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Estimation of friction distribution in partial-slip contacts from reciprocating full-sliding tests

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ABSTRACT

Methodology to estimate temporal and spatial variation of friction coefficient in partial slip contacts using reciprocating full-sliding tests is presented. Basic assumption is that surface modification and consequently the friction variation is a function of sliding distance. Friction coefficient is obtained as a function of sliding distance from the reciprocating full-sliding tests with low amplitude sliding at the contact interface. An analysis tool is used to estimate the slip in the interface for a given cycle. Cumulative slip at each point in the interface is used to predict the friction coefficient using friction curves generated from the full-sliding tests. Average value of the estimated friction coefficient variation was found to be in reasonable agreement with the values measured experimentally.

1. Introduction

Tightly clamped components that are subjected to cyclic loads typically have small amplitude sliding motion between two surfaces in contact leading to *fretting* [1,2]. Fretting effect initiates cracks at the interface of the sliding bodies due to very high stresses near the contact region. These cracks propagate due to bulk stresses acting in the components leading to their eventual failure at stress levels well below the fatigue (endurance) limit. This phenomenon is known as *fretting fatigue*. Practical situations wherein fretting influences the life of components include bolted and riveted joints; key-way shaft couplings and dovetail joints in turbines. Following factors influence contact tractions and hence contribute to the failure of the components: clamping (normal) loads, tangential loads, shape of the two contacting surfaces, material properties, friction coefficient and environmental conditions. While other factors have been studied in detail for partial slip contacts [3–7], more detailed studies are required for the effect of friction coefficient. In general, friction is due to combined effect of adhesion between the two surfaces, interlocking of asperities, asperity deformation, ploughing from wear particles and other factors [8–10]. These phenomena lead to surface modification with further sliding and a consequent change in friction coefficient. While the above explanations are qualitative, an estimate of the friction coefficient and its evolution at various points in the contact interface is needed for design and analysis.

Fig. 1 shows the schematic of an idealized contact between a punch and elastic half space. During fretting, if applied shear load (Q) is less than the average friction coefficient times the normal load (P), there is no gross sliding between the two bodies, leading to *partial slip* condition. Under these conditions, it can be shown that the contact zone will consist of a central *stick* region, in which there is no relative motion, surrounded by *slip* zones having sliding motion [4]. In the stick region, surface modification is negligible and the friction coefficient (μ) can be assumed not to vary significantly. Surface modification leads to a change in the friction coefficient in the slip region. Friction coefficient is different at different points in the interface because each point in slip region will slide with different amplitudes and further, these values vary with the number of sliding cycles. Therefore, μ varies with time as well as location (space). It is important to estimate this variation since the friction coefficient has a significant influence on the contact tractions.

In an experiment, we can only determine the mean value of the friction coefficient (μ) over the entire contact interface. Distribution of μ in the contact interface has to be estimated from this average value. Some of the researchers have developed models to predict this variation by assuming friction evolution to have linear [11] or exponential form [12]. However, actual form may be quite different and needs to be obtained from a sliding friction test or analysis. Hills & Nowell [4] proposed a bucket shaped curve representing the variation of friction coefficient in the contact region for a cylindrical contact. Dini & Nowell [13] established a numerical technique to predict the slip zone friction coefficient from the measured average. Their method is general and can be applied to any geometry provided that pressure distribution and stick/slip regime are known. However,

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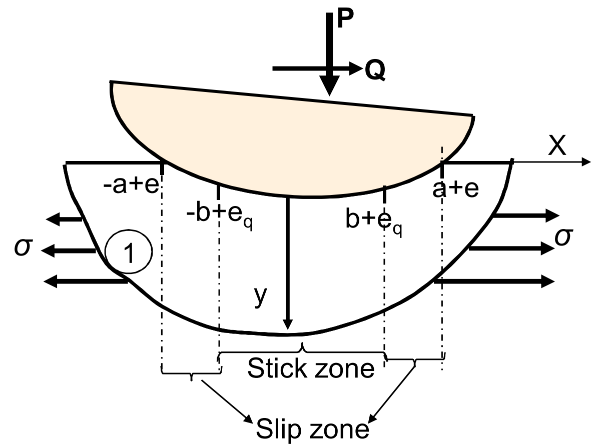

Fig. 1. Schematic representation of contact between a punch and elastic half space with the contact zone divided into central stick zone and the peripheral slip zones. Note that, for such two-dimensional contacts, P and Q are line loads (loads per unit depth).

they assume that μ is constant in the slip zone. This assumption is valid after a large number of cycles when friction coefficient has reached steady state value at every point in the slip region. Estimation of friction variation in the intermediate condition when it has not reached the asymptotic value in the slip region might also be necessary. Another way of estimating friction coefficient is by coupling Mindlin formalism with the measured friction dissipated energy and the measured tangential force [14,15]. The slip zone fiction coefficient can be estimated from the sliding transition from partial to gross slip condition using variable displacement method [16]. While they account for variation of friction coefficient with slip distance as well as normal force, their methodology is limited to Hertzian contact geometry. However, method based on sliding distance [4,13] is easily adaptable to random geometries.

This paper discusses a methodology to estimate temporal as well as spatial variation of friction coefficient (μ) using a combination of reciprocating full sliding tests and an analysis tool which can account for its spatial variation. It is essentially an attempt to predict the variation before the steady state distribution as predicted by Hills & Nowell [4] and Dini & Nowell [13]. Similar to their work, the underlying assumption here is that μ is a function of slip distance and the associated surface modification, for a given normal load. Friction coefficient (μ) as a function of sliding distance could be obtained from simple reciprocating full sliding tests. One drawback of the proposed method is that full sliding tests have to be conducted at the corresponding normal forces unlike energy methods [17] since the evolution depends upon the normal force. Using a previously developed analysis tool [18], contact tractions and slip are estimated at the interface. The friction coefficient, μ, at each point in the interface can then be modified according to the cumulative slip that it has experienced. Therefore, there are four aspects to this study: (1) obtaining μ as a function of sliding distance from reciprocating full sliding tests; (2) developing an analysis tool to study the effect of spatial variation of μ on contact tractions; (3) determining the variation of the μ along the contact interface with respect to time as well as space (location) using the results of the first two studies; (4) verification of average μ estimated from this variation with values measured from fretting tests. The analysis tool has been presented in detail elsewhere [18] and will only be briefly discussed here. This paper mainly focuses on the remaining three aspects. All the tests were conducted with flat specimens in contact with cylindrical pads (radius=250 mm) made from Al6061 T6 alloy. Pads and specimens were chosen to have same width to ensure plane strain conditions.

2. Experimental setup

Various test setups have been used for fretting studies and friction studies. Fretting experiments have been carried out earlier using proving ring [19–24] wherein tangential forces are measured indirectly using strain gages bonded to the pads. Due to strain gage bonding and the process of calibration of proving ring for each experiment, these setups are time consuming. Further, range of tangential loads that can be applied is limited. An external actuator could be used, but it is difficult to achieve a well controlled experiment due to the low displacement amplitudes involved [4]. Therefore, it is beneficial to utilize the compliance of the bridge or specimen or both in a bridge-type setup to generate necessary tangential load, as a reaction to the bulk load applied to the specimen. One such fixture used by Murthy et. al. [25] provides direct measurement of tangential load while minimizing the setup time and was taken as the basis for the fixture design discussed here. This fixture eliminates the need to recalibrate during every experiment. Friction evolution has also been studied using different setups [9] like pin-on-disk, ball-on-disk, block-on-ring etc. These setups might not exactly replicate friction evolution in fretting experiments because of oxide and nitride layer formation on the disk/ring during every rotation, when the pin is not in contact with a given location. Reciprocating test setup with small amplitude of oscillation avoids the oxide layer formation at the contact interface, which is closer to the condition during fretting. Further, it is convenient to use the same fixture for friction as well as fretting tests. Since many researchers have used plain fretting (reciprocating) experiments to evaluate friction evolution from dissipated energy [14,15], a similar adaptation was carried out for full sliding tests.

The rig (Fig. 2(a)) consists of a horizontal chassis resting on two reinforced vertical beams mounted on an *INSTRON 8801* single actuator fatigue testing machine. To simulate the hardware applications, it has been designed such that load transfer ratio (ratio of load transferred to the top support to the load applied at the bottom, F/F_o), is ≈50%. Specimen is clamped between the upper and lower grip and passes through a slot in the center of the chassis. Pads, one on either side of the specimen, are clamped on to the specimen through pad holder blocks. The pad is placed in the grooves in the center of the top and bottom portions of the pad holder block. The two portions are then bolted together. Each pad holder block is connected to the chassis by a set of thin plates (diaphragms). Normal load (P) is applied to the contact interface with the help of two threaded rods passing through free holes in the pad holder by tightening nuts on the rods against the block. It was measured by two load washers embedded between the pad holder and the nuts on each rod. The rods are located symmetrically on either side of the pad. To avoid out-of-plane moment, the loads on both the rods must be identical. Sum of the loads applied from the two rods gives the overall normal load (P). Since the pad holders are connected to the chassis with thin plates which act as membranes (diaphragms), most of the load applied to them in the horizontal direction (i.e., normal/transverse load) is transferred to the interface. Tangential or frictional load (Q) is generated at the specimen-pad interface since the diaphragms provide very good stiffness in the vertical direction (i.e., to in-plane loading).

This fixture can be used for fretting as well as the reciprocating full sliding friction tests (Fig. 2(b)). For fretting or fretting fatigue tests, both the ends of the specimen are clamped. When a load (F_o) is applied to the bottom of the specimen, part of it is transferred to the top support (F) and the remaining is transferred to the chassis through the specimen-pad contact. Since there are two pads, one on either side of the specimen, each will apply a tangential load of Q on the specimen. Consequently, $2Q$ is obtained directly as the difference between the loads measured by the bottom (F_o) and the top (F) load cells (Fig. 2(b)(i)). During the full sliding tests, the upper cross-head is unclamped and the upper end of the specimen is free. In this case, the load (F_o) applied at the bottom is entirely resisted by the friction between the