the first air chamber 100mm thick and the second 10mm thick (Figure 8) are presented.

Figures 9 and 10 present the experimental insulation curves in bands of 1/10 octave, obtained for several kinds of different glazed solutions.



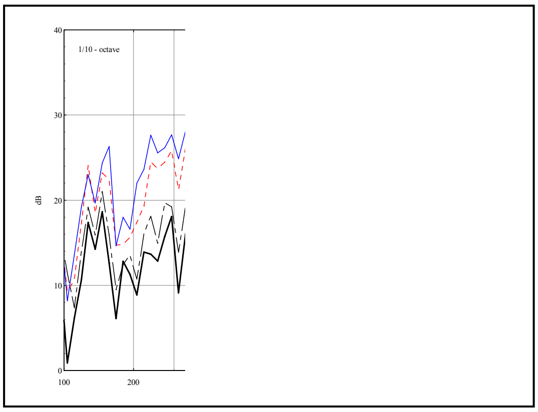


Fig. 9 – Insulation curves for single, double and triple glazed solutions

Fig. 10 – Insulation curves for single 8mm glazed solutions, with and without frame

Comparing the curves in Figure 9, one may find good agreement between the solutions for double 8+(100)+4 and triple 8+(100)+4+(10)+4 glazed solutions. It is also possible to see some agreement between single glazed solutions and multiple glazed solutions with small air chambers, especially for frequencies lower than 1000 Hz. Comparison between the insulation curves found with an opening frame and without a frame shows a much smaller coincidence dip in the situation without a frame. This difference may be explained by the smaller area of each panel used in the solution with a frame. Indeed, the coincidence dip is small when the ratio between the area and the thickness of the panel decreases, so the propagation of longitudinal plane waves is less. It is also possible to observe that only a small gap is needed to produce a big insulation dip.

5. CONCLUSIONS

Analysis of the results obtained reveals that, for very low frequencies, there is a marked dependency between the insulation curves for the different glazed solutions tested, and the form of propagation inside the receptor room. These results are not very surprising since the dimensions of each test room are significantly smaller than what is desirable (around 30 cubic metres for the receptor and 24 cubic metres for the emitting room).

The experimental results allow us to conclude that the predictive models described in this work are reasonably close to the experimental results for many of the simple and double glazed solutions. The biggest differences between the predicted and experimental insulation usually occur for multiple glazed panels that present relatively high global resonance frequencies (above 150 Hz). In the case of triple glass the experimental results are significantly different from those resulting from the models given.

Regarding the glazed solutions without frames, it can be concluded that the double glass only exhibits better insulation behaviour than single panels if the air chambers are close to or greater than 50mm thick, or if the air chambers are very small. With respect to double glass with two identical panels, the insulation dips for vibration eigenfrequencies are much greater than those found for solutions using glass of different thickness. With triple glass, in comparison with double glass, the improvements are not significant if we are looking at situations where the larger air chamber in the triple glazed solution is the same as the air chamber in the double glazed solution.

In practice, the construction high acoustic insulation double and triple glazing, with thick air chambers, is possible by making double windows with separate frames. This type of solution may be useful where it is necessary to improve the acoustic insulation of an existing façade.

Analysis of the glazed solutions with frames enables us to see that acoustic insulation is generally less than that provided by glazed solutions without frames. The insulation values of the solutions with frames improve if the mass of the frame is not small and is well sealed. The application of high insulation glazed solutions thus implies the use of a high quality frame. A window with a very small gap may suffer significant acoustic dips.

6. REFERENCES

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