DEVELOPMENT OF DESIGNS AND JUSTIFICATION OF THE PARAMETERS OF A SCRETTING DRUM WITH A DAMPER OF A SPINNING MACHINE

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ANNOTATION

The article provides an analysis of the features of existing designs of the discretizing drum of a spinning machine. The structural scheme and the principle of operation of the recommended composite discretizing drum with a shock absorber and sectional gear sets are presented. The results of theoretical studies of vertical vibrations of sectional toothed sets of a discretizing drum. On the basis of the obtained patterns of vibrations of gear sets and the constructed graphic dependencies, the main parameters of the drum are substantiated.

Keywords: Spinning machine, discretizing drum, fibers, tape, gear set, shock absorber, mass, stiffness, dissipation, oscillations, amplitude, amplitude, frequency.

АННОТАЦИЯ

В статье приводится анализ особенностей существующих конструкций дискретизирующего барабанчика прядильной машины. Представлена конструктивная схема и принцип работы рекомендуемого составного дискретизирующего барабанчика с амортизатором и секционными зубчатыми гарнитурами. Результаты теоретических исследований вертикальных колебаний секционных зубчатых гарнитур дискретизирующего барабанчика. На основе полученных закономерности колебаний зубчатых гарнитур и построенных графических завимостей обоснованы основных параметры барабанчика.

Ключевые слова: Прядильная машина, дискретизирующый барабанчик, волокна, лента, зубчатая гарнитура, амортизатор, масса, жесткость, диссипация, колебания, амплитуда, размах, частота.

INTRODUCTION

The essence of the discretization process lies in separating the tape into separate non-contacting fibers, in relative displacement and in distributing them over a very large length.
In the process of discretization, an extra-high thinning occurs, i.e. the tape is thinned by 3000-7500 times, and in the section of the discrete flow with ideal separation there are 2-6 non-contacting fibers. This is the difference between sampling and pulling. [1] Discretization is carried out in order to form a discrete flow of non-contacting fibers, which cannot perceive and transmit torque due to the rotation of the forming - twisting device - spinning chamber, thereby

creating conditions under which no false twisting occurs between the sampling device and the outlet pair. The discretization process is carried out in a discretizing device. At present, the design of the device of the BD-200 type machine is most widely used.

In another well-known design, the receiving drum assembly of the Maxilin type [2] is made of a steel pipe with an eight-start wave groove. The analyzes show that during single-start winding of the headset, the beard in this area is exposed to the tooth once per revolution of the drum. And during this period, it is possible to supply such a quantity of tape that the knot of the strand of fibers will be entirely fed into the working area and remain uncombed and turn into yarn. This certainly applies to a number of inclusion sizes commensurate with the feed per drum revolution, or less than it. Larger groups of fibers will be processed by the sampling roller. The disadvantage of the discretizing drum is: the low effect of the separation of weed impurities from the fiber, as well as the insufficient uniformity of the division of the fiber mass into individual fibers, especially with different fiber lengths.

Efficient Sampling Drum Design

A design of a composite sampling drum was developed, the scheme of which is shown in Fig.1. [3].

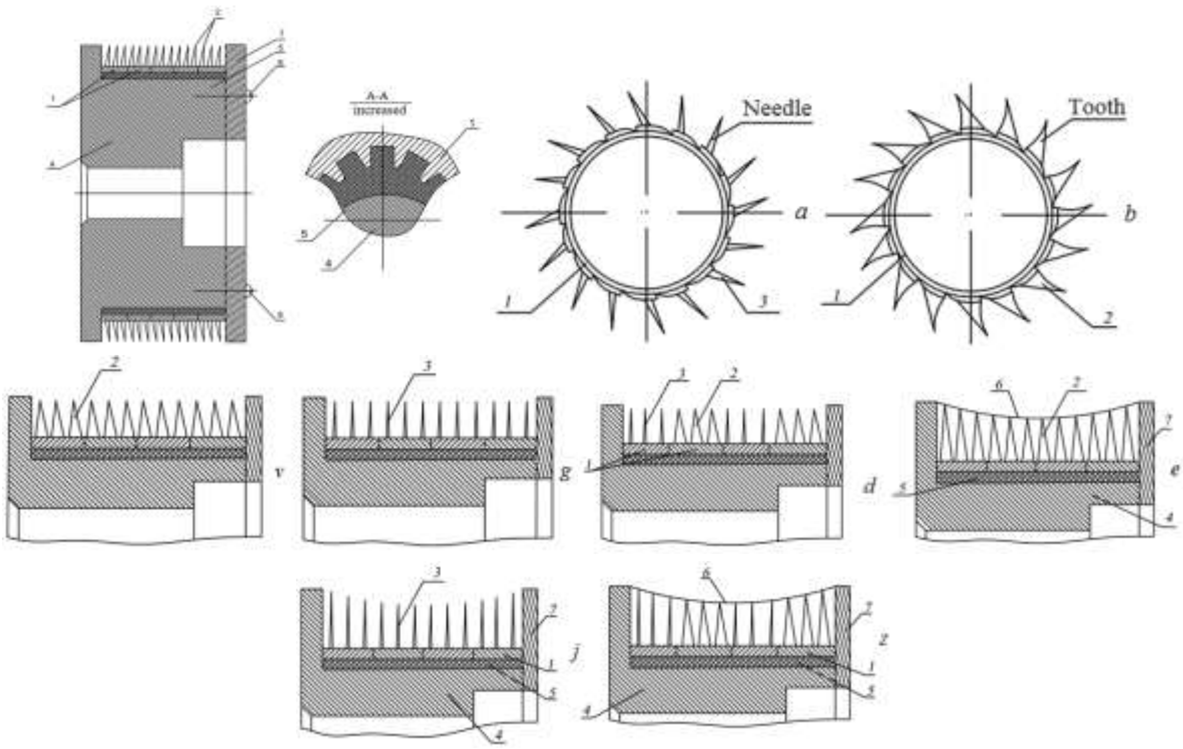


Fig 1 Discretizing drum of an rotor spinning machine

The structure works in the following way. When the discretizing drum rotates and due to the impact of the technological load from the fibers, the outer sleeve 1 with a toothed 2 or needle 3 headset perform additional high-frequency torsional vibrations due to their installation on the base 4 by means of an elastic (rubber) sleeve 5. By choosing the required rigidity of the elastic sleeve 5, one can obtain torsional vibrations with a certain amplitude and frequency, at which the efficiency of fiber separation from the main mass increases due to additional inertial forces.

At the same time, fiber breakage is significantly reduced. The feed tape with a height of 2.0 mm and a width of 9.0 mm has the highest density in its middle [3.4].
At the same time, due to the increased friction of the fibers along the side walls of the base 4 and the cover 7, the fibers remain from the main fibrous mass. Making the height of the teeth 2 and needles 3 installed along the edges of the drum greater by 0.4÷0.6 mm than the height of the teeth 2 and needles 3 along its middle allows overcoming the friction of the resistance of the fibers against the side surfaces, which increases the uniformity of movement and discretization of the fibers throughout drum width. To ensure efficient discretization of a tape of artificial fibers, the outer sleeves 1 with a toothed headset 2 are replaced by outer sleeves 1 with a needle-like headset 3.
When discretizing a tape consisting of a mixture of natural and artificial fibers, you can use the outer sleeve 1 of the toothed 2 and needle 3 headsets installed in alternation. Therefore, the proposed design is universal.
The proposed device allows increasing the efficiency and uniformity of discretization of fibrous material.

Calculation scheme and mathematical model of vertical oscillations of the gear set of the drum.
The parts of the discrete drum gear headsets are the same size. Therefore, their calculation schemes will be the same. The general calculation scheme is shown in Figure 2.
Lagrange's II-order equation [5,6] was used to generate the differential equation representing the oscillations of the gear set:

$$\frac{d}{dt}\left(\frac{\partial \tau}{\partial \dot{q}i}\right)-\frac{\partial \tau}{\partial qi}+\frac{\partial \eta}{\partial \dot{q}i}+\frac{\partial \eta}{\partial qi}=Q(qi) \qquad (1)$$

Here, q_i,$Q(q_i)$– generalized coordinates and generalized forces,
T, P - kinetic and potential energies; F is the dissipative function of the relay [7,8].

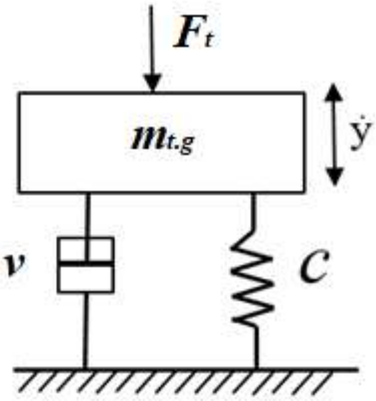


Figure 2. Calculation diagram representing the vertical vibrations of the gear set of the discrete drum

According to the calculation scheme, the potential, kinetic energies and dissipative function of the gear headset parts are determined from the following expressions: [5,8]:

$$P=\frac{1}{2}cy^2; \quad T=\frac{1}{2}m_g; \quad F=\frac{1}{2}v\left(\frac{dy}{dt}\right)^2 \qquad (2)$$

here, c,v are the coefficients of virginity and dissipation on the linear deformation of the rubber bushing under the headset parts; y - vertical shift; mass of τ_g–toothed headset part;

From the obtained expressions (2) were obtained products on the generalized coordinate [9,10]:

$$\frac{\partial P}{\partial y}=cy; \quad \frac{\partial F}{\partial y}=v\frac{dy}{dt}; \quad \frac{\partial T}{\partial \dot{y}}=m_g\dot{y}; \quad \frac{\partial T}{\partial y}=0$$

The yields over time are as follows [9]:

$$\frac{dy}{dt}\left(\frac{\partial T}{\partial \dot{y}}\right)=m=\frac{d^2y}{dt^2} \tag{3}$$

Generalized power [9];

$$F_Л=F_1+F_0\sin\omega t\pm\delta F_1 \tag{4}$$

By placing the obtained (3), (2) on (1), a differential equation was formed that represents the vertical oscillations of the gear set of the discrete drum [10,11];

$$m_g\frac{d^2y}{dt^2}+cy+v\frac{dy}{dt}=F_1+F\sin\omega t\pm\delta F_1 \tag{5}$$

The resulting (5) solution, which represents the free oscillations of the gear headset parts, is as follows [71,72];

$$y=\left(E_1\sin\omega_0 t+E_2\cos\omega_{0t}\right) \tag{6}$$

here $\omega_0=\sqrt{p_0^2-n^2}$; $p_0=\sqrt{\frac{c}{m_г}}$; $n=\frac{в}{2m_г}$;

The current solution for variant $F_1=0v$ $\delta F_1=0$ is the following expression for the forced oscillations according to [12]:

$$y=\frac{F_0\sin\left[\omega t-arctg\left(\frac{2n\omega}{p_0^2-\omega^2}\right)\right]}{m_г\sqrt{\left(p_0^2-\omega^2\right)^2+4\pi-\omega^2}} \tag{7}$$

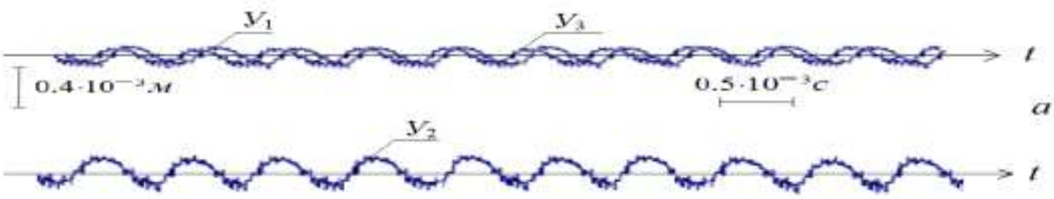
But the general solution of (5) was carried out on a computer using a random number generator, based on the Runge-Kutta program.

Analysis of the wobbling of the parameters of the discretizing drum on the nature of its vibrations. The expressive numerical solution (7) generated to obtain the oscillation laws of the discrete drum gear sets was performed separately for each gear set. Calculations were performed at the following values of the parameters:

$m_2=(1{,}2\div1{,}8)\cdot10^{-2}kg$; $n_д=(6.0\div7.5)\cdot10^{-3}$rev per min;

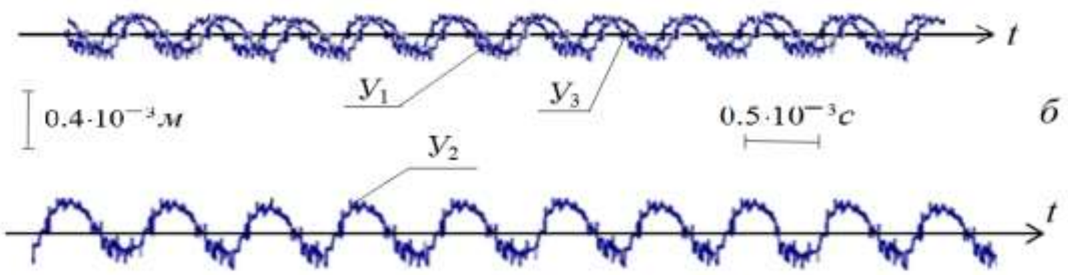
$F_1=(12\div24)sN$; $F_0=(2{,}3\div4{,}5)sN$; $\delta F_1=(0{,}25\div0{,}65)sN$;

$c=(0{,}08\div0{,}35)\cdot10^3 N/m$; $v=(1{,}3\div2{,}5)Nc/m$;|



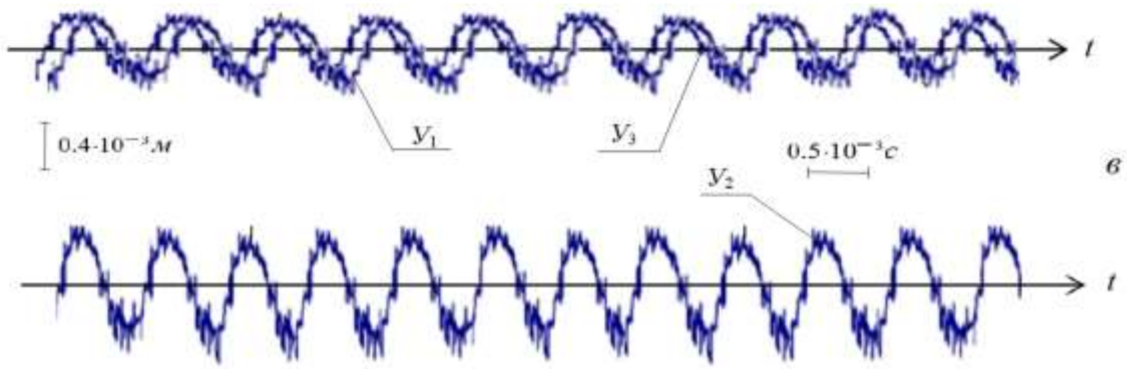
$F_л = (12 + 2.3\sin\omega t \pm 0.25)sN$; $c = 0.25\cdot10^3$ N/m;

$n_\partial = 6\cdot10^3$ rev per min; $v = 2{,}5$ Ns/m;



$F = (18 + 3.35\sin\omega t \pm 0.45)cH$; $c = 0.25\cdot10^3$ N/m;

$n_\partial = 6\cdot10^3$ rev per min; $в = 2{,}1$ Ns/m;



$F = (24 + 4.35\sin\omega t \pm 0.65)sN$; $c = 0.25\cdot10^3$ N/m;

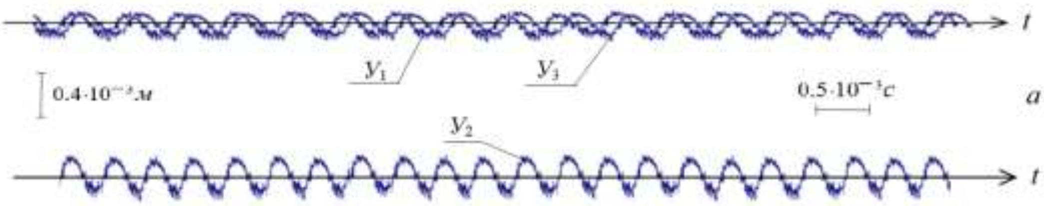
$n_\partial = 6\cdot10^3$ rev per min; $в = 2{,}1$ Ns/m;

Figure 3. The laws of vertical and vibration dependence of the laws of vertical vibration of discrete drum gear sets with the proposed composition are given.

In the obtained laws, the laws of vertical displacement of each of the three gear sets of the discrete drum are defined separately (Fig. 3, $У_1$, $У_2$, $У_3$, graphs). The figure shows the dependence of the proposed composition of the discretization of the drums on the technological laws of vertical vibration of the gear sets and the frequency of rotation of the drive. According to the analysis of the obtained laws, it was found that the amplitude of vibration of the gear set in the middle of the discrete drum is higher than $(15 \div 20)\%$ when the amplitude of vibration of the end gear set. The main reason for this is that it is taken as $F_2 = 1{,}2F_1 = 1.2F_3$, i.e. due to the magnitude of the technological resistance acting on the middle gear set.

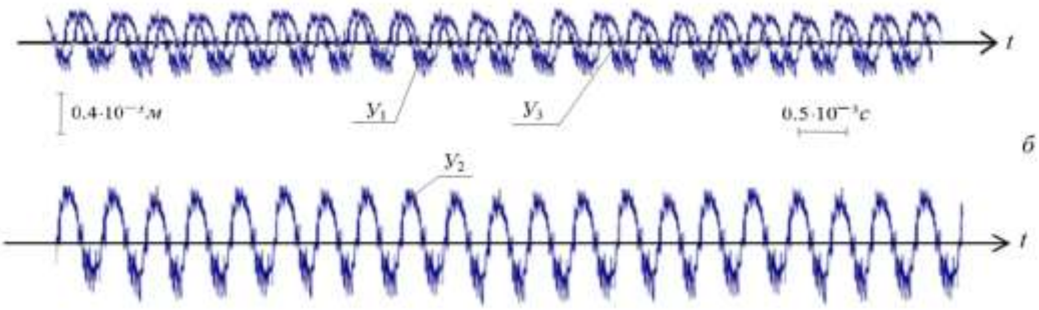
This is because the density of the tape fibers in the middle zone is higher. It should be noted that the vibration amplitudes of $У_1$, $У_2$, and $У_3$, also depend on the linear virginity coefficients of the rubber bushings on which the headsets are mounted. It can be seen from the laws in Figure 3 that as the technological resistance increases, the vibration amplitudes of the gear sets also increase accordingly (Figures 3, a, b, v-graphs). Their oscillation frequencies are almost equal to the rotational frequency of the drive during steady motion. In this case, the value of the oscillation frequency is $1.22\cdot10^2$ Gts. Correspondingly, when the drive speed was increased to $7.5\cdot10^3$rpm, the vibration frequencies of the gear sets increased proportionally and became equal to $1.42 \cdot 10^2$ Gts (Fig. 4, graphs a, b, v). In this case, the effect of the values of the vibration amplitudes of all three gear sets of the discrete drum is almost unchanged.

As a result of processing the obtained laws, graphs of the dependence of the vertical vibration coverage of gear sets on their parameters were constructed. Figure 5 shows graphs of the oscillation coverage of the discrete drum gear heads depending on the drive rotation frequency. Analysis of these coupling graphs showed that when the rotational speed of the drive increases from $4.5 \cdot 10^3$ rpm to $7.2 \cdot 10^3$ rpm, the vibration coverage of the two edge gears increases linearly from $0.130 \cdot 10^{-3}$ m to $0.615 \cdot 10^{-3}$ m at values ΔY_1 and ΔY_3 $F_1 = (12 + 2.3 \sin \omega t \pm 0.25)$ sN, where ΔY_1 and ΔY_3. The laws of change are almost the same, except that the oscillation phases shift.



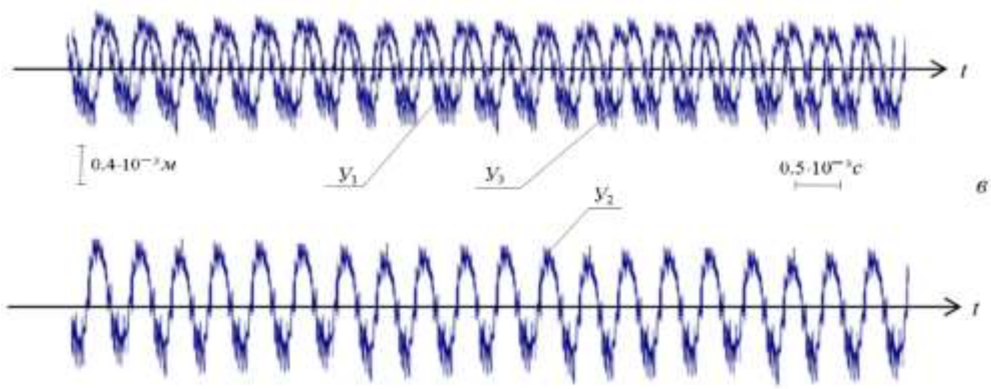
$F_л = (12 + 3.35 \sin \omega t \pm 0.25) sN$; $c = 0.25 \cdot 10^3$ N/m;

$n_\partial = 7.5 \cdot 10^3$ rev per min; $v = 2{,}1$ Ns/m;



$F_л = (18 + 3.35 \sin \omega t \pm 0.45) sN$; $c = 0.25 \cdot 10^3$ N/m;

$n_\partial = 7.5 \cdot 10^3$ rev per min; $v = 2{,}1$ Ns/m;

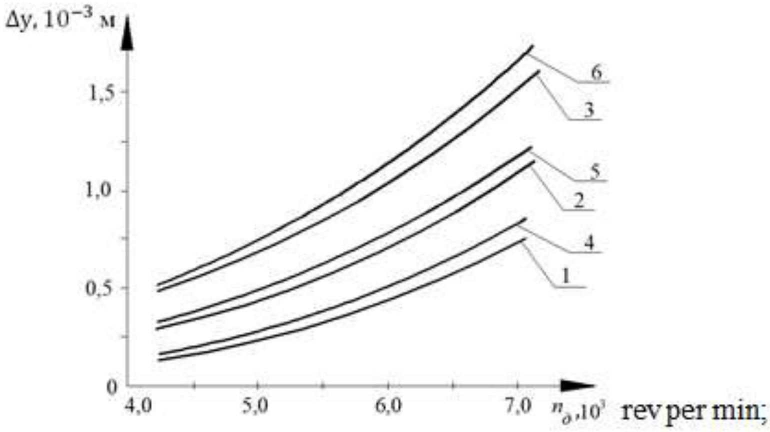


$F_л = (24 + 4.55 \sin \omega t \pm 0.6) sN$; $c = 0.25 \cdot 10^3$ N/m; $n_\partial = 7.5 \cdot 10^3$ rev per min; $v = 2{,}1$ Ns/m;

Figure 4. The dependence of the laws of vertical vibration of the discrete drum gear sets of the proposed composition on the technological and drive frequency is given.

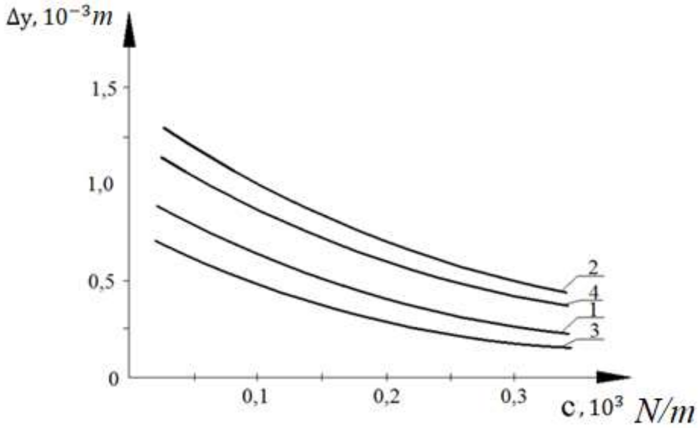
But the vibration coverage of the gear set in the middle will be greater than the vibration coverage values for the $F_2 = 12 \cdot F_1$ pieces. Correspondingly, the values of $\Delta Y_1 = \Delta Y_3$ increase in a nonlinear pattern from $0.485 \cdot 10^{-3}$ m to $1.42 \cdot 10^{-3}$ m. It should be noted that when the coverage values of the oscillations of the middle gear set are $F_1 = (12 + 2.3\sin \omega t \pm 0.25)$sN, respectively, the values of ΔY_2 increase in a nonlinear pattern from $0.138 \cdot 10^{-3}$ m to $0.79 \cdot 10^{-3}$ m.

When the technological resistance F_1=24,0 sN, the values of ΔY_2 increase from $0.54 \cdot 10^{-3}$ m to $1.62 \cdot 10^{-3}$ m. According to the results of experimental studies, the selection of the drive speed in the range of 10^{-3} rpm $(7{,}0 \div 7{,}5)$ to ensure that the vertical vibrations of the gear headset, as well as the difference between the teeth of the headset does not exceed $(1{,}0 \div 1{,}2) \cdot 10^{-3}$ m expedient. However, it is possible to increase the values of $p_д$ to $(10 \div 12) \cdot 10^3$ rpm by selecting the appropriate values of shock absorber virginity.



$1,2,3 - \Delta Y_1 = \Delta Y_3 = f(n_\partial)$; $4,5,6 - \Delta Y_2 = f(n_\partial)$; $1,4 - F_л = (12 + 2.3\sin \omega t \pm 0.25)$sN;
$2,5 - F_l = (18 + 2.35 - \sin \omega t \pm 0.45)$ sN; $3,6 - F_l = (24 + 4.3\sin \omega t \pm 0.65)$sN;

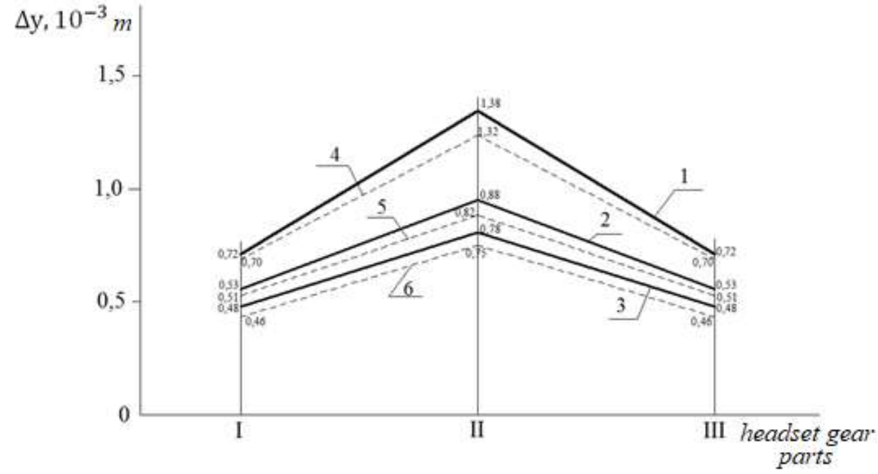
Figure 5. Graphs of the frequency dependence of the oscillation coverage of the discretion drum gear sets



1.3-I, III-section gear headset zones;
2-4-II- middle section gear headset zone;
$1,2 - F_л = (12 + 2.3\sin \omega t \pm 0.25)$sN; $3,4 - F_л = (18 + 2.35 - \sin \omega t \pm 0.45)$sN;
$n_\partial = 6 \cdot 10^3$ rev per min.

Figure 6. Graphs of the dependence of the vibration coverage of discrete drum gear sets on the coefficient of virginity of the rubber shock absorber

Figure 6 shows graphs of the dependence of the vibration coverage of discrete drum gear sets on the rubber shock absorber coefficient. According to the analysis of graphs, when the shock absorber height increases from $0,65 \cdot 10^3$ N/m to $0,35 \cdot 10^3$ N/m and F_1=12,0 cH, $\Delta y_2 = \Delta y_3$ values decrease from $0.62 \cdot 10^{-3}$m to $0.132 \cdot 10^{-3}$ m, respectively F_1=18,0 sN, the values of $\Delta y_1 = \Delta y_3$ decrease in a linear pattern from $0.93 \cdot 10^{-3}$ m to $0.26 \cdot 10^{-3}$ m (Fig. 2.5, Figures 1.3).



Gear headset parts

$1{,}4 - F_l = (24 + 4.5 \sin \omega t \pm 0.65)$cH; $2{,}5 - F_l = (18 + 3.35 - \sin \omega t \pm 0.45)$sN;

$3{,}6 - F_л = (12 + 2.3 \sin \omega t \pm 0.25)$cH; $1{,}2{,}3 - n_д = 7.5 \cdot 10^3$ rev per min; $c = 0.12 \cdot 10^3$ N/m; $в = 1.3$ Нс/м; $4{,}5{,}6 - n_д = 6.0 \cdot 10^3$ rev per min; $c = 0.25 \cdot 10^3$ N/m; $в = 2.1$ Ns/m;

I-first tooth headset part; II - part of the second gear set;

III-third gear headset part;

Figure 7. Graphs of the dependence of the parameters of the vibration coverage of the recommended discrete drum gear sets

The vibration coverage of the middle gear headset decreases from $1{,}28 \cdot 10^3$m to $0{,}43 \cdot 10^{-3}$m when F_1=12,0 sN. It can also be seen that when F_1=18,0 sN, the values of Δy_2 decrease in a nonlinear bond from $1{,}21 \cdot 10^3$ m to $0{,}51 \cdot 10^{-3}$m. It is therefore advisable to obtain the virulence values of the generalized shock absorber greater than $(0{,}2 \div 0{,}35) \cdot 10^{-3}$ N/m to ensure that the values of $\Delta y_1, \Delta y_2$ and Δy_3 are in the range of $(1{,}0 \div 1{,}2) \cdot 10^{-3}$m.

Figure 7 shows graphs of the dependence of the proposed discrete drum gear heads on the vibration coverage parameters. The analysis of the graphs shows that the coverage of vertical vibrations of the gear set in the middle zone can be seen as the technological resistance and the magnitude of the vibration coverage of the gears on both sides increase with increasing drive speed. However, it is possible to equalize the values of $\Delta y_1, \Delta y_2$ and Δy_3 by installing a separate shock absorber for each gear set and installing rubber bushings in the corresponding brackets.

Therefore, it is recommended to obtain a rubber bushing coefficient of virginity in the range of $(0.18 \div 0.20) \cdot 10^3$ Н/м for the shock absorbers of the two end gears and $(0.32 \div 0.35) \cdot 10^3$ Н/м for the shock absorbers of the average gear headset.

FINDINGS

An effective structural scheme of a composite sampling drum has been developed. On the basis of theoretical studies, the laws of vertical vibrations of the gear sets of the drum are obtained, and its parameters are substantiated.

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