NON-LINEAR VIBRATIONS AS A NEW DIAGNOSTIC TOOL - CRACK DETECTION EXAMPLE

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Summary

It is widely acknowledged from literature that material or structural imperfections of the shaft crack type are capable, in some cases, of generating coupled **non-linear** forms of lateral-axial-torsional vibrations in the rotating machine. Such information, however, pertains to small or model objects of that type. In the paper presented are investigations into the influence of the shaft crack on the dynamic state of a large, multi-supported rotating machine. Determined have been the cases where the crack is capable of inducing pronounced forms of vibrations, which can serve as basis for more reliable and effective diagnostics, or the cases, where the mechanism of couplings deteriorates, despite the existence of cracks with significant depths. Determined has been the degree of sensitivity of non-linear and mutually coupled system vibrations on crack propagation and conducted had been the assessment of their applicability as a diagnostic determinant of a the dynamic state in a large rotating machine.

Keywords: non-linear vibration, rotor dynamics, crack detection

Streszczenie

Znanym z literatury jest fakt, że imperfekcje materiałowe lub konstrukcyjne typu pęknięcie wału w niektórych przypadkach są w stanie wygenerować sprzężone drgania poprzeczno-wzdłużno-skrętne maszyny wirnikowej. Informacje te odnoszą się jednakże do małych lub modelowych obiektów tego typu. W pracy podjęte zostały badania wpływu pęknięcia wału na stan dynamiczny dużej, wielopodporowej maszyny wirnikowej. Określone zostały zarówno przypadki, dla których pęknięcie jest w stanie wywołać znaczące sprzężone formy drgań mogące być podstawą dla bardziej wiarygodnej i efektywnej diagnostyki, jak też i przypadki, dla których mechanizm sprzężeń zanika pomimo istnienia pęknięć o dużych głębokościach. Określony został stopień czułości nieliniowych i wzajemnie sprzężonych drgań układu na propagację pęknięcia oraz przeprowadzona została ocena ich użyteczności jako wyróżnika diagnostycznego stanu dynamicznego dużej maszyny wirnikowej.

1. INTRODUCTION

Issues of early detection of material and construction imperfections in rotating machines have been the merit of numerous publications for several years now [1-17,20,24,25]. Despite such abundant investigations conducted in that area all over the world there are still some issues unresolved. This regards particularly such problems as coupled forms of non-linear vibrations of multi-supported rotors induced by for example the shaft crack or the issue of appropriate determination of diagnostic determinant of such state.

Tracing the results of investigations on the assessment on the dynamic state of cracked rotors and early detection of that kind of defects [1-10] there can be several more general conclusions drawn, namely:

- directions of investigations tend to focus on the 1X and 2X harmonic responses in vibration spectra
- the 2X harmonic component is the most practical crack indicator to be implemented in a monitoring system [2,21]
- the 1X component of lateral vibrations is not sensitive to the crack depth, but the 1X component of axial vibrations and 2X component of lateral vibrations are very sensitive .

Obviously above conclusions are correct in principal, especially in specific cases analysed by authors of those publications. There can however be indicated situations, where the above conclusions are valid in a limited range and their generalisation may lead to significant errors in the analysis of the state of the object. This regards particularly large and multi-supported rotating machinery founded in slide bearings. One of the objectives of the present work is to indicate such cases.

Analysis of coupled, non-linear forms of vibrations of complex systems such as rotor-bearings-foundation requires development of qualitatively new research tools in the form of adequate models and computer codes capable of generating defected vibration spectra in a linear and moreover non-linear regime of operation of considered object. Question arises, why is the non-linear description in such case so important? The answer is, that only the non-linear description (even in the stable range of system operation) generates non-elliptical trajectories and vibration spectra, in the shape of which the considered effects can be coded in. Such tools together with investigations on that topic will be presented in the present paper. At the same time an attempt will be made to justify the thesis that in the case of large and complex rotating systems the coupled forms of vibrations can serve in many cases as a better diagnostic determinant of the dynamic state of the object.

2 RESEARCH TOOLS

In the classical rotating machine there can be discerned three principal sub-systems:

- rotor line with discs, clutches and imperfections like cracks or misalignments
- hydrodynamic journal bearings and labyrinth seals
- foundations with supports and bearings external fixings

Particularly difficult in theoretical modelling are slide bearings and labirynth sealings. At IFFM PAS in Gdansk there has been developed a so called diathermal model of heat transfer in bearings (the code DIATER), which consists of coupled Reynolds, energy and conduction equations and the model of hybrid lubrication in the case of supplying from the siphon pockets and possible bush skewness (the code IZOSLEW). The above bearings models have been described in detail in [23] or partially in [22,24,25] and hence will not be presented in the present paper. It is not even the intention of the latter.

The line of rotors with discs has been modeled by means of the FEM method featuring typical beam elements with 6 degrees of freedom in each node [16,17,22]. In order to account for a transverse shaft crack there has been applied, known from literature, model of the element with transverse crack due to Knott et al. [16,17] of the fully open – fully closed type.

A dynamic influence coefficient matrix of supporting structure and foundation has been determined by means of known commercial codes such as ABAQUS and ADINA, based on the FEM method.

A key issue is now to develop a general algorithm of calculations, both kinetostatic and dynamic, combining all mentioned above sub-systems of the entire system, and also development of the algorithm of dynamic calculations incorporating possible non-linear external excitations of the system and large displacements of shafts in bearings.

Obviously, one possible way of solving of such system of equations is the iterative process. In Fig. 1 presented is a general algorithm of calculations, where at each time step calculated are new coefficients of shaft stiffness with imperfections corresponding to their actual values of displacement. Iterative process is conducted up to the moment of obtaining a full convergence of results of calculation
and satisfactory accuracy. In effect we will obtain non-elliptical trajectories of displacement and spectra in selected nodes for all coupled forms of lateral-axial-torsional vibrations.

It is worth stressing at that point the most important issue, namely the way of combination of the shaft "breathing" process with its kinematic and dynamic displacements. Suggested algorithm of non-linear calculations for the entire system (Fig. 1) enables combination of a linear model of the element itself with the crack in the way enabling non-linear analysis of final results. It is possible as analysed dynamic and kinetostatic displacements of the entire line of rotors have a non-linear character. Therefore the process of opening and closing of a crack depends on its location on the line of kinetostatic deflections φ_k as well as instantaneous dynamic displacements φ_d as schematically presented in Fig. 2. In the assumed model it is possible to locate the crack on the circumference with respect to the plane of action of external excitation forces denoted by the angle α_p. In effect the process of crack "breathing" (opening and closing) is a very complex function of following parameters: φ_k, φ_d and α_p.

The influence of kinetostatic deflections of the line of rotors on the form of crack interaction can be of very significant importance in some cases. At relatively high values of kinetostatic deflections compared to instantaneous dynamic deflections the "breathing" process may cease and the crack may remain open or closed at all times.

Suggested algorithm of calculations and particularly the way of incorporation into the procedure of the model of element with the crack with account of kinetostatic and dynamic rotor shaft deflections in the linear and non-linear range of displacements opens qualitatively new possibilities of assessment of the state of rotating machine with shaft imperfections and forms, in author's opinion, an important element of novelty in the present work.

Based on the above model developed has been at the IFFM PAS in Gdańsk a suite of computer codes with a general name NLDW, consisting of a whole family of specific codes for the analysis of rotor dynamics, slide bearings and supporting structure – Fig. 3.

The system NLDW forms a basic research tool utilized in the present work. Due to apparent reasons there will not be discussed issues concerned with capabilities of that vast system nor the details related to the description of utilized equations and simplifying assumptions. Such information can be found in [18-20,22,24,25]

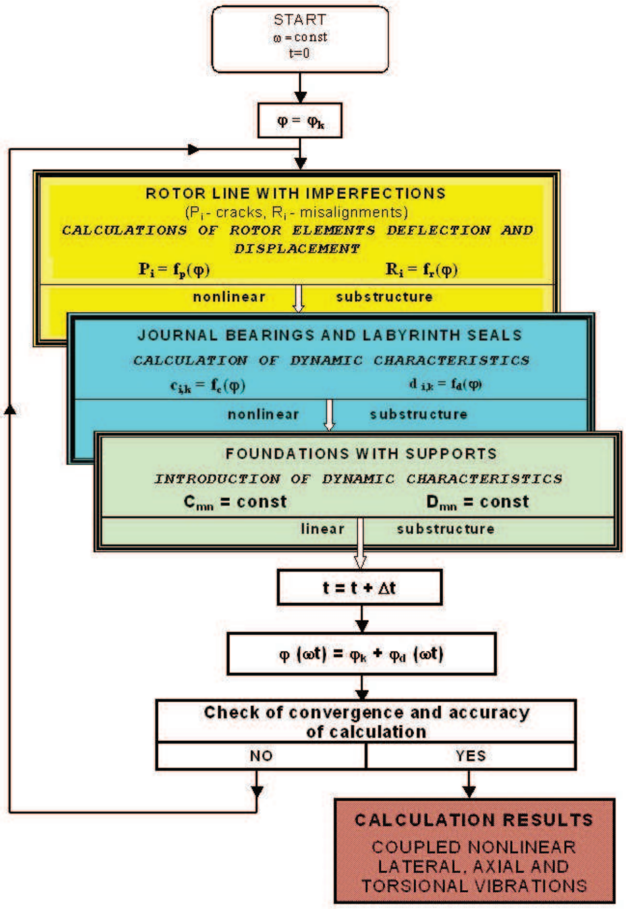

Fig. 1. A general algorithm of iterative procedure enabling non-linear analysis of coupled forms of vibrations with imperfections.

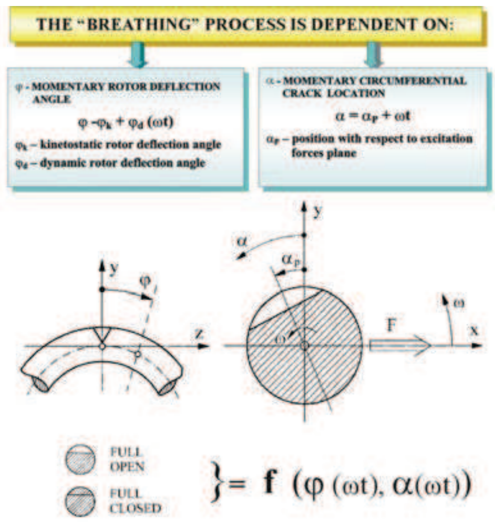

Fig. 2. Relation between the crack "breathing" process, the rotor kinetostatic and dynamic deflection and circumferential position.

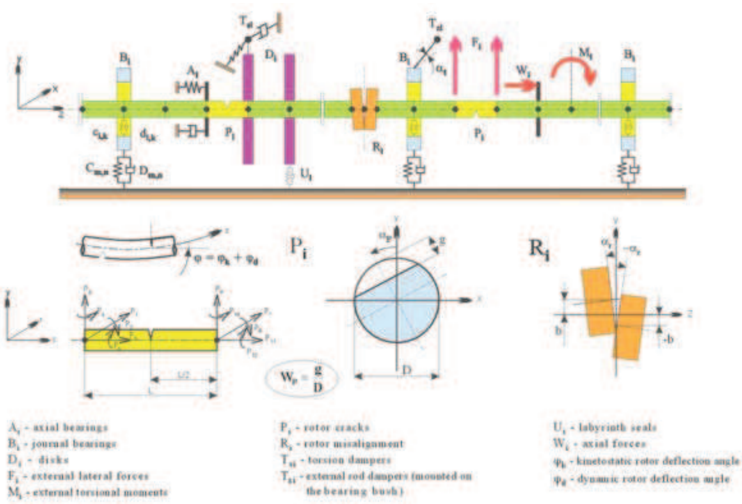

Fig.3. A computer system NLDW for non-linear analysis of rotor dynamics with imperfections, disks, slide bearings and supporting structure

3. OBJECT OF INVESTIGATIONS

Let's assume in further investigations the object presented in Fig. 4. It is a three-supported rotating machine with a shaft diameter d=0.1 m, its total length L=3.2 m with two discs of diameter D=0.4 m, supported in three slide bearings with cylindrical clearance. Matrices of dynamic influence of supporting structure have been experimentally measured. The system is influenced by two, counteracting in the phase domain, external excitation forces F1 and F2, which are the effect of action of mass imbalance M1=M2=0.0154 kg applied to both discs of the radius r =0.18 m. In Fig. 4 presented is FEM discretisation of the line of rotors in assumed object together with location of external excitation forces.

Let's assume now that in the system, defined in such a fashion, there can exist interchangeably two cracks, namely **crack1** situated in the rigid clutch, hence in the vicinity of a middle bearing No. 2 and **crack2** situated in the middle part of the line of rotors between bearings No.2 and 3, hence in the part where large shaft displacements are expected. Obviously, the cracks **crack1** and **crack2** will assume a different depth and will be situated in various locations on the circumference with respect to the plane of action of external excitation forces. Investigations will be conducted for four selected angles α_p of circumferential location of a crack, as illustrated in Fig. 4. In the present work assumed has been a definition of a relative crack depth $\mathbf{W}_p$ assumed as a ratio of the crack depth to a shaft diameter.

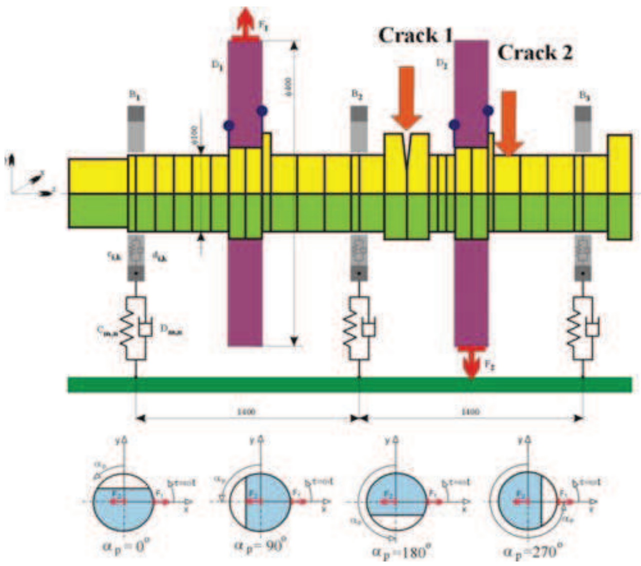

Fig. 4. FEM discretisation of a line of rotors of assumed object with localisation of cracks and specification of cases of their circumferential localisation.

4. INVESTIGATIONS INTO THE INFLUENCE OF CRACK LOCATION AND PROPAGATION

4.1. Influence of circumferential location of crack on transverse vibrations

Let's start our considerations from depicting the pattern of vibrations in the assumed object without cracks in the rotor shaft. Such case is regarded as a reference, very helpful in comparative analysis of cases with the crack. The results of calculations for the reference (base) case, in the form of so called diagnostic card, have been presented in Fig. 5. Calculations have been conducted for the system resonance velocity in the stable range of rotor operation (below the stability threshold limit). It is worth noticing here that assumed multi-supported rotor configuration generates dominant synchronous components of 1X type, but also generates notable super-harmonic components of 3X type (which is perceptible particularly in the case of absolute vibrations of the bearings bushes No.1).

Let's assume now, that in the assumed object we are dealing with a crack situated in the clutch of the line of rotors, i.e. with the case of a **crack1** , as denoted in Fig.4. Conducted has been a series of calculations for different crack depths and different circumferential angles of location of α_p. It turned out that for each value of the crack depth there can be discerned two characteristic cases, namely the least and the most circumferentially advantageous location of crack, from the point of view of relative and absolute vibrations of bearing nodes, which corresponds to the angles α_p =90º and α_p =270º, respectively. In Fig. 6 and Fig. 7 presented are diagnostic cards calculated for these two selected cases for the lateral vibrations.

As can be seen from the above figures the influence of circumferential location of a crack is very important, however very different in the case of absolute vibrations of bushes and relative vibrations of oil film. In the case of unfavourable location of a crack (α_p =90º) the amplitudes of absolute vibrations are over three times greater than in the case of the reference case. Interesting, that the same crack does not impose such a significant influence on relative vibrations of the oil film. It is sufficient, however, to shift the circumferential location of a crack toward a value of α_p =270º (the most favourable case) that the absolute vibrations of the bearings bush have become comparable to the base case, hence the case without the shaft crack.

An interesting observation here is a fact that in the vibration spectrum we do not observe noticeable components of 2X type, however, values of super-harmonic components of 3X type has increased compared to the reference case (compare Fig. 6 and Fig. 5). **This means, that there can exist cases, where the crack in the system reinforces already existent super-harmonic components and not necessarily generates a typical and expected spectrum component of 2X type.** The above conclusion can be very helpful in the case of monitoring systems of the state of the object, which bases generally on the analysis of spectrum components of 2X type (in the case of lateral vibrations).

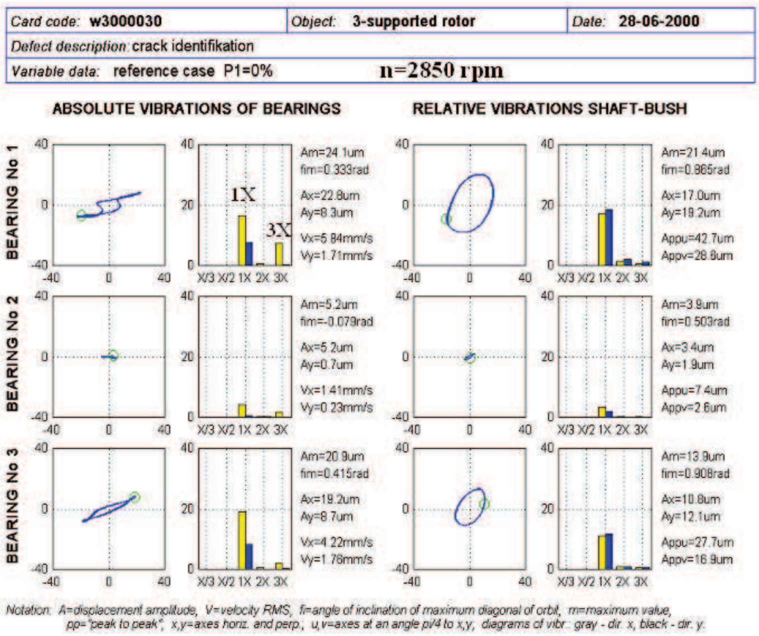

Fig.5. The results of calculations in the form of diagnostic card for the reference case without the crack. Trajectories and spectra of absolute bush vibrations and relative oil film vibrations for three bearings of multi-supported rotor from Fig. 4.

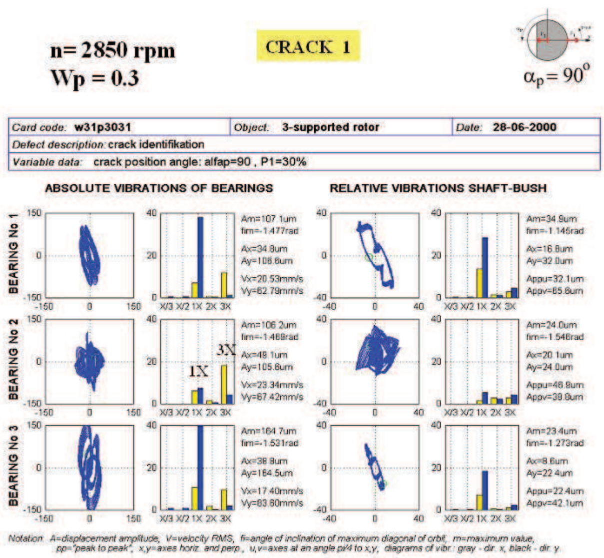

Fig. 6. Diagnostic card calculated for a crack with the depth of W_p=0.3 and the least favourable circumferential location α_p =90º.

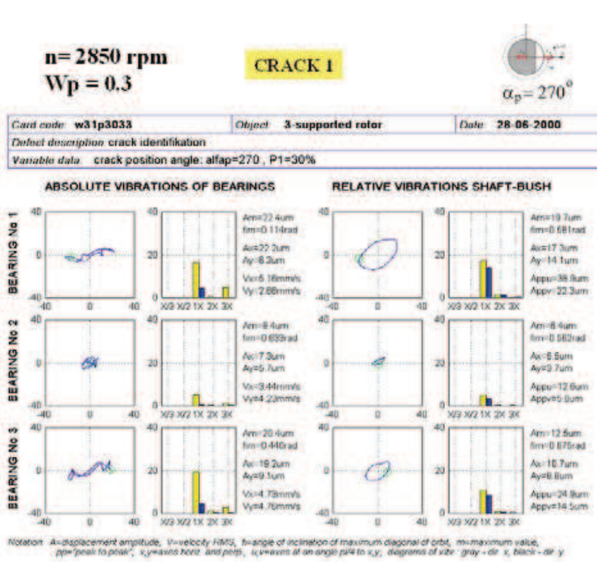

Fig. 7. Diagnostic card calculated for a crack with the depth of W_p=0.3 and the most favourable circumferential location α_p =*270º*

4.2. Coupled forms of vibrations induced by a crack

Let's proceed now to considerations related to coupled forms of vibrations and assessment of their applicability as a diagnostic determinant of a dynamic state of a large rotating machine. The first question to be answered is whether the rotor shaft crack is capable of generating, apart from apparent changes in distributions of lateral vibrations, significant axial and torsional vibration, despite the fact that in these directions there are no action of any external forces? In order to provide answer to that question conducted have been investigations, the results of which have been presented in Fig. 16-18.

From Fig. 8 it results that a crack with a moderate depth W_p=0.2 induced a second, very strong and dangerous resonance in the system **R2** at lower rotational velocity of the rotor. Let's see what will be the distribution of coupled forms of vibrations at a rotor velocity exactly corresponding to resonance **R2,** hence potentially the most dangerous situation. Let's consider the case of least and most favourable circumferential location of the crack and see whether the crack is capable of inducing serious couplings in vibrations. The results of calculations have been presented in Fig. 9 and Fig. 10. It results from them that **the crack under conditions of a strong resonance can render very strong couplings of system vibrations**, for example amplitude of axial vibrations in the vicinity of bearing No. 2 is about 80 μm, and hence is of the similar order than the amplitude of lateral relative vibrations of oil film of that bearing! It is interesting that the amplitude of lateral absolute

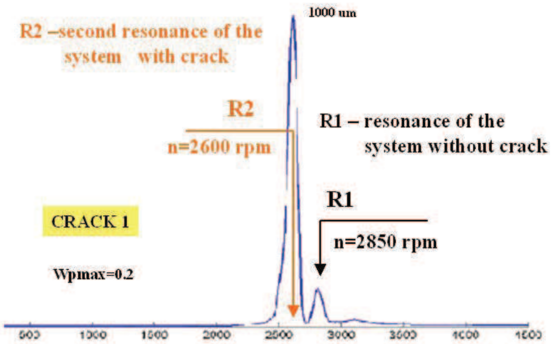

Fig. 8. Distribution of amplitude-frequency characteristics of the system for the crack situated in the clutch (crack1)with the depth W_p=0.2 and circumferential crack location α_p =90°.

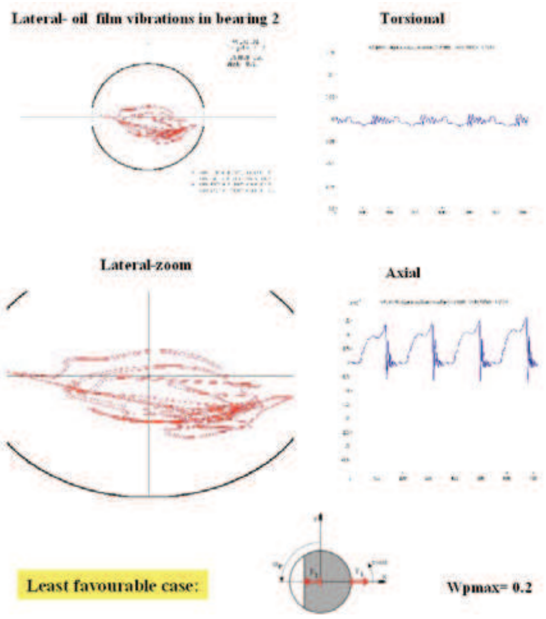

Fig. 9. Coupled forms of vibrations induced by the **crack1** with the depth W_p=0.2 calculated in the strong system resonance **R2** (n= 2600 rpm - Fig.16) for the cir-cumferential crack location **α_p =90º.**

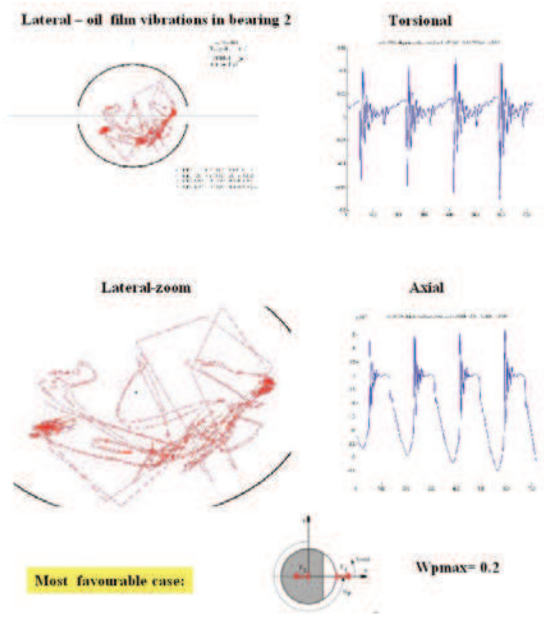

Fig. 10. Coupled forms of vibrations induced by the **crack1** with the depth W_p=0.2 calculated in the strong system resonance **R2** (n= 2600 rpm - Fig.8) for the circumferential crack location **α_p =270º.**

vibrations at that location is greater by one order of magnitude. **This confirms the earlier conclusion, that the crack more formidably influences absolute vibrations of the bush or shaft than the relative vibrations of oil film, which are simply constrained by the extent of the bearing clearance.** Obviously, such large values of amplitudes of absolute vibrations of the shaft or bushes are not allowable in practice due to the possibility of the failure of theentire system.

Another interesting conclusion stemming from Fig. 9 and Fig. 10 is related to the influence of circumferential location of a crack. It turned out that such influence still remains significant, however this time it is of a different tendency. Hitherto most favourable case turned out to become the worst case. That remark regards both the lateral vibrations and coupled axial and torsional vibrations. The above means, that the influence of circumferential location of the crack in complex machines cannot be unanimously determined, which apparently impedes its diagnostics.

5. DISCUSSION AND CONCLUDING REMARKS

Let's return now to the principal thesis in the paper and the question whether the defect of the shaft crack type is capable of generating significant coupled forms of vibrations in a complex, multi-supported rotating machine and can these vibrations be a better diagnostic determinant of the state of the object.

The question formulated in the above fashion does not, unfortunately, have a unanimous answer. Generated coupled forms of vibrations strongly depend on the location of the crack in the system as well as object operational conditions. There exist locations, where the couplings are important and can serve as basis for a more reliable diagnostics. Also during the object operation, for example under resonance conditions there can be recorded vibrations in the directions where external excitation forces are not acting. This regards particularly torsional vibrations. These are not only fragile toward propagation of a crack, but also attain, at greater values of crack depths, measurable values.

The crack along the line of rotors can be, however, situated in such a way that it will not be capable of inducing significant couplings in the system and hence the values of coupled amplitudes, such as axial-torsional, will become practically impossible to be measured. That remark pertains both to the object operation before and beyond the threshold of system stability. The development of oil whirls markedly impairs the coupled forms of vibrations.

In cases, where significant coupled forms of vibrations, due to crack, have been generated, their analysis can definitely supply more reliable information about the state of the object and at the same time serve as a better tool in early detection of the machine defect.

Summarising the considerations conducted in the present work we can conclude that in the case of large rotating machines a unanimous and early detection of defects of the shaft crack type is not always possible. There are situations, where the lack of characteristic components of the vibration spectrum (such as 2X type) or the lack of coupled forms of vibrations in the systems does not automatically implies that there are no cracks. This implies the necessity of individual treating and analysis of each object and also places high requirements to the contemporary diagnostic systems. Such requirements are even higher in the light of the fact that usually we do not know *a priori* about the crack location nor about amended resonance velocities of the object, amended by its influence, which indicates a potential possibility of ambiguous relations of the defect-symptom type and difficulties in identification of adequate diagnostic determinant of the state of the machine. On the other hand, however, modern methods of computer analysis, based on new models and computer codes, as shown in the results in the present paper, lay grounds for a better and safer diagnostics of emergency states of rotating machines in future.

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