ОПУБЛИКОВАННЫЕ СТАТЬИ В ПЕРЕВОДЕ НА АНГЛИЙСКИЙ ЯЗЫК

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Forced Vibrations of Planetary Gears with Elements of the Increased Flexibility*

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The paper continues the research of dynamics of planetary gears with elements of the increased flexibility. The basis for the analytical model and input equations is the system of nine differential equations of dynamics of the planetary mechanism received earlier. In the generalized forces the engine moment, the moment caused by torsion of an elastic shaft of a sun gearwheel, and elastic forces in meshing of gearwheels and on satellites axes are considered.

The planetary carrier's speed is assumed to be constant. The rigidity of the axle of the satellite in the tangential direction considerably exceeds the rigidity of the axle in the radial direction, therefore, the displacement in the tangential direction is absent. The moment of the engine changes under the harmonious law. In this case forced vibrations of a sun gearwheel and the satellite can be considered separately from forced vibrations of the satellite in the radial direction owing to flexibility of the satellite axle. Amplitude-frequency characteristics of these oscillations are constructed. Characteristics of vibration processes are expressed through mass-dimensional, kinematic and strength parameters of the planetary gear taking into account flexibility of its elements.

Influence of the relative height of the flexible axle section, gear ratio of the mechanism and module of gearing on the amplitude of forced vibrations and the position of the resonance area is investigated. Conclusions are drawn on limits of variation of these parameters and their influences on the structure of planetary gears.

Keywords: planetary gear, flexibility of elements, dynamic, vibrations.

Introduction

Planetary gears are widely used in many areas of engineering. The multi-satellite planetary gears have additional advantages in front of in-line gears [1-3]. The development of rational designs of planetary gears with elements with increased flexibility continues [4–6]. This makes it possible to reduce unevenness of load on satellites [7–9], but leads to the need to study additional strength in elements the elements having the increased flexibility [10] and also possible vibration processes [11–15]. The relevance of a problem increases at considerable loadings and high rotary speeds [16–18].

In work [19] the planetary gear of type k-h-v is presented by gearwheels as solid bodies (figure 1) and elastic linkages between them are modeled as a springs; system of equations based on Lagrange's equations of the second kind on nine generalized coordinates including: the displacement of the satellite mass centre is represented due to axial flexibility in radial direction y_{gi} and in tangential direction x_{gi}; the rotate angle of planet carrier φ_h; the relative angle of satellite φ_{gi} due to flexibility of teeth; the relative angle of gearwheel φ_a due to flexibility of gearwheel shaft фтв gearwheel teeth is represented. It is considered natural vibrations of gearwheels due to flexibility of teeth.

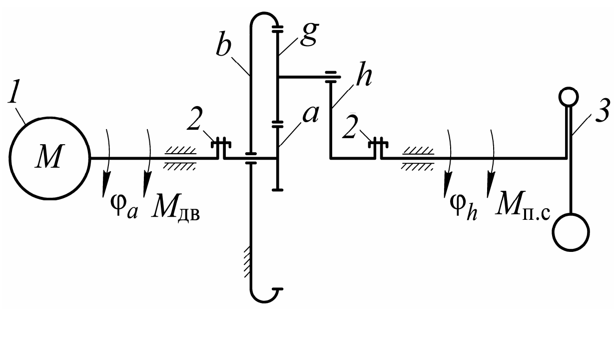

Fig. 1. The drive containing planetary gear: *b* - fixed central gearwheel, *g* - satellite, *h* - planet carrier, *a* - sun gearwheel, *1* - electric motor, *2* - coupling, *3* - actuator

The purpose of this work is the research of influence the increased flexibility of links on forced vibrations of planetary gears.

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Simplification of the original equations and the research of forced vibrations of the "sun gearwheel – satellite" system

In the generalized forces like in the general dynamic model [20] takes into account (figure 2): torque of engine $M_{дв}$; moment due to torsion of the elastic shaft of a sun gearwheel M_a; elastic forces in gear meshing of gearwheels. The moment $M_a = c_a \varphi_a$, where c_a is the torsional rigidity of the shaft. The elastic forces in bearing of satellite $F_{hgy} = c_{hgy} y_g$. Forses in gear meshing of motionless gearwheel b and satellite g $F_{nbg} = F_{ngb} = c_{bg} \Delta s_{nbg}$. Here c_{hgy} and c_{bg} are rigidity in gear meshing.

Since the force $F_{nbg} = F_{ngb}$ is directed along the line of contact located at an angle α_{gb} to the axis x_g, then the movement is due to the flexibility of the elements (figure 3) $\Delta s_{nbg} = \dfrac{\varphi_g r_g}{\cos \alpha_{gb}}$, where r_g is the satellite radius. The elastic force in the meshing of the sun gearwheel a and the satellite g is determined in a similar way.

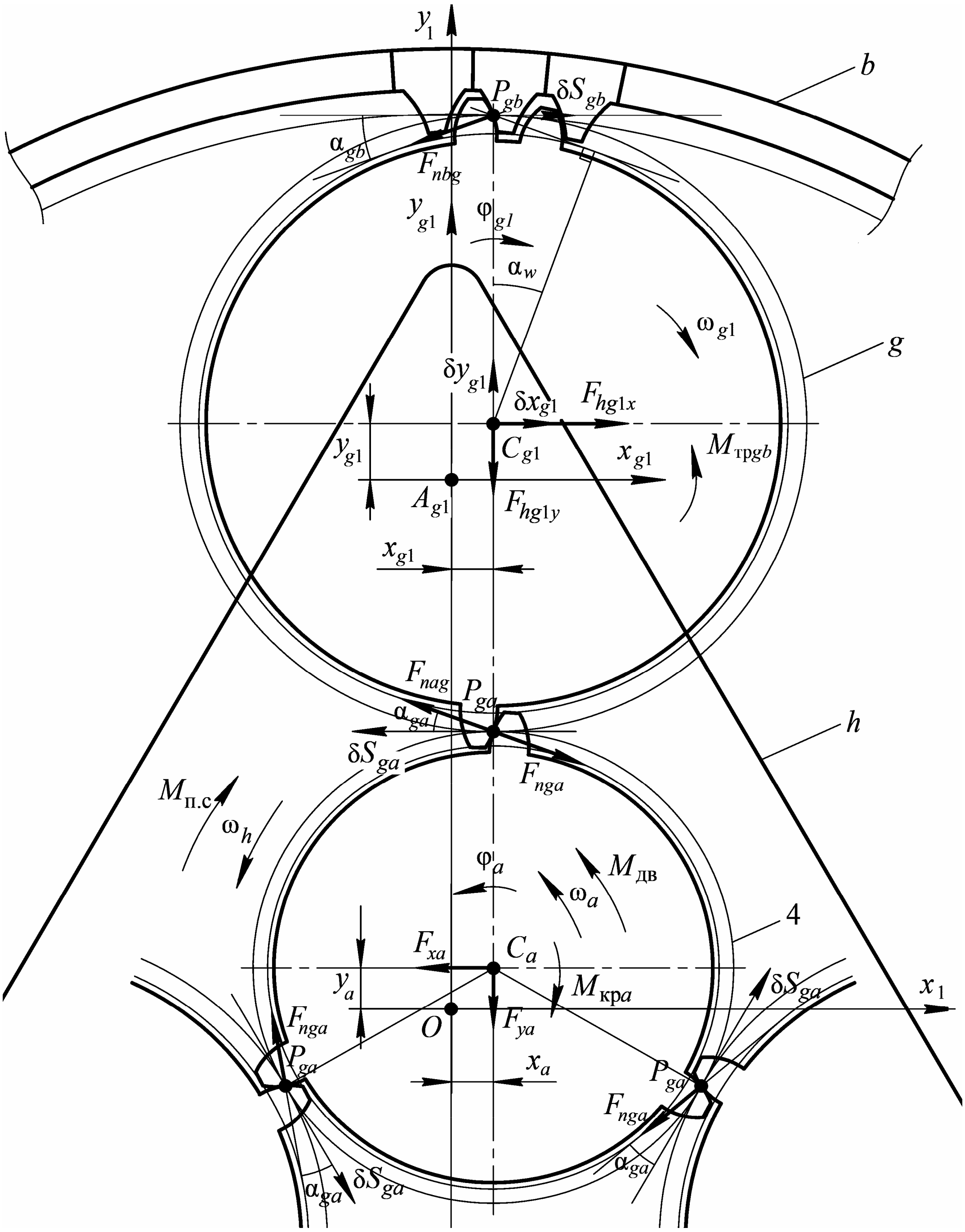

Fig. 2. The designe model with operating forces

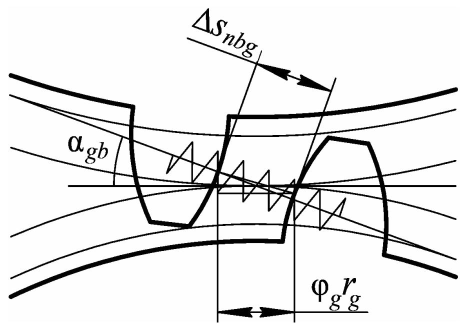

Fig. 3. Gear meshing “fixed central gearwheel – satellite”

To study forced vibrations of the planetary gear from a system of nine differential equations of second order, three equations were selected according to generalized coordinates φ_h, φ_a and y_{gi}:

$$I_g\ddot{\varphi}_h i_{hg} + I_g\ddot{\varphi}_g = \\ = -M_{mp} - c_{bg}\varphi_g r_g + c_{ga}\left(\varphi_a r_a - \varphi_a r_g\right) r_g;$$

$$I_a\ddot{\varphi}_h i_{ha} + I_a\ddot{\varphi}_a = \\ = M_0\cos pt - c_a\varphi_a - n_w c_{ga}\left(\varphi_a r_a - \varphi_g r_g\right) r_a;$$

$$2m_g\dot{x}\dot{\varphi}_h + m_g x\ddot{\varphi}_h + m_g\ddot{y}_g - m_g y_g\dot{\varphi}_h^2 = \\ = -c_{hgy}y_g - c_{bg}\varphi_g \operatorname{tg}\alpha_{gb} r_g + c_{ga}\left(\varphi_a r_a - \varphi_g r_g\right)\operatorname{tg}\alpha_{ga}.$$

It is believed that the speed $\omega_h = \dot{\varphi}_h = \text{const}$. The rigidity of the axle of the satellite in the tangential direction considerably exceeds the rigidity of the axle in the radial direction, therefore, the displacement in the direction of the axis x $x_g = \dot{x}_g = \ddot{x}_g = 0$. Torque of engine $M_{\text{дв}} = = M_0\cos pt$, where p is the frequency of the driving force. Then the equations are transformed to the following form:

$$I_g\ddot{\varphi}_g = -M_{\text{тр}} - c_{bg}\varphi_g r_g + c_{ga}\left(\varphi_a r_a - \varphi_g r_g\right) r_g; \quad (1)$$

$$I_a\ddot{\varphi}_a = M_0\cos pt - c_a\varphi_a - n_w c_{ga}\left(\varphi_a r_a - \varphi_g r_g\right) r_a; \quad (2)$$

$$m_g\ddot{y}_g - m_g y_g\dot{\varphi}_h^2 = -c_{hgy}y_g - c_{bg}\varphi_g \operatorname{tg}\alpha_{gb} r_g + \\ + c_{ga}\left(\varphi_a r_a - \varphi_g r_g\right)\operatorname{tg}\alpha_{ga}. \quad (3)$$

Here m_g is the mass of satellite; I_g, I_a are the moments of inertia of the satellite and sun gearwheel; n_w the number of satellites; r_a radius of the sun gearwheel; i_{gh}, i_{ha} are gear ratios.

Under the accepted assumptions, the system of equations is separated – the first two equations can be solved independently of the third. A particular solution of the system of equations (1), (2) can be found in the form $\varphi_g = A_{\varphi g}\cos pt$; $\varphi_a = A_{\varphi a}\cos pt$, from which

$$A_{\varphi g} = \frac{M_0 b_{12}}{\left(b_{11} - a_{11}p^2\right)\left(b_{22} - a_{22}p^2\right) - b_{12}^2};$$
$$A_{\varphi a} = -\frac{M_0\left(b_{11} - a_{11}p^2\right)}{\left(b_{11} - a_{11}p^2\right)\left(b_{22} - a_{22}p^2\right) - b_{12}^2}, \quad (4)$$

where $a_{11} = n_w I_g$; $a_{22} = I_a$; $b_{11} = n_w\left(c_{bg} + c_{ga}\right) r_g^2$; $b_{12} = -n_w c_{ga} r_g r_a$; $b_{22} = c_a + n_w c_{ga} r_a^2$.

From formulas (4) it is possible to determine natural frequencies of vibrations of the sun gearwheel system with satellites by equating to zero the denominator

$$a_{11}a_{22}k^4 - \left(b_{11}a_{22} + a_{11}b_{22}\right)k^2 + \left(b_{11}b_{22} - b_{12}^2\right) = 0.$$

Solution of the biquadratic equation:

$$k_{1,2}^2 = \frac{\left(b_{11}a_{22} + a_{11}b_{22}\right)}{2a_{11}a_{22}} \pm \\ \pm \frac{\sqrt{(b_{11}a_{22} + a_{11}b_{22})^2 - 4a_{11}a_{22}\left(b_{11}b_{22} - b_{12}^2\right)}}{2a_{11}a_{22}}.$$

For further analysis, we will use the relationship between the mass and size characteristics of the planetary gear [21]:

$$r_g = \left(0{,}5i - 1\right) r_a; \quad m_g = k_g m_a\left(0{,}5i - 1\right)^2;$$

$$m_a = k_a\rho\pi r_a^2 b_w; \quad I_g = \frac{m_g r_g^2}{2} = \frac{k_g m_a\left(0{,}5i - 1\right)^4 r_a^2}{2};$$

$$I_a = \frac{m_a r_a^2}{2}.$$

Here ρ is the density; b_w is gearwheel rim width; k_g – coefficient of filling of the satellite; i is gear ratio of the mechamism.

After substitution of expressions for mass and elastic coefficients a_{ij}, b_{ij} taking into account the fact that the rigidity of the gearing $c_{bg} = c_{ga} = 0{,}075Eb_w$; is the torsional rigidity of the shaft $c_a = GI_p/l_a$, where $G = E/\left[2\left(1 + \mu\right)\right]$ is the modulus of elasticity in shear, μ is Poisson’s ratio; $I_p = \pi d_a^4/32 = \pi r_a^4/2$ is the polar moment of inertia of the gearwheel shaft section, l_a is the length of the sun gearwheel axle, natural frequencies are

$$k_{1,2}^2=\frac{0,15E}{k_a\pi\rho m^2 z_a^2}\left[\frac{1}{k_g(0,5i-1)^2}+\frac{\pi}{0,6l'_a b'_w(1+\mu)}+n_w\pm\sqrt{\left[\frac{1}{k_g(0,5i-1)^2}-\frac{\pi}{0,6l'_a b'_w(1+\mu)}\right]^2+0,25n_w^2}\right].$$

Here the radius of the gearwheel is written in the form $r_a=mz_a$, where m is the gear module; z_a – number of the gearwheel teeth; relative values $l'_a=l_a/r_a$; $b'_w=b_w/r_a$.

Knowing the frequencies k_1 and k_2, expressions for the amplitudes, we rewrite in the form

$$A_{\varphi g}=\frac{M_0 b_{12}}{a_{11}a_{22}\left(p^2-k_1^2\right)\left(p^2-k_2^2\right)};$$

$$A_{\varphi a}=-\frac{M_0(b_{11}-a_{11}p^2)}{a_{11}a_{22}\left(p^2-k_1^2\right)\left(p^2-k_2^2\right)},$$

or $A_{\varphi a}=\mu A_{\varphi g}$.

Shape quotient $\mu=-M_0\left(b_{11}-a_{11}p^2\right)/b_{12}$.

Forced vibrations of satellite with axle of the increased flexibility

By substituting the found values $\varphi_g=A_{\varphi g}\cos\left(pt+\gamma\right)$ and $\varphi_a==A_{\varphi a}\left(\cos pt+\gamma\right)$ into equation (3), a particular solution is found: $y_g=A\cos\left(pt+\gamma_1\right)$. Amplitude

$$A=-M_0\left[\left(c_{bg}\operatorname{tg}\alpha_{gb}+c_{ga}\operatorname{tg}\alpha_{ga}\right)r_g b_{12}+\right.$$
$$\left.+c_{ga}r_a\operatorname{tg}\alpha_{ga}\left(b_{11}-a_{11}p^2\right)\right]/$$
$$/m_g a_{11}a_{22}\left(p^2-k^2\right)\left(p^2-k_1^2\right)\left(p^2-k_2^2\right).$$

Here $k^2=\left(c_{hgy}/m_g\right)-\dot\varphi_h^2$ natural frequencies of vibrations of satellite axle radial direction which is determined primarily by the bending rigidity of the satellite axle $c_{hgy}=3EI_x/l^3$. The cross-section of the axle with increased flexibility in the first approximation can be considered rectangular (figure 4). Then the moment of inertia of the section $I_x=bh^3/12$, where $b\approx D$, $h=\chi D$, χ is a fraction of diameter; $I_x=\chi^3D^4/12$. The diameter and length of the axle are expressed in terms of the diameter of the satellite: $D=\bar D r_g$; $l_{оси}=\bar l_{оси}r_g$. Then $c_{hgy}=\frac{E\chi^3\bar D^4 r_g}{4\bar l_{оси}^3}$, and natural frequency of vibrations of a satellite in the radial direction

$$k=\sqrt{\frac{E\chi^3\bar D^4}{4\bar l_{оси}^3 k_g\pi\rho\left(0,5i-1\right)b'_w m^2 z_a^2}-\frac{\omega_a^2}{i}}.$$

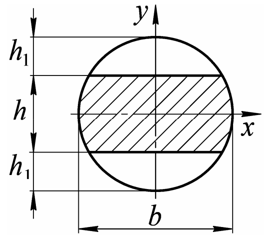

Fig. 4. Section of a flexible axles of the satellite

In figure 5 shows the dependences of the natural frequency of the satellite vibrations in the radial direction for different values of the input quantities. The graphs correspond to the following values of quantities: $\left[\sigma_F\right]=550$ MPa, $\bar D=1$, $\bar l_{оси}=2$, $b'_w=2$, $z_a=18$, $E=2.1\cdot10^{11}$ Pa; $\rho=7800$ kg/m^3; $k_g=0.8$, $\omega_a=100$ rad/s.

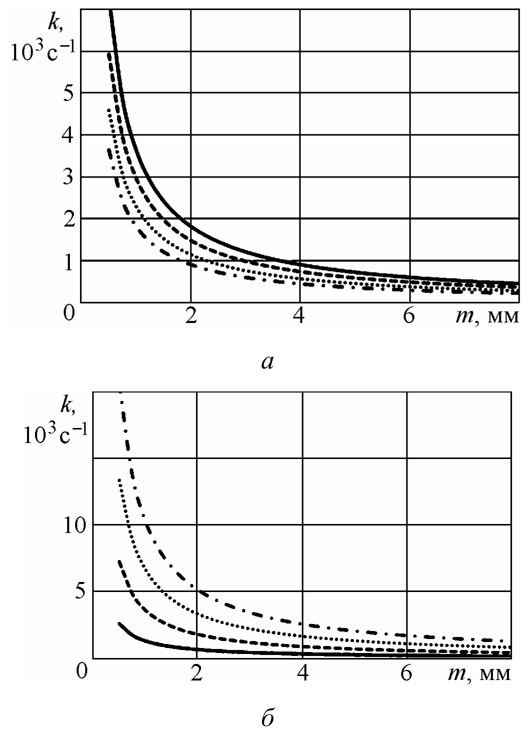

Fig. 5. Dependence of natural frequency of vibrations of the satellite in the radial direction from module m and from the gear ratio i: (*a*): ▬▬ - $i=4$, ▬ ▬ ▬ - $i=5$, ... - $i=7$, ▬.▬ - $i=10$; from the relative height of section of a flexible axles χ (*б*): ▬▬ - $\chi=0.25$, ▬ ▬ ▬ - $\chi=0.5$, ... - $\chi=0.75$, ▬.▬ - $\chi=1$

Sun gearwheel shaft torque $M=M_0\cos pt$, $M_0=M_h/i$. From the known relationship of the calculation of teeth for bending strength, the permissible moment $\left[M_h\right]$ on the planet carrier shaft

$\left[M_h\right]=\dfrac{2\left[\sigma_F\right]a_w n_w b_w m}{Y_F K_F}$, where $a_w=0,5ir_a$ is interaxial distance; Y_F – tooth form factor; K_F – unevenness of load in gear meshing factor [22]; $\left[\sigma_F\right]$ – allowable stress of bending.

After substitution of the gearing rigidity $c_{bg}=c_{ga}=0,075Eb_w$, of the expressions for the coefficients a_{ij}, b_{ij} and the torque M_0, amplitude

$$A=-\frac{Xp^2}{\left(k^2-p^2\right)\left(k_1^2-p^2\right)\left(k_2^2-p^2\right)},$$

where

$$X=\frac{0,15E\left[\sigma_F\right]\mathrm{tg}\,\alpha}{Y_F K_F k_g k_a \pi^2\rho^2}\frac{n_w}{m^3 z_a^4}\frac{1}{\left(0,5i-1\right)^2}.$$

Change of parameter X which characterizes the part of the satellite vibrations amplitude that does not depend on the natural and forced frequencies is shown in figure 6.

An amplitude-frequency characteristic of vibrations of the planetary gear is shown in figure 7. The graphs correspond to the following values of quantities: $\left[\sigma_F\right]=550$ MPa, $n_w=3$, $m=2$, $Y_F=3,75$, $K_F=1$, $l_a'=4$, $b_w'=2$, $z_a=18$, $E=2.1\cdot10^{11}$ Pa, $\rho=7800$ kg/m^3; $k_g=0.8$, $k_a=1$, $i=4$, $\omega_a=100$ rad/s.

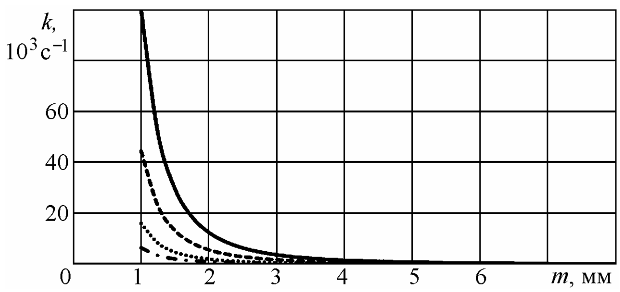

Fig. 6. Change of parameter X depending on module m and the gear ratio i: ▬▬ - i = 4, ▬ ▬ ▬ - i = 5, ... - i = 7, ▬ . ▬ - i = 10

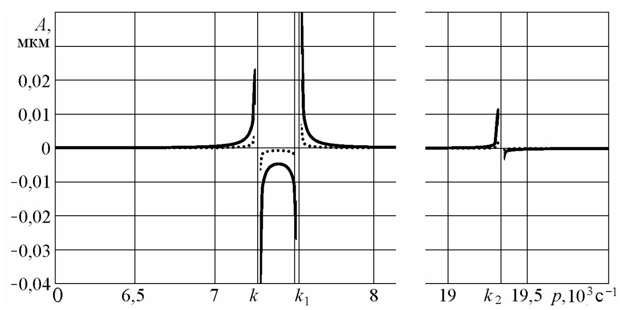

Fig. 7. An amplitude-frequency characteristic of vibrations of the satellite in the radial direction depending on the gear ratio i: ▬▬ - i = 4, ... - i = 7

Analysis of results

The features of the design scheme and the accepted assumptions made it possible to divide the analysis of forced vibrations of the planetary gear into the study of two independent systems: forced vibrations of the sun gearwheel and satellite due to the flexibility of the teeth and forced vibrations of the satellite in the radial direction due to the flexibility of the satellite axle.

The amplitude-frequency characteristic of the sun gearwheel and the satellite indicates that in the range of mass and size characteristics and gear ratios of planetary gears, which are most often encountered in practice, an increase in the amplitude of forced vibrations of this system is possible only with a significant value of the frequency of the driving force, and the first the frequency is in the region of 7000 rad/s, and the second is 19000 rad/s. In this regard, taking into account forced vibrations in planetary gears of this scheme can be relevant only in some high-speed gears.

The satellite axle flexibility significantly affects the natural frequency and amplitude of forced vibrations. So, the performed calculations and graphs indicate that a decrease in the relative height of the section along the χ satellite flexibile axle from 1 to 0.5 reduces the natural frequency of vibrations of the satellite in the radial direction by about three times, which increases the probability of resonance phenomena.

An increase in the gear ratio of the mechanism with a flexibile satellite axle also reduces the natural frequency of the satellite, but to a lesser extent. So, an increase in the gear ratio from 4 to 10 leads to a decrease in the natural frequency by about two times.

An increase in the engagement gear module has the strongest effect on the natural frequency of a satellite with a flexibile axle. A twofold increase in the engagement modulus leads to an approximately twofold decrease in the natural frequency.

The amplitude-frequency characteristic of a satellite with a flexibile axle indicates an increase in amplitudes near the resonant regions with a decrease in the gear ratio of the planetary gear. With an increase in the gear ratio to 7, a sharp decrease in amplitudes is observed near the resonance regions and in the region between the first and second frequencies.

Conclusions

An increase in the amplitude of forced vibrations of planetary gears with elements of increased flexibility is possible at a high frequency of the driving force. The probability of the appearance of resonance phenomena increases with a decrease in the height of the section of the flexible satellite axle in the radial direction but it is not typical for most planetary gears used in practice.

With an increase in the gear ratio the amplitude near the resonant regions decreases sharply. At the same time with an increase in the gear ratio the value of the natural frequency of the satellite with a flexibile axle decreases, which reduces the value of the resonant frequencies. With an increase in the gear module and consequently the diameters of the gearwheels, the risk of resonance phenomena increases.

The obtained dependences between the indicators of the strength of the engagement and the characteristics of the planetary gears make it possible to select rational values of the parameters of the planetary gear, ensuring its maximum load capacity with improved weight and dimensions.

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Вынужденные колебания планетарных передач с элементами повышенной податливости

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В статье продолжается исследование динамики планетарных передач с элементами повышенной податливости. Основой для расчетной схемы и исходных уравнений является полученная ранее система девяти дифференциальных уравнений динамики планетарного механизма. В обобщенных силах учтены момент двигателя, момент, обусловленный кручением упругого вала солнечной шестерни; упругие силы в зацеплении колес и в осях сателлитов.

Допускается, что скорость водила постоянна. Жесткость оси сателлита в тангенциальном направлении значительно превышает жесткость оси в радиальном направлении, поэтому перемещение в тангенциальном направлении отсутствует. Момент двигателя изменяется по гармоническому закону. В этом случае вынужденные колебания солнечной шестерни и сателлита можно рассматривать отдельно от вынужденных колебаний сателлита в радиальном направлении вследствие податливости оси сателлита. Построены амплитудно-частотные характеристики этих колебаний. Характеристики колебательных процессов выражены через массогабаритные, кинематические и прочностные параметры планетарной передачи с учетом податливости ее элементов.

Исследовано влияние относительной высоты сечения податливой оси, передаточного отношения механизма и модуля зацепления на амплитуду вынужденных колебаний и положение резонансной области. Сделаны выводы о пределах изменения указанных параметров и их влияния на конструкцию планетарных передач.

Ключевые слова: планетарная передача, податливость элементов, динамика, колебания.

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