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Coupled non-linear vibrations in multi-supported rotors founded on slide bearings

The paper contains a study of mutual interactions, which occur between lateral, torsional and axial vibrations in multi-supported rotors operating in the non-linear range. The object of investigations is a rotor founded on three slide bearings.

Presented have been the methods of modelling of the rotor cracks and bearings misalignment. These defects have been regarded as the factors generating the coupled forms of vibrations. In the work presented has been a part of work devoted to the influence of location of the angle of rotor crack on the system dynamical response. Applicability of phase vibration spectra analysis in diagnostics of such kind of defects has been indicated.

The paper presents the analysis of non-linear vibrations of real rotors carried out using a 3D heat transfer model of journal bearings. The method is adequate to describe the oil film properties and enables the use of the equations of motion for the entire system and thereby the description of instability behaviour of such objects.

In the work indicated are the capabilities of computer analysis in optimisation and diagnostics of technical objects such as power industry turbosystems. In the work presented is also a general discussion on the constraints emerging from modelling and mathematical description of different objects, not only of he power industry type.

1. Introductory remarks

Problems linked with the dynamics of rotating machinery are subject of investigations in several domestic and foreign research centres [1-10]. Despite abundant information on this topic, there are still numerous problems to be solved. This regards primarily to the analysis of non-linear vibrations and also coupled forms of lateral, axial and torsional vibrations. Analysis of this type of phenomena requires application of very advanced models and computer software. The problems becomes outmost difficult in the case of large power industry objects such as turbogenerators of large power. Coupled forms of vibration are generally a result of different kinds of couplings taking place in the system and interactions between construction and material imperfections. Presented here as example can

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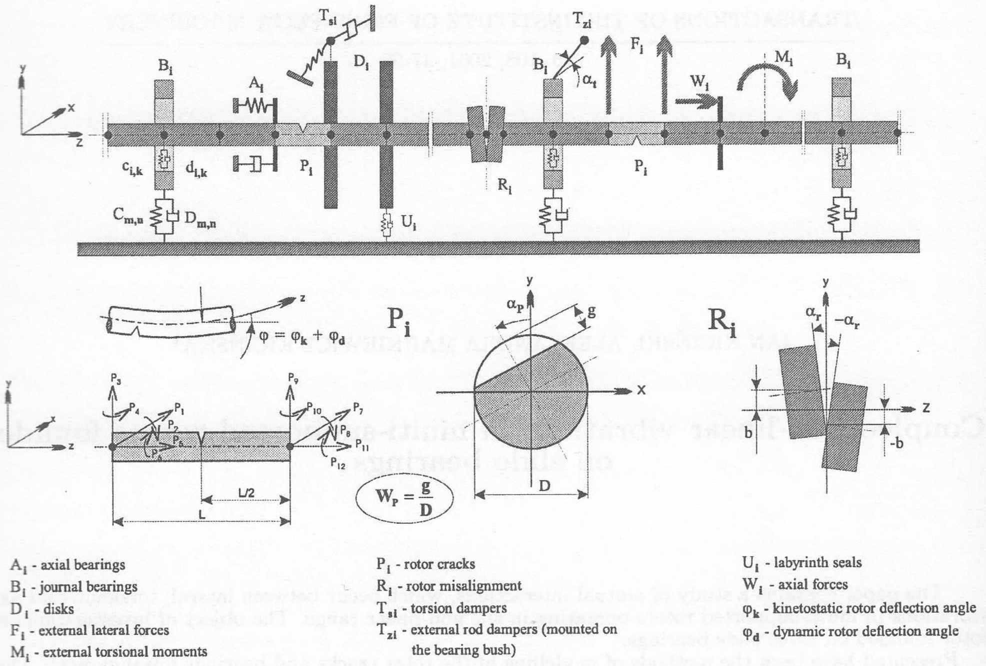

Fig. 1. Calculational capabilities of computer software from NLDW-70 suite for the analysis of rotor non-linear dynamics with imperfectionslike rotor cracks and misalignment.

be rotor cracks and misalignment of the rotor line. Modelling of such kind of imperfections and subsequently analysis of their influence on system operation represents undoubtedly one of contemporary challenges in this branch of science. Presented in this work problems provide an expressive example on the capability of proposed methods.

Obviously, the problems connected to non-linear analysis and influence of various kinds of imperfections do not explore contemporary problems of rotor dynamics by far, they are however the topic of interest of many research centres all over the world.

2. Research tools

At IFFM PASci in Gdańsk developed have been models and computer codes named NLDW – 70, which enable analysis in the non-linear range of lateral, axial and torsional vibrations in the systems of rotor-bearings type with account of imperfections such as rotor cracks and line of shafts misalignment – Fig. 1. Development of the model of the rotor-bearings-supports system using finite element method with 12 degrees of freedom and assumption of the model of cracks with account of the influence of actual shape of rotor line deflection φ and location of cracks $\pm\alpha_p$, and also model of misalignment inclusive of separated planes $\pm\alpha_r$, enables unique possibilities for the analysis of coupled lateral, axial and torsional vibrations rendered by imperfections such as cracks, misalignment or external excitations. Appearance of the cracks induce additional energy of elastic deformation, which is mathematically captured by adequately modified flexibility matrices. Assumed has been a two-position model of crack behaviour i.e. fully open or fully closed. Such model is widely used in the literature. A novelty in the assumed programme is that the process of closure and opening control is dependent both on the shape of kinetostatic line φ_k and also instantaneous „dynamic" shape of line of rotors φ_d rendered by external forces. Hence, during the shaft rotation, the opening or closure time of the crack is dependent on the actual shape of the line of rotors

$$\varphi = \varphi_k + \varphi_d.$$

Misaligned shaft sectors influence the element displacements in the direction of x axis by the distance a and in the direction of y axis by the distance b and rotation by the angle $\pm\alpha_r$ with respect to a vertical plane, Fig. 1. Making use of flexibility formulation of finite element method, the relations for additional bending and torsional moments can be derived, which exist in the system and in such way it is possible to construct adequately modified flexibility matrix of analysed element. A subsequent step is to develop a transformation matrix together with a matrix of stiffness for the finite element.

Above model enables modelling of not only misalignment alone, but also the degree of surface separation, for example in the case of couplings, by the angle $\pm\alpha_r$.

Problems of kinetostatics are solved based on a 3D model of heat transfer in bearings (so called elasto-diathermal model). NLDW – 70 code at each time step of iteration loop solves a non-linear equation of motion for the entire system and modifies at the same time the coefficients of stiffness and damping of oil film (as well as labyrinth sealings), and also modifies the matrices of stiffness reflecting the influence of cracks and misalignments. Such process is continued until adequate accuracy of calculations has been attained.

Details regarding the above models and computer codes have been presented in [5-6], [8-10]. In these works presented have also been information regarding experimental verification of developed research tools. This verification was referred both to laboratory objects and real ones (turbogenerators of 200 MW power). Developed research tools have been made reliable to the sufficient extent.

For the apparent reasons there is not any possibility for presentation of more specific details, both with respect to models itself and their verification. Non-linear analysis using such research tools as NLDW – 70 code enables advanced applications in diagnostics of industrial objects such as turbogenerators of large power. It has been made possible due to the fact that modelling of various kinds of defects in operations of such objects has become possible, and subsequent analysis of their symptoms on the basis of non-elliptic shapes of trajectories and vibration spectra. Non-elliptic and non-sinusoidal distributions of dynamical displacements

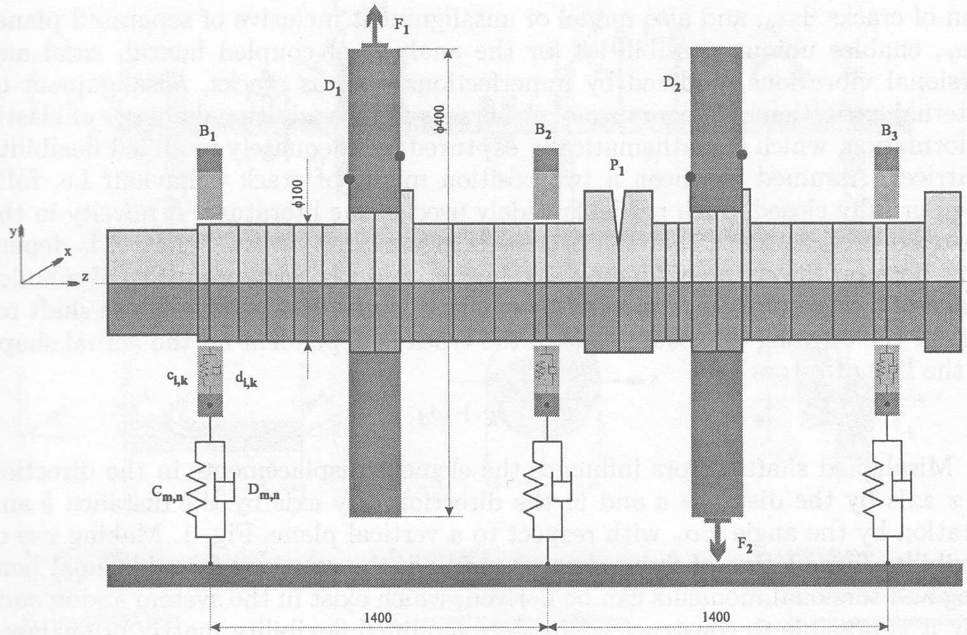

Fig. 2. Considered object – three-supported rotor with the crack P_1. Schematic of MES discretisation.

of selected nodes of the system suppress various kinds of defects, and their analysis by means of vibrational spectra enables development of catalogues of diagnostic relations.

3. Subject and aim of investigations

Let the object of the analysis be a three-supported rotor founded on slide bearings, Fig. 2. Such rotor can be found in the vibro-acoustic laboratory at IFFM PASci in Gdańsk. Based on the results of measurements conducted on this rotor, performed has been experimental verification of developed earlier theoretical models [6, 8, 9]. Let's assume now, that on our rotor, in the places of discs D_1 and D_2, there are acting forces F_1 and F_2 induced by a specified discs imbalance. Let the forces be displaced in phase by 180^0, which reinforces the system excited vibrations. Coefficients of support stiffness and damping $C_{m,n}$ and $D_{m,n}$ have been experimentally measured.

The aim of investigations was to determine the influence of inappropriate stiff coupling assembly (coupling both shafts) on the system dynamical properties. There can be imagined a situation, where due to non-uniform clamping bolts fixing there appear in the coupling a slot, which apparently changes the stiffness of the line of rotors. Let's try to model such situation using a model of a cracked

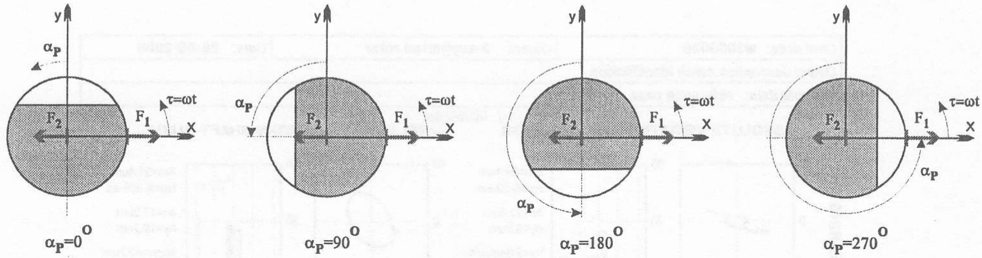

Fig. 3. Analysed cases of crack location angle α_p with respect to the plane of acting of external excitation F_1 and F_2.

rotor. The slot can be regarded as a closing and opening crack dependent upon the angle of line of shaft deflection φ and its location α_p. Fig. 1 and Fig. 2 present localisation of the coupling and a crack denoted by a symbol P_1.

There arises a question whether the angle of crack location (slot) with respect to the plane, where forces F_1 and F_2 are acting has an important influence of the system dynamical properties?. Or whether such influence comes from the depth of the crack defined by the indicator w_p (Fig. 1)?. An answer to these questions is the objective of the present work.

4. Results of calculations

Calculations using a NLDW-70 code have been conducted for different rotor rotational velocities n, different indicators of crack depth w_p and four angles of crack location α_p, as illustrated in Fig. 3. In effect obtained has been a vast material, of which selected parts will be presented below. Results will refer only to one rotational velocity $n = 2853$ rev/min (resonance velocity) and one value of crack indicator $w_p = 0.3$, which means a value of $P_1 = 30\%$. In order to enable direct comparisons, firstly conducted have been calculations for a rotor without cracks ($w_p = 0$, $P_1 = 0\%$). Results of calculations in the form of so called diagnostic card are presented in Fig. 4, and in the form of spatial trajectories of line of rotors vibrations in Fig. 5. This case will be regarded as a base case (reference). After a preliminary analysis of obtained results it turned out that from amongst four locations of cracks α_p presented in Fig. 3 a least favourable location can be recognised (largest amplitudes of vibrations) as well as most favourable one (lowest amplitudes of vibrations). These are the cases for $\alpha_p = 90^0$ and $\alpha_p = 270^0$ respectively. We will confine our attention to present only these cases. Results of calculations have been presented in Fig. 6÷9. As can be seen the influence of location of crack α_p turned out to be surprisingly large. Amplitudes of absolute vibrations of rotor and bearing bush are over 3 times greater for $\alpha_p = 90^0$ than for $\alpha_p = 270^0$. On the other hand, comparing Fig. 8 and 9 with the base case, i.e. without crack – Figs. 4 and 5 we can conclude that at the advantageous location of a crack slot even a crack as large as $P_1 = 30\%$ does

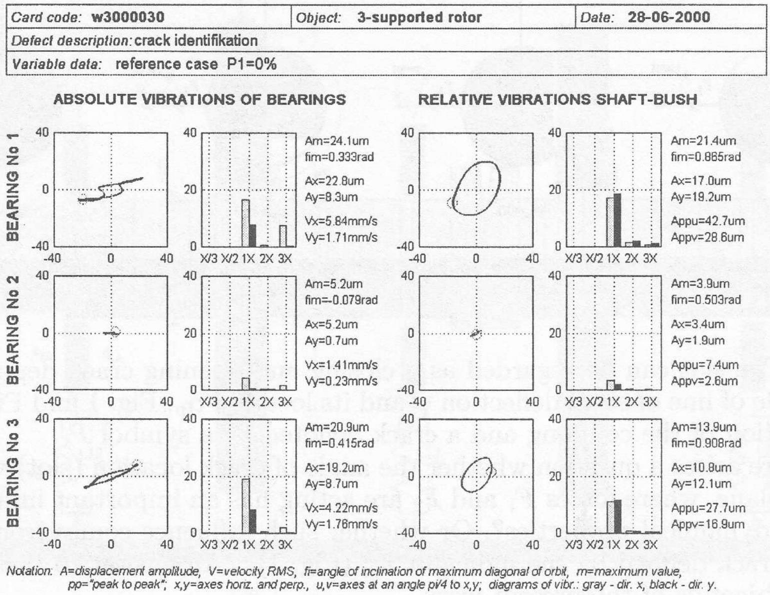

Fig. 4. Diagnostic card calculated for the case from Fig. 2 without rotor cracks – base case (reference).

not influence the extent of system vibrations. Figs. 10 and 11 present amplitude and phase spectra calculated for three cases: without a crack (upper graphs), with a crack for $= 90^0$ (middle graphs) and with a crack for $\alpha_p = 270^0$ (lower graphs). These spectra refer to the bearing bush No. 1 and only horizontal components Ax, fix. It turns out that even such large increase of system vibrations for the least favourable case ($\alpha_p = 90^0$) does not render a qualitative change in the distribution of amplitude vibration spectrum (Fig. 10). Important differences, however, can be found when analysing phase spectra (Fig. 11). This means that phase spectra are a better diagnostic determinant in the case of such defect. This conclusion can be of significant practical importance, especially in the light of the fact that amplitude spectra are regarded as traditional carriers of diagnostic information in such cases. Appearance of the crack in the system generates coupled form of lateral, axial and torsional vibrations. This is confirmed in Fig. 12. In the system there appear axial vibrations with the spectrum structure of 2X and 4X and torsional vibrations with the structure 2X (1X refers here to a frequency corresponding to rotor rotational velocity of $n = 2853$ rev/min). Some surprise can stem here from the fact, that in the vibration spectrum in the axial and torsional vibrations the component 1X does not exist (contrary to other lateral vibrations). This finding indicates towards another (apart form phase spectra) possibility of identification of defect of the rotor crack type.

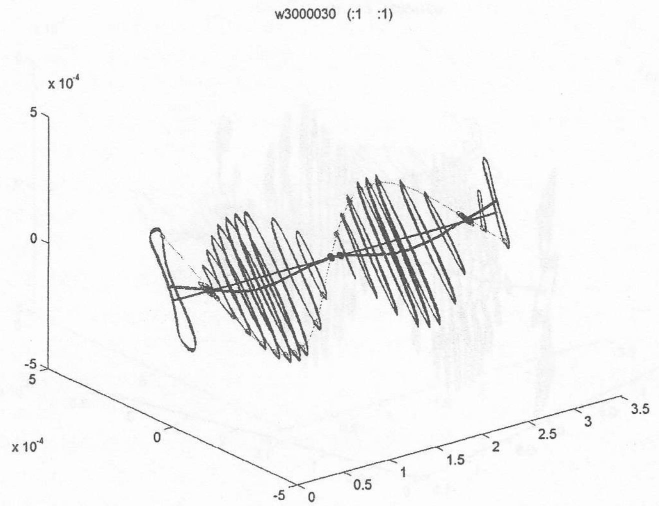

Fig. 5. Trajectories of vibrations of selected nodes of the line of rotors for the base case (without cracks) in comparison against a kinetostatic line of deflections.

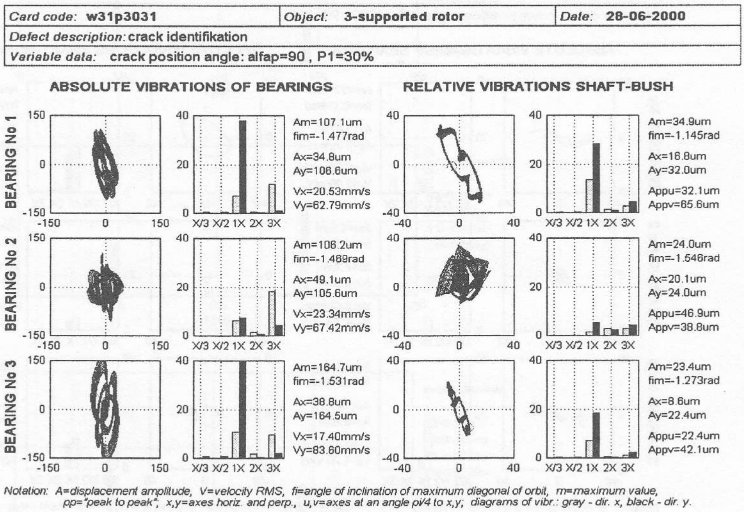

Fig. 6. Diagnostic card calculated for the cracked rotor – angle of crack location $\alpha_p = 90^0$. Least favourable case.

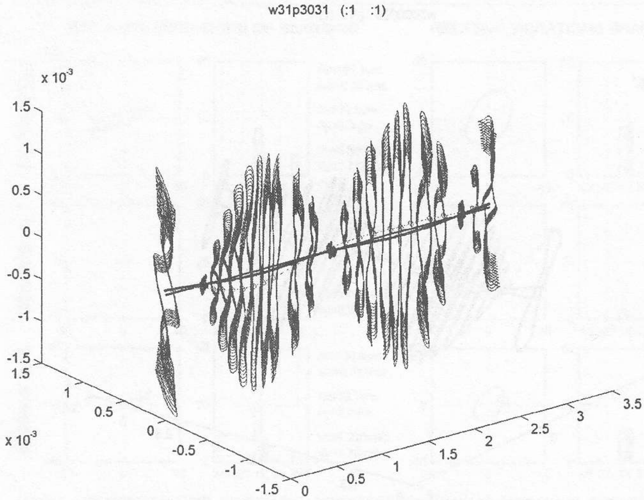

Fig. 7. Trajectories of vibrations of cracked rotor for $\alpha_p = 90^0$.

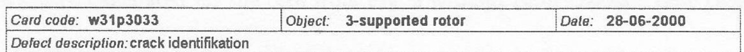

Fig. 8. Diagnostic card calculated for the cracked rotor – angle of crack location $\alpha_p = 270^0$ – most favourable case.

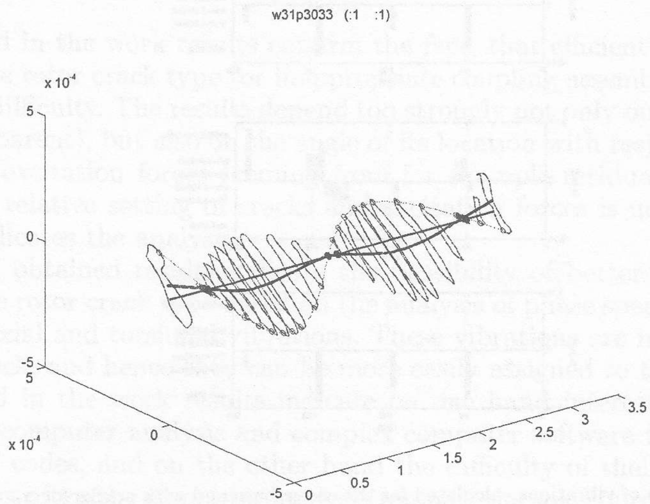

Fig. 9. Trajectories of vibrations of cracked rotor for $\alpha_p = 270^0$.

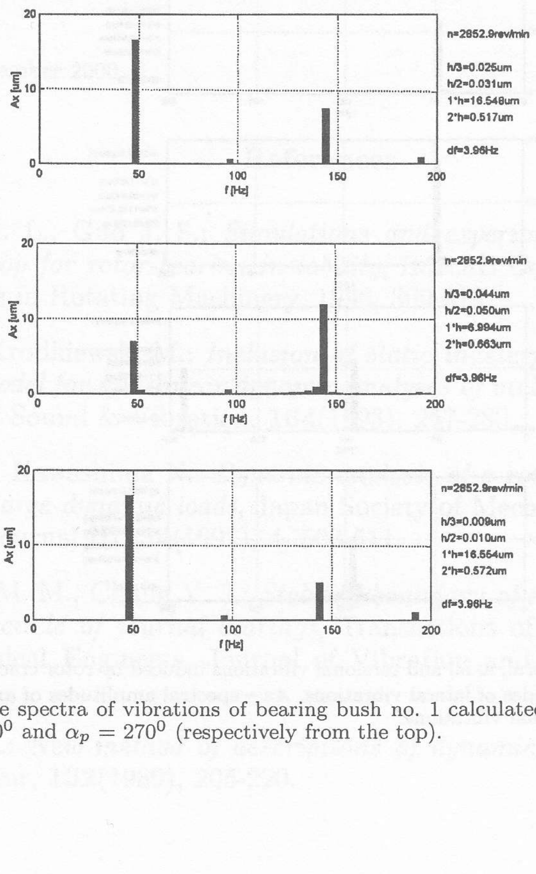

Fig. 10. Amplitude spectra of vibrations of bearing bush no. 1 calculated for the cases: base and with cracks for $\alpha_p = 90^0$ and $\alpha_p = 270^0$ (respectively from the top).

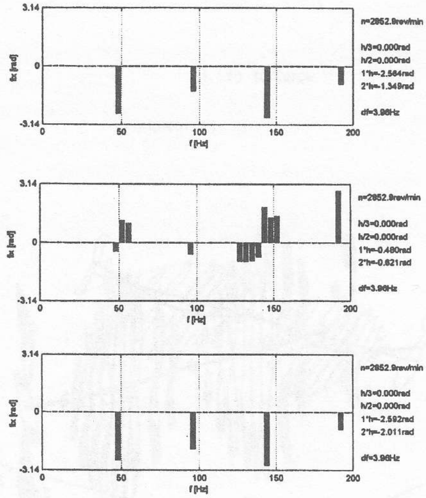

Fig. 11. Phase spectra of vibrations calculated for the cases: base and with cracks for $\alpha_p = 90^0$ and $\alpha_p = 270^0$ (respectively from the top).

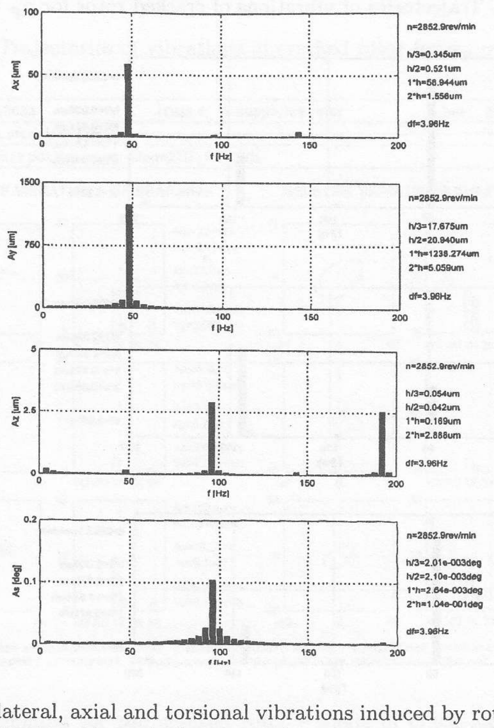

Fig. 12. Coupled spectra of lateral, axial and torsional vibrations induced by rotor cracks. Disk D_2, $\alpha_p = 90^0$, Ax, Ay – spectral amplitudes of lateral vibrations, Az – spectral amplitudes of axial vibrations, As – spectral amplitudes of torsional vibrations.

5. Concluding remarks

Presented in the work results confirm the fact, that efficient identification of defects to the rotor crack type (or inappropriate coupling assembly) is an issue of paramount difficulty. The results depend too strongly not only on the crack depth (which is apparent), but also on the angle of its location with respect to the plane of action of excitation forces (coming from for example residual imbalance). In practice the relative setting of cracks and excitation forces is not known, which dearly complicates the analysis.

However, obtained results indicate the possibility of better identification of defects of the rotor crack type based on the analysis of phase spectra and spectral analysis of axial and torsional vibrations. These vibrations are induced solely by the rotor crack, and hence they can be more easily assigned to this defect.

Presented in the work results indicate on one hand interesting possibilities of advanced computer analysis and complex computer software from the NLDW – 70 suite of codes, and on the other hand the difficulty of their interpretation. However, very often the computer analysis remains the sole source of information about this kind of phenomena, as it is difficult to imagine that for the sake of collection of appropriate data a real rotor from a large power industry object will be damaged in the way to simulate its cracks.

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References

[1] Adams M. L., Guo J. S.: *Simulations and experiments of non-linear hysteresis loop for rotor-bearing instability*, IMechE Conference Transactions, Vibrations in Rotating Machinery, 1996, 309-319.

[2] Ding J., Krodkiewski M.: *Inclusion of static indetermination in the mathematical model for non-linear dynamic analyses of multi-bearing rotor system*, Journal of Sound & Vibration, **164**(1993), 267-280.

[3] Hatori H., Kawashima N.: *Dynamic analysis of a rotor-journal bearing system with large dynamic loads*, Japan Society of Mechanical Engineers International Journal III, **34**(1991), 4, 503-511.

[4] Khonsari M. M., Chang Y. J.: *Stability boundary of non-linear orbits within clearance circle of journal bearings*, Transactions of the American Society of Mechanical Engineers, Journal of Vibration and Acoustics, **115**(1993), 303-307.

[5] Kiciński J.: *New method of descriptions of dynamic properties of slide bearings*, Wear, **132**(1989), 205-220.

[6] Kiciński J., Drozdowski R., Materny P.: *The non-linear analysis of the effect of support construction properties on the dynamic properties of multi-support rotor systems*, Journal of Sound & Vibration, **206**(1997), 4, 523-539.

[7] Muszyńska A., Franklin W. D., Bently D. E.: *Rotor active „anti-swirl" control*, Transactions of the American Society of Mechanical Engineers, Journal of Vibration, Acoustics, Stress and Reliability in Design, **110**(1988), 2, 143-150.

[8] Kiciński J., Markiewicz A.: *NLDW-70 codes for calculations of coupled forms of lateral, axial and torsional vibrations with imperfections of the type: shaft crack and elements misalignment*, IFFM PAS internal report, No. 357/2000 (in Polish).

[9] Kiciński J., Drozdowski R., Materny P.: *Nonlinear model of vibration in a rotor-bearings system*, Journal of Vibration and Control, **4**(1998), 519-540.

[10] Kiciński J., Haller R.: *Computer optimization of external fixation and initial clamping of a bushing with respect to static and dynamic properties of journal bearings*, Transaction of the ASME, Vol. 116, October 1994, 690-697.