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PRESSURE PULSE GENERATOR OF DIFFERENTIAL ACTION

New pressure pulse generator of differential action with increased vibration frequency has been designed. Its mathematical model is suggested and investigated as a component of vibration hydraulic drive for solid domestic waste compaction. The range of rational values of its main parameters relationships was found.

***Key words**: pressure pulse generator, vibration frequency, vibration hydraulic drive, compaction, solid domestic waste, mathematical modeling, rational parameters determination.*

Introduction

To create vibrations and to control their parameters in the working equipment of hydraulic drives, pressure pulse generators (PPG) of one- and two-stage design are currently used [1].

A disadvantage of the known one-stage PPG is their limited functional capabilities caused by narrow control range of their main parameters – frequency and amplitude of pressure pulses. When conditional passageway of the gate in one-stage PPG is increased, their size and general mass are also increased considerably [2, 3].

To the disadvantages of two-stage PPG belong their relatively big size, problems with synchronization of the first and second stages actuation [2, 3].

Problem statement, defining relationships

Therefore, a principally new PPG design was created that is an intermediate between one- and two-stage generators. It was named pressure pulse generator of differential action (PPGDA) and granted a useful model patent 29363 U [4].

The engineering result is increased frequency of the working fluid pressure oscillations. It is achieved by the introduction of an additional circular bore, connected with pressure line, to the PPG design.

This task is solved by providing PPG that comprises a housing, a spring-loaded valve, connecting pipes. The valve is of two-stage design. The first stage of a smaller diameter is pressed against the adjusting seat and the second stage of a larger diameter provides for a positive overlap of a circular bore connected with the discharge. The upper part of the valve second stage is located in the intermediate chamber formed in the housing body. Said upper part of the valve second stage includes longitudinal bores connecting the intermediate chamber with the circular bore. Besides, the upper part of the second stage is installed so that it can be in contact with the pushing plunger that is spring -loaded relative to the housing and its upper part extends to the above-valve chamber that, in its turn, communicates, via a channel, with an under-valve chamber and with the valve first stage located above it and being in constant communication with the pressure line. The first and second stages of the valve form a closed chamber with the housing, said chamber being in constant communication with the circular bore via a throttle. The diameters of the first and second stages and of the pushing plunger are related as $d_1<d_2<d_3$ correspondingly. The valve housing has additional upper circular bore connected with the pressure line. Overlap $h_в$ (the distance from the plunger

upper end to the upper face of the upper circular bore) makes 0,5…1 of the positive overlap h_n. In addition, the above-valve chamber is in continuous communication with discharge line through a variable throttle.

Fig. 1 shows the circuit of the pressure pulse generator of differential action.

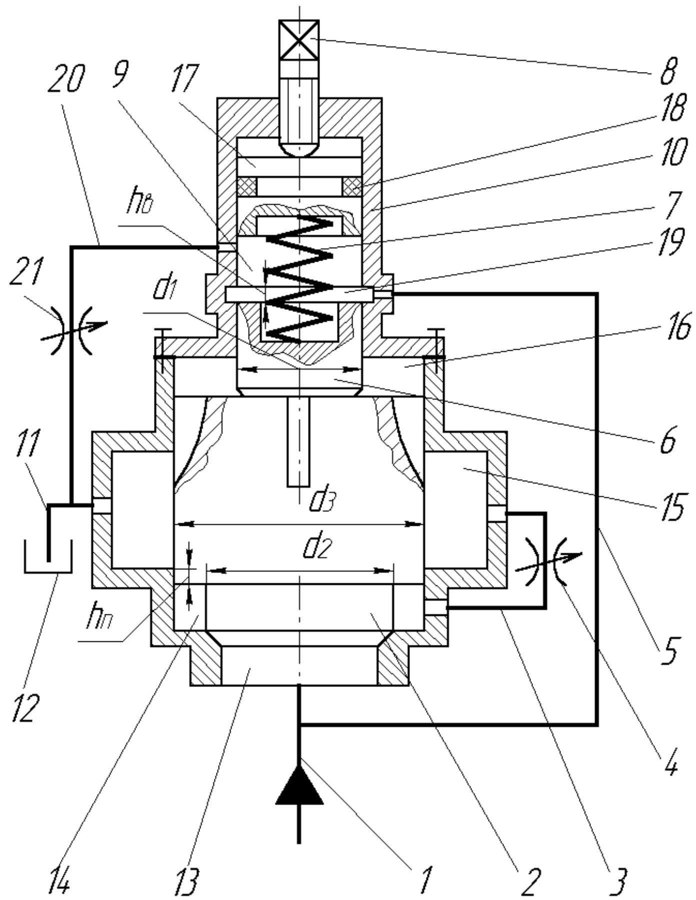


Fig. 1. Circuit of the pressure pulse generator of differential action

Pressure pulse generator of differential action includes valve 2 that is connected with pressure line 1 via under-valve chamber 13. Closed chamber 14, via line 3, communicates with circular bore 15 through variable throttle 4. Circular bore 15 of housing 10 is connected with intermediate chamber 16 through a milled passage in the valve body and with tank 12 via discharge line 11. Above valve 2 spring-loaded pushing plunger 6 is located. Screw 8 is intended for the control of the initial deformation of spring 7 through piston 17 with sealing ring 18. Above-valve chamber 9, via upper circular bore 19, is connected with pressure line 1 by means of line 5. Above-valve chamber 9 is also connected with discharge line 11 by line 20 through additional variable throttle 21.

Pressure pulse generator of differential action functions in the following way. In the initial position valve 2 is under the influence of the resulting force, caused by the difference of forces acting on the side of the valve 2 first stage of smaller diameter and on the side of the plunger 6 that contacts with the body of valve 2 on the other side, i.e. $R_1=P_1-P_2$, where $P_1 = p\pi d_1^2/4 + cx_0$ – pressure force from plunger 6, $P_2 = p\pi d_2^2/4$ – pressure force on the side of the first stage of valve

2 (p – current pressure in pressure line 1; c – strength of spring 7, x_0 – initial deformation of spring 7, created by screw 8 through piston 17, sealed by ring 18). As pressure in pressure line 1 increases, force P_2 is growing and when it exceeds P_1, first stage of valve 2 lifts from the adjusting seat. High-pressure fluid gets into the closed chamber 14 and pressure action is received by the entire cross section of valve 2. As a result, working pressure, created in pressure line 1 at that time, will act on the cross section of valve 2, i.e. on its second stage with diameter d_3. In this case the body of valve 2 will experience the force of the resultant component $R_2=P_3-P_1$, where $P_1 = p\pi d_1^2/4 + c(x_0 + x)$ – pressure force from plunger 6, $P_3 = p\pi d_3^2/4$ – pressure force from the second stage of valve 2, x – displacement of valve 2. As P_3 is higher than P_2, valve 2 will move sharply upwards (according to the drawing) in relation to valve 2. Plunger 6 will cover the upper circular bore 19, shutting off the above-valve chamber 9 from line 5 connected with pressure line 1. Then, under the action of the resultant force $R_3=P_3-P_1$, where $P_1 = c(x_0 + x)$ – pressure force from plunger 6, $P_3 = p\pi d_3^2/4$ – pressure force from the second stage of valve 2, valve 2 will continue its fast upward movement, passing the positive overlap of circular bore 15, connected with tank 12 via discharge line 11. Here, in order to eliminate counteraction to the displacement of plunger 6 and, therefore, also of valve 2, when they are moving upwards, a part of the fluid in above-valve chamber 19 is by-passed to the discharge line 11 via line 20 and additional variable throttle 21. In pressure line 1 pressure will drop to the discharge value p_d that is expedient to be used for intensification of vibration compaction processes of different materials and SDW in particular. Then, under the action of resultant force $R_4=P_1-P_3$ valve 2 moves to the seat of the first stage and effects positive overlap of circular bore 15 ($P_1 = c(x_0 + x)$ – pressure force from plunger 6, $P_3 = p_d\pi d_3^2/4$ – pressure force from the second stage of valve 2, p_d – discharge pressure that will be the same in under-valve chamber 13, closed chamber 14 and above-valve chamber 9 as a result of their being connected with tank 12 via discharge line 11). Then valve 2 moves further downwards passing the overlap of the upper circular bore 19, communicating above-valve chamber 9, via line 5, with pressure line 1. Further movement of the valve is effected by the resultant force $R_5=P_1-P_3$ ($P_1 = p_d\pi d_1^2/4 + c(x_0 + x)$ – pressure force from plunger 6, $P_3 = p_d\pi d_3^2/4$ – pressure force from the second stage of valve 2, p_d – discharge pressure that will be the same in under-valve chamber 13, closed chamber14 and above-valve chamber 9 as a result of their connection with tank 12 via discharge line 11). The rest of the fluid left in the closed chamber 14 is bypassed to the circular bore 15 via line 3 and variable throttle 4. In order to eliminate counteraction to the upward movement of valve 2, intermediate chamber 16 is in continuous communication with circular bore 15 via longitudinal bores. After valve 2 achieves extreme lower position (according to the drawing), pressure chamber 13 is separated from closed chamber 14, which results in further increase of the working fluid pressure required for the next operating cycle that is repeated periodically.

PPGDA belongs to the control equipment of hydraulic drives and can be used in the drives of vibration presses, testing rigs, construction and municipal vibration machines and the like.

Let’s consider PPGDA operation by the example of vibration hydraulic drive of the compaction plate for solid domestic waste (SDW) in garbage truck on the basis of the circuit proposed by the authors [5]. Due to the introduction of the pressure pulse generator into this circuit, vibration compaction method is realized. This makes it possible to increase SDW compaction coefficient. Fig. 2 presents design circuit of the hydraulic drive operation for the vibration method of SDW compaction using pressure pulse generator of differential action. The circuit presents the following geometrical, cinematic and power parameters: p_1, p_2, p_3, p_4 – pressures at the pump output, hydraulic cylinder input and at the filter input; W_1, W_2, W_3, W_4 – volumes of the conduits between the pump and directional control valve, between the directional control valve and hydraulic cylinder input, between the hydraulic cylinder output and directional control valve, between the directional control valve and the filter; Q_P actual flow rate of the pump; S_P – area of the conditional passageway of the directional valve port; S_F– surface area of the filtration element; k_F– specific capacity of the filter (not indicated in the drawing); μ_∂ – dynamic viscosity coefficient (not indicated in the drawing); D, d – diameters of the piston and of the rod; G_{pl} -weight of the compaction plate; G_c – hydraulic cylinder weight; G_{W1} – weight of a part of the waste located above the compaction plate; G_{W2} – weight of a part of the waste located outside the compaction plate (CP); F_{FP} – friction force between CP and the guides; F_W – friction force between SDW and the body; F_C – force developed by the cylinder; h_1, h_2 – heights of the lower and upper parts of CP; b – width of CP (not indicated in the drawing); δ – CP width; α – CP inclination angle; x – CP displacement; y – displacement of PPGDA closing element; d_1 – diameter of the pushing plunger; d_2 – diameter of the second stage of PPGDA closing element; d_3 – diameter of the second stage of PPGDA closing element; $m_к$ – mass of the closing element; c –stiffness of the spring; y_0 – initial deformation of the spring; h_n – positive overlap of the closing element, $h_в$ – overlap (distance from the upper end of the pushing plunger to the upper face of the upper circular bore), d_{thr1} – diameter of the throttle opening; d_{thr2} –diameter of the additional throttle opening .

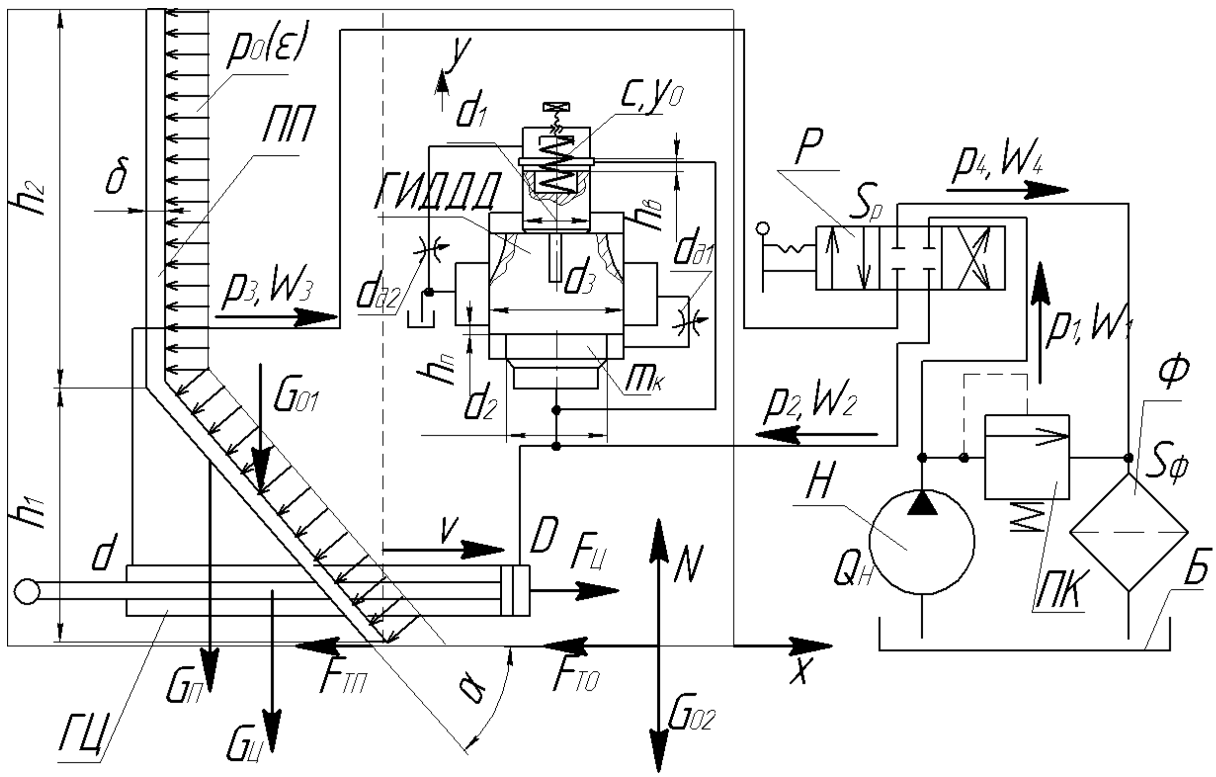


Рис. 2. Design circuit of the hydraulic drive for the vibration method of SDW compaction using pressure pulse generator of differential action

In the development of mathematical model of the garbage truck hydraulic drive operation of SDW compaction using pressure pulse generator of differential action the following assumptions were adopted [2, 3]: SDW compaction pressure depends on its relative deformation and is represented by a power function; movable parts of the working equipment for SDW compaction are assumed to be a single-mass system because CP and hydraulic cylinder body are rigidly connected with each other and hydraulic cylinder rod is rigidly connected with the garbage truck body, its mass being much higher than that of CP and hydraulic cylinder body (m_t=*2500* kg >> $m_{CP}+m_{HC}$=*300 kg*) and so is considered to be immovable; working fluid is assumed to be compressible and is characterized by a compressibility coefficient K; compressibility coefficient of the working fluid (WF) is not changing considerably with pressure change and therefore is considered to be constant; working fluid losses when it flows from high-pressure area to the low pressure area are directly proportional to the pressure differential at the boundary of these areas and is characterized by the working fluid flow coefficient σ; pressure value in the line between the filter and the oil tank can be ignored; dry friction in the movable elements of the hydraulic cylinder and PPG are ignored due to the absence of normal forces in friction pairs where clearance seal is used; general coefficient of SDW friction against steel, which is equal to the simple average of the component coefficients, is proportional to their percentage according to mass.

Taking the assumptions into consideration hydraulic drive operation of SDW compaction can be described by the corresponding systems of differential equations (1 – 6) and algebraic equations (7, 8) with corresponding limit conditions (9). Differential equation (1) describes working fluid losses at the pump – directional control valve section and takes into account actual pump output, WF flow rate through the directional control valve, WF losses while it flows from the high-pressure to the low-pressure area and deformation of the conduits. Differential equation (2) describes WF flow rate on the section between the directional control valve and hydraulic cylinder input and takes into account WF flow rate through the directional control valve, WF consumption for the hydraulic cylinder operation, WF consumption for opening of PPG closure, WF losses while it flows from the high- pressure to the low-pressure area and deformation of the conduits. Differential equation (3) describes WF consumption in the section between the hydraulic cylinder output and the directional control valve and takes into account WF consumption for hydraulic cylinder operation, WF flow rate through the directional valve, WF losses while it flows from the high-pressure to the low-pressure area and deformation of the conduits. Differential equation (4) describes WF consumption in the section between the directional control valve and the filter and takes into account WF flow rate through the directional control valve, WF consumption to open the PPG closure, WF flow rate through the filter, WF losses while it flows from the high-pressure to the low-pressure area and deformation of the conduits.

$$
\begin{cases}
Q_P = \mu S_P \sqrt{\dfrac{2(p_1 - p_2)}{\rho_{WF}}} + \sigma(p_1 - p_2) + KW_1 \dot{p}_1; & (1) \\
S_P \sqrt{\dfrac{2(p_1 - p_2)}{\rho_{WF}}} = \dot{x} S_{C1} + \sigma(p_2 - p_3) + KW_2 \dot{p}_2 + \dot{y}\dfrac{\pi}{4}\left(d_3^2 - 1(h_в - y) d_1^2\right) + & \\
+ 1(y - h_n)\mu\pi d_3 (y - h_n)\sqrt{\dfrac{2p_2}{\rho_{WF}}} + 1(y)\mu\pi\dfrac{d_{D1}^2}{4}\sqrt{\dfrac{2p_2}{\rho_{WF}}} + \mu\pi\dfrac{d_{D2}^2}{4}\sqrt{\dfrac{2p_2}{\rho_{WF}}}; & (2) \\
\dot{x} S_{C2} = \mu S_P \sqrt{\dfrac{2(p_3 - p_4)}{\rho_{WF}}} + \sigma(p_3 - p_4) + KW_3 \dot{p}_3; & (3) \\
\mu S_P \sqrt{\dfrac{2(p_3 - p_4)}{\rho_{WF}}} = k_Ф \dfrac{p_4}{\mu_D} S_F + \sigma p_4 + KW_4 \dot{p}_4; & (4) \\
p_2 S_{C1} - p_3 S_{C2} = m_P \ddot{x} + \pi D L \dfrac{\nu\rho}{\Delta}\dot{x} + p_O(\varepsilon) S_{P1} + Sign(\dot{x})(F_{FP} + F_{SW}); & (5) \\
p_2\left[1(y)\dfrac{\pi\left(d_3^2 - d_2^2\right)}{4} + \dfrac{\pi}{4}\left(d_2^2 - 1(h_в - y) d_1^2\right)\right] = m_к(\ddot{y} + g) + \pi d_3 L_1 \dfrac{\nu\rho}{\Delta_1}(\dot{y})^2 + c(y + y_0); & (6)
\end{cases}
$$

$$
\begin{aligned}
p_O(\varepsilon) = {} & 1774{,}117 + 0{,}09206\rho - 0{,}00257 p_{v1}\frac{S_{C1}}{S_P} - 38\frac{Q_P^2}{S_{C1}^2} - 0{,}4854\frac{x_{\max}^2}{S_P} - 0{,}001576\rho^2 - 63{,}06\frac{ST_{thr1}^2}{S_1^2} - \\
& - 1{,}066\cdot 10^{-10} p_{v1}^2\frac{S_{C1}^2}{S_p^2} + 9813{,}11 + 223{,}1\frac{x_{\max}}{\sqrt{S_P}} - 0{,}8612 p_{v1}\frac{S_{C1}}{S_P} - 8189\frac{Q_P^2}{S_{C1}^2} - 114{,}7\frac{x_{\max}^2}{S_P} - 0{,}03341\rho^2 - \quad ; \quad (7) \\
& - 12806\frac{S_{thr1}^2}{S_1^2} - 2{,}3\cdot 10^{-8} p_{v1}^2\frac{S_{C1}^2}{S_P^2}\,\varepsilon^{5586 - 18{,}6\rho - 1199\frac{Q_P^2}{S_{C1}^2} - 0{,}005134\rho^2 - 1942\frac{S_{thr1}^2}{S_1^2} - 3{,}306\cdot 10^{-9} p_1^2\frac{S_{C1}^2}{S_P^2}}
\end{aligned}
$$

$$
S_{P1} = b(h_1 tg\alpha + h_2);\; F_{Fpl} = f_C\left(G_{O1} + G_P + G_C + p_O(\varepsilon) S_{P2}\right);\; F_{SW} = f_O\left(G_O - G_{O1} + p_O(\varepsilon) S_T\right);
$$

$$
S_{P2} = \frac{bh_1}{tg\alpha};\; S_Б = \frac{(V_K - V)(b + 2(h_1 + h_2))}{b(h_1 + h_2)};\; G_{O1} = \frac{h_1 b(h_2 + h_1/2)\rho_O g}{tg\alpha};\; G_P = \left(h_2 + \frac{h_1}{\sin\alpha}\right) b\delta\rho_C g;
$$

$$
G_C = (m_C + S_{C1} x \rho_{WF}) g;\; G_O = V_K \rho_O g;\; m_P = (G_O + G_P + G_C)/g;\; S_{C1} = \pi D^2/4;\; S_{C2} = \pi\left(D^2 - d^2\right)/4; \quad (8)
$$

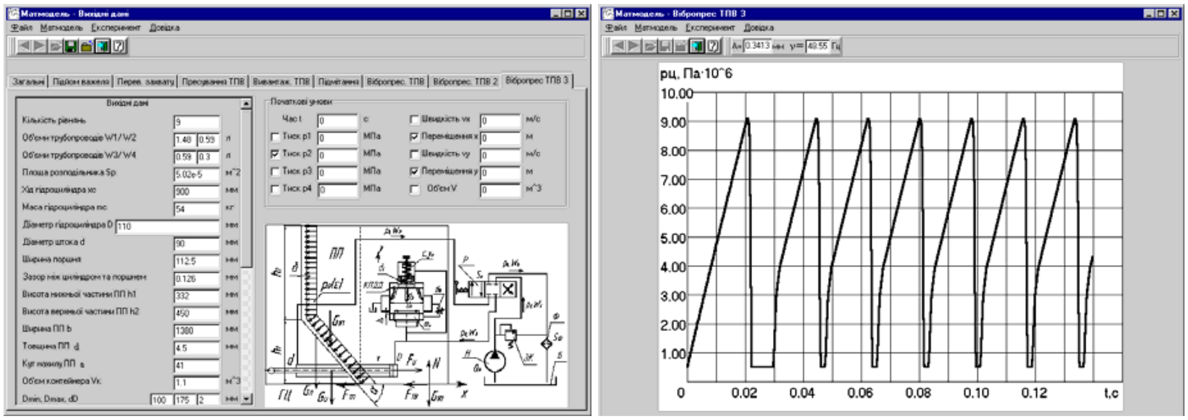
$$0 \le \{p_1, p_2, p_3, p_4\} \le p_{pv};\ 0 \le x \le x_{\max};\ 0 \le y \le y_{\max}, \qquad (9)$$

where **1**() – unit function; *Sign*() – sign unction; p_{pv} – response pressure of the pressure relief valve; x_{max} – maximal stroke of the hydraulic cylinder; y_{max} – maximal displacement of the PPG closing member to the stop; Δ_1 – value of the clearance between the closing member and PPG body; L_1 – length of the PPG closing member friction surface.

Differential equation (5) describes the compaction plate movement and takes into account the force developed by the hydraulic cylinder, inertia forces of the movable elements, viscous friction force; SDW reaction force in the process of compaction, force of the viscous friction of the compaction plate and SDW. Differential equation (6) describes the movement of the PPG closing member and takes into account the force created by pressure p_2, closing member inertia force, viscous friction force, spring elasticity force, weight of the closing member. Dependence (7) of SDW compaction pressure on its relative deformation ε, main parameters of the vibration compaction hydraulic drive being taken into account, was obtained experimentally [3].

System of differential equations (1 – 6), describing the dynamics of the garbage truck hydraulic drive during the operation of SDW compaction using PPGDA, is the system of ordinary differential equations that cannot be solved for higher derivatives. Besides, this mathematical model contains significant non-linearities. Non-linearity of these equations is manifested in that functions p_1, p_2, p_3, p_4 to be found are included into the differential equations in the form of expressions with fractional powers. In addition, some of the equations contain complex relationships that cannot be linearized by the ordinary expansion into a Taylor series. Presence of the logical functions in the above-mentioned equations (sign functions and a unit function) also complicates their analytical solution. The dependences of friction coefficients on the speed of the movable elements of hydraulic drives being investigated belong to the significant non-linearities, the functions of which have broken continuities. At present we do not know analytical methods for the transformation of the differential equations systems (1 – 6) to linear form.

The impossibility to solve the equations for higher derivatives and presence of significant non-linearities make it possible to draw a conclusion about the necessity to use numerical methods for the solution of the differential equation systems. So for the solution of differential equation systems (1 – 6) the method of Runge-Kutta-Fehlberg was used, that changes automatically the integration step while searching for computational errors, which provides higher accuracy of computations. This numerical method is realized in Delphi programming environment that enables numerical solution of the differential equations (1 – 6) and obtaining results in the form of graphs and tables. General view of the dialog window of the "Mathmodel" program for the investigation of the garbage truck collector hydraulic drive dynamics during SDW compaction operation using PPGDA is shown in fig.3.

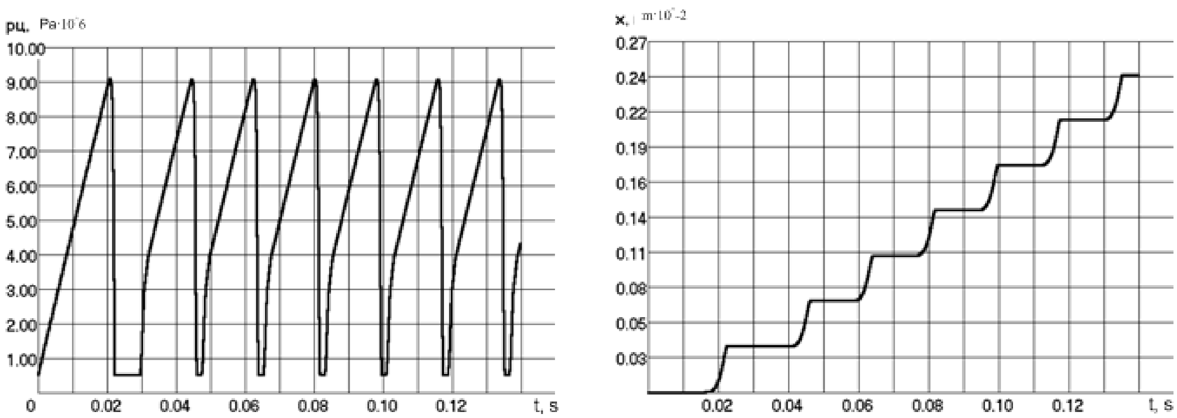


a) б)

Fig. 3. General view of the dialog window of the "Mathmodel" program for the investigation of the garbage truck hydraulic drive dynamics during SDW compaction operation using PPGDA: entering initial data (a); investigation results (b)

In the course of investigation the calculations were performed with integration step $h=10^{-4}$ and relative error $\varepsilon=10^{-16}$. Stability of the solution of differential equations systems was assured by checking the identity of results obtained for h values of integration steps and for the values of integration half-steps *h/2*.

During computer simulation, using the method of Runge-Kutta-Fehlberg and above-described mathematical model, results of the numerical investigation were obtained as to the processes occurring in the hydraulic drive during vibration compaction of SDW using PPG of differential action. Research results are shown in fig. 4. The created program enables automatic computation of vibration amplitude and frequency.



a) b)

Fig. 4. Pressure variations in the hydraulic cylinder pressure chamber p_c (a) and movements of the compaction plate *x* (b)

Fig. 5 presents the investigation results concerning the influence of $h_{в}/h_{п}$ relationship on vibration frequency *v*. On the results of the analysis of this relationship a conclusion can be made that for $h_{в}/h_{п}$=0,5…2 a 32,1%…10,2% increase of the vibration frequency can be observed. It was found that the best calculation method for $v=f(h_{в}/h_{п})$ relationship is quadratic regression equation given in the free space of fig. 6. In his case squared correlation coefficient has made R^2=0,9788, which is the evidence of adequacy of the obtained regression equation that can be used for the development of engineering calculation procedure for PPGDA parameters.

Subsequent increase of $h_в/h_n$ relationship does not lead to the growth of vibration frequency. As a range of rational $h_в/h_n$ relationship values $h_в/h_n$=0,5…1 could be recommended, which provides essential growth of vibration frequency (32,1%…20,9%), that is expedient to be used for intensification of vibration compaction processes of different materials and SDW in particular.

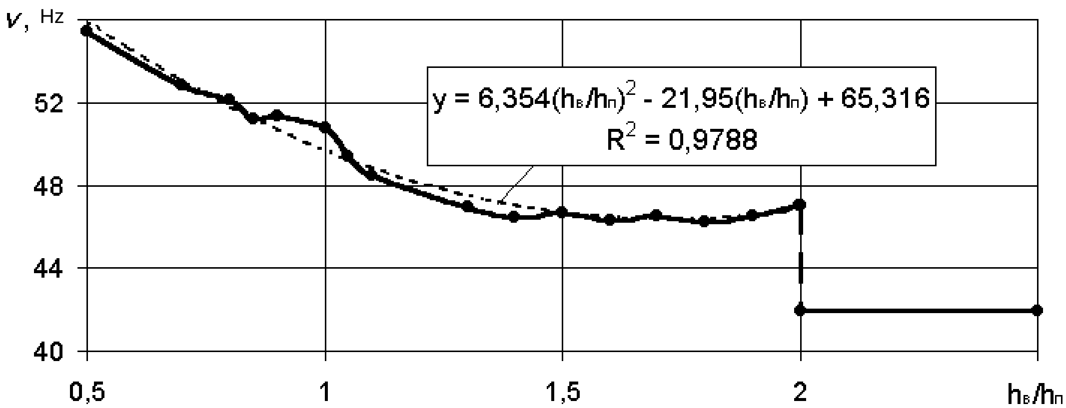


Fig. 5. Influence of $h_в/h_n$ relationship on vibration frequency ν

Conclusions

1. New pressure pulse generator of differential action (PPGDA) is designed. This new design (granted a Useful Model Patent of Ukraine 29363 U) makes it possible to increase vibration frequency of the working equipment of vibration machines on the basis of hydraulic drives, the rest of parameters being the same.

2. Mathematical model of vibration hydraulic drive for SDW compaction using pressure pulse generator of differential action is suggested. The model makes it possible to study the dynamics of said hydraulic drive in order to choose more rational parameters of its design.

3. On the basis of mathematical modeling the range of rational values for the relationship $h_в/h_n$=0,5…1 has been found, which provides significant increase (32,1%…20,9%) of vibration frequency ν that is expedient to be used for intensification of vibration compaction processes of different materials and SDW in particular.

4. It was determined that $\nu=f(h_в/h_n)$ relationship is adequately described by the quadratic regression equation that can be used for the development of engineering design procedure for PPGDA parameters calculation.

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