

Modeling of Dynamic Processes of the Main Executive Mechanism of the Hot-crank Press

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Abstract

The paper considers a dynamic model and its mathematical description of the central crank-slide mechanism of the hot-crank press. The peculiarity of the proposed model is ability to assess on its basis, the level of oscillatory processes at all levels of the mechanism: the crankshaft, connecting rod and the slider with calculation of the clearances in the joints of the crank-slider-crank and eccentric.

Keywords: Hot-crank press, clearance, functions of the position, dynamic model.

تمثيل الخطوات الديناميكية لآلية التنفيذ الرئيس لماكنة كبس المعدن على الحار

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الخلاصة

يتناول البحث النموذج الديناميكي والوصف الرياضي لماكنة الكبس على الحار نوع (الكرنك المركزي الانزلاقي). خصوصية النموذج المقترح هو القدرة على تقييم على اساسه مستوى العمليات التبدئية على جميع وحدات الماكنة: في العمود المرفقي، ذراع التوصيل والمنزلق مع حساب التوافقات في المفاصل ذراع التوصيل – المنزلق و ذراع التوصيل – عمود الكرنك.

الكلمات المفتاحية: ماكنة الكبس على الحار ذو نوع عمود المرفقي ، التوافقات ، دالة الموقع ، النموذج الديناميكي.

Introduction

Dynamic processes, their level and character are one of the main factors determining the efficiency, reliability, productivity and, ultimately, the quality of any mechanical system. The ability to estimate the parameters of dynamic processes in the mechanism, and incorporate them into its design, is the condition of creating a truly high quality, competitive products [1, 2]. The methods of calculating dynamic processes in the mechanism are well known [3, 4].

The aim of this paper is to create a dynamic model of the central crank- slide mechanism (the main actuator) of the hot-crank press (Fig.1). The proposed dynamic model simulated by means of CAD system. Results show the ability of creation and modernization of existing technological pressing by new schemes.

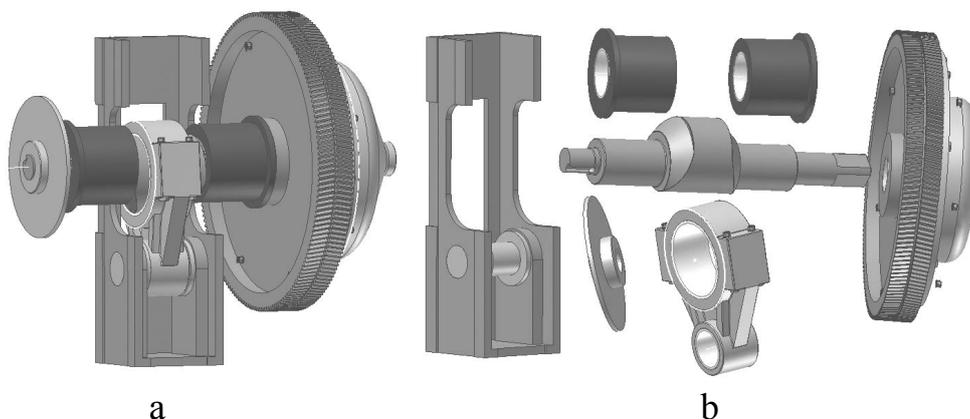


Fig.1. The main executive mechanism of the hot-crank press:
a – assembly, b - parts (knots)

Simulation and Modeling

The dynamic model of the slider-crank mechanism is shown in Fig.2. In its development, the following provisions are made:

- The dynamic model of the mechanism is based on its solid model, made in one of the 3D-modeling systems, containing a stress analysis module, for example, Inventor Professional.
- The mechanism model consists of three monolithic parts: the crankshaft, the connecting rod, and slider.
- Moments of inertia (J) and rigidities (c) of the model's elements are calculated on the basis of their 3D - models in any known solid modeling system. In our case it is the Inventor Professional.
- The coefficients of dissipation (ψ) are determined experimentally [3].
- The coefficients of damping (b) are determined in accordance with the results given in [4].
- The coefficients of friction (k_{tr}) are determined in accordance with the data given in [5].

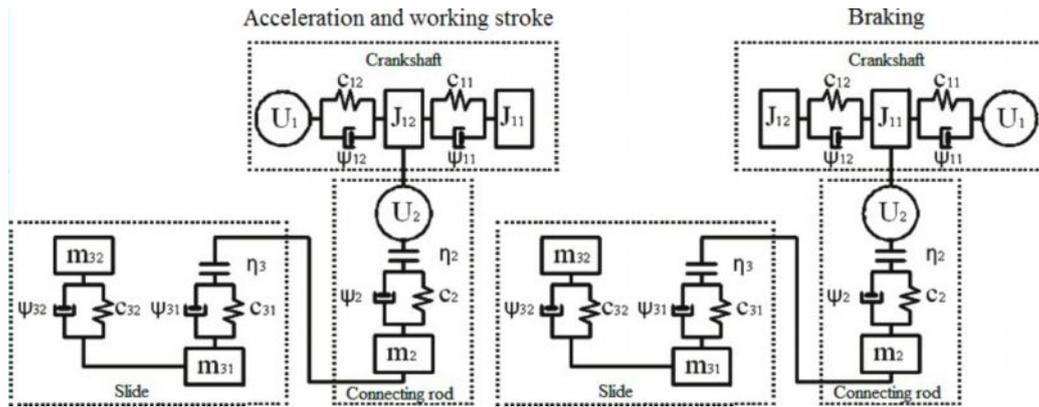


Fig.2. The dynamic model of the main executive mechanism of the hot-crank press

The crankshaft (Fig.3) performs rotational movement. The ideal movement (if we do not take into account elastic deformations and clearances) of its right side with the disk of the clutch is determined by the function U_1 .

The external loads are the moment of resistance of the connecting rod to rotation and the moments of friction in the bearings. During the operation the shaft experiences elastic deformations. A dynamic model of the shaft is a minimum of two masses connected by elastic dissipative bonds (Fig.2).

The connecting rod (Fig.4) performs in-plane parallel movement. The model takes into account only the translational component of its displacement. Accordingly, it is assumed that the connecting rod works only in tension - compression. The rigidity of the connecting rod compression is the rigidity of the connecting rod itself, the rigidity of the connecting rod tension is the rigidity of the mounting bolts of the cover.

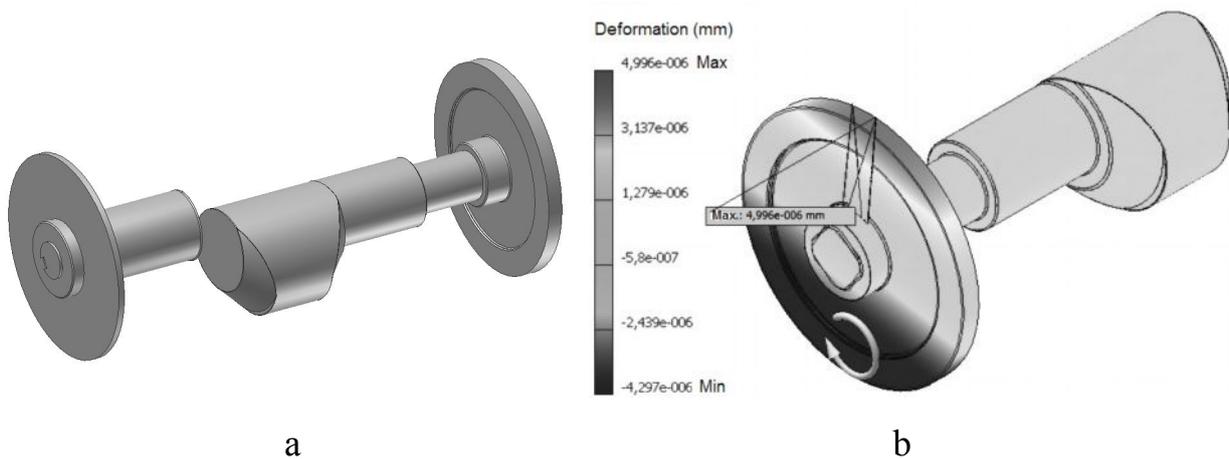


Fig.3. The crankshaft of the hot-crank press:
a - fragments of partitioning in dynamic modeling;
b - calculation of the rigidity of the 1st mass of 1st element of the model (C_{11})

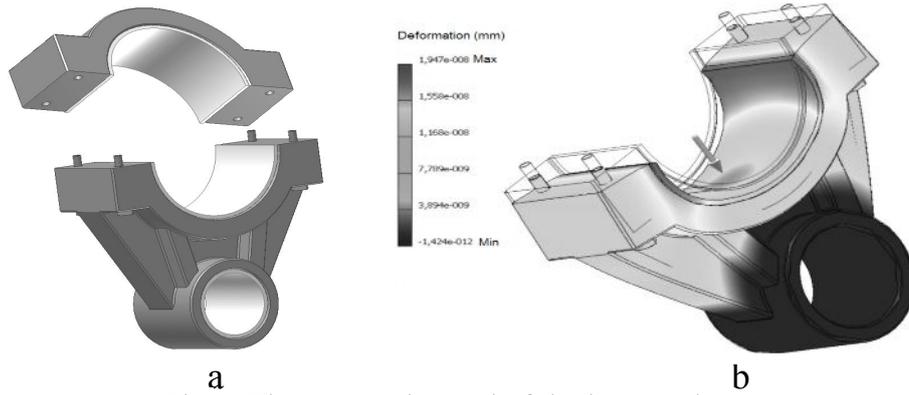


Fig.4. The connecting rod of the hot-crank press:
 a – fragments of partitioning with dynamic modeling;
 b - calculation of rigidity of 2-element model (c₂)

The linear displacement of the connecting rod U_2 relative to the angle of rotation of the crankshaft of the press x_{11} , velocity \dot{U}_2 , acceleration \ddot{U}_2 and the first transfer function U_2' are defined by the following dependencies: [6]

$$U_2 = R \cdot (1 - \cos(x_{11})) + L - \sqrt{L^2 - 2R \cdot \sin(x_{11})}; \quad U_2' = R \cdot \sin(x_{11}) + \frac{R^2 \cdot \sin(2x_{11})}{\sqrt{L^2 - 2R \cdot \sin(x_{11})}}; \quad \dots(1)$$

$$U_2'' = R \cdot \cos(x_{11}) + \frac{R^2 \cdot \cos(2x_{11})}{\sqrt{L^2 - 2R \cdot \sin(x_{11})}} + \frac{R^4 \cdot \sin(2x_{11})}{4 \cdot (L^2 - 2R \cdot \sin(x_{11}))^3}, \quad \dot{U}_2 = U_2' \dot{U}_1; \quad \ddot{U}_2 = U_2'' \dot{U}_1^2 + U_2' \ddot{U}_1,$$

Where:

- R - the radius of the crankshaft of the mechanism,
- L - the length of the connecting rod of the mechanism.

The movement of **the slider** (Fig.5) is back and forth. Dynamic processes occurring in the slider are modeled by a two-mass oscillatory system.

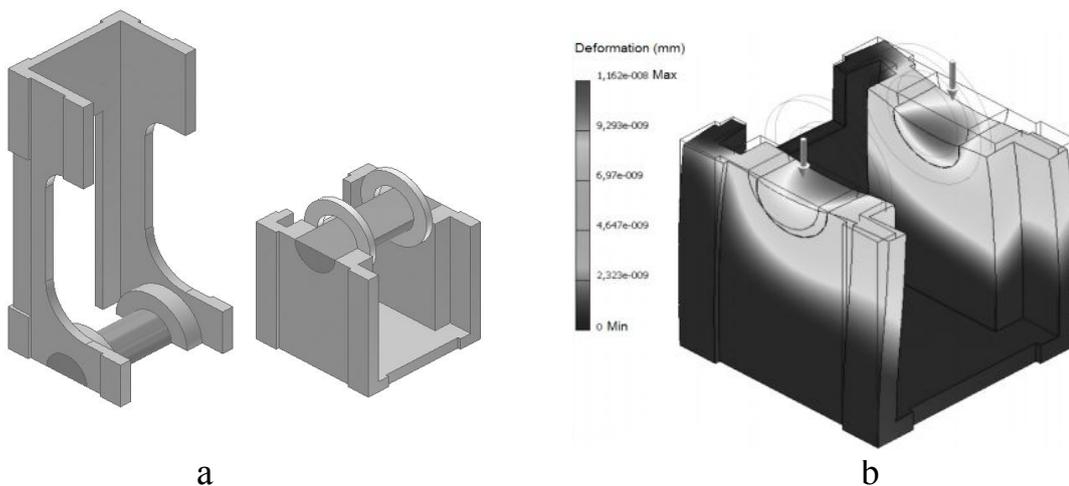


Fig.5. The slide of the hot-crank press:
 a - fragments of partitioning in dynamic modeling;
 b - calculation of the rigidity of the 1st mass of the 3rd element of the model (c₃₁)

The model parameters necessary for calculations are given in Table 1.

Table (1) the geometric and material properties of the model

Crank shaft (two - mass model)	
The moment of inertia to the eccentric J_{11}	592 kg.m ²
Rigidity to the eccentric c_{11}	$0.232 \cdot 10^{15}$ N.m/rad
The coefficient of dissipation ψ_{11}	0.4
The coefficient of friction k_{tr11}	0.03
The radius of crank shaft bearing (the first from the gear) ρ_{011}	0.28m
Moment of inertia after the eccentric (with a disc brake) J_{12}	1000kg.m ²
Rigidity after eccentric c_{12}	$0.3 \cdot 10^{15}$ N.m / rad
The coefficient of dissipation ψ_{12}	0.4
The coefficient of friction k_{tr12}	0.03
The radius of crank shaft bearing (the second from gear) ρ_{012}	0.28m
Connecting rod(single – mass model)	
Mass of connecting rod m_2	4881.392 kg
Rigidity of connecting rod in compression c_{2c}	$0.127 \cdot 10^{12}$ N.m
Rigidity of connecting rod in tension c_{2p}	$0.638 \cdot 10^{12}$ N.m
The coefficient of dissipation ψ_2	0.4
Code contact load k_2	0.2
Clearance of one-sided Crank- connecting rod θ_2	0.000073m
The slider (two – mass model)	
Mass of slide and shank m_{31}	4657.729 kg
Rigidity of slider in compression c_{31c}	$0.9 \cdot 10^{10}$ N.m
Rigidity of slider in tension c_{31p}	$0.47 \cdot 10^{10}$ N.m
The coefficient of dissipation ψ_{31}	0.4
Code contact load k_3	0.2
Clearance of one-sided connecting rod-slider θ_3	0.00005m
Mass of slider (Shank) m_{32}	3349.087 kg
Rigidity of slider (Shank) c_{32}	$0.734 \cdot 10^{11}$ N.m
The coefficient of dissipation ψ_{32}	0.4
The coefficient of friction k_{tr32}	0.05
The body (only elastic deformation in the area of stamping)	
Rigidity of body C_{cm}	$1.0 \cdot 10^{14}$ N.m
The coefficient of dissipation ψ_{cm}	0.4

Clearance is possible in connections (component parts of the mechanism are: the crankshaft – the connecting rod - the slider) (Fig.6). The contact surface of the links in their interaction within the clearance varies from zero to a maximum value, and the rigidity of the “link – clearance” system is changed respectively. The first step is the displacement of lubricant, then the compression of the remaining oil film, the contact deformation of the mating surfaces and, finally, the deformation of links. In Fig.6a this area is shaded. Naturally, all these processes occur simultaneously. But the determining role at this stage is played by any one of the given sequence. It is a problem to determine theoretically the law of rigidity change in the range from η to θ . In our model, we assume it to be linear (Fig.6b).

Where:

- θ - a clearance of one-sided view of the contact zone of deformation,
- η - one-sided clearance,
- U, x – the ideal and the real displacement of the link,
- c - link rigidity

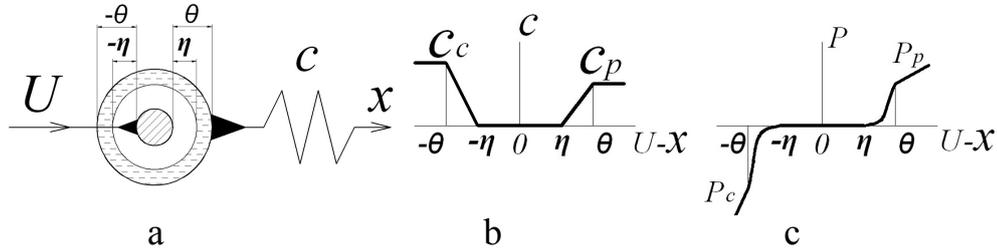


Fig.6. The scheme of the clearance (a), rigidity (b) and load (c) in the swivel:

Assuming $\theta > 0, |\theta - \eta| > 0, \eta = \theta \cdot (1-k)$, we have the following calculation dependencies to determine the rigidity and load:

The clearance is open. Elastic deformation is absent. Link rigidity $c=0$.

$$\begin{aligned}
 & -\eta \leq x - U \leq +\eta, \\
 & c = 0, \quad P = 0. \quad \dots\dots\dots(2)
 \end{aligned}$$

The clearance is selected compression. Link rigidity $c=c_c$.

$$\begin{aligned}
 & x - U < -\eta, \\
 & a = \frac{c_c}{\eta - \theta}, \quad b = \frac{c_c \eta}{\eta - \theta}, \quad c = a(x - U) + b. \quad \text{If } c \geq c_c \text{ or } k \leq 0.001, c = c_c \dots\dots\dots(3) \\
 & P = c(x - U + \eta).
 \end{aligned}$$

The clearance is selected tension. Link rigidity $c=c_p$.

$$\begin{aligned}
 & x - U > +\eta, \\
 & a = -\frac{c_p}{\eta - \theta}, \quad b = \frac{c_p \eta}{\eta - \theta}, \quad c = a(x - U) + b. \quad \text{If } c \geq c_p \text{ or } k \leq 0.001, c = c_p \dots\dots\dots(4) \\
 & P = c(x - U - \eta).
 \end{aligned}$$

Here θ is determined by the fit of the connection. The value of $\eta = \theta \cdot (1 - k)$, where $0.001 \leq k \leq 1$. When $k = 0.001$, it can be assumed that the rigidity changes abruptly from zero to a maximum value. In the connection there is a shock interaction of the links. When $k = 1$, free reconnection of spare parts in the connection is absent, which is typical for the fit with the preciseness of a clearance corresponding to the 7th, 6th and lower quality classes.

The movement of the disk of the clutch, rigidly connected to the crankshaft, is determined by the mode of the press operation. When operating in the automatic mode, the angular velocity of the press is constant and equal to ω . Accordingly, the function of $U_1 = \omega$

* t . If the press operates on a single stroke, then in the acceleration zone $t = 0 \dots t_p$, the angular velocity of the disk of the clutch varies from 0 to ω and further remains constant. Here and further,

t - the time of the mechanism operation,

t_p - the time of the acceleration of the crankshaft of the press after the clutch is turned on,

t_m - the time of the start of braking.

The angle of rotation of the crankshaft at the time t_p is φ_p and ranges from 3 to 15 degrees [7]. Assuming that the ideal velocity of the crankshaft varies according to the square law, we have the following dependences of the provisional zone from 0 to t_m :

$$t_p = \frac{2\varphi_p}{\omega}; \quad A = -\frac{3\omega^4}{4\varphi_p^3}; \quad B = \frac{3\omega^3}{2\varphi_p^2};$$

$$\text{If } 0 \leq t \leq t_p: \quad U_1 = \frac{1}{12}At^4 + \frac{1}{6}Bt^3; \quad \dot{U}_1 = \frac{1}{3}At^3 + \frac{1}{2}Bt^2; \quad \ddot{U}_1 = At^2 + Bt.$$

$$\text{If } t > t_p < t_m: \quad U_1 = \omega t - (\omega t_p - \varphi_p); \quad \dot{U}_1 = \omega; \quad \ddot{U}_1 = 0.$$

The equations describing the displacement of the crankshaft in the braking zone are not given here.

The links of the mechanism and, accordingly, the elements of its model, in the operation of the mechanism have the following external influence (Fig.6):

- Forces and moments of friction forces. The expressions (forms) for their calculations are given in [4].
- The effort of the counter balancer of the slider [7].
- Technological load. Its value can be obtained experimentally or analytically[8].
- The forces of the elastic deformation of the bed, acting on the zone of the discharge of the mechanism in its back stroke during the press.

In the system of the elements of the model the following internal forces act:

- Forces of dissipation [4].
- The forces of elasticity of the bonds of the model [4]. Their value can be defined as the product of the rigidity of the connection and its deformation (dependencies 2 - 4).

The effectiveness of the mechanism operation depends on the accuracy of performing the kinematic characteristics by its links and loads in them. The data for quality assessment can be presented in graphs (diagrams) and in a number of quality criteria (reliability, efficiency, durability). The most well-known of these criteria are maximum, average and mean square deviations of the kinematic characteristics of links from the ideal; maximum, average and mean square loads in the links of mechanisms, as well as a number of others [4, 9].

The mathematical model of the slider-crank mechanism is a system of five nonlinear differential equations of the second order:

Before the braking zone	In the braking zone
$\ddot{x}_{11} = -\frac{(P_{c11} + P_{\delta 11}) - (P_{c12} + P_{\delta 12})}{J_{11}} - \frac{(P_{c2} + P_{\delta 2})U'_2 + P_{mp11}}{J_{11}}$	$\ddot{x}_{11} = -\frac{(P_{c11} + P_{\delta 11}) + P_{mp11}}{J_{11}}$
$\ddot{x}_{12} = -\frac{(P_{c12} + P_{\delta 12}) + P_{mp12}}{J_{12}}$	$\ddot{x}_{12} = -\frac{(P_{c12} + P_{\delta 12}) - (P_{c11} + P_{\delta 11})}{J_{12}} - \frac{(P_{c2} + P_{\delta 2})U'_2 + P_{mp12}}{J_{12}}$
$\ddot{x}_2 = -\frac{(P_{c2} + P_{\delta 2}) - (P_{c31} + P_{\delta 31})U'_3 + P_{mp3}}{m_2}$	$\ddot{x}_2 = -\frac{(P_{c2} + P_{\delta 2}) - (P_{c31} + P_{\delta 31})U'_3 + P_{mp3}}{m_2} \dots\dots\dots(5)$
$\ddot{x}_{31} = -\frac{(P_{c31} + P_{\delta 31}) - (P_{c32} + P_{\delta 32})}{m_{31}} - \frac{P_{mp11} + P_{cm} + P_{bcm} + P_{ur} + P_{mex}}{m_{31}}$	$\ddot{x}_{31} = -\frac{(P_{c31} + P_{\delta 31}) - (P_{c32} + P_{\delta 32})}{m_{31}} - \frac{P_{mp11} + P_{cm} + P_{bcm} + P_{ur} + P_{mex}}{m_{31}}$
$\ddot{x}_{32} = -\frac{(P_{c32} + P_{\delta 32}) + P_{mp32}}{m_{32}}$	$\ddot{x}_{32} = -\frac{(P_{c32} + P_{\delta 32}) + P_{mp32}}{m_{32}}$

In equations (5) the symbols denote:

- U - the function of the position of the links of the mechanism (see dependence 1),
- U' - transfer functions,
- P_c - the forces of elasticity in the links, taking into account the clearances in their connections (see dependencies 2 - 4),
- P_b - the forces of dissipation [5],
- P_{mp} - the forces or moments of friction,
- P_{cm} and P_{bcm} - the forces of elasticity and dissipation arising in the deformation of the bed,
- P_{mex} and P_{ur} - the technological effort on the slider and the effort of its counter balancers.

Results and discussion

Fig.7 shows the graphs of dynamic load in the crankshaft press at pressing standard flange forgings in the single stroke mode (Fig.7a) and the diagram of reconnection of clearances in the “crank shaft – connecting rod” connection (Fig.7b). The latter is a normalized ordinate diagram of changing the connection rigidity from 0 to +1 (tension) and from 0 to -1 (compression). The values of the criteria of the mechanism’s performance quality calculated according to the proposed method are given in Table 2.

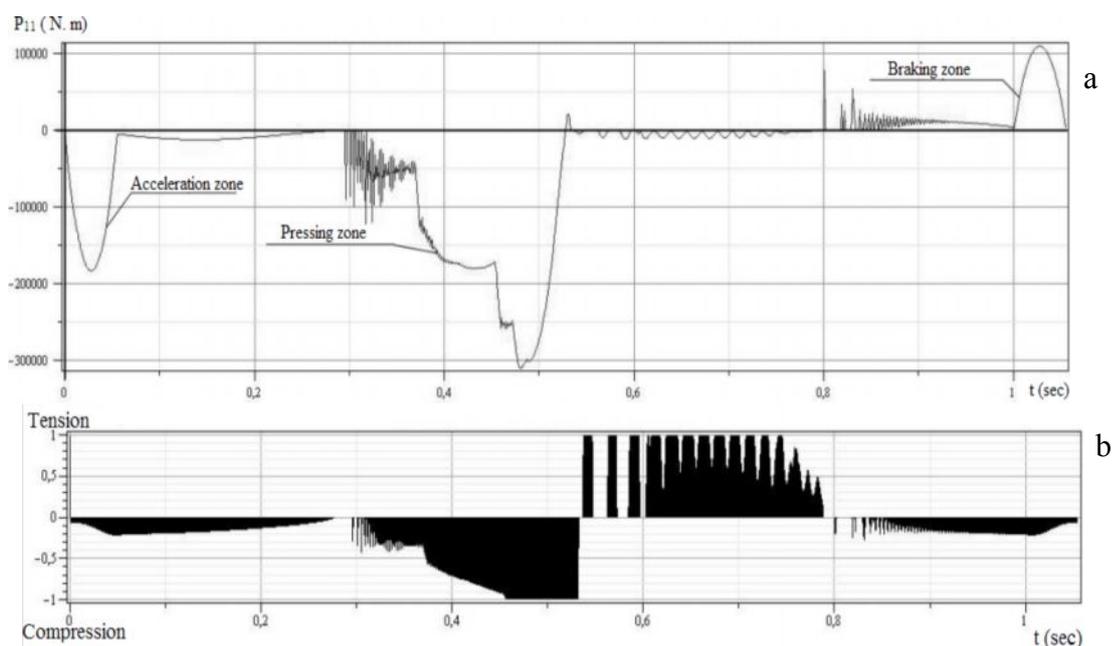


Fig.7. The results of investigating the dynamics of the hot-crank press: a - dynamic load in the crankshaft press, b – diagram of the reconnection of clearances in the “crank shaft – connecting rod” connection

Table (2) the values of the quality criteria of the mechanism.

Quality Criteria	Values		
	Real mechanism	Ideal mechanism	% discrepancy
Crankshaft			
Max. velocity (m / sec)	6.279	6.266	0.20
Max. acceleration (m/s ²)	169.635	169.635	0.00
Max. load (N)	$-3.095 \cdot 10^5$	$-3.166 \cdot 10^5$	2.29
Mean square load (N)	$1.028 \cdot 10^5$	$8.300 \cdot 10^4$	19.26
Mean load (N)	$5.553 \cdot 10^4$	$3.503 \cdot 10^4$	36.91
Connecting rod			
Max. velocity (m / sec)	1.115	1.104	0.98
Max. acceleration (m/s ²)	137.190	9.220	93.27
Max. load (N)	$-1.297 \cdot 10^7$	$1.301 \cdot 10^7$	0.31
Mean square load (N)	$2.602 \cdot 10^6$	$2.561 \cdot 10^6$	1.57
Mean load (N)	$8.534 \cdot 10^5$	$7.943 \cdot 10^5$	6.925
slider			
Max. velocity (m / sec)	1.115	1.102	1.166
Max. acceleration (m/s ²)	-811.509	-5.809	99.28
Max. load (N)	$-1.298 \cdot 10^7$	$-1.3015 \cdot 10^7$	0.269
Mean square load (N)	$2.608 \cdot 10^6$	$2.561 \cdot 10^6$	1.802
Mean load (N)	$8.439 \cdot 10^5$	$7.943 \cdot 10^5$	5.877

Conclusions:

1. The proposed dynamic model of a slider-crank mechanism and its software implementation can be used not only in calculating the indicators of reliability and serviceability of a specific mechanism, but also for a more accurate calculation of the reserve strength of its component parts, features of counterbalances, for optimization of performance clutches, brakes and other parameters.
2. The procedure of combining methods of the dynamic analysis of mechanisms and their solid modeling can significantly reduce the complexity of dynamic models and improve their accuracy.
3. The study of the dynamics of the hot-crank press on the basis of the proposed model allows estimating the influence of parameters of technological process of pressing on the level of dynamic processes in the press. The results of estimations are the modernization of the existing technological schemes pressing and the creation of new ones.

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