**Advances in Aircraft and Spacecraft Science**, *Vol. 5, No. 2 (2018) 251-258*
DOI: https://doi.org/10.12989/aas.2018.5.2.251 

# A virtual shaker testing experience: Modeling, computational methodology and preliminary results

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*(Received December 20, 2016, Revised January 11, 2017, Accepted September 9, 2017)*

**Abstract.** This work illustrates the progress of a TAS activity at exploring the challenges and the benefits of the Virtual Shaker Testing (VST) approach. The definition and the validation of new computational methodologies with respect to the state of the art were encouraged throughout this activity.

The shaker Finite Element (FE) model in lateral configuration was built for the purpose and it was merged with the SpaceCraft (S/C) FE model, together with the S/C-Shaker adapter. FE matrices were reduced through the Craig-Bampton method. The VST transient analysis was performed in MATLAB® numerical computing environment. The closed-loop vibration control is accounted for and the solution is obtained through the fourth-order Runge Kutta method.

The use of pre-existing built-in functions was limited by authors with the aim of tracing the impact of all the problems' parameters in the solution. Assumptions and limitations of the proposed methodology are detailed throughout this paper. Some preliminary results pertaining to the current progress of the activity are thus illustrated before the conclusions.

**Keywords:** virtual shaker testing; vibration tests; structural dynamics; S/C mechanical testing

## 1. Introduction

In the aerospace industry the base-shake sine testing is a well-known technique employed to ensure that the structure will survive the low frequency environment. Ideally, the input accelerations on the tested structure during the vibration test would be the same as the ones stated in the test specification. However, this ideal situation is not practical for several reasons. Among the others, one of the most important causes of discrepancies is given by the dynamic coupling between shaker and test article, which grows as the effective masses of the article increases. Since the numerical sine test-prediction is commonly obtained by imposing the ideal acceleration of the test specification to the base of the S/C, the numerically predicted structural responses of the tested structure are typically not in good agreement with the ones from the test.

The VST approach is a promising way to include the dynamic coupling between shaker and test article in the numerical sine test prediction. Basically, VST consists of a refined modeling

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http://www.techno-press.org/?journal=aas&subpage=7 ISSN: 2287-528X (Print), 2287-5271 (Online)

modelisation of the structure to be tested, the shaker with its seismic mass and the facility vibration control system altogether. The subsequent transient simulation will be representative of the vibration test on a specific facility, accounting for all the parameters which may impact or alter the dynamic behaviour of the payload (e.g. S/C-shaker dynamic coupling, choice of compression factor, sine sweep rate effect and beatings etc…). In principle, multiple VST simulations can be conducted by considering the same test item but referring to different test facilities. The results would help the program manager to choose the best solution for conducting the test campaign and mitigating the risks. Additionally, VST can limit the overtesting of large S/Cs; can favour the optimization of margins of safety and can lead to managing the dynamic coupling between specimen and shaker. Moreover, VST allows performing a physically consistent S/C FE model updating from test responses, the latter being a critical point for the launcher-S/C final Coupled Load Analysis (CLA).

To the best of the authors' knowledge, the ground-breaking VST simulation is the one by Apolloni and Cozzani (2007), followed by works (Ricci *et al.* 2008, Ricci *et al.* 2009). The idea at the basis of these works is to initially identify the parameters that may influence the test. A simplified model of the test item and of the shaker is thus considered under dynamic coupling. The vibration control system is also included in the modeling, by referring to an integrated computational environment. In these cases, the authors concluded that, by adequate upgrade of the controller, and acting on the parameters identified as critical for the dynamic coupling, it is possible to enhance the quality of the test and the performances of the facility.

More recently, two papers (Waimer *et al.* 2015a, Waimer *et al.* 2015b) were written about the experimental system identification of an electrodynamic shaker for VST, which is a topic that should be addressed in VST for reducing the error of modeling.

Di Pietro and Ladisa (2016) performed a VST study on a multiaxial shaker by referring to a dummy structure.

In the below sections the methodology followed during TAS on going VST activity is illustrated, together with some preliminary results.

## 2. Structural modelıng of the vst assembly

### 2.1 FE shaker and S/C models

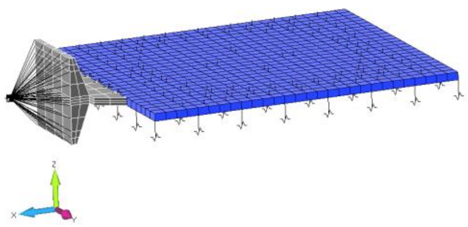


Fig. 1 The shaker moving parts

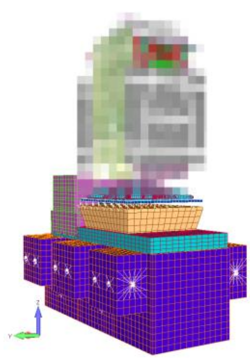


Fig. 2 Shaker + adapter + S/C assembly (lateral configuration)

The Finite Element Method (FEM) was employed in order to model the Shaker and S/C structures. The shaker is named Atlas and it is located in the facility of TAS-Cannes. This study is focused on Atlas lateral configuration, with excitation along the x-axis. The name of the S/C considered in this study will remain undisclosed and it will be hereafter called “the S/C”. The S/C weights slightly less than 4500 kg, having a center of gravity around 1.80 m. Figs. 1 and 2 illustrate the shaker moving parts and the complete VST assembly, respectively. Fig. 1 shows that the coil is modeled as a concentrated mass connected to the gripping jaw through a rigid element. Moreover, a number of rigid and springs elements are visible below the moving table and their function is to model the bearings’ stiffness along the directions perpendicular to the movement. In Fig. 2, several concentrated “large masses” FEs are placed at the tip of the suspension springs of shaker's seismic mass and they represent the rest of the world (clamped boundary condition for the frequency range of interest). No explicit clamp constraint is imposed to fix each one of these large masses since their substantial motionless is assured by their very large inertia.

### *2.2 FE Craig-Bampton reduction*

The VST assembly FE model consists of about 1.5E6 DOFs. It is understood that the model should be considerably reduced in order to make possible a VST run. A model reduction in Craig-Bampton format was thus performed by keeping the first 300 modes (being the frequency of the last mode close to 150 Hz) and considering 63 interface DOFs representing the coil, the pilots and some accelerometers that may take part in notching. This reduction, leading to a 365 DOFs model, was performed in MSC Nastran$^{TM}$ by using an internal TAS alter.

Fig. 3 provides the visual comparison of the number of FE nodes before (left) and after (right) the reduction. In the subfigure of the reduced model, the DOFs corresponding to the modes are not visible.

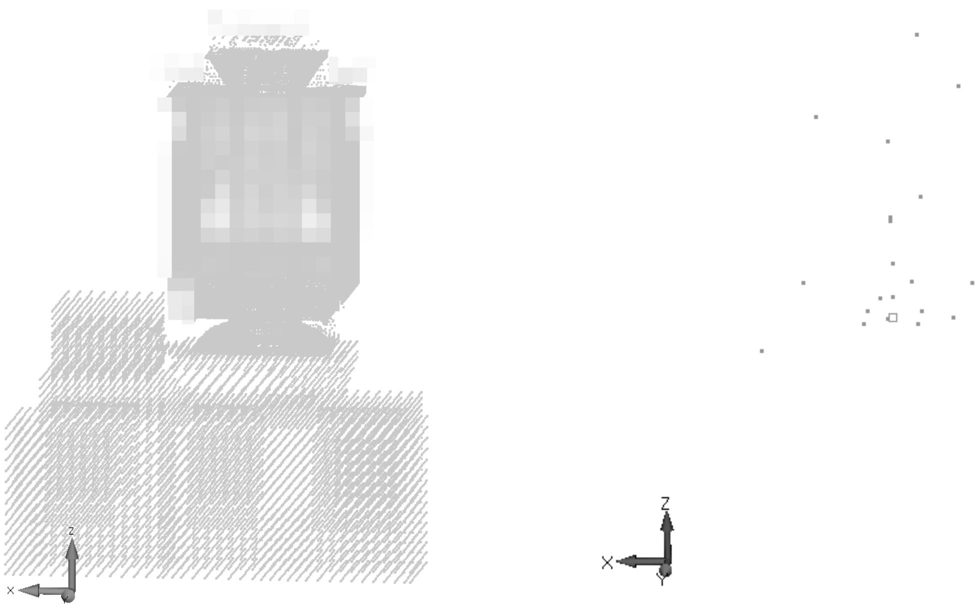



Fig. 3 VST assembly FE nodes before and after the reduction, from left to right

## 3. Computational methodology

A code was implemented in MATLAB® numerical computing environment in order to solve the transient problem of VST in the time domain. Here below is the list of the main capability of the code.

- Read the reduced stiffness and mass matrices written in OP4 format (output of MSC Nastran™ model reduction run).
- Generate the sweep forcing function according to a given starting frequency, sweep type (exponential or linear), sweep direction (up or down) and sweep rate.
- Impose the appropriate swept force to the coil and the corresponding reaction to the shaker body, taking into account the physical delay intrinsic of the electromechanical actuator.
- Solve the transient problem in the time domain, encompassing the swept frequency range of interest.
- Include the vibration control in the loop of the transient solution.
- Plot the results.

### 3.1 Numerical solution

The system to be solved is the following (Craig 1981)

$$\ddot{\boldsymbol{\eta}} = -\boldsymbol{C}_{\boldsymbol{gen}}\dot{\boldsymbol{\eta}} - \boldsymbol{K}_{\boldsymbol{gen}}\boldsymbol{\eta} + \boldsymbol{\Psi}^{\boldsymbol{T}}\boldsymbol{f}. \tag{1}$$

Bolt letters denote arrays, while the dot stands for differentiation with respect to time;

$\boldsymbol{\eta}$ is the vector of generalized displacements;

$\boldsymbol{\Psi}$ is the mass-normalized eigenvector matrix, being the mass matrix given by Craig-Bampton reduction of the VST assembly FE model;

$\boldsymbol{C}_{gen}$ is the modal damping diagonal matrix;

$\boldsymbol{K}_{gen} = \boldsymbol{\Psi}^T\boldsymbol{K}\boldsymbol{\Psi}$ is the generalized stiffness matrix, being $\boldsymbol{K}$ the stiffness matrix from the Craig-Bampton reduction of the VST assembly FE model;

$\boldsymbol{f}$ is the vector having the forces to be imposed (coil action and reaction).

The system in Eq. (1) is a $2^{nd}$ order Ordinary Differential Equation (ODE) system and it can be reduced to $1^{st}$ ODE system through standard algebraic manipulations. Thus, it can be solved numerically in the time domain through classical iterative methods. The $4^{th}$ order explicit Runge Kutta method is used in this work.

The amplitude of the sine sweep forcing function is updated at each oscillating period, according to the law of vibration control employed in the facility. The mechanical loading is thus imposed to the system with by setting a generic profile for pilots and the appropriate thresholds for notchers.

Finally, the physical accelerations vector $\ddot{\boldsymbol{a}}$ is given by

$$\ddot{\boldsymbol{a}} = \boldsymbol{\Psi}\ddot{\boldsymbol{\eta}}. \tag{2}$$

### 3.2 Limitations and assumptions of the methodology

The first hypothesis of the here presented computational methodology is the one of linear oscillations. Moreover, the use of modal damping (which simplify significantly the calculations, leading to a diagonal damping matrix) is one of the most significant limitation. In fact, it implies the assumption that all the VST assembly have the same damping ratio when a specific mode is considered.

It should be also mentioned that no physical modeling was introduced for the electromechanical phenomena implied by the coil's work. The force from the coil is imposed as pure mechanical loading, while the intrinsic delays of the electromechanical actuator are managed numerically by smoothing the application of the force variation requested by the vibration control. The amount and the type of smoothing was defined by the authors by correlating some test results with the numerical outputs.

## 4. Numerical results

The following results have to be intended as preliminary since the activity is currently in progress. Some numerical parameters related to the intrinsic delay of the electromechanical actuator are still under refinement. Moreover, there are some uncertainties on the damping values.In fact, knowing the damping of a S/C undergoing a VST analysis is not trivial. This is due to the fact that in the past, for convenience, the damping was evaluated to match the results from Frequency Response Analysis (steady-state) with the test responses (swept transient data). It follows that typically the damping values from heritage cannot be used in VST analysis without at least introducing the sine sweep rate effect correction (Lollock 2002, Roy and Girard 2012). Finally, in this case, the FE model of the shaker is not validated through test data, but it was built in accordance with the available geometrical, material, and component stiffness information. A few missing parameters were estimated through engineering judgment.

Some early comparison between VST numerical results and test results are provided through the following pictures, where the most interesting areas to be compared are circled.

The input profile set for the x-axis (lateral) qualification, incorporating some manual notching (ESA-ESTEC/ECSS 2013), is given in Fig. 4.

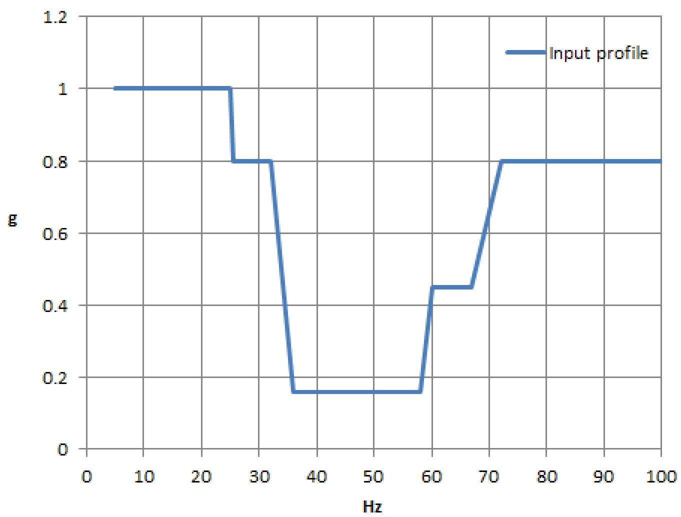


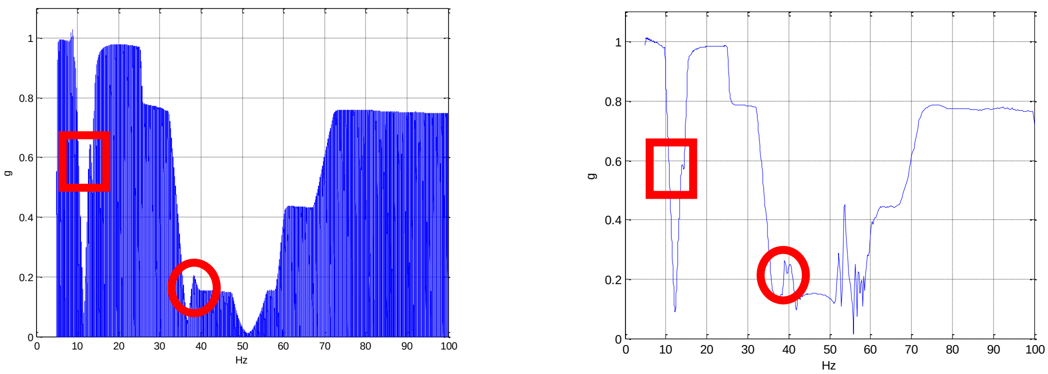

Fig. 4 Input profile for the qualification run (manual notching), x-axis

Fig. 5 shows the curves pertaining to a pilot which is located at S/C the base-ring.

Good agreement is confirmed between the VST prediction and the test data. More specifically, it should be emphasized the presence in both figures of some piloting deviancies strictly caused by transient phenomena which cannot be predicted through the conventional FRA sine test-prediction (circled areas). Fig. 6 provides a zoomed view of Fig. 5, with focus on the primary notch, where the beating due to the sine sweep rate effect (Nali and Bettacchioli 2016) is clearly noticeable. Fig. 7 illustrates the corresponding response of the accelerometer acting as notcher.

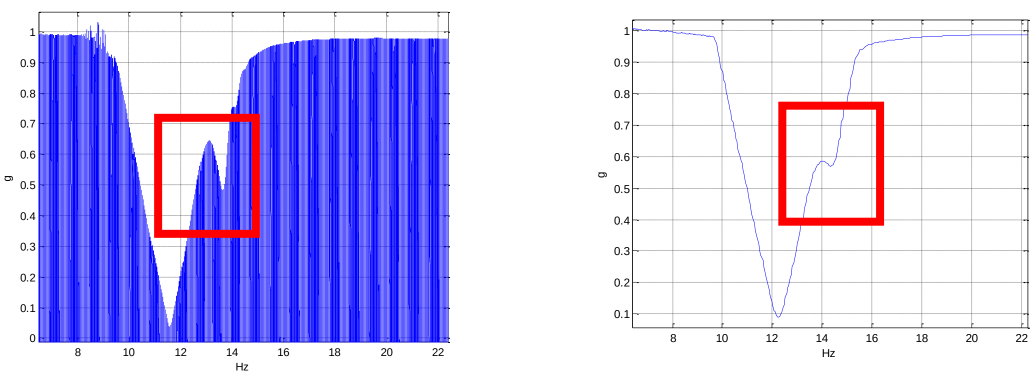
Fig. 8 is one more zoomed view of Fig. 5, with focus on another piloting deviancy unpredictable through FRA and occurring at higher frequency. Also, in this case, the VST prediction gives evidence of it. These transient behaviours typically comes together with significant disturbances of piloting. As a consequence of that, their proper prediction could help to tune the test parameters in order to mitigate the risks and save time (at the beginning of the test campaign, less low-level runs would be required for choosing the test parameters).

The plot comparison of cross-talks plots (given by the y and z-axis pilot acceleration) is not provided for sake of conciseness. It can be said that the maximal cross-talk acceleration is predicted by VST with satisfactory accuracy, even if the curves matching is not as good as for the excitation axis.



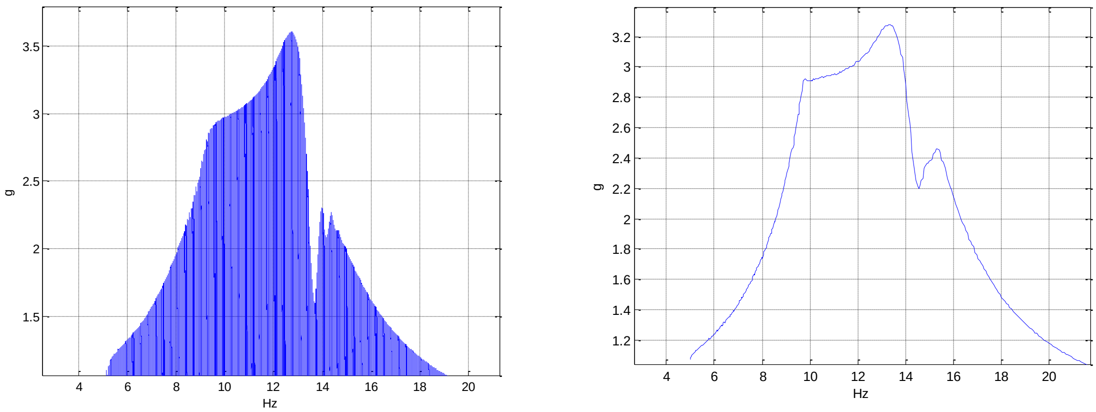

VST numerical simulation Qualification test of the S/C

Fig. 5 S/C pilot along the excitation (x-axis)



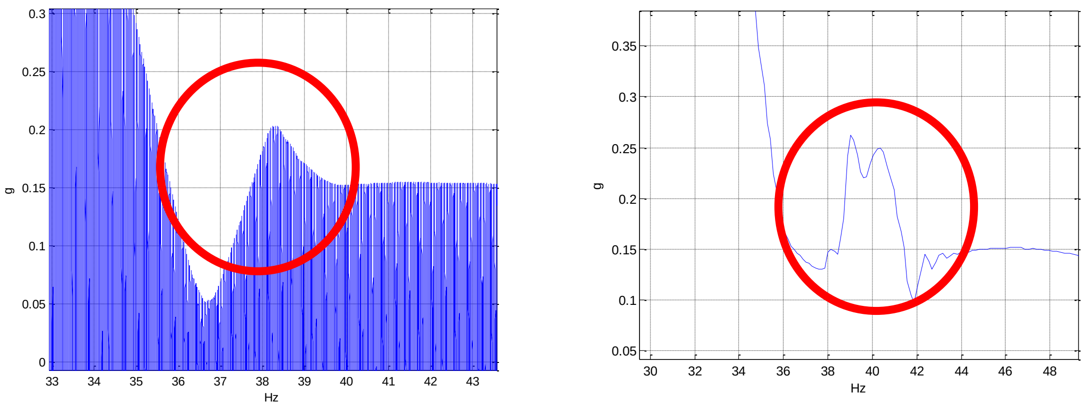

VST numerical simulation Qualification test of the S/C

Fig. 6 S/C pilot along the excitation (x-axis); zoom on primary notching




VST numerical simulation | Qualification test of the S/C

Fig. 7 Response of the accelerometer acting as notcher (x-axis); primary notching




VST numerical simulation | Qualification test of the S/C

Fig. 8 S/C pilot along the excitation (x-axis); zoomed view

## 5. Conclusions

The progress of a TAS activity aiming at exploring the benefits of the VST approach was illustrated in this work. VST numerical results show a good agreement with the test results. It is confirmed that the VST numerical sine test-prediction makes possible to anticipate before the test campaign some relevant issues due to the shaker-test article dynamic coupling (e.g., cross-talks) and to envisage the transient phenomena (e.g. the sine sweep rate effect). This could ease the choice of the test parameters before the test campaign, saving time and reducing the risks during the test. This is not feasible with the conventional sine test-prediction, which is performed through frequency response analysis and by imposing the hard-mounted boundary condition to the base of the S/C.

The use of structural damping instead of modal damping together with the further refinement of some parameters related to behaviour the electromechanical actuator might lead to increase of the accuracy of VST result.

## Acknowledgments

This work is supported by the European Space Agency, ESTEC under grant 4000115209/15/NL/LVH.

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