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IMPROVEMENT OF BICYCLE BALL-TYPE OVERRUNNING CLUTCH OPERATION

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Abstract: A comparative analysis of well-known literary sources has been conducted to identify and summarize the main shortcomings of existing freewheel ball clutches used in bicycles. The need to improve the operational characteristics of existing clutches in order to completely eliminate ball slippage during transitional modes of operation has been convincingly demonstrated. This is due to the fact that they transmit torque through the frictional forces between the jammed balls and the working surfaces of the semi-clutches.

KEYWORDS: bicycles; balls; bicycle clutches; surface contact

1 Introduction

Innovative engineering solutions have become essential to carrying out work processes without slippage of contacting elements and significantly improve the load capacity and lifespan of mechanical drives with overrunning ball clutches. These devices improve torque transmission, departing from traditional friction-based methods, and instead employ a sophisticated mechanism where spherical balls engage with specialized grooves in the driving and driven semi-clutches. By eliminating slippage between contacting surfaces, these advancements enhance the overall efficiency of mechanical systems and pave the way for extended durability across a myriad of applications, spanning from bicycles to industrial machinery and beyond. This approach leads to improved longevity and reliability to redefine performance standards.

2 Literature review.

The developed and current DSTU 2278-93 clearly defines the main requirements for various types of mechanical clutches. In the well-known research works [2-14], new overrunning ball clutches are considered at various levels, and their operational capabilities are investigated. A particularly large volume of theoretical and experimental research is devoted to the development of new freewheeling ball clutches (FWBC) for multispeed bicycle rear hubs. An analysis of the main operational parameters, technical project manufacturing, and metal clutch production has been carried out. An analysis of several scientific works dedicated to the development of a new type of rear hubs and their installation on production bicycles, as well as determining the performance of the new clutch under real operating

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conditions, has been conducted [2-14]. To carry out work processes without sliding of contacting elements and to significantly improve the load capacity and service life of mechanical drives with overrunning ball couplings. Such devices transmit the torque not due to friction, but by the engagement of the balls with the working surfaces of the special grooves of the driving and driven half-couplings. This completely eliminates sliding of the contacting surfaces and allows one to avoid the consequences of wear discussed in [15, 16]. In turn, the service life of the freewheel can be significantly increased.

2 Main material presentation.

As emphasized above, the purpose of these studies is to refine the contact surface areas and transmit loads through reliable engagement between the balls and the working surfaces of the semi-clutch grooves. Therefore, there is an urgent need for explanations of the processes involved in creating the working structure of the clutch, which is based on the developed principles of ball engagement located in grooves of various configurations and positions (Figures 1-4 and others).

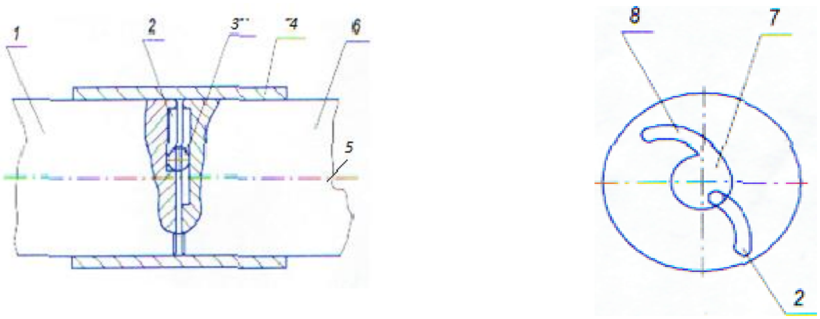


Fig. 1. Schematic diagram of the basic ball overrunning clutch: 1, 6 - shaft ends; 2, 7 and 8 - special grooves; 3 - balls; 4 - centring sleeve.

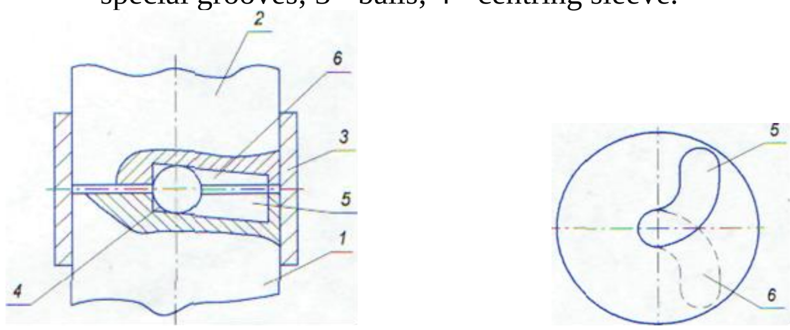


Fig. 2. Ball clutch with inclined grooves: 1, 2 - shaft ends; 3 - sleeve; 4 - ball; 5, 6 - inclined grooves.

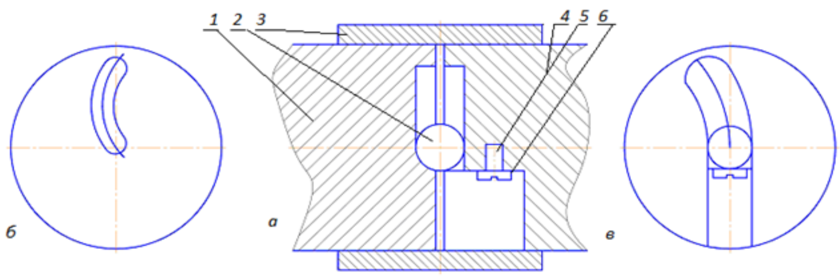


Fig. 3. Ball overrunning clutch with a spring: 1, 4 - shaft ends; 2 - ball; 3 - sleeve; 5 - screw; 6 - spring element.

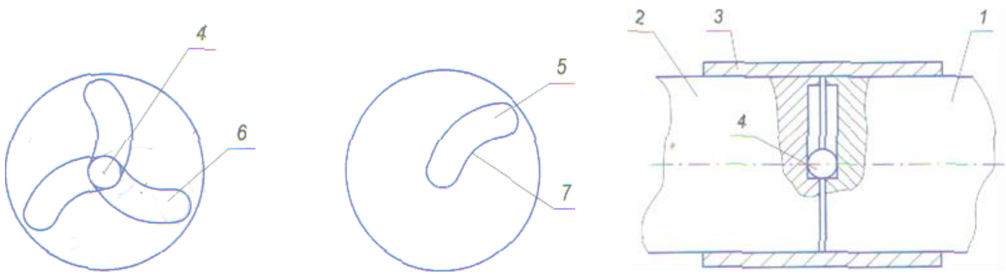


Fig. 4. Ball overrunning clutch with multiple grooves: 1 and 2 - shaft ends; 3 - sleeve; 4 - ball; 5 and 6 - grooves on the shaft ends.

Based on the general principles of connecting semi-clutches using balls (Figures 1-4) and other clutch designs developed by us, which have been patented, we proposed and practically investigated one of the developed clutches on a production bicycle. Here, the main working principle of this clutch is presented. Its explanation is provided using a simplified calculation scheme, which is shown in Figure 5 (where, on the left, a is its general axial cross-section of the simplified calculation scheme, and on the right, b is the view of the driven semi-clutch end in the working state, i.e., when the balls connect the semi-clutches and the clutch transmits the required torque).

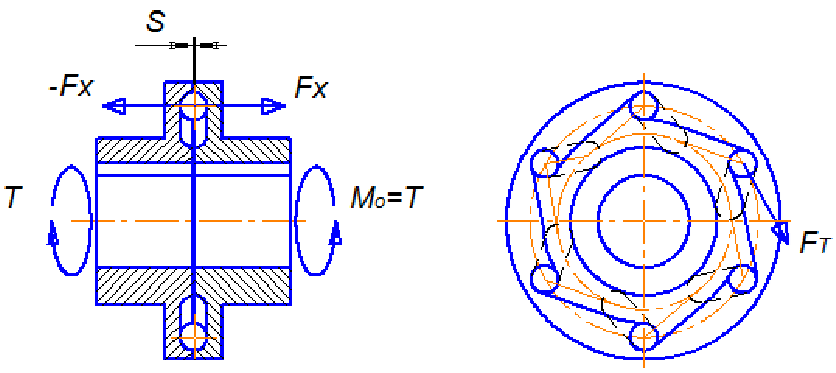
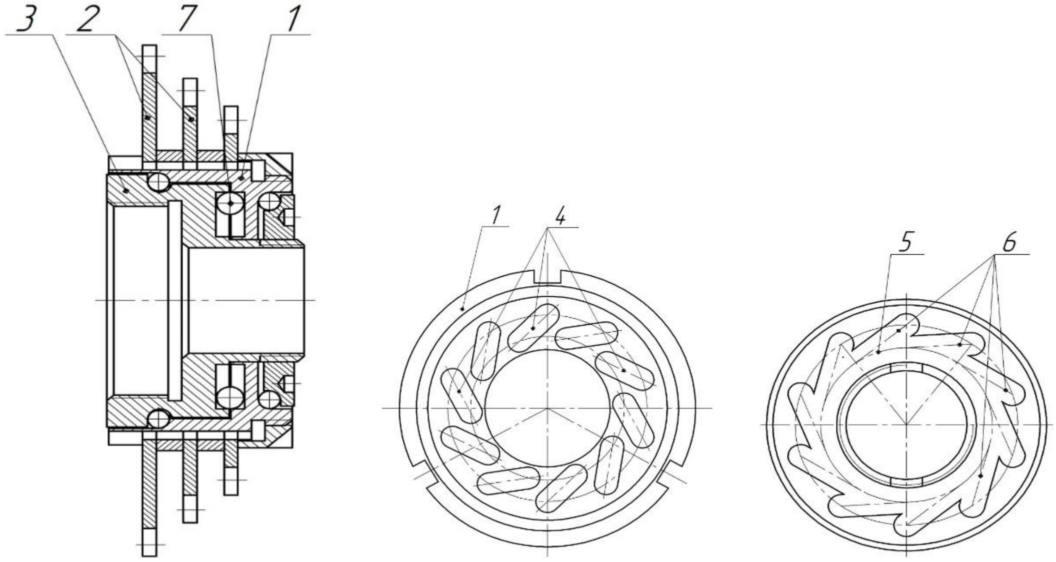


Fig. 5. Ball overrunning clutch with multiple grooves: 1 and 2 - shaft ends; 3 - sleeve; 4 - ball; 5 and 6 - grooves on the shaft ends.

From Figure 5, it is clear that the proposed clutch has certain design features: on the end face of the driving semi-clutch, grooves are made with displacement from the axis of rotation to the periphery of its flange, and on the end face of the driven semi-clutch, a circular groove and tangential grooves are made. All semi-clutch grooves contain balls, the number of which corresponds to the number and dimensions of the tangential grooves.

Using Figure 5, the working principle of such clutches can be explained. Firstly, when the balls are in position b, they transmit torque not through frictional forces between the drum and the star, but due to the engagement of the balls with the working surfaces of the tangential grooves of the semi-clutches. That is, when the driving semi-clutch rotates clockwise (see Figure 5), the balls, under the action of their own weight and centrifugal force, move from the center of rotation of the clutch to the periphery. Now they engage in the tangential grooves of the driven semi-clutch. Initially, there may be cases where not all balls simultaneously engage with the driven semi-clutch grooves; however, the clutch still transmits torque, but the load on the balls is distributed somewhat unevenly. The balls that do not engage with the grooves of the semi-clutch upon engagement quickly move into free tangential grooves, and their load becomes balanced. This is one drawback of such radial action freewheeling ball clutches, which has little impact on the functioning of the bicycle because it quickly disappears.

The positive results of the research on this clutch allowed for the development of a prototype for a three-speed bicycle rear hub, the simplified design of which is shown in Figure 6. This developed design can be applied to other bicycles by increasing the number of stars on their rear hubs. For this purpose, the number of sprockets is 2 (Figure 6) is adjusted to match the number of speeds (shifts).



a b c

Fig. 6. Overrunning ball clutches of the rear hub of the bicycle: a - cross-section of the overall view of the clutch; b - view from the end face of the driving semi-clutch; c - view from the end face of the driven semi-clutch.

The patented clutch (Figure 6) is made of metal, and its operational capability has been investigated. The clutch consists of the following elements: 1 - driven semi-clutch (housing), 2 - three stars, 3 - driving semi-clutch, 4 - inclined grooves on the inner surface of the housing, 5 - circular groove on the end face of the driven semi-clutch, 6 - tangential grooves to the circular groove, 7 – balls.

The operation of the clutch is also explained using Figure 6. Similar to any bicycle, one of the stars 2 (Figure 6, a) is engaged with the chain gear. During the movement of the chain, the working star 2 transmits torque to housing 1 through the splines. In the case of clockwise rotation of housing 1, the ball 7 moves in grooves 4 towards the periphery (Figure 6, b). When reaching the peripheral ends of grooves 4, balls 7 enter the tangential grooves 6 of the sleeve 3 and jam between grooves 4 and 6. The sleeve 3 starts rotating with the housing 1 and transmits torque. In the event of a change in the direction of rotation of the stars 2, or when the angular velocity of the sleeve 3 is greater than the angular velocity of the housing 1, the balls 7 are pushed out from the peripheral position by the side surfaces of the grooves 4 and 6 into the circular groove 5 of the sleeve 3. Then the balls 7 move only along the circular groove 5 and release the kinematic connection between the housing 1 and the sleeve 3. The torque from the pedals will not be transmitted, meaning the clutch enters a freewheel mode, while the driven wheel of the bicycle continues to rotate due to inertia. This occurs during the movement of the bicycle downhill. The functioning of the clutch was investigated during the operation of a production bicycle under nominal load, as well as with significant overload while moving along a test path with specially created obstacles. Field tests fully confirmed the operational capability of this new clutch design (Figure 6). In addition to field tests, a large volume of necessary research on the operation modes of the clutch was theoretically performed. An analysis of the motion laws of the balls relative to the working grooves of the semi-clutches was also conducted, resulting in the derivation of necessary formulas for the design and operation of each component of the clutch. Possible areas of wear of the grooves of the driven semi-clutch of the ball overrunning clutch under significant overloads were determined as well (see Figures 7 and 8).

Fig. 7. End views of the semi-clutches: 1 - driving, 2 - driven.

Experimental studies with triple overloading were conducted to determine the weakest points in terms of strength on the working surface of the semi-clutches. It was found that the most significant deformations during triple overloading are possible at the beginning of the entry of the balls into the grooves of the driven semi-clutch (Figure 7, position 2). This drawback of the device can be easily eliminated by rounding off the corners of the starting working surfaces of the driven semi-clutch (see Figure 7).

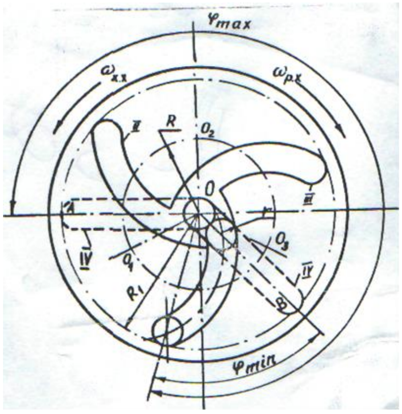
No significant deformation marks were found on the working groove surfaces of the driving semi-clutch. Therefore, it can be concluded that after slight changes to the starting working surface of the driven semi-clutch, the patented clutch will be fully operational.

For a better understanding of the motion transmission process by ball overrunning clutches, theoretical studies of their kinematic parameters have been conducted.

Fig. 8. Overrunning ball clutches of the rear hub of the bicycle: a - cross-section of the overall view of the clutch; b - view from the end face of the driving semi-clutch; c - view from the end face of the driven semi-clutch.

The necessary task for the first stage of theoretical research is to analyze the motion laws of the balls along the curved grooves, that is, to determine the kinematic dependencies describing the motion of the balls during transitional modes of operation of the new overrunning clutch. It is understood that during the engagement of the driving link of the freewheel clutch with the connecting element in the form of a ball, the relative arrangement of the semi-clutch grooves can vary. Among the many possible relative arrangements of the grooves, the most likely case is considered, where the connecting element is positioned at the upper part of the groove (position I, Figure 9). The second extreme possibility occurs when the connecting element, i.e., the ball, is positioned at *O* point of the IV[th] groove.

The first possible engagement of the clutch occurs when the driving link of the clutch rotates by an angle $\varphi_{\min,}$, while for the second case, the engagement of the clutch occurs when



the driving link rotates by an angle $\varphi_{\max.}$ (Figure 9).

Fig. 9. Scheme of possible coincident positions of the driving and driven semi-clutches: IV - the driving semi-clutch is depicted with a dashed groove, while the driven I, II, and III - have visible grooves.

From Figure 9, we can observe the values of the main angles of rotation of the semi-clutches:

$$\varphi_{\min} = \pi/z, \;\; \varphi_{\max} = \pi\left(\frac{5}{4} + \frac{1}{z}\right), \tag{1}$$

where z is an arbitrary number of grooves of the driven link of the clutch.

It should be emphasized that the minimum engagement time of this clutch corresponds to the first case (I), while the maximum engagement time corresponds to case II, where the initial position of the groove is horizontal. Similarly, the analytical schemes can be analysed for any number of grooves on the ends of the driven and driving links of such clutches.

The method for determining the velocity and acceleration of the connecting link is as follows. Assuming that for the first mutual arrangement of grooves, the driving link engages at the time when the connecting element (ball) is at point I (see Figure 9), the linear velocity V_3 will vary from zero at point *O* to its maximum value at point O_3.

In general, this velocity can be expressed by a known relationship of the form:

$$V_{3i} = \varpi_1 \rho_1, \tag{2}$$

where ϖ_1 is the actual value of the angular velocity of the driving link at the moment of acceleration of the system. It changes from zero to its maximum value depending on the characteristics of the driving motor. The law of change ϖ_1 during the acceleration period is selected from the torque function or the angle of rotation.

The angular velocity of the ball relative to its center O will be (see Fig. 9) next:

$$\omega_2 = \frac{\omega_1 \rho_i}{r} \tag{3}$$

If at the initial moment $\varpi_1 = \varpi_0 = 0,$ the mass of the system and the driving torque are constant, then the acceleration time and the angle of rotation will vary according to the following patterns:

$$\frac{d\varphi}{dt} = \sqrt{\frac{2T\varphi}{I}} = \frac{Tt}{I}; \tag{4}$$

$$t = \frac{I}{T} = \sqrt{\frac{2T\varphi}{I}} = \sqrt{\frac{2T\varphi}{T}}; \tag{5}$$

$$\varphi = \frac{Tt^2}{2I} \tag{6}$$

where T is the driving torque, I is the combined moment of inertia of the system, $\varphi, t -$ is the actual value of the angle of rotation of the driving link, and the time of engagement of the clutch. For the provided data, the dependence between the engagement time of the clutch and the geometric dimensions of the balls and grooves is characterized by the values in Table 1.

Table 1. The dependence of the engagement time of the clutch on geometric parameters

R	mm	2	4	6	8	10
$\frac{R+r}{R}$	-	2,00	1,50	1,30	1.25	1,20
T_{eng}	sec	3,14x10^{-3}	4,71x10^{-3}	4,08x10^{-3}	3,92x10^{-3}	3,77x10^{-3}

The findings stemming from our investigation into the engagement time of the ball clutch unequivocally validate its operational efficacy. Notably, our research highlights that the quantity and arrangement of grooves on the working surfaces of the driven half exert a significant influence on the clutch's engagement time. These insights underscore the importance of meticulous design considerations in optimizing clutch performance.

Our study reveals that while the radial-action ball clutch exhibits promising functionality, certain inherent drawbacks impede its widespread adoption. This recognition of limitations serves as a catalyst for driving further enhancements in operational performance. By identifying and addressing these shortcomings, we aim to bolster the applicability and effectiveness of radial-action ball clutches in diverse industrial settings.

The results derived from our investigations into radial-action clutches underscore the imperative of developing freewheeling axial-action ball clutches. This recognition of the need for advancement underscores our team's proactive engagement in addressing progressive challenges within the field. Through ongoing research and development efforts, we are committed to pioneering innovative solutions that push the boundaries of clutch technology, ultimately fostering enhanced performance, reliability, and adaptability across various applications.

CONCLUSION

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