Development of an Effective Resource-Saving Design and Methods for Calculating the Parameters of Gears with Compound Wheels

A. Djuraev, J.Kh. Beknazarov, Sh.Sh. Kenjaboev

Abstract*: *The article presents the design schemes and the principle of operation of the gear transmission with composite wheels and elastic elements. The technique and the results of the analysis to determine the degree of mobility of gears with composite wheels and elastic elements are given. The angular displacement of the composite gears and gear ratios are obtained by the analytical method. Graphic dependencies are constructed and the values of the system parameters are recommended.

Keywords*: *Gear, wheel, gear, elastic elements, gear ratios, degree of mobility, shear angle, bushing deformation, angular velocity, area, length, moment, shear modulus.

I. INTRODUCTION

This design of the gear transmission contains the driving and driven gears, the teeth of which gear transmit rotational motion from the shaft of the driving gear-pinion gear to the driven gear [1].

The disadvantage of this transmission is the lack of absorption of peak fluctuations in the loads (moments) on the transmission shafts when using the transmission in the drives of technological machines operating with variable loads. This leads to rapid wear of the teeth of the wheels, high noise, failure of the bearing bearings, thereby reducing the resource of operation of the gear transmission, especially at high speed operating modes.

In another design of the gear train to obtain a variable gear ratio, the gears are mounted on the shafts eccentrically [2].

II. RESEARCH METHODS

To reduce the peak values of load fluctuations on the transmission shafts, increase the service life, and also reduce noise, especially at high speed operating modes, the design of gear wheels, including elastic elements, has been improved.

The gear train contains (Fig. 1) a composite drive 1, driven 2 gears, which include a rim 3 and 4 with teeth, hubs 5 and 6 are rigidly mounted on shafts 7 and 8. Between rims 3, 4 and hubs 5 and 6 are rigidly mounted elastic (rubber) bushings 9, 11 and 10, 12. [3]

In this case, the gear wheel 2 and gear 1 have two rubber bushes inner 9, 10 and outer 11 and 12. The circular stiffness C_1 of the rubber bushings 9 and 12 are the same and are made of the same type of rubber, as well as the torsional stiffness C_2 of the rubber bushings (rubber grade) 10 and 11 are chosen the same, while $C_2 > C_1$.

The gear drive operates as follows: The rim 3 with the teeth of the driving gear 1 receives rotational movement from the drive motor (not shown in the figure), the shaft 7, the hub 5, through the elastic bushings 9 and 11. When the gear teeth of the wheels 1 and 2 are engaged, the rotational movement is transmitted to the rim 4 with teeth and through the elastic bushings 12 and 10 of the hub 6 and to the shaft 8. Under the influence of external loads and technological resistances, the torque on the shaft 8 changes. Moreover, the peak values of the moment fluctuations are absorbed by the elastic bushings 10 and 12. Further, additional peak loads are absorbed in the resilient bushings 9 and 11, the driving gear 1 (gear). In this case, the torques on the shafts 7 and 8 will be smoothed to some extent.

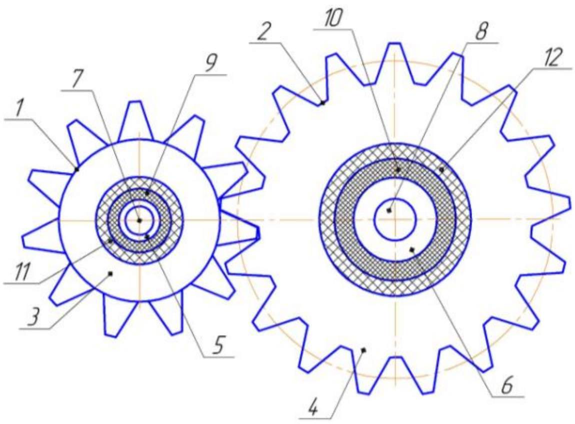


Fig. 1. Wheelwork

When transmitting moments between the gear 1 and the wheelwork 2, the rubber bushings 9 and 12 will be less deformed in the circular direction, since their circular stiffness's are chosen larger than the circular stiffness's of the rubber bushings 10 and 11.

In this case, the rubber bushings 10 and 11 will be deformed more due to $C_2 > C_1$. This provides a wide range of frequency of absorption of fluctuations in transmitted moments between gear 1 and gear 2

Revised Manuscript Received on November 05, 2019.

Anvar Djuraevich Djuraev (Tashkent, Uzbekistan) - Doctor of Technical Sciences, Professor, Department of general technical disciplines, Tashkent Institute of Textile and Light Industry, Tashkent city, Shohjahon street 5. e-mail: anvardjurayev1948@mail.ru

Beknazarov Jasur Kholmamatovich (Navoi, Uzbekistan) doctoral student of the Navoi branch of the Academy of Sciences of the Republic of Uzbekistan

Kenjaboev Shukurjon Sharipovich (Namangan, Uzbekistan) Doctor of Technical Sciences, Rector of Namangan Engineering Construction Institute

The gear transmission provides a reduction in peak values of load fluctuations on the transmission shafts, an increase in the service life, and also reduces noise, especially at high speed modes and a wide range of technological loads.

A. Structural analysis of gear mechanism with compound wheels.

The analysis of mechanisms is divided into: structural, kinematic and dynamic [4]. At the same time, a structural analysis of mechanisms is important, in which the degree of moving mechanism is determined. The degree of mobility of the mechanism will allow the determination of the number of moving links, kinematic pairs, as well as the number of drives. It is known that the degree of mobile planar mechanisms is determined by the Chebyshev formulas [5, 6]. It should be noted that when using the Chebyshev formula, the links of the mechanism are considered absolutely rigid, elastic links and bonds are not taken into account.

Figure 2 shows three options for the gear: a-gear consisting of gear 1 and wheel 2; b-driven wheel is made integral, has an elastic element; in-both gears are made integral.

Consider the structural analysis of the gear mechanisms shown in Fig. 1. The degree of mobility of the considered mechanisms is determined according to the technique given in [5,7]:

$$W=3n-2P_5-P_4=3\cdot2-2\cdot2-1=1 \qquad (1)$$

where, n – Is the number of moving links, P5 is the number of kinematic pairs of the fifth class, P4 is the number of kinematic pairs of the fourth class.

For all variants of gear mechanisms shown in Fig. 1, the degree of mobility is equal to unity [7,8]. At the same time, it is important to determine excess bonds in mechanisms that can lead to a significant decrease in the resource of work due to increased friction, vibration, unnecessary reactions, etc.

Excessive bonds are determined from the formula:

$$q=W-6n+5P_5+4P_4=1-6\cdot2+5\cdot2+4\cdot1=3 \qquad (2)$$

To reduce friction in kinematic pairs, increase the resource of the mechanism, we recommend the use of elastic elements in the mechanisms. For this, a formula is recommended that takes into account elastic elements when determining excess bonds in mechanisms [7]:

$$q=W-6n+5P_5+4P_4-n_y \qquad (3)$$

where, n_y - is the number of elastic elements in the mechanisms.

From formula (3) it is seen that each elastic element included in the mechanism reduces the excess bond by one.

Moreover, we have:

For option “a” in fig. 1: q = 3;

For option “b” in fig. 1: q = 2;

For option “c” in fig. 1: q = 1.

An analysis of the results shows that the third (Fig. 2 c) option is considered the most acceptable, in which the excess bond is reduced to unity. For the complete elimination of excess connections in the gear transmission, it is recommended that the bearings be mounted on the shaft of the driven gear in the housing by means of a rubber cushion.

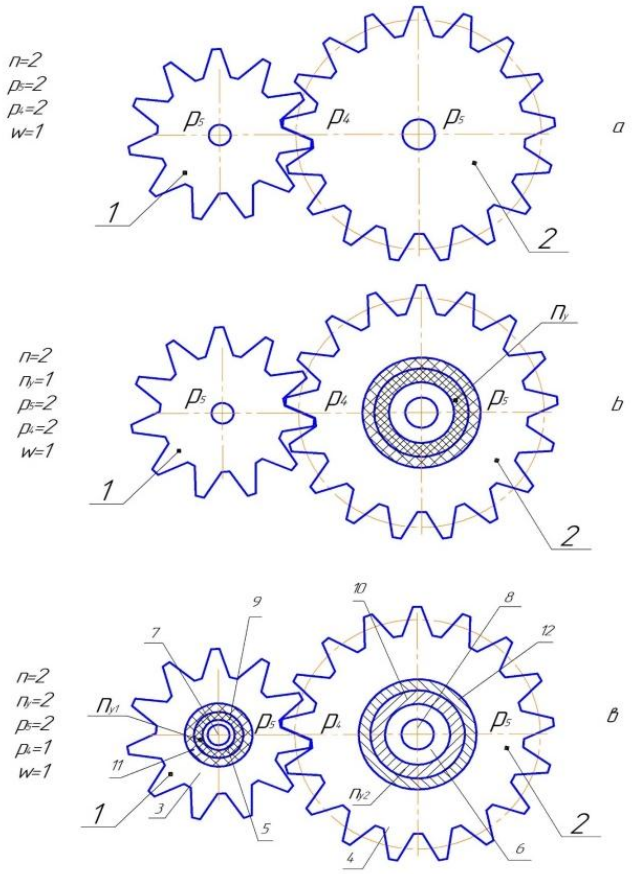


Fig. 2. Gear Schemes

Shift deformations of a shock absorber-sleeve of composite gears of a transmission

It is clear that, gear drives are widely used in the drives of technological machines [9]. The main disadvantage of these mechanisms is the rigid interaction of the teeth of the wheels when they are engaged, and the direct transmission of changes in the loads on the shafts of the gears. A new gear transmission scheme is recommended (see Fig. 2.b, c), in which the gear wheel 4 and gear 1 are made integral. The gear 1 is mounted on the shaft 2 by means of a shock absorber-sleeve 3, and the gear wheel 4 is mounted on the shaft 5 by means of a shock-absorber-sleeve 6. The thickness of the shock-absorbers-bushings 3 and 6 are selected according to the gear ratio:

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

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

where d_1, d_1' - the outer and inner diameters of the shock absorber-sleeve 3 gear 1; d_2, d_2' - the outer and inner diameters of the shock absorber-bush 6 of the wheel 4; R_1 and R_2 - the radii of the main circles of the gear 1 and wheel 4; ω_1 and ω_2 - corner speeds of gear 1 and wheel 4; U_{12} - gear ratio gear.

ISSN: 2278-3075, Volume-9, Issue-1, November 2019

III. ANALYSIS AND RESULTS

Fig. 3 is a diagram for calculating the shear strain of the shock absorber sleeve. When deriving the dependence of the angle of rotation of the shock absorber-sleeve on the magnitude of the external moment (the driving moment on the shaft and, accordingly, the moment of resistance from the gear when they are engaged), we select a thin-walled tube with a wall thickness and internal radius in the sleeve. Due to the deformation of the rubber of this tube, the outer sleeve of the shock absorber reinforcement will rotate by a small angle, and for the angle of shear of the rubber of the tube the equality is true [10]:

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

The surface area of the shear rubber selected tube:

$$F = 2\pi r l \tag{6}$$

Where, l - is the length of the sleeve (gear).

The circumferential shear force acting in rubber is:

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

Where, G - is the rubber shear modulus, N / m2.

The magnitude of the external torque (from the shaft moving and from the wheel as a moment of resistance):

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

From (8) we can obtain:

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

Integrating (9) in the redistribution and you can get:

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

In a similar way, one can obtain an expression for determining:

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

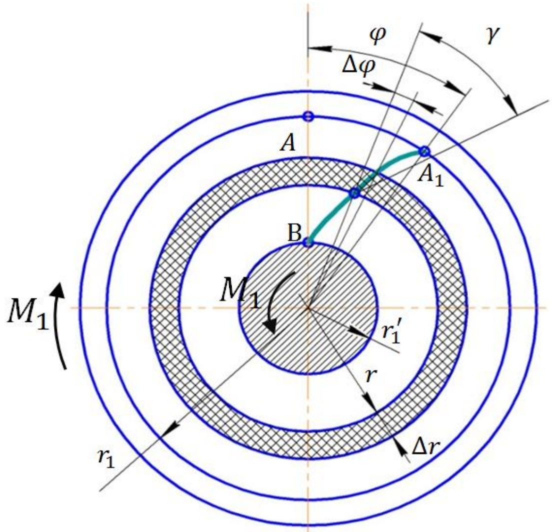
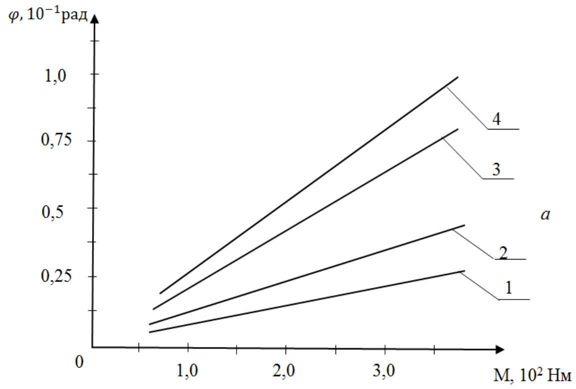


Fig. 3. The scheme for calculating the shear strain of the shock absorber-sleeve gear gear

In this case, the gear ratio of the gear is variable, it is determined from the following expression

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

Based on the numerical solution of (10) and (11) according to the following initial data, $M_1 = 8{,}2$ Nm; $M_2 = 6{,}3$ Nm; $\pi = 3{,}14$; $l = 24{,}2\cdot10^{-3}$ m; $r_1 = 3{,}8\cdot10^{-2}$ m; $r_2 = 5{,}6\cdot10^{-2}$ m, graphical dependences of the change in the angular shear deformations of the elastic rubber bushings of the gear wheels of the gears on the variation of the values of the rubber shear modulus and external torques (see Fig.4.) are plotted. An analysis of the obtained graphical dependences shows (see Fig. 4, a) that with an increase in the external moments M_1 and M_2, an increase in the strains of the angular shifts of the rubber bushings of the gears φ_1 and φ_2 is linear in nature [11, 12]. So, with an increase in M_1 and M_2, from 0.062 · 10^2 Nm to 0.37 · 10^2 Nm increases from 0.094 · 10^{-1} rad., to 0.24 · 10^{-1} rad with $r_1 = 2{,}1\cdot10^{-2}$ m and with $r_1 = 3{,}2\cdot10^{-2}$ m the angular shift φ_1 increases to 0.402 · 10^{-1} rad. Accordingly, for an elastic rubber sleeve of a gear φ_2 increases to 0.71 · 10^{-1} rad at $r_2 = 3{,}8\cdot10^{-2}$, $r_2 = 5{,}0\cdot10^{-2}$ and at $\varphi_2 = 1{,}03\cdot10^{-1}$, rad.



1, 2-$\varphi_1 = f(M_1)$; 3, 4-$\varphi_2 = f(M_2)$;
1- at: $r_1 = 2{,}1\cdot10^{-2}$ m; 2- at: $r_1 = 3{,}2\cdot10^{-2}$ m; 3- at: $r_1 = 3{,}8\cdot10^{-2}$m; 4-at: $r_1 = 5{,}0\cdot10^{-2}$ m

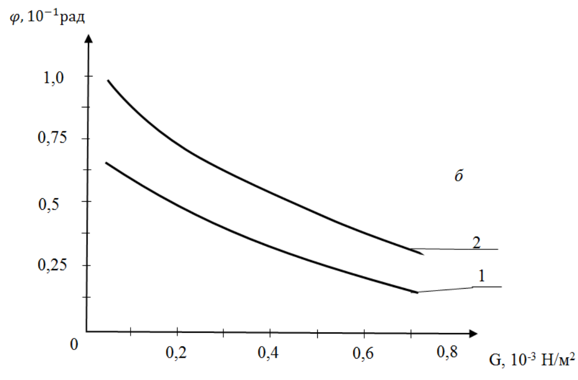


Fig. 4. Graphical dependence of the change in the strain of the angular shifts of shock absorbers - gear bushings and transmission wheels on the variation of the torques on the shafts (a) and the shear modulus (b)

To reduce the impact interactions of the teeth during engagement due to the large values of φ_1 and φ_2, the acceptable values of external moments are M_1 = (0.025 ... 0.028) $\cdot 10^2$ Нм, M_2 = (0.03 ... 0.036) $\cdot 10^2$ Nm. To reduce the values of the strain of the angular shifts of the rubber bushings, the shear modulus (0.33 ... 0.42) $\cdot 10^{-3}$Н/м^3 is considered appropriate (see Fig. 4).

IV. CONCLUSION

An effective resource is saving design of a gear transmission with composite wheels and elastic elements has been developed. The method recommendation determines the degree of sensitivity of gears with elastic elements. Studying the shear strain of the elastic elements of the wheels and obtaining a formula for determining the gear ratio of gears.

REFERENCES

1. Artobolevsky I.I. Theory of mechanisms and machines.- M .: Nauka, 1988. pp-640.
2. Frolov K.V. Theory of mechanisms and machines. -M .: Higher school, 1987. pp -496.
3. A.Jurayev, R.Kh. Maksudov Machine va mekhanlar nazariyashi 1-қism (Machine va mekhanlar tuzilish asoslari va kinematikashi) Kuv Ullanma ISBN 978-9943-11-945-1 “Fan va tehnologiya” nashirieti Toshkent 2019. pp -408.
4. A.Dzhuraev, Sh. Kenzhaboev Development of constructive schemes and the scientific basis for the analysis and synthesis of link mechanisms with elastic elements and flexible links of the drives of technological machines. Monograph Ed. Namangan 2019 p.267. ISBN 978-9943-5645-8-9.
5. A.Dzhuraev, Davidbaev B., Zulpiev S.M. Structural analysis of the lever-articulated coupling J. "FerPI ilmiy technique magazines", Fergana, 2009, No. 2, S.30-32.
6. A.S. 1370348 USSR, MKI F16H 21/00. Rocker mechanism / Magdiev M.A., Juraev A., Kuzibaev G.S., Mukhamedov D., Alimov P.T., Shukurov S.Z. IM and SS AN RU // B.I.-1988. Number 4.
7. A.S. 905549 USSR, MKI F16H 21/00. Rocker mechanism / Usmankhodzhaev Kh.Kh., Kuzibaev G.S., Malikov R.Kh., Magdiev M.A. IM and SS AN RU. // B.I. -1982. No. 6.
8. A.S. 1504430 USSR, MKI F16H 21/00, Rocker mechanism / Magdiev M.A. IM and SS AN RU // B.I.-1989. №32.
9. A.S. 1270462 USSR, MKI F16H 21/00. 06/19. Drawstring reversing mechanism / Magdiev M.A., Yuldashbekov S.A. IM and SS AN RU // B.I.-1986. №42.
10. A.S. 1379529 USSR, MKI F16H 21/00. Rocker mechanism / Magdiev M.A., Juraev A., Kuzibaev G.S., Malikov R.Kh. IM and SS AN RU // B.I. –1988. №9.
11. A.Djuraev, J.Kh.Beknazarov. Development of designs and methods for calculating gears with variable parameters and elastic elements. International journal of advanced research in science, engineering and technology Vol. 5, issue 5 , may 2018
12. O. O.Sayfidinov, J.Kh. Beknazarov. New prototype theory for cear transmission. Open Access Peer-Revieweed Journal 2(9) vol I February 2018

AUTHORS PROFILE

Anvar Djuraevich Djuraev (Tashkent, Uzbekistan) - Doctor of Technical Sciences, Professor, Department of general technical disciplines, Tashkent Institute of Textile and Light Industry, Tashkent city, Shohjahon street 5. e-mail: anvardjurayev1948@mail.ru

Beknazarov Jasur Kholmamatovich (Navoi, Uzbekistan) doctoral student of the Navoi branch of the Academy of Sciences of the Republic of Uzbekistan

Kenjaboev Shukurjon Sharipovich (Namangan, Uzbekistan) Doctor of Technical Sciences, Rector of Namangan Engineering Construction Institute