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Improving Methods of Wear Resistance in Heavy Loaded Sliding Friction Pairs

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Improvement of wear resistance and durability of machine elements with sliding friction pairs is the important tribological problems. The wear resistance has been determined with many configurative parameters, technological parameters, and operational parameters. In this study kinematics of cylindrical joint (CJ), whose motion is reciprocating and rotating, and influence of various parameters on wear resistance of friction pair was investigated.

1. INTRODUCTION

Sliding friction joints (SFJ) are widely applied into machine elements such as sliding bearing, cylindrical and spherical joints, and so on. The types of joint are "roller-plate", "bush-core" and "plate-plate". Their contact conditions are line contact or plane contact with friction. Wear resistance of such contact pairs is defined with many configurative, technological and operational parameters. The principal friction parameters are L/D ration (L: bush length, D: shaft diameter), roughnesses of contact surfaces Rz, hardness of contact surface HB, design performance DP, clearance 2Δ, angle of oscillation α, frequency of oscillation ν, and linear amplitude of oscillation $L\alpha$. In addition the important operational factors are contact pressure p, sliding velocity V_S, and environment temperature T_0.

The sliding friction joints (SFJ) are operated in air or in special condition such as in vacuum. For the purpose of reducing friction between contact bodies, lubricants such as solid lubricants, some coatings, self-lubricating materials, and greases are used. The tribological characteristics of greases are improved by anti-frictional additives such as polytetrafluoroethlene, molybdenum disulfide, graphite and so on. Self-lubricating materials are reinforced with fibers and powders. The best performance of improving the wear resistance in heavy loaded sliding friction joints (SFJ) were given with solid lubricant coatings on base of fabric layer and polymeric materials [1]. Sliding friction joints (SFJ) were generally operated under heavy loads, low velocity and at high or low temperature, therefore solid lubricating materials are used for lubricant.

In order to secure the reliability of sliding friction joints (SFJ), what should be taken account by designers are as follows: (1) required configurative and technological parameters, (2) the numerical values of those parameters, (3) the influence of each configurative, technological, and

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Table 1 Technical characteristics of cylindrical joints in industrial fields

	Industrial brunches & machines	Operational parameters		
		p, MPa	V_S, m/s	T_0, K
1	Food, domestic & printing brunches	1-5	0.001-0.1 0.5*	243-313
2	Railway transport	5-30	0.001-0.1	233-333
3	Transporters & combine	5-20 40*	0.01-0.05 0.5*	213-333
4	Tractors, trailers & tow cars	10-100 140*	0.01-0.05	213-333
5	Cranes, conveyers & transporters in heavy industry	5-50 100*	0.01-0.1	213-333
6	Ship	20-90 150*	0.001-0.1	233-333
7	Aviation	5-80 100*	0.01-0.1 1*	183-353 393*
8	Space	5-100 140*	0.001-0.1 3*	143-453 573*

* : peak value of parameters

Table 2 Operational condition of cylindrical joint (CJ)

	Oscillation cycles*	Operational parameters		
		p, MPa	V_S, m/s	T_0, K
Light loaded	4000-5000	p<10	V_S<0.1	253-353
Heavy loaded	4000-5000	10<p<100	V_S<0.1	193-453
Extreme loaded	<5000	100<p	0.1< V_S	T_0<193 453<T_0

* : Number of oscillation cycles for one operation

operational parameter on tribological characteristics.

Several characteristics of cylindrical joints (CJ) used in industrial field are shown in Table 1. Table 2 shows the conventional grade of operational conditions of cylindrical joints (CJ). The cylindrical joints (CJ) have been widely operated under the "heavy-loaded" operational condition. In laboratory test the operation under "heavy-loaded" condition are difficult, therefore we have not so much data about the influence of parameters on tribological characteristics of sliding friction joints (SFJ). In this study tests have been carried out under "heavy-loaded" operational conditions.

2. EXPERIMENTAL PROCEDURE

The tests were performed with cylindrical joint test machine [2]. The working stand consisted of four working steel blocks. A pair of shaft and bush specimens was fixed in each steel block. The steel blocks were interchangeable. Figure 1 shows the schematic of test machine. The test machine can work in intermittent pressure p=1-100 MPa sliding velocity V_S=0.0002-0.05 m/s and operational

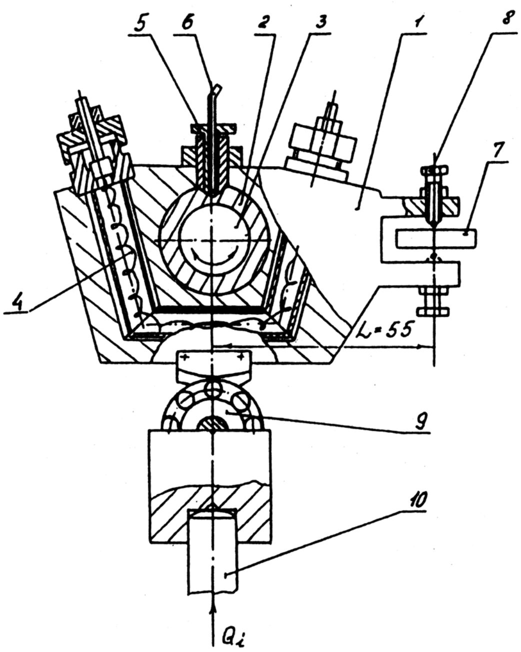
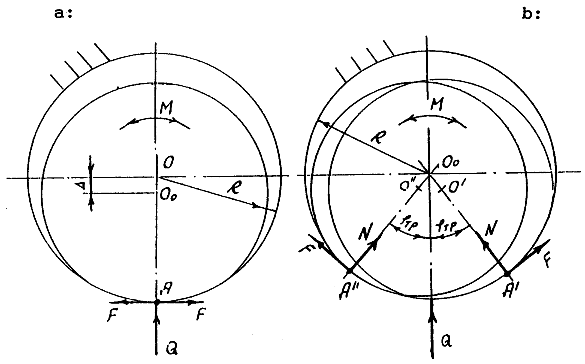


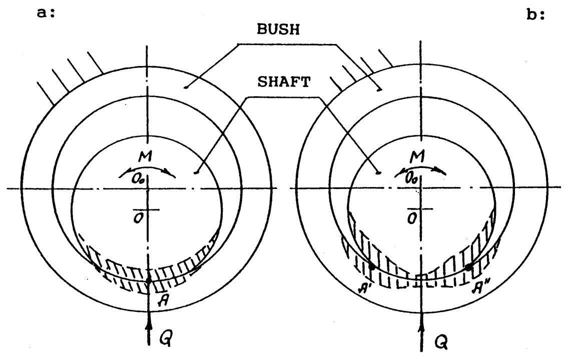
Fig.1 Schema of test apparatus

temperature T_0=173-532 K. Shaft and bush specimens were made of carbon steel with 0.45% carbon. The bush specimen (2) was fixed in the working steel block (1) by the fixture (5) in which a thermocouple (6) was connected. The shaft specimen (3) was fixed between two journal bearings. The friction pair (bush and shaft) was loaded by a rod of loading device (10) through a self-aligning double-row spherical ball bearing (9). In steel block a heater (4) was installed. The moment of friction is measured with tensometers (8). The signals of tensometers was transmitted to a tenso-intensifier, a oscillograph and a recorder. Tests were performed with solid lubricating coatings of FPF SLC consisting of epoxy resine and polyvinyl alcohol + frictin-polymer-forming



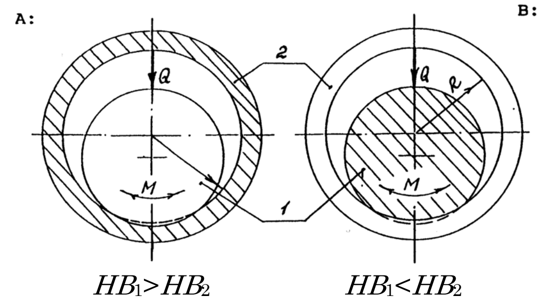
a) Single-area contact
b) Double-area contact

Fig.2 Kinematics of moving cylindrical joint (CJ)

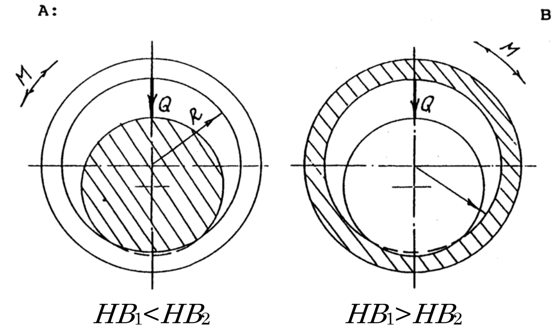


a) Single-area wear
b) Double-area wear

Fig.3 Projection diagram of wear



Design performance N1 (DP-1)



Design performance N2 (DP-2)

R : radius of bush
HB : Surface hardness
M : friction moment
Q : force
Subscript 1: shaft
Subscript 2: bush

Fig.4 Types of design performance (DP)

filler with a coating thickness h=200μm and thin-film coating of VNIINP consisting of MoS_2 + epoxy resin with a coating thickness h=20μm.

3. TEST RESULTS AND DISCUSSIONS

3.1 Kinematics and Design Performance of Joint

Kinematics of cylindrical joints (CJ) plays a significant role in wear forming. When the shaft turns against the fixed bush on some deflection angle by certain friction moment, the contact area of the shaft with the bush is transferred from the conventional equilibrium point (point A in Fig.2-a) to friction angle. In the beginning the shaft rotates against the bush without sliding and then the slide between the contacting surfaces starts. The angle of oscillation depends on the sliding velocity, friction coefficient, clearance between shaft and bush, and degree of freedom in the friction pair (rigidity of the system). Depending on those single zone contact of sliding and double zone contact of sliding (Figs.2 and 3), wear could be considered. The sliding distance of each element of pair is determined by design performance of the system. As shown in Fig. 4 there are four types of design performance (DP) of friction pair; the direct and inverse pairs where the shaft oscillates against the fixed bush, that is type DP-1, and the direct and inverse pairs where the bush oscillates against the fixed shaft, is type DP-2. Taking into account the kinematics of joint, the direct friction pair is the sliding friction joint, in which the harder sliding surface slides against the fixed softer surface. The inverse friction pair is the sliding friction joint, in which the softer sliding surface slides against the fixed harder surface. The sliding distance of each friction pair is different from the other one. The analyses of the unfolding-diagram of sliding distance S_f for one cycle of oscillation angle 2α in the cylindrical joint (CJ) indicates that in most cases the sliding distance of oscillating element is not equal to the sliding distance of non-oscillating element for one cycle of oscillation angle. The sliding distance of non-oscillating element would be larger than that of the oscillating element. Both thermal stress and wear of the sliding surface in the non-oscillating element would be higher than those in the oscillating element. Test results of friction pairs with different design performance indicated that wear of the inverse friction pair was greater than that of the direct pair at both room temperature and high temperature [3]. Note that when coating was employed on the oscillating surface, the term of running-in process decreased by 3-4 times in comparison with the case that the non-oscillating surface was coated.

The sliding distance during operation cycle can define the sliding distance of friction pair and the correct line wear intensity of lubricant material for each design performance. For a case of design performance DP-1, the sliding distance of shaft S_{f1} and the sliding distance of bush S_{f2} for one cycle of oscillation angle in the cylindrical joint (CJ) are designated as follows:

$$S_{f1}c = \begin{cases} 2\alpha R & : 0 < \alpha < 2\phi_0 \\ 4\phi_0 R & : 2\phi_0 < \alpha < 2(2\pi - \phi_0) \\ 2\alpha R - 8R(\pi - \phi_0) & : 2(2\pi - \phi_0) < \alpha < 4\pi \\ 8\phi_0 R & : 4\pi < \alpha < 2(4\pi - \phi_0) \end{cases} \quad (1)$$

$$S_{f2}c = 2\alpha R\,, \quad (2)$$

where

R : radius of bush

α : oscillation angle of cylindrical joint (CJ)

ϕ_0 : semi - angle of contact.

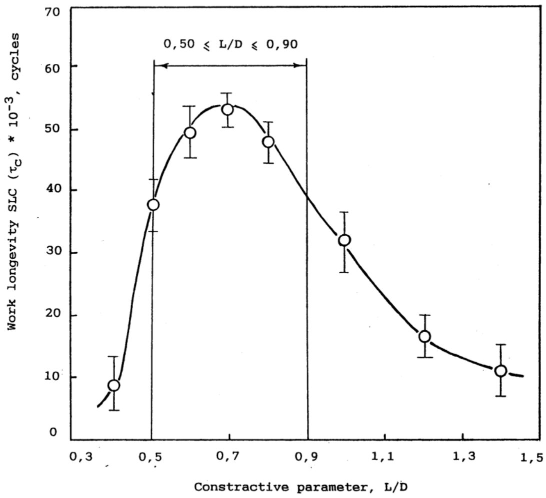
Sliding distance S_f for overall period of operation is

$$S_f = S_f c * N_{tc}\,, \quad (3)$$

where

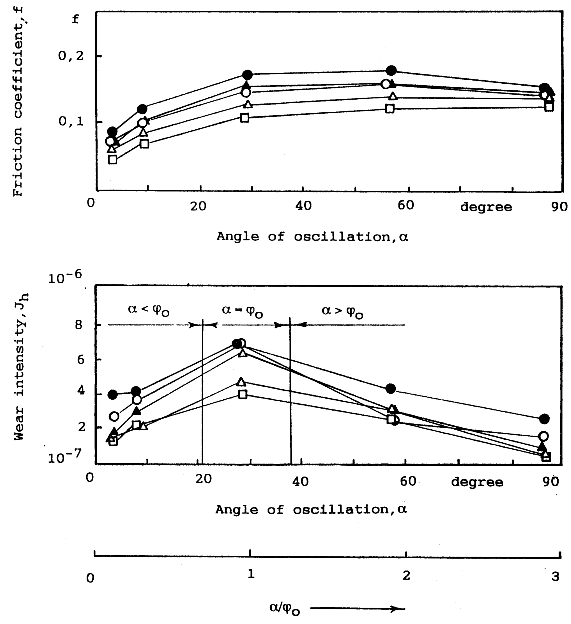
N_{tc}: number of total operation cycles of friction joint.

Parameter of line wear intensity J_h under settled friction region of contacting bodies is defined as:



SLC: MoS_2+epoxy resin (h=25μm)
p=40MPa, V_S=0.02m/s, T_0=293K

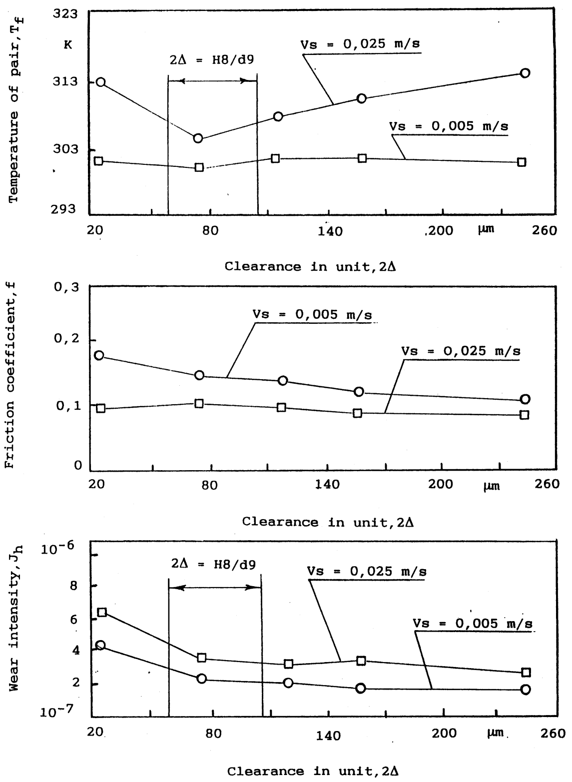
Fig.5 Relation between durability with solid lubricant coating (SLC) and configurative parameter L/D



● 2Δ=0.025mm ○ 2Δ=0.075mm
▲ 2Δ=0.112mm △ 2Δ=0.157mm
□ 2Δ=0.242mm

SLC: MoS_2+epoxy resin (h=25μm)
p=40MPa, V_S=0.005m/s, T_0=293K

Fig.6 Variation of friction coefficient f and wear intensity J_h



SLC: MoS_2+epoxy resin (h=25μm)
p=40MPa, V_S=0.005m/s, T_0=293K

Fig.7 Influence of clearance 2Δ on temperature T_0, friction coefficient f and wear intensity J_h

$$J_h = h_l / S_f \,, \tag{4}$$

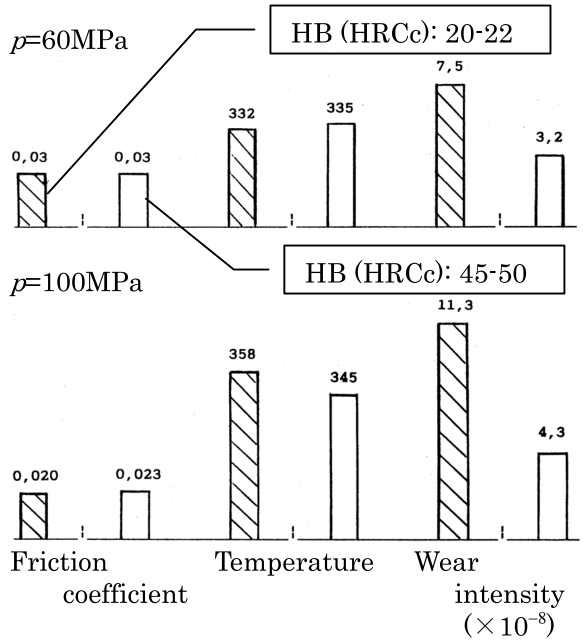
where h_l: wear of coated layer.

If we know these parameters and the thickness of anti-frictional material h and the frequency of operational cycles ν, we can determine work-time τ of lubricant material as follows:

$$\tau = \frac{h}{S_f \nu J_h} \,. \tag{5}$$

3.2 Influence of Constructional and Technological Parameters

Figure 5 shows the relation between durability of cylindrical joint (CJ) with solid lubricant coating against configurative parameter L/D. Test conditions were as follows: SLC–VNIINP, p=40 MPa, V_S=0.02 m/s, T_0=293 K. In L/D=0.5-0.9 the coated joint has high wear resistance. In L/D<0.5 the rigidity of the contact pair increased with increasing L/D. Figure 6 shows the variation of friction coefficient f and line wear intensity J_h of solid lubricant coating with oscillation angle α. In



SLC: TPF SLC (h=200μm)
V_S=0.02m/s, T_0=293K

Fig.8 Influence of clearance 2Δ on temperature T_0, friction coefficient f and wear intensity J_h

order to investigate the influence of the clearance 2Δ on tribological characteristics of cylindrical joint, tests were performed with SLC-VNIINP under p=40 MPa, V_S=0.005 m/s, and T_0=293 K. The wear intensity parameter has peak value in the oscillation angle α=20-40 degrees. The influence of oscillation frequency ν on wear was studied and great wear of friction pair was observed under the oscillation frequency when the oscillation angle α equal to the contact semi-angle ϕ_0. Judging from the kinematics of cylindrical joint (CJ) when α=ϕ_0 as shown in Figs.2 and 3, all points of sliding surfaces of both bush and shaft take part in friction and the wear particles did not go out from the contact area that led to great wear of coating.

Figure 7 shows the results tested under different values of clearance 2Δ, sliding velocity V_S. At high sliding velocity the minimum temperature was observed in 2Δ=60-110 μm and reduction of wear intensity of coating was also observed. That clearance would be suitable for a fitting of H8/d9 in less degree.

The influence of roughness parameter R_z of surfaces was studied with the coating FPF SLC under V_S=0.02 m/s, T_O=293 K. The results indicated that in all cases of pressure, friction coefficient, temperature of pair and line wear intensity increased remarkably in Rz> 1μm. Microcutting of coating material would take place in the cases.

In order to know the influence of surface hardness HB on tribological characteristics of friction joint tests were carried out with FPF SLC. As shown in Fig.8 the high hardness of substrate under the coating did not lead to the improvement of tribological characteristics of the friction pair. However, the wear rate of coating decreased with increasing hardness of the counter surface under p>60 MPa. Under low pressure p<40 MPa, the high hardness of the counter surface did not always led the reduction of wear rate of coating.

4. CONCLUSIONS

The following tribological recommendations for designing of sliding friction pairs were obtained.

1. Design performance of sliding friction joint (SFJ) should be selected similar to inverse friction pair. That is to say, the coating should be employed on moving surface when the sliding friction joint that the softer surface slides against the harder fixed surface is designed.
2. Configurative parameter L/D should be selected to L/D=0.5-0.9.
3. The clearance 2Δ should be selected to the fitting H8/d9.
4. Oscillation angle α for reciprocating-rotary motion of joint and linear amplitude $L\alpha$ for reciprocating motion of joint should be selected to be semi-angle contact for reciprocating- rotary motion of joint and semi-area contact for reciprocating motion of joint.
5. Roughness parameter Rz of surface should be Rz <0.9 μm.
6. In case under p>60 MPa, hardness of counterface should be more than 45 HRC.

REFERENCES

[1] Y.N.Drozdov, V.I.Klochikhin, F.G.Krymov: Proc. Int. Tribochem. Symp. Lanzhou, CHAS (1989), 223-229.
[2] V.I.Klochikhin: Friction and Wear (in Russian), 11-3 (1990), 480-489.
[3] V.I.Klochikhin: Thesis of Ph.D., IMASH, RAS, Moscow, (1991) 154.