

## PASSIVE CONTROL OF VIBRATIONS IN AN INSTRUMENTED CHARPY PENDULUM

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**Abstract.** *The dynamic fracture toughness ( $K_{Id}$ ) is a mechanical property that many times must be known during the design of mechanical structures subjected to dynamic loadings. A laboratory test that gives a precise assessment of this property is the instrumented Charpy impact test. The force vs. time record that is obtained from an instrumented test is a complex combination of: the true specimen response, inertial effects, vibration and elastic energy associated to the pendulum. The treatment of the vibration and elastic energy that comes from the pendulum presents some difficulties although they are well understood, and the treatment of these effects is generally based in the filtering of the signal emitted by the load cell that is placed in the pendulum hammer. This filtering process can be as problematic as the oscillations themselves because it can induce signal distortions. Viscoelastic dynamic absorbers of vibrations have been employed for passive control of vibrations and noise originated in mechanical systems. These devices are widely used in vibration control due to its high capacity to introduce reaction loads and dissipating vibration energy. When attached on geometrically complex structures these devices allow making a very efficient vibration passive control in a wide frequency band. The subject of this work was to minimize vibrations in an instrumented Charpy test machine to a level that their influence be negligible in the force vs. time record obtained from a test. A vibration passive control was implemented in the Charpy impact test machine tuned, by dynamic absorbers, at frequencies with larger influence in the obtained signal. In this way, more realistic dynamic fracture toughness results could be obtained. The employed methodology, developed by the GVIBS/CNPq research group, as can be seen in [1], [2] and [3], consisted of the following steps: modal analysis of the pendulum by finite element model; force vs. time test record analysis to define the frequency range of interest; numerical model validation through the experimental measurements; design of the viscoelastic dynamic absorbers, and, last, force vs. time record analysis when the passive vibration control was implemented.*

## 1 INTRODUCTION

In structural analysis and machine design the real mechanical behavior of the employed materials is necessary to know. Many mechanical components undergo dynamic loading as shocks and impacts. Then the assessment of the dynamic response of materials is very important, especially for those that are sensible to temperature and strain rate and can present brittle fracture when loaded dynamically.

The instrumented impact Charpy test allows a good and reliable dynamic fracture toughness determination because it imposes a high strain rate to the specimen. The obtained force vs. time record is a combination of the true material response, inertial effects, vibration effects and the elastic deformation energy associated to the Charpy testing machine, plus high frequency noise generated in the measurement chain. Then, all these effects must be known and separated for a precise characterization of  $K_{Id}$ .

Viscoelastic dynamic absorbers (VDAs) are commonly employed nowadays for noise and vibration control in mechanical systems. This is as a consequence of its capability to introduce reaction loads and dissipate vibratory energy in a frequency band where one or several natural frequencies are present. This type of control is more and more employed because its simplicity, constructive easiness and design advantages. They can reduce the vibration amplitude in a wide frequency range in complex geometries with a high efficiency [1].

The vibrations present in a force vs. time record obtained in an instrumented Charpy test are proposed to be reduced in this work. The methodology employed is the passive control of the characteristic modes of vibration in an instrumented Charpy test. Viscoelastic dynamic absorbers were employed to this purpose.

## 2 MATERIAL AND METHODS

### 2.1 Numerical model of the Charpy testing machine

The studied Charpy testing machine is owned by the Mechanical Properties Laboratory of the National University of Comahue (Neuquén – Argentina). The machine is an Avery Denison 6703/A/31042 model with 170/300 Joules (J) of capability. The numerical modeling by finite elements was performed using a commercial software Ansys® (version 12.1) using the Workbench module.

Two numerical models were developed. One of them was used for the Charpy pendulum and the other for the Charpy hammer. The mechanical properties used in the analysis are described in Table 1.

Model	Material	Young Modulus (GPa)	Poisson Ratio	Mass Density ( $\text{kg/m}^3$ )
Pendulum	Structural Steel	207	0,292	7850
Hammer	Tool steel	203	0,285	7860

Table 1: Mechanical properties considered in the numerical analysis.

### 2.2 Charpy pendulum

The 3D pendulum model is shown in Figure 1 (a). Some simplifications were implemented in the numerical model, according to Figure 1 (b). The pendulum head and the hammer, that are a lot more rigid than the reticulated pendulum structure, were considered as a point mass rigidly attached to the structure, Figure 2.

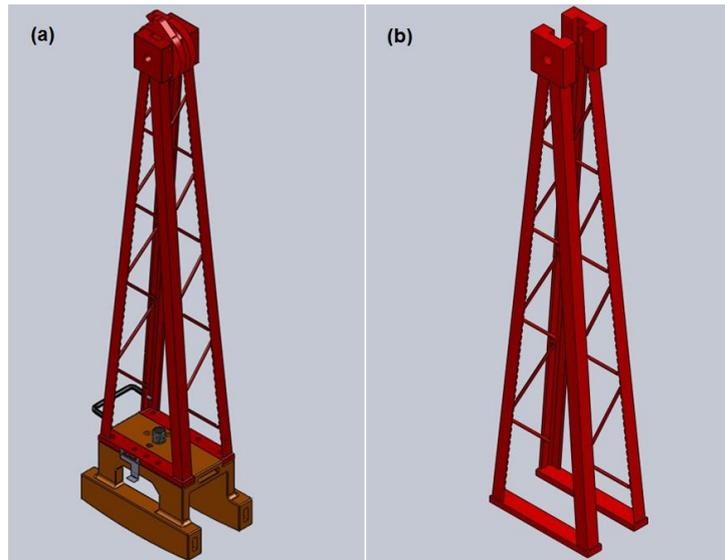


Figure 1: (a) 3D Charpy pendulum model; (b) simplified model.

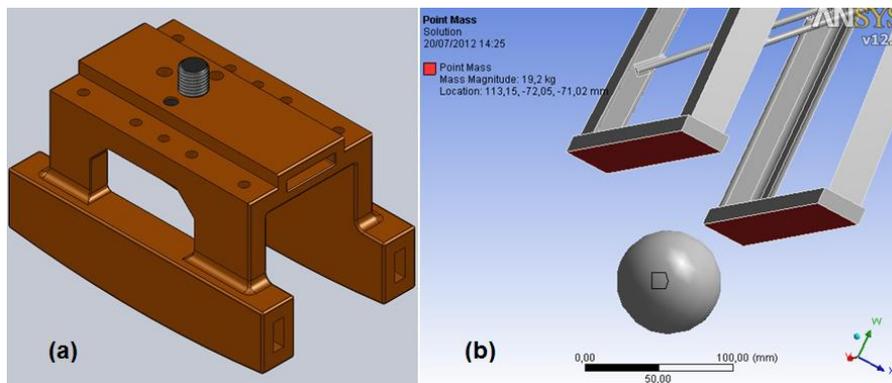


Figure 2: (a) Pendulum head plus hammer; (b) equivalent mass point.

The adopted boundary condition (BC), shown in blue in Figure 3, is known as cylindrical support, which have a free tangential degree of freedom and the axial and radial directions are constrained.

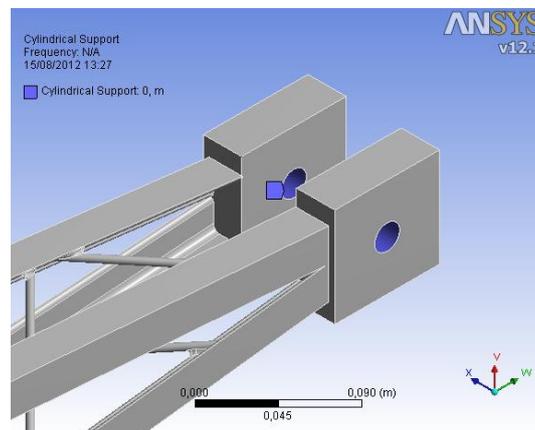


Figure 3: Surfaces where boundary condition was applied.

A detailed study on the size of elements and refining regions was performed for the model discretization, based on of the experience of the GVIBS/CNPq research group in finite element method (FEM). For this purpose, several simulations were performed taking into account the computational cost and the gain in terms of accuracy. Thus, a solution with high reliability and at the same time with a relatively low computational cost was obtained. Figure 4 shows the optimal model discretization obtained for the Charpy pendulum.

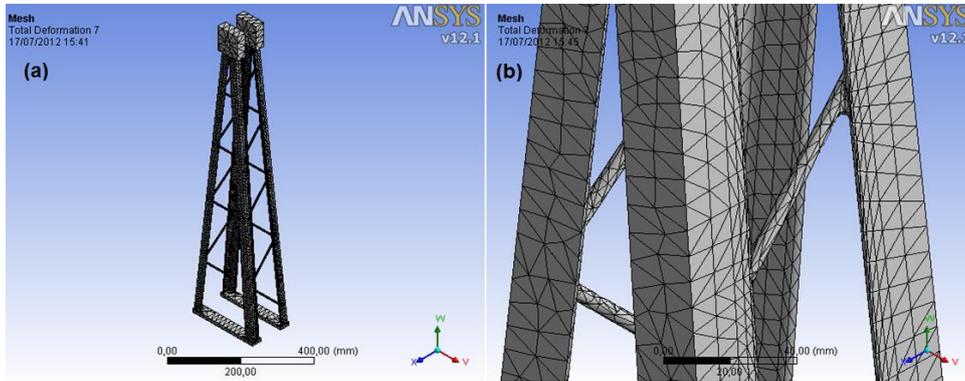


Figure 4: (a) Mesh model of the Charpy pendulum; (b) detail of the mesh model.

### 2.3 Charpy hammer

The 3D Charpy hammer model is shown in Figure 5 (a). Simplifications were also made for the numerical analysis, as shown in Figure 5 (b). A fixed support condition, shown in Figure 6 as a blue face, was adopted as the boundary condition. The procedure for meshing the Charpy hammer was similar to that adopted in the discretization of the Charpy pendulum. Figure 7 shows the hammer mesh model.

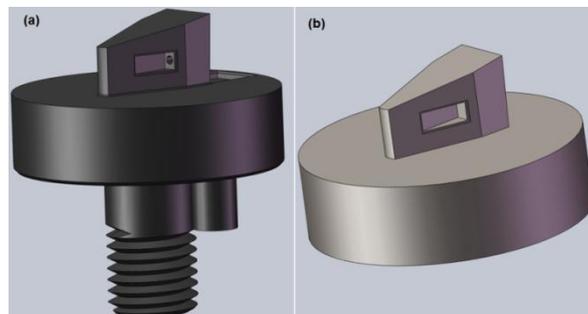


Figure 5: (a) 3D Charpy hammer model; (b) simplified model.

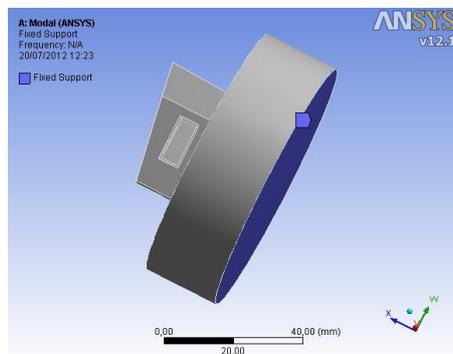


Figure 6: Surface where boundary condition was applied.

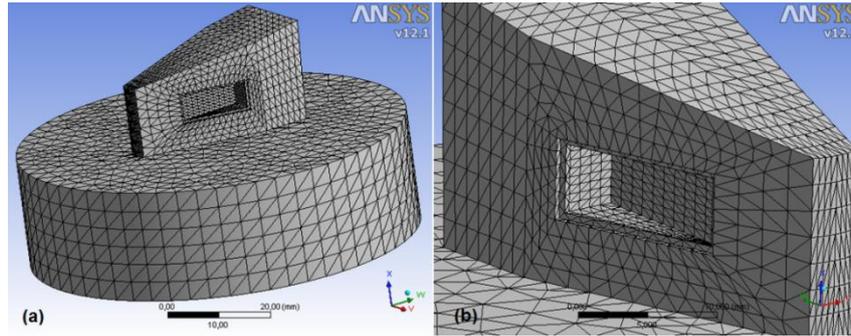


Figure 7: (a) Mesh model of the Charpy hammer; (b) detail of the mesh model.

Note: The simplifications adopted in the numerical analysis of the two models (pendulum and hammer systems) did not affect in any way the reliability of the analysis. Numerical studies with different degrees of simplifications showed that the simplified models perfectly characterized the originals ones.

#### 2.4 Calibration of the numerical model of the Charpy pendulum

The response of an accelerometer positioned at the arm near the pendulum support to a strike excitation was measured to validate the numerical model of the Charpy pendulum. The hit was produced at the end of the pendulum with a non-instrumented hammer. This validation of the numerical model was performed in this way due to lack of vibration measurement equipment.

#### 2.5 Analysis of the force vs. time curves

Once the modelling and calibration steps were completed, different response curves from instrumented Charpy tests (force vs. time records) were obtained and analyzed. The curves corresponded to brittle materials tests since, according to literature, the effects of vibration and elastic strain energy associated to the instrumented Charpy test machine are more significant in brittle materials.

This stage of the work had the following objectives: to know effects of pendulum vibration over the response of the tested specimen, to identify the natural frequencies excited during the test and its characteristic modes of vibration, and thus to define the frequency range of interest, besides the positioning of the control device being developed.

### 3 RESULTS AND DISCUSSION

#### 3.1 Charpy pendulum

The two first characteristic transversal vibration modes of the pendulum are shown in Figure 8, with values of : 24.79 Hz for the first mode, Figure 8 (a); and 284.34 Hz for the second mode, Figure 8 (b).

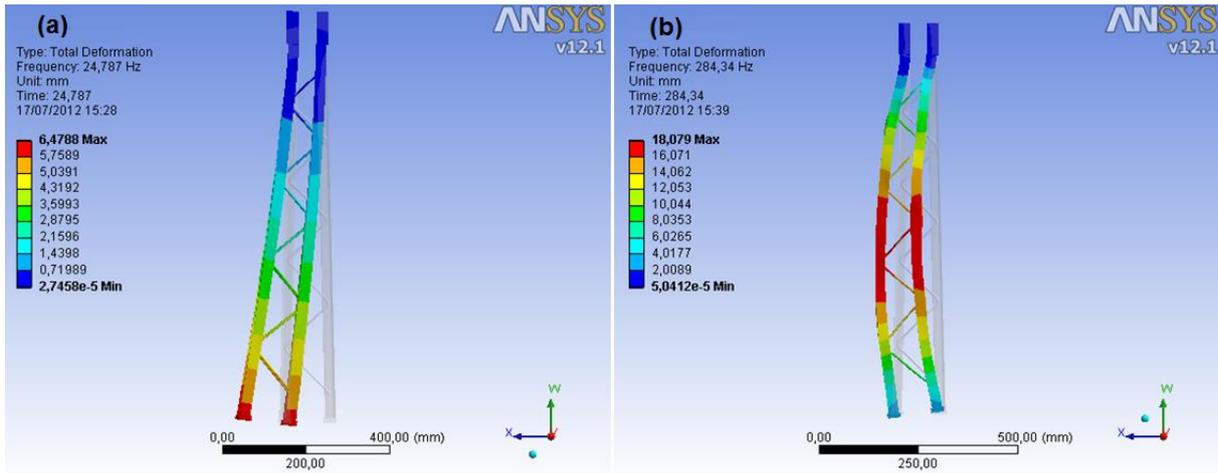


Figure 8: Modal shapes of the Charpy pendulum for natural frequencies: (a) 24.79 Hz; (b) 284.34 Hz.

The two first characteristic vibration modes of the pendulum in the movement direction are shown in Figure 9 with the natural frequencies of: 106.02 Hz for the first mode, Figure 9 (a); and 246.45 Hz for the second mode, Figure 9 (b).

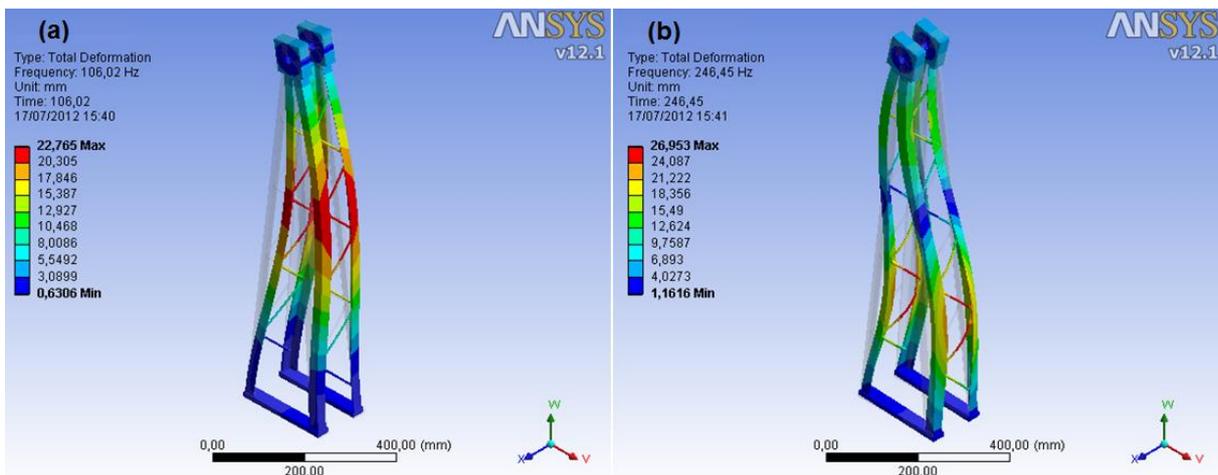


Figure 9: Modal shapes of the Charpy pendulum for natural frequencies: (a) 106.02 Hz; (b) 246.45 Hz.

Figure 10 shows the record, in the frequency domain, obtained to validate the pendulum modal numerical model. This curve corresponds to the acceleration response measured by the accelerometer fixed in the hammer, when the pendulum was subjected to an impact in an extreme. Although the curve shown in Figure 10 is not a typical frequency response function for typical modal analysis, but the information of the natural frequencies, taking account the quality which the impact load was applied, are present. Then, the natural frequencies experimentally measured, according to Figure 10, are: 25.83 Hz, 103.70 Hz, 236.30 Hz and 287.00 Hz.

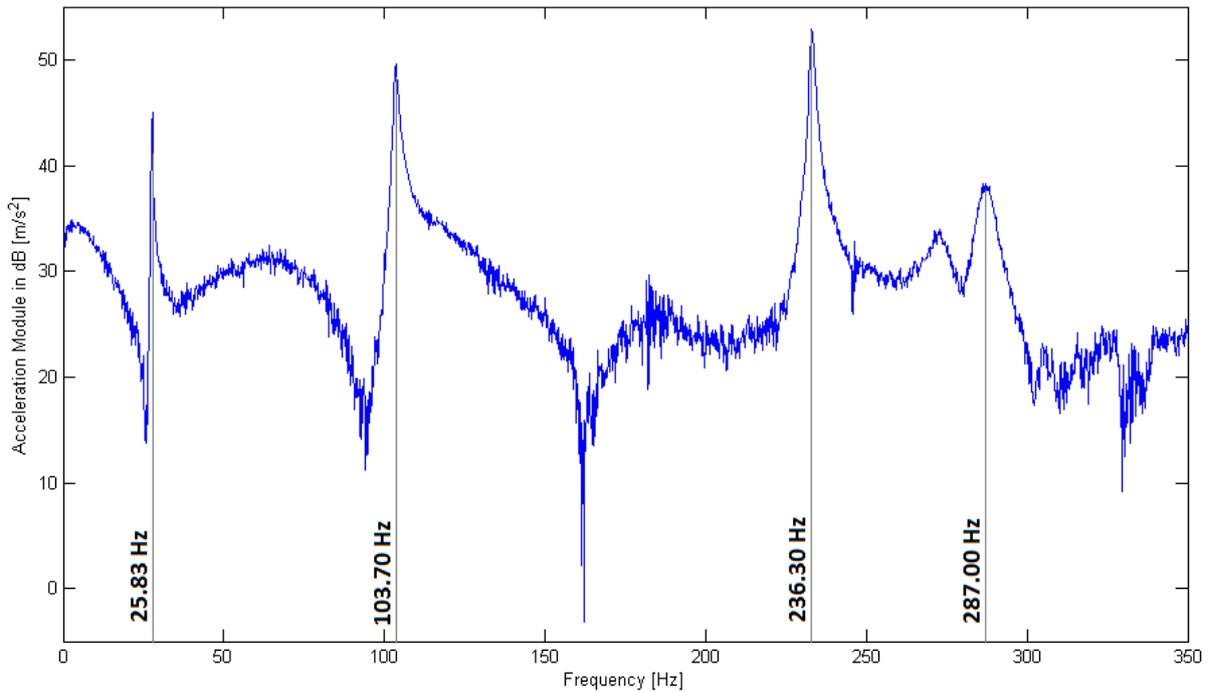


Figure 10: Response Charpy pendulum when subjected to the impact hammer blow.

Table 2 presents a comparison of the modal parameters (natural frequencies) obtained from the numerical model with those experimentally measured (Figure 10). The numerical model is considered verified because the errors for the four frequencies shown in Table 2 are lower than 5%.

Mode Shape	Experimental Frequency (Hz)	Numerical Frequency (Hz)	Error (%)
Figure 8 (a)	25.83	24.79	4.03
Figure 9 (a)	103.70	106.02	2.24
Figure 9 (b)	236.30	246.45	4.30
Figure 8 (b)	287.00	284.34	0.93

Table 2: Numerical and experimental natural frequencies comparison.

### 3.2 Instrumented Charpy test response curve (force vs. time record)

The frequencies corresponding to 103.7 and 236.3 Hz were expected to be measured because they correspond to the two first vibration modes in the direction of the pendulum movement.

Figure 11 shows one of the obtained force vs. time record. Figure 12 shows the frequency spectrum corresponding to this test, after the specimen fracture. Frequencies of 23.4 kHz, 46.7 kHz and 57.4 kHz can be identified. All the analyzed curves presented similar behavior to that shown in Figure 11.

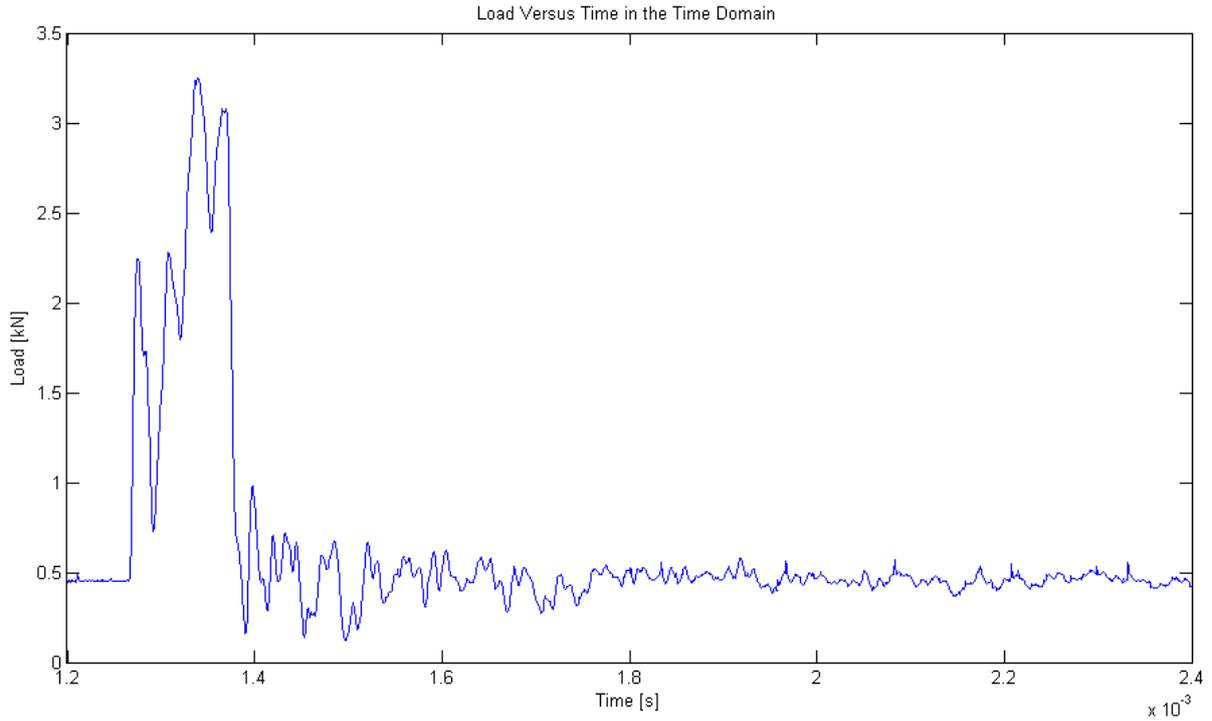


Figure 11: Force vs. time record without neutralizer.

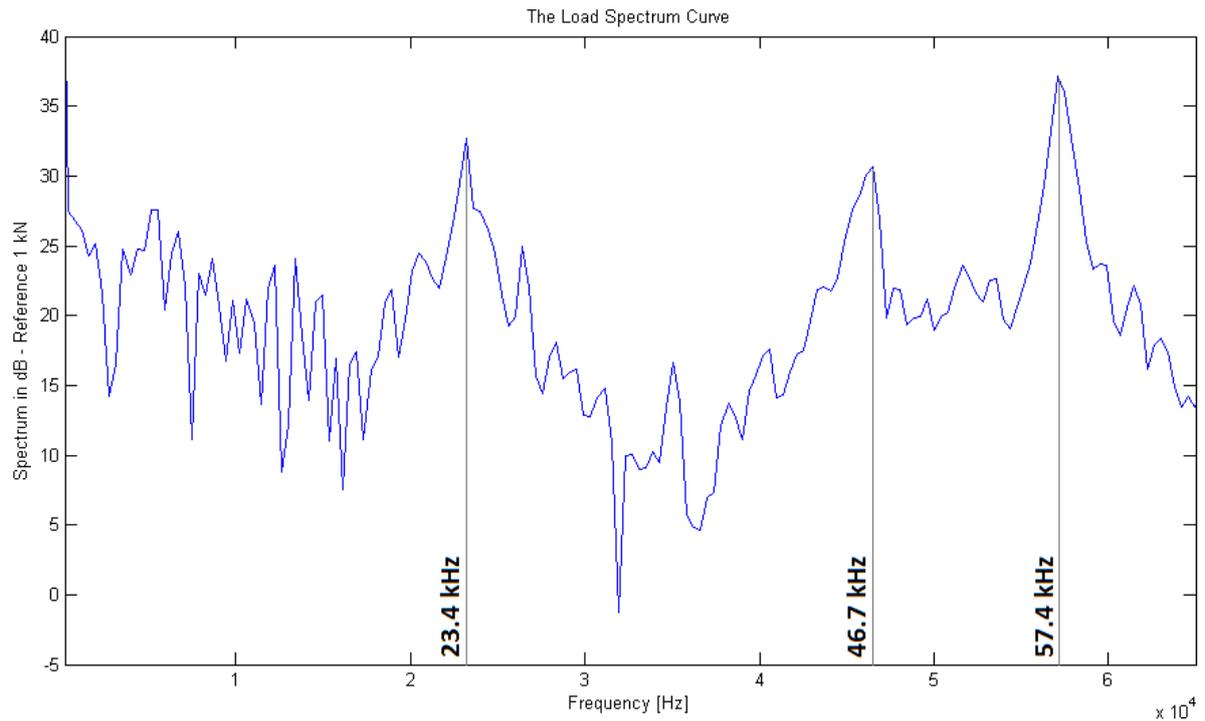


Figure 12: Spectrum of the force vs. time record – reference 1 kN.

This means that the oscillation frequencies observed in the force vs. time records do not correspond to the pendulum vibrations because they are many times higher (two orders of magnitude). In this way, and contrary to that stated in literature [4] [7], the oscillation problem in the force vs. time records is not related to the natural frequencies of the Charpy pendulum structure, at least for the analyzed records.

From the expressed above, three assumption were discussed: stress waves reflected by the hammer faces and captured by the strain gages, interaction between the specimen and the hammer, and vibrations of the hammer itself that are sensed by the strain gauges. The first assumption was rejected immediately because the calculated frequency for stress waves reflected by the wedge hammer faces is around 170 kHz, a lot higher than the measured ones.

The second one was also rejected because the oscillations continued after the specimen fracture. In this way, the third assumption was availed by means of a finite elements analysis of the hammer.

### 3.3 Charpy hammer

In the Charpy hammer modal analysis three natural frequencies were identified that corresponded to the observed ones in the experimental records (force vs. time records). The three identified natural frequencies (24; 50 e 59 kHz), are shown in Figure 13.

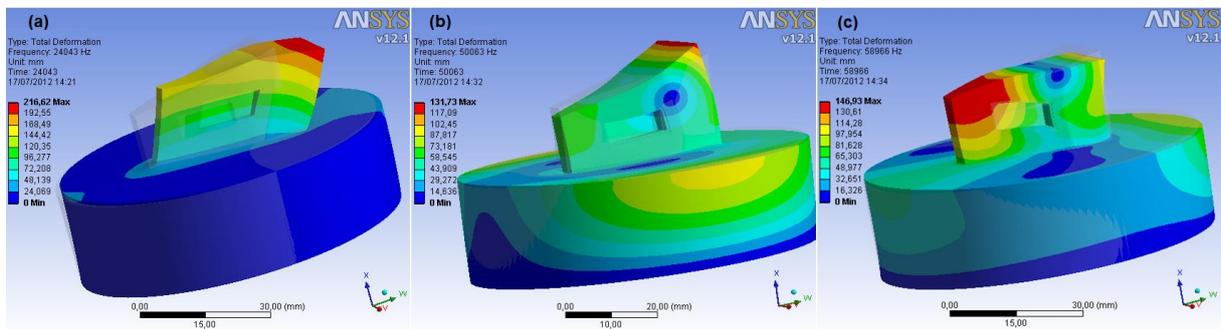


Figure 13: Modal shapes of the Charpy hammer for natural frequencies: (a) 24 kHz; (b) 50 kHz; (c) 59 kHz.

The strain gages located as shown in Figure 14 are capable to measure deformations in a large frequency band. The captured deformations associated to vibration modes correspond to frequencies of 23.4; 46.7 and 57.4 kHz that are very close to the calculated three natural frequencies (24; 50 e 59 kHz) of the hammer.

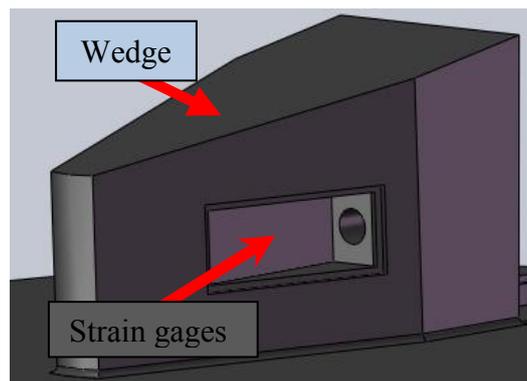


Figure 14; Fixing scheme of strain gages on the wedge Charpy hammer.

In this way, the third assumption is considered verified: vibrations of the hammer itself that are sensed by the strain gages. Consequently, the use of passive control by means of an adequately designed VDAs is the proposed solution to reduce the vibrations.

### 3.4 Design of the viscoelastic dynamic absorbers

The selected viscoelastic material was EAR C2003 whose dynamic characteristics are shown in the nomogram of Figure 15. The first frequency identified in Figure 12 (23.4 kHz) was selected as the project frequency (tuning). This selection, according to the GVIBS/CNPq experience, is based in the fact that controlling this frequency, the others higher are also controlled. This non-optimal control allows verifying this device capability, especially because the frequency range is very high for this type of passive actuators.

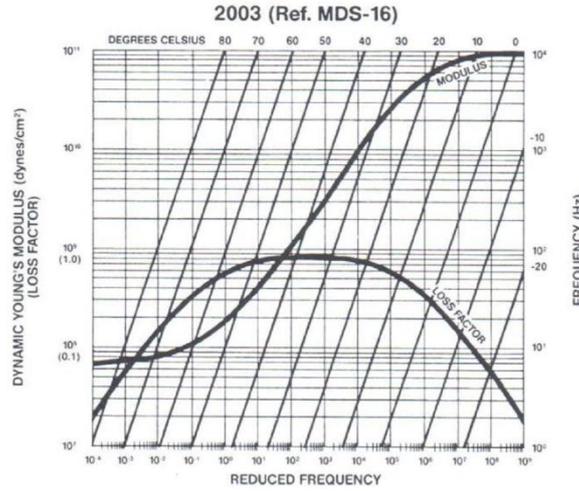


Figure 15: Nomogram of the EAR C2003 material.

In this way, the obtained Young modulus, for a frequency of 23.4 kHz and 25°C, was 5.9 GPa. The adopted shear modulus was  $G=E/3=1.97$  GPa.

The VDA mass of 22g was selected based on GVIBS/CNPq knowledge and experience (approximately 2 % of the Charpy hammer mass).

The antiresonance frequency of a viscoelastic dynamic absorber with one freedom degree is given, according to literature [1] and [2], by the expression:

$$\Omega_a = \sqrt{\frac{LG(\Omega_a)}{m_a}} \quad (1)$$

where  $L$  is the called VDA geometrical factor and possesses length dimension,  $G(\Omega_a)$  is the real component of the viscoelastic material shear modulus for the absorber natural frequency and  $m_a$  the absorber mass. In this way, when the excitation frequency equals that given by Eq. (1), both the dynamic rigidity and the dynamic mass, measured at the absorber base, adopt elevated values [1] Following this, the absorber was designed with a natural frequency equal to that of the system.

The geometrical factor is defined for the pure shear condition as [2]:

$$L = A/h \quad (2)$$

where  $A$  is the sheared area and  $h$  the height between loaded areas.

In this way, putting frequency, shear modulus and mass values into Eq. (1), the geometric factor ( $L$ ) can be obtained. The sheared area is obtained by Eq. (2) defining the height between the loaded areas (thickness of the viscoelastic material) as 1.75 mm.

Table 3 presents dimensions and characteristics of the designed absorber. The 3D VDA model and its assembling in the hammer are shown in Figure 16.

Design temperature	25 [°C]
Design Frequency ( $\Omega_a$ )	23,4 [kHz]
Mass of the VDA ( $m_a$ )	22 [g]
Geometric factor ( $L$ )	0,242 [m]
Height between load areas ( $h$ )	1,75 [mm]
Loaded area ( $A$ )	423 [mm <sup>2</sup> ]

Table 3: Characteristics and physical dimensions of the VDA.

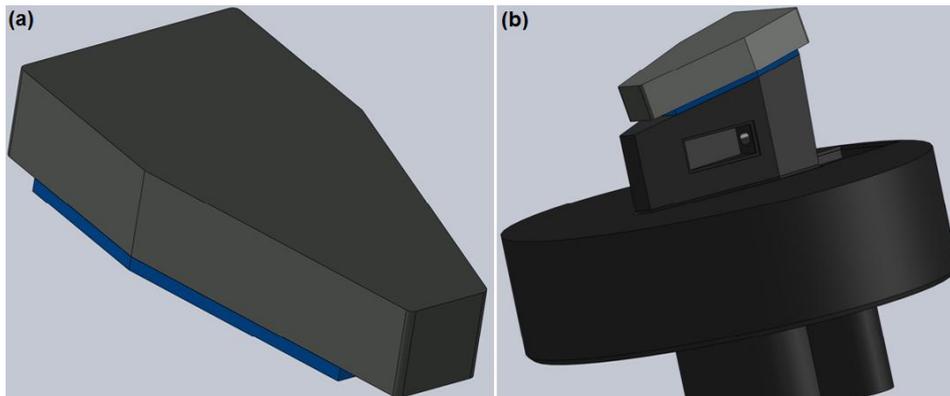


Figure 16: (a) 3D Viscoelastic dynamic absorber model; (b) compound system (hammer + VDA).

Once constructed, Figure 16 (a), the VDA was mounted on the Charpy hammer according to Figure 16 (b). Then, new force vs. time records were raised.

### 3.5 Force vs. time record with VDA

A of force vs. time record of an instrumented Charpy impact test with VDA attached to the wedge hammer is presented in Figure 17, and the corresponding spectrum is shown in Figure 18.

Figure 17 shows that the oscillations in the stage following specimen rupture were reduced. Another observation, as expected, is that the fluctuations in the record before the specimen rupture, which determines the dynamic fracture toughness, were also significantly reduced.

The Figure 19 shows the curves (force vs. time record) with and without dynamic absorber. A comparison between both curves shows the efficiency of this control device.

Even doing a simple test, it is possible to see that the vibration control can change the impulse quantities (the area under the curve force vs. time), which can change the measured magnitude that is related to the dynamic fracture toughness of the material under test.

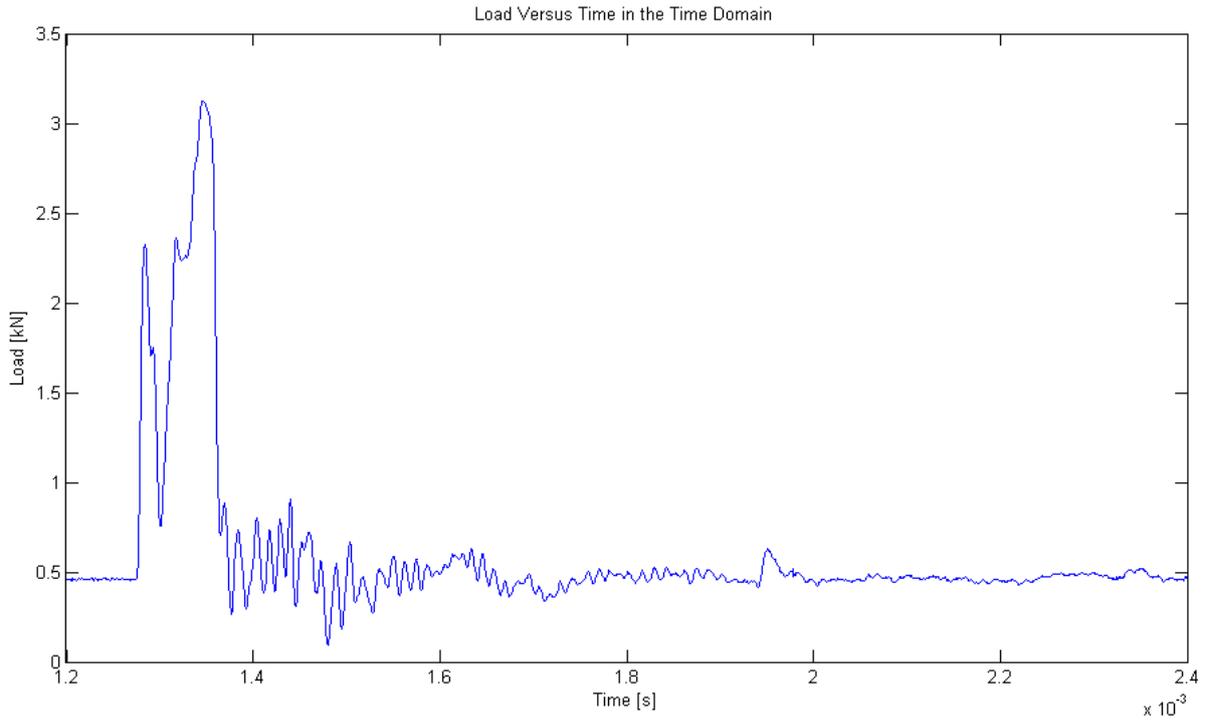


Figure 17: Force vs. time records with and dynamic absorber.

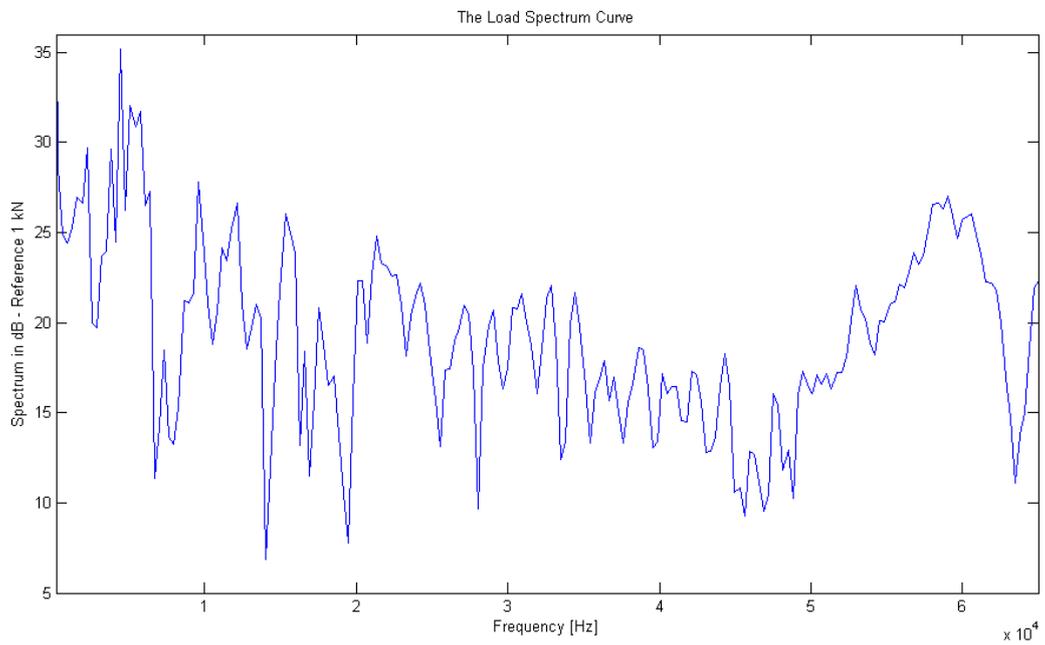


Figure 18: Spectrum of the force vs. time record with DVA – reference 1 kN.

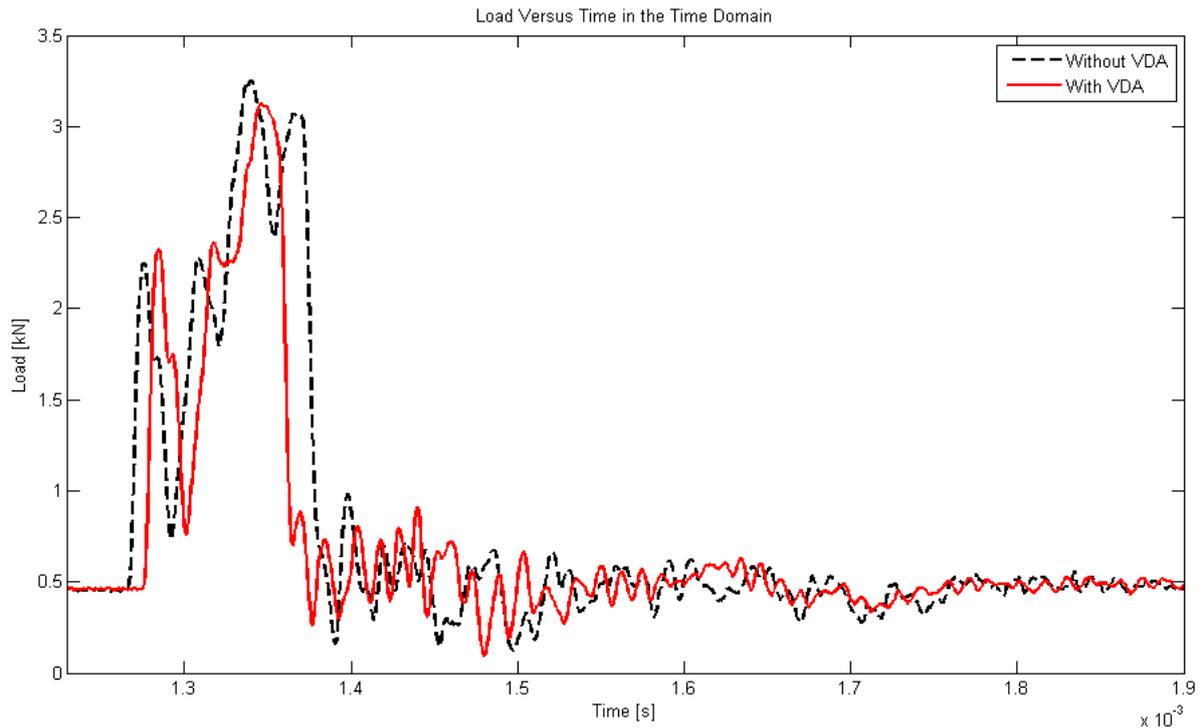


Figure 19: Force vs. time records with and without dynamic absorber.

#### 4 CONCLUSIONS

A numerical model of a Charpy pendulum was presented, and even with the lack of adequate equipment, experimentally validated. From the analysis of the obtained force vs. time records it was verified that some vibration modes in the hammer were excited during the test. A modal numerical analysis of the hammer was performed and its natural frequencies experimentally verified.

A viscoelastic dynamic absorber (VDA) was designed and built to control the vibrations in the impact hammer.

The designed VDA, although not optimized, showed able to reduce the vibration levels which allows a better force versus time records to obtain the dynamic fracture toughness.

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