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# DESIGN OF A VIBRATION MACHINE WITH BASE DISPLACEMENT CONTROL FOR STOCKBRIDGE DAMPER VIBRATION TESTS

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**Abstract.** *The action of wind on power lines generates mechanical vibrations that can damage cables and interrupt transmission. The Stockbridge damper is used to reduce these oscillations and extend the life of the lines. Experimental studies at Stockbridge use shakers to generate base excitations with constant velocities or accelerations. Tests with constant displacements are difficult to control on the shaker, especially in the resonant regions of the damper, revealing a little explored gap in Stockbridge studies. To fill this gap, a vibration machine capable of generating constant base displacements through the scotch yoke mechanism was developed. The construction of the machine involved a structured design methodology, as well as considerations and mathematical models. The functional implementation of the machine comprised dynamic tests with and without Stockbridge, in six different amplitudes of displacement, between 3 and 50 Hz of excitation. The results indicated the expected quadratic behavior of the base, as well as revealing the natural frequencies, the predominant regions of the vibration modes and the non-linear behavior of the damper. Therefore, these results validate the machine as a viable and affordable option for Stockbridge vibration testing.*

**Keywords:** *vibration machine, scotch yoke mechanism, Stockbridge damper.*

## 1. INTRODUCTION

The action of winds on electric power transmission lines promotes the formation and propagation of mechanical vibrations along these structures. These oscillations, in turn, cause progressive mechanical damage to the suspension points of the cables, which may cause them to break and interrupt the transmission of electrical energy (Labegalini et al, 1992).

The reduction of these damages and, consequently, the increase of the useful life of the lines can be made possible by suppressing the critical levels of vibration (Júnior, 2010). For this, vibration absorbing devices were developed, such as the Stockbridge damper. Figure 1 illustrates an asymmetrical model of this device, which consists of two inertial masses, a messenger cable and a clamp for fixing to the conductor cable.

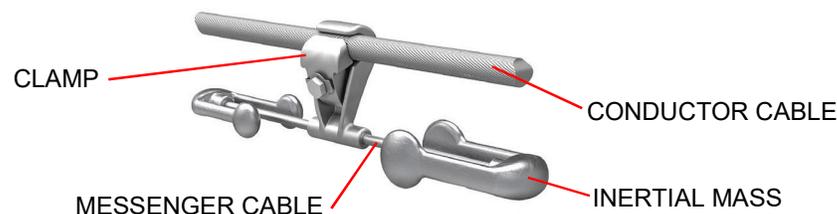


Figure 1 – Stockbridge damper coupled to a conductive cable (Pfisterer, 2020).

Several researchers have investigated the mechanical behavior of Stockbridge through experimental or numerical studies (Claren and Diana, 1969; Diana et al, 2003; Krispin et al, 2007; Barry, 2010; Barbieri and Barbieri, 2012; Kasap, 2012; Marchi, 2017; Vaja et al, 2018; Barbieri et al, 2019). These experimental studies employed shakers to generate harmonic and periodic base motions with constant velocities or accelerations.

However, the shaker has technical and operational limitations when acting as a Stockbridge excitation source. These gaps pave the way for the possibility of using other excitation mechanisms, until then, little explored when analyzing Stockbridge. Therefore, this work aimed to develop a vibration machine capable of generating constant base



It was found that shakers are the most used electrodynamic type exciters. On the other hand, Barbieri and Barbieri (2012) innovated by employing a mechanical type exciter. However, the adaptations and technical limitations of the machine restricted the tests. With regard to hydraulic exciters, there are no records of their use in the Stockbridge tests. In view of this, the search for other solutions focused on mechanical exciters. As a result, a potential use of mechanisms used in machines that test automotive shock absorbers was identified, for example, the scotch yoke mechanism.

Next, the project objectives were established in order to fill the identified technical gaps. These objectives included designing a mechanism capable of generating and maintaining constant the adjusted amplitude for different excitation frequencies, devising a solution to adjust different displacement amplitudes, employing materials with adequate functions and certified origin, designing components with resistance to fatigue, automating the machine through accessible industry 4.0 solutions, manufacturing the components according to the dimensioned specifications and tolerances, as well as developing a Human Machine Interface (HMI) to operationalize the tests.

To achieve these goals, the following performance specifications were defined:

- Test capacity for shock absorbers with a mass of less than 5.6 kg;
- Excitation range from 3 to 50 Hz, with a minimum resolution of 0.1 Hz;
- Adjustable and constant displacement range of up to 3.0 mm;
- Sufficient power to supply the excitation frequency and displacement amplitude requirements;
- Adequate stiffness to avoid resonances and significant vertical deformations;
- Abrasion resistant components;
- Fatigue resistance of dynamic components;
- Flexible configuration of tests to allow the coupling of sensors such as load cells and accelerometers;
- Control with local and remote monitoring of the excitation frequency of the displacement base;
- Remote HMI in order to develop automated tests.

## 2.2 Intermediate Steps

The diagram in Figure 3 summarizes the actions taken in the synthesis, analysis and selection steps.

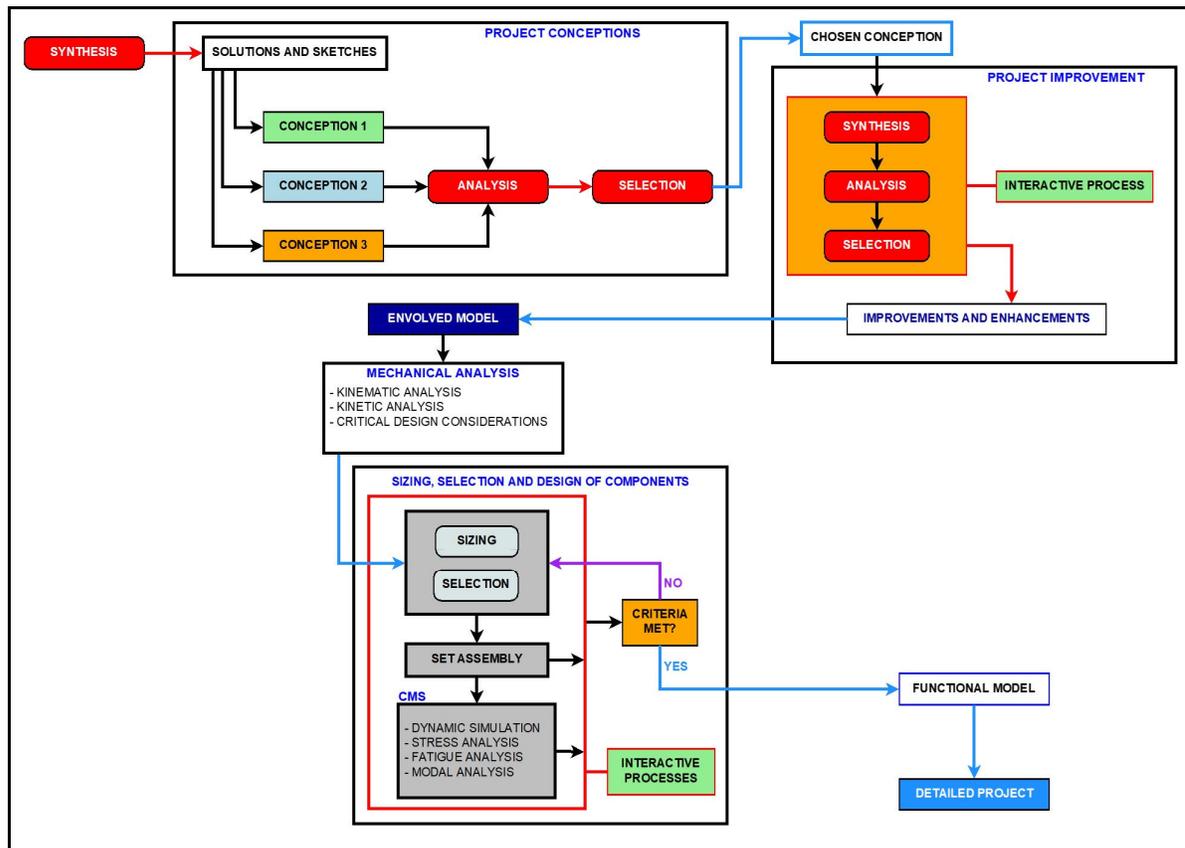


Figure 3 – Diagram of synthesis, analysis and selection steps.

Initially, from the idealization of some sketches, the most adequate conception was selected. This idea was put through an iterative process of improvements and refinements. As a result, the most involved design concept was obtained. Figure 4 illustrates this result.

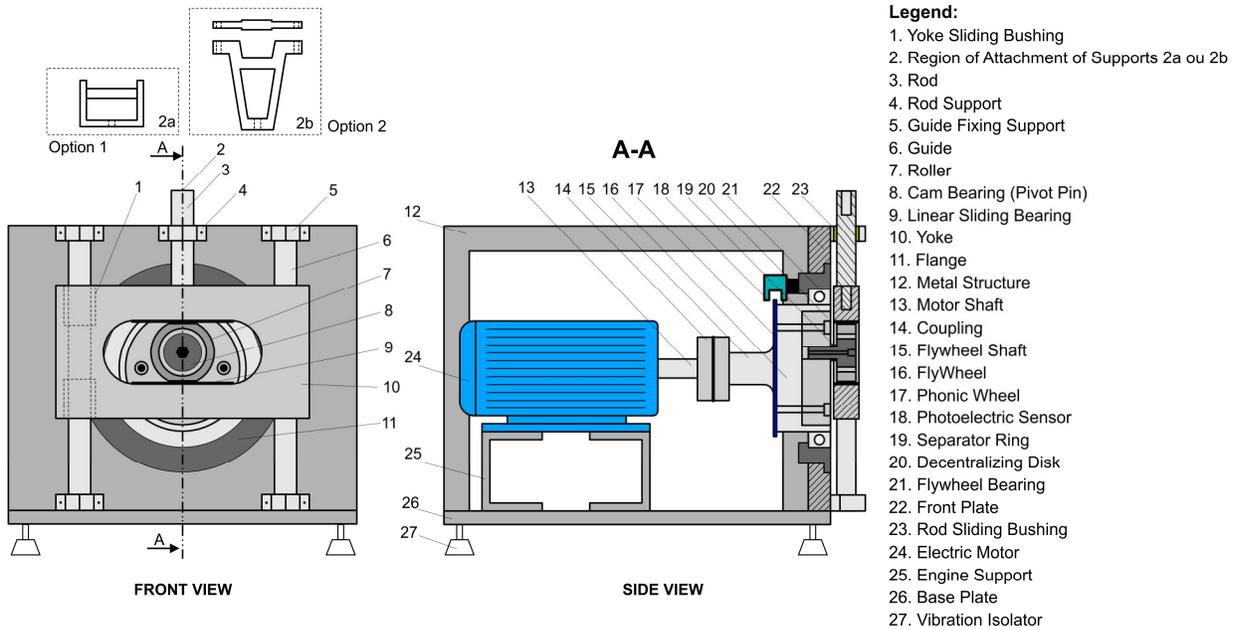


Figure 4 – Details of the evolved model.

Subsequently, the generated model was submitted to a mechanical analysis. This analysis involved kinematic and kinetic studies of the dynamic functioning of the model, as well as critical design considerations. One of them is related to the force exerted by Stockbridge due to base excitation.

The components of the scotch yoke system were dimensioned based on the most critical working condition, that is, the greatest mechanical effort. In this condition, it was assumed that the machine presents maximum values of excitation frequency, displacement amplitude and external load exerted by the Stockbridge.

The highest excitation frequency corresponds, according to the performance specifications, to 50 Hz. For the displacement amplitude, the maximum value of the specified amplitude range was admitted: 3.0 mm. Regarding the load, the largest force amplitudes occur when the Stockbridge is excited in the resonance condition. Thus, experimental force data were considered from tests conducted in Marchi (2017).

In this study, an asymmetric Stockbridge, with a mass equal to 5.6 kg, was tested in a shaker with excitations between 5 and 120 Hz and constant acceleration amplitudes. From these data, the greatest amplitude of force was identified: 367.92 N.

Based on safety coefficients recommended by Norton (2013), the force value was increased by a factor equal to 3. As a result, the value of 1110 N was considered as the maximum force amplitude generated by Stockbridge in the scotch yoke mechanism.

Following the methodology, based on the last model conceived and on the mechanical analysis carried out, the cycle of sizing, selection and design of the components was started. For this, it was necessary to adopt another iterative method, which consisted of steps of dimensioning, selection, assembly of the set and Computational Mechanical Simulation (CMS). Thus, with the help of the SolidWorks software, it was possible to design components that could fulfill the role of mechanical performance required in the dimensioned set.

The dynamic simulation was performed using the SolidWorks Motion tool. The use of this tool required the definition of some boundary conditions such as movement restrictions of the scotch yoke mechanism, external force exerted by the Stockbridge damper, gravity acceleration, parts materials, surface contact regions, friction coefficient and system excitation frequency.

Through the dynamic simulation, 720 dynamic load scenarios per machine rotation cycle were generated for the main components of the scotch yoke mechanism. Each kinetic scenario was discretized via FEM which, in turn, allowed the identification and analysis of critical stresses in each part. Figure 5 illustrates the stress analysis via FEM of the kinetic scenario with the highest load on the pivot pin, composing the generation and refinement of the mesh, determination of stresses, accompanied by iterations of convergence of a given parameter which, in this case, was the Von Mises equivalent stress in the critical region of the part.

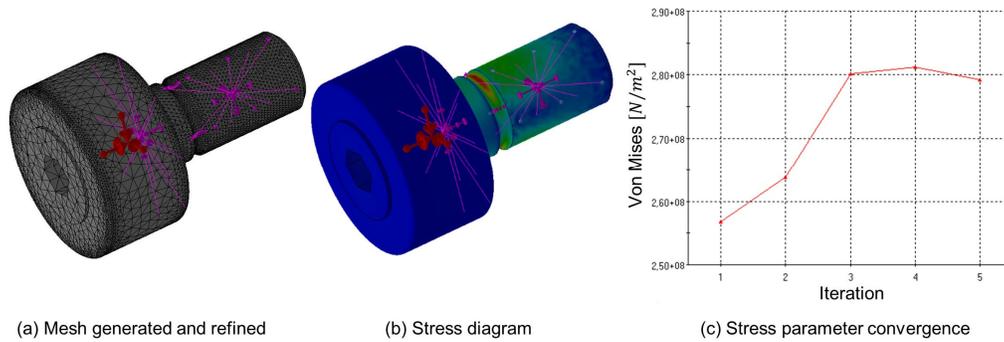


Figure 5 – Stress analysis via MEF of the pivot pin component.

Subsequently, these stresses were used to verify whether the calculated fatigue criterion was met. Furthermore, again with the help of the FEM, a modal analysis was performed to verify the presence of significant resonant frequencies in the excitation range of the machine.

Basically, five criteria needed to be met: assembly, power, elastic range, fatigue and critical resonant frequencies. If any condition was not met for a given part, then the iterative process repeated the steps of dimensioning, selection, assembly and computational mechanical simulation.

When all the major parts of the scotch yoke mechanism met the predicted criteria, a working model was obtained. This model, in turn, was able to proceed to the next stage of the project, called detailed design.

### 2.3 Final Steps

The detailed design phase involved the creation of technical drawings for each of the designed parts, as well as the assembled set. Figure 6 illustrates the resulting equipment, with the respective indication of the components.

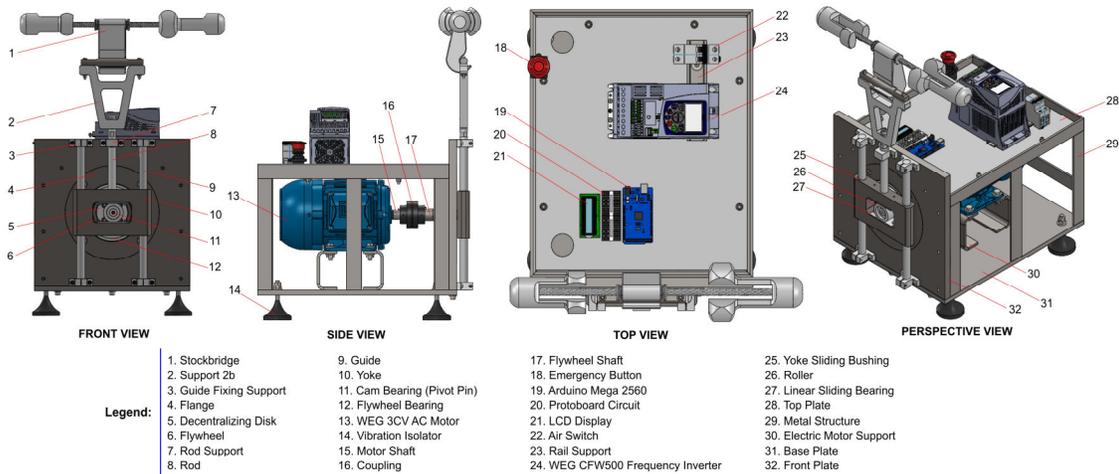


Figure 6 – Front, side, top and perspective views of the designed machine.

Following the steps, the necessary materials and accessories were acquired. Then, the parts were made from the appropriate manufacturing processes and according to the designed dimensions and tolerances.

From the manufactured parts and with the selected commercial components, the functional implementation of the machine was carried out through the following procedures:

- Assembly of manufactured and selected components;
- Verification of the integrity of the set, the presence of undesirable gaps and interference, the existence of misalignments, imbalances and incongruous decentralization;
- Calibration of the machine's rotation measurement sensor;
- Parameterization of the frequency inverter to drive the electric motor;
- Machine drive and remote control tests via HMI developed in Labview software;
- Dynamic tests of the machine with and without Stockbridge.

Once the tests were finished, improvements were implemented and the machine's functionality was identified, the methodology led to the final production stage. In this phase, to avoid oxidation, the parts were superficially protected. Figure 7 shows the final version of the equipment designed, manufactured and assembled.

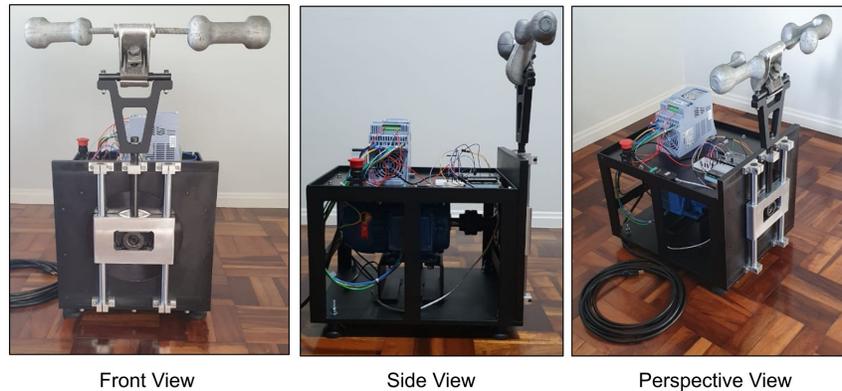


Figure 7 – Front, side and perspective views of the designed machine.

### 3. EXCITATION FREQUENCY MEASUREMENT SYSTEM

The need to monitor the rotation speed of the mechanism led to the development of a precise measurement system with low acquisition cost.

This system uses a photoelectric sensor, which detects the number of teeth on the phonic wheel in the adopted time interval. The detected teeth generate analog pulses, which are converted into digital pulses. Based on a conditioning circuit, the generated digital signals are corrected and subsequently processed in an electronic prototyping platform called Arduino.

The development of the conditioning circuit, as well as the calibration of the measurement system, required the construction of a test bench. Figure 8 illustrates this workbench and its respective elements.

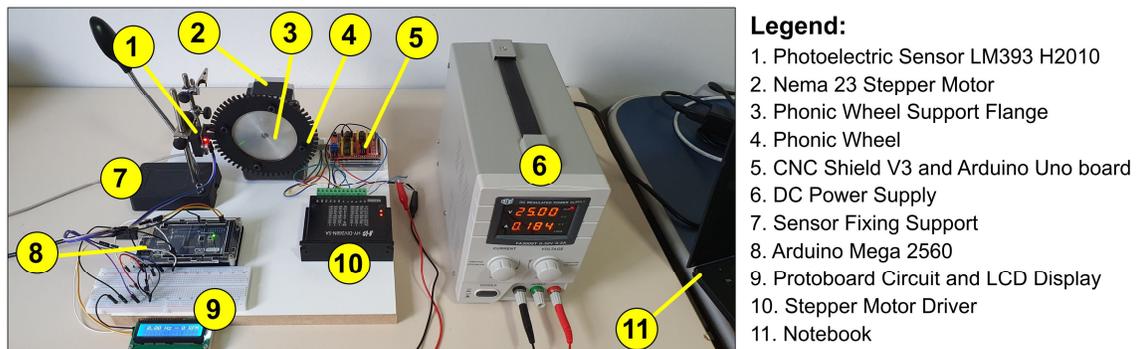


Figure 8 – Test bench for calibration of the machine's excitation frequency measurement system.

The bench in Figure 8 used as reference the precise rotations generated by a stepper motor, in which the phonic wheel was fixed. Thus, based on the known rotations of the stepper motor, between 60 and 3600 rpm, it was possible to analyze the digital signals of the photoelectric sensor through an oscilloscope. The distortions of these signals made it difficult for the Arduino to read them. Therefore, it was necessary to develop the conditioning circuit, providing the Arduino with consistent pulses proportional to the angular velocity of the phonic wheel.

Finally, through an algorithm, implemented in the IDE (Integrated Development Environment) of the Arduino, the corrected digital signals allowed to calculate with good precision the angular frequency of the motor shaft in rpm or Hz.

Complementarily, the measurement system has been improved to provide RPM readings on a remote device. For this, an HMI interface was developed in the LabView software.

### 4. DYNAMIC TESTS

Figure 9 shows the schematic design of the experimental bench used in the dynamic validation tests of the built machine and its components.



**Legend:**

1. Stockbridge Damper
2. Engineered Vibration Machine
3. Notebook with Test HMI
4. Arduino Duo
5. Notebook with Pulse Signals Software Brüel&Kjaer
6. Signal Acquisition System Type 3050A060 Brüel&Kjaer with 12 Channels
7. D-Link 8 Port Gigabit Switch

Figure 9 – Bench used in the dynamic tests of the machine.

Tests were performed with and without the presence of an asymmetric Stockbridge device. In both cases, the tests were conducted in six different ranges of displacement: 0.25 mm; 0.45 mm; 0.65 mm; 0.85 mm; 1.00 mm and 1.50 mm. For each displacement, the excitation frequency ranged from 3 to 50 Hz, with increments of 0.2 Hz, totaling 236 measurement points.

At each point, acceleration amplitudes were obtained in the time domain, with an acquisition interval of 16 s. The measured signals came from seven accelerometers, one fixed on the displacement base and the others along the damper. Figure 10 illustrates the placement of accelerometers on Stockbridge.



Figure 10 – Positioning of accelerometers along the Stockbridge.

The main technical specifications of accelerometers are organized in Table 1 such as type, sensitivity, frequency range and mass.

Table 1. Technical specifications of the accelerometers used in the tests.

| Accelerometer | Type             | Sensitivity [mV/(m/s <sup>2</sup> )] | Frequency Range [Hz] | Mass [kg] |
|---------------|------------------|--------------------------------------|----------------------|-----------|
| Base          | 4507 Brüel&Kjaer | 0.9920                               | 0.1 to 6000          | 0.0048    |
| 1             | 4507 Brüel&Kjaer | 1.010                                | 0.1 to 6000          | 0.0048    |
| 2             | 4507 Brüel&Kjaer | 1.003                                | 0.1 to 6000          | 0.0048    |
| 3             | 4507 Brüel&Kjaer | 1.004                                | 0.1 to 6000          | 0.0048    |
| 4             | 4507 Brüel&Kjaer | 1.040                                | 0.1 to 6000          | 0.0048    |
| 5             | 4507 Brüel&Kjaer | 0.9841                               | 0.1 to 6000          | 0.0048    |
| 6             | 4507 Brüel&Kjaer | 1.017                                | 0.1 to 6000          | 0.0048    |

Table 2 summarizes the acquisition parameters of the seven accelerometers in the Pulse Brüel&Kjaer signal manipulation software.

Table 2. Signal acquisition parameters in Pulse Bruel&Kjaer software.

| Parameters                              | Data    |
|---|---------|
| Points in Time Domain, number of points | 64000   |
| FFT Lines, number of lines              | 25000   |
| Sampling Frequency, Hz                  | 4096    |
| Sampling Interval, $\mu$ s              | 244.1   |
| Frequency Resolution, Hz                | 0.064   |
| Record Length, s                        | 15.625  |
| Time Weighting                          | Hanning |

In order to standardize and optimize the tests, given the number of points to be measured, a virtual HMI was developed on the Labview platform. This HMI, associated with the Arduino platform, made it possible to automatically control the activation, frequency step, maintenance of excitation, stop and machine rotation reading in accordance with the moments of measurement start and end of the Brüel&Kjaer signal acquisition system.

Then, the acceleration amplitudes in the time domain were converted to the frequency domain. These data, together with the excitation frequencies of the machine, were processed in Matlab software in order to generate evidence and graphical results.

### 5. PRELIMINARY RESULTS AND DISCUSSIONS

The experimental results are shown in Figures 11 to 13.

The graph in Figure 11 exemplifies the excitation thresholds maintained by the machine for 16 s, whether with or without Stockbridge.

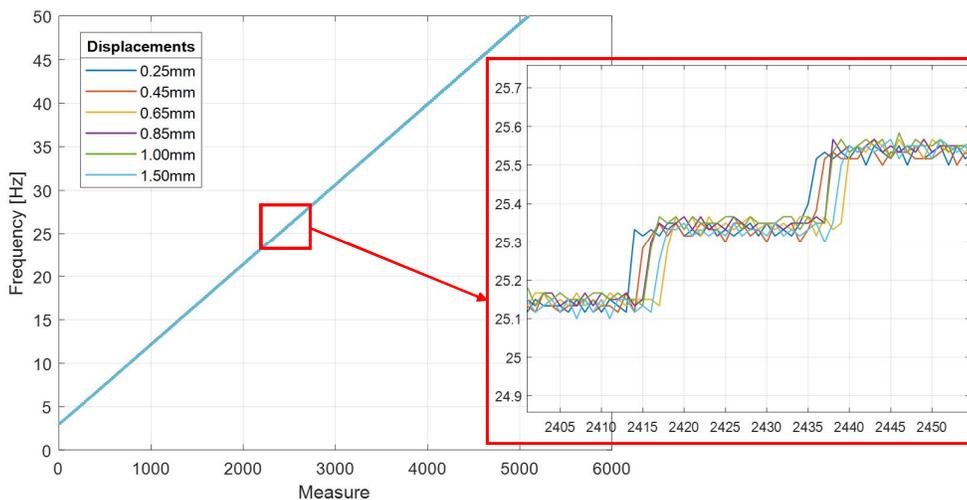


Figure 11 – Thresholds of excitation frequencies throughout the measurements.

The graphs in Figure 12 relate the experimental acceleration amplitudes of the base with the respective theoretical amplitude, in the considered frequency range, with and without the presence of Stockbridge.

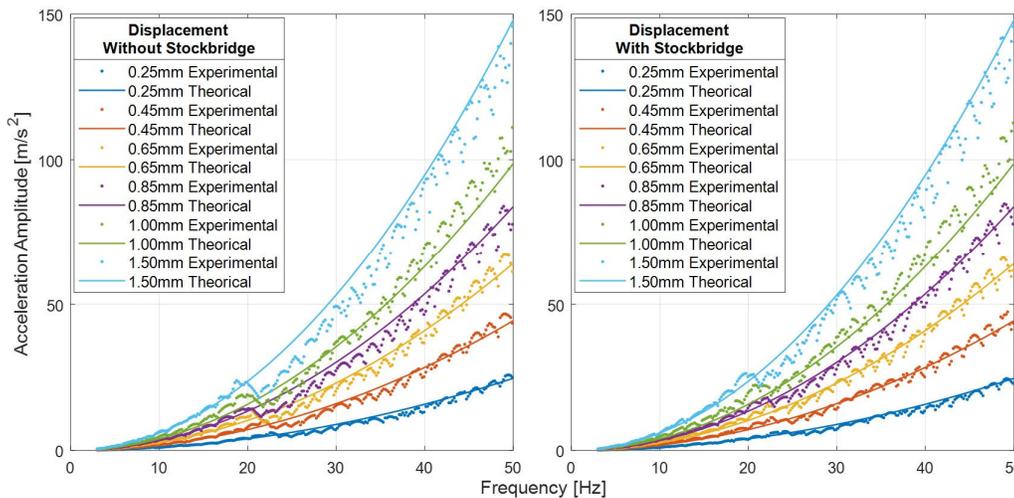


Figure 12 – Variation of acceleration amplitudes along the excitation range of the machine with and without Stockbridge.

Figure 13 corresponds to the graphs of the frequency response functions (FRF) of the respective analyzed point of the Stockbridge (Figure 10), for the six amplitudes of displacement and with the indication of the region where the four modes of vibration occur in the asymmetric Stockbridge tested.

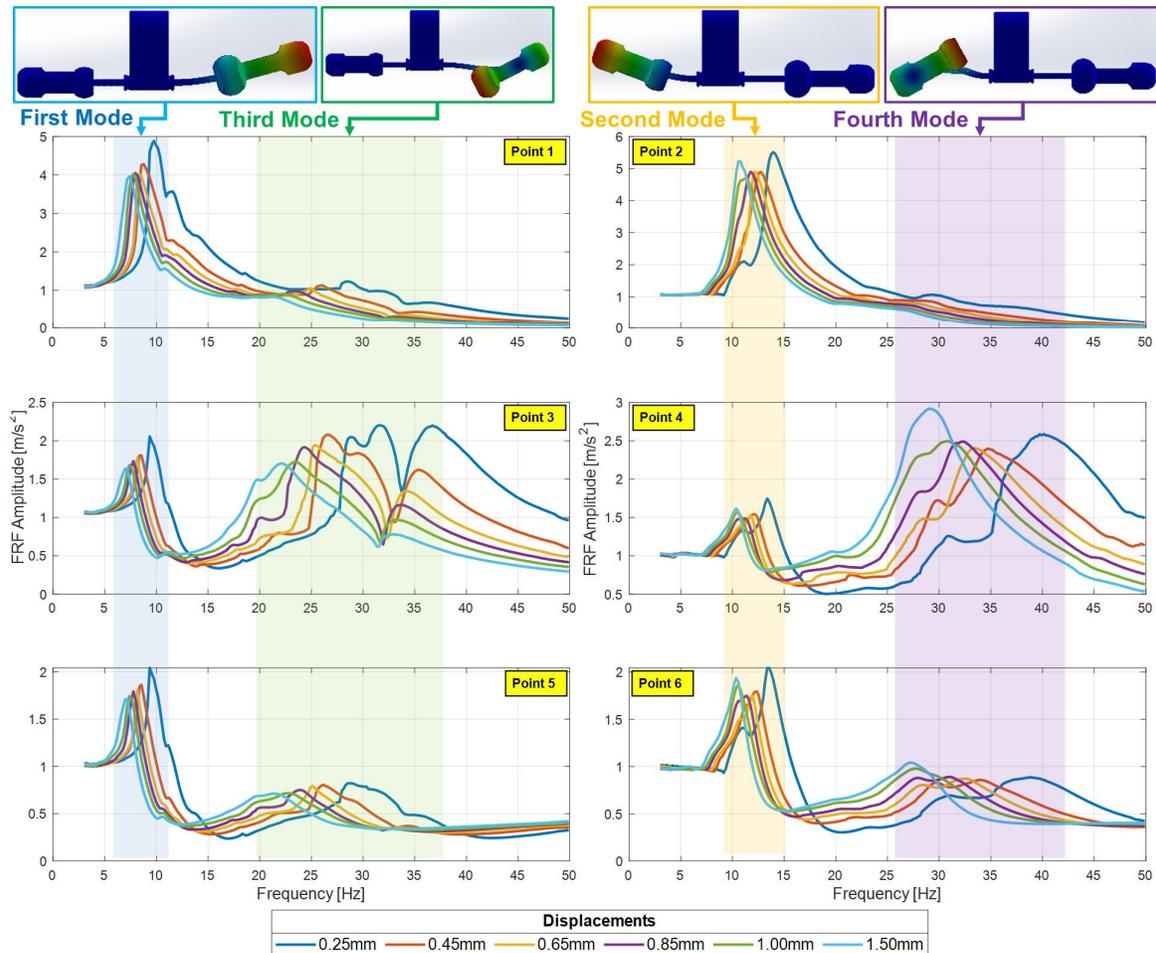


Figure 13 – Experimental FRFs of the asymmetric Stockbridge for different analyzed points, in different amplitudes of displacement and with the indication of the respective vibration modes.

The experimental results presented allow listing the following findings:

- Figure 11 shows the linear excitation behavior of the machine in the considered work range;
- The experimental data in Figure 11 reveals the machine's ability to keep the excitation practically constant throughout the measurements, in different amplitudes of displacement and load requests;
- The graphs in Figures 12 show that the maximum acceleration amplitudes of the base present an expected quadratic behavior along the excitation frequencies, either with or without the damper;
- The theoretical acceleration amplitudes, present in the graphs of Figure 12, constitute average parameters, which indicate a good convergence of the experimental data at low frequency and distortions for high machine excitations. Furthermore, these variations are accentuated with the increase in displacement amplitude;
- The FRFs in Figure 13 make it possible to identify the natural frequencies of the asymmetric Stockbridge used, as well as the regions where the vibration modes are more predominant. As expected for this damper, we have 4 natural frequencies. Two correspond to the higher inertial mass side, with the first vibration mode predominating at point 1 and the third at point 3. The other two resonant frequencies correspond to the lower inertial mass, with the second mode prevailing at point 2 and, at point 4, the fourth vibration mode;
- Figure 13 demonstrates the non-linear behavior of Stockbridge, given that the values of the natural frequencies decrease with the increase in the base displacement amplitudes;
- Finally, the FRFs show that the frequency bands around the first and second modes are narrow. As for the other two vibration modes, this range is wider. This indicates lower Stockbridge efficiency at low excitation frequencies.

## 6. CONCLUSIONS

This work aimed to develop a vibration machine capable of testing the Stockbridge device through constant base displacements.

Preliminary results show the machine's ability to generate base excitations in the specified working range, with excellent accuracy of excitation control and with an efficient and accessible monitoring system.

The increase in displacement amplitude, associated with high excitations, indicate variations in base acceleration amplitudes. However, these amplitudes showed the expected quadratic behavior, mainly at low frequencies and displacement amplitudes.

The dynamic and modal behavior presented by Stockbridge is compatible with the non-linear behavior expected for this device, as well as being in line with experimental results presented by other authors such as Tompkins et al (1956), Krispin et al (2007) and Vaja et al (2018).

Therefore, the developed machine proved to be a valid, promising and viable option for conducting Stockbridge shock absorber vibration tests. Valid because it presented the ability to generate results consistent with the expected dynamic behavior of Stockbridge. Promising because it allows extracting new information and conclusions based on the experimental data generated. And it is viable given that it consolidates unique characteristics into a single product: low manufacturing cost, easy maintenance, simple assembly, compact, robust, with automated and remote activation through accessible and integrable solutions from current industry 4.0, for example, the use of Arduino and Labview platforms.

## 7. ACKNOWLEDGEMENTS

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