Tribology International 102 (2016) 1–9

Contents lists available at ScienceDirect

Tribology International

journal homepage: www.elsevier.com/locate/triboint

An adaption of the Archard equation for electrical contacts with thin coatings

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ARTICLE INFO

Article history:
Received 7 December 2015
Received in revised form
4 May 2016
Accepted 5 May 2016
Available online 13 May 2016

Keywords:
Sliding
Ploughing
Numerical analysis
Indentation

ABSTRACT

For the wear resistance of sliding contacts geometry, contact force and coating system are strong factors of influence. In this paper a mathematical description of the available wear volume of frequently used hemispherical and U-type geometries is given. Smooth surface layers are worn abrasively and adhesively. In the first stage the flat geometry becomes ploughed, in the second stage adhesive wear dominates. The hardness of the surface depends on the substrate, underlayer and finish layer. A wear model based on a flat distribution of the Archard work density with consideration of the abrasive wear and the depth depending hardness for hemispherical and U-type geometries is derived. With theoretical and experimental methods an optimization of the Archard equation is discussed.

1. Introduction

An electrical connector is usually made of at least two contacts and an insulating case. There are different contact concepts, an often used concept is the combination of a rider and a flat contact (Fig. 1). For the rider contact different contact geometries are used like hemispherical and U-type geometries. The contact force is usually generated by an elastic deformation of the rider contact. In Fig. 1 a principal drawing for a rider/flat combination with the contact force F_N, the friction force F_R and the insertion length l is shown.

The contact resistance is the most important parameter for evaluating the contact quality, because a high contact resistance results in a high current heating. Holm found the fundamental connection between the contact force and the constriction resistance:

$$R \propto F^{-n} \tag{1}$$

With the exponent $n = 1/3$ for elastic deformation and $n = 1/2$ for plastic deformation [1].

If non-noble metals are used, the deformation behaviour of the coating affects the contact resistance [2]. If there is plastic deformation, the oxide layer can be broken through, more micro-asperities can be formed. For example, for tin layers a dependence on the contact geometry can be predicted, because different contact pressures would occur. So low contact radii would result in a better contact resistance performance, but the wear of the surface would be higher. Connectors are usually worn by mating cycles and vibrations. Wear and friction depend on factors like: substrate, coating system, topography, contact geometry, contact force, lubrication, environment.

There are basically four different wear mechanisms:

- Adhesive wear
- Abrasive wear
- Disruption
- Tribooxidation

Often various wear mechanisms occur at the same time. For example, for relative thick tin layers of about 10 µm a wear mechanism with abrasive wear in the first stage and adhesive wear in the second stage, when the rider contact penetrated the flat contact, is predicted [3,4].

2. Theoretical analysis

2.1. Strategy

The Archard equation is a simple model to describe adhesive wear with the available wear volume V, the contact force F, the wear track x, the hardness H and the dimensionless wear

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http://dx.doi.org/10.1016/j.triboint.2016.05.005
0301-679X/

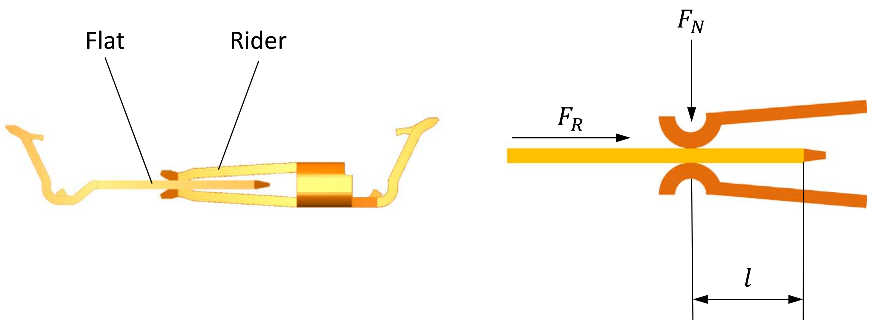

Fig. 1. Principal drawing of a rider/flat combination.

coefficient k [5,6]:

$$V = k \cdot \frac{F \cdot x}{H} \quad (2)$$

Gallego et al. presented a semi-analytical model to calculate fretting contact by computing [7]. McColl et al. developed finite-element models based on the Archard equation to calculate fretting wear [8,9]. The aim of this paper is to develop a simple wear model by using the Archard equation, that can be handled without special equipment.

Basseville et al. published a numerical three body wear model by utilizing the Archard equation. Wear particles are known for influencing the wear drastically [10]. The third body effect will not be considered in the following considerations, the model will be designed for materials that do not form wear debris.

Fouvry et al. have examined ceramic tribocouples under fretting conditions and have confirmed the linear decrease of the wear rate with the coating hardness [11]. For thin layers the hardness is also not constant, it depends on the surface thickness and the underlayer or substrate material. Equations for adhesive and abrasive wear for non-constant hardness data are not available yet and will be developed in this paper.

The following strategy for developing a lifecycle model will be pursued:

1. A mathematical analysis of the available wear volume of rider and flat geometries should be given.
2. The wear process should be divided and described in two stages:
   - Abrasive wear
   - Adhesive wear after abrasive wear finished
3. A lifecycle model based on the Archard equation with combining both wear stages should be developed.

2.2. Calculation of available wear volumes

Fig. 2 is an abstraction of spherical contact geometries with the outer radius r and the surface thickness s. For the calculation of the available wear volume for rider contacts with hemispherical and U-type geometries the equation for the semi-circle (Eq. (3)) is used. For the rider contact is defined that planar wear dominates and that the contact is worn when the substrate or underlayer is reached.

$$y = \sqrt{r^2 - x^2} \quad (3)$$

2.2.1. Available wear volume for hemispherical riders

The available wear volume for hemispherical riders (Fig. 3) can be estimated from Eq. (4). Hoppe got the equation in a similar

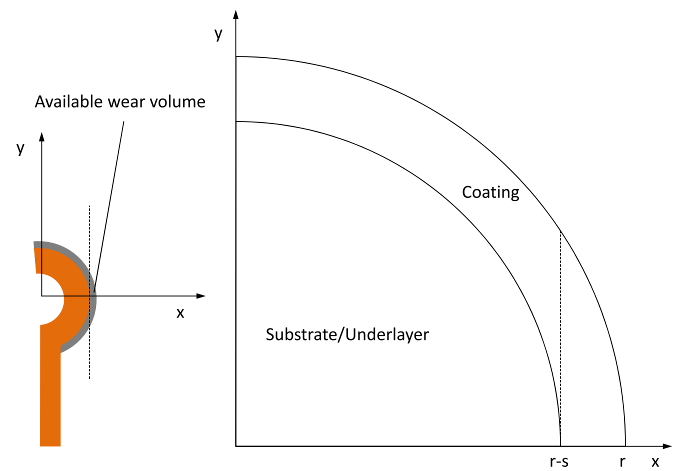

Fig. 2. Abstracted contact geometry.

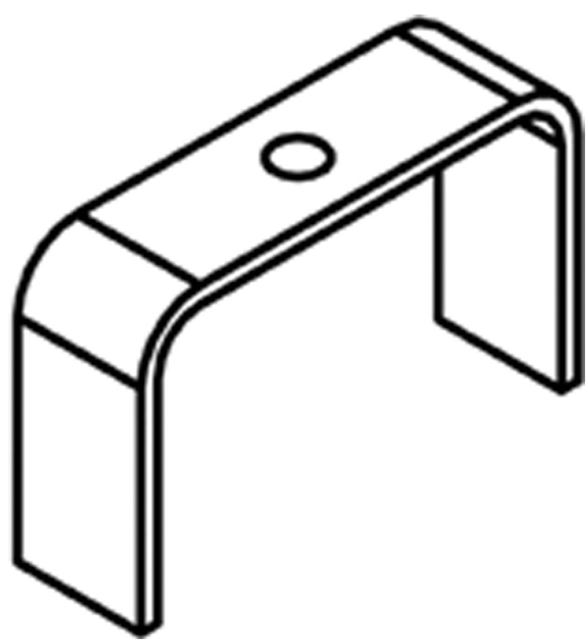

Fig. 3. Drawing of a hemispherical rider.

way [12].

$$V_h = \pi \int_{r-s}^{r} \left(\sqrt{r^2 - x^2}\right)^2 dx \quad (4)$$

Integrating from r to $r-s$ yields Eq. (5).

$$V_h = \pi \left(rs^2 - \frac{1}{3}s^3\right) \quad (5)$$