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# VIBRATION DIAGNOSTICS OF ROLLING MILLS BASED ON NONLINEAR EFFECTS IN DYNAMICS

This paper intends to describe nonlinear effects occurring in rolling mills dynamics. That is necessarily for vibrations damping and reliable diagnostics of rolling mills equipment under non-stationary working conditions. Three types of nonlinear effects are investigated taking place in drivelines and stands of different design, namely, transient torsional vibrations in hot rolling mills, chatter vibrations in tandem cold rolling mills and parametrical vibrations in high-speed wire and rod rolling mills. The procedure is proposed for natural frequencies identification when short transient torque signals restrict application of the Fourier transform. Examples are given on using nonlinear effects for wear diagnostics and vibrations control based on natural frequencies and modes analysis of multi-body systems.

Keywords: Rolling mills, nonlinear oscillations, chatter vibrations, identification

**Introduction.** The different types of rolling mills for production of steel strips, rods and tubes belong to the most dynamical large-scale industrial plants. A wide variety of drivelines designs are formed by multistage gearboxes, pinion stands, universal spindles, switchable couplings and the huge number of rolling stands types with 2, 4 and up to 20 rolls including compact rolling mills (blocks) with both vertical and horizontal rolls for wire and rod production.

Standard maintenance procedures usually implemented for majority of rotating machines (balancing, shafts alignment) are not applicable for rolling mills due to frequent changes of rolls and technological schedules. Harsh operating conditions of rolling mills affect vibration parameters and diminish implementation of standard diagnostics methods where loads and speeds are supposed to be stationary. That makes very important to understand the causes of different dynamical processes occurring under variable working conditions.

Research of nonlinear effects in rolling mills dynamics covers a wide range of domains in mechanics theory including elastic-plastic behavior of deformed metal, nonlinear torsional vibrations, chatter vibrations synchronization and parametrical axial vibrations of rolled rods in the multi-stand tandem mills where neighboring stands coupled by rolled metal. Interrelation of several nonlinear effects, which cause high dynamics, complicates the development of vibration damping methods in certain type of mills and appropriate diagnostics procedures.

This research intends to represent a systemized overview of nonlinear effects observed in steel rolling mills of different types. Three problems in rolling mills dynamics are discussed: 1) transient torsional oscillations in drivelines of hot rolling mills subjected to excessive wear (angular and radial backlashes); 2) chatter vibrations in continuous cold rolling mills; and 3) parametrical oscillations in high-speed compact reducing sizing mills with multistage gearboxes which structure is controlled by switchable servo couplings. The possible methods are proposed for rolling mills equipment diagnostics and resonance modes detection for vibration damping based on analysis of nonlinear dynamical effects.

**1. Transient torsional vibrations in the drivelines of hot rolling mills.** The drivelines of hot rolling mills are operated under high specific loads, harsh conditions and are characterized by increased wear. Backlashes as bilinear stiffness (Figure 1a) and frequent step-like loads cause the most severe failures in the drivelines. The typical geared driveline of hot rolling mill includes (Fig. 1, *b*): 1 – rolled strip; 2 – work rolls; 3 – spindles with the sliding pads or universal joints; 4 – pinion stand splitting drive torque; 5 – intermediate coupling; 6 – gearbox; 7 – motor shaft coupling; 8 – electric drive (5-12 MW).

Quick wear causes angular and radial backlashes which are opened before transient process for uncertain part $\delta_1$, then, coupling stiffness become equal to tg($\beta$).

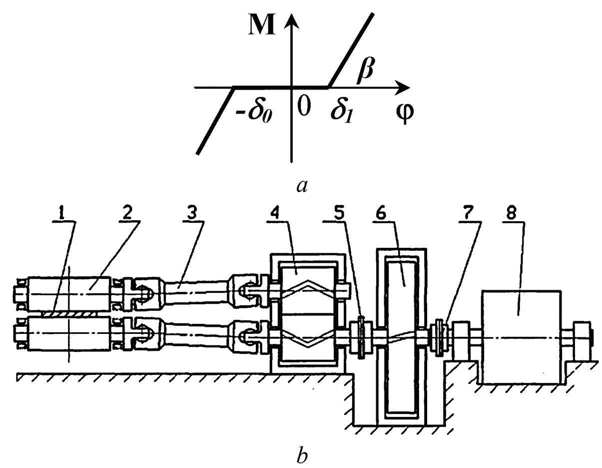


Figure 1 – Bilinear stiffness characteristic (*a*); geared driveline of the hot rolling mill (*b*)

Standard methods of vibration diagnostics [1], e.g. based on envelope spectrum analysis, require stationary drive speed and work load for defects recognition. It is almost impossible to provide constant conditions because of metal temperature and friction variation in the deformation zone. To solve these problems, a new approach is proposed for wear diagnostics based on detailed models (Fig. 2, *a*) of drivelines and their analysis in the range of natural frequencies of vibration [2]. Its implementation requires enough accurate identification of natural frequencies by the short and noised transient signals in order to compose reliable diagnostic rules.

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**1.1 Natural frequencies identification for short transient vibrations.** Torsional moments of inertia are determined by the equipment drawings and referred to the rolls rotation speed. To determine the elasticity coefficients with sufficient accuracy by the drawings is virtually impossible, because coupling parts have a complex shape, and there is a natural variation in modulus values. It is proposed to determine the elastic constants by the known analytical relations of the first and second harmonics of torsional vibration in the 3-mass system (Fig. 2, *b*).

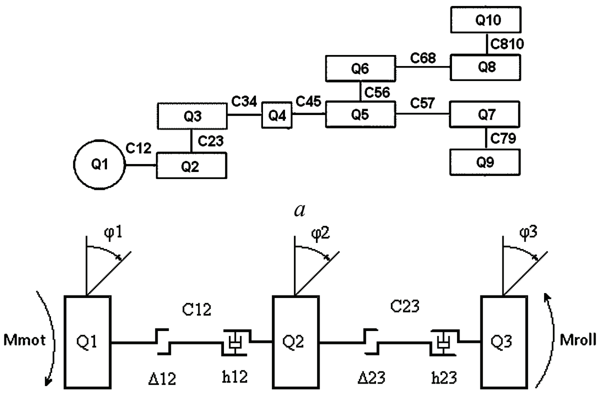



Figure 2. Design schemes of multi-body driveline of rolling mill: *a* – full; *b* – reduced to 3-mass model;
(Qi – the moment of inertia of rigid components; Ci – the stiffness of the massless elastic couplings; Δii – gaps; hii – damping coefficients; φi - angular displacement; Mmot, Mroll – motor and rolling torques)

Torque signals can be represented by the following expressions:

$$\begin{aligned} &F(t) = f_1(t) + f_2(t) + f_3(t);\\ &f_1(t) = (a_1 \cdot \cos(a_5 \cdot t) + a_3 \cdot \sin(a_5 \cdot t)) \cdot \exp(a_7 \cdot t);\\ &f_2(t) = (a_2 \cdot \cos(a_6 \cdot t) + a_4 \cdot \sin(a_6 \cdot t)) \cdot \exp(a_8 \cdot t);\\ &f_3(t) = a_9 \cdot (1 - \exp(a_{10} \cdot t)), \end{aligned} \qquad (1)$$

where $f_1(t)$ and $f_2(t)$ – 1st and 2nd natural frequency modes of damped oscillations, $f_3(t)$ – exponentially increasing rolling force (input load), $a_1...a_{10}$ – coefficients of approximation (Table 1).

Table 1 – Coefficients of transient signal approximation

| $a_1$ | $a_2$ | $a_3$ | $a_4$ | $a_5$ | $a_6$ | $a_7$ | $a_8$ | $a_9$ | $a_{10}$ |
|---|---|---|---|---|---|---|---|---|---|
| -147 | 85 | 191 | -31 | 11,99 | 20,91 | 22,17 | 3,65 | 112 | 36,7 |

Approximation required when short records of real transient process restrict application of the Fourier transform to determine natural frequencies. The useful parameters from the Table 1, reflecting the transient process, are: $a_5$, $a_6$ – first and second natural frequencies of the driveline (Hz), $a_9$ – static value of rolling torque (kN m), $a_{10}$ – rate of the rolling torque rise (characterized by rolls speed and metal reduction during the transient process).

Comparing the values of natural frequencies obtained from the equipment drawings and experimental data (Table 2), a notable difference can be seen in $\beta_1$ 16,66…12,03 Hz and $\beta_2$ 25,98…21,09 Hz.

The influence of moments of inertia on the natural frequencies is significantly lower than the influence of the stiffness. In turn, the stiffness (elasticity) is difficult to determine accurately, since parts often have a complex shape, different material components (different values of modulus), but it significantly affects the values of natural frequencies. Adjusted values of stiffness are determined by using the known relations of natural frequencies with parameters of 3-mass vibrating system:

$$\begin{cases} \beta_1 = \dfrac{1}{2\pi} \cdot d \cdot \sqrt{a-b};\\ \beta_2 = \dfrac{1}{2\pi} \cdot d \cdot \sqrt{a+b}; \end{cases}$$

$$a = (f + e \cdot c);\quad b = \sqrt{(f - e \cdot c)^2 + c}; \qquad (2)$$

$$c = \frac{C_{12}}{C_{23}};\quad d = \sqrt{\frac{C_{23}}{Q_2}};\quad e = \frac{Q_1 + Q_2}{2 \cdot Q_1};\quad f = \frac{Q_2 + Q_3}{2 \cdot Q3}.$$

Knowing the initial parameters $Q_1$, $Q_2$, $Q_3$ – calculated from drawings and $\beta_1$, $\beta_2$ – defined from real signals recordings, the adjusted values of stiffness $C_{12}$ and $C_{23}$ (Table 2) are determined by solving numerically the system of equations (2). Results of simulation are represented in Fig. 3.

Table 2. Initial and enhanced values of natural frequencies of 3-mass driveline

| Parameters | $Q_1\times10^3$, kg·m$^2$ | $Q_2\times10^3$, kg·m$^2$ | $Q_3\times10^3$, kg·m$^2$ | $C_{12}\times10^6$, N·m/rad | $C_{23}\times10^6$, N·m/rad | $\beta_1$, Hz | $\beta_2$, Hz |
|---|---|---|---|---|---|---|---|
| Calculated | 46,70 | 11,85 | 2,48 | 214,10 | 30,68 | 16,66 | 25,98 |
| Adjusted | | | | 68,77 | 32,81 | 12,03 | 21,09 |

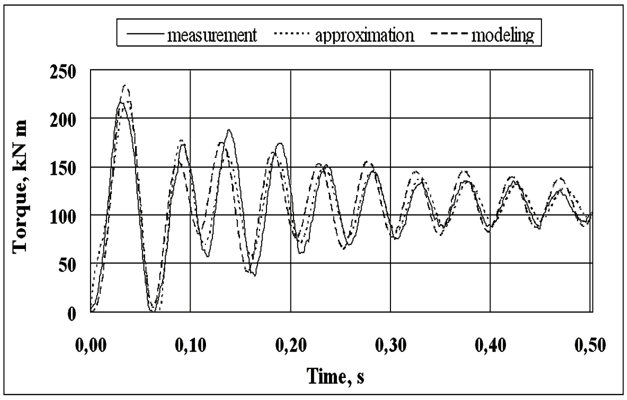



Figure 3 – Transient torque modeling with enhanced values of natural frequencies

The most significant factor affecting approximation is the non-linear behavior of torque when it crosses zero level. The phase shift appears in signals at this moment (after first period of oscillation) due to gaps opening and closing. Therefore, approximation should be restricted by logical conditions or measured data should be taken when all samples are above zero level.

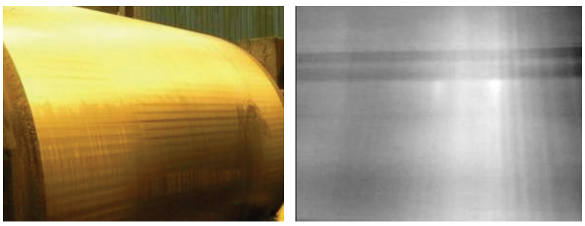
**1.2 Using transient torque signals for wear diagnostics.** Abovementioned nonlinear effects during transient processes can be used in wear diagnostics to overcome problems in non-stationary signals. For example, it can be used the difference of the first (second) period of transient signals for diagnostics purposes. Vibration signal needs low-pass filtering before analysis. The low-pass filter cutting frequency should be twice more than highest natural frequency of the oscillating system [2].

Torque amplification factor (TAF) is one of the main parameters in dynamic systems analysis. However, dynamic response of nonlinear systems depends on static rolling torque. It was shown that mill driveline exhibits higher TAF with decreasing static torque for different angular wear. This nonlinear effect is also used in driveline wear diagnostics (couplings, bearings). The distinctive advantage of proposed approach is that instead of full gaps, only their opened part can be identified which has influences on driveline dynamics. It was shown that radial gaps in bearings have the same influence on TAF as the angular ones. The highest TAF corresponds to opposite direction of shaft weight and teeth coupling reaction. The response of linear dynamic model is taken as reference values for wear diagnostics.

**2. Chatter vibrations in the tandem cold rolling mills.** Since the end of last century, chatter vibrations phenomena in the high-speed cold rolling mills is still intensively investigated because it significantly (by 20-30%) reduces annual plant productivity and strip quality. The most advanced tendencies in this domain of research were discussed in [3] and other studies. The main reason of vibration amplification in the tandem mills is considered the regeneration effect due to periodic variation of strip thickness and roughness [4-8] as well accounting viscoelastic properties of deformed steel strip [9].

Strategies of tandem mills control are based on modern approaches including adaptive models, fuzzy logic, neural nets for parameters prediction to meet very high demands on steel strips flatness, roughness and thickness tolerance (<5 microns). Nevertheless, sensors and actuators in conventional Automatic Gauge Control (AGC) systems because of narrow pass band (<10 Hz) are not able to control high frequency chatter vibrations. Nowadays, the only practical way to cancel chatter and prevent strip break is to drop down the mill speed by the alarms from vibration monitoring system. Such control impacts reduce overall mill productivity and affect strip quality due to transient processes excited in the drivelines.

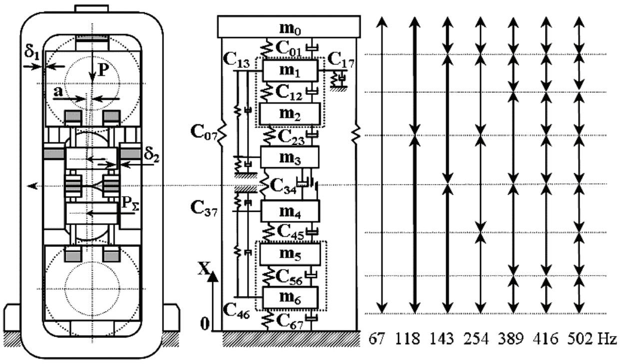
Chatter vibrations were investigated in the 5-stands high-speed 4-high cold rolling mill 2030 (rolls length in mm). The real-time chatter vibration monitoring and diagnostics system was developed for early detection of periodic defects on rolls (Fig. 4, *a*) and strip (Fig. 4, *b*).



*a* *b*

Figure 4 – Chatter marks on rolls (*a*) and strip (*b*)

It was shown by the detailed FEM analysis that 4-high rolling stand cumulative mass fraction is about 0.80-0.85 for the first 4-9 modes [10]. The composed model (Figure 5a), unlike other known studies, includes the work rolls bending units and backup rolls balancing units which are accounted as corresponding stiffness instead of external forces. Such approach allows fulfilling a dynamic analysis of the 4-high stand under the variable operating schedules. It is assumed symmetrical vibration for both service and drive sides of stand. Beside it, horizontal stability is analyzed of rolls chocks within the gaps $\delta_1$, $\delta_2$ and displacement $a$ within stand housing under the rolling force $P$ (Fig. 5, *a*). The effect of chatter regeneration due to strip thickness variation is investigated along with associated natural modes of stand vibrations (Fig. 5, *b*).




*a* *b*

Figure 5 – Dynamic model of 4-high stand with rolls bending units (*a*); natural modes of roll stack vibrations (*b*)

There are studies devoted to chatter regeneration in cutting tools dynamics [11]. The similarity of cutting machines and tandem rolling mills is that material is moving from one stand to subsequent stand and periodic defects have the influence with certain delay, but rolling parameters affect this delay significantly. Investigations of regeneration effect in rolling mills are mainly based on linear models or parametric excitation of strip due to variable stiffness in the rolls gap.

In distinction of known approaches, reliable chatter detection can be provided due to the commonly known physical phenomenon – synchronization of the mechanical oscillators via the couplings between them. The rolling stands of tandem mill are synchronizing by the elastic strip. Synchronization is accompanied by the effect of involving of close natural frequencies of adjacent stands and drift of their phases of vibration to certain value, identical in all stands. This effect results in decreasing of frequencies deviation from average value and increasing the correlation coefficients in adjacent stands. Frequencies and phases of chatter vibration begin to change even for minimal energy exchange between stands. Two main conditions for chatter early detection were derived as:

$$\sigma_n \le \sigma_{\min}; \quad K_{mean} \ge K_{\max}, \qquad (3)$$

where $\sigma_n$ – current root mean square (RMS) deviation of the principal modes frequencies in the adjacent stands within chatter frequency range (Hz); $\sigma_{min}$ – minimal difference equal to spectrum frequency step (Hz); $K_{mean}$ – mean correlation coefficient of vibration in the *n-1* pairs of adjacent stands; $n$ – number of stands; $K_{max}$ – maximal correlation coefficient for steady rolling without chattering ($K_{max}$ < 0.2-0.3).

**3. Parametrical oscillations of rolled rod in high-speed compact mills.** Nowadays, many enterprises producing long products like wire and rods, subjected to effect of vibration excitation in the high-speed compact mills with complicated geared drives. Increasing working speed above 110 m/s is limited by high probability of incidents due to vibrations in the constrained space between pairs of rolls (twisted by 90° in adjacent stands) and the subsequent loss of longitudinal stability.

The problem also lies in the fact that rolling mills designed by Morgan Construction Company (USA), in contrast to all previously studied mills, has a gearbox with variable structure providing certain technological advantages in rods rolling of various sections (Fig. 6).

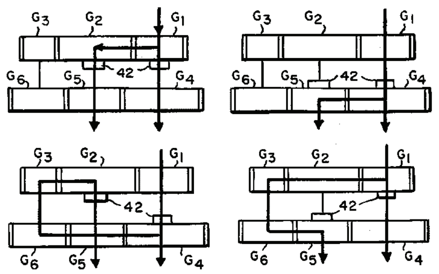



Figure 6 – Geared drive with variable structure of Reducing Sizing Mill (RSM)

The nature of this instability is still not clearly understood, because torque measurements during rolling in fully closed housing are very difficult. Dynamic analysis and vibration diagnostics of such type of gearboxes controlled by servo couplings are very complicated. Therefore, in-depth analytical study of driveline and rolled rod dynamics has been carried out [12].

A detailed kinematical scheme of reducing sizing mill (RSM) is represented in Fig. 7, *a*. Gearbox consists of four stages A, B, C, D and four stands 1…4 coupled by shafts. Dark points correspond to places of vibration sensors installation, M1…M9 are the controllable servo couplings to change gears ratios in accordance with rolling schedule.

In some previous studies of such type of mills on mathematical models [13, 14], authors addressing the dynamic phenomena by detuning from the resonance frequency bands proposed the following means: flexible couplings (change torsional stiffness); changes in the number of teeth (gears overlap); change in mass of rotating parts (moment of inertia). It was concluded that flexible couplings only briefly reduce peak loads in the gearbox. More effective is the change in the number of teeth of gears (for the same gear ratios), as the main cause of the oscillation is a periodic variation in the stiffness of the gears and couplings. The amplitude of the oscillation in gearing stiffness depends on the coefficient $e_R$ of teeth overlapping. Range considered in studies of change $e_R$ = 2.5...4.1 can reduce the dynamics of the driveline to acceptable values. The same result was obtained for split-path gearbox of slabbing mill where parametrical oscillations were investigated [15].

With an increase in the rolling speed on the wire and rod mills, defects become important in the driveline: the eccentricities of the work rolls; unbalance of components; misalignment of shafts; defects of mounting; wear of gears and couplings, breakage of teeth; defects of bearing assemblies. Beside the failures, defects cause kinematical perturbations affecting the accuracy of the rolled rod. Therefore, the main approach to solving the problem of increasing the rolling speed is the equipment condition monitoring and rod tension control [16-17].

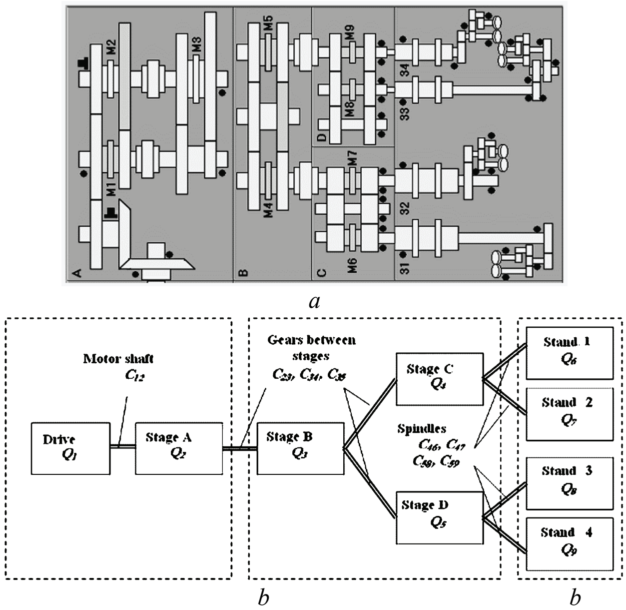



Figure 7 – Geared driveline of Reducing Sizing Mill (RSM): kinematical scheme of gearbox with stages A, B, C and servo couplings M1…M9 (*a*); calculation scheme (*b*); scheme of axial vibrations excitation (*c*)

Analytical multi-body model of RSM is shown in Fig. 7, *b*. A detailed analysis of natural frequencies and associated modes is represented in Table 3.

There are natural modes at the 22.7, 25.2, 31.8, 62.8 Hz where pairs of neighboring stands, 2-3, 3-4, 1-2, 2-3 accordingly, oscillate with opposite phases (relative amplitudes shown in frames).

Natural frequencies of transverse oscillations of rods are calculated with taking into account relation of elasticity modulus with temperature and section size are: 19.5…13.8 Hz (stands 1-2); 18.9…13.4 Hz (stands 2-3); 55.4…39.2 Hz (stands 3-4). Hence, if the frequency of natural mode will be twice less than excitation frequency, such modes may cause parametric oscillations. The calculation scheme of parametric oscillations is represented in Figure 7c. The corresponding equation of rod vibration is as following [18]:

$$C\frac{\partial^4 y}{\partial x^2}+(P_0+P\cdot\cos\Theta t)\frac{\partial^2 y}{\partial x^2}+m\frac{\partial^2 y}{\partial t^2}=0\,. \qquad (4)$$

where $P_0$ – mill setup tension force; $P$ – excitation force between stands; $y$ – displacement of rod from center line; $x$ – longitudinal coordinate of rod; $m$ – mass of rod between stands; $\Theta$ – excitation frequency; $C$ – stiffness of rod.

Then, initial equation can be reduced to standard Mathieu form:

$$\frac{d^2 x}{d\tau^2}+(a-2q\cdot\cos 2\tau)x=0\,. \qquad (5)$$

where $a$, $q$ – constants depending on rolling conditions (input and exit rod sections, reduction in stands, modulus of rod depending on metal temperature).

Further, stability of RSM when rods rolled in every pass schedule can be analyzed by standard Ince-Strutt diagram and mill control optimized to avoid resonance zones within the ranges of geared driveline working speed. Software is developed to calculate on-line stability conditions of rolling process for different reduction schedules.

Table 3 – Natural frequencies and modes of RSM geared driveline

| Masses | Natural frequencies and modes (Hz) | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 20.1 | 22.7 | 25.2 | 31.8 | 45.6 | 62.8 | 102.0 | 156.0 |
| Q1 (drive) | -0.324 | 0.073 | 0.001 | 0.001 | -0.09 | -0.001 | -0.059 | 0.015 |
| Q2 (stage A) | 0.130 | 0.017 | 0.001 | 0.001 | 0.187 | 0.003 | 0.854 | -0.530 |
| Q3 (stage B) | 0.073 | -0.040 | 0.001 | 0.001 | 0.406 | 0.005 | 0.452 | 0.835 |
| Q4 (stage C) | 0.167 | -0.232 | 0.001 | 0.001 | 0.346 | -0.653 | -0.147 | -0.089 |
| Q5 (stage D) | 0.205 | 0.090 | 0.001 | 0.001 | 0.575 | 0.662 | -0.204 | -0.120 |
| Q6 (stand 1) | 0.278 | -0.473 | 0.001 | -0.707 | -0.328 | 0.226 | 0.016 | 0.004 |
| Q7 (stand 2) | 0.278 | -0.473 | 0.001 | 0.707 | -0.328 | 0.226 | 0.016 | 0.004 |
| Q8 (stand 3) | 0.569 | 0.491 | -0.707 | 0.001 | -0.252 | -0.127 | 0.013 | 0.003 |
| Q9 (stand 4) | 0.569 | 0.491 | 0.707 | 0.001 | -0.252 | -0.127 | 0.013 | 0.003 |

**Conclusions.** The conducted research of nonlinear effects in rolling mills dynamics is the basis for condition monitoring and diagnostics of mechanical equipment and technological processes.

Using transient processes during metal biting in hot rolling mills allows developing the algorithms for diagnostics of angular and radial backlashes in drivelines. Short recordings of transient signals require special procedures for natural frequencies calculation since Fourier transform is not sufficient.

The multi-body dynamic models of cold rolling mills improve accuracy and diminish time lag in chatter vibration detection. Effect of neighboring stands synchronization is efficiently used in chatter monitoring system on industrial tandem mill 2030.

The multi-ratio gearbox of high-speed wire and rod rolling mill has advantages in production of variable sections of rods. Parametric oscillations of rod are excited due to out of phase natural torsional modes of vibrations in pairs of stands.

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