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STATISTICAL ENERGY ANALYSIS TO STUDY THE VIBROACOUSTIC OF A LAUNCHER VEHICLE: AN APPLICATION TO DIFFERENT FLIGHT CONDITIONS

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Abstract

Aim of this paper is the study of the sound pressure and the vibration levels of the upper part (fairing, AVUM, IS-3AVUM) of the VEGA launch vehicle during the liftoff and the transonic flight by the Statistical Energy Analysis (SEA).
The acoustic loads applied to the launch vehicle upon the liftoff and the transonic flight are broad band and random. They may be dangerous for the payload and the equipments.
A fast and cheap prediction of the sound pressure and of the vibration levels may be obtained by numerical procedures rather than performing measurements on real prototypes.
The classical numerical techniques (FEM, BEM) fail to solve high-frequency dynamic problems and a statistical approach seems more appropriate.
SEA is, at present, the most useful method for solving this kind of vibroacoustic problems, by providing information on the stored mechanical energy and on the dissipated mechanical power between subsystems.

INTRODUCTION

The acoustic loads applied to the launch vehicle upon the liftoff and the transonic flight, are broad band and random loads which may be dangerous for the payload and the equipments.
A fast and cheap prediction may be obtained by numerical procedures rather than performing measurements on real prototypes.
The classical numerical techniques (FEM, BEM) allow to predict the response of the mechanical systems at low frequency ranges.
A high frequency problem happens when the ratio between the characteristic wave length and the characteristic dimension of the system is very little respect to the unity.

It happens, for example, when a broad-band load forces a large and lightweight structure.
The classical numerical techniques fail to solve high-frequency dynamic problems, because the computational burden grows excessively, but also because the sensitivity of the numerical algorithms to uncertainties in the modal parameters increases with frequency so that the predicted response becomes meaningless. In this case a statistical approach seems more appropriate.
The Statistical Energy Analysis (SEA) is, at present, the most useful method for solving this kind of vibroacoustic problems, by providing information on the stored mechanical energy and on the dissipated mechanical power between modal subsystems [1-5]. In this paper the study of the sound pressure levels and the vibration levels at some points of the upper part (fairing, AVUM, IS-3AVUM) of the VEGA launch vehicle during the liftoff and the transonic flight is presented. The loads acting on the launcher are obtained by experimental measurements, SEA is the theoretical tool used to predict these levels by AutoSEA2 (ESI Group) software.

SEA THEORETICAL DESCRIPTION

Statistical Energy Analysis is a methodology to solve steady state vibroacoustic problems at high frequencies. The solution is obtained by dividing the studied mechanical system into subsystems representing groups of similar modes, where the modes of one group have a similar energy. A group of modes is an energy reservoir. Energy flows into each subsystems from external sources and it is balanced by the dissipated power and the power transferred to the other modal groups.
For a two degrees of freedom system (2DOF), the joint is conservative and massless, it is proved that the energy flow, averaged on frequency bands, between the two resonators, is proportional to the difference of total energies stored in the two oscillators. The governing equations of the method is valid for two groups of modes only under particular hypotheses: a list of these hypotheses is presented:

- all the modes of a subsystem must be similar (i.e. they must have almost the same energy, damping, coupling with the other subsystems and they must be almost excited by the same input power),
- the subsystems coupling must be conservative,
- the eigenfrequencies must be uniformly probable in the frequency range,
- the force exciting the subsystems must be random and not-correlated,
- the interactions between the subsystems must be weak.

The SEA equations of a system divided into M subsystems, can be written as follows:

$$P_{i,inj} = \omega \eta_i E_i + \omega \sum_{j=1, j\neq i}^{M} \left(\eta_{ij} E_i - \eta_{ji} E_j \right) \tag{1}$$

where i is the index of the subsystem, η_i and η_{ij} are the internal loss factors (ILF) and the coupling loss factors (CLF), respectively, and $P_{i,inj}$ is the power injected into subsystem i. Equations (1) represents the energy balance between the M subsystems:

$P_{i,d} = \omega \eta_i E_i$ is the power dissipated in subsystem *i* and $P_{ij} = \omega\left(\eta_{ij} E_i - \eta_{ji} E_j\right)$ is the power transmitted from subsystem *i* to subsystem *j* (figure 1).
The solution of the linear system (1) provides the energy stored in each subsystem.

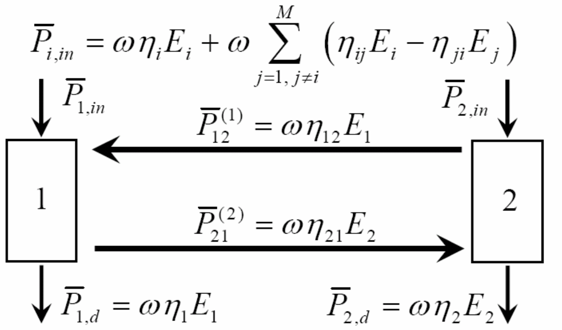

Figure 1 – Diagram of the energy flow balance of SEA

Three important parameters are used in the SEA method:

- The modal density of a system, $n(\omega) = N(\omega)/\Delta\omega$, is the number of subsystem modes in the unit frequency band.
- The modal overlap factor (MOF) is defined as follows:

$$m(\omega) = \omega\, n(\omega)\, \eta(\omega) \tag{2}$$

  η is the loss factor.
- The injected power is the mean power entering into a subsystem from an external source. The mean power injected in a point can be expressed by the following general expression:

$$P_{inj} = \frac{1}{2}\mathrm{Re}\left\{F V^*\right\} = \frac{1}{2}\left|F\right|^2 \mathrm{Re}\left\{\mathcal{M}^*\right\} \tag{3}$$

*F, V** and $\mathcal{M}$* are, respectively, the force, the complex conjugate velocity and the mobility at the driving point. Similar equation are used when acoustic or fluid-dynamic loads are considered.

MODEL DESCRIPTION

The SEA model of the studied launcher's part is built by characterizing a set of SEA subsystems. The choice of this set of subsystems depends on the material properties, the geometry of the physical prototype and the SEA requirements.
Figure 2 shows the whole set of subsystems. Figure 3 shows the line junctions between the structural subsystems and figure 4 shows the area junctions between the structural surface and the acoustic cavities. Table 1 summarizes the properties of each structural subsystem. Non structural mass are considered to describe the presence of the electronic components, stiffener bars and the liquid and gas tank. The considered ILF of the structural subsystems is 0.02, while the ILF of the acoustic cavity is 0.
AutoSEA2 software calculates the CLFs between the SEA subsystems by considering the shape and the materials.

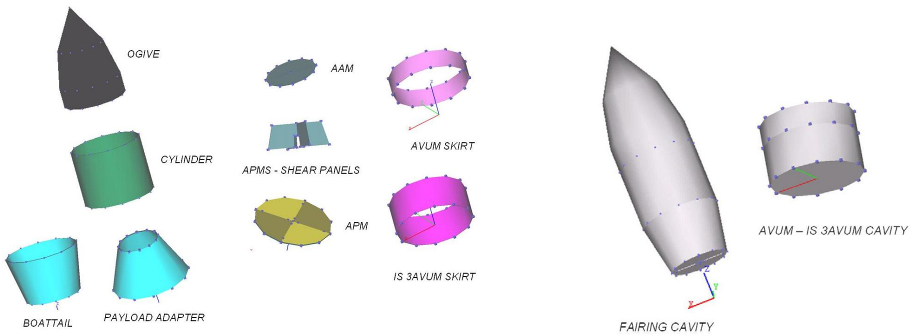

Figure 2 – SEA structural and acoustic subsystems

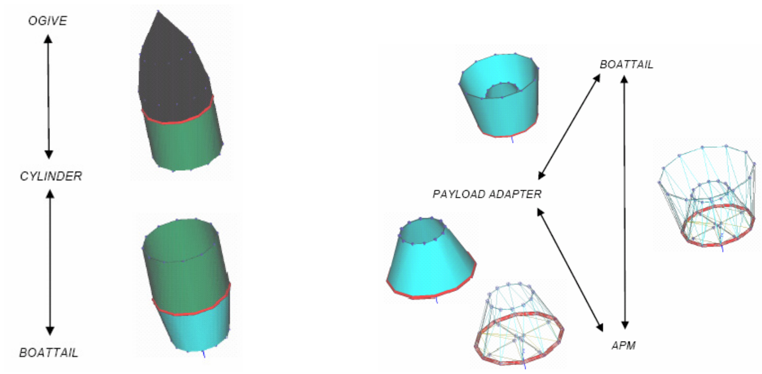

Figure 3 – SEA structural subsystems line junctions

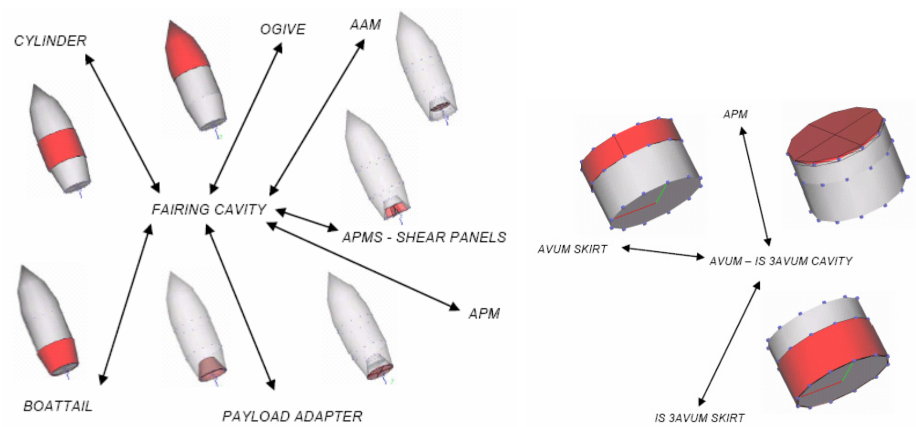

Figure 4 – SEA acoustic-structural subsystems area junctions

The classical relationship to calculate the CLFs between two structural subsystems *i* and *j* (line junctions) is:

$$\eta_{ij} = \frac{2 c_{B_i} L \tau_{ij}}{\pi \omega S_i}$$

where τ_{ij} is the wave transmission coefficient from the subsystem *i* to the subsystem *j*. For the junctions between structure and cavities (area junctions) the equation is:

$$\eta_{SC} = \frac{\rho_0 \, c \, \sigma}{\omega \, \rho_S}$$

where σ is the radiation efficiency.

Table 1 – Material properties of SEA structural subsystems

SUBSYSTEM	PROPERTY
OGIVES	thermal protection-carbon fibers-honeycomb-carbon fibers
CYLINDERS	thermal protection-carbon fibers-honeycomb-carbon fibers
BOATTAILS	thermal protection-carbon fibers-honeycomb-carbon fibers
PAYLOD ADAPTER	carbon fibers-honeycomb-carbon fibers
AAM	aluminum-honeycomb-aluminum
SHEAR PANEL	aluminum-honeycomb-aluminum
APM	aluminum-honeycomb-aluminum
AVUM SKIRT	ribbed shell: aluminum

On the line junctions APM-AVUM and AVUM-IS3AVUM is imposed a non structural mass by considering the gas tanks.

NUMERICAL RESULTS

SEA model assesment

Let us remember that the SEA model is valid as long as the SEA hypothesis are respected. About the limits on the injected power, the diffusivity of the considered loads guarantees the respect of the hypothesis on the force. The weak coupling is disappointed: in fact, often, the subsystems are two different parts of the same structures. The hypotheses which concerns the subsystems quality can be correlated with the value assumed by the MOF. Figure 5 shows the MOF of the considered subsystems: unless the acoustic cavities and the AVUM skirts, MOF>>1 happens after 1000Hz.

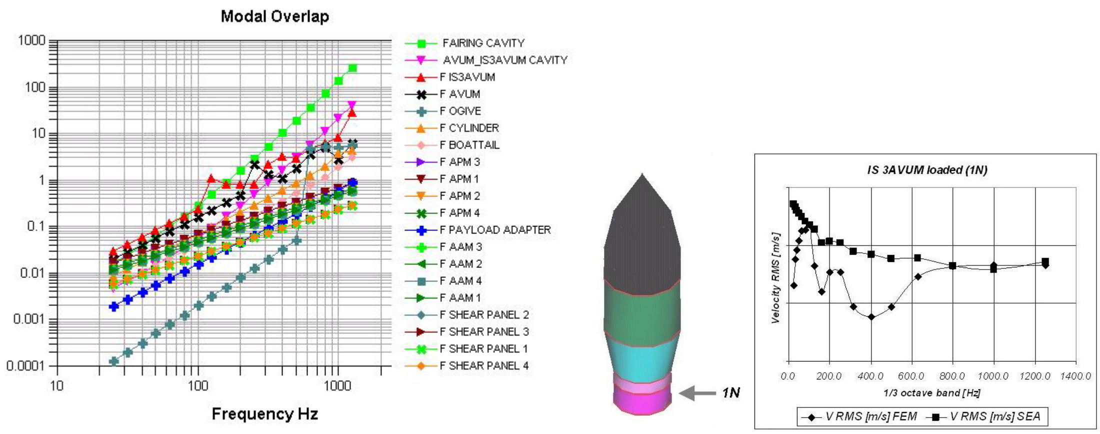

Figure 5 – modal overlap factor of the subsystems and FEM-SEA comparison

A numerical test is developed to study the SEA result reliability by the comparison of this result with an equivalent FEM solution. A finite element model of the studied system is loaded by a force with 1N amplitude and a flat spectrum in the interval 50-1250Hz. The force is imposed on the IS3AVUM. The average of the FEM results over all the nodes of the IS3AVUM is compared with the results given by AutoSEA2 for an equivalent load case. Figure 5 shows this comparison: particularly, a good agreement is reached only after 800Hz (all the results are presented in third octave band). This results seems to agree with the MOF behaviour.

SEA results

Two different load cases are considered in this paper. The first describes the acoustic environment upon the liftoff and the second during the transonic flight. The sound pressure level forcing the launcher is calculated by the measurements of a set of microphones, positioned around two scaled models, in two different tests.

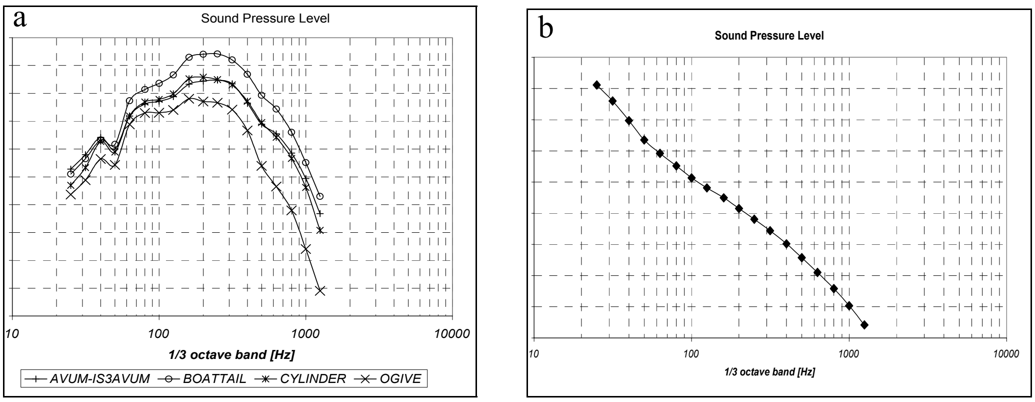

Figure 6 – Sound pressure level of the power injected: a) lift-off, b) transonic flight

During the liftoff the acoustic field loading the launch vehicle is supposed diffuse. Since the energy of a diffuse field is uniform and isotropic the sound pressure level acting over each SEA subsystem is calculated by averaging the energy measured by the microphones set in the neighborhood of the subsystems:

$$E_{EDF} = \sum_{i=1}^{4} E_i / 4 \quad \leftrightarrow \quad < p^2 >_{EDF} = \sum_{i=1}^{4} < p^2 >_i / 4 \tag{4}$$

In order to describe the acoustic field during the transonic flight an equivalent diffuse field is calculated by supposing that the energy of this field is equal to the mean energy measured during the transonic test.
A set of microphones measured the acoustic field close to a scaled model of the VEGA during a transonic test in a wind tunnel.
The acoustic field is measured in many different conditions: different orientations of the model respect to the fluid flow direction and different Mach numbers of the fluid flow.

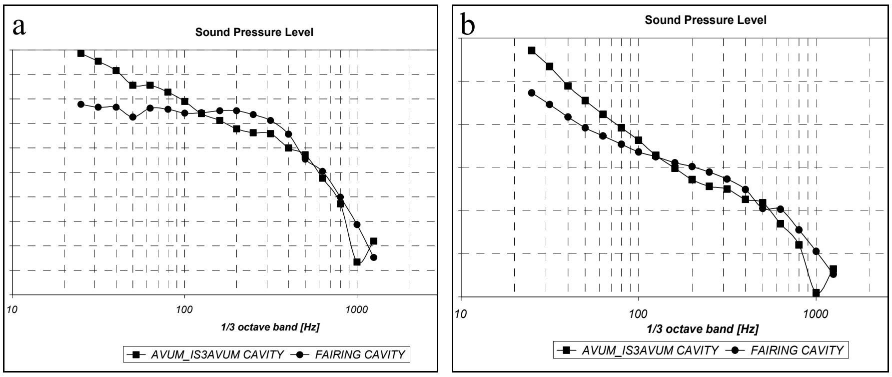

Figure 7 – Sound pressure level calculated by AutoSEA2: a) lift-off, b) transonic flight

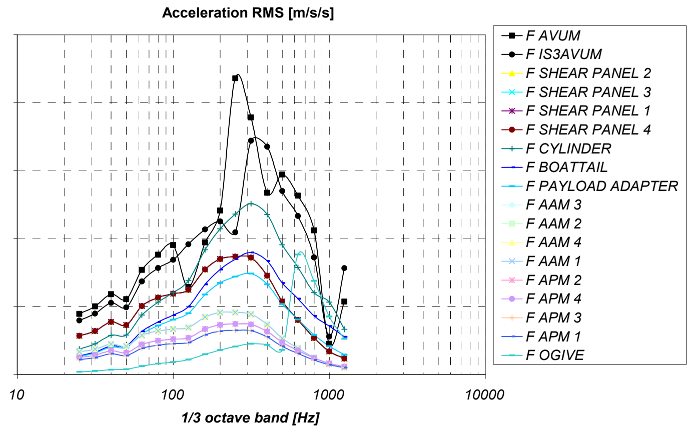

Figure 8 – Accelerations calculated by AutoSEA2 during the lift-off

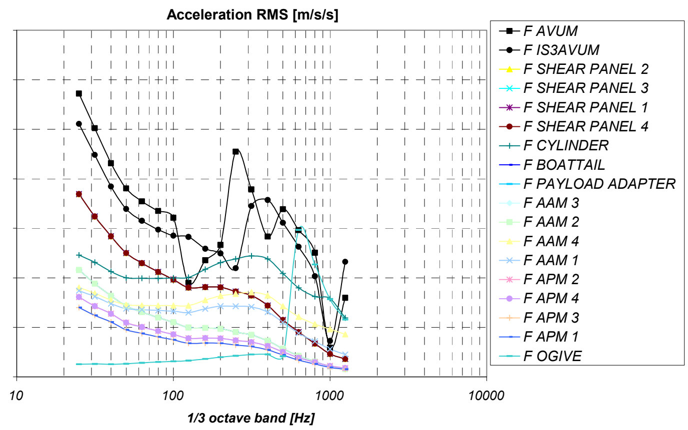

Figure 9 – Accelerations calculated by AutoSEA2 during the transonic flight

Since a diffuse acoustic field is defined as a wave field where the energy is uniform and isotropic, the equivalent diffuse field is calculated by averaging the measurements of the whole microphones during the different conditions:

$$E_{EDF} = \frac{\sum_{i=1}^{N\,Mach \times N\,angle \times N\,mic} E_i}{N\,Mach \times N\,angle \times N\,mic} \leftrightarrow \; < p^2 >_{EDF} = \frac{\sum_{i=1}^{N\,Mach \times N\,angle \times N\,mic} < p^2 >_i}{N\,Mach \times N\,angle \times N\,mic} \tag{5}$$

Therefore, a space average is considered together with all the flight conditions studied during the wind tunnel experiments. The results are shown in figure 6.
Figure 7-9 show the SEA results obtained by the software AutoSEA2. The sound pressure level into the acoustic cavities are shown in figure 7, the rms accelerations of the structural subsystems are shown in figure 8 and 9.

CONCLUSIONS

In this paper the vibro-acoustic field of a small launcher upper part is been studied by the SEA. This methodology has permitted to investigate the response of the system under particular kind of force (acoustic diffuse field) which can not be easily modeled by classical solution methods (FEM, BEM, …).
A simple test case is developed to prove the ability of SEA technique to give a good solution. The comparison between a FEM and a SEA solution shows that after around 800Hz the responses have a good agreement.
The spectrum of the load cases used are calculated by measurements obtained by two experiments on scaled models. The sound pressure level into the acoustic cavities and the rms acceleration of the structural subsystems are calculated by AutoSEA2 software and presented. A comparison with measurements or different numerical calculation is not available because of the sort of investigated problem.

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