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Refined mathematical model of vibroactivity of tube screw-rolling mill mandrel bar

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Abstract

The task concerning vibroactivity for refined dynamic model of mandrel bar acoustic positioning system of screw-rolling mill is solved. Difference equations of oscillatory motion are made and mathematical model of mandrel bar with account for cooperation with rolled pipe is refined. Analysis of vibroactivity and other dynamic processes in mechanical system of screw-rolling mill is carried out.

Keywords: *dynamic, vibroactivity, core, bloom, mass, stiffness, mandrel, oscillations, centering unit, variation in wall thickness, pipe.*

Introduction

While piercing of hollow billet the mandrel bar and rolled bloom on the axis of rolling are functionally hold by the complex of special supporting mechanisms called centering unit. On delivery ends of piercing mills of pipe-rolling plant alongside centering unit base and auxiliary operations are fulfilled, including: holding of fast-rotating, large and long mandrel bar along the geometrical axis of rolling; centering of piercing bloom that while rolling which has complex screw motion and vast inertial properties; delivery of rolled blooms from one mill to another ones [1].

For stable configuration of rolled pipes formation oscillations of rotating mandrel bar are of practical importance. For realization of necessary and stable technological operations of hollow billet piercing at the delivery end of piercing mills numerous guides, centering units, supporting and adjusting gears are used. Rotating long mandrel bar takes critical deadweight and changing in time dynamic loads from the piercing hollow billet. Because the mandrel bar is more flexible and inertial it causes in considered mechanical system critical in length and changing in time dynamic loads, which provide its bend along the piercing axis on sinusoidal forms.

As a result of uncontrolled wide oscillations the core along with mandrel moves in deformation zone along rolling axis (centering holder parallel goes off from overclamping of working rolls), causing variation in pipe wall thickness. It is obvious that dynamic phenomena arising in mandrel bar influences negatively on the history of hollow billet piercing that is reflected on bloom configuration and the quality of pipes.

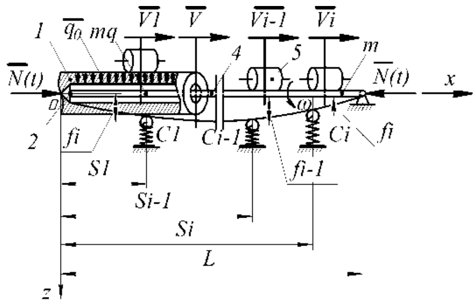
The ways of technological process intensification and upgrade points of rolled pipes dictate the necessary conditions for improving the design of mandrel bar centering unit, located along processing line and properly to delivery ends of screw-rolling mills.

The influence of different characteristics and technological peculiarities of pipe rolling on behavior of the system and quality of the finished product should be deeply studied in order to set scientifically based offers concerning the improvement of design of mandrel bar centering unit of piercing mills and pipe production technique.

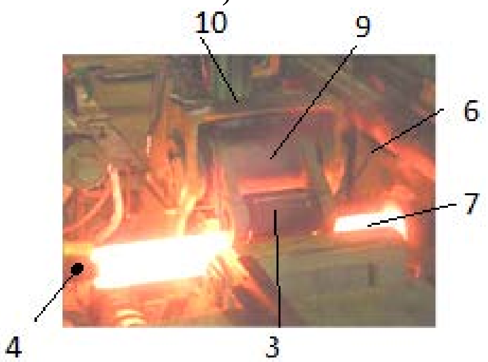
Solution of this task often determines refinement of design model and development of mathematical model of experimental TPA piercing mill which reflects sufficiently real processes taking place in the initial mechanical system.

In this work as subject of research the developed dynamic and mathematical models of holding device frame structure of TPA piercer plug are considered which differs from already known [1,2] in characteristics of moving updated centering units.

Change of degree of impact, mass of hollow billet and dynamic parameters of mandrel centering unit, traveling while piercing along the rolling axis, complicates considerably the description of dynamic processes.



a)



b)

Figure 1 Developed generalized dynamic model a) and construction of fixed non-updated moving enclosed centering unit b) of TPA piercer plug holding device (designed by OJSC "EZTM")

Research of developed dynamic model "framed structure – hollow billet" will allow to analyze dynamic behavior of framed structure with mandrel within all the process of hollow billet piercing and, based on the above, to solve the task concerning upgrading of mill centering units (figure 2).

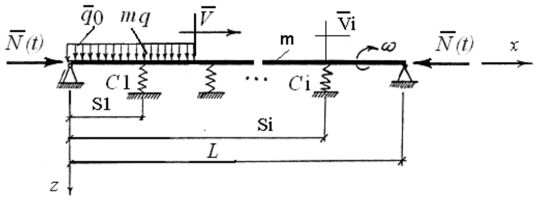


Figure 2 Design model of framed structure of piercer plug holding device

Let us assume the design model of framed structure of screw-rolling mill as hollow core of uniform section with hinged bearings on the ends and moving elastic supports (centering units) between them. The core rotating with angular rate ω around rolling axis x, is subjected to the force of piercing $\vec{N}(t)$ from the side of deformation zone. Herewith along the core with semi-constant rate of piercing $\vec{V}$ (uniformly) distributed load of rolled bloom with intensity $\vec{q}$ and mass per unit length m_q moves and rotates along with the core.

For building the refined system mathematical model and evaluating of dynamic condition of the core with mandrel and processes of hollow billet piercing let us use difference equation of oscillations of rotating uniform core on the elastic foundation of centering units [3,4].

$$m\frac{\partial^2 u_z}{\partial t^2}+\frac{\partial^2}{\partial x^2}\left[EI\frac{\partial^2 u_z}{\partial x^2}+f\left(\frac{\partial^3 u_z}{\partial x^2 \partial t}-\omega\frac{\partial^2 u_y}{\partial x^2}\right)\right]+$$
$$+\kappa\frac{\partial u_y}{\partial t}+cu_z=(m+m_q)\omega^2\left[e_1\sin(\omega t)+e_2\cos(\omega t)\right], \qquad (1)$$

where EI, m - bending stiffness of the core and its mass per unit length respectively; u_z – movings along z axis; c,k – coefficients of elastic foundation of centering units which depend on moving and speed respectively; f – coefficient reflecting viscous friction within the system while oscillation of mandrel bar; $e_1(x)$, $e_2(x)$

- eccentricities specifying the mass unbalance of mandrel bar round the axis of rolling.

Under the assumption of the fact that internal friction within the system is miserable as compared with technological and dynamic loads, and mass unbalance of mechanical system belongs to one subspace $e_1(x) = e_2(x) = e(x)$, according to [3,5], we may have the simple differenceequation

$$m\frac{\partial^2 u_z}{\partial t^2} + EI\frac{\partial^4 u_z}{\partial x^4} + cu_z = (m + m_q)\,\omega^2 e(x)\sin(\omega\, t), \tag{2}$$

where ω is angular velocity of rotation of mandrel bar round rolling axis.

Let us substitute the elastic foundation of mandrel bar with finite number i of yielding supports of movable centering units $\sum_{i=1}^{n} c_i u_y\ \delta\left(x - (s_i \mp V_i t)\right)$. Then the difference equation (2) is expressed as the following

$$m\frac{\partial^2 u}{\partial t^2} + EI\frac{\partial^4 u}{\partial x^4} + \sum_{i=1}^{n} c_i u\,\delta\left(x - (s_i \mp V_i t)\right) = (m + m_q)\omega^2 e(x)\sin(\omega t) \tag{3}$$

where $\delta(x - x_{oi})$ is Dirac delta function; s_i - design positions of supporting nodes of centering units along the axis of mandrel bar; V_i is the moving velocity of corresponding centering units of mandrel bar along the rolling pipe axis.

Taking into account that Dirac function has the following typical filter property, according to [5]

$$\int_a^b f(x)\,\delta(x - \xi)\,dx = f(\xi); \qquad [a < \xi < b],$$

in respect with changing of longitudinal compression force (piercing strain), according to [2] and to the law $N(t) = N_o + N_1\cos(\nu\, t)$ and impending bloom strain with intensity $\vec{q}(x,t)$ and elastic supports (movable centering units) the difference equation will be as following

$$m\frac{\partial^2 u_z}{\partial t^2} + EI\frac{\partial^4 u_z}{\partial x^4} + N(t)\frac{\partial^2 u}{\partial x^2} + \sum_{i=1}^{n} c_i u_z\ \delta\left(x - (s_i \mp V_i t)\right) - q(x,t) = (m + m_q)\omega^2\, e(x)\sin(\omega\ t), \tag{4}$$

where $q(x,t)$ is the degree of impact of piercing hollow billet on the mandrel bar which is of certain degree of heaviness and which may be found in accordance with [2,4] as following

$$q(x,t) = q_o - m_q\left(\frac{\partial^2 u}{\partial t^2} + V^2\frac{\partial^2 u}{\partial x^2} + 2V\frac{\partial^2 u}{\partial x\,\partial t}\right)\Bigg|_{x=V\,t} \tag{5}$$

Here q_0, m_q are the intensity and mass per unit length of rolled bloom respectively; v – is traveling speed (piercing) of a bloom along the mandrel bar; $\frac{\partial^2 u}{\partial t^2}$ - relative acceleration of a bloom; $V^2\frac{\partial^2 u}{\partial x^2}$ - translational acceleration of a bloom; $2V\frac{\partial^2 u}{\partial x\,\partial t}$ - Coriolis acceleration of rolled bloom.

In respect with typical shape of bent flexible bar while piercing of a bloom in a first approximation let us assume eccentricity equation of its axis towards the axis of piercing in sinusoidal form

$$e(x) = u_z = f(t)\sin\left(\frac{\pi x}{\ell}\right). \tag{6}$$

For solution of the equation (4) let us use the well-known scheme of Bubnov- Galerkin

$$\int_0^{\ell} L(u)\sin\left(\frac{\pi x}{\ell}\right)dx = 0. \tag{7}$$

Herein, for Bubnov- Galerkin scheme, according to [2,3], functional of difference equation (4) in respect with (5) and (6) will be expressed as

$$L(u) = m\frac{\partial^2 u_z}{\partial t^2} + EI\frac{\partial^4 u_z}{\partial x^4} + N(t)\frac{\partial^2 u_z}{\partial x^2} + \sum_{i=1}^{n} c_i u_z\ \delta\left(x - (s_i \mp V_i t)\right) - q(x,t) - (m + m_q)\omega^2\ f(t)\sin\frac{\pi x}{\ell}\sin(\omega\, t). \tag{8}$$

Inserting (8) into (7), in respect with natural mode shape orthogonality of mechanical system, after integrating and rearranging we will have the best possible difference equation of representation point movement on the axis of mandrel bar

$$\left[1+\frac{m_q}{2\pi m}\left(2\pi\frac{Vt}{\ell}-\sin\left(2\pi\frac{Vt}{\ell}\right)\right)\right]\frac{d^2 f(\mathrm{t})}{dt^2}+\frac{Vm_q}{m\ell}\left(1-\cos\left(2\pi\frac{Vt}{\ell}\right)\right)\frac{df(\mathrm{t})}{dt}+$$
$$+\left[\frac{EI}{m}\left(\frac{\pi}{\ell}\right)^4+\frac{2}{m\ell}\sum_{i=1}^{n}c_i\sin^2\left(\frac{\pi(s_i\mp V_i t)}{\ell}\right)-\frac{1}{m}\left(N_0+N_1\cos(\nu t)\right)\left(\frac{\pi}{\ell}\right)^2-\right.$$
$$\left.\frac{m_q}{m}\frac{V^2}{\ell^2}\pi\left(2\pi\frac{Vt}{\ell}-\sin\left(2\pi\frac{Vt}{\ell}\right)\right)-(1+\frac{m_q}{m})\omega^2\sin(\omega t)\right]f(\mathrm{t})=\frac{2q_0}{m\pi}\left(1-\cos\left(\pi\frac{Vt}{\ell}\right)\right). \qquad (9)$$

Dynamic of simulated bar system in respect with different modes of rolled pipes on the TPA 140 piercing mill is represented with numerous solutions of difference equation (9) in accordance with Runge-Koht method.

Derived curves, which are given on the figure 3, denote extremely unsatisfactory conditions for operation of delivery end devices of piercing mill. While realization of technological process there occurs formation of increased variations in wall thickness of blooms that in the following may be of congenital and hard to remove character.

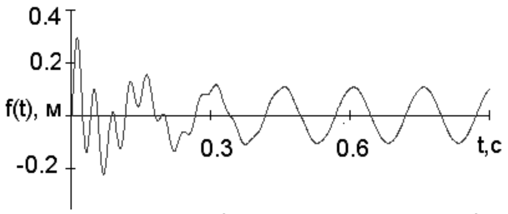


Figure 3 Dynamic of holding motion core of TPA 140 piercing mill mandrel (bar of 130 mm in diameter, material – steel 20)

With the help of mathematic modeling the influence of different characteristics of mandrel bar holding device and technological peculiarities of pipe manufacturing process on the behavior of developed dynamic model of mechanical system was established. Herein certain principal measures on updating of equipment, adoption of reasonable hollow billet piercing modes are developed and some suggestions on updating of pipes manufacturing processes, for example using TPA 140 piercing mill, are offered.

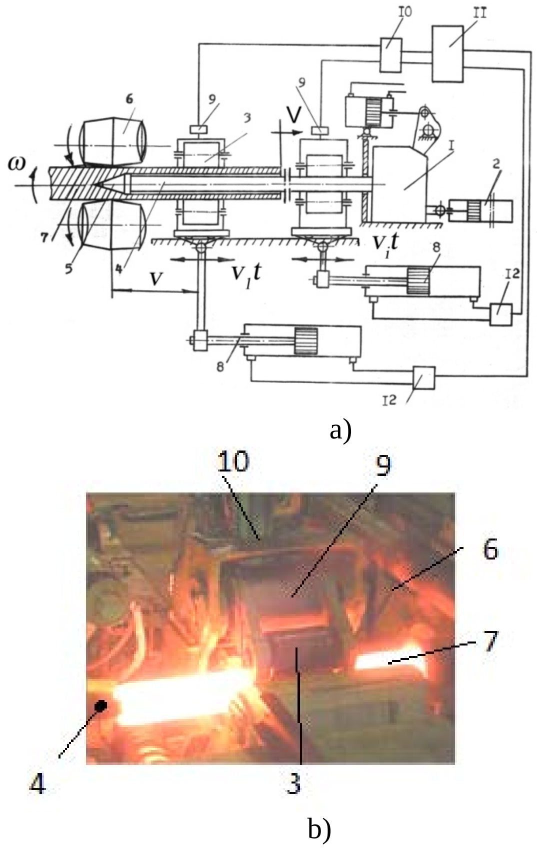


Figure 4 Holding device of TPA piercing mill mandrel bar with moving centering units a) and updated centering unit of mandrel bar of TPA140 piercing mill b)

Analysis and design of complex mechanical system vibroactivity is executed by step-by-step modeling of dynamic processes that allows to abandon from expensive and complex experimental investigations of TPA mills.

Analysis of refined and developed mathematical model of piercing mill mechanical system and further design of results obtained defines the influence of piercing velocity, relativities of system mass, piercing strain, degree of impact of rolled bloom, rate speed of the core and stiffness of moving supporting mechanisms (centering units) on vibroactivity of bar system mechanism of mandrel holder. Results of system vibroactivity calculations denote high instability of dynamic, susceptibility of dynamic model to changes of dynamic parameters of mechanical system and technological processes.

It is obvious that vibroactivity of bar system while realization of necessary technological piercing processes goes to mandrel located in overclamping of deformation zone, that leads to distortion of deformation zone shape and degeneration of quality of piercing blooms (tubes).

Realization of stable manufacturing processes of piercing is provided by choice of reasonable characteristics of system dynamic model and best performance of mills on the appropriate projecting phases and system operation.

Peculiarities of TPA 140 piercing mill operation denote the necessity of updated moving centering unit's system usage. For reduction of vibroactivity of core holding device of a mandrel the delivery end of TPA piercing mill (figure 4) should be updated, the chain of moving automatically installed centering units with longish body of centering roller should be placed along the rolling axis (developed construction of OJSC “EZTM”, Russia) [1,6].

Conclusions

1. Solution of a problem concerning vibroactivity for developed dynamic model of mandrel holding device of screw-rolling mill is given. Refined mathematical model for core holding device of a mandrel and dynamic model of mechanical system in respect with characteristics of supporting mechanisms of moving centering units of a mill delivery ends is made.

2. With the help of mathematical model of vibroactivity of mandrel holding device the best regimes of hollow billet piercing in respect with forecasting quality index of produced pipes and specifications of allowable vibroactivity of mill mandrel bar are determined.

3. Updating of TPA piercing mill delivery end with further installation along the piercing axis series of moving automatically installed centering units with longish body of centering rollers (l=900 – 1100mm) is suggested.

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