Acoustic vibration environment prediction for Amazonia 1 satellite using SEA method

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Abstract. The dynamic vibration environment applied to a satellite during its launch is predominantly acoustic. Therefore, the prediction of acoustic levels and the satellite and its onboard equipment vibroacoustic responses are mandatory in all project phases. In this sense, one of the most important tests to qualify a satellite for flight is the acoustic vibration test, whose levels are usually specified by launcher vehicle manufacturer. In the first phases of a project, several viability studies may be running simultaneously and important changes in structure design and even in concept can occur. Thus, a method that can produce analyses with low computational cost in a short period of time and with a reasonable accuracy would simplify the development process and reduce its costs. Usually, the acoustic analysis is performed by using the finite element method (FEM) for lower frequencies and statistical energy analysis (SEA) for higher frequencies. The present work describes the use of SEA method to perform the vibroacoustic analysis of Amazonia 1 satellite, a satellite developed by Instituto Nacional de Pesquisas Espaciais (INPE). The numerical results are compared to acoustic tests results performed at INPE on the satellite structural qualification model. Amazonia 1 is a remote sensing satellite designed by INPE to provide images on green, red, blue and near infrared (NIR) bands with resolution better than 70m with the goal to monitor forests, vegetation, agriculture, coasts, and others areas.

Keywords: acoustic vibration, satellite, SEA.

1 Introduction

The process involved in injecting a satellite on its nominal orbit is very energetic due to the need to overcome gravity, Earth rotation and to achieve enough kinetic energy to stay on orbit and not return to Earth, in this sense, a lot of vibrations are generated and transmitted to the satellite, mainly acoustically related vibrations, which are very important for satellite structure and onboard equipment. Satellite structures are usually designed based on finite element method (FEM), as most of nowadays structures due to the robust features related to the method and the high development stage of commercial codes. However, as the frequency increases, the number of elements needs to grow very fast in order to keep the analysis' accuracy and the size on computational matrix become unfeasible even for the modern computers, in the sense that the meshing, modeling and analysis become a non-optimal process. The most used method for high frequencies in aerospace structures is the statistical energy analysis (SEA), which is based on energy balance among the elements of a system, as described by Le Bot [1]. SEA approximations considers that inside a frequency bandwidth all energy of an element of the system is equally divided by its natural vibration modes, then the energy exchange between vibration modes of different elements are also equally distributed. The SEA method requires the knowledge of the loss factors of the system, and for aerospace structures Wijker [2] presents a very elucidating book with plenty of references and data regarding dissipation and coupling loss factors as well as modal densities.

This work presents a direct application of SEA method to the panels of Amazonia 1 satellite and compares the results with those obtained from its structural-acoustic qualification tests performed by INPE. Each panel was analyzed separately and no interaction between a panel and other element of the satellite was taken into account. Also, the influence of the air inside the satellite was neglected. The panel with simply supported boundary condition was excited by the acoustic diffuse field pressure applied on its outer face, as illustrated on fig. (1). The Amazonia 1 structural panels are sandwich panels composed of aluminum face sheets and honeycomb, the assembled equipment was considered by adding its masses to the panel density calculation. This simplified model, that do not take into account the internal interactions among all satellite elements is reasonable due to the following facts: the satellite structure design is box chapped with the panels looking direct to the acoustic chamber; the coupling between air and structure is weak; and, the test was performed in an acoustic chamber with closed loop pressure control.

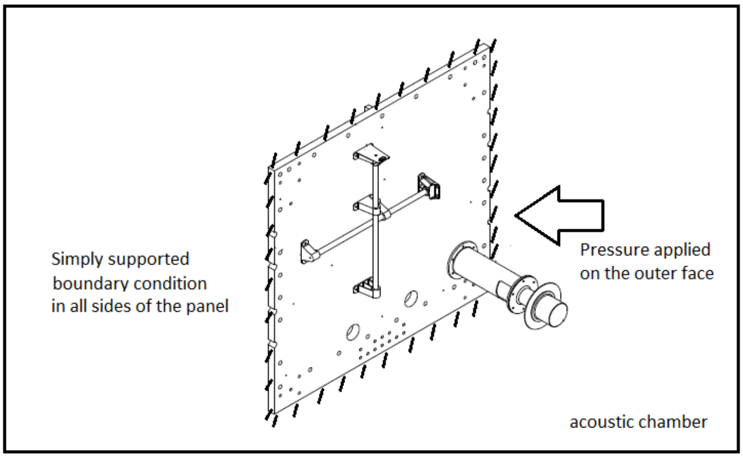

Figure 1. SEA model.

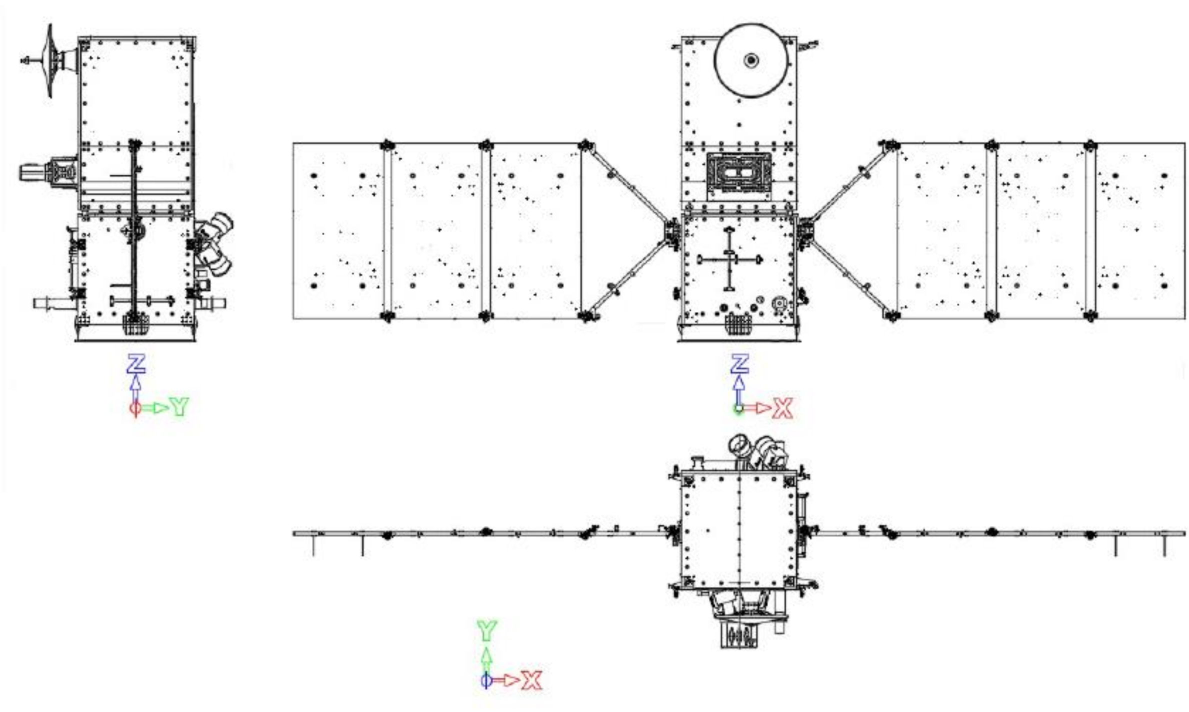

Figure 2. Amazonia 1 satellite overview in orbit configuration

Amazonia 1 is a satellite with 640 kg, around 2.5 meters high, 2 solar panels array of around 3.5 meters length, and divided in two box shaped modules. This satellite was launched on February 28th of 2021 with the mission to provide images from Brazil in 3 visible bands and 1 NIR band with a resolution of ~65 meters at nadir and a swath of ~850 km. Figure (2) presents an overview of the satellite. Amazonia 1 mission is the first satellite of the Multi Mission Platform (MMP) program from INPE, which is a program to develop a modular platform capable to hold a bunch of different missions in a modular concept, the MMP is the bottom box exhibit on fig. (2) and different payloads can be assembled over it, as the Amazonia 1.

2 SEA Model

SEA method was applied on each panel of the satellite independently, as explained on fig. (1), each SEA model used the acoustic energy of the chamber $[E_a]$, panel energy $[E_p]$ from which the acceleration responses were derived, structure to acoustic coupled loss factor $\eta_{s,a}$, structure loss factor η_s, panel modal density n_p and acoustic chamber modal density n_a. The acoustic energy was derived from pressure profile specification applied to the chamber and read on chamber microphones, the energy balance of the panel consists of the energy received from the chamber through its coupling loss factor and the energy dissipated by its structure loss factor, resulting in the following equation [2]:

$$\eta_s + \eta_{s,a} \quad \omega\left[E_p\right] = \omega \frac{n_s \eta_{s,a}}{n_a}\left[E_a\right], \tag{1}$$

where, ω is the frequency in rad/s. The acoustic energy is given by the relation of acoustic chamber pressure, volume, air density and speed of sound, while panel energy can be evaluated based on total mass and panel velocity (or acceleration/ω):

$$\left[E_p\right] = \frac{M}{\omega^2}\left[a^2\right], \tag{2}$$

$$\left[E_a\right] = \frac{V}{\rho c^2}\left[p^2\right], \tag{3}$$

where, M is the total mass of the panel including assembled equipment, $[a^2]$ is the mean square of acceleration which is given by the PSD acceleration on the band multiplied by the bandwidth Δf, ω is the center frequency, V, ρ and c are respectively the volume, air density and air speed of sound and $[p^2]$ is the mean square of pressure, calculated in the same way as the acceleration.

Combining equations (1), (2) and (3), the relation between panel acceleration PSD to chamber acoustic pressure PSD is given by:

$$\left[\frac{a^2}{\Delta f}\right] = \frac{V\omega^2}{M\rho c^2}\left(\frac{n_s}{n_a}\right)\left(\frac{\eta_{s,a}}{\eta_s + \eta_{s,a}}\right)\left[\frac{p^2}{\Delta f}\right]. \tag{4}$$

Conlon and Hambric [3] present the panel modal density and loss factor for sandwich panels, the acoustic model density by Kuttruff [4], critical frequency by Wijker [2], while the coupling loss factor structure acoustic is given by Crocker and Price [5]:

$$\eta_s = \omega \frac{S_s \quad h + t}{{c_{b,eff}}^2}\left[1 - \frac{1 / {c_b}^3 + 1 / {c_s}^3 \quad ^{-4/3}}{2 c_{b,eff} {c_b}^3}\right], \tag{5}$$

$$c_b = \sqrt{\omega} \quad D\big/\rho_s \quad ^{1/4}, \;\; c_s = \quad Gh\big/\rho_s \quad ^{1/2}, \;\; c_{b,eff} = \quad 1 / {c_b}^3 + 1 / {c_s}^3 \quad ^{-1/3}, \tag{6}$$

c_b, c_s and $c_{b,eff}$ are respectively the bending, shear and effective bending wave speeds, h and t are the thickness of the sandwich panel and face sheet, S_s, ρ_s, D and G are the panel area, density (including equipment mass), bending stiffness and shear modulus,

$$\eta_s = \begin{cases} 0.05, & f \leq 500Hz \\ 0.05\left(\dfrac{500}{f}\right), & f > 500Hz \end{cases} \tag{7}$$

$$n_a = \frac{4\pi f^2}{c^3}V + \frac{\pi f}{2c^2}S_a + \frac{1}{8c}P_a, \tag{8}$$

S_a and P_a are acoustic chamber total area and perimeter,

$$f_c = \frac{c^2}{2\pi}\frac{\sqrt{\rho_s / D}}{\sqrt{1 - c^2\rho_s / S_s}}, \tag{9}$$

$$\eta_{s,a} = \frac{\rho c \sigma}{\omega \rho_s}, \tag{10}$$

f_c the critical frequency, which is the frequency of the panel in which the wave speed is equal to the sound speed on the air, $\alpha=(f/f_c)^{0.5}$, and σ is the radiation efficiency, which can be approximated by

$$\sigma = \begin{cases} \dfrac{2\lambda_c\lambda_a}{S}\dfrac{f}{f_c}g_1\left(\dfrac{f}{f_c}\right) + \dfrac{P_p\lambda_c}{S}g_2\left(\dfrac{f}{f_c}\right) & f < f_c \quad ka, kb > 2 \\ g_1\left(\dfrac{f}{f_c}\right) = \begin{cases} \dfrac{4}{\pi^4}\dfrac{1-2\alpha^2}{\alpha\sqrt{1-\alpha^2}} & f < \dfrac{f_c}{2} \\ 0 & f > \dfrac{f_c}{2} \end{cases} & \\ g_2\left(\dfrac{f}{f_c}\right) = \dfrac{1-\alpha^2 \ \ln\left[\ 1+\alpha / 1-\alpha \ \right] + 2\alpha}{2\pi^{\ 2}\sqrt{1-\alpha^2}^{\ 3}} & \\ \dfrac{4P_p\lambda_c}{\pi^4 S}\left(\dfrac{f}{f_c}\right)^{0.5} & f < f_c, \quad ka, kb < 2 \\ \left(1 - \dfrac{f_c}{f}\right)^{-0.5} & f > f_c \end{cases} \tag{11}$$

3 Amazonia 1 Acoustic Qualification Test

Amazonia 1 satellite was qualified for acoustic vibration by testing a qualification model composed of the satellite structure subsystem and equipment dummies which respected the real mass, footprint and center of gravity position, fig. (3) shows the structural model inside the acoustic chamber. The tests were held at INPE's acoustic chamber on 2012 and followed an acoustic Sound Pressure Level (SPL) specification described on tab. (1), according to the Multi Mission Platform program.

The SPL specification was composed of an envelope of different commercial launcher vehicles requirements in order to provide flexibility to the MMP and qualify it to be launched by almost any available rockets in the market. The tests successfully proved the adequate dynamic responses of the platform and payload, which were capable to provide acceleration profiles to the onboard equipment and instruments compatible to the ones common used during equipment level qualification and acceptance campaigns.

The pressure profile during the tests was ensured by the chamber's control system, which used the four mics exhibited on fig. (3) to control the pressures applied directly to satellite for the total frequency band. The satellite model was fulfilled with accelerometers to record acceleration profiles from each equipment and structure control points. The satellite was isolated from the ground by the use of a trolley with pneumatics cushions and was positioned inside the chamber in an oblique angle to the walls to avoid any coupling. The acceleration results from three different panels showed on fig. (4) were used as they are representative for the entire satellite.

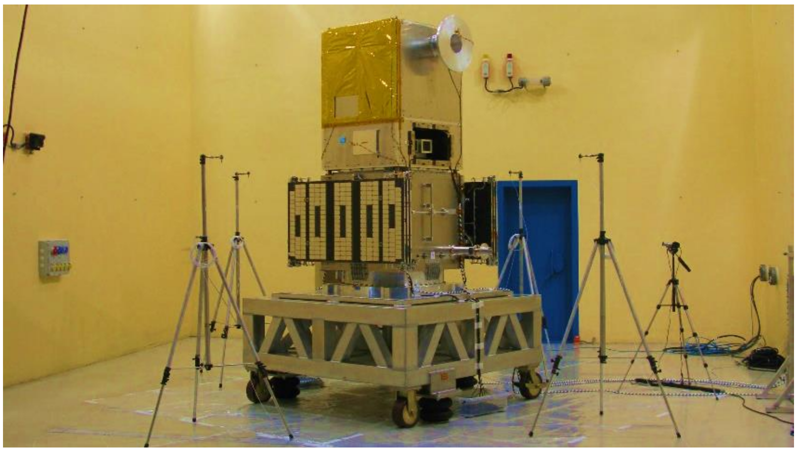

Figure 3. Amazonia 1 satellite structural model inside INPE’s acoustic chamber

Table 1. Amazonia 1 acoustic specification and recorded data. Specified OASPL 147.3dB, registered 147.4dB

Frequency Band (Hz)	Specification profile (dB)	Registered SPL (dB)	Frequency Band (Hz)	Specification profile (dB)	Registered SPL (dB)
20	127 +/-3	125	500	135 +/-3	135
25	128 +/-3	132.5	630	132 +/-3	132.5
31.5	128 +/-3	127.5	800	132 +/-3	132.5
40	129 +/-3	131	1000	132 +/-3	134
50	131 +/-3	131	1250	129 +/-3	130
63	135 +/-3	135	1600	129 +/-3	127
80	135 +/-3	136	2000	129 +/-5	125
100	135 +/-3	137	2500	124+/-5	122.5
125	138 +/-3	138	3150	124+/-5	120
160	138 +/-3	140	4000	118+/-5	118
200	138 +/-3	138	5000	118+/-5	118
250	137 +/-3	137.5	6300	118+/-5	116
315	135 +/-3	135	8000	118+/-5	116
400	135 +/-3	135	10000	118+/-5	115

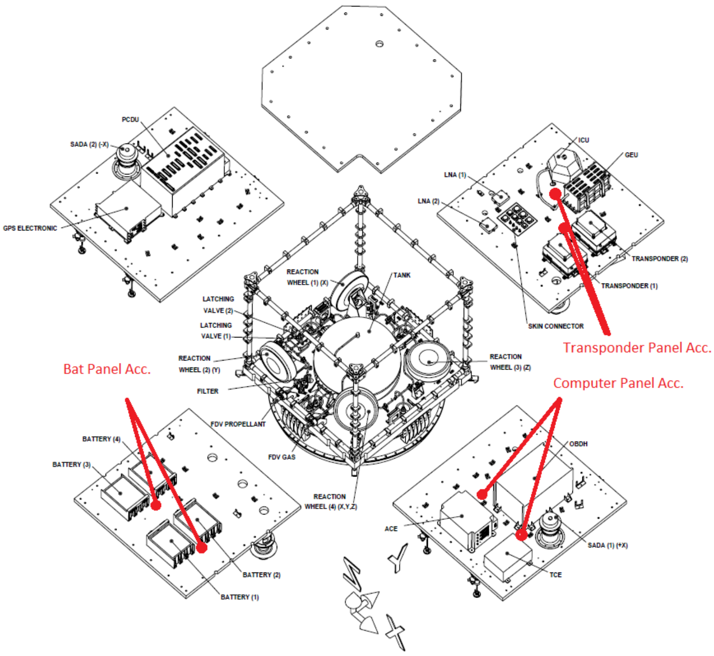

Figure 4. Used accelerometers inside Amazonia 1 qualification model.

4 Results

The results of the SEA method applied as described on this work are presented on figures (5), (6) and (7). Even for the presented model, which is very simple and of easy implementation, the results in average were very good from 100Hz (which is a very low frequency for SEA calculations) to around 10,000Hz. The model missed only the behavior of the first three natural modes of the panels, in frequencies around 100, 200 and 350Hz, and of frequencies below 100Hz. The results can be explained by the high modal density of the acoustic chamber, for example, in 100Hz the modal density is in the order of 7modes/Hz (chamber volume = 1764 m^3), which fulfilled SEA hypothesis of energy uniformity distribution among modes and energy fluxes based on modal interaction, closed loop pressure control of the chamber, that ensures the specified pressures on satellite and avoids the natural frequencies of the chamber, and that below panel first frequency, the panel behaves as a mass/spring system with force applied, that is governed by its stiffness and, therefore, responds very little at this band, which is a behavior not accounted on SEA model, in the same way that panels natural frequencies are not accounted either. The Grms found was close to the ones obtained with maximum error of 35% for battery panel and minimum error of 2% for transponder panel.

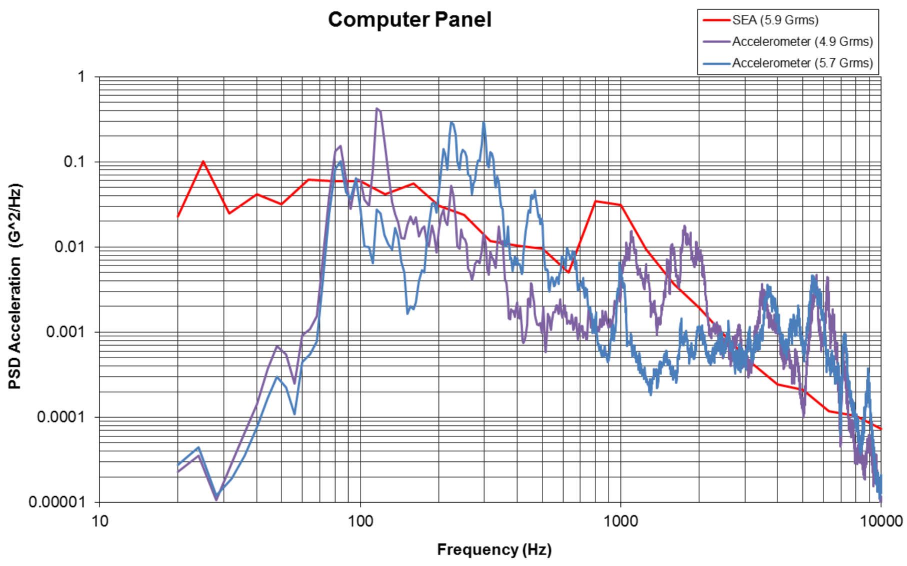

Figure 5. Computer Panel Acceleration PSD results.

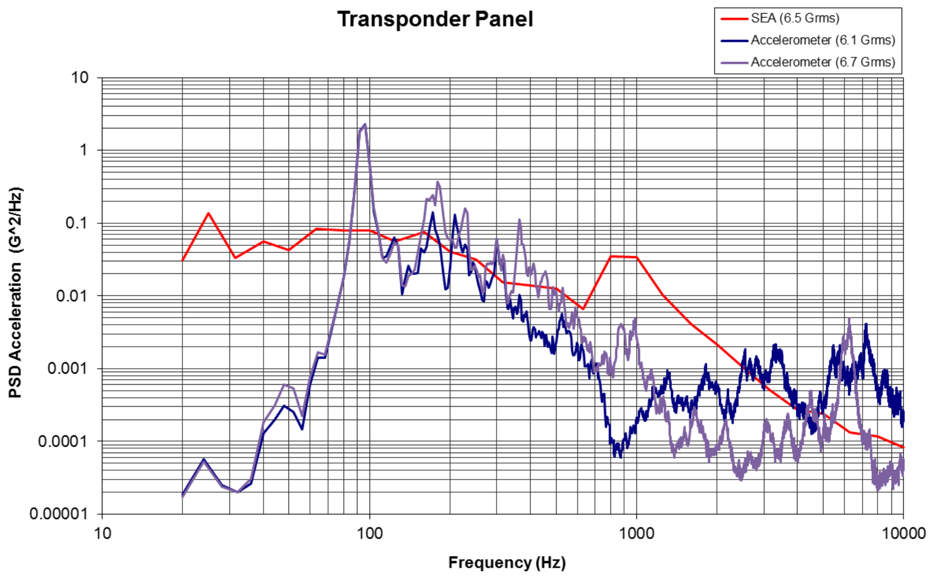

Figure 6. Transponder Panel Acceleration PSD results.

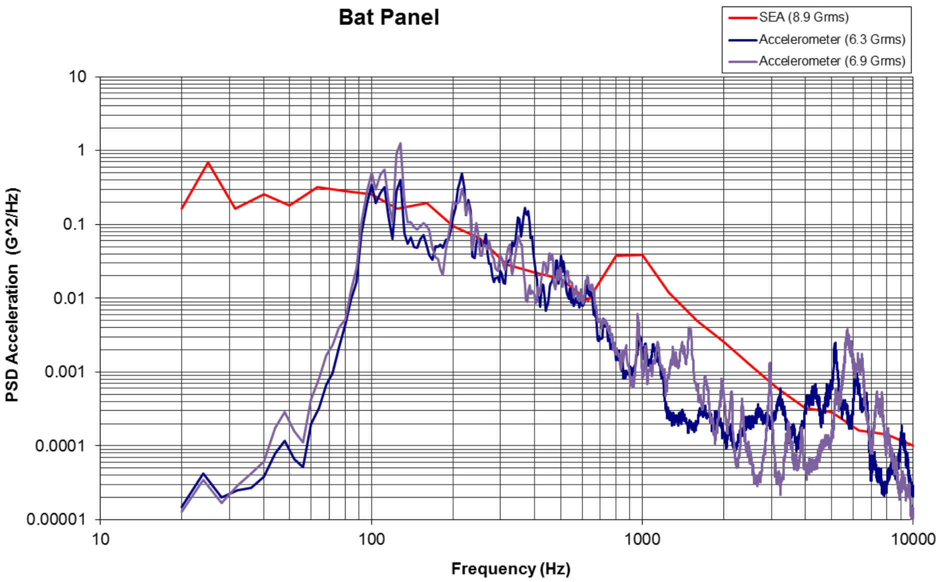

Figure 7. Battery Panel Acceleration PSD results.

5 Conclusions

As expected, SEA proved to be a powerful analytical tool to predict acoustic responses for aerospace structures and with a very simple implementation. The results of an acoustic test with closed loop control on pressure responses for large acoustic chambers are especially suitable for SEA method due to the high density of acoustic modes and the prevention of chamber modes influences. In this sense, the only major remaining uncertainties are regarded to the panel first modes, which can be predicted with simple analytic calculation of simple supported panels to achieve very reasonable results, as an example. The predicted Grms were also in good agreement with the registered ones during the tests.

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