

Rotor Balancing Using Parametric Excitation

Ricardo Ugliara Mendes ¹, Fadi Dohnal ²

¹ Laboratory of Rotating Machinery, Faculty of Mechanical Engineering, University of Campinas (UNICAMP), R Mendeleev 200, 13083-970, Campinas/SP, Brazil.

² Division for Mechatronics Lienz, University of Health Sciences, Medical Informatics and Technology (UMIT), Linker Iselweg 21, 9900 Lienz, Austria.

Abstract: The traditional balancing procedure consists of adding test masses and accelerating the rotor up to the critical speeds correspondent to the modes to be balanced. Recently, a novel method has been developed where the rotor is accelerated at the first critical speed only. The responses of higher modes are obtained by introducing a parametric excitation at the combination resonance frequencies of the rotor through the active magnetic bearings; resulting in an energy transfer from the first critical speed to the higher modes. This method reduces the balancing procedure time and the risk of high levels of vibration when the unbalance is critical and the rotor crosses higher critical speeds. However, the parametric excitation should be tuned very carefully to maximize the energy transfer to higher modes. Among the parameters to be tuned, the phase difference between the parametric excitation applied in the bearings is particularly important, once it can drastically influence the energy transfer. Therefore, this survey focuses on the influence of the phase difference between the bearings parametric excitation in the energy transferred from the lower critical speed to higher modes. The model of a flexible rotor sustained by active magnetic bearings is used in numerical simulations to evaluate the influence of the aforementioned parameter in the energy transfer to the second and third modes of the rotor.

Keywords: Influence Coefficient Balancing, Parametric Excitation, Parametric Combination Resonance, Sensitivity Analysis

INTRODUCTION

Balancing is a mandatory procedure for every rotor in industry and a review of the traditional balancing methods is found in the work of Kellenberger (1987). The widely used influence coefficient method consists of applying test masses to the balancing planes of the rotor and measuring its vibration responses close to the critical speeds to be balanced. The relations obtained from the measured vibrations and the test masses are used to calculate influence coefficients and, then, the correction masses. The main drawbacks of this methodology are the time consumed and the possibility of high levels of vibration when the original unbalance is critical.

Recently, Mendes and Dohnal (2018) proposed a novel method where the rotor must be accelerated up to the first critical speed only. In this method, energy is transferred from the first critical speed to higher modes via parametric excitation, as shown in Eq. (1). When the parametric excitation is applied in a combination resonance frequency $\nu^{kl,n}$ (correspondent to the difference between the damped frequencies of two modes: ω_k and ω_l), the system presents a behavior similar as if kinetic energy was transferred between the two modes involved; this effect was analytically proven by Dohnal (2012). A different approach for balancing rotors using the concept of parametric amplification was developed by Tresser et al. (2018). In Eq. (1), n is the order of the resonance frequency, ε is the amplitude of the parametric excitation and k is the stiffness value when the parametric excitation is turned off.

$$k_p(t) = k \left(1 + \varepsilon \sin(\nu^{kl,n} t + \phi) \right), \quad \nu^{kl,n} = |\omega_k - \omega_l| / n, \quad k, l = 1, 2, \dots \quad (1)$$

Tondl (1998) was the first to highlight the beneficial effects of parametric excitation at combination resonances. Dohnal (2012 and 2018) and Dohnal and Tondl (2013) presented, both theoretically and experimentally, the potential of parametric excitation at different combination resonances to increase the system damping. In these works, the parametric excitation was induced in the stiffness $k_p(t)$ of the active magnetic bearings (AMBs); the change in the bearing stiffness is obtained by changing the proportional gain of the PID controller, as shown in Eq. (1). In the cases involving a rotor supported on AMBs, the parametric excitations induced in the bearings were always in phase ($\phi = 0$, in Eq. (1)). Recently, the concept of increasing the system damping using parametric excitation was extended to rotors in fluid-film bearings by Chasalevris and Dohnal (2016).

In this work, the novel balancing method is briefly introduced, and the sensitivity of the kinetic energy transfer between modes to two parameters is analyzed: the phase difference between the parametric excitation induced in the AMBs (ϕ), and the amplitude of the parametric excitation (ε). The analyses are performed through numerical simulations of a flexible rotor with three critical speeds supported on two AMBs.

BALANCING PROCEDURE

The balancing procedure proposed by Mendes and Dohnal (2018) is very similar to the influence coefficients balancing procedure. In order to illustrate the correlation between both methods, the Campbell Diagram of the rotor used in the simulations (described in the next section) is presented in Fig. 1.

As mentioned before, the traditional balancing consists of several runs, each one with a test mass placed in a different balancing plane. In each run, the rotor response is measured as close as possible of the critical speeds of the modes to be balanced. That means that, in the traditional balancing, the rotor operates over the 1x line (black dashed line in the Campbell Diagram – Fig. 1) and the measurements are taken at the red dots (critical speeds); i.e. the rotor is accelerated up to each critical speed. The measurements acquired are used to calculate the influence coefficients, which are used to obtain the correction masses.

In the proposed balancing method, the rotor is accelerated at the first critical speed only (red dashed line at 65.93 rad/s). Once this rotational speed is reached and the response of the first critical speed is measured, parametric excitation at the combination resonance frequencies is used to induce a kinetic energy transfer to the higher modes (blue dots); to one mode at a time. Thus, the higher modes responses can be obtained without the need to accelerate the rotor further than the first critical speed. After the measurements are obtained, the processes of calculating influence coefficients and correction masses are the same used in the traditional balancing method. A limitation of the proposed method, in its present form, is that the relations between the responses of the system obtained via parametric excitation (blue dots) and by accelerating the rotor up to the correspondent critical speed (red dots) must be known. The implication of such limitation is that the method is only worth if more the one similar rotor must be balanced, than the same relations obtained for the first rotor can be used by the following ones. A detailed description of the method and its limitations are found in the work of Mendes and Dohnal (2018).

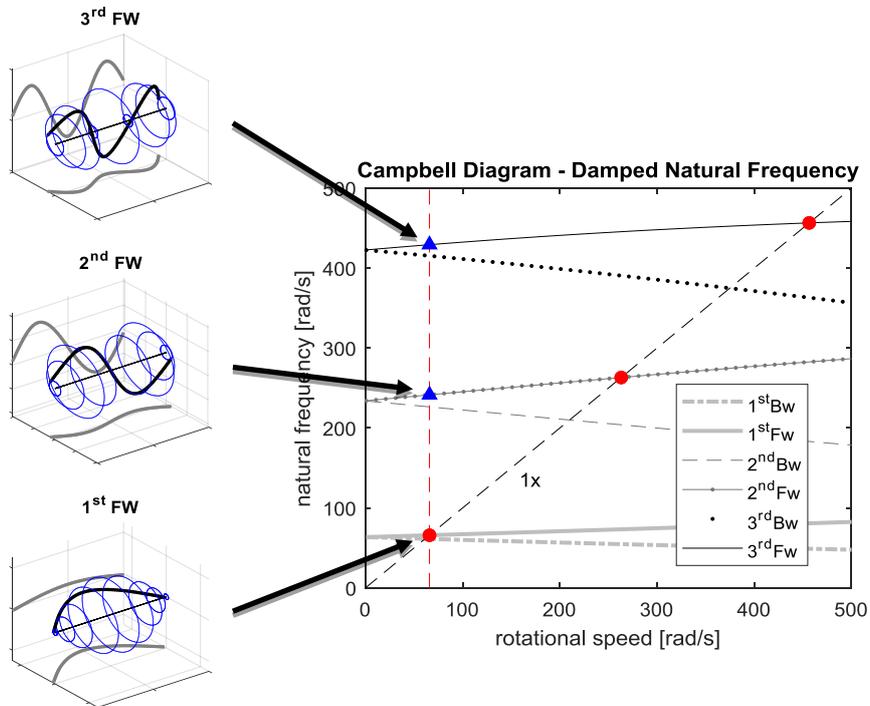


Figure 1 – Campbell diagram and mode shapes of the simulated rotor.

ROTOR MODEL

The sensitivity analysis to the phase difference between the parametric excitation in the bearings was performed using the rotor model described in the work of Mendes and Dohnal (2018). The finite element model of the steel shaft (400 mm length and 3.2 mm diameter) is composed by nine nodes (black numbers) and eight elements (red numbers). The beam elements used in the model consider the Timoshenko beam theory. The five discs have an external diameter of 63 mm and length of 24 mm. The unbalance distribution considered is presented in Tab. 1.

Two active magnetic bearings are placed at both shaft ends (nodes 1 and 9). The bearings have a nominal gap of 0.5 mm, a constructive constant of $4 \cdot 10^{-6} \text{ N.m}^2/\text{A}^2$ and work with a bias current of 0.04 A. A linearized magnetic force was considered in the simulations (Dohnal, 2012). Each axis of the AMBs is governed by a PID controller as presented in Eq. (2); where $K_p = 1760$, $K_i = 20000$, $K_d = 36.5$ and $T_f = 0.002$.

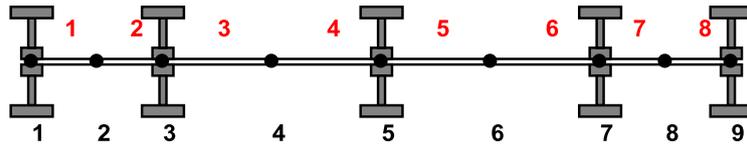


Figure 2 – Finite element model of the rotor simulated.

$$C(s) = K_p + \frac{K_i}{s} + \frac{K_d s}{T_f s + 1} \quad (2)$$

From the Campbell diagram of the rotor model, presented in Fig. 1, the first critical speed of the rotor is 65.93 rad/s, and the damped frequencies of the second and third modes at the first critical speed (blue dots) are 241.45 rad/s and 429.23 rad/s, respectively. Therefore, the combination resonance frequencies to transfer energy from the first mode to the higher modes, according to Eq. (1), are $v^{12,1} = 175.52$ rad/s and $v^{13,1} = 363.30$ rad/s. Since the unbalance is a forward excitation and the system under analysis is isotropic, it is reasonable to consider only forward modes in the balancing procedure.

Table 1 – Unbalance distribution considered in the simulations.

Disc	Node	Mass	Eccentricity	Angular Position
1	1	0.1 g	25 mm	0°
2	3	0.2 g	25 mm	230°
3	5	0.2 g	25 mm	120°
4	7	0.1 g	25 mm	180°
5	9	0.3 g	25 mm	270°

RESULTS AND DISCUSSION

The sensitivity analysis performed considered two parameters. First, the amplitude of the parametric excitation ε was evaluated from 0.1 to 0.9. It is important to recall that the parametric excitation was applied in both bearings with the same amplitude in every case simulated ($\varepsilon = \varepsilon_1 = \varepsilon_2$). The next parameter evaluated was the phase difference between the parametric excitation applied in both bearings ($\phi = \phi_1 - \phi_2$) for the range of 0° to 360°.

In the simulations performed, the system was rotating at the first critical speed (65.93 rad/s) and kinetic energy was transferred to the higher modes (blue dots at the Campbell Diagram – Fig. 1), at 241.45 rad/s and 429.23 rad/s, by inducing a parametric excitation simultaneously in both bearings. The results obtained when energy is transferred to the second mode are presented in Figs. 3a and 3b, and the results when energy is transferred to the third mode are presented in Figs. 3c and 3d.

From the results obtained for both modes (Figs. 3a and 3c), it can be concluded that, for a given phase difference between the bearings, the response amplitude of the mode receiving energy increases linearly with the increase of the amplitude of the parametric excitation. However, the increase in the amplitude changes as the phase difference between the bearings changes. In the specific case when the AMBs are used to induce the parametric excitation, it is important to analyze if the resulting current due to the amplitude used (ε) does not lead the bearing to a magnetic saturation; which could drive the system unstable.

The optimal value for the phase difference can be found in Figs. 3b and 3d, which present a side view of the same results. The second mode presents higher amplitude for a phase difference of 315°, and the third mode presents higher amplitude for a phase difference of 150°.

The real importance of properly tuning the phase difference between the bearings for each mode individually is clear when the parametric excitation amplitude is fixed, and the phase difference is analyzed. For the second mode, the increase in the displacement ranges from 2 (for $\varepsilon = 0.1$) to 9 (for $\varepsilon = 0.9$) when comparing the best (315°) and worse (135°) phase differences. Similarly, for the third mode, the increase in displacement ranges from 2 (for $\varepsilon = 0.1$) to 7 (for $\varepsilon = 0.9$) when comparing the best (150°) and worse (330°) phase differences.

Therefore, when the method is applied to real machines, the parametric excitation amplitude should be chosen very carefully to avoid driving the system unstable. Likewise, the phase difference between the bearings should be properly tuned to maximize the higher mode response, increasing the signal-to-noise ratio of the system response and ensuring a high-quality measurement.

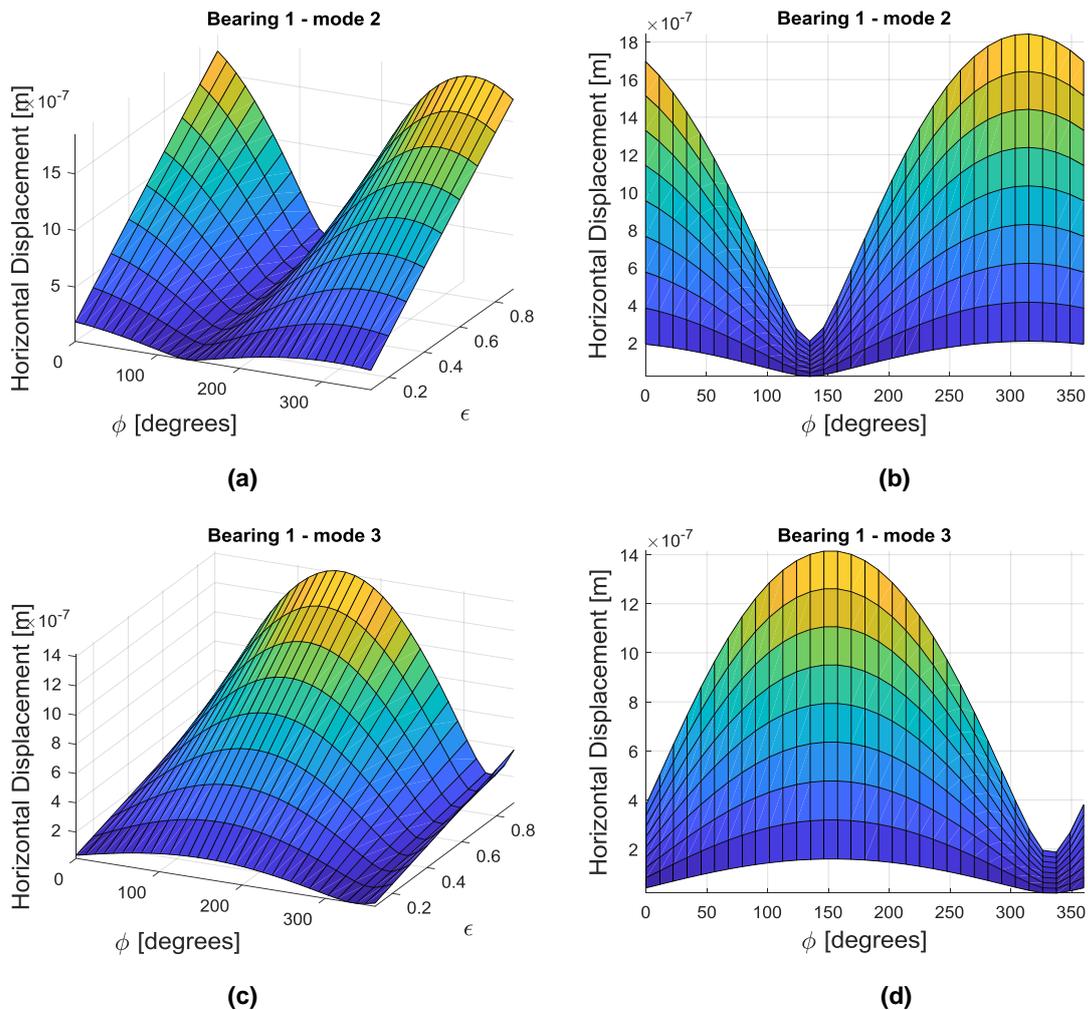


Figure 3 – Horizontal displacement at the AMB at node 1 in the presence of parametric excitation as a function of ϵ and ϕ : (a) and (b) second mode; (c) and (d) third mode.

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