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Dynamic Analysis of the VLS Fairing – Low-Frequency Numerical X Experimental Comparison

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Abstract: During the lift-off satellite launchers generate high acoustic pressure levels, due to the propulsion system of the engines. Even for small launchers, like the Brazilian VLS, values of sound pressure level of 140–160 dB OASPL can be reached on its higher structural parts. Such severe acoustic loads can affect the payload (or satellite) contained inside the fairing. In order to achieve a better knowledge of the dynamic behavior of fairings when submitted to lift-off acoustic loads, tests and numerical simulations should be performed. Deterministic finite element method can be used to predict the low-frequency dynamic behavior of space systems´ acoustic cavities. Acoustic reverberant chamber tests are used to simulate the diffuse acoustic pressure field generated by launchers. In this work a fluid-structure finite element model of the VLS fairing is built and submitted to an acoustic excitation of 145 dB OASPL. The predicted acoustic responses are compared to the results obtained during tests performed in the Acoustic Reverberant Test Facility (ARTF), a division of the National Institute of Space Research (INPE).

Keywords: Acoustics, Finite – Elements, Acoustic tests, VLS, Fairing

INTRODUCTION

The VLS is a conventional launcher with four stages, launched from an earth platform. During lift-off phase, VLS has a length of 19 meters, with a total mass of 50 tons and a thrust of 100 kN. Its propulsion is assured by solid propellant motors, in all stages, with a total propellant mass of 41 tons. The VLS allows putting satellites ranging from 100 to 350 kg in circular orbits at altitudes ranging from 250 to 1,000 km.

In flight missions, the VLS is subjected to various dynamic loads (Caneilles, 1998). In terms of acoustic solicitation, critical instants during a rocket launching, such as lift-off, transonic flight and maximum dynamic pressure phases, must be studied. These air-borne excitations, which have a broadband and random nature, expose the launcher's upper parts to extreme pressure loadings (Coyette et al. 1997). During lift-off, the noise generated by rocket boosters is reflected by the launch platform and re-injected into the launcher structure and payload compartment. Such severe acoustic levels, estimated on the higher parts of VLS launcher and ranging from 140 - 160 dB OASPL, can damage sensitive parts of the payload, destroying the satellite mission and wasting considerable amount of money. As such, a survey of the VLS fairing vibro-acoustic environment must be carried out, to determine its inner acoustic pressure levels. In this respect, it is very important to have reliable numerical tools that can predict the vibro-acoustic responses of launch vehicle systems to acoustic loads encountered in flight and that enable noise control engineers to optimize the vehicle design.

In this paper a low-frequency coupled technique was used to compute the VLS fairing behavior. The structural FEM/fluid FEM was applied to model the fairing body and its inner acoustic domain. The equivalent 145 dB OASPL excitation was applied on the fairing structural body and coupled calculations were done from 5 to 250 Hz, which yielded the acoustic cavity as well as the skin responses. A modal expansion reduction technique was applied to save computation time.

A very extensive work has been done to determine and manage the vibro-acoustic behavior of the VLS fairing. In view of validating the numerical prediction of the fairing vibro-acoustic behavior, an important test campaign was done to measure the acoustic noise inside the fairing cavity. The fairing experimental model was submitted to 145 dB OASPL in a 1,200 m^3 acoustic reverberant chamber and microphones were positioned in its inner acoustic domain to measure the acoustic responses.

This work discusses the low frequency modeling procedures applied for the prediction of the acoustic environment of the VLS fairing using fluid-structure coupling interaction. The acoustic test procedures, performed in the acoustic reverberant chamber are also described. Acoustic Theoretical x Experimental comparison is done and some conclusions are drawn. Finally, our road map on noise control techniques to optimize the fairing design is briefly discussed.

MODEL DESCRIPTION

VLS Fairing Description

Figures 1a and 1b show the Brazilian VLS fairing structure (Loures, 1991). The fairing is the structural compartment where the payload or satellite is placed, during the launcher mission. This structure has as function to give adequate aerodynamic shape to the launcher as well as protecting the payload. The VLS fairing has hammerhead type geometry (cone-cylinder-cone), with a maximum nominal diameter of 1.2 m and a height of 3.5 m. This structure is built on aluminum shells, reinforced by circular and longitudinal beams. Its exterior surface is lined with cork material and no acoustic blankets are provided inside the fairing cavity. The referred fairing has a total weight of 150 kg, including the weight of the aluminum structure, the weight of some functional components such as the electric and pyrotechnic components of the ejection system, mechanisms as well as the exterior cork liner.

Figure 1a: VLS fairing structure

Figure 1b: VLS fairing structure

Modeling methodology

In view of analyzing the sound insulation properties of the fairing structure and predicting the operational fairing cavity noise levels, both the dynamic displacements of the fairing structure as well as the acoustic pressure fields at both the interior and the exterior side of the fairing should be considered. In this study, however, the fluid-structure coupling interaction between the structural displacements and the exterior acoustic pressure field is neglected. The exterior acoustic pressure is assumed to be a known external excitation for the vibro-acoustic system, consisting of the fairing structure and the internal acoustic cavity.

The FE and BE methods are the most appropriate numerical techniques for the (low-frequency) dynamic analysis of this type of vibro-acoustic systems. However in this work, only the FE method is described.

FE based models for vibro-acoustic problems are most commonly described in an Eulerian formulation, in which the fluid is described by a single scalar function, usually the acoustic pressure, while the structural components are described by a displacement vector. The resulting combined FE/FE model in the unknown structural displacements and acoustic pressures at the nodes of, respectively, the structural and the acoustic FE meshes are (Craggs, 1973),

$$\left(\begin{bmatrix} K_S & K_C \\ 0 & K_A \end{bmatrix} + j\omega.\begin{bmatrix} C_S & 0 \\ 0 & C_A \end{bmatrix} - \omega^2.\begin{bmatrix} M_S & 0 \\ -\rho K_C{}^T & M_A \end{bmatrix}\right).\begin{Bmatrix} w_i \\ p_i \end{Bmatrix} = \begin{Bmatrix} F_{Si} \\ F_{Ai} \end{Bmatrix} \tag{1}$$

In comparison with a purely structural or purely acoustic FE model, the coupled stiffness and mass matrices are no longer symmetrical due to the fact that the force loading of the fluid on the structure is proportional to the pressure, resulting in a cross-coupling term K_C in the coupled stiffness matrix, while the force loading of the structure on the fluid is proportional to the acceleration, resulting in a cross-coupling term $M_C = -\rho K_C^T$ in the coupled mass matrix.

In deterministic models, the dynamic variables within each element are expressed in terms of nodal shape functions, which are usually based on low-order (polynomial) functions that are no local solutions of the governing dynamic equations. Since these low-order shape functions can only represent a restricted spatial variation, a large number of elements is needed to accurately represent the oscillatory wave nature of the dynamic response. A generally accepted rule of thumb states that at least 10 (linear) elements per wavelength are required to get reasonable prediction accuracy. Since wavelengths decrease for increasing frequency, the FE model sizes and the subsequent computational efforts and memory requirements increase also with frequency. As a result, the use of FE models is practically restricted to low-frequency applications. In comparison with uncoupled structural or acoustic problems, this practical frequency threshold becomes significantly smaller for coupled vibro-acoustic problems, since a structural and an acoustic problem must be solved simultaneously. Moreover, as mentioned above, the matrices in a coupled deterministic model are no longer symmetrical, so that less efficient non-symmetrical solvers must be used. As a consequence, the computational load, involved with the use of coupled FE/FE models for real-life vibro-acoustic engineering problems, such as the considered fairing problem, becomes already prohibitively large at very low frequencies.

In order to obtain coupled vibro-acoustic response predictions within reasonable computational efforts, the dimensions of the coupled FE/FE problem of the Eq. (1) have to be substantially reduced. The most commonly applied technique for such a model reduction is the modal superposition technique, which expresses the unknowns of the considered system in terms of a modal base, resulting in a set of unknown modal participation factors, whose size is much smaller than the size of the original set of unknowns. The most appropriate choice for the base functions are the modes of the coupled vibro-acoustic system. Again, the determination of these coupled modes with a non-symmetric eigensolver is a very time consuming procedure, which makes it for most vibro-acoustic problems a practically impossible calculation. The most commonly used alternative is a modal expansion in terms of uncoupled structural and uncoupled acoustic modes, which result from computationally efficient symmetric eigenvalue problems. However, the fact that uncoupled acoustic modes have a zero displacement component, normal to the fluid-structure coupling interface, implies that a large number of high-order uncoupled acoustic modes is required to accurately represent the normal displacement continuity along the fluid-structure interface. Hence, the benefit of a computationally efficient construction of the modal base is significantly reduced by the smaller model size reduction, obtained with an uncoupled modal base.

In the present FE/FE study, a modal expansion in terms of uncoupled structural and uncoupled acoustic modal bases has been used. On the one hand, structural wavelengths are usually much smaller than acoustic wavelengths, so that the structural FE mesh of the fairing should be finer than the acoustic FE mesh of the inner cavity. On the other hand, due to the continuity of the normal structural and fluid displacements along the fluid-structure coupling interface, both meshes should have comparable mesh densities, at least in the region of the fluid-structure coupling interface. In view of these two considerations and in order to keep the computational efforts within reasonable limits, the following modeling methodology has been adopted: A fine FE mesh of the fairing is used for the construction of the uncoupled structural modal base. The resulting modes are then projected onto a coarse FE mesh of the fairing structure. For the acoustic cavity FE mesh, the same mesh density is used along the fluid-structure coupling interface as the coarse mesh of the fairing structure, while the mesh density has been slightly decreased towards the central axis of the cavity. The uncoupled modes, resulting from this acoustic FE mesh, together with the projected structural modal base of the fairing structure fine mesh and the structural coarse mesh, have been used in a coupled FE/FE model. Note that the coarse structural mesh contains only the shell area of the fairing structure, while all reinforcing beams are omitted, since it is assumed that these stiffeners have no significant effect on the fluid-structure coupling interaction, while their presence would increase the computational load of the modeling process.

The displacement continuity of the structural and acoustic meshes (same density in the fluid-structure interface) was considered to perform the link and calculate the coupled dynamic skin displacements and acoustic cavity pressure responses.

Structural model

The fairing body was divided in five surfaces, as shown in figure 2a. The surface areas are discretized into 4-noded quadrilateral shell elements, while 2-noded beam elements are used for the circumferential and the axial stiffeners (also shown in figure 2b). To account for the mass of the cork blanket on the exterior fairing surface, a distribution of concentrated mass elements are attached to the fairing nodes.

Shell surface 1 has a thickness of 3.0 mm and is made on conventional aluminum (E=72 GPa, ν=0.29, ρ=2,750 kg/m^3), while the other four surfaces are 0.8 mm thick and made on aluminum alloy (E=72 GPa, ν=0.29, ρ=7,000 kg/m^3). Stiffeners are also made on conventional aluminum. Note that surface 1 is not shown on figure 1b. This surface simulates the interface between the fairing and the 4th stage skirt of the VLS.

FE fine mesh contains at least 6, 8 and 10 elements per structural wavelength in the frequency range up to, respectively, 350 Hz, 220 Hz and 150 Hz. The FE coarse mesh, these frequency limits are 200 Hz, 120 Hz and 80 Hz. Even though this coarse mesh has not enough spatial representation, fluid-structure analysis is done up to 220 Hz (accurate analysis bandwidth up to 150 Hz), applying the projection modes option (structural modal data base calculated using the fine FE mesh) technique to the coarse FE structural mesh.

The bottom edge of the surface 1 is assumed as clamped and the top of the surface 5 is modeled as a rigid structure.

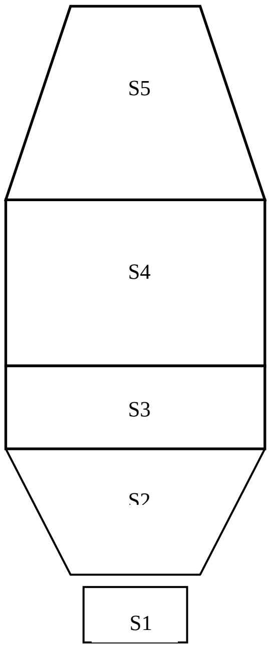

Figure 2a: surfaces of the fairing

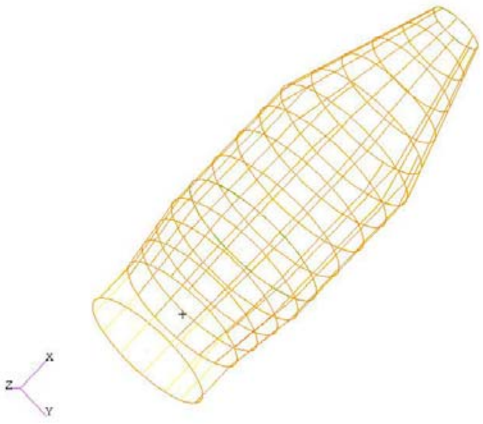

Figure 2b: location of the reinforcing beams

Table 1 describes the discrete properties of the fine and coarse structural meshes, used for the FE/FE calculations.

Table 1 - Discrete elements used for structural meshes

mesh		# shell el.	# beam el.	# mass el.	# nodes
FE fine	S1	4,000	240		
	S2	6,000	1,080		
	S3	2,000	360		
	S4	10,000	1,800		
	S5	12,000	1,672		
	total	34,000	5152	30,200	34,200
FE coarse	S1	2,250			
	S2	3,000			
	S3	750			
	S4	6,000			
	S5	7,500			
	total	19,500			19,650

A total of 174 structural modes in a frequency range up to 220 Hz have been identified using the fine mesh. Table 2 describes the main structural modes of the fairing body, calculated by the uncoupled modal analysis, in the range up to 150 Hz.

Table 2: Structural modes

mode	frequency (Hz)
first bending	38.637
first breathing	77.821
second breathing	92.694
first longitudinal	108.749
first torsion	125.772
second bending	150.735

As described in Table 2, the first structural bending mode of the fairing is identified at 38.6 Hz, while the second structural mode is at 150.7 Hz. Figures 3a and 3b show the referred structural modes.

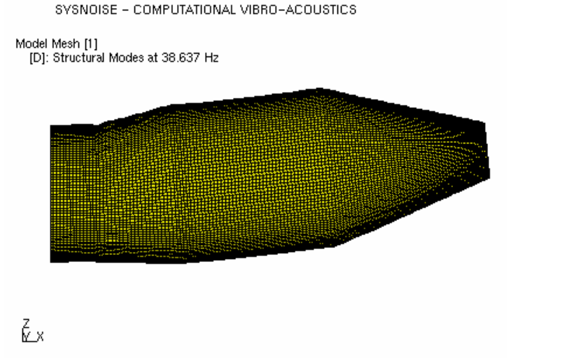

Figure 3a: First structural bending mode

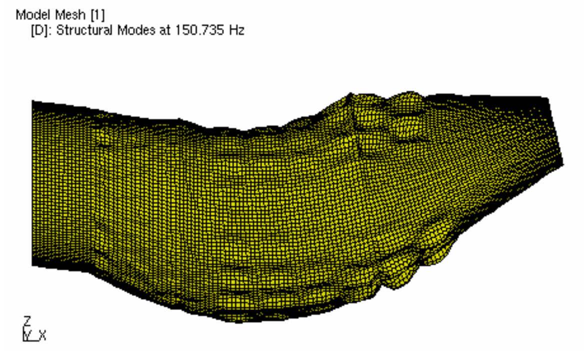

Figure 3b: Second structural bending mode

Acoustic model

The FE mesh for the fairing acoustic cavity consists of 119,577 nodes and 110,238 elements (106,050 8-noded hexahedral elements and 4,188 6-noded pentahedral elements). The cavity air has a mass density $\rho = 1.225$ kg/m^3 and a speed of sound c = 340 m/s. The bottom and top faces of the cavity are assumed to be acoustically closed. It is important to mention that the acoustic mesh generation took into account the cinematic continuity, on which at least on the fluid-structure interface the meshes must have identical density. A total of 80 acoustic modes in a frequency range up to 566 Hz have been identified using the acoustic mesh. As acoustic wavelengths are bigger than structural wavelengths and so, as uncoupled acoustic modes have a zero displacement normal to the fluid-structure interface, imply that a large number of high-order uncoupled acoustic modes is required to accurately represent the normal displacement continuity along the fluid-structure interface. That is why higher frequency range is used to describe the acoustic modal behavior of the fairing. Table 3 describes the acoustic modes in the frequency range up to 150 Hz.

Table 3: Acoustic modes

Mode	Frequency (Hz)
rigid body	0.000
first longitudinal	63.491
second longitudinal	112.129

Table 3 gives the natural frequencies of the first and second acoustic modes of the fairing cavity at 63.5 Hz and 112.1 Hz, respectively. Figures 4a and 4b show the referred acoustic modes.

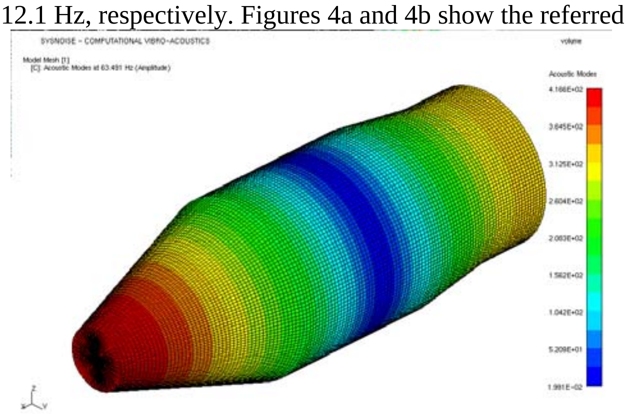

Figure 4a: First acoustic longitudinal mode

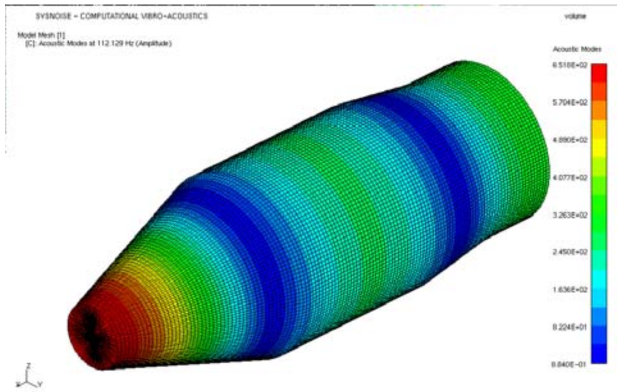

Figure 4b: Second acoustic longitudinal mode

Considering the described above, all the meshes and modal data basis needed to perform low-frequency calculations, using coupling fluid-structure FE technique are ready. Next steps are:

- excitation of the model, using an equivalent loading;

- generation of the coupled FE/FE model;

- FRA calculations for the both models.

Model excitation

In contrast with the aerodynamic noise during flight ascent, the nature of the lift-off acoustic pressure loading is close to a diffuse field excitation, having a (nearly) uniform pressure distribution (Coyette et al. 1997). A uniform exterior pressure loading is simulated by applying a normal point force on all nodes of the fairing shell elements. The force value is defined such that the total load is equivalent to a uniform pressure loading of 145 dB OASPL.

As such, uniform loading equivalent to the referred acoustic excitation is applied to the structural part of the model. Link of the acoustic and structural parts is done as well as the structural modal data base is projected to the coarse mesh. Finally, the FRA is computed from 5 to 220 Hz, in steps of 1 Hz for FE/FE model.

ANALYSIS RESULTS

In view of having a complete knowledge of the fairing dynamic vibro-acoustic behavior, the fairing structural skin as well as its inner acoustic domain responses should be presented. However, since this paper discusses only the comparisons of the acoustic results, the body structural displacements are not presented here. Below, the obtained results of the acoustic behavior applying vibro-acoustic low-frequency analysis technique predictions are presented.

FE/FE response calculations

A modal expansion in terms of 174 uncoupled structural and 80 uncoupled acoustic modes is used for the coupled response calculations. A modal damping of 1% is assigned to all structural modes. All calculations are performed with a frequency resolution of 1 Hz. Figure 5 plots the acoustic pressure spectra (up to 150 Hz) in two cavity positions of the fairing for the case of a uniform exterior pressure loading, using FEM/FEM coupling analysis. It can be seen that the low-frequency cavity acoustic pressure is dominated by the first longitudinal pressure mode around 63.5 Hz and the second longitudinal mode around 112.1 Hz (see figures 4a and 4b).

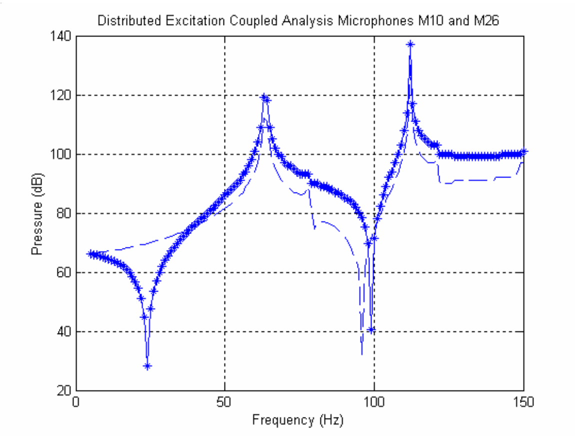

Figure 5: acoustic pressure spectra for uniform exterior pressure loading

Figure 6 presents the averaged acoustic spectra inside the fairing environment, up to 220 Hz.

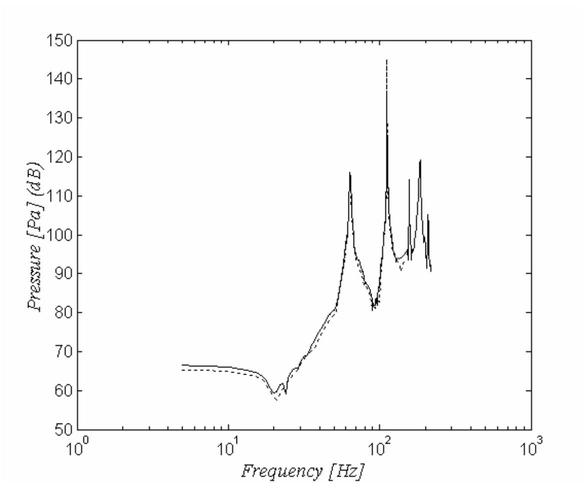

Figure 6: averaged acoustic spectra from 5 to 220 Hz

EXPERIMENTAL MODEL

Acoustic test

An extensive campaign of tests has been done, intending to manage the vibro-acoustic behavior of the VLS fairing, as well as to study and design noise reduction techniques to be applied to this space system. Among different tests, acoustic test was done at INPE-LIT Acoustic Reverberant Test Facility (ARTF).

The fairing structure was assembled as in-flight configuration and positioned inside the ARTF, which was excited with an acoustic diffuse field of 145 dB OASPL. Spectral distribution of this excitation, which simulates the generated acoustic pressure diffuse field during the VLS lift-off is shown in figure 7a. Eight control microphones were positioned along the acoustic reverberant chamber, as illustrated in figure 7b. These microphones feedback the acoustic chamber control system and the test specifications were assured. Four measurement microphones were located in different positions of the inner acoustic environment of the fairing (see Fig. 8). Data space averaging, taking into account the four measured SPL, was done and this averaged SPL is compared with the theoretical acoustic low-frequency responses, computed using virtual prototypes.

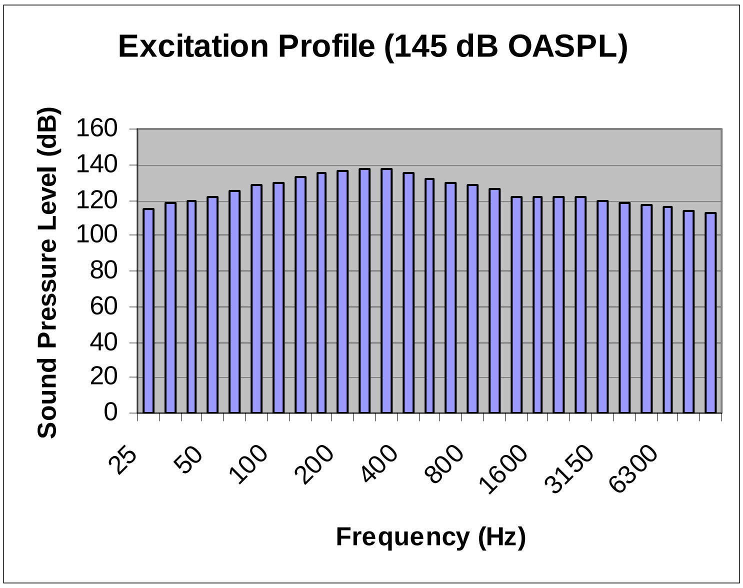

Figure 7a: VLS acoustic diffuse field for lift-off

Figure 7b: Fairing and control microphones inside the 1,200 m^3 ARTF

Figure 8 shows the measurement microphones, located inside the fairing cavity and data acquisition system, used to measure the SPL inside the fairing.

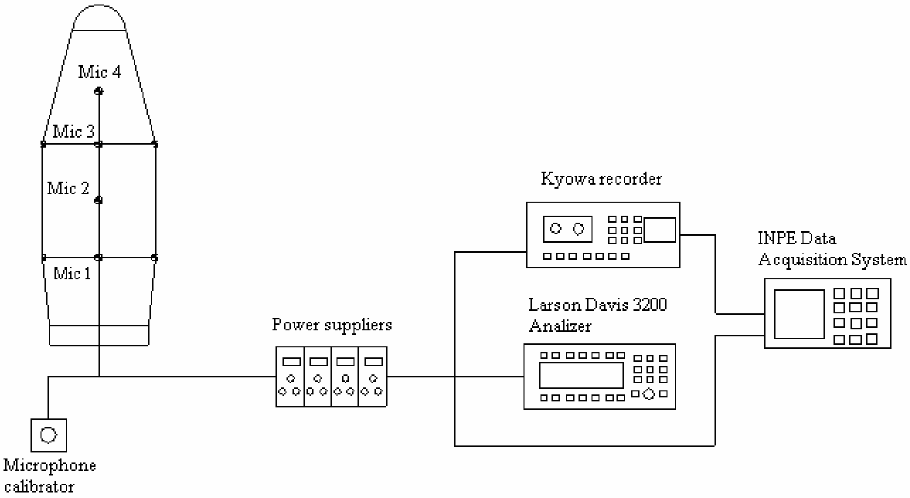

Figure 8: Data acquisition system and internal microphones location for the acoustic test

LOW-FREQUENCY THEORETICAL X EXPERIMENTAL COMPARISON

Deterministic computations yield approximated results, since low-frequency modeling techniques have been used (as described above). The calculated internal acoustic frequency response function shown in Fig. 6 may be transformed into a 1/3 octave band response to be compared with the experimental (measured) results. Figure 9 presents the 1/3 octave comparisons for the frequency bandwidths ranging from 31.5 up to 220 Hz.

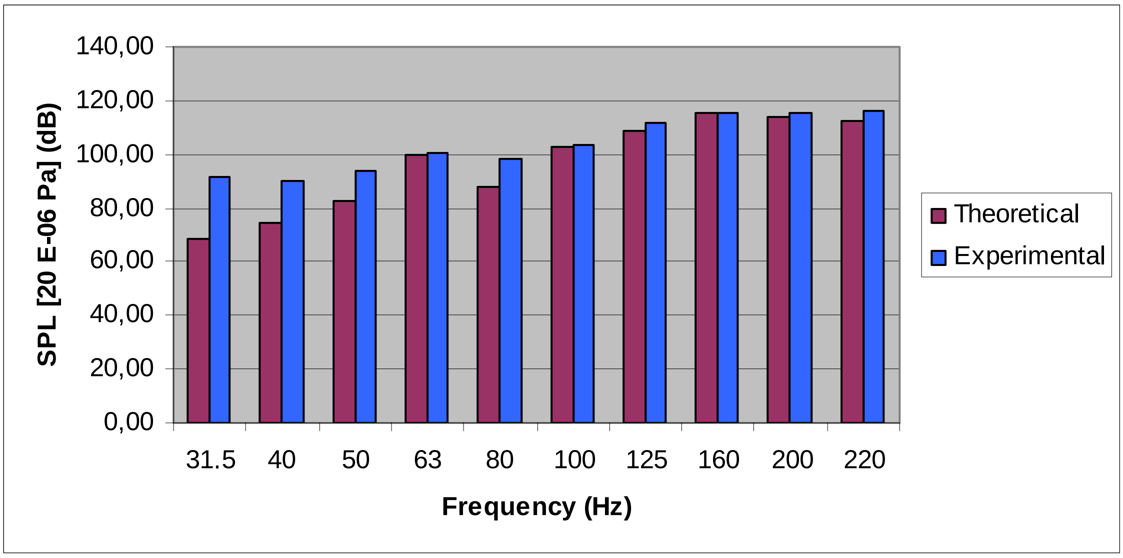

Figure 9: Low-frequency theoretical X experimental comparison

Note in Fig. 9 that acoustic experimental and calculated responses have good agreement, presenting more significant under predictions only on the low 1/3 octave bands 31.5, 40 and 50 Hz of 23 dB, 15 dB and 11 dB, respectively. However, in the regions where the cavity response is dominant (63 and 112 Hz), differences are not verified.

CONCLUSIONS

Vibro-acoustic virtual prototypes were used to preview the low-frequency acoustic behavior of the fairing cavity from 5 to 220 Hz, simulating the 145 dB OASPL diffuse pressure field, generated during the VLS lift-off.

A coupled deterministic technique, using FEM/FEM, was applied to the fairing problem. It was determined a reasonable low-frequency analysis band up to 150 Hz, considering accurate and efficient modeling techniques. The modal superposition technique was applied to obtain the FRA.

The VLS fairing was submitted to the estimated lift-off excitation 145 dB OASPL in an acoustic reverberant test at INPE-LIT ARTF. Internal acoustic pressure levels were measured by microphones distributed along the fairing cavity.

The experimental and calculated results were compared with good agreement, excepting for the frequencies below 50 Hz, where under predicted values were verified.

Our road map for the future:

High- and medium-frequency analysis may be done to complete the frequency bandwidth of interest (up to 8,000 Hz). For the high frequency analysis, coupled Statistical Energy Analysis (SEA) technique may be used. For the medium-frequency (or twilight zone) a redundancy analysis may be considered, using results from deterministic and statistical prediction techniques.

Virtual prototypes and experimental tests may be done to develop and design optimal acoustic noise control devices, using blanket absorption materials as well as Helmholtz Resonators.

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REFERENCES

Caneilles, J.L., Departement Analyse de Mission Arianespace – Evry (France), Environnement Mecanique en Vol sur les Charges Utiles Mises en Orbite par les Lanceurs Ariane, 1998.

Coyette, J-P; LECOMTE, C.; MEERBERGEN, K.. *Treatment of random excitations using SYSNOISE Rev. 5.3.1 – Documentation Theoretical Manual*, User's Manual and Validation Manual, LMS Numerical Technologies NV. 1997.

Loures, L.E.V.C, General Specification of the VLS Fairing, Document N° 590-600000/B1001, October 1989, Rev. August 1991.

A. Craggs, "An acoustic finite element approach for studying boundary flexibility and sound transmission between irregular enclosures", *Journal of Sound and Vibration* **30** (3), 343-357 (1973).

T.T. Do, "Vibro-acoustic modeling study of the Delta II 10-foot composite fairing", *Journal of the IEST*, november/december 1999, 26-33.

H. Defosse, M.A. Hamdi, "Vibro-acoustic study of Ariane V launcher during lift-off", *Proc. of Inter-Noise 2000*, vol 1, 9-14 (Nice, 2000).

B. Troclet, "Analysis of vibroacoustic response of launchers in the low and high frequency range", *Proc. of NOVEM 2000* (Lyon, 2000).

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