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DETERMINATION OF GEAR EXTENSION PARAMETERS OF A COMPOSITE FLEXIBLE ELEMENT

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Abstract. The article presents recommended options for extension schemes for gear transmissions. A new formula for determining the degree of freedom of gears is proposed. A method for eliminating redundant connections in the extension mechanism is shown. The results of calculating the shear deformation of the rubber bushing used as the elastic element of the transmission are given. The results of production tests of the proposed transmission design are also presented.

Key words: gear transmission, extension mechanism, elastic element, structure, degree of freedom, additional coupling, shear modulus, deformation, test, efficiency, resource.

Annotatsiya. Maqolada tishli uzatmalar uchun kengaytiruvchi sxemalarning tavsiya etilgan variantlari keltirilgan. Tishli uzatmalarning erkinlik darajasini aniqlash uchun yangi formula taklif etilgan. Kengaytiruvchi mexanizmdagi ortiqcha bog'lanishlarni bartaraf etish usuli ko'rsatilgan. Tishli uzatmaning elastik elementi sifatida qo'llanilgan rezina vtulkaning siljish deformatsiyasini hisoblash natijalari keltirilgan. Shuningdek, tavsiya etilgan uzatma konstruktsiyasining ishlab chiqarish sinovlari natijalari bayon etilgan.

Kalit so'zlar: tishli uzatma, kengaytiruvchi mexanizm, egiluvchan element, tuzilma, erkinlik darajasi, qo'shimcha mufta, siljish moduli, deformatsiya, sinov, samaradorlik, resurs.

Аннотация. В статье представлены рекомендуемые варианты расширительных схем для зубчатых передач. Предложена новая формула для определения степени свободы зубчатых передач. Показан способ устранения избыточных связей в расширительном механизме. Приведены результаты расчёта сдвиговой деформации резиновой втулки, используемой в качестве упругого элемента передачи. Также представлены результаты производственных испытаний предлагаемой конструкции передачи.

Ключевые слова. Зубчатая передача, расширительный механизм, упругий элемент, структура, степень свободы, дополнительная муфта, модуль сдвига, деформация, испытание, эффективность, ресурс.

Introduction

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.

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

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

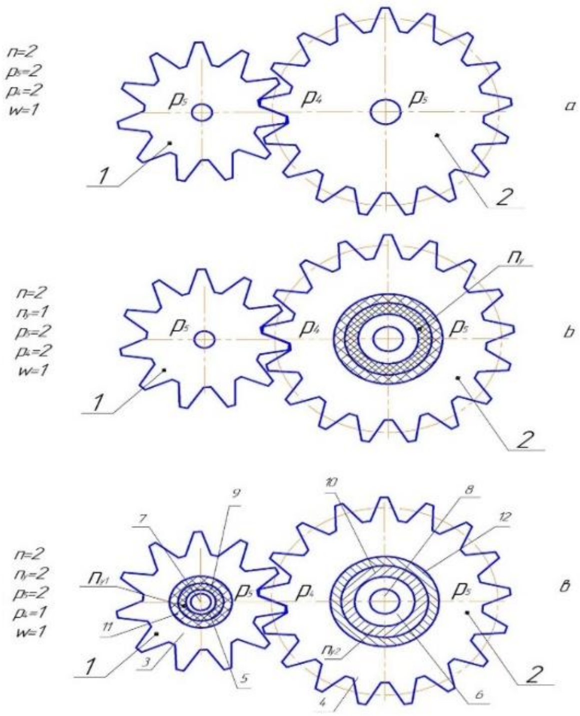

Figure 1. Gear extension scheme.

Excess links are defined by the following formula:

q=W-6n+5 P5+ 4P4=1-6•2+5•2+4•1=3 (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 (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 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.

$$\Delta_1 = \frac{d_1 - d_1'}{2} \quad \Delta_2 = \frac{d_2 - d_2'}{2} \tag{4}$$

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:

$$tg\gamma = \frac{\Delta\varphi_1 r}{\Delta r} \tag{5}$$

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

$$tg\gamma = \frac{\Delta\varphi_1 r}{\Delta r} \tag{5}$$

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

$$F = 2\pi rl \tag{6}$$

Rotational shear force of rubber in use, equality:

$$Q = GFtg\gamma = 2\pi rlG\frac{r\Delta\varphi_1}{\Delta r} \tag{7}$$

Where is the shear modulus of the rubber, N/m^2.

External torque value:

$$M_1 = Qr = 2\pi Glr^3\frac{\Delta\varphi_1}{\Delta r} \tag{8}$$

From (8) we can obtain the following:

$$\Delta\varphi_1 = \frac{\Delta r M_1}{2\pi Glr^3} \tag{9}$$

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

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

A similar expression can be obtained from [11]:

$$\varphi_2 = \frac{M_2}{4\pi Gl}\left[\frac{1}{r_2'^2} - \frac{1}{r_2^{\ 2}}\right] \tag{11}$$

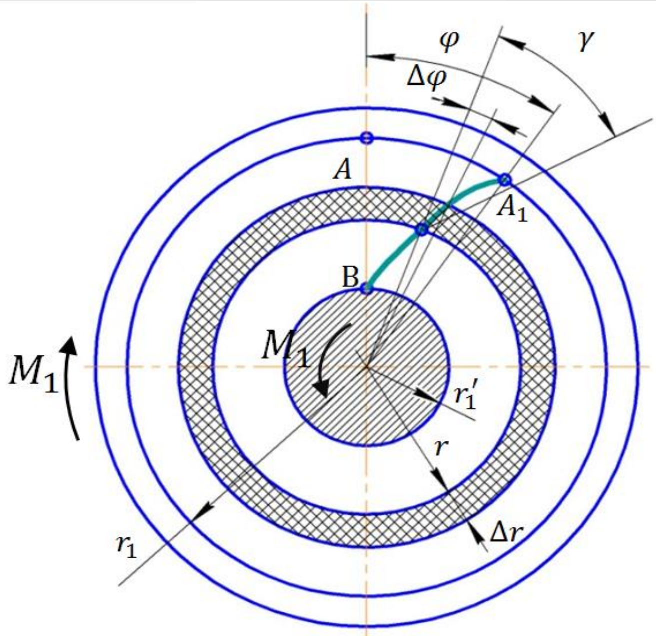

Figure 2. Scheme for calculating the bushing-shock absorber shear deformation in the gear transmission.

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

$$I_{12} = \frac{M_1\left[\frac{1}{{r'_2}^2} - \frac{1}{{r_2}^2}\right]}{M_2\left[\frac{1}{{r'_2}^2} - \frac{1}{{r_2}^2}\right]} \qquad (12)$$

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 *r_2=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].

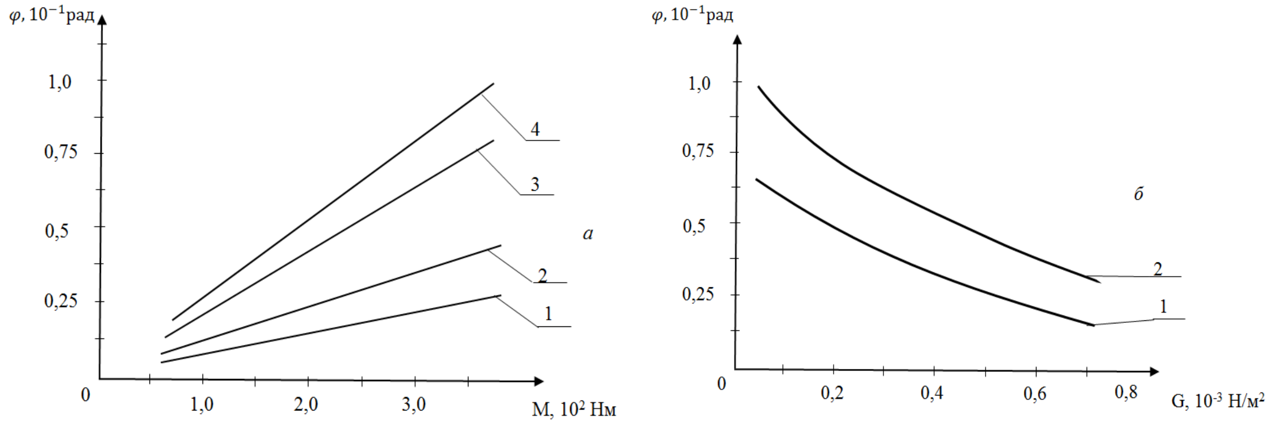

Figure 3. Graph of change of modulus of displacement of angular shear deformation of gear and bushing-shock absorber in gear.

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

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 · 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

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.

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.

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.

References:

[1]. Кенжабоев Ш. Джураев А. Бекназаров Ж Исследование деформации сдвига амортизатора-втулки составных зубчатых колес передачи. Фарғона водийси ҳудудларидаги маҳаллий хом-ашёлардан фойдаланиш асосида импорт ўрнини босувчи махсулотлар ишлаб чиқаришнинг долзарб масалалари. Наманган шахри 27-28 октябрь, 2018-йил.

[2]. Смелягин А.И. Структура механизмов и машин. Новосибирск.Изд. НГТУ, 2002-38 с.

[3]. Джураев А., Кенжабоев Ш. Структурнқй анализ и синтез кулисных механизмов с гибком звеном:Монография.-Т.: Фан, 2004.-120 с.

[4]. Давидбоев Б.Н., Джураев А., Зулпиев С.М., Давидбаева Н.Б. Структурный, кинематический и динамический анализ рычажно-шарнирной муфты с упругими элементами карданных механизмов. Монография, Изд “Фергана”, Фергана 2013, ISBN 978-9943-349-69-4, С.116, ФерПИ, 15.11.2012, №2-Баённома.

[5]. Джураев А., Зулпиев С. Структурный анализ рычажно-шарнирной муфты. Ж. «ФерПИ илмий техника журнали», Фергана, 2009, №2, С.30-32.

[6]. Файзиев И.Х., Джураева М.Ю.Распространинея структурной формулы Чебышева на механизмы с гибкими звенъями. // Динамика механизмов технологических машин. Науч. Труды / Им и СС Ан УзССР. Ташкент, Фан, 1983,-С.55-61.

[7]. Алимухамедов Ш.П., Соатов М.М. Структурный анализ и методы устранения избыточных связей механизма газораспределения автомобилных двигателей//.Ж. Вестник ТАДИ.-Ташкент.2017,-№2-С.8-12.

[8]. Djuraev A., Madrahimov Sh.H., Urinova S.I. Ways of reducing superfluous communications in battant mechanism of weaving looms. European science review, Austria, Vienna, 2016, № 1–2, 142-144 p.

[9]. Кенжабоев Ш. Джураев А., Мадрахимов Ш. Определение избыточных связи и в кинематических парах кулачковых механизмов. Тўқимачилик саноати корхоналарида ишлаб чиқаришни ташкил этишда илм-фан интеграциялашувини ўрни ва долзарб муаммолар ечими. Халқаро илмий-техникавий анжуман 2-қисм. 133-137 б., 27-28 июль 2017 йил. Марғилон.