

METHODOLOGY OF CARRYING RESEARCHES OF KINEMATIC PAIR OF PIVOING FRICTION

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Abstract: The paper presents the program and methodology of carrying out research on kinematic pair with pivoting friction. There was examined a node with pivoting friction in bearing concentrated connection flat surface-spherical bowl. During the research there were determined two types of characteristics – characteristic of friction moment as a function of load and experimentally designated characteristic of wear of the frictional kinematic pair. Both of the assigned characteristics are compared.

Keywords: MOMENT OF FRICTION, PIVOING FRICTION, FRICTION COEFFICIENT, WEAR

1. Introduction

The phenomenon that occurs between two bodies in contact in rest or in motion is called friction. As a result of this phenomenon there arises resistance to motion. There are various kinds of friction distinguished (figure 1).

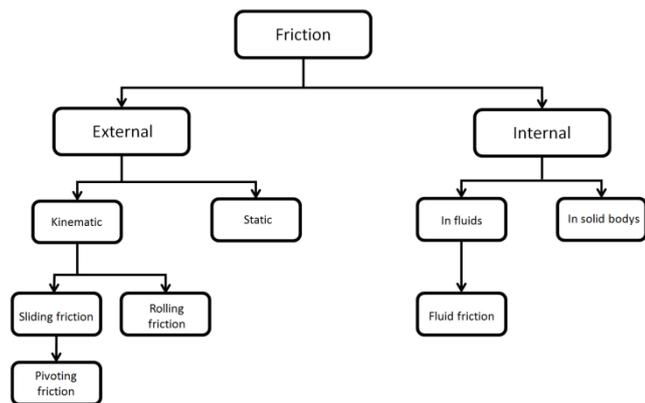


Fig. 1 Types of friction [4].

Machines constructors try to eliminate dry friction, replacing it with more advantageous fluid friction. Due to the fact that cooperating surfaces are not ideally smooth, on the top of the irregularities of the surface there could occur dry or boundary friction, whereas the cavities could be filled with lubricant. In such conditions the friction between the rough contact surfaces is so called mixed friction [2].

2. Pivoting friction

One particular case of sliding friction that occurs in slide bearings is pivoting friction, caused by continuous rotation or oscillation around the normal in point of contact between two cooperating surfaces. Pivoting friction occurs in the bearings: measuring timing devices, analogue scales, spindles, textile machinery etc. This paper attempts to analyze the conditions of pivoting friction for bearing model: flat surface - spherical bowl (figure 2). In this type of bearings obtaining fluid friction, even in case of total immersion of lubricating agent node (figure 3), is not possible due to the scattering of the gaps.

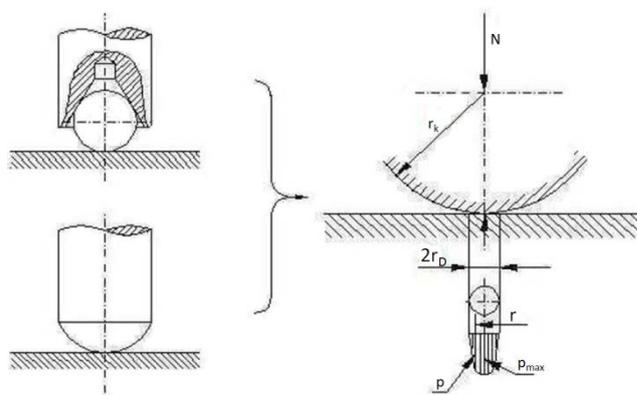


Fig. 2 Model of the bearing type spherical cap on plate; p - Hertz pressure, p_{max} - Hertz pressure maximum [3].

$$p_{max} = \frac{1}{\pi} \sqrt{\frac{6 \cdot N \cdot \frac{1}{r_k^2}}{\left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right)^2}} \tag{1}$$

where: ν - Poisson ratio, E - Young's modulus, N - axial force, r_o - contact radius, r_k - sphere radius.

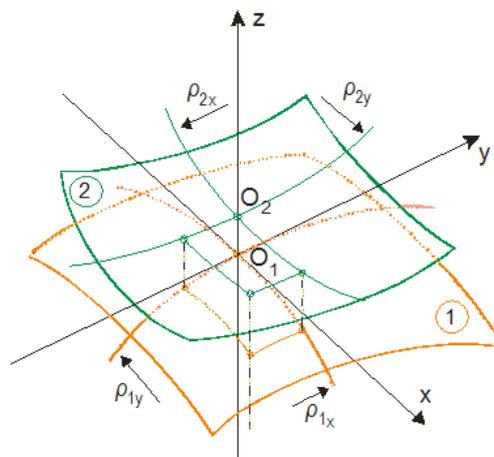


Fig. 3 General model of the slide bearing; r1, r2 – curvature radius of mating surfaces [5].

3. Experimental research model

In the initial examination there was used a bearing ball made of steel 100Cr6 of diameter 8mm and a flat plate made of Ertalon. The test was performed only in conditions of dry friction, without the use of the lubricant.

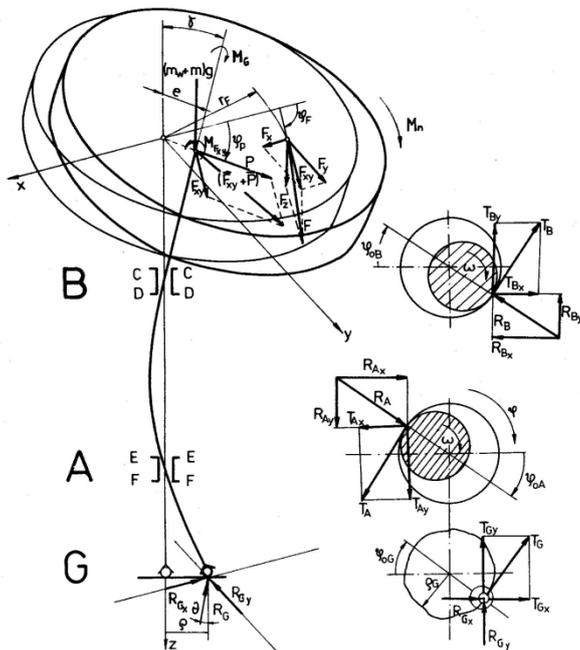


Fig. 4 Scheme of general model of radial thrust bearing [1].

4. Test bench

Figure 5 presents the test bench for determination of friction coefficient by the string twist angle measurement.

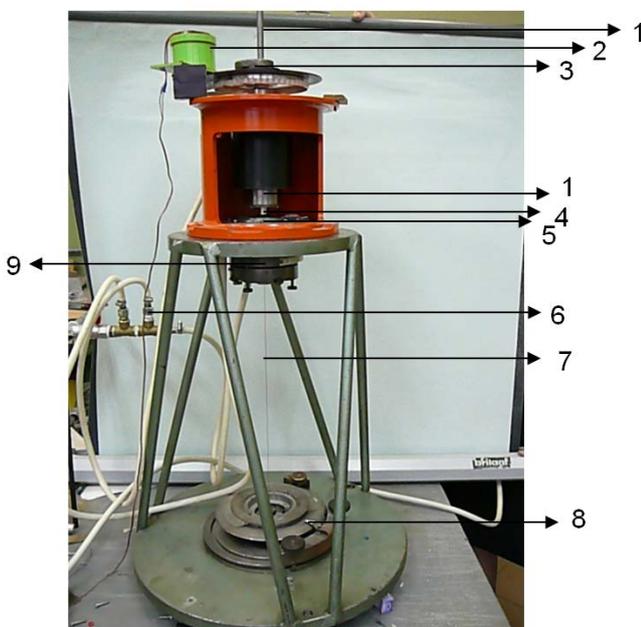


Fig. 5 Test bench for the friction moment and friction coefficient for the frictional pair ball joint – spherical cap; 1 – shaft, 2 – electric motor, 3 – weights, 4 – sphere, 5 – sample, 6 – compressed air supply, 7 – string, 8 – disc to read the angle of torsion, 9 – aerostatic table [6].

The test bench is made of two cylindrical radial bearings and axial bearing. The drive from electric motor (2) is transferred by the belt transmission on the shaft (1) which is ended by the ball joint. The load is changed by adding weights (3) on the shaft (1). The

tested ball joint (5) is placed on the aerostatic table (9) which is connected with the string (7). The second end of string is mounted to the table (8) which is used to twist string angle compensation. The angle of compensation is read from the scale plotted on the table (8).

5. Test bench researches

For the initial experiment there were selected bearing balls made of steel H-15 (100Cr6) of diameter 8 mm and a flat plate made of Ertalon (PA6-E). Experimental measurements were carried out by measuring the moment of friction as a function of load. In addition, the test results calculated due to the varying string angle verified friction torque values designated with signs of wear on the plate.

The test bench research was performed by the measurement of the torsional angle of the string as a function of load. The electric motor was started after mounting the plate from PA6-E and steel ball (100Cr6) to the test bench. The shaft rotated with constant speed – 36 rpm. Then the test bench was gradually loaded by 2 N in the range 7,09÷16,9N. After that the test bench worked in the inverse sequence. Weights were added for 500 cycles.

The smallest load due to the weight of shaft was 7,09 N and peak load was the sum of the weight of the shaft and weights. Each measurement was repeated in 10 measuring series in order to calculate the average value of the string twist angle. The moment of friction was determined according to the relationship:

$$M_t = 33,722 \cdot \mu \cdot p_{max} \cdot N \cdot r_k \cdot \left(\frac{1 - \nu_k^2}{E_k} + \frac{1 - \nu_p^2}{E_p} \right) \quad (2)$$

Track wear plate measuring scanned using an optical microscope with a mounted camera. Assuming that the diameter of the wear scar is 2r_o friction torque calculated from the relationship:

$$M_t = 2 \cdot \pi \cdot \mu \cdot p_{max} \cdot 7,1633 \cdot r_o^3 \quad (3)$$

6. Measurement results

Below is a sample list of friction torque for the combination of friction couple: plate from ertalon (PA6-E) – steel ball (100Cr6).

Tab. 1 Radius r_o of the wear traces on the plate in the function of load for the examined sample

m _w [kg]	m [kg]	g [m·s ⁻²]	d _k [m]	ν _k	ν _p	E _k ·10 ⁶ [Pa]	E _p ·10 ⁶ [Pa]	r _o ·10 ⁵ [m]
0,723	0,0	9,81	0,008	0,308	0,298	210000	180000	5,71
	0,2							6,19
	0,4							6,61
	0,6							6,98
	0,8							7,32
	1,0							7,63

Tab.2 Analytical values and values from traces of wear determined for the friction moment of the frictional pair ertalon-steel.

Load [N]	p _{max} [Pa]	Friction μ [-]	Average moment of friction [Nmm]	Average friction μ [-]	Moment of friction from trace [Nmm]
7,09263	1,04 E+09	0,067029	0,00059	0,067846	0,00052487
9,05463	1,13 E+09	0,062566	0,000721	0,059811	0,00067849
11,01663	1,21 E+09	0,054026	0,000845	0,053975	0,00076099
12,09786	1,27 E+09	0,051049	0,000994	0,051049	0,00089470
14,94063	1,33 E+09	0,049443	0,001152	0,049041	0,00104548
16,96541	1,39 E+09	0,046451	0,001286	0,046451	0,00115785



Fig. 6 Example trace of wear on the flat surface, $r_o=0,258$ mm, $S=0,209$ mm².

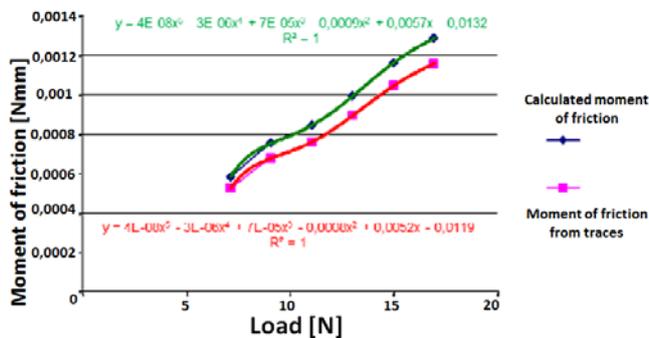


Fig. 7 Graph of moment of friction in function of load for the researches of friction couple PA6-100Cr6.

7. Summary

1. The moment of friction increases proportionally to the load.
2. The characteristic of the friction momentum as a function of load determined analytically is the same as the characteristic determined by the wear.
3. Differences in momentum of friction for analytical calculations and the values determined from the exploitation do not exceed 0,00012 Nmm in the entire load range.
4. The values determined analytically were calculated for the springy connection. The values of friction momentum determined from the wear include the impact of the plastic deformation and the exploitation.

5. References

1. Burcan J.: *Analiza oporów ruchu i jej wykorzystanie w projektowaniu drobnych mechanizmów precyzyjnych*, Politechnika Łódzka, Zeszyty Naukowe Nr 546, rozprawa habilitacyjna, Łódź, 1989,
2. Dietrich M.: *Podstawy konstrukcji maszyn*, Tom 2, WNT, Warszawa 1999,
3. Wozniak M., Ozuna G., De La Fuente P., Jozwiak P., Pawelski Z.: *Comparison of friction coefficient for selected car suspension elements*. International Virtual Journal for Science, Technics and Innovations for the Industry MTM. Year VIII, Issue 4/2014, pp.23-25, ISSN 1313-0226,
4. Lawrowski Z.: *Tribologia – Tarcie, Zużywanie i*

Smarowanie, Warszawa, PWN 1993,

5. Burcan J.: *Analiza oporów ruchu i jej wykorzystanie w projektowaniu drobnych mechanizmów precyzyjnych*. Zeszyty Naukowe nr. 546, Politechnika Łódzka, Łódź, 1989,
6. Wozniak M., Ozuna G., De La Fuente P., Jozwiak P., Pawelski Z.: *Comparison of researches of friction coefficient in concentrated contact for the stress: steel-steel and steel-magnesium alloys*. International Virtual Journal for Science, Technics and Innovations for the Industry MTM. Year VI, Issue 6/2013, pp.51-54, Bulgaria, 1-2.07.2013,