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Normal stiffness and damping at lightly loaded rough planar contacts

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ABSTRACT

Two features of contact between rough surfaces are localized interactions between asperities and the presence of air in the gap. This gives rise to a contact compliance, a static and local property and, when contacts vibrate, mechanical damping due to squeeze films of air. We examine the stiffness characteristics of clean surfaces in primarily elastic contact and both new worn surfaces in elasto-plastic contact. The squeeze-film damping can be related to the average gap between the surfaces. Experimental results are compared with analytical models. Qualitative and quantitative insight into both the stiffness and the gap can be obtained from the Greenwood–Williamson (1996) [1] paper that we are honoring at this symposium.

1. Introduction

The presence of surface roughness results in contact compliance at the interface between two surfaces. Also, the interface region consists mostly of air. Air can be drawn into and expelled out from the gap due to squeeze-film action, resulting in viscous damping. Other mechanisms of damping such as local slip, material hysteresis, plastic deformation and acoustic radiation can also be present but do not seem to be needed to explain most of the experimental results on which we will focus. We have found it important to include at least the normal stiffness for prediction of high frequency vibration, squeal, various aspects of thermo-mechanical contact as well as wear modeling.

Starting in the 1960s, considerable research was carried out on the stiffness and damping at joints between machine tool structural elements to better understand machine tool structural dynamics and stability as designs moved from cast iron to fabricated assemblies. A good summary of this work is provided by Back et al. [2]. Both exponential and power law load-deflection characteristics have been proposed and have been found to correlate with experimental data. The exponential form of the force-deflection relation follows directly from the pioneering work of Greenwood and Williamson (G–W) [1].

In this paper we explore, using our data and that of others, the contact stiffness and damping in relation to the Greenwood and Williamson paper and a second paper by Griffin, Richardson and Yamanami [3] written in 1966 on squeeze film damping. While squeeze film damping or "air pumping" has been noted as an important mechanism of joint damping since the 1960s, squeeze film damping models are usually applied to surfaces fully separated by air gaps. Somewhat surprisingly, we have not found any previous instances of a squeeze-film damping model being applied on rough surfaces in direct contact.

With this type of parameterization an interfacial layer of unspecified thickness is created to capture the damping and stiffness properties of the interface region. This approach allows the rough surface and gap region to be replaced by a smooth contact for modeling and analysis of fullscale machine elements in contact.

2. Elastic contacts

2.1. Contact stiffness

The stiffness characteristics of rough planar surfaces are highly nonlinear. However, in the case of an exponential distribution of asperity heights, it follows directly [4] from the G–W theory, that the linearized contact stiffness per unit area, k, at a nominal contact pressure, p, is $k=p/\sigma$ or $k\sigma=p$, where σ is the root mean square (rms) combined surface roughness. Persson and Fuller [5,6] have found the same result using a different approach based on multi-scale analysis. Polycarpou and Etsion [7] have shown that a Gaussian distribution of asperity heights can be approximated by

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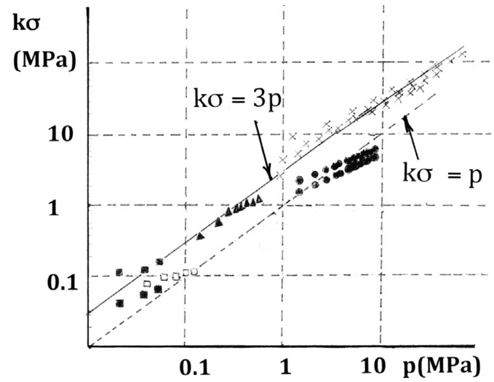

Fig. 1. Contact stiffness data (Hess and Wagh [13]; ▴ Polycarpou and Shi [10]; x Nuri [12]; ▫ Sherif and Kossa [11]; • Burdekin et al. [9]).

an exponential with an rms value of one third of the actual value. The contact stiffness can then be written as $k \approx 3p/\sigma$ or $k\sigma \approx 3p$. Other surface height distributions can lead to different stiffness-pressure relations [8].

In Fig. 1 we compare sets of measured stiffness values [9–13] as a function of pressure to the exponential, $k\sigma = p$, and the approximated Gaussian, $k\sigma = 3p$, G–W models. The data cover a range of more than three orders of magnitude in pressure and $k\sigma$. The rms roughness ranges from 0.15 to 2.0 μm and nominal contact pressure from 0.02 to 50 MPa. The two lines based on G–W capture nearly all of the data and provide a remarkably good estimate of the contact stiffness. While some of these pressures are rather low, we note that practical dry sliding contacts, e.g., most clutches and brakes, operate at nominal pressures ranging from a few tenths to two MPa, irrespective of the materials from which they are made. The mild steel electromagnetic clutches in our tests, described below, operate at a nominal pressure of 0.6 MPa. Stationary joints are more heavily loaded.

We have found, in applications to squeal and wear modeling [14–16], that the precise value of the contact stiffness, within a factor of two or three, is less important that the fact of its inclusion in the modeling.

In Fig. 1 there are departures from the linear increase of stiffness with pressure predicted by the G–W and the Persson theories. Sometimes this occurs within the same data set. Power law stiffness has been associated with both elasto-plastic deformations [17–20] and elastic contacts [21]. The paper by Paggi and Barber [21] provides an excellent review that is both historical and up to date. The power law stiffness suggested in [21] for an elastic contact has the kσ product proportional to p^n, where $0.8 < n < 0.9$. For elasto-plastic contacts, values of n in the range $0.5 < n < 0.7$ have been proposed[18]. The inverse proportionality of k to σ seems to be maintained in both types of contact.

2.2. Contact damping

We will focus on viscous squeeze film damping as the primary mechanism of damping. In this case, oscillatory motions, normal to the contact plane, cause air to be drawn into and expelled from the contact. In seeking data in the literature to test the squeeze film damping model, we found the experiments of Shi and Polycarpou [10] to be well-documented, providing both stiffness and damping data. They use an apparatus that employs impacts applied to the contact region, allowing stiffness and damping to be calculated from decaying vibrations. Their apparatus is based on a design by Serpe [14] and will be described more fully in Section 3 of this paper. Their stiffness measurements agree quite well with the G–W Gaussian approximation and were included in Fig. 1.

While the contacts we are considering are rough, we need to estimate the size of the gap. G–W [1] suggest an estimate of the gap, c, to be 2σ. However, the applied pressures in these tests between are quite low, between 0.15 and 0.51 MPa. This results in dimensionless pressures, p/E^*, between 0.14×10^{-5} and 0.48×10^{-5}, where E^* is the combined modulus of the steel surfaces of which the steel test samples are made. Using the dimensionless form of G–W (e.g., as given in [17]) we find the gap decreasing from 3.4σ to 3.1σ as the pressure increases over the range of the measurements.

The 1966 paper, by Griffin, Richardson and Yamanami (G–H–Y) [3], that is not as well known as G–W, deals with this problem. They analyze two geometries that we will use here. The first is a circular disc of radius, R separated by a uniform gap, c, above a planar surface. They show that the damping constant, b, can be written:

$$b = 48\mu R^3/\pi^3 c^3 \quad (1)$$

The viscosity of air is μ. The surfaces are parallel, with a gap, c.

The other geometry is a long rectangle of length, L, and width, w, where L»w. The damping constant is:

$$b = 96\mu L w^3/\pi^4 c^3 \quad (2)$$

The latter formulation will be used for the clutch samples in Section 3.

Although Shi and Polycarpou have not provided damping estimates based on theory we will compare their measurements with estimates based on G–H–Y. We will use a circular disc of the same area ($A = 1.82\ \text{mm}^2$) as their square plate. Their measurements and our calculations based on Eq. (1) for the critical damping ratio, ζ, where $\zeta = b/2(kAm)^{1/2}$, in per cent, are shown in Fig. 2. m ($=0.11$ kg) is the effective mass of one half of their test apparatus. The damping is generally low, only a few tenths of one per cent. The rms roughness, σ, is 0.19 μm. Based on the change in gap from 3.4σ to 3.1σ, the damping ratio is expected to increase by 32 per cent from the lowest to the highest pressure. The stiffness, k, being proportional to pressure, increases by a factor of 3.5 over the same range, resulting in a decrease of 87 per cent in the damping ratio due this factor. The combined effect of the decrease in gap and the increase in stiffness is a 30 per cent decrease in the theoretical damping ratio between the lowest and highest load.

Overall, the agreement with the G–H–Y model is very good. At the six higher pressures the squeeze film damping fully captures the damping, including the slight increase in damping ratio as the load decreases. At the two lower pressures, especially the lowest one, theory under-predicts the damping with the differences becoming significant.

Shi and Polycarpou [10] suggest that the much higher damping at the low loads is probably due to localized slip (they call it microslip) within the contact during impacts. They reach this

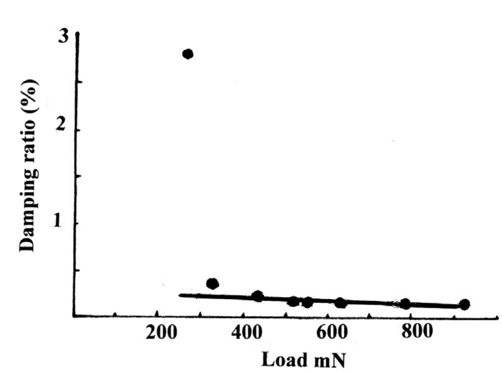

Fig. 2. Damping ratio versus pressure measurements (•from Shi and Polycarpou [10]). _____ Calculated damping ratio curve based on Griffin et al. [3].

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