№ 1 (132) / 2026
ISSN: 2520-6168
Vol. 132, № 1 / 2026
Техніка, енергетика, транспорт АПК
Technology, energy, agriculture transport AIC
UDC 621.787.4
DOI: 10.37128/2520-6168-2026-1-11

INCREASING THE WEAR RESISTANCE OF PARTS OF SOIL TILLING MACHINES BY THE METHOD OF HYDRO-IMPULSE SURFACE PLASTIC DEFORMATION

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Increasing the operational reliability of agricultural machinery components operating under conditions of intensive abrasive wear is a priority task of modern mechanical engineering. Surface plastic deformation using hydropulse devices is an effective method of forming a hardened layer, however, the stability of this process is significantly limited by the degradation of the geometric shape of the working tool.

The article analyzes the dynamics of the interaction of impact tips of various configurations (sphere, cone, trapezoid) with the treated surface. The evolution of the tool shape change under conditions of high-frequency cyclic loading and its influence on the energy parameters of the impact are considered. The study is based on the method of mathematical modeling of viscoelastic systems based on the Kelvin-Voigt model, which allows describing the hydropulse drive as a parametric system with a built-in pressure pulse generator.

In the course of the work, mathematical dependencies were determined for estimating the force of impact interaction, which took into account the elastic-plastic response of the part and the involutely small deformation of the tip. The kinematic characteristics of the piston-striker were calculated, in particular the maximum speed at the moment of contact and the rebound parameters. The use of the "elastic-concentrated model" of the energy carrier was justified to minimize the dimensions of the device while maintaining a high pulse frequency (up to 100 Hz). It was established that the use of the refined mathematical model allows adapting the processing modes to increase the wear resistance of agricultural machinery parts by 1.5–2 times.

The proposed approach, combining a refined mathematical model with adaptive control of the energy and geometric parameters of hydropulse loading, ensures the predictable formation of a hardened surface layer with enhanced wear resistance and stable properties over a wide range of operating conditions.

Keywords: *hydropulse device, surface plastic deformation, impactor, plastic deformation, mathematical modeling, pressure pulse generator, impact kinetic energy, forging, wear resistance, Kelvin-Voigt model.*

Eq. 20. Fig. 10. Ref. 15.

1. Problem formulation

The modern development of agricultural and general mechanical engineering is characterized by a constant increase in the requirements for the operational reliability of components and assemblies. The operating conditions of soil tillage equipment, in particular, constant abrasive action and significant dynamic loads, require the introduction of high-tech methods for increasing the wear resistance and fatigue strength of parts. One of the most promising and cost-effective areas in this area is surface plastic deformation (SPD) technology. Among SPD methods, a special place is occupied by the use of hydropulse devices, which allow generating significant

Article received 12.01.2026. Article accepted 12.02.2026. Article published 17.04.2026.

pulse loads [1,2]. This ensures the formation of a high-quality hardened surface layer (hardening) with specified physical and mechanical properties, which significantly extends the service life of machine parts.

However, the practical implementation of the potential of hydropulse hardening faces a number of technical difficulties. The efficiency of the process and the stability of the results obtained critically depend on the dynamic stability of the hydropulse drive and the preservation of the functional characteristics of the working tool (hammer). In real operating conditions, the striker is subjected to intense high-frequency cyclic loads. Such an impact leads to the gradual accumulation of microdamage and plastic deformation of the striker material, which inevitably leads to a change in its initial geometric shape [1,2].

A significant problem arises: the discrepancy between the actual geometry of the striker and the calculated parameters entails a chain reaction of negative consequences. In particular, the stability of the transfer of kinetic energy of the impact is disrupted, the area of contact interaction changes, which leads to unpredictable variations in the depth of the hardened layer and the heterogeneity of hardening over the entire machined surface. As a result, the overall quality of hardening decreases, which negates the advantages of the technology.

Thus, in modern mechanical engineering there is a clear scientific contradiction: on the one hand, it is necessary to provide high energy of the impact pulse for effective deformation of the surface of the part, and on the other hand, under the influence of this same energy, the tool loses its geometric stability and resource [2-4]. Resolving this contradiction requires research into the dynamics of the interaction of the striker with the machined surface, studying the processes of tool shape change, and developing mathematical models of the hydropulse drive.

2. Analysis of recent research and publications

The issue of increasing the durability of machine parts by surface hardening has been the focus of attention of many scientists for a long time. A significant number of fundamental works have been devoted to the study of the processes of pulse loading and deformation hardening of surfaces. In particular, the foundations of the dynamics of hydraulic impact systems are laid in the works of such scientists as Obertyukh R.R., Slabky A.V., Iskovich-Lototsky R.D. and others. The issue of contact interaction of bodies during high-speed collision was considered in detail in the classical Hertz theory, as well as in modern studies [2,3,4], where the main attention is focused on the energy balance between the kinetic energy of the impactor and the potential energy of deformation of the workpiece. It is generally accepted that the effectiveness of the hardening process directly depends on the kinetic energy of the impact and the current state of the working tool. In the works [3,4] it is substantiated that the striker of the hydraulic impulse device is a critically important element, since it directly interacts with the surface and transfers the energy of the hydraulic shock to it. However, despite significant successes in modeling hydraulic impulse systems, the issue of the evolution of the shape of the tool itself during long-term operation remains insufficiently studied.

Its durability depends on the ability of the material to resist plastic deformation, which, with repeated loading cycles, leads to irreversible changes in the geometry of the striker [3,4]. In particular, ball strikers are characterized by a gradual flattening of the contact surface, which transforms the initial point interaction into a planar one, and for conical ones, blunting or flattening of the apex. Such shape changes critically affect the stability of the hardfacing distribution in the metal and the quality of the hardened layer.

In addition, special attention is required to study the relationship between the frequency of pressure pulses and the dynamic response of the system, which is implemented in modern devices with built-in parametric pressure pulse generators (PGP). Unlike traditional vibration drives, such systems provide greater flexibility of technological settings, but require a refined mathematical description based on complex viscoelastic models, in particular the Kelvin-Voigt model [2-5]. Thus, the need to develop a mathematical apparatus that would take into account both the dynamics of the drive and the gradual deformation of the working tool makes this study relevant.

In the design of modern hydropulse systems, the impactor performs the function of a key link, providing direct kinetic interaction with the treated surface and implementing the directional transfer of hydroshock energy. The performance indicators and resource durability of this element are determined by its ability to withstand extreme dynamic loads that arise at the moment of high-speed collision of the tool with the part [2, 4].

Under conditions of cyclic loading-unloading with high frequency, the material of the working part of the striker is subjected to intense mechanical influences, which initiate the processes of plastic deformation. This leads to irreversible degradation of geometric parameters and changes in the physical and mechanical properties of the surface layer of the tool [3-5]. Understanding the mechanisms of such shape changes is a fundamental basis for optimizing the design parameters of working tips, rational selection of wear-resistant materials and development of accurate algorithms for predicting their service life [4]. Manifestations of plastic deformation of strikers are usually identified as local dents, residual curvatures of the working faces and radial flattening (especially for spherical and conical shapes), which tend to accumulate progressively over time [3,4]. The intensity of this process is a multifactorial function that depends on the magnitude of the generated impact energy, the rheological properties of the impactor material (in particular, the combination of hardness and impact toughness), the resistance to deformation of the processed material, as well as the initial geometric configuration of the tool contact zone [4-6]. To analyze the processes of wear and loss of geometric accuracy, we will consider the characteristic features of shape changes for the most common types of working tips.

Spherical (ball) striker

The spherical impactor (Fig. 1) due to its ideal geometric symmetry provides a uniform distribution of internal stresses in the material while observing the conditions of the central impact. In the initial phase of interaction, contact occurs at a point that allows achieving high values of instantaneous pressure necessary to overcome the yield point of the material of the processed part and the formation of the hardening zone [4-6].

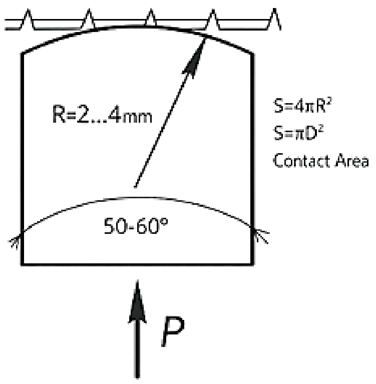


Fig. 1. Ball striker

However, in real operating modes of a hydraulic pulse drive, ideal conditions are often violated due to off-center impacts, dynamic rod beating or microstructural heterogeneity of the tool material itself. This leads to the occurrence of tangential stresses, which cause the formation of local defects: microdents, chips and cuts on the working surface.

Systematic high-frequency impact action initiates the processes of plastic flow of metal in the contact zone, which inevitably leads to gradual flattening (flattening) of the spherical surface. This process of geometry transformation is critical, since it changes the nature of the interaction from the initially point to planar.

Such a change in the contact area significantly reduces the specific pressure at a constant impact energy, which negatively affects the depth of strain hardening and the uniformity of the distribution of residual stresses in the surface layer of the part [4,5].

Conical striker

The conical impactor (Fig. 2) is characterized by a high ability to concentrate the energy of the shock pulse on the minimum contact area.

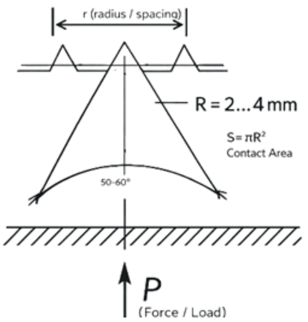


Fig. 2. Conical striker

Due to the sharp geometry of the tip, an extremely high specific pressure is created in the contact zone, which allows you to effectively overcome the forces of interatomic bonding in the surface layers of the processed material. This provides a high penetrating ability of the tool, which is critically important for the processes of perforation, local destruction or deep marking of hard materials [4, 5].

However, a significant gradient concentration of contact stresses in the narrow near-apex zone of the cone makes it most vulnerable to degradation processes. Under the action of cyclic dynamic loads, the stresses in the tip material often exceed its local yield strength, which initiates intense plastic deformation.

During long-term operation, the shape of the tip evolves: it gradually becomes blunt, undergoes radial flattening or microcracking due to the accumulation of fatigue microcracks. These structural changes lead to an increase in the actual contact area, which, at a constant pulse energy, significantly reduces the specific pressure and, as a result, sharply reduces the penetration efficiency and the quality of strain hardening of the surface [5].*Rounded top striker (rounded cone)*

The striker with a rounded top (Fig. 3) in its geometric configuration is a rational constructive compromise between spherical and conical tips. This shape allows you to combine the advantages of both types: it provides a higher concentration of the energy of the impact pulse compared to the sphere, but at the same time significantly reduces the gradient of peak contact stresses inherent in a sharp cone. Due to the radius of curvature, it is possible to achieve optimal distribution of forces in the contact zone, which contributes to the formation of a uniform layer of hardening with a lower probability of microfractures of the tool [4,5].

The use of a rounded cone allows stabilizing the strain hardening process due to a smooth change in the contact area during the tool's penetration into the part material. This ensures the effective occurrence of dislocation hardening processes in the near-surface layers [5].

However, under conditions of prolonged cyclic loading, the material of the rounded part is subject to fatigue and plastic redistribution. Plastic deformation in this case manifests itself in the form of a gradual flattening of the contact zone and degradation of the spherical segment of the tip. Such a shape change leads to a gradual deviation from the calculated impact dynamics: an increase in contact time and a decrease in peak pressure values. As a result, a decrease in the technological efficiency of processing is observed, which requires adjusting the operating parameters of the hydraulic pulse drive or replacing the working tip [4,5].

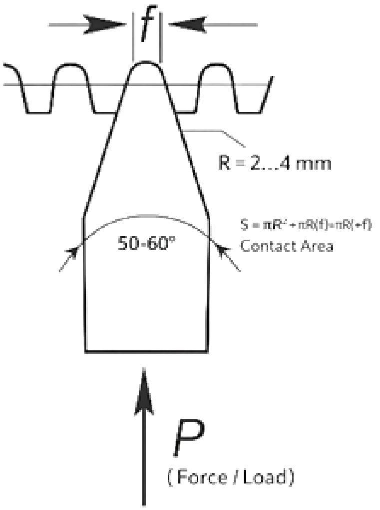


Fig. 3. A striker with a rounded top

Trapezoidal striker

The trapezoidal impactor (Fig. 4) is specially designed for technological operations that require the formation of extended deformation zones, the creation of wide technological grooves or the processing of significant surface areas in one working cycle. Due to the presence of a flat or slightly inclined working face, such a tool provides the transfer of hydraulic impact energy to a significantly larger contact area compared to other types of tips. This allows achieving high productivity when performing deformation strengthening operations on large flat surfaces [4-6].

A feature of the functioning of the trapezoidal shape is the specific distribution of contact stresses: their greatest concentration is observed not in the center, but along the perimeter of the working area (edge effect). This nature of the load requires high accuracy of positioning the impactor relative to the part and stability of the dynamic characteristics of the hydraulic drive.

During intensive operation, due to the significant interaction area and cyclic heating of the contact zone, the striker undergoes degradation changes. Plastic deformation manifests itself in the form of general flattening (settling) of the working face, loss of flatness and curvature of the side surfaces.

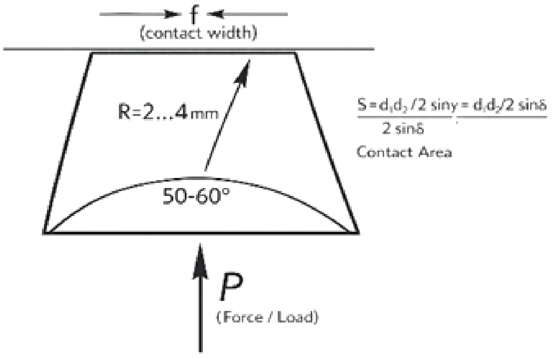


Fig. 4. Trapezoidal-shaped striker

A characteristic defect is the formation of characteristic notches along the edges of the working area, which is due to the edge concentration of stresses at the contact boundaries [5]. Such structural and geometric changes lead to an uneven distribution of hardening over the width of the treated area, which negatively affects the operational characteristics of the hardened surface layer.

A comprehensive study of the kinetics of shape change of strikers taking into account the accumulation of plastic deformations is a critically important stage in the design and modernization of hydraulic impulse systems. Analysis of the mechanisms of degradation of working surfaces allows not only to predict the residual resource of the tool, but also to scientifically substantiate the choice of materials with increased operational stability. In particular, the use of superhard alloys, materials with a high fracture toughness index and the application of special functional coatings that minimize adhesive wear and fatigue cracking are promising [5-8].

The physical and mechanical characteristics of the surface layer, in particular hardness, directly correlate with the duration of trouble-free operation of machine components and depend on the strength gradient and surface resistance to abrasive wear. For parts of soil tillage equipment operating in harsh environmental conditions, hardness testing is the main indicator of their potential durability and ability to resist plastic deformation [5, 6].

It has been established that under the dynamic influence of a spherical impactor of a hydropulse device, the nature of strain hardening (hardening) changes significantly depending on the pulse energy and rheological properties of the metal. The process of impact introduction of the tool initiates the redistribution of dislocations and grinding of the metal structure, which is reflected in the change in the microhardness profile with depth (Fig. 5). Understanding these patterns allows us to develop new design solutions that ensure the stability of the energy parameters of the impact action and high wear resistance of both the workpiece and the working body.

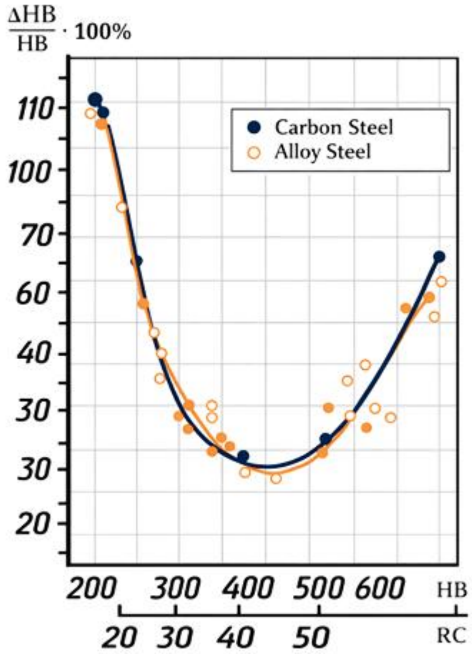


Fig. 5. Degree of maximum hardening for steels 50 and 65G

The transformation of the geometry of the working surface of the striker due to its plastic deformation directly correlates with the nature of the formation of the stress-strain state in the workpiece. The evolution of the shape of the contact zone leads to a significant redistribution of stresses, which, in turn, changes the kinetics of dislocation accumulation and the structure of the hardening zone (Fig. 6, 7). In particular, the deviation of the striker shape from the calculated one causes instability of the tool penetration depth and variability of the microhardness gradient along the depth of the surface layer [6-8].

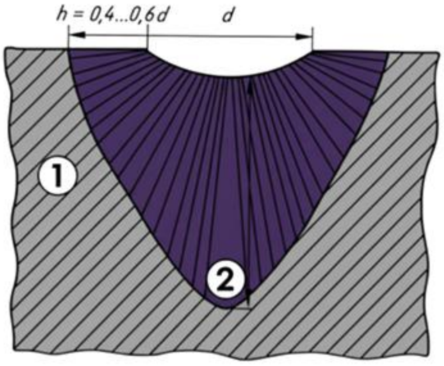


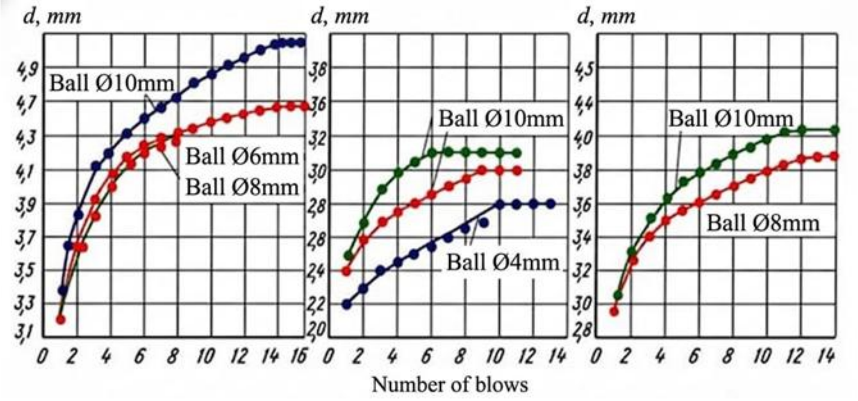
Fig. 6. Distribution of hardfacing zones and their depth with a ball-shaped striker:
1 – part 2 – deformation zone

It has been established that with an increase in the radius of curvature of the striker (for example, when the sphere is flattened), the area of elastic-plastic contact expands, which reduces the peak values of the contact pressure at constant kinetic energy. This leads to a decrease in the maximum depth of penetration of the striker and the dissipation of impact energy over a larger volume of metal, which can cause an insufficient level of hardening in the target hardening zones [6-8].

The study of the nature of the distribution of the hardened zone under the surface of the imprint (Fig. 7) confirms that to ensure the uniformity of the hardened layer, it is critically important to maintain the stability of the geometric parameters of the striker. Any deviations in the shape lead to a distortion of the isolines of the hardening distribution, which can provoke the appearance of zones with residual tensile stresses, which are potential foci of fatigue failure of the part during further operation [6]. Thus, the control of the shape change of the striker is a necessary condition for stabilizing the depth and quality of deformation hardening in high-performance hydraulic pulse technologies.

The assessment of the effectiveness of the impact effect in hydraulic pulse systems is based on the analysis of the kinetic energy generated by the working element (striker) immediately before the act of collision, as well as on the study of the dynamics of its transformation into the deformation energy of the workpiece. The quantitative indicator of this energy is determined by the mass of the moving parts of the drive and the square of their speed at the moment of contact, which is a determining factor for the formation of the depth and intensity of the hardened layer [4, 8].

An important component of the analysis is the high-speed deformation mode, which characterizes the intensity of the course of plastic processes in the metal. The high loading rate inherent in the hydropulse method initiates specific mechanisms of dislocation rearrangement of the crystal lattice, which allows achieving significant strengthening with minimal residual changes in the macrogeometry of the part [6,7]. In this case, the strain rate affects the material's resistance to plastic flow, which must be taken into account when adjusting the frequency characteristics of the pressure pulse generator.



a ***b*** ***c***

Fig. 7. The nature of the distribution of the riveted zone under the surface of the imprint:
a) steel 10; b) steel 50; c) steel 65G

In the process of interaction, the impact energy is distributed between the tool and the processed body, being spent on elastic recovery, plastic deformation and heat dissipation. Optimization of this energy balance by choosing rational modes of energy supply and parameters of the hydraulic link allows to ensure maximum efficiency of the deformation hardening process [7]. Thus, a complex combination of kinetic parameters of the impactor and rheological response of the material of the part is the basis for creating stable technological processes of surface-plastic deformation.

The generation of the required impact energy in devices for surface-plastic deformation is based on the use of various physical principles, the choice of which depends on the required frequency and amplitude of the pulses. The main methods are: the use of hydrodynamic fluid pressure, which allows for accurate dosing of energy through parametric GIT; the use of centrifugal forces arising from the rotation of unbalanced masses; and the use of vibrational oscillations of the working medium, which are transformed into shock pulses through the deforming tool [7, 8]. In addition, the energy efficiency of the process correlates with the design features of the impactor: its geometric configuration and the physical and mechanical properties of the material, in particular the hardness, which determines the ability to accumulate and transfer momentum without significant losses due to elastic dissipation.

At the microstructural level, the process of deformation strengthening of metals is based on the dynamics of the accumulation of dislocations, which act as the main carriers of plastic flow. Under the influence of the impact pulse, a series of dislocation loops are generated from the sources, which move along the slip planes of the crystal lattice. In the event of the first dislocation encountering an obstacle (grain boundaries, inclusions or other dislocations), the following loops, approaching the braking zone, cause progressive local distortion of the lattice [8].

The interaction of closely located dislocation areas creates fields of internal stresses that spread to adjacent microvolumes of the crystal. This leads to an intensification of the hardening process and a significant expansion of the volume of the deformed lattice around such clusters. As a result, a fine-grained structure with a high density of defects is formed, which provides a significant increase in the hardness and wear resistance of the hardened layer of the part [7-9].

3. The purpose of the article

The purpose of the study: to justify the choice of the optimal shape of the striker and calculate the operating parameters of the hydropulse device for stable surface hardening of parts.

To achieve the goal, the following tasks need to be solved:

- To analyze the influence of the geometric shape of the striker (sphere, cone, trapezoid) on the nature of the stress distribution and the accumulation of plastic deformations in the instrument itself.
- Describe using mathematical equations the process of movement of the striker inside the hydraulic impulse drive, taking into account the fluid pressure and spring stiffness.
- Determine a mathematical relationship that will allow you to accurately calculate the force of impact interaction between the tip and the part, taking into account the elastic properties of materials.
- Taking into account the information received, draw conclusions.

4. Results and discussion

The use of hydropulse drives for surface strain hardening processes with an operating frequency range of up to 100 Hz demonstrates significant advantages over traditional mechanical vibration drives. The main design and technological preferences of such systems are their high energy consumption with relatively compact dimensions, as well as the possibility of precise flexible adjustment of the amplitude-frequency parameters of the vibration loading of the impact element [8, 9]. Such adaptability is a critical condition for identifying and implementing optimal hardening modes that provide the required depth of hardening and surface microgeometry without the risk of overhardening or thermal destruction of the material.

According to the results of the analysis of existing technical solutions, the design schemes of hydropulse equipment for strain hardening are classified according to the principle of pulse source placement into two main types:

1. Systems with remote GIT, which allows servicing several actuators from one pumping station.
2. Devices with integrated GIT directly into the actuator, which minimizes hydraulic losses and increases system performance.

Design and engineering implementation of highly efficient hydraulic impulse devices requires a comprehensive study of their dynamic characteristics. The fundamental stage of development is the synthesis of adequate mathematical and dynamic models that allow describing transient processes in hydraulic and mechanical subsystems [8, 9]. The construction of a dynamic model of a device with a built-in parametric GIT is based on the analysis of the structural and calculation scheme (Fig. 8), which takes into account the masses of moving parts, the stiffness of elastic elements, viscous friction and nonlinear characteristics of the compressibility of the working fluid [9]. Such a comprehensive approach, combining theoretical modeling with subsequent experimental verification, is the key to creating reliable equipment for PPD.

To minimize the overall dimensions of the equipment, the most appropriate is to use structures with a GIT integrated into the actuator. Such a solution is especially effective when strengthening small-sized parts (Fig. 9). The design of a device with limited dimensions is based on the need to provide a given hardening energy at certain speed regimes, impact frequency and working fluid flow. Since the impulse energy is a determining criterion when selecting power components, it is recommended to combine the elastic elements of the GIT and the impact link to optimize the overall dimensions of the system [8, 10].

The key component that performs the power and distribution functions (Fig. 8) is the piston-hammer *(4)* with a tip *(4.2)* fixed to the rod *(4.1)*. It acts as a locking element of a single-stage GIT, which operates on the parametric principle of pulse generation. The sealing of the working cavities is carried out using chamfered and spool elements. In particular, the piston chamfer comes into linear contact with the seat *(5)*, which is adjacent to the inner surface of the sleeve in diameter [8, 9, 11].

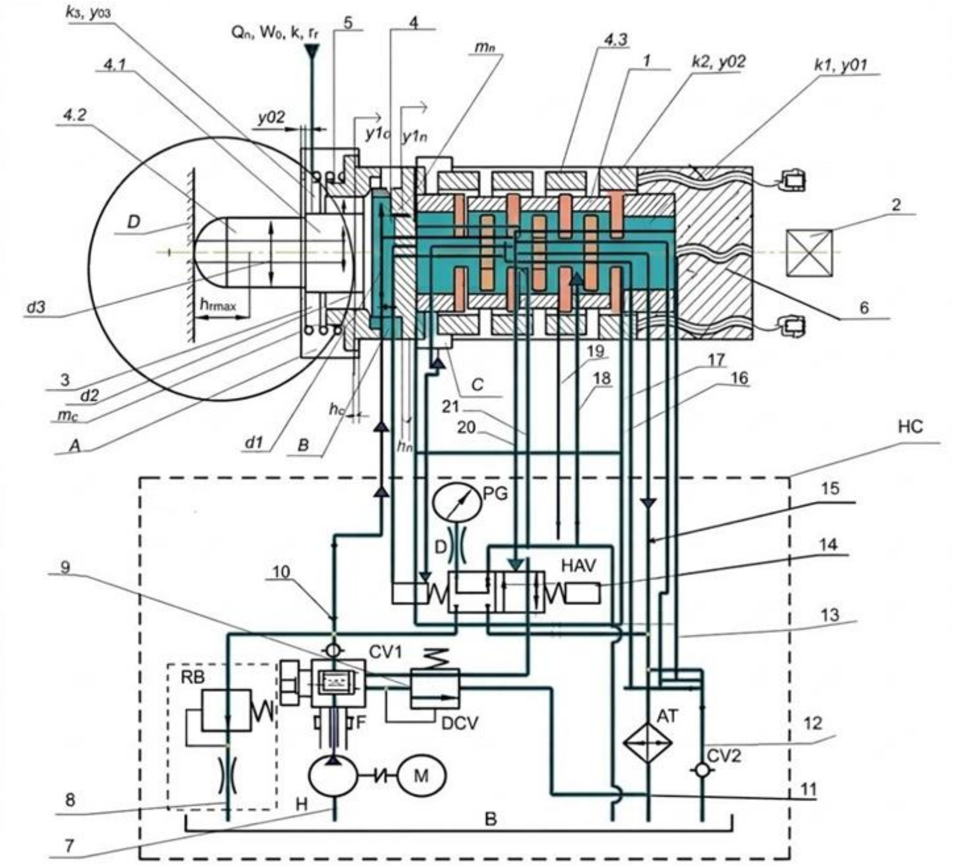


Fig. 8. Schematic hydro kinematic diagram of a prototype of a hydropulse device for deformation hardening of parts: 1 – spring; 2 – adjusting screw; 3 – spring; 4 – striker; 4.1 – rod; 4.2 – impact tip; 4.3 – elastic part of the piston; 5 – seat; 6 – cup; 7-21 – hydraulic lines; d_1, d_2 – diameter; d_3 – rod diameter; y_{02} –preliminary deformation; $h_д$ – additional overlap; $Q_н$, W_0, k, ρ_r – energy supply; k_1, y_{01} –spring stiffness and pre-deformation 1; k_2, y_{02} –k_2, y_{02} – stiffness and pre-deformation of the elastic part 4.3 of the piston-hammer 4; k_3, y_{03} – spring stiffness and pre-deformation 3; m_n i m_c –respectively, the combined masses of the piston-hammer; h_{hmax} – $h_{h_{max}}$ –maximum stroke of the piston - striker 1; y_{1c}, y_{1n} –the beginning (direction) of the forward stroke, respectively, of the seat 5 and the piston-hammer 4; A – pressure cavity; B – tank; C – intermediate cavity; D – workpiece; C – drain cavity;F – filter insert; CV1, CV2 – check valve; AT – radiator; DCV – safety valve; RB – flow regulator; H – hydraulic pump; HC – pumping station.

In the operating state of the system, the elastic part of the piston *(4.3)*, made in the form of a slotted spring, and the additional spring *(1)* installed in parallel have a preliminary deformation D_0.This ensures that the piston surface is located with a diameter d_2 at a distance of positive overlap x_0 relative to the drain channels. The initiation threshold (opening pressure) of the GIT is adjusted using a screw *(2)* in the cup *(6)*, which changes the compression of the spring *(1)*, as well as due to the elastic properties of the striker itself [8, 11].

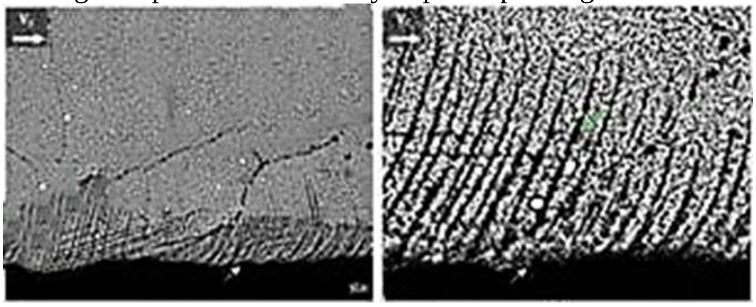
The structural and calculation diagram (Fig. 8) shows the main calculation parameters:

- Q_H –energy consumption from the pumping unit;
- V_A –total volume of the pressure main and cavity A;
- E andp_A – modulus of elasticity and current fluid pressure;
- C_1, D_1 and C_{43}, D_{43} –stiffness and deformation of the corresponding spring elements;
- $m_{уд}$ and $m_{сід}$ –combined masses of moving parts (firing and saddle);
- h_{max} – maximum piston stroke.

Fig. 9. General view of the hydraulic impulse drive striker

The machining process is accompanied by intense force interaction directly at the interface "tool - part". The most significant microstructural transformations are localized in subsurface zones at small depths relative to the newly formed surface. Under typical deformation conditions, there is a simultaneous action of thermal and mechanical factors, the gradient of which determines the final state of the material [10, 11]. The nature of the microstructure is significantly determined by the intensity of the external influence. With moderate machining, characterized by balanced thermal and force loads, the thickness of the hardened layer (the so-called resistance layer) is 11–15 µm (Fig. 10). Such hardening parameters are recorded when using tools made of polycrystalline cubic boron nitride and cemented carbides [9, 12]. It is characteristic that under moderate deformation the properties of the formed layer remain stable and similar, regardless of the specific combinations of thermomechanical factors inherent in different types of tool materials [10, 11].

In contrast, aggressive deformation modes cause much deeper structural disturbances. In this case, the depth of hardening reaches 24–28 µm, which is accompanied by the formation of specific “white layers” in the subsurface. The emergence of such structures indicates extreme gradients of temperatures and plastic flow rates, which leads to grain refinement to the nanoscale level and phase transformations [11, 12]. Understanding these mechanisms is key to ensuring the operational reliability of parts operating under conditions of increased wear.



a ***b***

Fig. 10. Microstructural integrity of the surface as a result of normal plastic deformation conditions; a – material resistance caused by intermediate thermal effects and low-intensity mechanical effects; b – details of deformation traces caused by low-intensity thermal effects and intermediate mechanical effects on the workpiece surface

The complete dynamic picture of each stage of the device's operation is described by a system of differential equations. These equations allow simulating the movement of the mechanical drive links and the balance of energy consumption in the hydraulic cavities at the same time, which is the basis for optimizing the parameters of the hydraulic drive [10, 11].

To optimize the analytical description and increase the efficiency of the numerical solution of the system of equations, the complex eight-stage operating cycle of the device is classified according to functional features into two main periods [10, 12]:

1. The period of direct travel (stages 1–5): characterized by active movement of inertial masses m_c and m_n under the action of increasing pressure of the energy carrier. The kinematics of this period is described in the coordinate system x_{cn} and x_{nn} (where the index «n» indicates the straight direction of movement) [10, 12].
2. The return period (stages 6–8): corresponds to the return of the masses to their original position after the act of impact interaction is completed. The movement is recorded by the coordinates x_{cz} and x_{nz} (where «z» indicates the reverse stroke), the countdown of which begins from the far right position of the working links.

When designing high-frequency hydraulic pulse systems, special attention is paid to minimizing the volume of the pressure cavity. $V_{\sum A}$, which is a necessary condition for the generation of high-frequency pulses. In view of this, when constructing the mathematical model, an elastic-concentrated model of the energy carrier was used. This assumption allows us to neglect the inertial properties (mass) of the fluid in cavity A, focusing on its elastic characteristics [11,12].

The hydraulic drive link in this study is presented as a classical viscoelastic model. It consists of an inertialess elastic component C_{or} and dissipative element k_{or}. Within this rheological approach, the hydraulic subsystem of the device is identified as a Kelvin-Voigt body, which allows for an adequate description of the damping and energy accumulation processes in the working environment [11, 12].

Throughout the cycle, the hydraulic link undergoes deformation at a variable rate, entering into dynamic interaction with the reduced masses. m_c and m_n. The relationship between the hydraulic and mechanical parameters of the system is established through the corresponding gear ratios, which are determined by the geometric parameters of the working chambers and the cross-sectional area of the links [11].

$$i_{oj} = A_j^2 \cdot A_0^{-2} \tag{1}$$

where A_j – cross-sectional area of the moving link of the device; A_0 –the cross-sectional area of the averaged pressure hydraulic line of the device drive.

$$A_o = \sum_{k=1}^{n}(l_k \cdot A_k)/\sum_{k=1}^{n} l_k \tag{2}$$

where l_k, A_k –respectively, the length and cross-sectional area k of the hydrochannel.

The elastic component of the hydraulic link, which within the framework of the adopted Kelvin-Voigt model is considered as inertialess, is characterized by the combined stiffness relative to the design area A_0. This parameter is crucial for assessing the dynamic response of the energy carrier to the high-frequency disturbance generated by the HIT [10, 11].

Mathematically, the stiffness of the hydraulic elastic element is determined according to the following functional dependence:

$$k_{or} = A_0^2 \kappa W_0^{-1} \tag{3}$$

To ensure the universality of the mathematical model and the possibility of its adaptation to various design parameters of the device, the calculated stiffness of the hydraulic link can be transformed relative to any characteristic cross-section. Based on the fundamental principles of the theory of hydraulic impulse systems [11,12], the recalculation of the stiffness for the selected cross-sectional area of the drive elements is carried out using the following analytical relationship:

$$k_{oj} = i_{oj} k_{or} \tag{4}$$

where $j = 1,2,...,n$ – the sequence number of the drive element link.

Based on the energy carrier model presented in fundamental works [11,12], the viscous resistance force parameter is used to describe energy losses in the hydraulic system F_{fo}. This force arises in an inertialess dissipative element that models internal friction and hydraulic losses in the pressure line.

Mathematically, the viscous drag force is calculated using the following relationship:

$$F_{fo} = c_o \cdot \dot{x}_{or} \tag{5}$$

where $x_{or} = dx_{or} / dt$ –strain rate x_{or} hydraulic link (Kelvin-Voigt bodies); $c_o = \pi\mu d_o/4 \approx 0{,}785\mu d_o$ – coefficient of viscous resistance to deformation of the hydraulic link (here μ is the dynamic viscosity of the energy carrier; $d_o = 2\sqrt{(A_0/\pi)}$).

To ensure the adequacy of the dynamic model and take into account energy losses in the moving couplings of the device, it is necessary to reduce the dissipative characteristics to the main working links. According to the provisions of the theory of hydraulic impulse systems [11, 12, 13], if the parameters of the dissipative element of the hydraulic subsystem are reduced to the characteristic cross-sectional areas of the moving parts of the drive, the calculation of the dissipative resistance force is carried out using the following dependence:

$$F_{fj} = i_{oj}^{-0,5} \cdot c_o(\dot{x}_{Aj} \pm \dot{y}_j) \tag{6}$$

where $x_{Aj} = x_{or} \cdot i_{oj}^{-0.5}$ – deformation of the hydraulic link, reduced to the cross-sectional area $A_{ji} \cdot \dot{y}_j$ –speed of the i-th drive link.

To describe the fundamental relationship between the state parameters of the working fluid and its deformation characteristics within the framework of the developed dynamic model, it is advisable to use the following scientific statement:

In the mathematical model of the hydraulic impulse drive, the current value of the pressure pr of the energy carrier in the averaged pressure main A_0 directly correlates with deformation x_{or} rhydraulic link. The dependence is based on the principles of linear elasticity of the medium and is determined by the ratio between the potential energy of a compressed fluid and its geometric volume [10, 11].

Mathematically, this relationship is described by the following dependence:

$$p_r = x_{or} \cdot k_{or} \cdot A_0^{-1} \tag{7}$$

To clarify the calculation scheme and simplify the mathematical modeling of the hydraulic impulse drive, the method of assumptions is used, which is based on the analysis of the significance of the influence factors. Based on the results of research [12, 13], when developing dynamic models of the hydraulic impulse drive systems, it is customary to ignore both the excess pressure of the energy carrier and the stiffness of the hydraulic link in the drain line. This is due to the fact that the pressure in the drain cavity approaches atmospheric, and its effect on the dynamics of the striker compared to the high pressure in the pressure chamber is negligible and does not introduce a significant error into the energy calculations.

Analytical comparison of functional dependencies for work processes allows you to establish a mathematical relationship between the critical parameters of the system state. In particular, a comparison of mathematical descriptions *(1.8)* and *(1.9)* allows us to determine the correlation between the pressure levels of “opening” (initiation) and “closing” of the GIT and the corresponding deformation changes of the hydraulic link [12, 13].

This ratio is the basis for determining the pulse amplitude and the operating frequency of the valve mechanism. It allows you to calculate the deformation x_{or} at the moments of changing the cycle phases, which is necessary to stabilize the hardening energy when the working fluid flow rate varies. The use of these dependencies provides an adequate description of hysteresis phenomena in the operation of the HIT, which directly affects the accuracy of setting the processing modes of parts [11, 13].

$$x_{0\max} = p_1 A_0 \cdot k_{or}^{-1} \tag{8}$$

$$x_{0k} = p_2 A_0 \cdot k_{or}^{-1} = x_{0\max} \cdot A_1 \cdot A_2^{-1} = x_{0\max} i_{21}^{0,5} \tag{9}$$

where $i_{21} = A_1^2 \cdot A_2^{-2}$ – internal gear ratio in the GIT between the areas of the working sections of the piston-hammer.

The dynamic stability of the operation of the hydropulse device is determined by the complex interaction of hydraulic and mechanical forces acting on its main links. An important parameter of the model is the gear ratio between the hydraulic energy and the mechanical movement of the piston-hammer *(4)* (Fig. 8), which determines the efficiency of the transformation of pressure into kinetic energy [13].

During the movement in the precision couplings of the piston-hammer *(4)* and its rods *(4.1, 4.3)* with the guide surfaces of the sleeve, a total resistance force R arises, which in its nature corresponds to semi-dry or semi-liquid friction. Dissipative processes in the system are described through the coefficients of viscous friction: c_1 — for the locking element of the piston-firing device in the axial bore of the sleeve, and c_3 — for the guide surfaces of the saddle. Taking these factors into account is necessary for accurate calculation of the pre-deformation acceleration rate and rebound energy [13, 14].

The central power parameter is the force F_y, which is generated at the moment of high-speed collision of the tip *(4.2)* with the surface of the workpiece *(D).* The magnitude and law of change of force F_y depend not only on the kinematics of the drive, but also on the physical and mechanical properties of the materials of the contact pair, including the elastic moduli, yield strength, and local hardness [13].

To derive a scientifically based dependence of the impact force F_y It is necessary to analyze the energy balance of the system. According to the fundamental principles of impact mechanics [11,13], the kinetic energy accumulated by the piston-hammer at the end of the return phase is transformed into deformation work at the moment of contact. This energy is distributed into the elastic component $\delta_{пд}$ (potential energy of elastic compression) and the plastic component $\delta_{плд}$, which is directly responsible for the formation of the hardened layer and the change in the surface microstructure [12].

$$E_k = 0{,}5 m_П \cdot V_{П\max}^2 \tag{10}$$

where $V_{n\max}$ –maximum speed of the striker piston 1 at the end of its return stroke; $m_{\max}$ – the mass of the impactor piston [12].

In the mechanics of high-speed hardening impact, the nature of the energy distribution is determined by the ratio of the elastic and plastic properties of the contact pair. Since in rational modes of PPD the magnitude of the plastic deformation of the part $\delta_{пл\partial}$ is significantly smaller than its elastic deformation $\delta_{п\partial}$ $(\delta_{пл\partial} < \delta_{п\partial})$, It can be argued that the accumulated kinetic energy of the piston-hammer is almost completely transformed into the potential energy of elastic compression of the metal [10, 12].

At the same time, according to the law of conservation of energy and taking into account the rebound phenomenon, a certain fraction of energy is spent on changing the kinematic state of the system after the impact. This process is accompanied by the movement of the piston-striker *(4)* and the elastic deformation of its power part in the direction of the forward stroke, which is denoted by the coordinate $y_{П}$. This redistribution of energy is due to the action of reactive forces of elastic recovery of the part, which initiate the reverse movement of the tool [11, 14].

For the mathematical description of this balance, the equation of the energy state of the system at the moment of maximum deepening of the striker is used:

$$E_{\mathrm{k}} = E_{к\partial} + E_{кв} = 0{,}5k_{Д} \cdot \delta_{ПД}^{2} + E_{к} \cdot k_{в}^{2} \tag{11}$$

or

$$0{,}5k_{Д} \cdot \delta_{ПД}^{2} = E_{к}(1 - k_{в}^{2}) = 0{,}5m_{П} \cdot V_{П\max}^{2}(1 - k_{в}^{2}), \tag{12}$$

where k_{∂} – local (contact) stiffness of the part *D* being processed; $k_{в}$ –coefficient of recovery for a partially elastic impact of the tip *(4.2)* on the part *D*. For a steel-on-steel impact, the coefficient $k_{в} = 5/9 \approx 0{,}59$ ($V_{Пmax} \leq 3$m/s);

At the moment of completion of the convergence phase, the accumulated potential energy of elastic deformation is transformed into mechanical work performed by the impact force of the tip *(4.2)* at the site of introduction of the tool into the material. $\delta_{п\partial}$. The calculation of this work is based on the fundamental principles of classical mechanics. In order to simplify the mathematical model, the assumption of collinearity of the vectors of the applied impact force is made. F_{y} and the vector of the maximum velocity of the piston-hammer $V_{Пmax}$ immediately before contact [14, 15].

Mathematically, the work of the impact force can be represented as an integral characteristic of the loading process:

$$F_{y} \cdot \delta_{ПД} = 0{,}5m_{П} \cdot V_{П\max}^{2}(1 - k_{в}^{2}), \tag{13}$$

where

$$F_{y} = 0{,}5m_{П} \cdot V_{П\max}^{2}(1 - k_{в}^{2}) \cdot \delta^{-1}_{ПД}. \tag{14}$$

Determining the maximum speed of the piston-hammer *(4)* at the moment of completion of its return stroke is a critical stage of the calculation, since it is this indicator that determines the magnitude of the impulse transmitted to the workpiece. The calculation is based on the previously accepted assumption of a predominantly elastic nature of the deformation, which allows us to relate the speed $V_{Пmax}$ із energy characteristics of the elastic elements of the drive [10, 15].

Mathematical estimation of speed is carried out by analyzing the dynamic equilibrium of masses under the action of the pressure of the energy carrier and the reaction of the spring components. According to studies [11, 14], this approach allows to take into account the inertia of the moving links and hydraulic losses that occur when changing the direction of movement.

$$E_{k} \cdot k_{в}^{2} = 0{,}5 \cdot V^{2}_{П\max} \cdot m_{П} \cdot k_{в}^{2} = 0{,}5 \cdot k_{2} \cdot h_{пв}^{2}, \tag{15}$$

where

$$V_{П\max} = \frac{h_{пв}}{k_{в}} \cdot \sqrt{\frac{k_{2}}{m_{П\Sigma}}} = h_{пв} \cdot \omega_{02} \cdot k_{в}^{-1}, \tag{16}$$

where $h_{пв}$ – the rebound stroke of the piston-hammer *(1)* after its interaction with part *D*; □$_{02}$ – natural frequency of mass oscillations $m_{П}$.

Substituting (1.9) into (1.7), we find:

$$F_{y} = 0{,}5 \cdot \omega_{02}^{2}(k_{в}^{-2} - 1)h_{пв}^{2} \cdot \delta_{п\partial}^{-1}, \tag{17}$$

or taking into account $k_{в} = 0.56$

$$F_{y} = 1{,}094 \cdot m_{П} \cdot \omega_{02}^{2} \cdot h_{пв}^{2} \cdot \delta_{п\partial}^{-1}. \tag{18}$$

In the phase of maximum penetration of the tip into the material of the workpiece, a state of dynamic equilibrium occurs. Dynamic force F_{y}, generated as a result of the shock pulse, is completely balanced by the elastic reactive force of the metal resistance. The latter is due to internal stresses that arise in the crystal lattice of the part (D) as a result of its elastic deformation [13-15]. According to the provisions of the mechanics of continuous media, this reactive force is directly proportional to the magnitude of the compression and the contact stiffness of the interaction zone.

Mathematically, this state can be represented as:

$$F_y = k_Д \cdot \delta_{пд} \tag{19}$$

The fundamental dependence given in [15] can be obtained by analytical transformation and algebraic simplification of equations (1.11) and (1.12). When constructing a mathematical model of the interaction process of the tip *(4.2)* with the workpiece *(D)*, it is advisable to consider this act as a classical contact of two spherical bodies. Such an assumption is quite legitimate, for example, in the strengthening treatment of cylindrical surfaces with a spherical indenter. According to the provisions of the Hertz theory of contact deformations, under the action of impact load in the contact zone, a local area of elastic-plastic compression is formed. Within the framework of the adopted geometric model, this zone is identified with a spherical segment. The radius of the base of this segment a, which determines the actual area of instantaneous contact, is calculated according to the following dependence [15]:

$$r = 0{,}88\sqrt[3]{F_y\,\frac{E_y^{-1} + E_Д^{-1}}{R_y^{-1} + R_Д^{-1}}}, \tag{20}$$

where E_y, $E_Д$ – Moduli of elasticity of materials of the tip *(4.2)* and parts *Д*; R_y, $R_Д$–radii of the impact part of the tip *(4.2)* and the part.

In the process of dynamic loading, the zone of elastic disturbance covers a certain area of both the workpiece *(D)* and the working tip *(4.2)*. However, to simplify the calculation scheme and increase its efficiency, it is advisable to take into account the gradient of the physical and mechanical properties of materials. Since the deforming element (tip) is usually made of high-alloy tool steels (for example, ShKh-15) and undergoes special heat treatment to achieve maximum hardness, its resistance to deformation significantly exceeds the similar indicator of the part material.

Given the high stiffness of the tip material and the significant gradient of the yield point of the mating elements, the model takes into account only the elastic-plastic response of the part, while the deformation of the tool is neglected. Thus, all the work of the shock pulse is directed to deformation changes exclusively in the workpiece *(D)*. This assumption allows us to identify the tip with a completely rigid indenter, which simplifies the determination of the insertion depth. $\delta_{пд}$ and calculation of the stress-strain state in the subsurface layers of the metal being strengthened [11, 12, 15].

The use of the rigid indenter model is a standard method for studying the processes of PPD, since it provides sufficient accuracy for engineering calculations in determining the kinetic energy necessary for the initiation of dislocation strengthening [11, 15].

A comparative analysis of the obtained dependencies with the results of previous experimental tests confirmed the high adequacy of the proposed mathematical model. Taking into account the relationship between the characteristics of the energy carrier and the geometry of the contact zone allows adapting the design of the device to the strengthening of a wide range of parts of agricultural engineering, increasing their wear resistance by 1.5–2 times.

5. Conclusion

According to the results of the study, it was established that the geometric stability of the working tool is a determining factor in ensuring the quality of surface deformation hardening. Analysis of the shape change of the impactors showed that the accumulation of plastic deformations in the apical zones leads to an uncontrolled increase in the contact area, a decrease in the specific pressure and a violation of the uniformity of the hardening.

A complex mathematical model of the hydropulse device has been developed, which is based on the elastic-concentrated representation of the energy carrier and the dynamics of a two-mass vibro-impact system, allowing for precise prediction of the kinetic energy of the pulse. The obtained analytical dependencies for calculating the force of the impact interaction, which are based on Hertz's theory and take into account the elastic properties of the contact pair, have confirmed their adequacy in comparison with experimental data.

The proposed methodological approach to the design of hydropulse systems with integrated GIT provides the possibility of flexible adjustment of frequency modes (up to 100 Hz). This guarantees the stability of the technological parameters of the tillage equipment even with a partial change in the tip profile, which is a necessary condition for increasing the operational reliability and durability of soil tillage equipment components under abrasive wear conditions.

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ПІДВИЩЕННЯ ЗНОСОСТІЙКОСТІ ДЕТАЛЕЙ ҐРУНТООБРОБНИХ МАШИН МЕТОДОМ ГІДРОІМПУЛЬСНОГО ПОВЕРХНЕВОГО ПЛАСТИЧНОГО ДЕФОРМУВАННЯ

Підвищення експлуатаційної надійності вузлів сільськогосподарської техніки, що працюють в умовах інтенсивного абразивного зношування, є пріоритетним завданням сучасного машинобудування. Поверхневе пластичне деформування з використанням гідроімпульсних пристроїв є ефективним методом формування зміцненого шару, проте стабільність цього процесу суттєво обмежена деградацією геометричної форми робочого інструмента.

У статті проаналізовано динаміку взаємодії ударних наконечників різної конфігурації (сфера, конус, трапеція) з оброблюваною поверхнею. Розглянуто еволюцію формозміни інструмента в умовах високочастотного циклічного навантаження та її вплив на енергетичні параметри удару. В основу дослідження покладено метод математичного моделювання в'язко-пружних систем на базі моделі Кельвіна-Фойгта, що дозволяє описувати гідроімпульсний привід як параметричну систему з вбудованим генератором імпульсів тиску.

В ході роботи визначено математичні залежності для оцінки сили ударної взаємодії, де враховано пружно-пластичний відгук деталі та нехтовно малу деформацію наконечника. Розраховано кінематичні характеристики поршня-ударника, зокрема максимальну швидкість у момент контакту та параметри відскоку. Обґрунтовано використання «пружно-зосередженої моделі» енергоносія для мінімізації габаритів пристрою при збереженні високої частоти імпульсів (до 100 Гц).

Встановлено, що використання уточненої математичної моделі дозволяє адаптувати режими обробки для підвищення зносостійкості деталей сільськогосподарського машинобудування у 1,5–2 рази.

Розроблений підхід поєднання уточненої математичної моделі та адаптивного регулювання енергетичних і геометричних параметрів гідроімпульсного навантаження забезпечує прогнозоване формування зміцненого поверхневого шару з підвищеною зносостійкістю та стабільністю властивостей у широкому діапазоні експлуатаційних умов.

Ключові слова: *гідроімпульсний пристрій, поверхневе пластичне деформування, ударник, пластична деформація, математичне моделювання, генератор імпульсів тиску, кінетична енергія удару, наклеп, зносостійкість, модель Кельвіна-Фойгта.*

Ф. 20. Рис. 10. Літ. 15.

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