УДК.621.825.(075.8)

V.O. Malaschenko, V.V. Fedik A.O. Borys
Lviv Polytechnic National University
Department of Machine Elements

FORCE INTERACTION IN THE ELEMENTS OF CLUTCH WITH PARALLEL GROOVES IN DRIVEN HALF SLEEVE

На основі попередніх досліджень розроблено досконалішу конструкцію кулькової муфти вільного ходу КМВХ осьової дії для стартерів двигунів внутрішнього згоряння, визначено силову взаємодію та максимальний обертальний момент для випадку, коли пази однієї півмуфти прямі, що спрощує технологію виготовлення.

Based on previous research, developed a more advanced design of ball freewheel for starters centerline of internal combustion engines, power-defined interaction for the case with parallel grooves and the maximum torque in a half sleeve, which simplifies manufacturing technology.

Statement of the problem. Freewheel are widely used in various vehicles for automatic connecting and disconnecting of shafts and torque transmitting in one direction only. Traditionally, overrunning roller clutches are used in such cases. However, they have several disadvantages and the limitation of the clutch durability and the torque magnitude due to rollers sliding relatively to the drum is most significant. This phenomena induce the improvement of freewheel mechanisms.

Analysis of previous research. Nowadays, there is a tendency to use roller freewheel for automatic connecting and disconnecting of shafts. As you know, these couplings have stringent technological and constructive requirements; therefore they are being replaced by ratchet mechanisms when possible, which also a number of following disadvantages: a loud noise at idle speed, intense backstops wear etc. Therefore, the problems directed towards the creation of advanced mechanical means (ball freewheels) are relevant. However, the designing and their further improvement requires a detailed analysis of their work, thorough research and obtaining of mathematical dependences between kinematic and power parameters, coupling switching time calculation.

Formulation of article objective. The purpose of the article, and, at this stage, the whole research of ball freewheel of axial action with parallel grooves on the powered half sleeve is the research of kinematic and power parameters of these devices, and the timing of clutch switching.

Some results of the research mentioned above are briefly described in this article.

The main material. Based on the improvement of overrunning clutches and previous experiments [5, 6, 9] the authors have developed a completely new ball freewheel of axial action and identified its kinematic and power parameters for the case when working grooves of driven half sleeve are made parallel to the axis of rotation of the clutch.

The suggested clutch (Fig. 1) is similar to the existing prototype and consists of: 1 – driving half sleeve that has a flange 2 with grooves 3; 4 – driven half sleeve with a cylindrical surface 5 with grooves 6; 7 – balls placed in these grooves, 8 - screw ring which rests on the spring 9, 10 – case of locking ring 11. Driven half sleeve includes a drive gear 12.

Installation dimensions of the clutch are concerted with sizes of splined shaft of the serial starter and its driving gear.

The main advantage of the new ball-overrunning clutch to existing technology is a significant simplification of its driven half sleeve manufacturing by cutting parallel grooves into the cylindrical surface. More significant is its advantage compared with overrunning roller clutches that are difficult to manufacture and require high accuracy in the assembly.

The working principle of such coupling is similar to the well-known [4, 11, 12, 12, 15], i.e. half sleeve 1 is associated through slanted splines 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.

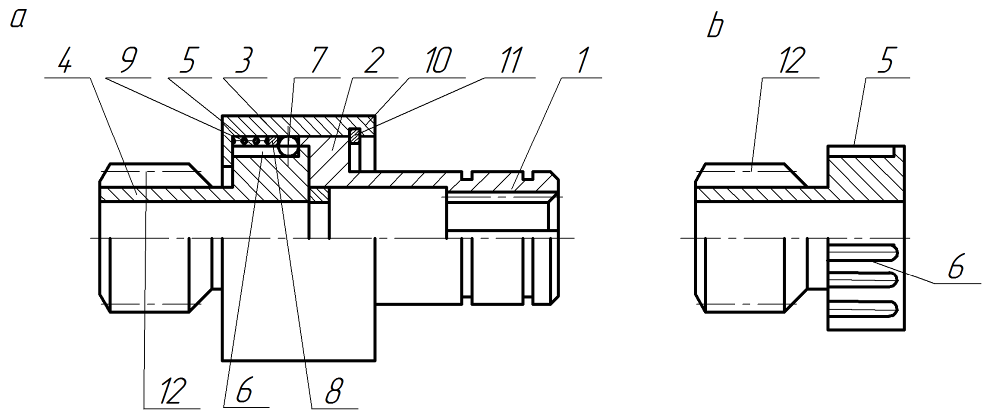

Fig. 1. Ball freewheel for starter of internal combustion engines:
a – general view of the clutch; b – driven half sleeve with driving gear and parallel grooves

One should mention that there are two principal modifications of such clutch depending on their design and operating mode. These are when the balls initially are either in driving or driven half sleeve.

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.

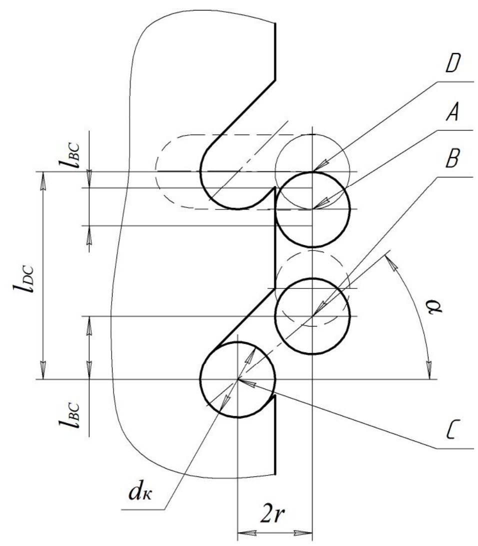

Fig. 2. Variants of ball positioning during the period of clutch switching in

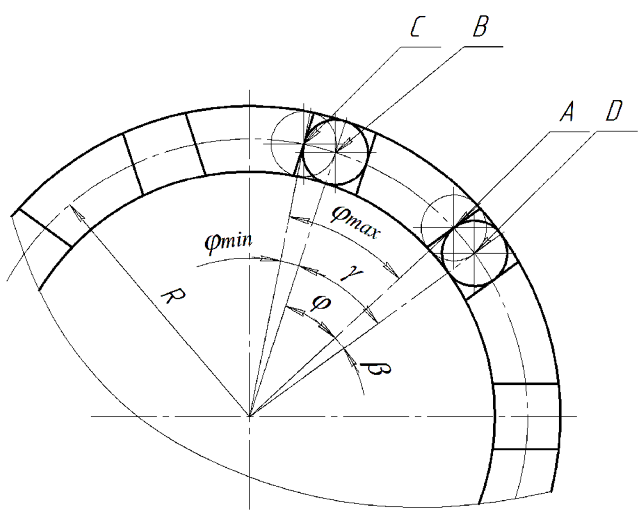

Fig. 3. Frontal view of the clutch, design scheme for angular displacement definition

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

$$j_{\min} = \frac{l_{BC}}{R} \approx \frac{2rtga}{R}\,. \tag{1}$$

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

$$t_{\min} = \frac{j_{\min}}{w_1} = \frac{2rtga}{w_1 R}\,. \tag{2}$$

Expressions for the maximum magnitude of angular displacement and time of relative movement of the grooves and balls is also apparent from Fig. 3:

$$j_{\max} = \frac{2rtga}{R} + \frac{2p}{z} - \frac{r}{R} = \frac{2zrtga + 2pR - rz}{Rz} = \frac{2pR + rz(2tga - 1)}{Rz}; \tag{3}$$

$$t_{\max} = \frac{j_{\max}}{w_1} = \frac{2pR + rz(2tga - 1)}{w_1 Rz}. \tag{4}$$

Here, ω_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 switching are in this range of calculations performed.

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

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.

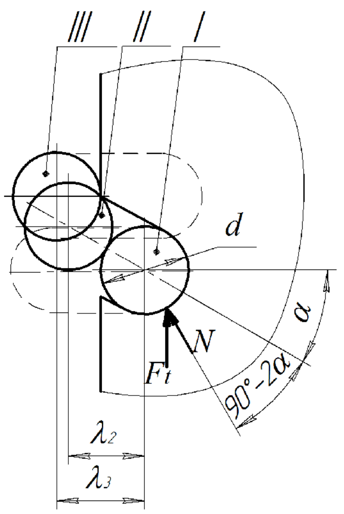

Fig. 4. Specific positions of clutch balls

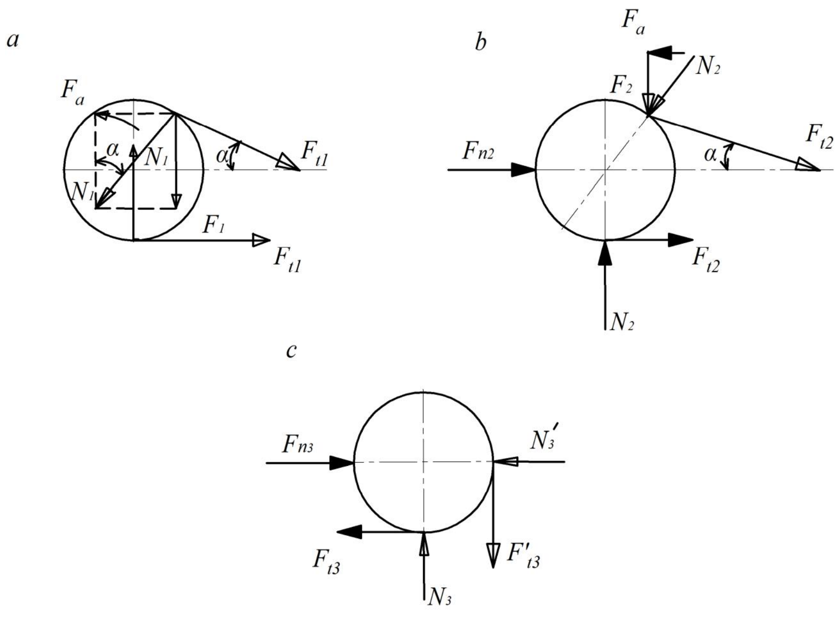

Fig. 5. Phases of clutch switching in: a – operating position; b – beginning of ball's rolling out of the groove; c – the ball have rolled out and being sliding across the end face.

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

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

For intermediate state (b) of the clutch we get

$$F_{t2}(\cos a + 1) - F_a - F_{n2} = 0 \;\; ; \;\; N_2 - F_2 - F_{t2}\sin a = 0 \;\; ; \;\; F_{n2} = {Gd\,l_2}\big/{8c^3 i_l}\,. \tag{6}$$

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

$$F_{n3} - F_{t3} - N_3' = 0 \qquad N_3 - F_{t3}' = 0 \;;\; F_{n3} = {Gd\,l_3}\big/{8c^3 i_p}\,. \tag{7}$$

Assume that torque on driving half sleeve is sustainable

$$F_1 = F_2 = T_p / R\,. \tag{8}$$

Fig. 4 suggests that $l_3 = \dfrac{d_k}{2}$ and $l_2 = 0{,}8\,l_3$, i.e. the maximum elastic force of the spring is

$$F_{n\max} = {Gda_k}\big/{16c^3 i_p}\,. \tag{9}$$

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); a – 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, l – 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

$$T_{\max} = \frac{k_n G d_n^4\, l_3\, f D_o}{16 D_n^3 i_p (\cos 2a + f \sin a)}\,. \tag{11}$$

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
$F_{n\max}$, H	0,01	1,11	3,42	8,40	16,62	31,01	52,32	136,20

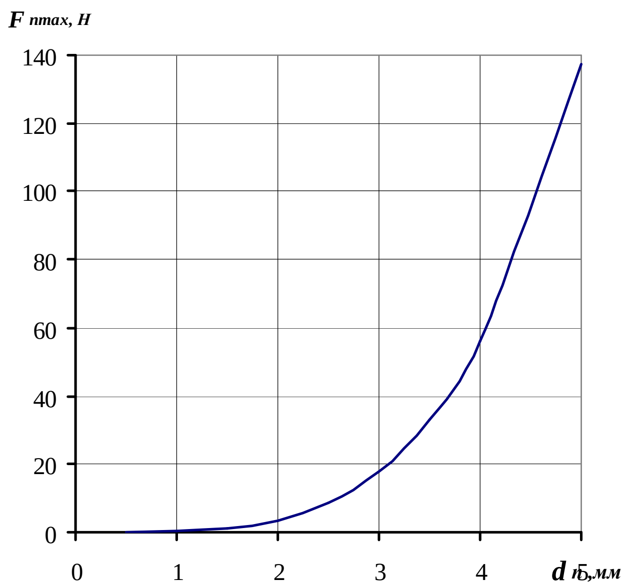

Fig. 6. Changing the elastic force depending on the diameter of the wire

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.

1. *ДСТУ 2278 – 93. Муфти механічні. Терміни та визначення. 2. Іванов М.Н. Детали машин. – М.: Высш. шк.., 1991. – 383 с. 3. Конічна обгінна муфта. Патент України № 29068А, 1999 / В.О. Малащенко, П.Я. Петренко, О.І. Сороківський. 4. Куликов Н.К. Расчет роликовых механизмов свободного хода на прочность // Машиностроение – 1952. – № 2. – С. 19–24. 5. Кулькова муфта вільного ходу. Патент України № 28884А, 1999 / В.О. Малащенко, О.І. Сороківський. 6. Малащенко В.О. Муфти приводів. Конструкції та приклади розрахунків. 2-ге вид. – Львів: НУЛП, 2006. – 196 с.; 2009. – 216 с. 7. Малащенко В.О., Сороківський О.І., Гомішин Я. Vol'nobezna gul'ockova spojka. Strojastvo Strojventvi. – 2001. – № 42. – С. 56–57. 8. Malascenko V., Sorokivskij O. The selection of Parametrs of a Coaster Ball Clutch and Recommendation for its Construction. Transaktions of the Universies of Kosise. – 2002. – № 2. – С. 1–6. 9. Малащенко В.О., Сороківський О.І. Навантаження напівмуфт кулькової муфти вільного ходу зосередженою осьовою силою / Машинознавство. – 2002. – № 5. – С. 46–48. 10. Малащенко В.О., Кравець І.Є., Сороківський О.І. Силова взаємодія між елементами кулькових муфт вільного ходу / Вісник Нац. ун-ту "Львівська політехніка". – 2003. – № 480. – С. 73. 11. Обгінна муфта. А. С. № 1423838, БІ № 34, 1988 / С.Г. Калінін, В.О. Малащенко, П.Я. Петренко. 12. Обгінна муфта. А.С. № 1423839, БІ № 34, 1988 / С.Г. Калінін, В.О. Малащенко, П.Я. Петренко. 13. Обгінна муфта. АС. № 1590747,БІ № 16, 1990 / С.Г. Калінін, В.О. Малащенко, П.Я. Петренко.15. Обгінна муфта. А.С. № 1691625, БІ № 42, 1991 / С.Г. Калінін, В.О. Малащенко, О.І. Сороківський.*