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Dynamic system model for ultrasonic lubrication in perpendicular configuration

Sheng Dong, Marcelo J. Dapino *

Smart Vehicle Concepts Center, Department of Mechanical and Aerospace Engineering, The Ohio State University, Columbus, OH 43210, USA

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ABSTRACT

Ultrasonic lubrication can be achieved by superimposing ultrasonic vibrations onto the relative sliding velocity between two surfaces. Ultrasonic vibrations are typically generated by a piezoelectric actuator. Relative to the macroscopic velocity, the vibrations can be longitudinal, transverse, or perpendicular. Often considered as a purely interfacial effect, ultrasonic lubrication is in fact a system phenomenon incorporating the dynamics of the actuator, sliding surfaces, and surrounding structure. This article presents a dynamic system model for ultrasonic lubrication configured in perpendicular mode, as experimentally measured with a modified pin-on-disc tribometer. The framework includes a lumped-parameter, dynamic model for the tribometer, an electromechanical model for the piezoelectric transducer used to generated the ultrasonic vibrations in the tribometer, and a "cube" model for the contact mechanics between asperities. Electrical impedance, system vibrations, and friction reduction are examined. Results show a strong match between experiments and simulations with errors lower than 10%. A parametric study is conducted to investigate the influence of driving voltage, macroscopic velocity, driving frequency, and signal waveform on ultrasonic friction reduction.

1. introduction

Friction is the resistance to the motion between two contacting surfaces when they slide or roll relative to each other [1]. Ultrasonic lubrication is achieved through superimposition of ultrasonic vibrations onto the macroscopic velocity. Ultrasonic vibrations can be applied in either of three directions: longitudinal, transverse, or perpendicular, as well as combinations thereof. Piezoelectric actuators are often employed to generate the vibrations.

For a given macroscopic velocity, higher vibration velocity results in greater friction reduction. In a harmonic sense, the magnitude of the vibration velocity is determined by the product of the amplitude and frequency of the vibration displacement. For this reason, compact ultrasonic transducers can be utilized to achieve high vibration velocity at small vibration amplitudes. Additionally, ultrasonic frequencies are significantly higher than the natural frequencies of the surrounding structures; this facilitates acoustic isolation of the transducer and prevents leakage of vibration energy.

Although several factors contribute to ultrasonic lubrication, a reduction in the effective contact pressure between asperities is the dominant factor for friction reduction when the vibrations are perpendicular to the macroscopic velocity. This study is focused on that mode of operation. The degree of friction reduction can be controlled by modulating vibration velocity, which is done by varying the actuator's drive voltage. Variable, solid state lubrication can be used in applications in which traditional lubrication is unfeasible (e.g., vehicle seats, space mechanisms) or where friction modulation is desirable (e.g., automotive steering or suspension components).

Both experimental and modeling studies have been conducted to understand and explain ultrasonic lubrication. For example, Littmann et al. [2,3] used a piezoelectric actuator generating vibrations at 60 kHz, making it slide longitudinally on a guide track. They also developed a mathematical relationship between velocity ratio and friction ratio, which indicates that a higher vibration velocity results in greater friction reduction. In their study, the velocity ratio was defined as the macroscopic velocity over the velocity of the ultrasonic vibrations. The friction ratio was defined as the friction force when ultrasonic vibrations are present over friction force without ultrasonic vibrations. It was proposed that a small velocity ratio leads to a low friction ratio, and hence effective friction reduction. As the velocity ratio increases, so does the friction ratio until a value of 1 is achieved and no further benefit from the ultrasonic vibrations is possible. Therefore, an increase in macroscopic sliding velocity moves the system towards

* Corresponding author.
E-mail address: dapino.1@osu.edu (M.J. Dapino).

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a friction ratio of 1 and reduces the effectiveness of the ultrasonic vibrations.

Kumar and Hutchings [4] studied the mechanisms of friction reduction with ultrasonic vibrations applied in directions longitudinal and transverse relative to the macroscopic velocity. Coulomb friction [5] was employed in their study, which assumes a constant coefficient of friction during sliding. The superposition of ultrasonic vibrations both in longitudinal and transverse directions changes the direction of instantaneous velocity so that the overall magnitude of the friction force is reduced. A similar explanation was proposed by Popov et al. [6] and Tsai and Tseng [7].

Bharadwaj and Dapino [8–10] and Gutowski and Leus [11] adopted Dahl's [12] friction model and built it into their dynamic models. Bharadwaj and Dapino [8–10] analyzed the influence of contact stiffness, global stiffness, mass, coefficient of friction, and signal waveform on friction reduction. Gutowski and Leus [11] simulated time-dependent friction as the output of the dynamic system, and obtained good agreement between the simulation and experimental data. However, in both studies, they treated contact stiffness as a constant value as opposed to a changing parameter when ultrasonic vibrations are present. They provided no physical explanation for the calculation of the contact stiffness, but presented it as a fitted variable for matching the experimental data. Nonetheless, these models were successful in explaining ultrasonic friction reduction with vibrations applied longitudinal to the sliding direction.

Teidelt et al. [13] extended Popov's modeling work from in-plane to out-of-plane. The pin-on-disc set-up was adopted, and ultrasonic vibrations were applied on the pin in a direction perpendicular to the disc surface. They employed the Coulomb friction model, and explained ultrasonic friction reduction as a result of reduced normal load. The change of the normal load was calculated as the product of the contact stiffness and the deformation in the vertical direction. Dong and Dapino [14–17] have reported experimental work on ultrasonic friction reduction in the perpendicular direction due to longitudinal vibrations, i.e., through the Poisson effect [14]. They also studied ultrasonic wear reduction at different speeds between stainless steel and aluminum using a modified pin-on-disc tribometer [15,16]. In addition, an elastic-plastic "cube" model was proposed to explain the mechanisms of ultrasonic friction and wear reduction when ultrasonic vibrations are employed in the direction vertical to the macroscopic velocity [17].

Prior work on ultrasonic lubrication has shown compelling evidence that the degree of friction reduction is intrinsically dependent on the ratio of the macroscopic sliding speed and the wave propagation speed. This dependency is at the core of why ultrasonic (as opposed to subsonic) vibrations are required for achieving ultrasonic lubrication. Much of the literature [2–4,6,14,17] on this topic considers pseudo-static (i.e., steady state) responses and ignores the dynamics of the system and how the system affects ultrasonic lubrication. Consideration of system dynamics is critical because the sliding and ultrasonic velocities are system-dependent. The focus of this paper is the development of a dynamic, system level model for a modified pin-on-disc tribometer. The approach presented here can be adapted to describe constructive details of other ultrasonic lubrication systems.

In order to explain ultrasonic lubrication at the system level, models must represent these components at three representative scales: a "cube" model for the asperities located at the interface incorporating contact and friction reduction mechanisms, an electromechanical model for the piezoelectric transducer, and a dynamic system model for the overall system. It is apparent that constructive details determine the dynamic response at each of the three scales. In lieu of a generic structure, the model presented here aims to represent the dynamics of a pin-on-disc tribometer system that is the subject of the experiments used to guide and validate the model. Generalization to other structures can follow from the relatively simple construct presented in this study. The model is described in Section 2, followed in Section 3 by a description of experiments conducted on a pin-on-disc tribometer. Model results and comparisons with experimental measurements are provided in Section 4. In Section 5, the model is used to investigate the dependence of ultrasonic lubrication on key system parameters: driving voltage, velocity, driving frequency, and signal waveform.

2. Model development

2.1. System dynamics model

As shown schematically in Fig. 1(a), a stationary pin is in contact with a rotating disc sample. The pin consists of a piezoelectric actuator and an acorn nut tip. The top end of the pin is fixed to a gymbal arm which is connected to a weight through a pulley system, which is employed to apply normal load to the contact interface.

A lumped-parameter, three-degree-of-freedom system model of the tribometer is shown in Fig. 1(b). The model consists of a piezoelectric transducer, three masses, springs, and dampers. Mass m_1 represents the acorn nut connected to the rotating sample (ground) via an axial stiffness (k_1), an axial damping coefficient (c_1), and a tangential stiffness (k_t). The relative velocity between the two surfaces is denoted v_r and the normal force at the interface is F_N. The effective dynamic mass, stiffness, and damping of the weight and pulley system are m_3, k_2 and c_2, respectively. The equivalent parameters for the piezo-actuator are m_0, k_p, and c_p. The actuation force generated by the actuator is F_P, acting on m_1 and m_2. The effective dynamic mass of the gymbal arm is represented by m_2. The displacement of the acorn nut, the gymbal arm, and the weight are x_1, x_2, and x_3, respectively.

The system equations can be written as

$$\begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2+m_0 & 0 \\ 0 & 0 & m_3 \end{bmatrix}\begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \\ \ddot{x}_3 \end{bmatrix} + \begin{bmatrix} c_1+c_p & -c_p & 0 \\ -c_p & c_p+c_2 & -c_2 \\ 0 & -c_2 & c_2 \end{bmatrix}\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \end{bmatrix} + \begin{bmatrix} k_1+k_p & -k_p & 0 \\ -k_p & k_p+k_2 & -k_2 \\ 0 & -k_2 & k_2 \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} = \begin{bmatrix} -F_P \\ F_P \\ -F_N \end{bmatrix}, \quad (1)$$

2.2. Piezoelectric transducer

Piezoelectric materials transform energy between electrical and mechanical domains. Although piezoelectric materials are intrinsically nonlinear, transducers based on these materials are often driven at low signal regimes under mechanical bias conditions, leading to approximately linear responses. This situation has been modeled extensively [18], as follows:

$$\mathbf{D} = \epsilon^{\mathbf{T}}\mathbf{E} + \mathbf{dT}, \quad (2)$$

and

$$\mathbf{S} = \mathbf{dE} + \mathbf{s}^{\mathbf{E}}\mathbf{T}, \quad (3)$$

where $\mathbf{D}$ is electric displacement, $\mathbf{T}$ is stress, $\mathbf{E}$ is electric field, $\mathbf{S}$ is strain, $\epsilon^{\mathbf{T}}$ is permittivity at constant stress, $\mathbf{s}^{\mathbf{E}}$ is mechanical compliance at constant electric field, and $\mathbf{d}$ is a piezoelectric coupling constant. No tensorial indices are used for simplicity since only the quantities in the poling direction (33) are taken into consideration.

A linear representation of the electromechanical coupling is shown in Fig. 2, where F is mechanical load, v is velocity, Z_m is blocked mechanical impedance, V is driving voltage, I is current,