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Vladyslav Protsenko[1], Volodymyr Malashchenko[2], Sylwester Kłysz[3], Oleksiy Avramenko[4]

[1] Department of Transport Technologies and Mechanical Engineering Department, Kherson State Maritime Academy, Ushakova ave. 20, Kherson, Ukraine, e-mail: 1904pvo@gmail.com, ORCID: 0000-0002-3468-4952
[2] Department of Technical Mechanics and Machine Dynamics, Lviv Polytechnic National University, 12, S. Bandery Str., Lviv, Ukraine, E-mail: volod.malash@gmail.com, ORCID 0000-0001-7889-7303
[3] Airworthiness Division, Air Force Institute of Technology, 6, Księcia Bolesława Str., Warsaw, Poland, e-mail: sylwesterklysz3@gmail.com, ORCID 0000-0002-4521-4972
[4] Department of Transport Technologies and Mechanical Engineering Department, Kherson State Maritime Academy, Ushakova ave. 20, Ukraine, Kherson, e-mail: nathblabird@gmail.com

SAFETY-OVERRUNNING BALL-TYPE CLUTCH PARTS CONTACT INTERACTION FEATURES

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Abstract. The article deals with the field of machinery, namely, with the protecting of devices for mechanical driving systems. Safety-overrunning clutches, operating on gearing principle, where safety and overrunning parts are mutually integrated, are perspectives for the building based on modular machines. This case is due to their compactness and low components, comparatively with combined constructions.New clutch design is investigated insufficiently. Particularly their calculation methods, namely parts contact stresses determination, developed deficiently. For ball-type overrunning clutches, contact strength calculations are well-developed, but its transference on new construction safety-overrunning clutches is impossible because of the difference between parts contact interaction in those clutches. The aims of the article are: to analyze created by authors safety-overrunning ball-type clutch parts contact interaction features; to propose on its base clutch construction improvement which could provide parts contact stresses minimization out of dependence with clutch manufacturing and assembling accuracy; taking into account Hertz contact interaction theory results, to obtain expressions for determining clutch parts loads and contact stresses. It is established that using safety-overrunning clutch grooves parallel to radius side surfaces is inexpedient. This can increase balls and internal semi-coupling grooves edge contact and significant contact stresses. To exclude the impact of clutch parts manufacturing and assembling accuracy on contact stresses in paper, proposed to incline grooves side surfaces at an angle to the semi-couplings radius, passing through the ball centre in diametric section. Comparatively,contact grooves inclination to radius with the edge allows decreasing contact stresses in 45–55 times. Further investigations should be focused on force parameters and operating characteristics justification for a clutch with inclined grooves proposed in this paper.

Keywords: contact strength, safety-overrunning clutch, torque, groove, edge contact, parts interaction, machine design.

Introduction

Assumptions that include machine parts calculation models often became a reason for increasing safety factors. This method does not always guarantee positive practice results obtainment insofar as parts deformations that cause deviations from nominal sizes often lead to significant changes in the design scheme, which can cause multiply parts stresses increasing that can exceed the specified safety factors.

Problem Statement

Safety-overrunning clutches, operating on gearing principle, where safety and overrunning parts are mutually integrated, are perspectives for the building based on modular machines. This case is due to their compactness and low components, comparatively with combined constructions. At the same time, this clutch design is investigated insufficiently. Particularly their calculation methods, namely parts contact stresses determination, developed deficiently.

Review of Modern Information Sources on the Subject of the Paper

Nowadays, overrunning and safety-overrunning clutches [1, 2] are usable for many types of machines, such as transport diesels and vehicles [3, 4], metal-cutting machines, ball mills [5, 6, 7], press-lines [8], etc. For ball-type overrunning clutches [4], methods of contact strength calculations are well-developed. But those methods transference on new safety-overrunning clutch construction [9] is impossible because of the difference between parts contact interaction in straight overrunning and safety-overrunning clutches.

Objectives and Problems of Research

The aims of the article are following: to analyze created by authors safety-overrunning ball-type clutch [9] parts contact interaction features; to propose on its base clutch construction improvement which could provide parts contact stresses minimization out of dependence with clutch manufacturing and assembling accuracy; taking into account Hertz contact interaction theory results, to obtain expressions for determining clutch parts loads and contact stresses.

Main Material Presentation

In ball-type overrunning clutches, parts strength calculations [4] considered that balls contact with plane-side surfaces of the grooves and accounting provides like for “ball-plane” contactcouple. Mentioned idealized approach does not think that grooves and balls contact takes place at the border of intersection plane side surfaces and external cylindrical surface 1 of internal semi-coupling at the *E* point (Fig. 1, where nominal dimensions are shown with dotted line). In the presence of semi-couplings dimensions deviations, it could become the following. During clutch operation, balls 2 huddling to external coupling internal cylindrical surface 3 with centrifugal force F_{ω}. Due to departures ES_B of groove width *B*, Es_h of inner diameter D_h, and ei_b ball diameter *d*, during clutch operation taking place ball centre displacement from point *A* to point A_1. This together with possible internal semi-coupling diameter *D* decreasing in the borders of tolerance by the amount of lower deviation ei_D, provides ball contact not with groove side surface but with the edge at the E_1 point. In comparison, it happens deviation of contact force acting line on the ψ angle, which leads to parts loading with force *N*instead of F_{Nt}. Therefore, in the presence of groove edge rounding (with radius *r*) “ball-plane” contact converted to “ball- cylinder” contact.

The mentioned circumstances are not so critical for ball overrunning clutches because contact with closed groove arc edge provides its linear connection, which is confirmed by tests (Fig. 2). In the new safety-overrunning clutches [9] balls contact with straight edges of internal and external semi-couplings and end of the disk 4, respectively at the points *E*, *K* and *M* (Fig. 1).

Therefore for safety-overrunning clutches, in which plane side surfaces are parallel to the radius in the diametric section, mentioned circumstances leads to multiple increasing of parts contact stresses (around point *E*), which could be determined by Eq. (1) [10].

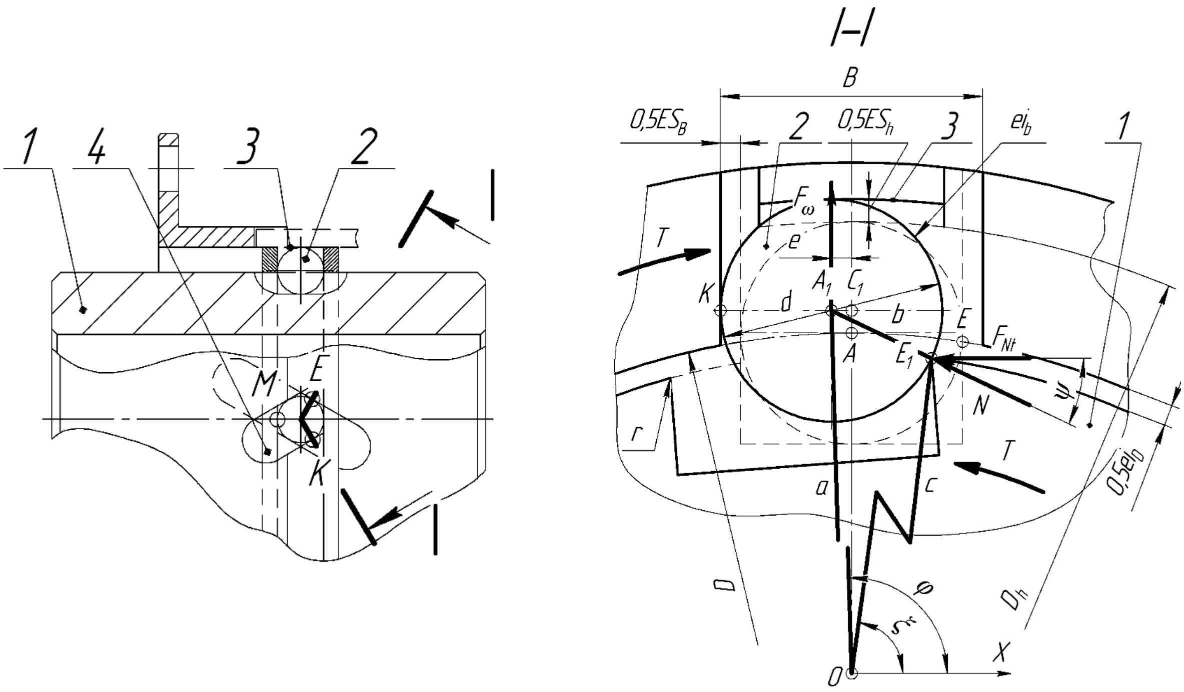
$$\sigma_H = Z_M \sqrt[3]{0.25N\left[\frac{\rho_1+\rho_2}{\rho_1\rho_2}\right]^2}, \tag{1}$$

where Z_M = 1755 MPa$^{2/3}$ – coefficient which takes into account parts mechanical properties (for steel parts); *N*–normal pressure force; $\rho_1 = 0.5d$ – ball radius; $\rho_2 = r$ – groove edge rounding radius in diametric section.

After substitution of known expressions in Eq.1, we obtain its more usable form Eq. 2.

$$\sigma_H = Z_M \sqrt[3]{\frac{2T\left(0.5d + r\right)^2}{zDd^2r^2\cos\alpha\cos\psi}}, \tag{2}$$

where T – torque; z –balls number in the clutch; D – balls centres location diameter; α – grooves to clutch axe inclination angle; ψ– normal force N deviation angle.



a b

Fig. 1. Parts interaction *(a)* and geometric calculation *(b)* schemes for safety-over running clutch

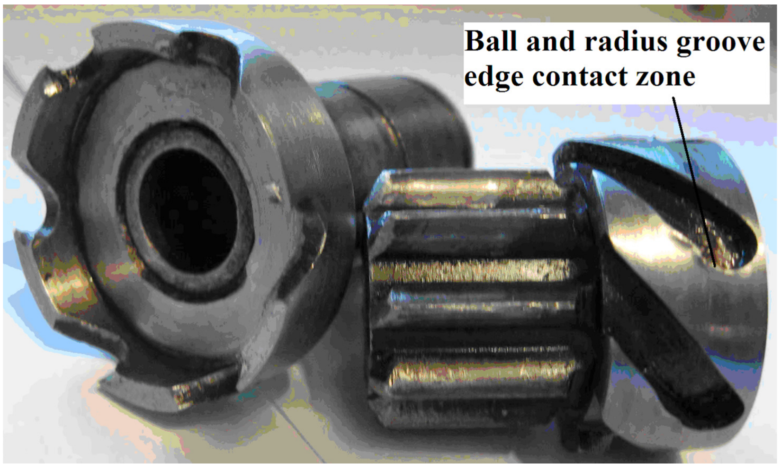


Fig. 2. Photograph of balls and groove radius edge contact zone in over running clutch semi-coupling

For contact stresses magnitude evaluation, it is necessary to determinate cos ψ, which needs some geometric calculations. For this purpose, we use the locked vector circuit's method [11]. For circuit *abc* equation will be following

$$a + b = c. \tag{3}$$

In projections into X axis will be

$$a\sin(90-\varphi)+b\cos\psi = c\cos\xi\,, \tag{4}$$

where from

$$\cos\psi = \frac{c\cos\xi - a\cos\varphi}{b}\,. \tag{5}$$

For other angles, we will have

$$\Delta OA_1E_1 : \xi = \varphi - \arccos\frac{a^2+c^2-b^2}{b}\,; \tag{6}$$

$$\Delta OC_1A_1 : \varphi = 90 + \arccos\frac{2a^2-e^2}{2a^2}\,. \tag{7}$$

For other dimensions following expressions are valid

$$a = OA_1 = OC_1 = OA + OC = 0.5(D + ES_h + ei_b)\,; \tag{8}$$

$$b = A_1B_1 = 0.5(d - ei_b)\,; \tag{9}$$

$$c = OB_1 = 0.5(D - ei_D)\,; \tag{10}$$

$$e = A_1C_1 = 0.5ES_B\,. \tag{11}$$

For mentioned factors on contact stresses effect illustration, calculations for the clutch with following main data have been performed: D = 58 h10($_{-0.12}$), d = 9.128 (±0.06), B = 9.128 H12($^{+0.15}$), D_h = 67.128 H11($^{+0.19}$), z = 8, α = 30º, T = 1.26 N·m. Taking into account that valid dimensions deviations are in the centre of tolerance fields (according to the law of normal distribution), in calculations been used its following values : ei_D = 0.060 мм, ei_b = 0, ES_B = 0.075 мм, ES_h = 0.095 мм.

Graph, shown in Fig. 3, demonstrates that acceptable contact stress values could be achieved in grooves edge radius values more than 0.8 mm. Edge rounding could be fulfilled only manually on practice, and its quality depends on performer professionalism. Therefore, the safety-overrunning clutch should be used such constructive methods, which could contact stresses value minimize out of dependence with manufacturing and assembling accuracy.

One of such methods is proposed in [4] rounded performance of grooves surface in perpendicular section. But this variant, firstly, is too complicated technologically because it is necessary to use profile cutters to manufacture and secondary it could become the reason for multiply normal force increasing (like in roller overrunning clutch [12]).

Another method to decrease contact stresses and exclude edge contact could be implemented through grooves side surfaces inclination at an angle $\beta$to the semi-couplings radius, passing through the ball centre in diametric section (Fig. 4).

Construction with inclined to radius grooves could bring the following advantages:

– ensuring balls contact with plane grooves side surfaces out of dependence on manufacturing and assembling accuracy;

– excluding the necessity of grooves edges rounding;

– excluding balls contact with internal surface 3 (in Fig. 4 is shown the gap between them), which could reduce friction in clutch overload operation period and reduce operating torque.

Using Fig. 4 scheme, it is possible to make up the following expressions for normal pressure forces

$$N_1 = N_{t1} = \frac{F_{Nt1}}{\cos\beta} = \frac{F_t}{\cos\alpha} = \frac{2T}{zD\cos\alpha\cos\beta}\,; \tag{12}$$

$$N_2 = N_{t2} + N_{\omega 2} = \frac{F_{Nt2}}{\cos\beta} + F_\omega\sin\beta = \frac{2T}{zD\cos\alpha\cos\beta} + F_\omega\sin\beta\,. \tag{13}$$

In Eq. 13 centrifugal component $F_\omega\sin\beta$ is only 6 % of normal force N_2 (in rotation speed 1500 rpm), making it possible to evaluate clutch parts strength by force N_1 (Eq. 12).

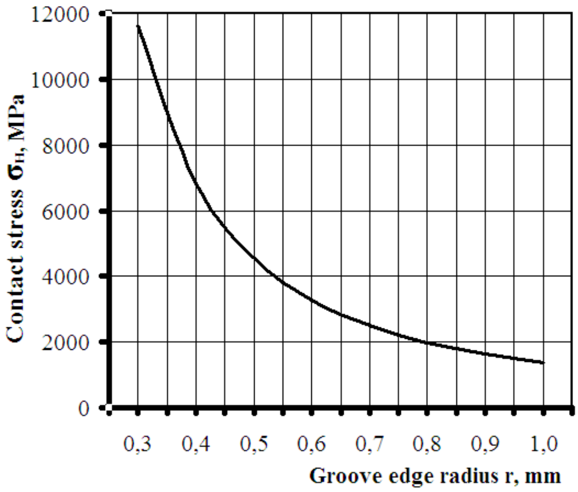


Fig. 3. Contact stresses value dependence on internal semi-coupling groove edge radius

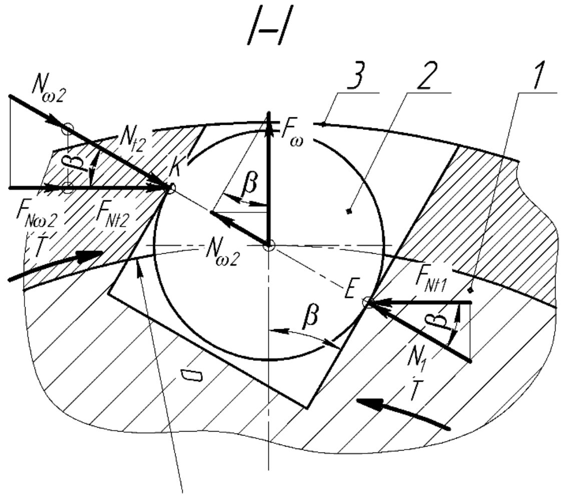


Fig. 4. Force diagram for coupling with edge inclined to radius

It makes possible to write on the base of Eq. 1 further expression for contact stresses evaluation

$$\sigma_H = Z_M \sqrt[3]{\frac{2T}{zDd^2 \cos\alpha \cos\beta}}\,. \tag{14}$$

Fig. 5 shows the graph of angle β impact on contact stresses around E point. This graph for a clutch with mentioned data can state the following:

– grooves inclination at an angle β to radius provides balls contact with plane-side grooves surfaces and through this significant contact strength out of dependence with manufacturing and assembling accuracy. Comparatively, with the edge, contact grooves inclination to radius allows decreasing contact stresses in 45–55 times (from 2000–2500 MPa to 44–45 MPa);

– angle β value has a minor impact on contact stresses magnitude – when magnified by 15 times, contact stresses increase by only 3.5 % (from 44.04 MPa to 45.58 MPa). Therefore the source of information for deciding on the angle β should not be the calculation of parts for contact strength, but other considerations, such as operation accuracy or load capacity.

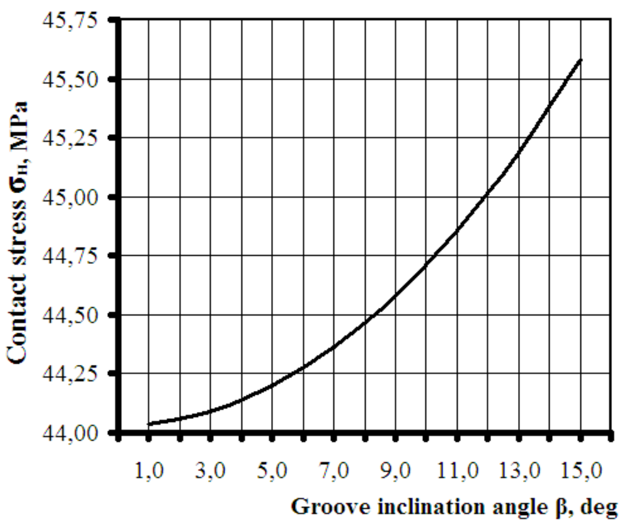


Fig. 5. Contact stresses value dependence on groove sinclination angle to semi-couplings radius

Conclusions

1. It is established that using in safety-overrunning clutch grooves with parallel to radius side surfaces is inadvisable as this can lead to balls and internal semi-coupling grooves edge contact and significant contact stresses increasing;

2. To exclude the impact of clutch parts manufacturing and assembling accuracy on contact stresses in paper proposed to incline grooves side surfaces at an angle β to the semi-couplings radius, passing through the ball centre in diametric section. Comparatively, with edge contact grooves inclination to radius allows decreasing contact stresses in 45–55 times (from 2000–2500 MPa to 44–45 MPa);

3. It is shown that angle βvalue has a minor impact on contact stresses magnitude – when magnified by 15 times (from 1º to 15º), contact stresses increase by only 3.5 % (from 44.04 MPa to 45.58 MPa).

4. Further investigations should be focused on force parameters and operating characteristics justification for a clutch with inclined grooves proposed in this paper.

References

[1] B. M. Hevko, I. V. Lootsiv, I. B. Hevko, et. al. *"Pruzhno-zapobizhni mufty: konstrukciya, rozrakhunky, doslidzhennya" ["Flexible-safety clutches: construction, calculation, study"]*. Ternopil, Ukraine: Publishing house "FOP Paljanyca V. A.", 2019. [in Ukrainian].

[2] V. O. Malashchenko, V. M. Strilets, O. R. Strilets, V. A. Fedoruk, *"Novy mufty mekhanichnikh privodiv mashyn" ["New couplings of machine mechanic drivings"]*. Rivne, Ukraine: Publishing house "NUWHP", 2019. [in Ukrainian].

[3] Y. Hanqi, Z. Chenyi, G. Bingzhao, "Fork-less two-speed I-AMT with overrunning clutch for light electric vehicle", *Mechanism and Machine Theory*. vol. 130. pp. 157–169, 2018. https://doi.org/ 10.1016/j.mechmachtheory.2018.08.019

[4] V. O. Malashchenko, P. Hashchuk, O. Sorokivsky, V. V. Malashchenko, *"Kulkovy mekhanyzmy vilnogo khodu" ["Ball-type freewheel mechanisms"]*. Lviv, Ukraine: Novy Svit-2000, 2012. [in Ukrainian].

[5] V. O. Malashchenko, G. P. Kunytskiy, *"Navantazhuvalna zdatnist velykomasovykh system" ["Load capacity of large-mass systems"]*. Lviv, Ukraine: Novy Svit-2000, 2020. [in Ukrainian].

[6] Y. Kharchenko, L. Dragun "Mathematical modeling of unsteady processes in electromechanical system of ring-ball mill", *Diagnostyka*, vol. 18(1). pp. 25–35, 2017.

[7] A. O. Borys, "Pokrashennya efektivnosti mekhanychyh pryvodiv zastosuvannyam obginno-zapobizhnykh muft" ["Improvement of the efficiency of mechanical drives by theuse of ball-type safety-overrunning cclutches"], Ph.D. dissertation, Lviv Polytechnic National University, Lviv, Ukraine, 2019. [in Ukrainian].

[8] K. Roczek, A. Krol, "Clutch-brake unit – principle of operation and basic diagnostic methods", *Diagnostyka*, vol. 19(1), pp. 33–39, 2018. https://doi.org/10.29354/diag/80975.

[9] V. Malashchenko, V. Protsenko, S. Klysz, V. NastasenkoandM. Babiy,"Force parameters and operating characteristics of safety-overrunning ball-typeclutch", *Diagnostyka*, vol. 22(3). pp. 43–50, 2021., https://doi.org/10.29354/diag/140000

[10] V. T. Pavlishche, Y. V. Kharchenko, A. F. Barvinsky, Y. G. Garshnev, *"Prikladna mekhanika" ["Applied Mechanics"]*. Lviv, Ukraine: Intelect-Zakhid, 2004. [in Ukrainian].

[11] V. A. Zinovjev, *"Teoriya mekhanizmov i mashyn" ["Mechanisms andmachines theory"]*. Moscow, USSR: Mashgiz, 1959. [in Russian].

[12] M. N. Pylipenko, *"Mekhanizmy svobodnogo khoda" ["Freewheel mechanisms"]*. Moscow, USSR: Mashinostoenie, 1966. [in Russian].