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Method of design calculation of a hydropulse device for strain hardening of materials

***Abstract**. The article had expounded of calculation project methodology to a device for strain hardening materials, his power element is connected in one design with a pressure pulse generator and a mechanical accumulator which accumulate potential energy and has the form two parallel installed slotted springs. Proposed calculation project methodology allows using simple dependencies to calculate all basic energy, force and geometrical parameters of the device, which was considered in the article.*

***Streszczenie.** W artykule wyjaśniono metodologię projektu obliczeniowego do urządzenia do utwardzania odkształceniowego materiałów, jego element mocy jest połączony w jednym projekcie z generatorem impulsu ciśnienia i akumulatorem mechanicznym, który gromadzi energię potencjalną i ma postać dwóch równolegle zainstalowanych sprężyn szczelinowych. Proponowana metodologia obliczeń projektowych pozwala wykorzystać proste zależności do obliczenia wszystkich podstawowych energii, sił i parametrów geometrycznych urządzenia, co zostało uwzględnione w artykule.. (**Metoda projektowania urządzenia hydropulacyjnego do odkształceniowego utwardzania materiałów**).*

Keywords: vibration damping; deformation; energy carrier; slotted spring.
Słowa kluczowe: tłumienie drgań; odkształcenie; nośnik energii; sprężyna z nacięciami.

Introduction

Strengthening of details by superficial plastic deformation, In particular, vibration damping treatment is devoted to quite a large number of works, however, in most cases, these works are devoted to processing technologies or investigation of the received surface layer [ ], and there are no work on the development of methods for design calculations of devices for deformation strengthening of materials practical [1-4]. Therefore, the development of a method for design calculation of the original design hydropulse device [5,6] is an actual scientific and engineering task.

Method of design calculation of a hydropulse device for deformation strengthening of materials It is based on the results of theoretical experiments on hydropulse drives and devices of various technical and technological purposes [1 – 3].

During the design calculations of the hydropulse drives and devices on their basis, the content and composition of the basic initial data is determined, in the first place, the purpose of the drive or the device, the required range of adjustment of operating parameters of the executive, such as, for example, the frequency of passage of pressure pulses generated by the pulse pressure generator (PPG) actuator or device, and the amplitude of the vibration of the actuator, the "opening" pressure of the PPG and the maximum inertial mass of the actuator of the actuator or device.

The described general initial data required for the design calculation of the hydropulse devices (or drives) for strain-reinforcing materials may be supplemented by additional data for specific circuitry and design features of the device.

Materials and research results

The method of a specific design calculation of a hydropulse device for strain hardening of materials will be considered for a device whose structural and calculation scheme is shown in Figure 1 [7]. The peculiarity of this device is that its power, elastic and distributive links are combined in such a way that simultaneously perform the functions of the GIT, power cylinder and mechanical battery-storage potential energy.

The power link of the device is a piston-drummer 1 on a stepped rod 1.1 which has a shock tip 1.2 (tool) installed. Piston part of the piston-drummer 1 from the side of the rod 1.1 is designed as distribution unit PPG with the first degree of sealing on the diameter of the facet d_1 the second degree of sealing of the spool type in diameter d_2 piston-drummer 1. The directional part of the piston-striker 1 is made in the form of a prickly spring 1.3. All elements of the piston-drummer 1 are one part that is located in the body (hydrocylinder) of the device (not shown in fig. 1).

The case part (the first degree of sealing of the distributing element of the PPG) of the piston-striker 1 interacts with the faces of the same dimensions formed in the stepped aperture of the floating saddle 2, which is designed as a cylindrical bush, the outer surface of which consists of three parts having different diameters and lengths. The directional part of the saddle 2, with the exact landing, is conjugated with the rigging of the device body by diameter d_2 (equal to the diameter of the spool part of the piston-striker 1). The largest diameter of the cylindrical part of the saddle 2 is a boom limiting the axial displacement of it h_c in the body of the device. The third part of the outer surface of the saddle 2 serves as a guide surface for a cylindrical coil spring 3, whose efforts create an initial seal in the first stage of the PPG [13,14,15].

In order to increase the level of potential energy of the device accumulating during the direct stroke of the piston-drummer 1, an additional cut-off spring 4 is installed in the inner part of the spigot spring 4. Preliminary deformation of the spindle springs 1.3 (y_{01}) and 4 (y_{02})you can independently adjust the screws 5 and 6, respectively 16, 17, 18].

When installing a device on a machine or other technological equipment, due to the installation effort, a small tension of the sprocket springs 1.3 and 4 is created in such a way that a slight clearance h_{02} (h_{02}=0,5…0,6 mm)between the device body and the end of the stem 1.1 larger in diameter, located in the pressure cavity A of the device (see Fig. 1). The degree of the rod 1.1 diameters d_3 is conjugated by the exact alignment with the surface of the hole of the device body. The length of this conjugation is not less $(1{,}5\ldots2)d_3$. To ensure coherence, the surface is diameter d_1, d_2 i d_3 machined from one installation on machines. Clearance h_{02} it is necessary to ensure that, during the interaction of the shock tip 1.2 with the surface of the processed part *Д*, the shock energy is guaranteed to transmit parts *Д*, not the device housing [19].

After the technological gap has been established h_{02}, with screws 5 and 6 the desired one is set according to the set pressure level p_1 «opening» PPG, preliminary deformation of the spring springs 1.3 and 4, respectively y_{01} and y_{02}. Pre-deformation y_{03} cylindrical spring 3 in this device is permanent and is created when the device is assembled.

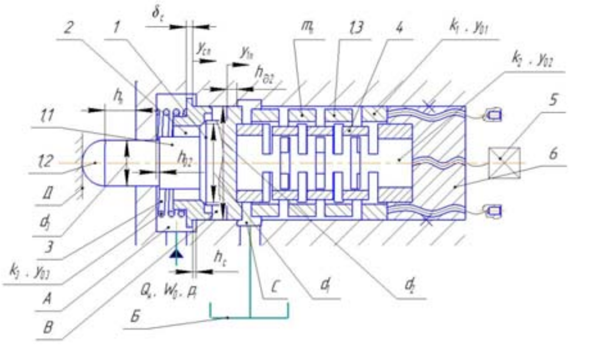

Fig.1. Structural calculation scheme of the hydropulse device for deformation strengthening of materials with built-in PPG

Axial displacement h_c saddles 2 – in essence the initial gap between the saddle hole and the end of the hole of the device body from the side of the rod part of the piston-drummer 1, consists of two parts, $h_c=h_{c_0}-h_{02}$, where $h_{c_0}=(2…2,3)h_{02}$ – the initial gap between the saddle hole 2 and the end of the hole of the device body before its installation in the technological equipment. Taking into account the made remark, $h_c=(1,3…1,0)h_{02}$.

The energy (working fluid) is fed into the pressure cavity A of the device. Between the first and second stages of the HIT sealing of the device an intermediate cavity B is formed, which is a positive block $h_{\partial¶}$ the second (spool) sealing degree of the PPG is separated from the drainage cavity C connected with the hydro-boat B hydrosystem, the device.

In order to ensure the normal operation of the cutting springs 1.3 and 4, there are gaps between their outer surfaces of the working parts of the rings and the guiding surfaces, the value of which is substantiated in the work [8]. The operating cycle of the device can be divided into a certain number of successive stages (phases), as is customary for hydropulse drives and vibrating machines at its base [8,9].

In order to simplify the mathematical description of these stages, they are united in two periods - the direct and reverse of the elements of the PPG or the executive unit of the hydropulse vibrator or device. For the purpose of detailed and correct from the physical point of view of the analysis of the working cycle of the hydropulse equipment, the direct and reverse moves of the executive part of this equipment graphically represent a conditional cyclogram in the form of graphs of energy pressure change, for example, in the pressure line of the PPG, and the displacements of the shut-off elements of the PPG and the executive link vibromachines, etc. . Since the first and subsequent pulses of pressure and displacement changes on this type of cyclograms must differ in shape and duration, then, as a rule, the first and subsequent pulses are depicted in the cyclograms.

The conditional cycle diagram of the operating cycle for the device under consideration (see Fig. 2) can be represented as three pressure changes graphs $p_r=f(t)$ in the pressure cavity A, displacement of the saddle 2 $y_c=f(t)$ and the stroke of the piston-drummer 1 $y_n=f(t)$ during the direct and reverse movements of the piston-drummer 1. The cyclogram contains the first and second impulses of the change p_r, y_c and y_n assuming that there is no transient process during the startup of the device, which in the real system necessarily takes place. To simplify the description of the processes of pressure change in the cavity A and the displacement of the saddle 2 and the piston-shock 1, the function $p_r=f(t)$, $y_c=f(t)$ and $y_n=f(t)$ are linear dependencies on all characteristic time intervals.

According to the proposed cyclogram when feeding the energy carrier to the pressure cavity A of the device under closed PPG pressure p_{r_A} the energy car begins to grow. When the pressure level $p_{r_A} \geq p_c$ (here p_c – the boundary pressure of the energy carrier, in which the piston-drummer 1 and saddle 2 begin to move), the saddle 2 and the piston-drummer 1 begin to move along the straight path as a whole.

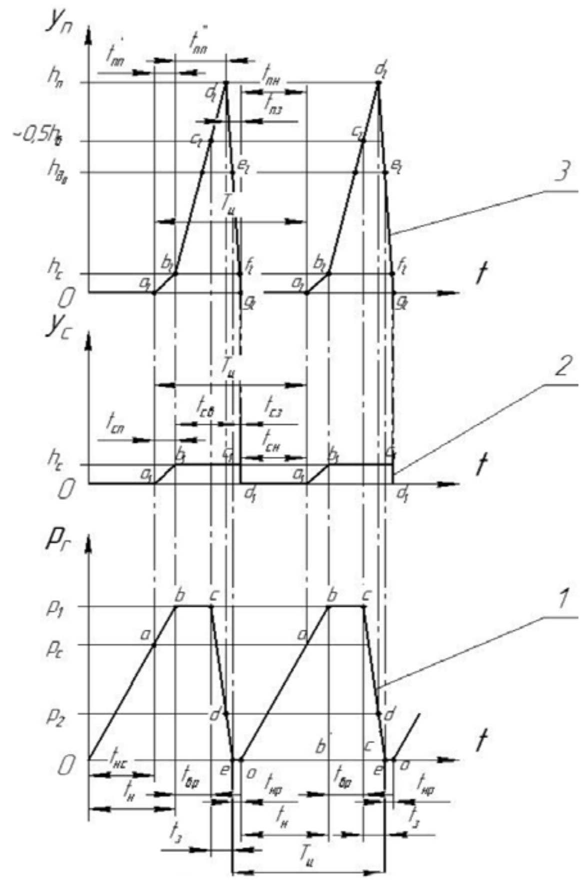

Fig. 2. Conditional graph of the working cycle of the device: curve 1 – $p_r=f(t)$, curve 2 – $y_c=f(t)$; curve 3 – $y_n=f(t)$

After moving the system saddle 2 - piston-drummer 1 at a distance $y_{c_{max}}=h_c$, saddle 2 with its hill, height δ_c, rests on the end of the hole, and the pressure in the cavity A rises to the level $p_{r_A} \geq p_1$ (here p_1 – pressure «opening» PPG). Tightness of the first degree PPG sealing is violated, the cavities A and B are combined and the action of the energy carrier under pressure p_1 extends to the working area of the cross-section $A_2=\pi(d_2^2-d_3^2)/4$ the second degree of sealing, which is larger than the working area $A_1=\pi(d_1^2-d_3^2)/4$ the first degree of PPG sealing because $d_3 < d_1 < d_2$ (see Fig. 1). The action of pressure is of magnitude p_1 to the square A_2 causes the accelerated movement of the piston-drummer 1 on the path of a positive overlap $h_\partial = h_{\partial_0} - h_c$ (h_{∂_0} –initial positive displacement of the spool part of the piston-striker 1, see Fig. 1). Saddle 2 at this moment is pressed to the end of the cutting device body with a spring 3 and does not move.

After passing the piston-striker 1 positive overlap h_∂ there is a connection of pressure A and intermediate B into cavities with drainage C and the movement of a piston-shock 1 on the way of a negative overlap $h_в = h_n - (h_{\partial_0} + h_c)$ (here h_n – full stroke of the piston-drummer 1). This stage of the motion of the piston-drummer 1 is the beginning of the «opening» of the PPG.

Since the intensity of the process of reducing the pressure level p_{r_A} in the cavity A during its connection with the drainage cavity C depends on the area of the open slit $A_щ = \pi d_2 (y_n - h_\partial)$ PPG, hydraulic resistance of the hydroline of the hydrosystem of the device and the feeder $Q_н$ hydraulic pump drive device, then you can assume that the pressure level $p_{r_A} = p_1$ will be kept for some time during the movement of the piston-drummer 1 on the way $h_в$ during his direct stroke. Approximately we can assume that preservation of the level of pressure $p_{r_A} = p_1$ continues during the displacement of the piston-drummer 1 during a direct stroke at a distance $\sim 0{,}5h_в$. The second half of the path $h_в$ the piston-drummer 1 passes in inertia (since at the time of opening the PPG it acquires a certain velocity), and the energy pressure in cavity A will decrease from the level $p_{r_A} = p_1$ to the level $p_{r_A} \le p_2$ (where p_2 – the "closure" pressure of the PPG). For achievement $p_{r_A} = p_2$ the reverse of the piston-drummer 1 begins on the path of a negative overlap $h_в$.

The feature of the considered hydropulse device for strain hardening of materials is a large area of the passage section of the open drain slot PPG $A_{щ_{max}} = \pi d_2 h_в$, which causes a rapid decrease in the energy of the energy at the moment of the reverse of the piston-drummer 1 practically to the level of drain pressure $p_{з_л} \ll p_2$. During the design calculation it can be assumed that $p_{з_л} = 0$.

Due to the previous deformation of the spring springs 1.3 and 4 and the twisted springs 3, a certain initial potential energy is stored in the system of the power unit and the integrated PPG, which can be estimated in accordance with the known [8] dependence:

(1) $E_{no}=0{,}5(k_1 y_{01}^2+k_2 y_{02}^2+k_3 y_{03}^2)$,

where k_1, k_2, k_3 – respectively, the stiffness of the inclined springs 1.3 and 4 and the twisted springs 3.

At the end of the direct stroke of the piston-striker 1, the potential energy of the power link of the device increases by the value:

(2) $\Delta E_n=0{,}5(k_1+k_2)h_п^2-k_3 h_c^2$.

The initial potential energy accumulated by the twisted spring 3 during the forward passage of the saddle 2 decreases by the values in $\Delta E_n=0{,}5k_3\left(y_{03}^2-h_c^2\right)$.

Since the total stiffness $k_{12}=k_1+k_2$ trench springs 1.3 and 4 are much more rigid k_3 twisted springs 3 ($k_{12} \gg k_3$), ana $h_c \ll h_п$, then a fraction of potential energy $0{,}5k_3h_c^2$ can be neglected and assume that

(3) $\Delta E_n=0{,}5(k_1+k_2)h_п^2$.

At the end of the reverse of the piston-shock 1 is the potential energy ΔE_n is expended against the forces of energy pressure, which create the resistance to the reciprocal movement of the piston-striker 1, the viscous friction forces in the guiding surfaces of the piston-striker 1 and the reverse displacement of the saddle 2. It is spent to work against the forces of energy pressure, which create resistance to the reciprocal movement of the piston-drummer 1, viscous friction forces in the guiding surfaces of the piston-drummer 1 and the reverse displacement of the saddle 2. Most of the potential energy ΔE_n transforms into kinetic energy $E_к$ the piston-striker 1 whose transformation can be approximated by the equation:

(4) $E_к=0{,}5m_п V_{п_{max}}^2=E_{п¶}+E_{кв}=0{,}5k_¶\delta_{п¶}^2+E_k k_в^2$,

where $m_п$ – the consolidated mass of the piston-drummer 1, which is the sum of the moving parts of the actual piston-drummer 1 and the perforated springs 1.3 i 4; $V_{п_{max}}$ – maximum speed of the piston-drummer 1 at the end of its reverse; $E_{п¶}=0{,}5k_¶\delta_{п¶}^2$ – the potential energy of the elastic deformation of the processed part; k_∂ – local (contact) stiffness details Д; $\delta_{п\partial} \gg \delta_{пл\partial}$ – elastic local (contact) deformation of the part Д, assuming that it is much more plastic local deformation opaf the rts Д; $E_{кв}=E_k k_в^2$ – the kinetic energy of the rebound of the piston-striker 1 in the direction of its direct motion $y_{пп}$; $k_в$ – coefficient of speed recovery for partial elastic impact of tip 1.2 on detail Д [5]. If $V_{n_{max}} \le 3$ m/s, then for impact the steel is a constant coefficient $k_в = 5/9 \approx 0{,}56$ [10].

Considering that during the return stroke impactor 1 medium-pressure cavity A can be taken close to the drain ($p_{r_A} \ll p_{зл}$),and the coefficient of friction in the guides of the piston-drummer 1 under the liquid friction regime does not exceed f=0,005 [11], the cost of potential energy ΔE_n to work against the forces of pressure and friction can be neglected, or, in the extreme case, to consider them in the form of a particle

(5) $\Delta E_{fp}=k_{fp}\Delta E_n$,

where $k_{fp}<1$ – coefficient of consumption of potential energy ΔE_n for work against forces of pressure and friction, for example, $k_{fp}=0{,}005\ldots0{,}01$. Under such an assumption, we can assume that the potential energy ΔE_n for the most part, becomes kinetic $E_к$:

(6) $\Delta E_n = E_к + \Delta E_{fp} = E_к + k_{fp}\Delta E_n$

where

(7) $E_к=\Delta E_n(1-k_{fp})$.

Components of the period $T_ц$ cycles of pulses of pressure $p_r(t)$, saddle moves 2 $y_c(t)$ and saddle moves 1 $y_n(t)$

can be expressed, in accordance with the adopted cyclogram (see Fig. 2), by a simple equation:

(8) $T_ц = t_{нр}+t_н + t_{вр}+t_з = t_{сп}+t_{св} + t_{сз}+t_{сн} = t'_{пп} + t''_{пп}+t_{пз}+t_{пн} = \nu^{-1}$

where

(9) $t_н = p_1 W_0/(Q_н)$ –

time of growth (set) of pressure in the pressure cavity A device from $p_{г_А}=0$ to $p_{г_А}=p_1$ [8, 9] (here: W_0 – the initial volume of the pressure cavity of the hydrosystem of the drive device; $Q_н$ – supply of hydraulic pump to the hydraulic system of the device; κ – isothermal module of elasticity of energy carrier); $t_{нр}$ – time of pressure shutdown at the level p_2 at the moment of interaction of the shock tip 1.2 with the surface of the machined part at the end of the reverse of the piston striker 1; $t_{вр}$ – time of the pressure shutdown in the cavity A at the level $p_{г_А}=p_1$ at the moment of acceleration of the piston-drummer 1 on the way $y_{пп} \approx h_{д¶}+0,5h_в$; $t_з$ – time for reducing the pressure in the hydraulic system of the device from the level p_1 to the level $p_{зл}=0$; $t_{сп}=t'_{пп}$ – time of joint motion of saddle 2 on the way h_c and a piston-drummer 1 ($t'_{пп}$) ; $t_{св}$ – time of saddle position 2 in spring condition 3 pressed (top of the graph, see fig. 2); $t_{сз}$ – return travel time of saddle 2 on the way h_c under the impact action of the piston-drummer 1 on the segment $y_{пз}=h_c$ his reverse; $t_{сн}=t_{нс}$ – time of saddle position 2 in its initial position (below the cyclogram), which can be assumed equal to time $t_{нс}=p_c W_0/(Q_н)$ – set pressure in the pressure cavity A hydraulic drive unit from $p_{г_А}=0$ to $p_{г_А}=p_c$); $t_{пп}=t'_{пп}+t''_{пп}$ – time of the direct stroke of the piston-drummer 1 on the way $h_п$ (here $t''_{пп}$ – time of the direct stroke of the piston-drummer 1 on the way $y_{nn} = h_\partial + h_в$); $t_{пз}$ – reverse time of the piston-drummer 1 on the way $h_п$; $t_{пн}$ – the time of the piston-striker 1 position in the initial position (below the cyclogram), which is the phase shift between the pressure pulses and the displacements of the saddle 2 and the piston-striker 1; ν – linear frequency of passage of pressure pulses, displacements (vibrations) of a piston-striker 1 and saddle 2.

As a conclusion from the foregoing, one can state that for the hydropulse devices for deformation strengthening of materials and, in particular, for the considered constructive scheme of the device (see Fig. 1), the basic initial data for the design calculation should be assigned:

- potential energy $\Delta E_п$ (see (3), which accumulates the power link (piston-drummer 1) of the device at the end of its direct drive (the charge);
- approximate consolidated mass $m_п$ power unit;
- maximum pressure «opening» $p_{1_{max}}$ or the range of its regulation $\left[p_{1_{min}}\ ,\ p_{1_{max}}\right]$;
- frequency range ν passage of pulses of pressure;
- minimum permissible positive overlap h_∂ spindle part of the shutter element (second degree of sealing) PPG;
- acceptable speeds of the energy carrier in the pressure $[V_н]$, drainage $[V_{зл}]$ and open cracks $[V_г]$ shutter element PPG;
- maximum deformation of the spring springs 1.3 and 4, respectively, $y_{01_{max}}$ and $y_{02_{max}}$ and springs 3, $y_{03_{max}}$;
- tentative maximum volume $W_{o_{max}}$ pressure cavity hydrosystem drive device.

The experience of designing and operating hydraulic drives of general machine-building applications and hydropulse drives [8] shows that the minimum positive displacement of spool shut-off and distribution elements is usually chosen within $2mm \le h_\partial \le 7mm$ depending on the accuracy of the conjugation of these elements with the guiding surfaces (the cuttings of the shells, sleeves, etc.), their sizes and requirements for the sealing of the slide overlap. Negative overlap $h_в$ spool shut-off and distribution elements are assigned in accordance with the maximum permissible speed $[V_г]$ the energy carrier through the open slit of the spool valve, which is determined by the differential pressure on this gap and the energy flow through it. In spool PPG, the hydropulse drives often have a negative overlap $h_в$ assign less or equal to positive:: $h_в \le h_\partial$ [8].

For the device under consideration, given that there is a positive overlap h_∂ is an important part of the full strength of the power line, on which the level of accumulated potential energy depends ΔE_n , and the factor of the spool sealing degree of PPG, it is expedient to take a positive overlap within $h_\partial = (4\ldots5)$ mm.

In order to minimize the axial dimensions of the device and ensure the interaction of the shock tip 1.2 of the piston-drummer 1 with the surface of the workpiece, an clearance $h_{02} = h_c \ge 0,6mm$.

According to the comments made, we take: $h_\partial = 4mm$; $h_в = 4mm$; $h_{02}=h_c=0,6mm$.

For the following data: $h_{\partial_0} = h_\partial + h_c = 4+0,6 = 4,6mm;$ $h_n = h_{\partial_0} + h_в = 4,6+4 = 8,6mm$.

From dependence (3) we obtain a formula for calculating the total rigidity of the spring springs 1.3 and 4

(10) $k_{12} = 2\Delta E_n / h_n^2;$

According to the known [1 - 3] for hydropulse drives formula:

(11) $p_{1_{max}} \ge \left[k_{12}\left(h_c + y_{01_{max}} + y_{02_{max}}\right)\right]A_1^{-1};$

and the scheme of the device (see Fig. 1), we find the desired area of the cross section of the first degree of the intermediate sealing PPG

(12) $A_1 \ge \left[k_{12}\left(h_c + y_{01_{max}} + y_{02_{max}}\right)\right]p_1^{-1};$

Taking into account that the spring springs 1.3 and 4 are high-rigid springs, whose forces vary in linear dependence on deformation in a relatively narrow range of deformation, to assign large values of the previous strains $y_{01_{max}}$ and $y_{02_{max}}$ not appropriate In addition, a significant amount $y_{01_{max}}$ and $y_{02_{max}}$ considerably increases the axial dimensions of the spring springs 1.3 and 4 and the device as a whole. In our opinion, the size of the previous deformations of these springs can be taken $y_{01_{max}} = y_{02_{max}} = y_{0П_{max}} = 4mm$.

Boundary pressure p_c energy can be estimated from inequality

(13) $p_1A_1 \le k_{12}(h_c + y_{01\max} + y_{02\max}) - k_3y_{03} = k_{12}(h_c + 2y_{0\Pi\max}) - k_3y_{03}$,

where

(14) $p_c \le \left[k_{12}\left(h_c + 2y_{on_{\max}}\right) - k_3y_{03}\right]A_1^{-1}$;

The purpose of the spring 3 (see Fig. 1) is to create a sufficient contact for the first-degree sealing of the PPG (saddle contact 2 with a plugged element of the piston-shock hammer sealing 1). Since the sealing between the trim sections of saddle 2 and the degree of the piston-shock 1 is amplified by the action of the working pressure of the energy carrier on the saddle 2 from the pressure cavity A, then the force of the spring 3 is required to create the initial contact sealing pressure, which is relatively small [8], which causes low rigidity k_3 and the previous deformation y_{03} of the spring 3, which, as already noted, can be created during the assembly of the device.

Comparing (11) and (14), we establish that the initial force of the spring 3

(15) $k_3y_{03} \le (p_{1_{\max}} - p_c)A_1$

where $p_{1_{\max}} = k_{12}(h_c + 2y_{on_{\max}})A_1^{-1}$ for accepted values $y_{01_{\max}}$, $y_{02_{\max}}$ and $h_c = h_{02}$

In the process of vibration shock the device of the oscillation of the saddle 2 must be carried out in synchrony with the oscillations of the piston-striker 1 (see Fig. 2) with the same period $T_ц$ (frequency ν). Such a saddle motion mode 2 is possible if its own circular frequency $\omega_{03} > \omega$, where $\omega = 2\pi\nu$ the circular frequency of the oscillations of the piston-drummer 1 is due to the frequency of passing v of the pressure pulses (the initial parameter is given). In other words, saddle 2 should fluctuate in resonance mode. In the resonance mode, the synchronous motion of the saddle 2 and the piston-shock 1 will be violated, and in the resonance mode, the oscillation amplitude (displacement) of the saddle 2 may exceed the value of the reciprocal movement of the piston-drummer 1 h_c, which can cause jamming of the guide surface of the saddle 2 in diameter d_2, conjugated to the corresponding surface of the hole (housing) of the device.

According to the results of theoretical and experimental studies of oscillatory systems, [11] is established that in the resonance regime the amplitude of the forced oscillations does not change (does not increase), if (in our case) $\omega_{03}\omega^{-1} = \sqrt{2}$.

If you do not take into account the elastic action on the saddle 2 of the rigidity of the pressure cavity of the hydrosystem of the device, then its own circular frequency ω_{03} oscillations of saddle 2 can be calculated from the known [9] dependence:

(16) $\omega_{03} = 2\pi\nu_{03} = \sqrt{k_3 m_2^{-1}}$,

where ν_{03} – linear frequency of own oscillations of the saddle 2; m_2 – seated mass of saddle 2 taking into account part of spring mass 3. Accepting $\omega_{03} = \sqrt{2}\omega = \sqrt{2} \times 2\pi\nu$, from (16) we find:

(17) $k_3 = 8\pi^2\nu^2 m_2 \approx 78{,}88\nu^2 m_2$

Since, as noted above, the sealing force of the spring 3 on the saddle 2 is amplified by the action of the energy of the pressure on it, then for the sake of convenience, the device is assembled, the value of the previous deformation y_{03} springs 3 is significantly larger $y_{on_{\max}}$ not expedient. During the design calculation, the device can be accepted $y_{03} = y_{on_{\max}} + (1...2)\text{mm} = 4 + (1...2) = (5...6)\text{mm}$.

Taking into account the above and the dependences (15) and (17), we obtain the formula for calculation p_c:

(18) $p_c \le p_{1_{\max}} - k_3y_{03}A_1^{-1}$.

According to the proposed cycle diagram of the device's working cycle (see Fig. 2), the pressure level p_c determines the phase shift $t_{нс}$ between impulses of pressure and displacements of saddle 2 and a piston-drummer 1:

(19) $t_{сн} = t_{нс} = t_н - t_{сп}$,

where

(20) $t_{сп} = t_н - t_{нс} = (p_{1_{\max}} - p_c)W_0/(Q_H\kappa)$,

where $t_{нс} = p_cW_0/(Q_H\kappa)$ [7, 8]. Obviously, for relatively low values k_3 and y_{03} in a relatively developed cross-sectional area A_1 the first degree of PPG sealing, the difference between pressure levels $p_{1_{\max}}$ and p_c the energy was small, which causes a short time $t_{сп}$ moving the saddle to a distance h_c, which is also significantly less than the turn $h_п$ piston-drummer 1. For the above reasons, it is possible to draw a preliminary conclusion that the saddle motion has little effect on the dynamics of the power unit of the device. This allows studying the dynamics of the device not to take into account the influence on the dynamic processes in the hydropulse drive device parameters of the vibrational motion of saddle 2.

As it was noted, the reverse of the piston-striker 1 will begin (see Fig. 2), when the pressure of the energy carrier in the pressure cavity of the hydraulic system of the device will decrease to the level of pressure "closure" of the PPG

(21) $p_{2_{\min}} \le k_{12}(h_c + 2y_{оп_{\max}} + h_п)/A_2$;

Comparing the expressions for $p_{1_{\min}}$ $p_{1_{max}}$ (see (11), (15) and $p_{2_{\min}}$, find the relationship between energy pressure levels during the opening and closing of the PPG device:

(22) $p_{2_{\min}} \le p_{1_{\max}} {A_1}/{A_2} + {k_{12}h_п}/{A_2}$;

The research of the hydropulse drives and the PPG [1-3] have established rational ratios (gear ratios) $U_{21}^{0,5} = A_1/A_2$ for different frequency pulses of pressure pulses created by PPG. Since in the device under consideration the upper level of the frequency of the passage of the pulses of pressure is recommended at the level $\nu = (60...100)$ Hz, then from our point of view the most expedient will be value $U_{21}^{0,5} = 0{,}45…0{,}5$ [8, 9].

If you neglect the size of the drain pressure of the energy carrier in the drainage system of the hydrosystem device drive, $p_{зл} = 0$, then according to the law of conservation of energy, for neglecting the cost of overcoming the forces of friction, work A_p pressure forces during the cycle $T_ц$ can be described on the basis of the conditional cyclogram (see Fig. 2) by the equation:

(23)
$$A_p = 0{,}5(p_c + p_{1_{\max}})A_1h_c + \left[p_{1_{\max}}(h_д + 0{,}75h_в) + 0{,}75p_{2_{\min}}h_в\right]A_2 =$$
$$= 0{,}5(p_c + p_{1_{\max}})A_1h_c + \left[p_{1_{\max}}(h_д + 0{,}75h_в)\right]A_2 + 0{,}75(p_{1_{\max}}A_1 + k_{12}h_п)h_в.$$

Suppose that work A_p in an hour $T_ц$ goes into the amount of potential $\Delta E_п$ (see (3) and kinetic energy $E_к$, (see (7), respectively, during the forward and reverse movements of the piston-drummer 1:

(24) $A_p \approx \Delta E_\text{п} + E_\text{к} = \Delta E_\text{п}(2 - k_{fp})$.

After simple transformations from (24), taking into account (23), we obtain the dependence for the calculation A_2 (cross-sectional area of the second (spool) degree of sealing of the steam element PPG):

(25) $$A_2 = \frac{\Delta E_n\left(2\text{-}k_{fp}\right)\text{-}0{,}5[\left(p_c+p_{1_{max}}\right)A_1h_c+1{,}5(p_{1_{max}}A_1+k_{12}h_n)h_\text{в}]}{p_{1_{max}}\left(h_{\P}+0{,}75h_\text{в}\right)};$$

Diameter dimensional value d_3 (see Fig. 1) of the piston-striker 1 rod 1.1 can be designed for design reasons to provide the required dimensions for the convenience of mounting at the end of the rod 1.1 drum tips 1.2 (tools) of various shapes and sizes. Obviously, the size of the rod 1.1 and the shock tip 1.2 must be such that their reduced stiffness significantly exceeds the total stiffness k_{12} perforated springs 1.3 and 4 and had more contact stiffness k_∂ processing details. Under these conditions, the elastic deformation of rod 1.1 during the shock interaction of the shock tip 1.2 with the surface of the part can be neglected and assumed that the material of rod 1.1 does not receive residual deformations, and the stresses arising in the cross sections of the stock are much smaller than the limits of the elasticity of its material. Usually, the limit $\sigma_\text{пр}$ of elasticity of materials, due to the difficulties of its definition in the reference books is not given. It is believed that the residual deformation, which corresponds to the elasticity of materials, does not exceed $\varepsilon_\text{зал}=(1...5)10^{-5}$ [10].

Material of stock 1.1 within the elastic deformation obeys Hooke's law [10]:

(26) $\sigma_\text{пр}=\varepsilon_\text{зал}E$,

where E – is the elastic modulus of the material of rod 1.1, as well as the piston-drummer 1, since they constitute one integral in the considered construction. If you use absolute strain of rod 1.1, then Hooke's law can be written in a different form:

(27) $\sigma_\text{пр}A_3=k_\text{ш}\Delta l_\text{ш}$

where $k_\text{ш}=EA_3/l_\text{ш}$ – strength of the rod is 1.1 in length $l_\text{ш}$ and the cross-sectional area $A_3=\pi d_3^2/4$; $\Delta l_\text{ш} = \varepsilon_\text{зал} l_\text{ш}$ – absolute strain of the rod 1.1.

The length of the rod 1.1 can be chosen for the following reasons: the part of the length of the rod 1.1 located in the pressure cavity A of the device is determined by the length of the cutter A (the pressure cavity of the device) in which it is necessary to place the length part of the saddle 2 and the spring 3. Directional part of rod 1, as a rule, should not be less 2 d_3 [8]. The length of the speaker from the device end of rod 1.1 should be sufficient to provide the required maximum stroke h_n piston drill 1 and mounting at this end of the drum tip 1.2 (tool). Thus, according to the above considerations, the length of the rod should not be less than $l_\text{ш}=(4...5)d_3$. For such a value $l_\text{ш}$ and recommend $\varepsilon_\text{зал}$, $\Delta l_\text{ш} = \varepsilon_\text{зал} l_\text{ш}=(1...5)10^{-5}(4...5)d_3=(4...25)10^{-5}d_3$. If we take into account the lower boundary $\varepsilon_\text{зал} = 1 \cdot 10^{-5}$, then $\Delta l_\text{ш} = (4\ldots5)10^{-5}d_3$.

In order to virtually eliminate the effect of a possible axial elastic deformation of rod 1.1 on the result of the deformation strengthening of the surface of the machined part D, the stiffness of rod 1 should be set to be substantially higher than the total stiffness k_{12} perforated springs 1.3 and 4, for example $k_\text{ш} = (30...50)k_{12}$.

Taking into account the above, depending on (27) taking into account (26), we obtain a formula for estimating the value of the diameter of the stock d_3:

(28) $d_3=(152...318)k_{12}E^{-1}$,

where d_3 – in mm; k_{12} – in N/mm; E – in MPa.

In the design process for small-sized devices, for which k_{12} may have order $\sim 10^4$ N/mm, the diameter value should be assigned to a numerical factor close to 152, and for large-sized devices with order $k_{12} \sim 10^3$ N/mm takes a numerical coefficient close to 318.

Diameters d_1 and d_2 the first and second stages of the sealing of the PPG (power link of the piston-drummer 1) can be calculated from simple formulas:

(29) $d_1 = 1{,}13\sqrt{A_1 + A_3}$;

(30) $d_2 = 1{,}13\sqrt{A_2 + A_3}$;

According to the conditional cycle graph of the working cycle of the device (see Fig. 2), the greatest duration in the pulse of pressure $p_r=f(t)$ has a time of increase (set) of pressure in the pressure cavity A of the device, which is determined by the dependence (9). Through time $t_\text{н}$ it is expedient to express other components of the cycle (see (8):

(31) $$T_\text{Ц}=K_{\text{Ц}_p}t_\text{н}=\nu^{-1}=\frac{K_{\text{Ц}_p}p_{1_{max}}W_0}{Q\kappa}$$

where $K_{\text{Ц}_p}=1+(t_\text{нр}+t_\text{вр}+t_\text{з})/t_\text{н}$ – a coefficient that can be conventionally called the cyclic coefficient of pressure pulse. By the graph of the graph (see Fig. 2) it is obvious that $K_{\text{Ц}_p} > 1$. To ensure the maximum set frequency ν_{max} passage of pressure pulses with the level of pressure «opening» PPG $p_{1_{max}}$ and the specified volume W_0 the hydraulic reservoir of the hydrosystem of the device according to (31) requires the delivery of a hydraulic pump:

(32) $Q_H=K_{\text{Ц}_p}\nu_{max}p_{1_{max}}W_0\kappa^{-1}$.

Sufficiently accurate value of the coefficient $K_{\text{Ц}_p}$ can be found as a result of the analysis of the mathematical model of the hydropulse drive of the device, the adequacy of which of the experimental sample of the device can be established by experimental research of the sample. The easiest estimate of the level $K_{\text{Ц}_p}$ can be set due to the relative values of individual time components of the cycle using the conditional cyclogram $p_r=f(t)$, introducing the concept of the magnitude of the pressure pulse:

(33) $\mu_{t_p}=T_\text{Ц}/ob'=1/(ob' \cdot \nu_{max})$,s/mm

where ob' – length of the segment ob' on the cyclogram, mm.

By measuring the corresponding segments on the graph (see Fig. 2), you can record:

(34) $t_\text{н}=\mu_{t_p}ob'$; $t_\text{нр}=\mu_{t_p}ea$; $t_\text{вр}=\mu_{t_p}b'c$; $t_\text{з}=\mu_{t_p}c'e$.

Substituting (34) into an expression for $K_{\text{Ц}_p}$, find its value in relative values:

(35) $K_{\text{Ц}_p}=1+(eo+b'c'+c'e)/ob'$.

Since the cycle diagram of the device's operating cycle is conditional, and the coefficient has an approximate estimated value, then in formula (32) it is expedient to enter a stock factor $К_з$=1,1...1,25:

(36) $Q_Н = К_3 К_{Щ_р} \nu_{max} p_{1_{max}} W_0 \kappa^{-1}$.

Time $t_з$ can be estimated by a dependence similar (9):

(37) $t_з = p_1 W_0 (\bar{Q}_Г \kappa)^{-1}$.

where $\bar{Q}_Г$ – average energy consumption through the open slit $A_Щ$ ГPPG device. Level $\bar{Q}_Г$ can be estimated through the relationship:

(38) $t_з / t_н = Q_Н / \bar{Q}_Г = ce' / ob' = \tau_{нз}$;

where

(39) $\bar{Q}_Г = Q_Н / \tau_{нз}$,

where $\tau_{нз}$ – relative time of reduction of energy carrier pressure in the device's hydrosystem from the level p_1 to the level $p_{зл} = 0$. According to the conditional cyclogram of the working cycle of the device $\tau_{нз} < 1$, and hence $\bar{Q}_Г > Q_Н$. By cost $\bar{Q}_Г$ the energy carrier can roughly determine the average energy velocity through the gap $A_Щ$, which should not exceed permissible $[V_Г]$ [11]:

(40) $V_Г = \bar{Q}_Г / \pi d_2 h_в \le [V_Г]$.

If the condition (40) is not satisfied, then you can increase it $h_в$ or d_2, but in this case should be synchronously changed d_1 and d_3.

The study and calculation of slit springs is devoted to work [8]. The authors of this work proposed an approximate formula for calculating the stiffness of the prickly spring

(41) $k_{ПП} = zE_{ПП} I_{зг} / k_\alpha R_{ПП}^3 n'$

where $z \ge 2$ – the number of openings in the inclined spring, most often $z = 2$; $E_{ПП}$ – modulus of elasticity material of a cutting spring; $I_{зг} = ab^3/12$ – moment of inertia of the section of the ring during bending (here «a» and «b» – respectively the width and length of the ring of the spring);; $R_{ПП} = 0{,}5(D_{ПП}\text{-}a)$ – average radius of the spring; n – number of working rings of a spring; $D_{ПП}$ – outer diameter of the spring; $k_\alpha = K_1\left[(\alpha/4) - (1-K_2)\tan(\alpha/4)\right]$, where $K_1 = E_{ПП} I_{зг} / G_{ПП} I_{кр} = 0{,}65K(K-0{,}63)$ – dimensionless ratio of stiffness of the ring during bending and twisting at $E_{ПП}/G_{ПП} = 2(1+\mu)$ (here $G_{ПП}$ – modulus of the bias (elasticity) of the spring material during torsion; μ=0,3 – poisson coefficient of spring material); α – the central angle of rotation of the rings of the spring one relative to the other without taking into account the width of the jumper between them (for $z = 2$, α=π);

$$K_2 = 1 - \frac{K_1 \sin\left(\frac{\alpha}{2}\right)\left[1+\tan\frac{\alpha}{4}\right]^2}{(K_1-1)\tan\frac{\alpha}{4} + (K_1+1)\left[1+\tan\left(\frac{\alpha}{4}\right)^2\left(\frac{\alpha}{4}\right)\right]}$$ – geometric coefficient; K=a/b; $I_{кр} = (K\text{-}0{,}63)b^4/3$ – moment of inertia of the section of the spring ring during torsion.

The strength of the cutting springs is checked by the energy theory of strength, calculated in a dangerous section of the ring of equivalent stress [11, 20]

(42) $\sigma_{екв} = \sqrt{\sigma_{зг}^2 + 3\tau_{кр}^2} \le [\sigma_{зг}]$,

where $\sigma_{зг} = M_{зг} / W_{зг}$; $\tau_{кр} = T / W_{кр}$ – respectively, the bending and torsion stresses in the dangerous sections of the rings of the spring, located at the points of transition of the rings to the jumpers; $M_{зг}, T$ – respectively, bending and twisting moments arising in spring cross sections under the action of its axial load F_{max}; $W_{зг} = ab^2/6$, $W_{кр} = (K-0{,}63)b^3/3$ [11] – respectively the moments of the resistance of the cross section of the ring during bending and torsion; $[\sigma_{зг}]$ – permissible spring bending stress tolerance.

In the design calculation of the trunks 1.3 and 4 devices (see Figure 1), the output formulas (41) and (42) can be simplified by introducing the following assumptions: b=a; the height of the jumper between the rings of the spring has the shape of a cube with an edge equal a; number of jumpers $n_п$ the perforated spring is connected with the number of its working rings n growing dependence

(43) $n_п = 2(n+1)$;

break springs 1.3 and 4 have two support rings, $(n_{оп} = 2)$, the thickness of each of them, in order to ensure their strength and rigidity, according to the recommendations given in the work [8], we accept $a_{оп} = 3a$, $z = 2$.

By making the corresponding transformations in formulas (41) and (42), according to assumed assumptions, we obtain simple dependencies for calculating the stiffnesses k_1 and k_2 and tests on the strength of the cutting springs 1.3 i 4:

(44) $k_i = (1{,}035 E_{ПП} a_i^4) / R_i^3 n_i$;

(45) $\sigma_{екв_i} = (1{,}22 F_{max} R_i) a_i^{-3} \le [\sigma_{зг}]$,

where is the index i=1 – for spring 1.3; index i=2 – for the spring 4; a_i – width of the ring of the spring 1.3 (a_1)and springs 4 (a_2); $R_i = 0{,}5(d_i\text{-}a_i)$ – average radius of cut springs: 1.3 -$R_1 = 0{,}5[(d_2 - 2\delta_1) - a_1]$; -$R_2 = 0{,}5[(d_4 - 2\delta_2) - a_2]$ (here δ_1, δ_2 are the lateral gaps between the outer surfaces of the inclined springs, respectively, 1.3 and 4 and their guiding surfaces, which exclude jamming of the springs during their working deformation [8]; d_4 – outer diameter of the spring surface of the spring 4); n_i – Number of working rings of a spring 1.3 (n_1) and 4 (n_2);($F_{max} = p_{1_{max}} A_2$) – maximum axial load of spring 1.3 and 4.

In order to simplify calculations and follow the recommendations of work [2], we can accept $\delta_1 = \delta_2 = \delta_{12}$, as well as from fig. 1 it is obvious that $d_4 = d_2 - 2\delta_{12} - 2a_1 = d_2' - 2a_1$ (here $d_2' = d_2 - 2\delta_{12}$). Inner

diameter of the prickly spring 4 $d_4^{вн}=d_4-2\delta_{12}-2a_2=d_4'-2a_2$, where $d_4'=d_4-2\delta_{12}$.
According to these remarks:

(46) $R_1=0,5(d_2'\text{-}a_1);$

(47) $R_1=0,5(d_4'\text{-}a_2)=\left[d_2'\text{-}2(a_1+\delta_{12})-a_2\right]=(2R_1-a_1-\delta_{12}-a_2),$

where $d_2'=2R_1+a_1$.

From the point of view of providing the same service life of the spring springs 1.3 and 4, since they are loaded at the same time and one by size of effort F_{max}, equivalent stresses $\sigma_{екв}$, which arise in cross-sections of springs, should be the same $\sigma_{екв_1}=\sigma_{екв_2}$.

Under this condition we can find from (45) that

(48) $R_1/R_2=a_1^3/a_2^3,$

where

(49) $a_2=a_1\sqrt[3]{(R_2/R_1)}.$

Width a_1 the ring of the slip spring 1.3 can be determined from formula (45), turning it into equality by introducing a stock factor $K_\sigma=[\sigma_{зг}]/\sigma_{екв}$, for example, at the level of 1.5 - 2.0, taking into account the fact that the spring material is loaded cyclically and works on endurance [8]:

(50) $\frac{[\sigma_{зг}]}{K_\sigma}=(1,22F_{max}R_1)a_1^{-3}=\left[1,22F_{max}0,5(d_2'-a_1)\right]a_1^{-3}.$

Solving with the Kardan formula [12] the cubic equation (50), which has only one real root, because the discriminant

(51) $D_1=\frac{\beta^2}{(\sigma')^2}\left[\frac{(d_2')^2}{4}+\frac{\beta}{27\sigma'}\right]>0,$

we will find

(52) $a_1=\sqrt[3]{\frac{\beta d_2'}{2\sigma'}+\sqrt{D_1}}+\sqrt[3]{-\frac{\beta d_2'}{2\sigma'}-\sqrt{D_1}},$

where $\beta=0,61F_{max}$; $\sigma'=[\sigma_{зг}]/K_\sigma$.

For similar considerations and the similar (52) formula, one can determine the width of the ring a_2 prickly spring 4:
where is the discriminator

(53) $a_2=\sqrt[3]{\frac{\beta d_4'}{2\sigma'}+\sqrt{D_2}}+\sqrt[3]{-\frac{\beta d_4'}{2\sigma'}-\sqrt{D_2}},$

(54) $D_2=\frac{\beta^2}{(\sigma')^2}\left[\frac{(d_4')^2}{4}+\frac{\beta}{27\sigma'}\right]>0,$

Comparing (51) and (54), we find that

(55) $d_4'=\sqrt{(d_2')^2-0,15\beta/\sigma'}.$

Stiffness k_1 and k_2 wedge springs 1.3 and 4 can generally be different, but in order to simplify design calculations, it is advisable to assign $k_1=k_2=0,5k_{12}$. Under such a condition from formula (45) we obtain dependences for determining the number of working rings of the inclined springs1.3 i 4:

(56) $n_1=2,07E_{ПП}a_1/(k_{12}R_1^3);$

(57) $n_2=a_1R_1^2n_1/(a_1R_2^2).$

Height b_k (see. Fig. 3) sealing the first degree chamfer sealing PPG similar to the valve internal combustion engines that operate in these conditions can be recommended within $b_k=2...4$ mm [13]. It is desirable to simplify the process of rubbing $b_k\le 2...2,5$ mm. The angle of the cone α_k saddle 2 (see Fig. 3), in order to prevent the jamming of the first degree of sealing PPG in saddle 2, is assigned within $\alpha_k=60^\circ...90^\circ$[8, 9]. Assuming the largest diameter of the conical chamfer b_k saddle 2 2 is equal d_1 and choosing $\alpha_k=60^\circ$, for an obvious dependence (see Fig. 3) we calculate the internal diameter $d_{П1}$ saddles 2:

(58) $d_{П1}=d_1-2b_k\tan(\alpha_k/2)=d_1-2b_k\tan 30°=d_1-1,15b_k.$

Height δ_c butter saddle 2 can be determined by the condition of the strength of the broom on the cut at the time of passing the saddle 2 distances h_c and the tightening of its bout at the end of the hole of the device body (see Fig. 1).

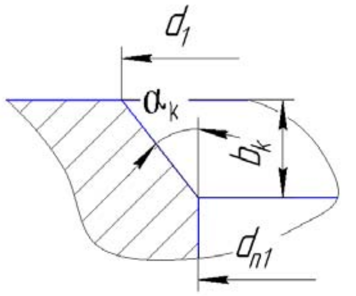

Fig.3. The shape of the sealing chamfer saddle device 2

In order to ensure the guaranteed strength of the saddle buckle 2 on the cut and due to the impact of the impact of the edges of the bout and the holes of the device body during their contact, force $F_{ск}$, which acts on the port saddle 2, we find the formula:

(59) $F_{ск}=k_д p_{1_{max}}\pi(d_2^2\text{-}d_{П1}^2)/4=1,57p_{1_{max}}(d_2^2\text{-}d_{П1}^2),$

where $k_д=2$ [11] – dynamic coefficient during shock interaction of the ends of the saddle bout 2 and the device casing.

The strength of the drum saddle 2 is cut:

(60) $\tau_з=F_{ск}/\pi d_2\delta_c\le[\tau_з],$

where

(61) $\delta_c\ge F_{ск}/\pi d_2[\tau_з]=0,5p_{1_{max}}(d_2-d_П^2/d_2)[\tau_з]^{-1},$

where $\tau_з$, $[\tau_з]$ – respectively, the design stresses of the cut in the transverse section of the saddle boot 2 and the permissible shear stress for the saddle material 2.

The researches of hydraulic drives [7] have established that the maximum allowable speeds of the energy carrier in the pressure hydraulic lines $[V_H]=6,5...9$ m/s for pressure $10\text{ MPa}\le p_r<15\text{ MPa}$ and drainage. $[V_{зл}]=2$ m/s.

If the device for deformation strengthening is connected to the hydro pump station using flexible hoses of high pressure, then, in terms of ease of operation of the device, it is advisable to discharge the pressure and discharge lines using the sleeves of one conventional passage, which is determined by the average maximum permissible speed of the energy carrier

(62) $[V_{НП}]=0,5([V_H]+[V_{зл}])=0,5(7,75+2)=4,88\approx 5\text{m/s},$

where $\left[V_H\right]_m = 7,75$ m/s – average permissible velocity of energy in the pressure line for $p_r = 10\text{MPa}$.

Conditional passage d_y the sleeves of high pressure of pressure and drainage hydroline are found by a simple formula [8, 21, 22]:

(63) $d_y = \sqrt{4\bar{Q}_{НГ} / \pi\left[V_{НП}\right]} \cong 0,5\sqrt{\bar{Q}_{НГ}}$,

where $\bar{Q}_{НГ} = 0,5(Q_H + \bar{Q}_Г) = 0,5Q_H(1+\tau^{-1}_{H3})$, m^3/s is the average energy consumption in pressure and drainage hydrolynamics of the device's hydrosystem.

The other (non-main) geometric dimensions of the device are in the design process according to generally accepted rules for the construction of the hydro equipment [11, 13], taking into account the peculiarities of the design of the hydropulse devices and drives [8, 9, 23].

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