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Wear of Electrical Contacts due to Small-amplitude Motion

Abstract: The "IBM wear model," which has previously been used to describe the wear of electrical contacts as a result of gross sliding motion, is used to describe contact wear in the case of small-amplitude sliding motion. This model was applied to a particular contact configuration on which a series of wear tests was performed. The empirical results of the tests are compared to the theoretical expressions and good agreement is found, thus indicating that the wear mechanism is essentially the same for both large- and small-amplitude sliding contact.

Introduction

In two previous papers [1, 2] a model was developed that describes the wear of electrical contacts produced by the gross sliding motion that occurs during the "making" and "breaking" of the contacts. Recently it has been found that nominally stationary contacts can also wear as a result of small-amplitude sliding produced by random vibration transmitted through the machine frame. These two modes of wear can be contrasted in the following manner. In the wear associated with the making and breaking of the contact, the magnitude of the relative sliding motion is greater than the length of the apparent contact area. In the wear produced in the nominally stationary condition, the amplitude of the relative sliding motion is less than the length of the apparent contact area. This latter type of wear may be considered wear under what is usually termed a "fretting" condition, and the former, wear under gross sliding conditions.

While a quantitative difference exists in the motion associated with these two modes of wear, the same basic wear mechanisms should be present in both cases [3]. Consequently, it is reasonable to expect that the same basic formulation used to develop the model for wear due to gross motion can be used to describe wear produced by small-amplitude motion. In this paper, such an approach is developed and the results compared with experimental data.

Theory

The model used to describe the wear of electrical contacts under gross sliding conditions is known in technical literature as the "IBM wear model" [4–7]. In this model the wear in a sliding system is governed by one or the other of the following equations:

$$dQ = CdN \tag{1}$$

or

$$d[Q/(\tau_{\max} W)^{9/2}] = CdN, \tag{2}$$

where Q is the maximum cross-sectional area of the wear scar taken in a plane perpendicular to the sliding direction; $\tau_{\max}$ is the maximum shear stress produced during sliding; W is the length of the apparent contact area taken in the direction of sliding; C is a constant dependent for a given system on the materials and lubricants used; and N is a measure of the amount of sliding. Specifically, N is the number of passes of sliding, where a pass is a distance of sliding equal to W.

In this formulation of the wear process, the nature of the predominant wear mechanism determines which equation is to be applied to a given system. For a wear process in which the energy going into wear remains constant as wear progresses, Eq. (1) is the appropriate equation. Equation (2) applies to a system in which the

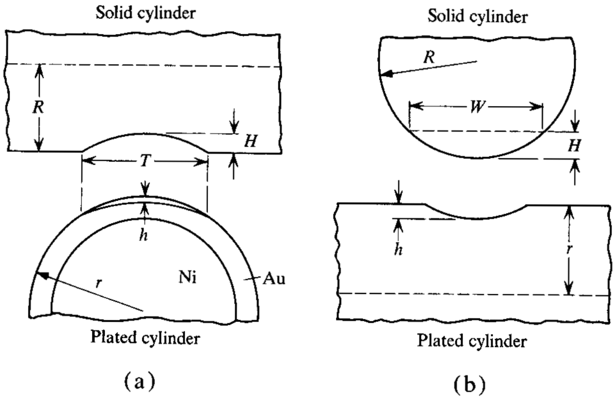


Figure 1 Diagram illustrating wear scars: (a) view in a plane perpendicular to the sliding motion; (b) view in a plane parallel to the motion and perpendicular to the axis of the solid cylinder.

energy varies as the stress level changes as a result of wear [8]. In Refs. 1 and 2 it was shown that the wear of unlubricated gold contacts is usually described by Eq. (1), while that of lubricated gold contacts frequently obeys Eq. (2). In general, the appropriate equation for a given system must be identified empirically. In a system in which both parts wear, two differential equations must be used to describe the wear, one for each part.

The dependence on motion in this formulation is associated with the determination of N. The relation can be described as follows. In most applications of these equations it is usually desirable to relate wear to some engineering factor, such as number of cycles, number of insertions and withdrawals of a contact, hours of operation, etc., rather than to the number of passes. To do this, the number of passes equivalent to such a unit of usage must be established. Mathematically, this conversion is made by means of the following expression:

$$dN = (w/W)\, dL, \tag{3}$$

where w is the total distance of sliding during a usage unit L. Substituting Eq. (3) into Eqs. (1) and (2) we obtain

$$dQ = C(w/W)\, dL \tag{4}$$

and

$$d[Q/(\tau_{\max} W)^{9/2}] = C(w/W)\, dL, \tag{5}$$

which are the equations used in engineering analysis.

Our hypothesis is that the same basic wear mechanisms are common to both modes of wear. Therefore we use this formulation to describe the wear of an electrical contact system for both gross and small-amplitude sliding. The difference in the amount of sliding is reflected in different values of w for the two modes, as can be illustrated by the following example.

Consider body A sliding back and forth across body B, for a total distance of travel S in a single cycle. Let L be the number of cycles. Then if $S/2$ is greater than W, $w = S$ for body A. For B, $w = 2W$ since each element of B experiences sliding only as A goes by. If $S/2$ is less than W, then for body A, $w = S$; for body B, $w = S$ also. Hence there is a difference in the values of w for gross and for small-amplitude sliding, and the equations have a means of taking into account the difference in the motions.

Experimental procedure

To explore the assumption of a common wear mechanism, a series of wear experiments was performed using the modified Bowden–Leben apparatus [1, 2, 4], which provides an oscillatory motion with a controlled amplitude. Three series of tests were performed on a particular contact configuration using the same material and the same load for each. These series differed in the amplitude of the sliding motion. In one series it was 0.002 in.; in the second, 0.010 in.; and in the third, 0.100 in.

The contact configuration consisted of two cylinders crossed at 90°; one cylinder had a radius of 0.028 in. and the other, 0.014 in. The smaller cylinder slid back and forth across the larger. The smaller cylinder consisted of a piece of nickel wire coated with approximately 125 μin. of gold plating (Vicker's microhardness 85 kg/mm^2). The larger cylinder was solid gold alloy with a microhardness of 75 kg/mm^2.

The tests were conducted under lubricated conditions (Monsanto OS-124 lubricant) and a normal load of 275 g. Wear measurements were made at various intervals from several hundred cycles of operation to over a million cycles. In general, several duplicate experiments were performed for each series. The dimensions of the wear scars were determined by means of both an optical microscope and a profilometer. The contact configuration, apparatus and techniques used in this investigation are the same as those discussed in Refs. 1 and 2.

Analysis and results

The approximate configuration of the wear scars was the same for all cases; a cylindrical groove was worn in the solid gold cylinder, while the plated cylinder wore to a larger radius. Typical scars are illustrated in Fig. 1 and photomicrographs are shown in Fig. 2. For such scars it has been shown [1, 2] that

$$Q^*_{\mathrm{S}} \approx T^3/16r, \tag{6}$$

$$Q^*_{\mathrm{D}} \approx Th/2, \tag{7}$$

$$W \approx T(R/r)^{\frac{1}{2}} \tag{8}$$

and

$$\tau_{\max} W \approx P(0.25 + \mu^2)^{\frac{1}{2}}/\pi T, \tag{9}$$

where P is the normal load, μ is the coefficient of friction and the Q^*'s are the cross-sectional areas of the scars taken in a plane perpendicular to the sliding direction; Q^*_s is the scar area for the solid cylinder and Q^*_p for the plated cylinder. In these expressions T is primarily a measure of the wear for the solid cylinder, while h is a measure for the plated cylinder. Although the Q^*'s represent the total cross-sectional areas of these scars, they do not necessarily represent the areas produced by wear. This is because the yield point of the gold used in these tests is exceeded when the load is first applied [9]. Therefore, the initial plastic flow contributes to the overall scar. If Q_{0s} and Q_{0p} are the areas associated with this plastic deformation, the areas associated with wear are

$$Q_s = Q^*_s - Q_{0s} \tag{10}$$

and

$$Q_p = Q^*_p - Q_{0p}. \tag{11}$$

For the materials used in this investigation, the plated cylinder is harder than the solid cylinder, so it is reasonable to expect that the initial plastic flow will occur mainly in the solid cylinder and will have the same shape as the wear scar there. Hence

$$Q_{0s} \approx T_0^3/16r \tag{12}$$

and

$$Q_{0p} \approx 0, \tag{13}$$

where T_0 is the width of the groove produced by the plastic deformation. Substituting for the Q_0's in Eqs. (10) and (11), we obtain the following expressions for the wear:

$$Q_s \approx (T^3 - T_0^3)/16r \tag{14}$$

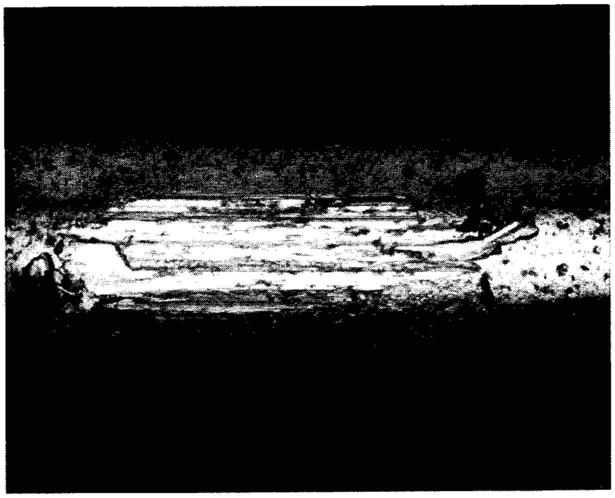
and

$$Q_p \approx Th/2. \tag{15}$$

For the purpose of these experiments it is convenient to measure usage in terms of cycles of operation and to define ϵ as the total distance traveled in a cycle. Hence $\epsilon/2$ would be the amplitude of sliding associated with the cycle. For the case in which $\epsilon/2 > W$, for the solid cylinder $w = \epsilon$; for the plated cylinder $w = 2W$. For the case in which $\epsilon/2 < W$, $w = \epsilon$ for both cylinders.

Having defined the Q's, $\tau_{\max}$, w and L, we can now integrate Eqs. (4) and (5) for this particular system. For the sake of brevity, however, only Eq. (5) is used, for it will be seen later that it is this equation which applies to the system. Therefore, by substitution of Eqs. (8), (9), (14) and (15) and the expressions for w into Eq. (5), the following equations result for the two members.

(a)



(b)

Figure 2 Typical photomicrographs of wear scars (110×): (a) scar on the solid cylinder; (b) scar on the plated cylinder.

For $\epsilon/2 > W$:

$$d(hT^{11/2}) = \frac{4(0.25 + \mu^2)^{9/4} P^{9/2}}{\pi^{9/2}} C_p\, dL, \text{ plated cylinder;} \tag{16}$$

$$Td[T^{9/2}(T^3 - T_0^3)] = \frac{16(0.25 + \mu^2)^{9/4} P^{9/2} r^{3/2} \epsilon}{\pi^{9/2} R^{1/2}} C_s\, dL, \text{ solid cylinder.} \tag{17}$$

For $\epsilon/2 < W$:

$$Td(hT^{11/2}) = \frac{2(0.25 + \mu^2)^{9/4} P^{9/2} r^{1/2} \epsilon}{\pi^{9/2} R^{1/2}} C_p\, dL, \text{ plated cylinder;} \tag{18}$$

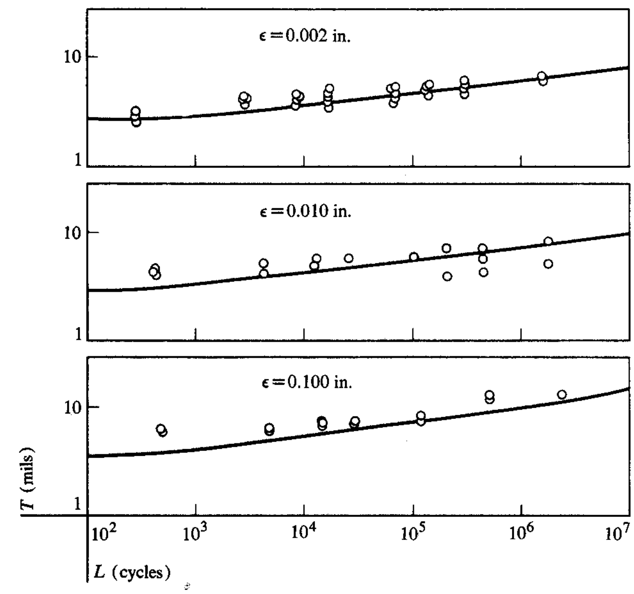
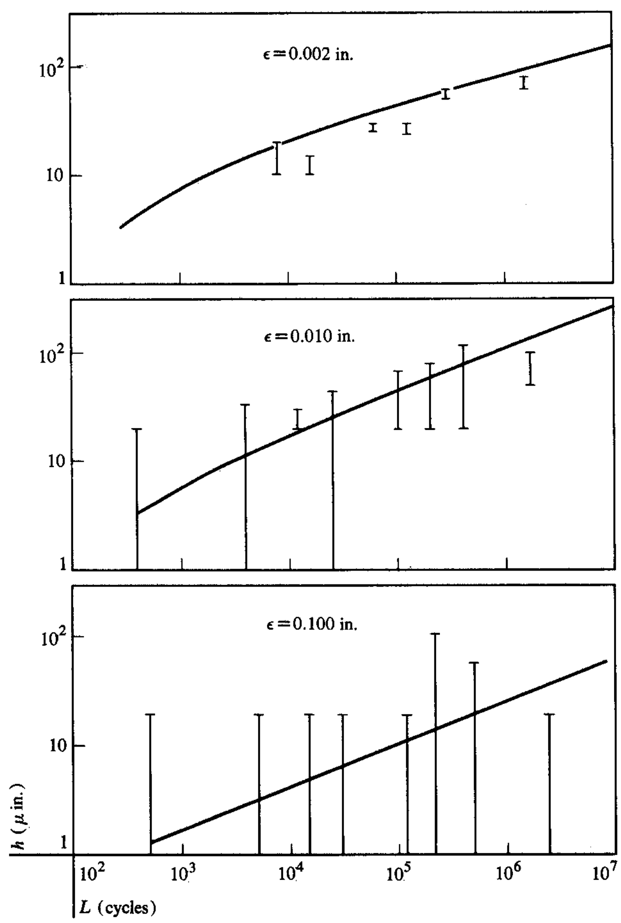


Figure 3 Variation of T with number of cycles.

Figure 4 Variation of h with number of cycles.



and Eq. (17) is again applicable for the solid cylinder. In these expressions C_p and C_s are the wear constants of Eq. (5), appropriate to the plated and solid cylinders, respectively.

These equations can be integrated by the techniques given in Refs. 1 and 2. For Eq. (17), i.e., for the solid cylinder in both cases, this integration results in the expression

$$T^{17/2}[1 - 0.926(T_0/T)^3 - 0.737(T_0/T)^{17/2}] = \frac{0.104(0.25 + \mu^2)^{9/2} P^{9/4} r^{3/2} \epsilon}{R^{1/2}} C_s L. \tag{19}$$

Depending on the ratio of $\epsilon/2$ to W, the following expressions are obtained for the plated cylinder:

$$h = \frac{T^3[0.883 - 0.818(T_0/T)^3 - 0.065(T_0/T)^{17/2}]R^{1/2}C_p}{4R^{3/2}\epsilon C_s}, \qquad \epsilon/2 > W; \tag{20}$$

$$h = \frac{T^2[1 - (T_0/T)^3]C_p}{8rC_s}, \qquad \epsilon/2 < W. \tag{21}$$

In Figs. 3 and 4 the experimental results of the test are given for the three different stroke lengths. The solid curves in these figures represent the appropriate theoretical expressions. For Fig. 3(a) they are Eqs. (19) and (21); for Figs. 3(b) and 3(c), Eqs. (19) and (20). These curves were obtained using a value of 2.94×10^{-19} in.$^{13/2}$-lb$^{9/2}$-cycle for C_p, 1.19×10^{-18} in.$^{13/2}$-lb$^{9/2}$-cycle for C_s, 0.60 for μ and 0.00025 in. for T_0.

The values of C_s and C_p were determined by fitting the experimental data at 3×10^5 cycles for the 0.002-in. stroke. The value of the coefficient of friction represents an average value obtained by performing auxiliary tests specifically to determine the coefficient of friction.

The value of T_0 was determined by assuming that the initial area of contact between the two cylinders was equal to the value of the normal load divided by the value of the hardness of the solid gold. This assumption was supported by measurements of the wear scar produced for one and two cycles of sliding.

In Figs. 3 and 4 the data for T represent values obtained from different tests. In the case of h, however, extreme values rather than specific values for each test are given. This is because the scars on the plated cylinder were small and quite irregular. Since it was extremely difficult to measure precisely the actual depth of the wear scar, upper and lower bounds for the wear scar were determined instead. The values shown represent the extremes not just for a single test, but for all the tests conducted.

In the case for which $\epsilon/2 = 0.100$ in., it is evident in Figs. 3(c) and 4(c) that there is poor agreement beyond 1.5×10^5 cycles. The reason for this is that the solid gold cylinder was not a complete cylinder, but rather a portion of cylinder mounted on some spring material, and at 1.5×10^5 cycles the gold was worn through and the spring material exposed. Consequently, after this point the nature of the system changes and good agreement would not be expected.

Discussion

While there is reasonable agreement between theory and experiment, the analytical expressions Eqs. (19), (20) and (21) are quite cumbersome for practical use. However, under normal engineering conditions it is usually found that for the magnitude of wear that is of concern, $T_0/T \ll 1$. If advantage is taken of this fact, simplified asymptotic expressions can be developed.

For example, consider the case of small-amplitude sliding, i.e., $\epsilon/2 < W$. In such a case, Eqs. (19) and (21) can be reduced to the following for $T_0/T \ll 1$:

$$T \approx K_s(\epsilon L)^{2/17}; \tag{22}$$

$$h \approx K_p(\epsilon L)^{4/17}, \tag{23}$$

where all constants for the system, i.e., load, radii, C's, etc. have been grouped into the K's. These expressions are now more tractable for engineering application.

These asymptotic expressions also clearly indicate that for small-amplitude motion, it is the total distance of sliding that is important, not specifically frequency, amplitude or time. This can be seen by noting that

$$\epsilon L = 2a\nu\phi, \tag{24}$$

where a is the amplitude of the oscillation, ν is the frequency and ϕ is the time. Consequently, if the product of these quantities is constant, the wear is the same.

Conclusions

From this study we conclude that the IBM wear model can provide a basis for describing the wear of electrical contacts as a result of small-amplitude motion. In addition, when the results obtained in this study are combined with the results obtained in previous studies, it is evident that this model provides a continuous means of describing the wear of such contacts for both small-amplitude and gross sliding conditions.

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