STRUCTURAL SENSITIVITY IN THE EVALUATION OF NATURAL FREQUENCIES OF VIBRO-ACOUSTIC SYSTEMS

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Abstract. *The treatment of coupled problems considering the fluid-structure interaction, as can be seen in vibro-acoustic systems, has been a constant target of research. The development of numerical analysis in vibro-acoustic models, a topic of this work, under defined boundary conditions, is important not only for understanding the physical phenomenon, but also to acquire sensitivity relative to factors that influence the vibro-acoustic modal analysis. Among the research objectives are considered: the appreciation of a simple formulation for the modal analysis of vibro-acoustic systems, the programming of this formulation for comparison with results from commercial software, and the modal sensitivity analysis on the structural change. This work uses a finite element discretization of the system, setting the modal analysis by means of a non symmetrical matrix formulation u-p, with the structural displacement u and the pressure p of the fluid. Then, we evaluate the influence of thickness in the evaluation of coupled natural frequencies. The set of results serves to the modal control of the vibro-acoustic system.*

Keywords*: vibro-acoustic, fluid-structure interaction, coupled system*

1. INTRODUCTION

In a fluid-structure coupled system under dynamic enforcement, the presence of the fluid domain influences the behavior of the structure and vice-versa, the vibration of the structure is influenced by the variation of the fluid pressure and the acoustic waves are sensitive to the variation of the structural displacement. In the context of free vibration, the natural frequencies and modes of the coupled system are different from the decoupled ones. See for example Moussou (2005), Marburg (2002) and Sigrist and Garreau (2007), among others.

The energy in a coupled mode is divided between the structure and the fluid. Usually the largest amount of energy stays either in the structure or in the fluid, which serves to classify the coupled mode as dominated by the fluid or by the structure, for example De Mello (2003).

Usually, a mode dominated by the structure is originated by an uncoupled structural mode that induces an acoustic mode in the fluid. Equally, a mode dominated by the fluid is an acoustic mode that induces a mode in the structure. The medium through which the fluid influences the movement of the structure is the pressure in the surface interface, as well as the movement of the interface surface modifies the acoustic domain.

The effect of the fluid pressure in the interface surface can be approximated with the term $\boldsymbol{f}_{s\Gamma_I}$, that is part of the excitation in the structural dynamic equation, constituted by vectors of surface and volume structural forces $(\boldsymbol{f}_{s\Gamma_I} + \boldsymbol{f}_{sB})$, according to Eqs. (1) e (2):

$$\boldsymbol{K}_{ss}\boldsymbol{u} + \boldsymbol{M}_{ss}\ddot{\boldsymbol{u}} = \boldsymbol{f}_{s\Gamma_I} + \boldsymbol{f}_{sB} \tag{1}$$

$$\boldsymbol{f}_{s\Gamma_I} = \int_0^L \boldsymbol{n}_s q \, dx \tag{2}$$

where $\boldsymbol{n}_s$ is the vector of shape functions in the structure. Forces in the interface surface of the structure are originated by the action of the fluid, being associated with the normal component of the surface force q to the pressure distribution on the interface.

Considering the method of the finite elements, see Dettemer and Perié (2006); it is possible to substitute q by the expression of nodal polynomial approach for each fluid element $\tilde{p} = \boldsymbol{n}_f^T \boldsymbol{p}$, where $\tilde{p}$ is the approaching of the scalar field of local pressures, $\boldsymbol{n}_f$ is the vector of shape functions of the fluid, and $\boldsymbol{p}$ is the vector of elementary nodal pressures, resulting Eq. (3):

$$\boldsymbol{f}_{s\Gamma_I} = \int_0^L \boldsymbol{n}_s \boldsymbol{n}_f^T \, dx \, \boldsymbol{p} \tag{3}$$

that represents the equilibrium condition in the interface. Substituting Eq. (3) in Eq. (1) furnishes Eqs. (4) and (5).

$$\boldsymbol{K}_{ss}\boldsymbol{u} + \boldsymbol{M}_{ss}\ddot{\boldsymbol{u}} + \boldsymbol{K}_{sf}\,\boldsymbol{p} = \boldsymbol{f}_s \tag{4}$$

$$\boldsymbol{K}_{sf} = -\int_0^L \boldsymbol{n}_s \boldsymbol{n}_f^T \, dx \tag{5}$$

The coupling of the structural domain with the fluid domain is imposed in the normal direction $\hat{n}$ of the interface surface, through an identity that guarantees the kinematical compatibility:

$$\dot{\boldsymbol{v}}_{\hat{\boldsymbol{n}}} = \ddot{\boldsymbol{u}}_{\hat{\boldsymbol{n}}} \tag{6}$$

that represents a slipping condition in the tangential direction to the interface.

Relative to the fluid domain, the fluid-structure coupling is described in terms of the pressure variations in the neighborhood of the structural domain according to the boundary condition given by Eq. (7), after usage of Eq. (6):

$$\frac{\partial p}{\partial \hat{n}} = -\rho_f \, \ddot{u}_{\hat{n}}, \quad \text{em } \Gamma_I \tag{7}$$

where ρ_f is the fluid density. After substitution of the component in the normal direction $\dot{v}_{\hat{n}}$ by $\ddot{u}_{\hat{n}}$, and considering the expression $\ddot{\tilde{u}} = \boldsymbol{n}_s^T \ddot{\boldsymbol{u}}$ to approximate the value of $\ddot{u}_{\hat{n}}$ by $\ddot{\tilde{u}}_{\hat{n}}$, or in discretized form by $\boldsymbol{n}_s^T \ddot{\boldsymbol{u}}$, we have Eq. (8):

$$\boldsymbol{M}_{ff}\,\ddot{\boldsymbol{p}} + \boldsymbol{K}_{ff}\,\boldsymbol{p} + \boldsymbol{M}_{fs}\ddot{\boldsymbol{u}} = \boldsymbol{f}_f \tag{8}$$

being the matrix with the interface terms expressed by Eq. (9):

$$\boldsymbol{M}_{fs} = \int_{\Gamma_I} \boldsymbol{n}_f \boldsymbol{n}_s^T \, d\Gamma_I \tag{9}$$

that allows to write the coupled system in semi-discretized form. Rewriting Eqs. (4) and (8) in united form:

$$\boldsymbol{K}_{ss}\boldsymbol{u} + \boldsymbol{M}_{ss}\ddot{\boldsymbol{u}} + \boldsymbol{K}_{sf}\,\boldsymbol{p} = \boldsymbol{f}_s \tag{10}$$

$$\boldsymbol{M}_{ff}\,\ddot{\boldsymbol{p}} + \boldsymbol{K}_{ff}\,\boldsymbol{p} + \boldsymbol{M}_{fs}\ddot{\boldsymbol{u}} = \boldsymbol{f}_f \tag{11}$$

that placed in a compact matricial form, it generates the coupled formulation ***u-p*** in displacement of the structure and pressure of the fluid:

$$\begin{bmatrix} \boldsymbol{M}_{ss} & \boldsymbol{0} \\ \boldsymbol{M}_{fs} & \boldsymbol{M}_{ff} \end{bmatrix} \begin{Bmatrix} \ddot{\boldsymbol{u}} \\ \ddot{\boldsymbol{p}} \end{Bmatrix} + \begin{bmatrix} \boldsymbol{K}_{ss} & \boldsymbol{K}_{sf} \\ \boldsymbol{0} & \boldsymbol{K}_{ff} \end{bmatrix} \begin{Bmatrix} \boldsymbol{u} \\ \boldsymbol{p} \end{Bmatrix} = \begin{Bmatrix} \boldsymbol{f}_s \\ \boldsymbol{f}_f \end{Bmatrix} \tag{12}$$

For the case of free vibrations, the second term of Eq. (12) is zero. The non symmetrical presentation of this formulation is its most important disadvantage, because it is not possible to use several efficient algorithms developed for symmetrical matrices. The major advantage of that formulation is its reduced number of degrees of freedom to model the fluid domain. This study intends to discover the influence of the structural thickness in the coupling of the fluid-structure system through the determination of the natural frequencies.

2. VIBRO-ACOUSTIC SYSTEM: ACOUSTIC CAVITY OVER PLATE

The programming for modal analysis of the coupled fluid-structure systems was accomplished in the program MATLAB©, because of the numerical and graphical available tools, enlarging the possibilities of the program

MEFLAB. The program of finite elements for academic research MEFLAB was developed into the MATLAB© environment, being in permanent development. For the study of simulations of the coupled systems and for matching the results after programming, the program Ansys® is used.

This section shows a fluid-structure system constituted by an acoustic cubic cavity on a plate; discussing the numerical tests of the employed formulations to calculate the natural frequencies and vibration modes by programming and simulation.

That system is analyzed with the objective of showing the weak coupling of the coupled modes, when the values of the frequencies of the uncoupled systems don't result close for the interval of frequencies considered.

2.1. Square plate

We study a square thin plate in bending of side 0,3048 m and thickness 3.2766 mm supported in the four vertexes. The used material shows the following properties: modulus of elasticity E=73,084x10^9 N/m^2, coefficient of Poisson ν=0,3 and density ρ=2821 kg/m^3.

The plate is represented through a mesh (8x8), counting 64 two-dimensional elements (denominated *SHELL63* in Ansys®) of 4 nodes each one and totaling 81 nodes. Additionally, it is considered that the displacements of the vertexes are equal to zero. This way the boundary conditions are u=v=r_z=0 entirely in the plate and still in the vertexes z=0, Fig. 1.

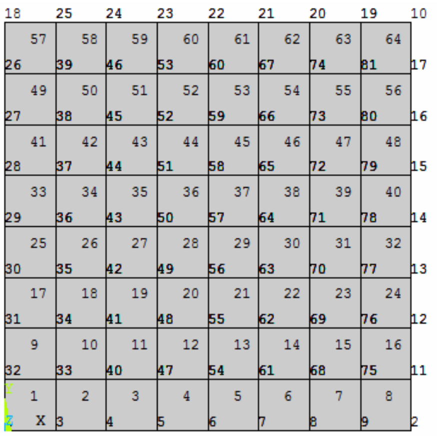


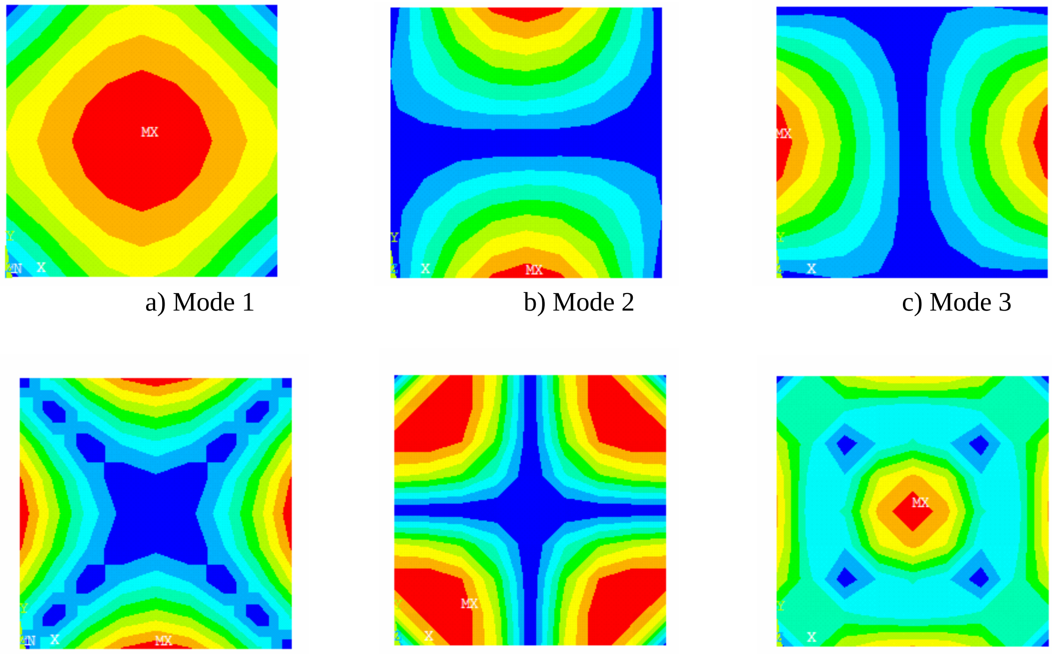
Figure 1. Mesh of the square plate with indication of the boundary conditions.

The natural frequencies obtained by programming are compared, in the Table 1, with analytical, experimental and simulated solutions. The average of the results variation obtained by programming relative to the analytical ones for the first three frequencies is equal to 0.23%, value considered appropriated if considered that the variation of similar results obtained by simulation is equal to 0,78%.

Table 1. Frequencies predicted in Hz of a square plate supported in the vertexes.

Mode	Analytical (Blevins. 1995)	Experimental (Reid. 1965)	MEF (Petyt and Mirza. 1972)	MEF Simulation	Variation (%) Simulation / Analytical	MEF Programming	Variation (%) Programming / Analytical
1	61.56	62	62.09	62.19	1.02	61.90	0.55
2	136.61	134	138.5	137.51	0.66	136.70	0.07
3	136.61	134	138.5	137.51	0.66	136.70	0.07
4		169	169.7	169.58		169.61	
5		330	340.0	339.44		335.43	
6		383	396.0	383.78		381.17	
7				442.42		439.77	
8				442.42		439.77	
9				606.53		600.50	
10				695.57		688.99	

It is observed in Figure 2 that there are two modes, the second and the third, with different modal patterns but with identical frequencies because of the symmetry.



a) Mode 1 b) Mode 2 c) Mode 3

d) Mode 4 e) Mode 5 f) Mode 6

Figure 2. Patterns of modes of the square plate supported in the vertexes.

2.2. Acoustic cubic cavity

In global matrix notation, after assembly of the elements, the purely acoustic cubic cavity is governed by one differential equation as given by Eq. (13), after canceling the structural elements and related ones of Eq. (12):

$$\boldsymbol{M}_{ff}\,\ddot{\boldsymbol{p}} + \boldsymbol{K}_{ff}\,\boldsymbol{p} = \boldsymbol{f}_f \tag{13}$$

The matrix of inertia or compressibility of the fluid $\boldsymbol{M}_{ff}$, the volumetric matrix of the kinetic energy of the fluid $\boldsymbol{K}_{ff}$ and the vector with the external excitations $\boldsymbol{f}_f$, are defined by:

$$\boldsymbol{M}_{ff} = \frac{1}{\rho_f c_s^2}\int_{\Omega_f} \boldsymbol{n}_f \boldsymbol{n}_f^T d\Omega_f \tag{14}$$

$$\boldsymbol{K}_{ff} = \frac{1}{\rho_f}\int_{\Omega_f} \boldsymbol{B}^T \boldsymbol{B} d\Omega_f \tag{15}$$

$$\boldsymbol{f}_f = \frac{1}{\rho_f}\int_{\Omega_f} \boldsymbol{n}_f P_B d\Omega_f \tag{16}$$

where c_s is the sound velocity in the fluid domain, P_B involves the body forces and,

$$\boldsymbol{B} = \begin{Bmatrix} \dfrac{\partial \boldsymbol{n}_f^T}{\partial x} \\ \dfrac{\partial \boldsymbol{n}_f^T}{\partial y} \\ \dfrac{\partial \boldsymbol{n}_f^T}{\partial z} \end{Bmatrix} \tag{17}$$

We study an acoustic cavity of side 0,3048 m. The internal fluid is air to 20ºC with density ρ_f=1.204 kg/m^3 and sound speed c_s=343.3 m/s.

The cavity is represented through a mesh (8x8x8) of hexaedrical solid elements (denominated as *FLUID30* in Ansys®) of 8 nodes each one, totaling 512 elements and 729 nodes. It is considered that the boundary conditions of the cavity are rigid walls, Figure 3.

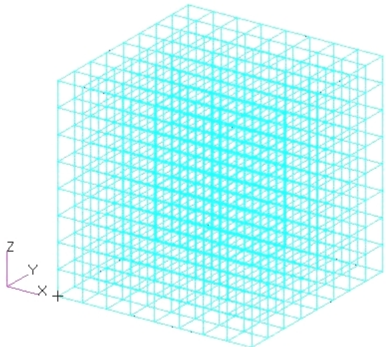


Figure 3. Acoustic cavity with boundary conditions of rigid wall in the faces.

The natural frequencies obtained by programming are compared, in Table 2, with analytical and simulated solutions. The analytical natural frequencies of the acoustic cavity are obtained through Eq. (13), (Blevins, 1995); where the walls of the cavity are considered as rigid with infinite impedance.

The average of the results variation obtained by programming and simulation relative to the analytical values is 1.56%, meaning the agreement of the programmed and simulated results.

$$f = \frac{c_s}{2}\sqrt{\left(\frac{i}{L_x}\right)^2 + \left(\frac{j}{L_y}\right)^2 + \left(\frac{k}{L_z}\right)^2} \qquad i = 0,1,2,\ldots;\ j = 0,1,2,\ldots;\ k = 0,1,2,\ldots \qquad (18)$$

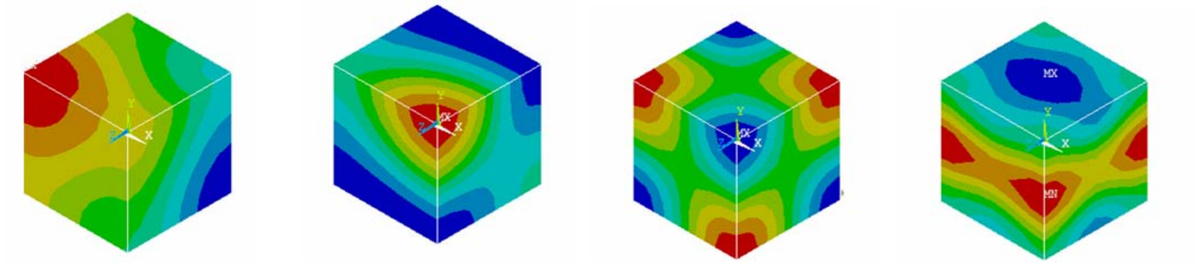
where,

f : natural frequency in Hz;
c_s : sound speed in air, equal to 343,3 m/s;
L_x, L_y, L_z : length, wide and height of the cavity, in this case all equal to 0.3048 m..

Table 2. Predicted frequencies of the acoustic cavity in Hz.

Mode	Notation (*i*,*j*,*k*)	Analytical (Blevins, 1995)	MEF Simulation	Variation (%) Simulation / Analytical	MEF Programming	Variation (%) Programming / Analytical
1	(0,0,0)	0.0	0.0	0.00	0.0	0.00
2	(0,0,1)	563.2	566.78	0.64	566.80	0.64
3	(0,1,0)	563.2	566.78	0.64	566.80	0.64
4	(1,0,0)	563.2	566.78	0.64	566.80	0.64
5	(0,1,1)	796.4	801.55	0.65	801.50	0.64
6	(1,0,1)	796.4	801.55	0.65	801.50	0.64
7	(1,1,0)	796.4	801.55	0.65	801.50	0.64
8	(1,1,1)	975.4	981.69	0.64	981.70	0.65
9	(0,0,2)	1126.3	1155.40	2.58	1155.40	2.58
10	(0,2,0)	1126.3	1155.40	2.58	1155.40	2.58
11	(2,0,0)	1126.3	1155.40	2.58	1155.40	2.58
12	(0,1,2)	1259.3	1287.00	2.20	1287.00	2.20
13	(0,2,1)	1259.3	1287.00	2.20	1287.00	2.20
14	(1,0,2)	1259.3	1287.00	2.20	1287.00	2.20
15	(1,2,0)	1259.3	1287.00	2.20	1287.00	2.20
16	(2,0,1)	1259.3	1287.00	2.20	1287.00	2.20
17	(2,1,0)	1259.3	1287.00	2.20	1287.00	2.20
18	(1,1,2)	1379.4	1406.20	1.94	1406.20	1.94
19	(1,2,1)	1379.4	1406.20	1.94	1406.20	1.94
20	(2,1,1)	1379.4	1406.20	1.94	1406.20	1.94

It is observed in Figure 4 the patterns of three modes with different natural frequencies.



a) Mode 2 b) Mode 3 c) Mode 8 d) Mode 9

Figure 4. Patterns of modes of an acoustic cubic cavity with different natural frequencies.

2.3. Acoustic cubic cavity over square plate

We study an acoustic cubic cavity over a square plate, modeled as a fluid-structure coupled system. The plate is modeled through two-dimensional elements, with dimensions and material specified in the section 2.1. The cavity is modeled with hexaedrical solid elements, with the dimensions of the cavity and fluid properties specified in the section 2.2.

It is considered that the displacements of the vertexes of the plate are equal to zero. This way, the boundary conditions are $u=v=r_z=0$ entirely in the plate and still $z=0$ in its vertexes. Also, the cavity is outlined by rigid walls.

The adopted simplification results in a mesh of 64 two-dimensional elements of four nodes each one and 512 hexaedrical solid elements of 8 nodes each one, totaling 576 elements and 810 nodes.

Table 3. Predicted frequencies I by programming of the cubic cavity over a square plate in Hz.

Plate	Cavity	Decoupled mode	Coupled mode	Coupled	Variation (%) coupled/decoupled	Variation Hz coupled/decoupled
	0.00	F1	1	0	0.00	0
61.90		S1	2	64.20	3.71	2.30
136.69		S2	3	136.50	-0.14	-0.19
136.69		S3	4	136.50	-0.14	-0.19
169.61		S4	5	169.50	-0.06	-0.11
335.43		S5	6	335.10	-0.10	-0.33
381.17		S6	7	380.90	-0.07	-0.27
439.77		S7	8	439.40	-0.08	-0.37
439.77		S8	9	439.40	-0.08	-0.37
	566.80	F2	10	569.90	0.55	3.10
	566.80	F3	11	569.90	0.55	3.10
	566.80	F4	12	570.20	0.60	3.40
600.50		S9	13	600.20	-0.05	-0.30
688.99		S10	14	688.60	-0.06	-0.39
688.99		S11	15	688.60	-0.06	-0.39
	801.50	F5	16	805.70	0.52	4.20
	801.50	F6	17	806.00	0.56	4.50
	801.50	F7	18	806.00	0.56	4.50
809.82		S12	19	809.30	-0.06	-0.52
	981.70	F8	20	986.80	0.52	5.10
1013.64		S13	21	1013.20	-0.04	-0.44
1013.64		S14	22	1013.20	-0.04	-0.44
1020.68		S15	23	1020.30	-0.04	-0.38
1039.04		S16	24	1038.70	-0.03	-0.34
	1155.40	F9	25	1161.30	0.51	5.90
	1155.40	F10	26	1161.30	0.51	5.90
	1155.40	F11	27	1161.40	0.52	6.00
1180.42		S17	28	1180.00	-0.04	-0.42
	1287.00	F12	29	1293.50	0.51	6.50
	1287.00	F13	30	1293.50	0.51	6.50

We can see in Table 3 the average variation of the first 30 coupled frequencies obtained by programming relative to the uncoupled systems; for frequencies predominantly structural S*i* (*i* = 1,2,...) results 0.15%, for frequencies predominantly fluid F*j* (*j* = 1,2,...) results 0.49% and for all frequencies 0.3%. The maximum value of variation is 6.50 Hz. These results highlight that there is a weak coupling.

It is observed in Table 4 the average variation of the first 30 coupled frequencies obtained by simulation relative to the decoupled values, for the predominantly structural frequencies is 1.02%, for the predominantly fluid frequencies is 0.12% and for all frequencies is 0.63%. The maximum variation is 8.3 Hz.

Table 4. Predicted frequencies II by simulation of the cubical cavity over a square plate in Hz.

Plate	Cavity	Decoupled mode	Coupled mode	Coupled	Variation (%) coupled/decoupled	Variation Hz coupled/decoupled
	0	F1	1	0	0	0
62.187		S1	2	70.479	13.34	8.3
137.51		S2	3	136.74	0.56	-0.77
137.51		S3	4	136.74	0.56	-0.77
169.58		S4	5	169.09	0.3	-0.49
339.44		S5	6	338.05	0.41	-1.39
383.78		S6	7	382.73	0.27	-1.05
442.42		S7	8	441.22	0.27	-1.2
442.42		S8	9	441.22	0.27	-1.2
	566.78	F2	10	567.89	0.2	1.11
	566.78	F3	11	567.89	0.2	1.11
	566.78	F4	12	569.07	0.4	2.29
606.53		S9	13	605.54	0.16	-0.99
695.57		S10	14	694.16	0.2	-1.41
695.57		S11	15	694.16	0.2	-1.41
	801.55	F5	16	802.27	0.09	0.72
	801.55	F6	17	803.27	0.21	1.72
	801.55	F7	18	803.27	0.21	1.72
821.05		S12	19	819.4	0.2	-1.65
	981.69	F8	20	982.5	0.08	0.81
1017.9		S13	21	1016.7	0.12	-1.2
1029.1		S14	22	1027.7	0.14	-1.4
1029.1		S15	23	1027.7	0.14	-1.4
1058.8		S16	24	1057.7	0.1	-1.1
	1155.4	F9	25	1155.9	0.04	0.5
	1155.4	F10	26	1156	0.05	0.6
	1155.4	F11	27	1156.4	0.08	1
1188.6		S17	28	1187.6	0.08	-1
	1287	F12	29	1287.4	0.03	0.4
	1287	F13	30	1287.4	0.03	0.4

2.4. Square plate of minor thickness

This is the square plate of section 2.1, but with a smaller thickness, 2.62128 mm (80% of the thickness of section 2.1).

The natural frequencies obtained by programming are compared in Table 5, with analytical and simulated solutions. The average of the results variation obtained by programming relative to the analytical ones for the first three frequencies is equal to 0.22%, considered adequate if we consider that the variation of similar results using the simulation is equal to 0.78%.

By analyzing the natural frequencies for the square plate of minor thickness compared to the frequencies of the square plate of the section 2.1, we can verify that the frequencies change in direct proportion to changes made in the plate thickness, i.e. if the thickness decreases 20%, all natural frequencies also decrease around 20%, that can be proven by Eq. (19), which originates from the elementary structural stiffness and mass expressions.

$$f = \sqrt{\frac{\boldsymbol{k}_{ss}^{e}}{\boldsymbol{m}_{ss}^{e}}} = \sqrt{\frac{E\,h^{3}\,[\cdots]}{\rho\,h\,a\,b\,[\cdots]}} = h\,[\cdots] \qquad (19)$$

A similar finding also occurs with the fluid medium, as can be seen in Eq. (20), where the natural frequencies change in direct proportion to the relative change of the speed of sound in the medium.

$$f = \sqrt{\frac{\boldsymbol{k}_{ff}^{e}}{\boldsymbol{m}_{ff}^{e}}} = \sqrt{\frac{\frac{a\,b\,c}{\rho_f}[\cdots]}{\frac{1}{\rho_f}a\,b\,c[\cdots]}} = c_s[\cdots] \tag{20}$$

Table 5. Predicted frequencies in Hz of a square plate with minor thickness

Mode	Analytical (Blevins, 1995)	MEF Simulation	Variation (%) Simulation / Analytical	MEF Programming	Variation (%) Programming / Analytical
1	49.25	49.75	1.02	49.52	0.55
2	109.29	110.01	0.66	109.35	0.05
3	109.29	110.01	0.66	109.35	0.05
4		135.66		135.68	
5		271.55		268.34	
6		307.02		304.93	
7		353.94		351.82	
8		353.94		351.82	
9		485.22		480.4	
10		556.46		551.19	

2.5. Acoustic cubical cavity over a square plate of minor thickness

It is presented in this section the acoustic cubical cavity over a square plate of minor thickness, modeled as a fluid-structure coupled system. The plate is modeled by two-dimensional elements, with dimensions and material specified in section 2.1. The cavity is modeled with hexahedrical solid elements with dimensions of the cavity and fluid properties specified in section 2.2.

All the considerations and simplifications as defined in section 2.3 are applied for the coupled system.

We can see in Table 6 the average variation of the first 30 coupled frequencies obtained by programming relative to the uncoupled values; for the predominantly structural frequencies Si (i = 1,2,...) is 0.30%, for the predominantly fluid frequencies Fj (j = 1,2,...) is 0.04% and for all frequencies is 0.20%. The maximum variation is 3.51 Hz.

Table 6. Predicted frequencies I by programming of the cubical cavity over a plate of minor thickness in Hz.

Plate	Cavity	Decoupled mode	Coupled mode	Coupled	Variation (%) coupled/decoupled	Variation Hz coupled/decoupled
	0	F1	1	0	0	0
49.52		S1	2	53.03	7.09	3.51
109.35		S2	3	109.16	-0.17	-0.19
109.35		S3	4	109.16	-0.17	-0.19
135.68		S4	5	135.56	-0.09	-0.12
268.34		S5	6	267.98	-0.13	-0.36
304.93		S6	7	304.67	-0.09	-0.26
351.82		S7	8	351.5	-0.09	-0.32
351.82		S8	9	351.5	-0.09	-0.32
480.4		S9	10	480.13	-0.06	-0.27
551.19		S10	11	550.69	-0.09	-0.50
551.19		S11	12	550.69	-0.09	-0.50
	566.80	F2	13	567.26	0.08	0.46
	566.80	F3	14	567.26	0.08	0.46
	566.80	F4	15	567.45	0.11	0.65
647.86		S12	16	647.43	-0.07	-0.43
	801.50	F5	17	801.66	0.02	0.16
	801.50	F6	18	801.91	0.05	0.41
	801.50	F7	19	801.91	0.05	0.41
810.91		S13	20	810.66	-0.03	-0.25
810.91		S14	21	810.66	-0.03	-0.25

816.54		S15	22	816.17	-0.05	-0.37
831.23		S16	23	830.86	-0.04	-0.37
944.33		S17	24	943.91	-0.04	-0.42
	981.70	F8	25	982.13	0.04	0.43
1102.58		S18	26	1102.16	-0.04	-0.42
	1155.40	F9	27	1155.62	0.02	0.22
	1155.40	F10	28	1155.62	0.02	0.22
	1155.40	F11	29	1155.82	0.04	0.42
1157.47		S19	30	1156.89	-0.05	-0.58

We can see in Table 7 the average variation of the first 30 coupled frequencies obtained by simulation relative to the uncoupled values; for the predominantly structural frequencies is 1,60%, for the predominantly fluid frequencies is 0,21% and for all frequencies is 1,09 %. The maximum variation is 12,43 Hz.

Table 7. Predicted frequencies II by simulation of the cubical cavity over a plate of minor thickness in Hz

Plate	Cavity	Decoupled mode	Coupled mode	Coupled	Variation (%) coupled/decoupled	Variation Hz coupled/decoupled
	0	F1	1	0	0	0
49.75		S1	2	62.184	12.43	25
110.01		S2	3	109.25	-0.76	0.7
110.01		S3	4	109.25	-0.76	0.7
135.66		S4	5	135.18	0.48	0.35
271.55		S5	6	270.21	1.34	0.49
307.02		S6	7	306.07	0.95	0.31
353.94		S7	8	352.83	1.11	0.31
353.94		S8	9	352.83	1.11	0.31
485.22		S9	10	484.25	0.97	0.2
556.46		S10	11	554.28	2.18	0.4
556.46		S11	12	554.28	2.18	0.4
	566.78	F2	13	568.97	2.19	0.4
	566.78	F3	14	568.97	2.19	0.4
	566.78	F4	15	569.51	2.73	0.5
656.84		S12	16	655.39	1.45	0.22
	801.55	F5	17	802.02	0.47	0.05
	801.55	F6	18	803.45	1.9	0.24
	801.55	F7	19	803.45	1.9	0.24
814.3		S13	20	813.17	1.13	0.14
823.26		S14	21	822.01	1.25	0.15
823.26		S15	22	822.01	1.25	0.15
847.06		S16	23	845.77	1.29	0.15
950.91		S17	24	949.58	1.33	0.14
	981.69	F8	25	983.42	1.73	0.17
1130.6		S18	26	1129.3	1.3	0.11
	1155.4	F9	27	1156.1	0.7	0.06
	1155.4	F10	28	1156.1	0.7	0.06
	1155.4	F11	29	1157	1.6	0.14
1175.6		S19	30	1173.9	1.7	0.14

For plates of thicknesses 0.00262128, 0.0032766 and 0.00393192 were obtained, from simulation results, the following values of RMS Variation Hz coupled/decoupled: 2.67, 1.9 and 1.58, respectively. With these results we construct the Figure 5, which indicates the non linear variation between the plate thickness and the RMS Variation Hz coupled/decoupled. As the plate thickness diminishes, the RMS value grows up.

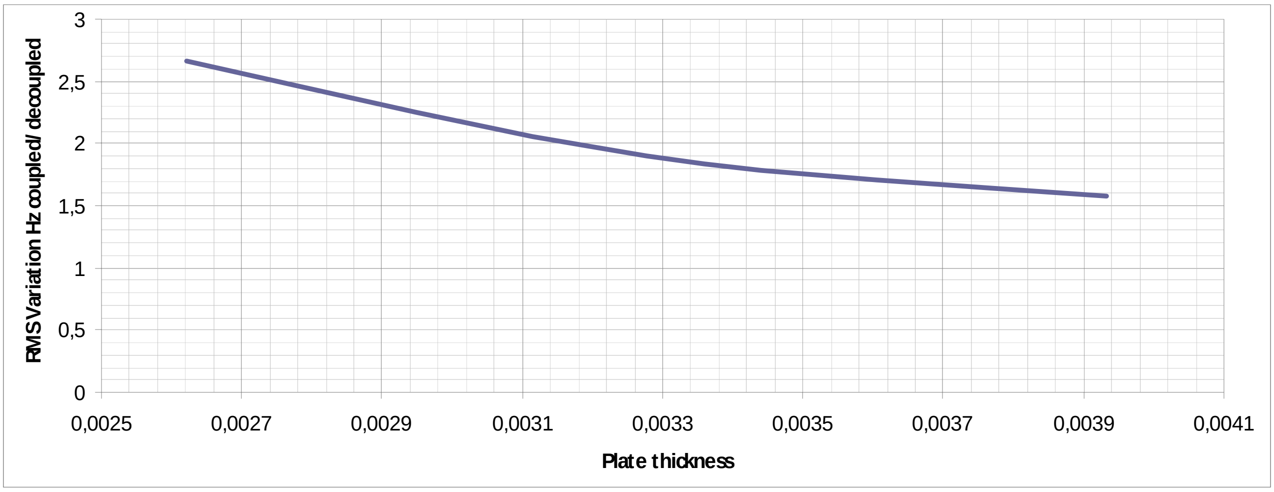


Figure 5. RMS Variation Hz coupled/decoupled as function of plate thickness.

3. CONCLUSIONS

Using both programming (Table 6) and simulation (Table 7) in structurally modified vibro-acoustic systems, as compared to the coupled system which adopts the structure of a square plate with the original thickness (section 2.3), we perceive changes in the coupled frequencies values, with greater variation of the predominantly structural frequencies, and minor variation of the predominantly fluid frequencies, as caused by a 20% reduction in the thickness of the square plate described in section 2.4.

4. ACKNOWLEDGEMENTS

This works counts with some scholarships: BIC-UFRGS (L.G.F., M.A.N.G.) and PIBIC-CNPq-AF/UFRGS (A.D.O.).

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