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R. R. Obertykh[1]
A. V. Slabkyi[1]
V. V. Ishchenko[1]

HYDROIMPULSE DRIVE OF A VIBRATION PRESS WITH A TWO-STAGE FIVE-LINE PRESSURE IMPULSE GENERATOR WITH PARAMETRIC CONTROL OF ITS SECOND STAGE

[1] Vinnytsia National Technical University

Analysis of theoretical and experimental studies of the hydraulic pulse drive of technological vibration and vibration-impact machines (vibratory cutting and deformation strengthening of machine part surfaces, hydro-pulse vibrators, etc.) showed that the hydro-pulse drive allows for the implementation of various vibration technologies in a wide range of technical applications and with adjustable vibration loads on objects of technological influence. Based on the results of a theoretical study of existing designs of vibration press equipment based on a hydraulic pulse drive, a schematic and structural diagram of a hydraulic pulse vibration press with a two-stage five-line pressure pulse generator with parametric control of its second cascade and an executive link with a drive of the vibrating table of the vibropress from a piston hydraulic cylinder and a load of the object of technological influence of the hydraulic pulse drive from a plunger hydraulic cylinder, the body of which is installed on the architrave of the vibrating press, and the plunger is kinematically connected to the movable crossbeam of the vibrating press.

The article presents a description of the operation of a newly developed design of a hydraulic pulse vibrating press with a two-stage five-line pressure pulse generator with parametric control of its second stage, and provides the main calculation dependencies according to which the main geometric and force parameters of the developed drive design can be calculated. This design solution expands the technical and technological capabilities of the hydraulic pulse drive, as it provides a high-speed mode of vibration movement of the executive link – the vibration table of the vibration press in combination with the hydraulic load of the object of technological influence from an additional plunger hydraulic cylinder installed on the architrave of the vibration press, the plunger of which drives the movable crossbeam of the vibration press.

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

Introduction

Vibropressing of large-sized workpieces of considerable weight made of refractory powders, for example, reaction-sintered silicon carbide (SIC/Q2), is carried out only with the help of vibrating presses, which are technological vibration (VM) and vibration-impact (VIM) machines equipped with mechanical, pneumatic, hydraulic or hydropulse drives [1 – 4]. An analysis of the advantages and disadvantages of these drives has shown that vibration pressing technologies for large-sized products of considerable weight made of refractory powder materials are best implemented on VM or VIM with a hydraulic pulse drive (HPD), for example, on inertial vibration press hammers (IVPH). [1, 2, 4], which provide high working forces (up to 320 kN and more) and a wide range of vibration load parameter adjustment (frequency – 1...100 Hz, amplitude – (0,1...10) 10^{-3} м) on the executive link of the vibration press. HPD, the idea and basic principles of which were developed by I. B. Matveev [2], is simple and reliable in operation and has a relatively low metal content.

Nowadays, the development of new schemes and designs for hydraulic impulse engines, theoretical and experimental research into dynamic processes in hydraulic impulse engines, and the creation of scientifically based methods for the design calculation of machines and devices based on HPD continues and develops mainly through the work of the VNTU hydraulic impulse engine scientific school, founded by I. B. Matveev.

The central and most important component of the HPD is a parametric-type pressure pulse generator (PPG) for the working fluid (energy carrier). The PPG generates pressure pulses in the working (pressure) chamber of hydraulic motors (hydraulic cylinders) – the executive link of the VM or VIM, as a result of which this link is set into vibrational motion, which causes a force vibration load on the technological object of influence, located, for example, on a vibration table, etc VM or VIM. Thus, by adjusting the parameters of pressure pulses (their frequency ν and pressure amplitude $\Delta p = p_1 - p_2$ of the energy carrier (here, respectively, p_1 and p_2 are the pressures of the energy carrier during the ‘opening’ and ‘closing’ of the PPG), the PPG controls the vibration load mode of the technological object of influence of the VM or VIM [1 – 6].

The expansion of the technical and technological capabilities of VM and VIM, primarily vibropresses, with HPD is determined by the perfection and functionality of PPG. Powerful VM or VIM with HPD usually use two-stage parametric PPG [1, 5, 6], therefore, the development of HPD with new, improved two-stage PPG to control its operating cycle is a relevant scientific and engineering task.

Research results

The basic hydrokinematic diagram of the HPD is shown in Fig. 1. The HPD consists of structural blocks – mechanical systems: PPG 1; drive executive link 2; controlled check valve 3. Parametric PPG 1 is built according to a two-stage scheme and consists of a servo drive 1.1 – a three-line two-position parametric single-stage PPG. Servo drive 1.1 via hydraulic valve 1.2, fine adjustment unit PPG 1, constructed from a parallel-mounted adjustable throttle 1.3 and check valve 1.4, and hydraulic valve 1.5 are connected to the control cavity 1.6 of the second cascade of PPG 1 in the form of a cylindrical spool 1.7 with five distribution edges, respectively located with equal-sized positive $h_{дз}$ and negative overlaps $h_{вз}$ relative to the cavities-bores in the body of the second cascade of PPG 1 (in Fig. 1, the body is not shown for convenience) of the pressure 1.8, piston hydraulic cylinder 2.1 – 1.9 and intermediate drain 1.10.

The recesses – cavities on the spool 1.7 in its closed (initial) position – are separated from the piston-bore by positive $h_{д}$ and negative $h_{вз}$ overlaps of the working edges of the spool 1.7 as follows: recess-cavity 1.11 from cavity-bore 1.8 by positive $h_{дз}$, and from the bore cavity 1.10 by a negative $h_{вз}$ overlap; the recess cavity 1.12 is separated from the bore cavity 1.9 by a positive $h_{дз}$ overlap, and from the drain recess cavity 1.13 by a negative $h_{вз}$ overlap. The drain cavity 1.13 with radial holes 1.14, central hole 1.15 and radial holes 1.16 is connected by a hydraulic line 1.17 to the tank *B* of the HPD. Cyclic hydraulic accumulator 4 with an initial volume W_{oa} in the initial closed position of the spool 1.7 is connected to the hydraulic channel-hydraulic line 1.18 by a bore-cavity 1.12.

The working fluid-energy carrier is supplied to PPG 1 HPD from the hydraulic pumping station (not shown in Fig. 1) of the vibrating press through hydraulic line 1.19 (in Fig. 1) near the position of this hydraulic line, the hydraulic pump flow $Q_н$, the isothermal modulus κ of elasticity of the energy carrier and its current pressure p_r level are indicated).

The parametric principle of operation of the second stage of PPG 1 – spool 1.7 is implemented by performing a step at the lower end of spool 1.7 (according to drawing Fig. 1) with a diameter $d_{cm} < d_з$ (here d_{cm} is the diameter of the step, and $d_з$ is the diameter of spool 1.7), the end of which is designed as a conical valve that interacts with the chamfer (ground along it) of the seat formed in the PPG 1 body (not shown in Fig. 1). The ground chamfer of the stage d_{cm} behind the closed PPG 1 separates the control cavity 1.6 of the second cascade of PPG 1 from the raised cavity 1.20.

An inertial stepped valve 1.21 is located in the central axial bore (opening) of the stage with a diameter d_{cm}. Its cylindrical part with a larger diameter is coupled with the surface of the central axial bore in the spool stage 1.7 with a diameter d_{cm} with a running precise fit (for example, $\varnothing d_{цк}\dfrac{H8}{f8}$, where $d_{цк}$ is the diameter of the cylindrical part of valve 1.21). A conical chamfer is formed on the end face of valve 1.21, lapped to the chamfer of the seat formed in spool 1.7. The cavity 1.22 above the valve (according to the drawing, see Fig. 1) is freely connected by openings 1.23 to the lifting cavity 1.20. The stroke of valve 1.21 is limited by spring ring 1.24. The opening of valve seat 1.21 (see Fig. 1) is freely connected to opening 1.15.

Through the upper end (according to the drawing, see Fig. 1), the spool 1.7 is loaded with a cylindrical coil spring 1.25, the preliminary deformation of which is adjusted by screw 1.26. The control cavity 1.6 is freely connected to the cavity 1.27 under the lower end (according to the drawing, see Fig. 1) of the valve 1.21.

The executive link of the drive 2 of the vibrating press consists of: a piston hydraulic cylinder 2.1, the rod 2.2 of which is rigidly connected to the vibrating table 2.3 and the piston 2.4 of the hydraulic cylinder 2.1. A mould 2.5 is fixed on the vibrating table 2.3 with the object 2.6 of technological influence of vibratory load, for example, powder material. The vibration load on object 2.6 also acts through punch 2.7 on plunger 2.8 of plunger hydraulic cylinder 2.9, which is fixed to the movable crossbeam of the vibration press (not shown in the drawing, Fig. 1).

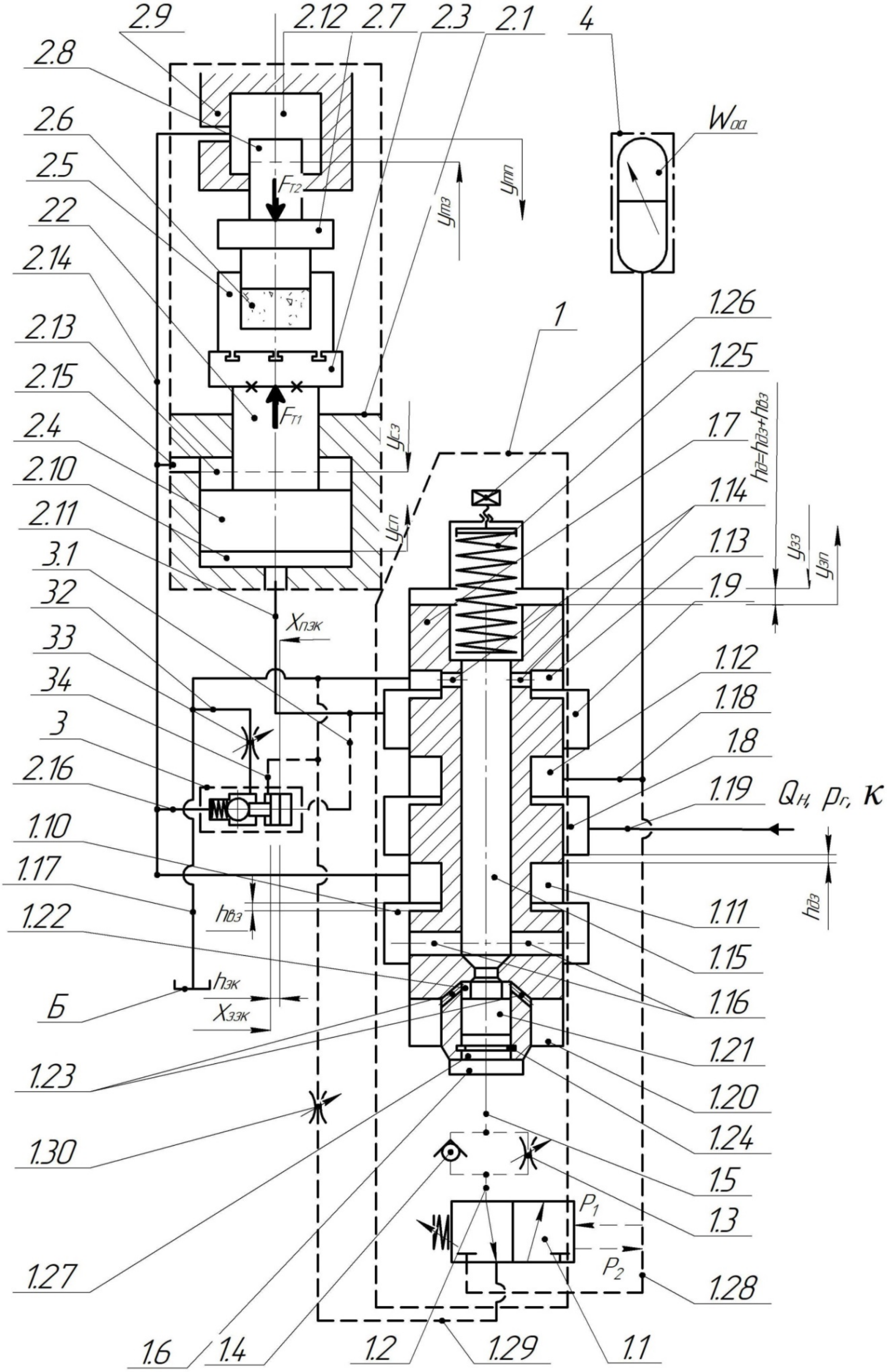

Fig. 1. – Schematic diagram of a hydraulic pulse drive for a vibrating press with a two-stage five-line pressure pulse generator with parametric control of its second stage

The energy carrier from PPG 1 is supplied to the piston cavity 2.10 of the piston hydraulic cylinder 2.1 through the hydraulic channel 2.11, which is permanently connected to the cavity-bore 1.9 of PPG 1. The working cavity 2.12 of the plunger hydraulic cylinder 2.9, the rod cavity 2.13 of the piston hydraulic cylinder 2.1, the inlet cavity of the valve part of the controlled check valve 3 are permanently connected via the hydraulic line 2.14 and hydraulic channels 2.15 and 2.16 to the bore cavity 1.11 of PPG 1.

The control hydraulic lines PPG 1 and the controlled check valve 3 on the schematic diagram of the HPD vibrating press hydraulic system are shown as thin dotted lines and perform the following functions, respectively: hydraulic line 1.28 – pressure hydraulic line supplying energy to servo drive 1.1 PPG 1; hydraulic line 1.29 – drain hydraulic line of servo drive 1.1 GIT1, on which an adjustable throttle 1.30 is installed for smooth regulation of the reverse speed of spool 1.7 of the second cascade of PPG 1; 3.1 – pressure hydraulic line for controlling the piston of the reverse control valve 3, on which an adjustable throttle 3.3 is installed; 3.4 – drain hydraulic line of the piston part of the reverse control valve. Hydraulic line 3.1 is connected to hydraulic line 2.11.

Before starting HPD the vibrating press, its entire hydraulic system (not shown in the drawing, Fig. 1) is connected via a start-up hydraulic distributor to tank B, as a result of which air is removed from all cavities of the hydraulic system, and the entire hydraulic system is filled with a power source with a pressure level equal to the drain, $p_r = p_{зл}$.

When the start distributor is switched to the closed position, the energy carrier flow is supplied to the cyclic hydraulic accumulator 4, charging it, and to the pressure chamber of the servo drive 1.1, acting on its shut-off and distribution link. As the pressure of the energy carrier in the hydraulic system HPD of the vibrating press increases to the ‘opening’ PPG 1 $p_r \geq p_1$ of the servo drive 1.1, the shut-off and distribution link switches from the right position to the left (according to the drawing, Fig. 1) and the flow of the energy carrier through the hydraulic valve 1.2, regulated by the throttle 1.3 and the hydraulic channel 1.5, enters under pressure $p_r = p_1$ (due to the small volumes of the cavities of the PPG 1, the pressure of the energy carrier in them almost instantly reaches the level p_1) into the control cavity 1.6 of the second cascade of PPG 1 and acts on the area $A_{cm} = \pi d_{cm}^2 / 4 \approx 0{,}785 d_{cm}^2$ of the stem (with a diameter of d_{cm}) of the spool 1.7.

During this process, the inertial step valve 1.21 closes under the action of pressure from the energy carrier $p_r = p_1$ on its lower end (according to the drawing, Fig. 1) and seals the lifting cavity 1.20, disconnecting it from the central drain hole 1.15. Sealing of the first stage of the spool 1.7 due to its separation from the seat, cavities 1.6 and 1.20 are connected and the pressure p_1 of the energy carrier acts on the entire cross-sectional area $A_з = \pi d_з^2 / 4 \approx 0{,}785 d_з^2$, causing a sharp increase in the switching force $F_з \gg k_{1,25} \cdot y_{01(1,25)}$ (here, respectively, $k_{1,25}$ and $y_{01(1,25)}$ the stiffness and preliminary deformation of spring 1.25. Spool 1.7 performs a working stroke $h_з = h_{дз} + h_{вз}$) switches to the upper (open) position (according to the drawing, Fig. 1).

As a result of switching spool 1.7, bore 1.12 connects to bore 1.9, and the charged cycle hydraulic accumulator 4 connects to piston bore 2.10 of piston hydraulic cylinder 2.1 via hydraulic channel 2.11. Almost simultaneously with the described process, the flow of energy carrier through the pressure hydraulic line 3.1, acting on the piston of the controlled check valve 3, closes it and thus, as a result of connecting the bore-cavity 1.8 with the bore -cavity 1.11 through hydraulic line 2.14 and hydraulic valve 2.15, the rod cavity 2.13 of the piston hydraulic cylinder 2.1 and the working cavity 2.12 of the plunger hydraulic cylinder 2.9 are directly connected to the hydraulic pump of the hydraulic system (not shown in the drawing, Fig. 1) of the vibrating press at a pressure of p_1.

Since $A_{пор} > A_{шт}$ (where $A_{пор} = \pi d_n^2 / 4 \approx 0{,}785 d_n^2$, $A_{шт} = \pi d_{шт}^2 / 4 \approx 0{,}785 d_{шт}^2$ respectively, are the cross-sectional areas of the piston 2.4 and rod 2.2 of the piston hydraulic cylinder 2.1), this causes a load from below on the object of technological influence 2.6 in matrix 2.5. At the same time, the action of the energy carrier with a pressure level p_1 and the cross-sectional area of the plunger 2.8 ($A_{пл} = \pi d_{пл}^2 / 4 \approx 0{,}785 d_{пл}^2$, where $d_{пл}$ – is the diameter of the plunger 2.8 of the plunger hydraulic cylinder 2.9) loads the object of technological influence 2.6 can be approximately estimated as follows: $F_{T1} \approx p_1 \cdot (A_{пор} - A_{шт})$; $F_{T2} \approx p_1 \cdot A_{пл}$. These forces in the process of operation of the HPD vibrating press hydraulic unit vary from maximum to minimum values, resulting in a two-sided force or vibration impact load on the object of technological influence.

After discharging the cyclic accumulator 4 and the direct movement of the vibrating table 2.7 (upwards according to the drawing, Fig. 1), the pressure of the energy carrier in cavities 1.9, 1.12, 2.10 and 1.20 decreases and at its level $p_r \leq p_2$ (here p_2 – the 'closing' pressure of PPG 1) servo drive 1.1 switches to its initial position (see Fig. 1). The lifting cavity 1.70 is connected via the check valve 1.4, servo drive 1.1 and throttle 1.30 to the drain line 1.17 with tank *B*. Under the action of spring 1.25, spool 1.7 moves to its initial position. During the return stroke of the spool 1.7, a slight increase in the pressure of the energy carrier in the cavity 1.20 opens the inertial valve 1.21, freely connecting cavity 1.20 through openings 1.23 with the central (drain) opening 1.15, resulting in a smooth landing of the spool stem 1.7 on the seat.

The connection of the piston cavity of the controlled check valve 3 with the drain cavity 1.13 through the negative overlap $h_в$ between the cavity 1.9 and the cavity 1. 13 causes it to open and connect cavities 2.12 and 2.13 via hydraulic line 2.14, hydraulic channels 2.15 and 2.16, adjustable throttle 3.3 and hydraulic valve 3.2 with drain hydraulic line 1.17. The presence of throttle 3.3 maintains a slightly elevated pressure level of the energy carrier in cavities 2.12 and 2.13, which ensures a guaranteed reverse stroke of the vibrating table with the object of technological influence 2.6. A new drive cycle begins from the initial position of all links of PPG 1 and the executive link of drive 2. Adjustable throttles 1.3 and 1.30 allow the speeds of the forward and reverse strokes of spool 1.7 of the second stage of PPG 1 to be adjusted within certain limits without diverting the energy $Q_н$ carrier flow to tank *B*, for example, through a flow regulator, which reduces the energy consumption required to exert a useful effect on the technological object.

A single-stage parametric PPG with three-line energy distribution (Fig. 2) can be used as a servo drive 1.1 PPG 1 in the described HPD vibrating press. The 'opening' pressure regulator in this servo drive 1.1 can be changed, in order to expand the technical capabilities of the HPD vibrating press, to a regulator with additional pressure p_2 regulation 'closing' PPG 1 as in a four-line parametric single-stage PPG [1, 7, 8] (Fig. 3). Other designs of servo drive 1.1 are also possible, for example, based on a slotted spring (SS) [3, 9, 14].

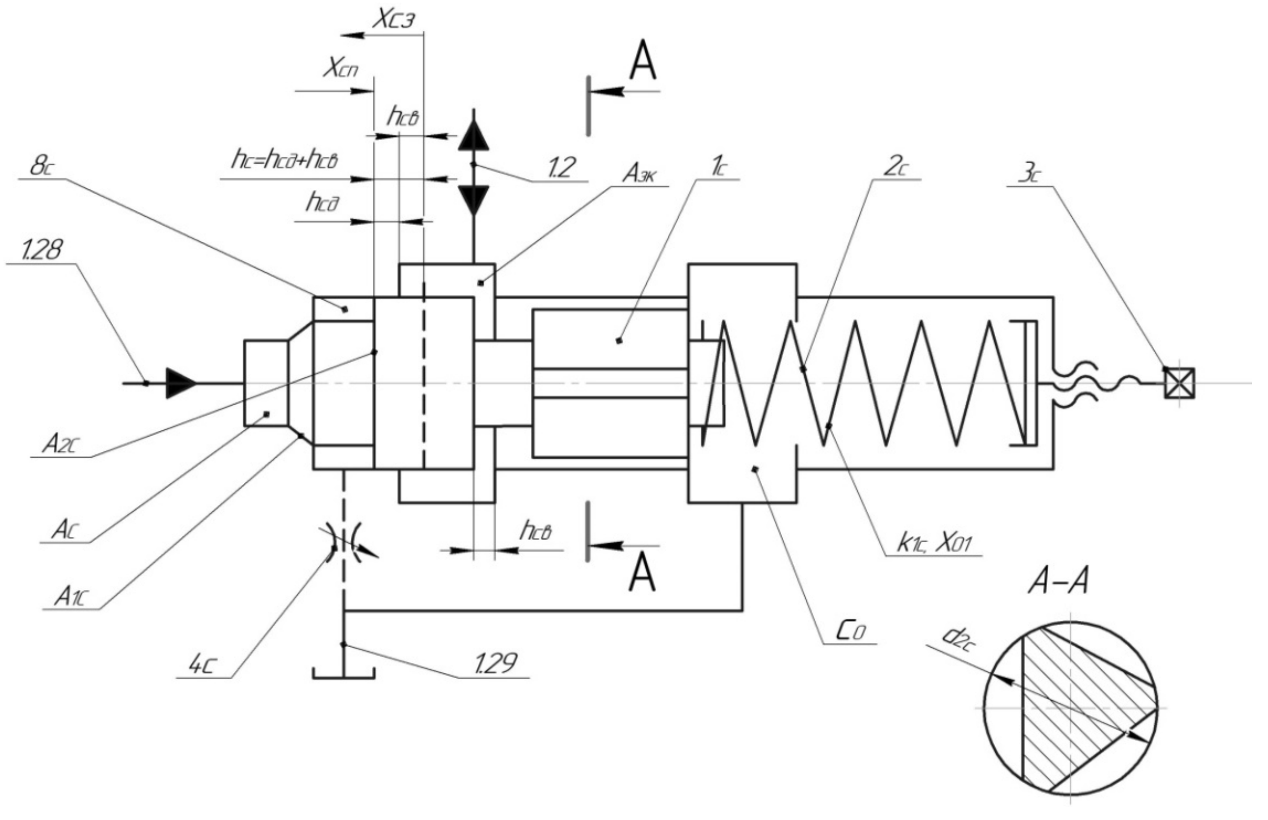

Fig. 2. – Servo drive 1.1 PPG 1 – single-stage three-line parametric pressure pulse generator

Servo drive 1.1 PPG 1, the structural and functional diagram of which is shown in Fig. 2, has a shut-off and distribution element 1_c with combined types of sealing of PPG stages – a valve in the first stage (with a chamfer ground on the seat) with an average cross-sectional area $A_к = \pi d_к^2 / 4 \approx 0,785 d_к^2$ and a spool valve in the second stage with a cross-sectional area $A_{2c} = \pi d_{dc}^2 / 4 \approx 0,785 d_{dc}^2$ (here d_{1c} and d_{2c} are the average diameter of the shut-off valve part of the shut-off and distribution element 1_c and the diameter of the spool valve part of this element, respectively). In the closed (initial) position, the positive overlap $h_{сд}$ of the spool part of the shut-off and distribution element 1_c separates the intermediate cavity B_c from the control cavity $A_{зк}$, which is connected by the PPG 1 adjustment block (see Fig. 1) hydraulic channel 1.2, and the intermediate cavity B_c in the initial position is isolated from the pressure cavity A_c of servo drive 1.1 by a ground chamfer along the middle diameter d_{1c} of the first sealing stage of servo drive 1.1. The control cavity $A_{зк}$ of the closed servo drive 1.1 through the negative overlap $h_{св}$ of the spool part of the shut-off and distribution element 1_c and the pin (see section A-A, Fig. 2) on the shank of this element is connected to the drain cavity C_c of servo drive 1.1, which is connected to drain hydraulic line 1.29 (see Fig. 1). The energy carrier A_c is supplied to the pressure cavity of servo drive 1.1 through hydraulic line 1.28 (see Fig. 1).

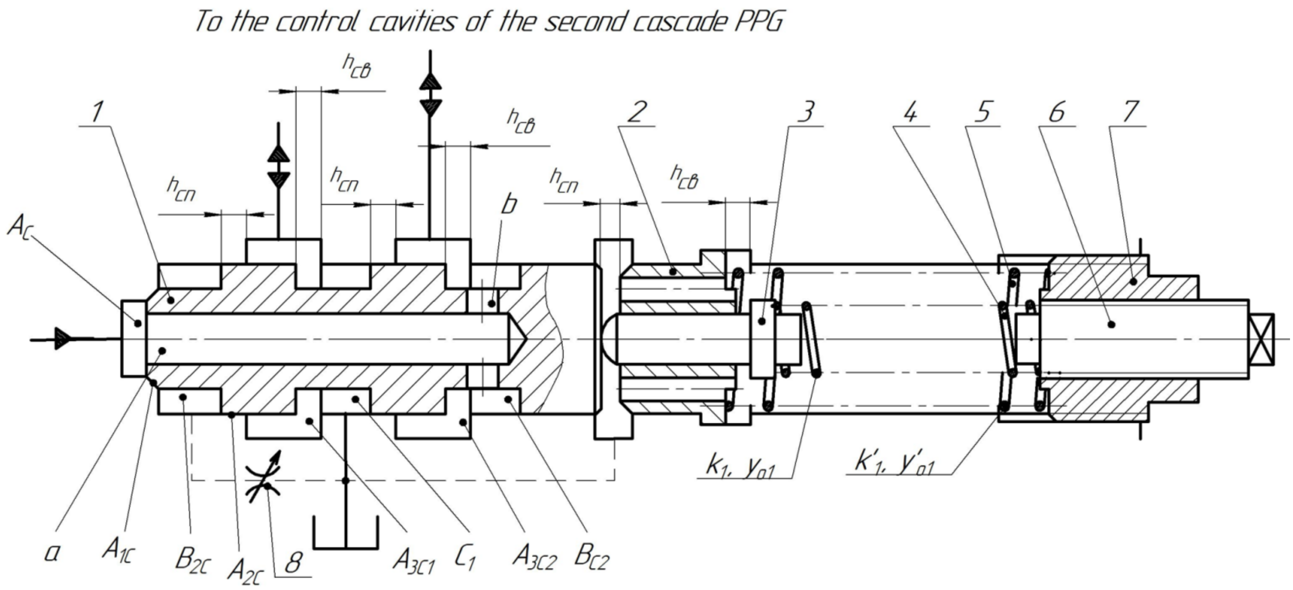

Fig. 3 – Single-stage four-line parametric PPG with energy carrier pressure level regulator for ‘opening’ and ‘closing’ the PPG

The shut-off and distribution element 1_c is loaded by a torsion spring 2_c, stiffness k_{1c}, and the ‘opening’ pressure p_1 regulator of the servo drive 1.1. The amount of preliminary deformation $x_{01c} = 0$ for $x_{01c\max}$ is adjusted using a screw 3_c (screw-nut transmission). The locking and distribution element 1_c locking mode at the end of its reverse stroke is regulated by a throttle 4_c

The pressure level for ‘opening’ p_1 servo drive 1.1 can be calculated using the following formula [1]:

$$p_1 \geq k_{1c} \cdot x_{01c} / A_{1c} \approx 1{,}27 k_{1c} \cdot x_{01c} \cdot d_{1c}^{-2}, \tag{1}$$

and the pressure of ‘closure’ depending on [3]

$$p_2 \leq k_{1c} \cdot (x_{01c} + h_c) / A_{2c} \approx 1{,}27 k_{1c} \cdot (x_{01c} + h_c) \cdot d_{2c}^{-2}, \tag{2}$$

where $h_c = h_{сд} + h_{св}$ – movement of the locking and distribution element 1_c.

By extracting the product $к_{1c} \cdot x_{01c}$ from formulas (1) and (2), we can establish a relationship between p_1 and p_1 та p_2:

$$p_2 \leq p_1 \cdot A_{1c} / A_{2c} + k_{1c} \cdot h_c / A_{2c} = p_1 \cdot U_{21}^{0.5} + k_{1c} \cdot h_c \cdot A_{2c}^{-2}, \tag{3}$$

where $U_{21} = A_{1c}^2 / A_{2c}^2 = d_{1c}^4 / d_{2c}^4$ – internal transmission ratio of the servo drive 1.1 [1 – 9].

During approximate calculations of parametric PPGs with 'opening' pressure p_1 regulators with elastic elements in the form of twisted cylindrical compression springs, component formula (3) $k_{1c} \cdot h_c \cdot A_{2c}^{-2} \ll p_1 \cdot U_{21}^{0,5}$ is neglected and the 'closing' pressure of the PPG is determined using a simplified dependence [10 – 12]:

$$p_2 \le p_1 \cdot U_{21}^{0,5} . \tag{4}$$

The parametric principle of operation of servo drive 1.1 is as follows: when the vibration press reaches the HPD due to the compressibility of the energy carrier at its pressure level $p_r \ge p_1$, the tightness of the shut-off and control element 1_c at the first stage is broken and it detaches from the seat, the cavities A_c and B_c connect. Due to the small volume of the cavity B_c, the pressure p_1 of the energy carrier in it almost instantly becomes equal and acts on the cross-sectional area A_{2c} of the spool part of the shut-off -regulating element $1_c(A_{c2} > A_{c1})$, which, moving quickly, passes through the positive overlap $h_{сд}$ connects the cavities A_c and B_c with the control $A_{зк}$ simultaneously disconnecting it from the drain cavity C_c. Usually, in order to avoid energy losses of the energy carrier during the direct movement of the shut-off and control element due to its transfer to the drain cavity C_c through incomplete passage of the negative overlap $h_{св}$, it is made smaller relative $h_{сд}$ to , as a rule $h_{св} = h_{сд} - (0,5...0,6)$ mm [1].

The direct movement of the shut-off and control element 1_c (opening of servo drive 1.1) causes the direct movement of spool 1.7 PPG 1 and the working movement of the executive link 2 of the vibrating press (see Fig. 1 and description of the HPD operation process), as a result of which the pressure of the energy carrier in the entire HPD hydraulic system decreases to the 'closed' level $p_r \le p_2$, causing the reverse movement of all HPD links, after which the operating cycle is repeated.

The working cycle of the vibrating press consists of many stages and phases, which are usually combined into forward (index 'п') and reverse (index 'з') strokes of the moving parts of the vibrating press during modelling and design calculations HPD– PPG 1, the executive link of the drive 2 and the controlled reverse valve 3. In Figs. 1 and 2, the directions of movement of the HPD links during their forward and reverse strokes are marked as follows: servo drive 1.1 (see Fig. 2) – $x_{сп}$ and $x_{сз}$; spool 1.7 of the second cascade of PPG 1 (see Fig. 1) – $y_{зп}$ and $y_{зз}$; piston hydraulic cylinder 2.1 with vibrating table 2.3 – $y_{сп}$ and $x_{ззк}$; plunger hydraulic cylinder 2.9 with punch 2.7 – y_{mn} and $y_{mз}$; piston of controlled check valve 3 – $x_{пзк}$ and $x_{ззк}$. Also, in Figs. 1 and 2, the maximum strokes of these HPD links are indicated near their directions of movement.

Well-known theoretical studies HPD of vibration (VM) and vibration-impact (VIM) technological machines [1–4] and the method of their design calculation are based on their dynamic and mathematical models, which are constructed on the basis of the principle (structural) diagrams of HPD (see Fig. 1 and Fig. 2) and indicative cycle diagrams of the working cycle of these HPDs [10–13].

Conclusions

1. A schematic diagram of HPD a vibrating press with a two-stage five-line PPG with parametric control of its second stage has been developed.

2. Parametric control of the second stage of PPG 1 is provided by the lower end of spool 1.7 of the second stage of PPG 1 (see drawing, see Fig. 1) of a stepped protrusion with a diameter $d_{cm} < d_з$ with a bevel seal, which allows, when opening spool 1.7 – the second stage of sealing of PPG 1, to use its entire cross-sectional area $A_з = \pi d_з^2 / 4 \approx 0,785 d_з^2$ to increase its opening speed and thereby PPG 1 and the technological capabilities of the HPD vibrating press in general.

3. Additional working edges of the spool 1.7 of the second cascade of the PPG 1 with identical positive $h_{дз}$ and negative $h_{вз}$ overlaps divide the working cycle of the HPD vibrating press into two circuits, one of which is powered by a flow of energy from the cyclic hydraulic accumulator 4, and the second from the hy-

draulic pump of the HPD hydraulic pump station (see Fig. 1), which allows the executive link of the vibrating press to use a piston hydraulic cylinder to drive its vibrating table into vibratory motion and ensure the reverse movement of the vibrating table and additional vibratory loading of the object 2.6 of technological influence using a plunger hydraulic cylinder 2.9 (see Fig. 1), thereby expanding the technological capabilities of the HPD vibrating press.

4. Dividing the working cycle of the HPD vibrating press into two circuits allows, to some extent, reducing the load on the hydraulic pump of the HPD hydraulic pumping station with variable pressure of the energy carrier, which increases the service life of the hydraulic pump.

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Obertykh Roman – Cand. Sc. (Eng.), associate professor, Professor of department industrial engineering , Vinnytsia National Technical University, https://orcid.org/0000-0003-2939-6582 , e-mail: obertyuh557@gmail.com.

Slabkyi Andrii – Cand. Sc. (Eng.), Associate Professor of department of industrial engineering, Vinnytsia National Technical University, https://orcid.org/0000-0001-9284-2296, e-mail: slabkiyandrey@gmail.com.

Ishchenko Vasyl –student of the department industrial engineering, Vinnytsia National Technical University.

Р. Р. Обертюх [1]
А. В. Слабкий [1]
В. В. Іщенко [1]

Гідроімпульсний привод вібропреса з двокаскадним п'ятилінійним генератором імпульсів тиску з параметричним керуванням його другим каскадом

[1] Вінницький національний технічний університет

За результатами виконаного теоретичного дослідження існуючих конструкцій вібропресового обладнання на базі   Аналіз теоретичних та експериментальних досліджень гідроімпульсного приводу технологічних вібраційних і віброударних машин (віброрізання та деформаційного зміцнення поверхонь деталей машин, гідроімпульсних вібраторів тощо) показав, що гідроімпульсний привод дозволяє реалізувати різноманітні вібраційні технології в широкому діапазоні технічного застосування та з регульованим вібронавантаженням об'єктів технологічного впливу. За результатами виконаного теоретичного дослідження існуючих конструкцій вібропресового обладнання на базі гідроімпульсного приводу розроблено принципову та конструктивну схеми гідроімпульсного вібропреса з двокаскадним п'ятилінійним генератором імпульсів тиску з параметричним керуванням його другим каскадом і виконавчою ланкою з приводом вібростола вібропреса від поршневого гідроциліндра та навантаженням об'єкта технологічного впливу гідроімпульсного приводу від плунжерного гідроциліндра, корпус якого встановлено на архітраві вібропреса, а плунжер кінематично зв'язаний з рухомою траверсою вібропреса.

В статті представлений опис роботи розробленої нової конструкції гідроімпульсного вібропреса з двокаскадним п'ятилінійним генератором імпульсів тиску з параметричним керуванням його другим каскадом та приведені основні розрахункові залежності, згідно яких можна розрахувати основні геометричні та силові параметри розробленої конструкції приводу. Таке конструктивне рішення забезпечує розширення технічних і технологічних можливостей гідроімпульсного приводу, адже забезпечує високошвидкісний режим вібропереміщення виконавчої ланки – вібростола вібропреса в поєднанні з гідравлічним навантаженням об'єкта технологічного впливу від додаткового плунжерного гідроциліндра, встановленого на архітраві вібропреса, плунжер якого приводить в рух рухому траверсу вібропреса

Ключові слова: генератор імпульсів тиску; герметизація; гідравлічна ланка; гідроімпульсний привод; прес; привод; подача; тиск; хід.

Обертюх Роман Романович – кандидат технічних наук, доцент, професор кафедри Галузевого машинобудування, Вінницький національний технічний університет, https://orcid.org/0000-0003-2939-6582, e-mail: obertyuh557@gmail.com.

Слабкий Андрій Валентинович – кандидат технічних наук, доцент, доцент кафедри галузевого машинобудування, Вінницький національний технічний університет, https://orcid.org/0000-0001-9284-2296 , e-mail: slabkiyandrey@gmail.com

Іщенко Василь Володимирович – магістр кафедри галузевого машинобудування, Вінницький національний технічний університет.