Substantiation of the methodology for calculating the design of a small-sized hydraulic pulse vibrator

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Abstract. A methodology for the design calculation of a new design of a hydraulic pulse vibrator with a valve pressure pulse generator of a parametric type has been developed, which allows determining its energy, power and geometric parameters.

Keywords: hydropulse device, calculating, frequency, amplitude, small-sized vibrator.

1. Introduction

One of the ways to develop vibration technology is to minimize the size without deteriorating the technical characteristics [1]. Analysis of the results of studies on the development of vibrators with different types of drives [1-8] has established that vibrators based on a hydraulic pulse drive using elastic elements of high stiffness [1, 9] have the smallest dimensions with significant oscillating power parameters. There is no generally accepted methodology for calculating the design of hydro-pulse vibrators, so it is necessary to carry out research to substantiate the methodology for calculating the corresponding new design of the vibrator in order to obtain the best technical and economic indicators.

2. Researches methodology

The methodology for calculating the design of a small-sized hydraulic pulse vibrator will be considered for the device whose structural and design scheme is shown in Fig. 1 [1, 10]. A feature of the design is the compactness of the structure due to the use of elastic elements of high rigidity and the construction of the structure based on a hydraulic pulse drive.

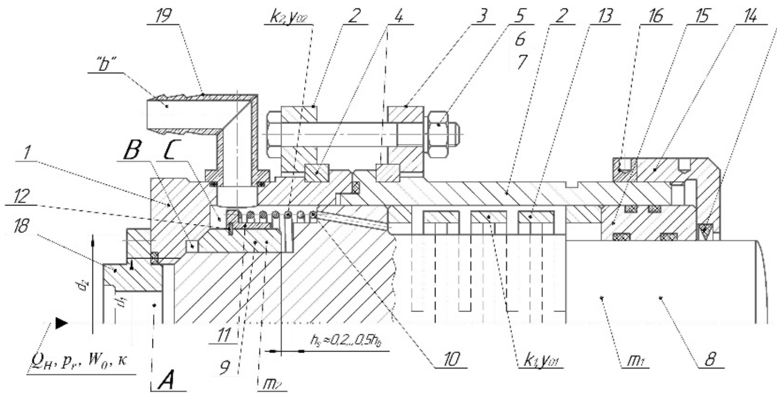

Fig. 1. Design scheme of a small-sized hydraulic pulse vibrator with a built-in valve pressure pulse generator

Hydraulic pulse vibrator with valve sealing stages of the pressure pulse generator (PPG), containing energy supply and discharge lines, cover 18, pressure pulse generator body 1, which is connected to the sleeve 2 by means of half rings 4, flanges 2 and 3, which are tightened through half rings 4 with bolts 5 and nuts 6, secured with spring washers 7. Plunger 8 on the valve part of which there is a valve sleeve 9, which is supported against the chamfer of the body 1 by a split spring ring 12, a stepped sleeve 11, coiled 10 and slotted springs13, a guide sleeve 15 and a cap nut 14, which is secured with a lock nut 16 and in which a dirt collector 17 is installed.

The maximum frequency of the pressure pulses $\nu_{max} = T_T^{-1}$ is determined by the power supply Q_H of the hydraulic pump of the hydraulic pumping station supplying the vibrator hydraulic system. The theoretical value of the flow rate Q_{H_T} can be found by the following formula [11]:

$$Q_H = K_{CT}\nu_{max} \cdot p_{1max}W_0\kappa^{-1} \cdot \eta_{OH}^{-1}, \tag{1}$$

where K_{CT} – theoretical cycle coefficient of pressure pulses; W_0 – the initial volume of the pressure cavity A of the hydraulic system of the vibrator; p_{1max} – maximum "opening" pressure PPG; κ – modulus of elasticity; η_{OH} – volumetric efficiency of the hydraulic pump (for a hydraulic pump of the gear pump $\eta_{OH} = 0{,}95...0{,}96$ [9, 10]). Since the cyclogram of the vibrator operating cycle is indicative (conditional), the K_{CT} has an estimating nature, which necessitates the introduction of a margin factor in Eq. (1) K_{SH}, the value of which can be refined based on the results of experimental studies based on the recommendations of [10], we accept $K_{SH} = 1.1$,..., 1.25, then the calculated value:

$$Q_H = K_{SH}Q_{H_\text{T}} = K_{SH}K_{CT}\nu_{max} \cdot p_{1max}W_0\kappa^{-1} \cdot \eta_{OH}^{-1}. \tag{2}$$

We will find the estimated value K_{CT} from the approximate cyclogram of the working cycle, using the concept of the scale of the pressure pulse $\mu_{tp} = T_T/00_1 = 00_1 \cdot \nu_{max}$ sec/mm (Fig. 2).

By measuring the segments on the indicative cyclogram of the vibrator's duty cycle $0a'$, $a'b'$, $b'c'$ and 00_1 (see Fig. 2), define:

$$t_H = \mu_{tp} \cdot oa', \quad t_{\text{BT}} = \mu_{tp} \cdot a'b', \quad t_{CT} = \mu_{tp} \cdot a'c, \quad t_{HT} = \mu_{tp} \cdot co', \quad T_T = \mu_{tp} \cdot oo'. \tag{3}$$

Taking into account Eq. (3) in the formula for $K_{CT} = [1 + (t_{BT} + t_{SH} + t_{HT})/t_H]$, calculate:

$$K_{CT} = 1 + \frac{a'b' + b'c + co'}{oa'}. \tag{4}$$

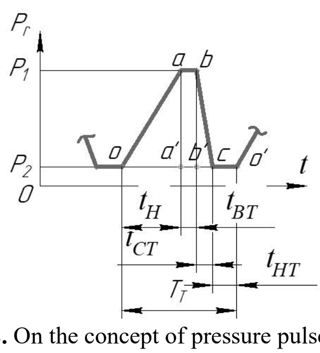

Fig. 2. On the concept of pressure pulse scale

According to the approximate cyclogram we have proposed: $\mu_{tp} = (00_1\nu_{max})^{-1} = (100\cdot32)^{-1}$ = 0.0003125 sec/mm, then, according to Eq. (4), $K_{CT} = 1.88$. Taking into account this value in Eq. (2) K_{CT} and K_{SH} and $\eta_{OH} = 0.95$,..., 0.96, find:

$$Q_H = (2{,}16\ldots2{,}46)\nu_{max} \cdot p_{1max} \cdot W_0 \cdot \kappa^{-1}, \tag{5}$$

where the average value is taken $\eta_{OH} = 0.955$.

To calculate the required cross-sectional area of a fully open PPG vibrator $A_{gap_{max}} = \pi d_2 h_b$, it is necessary to know the average energy flow rate through this gap $Q_{mH\Sigma}$ during the time t_{CT} of reduction of the energy carrier pressure in the hydraulic system of the vibrator from level p_1 to level p_2 (reduction of the deformation of the hydraulic link from x_{01} to x_{02}). By analogy with the known dependence [12], the time t_{CT} can be estimated by the formula:

$$t_{CT} = \frac{(p_1 - p_2)W_0}{Q_{mH\Sigma} \cdot \kappa}. \tag{6}$$

Whence, under the assumption of linearity of the function $p_r = f(t)$ [12] define:

$$Q_{mH\Sigma} = Q_H \cdot \frac{t_H}{t_{CT}} = Q_H \cdot \tau_{CT}, \tag{7}$$

where $\tau_{CT} = t_H/t_{CT}$ – is the relative time of energy carrier pressure reduction in the pressure cavity A of the vibrator (see Fig. 1) from level p_1 to level p_2. Since, according to the approximate cyclogram of the vibrator's operating cycle $t_{CT} < t_H$, then τ_{CT} >1 and $Q_{mH\Sigma} > Q_H$. During theoretical studies of the mathematical model of the vibrator, it is advisable $Q_{mH\Sigma}$ to coordinate the value with the calculated value $Q_{H\Sigma}^{max}$.

To avoid such negative phenomena as cavitation during the operation of the PPG vibrator, it is necessary to limit the speed V_{mH2} of the energy carrier through the gap $A_{gap_{max}}$ to the permissible one $[v_{H2}]$:

$$V_{mH2} = \frac{Q_{mH\Sigma}}{\pi d_2 h_8} \leq [V_{H2}]. \tag{8}$$

The energy carrier flow $Q_{m\Gamma\Sigma}$ also passes through the open gap $A_{1gap_{max}}$of the locking element of the first level of PPG sealing with a speed of:

$$V_{mH1} = \frac{Q_{mH\Sigma}}{\pi d_1 h_{b1}} \leq [V_{H2}], \tag{9}$$

where h_{b1} – negative opening of the closing element (plunger 8) of the first sealing level, as can be assigned $h_{в1}$=$h_в$, but provided that $V_{mH1} > V_{mH2}$, for $[V_{H1}] > [V_{H2}]$, for example, by assuming the flow rate of the energy carrier through the gap $A_{gap1_{max}} = \pi d_1 h_{b1}$ such as safety valves [12], or $h_{b1} \approx h_b + h_S = h_b + 0{,}4h_b = 1{,}4h_b$, then we can assume that $[V_{H1}]$=$[V_{H2}]$. When developing the mathematical model of the vibrator, we assumed that $0 \leq y_{1d} \leq h_b$ and $0 \leq y_{2d} \leq h_b$. The validity of the above assumptions should be verified during theoretical studies of the mathematical model of the vibrator and experimental studies of its prototype. As a certain margin of safety, let's take $h_{b1} = 1{,}4h_b$ and $[V_{H1}] = [V_{H2}]$, from Eqs. (8) and (9) for $[V_{m\Gamma2}] = [V_{\Gamma2}]$ and $[V_{mH1}] = [V_{H1}]$, find:

$$d_2 = 1{,}4d_1. \tag{10}$$

3. Results and discussion

According to the assumption structure adopted by us, as in [9, 13, 14], in the guides of the plunger 8 and the valve sleeve 9 (see Fig. 1) there is only a liquid friction mode, then, accordingly, the energy balance of the forward stroke of the locking elements of the first and second sealing stages of the PPG vibrator is determined by equation:

$$A_{mp} \geq \Delta E_{PE} + \Delta E_{SC} + \Delta E_{HL} + A_f, \tag{11}$$

where A_{mp} – average work of energy carrier pressure forces during the direct stroke of the mass $m_{1\Sigma}$ (plunger 8 and valve sleeve 9, see Fig. 1); ΔE_{PE} – increase in potential deformation energy of the slot spring 13; ΔE_{SC} – increase in the potential strain energy of the coiled spring 10 (see Fig. 1); ΔE_{HL} – increase in potential deformation energy of the elastic part of the hydraulic link; A_f – the total average work of friction forces during the straight stroke movement of the plunger 8 and the valve sleeve 9:

$$A_{mp} = p_{1max} h_b A_2 \tag{12}$$
$$\Delta E_{PE} = 0{,}5 k_1 h_b^2, \tag{13}$$
$$\Delta E_{SC} = 0{,}5 k_2 h_S^2, \tag{14}$$
$$\Delta E_{HL} = 0{,}5 k_{or} x_{01}^2 = 0{,}5 p_{1max}^2 A_0^2 k_{or}^{-1}, \tag{15}$$
$$A_f = F_{f_1} h_b. \tag{16}$$

According to the reasonable structure of assumptions adopted by us, the friction mode in the guides of the plunger 8 and the valve sleeve 9 is fluid, and $k_2 \ll k_1$, which allows us to neglect the components ΔE_{HL} and A_f in the energy balance Eq. (11) and write this equation in the form of inequality:

$$A_{TP} \geq \Delta E_{PE} + \Delta E_{HL}, \tag{17}$$

or taking into account Eqs. (12), (13) and (15):

$$p_{1max} h_b \cdot A_2 \geq 0{,}5 k_1 h_b^2 + 0{,}5 p_{1max}^2 A_0^2 \cdot k_{0r}^{-1} = 0{,}5 k_1 h_b^2 + 0{,}5 p_{1max}^2 \kappa^{-1} \cdot W_0, \tag{18}$$

where from:

$$A_2 \geq 0{,}5 (k_1 h_b p_{1max}^{-1} + p_{1max} \kappa^{-1} h_b^{-1} W_0), \tag{19}$$

or:

$$d_2 \geq 0{,}798 (k_1 h_b \mathrm{p}_{1max}^{-1} + \mathrm{p}_{1max} \kappa^{-1} h_b^{-1} W_0)^{0{,}5}, \tag{20}$$

where $A = \pi d_2^2/4 \cong 0{,}785 d_2^2$; A_0, k_{0r} – see Eq. (13) and (23); sign ≥ takes into account the neglect of components ΔE_{SC} and A_f in the Eq. (11).

The average diameter d_1 of the first level of sealing of the PPG vibrator is found from the Eq. (10): $d_1 = 0{,}714 d_2$.

The throughput of the vibrator PPG according to Eq. (8) depends $A_{gap_{max}} = \pi d_2 h_b$ on the energy carrier flow rate $V_{H2} \leq [V_{H2}]$ and is determined by the diameter d_2 and negative overlap h_b:

$$h_b \geq \frac{Q_{mH\Sigma}}{\pi d_2 [V_{H2}]}, \tag{21}$$

or taking into account Eq. (7):

$$h_b \geq \frac{Q_H \tau_{CT}}{\pi d_2 [V_{H2}]}. \tag{22}$$

The relative time τ_{CT} for reducing the pressure of the energy carrier in the pressure cavity A of the vibrator can be estimated based on the results of the analysis of experimental studies of the hydraulic pulse drive of vibrating and vibration-impact machines (oscillograms of changes in the pressure of the energy carrier in the pressure cavities of the hydraulic pulse drive [13-14]. According to this analysis, the average value is τ_{CT} = 2.3,..., 2.8 for the average frequency range of pressure pulses ν = (20...100) Hz and amplitude of $\Delta p = p_1 - p_2 =$ (9,..., 10) MPa. In accordance with the comments made, we will have:

$$h_b \geq \frac{(2{,}3\ldots2{,}8)Q_H}{\pi d_2[V_{H2}]} = \frac{K_{SR}(2{,}3\ldots2{,}8)Q_H}{\pi d_2[V_{H2}]}, \tag{23}$$

where K_{SR} = 1.1,…, 1.2 – reserve factor, which takes into account the indicative nature of the calculation h_b and $Q_{mH\Sigma}$. Taking into account the value of K_{SR}, find:

$$h_b = \frac{(0{,}8\ldots1{,}07)Q_H}{d_2[V_{H2}]}. \tag{24}$$

The cross-sectional area of the fitting screwed into the threaded hole of the cover 18 (Fig. 1) and the cross-sectional area of the opening of the high-pressure hose connecting the pressure cavity A of the vibrator to the hydraulic pumping station supplying the vibrator with energy carrier Q_H must be sufficient to allow the flow of energy carrier at a rate of $V_A \leq [V_{H1}]$ (or $[V_{H2}]$, as we have accepted that $[V_{H1}] = [V_{H2}] = [V]$):

$$V_A = \frac{Q_H}{A_A} = \frac{4Q_H}{\pi d_A^2} \leq [V], \tag{25}$$

where d_A – the diameter of the hole in the high-pressure hose, which can be considered the conditional passage of the vibrator. From Eq. (25) for $V_A \leq [V]$, we obtain:

$$d_A = \sqrt{\frac{4Q_H}{\pi[V]}} \approx 1{,}13\sqrt{\frac{Q_H}{[V]}}. \tag{26}$$

Through the open slot $A_{gap_{max}}$ of the first stage of sealing of the PPG of the vibrator passes the flow $Q_{mH\Sigma}$ of energy with a permissible speed $[V_{H1}] = [V]$, it is obvious that at this speed this flow of energy must pass through the cross-sectional area A_1, then from the equation of equality of speeds of the flow of energy:

$$[V] = \frac{4Q_{mH\Sigma}}{\pi d_1^2} = \frac{4Q_H}{\pi d_A^2}, \tag{27}$$

find:

$$d_1 = (1{,}52\ldots1{,}67)d_A, \tag{28}$$

where previously (see the text to (23)) $Q_{mH\Sigma} = (2{,}3 \ldots 2{,}8)Q_H$.
Substituting Eq. (28) into (10), we have:

$$d_2 = (2{,}13\ldots2{,}34)d_A. \tag{29}$$

Taking into account Eq. (29) in (23), we finally find:

$$h_b \approx \frac{(0{,}34\ldots0{,}50)Q_H}{d_A[V]}. \tag{30}$$

Taking $F_{TO} = F_{Tmax}$, to turn the inequality into an equality, we determine the required stiffness of the slotted springs 13, taking into account Eq. (28) (see Fig. 1):

$$k_1 = [(1{,}81\ldots2{,}19)p_{1max}d_A^2 - F_{Tmax}] \cdot y_{01}^{-1}. \tag{31}$$

In order for the coiled spring 10 (see Fig. 1) to work "in time" with the dynamic oscillatory process of the vibrator, its mode of operation must be resonant, which is determined by the condition [13]:

$$\omega_{02} = \sqrt{k_2 m_{1\Sigma}^{-1}} \geq \sqrt{2} \cdot 2\pi\nu_{max} \approx 8{,}88\nu_{max}, \tag{32}$$

whence $k_2 \geq 8\pi^2 m_{1\Sigma}^{-1}\nu_{max} \approx 78{,}85 m_{1\Sigma}^{-1}\nu_{max}$.

The initial force $F_{p10} = k_{02}y_{02max}$ of the spring 10 is determined y_{02max} by the deformation during the assembly of the vibrator, and the working force $F_{p10} = k_2(y_{02max} + h_s)$ is determined by the fact that during most of the forward and reverse strokes of the plunger 8 and the valve sleeve 9 they move as one ($m_{1\Sigma}$). Other geometrical dimensions of the vibrator are determined during the development of the vibrator design according to the generally accepted rules for the design of hydraulic and, in particular, hydraulic pulse machines, mechanisms and devices.

The method has been tested for a vibrator with a nominal "opening" pressure of PPG – 10 MPa, an amplitude of oscillations – 0.5,..., 2,0 mm and a frequency of passage of pressure pulses – 10,..., 100 Hz.

4. Conclusions

1) The developed method of design calculation of a hydraulic pulse vibrator with a valve PPG allows determining all the main energy, power and geometric parameters of the vibrator using relatively simple formulas.

2) According to the results of theoretical studies of the mathematical model of the vibrator after checking the adequacy of this model, correlating (clarifying) coefficients can be introduced into the calculation dependencies and formulas of the developed methodology for the design calculation of the vibrator in order to increase their correctness and accuracy.

3) The principles and approaches used in the developed design calculation methodology for determining the vibrator parameters can be the basic basis for constructing design calculation methods for other hydroimpulse vibrators and devices.

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Data availability

The datasets generated during and/or analyzed during the current study are available from the corresponding author on reasonable request.

Conflict of interest

The authors declare that they have no conflict of interest.

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