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Evaluation of the modal parameters of two shafts, one manufactured with a composite materials and the other manufactured with structural Steel, through the implementation of a technique of Operational Modal Analysis (OMA)

Evaluación de los parámetros modales de dos ejes, uno fabricado con materiales compuestos y otro en acero estructural, mediante la implementación de una técnica de Análisis Modal Operacional (OMA)

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ABSTRACT

The following paper focuses on obtaining the modal parameters of two shafts in operation. The research has been planned using a shaft with composite materials, carbon fibers in an epoxy matrix and an axis manufactured with structural steel A36. The objective is to assess and validate the implementation of a technique of Operational Modal Analysis (OMA) in axes of different materials, and at the same time, compare the dynamic answer of the elements involved in the research.

Keywords:

Operational Modal Analysis, modal trial, composite materials, composite shafts, modal parameters.

RESUMEN

El siguiente trabajo se centra en la obtención de los parámetros modales de dos ejes de transmisión de potencia en operación. El estudio está planteado en un eje fabricado con materiales compuestos, fibras de carbono en una matriz epóxica y un eje en acero estructural A36. El objetivo es, evaluar y validar la implementación de una técnica de Análisis Modal Operacional (OMA) en ejes de diferente composición estructural, y a su vez, comparar la respuesta dinámica de los elementos objeto de estudio.

Palabras clave:

Análisis Modal Operacional, Ensayo Modal, Materiales compuestos, Ejes compuestos, parámetros modales.

Introducción

In mechanical engineering, the study of power transmission shafts is essential, since they allow communication and the operation of the different mechanical components of a team. They are generally elements that are subjected to high torsional and shear stresses [1].

In view of this, and given the high industrial competitiveness that is present today, it is imperative to develop technologies that improve the response in relation to the effort and efficiency requirements in these mechanical components. Given this, and given the high industrial competitiveness that is present today, it is imperative to develop technologies that improve the response about the effort and efficiency requirements in these mechanical components. [4].

In this context, rotors with higher performance, high energy efficiency, and low weight are of greater interest in the industry [5]. In addition, performing vibration

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analysis and monitoring of mechanical equipment will extend its useful life and avoid catastrophic damage. Given this, the implementation of numerical models that allow having a vibratory characterization of these mechanical elements, and that generate efficiency in the operation of rotating systems, should be practical and profitable.

Currently, different techniques allow the dynamic characterization of equipment, in terms of the identification of its modal parameters (natural frequencies, modes of vibration, and damping coefficients). One of these techniques and that is used in greater proportion is the Experimental Modal Analysis (EMA, Experimental Modal Analysis), where the Frequency Response Functions (FRF) of an element under study are acquired through excitation by an established force, which allows its dynamic characterization [6,7]. But, in the application of this technique, the equipment to be analyzed must be removed from its place of operation and in a controlled environment. [8, 9].

The application of an Operational Modal Analysis (OMA, Operational Modal Analysis), allows to dynamically characterize the axis without stopping its operation. It is a technique used in civil engineering [10, 11, 12] and extrapolated to mechanical engineering [13¸14¸15]. This considers it as a viable and useful technique at an industrial level since the stoppage of equipment hurts a production line.

Thus, this document reports the dynamic characterization of two shafts of different structural composition using an OMA technique, one manufactured by carbon fibers in an epoxy matrix and the other, a shaft in structural steel A36.

Methodology

The development of the research began with the adaptation and manufacture of a carbon fiber shaft in an epoxy matrix, where its requirement for torsional load and operating speed is based on a structural steel shaft A36.

Shaft was designed and manufactured with similar geometry and shear stresses adjacent to A36 steel. Once we have the two axes, an EMA was performed, to obtain the modal parameters of each axis. This test will be used as a standard for the results obtained in the OMA.

Following this, the non-parametric technique was applied and in the frequency domain "Peak Picking" [16], obtaining the graphs of Averaged Normalized Power Spectral Density (ANPSD), which allow identifying the natural frequencies and damping coefficients corresponding to each vibration mode.

Finally, once the respective modal tests had been developed, the results obtained were analyzed in terms of the identification of the modal parameters, evaluating the viability of the OMA technique in axes of different structural compositions.

Manufacture of the shaft in composite materials

As a first measure, a composite shaft reinforced with carbon fibers and a polymeric matrix, epoxy type, was designed based on the torsional requirements of an A36 structural steel shaft with the same diameter and length. The optimal design of the axis was carried out with the help of a genetic algorithm, implemented in MATLAB. This algorithm resulted in the orientations that the filament winding must-have, the number of layers to apply in the manufacturing process, the mass of the shaft, and the safety factor of the critical torque.

To obtain the torsional requirements of the steel shaft, the allowable torque was calculated using Equation (1).

$$T_{adm} = \frac{\tau_{adm} * \pi * r^3}{2} \quad (1)$$

Where:

T_{adm} = Allowable torque
τ_{adm} =Allowable shear stress
r = Shaft diameter

With the torque value calculated for the steel shaft [33.277 Nm], the diameter [0.0127 m], the length of 0.56 [m], and for an operating speed of the shaft [1800 rpm], the design and manufacture of a shaft by winding carbon filaments, joined by an epoxy-type polymer matrix in a mold and with a fiber orientation of +/- 45 °, as shown in Figure 1.

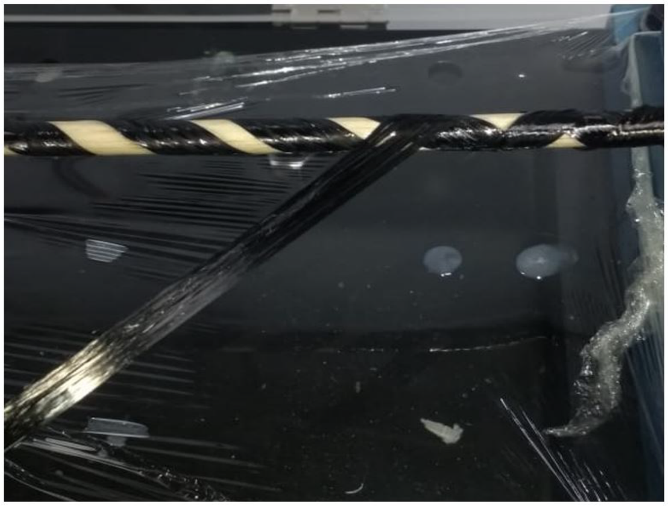

Figure 1. Manufacture of the shaft made of carbon fiber and an epoxy-type polymer matrix. Source: self-made

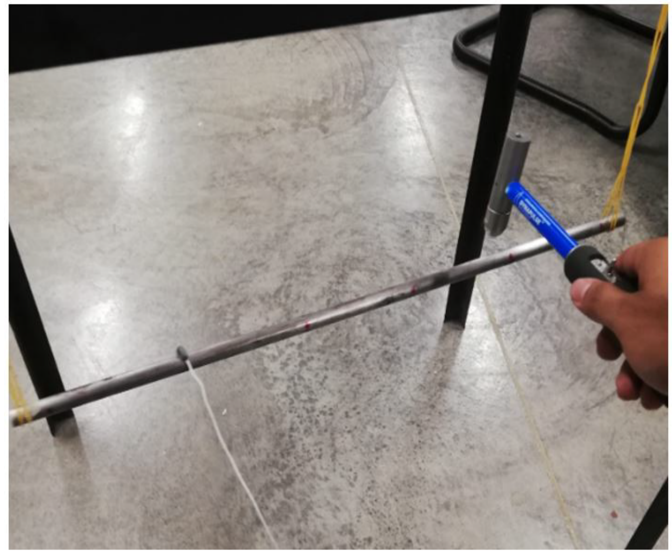

Figure 2. Experimental Modal Analysis (EMA), in free suspension condition for the A36 structural steel shaft. Source: self-made

Finally, the results obtained from the manufacture of the carbon fiber shaft were as follows:

• N ° of layers: 8
• Orientation of the fibers: +/- 45 °
• Approximate mass: 0.0101654 kg,
• Outside radius: 0.0127 m,
• Inner radius: 0.0082 m
• Critical Torque Safety Factor: 127,562

Execution of an Experimental Modal Analysis (EMA) in the two study axes.

The objective of the test is to obtain the modal parameters of the axis, that is, natural frequencies, damping, and modal shapes, through the Frequency Response Functions (FRF). To carry out the test, the free-free vibration condition is simulated, suspending the part (see Figure 2). The force is applied using a sensorized hammer and the response is measured with accelerometers.

Once the shaft is suspended, the points to be excited are defined (these points must largely cover the length of the study element).

The excitation is carried out with a Hammer model Dytran DYNAPULSE 5850B, configured with a sensitivity of 10,8mv/Lbf.

The vibratory response of the study element (the axis) is acquired through a uniaxial accelerometer model 3224B with sensitivity 11.51mv / g from the company DYTRAN Instruments Inc. The location of the sensor is evaluated so that it can record the modes of interest. to the best extent. An OR-35-Freq-8 data acquisition module from the OROS company is used as a data collector. The acquisition module is not only in charge of receiving the test signals, but also allows generating the graphical interface where the modal parameters of the axis and the configuration of the signal processing will be evidenced.

Once the modal parameters have been obtained through the EMA, with the help of a numerical model through finite element software, the experimental results will be theoretically contrasted.

Implementation of an OMA technique in the identification of natural frequencies and damping coefficients in the study axes

An algorithm focused on a non-parametric OMA technique called "Peak Picking" was executed. For this, two accelerometers (one in each axis support) were placed in the system, uniaxial reference 3056B2 from the company DYTRAN Instruments Inc, with a sensitivity of 99.68 mv / g and 100.44 mv / g, which deliver the vibration information of the axis understudy, to the data acquisition module OR-35-Freq-8 of the company OROS. This data is entered into the algorithm developed in MATLAB, and through digital signal processing, the ANPSD is obtained. There, the peaks with the highest energy content of the acquired data are identified that will be associated with the natural frequencies of the axis under study.

The coherence functions, calculated from the algorithm, allow validating these energy peaks as natural frequencies. This validation is based on data correlation. The transfer function or Frequency Response Function (FRF) describes the dynamic behavior of the system [H(ω)], is calculated taking as a reference an accelerometer that records the input information, ω[Xω_{ref} (ω)] and a second accelerometer that records the response of the system [output information, X(ω)]. Equation (2) represents this behavior [13].

$$|H(\omega)| = \frac{|X(\omega)|}{|X_{ref}(\omega)|} \quad (2)$$

The identification of the damping coefficients is based on the bandwidth method, where the damping fraction [ξ_n], is determined from the neighboring frequencies ξ_n [ω_2,ω_1] at resonant frequency [ω_n], [17]. Calculated according to Equation (3).

$$\xi_n = \frac{\omega_2 - \omega_1}{2\omega_n} \quad (3)$$

Results

Below are the results for the two axes under study.

A36 Structural Steel Shaft

Experimental Modal Analysis on A36 Structural Steel Shaft

In the first instance, a numerical model was implemented in a finite element software that allows estimating the natural frequencies of the axis, to plan the experimental test. Once planned, the experimental modal trial (EMA) was run.

Table 1 shows the results obtained by the EMA, for the first three-axis modes. Table 2 shows the comparison between the natural frequencies of the EMA, with those obtained from the numerical model through Ansys, once said model has been adjusted. The values obtained from the error, which measures the difference between the results, do not exceed 1.92% in any of the cases, which implies that the numerical model largely reproduces the physical model under this scenario.

Table 1. EMA Results on A36 Structural Steel Shaft.

Modes	Natural Frequencies [Hz]	Damping coefficients [%]
1	189,15	0,04
2	520,24	0,06
3	1016,20	0,05

Source: self made.

Table 2. Contrasting of theoretical and experimental natural frequencies.

	Finite element software	EMA	
Modes	Theoretical Natural Frequencies [Hz]	Experimental Natural Frequencies [Hz]	% of Error
1	185,63	189,15	1,9
2	510,45	520,24	1,92
3	997,04	1016,20	1,92

Source: self made.

Operational Modal Analysis in the A36 structural steel shaft

To carry out this test, the shaft was mounted in a Rotor kit with two supports at its ends, which will be the two measurement points. The assembly is evidenced in Figure 3.

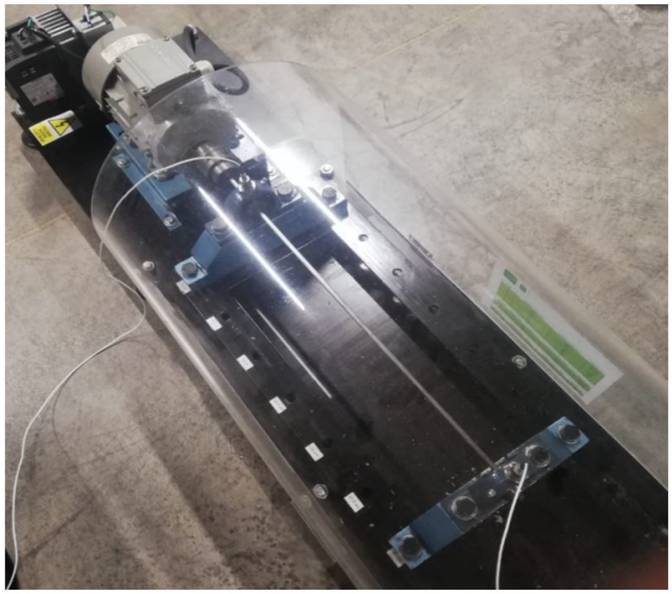

Figure 3. Test bench for the Steel Shaft. Source: self-made. Source: self-made
Source: self made.

Once the information from the two sensors was acquired, it was processed through the OMA (Peak Picking) algorithm and the ANPSD was obtained, which is evidenced in Figure 4.

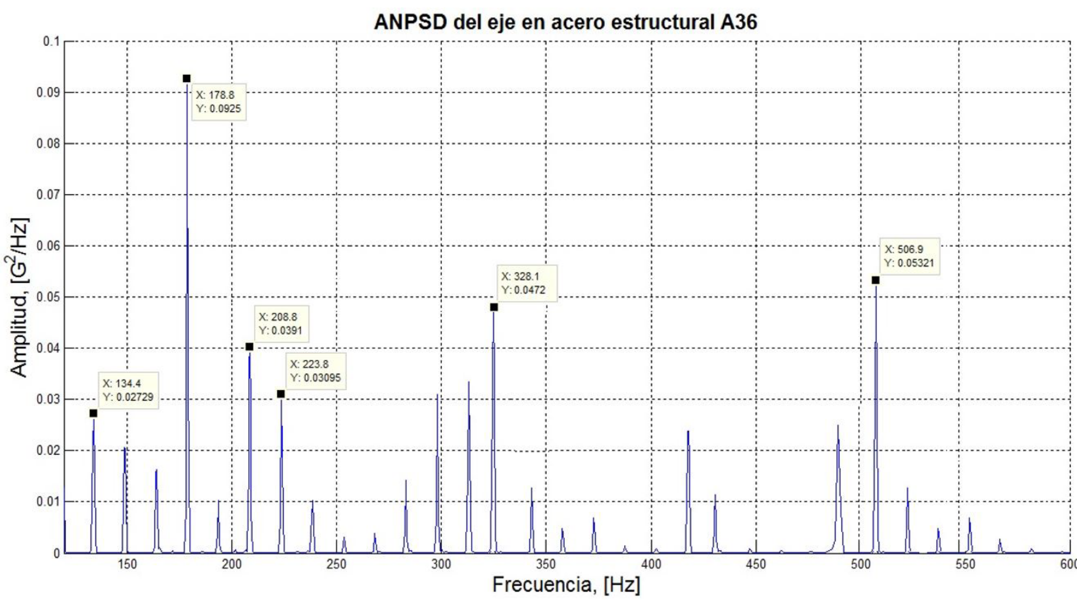

Figure 4. ANPSD of the A36 structural steel shaft. Source: self made.

The ANPSD of the steel, allowed to identify the first two peaks with the highest energy content of the signal in a frequency range between 60 Hz to 600 Hz, they are associated with the first two natural frequencies of the axis.

By obtaining the natural frequencies, the damping coefficients were calculated for each frequency, according to the equation (3).

Table 3 reports the two natural frequencies and the two damping coefficients associated with these frequencies, obtained by the OMA technique in the structural steel axis.

Table 3. OMA Results on A36 Structural Steel Shaft.

Modes	Natural Frequencies [Hz]	Damping coefficients [%]
1	178,8	0,279
2	506.9	0.099

Source: self made.

Comparison of theoretical results (EMA) vs Experimental (OMA) on the steel shaft.

A comparison of results was made regarding the natural frequencies and damping coefficients identified by the OMA technique concerning those of the EMA.

Table 4 shows the comparison of natural frequencies and their respective percentage of error.

Regarding the comparison of the results of the damping coefficients obtained in the experimental modal tests (OMA and EMA), evidenced in Tables 1 and 3; their values are in accordance with the system presented, and for both cases, they are less than 1%. Said comparison presents some difficulties when the related values are small because small differences imply high percentages of error.

Table 4. Comparison of the natural frequencies of the A36 structural steel axis.

Modes	EMA [Hz]	OMA [Hz]	% of Error
1	189,15	178,8	5,472
2	520,24	506,9	2,56

Source: self made.

Composite shaft

Experimental Modal Analysis on the carbon fiber axis

The same process recorded for the steel shaft was carried out. Figure 5 shows the EMA for the carbon fiber axis and the results obtained in this test are shown in Table 5.

Table 5. EMA results in the composite materials axis.

Modes	Natural Frequencies [Hz]	Damping coefficients [%]
1	112,1	5,31
2	307,06	4,95
3	583,09	4,82

Source: self made.

Figure 5. Experimental Modal Analysis (EMA), in free suspension condition for the axis of composite materials. Source: Self-made.

Operational Modal Analysis on the carbon fiber shaft

As in the previous case, the carbon fiber shaft was mounted on a kit Rotor with two supports at its ends, which will be the two measurement points, as shown in Figure 6.

Figure 6. Carbon fiber shaft test bench. Source: Self-made

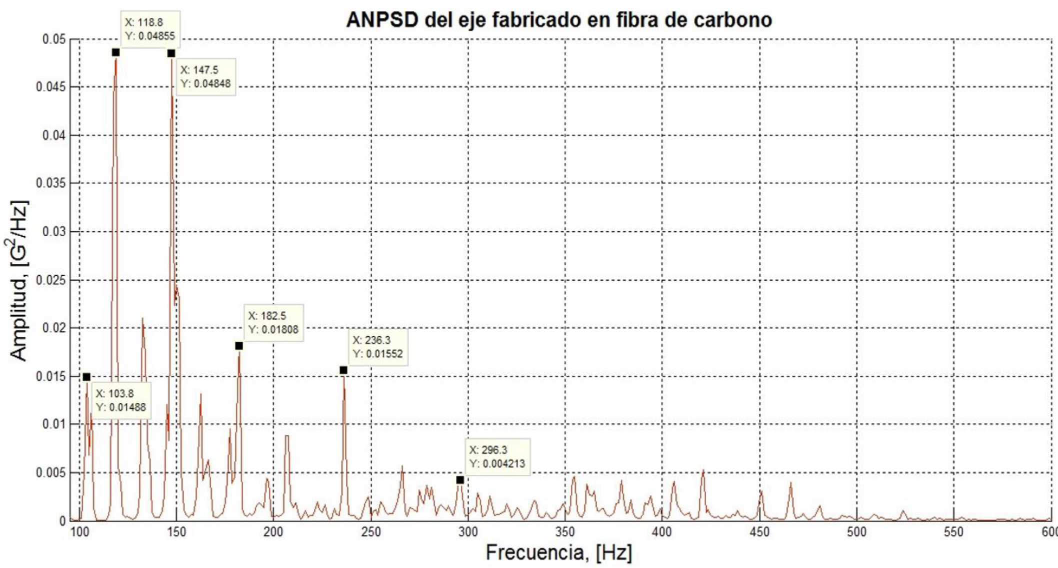

Figure 7. ANPSD of the shaft made of composite materials. Source: Self-made

For this case, the first natural frequency of the carbon fiber axis was identified, which is the first peak with the highest energy content of the ANPSD.

To calculate the damping coefficient, associated with the peak with the highest energy content in Figure 7, the neighboring frequencies were identified, zooming in on the peak of interest, as evidenced by Figure 8. The value of the damping coefficient was calculated from Equation (3).

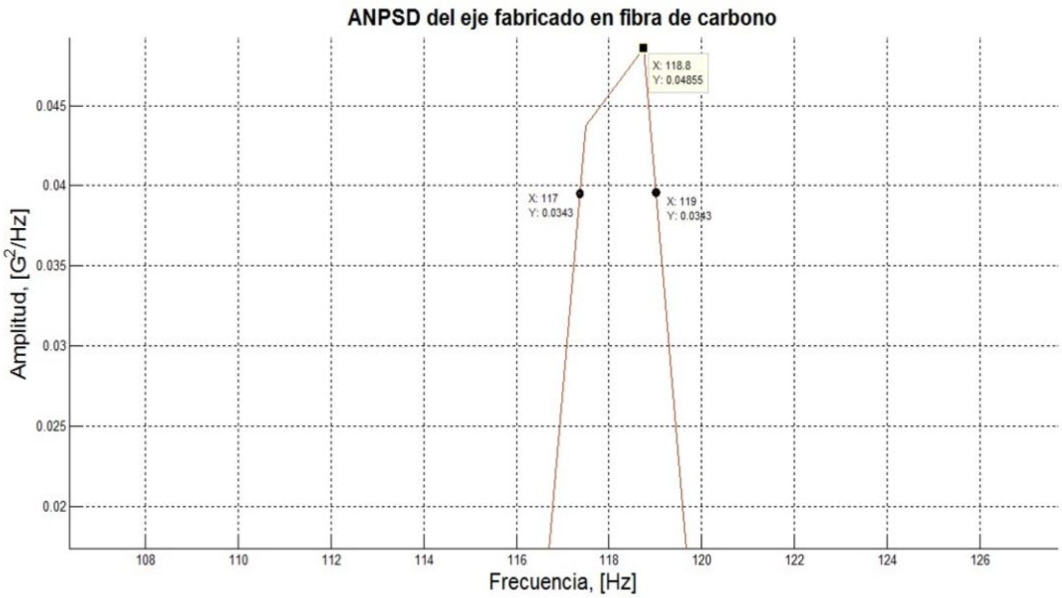

Figure 8. Identification of the neighboring frequencies associated with the first mode of vibration of the shaft made of composite materials.
Source: Self-made

This is how the first mode of said axis, made of composite material, has a natural frequency value of 118.8 Hz and a damping coefficient of 0.842%, estimated from the OMA

Comparison of Theoretical Results (EMA) vs Experimental (OMA) on the Carbon Fiber Axis

As in the previous case, a comparison was made of the results obtained from EMA and OMA. Table 7 presents the comparison of the natural frequencies and it is evident that the error is in an acceptable range, less than 6%. Table 8, reports the comparison of the results of the damping coefficients. In this case, the error is important and has to do with the interaction of the composite material shaft with the steel supports.

Table 7. Carbon Fiber Shaft Natural Frequency Comparison.

Modes	EMA [Hz]	OMA [Hz]	% of Error
1	112,1	118,8	5,98

Source: self made.

Table 8. Carbon fiber shaft damping coefficient comparison.

Modes	EMA [%]	OMA [%]	% of Error
1	5,31	0,842	84,14

Source: self made.

Conclusions

The Finite Element model developed by comparing the results obtained from the MEF with those of the EMA was validated. In natural frequencies, the maximum recorded error is less than 2%.

The OMA technique implemented for the identification of the first two natural frequencies of the structural steel axis mounted on the test bench described in this work was validated, through the comparison of the results obtained from OMA with those recorded from the EMA. The maximum error detected is less than 5.5%.

Regarding the calculation and identification of the damping coefficients using the OMA technique implemented, both for the steel shaft and for the composite material shaft, they differ greatly from the "theoretical value”. The influence that support systems have on said assessment becomes evident, especially when the rotor has a low mass compared to the components that support them. In this line, it is concluded that studies should continue to refine the evaluation of the damping coefficient of the system.

In the study of the present work, a shaft has been used whose mass is much less than a steel counterpart, and even much less than the elements that support it. In the same way, the transmission of vibration energy to the elements that support it was reduced, and therefore the difficulty of recording the vibration amplitudes of the shaft measured in non-rotating parts of the system increases. Thus, in Figure 7, the appearance of a series of peaks that are not related to the natural frequencies of the axis was evidenced and that generate a large energy contribution to the information obtained in the supports. Even so, it was possible to identify with an acceptable error percentage, the first mode of vibration of the composite materials shaft, validating the implementation of the technique in mechanical elements of said structural composition.

Based on the results and the analyzes carried out, it is established that the use of proximations is of interest for subsequent studies in order to acquire information on vibrations from different points of the axis, and generate a stabilization curve of results with respect to the points of measurement on non-rotating parts.

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