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Modelling of vibrations excited by formation and shearing of adhesive micro-junctions during sliding

B.L. Stoimenov and T. Mukai
Biologically Integrative Sensors Lab
Bio-mimetic Control Research Center, RIKEN
2271-130, Anagahora, Shimoshidami, Moryama-ku,
Nagoya 463-0003, JAPAN
E-mail: stoimenov@bmc.riken.jp

Abstract

Understanding of the mechanism of "rubbing" noise and low amplitude friction exited vibration generation in steady sliding can be helped by models describing the contact interactions. In the current paper we consider a simple microscopic contact model for surfaces in sliding, which is based on the adhesion theory of friction. In the proposed model, we consider that the formation and shearing of a junction contributes to a small change in the real contact area. The model incorporates random size and random spacing between junctions. We investigate the dependence of the instantaneous real contact area on the average size and number of junctions. We find that from the viewpoint of vibration reduction, it is advantageous if the real contact area needed to support given load is obtained as a sum of many small-sized micro-contacts, instead of few large-sized micro-contacts. The above result is in agreement with experimentally observed reduction of vibrations of a hard-disk slider after texturing.

1. INTRODUCTION

Friction-induced vibrations are a common occurrence in various kinds of machines, which have sliding components. Squealing brakes and cutting tool chatter are just two examples of large amplitude vibrations. Low amplitude friction exited vibrations are evidenced by the "rubbing" noise (Yokoi & Nakai, 1979; Stoimenov et al., 2004; LeBot et al., 2005) which can be heard and by friction force fluctuations, which can be measured in "steady sliding". Understanding the generation mechanism of "squeal noise" and "rubbing noise" can be helped by models describing the contact interactions.

In the past, various models have been proposed. Most models consider the dependence of the friction force on a macroscopic parameter. By far the most often used is the velocity dependence of friction, with the first such model used by Block (Block, 1940). Another macroscopic model is the one proposed by Oden and Martins, which defines normal and tangential interface laws (Oden & Martins, 1985). While, such models are well-suited to describe the phenomenon of stick-slip and squeal generation (Block, 1940; Tworzydlo, 1994), they are not appropriate for describing small perturbations around the steady sliding equilibrium position (Tworzydlo, 1994).

For understanding the mechanisms behind "rubbing noise" and micro-vibration generation in sliding, it is appropriate to consider the microscopic level contact interactions between the surfaces. Probably the best known microscopic model of contact is the Greenwood-Williamson model (Greenwood & Williamson, 1966). Many researchers (including one of the authors) have used it for comparison to sliding experiments, but it should be remembered that it is a static contact model, and the comparisons have limitations.

A microscopic model of sliding contact was proposed by Rice and Moslehy (Rice & Moslehy, 1997), who considered geometric excitation of the slider by roughness and wear debris sweeping through the contact. Each asperity or debris particle generated a displacement impulse with triangular time-profile. Their model, however, does not include the possibility of adhesion between the asperities of the contacting surfaces.

According to the adhesion theory of friction, (Bowden & Tabor, 1954) the normal load, applied in static contact is supported by many microscopic contacts, the sum of which forms the real contact area (Fig. 1). These micro-contacts are most likely to occur at the asperities of the microscopically rough surfaces and can be considered as welded junctions. The friction force is the sum of a ploughing term (not considered in the present study) and a shearing term, which arises from the force needed to shear the junctions and is given by:

$$F_s = \sum_{i=1}^{N} A_i \psi_i \qquad (1),$$

where A_i is the area of junction and ψ_i is its shear strength.

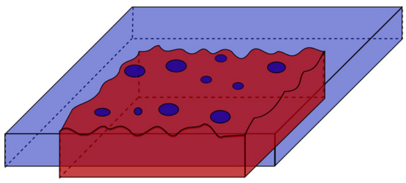

Figure 1 Real contact area formed by micro-contacts

This model of friction, proposed by Bowden and Tabor, is for quasi-static contact and does not consider the formation of junctions and their shearing during sliding.

Moreover, the same contact area can be achieved by a large number of small contacts or a small number of large contacts, but the adhesion theory makes no distinction, because it only considers the average friction force. The differences, however, become important, when friction fluctuations and friction-induced vibrations are considered.

In this paper we investigate the formation and shear of junctions as a possible mechanism for excitation of the tribo-system into vibration. Given the same average real contact area, the effect of the number of junctions and their size on the fluctuations of the real contact area during sliding is studied.

2. MODELLING

The model used in this paper is based on the adhesion theory of friction. It considers that junctions are formed and sheared during sliding and is an extension of a model first used (Rabinowicz, 1956) to extract information about the average size of junctions by autocorrelation analysis of friction fluctuations.

We assume that the junctions are formed when two truncated circular conical asperities slide over each other (Fig. 2). It is assumed that the asperities make contact over a flat region parallel to the plane of contact. The area of a junction at any given moment is given by the overlap between two circular areas and the "life history" of a junction as a function of relative displacement is shown in Fig.3 .

While the Rabinowicz model uses a triangular pulse to approximate the life of a junction, in the present paper we use the exact non-linear life history in which the engagement and disengagement of the asperities is gradual:

$$\begin{aligned} A_j(s) &= 2A_{seg}(s), & 0 \leqslant s \leqslant d \\ A_j(s) &= 2(A_{jmax} - A_{seg}), & d \leqslant s \leqslant 2\mathrm{d} \end{aligned} \qquad (2),$$

where

$$A_{seg}(s) = \frac{d^2}{4}\cos^{-1}\left(\frac{s}{d}\right) - \frac{1}{2}\sqrt{\frac{d}{2}(d-s) - \frac{s}{2}(d-s)^2} \qquad (3)$$

$$A_{jmax} = \frac{\pi d^2}{4} \qquad (4)$$

A_{seg} is the area of a segment of a circle, defined by height s and diameter d and A_{jmax} is the maximum area of a junction during its life.

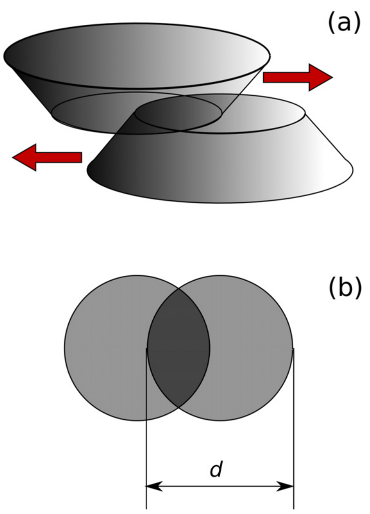

Figure 2 Truncated conical asperities in sliding contact (a) and their overlapping area (b).

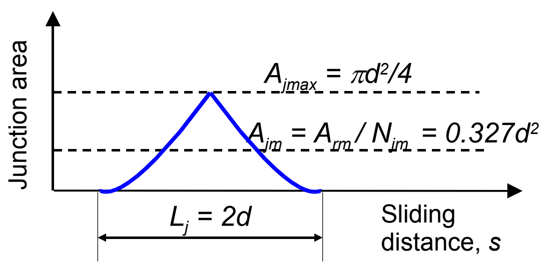

Figure 3 "Life history" of a junction

The life of a junction is given by:

$$L_j = 2d \quad (5).$$

From (2) and (3) the mean area of a junction during its life can be approximated by:

$$\bar{A}_j(s) = 0.327 d^2 \quad (6).$$

In the Rabinowicz model, the normal load is always supported by the the same number of junctions and as long as one junction diminishes it is replaced by another identical junction. In this way the junctions form at precisely regular moments in time and they all have exactly the same life. This model produces a constant contact area and and therefore constant friction force. In reality, however, the measurement of friction force always shows some fluctuation and Rabinowicz attributes them to the difference in junction strength. In this study we assume equal strength for all the junctions, but we extend the above model, by considering the formation of junctions as a random time event and by considering junctions with distributed size. As a result the instantaneous contact area is not constant, and this causes the friction force to fluctuate.

The mean real contact area, which supports the applied normal load, in sliding contact is given by:

$$\bar{A}_r(s) = \bar{N}_j(s)\bar{A}_j(a) \quad (7),$$

where $\bar{N}_j(s)$ is the mean number of junctions at any moment of sliding given by *s*, and $\bar{A}_j(a)$ is the is the mean area of a junction averaged over all junctions present in the contact area at any moment in sliding *s*. Here *(s)* is used to denote averaging along the direction of motion and *(a)*, averaging over the contact area.

If we assume that no significant changes occur on the surfaces during sliding, then spatial averaging can be replaced by temporal averaging and the mean area of a junction averaged over all junctions at any moment in sliding is equal to the mean area of a junction during its life:

$$\bar{A}_j(a) = \bar{A}_j(s) = A_{jm} \quad (8)$$

It is also true that the the mean number of junctions at any moment in sliding is equal to the number of junctions in static contact, i.e.

$$\bar{N}_j(s) = N_j(a) = N_{jm} \quad (9)$$

From (8) and (9) we can rewrite (7) as:

$$A_{rm} = N_{jm} A_{jm} \quad (10)$$

Thus, if we know the average value of the real contact area needed to support the applied load, and we suppose the average number of junctions, we can calculate the mean junction area A_{jm}, the mean asperity diameter d_m and the mean life of a junction L_{jm}:

$$A_{jm} = \frac{A_{rm}}{N_{jm}} \quad (11)$$

$$d_m = \sqrt{\left(\frac{A_{jm}}{0.327}\right)} \quad (12)$$

$$L_{jm} = 2 d_m \quad (13)$$

A more realistic model of the adhesive sliding would have to include junctions formed by asperities of different sizes. Asperity sizes are assumed to be normally distributed around the mean value, calculated in (12). In the current paper value of standard deviation of $\sigma = 0.2$ is used. In the simulation, in case of a negative value for *d* being drawn, the draw is repeated until a positive diameter is drawn.

We also consider that the junctions are formed randomly during sliding. Events, which occur randomly in a given amount of time, space, area, etc. are usually modelled by the Poisson distribution. Examples include the the number of Geiger counter clicks per second, the number of people walking into a store in an hour, and the number of flaws per metre of video tape. Thus, it is reasonable to assume that the number of junctions N_j formed during the period of average junction life L_{jm} is distributed by the Poisson distribution with mean N_{jm} . If the number of counts follows a Poisson distribution, then the interval between individual counts follows the exponential distribution. This fact is used to simulate a junction "birth" at random sliding distance interval s_j from the exponential distribution with mean value:

$$s_{jm} = \frac{L_{jm}}{N_{jm}} \quad (12)$$

The simulation takes the real contact area A_{rm} and the average number of junctions N_{jm} as inputs and the diameter of the asperities is obtained from the equation (12) above.

An example of a portion of the trace obtained for the instantaneous contact area as a sum of the area of all junctions is shown in Fig. 4 in the case of A_{rm} = $200\ \mu m^2$ and N_{jm} = 20. At any moment during sliding there are on average 20 junctions to support the applied load and the sum of their areas produces the instantaneous real contact area A_r:

$$A_r = \sum_{j=1}^{N_j(s)} A_j(s) \qquad (13).$$

The j-index is used here both to indicate 'junction' and as a summation index. Note that the number of junctions N_j is not constant, but depends on the relative sliding position s.

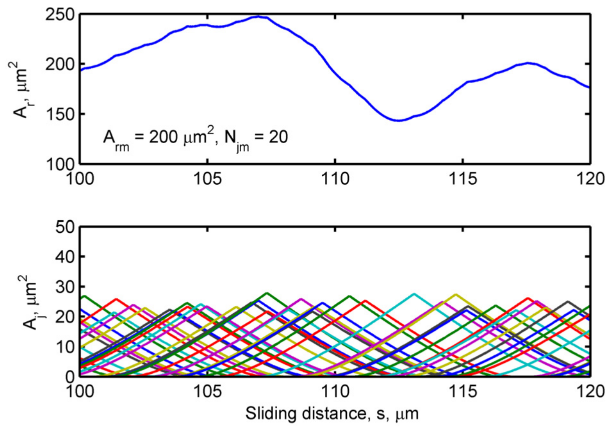

Figure 4 Instantaneous real contact area A_r as a sum of multiple junction areas A_j.

After the interactions at the contact interface leading to fluctuating contact area are considered, it is trivial to convert to force and to apply force excitation to the mechanical system holding the sliding pair together.

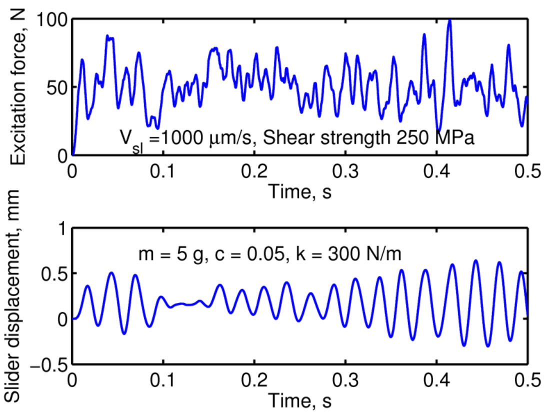

Figure 5 Excitation force and resultant vibration

Fig. 5 shows the result for a selected case when the interface is characterized by A_{jm} = $200\ \mu m_2$, N_{jm} = 20, sliding speed V_{sl} = $1\ mm/s$, shear strength of the junctions ψ = $250\ MPa$ and the dynamic system is characterized by mass m = $5\ g$, stiffness k = $300\ N/m$ and damping coefficient c = 0.05.

3. RESULTS

A representative simulation result for the fluctuation of the real contact area around a selected mean value for various number of supporting junctions is shown in Fig. 6. The mean real contact area of 200 μm^2 is formed by 10, 20, 30 40 and 50 junctions on average. The increase in the number of supporting junctions leads to a reduction of the contact area fluctuation, as can be seen from both area-distance plots and from the histograms.

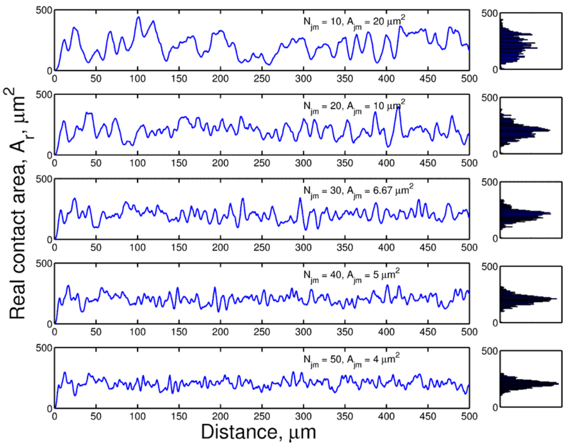

Figure 6 Real contact area traces for various average number of load-supporting junctions.

To quantize the fluctuation of the real contact area, we use the standard deviation (Fig. 7). The standard deviation of the instantaneous contact area σ_{Ar} decreases non-linearly with the increase of the number of supporting junctions. This trend is more obvious for large contact areas, and while the increase of the number of supporting junctions from 10 to 50 when contact area is 100 μm^2 results in a decrease of σ_{Ar} by 22 μm^2, the same increase of supporting junctions when contact area is 500 μm^2 results in a decrease of σ_{Ar} by 110 μm^2 .

The fluctuations of the simulated contact area were also analysed in the frequency domain by the power spectral density (PSD) function. Fig. 8 shows a representative example, when A_r = 200 μm^2 and the number of junctions is N_j = 20. The PSD resembles an exponential function – it shows a high relatively value at low spatial frequencies (long surface wavelength), which decays very quickly with the increase of frequency. The number of the supporting junctions and their relative size have a negligible effect on the PSD as seen from Fig. 9.

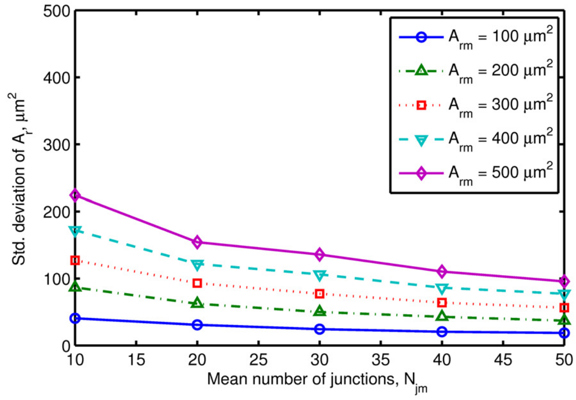

Figure 7 The standard deviation of the instantaneous contact area vs. the average number of supporting junctions.

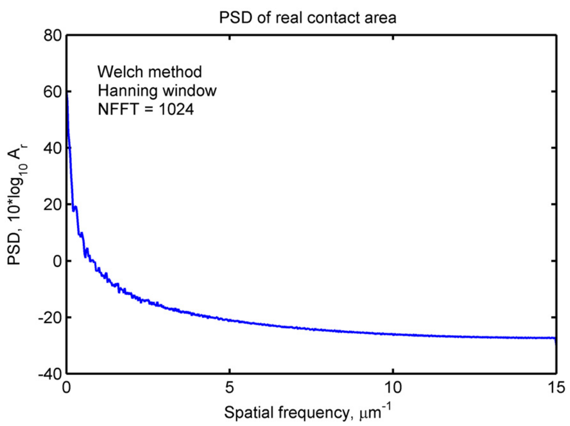

Figure 8 Power spectral density of the real contact area with mean 200 μm^2 and N_j = 20.

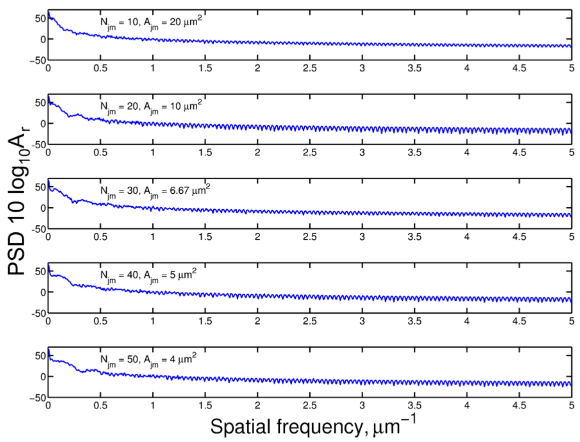

Figure 9 PSD of the real contact area with mean 200 μm^2.

4. DISCUSSION

The results of the simulation show that from the viewpoint of reducing the fluctuations of the real contact area and therefore the friction force induced vibrations, it is advantageous to have many small asperities on the surface, which can form a large number of small sized junctions. In practice this effect can be achieved by texturing the surfaces. Indeed, there are experimental results in the literature, which support this view. In studies on the hard disk head/disk interface (Xu at al., 2002; Zhou et al. 2005) a significant reduction of vibrations of a textured slider compared to a slider without texture was observed. This reduction of vibrations is usually attributed to the reduced friction of the textured surface, with the implicit assumption that low friction generates low vibrations and high friction – high vibrations. In fact, difference in the average value of friction is not enough to excite the holding system into vibration, but the fluctuating friction force can do it. And we believe that the presented model gives a better qualitative explanation of the observed reduction of vibrations for the textured sliders.

The simulation revealed that the spectrum of the fluctuating contact area has magnitude, which diminishes quickly with the increase of the spatial frequency and then stays constant. This observation can be readily compared to an observation by Kilburn (Kilburn, 1974) that the friction can be considered as a random process composed of a constant signal and superimposed white noise. The white noise has a constant magnitude at all frequencies. Our simulation results are qualitatively very similar to the experimental observation by Kilburn.

Although being able to provide some new qualitative understanding about friction induces vibrations, at this point the present model has a number of limitations. The individual junctions are assumed to always be parallel to the plane of sliding. This simplifies the model, but reduces its usefulness as vibrations in direction perpendicular to the direction of sliding are excluded. The proposed model may be thought as a model describing sliding under constant separation. In the future it would be useful to extend the model, so that asperities could make contact at an arbitrary angle to the plane of contact. This would add the possibility of a normal component of vibration, but would also add the need of loss of contact and impact considerations. At present the model is not ready for direct plug-in of values from measured surface profiles. Extension in that direction would enable direct quantitative comparison of model predictions and experimental results.

5. CONCLUSION

In order to gain insight into the vibration generation during steady sliding we used a microscopic contact model, which considers the life of junctions formed by adhesive micro-contacts. In comparison with previous models, in the present extended model the junctions are allowed to form at random in time and to have random size, which is closer to the reality of the interaction of rough surfaces.

The simulations suggests that adhesive sliding contact, in which the normal load is supported by a large number of small size micro-contacts would vibrate less that the same system in which the load is supported by the same real contact area distributed in few large micro-contacts.

This result agrees with experimental observations of reduced vibration of textured hard disk drive head sliders, compared to the non-textured ones.

6. REFERENCES

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