Research and modernization of the drive of cold-pilgering mills cage

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Abstract
Problems of modernization of cold-pilgering mills in a direction of both expansions of their technological possibilities, and working capacity increase are always actual. Coupled crank mechanisms used in the industry are often made without necessary estimation of level of their static indefinability that results not only in decrease in their reliability, but also influences qualitative characteristics of finished products.

Keywords: MODERNIZATION, COLD-PILGERING MILL DRIVE, STATIC DEFINABILITY, COUPLED CRANK MECHANISMs, PERFORMANCE FEATURES ANALYSIS

Constantly increasing requirements to the pipe geometry and properties of the materials are ideally suited by cold pilger rolling. Its vitality and competitiveness is also determined by the fact that it belongs to one of the few nonwaste technologies of materials processing by pressure, enabling: to control of the flow of metal in both the longitudinal and circumferential direction, which allows to control the geometry of the finished product with high rates of both the outer, and inner surfaces; to manage the structure of metal; to manufacture the pipes by different output parameters from the workpiece of standard size.

According to experts on the pipe rolling, metallurgical and engineering enterprises of Ukraine there are several hundred units for pipes cold pilger rolling by roll grooves and gauge rollers, so the extensive researches and proposals on their modernization in order to increase functionality and, therefore, increase their utilization rate, are relevant.

The cage of pipes cold pilger rolling unit (CPM), being a driven link of coupled crank mechanism is driven by the main motor through a gearbox or belt drive by gear wheels z_1 and z_1' (Fig. 1 a). These mechanisms, forming closed circuits, are a statically indeterminate system, the number of redundant connections of which is determined not only by the number of repetitive kinematic joints, but also their types. This forces to consider the deformations of links and kinematic joints elements at the analysis of drive operation.

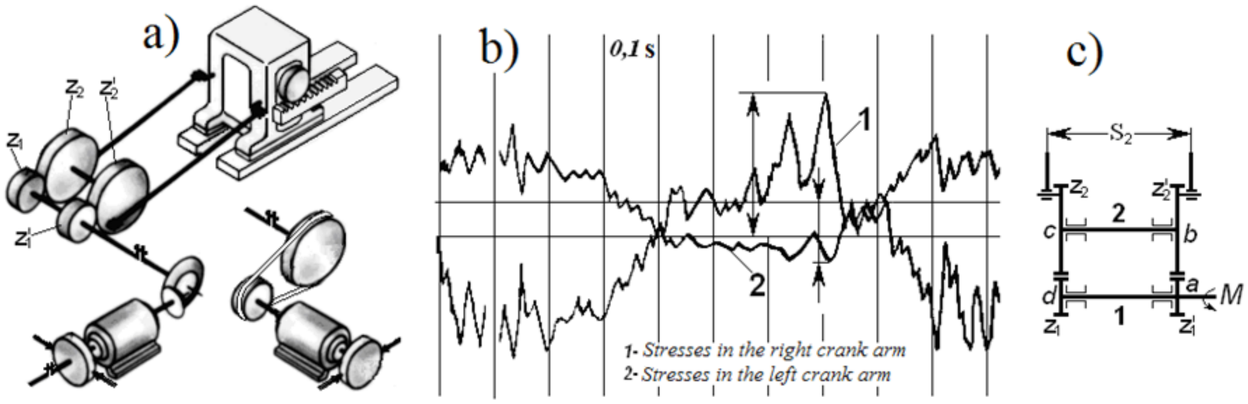

Figure 1. Kinematic scheme of the main drive of the mill CPM-32, the scope pattern of stresses in crank arms and the scheme of moment transmission from the coupled leading gear wheels to the cranks

As the shafts 1 and 2 can be elastically deformed, due to the asymmetry of power supply the transmission of power flow can occur both in the direction of ***aвcd*** and towards ***adcb*** (Fig. 1c). Establishment of the direction of moment transmission in drives of mills of different standard sizes is connected with their individual characteristics.

Moments of elastic forces in the shafts 1 and 2 can be represented by the following expressions:

$$M_1' = \varphi_1 \cdot c_1, \quad (1)$$

$$M_2' = \varphi_2 \cdot c_2, \quad (2)$$

where φ_1 φ_2 – twist angles of shaft 1 and shaft 2;

$$c_1 = \frac{GI_{P1}}{a}; \quad c_2 = \frac{GI_{P2}}{a} -$$

stiffness of shafts 1 and 2:

G – shearing modulus;

I_{P1} and I_{P2} – polar moments of inertia of shafts 1 and 2;

a – distance between the gear wheels z_1 and z_1' and z_2 and z_2'.

Fig. 1c shows that the moment required to perform the process, applied to the crank gear wheels is equal

$$M_2 = M_2' + M_1' \cdot i, \quad (3)$$

where $i = \frac{r_2}{r_1}$;

r_1 – the radius of the bottom line of teeth;

r_2 – the radius of the bottom line of crank-type wheels.

After some transformations of the expression (3), we obtain

$$M_2 = \varphi_2 \frac{GI_{P2}}{a}\left(1 + i^2 \frac{I_{P1}}{I_{P2}}\right). \quad (4)$$

Numerical analysis of the expression (4) shows that the its second element (for constructive solutions of mills drives CPM-32, CPM-55, CPM-75 and the CPM-90) are 10...20 times greater than the first one. On this basis in engineering calculations the moment M_2 can be determined from this relationship

$$M_2 = \varphi_2 \frac{I_{P1} G i^2}{a}. \quad (5)$$

When moving the stand the perceived by crank arms load is transmitted through the gear wheels to the drive shaft 1 with a suppleness, so the gear z_1' and crank-type wheel z_2' get a specific offset from the gear z_1 and crank-type wheel z_2. This shift, as follows from expression (5) is determined by the moment M_2, and stiffness of shaft 1. Mill stand within a certain angle is rotated around a vertical axis in one direction, and then under reversal of stress index - in the other one.

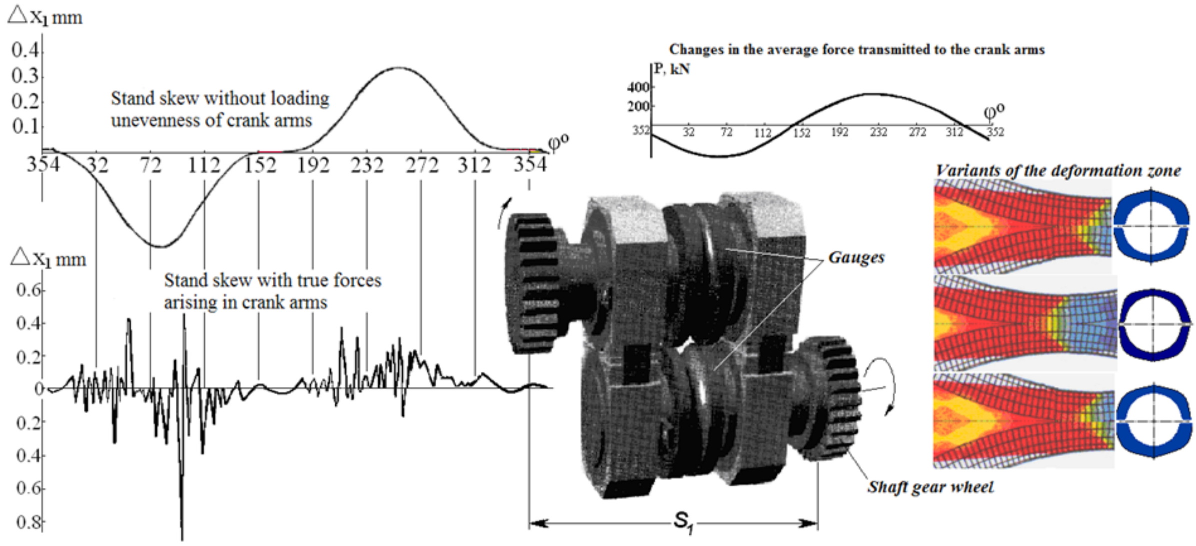

Figure 2. Stand skews and deformation zone distortion in the rolling process

When rolling the roll gear wheels on the rack bars the deformation zone is distorted as a result of stand skew (its yaw and pitch around the vertical axis), and binding of teeth can occur in extreme cases. This caused the need to set increased gaps in gear pairs "roller wheels - rails." They should be minimized since the increase in the gaps leads to a largely increased dynamic of interaction along the whole power circuit.

If we assume that crank arms stresses are the same, the same force P will affect each of the crank-type wheel, which will make a moment M_2 equal

$$M_2 = P \cdot r_0 \cdot \cos\beta\,, \tag{6}$$

where r_0 – radius of crank shaft;

β – angle between the crank shaft and perpendicular from the center of crank shaft rotation in the direction of crank arm.

At the joint considering of expressions (5) and (6), obtain

$$\varphi_2 = \frac{P \cdot r_0 \cdot \cos\beta \cdot a}{i^2 \cdot G \cdot I_{P1}}\,. \tag{7}$$

The study [5] shows the relation between twist angle φ_2 of the shaft 2 and the angle β and their influence on the movement Δx of a right shaft relatively to the left one. Stand skew is determined if the coordinates of the contact point of crank arms symmetry planes with axes of the link pins of their movable connection with the stand are known.

The additional movement of the center of leading roller gear wheel will be larger as the distance between the gear wheels S_1 is greater than the distance between the crank arms S_2.

Thus

$$\Delta x_1 = \Delta x \cdot \frac{S_1}{S_2}\,. \tag{8}$$

Dependences of value of Δh1, taking into account the differences in distance between the symmetry planes of the wrist pins and leading gears of rolls drive are shown in Fig. 2. From these graphs it is clear that if the resistance to movement of the stand varied smoothly, the skew (yaw) would take place smoothly, as it follows from the skew graph at an average stress. However, experimental studies show that the resistance to movement of the stand does not vary smoothly: in particular there is a sharp stresses change at the opening of the front or rear "throat" of the rolls (periods of supply and tilting of the "blank - finished pipe" system).

As a result of the sharp load increase or fall and unilateral application of the moment to the leading gear wheels the stand starts to oscillate around the vertical axis. The summation of all these processes and causes a significant difference between the modules of practical skew indicators and their theoretical values.

Partial elimination of this phenomenon was achieved by setting a more stringent system of power impact transmission in the line of main drive [9], [2].

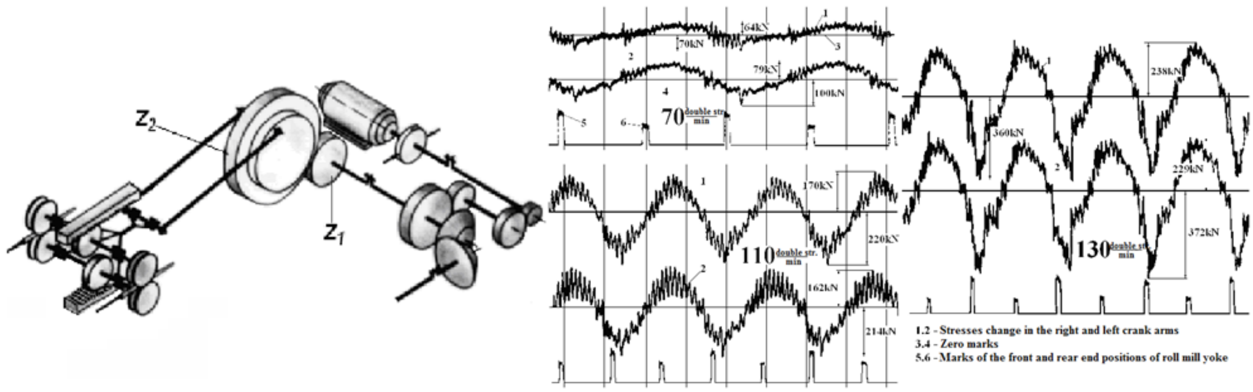

Figure 3. Kinematic scheme of the main mill drive HPT-2-40 and oscillograms of stresses in crank arms.

In a two line mills CPM-2-40 and CPM-2-90 the crank arms are placed between the axles of rolling. The simultaneous effect on the workpieces is made by two pairs of rollers arranged in a cassette, which is driven by the main motor through the gearbox by wheels z_1 and z_2. Wheel z_2 is mounted on the bearings, which inner diameter is greater than twice the radius of cranks of paired crank mechanism.

Fig. 3 shows the kinematic scheme of the main mill drive CPM-2-40 and and oscillograms of stresses in crank arms, measured in the manufacture of steel pipes with a diameter of 32 mm with wall thickness of 1.8 mm from the workpiece with a diameter of 57 mm with wall thickness of 3.5 mm. One-time supply of the workpiece was 18 mm, pace of work -70, 110 and 130 double strokes of stand per minute.

If in the crank arms of paired crank mechanism of mill CPM-32 the loading nonuniformity of crank arms reaches the two-triple values, the paired crank mechanism of mill CPM-2-40 will be characterized by the nonuniformity indicator of load distribution between the crank arms, which not significantly differ from one. However, the implementation of such a solution for single-strand mill is difficult not only for economic reasons. Placing the rolling axis between crank arms causes the application of eccentric crank mechanisms. The eccentric value depends on the diameter of the common shaft of crank-type wheels.

A number of the proposals to organize a more uniform loading of crank arms is known. Thus solution [4] includes setting one of the crank-type wheels, rotatable about the crank-type shaft. According to [11], the hub of the crank-type wheels gear proximal to the shaft drive end is performed by the hollow and elongated relatively to the ring in direction of the other gear and connected with the shaft at the location of the gear ring of crank-type wheel via a bearing mounted in the cavity, and is rigidly by an elongated end. It should be noted that with such a decision the control of branches stiffness values, which transfer the rotation to the gears of crank-type wheels must also be made.

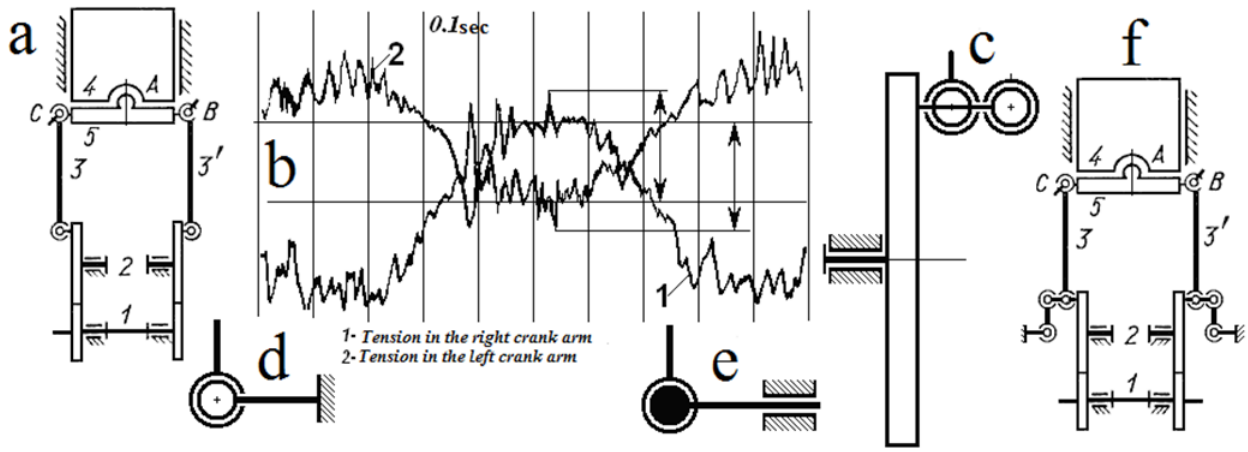

Figure 4. Statically determinated crank mechanism of mill CPM-32, oscillogram of stresses in crank arms and amplification schemes of cranks reference nodes

Close to the optimal one is the solution presented in Fig. 4a [6], under which the gear wheels, on which cranks are mounted, are set in cantilever fashion on individual support nodes 2 and stand bearing the rolls 4 is connected with crank arms 3 and 3′ by intermediate link 5.

The implementation of this solution was tested in a production environment [10], showing a number of positive effects (improving product quality, increasing the lifetime of the crank arm group elements).

However, the constraints imposed by the peculiarities of the implementation of maintenance solution, force to improve this structure.

To increase the load capacity of supports 2 the wrist pins are offered to be provided by additional spherical bearing nodes that allow to form a second support of the crank-type node forming with supporting frame a statically determinate group consisting of either a single unit (option *d* in Fig. 4), or two-link connection (option *e* in Fig. 4).

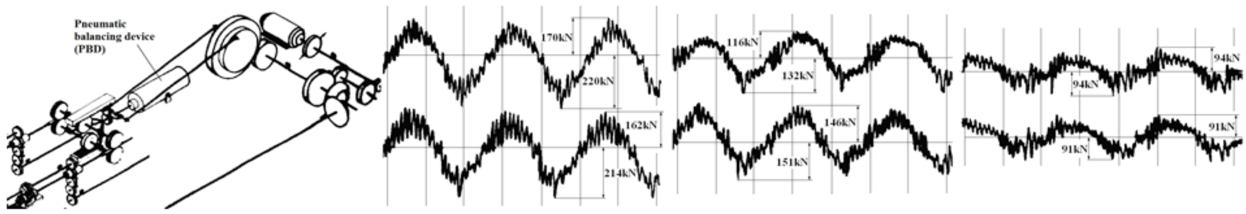

Figure 5. Oscillograms of stresses in crank arms at different degrees of balancing. The pace of work 110 double str./min

From the oscillograms shown in Fig.1, Fig.3 and Fig.4 it follows that the actual laws of links loading that provide reciprocating motion of stand, differ from the ideal ones, which are formed by the process loads and dynamic features of the crank mechanism. The the curves reflecting the net effect of links inertia technology, which frequency characteristics match the mill pace of work, are "stacked" by components causing these differences, which are formed due to the aforementioned factors.

Fig. 5 shows the study results of the most "dynamic mode" of the mill CPM-2-40 operation at various degrees of balancing. If the impact of PBD, localizing the inertial component of moving with variable speed masses, has provided a nearly twofold decrease in the level of crank arms loading and reduce of their loading unevenness, the ratio of swing amplitudes of vibrodynamic component remained almost unchanged, if not increased.

Great static and dynamic loads affect the drive crank arm directly from the part of the mobile work stand, which in some cases can lead to sudden loss of its stability within the elastic limit of mechanical system, according to the criteria set in studies [3, 8]. Providing the dynamic stability crank arms requires the exclusion of certain conditions of implementation of tubes pilger rolling, causing the emergence of various kinds of unwanted parametric oscillatory phenomena in the mechanical system of main power line. On the basis of a detailed study of the dynamic characteristics of the considered complex mechanical system the reduction of crank arms vibration activity can be achieved. Below is the study of parametric transverse oscillations of the crank-type group of main drive of mill CPM considering the unsteady interaction of the deformation and the inertial forces of the mobile work stand, which distinguishes it from the known solutions [1, 7, 11].

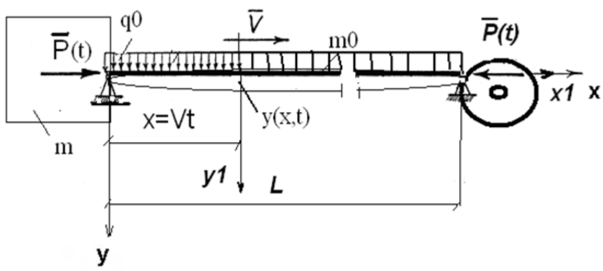

Figure 6. The calculation scheme of the crank arm of the main power line of mill CPM

The calculation scheme for the considered dynamical model of system is made taking into account the technological features of the cold rolling mills technology, reflecting the impact of the rolling force and intensity of the mass loading taking into account the movement of mill working stand (Fig. 6).

Consider the crank arm relative movement in the selected moving and fixed coordinate systems. Then the given calculation scheme is adequate to the actual manufacturing process of tube pilger rolling so it more deeply reflects the dynamic phenomena in the crank-type group of the main drive of mill CPM.

Oscillations of the crank arm power group of the mill CPM main drive in the transverse plane, according to the adopted calculation scheme (Fig. 6), are described by the differential equation

$$EI_z \frac{\partial^4 \acute{o}(x,t)}{\partial \tilde{o}^4} + m_0 \frac{\partial^2 y(x,t)}{\partial t^2} + \mathrm{P}(t) \frac{\partial^2 y(x,t)}{\partial x^2} = \begin{cases} q(x,t),\, 0 \le x \le \upsilon t; \\ 0,\, \upsilon t \langle x \le \ell, \end{cases} \tag{9}$$

where $y(x,t)$ – dynamic motions of the crank arm in transverse plane; EI_z - the flexural rigidity of the crank arm; $q(x,t)$ - the intensity of the crank arm loading by mass forces; $q(x,t)$ $-m_0$ – mass per unit length of the crank arm; ʋ – speed of the working stand movement.

The differential equation of transverse oscillations (1) for the crank arm is made of a mechanical system considering its inertia, intensity of the rolled pipe impact and time-varying inertia of stand working mill.

Analysis of experimental researches of power parameters of mills CPM [11] shows that the axial component of the rolling force affecting the crank arms group of mill CPM from the deformation zone, is periodic. Character of axial force change, in this case, advantageously corresponds to harmonic law

$$P(t) = P_0 + P_1 \cos(\omega t), \tag{10}$$

where P_0 – static component of axial forces of technological resistance; $P_1 = ma$ and ω - respectively dynamic component of forces and change frequency; m , a – respectively mass and acceleration of the working stand

There is a full basis to assume that in the first approximation, the conditions, described in [2] can be accepted. Consequently, under certain assumptions, note that the frequency of pipe rolling force approximately coincides with the frequency of the crank-type wheel rotation.

Then the differential equation of transverse oscillations of the crank arm image point (9), taking into account (10) will be rewritten as:

$$EI_z \frac{\partial^4 \acute{o}(x,t)}{\partial \tilde{o}^4} + m_0 \frac{\partial^2 y(x,t)}{\partial t^2} + \left(\mathrm{P}_0 + \mathrm{P}_1 \cos(\omega t)\right) \frac{\partial^2 y(x,t)}{\partial x^2} = \begin{cases} q(x,t),\, 0 \le x \le \upsilon t; \\ 0,\, \upsilon t \langle x \le \ell, \end{cases} \tag{11}$$

After the substitutions of mechanical system parameters and simplifications finally write the system of differential equations satisfying the conditions of the problem and adopted the mathematical model in the form of:

$$\frac{d^2 f_k(t)}{dt^2} + \Omega_k^2 \left[1 - 2\mu_k \cos(\omega t)\right] f_k(t) = \frac{2q_0}{k\pi m_0}\left[1 - \cos\left(\frac{k\pi\upsilon t}{\ell}\right)\right], \tag{12}$$

where Ω_k^2 – the squared frequency of oscillations of the crank arm, loaded by a static component of the shell rolling force P_0;

$\mu_k = \frac{P_1}{2\left[P_{\hat{e}\eth.\hat{e}} - P_0\right]}$ – coefficient of dynamic driving of the crank arm of main power line of mill CPM.

The dynamic driving coefficient μ_k and the free oscillations frequency of a crank arm Ω_k are determined by the conditions of mutual change of both the technological parameters of the deformation zone in cold pilger rolling, and the inertia parameters of the main drive of mill CPT.

The differential equation (11) is presented in the form of known parametric equations of Mathieu - Hill with the right side, which with a high degree of accuracy describes vibration activity of system “working stand - crank arm - power point.” The numerical solution of the differential equation in the Cauchy problem with the use of a Mathcad standard software package allows to get dynamic movements for the most common forms of crank arm oscillations.

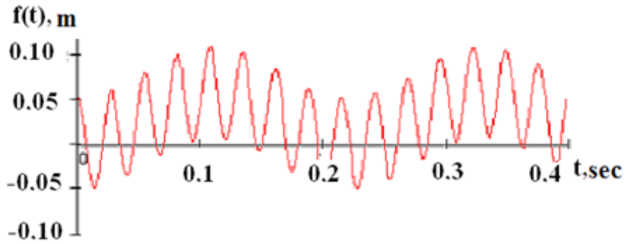

Figure 7. Dynamic movement of the crank arm of main drive of mill CPM-32 (workpiece ø38x1.8 - ø25x0.8 finished pipe, steel X18N10T)

Fig. 7 shows a fragment of the research operation of the crank arm of mill CPM-32 at the pipe rolling of steel X18N10T (workpiece ø38x1.8, finished pipe ø25x0.8), describing a dynamic process that takes place in the “working stand - crank arm - power point” system for the most characteristic first sinusoidal form of system oscillations. The confirmation of the study relevance is the results of experimental studies of the main drive of mill CPM-2-40, shown in Fig. 5. It can be seen that stresses changes in crank arms of its main drive indicate the presence of vibrodynamic processes in their formation.

It is shown that the given mathematical model of the "working stand - crank arm - power point" system with a high degree of confidence describes vibrodynamic processes in crank arms and significantly clarifies the description of all the main drive behavior. Note that in this case the picture construction of function stability of the "working stand - crank arm - power point" system of mill CPM, is reduced to the presentation and analysis of the well-known Ince - Stretta diagram (Fig. 8).

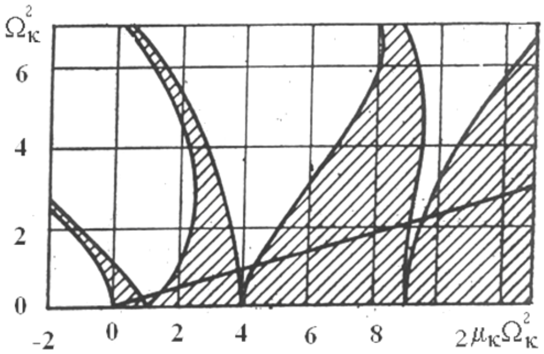

Figure 8. Diagram of function stability of main drive of mill CPM for generalized dynamic model of the "working stand - crank arm - power point" mechanical system

Diagram of function stability of the "working stand - crank arm - power point" system for the considered generalized dynamic model allows to analyze possible areas of parametric stability and to assess the role of crank arm vibration activity as part of the whole power line of mill CPM. One of the most important properties of the Mathieu - Hill equation is that at a certain ratio between its coefficients it has unlimited growing solutions. This provides the possibility to obtain a given ratio between the frequency of external force ω and the frequency of crank arms natural oscillations Ω. It is obvious that keeping the intensity of the impact on the rolled shell inertia parameters of crank arms and moving working stand is defined within the selected dynamic model of the considered mechanical system. Thus, the main factor limiting the rapidity of the main drive of mill CPM, and determining a low durability of its components and parts, is the significant manifestations of vibrodynamic processes. Basic dynamic loads are the result of significant inertial loads of stands elements, reciprocating and reciprocating rotational movements, and the presence crank arms of a certain rigidity in the main power line leads to additional high-dynamic components.

The ability to model the rolling modes in the design phase and the appointment of process parameters distinguishes the results obtained from the results of earlier known studies in the field of study of dynamic stability and vibration activity of the main drive equipment of mill CPM. In the future, the choice of rational modes of technological parameters of the pipe cold pilger rolling at the mill CPM must be carried out with the help of function stability diagram of the main drive (shaded areas). At the same time in order to determine the effect of the inertial loads leveling generated by mass of working stand on the values of dynamic loads in crank arms and shafting to this mass it is advisable to attach the relevant countervailing force. Balancing the inertia forces of the given mass of stand will cause a significant reduction (more than three times) of the main components of forces in crank arms. The moment of the wire shaft elastic forces significantly reduces. If at balancing of inertia forces of stand masses there is no balancing moment of inertia forces of its rotating masses, the significant reduce in the amount of dynamic forces in crank arms can not be achieved.

In view of the above, there are decisions to modernize the main drive line of mill CPM-32 by setting the crank arms of reduced mass, made by close tolerance forging from high-strength aluminum alloys [9, 11]. Furthermore, for balancing of loads on the crank arms of main drive wheel the uncoupling of crank-type wheels with the enhancement of support units and the attachment points of crank arm is provided (Fig. 9).

To increase the load capacity of supports crank wheel pins are equipped with additional spherical bearing units which allow to form a second additional support of crank-type unit. This group represents a statically definable structure consisting of either a single link or two-link connection. The crank arms, made of aluminum alloy, having the important non-linear characteristics and significant dissipative properties, perform the "extinguishing" of variables of working stand inert characteristics. Using connection with nonlinear characteristics leads to in the elimination of parametric resonance in the "working stand - crank arm - power point" system.

In order to ensure the convenience of repair works, maximum use of available spare parts and consumables in the proposed design the solution involving the use of bearings, operating at the units of the installation of crank-type wheels shaft and cranks connections with crank arms is implemented. Shaft-installation of crank-type wheels is performed on the basis of the connecting shaft, which provides a statical definability to the system, compact installation of the crank assembling unit in the supporting frame of mill camp and its joining it to stand without changing the existing methodology for conducting these operations.

Conclusions

1. Coupled crank mechanisms of the drive stand of most mills of pipes cold rolling are made by eccentric with asymmetrical (side) supply of torque to crank-type wheels. This leads not only to a significant distortion of the deformation zone, which leads to deterioration of production quality indicators, but also to the emergencies.

2. A number of decisions aimed at reducing the indicator of crank arms load unevenness is known. The closest to the optimal is the structural scheme, containing the separated crank-type wheels and an intermediate link of crank arms connection with supporting roll by movable stand.

3. A refined mathematical model of vibration activity for the "working stand - crank arm - power point" system is developed. Differential equation of parametric oscillations of mechanical system in the form of the Mathieu - Hill equation is made. A picture of parametric resonance for the crank arm main drive of mill CPM-32 is build.

4. Mathematical modeling of dynamic processes point to the need to optimize crank arm oscillations and reduce the vibration activity of system to the acceptable level. At that, the assessment of stability for the selected dynamic model of the mechanical system "working stand - crank arm - power point" was held on the basis of the basic Ince - Strutt diagram.

5. It is suggested and substantiated a number of technical solutions to modernize the crank-type node and crank arm of main drive for existing mills CPM (installation of crank arms from high-strength aluminum alloys and separation of crank-type wheels), aimed to eliminate the demonstrations of parametric oscillations in a mechanical system, reduce the dynamic loads and uneveness indicators of loading of working stand drive elements.

6. Having analyzed the experimental data, it was noticed that at the result of operation in a crank arms there are oscillating stresses of oscillatory character of alternating type, which in turn lead to the accumulation of damages or product destruction as a consequence. In the study of loadings patterns it was revealed that the stresses values are formed both a superposition (summing) of the main frequency, determined by the mill pace of operational (regime parameter) and an oscillatory process (constructive parameter), characterizing the properties of each crank arm. This results in a risk of the crank arm failure long before its planned lifetime.

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