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IDENTIFICATION OF DYNAMIC COEFFICIENTS IN A JOURNAL BEARING FROM EXPERIMENTAL UNBALANCE RESPONSES AND INVERSE PROBLEM

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Abstract. *Journal bearings have largely application in industrial turbomachinery. For most cases bearing stiffness and damping parameters governs the vibration characteristics in a rotor-bearing system. However, such coefficients are usually difficult to determine and this fact can produce inaccurate analysis in rotordynamic projects. In addition, during long term operation normal and abnormal machine conditions require its predictive monitoring. This work presents an experimental methodology to determine, simultaneously, the bearing dynamic coefficients and unbalance distribution by inverse problem based on unbalance response measurements and non-linear optimization techniques. The method adjusts the unbalance response simulated by a finite element model to measured curves at run-downs procedure. A hybrid method including genetic algorithm and a direct Nelder-Mead search was used for global optimization. A base formed by eigenvectors of an invariant primary system (real rotor and the initial constant bearing coefficients) was used to obtain the numerical unbalance response in the optimization environment. This base allows find the response of the real rotor (compound system), when bearing coefficients are changing in the optimization environment, in the equivalent way to structural modification techniques. This methodology allows solve the eigenvalue problem only once, reducing the computational time cost. An experimental rotor test rig was apply in this work. The rotor-bearing system include a flexible shaft supported by a plain journal bearing with fixed geometry. The collected data include phase and vibration amplitude for a rotation speed range passing through a critical rotation. The results for bearing parameters are compared to dynamic coefficients calculated by a commercial code. The unbalance magnitude and phase identified are compared to the trial mass applied and its position. The results are analyzed and discussed. Satisfactory results were obtained and the methodology shows to be applicable to identified, simultaneously, the bearing parameters and unbalance excitations.*

Keywords: *Journal Bearing, Optimization, Rotordynamics, Unbalance Response, Identification.*

1. INTRODUCTION

Fluid film bearing are widely used in industry turbomachinery and these components have high importance for reliable and uninterrupted operation. In most cases, bearing dynamic coefficients are the factors that governs the vibration characteristics. The importance of determining the bearings coefficient parameters in rotordynamic behavior prediction was mention in an early work was developed by Newkirk and Taylor (1925).

The bearing dynamic coefficient consideration is a critical and complex stage during rotordynamic projects. An inaccurate consideration may lead to erroneous determinations of critical rotation speed, unbalance responses and instabilities predictions. In addition, machines long term operation require predictive maintenance and the monitoring of variables associated directly or indirectly with the bearings condition is important for diagnostics and fault analysis.

The theoretical estimation of hydrodynamic bearing coefficients has always been considered sources of errors in the system behavior prediction. The dynamic parameters are obtained from the Reynolds differential equation. The analytical solution are feasible for the bearing coefficient determination by means of long bearing and short bearing consideration. (Meruane and Pascual, 2008). For complex bearing configurations such as the Tilting Pad bearings, it is necessary to use numerical methods to solve the Reynolds equation.

As a way to improve the prediction quality, some works proposed experimental methods to bearing coefficient determination. Lee and Hong (1989) and Tieu and Qiu (1994) proposed methods to determine bearing dynamic coefficients by using unbalance response measurements for different trial unbalance conditions and considering rigid rotors. De Santiago and San Andrés (2007a; 2007b) identified bearing parameters from impact response and unbalance responses. Tiwari and Chakravarthy (2006) proposed a method based in impulse responses that identified simultaneously the bearing dynamic coefficient and residual unbalance. Methods based in impulse excitation shown to be advantageous considering practical applications, due to the difficulty of exciting large rotors with impacts.

During the last decades, many works related to rotating systems have deal with nonlinear optimization technics as an inverse problem. This method allows determine the unknown parameters based on adjust of numerical to experimental data. Papadopoulos *et al.* (2008) proposed a method for journal bearing radial clearance identification. Bronkhorst (2010) implemented a methodology for flexible rotor balancing based in inverse problem and Genetic Algorithm (GA) for optimization routine and the unknown parameters were correctly identify.

The addition of bearing unknown parameters in the inverse problem increased the objective function complexity and a robust optimization method need be applied for global minima search. Hybrid methods that combine Genetic Algorithm (GA) with Nelder-Mead's Simplex Method are recommended (Renders and Flasse, 1996) to achieve a better performance on global searching. Aiming to determine simultaneously bearing dynamic coefficients and unbalance parameters, Kim *et al*. (2007) and Han *et al.* (2013) proposed different methodologies to identify unknown parameters by inverse problem. The works considered flexible rotors and bearing coefficients constants with rotational speed.

The use of optimization techniques based on hybrid methods shown to be promising. The disadvantage reported in papers (Han *et al.*, 2013; Kim *et al*. 2007) is the high computational time required, especially when it becomes necessary to solve the eigenvalues and eigenvectors problem for each GA individual to compose their frequency response. An alternative used in this work is the employment of the concepts applied by Espíndola and Silva (1992) and later by Dowbrawa Filho (2008). These concepts allows describing the dynamics of a compound system according to the generalized coordinates of the primary system. Thus, considering this concept, it is possible to describe the compound system in a modal sub-space of a primary system. This allows solving the eigenvalues and eigenvectors problem once, and these modal parameters are used to formulation of the compound system response. The consequence is gain a lot of computational time solving these problems.

The concept and methodology considered in this paper are presented in a previous work of Miliavacca *et al.* (2015). At this paper the bearing coefficient and unbalance parameters are simultaneously identified and numerical simulations based in a theoretical rotor-bearing system are developed. In the present paper, this methodology was adapted to allow considering the bearing coefficients varying with the rotation speed and the unbalance response collected in a laboratory test rig are used to determined unknown parameters.

2. METHODOLOGY

2.1 Mathematical model

The rotor equation can by derived from the application of *Lagrange's* equation as described by Lalanne and Ferraris (2001). Considering a steady state condition and a fixed rotation speed, Ω_{rpm}, the result rotor equation solvable using finite element method (FEM) is given by

$$[M]\{\ddot{q}(t)\}+\left(\left[C\left(\Omega_{rpm}\right)\right]+\left[G\left(\Omega_{rpm}\right)\right]\right)\{\dot{q}(t)\}+\left[K\left(\Omega_{rpm}\right)\right]\{q(t)\}=\{f(t)\}\,, \tag{1}$$

where $[M]$ represents the mass matrix, $[C]$ the viscous damping matrix, $[K]$ the stiffness matrix and $[G]$ the gyroscopic effect matrix. The vector q_i represents the nodal displacement given by $\{q_i\}=[u_i, w_i, \theta_i, \psi_i]^T$, where u and w are the displacements and θ and ψ the spins relative to X and Z directions, respectively. The modified Euller-Bernoulli beam element with two nodes and two degree of freedom per node is consider in this analysis and the elementary FEM matrices are well documented in Lalanne and Ferraris (2001) and Genta (2005).

The bearing are modeling with four stiffness coefficients (k_{xx}, k_{zz}, k_{xz}, k_{zx}) and four damping coefficients (c_{xx}, c_{zz}, c_{xz}, c_{zx}), speed dependent in general case. The unbalance excitation u_m is modeling as a mass m_u, with an eccentricity r_u situated in an angular position δ_m in $t=0$ seconds. Considering that the only excitation on the rotor came from mass unbalance, it is possible assume $\Omega=\Omega_{rpm}$ (Espíndola and Bavastri, 1997), and the rotor equation becomes

$$[M]\{\ddot{q}(t)\}+([C]+\Omega\,[G_1])\{\dot{q}(t)\}+[K]\{q(t)\}=\{f(t)\}\,, \tag{2}$$

where the matrix G_1 appears putting the rotation speed Ω_{rpm} in evidence on matrix G.

The identification methodology is described at Miliavacca *et al.* (2015). The early method is adapted to considered speed dependent bearing coefficients. The method considers the division of the system (Eq. 2) in primary and compound system. The primary system is defined as the real rotor properties and the initial bearing coefficients (previously), and the compound system (real system) includes the bearing stiffness and damping coefficients modification. The primary system is solving the eigenvalues and eigenvector problem once for initial and fixed values of the bearing stiffness and damping. The modal matrices are truncated inside a frequency range of interesting and the equation of motion of the compound system (primary system to which the increment of the bearing coefficients was included), is putting in the optimization environment. The aim is identify the unbalance parameters and the bearing modifications coefficients. The final values for bearing coefficients are obtained adding the optimal to the initial values after the optimization procedure. This theory is similar to structural modification concepts introducing by Brandon (1990).

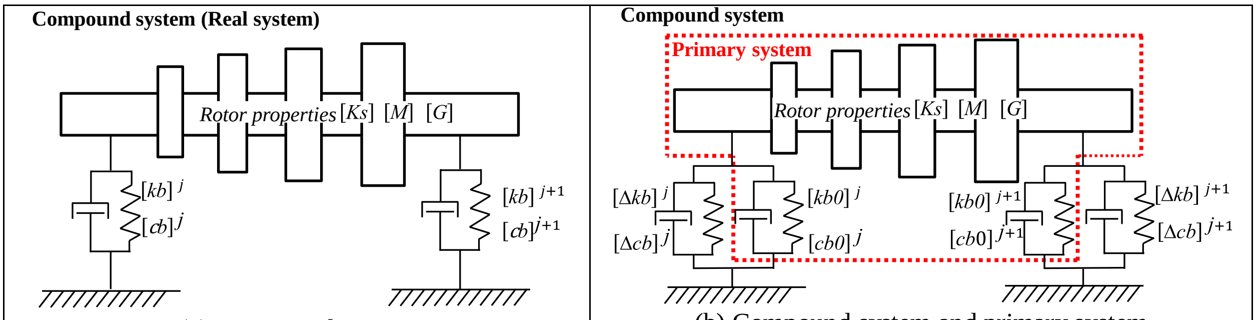
To define the primary and compound system, the stiffness and damping global matrices (Eq. 2) needs to be decomposed in shaft and bearing components, as

$$[K]_{n\times n}=[Ks]_{n\times n}+[Kb]_{n\times n} \quad \text{and} \quad [C]_{n\times n}=[Cb]_{n\times n}\,. \tag{3}$$

Here, $[Ks]$ is the global shaft stiffness matrix and $[Kb]$ and $[Cb]$ are the global bearing stiffness and damping matrices, respectively. The uppercase letters represent global matrices, with $n\times n$ dimension and n equal to the number of system degrees of freedom. No shaft damping is considered in this model. After, the bearings matrices can be decomposed in

$$[Kb]_{n\times n}=[Kb0]_{n\times n}+[\Delta Kb]_{n\times n} \quad \text{and} \quad [Cb]_{n\times n}=[Cb0]_{n\times n}+[\Delta Cb]_{n\times n}\,, \tag{4}$$

where $[Kb0]$ and $[Cb0]$ are the initial bearing stiffness and damping (related to the primary system) and $[\Delta Kb]$ and $[\Delta Cb]$ are the bearing stiffness and damping modification matrices. The bearing decomposition can be represented in a rotor-bearing system as Fig. 1, where $j=1$ *to* Nb and Nb is equal to the number of bearings with unknown dynamic coefficients. Figure 1 shows a simplified representation of the primary system delimited by the red dashed lines and compound system (real system). The bearing elementary matrices have dimensions *4 x 4* and are represented in lowercase letters.



(a) Compound system (b) Compound system and primary system

Figure 1. Rotor-bearing system with primary and compound system (simplified representation)

Considering Eq. (3) and Eq. (4), the equation for the primary system becomes

$$[M]\{\ddot{q}(t)\}+([Cb0]+\Omega\,[G_1])\{\dot{q}(t)\}+\big[[Ks]+[Kb0]\big]\{q(t)\}=\{f(t)\}\,, \tag{5}$$

and the motion equation of the compound system that includes the bearing modification matrices (the real system to be identified) can be rewritten by

$$[M]\{\ddot{q}(t)\}+([Cb0]+[\Delta Cb]+\Omega\,[G_1])\{\dot{q}(t)\}+\big[[Ks]+[Kb0]+[\Delta Kb]\big]\{q(t)\}=\{f(t)\}\,. \tag{6}$$

Taking Eq. (5) into the frequency domain by means of the Fourier transform and considering $\big[\widehat{M}\big]=\big[[M]\text{-}i[G_1]\big]$, results

$$\Big[-\Omega^2\big[\widehat{M}\big]+i\Omega[Cb0]+\big[[Ks]+[Kb0]\big]\Big]\{Q(\Omega)\}=\{F(\Omega)\}\,. \tag{7}$$

Representing Eq. (7) in a state-space (Ewins, 2000), results

$$\big[i\Omega[A]+[B]\big]\{Y(\Omega)\}=\{N(\Omega)\}\,, \tag{8}$$

where

$$\{Y(\Omega)\}=\begin{Bmatrix} Q(\Omega) \\ i\Omega\, Q(\Omega) \end{Bmatrix} \quad \text{and} \quad \{N(\Omega)\}=\begin{Bmatrix} F(\Omega) \\ 0 \end{Bmatrix}. \tag{9}$$

Considering the primary system, the eigenvalues problem in a state-space (Ewins, 2000) is defined as

$$[B][\Theta]=[\Lambda][A][\Theta] \quad \text{and} \quad [B]^T[\Psi]=[\Lambda][A]^T[\Psi]\,, \tag{10}$$

where,

$$[A]=\begin{bmatrix} [Cb0] & [\widehat{M}] \\ [\widehat{M}] & [0] \end{bmatrix}_{2n\times 2n} \quad \text{and} \quad [B]=\begin{bmatrix} \begin{bmatrix} [[Ks]+[Kb0]] & [0] \\ [0] & -[\widehat{M}] \end{bmatrix} \end{bmatrix}_{2n\times 2n}. \tag{11}$$

The model of the primary system was obtained only for unbalance excitation and the modal matrix is orthonormalized through $[A]$ matrix. The matrices $[\Theta]$ and $[\Psi]$ are compose by the left and right eigenvectors, respectively, and $[\Lambda]$ is the eigenvalues matrix.

On the other hand, the compound system the matrix $[\overline{A}]$ and $[\overline{B}]$ are given by

$$[\overline{A}]=[A]+[\widetilde{A}] \quad \text{and} \quad [\overline{B}]=[B]+[\widetilde{B}]\,, \tag{12}$$

where

$$[\widetilde{A}]=\begin{bmatrix} [\Delta Cb] & [0] \\ [0] & [0] \end{bmatrix}_{2n\times 2n} \quad \text{and} \quad [\widetilde{B}]=\begin{bmatrix} [\Delta Kb] & [0] \\ [0] & [0] \end{bmatrix}_{2n\times 2n}. \tag{13}$$

Considering the transformation $y(t)=\Theta\, p(t)$ and pre multiplying the Eq. (8) by Ψ^T, the motion equation of the compound system results

$$\left[i\Omega[I]+[\Lambda]+i\Omega[\Psi]^T[\widetilde{A}]\,[\Theta]+[\Psi]^T[\widetilde{B}][\Theta]\right]\{P(\Omega)\}=[\Psi]^T\{N(\Omega)\}\,. \tag{14}$$

Defining $[D]=\left[i\Omega[I]+[\Lambda]+i\Omega[\Psi]^T[\widetilde{A}]\,[\Theta]+[\Psi]^T[\widetilde{B}][\Theta]\right]$, the frequency response of the compound system is

$$\begin{Bmatrix} \{Q(\Omega)\} \\ i\Omega\{Q(\Omega)\} \end{Bmatrix}=\left[\Theta\left[D[\Omega]\right]^{-1}\Psi^T\right]\begin{Bmatrix} F(\Omega) \\ 0 \end{Bmatrix}. \tag{15}$$

According to Ewins (2000), if the model of the system has a high number of degree of freedom, it is possible works with a reduced number of the eigenvectors $[\Theta]$ and $[\Psi]$ and its equivalent eigenvalues, $[\Lambda]$. The model reduction consider a frequency range of interest and allows maintaining adequate response accuracy. The resulting truncated modal parameters are defined in this work as $[\widehat{\Theta}]$, $[\widehat{\Psi}]$ and $[\widehat{\Lambda}]$. The Eq. (15) is used the optimization environment to obtain the unknown parameters with accuracy and the only unknown parameters are the stiffness and damping modification coefficients inside the matrices $\widetilde{A}$ and $\widetilde{B}$ and unbalance excitation at vector $F(\Omega)$.

2.2 Nonlinear optimization problem

In the optimization routine, the error vector is defined as

$$\{d\}=\begin{Bmatrix} \left.\begin{matrix} Q^{num}{}_1 - Q^{exp}{}_1 \\ \vdots \\ Q^{num}{}_{Np} - Q^{exp}{}_{Np} \end{matrix}\right\} \Omega^1{}_{rpm} \\ \left.\begin{matrix} Q^{num}{}_1 - Q^{exp}{}_1 \\ \vdots \\ Q^{num}{}_{Np} - Q^{exp}{}_{Np} \end{matrix}\right\} \Omega^2{}_{rpm} \\ \vdots \\ \left.\begin{matrix} Q^{num}{}_1 - Q^{exp}{}_1 \\ \vdots \\ Q^{num}{}_{Np} - Q^{exp}{}_{Np} \end{matrix}\right\} \Omega^{Nr}{}_{rpm} \end{Bmatrix}_{N_p\times N_r}, \tag{16}$$

where Nr is the number of rotation speeds where the vibration value is read and Np is the number of reading positions (sensors). The experimental values Q^{exp} are obtained from an experimental set and the equivalent numerical Q^{num} from Eq. (15). Both experimental and numerical values in Eq. (16) are complex numbers.

Considering that the journal bearing coefficients are speed dependent these variables are assumed in this work varying with a quadratic function. As shown in Fig. 2, three different points were used for each bearing property to apply the interpolation function for each rotation speed considered at Eq. (16): Ω_{rpm}', the first rotation speed consider at unbalance response; Ω_{rpm}'', the central speed range; and Ω_{rpm}''', the highest rotation speed of unbalance response.

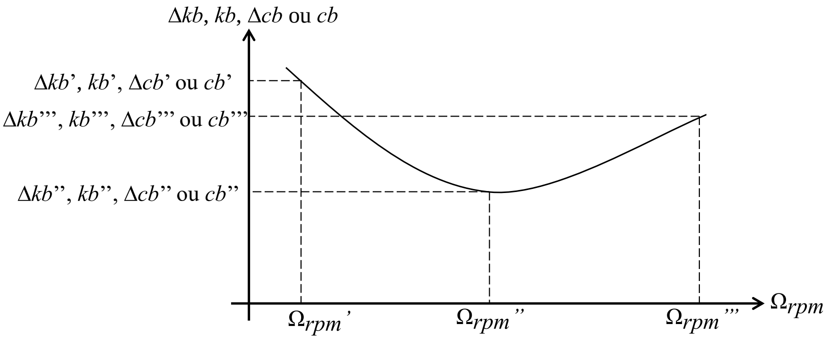


Figure 2. Bearing coefficient adjust by quadratic function

The objective function is defined as the quadratic error of the Eq. (19). Then, the classical nonlinear constrained optimization problem is defined as

$$
\begin{array}{ll}
\textit{minimize} & f_{obj}(x): R^n \rightarrow R \quad \text{and} \quad f_{obj}(x) = log_{10}\left(\sqrt{\{d\}^T \cdot (\{d\}^*)}\right)^2 \\
\textit{where} & (x) = \big[u_m{}^i,\ \delta_m{}^i,\ \Delta kb_{xx}{}^{'j},\ \Delta kb_{xx}{}^{''j},\ \Delta kb_{xx}{}^{'''j},\ \Delta kb_{zz}{}^{'j},\ \Delta kb_{zz}{}^{''j},\ \Delta kb_{zz}{}^{'''j},\ \Delta kb_{xz}{}^{'j},\ \Delta kb_{xz}{}^{''j},\ \Delta kb_{xz}{}^{'''j},\ldots \\
& \quad \Delta kb_{zx}{}^{'j},\ \Delta kb_{zx}{}^{''j},\ \Delta kb_{zx}{}^{'''j},\ \Delta cb_{xx}{}^{'j},\ \Delta cb_{xx}{}^{''j},\ \Delta cb_{xx}{}^{'''j},\ \Delta cb_{zz}{}^{'j},\ \Delta cb_{zz}{}^{''j},\ \Delta cb_{zz}{}^{'''j},\ldots \\
& \quad \Delta cb_{xz}{}^{'j},\ \Delta cb_{xz}{}^{''j},\ \Delta cb_{xz}{}^{'''j},\ \Delta cb_{zx}{}^{'j},\ \Delta cb_{zx}{}^{''j},\ \Delta cb_{zx}{}^{'''j},\ \ldots\big] \\
\textit{subject to} & 0 \leq u_m{}^i \leq u_m{}^i{}_{max}, \quad 0^o \leq \delta_m{}^i \leq 360^o \\
& \Delta kb^j{}_{min} \leq \Delta kb^{'j},\ \Delta kb^{''j},\ \Delta kb^{'''j} \leq \Delta kb^j{}_{max} \quad \text{and} \quad \Delta cb^j{}_{min} \leq \Delta cb^{'j},\ \Delta cb^{''j},\ \Delta cb^{'''j} \leq \Delta cb^j{}_{max}
\end{array}
\tag{17}
$$

for each bearing stiffness and damping elementary coefficient. At Eq. (17), n is the number of unknown parameters, $i=1$ *to* Nb and Nb is equal to the number of unbalance planes and $j=1$ *to* Nm and Nm is equal to the number of unknown bearings. The Euclidean norm applied in objective function allows include complex numbers, and the square of this value increases the convergence rate. In the most general case, each journal bearing is adjusted by 24 variables and each unbalance plane by 2 variables.

Figure 3 shows the general identification procedure. The experimental unbalance response is measured in a physical model and the instrument runout need to by subtracted. The rotor system is then modeled numerically assigning initial bearing coefficients (primary system). The modal parameters are obtained and truncated with an adequate number of modes. The optimization routine update the values of additional stiffness and damping for each bearing coefficient and each unbalance parameters until the stop criteria be reached.

Duo the objective function complexity the optimization routine starts with GA. The GA method continues until it reaches the maximum number of generations and the best individual is selected as the starting point for Nelder-Mead Simplex method. When the global optimum solution is identified the journal bearing and unbalance parameters are determined.

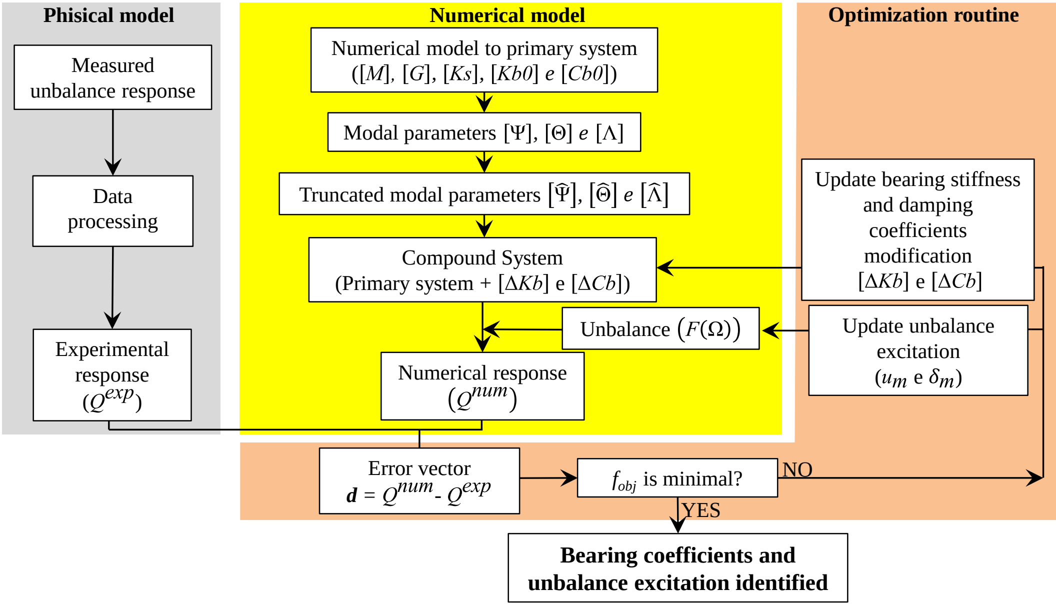


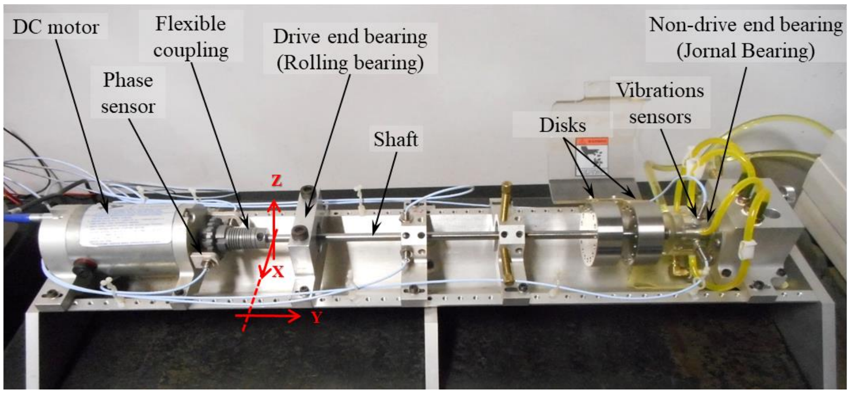
Figure 3. General identification procedure

3. EXPERIMENTAL STUDY

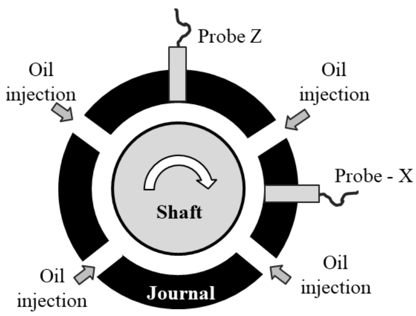
3.1 Description of the test rig and collected data

To validate the methodology experimentally, a test rig including a Bently Nevada RK-4 Rotor-kit was applied as shown at Fig. 4. A controllable DC motor is connected to one side of the shaft. A highly flexible coupling connects the shafts. At the drive end (DE), a rolling bearing was mounted on the shaft and. Two disks were installed near to the journal bearing in the non-drive end (NDE) and allow the insertion of unbalance mass. The journal bearing was fixed at the NDE and have four holes for oil injection as represented at Fig. 4.

The parameters of the rotor-kit system and journal bearing are listed in Tab. 1. The shaft Young's modulus was obtained experimentally by impact test measuring frequency response function and adjusting its equivalent numerical. The rolling bearing dynamic coefficient were analytically calculated according to Palmgren (1959) and are assumed to be known in this analysis. The magnitude of rolling bearing stiffness compared to other rotor dynamic characteristics allows to conclude that small discrepancies in its properties shows low influence at rotor dynamic response. The rotor was considered perfectly balanced and no shaft bow was considered. An unbalance mass was installed at inner disk as referenced at Tab. 1. The rotor-kit journal bearing has a plain and fixed geometry and an oil pump feeds the bearing with pressurized oil.



Teste rig



Journal bearing representation

Figure 4. Rotor-bearing test rig

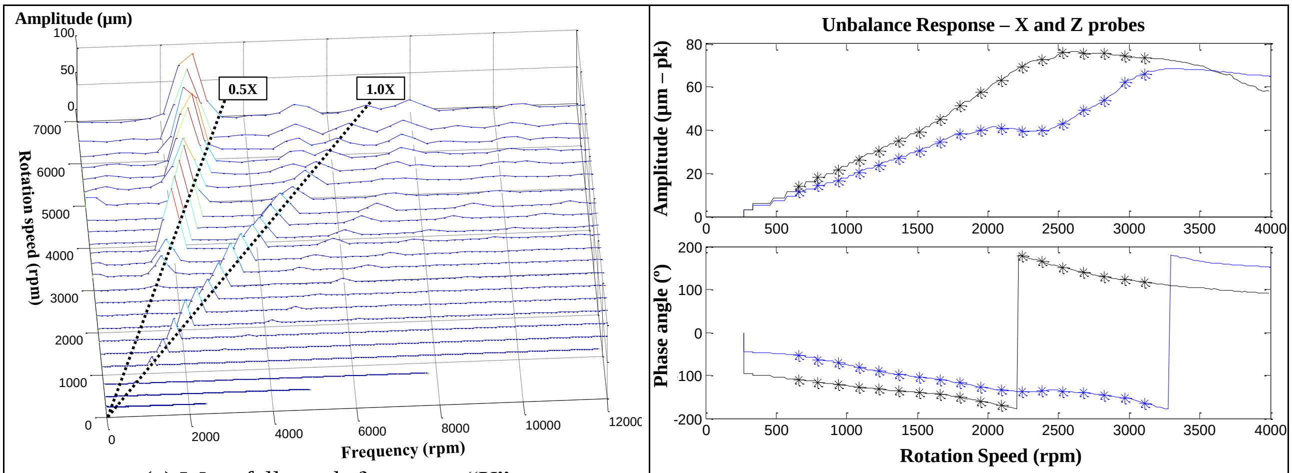
Aiming to know the limits of the rotor kit set, some evaluations were necessary prior to the unbalance response collection. These steps are essential in order to ensure the quality of the experimental data. The first evaluation was the electrical runout measured on the sensor track. The runout of the X and Z probe measures in slow rotation speed at journal bearing were, respectively, 10.55µm 0-p at -95.99º and 9.48µm 0-p at -0.69º. The values are considered appropriated for this application and these vectorial quantities as subtracted directly from the unbalance response readings.

Table 1. Parameters of the test rig

Shaft		**Rolling bearing coefficients**		**Journal bearing geometry**	
Shaft length (mm)	457	Rolling bearing position (mm)	50	Journal bearing position (mm)	444.3
Shaft diameter (mn)	10	k_{xx} ,k_{zz} (MN/m)	20	Shaft diameter (mm)	24.99
Young's modulus (GPa)	202	c_{xx} ,c_{zz} (N.s/m)	10	Bearing diameter (mm)	25.45
Poisson ratio	0.3	k_{xz}, k_{zx}, c_{xz},c_{zx}	0	Clearance (mm)	0.46
Specific gravity (kg/m³)	7747.15			Bearing lenght (mm)	25.4
Disks		**Unbalance mass(Disk 1)**		**Journal bearing lubricant**	
Width - all disks (mm)	25.0	u_m $(\times 10^{-6}\ kg.m)$	30	Injection pressure (psi)	7
Outside diameter - all disks (m)	75.0	δ_m (°)	0	Pumping temperature (ºC)	20
Disk 1 position (mm)	350.0	**Journal bearing static load**		Viscosity at @40ºC (cSt)	30.4
Disk 2 position (mm)	390.0	Static load (N)	14.8	Density (kg/m)	860

The proposed methodology considers only excitations from mass unbalance ($\Omega=\Omega_{rpm}$). So, just effects associated to running speed, denoted as "1X vibration", are mathematically modeled. The cross-coupling effects associated to this model of journal bearing causes the appearance of hydrodynamic instabilities when the rotation speed is raised. This phenomenon is characterized by the emergence of high vibration in frequencies at about "0.5X" the running speed. In this work, this phenomenon was identified experimentally by apply the Fast Fourier Transform (FFT) to one vibration signal measured in different rotation speeds. The results were grouped in the waterfall graph as showed at Fig. 5.a. The instability phenomena started in about 3100 rpm and are namely oil whirl and in about 5500 rpm the oil whip was evidence. These phenomena are well described at Genta (2005). Figure 5.a shows that until start the instability phenomenon, "1X" vibration components are predominant.

The rotor unbalance responses were measured at rundown procedures, and the recorded amplitude and phase angle without runout is showed at Fig.5.b. A rotation speed range starting at 659 rpm and finishing at 3111 rpm was selected. A total of 18 amplitude and phase vibration values, equally spaced, are chosen and marked as asterisk point at Fig. 5.b as unbalance response input for the optimization routine.



(a) Waterfall graph for sensor "X"

(b) Unbalance response measured

Figure 5. Vibration measured at position 444.3mm

3.2 Rotor model and algorithm variables

The rotor test rig showed at Fig. 4 was modeled in FEM. The shaft was modeled in 11 beam finite elements, and took into account the need for specific positioning of some nodes. Simulations performed with different number of elements indicated that this discretization was adequate.

The proposed methodology was apply to identified the unbalance magnitude and its angular position present at position 350 mm, and also the complete journal bearing dynamic coefficients, with four stiffness and four damping coefficients varying with the rotation speed. Table 2 shows the constraints for each bearing and unbalance property and the initial bearing dynamic coefficients. The initial values for principal stiffness and dumping coefficients were estimated from expected values and the cross-coupling dynamic coefficients are assumed with zero initial values. These values were used in the primary system for eigenvalues and eigenvectors calculation. Constrains descripted at Tab. 2 are assumed with values far enough from the real coefficients.

Considering the need to adjust a quadratic function for each stiffness and dumping property, the design project was composed by 26 variables. Due the objective function complexity, a GA population size of 10000 individuals and 20 generations were chosen for these analyses. The crossover rate was set in 85% and the mutation rate in 1%. After finishing the GA fixed number of generations, a Nelder-Mead Simplex Method is applies with a tolerance criteria of 1.0×10^{-8}.

Table 2. Constraints and initial values for the optimization routine

Variable	Lower bounds	Initial Values	Upper bounds
k_{xx} (N/m)	1.0×10^{2}	1.0×10^{5}	1.0×10^{7}
k_{zz} (N/m)	1.0×10^{2}	1.0×10^{5}	1.0×10^{7}
k_{xz} (N/m)	1.0×10^{-6}	0	1.0×10^{6}
k_{zx} (N/m)	1.0×10^{-6}	0	1.0×10^{6}
c_{xx} (N.s/m)	0	1.0×10^{3}	1.0×10^{5}
c_{zz} (N.s/m)	0	1.0×10^{3}	1.0×10^{5}
c_{xz} (N.s/m)	1.0×10^{-4}	0	1.0×10^{4}
c_{zx} (N.s/m)	1.0×10^{-4}	0	1.0×10^{4}
u_m (g.mm)	0.1	-	100.0
δ_m (º)	0	-	360.0

3.3 Results

Two analyzes were performed aiming to determine the unknown parameters and to show the influence of the modal reduction on the response: considering 100% and 70% of the eigenvectors. The final objective function value for both was similar and equal to the scalar -9.72. The minimum value for the truncated condition was achieved after 20699 seconds. Figure 6 represents the experimentally measured unbalance responses and the equivalents adjusted by the proposed methodology for the response composed with 70% of the eigenvectors. An adequate fit between the curves is observed.

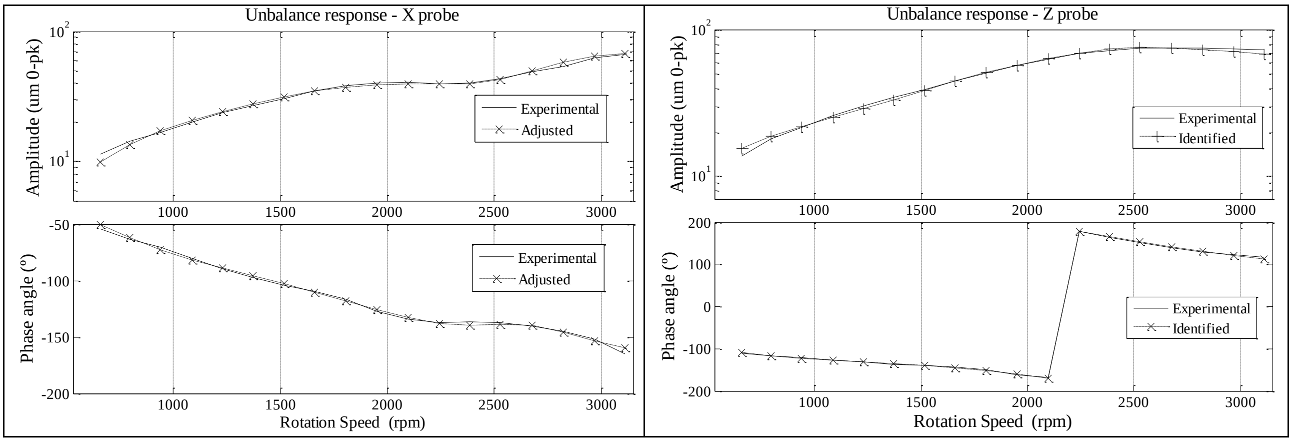


Figure 6. Experimental and numerically adjusted unbalance response

The results for the unbalance parameters and solution times are showed in Table 3. For variables associated with the unbalance the reference values are based on the mass added to the disk. As described, the residual unbalance wasn't measured prior to insertion of the test mass and the error showed on the results was attributed to this condition. However, the results obtained were close to the reference values.

Table 3. Results for unbalance parameters

Design variables	Reference values	Identified values (% error)	
		100% of eigenvectors	70% of eigenvectors
u_m (g.mm)	30	37.73 *(25.8%)*	37.67 *(25.6%)*
δ_m (º)	0	27.33 *(7.6%)*	27.11 *(7.5%)*
Solution time (seg.)	-	35613	20699

The coefficients of the hydrodynamic bearing determined by inverse problem were compared to those estimated numerically by means of the industrial code MAXBRG. This software is a finite element computer code that performs steady state hydrodynamic analysis for fluid journal bearings. The code input considered all geometry and lubricant properties from Tab. 1. Thermal effects and bearing surface deformation were not considered, and only static load was considered. Although MAXBRG code has been extensively tested with close agreement for industrial and conventional fixed geometry bearings, the test rig journal bearing has geometric peculiarities: the elevated diameter clearance and the journal holes for oil injection. However these numerical references were used only for comparative purposes with those obtained by proposed methodology. The results for bearing coefficient determination for the 70% of the eigenvectors applied and calculated by the industrial code are show at Fig. 7. Principal and cross stiffness and damping are plotted separately for better analysis.

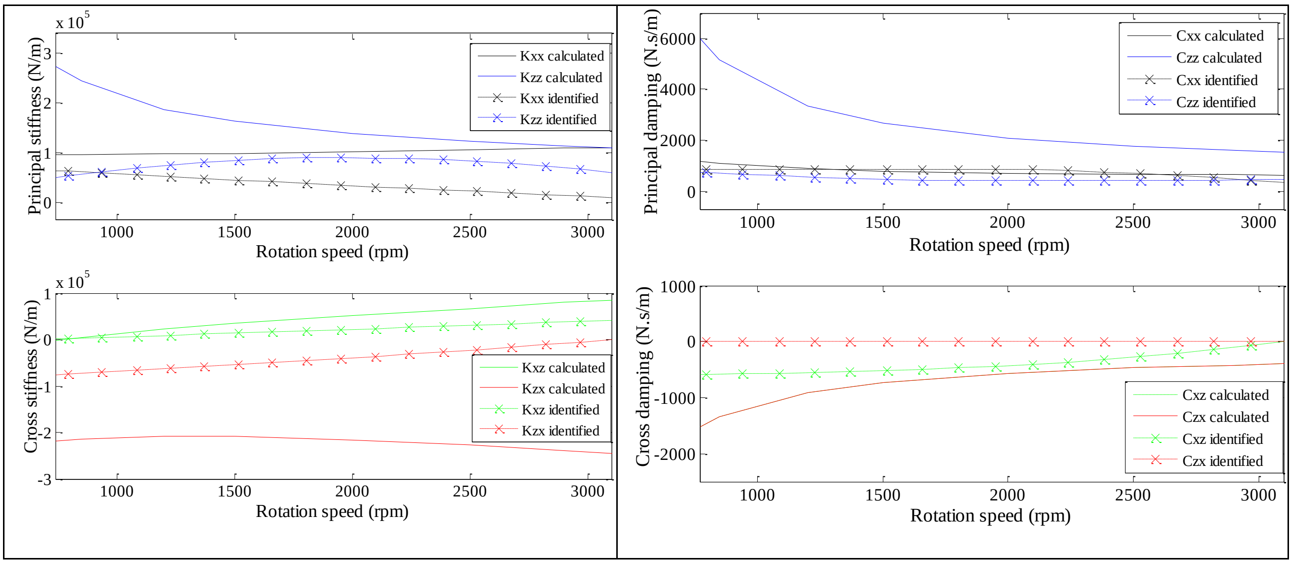


Figure 7. Calculated and identified journal bearing dynamic coefficients

For stiffness coefficients k_{zz} and k_{xz} the values determined experimentally via methodology approximated to those calculated numerically. The coefficient k_{xx} however, presented a spacing for higher rotations, maintaining the same order of magnitude between the values compared. For the cross stiffness parameter, k_{zx}, higher differences were also observed for higher rotations, while respecting the need for negative values according to numerical calculation. The damping coefficients obtained experimentally for parameters c_{xx} and c_{xz} approximated those obtained numerically. However, the comparison of the parameters c_{zz} and c_{zx} to their numerical equivalents showed differences.

The differences observed can be explained duo the difficult to modelling numerically the rotor kit journal bearing. The oil injection holes are positioned so that one of them is situated very close to the region of minimum thickness and maximum pressure of the oil film. This condition was not considered on the industrial code MAXBRG and could significantly affect the calculated dynamic coefficients. Another peculiar bearing geometrical condition is the elevated clearance between the journal bearing and the shaft. While industrial bearing usually consider clearances in the order of 0.2% of the shaft diameter, in this experimental application this ratio was about 2%. Another source of discrepancy between some parameters could be the static load considered to determine the calculated coefficients: in spite of the coupling has high flexibility, a small misalignment between shafts can change significantly the value of load.

In this way, some considerations of numerical modeling of the bearing may have caused errors in the reference values and discrepancies in the comparison of values identified through inverse problem. However coefficients are well fitted.

4. CONCLUSIONS

This work presents a method for simultaneous identification of journal bearing dynamic coefficients and mass unbalance distribution. The methodology was apply in an experimentally analysis with flexible rotor and a plain journal bearing.

The resolution of the eigenvalues problem just once associated with modal reduction improved significantly the computational processing time, without losing the quality of responses. The application of a hybrid optimization method allowed finding the objective function global minimum and allowed adjust precisely the experimentally and numerically unbalance responses. The unbalance parameters were identified with good agreement to the references. The four stiffness and four damping bearing coefficients varying with the rotation speed were identified. Some divergences between experimental and numerical calculated bearing properties were observed but are explained by the difficult in modeling the journal bearing chosen for the experimental analyses. The global results showed that the unknown parameters were correctly identified with agreement with the reference values. The correct modeling of known components like shaft and disks is important to not incorrectly identify unknown parameters. However these components have well-known models and experimental resources to obtain their properties. The methodology shows to be efficient and robust.

The identification algorithm has applicability in laboratory analyses and in real rotation machines. The methodology can be applied at bearing coefficient determination in new rotordynamic projects and rotor balancing machine and troubleshooting. Further investigation in real machines are required but the results are promising.

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