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On Optimization of Functional Parameters of Color Dimensions and Size Relations for Combating Resonance Disorders During Restoration

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ABSTRACT: The calculation of the dependent for determining the optimal values of the vibrational coefficients of the transmission functional equalizing gears by non-circular gears in the fight against resonant vibrations in the gearbox has been obtained.

KEY WORDS: Resonant vibration in the gearbox, optimal values of the vibration coefficients of gears.

From the point of view of the theory of oscillations, gear transmissions are a system with distributed parameters and have a large number of natural oscillation frequencies. This leads to the fact that in almost all modes the operation of the gearing is accompanied by the occurrence of oscillations at resonant frequencies [1].

Constructive solutions in the fight against vibrations and noise are known enough. A wider range of vibration damping in a gear reducer is its design, in which, in addition to constant natural vibrations, the gearbox has at least one gear train that creates additional low-frequency vibrations according to a certain law for one complete revolution of the driving wheel of this gear [2].

The paper [3] considers the issues of synthesis and design of gears with non-circular wheels to combat resonant vibrations of gears. In this case, the anti-resonance gear is one of the gear stages. The transfer function of this transmission changes according to a certain, for example, skew-symmetric law:

$$i(\varphi_1) = \frac{r[2+\cos(j_1\varphi_1)] + B\sin(j_1\varphi_1)}{u\cdot r[2+\cos(j_1\varphi_1)] - B\sin(j_1\varphi_1)} \tag{1}$$

where r- is the average radius of the centroid of the driving wheel, j1 - is the number of maximum values of the radius of the centroid of the driving non-circular wheel, φ 1 - is the angle of rotation of the driving non-circular wheel, B - is the coefficient, a constant value for a given gear, is selected depending on the coefficient of uneven movement of the mechanism [4] , u - the gear ratio of the equalizing gear with non-circular wheels.

The average radius of the centroid can be determined from the dependence

$$r = \frac{a_u}{u+1} \tag{2}$$

where aw - is the center distance of the equalizing gear with non-circular wheels, is determined by the well-known method for gears with round gears.

Coefficient [3] is predetermined

$$B = \frac{u\cdot r\sqrt{3}}{\delta}\left(\sqrt{u^2+2u+\delta^2+1} - u - 1\right), \tag{3}$$

Where δ - is the coefficient of non-uniform movement of the mechanism [4].

As a result of the use of one gear in the gearbox with non-circular wheels, the driven shaft of the gearbox will have a variable angular velocity, which in some cases is undesirable. To obtain a constant angular velocity of the output shaft of the gearbox, it is necessary to install two gears in it with non-circular gears. In this case, the transfer function of the second compensating gear must be interconnected with the transfer function of the first compensating gear by the dependence

$$i_2(\varphi_2) = \frac{1}{u_p\cdot i_1(\varphi_1)} \tag{4}$$

where up - is the gear ratio of the two gears as a whole,

φ 2 - the angle of rotation of the driven non-circular wheel of the first stage of the equalizing gear, it is also the angle of rotation of the driving non-circular wheel of the second stage of the gearbox

$$\varphi_2 = \int_0^{\varphi_1} i_0(\varphi_1)d\varphi_1 = \int_0^{\varphi_1} \frac{r[2+\cos(j_1\varphi_1)]+B\sin(j_1\varphi_1)}{u\cdot r[2+\cos(j_1\varphi_1)]-B\sin(j_1\varphi_1)}d\varphi_1 = \frac{B\cdot r\cdot(1+u)}{j_1(B^2+u^2r^2)}$$
$$\times\left\{\frac{3\cdot u\cdot r\cdot\left(1+\mathrm{tg}^2(0{,}5j_1\varphi_1)\right)}{B\cdot r\cdot(1+u)} + \left(\mathrm{lr}\left(\frac{r^2u-B^2)}{3\cdot u\cdot r+u\cdot r\cdot\mathrm{tg}^2(0{,}5j_1\varphi_1)-2B\cdot\mathrm{tg}(0{,}5j_1\varphi_1)}\right)+\right.\right.$$
$$\left.\left.+\frac{4B}{3u^2r^2-B^2}\left[\mathrm{arctg}\left(\frac{r\cdot u\cdot\mathrm{tg}(0{,}5j_1\varphi_1)-B}{3u^2r^2-B^2}\right)+\mathrm{arctg}\left(\frac{B}{3u^2r^2-B^2}\right)\right]\right\}. \tag{5}$$

From the course of machine parts [5], it is known that the main generator of oscillations in the gearbox (drive) is its high-speed stage. Therefore, we will consider oscillations arising in the first stage of the equalizing gear from non-circular gears (1)

We write the frequency of natural bending vibrations of the drive shaft [5]

$$\omega_c = \sqrt{\frac{g}{y_{c\,cm}}} \quad (6)$$

where g - is the free fall acceleration, yst - is the static deflection of the shaft due to the force of gravity of the wheel.

Angular frequency of forced bending vibrations of the transmission drive shaft by non-circular wheels

$$\omega_3 = \sqrt{\frac{48gEla_w i_1(\varphi_1)}{T_1 l^3[1+i_1(\varphi_1)]\sqrt{\operatorname{tg}^2\alpha+\operatorname{tg}^2\beta}}}, \quad (7)$$

where E - is the modulus of elasticity of the shaft material, İ - is the moment of inertia of the shaft section at the wheel landing site, T1 - is the torque on the shaft of the driving non-circular wheel, I - is the distance between the shaft supports of the driving non-circular wheel, and is the angle of α - engagement, is the angle of $\beta -$ inclination of the teeth.

The total frequency of bending vibrations of the transmission drive shaft with non-circular gears

$$\omega_2 = \sqrt{\frac{48\cdot E\cdot l\cdot g\cdot m\cdot a_w\cdot i_1(\varphi_1)\cdot(\omega_c^2-\omega_3^2)}{l^3\{\omega_c^2 l\cdot a_w m\cdot i_1(\varphi_1)+T_1[1+i_1(\varphi_1)]\sqrt{\operatorname{tg}^2\alpha+\operatorname{tg}^2\beta}\}}}, \quad (8)$$

where m - is the mass of the drive shaft assembly, e is the mass eccentricity of the drive shaft.

Solving together equations (6), (7) and (8) with respect $i_1(\varphi_1)$ to we obtain

$$i_1(\varphi_1) = \frac{\omega_2^2 l^3 T_1\sqrt{\operatorname{tg}^2\alpha+g^2\beta}}{48gEl\cdot m\cdot a_w(\omega_c^2-\omega_3^2)-l^3\omega_2\left(m\cdot a_n e\omega_c^2+T_1\sqrt{\lg^2\alpha+g^2\beta}\right)} \quad (9)$$

Equation (9) describes the function of the transmission ratio of non-circular gears depending on a number of geometric and kinematic parameters of the drive shaft system mounted on two bearings with the drive wheel in the middle.

Substituting into the left side of equation (9) the given function gear ratio, you can determine the value of the coefficients included in this function. For example, for function (1) we have

$$\frac{r[2+\cos(j_1\varphi_1)]+B\sin(j_1\varphi_1)}{u_1 r[2+\cos(j_1\varphi_1)]-B\sin(j_1\varphi_1)} = \frac{\omega_\Sigma^2 l^3 T_1\sqrt{\operatorname{tg}^2\alpha+g^2\beta}}{48gEI\cdot m\cdot a_w(\omega_c^2-\omega_3^2)-l^3\omega_\Sigma\left(m\cdot a_w e\omega_c^2+T_1\sqrt{\operatorname{tg}^2\alpha+\operatorname{tg}^2\beta}\right)} \quad (10)$$

From here (10) we determine the refined coefficient

$$B = \frac{r[2+\cos(j_1\varphi_1)]}{\sin(j_1\varphi_1)}\left[\frac{(1-u_1)\omega_\Sigma^2 l^3 T_1\sqrt{\operatorname{tg}^2\alpha+\operatorname{tg}^2\beta}}{l^3\omega_\Sigma^2 e\omega_c^2-48gEl(\omega_c^2-\omega_2^2)}\right] \quad (11)$$

Putting in equations (9) and (10) the value

$$i_1(\varphi_1) = \frac{1}{u_1}, \quad (12)$$

Let us determine the values of the frequencies 1 of forced bending vibrations of the shaft and the total frequency of bending vibrations of the drive shaft of the transmission by round gears, which corresponds to transmissions with average centroid radii of non-circular wheels. In this case, it is necessary to put J1 = 1.

By setting the frequency ω_2 and ω_C values, it is easy to determine the value of the gear ratio of the equalizing gear with non-circular wheels, and then, using formula (11), determine the value of the coefficient. B of the skew-symmetric transfer function (1) of the antiresonant gear.

Thus, a calculated dependence has been obtained to determine the optimal value of the coefficient of the skew-symmetric transfer function, as well as the optimal transfer function of equalizing gears with non-circular wheels, taking into account the natural and forced vibration frequencies of the leading stage to combat resonant vibrations in the gearbox.

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