STUDY OF THE INFLUENCE OF SUPPORT STRUCTURE ON THE ROTOR DYNAMIC BEHAVIOUR

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Abstract: *This works presents a study of the influence of a support structure with non-proportional damping on the dynamic behaviour of a rotor-bearings system. Rotating machinery is usually composed of several stationary and rotating components interacting with each other. Its main component is the rotor, which is supported by a group of bearings. In large rotating machinery, these bearings are hydrodynamics, and they are supported by a complex metallic structure. For small displacements, this structure has a linear behaviour in the majority of the cases; however, its damping is usually non-proportional, which increases the complexity of its analysis. In order to study the damping effect, a system of rotor-bearings-support structure was simulated, with a shaft modelled by the finite element method. This shaft is supported by a couple of hydrodynamic bearings, which were represented by their stiffness and damping coefficients, determined from the forces calculated through their finite volume models. The support structure was modelled using a group of lumped masses linked with springs and dampers. The rotor was excited by an unbalanced mass positioned in the middle of the shaft, and the vibration amplitude and phase were calculated using several rotation speeds. Different damping coefficients were applied to the support structure, representing different levels of non-proportional damping. The results show that the support structure damping affects mainly the shaft inside the bearing, and the non-proportional damping increased the coupling between horizontal and vertical shaft movements. Therefore, the correct damping modelling has been proved to be necessary in order to predict the rotating machine behaviour.*

Keywords: *Rotordynamics; Support Structure; Non-Proportional Damping*

1. INTRODUCTION

The Brazilian electrical power generation is based on hydroelectric and thermoelectric plants. These plants are usually composed by groups of rotating machinery, which includes turbines and generators. These elements are vital to energy generation; for that reason, it is important to understand their behaviour to get a cleaner and safer operation.

This particular type of rotating machinery is composed by a large number of components, like shafts, couplings, bearings, flow seals and support structure. The rotor of a turbine is usually composed by a shaft and groups of blades, and it is supported by bearings, which are hydrodynamic for large machines.

The bearings are supported by a structure, also known as foundation or pedestal. This structure is generally made of steel, and it is connected to large blocks or columns of concrete embedded in soil. These blocks or columns act as mass dampers, absorbing the rotor vibration transmitted to structure. This vibration excites the movement of the structure, interacts with bearings that send it back to the rotor. This dynamical interaction with the support structure affects the behaviour of the complete system (Rotor-Bearings-Support Structure), and because of that it is important to identify correctly the support characteristics. Several methods were employed to model the dynamic behaviour of foundations, from simplified models considering mass-spring systems or beams, to complex finite element models. Nevertheless, the models that showed better results are the experimental ones, built from the modal analysis of the frequency response function (FRF) measured in foundation structure.

Actual support structures are very complex, and although they can show a linear behaviour for small vibrations, only very simple structures are proportionally damped (Balmès, 1997). When damping is low and modal frequencies are reasonably far from each other, damping can be considered proportional. However, support structures of rotating machinery and all the elements connected to it usually present a high modal density with a high degree of damping, which increases the errors of considering a proportional damping. Therefore, it is important to include the effect of non-proportional damping to simulate the complete system properly.

1.1 Simulation of rotor-bearing-support structure system

Gasch (1976) proposed the inclusion of the foundation effects in a rotor-bearing system through its dynamical stiffness matrices, obtained by inverting the sum of the experimental receptance matrices of a foundation structure, and receptance matrices of the fluid film obtained from the stiffness and damping coefficients of the bearings.

Bachschmid et al. (1982) took several experimental tests in the foundation structure of a 660 MW turbo-generator, composed by concrete reinforced structure embedded in sand by a series of long pillars. The experimental data of the foundation were compared with the theoretical model one, composed by beam elements. They concluded that the mathematical model did not approach the dynamic behaviour of the actual structure, and the closest results were obtained after several adjustments of the parameters.

Cavalca (1993) used the Mixed Co-ordinates method to include foundation effects in a turbo-generator consisting of three rotors and seven bearings. The Mixed Co-ordinates method uses physical co-ordinates to the rotor and modal coordinates to the foundation structure. Cavalca et al. (2005) performed an investigation of the support structure applying the mixed co-ordinates method to determine the complete system response, and compared it to the experimental data. A good correlation was found in the calculation of natural frequencies, although, the amplitudes presented different results comparing experimental and theoretical models, because of poor accuracy in the calculation of the damping coefficients of the structure, which was improved later (Okabe, 2007).

1.2 Non-proportional damping

The non-proportional damping of structures has been studied for several years. The first approaches were only theoretical, and basically consisted in improving the simulation of a theoretical structure possessing this characteristic. Lately, the research evolved to the identification of non-proportional damping during the modal analysis of actual structures, and relates the experimental model to a theoretical model based on finite elements.

Özguven and Cowley (1981) developed a method for the calculation of receptance matrices of continuously and non-proportional damped plates, the results were compared to classical finite elements method. Özguven (1987) presented an improved method to calculate the receptance matrices of non-proportionally damped structures using the undamped modal data and damping matrix, and it could be applied to structures with frequency or temperature dependent damping properties. Ibrahimbogovic and Wilson (1989) developed an algorithm to solve iteratively coupled modal equations of linear structural systems with non-proportional damping. The algorithm eliminated the need for the computation of complex frequencies and mode shapes for use in dynamic modal analysis of structural system with non-proportional damping.

Balmès (1997) proposed a method to identify normal modes and the associated non-proportional damping matrix. The method was experimentally tested in the parameters identification of a testbed composed by triangular truss beams. Prells and Friswell (2000) studied the relation between proportional damping and general viscous damping, and created a measure of non-proportionality based on an orthogonal matrix, which represented the phase difference between the degrees of freedom of a system.

Kasai and Link (2002) developed an identification method of symmetric non-proportional damping parameters using undamped modal parameters. They highlighted the importance of using a combination of real normal modes, usually obtained by the application of Finite Element method, and a non-proportional damping matrix on the identification of structures. Naylor et al. (2004) presented the Resonant Decay Method to identify non-proportionally damped systems using multiple sources of excitation, based on extension of the force appropriation method. They tested the method simulating a non-proportionally damped plate model and identifying its modal parameters.

2. THEORETICAL MODEL

The mathematical modelling of the complete rotor-bearings-foundation system considers two subsystems: rotor-bearings and foundation. Each one of them is analysed, and the complete system response is obtained joining both subsystems dynamic equations.

The Mixed Co-ordinates method developed to simulate a complete system was based on a modification of the mechanical impedance method, to avoid numerical problems in the flexibility matrix inversion. This method represents the displacement vector of the structure connection nodes as independent variables using a modal approach (Cavalca, 1993). This transformation is applied on a mixed co-ordinates vector, physical coordinates for the rotor and modal coordinates for the foundation, which describes the complete system behaviour. This approach allows the selection of the most significant mode shapes in the rotor operation range.

According to Cavalca et al. (2005), to solve the motion equation of the complete system, it is necessary to establish the relation between physical and principal coordinates:

$$\{x_f\} = [\Phi]\cdot\{q\} \tag{1}$$

Where $\{x_f\}$ is the physical co-ordinates vector of the support structure, $\{q\}$ is the generalized (modal) coordinates vector, and $[\Phi]$ is the modal matrix (real eigenvectors).

Using Eq. (1), the foundation mechanical impedance matrix in modal coordinates can be obtained without the inversion of the flexibility matrix.

Neglecting the oil film inertia, the equation of the rotor-bearings subsystem becomes:

$$\begin{bmatrix}[M_R] & [0]\\ [0] & [0]\end{bmatrix}\begin{Bmatrix}\{\ddot{x}_r\}\\ \{\ddot{x}_m\}\end{Bmatrix}+\begin{bmatrix}[R_R]+[R_{rr}] & [R_{rm}]\\ [R_{mr}] & [R_{mm}]\end{bmatrix}\begin{Bmatrix}\{\dot{x}_r\}\\ \{\dot{x}_m\}\end{Bmatrix}+\begin{bmatrix}[K_R]+[K_{rr}] & [K_{rm}]\\ [K_{mr}] & [K_{mm}]\end{bmatrix}\begin{Bmatrix}\{x_r\}\\ \{x_m\}\end{Bmatrix}=\begin{Bmatrix}\{F_0\}\\ \{0\}\end{Bmatrix}+\begin{Bmatrix}\{0\}\\ \{F_f\}\end{Bmatrix} \tag{2}$$

Where $\{\ddot{x}_r\}$, $\{\dot{x}_r\}$, $\{x_r\}$ are the acceleration, velocity and displacement vectors of the rotor; $\{\ddot{x}_m\}$, $\{\dot{x}_m\}$, $\{x_m\}$ are the acceleration, velocity and displacement vectors of the bearings; $[M_R]$, $[R_R]$, $[K_R]$ are the mass, damping, stiffness matrices of the rotor; $[R_{rr}]$, $[R_{rm}]$, $[R_{mr}]$, $[R_{mm}]$ are the linear damping coefficient matrices of the oil film; $[K_{rr}]$, $[K_{rm}]$, $[K_{mr}]$, $[K_{mm}]$ are the linear stiffness coefficient matrices of the oil film; $\{F_0\}$ is the force due to rotor unbalance and $\{F_f\}$ is the force transmitted to the foundation.

A third equation is necessary to solve the linear system of equations, because the number of unknown variables ($\{x_r\}$, $\{x_m\}$ and $\{F_f\}$) is larger than the number of equations. The forces transmitted through the oil film on the rotor-foundation interface $\{F_f\}$ are determined through the modal approach of the Eq. (3).

Using the foundation motion equation in modal coordinates:

$$[m_f]\cdot\{\ddot{q}\}+[r_f]\cdot\{\dot{q}\}+[k_f]\cdot\{q\}=-[\Phi]^T\{F_f\} \tag{3}$$

Where $[m_f]$ is the modal mass matrix, $[r_f]$ is the modal damping matrix, $[k_f]$ is the modal stiffness matrix and $[\Phi]^T$ is the transposed modal matrix.

The foundation forces, transmitted by the oil film of the bearings, can also be written as:

$$[R_{mr}]\{\dot{x}_r\}+[R_{mm}]\{\dot{x}_m\}+[K_{mr}]\{x_r\}+[K_{mm}]\{x_m\}=\{F_f\} \tag{4}$$

Replacing Eq. (5) in Eq. (4):

$$[m_f]\{\ddot{q}\}+[r_f]\{\dot{q}\}+[k_f]\{q\}=-[\Phi]^T\cdot\left([R_{mr}]\{\dot{x}_r\}+[R_{mm}]\{\dot{x}_m\}+[K_{mr}]\{x_r\}+[K_{mm}]\{x_m\}\right) \tag{5}$$

If the modal shapes are independent, then the matrices, which contain the modal parameters determined through the modal analysis methods, are diagonal (Ewins, 1995). Expressing the foundation dynamic behaviour through its bearing connecting nodes, then the bearing nodes $\{x_m\}$ become the connection foundation nodes $\{x_f\}$.

After the determination of the foundation forces, the equation of the complete system is obtained replacing these forces obtained by Eq. (5) in Eq. (2), using the modal approach of Eq. (1):

$$\begin{aligned}&\begin{bmatrix}[M_R] & [0]\\ [0] & [m_f]\end{bmatrix}\cdot\begin{Bmatrix}\{\ddot{x}_r\}\\ \{\ddot{q}\}\end{Bmatrix}+\begin{bmatrix}[R_R]+[R_{rr}] & [R_{rm}][\Phi]\\ [\Phi]^T[R_{mr}] & [r_f]+[\Phi]^T[R_{mm}][\Phi]\end{bmatrix}\begin{Bmatrix}\{\dot{x}_r\}\\ \{\dot{q}\}\end{Bmatrix}+\\ &\begin{bmatrix}[K_R]+[K_{rr}] & [K_{rm}][\Phi]\\ [\Phi]^T[K_{mr}] & [k_f]+[\Phi]^T[K_{mm}][\Phi]\end{bmatrix}\begin{Bmatrix}\{x_r\}\\ \{q\}\end{Bmatrix}=\begin{Bmatrix}\{F_o\}\\ \{0\}\end{Bmatrix}\end{aligned} \tag{6}$$

The supporting structure can be represented only by its most significant modes, in the analyzed frequency range, independently of the degrees of freedom of the connection points (bearings).

This procedure is valid for orthogonal mode shapes that enable the diagonalisation of the mass and the stiffness matrices of the foundation by the modal matrix. The transmitted forces, written through the impedance matrix in principal co-ordinates, are substituted into Eq. (2); thus the equation of motion for the complete system is determined in Eq. (6).

In a complex supporting structure, only the modes that significantly contribute to the system response can be considered in the approach. As the only external excitation force considered in the model is the unbalance force, the modes that influence the rotor response should be inside its operational frequency range. Therefore, using this approach, the number of identified modes does not need to be equal to the number of degrees of freedom associated with the bearings. Consequently, the modal co-ordinates associated to the contributing modes will take place in the whole matrix of the system, substituting the bearings degrees of freedom connected to the structure.

3. TEST CASE

The test rig of LAMAR-Unicamp (Cavalca and Okabe, 2009) was chosen to be modelled and simulated. The shaft of the rotor was modelled by finite element method, using beam elements with circular cross sections (Nelson and McVaugh, 1976), discretized into 32 elements containing 33 nodes, and supported by two hydrodynamic cylindrical bearings, located at nodes 2 and 32 of the shaft, which were connected to the foundation by nodes 34 and 35 (Fig. 1). The rotor shaft had a length of 600 mm and 12 mm of diameter. The bearing journal had a diameter of 31 mm and length of 20 mm. A disk (mass) of 2.3 kg was attached in the shaft mid span, where it is possible to neglect gyroscopic effects.

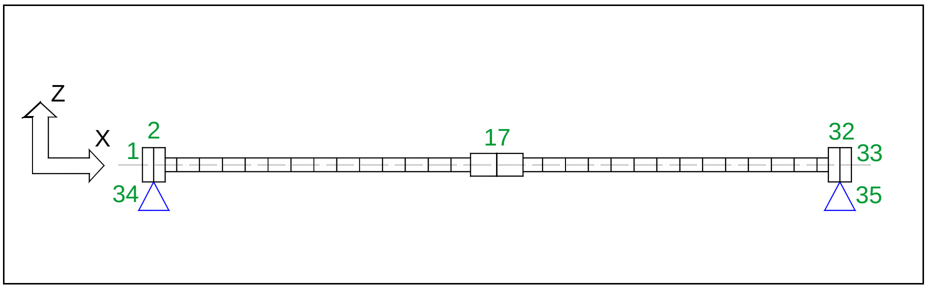


Figure 1: Finite element model of the rotor.

Damping and stiffness coefficients of hydrodynamic bearings were calculated using the finite volume method (Maliska, 2000), considering a finite bearing model with axial and circumferential flows. The synchronous response of the rotor was calculated using the mixed co-ordinates method to include the foundation effect. The simulation was performed in a rotation speed range from 0 to 70 Hz (439 rad/s).

The theoretical model of the support structure developed was composed by three masses interconnected with a set of springs and viscous dampers, which can be seen on Fig. 2. The main body (m_0) represents the bearing case, and it has cross-coupling stiffness and damping coefficients between other bodies (m_1 and m_2). The excitation forces were transmitted from the rotor to the main body through the hydrodynamic bearings.

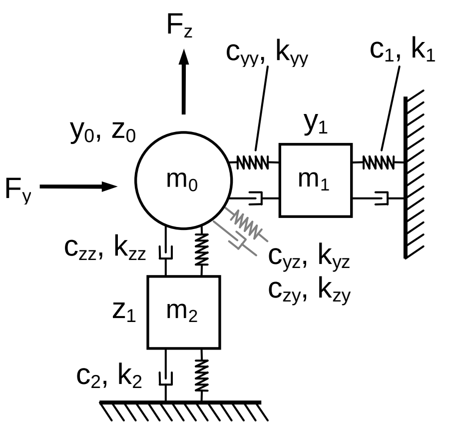


Fig. 2: Theoretical foundation structure (Okabe and Cavalca, 2008).

The equation system (Eq. (7)) was used to model the structure represented on Fig. 2, and the following values were adopted $m_0 = 5$ kg, $m_1 = 50$ kg, $m_2 = 200$ kg, $k_{xx} = 5\times10^5$ kN/m, $k_{xy} = k_{yx} = 1.5\times10^3$ kN/m, $k_{yy} = 1.5\times10^5$ kN/m, $k_1 = 1.5\times10^5$ kN/m and $k_2 = 1.5\times10^5$ kN/m, the proportional damping coefficients were $c_{xx} = 50$ Ns/m, $c_{xy} = c_{yx} = 0.15$ Ns/m, $c_{yy} = 15$ Ns/m, $c_1 = 15$ Ns/m and $c_2 = 15$ Ns/m. The non-damped natural frequencies of the flexible support structure using the mass and stiffness aforementioned were 4.3 Hz (Mode 1), 8.3 Hz (Mode 2), 27.9 Hz (Mode 3) and 52.8 Hz (Mode 4).

$$\begin{bmatrix} m_0 & 0 & 0 & 0 \\ 0 & m_0 & 0 & 0 \\ 0 & 0 & m_1 & 0 \\ 0 & 0 & 0 & m_2 \end{bmatrix} \begin{Bmatrix} \ddot{x}_0 \\ \ddot{y}_0 \\ \ddot{x}_1 \\ \ddot{y}_1 \end{Bmatrix} + \begin{bmatrix} c_{xx} & c_{xy} & -c_{xx} & -c_{xy} \\ c_{yx} & c_{yy} & -c_{yx} & -c_{yy} \\ -c_{xx} & -c_{xy} & c_1 + c_{xx} & c_{xy} \\ -c_{yx} & -c_{yy} & c_{yx} & c_2 + c_{yy} \end{bmatrix} \begin{Bmatrix} \dot{x}_0 \\ \dot{y}_0 \\ \dot{x}_1 \\ \dot{y}_1 \end{Bmatrix} + \begin{bmatrix} k_{xx} & k_{xy} & -k_{xx} & -k_{xy} \\ k_{yx} & k_{yy} & -k_{yx} & -k_{yy} \\ -k_{xx} & -k_{xy} & k_1 + k_{xx} & k_{xy} \\ -k_{yx} & -k_{yy} & k_{yx} & k_2 + k_{yy} \end{bmatrix} \begin{Bmatrix} x_0 \\ y_0 \\ x_1 \\ y_1 \end{Bmatrix} = \begin{Bmatrix} F_x \\ F_y \\ 0 \\ 0 \end{Bmatrix} \quad (7)$$

or $\left[M_F\right]\left\{\ddot{x}_F\right\}+\left[R_F\right]\left\{\dot{x}_F\right\}+\left[K_F\right]\left\{x_F\right\}=\left\{F_F\right\}$

Using Eq. (7), the support structure can be represented by a modal approach, multiplying each of the matrices by the modal matrix and its transposed:

$$\left[m_f\right]=[\Phi]\left[M_F\right][\Phi]^T; \quad \left[k_f\right]=[\Phi]\left[K_F\right][\Phi]^T; \quad \left[r_f\right]=[\Phi]\left[R_F\right][\Phi]^T+\left[R_{NP}\right] \quad (8)$$

The matrices [m_f], [r_f] and [k_f] were applied in Eq. (6) to provide the complete system response.

The matrix [R_{NP}] establishes the non-proportional damping of the support structure, and relates the coupling between modes, for instance, modes 1 and 2:

$$R_{NP} = \begin{bmatrix} 0 & c_{12} & 0 & 0 \\ c_{21} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \quad (9)$$

In order to make clear the influence of this kind of damping, it was determined that general damping matrix [r_f] would have only two off-diagonal elements different from zero, allowing the coupling of two modes each time. Then all combinations between the four modes (1-2; 1-3; 1-4; 2-3; 2-4; 3-4) were tested, using the symmetric combination ($c_{ij} = c_{ji}$). The following values of damping were tested: 0, 5, 10, 20, 25 and 30 $Ns \cdot kg^{-1}$.

The analysis of the combination of the first two structure mode shapes was extended, and the off-diagonal coefficients were also set in four combinations: c_{12} and c_{21} equal to zero (proportional damping), c_{12} equal to c_{21} and different of zero (symmetric), the upper coefficient c_{12} (30 $Ns \cdot kg^{-1}$) higher than c_{21} (5 $Ns \cdot kg^{-1}$) and, the lower coefficient c_{21} (30 $Ns \cdot kg^{-1}$) higher than c_{12} (5 $Ns \cdot kg^{-1}$).

4. RESULTS

The first simulation (fig. 3a and 3b) considered a rigid support structure; therefore, there were no displacements of the bearing case. Figure 3a shows the vibration amplitude of the disk located on the middle of the shaft, and it clearly demonstrates the first rotor resonance frequency around 24 Hz. A small difference between disk vertical and horizontal displacements reveals that it is most influenced by the shaft rigidity.

Figure 3b illustrates the journal vibration amplitude, as the rotor is symmetric the response is equal on both bearings; for that reason, just one of them was represented in all journal vibration graphs. An amplitude difference between directions of the journal vibration can be observed as a result of the bearing hydrodynamic coefficients, demonstrating the difference of stiffness and damping among directions.

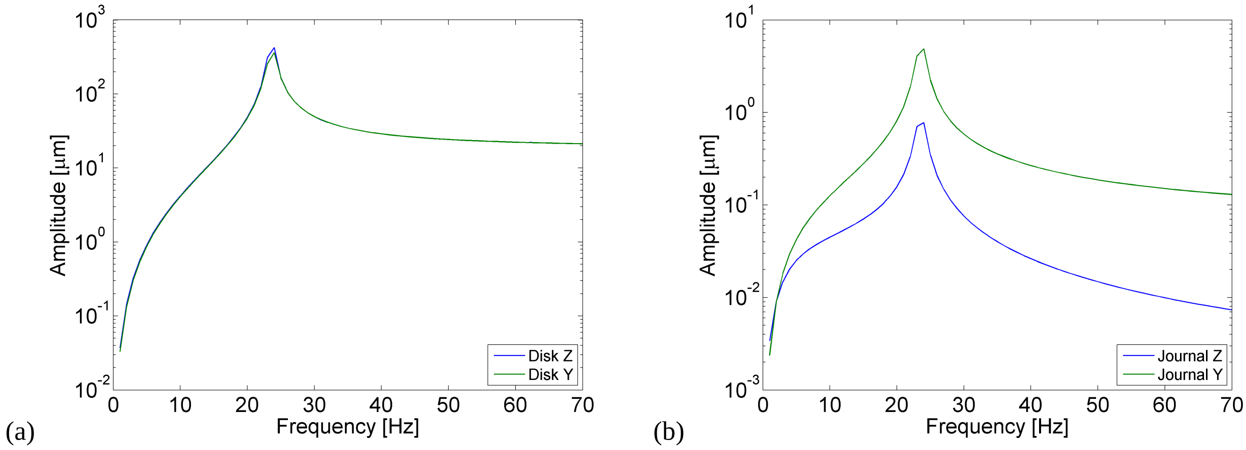


Fig. 3: Vibration amplitude of the rotor on a rigid foundation (a – disk; b – journal).

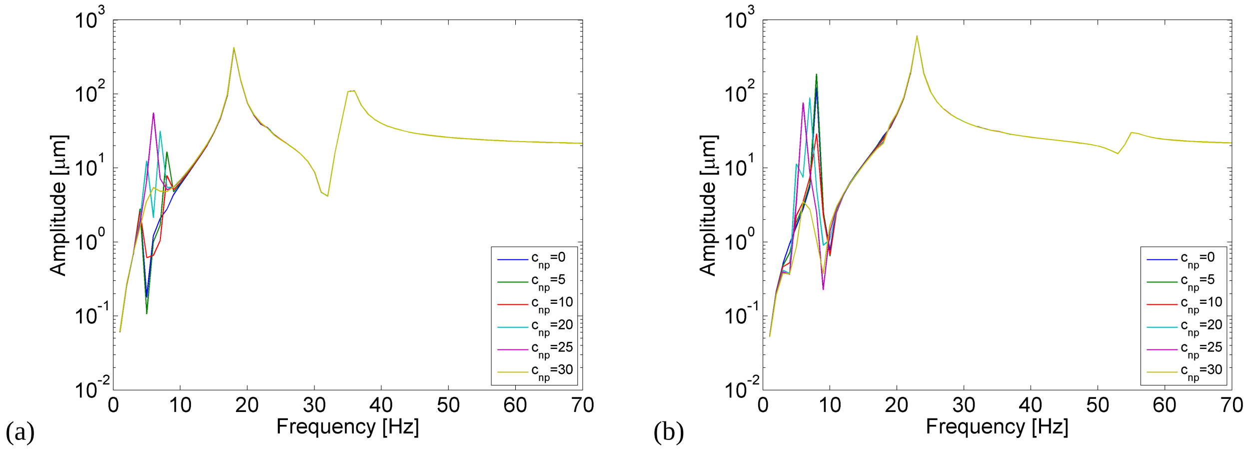


Fig. 4: Disk vibration amplitude (a – direction Y; b – direction Z) with coupling of modes 1 and 2.

Figure 4a shows the disk vibration amplitude on the horizontal direction (Y) using different damping coefficients, coupling the first two modes. It can be observed that an increase of the non-proportional coefficient (c_{np}) makes the second mode, which is vertical, appears on the horizontal response (c_{np} = 10). With this coefficient rising, the peaks of resonance grow (c_{np} = 20), transform into one peak (c_{np} = 25), and finally, the amplitude of this peak decreases (c_{np} = 30). On Fig. 4b, the same behaviour can be observed, but it is the first mode that appears on the vertical response, and there is not an increase of vibration amplitude in relation to the case of proportional damping (c_{np} = 0).

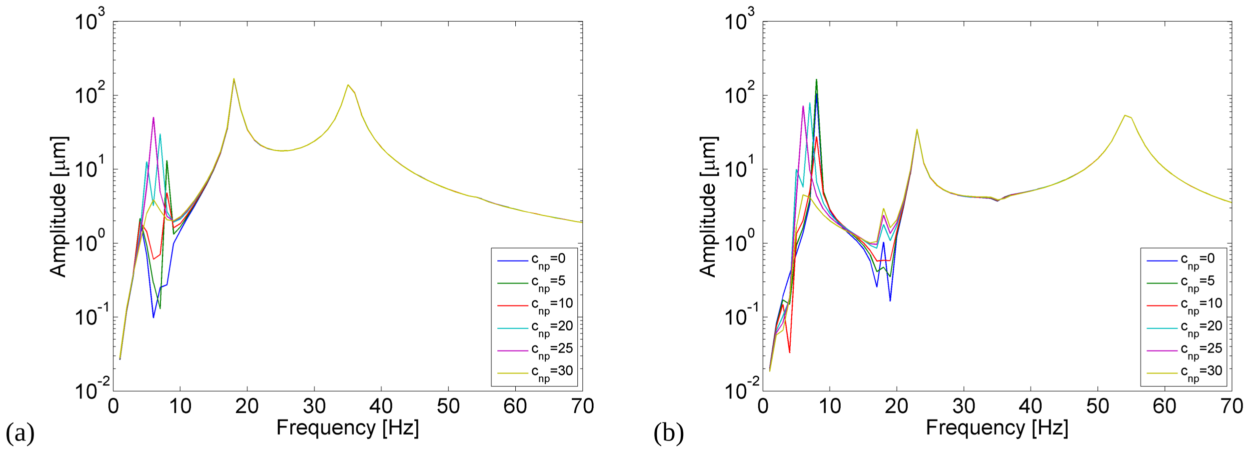
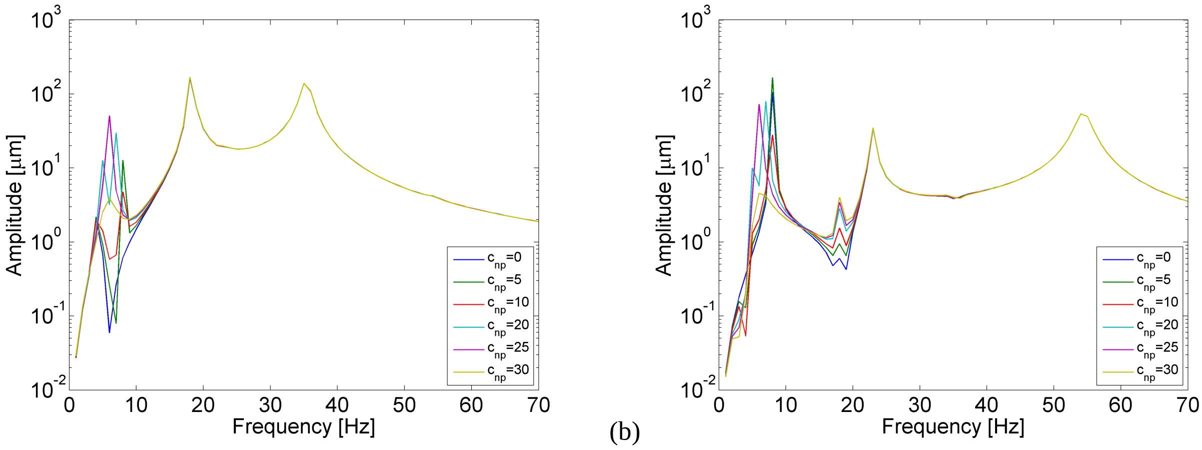


Fig. 5: Journal vibration amplitude (a – direction Y; b – direction Z) with modes 1 and 2 coupling.

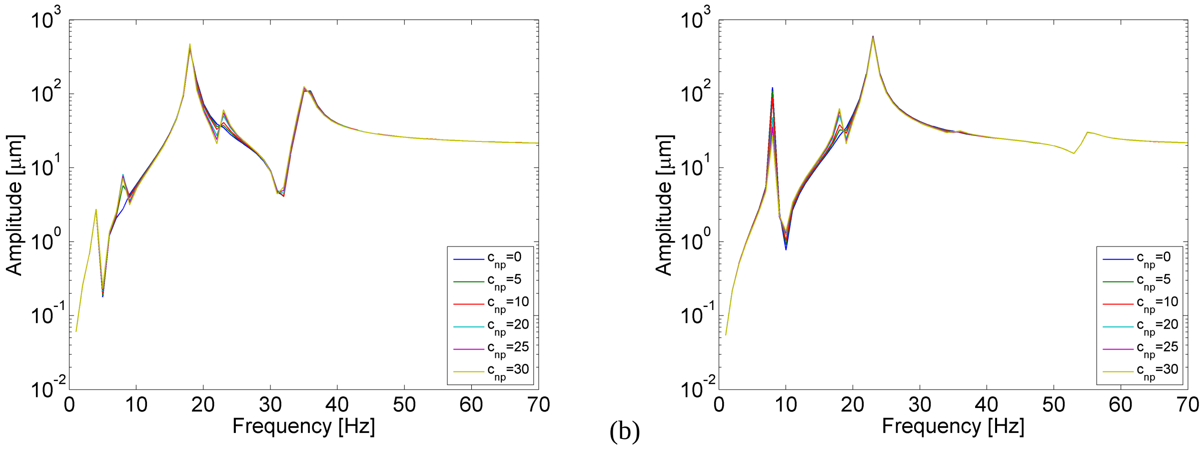
Figure 5a shows the vertical vibration amplitude of the journal, and there is the same coupling behaviour seen on Fig. 4a. A peak can be observed around 18 Hz, due to rotor resonance, and there is another peak around 37 Hz, which is related to the third mode of the support structure. Figure 5b shows the effect of coupling between the first two modes of support structure in the range below 10 Hz. The rotor resonance can be seen around 22 Hz and the fourth mode of the support structure around 55 Hz. The inclusion of the flexible support structure changed the rotor resonance frequency from 24 Hz to 18 Hz on the horizontal direction and 22 Hz on the vertical direction.



(a) (b)

Fig. 6: Bearing case vibration amplitude (a – direction Y; b – direction Z) with modes 1 and 2 coupling.

Figure 6a presents the horizontal vibration amplitude of the bearing cases, and it is very close to the result presented on Fig. 5a, which indicates that the hydrodynamic bearings employed are much stiffer than the support structure, and they transmitted almost all the vibration of the unbalanced rotor to the structure. Figure 6b shows the vertical response of the bearing cases, and there is a small difference on the resonance peak at 18 Hz between Fig. 5b and 6b, due to bearing stiffness.

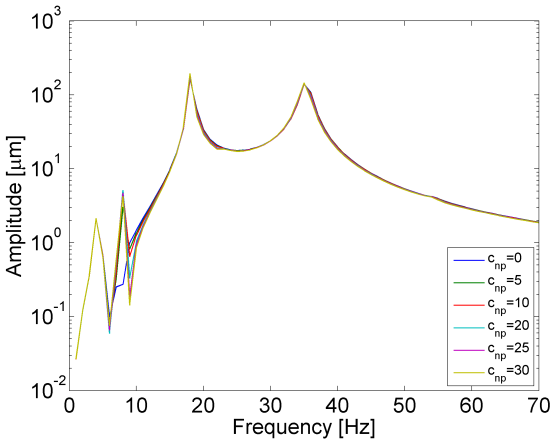


(a) (b)

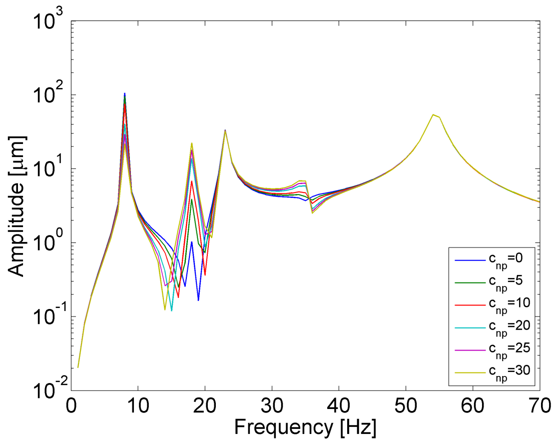
Fig. 7: Disk vibration amplitude (a – direction Y; b – direction Z) with modes 2 and 3 coupling.

Figure 7a shows the horizontal vibration amplitude of the disk positioned in the middle of the shaft, with the non-proportional coefficient damping (c_{np}) set between the second and third mode of the support structure. The damping effect is quite distinct from the one showed on Fig. 4a, where the coupling between the first two modes was clearly noticeable. The result of coupling of the second and third modes is the rising of two peaks, the first around 8 Hz due to the second mode of the support structure, and the second one around 23 Hz due to the structure third mode. Figure 7b illustrates the vertical response of the disk, and there is a rising of a peak around 18 Hz caused by the third mode of the support structure.

Figure 8a shows effect of the coupling between the second and third mode on the horizontal vibration of the journal, and it can be observed that the second mode of the structure appears with the non-proportional damping increase, while on Fig. 8b the rising of the structure third mode can be seen around 18 Hz.



(a)

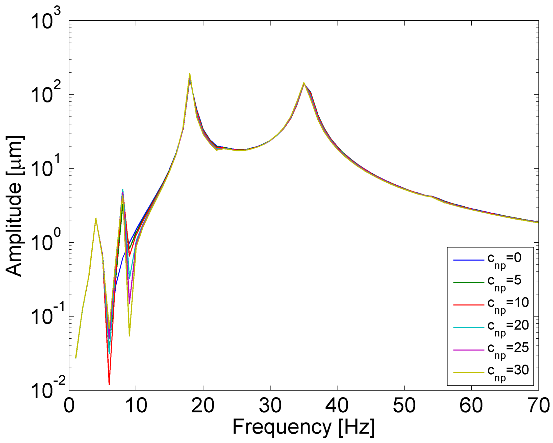


(b)

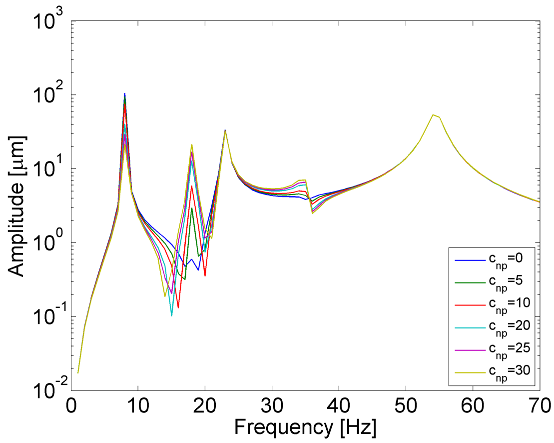
Fig. 8: Journal vibration amplitude (a – direction Y; b – direction Z) with modes 2 and 3 coupling.

Figure 9a and 9b shows the vibration amplitude of the support structure on horizontal and vertical direction, and the response on both directions is similar to the journal vibration, reinforcing that the hydrodynamic bearings are much stiffer than the support structure.

The other combinations of coupling between structure mode shapes were tested (Modes 1-3; 1-4; 2-4; 3-4), but none of them showed a noticeable change in the rotor response compared to the proportional damping case.

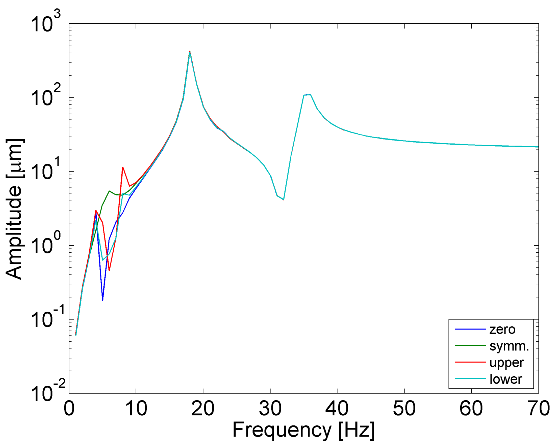


(a)

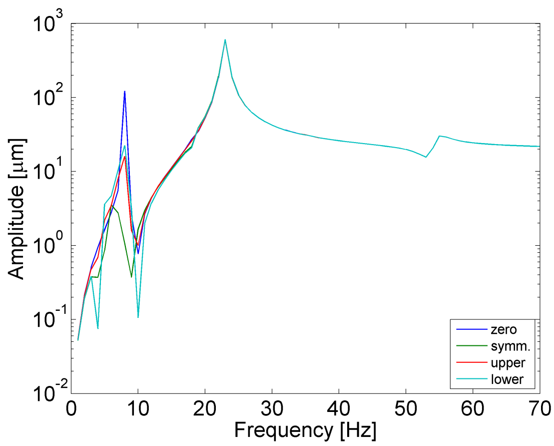


(b)

Fig. 9: Bearing case vibration amplitude (a – direction Y; b – direction Z) with modes 2 and 3 coupling.



(a)



(b)

Fig. 10: Disk vibration amplitude (a – direction Y; b – direction Z) with modes 1 and 2 coupling.

Figure 10a shows the influence of different configurations of non-proportional damping on the disk vibration amplitude. The symmetric non-proportional damping shows an increase of the first resonance peak compared to the proportional damping (zero), caused by the coupling of the first two modes of the support structure. A higher upper coefficient (upper = 30 $Ns{\cdot}kg^{-1}$ and lower = 5 $Ns{\cdot}kg^{-1}$) generates an increase of the influence of second mode on the disk response, while an increase of lower coefficient (upper = 5 $Ns{\cdot}kg^{-1}$ and lower = 30 $Ns{\cdot}kg^{-1}$) results on a small raise of second mode on the horizontal response. Figure 10b illustrates the influence of different damping configurations on the vertical response, and it can be seen that the peak of resonance caused by the second structure mode always decreases in comparison to the proportional damping (non-proportional coefficients equal to zero). The configuration with the lower coefficient increased generates a more evident anti-resonance on 4 Hz.

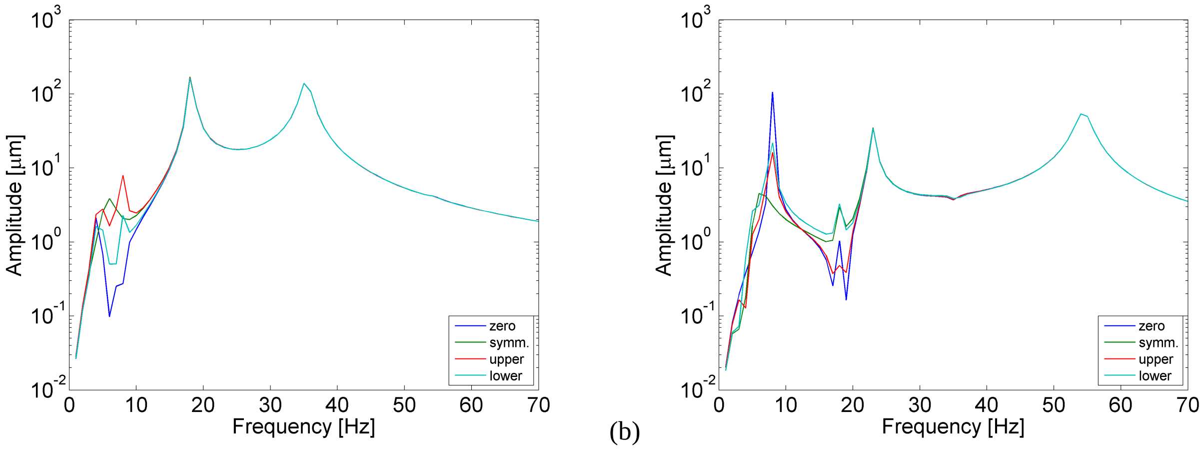


Fig. 11: Journal vibration amplitude (a – direction Y; b – direction Z) with coupling of modes 1 and 2.

Figure 11a shows the horizontal vibration amplitude of the journal inside the bearing, under the influence of different configurations of damping. The upper coefficient increases the coupling between the two first modes, and also increases the amplitude of the peaks. The application of the lower coefficient has the same kind of influence on the journal, however, the amplitude is lower. Resonance peak related to the rotor flexibility (18 Hz) and the third structure mode (37 Hz) remained unchanged. Figure 11b shows vertical vibration, and it is not clear the coupling between the two first structure modes as it was seen on the horizontal response, but it has the same tendency observed on Fig. 10b, where the proportional damping had the highest vibration amplitude.

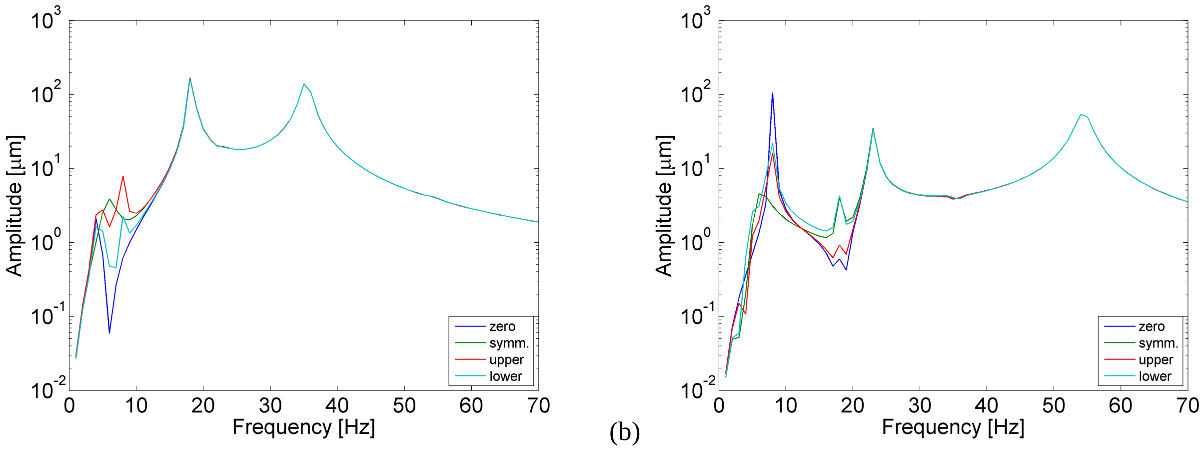


Fig. 12: Bearing case vibration amplitude (a – direction Y; b – direction Z) with modes 1 and 2 coupling.

Figures 12a and 12b show almost the same behaviour seen on Fig. 11a and 11b, what can be explained by the difference of stiffness between support structure and hydrodynamic bearings, which have stiffness varying from $8{\times}10^4$ to $3{\times}10^6$ N/m, and because of that almost all the vibration of the rotor is transmitted through the bearing to the support structure.

5. CONCLUSIONS

A simulation of a rotor-bearing-support structure system has been made in order to test the influence of non-proportional damping of the support structure using the mixed co-ordinates (modal and physical) method. The results of this simulation showed that this kind of damping can affect the rotor response, coupling the structure mode shapes.

The non-proportional damping is usually more difficult to model requiring more experimental data about the structure. Processing this data demands time and special procedures, so the first approach to include support structure effect on rotor response should consider a proportional damping. In the cases where difference between experiment and simulation persists, a more careful analysis should be made considering the coupling between close mode shapes, which have demonstrated a stronger effect on the rotor response. The next step to refine the response would be to consider the asymmetry of damping coefficients, and an optimization algorithm can be used to adjust these coefficients.

An important phenomenon observed with the application of non-proportional damping was the coupling of vertical and horizontal modes, which makes more difficult distinguishing the effect of each component (rotor, bearings and support structure) on the machine vibration. The utilization of structural modifications on the actual machine is recommended to highlight the effect of each component, when this procedure can be applied.

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