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POLISH ACADEMY OF SCIENCES

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

113

Selected papers from the International Conference
on *Turbines of Large Output*
devoted to 100th Anniversary of
Prof. Robert Szewalski Birthday,
Gdańsk, September 22-24, 2003

WYDAWNICTWO IMP PAN

GDAŃSK 2003

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

Appears since 1960

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Transactions of the Institute of Fluid-Flow Machinery have primarily been established to publish papers from four disciplines represented at the Institute of Fluid-Flow Machinery of Polish Academy of Sciences, such as:

- Liquid flows in hydraulic machinery including exploitation problems,
- Gas and liquid flows with heat transport, particularly two-phase flows,
- Various aspects of development of plasma and laser engineering,
- Solid mechanics, machine mechanics including exploitation problems.

The periodical, where originally were published papers describing the research conducted at the Institute, has now appeared to be the place for publication of works by authors both from Poland and abroad. A traditional scope of topics has been preserved.

Only original and written in English works are published, which represent both theoretical and applied sciences. All papers are reviewed by two independent referees.

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Editorial

These Special Issues of the *Transactions of Fluid-Flow Machinery*, Nos. 113 and 114 contain selected papers from the International Conference on *Turbines of Large Output* devoted to commemorate 100$^{\text{th}}$ Birthday Anniversary of Prof. Robert Szewalski. The conference was held in Gdańsk, Poland on September 22-24, 2003.

The Conference is a continuation of previous conferences held at the Institute of Fluid-Flow Machinery PAS in former years dedicated to technology of steam turbines. Series of conferences bearing the same name took place in the years 1962, 1965, 1968, 1993. In 1997 organised has been a conference on steam turbines and related topics, but with a slightly amended title – *Problems of Fluid-Flow Machinery.* At present at the Institute of Fluid Flow Machinery there are conduced research of fundamental character encompassing both the issues related to steam turbines and fundamentals of power engineering.

Organisers of the present conference have returned to the traditional name, i.e. Conference on *Turbines of Large Output* to mark the respect to the memory of Professor Robert Szewalski (1903-1993), the founder and a director of the Institute of Fluid-Flow Machinery for several years, and initiator of the conference series devoted to the problems of steam turbines.

The Editors are very grateful to the referees of the papers presented in this issue of the *Transactions of Fluid-Flow Machinery*: J. Badur, W. Batko, E. S. Burka, J.T. Cieśliński, P. Doerffer, A. Gardzilewicz, B. Grochal, J. Kiciński, G. Kosman, T. Król, J. Krzyżanowski, J. Mikielewicz, A. Neyman, W. Ostachowicz, R. Puzyrewski, R Rządkowski, J. Świrydczuk, M. Trela, T. Uhl and Z. Walczyk.

We appreciate cordially the manifested authors interest in our conference. Special thanks are conveyed to Ms 'Maya' Bagińska, for her fruitful editorial help related to directing the papers to reviewers and correspondence with the authors of contributions.

Prof. Jarosław Mikielewicz
Editor-in-Chief

Dr Edward Śliwicki
Managing Editor

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

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Coupled forms of vibrations in rotating machinery

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Abstract

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 methods of modelling of the selected defects have been presented (e.g. rotor cracks and bearings misalignments). These defects have been regarded as the factors generating the coupled forms of vibrations. In the paper a part of work devoted to the influence of circumferential location of the crack and kinetostatic and dynamic rotor deflection line on the system dynamical response has been presented. Applicability of coupled and phase vibration spectra analysis in diagnostics of such kind of defects has been indicated. 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 be rotor cracks and misalignment of the rotor line.

Keywords: Vibrations; Rotating machinery; Slide bearings; Computer analysis

1 Introductory remarks

The problems linked to the dynamics of rotating machinery and the analysis of various forms of vibrations are the topic of research in several scientific centres both in Poland and abroad. Despite such numerous information on that topic there are still many unresolved problems. That regards the analysis of nonlinear and coupled bending, axial and torsional vibrations in particular. Analysis of that type of phenomena requires application of very advanced models and computer codes. The problem becomes particularly difficult in the case of large power

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engineering objects, such as turbosets of large power. The coupled forms of vibrations are usually a result of different kinds of couplings present in the system as well as interactions connected with structural and material imperfections. Of importance here are rotor cracks and rotor line misalignments. Modelling of such kind of imperfections and the following analysis of their influence on the system operation is definitely one of contemporary challenges in that branch of science [1-13].

The main objective of this paper and accepted thesis are presented in Fig. 1.

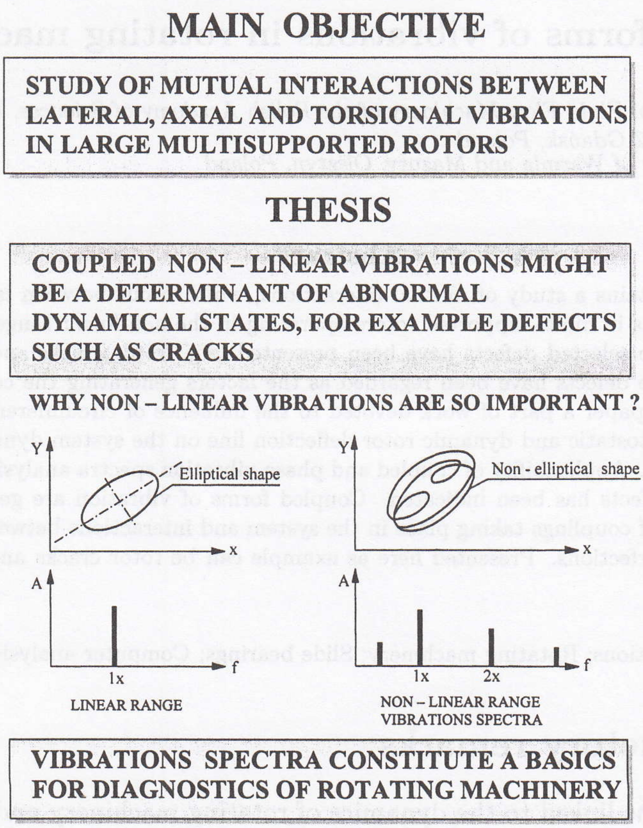


Figure 1. The main objective and thesis accepted in this paper.

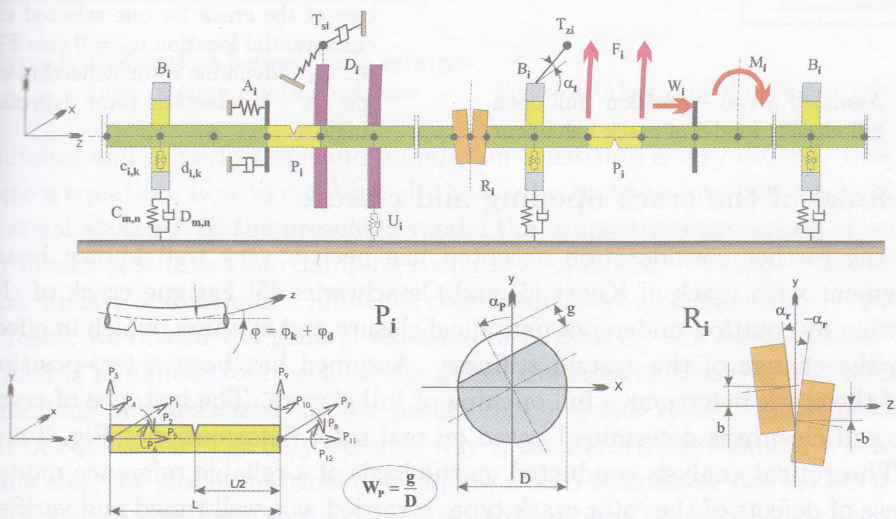
Spatial trajectories of vibrations of the line of rotors calculated for the generator's cracked rotor $P_1 = 30\%$ and circumferential location of crack $\alpha_p = 270°$ (coordinates in [m]).

2 Theoretical model

Presented in the work results have been obtained using a model together with the suite of computer codes NLDW-70, which have been described in detail in [10-13]. The code enables analysis of vibrations of the rotor-bearings-support system in the nonlinear range, i.e. calculates non-elliptic trajectories of selected node displacements for arbitrarily varying in time (but periodical) forces and external moments as well as a given system configuration (discs, dampers, journal and axial bearings, sealings, foundation, rotor imperfections of the crack and misalignment type) and enables construction of the vibration spectrum – Fig. 2.

Experimental verification of the assumed model forms a separate issue, which has been presented in [11, 13]. Presentation of the results of verification would fall far beyond the volumetric framework of the present paper. It is worth emphasis here that conducted also has been model verification on the real objects, that is power engineering turbosets of 200 MW power [14].

Due to the fact that the coupled forms of vibrations induced by the rotor crack are the topic of the present work, in the following body of the text presented will be the assumed FEM model of the element with a crack.



A_i - axial bearings
B_i - journal bearings
D_i - disks
F_i - external lateral forces
M_i - external torsional moments
P_i - rotor cracks
R_i - rotor misalignment
T_{si} - torsion dampers
T_{zi} - external rod dampers (mounted on the bearing bush)
U_i - labyrinth seals
W_i - axial forces
φ_k - kinetostatic rotor deflection angle
φ_d - dynamic rotor deflection angle

Figure 2. Calculational capabilities of a code NLDW-70 for nonlinear rotor dynamics with imperfections.

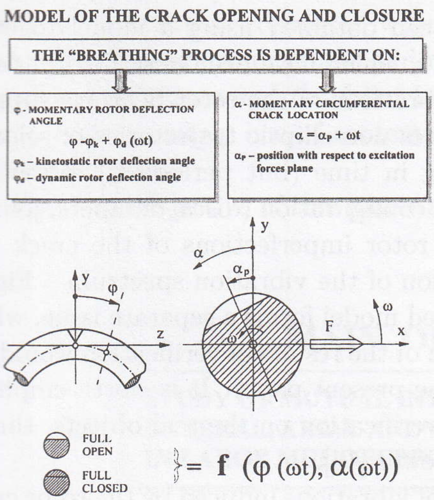


Figure 3. Assumed a two – position (full open, full closed) model of crack behaviour.

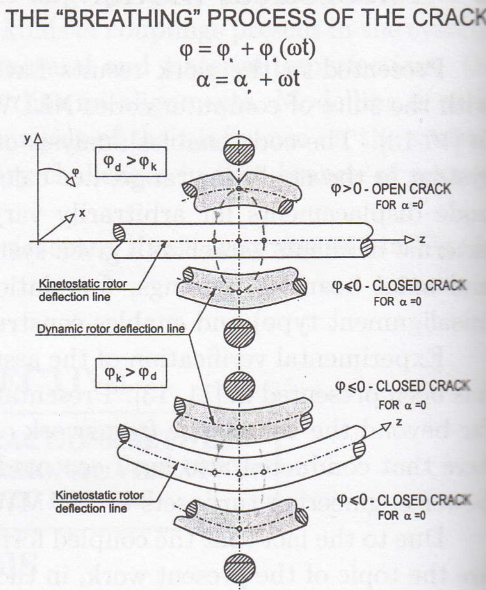


Figure 4. Explanation of the "breathing" process of the crack for one selected circumferential location $\alpha_p = 0$ (see Fig. 9). φ_d – dynamic rotor deflection angle, φ_k – kinetostatic rotor deflection angle.

2.1 Model of the crack opening and closure

For the further consideration accepted has been a very well known beam FEM element with crack of Knott [6] and Ostachowicz [5] Fatigue crack of the shaft during its rotation undergoes periodical closure and opening, which in effect leads to the change of the system stiffness. Assumed has been a two-position model of the crack behaviour – full opening or full closure. The instance of crack opening and closure is determined based on real rotor deformation – Fig. 3 and Fig. 4. Theoretical analysis conducted on the basis of a reliable reference model, in the case of defects of the rotor crack type, regarded as a well tuned and verified model of the real object without cracks, is often the only possible way of acquiring information about the behaviour of a system with a cracked rotor.

2.2 Nonlinear model. General equation of motion

The connections between main substructures of the large rotating machine, that means between journal bearings and labyrinth seals, rotor line with discs,

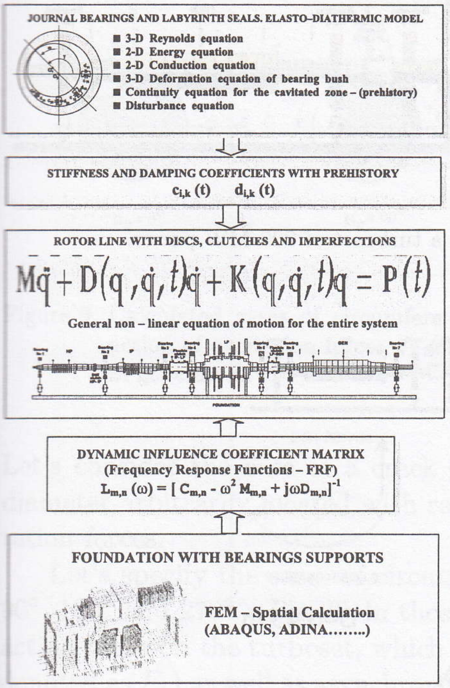


Figure 5. Connections between main substructures of large rotating machine.

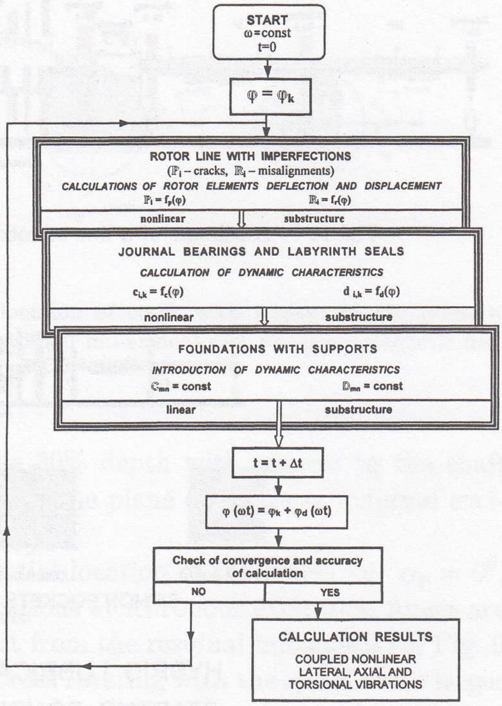


Figure 6. Algorithm of nonlinear calculation.

clutches and imperfections and foundation constitute a very difficult task. Arises here a equation, how to combine all these substructures into one coherent mathematical system? In the presented model the connections procedure is performed by means of stiffness and damping coefficients of journal bearing (which are highly non–linear) and dynamic influence coefficient matrix of foundation – Fig. 5. As a result we obtain the general equation of motion for the entire system – Fig. 5 which is nonlinear too, because the global matrices of damping and stiffness are functions of generalized motion parameters q, $\dot{q}$ and time t. How to solve such set of equations? The only possible way is the iterative procedure, where in each time step the global matrices of damping D and stiffness K are modified so as to sufficient convergence and accuracy of calculation is achieved.

3 Investigations on the basis of a large power engineering machine

Let the topic of our analysis be a turboset of 200 MW power schematically presented in Fig. 7. This machine has seven 2 – lobe bearings with 1÷4 siphon

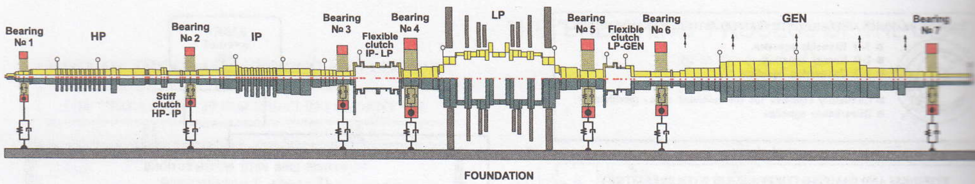


Figure 7. A scheme of a line of rotors in a turboset of 200 MW power.

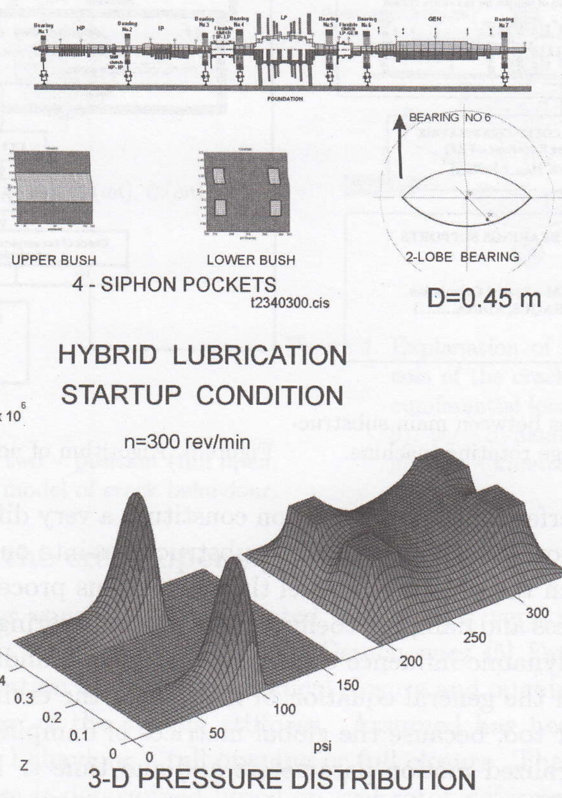


Figure 8. Examples of hybrid lubrication for start up conditions calculated for bearing no 6 of accepted object. Analysed case: hydrostatic pressure in lower bush is greater than hydrodynamic one (too large siphon pressure).

pockets. The selected examples of hybrid lubrication in bearing no 6 calculated for start up condition is presented in Fig. 8. Let's pass to the main subject under discussion. Let's assume that in the middle part of the generator (hence at the location of biggest kinetostatic deflections) the crack in the shaft P_1 occurred.

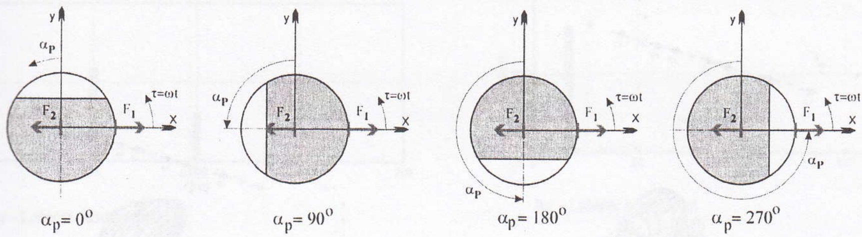


Figure 9. Calculated cases of circumferential location of crack with respect to the plane of action of excitation forces F_1 (from residual imbalance) and F_2 (from magnetic field in generator).

Let's consider the case of a crack of the 30% depth with respect to the shaft diameter arbitrarily located with respect to the plane of action of external excitation forces.

Let's specify the cases of circumferential location of the crack, i.e. $\alpha_p = 0°$, 90°, 180° and 270° – Fig. 9. In those locations synchronous excitation forces are acting there on the turboset, which result from the residual imbalance (in Fig. 9 denoted as F_1) as well as asynchronous forces rotating with the speed twice larger than the nominal speed excited by the influence of the magnetic field in the generator (in Fig. 9 denoted by F_2).

Interesting here will be the answer to a question whether the rotor crack and its circumferential location will have a significant influence on the dynamical state of the object and whether such type of rotor imperfection will be capable of inducing coupled forms of vibrations of such a large object?

The results of calculations in the form of spatial trajectories of dynamic displacements of selected nodes along the line of rotors have been presented in Fig. 10÷12.

Analysing the above results we can conclude, that in the case of a large power engineering machine, the crack in the generator's rotor and its circumferential location influence significantly the composition of system vibrations.

Some interesting information is supplied from the analysis of amplitude and phase spectra as well as analysis of coupled forms of vibrations induced by the crack in the generator.

In Fig. 13÷15 presented are amplitude and phase spectra in the middle of the generator, calculated for the base case without cracks, and the coupled forms of vibrations calculated for the generator's crack $P_1 = 30\%$ with the circumferential location of crack $\alpha_p = 0°$.

Analysing the obtained results it can be concluded that in the case of a large power engineering object the composition of the dynamical state, induced by the

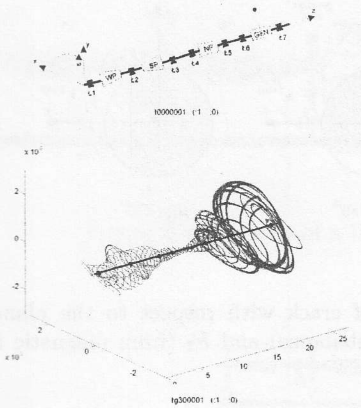


Figure 10. Trajectories of dynamical displacements of the line of rotors for the base case (without cracks). Kinetostatic deflection of the rotor and thermal displacements of supports have been neglected (coordinates in [m]).

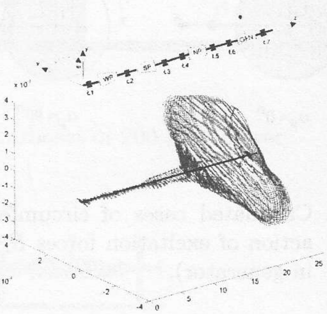


Figure 11. Spatial trajectories of vibrations of the line of rotors calculated for the generator's cracked rotor $P_1 = 30\%$ and circumferential location of crack $\alpha_p = 0°$ (coordinates in [m]).

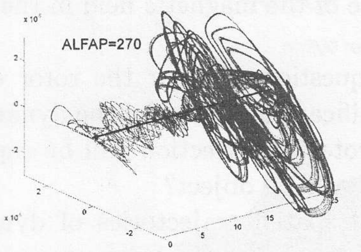


Figure 12. Spatial trajectories of vibrations of the line of rotors calculated for the generator's cracked rotor $P_1 = 30\%$ and circumferential location of crack $\alpha_p = 270°$ (coordinates in [m]).

crack is significantly more complex and hence more difficult in interpretation. Neglecting the case for $\alpha_p = 0°$ (the worst circumferential case), for which the system passed the threshold of stable operation, then even for the remaining cases of the crack location the identification of the crack by means of amplitude and phase spectra is not unanimous. Some information, however, is provided by the

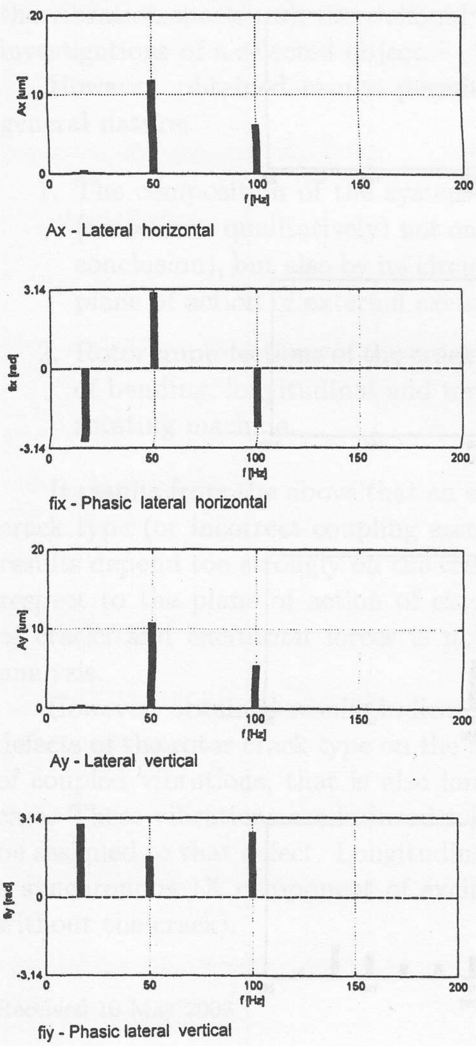


Figure 13. Components of amplitude spectra Ax, Ay and phase spectra fix and fiy calculated in the middle of the generator for the base case without cracks.

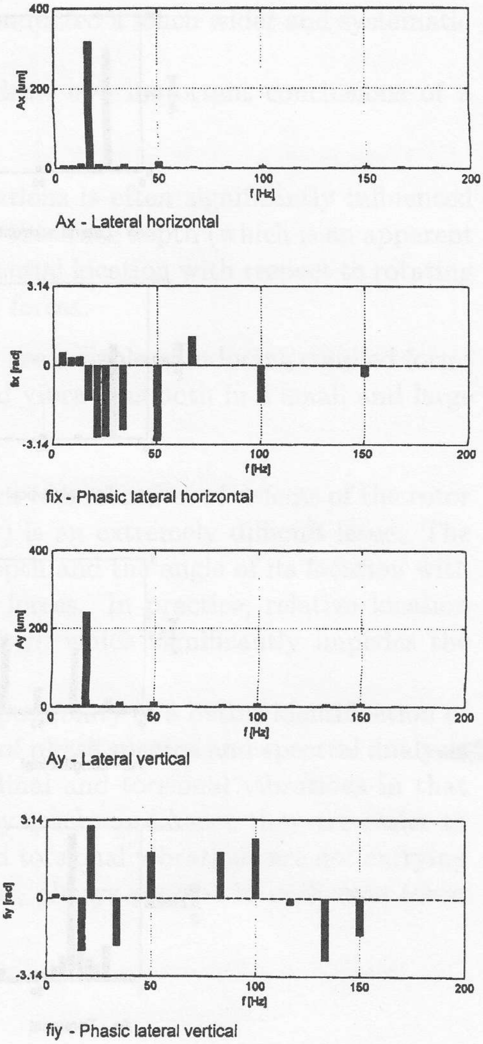


Figure 14. Components of amplitude spectra Ax, Ay and phase spectra fix and fiy calculated at the location of the generator's rotor crack ($P_1 = 30\%$) for the circumferential crack location $\alpha_p = 0°$.

coupled vibration spectra. The longitudinal vibrations generated by the crack can be of the same order as the lateral vibrations (neglecting the case of $\alpha_p = 0°$) even in locations somewhat distant to the crack. Characteristic are high values of higher harmonics of torsional vibration spectra (2X and 3X).

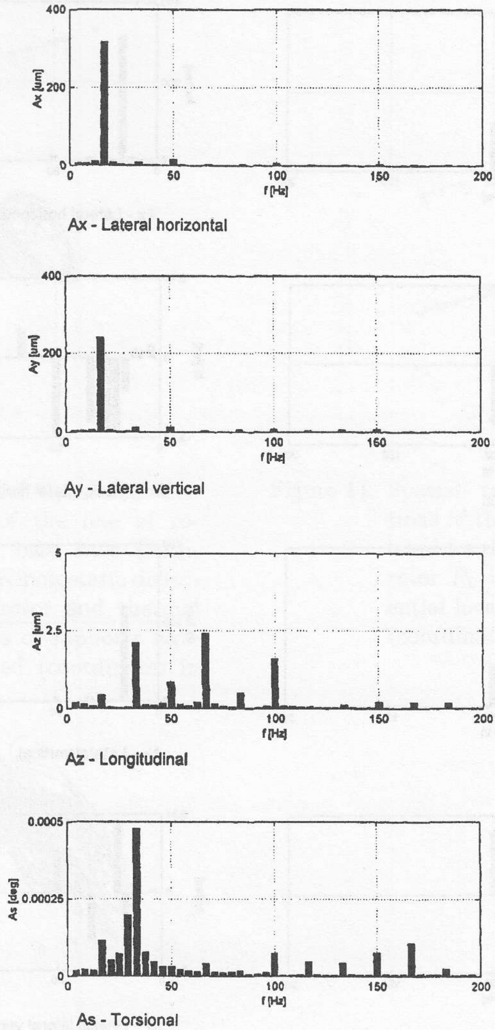


Figure 15. Coupled spectra of lateral, longitudinal and torsional vibrations induced by the crack in the generator. Calculations for $P_1 = 30\%$ and $\alpha_p = 0°$.

4 Concluding remarks

Presented in that section examples of calculations, referring to a large power engineering machine, cannot be regarded as a complete information source for the development of specific diagnostic relations.

It even was not the objective of presented considerations. For construction of reliable relations of the type defect (for example crack) – symptom (changes in the vibration spectrum), there should be conducted a much wider and systematic investigations of a selected object.

However, obtained results permit to draw two important conclusions of a general nature:

1. The composition of the system vibrations is often significantly influenced (sometimes qualitatively) not only by the crack depth (which is an apparent conclusion), but also by its circumferential location with respect to rotating plane of action of external excitation forces.

2. Rotor imperfections of the crack type are capable of inducing coupled forms of bending, longitudinal and torsional vibrations both in a small and large rotating machine.

It results from the above that an efficient identification of defects of the rotor crack type (or incorrect coupling assembly) is an extremely difficult issue. The results depend too strongly on the crack depth and the angle of its location with respect to the plane of action of external forces. In practice, relative location of cracks and excitation forces is not known, which significantly impedes the analysis.

However, obtained results indicate the possibility of a better identification of defects of the rotor crack type on the basis of phase spectra and spectral analysis of coupled vibrations, that is also longitudinal and torsional vibrations in that case. These vibrations are induced solely by cracks and hence they are easier to be assigned to that defect. Longitudinal and torsional vibrations are not carrying *a synchronous 1X component of excitations, always present in such case (even without the crack).*

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