POWER TURBO-SETS VIBRATION STATE DIAGNOSING

Mykola SHULZHENKO[1], Leonid METELEV[1], Yuriy EFREMOV[1], Slawomir BANASZEK[2]

[1]A.N. Podgorny Institute for Mechanical Engineering Problems NAS of Ukraine
Pozharsky Str. 2/10, UA-61046 Kharkov, Ukraine
fax +38 0572 944635, E-mail: shulzh@ipmach.kharkov.ua
[2]Szewalski Institute of Fluid Flow Machinery PAN
Fiszera 14, Gdańsk 80-231, Poland
phone +48 58 346 08 81, fax +48 58 341 61 44, E-mail: tolek@imp.gda.pl

The paper presents automatic system of turbo-sets vibration state analysis and diagnosing. The turbo-set shaft line is considered as main object for diagnosing. The system is equipped with sufficient sensors to measure vibratory signals from the rotor line and non-vibratory data. Computer system enables efficient real-time analysis and post-processing of measured values. Methods of operation states assesment are applied to provide analysis during the rotor turning, run-up/run-down, idle running and loading conditions. A special attention is payed to investigations of torsional vibrations as indicator of some defects. An example of computer simulation of shaft crack is presented showing their usefulness. A technique for the rotors torsional strains check and assessment at the turbo-set shaft line rotation is introduced into the presented system to determine fatigue phenomena.

Keywords: diagnostics, vibration state, technique and systems, power machinery, turbo-set.

DIAGNOZOWANIE STANU DYNAMICZNEGO TURBOZESPOŁÓW ENERGETYCZNYCH

Praca przedstawia automatyczny system analizy i diagnostyki stanu dynamicznego turbozespołów energetycznych. Linia wirników została przyjęta jaki główny obiekt diagnozowania. System jest wyposażony w odpowiednie czujniki mierzące sygnały drganiowe i inne wielkości. Komputerowa obróbka sygnałów umożliwia efektywną analizę w czasie rzeczywistym oraz postprocessing zmierzonych wartości. Zastosowane metody dają możliwość analizy w różnych stanach operacyjnych maszyny, jak obracanie wirnika, rozbieg/wybieg, bieg jałowy, bieg pod obciążeniem. Zwrócono specjalną uwagę na badania drgań skrętnych jako wyróżnika diagnostycznego niektórych defektów. Podany przykład badań symulacyjnych pęknięcia wirnika pokazuje ich użyteczność. Zaprezentowano technikę wyznaczania naprężeń skrętnych w celu określania zjawisk zmęczeniowych.

Słowa kluczowe: diagnostyka, stan dynamiczny, technologia i systemy, maszyny energetyczne, turbozespół.

1. INTRODUCTION

Nowadays the importance of problems regarding increase of steam power plant's and thermo-electric plant's turbo-set safety operation becomes very important. This is a result of the turbo-sets technical state when most of them completed their fleet life, their operational conditions at transient and peak modes in the conditions of power market which are nor foreseen by the design and realized, as a rule, with deviations from normal technological processes. Because of additional low-cyclic and high-cyclic loadings such modes result in accelerated accumulation of fatigue damages and appearance of latent defects, first of all in elements of the shaft line. Methods and techniques for the turbo-sets technical state analysis and diagnosing by vibration parameters are important for estimation of damages initiation and growth.

Collected numerical and experimental data show that appearance of the dangerous vibration is possible if the following take place: a crack of essential size in the rotor where the opening angle more than 0.25π or its depth equals 15-20% of the rotor diameter; damages in shaft couplings of rotors and non-centering of supports; in stability of the shaft line rotation and progress of self-exited oscillations; essential unbalance (mechanical or thermal) of shaft line rotating elements; breaking off or damage of rotor elements or its catching the stator; defects of elements and parts of the generator electromagnetic system [1-6].

Investigations [2-5] on the determination of information and diagnosing parameters and indicators of dangerous damages show that they influence on the change of the general vibration level of supports and rotor necks as well as on its separate harmonic and spectral components. Initiation and growth of most of dangerous defects and damages results first of all in specific changes of parameters of rotational harmonic component, in appearance and progress of parameters of the double rotational harmonic component and in the specific enrichment of vibration spectrum by high frequency (HF) components of 3, 4 and higher order at the progress of defects and damages till the dangerous level. Here, the specific enrichment of the spectrum by low frequency (LF) oscillations of the shaft line is observed at presence of elements defects and damages and under the influence of forces of hydrogasodynamical nature in the turbine flow passage and in the oil layer of plain bearings.

For example, if the amplitude and phase of the rotational component do not depend on operation modes (load, stem temperature, etc.) and the amplitude excesses essentially the amplitudes of other spectrum components, the possible reason is mechanical unbalance of shaft line rotational elements. In cases of their dependence on operation modes one can speak about a symptom of the thermal unbalance. Monotonic variation of rotational component parameters in time can indicate the crack growth, and stepwise one – breaking off or failure of turbo-set shaft line elements.

Presence of double rotational frequency harmonic component can be induced by the rotor double stiffness (because of presence and growth of the crack in the rotor or damage in shaft couplings), design peculiarities of generators bipolar rotor and effect of the electromagnetic field between rotor and stator.

High frequency vibration components arise, as a rule, simultaneously with rotational and double rotational components and can be indicators of the bending of rotor joints, rotor crack, rotor catching the stator, etc.

For determination of defects and damages of the diagnosed turbo-set it is necessary to form information-diagnosing arrays and trends of parameters taking into account operation modes and factors at continuous automotive analysis of these parameters. This is realized by the automotive system of turbo-sets vibration state analysis and diagnosing developed at the Institute for Mechanical Engineering Problems NASU. Experimental versions of the system operate on two power units 300 MW of Zaporozhje Steam Power Plant, at Kiev Thermo-Electric Plant-5 and Kharkov Thermo-Electric Plant-5. The block diagram of the system is presented in Fig. 1. To develop the system the experience of long-term investigations is used. Among them one can describe the following results.

Methods and developed equipment for contact and contact-less measurement of turbo-sets rotors and elements oscillations with utilization of eddy current converters with frequency modulated output signal which meet the requirements of Standards by their technical data. They have increased error probability for electromagnetic fields influence and do not require adjustment on the metal of object, type and length of the cable of the communication line and not sensitive for side metal that describes their advantages in the comparison with analogous devices of recognized companies. This has permitted to increase error probability and truth of obtained initial information on vibration parameters [7, 8, 9].

2. ASSESSMENT OF VIBRATION PARAMETERS

A method of assessment of the operation modes influence on the vibration parameters variation is described in [10, 11]. A method of identification of turbo-sets shaft line models calculating parameters by experimental amplitude-frequency characteristics of the diagnosing turbo-set which permits to build adequate calculating model of the system for numerical determination of its defects influence on the turbo-set's vibration parameters is described in [12].

Computer technologies of operation of the system of automotive analysis and diagnosing of vibration state in real time and post-analysis on the parameters correspondence with norms for vibrations at all operation modes to determine reasons of their deviations [13, 14].

Methods of the analysis and assessment of turbo-sets vibration state and reasons of its variation by amplitudes and phases of harmonic components of vibrations of rotors and shaft line bearings supports as well as by current spectral order characteristics and their variations in time (trends) [15].

A technique for the rotors torsional strains check and assessment at the turbo-set shaft line rotation which permits to determine fatigue phenomena [16, 17].

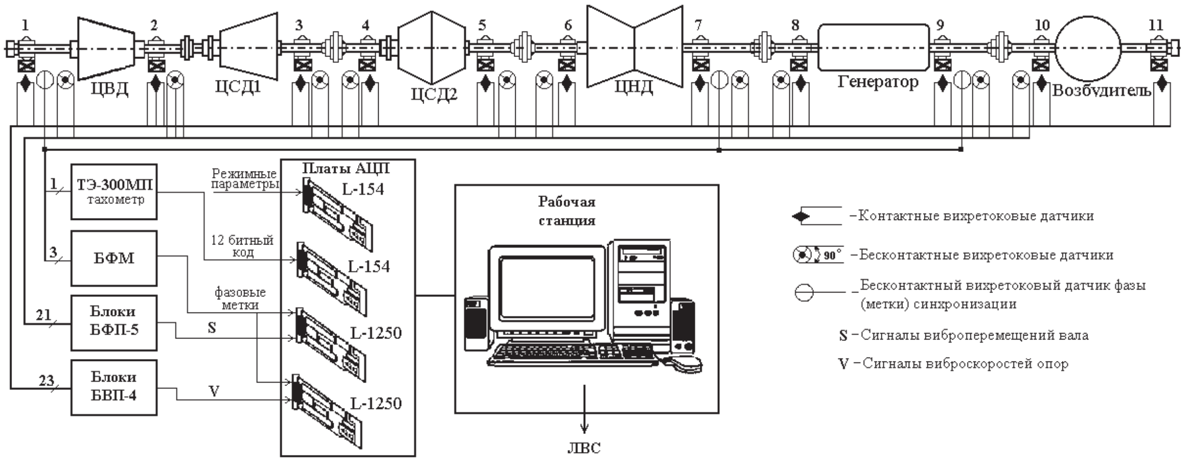


Fig. 1. The block diagram of the system.

Initial vibration data are signal of instantaneous values of relative displacements (gap variations) of all shaft line rotors necks in bearings recesses and vibrations of all bearings supports at least in two orthogonal directions. Collection of vibration signals is carried out simultaneously (in parallel by all channels), continuously and synchronously with the shaft line rotation. Synchronization of the signals receiving and treatment, check of rotation frequency and torsional strains are carried out by signals from marks on shaft line rotors. Necessary thermo-mechanical and non-vibration parameters are formed by signals from converters and measuring devices of the automatic process control system. The turbo-set shaft line is considered as main object for diagnosing because most probable defects and faults mentioned above are connected with its vibration state.

A technique of the turbo-set vibration state diagnosing at start-up procedures (rotor turning, speed-up–acceleration, no-load and loading) is intended for identification of thermal and mechanical rotor deflection, misalignment of rotors and supports as well as defects of mounting and repair.

Instantaneous values of necks displacements $S_{xi}(t)$ and $S_{yi}(t)$ are measured and registered continuously on the shaft line in each bearing at two orthogonal radial directions x and y, and maximal moduli of necks displacements are calculated

$$S_{i\max} = \max\sqrt{S_{xi}^2(t) + S_{yi}^2(t)} \qquad (1)$$

as well as their direction $\psi_{i\max}$. Orbital movements of all rotor necks in bearings and numerical values of $S_{i\max}$ and $\psi_{i\max}$ are displayed on the monitor. Following the experience of investigations the tolerated value of $S_{i\max}$ after the warm-up should not exceed 20 µm.

If after satisfied warm-up $S_{i\max}$ does not decrease till tolerated level, the reason can be non-removed defects of mechanical nature of mounting or repair. It can be seen from the mnemoline of the shaft line "bending" obtained on the monitor by $S_{i\max}$ of necks taking into account directions $\psi_{i\max}$ of their displacements in bearings (see Fig. 2).

A peculiarity of vibration state analysis and diagnosing at speed-up and acceleration modes is the following. They are carried out continuously in real time at variable rotational speed. Developed hardware, software and methods realize necessary speed of signals receiving and treatment as well as output of information in required amount on variation of controlled vibration parameters. This permits to receive and store frequency characteristics at speed-up and acceleration by general level of rotors and supports vibrations as well as by amplitude-phase-frequency characteristics of their harmonic components. Threshold levels of shaft and supports vibrations at speed-up are set by actual mean-statistic acceleration characteristics.

Rotors relative vibration displacement check on critical frequencies permits to prevent crushing of seals in the turbine flow passage and conserve present efficiency and reliability of the turbo-set operation. At the excess of the threshold level at least of one of rotors it is useful to interrupt the speed-up till clearing and removal of corresponding reasons. Besides, parameters of spectral and harmonic characteristics of turbo-set shaft line rotors and supports vibrations are used. To do it ordinal spectra (by numbers of harmonic components) are introduced which contain data on vibration amplitude and phase as functions of harmonic number which permits to check the variation of harmonics amplitudes and phases in the process of the turbo-set (see Fig. 3).

Operative assessment of the turbo-set state at operation modes (after loading and connection to the power network) is guaranteed by the way of continuous check of rotors and bearing supports vibration parameters on their correspondence with their normal levels.

At these modes information-diagnosing arrays and trends by adopted list of vibration parameters are formed. To do it samples of values of rotors relative vibration displacements and shaft line supports vibration speeds are synchronously formed each second with discreteness of 80 μs and values of vibration displacement span as well as root-mean-square value of vibration speed are calculated. By each of these parameters on the interval of 20 s arrays updated each second are formed. After averaging for 20 s their values are compared with normal values. The excess of the threshold level is indicated by the change of the color of the bar with simultaneous audible signal (see Fig. 4).

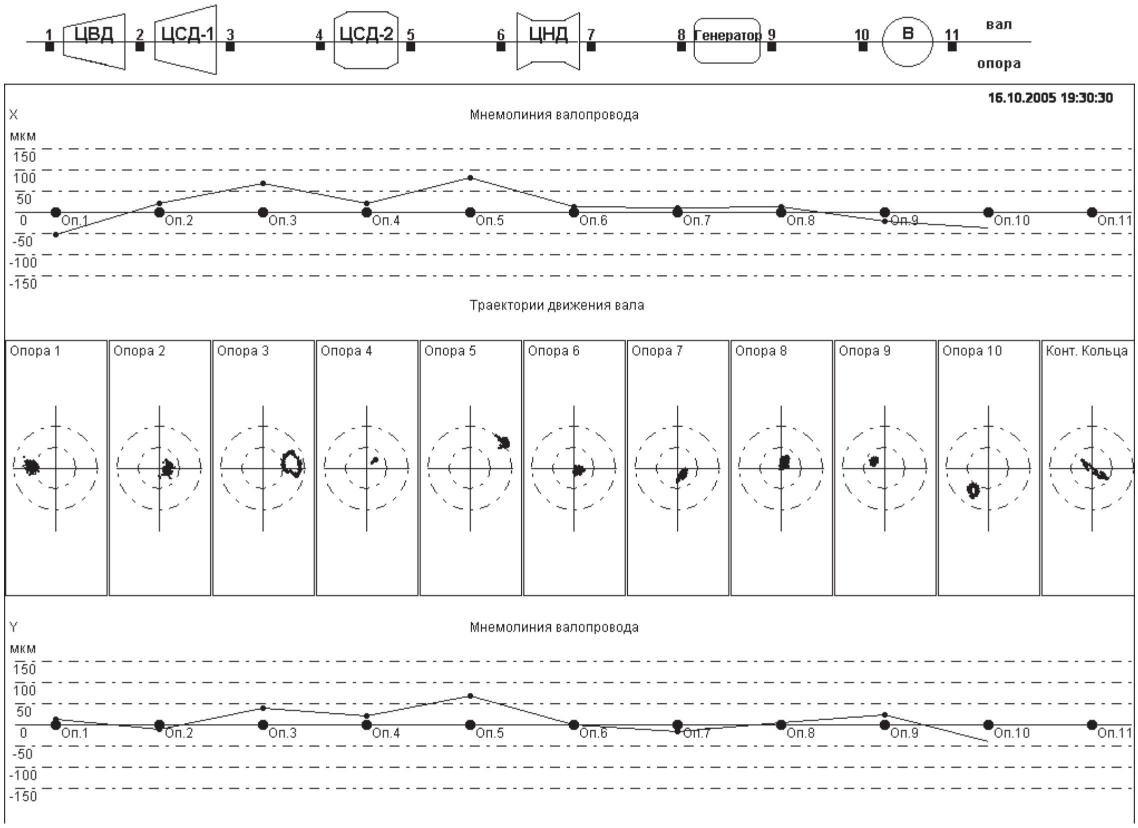


Fig. 2. Shaft line mnemoline.

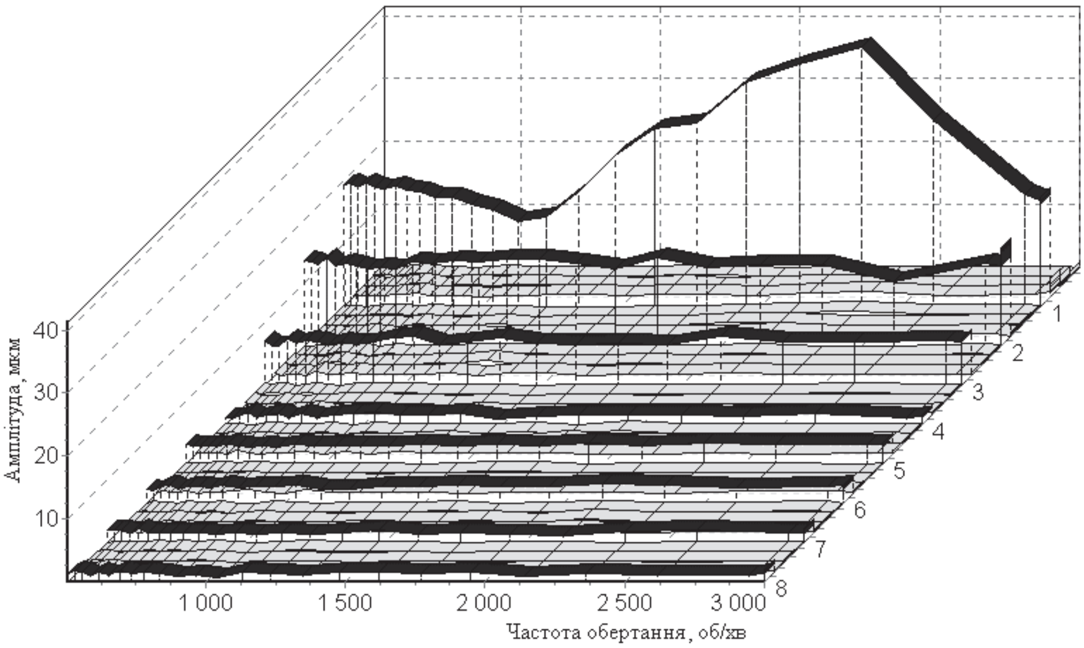


Fig. 3. Ordinal spectra of acceleration by harmonic components.

These arrays are initial to determine unexpected (stepwise) vibration variation because of breaking off or failure of shaft line rotating parts as well as sub-harmonic and super-harmonic components.

For the turbo-set vibration state diagnosing statistical arrays and trends of vibration parameters are formed with period of averaging of 20 s, 90 s, 1 h and 24 h. Using values averaged for 24 h trends are form for a month, a quarter, a season, a year, an inter-maintenance period, etc.

Besides, the system archives informational-diagnosing arrays and trends of the following parameters:

– amplitudes and phases of rotors vibration displacement rotation frequency harmonic component and bearing supports vibration speed, as well as their ratios by orthogonal directions of the same rotor necks and bearing supports;
– amplitudes and phases of double rotation frequency harmonic component of amplitude values for the 3-rd, 4-th and 5-th harmonics, as well as their ratios;
– amplitude of the sub-harmonic component with half rotation frequency of rotors and supports vibration displacement and vibration frequency;
– spectra of LF rotors and bearing supports vibrations in the frequency interval of (5– $F_0/2$) Hz (where F_0 – rotation frequency);
– spectra of HF rotors and bearing supports vibrations in the frequency interval higher $2F_0$.

By data of these arrays the spectral analysis of vibration signal with step of 3 Hz is carried out, besides the forming of the trajectory of rotors necks movement in plain bearings as well as phase portrait of the supports vibration speed is carried out (see Fig. 5).

The mentioned arrays of parameters are formed depending on rotation frequency, loading variation (active and reactive power and current of the generator rotor), steam parameters (pressure and temperature in the chamber of the regulating stage), temperature state of turbo-set elements including oil at input and output of supporting and thrust bearings and metal of the high-pressure cylinder (top-bottom), vacuum in the condenser, rotors and cylinders thermal expansions and some other parameters.

It permits to determine in time fast and slow vibration parameters variations taking into account influence of technological and operation factors and identify appearance of dangerous faults.

For the numerical analysis of defects influence on shaft line vibration parameters a dynamical model of the shaft line built by the way of the approach of results of vibration characteristics calculation to data of experimental investigations is used [12] (see Fig. 6).

Here the inverse problem of theory of oscillations is solved by searching of partial solutions. Nowadays special attention is paid to investigations of large turbo-sets torsional vibrations. Necessity of their investigations is determined by a number of accidents because of fatigue damages of rotors [4, 18, 19].

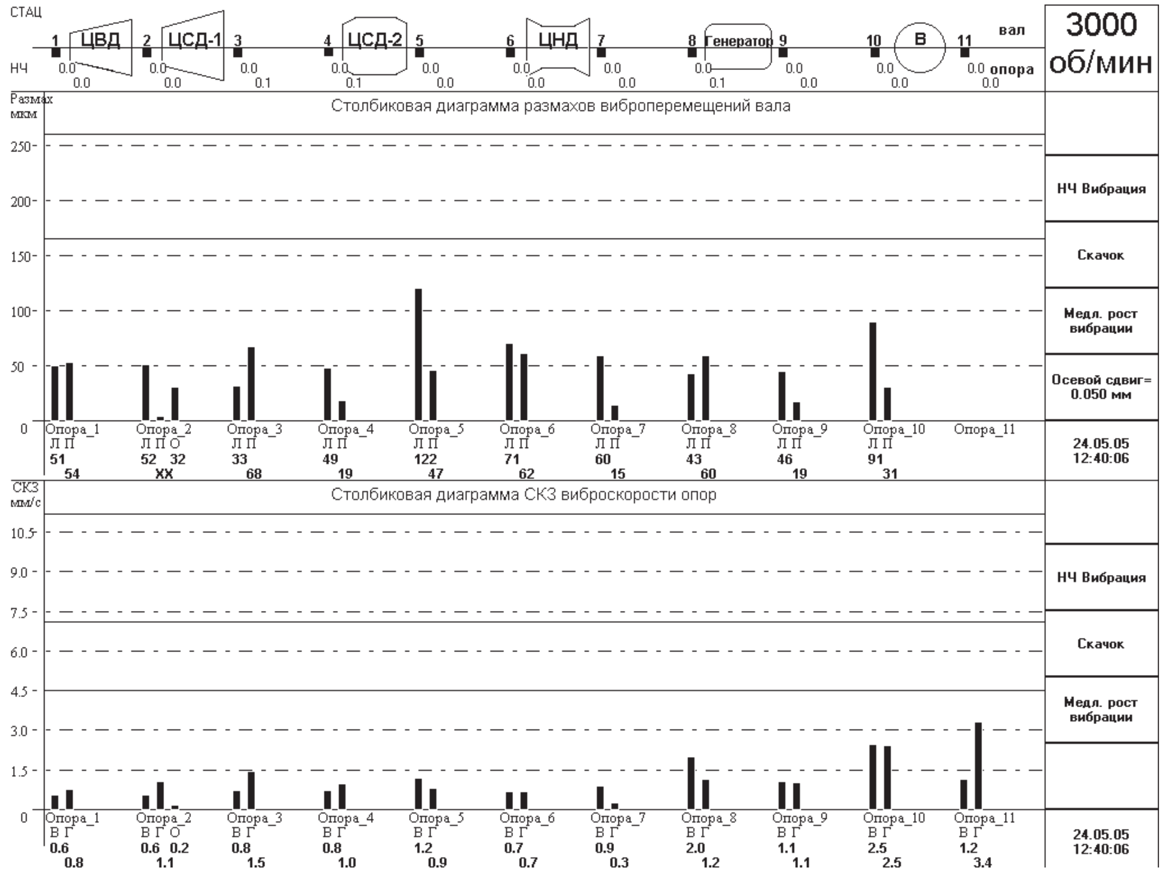


Fig. 4. Bar diagram of turbo-set vibration parameters.

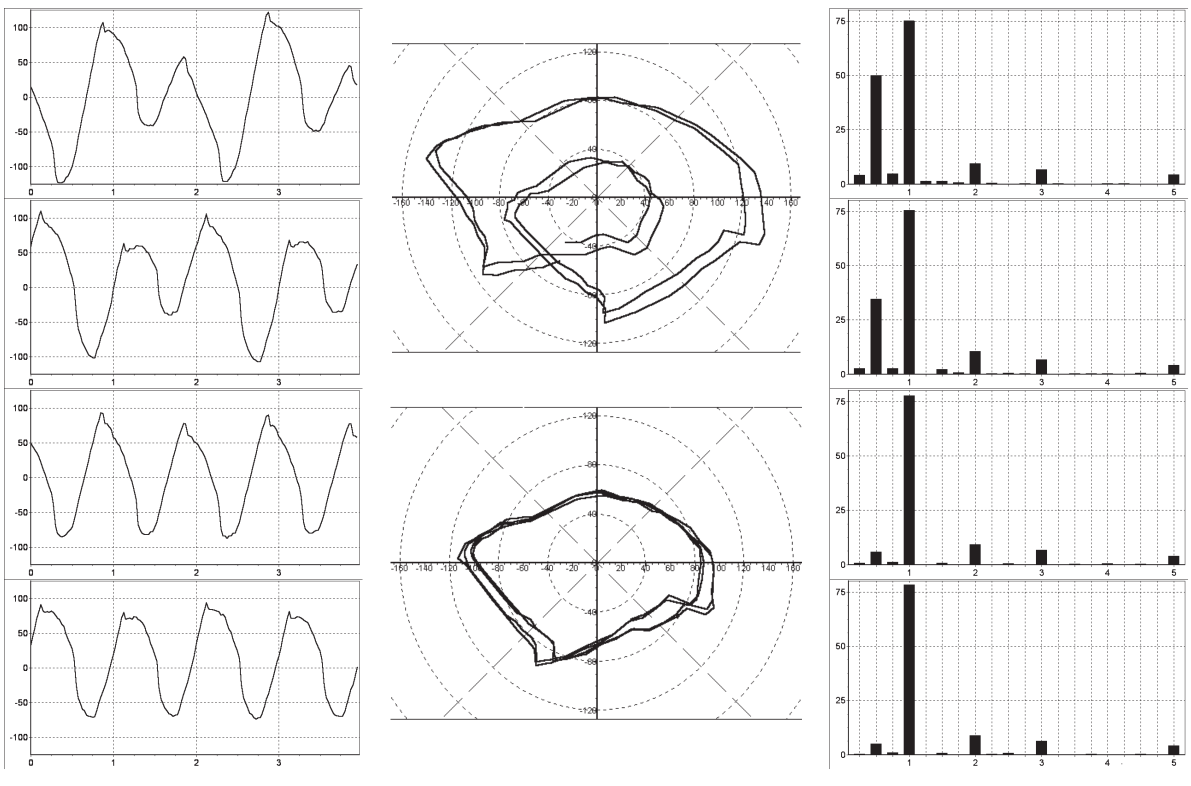
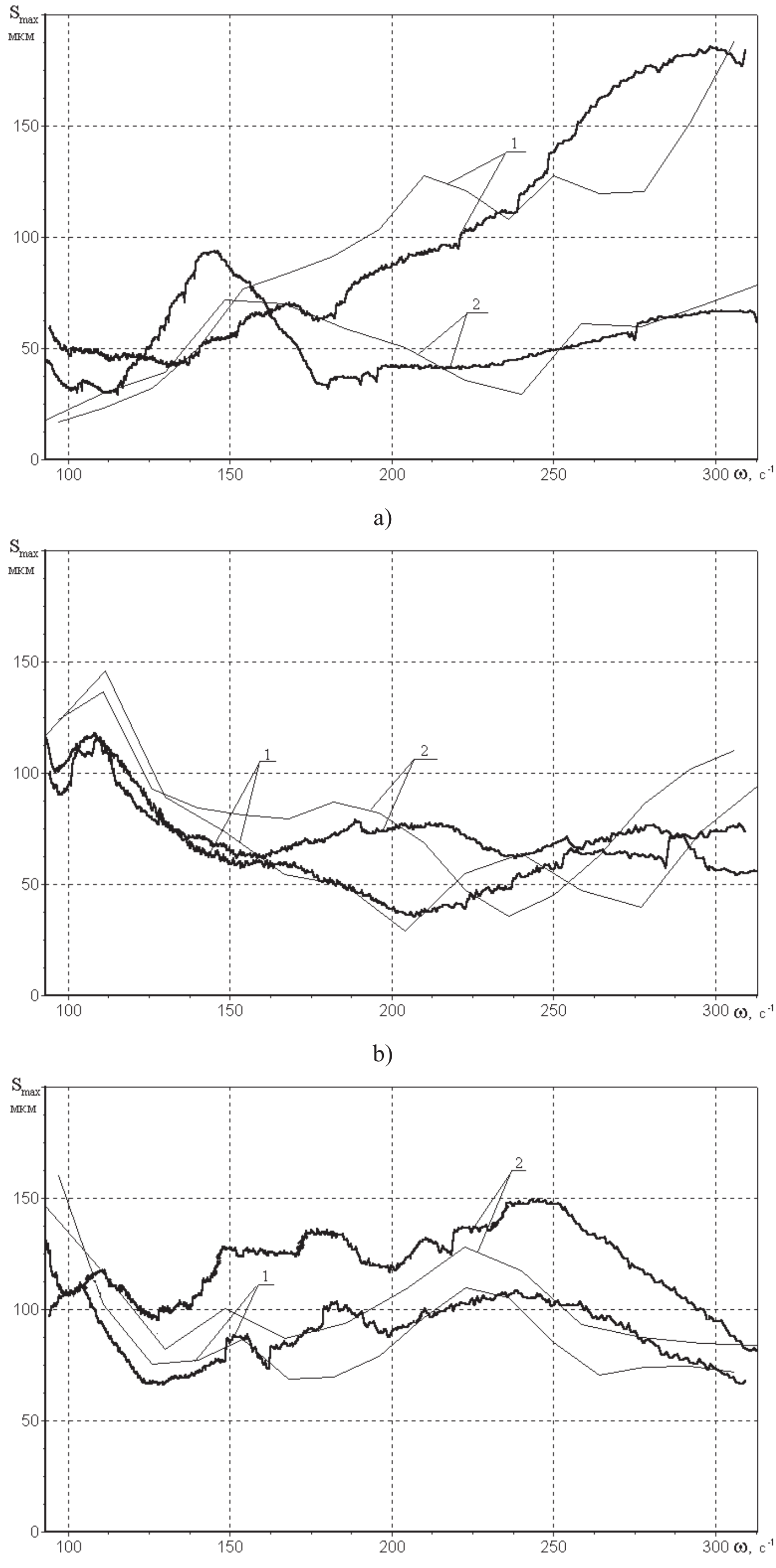


Fig. 5. Shaft line vibration characteristics.



c)

Fig. 6. Amplitude-frequency characteristics at acceleration:
a) support 1; b) support 4; c) support 7;
▬ – experimental values; — – calculated values:
1 – before repair, 2 – after repair

3. TORSIONAL VIBRATIONS AS DIAGNOSTIC INDICATOR OF SHAFT CRACK – COMPUTER SIMULATION

Investigations carried out at the Institute of Fluid-Flow Machinery PASci. (IFFM) show that by using methods of torsional vibrations spectral analysis it is possible to find initiation and growth of a cross crack [20]. They were based on simulation calculations. The NLDW computer code is utilized to perform the simulations. It is a part of computer code environment MESWIR that allows to model multifarious phenomena in rotating rotors founded on slide bearings. It bases on non linear methods to model interactions in the rotating machine between shaft, bearings and supporting structure. The core of this environment is strongly non-linear, diathermal model of slide bearing. This model encompasses Reynolds equation, energy equations, equations describing heat transfer in the lubricating gap, etc., giving as an output non-eliptical trajectories of displacements of nodes of the model and their spectra. A shaft is modeled with typical Timoshenko beam elements with 6 DOF's per node. It allows to take into account transverse, axial and torsional loads of the shaft, to obtain displacements in those directions and analyze couplings between them.

Crack is modeled with Knott model. It is typical two-state model of the slot. 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 [20-22].

$$K_e = T_t\,(L + L_d)^{-1}\,T \qquad (2)$$

where the index ***t*** denotes the matrix transposition and ***-1*** means matrix inversion.

The matrix of additional flexibilities $\boldsymbol{L_d}$ is calculated using the form of elastic deflection energy of the beam element, which emerges from the transverse crack [22]. It can be proven that it can be written in a form, where non-diagonal elements describe couplings between displacements in different directions:

$$\boldsymbol{L_d} = \begin{bmatrix} c_{11} & 0 & 0 & 0 & 0 & c_{16} \\ & c_{22} & 0 & c_{24} & c_{25} & 0 \\ & & c_{33} & 0 & 0 & c_{36} \\ & & & c_{44} & c_{45} & 0 \\ s & y & m & & c_{55} & 0 \\ & & & & & c_{66} \end{bmatrix} \qquad (3)$$

Crack depth is described by the crack coefficient W_P (4). 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.

$$W_P = \frac{a}{D} \qquad (4)$$

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 Fig. 7 [20].

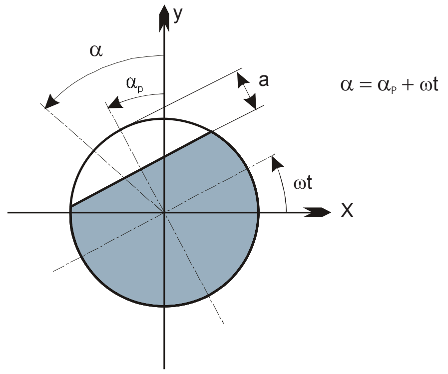


Fig. 7. Beam element with the crack turned with respect to the y axis by an arbitrary angle αp determining the initial location of the crack in the x-y plane.

This 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 .

As an example we can see the investigations of two different (separate) crack locations in the rotor of 200 MW turbogenerator (see fig. 8).

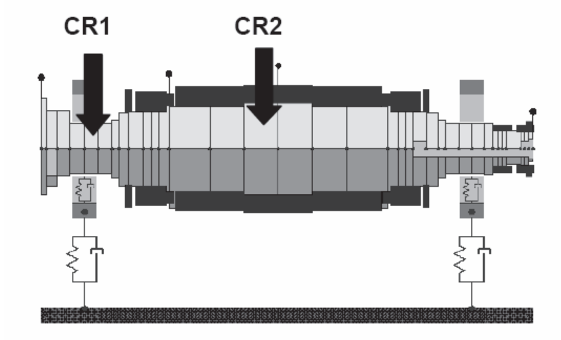


Fig. 8. The assumed crack positions in rotor of 200 MW turbo-set (scheme of generator - section between bearings No. 6 and 7).

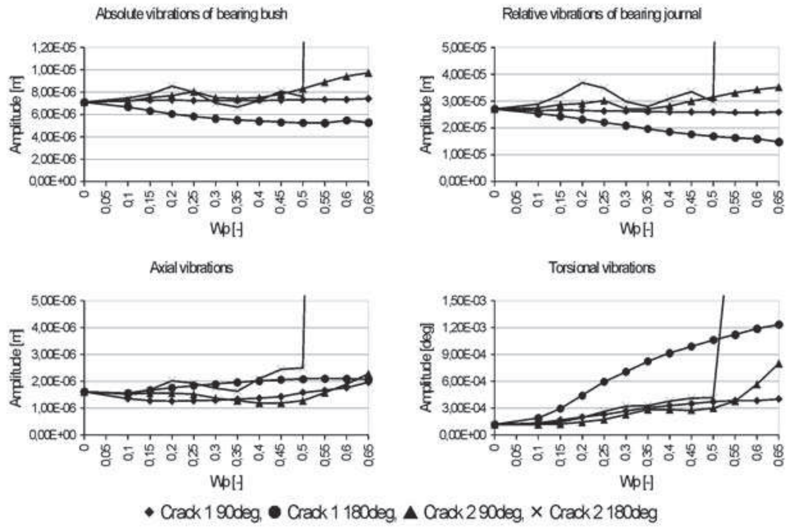


Fig. 9. The crack propagation influence on the machine dynamic state - amplitudes obtained for bearing No. 6 in selected cases

The analysis of fig. 9 leads to interesting conclusions. In some cases we can find some local resonances of bending vibrations for W_P=0.2-0.25. In other cases, bending vibrations doesn't exhibit such a phenomenon. They may even decrease. On the other hand, torsional vibration are the most sensitive to the crack. Their increase is the largest – in the case of CR2, $\alpha_P=180^O$ they increase over 30 times (not shown on the drowing because of the scale).

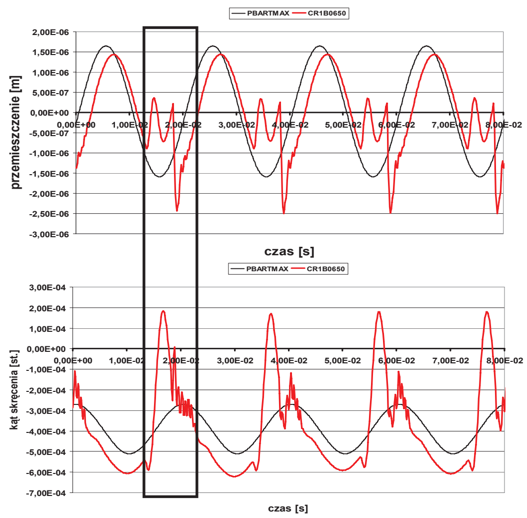


Fig. 10. The frame points the unsettled movement in axial (upper diagram) and torsional directions (lower diagrm) vibrations in bearing No.6 testifying coupling causd by crack. PBARTMAX – machine without defect, CR1B0650 – crack case No.1 (CR1), $\alpha_P=90^O$, W_P=0.65

It also have to be pointed the crack causes coupled vibrations. Fig. 10 shows time histories of axial and torsional vibrations of bearing No.6 (journal vibrations). Correlation between axial and torsional vibrations (fig. 9) testifies some kind of coupling and identifies crack presence rather clearly.

4. CLOSING REMARKS

At the Institute for Mechanical Engineering Problems NASU a technique of determination of residual non-elastic deformations and residual displacements in shaft couplings by marks distributed along the shaft line at torsional strains is developed and validated experimentally. Experimental investigations have shown that for the turbo-set T250/300-240 the twisting angle changes proportionally the power variation on the turbo-set shaft with the ratio of 10 angular minutes per 100 MW.

Finally, investigations carried out, informational-diagnosing data collected during a number of years of turbo-sets vibration state analysis and diagnosing system operation and a technique of dangerous damages determination permit to obtain initial data for operative warning on dangerous faults propagation and for expert system of increased vibrations reasons determination that represents the next step of the described diagnosing systems development.

REFERENCES

[1] Б. Т. Рунов.: *Исследование и устранение вибрации паровых турбоагрегатов*– М.: Энергоиздат, 1982.– 351 с.

[2] Н. Г. Шульженко, В. П. Билетченко, А. М. Ганжа: *Влияние расцентровки опоры на колебания и устойчивость валопровода* Вестн. ХГПУ. Динамика и прочность машин. Харьков, 1999. Вып. 53. С. 70–71.

[3] Н. Г. Шульженко, А. М. Ганжа, Ю. Г. Ефремов: *Спектральные характеристики вибропереме-щений валопровода при просадке опоры и раскрытии полумуфт* Пробл. машиностроения. 2000. 3, № 1–2. С. 77–82.

[4] Н. Г. Шульженко, Г. Б. Овчарова: *Влияние излома упругой оси ротора с поперечной трещиной на его вибрационные характеристики* Пробл. прочности. 1997. № 4. С. 82–89.

[5] Н. Г. Шульженко, Ю. С.: Воробьев *Численный анализ колебаний системы турбоагрегат-фундамент-основание* Киев: Наук. думка, 1991.– 232 с.

[6] А. С. Гольдин *Вибрация роторных машин* М.: Машиностроение, 2000.– 344 с.

[7] Н. Г. Шульженко, В. П. Билетченко, Л. Д. Метелев, В. И. Цибулько, Ю. Г. Ефремов, А. А. Беспрозванный, Н. И. Вова *Аппаратное обеспечение системы непрерывного вибромониторинга роторных агрегатов* Энергетика и электрификация. 2000. № 7(204). С. 35–38.

[8] Н. Г. Шульженко, Л. Д. Метелев, В. И. Цыбулько, А. И. Чугреев, Ю. Н. Гуров, Ю. Г. Ефремов: *Помехоустойчивые измерители вибрации. Вибрация машин: измерение, снижение защита*: Материалы 2-й Междунар. науч.-техн. конф. Донецк: ДонНТУ, 2004. С. 35–39.

[9] Н. Г. Шульженко, Л. Д. Метелев, В. И. Цыбулько, Ю. Г. Ефремов: *Помехоусто-йчивые измерители вибрации и их применение в динамике машин. Вибрации в технике и технологиях*. 2005. № 2(40). С. 104–107.

[10] N. G. Shulzhenko and Yu. G. Efremov: *Influence of operational modes of a powerful turboset on its vibrational state*/ Transactions of the Institute of fluid-flow Machinery: Selected papers from the International Conference on Turbines of Large Output. Gdańsk, 2003. P. 147–153.

[11] Н. Г. Шульженко, Ю. Г. Ефремов: *Разработка методики автоматизированной оценки влияния режимных факторов на вибрационное состояние турбоагрегата* Пробл. прочности. 2004. № 1. С. 147–156.

[12] Н. Г. Шульженко, Ю. Г. Ефремов, А. М. Ганжа: *Идентификация стержневых моделей валопроводов по данным экспериментальных исследований* Пробл. машиностроения. 2002. № 4. С. 67–71.

[13] М. Г. Шульженко, Л. Д. Мєтєльов, Ю. Г. Єфремов, В. Й. Цибулько: *Технології діагностування вібраційного стану багатоопорних роторних агрегатів* Автоматизація виробничих процесів у машинобудуванні та приладобудуванні. – Львів: Нац. техн. ун-т "Львівська політехніка", 2006. № 40. С. 296–305.

[14] Н. Г. Шульженко, Л. Д. Метелев, Ю. Г. Ефремов, В. И. Цыбулько: *Обеспечение надежности и ресурса энергетических турбоагрегатов использованием системы диагностики вибраций и предупреждения нештатных ситуаций*/ Энергетика и электрификация. 2004. № 5. С. 41 – 43.

[15] Н. Г. Шульженко, Л. Д. Метелев, Ю. Г. Ефремов, В. И. Цыбулько: *Анализ и диагностирование вибрационного состояния мощных турбоагрегатов* Енергетика та електрифікація. 2006. № 11. С. 30–38.

[16] Н. Г. Шульженко, Л. Д. Метелев, В. И. Цыбулько, Ю. Г. Ефремов: *Методика определения параметров крутильных деформаций роторов турбоагрегатов* Датчики и системы. 2004. № 1. 30–31.

[17] М. Г. Шульженко, Л. Д. Мєтєльов, В. Й. Цибулько, Ю. Г: Єфремов *Оцінка крутильних деформацій роторів агрегатів* Вібрації в техниці та технологіях.– Вінниця. 2006. № 1(43). С. 149–151.

[18] Haas G.: *Großchäden durch Turbinen-oder Generator-läufer, enstanden in Bereich bis zur Schlenderdrehzahl* Der Maschinen Schaden. 1977. 50, № 6. P. 195–204.

[19] B. Grabowski, O. Mahrenholtz: *Theoretical and experimental investigating of shaft vibrations in Turbo machinery excited by cracks* Energia Electrica. 1982. 59, № 10. P. 507–514.

[20] Kicinski J.: *Rotor Dynamics* The IFFM Pub-lisher, Gdansk 2006.

[21] Knott J. F.: *Fundamentals of fracture mechanics*, Butterworths, London 1973.

[22] Ostachowicz W. M., Krawczuk M.: *Coupled torsional and bending vibrations of a rotor with an open crack.* Archive of Applied Mechanics, 62 (1992), pp.191-201.