Research of sliding bearings with reverse friction pair and inlaid liners made of thermoplastic composite materials

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Abstract. In the article the authors reveal technical solutions to increase the efficiency of inverted plain bearings with typesetting inserts made of thermoplastic composite materials by determining their rational parameters. The working hypothesis of the research was the postulate that the efficiency of composite polymeric materials is determined not only by high physical and mechanical and other properties, but also by the rational design of the friction unit. In the article, the authors proposed the developed experimental design of the inverted plain bearing, which differs from the existing ones in that the plastic inserts of the plain bearing are made in the form of prisms with a rectangular cross section, one side of which is an arc. The authors present analytical dependences that allow to determine the load that is perceived by the most loaded liners under the simplest load condition and a purely radial load applied to a radial bearing having a contact angle. The article reveals the results of laboratory tests and found that the total linear wear of the bearings of the sliding bearing in the form of inverted friction pair at 100 hours of continuous operation in a grease environment is 20…25% less than the bearing in the form of direct friction pair.

1. Introduction
The efficiency of composite polymeric materials is determined not only by high physical [1] and mechanical [2] and other properties [3], but also by the rational design of the friction unit [4]. Sliding bearings made according to the shaft-sleeve scheme with a sleeve made of thermoplastic materials have become widespread [5]. This design provides ease of manufacture and assembly of the bearing [6], interchangeability and ease of repair [7], which in this case is to replace the worn sleeve with a new one [8]. In this case, the sleeve is usually pressed into the seat of the bearing assembly [9], which does not allow to obtain the required tension or clearance during operation [10] and makes it difficult to take into account the compensation in the connection of plastic and metal parts [11]. At the same time, the sleeve receives pressure from the shaft only on the surface [12], which is determined by the angle of contact.
(girth) [13]. As a result of such interaction the plug wears out in one place [14]. In addition, the spike of the shaft, which rotates in the sleeve [15], when the abrasive particles enter the gap [16], wears out, and in some cases its wear may exceed the wear of the sleeve [17]. Known designs of plain bearings in the form of inverted friction pair [18], where the plastic sleeve is mounted on the shaft and rotates with it inside the outer holder [19]. The advantage of this design is the absence of wear of the shaft when the abrasive enters the friction zone [20]. The disadvantage is unsatisfactory heat dissipation from the friction surface into the environment, which is caused by the continuity of the sleeve [21]. Known design of the inverted plain bearing, which consists of a housing with a clamp installed in it with rod inserts, which are located in the longitudinal grooves [22]. Landing of the rod inserts is carried out on a cylindrical surface so that the axes of the inserts do not coincide with the inner surface of the holder [23], so that the rods are fixed from falling out [24]. Any shaft design can be attached to the shaft [25]. In this embodiment, the bearing operates according to the scheme of the inverted friction surface [26].

The disadvantage of this design is the complexity of manufacturing individual elements – split locking rings, clips. Uneven wear along the height of the liners leads to an uneven change in the clearance in the bearing throughout the service life [27]. During the initial operation of the bearing, the increase in the gap in the connection is more intense than in the next [28]. Uneven wear of the inserts on the contact area causes a proportional increase in temperature in the friction zone. In this case, the calculation of the temperature regime, which is an important element in the design of plain bearings with plastic inserts [29], is a task of some complexity [30].

The aim of the research is to increase the efficiency of the use of inverted plain bearings with type-setting inserts made of thermoplastic composite materials by determining their rational parameters.

2. Materials and methods

At mutual sliding of the corresponding plastic inserts there is a wear caused by fatigue of material which is shown thanks to adhesive forces of coupling at inevitable heat formation. In the design of the bearing it is irrational to increase the working thickness of the liners, as the size of the liner is regulated by the size of the inner holder of the bearing assembly. In addition, the increase in the diameter of the cylindrical liners leads (due to the low thermal conductivity of the plastic) to a change in the thermal mode of operation of the bearing. Heat accumulates in the polymer and leads to deterioration of the mechanical properties of plastics. To eliminate these shortcomings, an experimental design of the inverted plain bearing was developed, which differs from the existing ones in that the plastic inserts of the plain bearing are made in the form of prisms with a rectangular cross section, one side of which is an arc. The diameter of the arc is determined by the size of the fixed holder, as a result of which the surface area of contact of the liner with the fixed holder always remains constant, regardless of their linear wear. The plain bearing (figure 1) consists of a housing 1, which is fixedly connected to the holder 2, which is fixed by a locking screw 5. Inside the holder 2, coaxially with it, is a holder 4, in the longitudinal grooves on the cylindrical surface of which are fixed plastic liners 3 by means of screws 9. The holder 4 (figure 1) is fixed on the shaft 11 by a key 6, and the thrust rings 8 on the shaft 11 and the seal 10 are held by the covers 7.

The operation of the bearing shaft 11 (figure 1) with the holder 4 and fixed inserts 3 is carried out due to rotation inside the holder 2. The presence of local wear of the holder 2 (figure 1) provides the possibility of rotation relative to the pressure zone of the shaft 11. The force of possible axial movement of the holder 4 (figure 1) in the form of rectangles made by cutting along the generating cylinder. These sections also have a spherical surface of friction of the liners on the holder 2 (figure 1), create obstacles to the rotation of the liners around its own axis.

The effort when installing the inserts in the holder 4 (figure 1), can further increase with the inevitable increase in temperature during operation of the bearing due to the increased coefficient of linear expansion of the polymer inserts. This fact, as well as the design solution contributes to the reliable fixation of the inserts in the holder 4. To simplify the fixation of the inserts in the holder 4 (figure 1), you can use a groove in the form of a dovetail, but this is due to the complexity of the technology of making a movable bearing holder.
3. Results and discussion

Determination of the load capacity of the sliding bearing in the form of an inverted pair is based on the calculation of the load distribution between the inserts (inserts), which is as follows. To the inner bearing holder of the sliding radial load application $F$, the inner holder is shifted relative to the outer on and the liners will be in position 0, 1, 2, … $n$ (figure 2). In the absence of an initial radial clearance in the bearing, the liners absorb the load $Q_0$, $Q_1$, …, $Q_n$. At the same time, deformations will occur in the places of contact of sliding bodies with the outer holder $\delta_0$, $\delta_1$, …, $\delta_n$. Placing the inserts relative to each other at an angle $\gamma$ the equilibrium equation is as follows:

$$F = Q_0 + 2 \cdot Q_1 \cdot \cos \gamma + 2 \cdot Q_2 \cdot \cos 2\gamma + \ldots + 2 \cdot Q_n \cdot \cos n\gamma.$$  \hspace{1cm} (1)

According to Srimeck’s theory, the relationship between load and deformation in this case can be expressed by the following relations:

$$\frac{Q^2}{\delta^2_0} = \frac{Q^2}{\delta^2_1} = \frac{Q^2}{\delta^2_2} = \ldots = \frac{Q^2}{\delta^2_n}, \quad \text{or} \quad \frac{Q^2}{\delta^2_0} = \frac{Q^2}{\delta^2_1} = \frac{Q^2}{\delta^2_2} = \ldots = \frac{Q^2}{\delta^2_n},$$  \hspace{1cm} (2)

$$Q_1 = Q_0 \cdot \left(\delta_1 \cdot \delta_0^{-1}\right)^{1.5}, \quad Q_2 = Q_0 \cdot \left(\delta_2 \cdot \delta_0^{-1}\right)^{1.5}, \quad Q_n = Q_0 \cdot \left(\delta_n \cdot \delta_0^{-1}\right)^{1.5}.$$  \hspace{1cm} (3)
the angle of the loaded zone decreases and the radial load is distributed within this range \( z = 0 \), and this change in engineering calculations can be neglected. \( F \) is taken into account by increasing the coefficient in the formula (7) constant and equal to 4.37. The appearance of a radial gap significantly affects the load distribution \( Q \) on the liner. In the case of a gap-free bearing, all inserts are loaded, which are located below the horizontal axis and the angle of the loaded area is \( 2\gamma = \pi \). With a radial gap \( \delta_r > 0 \) the angle of the loaded zone decreases and the radial load is distributed over a smaller number of liners, reloading each liner within the loaded area. The influence of the radial gap in engineering calculations is taken into account by increasing the coefficient in the formula:

\[
Q_0 = 5.2 \cdot F \cdot z^{-1}.
\]

More accurate calculations can use the formula:

\[
Q_0 = 4.37 \cdot \left( 0 + 40 \cdot \delta_r \cdot h^{-1} \right) \cdot F \cdot z^{-1}.
\]

where \( h \) – liner height; \( \delta_r \) – radial clearance.

In order to clarify the application of dependences (1), (2), (4) - (9) to calculate the number of inserts of inverted plain bearings, experimental studies were conducted, which consisted of the following. In the laboratory bearing (figure 3) were made several versions of the inner clips, which differed in the number of inserts and their sizes. In the process of testing the load in all variants of the internal clamps was applied the same: \( F = 800 \) H. According to formula (9), the values of the loads on the most loaded liner were determined at their variable number in the inner holder \( z = 4, 6, 8, 12 \). The tests were performed in grease. The sliding speed under these test conditions was constant and was \( \Theta = 0.392 \) m/s. The results of experimental studies of comparative tests of metal-polymer plain bearings showed (figure 4) that when the number of liners \( z = 4 \ldots 6 \) there is intense heat generation and loss of stability of polymer liners (melting and partial application of material on the sliding surface – the outer holder). This is due to the fact that the allowable gap \( \delta = 0.1 \ldots 0.15 \) mm in the bearing is insufficient when the number of inserts \( z = 1 \ldots 6 \). Due to the relatively high coefficient of linear expansion of the plastic liners at the specified gap, even with a slight increase in temperature in the friction zone, there is
a process of adhesion with partial jamming. Increasing the operating gap leads to an increase in the load on the liner. It is established that the most rational number of inserts in the inner holder of the inverted plain bearing is \( z = 8 \ldots 12 \). From figure 5 shows that under the same conditions at \( z = 8 \ldots 12 \) the moment of friction during running-in is less than at \( z = 4 \ldots 6 \). Evaluation of the effectiveness of graphite-modified coprolon-B in bearing assemblies was performed on wear resistance. For comparative evaluation, the bearing assembly of the initial cleaning apparatus of the Moreau corn harvester was taken. Four 11237K rolling bearings are mounted on the shaft. Experience of operation of the combine showed that installation of sliding bearings is not economic and does not reduce the vibration loadings which arise at work of the initial cleaning device.

![Figure 3. Design of the laboratory sliding bearing.](image1)

![Figure 4. Dependence of temperature in the friction zone of different design of the plain bearing on the test time T, hours: 1 - z = 4; 2 - z = 6; 3 - z = 8; 4 - z = 12; 5 - direct friction pair.](image2)

Frequency of rotation of a shaft of the initial cleaning device \( n = 200\text{min}^{-1} \) at nominal working loading on one bearing \( F = 600 \text{N} \). According to \([31, 32]\), the design dimensions of the plain bearing in the form of a direct friction pair are calculated.
Figure 5. Dependence of the moment of friction $M_i$ of various design of the sliding bearing on test time $T$, hours: $1 - z = 4$; $2 - z = 6$; $3 - z = 8$; $4 - z = 12$; $5 -$ direct friction pair.

Diameter of a basic site of a shaft (neck) $d_x = 35$ mm; wall thickness of the sleeve $S = (0.45...0.5) \cdot \sqrt{d_x} = 3$ mm; sleeve length $l = (1.0...1.3) \cdot d_x = 45.5 \approx 46$ mm. The experimental sliding bearing in the form of the inverted friction pair is installed on the basis of the standard case of the rolling bearing 11237K (figure 6). The outer friction holder 7 (figure 6) is pressed into the housing 8. The holder 7 (figure 6) can be installed in the housing by sliding landing, but in this case it must be provided to fix it from turning. The outer side surface of the holder 7 (figure 6), as well as the inner sliding surface is heat treated. The inner holder 5 (figure 6) is mounted on the shaft by means of a split conical sleeve with a clamping nut on the holder 4 are fixed eight inserts 6 of fluoroplastic-4 with a paraffin content of 6%. The thrust ring 2 (figure 6) prevents the displacement of the shaft with the holder 5 and is remote when determining the lateral gap between the ends of the inserts and the inner holder 5. In the cover 1 (figure 6) and the housing 8 are rubber-metal sealing sleeves.

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Figure 6. General view and elements of the laboratory sliding bearing in the form of an inverted friction pair based on a rolling bearing 11237K.

The results of laboratory tests showed that under the same conditions, the total linear wear of the bearings of the sliding bearing in the form of inverted friction pair at 100 hours of continuous operation in a grease environment is $20...25\%$ less than the bearing in the form of direct friction pair. This confirms the effectiveness of the use of plain bearings in the form of inverted friction pairs in the loaded components of machines and mechanisms for various purposes. It can be noted that the use of proposed solution in conjunction with advanced modified graphene-containing lubricants (PCF – polycarbonafluoride and GLS – graphene-like structures) of trademark «CFera» (manufacturer PKF Alliance LLC – Russia, St. Petersburg) will additionally increase the operational resistance of units.

4. Conclusion
The largest value of the initial load $Q_0$ acquires at $z = 15$, namely the initial load receives the most loaded liner of the sliding bearing placed in the plane of action of external radial loading at $\gamma = 0$. 
The magnitude of the initial load $Q_0$ within the angle $\gamma$ do not remain constant, but change relatively little $Q_0 = (4.3...4.43) \cdot F \cdot z^{-1}$ and this change in engineering calculations can be neglected and take into account that the coefficient is 4.37.

The appearance of a radial gap significantly affects the load distribution $Q$ on the liner, so with a radial gap $\delta_1 > 0$ the angle of the loaded zone decreases $2\gamma \leq \pi$ and the radial load is distributed over a smaller number of liners, reloading each liner within the loaded area.

The total linear wear of the inserts of the sliding bearing in the form of inverted friction pair at 100 hours of continuous operation in the environment of grease is $20...25\%$ less than the bearing in the form of direct friction pair.

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