Effect of wedge coupling in smooth cylindrical surfaces contact

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Abstract. The possibility of obtaining the wedge coupling effect in the contact of smooth cylindrical surfaces is considered in the article. Theoretical dependences are obtained to determine the magnitude of the reduced friction coefficient, realized in coupling of unrun and run-in cylindrical surfaces. The correctness of choice of the pressure law on cylindrical surfaces and the proposed theoretical dependencies is experimentally confirmed. Empirical values of the reduced friction coefficient for a "steel-steel" friction pair under oil lubrication conditions are obtained. It is shown that the value of the reduced friction coefficient in the proposed type of coupling of cylindrical surfaces increases 2.6...5.7 times in comparison with the actual friction coefficient.

1. Introduction
The principle of load transfer by friction forces is widely used in various mechanical devices: belt gearings [1–6]; free-wheel mechanisms [5, 7–13]; friction brakes [5, 13–18], etc. [4–5, 13, 19–27]. The load capacity of such devices is determined by the magnitude of the frictional force arising in the contact of their working surfaces.

To increase the frictional force, one can use the effect of wedge coupling of the working surfaces. In this case, kinematic pairs are used, which elements contact along prismatic surfaces resulting in the reduced friction coefficient is realized, which is determined by formula [4, 5]:

\[ f^\prime = \frac{f}{\sin \beta}, \]

where \( f \) – the actual friction coefficient; \( \beta \) – a half of the refracting angle of a prism.

However, in a number of cases, for example, in free-wheeling mechanisms, such design solution complicates the manufacturing and installation technology, and also increases frictional losses during the free running period, due to constant contact of their working surfaces.

It is proposed to obtain a wedge coupling effect in a kinematic pair which elements contact along smooth cylindrical surfaces.

2. Theoretical model
Let us consider the kinematic pair (figure 1) formed by an outer race with an inner cylindrical surface and a ring formed with arched bulges and on an outer cylindrical surface disposed at an angle \( \beta \). Contact in such kinematic pair occurs over the surfaces of the arched bulges.
In the general case, the ring is acted upon by pressing force $F_R$ and pressure $p(\phi)$. Under the influence of these loads, the ring is in equilibrium.

**Figure 1.** Design model of the kinematic pair: 1 – outer race; 2 – ring; 3, 4 – arched bulges.

Let us single out a surface element on the contact surface of the outer race and the ring:

$$ds = l r d\phi,$$  \hspace{1cm} (1)

where $l$ – the contact surface length; $r$ – the contact surface radius; $\phi$ is the current polar angle.

Elementary force of normal pressure on the element $ds$, with regard to expression (1) is defined as:

$$dF_N = p(\phi) ds = l r p(\phi) d\phi,$$  \hspace{1cm} (2)

where $p(\phi)$ – a function that describes the pressure law on contact surface.

Elementary frictional force on the element $ds$ with regard to expression (2), is defined as:

$$dF_f = f dF_N = f l r p(\phi) d\phi.$$

The expression for determining the total force of normal pressure can be found from the equilibrium condition of the ring:

$$F_N = F_R = 2 l r \int_{\frac{\pi}{2} - \beta}^{\frac{\pi}{2}} p(\phi) \cos \phi \, d\phi.$$  \hspace{1cm} (4)

The expression for determining the total frictional force can be written as:

$$F_f = 2 l f r \int_{\frac{\pi}{2} - \beta}^{\frac{\pi}{2}} p(\phi) d\beta.$$  \hspace{1cm} (5)

Then the reduced friction coefficient in such kinematic pair can be defined as:

$$f^* = \frac{F_f}{F_N}.$$  \hspace{1cm} (6)

From expressions (4) and (5), it is clear that the function $p(\phi)$ has a significant effect on the magnitude of the reduced friction coefficient.
With a sufficient accuracy for practical calculations, the nature of the pressure distribution for
unrun cylindrical surfaces can be approximated by a constant dependence, and the run-in ones – by a
cosinusoidal dependence [4–6, 13, 28].
Formulas for determining the reduced friction coefficient for these pressure distribution laws are
obtained (table 1).

**Table 1. Dependencies for determining the reduced friction coefficient.**

| The pressure distribution law | Formula |
|------------------------------|---------|
| Constant \( p(\varphi) = p_{\text{max}} = \text{const} \) | \( f^* = \frac{f\beta}{1 - \cos \beta} \). (7) |
| Cosinusoidal \( p(\varphi) = p_{\text{max}} \cos \varphi \) | \( f^* = \frac{4f (1 - \cos \beta)}{2\beta - \sin 2\beta} \). (8) |

The correctness of choice of the pressure law on cylindrical surfaces for each particular friction
pair must be checked experimentally.

3. **Experiment Procedure**

An experimental study of the reduced friction coefficients in the proposed kinematic pair was carried
out to verify the validity of the obtained formulas (7), (8) and to obtain their empirical values.

The wedge coupling effect of cylindrical surfaces (figure 2) was created by a kinematic pair formed
by a bushing and a semi-ring made with arched bulges disposed at angle \( \beta \). In the proposed kinematic
pair, the angle of the arched bulges is expedient to be taken within limits \( \beta = 15 \ldots 35^\circ \).

![Figure 2. A kinematic scheme of the experimental plant: 1 – bushing; 2 – semi-ring.](image)

During the experiment, a bushing, a semi-ring with a smooth cylindrical surface \( \beta = 90^\circ \) and semi-rings with angles \( \beta = 15^\circ, 25^\circ \) and \( 35^\circ \) were used. A coupling radius of the surfaces of the bushing and semi-ring is \( r = 28 \text{ mm} \), the coupling length is \( l = 10 \text{ mm} \). The experimental specimens of bushing and semi-rings were made of steel 14NiCr10 with subsequent heat treatment up to 58...62 HRC.

The experiments were carried out in the oil lubrication condition SAE30 under steady thermal
conditions with oil temperature \( t = (55 \pm 5) \text{ °C} \) and sliding speed \( v = 2 \text{ m/s} \).

The reduced coefficient of sliding friction \( f^* \) was taken as the investigated coefficient, the average
contact pressure \( p_{m} \) was taken as an independent coefficient.
Two series of experiments were carried out. In the first series, $f^\ast$ was defined in coupling of unrun surfaces, in the second series – in coupling of run-in surfaces. Before the second series of experiments, all specimens were run-in under load $p_m=0.5$ MPa for 15 hours.

During the experiment, the method of step pressure loading of the semi-ring from $p_m=0.5$ to 5.5 MPa per 1 MPa was used, and the friction torque $T_f$ in the coupling pair was measured. The loading was carried out to a pressure critical value at which signs of seizing appeared.

The experimental values of the reduced friction coefficients were found from:

\[
f^\ast = \frac{T_f}{nF_R}. \tag{9}
\]

4. Numerical Results and Discussion

Figure 3 shows a comparison of the experimental (solid lines) and theoretical (dashed lines) values of the reduced coefficient of sliding friction for unrun surfaces. The theoretical values of the reduced friction coefficients are obtained on the basis of formula (7), where $f$ was accepted from the experiment results of a semi-ring with a smooth cylindrical surface.

Figure 4 shows a comparison of the experimental (solid lines) and theoretical (dashed lines) values of the reduced coefficient of sliding friction for run-in surfaces. The value of the reduced friction coefficient was obtained on the basis of formula (8).

A comparison of the experimental and theoretical values of the reduced friction coefficients (figure 3 and 4) shows a fairly good coincidence of their qualitative and quantitative characteristics. The relative error for all specimens does not exceed 9.5\ldots18.7\% and only at one point – reaches 24\%.

This discrepancy is explained by the complexity of the accurate theoretical description of friction processes, especially under transient regimes [5–6].

5. Conclusion

The possibility of obtaining the wedge coupling effect by a simple and technological method - in contact with smooth cylindrical surfaces – is theoretically substantiated and experimentally confirmed.

It is shown that the use of such effect for a "steel-steel" friction pair with a real friction coefficient $f=0.04\ldots0.05$ [4–6], the value of the reduced friction coefficient increases 2.6\ldots5.7 times and can be $f^\ast=0.15\ldots0.38$ (for unrun working surfaces) and $f^\ast=0.10\ldots0.25$ (for the run-in working surfaces).
The proposed design solution providing an increase in the frictional force was used in eccentric free-wheel mechanisms, which made it possible to increase the reliability of their operation under conditions of pulsed loading [29].

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