ISSN 2076-2429 (print)
ISSN 2223-3814 (online)
Proceedings of Odessa Polytechnic University, Issue 1(65), 2022

MACHINE BUILDING

МАШИНОБУДУВАННЯ

UDC 621.941

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DYNAMIC VIBRATION DAMPERS FOR CONSOLE INSTRUMENTS

Г.О. Оборський, О.А. Оргіян, В.М. Тіхенко, Г.В. Баланюк, В.М. Колеснік. **Динамічні гасники коливань для консольного інструмента.** Максимальна ефективність динамічних гасників коливань (ДГК) досягається за оптимальних значень їх параметрів. Інерційні ДГК налаштовують за власною частотою та демпфуванням, причому оптимальні значення параметрів гасника залежать від спектрального складу подавлених коливань. Для нелінійних ДГК оптимальні значення параметрів (наприклад, зазору або попереднього стиснення) також залежать від інтенсивності зовнішніх впливів на об'єкт. Для багатоелементних ДГК слід оптимізувати кількість інерційних елементів гасника. Умови оптимального настроювання інерційного ДГК – це умови антирезонансу. У роботі досліджено налаштування ударного ДГК з в'язким тертям. Для визначення оптимального значення зазору між масою гасника та його корпусом вирішено нелінійну задачу методом точкових перетворень. Визначено вплив відхилень від оптимального налагодження ударного ДГК на розмах коливань. Встановлено, що малі відхилення від оптимуму викликають значний приріст розмаху коливань (у 2...5 разів), що призводить до необхідності налаштування ударного ДГК з високою точністю. У роботі оцінено чутливість віброгасника до похибки оптимізації параметрів. Також вивчені особливості налаштування багатоелементних ДГК, що вбудовуються в розточувальну борштангу для восьми типів гасників при варіюванні їх конструктивних особливостей. При дослідженнях частотних характеристик за міру ефективності прийнято співвідношення максимальних значень A_0^{max}/A_r^{max} де A_0^{max} – амплітуда коливань у системі без гасника. Встановлено вплив діаметрального зазору та зусилля осьового стиснення на ефективність ДГК різних типів.
Ключові слова: ударний гасник, чутливість налагодження, амплітуда, частота, зазор, борштанга, сила стиснення

H. Oborskyi, A. Orgiyan, V. Tikhenko, A. Balaniuk, V. Kolesnik. **Dynamic vibration dampers for console instruments.** The maximum efficiency of dynamic vibration dampers (DD) is achieved at optimal values of their parameters. Inertial DD are tuned according to their own frequency and damping, and the optimal values of the absorber parameters depend on the spectral composition of the suppressed oscillations. For nonlinear DD, the optimal values of parameters (for example, gap or precompression) also depend on the intensity of external influences on the object. For multi-element DDs, the number of damper inertial elements should also be optimized. The conditions for optimal tuning of the inertial DD are the conditions of antiresonance. In this work, the tuning of an impact DD with viscous friction has been studied. To determine the optimal value of the gap between the mass of the absorber and its body, a nonlinear problem was solved by the method of point transformations. The influence of deviations from the optimal setting of the shock DD on the amplitude of oscillations is determined. It has been established that small deviations from the optimum cause significant increments in the oscillation range (by 2...5 times), which leads to the need to tune the shock DD with high accuracy. The sensitivity of the vibration damper to the optimization error of its parameters is estimated in the work. In addition, the tuning features of multi-element DDs built into the boring bar for eight types of absorbers were studied with varying their design features. When studying the frequency characteristics, the ratio of the maximum values A_0^{max}/A_D^{max} was taken as a measure of efficiency, where A_0^{max} is the oscillation amplitude in the system without a damper. The influence of the diametral clearance and the axial compression force on the efficiency of various types of DDs has been established.
Keywords: shock absorber, tuning sensitivity, amplitude, frequency, gap, boring bar, compression force

Introduction

Numerous studies cited in the technical literature are devoted to designs, operating principles, calculations and optimal settings of vibration dampers or dynamic vibration dampers [1, 2, 3, 4]. To suppress harmonic, as well as narrow-band random oscillations, inertial DDs are used, in which the attached mass acts on the object of vibration damping through an elastic element, a pendulum connection, or by means of shock pulses. Under the conditions of action of broadband vibration disturbances,

DOI: 10.15276/opu.1.65.2022.01

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)

DD with friction and vibration absorbers are used (the latter are distinguished by the absence of an elastic connection between the attached mass and the object of vibration damping). At variable impact frequencies, it is recommended to use a DD with active elements that make it possible to adjust the absorber parameters [5].

Analysis of recent publications and problem statement

An increase in the number of DD elements creates additional antiresonances [6], which increase the efficiency of oscillation suppression [7]. The use of multi-element DD in systems with distributed parameters gives the greatest efficiency when installing elements in places where vibration effects are applied or in the antinodes of the forms of suppressed vibrations, as well as when matching the settings of individual elements [8, 9].

In metal-cutting machine tools, DDs with friction are most widely used, in the elastic elements of which elastic-damping materials are used [10, 11, 12]. Such DDs are used to reduce the vibration level of cantilever tools (boring bars, milling cutters, hones, etc.) and body parts of radial drilling, milling and other machines. Studies have shown that single-element DD with friction reduce the level of both bending and torsional vibrations of cantilever boring bars during fine boring, providing an increase in the stability of the dynamic system of the machine, reducing wear of cutters and roughness of the machined surface. Impact DD is most often used for boring bars, with multi-element DD, for example, Kennametal designs, being the most effective. Dry friction forces arising on the contact surfaces of the absorber elements cause stagnation that prevents effective suppression of low-level oscillations. The elimination of this disadvantage is achieved by using means for retaining lubricant between the elements of the DD [13, 14].

When evaluating the factors influencing the conditions for setting the DD for boring bars, one should take into account the characteristics of the influences that cause vibrations and the design features of vibration dampers. In addition to force and kinematic external influences on the mounting frame of the machine, the sources of oscillations are parametric disturbances. When processing a number of materials, mostly brittle, the boring cutter experiences an intense broadband random force effect. Impact absorbers are characterized by the action of friction forces on the attached mass. Finally, to increase the efficiency of DDs, multielement modifications of their designs should be used [15, 16].

The purpose of the study: improvement of DD designs and determination of the optimal setting under the conditions of external influences and change of parameters typical for finishing, boring and jig boring machines.

Research objectives: to achieve this goal, the following tasks were solved:

– optimal conditions for setting up a shock DD with friction;
– features of setting multielement DD;
– sensitivity of the deviations of the DD parameters to the range of oscillations.

The maximum efficiency of DD is achieved when the conditions for optimal tuning are met. Inertial DDs are tuned to the frequency of suppressed oscillations by setting the stiffness of the elastic element and the mass (moment of inertia) of the absorber. For a percussive DD, the gap size and mass are selected depending on the frequency and amplitude of the force that excites the vibrations of the object. The tuning of the DD with friction [17] is performed by selecting the frequency and attenuation from the condition of uniformity of the amplitude-frequency characteristic of the system, including the object and the DD, in the frequency range limited by the partial frequencies of the object and the vibration damper. The refinement of the relationships that determine the conditions for optimal tuning of the DD is achieved by taking into account the forces of inelastic resistance acting in the oscillatory system of the object, and the sensitivity of the frequency characteristics to the deviation of parameters from optimal values [18, 19].

Main part

Conditions for tuning shock DD with friction.

The problem of optimizing the parameters of the shock DD was solved under the assumption of the linear nature of the friction forces that arise between the absorber and the object of vibration damping (Fig. 1, *a*).

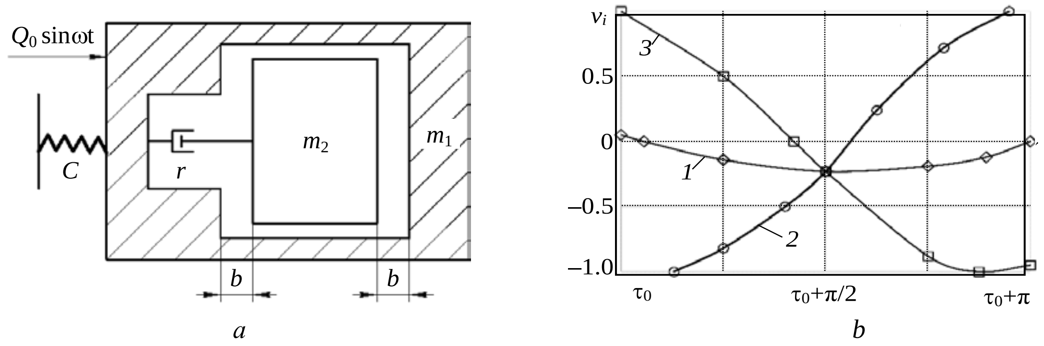


Fig. 1. Shock DD with friction: *a* – design scheme; *b* – the influence of the deviation of parameters from the optimum on the range of fluctuations; *1* – $\tau_0 = \tau_0^* = 1.67$; *2* – $\tau_0 = 1.68$; *3* – $\tau_0 = 1.66$

The equations of motion of such a system in the interval between impacts have the form:

$$m_1\ddot{y}_1 + r(\dot{y}_1 - \dot{y}_2) + cy_1 = Q_0 \sin\omega t,$$
$$m_2\ddot{y}_2 + r(\dot{y}_2 - \dot{y}_1) = 0, \qquad (1)$$

where: m_1 – weight of the boring bar;

m_2 – mass of DD;

c – boring bar stiffness;

r – dissipation coefficient;

Q_0 – damping coefficient;

ω – disturbance frequency.

We solve this problem using the method of point transformations.

After passing to dimensionless variables and parameters:

$$\tau = \omega t\,,\ \upsilon_1 = \frac{m_1\omega^2 y_1}{Q_0},\ \upsilon_2 = \frac{m_1\omega^2 y_2}{Q_0},\ \mu = \frac{m_2}{m_1},\ \kappa = \frac{r}{m_1\omega_1},\ \omega_0^2 = \frac{c}{m_1\omega^2},\ \Delta = \frac{m_1 b\omega}{Q_0}, \qquad (2)$$

equations (1) take the form:

$$\ddot{\upsilon}_1 + \kappa(\dot{\upsilon}_1 - \dot{\upsilon}_2) + \omega_0^2\upsilon_1 = \sin\tau,$$
$$\mu\ddot{\upsilon}_2 + \kappa(\dot{\upsilon}_1 - \dot{\upsilon}_2) = 0. \qquad (3)$$

These equations must be supplemented by equations describing the impact interaction of the absorber with the object at the moment of time τ_0:

$$\left|\upsilon_1(\tau_0) - \upsilon_2(\tau_0)\right| = \Delta,$$
$$(1+\mu)\dot{\upsilon}_1'(\tau_0) = (1-\mu R)\dot{\upsilon}_1(\tau_0) + \mu(1+R)\dot{\upsilon}_2(\tau_0),$$
$$(1+\mu)\dot{\upsilon}_2'(\tau_0) = (1+R)\dot{\upsilon}_1(\tau_0) + \mu(1-R)\dot{\upsilon}_2(\tau_0), \qquad (4)$$

where: *R* – velocity recovery coefficient upon impact, and the dashed lines indicate the values of velocities after impact.

In the interval between impacts, we find the solution of the linear system of equations (3):

$$\upsilon_1 = A_1 \sin\tau + B_1 \cos\tau + \sum_{i=1}^{4} G_i \exp(\nu_i\tau),$$
$$\upsilon_2 = A_2 \sin\tau + B_2 \cos\tau + \sum_{i=1}^{4} L_i \exp(\nu_i\tau), \qquad (5)$$

where: ν_i – roots of the frequency equation.

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)

The coefficients of the particular solution of the inhomogeneous problem A_k, B_k are determined after substituting (5) into equations (3). Of the eight coefficients of the general solution of a homogeneous problem, only four are independent, since G_i and L_i are interconnected by equations of forms. From three equations (4) and two symmetry conditions for the simplest periodic regime, we find four coefficients L_i and the smallest positive root of the equation $\left|\upsilon_1(\tau_0)-\upsilon_2\right|(\tau_0)=\Delta$. Assuming the impact to be elastic and the dissipation to be small, considering oscillations near the resonance, and representing the coefficients of the solution as expansions in powers of κ, we arrive at the relation:

$$\tau_0 = \frac{4\mu\Delta}{\pi}, \tag{6}$$

and we obtain the expressions υ_1 and υ_2.

We consider optimal values of the absorber parameters at which the amplitude of the first harmonic of the function $\upsilon_1(\tau)$ vanishes. From here we find an equation for determining τ_0^* – the value of τ_0 corresponding to the optimum conditions:

$$\kappa\sin\tau_0^* + \mu\cos\tau_0^* = 0,$$

For small κ, we obtain the expression (7):

$$\tau_0^* = \frac{\pi}{2} + \frac{\kappa}{\mu}, \tag{7}$$

which, together with (6), leads to the condition that the gap must satisfy:

$$\Delta^* = \frac{\pi^2}{8\mu} + \frac{\pi\kappa}{4\mu^2}. \tag{8}$$

Returning to the dimensional parameters of the system using relations (2), we obtain:

$$b^* = \frac{\pi Q_0}{4m_2\omega^2}\left(\frac{\pi}{2} + \frac{r}{m_2\omega}\right). \tag{9}$$

To assess the effect of deviations from the optimum on the range of fluctuations, numerical calculations of the function $\upsilon_1(\tau)$ were carried out for three values of τ_0: optimal and deviating from it by ±0.01 (Fig. 1, *b*). Small deviations from the optimum cause large increments in the oscillation range, which means that the tuning of the shock DD must be performed with high accuracy.

Let us present the results of the tuning features of multielement DDs.

Multi-element vibration dampers of the cantilever tool are installed near the cutting zone (Fig. 2).

a *b*

Fig. 2. Placement of DD: *a* – in front of the incisor, *b* – and behind the incisor

The dimensions of the attached mass are limited by design. However, an increase in the DD mass in excess of 10% of the tool mass is accompanied by a slight decrease in the level of vibrations.

This regularity was verified experimentally during fine boring with cantilever boring bars, the diameter d1 of which varied within 15…70 mm. The nature of the change in the amplitude of oscillations depending on the ratio of the mass of the absorber to the mass of the boring bar is shown in Fig. 3.

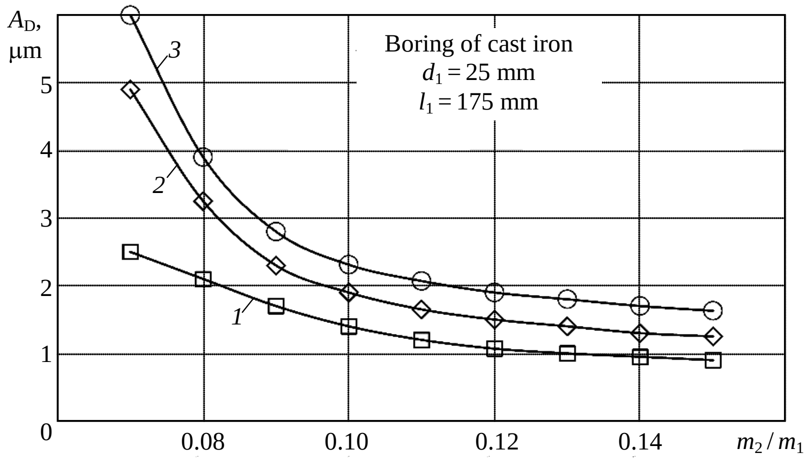
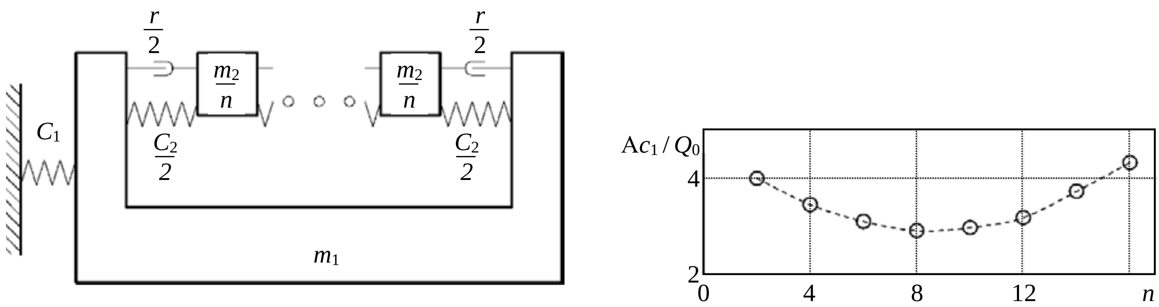


Fig. 3. Dependence of the oscillation amplitude on the ratio of the total mass of the DD to the mass of the boring bar: *1* – *t* = 0.05 mm; *2* – *t* = 0.1 mm; *3* – *t* = 0.2 mm

With limited dimensions, the necessary increase in the mass of the absorber is achieved by manufacturing its elements from heavy alloys. Assuming that the total mass of a multi-element DD remains constant, let us consider the influence of the number n of absorber elements on its characteristics. When connecting the elements of the DD in series, we solve the system of equations of motion:

$$m_1\ddot{y} + r\dot{y} - \frac{r}{2}(\dot{y}_1 + \dot{y}_n) + c_1 y - \frac{C_2}{2}(y_1 + y_n) = Q_0 \sin \omega t,$$
$$\frac{m_2}{n}\ddot{y}_i + \frac{r}{2}(2\dot{y}_i - \dot{y}_{i-1} - \dot{y}_{i+1}) + \frac{C_2}{2}(2y_i - y_{i-1} + y_{i+1}) = 0, \tag{10}$$
$$(i = 1, 2, \ldots, n;\ y_0 = y_{n+1} = y),$$

corresponding to the linear model (Fig. 4, *a*).



a *b*

Fig. 4. DD with series connection of elements: *a* – calculation scheme; *b* – calculated dependence of the amplitude of forced oscillations on the number of elements

The eigenfrequencies of the DD can be calculated from the formula:

$$\omega_i^2 = \frac{2nC_2}{m_2}\sin^2\frac{i\pi}{2(n+1)}, \tag{11}$$

from which it follows that the length of the spectrum $\omega_n - \omega_1 = 2\sqrt{\frac{nC_2}{m_2}} \cdot \sin\left[\frac{\pi(n-1)}{4(n+1)}\right]$ grows about as $\sqrt{n}$. Numerical study of the dependence of the DD efficiency on *n* for the constant *m* shows that the

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)

amplitude of the solution $y = A\sin(\omega t + \varphi_0)$ Eqs. (10) depends on the number of elements nonmonotonically (Fig. 4, *b*). Therefore, the optimization problem n arises for multielement DD.

Comparative experimental studies of the efficiency of single-element and multi-element DD of different types, schematically depicted in Figs. 5.

Impact DHA is a cylinder freely inserted into the cavity of the drill rod (Fig. 5, *a*); optimization for it is carried out by choosing the diametrical gap 2*b*. In DD (Fig. 5, *b*) the backlash is filled with elastic-damping material (rubber or hydroplastic); the optimal operating conditions of such a damper are set by a joint change in the size of the gap 2*b* and the characteristics of the material filling the gap.

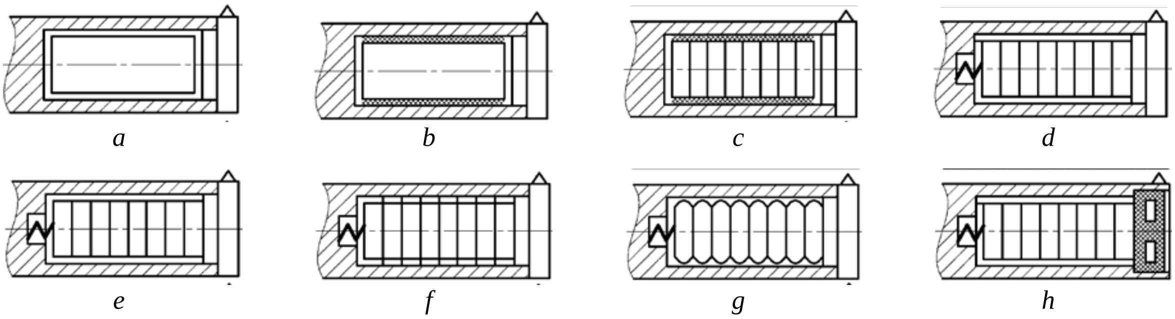


Fig. 5. Types of investigated DD for cantilever borscht: *a*, *b* – one-element; *c*, *d*, *e*, *f*, *h* – multi-element

In the multi-element vibration damper in Fig. 5, *c* cylinder dissected into disks. In comparison with DD (Fig. 5, *d*) there is an additional opportunity to increase efficiency by changing the number of n elements. The efficiency of DD type 4, similar to the Kennametal model, also depends on the magnitude of the axial force *N* compressing the disks. DD (Fig. 5, *e*), proposed differs in that elastic gaskets covered with a viscous substance separate the disks from each other. In DD (Fig. 5, *f*) the dividers are rigid partitions. For these dampers, in addition to the above parameters, the characteristics of the gaskets and viscous substance are also optimized.

In order to determine the direction of action of the shock pulses transmitted by the elements of the damper to the drill rod, the design of the DD (Figs. 5, *g*) with annular protrusions on the discs. The dependence of the damping capacity of the spindle with a rod on the speed leads to the need for appropriate adjustment of the DD, which can be automated, for example, using the design (Fig. 5, *h*).

The vibration damper with annular protrusions on the disks *1* (Fig. 6) is compressed by a spring *4* through the pusher *5*. The spring is held by a threaded cover *2*, which is screwed into the body of the drill rod increases the force *N* of axial compression. The position of the cover is fixed by a lock nut *3*. The location of the protrusions in the middle of each disk ensures the passage of the line of action of the shock pulse through the center of gravity of the disk, i.e. in the direction of the suppressed bending oscillations of the drill rods. The shape of the cross section of the protrusion and the width of the site on which the contact is made during the impact interaction are selected depending on the characteristics of the material from which the disks are made.

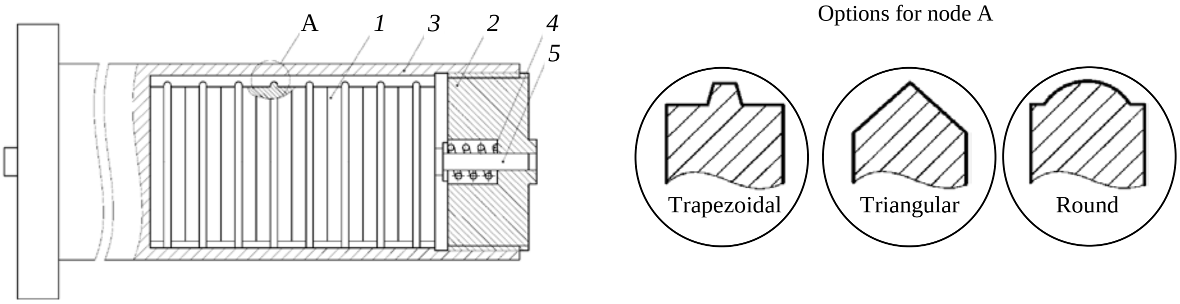


Fig. 6. DD of shock action with ledges on disks

When testing DD (Fig. 5, *d*, *e*, *f*, *c*) the magnitude of the axial compression force was determined using a calibration device (Fig. 7).

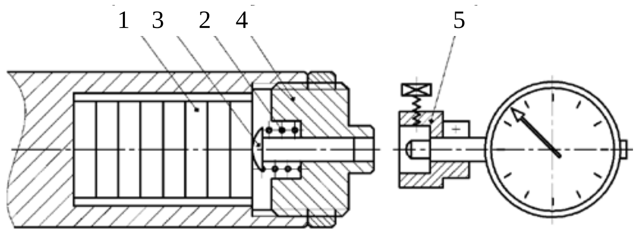


Fig. 7. Multi-element DD with a device for calibration of axial compression force

When the force N of axial compression of the spring *2* transmitted to the disks *1* changes, the pusher *3* moves in the hole of the threaded cover *4*. The magnitude of this movement is measured by an arrow mounted in the housing *5* of the device fixed when adjusting on the shank of the threaded cover. The calibration of the device is performed with the threaded cover removed from the drill rods by loading the pusher through the dynamometer.

The dependence of the amplitude A_D of the oscillations of the drill rod with DD on the number of damper elements (Fig. 8) is nonmonotonic in both test methods. The use of DD with an optimal value of *n* equal to (8) reduces the amplitude of oscillations during cutting by 3...5 times compared with a single-element damper.

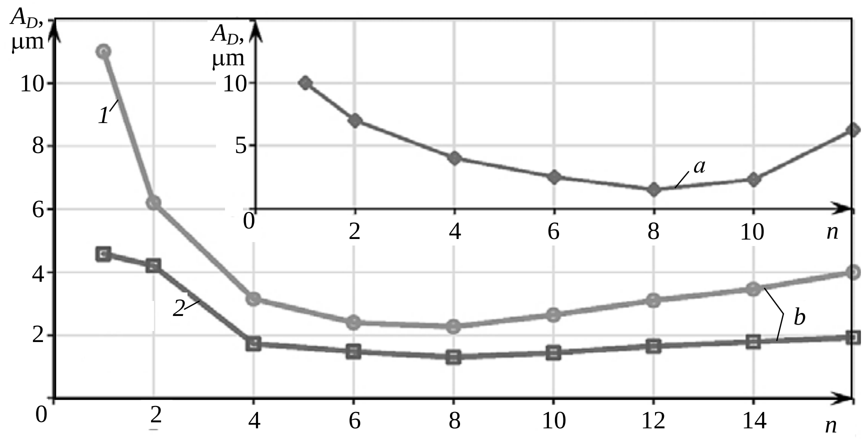


Fig. 8. Dependences of the amplitude of oscillations on the number of elements of DD:
a – by frequency response; *b* – when boring; 1 – d_1=50 mm, l_1=475 mm; and 2 – d_1=25 mm, l_1=175 mm

In the study of frequency characteristics, the ratio of the maximum values of A_0^{max}/A_D^{max}, where A_0 is the amplitude of oscillations in a system without a damper, was taken as a measure of the efficiency of DD. According to the changes in the efficiency of DD during the joint variation of the diametrical gap and the axial compression force of the disks, the optimal values of these parameters were set (Fig. 9): for DD type 4 $2b_{opt} = 0.3$ mm and $N_{opt} = 6.5$ N. Near the optimum parameters, which creates favorable conditions for the operation of multi-element DD.

The optimal values of 2*b* opt obtained in the drilling experiments and determined by the frequency response are close, and N_{opt} are markedly different. When boring, the dependence of N_{opt} on the mass *m* of the damper was established (Table 1). These data were obtained for 8-element DD at $m_2 = 0.12m_1$.

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)

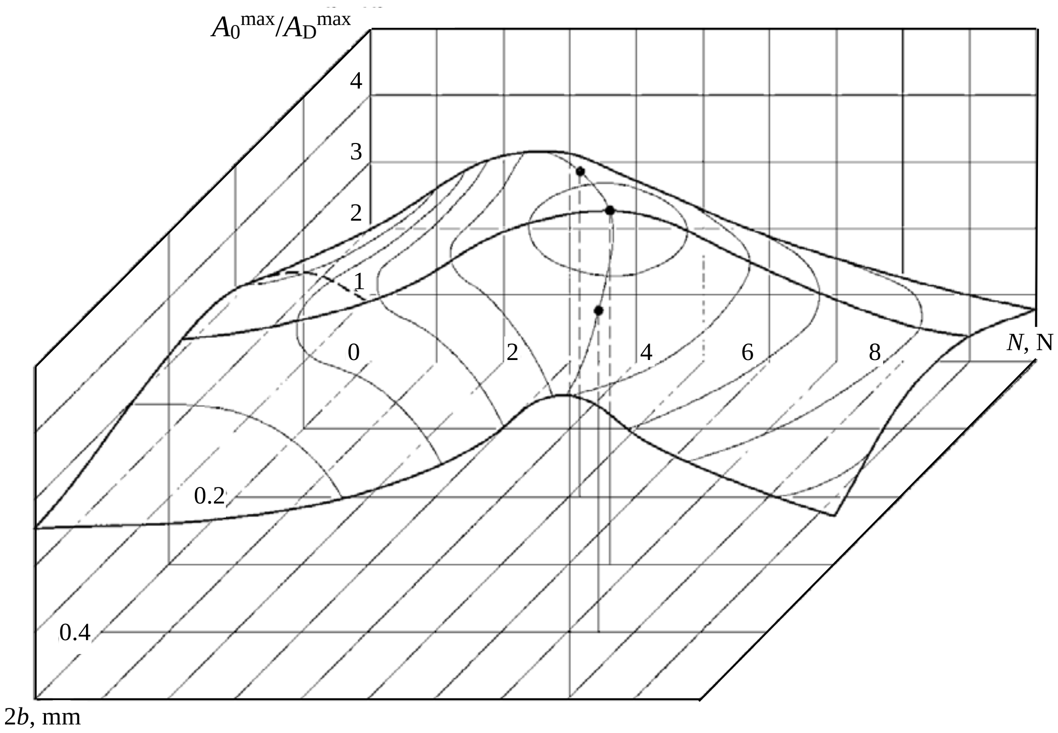


Fig. 9. Influence of diametrical gap 2b and axial compression force on the efficiency of DD of the type shown in Fig. 5, *d*

Table 1

Optimal values of axial compression force

m_2, g	N_{opt}, N	m_2, g	N_{opt}, N
50	11	500	30
100	17	1000	37
200	22	2000	40

Comparisons of the efficiency of DD of different types (Fig. 10) in terms of frequency response and ultimate susceptibility to boring show a significant advantage of multi-element dampers over single-element ones. Single-element DDs reduce the amplitudes of oscillations in resonance and increase the value of the ultimate pliability of borshtangs by 3...4 times, and multi-element – by 5...10 times. Among the tested DD, the type 7 damper is the most effective, providing a tenfold increase in the maximum yield strength when boring compared to drill rods not equipped with DD.

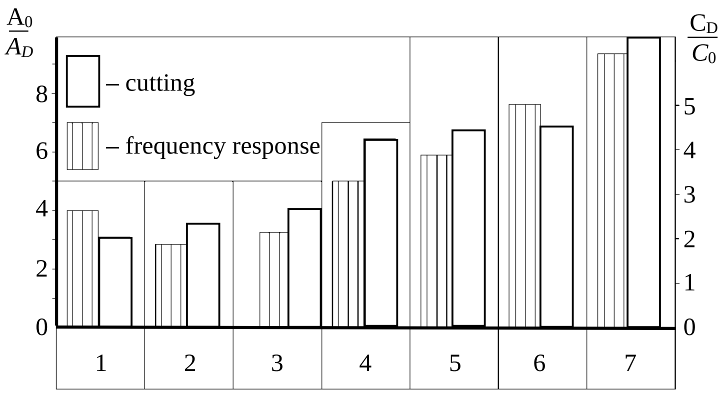


Fig. 10. Comparison of the effectiveness of DD of different types shown in Fig. 5

Conclusions

Vibrations during cutting reduce the tool life. At present, there is no single method to eliminate the cause of the excitation of oscillations. However, intensive study of the sources of the appearance of vibration regimes can significantly reduce the harmful effects of mechanical vibrations. Vibration control is especially important during high-precision finishing on boring and jig boring machines. Damping of oscillations by means of shock-type dampers is widely implemented in production. The correct setting of vibration dampers depends on the characteristics of external influences. Impact dampers are characterized by the action of friction force on the attached mass. Multi-element modifications of their designs should be used to increase the efficiency of DD. The problem of setting the shock DD with friction is solved in the work. It is established that small deviations from the optimum cause significant increments in the amplitude of oscillations. This conclusion indicates the need for high-precision DD adjustment.

One of the important tasks for practice is the calculation of the parameters of the DD setting under the action of parametric perturbations. For multi-element DD set the optimal values of the parameters: the number of masses $n = 8$, the gap size of 0.3 mm, and the compressive forces of the disks $N = 6.5$ N.

It is established that near the optimum the efficiency of DD weakly depends on the change of parameters, which creates favorable conditions for the operation of multi-element DD.

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Received January 23, 2022

Accepted March 05, 2022