

Virtual shaker testing at V2i: measured-based shaker model and industrial test case

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Abstract

During high level vibration test on a heavy specimen, the test engineer is often facing difficulties to pass properly the specified vibration level due to coupling between the specimen under test and the electrodynamic shaker.

The present paper highlights the methodology followed to develop a virtual shaker testing simulator. The first step involves the dynamic identification of a 80 kN shaker performed thanks to measurements (modal analyses and sine sweeps). The second step is the definition of the physic included in the simulator and the translation of the electromechanical equations in a home-made simulator. Controller developed by SIEMENS LMS and supplied to V2i in the framework of the AOC research project (“Advanced Operational Certification”, Walloon Region funding) is introduced to close the loop. An industrial test case is described to demonstrate the ability of the simulator to deal with real complex structures.

1 Introduction

The feasibility to perform physical tests in the field of vibration testing cannot be entirely a priori assessed examining the shaker capabilities. Some couplings between the specimen under test and the shaker itself are well present and are emphasized in the case of high loads and/or high mass compared to the moving mass of the shaker. There exists a demand from the test supplier to foresee such behavior prior to the test to manage adaptations in the control strategy and/or to propose and justify level reduction.

Virtual shaker testing includes approaches to simulate the coupled behavior thanks to an electromechanical model of the shaker for which input voltage is assigned by a closed-loop controller. Previous work carried out on this topic [1-4] represents the shaker as a simple lumped-mass model and shows that such type of model is sufficient to represent accurately the coupled behavior of interest. Only the vertical degrees of freedom are taken into account in these studies. In [5] the torsion and in-plane rotation degrees of freedom are included to expand the shaker coordinate space. In fact, low damped torsion mode is observed by measurement and has to be incorporate in the model. The studied electrodynamic shaker (Ling 80kN) is installed in V2i premises. The performed shaker parameter tuning is based on measurements (hammer-impact modal analyses and shaker sine sweeps). The created simulator allows coupling either a reduced-order model of a specimen under test either a mass-spring equivalent system. The reduced-order model includes the dynamic behavior of the specimen by use of the Craig-Bampton reduction method [6]. The system is closed by a controller model developed by Siemens/LMS that mimic the behavior of the hardware

used to control the shaker. All the parts linked together allow the engineer predicting the response of a system prior to the test, foreseeing possible difficulties and searching strategies to cope with these problems (different control parameters, control accelerometer position, level reduction, etc).

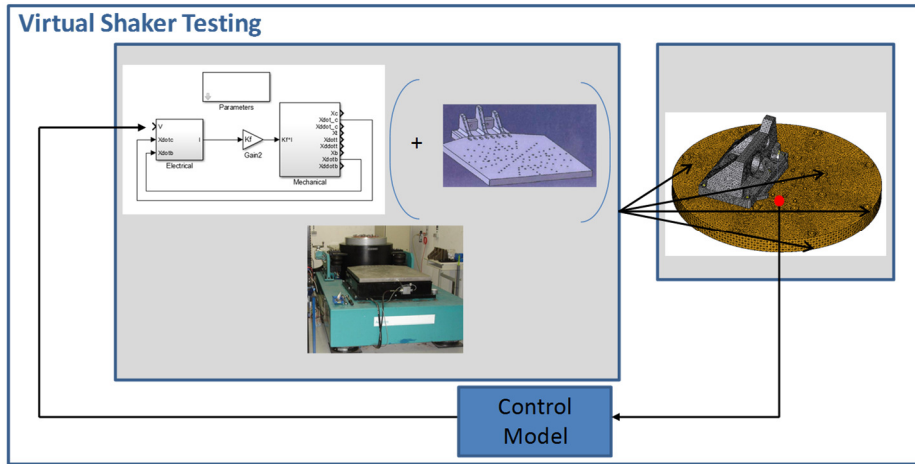


Figure 1: Virtual Shaker testing principle

Firstly, the article describes the methodology to define the coupled electromechanical shaker model. Modal characteristics and sine sweep responses are analyzed to update the lumped-mass model. The methodology followed to couple the specimen model to the shaker simulator is presented. Secondly an industrial test case demonstrates the capabilities of the simulator to deal with complex large model.

2 Virtual shaker testing simulator

2.1 Electromechanical model of the shaker

Similarly to the philosophy adopted in [1-4], a lumped-mass model is taken as basis of representation of the mechanical part of the electrodynamic shaker. The latter is coupled to a RL model of the electric part in such a way that the force acting on the coil and the induced back-electromotive force are correctly introduced. The model is shown in Figure 2.

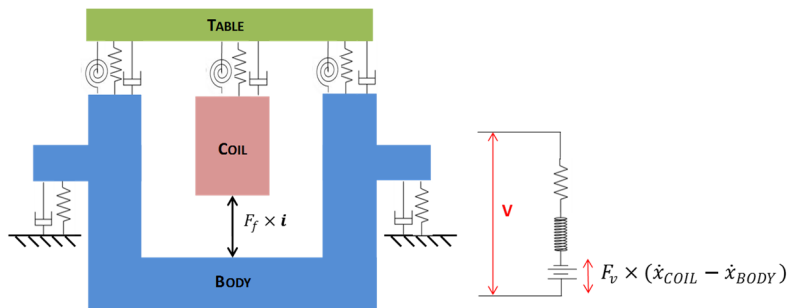


Figure 2: Electromechanical model of shaker

To the best author's knowledge, all the models found in the literature consider only the vertical degrees of freedom. From the experience gained by the V2i engineers, the information about transversal and torsional table behavior is important to know due to the presence of bending and torsional mode of the shaker top interface. The excitation of these modes can induce large transversal accelerations imposed to the specimen.

To be able to represent these phenomena, the torsion and the plane rotations are added in the set of degrees of freedom, which results in an mechanical model including not 3 degrees of freedom but 7, i.e.:

$$\mathbf{x} = [z_{Coil} \quad z_{table} \quad z_{body} \quad \theta_{z,table} \quad \theta_{x,table} \quad \theta_{y,table} \quad \theta_{z,Coil}]^T \quad (1)$$

The only degree of freedom of the electric model is the current i .

The electromechanical model is defined by the values of the stiffness $[\mathbf{K}]$, mass $[\mathbf{M}]$, damping $[\mathbf{C}]$ of the different mechanical parts, by the coil resistance R and inductance L and by the coupling terms F_f and F_v (if SI units are used, the force to current constant and the voltage to velocity constant are equal).

The resulting equation system is:

$$\begin{bmatrix} \mathbf{M} & \mathbf{0} \\ \mathbf{0} & 0 \end{bmatrix} \ddot{\mathbf{q}} + \begin{bmatrix} \mathbf{C} & \mathbf{0} \\ \mathbf{F}^T & L \end{bmatrix} \dot{\mathbf{q}} + \begin{bmatrix} \mathbf{K} & -\mathbf{F} \\ \mathbf{0} & R \end{bmatrix} \mathbf{q} = \begin{Bmatrix} 0 \\ V \end{Bmatrix} \quad (2)$$

Where:

$$\mathbf{q} = [\mathbf{x} \quad i]^T \quad (3)$$

$$\mathbf{F} = [F \quad 0 \quad -F \quad 0 \quad 0 \quad 0 \quad 0]^T. \quad (4)$$

During vibration tests on the studied shaker, the presence of torsional and in-plane rotation vibration modes are observed even if a priori only vertical loads are applied to the coil. An explanation can be found in a non-perfect alignment and centering of the coil axis with the structural direction. Non-diagonal terms are introduced in the mass matrix to introduce these effects:

$$M = \begin{bmatrix} m_c & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & m_t & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & m_b & 0 & 0 & 0 & 0 \\ -\frac{m_t \times e_{t,z}}{J_{z,t}} & 0 & 0 & (J_{z,t} + J_s) & 0 & 0 & 0 \\ -\frac{m_t \times e_{t,x}}{J_{x,t}} & 0 & 0 & 0 & J_{x,t} & 0 & 0 \\ -\frac{m_t \times e_{t,y}}{J_{y,t}} & 0 & 0 & 0 & 0 & J_{y,t} & 0 \\ -\frac{m_c \times e_{c,z}}{J_{z,c}} & 0 & 0 & 0 & 0 & 0 & J_{z,c} \end{bmatrix} \quad (5)$$

In order to simulate the shaker when positioned in horizontal configuration, a finite element model (Figure 3) is created considering :

- shell elements for the slip table;
- “spring-damper” local elements for the bearings (green elements in Figure 3);
- “spring” elements representing the sticking effect of the oil circulating below the slip table (green elements in Figure 3).

The model is then linked to the lumped mass model of the shaker (left part of Figure 3).

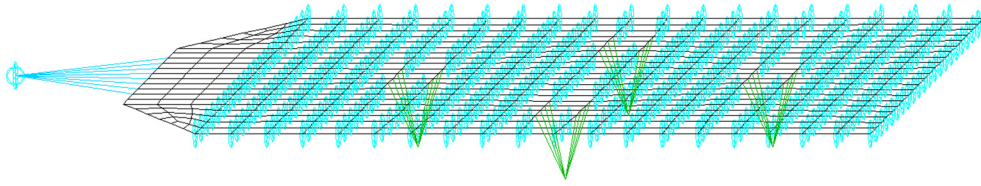


Figure 3: Finite element model of the shaker in horizontal configuration

2.2 Shaker identification

For both vertical and horizontal configurations, two sets of measurements were taken in order to collect reference data for the purpose of model updating:

- 1) Hammer impact testing with the shaker at rest (but with the oil circulating below the slip table): these measurements are taken in order to obtain an experimental modal model of the shaker. The tools available in the Test.Lab environment (Siemens LMS Software) [7] are used for this purpose.
- 2) Sine sweep 0.5 g at 1 Oct/min between 5 Hz and 2500 Hz (usual frequency range for an electrodynamic shaker): in complement of the modal analyses, these measurements allow characterizing the complete coupling between the different parts of the system (electrical, mechanical and the control one).

As stated in the introduction, the identification process is applied to the 80 kN shaker installed in the V2i premises but the methodology is relevant for all the electrodynamic shakers.

2.2.1 Vertical configuration shaker identification

The hammer impact testing analysis is performed during a maintenance of the shaker what gives access to the top part of the coil (Figure 4). The impact is imposed in different directions on the top part of the table.



Figure 4: Hammer impact testing on the shaker in vertical configuration

The Figure 5 to Figure 7 show the mode shapes and the modal characteristics of three identified modes. The two first are in-plane rotation of the top table and the third mode is a pure torsion of the top part. Having no possibility to position accelerometers on the bottom part of the coil, the typical coil mode (out-of-phase deformation between the coil and the table) cannot be retrieved from the data set. Some peaks are observed on the weighted sum of the Frequency Response Functions (Figure 8) at high frequency but the identified mode shapes have higher spatial frequency non-representable by the adopted discretization of the shaker model.

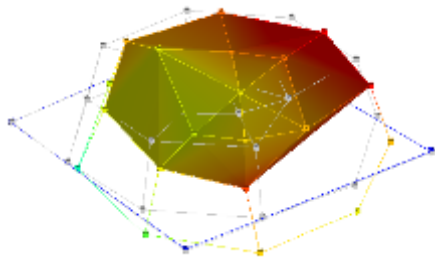


Figure 5: Vertical configuration impact testing
Frequency 102 Hz - damping 8%

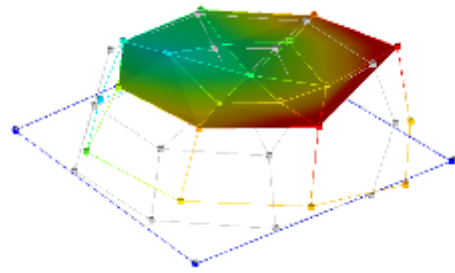


Figure 6: Vertical configuration impact testing
Frequency 111 Hz - damping 8%

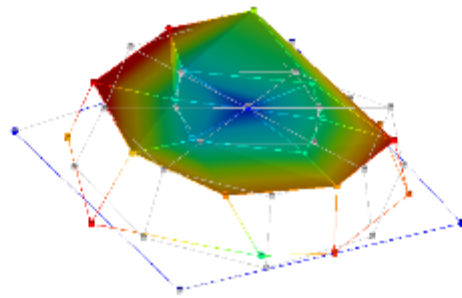


Figure 7: Vertical configuration impact testing - Frequency 434 Hz - damping 0.8%

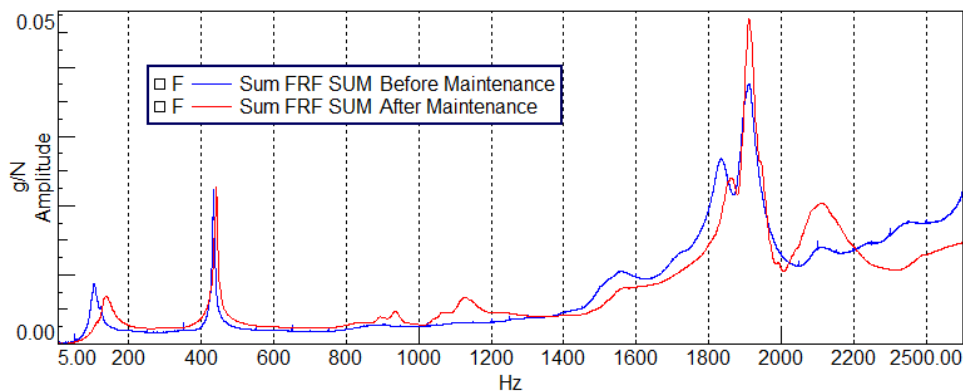


Figure 8: Vertical configuration impact testing - Weighted sum of the frequency response functions

The modal analysis has been performed twice: before and after the maintenance (with dismounting and remounting of the moving parts). The two measurements show frequency shifts of the in-plane and the torsion mode without any change in the peak sharpness what shows the sensitivity of the modal features with respect to the coil mounting (Figure 8).

The signature of the torsion mode is also clearly identifiable on the sine sweep responses (Figure 9), which demonstrates the activation of the mode even if, a priori, no vertical modal mass is present in the identified torsion mode. A second torsion mode at around 900 Hz with a higher level of damping is observed in the acceleration spectrum what is non-clearly identified by the modal analyses. The activation of the static electromagnetic field influences the modal content mainly in slightly modifying the resonance frequency values.

The signature of the coil mode is detected on the “table acceleration to drive voltage” FRF (see Figure 16) (high acceleration at the top part for a low input voltage).

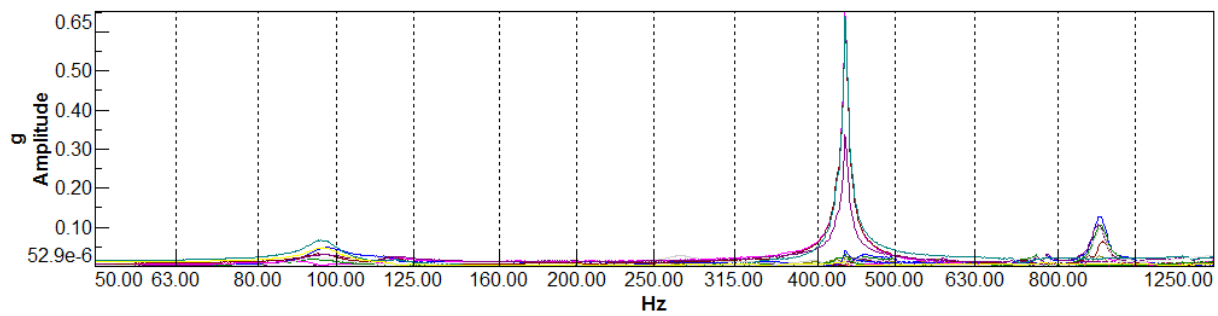


Figure 9: Vertical configuration sine sweep – Accelerations on sensors placed on the shaker top table

2.2.2 Horizontal configuration shaker identification

The measurement points for the hammer impact on the shaker in horizontal configuration are chosen to obtain a sufficient spatial discretization of the slip table. The system is impacted at the lower right part in the three structural directions as shown in Figure 10. A set of modal characteristics is identified. Three among the identified mode shapes are shown in Figure 11 to Figure 13. However, as confirmed by the sine sweep acceleration spectra, the most interesting mode is the pumping mode observed at 886 Hz (for the shaker at rest – see Figure 13) for which a node of vibration is present just where the slip table is connected to the shaker.

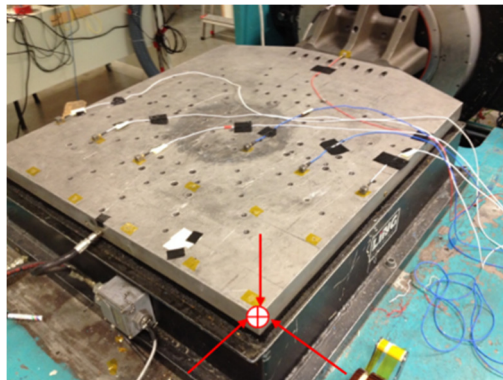


Figure 10: Horizontal configuration - Hammer impact testing setup

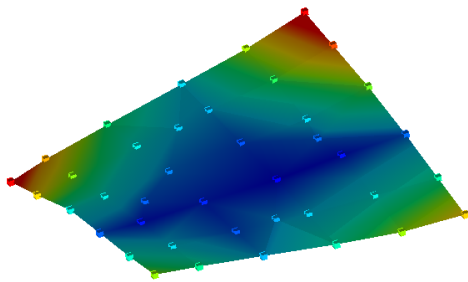


Figure 11: Horizontal configuration impact testing
Frequency 528 Hz - damping 1.9%

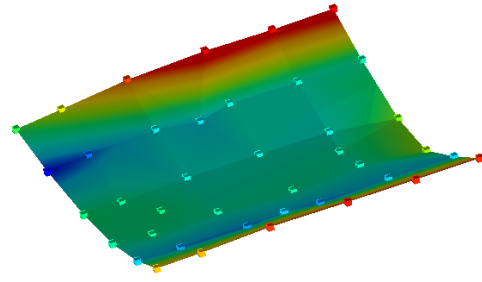


Figure 12: Horizontal configuration impact testing
Frequency 691 Hz - damping 1.5%

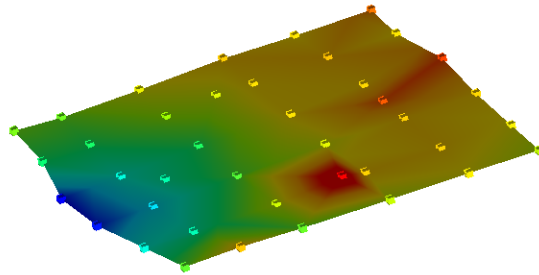


Figure 13: Horizontal configuration impact testing Frequency 886 Hz - damping 1.0%

Figure 14 shows the shaker table acceleration to drive voltage Frequency Response Function when a constant acceleration is imposed on a point located in the slip table center. Due to the effect of the electromagnetic field on the system and the fact that the excitation level is higher, the pumping mode is observed at a lower frequency (735 Hz) in comparison to the modal features of the shaker at rest (886 Hz). The FRF curve allows detecting an additional resonance at around 920 Hz which mode shape can be an out-of-phase movement between the moving part of the shaker and the slip table (non-observable).

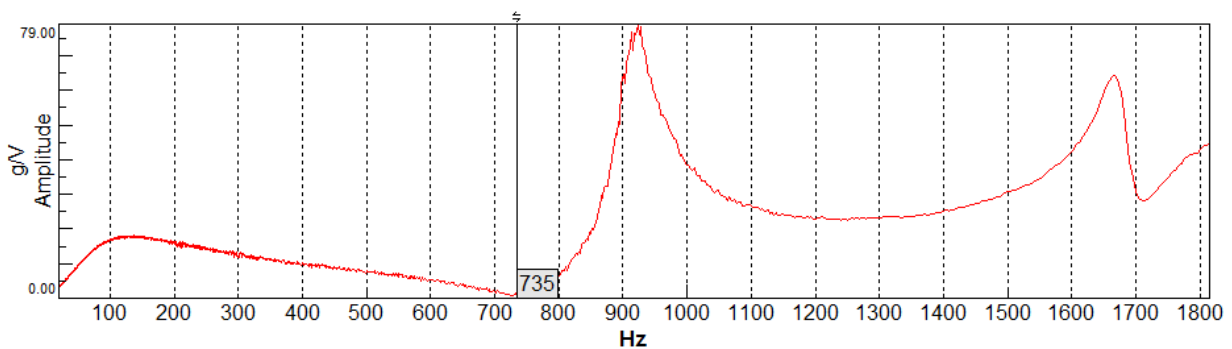


Figure 14: Horizontal configuration sine sweep: Shaker table acceleration (point located at the junction between the slip table and the shaker table) to drive voltage FRF

2.3 Shaker model updating

Starting from guessed values assessed thanks to information found in the shaker data sheets and from geometric measurements, the model updating process is mainly conducted by manual sensitivity analyses. The objective is :

- To minimize the least-square difference between the measured and simulated frequency responses functions;

- To minimize the difference between the measured and simulated modal characteristics (resonance frequencies and mode shapes).

Figure 15 shows the modal shapes computed on the updated lumped mass model of the shaker. The first mode is the “isolation mode” (rigid body mode of the shaker with respect to the mounting). The second mode is the “suspension mode” characterized by a relative movement between the body and the moving part (coil and table). The third and the fourth mode shapes are characterized by an in-plane rotation of the top table (orthogonal modes). The fifth mode shape is an out-of-phase torsion motion between the table and the coil. In the last mode, the coil moves out-of-phase with the table. In comparison with the modal characteristics highlighted by the measurements, the agreement is very good (frequency, shape and damping).

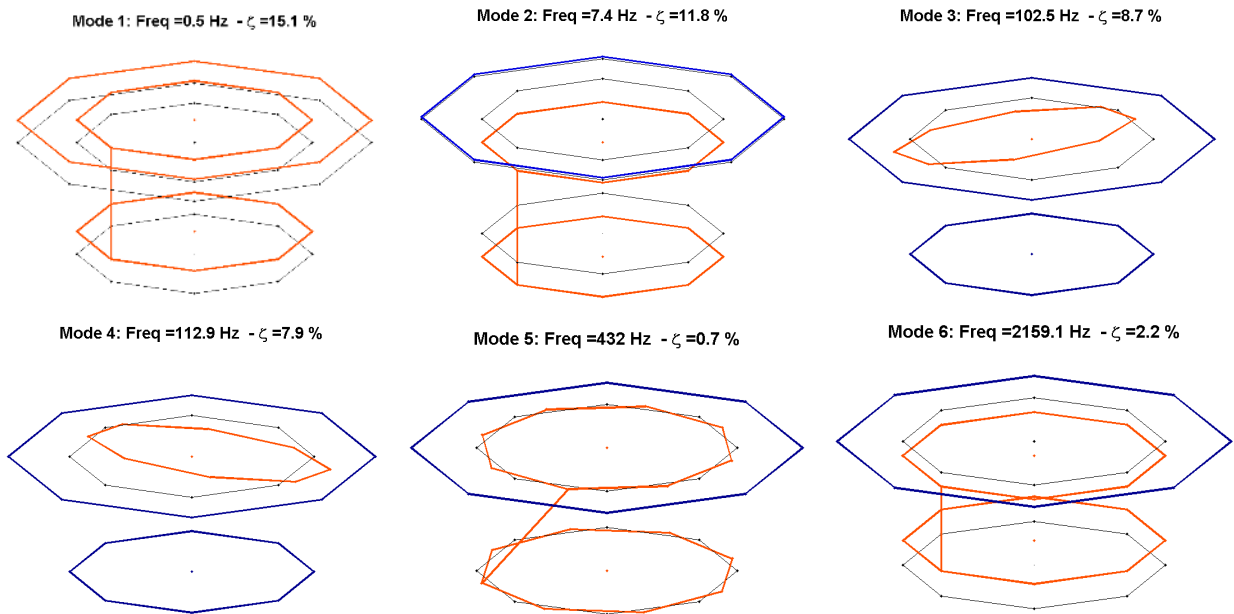


Figure 15: Updated shaker lumped mass model – Mode Shapes – In black the non-deformed structure composed of the coil, the table and the body and the deformations colored according to the normed amplitude (blue zero amplitude to red for maximal amplitude)

Figure 16 and Figure 17 superpose the Frequency Response Functions coming from the measurements and the ones computed with the updated electromechanical model. The quality of the representation of the electromechanical coupling is then demonstrated.

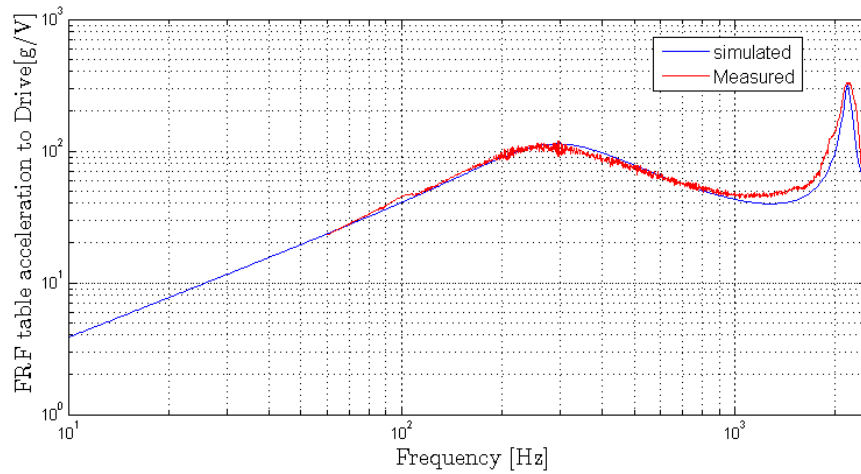


Figure 16: Comparison between measured and simulated table vertical acceleration to drive voltage Frequency Response Function

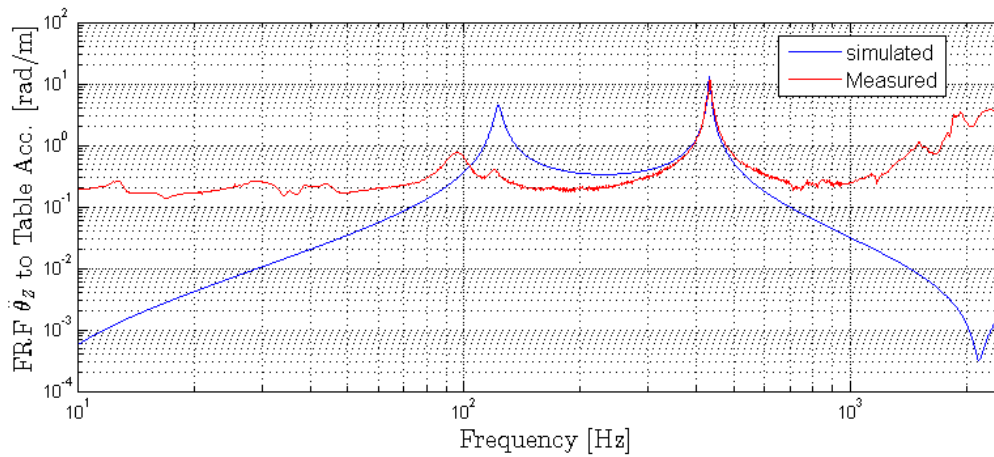
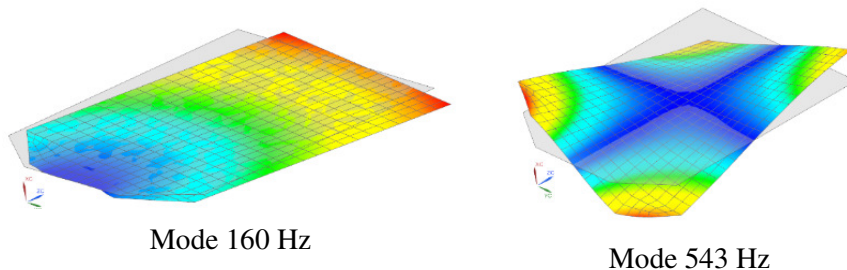


Figure 17: Comparison between measured and simulated table acceleration (rotation around the vertical axis) to table vertical acceleration Frequency Response Function

Figure 18 shows some of the updated mode shapes of the shaker in horizontal configuration. The Modal Assurance Criterion comparing the experimental and numeric mode shapes is given in Figure 19. The correlation is not perfect for all the modes. However, only the modes with high specific mass along the slipping direction are important for the purpose of coupling phenomena representation, which is achieved with the good correlation for the pumping mode (MAC of 0.96). Figure 20 demonstrates the good agreement between the measured and computed Frequency Response Functions.



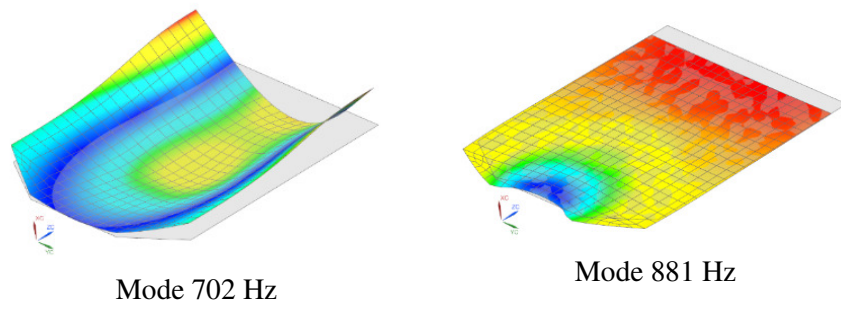


Figure 18: Updated slip table model – Significant Mode Shapes

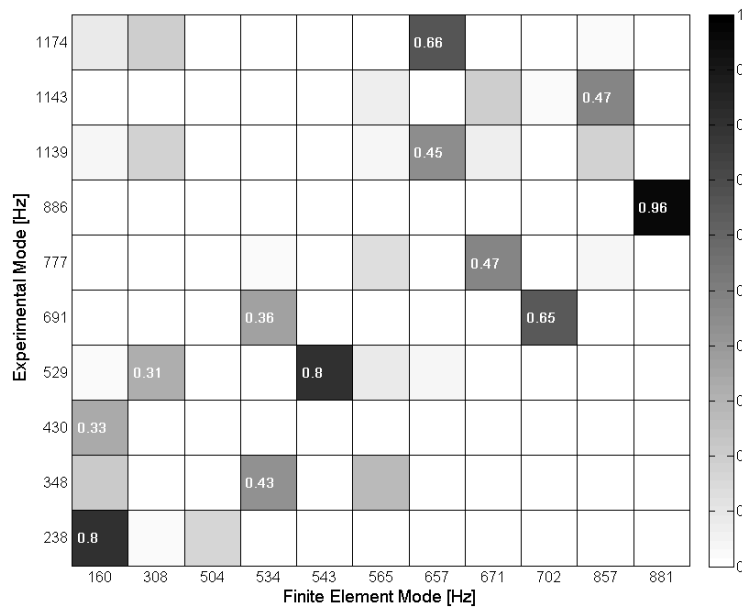


Figure 19: Modal Assurance Criterion between the experimental and numeric mode shapes of the shaker in horizontal configuration

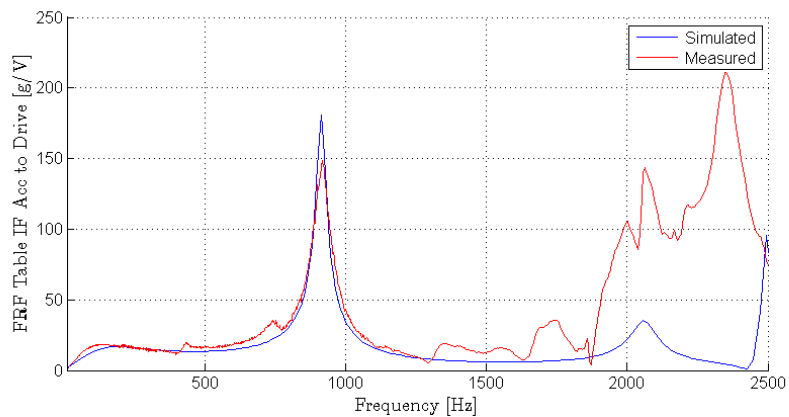


Figure 20: Superposition of the measured and simulated FRF of the acceleration in a central point to the Drive Voltage

2.4 Model of the controller

A controller model that mimics the LMS hardware behavior was developed by SIEMENS LMS and supplied in the framework of AOC research project. The controller described in [2] allows imposing a sine sweep up to 400 Hz. The controller behavior can be tuned thanks to parameters such as the compression factor and the sine sweep rate that are available in the Test.Lab interface.

The controller gives the drive voltage level to inject in the electromechanical system in order to follow as close as possible the reference signal.

2.5 Specimen integration

The previously described parts (updated electromechanical part and the controller model) are coupled and assembled in a unique environment. The customer test specifications impose controlling the injected acceleration level either on the shaker table either on a specific location on the specimen under test. A model of the specimen itself has to be incorporated in the simulator to consider the complete coupling.

Two options are developed: a one degree of freedom mass-spring system or a Craig-Bampton reduced-order model [8].

- **One degree of freedom model:** The mass, the stiffness between the concentrated mass and the shaker table and the modal damping percentage have to be defined. This simple model is suitable when the resonance frequencies are well separated and when the analysis is focused on a simple assessment on a small frequency range.
- **Reduced-order model:** A detailed finite element model of the specimen has to be realized and possibly updated if experimental data are available. The reduced-order model is computed in retaining the vertical and rotation degrees of freedom on an interface node. In the framework of this research, the Samcef software [8] is used for the specimen finite element creation and for the reduced-order model computation. The reduced mass and stiffness matrices are as follows:

$$M_{SE} = \begin{bmatrix} M_{FF} & M_{FI} \\ M_{FI} & M_{II} \end{bmatrix} \quad (5)$$

$$K_{SE} = \begin{bmatrix} K_{FF} & 0 \\ 0 & K_{II} \end{bmatrix} \quad (6)$$

The transformation matrix for selected points is determined to retrieve the responses at these points at any time from the reduced coordinates. The damping is considered in the reduced modal space as follows:

$$C_{II} = \text{diag}(\varepsilon_i \times 2 \times \omega_i \times \mu_i) \quad (7)$$

The specimen reduced matrices are transformed into state-space model for computational time reasons.

The reacting loads between the specimen and the shaker are computed and considered to couple the two parts together. The reacting loads are defined as output of the state-space model as follows:

$$F_{reac} = \underbrace{\left\{ M_{FI} \times \left[-\frac{K_{II}}{M_{II}} \quad -\frac{C_{II}}{M_{II}} \right] \right\}}_C \times \underbrace{\begin{Bmatrix} p \\ \dot{p} \end{Bmatrix}}_D + \underbrace{\left\{ M_{FI} \times \frac{M_{FI}}{M_{II}} \right\}}_D \times \ddot{q}_F \quad (8)$$

With p, \dot{p} , the reduced modal coordinates.

A Dormand-Prince time integration scheme is used to resolve the system.

A Graphical User Interface allows managing all the processes of the virtual shaker testing.

3 Test case

Two academic test cases are presented in [5] demonstrating the performances of the simulator to deal with small simple structures. These examples show that the simulator is able to predict some pathological cases such as when the control is located on a vibration node.

The application of the virtual shaker testing process on an industrial case is described in the present paper. The considered structure is a luminaire named “Omniblast” commercialized by the Schreder group (Figure 21). In the framework of the mechanical certification, the luminaire has to pass some vibration tests to demonstrate the strength of the assembly against loads such as the ones coming from the wind amplified by the mounting shaft dynamics. The highly accelerated life test consists in imposing a given level of acceleration (according to the considered norm) at a frequency corresponding to its first resonance frequency during a given number of cycles.

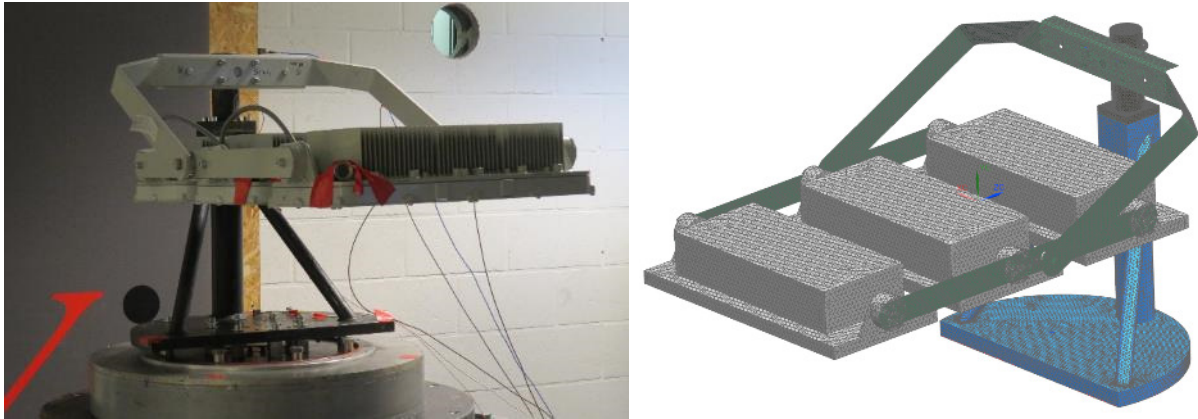


Figure 21: Omniblast luminaire (Schreder): left - installed on the 80 kN shaker and right – finite element model (number of degrees of freedom: 650 000)

The physical tests were conducted in the V2i premises. The analysis of the recorded acceleration responses allows determining the main modal characteristics such the resonance frequency and the estimation of the modal damping thanks to the circle fitting method. The finite element model of the idealized luminaire is created in the Siemens NX environment and the reduced-order model including the first 7 modes and retained nodes at the bottom of the support are build. The transformation matrix is computed at points where accelerometers are placed during the physical vibration test, which allows visualizing the behavior of the structure and performing comparisons. These additional nodes can also be considered as the control point. Part of the dynamic of the structure is then introduced in the control loop. In order to achieve a reasonable correlation between the numeric model and the recorded measurements, special care is taken in modeling the assembly between the different parts of the structure.

The considered norm (published by the IEC institute) for this particular test prescribes to impose a constant 0.5g acceleration applied to the connection of the luminaire with the pole shaft (support in black in Figure 21). Previously and just after the endurance, a sine sweep at 1 Oct/min is performed first to identity the most relevant resonance frequencies and second to judge the good health of the structure (comparison between the frequency signature before and after test).

Figure 22 and Figure 23 compare the measured and simulated signals (drive voltage and acceleration on a point located on the luminaire fork). The agreement is quite accurate. It has to be noted that the availability of measure data prior to the simulation helps to a good prediction of the dynamic amplification.

The time spent to simulate the sine sweep between 3 and 50 Hz is around 580 s (Intel i7 @2,50GHz,16 Go RAM) to be compared to a test duration of around 240 s.

The simulation considers the whole dynamic as being linear. The effect of nonlinearities such as some contacts, nonlinear stiffnesses or large deformations cannot be foreseen. Clear nonlinearity is observed in the acceleration spectra at around 40 Hz (non-symmetry of the spectra around the resonance that can be explained by the presence of a softening effect in the system), Figure 23 . This phenomenon is completely neglected in the simulated responses.

No difficulties are expected for these kinds of vibration tests. The feasibility is well assessed prior to the test thanks to simple formulas (given the maximal force to be compared to the shaker maximal reachable load). This test case is conducted in order to demonstrate the capability of the virtual shaker simulator to deal with complex real structure.

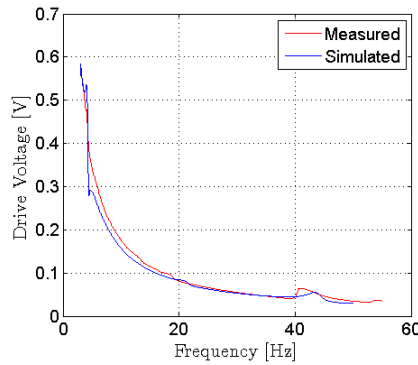


Figure 22: Comparison between the measured and simulated Drive Voltage

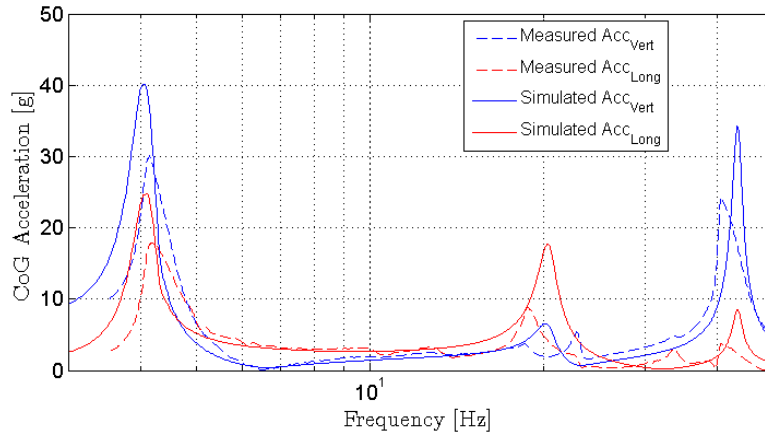


Figure 23: Comparison between the measured and simulated accelerations at the luminaire center of gravity

4 Conclusions

Thanks to the introduction of additional degrees of freedom, the developed virtual shaker testing tool allows predicting the dynamic behavior of the coupled shaker-specimen assembly (translations as well as rotations). Difficulties can be detected and solutions can be tested in a numerical way. The risks of non-appropriate vibration test imposed to the specimen can be lowered.

The present work focuses on the 80 kN shaker installed at V2i premises but the updating methodology from experimental measurements can be extended to all types of electrodynamic shakers. At this development stage, only sine sweep test is available.

In further work, additional features will be added such as the control strategy (average and maximal) on multiple points of the specimen.

Acknowledgements

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