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Complex study of dynamics of automatic pipe-rolling plant

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Abstract
The complex solution of a problem of dynamics for the chosen "working stand - shell - mandrel-bar" model of automatic pipe-rolling plant TPA taking into account time-varying of the system mass is considered. The differential equations of the movement for elastic elements of the chosen multi mass model of a working stand and specified differential equations of longitudinal oscillations of a mandrel with a bar taking into account change in time of mechanical system mass of automatic mill TPA are worked out. Results of research of dynamics of interaction of the rolled shell with elements of a working stand of the mill taking into account their elasticity are given. Expressions for the corresponding form of oscillations of deformable elements of a working stand of the automatic mill are obtained. Features of dynamics of the mill working stand elements are revealed. Dynamic functioning of a mandrel with a bar in the deformation zone and elastic system of the mechanism of holding of the mill mandrel are presented taking into account influence of non-stationary forces of technological resistance and variability of inertness of the attached mass of the piercing hollow billet. Nature of behavior of the mandrel holding mechanism is established taking into account change in time of the rolled pipe mass and non-stationary cyclic influence from the deformation zone. The complex analysis of non-stationary oscillations of "shell– mandrel –bar" mechanical system is represented by the coefficient of system dynamism.

Key words: AUTOMATIC MILL, WORKING STAND, SHELL, MANDREL, BAR, VIBRATION, ROLLING FORCE, DYNAMICS, VARIABLE MASS, DIFFERENTIAL EQUATION, LONGITUDINAL RIGIDITY, THICKNESS VARIATION, FORCED OSCILLATIONS, AMPLITUDE, FREQUENCY, MESHCHERSKY'S TASK

Introduction

Pipe-rolling plants with automatic mill are designed to produce hot-rolled seamless pipes of a wide range. The ability to quickly switch from one type of pipe production to another one defines the high efficiency of using TPA with automatic mill, while not excluding rolling of tubes of small batches.

Increasing production rate of seamless pipes on mills in the production line of pipe-rolling plants (APP) entails tightening the operation mode, both the main and auxiliary equipment.

Automatic mill is the most narrow place due to the current structural features of the sequence diagram in implementing the required processes in the production of seamless pipe in TPA production line [1]. Automatic mill distinctive feature is the presence of significant dynamic processes. Dynamic load in the mill and drive line at the stage of the workpiece grip is 3...4 times higher than the load at the steady rolling process.

Such character of loading reduces durability and reliability of the main equipment of automatic mills.

An analysis of literary sources

In the known studies of automatic mills dynamics of impact interaction of the workpiece and the rolls when gripping is considered without taking into account the elastic linkages of the system base equipment or the grip process is studied from an energy point of view apart from the steady rolling process. These assumptions lead to significant differences in the design loads of automatic mills elements from loads obtained in experimental studies.

In order to form a stable geometry of pipes rolled on automatic mill TPA, the stable dynamics of the base equipment has a practical value. Among the set of loads acting on the blocks and parts of automatic mill, the least unstudied are significant in size and non-stationary time-varying dynamic loads.

Problem statement

Experience of operation of domestic automatic mills TPA shows that when the forced feeding of the shells into mill grooves (accelerated feeding of the shell into the deformation zone is provided), there is some improvement in the grip of the shell by working rolls. The gripping of the shell by the rolls of the automatic mill, among other things, is significantly complicated by the fact that the shell during interacting with the working rolls is in contact with the mandrel and the bar system of the mandrel holding mechanism in the groove. These special conditions, along with all other factors, form the initial conditions of the technological process and affect the complex stress-strain state of the elements of the working stand and the rolled shell in the deformation zone, and form the nonstationary dynamics of the mill equipment in general. In this case, the forced supply of the rolled shell to the deformation zone of the mill by shell pushing aggravates the character of the dynamic processes formation. Determination of the real load spectrum allowed us to obtain recommendations for improving the automatic mill equipment in order to increase its technological capabilities, reliability and durability.

A number of works have been devoted to the investigation of complex dynamic processes of interaction between the rolled shell and rolls and the mandrel of mills [1 - 4].

In this case, a mathematical model of the non-linear process of the nonstationary interaction of rolled metal with mill rolls was considered in [3], the analysis of which later allowed us to obtain an expression for the corresponding simplified form of the dynamic load during transient processes.

This work was carried out in a complex formulation on the basis of the development of the previously accepted calculation scheme and the mathematical model of the dynamics of the basic units of the automatic mill. A definite attempt has been made to establish the influence of the basic parameters of the interaction of working rollers with a shell, a mandrel, and a mandrel holding mechanism. Obviously, the proposed approach is more correct and necessary for studying complex dynamic phenomena in the base elements of the working stand and the output side of the automatic mill.

It is known that for the implementation of stable technological operations of rolling the shell on the mill, by stabilizing the dynamics of the work stand and vibrating activity of the bar of the mandrel holding mechanism, numerous restraining, guiding, centering and thrust-adjusting mechanisms are used on the output side of domestic automatic mills [1, 4]. The mandrel holder perceives significant static and time-varying non-stationary dynamic loads on the side of the rolled shell. Due to the fact that the mandrel holder has considerable flexibility and a large mass, in the considered mechanical system significant in size and time-varying non-stationary dynamic loads occur; they cause its longitudinal oscillations along the rolling axis of the shell according to the corresponding harmonic forms. The mandrel together with the bar of its holding mechanism oscillates in the deformation zone. The center pilot of the mandrel deviates from the design position in the deformation zone (groove gorge of the working rolls) that often causes not standardized wall thickness variation of the shell (pipe) by technical documentation. The intensity of the non-stationary interrelated action from the side of the working stand and the deformation zone on the mandrel, the time varying in the inertia of the pipe and the stiffness parameters of the mandrel holder significantly complicate the description of the dynamic processes (Figure 1).

Ways of intensification of the technological process and issues of improving the quality of the rolled pipes dictate the conditions for improving the designs of all basic mechanisms of the working stand and the mandrel holding system.

Work objective

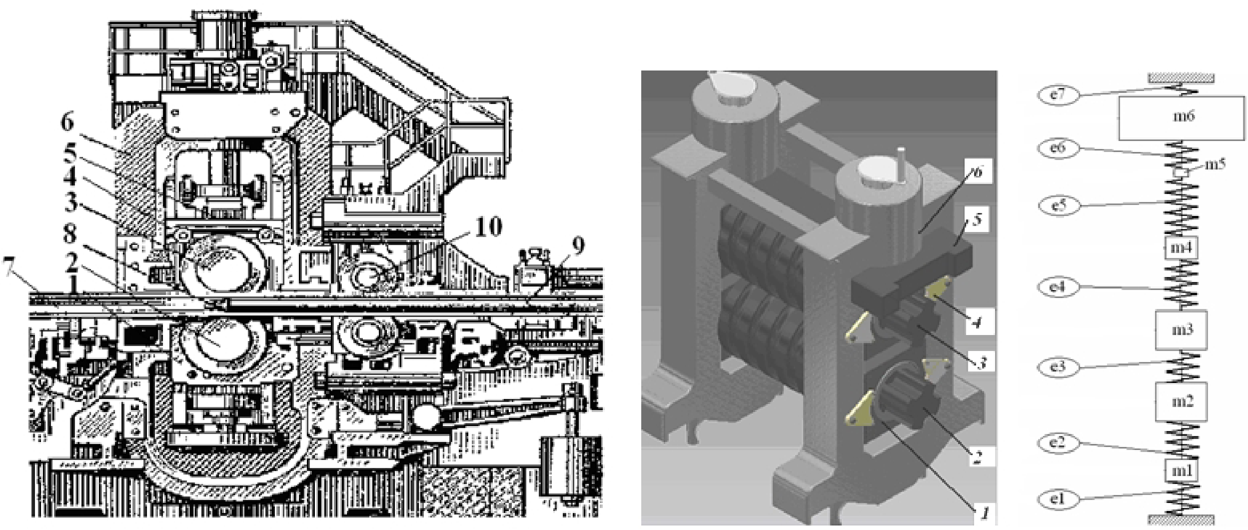
The objective of the work is to study the developed dynamic model "working stand - tube billet - mandrel – bar system", which will allow us to analyze the dynamic state of the elements both of the work stand and the bar system with the mandrel during the whole process of rolling the pre-pierced shell and on the basis of the above, to develop radical proposals for a comprehensive modernization of equipment at the output side of the mill.

Adjustment of the required parameters of the deformation zone is carried out in a complex way: by setting the working rolls with a wedge mechanism, the position of the mandrel on the roll gorge, the fixed-adjusting mechanism and the modernized intermediate wires installed along the entire output side of the automatic mill [4].

Method for solving the problem

In order to form scientifically grounded proposals for improving the design of the working stand, the equipment for the output side of the mill and the technology of producing pipes in mills, it is necessary to study more deeply the influence of various parameters and features of the technological process of rolling shells on the behavior of the entire mechanical system of the mill and the quality of the finished product.

The solution of this problem causes some adjustment of the calculation scheme and the further development of the mathematical model of the investigated system "working stand-mandrel-bar" the most accurately reflecting the actual processes occurring in the initial mechanical system of the automatic mill.



a) b) c)

Figure 1. Automatic pipe-rolling plant

a) three-dimensional model b) generalized dynamic system of elements of the working stand c) 1 - bottom chock; 2 – bottom roll; 3 – top roll; 4 - bottom chock; 5 - wedge mechanism for moving the top roll; 6 - stand of the working stand; 7- shell; 8- mandrel; 9 - mandrel bar; 8 - register plate; 9 - wedge pressure device; 10 - stripper rolls of the shell

In the present work, the developed dynamic and mathematical models "working stand-mandrel-mandrel bar" of an automatic mill are considered in the complex as an object of research.

This work differs from the known works [1-4] by complex approach to the study of interrelated dynamic processes with subsequent consideration of the time-varying inert characteristics of the rolled shell and the cyclically changing non-stationary technological loads acting on the side of the deformation zone of the working stand of the mill.

We turn to the compilation of mathematical models of the subsystems under consideration in order to investigate the interaction of the shell with the working stand units, the mandrel and the holding mechanism of the automatic mill bar system.

Mathematical models of the initial mechanical system of the automatic mill are represented in the form of mathematical models of interrelated subsystems: differential equations describing the behavior of the multi-mass model of the elements of the mill working stand (Fig. 1); differential equation describing the behavior of the selected model of the bar holding bar mechanism of the mandrel (Fig. 3).

Next, we proceed to simplifying the multi-mass dynamic model of the TPA mill stand and to the solution of the problem (Fig. 1b). The multi-mass dynamic model of the system is reducible to the three-mass model of the system using the technique [2, 3]. We present the problem in the basic statement of the problems of the mechanical system dynamics with taking into account of certain initial conditions.

Differential equations of motion of the elements of the working stand of cold reducing mill (CRM) in the first approximation for the accepted three-mass model of the system will be written as follows:

$$\begin{cases} M_1 \dfrac{d^2 y_1(t)}{dt^2} = P - C_{01} y_1(t) - C_{12}\left(y_1(t) - y_2(t)\right) - \mu_1 \dfrac{dy_1(t)}{dt}; \\ M_2 \dfrac{d^2 y_2(t)}{dt^2} = -P - C_{12}\left(y_2(t) - y_1(t)\right) - C_{23}\left(y_2(t) - y_3(t)\right) - \mu_2 \dfrac{dy_2(t)}{dt}; \\ M_3 \dfrac{d^2 y_3(t)}{dt^2} = -C_{30} y_3(t) - C_{23}\left(y_3(t) - y_2(t)\right) - \mu_3 \dfrac{dy_3(t)}{dt}, \end{cases} \qquad (1)$$

where M_i ($i = 1,3$) – given masses of elements of automatic mill working stand; C_i ($i = 1,3$) –given stiffness of the elements of mill working stand; y_i($i = 1,3$) – dynamic displacement of the elements of mill working stand in the vertical plane; $P(t) = P_0 + P_1 \sin(\omega t)$ – the vertical harmonic component of the pipe rolling force – frequency billet ω (P_0 static and P_1 amplitude value of the pipe rolling force).

Differential equations (1) are compiled and presented in the formulation of the Cauchy problem, which describes with sufficient accuracy the forced oscillations of the elements of the elastic subsystems of working stand of the automatic mill. Further, the solution of the system of differential equations (1) is implemented numerically using the Runge-Kutta method for the most common forms of oscillations of the "working stand-shell" system. Let us perform a more accurate calculation for the selected model of the forced oscillations of the elastic subsystems of the rolls of the automatic mill working stand on the basis of the initial data taken from [3, 4, 8]. Dynamic features and conditions for the functioning of the elements of the elastic subsystems of the rolls space (changes in the gap between the rolls) of the working stand of the automatic mill TPA 350 are shown in Fig. 2.

Analysis of calculation results (Fig. 2) and experimental studies show that the differential equations (1) with a sufficiently high degree of accuracy describe the forced oscillations of the elements of the elastic subsystems of the inter-rolls space of the rolls unit of the mill working stand. However, the obtained results are insufficient to establish the real reasons for the formation of the wall thickness difference in the automatic mill TPA. Next, we turn to the study of the longitudinal oscillations of the mandrel with the bar of its holding mechanism (Fig. 3). We turn to the adjusted solution of the problem of forced oscillations of a mandrel with bar system of its holding mechanism on the output side of the automatic mill TPA taking into account time-varying of the mass of the system.

To construct a correct model of the dynamic state of the system and the subsequent analysis of the mechanism of formation of the given wall thickness variation, we use the corresponding differential equation for the longitudinal motion of the mandrel with the bar along the rolling axis [2, 6].

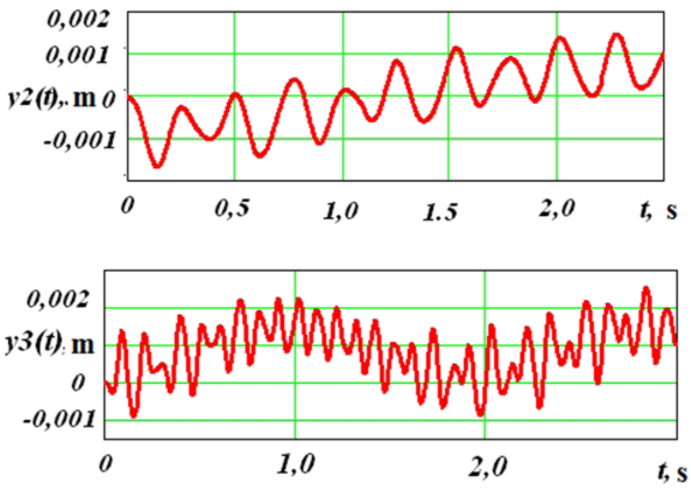
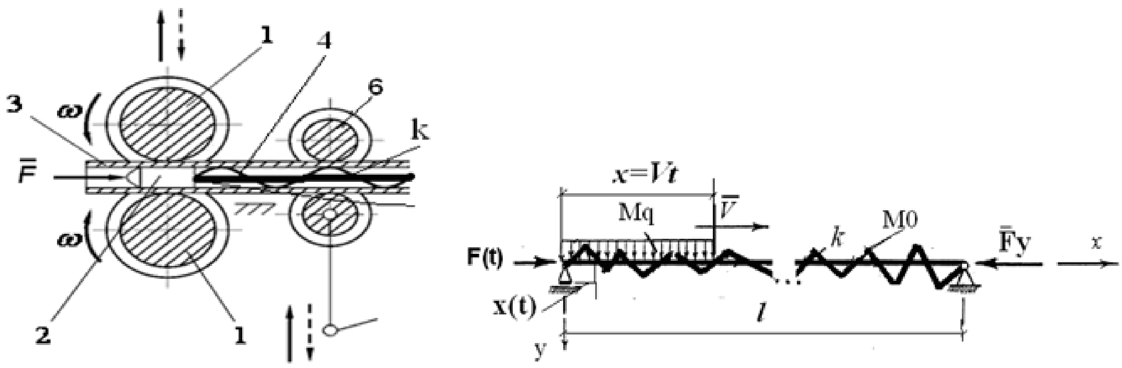


Figure 2. Dynamics of the elements of the rolls unit of the working stand of automatic mill TPA 350 (rough tube with diameter of 320 × 10 mm, material - steel 13GMF): $y_2(t)$ – bottom roll moving; $y_3(t)$ – top roll moving

We assume the bar elasticity force in the longitudinal direction of the rolling axis according to Hooke's linear law within the assumptions that the elastic bar with the mandrel is held on the rolling axis of the centering wires:

$$F(t) = kx(t).$$

Here k – given longitudinal stiffness of all the mandrel holding mechanism units (rigidity of the elastic systems of all units of the mill outlet side in the direction of rolling axis).



a) b)

Figure 3. a) Mandrel holding mechanism of automatic mill TPA; b) design scheme of the mechanical system

Under the assumption that the internal friction in the mechanical system is small compared to the cyclic technological load $F_0 \sin(\omega t)$ and nonstationary dynamic loads are insignificant according to [5, 6], we obtain a differential equation of the longitudinal movement of the mandrel with the bar.

$$M(t)\frac{d^2x(t)}{dt^2} + \frac{dM(t)}{dt}\frac{dx(t)}{dt} + kx(t) = F_0 \sin(\omega t)H(t), \qquad (2)$$

Where $x(t)$ – longitudinal displacement of mandrel in the deformation zone along the rolling axis; $M(t)$ – attached system mass variable in time taking into account the initial mass of the mandrel holding mechanism M_0; $H(t)$ – Pulse Heaviside function; ω – the frequency of the change in the driving force (according to [5, 6] is equal to the frequency of the wall thickness variation after the piercing mill TPA).

Let us note that the time-varying mass per unit length of rolled shells causes a change in the inertia of the whole mandrel holding mechanism, which largely determines the nature of non-stationary dynamic processes in the mechanical system.

Based on the results of a number of investigations of the automatic mill [1-3], the law of mass change of the system taking into account the variability in time of the mobile tube mass takes the following form:

$$M(t) = M_0 + M_q \left.\frac{x}{l}\right|_{x=\upsilon t} = M_0(1+\gamma t),$$

Where $\gamma = \frac{M_q}{M_0}\frac{\upsilon}{l}$ – index of inertness (mass change rate) of the mechanical system ($\gamma > 0$ the mass of the system increases); M_q $m_q l$ – mass of the rolled shell; υ – speed of movement (rolling) of the shell along the mandrel bar; $M_0 = m_0 l$ – initial mass of the bar system.

Based on studies of the dynamics of a body of variable mass of the formulation of the fundamental problem of I. V. Meshchersky [4], we take into account the important reactive add $\frac{dM(t)}{dt}\frac{dx(t)}{dt}$ of the inertial load of the rolled pipe in equation (1).

Therefore, to analyze the corresponding part of equation (2) taking into account the change in the mass of the system, the Cauchy problem under certain initial conditions is formulated. Under the assumption that there is internal viscous friction in the mechanical system in comparison to the cyclic technological loads according to [5, 6], we obtain the differential equation of the longitudinal movement of the mandrel with the bar. Then the differential equation of the longitudinal oscillations of the mandrel with the bar (2) taking into account the law of the tube mass variation in time in the statement of the Cauchy problem acquires the form:

$$\begin{cases} M_0(1+\gamma t)\dfrac{d^2x(t)}{dt^2}+(M_0\gamma\varepsilon+\mu)\dfrac{dx(t)}{dt}+kx(t)=F_0\sin(\omega t)H(t); \\ x(0)=x_0; \qquad \dfrac{dx(0)}{dt}=\upsilon_0. \end{cases} \tag{3}$$

Next, we proceed to determine the dynamic coefficient under the action of the reactive component. Taking into account the action of the reactive force, we solve the differential equation (5) for given initial conditions of the problem. After dividing the parts of equation (3) by M_0 we write:

$$\left(1+\gamma t\right)\frac{d^2x(t)}{dt^2}+\frac{M_0\gamma\varepsilon+\mu}{M_0}\frac{dx(t)}{dt}+\omega_0^2x(t)=\frac{F_0\sin(\omega t)}{M_0}H\left(t\right), \tag{4}$$

where $\omega_0^2=k/M_0$ – the square of the frequency of free longitudinal oscillations of the mandrel with the bar; μ – coefficient of viscous resistance of the system; ε – coefficient of internal friction in the mechanical system.

Analysis of the results

The inhomogeneous differential equation (4) is compiled and represented in the formulation of the basic Cauchy problem, which describes the forced oscillations of the mandrel with the bar of the piercing mill with sufficient accuracy. Further, the solution of the differential equation (4) is implemented numerically by the Runge-Kutta method for the most common forms of oscillations of the subsystem "shell -mandrel-bar".

Let us perform a more accurate calculation on the basis of the compiled mathematical model of the problem of forced oscillations of a mandrel with a bar in the example of an automatic mill TPA 350. To do this, we take the following initial data for automatic mill: $l=12,5$ m; $\upsilon=4$ m/s; $M_0=m_0l$; $M_q=m_ql$; $m_0=150$ kg/m; $m_q=100$ kg/m; $k=24\cdot10^6$ N/m; $\varepsilon=1$, $\mu=0,6$ Ns/m $t\in[0;6]c$. The results of numerical analysis of the differential equation (4) of the longitudinal oscillations of the mandrel together with the bar of its holding mechanism during rolling of rough pipes (liners) with a diameter of 320 × 10 mm, material - 13GMF steel on automatic TPA 350 are shown in Fig. 4.

The results of numerical analysis of the differential equation (4) of the longitudinal oscillations of the mandrel with the bar of its holding mechanism during rolling of rough pipes (shells) with a diameter of 320 × 10 mm (material - 13GMF steel) on automatic mill TPA 350 are shown in Fig. 4.

The calculated curve shown in Fig. 4 indicates the extremely unsatisfactory cyclic operating conditions of the mandrel with the bar during the process of shell rolling on the mill. The amplitude of the forced longitudinal oscillation of the mandrel in the rolls gorge of the working stand is 0.03-0.032 m, which exceeds the permissible adjustment values of the longitudinal movements of the mandrel in the groove of the deformation zone. Note that the adjustment value according to the grooving and the rolling table is 0.020 m.

Thus, when the reactive force acts, the nonstationary behavior of the "shell -mandrel-bar" system changes respectively. At the same time, the maxima of the dynamic coefficient become opposite to those that have been previously obtained for analogous models without taking into account the reactive force [3, 5].

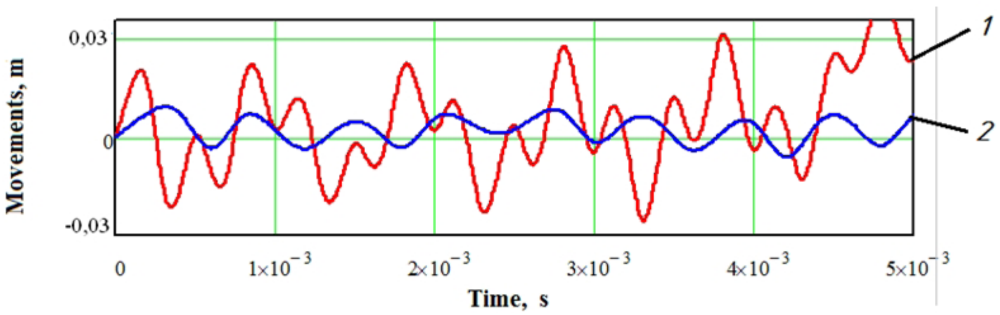


Figure 4. Longitudinal oscillations of mandrel with bar of automatic mill TPA 350 (rough tube (shell) 320 × 10 mm, material - steel 13GMF):

1 - before modernization of the mandrel holding mechanism; 2 - after the modernization of the mechanical system

Calculations show that under the action of reactive force in the system "shell (tube) - mandrel - bar" with a linearly increasing mass, the first maximum of the dynamic coefficient isK $K_{\partial} < 2$, and the subsequent maxima are less than the first. Consequently, there is a certain stabilization of the dynamics of the "shell (tube) - mandrel-bar" system, which is close to the results of experimental studies of the automatic mill TPA 350 [1, 4].

An analysis of the results of a complex calculation of the dynamics of an automatic mill indicates that the mathematical models compiled reliably of the describe nonstationary dynamic phenomena in the system "shell -working stand-mandrel-bar". The amplitude-frequency characteristics of dynamic processes when longitudinal oscillations of the system during the entire process of rolling the shell on the automatic mill TPA 350 exceed the permissible level of oscillations of the mechanical system.

In the course of implementation of the technological process, the dynamism of the system "shell (tube) - working stand - mandrel - bar" entails the formation of an increased wall thickness variation of the shell, which further is complex and difficult to remove. It is obvious that taking into account the dynamics of the mill working stand, the intensity of the impact from the deformation zone and the variability of the mass of the rolled shell approaching with the rolling speed $\vec{\upsilon}$ are the defining parameters in the framework of considered model of the problem.

The possibility of complex mathematical modeling of different modes of shell rolling at the stage of designing the technological processes of pipes rolling on an automatic mill TPA 350 significantly distinguishes the results obtained from the results of earlier known works in the field of studying the dynamics and vibrational activity of the elastic subsystems of the working stand and the elements of the mandrel holding mechanism [1, 4]. Fig. 5 shows the change of the wall thickness variation (thickness gauges of ultrasonic testing) along the length of the batch of roughing pipes (cartridges) rolled on an automatic mill TPA 350

Fig. 5 shows the variation in the difference (ultrasonic thickness measurements) along the length of the batch of roughing pipes (shells) rolled on an automatic mill TPA 350.

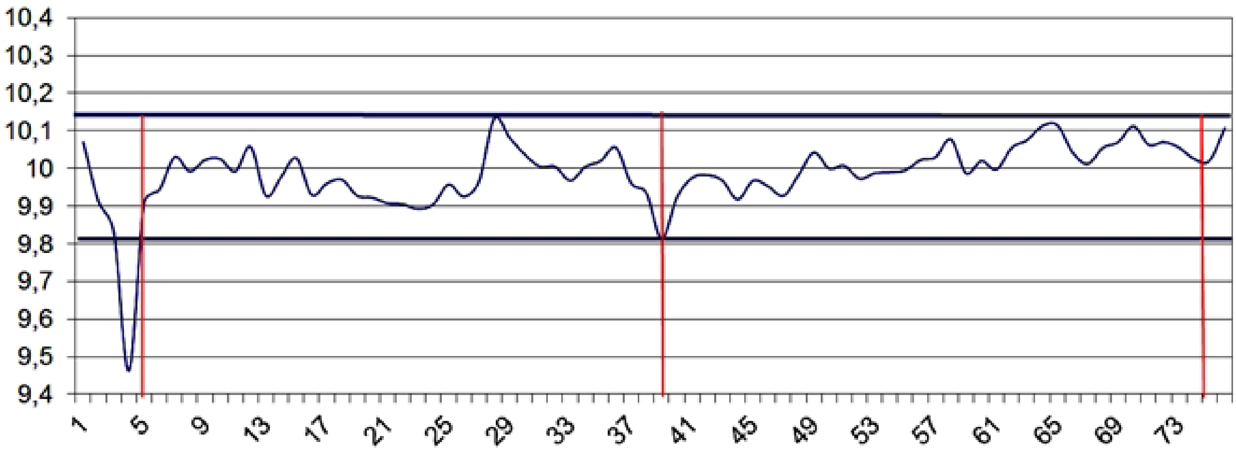


Figure 5. Longitudinal development drawing of the average thickness of the rough pipe wall (diameter 320 × 10 mm, material - steel 13GMF) after the automatic mill TPA 350 (red indicates the periods of high wall thickness variability of the system)

Excluding the influence of a number of other factors ("temperature wedge") when rolling on a longitudinal rolling mill, we can distinguish the effect of the oscillatory movements of the "shell- working stand-mandrel-bar" system. At the same time, its period of oscillations and the commensurate amplitude of oscillations with oscillations of the mechanical system "shell-working stand-mandrel-bar" of automatic mill in each respective period are clear.

In general, without taking into account the change in the magnitude of the longitudinal wall thickness variation due to the temperature change along the ends of the rolled pipe, it can be concluded that the additional change in longitudinal wall thickness variation (Fig. 5) should be done by a significant amount. So, when rolling a rough pipe with a diameter of 320 × 10 mm (material - steel 13GMF), the longitudinal component of the wall thickness variation increased by another 0.3 mm.

The results dictate the need for modernization of the mandrel bar holding mechanism and changing the design of the mill working stand by creating a prestressed stand, installing rolls on the basis of rolling bearings, anti-bending devices of working rolls of modern roller guides.

Conclusions

The design schemes of the interconnected mechanical subsystems of the automatic mill TPA have been clarified and mathematical models of the dynamics of the system "working stand-shell-mandrel-bar" have been developed within the framework of adopted models. The variability in time of the mass of the rolled pipe, the cyclic nature of the technological load of the deformation zone, the elasticity parameters and the dissipation of the base elements of the automatic mill working stand and the mandrel holding mechanism are taken into account.

The study of the dynamics of the mandrel holding mechanism showed that, when the intensive impact from the deformation zone on the working stand and the mandrel, taking into account the variability in time of the mass of the rolled shell is the most important factor in determining the dynamism of the system. It is established that the dynamics of the automatic mill TPA 350 are changed monotonically during longitudinal oscillations of the mandrel in the deformation zone. In this case, the reactive force and other inert characteristics of the system have a significant impact on the dynamics of the mandrel with the bar and the working stand of the mill, which ultimately determines the basic parameters of the mechanism for the formation of the shells wall thickness variation.

Results of numerical solution of differential equations of oscillations of the multi-mass model elements of elastic subsystems of the working stand and longitudinal oscillations of the mandrel with a bar of an automatic mill TPA 350 are presented. This allowed to evaluate the amplitude-frequency characteristics in complex of both the mill working stand rolls and the longitudinal displacements of the mandrel itself in the deformation zone taking into account the variability in time of the mass of the rolled shell. It is found that the amplitude of the forced vibrations of the working stand in the respective phases reaches 0.0023 m, and the displacement of the mandrel over the wedge of the working rolls is 0.032 m. The obtained amplitude greatly exceeds the permissible values of the stand parameters of the automatic mill TPA 350 deformation zone and leads to an increased wall thickness variation in the shells.

At the stage of designation of technological processes for shells rolling, by means of complex mathematical modeling of the dynamics of the working stand and the study of longitudinal oscillations of the mandrel with its holding mechanism, rational rolling modes have been established by calculation taking into account the predicted indices of the shells wall thickness variation and finished pipes. Calculations show that, for example, when rolling shells with a diameter of 320x10, the material is 13GMF steel, a stable rolling process is performed at the optimum rolling speed of the shell on the automatic mill TPA 350 of no more than 3.2 m/s.

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