

The Influence of Vibration on Friction

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ABSTRACT

This paper summarizes results from the author's work on friction in dry sliding contacts in the presence of vibration. A number of idealized models of smooth and rough contacts are examined. It is shown that vibration can cause up to a 10% reduction in average friction even with continuous contact. A larger reduction in friction occurs when there is intermittent contact loss. This is found to be true for both elastic and plastic contacts, and for adhesive and plowing mechanisms of friction. The results of this work are compared and validated with measurements from experiments. The results presented are fundamental, but applicable to machine components with contacts including brake systems.

INTRODUCTION

Surfaces in contact are often subjected to dynamic loads and associated contact vibrations. The dynamic loading may be generated either external to the contact region, as in the case of unbalanced or misaligned moving machinery components, or within the contact region, from sliding surface roughness-induced vibration. Vibrations may be undesirable from the point of view of the stresses that are induced or noise that is generated. Furthermore, vibrations can reduce friction and therefore the performance of systems such as brakes and clutches. Test results of brake materials obtained from different test machines may vary due to differences in vibratory environments inherent to the test machine. In addition, brake system test results may vary due to a changing vibration environment from, for example, a worn or wearing dynamometer bearing.

This paper explores the influence of vibration and dynamic loading on friction in dry sliding contacts. Elastic and plastic contact models are presented and both adhesion and plowing mechanisms of sliding friction are considered. Instantaneous and average normal and friction forces are computed and analyzed. Vibration is found to cause up to a 10% reduction in average friction with continuous contact and even larger reductions with intermittent contact loss. Remarkably, this is found for contact systems in which adhesion is the primary mechanism of friction as well as for systems in which plowing is the primary mechanism of friction. The reader is referred to other papers [1-5] for details on the various models presented. The significance of the effect of vibration on friction in brake systems is a potential reduction in braking performance, as well as a source for discrepancy between test results from dynamometers with different vibration environments.

ADHESIVE FRICTION AT HERTZIAN CONTACTS

For the first contact system model, the dynamic behavior of a circular elastic Hertzian contact under dynamic excitation is examined. The system model is shown in Fig. 1.

The rider has mass, m , and is in contact with a flat surface through a nonlinear stiffness and a viscous damper. The lower flat surface moves from left to right at a constant speed, V . The friction force, F , acts on the rider in the direction of sliding. The rider is constrained to motion normal to the direction of sliding. The model accommodates the primary normal contact resonance. The contact is loaded by its weight, mg , and by an external load, $P = P_0(1 + \alpha \cos \Omega t)$, which includes both a mean and a simple harmonic component. The normal

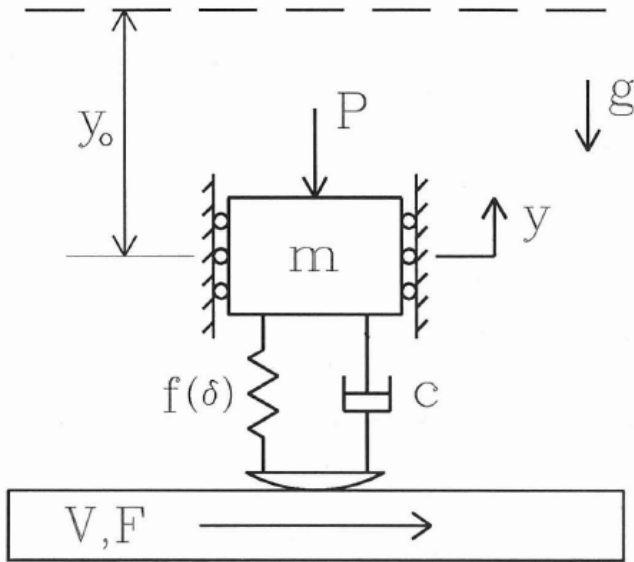


Figure 1 Dynamic model of Hertzian contact.

displacement, y , of the mass is measured upward from its static equilibrium position, y_0 . The equation of motion during contact, obtained from summing forces on the mass is

$$m\ddot{y} + c\dot{y} - f(\delta) = -P_0(1 + \alpha \cos \Omega t) - mg \quad (1)$$

where c is a damping constant, δ is the contact deflection and $f(\delta)$ is the restoring force from Hertzian contact mechanics defined as

$$f(\delta) = \frac{4}{3} E' R^{1/2} \delta^{3/2} = K_1 (y_0 - y)^{2/3}, \quad y_0 = \left(\frac{P_0 + mg}{K_1} \right)^{2/3} \quad (2)$$

Where E' is the effective modulus of elasticity, R is Hertzian contact radius, and K_1 is a spring constant dependent on E' and R . Equation (2) shows that the elastic contact stiffness is nonlinear. An approximate steady-state solution to the nonlinear system defined by equation (1) has been obtained [1] using the perturbation technique known as the method of multiple scales.

From Hertzian theory, the contact area, A , is proportional to the contact deflection, $(y_0 - y)$. Based on the simple adhesion theory of friction, the instantaneous friction force, F , is assumed to be proportional to the area of the contact, A . Therefore,

$$\frac{F}{F_0} = \frac{A}{A_0} = 1 - \frac{y}{y_0} \quad (3)$$

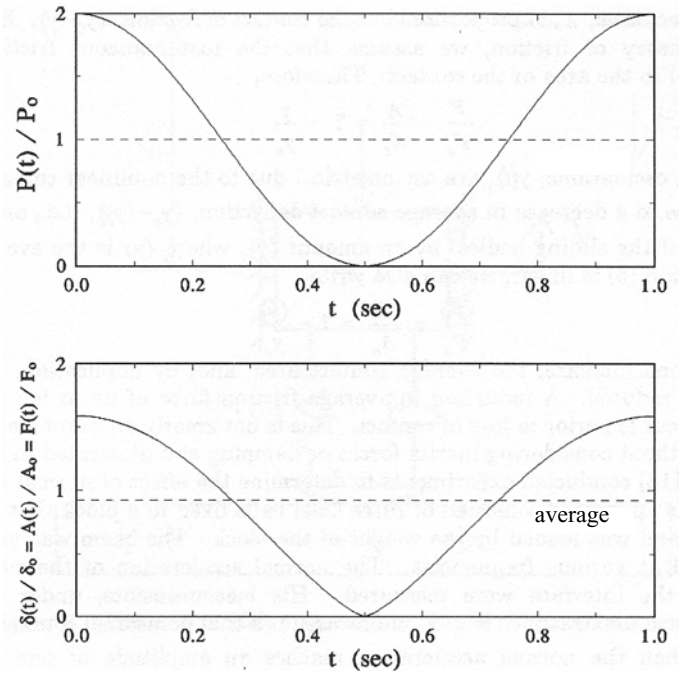


Figure 2 Instantaneous and average normal load, contact area and friction for Hertzian contact.

where F_0 and A_0 are the friction and area of contact at static equilibrium. The normal oscillations, $y(t)$, are asymmetrical due to the nonlinear contact stiffness, and give rise to a decrease in average contact deflection, $\langle \delta \rangle = (y_0 - \langle y \rangle)$, (i.e., an increase in separation of the sliding bodies) by an amount $\langle y \rangle$, where $\langle y \rangle$ is the average of $y(t)$. Figure 2 illustrates the asymmetrical fluctuation in contact deformation, contact area and friction due to harmonic normal load oscillations. Since equation (3) is linear, average values can also be written as

$$\frac{\langle F \rangle}{F_0} = \frac{\langle A \rangle}{A_0} = 1 - \frac{\langle y \rangle}{y_0} \quad (4)$$

As the magnitude of the normal vibration increases, the average contact area and the average friction are reduced. A reduction in average friction force of up to 10% was shown to occur [1] prior to loss of contact.

Godfrey [6] conducted experiments to determine the effect of normal vibration on friction. His apparatus consisted of three steel balls fixed to a block that slid along a steel beam and was loaded by the weight of the block. The beam was vibrated by a speaker coil at various frequencies. The normal acceleration of the rider and the friction at the interface were measured. His measurements, under dry contact conditions, are illustrated in Fig. 3. Assuming contact loss begins to occur when the normal acceleration reaches an amplitude of one g, one can superimpose the friction reduction predicted by our model as indicated by the heavy line. Reasonably

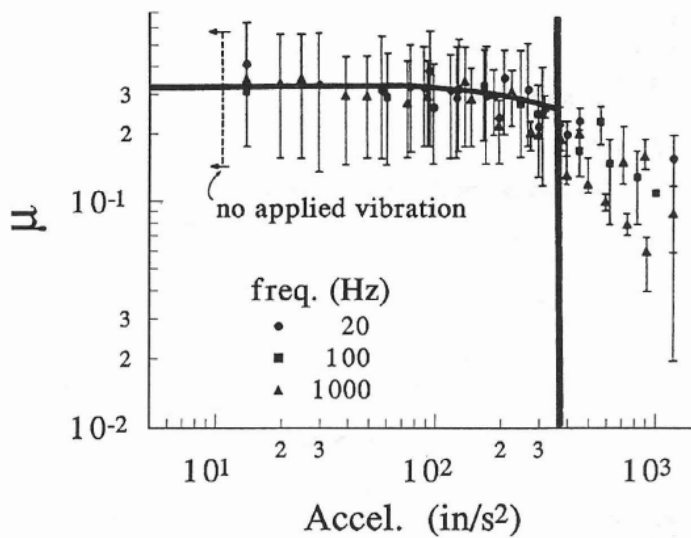


Figure 3 Measurements from Godfrey (1967) showing the effect of vibration on friction; —, model prediction.

good agreement is obtained. At higher normal accelerations, where there is progressively more intermittent contact loss, a larger reduction in friction occurs.

SURFACE ROUGHNESS-INDUCED VIBRATION

The dynamic behavior of continuously sliding Hertzian contacts under random roughness-induced base excitation has also been examined [2, 3]. The system model is shown in Fig. 4. At sufficiently high loads, such contacts can be represented by a smooth Hertzian contact [7]. The role of the surface roughness in this model is only to provide a base excitation as it is swept through the contact region.

By restricting the effective surface roughness input displacement to stationary random processes defined by the spectral density function $S_{y_i y_i}(k) = L\pi^{-1}k^{-4}$ (where L is a constant and k is the surface wavenumber), the Fokker-Planck equation can be used to obtain the exact stationary solution.

As in the case with applied harmonic normal loading, a decrease in the mean contact compression under dynamic loading is found which leads to a reduction in the mean contact area and the average friction force, under the assumption that the instantaneous friction force is proportional to the instantaneous area of contact. Based on the analysis in [2] the reduction in average friction force when vibration amplitudes approached the limit of contact loss was found to be about 9%.

A pin-on-disk system with a steel against steel Hertzian contact, excited by surface irregularities, was used to obtain measurements of average friction at various sliding speeds. The normal vibrations were found to increase

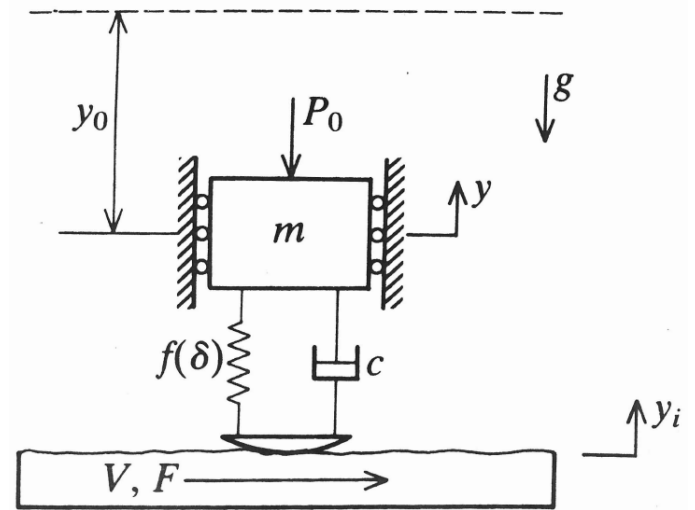


Figure 4 Hertzian contact model with roughness-induced vibration.

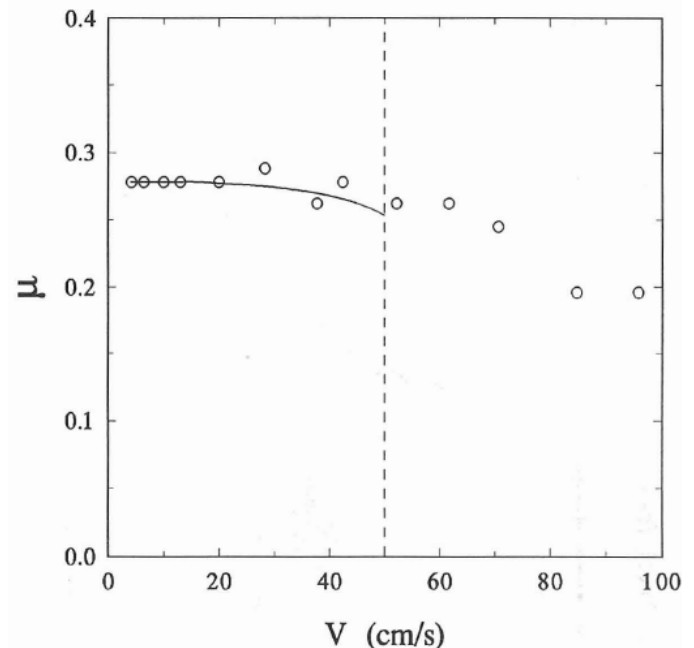


Figure 5 Average coefficient of friction at various sliding speeds: o, measurements; —, theoretical.

with sliding speed. The analysis was compared with the experiments by adjusting the parameter, L , so that the analytical model gave initial loss of contact at the same speed (i.e., at 50 cm/s) as observed during the tests. The computed results are shown together with the measurements in Fig. 5. The measurements show a decrease in friction with increasing sliding speed. The measurements illustrate that at speeds above those associated with initial loss of contact, even larger reductions in average friction can occur.