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SAFETY-OVERRUNNING CLUTCH ELASTIC TORQUE OPERATION CHARACTERISTIC STUDY

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Abstract

The article deals with ball-type safety-overrunning clutches' elastic characteristics in overload mode. Main semi-couplings geometric parameters, particularly ball stroke and semi-couplings twisted angles, affecting operation speed and clutch elastic torque considered and obtained expressions for this parameters determination. Main clutch parts force interaction features in the final overload operation period analyzed and ratios for the clutch final period load are obtained. Based on current and previous studies, the expressions for elastic torque in the overload operation period are estimated. Obtained results could become the tool for the dynamics analysis of studied clutch-equipped driving operation during the clutch reengagement in overload mode. It is shown the expedience of using in high-speed driving clutches with large values of grooves to clutch axis inclination angle to decrease dynamic loads in driving, because in little and middle angles values provide nosedive dropping of final stage torque which can become a source of intense oscillations in the driving equipped with clutch in overload mode.

Keywords: safety-overrunning clutch, overload, elastic characteristic, engaging, operation mode

1. ACTUALITY AND STATEMENT OF THE PROBLEM

Plenty of contemporary created and studying machines are now equipped with overrunning clutches [1-4]. Most of them require dynamic load estimation, which needs information about driving elements torsion stiffness, wear etc. Therefore implementation in these mechanisms of new types of transmission links needs evaluation of their characteristics, particularly operation elastic torque. Thereby estimation of a dynamic characteristic of new promising driving elements is an important problem for machinery science.

2. ANALYSIS OF RECENT RESEARCH AND PUBLICATIONS

Friction-operated overrunning clutches are still in use in widespread driving, particularly in internal combustion engine starters' mechanisms. Most commonly used in modern driving are the roller-type clutches. This type of clutch design method has been the focus of sustained research attention. Thus, in [5] researched the failure mechanisms of the clutch and proposed a way of its service period prolongation by contact zone stiffness magnification. That became a source for driving equipped with clutch dynamic model development and analysis. In [6] contact interaction of roller-type clutch by finite element method was studied. In [7] overrunning clutch design was researched using fuzzy theory. In [8] presented a numerical experiment with an overrunning clutch and studied the dynamics of its elements jamming. In [9] it is shown that using sprag-type clutches with heterogeneous wedges instead of rollers provides significant structure simplification. The latest research [10-15] directed to improve its characteristics despite well-studied disadvantages [16-20] that have their source in the friction energy transfer principle.

Ball-type gearing-operated overrunning and safety overrunning clutches with grooves sides parallel to semi-couplings radius (straight grooves) and with grooves sides inclined to the radius (inclined grooves) proposed [16, 17] and developed [18-20] by authors have much more load capacity than friction-operated structures due to their operation principle and are promising for implementation in driving mechanisms, especially large-mass and heavy-loaded.

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Mentioned ball-type clutches operation in overload mode can be divided into two periods – initial when maintained balls engagement with both semi-couplings grooves, and final, when balls get out of engagement with external semi-coupling grooves and provide them the opportunity for free rotation (idling period).

Clutch operation in the initial period was studied in [18] and [20] where elastic torque ratios were obtained. Particularly, in [18] studied clutch with straight and in [20] – with inclined grooves. In [19] inclined grooves sides in gearing-operated clutches implementation was justified.

However, a final period pattern still needs investigation.

The aim of the article is to study of safety-overrunning clutches with straight and inclined grooves sides overload mode final operation period.

Tasks solving in the paper are the following:
- to get expressions for semi-coupling elements' main geometric parameters calculation;
- to obtain ratios for clutch semi-couplings twisted angles for both operation periods;
- to derive equations for clutch elements loads and elastic torque in final operation and idling periods;
- to built clutch characteristics - elastic torque from twisted angle dependence;
- to analyze the impact of clutch constructive parameters on its elastic torque in the final operation period.

3. STATEMENT OF THE MAIN MATERIAL

In the initial period (geometry and force parameters of it further marked with «b» index) external semi-coupling grooves 1 and internal semi-coupling grooves 2 are in gear with balls. Balls move along grooves, relocating spring-loaded ring 3 and compressing its spring. In this period each ball center moves from O point to O1.

After that begins the final period (parameters of it marked with «e» index), when balls roll over B edge of external semi-coupling grooves 1 and get out of gear with it, staying in grooves 2 of internal semi-coupling.

For internal semi-coupling, we could write following (fig. 1):

$$KO_3 = \frac{O_3A}{\cos\alpha} = \frac{d}{2\cos\alpha}; \qquad (1)$$

$$h_g = KO_3 - GO_3 = \frac{d}{2}\left[\frac{1-\cos\alpha}{\cos\alpha}\right]; \qquad (2)$$

$$\delta_g = GH = \frac{d}{2}\left[1-\cos\alpha\right], \qquad (3)$$

where α is the angle of oppositely directed grooves to the clutch axis inclination in the internal and external semi-couplings; *d* is ball diameter.

Respectively, internal semi-coupling 2 groove length along its axis and milling cutter stroke will be define as:

$$L^{\mathrm{int}}_{n\min} = 3d + 4h_g = d\left[1+\frac{2}{\cos\alpha}\right]; \qquad (4)$$

$$H^{\mathrm{int}}_{mill} = O_3O_3' = L^{\mathrm{int}}_g - d = \frac{2d}{\cos\alpha}. \qquad (5)$$

Similarly for external semi-coupling groove 1 milling cutter stroke will amount like:

$$H^{ext}_{mill} = A'A'' = 2h_g + 2d - 2\delta_g = d\left[\frac{1+\cos^2\alpha}{\cos\alpha}\right]. \qquad (6)$$

And external semi-coupling groove length will be following:

$$L^{ext}_g = H^{ext}_{mill} - h_g. \qquad (7)$$

Ball full stroke in groove and along clutch axis when clutch actuated (both in overload and overrunning clutch modes):

$$h_{op} = h_b + h_e = d + 2h_g = \frac{d}{\cos\alpha}; \qquad (8)$$

$$\lambda_{op} = \lambda_b + \lambda_e = h_{op}\cos\alpha = d. \qquad (9)$$

Arc length that corresponds to clutch twist angle and clutch full operation twist angle will constitute:

$$l_{op} = l_b + l_e = h_{op}\sin\alpha = dtg\alpha; \qquad (10)$$

$$\varphi_{op} = \varphi_b + \varphi_e = \frac{2l_{op}}{D} = \frac{2d}{D}tg\alpha, \qquad (11)$$

where *D* is balls center arrangement diameter.

For clutch operation initial period we will have following ball stroke and clutch twist angle:

$$\lambda_b = 0,5d(1+\sin\alpha); \qquad (12)$$

$$h_b = \frac{\lambda_b}{\cos\alpha} = \frac{d(1+\sin\alpha)}{2\cos\alpha}; \qquad (13)$$

$$l_b = \lambda_b tg\alpha = 0,5d(1+\sin\alpha)tg\alpha; \qquad (14)$$

$$\varphi_b = \frac{2l_b}{D} = \frac{d}{D}(1+\sin\alpha)tg\alpha. \qquad (15)$$

Clutch elastic torque at the initial period [10, 12] can be defined by (16) for clutch with straight grooves sides and by (17) for clutch with inclined grooves sides

$$T^{sgs}_{bi} = \frac{D(F_{sp}+C_{sp}\lambda_i)}{4tg\alpha}\times \left(f\left[ctg\alpha + \frac{zF_\omega}{(F_{sp}+C_{sp}\lambda_i)} + 2tg\alpha\right]+1\right); \qquad (16)$$

$$T^{igs}_{bi} = \frac{D(F_{sp}+C_{sp}\lambda_i)}{4tg\alpha\cos\beta}\times \left[\begin{array}{c} 2\,ftg\alpha\left(1+\cos\beta\right)+ \\ +\cos\beta\left(1-\dfrac{zF_\omega}{2(F_{sp}+C_{sp}\lambda_i)}\sin 2\beta\sin\alpha\right)\end{array}\right], \qquad (17)$$

where *z* is the total balls in clutch number; m_b is ball mass; f is sliding friction coefficient; ω is clutch angular velocity; C_{sp} is spring rigidity; $F_\omega = 0,5m_b\omega^2D$ - ball centrifugal force, $\lambda_i = 0...\lambda_b$ ($\varphi_i = 0...\varphi_b$).

The results obtained below make it possible to simulate clutch elastic characteristics in the initial period of its operation in overload mode.

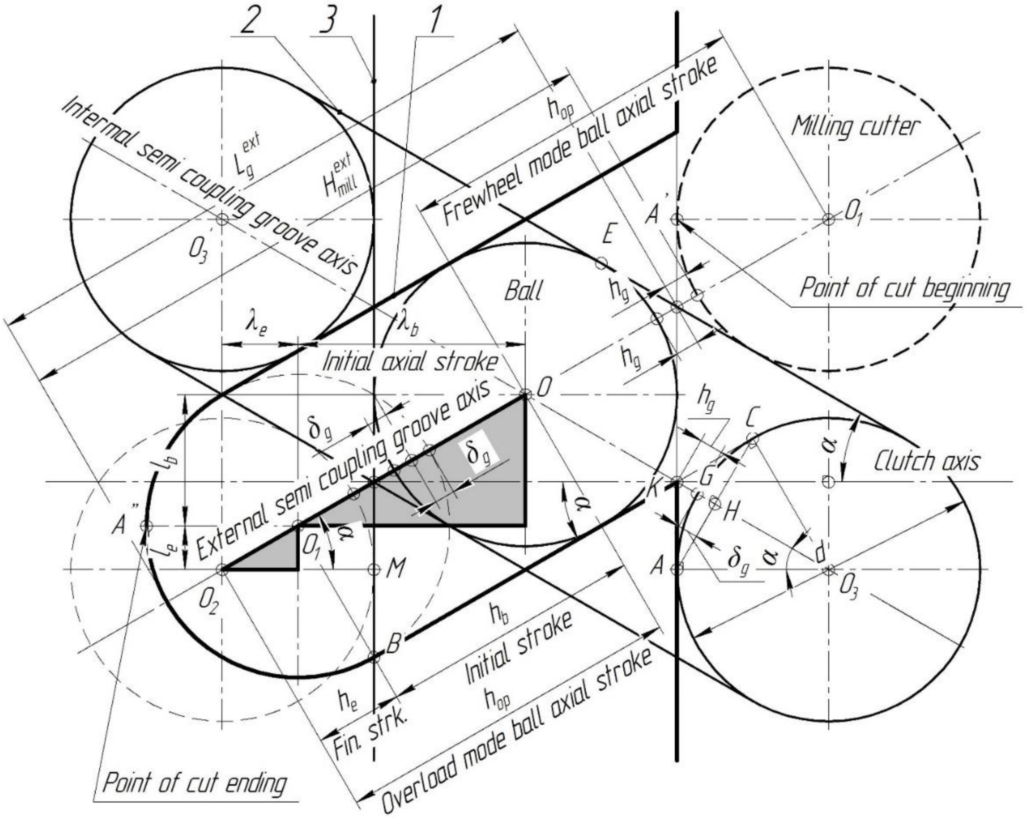

Fig. 1. Clutch grooves geometry diagram

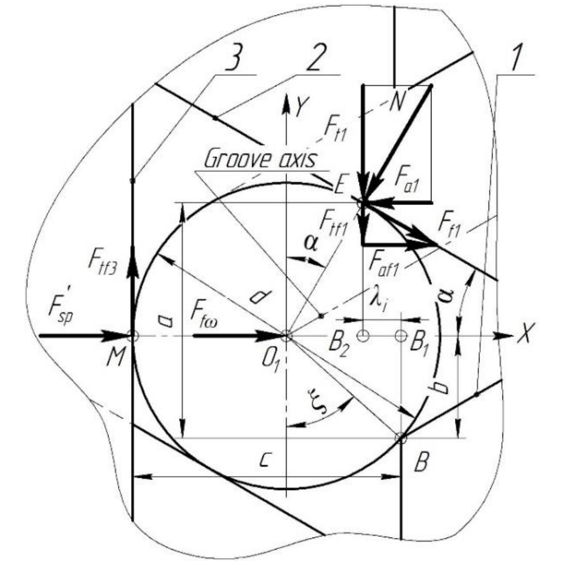

Fig. 2. Clutch with straight grooves sides element force interaction scheme at final operation period

For geometric and kinematic parameters of clutch in final period of its operation the following relations are valid:

$$\lambda_e = \lambda_{op} - \lambda_b = 0,5d\left(1-\sin\alpha\right). \tag{18}$$

$$h_e = \frac{\lambda_e}{\cos\alpha} = \frac{d(1-\sin\alpha)}{2\cos\alpha}; \tag{19}$$

$$l_e = \lambda_e tg\alpha = 0,5d(1-\sin\alpha)tg\alpha; \tag{20}$$

$$\varphi_e = \frac{2l_e}{D} = \frac{d}{D}(1-\sin\alpha)tg\alpha. \tag{21}$$

To get an expression to define clutch elastic torque at the final operation period consider the scheme of its parts force interaction in this stage. The ball moves then under the normal force N action from groove 2 internal semi-coupling pressure (fig. 2).

Tangential component F_{t1} creates exactly clutch elastic torque. Spring-loaded ring 3 puts pressure on the ball creating regenerative moment. Friction forces F_{tf1}, F_{af1}, F_{tf3}, $F_{f\omega}$ also prevent ball capsizing over the edge B. Ball equilibrium equation there will be following:

$$\begin{gathered}\sum M_B = (F_{t1} + F_{tf1})\lambda_i + F_{a1}a - \\ -(F_{sp}^{/} + F_{f\omega})b - F_{tf3}c - F_{af1}a = 0,\end{gathered}$$

or (22)

$$\begin{gathered}(F_{t1} + F_{t1}\, ftg\alpha)\lambda_i + F_{t1}tg\alpha a - \\ -(F_{sp}^{/} + F_{f\omega})b - F_{sp}^{/} fc - F_{t1} fa = 0.\end{gathered}$$

From the last equation we get

$$F_{t1} = \frac{F_{spi}^{/}(b+fc) + fF_{\omega}b}{\lambda_i(1+ftg\alpha) + a(tg\alpha - f)}, \tag{23}$$

where $F_{spi}^{/} = F_{spi} / z$ is partial spring force, acting each ball; *a, b, c* are the arms of the forces; λ_i – current ball displacement value along clutch axis.

Then clutch with grooves straight sides elastic torque at the final operation period can be determined from the expression:

$$T_{ei}^{sgs} = zF_{t1}\frac{D}{2} = \frac{0,5D\left[\left\{F_{sp}+C_{sp}(\lambda_b+\lambda_i)\right\}(b+fc)+zfF_\omega b\right]}{\lambda_i(1+ftg\alpha)+a(tg\alpha-f)}. \quad (24)$$

For clutch with inclined grooves sides rightly follow ratios (fig.3).

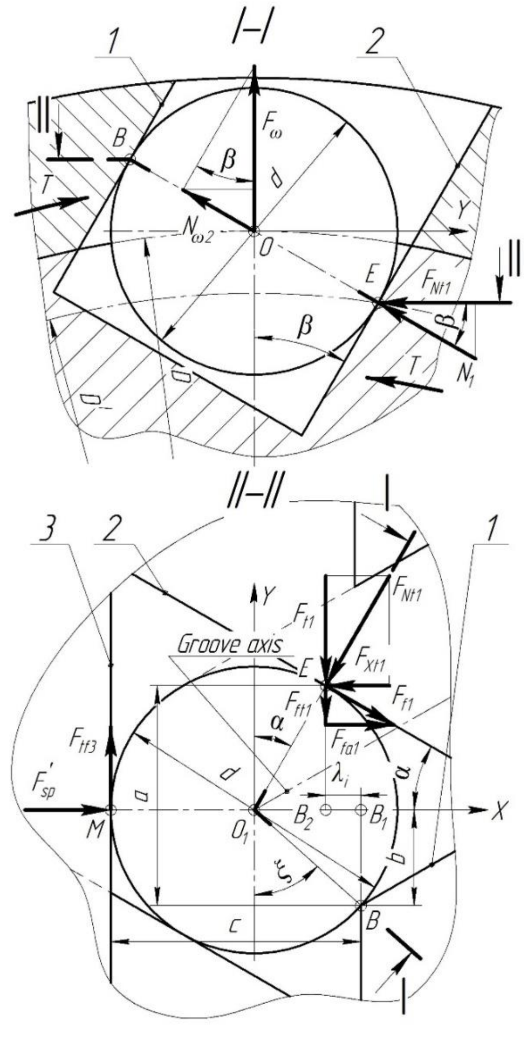

Fig. 3. Clutch with inclined grooves sides element force interaction scheme at final operation period

$$\sum M_B = (F_{t1}+F_{tf1})\lambda_i + F_{Xt1}a - F'_{sp}b - F_{tf3}c - F_{af1}a = 0,$$

or (25)

$$F_{t1}\left[(1+ftg\alpha)\lambda_i + (tg\alpha - f)a\right] - F'_{sp}(b+fc) = 0;$$

$$F_{t1} = \frac{F'_{sp}(b+fc)}{\lambda_i(1+ftg\alpha)+a(tg\alpha-f)}; \quad (26)$$

$$T_{ei}^{igs} = zF_{t1}\frac{D'}{2} = \frac{0,5D'\left\{F_{sp}+C_{sp}(\lambda_b+\lambda_i)\right\}(b+fc)}{\lambda_i(1+ftg\alpha)+a(tg\alpha-f)}, \quad (27)$$

where $D' \approx D - d\sin\beta$ is coerced diameter.

Corresponding distances could be determined as:

$$c = MO_1 + O_1B_2 + B_2B_1 = 0,5d + 0,5d\sin\alpha + \lambda_i = 0,5d(1+\sin\alpha)+\lambda_i; \quad (28)$$

$$\sin\xi = \frac{O_1B_1}{O_1B} = \frac{0,5d\sin\alpha+\lambda_i}{0,5d} = \sin\alpha + 2\frac{\lambda_i}{d}; \quad (29)$$

$$a = O_1E\cos\alpha + O_1B\cos\xi = 0,5d\left[\cos\alpha + \sqrt{1-\left[\sin\alpha+2\frac{\lambda_i}{d}\right]^2}\right]; \quad (30)$$

$$b = BB_1 = O_1B\cos\xi = 0,5d\cos\xi = 0,5d\sqrt{1-\left[\sin\alpha+2\frac{\lambda_i}{d}\right]^2}; \quad (31)$$

$$\lambda_i = (\varphi_i - \varphi_b)\frac{D}{2tg\alpha}. \quad (32)$$

After balls and grooves 1 engaging finish it will move between ring 3 and external semi-coupling ends, clamped by spring, sliding on external semi-coupling internal cylindrical surface pressed against her by centrifugal force.

Than clutch idling torque will constitute:

$$T_{idl} = D\left(\left[F_{sp}+C_{sp}\lambda_{op}\right]\frac{k}{d} + 0,5zfF_\omega\right), \quad (33)$$

where k = 0,0015 mm – rolling friction coefficient.

Torque (33) will load clutch-equipped shafts in the twisted angle $\varphi_i = \varphi_{op}...2\pi/z$ till balls and grooves start new engagement.

4. DISCUSSION OF RESULTS

Thus, clutch elastic torque for twist angles $\varphi_i = 0…\varphi_b$ (15) determined by expressions (16) and (17) for clutches with straight and inclined grooves respectively. Than for twist angles $\varphi_i = \varphi_b…\varphi_{op}$ by (24) and (27), and after clutch disengaged shafts, in $\varphi_i > \varphi_{op}$ – by expression (33).

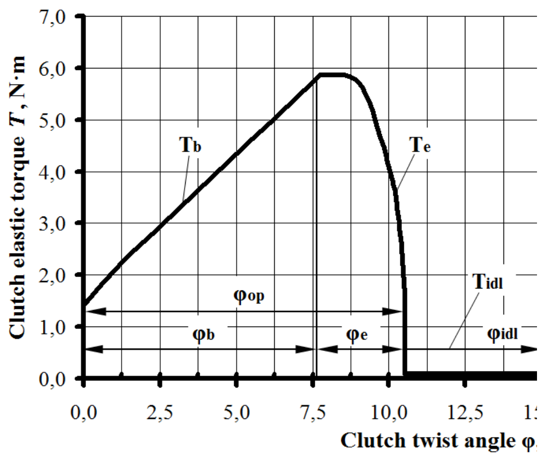

Fig. 4. Clutch elastic torque of semi-couplings twisted angle dependence (straight grooves sides, α = 30º)

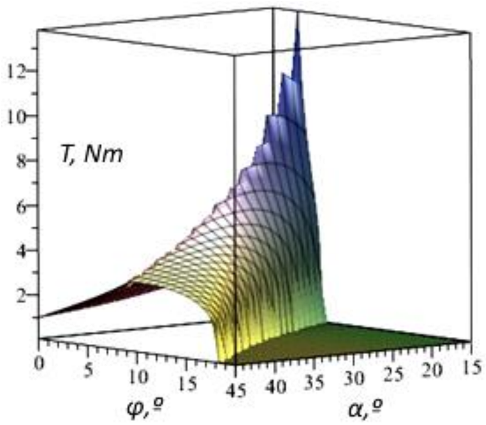

Fig. 5. Influence of grooves to clutch axis inclination angle on clutch elastic torque (clutch with straight grooves sides)

To illustrate the mutual effect of design parameters' influence on clutch elastic torque, simulation for the clutch with the following defining dimensions and characteristics have been performed: ball center arrangement diameter D = 60 mm, ball diameter d = 9.525 mm (from standard bearing), balls quantity in clutch z = 6, grooves to clutch axis inclination angle α = 15…45º, rotation frequency n = 1500 rpm, friction coefficients f = 0.10, spring tightening initial force F_{sp} = 50 N, spring rigidity C_{sp} = 20 N/mm, grooves sides to radius inclination angle β = 5…45º.

Simulation results are shown in Fig. 4 to Fig. 6. Fig. 4 illustrates clutch witch straight grooves sides elastic torque of semi-couplings twisted angle dependence (α = 30º) where are operation intervals boundaries indicated.

In fig. 5 shows built obtained expressions clutch with listed parameters elastic characteristic which illustrates grooves to clutch axis inclination angle α influence on elastic torque nature change. Results, received by this simulation show that grooves to clutch inclination angle α value affect on clutch elastic torque tangibly. Particularly little and middle angles α = 15…25º provide the highest load capacity but final stage torque T_e dropping of nosedive. In clutch overload mode operation this phenomenon can lead to an increase of the semi-coupling relative speed in the interval between balls and grooves engagements which in turn can become a source of intense oscillations in the driving equipped with clutch. This brings us to the result of the expedience of using clutches with large values of α about 40…45º to decrease dynamic loads in driving.

The graph shown in fig. 6 illustrates the minor impact of grooves sides to radius inclination angle β on clutch elastic torque variation in overload mode. So, there is no reason to increase the value of β angle because it can lead to clutch parts overload without significant load capacity or another clutch characteristics magnification.

Fig. 7 data gives an opportunity to confirm the lack of balls diameter to its centers' arrangement diameter ratio on final stage clutch elastic torque changing nature influence. The balls diameter increasing provides only clutch load capacity magnification.

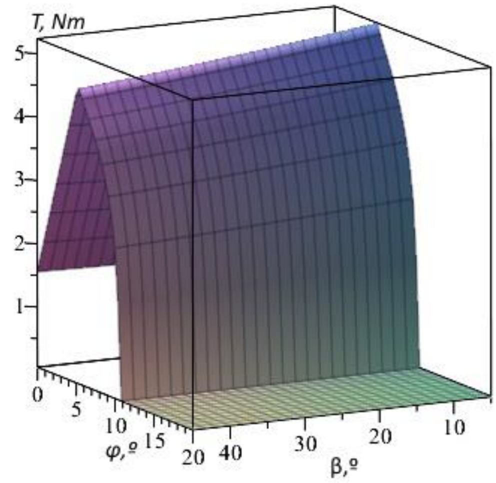

Fig. 6. Influence of grooves sides to clutch radius inclination angle on clutch elastic torque (inclined grooves sides, α = 30º)

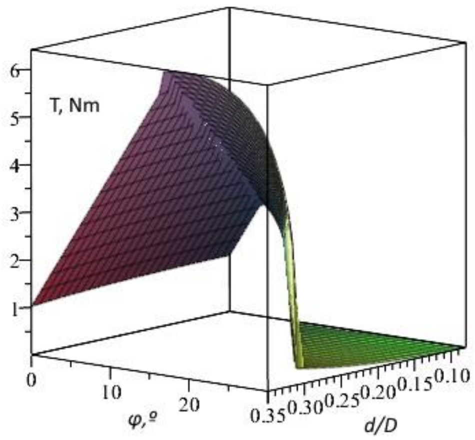

Fig. 7. Influence of balls diameters to its center's arrangement diameter ratio on clutch elastic torque (inclined grooves sides, α = 40º, β = 30º)

5. CONCLUSIONS

1. Ball-type safety-overrunning clutch with straight and inclined to radius grooves sides elements force interaction in the final clutch operation period studied in the article. On this basis, the expressions for clutch torque elimination in this period were obtained.
2. The presented results gave an opportunity for estimation of clutch elastic torque with semi-couplings twisted angle relation which could become the tool for the dynamics analysis of studied clutch equipped driving operation during the clutch reengagement in overload mode.
3. It is shown the expedience of using in high-speed driving clutches with large values of α about 40…45º to decrease dynamic loads in driving, because in little and middle angles α = 15…25º provide nosedive dropping of final stage torque

Te which can become a source of intense oscillations in the driving equipped with clutch in overload mode.

4. Grooves sides to radius inclination angle β have the minor impact on clutch elastic torque variation in overload mode. That's why its value can be selected taking into account technologic features.
5. It is confirmed that the ball's diameter to its center arrangement diameter ratio has no influence on the final stage clutch elastic torque changing nature. The balls diameter increasing provides only clutch load capacity magnification.

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Declaration of competing interest: *The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.*

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