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SHAKER STRUCTURE INTERACTION: OVERVIEW AND UPDATED RESULTS

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Due to its versatility, the electrodynamic exciter (aka electrodynamic shaker) has become the preferable excitation device in structural dynamic tests. An electrodynamic exciter allows the experimentalist to excite the structure under test with almost any kind of input signal: sinusoids, random, transient, in either open loop or in closed loop setups. It is also known, however, that since the shaker needs to be mechanically attached to the structure it is almost impossible to avoid some sort of interaction between them. This paper presents a comprehensive analytical and experimental study on the interaction between electrodynamic exciters and the structure under test. The causes and effects of this interaction on the obtained data quality are discussed. The experimental results confirm most of the phenomena observed in the analytical section, such as: (i) the input force drop-off and (ii) the influence of the shaker / power amplifier assembly on the structure's response. Finally, the efficiency of FRF estimators (H_1, H_2, H_3 and H_v) is verified experimentally as applied to shaker tests in the vicinities of structure's resonances, whereby the input force drop-off phenomenon occur and consequently, the signal-to-noise ratio may become critical, resulting in completely inappropriate FRFs.

1. Introduction

Experimental modal analysis has been a classic technique for obtaining the dynamic characteristics of structures, such as, resonant frequencies, mode shapes and modal damping, providing the basis for further analysis such as: response to a given dynamic load, aeroelastic prediction, model updating and validation, and active or passive control design.

Among the available techniques to excite a structure, the use of a shaker is probably the most popular, thanks to the wide capability of exciting with different signals: periodic, random or transient. However, since the shaker needs to be mechanically attached to the structure, it is nearly inevitable that some sort of interaction will occur between them [1]. The causes and effects of this interaction have been a matter for the experimentalists since the very beginning of modal analysis [2-3] and is still a relevant research topic, in both open loop and in closed-loop shaker testing [4-8].

In particular, Oliveira & Varoto [9] presented a study on this matter paying special attention to the force drop-off phenomenon and making a brief review of most of the references cited above. Recently, Lang [6] approached the subject from the point of view of the shaker's performance and Peres *et al* [10] presented several practical aspects on setting-up the excitation device. In their papers shaker performance is addressed only with respect to the shaker's mechanical features and the power amplifier voltage/current capacities, neglecting the dynamics of the structure. Comstock [11] proposes a feedback process based on the armature acceleration to improve force performance in modal tests.

The aim of the present paper is to perform an analytical and experimental study on shaker/structure interaction, reviewing some of the most important concepts and bringing some new issues to light.

2. Analytical Models and Theoretical Background

The theoretical development described in details by McConnell & Varoto [1], considers the shaker's armature dynamics as well as the electrodynamic relationships with the power amplifier. Also, the interaction between shaker and the structure concerned.

A simple 1 DoF shaker model is a fair hypothesis considering shakers are typically operated bellow the armature resonance, *i.e.*, within its usable frequency range[1,5,9]. The frequency response for the shaker is determined by the 1 DoF system resonance. This subject will be revisited in the experimental section, where different tests are performed to characterize the shaker dynamics. Figure 1 shows the electromechanical model used by Olsem [12] and McConnell & Varoto [1] to describe the electromagnetic coupling on the armature-coil system. This electromechanical coupling is governed by several parameters, for example, the armature mass m_a, stiffness k_a and damping c_a, the coil resistance R and inductance L, and the input voltage signal $E(t)$ and the back electromagnetic voltage E_{bemf}.

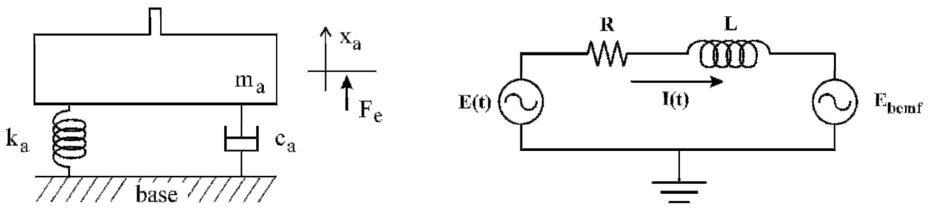


Figure 1. Electromechanical shaker model

The equations that govern this electromechanical system are given as:

$$m_a\ddot{x} + c_a\dot{x} + k_a x = F_e(t) \tag{1}$$

$$RI + L\dot{I} + E_{bemf} = E(t) \tag{2}$$

It is important to state here that the electromagnetic force $F_e(t)$, as well as the E_{bemf} voltage are a function of the power amplifier's mode of operation. Power amplifiers usually have two modes of operation, current and voltage mode. These modes establish the voltage vs. current relationships during exciter operation. The basics of each mode of operation are: (i) in the current mode of operation, the relationship between the input voltage (E_i) to the amplifier and its output current (I) to the shaker is the linear relationship $I(\omega)=G_iE_i(\omega)$ where G_i is the amplifier current mode gain, leading to an excitation force that is proportional to the voltage sent to the amplifier; on the other hand, (ii) in the voltage mode, the voltage sent to the shaker coil is proportional to E_i, resulting in a net voltage that is dependent on the E_{bemf}., which is dependent on the armature velocity $E_{bemf}(t) = K_v\dot{x}(t)$. Therefore, the coupled systems of equation (3) and (4) should be solved in order to derive the relationship between E_i and I_o (current and voltage amplitudes, respectively) or even F_e.

$$(k_a - m_a\omega^2 + jc_a\omega)X = K_f I_0 \tag{3}$$

$$(R + j\,L\omega)I_0 + j\,K_v\omega X = E_0 \tag{4}$$

Notice that the relationship between the E_{bemf} and the shaker table velocity neglects nonlinear components due to great armature displacements as described by Tomlinson [3].

Figure 2 shows the analytical bare table accelerance FRFs for the current and voltage modes of operation of an available vibration exciter, normalised by the first armature resonance. The difference between the two modes of operation is clear. In the current mode FRF (solid line), once the table passes through its mechanical resonance, the FRF amplitude becomes constant, and this implies, in principle, a reliable condition for exciter operation, as stated in [1]. The other two FRFs depicted in Fig. 2 (dashed and dotted) correspond to the bare table FRF in the voltage mode of operation. It is severely damped when compared to the current mode, mostly due to the high electromagnetic damping. Also, the two voltage mode FRFs shown in Fig. 2 are different in the sense that they use different values for the coil resistance. It is seen that the smaller resistance yielded a magnitude closer to the current mode FRF for higher frequencies.

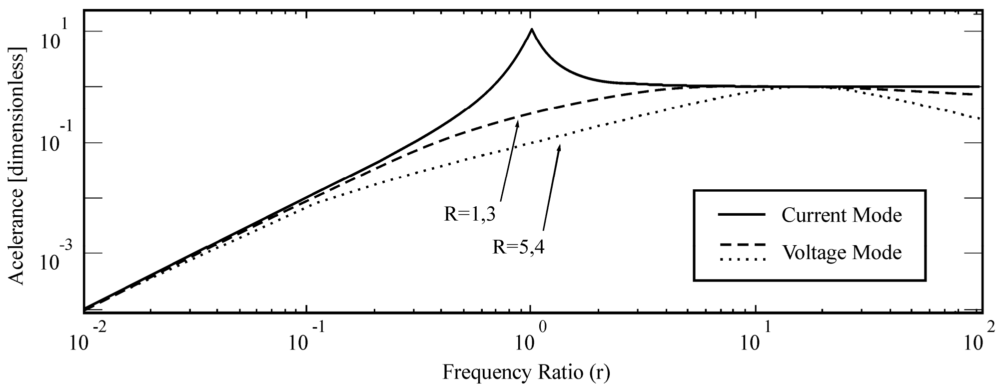


Figure 2. Bare table accelerance FRFs around armature resonance.

3. Shaker/Structure Interaction: Numerical Simulation

In this section the interaction between the shaker and the structure is approached analytically. Initially the basic equations are derived and the force drop off phenomenon is illustrated. If one assumes that the structure is connected to the shaker through a rigid connector, the same acceleration will occur at the interface point, *i.e.* $\ddot{x}_a(t) \cong \ddot{x}_s(t)$ [1,13], where the indices *a* and *s* refer to the armature and DoFs structure, respectively.

These accelerations can be written in the frequency domain as:

$$\begin{cases} \ddot{X}_a(\omega) = H_{aa}(\omega)\left(F_e(\omega) - F_1(\omega)\right) \\ \ddot{X}_S(\omega) = H_{11}(\omega)\left(F_1(\omega)\right) \end{cases} \tag{9}$$

where $H_{aa}(\omega)$ is the shaker electromechanical driving point FRF and $H_{11}(\omega)$ is the structure driving point FRF, $F_e(\omega)$ is the force generated by the armature in its magnetic field and $F_1(\omega)$ is the force delivered to the structure at the attachment point. From equation (9) an expression for the force delivered to the SUT can be derived:

$$F_1(\omega) = F_e(\omega)\frac{H_{aa}(\omega)}{H_{aa}(\omega) + H_{11}(\omega)} \tag{10}$$

Figure 3 shows the force delivered by the shaker to a structure with arbitrary modal properties. This result illustrates the drop off on the force transmitted to the system in the vicinity of the system's natural frequencies. It also can be observed that the magnitude of the drop off on the force varies with different modal masses, which are respectively: $M_1 = 4.0$ kg, $M_2 = 0.2$ kg, and $M_3 = 2.0$ kg. By observing Fig. 3, it can be stated that the lower the modal mass, the deeper the drop on the excitation force.

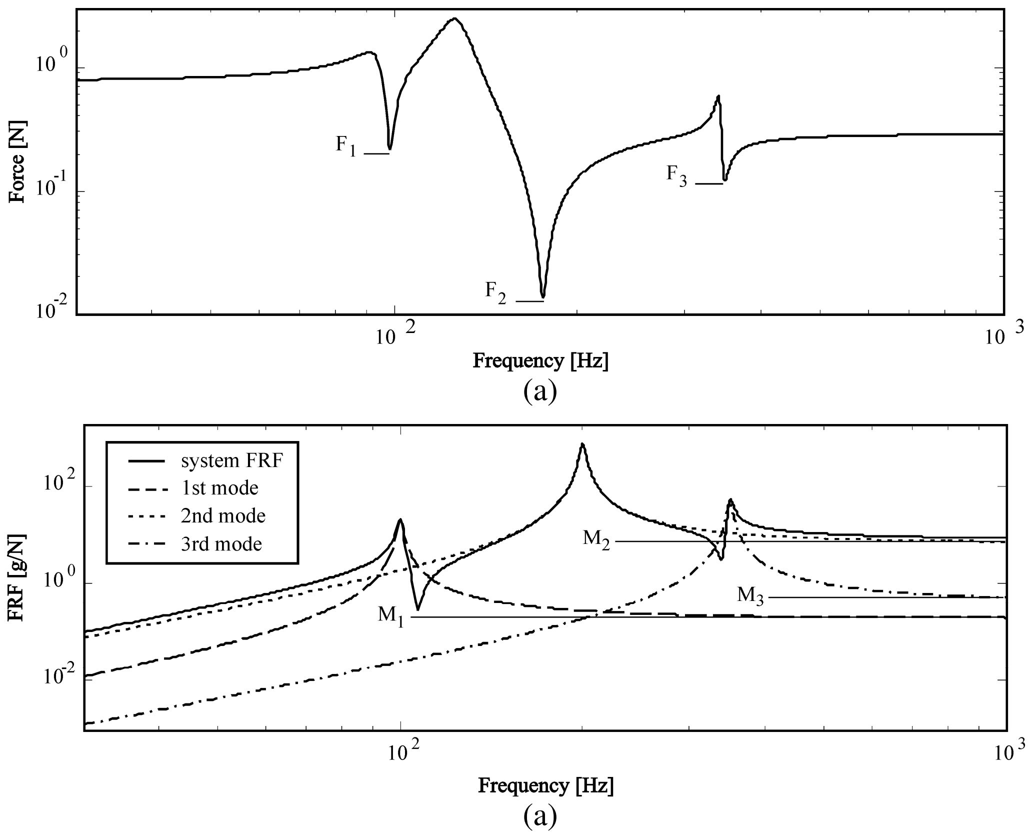


Figure 3. Shaker structure interaction (a) force drop off (b) SUT modes response and modal superposition

4. Experimental Validation

In this section, the validity of models reviewed and proposed earlier is discussed through experimental results. Initially, experiments concerning the shaker itself are conducted. Next, the main phenomena involving shaker/structure interaction are illustrated with some experiments conducted in a complex structure.

4.1 Shaker Electrodynamic Model

In order to observe the phenomena described before by analytical models of the shaker itself, three experiments were conducted: (i) the armature bare-table, (ii) impact and (iii) the shaker vs. shaker tests. The current test setup includes: Modal Shop's SmartAmp™ 2100E21-400 and shakers model 2025E and K2007E01, PCB force transducer model 208C01, PCB accelerometer model 352A78, PCB hammer kit model 086C03, LMS SCADAS Mobile running on LMS Test.Lab.

The bare-table test consists of sending a wideband signal to the shaker coil and measuring the FRF from table acceleration to the voltage sent to the power amplifier. The resulting FRFs can be seen in Fig. 4. Both shakers, Modal Shop 2025E and K2007E01, have been tested and, as it can be seen, present similar frequency domain characteristics, with low-frequency resonance around 20Hz and high-frequency armature resonance at about 9.5kHz. In this type of test, it is possible to clearly identify the high-frequency resonance, while the low-frequency one is heavily damped by the amplifier voltage mode. In order to better understand the pure mechanical behaviour of the exciter, other types of tests can be conducted, such as the impact test.

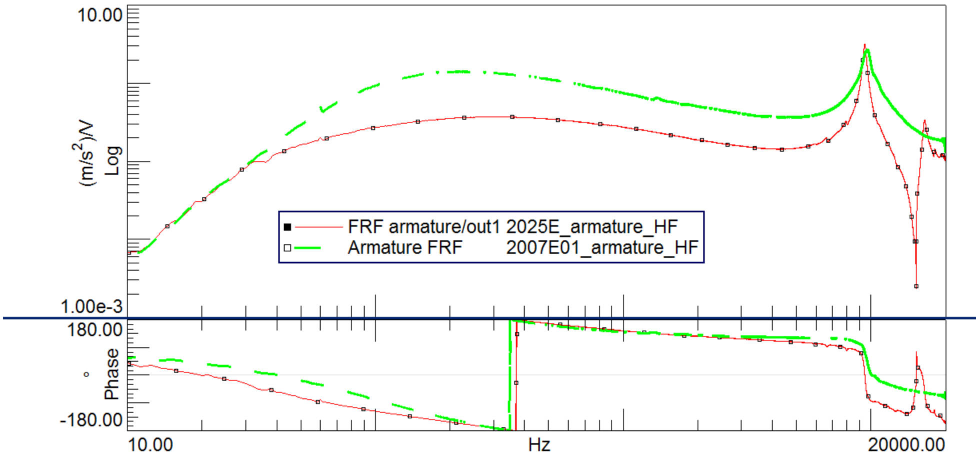
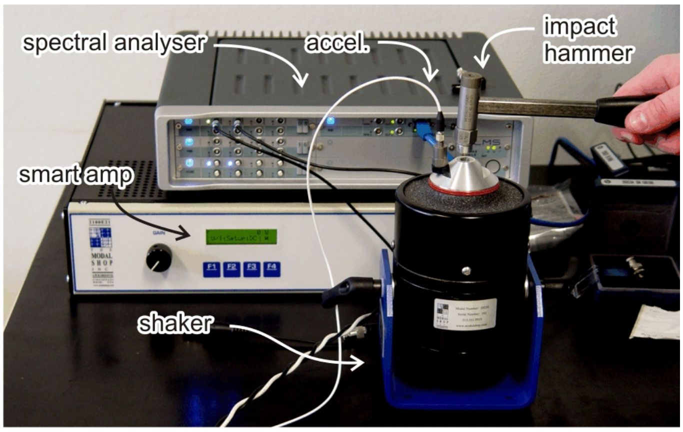
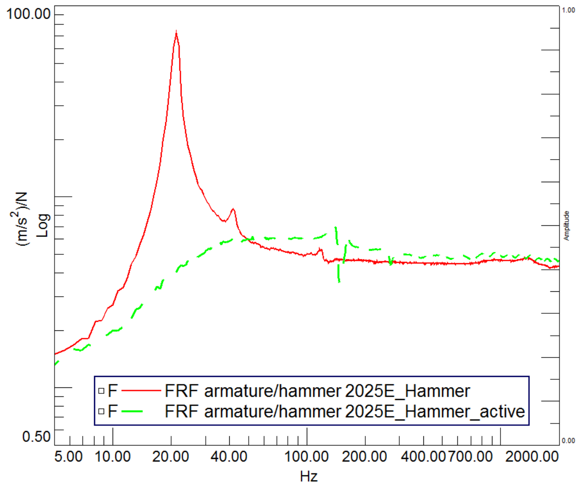


Figure 4. Armature responses to coil excitation

The armature impact test (Fig. 5a) consists of installing an accelerometer on the armature top and impacting it with an instrumented hammer. In this way, not only the electromechanical but also the mechanical model of the shaker can be validated and/or obtained experimentally. The resulting FRFs are depicted in Fig. 5(b), that shows the passive (amplifier off) and active (amplifier on) FRFs for armature acceleration over hammer tip force. The effect of the electromagnetic damping introduced by the amplifier is clear in Fig. 5(b), where the FRFs for the Modal Shop 2025E shaker are shown.



(a)



(b)

Figure 5. (a) Armature impact test setup and **(b)** Armature impact response for active and open coil

Figure 6 shows the shaker vs. shaker test setup. Here, one shaker is activated by the power amplifier, while the other one can be passive (open coil) or active (powered by another amplifier). The electromechanical FRFs, for coil voltage over armature force excitation are shown in Fig. 7. These measurements allow the identification of the constants K_v and K_f mentioned in section 2. The effect of the E_{bemf} can also be seen on Fig. 7, besides damping out the resonance peak, it also decrease the voltage output for off-resonance frequencies by an order of magnitude. Last but not least, the plot shows the phase angle between force and voltage, which obeys a 1 DoF system FRF, shifting from -90^{o} to $+90^{o}$ after the resonance.

4.2 Shaker/Structure Interaction

In order to demonstrate some of the phenomena discussed above in a real scenario, the setup shown in Fig. 8 was created. The car door was excited with the 2025E shaker, using burst random at

80%, rectangular windows on input and output channels and voltage mode amplifier. A total of 30 averages were calculated for both H_1 and H_v FRF estimators, in order to show the vulnerability of H_1 to the force drop-offs.

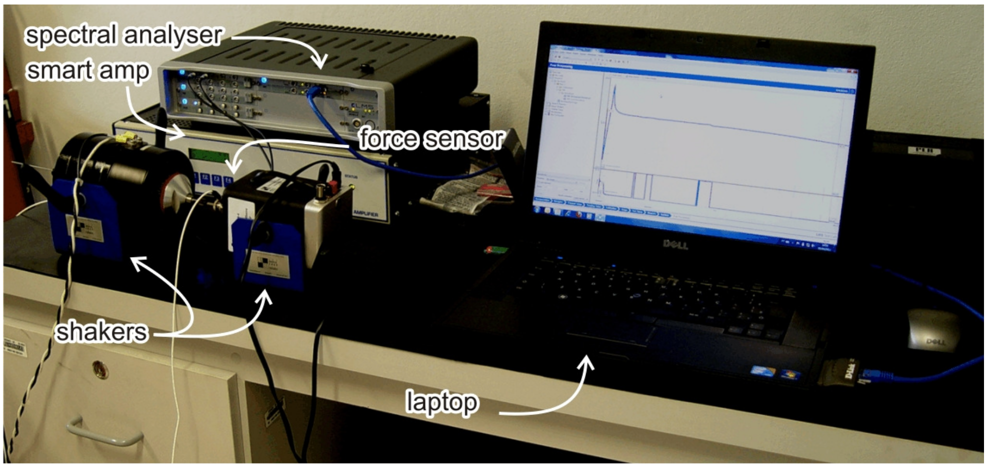


Figure 6. Shaker vs. shaker setup

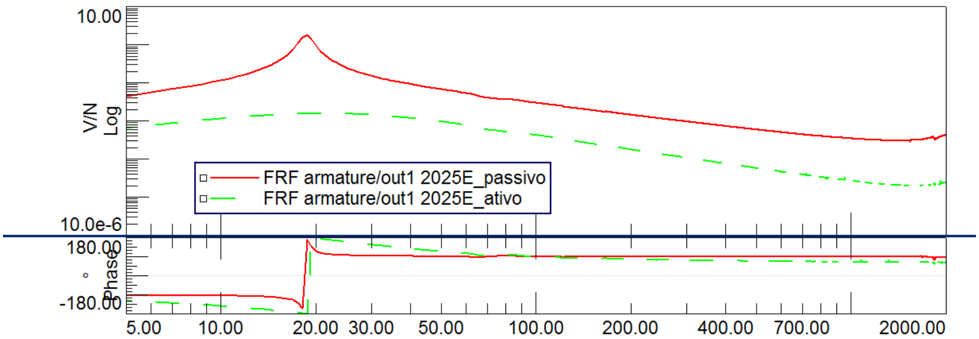


Figure 7. Coil electrical response to armature input force

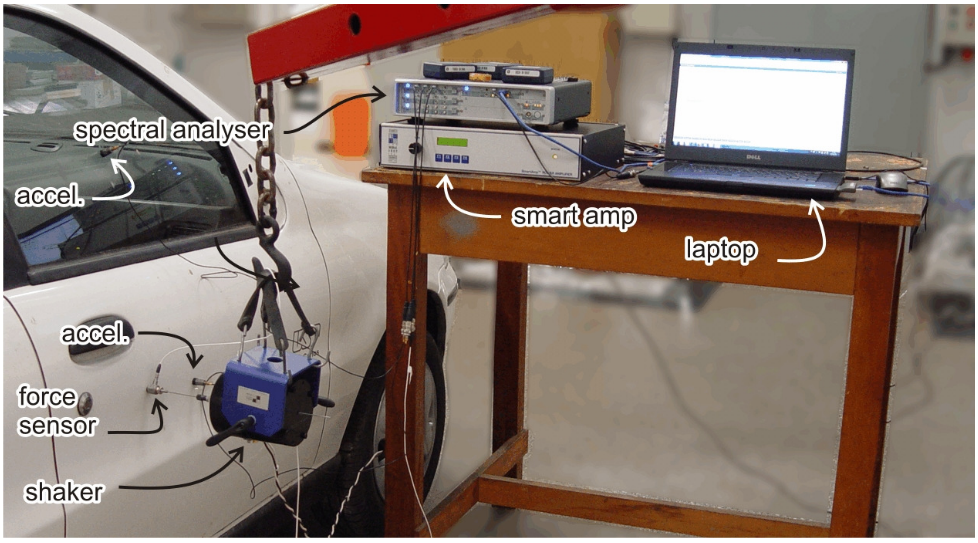


Figure 8. FRF measurement setup – car door.

Figure 9 shows a comparison of the power spectrum densities for the input forces when the shaker is attached either to the mid-door or to the window (and the same voltage and amplifier gain are used). The main difference between these two approaches is the impedance of the excitation point, which is higher at the window (as well as more damped). The armature motion when attached to the window will be influenced by the structure impedance, resulting in lower velocity relative to the shaker body. Consequently, the force drop-offs will be less noticeable, which will minimize the negative effects resulting from this phenomenon.

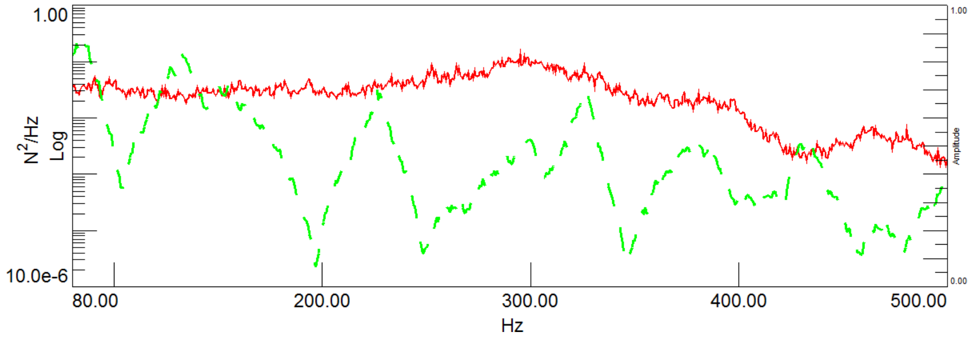


Figure 9. Input force PSD: (- -) mid-door (–) window

Figures 10 and 11 show comparisons of FRFs and Coherence measured on the car setup using H_1 and H_v FRF estimators. While Fig. 10 shows the measurements for the mid-door excitation point, Fig. 11 shows the data for the window location. The coherence functions shown in Fig. 10 illustrate such negative effects. Due to the drops in excitation force, the reference signal is more susceptible to noise, which is a problem when using H_1 estimators, hence the low coherence values. The quality of the measurement can be improved by using the H_v estimator. The best approach, however, is to chose a better point of shaker attachment. A good rule of thumb would be choosing a "harder" point (while avoiding nodal lines), rather than a point of maximum displacements. This result can be seen in Fig. 11 by measuring at the mid-door position and exciting at the window: the H_v estimator gives better coherence values than H1, while there are no significant discrepancies when comparing H_1 and H_v estimators for the FRF.

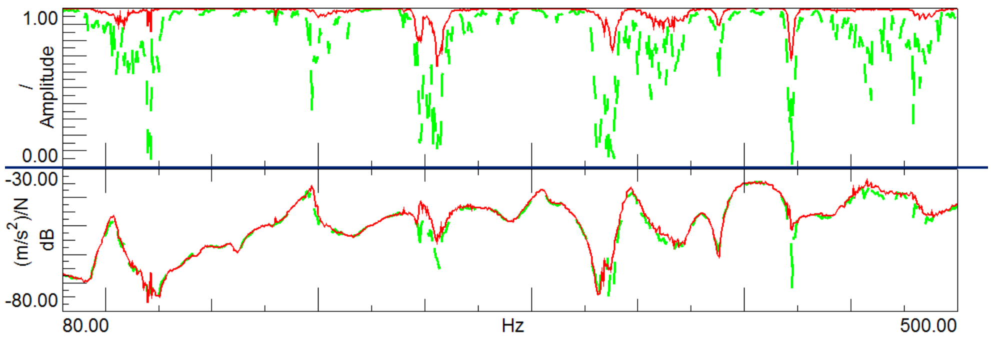


Figure 10. Coherence and FRF - door excitation / window response: (- -) H_1 estimator and (–) H_v estimator

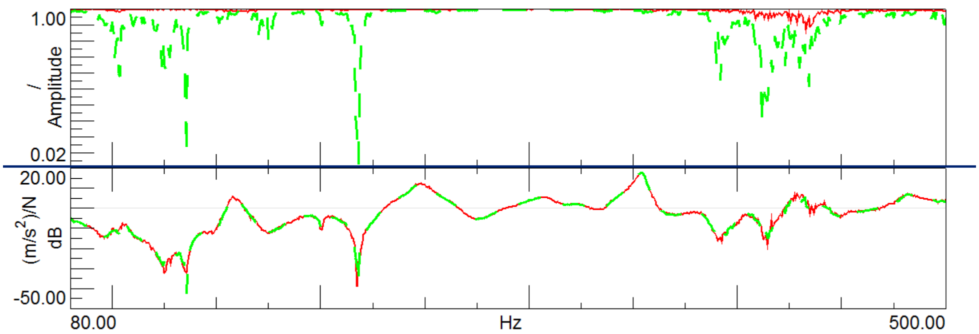


Figure 11. Coherence and FRF - window excitation / door response: (- -) H_1 estimator and (–) H_v estimator

5. Conclusions

This paper presented a review on some of the key aspects regarding shaker excitation in vibration testing both, analytically and experimentally. Some experiments have been performed in order to validate the hypothesis adopted during the shaker modelling, such as the 1DoF mechanical model and the electromechanical coupling. The shaker/structure interaction results in the well known force drop-off phenomenon that was predicted by the analytical model and observed during the experiments. As a result, one should always take special attention to excitation test setup, details such as the proper shaker placement, the correct use of stingers, the proper choice of excitation signals, windows and FRF estimators, etc. are crucial for acquisition of quality data.

6. Acknowledgements

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