УДК 621.825.(075.8) DOI: 10.30977/BUL.2219-5548.2021.95.0.124

# LOADING CAPACITY OF BALL BENDING CLUTCH WITH STRAIGHT SLOTS

**Malachschenko V. O.[1], Orel O. V.[2], Fedik V. V.[3]**
**[1]Lviv Polytechnic National University Department of Machine Elements**
**[2]Kharkiv National Automobile and Road University**
**[3]Drohobych College of Oil and Gas**

***Abstract.*** *Based on previous research results, a more advanced design of ball coupling of freewheel (BCFH) of axial action for starters of internal combustion engines is developed, force interaction and maximum torque for the case when the grooves of one half-clutch are straight, which simplifies manufacturing technology. Calculation schemes for different positions of the main elements are offered, loading of all working surfaces of the new coupling taking into account friction of balls with lateral surfaces of grooves is established.*
***Key words:*** *ball coupling, straight grooves, coupling, ball friction, bypass coupling.*

## Introduction

Freewheel is known to be widely used in various vehicles to automatically connect and disengage shafts and transmit torque in only one direction. In such cases, overtaking roller couplings are now used, which have a number of disadvantages, and the limitation of the strength of the elements in the engagement and the amount of torque due to the sliding of the rollers relative to the working surface of the drum is the most significant. This phenomenon leads to the intensive operation of freewheel mechanisms, the repair operations of which are often difficult. Analyzing the existing literature sources, we conclude that these couplings also have strict technological and design requirements. Therefore, when possible, they are replaced by ratchet mechanisms, which also have a number of significant disadvantages. This is primarily a loud noise at idle, intense idling, and so on. This situation creates urgent problems aimed at developing advanced mechanical means. Such mechanisms have been developed as freewheel ball clutches or ball overrunning clutches. But the design and their further improvement requires a detailed analysis of the principle of operation, a thorough theoretical study to obtain mathematical dependencies that describe the basic kinematic and force parameters. Such mathematical expressions are needed to calculate the time of connection and disconnection of the ends of the drive shafts of machines.

## Analysis of publications

It is known that mechanical couplings are widely used in drives of machines and mechanisms. Therefore, currently developed and conducted research on their design features, load capacity, developed appropriate standards, etc. [1, 2, 4]. A number of new freewheel ball couplings have been developed and patented [3, 5, 11–13, 15], their design, main advantages, areas of application, principle of operation, etc. are described. Many experimental researches on practical definition of their operational indicators are carried out. A special place is now occupied by new type couplings, which are allocated to a separate subclass and are called ball overtaking radial and axial action. These types of couplings are also constantly improved based on their existing technical characteristics [6–10,14]. Such and other scientific works are basic for a new ball overrunning clutch, in which the grooves of one of the half-couplings are made along its generating outer surface, or they are parallel to the geometric axis of the shaft.

## Purpose and task statement

The purpose of this article is to develop calculation schemes for the study of kinematic and force parameters of new ball bypass couplings of simplified design by performing parallel to the axis of the shaft grooves of one of the half-couplings. The results of the above studies are briefly described in this article.

## Building a research system

Based on the results of improving the process of clutch ends of ball bypass couplings and known previous experiments [5, 6, 9], a new free axial action of the ball was developed and its kinematics and force parameters were determined for the case when the working grooves of the driven clutch are parallel to the clutch axis.

The new principle of adhesion (Fig. 1) is similar to the existing prototype. It consists of: 1 – leading half-coupling, which has a flange 2 with grooves 3; 4 – driven half-coupling with a cylindrical surface 5, which is also made of grooves 6; 7 – balls placed in these grooves, 8 – rings resting on the spring 9; 10 – housing and retaining ring 11. The drive half-clutch has teeth 12 for connecting it to the motor drive. The setting dimensions of the clutch are consistent with the parameters of the spherical shaft of the serial starter and its drive transmission.

The manufacture of grooves parallel to the axis of the shaft is the main advantage of the new ball bypass coupling, which significantly simplifies its manufacture. Therefore, there is an urgent need to conduct these studies of the operability of a mechanical drive with such a new clutch. Its more significant advantage is in comparison with splints with motor rotor roller of the basic internal combustion engine starter that transmits torque to it. Balls 7 that were previously situated in grooves of driven half sleeve 4 begin to roll due to springs 9 in curved grooves 3 that are already moving. As a result, coupling starts to rotate as a single unit and acquires a constant angular velocity, i.e. the coupling goes to working condition. Before starting the driven half sleeve, pulling relay brings clutch to the flywheel on crankshaft of the internal combustion engine where the drive gear 12 meshing with the flywheel teeth crown. After that the main engine starts. Once the engine is running, the speed of rotation of the crankshaft equals to 84...136 rad/s, which is significantly more than the speed of starter rotor rotation, and so the drive gear immediately disconnects from the flywheel crown followed by balls rolling on inclined surfaces of the grooves back to the grooves of driven half sleeve 4. Clutch disconnects and switches to freewheel mode.

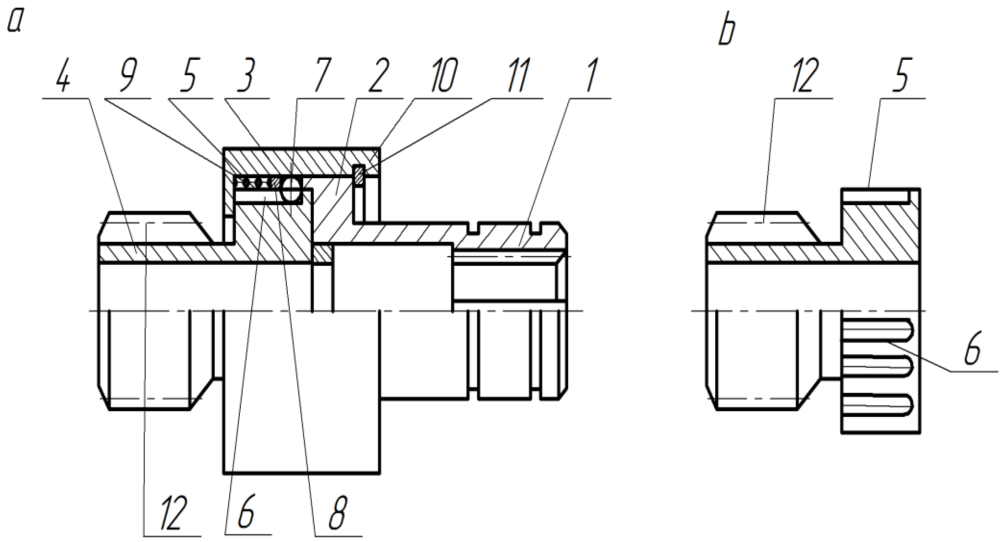


Fig. 1. Ball freewheel for starter of internal combustion engines: a – general view of the clutch; b – driven half sleeve with driving gear and parallel grooves

It is important to note that there are two fundamental modifications of this clutch depending on the design of the coupling and its mode of operation. This is when the balls first move through the grooves, or are completely engaged.

Possible ball position relatively to the grooves half couplings is shown in Fig. 2.

The maximum and minimum turning angle of the clutch is shown in Fig. 3.

Fig. 3 shows that the minimum angular displacement of the half sleeve before cluch switches can be defined as

$$\phi_{\min} = \frac{l_{BC}}{R} \approx \frac{2rtg\alpha}{R}. \quad (1)$$

$$t_{\min} = \frac{\phi_{\min}}{\omega_1} = \frac{2rtg\alpha}{\omega_1 R}. \quad (2)$$

$$(t_{min} \le t_i \le t_{max}).$$

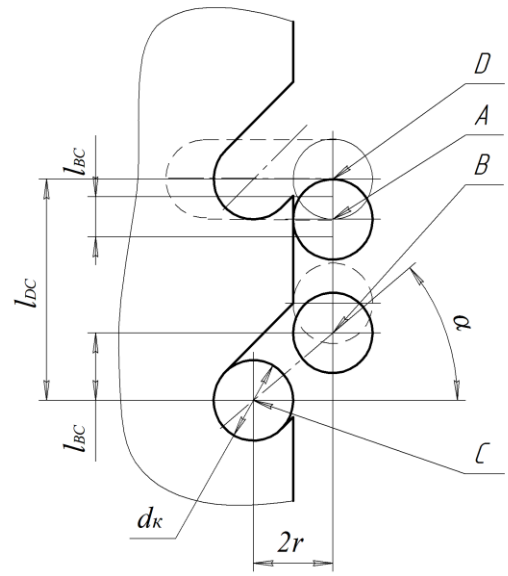


Fig. 2. Variants of ball positioning during the period of clutch switching in

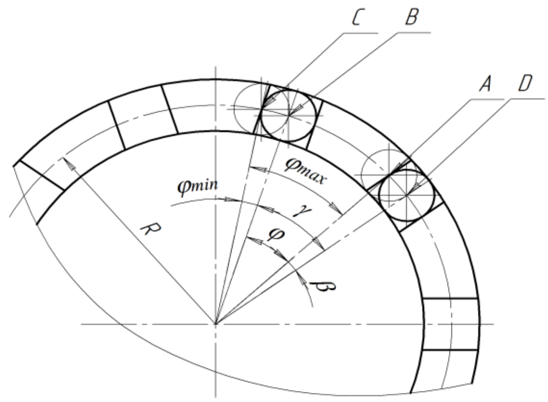


Fig. 3. Frontal view of the clutch, design scheme for angular displacement definition

Assuming that the motion is uniform on the $l_{BC}$ interval, the minimum time to overcome it will be

Expressions for the maximum magnitude of angular displacement and time of relative movement of the grooves and balls is also apparent from Fig. 3: switching are in this range of calculations performed.

$$\phi_{\max} = \frac{2rtg\alpha}{R} + \frac{2\pi}{z} - \frac{r}{R} = \frac{2zrtg\alpha + 2\pi R - rz}{Rz} = \frac{2\pi R + rz(2tg\alpha - 1)}{Rz}; \quad (3)$$

H

ere, $\omega_1$ - is the constant angular velocity of the driving half sleeve, other parameters appearing in (1)–(4) are shown in Fig. 2 and 3.

All other possible positions of balls relatively to the grooves of the half sleeves and various values of angles of clutches

$$t_{\max} = \frac{\phi_{\max}}{\omega_1} = \frac{2\pi R + rz(2tg\alpha - 1)}{\omega_1 Rz}. \quad (4)$$

The fact we changed the direction of grooves in driven half sleeve leads us to the necessity of the power calculation clarification. Fig. 4 shows the clutch operating modes (I, II, III), and Fig. 5 shows the phases of clutch switching considering force interactions on surfaces of balls and half sleeves.

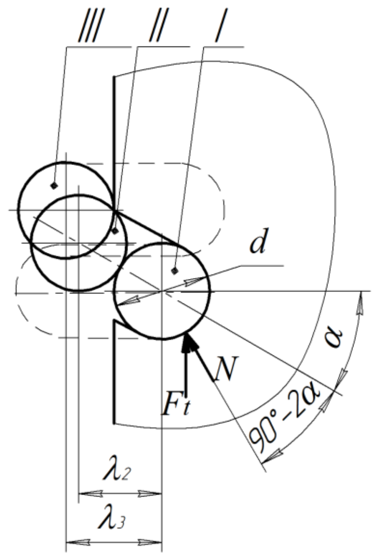


Fig. 4. Specific positions of clutch balls

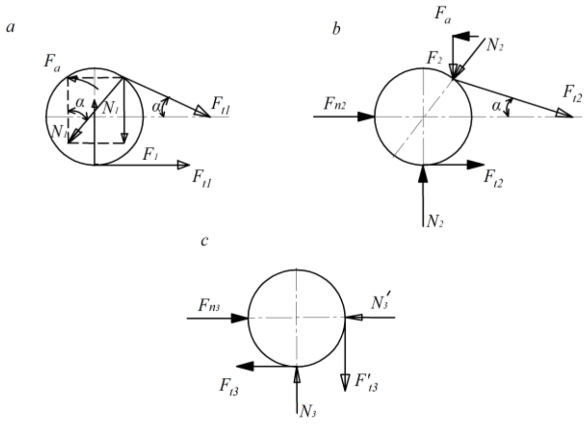


Fig. 5. Phases of clutch switching in: a – operating position; b – beginning of ball's rolling out of the groove; c – the ball have rolled out and being sliding across the end face

Considering the ball balance for working condition (a) of the clutch we obtain

$$F_{t1}(\cos\propto+1)-F_a=0;$$
$$N_1-F_1-F_{t1}\sin\propto=0. \quad (5)$$

For intermediate state (b) of the clutch we get

$$F_{t2}(\cos\propto+1)-F_a-F_{n2}=0;$$
$$N_2-F_2-F_{t2}\sin\propto=0;$$
$$F_{n2}=\frac{Gd\lambda_2}{8c^3 i_l}. \quad (6)$$

Finally, for idling speed of the clutch (c) we have

$$F_{n3}-F_{t3}-N_3^{'}=0;$$
$$N_3-F_{t3}^{'}=0;$$
$$F_{n3}=\frac{Gd\lambda_3}{8c^3 i_p}. \quad (7)$$

Assume that torque on driving half sleeve is sustainable

$$F_1=F_2=T_p/R. \quad (8)$$

Fig. 4 suggests that $\lambda_3=\frac{d_k}{2}$ and $\lambda_2=0{,}8\lambda_3$, i.e. the maximum elastic force of the spring is

$$F_{n\max}=\frac{Gd\alpha_k}{16c^3 i_p}. \quad (9)$$

Here, in expressions (5)–(9) we used following denotations: $N_i, F_{ti}$ – normal pressure and friction arising between the balls and working surfaces of the clutches grooves ($i$ = 1, 2, 3); $\alpha$ – the incline of the grooves, $F_a$ –axial component of the force $F$; $F_n$ – elastic force of the spring; $G$ – modulus of elasticity of the second kind; $d$ – diameter of the wire, $\lambda$ – the axial deformation of the spring, $d_k$ – balls diameter, $c$ - spring characteristics, $i_p$ –the number of working coils, $T_p$ – rated torque, $R$ – the radius of the balls centers circle.

The paper also received the maximum amount of torque that can transfer clutch considering rolling friction between bodies and work surfaces grooves driven half. For certain structural factors of the value determined by of calculations performed

$$T_{\max}=\frac{k_n G d_n^4 \lambda_3 f D_o}{16 D_n^3 i_p(\cos 2\alpha+f\sin\alpha)}. \quad (10)$$

Here, $k_n$ – coefficient of load changes; $d_n$ – wire diameter springs; $f$ – the coefficient of friction $i_p$ – the number of turns of the spring, the rest of the parameters above.

QUANTITATIVE ANALYSIS OF CHANGES IN MAXIMUM LOAD

| $d_n$, мм | 0,5 | 1,5 | 2,0 | 2,5 | 3,0 | 3,5 | 4,0 | 5,0 |
|---|---|---|---|---|---|---|---|---|
| $F_{n\max}$, Н | 0,01 | 1,11 | 4,42 | 8,40 | 18,62 | 31,01 | 58,32 | 138,60 |

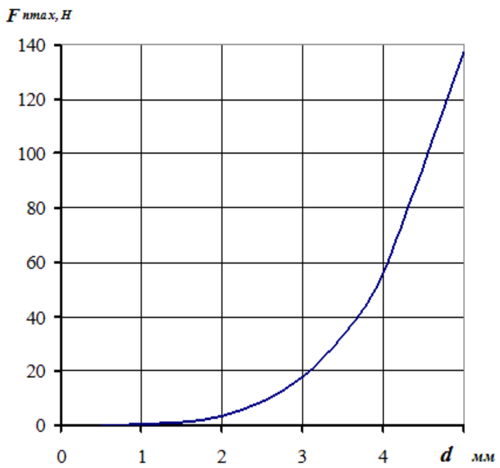


Fig. 6. Changing the elastic force depending on the diameter of the wire

## Conclusions

Analytical expressions we obtained have practical significance. They allow you to set the time of clutch switching on and off (1)–(4) and to determine the force interaction between its elements (5)–(9).

Theoretical studies conducted and the analytical expressions obtained is an essential foundation for further studying of opportunities for overrunning ball clutches application in the vehicle drives with internal combustion engines.

According to the graph (Fig. 6) can easily choose the specific parameters of the spring even during operation of clutch.

## References

1. DSTU 2278 - 93. Mechanical couplings. Terms for definition.
2. Ivanov M.N. Machine parts. - M.: Higher. шк ., 1991. – 383 с.
3. Conical bypass coupling. Patent of Ukraine № 29068A, 1999 / V.O. Malachschenko, P.Ya. Petrenko, OI Sorokovsky.
4. Kulikov N.K. Calculation of freewheel roller mechanisms for strength. Mechanical Engineering, 1952, –№ 2. – P. 19–24.
5. Freewheel ball clutch. Patent of Ukraine № 28884A, 1999 / V.O. Malachschenko, O.I. Sorokovsky.
6. Malachschenko V.O., Drive clutches. Constructions and examples of calculations. - Lviv. NU LP, 2006. – 196 p. and 2009. – 216 p. (2nd edition).
7. Malachschenko V.O., Sorokovsky O.I., Gomishin I. Vol'nobezna gul'ockova spojka.Strojastvo Strojventvi. – 2001. – № 42. – S.56–57.
8. Malachschenko V., Sorokivskij O. The selection of Parametrs of a Coaster Ball Clutch and Recommendations for its Construktion. Transactions of the Universies of Kosise. – 2002. – № 2. – P. 1–6.
9. Malachschenko V.O., Sorokovsky O.I. Load of freewheel ball couplings with concentrated axial force / Mechanical Engineering. – 2002. – № 5. – P. 46–48.
10. Malachschenko V.O., Kravets I.E., Sorokovsky O.I. Power interaction between elements of freewheel ball couplings / Bulletin of Lviv Polytechnic National University. – 2003. – № 480. – P. 73.
11. Bypass clutch. A. S. № 1423838, BI № 34, 1988 / S.G. Kalinin, V.O. Malachschenko, P.Ya. Petrenko.
12. Bypass coupling. A.S. № 1423839, BI № 34, 1988 / S.G. Kalinin, V.O. Malachschenko, P.Ya. Petrenko.
13. Bypass coupling. AC. № 1590747, BI № 16, 1990 / S.G. Kalinin, V.O. Malachschenko, P.Ya. Petrenko.
14. V.O. Malachschenko, E.S. Wenzel, A.O. Boris. Possibilities of performance of safety functions by the ball overtaking coupling of axial action. - Kharkiv. Bulletin of the National Technical University "KhPI", № 1, 2020. - P. 61-65.
15. Bypass coupling. A.S. № 1691625, BI № 42, 1991 / S.G. Kalinin, V.O. Malachschenko, O.I. Sorokivskij.



**Malachschenko V.O.**[1] - Doctor of Technical Sciences, Professor, Lvov Polytechnic National University, Head of the Department of Technical Mechanics and Dynamics of Machines,
Lviv, 32, S. Bandery Street, room 305,
tel. 032-2580678604504, volod.malash@gmail.com
**Orel O.V.**[2] Kharkiv National Road University, st. Yaroslav the Wise, 25, 61002, Kharkiv,
tel. (057) 707-36-58, oav1980@gmail.com
**Fedik V.V.**[3] - Ph.D., lecturer Highest category, Drohobych College of Oil and Gas, Drohobych, street Hrushevskoho, 52, tel. 0673416091, fedikvasil82 @ gmail.com


**Навантажувальна здатність муфти з прямими пазами однієї півмуфти**


***Анотація.*** *На основі попередніх результатів досліджень розроблено більш досконалу конструкцію кулькової муфти вільного ходу осьової дії для стартерів двигунів внутрішнього згоряння, визначено силову взаємодію та максимальний обертальний момент для випадку, коли пази однієї півмуфти є прямими, що спрощує технологію виготовлення. Запропоновано розрахункові схемі для різних положень основних елементів, установлено навантаження всіх робочих поверхонь нової муфти з урахуванням тертя кульок із боковими поверхнями пазів.*



***Ключові слова:*** *кулькова муфта, прямі пази, муфта, тертя кульок, обгінна муфта.*



**Малащенко В.О.**[1] д.т.н., професор, Національний університет «Львівська політехніка», завідувач кафедри «Технічна механіка та динаміка машин»,
м. Львів, вул. С. Бандери, 32, кім. 305,
тел. 032-2580678604504,
volod.malash@gmail.com
**Орел О.В.**[2], к.т.н., доцент, Харківський національний автомобільно-дорожній університет, м. Харків, вул. Ярослава Мудрого,25. oav1980@gmail.com, (057) 707-36-58
**Федик В.В.,**[3] к.т.н., викл. вищої категорії, Дрогобицький коледж нафти і газу, м. Дрогобич, вул. Грушевського, 52, тел. 0673416091, fedikvasil82@ gmail.com.