Mechanical Systems and Signal Processing 58-59 (2015) 101–114

Contents lists available at ScienceDirect

Mechanical Systems and Signal Processing

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

Estimation of coefficient of friction for a mechanical system with combined rolling–sliding contact using vibration measurements

Sriram Sundar, Jason T. Dreyer, Rajendra Singh *

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

ARTICLE INFO

Article history:
Received 10 June 2013
Received in revised form
14 November 2014
Accepted 27 November 2014
Available online 13 January 2015

Keywords:
Combined rolling–sliding contact system
Cam–follower
Friction experiment
Mixed lubrication regime
Nonlinear dynamics

ABSTRACT

A new dynamic experiment is proposed to estimate the coefficient of friction for a mechanical system with a combined rolling–sliding contact under a mixed lubrication regime. The experiment is designed and instrumented based on an analogous contact mechanics model, taking into consideration the constraints to ensure no impact and no sliding velocity reversal. The system consists of a cam (rotating with a constant speed) having a point contact with a follower that oscillates about a frictionless pivot, while maintaining contact with the cam with the help of a well-designed translational spring. The viscous damping elements for contact are identified for two different lubricants from an impulse test using the half-power bandwidth method. Dynamic responses (with the cam providing an input to the system) are measured in terms of the follower acceleration and the reaction forces at the follower pivot. A frequency domain based signal processing technique is proposed to estimate the coefficient of friction using the complex-valued Fourier amplitudes of the measured forces and acceleration. The coefficient of friction is estimated for the mechanical system with different surface roughnesses using two lubricants; these are also compared with similar values for both dry and lubricated cases as reported in the literature. An empirical relationship for the coefficient of friction is suggested based on a prior model under a mixed lubrication regime. Possible sources of errors in the estimation procedure are identified and quantified.

1. Introduction

Friction plays a significant role in the dynamics of mechanical systems under sliding contacts [1–7]. The friction force is often modeled using the Coulomb formulation, though the analyst must judiciously select the value of the coefficient of friction (μ). In many prior experimental studies as summarized by Persson [8], μ is found from a simple and pure translational sliding contact (without rolling) system. For instance, Espinosa et al. [9] used a modified Kolsky bar apparatus, while Hoskins et al. [10] used a sliding block of rocks to estimate the normal and friction forces. Furthermore, translational sliding experiments were employed by Worden et al. [11] to estimate the dependence of friction forces on displacement and velocity, and then by Schwingshackl et al. [12] to model the non-linear friction interface. Several investigators have also

* Corresponding author. Tel.: +1 614 292 9044.
E-mail address: singh.3@osu.edu (R. Singh).

http://dx.doi.org/10.1016/j.ymssp.2014.11.015

conducted friction experiments on rotating systems such as a pin-disk apparatus [13,14], two rotating circular plates [15], and a radially loaded disk-roller system [16,17]. Also, Kang and Kim [18] determined the Coulomb friction insight stabilization equipment using torque and angular displacement characteristics, while Povey and Paniagua [19] estimated the bearing friction for a turbo machinery application. Such pure sliding contact experiments cannot be employed to estimate μ for a system with combined rolling–sliding contact since the kinematics is different. Radzimovsky et al. [20] conducted experiments on gears to determine the instantaneous μ over a mesh cycle. However, none of the prior combined rolling–sliding contact experiments rely on vibration measurements. Furthermore, the conventional or direct approaches given in the literature [9–19] focus on estimating the μ for only a pure sliding contact systems (without rolling). Since the kinematics of such systems is generally complicated, new effort is needed to estimate the μ from vibration measurements (measured forces and acceleration) under certain conditions.

Some researchers have experimentally studied cam–follower mechanisms [21,22] from the stability and bifurcation perspective under impacting conditions. In contrast, a cam–follower mechanism with rotational sliding contact (with no impacts) is used to experimentally determine μ in this study. Since μ cannot be directly measured from vibration experiments, an analogous contact mechanics model [23] is developed to aid the process. The goal is to vary the surface roughness, lubrication film thickness, contact pressure and velocities at contact (sliding and entrainment). The proposed system could then be utilized to simulate the contact conditions seen in drum brakes and geared systems.

2. Problem formulation

Fig. 1 shows the mechanical system with an elliptic cam (with semi-major and minor axes as a and b, respectively). Though the kinematics of combined rolling–sliding contact systems are complex compared to systems with pure sliding contact, this is one of the simplest systems which one can devise to measure the coefficient of friction in such systems which would allow controlled measurements of the reaction forces and system acceleration. The cam is pivoted at E along its major axis with a radial run out, e, from its centroid (G_c, with subscript c denoting cam). The angle made by the end point of the major axis (A) with the horizontal axis ($\hat{e}_x$) is $\Theta\,(t)$, which is an excitation to the system (where t represents the time). The equation of the elliptic cam is given by the following, where r is the radial distance from G_c to any point on the circumference of the cam, and Δ is the polar angle of that point,

$$r(\Delta)=\frac{ab}{\sqrt{[a\ \sin\,(\Delta)]^2+[b\ \cos\,(\Delta)]^2}}. \tag{1}$$

The cam is in a point contact (at O_c) with the follower (at O_b, with subscript b denoting follower), which consists of a thin cylindrical dowel pin (of radius r_d) attached to a bar (of length l_b) of square cross-section (of width w_b). The center of gravity of the follower lies at G_b at a distance of l_g from the pivot point P (using roller bearings) which is at d_y distance above the ground. The follower is supported by a linear spring (k_s) along the vertical direction ($\hat{e}_y$), which is at a distance of d_x from P as shown in Fig. 1. The angular motion of the follower is given by $\alpha(t)$ in the clockwise direction from the $\hat{e}_x$ axis; it is also the only dynamic degree-of-freedom of the system. The contact mechanics at O between the cam and the follower is represented by non-linear contact stiffness (k_λ) and viscous damping (c_λ) elements. Viscous damping is valid in this study since the system is designed to not lose the contact at any point of time, and the indentation velocity is low (no impacts), and hence the contact damping force is insignificant (compared to contact stiffness force) regardless the damping

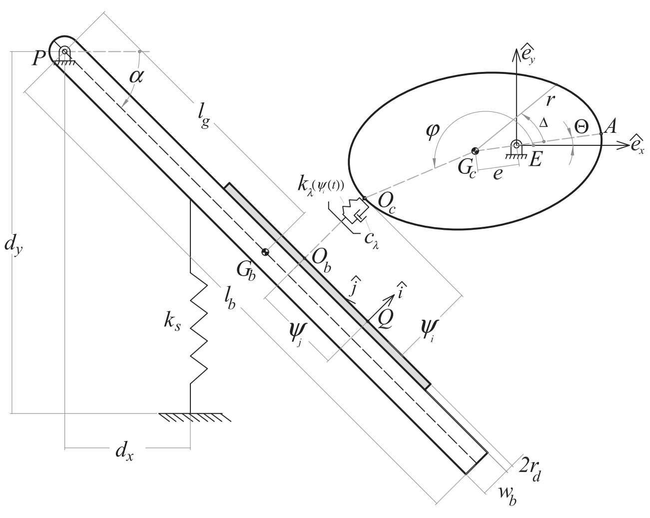

Fig. 1. Example case: a mechanical system with an elliptic cam and follower supported by a lumped spring (k_s).