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Peculiarities of formation of nonsteady processes during impact interaction of hollow billet with rolls of automatic mill

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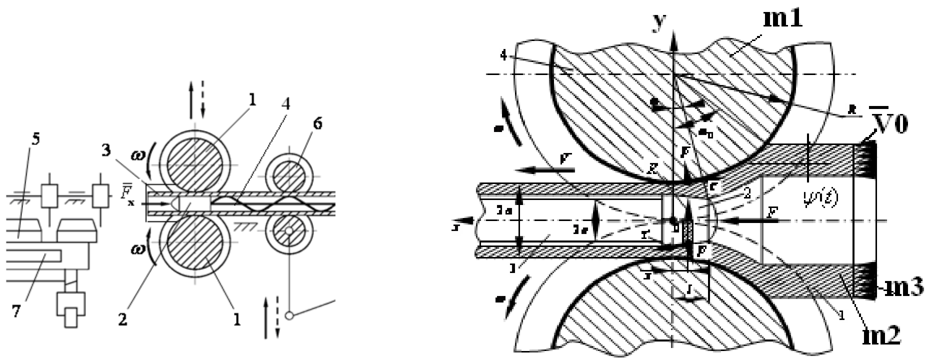
Abstract
The results of researches of dynamic of nonsteady impact interaction of the hollow billet with rolls of automatic mill are presented in the work. Refined mathematical model of nonsteady interaction of working rolls of automatic mill with rolled billet is developed. The peculiarities of behavior of mechanical system are determined. The expressions for determination of mechanical system responsiveness in respect with corresponding parameters of impact impulse and dynamic interaction of the billet with working rolls of automatic mill.
Key words: HOLLOW BILLET, TUBE, TECHNOLOGY, ROLL, GROOVE, MANDREL, AUTOMATIC MILL, DYNAMICS, DEFORMATION ZONE, IMPACT IMPULSE, MASS, MATHEMATICAL MODEL, PRESSURE, CRUMBLING, DYNAMIC DEFLECTION

Growing work speed of pipe-rolling plants (PRP) causes stiffening of functioning conditions of main and auxiliary equipment of the mills of all technological line. Automatic mill in view of some reasons is the tightest place in the line of PRP at the realization of required technological processes of seamless tubes manufacturing on the PRP [1].

Transition processes are followed by significant impact loads during rolling of billets on the automatic mill. They arise, as a rule, during biting of a tube with massive working rolls.

It should be mentioned that at forced feed of the billet into the grooves of a mill (there provided billet feed by the pusher into the deformation zone) there appears certain improvement of billet biting by working rolls. Biting of the billet by automatic mill rolls among other thing is considerably complicated by the fact that the billet when hitting the roll simultaneously interacts with the mandrel in the groove and all the framed structure of mandrel holding mechanism (fig. 1). These conditions together with other may form initial conditions of the process and complex strain-stress state of the metal in the deformation zone and nonsteady dynamic processes in the elements of reduction mill and main drive gear of the mill.



a) b)

Figure 1. The schemes of automatic mill of pipe rolling plant and interaction of the hollow billet with rolls: 1 – roll; 2- mandrel; 3 – hollow billet; 4 – mandrel core; 5 - fore plate; 6 - stripper rolls; 7 – pusher of the billet.

Among the total amount of dynamic loads, acting in the main line of automated mill of pipe rolling plant (PRP), the least explored is significant in value short impact loads, caused by the interaction of hollow billet with grooved rolls during forced metal biting. Let us mark that forced billet biting by the rolls of the mill along the rolling axis is provided by pneumatic billet pushers.

The range of works [2-4] is devoted to research of similar dynamic processes of impact interaction of the billet with rolls of longitudinal rolling mill.

Herein mathematical model of non-linear process of impact interaction of the billet with rolls of longitudinal rolling mill is suggested in the work [2], linearization of which allowed to obtain expressions for corresponding simplified form of impact impulse.

This work is fulfilled on the base of development of excepted mathematical model, where there was taken an attempt to determine the main parameters of impact interaction of working rolls of automatic mill with the hollow billet in non-linear statement of a problem.

It is obvious that suggested approach is more adequate and convenient during study of complex dynamic phenomena in the elements of main drive gear of automatic mill (PRP).

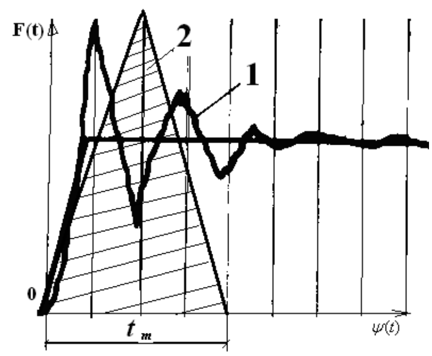


Figure 2. The character of formation of impact force during interaction of the hollow billet with rolls of automatic mill: 1 - rolling pressure; 2 – averaged form of impact impulse of the billet against the roll

For investigation of the impact interaction of the billet with working rolls of automatic mill we will use the system of connected differential equations

$$\begin{cases} m_1 \dfrac{d^2 y(t)}{dt^2} + c y(t) = F(t)\cos(\dfrac{\alpha}{2}); \\ m[\dfrac{d^2 y(t)}{dt^2} + \dfrac{d^2 \psi(t)}{dt^2}] = -F(t)\cos(\dfrac{\alpha}{2}); \end{cases} \quad (1)$$

$$\begin{cases} m_1 \dfrac{d^2 x(t)}{dt^2} + c x(t) = F(t)\sin(\dfrac{\alpha}{2}); \\ m[\dfrac{d^2 x(t)}{dt^2} + \dfrac{d^2 \psi(t)}{dt^2}] = -F(t)\sin(\dfrac{\alpha}{2}); \end{cases} ; \quad (2)$$

$$F(t) = \eta\left[\psi(t)\right]^n . \quad (3)$$

Here $m = m_2 + m_3$ – respectively the sum of mass of billets and elements of pusher device; m_1 – the mass of roll led to the point of impact; $c = c_x = c_y$ – stiffness of the roll along the axis of coordinates; $y(t), x(t)$ – dynamic deflection of the roll in the point of interaction with the billet; $\psi(t)$ – local crushing of the billet in the area of contact with rolls; $F(t)$ – the force of impact interaction of the pipe against the roll (impact impulse); α – nip angle of the billet by rolls; η and n - some constants, which depend on the mechanical properties of the material and the shape of hitting bodies (hollow billet and roll), which are determined experimentally [3], or on the base of strict theoretical reasoning, given in the work [3,4]. The solution of differential equations (1) and (2) we present in the Cauchy's setting of the problem in respect with defined initial problem specification

$x(0) = 0;\ \dot{x}(0) = 0;\ y(0) = 0\ ;\ \dot{y}(0) = 0\ ;\ \psi(0) = 0\ ;\ \dot{\psi}(0) = \upsilon_0^*$,

where υ_0^* – initial velocity of billet hitting with the rolls (equals the velocity of moving of billet pusher).

Using Duhamel's generalized integral [3, 4, 7] the expression for determination of dynamic deflection of the roll, local crushing of the billet we will represent through the function $F(t)$ as follows

$$y(t) = \frac{1}{\sqrt{cm_1}} \int_0^t F(\tau) \cos(\frac{\alpha}{2}) \sin[\sqrt{c/m_1}(t-\tau)] d\tau \ ; \qquad (4)$$

$$x(t) = \frac{1}{\sqrt{cm_1}} \int_0^t F(\tau) \sin(\frac{\alpha}{2}) \sin[\sqrt{c/m_1}(t-\tau)] d\tau \ ; \qquad (5)$$

$$\psi(t) = \upsilon_0^* t - \int_0^t F(\tau)[\frac{t-\tau}{m} \frac{1}{\sqrt{cm_1}} \sin[\sqrt{c/m_1}(t-\tau)]] d\tau \qquad (6)$$

By substitution (5) and (6) into (3) we will obtain the integral equation regarding impact interaction of the hollow billet with rolls of automatic mill

$$F(t) = \eta \upsilon_0^* \left\{ t - \int_0^t F(\tau)[\frac{t-\tau}{m} + \frac{1}{\sqrt{cm_1}} \sin[\sqrt{c/m_1}(t-\tau)]] d\tau \right\}^q . \qquad (7)$$

For the convenience of task solution let us apply to the equation (7) method of stepwise approximations, having chosen the solution of corresponding basic task, which was stated in the work [2] as initial approximation

$$F(t) = \frac{\upsilon_0^*}{c_1 \gamma_1 \gamma_2 (\gamma_2^2 - \gamma_1^2)} \left\{ [\frac{\gamma_1}{mc_1}(1+\frac{1}{\beta}) - \gamma_1^3] \sin(\gamma_2 t) - [\frac{\gamma_2}{mc_1}(1+\frac{1}{\beta}) - \gamma_2^3] \sin(\gamma_1 t) \right\}, \qquad (8)$$

where $\gamma_{1,2} = \sqrt{\frac{A}{2} \pm \sqrt{\frac{A^2}{4} - B}}$; $A = \frac{1+\beta+k^2 mc_1}{mc_1\beta}$; $B = \frac{k^2}{mc_1\beta}$; $\beta = \frac{m_1}{m}$: $k^2 = \frac{c}{m}$, (9)

where c_1 – coefficient of proportionality in the linear dependence from the function of impact force interaction of the billet against the roll ($\psi = c_1 F$).

From [4,5] it is known that suggested process of creation of stepwise approximations at rather tight step of iteration converges with accurate solution. The estimation of results converge in the set limits of tolerances of task solution lies within the limits +/–10 %.

Let us proceed to the experimental determination of constants of impact impulse η and n, mainly depending on the mechanical properties of the material and shape of the hitting bodies (hollow billet and roll).

Analysis of literary sources [1-3] according to the determination of metal pressure on the rolls of automatic mill and theoretical researches of technological loads showed that these parameters for rolling process of pipes depend mainly on the billet diameter, degree of its deformation and the velocity of billet rolling. It is characteristic that in result of billet hitting against the roll, the rolling force increases in a short moment significantly and further it reduces till nominal values and changes comparatively by non-significant range.

During rolling of heavy wall pipes with relatively large mass (500-800 kg), dynamic components of the rolling force of the billet appear to be dominant.

The results of interpretation of oscillogram [6], processed by means of mathematical statistics are the most probable averaged values within two passes of the value by metal pressure on the rolls and the moment on the spindle of the main gear drive are presented in the table 1.

Table 1 The results of experimental researches of the driveline of automatic mill PRP 140

No	Slug diameter, mm	Steel grade	Size of hollow billet mm × mm	Rolling pressure, kN	Moment on the upper spindle kN m
1	130	Х13Н10Т	Ø 138 × 13.0	730/854	79
2	100	ШХ15	Ø 103 × 12.5	560/790	68
3	110	40Х	Ø 114 × 11.0	590/810	57
4	140	30ХГСА	Ø 148 × 13.5	760/875	74
5	110	Ст.20	Ø 114 × 11.0	550/780	54

Measurements of forces were fulfilled during billet rolling in accordance with the hardest and developed manufacturing routes basing on the pipe rolling table on the automatic mill PRP 140. By using of oscillograms we may experimentally find the corresponding values of constants of impact impulse η and n. For example, during rolling of the billet Ø 114 × 11,0 from the steel St. 20, the cumulative pressure of metal on the rolls is 1330 kN and by means of leveling of peak values of metal pressure on the rolls in respect with crushing of billet entry section we find that η =1,32 и n =3,45.

Oscillogram of experimental research and its processing give definite representations about the character of formation of technological load. It is seen that they are of nonsteady dynamic nature, especially during billet biting by the rolls of automatic mill (fig.3).

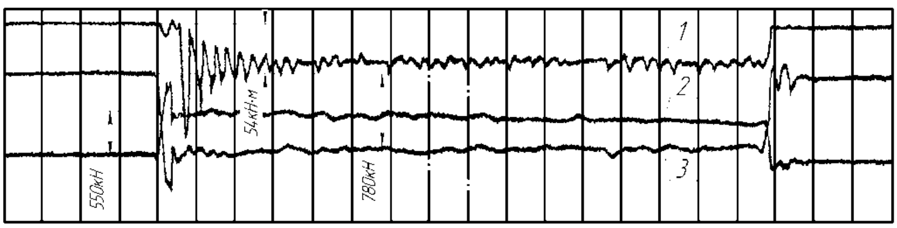


Figure 3. Oscillogram of energy power parameters of main drive gear of automatic mill PRP 140 (hollow billet Ø 114 × 11.0; steel 20, time interval – 0.1 s): 1 – moment on the bottom spindle; 2, 3 – rolling pressure.

Herein dynamic factor (correlation of maximum moment to the averaged value of moment during steady-state process of rolling) comprises 3.2 – 3.7.The time of action of the first three peaks of dynamic load makes 4 – 5 % from the machine time. Maximum value of the rolling pressure, fixed on the oscillograms, is equal to 1400kN, moment on the spindle of the working roll - 240 kN m.

Experimental investigation showed that forces and moments in the first and second passes hardly differ in their values. This is explained by the fact that if in the first pass of the billet through the rolls the deformation is higher, then in the second one increase of deformation stress happens because of noticeable reduction of metal temperature.

Further after determination of corresponding coefficients, we will proceed to the analytical determination of the value of impact impulse.

Substituting the expression (8) into the summand of integral equation (7) we will obtain approximate representation for the required function of impact impulse of the billet against the roll as follows

$$F(t)=\frac{\eta(\upsilon_0^*)^n}{[c_1(\gamma_2^2-\gamma_1^2)]^n}\{c_1(\gamma_2^2-\gamma_1^2)t+[\frac{1}{mc_1}(1+\frac{1}{\beta})-\gamma_1^2]$$
$$[(\frac{1}{m\gamma_2^3}+\frac{1}{m_1\gamma_2(\gamma_2^2-\frac{c}{m_1})})\sin(\gamma_2 t)-\frac{1}{\sqrt{cm_1}(\gamma_2^2-\frac{c}{m_1})}\sin(\sqrt{c/m_1}t)-\frac{t}{m\gamma_2^2}]-$$
$$-[\frac{1}{mc_1}(1+\frac{1}{\beta})-\gamma_2^2][(\frac{1}{m\gamma_1^3}+\frac{1}{m_1\gamma_1(\gamma_1^2-\frac{c}{m_1})})\sin(\gamma_1 t)-$$
$$-\frac{1}{\sqrt{cm_1}(\gamma_1^2-\frac{c}{m_1})}\sin(\sqrt{c/m_1}t)-\frac{t}{m\gamma_1^2}]\}. \quad (10)$$

Consequently, the time of achievement of maximum impact impulse of the billet against the roll of automatic mill t_m is determined from the equation (10) when $F(t)=0$.

It is obvious that in this case billet impact force against the roll of the mill is determined from the (10) and at $t=t_m$ equals $F(t_m)=F_m$.

We will mark that substitution of the function of impact impulse (10) into the formulas (5) and (6) gives corresponding dependences for dynamic bending of the roll and local crushing of contact billet.

However, due to some difficulties of mathematical character (phase quadrature) in the (5) and (6), functions $x(t)$, $y(t)$ and $\psi(t)$ are to be determined from the system of equations (1) and (3).

Considering that $\psi(t)=\sqrt[n]{F(t)/\eta}$ for small angles of billet biting ($x(t)=0$) we will get the expression for determining of dynamic bend of working roll during hit from the side of rolling billet

$$y(t)=\frac{1}{c}(1+\beta)F(t)-\frac{m_1\upsilon_0}{cc_1(\gamma_2^2-\gamma_1^2)}$$
$$\{[\frac{1}{mc_1}(1+\frac{1}{\beta})-\gamma_1^2][(\frac{1}{m\gamma_2^2}+\frac{\gamma_2}{m_1(\gamma_2^2-\frac{c}{m_1})})\sin(\gamma_2 t)-$$
$$-\frac{\sqrt{c}}{\sqrt{m_1^3}(\gamma_2^2-\frac{c}{m_1})}\sin(\frac{c}{m_1}t)-[(\frac{1}{m\gamma_1^2}+\frac{\gamma_1}{m_1(\gamma_1^2-\frac{c}{m_1})})\sin(\gamma_1 t)-$$
$$-\frac{\sqrt{c}}{\sqrt{m_1^3}(\gamma_1^2-\frac{c}{m_1})}\sin(\frac{c}{m_1}t)\}.$$
(11)

The character of changing of technological loads in the zone of interaction of the billet with rolls depending on the mass and velocity of its feed by the pusher in the deformation zone is conditioned by nonsteady history of rolling. In this regard to reveal strength reserves of the rolls and elements of main drive gear line of automatic mill, it is necessary to know maximum values and the character of changing of billet rolling force.

One may judge about the adequacy of the obtained results to the real process of impact interaction of the billet with roll of automatic mill PRP, comparing for example, maximum contact force, which is determined by the equation (11) with F_m value obtained by the approximation formula in accordance with [2, 3].

$$F_m=\upsilon_0^*\sqrt{mb^*\sigma_c[tg(\alpha)+ctg(\alpha)(1-k)}, \quad (12)$$

where b^* –given billet “width” in the zone of contact with a roll; σ_c – averaged value of yield point of the pipe material; α – angle of biting of the pipe by rolls; k – coefficient of reduction of the velocity during hitting of the pipe against the roll (may be determined experimentally).

The last expression is obtained on the base of force equality of plastic crushing and billet kinetic energy. This expression gives the upper bound of maximum force of impact interaction of the billet with automatic mill roll.

As an example let us consider the hit of the billet (hollow billet after piercing mill) with mass m =580 kg against the working roll of automatic mill PRP 140, herein the roll mass led to point of impact is equal to m_1 =6500 kg and the velocity of forced fed of the billet on the rolling axis makes υ_0^* =2 m/s.

The calculations show that maximum contact force of the billet against the roll, under the methodology, which was suggested, was 1180 kN, and uder the simplified one – 1235 kN. Fractional accuracy of calculations is about 12%, which suggests rather good reproducibility.

We may mark that the developed mathematical model (1)-(3) and fulfilled analysis may be useful during investigation of similar complex dynamic systems of longitudinal rolling mills.

Conclusions

The results of research of impact interaction of the hollow billet with rolls of automatic pipe-rolling plant are presented.

Refined mathematical model is developed and main parameters of dynamic interaction of working rolls of automatic mill with rolled billet are revealed.

Expressions for corresponding form of impact impulse and dynamic composite of metal pressure on working rolls of automatic mill are obtained.

The values and character of changes of impact loads in the area of billet interaction with rolls depending on the mass and velocity of its fed into the grooves of deformation area are determined.

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