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METHOD OF DESIGN CALCULATION OF A PARAMETRIC SINGLE-STAGE PRESSURE PULSE GENERATOR WITH ADJUSTABLE PRESSURE "CLOSING"

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On the basis of a qualitative analysis of the structure and principle of operation of a parametric single-stage pressure pulse generator with adjustable pressure "closing" p_2, the design of a prototype of the generator was developed. The article describes the principle of operation of the developed structure and describes the design of structural elements and their interaction with other structural elements from the justification. The design calculation methodology is developed on the basis of generally accepted engineering and scientific principles and experience in the design and operation of equipment based on pneumatic and hydropulse drive of vibration and vibration shock machines. The paper presents a sample of initial data that is necessary for the development of a design calculation methodology for a parametric single-stage pressure pulse generator with adjustable "closing" pressure p_2.

Based on the analysis of the design of the prototype of the generator, the accepted initial (input) data and the experience of designing a hydropulse drive and pressure pulse generators given in the literature sources, a method of design calculation of a parametric single-stage pressure pulse generator with adjustable pressure "closing" p_2 has been developed and its main energy, power and geometric parameters of the design of the prototype of both a hydropulse drive and a generator have been determined pressure pulses. The presented mathematical dependencies allow you to change the parameters of the pressure pulse generator to the corresponding technical and economic parameters of the technological process.

The paper also substantiates the use of the concept of "cyclic pressure pulse coefficient", the exact value of which can be found only based on the results of experimental studies of a prototype of a pressure pulse generator. For the design calculation, based on the recommendations of the researchers of the hydropulse drive, a margin factor within 10 ... 25 percent of the face value.

Keywords design calculation methodology; pressure pulse generator; sealing; hydraulic link; hydraulic pulse drive; press; drive; feed; pressure; stroke.

Introduction

The analysis of the results of experimental studies of the HPD [1 – 7] shows that in the well-known designs of pressure pulse generators (PPG), the pressure of the energy carrier p_1 "opening", and the pressure p_2 "closing" is clearly regulated , due to the inertia of the flow of the energy carrier and excessive movement of the shut-off and control element (no limitation of its working stroke h) PPG fluctuates within relatively wide limits, usually in the direction of decreasing from the calculated value. This nature of pressure change p_2 affects the stability of the frequency v of pressure pulses and their amplitude $\Delta r = p_1 - p_2$, and this worsens the quality of the technical characteristics of the PPG and the HPD as a whole.

The described problem is to some extent solved in the design of a parametric single-stage PPG, for example, according to the scheme of connection "combined" to the actuators of the hydropulse drive (HPD). The structural diagram of this PPG, for example, as a servo drive of a two-stage PPG, is shown in Fig. 1 [8]. The pressure regulator p_2 "opening" consists of two concentrically mounted springs *4* and *5*, placed in the boring of the regulator cup (the glass is conditionally not shown), while the spring *4* through the plunger *3* loads the shut-off and control element *1* of the PPG. Plunger *3* is located in the axial boring of the movable stepped sleeve *2*, pressed in its original position through the shoulder to the PPG body (conditionally not shown) by a spring *5*. Preliminary deformation of the spring *4* (y_{01}) and *5* (y'_{01}) is adjusted, respectively, by screws 6 and 7. Screw 6 is placed in the axial threaded hole of screw 7. Structurally, the assembly of the regulator of energy carrier pressure levels p_1 and p_2 (parts 2 – 7, the regulator housing is conditionally not shown) is designed as a separate assembly unit [8].

The dimensional chain of the axial assembly of SRE *1* and bushing *2* is calculated in such a way that in the initial position, the gap between the right (according to the drawing) end of SRE *1* and the left end of the bushing *2* is equal to the positive overlap h_{sp} of the spool part of the SRE *1*. The stroke of the bushing is equal to the negative overlap h_{sv} SRE *1*.

The main frontal section of the structural scheme without overall dimensions and dimensions of the fittings of the conjugations of a parametric single-stage PPG with adjustable pressure "closing" p_2 in the butt design, for example, for connecting to the distribution parallelepiped [2, 3] of the flow of the energy carrier HPD (there may be a so-called "pipe" [2, 3] version), is shown in Fig. 2 . The design of the PPG is built from three assembly units (blocks) - distribution of the energy carrier, throttle for adjusting the landing mode of the shut-off and control element (SRE) and the regulator of the pressure levels "opening" p_1 and "closing" p_2 of the PPG.

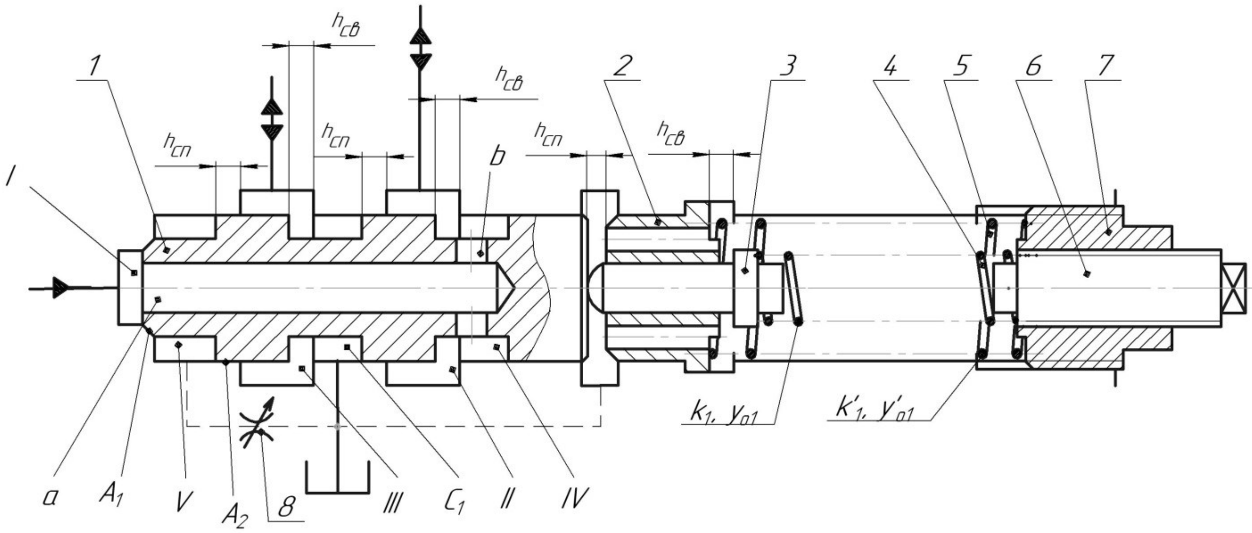

Fig. 1. Structural diagram of a parametric single-stage PPG with pressure regulation p_2 "closure"

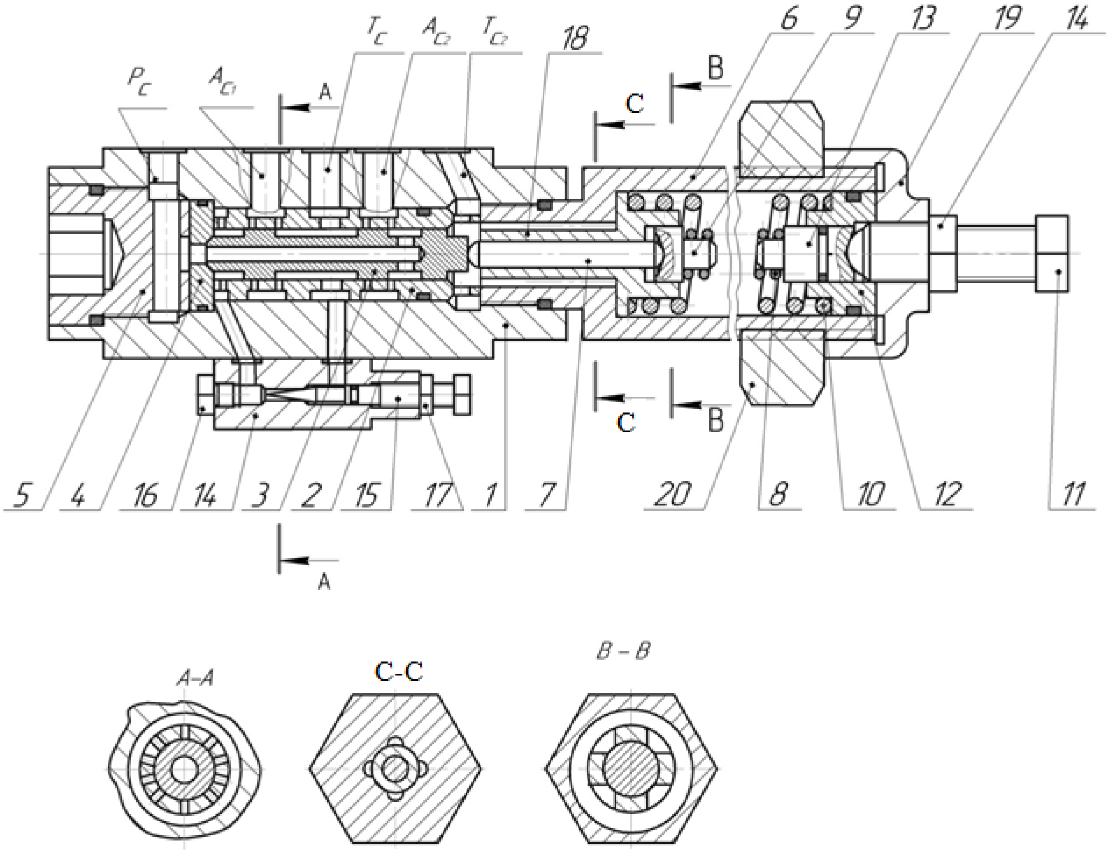

Fig. 2. Main frontal section of the structural diagram of a parametric single-stage PPG with adjustable "closing" pressure *p2*

The outlets A_{c1} and A_{c2}, respectively, can be connected according to the scheme "at the inlet" with the cyclic accumulator and the first plunger hydraulic cylinder, and in the case of using a piston hydraulic motor in VM (or VIM), with its piston cavity, and according to the scheme "at the outlet" with the second plunger hydraulic cylinder, or in the case of using a piston hydraulic motor as an actuator link of the HPD, with the rod cavity of this hydraulic motor (hydraulic cylinder). In case of use developed by PPG as a servo drive of a two-stage PPG, the outlets A_{c1} and A_{c2} are connected to the control cavities of the second stage according to the corresponding schemes [3]. In sleeve *2* there is a valve-spool shaped SRE 3. The spool part of the shut-off element *3* has four working edges, through which a row (*no*) of radial holes of small (calculated) diameter $d_{oz} = h_{sv}$, located in a strictly radial plane (see section *A - A* in Fig. 2) in sleeve *2* positive h_{sp} and negative h_{sv} overlap SRE *3*. Number of no holes, for example (*Ø2,0... 4.0*) *mm*, determined by the condition of providing the required passage area of a fully open spool slot SRE *3*, $A_{sh} = \pi \cdot d_2 \cdot h_{sv} = = \pi \cdot d^2_{oz} \cdot n_o / 4 \approx 0.785 \cdot d^2_{oz} \cdot n_o = = 0.785 \cdot h_{2sv} \cdot n_o$). The valve part (conical chamfer with a cone angle of $60° \pm 30'$, the average diameter of which corresponds to the cross-sectional area A'_1 (here $A'_1 \equiv f_1$, see Fig. 1) SRE *3* is in contact with the seat *4* installed in the boring of the body *1*, coaxial with the boring for the sleeve *2*, and fixed with a threaded plug *5* with a central blind hole connected by a through radial hole in the plug *5* with a pressure hydraulic channel *Rs for* supplying the energy carrier to the PPG.

With the threaded part of the cup *6* of the regulator of the pressure levels "opening" p_1 and "closing" *p2, the* PPG sleeve *2* is pressed against the seat *4*, and the SRE *3* is loaded with a spring *8* through the pressure plunger *7*, the ends of which rest on supports *9*, which are in contact with the spherical ends of the pressure plunger *7* and a ball rolled into the end of the adjusting plunger *13*, located in the tubular piston *12*, which, according to the running fit, is mated to the surface of the central axial boring in the cup *6*. The tubular piston *12* is the right (according to the drawing, see Fig. 2) spring support *10* of the pressure regulator p_2 "closing" the PPG. The left end (according to the drawing, see Fig. 2) of the spring *10* rests on the flange of the stepped sleeve *17*, which, according to the running fit, is mated to the surface of the through axial hole in the part of the cup *6*, which is screwed into the body *1*. In the central axial opening of the stepped sleeve *17* according to the running fit, a pressure plunger 7 is installed. Between the left end of the SRE *3* and the right end of the stepped sleeve *17* (according to the drawing, see Fig. 2) in the initial position, a gap h_{SV} is provided, equal to the positive overlap of SRE *3*, after which on SRE *3*, in addition to the action of the spring *force 8* on it, another force is added from spring *10* - the pressure regulator p_2 "close" the PPG. Springs *8* and *10*, so that in the event of a breakdown of one of the springs, there is no jamming with debris that can get stuck between the turns of the other spring, wound in different directions; One has the direction of the right helical line, and the second has the direction of the left. In addition, this method of manufacturing to some extent compensates for possible skewed lateral forces from springs *8* and *10*. The pre-deformation *y01* (see Fig. 1) of the spring *8* – the pressure regulator *p1* of the "opening" of the PPG, is changed by means of an adjusting screw 11 screwed into the tubular screw *16*, which is connected to the cup 6 by a thread on its outer surface. The tubular screw *16* changes the pre-deformation y'_{01} through the tubular piston 12 (see Fig. 1) of the spring *10* - the pressure regulator p_2 "closes" the PPG. The position of the screw 11 and the tubular screw *16* is fixed, respectively, with lock nuts 14 and 15.The axial holes *"a"* in the cup *6* connect the boring of springs 8 and 10 with the intermediate drain cavity of the PPG, which with the hole T_{c2} is connected to the drain cavity of the hydroelectric power plant for power supply with the energy carrier HPD VM or VIM.

The throttle for adjusting the landing mode SRE *3*, during its reverse stroke, consists of a choke body *18* docked to the body *1 of* the PPG, a needle *19*, which has a conical (needle), cylindrical and threaded parts, and a threaded plug *20*. The central axial hole of the body *18* of the throttle has a stepped cylindrical shape, the smaller diameter of which is covered by a needle *19*, is connected by two radial hydraulic channels to the corresponding bores of the body *1*: with the intermediate cavity *B2* SRE *3* PPG and the drain hole T_c (see Fig. 1). The threaded hole, closed with a plug *20*, is designed to install a sensor for registering changes in pressure in the intermediate cavity SRE *3*. The position of the needle *19* is fixed with a lock nut *21*.

Research results

To develop a method for the design calculation of a parametric single-stage PPG with adjustable pressure "closing" p_2 , the basic initial data should be enough to calculate all the energy, power, kinematic and geometric parameters of HPD and PPG, which allow developing the design of a prototype (DP) of both HPD and PPG. The experience of design, experimental research and production of a prototype of HPD and PPG shows, basically, these are the following data [6, 9 -14]:

– type of hydraulic pump of the pumping (or pumping and accumulatory) power station of the HPD. Based on the experience of operation of the HPD [2 – 6], hydraulic pumps of the NSH type have proven themselves best in the HPD, since they do not have suction and discharge valves that quickly fail when operating under conditions of pulsating pressure with a significant amplitude [2 – 6]. Supply Q_n hydraulic pump in the methods of design calculation of the HPD of technological VM and VIM is the calculated value, so a specific type of hydraulic pump is selected after determining Q_n;

– the nominal pressure of the "opening" of the PPG, as a rule, is prescribed $p_1 = 10\ MPa$ (in necessary cases, $p_1 > 10$ MPa can be prescribed, for example, when it is necessary to develop a compact mechanism, since modern pumps of the NSH type can operate with an energy pressure of up to *16 MPa*);

– the frequency control range of pressure pulses ν is determined by the supply *Qn* of the hydraulic pump, the initial total volume $W_{0\Sigma}$ of the pressure cavity *A* of the hydraulic system HPD (this includes the volume *Wa* of the cyclic accumulator, if it is present in the hydraulic system of the drive), the internal gear ratio *u21* PPG and its conditional passage. Approximately, we assign $\nu = (10... 100)\ Hz$ ($\nu_{max} = 100\ Hz$);

– approximately assign the preliminary internal gear ratio of PPG $u_{21} = 0.06 ... 0.12$ then by $\theta = 1$ and $\xi = 1$, the pressure of the "closing" of the PPG will be regulated in the range of $p_2 \approx (0.49 ... 0.69)\ p_1$;

– approximate ranges for adjusting the preliminary deformation of springs: 8 – $y_{01}=(5.0... 18.0)\cdot 10^{-3}\ m$; 10 – $y'_{01} = (5.0... 18.0)\cdot 10^{-3}\ m$ (see Fig. 1);

– it is recommended to choose the grade of working fluid (energy carrier) for GL HPD and PPG from a number of mineral oils used for hydraulic pumps of the NSH type. For gear pumps, it is recommended to use high-quality hydraulic oils with anti-wear additives (class HLP/DIN 51524-2 or HVLP/DIN 51524-3) and viscosity corresponding to the operating temperature (most often ISO VG 32, 46, 68). They provide wear protection, oxidative stability and reliable operation at high pressures;

– average modulus of elasticity of the energy carrier $\kappa = 1.45\cdot 10^3\ MPa$;

– approximate values of moving consolidated masses m_3 (SRE *3*), parts 7 (m_7), 9 (m_7), 17 (m_{17}) and springs 8 and 10 (~ 0.3 m pr is taken into account);

– permissible flow rates of energy carrier in pressure $[v_p]$ and drain $[v_{zl}]$ hydraulic lines of the hydraulic system HPD and PPG, which are assigned according to the recommendations [15];

– grades of structural steels of the main parts of PPG (see Fig. 2.2): hull *1* – steel *45*, or *40X*, *34... 42 HRCe* (hardening in lubricant, tempering); SRE *3* – steel *ШХ15*, *62... 64 HRCe* (quenching in lubricant, tempering); sleeve *2* - steel *ШХ15, 58... 60 HRCe* (hardening in lubricant, tempering); saddle *4* - *steel ШХ15, 58... 60 HRCe* (quenching in lubricant, tempering); springs *8* and *10 – steel 60C2VA, 53... 57 HRCe* (quenching in lubricant, tempering). The materials of other parts of the PPG and the type of their heat treatment during the development of its PD are assigned based on the results of a detailed analysis of their service purpose and working conditions;

– methods of organizing communications between the links of the HPD and PPG and the hydraulic pumping (hydraulic pump-accumulator) power supply station of the PPG are selected by the designer in accordance with the conditions of specific technological processes that are implemented with the help of the HPD of a specific VM or VIM;

– approximate initial total volume $W_{0\Sigma}$ of the pressure cavity *A* of the hydraulic system of the HPD and the volume *Wa* of the cyclic hydraulic accumulator, if it is available in the hydraulic system of the drive, respectively: $W_{0\Sigma} = 1\cdot 10^{-3}\ m$; $W_a \approx 0.5\cdot 10^{-3}\ m$;

– the qualities of accuracy of conjugations and details of PPG are assigned by the designer during the development of the PPG PD based on the results of a detailed analysis of their service purpose, physical principles of functioning and working conditions.

If necessary, in addition to the above data, additional clarifying data can be added to this list, and it is also necessary to take into account the acquired experience in the design and operation of VMs and VIM with a hydropulse drive [1 – 14] and technical indicators of the purpose.

The QH supply of the hydraulic pump is the main parameter, which, with fixed other parameters of the operating cycle of the drive (p_1, p_2, $W_{0\Sigma max} = W_{0\Sigma} + W_a$, κ, ν_{max}, η_{on} and A'_1), determines the upper limit of the frequency νmax of pressure pulses generated by the PPG.

Experimental studies of the HPD [2 – 6] have established that at the time of completion of the discharge of the energy carrier from the cyclic accumulator (conditionally not shown in Fig. 2) into the cavity of the first executive hydraulic motor (conditionally not shown in Fig. 2), the HPD through the hydraulic channel A_{c1} PPG (see Fig. 2), there is no pressure holding at the level *of* p_1. Hydraulic channel A_{c1} The PPG is connected according to the scheme "at the inlet" to the first executive hydraulic motor. The discharge of the cyclic accumulator begins immediately after the opening of the SRE *3* PPG, and the final level of the discharge pressure of the cyclic accumulator is equal to p_2 of the pressure of the "closing" of the PPG. The increase in pressure in the cyclic accumulator to *the level* p_1 occurs when the SRE is closed *3* PPG. can be evaluated by classical dependence [2, 3]:

$$t_n = p_1\cdot W_{0\Sigma max} / (\kappa\cdot Q_n). \quad (1)$$

During the same time t_n, *the* pressure of the energy carrier rises to the level p_1 in the cavity of the second actuator hydraulic motor (conditionally not shown in Fig. 2) the HPD, which is connected according to the scheme "at the output" to the PPG through the hydraulic channel A_{c2} (see Fig. 2). It should be noted that the dependence (1) characterizes the first pressure pulse of the energy carrier after switching on the PPG without taking into account transients. For subsequent pressure pulses, When the oscillatory process of change in the pressure of the energy carrier stabilizes, it approximately occurs after the second pressure pulse [3 – 12], to calculate the time *TN*, the formula should be used:

$$t_n = \Delta p \cdot W_{0\Sigma} / (\kappa \cdot Q_n), \tag{2}$$

At the moment of opening the SRE *3* PPG, the process of charging the cycle accumulator and the working stroke of the moving link of the second actuator hydraulic motor (plunger or piston of the hydraulic cylinder) continues until the SRE 3 is completely switched. Based on these considerations, the total charging time *of* the cycle accumulator is equal to the time $t_{zca} = t_{rhgd2}$ of the full working stroke of the moving link of the second executive hydraulic motor, can be given as the sum

$$t_{zca} = t_{rhgd2} = t + t_{pg}, \tag{3}$$

where t_{pg} is the switching time (forward travel) of SRE *3* PPG. The full period *Tt of* pressure fluctuations in the pressure cavity *A1* (see Fig. 1) can be represented as a dependence

$$T_t = t_{zca} + t_{rca} + t_{wit} = t_n + t_{pg} + t_{rca} + t_{wit} = t_n \cdot [1 + (t_{pg} + t_{rca} + t_{wit})] / t_n = K_{ct} \cdot t_n = v^{-1}, \tag{4}$$

where $K_{ct} = [1 + (t_{pg} + t_{rca} + t_{wit})] / tn$ is the cyclic coefficient of pressure pulses generated by the HIT; $t_{rca} = t_{zhgd2}$ *is the* total discharge time of the cyclic accumulator, equal to the time t_{zhgd2} of the full reverse stroke of the moving link of the second actuating hydraulic motor; T_{weet} is the time of holding the pressure of the energy carrier at the level of p_2 after the completion of the reverse stroke of the SRE *3*. In the absence of pressure holding at the level *of* p_1, the duration of pressure pulses in the cyclic accumulator and the cavity under the moving link of the second actuator hydraulic motor

$$t = t + t_{Rca} = t + t + t_{zhgd2}. \tag{5}$$

The throughput of the PPG is determined by the cross-sectional area A'_1 of the first stage of PPG sealing (see Fig. 1 and Fig. 2). The diameter $d_1 = (4(A'_1 / \pi)^{0.5} \approx 1.13 \cdot (A'_1)^{0.5}$, if it is counted relative to the average diameter of the ground chamfer of the valve part of the SRE 3 (see Fig. 1 and Fig. 2), is actually the diameter *du* of the conditional passage of the PPG, for the determination of which it is necessary to know the supply of the hydraulic pump Q_n pumping and accumulator power station of the HPD. The theoretical value of the supply *of* Q_{nT} can be calculated according to the formula known for the HPD [3]

$$Q_{nT} = K_{ct} \cdot v_{max} \cdot p_{1max} \cdot W_{0\Sigma max} \cdot \kappa^{-1} \cdot \eta^{-1}_{oh}, \tag{6}$$

where K_{ct} is the cyclic coefficient of pressure pulses; ηon is the volumetric efficiency of the hydraulic pump (for hydraulic pumps of the NSH type $\eta_{on} = 0.95 ... 0{,}96$ [12]). The exact value of K_{ct} can be found only based on the results of experimental studies of the State Land Inspection Center, so it is advisable to enter into the formula (6) the reserve factor K_{zv}, the value of which can be specified based on the results of experimental studies of HPD and PPG. It is recommended that [3] take $K_{sv} = 1.10... 1.25$ then the dependence (6) will take the form:

$$Q_n = K_{sv} \cdot K_{ct} \cdot v_{max} \cdot p_{1max} \cdot W_{0\Sigma max} \cdot \kappa^{-1} \cdot \eta^{-1}_{on} = (1.10... 1.25) \cdot K_{ct} \cdot v_{max} \cdot p_{1max} \cdot W_{0\Sigma max} \cdot \kappa^{-1} \cdot \eta^{-1}_{on}. \tag{7}$$

Guaranteed shock-free contact of the SRE *3* PPG with the saddle *4* during its reverse movement is provided under the condition of the oscillatory movement of the SRE *3* V in the resonant mode [3]:

$$\omega_{max} / \omega_{\Sigma 1} \geq \sqrt{2}, \tag{8}$$

where $\omega_{max} = 2\pi v_{max}$ is the circumferential frequency of oscillations SRE *3*, which should be taken equal to the frequency of pressure pulses $\omega_{\Sigma 1} = \sqrt{(k_{\Sigma 1}/m_{1\Sigma})}$ is the natural circumferential frequency of the SRE *system 3* – GL HPD; $m_{1\Sigma} = m_{SRE} + \rho_e \cdot W_{0\Sigma max}$ *is* the total aggregate mass of SRE *3,* taking into account the mass of the energy carrier in the volume $W_{0\Sigma max}$ (here $m_{SRE} = m_3 + m_7 + m_{17} + 0.3(m_8 + m_{10})$ is the total aggregate mass of SRE *3*, which consists, respectively, of the masses of the parts (see Fig. 2) 3, 7, 17, 8 and 10 and ρe *is* the density of the energy carrier); $k_{\Sigma 1} = k_1 + A^2{}_2 \cdot \kappa / W_{0\Sigma max}$ is the summary total stiffness of the elastic link of the PPG HPD VM (or VIM), taking into account the rigidity of the GL relative to the area *A2* of the cross-sectional section of the spool part of the SRE *3* PPG (see Fig. 1 and Fig. 2).

According to condition (8), the stiffness k_1 *of* the spring 8 of the pressure regulator *p1* "opening" of the PPG must satisfy the inequality

$$k_1 \leq 2\pi^2\, m_{1\Sigma} \cdot v^2_{max} - (A^2{}_2 \cdot \kappa / W_{0\Sigma max}). \tag{9}$$

At the given nominal pressure p_1 "opening", the range of change in the pre-strain [y_{01min}, y_{01max}] of the spring *8* of the pressure regulator p_1 and the internal gear ratio u_{21} PPG [8]:

$$p_2 \le p_1 \cdot A'_1/A_2 + (k_1 \cdot h_c + k'_1 \cdot h_{St}) / A_2 + k'_1 \cdot y'_{01} / A_2, \qquad (10)$$

$$p_2 = p_1 \cdot u_{21}^{0.5} \cdot (1 + \theta_1 \cdot \xi) + k_1 \cdot h_c \cdot A^{-1}{}_2 \cdot (1 + \theta_1 \cdot h_c / h_{sv}). \qquad (11)$$

According to the formulas (10) and (11), neglecting the sum of its terms $(k_1 \cdot h_c + k'_1 \cdot h_{sv}) / A_2 + k'_1 \cdot y'_{01} / A_2$, the rigidity *of* k_1 can be estimated by the inequality

$$k_1 \le u_{21}^{0.5} \cdot (1 + \theta_1 \cdot \xi) = u_{21}^{0.5} \cdot p_1 \cdot A_2 \cdot y^{-1}{}_{01m}, \qquad (12)$$

where $y_{01m} = 0,5\ (y_{01min} + y_{01max})$ is the average value of the preliminary deformation of the spring *8* of the pressure regulator, corresponding to the nominal pressure *p1* of the "opening" of the PPG.

Equating (9) and (12) from the resulting quadratic equation, taking the positive root (the negative root has no physical content), we find the dependence for calculating the area A_2 of the cross-section and the diameter d_2 of the spool part of SRE *3*:

$$A_2 = 0,5 \cdot u_{21}^{0,5} \cdot p_1 \cdot W_{0\Sigma max}\{[1 + 8\pi^2 \cdot v^2{}_{max} \cdot \kappa \cdot y^2{}_{01} / (u_{21} \cdot p^2{}_1\, W_{0\Sigma max})]^{0,5} - 1\}\ (y_{01m} \cdot \kappa)^{-1}; \qquad (13)$$

$$d_2 = (4(A_2 / \pi)^{0,5} \approx 1,13 \cdot (A_2)^{0,5}. \qquad (14)$$

The average cross-sectional area A'_1 along the chamfer of the valve part of SRE *3* (the first stage of PPG sealing) is calculated according to the given gear ratio u_{21} based on the formula (10):

$$A'_1 = u_{21}^{0.5} \cdot (1 + \theta_1 \cdot \xi) \cdot A_2. \qquad (15)$$

Theoretical and experimental studies of HPD and PPG [2 – 6] have established that the average Q_{mg} of energy consumption through the open gap of PPG $A_{sh} = \pi \cdot d_2 \cdot h_{sv}$ (see Fig. 1) is associated with the supply of the hydraulic pump Q_n to the pumping and accumulator power station HPD by dependence

$$Q_{mg} = (Q_n + Q_{am}) \cdot t_n / t\Delta p = (Q_н + Q_{am}) \cdot \tau_{\Delta p}, \qquad (16)$$

where Q_{am} is the average supply of the cyclic accumulator; $\tau_{\Delta p} = tн / t_{\Delta p}$ is the relative time of reducing the pressure of the energy carrier in the cyclic accumulator and in the second actuator hydraulic motor of the HPD from level p_1 to level p_2, and $t_{\Delta p} < t_n$ ($\tau_{\Delta p} > 1$), from which it follows that $Q_{mg} > (Q_n + Q_{am})$; $t_{\Delta p} = t_{rca} = t_{zhgd}$ is the time of reducing the pressure of the energy carrier from the level *of* p_1 to the level *of* p_2.

In order to avoid negative phenomena during the operation of the PPG, such as cavitation, etc., and according to the rules for designing hydraulic drives [15 – 16], the average speed vm_g of the energy carrier through the open gap A_{sh} PPG should not exceed the permissible $[v_r]$ (due to the impulse nature of the energy carrier flow, it is possible to assume $[vp]_{max} = 15$ *m/s* [16]).

$$v_{mg} = Q_{mg} / (\pi \cdot d2 \cdot h_{sv}) \le [v_p] \qquad (17)$$

from where

$$Qm_g \le \pi \cdot d2 \cdot h_{sv} \cdot [vr]. \qquad (18)$$

The negative overlap h_{sv} of the spool part of the SRE *3 – no* of small radial holes d_{oz} in the sleeve *2*, in order to reduce energy consumption during the opening of the HIT due to the flow of the working fluid from the pressure cavity to the drain cavity, is assigned [2 – 6] less than the positive overlap h_{sp} by ~ *(0.5...1.0) mm* depending on the quality of accuracy of the conjugation of the spool part of the SRE *3* with its sleeve *2*. Usually, the conjugation of surfaces of the type SRE *3* – sleeve *2* is performed according to the qualities of at least *5 – 6*, according to the running fit, for example, $\emptyset d_2\ H6 / g6$, which allows to ensure reliable sealing with a positive overlap $h_{sp} = (2.5.....4.5)$ *mm* [3 – 6].

For the developed PPG, it was assumed *that* $d_{oz} = h_{sv}$, ($A_{sh} = \pi \cdot d_2 \cdot h_{sv} = \pi \cdot d^2{}_{oz} \cdot n_o / 4 \approx 0.785 \cdot d^2{}_{oz} \cdot n_o = 0.785 \cdot h^2{}_{sv} \cdot n_o$, see Fig. [8], that at $h_{sv} = 4.0$ *mm* allows us to assign $h_{sp} = 4.5$ *mm* and the stroke of the SRE *3* $h = h_{sp} + h_{sv} = 4.5 + 4.0 = 8.5$ *mm*. With these data and with the dependence for A_{sh}, we find a simple formula for calculating the required number of no radial holes in sleeve *2*:

$$no = 4 \cdot d_2 / h_{sv}. \qquad (19)$$

The total maximum flow rate $Q_{гmax} = (Q_{нmax} + Q_{amax})$ of the energy carrier passing through the area A_{sh} of a fully open gap SRE *3* PPG can be estimated by a formula similar to (19):

$$Q_{гmax} = (Q_{нmax} + Q_{mamax}) \cdot \tau_{\Delta p} = \pi \cdot d_2 \cdot h_{sv} \cdot [v_p]_{max}, \qquad (20)$$

where experimental studies of the State Land Registration Committee of the State Inspectorate of Industrial Engineering and Labor Statistics [2 – 6] established that $Q_{mamax} = (2... 3)\ Q_{нmax}$ is the supply of energy carrier by a cyclic hydraulic accumulator to the cavity of the executive hydraulic motor during the time t_{rc} of discharge of the accumulator, which corresponds to the maximum possible supply of the hydraulic pump; the relative time $\tau_{\Delta p} = t_н / t_{\Delta p} > 1$ decrease in energy carrier pressure from the level p_1 to the level *of* p_2 can be estimated by the following considerations: 1) with an increase in $Q_{нmax}$ with the scheme of connection of the PPG to the hydraulic motor "at the output", the time *tn* decreases (see Fig. (1); 2) the time $t_{\Delta p}$ in any scheme of connection of the PPG is mainly determined by the speed v_{mg} of the flow of energy carrier through the area $A_{sh};$ 3) it can be assumed that $\tau_{\Delta p} \approx \approx [v_r]_{max} / [v_r]_{nom} = 15 / (5... 8) \approx 2... 3$ (here $[v_r]_{nom} = (5... 8)\ m/s$ is the nominal permissible flow rate of the energy carrier in pressure hydraulic lines at pressure *(10... 12) MPa* [15 – 16].

Based on the dependence (21) and the formula (20) and the above considerations, after simple algebraic transformations, we obtain a formula for calculating the maximum supply Q_{hmax} of a hydraulic pump, corresponding to the maximum allowable speed $[v_p]_{max}$ of the energy carrier:

$$Q_{нmax} = (1.04... 2.08)\cdot d^2{}_2{}^{\cdot[}v_p]_{max}\cdot n^{-1}{}_o. \tag{22}$$

The found value of $Q_{нmax}$ should be compared with the calculated value of $Q_н$ according to the formula (9) and according to the average value of $Q_{нm} = 0.5\cdot(Q_{нmax} + Q_н)$, select a specific model of a hydraulic pump of the NSH type.

Guided by the established recommendations [8], the stiffness of the spring 10 (see Fig. 2) we calculate $k'_1 = \theta_1\cdot k_1 = 0.8\cdot k_1$, where $\theta_1 = 0.8$. The preliminary strain $y'_{01} = \xi\cdot y_{01} = 1.05\cdot y_{01}$ *of* spring 10 is determined according to the initial data and the formula ($y'_{01} = \xi\, y_{01}$), taking $\xi = 1.05$.

The height b_s of the ground chamfer of the valve part (first stage of sealing) SRE *3* PPG, at which the required wear resistance of this part of SRE 3 is ensured, is found according to the formula given in the paper [3], having previously determined the average driving force $F_{rcm} = 0.5\cdot\{\ k_1\cdot[h + \theta 1\cdot h_{sp} + y_{01}(1 + \theta_1\cdot\xi)] - p_2\cdot A_2\}$ of the movement of SRE *3* on its reverse course:

$$b_s \geq 1.38\cdot d_1\cdot -1\sqrt{[F_{pcm}/(d_1^2\cdot\sigma_{\text{пр}})] + 1}, \tag{23}$$

where σ_{pr} is the elastic limit of the saddle material *4* PPG. The formula (23) is valid for the angles of the cones $\alpha = 60^o$ of the valve part of the SRE *3* and the seat *4*.

The parameters of coiled springs 8 and 10 are calculated according to the standard methodology given in DSTU 13764-68, DSTU 23776-68), and all other geometric dimensions of the PPG are determined by the designer during the development of the PPG PD according to the generally accepted rules for the design of hydraulic and, in particular, hydropulse machines, mechanisms and devices [1 – 16].

After the development of the *DP* PPG, it is also necessary to check for strength (for cutting and creasing) the cutting of plugs *5* and *20* (see Fig. 2), which are subjected to cyclic load due to the action of pulsating high pressure on these parts with an amplitude $\Delta p = p_1 - p_2$. According to the accepted input data, the material of case *1* is steel *45*, or *40X*, *34 ... 42 HRCe* (hardening in lubricant, tempering). From the same material and heat treatment, it is advisable to make a body *18* of the throttle for adjusting the landing mode SRE *3*, and for plugs *5* and *20*, use steel *45*, *240 ... 250 HB* (improvement, tempering). Since, in accordance with the intended type of heat treatment, the materials of cases *1* and *18* are more durable, the calculation of the cutting stresses τ_{zr} and crushing σ_{zm} should be performed for cutting plugs *5* and *20* according to the formulas given in the textbooks from the course "Machine Parts" [17, 18]:

$$\tau_{zr} = F_{japmax} / (\pi\cdot d_j\cdot k_{pr\cdot Hjp}\cdot k_{mp}) \leq [\tau_{zr}]; \tag{24}$$

$$\sigma_{zm} = 4\cdot F_{japmax} / [\pi\cdot(d^2{}_j - D^2{}_{1k})\cdot z_{jp}] \leq [\sigma_{zm}], \tag{25}$$

where $F_{japmax} = p_1 A_{jp}$ is the maximum axial force acting on the *area* A_{jp} of the cross-section of plugs *5* and *20* (j = *5*; *20*); D_j – outer diameter of cork cutting *5* and *20*; k_{pr} is the coefficient of completeness of the slicing ($k_{pr} \approx 0.87$ – for metric triangular slicing); H_{jp} is the cutting height of corks *5* and *20*; $k_{mp} \approx 0.55$ is the coefficient of uneven distribution of the load along the turns of the slicing; D_{1k} is the inner diameter of the slicing in bodies *1* and *18* ($k = 1; 18$); $z_{jp} = H_{jp} / p_j$ is the number of turns on the twisting length of the cut (cutting length) of corks *5* and *20*; *pj* is the step of cutting corks *5* and *20* ($j = 5; 20$); $[\tau_{zr}] = 125\ MPa$, $[\sigma_{zm}] = 610\ MPa$ – respectively, the permissible shear and crease stresses for steel *45* under pulsating load mode [19, 20]. If necessary, other types of verification calculations can be performed, for example, calculation of the reliability of the PPG operation with the determination of the main reliability indicators [17, 18] – reliability, durability, maintainability and safety.

The main parameters of the PD of a parametric single-stage PPG with adjustable pressure "closing" $p2$, determined according to the described design calculation methodology, are given in Table 1.

Table 1

Main design parameters of the design of the prototype of the PPG

Parameter name	Unit Measurement	Numeric parameter value
Nominal (average) consumption Q_{nm} of energy carrier	m^3/s	$1{,}14 \cdot 10^{-3}$
Nominal pressure of the "opening" p_1	MPa	10
Frequency control range v of pressure pulses at nominal flow rate Q_{nm}	Hz	10...100
Maximum cross-sectional area of the throttle adjustment of the landing mode SRE *3*	m^2	$5 \cdot 10^{-8}$
Spring stiffness: *5* - pressure regulator "opening" p_1 - k_1	N/m	6,08·104
10 – pressure regulator "closing" $p_2 - k'_1$	N/m	4,86·104
Specified limits for adjusting the pre-deformation of springs: *8* – opening pressure regulator" – y_{01}	m	$(5...17) \cdot 10^{-3}$
10 – pressure regulator "closing" – y'_{01}	m	$(5...18) \cdot 10^{-3}$
The diameter of the conditional passage of the PPG *du is* equal to the diameter d_1 of the first stage of PPG sealing	m	$12 \cdot 10^{-3}$
Diameter d_2 of the second stage of PPG sealing	m	$18 \cdot 10^{-3}$
The refined internal gear ratio *of the PPG* u_{21} by $\theta = 0.8$ and $\xi = 1.05$	–	0,058
The nominal pressure of the "closing" p_2, determined at the specified values p_1, u_{21}, θ and ξ	MPa	4,42
Positive overlap of SRE *3* h_{sp}	m	$4{,}5 \cdot 10^{-3}$
Negative overlap of SRE *3* $h_{sv} = d_{oz}$	m	$4{,}0 \cdot 10^{-3}$
Full stroke of the SRE *3* $h = h_{sp} + h_{sv}$	m	$8{,}5 \cdot 10^{-3}$
Estimated number of radial holes in sleeve *2*	–	18
Summary total stiffness of the elastic link of PPG, HPD $k_{\Sigma 1} = k_1 + k_{GL} = k_1 + A^2{}_2 \cdot \kappa / W_{0\Sigma max}$	N/m	1,23·105
Consolidated total hardness of GL k_{GL} at $W_{0\Sigma max} = 1.5 \cdot 10^{-3}\ m^3$	N/m	6,25·104

Conclusions

The developed method for the design calculation of a parametric single-stage PPG with adjustable pressure "closing" p_2 allows you to determine all the main energy, power and geometric parameters of the generator and can be used for the design calculation of single-stage PPG of other types and schemes of connection to the actuator hydraulic motors (hydraulic cylinders) of the hydropulse drive of vibration or vibration shock machines.

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Методика проєктного розрахунку параметричного однокаскадного генератора імпульсів тиску з регульованим тиском «закриття»

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На основі якісного аналізу будови та принципу дії параметричного однокаскадного генератора імпульсів тиску з регульованим тиском «закриття» р2 розроблена конструкція дослідного зразка генератора. В статті описано принцип роботи розробленої конструкції та описано з обґрунтування конструктивне виконання елементів конструкції та їх взаємодія з іншими елементами конструкції. Методика проектного розрахунку розроблена на базі загальноприйнятих інженерно-наукових засад та досвіді проектування і експлуатації обладнання на базі пнемо- та гідроімпульсного приводу вібраційних та віброударних машин. В роботі представлений зразок початкових даних, що необхідний для розроблення методики проєктного розрахунку параметричного однокаскадного генератора імпульсів тиску з регульованим тиском «закриття» р2.

Грунтуючись на аналізі конструкції дослідного зразка генератора, прийнятих початкових (вхідних) даних і наведеному в літературних джерелах досвіді проєктуванні гідроімпульсного приводу та генераторів імпульсів тиску, розроблено методику проєктного розрахунку параметричного однокаскадного генератора імпульсів тиску з регульованим тиском «закриття» р2 та визначено його основні енергетичні, силові та геометричні параметри конструкції дослідного зразка як гідроімпульсного приводу так і генератора імпульсів тиску. Представлені математичні залежності дозволяють змінювати параметри генератора імпульсів тиску під відповідні техніко-економічні параметри технологічного процесу.

Також в роботі обґрунтовано використання поняття «циклового коефіцієнту імпульсу тиску», точне значення якого може бути знайдено тільки за результатами експериментальних досліджень дослідного зразка генератора імпульсів тиску. Для проектного розрахунку, ґрунтуючись на рекомендаціях дослідників гідроімпульсного приводу, використовується коефіцієнт запасу у межах 10…25 відсотків від номінального значення.

Ключові слова: методика проєктного розрахунку; генератор імпульсів тиску; ущільнення; гідравлічна ланка; гідроімпульсний привід; прес; привід; подача; тиск; удар.

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