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Numerical and experimental analyses of sound transmission loss of double-glazed unit

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Abstract: The acoustic performances of building elements such as windows are performed in laboratory according to standards. In addition to the high cost of the experimental tests, the measurements at low frequencies, face some difficulties such as the lack of reproducibility, the diffuseness of the acoustic field and the effect of the modal behaviour of the rooms. To overcome this, a numerical analysis of the transmission loss of a double-glazing structure is developed in this work. To this end, four numerical configurations, based on experimental conditions, are proposed. The differences concern the modeling of the emitting and receiving rooms. The numerical model used for the double-glazing was calibrated through the Experimental Model Analysis. Results show that there is a significant effect of the rooms on the transmission loss at low frequencies and so of its properties such as the acoustic absorption. A comparison with experimental results is also established to validate the chosen configuration with which parametric analyses are carried out, however, results are not presented in this paper.

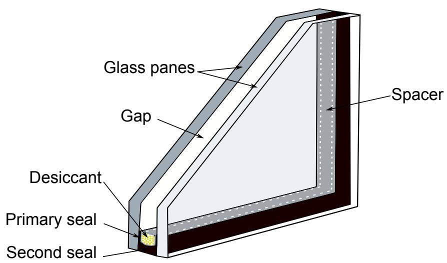
Keywords: Double-glazing, Transmission Loss, Experimental Modal Analysis, Numerical analysis.

INTRODUCTION

Double-glazing is often used in noise reduction since a relatively high transmission loss is achieved by the introduction of an acoustic gap between the panels. Due to various effects of fluid-structure interaction, boundary conditions and the modal behaviour in low frequencies, the prediction of the acoustic performance of the double wall remains complicated. In the literature, Tadeu et al. studied experimentally the transmission loss through single, double and triple glazing (Tadeu and Mateus, 2001), while, in Xin and Lu (2011), the effect of boundary conditions on this acoustic indicator of double panels was investigated. A comparison of different types of glass can also be found in Miskinis et al. (2015). Regarding the numerical prediction approaches, several methods are available. The choice of the method depends on the computational cost and the considered frequency band. The Finite Element Method (FEM) is suitable to treat problems in low frequencies. For example, Larbi et al. used it to develop a model of a sandwich plate with viscoelastic core (Larbi et al. 2016). Another example can be found in Sgard et al. (2000) which studied the prediction of sound transmission through double-wall with elastic porous lining.

NUMERICAL CALIBRATION OF THE DOUBLE-GLAZING FROM EMA

In this section, we are interested in the calibration of a numerical model of the double-glazing through the Experimental Modal Analysis (EMA) . The studied structure presented by Figure 1 is an Insulting Glass Unit. It consists of two glass and a separating edge which forms an air or gas gap. The complexity is located along the edge. In fact, the stainless hollow steel spacer is filled at least half by molecular sieve in the form of beads called the desiccant and used to dry out the cavity. In addition, a dual seal combination has a primary seal of polyisobutylene between the spacer and the panes, and a secondary seal of silicone around the outside edge is used to glue all the parts. So, to calibrate a model of the group "spacer + desiccant + seal", an EMA is carried out for the full system in order to determine the mechanical properties (E, ν and ρ) for an equivalent homogeneous isotropic material for this group.



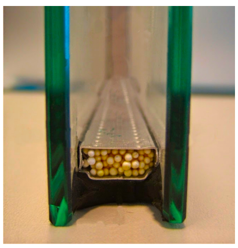


Figure 1 – Details of the Insulting Double Unit

Experimental Modal Analysis

For performing the EMA, a home-made Matlab toolbox has been developed and includes several approachs like LSCF and its polyreference version. The suspended double-glazing (see Figure 2), composed of two glass of 6 mm and 4 mm separeted by 18 mm of argon, is meshed into 153 nodes in order to obtain a good visualisation of the mode shapes. The application of the method implemented in the toolbox requires the measurement of the Frequency Response Function (FRF) at different points of the structure after its excitement. In this work, the roving hammer test is used with an impact hammer and three reference accelerometers.

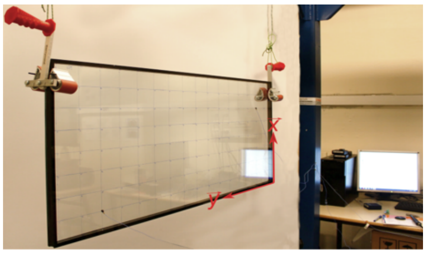


Figure 2 – Suspended double-glazing

Numerical calibrated model

The double-glazing experimentally tested is calibrated numerically with the finite element method. The numerical model consists of two 2D shells representing the two panels separated by a 3D cavity filled with argon. An equivalent solid for the group "spacer + desiccant+ seal" is defined. Quadrilateral elements for the glass panels and hexahedral ones for the fluid and the equivalent spacer are used for the finite element discretization, and their sizes are controlled by the wavelength of each subdomain (see Figure 3) .

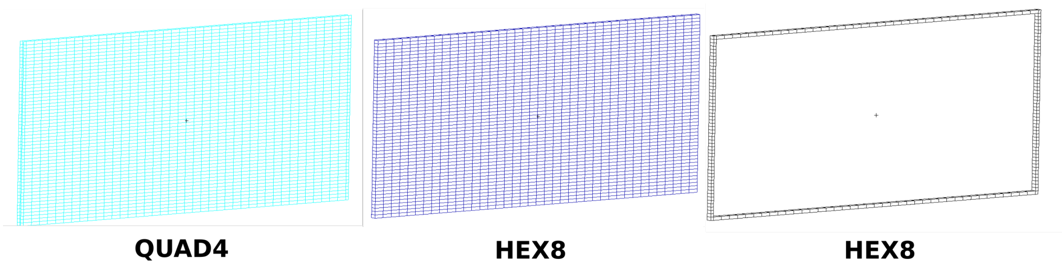


Figure 3 – Numerical meshing of the IGU

Results and discussion

Table 1 shows the comparison of the first ten natural frequencies of the structure obtained from the experimental and numerical analysis after calibration of the numerical model on the first natural frequency. For a such complex structure, results are considered satisfying since the maximum difference of natural frequency is 6.8 %, observed for Mode 8. Therefore, this calibrated model is used to evaluate numerically the transmission loss.

Mode	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
Experimental	29.9	37.7	41.2	56.4	60	75.5	93.9	94	98.4	117.6
Numerical	29.8	36.6	38.6	56.4	63.8	72.9	98.9	100.4	102.7	113.9
Gap (%)	0.4	2.9	6.3	0.1	6.4	3.4	5.4	6.8	4.4	3.1

Table 1 – Experimental and numerical eigenfrequencies of the double-glazing

DETERMINATION OF THE SOUND TRANSMISSION LOSS

The airborne sound insulation of building elements is evaluated in laboratory according to standards ISO 10140. Measurement are done between two adjacent reverberation rooms. In low frequencies, tests face some difficulties such as the lack of reproducibility, the diffuseness of the acoustic field and the effect of the modal behaviour of the rooms. To study these different problems, four numerical configurations have been carried out. In the first one (Figure 4.a), a realistic laboratory configuration is considered. It consists in two reverberant rooms modelled with a spherical acoustic source in the emitting one. In fact, standards assume that the acoustic excitation is a diffuse field for the entire frequency band of measurement. However, this condition is respected only beyond the "Schroeder frequency" which marks the limit between the modal and the diffuse field behaviour. So, a second configuration (Figure 4.b) considering the source side as a diffuse field even for the very low frequencies while modelling the receiving room is considered. Numerically, this condition is ensured from the superposition of set of plane waves with random phase. For the third configuration (Figure 4.c), to avoid the modal behaviour of the receiving room, an acoustic radiation in a half-space while modelling the emissive room is considered. The Rayleigh condition, which references to a vibrating structure in the plane of a rigid baffle, is used. Finally, the fourth configuration (Figure 4.d) combines the diffuse field in the source side and the Rayleigh radiation in the receiving one. The advantages of this model is a significant reduction in the model size, but considers perfect experimental conditions in both sides of the structure under test.

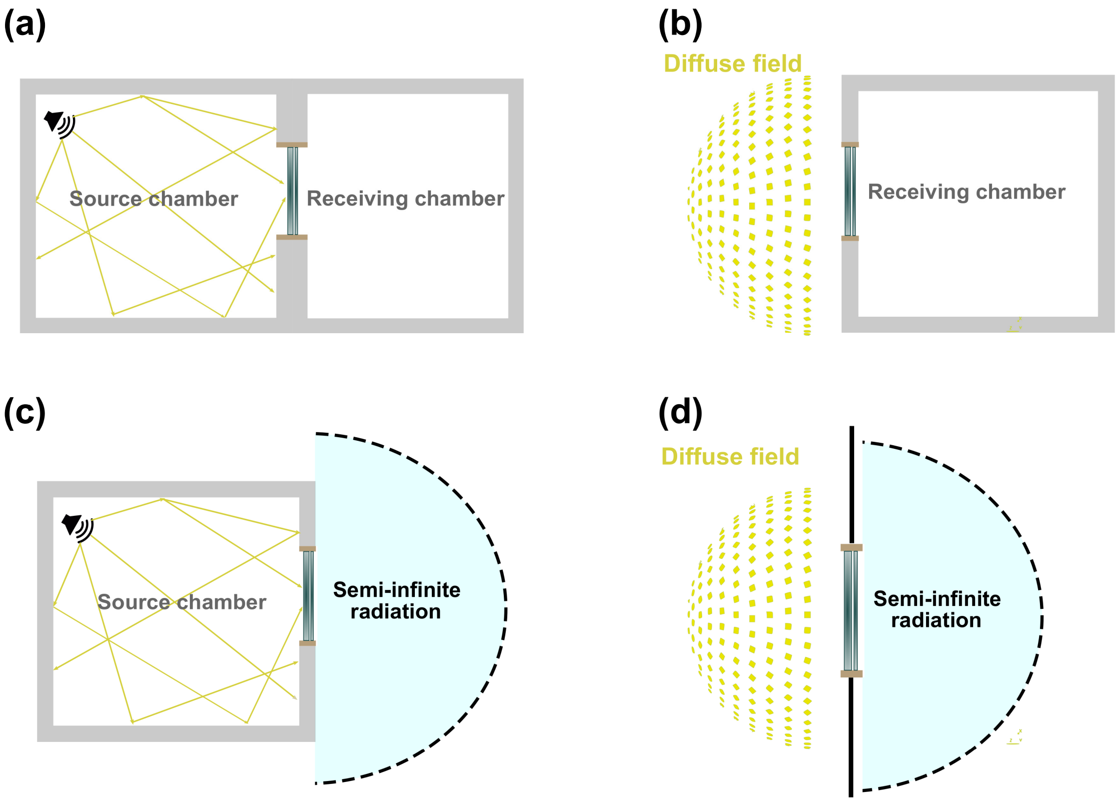


Figure 4 – Numerical configurations considered for the prediction of the TL

RESULTS OF THE TRANSMISSION LOSS

The analyses have been carried out to investigate the sound transmission loss through the IGU 6/18/4 (6 mm glass panel in the emitting side, 18 mm argon cavity and 4 mm glass panel in the receiving side) calibrated in the previous section. A damping coefficient of 1 % is used for all the materials of the structure. The latter is fully clamped at its edges in the common wall of the rooms. The emitting and receiving rooms dimensions are $5\times4.5\times3.25$ m^3 and $5\times4\times3.25$ m^3, respectively (Fig. 5).

To compare the numerical cost of the four configurations, a comparison of the degrees of freedom (DOF) with considering the size of each model is given in Table 2. With respecting the meshing conditions for the structure and the fluid domains, the first case with the two acoustic rooms has 189 $\times10^3$ DOF which is around 5 times the DOF of the case with free field in the both sides of the double glazing.

Comparison of the TL of the four configurations

To study the intrinsic acoustic response of the IGU without any influence of environmental factors, the narrow band transmission loss of the structure predicted with the perfect free-fields configuration is presented in Fig. 6. The result is compared to the one obtained with the impedance approach given by Heckel (Heckl M., 1981). As it can be observed,,

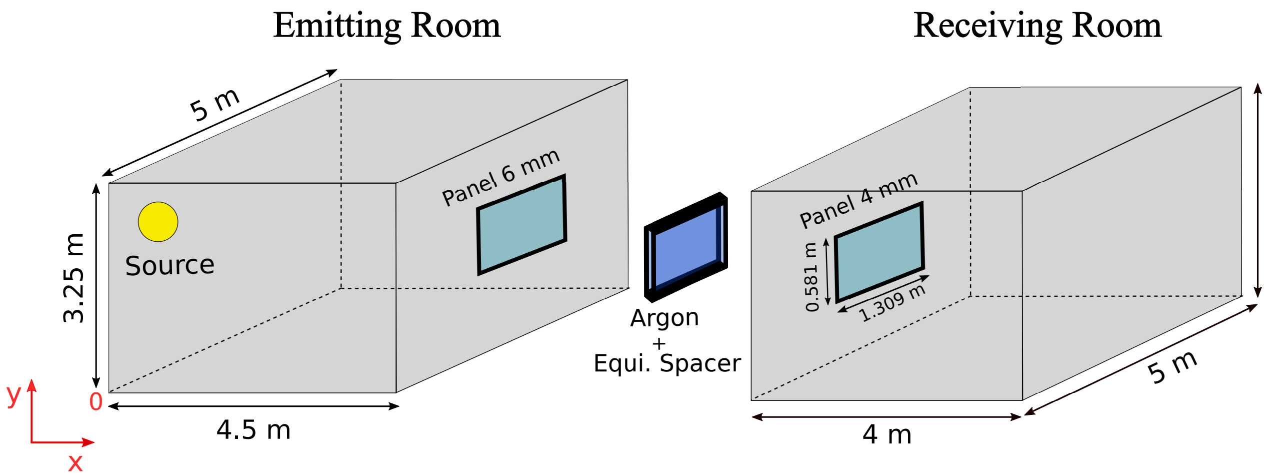


Figure 5 – Emitting and receiving rooms connected through the IGU.

Table 2 – DOFs of the four configurations.

Configuration	1	2	3	4
DOFs ($\times 10^3$)	189	110	115	36

the theorical TL is higher than the numerical result except in the restricted frequency band around 175 Hz. This has been pointed out by (Heckl M., 1981) who stated that the impedance theory can only be applied qualitatively in the case of the double glazing of windows. This was explained by the fact that, in addition to the small size of this type of strcuture, the argon cavity between the panels has a significant lateral resonances. In the region of coincidence of theoretical and numerical results, a characteristic phenomenon of the double-partition system is the mass-air-mass frequency. At this frequency, the two plates move out of phase and the effect of the cavity on the plates is mostly one of added stiffness (Larbi et al. 2016). This frequency is similar to the mass-air-mass resonance of unbounded double panels, so it can be calculated approximately by the expression given in (Heckl M., 1981):

$$f_{mam} \cong 1800\sqrt{\frac{m_1+m_2}{d\,m_1 m_2}} \tag{1}$$

where d is the depth of the acoustic cavity between the panels in mm, and m_1 and m_2 are the surface masses of the panels in kg/m^2. For the present structure, this frequency is around 174 Hz and the corresponding modal shape is illustrated in the Fig. 7. The other dips in the numerical TL (Fig. 6) correspond to the eigenfrequencies of the double glazing whose density increases as the frequency increases.

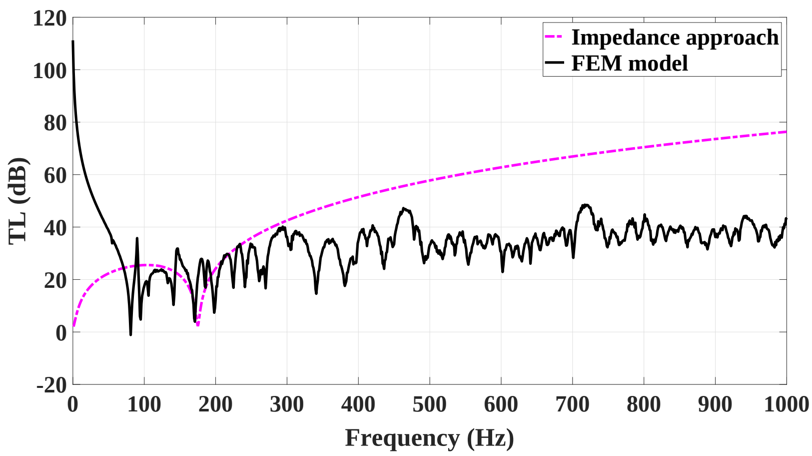


Figure 6 – Comparison of the TL between the impedance approach and the FEM (4^{th} configuration).

The coupling of the structure to acoustic rooms change its response and so the transmission of the sound due to two related factors: the first one is the closeness of the eigenfrequencies of the rooms and those of the structure, the second

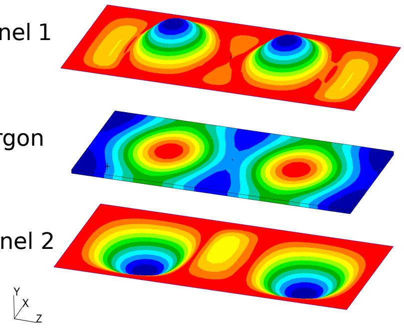


Figure 7 – Fluid pressure and panels normal displacement at the mass-air-mass frequency (174 Hz).

is the spatial matching between the distributions of the displacements and the acoustic pressure over the structure (Fahy F., 1985). To investigate the influence of the rooms, the first three configurations are compared to the forth one. Theoretically, Schroeder has shown that the acoustic response of an acoustic enclosure presents two zones: the modal region and the high modal density region (Schroeder, M. R., 1996). The limits between the two zones is known as the "Schroeder frequency f_{Sch}", it depends on the acoustic absorption and defined by (Schroeder, M. R., 1996) as:

$$f_{Sch} = 2000\sqrt{(0.16/A)} \tag{2}$$

A perfect rigid walls are modeled for the rooms. Indeed, even if this case is an extreme situation, this study will highlight the significant effect of the acoustic absorption in the rooms. From the comparison between the four configurations in narrow band presented in Fig. 8-a, it can be observed that in very low frequencies, there are visible and separated natural frequencies of the rooms. As a result, intense dips and fluctuations marked the TL in addition to those presenting the modal behavior of the structure. As the frequency increases, this behavior becomes less significant due to the increase of the modal density of the rooms, and so, the TL curves of the configurations with one or two rooms become smoother. From the 1/3-octave spectrum presented in Fig. 8-b, it can seen that the results from all the configurations have the same trend for frequencies above 250 Hz, although, a significant difference is observed between them. For example, the difference between the case of the acoustic suit and the perfect free-fields reaches 14.5 dB in 500 Hz.

Comparison with experimental measurements

In order to validate the 4^{th} configuration and the calibrated model of the sealing system of the double glazing, we compare the numerical results with experimental data from the literature. The frequency band depends on the available experimental data. We have chosen to predict the transmission loss of the system up to 500 Hz when the experimental data starts at 50 Hz, and up to 630 Hz when measurements are made for frequencies above 100 Hz.

The experimental data presented in this section are derived from measurements made in accordance with the standards ISO 10140, (2010). The typical installation of double glazing in the opening of the wall separating the two rooms is shown in Fig. 9. A layer of mineral wool is inserted into the partition of the wall and the wooden opening to reduce the solid sound transmission. After the installation of the double glazing, a compressible foam and a mastic seal are applied to the entire perimeter to eliminate sound leaks. The measurement results are presented in third-octave bands according to the recommendations of the standard (ISO 10140, 2010).

For the results issued from Foret et al (2009) work, the type of edge sealant system of the structure is detailed, however, it is unknown for the data issu from the work of Assaf R. (2015). In fact, the majority of the globally manufactured IGU are dual sealed especially in Europe with about 85-90 % of commercial glazing units . To this end, the calibrated properties of our model, which is a dual sealed one, are used for the present models.

The results from the work of Foret et al (2009) are illustrated in by Fig. 10. In total, seven 4/16/4 IGU have been tested with normalized dimensions which are 1.48 m wide by 1.23 m high. The difference between them lies in the sealing system, i.e. the type of the spacer and the two sealing barriers. The Fig. 10-a shows the TL of the different configurations tested from 100 Hz to 630 Hz. Results show that the acoustic performance of the system depends on the sealing system with a difference that can reach about 5 dB at 200 Hz, corresponding to the resonance region. The first IGU that has been tested experimentally has the same sealant system of our model calibrated previously. For that, a comparison of the result of the numerical model and that of the first system is presented in Fig. 10-b. A good agreement can be noticed between the two approaches over the whole frequency range with a maximum difference of 1.5 dB at 315 Hz. In order to explore the representativity of our calibrated model, the numerical TL is compared to the average of the seven series of the experimental data (Fig. 10-b). It can be seen that the numerical curve falls within the standard deviation from the mean of the experimental results except for the frequencies 125 Hz and 315 Hz. The degradation of the acoustic performance due to the effect of the resonance mass-air-mass can be clearly distinguished which is estimated at 228 Hz for the present

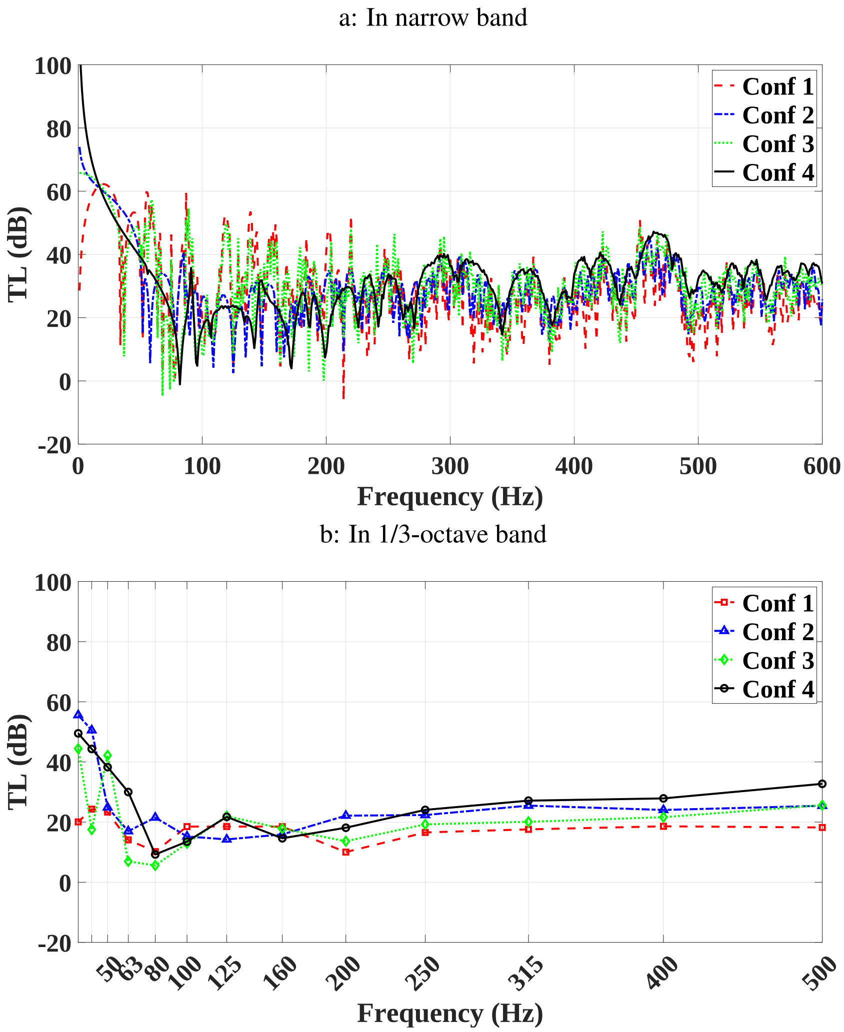


Figure 8 – Comparison of TL of the IGU calculated with the four configurations (Perfect rigid walls).

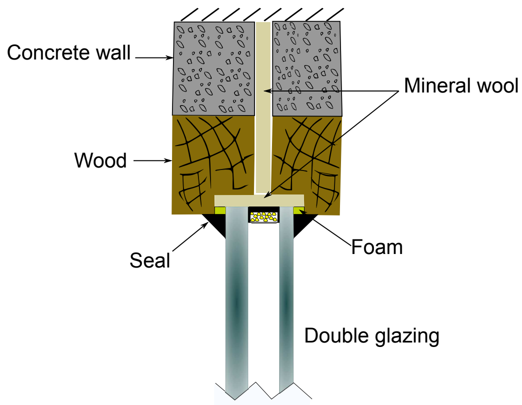


Figure 9 – Mounting details of double glazing in the wall for the acoustic test.

system.

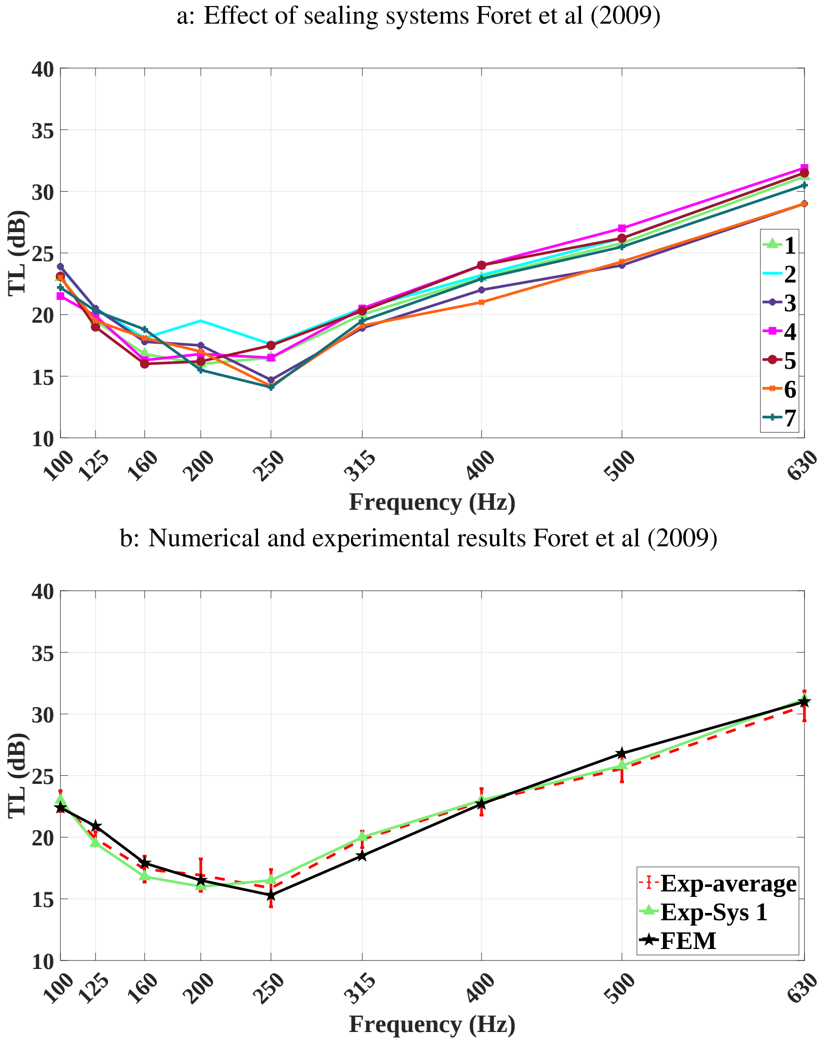


Figure 10 – Comparison of the numerical and experimental transmission loss of a normalized 4/16/4 IGU.

The second IGU studied is composed of two glass of 6 mm separated by 12 mm by a cavity fulled with argon and tested by Assaf R. (2015). The dimensions of the glass are 1.13×1.38 m^2 whose sealing system is not specified. Fig. 11 presents the comparison between numerical and experimental transmission loss. Even with the absence of information (type of the sealent system), the both curves follow the same trend over the frequency range of interest except for the 80 Hz frequency. For this glazing composition, the analytical mass-air-mass frequency gives a value of 215 Hz (Fahy F., 1985), which corresponds to the dip in the TL at 200 Hz for both results.

The comparison with experimental results of the two double glazing composition show a good agreement with the numerical transmission loss for the whole frequency range even for those below 100 Hz. As a consequence, the calibrated model of the Equivalent Spacer can be considered satisfying acoustically and the 4^{th} configuration leads to predict properly the acoustic performances of the structure independently to the rooms.

CONCLUSIONS

The double-glazing is calibrated numerically with the Experimentally Modal Analysis. The established model is used to carry out numerical acoustic analyses. The results show that the presence of the acoustic rooms has a significant influence on the TL at very low frequencies. However, above 125 Hz, all TL converge. The fourth configuration is thus preferred since it is the less expensive configuration in terms of computational cost. Finally, it can be noted that in addition to the experimental results presented in this paper, parametric analyses have been performed but are not presented here.

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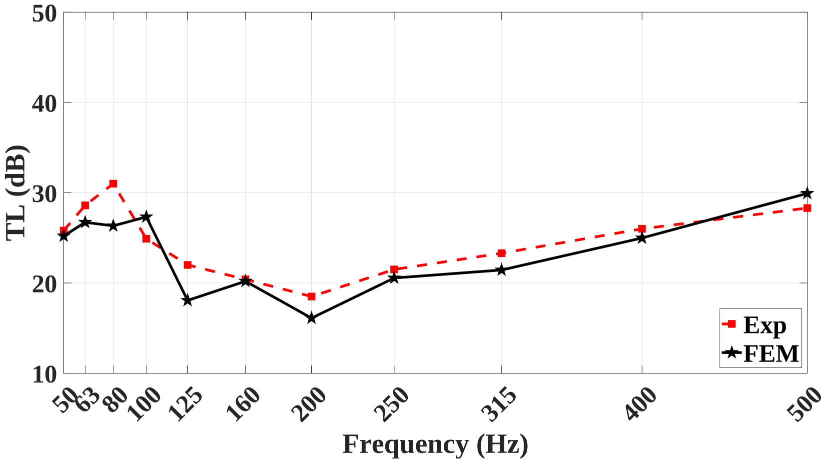

Figure 11 – Comparison of the numerical and experimental transmission loss Assaf R. (2015) of 6/12/6 IGU.

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