719. Diagnostic research of rotor systems with variable inertia moment

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Abstract. The main reason of a breakdown for bearings, operating in rotor systems with dynamic loads, is rotor unbalance causing heavy loads especially during the rotational resonance speeds at run ups and coast downs when operation speed of a rotor system exceeds the resonance speed. Dynamic loads of rotor systems with rotor variable inertia moment affecting bearings, when the rotor‘s rotation speed is constant as well as during the run up and coast down periods, are significantly lower in comparison to instant inertia moment rotors. Experimental research of a flexible rotor with variable inertia moment performed by measuring rotor’s vibration displacements with proximity probes at large rotation speed interval exceeding resonance. The analysis of results allowed determining the optimal acceleration-deceleration modes and variable dynamic stiffness of rotating system. Experimental research has been carried out with an original rotor system testing facility and a high speed multi-channel vibration signal analyzer unit.

Keywords: variable inertia moment rotor, vibration displacements, resonance, kinetic orbits.

Introduction

As practice of experimental research with rotor systems developed, the main cause of a breakdown for rotor systems with variable inertia moment and their bearings is related to extremely large unbalance of rotors, causing heavy loads during the resonance speed [2, 10]. Kinetic energy of a rotor, rotating during the resonance, is transformed into mechanical energy of transverse vibrations, emerging in extremely large transverse vibration displacement amplitude of a rotor [1, 4, 5]. At the beginning of an operation, unbalance of technological machines’ rotors is acceptable (in compliance with ISO 1940) [3], however, unbalance varies due to technological cause (machines of chemical industry operate in an aggressive environment and etc.). Negative effect of an unbalance is evident during the machine starting and stopping procedures, when a technological nominal rotation speed of a rotor system exceeds the resonance speed. Analysis of theoretical and experimental research allowed developing optimal starting and stopping modes of rotor systems so that vibration intensity during the resonance is the lowest. Resonance vibrations of rotor systems are related to a variable rotor’s dynamic stiffness. Experimental research has been carried out using an original rotor system with variable inertia moment testing facility and a high speed multi-channel vibration signal analyzer unit. Authors of the articles [6-8] present modeling principles for flexible rotors. Received results can be applied to the modeling of flexible rotor systems.

Object of research – analysis of intensity for transverse vibration displacements of a rotor system with variable mass moment of inertia during the resonance. To obtain the objective, measurements of transverse vibrations of a flexible rotor with variable inertia moment by changing the speed of a rotor’s starting-stopping points were performed; analysis of measurement results and the most informative vibration displacement data formats were determined. In compliance with the research results, starting and stopping modes were determined, in the presence of which dynamic loads, affecting the bearings, are the lowest during the resonance speed.

The subject of research

In order to run the dynamic and diagnostic testing, an original mechatronic rotor system's with variable inertia moment testing facility was designed, as given in Fig. 1. The testing facility comprises of: EM – electric motor of direct current, the greatest rotation speed of which is 9300 rpm, 1 – flexible rotor, rotating in journal bearings, 2 – disk with chains, imitating a variable inertia moment, proximity probes (1XY and 2XY) with electronic signal transmitters, KP – keyphasor probe and etc. Facility's specifications: rotor system rotates in a journal bearings; rotation speed of a rotor is changed by the electronic control unit placing acceleration, which is the same at the time of run up and coast down.

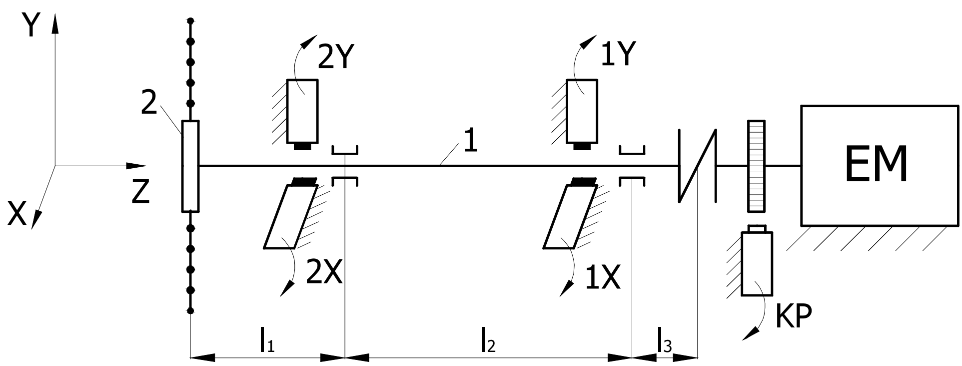


Fig. 1. Dynamic of a rotor system's with variable inertia moment testing facility

Multi-channel vibration signal analyzer unit OROS Mobi-Pack OR-36 and computer software OROS NVGate V7.00 and OROS ORBIGate V3.00 were used for recording and analyzing the signal. Other items, such as proximity probes (model PR6423/005-141-BNC, sensitivity 8000 V/m), optical tachometer ORAC-TACO02, for measurement of a rotation speed, and electronic signal converters were used for the analysis of vibration signals. Analysis of measurement results was carried out with the choice of the most informative data formats of absolute vibration displacement signals: trend during run up and coast down, vibration displacements versus order spectrum, Bode diagrams.

Methodology of experimental research

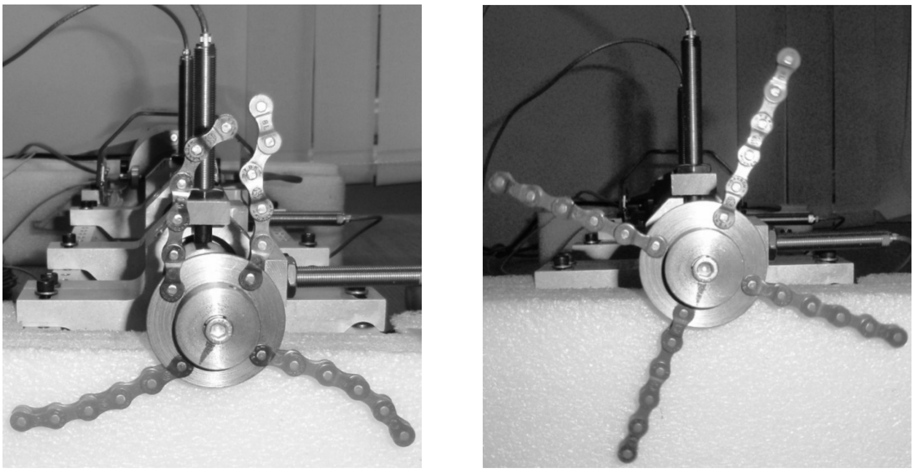
In order to measure resonance speed and intensity of vibrations of a rotor system during the resonance speed, experimental measurement of vibrations was carried out by speeding the rotor system up to a maximum rotation speed, which is 9300 rpm, changing the run up-coast down acceleration. The highest resonance speed determined to 3840 rpm, second – exceeds 8900 rpm. For a further performance of rotor dynamics a rotation speed interval was chosen as follows: from 250 rpm to 5000 rpm, where the first resonance speed is 3840 rpm, as this is typical to technological machines. Rotation speed of a rotor were changed discreetly during the rotor run up and coast down – angular acceleration varied as follows: 200, 400, 600, 800, 1000, 1200, 1400 rad/s^2 i.e. run up and coast down of a rotor system was controlled by the automatic system's measurement sensor.

Test results and their analysis

As given in Fig. 2, at the start of a rotor system with disk having flexible chains, position of chains is unstable at a low rotation speed, which is 250 rpm. Such rotation of a rotor is known as having a variable inertia moment which is $(1{,}358\text{-}2{,}156)\cdot 10^{-4}$ kg m^2. Calculation of inertia moment, based on a formula below: Eq. (1).

$$I = \int r^2 dm \tag{1}$$

When a rotor reaches a rotation speed of 700 rpm, inertia moment of a disk with flexible chains becomes the highest ($2{,}156\cdot 10^{-4}$ kg m^2), as the chains partially stretch being affected by centrifugal force, as shown in Fig. 3. Having increased the rotation speed, position of chains changed due to aerodynamic air resistance force, thus making mass moment of inertia to change.



250 rpm 700 rpm

Fig. 2. Position of chains of a flexible rotor system's variable inertia moment with different rotation speed

Object of research – determine how displacement of rotor's transverse vibrations changes during the resonance with run up and coast down of the rotor system. Fig. 3 and 4 presents diagrams with data stating that shaft's displacement during the resonance speed increases with 1000, 1200, 1400 rad/s^2 at the moment of starting, while during the stopping – remains constant. This occurs due to stiffness of nonlinear system. It has been experimentally proven that with the run up and coast down (during resonance speed), the form of rotor's vibrations, given in Fig. 5, changes. This is affected by a changing inertia moment: with run up, the chains are bent towards one side, while when coast down – backwards in respect to rotating shaft position. Two cases of acceleration variation: 600 and 1200 rad/s^2 were chosen for a further rotor's dynamic research during resonance, as in the presence of the above mentioned rotor's acceleration modes, displacements of vibrations varies: with 600 rad/s^2 – vibration displacements amplitude is the smallest, while with 1200 rad/s^2 – vibration displacements amplitude is the largest.

From the vibration displacements versus order spectrum is evident that vibrations, stimulated by the rotors unbalance (1X), dominate during resonance, however displacements of 2X, 3X and 4X frequency vibrations, related to mechanical spaces in bearing units, are also seen. This is a standard exploitation case of industrial machines.

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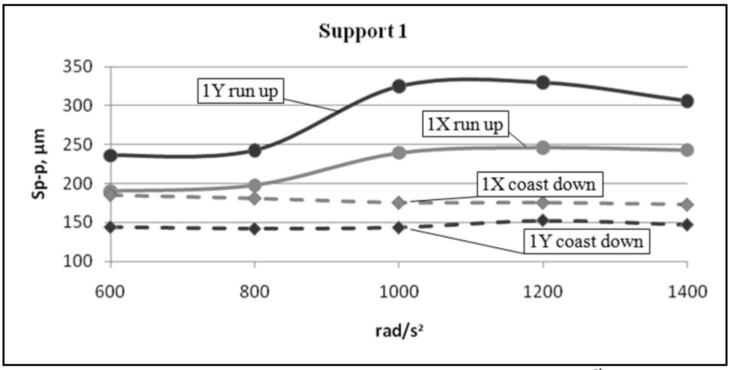

Fig. 3. Variation diagrams of rotor's vibrations displacements *Sp-p* at the 1st support received during the resonance speed: with run up (1X proximity probe and 1Y proximity probe) and coast down (1X proximity probe and 1Y proximity probe)

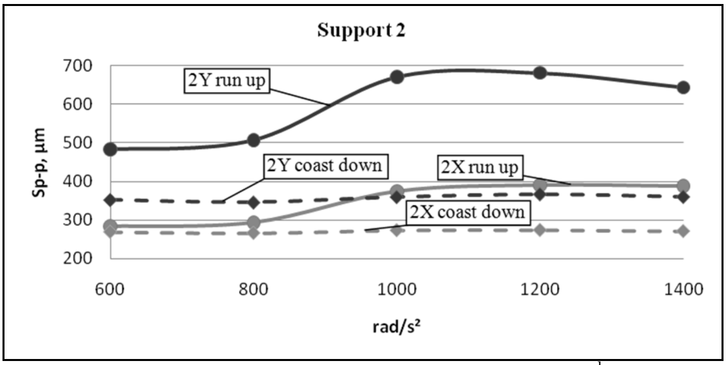

Fig. 4. Variation diagrams of rotor's vibrations displacements *Sp-p* at the 2nd support received during the resonance speed: with run up (2X proximity probe and 2Y proximity probe) and coast down (2X proximity probe and 2Y proximity probe)

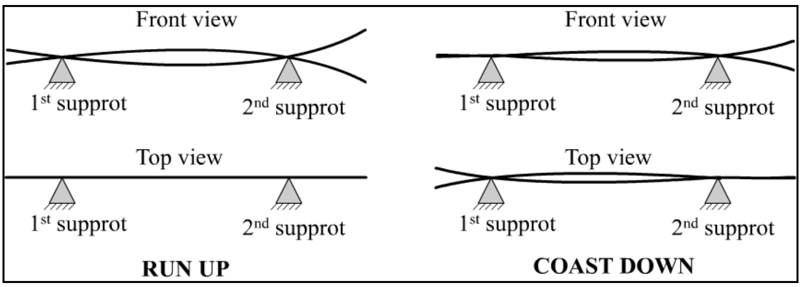

Fig. 5. Rotor's first vibration forms received during the resonance speed with run up and coast down of a rotor

Due to rotor system's nonlinear vibration amplitude-frequency record, vibration amplitudes with run up and coast down of a rotor during the resonance speed are different. Vibration displacements are always higher with running up the rotor (during resonance speed) and vice versa – with the coast down of a rotor, vibration displacements are lower [9].

Vibration trend, given in Fig. 6, presents all vibration *Sp-p* values of measurement points' as well as values for starting and stopping of a rotor. It is evident from *Sp-p* displacements that shaft's displacements most of all increase during the resonance speed with a run up. Here, 1,2,3 and 4 *Sp-p* values of vibration displacements are respectively 1 - 1X, 2 - 1Y, 3 - 2X, 4 - 2Y with run up and coast down of the machine.

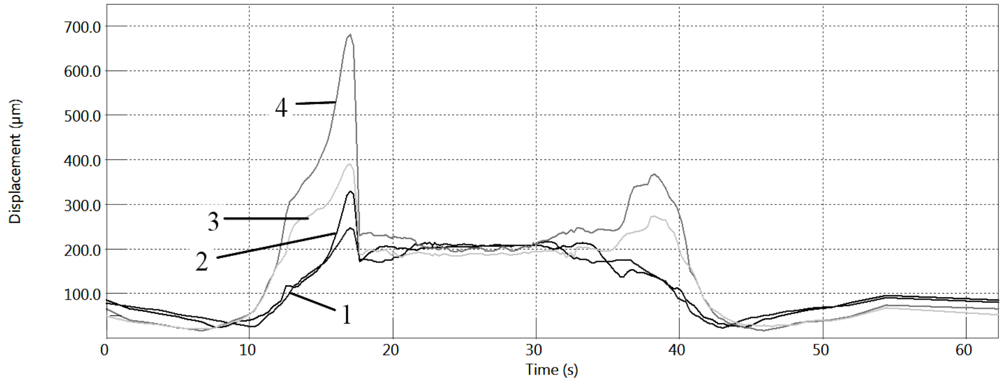

Fig. 6. Trend 1200 rad/s^2. 1 - 1X proximity probe, 2 - 1Y proximity probe, 3 - 2X proximity probe, 4 - 2Y proximity probe

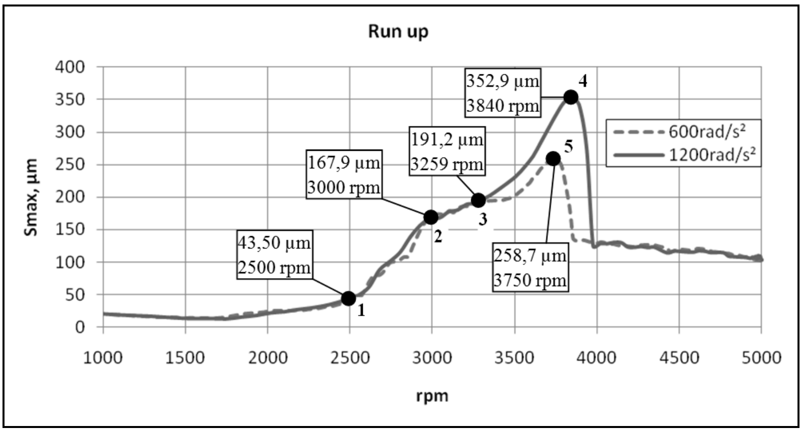

Fig. 7. *Smax* dependency on rotation speed during run up with different accelerations

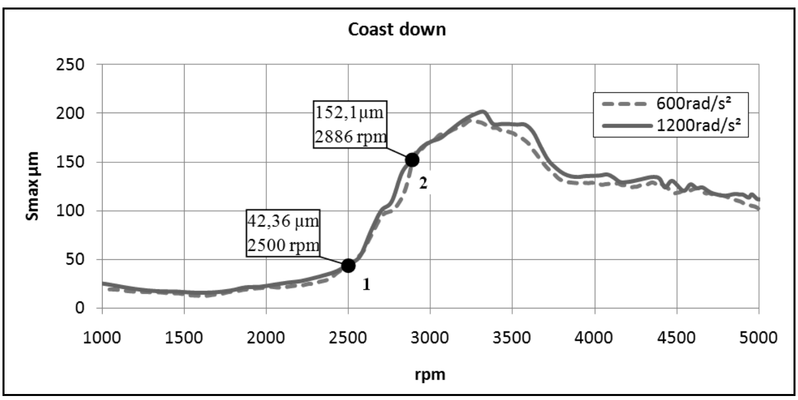

Fig. 8. *Smax* dependency on rotation speed during coast down with different accelerations

Points 1, 2, 3 and 4 marked in Fig. 7 and Fig. 8, given below, indicate an increase of vibrations intensity with run up in Fig. 7 and decrease with coast down in Fig. 8. When stopping the machine points 3, 2 and 1 being marked correspond to vibration displacements and rotation speed of a rotor. It is worth noting that acceleration of a run up has the greatest impact on dynamic stiffness of a rotor system with variable moment of inertia. The most intense increase of vibration displacements is determined between 1 and 2 point, when speed changes at 500 rpm and between 3 - 4 and 3 - 5 points, when a resonance is reached at 3840 rpm (point 4) and 3750 rpm (point 5). When diagnosing a rotor system in a stopping mode, such phenomena does not exist, as electric motor operates in a generating mode, distinguished by a high damping power.

Having a machine operating in a varying operational cycle with frequent starting, intensity of vibrations during resonance depends on acceleration of a run up. Such data formats allow determining an optimal acceleration of a run up based on vibrations intensity for technological machines of various purposes.

Conclusions

Using an original rotor system’s with variable inertia moment dynamics testing facility and vibrations measurement and analysis technique, based on information technologies, it has been determined that:

1. First resonance speeds of rotor system with variable inertia moment is at 3840 rpm and at 8900 rpm.
2. Dependence of rotor system’s vibration intensity on rotor’s run up and coast down acceleration.
3. Selected a run up mode, where intensity of vibrations is the lowest during resonance.
4. The main parameters of a dynamic model used for theoretical research were identified.

References

[1] **Barzdaitis V., Činikas G.** Monitoring and diagnostics of rotor machines. Kaunas: Technologija, 1998, 364 p.

[2] **Vekteris V., Čereška A., Jurevičius M.** Diagnostic research of adaptive hydrodynamic segmental bearings with elastic ring in a rotor system. Mechanika, No. 2(82), 2010, p. 58–62.

[3] ISO 1940-1:2003, Mechanical vibration - Balance quality requirements for rotors in a constant (rigid) state. Part 1: Specification and verification of balance tolerances.

[4] **Bently Donald E.** Fundamentals of rotating machinery diagnostics / Donald E. Bently, with Charles T. Hatch; Edited by Bob Grissom. Minden: Bently Pressurized Bearing Press, 2002, 726 p.

[5] **Muszyńska Agnieszka (Agnes)** Rotordynamics. Boca Raton: Taylor & Francis, 2005, 1075 p.

[6] **S. K. Laha, S. K. Kakoty** Non-linear dynamic analysis of a flexible rotor supported on porous oil journal bearings. Communications in Nonlinear Science and Numerical Simulation, Volume 16, Issue 3, March 2011, p. 1617-1631.

[7] **Chris A. Papadopoulos, Pantelis G. Nikolakopoulos, George D. Gounaris** Identification of clearances and stability analysis for a rotor-journal bearing system. Mechanism and Machine Theory, Vol. 43, 2008, p. 411-426.

[8] **J.-J. Sinou** Non-linear dynamics and contacts of an unbalanced flexible rotor supported on ball bearings. Mechanism and Machine Theory, Vol. 44, 2009, p. 1713–1732.

[9] **Augustaitis Vytautas Kazimieras** Basis of Mechanical Vibrations. Vilnius: Žiburys, 2000, 319 p.

[10] Methods of diagnostics of rotors with supports of variable stiffness and means for implementation of fault prevention. Research work Nr. 8404, Nr. 8527, Kaunas University of Technology – UAB „Arvi Fertis“, 2007-2011.