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Short communication

Measured-based shaker model to virtually simulate vibration sine test

Sébastien Hoffait[a,*], Frédéric Marin[a], Daniel Simon[a], Bart Peeters[b], Jean-Claude Golinval[c]

[a] V2i, Belgium
[b] Siemens PLM Software, Belgium
[c] LTAS Ulg, Belgium

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ABSTRACT

During high level vibration test on a high mass specimen, the test engineer is often facing difficulty to pass properly the specified vibration level due to coupling between the specimen and the shaker. The present paper present a methodology to define a virtual shaker testing simulator. The first step involves the dynamic identification of a 80 kN shaker performed thanks to measurements (modal analysis and sine sweep). The second step is the definition of the physic represented in the simulator and the translation of the electromechanical equations in a home-made simulator. Controller developed by SIEMENS LMS and supplied to V2i for a use in the framework of the AOC project is introduced to close the loop. Two test cases are described to demonstrate the possibilities offered by the simulator.

1. Introduction

The feasibility to perform physical tests in the field of vibration testing cannot be entirely a priori assessed examining the shaker capability. In case of high load (injected level or mass), the test engineer can have to deal with couplings between the tested specimen and the shaker. There exists a demand from the test supplier to foresee such behavior prior to the test. By their contribution in several research [1], the space industry has shown its interest in order to reduce the risks during the qualification test campaign. The final goal is to give tools to manage adaptation in the control strategy and/or to justify levels reduction to protect both the shaker and the tested specimen.

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. Refs. [2–4] present shaker modeling considering simple lumped-mass model. Such type of model is refined enough to represent a part of the coupling. Only the vertical degrees of freedom are considered in these studies and the transversal effects are neglected. The present work proposes to include the torsion and rotational degrees of freedom of the top part of the armature to complete the shaker representation. The measures performed on a 80 kN electrodynamic demonstrate the need to introduce additional degrees of freedom. In fact, low damped torsion mode is observed that can induce very large non-desired transverse accelerations.

* Corresponding author.
E-mail addresses: s.hoffait@v2i.be (S. Hoffait), f.marin@v2i.be (F. Marin), d.simon@v2i.be (D. Simon), bart.peeters@siemens.com (B. Peeters), jc.golinval@ulg.ac.be (J.-C. Golinval).

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The present paper describes first the methodology to define the coupled electromechanical shaker model. Modal characteristics and sine sweep responses are analyzed to update the lumped-mass model. The controller introduced in the simulator is briefly described. The methodology followed to couple the specimen model to the shaker simulator is presented.

Secondly the Graphical User Interface functionalities are listed and two test cases, one of which is compared to measurements, are described.

The paper ends with some discussions about the possibilities offered by the simulator and the perspectives of improvement.

2. Electromechanical model of the shaker

Similarly to the philosophy adopted in Ref. [2–4], a lumped-mass model is considered to represent the mechanical part of the 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 Fig. 1

To the best of the author's knowledge, only the mechanical vertical degrees of freedom are considered in the model presented in the literature. In the present paper, the torsion and the plane rotations are added in the set of degrees of freedom, which results in a mechanical model including 7 degrees of freedom, i.e.:

$$\boldsymbol{x} = \begin{bmatrix} z_{Coil} & z_{table} & z_{body} & \theta_{z,table} & \theta_{x,table} & \theta_{y,table} & \theta_{z,Coil} \end{bmatrix}^{\boldsymbol{T}}$$

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

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

The resulting equation system is:

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

where $\boldsymbol{q} = [\boldsymbol{x} \quad i]^{\boldsymbol{T}}, \boldsymbol{F} = [F \quad 0 \quad -F \quad F_\theta \quad 0 \quad 0 \quad 0]^T$.

Note that a coupling term F_θ applied between the coil torsion degree of freedom and the current is necessary in order to achieve a good correlation of the model with the observed torsion behavior of the electrodynamic shaker.

In order to simulate the shaker when positioned in its horizontal configuration, a finite element shell model of the slip table is linked to the electromechanical model of the shaker. The bearings are represented by "spring-damper" elements and the oil film interaction is taken into account thanks to local spring elements.

3. Shaker identification

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

1) Impact testing in order to perform modal analyses of the shaker when at rest. The Least Square Complex Exponential method implemented in the LMS software [6] was used for this purpose.

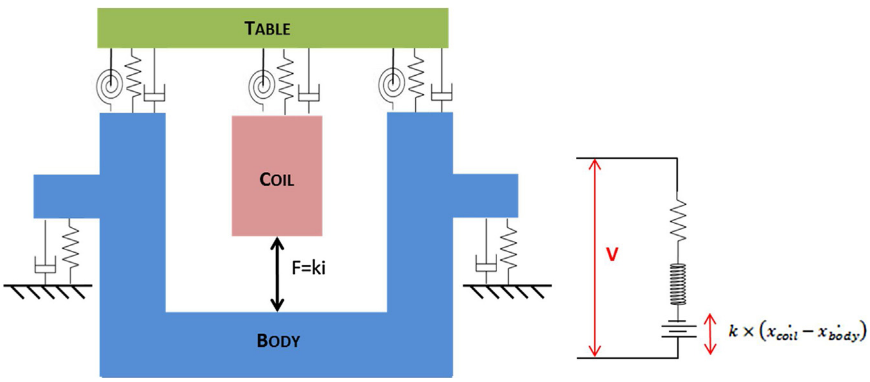

Fig. 1. Electro-mechanical model of shaker.

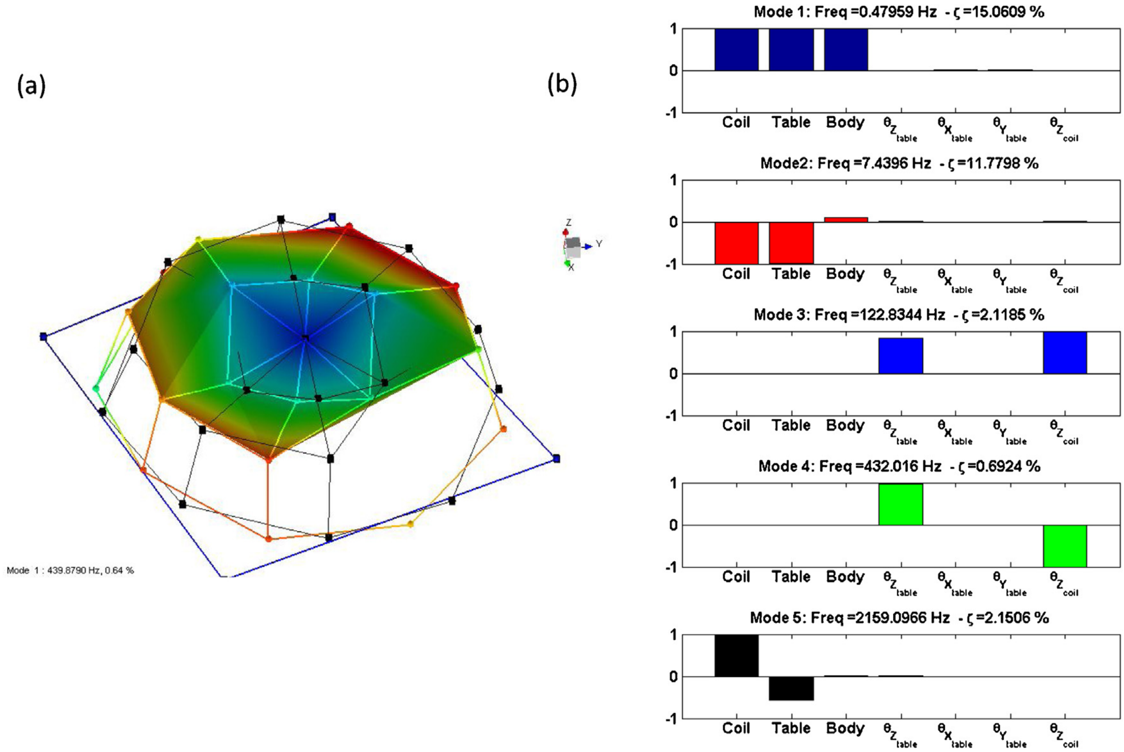

Fig. 2. Shaker (vertical configuration) modal characteristics: (a) Experimental torsion mode of the shaker table (the top part rotates while the shaker body stay non deformed) and (b) mode shape of the shaker model (the normed modal deformation amplitudes are represented for each degree of freedom and for each mode).

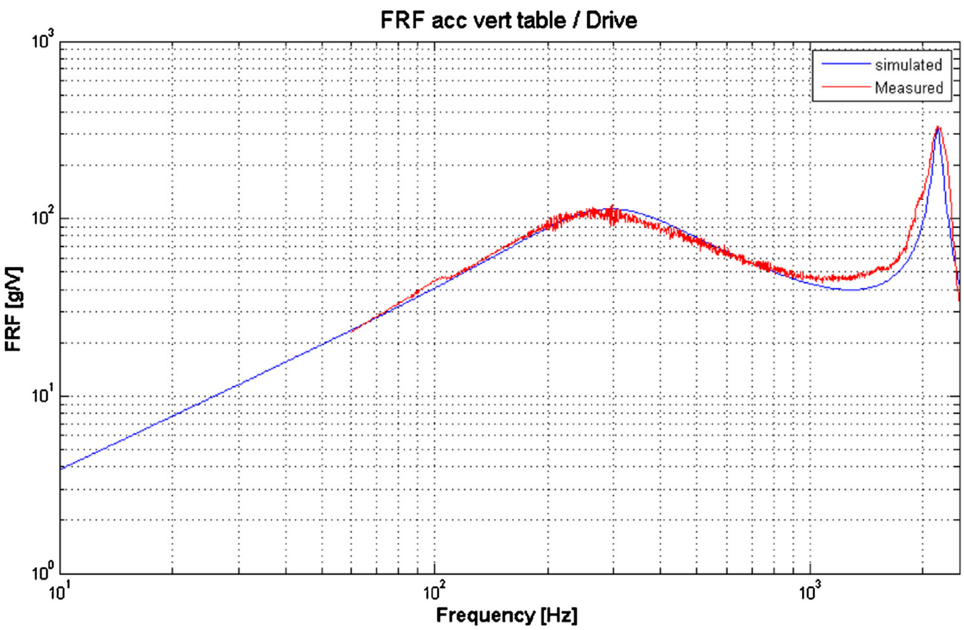

Fig. 3. Acceleration on drive signal frequency response function comparison.

2) Sine sweep between 5 and 2500 Hz in order to characterize the electric signal and the electromechanical coupling.

The major results of this measurement campaign are listed here below:

- A low-damped torsion mode at around 440 Hz is determined (vertical configuration—Fig. 2(a)). The modal deformation is characterized by a rotation of the entire top part of the table. The signature of this mode is also clearly identifiable on the sine sweep responses; what demonstrated the possible activation of the mode even if, a priori, no vertical modal mass is present in the observed torsion mode.
- The coil mode (at 2160 Hz) is not identified by the modal analysis due to the impossibility to place accelerometer on the bottom part of the coil. However its signature is present on the “table acceleration to drive voltage” frequency response function.
- A problematic pumping mode of the slip table (horizontal configuration) is identified at 890 Hz. This mode is characterized by a node of vibration located at the junction between the slip table and the shaker.
- Sine sweeps are performed for different gain value of the shaker amplifier in order to identify the gain parameter to consider in the simulator control loop.

4. Shaker model updating

Starting from initial parameter values assessed (shaker data sheet and geometric measurements), model identification is performed first by manual sensibility analyses and secondly by using optimization methods to minimize the least-square difference between the measured and computed frequency response functions. Updating of the shaker in horizontal configuration focuses on the correlation of the pumping mode.

In order to allow torsion and in plane rotation modes to be activated, some perturbation terms are introduced in the mass matrix.

Fig. 2(b) shows the shaker modal characteristics in vertical configuration (frequencies, modal damping and normed mode shapes; the amplitude of the mode shape for each of the 7° of freedom is presented). The torsion mode in the model is at 432 Hz. Two torsion modes (“in phase” at 123 Hz and “out of phase” at 432 Hz) are observed as shown in Fig. 2 but are not detectable by the measurements.

Fig. 3 demonstrates the quality of the model updating by a superposition of the “table acceleration to drive voltage” frequency response functions measured and simulated with the virtual shaker.

The pumping mode is correlated with a Modal Assurance Criterion (MAC—[5]) of 95% between the measured and simulated mode shape vectors. The other slip table modes are not correlated with the same level of quality. Moreover only the modes with a high specific mass along the slip direction are important for the purpose of representing the coupling phenomena.

5. Model of the controller

A controller model that mimics the LMS hardware behavior was developed by SIEMENS LMS and supply for an use in the framework of a research project. The controller described in [2] allows imposing a sine sweep up to 400 Hz. A set of control parameters similar to the ones to be defined during a vibration test (compression factor, sine sweep rate) is parameterizable.

6. Specimen coupling

The created virtual shaker simulator allows coupling either one degree of freedom mass-spring system either a Craig-Bampton reduced order model [8] to the shaker model.

- One degree of freedom model: The mass, the stiffness between the concentrated mass and the shaker table and the damping percentage has to be defined. This simple model is suitable when the resonance frequencies are well separated and that the analysis is focused on a particular mode shape response.
- Reduced order model of a specimen: A detailed finite element model of the specimen has to be realized and a super-element is computed by retaining the vertical and rotation degrees of freedom of an interface node. The Samcef software [7] is used for the specimen finite element creation and for the reduced order model computation. The transformation matrix for selected points is determined to retrieved the responses at theses points at any time from the reduced degrees of freedom. When horizontal configuration is considered, the finite element model of the specimen is connected to the slip table finite element model by classical connection method (rigid connections between nodes). The reduced order model of the assembly is then computed.

All the mathematical models (shaker, specimen and controller) are assembled in an unique environment. The reacting loads between the specimen and the shaker are computed and considered to couple the two parts. The acceleration defined as the controlled signal is injected as input in the controller box and determined the drive voltage to ensure that the requested level is reached.

A Dormand-Prince time integration scheme [9] is used to resolve the system.

7. Application examples

7.1. Graphical user interface

A graphical user interface is developed to manage all the processes of virtual shaker testing. The interface allows the users to define the control parameters, to import super-element or define mass-spring system, to assign damping ratio to modes, to define the control strategy on the table or one node of the super-element. Some functionalities for results visualization and saving are also available.

7.2. Test Case 1: beam on shaker in vertical configuration

An horizontal steel beam fixed on the shaker table in its vertical configuration is considered (Fig. 4(a)). Sine sweep is performed imposing the controlled signal either at the beam basis either at a middle point of the beam (0.1 g imposed between 10 and 400 Hz, sine sweep rate of 2 Oct/min).

The measured frequency response functions are used to assess the damping ratio of the first modes using the half-power method [10]. The finite element model of the beam is created. A super-element including four internal modes are computed with the vertical and torsion degrees of freedom of the beam basis as retained degrees of freedom.

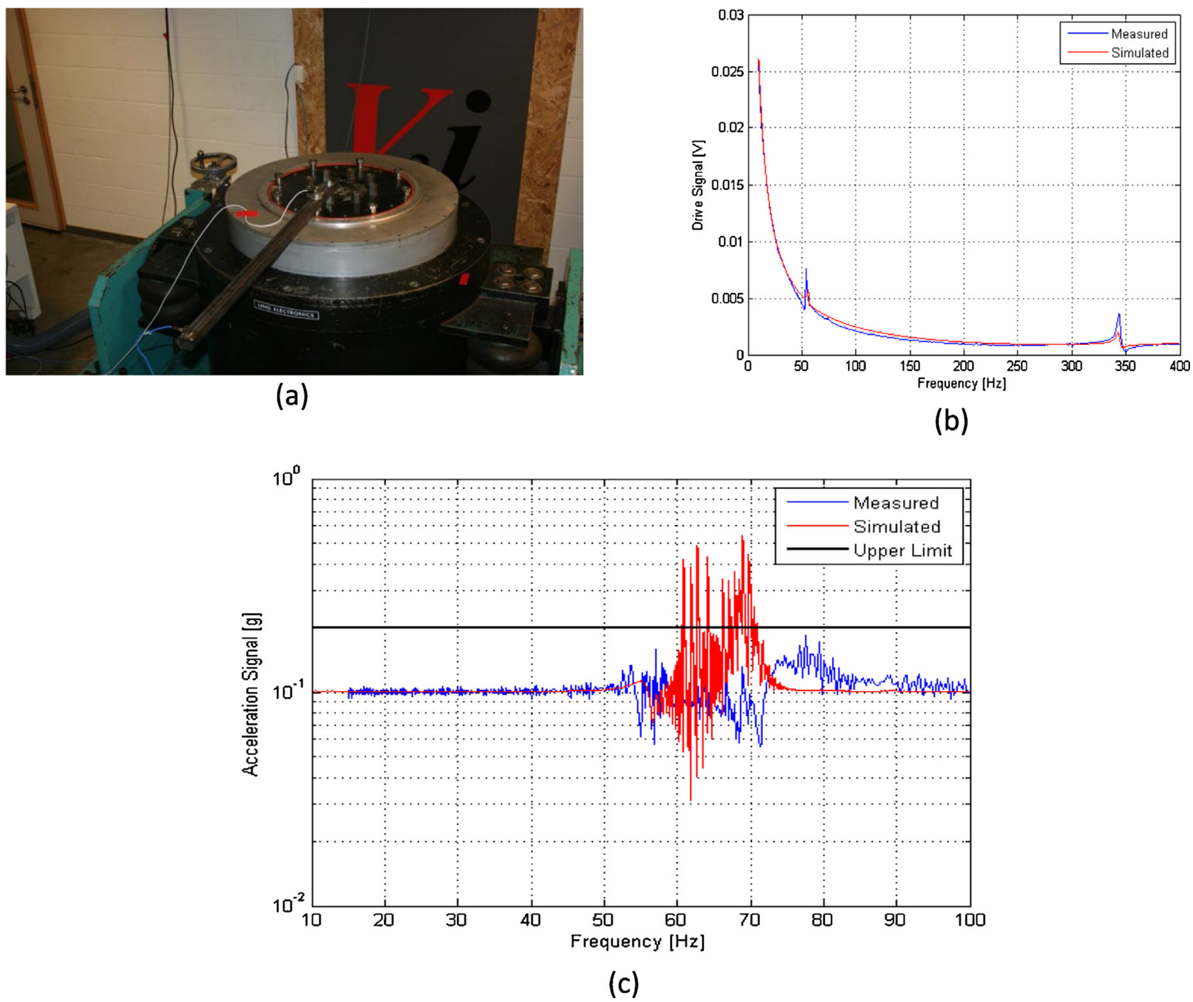

Fig. 4. Test Case 1: (a) Set-up—Steel beam fixed on the table the 80 kN shaker installed in V2I premises, (b) Drive Voltage Comparison and (c) Control acceleration Comparison.

Fig. 4(b) compares the drive voltage measured and simulated with the virtual shaker simulator when the control is considered at the beam basis. The curves are superimposed but the peaks observed when the system is passing through the resonance frequencies are slightly underestimated.

The computational time to simulate the whole frequency range is about 8 min (16 Go RAM), namely a factor 2,8 in comparison with the real test duration.

When the control is considered at a middle point of the beam, the physical test does not pass the anti-resonance frequency (automatically stopped by the controller due to detection of upper limit exceeding and/or low level of acceleration on the control accelerometer). The difficulty to perform the test is well predicted by the simulator. The simulation shows multiple peaks exceeding the control acceleration (Fig. 4(c)). Some control parameter modifications can be tested to try improving the control behavior. As it is observed in the physical test, an increase of the compression factor when the anti-resonance approaches helps the controlled acceleration to stay inside the admissible limits. This improvement is tested on the simulator.

7.3. Test Case 2: Truss structure on shaker in horizontal configuration

A second test-case allows demonstrating the possibility to simulate specimen placed on the slip table. A truss steel structure is now considered. The feasibility to impose the control signal at a point located at a middle position of an intermediate beam (node 500098 in Fig. 5(a)) is assessed.

Fig. 5(b) shows the accelerations determined by the simulator. Peaks exceeding the upper limit are observed. If this vibration test is performed, it is highly probable that the test will be interrupted by the controller. High input current that can exceed the limit is also foreseen. Some modification of the control parameters at the proximity of the problematic frequency range can improved the controller behavior and are tested. Prior to the test, the problems can be predicted and improvement solutions can be found.

8. Conclusion

The developed virtual shaker testing tool allows predicting the dynamic behavior of the coupled shaker-specimen assembly. Difficulties can be detected and solutions can be tested in a numerical way. The risks of non-appropriate vibration test imposed to specimen can be lowered.

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

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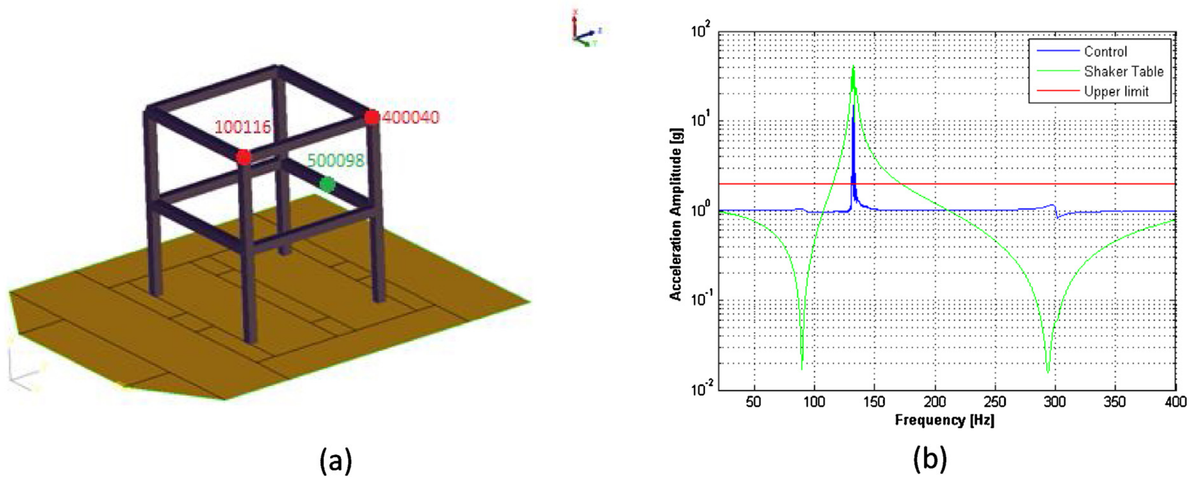

Fig. 5. Test Case 2:(a) Truss steel structure placed on the slip table model and (b) Simulated Accelerations.

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