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EXPERIMENTAL MODAL ANALYSIS APPLIED TO A FOUNDATION STRUCTURE

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Abstract. *Rotating machines are associated with the great leaps in the technological and social development of humanity. The functioning of pumps, compressors, turbines and engines are critical factors to guarantee any productive process. Hence the need to ensure greater reliability for rotating machines. One of the factors that help in this regard is the characterization of foundation structures, since their behavior directly affects the vibrational response of the machine. The present study details the realization of an Experimental Modal Analysis in a metallic foundation structure, and the effects on its modal parameters when changing its mass and stiffness by fixing metal beams on it. From the presented results, it can be noticed a considerable variation in the modal parameters related to the lower natural frequencies.*

Keywords: *Rotordynamics, Foundation Structure, Experimental Modal Analysis, Modal Parameters Variation.*

1. INTRODUCTION

Rotating machines are defined from an element that rotates around its center and with a position relative to another fixed element. The main feature of this type of machine is the ability to convert input energy into different types of mechanical output energy.

The study of rotordynamics dates back the 19th century, through several researchers observing the behavior of machines in operation. The main studies from this time Rankine (1869), Laval (1883), Dunkerley and Föppl (1895) and Jeffcott (1919), which introduced the basic concepts and principles of rotordynamics. The technological development of the twentieth century made possible an advance in the research of such systems, including the use of the finite element method developed by Archer (1963). This method consists of subdividing geometries into small elements, obtaining the solution of the equations of motion of the dynamic system numerically. The method considers that the total energy of the system is given by the sum of the energy of each element.

The evolution of studies on modeling shafts and bearings, with the implementation of numerical models for each of these components, has led several research groups to develop software dedicated to the modeling of rotating systems. In this context, Tuckmantel et al. (2011) used different models to integrate a finite element modeling of a rotor (Nelson, 1980), taking into account the models of the bearings (Machado and Cavalca, 2009, Daniel and Cavalca, 2013, Tshura and Cavalca, 2020), and their support structure (Cavalca et al., 2005).

Regarding the study of the behavior of foundations structures, in particular, it began in the 1960s, when Weber (1961) applied the transfer matrix method to a model with two beams, which represented the rotor and a table-type foundation. Later, Wilson and Brebbia (1971) presented a new solution to represent the vibratory behavior of steel foundations in rotating machines using the finite element method to obtain the mass and stiffness matrices of the system. In order to obtain an alternative means of including the foundation effects in the rotor, Cavalca (1993) implemented the mixed coordinate method, associating the physical coordinates of the rotor and the modal coordinates of the foundation. Cavalcante (2001) used the mixed coordinate method to integrate the parameters of a foundation structure, obtained using the classical modal analysis methodology, to a bearing rotor model. Okabe (2007) investigated the effects of the support structure and hydrodynamic bearings on the behavior of a rotating machine. The modal parameters of the foundation were defined through a modal analysis of FRF's.

The present work is inserted in the context of a larger project, where a theoretical-experimental model is proposed for the interaction between rotor-bearings-structure, where the classic Finite Element Method is used for rotor modeling. And with regard to hydrodynamic bearings, a numerical approach is used to calculate nonlinear hydrodynamic forces, through the solution of the Reynolds Equation using the Finite Volume Method; and finally, the experimental analysis of the foundation structure was in charge of this work. The idea is to unite the theoretical response of the rotor-bearing system with the experimental response of the foundation, in order to identify how the vibration iteration occurs between the two systems.

Within this scenario, the main objective of this work is to carry out an experimental modal analysis of a metallic structure used as a foundation for a rotating machine. The paper analyzes the modal parameters of the foundation structure and also the influence of changing the mass and stiffness of the foundation by fixing metal beams on it, through the evaluation of its modal parameters.

2. METHODOLOGY

In order to contextualize the study on modal analysis, it is important to detail some theoretical concepts essential for understanding the theme. This section presents the main principles of vibrations involved and the methods of modal identification most used in the area.

For a non-conservative system with n degrees of freedom, the equation of motion in matrix form, for free vibration, is given by:

$$[M]\ddot{x} + [C]\dot{x} + [K]x = 0 \tag{1}$$

being $[M]$ the mass matrix, $[K]$ the stiffness matrix and $[C]$ the damping matrix of the system.

Knowing that this type of equation has a solution of type $\{X\}e^{st}$, and taking this solution with its derivatives in Eq.(1), we get:

$$([M]s^2 + s[C] + [K])\{X\} = \{0\} \tag{2}$$

Eq.(2) defines an eigenvalue and eigenvector problem. Therefore, the trivial solution is not of interest for the vibratory movement, so that:

$$det([M]s^2 + s[C] + [K]) = 0 \tag{3}$$

The expansion of this determinant leads to the characteristic polynomial. The roots of this polynomial are the eigenvalues of the system, which has the form of Eq.(4). For dissipative systems, we have relations (5) and (6) for each eigenvalue.

$$s_r = \sigma_r + j\omega_{dr} \tag{4}$$

$$\sigma_r = -\xi_r\omega_r \tag{5}$$

$$\omega_{dr} = \omega_r\sqrt{1 - {\xi_r}^2} \tag{6}$$

where ξ_r is the damping factor, ω_r is the natural frequency and ω_{dr} is the damped natural frequency, all associated with the eigenvalue r of the system.

Thus, from the information contained in the eigenvalues is possible to obtain the natural frequencies and the damping factors of the system. The eigenvectors represent the vibration modes of the system, obtained by substituting the eigenvalues obtained in Eq.(2).

From the definition of the modes normalized by the mass matrix ([φ]), it can be written that for the damping matrix we have the following relation:

$$\bar{C} = [\phi]^T[C][\phi] \tag{7}$$

In terms of damping, the matrix $\bar{C}$ is not necessarily diagonal, and the condition for the diagonalization is the possibility of its representation through a linear combination of the matrices $[M]$ and $[K]$. This type of damping is known as proportional structural damping.

This type of damping is defined in terms of the constants α and β, which are determined using specific model adjustment methods.

$$[C] = \alpha[M] + \beta[K] \tag{8}$$

Finally, the free-motion matrix equation of a system of "n" degrees of freedom, with proportional type damping, can be described as:

$$[M]\{\ddot{x}_{(t)}\} + (\alpha[M] + \beta[K])\{\dot{x}_{(t)}\} + [K]\{x_{(t)}\} = 0 \tag{9}$$

So, the damping matrix is also diagonalized by the modal matrix, so that:

$$\bar{C} = \alpha[I] + \beta[\omega_r^2] \tag{10}$$

As for this case $[\phi]$ diagonalizes $[C]$, one can analyze an n-degree of freedom system as being constituted by n independent 1 degree of freedom systems in the modal coordinates q_i, that is, in the same way they will be obtained n dampened vibrating modes.

$$\ddot{q}_r + (\alpha + \beta\omega_r^2)\dot{q}_r + \omega_r^2 q_r = 0 \tag{11}$$

3. SYSTEMS WITH NON-PROPORTIONAL DAMPING

For the case where $[\phi]$ does not diagonalize $[C]$, the equivalent damping matrix, after transformation by the modal matrix, results in a complete matrix, which implies n equations of motion coupled in terms of speed, but decoupled in other terms.

$$\ddot{q}_r + \omega_r^2 q_r + \sum_{j=1}^{n} c_{jr}\dot{q}_r = 0 \quad (12)$$

Therefore, it is clear that if $[C]$ cannot be described as a linear combination of the matrices $[M]$ and $[K]$, the advantage of transforming physical or geometric coordinates into main or modal coordinates is lost. For this case, there is a similar approach, in which the equation of motion remains the same, however a state vector is introduced which is presented as follows:

$$u_{(t)} = \begin{Bmatrix} \{x_{(t)}\} \\ \{\dot{x}_{(t)}\} \end{Bmatrix} \quad (13)$$

With this state vector, we can rewrite the equation of motion in matrix form as:

$$\begin{bmatrix} [C] & [M] \\ [M] & [0] \end{bmatrix} \begin{Bmatrix} \{\dot{x}_{(t)}\} \\ \{\ddot{x}_{(t)}\} \end{Bmatrix} + \begin{bmatrix} [K] & [0] \\ [0] & -[M] \end{bmatrix} \begin{Bmatrix} \{x_{(t)}\} \\ \{\dot{x}_{(t)}\} \end{Bmatrix} = \begin{Bmatrix} \{0\} \\ \{0\} \end{Bmatrix} \quad (14)$$

Since [0] is a null matrix of order n, the equation can also be written as:

$$[A]\{\dot{u}_{(t)}\} + [B]\{u_{(t)}\} = \{0\} \quad (15)$$

Adopting a solution of the type $\{x_{(t)}\} = \{X\}e^{st}$, the system with 2n 1st order equations in the state space would have the following solution:

$$u_{(t)} = \begin{Bmatrix} \{x_{(t)}\} \\ \{\dot{x}_{(t)}\} \end{Bmatrix} = \begin{Bmatrix} \{X\} \\ s\{X\} \end{Bmatrix} e^{st} = \{U\}e^{st} \quad (16)$$

$$\dot{u}_{(t)} = \begin{Bmatrix} \{\dot{x}_{(t)}\} \\ \{\ddot{x}_{(t)}\} \end{Bmatrix} = \begin{Bmatrix} s\{X\} \\ s^2\{X\} \end{Bmatrix} e^{st} = s\{U\}e^{st} \quad (17)$$

Replacing the proposed solutions, in Eq.(14):

$$\left[s[A] + [B]\right]\{U\} = \{0\} \quad (18)$$

Eq.(18) presents a generalized eigenvalue and eigenvector problem, whose solution provides 2n eigenvalues and 2n eigenvectors.

Considering that eigenvalues appear in conjugated complex pairs, s_r and s_r^*, the corresponding eigenvectors will be:

$$\{\Psi_r'\} = \begin{Bmatrix} \{\Psi_r\} \\ \{\Psi_r\}s_r \end{Bmatrix} \quad (19)$$

$$\{\Psi_r'^*\} = \begin{Bmatrix} \{\Psi_r^*\} \\ \{\Psi_r^*\}s_r^* \end{Bmatrix} \quad (20)$$

Applying the orthogonality properties, we have that:

$$\{\Psi_s'\}^T[A]\{\Psi_r'\} = 0 \quad (21)$$

$$\{\Psi_s'\}^T[B]\{\Psi_r'\} = 0 \quad (22)$$

After applying the coordinate transformation, shown in Eq.(23), in differential Eq.(15), and multiplying all terms by $[\Psi']^T$, we have Eq.(24), which is the equation of motion in the form of state in the domain modal.

$$\{u_{(t)}\} = [\Psi']\{q_{(t)}\} \quad (23)$$

$$[a_r]\{\dot{q}_{(t)}\} + [b_r]\{q_{(t)}\} = \{0\} \quad (24)$$

Thus, a set of 2n unbound equations is obtained, equivalent to a system of 2n equations of a degree of freedom. Considering each solution as follows:

$$q_{r\,(t)} = \bar{Q}_r e^{s_r t} \quad (25)$$

The response for free vibration is then calculated by replacing the coordinate transformation equation and the state equation:

$$\{u_{(t)}\} = \sum_{r=1}^{2n} \{\Psi_r\}\,\bar{Q}_r e^{s_r t} \quad (26)$$

$$s_r = \frac{-b_r}{a_r} \quad (27)$$

As presented, it is possible to obtain the system's temporal response once its characteristics are known. However, there is a problem with modeling this system, specifically with regard to the matrices $[M]$, $[C]$ and $[K]$.

Even with the use of a theoretical modal analysis, the same problems are encountered. However, in an experimental modal analysis (EMA), with the frequency response function (FRF), there are methods to obtain the modal parameters that facilitate the process.

4. FREQUENCY RESPONSE FUNCTION

An FRF can be defined as a proportionality relationship, between an input and an output, that describes the system's behavior. Analyzing an FRF, its outputs can be given as a function of displacement (compliance), speed (mobility) or acceleration (accelerance). The expression of an FRF matrix can be seen in Eq.(28).

$$\left[H_{(\omega_k)}\right] = \sum_{r=1}^{2N} \frac{[A]_r}{j\omega_k - \lambda_r} \tag{28}$$

where N denotes the number of modes, $[A]_r$ is defined as the residual matrix for mode r and λ_r the eigenvalue for mode r.

As reported in literature (He and Fu, 2001), there are several methods for the identification of the modal parameters that use FRF. Each identification method has its particularities and may also differ with respect to the number of signal inputs and outputs used. Among the most common classic methods, it can be mentioned the *peak-picking*, *circle-fit*, *line-fit*, *rational fraction polynomial*, or RFP, and *least-squares complex exponential*, or LSCE.

4.1 Identification method

For the identification of the structure modal parameters, the method used in this work is the Least Squares Method for Complex Exponentials (LSCE), usually the method most applied to systems with multiple degrees of freedom. This method explores the relation between an impulse response function (IRF) of a system with n degrees of freedom and its complex poles and residues through a complex exponential.

Starting from the transfer function, the inverse Laplace transform of this function is the IRF. Although the data is real, the residues and roots represent complex variables. It is possible to demonstrate that the imaginary parts cancel out because of the complex conjugates. The next step is to estimate the roots and residues of the sampled data.

Starting from the Prony equation, as demonstrated by He and Fu (2001), and developing it in a summation, we will have respectively:

$$\beta_0 + \beta_1 z_r + \beta_2 z_r^2 + \cdots + \beta_{2N-1} z_r^{2N-1} + \beta_{2N} z_r^{2N} = 0 \tag{29}$$

$$\sum_{k=0}^{2N} \beta_k h_k = \sum_{k=0}^{2N} \beta_k \sum_{k=0}^{2N} {}_rA_{ij} z_r^k \tag{30}$$

β being the coefficient to be estimated by the Prony equation and z_r the conjugate of the roots s_r.

From Eq.(29), we know that the right side of the equation becomes zero when z_r is the root of the polynomial. This leads us to a relation between the β coefficients and the IRF samples:

$$\sum_{k=0}^{2N} \beta_k h_k = 0 \tag{31}$$

This relation offers a numerical alternative to estimate β coefficients. In the Eq.(29), we can consider β_{2N} to be one. Taking a set of 2N samples from an IRF, the previously mentioned relationship is obtained. Taking 2N sets of 2N of samples from an IRF, 2N linear equations are obtained. The coefficients can be estimated from the equations system presented as 32:

$$\begin{bmatrix} h_0 & h_1 & h_2 & \cdots & h_{2N-1} \\ h_1 & h_2 & h_3 & \cdots & h_{2N} \\ \vdots & \vdots & \vdots & \vdots & \vdots \\ h_{2N-1} & h_{2N} & h_{2N+1} & \cdots & h_{4N-2} \end{bmatrix} \begin{Bmatrix} \beta_0 \\ \beta_1 \\ \vdots \\ \beta_{2N-1} \end{Bmatrix} = \begin{Bmatrix} h_{2N} \\ h_{2N+1} \\ \vdots \\ h_{4N-1} \end{Bmatrix} \tag{32}$$

As the β coefficients are known, the Prony equation can be solved to find the roots z_r. These roots are related to the complex eigenvalues of the system, or s_r. The roots s_r are determined by the natural non-damped frequencies, ω_r, damping factors, ξ_r, and modal shapes, that can be defined respectively by:

$$\omega_r = \frac{1}{\Delta}\sqrt{ln(z_r)ln(z_r{}^*)} \tag{33}$$

$$\xi_r = \frac{-ln(z_r z_r{}^*)}{2\omega_r \Delta} \tag{34}$$

$$\begin{bmatrix} 1 & 1 & \cdots & 1 \\ z_1 & z_2 & \cdots & z_{2N} \\ \vdots & \vdots & \ddots & \vdots \\ {z_1}^{2N-1} & {z_2}^{2N-1} & \cdots & {z_{2N}}^{2N-1} \end{bmatrix} \begin{Bmatrix} {}_1A_{ij} \\ {}_2A_{ij} \\ \vdots \\ {}_{2N}A_{ij} \end{Bmatrix} = \begin{Bmatrix} h_0 \\ h_1 \\ \vdots \\ h_{2N-1} \end{Bmatrix} \quad (35)$$

being Δ the time step in which the data was acquired and A the residual matrix.

5. TEST RIG AND EXPERIMENTAL PROCEDURE

Based on ACI 351.3R-04 of the American Concrete Institute (ACI), the experimental bench used in this work can be classified as a structure with springs mounted on a block, as shown in Figure 1.

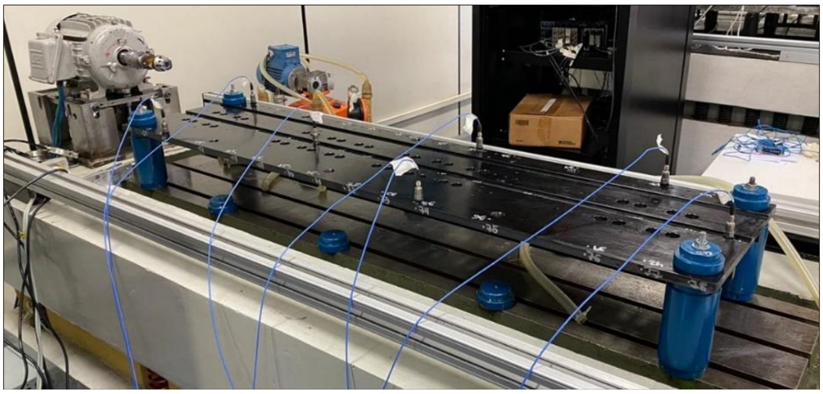

Figure 1. Assembly of the metallic foundation structure on the test rig used in the tests.

As previously mentioned, one of the current studies in this test rig is the analysis of the rotor-bearings-foundation iteration, that is, the evaluation of the influence of the characteristics of the foundation in the dynamic response of the rotor-bearings system. However, during a preliminary analysis of the foundation structure, it was found that its natural frequencies were very high, with only the first frequency in the operating range possible to be used in a rotor on this test rig.

In view of this fact, it was decided to manufacture a new foundation structure, which had lower natural frequencies, in order to enable the operation of the rotating system at rotation speeds that coincide with different natural frequencies of the foundation structure. After several simulations, it was found that a foundation thickness of 12mm (the original foundation has 20mm thick) was the one that best fit the requirements. Therefore, a new foundation structure with a thickness of 12mm was manufactured, which is the main object of study of this work.

In this scenario, there was a need to make an experimental modal analysis of this new structure. Allemang (1999), says that the four basic assumptions that a structure must obey to go through the process of experimental modal analysis are: the structure must be assumed as linear, must be invariant in time, must obey Maxwell's Reciprocity Principle and must be observable. That guaranteed, the procedure for performing an EMA can be basically divided into preparing the experiment, acquiring data, pre-processing the data and identifying the modal parameters.

For the tests carried out in this work, the first step of the experimental procedure was to mark 81 analysis points on the structure, 45 points on the upper face, in order to measure the vibration in the vertical direction, here called the 'z' direction, 30 points on the major side of both sides, to measure vibration in the lateral direction, here called 'y' and 6 points on the minor side also of both sides, to measure vibration in the axial direction, here called 'x' . After the structure was properly marked, the accelerometer fixation locations were defined. The accelerometers were fixed in such a way that they were kept in a fixed position regardless of the direction of impact. The mass of the foundation is 28.2 kg, while the mass of the metal beams is 0.7 kg for each of the four used.

The acquisition board is from National Instruments® model USB 6361 with 16 analog inputs, the accelerometers were those of the type PCB ICP, with sensitivity of 10 mV/g and frequency range of 0.5 to 10kHz, and the conditioner signal used was from the PCB with 8 channels and BNC inputs.

After the accelerometers were fixed, using an impact hammer, all points marked on the structure are subjected to an excitation force, and the number of five impacts is defined to check a measurement. The algorithm developed for the acquisition had a control that prevented the strikes with double impact from being computed.

The program used to read the information provided by accelerometers and impact hammer was developed on the LabView® platform. The algorithm uses the information of excitation force and acceleration as input, and outputs the FRF’s, as well as coherence graphs and Nyquist circle plots. All this information is obtained for each of the accelerometers and for each of the points in the structure. The sampling rate of the experiments was of 10kHz, and the windowing was exponential, with a determined coefficient of 0.01. No filter was used to process the data, since higher sampling rates were tested and no difference was observed in the results.

Once the acquisition has been made, the first stage of data analysis refers to the grouping of information regarding the geometry of the structure, with regard to the impact points and positioning of the accelerometers. As the accelerometers used are uniaxial, 3 accelerometers are needed, positioned on the axes defined as x, y and z, to compose a node to be analyzed.

Other relevant information refers to the direction of application of the excitation force. Being considered the same reference adopted for the geometry of the structure and positioning of the accelerometers.

To obtain the modal parameters of all performed experimental modal analysis tests, the Matlab® software was used, which, from its version made available in 2017, incorporated a toolbox that allows the modal analysis of bodies and structures to be carried out. Within this toolbox, the modalfit function is responsible for extracting the modal parameters by analyzing the system's FRFs.

The modalfit function estimates natural frequencies, damping factors and modal forms. It is possible to choose between the LSCE and Peak Picking identification methods for extracting the modal parameters, as well as the possibility of defining a specific frequency range to search for such parameters. The number of modes fetched for a set of FRF's is also a function input parameter. However, for all the analyzes performed in this work, the identification method used was LSCE, as it is the most suitable to be applied in systems with multiple degrees of freedom.

Finally, with the use of the EasyANIM package, a plugin to be used in the Matlab® software, it was possible to visualize the respective vibrating modes of the structure. This plugin was developed by the Department of Theoretical Mechanics, Dynamics and Vibrations, Faculty of Engineering, University of Mons (Kouroussis et al., 2012). Through information on the geometry of the structure and its modal parameters, the program graphically displays the modal shapes of each analyzed mode.

6. RESULTS AND DISCUSSIONS

The new manufactured foundation structure, which is used in this work, has a smaller thickness compared to the previous one. In addition, it was also thought to make some adaptations to this structure, in order to allow simulating other effects that may be interesting for future studies. Grooves were included in the lower part of the structure, where it is possible to attach some elements (small bars of metallic material), changing the structure's mass and stiffness distribution, making it possible to include anisotropies in this structure. In this way, it is possible to simulate situations where there are anisotropic effects on the foundation and how these effects are transmitted to the rotor-bearings system.

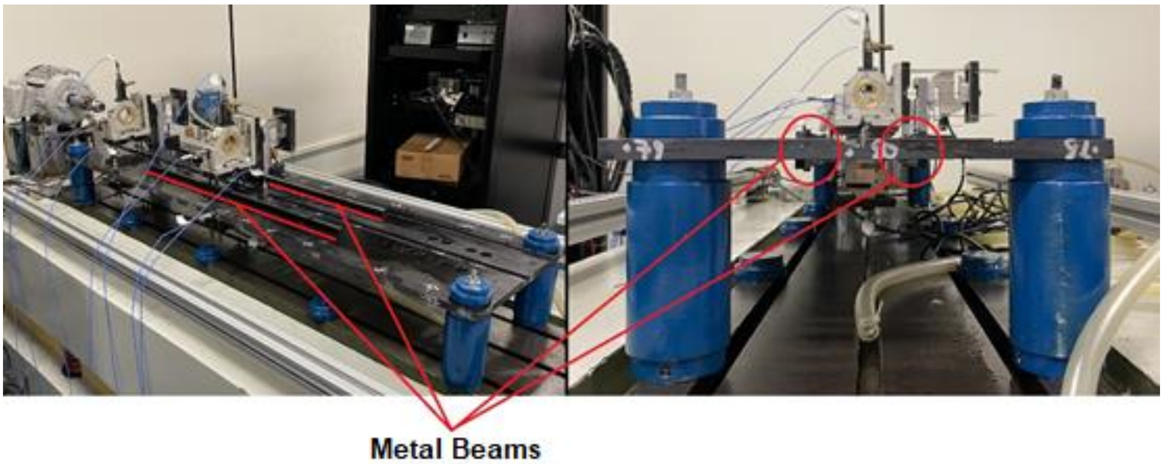

Figure 2. Detail of the fixing position of the metal beams to the metallic foundation structure.

Therefore, the idea of the results presented in this work is to evaluate the real influence of these changes, through the addition of metal beams above and below the central portion of the structure, in its modal parameters. It is important to point out here that the analyzes will be made considering also the bearing houses mounted on the structure, since it is these houses that will actually make the subsequent interaction between foundation and shaft-bearing system. Finally, it is worth mentioning that the structure is supported by four rigid supports at its ends. Figure 2 shows the mounted system, highlighting the introduction of the metal beams.

Due to the large volume of data and large amount of graphs of the research, it was impossible to place all the information in this article and the complete work can be analyzed in Gusmão (2020). However, as an example, the graph in Figure 3 shows an FRF (left) and a coherence graph (right) obtained from data acquisition by the program developed in LabView. The coherence graph is extremely important for the analysis of the frequency range to be analyzed, as the coherence values close to "1" guarantee the reliability of the acquired data.

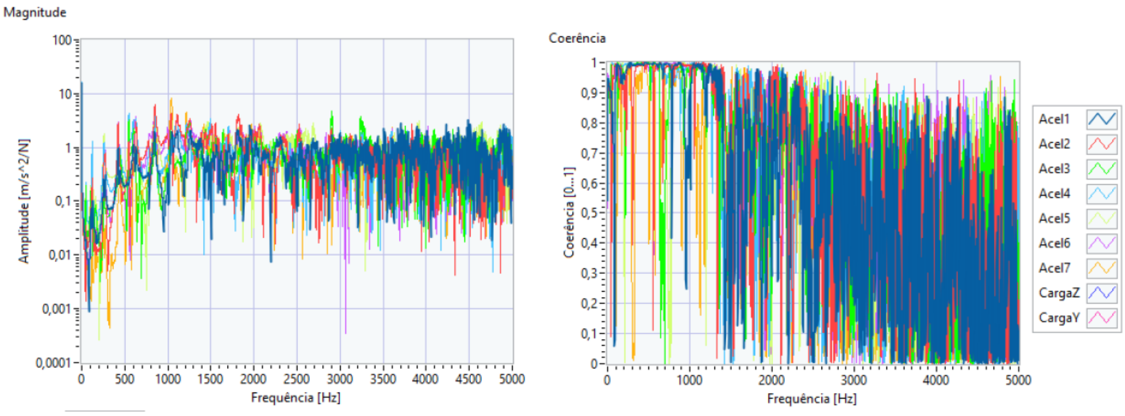

Figure 3. FRF (left) and coherence (right) graphs obtained in the data acquisition process.

The acquired data were compared with a model developed in commercial software in order to validate the experimental procedure. As shown in Gusmão (2020) the modal parameters arising from the simulation are quite similar to those obtained by the experiment.

As previously said, the purpose of fixing the metal beams to the foundation structure is to cause a change in its stiffness and mass. This opens up the possibility of simulating failure conditions that may exist in foundation structures that support rotating systems. With this in mind, the tests were carried out and table 1 was prepared to compare the natural frequencies before and after the addition of the metal beams.

Table 1. Comparison of the natural frequencies of the structure with and without the metal beams.

Modal Shapes	**Natural frequency - Without metal beams [Hz]**	**Natural frequency - With metal beams [Hz]**	**Percentage difference [%]**
1º	37,13	47,14	26,96
2º	57,66	57,79	0,23
3º	80,15	80,87	0,89
4º	93,21	94,26	1,13
5º	124,37	125,48	0,89
6º	133,07	134,88	1,36
7º	147,57	145,76	1,27
8º	152,88	155,84	1,94
9º	176,96	183,08	3,46

An aspect that draws attention in the analysis of table 1 is the significant increase in the natural frequency associated with the first modal form, going from 37.13Hz to 47.14Hz after adding the metal beams. However, the modal form, for this case, had no apparent changes, as shown in Figure 4. All figures that represent frames of animations of modal shapes have videos, whose links are present in the attachment of this work.

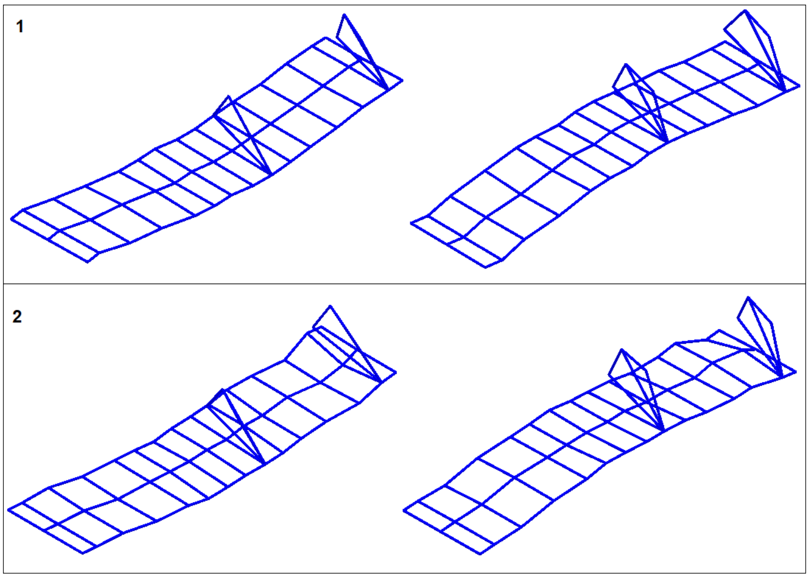

Figure 4. Comparison of the animations of the first mode of vibration of one of the accelerometers through EasyAnim (1 - Without fixed metal beams. 2 - With fixed metal beams).

In a general aspect, it is possible to observe a small increase in the natural frequency in most modes of vibration, eight of the nine compared modes, which indicates the possibility of identifying changes in the mass and stiffness of the foundation through the technique used, therefore, there is the possibility to identify foundation failures through an EMA.

Another interesting analysis is the change occurred in the modal forms of some natural frequencies of the structure. Analyzing the frequency range from 47.14Hz to 145.76Hz, it was possible to observe that among the 3 modal shapes found, two, which were composed by a combined bending modal shape of the structure together with a lateral deformation of the bearing houses, the fourth and sixth modes of vibration presented in table 1, have been transformed into combined torsional modal shapes of the structure together with a lateral deformation of the bearing houses. This can be justified by the fact that the increase in stiffness occurred in the longitudinal portion of the structure after the metal beams were added, making it difficult to bend the structure. The fifth modal form, which did not change, presented only a deformation of the bearing houses, before and after the addition of the metal beams. Figure 5 helps to observe the phenomenon of changing the bending modes by torsion modes when adding metal beams to the structure.

This increase in the stiffness of the central portion of the structure also justifies the significant increase in the natural frequency associated with the first modal shape of the system, which is a bending mode. After the beams were fixed, there was an increase of 10Hz, or almost 20%, in the natural frequency of the first mode of vibration in the system.

In modes with higher frequencies (modes seven to nine of table 1), the system already has enough energy to overcome the increase in stiffness, and again presents combined bend modal shapes of the structure together with a lateral deformation of the bearing houses.

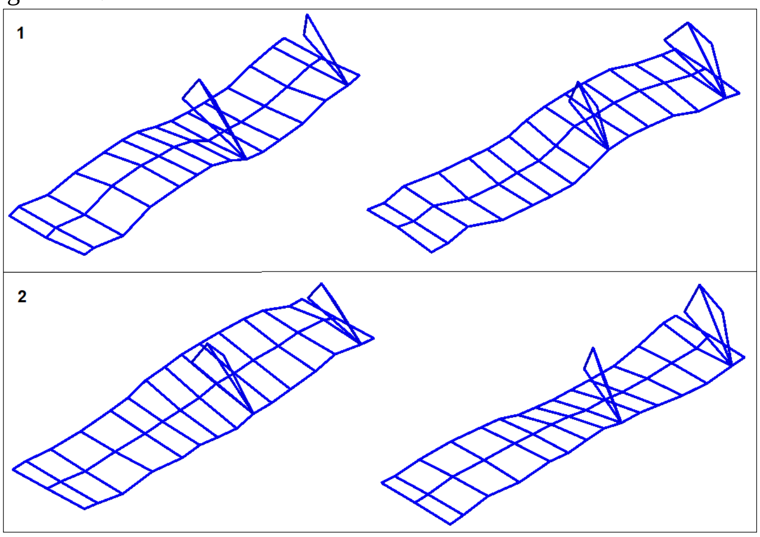

Figure 5. Comparison of the deformed of the fifth mode of vibration of one of the accelerometers through EasyAnim (1 - Without fixed metal beams. 2 - With fixed metal beams).

The last modal parameter analyzed is the system's damping factor. Table 2 presents damping factor values for the system before and after adding metal beams to the foundation structure.

Table 2. Comparison of the damping factors of the structure with and without the metal beams.

Modal Shapes	Damping factors - Without metal beams [%]	Damping factors - With metal beams [%]	Percentage difference [%]
1º	1,83	3,70	102,19
2º	3,28	4,80	46,34
3º	1,85	2,54	37,30
4º	5,53	4,17	24,59
5º	3,62	3,05	15,75
6º	2,96	4,68	58,11
7º	4,55	3,10	31,89
8º	3,46	6,55	89,31
9º	2,52	2,56	1,59

Although the identification of damping factor contains greater uncertainty, since the vast majority of available identification methods have greater difficulty in identifying this parameter, it is possible to observe, in general, an increase in damping of most vibrating modes, six of the nine compared modes, of the system after the metal beams are added to the foundation structure.

7. CONCLUSIONS

This paper presented a detailed description of an experimental modal analysis procedure performed on a foundation structure, used in rotating systems, making it possible to make comparisons about the modal parameters, even with the visualization of their modes of vibration.

It was analyzed changes in the mass and stiffness of the foundation, through the addition of small metal beams to the structure, at a specific point, in the case analyzed in the center of the structure. It was possible to observe a significant increase in the natural frequency of the first modal form. This was due to the fact that the first mode of vibration is a bending mode, and the increase in stiffness occurred in the longitudinal direction in the center of the structure. This configuration change also caused combined modes between the structure and the bearing houses to change from bending modes to torsion modes. There was also an increase in damping factors, for most modes, after the inclusion of this small metal beams in the central region of the structure.

The objectives of the work were successfully achieved. It was possible to observe, through the proposed experimental modal analysis process, with a certain degree of reliability, that the acquisition of the new foundation structure for the experimental test rig, meets the specified objectives: to reduce its natural frequencies for feasible regions to operate the rotating system; enable the simulation of possible foundation failures by adding the small fabricated metal beams.

Based on these conclusions, suggestions for future work are:

Perform the tests using an acquisition board and a conditioner with a greater number of channels, in order to allow the use of a greater number of accelerometers simultaneously.

Evaluate a rotating system, based on the structure analyzed in this work, in order to verify the rotor-bearing-structure iteration. In addition, to evaluate how the inclusion of small metal beams influences the behavior of the rotating system.

8. ATTACHMENT

Link to the animation related to Figure 4 – https://youtu.be/C-9o22zD19A
Link to the animation related to Figure 5 – https://youtu.be/E3rnIZNTFbA

9. ACKNOWLEDGMENT

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