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DYNAMICS AND DIAGNOSTICS OF THE ROLLING MILLS DRIVELINES WITH NON-SMOOTH STIFFNESS CHARACTERISTICS

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ABSTRACT

The rolling mills drivelines are under investigation. Their dynamics is characterized by the extremely high torque amplification factors. The main problem is to identify non-smooth piece-wise linear stiffness characteristics in the multi-body system produced by the angular and radial backlashes. Different smoothening functions are analyzed. Frequency domain is used to determine amplitudes and phases of natural frequencies and harmonics due to angular and radial backlashes appearance. It was shown that interrelations between the static torsional loads and dynamic responses can be utilized for diagnostics.

INTRODUCTION

The rolling mills drivelines are operated under the extremely high loads and simultaneously are characterized by the increased wear. Backlashes and step-like impulse loads during the hot metal rolling cause the most frequent failures in the drivelines. Standard methods of vibration diagnostics based on envelope curve spectrum analysis require stationary drive speed and load for signal averaging. It is quite difficult to provide constant load in the rolling mills because of metal temperature and friction forces 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 avoiding inconveniences of standard diagnostics methods.

Different kinds of backlashes in the rolling mills equipment are given in the Fig.1, where following notation are used: F – force (torque), δ – generalized coordinate. Function like in the Fig.1a describes clearance with a dead zone in the driveline couplings caused by wear or assembly errors (both positive and negative δ). In general, transient process calculation supposes that the opened part of gap b may not be equal to a closed part a (the whole gap is $a+b$). The Fig.1b describes "softening" stiffness function for bearings, housing and bolting (positive δ). The fracture point means stiffness decreasing when gap is opening between gearbox housing and bearing cover under the action of severe shock torsional vibrations. A "hardening" function in Fig.1c shows the 4-high stand rolls stack vertical stiffness for low and high rolling loads (positive δ).

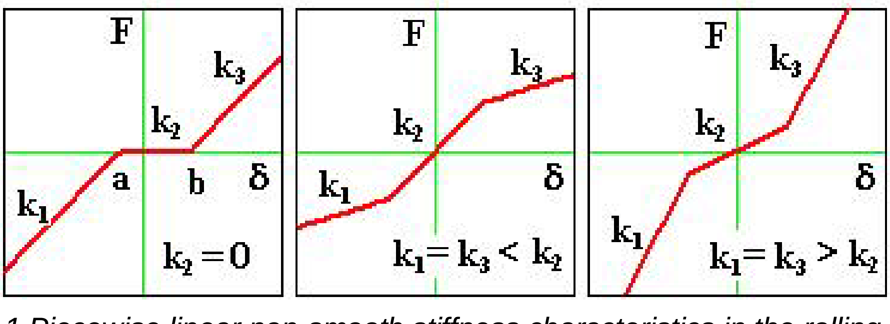

Fig. 1 Piecewise linear non-smooth stiffness characteristics in the rolling mills

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Angular wear increases torque amplification factor (TAF) in all driveline 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 technical condition. The prognostic maintenance based on gaps diagnostics is complicated because of short period of transient process when the work torque is appearing and the backlashes are closing. Under the full load it is difficult to determine the backlashes exactly by the modulation characteristics of the vibration signal.

1. DRIVELINES DYNAMICS

An analysis of clearance non-linearity and vibration impacts in torsional systems was conducted in [1] and in many other works. Unfortunately, there are a few works on rolling mills drivelines having certain particularities.

The hot rolling mills spring-mass model (see Fig.2) 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). System has a branched structure due to spindles.

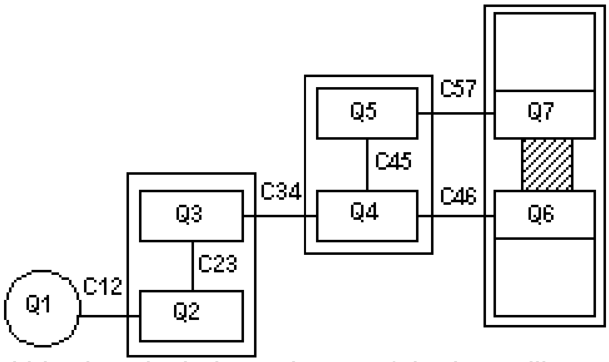

Fig. 2 The multi-body calculation scheme of the hot rolling mill (one stand)

The WR's coupling via the rolled strip is not considered during transient process simulation. The every coupling has an uncertain part of gap opened before transient. First of all, it concerns spindles, which are, usually, under- or over-weighted because of imperfect spring type balancing units. Even ±5% error of spindles weight (0.2-2 KN) tuning will cause their periodical (twice per rotation) vertical motion and gaps opening. Beside it, steel strip higher velocity than the WR linear velocity will also produce gaps in the spindle couplings and pinion stand.

The analytical research of nonlinear multi-body systems, usually, assumes reducing the initial system to less DOF. That is possible, but inefficient, if the every coupling diagnostics is required. Then, those coupling is necessarily even, if its contribution to overall dynamics is not significant.

A well known earlier fact that backlashes, when are opening, cause high frequency vibrations, was also confirmed by this research. But it was cleared up what the frequencies exactly appear in the signals of torque and bearings vibration and the how they could be used for backlashes diagnostics. It has been shown that higher natural frequencies associated, but not equal, with partial frequencies of the torsional system will appear in torque spectrum. Also, as it is known, the higher harmonics of the main frequency will be produced by nonlinearities. Such regularities have been taken into consideration for wear diagnostics methods. The main idea is to compare linear system as reference with a response of a nonlinear real driveline in the range of natural frequencies.

1.1 Static load influence on dynamic response

Torque amplification factor (TAF) is the main parameter, usually, used for system dynamics estimation. However, for nonlinear systems, dynamic response depends on static load (rolling torque). It was shown (see Fig. 3) 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, is not taken into account for the durability calculations in the rolling mills.

The field torque measurements were fulfilled in the industrial hot rolling mill in the corresponding motor shaft. The newly designed in the Iron and Steel Institute 8-channel telemetry system was utilized. 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. Results of torque measurements are represented in Fig. 4 where we can see the same nonlinear relation between static load Mst and dynamic response Mmax and TAF.

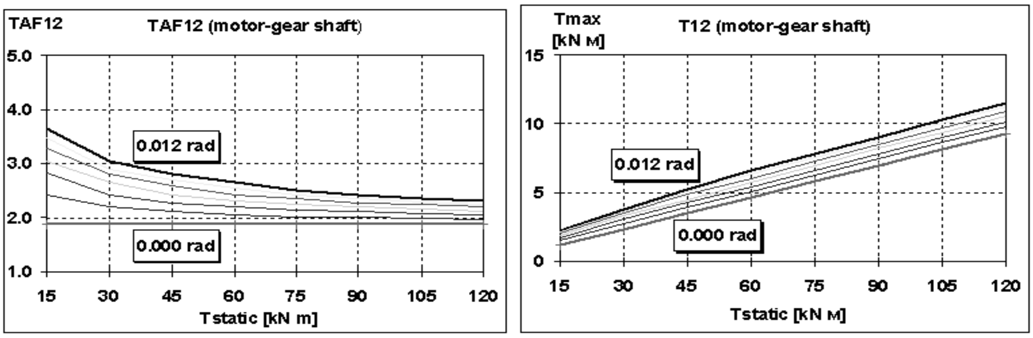

Fig. 3 Static torque influence on TAF and peak torque Tmax in motor shaft (C12) (Simulation)

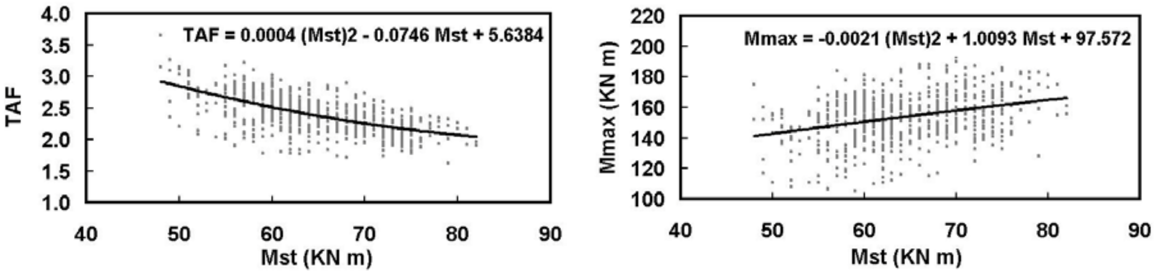

Fig. 4 Static torque influence on TAF and peak torque Mmax in motor shaft (C12) (Telemetry measurements with the strain gauges)

1.2 Smoothening functions

Because of backlashes are described by a non-analytical and non-differential discontinuous function of logical type (see below), it worsens model numerical simulations. Therefore, some smoothening functions were analyzed and issues of their implementation were discussed. They are as following (see Fig.5):

1. Logical type: $$g_1(\delta) = \begin{cases} k_3(\delta - b) + k_2 b \Rightarrow b < \delta \\ k_2\delta \Rightarrow -a \le \delta \le b \\ k_1(\delta + a) - k_2 a \Rightarrow \delta < -a \end{cases}$$

2. Polynomial type: $$g_2(\delta) = a_1 \cdot \delta + a_2 \cdot \delta^2 + a_3 \cdot \delta^3$$

3. Arc-tangent type: $$g_3(\delta) = \delta \cdot a_0 \cdot \arctan(\sigma \cdot |\delta|)$$

4. Hyperbolic-tangent type: $$g_4(\delta) = \delta \cdot a_0 \cdot \tanh(\sigma \cdot |\delta|)$$

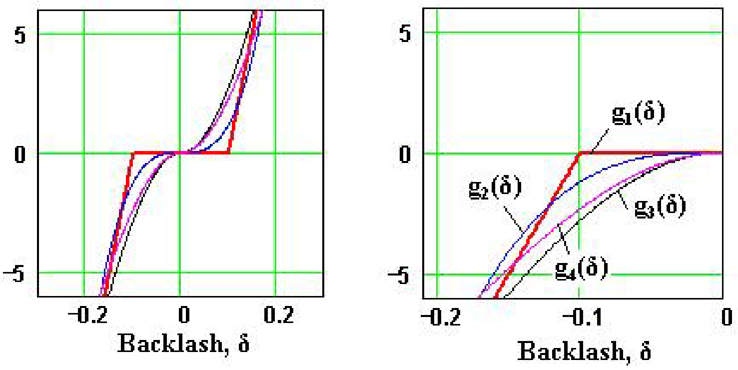

Fig. 5 Non-smooth stiffness characteristic approximation with continuous functions

Adjusting of the approximation functions $g_1(\delta)$-$g_4(\delta)$ is carrying out by the following parameters a_0, a_1, a_2, a_3, σ. Although a flexibility of such functions, there are certain restrictions for their implementation. Only small dimensionless values of gaps (0.01-0.1) and stiffness (10-100) combinations are available for accurate approximation. Otherwise, scaling factors for transition to model parameters should be introduced to get acceptable accuracy. A linear component (a_1) in $g_1(\delta)$ is responsible for gap size, while the cubic component (a_3) for stiffness approximation. The values $a_1<0.01$ do not affect curvature near the zero point. A square component (a_2) gives possibility to simulate with a polynomial $g_2(\delta)$ function the asymmetry in gap opening conditions (it corresponds to $a \neq b$ in $g_1(\delta)$). Coupling preloading conditions are also available due to square component (a_2) in $g_2(\delta)$ when symmetry point is shifted beyond the initial point of coordinates. However, when $a_2 > \sqrt{3 \cdot a_1 a_3}$ or $a_1 < 0$, two points - maximum and minimum - appear in the $g_2(\delta)$ graph instead of one saddle point. That has to be taken into account during parameters tuning.

The every coupling in the driveline with its unique gap and stiffness values requires special function for approximation. Right function choice depends not only on gap and stiffness values, but is also related to actual torque amplitudes. For example, polynomial type function $g_2(\delta)$ is more accurate near the fraction points of stiffness curve (within the $\pm 2(a+b)$ range), then, it crosses the original function $g_1(\delta)$ and begins to deviate significantly from it. On the other hand, the arc-tangent $g_3(\delta)$ and hyperbolic $g_4(\delta)$ functions (the are similar in behavior) are more accurate for large amplitudes far from fracture points (beyond the $\pm 2(a+b)$ range). So, there are no general recommendations for any cases.

The interaction of a_0 and σ is not fully understood and actual limits have not yet been determined definitely. For sure, a smaller σ value corresponds to smaller stiffness, but smaller a_0 fits larger gap. The larger the σ value, the closer is the approximated $g_3(\delta)$, $g_4(\delta)$ curve to the original piecewise linear function $g_1(\delta)$.

1.3 Frequency domain analysis of smoothening functions

Effect of smoothening functions on the frequency response of an oscillator with clearance non-linearity was investigated in [2]. The Nonlinear Identification through Feedback Outputs (NIFO) technique was also used in [3] to estimate the nominal linear FRFs for SDOF system using three different generating functions to describe the modulation in frequency response: $\Delta y^{p+(n/m)}$, where Δy is the relative motion across the nonlinear element and n and m are integers such that $m > n$. In order to estimate here the influence of different approximation functions (g_1 - g_4) on frequency response functions (FRF), some calculations were carried out on a SDOF system (see Fig.6).

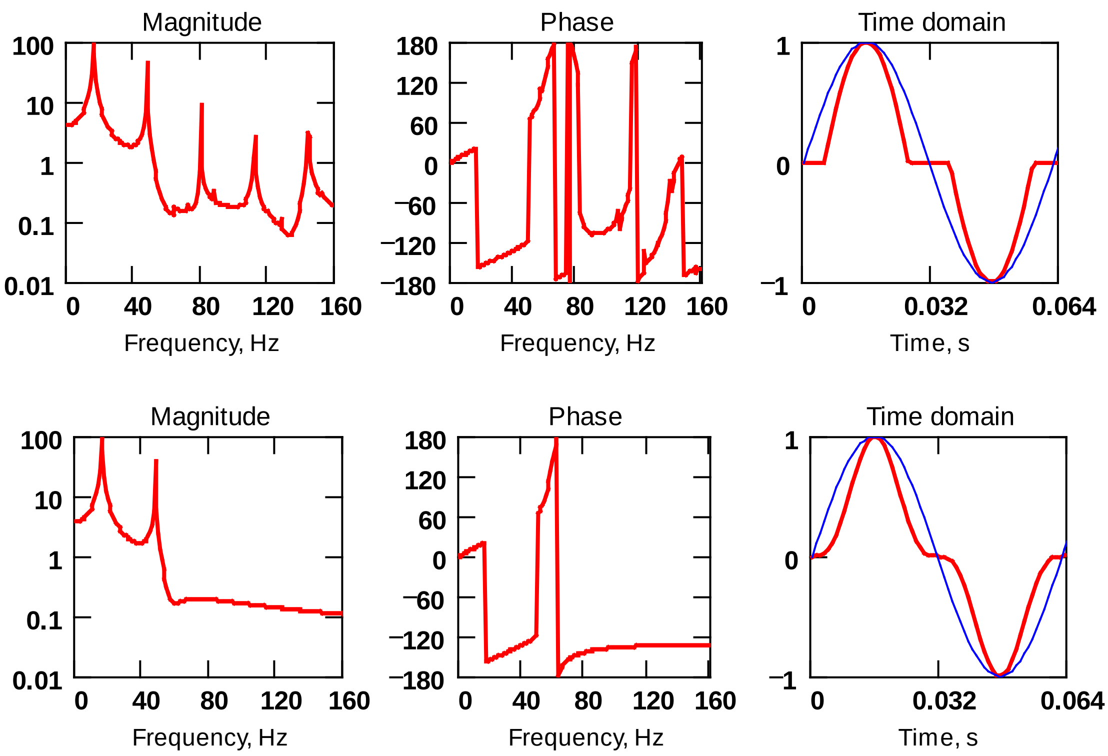

Fig. 6 FRF and time domain signal for g_1 (logical) and g_2 (polynomial) functions

We can see that piecewise function generates more harmonics and irregular phase spectrum. The same output exhibit g_3 and g_4 functions (not shown). Polynomial function gives only two harmonics in FRF and clearer phase portrait. Again, it depends on actual amplitudes of torque and gap value. Than the more trajectory beyond the stiffness fracture points (larger amplitudes), the less influence of nonlinearities. That coincides with experimental results.

2. BACKLASHES DIAGNOSTICS.

The torsional natural frequencies, calculated on investigated rolling stand driveline, were as following: 12, 15, 20, 34, 45 and 81 Hz. As it was expected, measured torque signal had peaks at the lower frequencies and the vibration had them in the higher range. Spectrum peaks amplitudes (A12…A81 Hz) and according phases at the natural frequencies (see Fig. 7) were taken for different gearbox and spindles wear (angular backlashes) by the torque signals.

2.1 Angular gap diagnostics

Diagnostics algorithms are 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 in 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 measurement.

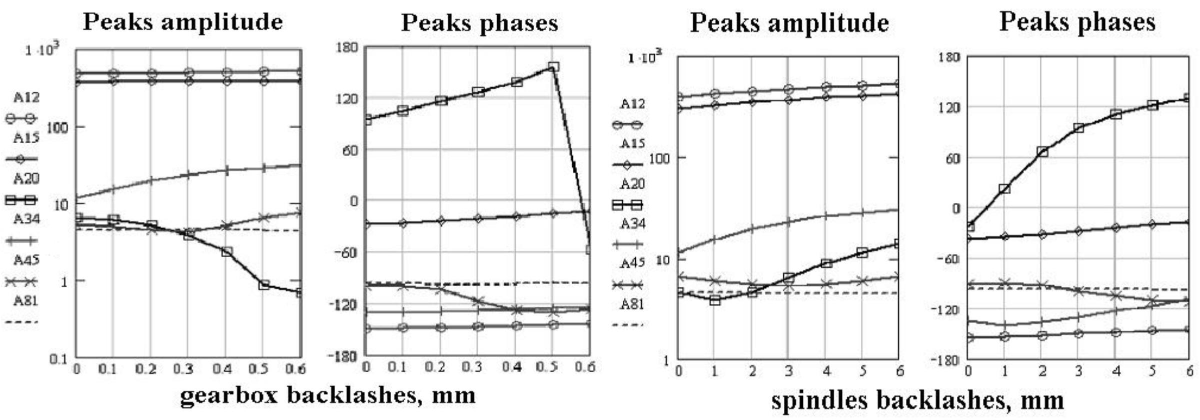

Fig. 7 Variation of the torsional natural frequencies and phases due to backlashes in the gearbox and spindles

Calculation of FRFs is not a problem for analytical linear models. On the other hand, experimental time-domain data converting into the frequency domain and corresponding FRFs, depending on the types of external excitation, may not be perfect over the whole frequency range. For example, for the transient impacts, signal-to-noise ratios tend to be poor, but this will mostly affect the FRFs at non-resonant frequencies. Another difficulty in FRF comparison is that FRFs are very sensitive to sensors and exciters placement. That restriction is avoided due to a naturally constant strip impact in the WRs and, more importantly, stable position of torque sensor on the shaft.

2.2 Bearings diagnostics

Radial wear and backlashes are the most important maintenance parameters because they cause transient shaft motion within gaps and significant strains appear on the marginal parts of teeth. 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. Therefore it is important to diagnose the bearings radial backlashes and the housing gap opening.

The calculation scheme of shaft, bearings and housing nonlinear system is represented in Fig.8a. Bearing backlash is the first stage of wear. The next stage is the housing gap opening, which may lead to teeth fraction. During the bearing wear its natural frequency is decreasing slowly. Then, bolting plastic deformation begins and the main frequency amplitude falls down. Also, higher harmonics appear. During the housing gap opening phases of all harmonics change significantly.

Beside the frequency domain it is useful for diagnostics purposes to obtain time domain trajectory of shaft center during the transient process (see Fig.8b). The initial position of shaft was in the bottom. The places where trajectory is out of geometrical circle correspond to bearing deformation and possible damages, if it exceeds the elastic limit of material. It gives a possibility to determine a relation of radial dynamical loads in the bearing and housing by the wear. For example, a certain limit of load (about 1000 KN) will be overrun at the 1.8 mm wear of bearing (see Fig.8c). It allows predicting a maintenance period for the rolling mills gearboxes and pinioning stands.

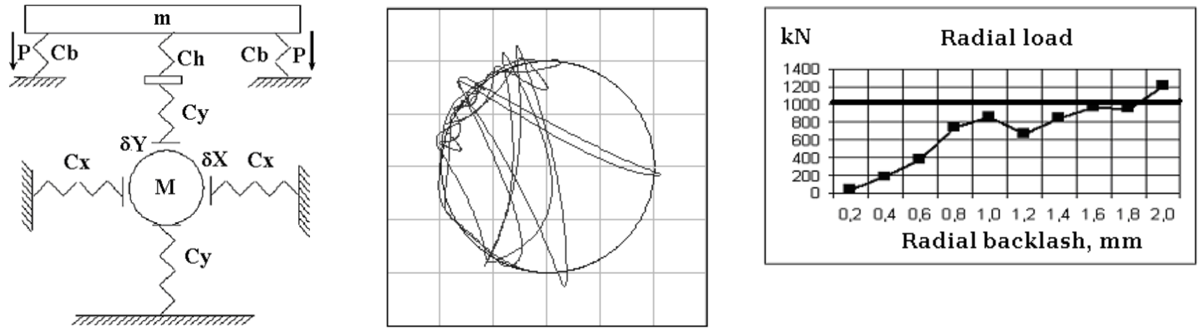

Fig. 8 Calculation scheme of shaft-bearing-housing nonlinear system, shaft motion trajectory and maximal radial loads on bearing

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

CONCLUSIONS

Angular and radial backlashes are the most important parameters of the rolling mills drivelines technical condition and maintenance. Transient torsional nonlinear vibration initiated by the step-like rolling load may be used for wear diagnostics as in time domain, so 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. Some features of nonlinear torsional vibration, such as static load and dynamic torque relation, may be used for backlashes diagnostics. Natural frequencies and their higher harmonics amplitudes and phases are used in algorithms for driveline, gearboxes bearings and bolting diagnostics. The smoothing functions, in general, affect those frequency response regimes that are influenced by the stiffness curves fractions. However, the peak values and TAF during transient appear to be insensitive to the choice of smoothing function.

REFERENCES

[1] T.C. Kim. *Analysis of Clearance Non-Linearities and Vibro-Impacts in Torsional Systems*, PhD. Thesis, Ohio State University, 2003.

[2] T. C. Kim, T. E. Rook and R. Singh. Effect of Smoothening Functions on the Frequency Response of an Oscillator with Clearance Non-Linearity *Journal of Sound and Vibration*, 2003.

[3] D. E. Adams and R. J. Allemang. Polynomial, Non-Polynomial, and Orthogonal Polynomial Generating Functions for Nonlinear System Identification, *ISMA2000*, Belgium, 2000.