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NONLINEAR VIBRATIONS AND BACKLASHES DIAGNOSTICS IN THE ROLLING MILLS DRIVE TRAINS

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Abstract

The angular and radial backlashes due to intensive wear are the most distinctive feature of the cold and hot rolling mills drive trains. It causes nonlinear torsional vibration and significant torque amplification. It leads to equipment failures but it could be used for wear diagnostics in the range of natural frequencies of the drive trains during the transient processes. The static load and dynamic response interrelation, non isochronisms and other nonlinear system features are used for wear diagnostics under the non stationary loads and speeds.

Key words

Rolling mills, torsional oscillations, backlashes diagnostics.

1 Introduction

The rolling mills drive trains work under the extremely high loads and are characterized by the increased wear. Backlashes and frequent step-like impulse loads during the hot metal rolling causes the most numerical failures in the drive trains. Standard methods of vibration diagnostics based on envelope curve spectrum analysis require stationary drive speed and load for averaging. It is quite difficult to provide constant load in the rolling mills because of metal temperature and friction conditions variation in the work rolls gap. Therefore the new approach is proposed for wear diagnostics based on torque and vibration transient processes analysis. It allows to avoid inconveniences of standard diagnostics methods.

Detailed research in rolling mills dynamics and model analysis see in [Gudehus, 1983].

2 Rolling mills drive trains schemes

The most common type of the drive trains for hot rolling and some temper cold rolling mills see in Fig.1.

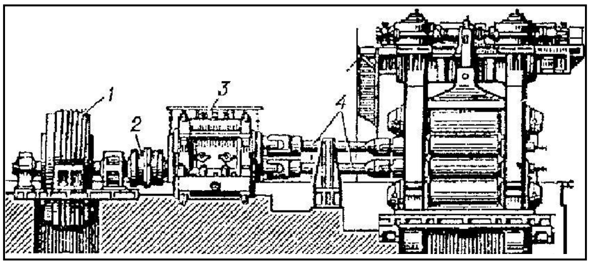


Figure 1. Hot rolling mills drive train.
(1) Motor (2) Coupling (3) Pinion stand (4) Spindles

Almost 70-80% of overall angular backlashes in the drive train exist in spindles couplings. The rest of backlashes are distributed in other gears and couplings. It is important to note that spindles backlashes influence on torsional vibration is very dependant on tuning accuracy of the weight compensation mechanism.

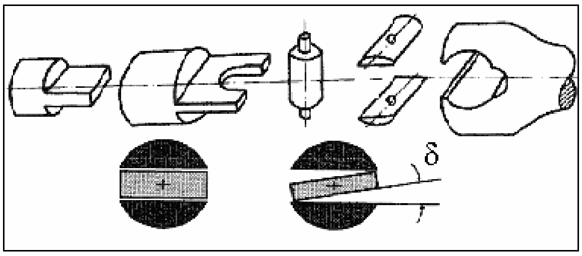
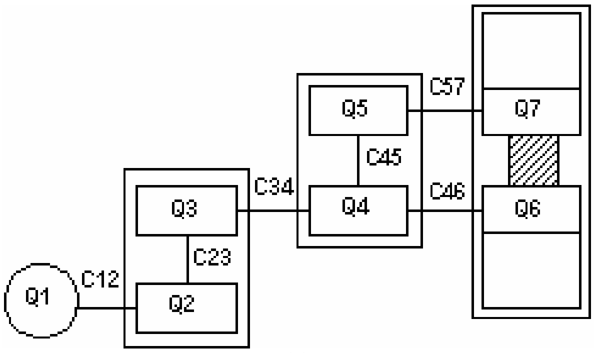


Figure 2. Spindle sliding block wear and backlash (δ)

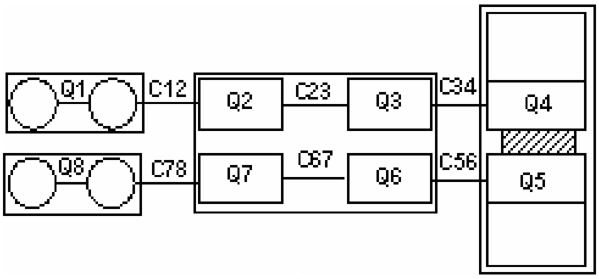
Even ±5% tuning error of spindles weight (0.2-2 kN) will cause spindles vertical periodical (twice per one rotation) motion and backlashes opening. Beside it significant misbalance appears in the rolls gap due to spindles and work roll axis misalignment after the rolling torque step-like raising.

3 Drive train spring-mass models

The different drive trains spring-mass models schemes are shown in the Fig. 3.



(a)



(b)

Figure 3. Rolling mills drive trains schemes.
(a) Hot rolling mill 4-high stand with single drive
(b) Cold rolling mill 4-high stand with twin-drives

The hot rolling mills spring-mass model (see Fig.3(a)) has the following elements: direct current or synchronous electric motor (Q1), intermediate gear couplings (C12, C34), gearbox (Q2, C23, Q3), pinion stand (Q4, C45, Q5), two spindles (C46, C57) and the rolling stand itself with two pairs of upper (Q7) and lower (Q6) work rolls (WR) and backup rolls (BUR) which moment of inertia is reduced to the WR axis. System has branched structure due to spindles.

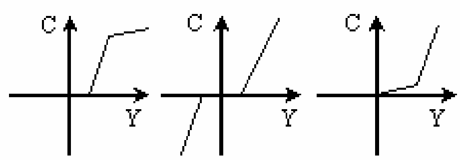
The cold rolling mills (see Fig. 3(b)) differ by the twin drives for every pair of WR and BUR and two separated rotors for every electric motor (Q1, Q8). The gearbox (Q2, Q3, Q6 and Q7) does not connect upper and lower WR drive trains but designed in whole housing. It has train structure with the upper and lower motors as the masses at both ends.

Models are not complicated here by the periodical gears stiffness as the short transient processes are investigated (about 0.2-0.5 s).

The particular element in the dynamic model is metal sheet in the rolls gap (shown shaded in Fig. 3). Usually, it is considered for torsional vibration analysis that the friction contact between WR and rolled metal acts like ideal kinematical coupling. But modern rolling theory gives formulas for torque calculation where the slipping zones are fairly admitted. Such contradiction was discussed [Marjuta, Krot, 1997] for energy transferring process by friction in the rolling mills and the new approach was proposed based on kinetic energy saving law. In this work we used conventional model for research where the rolling torque is considered as an external load for torsional vibration.

4 Backlashes as nonlinear stiffness

Different kinds of backlashes in the rolling mills equipment are given on graphs in the Fig. 4 where denoted: C – stiffness, Y – generalized coordinate of motion. Function in the Fig. 4(a) describes the summary stiffness for bearings and its bolting. The first fracture point means backlash in bearing itself. The second fracture point means deformation when the gap between housing and cover are opening during shock torsional vibrations in the gearboxes and pinion stands. Function in the Fig. 4(b) describes conventional dead zone in the drive train couplings caused by wear or assembly errors. The Fig. 4(c) shows the 4-high stand rolls stack vertical stiffness for low and high rolling loads.



(a) (b) (c)

Figure 4. Different kinds of backlashes in the rolling mills equipment:
(a) Bearings with bolting (b) Coupling (c) Rolls stack

Angular wear increases torque amplification factor (TAF) in all drive train elements and if overload occurs it consequently leads to sudden failures. Therefore backlashes should be considered as the main parameters of the rolling mills drive trains condition. The maintenance based on its diagnostics is complicated because of short period of transient process when the full torque is appearing and the backlashes are closing. Under the full torque load it is difficult to determine the exact backlashes by the modulation characteristics of the vibration.

5 Nonlinear dynamics simulation

The analytical research of nonlinear multi-body systems usually assumes reducing the initial system to less DOF. That is possible but inefficient as it requires the analysis of every wear influence for its diagnostics even if it has a little contribution to the overall energy of torsional vibration (see Fig. 5). Therefore numerical models (see Fig. 3) with the enough detailed springs and masses were identified and used for wear diagnostics.

Model identification has been fulfilled in time domain and also in a frequency domain. Well known earlier fact that backlashes when opened cause high frequency vibrations was confirmed by simulation. But it was cleared up what the frequencies exactly appear in the signals of torque and bearings vibration and the way it could be used for backlashes diagnostics.

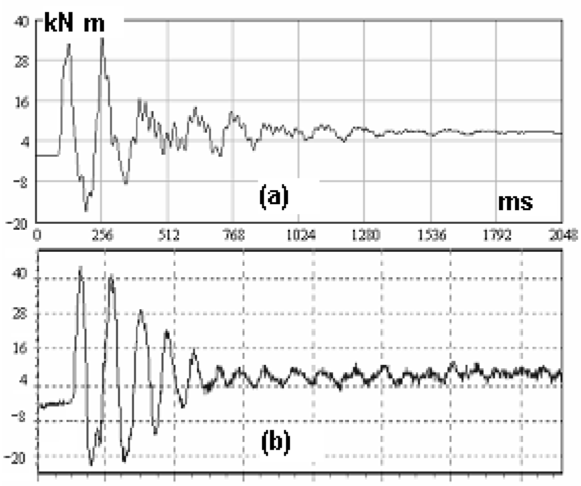
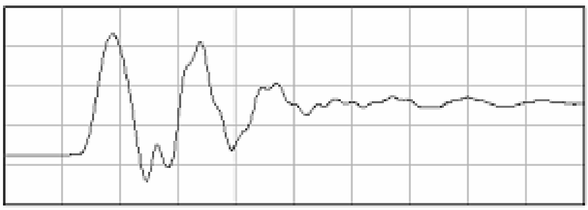


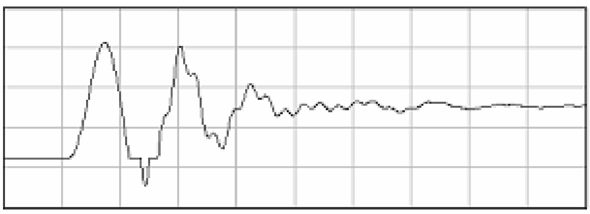
Figure 5. Nonlinear torsional vibration:
(a) Calculation (T12) (b) Measurement

The model simulation has shown that higher natural frequencies associated but not equal with partial frequencies of the system torsional vibrations will rise in torque spectrum. Also as it is known from theory the 2nd and 3rd harmonics of the main natural frequency will appear too. Such regularities have been taken for wear diagnostics methods.

For example, torque curve may exhibit non isochronisms and different kind particularities which are used for wear diagnostics (see Fig. 6).



(a)



(b)

Figure 6. Nonlinear effects in the drive train:
(a) Wear in the gearbox (b) Wear in the couplings

Torque amplification factor (TAF) is one of the parameters for system dynamics estimation. But for nonlinear systems dynamic response depends on static load (rolling torque). It was shown (see Fig. 7) that with less torque Tstatic we obtain bigger TAF for different angular wear (0.000…0.012 rad). Such nonlinearity is almost invisible for Tmax curves and usually it is not taken into account for the durability calculations in the rolling mills. But it has influence for low Tstatic loads.

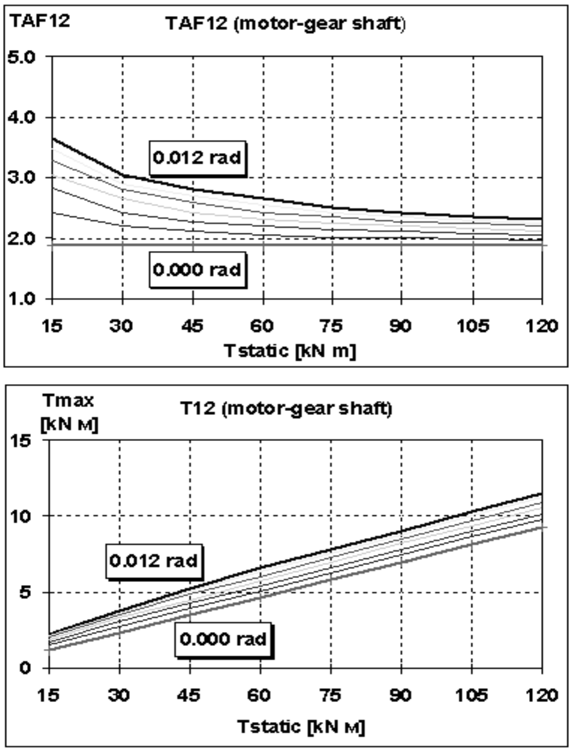


Figure 7. Nonlinear static torque influence:
(a) TAF on motor-gear shaft (b) Peak torque Tmax

6 Field measurements technique

The possibility to fulfill measurements in the industry rolling mills aids to understand and estimate numerically the influence of wear (angular and radial backlashes) on vibration and torque signals parameters.

Torque measurements were fulfilled with the 8-channel telemetry system newly designed in the Iron & Steel Institute. Vibration was measured with the 4-channel signal conditioner (PCB Piezotronics model 48A22) and IMI Sensors accelerometers (model 603C01). The special software was used for signals recording and FFT transform in conjunction with low-pass filtering and other signal processing procedures.

The calculated natural frequencies of the investigated rolling stand drive train were as following: 12, 15, 20, 34, 45 and 81 Hz. As it expected torque signal has peaks at the lower frequencies and the vibration has at the higher range (see Fig. 8). No one frequency of teeth couplings or bearings was in this range. A little peak was observed in the torque signal at the 6-7 Hz near the rotating frequency.

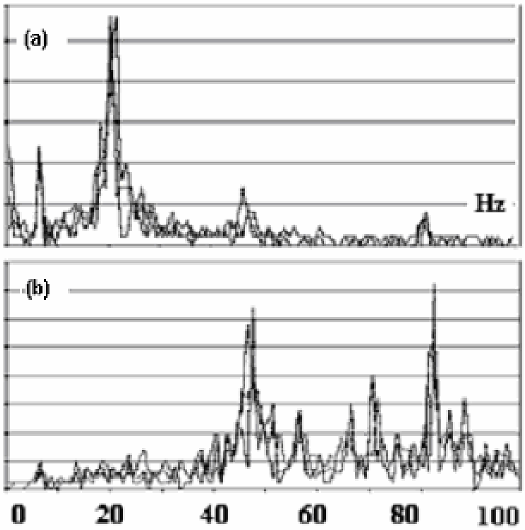


Figure 8. Measured signals spectrums (averaged): (a) Torque (b) Vibration

The difference between the calculated and measured natural frequencies does not exceed about 1-2%. It is enough taking into account the low frequency range and time of recording.

7 Drive train wear diagnostics

As it could be seen from Fig. 9 non isochronisms appeared if the first period (marked by circle) and the next periods of torque and vibration oscillations were compared. The period length depends on wear in couplings and may be determined for diagnostics purposes. Vibration signal needs low-pass filtering before analysis. The low-pass filter cutting frequency should be twice more than highest natural frequency.

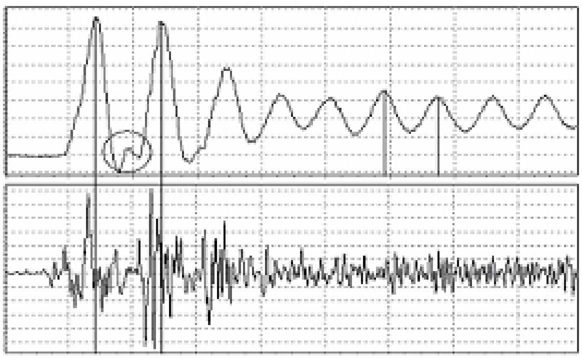


Figure 9. Non isochronisms in the Torque (upper) and Vibration (lower) signals due to wear.

The spectrum peaks amplitudes (A12…A81 Hz) and according phases at the natural frequencies (see Fig. 10) were obtained for different gearbox and spindles wear (angular backlashes) by the torque signals.

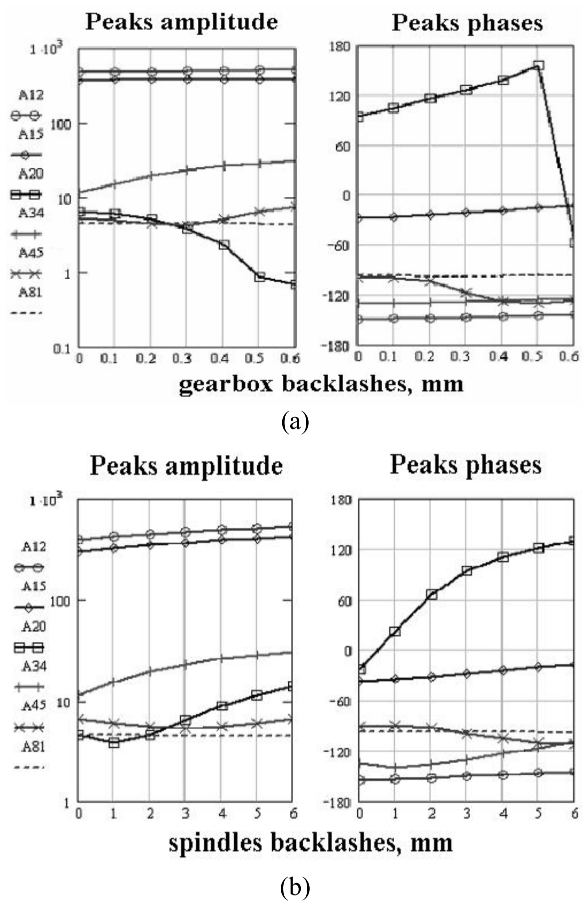


Figure 10. Natural frequencies and wear relations: (a) Natural frequencies by the gearbox wear (b) Natural frequencies by the spindles wear

Diagnostics algorithms based on combination the amplitudes and phases at the different natural frequencies. In this case variable A20 Hz and its phase are most sensitive to wear by torque signal. After the 2.5 mm wear in spindles A45 Hz amplitude becomes less than A20 Hz. In such manner other variables may be analyzed to build diagnostics algorithms which differ for other points of torque or vibration measurement [Krot et al., 2007].

8 Bearings and bolting diagnostics

Radial wear and backlashes are the most important maintenance parameters as it cause during transient processes shafts angular motion and significant strains rising on marginal parts of teethes. Beside it bearing's housing bolts have plastic deformation due to shock vibration. The screwing up is the standard maintenance operation for the rolling mills gearboxes and pinion stands. All nuts are fixed by welding but the gap appears due to screws plastic deformation. Therefore it is an important problem to diagnose the bearings radial backlashes and the housing gap opening.

The calculation scheme for shaft, bearings and housing nonlinear vibrations is represented in the Fig. 11.

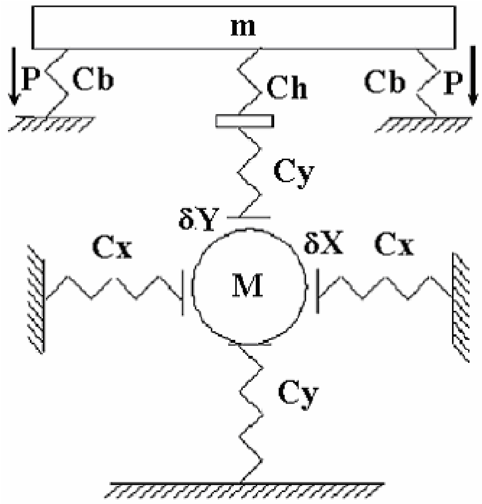
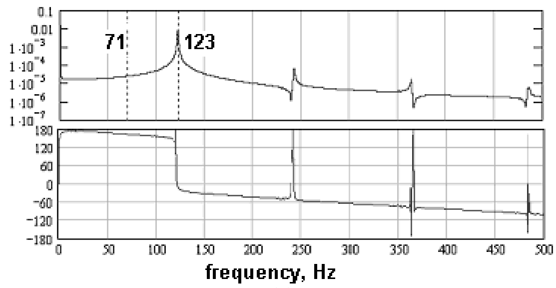


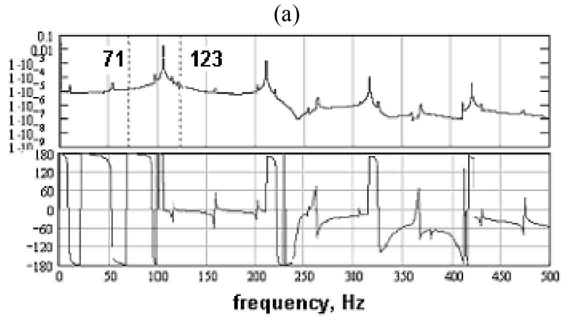
Figure 11. Calculation scheme of shaft, bearing and housing interaction.

There it is denoted: Cy, Cx – vertical and horizontal bearing nonlinear stiffness; δX, δY – radial backlashes, Cb, Ch – bolting and housing nonlinear stiffness; m, M – housing and half shaft masses; P – forces of every side bolting.

It was investigated the influence of bearing radial wear (0.2…0.6 mm) and housing gap (0.05…1.05 mm) due to bolting deformation. The calculated range of the shaft main natural frequency was about 71…123 Hz for such wear. Vibration signal point was on the shaft housing in vertical direction. Amplitude and phase diagrams are represented in Fig. 12.



(a)

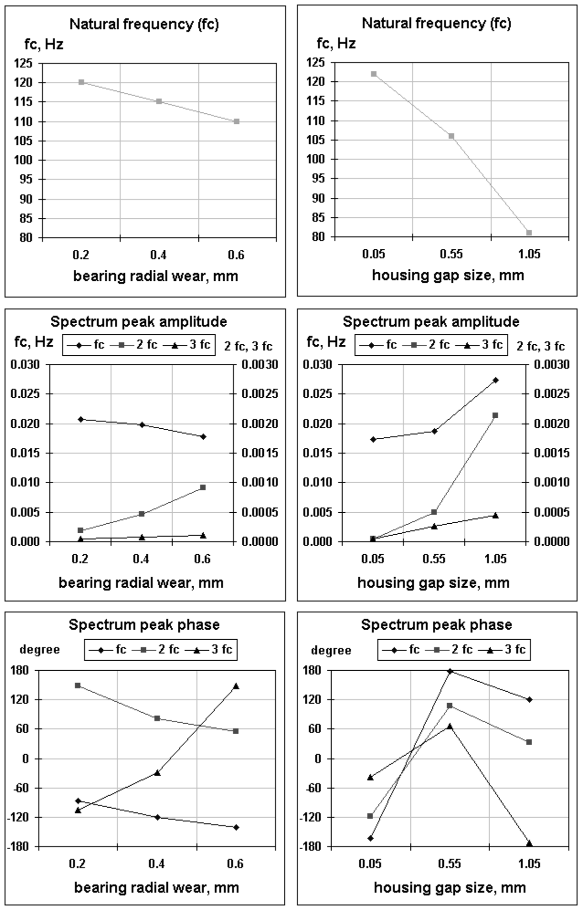


(b)

Figure 12. Bearing natural frequency shift due to wear: (a) Bearing wear - 0.2 mm (b) Bearing wear - 0.4 mm

The main natural frequency, spectrum amplitude and phase graphs for three bearings wear and housing gap see in Fig. 13 with the same scale. High harmonics amplitudes of main frequency have another scale (on the right side).

Bearing backlash is the first stage of wear. The next one is the housing gap opening which may cause teethes fraction. During the slow bearing wear its natural frequency has less decreasing than for bolting plastic deformation. The main frequency amplitude falls down and high harmonics rising up (see Fig. 13(a)). During the housing gap opening more than 0.55 mm all phases have fracture point (see Fig. 13 (b)).



(a) (b)

Figure 13. Diagnostics functions for bearing and bolting wear:
(a) Bearing wear 0.2-0.6 mm
(b) Housing wear 0.05-1.05 mm

All above mentioned regularities depend on spring-mass parameters of gearbox shafts, bearing, housing and torsional load.

Beside the frequency domain it is useful for diagnostics purposes to obtain time domain trajectory of shaft center during the transient process (see Fig. 14). The initial position of shaft was down. The pieces of trajectory which are out of geometrical circle mean bearing deformation and possible damages if it more than the elastic limit.

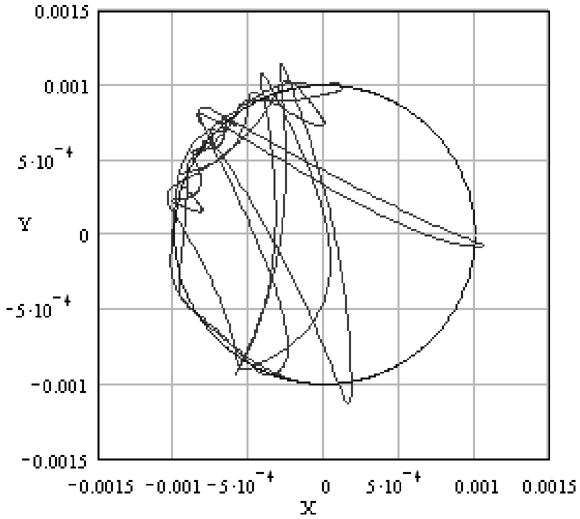


Figure 14. Calculation results of shaft center moving with the bearing and housing wear (±0.001 m).

Also model gives the possibility to build the relation of radial dynamical loads in the bearing and housing by the wear. For example see Fig. 15, where the certain limit of load (about 1000 kN) will be overrun at the 1.8 mm wear of bearing. It allows the prediction of maintenance period for the rolling mills gearboxes and pinion stands.

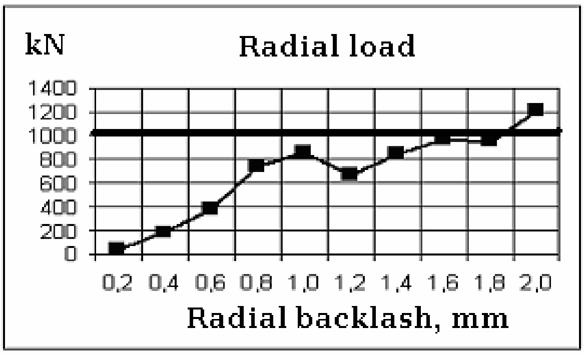


Figure 15. Radial load in the bearing by the wear.

It was shown by the calculations that radial backlashes have the same influence on torsional nonlinear vibration as the angular ones. The most dynamic loads at the transient process appear when the shaft weight force and teeth coupling reaction have opposite directions.

The future topics of research are the rolling stands housing wear and work rolls bearing defects diagnostics.

9 Conclusions

Angular wear and radial backlashes are the most important parameters of rolling mills drive trains condition and maintenance.

Torsional nonlinear vibration during the step-like rolling load transient process may be used for wear diagnostics in time domain and in frequency domain.

The main idea of backlashes diagnostics is to compare dynamic response of real system with the signals simulated by the linear model.

It is necessary to take into account TAF by the Tstatic relation for the durability calculations moreover with the low torsional loads.

Nonlinear torsional vibration properties such as static load and dynamic torque interrelation, non isochronisms are the features used for backlashes diagnostics.

Natural frequencies and its higher harmonics amplitudes and phases are used in algorithms for bearing backlashes and bolting wear diagnostics.

Designed model allows calculating the radial dynamical loads and damages in the bearings and housing by the wear. It helps to predict maintenance period for rolling mills gearboxes and pinion stands.

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