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Functioning features of the mill power line of pipes screw rolling

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Abstract
Some functioning features of the mill power line of pipes screw rolling of the pipe-rolling unit are considered. The developed dynamic model of a mill of pipes screw rolling is accepted. The differential equations of the movement of masses on the example of the main drive of a piercing mill taking into account the influence of variability of working angles and kinematics of universal hinges of a spindle are worked out. The specified solution of the differential equations of the movement of the power line of a mill is provided. Conditions of steady functioning of the power line and a mill working rolls of screw rolling of pipes taking into account variability of working angles of dual hinges are defined.
Keywords: MILL, SCREW ROLLING, PIERCING, DRIVE, POWER LINE, HINGE, SPINDLE, DYNAMICS, PIPE, INERTIA MOMENT, STABILITY, VIBRATION

Introduction

The high level of dynamic loads under the technological processes considerable rates in the mills power line drives of pipes screw rolling of the pipe-rolling unit (PRU) definitely affects the reliability and durability of elements, as well as the possible growth of mills productivity. High-speed and technological operating modes of pipes screw rolling mills correspond to the lower level of technical characteristics at best. They demand if not full replacement, but at least, radical modernization.

The average fault time caused by accidental failure of drive line details only of heavy loaded mills of pipes screw rolling, for example the piercing mills of pipe rolling plant 350 constructed on the basis of massive universal spindles, is about 10 - 12% of the general operation time. The fracture mode of the main power line elements of the piercing mills of screw rolling shows that the considerable single peak overloads as well as periodic effect of regular dynamic loads are the main reasons of details breakage. Not only peak values of dynamic loads, but also the nature of force impact from the rolled pipes, according to technological tasks of PRU, affect the formation of a complex picture of interaction of the worked pipe workpieces with system of the main and auxiliary mechanisms of the main power line of pipes screw rolling mill, according to technological tasks of PRU. As a rule, when pipes rolling, the peaks of cyclic dynamic loads in the power line elements of piercing mills repeat each time during transition processes period (workpiece gripping and pipe dropping). It should be noted that it is one of the reasons of premature failure of details, the main drive units. The massive universal spindles on the basis of dual Guk's hinges in the line of the piercing mill drive are the defining factors of dynamic loads formation.

Problem statement

Known method of the screw rolling mill line calculation do not consider the impact of massive spindles working angles change based on the dual Guk's hinges and technological loads nature on the features of their functioning [1, 2]. For these reasons, the development of researches directed on stabilization of dynamic characteristics of the screw rolling mill drive line is represented as relevant. In order to determine the vibroactivity arising in some transmission elements of pipes screw rolling mills, let us carry out the preliminary analysis of sources and basic designs of actuating devices on the example of the main drive of piercing mills of PRU (Fig. 1)

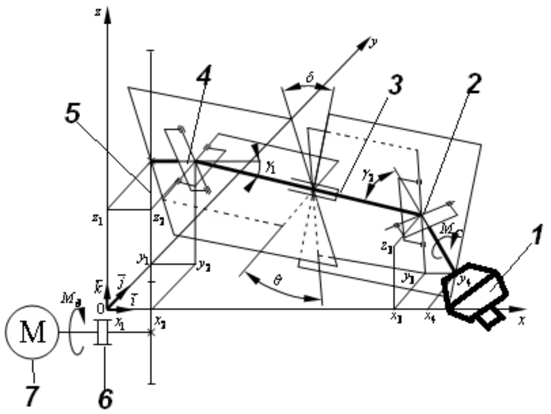

Figure 1. Kinematic diagram of the drive power line of the piercing mill of PRU with universal spindles: 1 - working roll; 2, 4 – Guk's hinge; 3 – spindle shaft; 5 – reducer; 6 – coupling; 7 – electric motor

The dynamic loads formation in the drive of a piercing mill is caused by functioning features of dual gimbal drive of universal spindles. Let us note that the kinematics of universal spindles is similar to kinematics of widespread gimbal drive on the basis of Guk's hinges. The functioning characteristic of the mill power line is that the elements, at that, rotate with periodically changing angular speed and possess a certain degree of run unevenness [3, 6]. Therefore, the dynamic loads formation nature and further functioning of the piercing mill working rolls drive mainly is defined by impact of unevenness of angular speed and angular acceleration of universal spindles. Let us notice that fixing of piercing mills working rolls within working angles of pipe workpiece supply is implemented according to the existing scheme in the working stand housing drum and is not a rigid construction. In the course of piercing, it leads to working angles change and further development of unevenness of this mechanical system run.

The analysis of dynamic processes in the main drives of a row of heavy loaded piercing mill shows that hinges kinematic parameters change affects the change of the speed component of the working rolls movement. Therefore, when implementing of basic processes of pipe workpiece piercing, as a rule, technological resistance forces change periodically, reaching the maximum and minimum value one-by-one.

Let us note that nonuniform rotation of mill working rolls excites the additional dynamic loads in the elements of the power line. It considerably reduces durability of the drive units, negatively influences the run of stable technological process of piercing and finished pipes quality.

Work objective

The objective of the dynamic analysis and synthesis of the piercing mills main drive containing universal spindles is the selection of the rational parameters of spindles providing the minimum unevenness of working rolls rotation and steady dynamics of all the power line.

The method of task solution

For the solution of a task of the analysis and synthesis of the piercing mill drive, the four-mass dynamic model presented in Figure 2 is accepted. When generating of differential equations of the movement of piercing mill mass, let us use the method of equivalent rigidity for the selected model of mechanical system [6]. Let us take the following main assumptions into account: the discrete masses of mechanical system of the piercing mill drive have the constant inertia moment; the external moments are constant; the provision of a drive line component in space is permanent; axes of universal hinges crosspieces are absolutely rigid; the friction in universal hinges is absent; there is no dissipation of energy in the main drive of the mill.

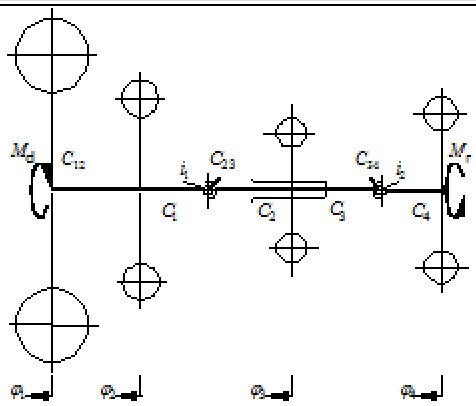

Figure 2. Dynamic model of piercing mill transmission with universal spindles

Let us make a system of the differential equations of movement for the corresponding transmission masses of the piercing mill within the accuracy of second order of smallness taking into account the mentioned assumptions and expressions for transmitting ratios of universal spindle hinges and the drive line given in papers [2, 6]. This differential equations system is of the form:

$$I_1\ddot{\varphi}_1 + k_{11}\varphi_1 - k_{12}\varphi_2 = M_1; \quad (1)$$

$$I_2\ddot{\varphi}_2 - k_{21}\varphi_1 + k_{22}\varphi_2 - k_{23}\varphi_3 = M_2; \quad (2)$$

$$I_3\ddot{\varphi}_3 - k_{32}\varphi_2 - k_{33}\varphi_3 - k_{34}\varphi_4 = M_3; \quad (3)$$

$$I_4\ddot{\varphi}_4 - k_{43}\varphi_3 + k_{44}\varphi_4 = M_4, \quad (4)$$

where I_1, I_2, I_3, I_4 – reduced inertia moments of reducer tooth-wheels, universal spindles and working rolls on drive shaft of the piercing mill;

$$k_{11} = k_{12} = k_{21} = c_{12}; \quad k_{22} = c_{12} + c_{23}\left[1 + 0.5n\varepsilon(1-\lambda_1)\cos 2\omega t\ \right]; \ k_{23} = c_{23}\left[1 + 0.5n\varepsilon(1-\lambda_1)\cos \omega t\right];$$

$$k_{32} = c_{23}(1 - 0.5n\varepsilon\lambda_1\cos 2\omega t); \ k_{33} = c_{23}(1 - 0.5n\varepsilon\lambda_1\cos 2\omega t + c_{34}\left[1 + 0.5\varepsilon(1-\lambda_2)\cos 2(\omega t + \nu\right];$$

$$k_{34} = c_{34}\left[1 + 0.5\varepsilon(1-\lambda_2)\cos 2(\omega t + \nu)\right]; \ k_{43} = k_{44} = c_{34}\left[1 - 0.5\varepsilon\lambda_2\cos 2(\omega t + \nu)\right]$$

- equivalent rigidity of the piercing mill drive line elements;
$M_1 = M_d$; $M_2 = -0.25c_{23}n\varepsilon\sin 2\omega t$; $M_3 = 0.25c_{23}n\varepsilon\sin 2\omega t - 0.25c_{34}\varepsilon\left[\sin 2\nu + \sin 2(\omega t + \nu\right]$; $M_4 = M_r + 0.25c_{34}\varepsilon\left[\sin 2\nu + \sin 2(\omega t + \nu\right]$ – equivalent moments of the piercing mill drive line.
Where φ_i – full angles of system masses rotation; $n = \gamma_1^2/\gamma_2^2$; $\varepsilon = \gamma_2^2$; γ_1, γ_2 – angles between axes of the spindle device shafts of piercing mill drive; $\nu = \theta - \delta$; θ – angle of a relative turn of spindle shaft universal hinges half-coupling; δ – angle between the basic planes of universal hinges (Fig. 1); ω – normalized angular velocity of the drive shaft rotation (motor armature) of the main drive; M_d – drive moment of the motor of the main drive; M_r – the moment of technological resistance when piercing of pipe workpiece.

In order to solve the system (1) – (4), containing exclusively variable component, let us put in the new variables ψ_i, connected with φ_i by the ratio $\varphi_i = \psi_i + \psi_{i0}$, considering $\psi_{i0} = const$.
The constant ψ_{i0} are found from the algebraic equations system on the basis of characteristic feature of the piercing mill power line functioning, that is $M_d = -M_r = M$.
It is necessary to outline that when first approximating, ψ_{i0} respectively will be

$$\psi_{10} = 0, \quad \psi_{20} = M/c_{12}, \quad \psi_{30} = M(c_{12}^{-1} + c_{23}^{-1}), \quad \psi_{40} = M(c_{12}^{-1} + c_{23}^{-1} + c_{34}^{-1}) + 0,25\varepsilon\sin 2\nu.$$

After necessary transformations (1) – (4), we will obtain the system of heterogeneous linear differential equations of the second order with periodic coefficients. Each equation of the system of the heterogeneous linear differential equations characterizing the system mass movements can be reduced to canonical (standard) form of the Mathieu equation by certain transformations. It is known that the solution of the Mathieu equations with periodic coefficients is followed by specific zones of instability and is connected with some difficulties of mathematical nature [1, 6].

The analysis of papers [1 – 3] showed that the solution of the Mathieu equation by only 1 - 2% differs from the basic task solution at small value of constants under periodic coefficient and research of the movement far from instability zones. Let us note that the components with periodic coefficient are neglected in the canonic Mathieu equation.
Therefore, taking into account the above mentioned assumptions let us give to the system of the differential equations (1) – (4) the following form:

$$I_1\ddot{\psi}_1 + c_{12}\psi_1 - c_{12}\psi_2 = M_{1s}\sin 2\omega t + M_{1c}\cos 2\omega t; \quad (5)$$

$$I_2\ddot{\psi}_2 - c_{12}\psi_1 + (c_{12} + c_{23})\psi_2 - c_{23}\psi_3 = M_{2s}\sin 2\omega t + M_{2c}\cos 2\omega t; \quad (6)$$

$$I_3\ddot{\psi}_3 - c_{23}\psi_2 + (c_{23} + c_{34})\psi_3 - c_{34}\psi_4 = M_{3s}\sin 2\omega t + M_{3c}\cos 2\omega t; \quad (7)$$

$$I_4\ddot{\psi}_4 - c_{34}\psi_3 + c_{34}\psi_4 = M_{4s}\sin 2\omega t + M_{4c}\cos 2\omega t, \quad (8)$$

where $M_{1s} = 0;\ M_{2s} = -0.25c_{23}n\varepsilon$;

$$M_{3s} = 0.25\varepsilon[c_{23}n - c_{34}\cos 2\nu - 2M(1-\lambda_2)\sin 2\nu];$$

$$M_{4s} = 0.5(c_{34}\cos 2\nu + M\lambda_2 \sin 2\nu);$$

$$M_{1c} = 0;\ M_{2c} = 0.5nM(1-\lambda_1); \quad M_{3c} = 0.25\varepsilon\{2M[n\lambda_1 + (1-\lambda_2)\cos 2\nu] - c_{34}\sin 2\nu\};$$

$$M_{4c} = 0.5\varepsilon(c_{34}\sin 2\nu - M\lambda_2\cos 2\nu)$$

The direct solution of the system of the equations (5) – (8) causes the lengthy dependences. It causes certain inconvenience for implementation of dynamic synthesis of the mill drive. Therefore, let us find the task solution in the form of decomposition by forms of system natural vibrations, presenting the equation in the form

$$\psi_i = \sum_{k=1}^{n0} q_c a_{iv}, \quad (9)$$

where q_c – main coordinates; a_{iv} – the shift of i-th mass of piercing mill dynamic model at the corresponding form of mechanical system natural vibration. Let us consider separately the movement of dynamic model mass under the forces impact proportional to harmonious functions $\sin(2\omega t)$ and $\cos(2\omega t)$. After necessary transformations, we obtain

$$\psi_{is} = \sum_{j=1}^{n0} M_{js} h_{ji} \sin(2\omega t); \quad (10)$$

$$\psi_{ic} = \sum_{j=1}^{n0} M_{jc} h_{ji} \cos(2\omega t). \quad (11)$$

In (10) and (11), h_{ji} are known for this selected dynamic model of the piercing mill drive of number; they are calculated by the formula

$$h_{ji} = \sum_{k=1}^{n0} a_{jk} a_{ik} / [m_k (p_k^2 - 4\omega^2)],$$

where $m_k = \sum_{i=1}^{n0} J_i / a_{ik}^2$ – the generalized weight at k-form of natural vibrations for the accepted dynamic model of the drive line; p_v – the frequency of natural vibrations of four-mass dynamic system.

The mass movement of the accepted dynamic model of the piercing mill main drive because of its linearity is expressed by the sum

$$\psi_i = \psi_{is} + \psi_{ic}, \quad (12)$$

or

$$\psi_i = A_i \cos(2\omega t - \theta_i), \quad (13)$$

where $A_i = \sqrt{A_{is}^2 + A_{ic}^2}$; $\theta = arctg(A_{is} / A_{ic})$;

$$A_{is} = \sum_{j=1}^{n0} M_{is} h_{ji};\ A_{ic} = \sum_{j=1}^{n0} M_{ic} h_{ji}.$$

Therefore, the angular rate of mass I_4 replacing a working roll of the mill is

$$\omega_4 = 2\omega A_4 \cos(2\omega t - \theta_4), \quad (14)$$

where

$$A_4 = \sqrt{(M_{2s}h_{24} + M_{3s}h_{34} + M_{4s}h_{44})^2 + (M_{2c}h_{24} + M_{3c}h_{34} + M_{4c}h_{44})^2}$$
(15)

From the developed mathematical model of system, it follows that the dynamic processes formation in the line of the piercing mill drive on the basis of gimbal drive is mainly defined by parameters of universal spindles.

It should be noted that the research of universal spindles kinematics of the piercing mill drive line causes certain difficulties. These universal spindles are made in the form of dual spatial gimbal drive.

The elements of hinges kinematic pairs load the inertia forces and the moment of the inertia forces of the spindle. Let us use the hinges kinematics on the basis of single gimbal drive in order to calculate the synchronous and asynchronous dual drives of the universal spindle of the piercing mill power line with a massive transmission shaft.

Let us pass on to the analysis of standard drive universal spindle kinematics on the basis of spatial gimbal drive (Fig. 1), which consists of two hinges, for specification of development nature and further synthesis of dynamic processes. The dependence between angular movements of two gimbal drive adjacent shafts connected by universal hinges, generally if $\psi_i \neq 0$, apparently from works [1, 6], is expressed by a ratio

$$tg(\varphi_{i+1}+\psi_{i+1})=\frac{1}{\cos\gamma_i}tg(\varphi_i+\psi_i)\,, \quad (16)$$

where ψ_i – the angular between the plane of the drive shaft leading fork and the plane (phase angular), where there are axes of the drive shafts connected by $i-$th hinge.

Therefore, the dependence between angular movements of the transmission and driven shaft of dual gimbal drive of the drive, in that specific case, is defined by expression

$$tg(\varphi_3+\psi_3)=\frac{1}{\cos\gamma_2}tg(\varphi_2+\psi_2)\ . \quad (17)$$

The angular rate of the working roll ω_3 (for a case, where $\psi_1=0$ is given in paper [6]) is determined by differentiation of expression (17), presented by ratio

$$\omega_3=\omega_2+2c_{21}\omega_2\cos2\psi_2\cos2\varphi_2-2c_{21}\omega_2\sin2\psi_2\sin2\varphi_2+$$
$$+e_{21}\cos2\psi_2\sin2\varphi_2\dot{\gamma}_2+e_{21}\sin2\psi_2\sin2\varphi_2\dot{\gamma}_2\,,$$
(18)

where

$$\omega_2=\omega_1+2c_{11}\omega_1\cos2\varphi_1+e_{11}\frac{d\gamma_1}{dt}\sin2\varphi_1\,,$$

$$c_{11}=\frac{1}{\cos\gamma_1}\left(\frac{tg^2\gamma_1}{4}-\frac{tg^4\gamma_1}{4}+\ldots\right);$$

$$c_{21}=\frac{1}{\cos\gamma_2}\left(\frac{tg^2\gamma_2}{4}-\frac{tg^4\gamma_2}{4}+\ldots\right),$$

$$e_{11}=\frac{tg\gamma_1}{2\cos\gamma_1}-\frac{tg^3\gamma_1}{4\cos\gamma_1}+\ldots;$$

$$e_{21}=\frac{tg\gamma_2}{2\cos\gamma_2}-\frac{tg^3\gamma_2}{4\cos\gamma_2}+\ldots\quad .$$

Using results of the paper [6] and (18), let us present the values $\sin2\varphi_2$, $\cos2\varphi_2$ by the following approximate expressions

$$\sin2\varphi_2=\sin2\varphi_1+c_{11}\sin4\varphi_2+\ldots;$$
$$\cos2\varphi_2=\cos2\varphi_1-c_{11}\cos2\varphi_1+\ldots\ .$$

Only universal spindles with working angles $\gamma_i\le20^\circ$ are used in drives of the majority domestic piercing mills. It is characteristic that in this case, the coefficients $c_{11},c_{21},e_{11},e_{21}$ are rather small values. Whereby, in further transformations, the component with coefficients $c_{11}^2,c_{11},c_{11}e_{11},\cdots$ is neglected.

So, having substituted the values $\omega_2,\sin2\varphi_2,\cos2\varphi_2,$ from [6] in a ratio (18), having carried out necessary transformations and rejecting values of the second order of smallness (components with coefficients $c_{11}^2,c_{11},c_{11}e_{11},\ldots$ and so forth), let us write down an approximate formula for the angular rate of the mill working roll in the form

$$\omega_3=\omega_1+2c_{11}\omega_1\cos2\varphi_1+e_{11}\sin2\varphi_1\dot{\gamma}_1+$$
$$+2c_{21}\omega_1\cos2(\varphi_1+\psi_2)+e_{21}\sin2(\varphi_1+\psi_2)\dot{\gamma}_2 \quad (19)$$

Having differentiated on time the equation (19) on three parameters φ_1, γ_1,γ_2, which are functions of time,

$$\varepsilon_3=\frac{\partial\omega_3}{\partial\varphi_1}\frac{d\varphi_1}{dt}+\frac{\partial\omega_3}{\partial\gamma_1}\frac{d\gamma_1}{dt}+\frac{\partial\omega_3}{\partial\gamma_2}\frac{d\gamma_2}{dt}$$

we obtain the angular acceleration ε_3 of the working roll in the form

$$\varepsilon_3=\varepsilon_1+2c_{11}\varepsilon_1cos2\varphi_1-4c_{11}\omega_1^2\sin2\varphi_1+2e_{11}\omega_1cos2\varphi_1\dot{\gamma}_1-$$
$$-4c_{21}\omega_1^2\sin2(\varphi_1+\psi_2)+2e_{21}\omega_1cos(2\varphi_1+\psi_2)\gamma_2+$$
$$+2\frac{\partial c_{11}}{\partial\gamma_1}\omega_1cos2\varphi_1\dot{\gamma}_1+\frac{\partial e_{11}}{\partial\gamma_1}\sin2\varphi_1\left(\dot{\gamma}_1\right)^2+e_{11}\sin2\varphi_1\ddot{\gamma}_1+$$
$$+2\frac{c_{21}}{\partial\gamma_2}\omega_1\cos2\left(\varphi_1+\psi_2\right)\dot{\gamma}_2+\frac{\partial e_{21}}{\partial\gamma_2}\sin2(\varphi_1+\psi_2)\left(\dot{\gamma}_2\right)^2+$$
$$+e_{21}\sin2(\varphi_1+\psi_2)\ddot{\gamma}_2\,,$$
(20)

where

$$\frac{\partial c_{11}}{\partial\gamma_1}=\frac{tg\gamma_1}{2\cos^3\gamma_1}-\frac{tg^3\gamma_1}{2\cos^3\gamma_1}-\frac{tg^5\gamma_1}{2\cos\gamma_1}+\ldots;$$

$$\frac{\partial c_{21}}{\partial\gamma_2}=\frac{tg\gamma_2}{2\cos^3\gamma_2}-\frac{tg^3\gamma_2}{2\cos^3\gamma_2}-\frac{tg^5\gamma_2}{2\cos\gamma_2}+\ldots;$$

$$\frac{\partial e_{11}}{\partial\gamma_1}=\frac{1}{2\cos^3\gamma_1}\left(1-\frac{tg^2\gamma_1}{2}-\frac{13tg^4\gamma_1}{8}+\ldots\right);$$

$$\frac{\partial e_{21}}{\partial\gamma_2}=\frac{1}{2\cos^3\gamma_2}\left(1-\frac{tg^2\gamma_2}{2}-\frac{13tg^4\gamma_2}{8}+\ldots\right).$$

Let us consider two special cases of functioning of spatial universal spindles of the piercing mill main drive with dual gimbal drive. These spindles are most often met in practice of the power line operation. For universal spindles of the piercing mill drive line with dual gimbal drive, $\gamma_1=\gamma_2;\dot{\gamma}_1=\dot{\gamma}_2;\psi_1=0;\psi_2=0,$ is correct. In this case, gimbal forks are installed on the transmission shaft at an angle 90^0 ($c_{11}=c_{21},e_{11}=e_{21},\cos2\psi_2=1$). Therefore, let us write down a formula for the angular rate of a working roll in the form:

$$\omega_3=\omega_1+4c_{11}\omega_1\cos2\varphi_1+2e_{11}\sin2\varphi_1\dot{\gamma}_1\,. \quad (21)$$

Thus, for this case, at the preset angles $\gamma_i \leq 20^\circ$, rotation unevenness of working rolls will be maximum.

For universal spindles of piercing mill transmission with dual gimbal drive, $\gamma_1 = \gamma_2; \dot{\gamma}_1 = \dot{\gamma}_2; \psi_1 = 0; \psi_2 = 90^\circ$, i.e. gimbal forks are installed on the transmission shaft in one plane ($c_{11} = c_{21}, e_{11} = e_{21}, \cos 2\psi_2 = -1$). For this case, let us write down a formula for the working roll angular rate in the form:

$$\omega_3 = \omega_1 + 2c_{11}\omega_1 \cos 2\varphi_2 + $$
$$+ e_{11} \sin 2\varphi_2 \dot{\gamma}_1 - 2c_{21}\omega_1 \cos 2\varphi_2 - e_{21} \sin 2\varphi_1 \dot{\gamma}_2 = \omega_1$$
(22)

Therefore, the angles γ change has no noticeable impact on kinematics of universal spindles in synchronous gimbal drive of the piercing mill drive line ($\gamma_1 = \gamma_2; \dot{\gamma}_1 = \dot{\gamma}_2; \psi_1 = 0; \psi_2 = 90^\circ$).

The analysis of this work results shows that it is possible to be limited to average values γ_1 calculations in the kinematic analysis of the piercing mill gimbal drive at small values of angle $\gamma_1 \leq 20^\circ$ and small change rates of this angle $\dot{\gamma}_1 > \omega_1$, and the impact of angle γ_1 variability at small values $\dot{\gamma}_1 \leq (0{,}05 \ldots 0{,}1)\omega_1$ may be neglected. The calculation by formulas (19) and (20) should be carried out only at high rates of change $\dot{\gamma}_1 >> \omega_1$, especially in high-speed gimbal drives. It is reasonably to use more exact ratio for variable components of angular rate and angular acceleration of the driven working roll of the piercing mill at big angles $\gamma_1 \geq 20^\circ$.

The obtained dependences (19) and (20) are universal and can be used with a sufficient accuracy for the analysis and synthesis of dynamics of the piercing mill main drive with dual universal hinges.

It should be noted that in drives of the piercing mills row, the working angles $\gamma_1 \geq 20^\circ$ are applicable. In this regard, there are additional components, which quantity depends on the number of variable angles γ_i in drive elements, in formulas for angular rates and accelerations. In this case, the equation of the movement of the drive line of a mill working roll (13) definitely differs from the known Mathieu equation, since the periodic coefficients are not small.

The research (13) and (14) allows establishing of some main reasons for occurring of complex vibrations of working roll axis in combination with the drum in the housing window; in turn, they affect significantly the dynamics of the entire piercing mill.

The obtained results analysis shows that the unevenness of the power line run with dual universal hinges is the main reason for discrepancy of experimental and known theoretical researches results of the piercing mill drive line. It is necessary to emphasize that the certain alternate areas of steady and unstable solutions are inherited in the differential equations (5) – (8) because of the available periodic coefficients. From (13) and (14), it can be seen that the turn angles of the main drive spindle hinges (parameters n and ν) have impact on the value of vibration amplitude A_4. Having selected the last of conditions $A_4 = 0$, it is possible to achieve the quasiuniform rotation of working rolls at first approximation.

The specified analysis and synthesis of vibroactivity of the piercing mill power line should be made on the basis of research of the drive generalized dynamic model. In this case, the stability of functioning of driving line elements and piercing mill working rolls of the pipe-rolling unit is defined on the basis of known prerequisites for the mathematical theory of stability and asymptotic methods of differential equations research according to [5, 7]. At that, from the equation (15), it is necessary to determine the areas of parametrical stability of system depending on the piercing mill operation modes, a spatial location and angle sizes of the spindle device elements of the main transmission.

The approximately twofold decrease in level of system acceleration capacity is reached by using of recommended sizes of working angles γ_i, for example, for the piercing mill of the pipe-rolling unit TPA 350 within 10°30' ÷ 11°30', and the corresponding rotation frequency of the main drive elements. In this case, at the same time, the noticeable decrease in vibroactivity of the drive line elements of the working rolls is observed. However, the further increase in working angles of spindles and rotation frequency of the piercing mill drive line of TPA 350 causes the significant increase of dynamic loads. It demands the introduction of additional measures for stabilization of vibroactivity in the mechanical system “the drive line – working rolls”.

Conclusions

1. It is established that known methods of dynamics calculation of the mill drive line of pipes screw rolling (piercing mill) do not consider some features of the mill working roll behavior, and in particular, the time change of axes deviation angles

of spindles spatial hinges.

2. The specified differential equations of the movement of the piercing mill main drive elements taking into account the time change of working angles size and unevenness of kinematic parameters of spindle hinges are worked out.

3. The impact of basic parameters of universal spindle with dual hinges on the dynamic loads formation of the piercing mill drive is established.

4. The stabilization of the run of the piercing mill main drive line is achieved at the first approximation by working roll amplitude minimization and the selection of spindle hinges parameters.

5. It is found out that the variability of dual hinges working angles of the drive universal spindle is the main reason of parametrical vibrations in mechanical system of the piercing mill power line.

6. The obtained results can be applied in drives of similar machines and mechanisms with universal spindles on the basis of Guk's hinges.

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