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DEVELOPMENT OF NEW BALL SAFETY COUPLINGS AND JUSTIFICATION OF THE BASIC TECHNICAL PARAMETERS WHICH ENSURE THE RELIABILITY OF THE TECHNICAL WORK

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The modern technology faces the task of improving the operational reliability of the working bodies and drives of machines. One of the ways to solve this problem is to develop and apply high precision and low dynamic couplings.

The dynamic loads that occur when the existing couplings are triggered result in considerable shock loads, which results in rapid wear of the coupling surfaces and shortening of service life.

In view of this, the development of new protective coupling designs that reduce shock loads and increase the reliability and durability of machinery mechanisms is relevant.

Keywords: coupling, safety coupling, reliability, shock loads, durability, performance.

1. Introduction

Important contribution to the formation of the scientific bases of theoretical calculations of the parameters of the safety couplings, methods of experimental studies, as well as their synthesis and design have been made by well-known scientists: Reshetov D.M., Polyakov V.S., Gevko B.M., Liubin M.V., Nagornyak G.S., Gevko R.B. and other.

Synthesis of structural and kinematic circuits of ball, cam and planetary safety couplings, methods of their calculation in combination with the nature of the change of the moment of resistance at the operating device are given in the Nagornyak G.S. and Lutivas I.V. handbook [1].

The question of profiling the wells under the balls to ensure a constant actuation of the safety couplings is solved in the work of Nagornyak G.S. [2].

The investigations on safety couplings were made in [3, 6, 7].

The method of calculating the safety couplings with the determination of the relationship between the power (half couplings` force of compression, torque)

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and the design parameters of ball and combined safety couplings is set out in the works of Gevko B.M. and Gevko R.B [4, 5].

The purpose of the work is to reduce the level of shock loads on the drives of machines by substantiating and selecting rational parameters of new designs of low-dynamic ball safety couplings.

Thus, it can be concluded from the analysis that the overwhelming majority of the designs of safety couplings and works related to the determination of their rational and optimal design, kinematic and dynamic parameters are aimed at identifying local problems and do not take into account the basic set of requirements, which should comply with safety couplings, namely: precision actuation; reliability and durability in operation (minimal wear on the engagement elements), as well as ease use.

2. Equipment and methods

Experimental research program

A program of experimental research was developed in accordance with the basic tasks of research, as well as on the basis of theoretical calculations. Research program provides:

- development of technical documentation and production of prototype models of ball safety couplings with possible adjustment of the parameters of structural elements that significantly affect the functioning of the coupling;
- development and production of an experimental stand for conducting research on determining the parameters of the actuation of the safety couplings;
- conducting of static and dynamic researches with determination of influence of structural and kinematic parameters of safety couplings on character of their actuation and change of magnitude of torque;
- conducting experimental studies using mathematical planning of a multifactorial experiment;
- comparative analysis of the results of theoretical and experimental studies on the value of torque when actuating couplings and the angle of relative rotation of the couplings after their opening to re-contact;
- carrying out of production tests of the developed designs of ball safety couplings in real conditions of operation.

3. Results and discussion

The basic directions of improvement of designs of ball safety couplings are reduction of shock loads at relative rotation of leading and slave links, increase of accuracy and stability of work of couplings at their high durability.

Reduction of shock loads during the towing of half couplings is provided, usually, by the use of damping elements, the structural use of which is quite wide. The typical options of their implementation are considered below.

In Fig. 1. shows a ball-joint coupling [8] comprising a hub 1, a leading 2 and a driven 4 half-coupling, interconnected by balls 3. A driven half-coupling with a disk 7 is compressed by a spring 6. The spring is covered by a shock-absorbing sleeve 5, which is made of elastic material in the form of two flanges connected with each other in outer diameter by a corrugated surface.

In case of overload and relative rotation with axial displacement of the half couplings, the shock-absorbing sleeve counteracts the instantaneous compression of the spring, thus damping the axial and circular shock. This effect is achieved by using a shock absorber sleeve made of polymers characterized by elastic-viscous deformation.

The main disadvantage of such a clutch is the short-lived cushioning sleeve due to changes in the characteristics of polyamides when they are subjected to alternating cyclic loads.

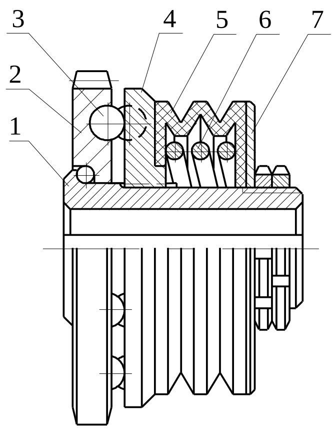

Fig. 1. Ball bearing coupling with corrugated shock absorber sleeve

The use of rubber damping disks 1 and 3 on both sides of the movable half coupling 2 (Fig. 2) provides a reduction of the inertial displacements of the movable half coupling towards the compression of the spring and also promotes the increase of the torque transmitting the coupling [8]. An internal damping disc provides the axial impact damping when re-engaging the half couplings. The main disadvantage of such a clutch is the mismatch between the time of deformation of the metal springs and the rubber damping disks. Therefore, at high torques, the inner rubber disk does not have time to return to its initial stress state after deformation, while the movable half-coupling will reciprocate in the axial direction

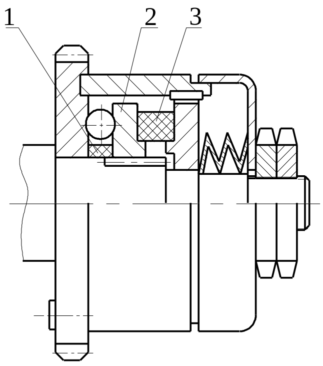

Fig. 2. Ball bearing coupling with double-sided arrangement of damping disks

On the basis of the conducted power analysis, an analytical dependence is derived to determine the torque that transmits the safety clutch from its design parameters and relative to the position of the half-coupling [2].

$$T_M = \frac{RC\left[\delta_0 \sqrt{r^2 - (\sqrt{h_l \cdot (2r - h_l)} - R_\varphi)^2} - r + h_l\right]}{tg\left\{\arcsin\left(\frac{\sqrt{r^2 - (\sqrt{h_l (2r - h_l)} - R_\varphi)^2}}{r}\right) - \rho\right\}}, \qquad (1)$$

where R – the radius location of the balls relative to the axis of the coupling;

C – the rigidity of the spring;

r – the radius of the ball;

δ_0 – pre-tensioned spring;

h_l – the value of the recess of the ball;

φ – displacement angle of the half-coupling in the circular direction;

ρ – the friction angle movable joints of half-coupling.

Power analysis of the elastic element (annular spring)

The system (Fig. 3) is once statically indeterminate.

The canonical equations of the force method will be:

$$\delta_{11} X_1 + \Delta_{1F} = 0, \qquad (2)$$

where X_1 – unknown effort; δ_{11}, Δ_{1F} – canonical equation coefficients.

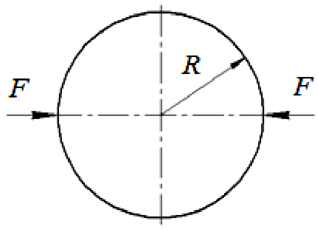

Fig. 3. The scheme of loading of the elastic element by two forces

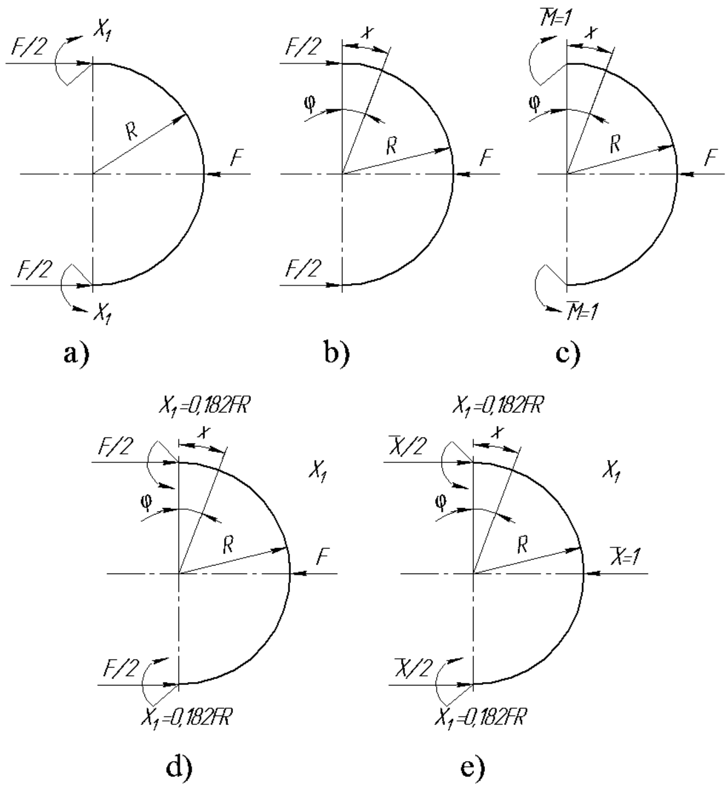

Fig. 4. Calculation schemes: a) equivalent system; b) forces diagrams for determining the load moment; c) schemes of forces for determination of single moment; d) schemes of total bending moment; e) forces diagrams for determining the total single moment

Load bending moment equation (Fig. 4, b):

$$M_{ZF} = \frac{F}{2}(R - R \cdot \cos\varphi) = \frac{FR}{2}(1 - \cos\varphi)\,, \quad (3)$$

where F – applied force.

Single bending moment equation (Fig. 4, c):

$$\overline{M}_Z = 1. \quad (4)$$

Then the coefficients of the canonical equation of the method of forces are found by means of the Mora integral:

$$\delta_{11} = \int_0^{\pi/2} \frac{\overline{M}_Z \cdot \overline{M}_Z}{EI_Z} dx = \int_0^{\pi/2} \frac{1 \cdot 1}{EI_Z} R\, dx = \frac{R}{EI_Z} \cdot \varphi\Big|_0^{\pi/2} = \frac{R \cdot \pi}{2EI_Z}, \tag{5}$$

where EI_Z – applied force.

$$\Delta_{1F} = \int_0^{\pi/2} \frac{M_{ZF} \cdot \overline{M}_Z}{EI_Z} dx = \int_0^{\pi/2} \frac{\frac{FR}{2}(1-\cos\varphi) \cdot 1}{EI_Z} \cdot R\, d\varphi = \frac{FR^2}{2EI_Z}\left[\frac{\pi}{2} - 1\right]. \tag{6}$$

Substituting the found coefficients into the canonical equations, we find an unknown effort:

$$x_1 = -\frac{\Delta_{1F}}{\delta_{11}} = -0{,}182FR. \tag{7}$$

Bending moment equation for the loaded system (Fig. 4, d):

$$M_{ZF} = -x_1 + \frac{F}{2}(R - R \cdot \cos\varphi) = -0{,}182FR + \frac{FR}{2}(1-\cos\varphi). \tag{8}$$

Bending moment equation for a single system (Fig. 4, e):

$$\overline{M}_Z = -x_1 + \frac{\overline{x}}{2}(R - R \cdot \cos\varphi) = -0{,}182R + \frac{R}{2}(1-\cos\varphi). \tag{9}$$

Then the deformation at the site of application of single force will be equal to:

$$\Delta = \int_0^{\pi/2} \frac{M_{ZF} \cdot \overline{M}_Z}{EI_Z} dx = \frac{R}{EI_Z}\int_0^{\pi/2} (-0{,}182FR + \frac{FR}{2}(1-\cos\varphi) \cdot (-0{,}182R + \frac{R}{2} \cdot$$
$$\cdot (1-\cos\varphi))d\varphi = \frac{R^3}{EI_Z}\int_0^{\pi/2} (-0{,}182F + \frac{F}{2}(1-\cos\varphi)) \cdot (-0{,}182 + \frac{1-\cos\varphi}{2})d\varphi, \tag{10}$$

where I_z – the moment of inertia of the cross-section relative to the z axis (Fig. 5).

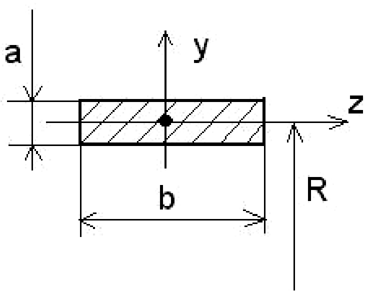

Fig. 5. Cross section of the ring

Where the force applied to the ring will be determined by:

$$F = \frac{\Delta \cdot E \cdot I_Z}{R^3 \cdot \int_0^{\pi/2} (-0{,}182 + \frac{1-cos\varphi}{2}) \cdot (-0{,}182 + \frac{1-cos\varphi}{2}) d\varphi} . \quad (11)$$

In Fig. 6 presents a ball clutch with bearing radial engagement elements, consisting of a drive 1 and a driven 2 half couplings, balls 3, which are located in the openings of the drive coupling, and pressed by a spring 4 to the driven half coupling. The spring is supported by radial displacement by pins 5, and axial displacement by a corkscrew 6.

The safety coupling works as follows. Torque is transmitted from the drive half clutch 2 through the bearing balls 6, which are pressed into the holes by a spring 4 on the driven half clutch 1.

The driven half coupling is slowed down in case of overload. The lead half coupling and the spherical balls continue to rotate, which causes the bearing balls to come out of engagement with the holes of the driven half coupling. The bearing balls are moved in radial grooves towards the axis, which leads to deformation of the springs. At this point, the balls are rolled over the driven half-clutch.

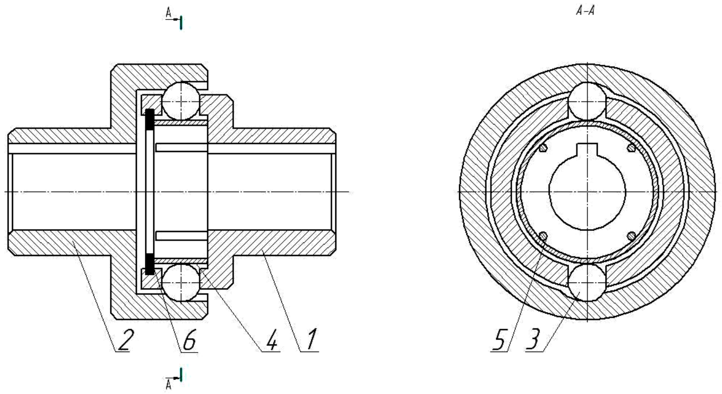

Fig. 6. Ball bearing coupling with bearing radial engagement

The balls approach the next arc grooves with further relative rotation. The arc groove on the side of the ball is made less inclined for a smoother entry into the hole. The ball exits the hole again under the influence of too much torque. By reducing the amount of torque, under the action of the springs 4, the balls 3 resume contact with the grooves of the inner surface of the housing of the driven half coupling, ie restores the functioning of the device. Compensation of the shaft joint inconsistency is due to the deformation of the spring 4. The proposed safety ball joint is characterized, in addition to the function of protecting the machine components from overload, by the ability to compensate for the displacement of the connected shaft.

In Fig. 7 depicts an embodiment of a ball bearing coupling with radial engagement elements. In this case, the clutch differs from the previous one in that it is mounted not on two shafts, but on the shaft and chain gear, the holes for the balls are made on the outer surface. The presented safety coupling (Fig. 7) consists of an asterisk 2 in which the holes for the balls 6, which are pressed by a spring 5 into the holes, which are made on the outside of the hub 3.

The ring 4 is installed from the axial displacement of the spring, and from the axial displacement of the sprocket there are corkscrew rings 7. This safety ball coupling works similar to the previous coupling structure.

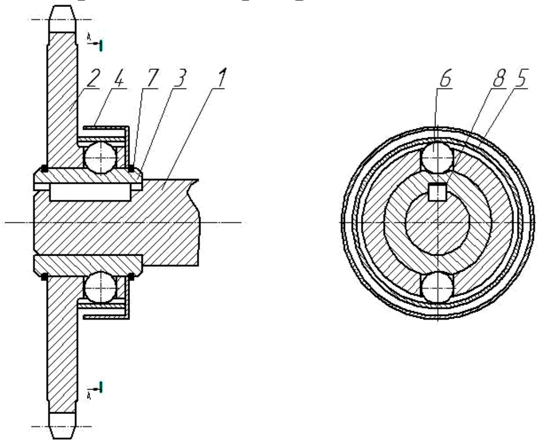

Fig. 7. Ball bearing coupling with bearing radial engagement elements

Results of static studies of safety couplings

The purpose of static experimental studies of designed and manufactured ball safety couplings was to determine the nature of their operation, to determine the maximum torque at two stages of operation of the couplings and to perform a comparative analysis between the results of theoretical and experimental studies.

Pre-couplings were mounted on the P5 bursting machine, for which special fixing ends were made, which were connected on one side with half couplings and on the other with grips of the bursting machine. The general view of this stand, on which the couplings are installed, is shown in fig. 8.

With the help of the loading mechanism the moment of resistance to the coupling was increased, the value of which was fixed on the power scale. In this case, the relative displacement of the half couplings was determined using an angle scale.

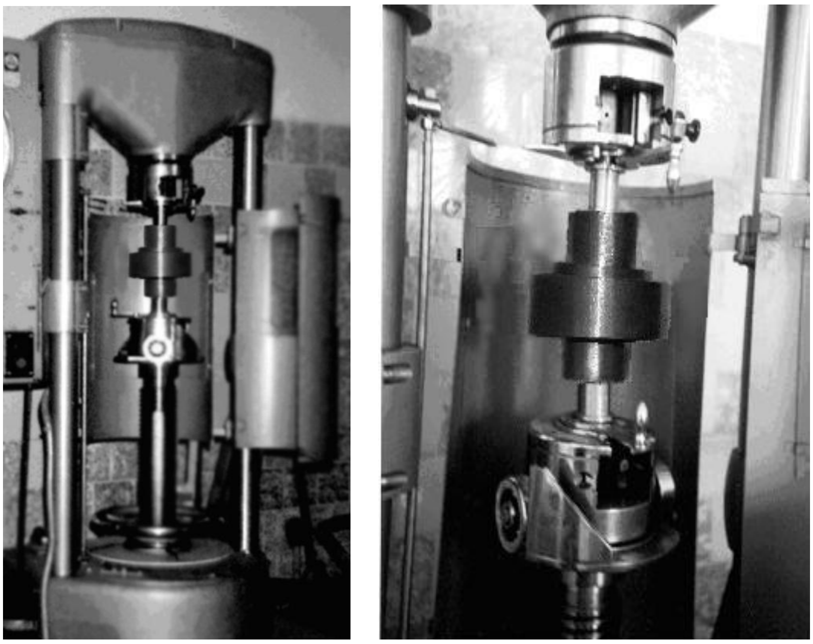

Fig. 8. General view of stand P5 with clutch

The results of studies showed, that the angular rotation of the half-coupling occurred at maximum torque, which dropped sharply when the angular displacement of the half-coupling occurred. This fully confirms the results of the theoretical studies that have been presented on the basis of the analysis of the preceding formulas. That is, the torque drops sharply when the angular displacement of the half coupling occurs.

Thus, during the studies recorded the maximum torque T_m , in ten times the repetition at different positions of the wells.

For experimental research, couplings were made with the following design parameters.

For coupling with radial engagement elements: C = 7530; 10900; 16400; 19100 N/m; R = 0.023 m; r = 0.0078 m; δ_0 = 0.01 m; h_l = 0.003 m.

These parameters were substituted for theoretical dependencies, which determined the maximum torque on the couplings. It was assumed that the coefficient of friction is f = 0.17.

According to the results of the research, it is established that for couplings with radial engagement elements, these parameters are: coefficient of accuracy of triggering γ_a = 1.27; torque T_m = 67.4 N·m; average quadratic deviation σ = 5.88 N·m; coefficient of variation υ = 8.7%.

The results of theoretical (solid lines) and experimental studies (dashed lines) are presented in Fig. 9 are clearly linear in nature. The upper pairs of graphs represent the main torque of the clutch when overload occurs.

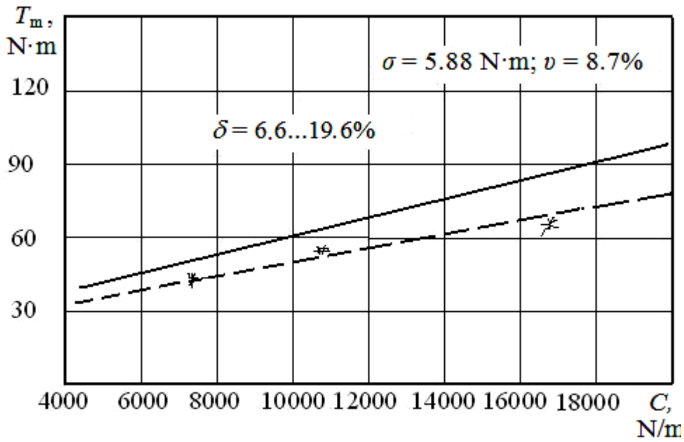

Fig. 9. Dependence of change of a torque of a coupling with radial elements of engagement on spring stiffness: δ - the error between the results of theoretical and experimental studies

Since the coefficient of accuracy of the coupling is determined by the main torque transmitting the coupling, a comparative analysis of the results of theoretical and experimental studies is performed for a pair of graphs.

Analysis of the graphical dependencies shown in Fig. 9 shows that the error between the results of theoretical and experimental studies is within 6.6…19.6% for different values of the experiment results.

Thus, according to the results of the experiment, it was established that the analytical dependencies (Fig. 9) were deduced in advance, in order to determine the torque transmitting the coupling adequately reflect the actual triggering processes of the developed couplings. Therefore, these analytical dependencies can be used in the engineering design of different sizes of such couplings.

In this case, given one constant parameter, you can determine the others, based on the required torque, which must transmit the clutch.

Experimental studies of the designed and manufactured ball safety couplings were carried out on the stand according to the procedure given above. Studies have also been conducted for ball bearing couplings with a change in the radius of the balls *R* and with different radii of balls *r*.

The results of the experimental studies are rearranged in Fig. 11 and 12.

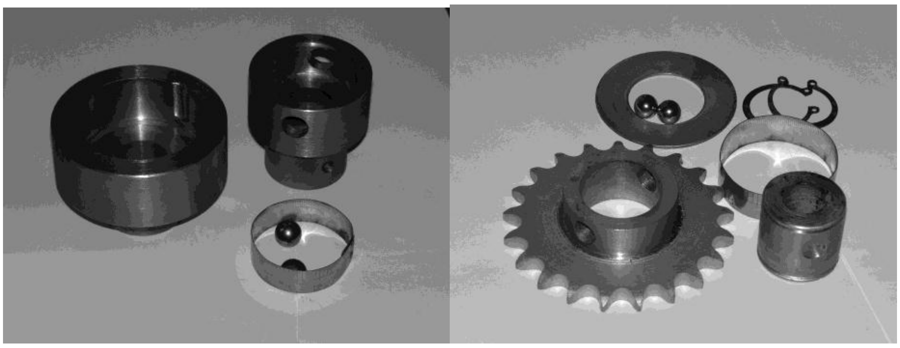

Fig. 10. General view of ball bearing coupling with radial engagement elements

The tendency of the influence of the parameters *R*, and *r* on the value of T_m for a clutch with radial engagement elements is similar to the one previously considered.

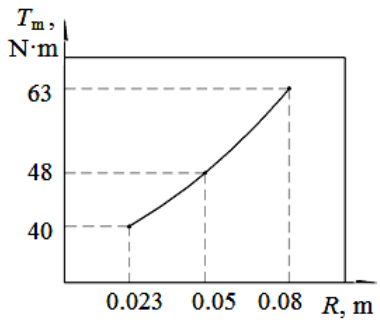

Fig. 11. Dependence of influence of radius of arrangement of balls on the value of torque

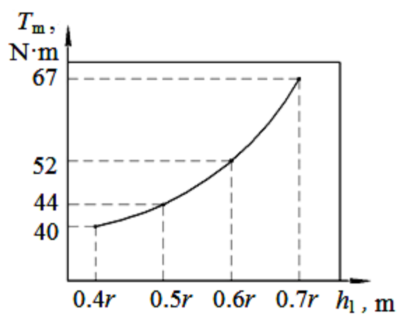

Fig. 12. Dependence of the influence of the depth of the holes on the torque value

For dependence $T_m = f(C)$ – 6.6% – 19.6%.

For dependence $T_m = f(R)$ – 7% – 18.5%.

For dependence $T_m = f(h_l)$ – 7.5% – 17%.

Thus, the results of experimental studies of the designed coupling have confirmed the validity of theoretical provisions that can be used to justify and select rational parameters of the designed coupling structures and their engineering design.

4. Conclusions

On the basis of the design schemes, experimental designs of low-dynamic ball coupling ball joints are designed and manufactured. On the basis of the milling machine a stand for research and determination of the operational characteristics of the safety couplings was developed, as well as the method of conducting the research.

According to the results of static studies, it is established that the coefficient of accuracy of triggering is $\gamma_a \approx 1.27$. Based on the statistical processing of torque scattering, it is established that average quadratic deviation is $\sigma = 5.88$ Nm, and the coefficient of variation – $\upsilon = 8.7\%$. The error between the results of theoretical and experimental studies is $\delta = 6.6 \ldots 19.6\%$.

Further research will be aimed at improving the design of ball safety couplings by increasing the number of balls to four. It is planned to investigate the change in torque during the operation of the clutch under shock loads on the drive of the machine.

R E F E R E N C E S

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