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THE INFLUENCE OF A FLY-WHEEL IN A ROTOR-BEARING SYSTEM ON ITS CRITICAL SPEEDS

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ABSTRACT

The paper highlights the influence of the presence of a fly-wheel on a rotor-bearing system in the removal of the critical speed in a rotational speed interval. The stiffness of the bearings and the damping introduced by them, the gyroscopic effect on the system elements are taken into account. The quantification of the dynamic influence of the fly-wheel is carried out with the help of Campbell diagrams in the finite element analysis program Ansys Workbench, taking into account gyroscopic effects.

KEYWORDS: gyroscopic effect, rotor-bearing system, Campbell diagram

1. INTRODUCTION

The dynamics of the rotors and the stability of the rotating machines plays an important role in improving the security of these systems [2].

Two phenomena in rotor dynamics are dangerous and can lead to unacceptable vibration levels. These are critical rotational speeds and linearly unstable regimes, sometimes having the consequences of catastrophes [2].

In the literature, there is an important concern about the study of rotor-bearing systems with high rotational speeds, taking into account the stiffness of the bearings and the damping introduced by them [1]-[8].

Programs have been developed for the analysis of the influence of the gyroscopic effect on the rotors, in the Matlab environment [2] or using FEM [9].

The stable operation of a high-speed rotating rotor-bearing system is dependent on the internal damping of its materials. A finite element calculation model of a rotor-shaft system based on a 3D high-order element (Solid186 in Ansys Workbench) is introduced in reference [9] to study the turbocharger rotor-bearing system in a temperature field.

The gyroscopic effects arise from the presence of rotating masses mounted on the rotor and their orientation with respect to the bearing center line. This rotor, along with the bearing support conditions continuously causes forward and backward whirl phenomenon in the rotor and induces fatigue loading in the bearings.

The phenomenon of vibration is a common effect for all bodies that have a mass and elasticity [1], [4].

Unlike the general structural systems, the rotor does not have an actual vibrational motion, but a precessional motion around the undeformed axis of the rotor. When the bearings are isotropic, at a certain rotational speed the deformed rotor remains unchanged during motion, and the centers of gravity of the cross sections describe circular precession orbits in space. The motion appears as a vibration when the projection of the rotor's displacement is measured in a certain fixed direction in space [8].

Rotor dynamics is different from structural vibration analysis because it includes operation at high speeds, gyroscopic moments, Coriolis forces, critical speeds and the establishment of stability/instability domains [1], [3].

The gyroscopic effects on critical speeds of a rotor system supported in bearing can be studied by means of a Campbell diagram. In this paper, gyroscopic effect of a rotor supported on bearings is studied by means of Campbell diagrams using Ansys program. The numerical study is carried out by modelling of the rotor-bearing system using finite element tacking into account mass, stiffness, damping and gyroscopic matrices [10]. The solution is obtained by solving the assembled equations of motion (1):

$$M\ddot{q} + \left(C + \Omega G\right)\dot{q} + Kq = 0 \qquad (1)$$

Where q is the generalized coordinate matrix, M is the mass matrix, G is the gyroscopic skew-symmetric matrix, K is the symmetric stiffness matrix, C is the damping matrix, Ω is the spin speed.

The natural frequencies of a rotating shaft are the function of angular velocities and they are influenced by gyroscopic effects of rotor system and by bearing stiffness and damping.

For the forward modes, their natural frequencies increase with increasing spin speed, while for the backward modes, their natural frequencies decrease

Critical speeds, which occur when the rotor spin-speed matches with its natural frequencies, can be identified using the Campbell diagram.

For a rotor supported by orthotropic bearings, the forward natural frequencies increase with the rotor speed, while the backward frequencies decrease with rotor speed [2], as can be seen in Figure 1.

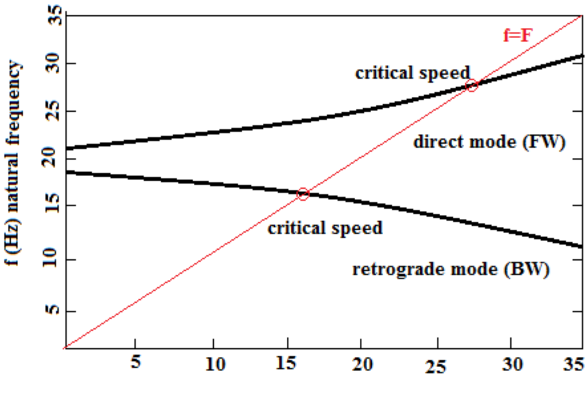

Fig. 1. *Example of a Campbell diagram for a rotor with orthotropic supports [2]*

The precession of the rotors is described by direct "forward" modes, FW, and indirect "backward" precession modes, BW. The bearing anisotropy produces pairs formed between a direct and an indirect mode. The gyroscopic effects lead to the distancing the natural frequencies forward-FW, backward-BW in a pair, making it more flexible on the BW mode and stiffening it on the FW mode [2].

Although from the analytical approach there is an analogy between the precession movement and the vibratory movement, the practical influences are different. At a critical speed, if the deformations are not limited, a rotor bends permanently rather than breaking due to fatigue, a phenomenon encountered in shafts that perform transverse vibrations [6].

The results of the modal analysis in the case of the rotors show damped and undamped frequencies. The real component of a complex frequency gives us the measure of stability, it is the exponent from the expression of the damped free vibration; a negative number indicates that the mode is stable [6].

A critical speed (critical angular speed) of order k is the speed at which a multiple of the rotor's angular speed coincides with one of its own natural frequencies.

Some important things about Critical speeds are:

- They occur at well-defined values of the rotation frequency;
- The amplitude grows linearly in time if no damping is present. In general, in practice, the critical speed can be maintained within reasonable limits [2].

2. THE ROTOR-BEARINGS SYSTEM ANALYZED

There are situations in practice where the rotors do not rotate at very high speeds, but they have wheels and fly-wheels; the gyroscopic effect is important in these systems and influences the critical speeds.

The analysis of the rotor was carried out in the Modal module of the Ansys Workbench program, which allows to obtain the Campbell diagrams with critical speeds of the first and second order and the establishment of the domains of instability for a selected range of rotational speeds. The natural frequencies and the natural modes of vibration for the rotors in this type of analysis are dependent on the rotor speed.

The scheme of the analyzed system is represented in Figure 2, and its parameters are given in Table1.

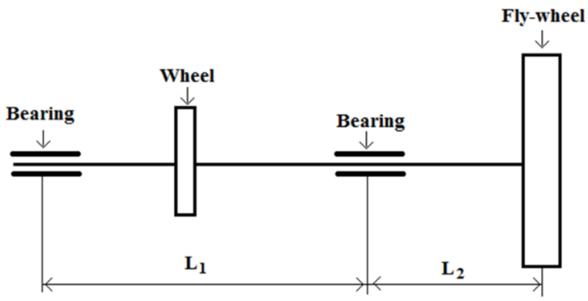

Fig. 2. *Scheme of the analyzed system*

Table 1. *Parameters of the rotor system*

System parameters	
L1	L1= 0.560 m
L2	L2= 0.100 m
Shaft radius	r = 0.010 m
Shaft density	ρ = 7810 kg/m^3
Shaft Elastic modulus	E = 2x10^{11} N/m^2
Wheel mass	mw = 2.7251 kg
Fly wheel mass	M_{fw}=4.883 kg
Wheel Polar mass moment of inertia	7.7217x10^{-3} kg m^2
Fly wheel Polar mass moment of inertia	2.441x10^{-2} kg m^2

The rotating shaft is supported by bearings and assumed to be as damped support. The stiffness and the damping effects of the bearing supports are simulated by springs and viscous dampers (kyy = 10000 N/mm, kxx = 10000N/mm, dyy = 10 Ns/mm and dxx = 10Ns/mm) in the two transverse directions.

The stiffness and the damping properties of the bearing were introduced by using Ground to Solid connection in Ansys Workbench (Modal), Figure 3.

The finite element discretization (mesh) is represented in Figure 4 (for the system with fly-wheel - 8484 nodes and 1549 elements).

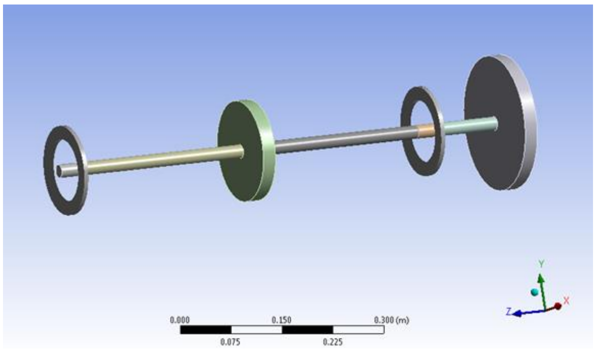

Fig. 3. *System with bearings described using Ground to Solid connection*

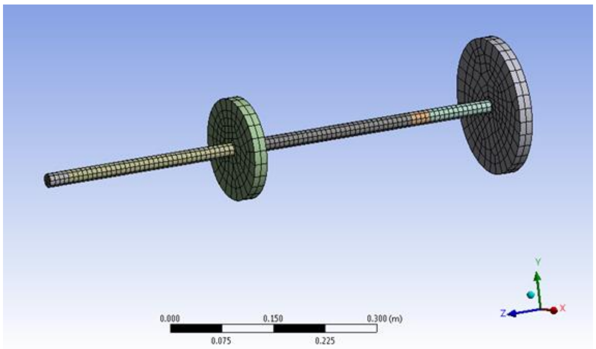

Fig. 4. *System mesh - 8484 nodes and 1549 elements*

The Modal module of the Ansys Workbench program was used, which gives the possibility of obtaining Campbell diagrams for a range of shaft speed variation.

The type of element used in discretization is SOLID 186 with 20 nodes and 3 degrees of freedom per node (the translations along the three axes x, y and z), Figure 5.

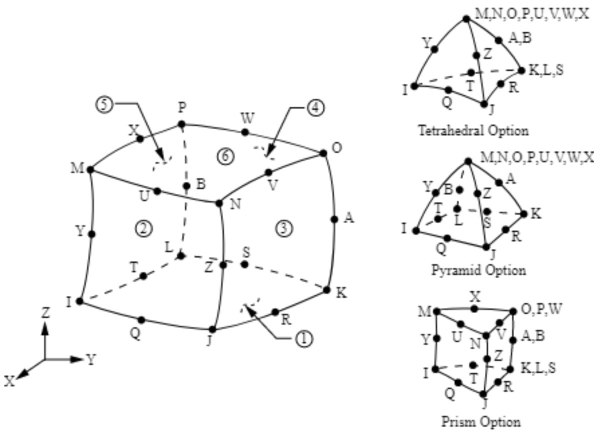

Fig. 5. *The element SOLID 186 [10]*

Two cases were analyzed in the range 0-1500 rpm:

- The system with the fly-wheel on the shaft.
- The system without the flywheel on the shaft.

The bearings were considered with isotropic behavior in both analyzed cases; instability modes are not present in the range 0-1500rpm. Ten natural frequencies and modes were taken into account.

The presence of the fly-wheel on the shaft influences its natural vibration (frequencies and modes) and removes the occurrence of a critical speed in the range 0-1500 rpm, Figure 6 and Table 2.

A forward and a backward modes are given in the APPENDIX for the system with Fly-wheel.

The absence of the flywheel on the shaft influences the natural frequencies and modes of vibration and leads to the appearance of a critical speed in the range 0-1500 rpm at 1434.8 rpm, Figure 7 and Table 3.

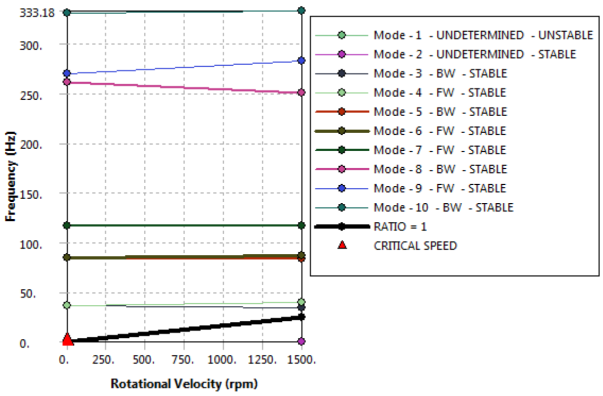

Fig. 6. *Campbell diagram of the system with Fly-wheel*

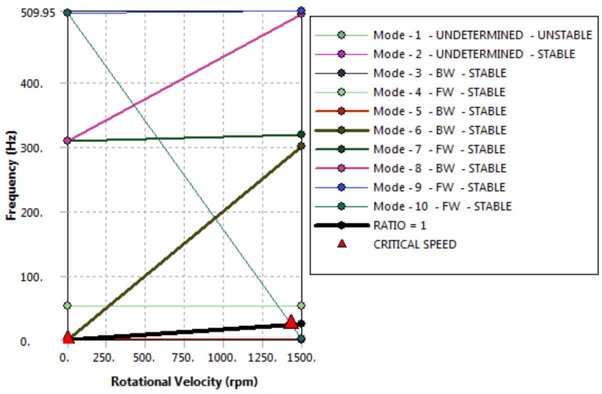

Fig. 7. *Campbell diagram of the system without Fly-wheel*

Table 2. *Mode stability and critical speeds of the system with Fly-wheel*

Mode	Whirl Direction	Mode Stability	Critical Speed	0. rpm	1500. rpm
1.	UNDETERMINED	UNSTABLE	NONE	0. Hz	0. Hz
2.	UNDETERMINED	STABLE	0.22956 rpm	3.826e-003 Hz	3.826e-003 Hz
3.	BW	STABLE	NONE	36.852 Hz	34.475 Hz
4.	FW	STABLE	NONE	36.862 Hz	39.204 Hz
5.	BW	STABLE	NONE	84.989 Hz	83.402 Hz
6.	FW	STABLE	NONE	85.05 Hz	86.727 Hz
7.	FW	STABLE	NONE	116.56 Hz	116.56 Hz
8.	BW	STABLE	NONE	261.9 Hz	250.18 Hz
9.	FW	STABLE	NONE	270.21 Hz	282.86 Hz
10.	BW	STABLE	NONE	331.29 Hz	333.18 Hz

Table 3. *Mode stability and critical speeds of the system without Fly-wheel*

Mode	Whirl Direction	Mode Stability	Critical Speed	0. rpm	1500. rpm
1.	UNDETERMINED	UNSTABLE	NONE	0. Hz	0. Hz
2.	UNDETERMINED	STABLE	0.19707 rpm	3.2845e-003 Hz	3.2845e-003 Hz
3.	BW	STABLE	NONE	52.664 Hz	52.663 Hz
4.	FW	STABLE	NONE	52.68 Hz	52.681 Hz
5.	BW	STABLE	NONE	0. Hz	1.0707 Hz
6.	BW	STABLE	NONE	0. Hz	300.25 Hz
7.	FW	STABLE	NONE	308.81 Hz	317.53 Hz
8.	BW	STABLE	NONE	308.89 Hz	504.93 Hz
9.	FW	STABLE	NONE	507.19 Hz	509.95 Hz
10.	FW	STABLE	1434.8 rpm	507.44 Hz	1.9491

3. CONCLUSIONS

The present paper highlights the using of a fly-wheel in the rotor-bearing systems, with bearings having isotropic behavior, in order to remove the danger of the presence of a critical speed in a determinated range of rotational speeds.

The work practically highlights the name of the fly-wheel as a motion uniformity element.

So, the presence of a fly-wheel can “control” the critical speeds of a rotor-bearing in a certain range of rotational speeds.

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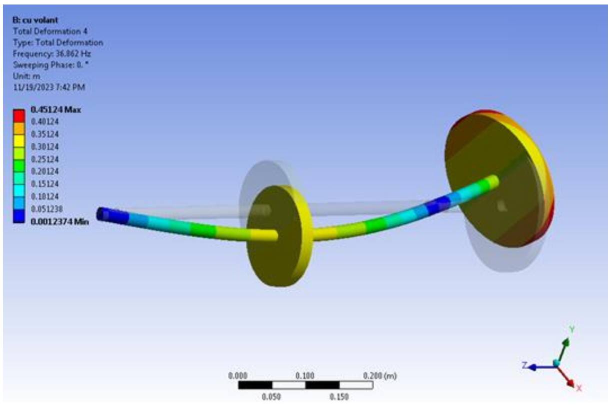
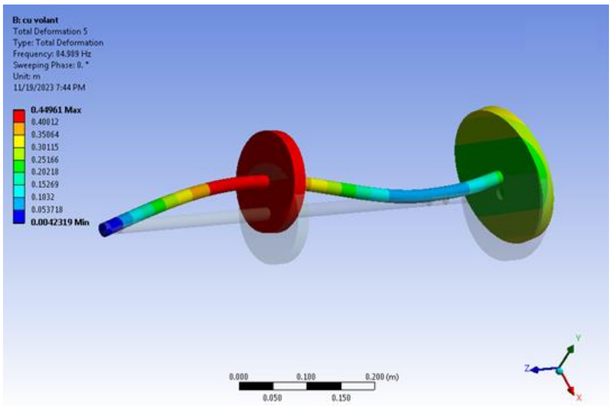
APPENDIX



A Forward (FW) natural mode



A Backward (BW) natural mode