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Influence mechanism of morphological parameters on tribological behaviors based on bearing ratio curve

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ABSTRACT

The influence mechanism of the contact surface morphology statistical parameters, such as S_a, S_q, S_{sk}, and S_{ku}, on the friction and wear properties has not been well explained or understood. This paper establishes formulas between the bearing area ratio and friction coefficient for dry and lubricated friction, investigates the relationship between the bearing area ratio curve and roughness parameters, and subsequently explains the influence mechanism of the roughness parameters on the friction and wear by the bridge of the bearing area ratio. Finally, dry and lubricated pin-on-disc tests were performed on samples with different roughness parameters, and the experimental results verify that the established formulas and the explanation of the influence mechanism are feasible and reasonable.

1. Introduction

In the primary stage of friction and wear, the surface morphology has a significant impact. Surface morphology refers to the geometry of the object surface, broadly speaking, including the roughness, waviness, shape errors and texture. Machinery and equipment wear, lubrication, friction, vibration and noise, fatigue, sealing, and assembly properties are all related to the surface morphology of contact. Therefore, the study of the influence of the surface topography on the friction and wear behavior has great significance.

Most studies of the effect of the surface morphology on friction and wear have been focused on surface modification, coating, and the surface texture [1–3]. There have been many studies analyzing the influence of the surface roughness for lubrication, friction and wear [4–6], while studies of the correlation between the surface topography statistical parameter of the asperity height distribution and the friction and wear are few. In [2], where the effect of the surface roughness parameters on the mixed lubrication characteristics was investigated by modeling, it was found that the skewness and kurtosis have a great effect on the contact parameters of mixed lubrication. Sedlaček M, Podgornik B, and Vižintin J [8,9] conducted dry and lubricated pin-on-disc tests under different conditions with 100Cr6 steel plate samples with different average surface roughnesses, and they obtained experimental results correlating the roughness parameters (R_a, R_q, R_{sk}, R_{ku}) and the friction and wear. Tayebi [10] adopted the Pearson distribution, based on the CEB model, and they discussed the effects of R_{sk} and R_{ku} on the static friction coefficient, determining that surfaces with a high kurtosis and positive skewness exhibit a lower static friction coefficient compared to the Gaussian case. Although much experimental work has been performed in the field of surface roughness and the topography of contact surfaces with respect to the correlation between the surface roughness and friction [7–11], the mechanism of the effect of the contact surface topography statistical parameters on the friction and wear has not yet been well explained or understood.

For this reason, the present research establishes functions correlating the dry or lubricated friction coefficient and the bearing area ratio. It analyzes the relationship between the contact surface asperity statistical parameters and the bearing ratio curve, and confirms by experiment that by using functions of the dry or lubricated friction coefficient and bearing area ratio, the mechanism of the effect of the contact surface topography statistical parameters on the friction and wear would be understood more easily, and the function formula correctness would be verified.

2. Functions of dry or lubricated friction coefficient and bearing area ratio

2.1. Function of dry friction coefficient and bearing area ratio

According to the classical friction laws, the frictional force is directly proportional to the normal load. At this constant of proportionality, the friction coefficient is constant and independent of the apparent contact area, surface roughness, and sliding velocity, but dependent on the material properties [13].

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However, a large number of experiments have shown that, in the early stage of friction and wear, the friction coefficient is greatly affected by the surface topography parameters, and it is closely related to the real contact area. After a systematic experimental study, Bowden and Tabor [14] put forward a more complete theory of adhesive friction, given by

$$F_f = A_r \tau_s + A_p \tau_p \tag{1}$$

where A_r and A_p denote the real contact area and the furrow area, respectively, τ_s is the contact force of a unit area, and τ_p is the furrow force of a unit area.

The coefficient of friction has nothing to do with the nominal contact area, but is related to the real contact area. The peak of most of the metal surfaces, $\frac{A_p}{A_r}$, is very small, so that the second term $A_p \tau_p$ can be neglected, which then gives us

$$F_f = A_r \tau_S \tag{2}$$

$$B(z) = \sum_i A_{ri}(z)/A \tag{3}$$

where $B(z)$ is the bearing area ratio, z is the height of the rough particles, $A_{ri}(z)$ is the real contact area of the number i peak at a certain height, and A is the nominal area.

Substituting Eq. (3) into Eq. (2), we have

$$F_f(z) = \tau_s \sum_i A_{ri}(z) = \tau_s{}^*A^*B(z) \tag{4}$$

Therefore, the function of the dry friction coefficient and the bearing area ratio is established:

$$\mu(z) = \frac{F_f}{F_n} = \frac{\tau_f{}^*A^*B(z)}{F_n} \tag{5}$$

Eq. (5) shows that the dry friction coefficient has a positive correlation with the bearing area ratio.

2.2. Function of the friction coefficient of mixed lubrication and the bearing area ratio

An important feature of mixed lubrication is that most of the normal load is shared by the lubricant film, so that the friction coefficient of the mixed lubrication is given by

$$\mu = \frac{F_f}{F} = f_h \frac{F_h}{F} + f_c \frac{F_c}{F} \tag{6}$$

where, f_h and f_c denote the friction coefficients of the EHD film and asperity contact, respectively, F_h is the EHD film load, and F_c is the asperity contact load.

According to Tallian's calculation model of an elastohydrodynamic lubrication film [15] and assuming that the lubrication film is in accordance with the Hertz rules,

$$F_h = \frac{2}{\pi} p_{\max} A(1 - A_c/A) = \frac{2}{\pi} p_{\max} A[1 - B(z)] \tag{7}$$

where $p_{\max}$ is the maximum pressure of the lubrication film.

For a completely elastic contact, the summit contact load is

$$F_c = \frac{E'}{4\pi^2} A\sigma_\theta I\left(\frac{h}{\sigma}, \alpha\right) = \frac{E'}{2\pi} NA \int_h^\infty (z - h)\phi(z)dz \tag{8}$$

The asperity height distribution is $F(z) = 1 - B(z)$, so that the asperity height distribution density is

$$\phi(z) = \frac{dF(z)}{dz} = \frac{d[1 - B(z)]}{dz} = -B'(z) \tag{9}$$

Substituting Eq. (7) and Eq. (8) into Eq. (6) and using Eq. (9), we will obtain the friction coefficient of the mixed lubrication under a completely elastic asperity contact as

$$\mu_1(z) = f_h \frac{2}{\pi F} p_{\max} A[1 - B(z)] - f_c \frac{E'}{2\pi F} NA \int_h^\infty (h - z)B'(z)dz \tag{10}$$

For the completely plastic asperity contact, the asperity contact load is

$$F_c = \sigma_s A \frac{B(z)}{1 - B(z)} \tag{11}$$

Substituting Eq. (7) and Eq. (11) into Eq. (6), we can obtain the friction coefficient of the mixed lubrication under a fully plastic asperity contact

$$\mu_2(z) = f_h \frac{2}{\pi F} p_{\max} A[1 - B(z)] + f_c \frac{\sigma_s A}{F} \frac{B(z)}{1 - B(z)} \tag{12}$$

In fact, the asperity deformation includes both elastic and plastic aspects. The real friction coefficient should be somewhere between μ_1 and μ_2 when $\frac{h}{\sigma} \geq 1$, and the partial elastohydrodynamic lubrication film bears most of the load [15]. Christensen calculated the ratio between the total load and fluid load for plane slider bearings under the mixed lubrication condition, assuming the asperity contact to be fully plastic. As the ratio of the minimum nominal film thickness to the maximum height of the peaks at the slider outlet approaches zero, the lubrication film can still withstand a considerable load [16]. Therefore, $\frac{F_c}{F}$ becomes weak. Under certain conditions, if the asperity contact friction coefficient is neglected, Eq. (6) can be simplified as

$$\mu(z) \approx f_h \frac{2}{\pi} p_{\max} A[1 - B(z)] \tag{13}$$

Eq. (13) shows that the mixed lubrication coefficient has a negative correlation with the bearing area ratio.

3. Experiments

To verify the correlation between the dry or lubricated friction coefficient and the bearing area ratio when it is based on Eq. (5) and Eq. (13), the mechanism of the effect of each of the statistical topography parameters on the friction and wear performances must be analyzed, and the friction and wear results of the experimental samples with different roughness parameters should be compared for analysis.

3.1. Materials and methods

To obtain samples of diverse surface topography parameters with the same surface material asperity by a mechanical material removal method, No. 45 steel disc samples were prepared with different average surface roughnesses using different grades of grinding, polishing, turning and milling. Usually, turning produces a high-peaked surface with positive skewness, while grinding or milling processes produce grooved surfaces with a negative skewness and high kurtosis value [7]. The HRC of No. 45 steel is 42–46, and it more easily undergoes wear than 100Cr6. The dimensions of the steel plates were Φ28 mm×10 mm (thickness). It was guaranteed that the upper and bottom surfaces had a higher degree of parallelism to reduce a potential source of error.

Surface topography measurements were performed by a TALYSURF CLI 1000 (Taylor-Hobson, England) in contact mode. The sampling interval, orientation and location of the measurement had some influence on the surface topography parameters values. The results of the samples prepared by turning were significantly affected by the direction and location of the measurement. The surface pattern produced by the turning tool was virtually concentric around the center of the specimen. The farther away the pattern was from the center, the greater the cutting line speed, which caused changes in the surface roughness. The surface pattern produced by grinding was unidirectional in nature, parallel and vertical. The samples with a larger interval of measurement exhibited significant effects on the measurement results. There was no significant influence on the roughness parameter

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