VIBRO-ACOUSTIC STUDY OF COROT: ANALYSIS AND TEST CORRELATION

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ABSTRACT

This paper deals with the different steps of the acoustic study performed on COROT instrument covering from analytical simulation aspects until qualification by acoustic test. Acoustic load during launch is particularly constraining for the space telescope COROT due to its architecture.

First, the results of the vibro-acoustic analysis performed using ASTRYD™ in order to predict the responses of the spacecraft to acoustic pressure during launch are presented. The calculation method and the obtained responses of the structure submitted to a diffuse field are exposed. In addition, physical measurements during the qualification test in the reverberating acoustic chamber are presented.

At last, the correlation between the simulation results and the measured test results is analyzed. Acoustic responses were globally well predicted by the simulation, in particular for structures strongly excited by acoustic pressure. In definitive, this correlation shows the relevance and the limitations of the current release of ASTRYD™ in vibro-acoustic analysis for test prediction.

1. INTRODUCTION

During launch, spacecraft are submitted to severe acoustic environment generated by noise of the engines' firing, blast wave and aero dynamic phenomena. This acoustic load is one of the dimensioning cases of spacecraft structures. For structures presenting large light surfaces, which is the case of the space telescope COROT, this load is particularly constraining. Therefore it must be characterized and taken into account in the design. Finally, at the end of the spacecraft development, the resistance of the structure to this environment must be proven by test.

After a brief presentation of the specimen, results for each step of the acoustic study are given including correlation between analysis and test.

The results presented hereafter concern merely the COROT instrument. However, the same study was performed at satellite level giving similar results, although not presented in this paper.

2. PRESENTATION OF COROT

2.1. The mission

COROT stands for COnvection ROtation and planetary Transit. The COROT space telescope is a cooperative mission between the French space agency CNES (responsible for the mission, the satellite and the instrument development), the CNRS and international partners ESA, Austria, Belgium, Brazil, Germany and Spain. COROT has the dual goal of carrying out the first systematic astero seismological observations of stars other than our Sun and conducting a survey searching for planets like ours at an unprecedented level of accuracy.

Following its launch into a near perfect orbit on December the 27th 2006 by Soyuz 2-1B, the COROT satellite has been busy carrying out both in-orbit verification and its first scientific observations on its polar orbit, at the altitude of 896 km. 'Corot-Exo-1b' is the first exo-planet discovered by COROT.

2.2. The instrument

The COROT payload is mounted on a platform belonging to the PROTEUS family, which is designed for 500 kg class satellites, operating in low earth orbit.

The instrumental concept consists on a white light photometer (seismology) with spectra dispersion (exo-planets). The payload structure is composed of the four main products:

- COROTEL, the telescope consisting of the afocal part with the two parabolic mirrors off axis, the external cylindrical baffle and the one-shot cover design to protect the telescope.
- COROTCAM, the camera composed of the focal unit including the dioptric objective, two

proximity electronics units, a shielding, a prism and the focal unit thermal control.

- COROTCASE, the equipment bay, which includes the mechanical structure with 4 sidewalls, a set of fine thermal control components, the scientific processing units, the housekeeping units and the two converters for instrument power supply

These substructures can be easily recognized in the figure below:

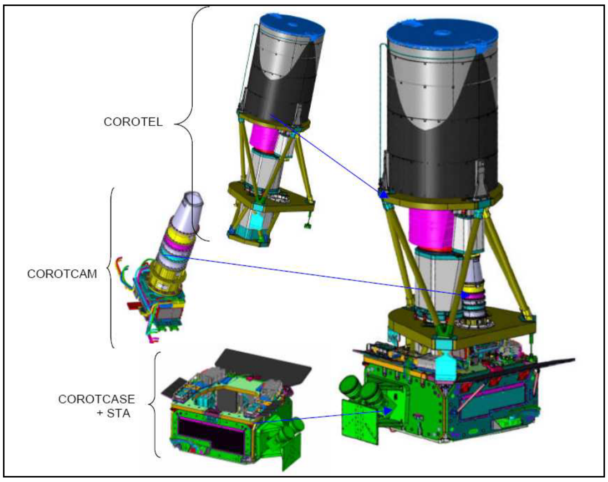


FIG 1. COROT architecture

3. VIBRO-ACOUSTIC SIMULATION

3.1. Simulation technique

In order to predict the responses of the payload to acoustic pressure during launch, a vibro-acoustic analysis was realized using ASTRYD™. This software, developed by 01dB Metravib and CNES, solves fluid-structure coupling problems in the time domain and belongs to the so-called "FEM+BEM" techniques, a deterministic method which combines the Finite Element Method and Boundary Element Method. Therefore, the analysis is based on two different models. The dynamic model allows representing the dynamic behaviour of the structure in the desired frequency range. The acoustic model represents the geometric surfaces of the structure which are in contact with the fluid. The resolution of the coupled problem is performed in the modal base of the structure by modal superposition techniques after projection of this base on the acoustic mesh: we attribute to each node of the acoustic mesh a modal displacement corresponding to a combination of modal displacements (normal to the acoustic surface) of nodes belonging to the mechanical mesh, which have been associated geometrically to these acoustic nodes.

3.2. Simulation in diffuse field

During acoustic test, the structure is excited by a diffuse field, which results, in theory, from the superposition of an infinite number of statistically independent plane waves, coming from all directions of the space.

To simulate the response of COROT in the acoustic test, the so called direct calculation method was applied. The diffuse field is modelled by a finite number of plane waves, generated by as many remote acoustic sources distributed in the space. In our case, the number of plane waves was 16, which gives satisfactory results.

For each plane wave, ASTRYD™ computes not only the pressures and accelerations on the acoustic mesh, but also the modal coordinates η of the response of the structure. This allows building the response of the structure on the nodes of the dynamic model, and for any other physical value given by modal analysis on the dynamic model, such as displacements, forces or stresses, by modal superposition. Therefore, the transfer function between the plane wave number i and the physical value v, at node P of the dynamic model, and at pulsation ω, is given by:

$$H_v^i(P,\omega) = \sum_{k=1}^{N_{modes}} h_\eta^i(k,\omega).\varphi_v(k,P)$$

where:

- $h_\eta^i(k,\omega)$ is the transfer function between the acoustic source number i and the modal coordinate η of the eigenmode k, at pulsation ω, computed by ASTRYD™ ;

- $\varphi_v(k,P)$ is the k^{th} modal vector of the physical value v at node P, given by modal analysis.

Once the ASTRYD™ computation has been performed for every plane wave, the response of the structure in diffuse field can be obtained by quadratic superposition of the contribution of each plane wave. The power spectrum density (PSD) of the response of the physical value v, at node P and at pulsation ω, under the diffuse pressure field defined by its PSD $S_{p_0p_0}$, is then given by:

$$S_{vv}(P,\omega) = R^2 \sum_{i=1}^{N_{pw}} \alpha_i \left|H_v^i(P,\omega)\right|^2 .S_{p_0p_0}$$

where:

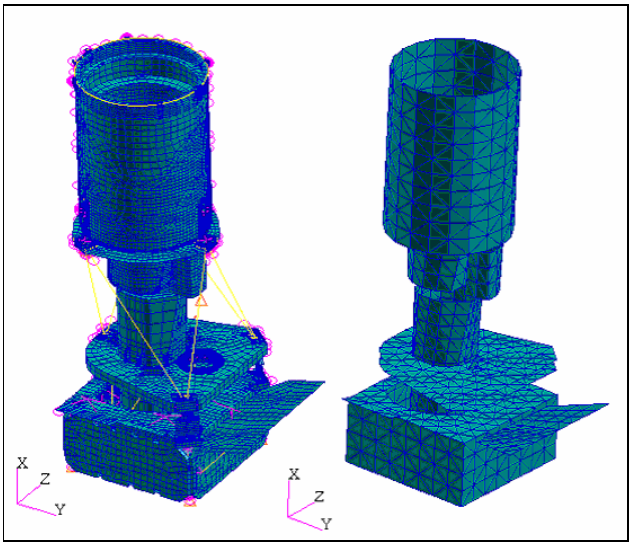
-R is the distance of the remote 16 acoustic sources,

-α_i is a coefficient proportional to the solid angle attributed to the plane wave number i ($\sum_{i=1}^{N_{pw}} \alpha_i = 1$)

As can be seen, this method requires 16 ASTRYD™ computations in all, and then all kinds of results are available.

3.3. Description of the models

Two different meshes of the COROT instrument were prepared for the need of the computations, as shown on figure 2:



COROT dynamic mesh   COROT acoustic mesh

FIG 2. COROT models for simulations with ASTRYD™

From the acoustic point of view, the acoustic model used for the simulations comprises only 'wet' surfaces, that is, surfaces in interaction with the surrounding air. Generally, this type of meshes involves all the surfaces of the primary structure. Equipment surfaces were not modelled (being lumped masses in the FEM model their dynamic behaviour will not be taken into account in the vibro-acoustic computation), neither the inner surfaces of the structure. The resulting acoustic mesh has a total of 1842 triangular elements, all of them 'thin' elements, that is, wet on both sides. In principle with this mesh characteristics, analysis up to 790 Hz could be led. However, the version of the software which was used, is able to manage a maximum of 1000 modes. Consequently, computation had to be limited to 450 Hz and the fourth octave band is not completely covered.

From the dynamic point of view, an updated model is desirable since acoustic simulations depend strongly on the precision of the dynamic model. However, as the simulations were realised before mechanical testing in the framework of its preparation, no updated model was available. Nevertheless, the model revealed itself as of quite good quality at the dynamic test, particularly in terms of lateral frequency, as shown in the table below:

	Predicted by FEM	Encountered at Sinus test
1st lateral (Y)	38.8 Hz	38 Hz
2nd lateral (Z)	45.1 Hz	45 Hz
1st longitudinal (X)	79.5 Hz	96 Hz

TAB 1. First frequencies predicted and measured

In terms of longitudinal frequency the model was not so good due to the deficiency of stiffness in the telescope model. Consequently the obtained responses from the analysis in this direction could not be very realistic.

This model is composed of 61703 elements and 46320 nodes and represents a total mass of 257 Kg.

4. VIBRO-ACOUSTIC SIMULATIONS

The technique and the models described in paragraph 3 were used to perform the acoustic test predictions. For the computation, 915 modes were taken into account and a modal damping of 1% was assumed for all these modes.

The objective of this simulation was to identify possible problems that may appear during qualification in the acoustic chamber. Consequently, the obtained results were compared to random requirement for each of the subassemblies. A typical output of this analysis is a graph like the one exposed in figure 3, which shows the PSD acceleration curves predicted with ASTRYD™ at the focal unit in COROTCAM and its interfaces and they are compared to the random qualification environment :

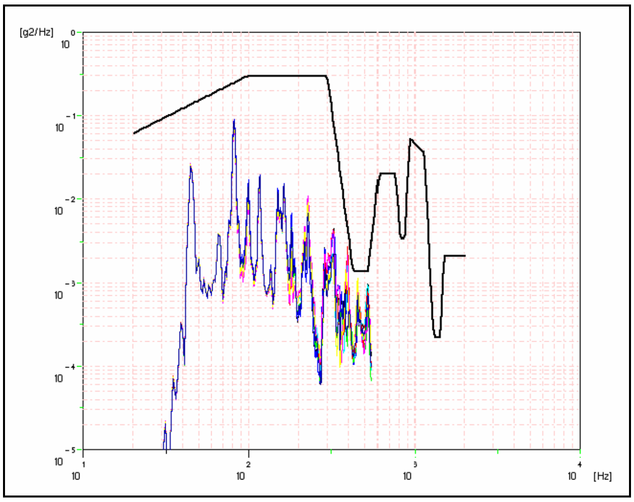


FIG 3. Predicted levels in the focal unit (COROTCAM) and random requirement level

Globally, the fulfilment of the requirements in term of effective values was confirmed. In spite of the limitation in frequency of the study, the obtained values were always considerably lower than the specified values.

In the frequency domain, the random requirements were exceeded by analysis' results in different frequencies, but each case was studied and declared not critical for different reasons specific to each case.

5. VIBRO-ACOUSTIC TEST

The acoustic test of the COROT instrument was performed in the MVS reverberation chamber of Intespace (Toulouse) in October 2005. The test instrumentation consisted of several accelerometers, extensometric gauges and force cells, making a total of 120 measuring channels and 4 control microphones around the structure.

The levels to be applied for acoustic qualification were those indicated in Soyouz User's Manual and given in the table below:

Octave Band Center Frequency (Hz)	Qual. level (dB)
63	135
125	137
150	139
500	137
1000	128
2000	124
Overall	145

TAB 2. Pressure levels to be applied for COROT acoustic test

During acoustic test, the specified diffuse field is supposed to be obtained by the multiple reflections of the pressure waves generated by the horns of the chamber. But the dimensions of the room being finite, although large, we have to take into account, that simulating by test a diffuse field is not easy in the low frequency range.

The acoustic test was realised without any problem: Required levels were accomplish and no damage was identified in the structure.

Once again, the measured DSP acceleration curves were compared to random requirement for each of the subassemblies, as shown in picture 4:

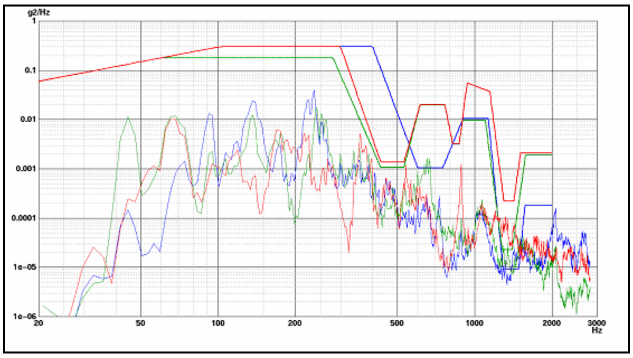


FIG 4. Measured levels in the dioptric objective (COROTCAM) and random qualification level

The ratio between the maximum acceleration reached in the transient signal and the effective value of the signal was also studied. We could state that this ratio staggers between 4.0 and 5.3 for the majority of the accelerometers, with an average of 4.7 and peaks up to 7.56. However the proportion values which exceeded 3σ is 0.25 % of the time in average with variations of 0.04 and 0.44 % depending on the sensors. Therefore, we can conclude that acceleration bounded to 3σ over 99.75% of the time in average which is consistent with a Gaussian distribution.

We can also highlight that for all the electronic units and for the different elements of the camera that had undergone a qualification test in random environment, these tests revealed themselves as qualifying with respect to the levels seen in the acoustic environment at instrument level.

For the baffle, random vibration test performed were qualifying in terms of interface levels, but not in terms of internal responses. The qualification of the baffle was only achieved during the acoustic tests. This equipment was strongly excited in the acoustic test at instrument level in the octave band centred at 250Hz. As a conclusion from this test, a dispensation of -2dB in this frequency range was requested from the launcher authorities for the acoustic qualification test at satellite level in order to avoid the useless reproduction of this solicitation.

6. MEASUREMENT AND ANALYSES CORRELATION

In the following figures the correlation between simulation results and acoustic test measurements are presented. Globally, when the acoustic excitation is high, the simulation achieves to predict rather well the levels measured in the acoustic chamber.

First of all, for one of the converters for instrument power supply BCVETN, situated on the panel –Z of the COROTCASE the out of plane accelerations Power Spectral Density measured and calculated curves are shown in the figure below:

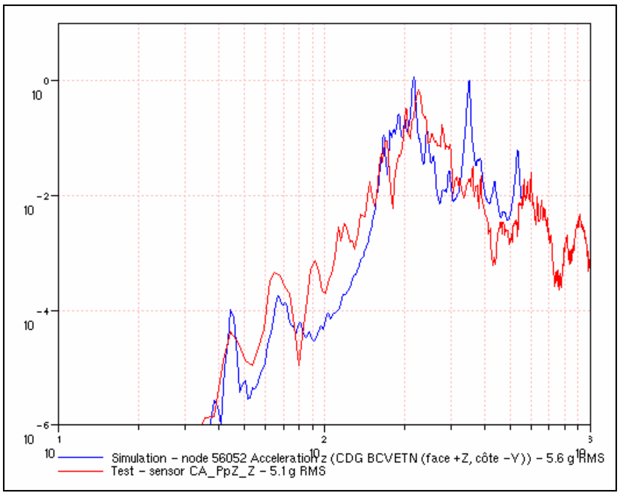


FIG 5. Correlation test/simulation on the BCVETN out of plane (COROTCASE)

The overall aspect of the computed curve fits the measured one. The response of this unit on the second lateral mode in the normal direction of the panel is rather well simulated. The frequency is good and the level is slightly too high. We measured 4.10^{-5} g^2/Hz while the analysis predicts 8.10^{-5}, which makes a difference of 3 dB. However the overall RMS acceleration level predicted is greater than the computed one, partly due to the occurrence of a peak around 350Hz which is inexistent in the measurements.

A similar phenomenon is also visible in the following graph, which shows the response on the digital processing units DPU1 and DPU2, located on the panel –Y of the COROTCASE:

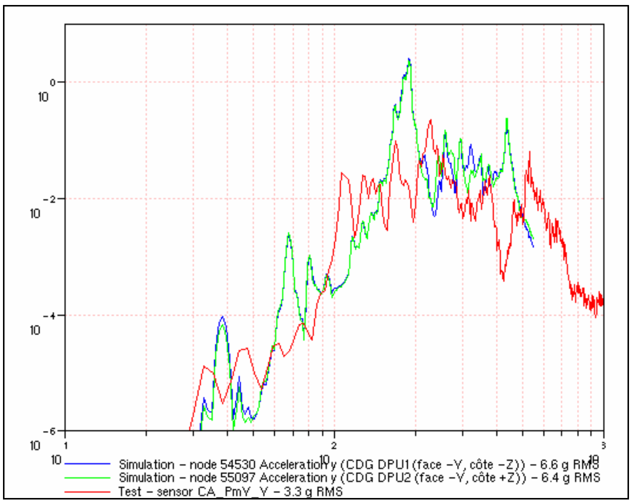


FIG 6. Correlation test/simulation on the DPU1 and DPU2 out of plane (COROTCASE)

The explanation is the presence of the balancing mass on this panel for the acoustic test. This mass was not taken into account in the FEM. Therefore, in the calculated modal base we find this mode at 190 Hz, which is a mode of this panel of the equipment-bay and which disappears when adding the mass for the test.

On the contrary, in-plane responses of the same unit BECVT are much less excited and therefore the correlation is worse:

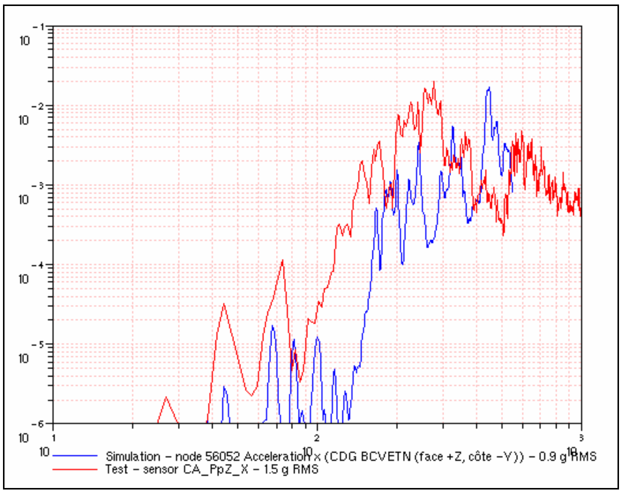


FIG 7. Correlation test/simulation on the BCVETN in-plane (COROTCASE)

Concerning the first lateral mode of the structure a frequency shift of about 7 Hz is observed for the measured responses with respect to the analytical ones, as shown clearly in the figure below:

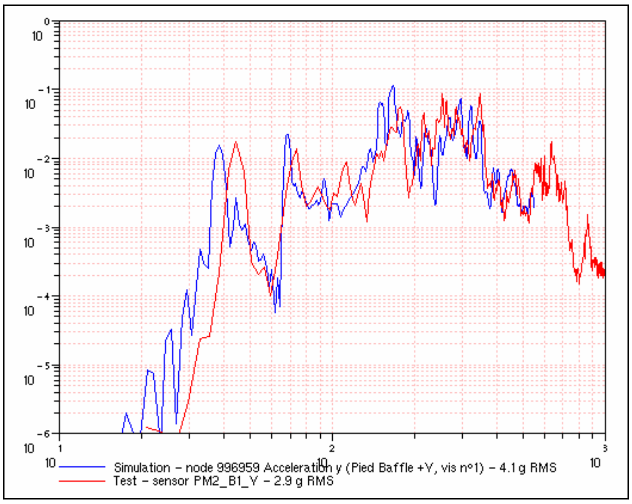


FIG 8. Correlation test/simulation on the M2 Plate near the baffle IF in Y direction (COROTCASE)

Being this mode well predicted by the FEM, as it was clearly identified at sinus test, this shift cannot be the consequence of a modelling error. One probable explication would be to link this shift to a non-linearity manifested in the acoustic test or simply that this mode is not responding in acoustic and only the second lateral mode is.

In the figure below, once more the good correlation between simulation and test results is illustrated by the curves of PSD acceleration on the baffle in lateral direction:

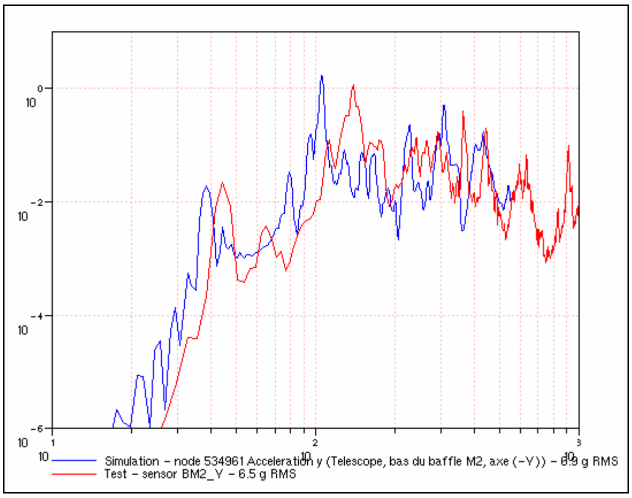


FIG 9. Correlation test/simulation on the baffle in lateral direction

The overall aspect of the computed curve fits quite well the measured one and the overall predicted level is just 0.4 gRMS above the overall measured level, which represent an acceptable difference of 6%.

On the contrary, predicted longitudinal responses on the baffle, which present a much lower level, are in worse agreement with the measurements, as shown in the following figure:

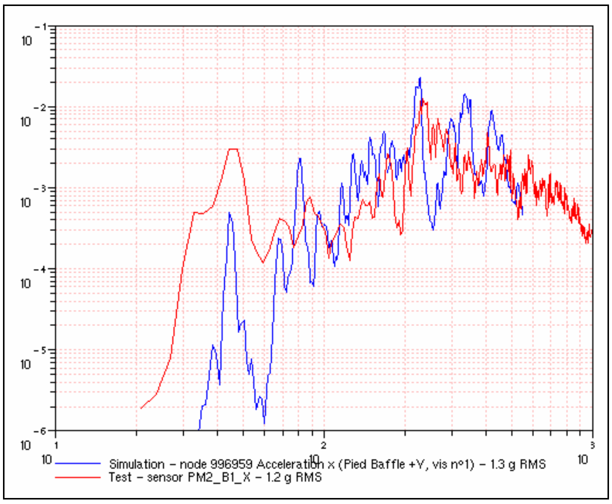


FIG 10. Correlation test/ simulation on the baffle in longitudinal direction

This is due to the lack of representativeness of the dynamic model in the longitudinal direction.

Another component strongly excited by acoustic environment is the focal unit radiator. As illustrated in figure number 11, responses at this part of the structure predicted by the simulation are in very good agreement with test results:

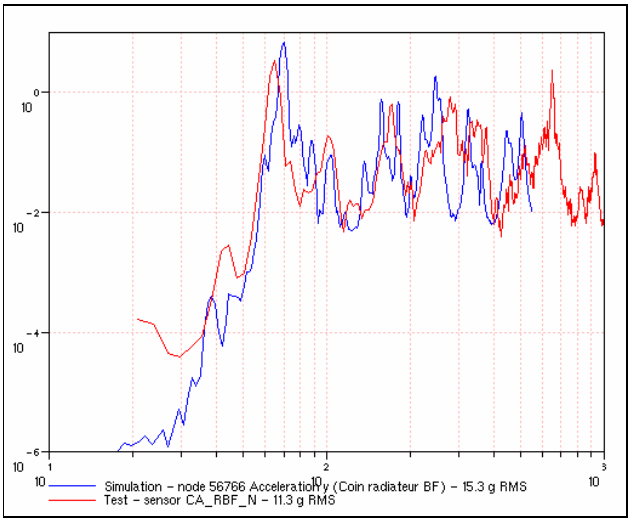


FIG 11. Correlation test/simulation on the focal unit radiator Y direction (COROTCASE)

A brief correlation of the overall RMS acceleration is shown in table 3 in different locations of the structure. It can be appreciated that simulation results are in general overestimating measured values, so simulation can be considered conservative with respect to the test. However the absolute difference between both values is not negligible as it exceeds often 3dB.

For the highest acceleration level measured (18gRMS at the obturator hinge), the predicted level is slightly underestimated but the difference is only 0.5 dB.

	Axis	gRMS Analysis [0-450Hz]	gRMS Test [0-450Hz]	Diff. [gRMS]	Diff. [dB]
COROTCASE -Y panel	X	0.8	1.3	-0.5	4.22
	Y	6.6	2.7	3.9	7.76
	Z	1	0.8	0.2	1.94
COROTCASE +Z panel	W	0.9	1.1	-0.2	1.74
	Y	0.9	0.8	0.1	1.02
	Z	5.6	4.7	0.9	1.52
COROTEL Top plate	X	1.3	0.8	0.5	4.22
	Y	4.1	2.5	1.6	4.30
	Z	2.5	1.5	1	4.44
BAFFLE Obturator hinge	Y	16.9	18	-1.1	0.55
	Z	12.7	5	7.7	8.10
COROTCAM	X	2	1.2	0.8	4.44
	Y	2.3	1	1.3	7.23
	Z	1.6	0.7	0.9	7.18

TAB 3. Acceleration levels

To sum up, responses which were correctly predicted can be recapitulate sum up as follows:

- Normal responses in the equipment-bay panels
- Responses in the Focal Unit radiator and in the thermal bus
- Responses on the top plate of the telescope
- Responses in lateral direction on the baffle

On the contrary, the following responses were not well predicted:

- In-plane responses in the-equipment bay panels
- Responses in longitudinal direction on the baffle
- Reponses in the camera. This fact is probably linked to the absence of acoustic modelling of the camera. Being condensed in the dynamic model, its surfaces were not taken into account in the acoustic mesh.

7. CONCLUSION

The simulation relies mainly on the correctness of the dynamic finite element model. Moreover, it implies an idealization of the diffuse field, while tests are actually performed in reverberating chambers. For these reasons, discrepancies between predicted and measured responses were encountered.

Nevertheless, acoustic responses were globally well predicted by the simulation, in particular for structures strongly excited by acoustic pressure.

In definitive, this correlation shows the relevance of the ASTRYD^{TM} vibro-acoustic analysis for test prediction.

8. REFERENCES

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