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LOAD CAPACITY AND DESIGN PARAMETERS OF BALL-TYPE SAFETY-OVERRUNNING CLUTCH WITH INCLINED GROOVES SIDES

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Abstract

Article deals with safety-overrunning clutches for mechanical transmissions. Modern design of safety-overrunning clutch with grooves sides inclined to semi-coupling radius has been described and researched in the article. It has practical value for creation modular-type machines. On the basis of the theoretic studies, the expressions for obtaining the main specific operation parameters have been proposed: rating torque, beginning and ending operation torques. As the result of the studies, the equations for estimation the clutch main operation characteristics have been received - rating torque exceeding coefficient, coefficients of clutch accuracy and sensitivity. On account of modeling and comparison with clutch where grooves sides are parallel to the radius made a number of important conclusions. The analysis performed demonstrates that clutches with inclined to radius grooves sides in general have higher operation characteristics compared with clutches with parallel to radius grooves sides, particularly higher accuracy coefficient and lower rating torque exceeding coefficient. Obtained results make it possible to recommend for highly loaded large-mass systems clutches with low values of grooves to clutch axe and grooves sides to radius inclination angles, because it provides balls contact with plane sides grooves surfaces and through this allows to decrease contact stresses compared with clutches with grooves sides parallel to radius; allows to provide high load capacity with low rating torque exceeding in overload mode; in clutches with inclined to radius grooves sides friction impact manifests less in operation with high rotation frequency.

Keywords: safety-overrunning clutch, torque, overload, semi-couplings, freewheel clutch, operation sensitivity, operation accuracy

1. ACTUALITY AND STATEMENT OF THE PROBLEM

Modern machines operate at high speeds and workloads, frequent changes in operating modes and overloads [1-3]. In such conditions, the development of devices for changing the speed and protection against overloads, capable with high load capacity and stability of characteristics to provide effective protection against overload is an important problem for mechanical engineering science.

2. ANALYSIS OF RECENT RESEARCH AND PUBLICATIONS

Created many years ago freewheel clutches, based on friction torque transfer principle have well studied disadvantages. Despite of that, such constructions are still using in modern engineering, particularly in vehicles [4-7]. Safety-overrunning clutches usually built on its basis by sequential joining of safety clutch [8]. New working principle in freewheel clutches - torque transfer through gearing balls and semi-couplings grooves is the source for new constructions creation [8-10]. On the basis of that principle new design of safety-overrunning clutch, with safety and freewheel parts are integrated, was developed and researched by authors [11-13]. In this clutch grooves with parallel to radius side surfaces can lead to balls and internal semi-coupling grooves edge contact and significant contact stresses increasing. To exclude impact of clutch parts manufacturing and assembling accuracy on contact stresses it is proposed [14] to incline grooves sides surfaces at an angle β to the semi-couplings radius, passing through the ball centre in diametric section. New clutch scheme is shown in Fig. 1 and Fig. 2.

The goal of the paper is evaluation of safety-overrunning clutch with inclined grooves sides load capacity, force parameters and operation characteristics.

Tasks solved in work are following:

- to estimate clutch load capacity, to get equations for its parts loads and rating torque determination;
- to derive equations for operation clutch operation beginning torque, and maximal operation process torque in overload mode;
- to value new clutch main operation characteristics - coefficients of accuracy sensitivity and rating torque exceeding in overload mode;
- to analyze constructive parameters influence on clutch basic operation characteristics;
- to outline the scope of new safety-overrunning clutches.

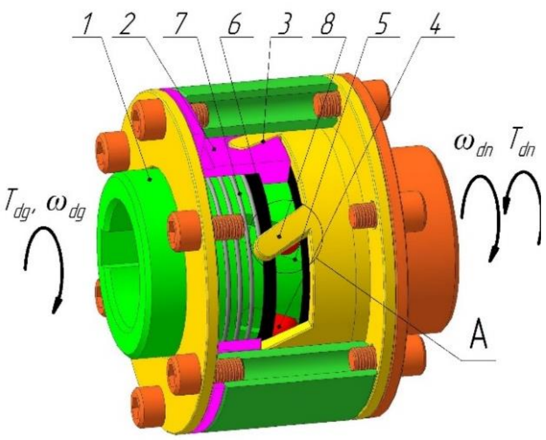

Fig. 1. Safety-overrunning clutch with inclined grooves sides scheme:
1 – internal semi-coupling (driving); 2 – external semi-coupling (driven); 3 – external semi-coupling groove; 4 – ball; 5 – internal semi-coupling groove; 6 – ring; 7 – overload spring; 8 – crown bush lug

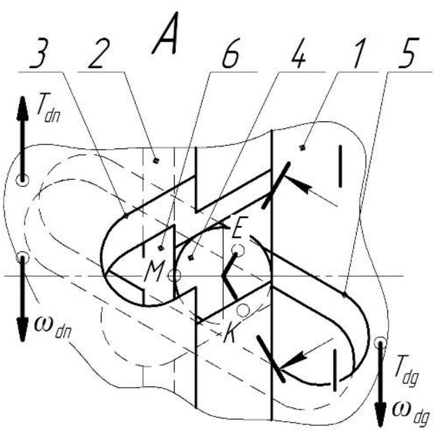

Fig. 2. Inclined grooves sides scheme (view *A* by fig. 1)

3. STATEMENT OF THE MAIN MATERIAL

Presented in article researches have been received with following admissions:

- the angles of oppositely directed grooves to the clutch axis inclination in the internal and external semi-couplings are peer in value and equal to α;

- the clutch details deformations are minimal and do not have tangible impact on the geometry, and the load transfer between them, so the balls are loaded evenly;
- forces are attached to the balls centers;
- when the clutch is operating, the balls have no rotation;
- spring rigidity during clutch operation is constant.

During the transferring the rating torque *T*, each ball stays in balance under the action of a system of converging forces (Fig. 2, Fig. 3), which are applied from grooves 5, 3 and ring 6 surfaces respectively at points *K*, *E*, and *M*.

At this points conjugate clutch parts acting on ball with forces N_1, N_2 and $F_{sp}^{/}$ (Fig. 3). Those forces value is due to clutch load and speed mode - *T* torque is the origin of tangential forces F_{ti} and its components F_{Nti}, N_{ti}, F_{Xti}:

$$F_{t1} = F_{t2} = \frac{2T}{zD}; \tag{1}$$

$$F_{Nt1} = F_{Nt2} = \frac{F_{ti}}{\cos\alpha} = \frac{2T}{zD\cos\alpha}; \tag{2}$$

$$F_{Xt1} = F_{Xt2} = F_{ti} tg\alpha = \frac{2T}{zD} tg\alpha; \tag{3}$$

$$N_{t1} = N_{t2} = \frac{F_{Nti}}{\cos\beta} = \frac{2T}{zD\cos\alpha\cos\beta}, \tag{4}$$

where *z* is the total balls in clutch number; *D* is the diameter of balls centers location circle.

Centrifugal forces F_ω, acting on balls due to grooves sides to radius inclination generate following components:

$$N_{\omega 2} = F_\omega \sin\beta = m_b \omega^2 D \sin\beta; \tag{5}$$

$$F_{N\omega 2} = N_{\omega 2}\cos\beta = \tag{6}$$

$$= F_\omega \sin\beta\cos\beta = 0,5 F_\omega \sin 2\beta;$$

$$F_{X\omega 2} = F_{N\omega 2}\sin\alpha = 0,5 F_\omega \sin 2\beta \sin\alpha \tag{7}$$

where $F_\omega = m_b \omega^2 D$ is centrifugal force acting on ball; m_b is ball mass; ω is clutch angular velocity.

Then full forces N_1 and N_2, which load balls and grooves constitute:

$$N_1 = N_{t1} = \frac{F_{Nt1}}{\cos\beta} = \frac{2T}{zD\cos\alpha\cos\beta}; \tag{8}$$

$$N_2 = N_{t2} + N_{\omega 2} = \frac{F_{Nt2}}{\cos\beta} + F_\omega \sin\beta = \frac{2T}{zD\cos\alpha\cos\beta} + F_\omega \sin\beta; \tag{9}$$

By force N_2 clutch parts contact stresses around *K* point (Fig. 3) could be determined according by well-known expression [15]:

$$\sigma_H = Z_M \sqrt[3]{\frac{N_2}{d^2}}, \tag{10}$$

where Z_M = 1755 MPa$^{2/3}$ is the coefficient which takes into account parts mechanical properties (for steel parts); *d* is ball diameter.

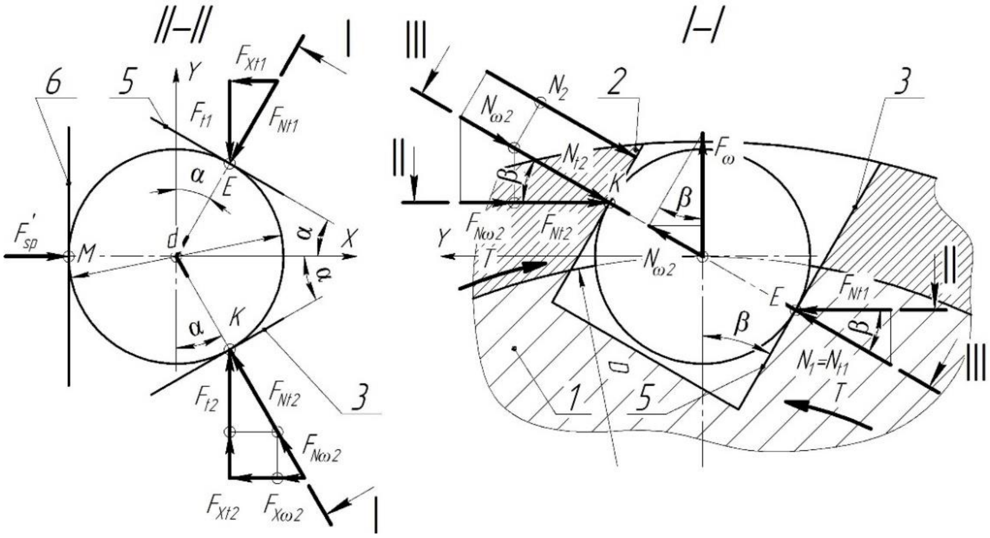

Fig. 3. Clutch parts force interaction diagram

Therefore rating torque that the clutch is capable to transfer could be determined by the ratio (12) and spring force by the ratio (13), usable for clutch designing:

$$T=\frac{F_{sp}-0,5zF_{\omega}\sin 2\beta\sin\alpha}{4tg\alpha}D=$$
$$=\frac{F_{sp}D}{4tg\alpha}\left[1-z\frac{F_{\omega}}{2F_{sp}}\sin 2\beta\sin\alpha\right]; \quad (12)$$

$$F_{sp}=\frac{4Ttg\alpha}{D}+0,5zm_b\omega^2 D\sin 2\beta\sin\alpha\,. \quad (13)$$

For beginning of clutch overload mode operation it is necessary to make torque T_b, which must exceed rating torque T, balls in grooves 3 and 5 friction forces torque T_{fg} and balls on ring 6 friction forces torque T_{fr}. Then the torque of safety part operation beginning will be:

$$T_b=T+T_{fg}+T_{fr}\,. \quad (14)$$

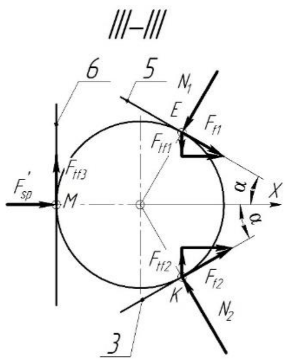

Fig. 4. Ball force diagram at the clutch overload mode beginning

The torque T_{fg} is generating by tangential components F_{tfi} of friction forces F_{fi} (Fig. 4). According to this could be written following equations:

$$T_{fg}=\frac{F_{tf1}+F_{tf2}}{2}zD=$$
$$=\frac{F_{f1}\sin\alpha+F_{f2}\sin\alpha}{2}zD=$$
$$=\left(N_1+N_2\right)\frac{f_g\sin\alpha}{2}zD= \quad (15)$$
$$=\left(\frac{4T}{zD\cos\alpha\cos\beta}+F_{\omega}\sin\beta\right)\frac{f_g\sin\alpha}{2}zD=$$
$$=\frac{2Ttg\alpha}{\cos\beta}f_g+\frac{zDF_{\omega}\sin 2\beta\sin\alpha}{4\cos\beta}f_g;$$

$$T_{fr}=F_{sp}\frac{D}{2}f_r\,. \quad (16)$$

where f_g and f_r is the sliding friction coefficients for balls on grooves 3 and 5 and ring 6 surfaces respectively.

Then, taking into account equations (14)…(16) and substituting to them ratios (12) and (13), we obtain more useful expression (17) for clutch safety part operation beginning torque:

$$T_b=T+T\frac{2tg\alpha}{\cos\beta}f_g+$$
$$+\frac{0,5zDF_{\omega}\sin 2\beta\sin\alpha}{2\cos\beta}f_g+F_{sp}\frac{D}{2}f_r. \quad (17)$$

For equivalence friction coefficients in grooves and on ring ($f_g=f_r=f$) we obtain expression (18).

Cognition obtained the equations (12) and (18) makes it possible to derive expressions for rate the clutch safety part basic operation characteristics, particularly rating torque exceeding coefficient k_e (19) and accuracy coefficient γ_a (20)

$$T_b=\frac{F_{sp}D}{4tg\alpha\cos\beta}\times$$
$$\times\left[2\,ftg\alpha\left(1+\cos\beta\right)+\cos\beta\left(1-z\frac{m_b\omega^2 D}{2F_{sp}}\sin 2\beta\sin\alpha\right)\right]; \quad (18)$$

$$\gamma_m = \frac{\left[2 f_m tg\alpha \left(1+\cos\beta\right) + \cos\beta \left(1 - z \dfrac{m_b \omega^2 D}{2F_{sp}} \sin 2\beta \sin\alpha \right) \right]}{\left[2 f_n tg\alpha \left(1+\cos\beta\right) + \cos\beta \left(1 - z \dfrac{m_b \omega^2 D}{2F_{sp}} \sin 2\beta \sin\alpha \right) \right]}; \quad (19)$$

$$k_e = \frac{\left[2 f tg\alpha \left(1+\cos\beta\right) + \cos\beta \left(1 - z \dfrac{m_b \omega^2 D}{2F_{sp}} \sin 2\beta \sin\alpha \right) \right]}{\cos\beta \left[1 - z \dfrac{m_b \omega^2 D}{2F_{sp}} \sin 2\beta \sin\alpha \right]}; \quad (20)$$

where f is the sliding friction coefficient (f – medium, f_n – minimum, f_m – maximum).

At the end of clutch operating in overload mode spring 7 (Fig. 1) reaches the deformation (21) [13].

$$\lambda_0 = 0,5d(\sin\alpha + 1) \cdot \quad (21)$$

For such deformation creation, clutch must be loaded with maximal operation torque T_{max} (22). Therefore sensitivity coefficient γ_s could estimate by ratio (23).

$$T_{\max} = \frac{(F_{sp} + C_{sp}\lambda_0)D}{4tg\alpha\cos\beta} \times$$
$$\times \left[\begin{array}{l} 2 f tg\alpha \left(1+\cos\beta\right) + \cos\beta \times \\ \times \left(1 - z \dfrac{m_b \omega^2 D}{2(F_{sp} + C_{sp}\lambda_0)} \sin 2\beta \sin\alpha \right) \end{array} \right]; \quad (22)$$

$$\gamma_s = \frac{F_{sp}}{(F_{sp} + C_{sp}\lambda_0)} \times$$
$$\times \frac{\left[2 f tg\alpha \left(1+\cos\beta\right) + \cos\beta \left(1 - z \dfrac{m_b \omega^2 D}{2F_{sp}} \sin 2\beta \sin\alpha \right) \right]}{\left[2 f tg\alpha \left(1+\cos\beta\right) + \cos\beta \left(1 - z \dfrac{m_b \omega^2 D}{2(F_{sp} + C_{sp}\lambda_0)} \sin 2\beta \sin\alpha \right) \right]}. \quad (23)$$

To judge the mutual effect of design and clutch characteristics in overload mode, calculations for the clutch with following main parameters have been performed: balls center arrangement diameter D = 60 mm, ball diameter d = 9.525 mm (from standard bearing), balls amount in clutch z = 6, grooves to clutch axe inclination angle α = 30º, rotation frequency n = 1500 rpm, friction coefficients f = 0.10, f_m = 0.15, f_n = 0.05, spring tightening initial force F_{sp} = 50 N, spring rigidity C_{sp} = 20 N/mm, grooves sides to radius inclination angle β = 5…45º. Modeling results are shown in Fig. 5 to Fig. 13.

4. DISCUSSION OF RESULTS

Graphs, shown further illustrate grooves sides to radius inclination angle β on torques (12), (18), (22) (Fig. 5) and coefficients (19), (20), (23) (Fig. 7), which makes it possible to estimate the clutch operation parameters and it parts contact stresses (Fig. 6).

Their analysis gives an opportunity to draw a conclusion that in order to increase the clutch load capacity and contact stresses decreasing, the semi-couplings grooves sides must be made with less inclination to radius angles.

When β value increases from 5º to 45º clutch load capacity decreases by 13.4% (rating torque T decreases from 1.264 N·m до 1.095 N·m) (Fig. 5), accordingly operation beginning torque T_b decreases from 1.564 N·m to 1.458 N·m, and contact stress σ_H (10) increases by 60% (from 55 MPa to 88 MPa) (Fig. 6).

Increasing β inclination angle also have impact on clutch operating characteristics (Fig. 7). Particularly, clutch operating accuracy getting a little worse because coefficient of accuracy γ_a increases by 8.2 % (from 1.384 to 1.497), rating torque exceeding coefficient k_e increases by 7.5% (from 1.238 to 1.331). Clutch overloads sensitivity also getting a little worse –coefficient of sensitivity γ_s decreases by 10.3% (from 0.214 to 0.192).

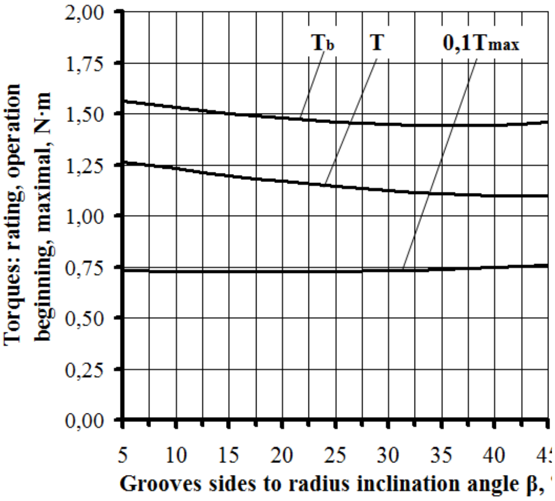

Fig. 5. Influence of grooves sides to radius inclination angle on the clutch load capacity

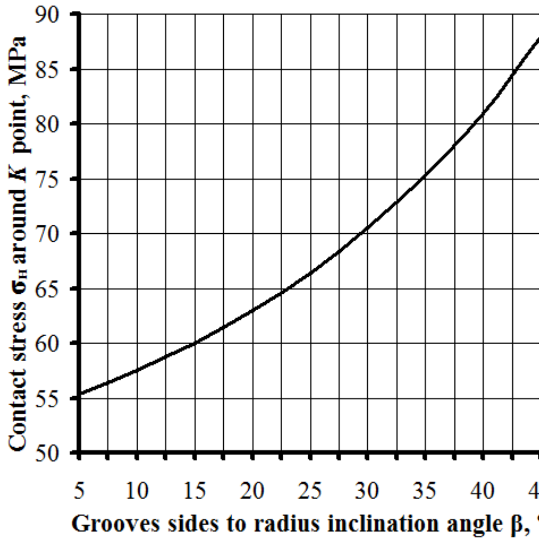

Fig. 6. Influence of grooves sides to radius inclination angle on clutch parts contact stress around K point

Graphs shown in Fig. 8 to Fig. 10 illustrate grooves to clutch axe inclination angle α influence on main operation characteristics of clutches with grooves sides parallel to radius (built with dotted lines) and sides inclined (β = 5º) to radius (built with solid lines). Results, obtained by modeling demonstrate that angle α value impacts on clutch characteristics significantly. Particularly small grooves to axe inclination angles $\alpha = 5…10°$ (Fig. 8) provide highest load capacity and lowest torque loads while operation (lowest coefficient k_e value). So, clutch with inclined to radius grooves sides provides rating torque coefficient exceeding k_e in 2.15…1.55 times less than coupling with parallel grooves sides (1.035…1.071 against 2.223…1.665).

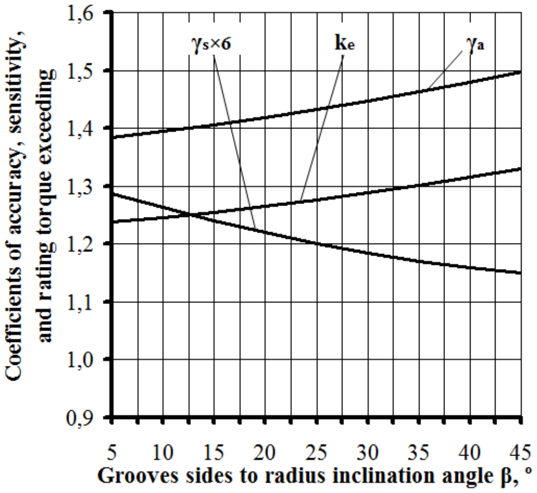

Fig. 7. Influence of grooves sides to radius inclination angle on accuracy, rating torque exceeding, and clutch sensitivity coefficients

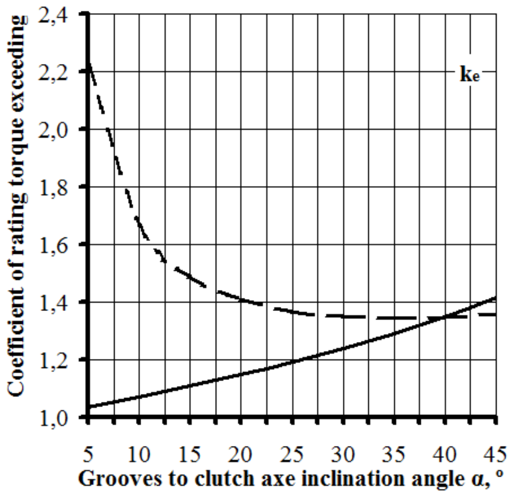

Fig. 8. Influence of the semi-couplings grooves inclination angle on the rating torque exceeding coefficient

For greater angle α values this advantage getting lower – when α = 25º, k_e coefficient of clutch with inclined to radius grooves sides getting value only by 15% less than of clutch with parallel to radius grooves sides, and when α = 40º torque exceeding coefficients for both clutches became equal ($k_e \approx 1.350$). This makes it possible to recommend for highly loaded large-mass systems clutches with low values of α and β angles – it makes possible to provide high load capacity with low rating torque exceeding in overload mode.

From the point of view for friction coefficients value sensitivity (coefficient of accuracy γ_a to one proximity), clutch with inclined to radius grooves sides also demonstrates advantages (Fig. 9). So, when α = 5…20º, its coefficient of accuracy is by 65.0…6.5% lower than of clutch with parallel to radius grooves sides (1.068…1.259 against 1.759…1.340). Around α = 25º coefficients of accuracy for both types of clutches became equal ($\gamma_a \approx 1.315$). With further angle α value increasing operation accuracy of clutch with parallel to radius sides exceeds this characteristic of clutch with inclined grooves sides.

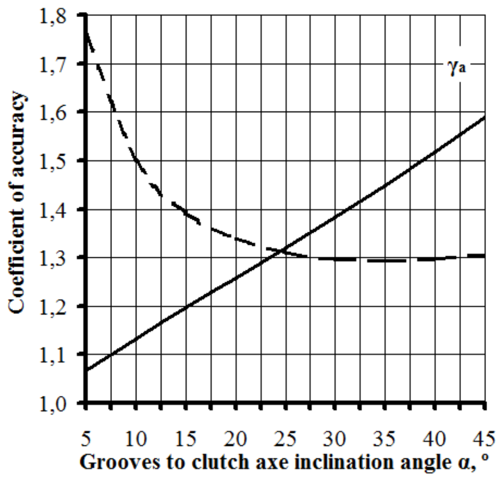

Fig. 9. Influence of the semi-couplings grooves inclination angle on the accuracy coefficient

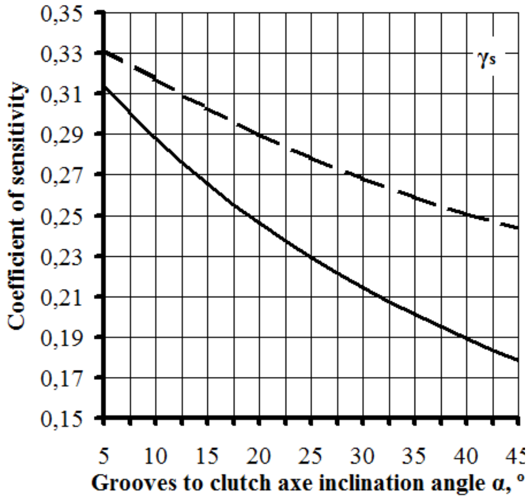

Fig. 10. Influence of the half-couplings grooves inclination angle on the sensitivity coefficient

Clutch with inclined to radius grooves sides has lower sensitivity to overloads than clutch with parallel grooves sides (Fig. 10) – when α = 5…45º first one has by 5…18% higher coefficient of sensitivity γ_s then second one (0.332…0.244 against 0.314…0.179).

Graphs presented in Fig. 11 to Fig. 13 demonstrate the influence of the rotation frequency n on clutch accuracy, sensitivity and rating torque exceeding coefficients. Their analysis makes possible to state that generally clutch with inclined grooves sides characteristics less sensitive to rotation frequency n change. This can be explained by the

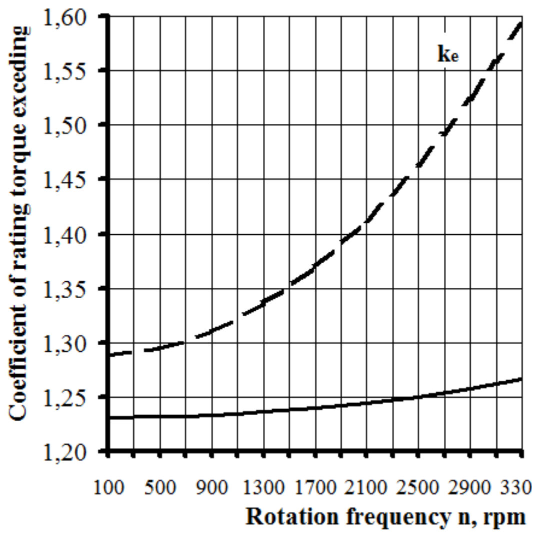

Fig. 11. Influence of the clutch rotating frequency on the rating torque exceeding coefficient

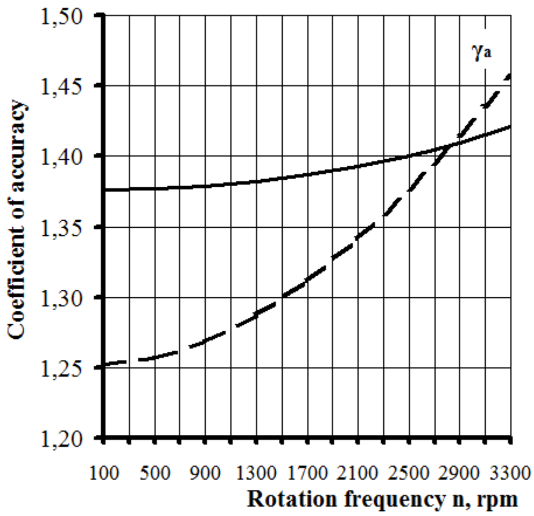

Fig. 12. Influence of the clutch rotating frequency on the accuracy coefficient

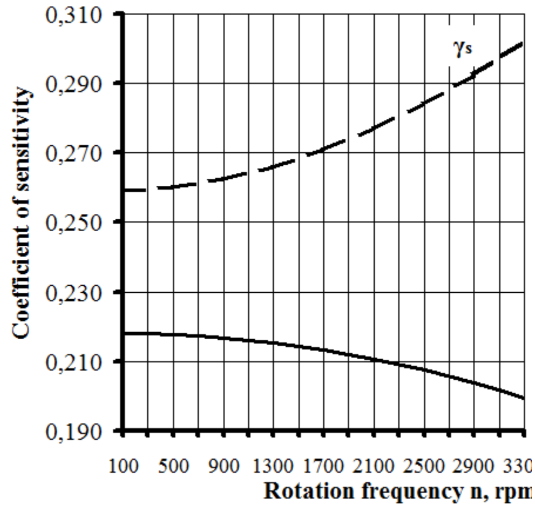

Fig. 13. Influence of the clutch rotating frequency on the sensitivity coefficient

fact is in clutch with parallel to radius grooves sides centrifugal force F_ω fully accept buy crown bush lugs 8 internal surface (Fig. 1) and causes appropriate friction force which increases beginning operation torque T_b. Instead of this, in clutch with grooves sides inclined to radius, friction force while operation produces only by component $N_{\omega 2} = F_\omega \sin\beta$, that at $\beta = 5°$ gives friction reducing by 90%. Mentioned design feature define change of analyzed clutch characteristics. In rotation frequency interval $n = 100…3300$ rpm rating torque exceeding coefficient k_e (Fig. 11) of clutch with inclined grooves sides is by 4.7…25.8% lower than of clutch with parallel to radius sides (1.231…1.266 against 1.289…1.592). In mentioned rotation frequency interval coefficient of accuracy γ_a (Fig. 12) of coupling with inclined to radius sides is more stable – it changes only by 3.3% (from 1.376 to 1.421), and of clutch with parallel to radius grooves this coefficient changes by 16.4% (from 1.252 to 1.457), coefficient of sensitivity γ_s (Fig. 13) respectively by 8.3% (from 0.218 to 0.200) and by 16.6% (from 0.259 to 0.302).

5. CONCLUSIONS

1. Main parts force interaction of new ball-type safety-overrunning clutch with inclined to radius grooves sides researched in the paper. On the basis of the theoretic studies, the expressions for obtaining the main specific operation parameters have been proposed: rating torque, beginning and ending operation torques. As the result of the studies, the equations for estimation the clutch main operation characteristics have been received - rating torque exceeding coefficient, coefficients of clutch accuracy and sensitivity. Mentioned equations have practical value for presented modern clutch designing.
2. It is shown that clutches with inclined to radius grooves sides in general have higher operation characteristics compared with clutches with parallel to radius grooves sides, particularly higher accuracy coefficient and lower rating torque exceeding coefficient.
3. Obtained results make it possible to recommend for highly loaded large-mass systems clutches with low values of grooves to clutch axe and grooves sides to radius inclination angles by the following reasons:

– it provides balls contact with plane (quasi-plane) sides grooves surfaces and through this allows to decrease contact stresses compared with clutches with grooves sides parallel to radius;

– it makes possible to recommend for highly loaded large-mass systems clutches with low values of α and β angles – it allows to provide high load capacity with low rating torque exceeding in overload mode;

- in this type of clutch friction impact manifests less in operation with high rotation frequency.

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