THE ARCHIVE OF MECHANICAL ENGINEERING

VOL. LIV 2007 Number 2

DOI: 10.24425/ame.2007.131546

Key words: monitoring, failure, experiment and simulation, gear driver, roller bearings, FEM

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MODELING AND TESTING OF GEAR DRIVER WITH ROLLER BEARINGS

Technical system for condition monitoring and failure diagnostics of gear driver with roller bearings was tested in situ. The experimental measurement data of rotors shaft vibration displacements were introduced into physical model of gear teeth meshing dynamics. The modelling and simulation of teeth failures in gear driver with roller bearings was performed by finite element method. The experimental and simulation results were used in identification and elimination of sources of gear teeth damages and bearings failures.

1. Introduction

Condition monitoring and failure diagnostics technology of heavy duty rotating systems with antifriction bearings makes it possible to avoid accidents in machine operation. The damaged gears or bearings may lead to catastorphic failure of a machine running in long term continuous operation mode. In paper [1], a computer simulation for static and dynamic behaviour of mesh gear teeth is presented. This includes aspects of parametric modeling of involute spur gear pairs in meshing process, static, dynamic and kinematic conditions of meshing gear teeth and flexibility algorithm of static and dynamic contact finite element method. In the study, the authors also analyse the results of the contact force and dynamic load of meshing gear

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teeth with the consideration of elastic deformation of gear teeth, meshing impacts due to stiffness variation of gear teeth and simultaneous variation of meshing tooth pairs. In paper [2], one proposes the use of vibroacoustic method for analysing of gear drives vibration. The method is based on the idea that early gear faults or any other irregularities of working gear drives can be determined by the analysis of emitted vibration. The analysis is based on the observation of the characteristic frequencies in the frequency spectrum, which can be obtained with a proper transformation of the recorded signal. Paper [3] concerns the dynamics following the operation of gear power transmission. The experimentally confirmed dependences are proposed for determination of the dynamic parameters of gear mesh loading. The results could be used for strength evaluation of gear power transmissions and for elaborating methods of elimination of vibration. Paper [4] presents the problems of diagnostics of modern machines comprising gear drivers with antifriction bearings. The wide range of vibration frequencies, caused by machines and bearing elements, is divided into four frequency intervals. Together with diagnostics results, one introduces the results of in situ analysis of monitoring data acquired during experimental testing of the planetary gear power transmission, belt drivers, rotor with roller bearings, etc.

The aims of the study described in this paper were: to carry out dynamic research of a gear driver with roller bearings by applying information technologies and to identify the causality of involute gears' defects; to carry out experimental research in situ in order to determine displacement of a gear driver rotor shaft as maximum displacement value relative to the bearing housing, and to evaluate the quality of teeth meshing; to design physical model of rotor with gearing and to simulate contact stresses of gears teeth versus teeth meshing inaccuracies and finally prevent failures of the machine.

2. The object of research

Diffusion machines are used in sugar production industry. The machine comprises a critical (key element in technological diagram) low-speed gear driver with high torque, as shown in the kinematical scheme in Fig. 1: driving spur gears z_3, z_3' and driven z_4 with module m = 20 mm, $z_3 = z_3' = 17$ and $z_4 = 115$. Rotation speed of driving gears z_3 and z_3' is low 15.45 rpm, contact ratio is 1,6. Rotors are rotating in radial double row spherical roller bearings with cylindrical bore (SKF 22230 CC/W33).

During continuous long term operation of the machine, the gears z_3, z_3' and z_4 and bearings generated high dynamic forces: the contact surfaces of the gears teeth were damaged, as showed in Fig. 2.

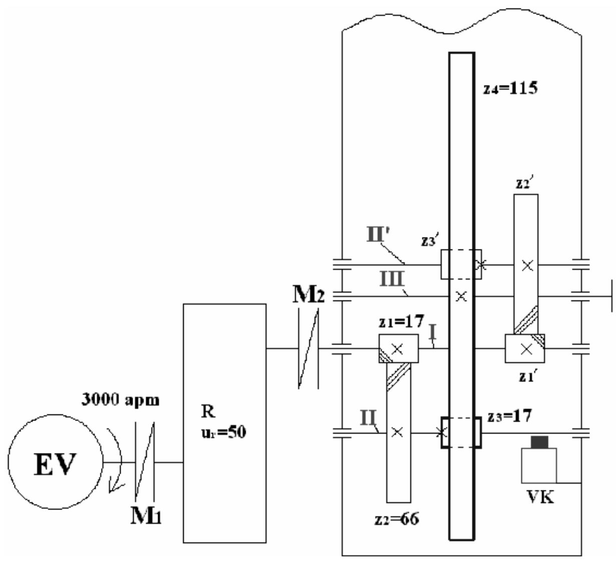

Fig. 1. Kinematics' diagram of gear driver in diffusion machine: EV – electromotor; M1 and M2 – elastic couplings; R – reducer; VK – sensor of shaft vibration displacements, I, II, and III rotors

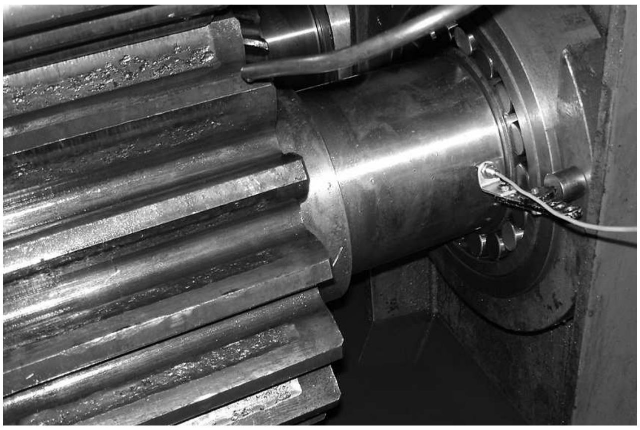

Fig. 2. The view of driving gear z_3 teeth damages and the II rotor shaft displacement and vibration displacement measurement with contactless sensor

3. Dynamics of gear driver with antifriction bearings

Experimental research on dynamics of the 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 the II rotor shaft located near bearing as shown in Fig. 2. Experimental research was carried out with RB 6423 (Epro, Germany) contactless sensor (VK) BNC 3500 (USA) signal analyzer module. The view of experimental setup and measurement results are presented in Fig. 2,…, Fig. 4.

The measurement results of the driving rotor shaft II show that antifriction bearing support has low radial stiffness and large mechanical looseness. The vibration displacement amplitude $\boldsymbol{s_p}$ of the driving gear z_3 rotor shaft in horizontal direction is 150÷200 µm (Fig. 3 – Fig. 4) and 120÷150 µm. in the vertical direction. These experimental tests results have shown that during the operation of a gears z_3, z_3' and z_4, the teeth base pitch permanently changes, and that causes significant changes in teeth meshing pitch point of the involute teeth system.

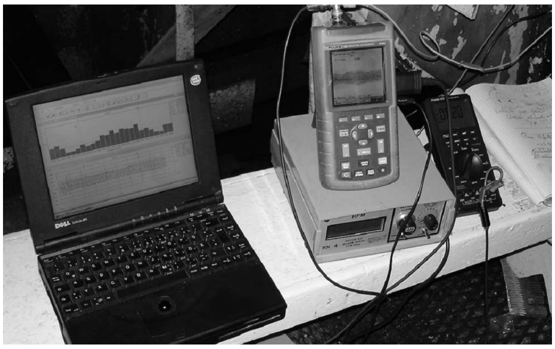

Fig. 3. The view of measurement unit

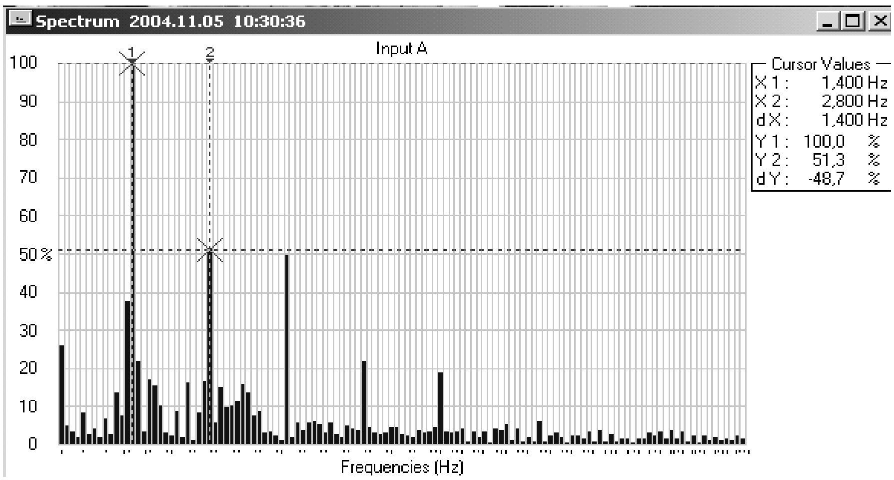

Fig. 4. The horizontal vibration displacements spectrum of the rotor II driving shaft: vibration displacement amplitude of 1,4 Hz frequency indicates teeth meshing process (vibration displacement values are expressed in % of the maximum value)

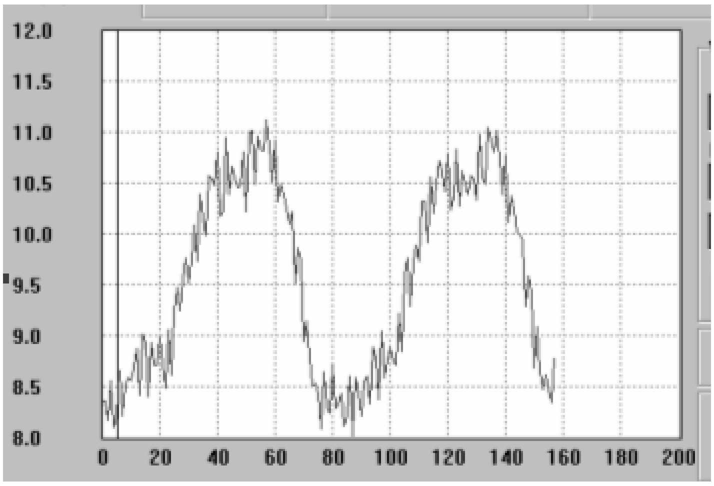

Fig. 5. Displacement (eccentricity) and vibration displacement time plot of rotor III (with gear z_4) shaft during two rotations: displacement measured in volts, in the horizontal axis – time measured in seconds

As shown in Fig. 5, rotor III with the driven gear z_4 cause large eccentricity, and thishas a negative influence on teeth meshing. The resulting incorrect teeth meshing increases dynamic forces that additionally loade teeth and finally increase contact stresses that reaches inadmissible values.

These experimental results were applied in the modelling, in order to calculate the function of contact stresses versus deformations of driving rotor II with gear z_3.

4. Modeling and simulation of the teeth meshing

Modelling and mathematical simulation of a gear driver was carried out in accordance to the designed physical model and experimental tests applying the FEM. Then, contact stresses of meshing teeth were calculated. The task was solved applying ANSYS software. The mathematical model was developed after accepting the most unfavourable conditions for teeth meshing of gearz_3, z_3' and z_4, – with one pair of teeth in contact, and changing shaft vibration displacement amplitude from 25 μm up to 250 μm.

Geometric model of the gear driver was developed using SOLID WORKS graphical software and ANSYS scripts, assigning for drive gear z_3 and driven gear z_4 identical real strength parameters of materials: elasticity module $E = 2 \cdot 10^{11}$ Pa and Poisson's ratio $\mu = 0.3$. Geometric model of gear teeth was divided into tetrahedral and hexahedral 3D elements of SOLID95 as presented in Fig. 6.

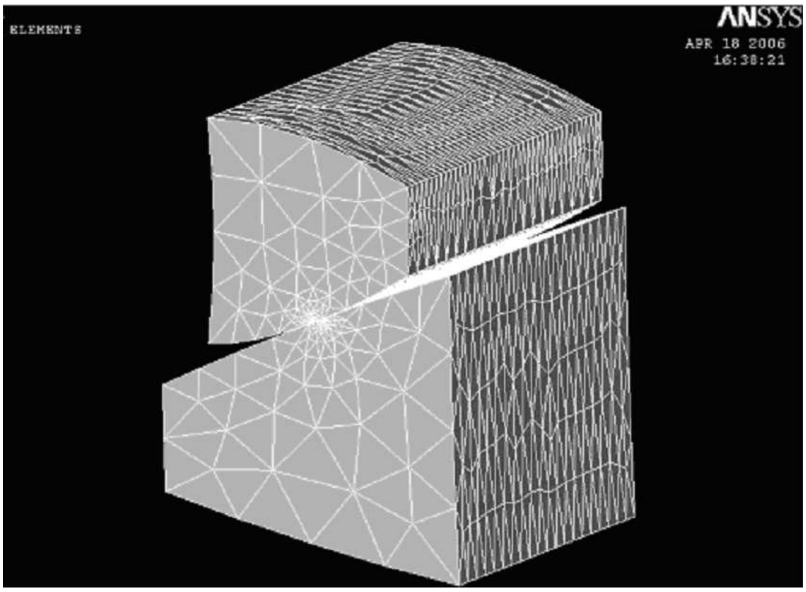

Fig. 6. Physical model of teeth meshing in driving z_3 and driven z_4 gears in FE with pitch point

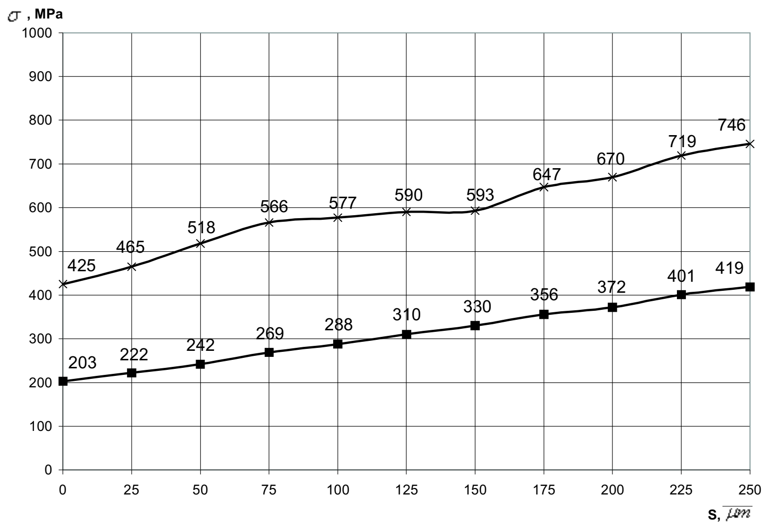

Fig. 7. Simulation results of the contact stresses of driving (1) and driven (2) gear meshing teeth versus displacements of rotor at the bearing

Modelling and simulation of meshing teeth in gears z_3 and z_4 by FEM led to the following conclusions (illustrated in Fig. 7): under ideal tooth meshing, when vibration displacement amplitude of the rotor II shaft equals 0, the maximum contact stresses at the surface of teeth is low – in driving tooth 201 MPa (1, Fig. 7) and in the driven tooth 400 MPa (2, Fig. 7). When vibration displacement amplitude of the rotor II shaft reaches 200 μm, the maximum contact stresses of teeth increases significantly – and equals in the driving tooth 372 MPa (1, Fig. 7) and in the driven tooth 647 MPa (2, Fig. 7).

On the basis of gearing design results, it was accepted that admissible contact stresses of a teeth in gears z_3 and z_4 is σ_{adm} = 450 – 600 MPa. The simulation results indicate that, when rotor II shaft at bearing vibration displacement amplitude s_p exceeds 200 μm, the contact stresses of the meshing teeth exceeds the admissible value of σ_{adm}. This amplitude of vibration displacement is the main reason for damage of the involute teeth.

5. Conclusion

The authors carried out theoretical and experimental research on dynamics of a gear driver in a diffusion machine. Mathematical modelling of gear driver was designed using FEM and simulation was performed by using the ANSYS software. Experimental and simulation results confirmed that vibration displacement of the driving gear with rotor II shaft is the main parameter that describes technical condition of involute teeth meshing quality. Damages of working surfaces of involute teeth in gears z_3 and z_4 were caused by high radial displacement of the rotor II shaft or the roller bearing: e. g. the shaft displacement 200 μm measured at the roller bearing location indicated unacceptable teeth meshing quality in the gearing.

Manuscript received by Editorial Board, March 20, 2007

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Modelowanie i badania napędu zębatego z łożyskami wałeczkowymi

S t r e s z c z e n i e

Przedmiotem pracy były testy in situ systemu technicznego do monitorowania i diagnostyki uszkodzeń napędu zębatego z łożyskami wałeczkowymi. Eksperymentalne dane pomiarowe dotyczące przemieszczeń wibracyjnych wałów wirników wprowadzono do fizycznego modelu dynamiki współpracy zębów napędu. Modelowanie i symulacja uszkodzeń zębów w napędzie z łożyskami wałeczkowymi była wykonana przy użyciu metody elementów skończonych. Wyniki eksperymentów i symulacji zostały wykorzystane do identyfikacji i eliminacji przyczyn uszkodzeń zębów i awarii napędu.