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The Dynamic Vibrations Hydraulic Quencher

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Summary. A review and analysis were carried out of the developed hydraulic system for quenching dynamic oscillations. The mathematical model was made for determining the time delay in operation of the hydraulic system for dynamic quenching of oscillations. A calculation was made of soil cleavage period and operation time delay of dynamic oscillations quencher, from which it is possible in theory to establish the ability of the hydraulic system for dynamic oscillations quenching to operate in time. The hydraulic system was developed for dynamic oscillations quenching inside the working unit to prevent the transmission of vibrations to the base of the machine. An analysis was performed of the damper's means and the method of dynamic damping on which the hydraulic system was developed for dynamic oscillations quenching. The mathematical sequence was built for the determination of operation time delay of the quencher. The parameters have allowed to construct the experimental model of hydraulic system for dynamic oscillations quenching.

Key words: quencher, dynamic oscillations, hydraulic system, time lag response, soil cleavage period.

INTRODUCTION

The construction often includes works that cannot be performed by conventional machines. In this case, using a special technique, which is equipped by active action engines and used for ground operations can be consider as a solution.

One of the disadvantages of this special technology is the transmission of vibrations from the working body to base machine, accompanied by the premature wearing down of the basic machine parts which do not participate in the ground destruction.

To solve the above-mentioned problem amortizable equipment can be used to isolate vibrations caused by the working body movement from the base machine.

Given that the current cushioning devices are not effective enough, there is demand for the construction of a new hydraulic system for dynamic oscillations quenching as well as mathematical model for the determination of the main characteristics of the new system, namely: determining the time delay in the operation of the hydraulic system for dynamic oscillations quenching, its evaluation, timeliness, triggering of specified dynamic fluctuations.

PURPOSE OF WORK

Based on the results of the dynamic analysis of the method and vibration cushioning devices the work aimed to develop:

- A mathematical model for determining the time delay in the operation of the hydraulic system for dynamic oscillations quenching;
- A hydraulic construction developed by the authors for extinguishing dynamic oscillations that arise inside the working part in order to make their transmission to the base machine impossible.

THE MAIN MATERIAL

In engineering it is often necessary to damp the vibration transmitted to the machine from its working equipment. Basically, it concerns machines with dynamic (active) working tools. As a result of work, the vibrations can be transmitted to the machine and cause destruction. To prevent transmitting the vibrations from the working body to the base machine the so-called springy elements are used, which try to put out the vibrations transmitted to the base machine. The main method of vibration damping is called – a method of dynamic vibration damping [3, 4, 5, 15, 19].

The method of dynamic vibration damping is based on the application of additional protective devices to change the vibration properties in a given object. The work of the dynamic quencher is based on putting out the force transmitted to the object. This dynamic quenching differs from another method of reducing vibration, characterized by imposition of the additional kinematic object linkages, such as fixing some of its points [2, 19].

Changing the vibrating condition of the base machine when joining a dynamic quencher can be achieved by the redistribution of vibrational energy from the object to the quencher and in the direction of the increasing energy dissipation fluctuations [13, 20]. The former is implemented by changing the system's settings of the object-quencher over the frequencies of the vibration excitation by correcting the elastic-inertial properties of the system. In this case, the devices attachable to the object are called inertial dynamic quenchers. An inertial quencher is used to suppress harmonic mono or narrowband random fluctuations.

During a vibration's load of the wider frequency range the second method is preferred. It is based on increasing dissipative properties of the system by attaching additional specific damping elements to the object. Dynamic quencher dissipative types are called vibration absorbers [3]. The combined ways of the dynamic quenching using both correction elastic-inertial and dissipative properties of the system are also possible. In such cases we talk about the dynamic quencher of friction [7, 16].

When implementing dynamic quencher counteraction fluctuations the object is affected by reactions that are passed to it from the attached bodies. For this reason, considerable effects with limited amplitudes of the masses adjusted can be achieved only by large mass (moment of inertia) of connected bodies, typically ≈ 5...20% of the reduced mass (moment of inertia) primary system with an appropriate form of vibrations within the frequency quenching which is performed, respectively [3].

Typically, a dynamic quencher is used to achieve a local effect: a reduction of the object's vibration in the parts where the quencher is fastened. Often it may be associated even with the deterioration of the object's vibration in the other less appropriate places.

a *b*

Fig. 1. The overall look of shock absorbing passive elements: *a* – spring (of vehicle), *b* – spring

Dynamic quencher can be constructively implemented based on the passive elements (Fig. 1) (masses, springs, shock absorbers, dampers) and the active ones, which have their own power sources.

In the last case we are talking about the use of automatic control units that use elements driven by an electric, hydraulic or pneumatic system. Their combination of passive devices is successful, an example of which is the shock absorber shown in Fig. 2.

Fig. 2. General view of the shock absorber

The utilization of the active elements expands the possibilities of dynamic vibration suppression, because it allows to conduct continuous adjustment of the parameters of dynamic quencher as a function of excitation acting, and thus to perform quenching in conditions of changing vibration loads. A similar result can be achieved sometimes by means of passive devices with nonlinear characteristics.

Shock absorber is the device which converts mechanical energy into thermal and is used for vibration damping and shock absorption. It bumps acting on the casing (frame). Shock absorbers are used in conjunction with elastic elements: springs (of a vehicle), pillows, etc.

Hydraulic shock absorbers have become the most commonly used. In a hydraulic shock absorber the resistance force depends on the speed of rod movement. Oil is the working medium. The principle of operation of the shock absorber is based on the reciprocating movement of the piston shock absorber, which through a small hole puts oil from one chamber to another, converting mechanical energy into the heat energy.

Today, the generally used solutions for vibration damping devices are those which utilize the hydraulic elements. Hydraulic dampers as opposed to friction ones, have the longer duration of work and can quench a small oscillation amplitude.

Fig. 3 shows the design of the damper, which contains an adjusting screw. This allows to increase the resistance value of the liquid flowing through the channel and thus control the damping, according to [9, 12, 13].

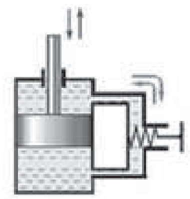

Fig. 3. Liquid damper's adjusting screw

Nowadays, the most promising system is the hydraulic vibration damping one, which is designed based on hydraulic shock absorbers and dampers.

In the design of hydraulic systems for blanking the oscillations the dynamic characteristics of systems must be considered, in particular, the transfer speed signals and the total system performance, pressure fluctuations in various points of the system (including hydraulic shocks), sustainability and quality of system transients [1, 11].

Movement of actuating mechanism always comes with some delay in relation to the input signal. The identification of the delay's amount allows to determine the system's dynamic, the total response time and the need to introduce appropriate units to compensate for the delay calculation, depending on the frequency control signal and set according to the corresponding pulses [8, 17, 9].

To perform the system's calculations it is necessary to know the basic system parameters, including the size of pipelines, hydraulic and mechanical resistance properties of the working fluid and hydraulic machines as well as hydraulic power source characteristics [6].

Total delay time of the system's response can be defined as a first approximation by the formula:

$$t_d = \frac{\Delta V + V_1}{Q_h + 0,5Q_b}, \tag{1}$$

where:
ΔV – reduces the volume of liquid in the system by increasing the pressure on the value of Δp, m^3, V_1 – volume of fluid required to fill additional volumes in the system, m^3, Q_b – leak in the system for working pressure, m^3/s, Q_h – the nominal flow rate in the system, m^3/s.

In this case Q_h and V_1 are determined from the formula:

$$Q_h = \frac{17,1N_h}{P_h}, \tag{2}$$

where:
N_h – hydraulic drive power, kW, P_h – nominal pressure of hydraulic system MPa.

The volume of the fluid required to fill the additional volume in the system V_1 is 5...10% of the total volume of fluid in the hydraulic system V, m^3. The total volume of fluid in the hydraulic system is calculated as follows:

$$V = V_c + V_e, \tag{3}$$

where:
V_e – the amount of hydraulic fluid that is equipped in hydraulic system, m^3, V_c – the amount of hydraulic fluid that is in the pipeline hydraulic system Eq. (3), which is calculated by the equation:

$$V_c = \frac{\pi D^2}{4} L, \tag{4}$$

where:
L – total length of pipelines, m, D – internal diameter hydraulic of the pipeline, m^2, calculated by the formula:

$$D = 4,5\sqrt{Q_h / W}, \tag{5}$$

where:
W – speed liquid in the hydraulic system at a prescribed pressure, m/s.

$$V_1 = (0,05...0,1)V. \quad (6)$$

At first approaching the delay operation of the system Eq. (1) we obtain the equation:

$$t_d = \frac{\Delta V + V_1}{Q_h} \cdot \frac{1}{1 - \frac{Q_b}{2Q_h}}. \quad (7)$$

In the view of that:

$$Q_b = K_b P, \quad (8)$$

where:
P – operating pressure in the system, MPa, K_b – the coefficient of leakage of liquid, received from the equation:

$$K_b = j\frac{61,2N_h}{P^2}, \quad (9)$$

where:
j – coefficient of the changes in the units of measurement of l/min in m^3/s and 0.278 matter.

In this case, reduction in the volume of liquid in the system while increasing pressure by the value of Δp is calculated as follows:

$$\Delta V = \delta S_1 L, \quad (10)$$

where:
δ – coefficient of decrease of the liquid, which depends on the operating pressure; S_1 – cross-section inner diameter hydraulic of the pipeline, m^2.

Moreover, S_1 is calculated as follows:

$$S_1 = \frac{\pi D^2}{4}. \quad (11)$$

Finally, after determining equations will get simplified the delay triggering and performance will look like:

$$t_d = \frac{\delta S_1 L + V_1}{Q_h - 0,5K_b P}. \quad (12)$$

From the dependence it is obvious that to reduce the time delay triggering it is necessary:

1. Working channels and pipelines should be as short and rigid as possible;
2. Volume losses should be lowered to a minimum;
3. Pump capacity should be significant.

In general, the performance of the dynamic oscillations quencher is determined for each particular system, provided that the signal can be transmitted with a specific delay, but performance must be such as not to violate the stability of the whole circuit [18].

To achieve the stated conditions, we need to find out the time of soil cleavage, i.e. the time at which the dynamic ripper makes one complete cycle of the movement T_c [14], which is inverse to the average oscillation frequency maxima of cutting the soil, and is given by:

$$T_c = \frac{1}{\bar{n}_m} \text{ s}, \quad (13)$$

$$\bar{n}_m = \frac{\bar{n}_0}{0,63...0,87} \text{ 1/s}, \quad (14)$$

where:

$$\bar{n}_0 = (2,0...2,8)\frac{W_w}{H} \text{ 1/s}$$

H – the average oscillation frequency of cutting soil,
m - loosening depth,
W_w – speed of the working body.

Let us etermine the dependence of the time of delay of triggering quencher dynamic fluctuations of hydraulic parameters [2, 19].

Suppose that a dynamic body is working in rocky soil at the depth of H = 0,3 m, in which case the rate of dynamic body is

$$W_w = 2 \text{ m/s}.$$

First of all determine the time of soil cleavage:

$$T_c = \frac{1}{\bar{n}_m} = \frac{1}{15,3} = 0,09 \text{ s}. \quad (15)$$

Determine the relationship Eq. (14) between the middle frequency oscillation cutting forces:

$$\bar{n}_m = \frac{\bar{n}_0}{0,63...0,87} = \frac{13,3}{0,87} = 15,3 \text{ 1/s}, \quad (16)$$

$$\bar{n}_0 = 2 \cdot \frac{2}{0,3} = 13,3 \text{ 1/s}. \quad (17)$$

The initial data of the hydraulic system for blanking dynamic oscillations: N_h = 100 kW, P_h = 25 MPa, P = 30 MPa, L = 20 m, W = 4.25 m/s.

Let us carry out the calculation:

$$Q_h = \frac{17{,}1 \cdot 100}{25} = 68{,}1 \text{ m}^3\text{/s}, \quad (18)$$

$$D = 4{,}5 \cdot \sqrt{68{,}1/4{,}25} = 18 \text{ mm}, \quad (19)$$

$$V_{\text{con.}} = \frac{3{,}14 \cdot 0{,}018^2}{4} \cdot 20 = 0{,}0051 \text{ m}^3, \quad (20)$$

$$V_1 = (0{,}0051 + 0{,}0549) \cdot 0{,}1 = 0{,}006 \text{ m}^3, \quad (21)$$

$$S_1 = \frac{3{,}14 \cdot 0{,}018^2}{4} = 0{,}00026 \text{ m}^2, \quad (22)$$

$$K_b = 0{,}278 \frac{61{,}2 \cdot 100}{30^2} = 1{,}89 \text{ kW/MPa}^2, \quad (23)$$

$$t_{\text{d}} = \frac{30 \cdot 0{,}00026 \cdot 20 + 0{,}006}{68{,}1 - 0{,}5 \cdot 1{,}89 \cdot 30} = 0{,}004 \text{ s}. \quad (24)$$

From the resulting example we can conclude that the system performance satisfies oscillation quenching specified condition of this dynamic device's triggering delay is lower than 15 % of the soil cleavage time.

By changing the parameters of hydraulic system dynamic quenching fluctuations – namely the supply of hydraulic fluid in the system and reduction of fluid volume ratio, and substituting them in the present calculation, we obtain the dependence of the delay in the operation of the hydraulic system dynamic quenching oscillation (speed) on the hydraulic fluid supply system (Fig. 4) and on the coefficient reduction of fluid (Fig. 5).

The hydraulic quencher dynamic fluctuations were developed to improve the efficiency of dynamic quenching of oscillations, which is based on the use of the above-mentioned system.

The dynamic oscillations quencher (Fig. 6, 7) works as follows [10].

Vibrations that are transmitted from the working body to the base machine are blanked with the help of dynamic fluctuations quencher *1*. At active work, the oscillatory body rod *3* tries to

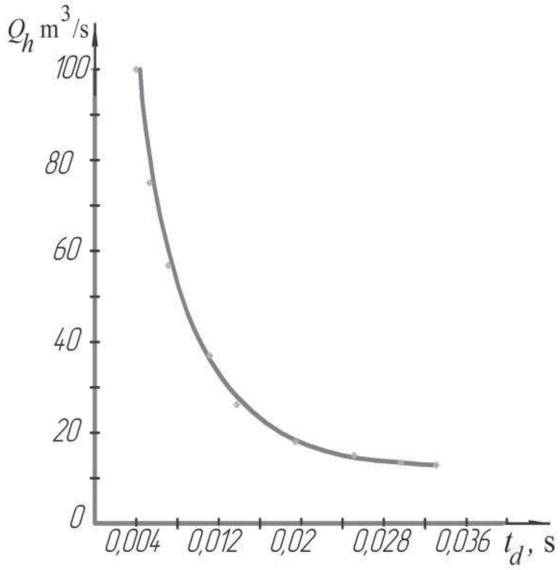

Fig. 4. Graph of the time delay dependence during the operation of hydraulic blanking dynamic fluctuations on hydraulic fluid supply.

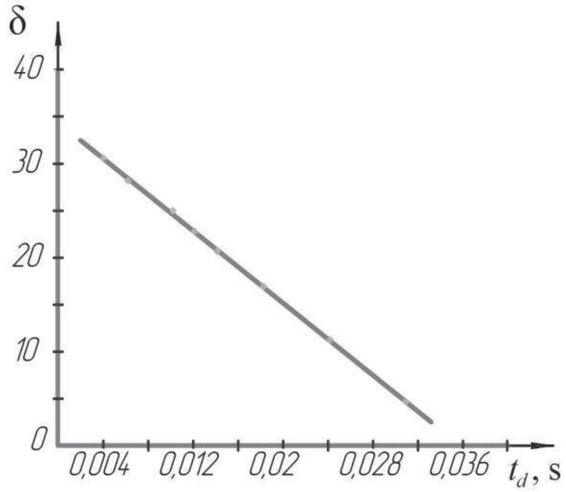

Fig. 5. Graph of the time delay dependence during the operation of hydraulic blanking dynamic fluctuations on the fluid amount reduction coefficient

reproduce vibrational motion in the housing *2*. However, when the direction of the vibrational motion is directed, for example, left by movement of fluid through the holes throttled *6* plunger *4*, *8* rods blocking valve and plunger valve *11* are moving blocking right. Thus blocking rods *8* press the valve rods washer *7*. Meanwhile, blocking plunger valve *11* presses the plunger *4*, blocking throttling holes *6*, so that the liquid begins to flow through passage the holes *12*, which extinguish the movement of the rod *3* left. Once the

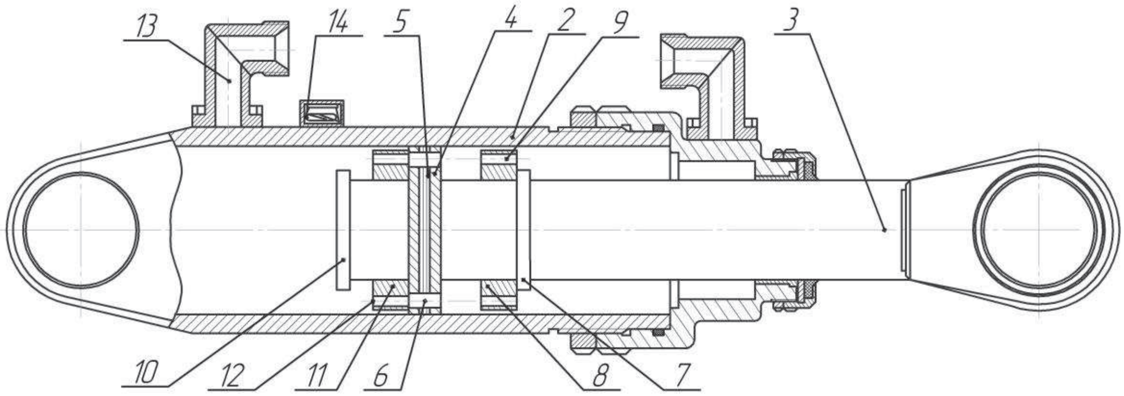

Fig. 6. Quencher of dynamic fluctuations

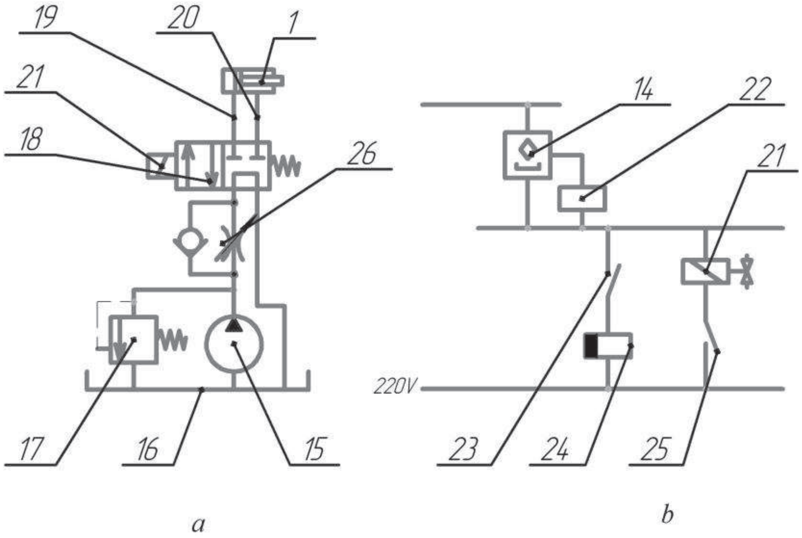

Fig. 7. The hydraulic circuit control in dynamic oscillations quencher *a* and electrical circuit control distributor *b*

plunger *4* reaches the reed switch *14*, the magnetic field of the magnet constant step *5* shut reed contacts *14* (Fig. 7, *b*). The signal will go to detention relay *22*, which shut normally open contact *23*. He, in turn, the delay relay exclusion 24 is turned on. After that, delay relay exclusion *24* shuts contact *25*, which starts the electromagnetic control 21. The electromagnetic control *21* toggles the distributor *18* in the left position. Hydraulic pump *15* works through the variable orifice with check valve *26*, which regulates the supply of hydraulic fluid distributor *18* and pressure line *19* takes an additional portion of the fluid in the plunger cavity quencher dynamic fluctuations *1*. Excess fluid flows out of rods quencher dynamic cavity oscillations *1* through the drain line *20* and distributor *18* to the tank of hydraulic fluid *16*. When plunger *3* moves to the right by moving the working fluid through the throttling holes *6*, *8,* rods blocking valve and plunger valve *11* move left. Valve blocking plunger *11* presses washer *10* and valve blocking rods *8* press plunger *4* blocking throttling hole *6*. As a result, the liquid begins to flow through passage *9* , putting the movement of the rod *3* right. Once the plunger *4* moves away from the reed switch *14* and the magnetic field of the magnet constant step *5* stops to influence the reed switch *14* (reed switch contacts open up *14*), the signal ceases to be submitted normally to open contact *23*. This, in turn, relays switching delay open up exclusion *24*. It will work for a while, allowing the hydraulic pump *15* to submit several additional portions of the liquid in the plunger cavity oscillations quencher *1*-for-*4* plunger removal of reed switch *14*. After exclusion of relay contact delay *24,* the *25*, *21* will control the electromagnetic switch distributor *18* in the far right position, then feed additional portion of the liquid to the quencher dynamic fluctuations *1* end. Hydraulic fluid is fed to the quencher through the pipe *13*.

Because of the dynamic fluctuations quencher *1* reduces dynamic fluctuations in the base machine, during the working bodies active action.

CONCLUSIONS

1. Based on the analysis of dynamic cushioning devices for vibration in the hydraulic system, the mathematical model of the process of determining the delay time of their operation is worked out, which allows to design these systems.
2. Based on the values, the dependency graphs are built of: time delay operation dependence of the hydraulic system dynamic oscillations on quenching coefficients of reduction of fluid and supply of hydraulic fluid. By changing these parameters the system was designed for adjusting the hydraulic damping system dynamic oscillations.
3. A new design was performed of hydraulic blanking of dynamic fluctuations with the ability to change the filling parameters.
4. The design provides adjustable vibration base machine for vibrations attachments.

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ГИДРАВЛИЧЕСКИЙ ГАСИТЕЛЬ ДИНАМИЧЕСКИХ КОЛЕБАНИЯХ

Аннотация. Проведен обзор и анализ существующих амортизирующих устройств. Рассмотрен метод динамического гашения колебаний. Составлена математическая модель определения времени запаздывания срабатывания гидроавтоматической системы гашения динамических колебаний, значение которого позволяет создавать гидроавтома-тические системы гашения колебаний вовремя реагирующих на динамические внешние возмущения. Проведен расчет периода скалывания грунта при работе рыхлителя и времени запаздывания срабаты-вания гасителя динамических колебаний, на основе которых устанавливается способность гидроавтоматической системы гашения дина-мических колебаний вовремя срабатывать обеспечивая предотвращение передачи колебаний к базовой машине. Разработана гидроавтоматическая система гашения дина-мических колебаний, возникающих на рабо-чем органе, для предотвращения передачи этих колебаний к базовой машине.

Ключевые слова: гаситель, динамические колебания, гидроавтоматическая система, время запаздывания срабатывания, период скалывания грунта.