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Modelling of rotor dynamics caused by of degrading bearings

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1. Introduction

Many aspects of rotary systems behaviour should be considered in their exploitation. Technological machines and systems are affected by various internal and external exciters. Therefore it is vital to use proper methods of condition monitoring aiming to control and predict processes running in those machines.

Measurements of vibrations are widely used for condition monitoring and faults detection of rotary systems [1-6]. It is very useful tool for extending the lifetime of technological machines and preventing serious accidents. In most cases vibration monitoring is limited to measurements of vibrations in several points of a machine (in many applications – on bearing cases). Therefore it may be complicated to rely only on the experimental data especially for prediction of rotary systems behaviour and changes of reliability caused by gradual degradation of their elements.

Numerical modelling of rotary systems dynamics is useful not just for design, but also for prediction of their parameters (e.g. changes of dynamics caused by long term exploitation) during exploitation [1, 2, 5, 7, 8].

2. Dynamics of rotor supported by hydrodynamic bearings

There are known several methods of analytical or numerical modelling of rotary systems dynamics and other physical phenomena [1, 2, 6-16]. Applications of complex 3D models are rather computationally expensive. Therefore some authors are seeking to improve relatively simpler models of rotary systems and their elements by introducing and analysing complex matrix of nonlinear damping and stiffness. In this case modelling of real rotary systems brings very large and complex matrixes what are quite complicated to operate and analysis of those models requires lot of time. In other cases complex models of nonlinear forces of excitation, acting in rotors supports are introduced [1, 5, 6]. Both methods have their strong and weak sides.

The second method of modelling is applied in this research. It allows usage of relatively small and simple set of matrixes however generates acceptable results [1, 5]. Naturally, if complexity of such model rises (they are applied for analysis of dynamics of complex real rotary systems), special methods and techniques of modelling have to be applied [1, 5, 6, 7].

Therefore the model, including masses of rotor disk and beam elements (Fig. 1), stiffness and damping of hydrodynamic bearings, is composed by means of the FEM. Nonflexible rotor rotating on two supports (rotor of a centrifugal pump is used and a prototype for this model), characterized by stiffness and damping elements is divided into nineteen elements (supports are located on the first and the last element). It is assumed that stiffness, material and other properties are not changing in any of described elements.

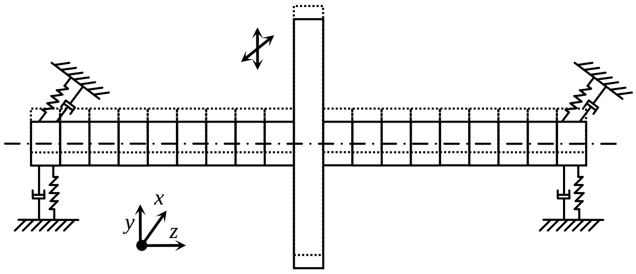


Fig. 1 General view of rotor model with finite beam and disk type elements

Dynamical model of rotor vibrations assumes that every element has four degrees of freedom (linear degrees of freedom in *x* and *y* directions (Fig. 1) and two angular in planes *yz* and *xz*). Forces of damping, caused by changes of properties of the system elements materials (internal friction) and generated in sliding bearings are also included.

General equation of the rigid rotor vibration dynamics can be given in the form

$$\left([M]+[M']\right)\{\ddot{U}\}+\Omega\,[G]\{\dot{U}\}+[C]\{\dot{U}\}+ \\ +[K]\{U\}=[P]+[P_H(A,\Omega,t)] \quad (1)$$

here $[M]$ is matrix of masses of rotor elements, $[M']$ is matrix of masses describing angular oscillation of rotor elements, $[G]$ is gyroscopic matrix, $[C]$ is matrix of damping, $[K]$ is matrix of stiffness, $\{U\}$ is array of linear movement of rotor elements, $[P]$ is matrix of forces, affecting rotor elements, $[P_H]$ is matrix of hydrodynamic forces (what are function of time, vibration amplitude and frequency of rotor rotation) and Ω is angular frequency of rotor rotation.

Hydrodynamic forces, acting in journal bearings should be calculated in order to evaluate impact of bearing on rotor vibrations [9-11, 15, 16]. Muszynska and others [1-3] provide methods for the calculation of hydrodynamic forces. It is assumed in the proposed model that hydrodynamic forces are changing together with the frequency and amplitude of vibrations of rotor elements, situated on sliding bearings.

Forces, acting elements on hydrodynamic bearings in *x* and *y* directions may be described as follows

$$\left.\begin{aligned} P_x &= -\frac{12\pi\eta l R^3 A_c}{c^3\left(1-\varepsilon^2\right)^{2/3}\left(2+\varepsilon^2\right)}\left(\left(1-\varepsilon^2\right)\Omega-\left(2+\varepsilon^2\right)\omega\right)\sin\Omega\ t \\ P_y &= -\frac{12\pi\eta l R^3 A_c \varepsilon}{c^3\left(1-\varepsilon^2\right)^{2/3}\left(2+\varepsilon^2\right)}\left(\left(1-\varepsilon^2\right)\Omega-\left(2+\varepsilon^2\right)\omega+3\varepsilon^3\left(\frac{\varepsilon^2\Omega}{2+\varepsilon^2}+\omega\right)\right)\cos\Omega\ t \end{aligned}\right\} \tag{2}$$

here l is length of the bearing, η is dynamic viscosity of lubricant, R is radius of rotor journal, A_c is characteristic gap of the bearing (in the model it is equal to amplitude of vibrations, which is function of time), ε is eccentricity ration ($\varepsilon = e/c$, $\varepsilon \to 1$ for modelled case), ω is angular velocity of rotor orbiting and c is radial gap of the bearing.

It is assumed what cavitation in bearings does not occur. Angular velocity of the rotor rotation Ω is considered to be a function of time and angular velocity of orbiting ω is as function of Ω, maintaining the condition of oil film stability [1].

Condition of the stable oil film [1]

$$\Omega < \frac{\left(4-\varepsilon^2\right)\left(2+\varepsilon^2\right)}{\varepsilon^4-2\varepsilon^2+4}\omega \tag{3}$$

Damping of vibrations is calculated applying coefficients of external and internal (structural) damping. In this case forces of external friction are

$$P_{ex}(u) = c_{ex}\dot{u} \tag{4}$$

here $P_{ex}(u)$ is external resistance force, c_{ex} is coefficient of damping and $\dot{u}$ is velocity of vibrations in certain direction (x or y).

Damping of vibrations due to internal friction in the material of rotor is described by internal coefficient of damping which is a function of vibrations amplitude. It has been assumed that structural damping of vibration is quite low and does not depend on frequency of vibrations. Therefore average coefficient of damping characterising quantity of vibration energy, dissipated in the rotor, may be calculated

$$c_{in,av} = \frac{\delta}{\pi}\sqrt{mk} \tag{5}$$

here δ is logarithmic decrement, m is mass of the rotor and k is stiffness of the rotor. δ is a function of material properties, internal tension and therefore is the function of vibration amplitude [14]

$$\delta = \frac{2^{\eta_m+1}\upsilon_m(\eta_m-1)A^{\eta_m-1}}{\eta_m(\eta_m+1)} \tag{6}$$

here η_m and υ_m are coefficients describing properties of rotor material, A is amplitude of vibrations.

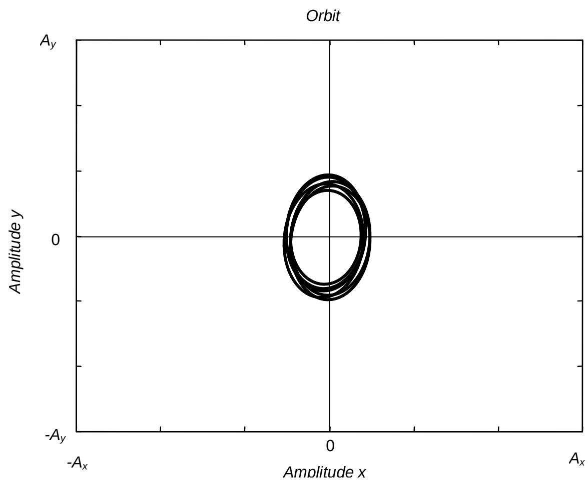
Therefore matrix of damping is [12]

$$\left[C\right] = \frac{2^{\eta_m+1}\upsilon_m\left(\eta_m-1\right)A^{\eta_m-1}}{\pi\eta_m\left(\eta_m+1\right)}\left[\Lambda\right] \tag{7}$$

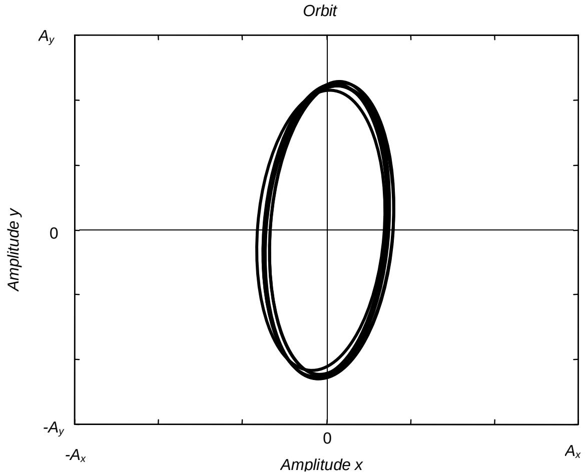
where $\left[\Lambda\right]$ is matrix of properties of rotor material. Elements of this matrix are calculated according to the formula (6).

3. Results of modelling

Analysis of the presented model involved typical cases of hydrodynamic bearings degradation. Gaps of the both bearings were analysed corresponding to the sizes typical for new bearings and to the maximal allowed (worn), as well as the number of transitional cases. The numerical analysis was carried out applying the Runge-Kutta method.



a

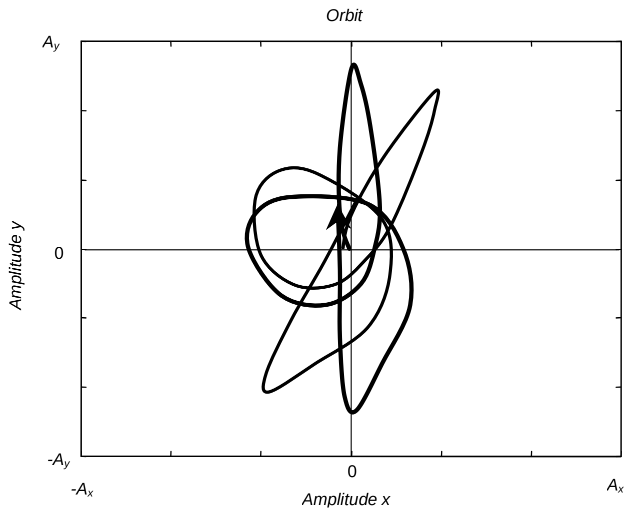


b

Fig. 2 Selected dynamic orbits (in the plane *xy*) of bearing journal centre showing impact of hydrodynamic forces: a – gap in the bearing is typical for common exploitation range, b – gap in the bearing is maximal allowed for machine exploitation

The dynamic orbits are modelled in order to obtain information concerning periodic or nonperiodic behaviour of rotary system. Results of modelling (Fig. 2) show increasing orbits amplitudes in condition of gradual degradation of bearing surfaces. It can be stated what those results (shapes of journal orbiting and development of amplitudes of journal displacements) correspond to the data of experimental measurements [1-3].

Fig. 3 presents complex dynamic orbits of the rotor journal. Shape of the orbit plot is partly caused by some limitations of the model and the objective to shorter computing time.



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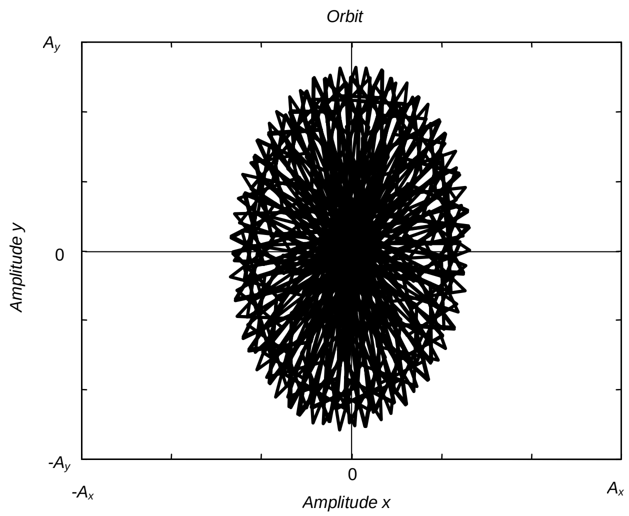

b

Fig. 3 Dynamic orbits of degrading system (gaps in bearings are maximal allowed for machine exploitation): a – damaged bearing orbits, b – complex orbit

The spectrum of vibrations was analysed by using the fast Fourier transformation. Situations representing gradual increasing of the bearing gap (degradation of bearing surface) were modelled. Changes of spectrum of vibrations in the vertical direction (Fig. 4) show typical behaviour of degrading rotary system [1]. Spectrum components corresponding to frequencies of rotor rotation and orbiting (in this case $\omega \approx \Omega/3$) can be noticed. Spectrum components representing combinations of rotary and orbiting frequencies start appearing while gaps of bearings are increasing and the influence of hydrodynamic forces Eq. (2) is also increasing.

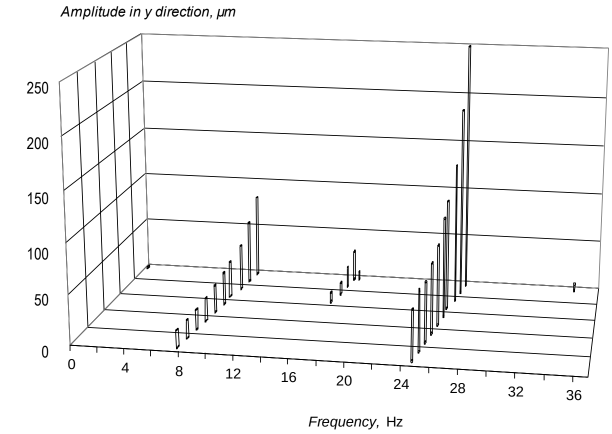

Fig. 4 Modelled spectrum of vibrations in the vertical direction showing the case of increasing gaps of hydrodynamic bearings

Fig. 5 presents amplitude – frequency characteristics of the rotor journal vibrations. Curves *1-3* represent the increase of bearings gaps (the smallest gap – curve *1* and the largest – curve *3*).

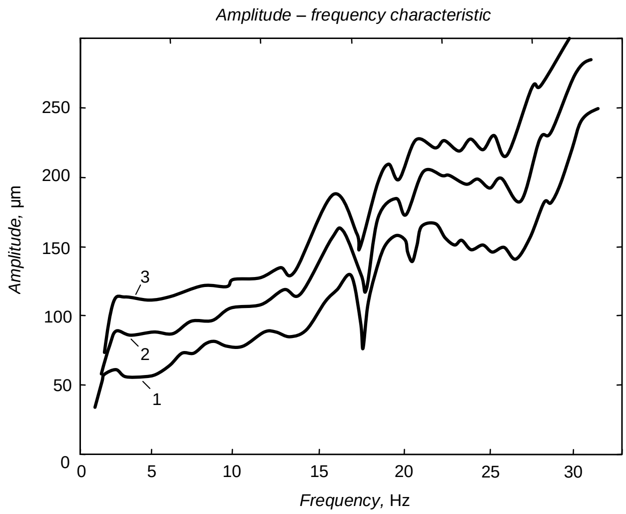

Fig. 5 Amplitude – frequency characteristics in case of increasing gaps of hydrodynamic bearings

4. Conclusions

Dynamics of a rotary system with rigid rotor is modelled simulating gradual degradation of hydrodynamic bearings. The following conclusions may be presented from the modelling results.

1. Regardless to the limited precision and other limitations of the model, results of modelling involve calculations of nonlinear hydrodynamic forces and corresponds to typical ones obtained from experimentally studied cases [1-3] quite well.

2. Presented model, updated with experimentally measured parameters of real machines, can be used for modelling of rotary system vibrations caused by complicated processes of degradation, typical for technological machines. Therefore it is useful for prediction of machines reliability.

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GUOLIŲ DILIMO ĮTAKOTAS ROTORIAUS DINAMIKOS MODELIAVIMAS

R e z i u m ė

Tikslus realių technologinių sistemų dinamikos modeliavimas yra sudėtingas uždavinys, reikalaujantis nemažų žmogaus ir aparatinių išteklių. Todėl, siekiant padidinti modeliavimo tikslumą nekuriant nepagrįstai sudėtingų ir sunkiai analizuojamų modelių, rotorinės sistemos modelius reikia koreguoti remiantis eksperimentinių tyrimų rezultatais.

Straipsnyje pateikti standaus rotoriaus virpesių, sukeltų hidrodinaminių guolių dilimo, modeliavimo rezultatai. Modelis sudarytas naudojant netiesinių hidrodinaminių jėgų išraiškas ir eksperimentinių tyrimų metu nustatytus rotorinės sistemos parametrus.

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MODELLING OF ROTOR DYNAMICS CAUSED BY OF DEGRADING BEARINGS

S u m m a r y

Precise modelling of dynamics of complex technological systems is complicated problem requiring significant human and computational resources. Therefore, aiming to create model of rational size and complexity, some adjustments of models should be done applying experimentally obtained parameters of rotary systems and their elements.

This paper presents results of modelling of vibrations of rigid rotor caused by the degradation of hydrodynamic bearings. Model is composed applying equations of nonlinear hydrodynamic forces and measured parameters of a real rotary machine.

Keywords: rotary system, vibration, bearing degradation.

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