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Modeling and diagnostics of the gear power transmissions

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1. Introduction

Rotating systems are the main mechanical units of power generating and technological machines. The integral part of modern machines with rotating systems introduces IT based condition monitoring, protection, failure diagnostics and expert systems for prediction of unexpected failures and based on vibration and technological parameters measurement. To increase technical condition monitoring and vibration severity evaluation accuracy in practice with the identification of vibration sources the modeling and simulation of rotor systems are inevitable processes [1 - 7].

Systematic machine vibration monitoring and failure diagnostics prevent from unexpected failures of machines. The most frequently occurring breakdown of rotor systems - failure of bearings and rotor-stator part rub [1]. Authors evaluate diagnostics methods for high speed rotors with rolling bearings [2]. The papers [8 - 10] concern vibration monitoring and failure diagnostics of low rotation speed, but high torque rotors and gear transmission rotors, for the identification of rolling bearings defects. The gears teeth meshing malfunctions are estimated through measurement of rotors radial and axial vibrations displacement in situ and modeling and simulations of designed results. The authors present new research data on gear transmission dynamics and diagnostics.

The heavy duty gear power transmission have been modeled and tested in situ to protect unexpected failures [11, 12].

This paper presents effectiveness of new designed diagnostic method for prediction of the unexpected failures of low rotation speed high torque gear transmission with antifriction bearings.

2. Objects of research

High mass horizontal axis cylinder machine kinematic scheme is presented in Fig. 1. The heavy loaded low speed gear transmission elements are pinion z_1 and gear z_2, four low rotation speed cylinders 1K, 2K, 1D, 2D and eight roller bearings: four on the left cylinder side 1K1, 1K2, 2K1, 2K2 and four on the right cylinder side 1D1, 1D2, 2D1, 2D2. The impacts generated by pinion z_1 teeth meshing are the main sources of vibration. The teeth module is 16 mm, $z_1 = 25$ and $z_2 = 176$ teeth. The rotational speed of the pinion z_1 is low ~38 rpm, rotation torque is high 8250 Nm and the speed of asynchronous electric motor rotor is 1500 rpm. All rotors rotate in double row spherical roller bearings SKF 22228 CCK/W33 with a tapered bore H3128.

The operation data indicated that the main defect of low speed rotors is the failure of bearing elements: rolled bearings, a tapered bore wrenched on an inner ring and shaft and mechanical looseness in the outer ring-case housing.

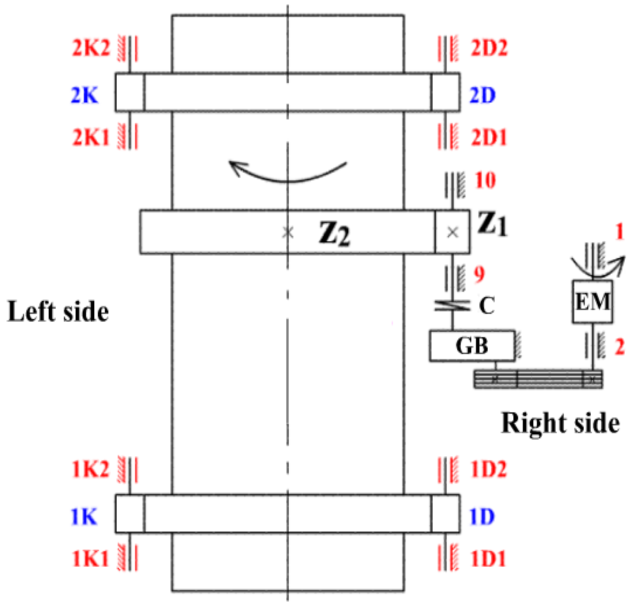

Fig. 1 Kinematic scheme of high mass horizontal cylinder machine: EM – asynchronous electric motor, GB – gear box, C – elastic coupling, z_1 and z_2 – driving and driven spur gears; 1D, 1K, 2D and 2K – cylindrical wheels; D1, 1D2, 1K1, 1K2, 2D1, 2D2, 2K1 2K2, 9 and 10- double row spherical roller bearings SKF 22228 CCK/W33 with a tapered bore H3128

The more sophisticated low rotation speed gear power transmission is operating in sugar production diffusion machine, as shown in Fig. 2: driving spur gears $z_3 = z_3' = 17$ with 15.45 rpm and driven $z_4 = 115$, contact ratio 1.6 and module $m = 20$ mm. The radial double row spherical roller bearings with cylindrical bore SKF 22230 CC/W33 are used. During continuous long term operation teeth surfaces of the gears z_3, z_3' and z_4 were seriously damaged (Fig. 3).

The new diagnostics method was designed to identify the defects of rotors antifriction bearings and gears.

3. The high mass horizontal cylinder machine vibration testing in situ

During the machine operation defects of the gears' z_1 and z_2 teeth surface were identified. For the evaluation of rotor's with z_1 gear transmission technical condition, failure diagnosis and identification causality of damages, the experimental research and theoretical modeling was carried out. Absolute vibration velocity and acceleration, and rotors radial displacement were measured for

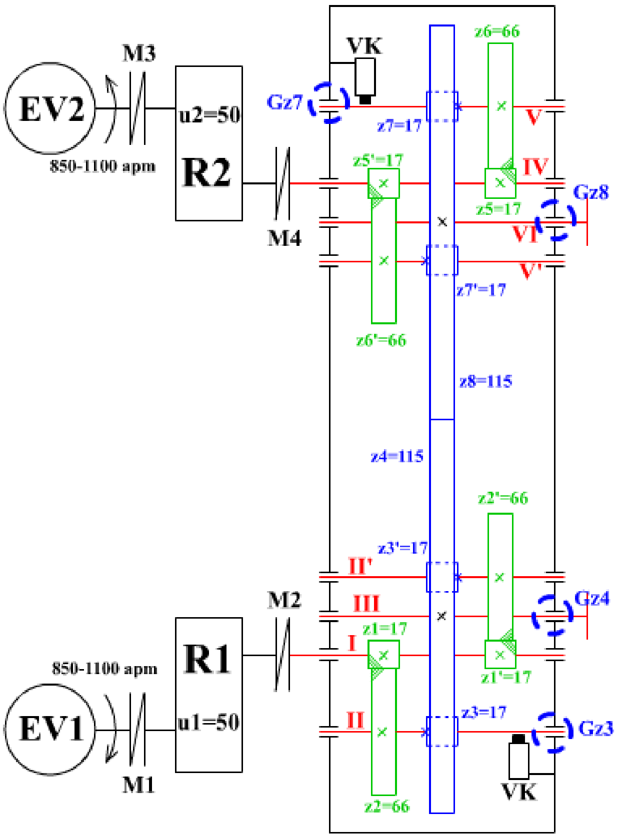

Fig. 2 Kinematic scheme of diffusion machine: EV1 and EV2 – electric motors; M1, M2, M3 and M4 – elastic couplings; R1 and R2 – gear boxes; I, II, II‘, III, IV, V, V‘ and VI – low rotational speed rotors with gears z_1, z_2, z_1‘, z_2‘, z_5, z_5‘, z_6, z_6‘, z_3, z_3‘, z_4, z_7, z_7‘ and z_8; VK – shaft vibration displacement measurement sensors (proximity probes), Gz3, Gz4, Gz7 and Gz8 – radial double row spherical roller bearings with cylindrical bore SKF 22230 CC/W33

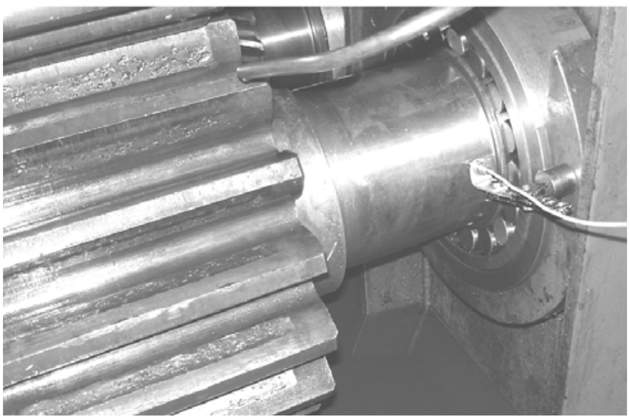

Fig. 3 The damaged z_3 teeth and vibration displacement measurement sensor

the 9th and the 10th bearings housings (Fig. 1). The measurement results were analyzed using Adash Compressed Time (ACMT) method [10]. The measurement of shaft radial vibration displacement were based on shaft displacement measurements with contactless sensors following methods of discreet modeling designed at the Klaipeda University and Kaunas University of Technology.

Condition monitoring of bearings continues systematically. Mechanical vibration measurement data showed that conventional vibration testing methods were ineffective in diagnostics of low rotation high power machines, because the vibration signal was stochastic. The comparison of 9th bearing housing vibration V_{RMS} parameters of the rotor with defective bearing and new one is illustrated in Table 1. The FFT vibration velocity and acceleration spectra are shown in Figs. 4 - 7 and the vibrations velocity and acceleration time in Figs. 8 - 11. At the comparison of vibration data with a damaged bearing and with new bearing outstanding difference between vibrations parameters was not noticed, Table 1.

For the gears z_1 and z_2 vibration amplitudes of teeth meshing frequency ~15.6 Hz and harmonics (~31.1 Hz, ~47 Hz, ~63 Hz, ~78 Hz, ~94 Hz, ~109 Hz,

Table 1

Vibrations velocity V_{RMS} values of 9th and 10th bearings housing

Date	Vibration velocity (10 - 1000 Hz) V_{RMS}, mm/s (V-vertical, H-horizontal vibrations)			
	9V	9H	10V	10H
9th defective bearing	2.01	7.82	1.94	4.29
After 9th bearing replacement	2.18	7.99	1.83	5.32

RMS - root mean square

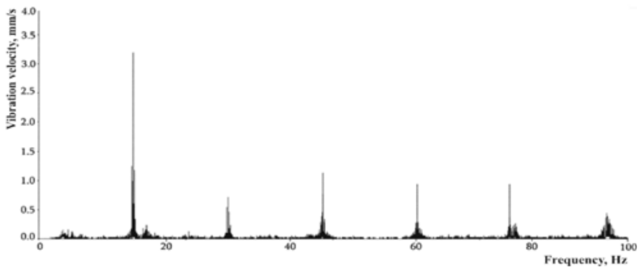

Fig. 4 Horizontal vibration velocity spectrum of defective 9th bearing

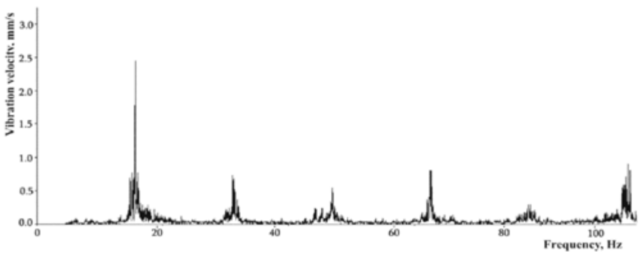

Fig. 5 Horizontal vibration velocity spectrum of a new 9th bearing

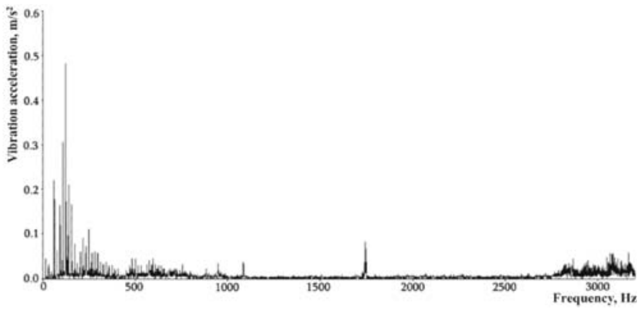

Fig. 6 Horizontal vibration acceleration spectrum of defective 9th bearing

~125 Hz) are dominating, Figs. 4 and 5. Low vibration acceleration amplitudes (up to 0.5 m/s^2) dominate in horizontal vibrations acceleration spectrum of the 9th bearing housing (Figs. 6 and 7) and are generated by the gear transmission.

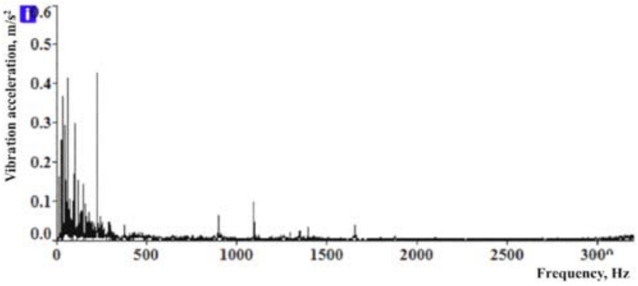

Fig. 7 Horizontal vibration acceleration spectrum of new 9th bearing

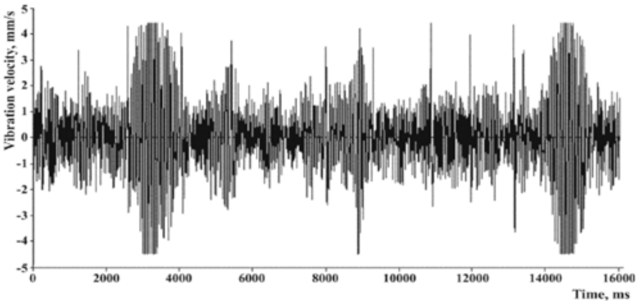

Fig. 8 Horizontal vibration velocity time-base plot of defective 9th bearing

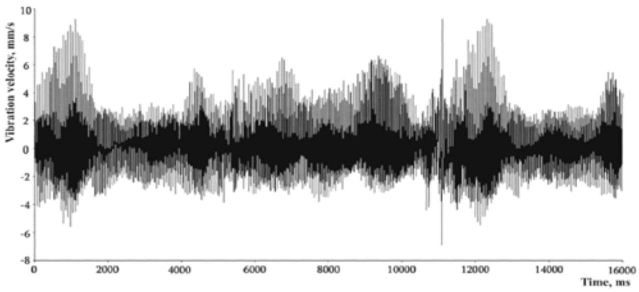

Fig. 9 Horizontal vibration velocity time-base plot of new 9th bearing

Vibration of the 9th bearing housing, which are generated by a gear's z_1 and gear z_2 teeth mesh dominate in the vibration velocity and acceleration time-base plots (Figs. 8 and 11). The impacts from large diameter segmented gear z_2 are excited periodically every ~11th second of each full rotation.

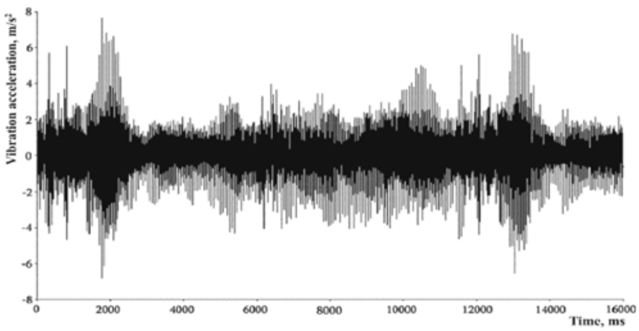

Fig. 10 Horizontal vibration acceleration time-base plot of defective 9th bearing

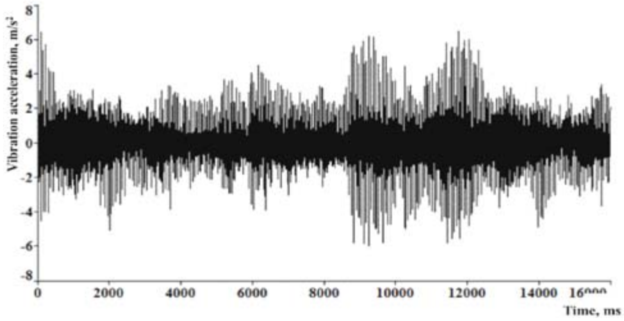

Fig. 11 Horizontal vibration acceleration time-base plot of a new 9th bearing

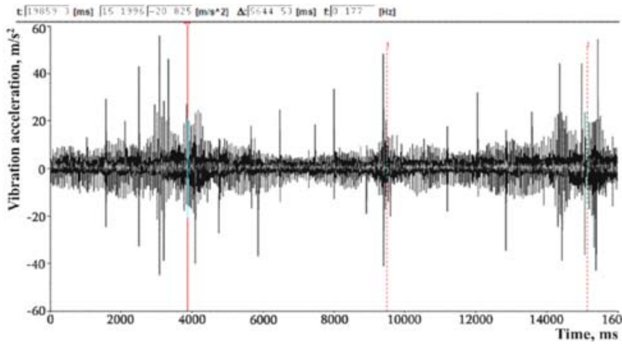

Fig. 12 Horizontal vibration acceleration time-base plot of defective 9th bearing interpreted by the ACMT method

The vibration data FFT formats and V_{RMS} values, presented, are uninformative in condition monitoring and failure diagnostics of defective 9th bearing. The defective 9th bearing (rolled bearing's elements and damaged tapered bore wrenches in the bearing and on the shaft cylindrical surface) generates low amplitude vibration, which were impossible to be identified with ordinary vibration data formats. The more advanced ACMT (Adash **C**ompressed Time Method) method based on measurements of bearings housings vibration accelerations were used as shown in Figs. 12 and 13. The diagrams show vibration of defective and new bearing. Shocks generated due to the change of teeth meshingimpacts when rolling tracks of the bearing are rolled or bearings housings elements have heavy damages. However, the ACMT method with vibrations acceleration evaluates the severance of teeth meshing but did not allow identify the defects in 9th bearing. Diagrams of ACMT vibrations velocity time-base plots show limited information for diagnostics.

Diagrams of vertical vibrations displacements of a shaft to the 9th bearing support, before and after replace-

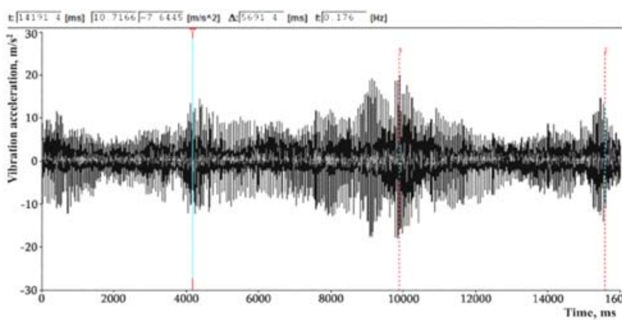

Fig. 13 Horizontal vibration acceleration time-base plot of new 9th bearing interpreted by the ACMT method

Table 2

Low speed rotor with a gear z_1 radial vibration displacements zero-to-peak values

Direction	Rotors radial displacements zero-to-peak values s_{o-p}, µm			
	9th bearing shaft	10th bearing shaft	9th bearing shaft	10th bearing shaft
	9th defective bearing		9th new bearing	
Vertical	132	79	67	63
Horizontal	118	52	38	41

ment of the defective 9th bearing, are presented in Figs. 14 and 15. In case of a defective bearing, the highest measured radial vibrations displacement (relative to 9th bearing support) value s_{o-p} reached 132 µm and with new bearing, it reduced ~2 times and reached 67 µm. Increased radial displacements of the shaft changed the position of teeth mesh pole of a gear z_1 and z_2.

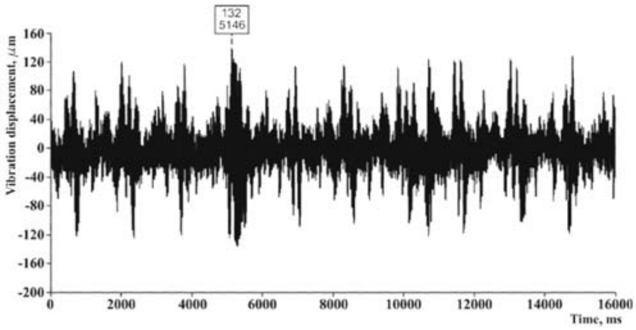

Fig. 14 Radial displacements of the rotor with gear z_1 relative to bearing housing time-base plots with 9th defective bearing

Identification of the defects in the 9th bearing was successful only with the measurements of rotor radial vibration displacements peak-to-peak s_{p-p} or zero-to-peak s_{o-p} values with contactless sensors. Sensors were fixed near the bearing of low speed rotor with a gear z_1 and upon carrying out modeling and theoretical calculations. Measurement results of rotor radial vibration displacements s_{o-p} in horizontal and vertical directions are presented in Figs. 14 and 15. Radial vibration displacements of the rotor's measured before and after replacement of the 9th defective bearing by a new, are presented in Table 2.

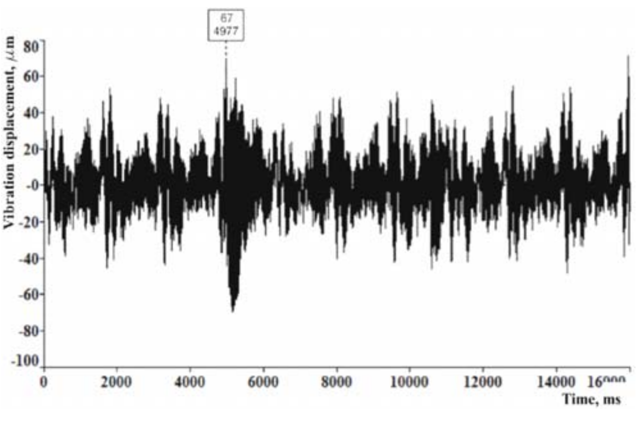

Fig. 15 Radial displacements of the rotor with gear z_1 relative to bearing housing time-base plots with 9th new bearing

4. Sugar production diffusion machine vibration testing in situ

Experimental testing of gear driver in situ of II rotor shaft with gear z_3, were carried out under variable load from 80% to 110% of rated load values. The aim of the research was to determine the displacement and vibration displacement of rotors II shaft located near bearing as was shown in Fig. 3. Experimental research was carried out with RB 6423 contactless sensor and signal analyzer module DMA4 (Epro, Germany). The experimental measurement results are presented in Figs. 16 - 18. The results of vibration displacement measurement for the driving rotor shaft II show that antifriction bearing support has low radial and axial stiffness and large mechanical looseness.

The vibration displacement amplitude s_p of the driving gear z_3 rotor shaft measured in a horizontal (radial) direction is 150 - 200 µm and in the vertical 120 - 150 µm. The axial displacement of rotor II is high with low frequency 0.01217 Hz, Fig. 17. These results determined that during the operation of a gears z_3, z_3' and z_4, the teeth base pitch permanently changes its position that caused valuable changes in teeth meshing pitch point positions of the involutes teeth surfaces.

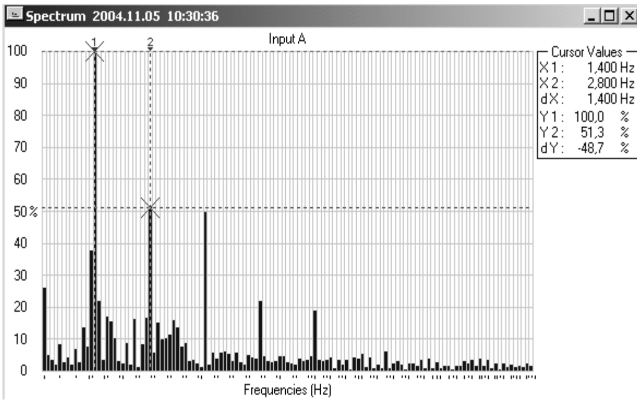

Fig. 16 The rotor II shaft horizontal (radial) vibration displacement spectrum: 1.4 Hz frequency teeth meshing shocks dominated

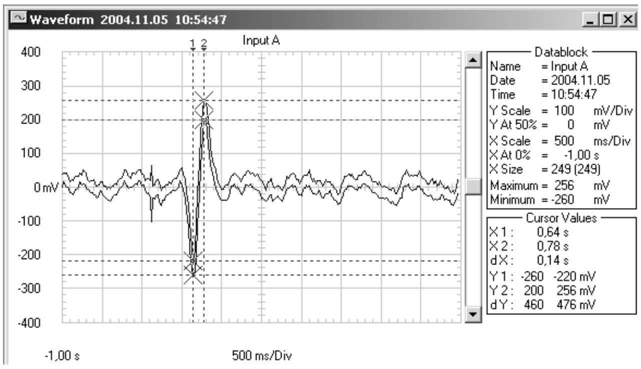

Fig. 17 The rotor II shaft axial displacement plot during one full rotation of rotor III (0.73 rpm)

As shown in Fig. 18, the rotor III with driven gear z_4 provides large eccentricity values (up to 0.190 mm) that negatively influences teeth meshing; dynamic forces in teeth were increased and finally contact stresses reached inadmissible values. These experimental results were applied in the modeling to simulate contact stresses in the gears. The contact stresses versus deformations of driving rotor II with gear z_3 were calculated.

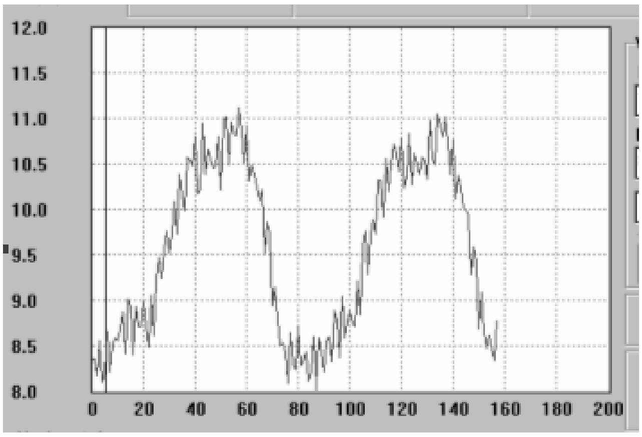

Fig. 18 Displacement (eccentricity) and vibration displacement time plot of rotor III (with gear z_4) shaft during two rotations: in the vertical axis – shaft displacement measured in volts, in the horizontal axis – time measured in seconds

5. Rotors supervising, mathematical modeling and diagnostics method (RSMMD)

The analysis of theoretical and experimental research materials showed that there are no finished and reliable diagnostics technologies for defects identification of low speed rotors with antifriction bearings and gear transmissions. Therefore the authors designed and validated new diagnostics method RSMMD as showed in Fig. 19. It has the main difference that vibration measuring is performed with contact less sensors instead accelerometers

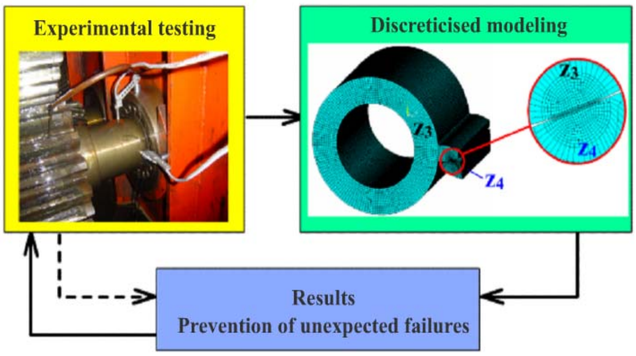

Fig. 19 The scheme of RSMMD method used for diagnostics of low speed rotor

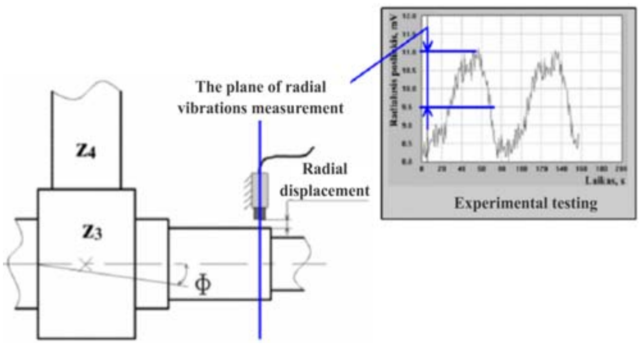

Fig. 20 The shaft radial displacement measurement scheme

or vibration velocity seismic transducers. The vibration displacement measurements of rotor are performed near the bearing as showed in Fig. 3. The measured vibration displacement amplitude zero-to-peak $s_{o\text{-}p}$ is involved in mathematical model and using mathematical simulation it is desirable to analyze gears teeth meshing conditions as showed in Fig. 20.

6. Modeling and numerical example

Modeling of high mass horizontal cylinder machine and diffusion machine's gear transmissions and tooth contact is carried out by the FEM with ANSYS software. Experimentally measured radial displacements $s_{o\text{-}p}$ values of cooler's gear z_1 and diffusion machine's rotor gear z_3 were integrated into discreticised models (separates models). The models were made under least favorable conditions of teeth meshing, i.e. when only one pair of teeth were in contact and the both machines are fully loaded. The models were settled with a change of rotor's radial displacement from 0 to 500 µm in the case of cooler and from 0 to 250 µm in the case of diffusion machine. The models were solved when both rotors materials have linear stiffness characteristics. Geometric models of teeth are divided into lower order finite elements of hexahedron SOLID 45 type, having eight nodes and three degrees of freedom (displacements in respect of three coordinates) as showed in Fig. 21. For the reason that the modeled gear transmissions are of low speed, the solutions were worked out in static

$$[K]\{U\} = \{F\} \tag{1}$$

where $[K]$ is the stiffness matrix of finite element; $\{U\}$ is the nodal element displacement vector; $\{F\}$ is the load vector.

Nodal displacement vector $\{U\}$ is composed of

$$\{U\} = \lfloor u_x \quad u_y \quad u_z \rfloor \tag{2}$$

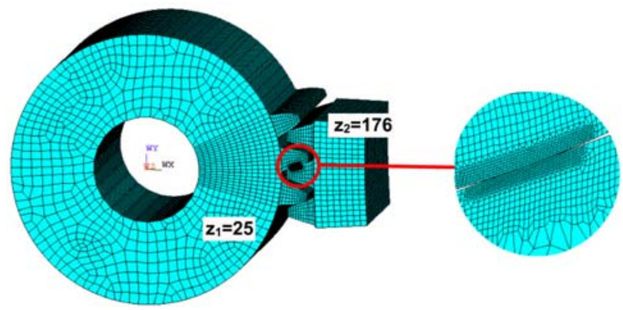

Fig. 21 Main view of gears teeth meshing of high mass horizontal cylinder machine

Having solved the teeth contact discreticised models, it was determined, that in case of a rotor's radial displacements, located next to support of a bearing, are equal to 0, teeth contact stresses reach 341 MPa value in the case of cooler, and 378 MPa in the case of diffusion machine. When radial displacement of the rotor of high mass horizontal cylinder machine exceeds the value of 130 µm and 100 µm in the case of diffusion machine, permissible contact stresses value (σ_{adm}=~500 MPa) is exceeded, therefore plastic deformations occur in the teeth, and the teeth are damaged. Radial displacement of the rotor is limited by values 130 μm of high mass horizontal cylinder machine and 100 μm of diffusion machine in continuous long term operation. A diagram, presented in Fig. 22, can be used in order to determine the residual resources of service for rotors only if bearing's rolling tracks and rolling elements are rolled or damaged the tapered bore-shaft outer ring-case mounting tolerance of the bearing housings.

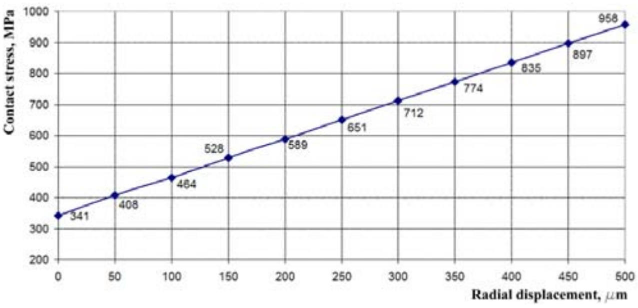

Fig. 22 The teeth z_1 and z_2 contacts stress values versus rotor's radial displacement s_{o-p} measured values

7. Conclusions

Experimental research confirms that ordinary vibration research methods with standard vibration data formats (root mean square values, FFT spectrum, time-base plots, etc.) are not suitable for the evaluation of technical condition of low speed rotors with gears and antifriction bearings. It is also impossible to identify defective bearing of a low speed rotation rotor with gears using ACMT method. Diagrams of ACMT vibrations acceleration time-base plots allowed only the identification of shocks generated by the impacts of gears teeth meshing.

Diagnostics and evaluation residual performance of low speed rotor is possible using absolute (or relative) measurements values of radial displacements s_{o-p} or s_{p-p} provided by contact less sensors.

Theoretical modeling together with experimental research of dynamics of a gear driver and roller bearings in low rotation speed machines is realized as a new Rotors Supervising, Mathematical Modeling and Diagnostics method.

Experimental and simulation results approved that low frequency vibration displacement values are the main parameters that describe technical condition of involute teeth meshing quality.

References

1. **Barzdaitis, V.; Činikas, G.** 1998. Monitoring and Diagnostics of Rotating Machines. Kaunas: Technologija, 364p. (in Lithuanian).
2. **Gottvald, J.** 2010. The calculation and measurement of the natural frequencies of the bucket wheel excavator SchRs 1320/4x30, Transport 25(3): 269-277.
3. **Jonušas, R.; Juzėnas, E.; Juzėnas, K.** 2010. Analysis of some extreme situations in exploitation of complex rotory systems, Mechanika 1(81): 53-57.
4. **Lazar, B.; Wojnar, G.; Mady, H.; Czech, P.** 2009. Evaluation of gear power losses from experimental test data and analytical methods, Mechanika 6(80): 56-83.
5. **Lipovszky, G.; Solyomvari, K.; Varga, G.** 1990. Vibration testing of machines and their maintenance. Joint edition published by Elsevier Science Publishing, Amsterdam, the Netherlands and Akadėmiai Kiado, Budapest, Hungary, 303p.
6. **Muszynska, A.** 2002. Rotordynamics, CRC Press, Boca Raton, FL, Taylor &Francis, London, 2005. 1075p.
7. **Bently, D.E.; Hatch, C.T.** Fundamentals of rotating machinery diagnostics, Bently Pressurized Bearing Press, Minden, Nevada, USA. 726p.
8. **Mažeika, P.; Barzdaitis, V.; Didžiokas, R.; Žemaitis, V.** 2006. Modeling and simulation of gearings in the diagnostics medium. Proceedings of the 6th international conference "Vibroengineering 2006". ISSN 1822-1262: 139-142.
9. **Mažeika, P.; Barzdaitis, V.; Bogdevičius, M.; Didžiokas, R.** 2007. Realiability of rotating system with gearings. Proceedings of VI international conference „The improvement of the quality, reliability and long usage of technical systems and technological processes". ISBN 966-330-028.- Hurghada, Egypt: 40-43.
10. **Mažeika, P.; Didžiokas, R.; Barzdaitis, V.; Bogdevičius, M.** 2008. Dynamics and reliability of gear driver with antifriction bearings, ISSN 1392-8716, Journal of Vibroengineering 10: 217-221.
11. **Mažeika, P.; Žemaitis, V.; Didžiokas, R.** 2005. Diagnostics of gear power and other transmissions rotating on rolling bearings, ISSN 1648-8776, Journal of Young Scientists 2(6), Šiauliai, Lithuania: 96-105 (in Lithuanian).
12. **Airapetov, E.L.; Aparkhov, V.I.; Evsikova, N.A.; Melnikova, T.N.; Filimonova, N.I.** 1995. The model of teeth contact dynamical interaction in the spur gearing. Proceedings of Ninth World Congress on the Theory of Machines and Mechanisms, IFToMM, Politechnico di Milano, Italy, vol.1: 459-461.

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KRUMPLINIŲ PAVARŲ MODELIAVIMAS IR DIAGNOSTIKA

R e z i u m ė

Straipsnyje pateikiami eksperimentiniai ir teoriniai lėtaeigių krumplinių pavarų (iki 40 aps/min) krumplių kabinimosi kokybę įvertinantys diagnostiniai tyrimai. Tyrimų rezultatai gauti veikiant mašinai ilgalaikės nepertraukiamos natūralios eksploatacijos sąlygomis. Lėtaeigis rotorius su cilindriniu krumpliaračiu sukasi dvieiliuose ritininiuose riedėjimo guoliuose su kūgine įvore. Sukurtas krumplių susikabinimą įvertinantis fizikinis modelis. Eksperimentinių tyrimų rezultatai gauti bekontakčiais poslinkių matavimo jutikliais ir panaudoti kuriant teorinį modelį. Rezultatai patvirtina skaičiavimo duomenis, leidžiančius įvertinti krumplių susikabinimo techninę būklę ir išvengti rotorinės sistemos netikėtų gedimų, veikiant ilgalaikiame nepertraukiamame darbo režime.

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MODELING AND DIAGNOSTICS OF THE GEAR POWER TRANSMISSIONS

S u m m a r y

This article presents experimental and theoretical research data based on mechanical vibrations of low speed rotor system (up to 40 rpm) with a gear transmission and double row spherical roller bearings with tapered bore. The rotor system experimental and theoretical modeling research analysis results confirm effective technical conditions monitoring and failures diagnostics method of low speed rotor system with antifriction bearings. The modeling and simulation acquired results generate determinations of residual operation time and service of gear transmissions and roller bearings. Authors presented a method of low rotation speed rotor vibration severity evaluation with validation data acquired by the experimental data of continuous long term running machine in situ.

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МОДЕЛИРОВАНИЕ И ДИАГНОСТИКА ЗУБЧАТЫХ ПЕРЕДАЧ

Р е з ю м е

Статья оценивает техническое состояние тихоходных (до 40 оборотов в минуту) цилиндрических зубчатых передач с роликовыми радиальными сферическими двухрядными подшипниками с коническим отверстием. Результаты исследований получены экспериментальным путем многолетней практикой диагностики технологических машин на предприятиях. Создана и валидирована теоретическая модель оценивающая качество зацепления зубьев цилиндрических зубчатых передач в зависимости от технического состояния подшипников качения ротора. Результаты исследований позволяют оценить долговечность безотказной работы роторной системы с цилиндрическими зубчатыми передачами, работающей в непрерывном продолжительном режиме.

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