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VIBRO - ACOUSTIC LOW FREQUENCY ANALYSIS OF THE VLS EQUIPMENT BAY USING FINITE ELEMENTS

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Abstract

Satellite launchers produce high intensity acoustic excitation. Even a small launcher, like the Brazilian VLS, can generate values of sound pressure level of 140–150 dB. Such severe loads can affect not only the payload but also the equipment bay, which contains some important control equipment. In order to achieve a better knowledge of the dynamic behavior of the equipment bay tests and numerical simulation should be performed. In this work the acoustic dynamic behavior of the equipment bay is analyzed in the low frequency range using the finite element method. The need of a coupled analysis is verified and some passive strategies to reduce the sound energy level inside the cavity are also simulated in the numerical model.

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INTRODUCTION

During take off satellite launchers generate and suffer very strong acoustic loads. Typical values of overall sound pressure level (OSPL) lies in the range 140-150 dB even for a small launcher like the Brazilian VLS, a solid propellant 50 tons rocket. These loads could affect the launcher payload and other internal equipment also. Therefore, the prediction of the dynamic acoustic behavior of some launcher cavities is of great importance [7]. In the low frequency range the finite element method could efficiently predict the dynamic behavior of coupled systems [2]. It was shown that the structure - fluid coupling in the VLS fairing system is weak and so an uncoupled analysis could give reasonable results with small computational cost [4].

The most used passive methods to reduce the acoustic loads into launchers fairings are Helmholtz resonators and acoustic absorber blankets [6].

In this work we evaluated the fluid – structure coupling in the VLS equipment bay by comparing results from coupled and uncoupled eigenanalysis. Also a preliminary analysis of the internal sound pressure level was done by submitting the system to an harmonic uniformly distributed radial load corresponding to an external sound pressure level of 150dB. An impedance boundary condition was included in order to simulate the placing of acoustic blankets on the walls. The sound pressure levels calculated with and without impedance were then compared. All calculations were performed using the finite element code ANSYS [1].

STRUCTURE GOVERNING EQUATIONS

The displacements of a linear elastic domain limited by a boundary Γ $(\Gamma = \Gamma_1 \cup \Gamma_2)$ with viscous damping are governed by:

$$\sigma_{ij,j}(u) - \rho_s \frac{\partial^2 u_i}{\partial t^2} + d \frac{\partial u_i}{\partial t} = 0 \tag{1}$$

subject to the following boundary conditions:

$$\sigma_{ij,j}(u) n_j = F_i \qquad on \qquad \Gamma_1 \tag{2}$$

$$u_i = 0 \qquad on \qquad \Gamma_2 \tag{3}$$

Where ρ_s is the structure specific mass, *u* is the displacement vector, *t* is the time, *n* is a vector normal to the boundary, *d* is the damping constant and *F* is the force vector acting on the domain boundary. The problem defined by the equation (1) and the boundary conditions (2) and (3) can be discretized by the finite element method [8]:

$$[M]\{\ddot{u}\} + [D]\{\dot{u}\} + [K]\{u\} = \{F\} \tag{4}$$

Where $[M]$, $[D]$ and $[K]$ are respectively the mass, damping and stiffness matrices, $\{u\}$ is the nodal displacement vector and $\{F\}$ is the nodal external forces vector.

FLUID GOVERNING EQUATIONS

A fluid domain bounded by a surface Ω_f $(\Omega = \Omega_1 + \Omega_2 + \Omega_3 + \Omega_4)$, is governed by the Helmholtz equation [1],[2]:

$$\nabla^2 p - \frac{1}{c^2}\frac{\partial^2 p}{\partial^2 t} + \left(\frac{r}{\rho_f c}\right)\frac{1}{c}\frac{\partial p}{\partial t} = Q \tag{5}$$

Which is subjected to the following boundary conditions:

$$p = \overline{p} \qquad on \qquad \Omega_1 \tag{6}$$

$$\frac{\partial p}{\partial n} = \overline{v}_n \qquad on \qquad \Omega_2 \tag{7}$$

$$p = \overline{Z} v_n \qquad on \qquad \Omega_3 \tag{8}$$

Where ρ_f is the fluid specific mass, r is the characteristic impedance of the absorbent material, p is the fluid pressure; c is the sound velocity within the fluid; t is the time, Q is the term of the acoustic sources within the fluid volume, n is the unit normal to the boundary, $\overline{Z}$ is the impedance on the boundary, v_n is the prescribed boundary normal velocity and ∇ is the Laplacian operator. The upper bar indicates prescribed values. The term $\left(\frac{r}{\rho_f c}\right)$ is called boundary absorption coefficient (β), which is commonly frequency dependent.

The boundary value problem defined by the equation (5) and subjected to the boundary conditions (6), (7) and (8) can be discretized by the finite element method:

$$[H]\{p\} + [C]\{\dot{p}\} + [E]\{\ddot{p}\} = \{Q\} \tag{9}$$

Where [E], [H] and [C] are the compressibility, volumetric and acoustic damping matrices respectively and $\{Q\}$ is the excitation vector (acoustic sources).

FLUID-STRUCTURE INTERACTION

When the fluid – structure coupling is considered, the fluid pressure acts on the "wet" part of the structure (the interface). Such force is then included on the right hand side of the equation (1). For the fluid domain the coupling implies a kinematic compatibility in the interface, included by the following boundary condition:

$$\frac{\partial p}{\partial n} = -\rho_f n.\ddot{u} \tag{10}$$

Where ρ_f is the fluid specific mass. The coupled system matrix equation is [2]:

$$\begin{bmatrix} [M] & [0] \\ \rho_f [L]^T & [E] \end{bmatrix} \begin{Bmatrix} \{\ddot{u}\} \\ \{\ddot{p}\} \end{Bmatrix} + \begin{bmatrix} [D] & [0] \\ [0] & [C] \end{bmatrix} \begin{Bmatrix} \{\dot{u}\} \\ \{\dot{p}\} \end{Bmatrix} + \begin{bmatrix} [K] & -[L] \\ [0] & [H] \end{bmatrix} \begin{Bmatrix} \{u\} \\ \{p\} \end{Bmatrix} = \begin{Bmatrix} \{F\} \\ \{Q\} \end{Bmatrix} \tag{11}$$

Where $[L]$ is the fluid – structure interface matrix.

SYSTEM ANALYZED

A general view of the VLS is shown in the figure (1) [5]. The equipment bay is placed under the faring (payload) and the 4th stage.

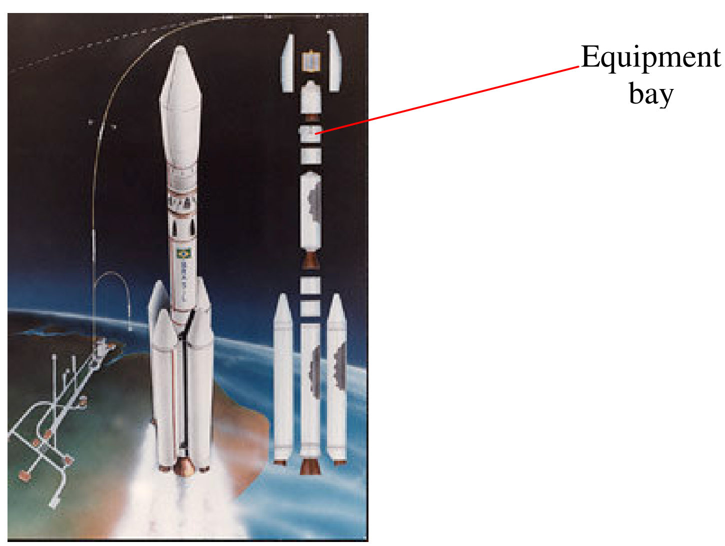


Figure 1 – VLS

FINITE ELEMENT MODEL

The equipment bay is a cylindrical shell with a trussed arm. The cylinder is closed in its bottom part by a circular plate and in the top by the 4th stage. The structural and fluid meshes can be observed in the figure (2). Only half part of the structure is displayed to better show its internal structure.

The cylinder is assumed clamped in its tips. Only nodal vertical displacements are allowed in the upper part where a vertical pre-load of 12446 N was applied (the upper stages weight). The acoustic cavity is considered rigid in the interface with the 4th stage (its upper part) and flexible in all others walls, where the fluid – structure interface boundary condition is applied. The finite element model characteristics are displayed in the table (1).

	Structure	Fluid
Sound velocity	-	340 m/s
Specific mass	2768 kg/m^3	1.225 kg/m^3
Elasticity modulus	0,75 x 10^{11} N/m^2	-
Degrees of freedom	4658	1881
Element used	Linear plate with 4 nodes	Linear hexahedral with 8 nodes
Element degrees of freedom per node	3 translations and 3 rotations	1 (pressure)

Table 1 – Finite element model characteristics

Figure 2 – Structural (left) and fluid (right) meshes

NUMERICAL RESULTS

Natural frequencies and mode shapes were calculated for the uncoupled structure, the uncoupled fluid and for the coupled system. The frequencies are shown in the table (2) and some of the uncoupled and coupled mode shapes are displayed in figure (3). Double modes are indicated in table (2) with (d). It can be noticed from table (2) that the structure modes dominate the behavior of the coupled system at least until 200 Hz. Also, the coupled and uncoupled structure mode shapes displayed in figure (3) are quite similar. So the structure affects severely the acoustic fluid behavior while the fluid influence in the structure dynamics is quite small.

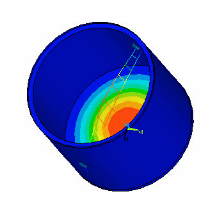
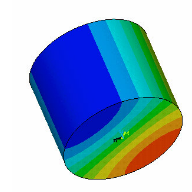
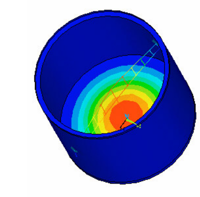
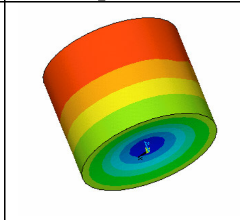
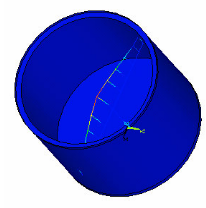
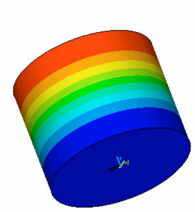
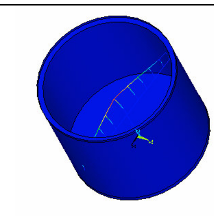
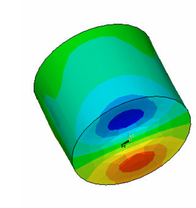
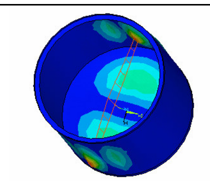
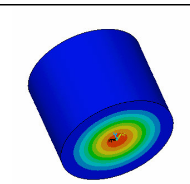
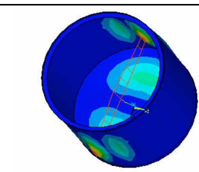
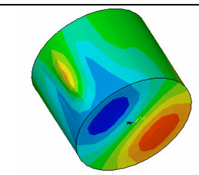
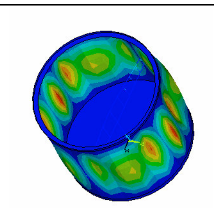
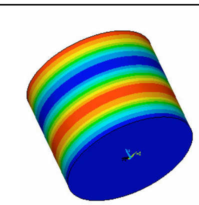
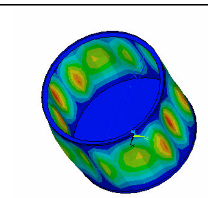
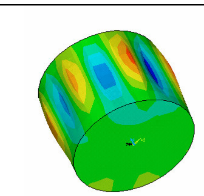
Mode	Uncoupled structure	Uncoupled fluid	Coupled structure	Coupled fluid
1	 36.94 Hz	 198.76 Hz	 37.63 Hz	 37.63 Hz
2	 39.95 Hz	 215.21 Hz	 39.95 Hz	 39.95 Hz
6	 76.91 Hz	 415.20 Hz	 76.88 Hz	 76.88 Hz
21	 237.32 Hz	 678.92 Hz	 236.52 Hz	 236.52 Hz

Figure 3 - Calculated mode shapes

Frequencies (Hz)			
Mode	Uncoupled structure	Uncoupled fluid	Coupled system
1	36.94	198.76 (d)	37.63
2	39.95	215.21	39.95
3	67.58	292.96 (d)	66.71
4	67.85	330.06 (d)	66.99
5	70.99	394.02 (d)	72.11
6	76.91	415.20	76.88
7	98.05	438.73	98.05
8	100.72 (d)	454.63 (d)	99.96 (d)
9	117.93	467.66	117.92

Table 2 – Eigenfrequencies (Hz)

An impedance boundary condition was applied on the same walls where the fluid – structure coupling has been included and a vibro – acoustic harmonic analysis was then performed. Nodal harmonic forces of 5.50 N corresponding to an external sound pressure level of 150 dB were applied on all cylindrical shell nodes. A slice of the structure with the applied forces is shown in the figure (4).

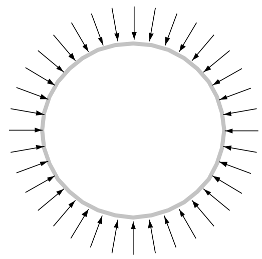


Figure 4 – Structural loads

The frequency band covered by the excitation was 0.5 Hz to 300Hz. To obtain a good description of the peaks, the pressure responses were calculated at each 0.25 Hz. The values of the boundary absorption coefficient (β) considered in the analysis were taken from [3] and are displayed in the table (3). The response curves were calculated with and without impedance. A damping of 1% for the coupled system was included in both cases. A pressure response curve is plotted in figure (5).

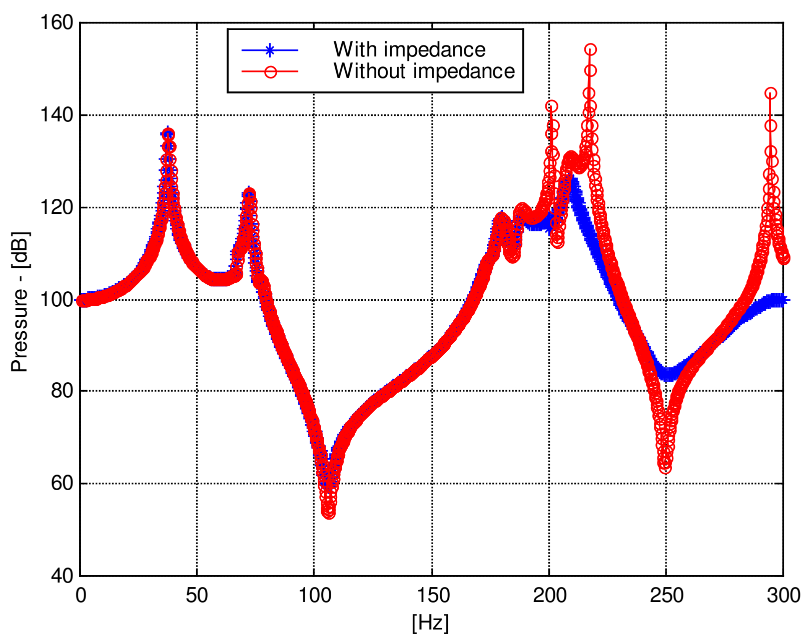


Figure 5 – Pressure response curve

Frequency band (Hz)	β
0-100	0
100-200	0.0375
200-300	0.0625

Table 3 – Boundary absorption coefficient values

CONCLUSIONS

The vibro – acoustical behavior of the VLS equipment bay was analyzed using the finite element method. In a low frequency band (0-300 Hz) the coupled modes are predominantly structural and the structural modes are severely affected by the local modes corresponding to each one of structure's three main components (the circular plate, the cylindrical shell and the trussed internal arm).

A coupled harmonic analysis was performed applying uniformly distributed forces on the cylindrical shell equivalent to an external sound pressure level of 150 dB. When the impedance boundary condition was included the calculated pressure response show a maximum reduction of 45dB in the last peak. Although not all peaks were reduced with such intensity, the impedance introduced on the cavity walls was able to keep the internal sound pressure levels bellow 127 dB in the 0-300 Hz band. Moreover, the sound absorption coefficient increases in higher frequencies and so stronger reductions can be expected for higher frequencies. In future works a better approximation for the real acoustical excitation during the lift off considering its randomic nature should be used. The use of Helmholtz resonators to reduce the internal sound pressure level around specific low frequency peaks should also be analyzed.

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