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doi:10.15199/48.2019.04.16

Analysis of hydraulic vibration drive machine for vibration abrasive processing

***Abstract**. In this work the original design of the vibration hydraulic cylinder of the drive of machine for the vibration abrasive machining is proposed, the structural and calculation scheme of the plunger vibration hydraulic cylinder is considered, also its dynamic and mathematical models of the work flow are presented and their mathematical analysis is performed.*

***Streszczenie.** W pracy przedstawiono oryginalny projekt wibracyjnego siłownika hydraulicznego napędu maszyny do wibracyjnej obróbki ściernej, przedstawiono schemat konstrukcyjny i obliczeniowy siłownika hydraulicznego z tłokiem wibracyjnym, przedstawiono również jego dynamiczne i matematyczne modele przepływu sterowania, a także dokonano ich analizy matematycznej. **(Analiza hydraulicznego wibracyjnego napędu do wibracyjnej obróbki ściernej)**.*

Keywords: hydraulic cylinder, hydraulic drive, drive for the vibration abrasive machining, dynamic and mathematical models.
Słowa kluczowe: siłownik hydrauliczny, napęd hydrauliczny, napęd wibracyjnej obróbki ściernej, modele dynamiczne i matematyczne.

Introduction

Equipment for vibration abrasive processing (VAP) is used in machine-building for performing, for example, cleaning of cast, forged, punched and heat-treated billets, parts and semi-finished products of various purposes and a wide range of products (see next chapter), [1]. Cleaning the workpieces in many cases precedes the implementation of further technological operations and plays an important role in the overall process of manufacturing parts.

Such processes and machines and their implementation are well known in the machine-building industry. But most of them have a significant drawback - relatively small performance. Therefore, the problem of finding ways to increase the productivity of these machines is an important problem.

Analysis of literary data and problem statement

There are many types of VAP, among the most effective are:

Spindle vibration abrasive treatment (fig. 1) [2] is the process of surface treatment of parts that come in contact with a vibrating abrasive working environment.

Such processing is finished and does not imply changes in the accuracy of the sizes. Treatment is carried out by removing the smallest particles of the metal, its oxides, or the plastic deformation of the surface to be treated as a result of relative slip and collision with a sufficiently high speed of the treated surface and particles of the working medium. Details are given the rotation (by attaching it to the spindle of the machine) at a speed of 0.5 -1 to 7-15 m/s. The working environment is exposed to directed vibrations with a frequency of 1500 - 2000 min^{-1} and an amplitude of 1 - 5 mm (fig. 1) [1]. Chemically and surface-active substances can be used to intensify the processing process. The large working space, where the working environment, allows the processing of parts in the entire volume of the camera, while excluding the need for a rigid connection of the treated surface with the processing environment, and, in addition, create the preconditions for automating the process and increasing the efficiency of the operation as a result of multi-spindle (batch) processing. The spindle may be arranged vertically, horizontally or at different angles. Vibration abrasive processing by spindle allows to carry out grinding-polishing, cleaning, strengthening operations, removing of scorings, and others. The machine proposed by the author has good preconditions for ensuring high productivity, but the mechanical drive does not allow to provide a wide range of vibration parameters with the provision of constant power during processing.

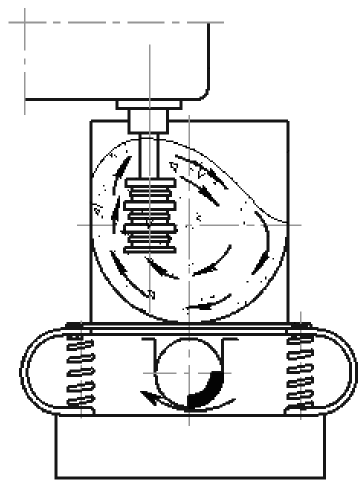

Fig. 1. Scheme of the process of spindle vibration abrasive processing: 1 - spindle of the machine; 2 - working chamber (bunker); 3 - vibration installation; 4 - the workpiece

Vibration abrasive electrochemical treatment (fig. 2a). The essence of it is as follows: the component is connected to the positive pole of the current source and immersed in a vibrating abrasive medium, which is dampened by the electrolyte of the required composition, and the hopper is connected to the negative pole of the current source. When the current source is switched on, there is an electrochemical (anode) dissolution of the surface layer of the workpiece and continuous mechanical removal by the abrasive products of the anode dissolution products.

Vibration thermo-mechanical treatment (fig. 2b). The essence of this type of VAP is to heat the machined parts, which is provided by the design of the working chamber (to create a high temperature, mainly used electric heaters). Cooling in vibration units is carried out with the help of a system of circulation of compressed air or lubricating and cooling fluid. Vibration thermo-mechanical treatment can be used not only as a strengthening operation, but also as a stabilizing treatment, as well as for the application of certain types of pellicles [2]. The machine proposed by the author has a mechanical vibration drive, which has low efficiency.

Magnetic abrasive processing (fig. 2c). Its feature is that in the working area of the vibration unit a permanent or alternating force magnetic field is created that is directed along the axis of the circulating motion of the working medium, which moves in the chamber under the action of its oscillations. Processed ferromagnetic parts are installed along magnetic lines of force, suspended and braked in a moving environment by a magnetic field. The working chamber is executed from diamagnetic material (stainless steel, duralumin, etc.); The protection of machined parts from mutual collisions in the processing process is as a result of their mutual repulsion with the same name magnetized their ends [3,4]. The machine proposed by the author has electromagnetic vibration drive, which has low efficiency when used for the bulky vibration bunker. Therefore, this machine can not be used for simultaneous processing of a large number of parts.

Vibro-mechanical machining (fig. 2d) [5,9,10]. The essence of this kind of processing is that the vibration load (can be two or three-dimensional (spatial)) is applied to the working chamber (bunker), which allows to reproduce the trajectories of the motion of the arbitrary working medium. The use of this type of VAP provides increased accuracy of reproduction of the specified parameters of the processing process, which significantly extends the scope of its use, especially for details of complex configuration [2]. The machine proposed by the author has a mechanical vibration drive, which has low efficiency.

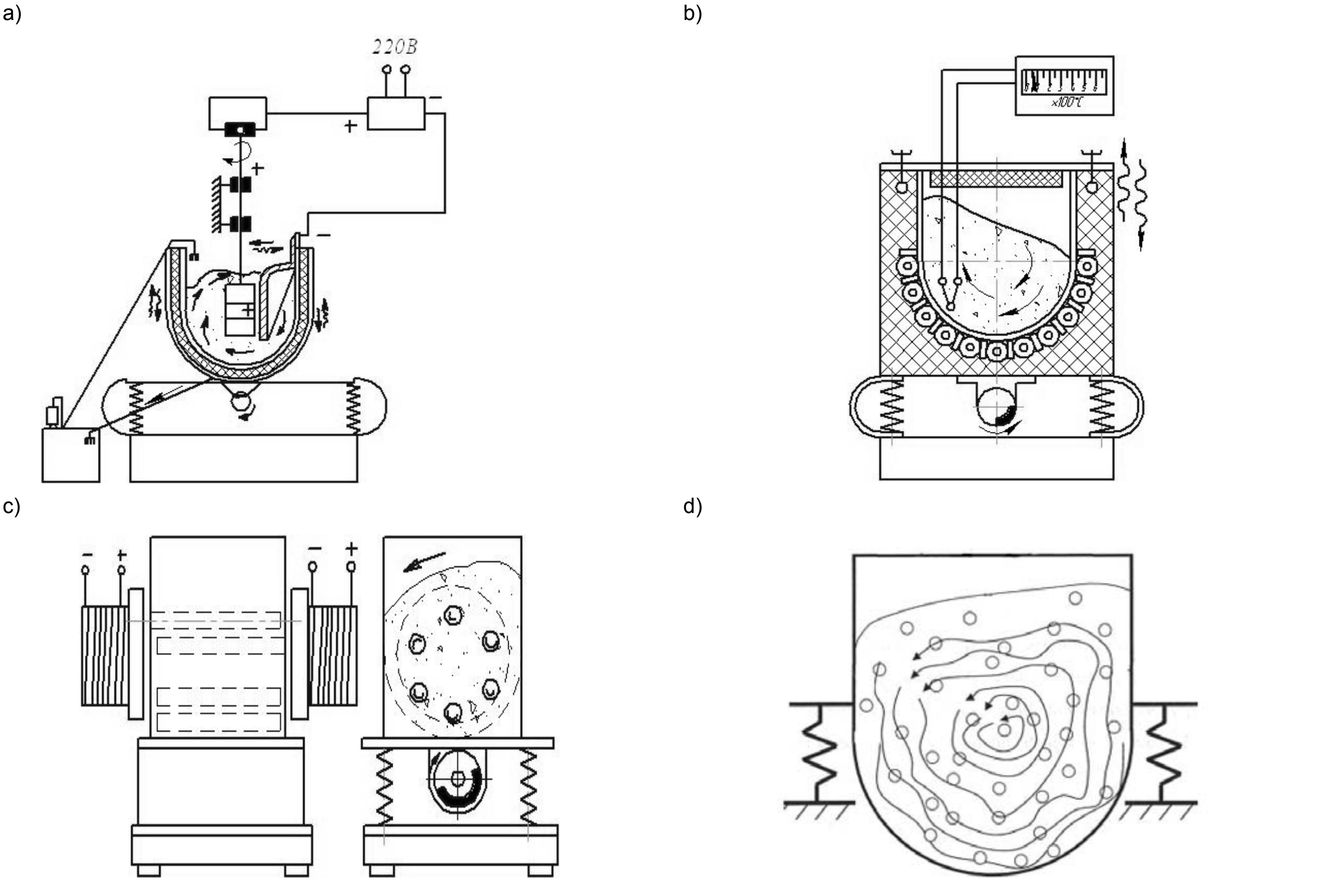

Fig. 2. Types of vibration abrasive processing: a) Vibration abrasive electrochemical treatment; b) Vibration thermo-mechanical treatment; c) Magnetic abrasive processing; d) Vibro-mechanical machining

Purpose and objectives of the research

Various drives are used for the operation of the VAP equipment: most often mechanical, electric, combined, less often hydraulic and pneumatic. But mechanical and electromagnetic drive does not allow to provide a wide range of vibration parameters with the provision of constant power during processing. Authors of the works [6,7] have established that the vibrating hydraulic drive has the smallest size among other types of actuators with the same power output, and is capable of generating wide range vibrations. At the same time, the hydraulic drive is relatively complex and contains a lot of constituent elements. Thus, the hydraulic impulse drive for equipment for the VAP should consist of a pumping station, a pressure pulse generator (PPG), an executive hydraulic cylinder that generates a vibration load and a working link (vibrating hopper).

The purpose of the research is to develop a universal compact hydraulic vibration drive for use in the machines considered in the previous chapter. But most importantly that hydraulic vibration drive allow to provide a wide range of vibration parameters with the provision of constant power during processing and it will increase the machining performance.

Materials and methods of research

In order to get a more compact drive, we were asked to combine the executive hydraulic cylinder and PPG in one node. As a result, we created a plunger vibrating hydraulic cylinder, the structural-calculation scheme of which is shown in fig. 3.

According to the scheme, the plunger vibrating hydraulic cylinder works as follows. In the initial position, the plunger 1 (executive link), which acts as a spool, is pressed with a spring 2 to a ball 3 that overlaps the plunger channel. With increased pressure in the cavity A, the instantaneous increase of force on the stationary shutter element 3, which is made in the form of a ball, is instantaneously increased to the pressure p_1 of the action.

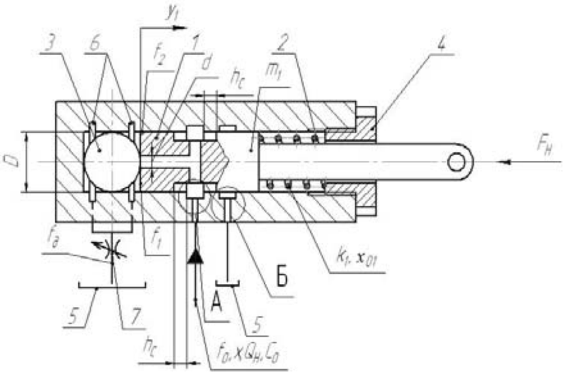

Fig. 3. Plunger vibrate cylinder (structural- calculation scheme): 1 – plunger; 2 – a spring; 3 – a ball; 4 – nut; 5 – hydraulic tank; 6 – ring ducts; 7, 8 – hydraulic chokes

The indicated increase is due to the step change in the lifting area from $S_{n1} = \pi d^2 / 4$ to $Sn_2 = \pi D^2 / 4$ ($S_{n2} > S_{n1}$) at the moment of breaking the sealing of the landing of the locking element 3 on the saddle in the plunger 1. As a result, the lifting effort increases from $P_{H1} = p_1 S_{n1}$ to $P_{H2} = p_1 S_{n2}$. The P_{H2} effort is usually considerably greater than the force setting of the spring $P_{np} = P_{H1}$, which presses the shutter element 3 to the saddle in its original position. Under the action of P_{H2}, the plunger moves to the right by connecting the cavity A with the cavity of the drain B, an instantaneous drop in pressure in the hydraulic system to the value p_2. At the same time, the force on the locking element 3 decreases to the configurable value $P_{H1} = p_2\ S_{n2}$ and the spring can return the plunger 1 to its original position, pressing it to the ball 3. Then the cycle repeats itself.

The regulation of the frequency of operation of the PPG is due to the change in the stiffness of the spring 2 with the help of a nut 4. The remainder of the liquid from the chamber in which the ball 3 is located will be displaced in the tank 5 through the ring ducts 6 which are connected by a channel with a drain line. The volume of the fluid that will jerk out of the cavity in which the ball 3 is located is controlled by the throttle 7, which is an additional mechanism of frequency control. The plunger vibrate cylinder has several advantages: technological, ease of design, easy control of the parameters of the generated vibration load. The dynamic model of the vibratory hydraulic cylinder is shown in fig. 4.

During the compilation of the structural and design scheme and the dynamic model of the plunger vibrate cylinder, the following designations were used:

m_1 – is the weight of the plunger and useful weight;
K_1, K_0, – linear stiffness of spring 2 (fig. 3) and the hydraulic line respectively;
C_1, C_0 – viscous friction in a hydraulic cylinder;
R_1 – "dry" friction of elements of the hydraulic cylinder;
Q_N - pump feed;
F_H – external load force;
f_1, f_2 and $f_{\partial p}$ – are the cross-sectional areas of the contact surfaces of the plunger and the throttle, respectively;
x_1 – displacement of mass m_1 along the abscissa;
x_{01} – compression of the spring;
l_1, l_2, l_3 – the length of the hydraulic line.

When constructing a complete dynamic model, the following assumptions were adopted:

- taking into account the peculiarities of the hydraulic system construction with the highest possible rigidity achieved with minimal volumes of hydraulic lines, the working fluid is represented by an "elastic, concentrated model" [6,11,12];

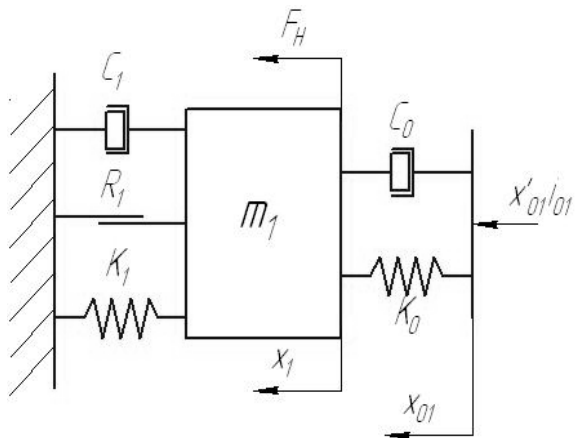

Fig. 4. Dynamic model of the vibrating hydraulic cylinder

- the walls of the channels and pipelines are absolutely rigid;
- wave processes in hydraulic lines, due to their not significant length, do not affect the dynamics of the system;
- shock interactions of masses in the initial equations of motion are not taken into account;
- leaks of the working fluid through the gaps between the conjugated elements are insignificant compared to the supply of the hydraulic pump and the original equations of motion are not taken into account;
- the hydrodynamic forces acting on the shutter element of the PPG are small in comparison with the forces of displacement, elastic return and resistance;
- reduced volumetric modulus of elasticity of the hydraulic system is taken constant throughout the operating cycle of the vibration installation;
- the hydraulic resistance of the drainage line is small and therefore the initial dependencies are not taken into account.

In order to obtain the maximum rigidity of the hydraulic system, the volume of pressure in entering hydraulic line and cavities is reduced to a minimum, which allows the weight of the fluid in the pressure of entering hydraulic line neglected by replacing it with massless springs with stiffenings K_0, K_1 and dampers with viscosity C_1, C_0 [7,8].

The hydraulic link in the form of a massless spring K_0 and a damper with a viscosity C_0, which deforms at a constant velocity $V_0 = Q_H / f_0$ [5], where Q_H = const - delivery of the hydraulic pump; $f_0 = \sum_1^n (l_i \cdot f_i) \Big/ \sum_1^n l_i$ - averaged area of pressure hydraulic line; l_i and f_i, respectively, the length and cross-sectional area of the individual channels of the pressure cavity of the vibration hydraulic cylinder interacts with the mass m_1 due to the transfer ratio i_{01}. The elastic and viscous resistance to the displacement of the mass m_1 is determined by the coefficients of stiffness K_1, K_0 and the viscosity C_1, C_0, and the dry friction - by force R_1, in addition to the working link also acts the force F_H of the technological load.

The stiffness of the hydraulic system on the basis of the Hooke's law for the fluid is determined by the known dependence [6,13-15] for each section of the hydraulic system in the form:

(1) $$K_{01} = \frac{f_0^2 \cdot \chi}{W_o + f_1 \cdot x},$$

where: χ – combined volumetric modulus of the hydraulic system; W_0 – initial volume of the pressure cavity of the hydraulic system; f_1 – area of the cross-section of the plunger; x – coordinate of movement m_1.

As a rule $W_o \geq f_1 \cdot x$, then $K_0 \approx f_0^2 \cdot \chi / W_o = const$.

The transfer ratio i_{01} can be calculated provided that the instantaneous pressure of the energy carrier which applied to the hydraulic link is equal to the pressure which applied to the cross-sectional area of the corresponding drive's line:

(2) $$\frac{K_{01} \cdot x_{01}}{f_0} = \frac{i_{01} \cdot K_{01} \cdot (x_{01} - x_1)}{f_1}.$$

From the formula (2), we will output the value

(3) $$i_{01} = \left(\frac{f_1}{f_0}\right) \cdot \left[\frac{x_{01}}{(x_{01} - x_1)}\right],$$

where x_{01} – deformation of the hydraulic link.

The product $i_{01}K_{01} = K_{11}$ is the rigidity coefficient of hydraulic system cocked to m_1. The operating cycle of the vibration installation can be divided into two stages – the direct and the return of the working link. Using the Dalamer principle, on the basis of the dynamic model of the workflow (fig. 3), we write the equation of motion of the mass m_1 in the direct motion in the form

(4) $$\begin{gathered} m_1\ddot{x}_1 = i_{01}k_0\left(x_{f1} - x_1\right) - k_1\left(x_1 + x_0\right) - \\ -i_{01}^{0,5}C_0\left(\dot{x}_{f1} - \dot{x}_1\right) - C_1\dot{x}_1 - R - F_H; \\ 0 \leq x_1 \leq h_c;\ x_0 \leq x_{0\max}. \end{gathered}$$

To obtain an unambiguous solution of equation (4), it is necessary to add to it the equation of expenditure:

(5) $$Q_H = \dot{x}_0 k_0 f_0^{-1} W_0 + \zeta_1 \pi d h_c \sqrt{\left(2k_0/\rho\right) \cdot \left(x_{f1} i_{01} f_1^{-1}\right)} + Q_{op},$$

where ζ_1 – the coefficient of flow due to the negative overlap of the valve; x_{f1} – the deformation of the hydraulic link is cocked to the area f_1.

On the other hand, using the well-known technique [7, 8,1], the pattern of pressure change in the cavity of the hydraulic cylinder $p(t)$ at the hydraulic device's operation stage can be written as

(6) $$p(t) = K_0 \frac{x_0 - x_1}{f_2}.$$

For the dynamical model considered (fig. 4), the displacement

(7) $$x_0 = \frac{Q_H}{f_{pl}}\left(\tau_{зат} + t\right).$$

At the moment of the detention, $\tau_{зат}$, corresponding to the beginning of the displacement of the mass m_1 ($x_1 = 0$), which deforms the elastic link K_0 by the effort $R_m = K_0 x_0$ ($\tau_{зат}$). This effort is determined by the efforts of the previous tension $P_0 = K_0 x_{01}$ springs of elastic return.

Then the equation of motion of mass m_1 can be written in the form

(8) $$\begin{gathered} m_1 \frac{d^2 x_1}{dt^2} + \xi_0 \frac{d^2 x_1}{dt^2} + (C_0 + C_1)\frac{dx_1}{dt} + \\ +(K_0 + K_1)x_1 = K_0 x_0. \end{gathered}$$

Using the linearization method given in [7,8], we obtain a solution of equation (4)

(9)

$$x_1 = \frac{K_0 \cdot Q_H}{(K_0 + K_1) \cdot f_2} \cdot t - -\frac{K_0 \cdot Q_H}{(K_0 + K_1) \cdot f_{пл}} \sqrt{\frac{m_1}{K_0 + K_1}} \cdot \frac{e^{\sqrt{\frac{K_0 + K_1}{m_1}} \cdot t}}{\left(\sqrt{1 - \beta^2}\right)^3} \cdot \sin G \cdot t,$$

where: $\beta = \frac{a}{2}\sqrt{(K_0 + K_1) \cdot m_1}$.

Substituting equation (5) in (6) we obtain expression of pressure change in the cavity of the hydraulic cylinder

(9) $$\begin{gathered} p(t) = \frac{K_0 \cdot x_{деф}}{f_2} + \frac{K_0 \cdot K_1}{K_0 + K_1} \cdot \frac{Q_H}{f_2} \cdot t - \\ -\frac{K_0^2 \cdot Q_H}{(K_0 + K_1) \cdot f_2^2} \cdot \sqrt{\frac{M_{пр}}{K_0 + K_1}} \cdot \sin\sqrt{\frac{K_0 + K_1}{m_1}} \cdot t. \end{gathered}$$

For such vibrational hydraulic drives, the maximum pressure in the cavity of the hydraulic cylinder corresponds to the pressure p_1 for adjusting the actuation of the valve of the PPG. Period of pulsations of pressure in the cavity of the working hydraulic cylinder $T = t + t_{ск} + t_1$, where $t_{ск}$ – time of pressure drop by the PPG; t_1 – time of filling the volume of the hydraulic system at $p = p_2 \approx 0$ – the time of attenuation of the oscillation of the executive link after braking at the end of the reverse.

The pressure drop time t_{ck} is determined by the parameters of the PPG (vibration exciter) and the volume V_c of the hydraulic system. In similar PPG, pressure relief is performed at a significant speed $(dp/dt)_{average}$ (average speed approximately 10^9 - 10^{10} *Pa/s*). The time of pressure drop is determined by the dependence

$$t_{ск} = \frac{p_{\max}}{(dp/dt)_{\text{average}}}.$$

The increase in pressure Δp is achieved mainly due to the fluctuation of the pressure of the fluid in the first quarter of the first oscillatory cycle with an amplitude value

$$A_p = \frac{K_0^2 \cdot Q_H}{(K_0 + K_1) \cdot f_2^2} \cdot \sqrt{\frac{m_1}{K_0 + K_1}}.$$

We use assumptions

$$\frac{K_0 \cdot K_1}{(K_0 + K_1) \cdot f_2^2} \cdot t_{(0)} = 0.$$

Then the time τ_0 of reaching the pressure p_1 is expressed by (9). The solution of this equation is

$$\tau_0 = \sqrt{\frac{m_1}{K_0 + K_1}} \cdot \arcsin\frac{\Delta p}{A_p}.$$

Taking into account the assumed assumption of solving these equations, we will allow us to determine the displacement of the plunger, the pressure variation in the cavity of the hydraulic vibration cylinder, and hence the effort it develops.

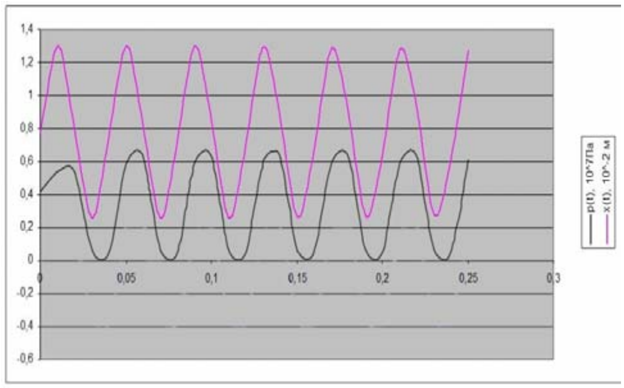

Fig. 5. Graphs of displacement of the plunger *x(t)* and pressure changes in its cavity *p (t)*

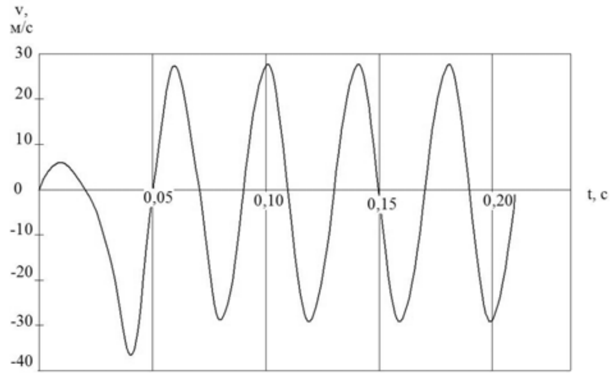

Fig. 6. Chart of speed change *v (t)* of the hydraulic cylinder's plunger

Research results

By solving equations 4 and 5 for the initial data presented in Table 1, we obtained the graph of displacement of the plunger of the vibration hydraulic cylinder and the graph of pressure change in the hydraulic drive, shown in fig. 5.

From fig. 5 it is seen that the cylinder plunger moves at a frequency of 25 Hz and an amplitude of 6 mm, the maximum pressure on the valve of PPG $p_1 = 13 \cdot 10^6$ Pa, and the opening pressure $p_1 = 3 \cdot 10^6$ Pa.

Table 1. Input data for the analysis of a computer mathematical model of a vibrating hydraulic cylinder

K_0, N/m	Hardness of the hydraulic spring	$24 \cdot 10^6$
K_1, N/m	Hardness of the spring	$1.02 \cdot 10^5$
Q_H, m^3/s	Pump supply	0.023
f_{pl}, m^2	Plunger cross-sectional area	0.0035
M_1, kg	Weight of the plunger with the weight attached to it	82
X_0, m	Previous strain of the spring	0.04

The chart of speed change *v(t)* of the hydraulic cylinder's plunger is presented in fig. 6, which is the result of differentiation of movement. From fig. 5 it is seen that the plunger moves with a maximum speed of 30 m/s.

An important task in the further work in this area of research should be to create a prototype and perform a comparison of theoretical values with practical [16-18].

Conclusions

The authors were tasked to develop the drive for machines for vibration abrasive processing. Based on the analysis of literary sources, it has been established that the mechanical and electromagnetic actuators, that most commonly used in such machines are not sufficiently effective for generating vibrations in the wide-area range with constant power and in large-scale vibration bins used in mass production for the simultaneous processing of a large number of parts.

The authors suggested that a compact hydraulic vibrating drive should be used to solve this problem, which could create vibrations in a wide range with high-power.

The proposed structural scheme of the plunger vibration hydraulic cylinder, on the basis of which was developed a dynamic and mathematical model of the workflow. As a result we obtained the graph of displacement of the plunger of the vibration hydraulic cylinder and the graph of pressure change in the hydraulic drive.

On the basis of the offered equipment (which was examined in the first chapter of the article) it is possible to construct a more compact vibration hydraulic drive with a very wide range of regulation of the generated vibration load. Also the plunger vibratory hydraulic cylinder can be used as an element of a vibrating actuator for a variety of equipment (VAP equipment, vibrating conveyors, vibratory loading/ unloading devices, etc.).

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