Selection of Polymer Materials for Micro Slide Bearings With Respect to Minimization of Resistance to Motion

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ABSTRACT Values of static and kinetic friction coefficient in micro slide bearings, consisting of journals and bushings (with the diameter of 3.5 mm) made of pairs of 5 polymer materials (16 chosen combinations of PA11, ABS, PC, PS and PETP), were determined. A method of measuring static and kinetic friction coefficient, at various values of: load, rotational speed, and standstill time, as well as the structure of a dedicated test rig, were discussed. Chosen mechanical properties of polymers measured using indentation method, and conclusions from the conducted experiments, are shown. The presented results provide key information for a proper choice of polymer materials, which are to work as a friction node. The biggest observed difference in the moment of friction among the tested slide bearings reached almost 40% (with respect to the highest value).

INDEX TERMS Polymer, friction coefficient, tribology, slide bearing, 3D printing.

I. INTRODUCTION

Due to technological reasons: injection, 3D printing, polymers are more and more often implemented in manufacturing machines and devices [1]–[5]. For many years, the dominating ones were interacting pairs polymer - steel and polymer – anodized aluminum alloys. Currently, mainly due to the development of 3D printing, one may encounter devices, in which there are mating polymer pairs of friction elements in the form of: slide bearings, gear wheels, guideways. A variety of polymers available on the market constitutes a significant problem for a designer, whose task is, among other things, their optimal choice with respect to tribological reasons. Tribological aspects of polymers in mechanisms and micromechanisms have been found to be a major issue in durability of those devices [6]–[9]. In the available literature, however, there are few results of tribological studies of polymer-polymer materials [10]–[12]. Companies manufacturing polymers introduce their own designations and keep the chemical composition and technological process of manufacturing secret. Currently, there is a lack of com-

pany catalogues recommending mutual matching of polymer-polymer materials. Due to extremely labor-intensive studies, and the amount of the possible machining of polymer pairs, this work presents results of studies of 5 mutually matched materials of slide bearings, fabricated by injection molding from the following popular polymers applied in journals and bushings of non-lubricated slide bearings: Polyamide 11 (PA11), Polystyrene (PS), Polycarbonate (PC), Acrylonitrilebutadienestyrene (ABS), Polyethyleneterephthalate (PETP). Additionally, we propose a custom test rig for tribological studies of micro slide bearings, due to numerous limitations of so far used methods [13]–[25].

II. TEST RIG

Experimental studies were conducted on a specially designed test rig (Fig. 1). The principle of operation of the test rig was based on the PN-EN ISO 8295 standard. However, some modifications of the standard were introduced in order to measure friction between cylindrical elements - the shape of actual slide bearings.

The test rig was built on a solid concrete base (10). The journal of the studied bearing (5) was fixed in the

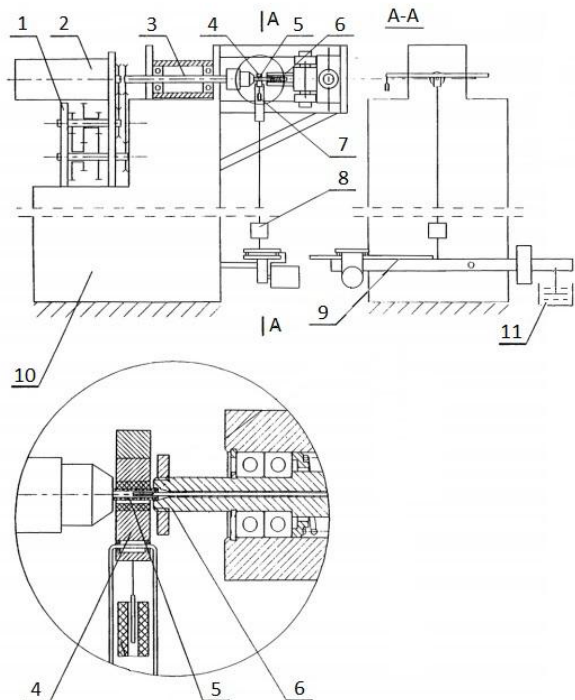The associate editor coordinating the review of this manuscript and approving it for publication was Wuliang Yin[ID].

FIGURE 1. Test rig for studying friction in miniature journal bearings.

FIGURE 2. Tested miniature journal bearing.

FIGURE 3. Principle of measuring the friction coefficient.

self-centering holder of the shaft (3) powered by a DC motor (2) and a gearbox with high ratio (1). The other end of the bearing journal was supported by a rotatable holder (6) with an adjustable position. The bushing of the bearing was fixed in the socket in the shape of a horizontal beam with an opening (4), loaded through a vertical string by a lever system (9). An oleo-dynamic vibration damper (11) was attached to the lever system. On the string, a strain gauge (8) was mounted, enabling a precise set of the force loading the studied friction node. Near the arm of the socket, an induction displacement sensor (7) was attached, allowing to determine an instantaneous moment (coefficient) of friction of the studied bearing. The distance between the rotation axis of the bearing, and the axis of displacement measurement by means of the induction sensor, could be altered, depending on the magnitude of the predicted moment of friction, and its exact value is a parameter crucial for a proper determination of the moment of friction.

The motor of the test rig was controlled using an electronic system mediating the transmission of control signals from dedicated computer software, which was also responsible for acquiring measurement data in the course of operation of the studied bearing. On one of the wheels of the transmission, an optoelectronic sensor measuring the rotational speed was attached, enabling a precise setting of slip speeds of the studied bearing.

Micro slide bearings (Fig. 2), made entirely of polymers, operating without any lubricant, were studied. For the studies,
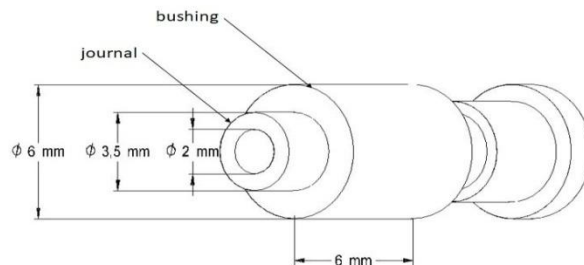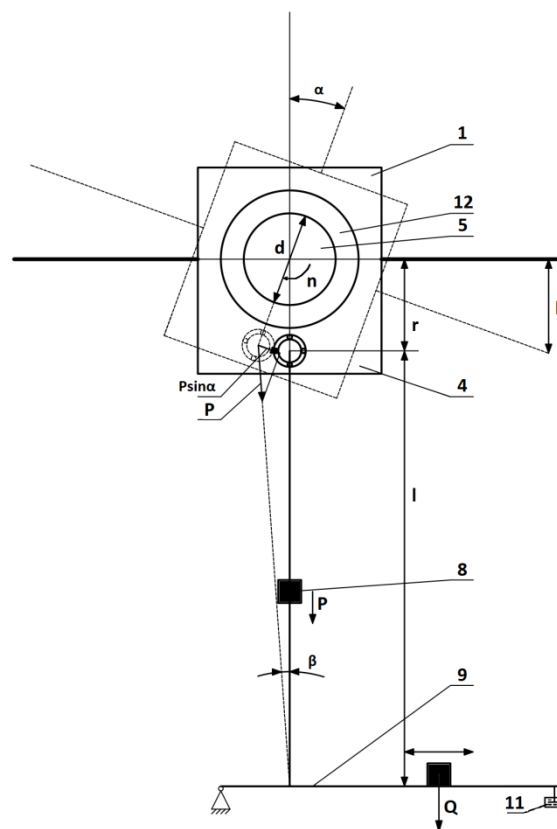
journals and bushings with diameters giving a lateral play within the range from 130 to 140 μm, were chosen.

The procedure of measuring the friction coefficient (Fig. 3) obtained a patent [26].

The moment of friction between the rotating journal (5), at the rotational speed n, and the bushing (12) equals:

$$M_t = 1/2 \, Pd\mu \tag{1}$$

and is balanced by a moment

$$M_z = Pr\sin\alpha \tag{2}$$

As a result of balance of moments $M_t = M_z$, we obtain

$$\mu = 2 \, r\sin\alpha/d \tag{3}$$

where:

μ – friction coefficient between the journal and the bushing,

P – force loading the bearing from 5 to 30 N, thanks to the moving, on a one-sided lever, weight Q, measured by the strain gauge force sensor (8),

$d = 3.5$ mm – journal diameter,

$r = 7$ mm – distance between: bearing axis and the point of attaching the string loading the bushing socket (4) by P.

α – angular displacement of the socket under the influence of Mt,

$l = 1070$ mm – the length of the string loading the socket of the bearing bushing.

Angle α was determined from measuring: height h by an induction displacement sensor, and distance from the bearing axis to the point of attaching the induction displacement sensor (7). The zero-point moment of friction was found by rotating the journal clockwise and counterclockwise, assuming that the friction coefficient was identical in both directions of rotation (see Fig. 6). In the designed test rig, the length of the string l was chosen in such a manner, that the influence of angle β on the error of measurement could be disregarded: for the friction coefficient $\mu = 0.5$ angle $\beta = 15$', so $\cos\beta = 0.99999$.

Having programmed the test rig and conducted a preliminary measurement of the friction coefficient for comparative purposes and to eliminate system errors, additional studies using displacement method were conducted. A string with two identical weights, one of which was immersed in water, had been mounted on the bushing. Measuring the depth of the weight immersion in water, buoyant force was determined, and after calculations, friction coefficient was obtained. The results of both measurements were within the range of 3-sigma measurement uncertainty.

Parameters of the studies:

- load: 5; 6; 8; 10; 13; 16; 20; 25; 30 N,
- linear speed of the journal in reference to the bushing: 4.3; 9.7; 18; 29.2; 39.6; 51.5; 59.9; 79.3; 101.6 μm/s,
- standstill time: 1; 2; 5; 15; 30; 60; 300; 900 s.
- temperature of 20-25°C and relative humidity of 40-60%

The course of each study was as follows,

a. assembly of the bushing and the journal of the bearing in the holders,
b. presetting parameters of the study into the control software (rotational speeds, loads, standstill times),
c. starting the software,
d. configuring a set load value,
e. a trial start-up and stop of the bearing,
f. standstill under load for a preset time period,
g. starting the bearing with the first preset speed,
h. registering the build-up of moment of friction, interrupting the contact (with the torque corresponding to static friction) and kinetic friction within the initial period,

i. interrupting data acquisition and stopping the motor.

The average value of friction coefficient μ of a given material pair was determined as the arithmetic mean of friction coefficients, for each combination of a loading force and slip speed, with the studied standstill time of a given friction node. Based on these values, a standard deviation of friction coefficient in a given study was calculated, in order to assess data scatter.

Other tribological properties of the tested bearings have not been studied, except only for the fact that 24 hours of operation did not result in a significant increase of the moment of friction. After the measurements, a wear as deep as 20 μm was observed.

As commonly known, selection of appropriate (with respect to its mechanical and thermal properties) material for the journal and the bushing of a bearing is only the first stage. Many other parameters impact the resistance to motion; for instance, roughness should be taken into account. In this study polymer samples fabricated by injection molding with no further processing were used.

Roughness of the samples was analyzed by means of Atomic Force Microscopy (AFM) over the area of 20 μm x 20 μm. After leveling the cylindrical surface, the roughness parameter Ra was calculated for all of the samples. The average values in μm were: PA11 Ra = 0.197; PC Ra = 0.049; PS Ra = 0.093; ABS Ra = 0.192.

Mechanical properties of polymers: hardness, elastic modulus, viscosity — appearing often in frictional models created for simulation purposes, e.g. in [27], were determined by means of a Hysitron Tryboscope. The tests were performed on the flat side of the bush using a Berkovitch tip. A trapezoidal load profile was employed, with maximum load of 200 μN and loading/unloading speed of 40 μN per second. Additionally, the maximum load was held for 240 seconds in order to observe a creep of the polymer, which enabled calculation of its viscous properties according to the VEP model [28]. Every sample was measured 7 times at different regions on the bushing. The viscosity of the sample was calculated from [29]:

$$h^{CREEP}(t) = \frac{P_{max}^{\frac{1}{2}}}{(\alpha_3 \eta)^{\frac{1}{2}}} + h^{LOAD}(t_r) \qquad (4)$$

where:

$h^{CREEP}(t)$ — indentation depth during the hold-load segment of indentation,

P_{max} — maximal load during indentation,

α_3 — dimensionless geometrical constant; for Berkovitch tip it is equal to 4.4 [29],

η — viscosity,

$h^{LOAD}(t_r)$ — indentation depth at the end of the loading segment.

The elastic modulus was derived from plain strain modulus [28] as:
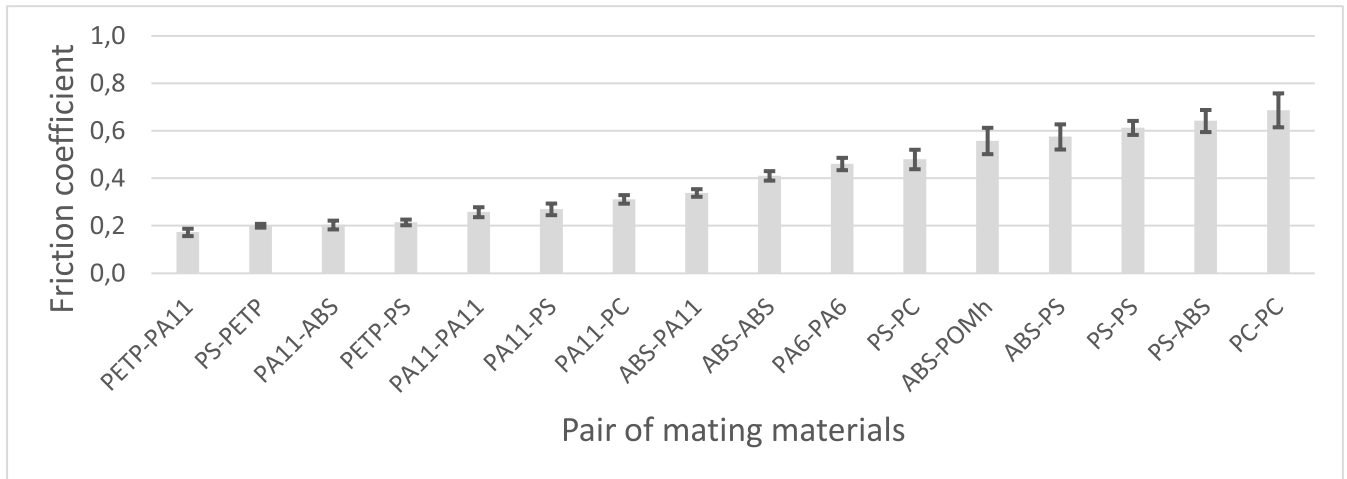
$$E' = \frac{E}{1 - \nu^2} \qquad (5)$$

FIGURE 4. Average static friction coefficients between the journal and the bushing made of the studied polymers (with standard deviation); the material of the journal is the first of the two enumerated materials.

where:

$E'-$ plain strain modulus,

E – elastic modulus,

$\nu-$ Poisson ratio.

The plain strain modulus was calculated using the unloading slope by selecting the material properties in such a way that the modeled curve would fit the experimental curve [29]:

$$h^{UNLOAD}(t) = k^{\frac{1}{2}}\left(\frac{t_r^{\frac{3}{2}} - (2t_r + t_c - t)^{\frac{3}{2}}}{\frac{3}{2}(\alpha_3 \eta)^{\frac{1}{2}}} \right.$$

$$\left. + \frac{(2t_r + t_c - t)^{\frac{1}{2}} - t_r^{\frac{1}{2}}}{(\alpha_2 E')^{\frac{1}{2}}} \right) + h^{CREEP}(t_r + t_c) \quad (6)$$

where:

$h^{UNLOAD}(t)-$ indentation depth during the unloading in time t,

K – unloading rate,

t_r- loading and unloading time,

t_c- time of hold-load segment of indentation,

α_2- dimensionless geometrical constant; for Berkovitch tip equal to 4.4.

The hardness of the polymers was calculated using the Oliver and Pharr method [30] as:

$$H = \frac{P_{max}}{A(h_c)} \quad (7)$$

where the contact area $A(h_c)$ is calculated as:

$$A(h_c) = 24.56 h_c^2 \quad (8)$$

and the contact depth h_c is calculated as:

$$h_c = h_{max} - \frac{\varepsilon P_{max}}{S} \quad (9)$$

where:

$h_{max}-$ maximum contact depth,

TABLE 1. Averaged values of the measured mechanical properties of polymers with respective standard deviation of measurements.

	ABS	PA11	PC	PS	PETP
Hardness (GPa)	0.55 +/-0.15	0.40 +/-0.16	0.42 +/-0.12	0.36 +/-0.11	0.27 +/-0.07
Young Modulus (GPa)	1.73 +/-0.33	1.30 +/-0.42	1.47 +/-0.51	1.46 +/-0.33	1.22 +/-0.47
Viscosity (TPa s²)	145 +/-45	276 +/-77	76 +/-16	109 +/-18	196 +/-61

$\varepsilon-$ geometrical constant; for Berkovitch tip equal to 0.75 [31],

S – slope of the unloading curve.

III. RESULTS

Journals and bushings made of 5 thermoplastic polymers PA11, ABS, PC, PS and PETP were used. 16 material combinations were studied. The obtained average values of the respective friction coefficients, along with the relevant standard deviation, are presented in Fig. 4 (static friction coefficient) and Fig. 5 (kinetic friction coefficient).

Elastic modulus, hardness and viscosity of each of the studied polymers are given in Table 1, calculated on the basis of transformed Eq. (5), (7), (4), respectively.

An exemplary course of variations of the friction coefficient during each experiment is presented in Fig. 6. At the first stage, the journal sticks to the bushing without any mutual slip; it is a stage of undeveloped friction: the friction coefficient apparently increases, reaching its maximal value μ_s. Then, the journal starts to slip in the bushing; it is a stage of mixed friction. Finally, as the rotational speed of the journal stabilizes, the coefficient of friction settles at the value μ_k. The non-zero value of the friction coefficient at the beginning, results from the measurement setup in which
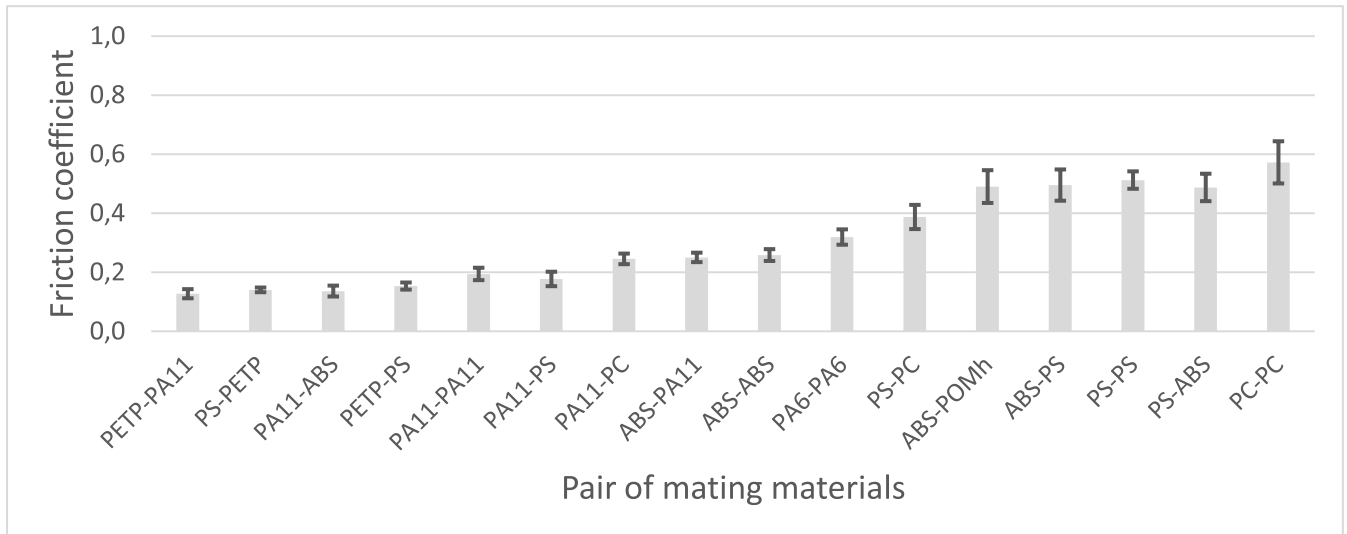
FIGURE 5. Average kinetic friction coefficients between the journal and the bushing made of the studied polymers (with standard deviation); the material of the journal is the first of the two enumerated materials.
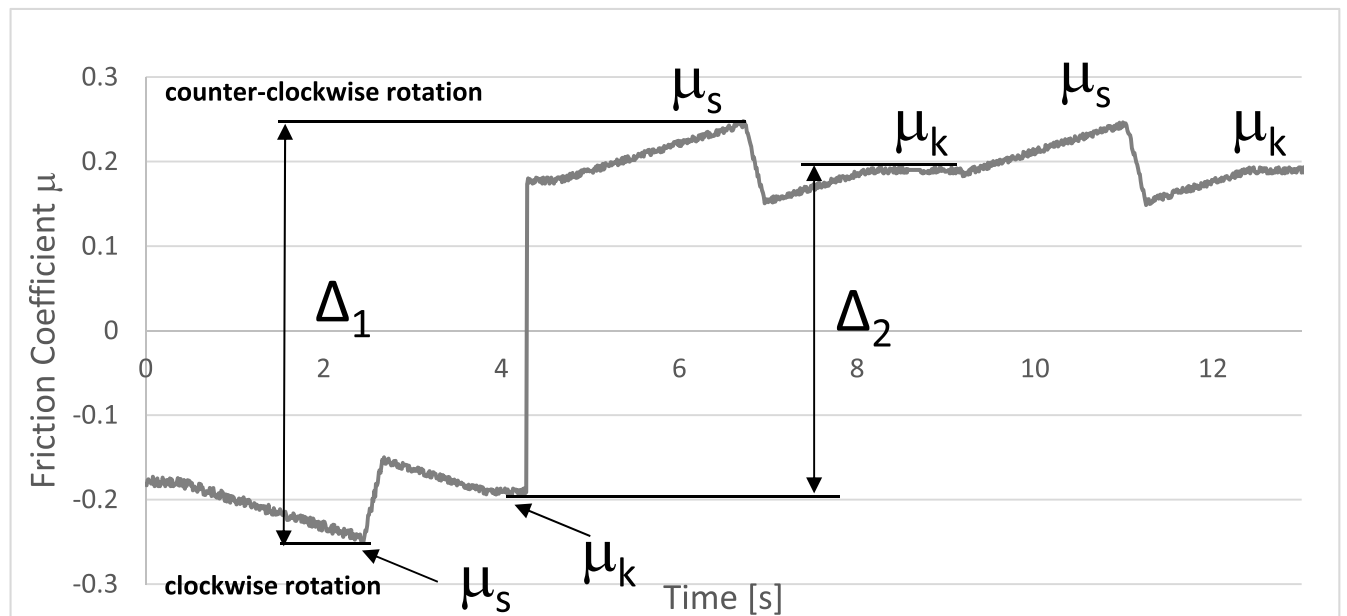
FIGURE 6. Exemplary course of variations of the friction coefficient

the zero-point is found by clockwise and counterclockwise rotation of the journal.

IV. DISCUSSION

Friction is one of the most crucial factors impacting the resistance to motion. Its impact is greater in the case of miniature slide bearings, where aerodynamic drag and inertia forces play a much lesser role.

None of the tested samples was subjected to any further processing after having been fabricated by injection molding. Such condition of their surface is the most probable in the case of industrial applications.

Pairs of materials featuring the lowest friction were PETP-PA11, PETP-PS and ABS-PA11, while the highest friction coefficient was measured for ABS-PS, PS-PS and PC-PC combinations. Pairs consisting of the same material on both elements feature high friction coefficients, as compared to mixed materials. Kinetic friction for all the measured pairs is lower than static friction, and the pairs that yield low static friction coefficient feature also low kinetic friction coefficient.

The overview of values of friction coefficients for different combinations of mating materials, enables an optimal selection of materials while designing a friction node, where
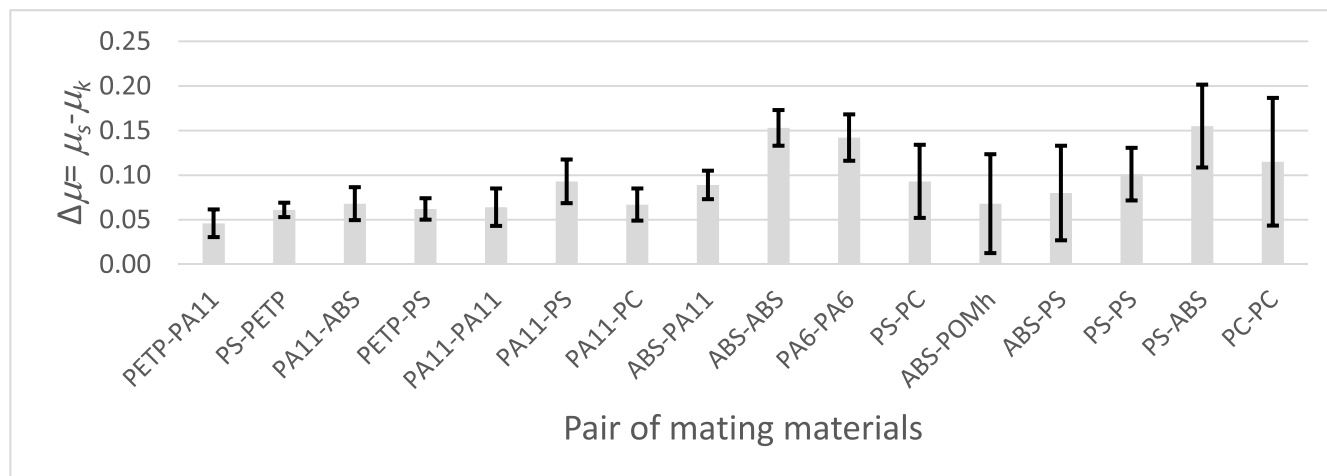
FIGURE 7. Difference between static and kinetic friction coefficients (with standard deviation); the material of the journal is the first of the two enumerated materials.

the materials chosen for a bearing journal and a bushing are characterized by an advantageous friction coefficient.

In friction nodes, one ought to avoid using pairs of materials which are the same, or very similar, due to adhesive interaction increasing friction coefficient. Thus, in many cases, cheap materials which are properly chosen, will mate much better than more expensive ones, yet the same, while used for a journal and a bushing at the same time.

Statistically significant differences in values of moment of friction for the same pairs of materials, depending on which a journal, and which a bushing was made of, were not observed. The analysis was done by t-test comparison of groups at *p*-value of 0.05.

V. CONCLUSION

As confirmed by numerous examples in the world of technology, the choice of an appropriate pair of materials mating friction-wise, is far from easy. Quite often, the worst solution is accepted, i.e. making both mating elements of the same material. Owing to the presented results of the study, one may optimally select polymer materials which constitute friction pairs, characterized by low value of friction coefficient, as well as low difference between the static and kinetic friction coefficient (as defined in Fig. 6), whose values are presented in Fig. 7.

Knowledge of mechanical properties of materials used for slide bearings, and potentially ball bearings, allows us to predict the durability of the bearings. These predictions can be done by using modeling based on the Finite Element Method.

It is planned to further the research with regard to determining the significance of the influence of the following parameters: rotational speed, load, standstill time. These parameters will be altered in a controlled manner, while conducting experimental works. It is also intended to study a wider group of polymers.

As proven by the reported results, appropriate selection of polymer materials for micro slide bearings may minimize their resistance to motion even by ca. 40%.

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