Numerical Modeling of a Vibration Test Platform with Imbalance CILAMCE-2023

Pedro Augusto de Almeida[1], Marcos Paulo Miranda Costa[2] Suzana Moreira Ávila[3]

[1]*Programa de Pós-Graduação em Sistemas Mecatrônicos - PPMEC, Universidade de Brasília - UnB*
Campus Universitário Darcy Ribeiro, Brasília-DF, 70910-900, Brasil
pedrogutojr@gmail.com
[2 3]*Faculdade do Gama - FGA, Universidade de Brasília - UnB*
St. Leste Projeção A - Gama Leste, Brasília - DF, 72444-240, Brasil
markosmcosta@hotmail.com

Abstract. The study of vibrations caused by imbalances is crucial in industry and engineering, as imbalance can cause vibrations and material fatigue. Motors exhibit vibrations due to imbalance, which can be caused by imperfections in manufacturing, design flaws and maintenance. This study analyzes the vibratory behavior of a sliding bearing on a rotating shaft with rotary imbalance through Finite Element simulations. The numerical results were compared to experimental results obtained on a bench using an accelerometer. The analyses were performed using MATLAB and Ansys software. The study aims to contribute to the understanding of vibrations in rotating systems and the development of prevention and correction techniques for imbalance. The comparison between experimental and numerical results enabled the calibration of the model in terms of its boundary conditions, particularly with regard to the supports of the piece. Initially, it was believed that a perfect fixed support would be sufficient, but the numerical results showed discrepancies in relation to the experimental results. Therefore, the system was modeled with the introduction of an elastic support, and through manual adjustments in stiffness, it was possible to obtain good correspondence between the numerical and experimental results, adapting the model of the bearing to its real counterpart.

Keywords: motor imbalance, numerical simulation, finite element method, experimental analysis.

1 Introduction

The study of vibrations due to imbalances is one of the fundamental areas of interest in contemporary industry and engineering. When an asymmetric part, either in terms of mass or geometry, begins to rotate, imbalanced radial forces arise capable of causing vibrations and fatigue of the materials (NIGRO [1]). According to RAO [2], most drive motors present vibrations as a consequence of their inherent imbalance. This observed imbalance, in turn, is attributed to imperfections in the manufacturing processes of the constituent materials, design flaws and/or poor maintenance.

In this context, the present work aims to carry out the modal, harmonic and transient analysis of a sleeve bearing coupled to a rotating shaft, whose rotation frequency is established by a single-phase induction motor. The vibratory behavior of the bearing exposed to a rotational imbalance was evaluated based on Finite Element simulations, while the numerical results were also compared with those obtained experimentally (on a Lima's vibration test platform [3]) using a piezoelectric accelerometer. The comparison allowed the numerical model to be calibrated in terms of the definition of its elements, the mesh and the boundary conditions to which it was submitted so that there was equivalence between the model and the bearing belonging to the test bench. The analyzes were conducted using the software MATLAB and Ansys.

2 Methods

The study was carried out in two main stages: The first consisted of collecting experimental data, associated with the bearing's vibratory response when the shaft to which it is coupled was subjected to rotational imbalance. The second stage comprised the calibration of the numerical model of the bearing based on data from the test bench, followed by the modal, harmonic and transient analysis of the system.

The experimental procedure began by calibrating the piezoelectric accelerometer using the portable accelerometer calibrator from PCB Piezotronics, model 394C06. After calibrating the piezoelectric accelerometer, the acceleration was measured to quantify the vibration at different measurement points. The data were acquired using the National Instruments data acquisition board, model NI9234 - 4 channels. The processed data from the board are sent to the computer for data processing in the LabVIEW software.

The data acquisition frequency was set at 2048 Hz, with a test duration of 10 seconds. The setup includes a 0.5 hp induction motor with a voltage of 220 Volts and a rotation frequency of 3520 RPM; two SKF ball bearing units, model SY 1" WDW [4]; a 1-inch shaft with an attached disk that generates the imbalance through sets of nut and bolt. The Fig. 1 illustrates the assembled test rig:

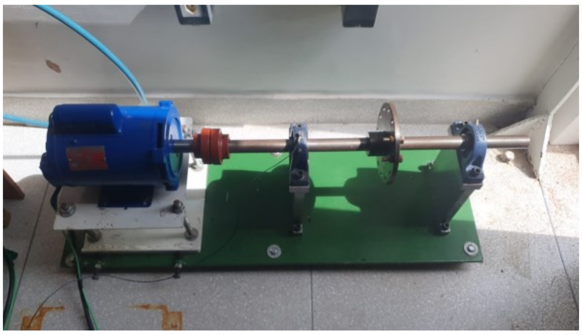

Figure 1. Vibration test platform with imbalance.

For the finite element numerical modeling, the Ansys software was employed. In this type of modeling, the most complex model doesn't always become the correlated one, as it requires evaluating the complexity vs. performance trade-off. Thus, two numerical models were created: one with only the cast iron bearing housing and another model incorporating the aluminum supports, as shown in Fig. 2. In addition to the geometry to be tested, the material properties are required, as shown in Tab. 1:

Table 1. Material properties

Property	Cast iron	Aluminum
Density	7400 kg/m³	2770 kg/m³
Young's Modulus	138 GPa	71 GPa
Poisson's Ratio	0,26	0,33
Yield Strength	250 MPa	280 MPa
Ultimate Strength	735 MPa	310 MPa

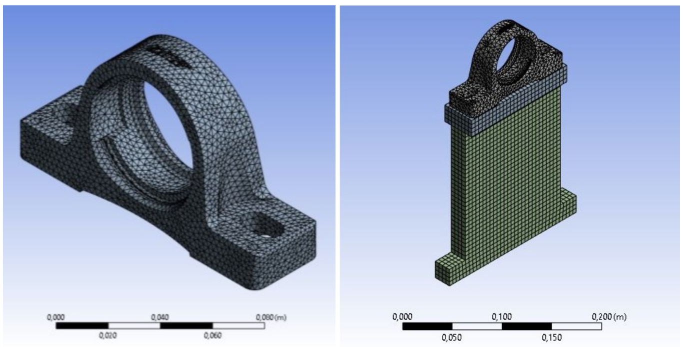

Figure 2. Numerical models.

For the first numerical model, tetrahedral elements were applied to the mesh, along with local refinement using a body sizing of 0.0025 m. In the second model, a combination of tetrahedral (for the bearing housing) and hexahedral (for the supports) mesh elements were used, limited to a body sizing of 0.005 m. It is worth noting that this refinement was adjusted to maximize the number of elements and nodes allowed by the student license.

Regarding the boundary conditions, two types of supports were used: fixed support and elastic support. In the modal analysis, six vibration modes were extracted. For the harmonic analysis, in addition to varying the supports, a force of 10 N was applied to the inner face of the top of the bearing, corresponding to the maximum amplitude extracted from the results of the experimental test. In the transient analysis, in addition to varying the supports, a harmonic force corresponding to the rotation of the shaft coupled to the motor was applied. This harmonic force follows eq. (1), where "A" represents the force amplitude (the same 10 N from the harmonic analysis), θ is the rotation frequency of the motor (3520 RPM or 58.67 Hz), and "t" is time in seconds, ranging from 0 to 10 s, with 500 sub steps.

$$F = A\sin(2\pi\theta t) \tag{1}$$

3 Results

The experimental data were measured at 4 different measurement points: bearing base, bearing top, support base, and motor base. Another modification was made to the system configuration: balanced or imbalanced. The result were 6 text files containing 2 columns: the first with time in seconds, and the second with acceleration in G. After data processing using MATLAB software, it was possible to plot each of the acceleration time history. Figure 3 illustrates the measured spectrum in the time domain for the case of the balanced system with an accelerometer set at the top of the bearing housing.

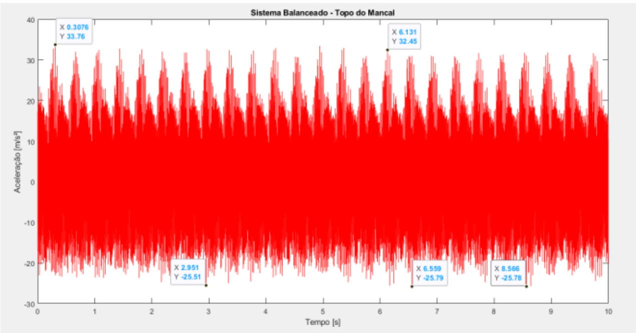

Figure 3. Acceleration [m/s²] vs. Time [s] for the Balanced System with Accelerometer at the Top of the Unit.

Subsequently, to obtain the frequency domain, the *pwelch* function was employed, which provides an estimate of power spectral density using a discrete time signal via Welch's method. The Fig. 4 illustrates the frequency domain estimated response for the case of the imbalanced system with an accelerometer at the motor base.

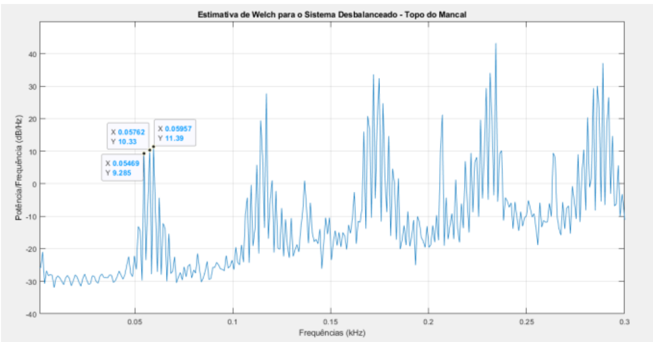

Figure 4. Power spectral density estimate for the Imbalanced System with Accelerometer at the Top of the Unit.

An interesting observation when comparing the two responses is the phenomenon of beating. In the time domain, as seen in Fig. 3, it manifests as a periodic variation in vibration, resulting in sound. This pattern arises because the natural frequency of the motor and the vibration frequency detected by the accelerometer are slightly different or very close. Such variation is visualized in the frequency domain by the presence of three initial peaks in Fig. 4.

For the remaining accelerometer measurement points, the data collection and analysis procedure were the same, so that the maximum acceleration amplitudes and their corresponding peak frequencies are shown in Tab. 2. Subsequently, the RMS values was calculated for each measurement point.

Table 2. Acceleration peaks for two of the measured points

Accelerometer on Support Base		**Accelerometer on Engine Base**	
Acceleration vs. Time – Balanced System		Acceleration vs. Time – Balanced System	
Positive peak	35.71 m/s²	Positive peak	48.54 m/s²
Negative peak	-31.05 m/s²	Negative peak	-51.41 m/s²
Acceleration vs. Time – Imbalanced System		Acceleration vs. Time – Imbalanced System	
Positive peak	41.69 m/s²	Positive peak	52.47 m/s²
Negative peak	-33.70 m/s²	Negative peak	-48.29 m/s²
Welch power spectral density estimate		Welch power spectral density estimate	
Positive peak	57.62 Hz	Positive peak	57.62 Hz
Negative peak	57.62 Hz	Negative peak	57.62 Hz

It's worth noting the close proximity of the measured frequency, 57.62 Hz, to the motor's rotation frequency of 58.67 Hz. Furthermore, it was observed that the introduction of imbalance into the system through the eccentric masses led to an increase in vibration peaks.

In the Ansys Mechanical environment, when incorporating the numerical modeling, the goal was to align each of the models with the experimental results. This alignment was achieved by adjusting the boundary conditions, modifying the supports, and altering the loading conditions of the analyses.

For the bearing housing model, it was observed that employing a perfect fixed support resulted in high natural frequencies through modal analysis. Furthermore, when determining the most influential mode through harmonic analysis, it yielded a mode different from what was observed experimentally.

The solution was to employ an elastic support with a specified stiffness. This approach rectified the discrepancies in modal and harmonic analyses. However, for transient analysis, it led to a divergent curve, as depicted in Fig. 5:

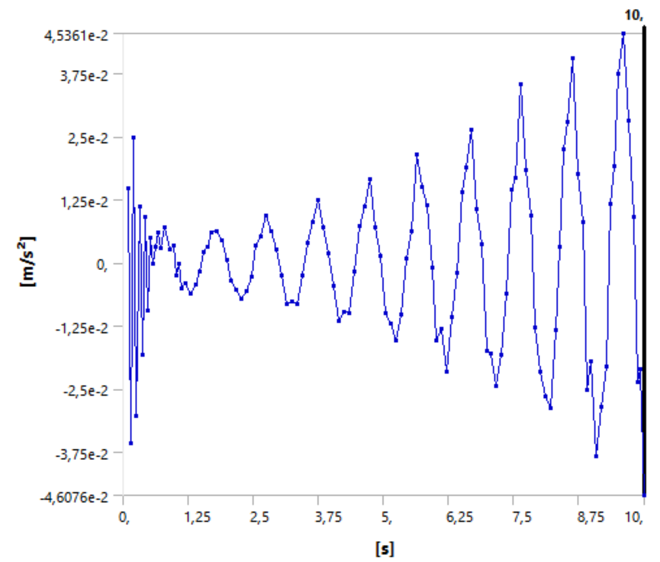

Figure 5. Vibratory response in the transient analysis of the first model with elastic support.

Using the same calibration procedure, the analyses were conducted again, this time with model 2, by varying the support and applying contact restraints between the components. For the perfect fixed support, the response values differed significantly from the experimental outcome. Once again, the solution was to apply the elastic support, which yielded the best correlation.

In both the modal and harmonic analyses, the relevant vibration mode was the one with a frequency of 57.62 Hz, matching the frequency estimated through experimental Welch analysis. Moreover, during the transient analysis, the steady-state response was more satisfactory and demonstrated the anticipated vibrational behavior, as illustrated in Fig. 6:

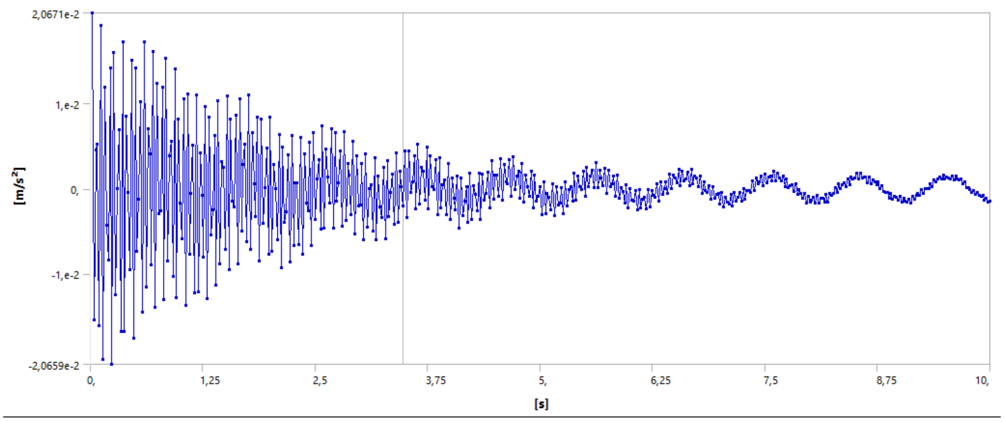

Figure 6. Vibratory response in the transient analysis of the second model with elastic support.

4 Conclusions

The present study aimed to conduct modal, harmonic, and transient analyses of a sleeve bearing coupled to a rotating shaft, evaluating its behavior in response to rotational imbalance. The calibration of the numerical model was performed through the analyses conducted adjusting boundary conditions, especially in terms of defining the supports for the component. It was observed that through model refinement, the virtual model can become correlated with the real-world behavior.

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