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Double glazing vibroacoustic behaviour

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The dynamic behaviour of a double glazing system requires the modelling dynamic behaviour of panels, air cavity, and fluid structure interaction. The associated energy functional formulation is established. Its discretisation by finite element methods leads to dynamic system equation. Modal approach, based on the decomposition of the modal pressure in a statical pressure and dynamical one, permits the determination of eigenfrequencies and eigenmodes of coupled system and the resolution of dynamical equations projected on the modal basis. Vibratory analysis allows to conclude the existence of two eigenmodes families for which the vibroacoustic coupling between panels and air cavity is absent for the first one and strong for the second one. Moreover, the established dynamic responses of a double glazing system show an important reduction of energy transmitted in comparison with simple panels.

INTRODUCTION

Research in vibro-acoustics has raised a considerable interest since the beginning of the last century. Many studies in this field investigated the problem of transparency to the aerial sounds in simple panels [1, 2, 3, 4, 5, 6, 7]. Since the middle of the 20th century, researchers have started investigating double panel systems. Double-glazing has become very widely used in building and industries such as car making, ship building and aeronautic and space crafts.

London [8] dealt with the double panel system by the use of a simple model of two infinite plates separated by a perfect fluid. Later, Sewell [9] studied the acoustic transmission through a simple panel and a double panel both hinged on an infinite baffle. Fahy and Mason [10] developed a simple theoretical model method representing a double panel mounted by springs.

To solve the acoustic transmission problem through double panels, Bouhioui [11] developed a mixed method, based on finite element formulation for both the structure and the internal fluid and an integral equation formulation for the outside fluid.

Tang [12] analysed the noise reduction at low frequency by lining the cavity between two panels of a finite double panel structure with porous material. He found that the sound insulation of sandwich panels with air cavities containing and porous material is more effective than that of the single-layered panel at low frequency.

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Fonseca [13] studied the vibro-acoustic behaviour of the double panel by the active control of the sound transmission and concluded that the sound transmission through the double panel is larger than that through a single panel at the resonances of the coupled eigenmodes. At frequencies where the double panel radiates much sound energy, the acoustic energy in the cavity is also high. In order to avoid those frequencies and achieve a considerable reduction of the sound transmission over a large frequency band, active control simulations is accomplished for both symmetrical and asymmetrical double panel systems.

Tadeu [14] presented an experimental result on sound insulation of glazed openings. The laboratory experiments were performed placing the test specimens between two relatively small rooms. The effect of the type of fixing frame, the air gap between the panels and the glass panel thickness are studied and compared with analytical models. A difference is found between experimental results and analytical models.

However, despite the progress made in the practical aspect of vibro-acoustics, researches are not able to take account of all complex phenomena producing in a double panel system. In fact, the vibro-acoustic behaviour analysis of a fluid cavity confined between two panels, involves many aspects of mechanics such as acoustics, structure calculating and fluid-structure interaction. These require the use of the numerical methods especially the finite element method to solve this complex problem.

In this paper, we adopt a mixed numerical model based on a classical finite element formulation for the structure and the internal fluid, and a boundary element formulation for the external fluid. The developed model permits the determination of the vibro-acoustic modes for a double panel system and therefore, the establishment of the dynamic responses by modal recombination. The numerical results are presented and discussed.

1. DYNAMIC EQUATIONS

The double panel system (Fig. 1) consists of a fluid cavity confined between two thin plates. This assembly is solidly fixed to a rigid baffle which separates the acoustic domain into two semi infinite fluid domains V_1 and V_2.

Fluid structure interaction problems are more often resolved in the Fourier space.

The Love-Kirchhoff flexion equation of the thin plates may be written as follows [11, 15, 16]:

$$\begin{cases} D_i\Delta^2 \mathrm{w}_i - \omega^2\sigma_{si}\mathrm{w}_i = \mathrm{F}_i & \text{in } \Sigma_i \ (i=1,2); \\ \mathbf{C}_{pi}\left(\mathrm{w}_i\left(\mathrm{x},\mathrm{y}\right)\right) = 0 & \text{on the edge } \partial\Sigma_i \ (i=1,2), \end{cases} \tag{1}$$

where w_i, Σ_i and $\partial\Sigma_i$ are respectively the displacements, the middle surface and the edges of the plate index *i*, σ_{si} and D_i are respectively density and flexion modulus of plate index *i*, and $\mathbf{C}_{pi}$ represents the boundary conditions operator. F_i is the pressure applied on the plate index *i*, with

$$\begin{cases} F_1 = p_1 + p_s - p & \text{on the plate } \Sigma_1; \\ F_2 = p - p_2 & \text{on the plate } \Sigma_2, \end{cases} \tag{2}$$

where p_s is the pressure generated by the acoustic source, p is the pressure exerted by the fluid cavity on each plate and p_i is the pressure radiated by the plate index *i*.

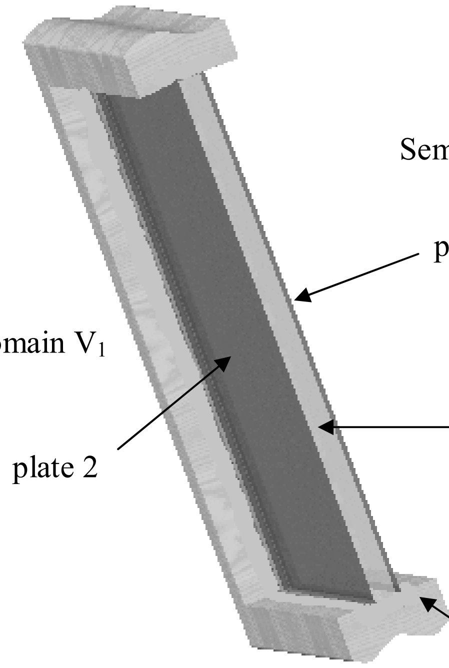


Fig. 1. Double panel system

The pressure in the cavity, solution of Helmholtz equation, may be written [9, 11, 15, 16]:

$$\Delta p + \frac{\omega^2}{c^2} p = 0, \tag{3}$$

where p and c are respectively fluid pressure and sound velocity.

The two coupled vibro-acoustic conditions between fluid and plates may be written as follows [9, 11, 15, 16, 17]:

$$\frac{\partial p}{\partial n_i} - \omega^2 \rho_f w_i = 0 \text{ on } \Sigma_i \ (i = 1,2), \tag{4}$$

$$-p\vec{n}_i = [\sigma_i]\vec{n}_i \text{ on } \Sigma_i \ (i = 1,2), \tag{5}$$

where ρ_f is the fluid density. $[\sigma_i]$ is the stress tensor in the plate index *i*. $\vec{n}_i$ is the outer normal of fluid related to the plate *i*, with $n_1 = -n_2$.

The fluid in the acoustic domains V_i (i = 1, 2) satisfied the acoustic wave equation

$$\left(\Delta + k_i^2\right)p_i = 0 \text{ in } V_i \ (i = 1,2) \qquad (6)$$

with the coupling conditions:

$$\frac{\partial p_i}{\partial n} = \rho_i \omega^2 w_i \text{ on } \Sigma_i , \qquad (7)$$

$$\frac{\partial p_i}{\partial n} = 0 \text{ on the baffle} \qquad (8)$$

and the Summerfield radiated condition

$$\lim_{r\to\infty} r\left|\frac{\partial}{\partial r} - jk_i\right| p_i = 0 . \qquad (9)$$

2. FUNCTIONAL OF ENERGY

The functional of energy of a double panel system is the sum of all elementary functional energy of each component [11, 16, 17, 18] which may be written:

$$\begin{aligned} U\left(w_1, w_2, p\right) = & \frac{1}{2}\int_{\Sigma_1}\{\varepsilon_1\}^T\left[D_1\right]\{\varepsilon_1\}d\Sigma_1 - \frac{1}{2}\omega^2\int_{\Sigma_1}\sigma_{s1}w_1^2 d\Sigma_1 - \frac{1}{2}\omega^2 B_1\left(w_1, w_1\right) \\ & + \frac{1}{2}\int_{\Sigma_2}\{\varepsilon_2\}^T\left[D_2\right]\{\varepsilon_2\}d\Sigma_2 - \frac{1}{2}\omega^2\int_{\Sigma_2}\sigma_{s2}w_2^2 d\Sigma_2 - \frac{1}{2}\omega^2 B_2\left(w_2, w_2\right) \\ & + \frac{1}{2\omega^2}\int_{V_f}\frac{1}{\rho_f}\left(\overrightarrow{\text{grad}}\, p\right)^2 dV_f - \frac{1}{2}\int_{V_f}\frac{1}{\rho_f c^2}p^2 dV_f \\ & + \int_{\Sigma_1} w_1 p\, d\Sigma_1 - \int_{\Sigma_2} w_2 p\, d\Sigma_2 - \int_{\Sigma_1} w_1 p_s\, d\Sigma_1 \end{aligned} \qquad (10)$$

where $\{\varepsilon\}$ and $[D]$ are respectively the second derivative displacement vector and the elastic matrix of plate:

$$\{\varepsilon\}^T = \left\langle -\frac{\partial^2 w}{\partial x^2},\ -\frac{\partial^2 w}{\partial y^2},\ -2\frac{\partial^2 w}{\partial x \partial y}\right\rangle ; \quad [D] = \frac{Eh^3}{12\left(1-\nu^2\right)}\begin{bmatrix} 1 & \nu & 0 \\ \nu & 1 & 0 \\ 0 & 0 & \frac{1-\nu}{2} \end{bmatrix}, \qquad (11)$$

and B is the bilinear auto-influence acoustic impedance form:

$$B\left(w,w\right) = \rho\int_\Sigma\int_\Sigma w(Q)\,G(M,Q)\,w(M)\,d\Sigma\, d\Sigma , \qquad (12)$$

where $G(M,Q)= \frac{e^{jkR}}{2\pi R}$ is the free Green function of the Helmholtz's equation which satisfies the Sommerfeld condition equation (9).

Displacements w_1 and w_2 in each panel and pressure p in fluid cavity make stationary the functional energy U and are the solution of the problem.

The functional of energy is discretisated by Discret Kirchoff Triangle finite elements for the plates, prismatic finite elements for the fluid cavity, triangular finite elements for the fluid-structure interaction and triangular element for boundary external fluid [16]. This discretisation by finite element method leads to the dynamic equation:

$$\begin{bmatrix} [K_1]-\omega^2\left([M_1]+[B_1]\right) & 0 & [C_1] \\ 0 & [K_2]-\omega^2\left([M_2]+[B_2]\right) & -[C_2] \\ [C_1]^T & -[C_2]^T & \frac{1}{\rho_f\omega^2}\left([H]-k_0^2[Q]\right) \end{bmatrix} \begin{Bmatrix} \{W_1\} \\ \{W_2\} \\ \{P\} \end{Bmatrix} = \begin{Bmatrix} \{F_1\} \\ 0 \\ 0 \end{Bmatrix}, \quad (13)$$

where:$[K_1]$ and $[K_2]$ are respectively the rigidity matrix of plates index 1 and 2, $[M_1]$ and $[M_2]$ are respectively the mass matrix of plates index 1 and 2, $[Q]$ is the fluid compressibility effect matrix, $[H]$ is the fluid inertial effect matrix, $[B_1]$ and $[B_2]$ are respectively the acoustic impedance matrix associated to the acoustic domains V_1 and V_2, $[C_1]$ and $[C_2]$ are respectively the coupling matrix between fluid-plate 1 and fluid-plate 2, $\{W_1\}$ and $\{W_2\}$ are respectively the nodal displacement vector of the plate 1 and 2, $\{P\}$ is the nodal pressure, $\{F\}$ is the nodal force vector applied to plate 1.

For the resolution of the dynamic equation (13), we adopt modal approach. First, the eigenmodes and eigenfrequencies of the system are established. Then, the dynamic equation (13) is projected in the modal basis, resolved and dynamic responses are therefore obtained by modal recombination.

In fact, in the first step, we determinate the eigenmodes and the eigenfrequencies of the system composed only with panels and fluid cavity. There is solution of the dynamic equation on plate's modal basis and fluid cavity's modal basis:

$$\begin{bmatrix} \left[\omega_{i,S1}^2-\omega^2\right] & [0] & [\mathbf{C}_1] \\ [0] & \left[\omega_{i,S2}^2-\omega^2\right] & -[\mathbf{C}_2] \\ [\mathbf{C}_1]^T & -[\mathbf{C}_2]^T & \frac{1}{\rho_f\omega^2}\left[k_f^2-k_0^2\right] \end{bmatrix} \begin{Bmatrix} \{\mathbf{W}_1\} \\ \{\mathbf{W}_2\} \\ \{\mathbf{P}\} \end{Bmatrix} = \begin{Bmatrix} \{0\} \\ \{0\} \\ \{0\} \end{Bmatrix}. \quad (14)$$

The use of B. Irons transformation [19] needs the decomposition of the fluid cavity's eigenmodes in a static mode and dynamic modes. So, the compressibility matrix $\left[k_f^2\right]$ and the pressure vector $\{\mathbf{P}\}$ can be defined as follow:

$$\left[k_f^2\right] = \begin{bmatrix} 0 & \\ & h \end{bmatrix};\ \{\mathbf{P}\} = \begin{Bmatrix} P_0 \\ P_r \end{Bmatrix}, \quad (15)$$

where h is a diagonal matrix composed of fluid cavity's eigenfrequencies, P_0 is the static modal pressure and P_r is the dynamic modal pressure.

The coupling matrix between the plate index *i* and the fluid cavity may be written:

$$[\mathbf{C}_i]=\left[\{c_i\},C_{ri}\right], \tag{16}$$

where $\{c_i\}$ is a coupling vector between the plate index *i* and the static modal pressure and C_{ri} is the coupling matrix between the plate index *i* and the dynamic modal pressure.

We suggest the following change of variable:

$$P_r=\sqrt{\rho_f c^4}\,\sqrt{h}\,Q_r\,. \tag{17}$$

The dynamic equation (14) may be written in an eigenvalue problem form:

$$\left([K]-\omega^2\,[M]\right)\{U\}=\{0\}, \tag{18}$$

where

$$[K]=\begin{bmatrix} \left[\omega^2_{i,S1}\right]+\dfrac{1}{\rho_f c^2}\{c_1\}\langle c_1\rangle & -\dfrac{1}{\rho_f c^2}\{c_1\}\langle c_2\rangle & 0 \\ -\dfrac{1}{\rho_f c^2}\{c_2\}\langle c_1\rangle & \left[\omega^2_{i,S2}\right]+\dfrac{1}{\rho_f c^2}\{c_2\}\langle c_2\rangle & 0 \\ 0 & 0 & \left[\omega^2_f\right] \end{bmatrix}, \tag{19}$$

$$[M]=\begin{bmatrix} I+\rho_f C_{r1}h^{-1}C^t_{r1} & -\rho_f C_{r1}h^{-1}C^t_{r2} & -\sqrt{\rho_f}C_{r1}h^{-1/2} \\ -\rho_f C_{r2}h^{-1}C^t_{r1} & I+\rho_f C_{r2}h^{-1}C^t_{r2} & \sqrt{\rho_f}C_{r2}h^{-1/2} \\ \sqrt{\rho_f}h^{-1/2}C^t_{r1} & \sqrt{\rho_f}h^{-1/2}C^t_{r2} & I \end{bmatrix};\quad \{U\}=\begin{Bmatrix}\{\mathbf{W_1}\}\\ \{\mathbf{W_2}\}\\ \{Q_r\}\end{Bmatrix}. \tag{20}$$

Then, in the second step, the dynamic equation (13) is projected on the eigenmodes basis of the coupled system and may be written as follow:

$$\left(-\omega^2[M]+[K]+j[A]\right)\{X\}=\{F(\omega)\}, \tag{21}$$

where [M] and [K] are respectively the generalized mass matrix and the generalized stiffness matrix, {F} is generalized force vector, [A] is the generalized modal damping matrix.

Plate's dissipation of energy is modelled by an hysteretic damping [20] in stiffness matrix's dynamic equation (13) and fluid cavity's dissipation of energy is modelled by a complex acoustic wave number $k_0^2\left(1+j\,\eta_f\right)$ [21] in dynamic equation (13).

3. NUMERICAL RESULTS

3.1. Vibro-acoustic analysis

The double glazing system investigated corresponds to an air cavity confined between two clamped glass plates. The geometrical and mechanical system features are presented below in Table 1.

Table 1. Geometrical and mechanical system features

Glass plate		**Air cavity**	
Gap between plates (m)	0.012	Height (m)	0.012
Length (m)	1.48	Length (m)	1.48
Width (m)	1.23	Width (m)	1.23
Thickness (m)	0.004	Density (Kg/m3)	1.2
Young modulus (MPa)	$7.2\ 10^4$	Sound Velocity (m/s)	341
Density (kg/m^3)	2500		
Poisson ratio	0.22		

In order to characterize the vibroacoustic behaviour of the coupled system's, we determine the eigenfrequencies and the eignmodes in the frequency band between 0 and 1000 Hz. Table 2 presents the first four eigenfrequency values of a clamped plate only, air cavity only and double glazing system composed of clamped plates.

Table 2. Eigen frequency values

Mode order	**Plate only**		**Air cavity only**		**Double glazing**	
	(m,n)	Frequencies (Hz)	(m,n,l)	Frequencies (Hz)	Frequencies (Hz)	Plates deformed shape
1	(1, 1)	20.4	(0,0,0)	0.0	20.43	in phase
2	(2, 1)	36.8	(1,0,0)	118.1	20.57	in opposition
3	(1, 2)	45.9	(0,1,0)	142.9	28.49	in opposition
4	(2, 2)	60.8	(1,1,0)	186.1	36.84	in phase

The double glazing modal analysis shows the presence of two eigenmodes groups. For the first one, the two plates vibrate in phase (mode number 1 and 4) and the fluid is not affected. For the second group, plates vibrate in opposition and the fluid is greatly affected. The coupling is strong and very important.

The modal deformed shape for both plates and air cavity for the first group of eigenmodes and the second group are presented respectively in Table 3 and Table 4.

Table 3. First group of deformed modal shape

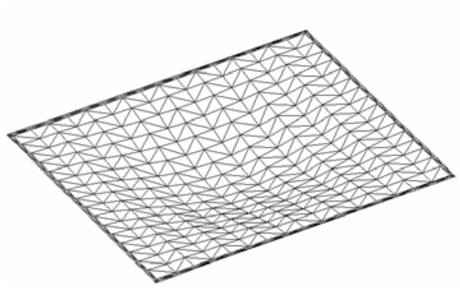
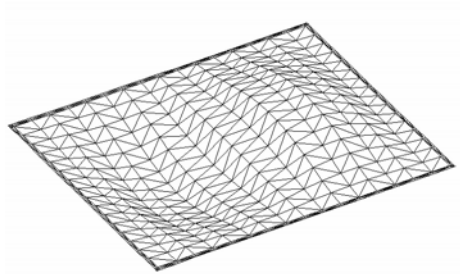
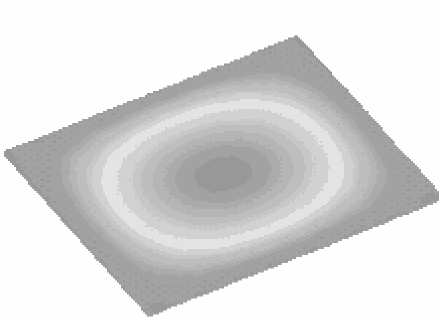
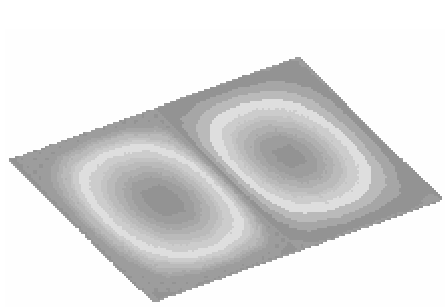
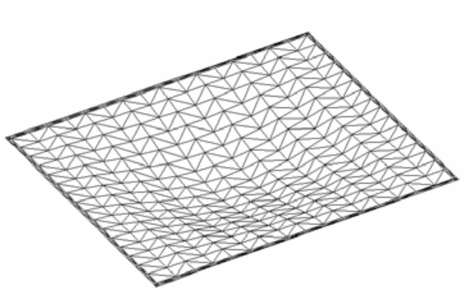
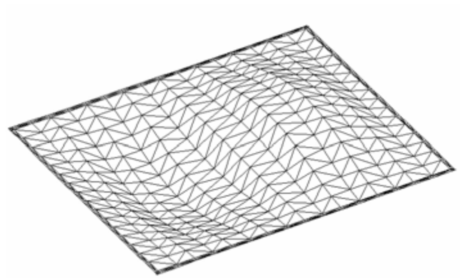
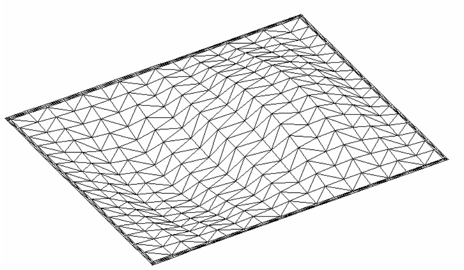
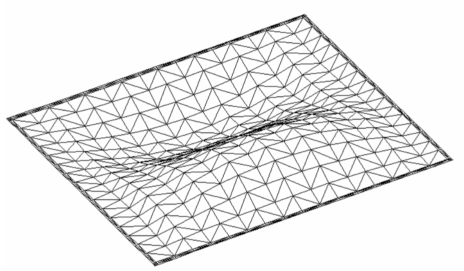
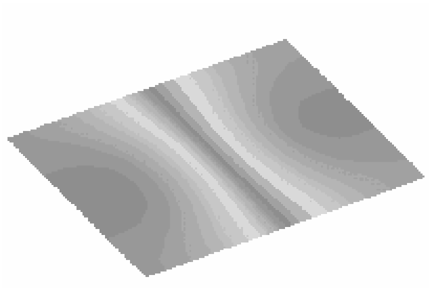
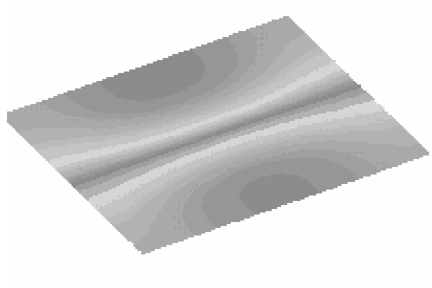
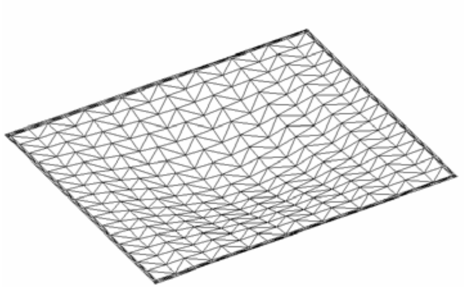
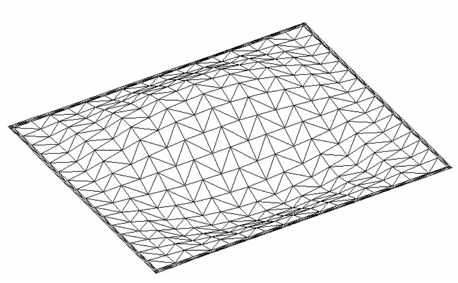
Mode 1 $f_1 = 20.43$ Hz	Mode 4 $f_4 = 36.84$ Hz
deformed modal shape for plate index 1	
	
modal pressure in air cavity	
	
deformed modal shape for plate index 2	
	

Table 4. Second group of deformed modal shape

Mode 2 $f_2 = 20.57$ Hz	Mode 3 $f_3 = 28.49$ Hz
deformed modal shape for plate index 1	
	
modal pressure in air cavity	
	
deformed modal shape for plate index 2	
	

The practical effect of the coupling on the conception quality of the system is the attenuation of vibration in the second plate. In fact, we always have two eigenmodes at the same eigenfrequency value, one in phase and the other in opposition. They are expected to generate this attenuation via the fluid.

3.2. Dynamic responses

This part deals with the dynamic responses of a double panel system. The features of the system are the same as those mentioned before in table 1. The system is exited by an incident wave exerted on plate index 1. The numerical results obtained in the case of a double panel system were compared to those of a simple panel system.

In fact, the two important quantities in vibroacoustic are respectively the acoustic transmission loss factor and the insertion loss. The acoustic transmission loss factor noted TL is given by the formula:

$$\mathrm{TL}(\omega) = 10 \ \ \mathrm{Log}_{10}\left(\frac{\mathsf{P}_{inc}}{\mathsf{P}_{rad}}\right) , \tag{22}$$

where P_{inc} is the incident power and P_{rad} is the power radiated by the second plate in the semi infinite fluid domain V_2.

The insertion loss noted IL is defined as the ratio between the radiated sound power of a simple and a double panel system. It is given by the formula:

$$\mathrm{IL}\left(\omega\right) = 10 \ \ \mathrm{Log}_{10}\left(\frac{\left(\mathsf{P}_{rad}\right)_{simple}}{\left(\mathsf{P}_{rad}\right)_{double}}\right) . \tag{23}$$

The acoustic transmission loss factor for both simple panel and double panel systems is plotted in figure 2.

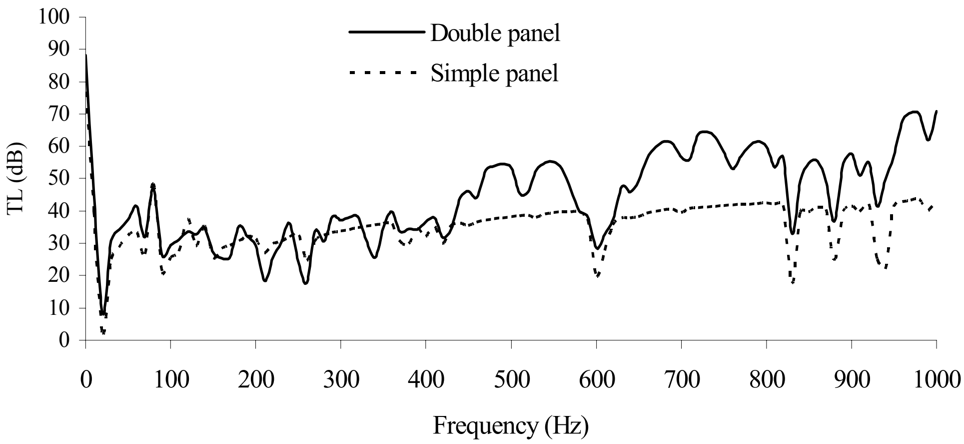


Fig. 2. Acoustic transmission loss factor

The minimums of the TL correspond to the excited modes. We note that double and simple panels have the same minimums for some frequencies located at about 600, 830, 880, 930 Hz and which are close to the first eigenfrequencies group, when the two plates vibrate in phase and there is no cavity influence

Figure 3 shows the calculated insertion loss of the double panel system.

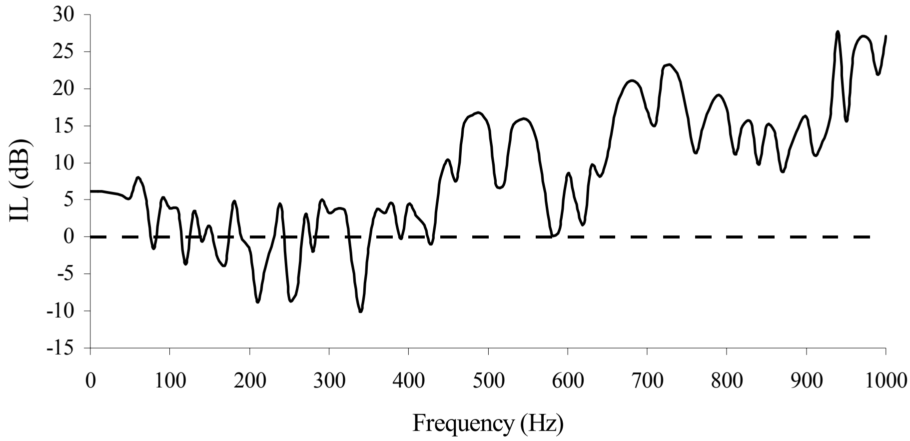


Fig. 3. Insertion loss

Dynamic results show the advantages of the use of a double panel system in comparison of a simple panel. In fact, for a large band of frequency, the energy transmitted by a double panel system is less important than the one transmitted by a simple panel. Only the insertion loss curve shows a bad sound insulation character of a double panel system in the mass-air-mass frequency region (the mass-air-mass resonance occurs at 250 Hz [13, 21]).

CONCLUSIONS

The vibratory analysis performed for a double glazing system shows the existence of two modes groups in which plates vibrating in phase without fluid coupling for the first group and in opposition with a strong fluid coupling for the second one. Moreover, dynamic analysis results for a double panel system show the great influence of the fluid cavity on plate vibration and which generates an important reduction for energy transmitted.

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