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LOADING OF THE MAIN ELEMENTS OF THE FREE WHELL BALL COUPLING

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Abstract. The article examines the design features and the principle of its operation of the overrunning ball clutch. A methodology for calculating transverse forces, bending moments and axial deformations of the main elements of the overrunning ball clutch is proposed. The given materials allow designers to rationally choose the materials for manufacturing and the main geometric parameters of the given overrunning ball clutch.

Keywords: coupling, freewheel coupling, stress, half-coupling, half-coupling connection.

Introduction

Couplings are responsible parts of machines and mechanisms that significantly affect the level and nature of the load on the kinematic chains of the mechanical drive. Also, they often perform the functions of protective devices of the responsible mechanical drives against overloads, as well as the functions of regulators of the speed of movement and the direction of energy transmission.

Problem Statement

Most often, couplings are used to connect shafts that have a common geometric axis of rotation or their geometric axes can be easily centered. Along with that, couplings are used to connect non-coaxial shafts, the geometric axes of which have radial, axial and angular displacement, for example, in subway trains.

For automatic adjustment of technological processes, free-wheel couplings are widely used, which allow to automatically connect and disconnect shafts without stopping the engine, as well as to transmit torque only in one direction.

Review of Modern Information Sources on the Subject of the Paper

The main known studies of free-running clutches relate to roller and ratchet mechanisms, which are described in the works of: V. S. Polyakova, M. M. Ivanova, S. G. Nagornyaka, O. A. Rakhovsky [8]. A number of designs of free-running ball couplings were developed, which are protected by patents for inventions and utility models at Lviv Polytechnic National University. These designs were studied in the works of V. V. Malashchenko, I. E. Kravts, V. V. Malashchenko, O. I. Sorokivskyi, and A. O. Borys. and others [1–5, 9–11].

Objectives and Problems of Research

The purpose of the work is to develop a methodology for calculating the strength of new free-running ball couplings that will allow rational selection of the main geometric parameters of half-couplings during their design.

Main Material Presentation

The main details of the design of the overrunning ball clutch are shown in fig. 1. The coupling consists of a leading semi-coupling 1, on the end of which grooves 4 are made with an offset from the geometric axis of the coupling to the periphery, and a driven semi-coupling 2. An annular groove 5 and tangential grooves 6 are made on the end of the driven half-coupling. Grooves 6 tangentially depart from the annular groove 5. Balls 3 are placed in the grooves of the driving half-coupling. The number of grooves corresponds to the number of balls and depends on the geometric parameters of the coupling and its load capacity. The depth of the half-coupling grooves also agrees with the diameter of the ball.

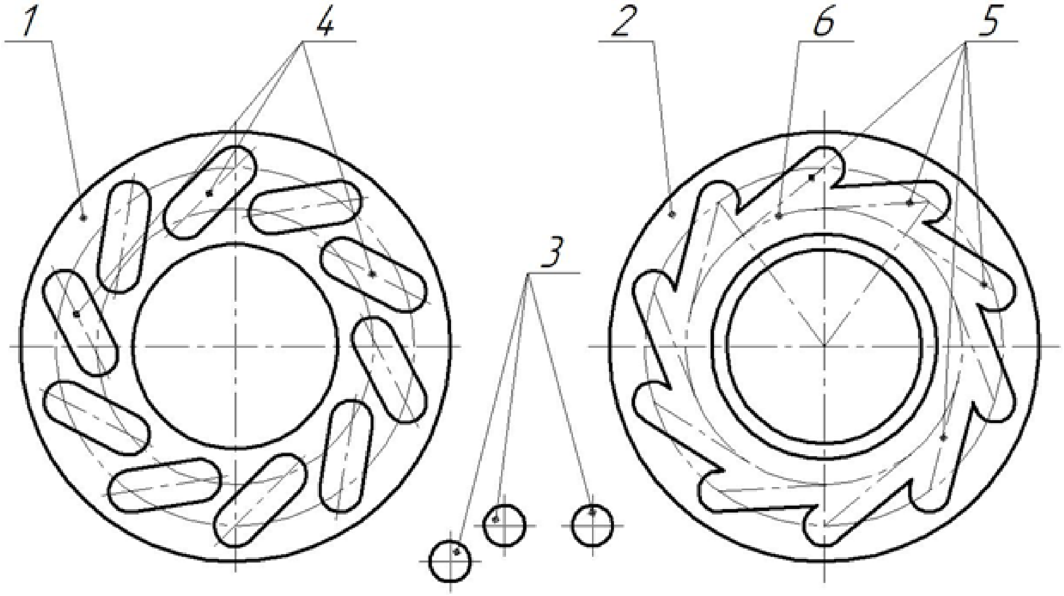

Fig. 1. The design of the overrunning ball clutch

The principle of operation of the free-running ball coupling is as follows. During the clockwise rotation of the driving half-coupling 1 (Fig. 1), the balls 3 under the action of centrifugal force are pushed into the nearest tangent groove 5 of the driving half-coupling 2. Then, under the pressure of the side surfaces of these grooves of the half-couplings 1 and 2, balls 3 reach their peripheral ends and transfer the load to the driven half-coupling 2, which starts to rotate. The mechanism will work in the working mode and transmit torque from the engine to the working body of the machine.

During the rotation of the driving half-coupling in the opposite direction, the balls 3 are pushed out by the side surfaces of the grooves 4 and 5 of the half-couplings 1 and 2 into the annular groove 6. Sliding along the annular groove 6, the balls 3 break the connection and put the coupling into idle mode. Automatic disconnection of the shafts also occurs in the case when, for technical reasons or due to the technological process, the driven half-coupling 2 will have a greater angular speed than the leading half-clutch 1, i.e. power transmission in the reverse direction is impossible.

Preliminary analysis revealed that the weakest point of ball couplings is the shape of the peripheral ends of the half-coupling grooves, which transmits torque from the driving to the driven link. The magnitude of the stresses and their type depend on the nature of the contact of the balls with the peripheral surfaces of the grooves of the half-coupling. Therefore, the shape of the grooves, especially their peripheral ends, significantly affects the distribution of the load of the contacting elements touching on different surfaces or at points. The maximum contact surface area can be obtained with spherical ends of the grooves. In this case, the radius of the sphere of the groove should be close to the radius of the ball. Then, taking into account the size of the gap between the half-couplings, the contact surface will be almost a full half of the surface of the ball. The calculation scheme of the forces acting on the ball in engagement is shown in Fig. 2.

Consider the case when there are three or more balls in the engagement of the clutch. For the calculation, it can be assumed that the axial force F_y, acting from the side of the ball on the driven half-coupling, is distributed symmetrically around a circle with a radius R (Fig. 3).

To calculate the free-running ball coupling, we will use the classical well-known theory. This theory makes it possible to calculate the strength of half-couplings as circular plates of constant thickness h with a central hole, which are symmetrically loaded with an axial force [6, 7].

It is known that in this case, the transverse force Q and bending moments M_r and M_Q act in the sections of the half-coupling element bounded by the angle $d\varphi_0$. The normal stresses σ_r and σ_Q vary with the thickness of the half-coupling as well as linearly. The maximum values of normal stresses near the surfaces of the half-coupling are determined by the following formulas [6, 7]:

$$\sigma_r = \pm \frac{6M_r}{\square^2};\ \sigma_\theta = \frac{6M_\theta}{\square^2}, \tag{1}$$

where M_r, M_θ – are bending moments related to the unit length of the half-coupling element (plus refers to the lower side of the half-coupling).

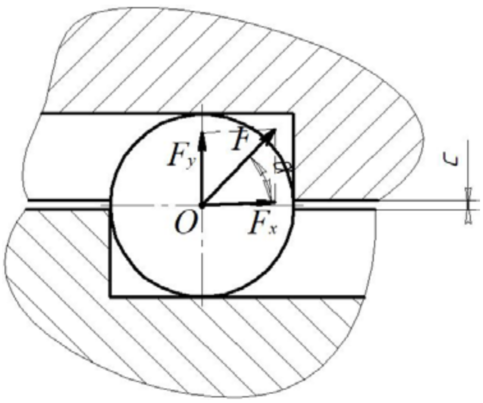

Fig. 2. Forces acting on the ball

To solve the given problem, we first determine the initial parameters from the boundary conditions at the edges of the semi-coupling. Since the half-couplings are clamped along the inner radius r, we get:

$$w(r) = M_j = 0;\ F_i = F_x;\ M_r(r_1) = 0, \tag{2}$$

where F_i, M_r – forces and moments with which the semi-coupling is loaded; $w(r)$ – deflection of the half coupling (up).

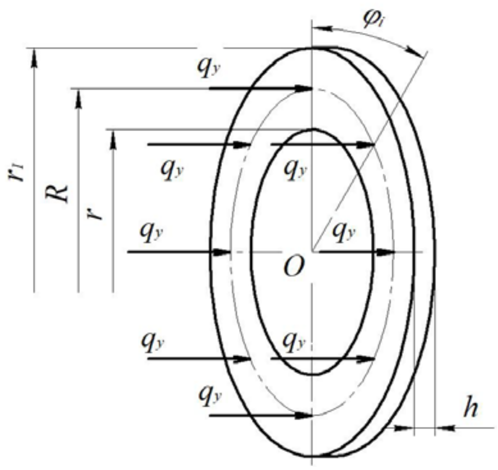

Fig. 3. Forces acting on the ball in the engaged state

For this case, using the general equations for the calculation of round plates [6], [7], we will obtain a solution in the form of the following system:

$$\left.\begin{aligned} &F_i = F(r) + \delta F_y; \\ &M_r = \phi_{21} F(r) + \phi_{22} M(r) + \delta \phi_{21}(R) F_y; \\ &M_\theta = \phi_{31} F(r) + \phi_{32} M(r) + \delta \phi_{31}(R) F_y; \\ &w = w(r) + \phi_{51} F(r) + \phi_{52} M(r) + \delta \phi_{51}(R) F_y; \\ &F_y = 2\pi R q_y, \end{aligned}\right\} \tag{3}$$

where, the coefficient $\delta = 0$ when $r_i < R$ and $\delta = 1$ when $r_i \geq R$; q_y – half-coupling load distributed along the radius R; $\varphi_{ij}(r)$ – influence functions defined by the following expressions [6]:

$$\left.\begin{aligned} &\phi_{21} = \frac{1}{8\pi}\left\{2(1+v)\,ln\frac{r_i}{r} + (1-v)\left[1 - \left(\frac{r}{r_i}\right)^2\right]\right\}; \\ &\phi_{22} = \frac{1}{2}\left[1 + v + (1-v)\left(\frac{r}{r_i}\right)^2\right]; \\ &\phi_{31} = \frac{1}{8\pi}\left\{2(1+v)\,ln\frac{r_i}{r} - (1-v)\left[1 - \left(\frac{r}{r_i}\right)^2\right]\right\}; \\ &\phi_{32} = \frac{1}{2}\left[1 + v - (1-v)\left(\frac{r}{r_i}\right)^2\right]; \\ &\phi_{51} = \frac{r^2}{8\pi D}\left\{\left[1 + \left(\frac{r}{r_i}\right)^2\right] ln\frac{r_i}{r} + 1 - \left(\frac{r_i}{r}\right)^2\right\}; \\ &\phi_{52} = \frac{r^2}{4D}\left[\left(\frac{r}{r_i}\right)^2 - 1 - 2\,ln\frac{r_i}{r}\right], \end{aligned}\right\} \tag{4}$$

where $D = \frac{Eh^3}{12(1-v^2)}$ – the cylindrical stiffness during bending of the half-coupling; r_i – variable radius; v – Poisson's ratio.

Let's determine the initial parameters based on the boundary conditions at the edges of the semi-coupling:

– when $F(r_1) = 0$ we have $F(r_1) = F(r) + F_y = 0$; $F(r) = -F_y$;

– at $M_r(r_1) = 0$, we get:

$$\begin{aligned} M_r(r_1) &= \phi_{21}(r_1)F(r) + \phi_{22}(r_1)M_r(r) - \phi_{21}(R)F_y = \\ &= F_y[\phi_{21}(r_1) - \phi_{21}(R)] + \phi_{22}(r_1)M_r(r) = 0. \end{aligned} \tag{5}$$

Using the influence functions (4), we obtain from the last equation:

$$M_r(r) = F_y \frac{\left\{2(1+v)\left(ln\frac{R}{r} - ln\frac{r_1}{r}\right) + (1-v)\left[\left(\frac{r}{r_1}\right)^2 - \left(\frac{r}{R}\right)^2\right]\right\}}{4\pi\left[1 + v + (1-v)\left(\frac{r}{r_1}\right)^2\right]}. \tag{6}$$

Taking into account the initial parameters, we reduce the system of equations (3) to the form:

$$\left.\begin{aligned} &F(r) = -F_y; \\ &F(r_1) = F(r) + F_y = 0; \\ &M_r = F_y[\phi_{21} - \delta\phi_{21}(R)] + \phi_{22}M_r(r); \\ &M_\theta = F_y[\phi_{31} - \delta\phi_{31}(R)] + \phi_{32}M_r(r); \\ &w = F_y[\phi_{51} - \delta\phi_{51}(R)] + \phi_{52}M_r(r). \end{aligned}\right\} \tag{7}$$

After substituting the influence functions (4) into the equation of the system (7), we obtain the formulas for determining the bending moments and the amount of deflection of the driven half-coupling:

$$M_r = \frac{F_y}{8\pi}\left\{2(1+v)\left(ln\frac{r_i}{r} - \delta\, ln\frac{R}{r}\right) + (1-v)\left[\delta\left(\frac{r}{R}\right)^2 - \left(\frac{r}{r_i}\right)^2\right]\right\} + \frac{M_r(r)}{2}\left[1+v+(1-v)\left(\frac{r}{r_i}\right)^2\right];$$

$$M_\theta = \frac{F_y}{8\pi}\left\{2(1+v)\left(ln\frac{r_i}{r} - \delta\, ln\frac{R}{r}\right) - (1-v)\left[\delta\left(\frac{r}{R}\right)^2 - \left(\frac{r}{r_i}\right)^2\right]\right\} + \frac{M_r(r)}{2}\left[1+v-(1-v)\left(\frac{r}{r_i}\right)^2\right]; \quad (8)$$

$$w = \frac{F_y r^2}{8\pi D}\left\langle\left\{\left[1+\left(\frac{r}{R}\right)^2\right] ln\frac{r_i}{r} + 1 - \left(\frac{r_i}{r}\right)^2\right\} - \delta\left\{\left[1+\left(\frac{R}{r}\right)^2\right] ln\frac{R}{r} + 1 - \left(\frac{R}{r}\right)^2\right\}\right\rangle + \frac{M_r(r)r^2}{4D}\left[\left(\frac{r}{r_i}\right)^2 - 1 - 2\, ln\frac{r_i}{r}\right].$$

The largest bending moments act at the point of attachment of the half-coupling, in the case $r_i = r$:

$$M_r(r) = F_y\frac{\left\{2(1+v)\left(ln\frac{R}{r} - ln\frac{r_1}{r}\right) + (1-v)\left[\left(\frac{r}{r_1}\right)^2 - \left(\frac{r}{R}\right)^2\right]\right\}}{4\pi\left[1+v+(1-v)\left(\frac{r}{r_1}\right)^2\right]}; \quad (9)$$

$$M_\theta(r) = \frac{F_y v}{4\pi}\frac{\left\{2(1+v)\left(ln\frac{R}{r} - ln\frac{r_1}{r}\right) + (1-v)\left[\left(\frac{r}{r_1}\right)^2 - \left(\frac{r}{R}\right)^2\right]\right\}}{\left[1+v+(1-v)\left(\frac{r}{r_1}\right)^2\right]} \quad (10)$$

The largest deflections occur near the outer edge of the half coupling, in the case $r_i = r_1$:

$$w(r_1) = \frac{F_y r^2}{8\pi D}\left\langle\left\{\left[1+\left(\frac{r_1}{r}\right)^2\right] ln\frac{r_1}{r} + 1 - \left(\frac{r_1}{r}\right)^2\right\} - \left\{\left[1+\left(\frac{R}{r}\right)^2\right] ln\frac{R}{r} + 1 - \left(\frac{R}{r}\right)^2\right\}\right\rangle + \frac{F_y r^2}{16\pi D}\frac{\left\{2(1+v)\left(ln\frac{R}{r} - ln\frac{r_1}{r}\right) + (1-v)\left[\left(\frac{r_1}{r}\right)^2 - \left(\frac{r}{R}\right)^2\right]\right\}}{\left[1+v+(1-v)\left(\frac{r}{r_1}\right)^2\right]}\left[\left(\frac{r_1}{r}\right)^2 - 1 - 2\, ln\frac{r_1}{r}\right]. \quad (11)$$

The resulting equations (9)–(11) are essential for solving the applied problem of determining the main characteristics of free-running ball couplings in the case of simultaneous engagement of several balls.

As an example, let's calculate the developed free-running ball coupling with the following geometric dimensions: the half-coupling is fixed along the radius r = 10 mm; Poisson's ratio for steel v= 0.3 (other dimensions are given above).

The cylindrical stiffness of the semi-coupling in this case is equal to

$$D = \frac{2.1\cdot10^{11}\left(1.5\cdot10^{-3}\right)^3}{12(1-0.3^2)} = 64.9 \text{ N·m}.$$

According to formulas (9) and (11), we determine the largest bending moments in the places of attachment of half-couplings:

$$M_r(r) = -26.34 \text{ (N·m)}; \quad M_\theta(r) = -7.9 \text{ (N·m)}.$$

The largest deflection near the outer edge of the half-coupling due to the action of the axial force according to formula (11) is equal to

$$w(r_1) = 6.993\cdot10^{-6} \text{ m}.$$

The resulting half-coupling deflection is so small that it can be neglected. The maximum values of normal stresses in the cross-section of the half-coupling are determined by the system of equations (1):

$$\sigma_r = \frac{6 \cdot 26.3358}{(1.5 \cdot 10^{-3})^2} = 70.23 \text{ MPa}; \ \sigma_\theta = \frac{6 \cdot 7.9}{(1.5 \cdot 10^{-3})^2} = 21.07 \text{ MPa}.$$

To ensure the strength of the free-wheel coupling, these stresses are compared with the permissible ones, and the most appropriate material for the manufacture of half-couplings is selected. The largest half-coupling deflections should not exceed 10 % of the ball diameter. In the opposite case, the thickness of the half-coupling should be increased, or the half-coupling should be fixed with an additional element in the axial direction.

During operation of a overrunning ball coupling in the most unfavorable case, the full load can be transmitted by the coupling with only one ball engaged between the coupling halves. If we assume that there is uniform contact between the ball and the surface of the half-coupling groove, then the uniform forces acting on the ball will be directed to the center of the area of the contact surface. This occurs in the case of the spherical shape of the peripheral ends of the grooves. Let's first decompose the uniform force F into two components: F_t and F_y (Fig. 4). The axial force F_y, defined above, is directed perpendicular to the end surfaces of the half-couplings, as a result of which transverse forces and bending moments occur in their cross-sections, which, under certain conditions, can lead to undesirable values of axial deformations and disruption of the reliable engagement of the ball with the grooves of the half-couplings. To determine the values of forces and moments, consider a semi-coupling as a ring of constant thickness loaded with an axial force F_y.

The concentrated axial force F_y is balanced by the distributed load q acting according to the law shown in Fig. 4, where R is the radius of the circle of the centers of the balls in the working condition; r is the inner radius of the annular surface of the driven half-coupling [6].

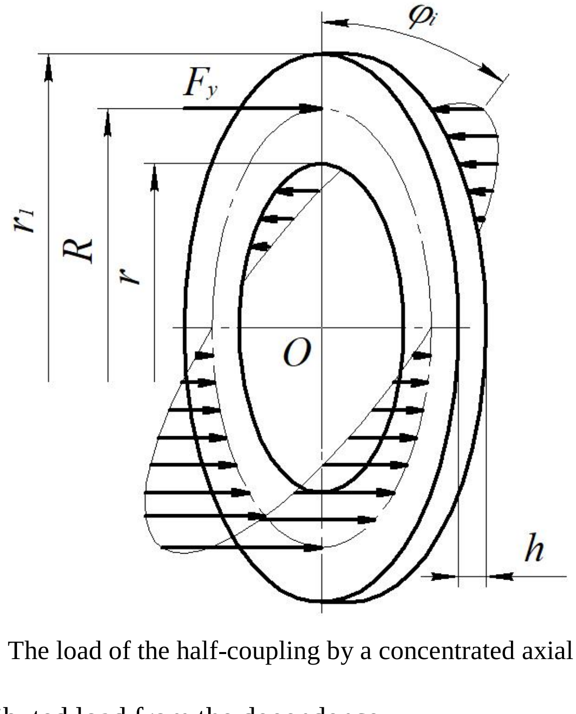

Fig. 4. The load of the half-coupling by a concentrated axial force

We determine the distributed load from the dependence

$$q = \frac{F_y}{\pi r}(0.5 + \cos \phi_i), \tag{12}$$

where φ_i – the angular coordinate of the current section of the ring.

Note that if the semi-coupling is reduced to an equivalent ring (Fig. 4), then according to the classical theory of the strength of the ring, we have reason to assert that transverse force and bending moments act in its cross-sections.

Using the general equations for the calculation of rings [6], we will obtain expressions for determining: transverse force, bending moments and displacement in the form of the following system of equations:

$$\left.\begin{aligned}
&Q=\frac{F_y}{\pi}\left(\frac{\pi-\phi_i}{2}-\sin\phi_i\right);\\
&M_r=F_yR\left(-\frac{1}{2\pi}\right)[(\pi-\phi_i)\sin\phi_i-0.5\cos\phi_i-1];\\
&M_\theta=F_yR\frac{1}{2\pi}\left[(\pi-\phi_i)\left(1-\frac{r}{R}\cos\phi_i\right)-\left(2-0.5\frac{r}{R}\right)\sin\phi_i\right];\\
&u=\frac{F_yR^3}{EJ_y}\left(-\frac{1}{2\pi}\right)\frac{r}{R}\left\{(1+\lambda)\left[1-0{,}5\left(\frac{\pi^2}{3}-\frac{3}{4}-\pi\phi_i+0.5\phi_i\right)\cos\phi_i-\right.\right.\\
&\left.-\frac{1}{2}(\pi-\phi_i\sin\phi_i)\right]+\lambda\left[\frac{r}{R}+\frac{\pi^2}{6}-\frac{R}{r}\frac{(\pi-\phi_i^2)}{2}+0{,}5\left(2\frac{R}{r}+3\frac{r}{R}\right)\cos\phi_i-\right.\\
&\left.\left.-\frac{R}{r}(\pi-\phi_i)\sin\phi_i\right]\right\},
\end{aligned}\right\}\qquad(13)$$

where $\lambda=\frac{EJ_y}{GJ_\kappa}$ – the ratio of stiffnesses and moments of inertia during bending and twisting; Q – the transverse force in the section; M_r i M_θ – bending moments in sections; u – cross-sectional displacement of the half-coupling; E, G – modulus of elasticity and shear modulus of the material; J_y, J_κ – moments of inertia of the section relative to the axis and torsion.

The moments of inertia of sections are determined by the formula

$$J_y=J_\kappa=\frac{(r_1-r)\square^3}{12},\qquad(14)$$

where h – the thickness of the half coupling; r_1, r – its outer and inner radii.

The largest displacement (deformation) of the half-coupling in relation to the position of the ball will occur at the point of application of the axial force F_y, at $\varphi_i=0$. As a result, the final expressions for determining the transverse force, bending moments and displacement will be written in a simplified form:

$$\left.\begin{aligned}
&Q=\frac{F_y}{2};\\
&M_r=\frac{3}{4\pi}F_yR;\\
&M_\theta=F_yR\frac{1}{2}\left(1-\frac{r}{R_i}\right);\\
&u=\frac{F_yR^3}{EJ_y}\left(-\frac{1}{2\pi}\right)\frac{r}{R}\left\{(1+\lambda)\left[1-0.5\left(\frac{\pi^2}{3}-\frac{3}{4}\right)+\right.\right.\\
&\left.\left.+\lambda\left[\frac{r}{R}+\frac{\pi^2}{6}-\frac{R}{r}\frac{\pi}{2}+0.5\left(2\frac{R}{r}+3\frac{r}{R}\right)\right]\right]\right\}.
\end{aligned}\right\}\qquad(15)$$

The system of equations (15) makes it possible to determine the largest displacements and forces occurring at the point of application of the axial force. The value of these parameters significantly affects the nature of the ball and half-coupling engagement. With their increase, the half-couplings are unzipped, which can lead to a violation of the process of connecting the shafts. The normal stresses σ_r and σ_θ vary along the thickness of the half-coupling according to a linear law with maximum values near its surface:

$$\sigma_r = \frac{6M_r}{(r_1 - r)\square^2}, \tag{16}$$

$$\sigma_\theta = \frac{6M_\theta}{(r_1 - r)\square^2} \tag{17}$$

It is advisable to use equations (15)–(17) to solve problems with determining the main characteristics of ball couplings for the most unfavorable case, when only one ball is engaged.

We will adapt the developed methodology to the quantitative analysis of the bicycle ball coupling with the following geometric parameters: half-coupling thickness h = 1.5 mm; its outer diameter d = 49 mm; the load with the maximum axial force F_y = 800 N occurs along the radius R = 20 mm; the inner radius of the semi-coupling r = 6.5 mm; modulus of elasticity for steel $E = 2.1\times10^5$ MPa; shear modulus for steel $G = 8\times10^4$ MPa.

According to the first three equations of system (15), we determine the largest bending moments and transverse force in the cross-sections of half-couplings:

$$Q = 800\cdot 0.5 = 400 \text{ N};$$

$$M_r = 800 \cdot 20 \cdot 10^3 \cdot 3 / 4 / 3{,}14 = 3.82 \text{ N·m};$$

$$M_\theta = 800\cdot 20\cdot 10^{-3}\cdot 0.5\left(1-\frac{6.5}{20}\right) = 5.4 \text{ N·m}.$$

The moments of inertia of the half-coupling sections are determined by the well-known equation (14)

$$J_y = J_\kappa = \frac{(24.5-6.5)10^{-3}\left(1.5\cdot 10^{-3}\right)^3}{12} = 5.0625\cdot 10^{-12} \text{ m}^4.$$

Ratio of stiffnesses and moments of inertia during bending and twisting

$$\lambda = \frac{2.1\cdot 10^5}{8\cdot 10^4} = 2.625.$$

Displacement of the ring section according to the last formula of the system (15)

$$u = \frac{800\left(20\cdot 10^{-3}\right)^{3}{}^{3}}{2.1\cdot 5.0625\cdot 10^{-1}}\left(-\frac{1}{2\cdot 3.14}\right)\frac{6.5}{20}\left\{(1+2.625)\left[1-0.5\left(\frac{3.14^2}{3}-\frac{3}{4}\right)+\right.\right.$$

$$\left.\left.+2{,}625\left[\frac{6{,}5}{20}+\frac{3{,}14^2}{6}-\frac{20}{6{,}5}\frac{3{,}14}{2}+0{,}5\left(2\frac{20}{6{,}5}+3\frac{6{,}5}{20}\right)\right]\right]\right\} = = -2.684\cdot 10^{-4} \text{ m}.$$

The minus sign means that the displacement of the half-coupling occurs down the Y-axis. According to equations (16) and (17), the maximum values of normal stresses in the cross-section of the half-coupling are:

$$\sigma_r = \frac{6\cdot 3.82}{(24.5-6.5)10^{-3}\left(1.5\cdot 10^{-3}\right)^2} = 566 \text{ MPa};$$

$$\sigma_\theta = \frac{6\cdot 5.4}{(24.5-6.5)10^{-3}\left(1.5\cdot 10^{-3}\right)^2} = 800 \text{ MPa}.$$

The results confirm the practical value of the proposed methodology for calculating the strength of half-couplings of a new overrunning ball coupling.

The case when there is only one ball in the engagement can be considered as limiting, and the values of stresses and deflections for it can be determined only to identify their maximum probable values. In reality, in practice, more often than not, several balls will be engaged. Therefore, the working stresses in the cross-sections and the deflections of the half-couplings when they are symmetrically loaded will be much smaller. It should be noted that the real stresses and deflections of half-couplings will always be in the interval between their minimum and maximum values, approaching the lower limit with an increase in the number of balls that are simultaneously engaged.

Conclusions

The article proposes a methodology for calculating the strength of one of the structures of overrunning ball couplings, which allows the selection of the main elements of these couplings for use in drives of machines and mechanisms. According to this method, it is possible to calculate working transverse forces, bending moments, and axial deformations of the main elements of ball couplings, as well as to evaluate the strength of its elements.

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