Integration of NVH testing:
A case study on a spatial structure.

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This aerospace mechanical support structure was subject to a comprehensive set of experimental and analytical mechanical engineering methods. Also the correlation between the results from the different used methods is discussed extensively.

In a first step a FEM model was created and the analytical structural modes were calculated. These FEM model results were then utilized in pre-test analysis tools to obtain the optimum excitation and response locations for the EMA experimental modal. A full experimental modal measurement sequence was carried out and the results were correlated with the analytical FEM model results.

In a second step the structure was tested on a shaker, using standard vibration control Random and Sine excitation methods.

1. Introduction to the special mechanical structure

The structure under test is a part of the global facility designed by Centre Spatial de Liège for the ground mechanical qualification of the science payload of Herschel Space Observatory*.
The three Herschel Focal Plane Units† will be launched in cold conditions (~ 4 K). Therefore, ground qualification at cryogenic temperature (< 20 K) is required.
To reach such low temperature a vacuum environment is required, due to force limitation of the shaker (maximum 200 kN in sine mode); the weight of a vacuum vessel surrounding the instrument can not be included in the test set-up. So the solution is the mechanically linking the moving part of the shaker (i.e. slip table or head expander), being in ambient conditions, to the structure inside the vacuum chamber in cold environment (see figure 1.1). This mechanical link is what we call the Vibration Test Adapter and is the subject of the study presented in this article.

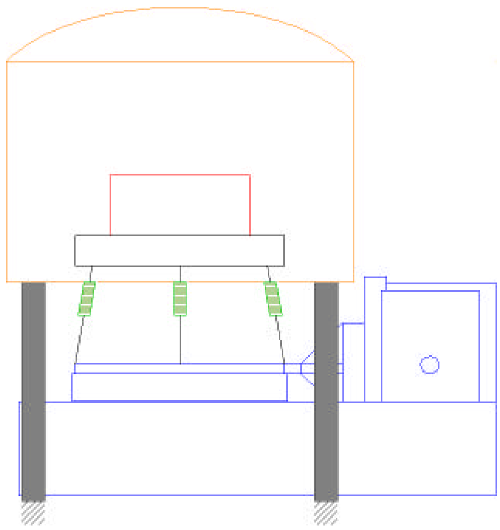


Figure 1.1 - Vacuum vessel ; Shaker ; Supporting frame ; Leak-tightness device ; VTA ; Specimen

For more information about the global facility see [1].

The VTA has been designed according to the following target specifications:

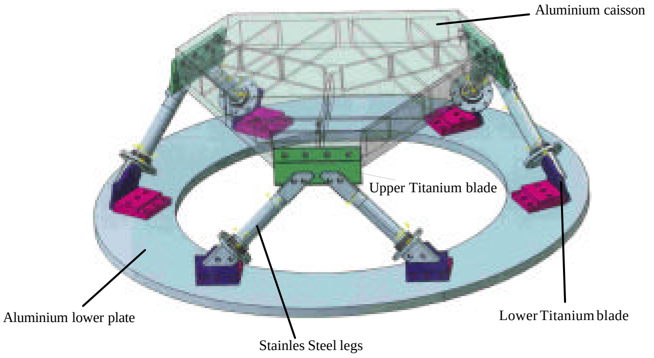
- Increasing of the first resonant frequency (target: above 200 Hz)
  To allow a good control of the input level during the qualification test
- Limiting of the structural mass
  To be able to excite with the required qualification level of 20 g in sine

* Fourth ESA cornerstone mission to be launched in 2007 by Ariane 5
† HIFI, PACS and SPIRE

- Reduction of conductive heat flux through the supporting legs
  A temperature lower then 20 K on the instrument could be reached
- Acceptable thermo -elastic behaviour
- Imposed geometrical dimensions (volume of instruments, interface with vacuum vessel, size of shaker, etc…)

The final design is presented at

Figure 1.2.
Six stainless steel legs link a lower aluminium disk to an upper aluminium caisson thanks to titanium flexible blades. This lower plate is attached either on the slip table or on the head expander of the shaker. This pseudo-isostatic structure has acceptable thermo -elastic behaviour from room temperature to 20 K. The cross section of the legs is reduced to minimize the conductive heat flux.



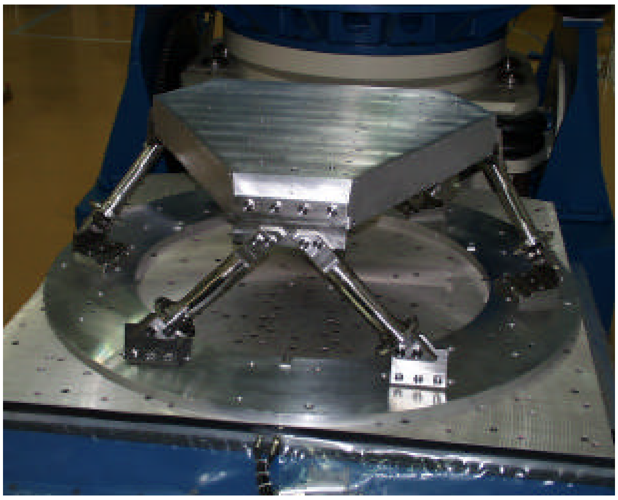


Figure 1.2 - Sketch and photo of VTA

2. Finite Element Analyses

A Finite Element Model was realised using Samcef software (see [2]).
This FEM of the VTA used shell elements (for the upper caisson and the blades) and beam elements (for the connecting hollow tubes). Each vertex of the model has six Degrees Of Freedom (i.e. 3 translations and 3 rotations). The FEM has the following characteristics:

- 6120 DOF
- 1017 nodes
- 1188 elements

On this model we computed a modal analyse, as boundary conditions we fixed the 6 DOF of each node on the lower side of the lower blades and we simulated the instrument by a mass of 80 kg at 178 mm rigidly connected to the VTA (see Figure 2.1).

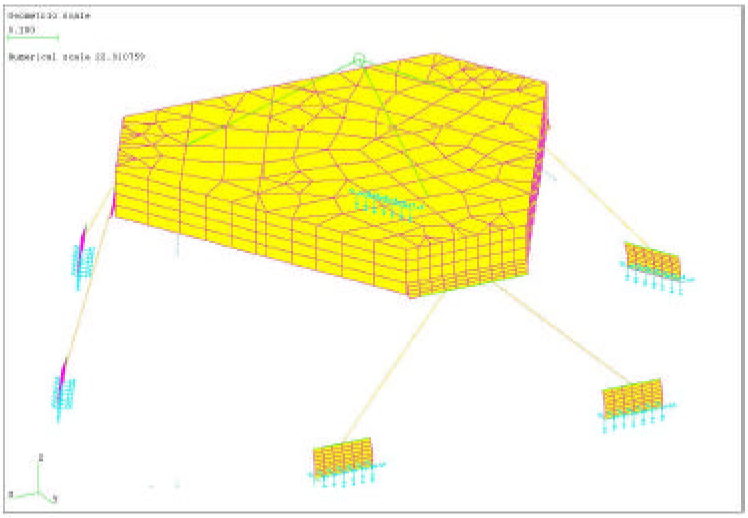


Figure 2.1 - FEM of the VTA

The computed resonant frequencies for such model are:

- 747Hz, caisson alone in free-free conditions
- 257 Hz along X horizontal axes (see Figure 2.2)
- 257 Hz along Y horizontal axes
- 266 Hz along Z vertical axes (see Figure 2.3)

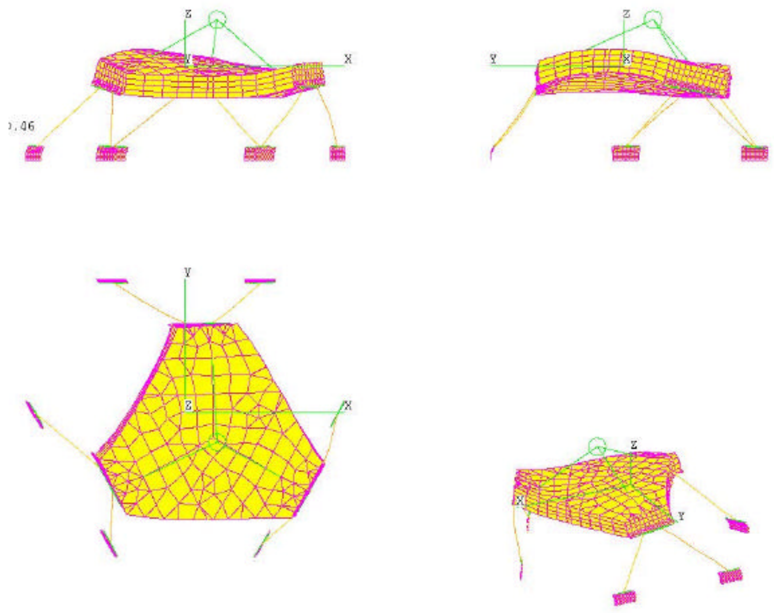


Figure 2.2 - First horizontal mode (loaded)

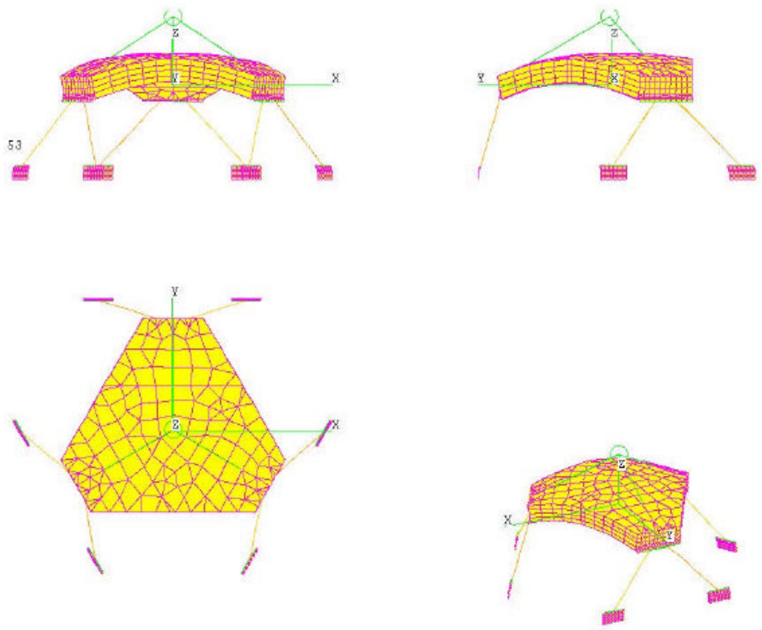


Figure 2.3. - First vertical mode (loaded)

The given mass of the VTA from the FEM is 231 kg; the measured mass of the built VTA is 236 kg.

From these calculations we can conclude that our goal to have the first resonant frequency above 200Hz within the set forth weight constraints is achieved.
Now we decided to compute the frequencies of the VTA without the instrument, which correspond to the configuration we will be focussing on in the following paragraphs. The results are:

- 348 Hz along Z vertical axes
- 365 Hz along X horizontal axes
- 365 Hz along Y horizontal axes

3. Pre-test analysis

In preparation for the Experimental Modal Analysis test, a pre-test analysis was executed based on the above mentioned FEM model. The calculated FEM mode shapes are used to select optimal locations and directions for measuring the experimental mode shapes and pairing them with the analytical mode shapes. Finding optimal measurement locations was based upon analysing the highest Average Driving Point Residues. This calculation was done using the FEMTools program (see [1]), and results are shown below in Figure 3.1. The optimal measurement locations are indicated in red colour.

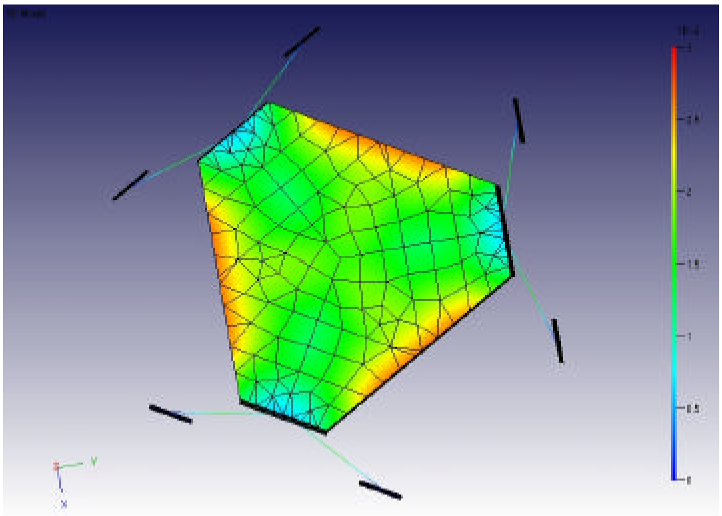


Figure 3.1.

4. First Modal Analysis: structure on slip table, free from shaker

Based upon pre-test FEMTools analysis the caisson structure is instrumented as follows:

- All measurement points should as much as possible match the FEM nodes.
- 10 tri-axial accelerometers where put onto the caisson and will be used as well for the Experimental Modal Analysis as for the Vibration control test.
- Hammer excitation in a large number of points will be used to obtain the required FRF's.

The experimental modal analysis was carried out using the SmartOffice product (see [1]). The structure was fixed to the slip table of the shaker slip table, with the shaker detached. Under these conditions following results were obtained:

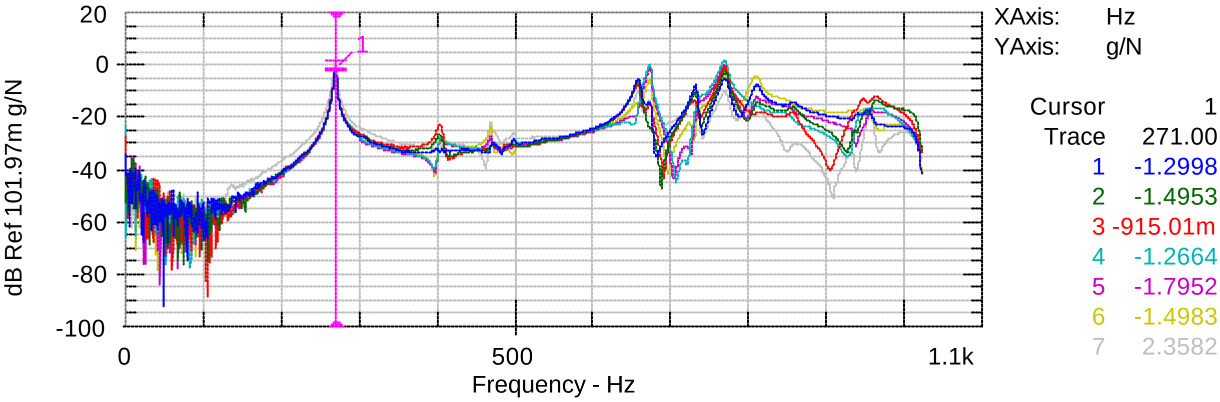


Figure 4.1.

- Multiple Z-responses related to excitation point 29Z, vertical resonant frequency at 271 Hz (see Figure 4.1).

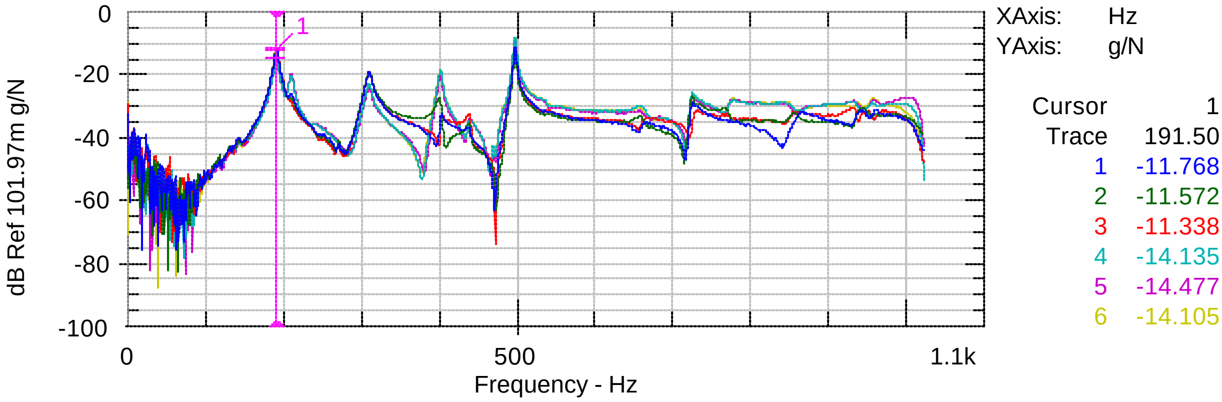


Figure 4.2.

- Multiple X-responses related to excitation point 29X, vertical resonant frequency at 192 Hz (see Figure 4.2).

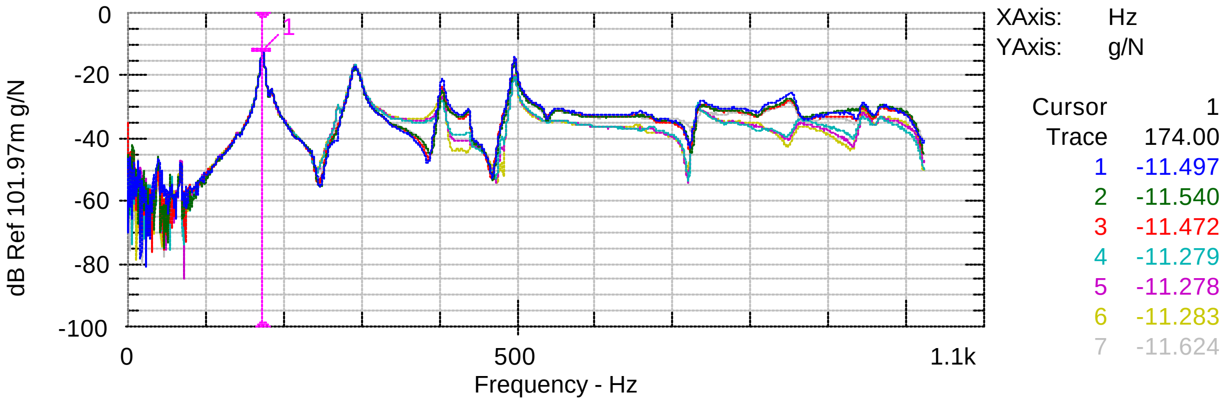


Figure 4.3.

- Multiple Y-responses related to excitation point 29Y, vertical resonant frequency at 174 Hz (see Figure 4.3).

The obtained Mode Shapes were:
- 174 Hz : pure Y translation of the caisson
- 192 Hz : pure X translation of the caisson
- 271 Hz : pure Z bending of the caisson (see Figure 4.4.)

It important to note that between the X, Y and Z axis a minimum interaction and cross talk existed.
However, the obtained experimental results differ fundamentally from the calculated FEM results (see Table 4.1).

Mode	FEM	EMA	%
X translation	365	192	90
Y translation	365	174	110
Vertical Bending	348	271	28

Table 4.1.

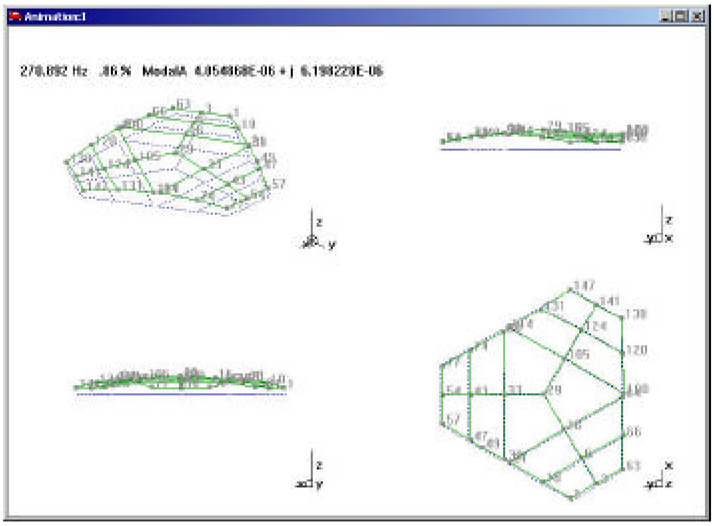


Figure 4.4.

5. Second Modal Analysis: Shaker connected, Y direction

Not being able to explain the large discrepancies between the FEM and EMA, we moved on to perform the EMA in the configuration we will be testing the final structure: we connected the shaker in the Y direction.

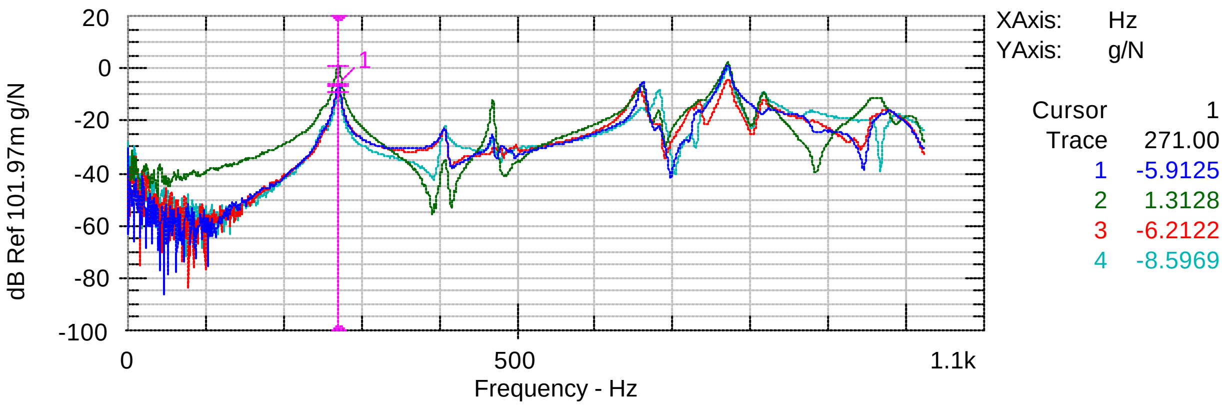


Figure 5.1.

- Multiple Z-responses related to excitation point 29Z, vertical resonant frequency at 271 Hz (see Figure 5.1.).

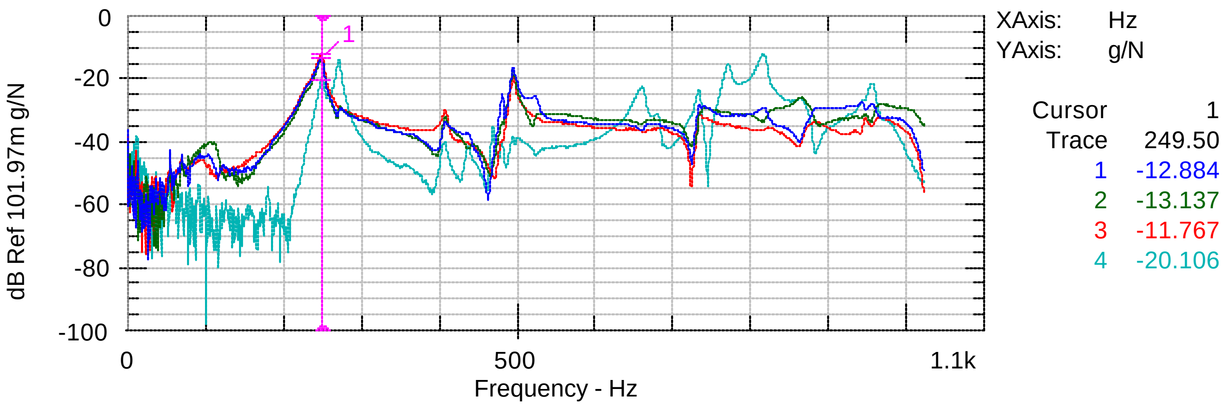


Figure 5.2.

- Multiple Y-responses related to excitation point 29Y, horizontal resonant frequency at 249 Hz (see Figure 5.2).

Now, it looks like that changing the boundary conditions of the slip table, the vertical bending mode was not changed in frequency. However, we noted a drastic move upwards of the frequency of the horizontal translation mode shape from 174 Hz to 250 Hz.

A similar test with connecting the shaker in the X direction resulted in similar changes (see Figure 5.3).

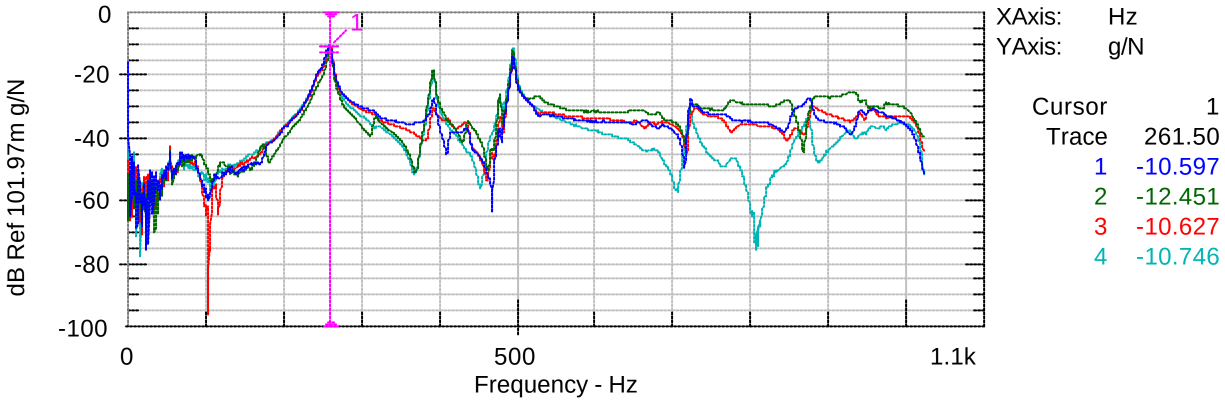


Figure 5.3.

6. Forth Modal Analysis: caisson Free-Free

The Free-Free test of the caisson structure indicated a resonant frequency of 710 Hz (see Figure 6.1), similar to what was found with the FEM calculations.

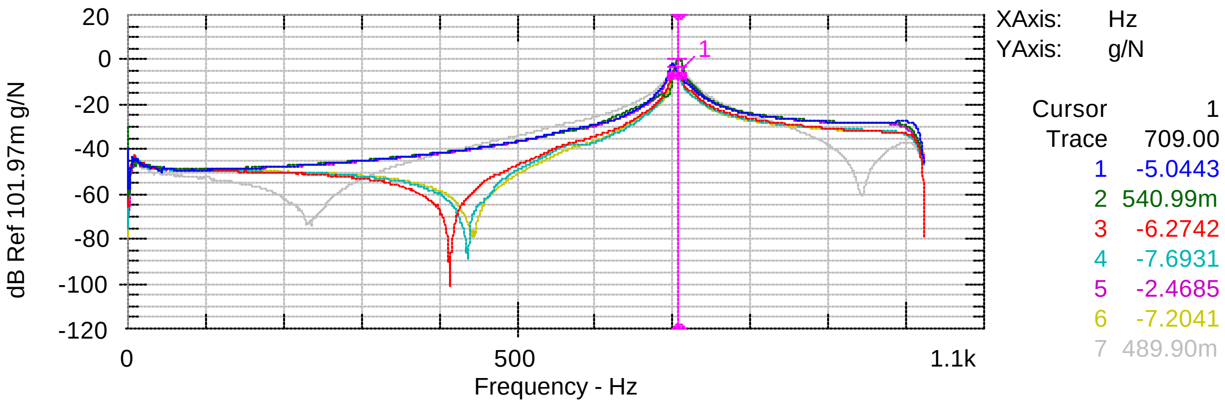


Figure 6.1.

7. Using what we learned to update the FEM model

Based upon our experience with the different experimental modal tests, we applied a FEM– EMA correlation and updating tool to get a better understanding of which parameters into the FEM model needed to be change in order to get better correlation between FEM and EMA.
In a first step the EMA measurement points were matched with the according FEM nodes (see Figure 7.1.). Based upon the fact that we had chosen the measurement points coinciding with FEM nodes this was easy to accomplish.

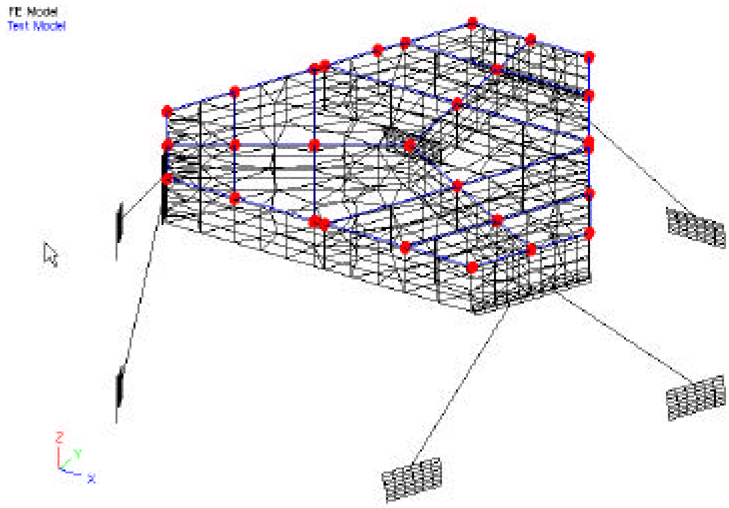


Figure 7.1.

In a second step the MAC (Modal Assurance Criteria) were calculated between the modes obtained in FEM and EMA (see Figure 7.2.). Upon a criterion of 75% MAC, 4 modes correlated very well.

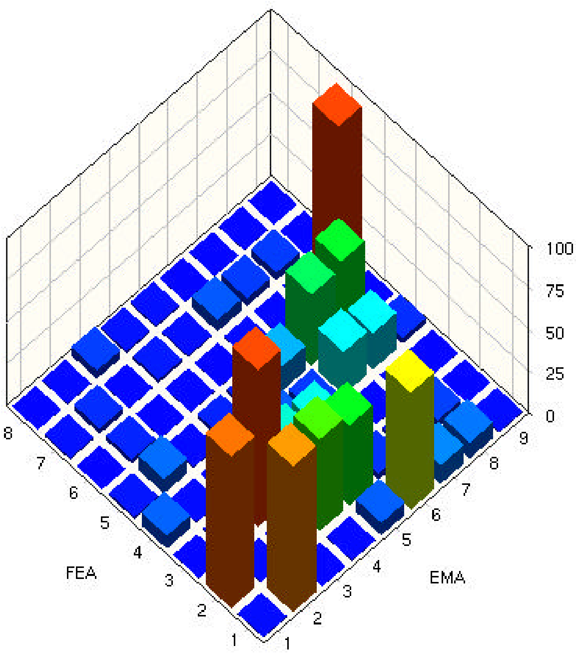


Figure 7.2.

- Mode 1 and 2, the X and Y translations, were reversed between the FEM and EMA results.
- The vertical bending was quite identical.
- The fourth mode was a rotation around the vertical axis.

The resulting frequencies were quite different and needed to be updated.

Knowing from previous analysis that:

- the caisson structure was well modelled, because the first resonant frequency from FEM and EMA quite well coincided
- the frequencies of the translation modes changed a lot when changing between different configurations

The main goal was to bring the X and Y translation modes into the 240 – 250 Hz range, the vertical bending mode into the 270 Hz range.

We concentrated on updating the FEM model:

- Case 1: allowing major stiffness changes into the legs, connecting the caisson structure to the slip table
- Case 2: changing the boundary conditions on the legs into flexible connections.

7.1. Case 1: Stiffness changes into the legs

Based upon a FEA -MEA updating process in 5 iterations, indicating a stiffness change of up to 50% into the six legs connecting the caisson structure to the slip table, the 4 resonant frequencies of the most correlated FEA modes came into the range of the EMA test.

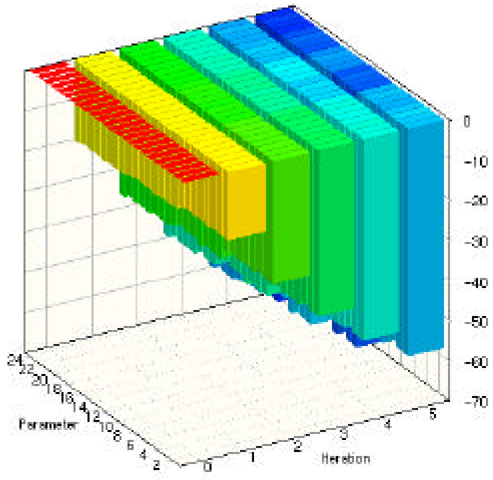


Figure 7.3

Mode	FEM	Hz	EMA	Hz	Error %	Mac
1	1	244	1	249	-2	89
2	2	254	2	261	-3	89
3	3	280	3	271	3	94
4	6	499	9	440	13	95

Table 7.1.

7.2. Case 2: Changing the boundary conditions

In this second case we exchanged the fixed boundary conditions with springs. Based upon an updating process, we obtained that springs of 1^E8 moved the 4 resonant frequencies of the most correlated modes came into the range of the EMA test.

Mode	FEM	Hz	EMA	Hz	Error %	Mac
1	1	238	1	249	-4	86
2	2	238	2	261	-9	90
3	3	288	3	271	6	95
4	6	450	9	440	2	95

Table 7.2.

Based upon an analysis of the new mode shapes and Mac values we expect that a combination of both above mentioned changes will be necessary to obtain an improved correlation between FEM and EMA.

8. Results of swept sine and random tests

Now we performed (in the same conditions as paragraph 5 with the input controlled on the slip table) a low level sine sweep (0.5 g ; 5-2000 Hz) on the VTA to identify the frequencies which we have to deal with during the tests on the instruments. We finally obtained 217 Hz for first horizontal axis (Y in our tests X in the Samcef model) and 229 Hz for the second horizontal axis.

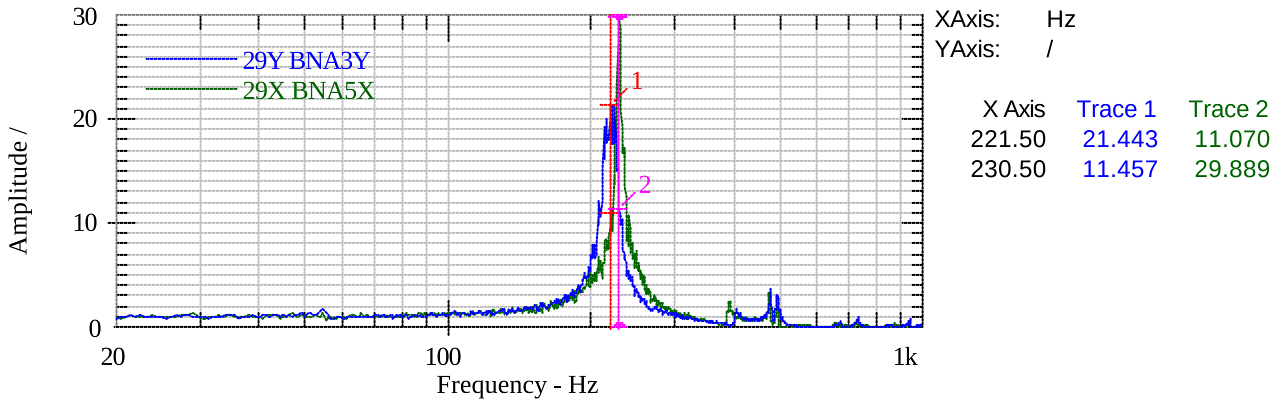


Figure 8.1.

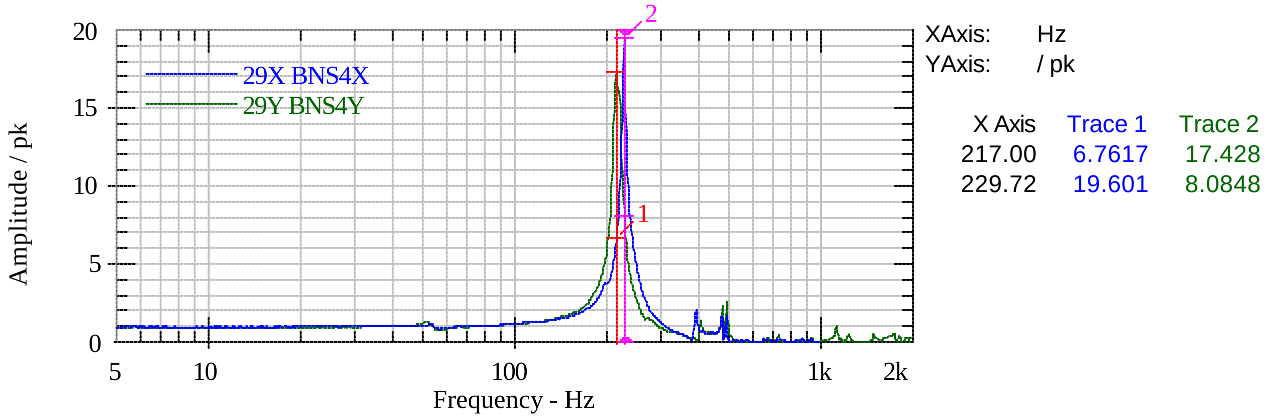


Figure 8.2.

We also performed a low level random on the VTA to verify if the frequencies were stables (some little difference were expected with respect to sine test). We obtained 221 Hz and 230 Hz, so a good stability was concluded

9. Conclusions

The results of this series of tests were good from a vibration control point of view. We did indeed we see quite high frequencies (above 200 Hz) with almost only translation modes, so we should be able to apply the requested excitation spectra to the base of the instruments to qualify (i.e. using a notching - control philosophy).
The next steps are:

- Conduct a frequency identification test of vibration test along the three axes of excitation.
- Perform the same series of tests on the VTA loaded with a dummy mass of 80 kg (decreasing of the frequencies is suspected, around 25 %)
- Qualify the VTA loaded by imposing the high-level sine and random tests (some settling is expected)
- Perform qualification series of tests with the VTA upper caisson structure in cold environment (increasing of the frequencies is expected due to mechanical properties of the materials in cryogenic condition)

To address the differences between the sine sweep excitation and the modal identification results, we will add some clamping device near the base of the VTA to improve the connection with the slip table (or head expander), since the results of the comparison of the FEM and the measurements suggest that the boundary conditions are the most driving parameter for the improvement of the frequencies.

Reference

[1] A. Cucchiaro, C. Grodent, P. Jamotton, J-S. Servaye, C. Delrez, Cold Vibration Facility at CSL premises. 22nd Space Simulation Conference *Mission Success Through Environmental Test*

[2] C. Delrez (AMOS), Structural analysis report. Internal technical note TN-CSL-CRYOV-01001

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