23rd ABCM International Congress of Mechanical Engineering
December 6-11, 2015, Rio de Janeiro, RJ, Brazil

IDENTIFICATION OF BEARING DYNAMIC COEFFICIENTS FROM UNBALANCE RESPONSES MEASUREMENTS AND NONLINEAR OPTIMIZATION TECHNIQUES

Alcemir Miliavacca
Universidade Federal do Paraná
alcemirm@yahoo.com.br

Rodrigo Bubniak Silvério
Universidade Federal do Paraná
r.bubniak.silverio@gmail.com

Carlos Alberto Bavastri
Universidade Federal do Paraná
bavastri@ufpr.br

Abstract. *Bearing dynamic features are important parameters that governs the vibration characteristics in rotating machines. However, these coefficients are usually difficult to determine and this fact can produce inaccurate overall results. This work proposes a methodology to determine, simultaneously, the bearing dynamic coefficients and unbalance distribution by structural modifications and non-linear optimization techniques. The method adjusts the unbalance response simulated by a finite element model to measured curves at run-downs procedure. As the objective function is multimodal, 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 dramatically the computational time cost. A numerical simulation was performed and the results are presented and discussed. An analysis on the GA optimization parameters, as well as the value of the initial bearing parameters in the eigenvalue problem was also performed. The methodology shows to be efficient and reliable to identified, simultaneously, the bearing parameters and unbalance excitations.*

Keywords: *Bearing, Optimization, Rotordynamic, Unbalance Response, Structural Modification*

1. INTRODUCTION

Rotation machines like pumps, turbines, generators and compressors are widely used in the industry and its application is usually related to critical production stages. The rotordynamic subject has high importance for reliable and uninterrupted operation of these machines.

The prediction of dynamic machine behavior is essential for proper design and machine vibration control. Especially by the increase in demand of high speed and lightweight rotors, bearing dynamic coefficients are the main factors that governs the vibration characteristics. Newkirk and Taylor (1925) performed an early work that mentions the importance of determining the bearings coefficient parameters in rotordynamic behavior. In that paper, the authors investigated instability phenomena and the causes are associated to the bearing dynamic characteristics.

The bearing coefficient consideration is a critical and complex stage during rotordynamic analysis. Historically, the theoretical estimation of these parameters has always been considered sources of errors in the system behavior prediction. The dynamic stiffness are obtained analytically by Hertz for rolling bearing types and the damping coefficients are small. For hydrodynamic bearings the analytical solution for bearing coefficients are only feasible for specifically bearing geometries. For more complex bearing configurations such as the Tilting Pad bearings it is necessary to use numerical methods.

In the last decades, researchers around the world have proposed methods for the experimental determination of the bearings coefficients as a way to improve the prediction performance of these systems. Lee and Hong (1989) and Tieu and Qiu (1994) identified the bearing dynamic coefficients by using unbalance response measurements for different trial unbalance conditions, for rigid rotor. 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 on unbalance excitation shown to be advantageous when compared to impulse excitations because it is easier excite large and real rotors.

In past few years, nonlinear optimization technics have been introduced in rotating systems analysis. These solutions allows determine the unknown parameters based on adjust of numerical to experimental data. However, the complexity of the objective function for these kind of problems require the use of robust optimization methods like Genetic Algorithm (GA). The reported disadvantage of this method is the required time to find the solution. Hybrid methods that combine GA with Quasi-Newton or Nelder-Mead's Simplex Method are applied (Renders and Flasse 1996) to achieve a better performance on global searching. In a recent work, Bronkhorst (2010) implemented a methodology for flexible rotor balancing. It was based on a hybrid optimization method that adjust numerical to experimental unbalance response data. Kim *et al.* (2007) and Han *et al.* (2013) proposed different methodologies for simultaneous bearing and unbalance parameters identification on flexible rotors to reduce the computational time expense.

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 still 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) for passive vibration control through dynamic neutralizers. 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.

2. ROTOR MODELING

A single rotordynamic system is composed by shaft, disks and bearings. The rotor equation can by derived from the application of *Lagrange's* equation to each rotor component (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

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

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 $\{q_i\}=[u,\ w,\theta,\psi]^T$, 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. The elementary FEM matrices are well documented in Lalanne and Ferraris (2001) and Genta (2005).

The bearing are generally modeling as support elements, with four stiffness coefficients (k_{xx}, k_{zz}, k_{xz}, k_{zx}) and four damping coefficients (c_{xx}, c_{zz}, c_{xz}, c_{zx}). 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. Bearings coefficients and unbalance are represented at Fig. 1. The bearing coefficients and unbalance forces are added directly to the node where the bearing and unbalances masses are considered on the model.

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

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

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

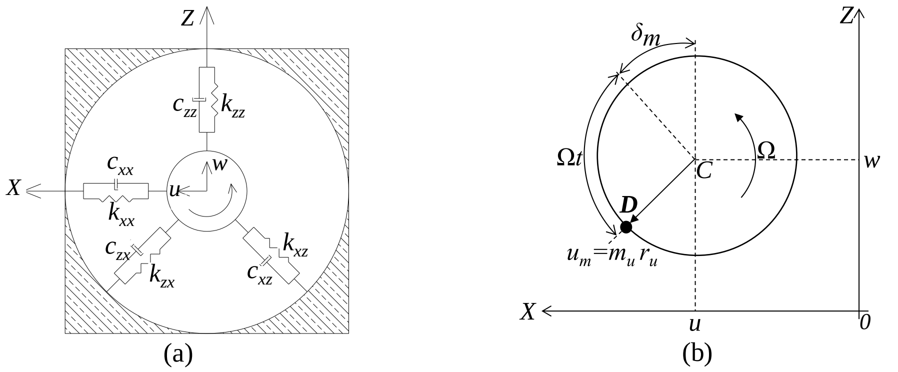


Figure 1. Bearing stiffness and damping coefficients (a) and unbalance representation (b)

3. METHODOLOGY

3.1 Primary and Compound System

The methodology is based on structural modifications and the fallowing summarized steps are applied:

- Step 1 - The real rotor properties to which the initial bearing coefficients are defined (primary system);
- Step 2 - The eigenvalues and eigenvectors of the primary system are found once;
- Step 3 - Modal matrices are truncated;
- Step 4 - 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;
- Step 5 - Bearing coefficients are obtained adding the optimum modification values (from step 4) to the initial values.

Equation (2) can be divided in primary and compound system. In the proposed methodology, the concept of the primary system is introducing to solving the eigenvalues and eigenvector problem once for initial and fixed values of the bearing stiffness and damping. The final values for these 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 matrices (Eq. 2) needs to be decomposed as

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

where $[Ks]$ is the global shaft stiffness matrix and $[Kb]$ and $[Cb]$ are the global bearing stiffness and damping matrices, both with $n\times n$ dimension, where n is the number of system degrees of freedom. No shaft damping is considered.

Further, 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 (used in the so called primary system) and $[\Delta Kb]$ and $[\Delta Cb]$ are the bearing stiffness and damping modification matrices that will be used in optimization environment. So, 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 including the bearing modification matrices, the motion equation of the compound system (the real system to be identified) can be rewritten by

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

The rotor-bearing system and the bearing decomposition can be represented as Fig. 2 where $j=1$ *to* Nb and Nb is equal to the number of bearings with unknown coefficients. Figure 2 shows the primary system delimited by the red dashed lines and compound system (real system). The bearing matrices represented at Fig. 2 are the elementary bearing matrices and have dimensions *4 x 4*, with coefficients represented as Fig. 1.

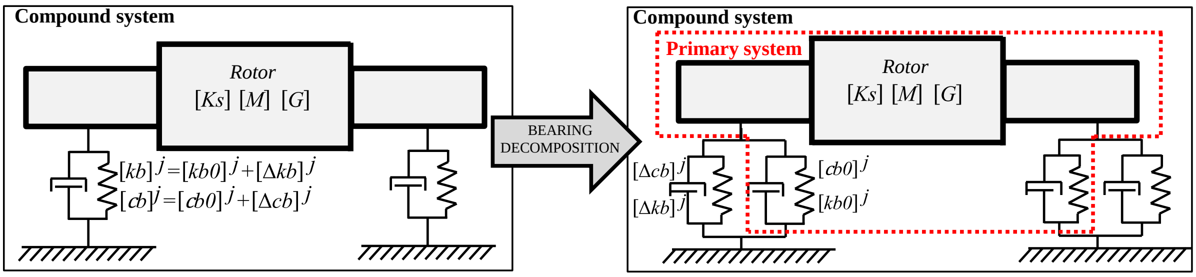


Figure 2. Rotor-bearing system with primary and compound system (simplified model)

Applying the Fourier transform to the primary system equation (Eq. 5) and considering $\big[\widehat{M}\big]=\big[[M]\text{-}i[G_1]\big]$, results

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

In a state-space (Ewins, 2000), the Eq. (7) is given by

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

where

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

The eigenvalues problem in a state-space (Ewins, 2000) is defined as

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

where,

$$[A]=\begin{bmatrix} [Cb0] & [\widehat{M}] \\ [\widehat{M}] & [0] \end{bmatrix}_{2n\times 2n} \tag{11}$$

and

$$[B]=\left[\begin{bmatrix} [[Ks]+[Kb0]] & [0] \\ [0] & -[\widehat{M}] \end{bmatrix}\right]_{2n\times 2n}. \tag{12}$$

The matrices $[\Theta]$ and $[\Psi]$ are compose by the left and right eigenvectors, respectively, and $[\Lambda]$ is the spectral matrix. The modal model of the primary system was obtained only for unbalance excitation and the modal matrix is orthonormalized through $[A]$ matrix. For the other hand, the compound system the matrix $[\overline{A}]$ and $[\overline{B}]$ are given by

$$[\overline{A}]=[A]+[\widetilde{A}] \tag{13}$$

and

$$[\overline{B}]=[B]+[\widetilde{B}], \tag{14}$$

where

$$[\widetilde{A}]=\begin{bmatrix} [\Delta Cb] & [0] \\ [0] & [0] \end{bmatrix}_{2n\times 2n} \tag{15}$$

and

$$[\widetilde{B}]=\begin{bmatrix} [\Delta Kb] & [0] \\ [0] & [0] \end{bmatrix}_{2n\times 2n}. \tag{16}$$

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

$$\left[i\Omega[I]+[\Lambda]+i\Omega[\Psi]^T[\widetilde{A}]\,[\Theta]+[\Psi]^T[\widetilde{B}][\Theta]\right]\{P(\Omega)\}=[\Psi]^T\{N(\Omega)\}. \tag{17}$$

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

$$\begin{Bmatrix} \{Q(\Omega)\} \\ i\Omega\{Q(\Omega)\} \end{Bmatrix}=\left[\Theta[D[\Omega]]^{-1}\Psi^T\right]\begin{Bmatrix} F(\Omega) \\ 0 \end{Bmatrix}. \tag{18}$$

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 in the modal matrices, $[\Theta]$ and $[\Psi]$, and its equivalent eigenvalues, $[\Lambda]$, inside a frequency range of interest and maintaining adequate accuracy with a reduced number of equations (Ewins, 2000). The resulting truncated modal parameters are denominated $[\widehat{\Theta}]$, $[\widehat{\Psi}]$ and $[\widehat{\Lambda}]$.

In Eq. (18), the only unknown parameters are the stiffness and damping modification coefficients and unbalance excitation. The Eq. (18) is used the optimization environment to obtain the unknown parameters with accuracy.

3.2 Optimization problem

The objective function is defined as the difference between the experimentally response and the set numerical response by changing the variables that compose the vector design variables (bearing and unbalance parameters). The error vector is defined as

$$\{d\}=\left\{\begin{matrix} \left.\begin{matrix} Q^{num}_{1} - Q^{exp}_{1} \\ \vdots \\ Q^{num}_{Np} - Q^{exp}_{Np} \end{matrix}\right\} \Omega^{1}_{rpm} \\ \left.\begin{matrix} Q^{num}_{1} - Q^{exp}_{1} \\ \vdots \\ Q^{num}_{Np} - Q^{exp}_{Np} \end{matrix}\right\} \Omega^{2}_{rpm} \\ \vdots \\ \left.\begin{matrix} Q^{num}_{1} - Q^{exp}_{1} \\ \vdots \\ Q^{num}_{Np} - Q^{exp}_{Np} \end{matrix}\right\} \Omega^{Nr}_{rpm} \end{matrix}\right\}_{N_p \times N_r}, \tag{19}$$

where Np is the number of reading positions (sensors) and Nr is the number rotation speeds where the vibration value is read. The experimental values Q^{exp} are obtained from an experimental set and the equivalent numerical Q^{num} from Eq. (18). Both experimental and numerical values in Eq. (19) are complex.

Then, the classical nonlinear constrained optimization problem is defined as

$$\begin{array}{ll} minimize & f_{obj}(x): R^n \rightarrow R \quad \text{and} \quad f_{obj}(x) = log_{10}\left(\sqrt{d^T \cdot (d^*)}\right)^2 \\ where & (x) = \left[u_m{}^i, \delta_m{}^i, \Delta kb_{xx}{}^j, \Delta kb_{xz}{}^j, \Delta kb_{zx}{}^j, \Delta kb_{zz}{}^j, \Delta cb_{xx}{}^j, \Delta cb_{xz}{}^j, \Delta cb_{zx}{}^j, \Delta cb_{zz}{}^j\right] \quad , \\ subject\ to & u_m{}^i \geq 0, \quad 360^o \geq \delta_m{}^i \geq 0^o \\ & \Delta kb^j{}_{min} \leq \Delta kb_{xx}{}^j, \Delta kb_{zz}{}^j, \Delta kb_{xz}{}^j, \Delta kb_{zx}{}^j \leq \Delta kb^j{}_{max} \\ & \Delta cb^j{}_{min} \leq \Delta cb_{xx}{}^j, \Delta cb_{zz}{}^j, \Delta cb_{xz}{}^j, \Delta cb_{zx}{}^j \leq \Delta cb^j{}_{max} \end{array} \tag{20}$$

where 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.

3.3 Optimization routine

Figure 3 shows the flowchart for the proposal methodology. The experimental unbalance response is measured in a physical model. The rotor system is then modeled numerically assigning initial bearing coefficients. The modal parameters are obtained and truncated with an adequate number of modes. The optimization routine update the values of additional stiffness and damping bearing coefficients and unbalance parameters until the stop criteria be reached.

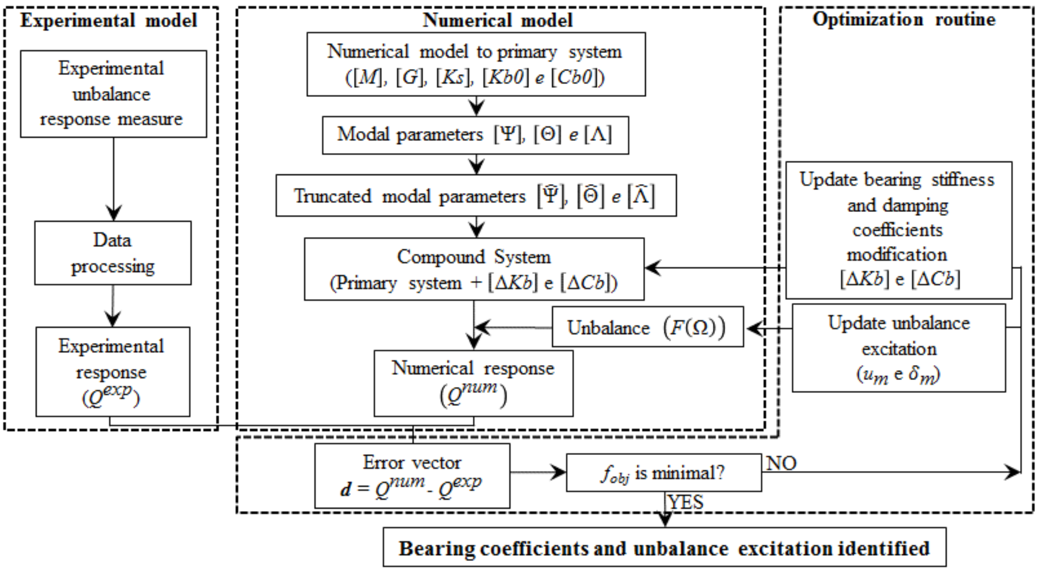


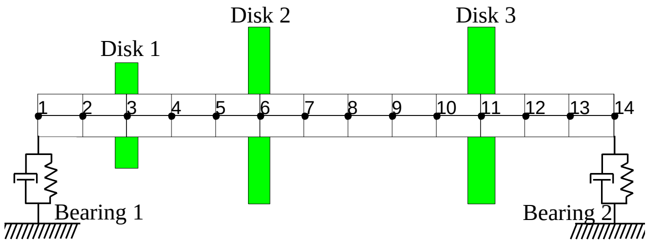
Figure 3. Flowchart of the methodology

The optimization code starts with GA the duo the objective function complexity. The code begins with the random generation of the initial population through the problem-imposed constraints. From this population, the GA operators are applied. A crossover rate of 85% and mutation rate of 1% are selected. The population size and the number of generations are determined depending on the complexity of the problem and more detail will be shown in the further analysis. In order to favor the best individuals an elitism of 2,5% of the population is chosen. The GA method continues until it reaches the maximum number of generations and the best individual is selected as the starting point for direct search method. The Nelder-Mead Simplex Method is applies after the GA method, and a tolerance criteria of 1 x 10^{-8} is chosen.

4. SIMULATIONS

To validate the proposed methodology a numerical evaluation was performed. All the simulations contained in the fallowing sections are based in the Lalanne's rotor model (Lalanne and Ferraris, 2001), as detailed in Fig. 4.

FEM simulations provided all input frequency responses used in methodology validation. Modified Euler-Bernoulli beam elements were considered and the rotor discretization are shown Fig. 4.



Shaft	
Shaft Length (m)	1,3
Shaft diameter (m)	0,1
Young's modulus (GPa)	200
Poisson ratio	0,3
Disks	
Width - all disks (m)	0,05
Diameter - Disk 1 - node 3 (m)	0,12
Diameter - Disk 2 - node 6 (m)	0,20
Diameter - Disk 2 - node 6 (m)	0,20

Bearing coefficients		
Brg 1 (node 1) and Brg 2 (node 14)		
k_{xx}	(MN/m)	50
k_{zz}	(MN/m)	70
c_{xx}	(N.s/m)	500
c_{zz}	(N.s/m)	700
k_{xz}, k_{zx}, c_{xz}, c_{zx}		0
Unbalance (Disk 2 - node 6)		
u_m	($\times 10^{-6}$ $kg.m$)	200
δ_m	(º)	0

Figure 4. Lalanne's rotor model (Lalanne and Ferraris, 2001)

4.1 Objective function evaluation

The behavior of the objective function according to the proposed methodology was analyzed graphically. The rotor was modified to make it symmetrical with $k_{xx}= k_{zz}=70\ MN/m$ and $c_{xx}= c_{zz}=700\ N.s/m$ for both bearings (nodes 1 and 14). The objective was decrease the number of design variables, since it is only possible plot two design variables in a three-dimensional graphic. The unbalance response for this rotor was simulated into a rotation speed range of 2500 to 4500 rpm, getting 30 amplitude and phase vibration values, equally spaced in the mentioned speed range. The vibration readings are related to the node 12, for both directions. The unknown parameters (design variables) considered are principal stiffness at bearing 2 and unbalance (u_m) at disk 2. The bearing 1 coefficients were assumed to be known. The initial value for bearing 2 principal stiffness is *0,1MN/m*. Figure 5 presents the objective function values values versus stiffness and unbalance conditions.

As show in Fig. 5, the objective function for this problem has many local minimum. The global optimum solution has a narrow profile. As the number of design variables increases, a more complicated objective function is expected. A robust global optimization method is essential to these kind of problems.

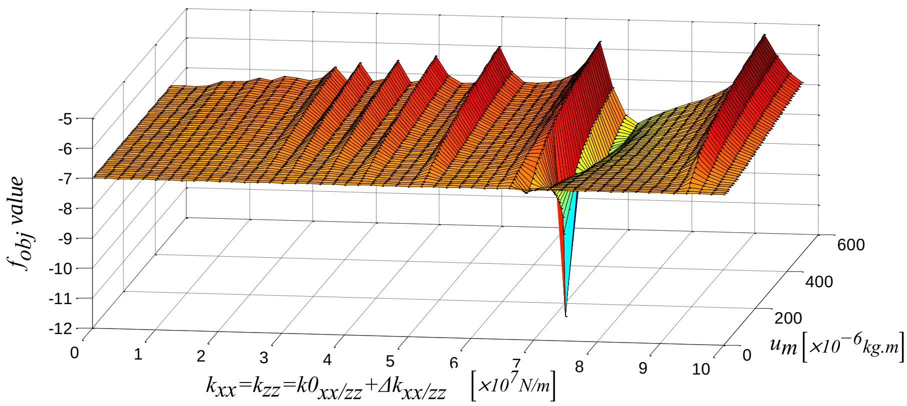


Figure 5. Objective function behavior

4.2 Determination of bearing 2 coefficients and disk 2 unbalance excitation

The proposed methodology was apply to determine the bearing 2 coefficients (node 14) and the unbalance excitation at disk 2 (Fig. 4). The unbalance response was measured at node 12 in both directions. A rotation speed range used was

2500 to 4500 rpm. Thirty amplitude and phase vibration versus rotation speed points, equally spaced in rotation speed were considered as experimental input data. The design project was composed by $u_m, \delta_m, \Delta k_{xx}, \Delta k_{zz}, \Delta c_{xx}, \Delta c_{zz}$, and subjected to the constrains $\left(1\times10^5 N.m \le k_{xx}, k_{zz} \le 1\times10^9 N.m\right)$, $\left(0 \le c_{xx}, c_{zz} \le 1\times10^3 N.s/m\right)$, $(0 \le u_m \le 500\, g.mm)$ and $(0 \le \delta_m \le 360^o)$.

The first analysis considered different GA parameters. The population size and number of generations were changed, and the initial values for principal bearing stiffness and damping coefficients were fixed in 1×10^6 *N/m* and *1000 N.s/m*. The results were summarized in Tab. 1. The identified values compared to the reference indicated very good results for populations size greater than 300 individuals, with error lower than 0,1% for all bearing coefficients and unbalance parameters. A population size of 300 individuals shows to be inadequate for this problem solution.

Table 1. Results for different population size and number of generations

Design variables	Reference values	Number of generations	Identified values (*% error*) Size of population (number of individuals) Pop. = 1300	Pop. = 700	Pop. = 300
k_{xx} *(MN/m)*	50	20	50,00 *(0,0%)*	50,00 *(0,0%)*	50,00 *(0,0%)*
		15	50,00 *(0,0%)*	50,00 *(0,0%)*	10,28 *(-79,4%)*
k_{zz} *(MN/m)*	70	20	69,97 *(0,0%)*	69,97 *(0,0%)*	69,97 *(0,0%)*
		15	69,97 *(0,0%)*	69,97 *(0,0%)*	43,92 *(-37,3%)*
c_{xx} *(N.s/m)*	500	20	499,40 *(-0,1%)*	499,49 *(-0,1%)*	557,00 *(11,4%)*
		15	499,49 *(-0,1%)*	499,45 *(-0,1%)*	0,00 *(-100%)*
c_{zz} *(N.s/m)*	700	20	699,90 *(0,0%)*	699,85 *(0,0%)*	0,04 *(-100%)*
		15	699,85 *(0,0%)*	699,85 *(0,0%)*	0,00 *(-100%)*
u_m *(g.mm)*	200	20	199,97 *(0,0%)*	199,97 *(0,0%)*	199,83 *(-0,1%)*
		15	199,97 *(0,0%)*	199,98 *(0,0%)*	119,19 *(-40,4%)*
δ_m *(º)*	0	20	0,00 *(0,0%)*	360,00 *(0,0%)*	0,00 *(0,0%)*
		15	360,00 *(0,0%)*	360,00 *(0,0%)*	360,00 *(0,0%)*
Solution time (sec)	-	20	8568	4738	2298
		15	6906	4106	1753

The second analysis group considered different initial values for principal bearing stiffness and damping coefficients and different percentage of eigenvectors used to calculate system unbalance response. The obtained results are described at Tab. 2. The GA population size of 700 individuals and a number of generations of 15 were chosen as a standard for these analyses.

For the initial bearing stiffness coefficients of 1×10^6 *N/m* and 1×10^8 *N/m* the error was too low for all cases, even with only 50% of eigenvectors at the response. For the initial stiffness of 1×10^3 *N/m*, the results showed good agreement except for the damping coefficients. When the initial stiffness was 1×10^9 *N/m*, the results showed very good agreement when 100% of the eigenvectors were used, but no convergence when eigenvectors were 50% truncated. Possibly, the imposed initial high bearing stiffness moves the primary system natural frequencies too away of the real natural frequencies, and a large number of eigenvectors were necessary to compose the system response. Although tested in this work to known the methodology boundary, the initials values of 1×10^3 *N/m* and 1×10^9 *N/m* are not suitable for a practical identification simulation, and these values would be as close as possible than the expected. Also, a reduced solution time is possible when reducing the number of eigenvectors used at the response.

Table 2. Results for different initial bearing coefficients and eigenvectors percentage used in the response

Design variables	Reference values	% of eigenvectors used	Identified values (*% error*) Initial values for principal bearing stiffness and damping $kb0_{xx/zz}=1x10^3$ *N/m* $cb0_{xx/zz}=100$ *N.s/m*	$kb0_{xx/zz}=1x10^6$ *N/m* $cb0_{xx/zz}=100$ *N.s/m*	$kb0_{xx/zz}=1x10^8$ *N/m* $cb0_{xx/zz}=100$ *N.s/m*	$kb0_{xx/zz}=1x10^9$ *N/m* $cb0_{xx/zz}=100$ *N.s/m*
k_{xx} *(MN/m)*	**50**	100%	50,00 *(0,0%)*	50,00 *(0,0%)*	50,00 *(0,0%)*	50,00 *(0,0%)*
		50%	49,98 *(0,0%)*	49,98 *(0,0%)*	49,92 *(-0,2%)*	*No convergence*
k_{zz} *(MN/m)*	**70**	100%	69,99 *(0,0%)*	69,97 *(0,0%)*	70,00 *(0,0%)*	69,99 *(0,0%)*
		50%	69,61 *(-0,6%)*	69,62 *(-0,5%)*	69,89 *(-0,2%)*	*No convergence*
c_{xx} *(N.s/m)*	**500**	100%	496,20 *(-0,8%)*	504,00 *(0,8%)*	500,01 *(0,0%)*	496,10 *(-0,8%)*
		50%	461,00 *(-7,8%)*	488,89 *(-2,2%)*	496,42 *(-0,7%)*	*No convergence*
c_{zz} *(N.s/m)*	**700**	100%	698,20 *(-0,3%)*	698,00 *(-0,3%)*	700,03 *(0,0%)*	700,00 *(0,0%)*
		50%	782,00 *(11,7%)*	695,84 *(-0,6%)*	704,70 *(0,7%)*	*No convergence*
u_m *(g.mm)*	**200**	100%	199,95 *(0,0%)*	199,96 *(0,0%)*	200,00 *(0,0%)*	200,00 *(0,0%)*
		50%	199,51 *(-0,2%)*	199,52 *(-0,2%)*	199,91 *(-0,2%)*	*No convergence*
δ_m *(º)*	**0**	100%	359,96 *(0,0%)*	0,00 *(0,0%)*	360,00 *(0,0%)*	360,00 *(0,0%)*
		50%	0,00 *(0,0%)*	0,00 *(0,0%)*	0,00 *(0,0%)*	*No convergence*
Solution time (sec)	-	100%	4239	3410	3801	5041
		50%	2193	1616	1938	*No convergence*

5. CONCLUSIONS

This paper proposes an effective and robust method for simultaneous identification of bearing dynamic coefficients and unbalance information in a flexible rotor-bearing system. The methodology has the flexibility to incorporate any number of bearings and balancing planes.

The application of a hybrid optimization method allows finding the objective function global minimum. For the example proposed, the methodology provided good results for an adequate GA population size using a reduced number of generations. A modal reduction by the eigenvectors truncation is possible without losing the quality of responses. The resolution of the eigenvalues problem just once associated with modal reduction improved significantly the computational processing time. On the other hand, if the high stiffness value is used as the initial value, the boundary condition can be very strong and the base formed may not adequately represent the dynamic of the rotor. However, a wide range of initial bearing coefficients could be used to formulate the real rotor response. The global results showed that the unknown parameters were correctly identified with very good agreement to the reference values. The methodology shows to be efficient and robust.

The identification algorithm has applicability on field to identify bearing dynamic and unbalance. A practical way of acquiring real machine vibration data could be during a machine run-down procedure, using the available machine instrumentation. Further investigation in a test rig are required in the next work but the results are promising.

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