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The New Design and Parameters of the Preparation of Gears

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Abstract:

Objective. The article presents options for extension circuits with gears with the recommended content. A new formula for determining the degree of freedom of gears has been proposed. Here's how to eliminate redundant connections in an extension. The results of the calculation of the shear deformation of the rubber bushing of the elastic element of the gear are given. The results of the recommended transmission production tests are given.

Methods. In the process of research, higher mathematics, theory of machines and mechanisms, theory of oscillations, dynamics of machines, test methods of mechanical engineering and technological machines were used

Results. This is due to the fact that the rational parameters and operating mode of the device are determined based on the analysis of the created models and laws of motion.

Conclusions: The generalized formula for determining the degree of excitability of flexible gear mechanisms was proposed. At the same time, a method of eliminating redundant connections in flat mechanisms was developed.

Keywords: Gear, extension, flexible element, structure, degree of freedom, plus coupling, shear modulus, deformation, test, efficiency, resource.

Methods. The analysis of mechanisms is divided into: structural, kinematic and dynamic [1]. In this case, a structural analysis of the mechanisms is important, in which the level of the moving mechanism is determined. The degree of mobility of the mechanism allows to determine the number of moving links, kinematic pairs, as well as the number of transmissions. It is known that the degree of motion of mechanisms is determined by Chebyshev's formula [2,3]. It should be noted that when using the Chebyshev formula, the joints of the mechanism are absolutely rigid, elastic joints and connections are not taken into account.

Figure 1 shows the gears in 3 different variants: a-gear transmission consists of a gear 1 and a wheel 2; the b-steering wheel structure has a flexible element; v-both gears are also made integral.

Let us consider a structural analysis of the gear mechanisms shown in Figure 1. The degree of mobility of the considered mechanisms is determined according to the methodology presented in the works [4;5]:

$$W=3n-2P5-P4=3•2-2•2-1=1 \quad (1)$$

Where n is the number of moving links,
P5 is the number of fifth grade kinematic pairs,
P4 is the number of fourth-class kinematic pairs.

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Results.

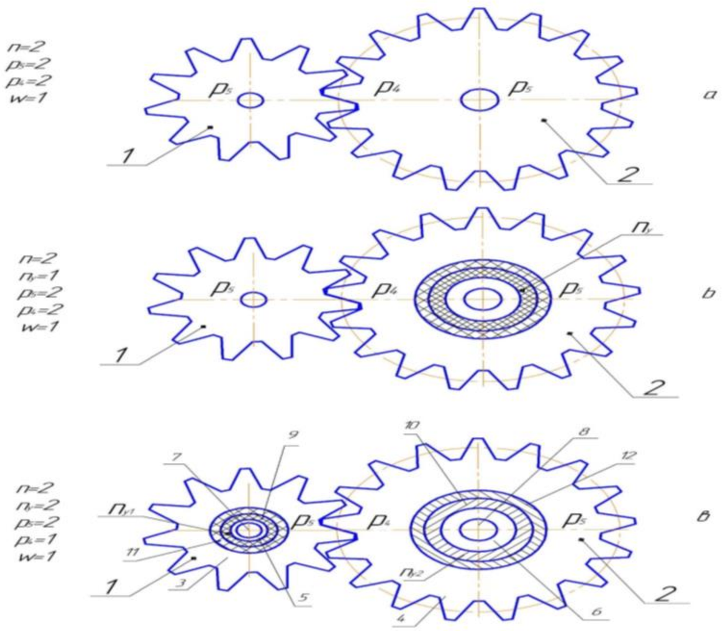
For all variants of gear mechanisms shown in Figure 1.1, the degree of movement is equal together [6;7]. However, it is important to identify redundant connections in mechanisms that can lead to a significant reduction in work resource due to increased friction, vibration, and unnecessary reactions.

Figure 1. Gear extension scheme

Excess links are defined by the following formula:

$$q=W-6n+5\,P5+\,4P4=1-6\cdot 2+5\cdot 2+4\cdot 1=3 \qquad (2)$$

We recommend the use of elastic element mechanisms to reduce the friction in the kinematic pair,



to increase the working resource of the mechanism. To do this, a formula is recommended that takes into account the elastic elements in the detection of excess links in the mechanisms [8;9]:

$$q=W-6n+5\,P5+\,4P4-ny \qquad (3)$$

Where, is the number of elastic elements in the ny-mechanisms.

As can be seen from formula (3), each elastic element inserted into the mechanism reduces the excess bond to one.

Also:

Figure 1 for option “a”: q = 3;

Figure 1 for option “b”: q = 2;

Figure 1 for option “v”: q = 1.

Analysis of the results shows that the third connection option (Fig. 2, c) is the most optimal, in which the excess connection is reduced to one.

To completely eliminate excess connections in the gear transmission, it is recommended that the drive gear be fitted with rubber bushings to the housing for mounting on the bearing on the rotation shaft.

It is known that gear transmissions are widely used in technological machines [10].

The main disadvantages of these mechanisms are the rigid interaction that binds the teeth of the wheels together and the transfer of changes in loads directly to the shafts of the gears.

In the proposed new scheme (see Fig. 2.1-b, v.) The gear 4 and the gear 1 are integral. The gear 1 is fastened to the shaft 2 by means of the shock absorber-bushing 3 and the gear 4 is fastened to the shaft

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5 by the shock absorber-bushing 6.

In this case, the thickness of the shock absorber-bushings 3 and 6 is selected according to the transmission ratio.

$$\Delta_1 = \frac{d_1 - d_1'}{2} \quad \Delta_2 = \frac{d_2 - d_2'}{2} \qquad \textbf{(4)}$$

From:

$$U_{12} = \frac{\omega_1}{\omega_2} = \frac{R_2}{R_1} = \frac{\Delta_2}{\Delta_1};$$

where - the outer and inner diameters of the shock-bushing 3, gear 1; - outer and inner diameters of shock absorber-bushing 6, wheel 4; and the radii of the main circles of -shesternya 1 and wheel 4; and the angular velocities of -shesternya 1 and wheel 4; -transmission ratio.

Figure 2 shows the bushing-shock absorber shift on the gear transmission deformation calculation scheme is presented. Due to the deformation of this bushing rubber, the outer bushing of the shock absorber armature rotates at an angle, the sliding angle of the bushing rubber is equal to:

$$tg\gamma = \frac{\Delta\varphi_1 r}{\Delta r} \qquad (5)$$

In this case, the sliding surface of the selected bushing rubber will be as follows:

$$tg\gamma = \frac{\Delta\varphi_1 r}{\Delta r} \qquad (5)$$

Where l is the length of the bushing (shesternya).

$$F = 2\pi r l \qquad (6)$$

Rotational shear force of rubber in use, equality:

$$Q = GFtg\gamma = 2\pi r l G \frac{r\Delta\varphi_1}{\Delta r} \qquad (7)$$

Where is the shear modulus of the rubber, N / m2.

External torque value:

$$M_1 = Qr = 2\pi G l r^3 \frac{\Delta\varphi_1}{\Delta r} \qquad (8)$$

From (8) we can obtain the following:

$$\Delta\varphi_1 = \frac{\Delta r M_1}{2\pi G l r^3} \qquad (9)$$

By redistribution from (9) the following can be obtained:

$$\varphi_1 = \frac{M_1}{2\pi G l} \int_{r_1'}^{r_1} \frac{dr}{r^3} = \frac{M_1}{2\pi G l} \left[\frac{1}{2(r_1')^2} - \frac{1}{2(r_1)^2} \right] \qquad (10)$$

A similar expression can be obtained from [11]:

$$\varphi_2 = \frac{M_2}{4\pi G l} \left[\frac{1}{r_2'^2} - \frac{1}{r_2^{\ 2}} \right] \qquad (11)$$

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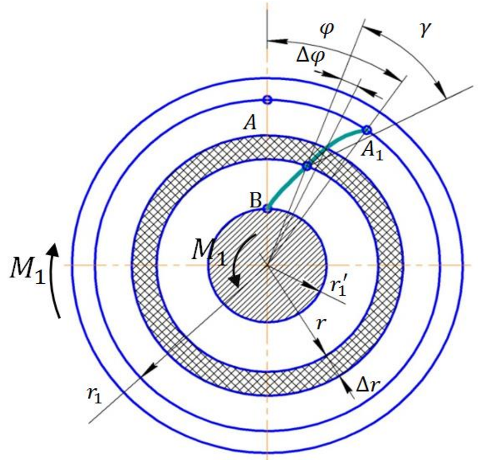
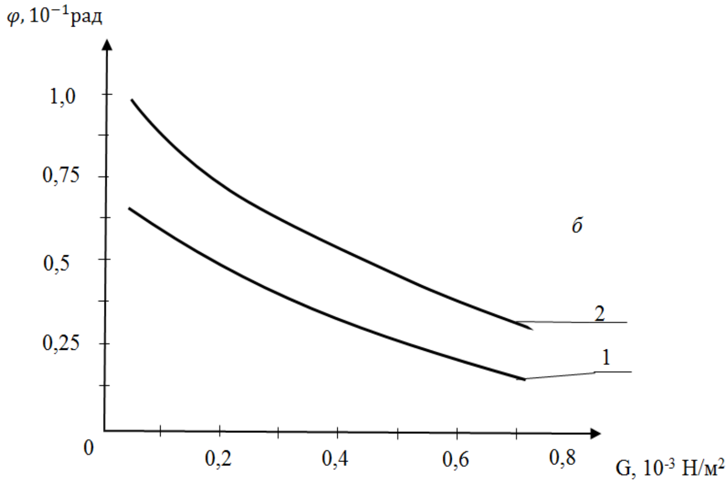


Figure 2. Scheme for calculating the bushing-shock absorber shear deformation in the gear transmission

In this case, the gear ratio is variable and is determined from the following formula

$$И_{12} = \frac{M_1\left[\frac{1}{r_2'^2} - \frac{1}{r_2^{\,2}}\right]}{M_2\left[\frac{1}{r_2'^2} - \frac{1}{r_2^{\,2}}\right]} \tag{12}$$

The initial data are as follows: M_1 = 8.2 Nm; M_2 = 6.3 Nm; p = 3.14; l = 24.2 · 10-3 m; r_1 = 3.8 · 10-2 m; Calculations were performed on r2 = 5,6 · 10-2 m and on the basis of them a graph of the laws of



variation of the values M of the modulus of displacement of the rubber shear and the external torque of the rubber bushings on the gears was obtained (Fig. 3 a) [12].

1, 2-ph1 = f (M1); 3, 4-ph2 = f (M2); When 1-r1 = 2.1 · 10-2 m; When 2-r1 = 3.2 · 10-2 m; When 3-r1 = 3.8 · 10-2 m; When 4-r1 = 5.0 · 10-2 m

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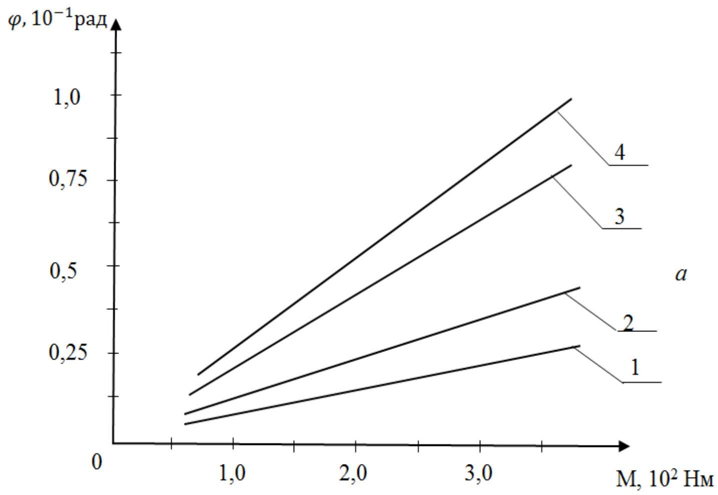

Figure 3. Graph of change of modulus of displacement of angular shear deformation of gear and bushing-shock absorber in gear

As the external torque M_1 and M_2 increase, the angular shear deformation of the rubber bushings on the gears increases in linear motions. It is recommended to obtain the external torques M_1 and M_2 at values M starting from $0.062 \cdot 10^2$ Nm to reduce the shocks that occur when gears engage at relatively large values M of (1 and (2) (Figure 3 b).

Given a torque of $0.37 \cdot 10^2$ Nm (1 increases from $0.094 \cdot 10^{-1}$ rad to $0.24 \cdot 10^{-1}$ rad, the angular displacement in m and m increases to 0.402 · 10-1 rad. Accordingly, the gear wheel has a flexible rubber the sliding angle of the bushing increases to $0.71 \cdot 10^{-1}$rad,,, rad.

It is recommended to take the external torques $M_1 = (0.025\ldots 0.028) \cdot 10^2$ Nm, $M_2 = (0.03\ldots 0.036) \cdot 10^2$ Nm to reduce the shocks caused by the coupling of the gears at relatively large values M of 1 and (2). Angle of rubber bushings it is advisable to obtain the shear modulus at values $M_1$of (0.33 ... 0.42) · 103 N / m3 to reduce the amount of shear deformation.

Conclusions: a) The generalized formula for determining the degree of excitability of flexible gear mechanisms was proposed. At the same time, a method of eliminating redundant connections in flat mechanisms was developed.

b) The formulas for determining the angular shear deformations of the flexible bushings of the gears were obtained. Graphs of the dependence of the angular deformations of the flexible bushings of the gears on the torque moments on their shafts were constructed.

c) In the extension, the recommended values of the parameters were taken into account, taking into account the reduction of the impact on the interaction of the wheel teeth.

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