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# Development of Design and Calculation of Frictional Force in Rotational Kinematic Pair of the Fifth Class with Longitudinal Grooves

**A.Djuraev, Sh.Sh.Kenjaboyev, A.Akbarov**

Professor, Department of the Machine and Service, Tashkent Institute of Textile and Light Industry, Tashkent, Uzbekistan
Rector of the Namangan Engineering and Construction Institute, Namangan, Uzbekistan
Teacher, Department of the Technological Machines and Equipment, Namangan Engineering and Construction Institute, Namangan, Uzbekistan

**ABSTRACT**: The results of the numerical solution of the problem for determining the moment of frictional forces in a rotary kinematic pair of the fifth class with longitudinal grooves on the surface of the axis are presented in order to preserve and use the lubricant, which allows an increase in the service life. Based on the analysis of the obtained graphical dependences of the parameters, the necessary values are substantiated.

**KEYWORDS**: Kinematic pair, groove, pin, cylinder, friction, torque, width, length, lubrication.

## I.INTRODUCTION

In the general theory of machines and mechanisms, a kinematic pair is used to refer to the movable connection of two contiguous links. The relative motion of each link of the kinematic pair is subject to restrictions that depend on the way in which the links of the pair are connected. These constraints are called coupling conditions in kinematic pairs [1]. Depending on the imposed restriction of the motion of the links, the kinematic pairs are divided into five classes. If five of the six movements are imposed on the motion of the links forming the kinematic pair (in the space the free link has 6 motions), then this kinematic pair will be of the fifth class. The relative motion of the links forming the kinematic pair can be either rotational or translational.

In a known rotational kinematic pair of the fifth class formed by two cylinders in constant contact. In this case, the bead of the inner cylinder prevents the movement of one cylinder relative to the other in the axial direction, but does not prevent the rotation of one of them relative to the other [1].

The main disadvantage of this design of rotary kinematic pair of the fifth class is a low service life due to significant friction and wear of the kinematic pair elements, especially at high speed operation modes, and at the same time the accuracy of motion decreases and noise appears.

In another known design, a fifth-class rotary kinematic pair comprising a cylinder (shaft) with beads installed in a cylinder (sliding bearing), the shaft rotating relative to the stationary bearing, which constitute a rotational kinematic pair of the fifth class. To increase the service life, there is an opening in the bearing housing, through which a lubricating fluid is supplied, which reduces the friction and wear of the elements (cylindrical surfaces) of the kinematic pair [2, 3, 4, 5]. The disadvantage of this design is a low service life due to insufficient uniform lubrication of the entire area of the kinematic pair elements.

## II. LITERATURE SURVEY

In order to increase the service life and increase the accuracy of the transmission of motion by reducing the frictional force in the kinematic pair of the fifth class, an improved design has been developed.

The structure consists of an inner cylinder (shaft) 1 mounted in a cylindrical hole 2 of the body 3, which are constantly in contact and the cylinder 1 is only able to rotate relative to the fixed cylinder 2 of the housing 3 (Fig. At the same time, their relative movement along the axis of the cylinders 1 and 2 is not due to the execution of the beads 4 of the cylinder (shaft) 1. The cylinder 1 has longitudinal grooves 5 with a certain pitch and depth, having the shape of a part of the circle in cross section. The cylinder 2 of the housing 3 has an opening 6 for supplying the lubricant material to the elements of the kinematic pair.

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The design works as follows. During operation, the cylinder (shaft) 1 rotates relative to the cylinder 2 of the housing 3. The lubricant is supplied through the opening 6 into the elements of the kinematic pair. In this case, the lubricant reduces the friction and thus the wear of the cylinders 1 and 2. The surplus of the lubricant is stored in the grooves 5. Due to the centrifugal forces, the lubricant will constantly supply the friction zone between the cylinders 1 and 2. The barrels 4 of the cylinder 1 eliminate the longitudinal relative movements cylinders 1 and 2. Due to the grooves 5, the contact area between the cylinders 1 and 2 decreases, thereby reducing the friction and wear of the cylinders 1 and 2. The design allows for an increase in the working life of the kinematic pair (hinge).

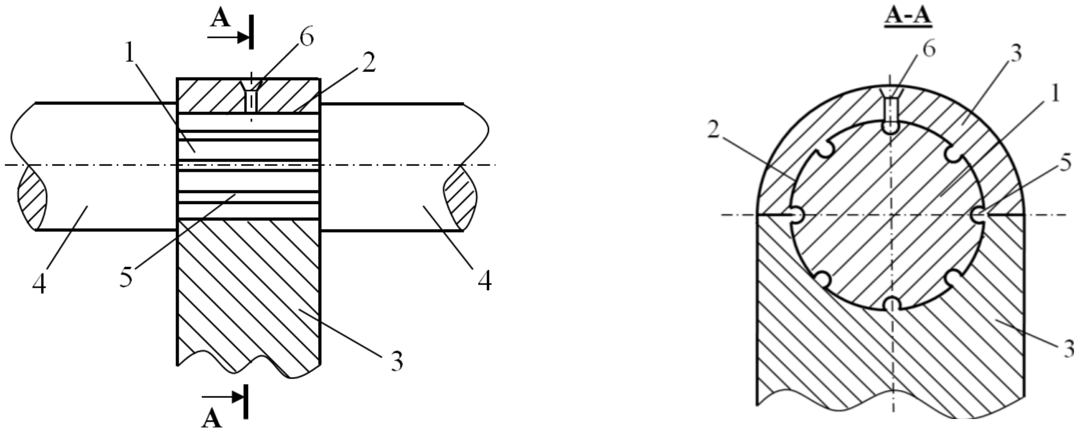


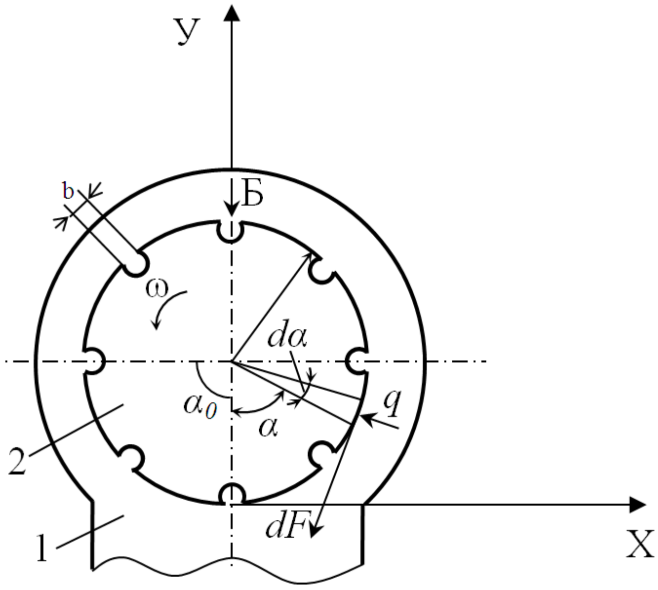
Fig. 1. Rotary kinematic pair of the fifth class with longitudinal grooves of the axis

## III. METHODOLOGY

In the known method for calculating the friction torque in a rotational kinematic pair of the fifth class, two hypotheses are used [1, 6, 7]. We use the first hypothesis according to the calculation scheme shown in Fig. 2, taking into account the equilibrium condition of the pressure force Pk and the projection of the frictional force in the elementary contact zone on the OY axis, one can determine the expression

$$P = 2ql\left[r\sin\left(\frac{\pi}{2}-\frac{i}{2}\arcsin\frac{b}{r}\right)-\frac{ib}{2}\right] \tag{1}$$

where, q-is the pressure force, l- is the length of the kinematic pair, r- is the radius of the cylinder, i -is the number of grooves, and b- is the width of the groove.



1-axis (cylinder), 2-pin (housing)
Fig. 2. Calculation scheme for determining the moment of friction

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It should be noted that the force P of the force of the weight of the body, the spigot on the axis (cylinder), basically operate in the girth zone π.
Taking into account the reaction force of the trunnion on the axis (cylinder), the moment from the frictional force is calculated from the following expression

$$M_{mp} = fPr\frac{\frac{\pi}{2}-\frac{i}{2}\sin\frac{b}{r}}{\sin\left(\frac{\pi}{2}-\frac{i}{2}\sin\frac{b}{r}\right)} \tag{2}$$

Taking $\sin\frac{b}{r}\approx\frac{b}{r}$, and taking into account (1) and coefficient "a" taking into account the degree of lubrication, we obtain

$$M_{mp} = 2fqal\left[r\sin\left(\frac{\pi}{2}-\frac{ib}{2r}\right)-\frac{ib}{2}\right]\cdot\frac{\frac{\pi}{2}-\frac{ib}{2r}}{\sin\left(\frac{\pi}{2}-\frac{ib}{2r}\right)} \tag{3}$$

where, f-coefficient of friction of steel on steel.
For a certain value of the force P, the value q can be calculated from expression (1).
Analysis of formula (2) shows that in the absence of longitudinal grooves on the axis (cylinder), b = 0, then we have

$$M_{mp}^{/} = fPra\frac{\frac{\pi}{2}}{\sin\frac{\pi}{2}} \tag{4}$$

If the angle of contact between the pin and the cylinder occurs at an angle α0, then we have

$$M_{mp}^{//} = fP\,ra\frac{\alpha_0}{\sin\alpha_0} \tag{5}$$

The resulting expression (5) corresponds to the well-known formula given in [1] according to the first hypothesis. It should be noted that expression (5) is a particular case of formula (3).

## IV. RESULTS

The initial data of the parameters of the rotational kinematic pair of the fifth class were chosen for the mechanisms and machines of the textile, light, cotton-ginning industry and agricultural machinery: P=(2,0÷6,0)·$10^2$ N ; l=(1,0÷8,0)·$10^{-2}$ m; r=(1,0÷4,0)·$10^{-2}$ m; f=0,05÷0,1; i=8,0÷16,0; a=0,4÷0,7; b=(0,5÷3,0)·$10^{-3}$ m.
The recommended design of the rotational kinematic pair significantly reduces the torque from the frictional forces, thereby increasing the service life of the mechanism and the machine. At the same time, the design parameters and external load influence the reduction of the torque from the frictional forces. In Fig. Figure 3 shows the graphs of the variation of the torque versus friction forces on the variation of the width of the longitudinal grooves on the surface of the cylinder (axis) of the kinematic pair. It can be seen from them that an increase in the width of the grooves leads to a decrease in Mmr by a nonlinear regularity. This is explained by the fact that with increasing values of "b", the contact area of the surfaces of the kinematic pair also decreases. In this case, an increase in the number of grooves significantly reduces the friction torque. Thus, with an increase in the groove width from 0.5 · $10^{-3}$ m to 3.0 · $10^{-3}$ m at i = 8.0, the friction moment decreases from 30 Nm to 9.9 Nm, and at i = 16.0, the value Mt decreases from 14.5 Nm to 2.9 Nm. For the machines and mechanisms of textile, light industry and agricultural production to increase the resource of kinematics up to 20%, it is recommended to take i = 10,0 ÷ 12,0 and b = (0,8 ÷ 2,0) · $10^{-3}$ m.
It is important to study the influence of the width and radius of the axis of the kinematic pair on the character of the change in the frictional moment (see Fig. 4). The increase in r and l proportionally increases the value of Mtr according to the linear regularity. Therefore, to reduce the friction in the kinematic pair, it is expedient to reduce r and l to possible values.

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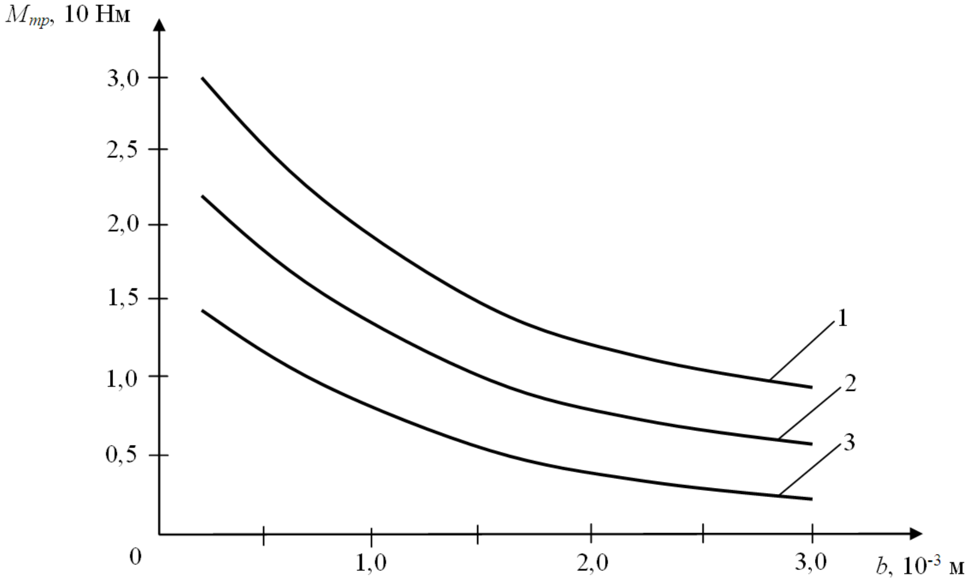
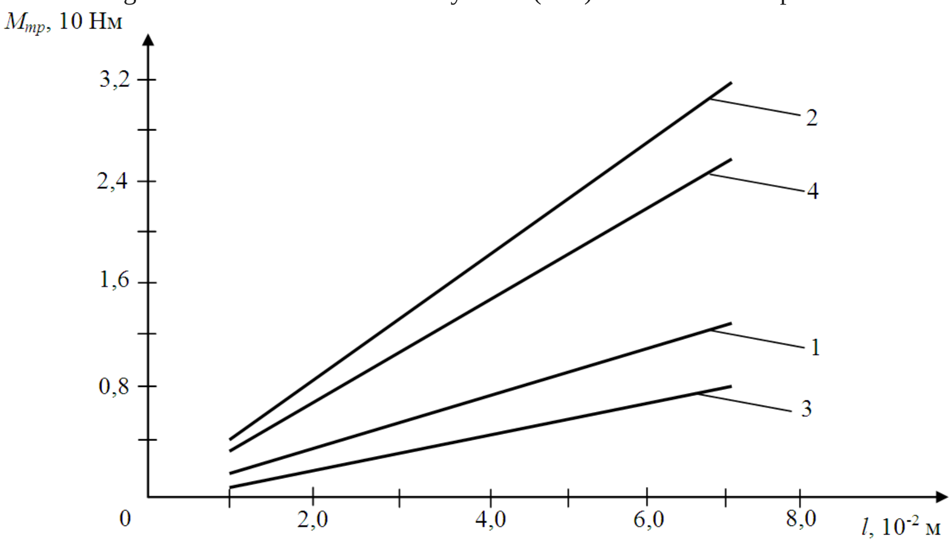



1-at i = 8.0; 2-at i = 12.0; 3-at i = 16.0

Fig. 3. Graphical dependences of the change in the torque of the frictional force on the width of the longitudinal grooves on the surface of the cylinder (axis) of the kinematic pair




1, 3-for the proposed variant; 2, 4-for the existing version

1, 2-at r = 4,0 · 10-2 m; 2-at 1, 2-at r = 3,0 · 10-2 m

Fig. 4. Comparative dependence of the change in the torque on the frictional forces on the change in the length of rotational kinematic pairs

It should be noted that the load P in the main acts on the lower half of the surface of the cylinder, especially when P is formed from the weight of the pin. The greater the load P, the greater the value of Mtr. At the same time, the grooves on the surface of the cylinder not only reduce the contact area of the surfaces of the journal and the cylinder, but also provide a source of conservation and use of the lubricant. The degree of lubrication is taken into account by the coefficient "a". In Fig. Figure 5 shows the graphs of the variation of the frictional moment versus the change in the load P for different degrees of lubrication of rubbing surfaces. Thus, with a change in the external load (1.0 ÷ 8.0) · 102 N, the moment of frictional forces at a = 0.70 increases to 40.5 Nm according to linear regularity, and at a factor of 0.40, Mtr increases only up to 15.4 Nm. Therefore, the recommended lubrication values are a≤ (0,4 ÷ 0,5).

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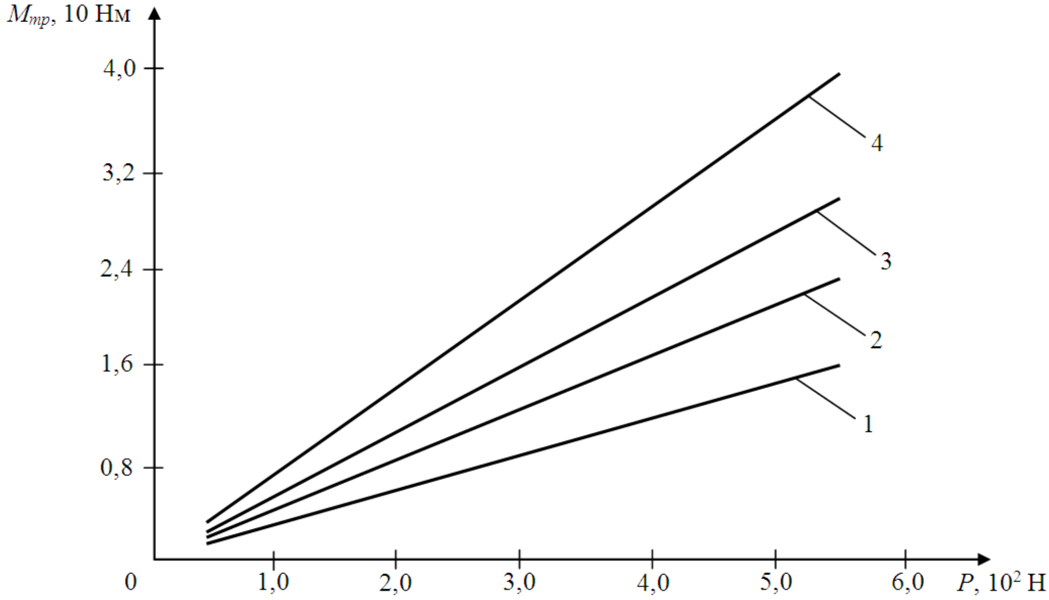


1-at a = 0.4; 2-at a = 0.5; 3-at a = 0.6; 4-at a = 0.7

Fig. 5. Graphical dependences of the change of the moment on the frictional force in the kinematic pair with the grooves in the cylinder (axis) from the increase in the load from the pin

## V. CONCLUSION

The analysis of the work of rotational kinematic pairs of the fifth class is carried out, their shortcomings are determined. An effective scheme for constructing a rotational kinematic pair of the fifth class with grooves on the surface of the cylinder (axis) was developed. A method for calculating the torque from frictional forces in a kinematic pair is recommended. Based on the numerical solution of the problem, graphical dependencies of the parameters are constructed and their rational values are recommended.

## REFERENCES

[1] IIArtobolevskyI.I. Theory of mechanisms and machines. M.: Science, 1988. p. 21-26.
[2] A.Djuraev et al. Theory of Mechanisms and Machines. T.: G. Guloma, 2004. p. 29-30.
[3] A.Kukarina, A.Mronov. About the influence of bearing wear on the strength of the shafts of shafts of vessels. Vestnik ASTU, Russia, 2013, №1. p. 23-26.
[4] V.A. Yudin, Yu.V. Vorobiev. Friction in frictional rolling with a sliding roller pair. Izvestiya VUZov, Mashinostroenie, 1967, №1, p.31-35.
[5] I.Panova. Wear of tribo-plastics in sliding bearings. Internet-journal "NAUKOVODENIE" Issue 2, 2014.
[6] N. Blokhin, V. Danilov, Yu. Musaev. Coefficients of Friction of Rolling of Smooth Cylinders at the Ultimate Bearing Capacity of the Oil Layer.- Mashinovedenie, 1972, no.3, p.85-89.
[7] Kragelsky I. Friction and wear. -M .: Mechanical Engineering, 1968. 430 p.