PIPE AND TUBE PRODUCTION

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Some Peculiarities of Operation of Cold Pilgering Tube Mills

An engineering technique to simplify a developed dynamic model is proposed, which includes the combination and development of the known techniques, moreover, the active mechanical bonds in the original dynamic model of a cold pilgering mill are shown, which are causing the occurrence of strongly pronounced parametric processes in the studied system. The dynamics of simplified dual-mass cold pilgering mill models with combined parameters and periodically varying weight characteristics was studied. A differential equation of parametric oscillations in the drive power line was composed. The solution of the differential equation allows us to estimate the dynamic displacements of the considered drive of the cold pilgering mill working stand for the most common types of oscillations of a mechanical system. Since the differential equation (8) is presented in the formulation of Cauchy base problem, for the most common first form of oscillations of a mechanical system, the problem is solved numerically by means of the standard software product using Runge – Kutta method. Dynamic peculiarities of the operation of the drive of the cold pilgering mill working stand are presented in the form of angular displacements of the drive shaft of the mechanical system. The dynamics calculation of the working stand drive of CPM 32-3 (tube rolling on the route 38x3.8 → 19.1x2.1, material - steel X18H10T) was carried out. The reasons for the occurrence of parametric oscillations in the drive system were identified and zones of dynamic instability of the mechanical operation of the cold pilgering mill drive system were established allowing the selection of optimal rolling conditions at the design stage of technological processes.

Keywords: *cold rolling, tube, mill, power line, main drive, dynamic model, simplified design scheme, mathematical model, parametric oscillations, differential equation, angular oscillations, dynamic instability*

The need to establish the essence of many dynamic phenomena and the prediction of peak values of the loads, both during the design or reconstruction stages of cold pilgering mills, and their operation using advanced calculation methods remain relevant.

A flow sheet of the most common design of the cold pilgering mill is shown in Figure 1. The cold pilgering mill consists of a number of mechanisms, the kinematic and force connections of which are carried out by the developed system of rolls. From the technological peculiarity of the process of cold rolling of pipes, it follows that cold pilgering mill mechanisms operate in various continuous or periodic (cyclic) modes. During periods when the stand passes the extreme positions, the translational and rotational motions of the “billet-ready-made product” system are periodically carried out by means of the respective feeding chucks and rotation of the distributing and feeding mechanism (hereinafter DFM) . Synchronization of the cycle operations of feeding and rotation with the stand position is implemented by a feeding and turning unit (hereinafter FTU).

The cold pilgering mill systems are combined into the following main units such as the twinned slider-crank mechanism for moving the working stand; power plant including an electric motor, an angular gearbox and a brake; inlet gearbox distribution and feeding mechanism; washer shaft of a cam-and-lever mechanism; connecting shafts and couplings. For the ease of the analyzing dynamic phenomena in a chain, all units can be combined into the so-called main power line of cold pilgering mill [2].

Peak values of dynamic loads and features of their occurrence are determined by the lowest natural vibration frequencies of the same mechanical system. In some cases, when the frequencies of the eigen-oscillations of multi-mass mechanical systems shall be calculated, the inconveniences of various kinds (including computational) appear that make it difficult to obtain the reliable data. At the same time, it becomes difficult to identify the main factors determining the development of various dynamic processes in the cold pilgering mill [2, 3].

There are some well-known methods for simplifying developed dynamic models of complex multi-mass mechanical systems, which imply the possibility of reducing to a system consisting of simple generally “connected” with each other series partial systems with one degree of mobility [4–8].

The nature of oscillations in simple or, as they are called, partial subsystems shall be known to analyze physical phenomena and dynamic processes in a complex mechanical system of the cold pilgering mill. Each of these proposed partial mill systems is the simplest dual-mass system. The main parameters of mechanical system of the CPM 32-3 are given in Table 1, the dynamic model of which is based on the initial scheme (see Fig. 1).

The initial seven-mass dynamic model with six degrees of freedom can be reduced to models with a smaller number of the same [1, 8]. The cyclic frequencies of eigen-oscillations were determined for various models of the main power line of CPM-32 using the standard package of programs.

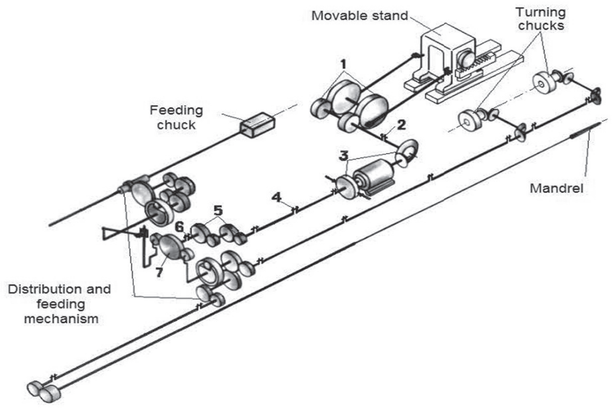

Figure 1. The kinematic scheme of cold pilgering mill with a feeding and turning mechanism of the gearbox type: 1 – twinned crank-slider mechanism for working stand moving; 2 – power plant; 3 – electric motor and angle gearbox with a brake; 4 – drive shaft; 5 – gearbox distribution and feeding mechanism; 6 – cam and lever mechanism and its connecting shafts and couplings; 7 – distributor shaft

The modern direction of up-gradation of CPM-32, CPM-55 and CPM-75 mills envisages the installation in the power line of the mill of an epicyclic turning and feeding mechanism (hereinafter ETFM) of a new generation with independent drives of translational and rotational movements of the "billet-ready-made pipe" and rational working stand [2]. A periodic action mechanism made on the basis of epicyclic converters with a pneumatic control system is used in the feeding and rotation circuits, in contrast to those used in DFM instead of a cam-lever mechanism and free-wheel mechanisms.

The efficiency of modernization was ensured by maximizing the use of DFM elements for up-gradated mills (screws, nuts, feed chucks; elements with developed stereometry of turning chains, etc.). It should be noted that the new elements of the epicyclical turning and feeding complex did not require additional structural components and were installed on the places of the dismantled gear type DFM .

A significant demonstration of this solution was the division of the mechanical systems of the mill into a number of independent power circuits, the interaction of which was carried out only by a program that was "wired" in the pneumatic system of the ETFM control unit. As a result, the parameters of the main power line have been changed significantly (Fig. 2).

The variation in the work speed of the cold pilgering mill due to the requirements of the technological process is undoubtedly related to the dynamic characteristics of the working stand drive operation, which is confirmed by the results of many experimental studies [1, 2, 7].

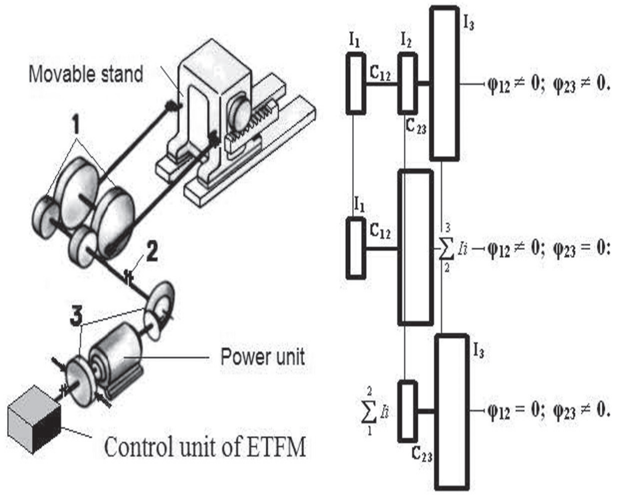

Figure 2. The kinematic scheme, the dynamic model and its partial systems of the main power line of cold pilgering mills with epicyclic turning and feeding mechanism: 1 – twinned slide and crank mechanism for the working stand moving; 2 – drive shaft of the power plant; 3 - electric motor with angular gearbox and brake

From experimental studies [1, 2], it can be seen that the maxima of the dynamic loads are repeated every time during the period of distinctive transient processes (Fig. 2, 3).

Dynamic loads in the steady-state operating mode of CPM 32-3 stand drive are periodic. The amplitude and frequency characteristics of the system are the closest to those of a dual-mass simplified dynamic model of the main drive of the mill.

The resulting simplified dynamic models of the system shall be studied further in order to analyze periodically changing in time dynamic loads (parametric phenomena) arising in the steady operating mode of the working stand drive of the cold pilgering mill.

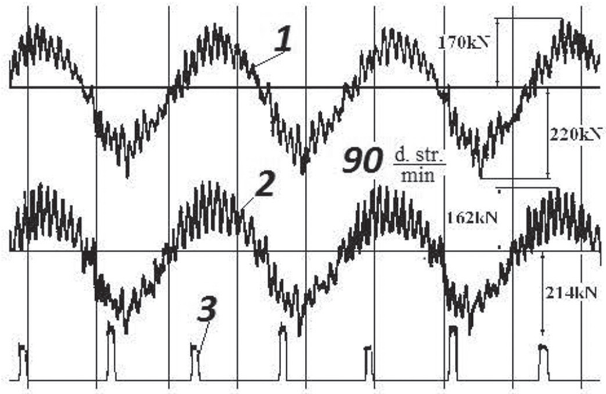

Figure 3. Parametric oscillations of connecting rods of power line of CPM 32-3 when rolling a pipe on the route 38x3.8 → 19.1x2.1, material - steel X18H10T: 1, 2 - oscillations of connecting rods of the working stand drive; 3 - time mark 0.! s

According to [1, 9], the dual-mass simplified partial dynamic model of the main drive of the cold pilgering mill is described by the differential equations of motion of the system in the following form:

$$I_1 \frac{d^2\varphi_1}{dt^2} + c(\varphi_1 - \varphi_2) = -M_1; \qquad (1)$$

$$I_2 \frac{d^2\varphi_2}{dt^2} - c(\varphi_1 - \varphi_2) = M_2,$$

where M_1 – torque of technological resistance from the deformation zone; M_2 – motor shaft torque of the main drive of the cold pilgering mill; φ_1, φ_2 – angular displacement of the masses of the selected mechanical system; I_1, I_2 – moments of inertia of the respective masses of the mechanical system: c – equivalent torsional rigidity of the connection between the masses of the system.

If we neglect the deformation of the connecting rods of the working stand drive and their masses are replaced by the corresponding masses, one of which is attached to the crank wheel, and the other one is attached to the movable working stand of the mill, a fairly acceptable design scheme can be obtained.

It should be highlighted that the moment of inertia of the added mass of the connecting rod part m_1R^2 shall be added to the moment of inertia of the crank wheel:

$$I_1 = I + m_1R^2 \qquad (2)$$

Where I – moment of inertia of the crank wheel of the main drive of the mill; R – radius of the crank wheel.

In the framework of the considered dual-mass dynamic model of the main drive of the cold pilgering mill, the periodically varying given moment of inertia of the "working stand-connecting rod-crank wheel" system considering the above is determined as follows:

$$I_1 = I + m_1R^2 + \frac{1}{2} m_2R^2(1 - \cos(2\omega t)) \qquad ,(3)$$

Where m_2 – the total weight of the working stand and the corresponding part of the connecting rod;

ω – the steady-state value of the angular velocity of the crank wheel.

Consequently, according to [3, 7], as per the system of equations of the drive elements oscillations (1), we get the following:

$$\frac{d^2\varphi}{dt^2} + \frac{c(\overline{I}_1 + I_2)}{\overline{I}_1 I_2}\varphi = \frac{M_1}{I_2} + \frac{M_2}{I_1} \qquad , (4)$$

where $\varphi_1 - \varphi_2 = \varphi$; $\overline{I}_1 = I + m_1R^2 + \frac{1}{2}m_2R^2$ is some average value of the reduced moment of inertia of the crank wheel, connecting rods and the working stand of the mechanical system.

Substituting the formula (3) into the differential equation (4), the following shall be written:

$$\frac{d^2\varphi}{dt^2} + c\left[\frac{1}{I_2} + \frac{1}{\overline{I}_1\left(1 - \frac{m_2R^2}{2\overline{I}_1}\cos(2\omega t)\right)}\right]\varphi = \frac{M_1 I_2 + M_2 \overline{I}_1}{I_2 \overline{I}_1} \qquad . (5)$$

From equation (5), it can be seen that the coefficient of the function φ depends on time t. This is different from the cases considered in [1–9]. It should be noted that within the framework of the considered problem of dynamics of a working stand drive of the cold pilgering mill, the variability of the coefficient is associated with periodic changes in the system's inertia moment and not with the system stiffness coefficient.

For the convenience of analyzing the dynamic phenomena in the main drive of the mill, a certain simplification of equation shall be made (5). Due to the small fraction $\frac{m_2R^2}{2\overline{I}_2}$ compared with the unit, it can be assumed that:

$$\frac{1}{1 - \frac{m_2R^2}{2\overline{I}_1}\cos(2\omega t)} \approx 1 + \frac{m_2R^2}{2\overline{I}_1}\cos(2\omega t) \qquad . (6)$$

Then the original differential equation (5) takes the following form:

$$\frac{d^2\varphi}{dt^2} + c\left(\frac{1}{I_1} + \frac{1}{I_{2ñð}}\right)\left[1 - \frac{I_2 m_2 R^2}{2\overline{I}_1 I_2 (I_2 + \overline{I}_1)}\cos(2\omega t)\right]\varphi = \frac{M_1 I_2 + M_2 \overline{I}_1}{I_2 \overline{I}_1} \qquad . (7)$$

Differential equation (7) is conveniently represented in the form of parametric Mathieu – Hill equations with the right-hand side [7, 9].

After substitutions of the parameters of the mechanical system dynamic model and some transformations of the differential equation (7), the equation of the adopted generalized mathematical model of the cold pilgering mill drive is obtained, which meets the conditions of the set problem in the following form;

$$\frac{d^2\varphi}{dt^2}+\Omega^2\left[1-\mu\cos(2\omega t)\right]\varphi=\frac{M_1I_2+M_2\bar{I}_1}{I_2\bar{I}_1},\quad \Omega^2=c\left(\frac{1}{I_2}+\frac{1}{\bar{I}_1}\right) \qquad (8)$$

Where $\Omega^2 = c(\frac{1}{I_2}+\frac{1}{\bar{I}_1})$ – the square of the frequency of eigen-oscillations of the mechanical drive system of the working stand; $\mu=\frac{I_2m_2R^2}{2\bar{I}_1I_2(\bar{I}_1+I_2)}$ – coefficient of dynamic excitation of the dual-mass model of the drive of the working stand of the mill.

Coefficient of the dynamic excitation of the system μ and frequency of eigen-oscillations of the main drive Ω are determined from the characteristic conditions of the mutual change of the dynamic parameters of the cold pilgering mill.

The solution of the differential equation (8) allows us to estimate the dynamic displacements of the selected drive element of the cold pilgering mill-working stand for the most common modes of vibration of a mechanical system. Since the differential equation (8) is presented in the formulation of the Cauchy basic problem, for the most common first form of oscillation of a mechanical system, the problem shall be solved numerically using the Runge – Kutta method in Matcad software. Let us carry out the calculation of the dynamics of the working stand of the mill for the selected dynamic model of the mechanical system using the example of CPM 32-3 [1, 2]. Dynamic features of the operation of the working stand drive of CPM 32-3 are presented in the form of angular displacements of the drive shaft in the mechanical system (Fig. 3).

A comparison of the results of numerical calculation (Fig. 3) and experimental studies [1, 2] shows that the differential equations (8) with a sufficiently high degree of accuracy describe the forced vibrations of the working stand drive of CPM 32-3. The results obtained are relatively convergent with the results of experimental studies (Fig. 3). The amplitude and frequency characteristics of dynamic processes with parametric oscillations of the main working stand drive during the implementation of the entire tube rolling process of 38x3.8→19.1x2.1, material – steel X18H10T at CPM 32-3 exceeds the acceptable level of the vibration activity of the mechanical system.

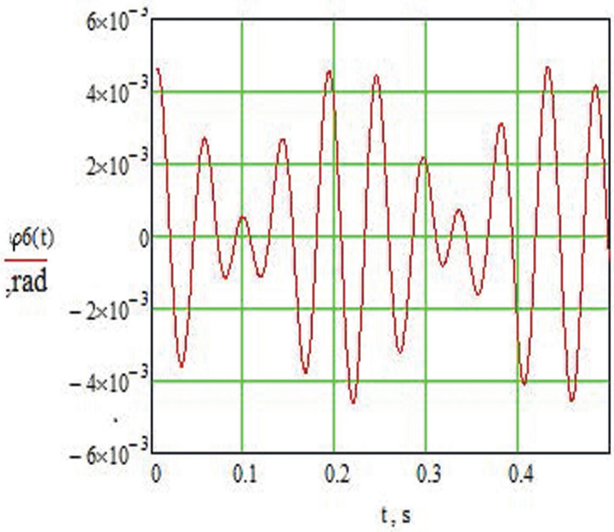

Figure 3. Dynamic of the working stand drive of CPM 32-3 (tube rolling on the route 38x3.8→19.1x2.1, material –steel X18H10T): $\varphi_6(t)$ – angular displacements of the drive shaft of the working stand

The developed mathematical model significantly clarifies the behavior of the mechanical system and reliably describes the processes in the working stand drive of CPM 32-3. Due to the fact that the dynamics of the working stand drive of the cold pilgering mill is shown in the form of the Mathieu – Hill equation, the parametric stability of the mechanical system behavior in a generalized form is convenient to estimate based on the Ains – Strett stability diagram [9].

It should be noted that the stability diagram of the system operation for the considered generalized dynamic model of the cold pilgering mill allows analyzing possible areas of parametric stability of the working stand drive line operation and can be used during selection of rational tube rolling modes. The corresponding areas of dynamic stability of the system are highlighted by the shaded areas in the diagram. When using Ains – Strett stability diagram [9], it should be taken into account that in the considered case, the frequency of changes in the stiffness coefficient of the system is equal to 2ω (not to ω). Therefore, the main area of parametric resonance of drive corresponds to $2\omega/\Omega=2$, i.e. $\omega/\Omega=1$.

From the experience of operation of the majority of domestic CPM [1, 8], it is known that in some modes of mill operation, significant parametric oscillations are observed. The possibility of simulating quasi-harmonic oscillations in the main drive and the possibility of selection of the optimal modes of high-quality pipes rolling at the design stage of technological processes significantly distinguishes the results obtained from the results of previously known works.

Conclusion

1. An engineering technique for simplification of a developed dynamic model including the combination and development of well-known techniques has been used. In addition, the active mechanical connections in the original dynamic model of the cold pilgering mill causing the pronounced parametric processes in the system under study have been revealed.

2. The dynamics of simplified dual-mass cold pilgering mill models with combined parameters and periodically changing mass characteristics has been studied.

3. The reasons for the occurrence of parametric oscillations in the drive system have been identified and dynamic instability zones of the operation of cold pilgering mill drive have been established, which allows for the selection of optimal rolling conditions at the design stage of pipe production processes.

4. The obtained results of studies of the dynamics of simplified dynamic models of cold pilgering mill with combined parameters and periodically changing mass characteristics are quite applicable to similar mechanical systems.

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