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CRACK INDICATORS FOR LARGE POWER TURBO-SET MONITORING AND DIAGNOSTIC SYSTEM

SUMMARY

Active elements in power machine may be controlled with a signals from diagnostic system to stabilize it's operating in a case of emergency work. It needs to identify specific indicators for particular classes of defects. The paper presents possibilities to define such indicators for transverse crack occurring in the rotor of 200 MW turbo-set. The investigations were performed in the way of computer simulations utilizing the NLDW computer code. It bases on non linear model of bearings and Knott crack model. Crack propagation influence on the machine's dynamic state was examined. A resonance being a function of crack depth was detected. The amplitude and phase spectra were discussed as crack indicators as well as torsional/axial vibrations couplings.

Keywords: *rotor dynamics, diagnostics, crack, turbo-set, rotating machinery*

WYRÓŻNIKI PĘKNIĘCIA DLA SYSTEMU MONITORUJĄCO-DIAGNOSTYCZNEGO TURBOZESPOŁU DUŻEJ MOCY

W celu umożliwienia awaryjnej pracy maszyny energetycznej jej elementy aktywne mogą być sterowane sygnałami z systemu diagnostycznego. Wymaga to określenia odpowiednich wyróżników diagnostycznych dla poszczególnych klas defektów. Niniejsza praca przedstawia możliwości określenia takich wyróżników dla poprzecznego pęknięcia wirnika turbozespołu o mocy 200 MW. Badania przeprowadzono na podstawie symulacji komputerowych programem NLDW, bazującym na nieliniowym modelu łożysk ślizgowych i modelu pęknięcia według Knotta. Badano wpływ propagacji pęknięcia na stan dynamiczny maszyny. W wyniku badań wykryto rezonans w funkcji głębokości pęknięcia. Przedyskutowano wykorzystanie widm amplitudowych i fazowych oraz sprzężeń drgań osiowo-skrętnych jako wyróżników diagnostycznych pęknięcia.

1. INTRODUCTION

It is well known that imperfections and defects occurring in rotating machine can change it's dynamic state. We can say the machine dynamic state is a symptom of defect. This is the base of vibrodiagnostics, and relations between defects and symptoms we call "diagnostic relations". Those relations may be implemented into a "knowledge base" of some intelligent systems monitoring dynamic state of working machine. They enables the system to make an appropriate conclusion about the state of machine, especially if there is a possibility of defect [1]. In some cases the system is possible to identify particular defect and activate specific procedures to assure the work of the machine and/or inform the service about it.

If some controllable elements are implemented in the machine structure we are able to vary their properties. We can use that possibility to stabilize the machine's vibrations, especially if it is failed. Of course, it wouldn't be reasonable to run broken machine, but under some conditions it could give more time to decide about it's further operating or to make some additional diagnostic tests for instance. It needs to define an appropriate procedures for emergency work basing on implemented diagnostic system and it's knowledge base. They probably would be different for different classes of defects. So, besides diagnostic relations such a diagnostic and controlling system needs to define specific indicators (discriminators) for particular classes of defects.

The paper presents possibilities to define such diagnostic discriminator for transverse crack occurring in the rotor of 200 MW turbo-set. The investigations were the part of large research project granted by Polish Committee of Scientific Researches (KBN). The project has been performed by seventeen research teams from Poland. Creating the diagnostic relations was performed in the way of computer simulations – the only reasonable way to know the dynamic state of so large critical machine with cracked shaft. It is so called "model based diagnostics". Computer code utilized to perform the simulations is presented. Then the results and a discussion about the crack propagation influence on the machine's dynamic state is performed. At the end of the paper we discuss the possible indicators of crack in the machine.

2. COMPUTER CODE

The computer code that we use to simulate the rotating machinery dynamic state is called NLDW (which are the first letters of Polish phrase "Non Linear Rotor Dynamics"). It is originally invented in IFFM PASci. A diathermic model of heat transfer in slide bearings is it's base. That model contains Reynolds equations, energy and conductivity equations and hybrid lubrication model. The rotor line is modeled with FEM using typical beam elements with 6 DOF's per node. The NLDW code has got a unique possibility to take into account the kinetostatic displacements of the rotor line.

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Dynamic flexibility matrices of the slide bearings support have to be applied. It has to be pointed that equations describing slide bearings model are strongly non-linear. It is fundamentally important to our further considerations. Only non-linear model is possible to generate non-elliptical vibration trajectories (either of the shaft nodes or bearing bush). As we treat such trajectories with an FFT analysis we can obtain a vibration spectra. The symptoms of the defects occurred in the machine can be "hidden" in those spectra. It is essential for model based diagnostics. We could find no symptom with a linear model, because in such case we obtain elliptical trajectories with only one – synchronous spectral line [2].

Figure 1 presents the NLDW algorithm. Because of non-linear equations it is an iterative process. In each time step the stiffness and damping coefficients are calculated for all of the bearings. The shaft elements stiffness matrices are modified due to modeled imperfections and their actual displacements. The process is running until it is convergent and the results are obtained with defined accuracy. As the effect we can get a non-elliptical trajectories and vibration spectra for chosen nodes for all coupled lateral-axial-torsional vibration forms.

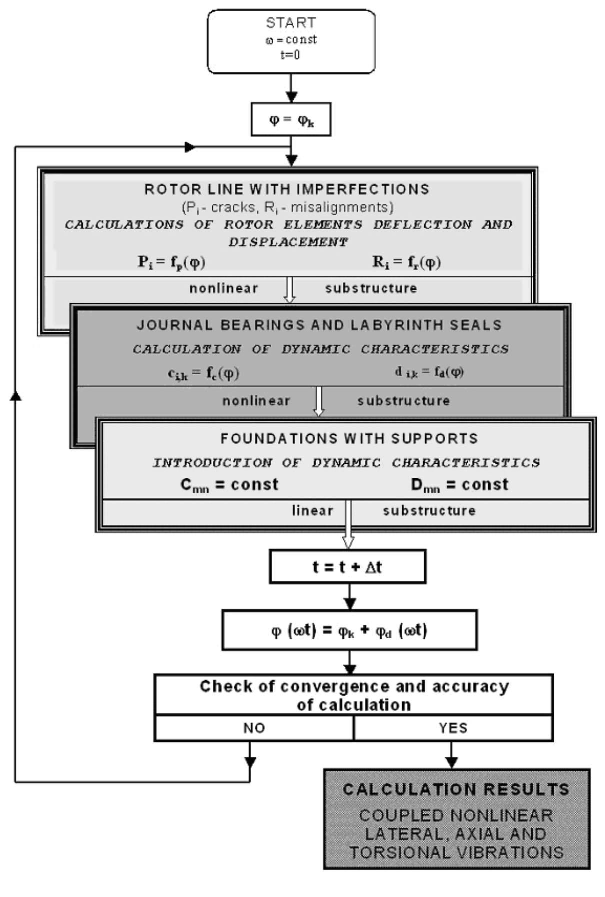

Fig. 1. The NLDW algorithm

Crack is modeled by well known Knott model [3, 4]. It is typical "two-state" model (full close/full open). The appearance of the crack amends the form of the stiffness matrix in such a way that additional influences appear, which are responsible for coupling of the bending and axial vibrations as well as bending and torsional or two-dimensional bending vibrations.

Transformation of the stiffness matrix of the beam element with a crack to the system turned by an arbitrary angle α has been presented in Figure 2 [5].

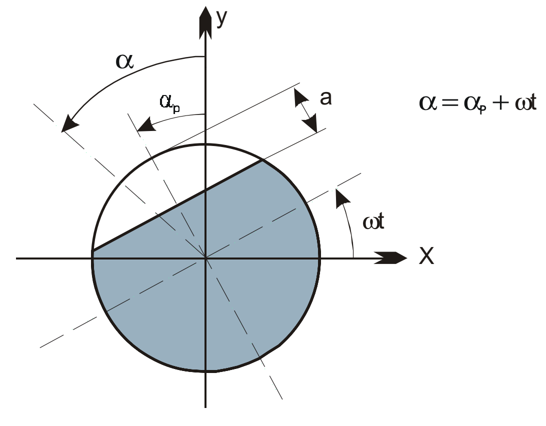

Fig. 2. Beam element with the crack turned with respect to the y axis by an arbitrary angle αp determining the initial location of the crack in the x-y plane

Such angle is a sum of the angle describing initial location of the crack with respect to the reference system α_P and the angle resulting from rotation of the shaft ωt. Applying simple mathematical transformations we obtain the stiffness matrix for the beam element with the crack turned in the x-y plane by the angle α_P.

Crack depth is described by the crack coefficient W_P (1). It can be also called a "relative crack depth" which is absolute crack depth divided by the shaft diameter (both in the length units). It is convenient to user to apply it into NLDW data.

$$W_P = \frac{a}{D} \tag{1}$$

3. THE OBJECT AND METHODOLOGY OF RESEARCH

The object of the research is a 200 MW turbo-set. The model was identified and tuned to the parameters related to the particular real machine working in one of Polish power plants. Figure 3 presents the FEM discretisation of the model:

Two different axial crack positions were taken into consideration (separately):

Crack 1 (CR1) – in the bearing No. 6 (in the node of lateral vibrations),
Crack 2 (CR2) – in the middle of the generator shaft span.

In the second case the slot is placed in the region where the kinetostatic line has the greatest deflection, so this case is expected to generate a greater crack "breathing" effects and to have a greater influence on the rotor dynamic state than the first one (Fig. 4). Rotating speed of 3000 rpm was assumed.

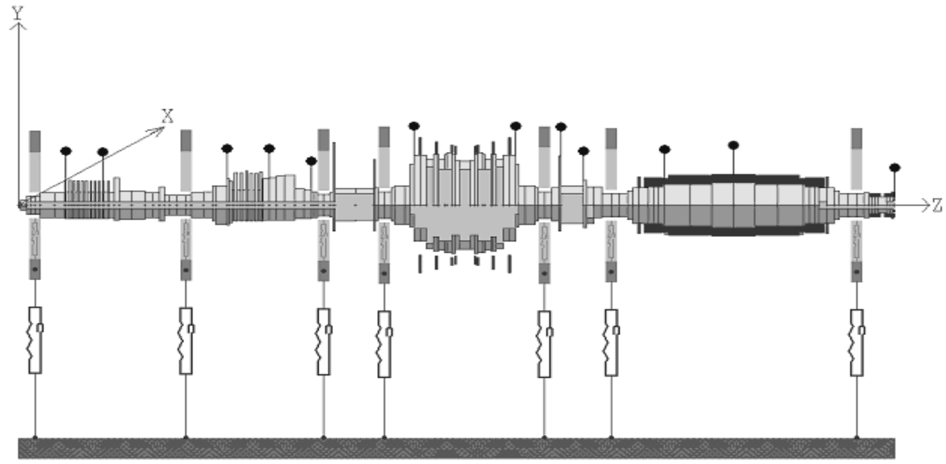

Fig. 3. The rotor line discretisation

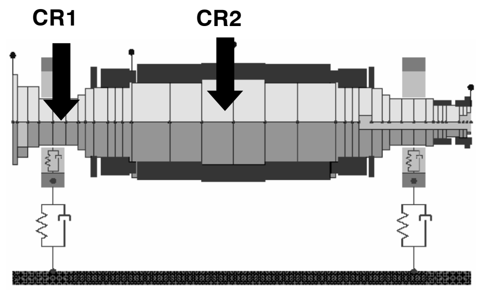

Fig. 4. Crack axial positions

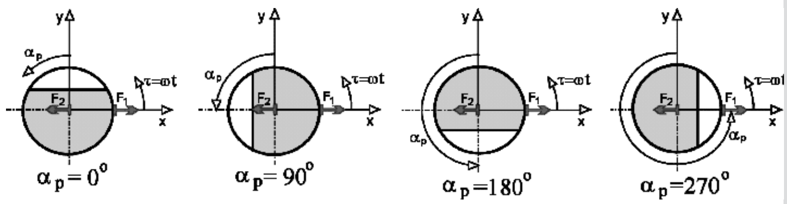

Fig. 5. Circumferential positions of the crack

In each of axial positions 4 different angular crack positions were taken into consideration. They are described by the α_P angle related to the Y axis (Fig. 5). We can say they have also a different phase related to the key-phasor. A crack rotates due to the rotor moving so we may assume α_P is a starting angle. It means that the crack opening or closing time (or rotation angle) is different for different α_P angles. Therefore phenomena caused by the crack have different phase. However rotor is forced by the system of time-varying forces and moments, thus the resultant force is time-varying too. It causes different (and perhaps also time-varying) phase between crack and resultant force or moment. So that the rotor dynamic state is not depending on the kinetostatic deflection only. Dynamic "breathing" effects interact with effects caused by temporary values of external forces and moments. They may let down or amplify each other.

4. CRACK PROPAGATION INFLUENCE ON THE MACHINE'S DYNAMIC STATE

Figure 6 presents some chosen node amplitudes changes due to the crack propagation. Diagrams were made for vibrations of bearing No. 6. Comparison of presented diagrams shows differentiation of the rotor dynamic state due to the crack position.

If crack occurs in the lateral vibration node (CR1) the amplitudes of the lateral vibrations are practically constant. In some particular cases vibrations can even decrease. Crack in the middle of the generator shaft span (CR2) generates other phenomena. The amplitudes increase especially for greater crack depths. The greatest amplitude increase (over 30 times for torsional vibrations) occurs in the case of $\alpha_P = 180^o$ and $W_P > 0.55$. It is supposed to be a result of very strong lateral/axial/torsional vibration coupling. But in the other hand, if crack has another α_P angle, it doesn't generate so strong couplings. So, the crack dynamic effects are suspected to interact with external forces, as it was mentioned before. So it has to be pointed that it is possible that relative crack and forces position in this case is specific and may increase the crack dynamic effect.

The next phenomenon to be mentioned is a resonance. In this case it is probably the function of the crack depth and it occurs for CR2 case and the crack coefficient $W_P = 0.2$–0.25. It is very important. There are a lot of publications describing the cracked shaft natural frequency decrease due to increasing rotating speed. Our calculations were carried out with constant rotating speed. The crack coefficient is the only value that was varied. That's why we can say this resonance is caused by the crack propagation. It is interesting phenomenon and it has to be analyzed more particular in the further works.

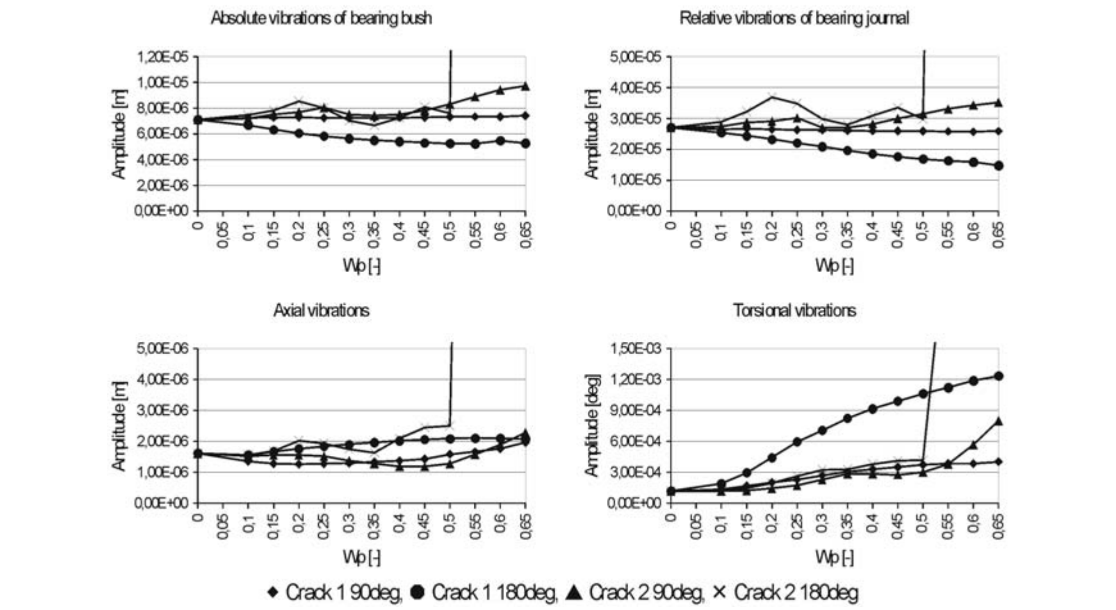

Fig. 6. The crack propagation influence on the machine dynamic state (amplitudes obtained for bearing No. 6)

5. DIAGNOSTIC INDICATORS OF THE CRACK IN ROTATING SHAFT

A presence of 2X harmonic is pointed as typical indicator of crack in rotating shaft. As we can see at Figure 7 they are present also if we haven't crack ($W_P = 0$). In this case a superior harmonic is caused probably by slide bearings instability [6]. Figure shows a case that second harmonics didn't grow due to the crack depth propagation (we can observe growing second harmonics in torsional vibrations [7]). It means that 2X component may be not so good crack indicator in so complicated shafts (multi-supported and subjected to very complicated system of forces and moments).

Figure 7 shows also a phase spectra. They seem to be more proper crack indicators. Their changes are significant but their dependence on crack propagation in the shaft isn't well identified yet. So determination of phase spectra dependence on crack depth is to be a subject of further works.

Let us pay our attention on axial and torsional vibrations. As we mentioned in section 4, torsional vibrations are the most sensitive to crack propagation. So they seem to be better diagnostic indicator of the crack than lateral vibrations. But let us try to analyze axial and torsional vibrations together. Figure 8 presents the diagrams of axial and angular displacements of node related to journal of bearing No. 6 as a time functions. We can see clearly correlations of the effects caused by the crack in the axial and torsional vibration mode. It testifies a kind of coupling between those vibrations. This coupling is possible to recognize even in the case of weak crack propagation effects on transverse vibrations. We hope that the analysis of couplings between the directions of vibration gives the most proper information about the crack in the rotating machine.

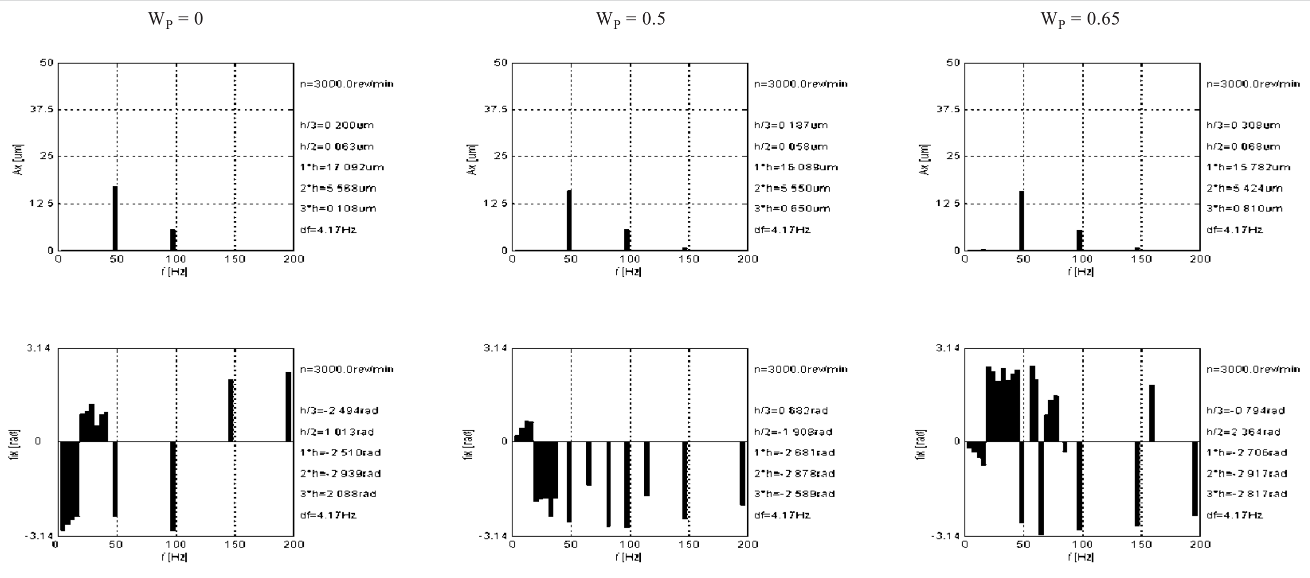

Fig. 7. The amplitude and phase spectra of relative journal lateral vibrations

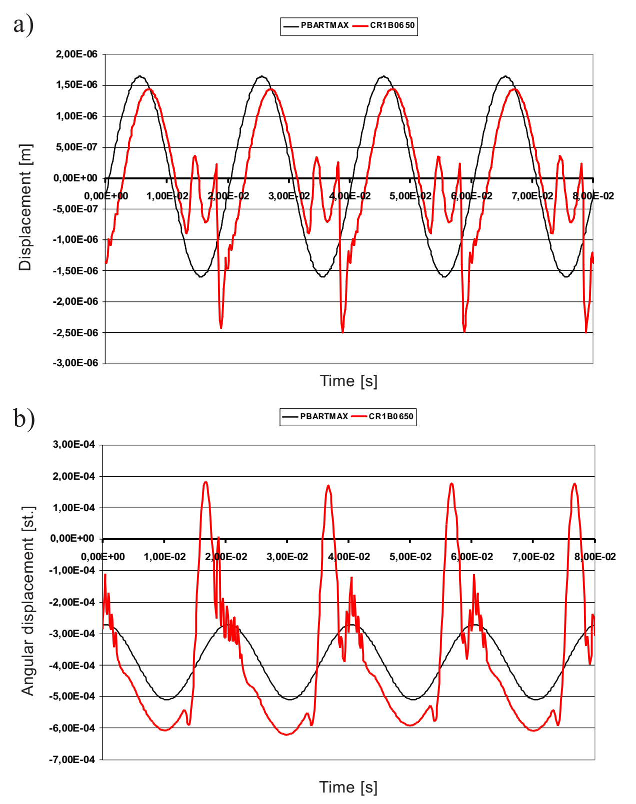

Fig. 8. Coupling between axial (a) and torsional (b) vibrations; bearing No. 6 Crack 1, $\alpha_P = 90^o$

6. CONCLUSIONS

Active elements implemented in power machine may be controlled with a signals generated by diagnostic system. That's why we need to determine a proper indicators for particular classes of defects. In a case of crack the couplings between directions of vibrations seem to be the most hopeful indicators. Torsional/axial vibration coupling looks to be the most visible one. It requires to have a system measuring torsional and axial vibrations mounted on the machine. We can also imagine another quantities that are expected to be an auxiliary indicators – an amplitude and phase spectra for instance. Phase spectra still need to be properly determined and we hope it will be a subject of our further works. We would like to find some correlations between spectra and crack depth and/or location.

There are also another ways to identify a crack in rotating shaft (a run-down test for instance). Nevertheless, if some kind of signal is expected to control any active elements in power machine (active bearing bushes, active elements in supporting structure and even a lever oil in hybrid lubrication), it need to be generated "on-line" and utilized to control immediately. We need controlling (diagnostic) system to separate the signals caused by different defects and control the machine in a way that is specific for particular class of defect. We have shown some phenomena occurring in cracked rotating shaft that may generate signals utilized as crack indicators. The same signals may give an information to the service about the crack and diagnostic system may run some procedures to secure the safety of the machine. Another question is what kind of active elements we may implement in the large power machine. We think the answer on this question is not simple and it needs to undertake a lot of interdisciplinary research projects.

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