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Non-linearity Identification of Composite Materials by Scalable Impact Modal Testing

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Nomenclature:

DOF Degree of Freedom
EMA Experimental Modal Analysis
FEA Finite Element Analysis
FFT Fast Fourier Transformation
FRF Frequency Response Function
SAM Scalable Automatic Modal Hammer
SLDV Scanning Laser Doppler Vibrometry

Abstract

The aim of experimental modal analysis is to determine the structural dynamic characteristics of a given component or assembly. However, modal models are based on linear systems of equations and assume material orthotropy and linear stiffness components. Many industrial elements made of highly-complex, composite materials, do not accomplish these assumptions due to their non-linear material behavior. One practical measurement method is performing iterative modal analyses; this is, measurements at different force input levels. Several iterations lead to the knowledge of different points of the structural force/response spring curve and how this behavior affects the modal test.

In this paper, a novel Scalable Automatic Modal hammer (SAM) is presented. The SAM allows exciting the structure with precisely adjustable and reproducible force amplitudes. The test device is designed in a way that only the inertia mass of the hammer tip impacts the structure with a finely amplitude-adjustable Dirac impulse. The non-linear behavior of composite materials and jointed structures can be investigated with the SAM in terms of impact force-depending natural frequencies and damping ratios. This leads to an increase in the accuracy of the experimental data and therefore, a more straightforward modal model correlation in regards to the real structure.

Keywords

Experimental modal analysis, non-linear material, automatic modal hammer, composite material, validation

Introduction

The general definition of experimental modal analysis (EMA) is that it is a process whereby a structure is described in terms of its dynamic properties, i.e. eigenfrequencies, damping ratios and mode shapes. Systems of equations are used for this purpose, where each equation describes the response of one of the degrees of freedom (DOFs) of the studied structure. The DOF represents the minimum number of independent motions required to define the positions of all parts of a system at an instant of time. The equation describing damped oscillatory motion for one degree of freedom in an arbitrary direction of space u takes the form [1]:

$$m\ddot{u} + c\dot{u} + ku = 0$$

Systems of motion equations, describing all the DOFs of the studied system, would take the matrix form:

$$[M]\{\ddot{u}\} + [C]\{\dot{u}\} + [K]\{u\} = 0$$

As a result, the vibrational behavior of any structure can be ideally described in terms of this system of linear equations. Thus, it is generally said that the basis of modal analysis assumes linearity, an assumption which has two main implications in the present context [1]:

1) That doubling the magnitude of the excitation would simply result in a doubling of the response, and so on, and
2) That if two or more excitation patterns are applied simultaneously then the response thus produced will be equal to the sum of the responses caused by each excitation individually [1].

Assuming the studied structure is linear means assuming that the distributions of mass [M] and stiffness [K] are linear throughout the structure as well. While this is generally accomplished in most academic examples such as beams and plates, some real industrial structures pose many challenges on their accurate analysis. Parts made of innovative composite materials, or bolted/joint structures do not fit well into this linear model and their dynamic properties are influenced by many other factors.). Frictional interfaces are known to induce non-linear behavior and mechanical joints cause variability in terms of resonance frequencies as well as amplitude levels [2]. These tend to be characterized in a non-linear force/response rate curve (Fig. 1

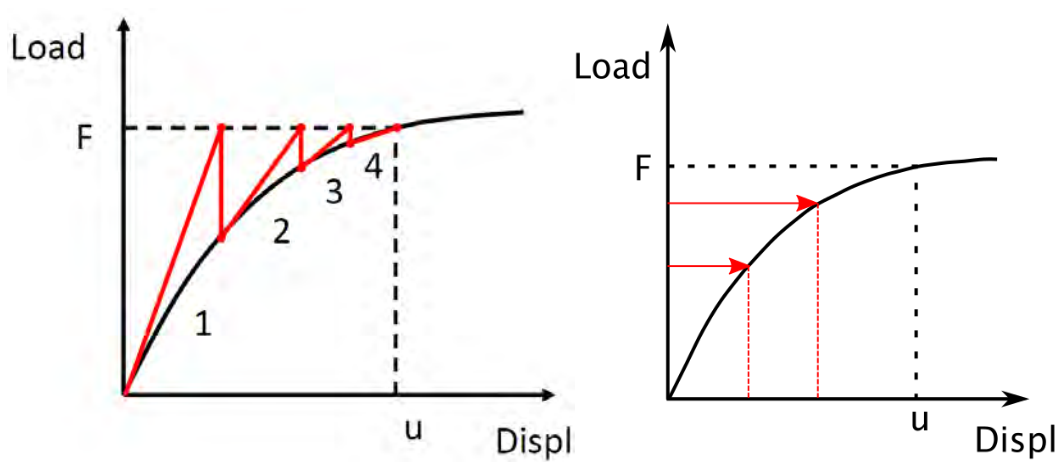

Fig. 1: Left – An example of a Newton-Raphson iterative modal analysis analysis for one increment of load (four iterations) at a given force/spring rate curve of a non-linear structure [3]. Right – Example of an experimental modal measurement of a non-linear structure in incremental force levels with an automatic modal hammer.

These difficulties can be overcome by iterative computations when performing FEA-based modal analysis (Fig. 1, left). There exist several methods to carry out non-linear computations based in incremental-iterative methods [3]. However, modal analysis performed via FE requires a continuous validation process of the simulation models with experimental data in order to improve the simulation results [4, 5]. A working hypothesis is that these iterative FE modal models can be correlated through iterative EMA data (Fig. 1, right).

The Scalable Automatic Modal Hammer (SAM)

Until recent years, experimental modal analysis could only be performed by means of handheld hammers and piezoelectric sensors. This is not only a time-consuming approach, but it results in different and inaccurate force inputs at each hammer impact. This problem could be solved by averaging the measurements many times, with the side effect of dramatically increasing the measurement time, especially when a large number of DOFs were involved. The last years have seen the popularization of Scanning Laser Doppler Vibrometry (SLDV) systems for many kinds of vibration testing, which allow effortlessly analyzing hundreds of DOFs in a single experiment. Many other excitation techniques have been developed during the last decades, such as semi-automatic modal hammers, electrodynamic/piezoelectric shakers and shaker-based, non-contact magnetic exciters.

The automatic alternatives currently available in the market fail to allow a fine-tuned, robust and truly repeatable impacting force adjustment. In most cases, the experimental repeatability will be an issue due to the fact that the human factor is not fully eliminated during the hammer positioning procedures and force adjustment or re-adjustment throughout the measurement. In SLDV experiments with hundreds of DOFs, issues related to operator fatigue easily arise. Electrodynamic shakers have issues hindering their usability at frequencies higher than 10 kHz and require changes to be done in the tested structure, like threads for the attachment of the stinger. Non-contact magnetic exciters, such as those described in [6] are based in electrodynamic shakers and thus, they suffer of the aforementioned limits in their spectral ranges [7].

The Scalable Automatic Modal Hammer (NV-TECH, Steinheim a.d. Murr, Germany), called SAM on its acronym in German language, has been developed and patented with the purpose of iterative experimental modal analysis in non-linear structures (Fig. 2). This test device allows the characterization of dynamic properties of non-linear structures through iterative modal testing, such as parts made of composite materials, automotive brake pads, gas turbine blades, etc., by the precise and adjustable setting of different impact force amplitudes.

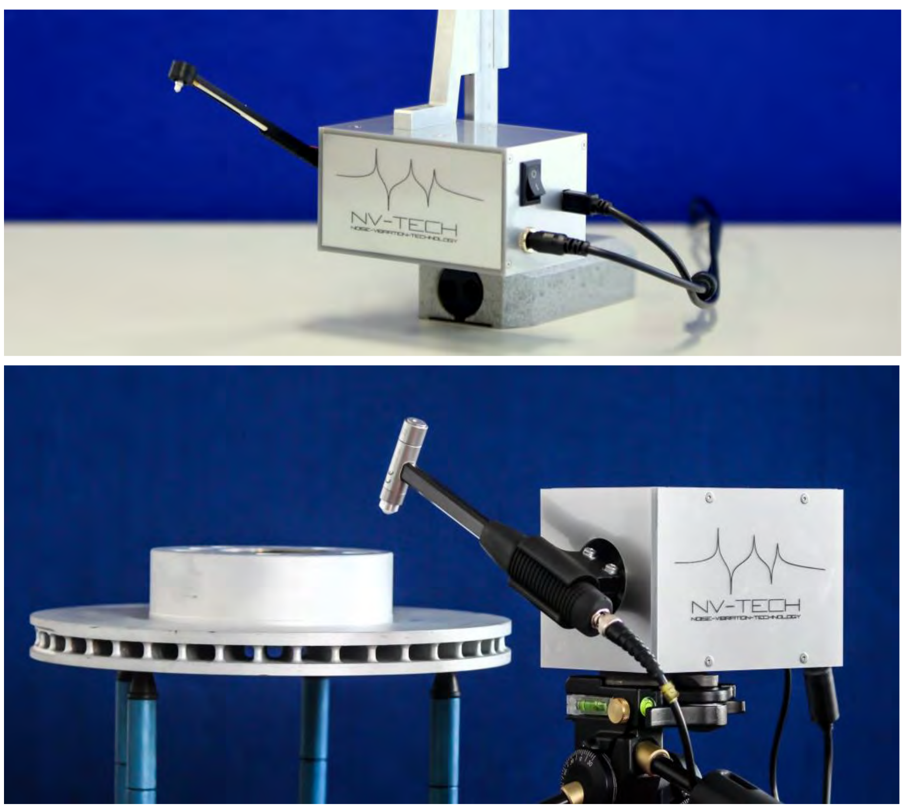

Fig. 2: Two sizes of SAM. Top – SAM1, for the analysis of light-weight structures (brake pads, turbine blades, material probes, music instruments, etc.) up to more than 20 kHz. Down– SAM2, for the analysis of heavier structures, (brake discs, blade assemblies, alternators, frames, rotors, etc.) up to 12 kHz.

The SAM is composed mainly of two parts, a stepper motor (PANdrive, Trinamic Motion Control GmbH & Co. KG, Germany), encased in acrylic plastic for security reasons, and a hammer handle. The hammer handle is rigidly attached to the stepper motor axle. Due to the complexity and design requirements of the hammer handle design, the handles for SAM1 and SAM2 are manufactured by the 3D printing of ABS plastic. Both handles for SAM 1 and SAM2 feature a slot where a commercial mini modal hammer model 086E80 (PCB Piezotronics, Depew, NY, USA) for SAM1, or a model 086C03 modal hammer for SAM2, can be rigidly inserted and fixed. The hammer handles have a functional design. They avoid the loosening and subsequent accidental turning of the hammer tip during its operation. Failure to avoid the turning causes that the cylinder where the piezocrystals and microelectronics are contained impacts the structure directly. This leads most of the times to irreversible damages in the force sensor. The hammer handle is rigid enough to lead the correct direction of the force sensor but, at the same time, it is flexible enough to induce a free and reactionless impact [7]. A computer program, which contains the instructions the SAM follows, is pre-programmed on a PC and transferred via USB cable to the stepper motor.

The reliability of the SAM in regards to its impact repeatability and reproducibility has been already discussed in [7] and [8] among others, therefore this discussion lies out of the scope of this paper.

Materials and Methods

The chosen structure for the analysis is a brake pad. These kinds of parts are made of, at least, two joint components: the friction material and the backing plate. The friction material is a highly complex compound of binders, lubricants, resins, abrasive components, glass, reinforcing fibers, etc. [9]. All of them are proprietary information of each manufacturer. Therefore, any brake pad in the market is a good example of a non-linear structure.

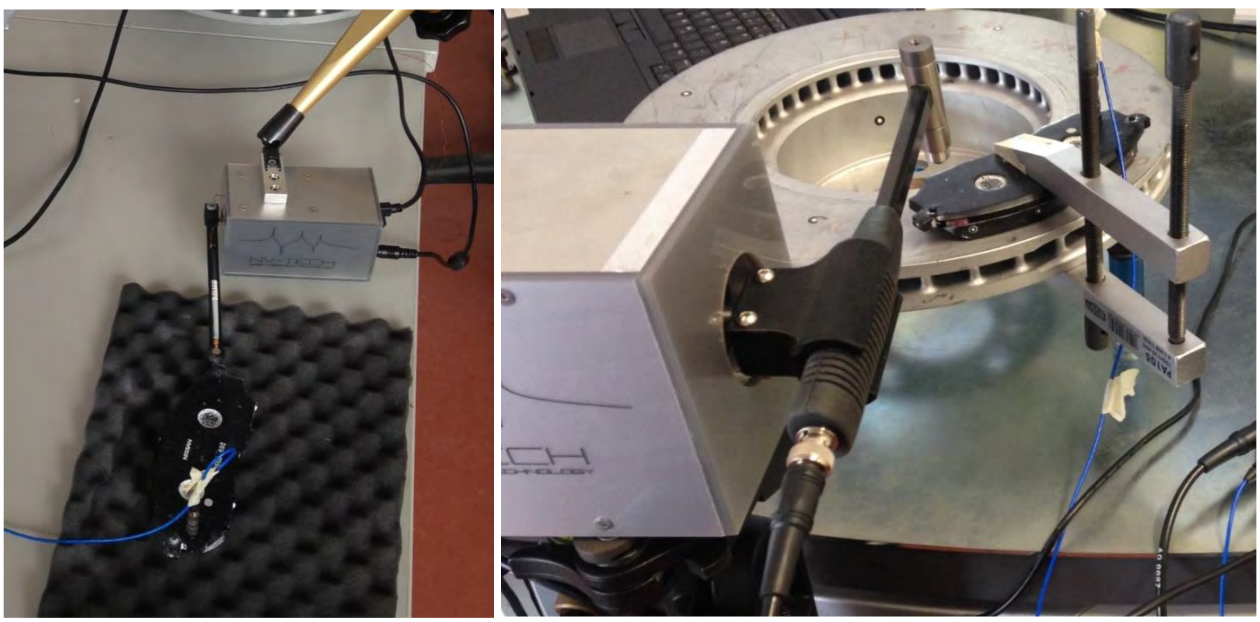

Fig. 3: Left – Test A, free-free brake pad impacted with the SAM1 in a range between 5 and 100 N. Right – Test B, clamped pad impacted with the SAM2 in a range between 80 and 2000 N.

Two tests have been performed for two different setups for the same brake pad. The output response signal was in both cases collected by means of a 5.8-gram single-axis accelerometer model 352C04 from PCB Piezotronics, Inc. A single Frequency Response Function (FRF) was for each case computed between the input and output points at each force level. Each FRF was computed with 10 averages at each force level for all tests.

Test A consisted on the analysis of a brake pad placed on a foam mat and impacted with the SAM1 in a force range between 5 and 100 N (Fig. 3, left). Test B consisted on the analysis of this same brake pad, but in this case clamped to a larger assembly, and impacted with the SAM2 in a force range between 80 and 2000 N (Fig. 3, right). The brake pad was clamped to a heavy brake disc in Test B because experience showed that the pad would pop out of its initial position when impacted with the SAM2 on free-free conditions. The entire assembly was supported by magnetic feet with rubber caps at their ends. As indicated, the SAM2 is appropriate for testing heavier structures and assemblies, such as the used in Test B.

Test A: Results and Discussion

Fig. 4 shows a general plot of the FRFs obtained for impact forces ranging between 5.2 and 98 N with the SAM1. Lighter shades of blue in the plot indicate increasing impact force amplitudes. It can be observed with bare eye how there are slight variations in damping between the lightest and darkest plots. FRFs derived from measurements with impact forces below 20 N lead to very noisy results at frequencies higher than 10 kHz. Zooming in on each peak allows observing better those shifts (Fig. 5). Linear scaling is used in Fig. 5 for a better visualization. The damping ratio of the first mode (1.97 kHz) decreases with increasing impact forces, the peak increases its height as a result. The damping ratio of the third mode (5.87 kHz) increases with increasing impact forces, the peak shortens as a result. There are no measurable eigenfrequency shifts in in this test.

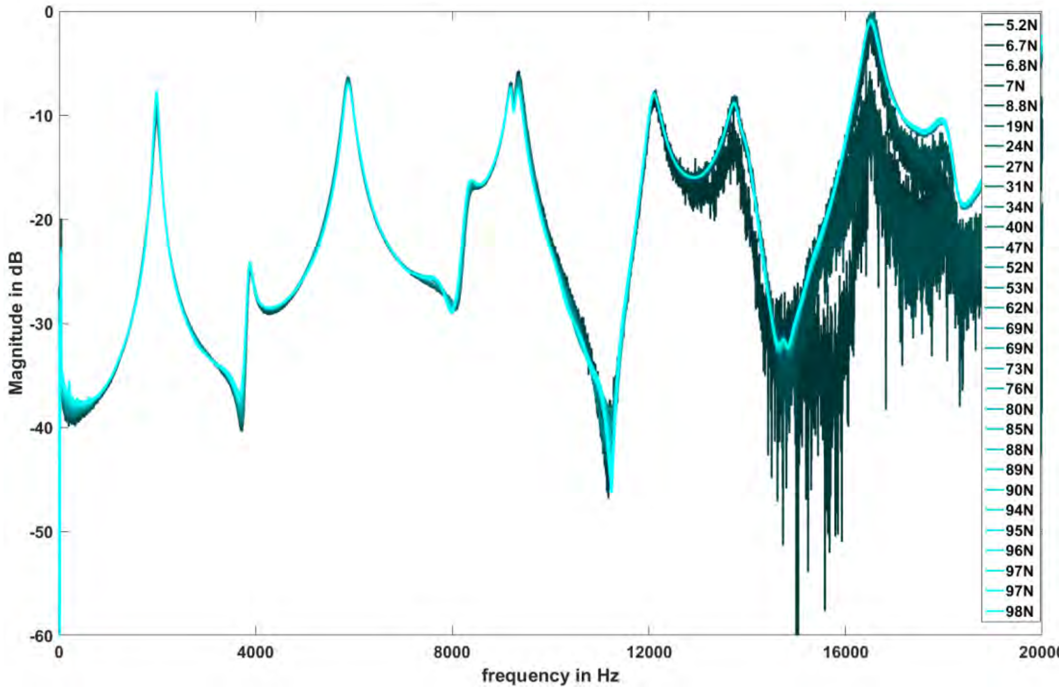

Fig. 4: Plot in dB scaling of the FRFs obtained during Test A. Lighter shades of blue indicate increasing impact force amplitudes.

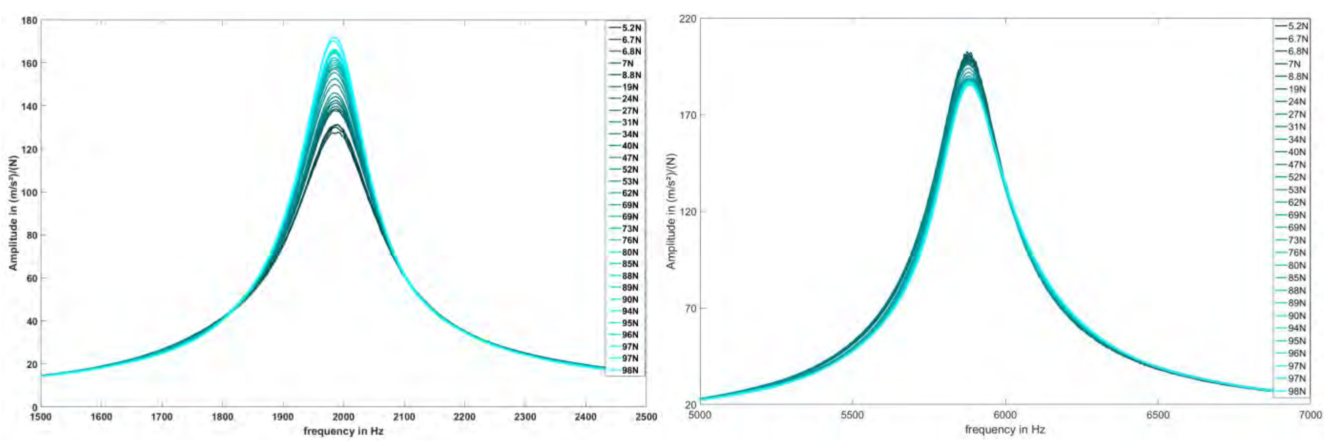

Fig. 5: Zoomed views in linear scaling of Fig. 4 at the ranges between 1.5-2.5 kHz (left) and between 5-7 kHz (right). Linear scaling is used for simplicity.

The changes in damping ratios have been accounted with the half power method, also popularly known as the "-3 dB method", for all modes on this analysis and compared for growing impact force amplitudes (Fig. 6). The higher differences can be found for the first two modes. With an overall decrease of 0.5%, the first mode is the most influenced by increasing impact force amplitudes.

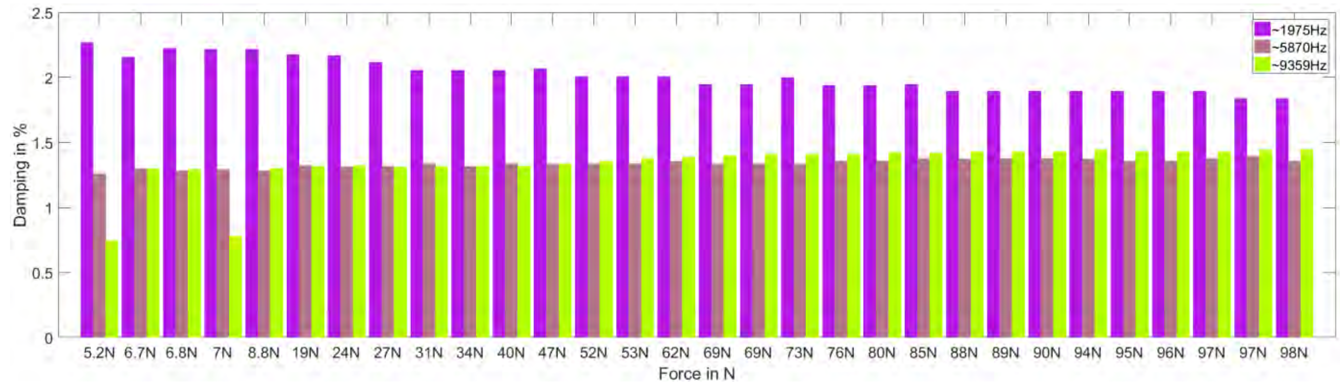

Fig. 6: Progression of the damping ratio of modes 1, 3 and 5 with respect to the impact force amplitude. Note the progressive decay in damping with increasing forces for mode 1 at 1.97 kHz, and the slight increase in damping with increasing forces for mode 5 at 9.35 kHz.

Test B: Results and Discussion

Fig. 7 shows a plot of the FRFs obtained for impact forces ranging between 80 N and 2000 N with the SAM2 for the spectrum range of interest between 0 and 3.5 kHz. The entire range is not shown due to the lack of space. The increasing shades of blue in the plot indicate increasing impact force amplitudes. Note the chaotic behavior of the FRFs caused by the clamping of the analyzed brake pad, in line with the results described in [10]. It can be observed with bare eye how all peaks present noticeable variations in damping and eigenfrequency shifts.

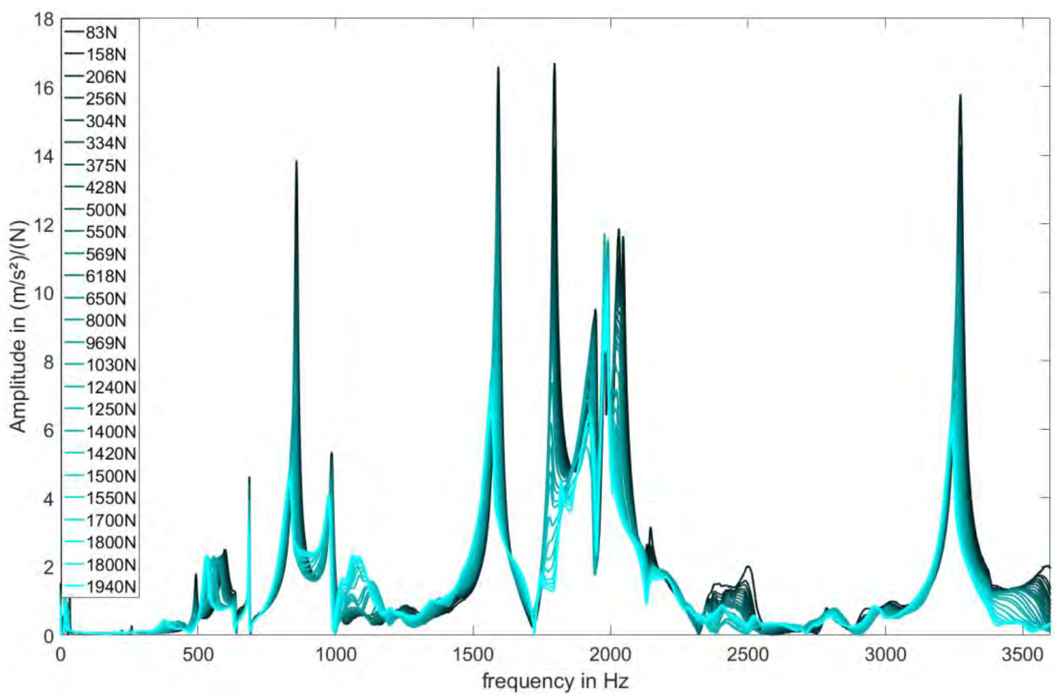

Fig. 7: Plot in linear scaling of the FRFs obtained during Test B. Lighter shades of blue indicate higher impact force amplitudes.

Figures 7 and 8 show how higher impact force amplitudes generally lead to a shift to lower eigenfrequencies and generally higher damping values (the peak amplitude decreases) for almost all modes. There are a few exceptions, such as mode 6 (5.8 kHz) which presents a remarkable damping progression, growing in the force range between 0 and 0.5 kN and decreasing thereafter.

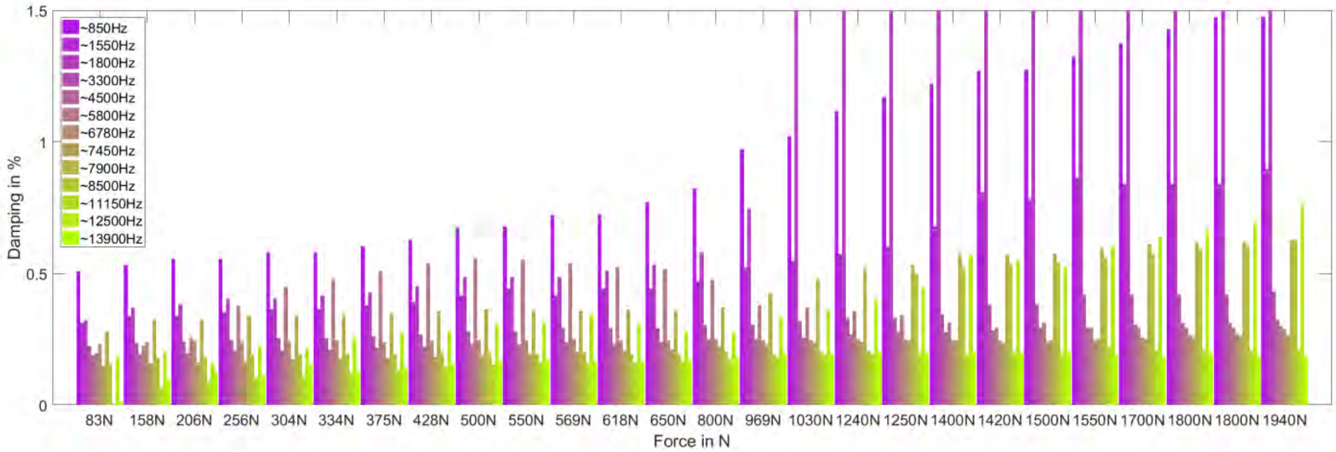

Fig. 8: Progression of the damping ratio of some of the modes encountered in this analysis with respect to the impact force amplitude.

A zoomed view of mode 2 is shown in Fig. 9, where the combined frequency shifting and increase in damping can be clearly seen. By plotting the different peak values of each FRF a curve can be generated, which is a representation of the force/response spring curve for this resonant mode, for this specific test and setup.

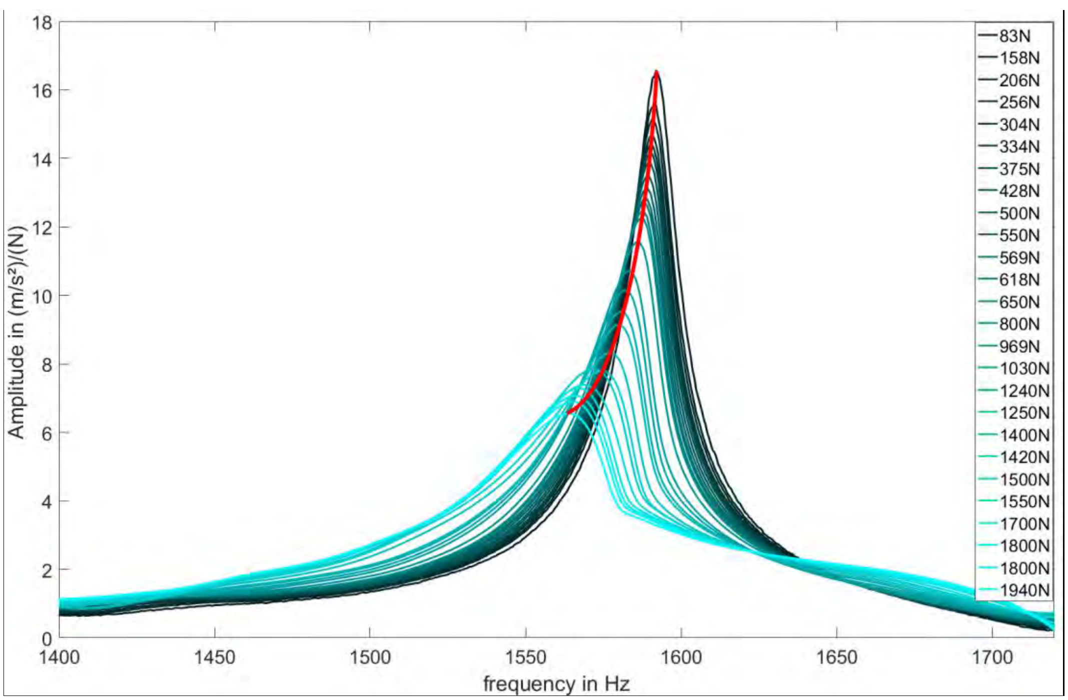

Fig. 9: Zoomed view of Fig. 7 at the range between 1.4-1.7 kHz. The shift between the eigenfrequencies obtained at 83 and 1940 N is of approximately 27 Hz. A red curved line has been added as a representation of the force/response spring curve for this given resonant mode on this experimental setup.

Conclusions and Further Work

In this paper, the study of non-linear structures such as brake pads by means of the SAM has been investigated. Both tests prove an inherent force-depending damping distribution in the analyzed free-free and clamped brake pad, which can be analogous to many other composite, highly non-linear structures. The differences in damping are lower in Test A than in Test B, two reasons are thought to have a clear influence:

1) The pad is set on free-free conditions in Test A, whereas it is clamped against a larger structure in Test B. There are friction-related damping effects which have not been taking into account in this investigation. The aim of this paper was proving the capabilities of the SAM in regards to the investigation of force-dependent damping, and not the modal characterization of the pad.
2) Figure 10 shows in a graphical way the hypothesis that the effects in the obtained damping ratios are smaller, even negligible, when small forces and small displacements take place. It is possible that the excitation range of the SAM1 (5 to 100 N) falls in the quasi-linear section of this non-linear force/response curve and therefore, the differences between extreme force values are small. Instead, on a range with higher force loads, such as those inflicted by the SAM2, which lead to higher displacements, the non-linear effects would increase its importance and have a clear impact on the progression of damping ratios as shown in Fig. 8.

Both models of SAM are currently applied in several research projects devoted to the research in non-linear material probes and aerospace and automotive components. The results derived of these investigations will be used to prove or disparage the hypotheses proposed in this paper. The SAM is constantly subject of improvements and revisions, all oriented to increase the testing precision and usability of the device.

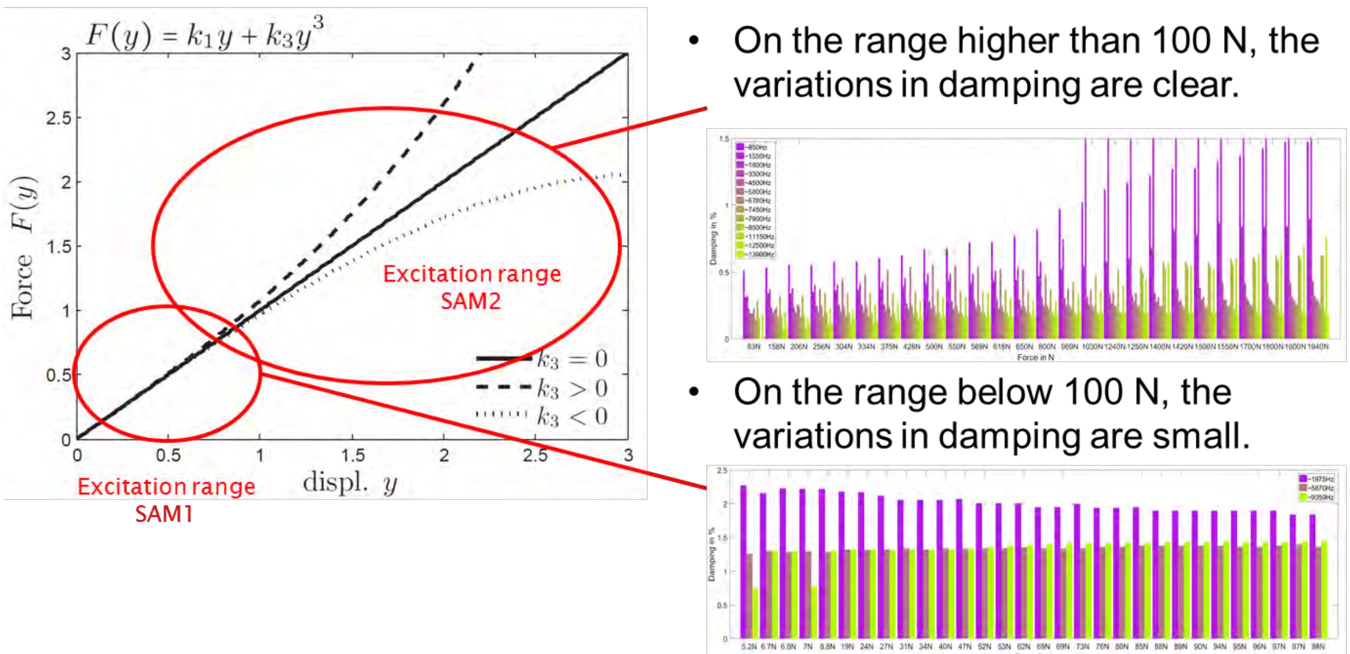

Fig 10: Non-linear force-response curves [11] in relation with the excitation ranges capable by the SAM1 and SAM2.

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